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(54) HEAT EXCHANGER COMPRISING WAVE-SHAPED FINS

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(57) **ABSTRACT**

A heat exchanger for exchanging heat between a first fluid and a second fluid is disclosed; wherein the heat exchanger has improved thermal efficiency and low fluid back pressure. The heat exchanger comprises conduits for conveying the first fluid, wherein the conduits include a plurality of flow passages. The flow passages are defined by a plurality of fins that are continuous along the direction of flow of the first fluid. Each fin includes a wave-shaped region, and adjacent fins sub-divide each flow passage into first and second sections that are interposed by a third section. The wave-shape of the fins creates a continuously varying cross-sectional area for each third section. The variation of the cross-sectional area of the third section, coupled with the wave-shape of the fins, induces a swirl flow between the third section and each of the first and second sections. This swirl flow improves the efficiency of the overall convection heat transfer in each conduit. Further, the overall cross-sectional area of each conduit remains constant even as the cross-sectional areas of individual flow passages changes, which mitigates the development of back pressure in the flow of the first fluid.

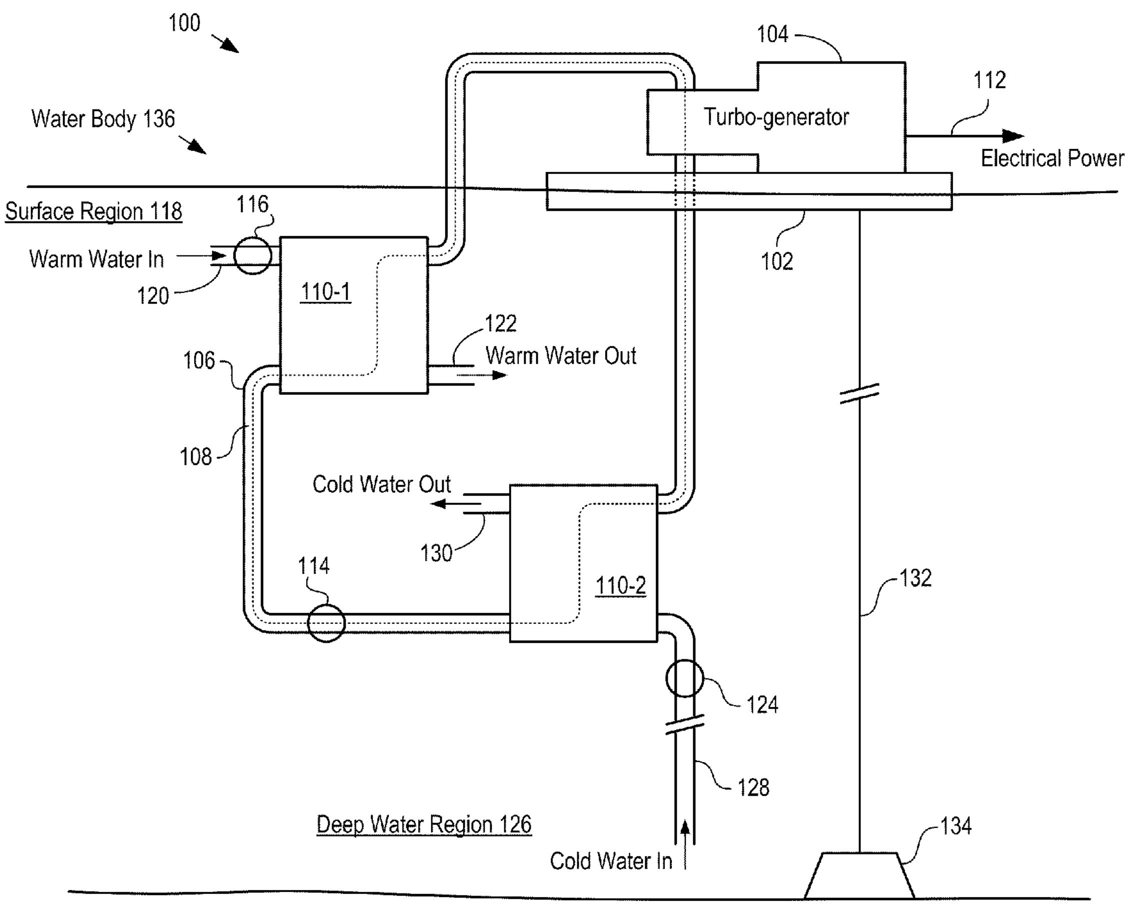
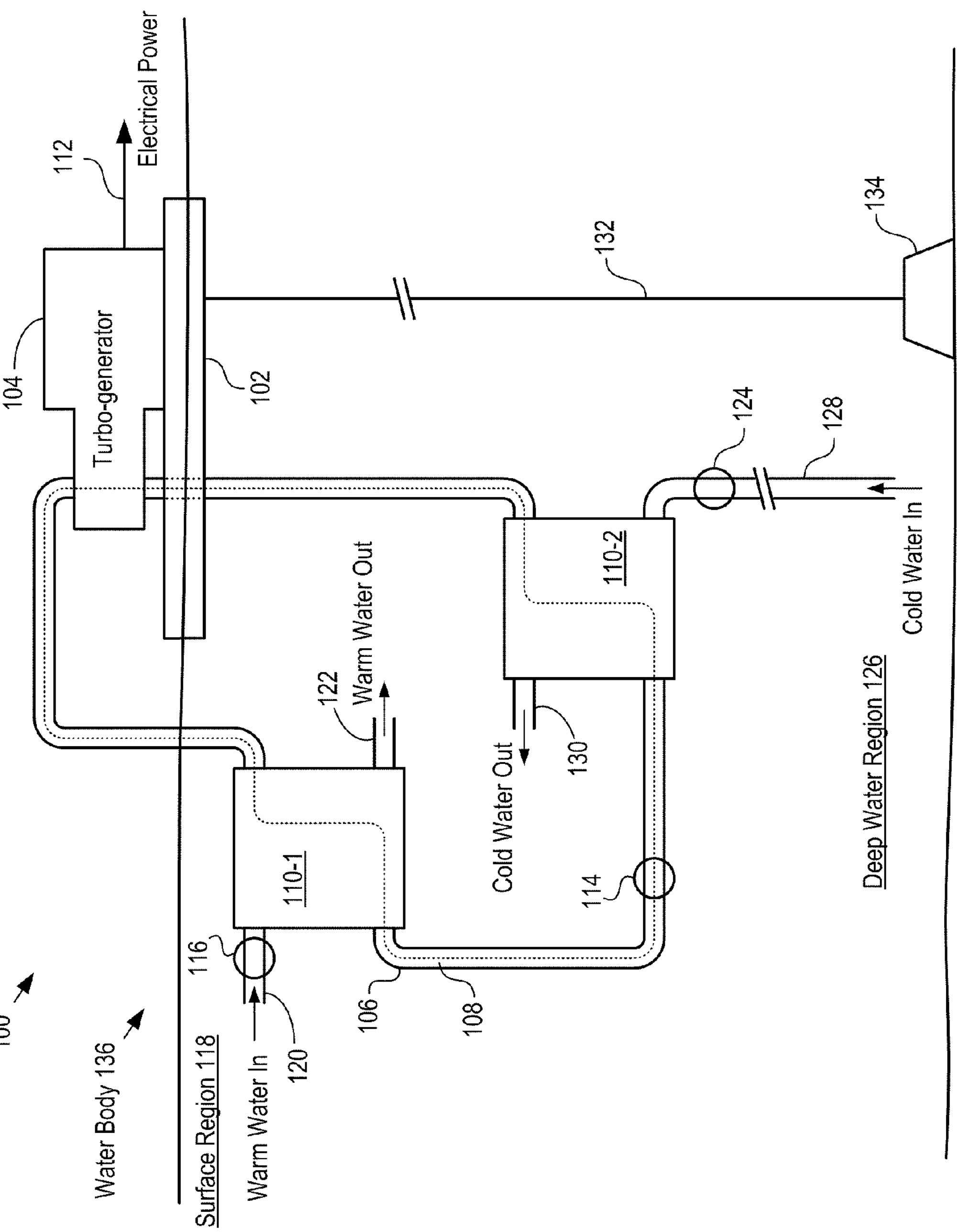


FIG. 1


100
Water Body 136
Surface Region 118
Warm Water In
120
116
110-1
106
108
122
Warm Water Out
114
Cold Water Out
130
110-2
Deep Water Region 126
Cold Water In
124
128
Turbo-generator
104
102
112
Electrical Power
132
134

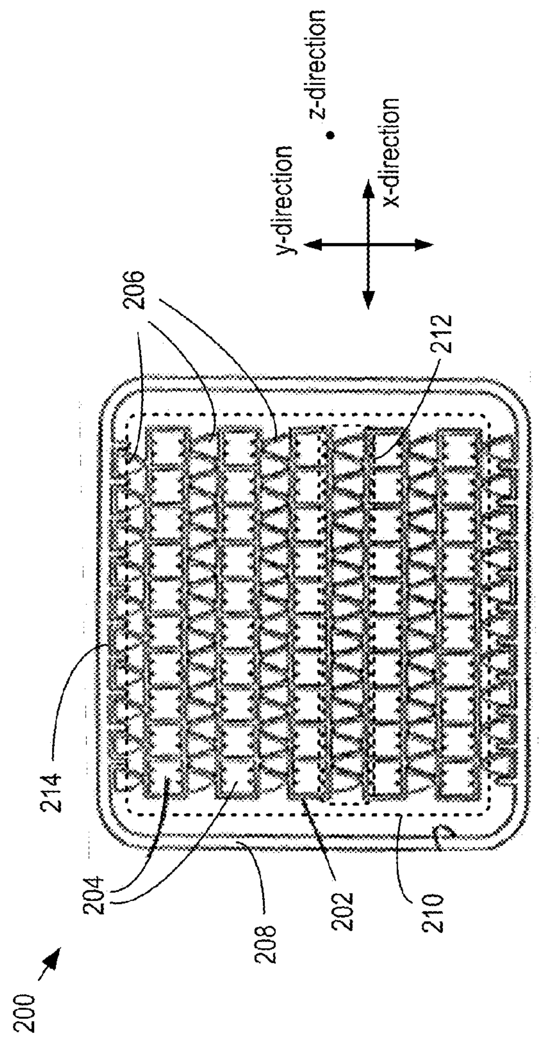


FIG. 2A (Prior Art)

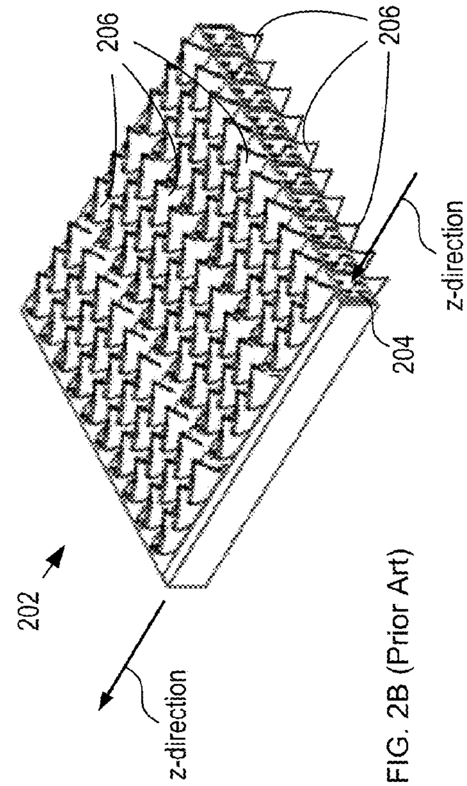


FIG. 2B (Prior Art)

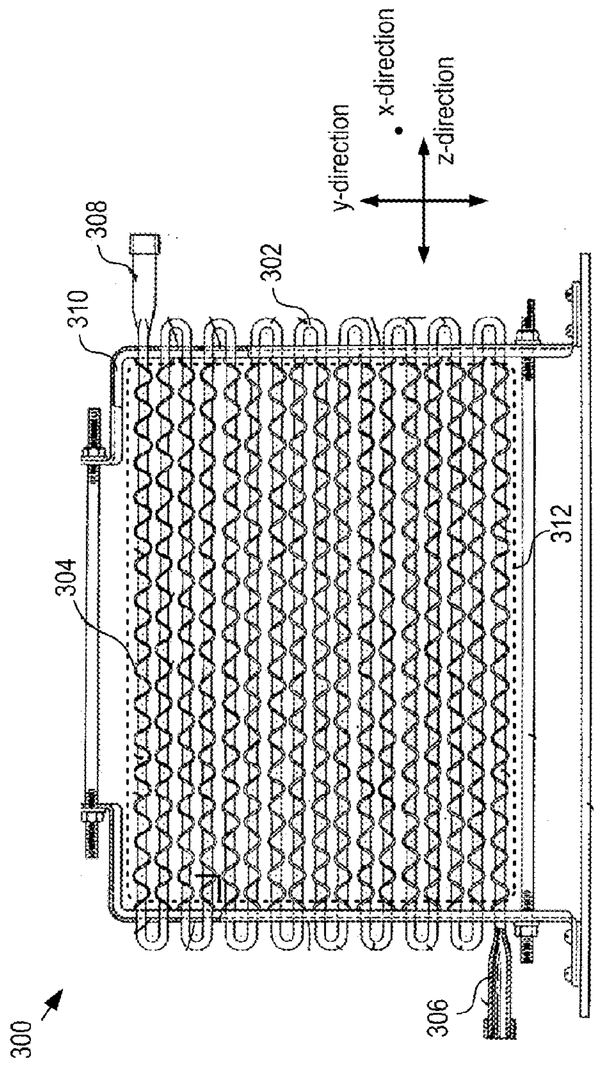


FIG. 3A (Prior Art)

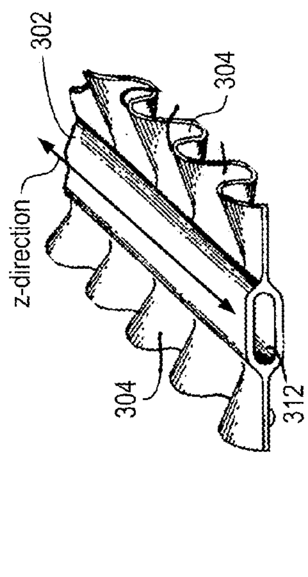


FIG. 3B (Prior Art)

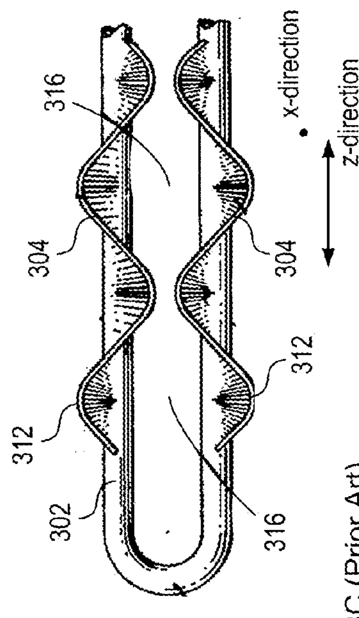


FIG. 3C (Prior Art)

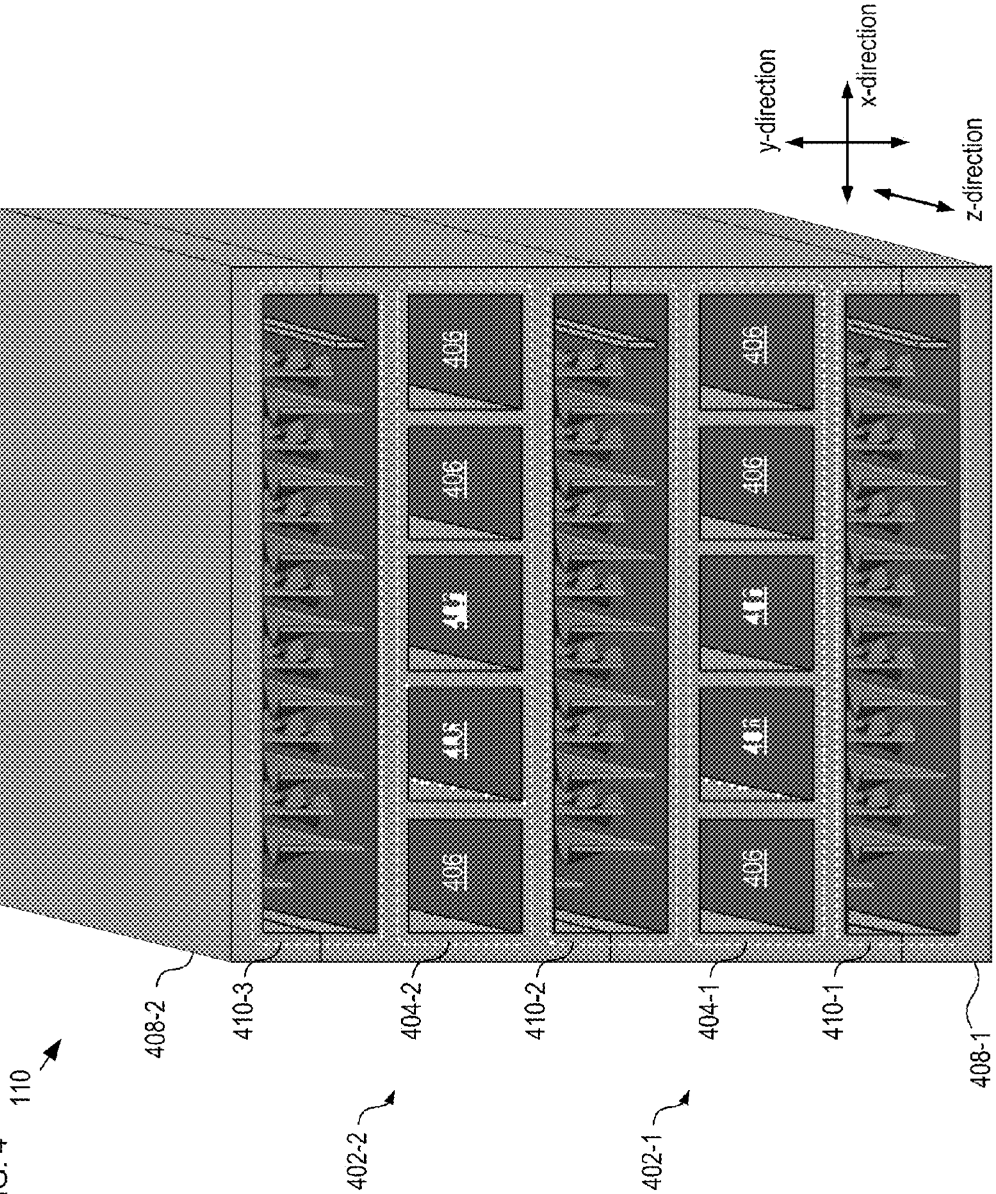

y-direction
x-direction
z-direction
406
406
406
406
406
406
110
408-2
410-3
404-2
410-2
404-1
410-1
408-1
402-2
402-1

FIG. 4

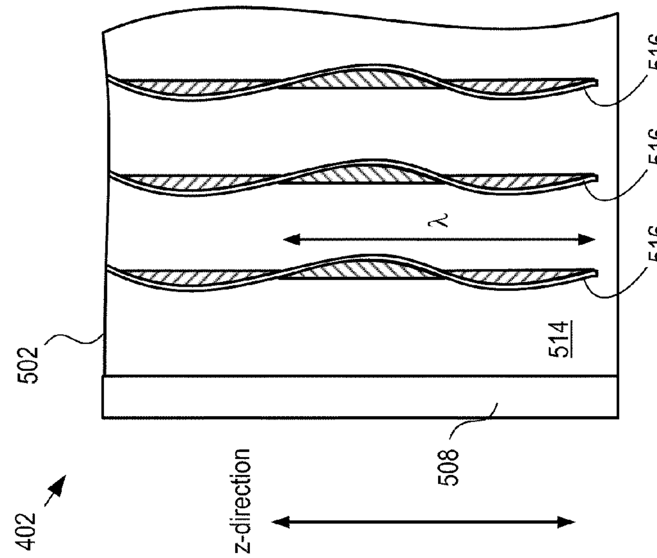


FIG. 5C

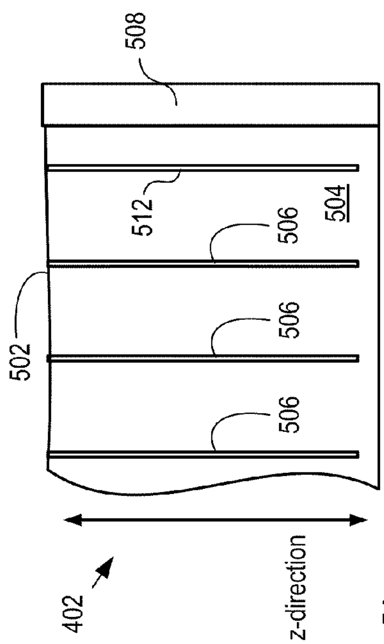


FIG. 5A

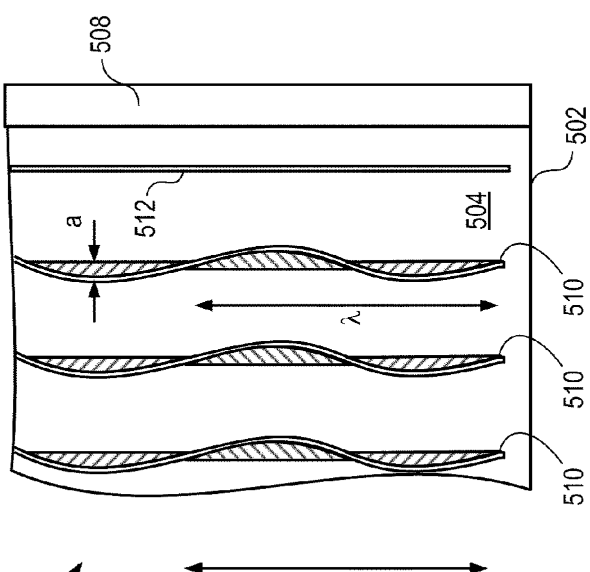


FIG. 5B

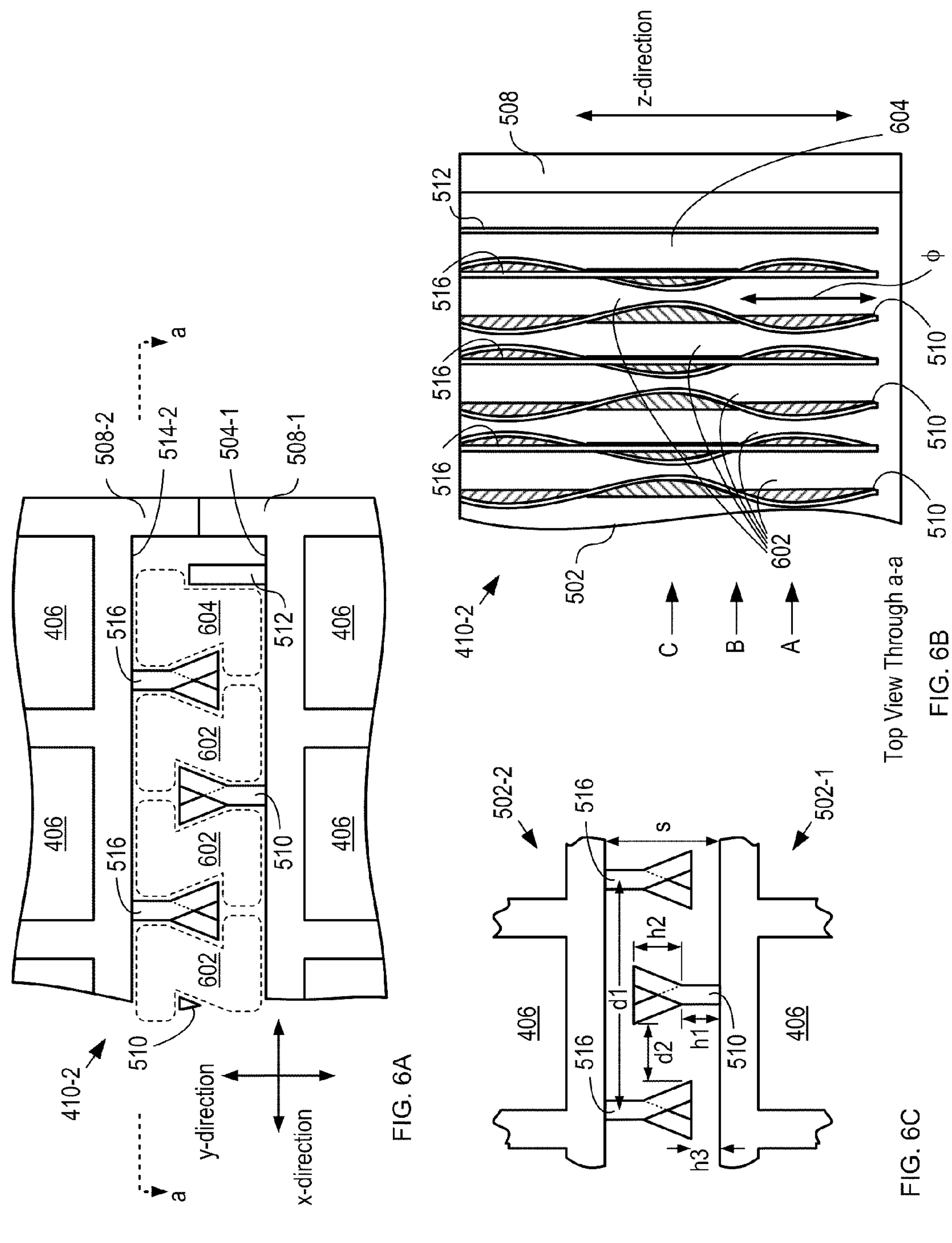


FIG. 6A

FIG. 6B

FIG. 6C

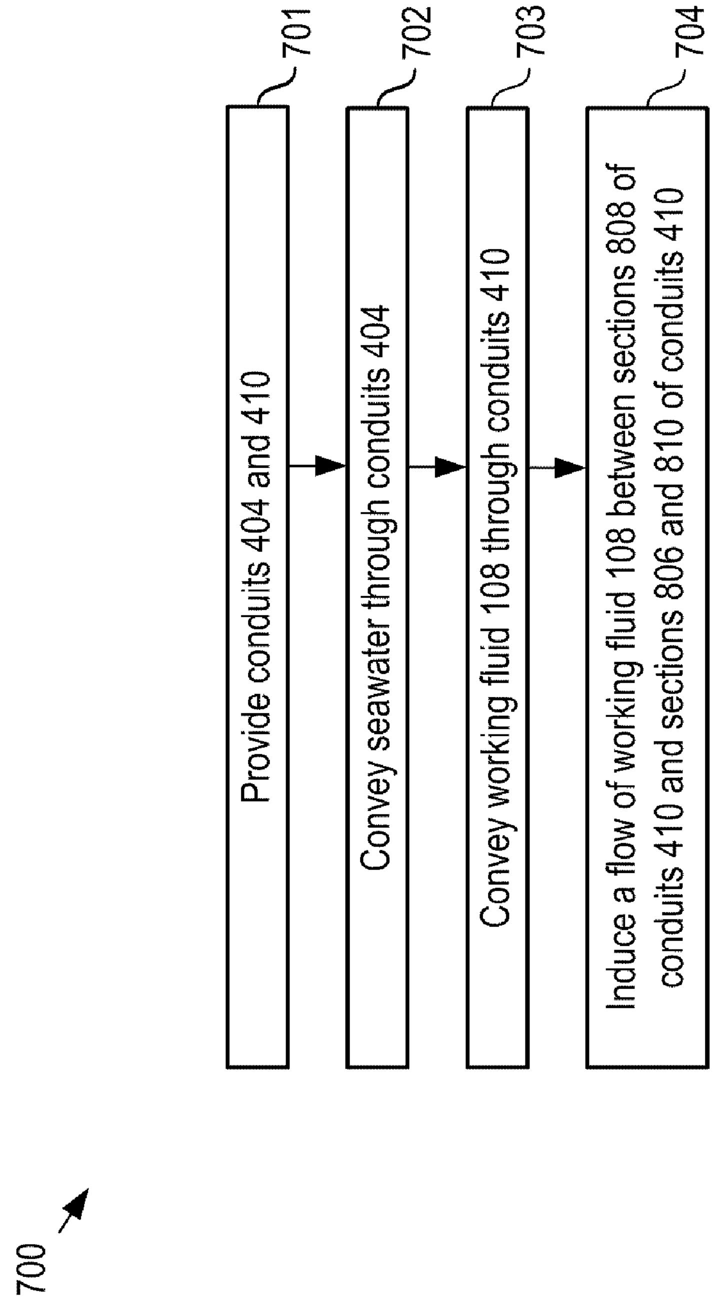

700
Provide conduits 404 and 410
701
Convey seawater through conduits 404
702
Convey working fluid 108 through conduits 410
703
Induce a flow of working fluid 108 between sections 808 of conduits 410 and sections 806 and 810 of conduits 410
704

FIG. 7

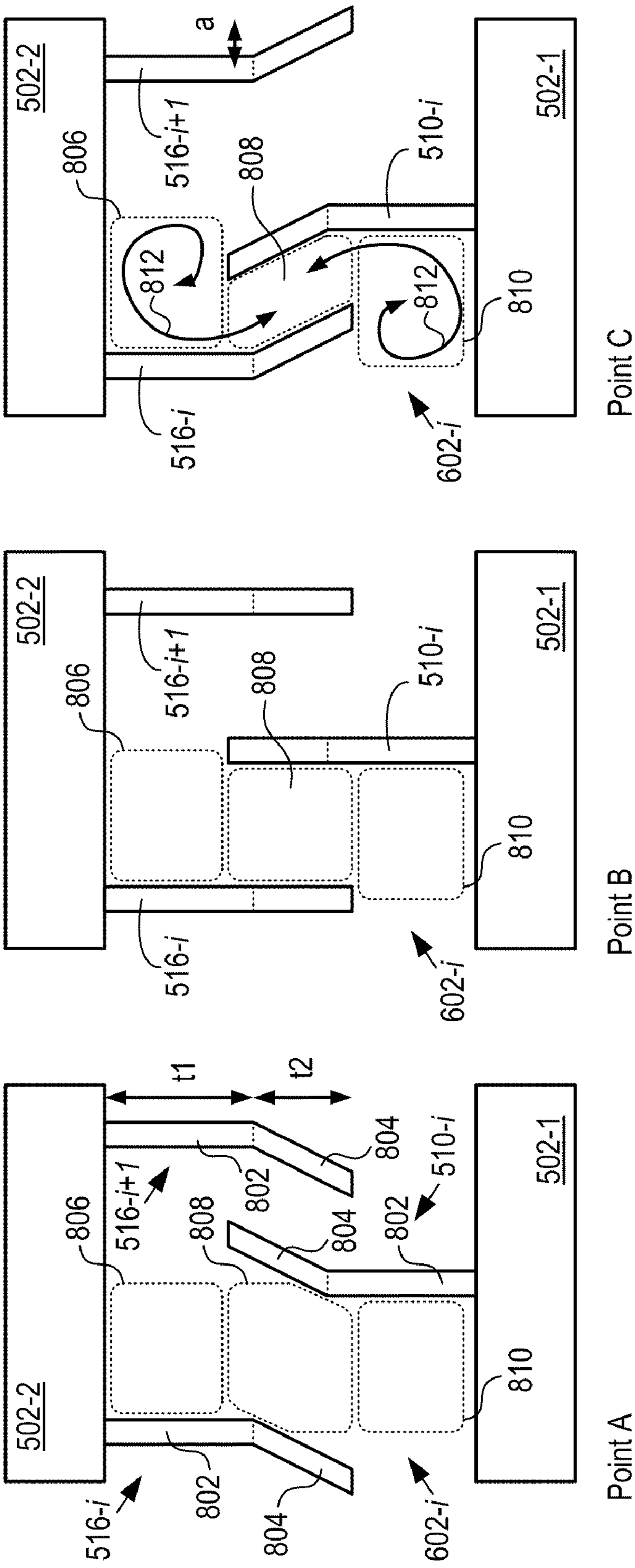


FIG. 8A

FIG. 8B

FIG. 8C

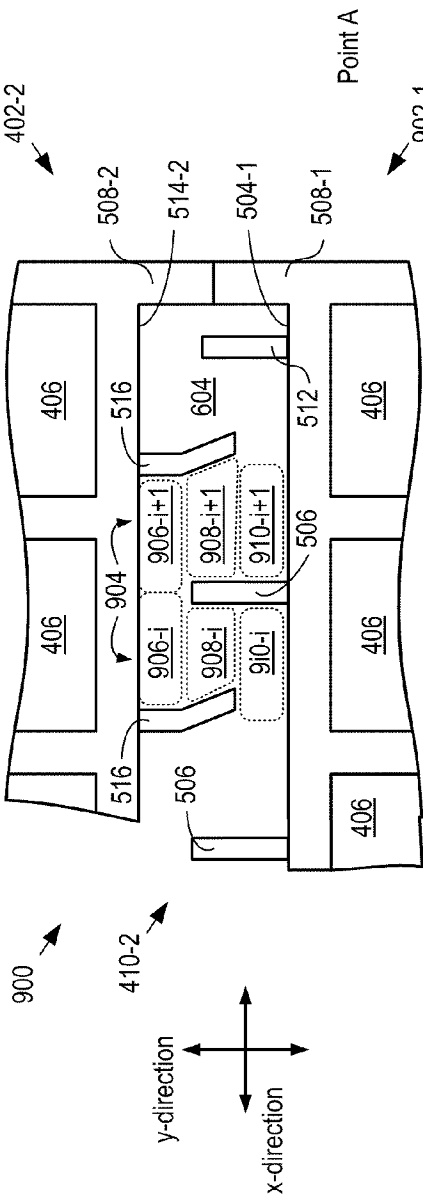

402-2
902-1
Point A
508-2
514-2
504-1
508-1
406
516
604
512
904
906-i+1
908-i+1
910-i+1
506
906-i
908-i
9i0-i
900
410-2
y-direction
x-direction

FIG. 9A

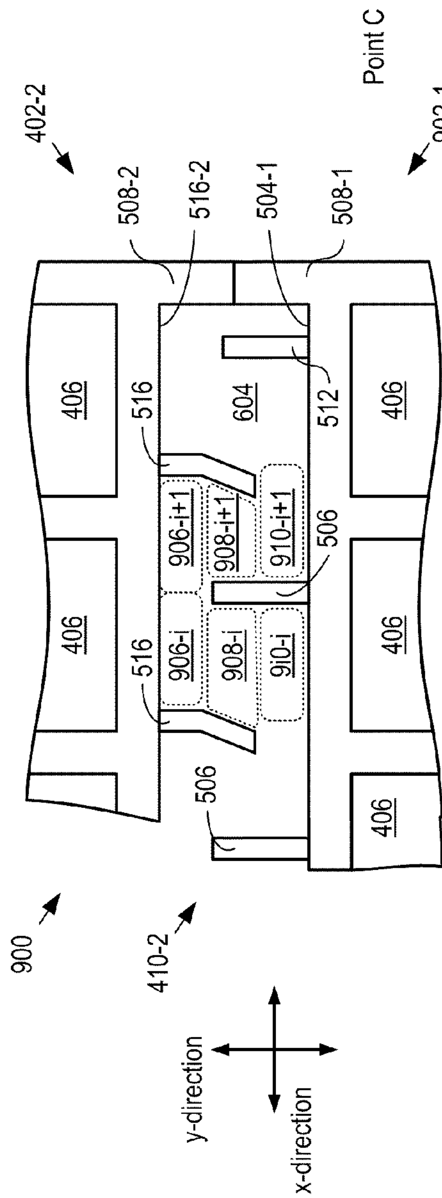

402-2
902-1
Point C
508-2
516-2
504-1
508-1
406
516
604
512
906-i+1
908-i+1
910-i+1
506
906-i
908-i
9i0-i
900
410-2
y-direction
x-direction

FIG. 9B

HEAT EXCHANGER COMPRISING WAVE-SHAPED FINS

FIELD OF THE INVENTION

[0001] The present invention relates to energy conversion in general, and, more particularly, to heat exchangers.

BACKGROUND OF THE INVENTION

[0002] The Earth's oceans are continually heated by the sun and cover nearly 70% of the Earth's surface. The temperature difference between deep and shallow waters contains a vast amount of solar energy that can potentially be harnessed for human use. In fact, it is estimated that the thermal energy contained in the temperature difference between the warm ocean surface waters and deep cold waters within ±10° of the Equator represents a 3 Tera-watt (3×10^{12} W) resource.

[0003] The total energy available is one or two orders of magnitude higher than other ocean-energy options such as wave power, but the small magnitude of the temperature difference makes energy extraction comparatively difficult and expensive, due to low thermal efficiency.

[0004] Ocean thermal energy conversion ("OTEC") is a method for generating electricity which uses the temperature difference that exists between deep and shallow waters to run a heat engine. A heat engine is a thermodynamic device placed between a high temperature reservoir and a low temperature reservoir. As heat flows from one reservoir to the other, the engine converts some of the heat to work. This principle is used in steam turbines and internal combustion engines. Rather than using heat energy from the burning of fuel, OTEC power draws on temperature differences caused by the sun's warming of the ocean surface.

[0005] One heat cycle suitable for OTEC is the Rankine cycle, which uses a low-pressure turbine. Systems may be either closed-cycle or open-cycle. Closed-cycle systems use a fluid with a low boiling point, such as ammonia, to rotate the turbine to generate electricity. Warm surface seawater is pumped through a heat exchanger where the low-boiling-point fluid is vaporized. The expanding vapor turns the turbo-generator. Then, cold, deep seawater—pumped through a second heat exchanger—condenses the vapor back into a liquid, which is then recycled through the system. Open-cycle engines use the water heat source as the working fluid.

[0006] As with any heat engine, the greatest efficiency and power is produced with the largest temperature difference. This temperature difference generally increases with decreasing latitude (i.e., near the equator, in the tropics), but evaporation prevents the surface temperature from exceeding 27° C. Also, the subsurface water rarely falls below 5° C. Historically, the main technical challenge of OTEC was to generate significant amounts of power, efficiently, from this very small temperature ratio. But changes in the efficiency of modern heat exchanger designs enables performance approaching the theoretical maximum efficiency.

[0007] OTEC systems have been shown to be technically viable, but the high capital cost of these systems has thwarted commercialization. Heat exchangers are the second largest contributor to OTEC plant capital cost (the largest is the cost of the offshore moored vessel or platform). The optimization of the enormous heat exchangers that are required for an OTEC plant is therefore of great importance and can have a major impact on the economic viability of OTEC technology.

[0008] The primary function of a heat exchanger is to efficiently transfer thermal energy from one media to another. Heat exchangers of various types are widely used across many applications such as in air conditioners, industrial chemical plants and power generation. For OTEC systems, while there are many conventional heat-exchanger designs that can be considered, there are, as a practical matter, no good choices.

[0009] Almost all heat exchangers can be classified by two types of geometries: Shell and Tube Heat Exchangers or Compact and Extended Surface Heat Exchangers. Of the Compact and Extended Surface Heat Exchanger geometry, two common types are Plate-Fin Heat Exchangers and Extruded-Fin Heat Exchangers. Plate-fin heat exchangers are typically used in the Cryogenics Industry and extruded-fin heat exchangers are most commonly used in the Automotive Industry.

[0010] Conventional shell and tube heat exchangers are widely available for marine use. But the overall heat transfer coefficient, U, that is associated with reasonable pressure drops for OTEC is typically below 2000 W/m^2K. This drives the size and cost for this type of heat exchanger and reduces its economic viability.

[0011] Plate-fin heat exchangers are assembled as an array of stacked aluminum plates with thin corrugated sheets of fins between the plates. The entire array is then brazed together to form a heat exchanger core. Unfortunately, plate-fin heat exchangers are undesirable for many applications because of high material and fabrication costs. Second, brazed joints are poorly suited to applications in which corrosive media are used. In OTEC applications, brazed joints are particularly susceptible to galvanic corrosion when exposed to seawater. Third, plate-fin heat exchangers are typically characterized by low thermal and/or flow efficiency. Such designs suffer from varying amounts of fluid flow resistance, and still do not eliminate manufacturing cost and corrosion issues for very large scale assemblies, however. Fourth, the small passages found in typical plate-fin heat exchangers are prone to biofouling. Fifth, maintenance, such as refitting, repair, and refurbishment, on plate-fin heat exchangers is challenging due to the difficulty of accessing their internal regions.

[0012] Conventional extruded-fin heat exchangers are made from standard aluminum extrusions with thin aluminum sections to increase thermal efficiency. Extruded-fin heat exchangers are typically much less expensive to produce than plate-fin heat exchanger cores because of the elimination of the additional fin component and brazing operations. The trade-off with this design is that they typically have lower thermal performance due to a boundary layer created in the fluid(s) being used. This boundary layer is a physical phenomenon that can only be eliminated by disrupting the fluid flow within the heat exchanger core to create stirring turbulences. This issue has been studied for decades and has been solved for various applications in different ways with limited success.

[0013] Typically, plain fins are mounted on the top and bottom of the extruded channels. These fins are much like the rectangular fins in the plate-fin heat exchanger because they are straight and uninterrupted along the full length of flow. Fins that are straight along the flow length tend to develop fluid boundary layers that are quite thick, which results in lower values of the heat transfer coefficient. A plain fin arrangement will exhibit relatively low pressure drop but have relatively low heat transfer. More complex fin designs that provide disruptions to fluid flow can improve heat transfer; however, complex fin designs suffer from a higher pressure drop through the heat exchanger.

[0014] With today's growing need for energy, using a renewable constant source is a desirable solution. As a consequence, there is a renewed interest in OTEC power plants. But development of an OTEC heat exchanger that accommodates high flow rates while minimizing pumping parasitic losses and offering long life in the ocean environment remains elusive.

SUMMARY OF THE INVENTION

[0015] The present invention provides a heat exchanger having higher heat transfer efficiency than heat exchangers known in the prior art. Heat exchangers in accordance with the present invention are particularly well-suited for use in OTEC systems.

[0016] An embodiment of the present invention comprises first conduits for conveying a first fluid through the heat exchanger and second conduits for conveying a second fluid through the heat exchanger, wherein the first fluid and second fluid are thermally coupled by the heat exchanger. The first conduits include flow passages that induce turbulence in the flowing first fluid without significantly increasing fluid back pressure. The turbulence is induced by wave-shaped fins that project into each first conduit to form a plurality of flow passages. The wave-shaped fins are continuous along the direction of fluid flow. Adjacent pairs of wave-shaped fins define three sections in each flow passage: first and second sections that are interposed by a third section. The wave-shape of the fins results in a continuous variation of the cross-sectional area of the third section along the direction of fluid flow. As this cross-sectional area changes, the first fluid flowing through the flow passage is forced to exchange between the third section and each of the two remaining sections of the flow passage. This exchange of fluid between the three sections induces a swirl, or vortex, flow in the first and second sections, which increases the overall convection heat transfer in the flow passage.

[0017] The fins are arranged within each first conduit such alternating fins project into the first conduit from opposite surfaces and so that the wave shapes of adjacent fin pairs are offset by a phase difference. This phase difference leads to a continuously periodic change on the cross-sectional area of the third section along the length of the flow passage. As the cross-sectional area shrinks, first fluid is "squeezed" from each third section into the first and second sections of each flow passage. As the cross-sectional area of the third section increases, first fluid is drawn back into the third section from the other two sections. Further, the wave-shape of the fins defines a shape of the third section that induces the first fluid to swirl as it enters and exits the first and second sections. This swirl flow creates turbulence that enhances heat transfer between the first fluid and walls of the flow passages.

[0018] It is a further aspect of the invention that the first conduits avoid inducing a significant fluidic back pressure while conveying the first fluid through the heat exchanger. Increased back pressure of the fluid is mitigated by the fact that the overall cross-sectional area of the first conduits remains the same even while the cross-sectional areas of individual flow passages within it change. The consistency of overall cross-sectional area of the conduits results from the complimentary nature of adjacent flow passages within them. Specifically, as the cross-sectional area of a first flow passage is shrinking, the cross-sectional area of its adjacent flow passages is increasing by a commensurate amount. As a result, the sum of the cross-sectional areas of all flow passages in a given first conduit remains constant.

[0019] An embodiment of the present invention comprises a heat exchanger that thermally couples a first fluid and a second fluid, wherein the heat exchanger comprises: (1) a first plate comprising a first material that is thermally conductive, wherein the first plate comprises; (i) a first conduit for conveying the first fluid along a first direction, wherein the first conduit comprises at least one first channel; and (ii) a first plurality of first fins, wherein each first fin is continuous along the first direction, and wherein each first fin comprises a first fin portion that has a first periodic shape having a first periodicity along the first direction; and (2) a second plate comprising the first material, wherein the second plate comprises; (i) a second conduit for conveying the first fluid along the first direction, wherein the second conduit comprises at least one second channel; a (ii) second plurality of second fins, wherein each second fin is continuous along the first direction, and wherein each second fin comprises a second fin portion that has the first periodic shape having the first periodicity along the first direction; wherein the first plurality of first fins and the second plurality of second fins collectively define a plurality of passages for conveying the second fluid along the first direction, and wherein each of the plurality of passages comprises a first section, second section, and third section; and wherein the cross-sectional area of the first section varies along the first direction based on the first periodicity, and wherein the variation of the cross-sectional area of the first section induces flow of the second fluid between the first section and each of the second section and third section.

BRIEF DESCRIPTION OF THE DRAWINGS

[0020] FIG. **1** depicts a schematic diagram of an OTEC power generation system in accordance with an illustrative embodiment of the present invention.

[0021] FIG. **2**A depicts a schematic drawing of a first heat exchanger in accordance with the prior art.

[0022] FIG. **2**B depicts a perspective view of a plate **202**.

[0023] FIG. **3**A depicts a schematic drawing of a second heat exchanger in accordance with the prior art.

[0024] FIG. **3**B depicts a perspective view of a section of tube **302**.

[0025] FIG. **3**C depicts a schematic of a portion of heat exchanger **300**.

[0026] FIG. **4** depicts a perspective view of a heat exchanger in accordance with the illustrative embodiment of the present invention.

[0027] FIG. **5**A depicts a top view of a portion of plate **402** prior to the formation of a wave-shape on its fins in accordance with the illustrative embodiment of the present invention.

[0028] FIG. **5**B depicts a top view of a portion of plate **402**, after the formation of a wave-shape on its fins.

[0029] FIG. **5**C depicts a bottom view of a portion of plate **402**, after the formation of a wave-shape on its fins.

[0030] FIG. **6**A depicts a representational cross-sectional view of conduit **410-2** in accordance with the illustrative embodiment of the present invention.

[0031] FIG. **6**B depicts a top view, through line a-a of FIG. **6**A, of conduit **410-2** in accordance with the illustrative embodiment of the present invention.

[0032] FIG. **6**C depicts a detailed cross-sectional view of a portion of conduit **410-2**.

[0033] FIG. **7** provides operations of a method for thermally coupling a first fluid and a second fluid in accordance with the illustrative embodiment of the present invention.

[0034] FIG. **8**A depicts a flow passage **602**-*i* within in conduit **410-2**, at point A depicted in FIG. **6**B.

[0035] FIG. **8**B depicts a flow passage **602**-*i* within in conduit **410-2**, at point B depicted in FIG. **6**B.

[0036] FIG. **8**C depicts a flow passage **602**-*i* within in conduit **410-2**, at point C depicted in FIG. **6**B.

[0037] FIG. **9**A depicts a representational cross-sectional view through a portion of a heat exchanger at a first point along the direction of fluid flow, in accordance with an alternative embodiment of the present invention.

[0038] FIG. **9**B depicts a representational cross-sectional view through a portion of a heat exchanger at a second point along the direction of fluid flow, in accordance with the alternative embodiment of the present invention.

DETAILED DESCRIPTION

[0039] FIG. **1** depicts a schematic diagram of an OTEC power generation system in accordance with an illustrative embodiment of the present invention. OTEC system **100** comprises turbogenerator **104**, closed-loop conduit **106**, heat exchanger **110-1**, heat exchanger **110-2**, pumps **114**, **116**, and **124**, and conduits **120**, **122**, **128**, and **130**. OTEC system **100** is deployed in water body **136** wherein a suitable temperature difference exists between water near the surface and water located at a deep level of water body **136**.

[0040] Turbo-generator **104** is a conventional turbine-driven generator. Turbogenerator **104** is mounted on floating platform **102**, which is a conventional floating energy-plant platform. Platform **102** is anchored to the ocean floor by mooring line **132** and anchor **134**, which is embedded in the ocean floor. In some instances, platform **102** is not anchored to the ocean floor but is allowed to drift. Such a system is sometimes referred to as a "grazing plant."

[0041] In typical operation, pump **114** pumps a primary fluid (i.e., working fluid **108**), in liquid form, through closed-loop conduit **106** to heat exchanger **110-1**. Ammonia is often used as working fluid **108** in OTEC systems; however, it will be clear to one skilled in the art that any fluid that evaporates at the temperature of the water in surface region **118** and condenses at the temperature of the water in deep water region **126** is suitable for use as working fluid **108** (subject to material compatibility requirements).

[0042] Heat exchanger **110-1** and **110-2** are configured for operation as an evaporator and condenser, respectively. One skilled in the art will recognize that the operation of a heat exchanger as evaporator or condenser is dependent upon the manner in which it is configured within system **100**. Heat exchanger **110** is described in detail below and with respect to FIGS. **4** through **7**C.

[0043] In order to enable its operation as an evaporator, pump **116** draws warm secondary fluid (i.e., seawater from surface region **118**) into heat exchanger **110-1** via conduit **120**. At heat exchanger **110-1** heat from the warm water is absorbed by working fluid **108**, which induces working fluid **108** to vaporize. After passing through heat exchanger **110-1**, the warm water is ejected back into water body **136** via conduit **122**. In a typical OTEC deployment, the water is surface region **118** is at a substantially constant temperature of approximately 25 degrees centigrade (subject to weather and sunlight conditions).

[0044] The expanding working fluid **108** vapor is forced through turbogenerator **104**, thereby driving the turbogenerator to generate electrical energy. The generated electrical energy is provided on output cable **112**. Once it has passed through turbogenerator **104**, the vaporized working fluid enters heat exchanger **110-2**.

[0045] At heat exchanger **110-2**, pump **124** draws cold secondary fluid (i.e., seawater from deep water region **126**) into heat exchanger **110-2** via conduit **128**. The cold water travels through heat exchanger **110-2** where it absorbs heat from the vaporized working fluid. As a result, working fluid **108** condenses back into liquid form. After passing through heat exchanger **110-2**, the cold water is ejected into water body **136** via conduit **130**. Typically deep water region **126** is 1000+ meters below the surface of water body **136**, at which depth water is at a substantially constant temperature of a few degrees centigrade.

[0046] Pump **114** pumps the condensed working fluid **108** back into heat exchanger **110-1** where it is again vaporized; thereby continuing the Rankine cycle that drives turbogenerator **104**.

[0047] FIG. **2**A depicts a schematic drawing of a first heat exchanger in accordance with the prior art. Heat exchanger **200** comprises a plurality of plates **202**, and housing **208**. Heat exchanger **200** is in accordance with a heat exchanger described by T. F. Brise in U.K. Patent Application GB2424265, which was published Sep. 9, 2006, and which is incorporated by reference herein.

[0048] Each plate **202** is an extruded body that comprises a plurality of channels **204**, which convey a first fluid along the z-direction as shown. In some cases, each of channels **204** comprises projections directed inward from the surface of the channel. These projections increase the heat transference between the first fluid and the plate material.

[0049] Plates **202** are stacked and welded together to form heat exchanger core **210**. Adjacent plates in heat exchanger core **210** collectively define conduits **212** for conveying a second fluid along the z-direction. Each plate **202** further comprises a plurality of fin segments **206**, which depend from the upper and lower surfaces of the body of plate **202** and project into conduits **212**.

[0050] Each fin segment **206** is a discrete element that is formed from fins that are initially continuous along the z-direction. To form fin segments **206**, these initially continuous fins are "segmented and the fin segments twisted so that at least the upper parts of the fin segments are at an angle" with respect to the z-direction (i.e., the longitudinal axis of the tube). According to Brise, "Segmenting the fins and twisting the fin segments creates lateral flow paths across the surface of the tube along which coolant can flow."

[0051] Heat exchanger core **210** is located within heat exchanger **200** via housing **208**. Housing **208** locates heat exchanger core **210** by means of seats **214**, which receive the uppermost and lowermost fin segments that project from the heat exchanger core. Once heat exchanger core **210** is positioned within sleeve **208**, weld joints are formed at seats **214** to firmly fix the heat exchanger core in place.

[0052] FIG. **2**B depicts a perspective view of a plate **202**. As is readily seen from the figure, the top and bottom outer surfaces of plate **202** are covered by rows of individual fin segments **206**. These fin segments are arranged along the z-direction in rows.

[0053] As the second fluid flows along the z-direction, the discontinuities between fin segments **206** create turbulence that enhances heat transfer between the second fluid and plate **202**.

[0054] Unfortunately, heat exchangers such as heat exchanger **200** have several disadvantages. First, in many cases the formation of fin segments **206** shears off much of the base of each fin segment. As a result, fin segments **206** are highly susceptible to fracture during use. Fractured fins are likely to lodge in conduits **212** and restrict the flow of the second fluid through the heat exchanger.

[0055] Second, when the plates **202** are stacked together to form conduits **212**, fin segments **206** are interlocked such that they nearly completely obstruct the flow path for the second fluid. As a result, fin segments **206** cause a large pressure-drop through conduits **212**.

[0056] Third, it is well-known to those skilled in the art that discontinuities, such as those between fin segments **206**, significantly increase back pressure.

[0057] Fourth, it is well known that discontinuities in a flow path are prone to impurity build-up, which leads to fouling during use. For example, the use of serrated fins, such as those included in heat exchanger **200**, is discouraged in many heat exchanger design handbooks, such as "*Heat Exchanger Design Handbook*," written by T. Kuppan. In general, it is difficult and expensive to clean fouled fluid conduits of a heat exchanger, and nearly impossible for compact heat exchangers.

[0058] FIG. **3**A depicts a schematic drawing of a second heat exchanger in accordance with the prior art. Heat exchanger **300** comprises tube **302**, ruffled fins **304**, inlet **306**, outlet **308**, and frame **310**. Heat exchanger **300** is in accordance with a heat exchanger described by W. E. McCullough in U.S. Pat. No. 2,347,957, which was issued May 2, 1944, and which is incorporated by reference herein.

[0059] Tube **302** is a tubular-shaped conduit for conveying a first fluid along the z-direction as shown. Tube **302** is formed by flattening larger extruded tubing stock to form a central tube (i.e., tube **302**) having attached flat projections.

[0060] These flat projections are run through rollers, or other suitable apparatus, to ruffle the projections into ruffled fins **304**. As a result, ruffled fins **304** are wavy projections that extend laterally from tube **302**.

[0061] FIG. **3**B depicts a perspective view of a section of tube **302**. Tube **302** comprises conduit **312**, which conveys the first fluid along the z-direction. To form heat exchanger core **312**, tube **302** is bent multiple times to form a continuous serpentine-shaped coil. In order to facilitate bending tube **302**, portions of ruffled fins **304** are cut away near the regions of the bends. Heat exchanger core **312** is mounted into frame **310** such that each of the plurality of straight runs of tube **302** is oriented along the z-direction, and ruffled fins **304** project along the x-direction, as shown.

[0062] FIG. **3**C depicts a schematic of a portion of heat exchanger **300**. As evidenced by FIG. **3**C, the bends of tube **302** are judiciously placed so that the crests **312** of ruffled fins **304** collectively define funnel-shaped "air scoops **314**" that collectively form a plurality of "Venturi throats." In other words, tube **302** is bent such that the wave of ruffled fins **304** of a first straight section is 180 degrees out of phase with the waves of adjacent straight sections. For example, referring to FIGS. 7 and 11 of the patent, McCullough discloses that for adjacent straight runs, the crests of the ruffled fins **304** are located "extending away from each other and beyond the width of the tube section 11. Due to this construction each of these ruffles acts as an air scoop and between them is an opening having a general funnel shape which narrows down to a flat opening 24 between the runs of tube section . . . " And, "The cross section through these ruffles (FIG. 16) shows that the air stream first impinges forcefully against the fins 22 and 23, then rushes with increased velocity through the Venturi throat, 24, and expands to make contact with the ruffles 25 and 28." See e.g., McCullough, page 2, second column, lines 3-20.

[0063] It is clear from this description that heat exchanger **300** is suitable only for exchanging heat between the first fluid and air. Further, heat exchanger **300** conveys the first fluid along the z-direction and the second fluid (i.e., air) along an orthogonal direction (i.e., the x-direction, as shown). As a result, the interaction length of heat two fluids in heat exchanger **300** is extremely short, which limits the efficiency of heat transfer. Still further, the 180 degree bends in the flow-path of the first fluid lead to a large pressure drop that reduces the efficiency of the heat exchanger and limits the size of such heat exchangers.

[0064] FIG. **4** depicts a perspective view of a heat exchanger in accordance with the illustrative embodiment of the present invention. Heat exchanger **110** comprises plates **402-1** and **402-2** and end plates **408-1** and **408-2**. Although the illustrative embodiment comprises two plates **402**, it will be clear to one skilled in the art, after reading this specification, how to specify, make, and use alternative embodiments of the present invention that comprise any practical number of plates **402**. FIG. **4** is described with continuing reference to FIG. **1**.

[0065] Heat exchanger **110** thermally couples working fluid **108** and seawater taken from a region of the water body **136**. For example, heat exchanger **110-1** acts as an evaporator that heats working fluid **108** by transferring heat from warm seawater of surface region **118**. In similar fashion, heat exchanger **110-2** acts as a condenser that cools vaporized working fluid **108** by thermally coupling it with cold seawater of deep water region **126**.

[0066] Each of plates **402-1** and **402-2** (collectively referred to as plates **402**) conveys seawater along the z-direction via conduits **404-1** and **404-2**, respectively. Each of conduits **404-1** and **404-2** comprises a plurality of channels **406**. In the illustrative embodiment, plates **402** are extruded plates of aluminum alloy. In some embodiments, plates **402** are made of another suitable material, such as aluminum, composite materials, graphite, graphite foam, and the like. It is preferable that plates **402** be made of a material that is substantially corrosion-free when exposed to seawater and/or common working fluids. Channels **406** are fluidically coupled to closed-loop conduit **106** via manifolds (not shown for clarity).

[0067] End plates **408-1** and **408-2** (collectively referred to as end plates **408**) mate with plates **402-1** and **402-2**, respectively. Plates **402** and end plates **408** collectively define a heat exchanger core that conveys both working fluid **108** and seawater along the z-direction while also fluidically isolating the fluids from one another.

[0068] Plates **402** are stacked together with end plates **408** to collectively define conduits **410-1**, **410-2**, and **410-3** (collectively referred to as conduits **410**). Each of conduits **410**

conveys working fluid **108** through heat exchanger **110** along the z-direction. Conduits **410** are described in more detail below and with respect to FIG. **6**.

[0069] It should be noted that although the illustrative embodiment conveys seawater through conduits **404** and working fluid through conduits **410**, in some embodiments seawater is conveyed through conduits **410** and working fluid is conveyed through conduits **404**. In some alternative embodiments, a secondary fluid other than seawater is conveyed through the heat exchanger.

[0070] Plates **402** and end plates **408** are joined with a substantially galvanic corrosion-free joint, such as a friction-stir welding joint. In some embodiments, plates **402** and end plates **408** are joined using a different joining technology.

[0071] FIG. **5**A depicts a top view of a portion of plate **402** prior to the formation of a wave-shape on its fins in accordance with the illustrative embodiment of the present invention. Plate **402** comprises body **502**, straight fins **506**, and partition **512**. Plate **402** is representative of each of plates **402-1** and **402-2**.

[0072] Body **502** is the central portion of plate **402**, which comprises conduit **404**. On each end (left and right ends, as depicted in FIG. **4**), body **502** comprises a sidewall **508**. Sidewall **508** projects above and below body **502** a distance necessary to provide a desired amount of clearance along the y-direction above fins **510** when plate **402** is joined with another plate **402** or an end plate **408**. Sidewall **508** also provides a surface for joining plate **402** with another plate **402** or an end plate **408**.

[0073] Straight fins **506** are projections that project from surface **504** of plate **402**. Straight fins **506** are normal to surface **504**.

[0074] Partition **512** is a straight wall that forms a sidewall for an end flow passage of conduit **410**, as described below and with respect to FIGS. **6**A and **6**B.

[0075] Straight fins **506**, sidewall **508**, and body **502** are contiguous portions of a single extrusion that forms plate **402**.

[0076] FIG. **5**B depicts a top view of a portion of plate **402**, after the formation of a wave-shape on its fins.

[0077] Straight fins **506** are formed into fins **510** through a conventional forming process, such as stamping, rolling, crimping, or hot forming. For example, a pair of stamping dies, having a desired wave shape, can be used to press this desired wave shape into straight fins **506**. Alternatively, rollers or cams having a suitable forming surface can be rolled along the sides of straight fins **506** to deform them into fins **510**. Such a rolling process offers an ability to form fins **510** in a continuous manner. Further parallel sets of rollers or cams enable the formation of a plurality of fins **510** at the same time.

[0078] Once straight fins **506** have been shaped, fins **510** are characterized by the desired wave shape along the z-direction, wherein the wave shape is that of a sinusoid having amplitude a and wavelength λ. In some embodiments, fins **510** are characterized by a periodic shape other than a sinusoid, such as triangular or chevron-shaped patterns.

[0079] FIG. **5**C depicts a bottom view of a portion of plate **402**, after the formation of a wave-shape on its fins. Plate **402** further comprises fins **516**, which project from bottom surface **514** of body **502**. Fins **516** are analogous to fins **510**; however, fins **516** are formed such that, when plates **402-1** is flipped over and plates **402-1** and **402-2** are joined, the wave shapes of fins **510** and **514** nest such that there is a relative phase, ϕ, between them along the z-direction. In the illustrative embodiment, ϕ has a magnitude of approximately 180 degrees. In some embodiments, ϕ has a magnitude of approximately 90 degrees. In some embodiments, ϕ can have any magnitude within the range of 0 degrees to 360 degrees. The relationship of ϕ to the performance of heat exchanger **110** is discussed in more detail below and with respect to FIGS. **7** and **8**A-C.

[0080] In similar fashion to conduit **410-2**, conduit **410-1** is collectively defined by plate **402-1** and end plate **408-1**. The top surface of end plate **408-1** is analogous to surface **504**. Further, end plate **408-1** comprises fins **510** that project into conduit **410-1** from its top surface. The fins **510** of end plate **408-1** nest with fins **516** that project from surface **514** of plate **402-1** to form a plurality of flow passages in conduit **410-1**. Still further, the top surface of end plate **408-2** is analogous to surface **514** and end plate **408-2** comprises fins **516** that project into conduit **410-3** from its bottom surface. The fins **516** of end plate **408-2** nest with fins **510** that project from surface **504** of plate **402-2** to form a plurality of flow passages in conduit **410-3**. As a result, conduits **410-1** and **410-3** are analogous to conduit **410-2**.

[0081] FIGS. **6**A and **6**B depict a representational cross-sectional and top view through line a-a, respectively, of conduit **410-2** in accordance with the illustrative embodiment of the present invention. When plates **402-1** and **402-2** are joined, fins **510** project from surface **504-1** into conduit **410-2**. Fins **510** nest with fins **516** that project from surface **514-2** into conduit **410-2**. As a result, fins **510** and **516** collectively define flow passages **602** within conduit **410-2**. In addition, flow passage **604** is defined by the right-most (as depicted in FIG. **6**A) fin **516** and partition **512**, which projects from surface **504-1**. In similar fashion, although not shown for clarity, left-most fin **510** and partition **512** that projects from surface **514-2** collectively define a second flow passage **604**.

[0082] FIG. **6**C depicts a detailed cross-sectional view of a portion of conduit **410-2**. Each of fins **510** and **516** has substantially the same dimensions. While the specific dimensions of fins **510** and **516** are dependent upon the desired characteristics of heat exchanger **110**, the dimensional relationships of fins **510** and **516** influence the manner in which working fluid **108** flows through conduit **410-2**.

[0083] Each of fins **510** and **516** is characterized by a wave starting height of h**1**. The portion of each fin that comprises its wave shape is denoted as h**2**. In the illustrative embodiment, h**2** is greater than h**1** although it will be clear to one skilled in the art, after reading this specification, how to specify, make, and use alternative embodiments of the present invention wherein h**2** is not larger than h**1**.

[0084] Fin-to-base clearance h3 denotes the clearance between the fins and surfaces **504** and **514** of conduit **410-2**. The fin-to-base clearance is determined by the difference between the separation, s, between surfaces **504** and **514** and the sum of h**1** and h**2**.

[0085] Each of the pluralities of fins **510** and **514** are characterized by the same fin-periodicity, d**1**. The wave-to-wave clearance, d**2**, is based on the fin-periodicity, d**1**, and the wave amplitude, a.

[0086] Judicious selection of h**1**, h**2**, h3, s, d**1**, d**2**, a, λ, and ϕ enables the design of conduits **410** that have high heat transfer efficiency.

[0087] It should be noted that prior art attempts to incorporate wavy channels (e.g., wavy fins) into conventional heat exchangers have utilized channels whose cross-sectional area remains constant along the direction of fluid flow. These channels are wavy in that they periodically abruptly change the flow direction of the fluid. Examples of such prior-art heat exchangers are found in "Forced Heat Convection in Wavy Fin Channel," by Yang, et al., published in *The Journal of Thermal Science and Technology*, Vol. 3, pp. 342-354, (2008). A boundary layer, which forms in the flowing fluid, will separate and reattach at alternative periods of the wave thereby resulting in vortices around the flow channels. While these vortices improve the heat transfer coefficient of the heat exchanger, such heat exchangers are subject to a significant pressure drop (i.e., fluidic back pressure) through the length of the flow channels.

[0088] FIG. **7** provides operations of a method for thermally coupling a first fluid and a second fluid in accordance with the illustrative embodiment of the present invention. Method **700** begins with operation **701**, wherein heat exchanger **110** is provided. FIG. **7** is described herein with continuing reference to FIGS. **1**, and **4**-**6**C and further reference to FIGS. **8**A-C.

[0089] FIGS. **8**A-C depict a flow passage **602**-*i* within in conduit **410-2** at different points along the z-direction. The shape of flow passage **602**-*i* is based on the relative orientations of shoulders **802** and wave portion **804** of fins **516**-*i* and **510**-*i*, respectively. Flow passage **602**-*i* comprises sections **806**, **808**, and **810**. The shape and cross-sectional area of section **808** changes along the length of conduit **410-2** and is defined by the configuration of fins **510**-*i* and **516**-*i* at each point along the z-axis.

[0090] At operation **702**, conduits **404** are fluidically coupled to a region of water body **136**. As a result, seawater is conveyed through heat exchanger **110** via conduits **404**.

[0091] At operation **703**, conduits **410** are fluidically coupled to working fluid **108**. As a result, working fluid **108** is conveyed through heat exchanger **110** via conduits **410**. Both the seawater and working fluid are conveyed along the z-direction, as described above; therefore, the interaction length between them is substantially the length of heat exchanger **110**. A long interaction length enables a highly efficient transfer of thermal energy between the seawater and working fluid.

[0092] At operation **704**, changes in the cross-sectional areas of sections **808** induce flow **812** of working fluid. These changes induce the flow of working fluid **108** between section **808** and sections **806** and **810** of each flow passage in the conduit.

[0093] FIG. **8**A depicts flow passage **602**-*i* at point A, when wave portions **804** of fins **510**-*i* and **516**-*i* are separated by a maximum distance. At this point along conduit **410-2**, section **808** of flow passage **602**-*i* has the largest cross-sectional area.

[0094] FIG. **8**B depicts flow passage **602**-*i* at point B, when fins **510**-*i* and **516**-*i* are straight. At this point along conduit **410-2**, section **808** of flow passage **602**-*i* has a smaller cross-sectional area than that depicted in FIG. **8**A.

[0095] FIG. **8**C depicts flow passage **602**-*i* at point C, when fins **510**-*i* and **516**-*i* are separated by a minimum distance. At this point along conduit **410-2**, section **808** of flow passage **602**-*i* has the smallest cross-sectional area.

[0096] As working fluid **108** flows through conduit **410-2** from point A to point B to point C, the working fluid is squeezed from section **808** into each of sections **806** and **810**. This creates turbulence in and around sections **806** and **810** due to flow **812**. In some embodiments, flow **812** is a swirl flow. In some embodiments, flow **812** is a vortex flow. The turbulence in these sections significantly increases the thermal transfer efficiency of heat exchanger **110**.

[0097] It should be noted that the combined cross-sectional area of passages **602** and **604** remains the same throughout the length of heat exchanger **110**. This is readily seen by the fact that as the cross-section of section **808**-*i* becomes smaller, its reduction in size is offset by a commensurate increase in the cross-sectional area of **808**-*i*+1. The shift in cross-sectional area between neighboring flow passages is responsible for advantageously inducing turbulent flow in sections **806** and **810** of each flow passage. The conservation of the overall cross-sectional area of conduits **410** affords embodiments of the present invention additional advantage since the improved thermal transfer efficiency accrues without incurring an increase in flow resistance through the conduits. Still further, the lack of discontinuities in flow passages **602** and **604** enables heat exchanger **110** to avoid fluid flow interruptions inherent in prior-art heat exchangers, such as heat exchanger **200**. Such fluid flow interruptions greatly increase fluid back pressure in these prior-art systems.

[0098] FIGS. **9**A and **9**B depict representational cross-sectional views through a portion of a heat exchanger, at two points along the direction of fluid flow, in accordance with an alternative embodiment of the present invention. Heat exchanger **900** is substantially identical to heat exchanger **400**; however, in heat exchanger **900**, plate **402-1** is replaced by plate **902-1**.

[0099] Plate **902** is analogous to plate **402** prior to the formation of a wave-shape on its fins. In other words, plate **902** comprises fins **506**, rather than fins **510**, and a top view of plate **902** would be analogous to the top view of plate **402** depicted in FIG. **5**A.

[0100] When plates **902-1** and **402-2** are joined as shown in FIGS. **9**A and **9**B, fins **506** project from surface **504-1** into conduit **410-2**. Fins **506** nest with fins **516** that project from surface **514-2** into conduit **410-2**. As a result, fins **506** and **516** collectively define flow passages **904** within conduit **410-2**.

[0101] Each of flow passages **904** comprises sections **906** and **910**, which are interposed by section **908**. Sections **904**, **906**, and **910** are analogous to sections **806**, **810**, and **812**, described above and with respect to FIGS. **8**A-C.

[0102] In FIG. **9**A, the cross-sectional view is taken at a point analogous to point A of the heat exchanger, as described above and with respect to FIG. **6**B. In other words, the wave-shapes of fins **516** are depicted at their point of right-most deflection.

[0103] In FIG. **9**B, the cross-sectional view is taken at a point analogous to point C of the heat exchanger, as described above and with respect to FIG. **6**B. As a result, fins **516** as depicted in FIG. **9**B are 180 degrees out of phase with those depicted in FIG. **9**A.

[0104] As working fluid **108** flows through conduit **410-2** from point A to point C, the working fluid is squeezed from section **908** into each of sections **906** and **910**. This creates turbulence in and around sections **906** and **910** due to flow into and out of section **908**. In some embodiments, this flow is a swirl flow. In some embodiments, this flow is a vortex flow. As for heat exchanger **110**, the turbulence in sections **906** and **910** significantly increases the thermal transfer efficiency of heat exchanger **900**.

[0105] It should be noted that the combined cross-sectional area of passages **904**, like that of passages **602**, remains the same throughout the length of heat exchanger **110**. This is readily seen by the fact that as the cross-section of section

908-*i* becomes smaller, its reduction in size is offset by a commensurate increase in the cross-sectional area of section 908-*i*+1.

[0106] It is to be understood that the disclosure teaches just one example of the illustrative embodiment and that many variations of the invention can easily be devised by those skilled in the art after reading this disclosure and that the scope of the present invention is to be determined by the following claims.

What is claimed is:

1. A heat exchanger that thermally couples a first fluid and a second fluid, wherein the heat exchanger comprises:

(1) a first plate comprising a first material that is thermally conductive, wherein the first plate comprises;

(i) a first conduit for conveying the first fluid along a first direction, wherein the first conduit comprises at least one first channel; and

(ii) a first plurality of first fins, wherein each first fin is continuous along the first direction, and wherein each first fin comprises a first fin portion that has a first periodic shape having a first periodicity along the first direction; and

(2) a second plate comprising the first material, wherein the second plate comprises;

(i) a second conduit for conveying the first fluid along the first direction, wherein the second conduit comprises at least one second channel; and

(ii) a second plurality of second fins, wherein each second fin is continuous along the first direction;

wherein the first plurality of first fins and the second plurality of second fins collectively define a plurality of passages for conveying the second fluid along the first direction, and wherein each of the plurality of passages comprises a first section, second section, and third section; and

wherein the cross-sectional area of the first section varies along the first direction based on the first periodicity, and wherein the variation of the cross-sectional area of the first section induces flow of the second fluid between the first section and each of the second section and third section.

2. The heat exchanger of claim **1** wherein each second fin is substantially straight along a second direction that is orthogonal with the first direction.

3. The heat exchanger of claim **1** wherein each second fin comprises a second fin portion that has the first periodic shape having the first periodicity along the first direction.

4. The heat exchanger of claim **3** wherein the first plurality of first fins and the second plurality of second fins are arranged along the first direction with a phase offset.

5. The heat exchanger of claim **4** wherein the phase offset is 90 degrees.

6. The heat exchanger of claim **4** wherein the phase offset is 180 degrees.

7. The heat exchanger of claim **4** wherein the phase offset is 180 degrees.

8. The heat exchanger of claim **1** wherein each first fin further comprises a third fin portion that is substantially straight along the first direction, and wherein the third fin portion interposes the first plate and the first fin portion.

9. The heat exchanger of claim **8** wherein each second fin further comprises a fourth fin portion that is substantially straight along the first direction, and further wherein the fourth fin portion interposes the second plate and the second fin portion.

10. The heat exchanger of claim **8** wherein the first fin portion has a first height and the third fin portion has a second height, and wherein the first height and the second height are substantially equal.

11. The heat exchanger of claim **8** wherein the first fin portion has a first height and the third fin portion has a second height, and wherein the first height and the second height are unequal.

12. The heat exchanger of claim **1** wherein the variation of the cross-sectional area of the first section induces turbulence in the second fluid in at least one of the second section and third section.

13. The heat exchanger of claim **1** wherein the variation of the cross-sectional area of the first section induces a swirl flow of the second fluid in at least one of the second section and third section.

14. The heat exchanger of claim **1** wherein the variation of the cross-sectional area of the first section induces a vortex flow of the second fluid in at least one of the second section and third section.

15. A heat exchanger that thermally couples a first fluid and a second fluid, wherein the heat exchanger comprises:

a first conduit for conveying the first fluid along a first direction; and

a second conduit for conveying the second fluid along the first direction, wherein the second conduit comprises plurality of fins that are continuous along the first direction, and wherein each of the plurality of fins comprises a first fin portion that has a first periodic shape having a first periodicity along the first direction;

wherein the plurality of fins collectively define a plurality of passages in the second conduit, and wherein each of the plurality of passages comprises a first section, second section, and third section, and wherein the first section has a cross-sectional area that varies along the first direction based on the first periodicity, and further wherein the variation of the cross-sectional area of the first section induces flow of the second fluid between the first section and each of the second section and third section.

16. The heat exchanger of claim **12** wherein the variation of the cross-sectional area of the first section induces turbulence in at least one of the second section and third section.

17. The heat exchanger of claim **12** wherein the variation of the cross-sectional area of the first section induces swirl flow in at least one of the second section and third section.

18. The heat exchanger of claim **12** wherein the variation of the cross-sectional area of the first section induces vortex flow in at least one of the second section and third section.

19. A method comprising:

conveying a first fluid in a first conduit, wherein the first conduit conveys the first fluid along a first direction;

conveying a second fluid in a plurality of passages, wherein the second fluid and first fluid are thermally coupled via the first conduit and the plurality of passages, and wherein each of the plurality of passages is continuous along the first direction and comprises a first section, second section, and third section, and wherein the cross-sectional area of the first section varies along the first direction based on the first periodicity; and inducing a first flow of the second fluid between the first section and each of the second section and third section, wherein the first flow is induced by the variation of the cross-sectional area of the first section along the first direction.

20. The method of claim **19** wherein the first flow is induced such that a turbulent flow is induced in at least one of the second section and third section.

21. The method of claim **19** wherein the first flow is induced such that a swirl flow is induced in at least one of the second section and third section.

22. The method of claim **19** wherein the first flow is induced such that a vortex flow is induced in at least one of the second section and third section.

23. The method of claim **19** further comprising providing the first conduit and second conduit, wherein providing the first conduit and second conduit are provided by operations comprising:

providing a first plate, wherein the first plate comprises the first conduit and a first plurality of first fins;

providing a second plate, wherein the first plate comprises a second conduit for conveying the first fluid along the first direction, and wherein the second conduit comprises at least one second channel, and further wherein the second plate comprises a second plurality of second fins;

wherein the first plate and second plate are provided such that the first plurality of first fins and second plurality of second fins collectively define the plurality of passages.

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