US 20160123637A1

(19) United States
(12) Patent Application Publication
MORENO et al.

(10) Pub. No.: US 2016/0123637 A1
(43) Pub. Date: May 5, 2016

(54) **TWO-PHASE HEAT EXCHANGER FOR COOLING ELECTRICAL COMPONENTS**

(71) Applicant: **Alliance for Sustainable Energy, LLC**, Golden, CO (US)

(72) Inventors: **Gilbert MORENO**, Lakewood, CO (US); **Jana R. JEFFERS**, Arvada, CO (US); **Kevin BENNION**, Littleton, CO (US); **Charles KING**, Golden, CO (US); **Sreekant NARUMANCHI**, Littleton, CO (US)

(21) Appl. No.: **14/926,594**

(22) Filed: **Oct. 29, 2015**

Related U.S. Application Data

(60) Provisional application No. 62/069,983, filed on Oct. 29, 2014.

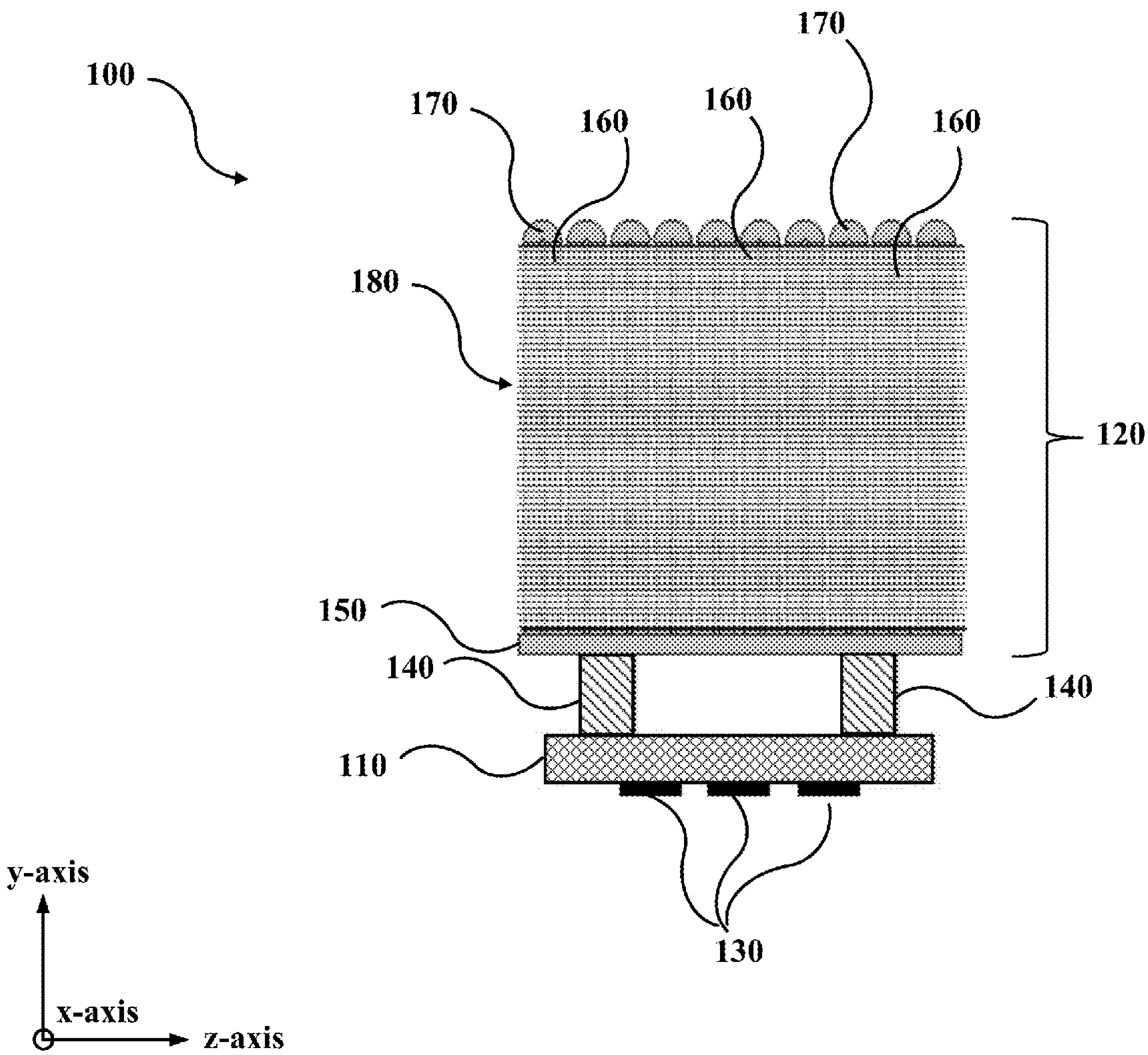
Publication Classification

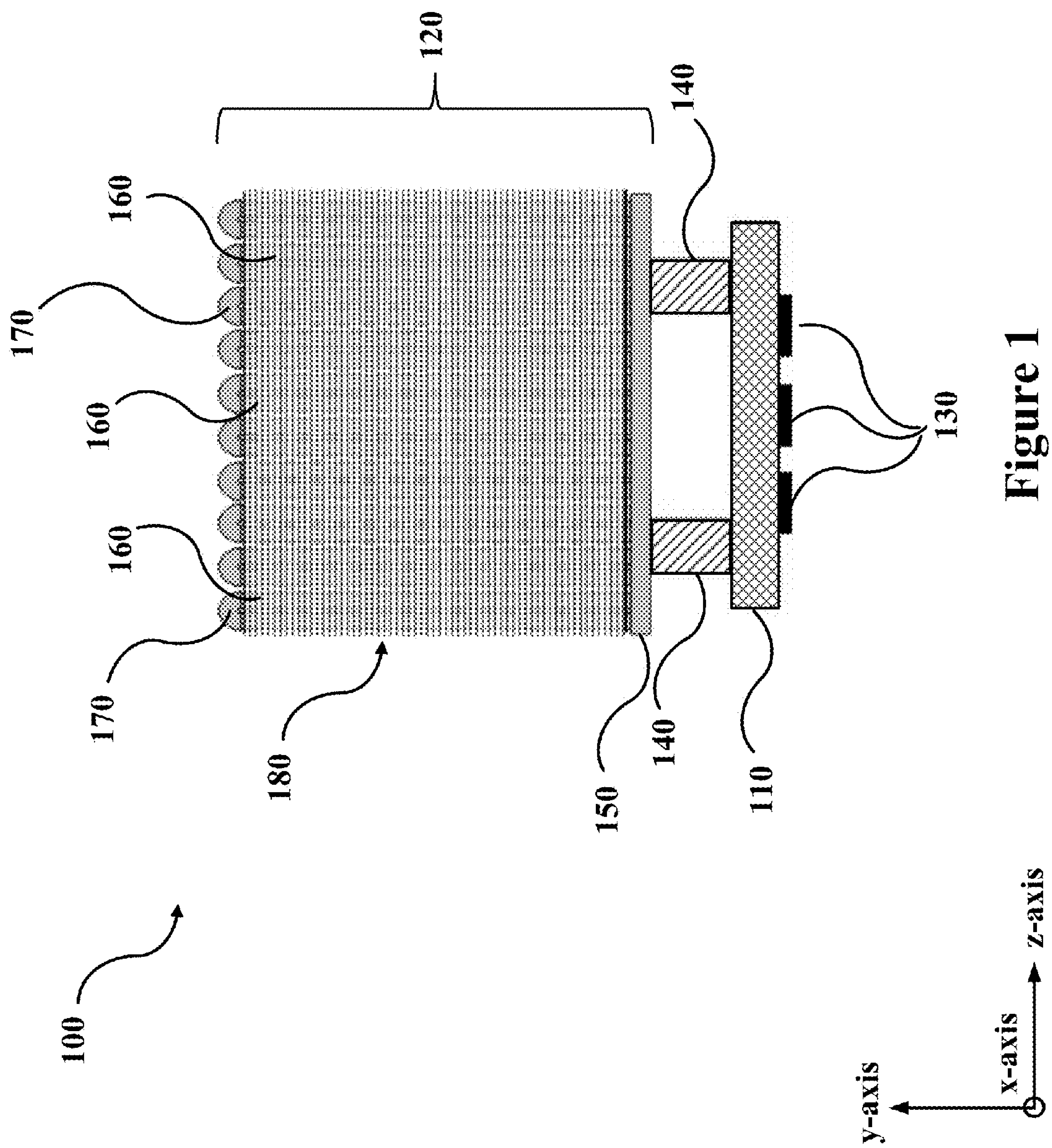
(51) **Int. Cl.**
F25B 39/00 (2006.01)

(52) **U.S. Cl.**
CPC ***F25B 39/00*** (2013.01); *F25B 39/02* (2013.01)

(57) **ABSTRACT**

Described herein are passive cooling systems utilizing two-phase heat-transfer fluids for transferring heat from one or more heat sources.





120
140
160
170
160
160
130
170
180
150
140
110
100
z-axis
x-axis
y-axis

Figure 1

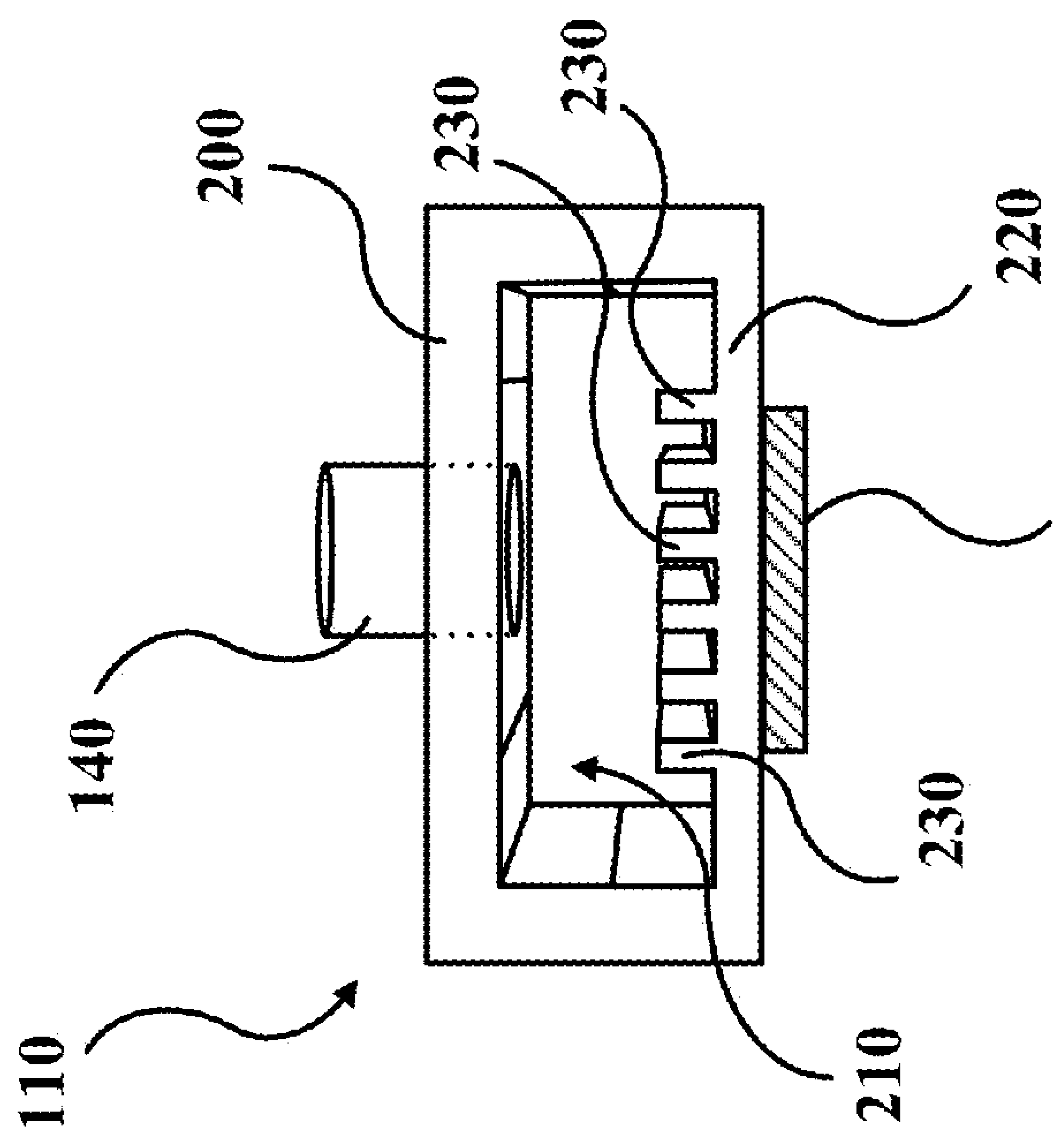


Figure 2b

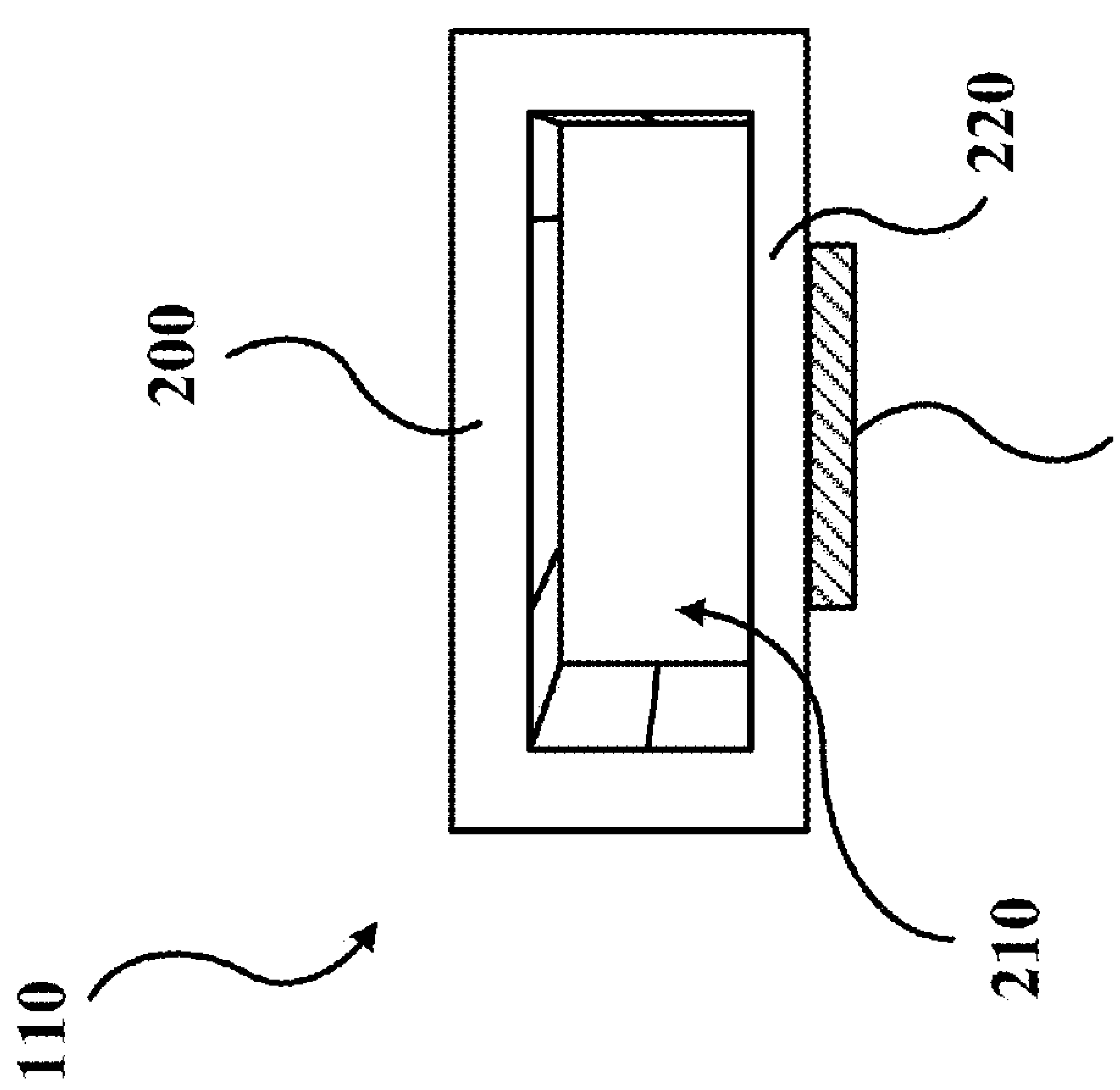
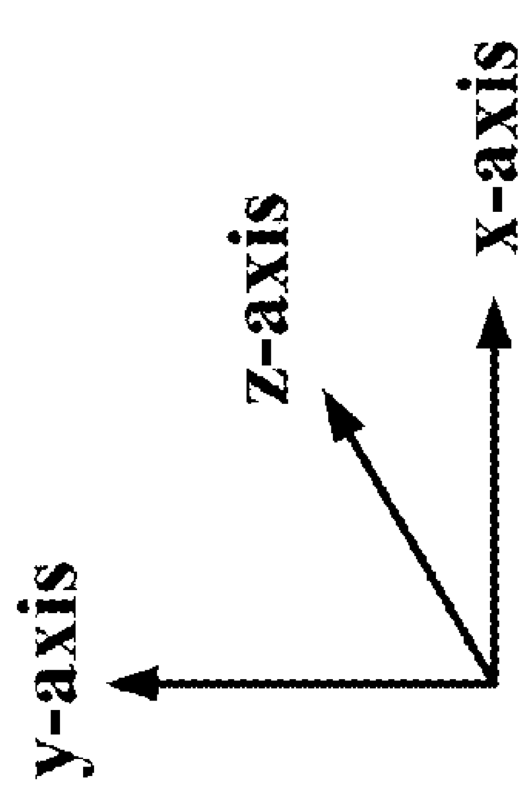


Figure 2a



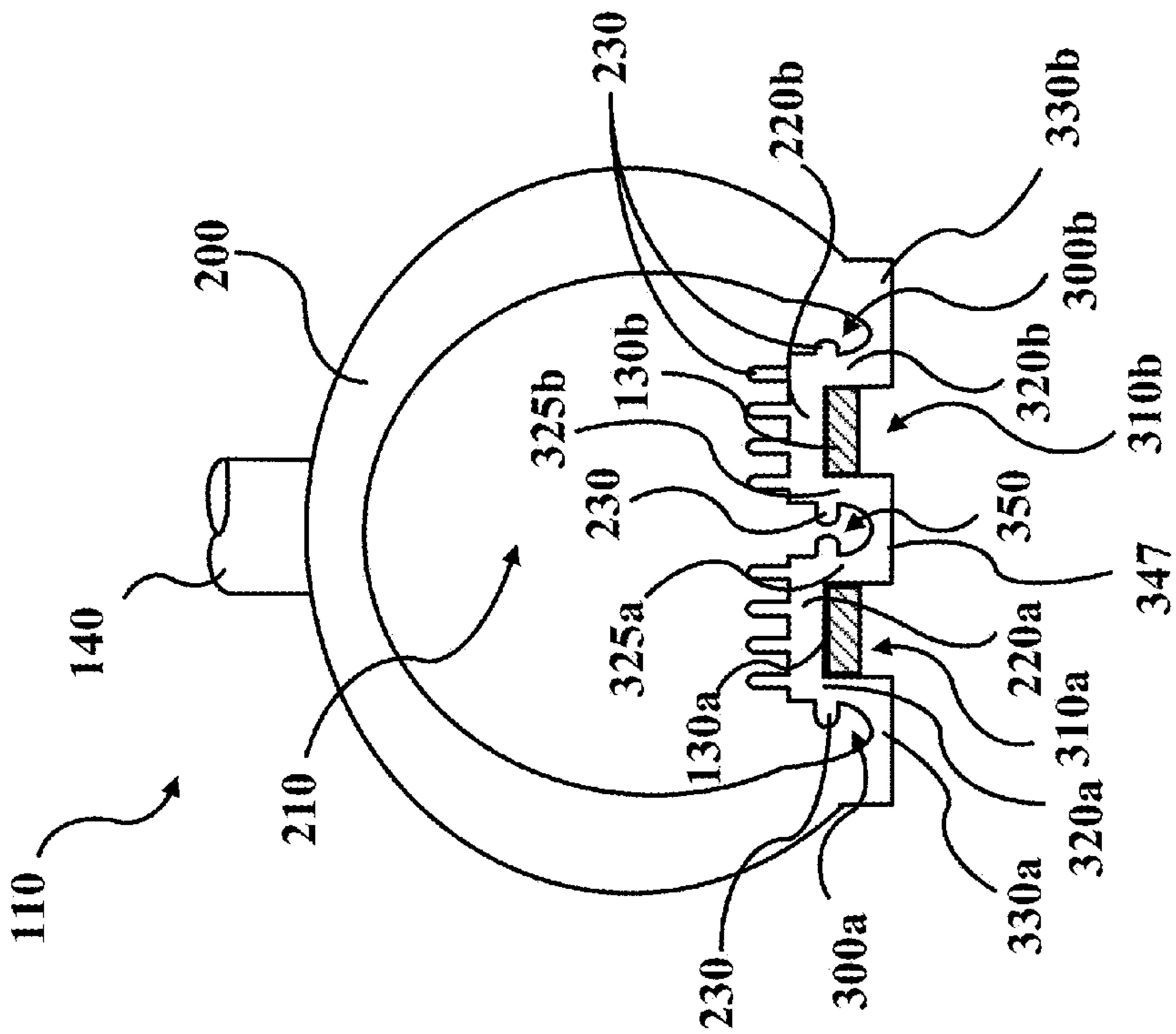
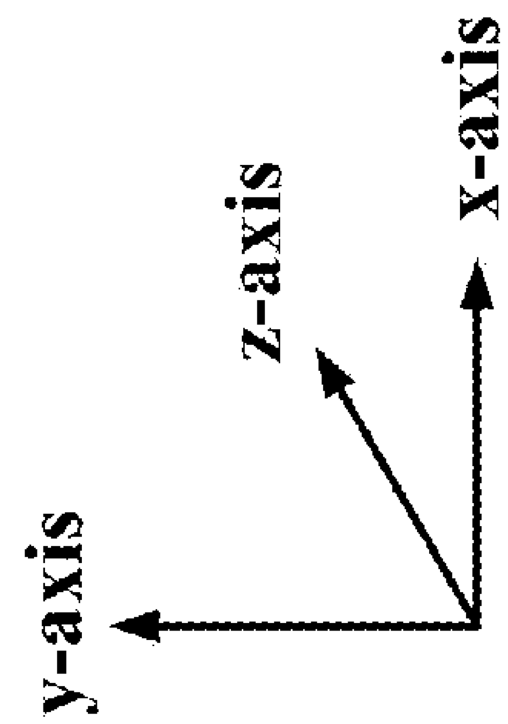


Figure 3b



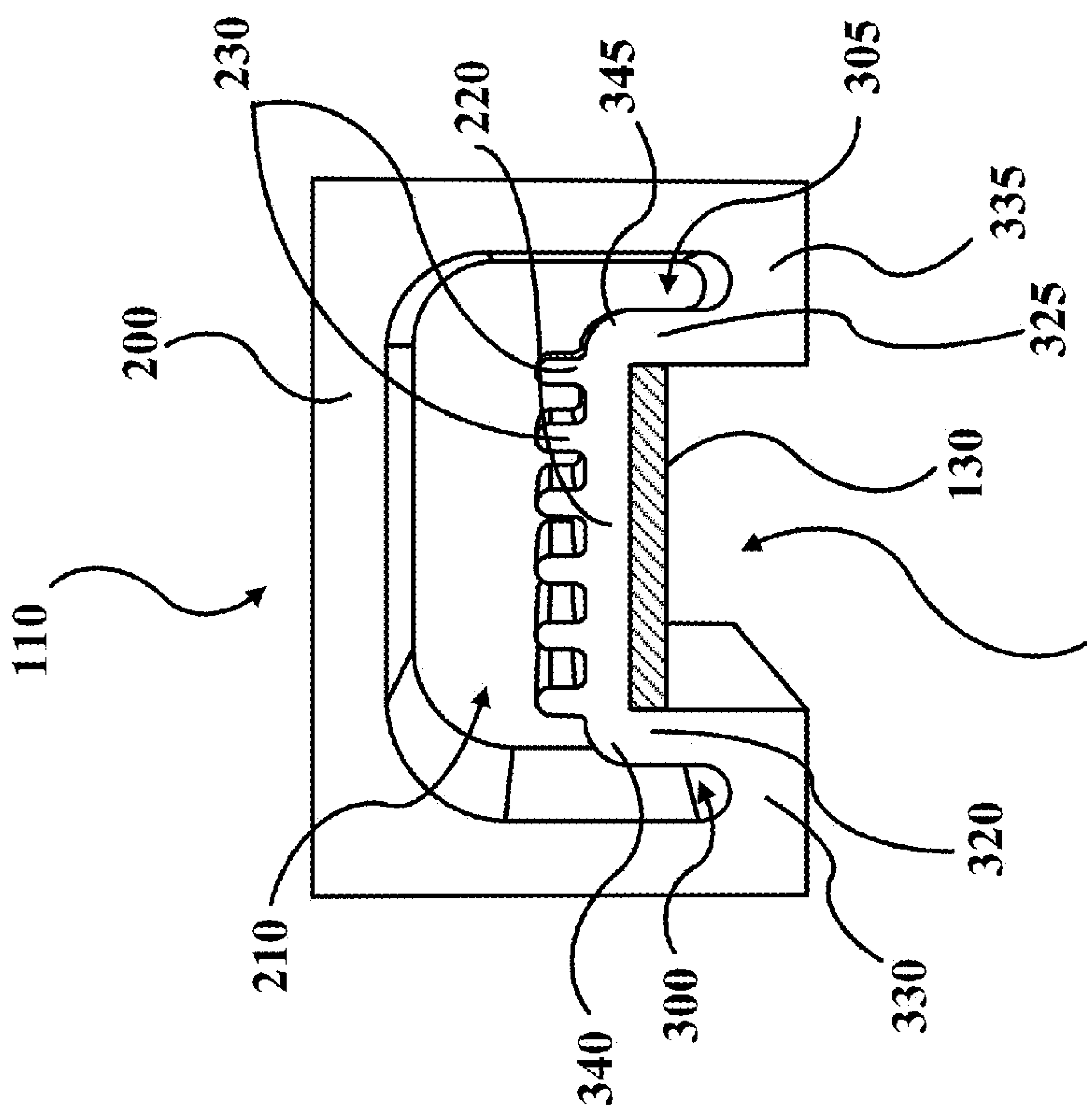


Figure 3a

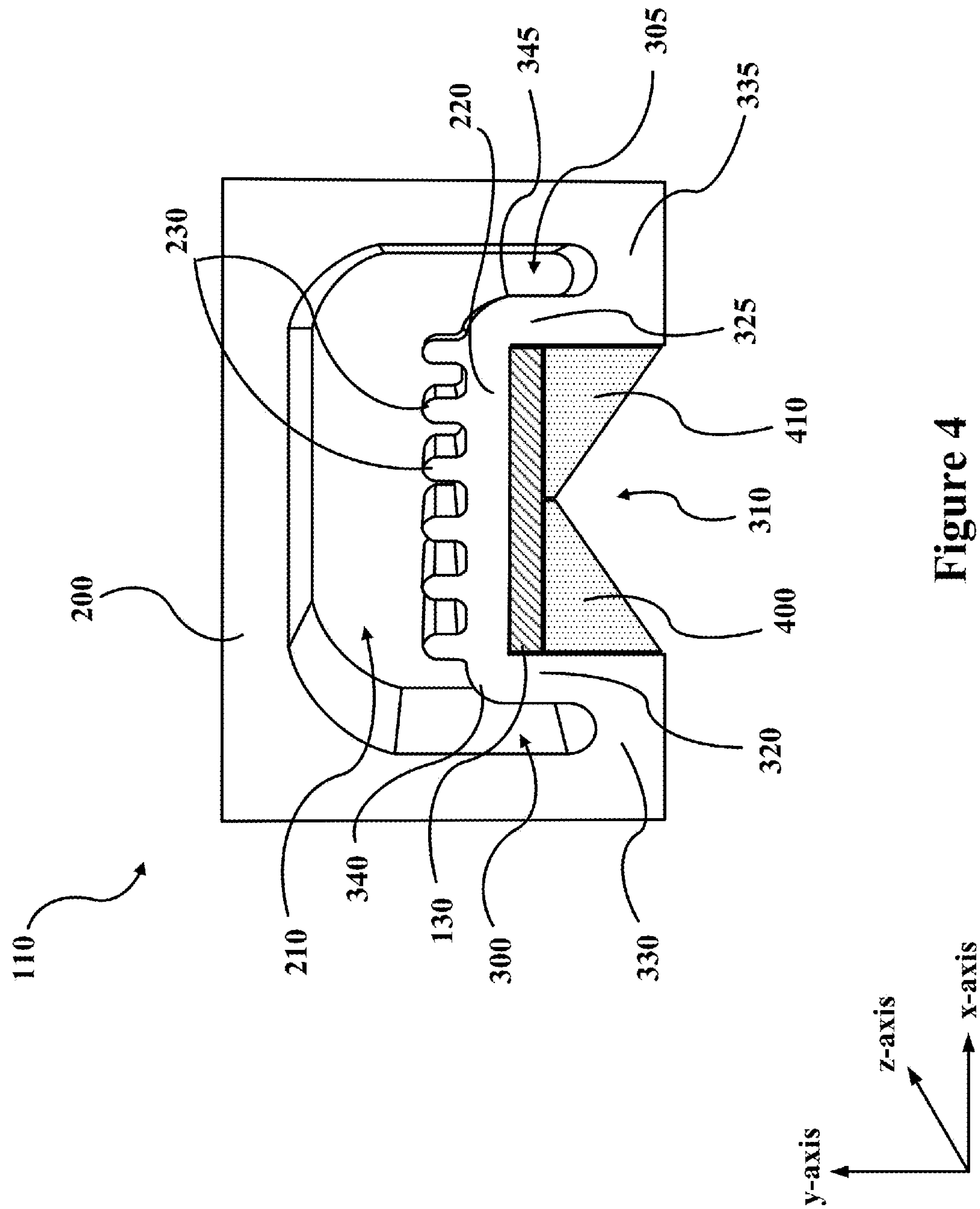

110
200
210
220
230
300
305
310
320
325
330
335
340
345
130
400
410
y-axis
z-axis
x-axis

Figure 4

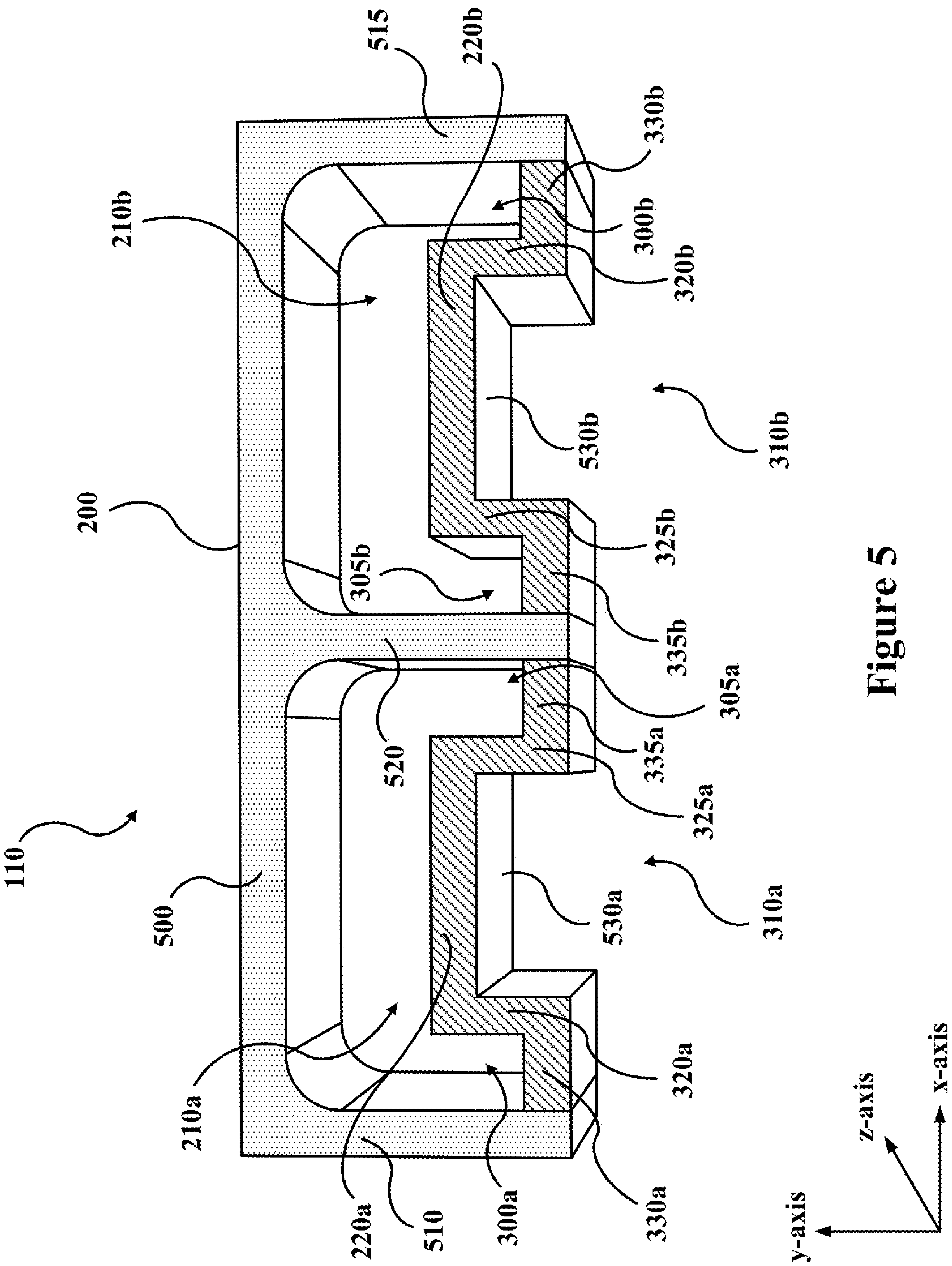

110
500
200
515
210b
220b
330b
300b
320b
530b
310b
305b
325b
335b
520
305a
335a
325a
530a
310a
210a
320a
220a
510
300a
330a
y-axis
z-axis
x-axis

Figure 5

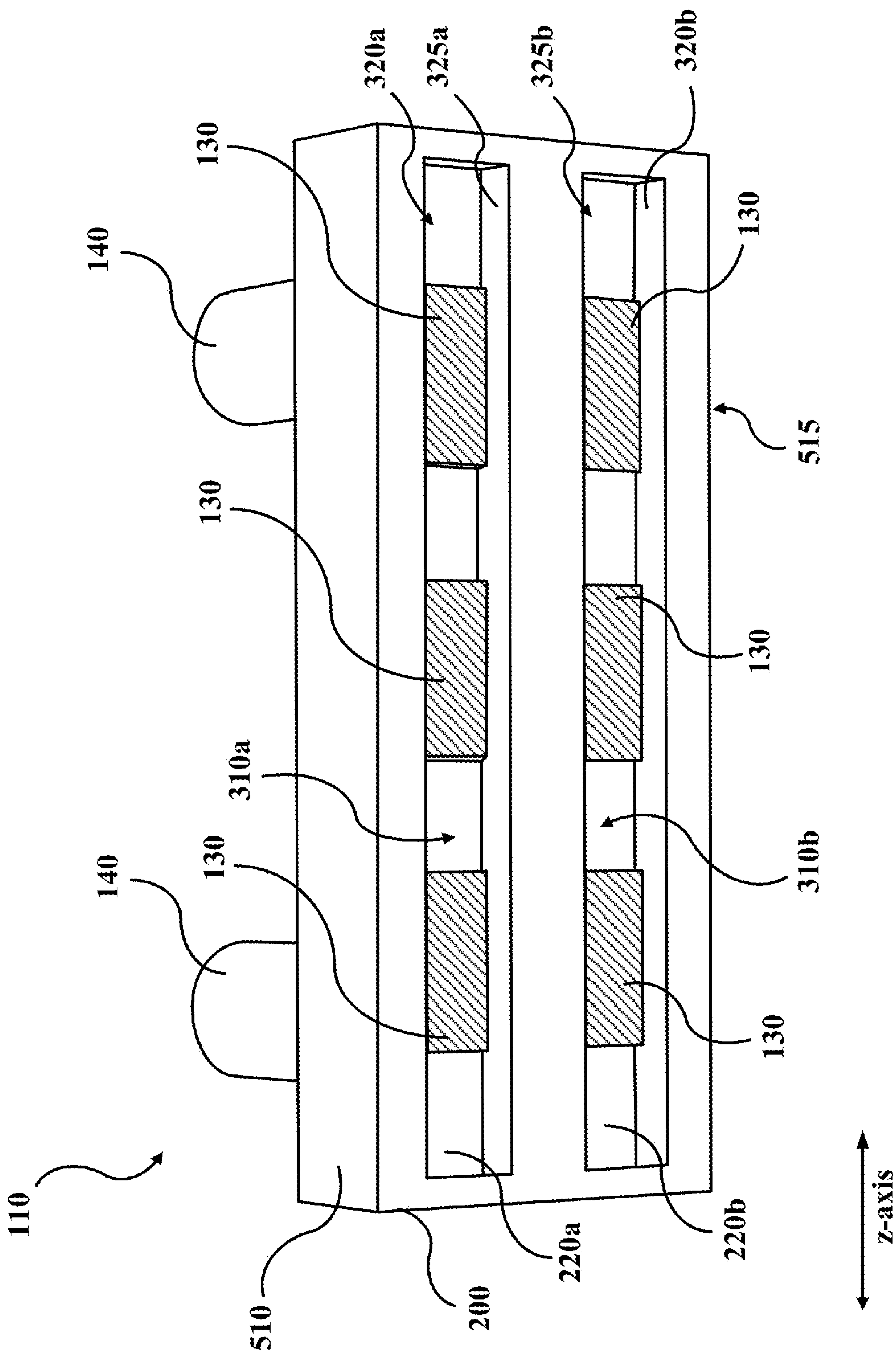

110
140
140
130
130
130
130
130
130
320a
325a
325b
320b
310a
310b
515
510
200
220a
220b
z-axis

Figure 6

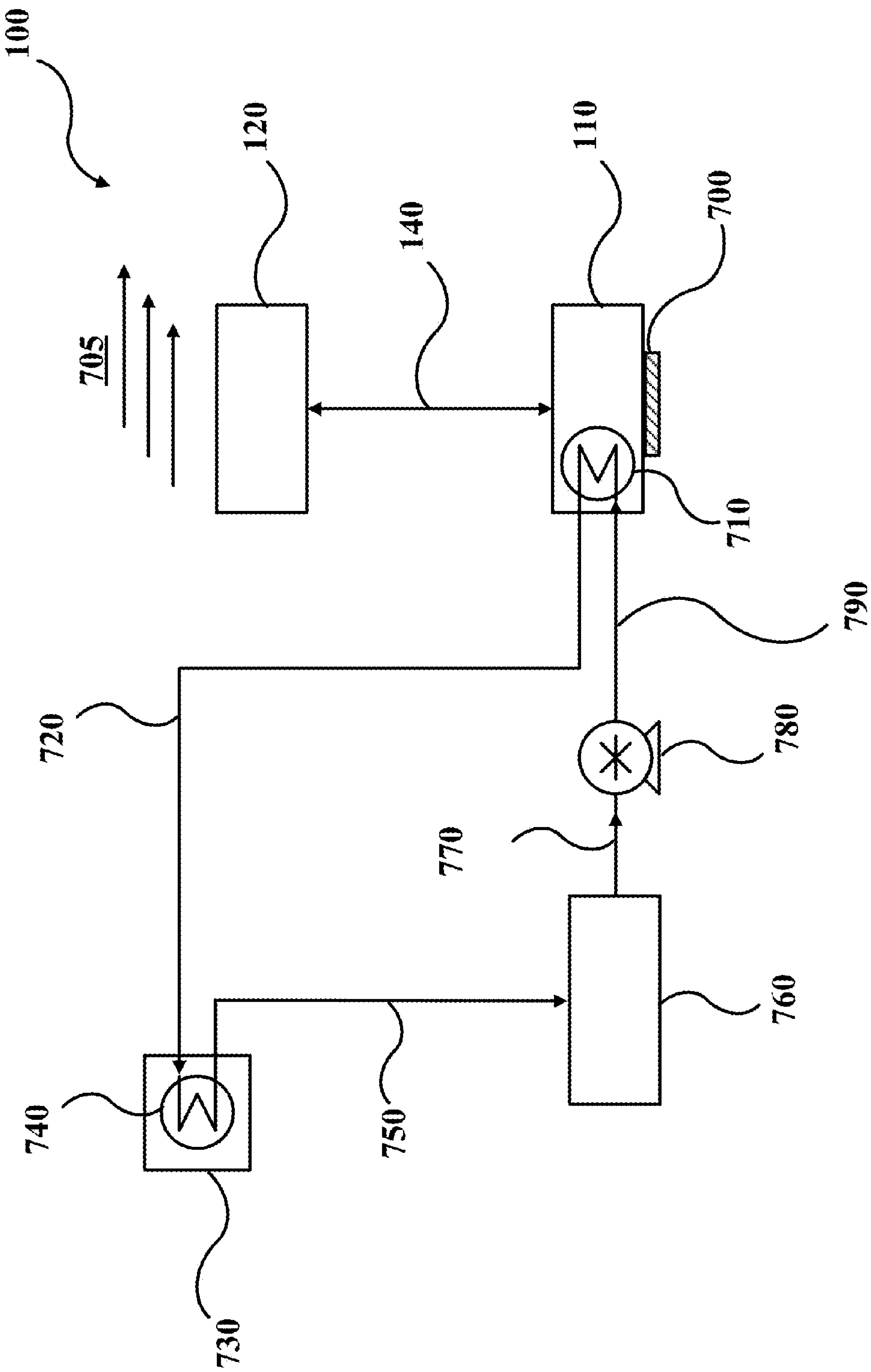


Figure 7

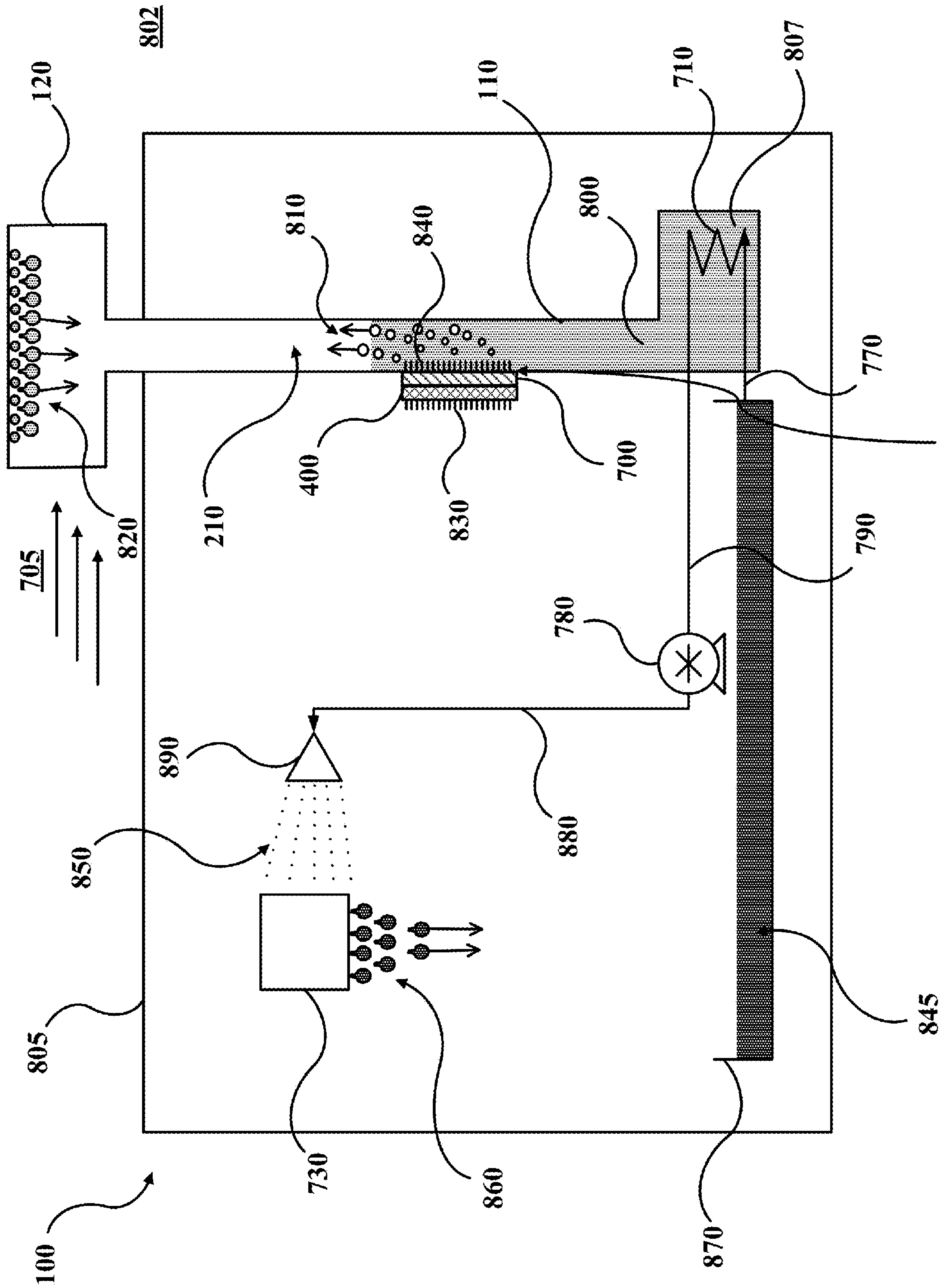

802
120
110
710
807
810
840
800
770
220
700
400
830
210
820
705
790
780
890
880
850
805
845
730
860
870
100

Figure 8

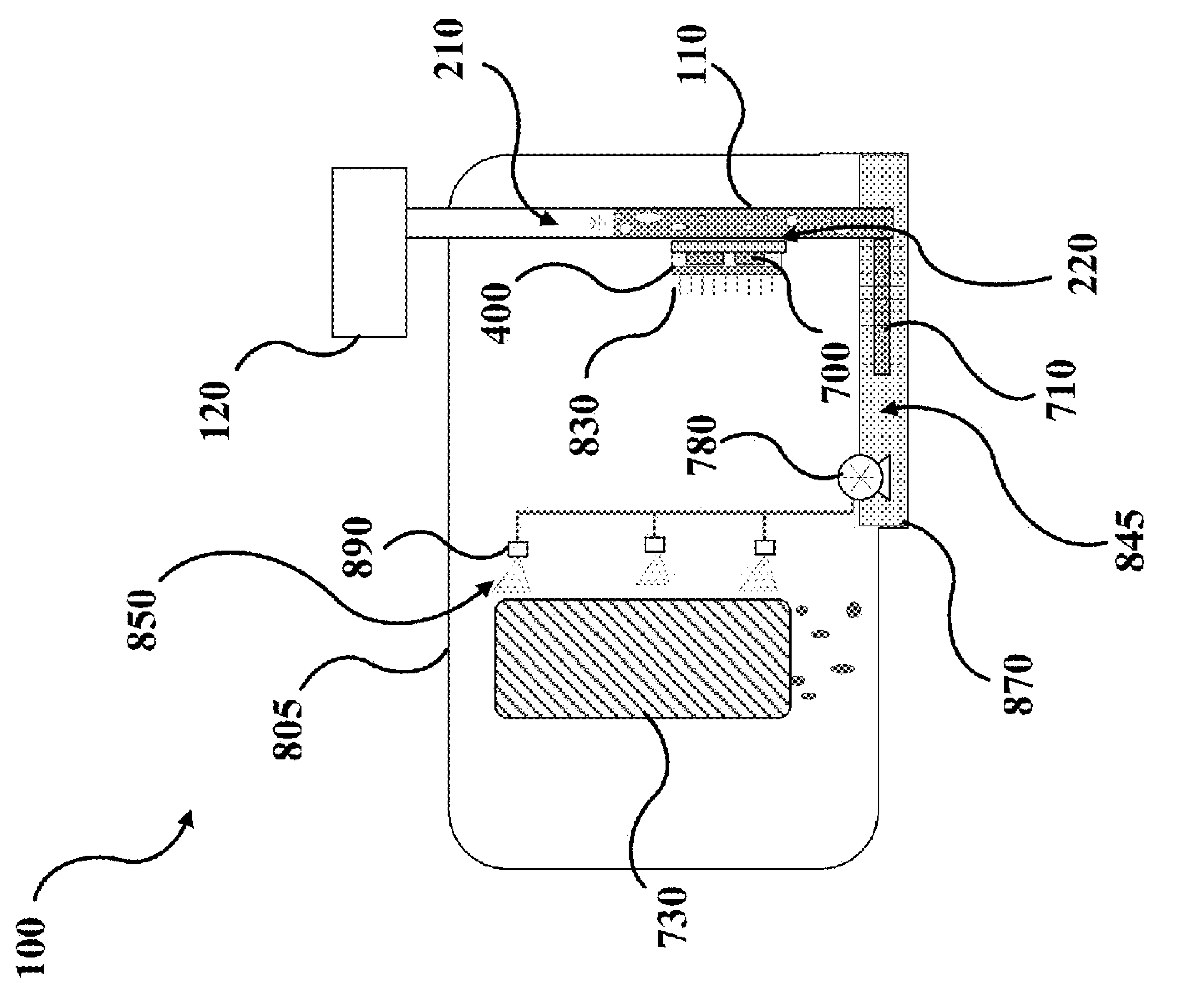


Figure 9a

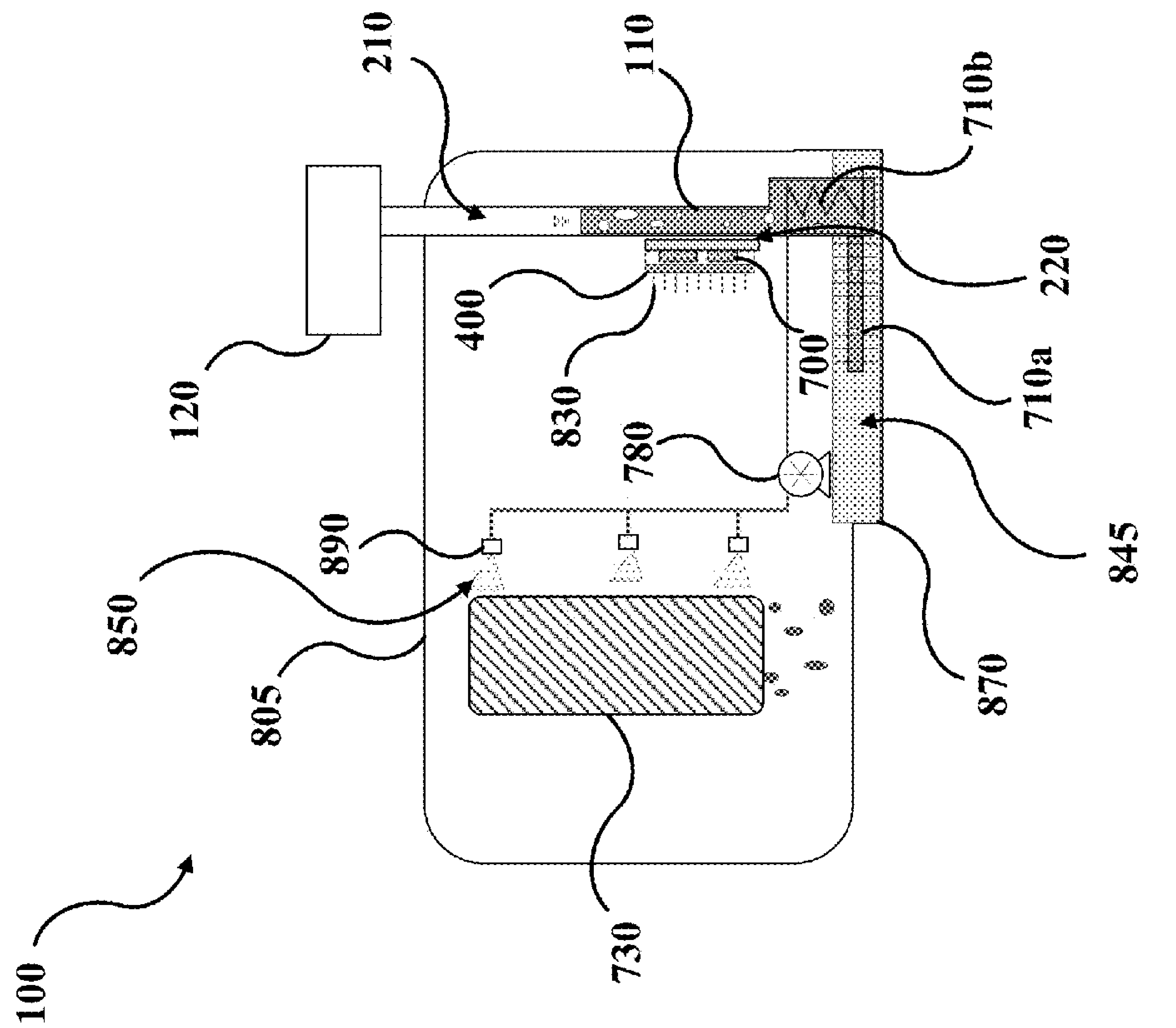


Figure 9b

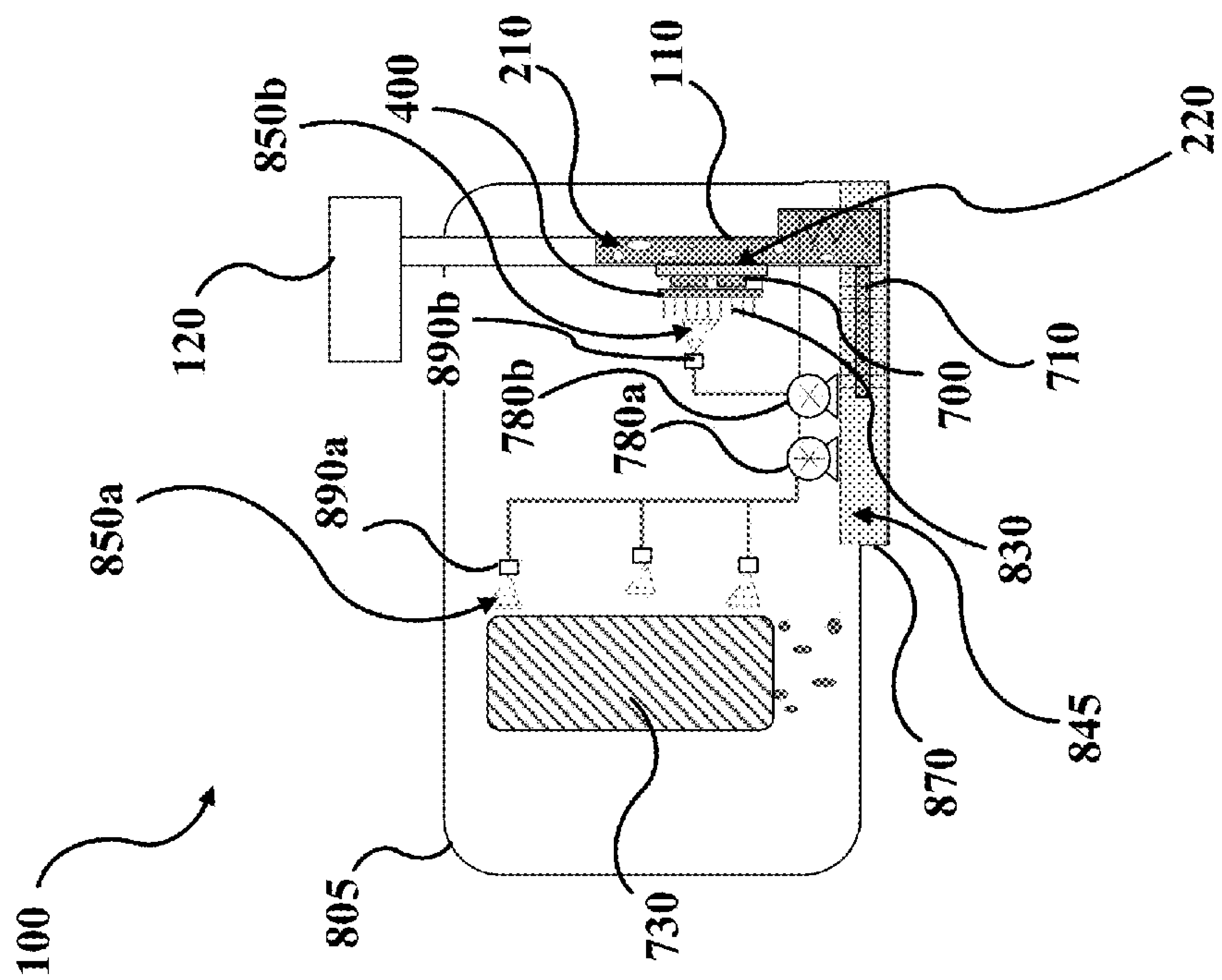

100
805
730
850a
890a
870
845
780a
780b
890b
830
700
710
120
850b
400
210
110
220

Figure 10

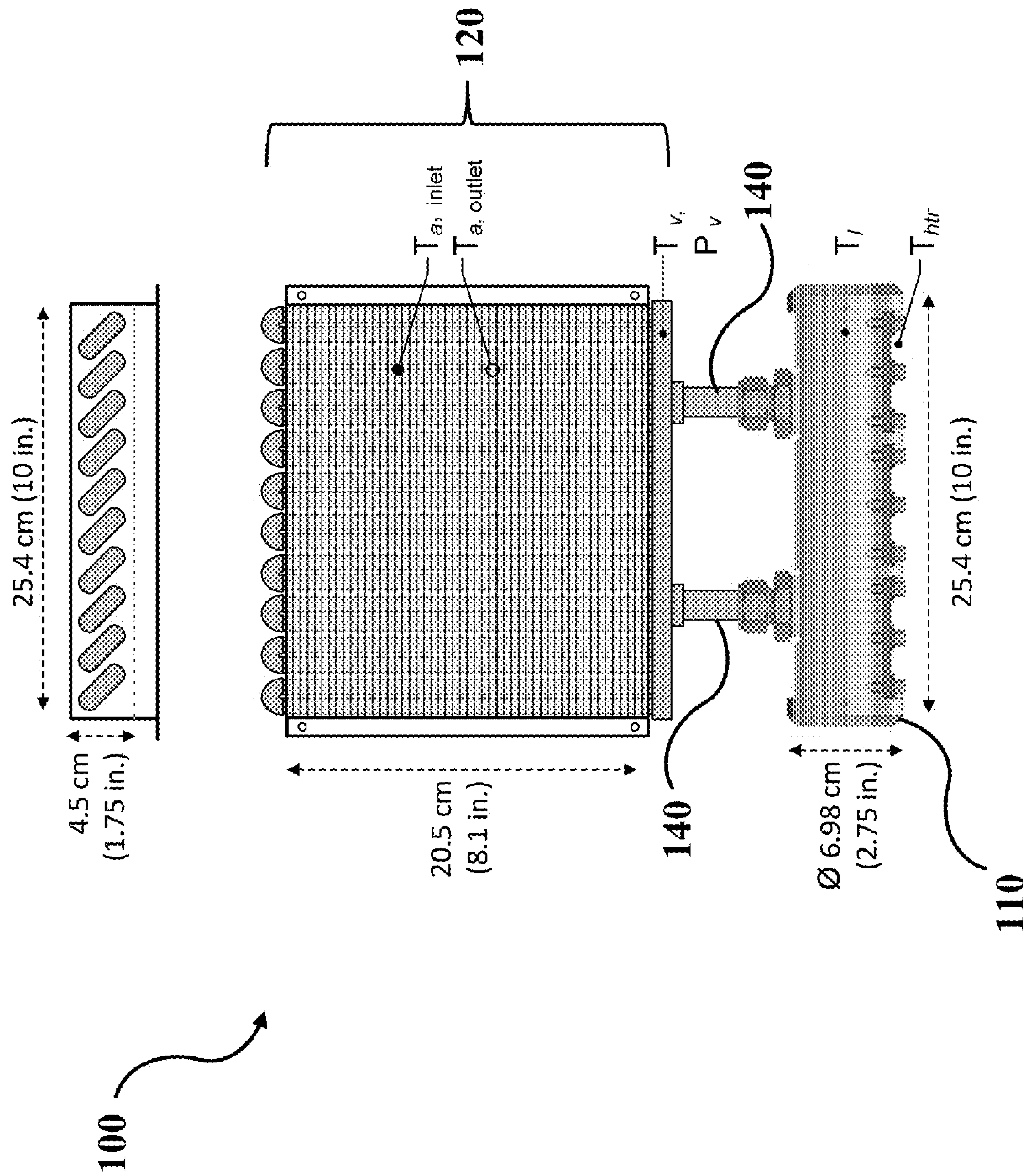

100
25.4 cm (10 in.)
4.5 cm (1.75 in.)
120
$T_{a,\ inlet}$
$T_{a,\ outlet}$
20.5 cm (8.1 in.)
T_{v},
P_{v}
140
140
T_{l}
T_{htr}
Ø 6.98 cm (2.75 in.)
25.4 cm (10 in.)
110

Figure 11

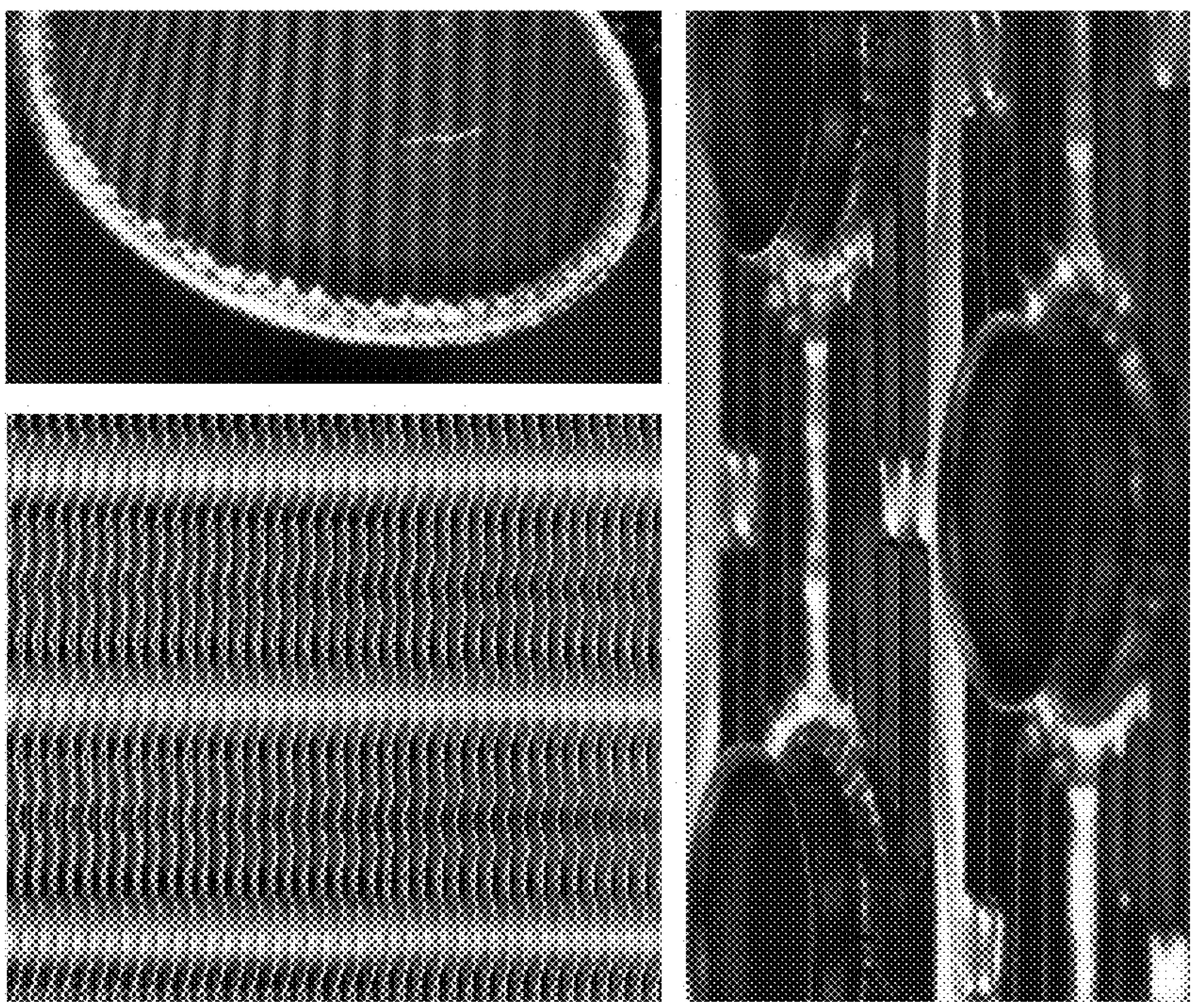


Figure 12

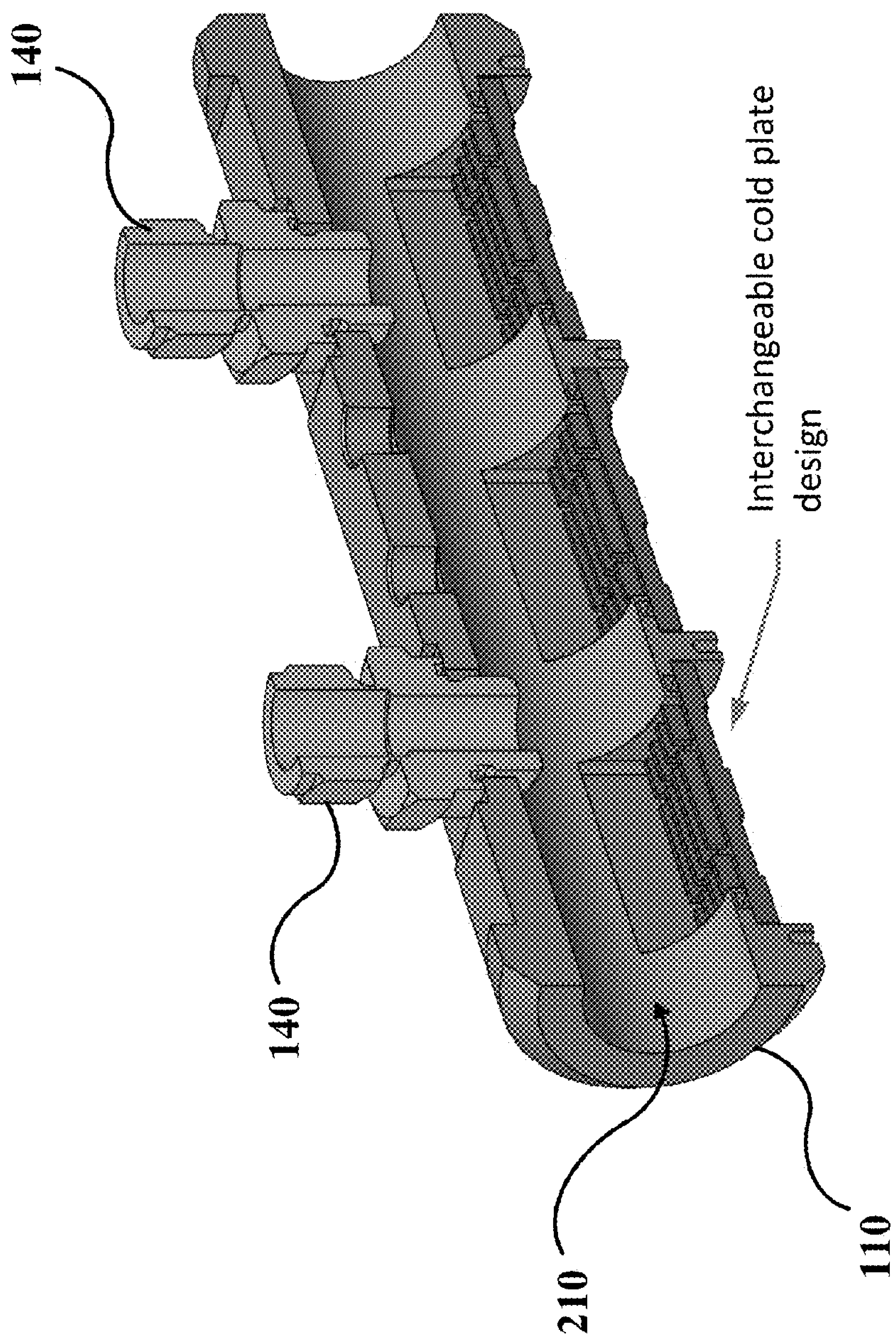


Figure 13

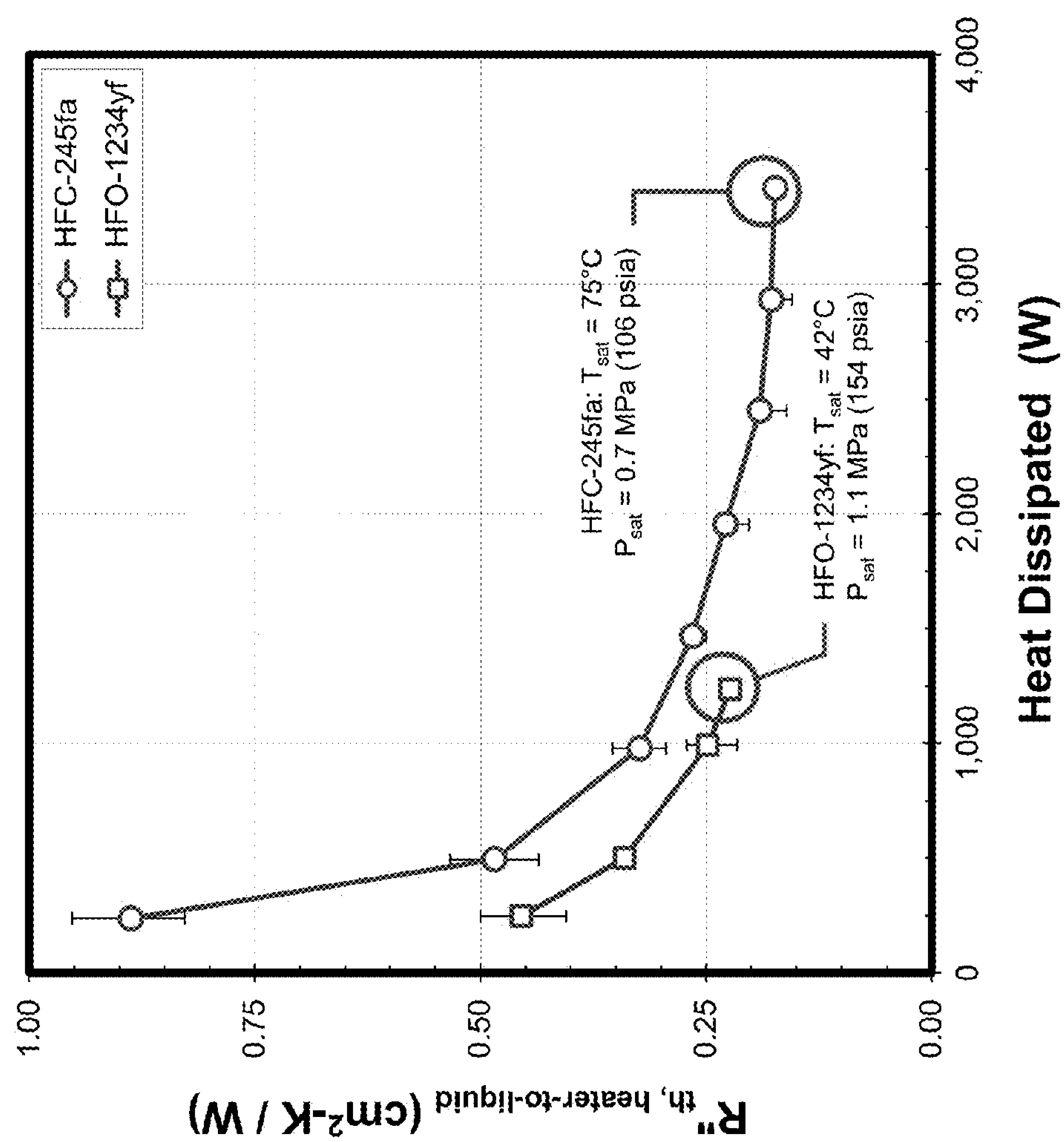

HFC-245fa
HFO-1234yf
HFC-245fa: T_{sat} = 75°C
P_{sat} = 0.7 MPa (106 psia)
HFO-1234yf: T_{sat} = 42°C
P_{sat} = 1.1 MPa (154 psia)
R"$_{th, heater-to-liquid}$ (cm^2-K / W)
1.00
0.75
0.50
0.25
0.00
0
1,000
2,000
3,000
4,000
Heat Dissipated (W)

Figure 14

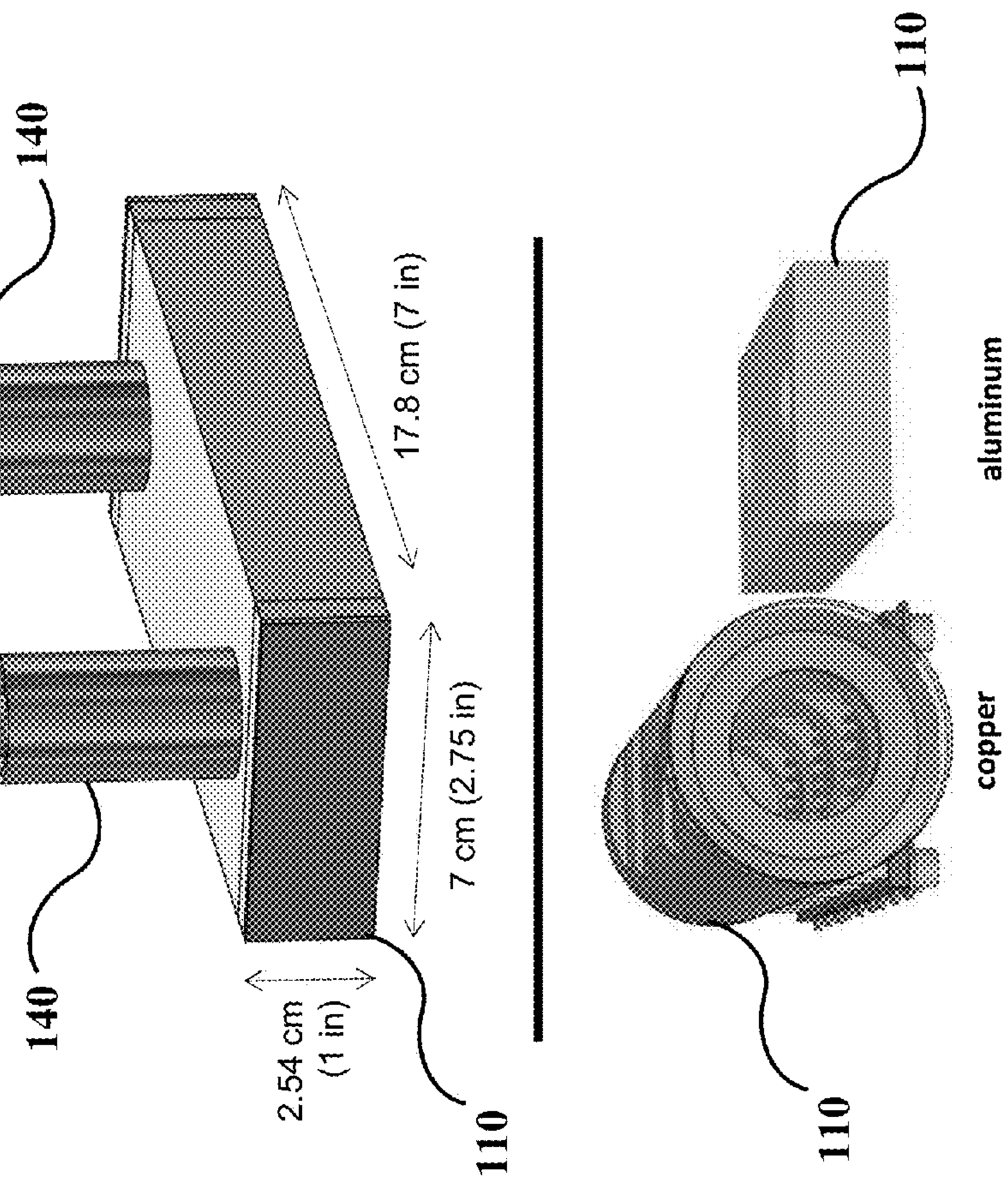

140
140
17.8 cm (7 in)
7 cm (2.75 in)
2.54 cm
(1 in)
110
110
110
copper
aluminum

Figure 15

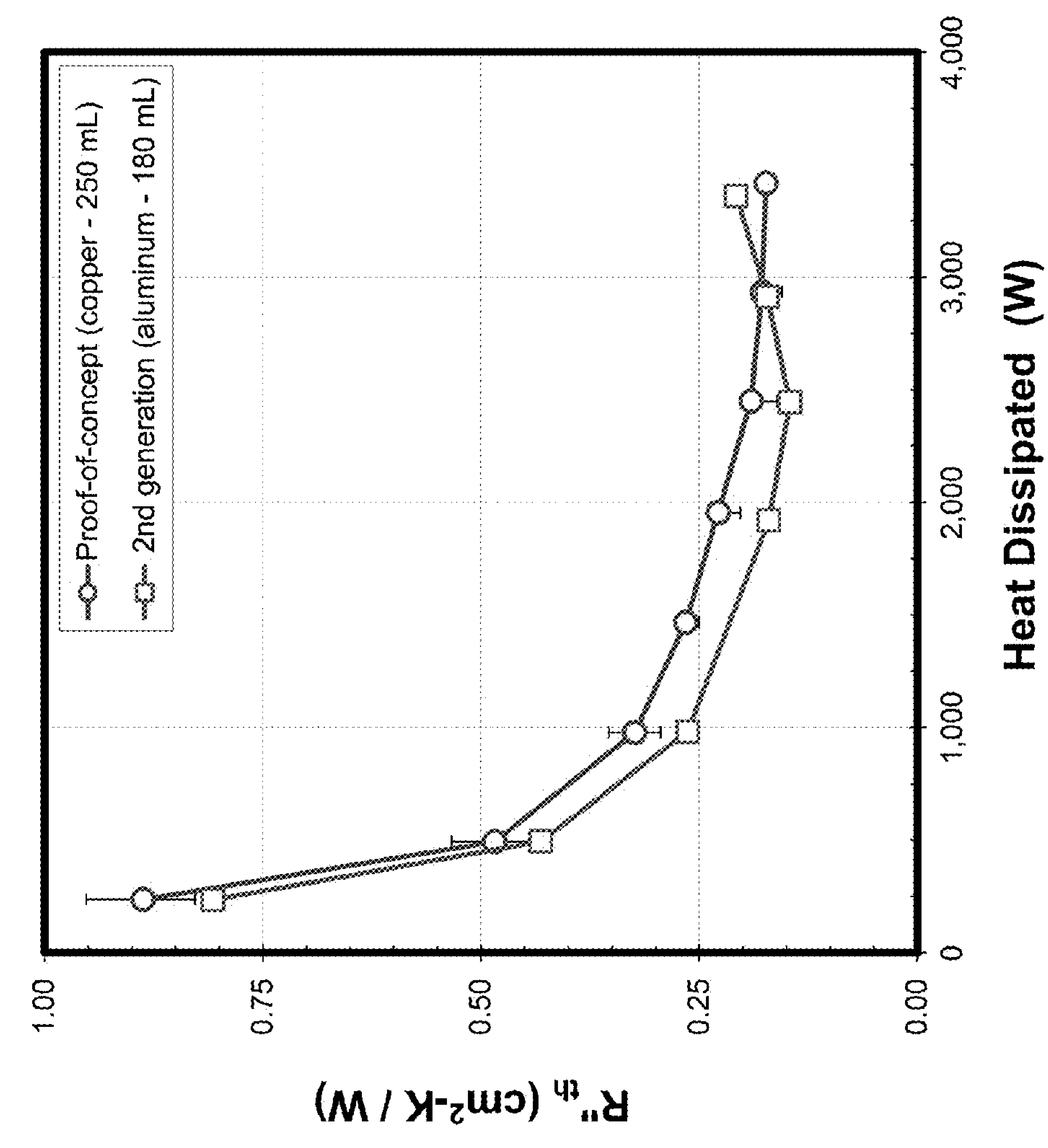

Proof-of-concept (copper - 250 mL)
2nd generation (aluminum - 180 mL)
R''th (cm²-K / W)
1.00
0.75
0.50
0.25
0.00
Heat Dissipated (W)
0
1,000
2,000
3,000
4,000

Figure 16

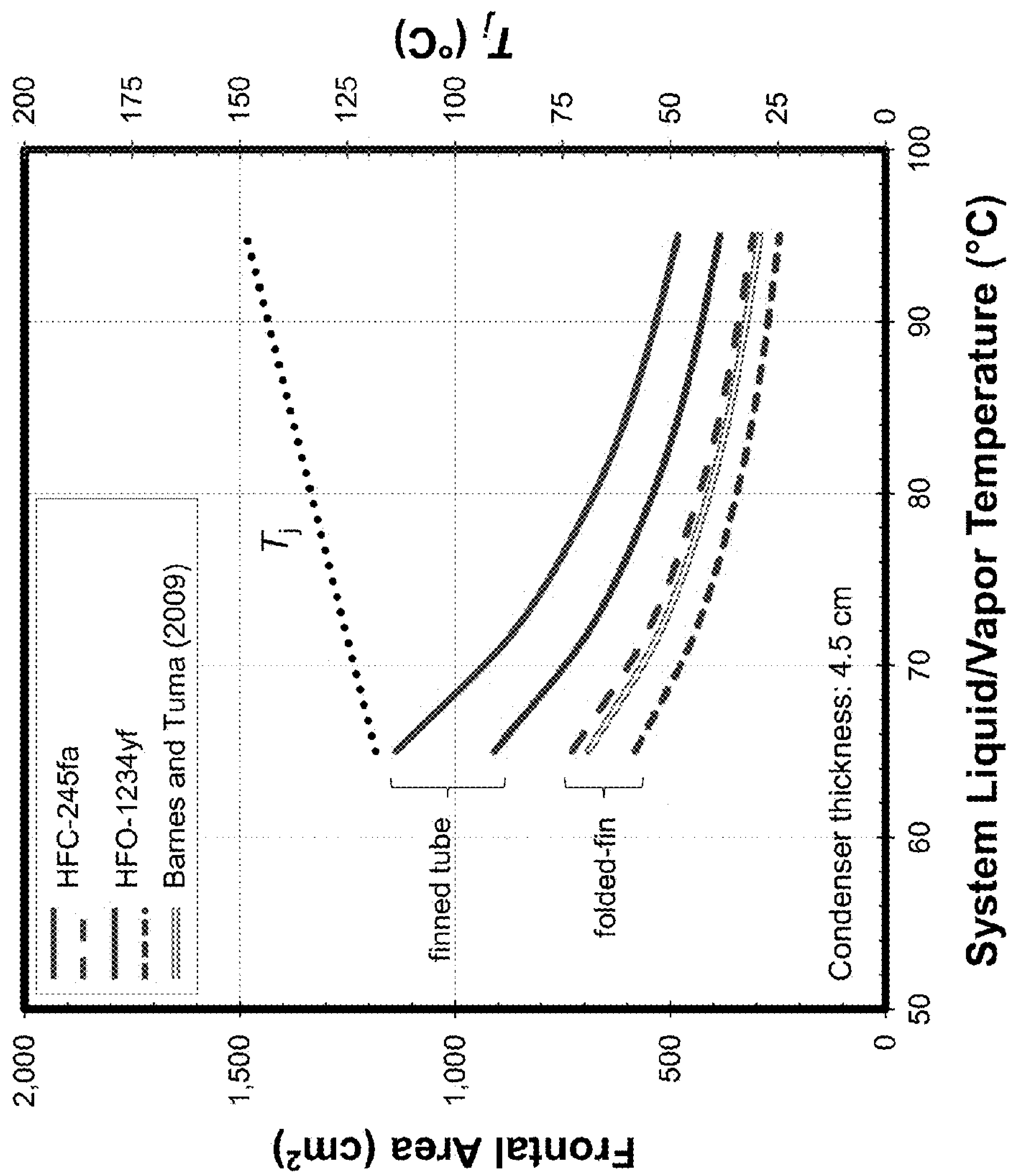

HFC-245fa
HFO-1234yf
Barnes and Tuma (2009)
T_j
finned tube
folded-fin
Condenser thickness: 4.5 cm
Frontal Area (cm²)
2,000
1,500
1,000
500
0
T_j (°C)
200
175
150
125
100
75
50
25
0
50
60
70
80
90
100
System Liquid/Vapor Temperature (°C)

Figure 17

TWO-PHASE HEAT EXCHANGER FOR COOLING ELECTRICAL COMPONENTS

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application claims priority to and the benefit of U.S. Provisional Application No. 62/069,983, filed on Oct. 29, 2014, entitled "Two-Phase Heat Exchanger for Power Electronics Cooling" and is herein incorporated by reference in its entirety.

CONTRACTUAL ORIGIN

[0002] The United States Government has rights in this invention under Contract No. DE-AC36-08GO28308 between the United States Department of Energy and the Alliance for Sustainable Energy, LLC, the Manager and Operator of the National Renewable Energy Laboratory.

BACKGROUND

[0003] The size, weight, and cost of power electronic components are factors that influence the cost of hybrid and electric vehicles. According to a report from the Oak Ridge National Laboratory (Energy and Environmental Analysis, Inc., 2007, "Technology and Cost of the MY2007 Toyota Camry HEV—Final Report," Technical Report No. ORNL/TM-20071132), power electronics accounts for up to 40% of the total traction drive cost in hybrid vehicles. Increasing vehicle electrification, in an effort to reduce the nation's dependence on foreign oil, requires making electric drive vehicles cost competitive with conventional gasoline powered vehicles. For these reasons, the Department of Energy (DOE) has defined cost, power density, specific power, and efficiency technical targets aimed at decreasing the cost and increasing the efficiency of these components. According to the DOE, reaching these cost targets requires a 4.1-fold and 2.5-fold reduction to the cost of power electronics and electric motors, respectively. Additionally, the DOE has highlighted three strategies to enable reaching their technical targets: 1) fully integrated components (i.e., integrating the power electronics with the motor), 2) using wide-bandgap (WBG) semiconductors, and 3) using non-rare earth element motors. One means of reducing the cost of hybrid and electric vehicles is through reduced cost, weight, and size of automotive power electronics.

[0004] However, heat dissipation is a limiting factor in reducing the size and cost of the power electronic devices. Current power electronic semiconductor devices are sized larger to spread heat and thus allow for reliable operation. Significant cost reductions can be achieved by decreasing the size of these semiconductor electronic devices. Increasing the heat dissipation through the use of highly efficient cooling schemes allows for greater power density (heat per volume), which in turn reduces the size, weight, and cost of power electronics.

[0005] It is estimated that almost 6 million hybrid electric vehicles (HEV) have been sold worldwide and their sales are expected to grow in future years. This increasing trend towards vehicle electrification has also increased the demands for plug-in hybrid electric (PHEVs) and electric vehicles (EVs). All electric vehicles (e.g., HEV, PHEV, and EV) require power electronic systems and thus could benefit from improved cooling technology methods and systems. Two-phase cooling offers some of the highest heat-transfer rates, higher than those possible with conventional, single-phase liquid cooling systems typical of automotive heat exchangers. Although some researchers have developed two-phase cooling systems for automotive electronics (see U.S. Pat. No. 6,993,924 and U.S. Patent Application Publication No. 2012/0267077), none of these concepts are currently used in automotive power electronics cooling systems. Thus, there remains a need for improved two-phase cooling system designs that can meet technical requirements, while also reducing cost, to enable electric vehicles a more effective entry into current automotive markets.

SUMMARY

[0006] An aspect of the present invention is an evaporator having a first wall, where the first wall has an external surface, a first edge, and a second edge. The first wall, the first edge, and the second edge are substantially parallel to a plane, and the first edge and the second edge are substantially parallel to each other. The evaporator also has a second wall extending from the first edge and the external surface, and the second wall is substantially perpendicular to the plane, and substantially parallel to the first edge. The evaporator also has a third wall extending from the second edge and the external surface, and the third wall is substantially perpendicular to the plane, and substantially parallel to the second edge. The external surface of the first wall, the second wall, and the third wall form a passage substantially parallel to the first edge. The passage is configured to contain at least one heat source, and at least the first wall is configured to be in thermal communication with the at least one heat source.

[0007] In some embodiments of the present invention, the evaporator may have a conducting element, where the conducting element is positioned within the passage. The conducting element is configured to be in thermal communication with the at least one heat source, and the conducting element is in thermal communication with at least one of the second wall or the third wall. In some embodiments of the present invention, a conducting element may be a block of material with a substantially triangular cross-section, where the block of material has a first side that is in thermal communication with either the second wall or the third wall, and the block has a second side that is configured to be in thermal communication with the at least one heat source.

[0008] In some embodiments of the present invention, a conducting element may include a first block of material with a substantially triangular cross-section, and a second block of material with a substantially triangular cross-section. The first block of material may have a first side that is in thermal communication with the second wall, and a second side that is configured to be in thermal communication with the at least one heat source. The second block of material may have a first side that is in thermal communication with the third wall, and a second side that is configured to be in thermal communication with the at least one heat source.

[0009] In some embodiments of the present invention, a first wall of an evaporator may have an internal surface with a surface area extender extending from the internal surface. The internal surface may be substantially parallel to the external surface, and the surface area extender may be substantially perpendicular to the plane. In some embodiments of the present invention, a surface area extender may be a fin, where the fin is substantially parallel to the first edge. In some embodiments of the present invention, the internal surface and the external surface may define a width of the first wall, where the fin may be at a height that is approximately equal to the width.

[0010] In some embodiments of the present invention, the second wall may terminate with an edge, the third wall may terminate with an edge, and the edge of the second wall and the edge of the third wall may be positioned below the external surface. The evaporator may also include a first tab extending from the edge of the second wall, where the first tab is substantially perpendicular to the second wall, and extends away from the passage. The evaporator may also include a second tab extending from the edge of the third wall, where the second tab is substantially perpendicular to the third wall, and extends away from the passage.

[0011] In some embodiments of the present invention, an evaporator may have a housing where the housing has a first end connected to the first tab, a second end connected to the second tab, and at least one wall physically connecting the first end to the second end. The housing may form a first interior channel between the interior surface of the first wall and the housing. The housing may form a second interior channel between the second wall and the housing, and the housing may form a third channel between the third wall and the housing. In some embodiments of an evaporator, the first wall, the second wall, the third wall, the first tab, the second tab, and the housing may be a single piece of material. A single piece of material may be aluminum.

[0012] In some embodiments of the present invention, the first wall, the second wall, the third wall, the first tab, and the second tab of an evaporator may all be a first single piece of material. The first single piece of material may be aluminum. The housing may be a second single piece of material. The second single piece of material may be copper.

[0013] A further aspect of the present invention is a cooling system having an evaporator configured to cool a first heat source, a condenser in fluid communication with the evaporator, a refrigerant contained within the condenser and the evaporator, and a circulating liquid system. The circulating liquid system may include a liquid contained within the circulating liquid system, a liquid reservoir, a pump in liquid communication with the liquid reservoir, and a first heat exchanger in liquid communication with the pump, where the first heat exchanger is configured to deliver heat from the liquid to the refrigerant. The circulating liquid system may also include a second heat exchanger in liquid communication with the first heat exchanger. The second heat exchanger may be configured to deliver heat from a second heat source to the liquid, and the second heat exchanger may be in liquid communication with the liquid reservoir. The pump may circulate the liquid through the first heat exchanger, the second heat exchanger, and the liquid reservoir.

[0014] In some embodiments of the present invention, the second heat exchanger may have a spray nozzle, such that the liquid may be sprayed onto the second heat source. The liquid reservoir may have a liquid collection pan configured to collect oil heated by the second heat source. In some further embodiments of the present invention, the evaporator and the circulating liquid system may be fluid-sealed within a container operating at a first average temperature. In still further embodiments, the condenser may operate at a second average temperature that is less than the first average temperature.

BRIEF DESCRIPTION OF THE DRAWINGS

[0015] Exemplary embodiments are illustrated in referenced figures of the drawings. It is intended that the embodiments and figures disclosed herein are to be considered illustrative rather than limiting.

[0016] FIG. **1** illustrates a cooling system for removing heat from one or more heat sources, for example power electronics, according to exemplary embodiments of the present invention.

[0017] FIGS. **2***a* and **2***b* illustrate evaporators for removing heat from one or more heat sources, for example a power electronics module, according to exemplary embodiments of the present invention.

[0018] FIGS. **3***a* and **3***b* illustrate evaporators for removing heat from one or more heat sources, for example power electronics modules, according to exemplary embodiments of the present invention.

[0019] FIG. **4** illustrates an evaporator for removing heat from one or more heat sources, for example a power electronic module, according to exemplary embodiments of the present invention.

[0020] FIG. **5** illustrates an evaporator for removing heat from two or more heat sources where the evaporator includes multiple parallel internal channels, and constructed from more than one single piece of material joined together, according to exemplary embodiments of the present invention.

[0021] FIG. **6** illustrates an evaporator having two external passages configured to receive six heat-generating elements, according to exemplary embodiments of the present invention.

[0022] FIG. **7** illustrates a system for simultaneously removing heat from two different heat sources, for example a power electronics module and a motor, according to exemplary embodiments of the present invention.

[0023] FIG. **8** illustrates a system for simultaneously removing heat from two different heat sources, for example a power electronics module and a motor, utilizing recirculating cooling oil, where the oil is sprayed onto the motor, according to exemplary embodiments of the present invention.

[0024] FIGS. **9***a* and **9***b* illustrate systems for simultaneously removing heat from both power electronics and a motor, utilizing a recirculating cooling oil loop, according to exemplary embodiments of the present invention.

[0025] FIG. **10** illustrates a system for simultaneously removing heat from both power electronics and a motor, utilizing a recirculating cooling oil system, according to exemplary embodiments of the present invention.

[0026] FIG. **11** illustrates a schematic of a passive, two-phase cooling system, according to exemplary embodiments of the present invention.

[0027] FIG. **12** illustrates images of a finned-tube condenser (top left), a rifled tube condenser (top right), and a louvered-fin condenser (bottom), according to exemplary embodiments of the present invention.

[0028] FIG. **13** illustrates a cross-sectional view of an evaporator incorporating an interchangeable cold plate design, according to exemplary embodiments of the present invention.

[0029] FIG. **14** illustrates experimental data for thermal resistances to heat flow versus heat dissipated for two refrigerants for an evaporator.

[0030] FIG. **15** illustrates mechanical drawings of two evaporators for cooling one or more heating sources, according to exemplary embodiments of the present invention.

[0031] FIG. **16** illustrates experimental data of thermal resistances to heat flow versus heat dissipated for two evaporators.

[0032] FIG. **17** illustrates estimates of condenser sizing requirements for various two-phase heat-transfer fluid operating temperatures, according to exemplary embodiments of the present invention.

REFERENCE NUMBERS

[0033] **100** . . . cooling system
[0034] **110** . . . evaporator
[0035] **120** . . . condenser
[0036] **130** . . . heat source
[0037] **140** . . . vapor/condensate line
[0038] **150** . . . manifold
[0039] **160** . . . tube
[0040] **170** . . . bend
[0041] **180** . . . fins
[0042] **200** . . . housing
[0043] **210** . . . main channel
[0044] **220** . . . first wall
[0045] **230** . . . surface area extender
[0046] **300** . . . first side channel
[0047] **305** . . . second side channel
[0048] **310** . . . passage
[0049] **320** . . . second wall
[0050] **325** . . . third wall
[0051] **330** . . . first tab
[0052] **335** . . . second tab
[0053] **340** . . . first edge
[0054] **345** . . . second edge
[0055] **347** . . . connecting tab
[0056] **350** . . . shared side channel
[0057] **400** . . . first conducting element
[0058] **410** . . . second conducting element
[0059] **500** . . . spanning wall
[0060] **510** . . . first side wall
[0061] **515** . . . second side wall
[0062] **520** . . . dividing wall
[0063] **530** . . . insert
[0064] **700** . . . power electronics module
[0065] **705** . . . cooling fluid
[0066] **710** . . . first heat exchanger
[0067] **720** . . . first return line
[0068] **730** . . . motor
[0069] **740** . . . second heat exchanger
[0070] **750** . . . second return line
[0071] **760** . . . liquid reservoir
[0072] **770** . . . first supply line
[0073] **780** . . . pump
[0074] **790** . . . second supply line
[0075] **800** . . . refrigerant
[0076] **802** . . . outside environment
[0077] **805** . . . containment system
[0078] **807** . . . evaporator reservoir
[0079] **810** . . . refrigerant vapor
[0080] **820** . . . refrigerant condensate
[0081] **830** . . . first surface area extenders
[0082] **840** . . . second surface are extenders
[0083] **845** . . . oil
[0084] **850** . . . oil spray
[0085] **860** . . . heated oil
[0086] **870** . . . oil collection pan
[0087] **880** . . . third supply line
[0088] **890** . . . spray nozzle

DETAILED DESCRIPTION OF SOME EMBODIMENTS

[0089] The present disclosure may address one or more of the problems and deficiencies of the prior art discussed above. However, it is contemplated that some embodiments as disclosed herein may prove useful in addressing other problems and deficiencies in a number of technical areas. Therefore, the embodiments described herein should not necessarily be construed as limited to addressing any of the particular problems or deficiencies discussed herein.

[0090] The methods and systems described herein utilize a two-phase cooling strategy with other unique features (e.g. dual-side cooling, increased heat spreading, and extrudable fabrication design) that have the potential to significantly increase power densities and thus reduce the size and cost of power electronics. The high heat-transfer capacity and isothermal characteristics of two-phase heat-transfer enables the cooling systems described herein to outperform conventional automotive cooling systems that employ single-phase liquid cooled heat exchangers. This has been demonstrated through a combination of experimental work and modeling analysis. The two-phase cooling concepts described herein utilize a passive cooling approach (e.g. no pumping or compression of the two-phase heat-transfer fluid contained within the condenser and the evaporator) and an evaporator design that increases heat dissipation, reliability, and efficiency, resulting in reduced cooling system size and/or weight. The increased heat dissipation provided by the two-phase based cooling system has the potential to increase power density and reliability. Increasing the power density and reliability of automotive power electronics are paths to reducing the cost of electric-drive vehicles (e.g., hybrids, plug-in hybrids, all electric, and fuel cell).

[0091] The cooling systems described herein utilize a two-phase heat-transfer fluid cooling system. The term "two-phase heat-transfer fluid" refers to a fluid that is reversibly cycled between the fluid's liquid phase and vapor phase. By utilizing a two-phase heat-transfer fluid within a cooling system having an evaporator and a condenser, heat may be removed from a heat-generating element (e.g. power electronics) by vaporizing the heat-transfer fluid in the evaporator and subsequently condensing the fluid in the condenser. Since this process utilizes the latent heat of vaporization of the two-phase heat-transfer fluid, such a system can be very efficient in terms of energy removed from the heat-generating element per unit mass of heat-transfer fluid utilized, and in terms of energy removed per unit surface area of the evaporator and/or condenser. Examples of two-phase heat-transfer fluids that may be used in cooling systems described herein include various refrigerants/coolants including but not limited to R-134a, R-1234yf, R-245fa, HFE-7000, HFE-7100, Novec 649, and/or any other suitable fluid capable of reversibly vaporizing and condensing.

[0092] FIG. **1** illustrates an exemplary cooling system **100** that includes an evaporator **110** and a condenser **120**, connected by two vapor/condensate lines **140**, and configured to cool a plurality of heat sources **130** (e.g. electronics). In this example, the condenser **120** is constructed using multiple tubes **160**, with each tube oriented in a substantially vertical position (e.g. along the y-axis and relative to the XZ-plane). However, the vertical position is for illustrative purposes, and other examples of cooling systems may position condenser tubes at an angle other than 90 degrees from horizontal (where horizontal corresponds to the XZ-plane in FIG. **1**). For example, a condenser may be provided with tubes at an angle of about 45 degrees to about 90 degrees, relative to horizontal. In other embodiments, a two-phase evaporative cooling system may have a condenser that is slanted to reduce the height of the system while increasing the frontal condenser area. This exemplary design may also induce liquid flow to the evaporator during time of need, for example, during high acceleration situations and/or during hill-climbing. Each tube **160** may be a hollow tube, pipe, duct and/or any other conduit suitable for transferring vapor and/or liquid through a hollow, internal portion of the conduit. For example, a condenser may include one or more substantially parallel conduits, where each conduit has at least one outside wall defining a closed, cross-sectional shape including a circle, an ellipse, a square, a rectangle, and/or any other suitable cross-sectional shape configured to contain and/or transfer vapor and liquid within the conduit.

[0093] FIG. **1** illustrates that a first end of teach tube **160** may begin at a common manifold **150**, with a hollow, internal portion that runs the length of the manifold **150**, so that the manifold **150** disperses vapor flowing from the evaporator **110** through all or most of the tubes **160** of the condenser **120**. A manifold **150** may be utilized to maximize vapor distribution across all of the tubes **160** to increase the surface area available for condensing vapor within the condenser **120**. FIG. **1** shows each tube **160** terminating with a second end that is physically attached to a bend **170** that connects each tube **160** to an adjacent tube **160**. Thus, the condenser **120** pictured in FIG. **1** has multiple tubes **160**, where each tube **160** is in fluid communication with a manifold **150**, and pairs of tubes are joined and in fluid communication at a high point (e.g. along the y-axis) by a bend **170**.

[0094] FIG. **1** illustrates a condenser **120** that utilizes bends **170** to connect neighboring tubes **160** to one another. Alternatively, the tubes **160** of a condenser may terminate at a second common manifold. Or each tube **160** may terminate with a plug or cap. In other cases, the bends **170** may be solid bends (versus hollow) to provide structural support only, and which do not allow fluid communication between neighboring tubes **160**. In still other embodiments, each tube **160** may terminate with a capping block that is common to all of the tubes **160**.

[0095] FIG. **1** illustrates that each tube **160** may be positioned with a space or gap between itself and its neighboring tubes **160**. In some cases, spaces (or gaps) may be provided between neighboring tubes **160** of a condenser **120** to enable a cooling fluid to flow between the tubes **160**; e.g. natural or forced convective flow. In same examples, the cooling fluid used may be air and/or any other suitable gas. Spaces between the tubes **160** may increase the surface area available for heat-transfer and/or help maximize the fluid velocity, which in turn, may increase the heat-transfer rate achievable by the condenser, resulting in more efficient heat-transfer, condensation, and allow for smaller condenser sizes. In addition, the outside surface area available by a condenser **120** for contact with a cooling fluid may be increased by the use of fins **180**. Gaps are provided between neighboring fins **180**, so as to provide a flow path for a cooling fluid to pass over and between the fins.

[0096] In still other examples, a condenser for a cooling system may include any suitable heat-exchanger configured to be cooled by an external cooling fluid flowing over one or more outside surfaces of the condenser. For example, condenser types include brazed folded-fin, finned-tube, flat plate, shell and tube, dimpled, and/or any other suitable heat exchanger. However, all of these condenser examples are provided for illustrative purposes and the specific design of a condenser will be determined by the specific application and environment.

[0097] FIG. **1** illustrates two vapor/condensate lines **140** physically connecting the evaporator **110** and the condenser **120**. Again, this is for illustrative purposes, for example when the condenser **120** and the evaporator **110** are provided as separate units, which are installed in different locations within a vehicle. In this case, one or more vapor/condensate lines **140** may be used to provide a fluid path for the two-phase heat-transfer fluid between the condenser **120** and the evaporator **110**. A vapor/condensate line **140** may be a cylindrical pipe or tube and/or any other conduit suitable for transferring vapor and/or condensate between the condenser **120** and the evaporator **110**. To minimize pressure drops and/or liquid entrainment by the vapor produced in the evaporator **110**, a plurality of vapor/condensate lines **140** may be utilized to increase the total open surface area available for vapor and/or condensate flow between the condenser **120** and evaporator **110** (e.g. to lower the fluid velocity through the vapor/condensate line **140**.) However, in other applications, a single vapor/condensate line **140** may connect the condenser **120** and the evaporator. For example a single vapor/condensate line **140** may have essentially the same length (z-axis) and width (x-axis) as the condenser **120** and/or the evaporator **110**, and have a height (y-axis) necessary to link the condenser **120** to the evaporator **110**. FIG. **1** illustrates the vapor/condensate lines **140** as straight, vertical conduits. However, this is not intended to be limiting, and a vapor/condensate line **140** may include one or more bends or turns, as needed, for a specific vehicle's configuration and/or space limitations. FIG. **1** illustrates an example where the condenser **120** is positioned above (in the y-axis direction) the evaporator **130**. Other embodiments of cooling systems may reverse the position where the condenser **120** is positioned below the evaporator **110** or at substantially the same height. In some cases, a wicking material may be provided to assist with the transfer of condensate from the condenser **120** to a higher-positioned evaporator **110**.

[0098] The evaporators shown in FIGS. **2***a* and **2***b* may be visualized as cross-sectional views, in the XY-plane, of the evaporator **110** shown in FIG. **1**. FIG. **2***a* illustrates an embodiment of an evaporator **110**, which includes a housing **200** defining an internal channel, referred to herein as a main channel **210**. This exemplary evaporator **110** has a rectangular cross-section with two relatively long, horizontal walls (along the x-axis) and two shorter vertical walls (along the y-axis). Each wall has an internal surface facing the main channel **210** and an external surface facing the outside environment of the evaporator **110**. Thus, each wall has a thickness defined by its respective internal surface and external surface. For example, depending on the system's operating pressure, a wall thickness may be about 1 mm to about 10 mm. Both FIGS. **2***a* and **2***b* show a heat source **130** in thermal communication with a first wall **220**. The term "thermal communication" is used to indicate that the heat source **130** and the first wall **220** may exchange heat between each other and are not necessarily in direct physical contact with each other. For example, there may be an intervening layer positioned between the heat source **130** and the first wall **220**, for example a thermal grease, a thermal interface material, a bonding interface material, an adhesive, and/or any other suitable material to facilitate contacting the heat source **130** to the evaporator housing **200**. Examples of bonding interface materials that may be placed between the heat source **130** and the first wall **220** of the housing **200** of the evaporator **110** include thermoplastics, solder, brazing, and/or sintered silver. In other cases, there may be no intervening layer positioned between the heat source **130** and the first wall **220**, and the heat source **130** may be in direct contact and thermal communication with the first wall **220** of the evaporator **110**. Heat-transfer between the heat source **130** and the first wall **220** may occur by conductive and/or radiant heat-transfer.

[0099] A condenser **120** for a cooling system **100** may either be cooled using air forced convection cooling (i.e., fan), liquid cooling, or natural convection cooling. In a liquid cooling configuration, the heat absorbed by the two-phase heat-transfer fluid may be utilized to provide heating to the cabin of a vehicle during cold environmental conditions. This configuration may be more suited to electric vehicles where there is no combustion engine to provide the heating capacity. In some embodiments of a natural cooling configuration, the condenser fins **180** may be integrated into the transmission/transaxle body to make use of the larger surface areas of the transmission/transaxle.

[0100] FIGS. **2***a* and **2***b* illustrate the first wall **220** as being a longer, horizontal wall (relative to the vertical wall). However, this is for illustrative purposes only. The heat source **130** may alternatively be placed along a shorter, vertical wall. In addition, the housing **200** is not limited to rectangular cross-sectional shapes. Any other suitable cross-sectional shape may be utilized, depending on the application, and includes square, triangular, hexagonal, circular, elliptical, and/or any other suitable cross-sectional shape. FIGS. **2***a* and **2***b* also illustrate a single heat source **130** associated with an evaporator **110**. However, an evaporator **110** may be in thermal communication with and cool one or more heat sources **130**, and multiple heat sources **130** may be in thermal communication with more than wall of the evaporator **110**. For example, multiple heat sources **130** may be placed in thermal communication and/or physical contact with substantially all or most of the outer periphery of the housing **200** of an evaporator **110**.

[0101] Comparing FIGS. **2***a* and **2***b*, FIG. **2***b* illustrates a vapor/condensate line **140**, for connecting the evaporator **110** to a condenser (not shown). Thus, the example of FIG. **2***b* may include two additional walls (not shown), one at each end of the evaporator **110** along the z-axis to provide a fluid seal. FIG. **2***a* does not illustrate a vapor/condensate line **140**, although one could be included, as shown in FIG. **2***b*. FIG. **2***a* illustrates an evaporator **110** where the main channel **210** is the conduit for transferring vapor and condensate between the evaporator **110** and the condenser (not shown). This concept will be described in more detail when describing FIG. **8** below. FIG. **2***b* also illustrates that the inside surface of the first wall **220** may include one or more surface area extenders **230**. Surface area extenders **230** increase the surface area available for heat-transfer from the heat source **130**, through the first wall **200**, to the two-phase heat-transfer fluid (not shown) contained within the evaporator **110**. Increasing the surface area of the internal surface of the first wall **220** may increase the heat-transfer rate achievable by the evaporator **110**, and thus, reduce the evaporator's size and/or cost. The surface area extenders **230** of FIG. **2***b* are shown as relatively short, rectangular blocks of material positioned substantially parallel to the long-axis (z-axis) of the evaporator **110**. In addition, the surface area extenders **230** of FIG. **2***b* are shown as having heights (y-axis) and widths (x-axis) that are relatively equal. However, these characteristics are not intended to be limiting. For example, the height to width ratio of a surface area extender **230** may be about 0.1 to about 10. In other examples, the height to width ratio of a surface area extender **230** may be about 0.5 to about 5. In other examples, a surface area extender **230** may take a different shape. For example, a surface area extender **230** may have a rounded surface at its apex (see FIG. **3***a*) versus the sharp-cornered, squared apex as shown in FIG. **2***b*. Or a surface area extender **230** may be positioned parallel to a different axis, for example, parallel to the x-axis (the shorter axis) instead of the y-axis (the longer axis) as shown in FIG. **2***b*. In yet other cases, a plurality of surface area extenders may be provided that are aligned along two or more axes such that the surface area extenders intersect each other; e.g. in a cross-hatch pattern. In still other cases, a surface area extender **230** may be substantially the same dimension in all or most directions in the XZ-plane. For example, instead of being longer in the z-axis direction and relatively short in the x-axis direction (e.g. rectangular strips) as shown in FIG. **2***b*, a surface area extender **230** may be in a shape that is substantially equal in length in the x-axis direction, the z-axis direction, and all or most other directions in the XZ-plane. Examples of such an embodiment include a surface area extender **230** in the shape of a bump, wedge, pyramid, knob, cylinder, cone, and/or any other suitable 3-dimensional shape extending from the internal surface of the first wall **220** into the main channel **210** of the housing **200** of the evaporator **110**.

[0102] FIGS. **2***a* and **2***b* illustrate a housing **200** constructed of a single piece of material. This is for illustrative purposes only, and in some cases, a housing **200** may be constructed from more than one piece of material that are joined together such that the connecting points of the various pieces make a fluid seal; e.g. welded, soldered, glued, etc. This concept will be discussed in more detail when describing FIG. **5**. However, FIGS. **2***a* and **2***b* illustrate an evaporator **110** where the housing **200** is constructed from a single piece of material made, for example, by an extrusion process. The evaporator **110** may also be fabricated through a machining or casting process or by using 3D printing technology. Examples of suitable materials for the housing **200** include aluminum, aluminum alloys, and/or copper.

[0103] FIG. **3***a* illustrates another embodiment of an evaporator **110** for a cooling system. A heat source **130** is in thermal communication with the external surface of a first wall **220** and the first wall **220** has an internal surface facing a main channel **210** of an evaporator **110**. In addition, as in previous examples, a plurality of surface area extenders **230**, in the form of rounded projections extend from the internal surface of the first wall **220** into the main channel **210**. However, in this example, the first wall **220** does not take the shortest path, in the x-axis direction, to intersect with the vertical walls (in the y-axis direction) of the housing **200**. Instead, the first wall **220** terminates at a first edge **340** and a second edge **345** before intersecting/contacting the vertical side walls of the housing **200**. Both the first edge **340** and the second edge **345** are positioned substantially parallel to the long-axis (z-axis) of the housing **200** of the evaporator **110**. Thus, FIG. **3***a* illustrates a first wall **220** that is not in direct physical contact with the vertical side walls of the housing **200**. Instead, the first wall **220** is joined to the housing **200** indirectly, first by a downward and outward extending second wall **320** that begins at, and is joined to, the first edge **340** of the first wall **220**, and a first tab **330** that extends horizontally from the second wall **320** to intersect with the housing **200**. Thus, the second wall **320** and the first tab **330** complete one connection from the first wall **220** to the housing **200** (in this example on the left side of FIG. **3***a*). A complementary set of elements is also provided for connecting the first wall **220** to the housing **200** to provide an evaporator **110** this is fluid-sealed from the outside environment. Thus, the first wall **220** is also connected to the housing **200** by a third wall **325** that begins at, and is joined to, the second edge **345** of the first wall **220**, and a second tab **335** that begins at, and is joined to, the third wall **325** that extends horizontally from the third wall **325** to intersect with the opposite side of the housing **200** (in this case on the right side of FIG. **3***a*).

[0104] Referring again to FIG. **3***a*, the second wall and the third wall (**320** and **325**), the pair of tabs (**330** and **335**) and the housing **200** together form a pair of side channels (**300** and **305**) that are positioned on either side (either edge **340** and **345**) of the first wall **220**. The side channels (**300** and **305**) also extend below the main channel **210** and the internal surface of the first wall **220**. Thus, the side channels (**300** and **305**) and the second and third walls (**320** and **325**) extend the surface area available for heat-transfer from the heat source **130** to the two-phase heat-transfer fluid (not shown) contained within the evaporator **110**. The evaporator **110** embodiments illustrated in FIGS. **3***a* and **3***b* provide additional pathways for heat removal from the heat source **130** to the two-phase heat-transfer fluid (not shown) contained within both the main channel **210** and the side channels (**300** and **305**). In other words, in addition to transferring heat from the heat source **130** through the first wall **220** to the two-phase heat-transfer fluid (not shown), heat may also be transferred from the heat source **130**, through the second and third walls (**320** and **325**) to the two-phase heat-transfer fluid contained in both of the side channels (**300** and **305**). A "main channel" may also be referred to herein as a "first channel". A "first side channel" may also be referred to herein as a "second channel". A "second side channel" may also be referred to herein as a "third channel".

[0105] Referring again to FIG. **3***a*, the first wall **220**, the second wall **320**, and the third wall **325** also define the boundaries of a passage **310** positioned on the exterior of the housing **200** of the evaporator **110**. This passage **310** provides a space in which one or more heat sources **130** may be positioned. In the example of FIG. **3***a*, the width (in the x-axis direction) of the heat source **130** is about the same, or slightly smaller, than the width of the passage **310**. By setting these widths to about the same amount insures that the lateral edges of the heat source **130** (left and right edges of FIG. **3***a*) are in thermal communication, and possibly in direct physical contact, with the second wall **320** and third wall **325** of the evaporator **110**. Gaps (e.g. air gaps) between the lateral edges of the heat source **130** and the second wall **320** and/or third wall **325** may increase the thermal resistances between these elements and, as a consequence, reduce the potential heat-transfer benefits of the side channels (**300** and **305**) and the second wall **320** and third wall **325**. Thus, in some cases, the lateral edges of the heat source **130** may be in direct physical contact (and thermal communication) with the second wall **320** and the third wall **325**. In other cases, the lateral edges of the heat source **130** may be in thermal communication, where conductive heat-transfer is increased through the use of a material positioned between the lateral edges of the heat source **130** and the second wall **320** and third wall **325**. For example, there may be an intervening layer positioned between the lateral edges of the heat source **130** and the second wall **320** and third wall **325**, for example a thermal grease, a thermal interface material, a bonding material, an adhesive, and/or any other suitable material to help facilitate contacting the heat source **130** to the second wall **320** and third wall **325**. Other examples of materials that may be placed between a heat source **130** and the second wall **320** and third wall **325** of an evaporator **110** include thermoplastics, solder, and/or sintered silver. In still other examples, a small gap may be provided between the second wall **320** and the lateral edge of a heat source **130** and/or between the third wall **325** and the lateral edge of a heat source **130** to allow for thermal expansion/contraction of the cooling system **100** components.

[0106] FIG. **3***b* illustrates another exemplary embodiment of an evaporator **110**. In this example, the evaporator **110** has a housing **200** with a substantially cylindrical shape and substantially circular cross-section and two passages (**310***a* and **310***b*) for placing a plurality of heat sources (**130***a* and **130***b*). A vapor-condensate line **140** is also provided so that vapor and condensate may flow between the evaporator **110** and the condenser (not shown). The two passages (**310***a* and **310***b*) are aligned substantially parallel to one another and aligned with each other along the long axis (z-axis) of the cylindrical housing **200**. Both passages (**310***a* and **310***b*) are configured to contain at least one heat source (**130***a* and **130***b*). Heat source **130***a* is in thermal communication with a first wall **220***a*, a second wall **320***a*, and a third wall **325***a* such that heat is removed from heat source **130***a* through at least one of the walls, and transferred to the two-phase heat-transfer fluid contained within the evaporator **110**. Similarly, heat source **130***b* is in thermal communication with a first wall **220***b*, a second wall **320***b*, and a third wall **325***b* such that heat is removed from heat source **130***b* through at least one of the three walls, and transferred to the two-phase heat-transfer fluid contained within the evaporator **110**. Further, the housing **200**, first tab **330***a*, and second wall **320***a* form the boundaries of first side channel **300***a*. In addition, the housing **200**, first tab **330***b*, and second wall **320***b* form the boundaries of first side channel **300***b*. In addition, both heat sources (**130***a* and **130***b*) may transfer heat through their respective third side walls (**325***a* and **325***b*) to a shared side channel **350**, whose boundaries are defined by the third side walls (**325***a* and **325***b*) and connecting tab **347**.

[0107] Thus, heat-transfer in the evaporator **110** of FIG. **3***b* may occur from the heat sources (**130***a* and **130***b*) through their respective first walls (**220***a* and **220***b*) to the two-phase heat-transfer fluid contained in the main channel **210**, and also through each heat source's respective second wall (**320***a* and **320***b*) and third wall (**325***a* and **325***b*) to the two-phase heat-transfer fluid contained within their corresponding first side channel (**300***a* and **300***b*) and the shared side channel **350**. FIG. **3***b* also illustrates a plurality of surface area extenders **230** projecting vertically from the first walls (**220***a* and **220***b*) and projecting horizontally from the second walls (**320***a* and **320***b*) and the third walls (**325***a* and **325***b*) into the side channels (**300***a*, **300***b*, and 350).

[0108] Referring again to FIG. **3***b*, the lower portion (in the y-axis direction) of the housing begins (starting at the left) with a first tab **330***a* horizontally connected to the housing **200** and a second wall **320***a*. The second wall **320***a* extends vertically and connects with a first edge of a horizontally oriented first wall **220***a*. The first wall **220***a* extends horizontally to terminate at a second edge from which a third wall **325***a* extends downward and vertically to terminate at a horizontally oriented connecting tab **347**. The connecting tab **347** connects the third wall **325***a* to the third wall **325***b*. Third wall **325***b* extends vertically and connects at a second edge of first wall **220***b*, which extends horizontally to terminate at a second edge. Second wall **320***b* then extends vertically and downward from the second edge to connect with first tab **330***b*, which extends in the horizontal direction to connect with the housing **200**. All of these elements are shown in FIG. **3***b* as being constructed from a single piece of material, including the housing **200**. This may be achieved by forming the evaporator **110** using an extrusion process utilizing materials such as copper, aluminum, and/or aluminum alloys. Alternatively two or more of the elements shown in FIG. **3***b* may be manufactured as individual pieces, which are subsequently joined together; e.g. welded, soldered, glued, etc.

[0109] FIGS. **3***a* and **3***b* illustrate embodiments that have one and two passages **310** respectively. Other embodiments may have more than two substantially parallel passages, for example 3, 4, 5, 6, 7, 8, 9, or 10 passages, to provide additional space for a greater number of heat sources, or to provide space for physically larger heat sources. Further, the embodiment of FIG. **3***b* may include caps or plugs positioned at the ends of the evaporator **110** (in the z-axis direction) so as to seal the evaporator **110** from the external environment.

[0110] FIG. **4** illustrates another element of a cooling system to further increase the heat-transfer capabilities of a cooling system: conducting elements (**400** and **410**). As in previous figures, the exemplary embodiment of FIG. **4** shows an evaporator **110** with a housing **200** with external walls that define a main channel **210**, a first side channel **300**, and a second side channel **305**. The first side channel **300** is defined by the housing **200**, a first tab **330**, and a second wall **320**. The second side channel **305** is defined by the housing **200**, a second tab **335**, and a third wall **325**. A first wall **220** is positioned between the second wall **320** and the third wall **325** at connecting edges **340** and **345**, respectively. Thus, heat generated by the heat source **130** may be transferred through three separate contact surfaces to the two-phase heat-transfer fluid: the first wall **220**, the second wall **320**, and the third wall **325**. However, as in previous examples, the heat source **130** of FIG. **4** has a relatively large width in the x-axis direction and relatively short thickness in the y-axis direction. Thus, more surface area (per unit length in the z-axis direction) is available for heat-transfer through the first wall **220** of the evaporator **110**, and significantly less surface area (per unit length in the z-axis direction) is available through the remaining two pathways: through the second wall **320** and the third wall **325**.

[0111] However, placement of one or more conducting elements, two in this case (a first conducting element **400** and a second conducting element **410**) takes advantage of a fourth remaining surface of the heat source **130**, the downward facing surface of the heat source **130** (in the XZ-plane). In the example of FIG. **4**, the first conducting element **400** and the second conducting element **410** are positioned within the passage **310** such that the heat source **130** is positioned between the external surface of the first wall **220** and the conducting elements (**400** and **410**). The first conducting element **400** and the second conducting element **410** may be in direct physical contact with the lower surface of the heat source **130**. Alternatively, the first conducting element **400** and the second conducting element **410** may be in thermal communication with the heat source **130**, where another material is placed between the conducting elements (**400** and **410**) and the lower surface of the heat source **130**. For example, a thermal grease, a thermal interface material, a bonding interface material, an adhesive, and/or any other suitable material to facilitate contacting the heat source **130** to the first conducting element **400** and the second conducting element **410** may be used. Examples of bonded interface materials that may be placed between the heat source **130** and the conducting elements (**400** and **410**) include thermoplastics, solder, and/or sintered silver.

[0112] The first conducting element **400** and the second conducting element **410** of FIG. **4** are illustrated as opposing triangular wedges. However, alternative embodiments could utilize one or more conducting wedges positioned within the passage **310**. For the case of a single conducting element, the conducting element could be sized to substantially fill the passage **310**. Generally, it is desirable to maximize the amount of surface area of the heat source **130** that is in thermal communication and/or in direct physical contact with the adjacent surface(s) of the conducting element(s) (**400** and **410**). Therefore, in some embodiments, the exposed surface area (in the XZ-plane) of the heat source **130** and the adjacent surface area (in the XZ-plane) of a conducting element may be substantially equal. In addition, it is generally desirable to maximize the amount of surface area (in the YZ-plane) of the external surfaces of the second wall **320** and the third wall **325** that is available for heat transfer from the heat source **130**. Thus, triangularly-shaped conducting elements (**400** and **410**) achieve both of these criteria, while also reducing the amount of material used to construct the conducting elements (**400** and **410**). Such a triangular configuration may be achieved using two separate, opposing conducting elements (**400** and **410**) as shown, or alternatively, the same triangular shape may be achieved using a single piece of material. Use of more than one conducting element may provide easier placement of the conducting elements within the passage **310** by providing some "room" for movement of the conducting elements (**400** and **410**) relative to each other and within the passage **310**. Alternatively, conducting elements (**400** and **410**) may be placed in close contact with the outside surfaces that define the passage **310** and the heat source **130**. A space or gap may be left between the neighboring conducting elements (**400** and **410**). Such a gap may be provided to account for differences in thermal expansion and/or to allow for electrical connections from the heat source **130** to a control module (not shown). Conducting elements may be constructed from any suitable material with higher thermal conductivities, including but not limited to copper and/or aluminum. Thermally conductive ceramics such as aluminum nitride and/or silicon nitride may also be used as the conductive elements (**400** and/or **410**).

[0113] As shown in FIG. **4**, one or more conducting elements (e.g. **400** and **410**) may be positioned along the x-axis within the passage **310** to increase heat-transfer from the one or more heat sources (e.g. **130**) positioned within the passage **310**. Alternatively, or in addition to, one or more conducting elements may also be positioned within the passage **310** along the long axis of the housing **200** (z-axis direction), with or without out spaces between neighboring heat sources along the long axis of the housing **200**.

[0114] FIG. **5** illustrates that an evaporator **110** may include one or more main channels positioned substantially parallel to one another; e.g. where each long axis of each main channel are parallel (relative to the z-axis). In addition, FIG. **5** illustrates that an evaporator **110** may be constructed from individual pieces that are later connected together; e.g. welded, soldered, glued, etc. For example, as shown in FIG. **5**, a housing **200** may be constructed as a first piece that defines some of the boundaries of two separate main channels, **210***a* and **210***b*, where the two main channels share a spanning wall **500** and are divided vertically from each other by a dividing wall **520**. Each main channel (**210***a* and **210***b*) has a lower boundary defined by an insert, **530***a* and **530***b*, respectively. Each insert (**530***a* and **530***b* respectively) includes a first tab (**330***a* and **330***b* respectively) in physical contact with a side wall (**510** and **515**) of the housing **200**. Both first tabs (**330***a* and **330***b*) extend horizontally from the respective side walls (**510** and **515**) and terminate at a second wall (**320***a* and **320***b*) which extend upwards into its respective main channel (**210***a* and **210***b*) to terminate at a first edge of a first wall (**220***a* and **220***b*). Each first wall (**220***a* and **220***b*) extends horizontally to end at a second edge and connects with a third wall (**325***a* and **325***b*), which extends vertically downward to contact a second tab (**335***a* and **335***b*), where each second tabs (**335***a* and **335***b*) extends horizontally to contact the dividing wall **520**. As a result, each insert (**530***a* and **530***b*) provides three walls, the first wall (**220***a* and **220***b*), the second wall (**320***a* and **320***b*), and the third wall (**325***a* and **325***b*), that provide surfaces for contacting with one or more heat sources (not shown). In addition, each insert provides surfaces that, together with the housing **200**, define the side channels of the evaporator **110**; e.g. a first side channel (**300***a* and **300***b*) and a second side channel (**305***a* and **305***b*).

[0115] Referring again to FIG. **5**, the inserts (**530***a* and **530***b*) may be provided as separate components to the housing **200** if, for example, the material used for the inserts is significantly more expensive than the material used for the housing. For example, the housing **200** may be fabricated using aluminum, while the inserts (**530***a* and **530***b*) may be fabricated using copper. FIG. **5** shows the concept of using two inserts (**530***a* and **530***b*) in conjunction with a housing **200** to yield two parallel main channels (**210***a* and **210***b*) within a single evaporator **110**. However, a single insert or more than one insert could alternatively be used, together with one or more housings to produce an evaporator with one more main channels. Regardless of the complexity of the evaporator and/or number of component pieces used, the inserts are attached to the housing to create a fluid seal. Methods for attaching include welding, soldering, and/or gluing.

[0116] Further, one or all of the internal surfaces of an evaporator may have one or more coatings and/or surface treatments to further enhance heat-transfer. Coatings that may be added to the boiling/condensing surfaces (wetted area) of an evaporator that may increase phase change heat-transfer include microporous and nanoporous coatings. These coatings may be sintered, brazed or epoxied to surfaces and may be constructed from copper and aluminum nano- or micron-sized materials.

[0117] FIG. **6** illustrates a perspective view of the lower portion of an evaporator **110** similar to that shown in FIG. **5**, with inserts and a housing **200** constructed from a single piece of material (e.g. copper or aluminum). FIG. **6** illustrates two substantially parallel passages, **310***a* and **310***b*, aligned along the z-axis direction. There are three heat sources **130** positioned within each passage, each separated from its neighbors by gaps.

[0118] The features and elements of some of the embodiments described herein, provide a number of advantages and benefits, some of which are described in more detail below

1. Dual-Sided Cooling of the Heat Source (e.g. One or More Power Modules) Utilizing an Indirect Two-Phase Cooling Scheme for Increased Heat Dissipation.

[0119] The cooling systems described herein enable heat to be removed from multiple sides of the power modules to increase heat dissipation and to allow for smaller cooling systems. The heat on the top-side of the modules may be transferred to the evaporator through direct contact with the evaporator and the conducting elements also provide a heat conduction path through the backside of the power modules to the evaporator.

2. Increased Heat Spreading within the Evaporator for Increased Heat-Transfer.

[0120] The cooling systems described herein combine the use of a two-phase heat-transfer with increased surface due to the incorporation of side channels positioned within the evaporator on both lateral sides of the power modules. These side channels allow the two-phase heat-transfer fluid (e.g. refrigerant) to evaporate/boil on the lateral sides of the power modules, which promotes heat spreading through the evaporator housing and increases the wetted surface area in contact with the two-phase heat-transfer fluid. Therefore heat is removed, within the evaporator, via evaporation/boiling through the top and lateral sides of the power modules (and bottom sides due to the use of conducting elements), which significantly increases heat-transfer rates, allowing for a more compact cooling system.

3. Passive (e.g. No Pumping) Two-Phase Scheme Allows for Increased Efficiency and Reliability

[0121] The two-phase cooling systems may dissipate large amounts of heat without the need for a pump or compressor (e.g. passive). The two-phase heat-transfer fluid within the system may be transported via gravity, vapor pressure and capillary wicking through an enhanced surface (e.g., microporous coating). Current vehicle cooling systems require a pump to circulate fluid between and/or through their heat exchangers to provide cooling of the heat sources (e.g. power electronics). Alternatively, a passive two-phase cooling system may provide better cooling capacity without the need for a mechanical device to circulate the heat-transfer fluid. Eliminating a pump from a cooling system may result in increased efficiency (e.g. reduced power consumption by the system), increased reliability (e.g. no moving parts), and reduced manufacturing and operating costs.

4. Extrudeable Evaporator Design for Low-Cost Manufacturing

[0122] The one-dimensional features of the evaporator and the use of appropriate metals (e.g. aluminum) may allow for the evaporators described herein to be fabricated using an extrusion process. Extruding is a cost-effective method for fabricating components.

5. Limited Inclination Effects on Performance

[0123] The compact designs and configurations of the cooling systems described herein, combined with the placement of an evaporator below the condenser, may minimize the effect of vehicle inclination on cooling performance. This is especially true for inclination/rotation along the z-axis (see FIG. 3*b*).

6. Condenser and Evaporator May be Constructed as One Unit or Split to Allow for Flexibility.

[0124] Some of the cooling systems described herein may be provided with the evaporator and condenser as a single, combined unit, for example, with the evaporator may be the lower manifold on the condenser. Alternatively, a cooling system may also be configured so that the condenser and evaporator are separate and are connected through tubing/piping. This flexibility (e.g. combined or separate) provides options on the placement of the condenser to suit different applications.

7. Increased Heat-Transfer Via the Use of Finned Surfaces and Enhanced Surface Coatings.

[0125] An evaporator may incorporate surface area extenders (e.g. finned structures) and boiling enhancement coatings strategically placed within the evaporator to further improve heat-transfer and reduce the size of the cooling system. Other embodiments may comprise boiling enhancement coatings and/or surface coating techniques that have been proven to enhance evaporation/boiling heat transfer coefficients and critical heat flux values by as much as 430% and 120%, respectively.

[0126] The power electronics and electric motor (PEEM) in most electric-drive vehicles are often cooled using a dedicated, low-temperature water-ethylene glycol (WEG) cooling loop. Therefore, in hybrid-electric vehicles, there are typically two WEG-based coolant loops—a low-temperature (65° C.) loop to cool the PEEM and a high-temperature (105° C.) loop to cool the internal combustion engine (ICE). These systems tend to be relatively complex, inefficient, and prone to operation issues. Thus, a two-phase cooling system that cools both the power electronics and the electric motor (PEEM) of an electric-drive vehicle would be advantageous. Thus, some of the condenser/evaporator embodiments described herein may be well suited to replace the low-temperature pumped coolant system currently used in electric-drive vehicles with a smaller, lighter, more cost-effective passive (no pump or compressor), two-phase-based cooling system. Some of the cooling systems described herein may increase power densities and lower component operating temperatures by utilizing higher heat-transfer rates of two-phase cooling. For power electronics, increased power densities may result in smaller semiconductor sizes and numbers. For the motor, lower operating temperatures may decrease or eliminate expensive, rare-earth elements used in the motor's permanent magnets. A reduction in the semiconductor size and number and a reduction in rare-earth motor materials may reduce the cost of the electric traction-drive system in electric-drive vehicles. Additionally, an efficient thermal management strategy may be an effective means of improving performance and reliability.

[0127] FIG. **7** illustrates a cooling system **100** that utilizes an evaporator **110** and a condenser **120** similar to those described previously for FIGS. **1-6**. A cooling fluid **705** is shown for the transfer of heat from the condensing two-phase heat-transfer fluid (not shown) to the external environment. For example, a cooling fluid **705** may be a relatively low temperature air flow passing over an external surface of the condenser **120**. In this example, the evaporator **110** and the condenser **120** are illustrated as separate units fluidly connected by a vapor/condensate line **140**. The evaporator **110**, condenser **120**, and vapor/condensate line **140** contain the two-phase heat-transfer fluid (not shown) for transferring heat from a power electronics module **700**, which has at least one surface in thermal communication with an external surface of the evaporator **110**.

[0128] In addition, FIG. **7** illustrates a cooling system **100** that cools a motor **730**, in addition to the power electronics module **700**. The motor **730** is cooled using a heat-transfer liquid (not shown), where the heat-transfer liquid is circulated through a loop by a pump **780**. Examples of heat-transfer liquids that may be used to cool the motor **730** include oils such as mineral oils, and synthetic materials. Examples of oils typically used to cool the electric motor include lubricating transmission oil. The suction side of the pump **780** is connected to a liquid reservoir **760** by a first supply line **770**. The discharge side of the pump **790** circulates the heat-transfer liquid through a first heat exchanger **710** that is in thermal communication with the two-phase heat-transfer fluid contained in the evaporator **110**. Thus, heat may be transferred from the heat-transfer liquid (circulated by the pump **780**) to the two-phase heat-transfer fluid (contained in the evaporator **110**) such that the two-phase heat-transfer fluid vaporizes and travels through the vapor/condensate line **140** to condense in the condenser **120**. Cooled heat-transfer liquid then exits the first heat exchanger **710** and circulates through a first return line **720** to a second heat exchanger **740** that is configured to be in thermal communication with the motor **730**. The motor **730** is at a higher temperature than the heat-transfer liquid flowing through the second heat exchanger **740** and, therefore, heat is transferred from the motor **730** to the heat-transfer liquid by the second heat exchanger **740**. The heated heat-transfer liquid is then delivered back to the liquid reservoir **760** through a second return line **750** where the process may be repeated.

[0129] The first heat exchanger **710** may be a tube, pipe, and/or coil with an inlet that passes through a sidewall of the evaporator **110** to a portion of the first heat exchanger **710** that is within an inside volume of the evaporator **110** and at least partially submerged in the two-phase heat-transfer fluid. Thus, the heat-transfer fluid (not shown) may pass through an inlet into the submerged portion of a tube, pipe, and/or coil to transfer heat to the two-phase heat-exchange fluid, and then exit from the evaporator **110** through an exit to the first return line **720**. In some examples, the first heat exchanger **710** may have fins to increase the surface are available for heat exchange. In still other embodiments, at least a portion of the evaporator **110** may be positioned directly in the liquid reservoir **760** such that at least a portion of the outside surface of the evaporator **110** may be submerged and in direct physical contact with the heat-transfer liquid present within the liquid reservoir **760**. Alternatively, a heat-transfer liquid may be circulated through a first heat exchanger **710** that shares a surface with the evaporator **110** such that heat may be transferred from the heat-transfer liquid to the two-phase heat-transfer fluid contained in the evaporator **110**. In some case, the pump **780** may be positioned within the liquid reservoir **760** and may be partially or completely submerged in the heat-transfer liquid. Examples of types of pumps that may be used include positive-displacement pumps such as gear pumps, piston pumps, and/or any other suitable pump.

[0130] The second heat exchanger **740** may be a closed system, for example, a heat exchanger that is built to be an integral part of the motor **730**, such as a volume or space in thermal communication with at least one surface of the motor **730**. For example, the heat-transfer liquid may be passed through a volume or space created by an external housing positioned around at least one outside surface of the motor **730**, where the housing has an inlet and an outlet for directing the heat-transfer liquid into the housing, and across at least one outside surface of the motor **730** to receive heat from the motor. The heated heat-transfer liquid may then be directed out of the volume or space through an exit to the second return line **750** leading to the liquid reservoir **760**. Alternatively, the second heat exchanger **740** may be, or include, an open system. For example, an open system design for a second heat exchanger **710** may include a first return line **720** that terminates with at least one spray nozzle, jet, and/or opening, which may direct the heat-transfer liquid onto at least one outside surface of the motor **730**. Thus, the heat-transfer liquid may exit the at least one spray nozzle, jet, and/or opening as a spray at a relatively high exit velocity and/or at a high pressure drop. Alternatively, the heat-transfer liquid may exit the at least one spray nozzle, jet, and/or opening at a relatively low velocity and low pressure drop, for example as a liquid that essentially falls by the influence of gravity to impinge on an outside surface of the motor **730**.

[0131] The first supply line **770**, second supply line **790**, first return line **720**, and the second return line **750** may be constructed from one or more pipes, tubes, ducts, channels, and/or any other suitable conduit for transferring liquid from one point to another. These conduits may be constructed from copper, aluminum, stainless steel, and/or any other suitable material of construction.

[0132] A liquid reservoir **760** may be a closed container, such as a tank. In such a case, the second return line **750** may be connected to the liquid reservoir **760** by a fitting. Similarly, the first supply line **770** may be connected to the liquid reservoir **760** by a fitting. Heat transfer liquid may then enter and exit the tank through these nozzles. Alternatively, a liquid reservoir **760** may be an open-toped tank, such as an oil pan. A liquid reservoir **760** in the form of an oil pan may be well suited for open-system embodiments of the second heat exchanger **740**, where for example, the heat-transfer liquid is sprayed onto at least one outside surface of a motor **730**.

[0133] In summary, FIG. **7** illustrates a cooling system **100** where heat is removed from a motor **730**, transferred by a heat-transfer liquid to a two-phase heat-transfer fluid, which also removes heat from a power electronics module **700**. Subsequently, the total heat removed from the motor **730** and from the power electronics module **700** may be transferred from the cooling system **100** to the environment by heat-transfer from the condensing two-phase heat-transfer liquid in the condenser **120** to the external environment (e.g. to cooling fluid **705**).

[0134] FIG. **8** illustrates one embodiment of the general cooling system illustrated in FIG. **7**. In the example of FIG. **8**, most of the elements of the cooling system are positioned within a containment system **805**, which provides a volume that is separated from the outside environment **802**. So, a motor **730**, an oil collection pan **870**, a pump **780**, an evaporator **110**, a first heat exchanger **710**, a second heat exchanger (e.g. spray nozzle **890**), and a power electronics module **700** are positioned within the containment system **805**. The power electronics module **700** is positioned to be in thermal communication with at least one outside surface of the evaporator **110** to enable the removal of the heat generated by the power electronics module **700**. In addition, the power electronics module **700** is positioned between the evaporator **110** and at least one conducting mass **400**, which has a set of first surface area extenders **830** to promote the transfer of at least some heat generated by the power electronics module **700** to the internal volume of the containment system **805**. The evaporator **110** has at least one main channel **210** positioned vertically and in fluid communication with a condenser **120** and an evaporator reservoir **807**. A refrigerant **800** (a two-phase heat-transfer fluid) is contained within the evaporator **110**. A portion of the main channel **210** extends out of the containment system **805** into the outside environment, and terminates in fluid communication with a condenser **120**. At least one outside surface of the condenser **120** may be contacted by a cooling fluid **705** (e.g. a relatively cool flow of air), causing the refrigerant vapor **810** to condense within the condenser **120** to produce refrigerant condensate **820**, which is then transferred by gravity back to the evaporator **110**. A first wall **220** of the evaporator **110**, corresponding to a portion of the evaporator **110** that is in thermal communication with the power electronics module **700**, may have a second set of surface area extenders **840** that project into the liquid volume of the refrigerant **800** (e.g. two-phase heat-transfer fluid). This second set of surface area extenders **840** may increase the surface area available for transferring heat from the power electronics module **700** to the refrigerant **800**.

[0135] The cooling system **100** of FIG. **8** also illustrates a recirculating oil system for removing the heat generated by a motor **730**. The recirculating oil system includes an oil collection pan **870** configured to receive heated oil **860** that has been applied to the outside surfaces of the motor by a spray nozzle **890**. The spray nozzle **890** applies an oil spray **850** onto at least one outside surface of the motor. Thus, the motor transfers heat to the oil, cooling the motor, and the heated oil **860** then falls by gravity into the oil collection pan **870**. The oil **845** in the collection pan **870** is then pulled into a first supply line **770** by the suction created by pump **780**. The first supply line **770** directs the oil through a first heat exchanger **710**. This exemplary first heat exchanger **710** is constructed from a length of pipe and/or tubing positioned within the evaporator reservoir **807**, and is at least partially submerged in the refrigerant **800**. Thus, the heat removed from the motor by the oil **845** is transferred to the refrigerant **800**, which vaporizes and is transferred to the condenser **120** and transferred to the cooling fluid **705** in the outside environment **802**.

[0136] After transferring heat from the oil **845** to the refrigerant **800**, the oil is transferred to the pump **780** by suction through the second supply line **790**. The pump **780** then discharges the oil into the third supply line **880**, which transfer the oil to the spray nozzle **890**, which distributes the oil spray **850** onto the motor **730**.

[0137] Thus, some embodiments of a two-phase cooling system may provide cooling of both the power electronics and the electric motor (PEEM) of an electric-drive vehicle, where the cooling system includes a condenser and an evaporator with a two-phase heat-transfer fluid contained therein and a recirculating heat-transfer liquid system. Such a system may provide the following benefits:

[0138] 1. Two-phase cooling may increase heat transfer and thus decrease the temperature of the heat-transfer liquid utilized to cool the motor. Lowering the operating temperature of the heat-transfer liquid may allow the motor to operate at lower temperatures which may enable reducing or eliminating rare-earth (expensive) elements used in permanent magnet motors. Eliminating rare-earth elements may result in significant cost reductions. Some of the proposed cooling concepts may allow for lower temperature heat-transfer oil to be used to cool the PEEM while still maintaining a more-elevated bulk temperature for the lubricating oil; higher heat-transfer oil temperatures are beneficial to the transmission/transaxle gears and bearing because of lower heat-transfer liquid viscosities).

[0139] 2. The combination of two-phase cooling and heat-transfer liquid jet-impingement cooling may increase heat dissipation and thus increase power densities. Higher power densities may reduce the mass of semiconductor material used and the quantity of semiconductor units, which may reduce the costs of such systems.

[0140] 3. In some embodiments, a cooling system may be combined with and cool high-temperature, wide-bandgap (WBG) power electronic modules. The increased efficiency and high-temperature capabilities of WBG-based power electronics makes them a promising technology for future electric-drive vehicles.

[0141] 4. In further embodiments, a cooling system may be applied to electric traction-drive systems where the power electronics are integrated into the motor. The trend in electric-drive vehicles is to place the power electronics closer to the motor since this offers advantages in terms of system packaging and performance.

[0142] 5. Some embodiments of a cooling system may eliminate the typical pumped, low-temperature WEG coolant loop and replace it with an efficient, passive (no pump or compressor) two-phase heat-transfer fluid cooling system.

[0143] 6. In still further embodiments, a cooling system may provide an option of using the waste heat from the APEEM, for cabin heating. This option would likely require a liquid-cooled condenser. This strategy may be more suited for all electric vehicles where there is no ICE to provide the heating capacity.

[0144] Thus, some embodiments of cooling systems include a two-phase heat-transfer fluid cooling system that cool the PEEM and is integrated within the vehicles transmission/transaxle. Further examples of the present invention are provided below.

EXAMPLES

Example 1

Electric Motor Cooling

[0145] Two examples that illustrate some of the features of cooling systems **100** for simultaneously cooling both power electronic modules **700** and a motor **730** are shown below in FIGS. **9***a* and **9***b*. The configuration shown in FIG. **9***a* cools a heat-transfer liquid, oil **845**, in an oil collection pan **870** using a natural convection heat exchanger, first heat exchanger **710**, configured to exchange heat with the evaporator **110**. The first heat exchanger **710** is illustrated as a horizontal extension of the evaporator **110**, substantially submerged in the oil **845** contained within the oil collection pan **870**, and includes fins for increasing the surface area in contact with the oil **845**. In this example, a pump **780** draws oil **845** from the oil collection pan **870** and impinges it onto the motor **730** in the form of an oil spray **850**. The example shown in FIG. **9***b* uses natural convective heat transfer in a first portion of the first heat-exchanger **710***a*, similar to the example shown FIG. **9***a*. However, the example of FIG. **9***b* also utilizes forced convection in second portion of the first heat-exchanger **710***b* to provide additional cooling of the oil **845** impinged onto the motor **730**. The forced convection portion of the first heat-exchanger **710***b* may be designed to more effectively reduce the temperature of the oil that is impinged onto the motor **730** (i.e., $T_{oil,\ motor} < T_{oil,\ bulk}$). The second portion of the first heat exchanger **710***b* includes an inlet for drawing oil **845** from the oil collection pan **870** into a section of tubing or piping positioned within the two-phase heat-transfer fluid contained in the evaporator **110**.

[0146] Decreasing the motor's operating temperature has the potential to decrease or eliminate expensive, rare-earth elements typically used in motors. Rare-earth elements are currently utilized in the motor magnets to increase the motor's operating temperature. The less-effective natural convection heat exchanger/evaporator allows the ATF at the reservoir to operate at higher temperatures (e.g., 70° C. –90° C.) which is beneficial to the gears and bearing due to lower ATF viscosities at higher temperatures.

[0147] FIGS. **9***a* and **9***b* illustrate most of the cooling system **100** components housed within a containment system **805**, with the exception of the condenser **120** and a portion of the main channel **210** of the evaporator **110**. These examples also show at least one power electronics module **700** positioned in thermal communication with a first wall **220** of the evaporator **110**. The power electronics module **700** is sandwiched between the first wall **220** of the evaporator **110** and a conducting element **400** having one or more surface area extenders **830**.

Example 2

Power Electronics Cooling

[0148] FIG. **10** illustrates further embodiments of a proposed cooling system **100** for cooling a power electronics module **700** and a motor **730**. The majority of the heat generated by the power electronics module **700** may be dissipated an evaporator **110** in conjunction with a condenser **120**, utilizing a two-phase heat-transfer fluid contained therein. A surface of the power electronics module **700** may be placed in thermal communication with a first wall **220** of the evaporator **110**. Vaporized two-phase heat transfer fluid may travel through the evaporator's main channel **210** to deliver the vapor to the condenser **120** where it condenses. This two-phase cooling approach may provide the sole means of heat dissipation from the power electronics module **700**, as is shown above in Example 1. However, additional cooling of the power electronics **700** may be provided by impinging oil **890***b* onto an outside surface of the power electronics module **700**. As shown in FIG. **10**, an outside surface of the power electronics module **700** may include a thermally conductive element **400** in thermal communication with a plurality of surface area extenders **830**. For example, a thermally conductive element **400** may include finned surfaces and/or enhanced surfaces to augment heat transfer. Impinging oil **890***b* directly onto the thermally conductive element **400** may improve heat transfer (through the use of finned and enhanced surfaces) and/or prevent or minimize contacting the power electronic module **700** with the oil **845**.

[0149] Impinging oil **890***b* onto an outer surface of a power electronics module **700** may be accomplished using two pumps—a first pump **780***a* for the motor and a second pump

780*b* for the power electronics module **700**. Such a configuration may provide additional cooling capacity during high peak power demand periods. However, a single pump may be provided in other embodiments to oil to both the power electronics module **700** and to the motor **730**. For either cases (one or two oil pumps), it may be possible to further reduce the oil **845** temperature prior to contacting the power electronics, by contacting the oil with at least one outside surface of the evaporator **110** prior to contacting an outside surface of the power electronics module **700**. For example, this may be accomplished by contacting the oil **845** on one or more outside surfaces of the evaporator **110** above the power electronics module **700**, after which the oil flows by gravity over the power electronics module **700**. The oil **845** may then transfer heat to the evaporator **110** and vaporize the two-phase heat-transfer fluid contained therein to reduce the oil temperature. The cooled oil **845** may then be sprayed **850***b* onto the power electronics to provide additional cooling. The heated oil will then gravity drain back to the oil collection pan **870**. Thus, the example of FIG. **10** illustrates one or more spray nozzles **890***a* for distributing oil (e.g. in the form of an oil spray **850***a*) onto surfaces of the motor **730** for cooling the motor.

[0150] FIG. **10** illustrates most of the cooling system **100** components housed within a containment system **805**, with the exception of the condenser **120** and a portion of the main channel **210** of the evaporator **110**. This example also shows at least one power electronics module **700** positioned in thermal communication with a first wall **220** of the evaporator **110**. The power electronics module **700** is sandwiched between the first wall **220** of the evaporator **110** and a conducting element **400** having one or more surface area extenders **830**.

[0151] In any of the examples described above, a feasible alternative may include replacing the oil spray/impingement concepts with a jet impingement, oil-dripping, or channel-flow type cooling strategy. In a channel flow configuration, a pump may circulate a heat-transfer liquid (e.g. oil) through a heat exchanger mounted on one or more external surfaces of the power electronics module. In other examples, oil spray concepts may be utilized in conjunction with a jet impingement, oil-dripping, or channel flow configurations.

[0152] In some embodiments, a cooling system having a condenser and an evaporator may operate at a pressure ranging from close to absolute vacuum to about 300 psia. In other embodiments, a cooling system having a condenser and an evaporator may operate at a pressure ranging from close to 200 psia to about 300 psia. A two-phase heat-transfer fluid, may have a boiling point of about 50° C. to about 100° C. at a pressure of one atmosphere. A heat-transfer liquid (e.g. oil) may operate at a temperature of about 90° C. to about 120° C. A heat-transfer liquid may have a high operating temperature, T_H, and a low operating temperature, T_L, where the difference between T_H and T_L is from about 1° C. to about 10° C. In some examples of a cooling system, the temperature difference between the operating temperature, T_1, of a two-phase heat-transfer fluid and the high operating temperature, T_H, of a heat-transfer liquid (e.g. oil), may be a ΔT of about 5° C. to about 20° ($\Delta T = T_H - T_1$).

[0153] As used herein, the terms "about" and "substantially" refer to a plus and/or minus deviation of 5% around the numerical value stated. For example, "about 100° C." refers to a temperature from 95° C. to 105° C. "About 200 psia" refers to a pressure from 190 psia to 210 psia. The term "substantially vertical" refers to an element positioned at an angle of 85.5 degrees to 94.5 degrees relative to a reference plane.

Example 3

Testing of a Passive, Two-phase Cooling System

[0154] Experimental Apparatus: FIG. **11** illustrates a schematic of a cooling system tested. Two different evaporator design were tested—a copper cold plate-based design and an aluminum-based design. Both evaporators were designed to cool six Delphi discrete power modules/switches. The six power modules represent a voltage source inverter system required to power an automotive electric machine/motor. Further details of the the evaporator designs tested are provided below.

[0155] Two condensers were tested, one with plain tubes and one with rifled tubes. The rifled features, shown in FIG. **12**, were evaluated as a means of enhancing condensation heat transfer. Per the manufacturer, the condensation-side surface areas were 1,190 cm^2 and 1,250 cm^2 for the plain and rifled condensers, respectively. The two condensers were finned-tube, air-cooled type heat exchangers. The fins were louvered and fabricated from 0.15-mm-thick aluminum sheets. The total air-side surface area was calculated to be approximately 29,000 cm^2. Twenty 0.95-cm (⅜-in.) outer-diameter copper tubes were arranged in two rows (see FIG. **11**). Two tubes, one from each row, were connected at the top via U-bend fittings. A manifold header connected all tubes at the lower end.

[0156] Heat from the system was rejected to air by means of a 17.8-cm (7-in.) diameter automotive axial fan mounted to the condenser. The fan operated on 12 V and consumed 38 W of power, calculated by voltage and current measurements. Two 2.54-cm-inner-diameter tubes connected the condenser manifold to the evaporator. The tubes were sized large enough to prevent falling liquid from restricting/blocking the rising vapor.

[0157] The system was designed with a maximum operating pressure of 1.03 MPa (150 psi). Finite element structural analysis was used to design the evaporators to allow for elevated-pressure operation. The condensers were purchased and had a 2 MPa pressure rating. Once the condensers and evaporators were assembled, hydrostatic pressure-tests were conducted to verify the system's pressure rating. The cooling system was instrumented with sensors to measure system pressure and temperatures. System vapor pressure was measured using a calibrated absolute pressure transducer. System temperatures were measured using calibrated K type thermocouples for the inlet-air, outlet-air, liquid, vapor, and heater temperatures.

[0158] Experimental Procedures: Six Watlow ceramic heaters were used in place of the Delphi power modules. The dimensions of the ceramic heaters (25 mm×15 mm×2.5 mm) were similar to those of the Delphi modules. Each heater generated up to about 580 W of heat (total power for six heaters: 3,500 W). The ceramic heaters were externally attached to the evaporators using thermally conductive grease as the thermal interface material. The temperatures of the heaters were measured via thermocouples embedded within the ceramic heaters.

[0159] After the heaters were attached to the evaporator, the system was charged using a transfer tube that contained a measured amount of saturated, oil-free refrigerant. Prior to charging the system with the refrigerant, the air in the system was removed using a vacuum pump. Pressure within the system was allowed to decrease to about 2 Pa to ensure that most of the air was evacuated. A valve between the transfer tube and the system was then opened, allowing the refrigerant to drain into the system. Once the refrigerant was transferred, the valve between the transfer tube and the system was closed, and the transfer tube was disconnected from the system. Measurements of the vapor temperature and pressure confirmed saturated conditions, verifying that no air was present within the system.

[0160] Experiments were initiated after saturated conditions were verified. First, the fan was powered to pull ambient air (Ta inlet=25° C.) through the condenser. The six heaters were then powered using two Agilent direct current power supplies. Heater loads ranged from 250 W to 3,500 W. The system was allowed to reach equilibrium conditions (about 15 minutes) at each power level before increasing the power. Measurements of the system and heater temperatures combined with heat load measurements were used to compute the evaporator and condenser thermal resistances at various power levels.

[0161] Uncertainty Analysis: Analysis was conducted to quantify the uncertainty in the measured experimental values according to the procedures known in the art. The procedure consisted of gathering systematic and random uncertainties in all measured variables. The propagation-of-error equation was then used to estimate the uncertainties in the calculated values. The uncertainty for the thermal resistance values is estimated to be approximately ±9%. After calibration, the uncertainty in a thermocouple was conservatively estimated at ±0.1° C., while the uncertainties in the heat measurements were estimated to be ±1%. All stated uncertainties were calculated at a 95% confidence level.

[0162] Results: Experiments were conducted to measure the thermal performance of a passive two-phase cooling system. This included measuring the thermal resistance of the condenser and evaporator. The thermal performance data combined with the volume, weight, and parasitic power requirements of the two-phase cooling system were then compared with equivalent metrics of a conventional, water-ethylene glycol-based (WEG) cooling system.

[0163] Experiments were conducted to measure the thermal resistance of two evaporator designs—a copper cold plate-based design and an advanced all-aluminum-based design. The evaporator's specific (area-weighted) thermal resistance was defined per Equation 1.

$$R''_{th,evaporator} = \frac{(\overline{T_{htr}} - T_l)}{\text{Total power}} \times A_{htr} \quad (1)$$

[0164] where $\overline{T_{htr}}$ is the average temperature of the six ceramic heaters as measured by the thermocouples embedded within each heater, and T_l is the refrigerant liquid temperature. The total power is the total heat dissipated by the system and A_{htr} is the total surface area of the six heaters (22.5 cm^2).

[0165] Evaporator Thermal Performance (Concept 1): The initial copper cold plate design consisted of a steel cylinder and three copper cold plates and was charged with 250 cm^3 of refrigerant (see FIG. **13**). This evaporator's flexible design also allows testing of different boiling/evaporation enhancement techniques (e.g., boiling enhancement coatings and finned surfaces). The copper cold plates were machined with 4-mm-tall fins, each of which has a wetted surface area of about 55 cm^2. Because two-phase cooling provides high heat transfer rates, the fins are not expected to have a significant effect on the evaporator thermal resistance. However, the fins were included as a means of increasing surface area in an attempt to delay dry-out.

[0166] The specific thermal resistance of the copper cold plate evaporator is plotted versus the total heat dissipated in FIG. **14**. Results for refrigerants HFC-245fa (330 grams) and HFO-1234yf (280 grams) are provided. As shown in FIG. **14**, the heater-to-liquid resistances for both refrigerants decrease with increasing heat dissipation. This effect is associated with an increased contribution from two-phase heat transfer (i.e., boiling/evaporation) at higher power levels, which improves thermal performance. Tests were performed with increasing and decreasing heat fluxes to evaluate for hysteresis effects. Results showed that using an increasing or decreasing heat rate had little effect on the thermal performance of the cooling system.

[0167] With 250 cm^3 of HFC-245fa, the system was capable of dissipating 3.5 kW of heat under steady-state conditions without reaching dry-out (i.e., critical heat flux). The refrigerant-volume-to-heat-dissipated ratio, at maximum heat load, was 71 cm^3/kW. At 3.5 kW, the heaters were at their maximum allowable power rating, thus it was not possible to test at higher heat loads. The 3.5 kW of heat dissipation is noteworthy because it was a conservative estimate on the inverter heat dissipation requirement for a 55-kW electric traction-drive system. With HFO-1234yf, tests were limited to lower heat loads due to HFO-1234yf's higher operating pressures. At 1.25 kW of heat dissipation, HFO-1234yf's saturated temperature reached 42° C., which corresponded to a pressure of about 1.1 MPa—the maximum operating pressure of the system. Increasing the heat dissipation with HFO-1234yf required increasing the condensing and/or pressure capacity of the system. Compared at the same heat loads, HFO-1234yf produced thermal resistance values that were 22%-47% lower than those produced with HFC-245fa. HFO-1234yf's higher heat transfer coefficients allowed it to outperform HFC-245fa.

[0168] Evaporator Thermal Performance (Advanced Design): Features to improve the evaporator's performance and reduce its size were identified and incorporated into an advanced evaporator design. The features were intended to reduce the evaporator thermal resistance while utilizing low-cost fabrication techniques and materials (e.g., aluminum). The advanced evaporator concept used an indirect cooling approach to be compatible with conventional power electronic packages (i.e., silicon on ceramic substrate). A simple schematic of the advanced evaporator design is shown in FIG. **15**. The evaporator was designed to cool six Delphi discrete power switches. The advanced design reduced the refrigerant volume to 180 cm^3 (HFC-245fa=240 grams, HFO-1234yf=200 grams). For comparison, the 2010 Toyota Camry requires 510 grams of refrigerant (HFC-134a) for the cabin's air-conditioning system. Also, typical automotive inverter coolant systems utilize several liters of WEG (i.e., antifreeze). Reducing the refrigerant quantity is an effort to reduce system cost, weight, and size. As shown in FIG. **15**, the advanced design is more compact as compared with the previous copper-based design.

[0169] The specific thermal resistance of the advanced evaporator design is compared in FIG. **16** to the thermal performance of the previous copper cold-plate design. The performance curves shown in this figure were obtained using HFC-245fa. The results demonstrated that the advanced design can dissipate automotive-scale power electronic heat loads (3.5 kW) with only 180 cm^3 of refrigerant. Additionally, the unique features of the advanced evaporator design allowed the aluminum evaporator to provide equal or lower thermal resistance values as compared with the initial evaporator design that utilized copper cold plates. Further reductions to the thermal resistance can be obtained if the advanced design is fabricated entirely out of copper. However, in an effort to reduce cost and weight, the use of aluminum is seen as a more practical option.

[0170] At higher heat loads (≥3 kW), fluctuations in the pressure and temperature were observed for tests with the advanced evaporator design. These fluctuations resulted in an increase in thermal resistance. The fluctuations are believed to be associated with the compact size of the advanced evaporator design that restricts vapor from exiting the evaporator. Applying microporous coatings to the evaporator's boiling surface may reduce or eliminate the fluctuations. Microporous coating structures provide passive refrigerant transport via wicking and may reduce the amount of vapor generation and these effects are believed to potentially suppress pressure-temperature fluctuations in passive, two-phase systems.

[0171] It is important to note that the thermal performance results provided in FIGS. **14** and **16** were obtained using ceramic heaters attached via thermal interface material. These heaters were used as a substitute for actual power modules attached via bonded interface materials (e.g., thermoplastics, solder). Therefore, the thermal resistance values are not necessarily indicative of the thermal performance in actual applications. Also, the thermal performance results provided in FIGS. **14** and **16** were obtained using non-coated evaporator surfaces. The use of boiling enhancement coatings such as 3M's copper microporous coating within the evaporator may enable both increased heat dissipation and lower thermal resistance values.

[0172] The junction-to-liquid thermal resistance of the advanced evaporator design was simulated through finite element analysis (FEA). A computer-assisted design (CAD) model of the Delphi power modules bonded to the advanced evaporator design was first developed. The CAD model incorporated all the thermal resistance interfaces within the module stack, including the solder layers within the power electronics module and a thermoplastic-type bonded interface between the power electronics module and the evaporator. The model was then imported into ANSYS Workbench for thermal analysis. The FEA imposed heat transfer coefficient boundary conditions to simulate boiling heat transfer from enhanced surfaces within the evaporator. The boiling heat transfer coefficients for HFC-245fa and HFO-1234yf were experimentally measured with enhanced surfaces (3M microporous boiling enhancement coating) at various saturated temperatures. The measured heat transfer coefficient values exceeded 100,000 W/m^2-K within a wide heat flux range. Moreover, the performance of the two refrigerants was found to be similar for boiling on microporous-coated surfaces. Because uniform boiling may not occur with the evaporator surfaces, a conservative estimate on the heat transfer coefficients was imposed (50,000 W/m^2-K) for these simulations.

[0173] The specific thermal resistance results (junction-to-liquid) as predicted by FEA are provided in Table 1. The insulated gate bipolar transistor's area and maximum temperature (i.e., junction temperature) combined with the refrigerant temperature were used to calculate the thermal resistance values. Two evaporator designs were analyzed—aluminum-based and copper-based. The aluminum fabrication is a more practical design while the copper fabrication is more of a concept aimed at reducing the package stack resistance. For comparison, the thermal resistance (junction-to-liquid) of the 2008 Lexus Hybrid double-sided and liquid (WEG)-cooled power module are also shown in Table 1. The thermal resistance value of 0.33 cm^2-K/W was calculated using performance data known in the field.

TABLE 1

The FEA-estimated evaporator (junction-to-liquid) thermal resistances for aluminum and copper evaporators. Performance data for an automotive cooling system are provided for comparison.

	R''_{th} (cm^2-K/W)	(% R''_{th} reduction
Lexus Hybrid (2008) double-side cooled modules	0.33 [11, 12]	—
Two-phase: aluminum baseplate (finite element analysis results)	0.14	58%
Two-phase: copper baseplate (finite element analysis results)	0.12	65%

[0174] The FEA-predicted thermal resistance values of the aluminum (0.14 cm^2-K/W) and copper (0.12 cm^2-K/W) evaporator modules are about 58% and 65% lower, respectively, than that of the double-sided, WEG-cooled 2008 Lexus system. (The 2013 Toyota Camry Hybrid uses a similar power modules and cooling system.) These thermal resistance reductions translate to a 139% and 189% increased heat flux capacity. Experimentally measuring the thermal resistance of the advanced evaporator with a power module was not possible due to equipment power limitations. An immersion cooling (two-phase) strategy may allow for even greater thermal enhancements. However, an indirect cooling approach was used in this case to allow for its use with more traditional power modules and to alleviate some reliability concerns associated with immersing electronics in a refrigerant.

[0175] Condenser Thermal Performance and Analysis: The unit-thermal resistances for the condensers are provided in Table 2 for both refrigerants. The vapor-to-air resistance includes the condensation-side and the air-side resistances and was defined per Equation 2.

$$R''_{th,condenser} = \frac{(T_v - \overline{T}_a)}{\text{Total power}} \times Condenserfrontal\ \text{area} \tag{2}$$

[0176] where T_v is the refrigerant vapor temperature and $\overline{T}_a$ is the average inlet air temperature as measured by two thermocouples just upstream of the condenser. The stated condenser performances are for a passive condenser design cooled via an automotive axial fan. The total fan parasitic power was measured to be 38 W. An estimate of the air flow rate was calculated by imposing an energy balance on the air-side (i.e., inlet and outlet to the condenser) and the total heat dissipated. Assuming steady-state conditions and incorporating the appropriate air thermal properties, the estimated air volumetric flow rate is 0.12 m^3/s (250 ft^3 per minute).

TABLE 1

Condenser thermal resistance values calculated using Equation 2.

Refrigerant	R''_{th} (cm^2-K/W) Plain	Rifled	Rifled enhancements
HFC-245fa	9.30	7.58	18%
HFO-1234yf	8.12	6.06	25%

[0177] Condensation-side enhancements from the rifled tubes were found to reduce the overall condenser thermal resistances by 18% and 25% for HFC-245fa and HFO-1234yf, respectively. The results show that the rifled structures are more effective at enhancing condensation heat transfer with HFO-1234yf. Moreover, better performance is achieved with HFO-1234yf. For the plain tube condenser, HFO-1234yf yielded thermal resistance values that were about 13% lower than those with HFC-245fa. For the rifled tube condenser, HFO-1234yf yielded thermal resistance values that were about 20% lower than those with HFC-245fa.

[0178] An analysis was conducted to estimate condenser size based on operating conditions, a maximum allowable cooling system temperature (i.e., vapor and liquid), and the experimentally measured condenser thermal resistance values provided in Table 1. The operating conditions used for this analysis were: 3.3 kW of steady-state heat dissipation (estimated requirements for a 55-kW traction drive inverter) and 43° C. inlet air temperature. The estimated condenser frontal area (i.e., frontal footprint) requirements are plotted versus the system temperature in FIG. **17**. In the figure, the lower system temperature of 65° C., used for this analysis, corresponds to the coolant (i.e., WEG) temperature typical of existing automotive power electronics systems. The solid line curves for both refrigerants are the sizing requirements for the finned-tube condenser per the performance data reported in Table 1. The T_j curve in FIG. **17** represents the maximum junction temperature and was calculated using the thermal resistance data for an aluminum evaporator and a six power module/switch system configuration.

[0179] Because typical automotive condensers are constructed using a brazed, folded-fin design, a simplified analysis was conducted to estimate the sizing requirements for a folded-fin type condenser. For this analysis, the ratios of the air-side surface area to frontal surface area were measured for finned-tube and folded-fin condensers. The air-side-to-frontal-surface-area ratios were calculated to be 55.5 mm^2/mm^2 for the finned-tube and 86.7 mm^2/mm^2 for folded-fin condensers, assuming a condenser thickness of 4.5 cm for both cases. The folded-fin frontal area requirements were estimated by matching the air-side surface area for the folded-fin design to those of the finned-tube design at the various system temperatures. The folded-fin frontal area requirements were then calculated via the surface area ratio (86.7 mm^2/mm^2) and the results are shown FIG. **17**. This simplified analysis assumes that the air-side is the dominant thermal resistance, and air velocities are similar to those provided by the fan used in this study. For reference, the performance of a folded-fin condenser operating in a thermosyphon configuration, as reported by Barnes and Tuma ("Practical Considerations Relating to Immersion Cooling of Power Electronics in Traction Systems," *IEEE Transactions on Power Electronics*, (25: 9), 2010; pp. 2478-2485.), is provided in FIG. **17**. In that study, a condenser thermal resistance of 4.6 cm^2-K/W was reported using a hydrofluoroether fluid and an air velocity and pressure drop of 2.2 m/s and 44 Pa, respectively. The estimated condenser sizing requirements, per the performance reported by Barnes and Tuma, are similar to the results predicted for HFC-245fa.

[0180] The size requirements of the condenser can then be estimated using FIG. **17** and a maximum allowable system temperature. For low-temperature applications typical of existing automotive power electronics coolant temperatures and conventional silicon devices, the use of HFO-1234yf is recommended because it provides better performance. At a system temperature of 80° C. and with HFO-1234yf, the condenser frontal size requirements are estimated to be about 325 cm^2 with a maximum T_j of about 125° C. HFC-134a may also be considered in this case because it is a refrigerant currently used in automotive systems and because prior studies have shown that its heat transfer performance (boiling and condensation) is similar, if not slightly better than, that of HFO-1234yf.

[0181] Operating at higher system temperatures has advantages because it allows for a more compact condenser. However, higher temperature operation may be dependent on the development of higher temperature auxiliary electronic devices (e.g., capacitors) that are packaged within the inverter. With this in mind, if higher system temperatures are practical, HFC-245fa is recommended due to its higher critical temperature (T_C=154° C.) and lower operating pressures. At a system temperature of 95° C. and a T_j=148° C., the condenser size requirements would decrease to about 310 cm^2.

[0182] Two-Phase and Baseline Cooling Metric Comparisons: Analysis was conducted to compare the coefficient of performance (COP), volume, and weight of a passive two-phase cooling system with that of a conventional automotive WEG-based (single-phase) power electronics cooling system.

[0183] Passive Two-Phase Inverter Cooling System Metrics: The condenser, evaporator, and refrigerant were included in the volume and weight metrics for two-phase cooling system. The metrics for an aluminum evaporator were directly measured from the prototype that was fabricated and tested. The condenser metrics were estimated based on the condenser finned-frontal-area requirements from FIG. **17** assuming 80° C. (HFC-245 assumed) maximum system (liquid and vapor) temperatures. The additional volume of the headers/manifolds and specific weight (weight/volume) of an automotive condenser was measured and used to compute the total volume (finned and headers) and the weight of the condenser for the passive two-phase cooling system. The 240 grams (HFC-245 assumed) of refrigerant was included in the total system weight. The refrigerant volume was not included in the total volume measurement because the refrigerant is contained within the evaporator and thus its volume is accounted for in the evaporator volume. The thermal resistance of an aluminum evaporator (from Table 1) combined with the total parasitic power of the system (38-W fan parasitic power) was used to compute a COP value per Equation 3.

$$COP = \frac{1}{R''_{th} \times \text{Parasitic power}} \tag{3}$$

[0184] The volume and weight of the fan and associated piping (connecting the evaporator to the condenser) were not included in the total volume and weight calculations.

[0185] Baseline, WEG-Based Inverter Cooling System Metrics: Conventional automotive WEG-based inverter cooling systems consists of a cold plate to directly cool the inverter, radiator, pump, fan, WEG, and associated piping. Estimating the metrics for a baseline, the WEG-based automotive inverter cooling system is not a straightforward procedure because these systems are also designed to cool the electric motor. Therefore, the radiators in these inverter cooling systems are sized to dissipate the combined heat loads of the inverter and the electric motor. Because the proposed two-phase cooling system only considers cooling the inverter, analysis was conducted to size a radiator to dissipate heat loads for the inverter only, per procedures known in the art. The radiator was sized to dissipate 3.3 kW of heat with a maximum inlet air temperature of 43° C.—the same conditions used to size the two-phase condenser. Additionally, a 10 L/min WEG flow rate was assumed within the radiator. From this analysis, the finned radiator volume, fan pressure drop and flow rate, and pump pressure drop were estimated. The additional volume of the headers/manifolds and specific weight (weight/volume) of an automotive radiator was measured and used to compute the total volume (finned and headers) and the weight of the inverter-only radiator. The air-side pressure drop and flow rate were used to compute the fan parasitic power assuming 20% fan efficiency.

[0186] The 2012 Nissan Leaf inverter (80-kW motor) cold plate/heat exchanger was used to compute the volume, weight, and WEG pump parasitic power metrics for the inverter cold plate. The volume of the cold plate was directly measured and then scaled down by the ratio 55 kW/80 kW (80 kW for the Nissan Leaf and 55 kW for the two-phase cooling system). The weight was computed assuming the density of aluminum (6061) and a 60% cold plate void space (i.e., WEG channels).

[0187] Experiments were conducted to measure the parasitic power required to circulate WEG (10 L/min flow rate and 60° C.) through the 2012 Nissan Leaf inverter cold plate. The power required to circulate WEG through the inverter cold plate and radiator was used to compute the total pump parasitic power requirements assuming a 60% pump efficiency. The COP of the baseline system was calculated per Equation 3 using the total parasitic power (fan and pump) and the specific thermal resistance (junction-to-liquid) of the 2012 Nissan Leaf.

[0188] The volume of the WEG, contained within the Nissan Leaf cold plate was measured and doubled in an attempt to account for the total WEG volume in both the cold plate and radiator. The total WEG volume was computed to be approximately 1.6 L. The WEG's volume was not used when computing the overall volume of the system because the WEG is contained within the system components and thus its volume is accounted for in the cold plate and radiator volume. However, the WEG volume was used to compute the WEG weight via the specific weight properties of WEG. The weight of the WEG was then utilized to compute the overall baseline system weight. The volume and weight of the fan, pump, tubing, and WEG within the tubing were not included in the total volume and weight calculations for the baseline cooling system.

[0189] As shown in Table 3, utilizing a passive two-phase cooling system reduced the weight by an estimated 41% and increase the COP by an estimated 127%. The volumes of the two cooling systems were nearly identical; however, more components were excluded from the baseline system (i.e., fan, pump, piping, and additional WEG volume and weight) as compared with the two-phase system (i.e., fan and piping) and thus reductions to the volume are also likely. The majority of the reductions to the system are associated with reducing the amount of coolant/refrigerant and reducing the size of the inverter/power module heat exchanger. The advanced aluminum evaporator 33% the volume of the scaled-down (by 55 kW/80 kW) 2012 Nissan Leaf cold plate. Moreover, the two-phase cooling system contains about 17% less aluminum (by weight) as compared to the WEG-based cooling system.

TABLE 2

Percent reductions to the volume and weight and percent increases to the COP and heat dissipation using a passive two-phase cooling system.

Volume reduction	Weight reduction	COP	Heat dissipation increase
−2%	41%	127%	139%

[0190] It should be noted that the size of the condenser can be reduced and the volume and weight reductions listed in Table 3 would increase, if the two-phase coolant system is allowed to operate at higher temperatures (FIG. 7). In this analysis, the maximum two-phase temperatures (vapor and liquid) were kept below 85° C. to enable the use of low-temperature capacitors.

[0191] Conclusions: Experiments and analysis have been conducted to evaluate the use of two-phase cooling (passive) for automotive power electronics. The following summarizes some of the results from this work:

[0192] It was demonstrated that a passive two-phase cooling system can cool six Delphi discrete power modules, dissipating about 55 kW inverter-scale heat loads (~3.5 kW) with only 180 mL of refrigerant.

[0193] The proposed passive and indirect two-phase cooling approach may reduce the junction-to-liquid thermal resistance by 58% to 65%. These reductions to the thermal resistance translate to a 139% to 189% increase in the device heat dissipation capabilities.

[0194] Analysis indicates that using a passive two-phase cooling approach may reduce the weight (by 41%) and significantly improve the COP (by 127%) of the inverter thermal management system. Additional improvements may be obtained if the two-phase cooling system is allowed to operate at higher vapor-liquid temperatures.

[0195] The foregoing discussion and examples have been presented for purposes of illustration and description. The foregoing is not intended to limit the aspects, embodiments, or configurations to the form or forms disclosed herein. In the foregoing Detailed Description of Some Embodiments for example, various features of the aspects, embodiments, or configurations are grouped together in one or more embodiments, configurations, or aspects for the purpose of streamlining the disclosure. The features of the aspects, embodiments, or configurations, may be combined in alternate aspects, embodiments, or configurations other than those discussed above. This method of disclosure is not to be interpreted as reflecting an intention that the aspects, embodiments, or configurations require more features than are expressly recited in each claim. Rather, as the following claims reflect, inventive aspects lie in less than all features of a single foregoing disclosed embodiment, configuration, or aspect. While certain aspects of conventional technology have been discussed to facilitate disclosure of some embodiments of the present invention, the Applicants in no way disclaim these technical aspects, and it is contemplated that the claimed invention may encompass one or more of the conventional technical aspects discussed herein. Thus, the following claims are hereby incorporated into this Detailed Description of Some Embodiments, with each claim standing on its own as a separate aspect, embodiment, or configuration.

What is claimed is:

1. An evaporator comprising:
a first wall comprising:
an external surface;
a first edge; and
a second edge, wherein:
the first wall, the first edge, and the second edge are substantially parallel to a plane, and
the first edge and the second edge are substantially parallel to each other;
a second wall extending from the first edge and the external surface, substantially perpendicular to the plane, and substantially parallel to the first edge; and
a third wall extending from the second edge and the external surface, substantially perpendicular to the plane, and substantially parallel to the second edge, wherein:
the external surface of the first wall, the second wall, and the third wall form a passage substantially parallel to the first edge,
the passage is configured to contain at least one heat source, and
at least the first wall is configured to be in thermal communication with the at least one heat source.

2. The evaporator of claim **1**, further comprising a conducting element, wherein:
the conducting element is positioned within the passage,
the conducting element is configured to be in thermal communication with the at least one heat source, and
the conducting element is in thermal communication with at least one of the second wall or the third wall.

3. The evaporator of claim **2**, wherein:
the conducting element comprises a block of material with a substantially triangular cross-section,
the block of material has a first side that is in thermal communication with either the second wall or the third wall, and
the block has a second side that is configured to be in thermal communication with the at least one heat source.

4. The evaporator of claim **2**, wherein the conducting element comprises:
a first block of material with a substantially triangular cross-section; and
a second block of material with a substantially triangular cross-section, wherein:
the first block of material has a first side that is in thermal communication with the second wall, and a second side that is configured to be in thermal communication with the at least one heat source, and
the second block of material has a first side that is in thermal communication with the third wall, and a second side that is configured to be in thermal communication with the at least one heat source.

5. The evaporator of claim **1**, wherein the first wall further comprises an internal surface and the evaporator further comprises:
a surface area extender that extends from the internal surface, wherein:
the internal surface is substantially parallel to the external surface, and
the surface area extender is substantially perpendicular to the plane.

6. The evaporator of claim **5**, wherein the surface area extender comprises a fin, wherein the fin is substantially parallel to the first edge.

7. The evaporator of claim **6**, wherein:
the internal surface and the external surface define a width of the first wall, and
the fin comprises a height that is approximately equal to the width.

8. The evaporator of claim **1**, wherein:
the second wall terminates with an edge,
the third wall terminates with an edge,
the edge of the second wall and the edge of the third wall are positioned below the external surface, and
the evaporator further comprises:
a first tab extending from the edge of the second wall, substantially perpendicular to the second wall, and away from the passage; and
a second tab extending from the edge of the third wall, substantially perpendicular to the third wall, and away from the passage.

9. The evaporator of claim **8**, further comprising a housing, wherein:
the housing comprises:
a first end connected to the first tab;
a second end connected to the second tab; and
at least one wall physically connecting the first end to the second end, wherein:
the housing forms a first interior channel between the interior surface of the first wall and the housing,
the housing forms a second interior channel between the second wall and the housing, and
the housing forms a third channel between the third wall and the housing.

10. The evaporator of claim **9**, wherein the first wall, the second wall, the third wall, the first tab, the second tab, and the housing are all a single piece of material.

11. The evaporator of claim **10**, wherein the single piece of material is aluminum.

12. The evaporator of claim **9**, wherein the first wall, the second wall, the third wall, the first tab, and the second tab are all a first single piece of material.

13. The evaporator of claim **12**, wherein the first single piece of material is aluminum.

14. The evaporator of claim **13**, wherein the housing is a second single piece of material.

15. The evaporator of claim **14**, wherein the second single piece of material is copper.

16. A cooling system comprising:
an evaporator configured to cool a first heat source;
a condenser in fluid communication with the evaporator;
a refrigerant contained within the condenser and the evaporator; and
a circulating liquid system comprising:
a liquid contained within the circulating liquid system;
a liquid reservoir;

a pump in liquid communication with the liquid reservoir;

a first heat exchanger in liquid communication with the pump, wherein the first heat exchanger is configured to deliver heat from the liquid to the refrigerant; and

a second heat exchanger in liquid communication with the first heat exchanger, wherein:

the second heat exchanger is configured to deliver heat from a second heat source to the liquid,

the second heat exchanger is in liquid communication with the liquid reservoir, and

the pump circulates the liquid through the first heat exchanger, the second heat exchanger, and the liquid reservoir.

17. The cooling system of claim **16**, wherein the second heat exchanger comprises a spray nozzle, wherein the liquid is sprayed onto the second heat source.

18. The cooling system of claim **17**, wherein the liquid reservoir comprises a liquid collection pan configured to collect oil heated by the second heat source.

19. The cooling system of claim **16**, wherein the evaporator and the circulating liquid system are fluid-sealed within a container operating at a first average temperature.

20. The cooling system of claim **19**, wherein the condenser operates at a second average temperature that is less than the first average temperature.

* * * * *