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(71) **Applicant** *(for all designated States except US)*: **CARRIER CORPORATION** [US/US]; One Carrier Place, Farmington, Connecticut 06034 (US).

(72) **Inventors; and**
(75) **Inventors/Applicants** *(for US only)*: **TARAS, Michael F.** [US/US]; 5424 Springview Drive, Fayetteville, New York 13066 (US). **JOARDAR, Arindom** [IN/US]; 416 Deerfield Road, Apt. 3, East Syracuse, New York 13057 (US). **ESFORMES, Jack L.** [US/US]; 5896 Invincible Drive, Jamesville, New York 13078 (US). **DUNSHEE, Kevin B.** [US/US]; 221 Mallard Drive, Camillus, New York 13031 (US).

(74) **Agent**: **HABELT, William W.**; Cantor Colburn LLP, 20 Church Street, 22nd Floor, Hartford, Connecticut 06103 (US).

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(54) **Title**: MULTIPLE TUBE BANK FLATTENED TUBE FINNED HEAT EXCHANGER

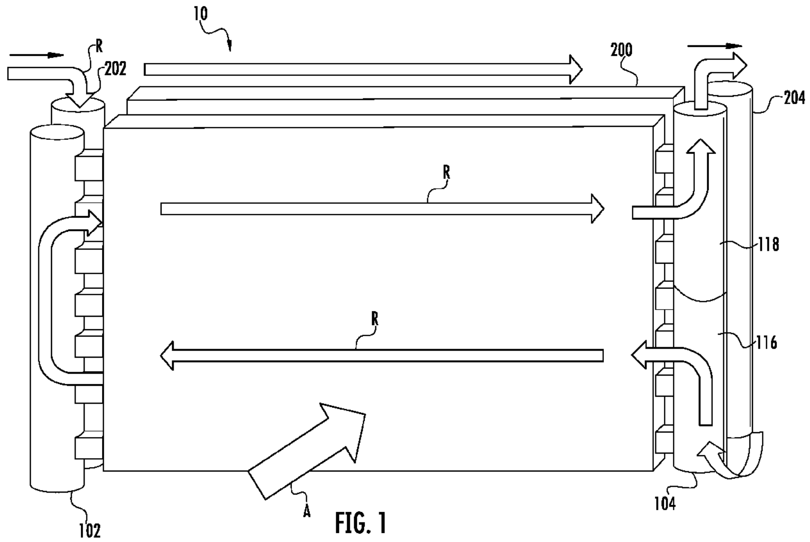

FIG. 1

(57) **Abstract**: A multiple tube bank heat exchanger includes a first tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship and a second tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship. The second tube bank is disposed behind the first tube bank with a leading edge of the second tube bank spaced from a trailing edge of the first tube bank. A continuous folded plate fin extends between the first and second flattened tube segments of both of said first tube bank and said second tube bank.

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MULTIPLE TUBE BANK FLATTENED TUBE FINNED HEAT EXCHANGER

Cross-Reference to Related Application

[0001] Reference is made to and this application claims priority from and the benefit of U.S. Provisional Application Serial No. 61/416,145, filed November 22, 2010, entitled "Multiple Tube Bank Flattened Tube Finned Heat Exchanger", which application is incorporated herein in its entirety by reference.

Field of the Invention

[0002] This invention relates generally to heat exchangers and, more particularly, to flattened tube and fin heat exchangers.

Background of the Invention

[0003] Heat exchangers have long been used as evaporators and condensers in heating, ventilating, air conditioning and refrigeration (HVACR) applications. Historically, these heat exchangers have been round tube and plate fin (RTPF) heat exchangers. However, flattened tube plate fin heat exchangers are finding increasingly wider use in industry, including the HVACR industry, due to their compactness, structural rigidity, lower weight and reduced refrigerant charge requirement, in comparison to conventional RTPF heat exchangers.

[0004] A typical flattened tube plate fin heat exchanger includes a first manifold, a second manifold, and a single tube bank formed of a plurality of longitudinally extending flattened heat exchange tubes disposed in spaced parallel relationship and extending between the first manifold and the second manifold. Additionally, a plurality of plate fins are disposed between each neighboring pair of heat exchange tubes for increasing heat transfer between a fluid, commonly air in HVACR applications, flowing over the outer surface of the flattened tubes and along the fin surfaces and a fluid, commonly refrigerant in HVACR applications, flowing inside the flattened tubes. In an embodiment of flattened tube commonly used in HVACR applications, termed multi-channel, mini-channel or micro-channel tubes, the interior of the flattened tube is subdivided into a plurality of parallel flow channels. For example, U.S. Pat. No. 6,964,296 shows a flattened tube flat plate fin heat exchanger in both a single tube bank and a double tube bank embodiment with horizontal tube runs and vertically extending flat plate fins.

[0005] A concern associated with the use of flattened tube heat exchangers in HVACR applications is poor drainage of condensate/water from the surface of the flattened tubes. The retention of condensate/water can be particularly problematic in flattened tube heat exchangers having horizontal tubes with high fin density and close tube spacing. In such constructions, condensate/water tends to collect on the flat horizontal surfaces of the heat exchange tubes in the spaces between the densely packed fins. The condensate/water collecting on the external surfaces of the heat exchanger tubes acts as an electrolyte and tends to accelerate corrosion and pitting of the tube surface. Condensate/water retention on the horizontal surface of the heat exchanger tube may also result in increased airside pressure drop and reduced air flow, as well as cause an undesirable condensate blow-off effect. Any condensate/water collecting on the horizontal tube surface also constitutes a layer of added thermal resistance to heat transfer on the airside of the heat exchange tubes.

[0006] Accordingly, the need exists for a flattened tube finned heat exchanger that is substantially free draining of condensate/water off the horizontal flat surface of the flattened horizontally extending flattened heat exchange tubes. The desire also exists for a flattened tube finned heat exchanger that is substantially free draining of condensate/water, while also achieving enhanced thermal performance.

Summary of the Invention

[0007] In an aspect, a heat exchanger includes a first tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship, a second tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship, the second tube bank disposed behind the first tube bank with a leading edge of the second tube bank spaced from a trailing edge of the first tube bank, and a continuous folded fin plate extending between the first and second flattened tube segments of both of the first tube bank and the second tube bank. The continuous folded fin plate may comprise a louvered plate having a first louvered section extending between the first and second flattened tube segments of the first tube bank and a second louvered section extending between the first and second flattened tube segments of the second tube bank. The continuous folded plate fin may further comprise a transition section between the first louvered section and the second louvered section, the transition section positioned between a trailing edge of the first tube bank and a leading edge of the second tube bank. The louvers of the first louvered section are oriented at a forward angle and the louvers of the second louvered section are oriented at a back angle. The transition section may include condensate drainage notches.

[0008] In an aspect, a heat exchanger is provided for passing a refrigerant in heat exchange relationship with an air flow passing through an air side of the heat exchanger. The heat exchanger includes a first tube bank including a plurality of flattened tube segments extending longitudinally in spaced parallel relationship, and a second tube bank including a plurality of flattened tube segments extending longitudinally in spaced parallel relationship, said second tube bank disposed downstream with respect to the first tube bank with a leading edge of the second tube bank spaced from a trailing edge of the first tube bank, wherein the air flow passes first transversely across the flattened tube segments of the first tube bank and then passes transversely across the flattened tube segments of the second tube bank, and the refrigerant flows first through the flattened heat exchange tube segments of the second tube bank and then through the flattened heat exchange tube segments of the first tube bank. In an embodiment, the heat exchange tube segments of the second tube bank are arranged in an in-line arrangement with the heat exchange tube segments of the first tube bank. In an embodiment, the heat exchange tube segments of the second tube bank are arranged in a staggered arrangement with the heat exchange tube segments of the first tube bank.

[0009] In an aspect, a parallel-counterflow heat exchanger is provided for passing a refrigerant in heat exchange relationship with an air flow passing through an air side of the heat exchanger. The heat exchanger includes at least a first tube bank and a second tube bank, each of said tube banks having a first pass including a first plurality of flattened heat exchange tube segments extending longitudinally in spaced parallel relationship and a second pass including a second plurality of flattened heat exchange tube segments extending longitudinally in spaced parallel relationship. The air flow passes first transversely across the flattened tube segments of said first tube bank, and passes second transversely across the flattened tube segments of said second tube bank. The refrigerant flows first through the first pass of the first tube bank, then through the first pass of the second tube bank, then through the second pass of the second tube bank, and then through the second pass of the first tube bank. In an embodiment, well adapted for use as an evaporator, the first plurality of heat exchange tube segments of the first pass of the first tube bank collectively define a first refrigerant flow area, the first plurality of heat exchange tube segments of the first pass of the second tube bank collectively define a second refrigerant flow area; the second plurality of heat exchange tube segments of the second pass of the second tube bank collectively define a third refrigerant flow area, and the second plurality of heat exchange tube segments of the second pass of the first tube bank collectively define a fourth refrigerant flow area. The respective refrigerant flow areas becoming progressively larger from the first refrigerant flow area to the second refrigerant flow area to the third refrigerant flow area to the fourth refrigerant flow area.

Brief Description of the Drawings

[0010] For a further understanding of the disclosure, reference will be made to the following detailed description which is to be read in connection with the accompanying drawing, where:

[0011] FIG. 1 is a diagrammatic illustration of an exemplary embodiment of a multiple tube bank, flattened tube finned heat exchanger as disclosed herein;

[0012] FIG. 1A is a side elevation cut out illustrating the fin and tube arrangement of the flattened tube finned heat exchanger of FIG 1;

[0013] FIG. 2 is a sectioned plan view of a portion of an exemplary embodiment of a multiple tube bank, flattened tube finned heat exchanger taken generally along line 2-2 of FIG. 1;

[0014] FIG. 3 is a sectioned side elevation view of the embodiment of the multiple tube bank, flattened tube finned heat exchanger of FIG. 2;

[0015] FIG. 4 is a sectioned side elevation view of an alternate embodiment of the multiple tube bank, flattened tube finned heat exchanger of FIG. 3;

[0016] FIG. 5 is a perspective view of a single fold of the continuous folded fin plate of the heat exchanger disclosed herein;

[0017] FIG. 6 is a section plan view through the fins of the single fold of the folded fin plate shown in FIG. 5;

[0018] FIG. 7 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating an exemplary cross-counterflow refrigerant circuit;

[0019] FIG. 8 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating another exemplary cross-counterflow refrigerant circuit;

[0020] FIG. 9 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating another exemplary cross-counterflow refrigerant circuit;

[0021] FIG. 10 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating another exemplary cross-counterflow refrigerant circuit;

[0022] FIG. 11 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating another exemplary cross-counterflow refrigerant circuit;

[0023] FIG. 12 is a diagrammatic view of the multiple tube bank heat exchanger disclosed herein illustrating another exemplary cross-counterflow refrigerant circuit;

[0024] FIG. 13 is a diagrammatic view of a multiple tube bank heat exchanger disclosed herein illustrating a parallel-counterflow refrigerant circuit; and

[0025] FIG. 14 is a sectioned side elevation view of an exemplary embodiment of a multiple tube bank flattened tube finned heat exchanger showing a first and a second tube bank arranged in a staggered heat exchange tube arrangement.

Detailed Description of the Invention

[0026] Referring initially to FIGs. 1 and 2 of the drawing, there is depicted an exemplary embodiment of a multiple bank flattened tube finned heat exchanger 10 in accordance with the disclosure. As depicted therein, the heat exchanger includes a first tube bank 100 and at least a second tube bank 200 that is disposed behind the first tube bank 100. Each of the first tube bank 100 and the second tube bank 200 includes a first manifold 102, 202 extending along a vertical axis and a second manifold 104, 204 also extending along a vertical axis. It is to be understood that the multiple bank flatted tube finned heat exchanger 10 disclosed herein may include more than two tube banks.

[0027] The first manifold 102 and the second manifold 104 of the first tube bank 100 are spaced apart from one another with a plurality of heat exchange tube segments 106, that is at least a first and a second tube segment, extending longitudinally in spaced parallel relationship between and connecting the first manifold 102 and the second manifold 104 in fluid communication. Similarly, the first manifold 202 and the second manifold 204 of the second tube bank 200 are spaced apart from one another with a plurality of heat exchange tube segments 206, that is at least a first and a second tube segment, extending longitudinally in spaced parallel relationship between and connecting the first manifold 202 and the second manifold 204 in fluid communication. The neighboring manifolds at the ends of the tube banks 100, 200 may be separate manifolds, such as depicted in FIG. 2 with respect to first manifold 102 and first manifold 202, connected by appropriate piping as necessary to accommodate the particular refrigerant circuitry incorporated into the heat exchanger 10, or the neighboring manifolds at the ends of the tube banks 100, 200 may be combined or integrated into a single manifold, such as depicted in FIG. 2 with respect to second manifold 104 and second manifold 204, subdivided into chambers as appropriate, which chambers are fluidly interconnected internally within the single manifold as necessary to accommodate the particular refrigerant circuitry incorporated into the heat exchanger 10.

[0028] Each of the heat exchange tube segments 106, 206 comprises a flattened heat exchange tube having a leading edge 108, 208, a trailing edge 110, 210, an upper flat surface 112, 212, and a lower flat surface 114, 214. The interior of each of the heat exchange tube segments 106, 206 may be divided by longitudinally extending interior walls into a plurality of parallel flow channels 120, 220 that establish fluid communication between the respective headers of the first and the second tube banks 100, 200. The second tube bank 200 is disposed behind the first tube bank 100, with respect to the airflow direction, with the leading edge 208 of the second tube bank 200 spaced from the trailing edge 110 of the first tube bank 100 by a relatively narrow gap, G, of about 2 to 4 millimeters (mm) (0.08 to 0.15 inches). In the depicted embodiment, each of the heat exchange tube segments 106, 206 of the first and second tube banks 100, 200, respectively, comprises a multi-channel tube having an interior flow passage divided by interior walls into a plurality of discrete flow channels 120, 220 that extend longitudinally the length of the tube from an inlet end of the tube to the outlet end of the tube. However, it is to be understood that the tube banks 100, 200 could include serpentine tubes with the heat exchange tube segments 106, 206 being parallel linear tube segments connected by U-bends or hairpin turns to form a serpentine tube connected at its respective ends between the first manifold and the second manifold of the tube bank.

[0029] The flattened tube finned heat exchanger 10 disclosed herein further includes a plurality of folded fin plates 20. Each folded fin plate 20 is formed of a single continuous plate of fin material tightly folded in a ribbon-like fashion thereby providing a plurality of closely spaced fins 22 that extend generally orthogonal to the flattened heat exchange tubes 106, 206. Typically, the fin density of the closely spaced fins 22 of each continuous folded fin plate 20 may be about 18 to 25 fins per inch, but higher or lower fin densities may also be used. The depth of each of the ribbon-like folded plate 20 extends from the leading edge 108 of the first tube bank 100 to the trailing edge of 210 of the second bank 200 and spans the gap, G, between the trailing edge 110 of the first tube bank 100 and the leading edge 208 of the second tube bank 200. In an embodiment of the heat exchanger 10, the manifolds, heat exchange tubes and fins are all made of aluminum or aluminum alloy material.

[0030] Referring now to FIGs. 5 and 6, a single fold of the continuous ribbon-like folded fin plate 20 is depicted and shows two fins 22. Each fin 22 has a first section 24, a second section 26 and a third section 28. When installed in the heat exchanger 10, the first section 24 is disposed within the first tube bank 100, the third section 28 is disposed within the second tube bank 200, and the second section

26 spans the gap, G, between the trailing edge 110 of the first tube bank 100 and the leading edge 208 of the second tube bank 200. The first and second sections 24 and 28 may be provided with louvers 30, 32 formed in a conventional manner in the material of the fin plate 20. In the depicted embodiment, the louvers 30 formed in the first section 24 are forward angled relative to the direction (indicated by the arrow in FIG. 6) of flow along the fins 22 and the louvers 32 formed in the third section 28 are back angled relative to the direction of flow along the fins 22. In the depicted embodiment, both the louvers 30 in both first section 24 and louvers 32 in the third section 28 of each fin 22 are forward and back angled, respectively, at a desired louver angle, A. The second section 26 comprises a turn-around louver 34 that provides a transition between the forward angled louvers 30 of the first section 24 and the back angled louvers 32 of the third section 28 of each fin 22. When installed in the heat exchanger 10, the second section 26 spanning the gap, G, between the first tube bank 100 and the second tube bank 200 provides a drainage path for condensate/water collecting on the horizontally disposed upper flat surfaces 114, 214 of the flattened heat exchange tube segments 106, 206.

[0031] In the embodiment of the heat exchanger 10 depicted in FIG. 3, the first tube bank 100 and the second tube bank 200 have the same number of heat exchange tube segments 106, 206 extending longitudinally and horizontally in spaced parallel relationship between their respective manifolds at the same tube spacing. Additionally, the heat exchange tube segments 206 are disposed in direct alignment with the heat exchange tube segments 106. In this arrangement, a continuous folded plate fin 20 is disposed between each pair of adjacent aligned sets of tube segments 106, 206.

[0032] However, the first tube bank 100 and the second tube bank 200 need not have the same number of heat exchange tube segments 106, 206. Rather, the number of heat exchange tubes 106 in the first tube bank 100 may have a different number of heat exchange tubes 206 in the second tube bank 200. For example, in the embodiment of the heat exchanger 10 depicted in FIG. 4, every other heat exchange tube 106 in the first tube bank 100 has been removed such that the first tube bank 100 has a lower number of heat exchange tube segments 106 than the number of heat exchange tube segments 206 of the second tube bank 200. In this particular embodiment, every other tube segment 206 of the second tube bank 200 is directly aligned behind a tube segment 106 of the first tube bank 100, while the remaining tube segments 206 do not have a corresponding tube segment 106 disposed upstream thereof.

[0033] It should be noted that in embodiments wherein the first and second tube banks 100, 200 have an unequal number of tube segments 106, 206, a plurality of continuous folded fin plates 20 may extend through both tube banks with the number of continuous fin plates 20 determined such that a continuous fin plate 20 is disposed between each pair of adjacent heat exchange tube segments of the tube bank having the greater number of heat exchange tube segments 106, 206. However, alternatively, some of the fin sections in the tube bank having the lower number of heat exchange tube segments may be removed so that some of the folded fin plates, but not all, extend only from the leading edge to the trailing edge of the tube bank having the greater number of heat exchange tube segments.

[0034] The tube width may also be different between the heat exchange tube segments 106 positioned in the first tube bank 100 and the heat exchange tube segments 206 positioned in the second tube bank 200. In general, the widths of the heat exchange tube segments 106, 206 of the multiple bank heat exchanger 10 would typically range from 12 to 32 mm (about 0.5 to 1.5 inch). Since the number of flow channels 120, 220 within the flattened heat exchange tube segments 106, 206, respectively, typically varies directly with tube width, the number of flow channels 120, 220 in each of the heat exchange tube segments 106, 206, respectively, may be different and be tailored to the refrigerant thermo-physical properties, such as density.

[0035] For instance, for condenser heat exchangers in air conditioning or refrigeration applications, the heat exchange tubes of the downstream tube bank with respect to the refrigerant flow, which in the embodiment depicted in Figs. 1 and 3, would be the first tube bank 100, would have lower width tube segments 106, as compared to the width of the heat exchange tube segments 206 of the second tube bank 206, i.e. the upstream tube bank with respect to refrigerant flow, to accommodate the refrigerant condensing process and maintain desired refrigerant flow velocity in the channels of the heat exchange tube segments in the downstream tube bank for the appropriate balance between heat transfer and pressure drop characteristics.

[0036] Reducing the number of heat exchange tube segments in the downstream tube bank with respect to refrigerant flow, whether or not in conjunction with reducing the width of the heat exchange tube segments in the downstream tube bank, relative to the number of heat exchange tube segments in the upstream tube bank with respect to refrigerant flow, is also effective in accommodating the refrigerant condensing process and maintaining desired refrigerant flow velocity in the flow channels of the heat exchange tube segments in the downstream tube bank for the appropriate balance between heat transfer and pressure drop characteristics. Additionally, the cross-sectional flow area through the heat exchange tube segments of the downstream tube bank with respect to refrigerant flow may reduced by decreasing the cross-sectional flow area of the multiple flow channels or decreasing the number of the flow channels to accommodate the condensing refrigerant flow, while maintaining outside geometrical characteristics identical to those of the upstream tube bank with respect to refrigerant flow.

[0037] These concepts can be applied to evaporator heat exchangers in air conditioning or refrigeration applications in the reversed manner. For example, for evaporator heat exchangers, the heat exchange tubes of the upstream tube bank with respect to the refrigerant flow could have lower width tube segments as compared to the width of the heat exchange tube segments of the downstream tube bank with respect to refrigerant flow.

[0038] In conjunction with multiple tube banks having different width heat exchange tube segments, the louvered fins 22 of the continuous folded fin plates 20 may be asymmetrical with the first section 24 and the third section 28 sized differently to match the respective tube widths of the first and second tube banks 100, 200. If the heat exchange tube segments 106, 206 have the same tube width, then the fins 22 are with the first and third fin sections 24, 28 being of equal length and the second sections 26, i.e. the turn-around louver section, of the fins 22 being centrally located between the first and third sections 24, 28. However, for a heat exchanger configuration having tube banks of dissimilar tube width, the second section 26, i.e. the turn-around louver section, is not positioned centrally in the fin 22, but rather is located off-center in the fin between the first section 24 and the third section 28 of the fin.

[0039] Nevertheless, the second section 26, i.e. the turn-around louver section, should again be aligned to span the gap, G, between the trailing edge of the first tube bank 100 and the leading edge of the second tube bank 200, such as illustrated in FIGs. 3 and 4, because aligning the turn-around louver sections 26 of the louvered fins 22 with the gap, G, between the tube banks provides improved condensate/water drainage from the surface of the flattened heat exchange tube segments. Designs with presence of notches at the turn-around louver will necessitate use of asymmetrical louver bank.

[0040] Referring now again to FIG. 1, the multiple tube bank flattened tube finned heat exchanger 10 will be described as configured as a condenser heat exchanger in a refrigerant vapor compression system of an air conditioning unit, transport refrigeration unit or commercial refrigeration unit. In such applications, refrigerant vapor from the compressor (not shown) of the refrigerant vapor compression system (not shown) passes through the manifolds and heat exchange tube segments of the tube banks 100, 200, in a manner to be described in further detail hereinafter, in heat exchange relationship with a cooling media, most commonly ambient air, flowing through the airside of the heat exchanger 10 in the direction indicated by the arrow labeled "A" that passes over the outside surfaces of the heat exchange tube segments 106, 206 and the surfaces of the folded fin plates 20.

[0041] The multiple tube bank flattened tube finned heat exchanger 10 depicted in FIG. 1 has a cross-counterflow circuit arrangement. The air flow first passes transversely across the upper and lower horizontal surfaces 112, 114 of the heat exchange tube segments 106 of the first tube bank and then passes transversely across the upper and lower horizontal surfaces 212, 214 of the heat exchange tube segments 206 of the second tube bank 200. The refrigerant passes in cross-counterflow arrangement to the airflow, in that the refrigerant flow passes first through the second tube bank 200 and then through the first tube bank 100. In the process, the refrigerant passing through the flow channels of the heat exchange tubes

106, 206 rejection heat into the airflow passing through the air side of the heat exchanger 10. The multiple tube bank flattened tube finned heat exchanger 10 having a cross-counterflow circuit arrangement yields superior performance, as compared to the crossflow or cross-parallel flow circuit arrangements.

[0042] More specifically, in the embodiment depicted in FIG. 1, the refrigerant flow, designated by the label "R", passes from the refrigerant circuit (not shown) into the first manifold 202 of the second tube bank 200 and is distributed amongst the heat exchange tube segments 206 to flow therethrough into the second manifold 204 of the second tube bank 200. The refrigerant collecting in the second manifold 204 of the second tube bank 200 then passes into a lower section 116 of the second manifold 104 of the first tube bank 100 and is distributed amongst a first portion of the heat exchange tube segments 106 to flow therethrough into the first manifold 102 of the tube bank 100. The refrigerant passes from the first manifold 102 into a second portion of the heat exchange tube segments 106 and flows therethrough into an upper portion 118 of the second manifold 104 of the first tube bank 100 and is directed thereform back into the refrigerant circuit of the refrigerant vapor compression system (not shown). Thus, the refrigerant circuit of the embodiment of the multiple bank heat exchanger depicted in FIG. 1 is a single pass - two pass, cross-counterflow refrigerant circuit.

[0043] Referring now to FIGs. 7-11, various other exemplary embodiments of acceptable refrigerant circuit arrangements of the cross-counterflow, two tube bank heat exchanger 10 are illustrated schematically. In each of FIGs. 7-10, the flow of air though the air side of the heat exchanger is in cross-counterflow as indicated by the arrow "A" and first through the first tube bank 100 and then through the second tube bank 200 as discussed previously with respect to FIG. 1. The flow of refrigerant, indicated by the arrow "R", is first through the second tube bank 200 and then through the first tube bank 100 in overall cross-counterflow to the air passing through the air side of the heat exchanger 10. In the embodiment depicted in FIG. 7, the refrigerant flow circuit comprises a single pass-single pass, cross-counterflow refrigerant circuit. In the embodiment depicted in FIG. 8, the refrigerant flow circuit comprises a two pass-two pass cross-counterflow refrigerant circuit. In the embodiment depicted in FIG. 9, the refrigerant flow circuit comprises a single pass-two pass variation of the single pass-two pass cross-counterflow refrigerant circuit of FIG. 1. In the embodiment depicted in FIG. 10, the refrigerant flow circuit comprises a two pass-three pass cross-counterflow refrigerant circuit.

[0044] Referring now to FIGs. 11 and 12, exemplary embodiments of acceptable refrigerant circuit arrangements of the cross-counterflow, three tube bank heat exchanger 10 are illustrated schematically. In each of FIGs. 11-12, the flow of air through the air side of the heat exchanger is in cross-counterflow as indicated by the arrow “A” and first through the first tube bank 100, then through the second tube bank 200, and lastly through the third tube bank 300. The flow of refrigerant, indicated by the arrow “R”, passes first through the third tube bank 300, then through the second tube bank 200, and lastly through the first tube bank 100 in overall counter flow to the air passing through the air side of the heat exchanger 10. In the embodiment depicted in FIG, 11, the refrigerant flow circuit comprises a single pass-single pass-single pass cross-counterflow refrigerant circuit. In the embodiment depicted in FIG, 12, the refrigerant flow circuit comprises a single pass-two pass-three pass cross-counterflow refrigerant circuit.

[0045] Referring now to FIG. 13, there is depicted a multiple tube bank flattened tube heat exchanger 400 having three tube banks 100, 200, 300 in a parallel-counterflow arrangement. In the embodiment depicted in FIG. 13, the flow of air through the air side of the heat exchanger, as indicated by the arrow “A”, passes first through the first tube bank 100, then through the second tube bank 200, and lastly through the third tube bank 300. Each of the tube banks 100, 200, 300 comprises a two-pass tube bank having a lower pass 130, 230, 330, respectively, and an upper pass 140, 240, 340, respectively. The flow of refrigerant, indicated by the arrow “R”, is first generally parallel to the flow of air through the heat exchanger 400 and then generally counter to the flow of air through the heat exchanger 400. The refrigerant passes first through the lower pass 130 of the first tube bank 100, then through the lower pass 230 of the second tube bank 200, then through the lower pass 330 of the third tube bank 300, then through the upper pass 340 of the third tube bank 300, then through the upper pass 240 of the second tube bank 200, and lastly through the upper pass 140 of the first tube bank 100 and back into the refrigerant circuit (not shown). In the embodiment of the heat exchanger 400 depicted in FIG. 13, the number of heat exchange tube segments varies progressively amongst the passes 130, 230, 330, 340, 240, 140 with the least number of heat exchange tube segments being in the lower pass 130 of the first tube bank 100, then increasing progressively through the pass 230, 330, 340, 240 to the upper pass 140 of the first tube bank 100 which has the greatest number of heat exchange tubes. Multiple bank flattened tube heat exchanger 400 is particularly suitable for application as an evaporator in a refrigerant vapor compression system due to the progressively increasing refrigerant flow area provided as the refrigerant flows through the various passes of the of the three tube banks 100, 200, 300 in the progression of passes of increasing tube number as herein described as this arrangement accommodates the change in density of the refrigerant as the refrigerant passes through the evaporator. It has to be understood that in case the refrigerant condensation process is to take place inside the heat exchanger tubes, the number of tubes in each segment can progressively decrease.

[0046] With various well-optimized circuits, such as exemplified by the refrigerant circuits discussed herein, heat transfer performance was improved without noticeable penalty in the refrigerant side pressure drop or fan power. The enhanced performance of the multiple tube bank, flattened tube finned heat exchanger 10 as disclosed herein permits coil volume & face area of the heat exchanger to be reduced up to 25% as compared to conventional single bank flattened tube heat exchangers.

[0047] Referring now to FIG. 14, there is depicted an exemplary embodiment of a multiple bank flattened tube folded fin plate heat exchanger 50 having a first tube bank 510 having a plurality of multi-channel heat exchange tube segments 512 and a second tube bank 520 having a plurality of multi-channel tubes 522. The heat exchange tube segments 512 of the first tube bank 510 extend in parallel spaced relationship between a first manifold (not shown) and a second manifold (not shown) as in the manner discussed earlier with respect to the first tube bank 100 of the heat exchanger 10. Similarly, the heat exchange tube segments 522 of the second tube bank 520 extend in parallel spaced relationship between a first manifold (not shown) and a second manifold (not shown) as in the manner discussed earlier with respect to the second tube bank 200 of the heat exchanger 10. However, in the heat exchanger 50, the second tube bank 520 is arranged in spaced relationship at a relatively narrow gap, G, downstream with respect to airflow through the air side of the heat exchanger 50 of the first tube bank 510 with the heat exchange tube segments 522 disposed in staggered relationship with the heat exchange tube segments 512 of the first tube bank 510.

[0048] Additionally, a first plurality of folded fin plates 530 is provided in the first tube bank 510 with one folded fin plate 530 installed between and in heat transfer relationship with each pair of neighboring heat exchange tube segments 510 and a second plurality of folded fin plates 540 is provided in the second tube bank 520 with one folded fin plate 540 installed between and in heat transfer relationship with each pair of neighboring heat exchange tube segments 520. Each of the folded fin plates 530, 540 comprises a continuous ribbon-like folded plate defining a plurality of fins 532, 542, respectively, that extended generally orthogonal to the heat exchange tube segments 512, 522, respectively. Each of the fins may comprise a louvered fin. Each folded fin plate 530 extends from the leading edge to the trailing edge of the heat exchange tube segments 512 of the first tube bank 510, but does not extend into the second tube bank 520. Each folded fin plate 540 extends from the leading edge to the trailing edge of the heat exchange tube segments 522 of the first tube bank 520, but does not extend into the first tube bank 510. The terminology used herein is for the purpose of description, not limitation. Specific structural and functional details disclosed herein are not to be interpreted as limiting, but merely as basis for teaching one skilled in the art to employ the present invention. Those skilled in the art will also recognize the equivalents that may be substituted for elements described with reference to the exemplary embodiments disclosed herein without departing from the scope of the present invention.

[0049] While the present invention has been particularly shown and described with reference to the exemplary embodiments as illustrated in the drawing, it will be recognized by those skilled in the art that various modifications may be made without departing from the spirit and scope of the invention. It is also to be understood that the heat exchanger disclosed herein may be utilized in connection with refrigerant vapor compression systems used in air conditioning, heat pump, and refrigeration applications. Therefore, it is intended that the present disclosure not be limited to the particular embodiment(s) disclosed as, but that the disclosure will include all embodiments falling within the scope of the appended claims.

We Claim:

1. A heat exchanger comprising:

a first tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship;

a second tube bank including at least a first and a second flattened tube segments extending longitudinally in spaced parallel relationship, said second tube bank disposed behind said first tube bank with a leading edge of the second tube bank spaced from a trailing edge of the first tube bank; and

a continuous folded plate fin extends between the first and second flattened tube segments of both of said first tube bank and said second tube bank.

2. The heat exchanger as recited in claim 1 wherein said continuous folded plate fin comprises a louvered plate having a first louvered section extending between the first and second flattened tube segments of the first tube bank and a second louvered section extending between the first and second flattened tube segments of the second tube bank.

3. The heat exchanger as recited in claim 2 wherein said continuous folded plate fin further comprises a transition section between the first louvered section and the second louvered section, the transition section positioned between a trailing edge of said first tube bank and a leading edge of said second tube bank.

4. The heat exchanger as recited in claim 3 wherein said transition section includes condensate drainage apertures.

5. The heat exchanger as recited in claim 3 wherein the first louvered section and second louvered section are symmetrical.

6. The heat exchanger as recited in claim 3 wherein the first louvered section and the second louvered section are asymmetrical.

7. The heat exchanger as recited in claim 2 wherein the louvers of the first louvered section are oriented at a forward angle and the louvers of the second louvered section are oriented at a back angle.

8. The heat exchanger as recited in claim 5 wherein said continuous folded plate fin further comprises a transition section between the first louvered section and the second louvered section, the transition section positioned between a trailing edge of said first tube bank and a leading edge of said second tube bank.

9. The heat exchanger as recited in claim 8 wherein said transition section includes condensate drainage apertures.

10. The heat exchanger as recited in claim 1 wherein said first tube bank and said second tube bank comprise tube banks having an equal number of flattened tube segments.

11. The heat exchanger as recited in claim 1 wherein one of said first tube bank and said second tube bank comprises a tube bank having a lesser number of flattened tube segments than the tube bank of the other of said first tube bank and said second tube bank.

12. The heat exchanger as recited in claim 1 wherein the flattened tube segments of said first tube bank have a first tube width and the flattened tube segments of said second tube bank have a second tube width being different from the first tube width.

13. The heat exchanger as recited in claim 1 wherein each flattened tube segment of said first tube bank comprises a multi-channel tube segment having a first plurality of parallel flow channels and each flattened tube segment of said second tube bank comprises a multi-channel tube segment having a second plurality of parallel flow channels, the second plurality of parallel flow channels being different in number from the first plurality of parallel flow channels.

14. A heat exchanger for passing a refrigerant in heat exchange relationship with an air flow passing through an air side of the heat exchanger, comprising:

a first tube bank including a plurality of flattened tube segments extending longitudinally in spaced parallel relationship; and

a second tube bank including a plurality of flattened tube segments extending longitudinally in spaced parallel relationship, said second tube bank disposed downstream with respect to said first tube bank with a leading edge of the second tube bank spaced from a trailing edge of the first tube bank;

wherein the air flow passes first transversely across the flattened tube segments of said first tube bank and then passes transversely across the flattened tube segments of said second tube bank, and the refrigerant flows first through the flattened heat exchange tube segments of said second tube bank and then through the flattened heat exchange tube segments of said first tube bank.

15. The heat exchanger as recited in claim 14 wherein the heat exchange tube segments of said second tube bank are arranged in an in-line arrangement with the heat exchange tube segments of said first tube bank.

16. The heat exchanger as recited in claim 14 wherein the heat exchange tube segments of said second tube bank are arranged in a staggered arrangement with the heat exchange tube segments of said first tube bank.

17. A heat exchanger for passing a refrigerant in heat exchange relationship with an air flow passing through an air side of the heat exchanger, comprising:

at least a first tube bank and a second tube bank, each of said tube banks having a first pass including a first plurality of flatted heat exchange tube segments extending longitudinally in spaced parallel relationship and a second pass including a second plurality of flatted heat exchange tube segments extending longitudinally in spaced parallel relationship;

wherein the air flow passes first transversely across the flattened tube segments of said first tube bank, and passes second transversely across the flattened tube segments of said second tube bank, and the refrigerant flows first through the first pass of the first tube bank, then through the first pass of the second tube bank, then through the second pass of the second tube bank, and then through the second pass of the first tube bank.

18. The heat exchanger as recited in claim 17 wherein the first plurality of heat exchange tube segments of the first pass of the first tube bank collectively define a first refrigerant flow area, the first plurality of heat exchange tube segments of the first pass of the second tube bank collectively define a second refrigerant flow area; the second plurality of heat exchange tube segments of the second pass of the second tube bank collectively define a third refrigerant flow area, and the second plurality of heat exchange tube segments of the second pass of the first tube bank collectively define a fourth refrigerant flow area, the respective refrigerant flow areas becoming progressively larger from the first refrigerant flow area to the second refrigerant flow area to the third refrigerant flow area to the fourth refrigerant flow area.

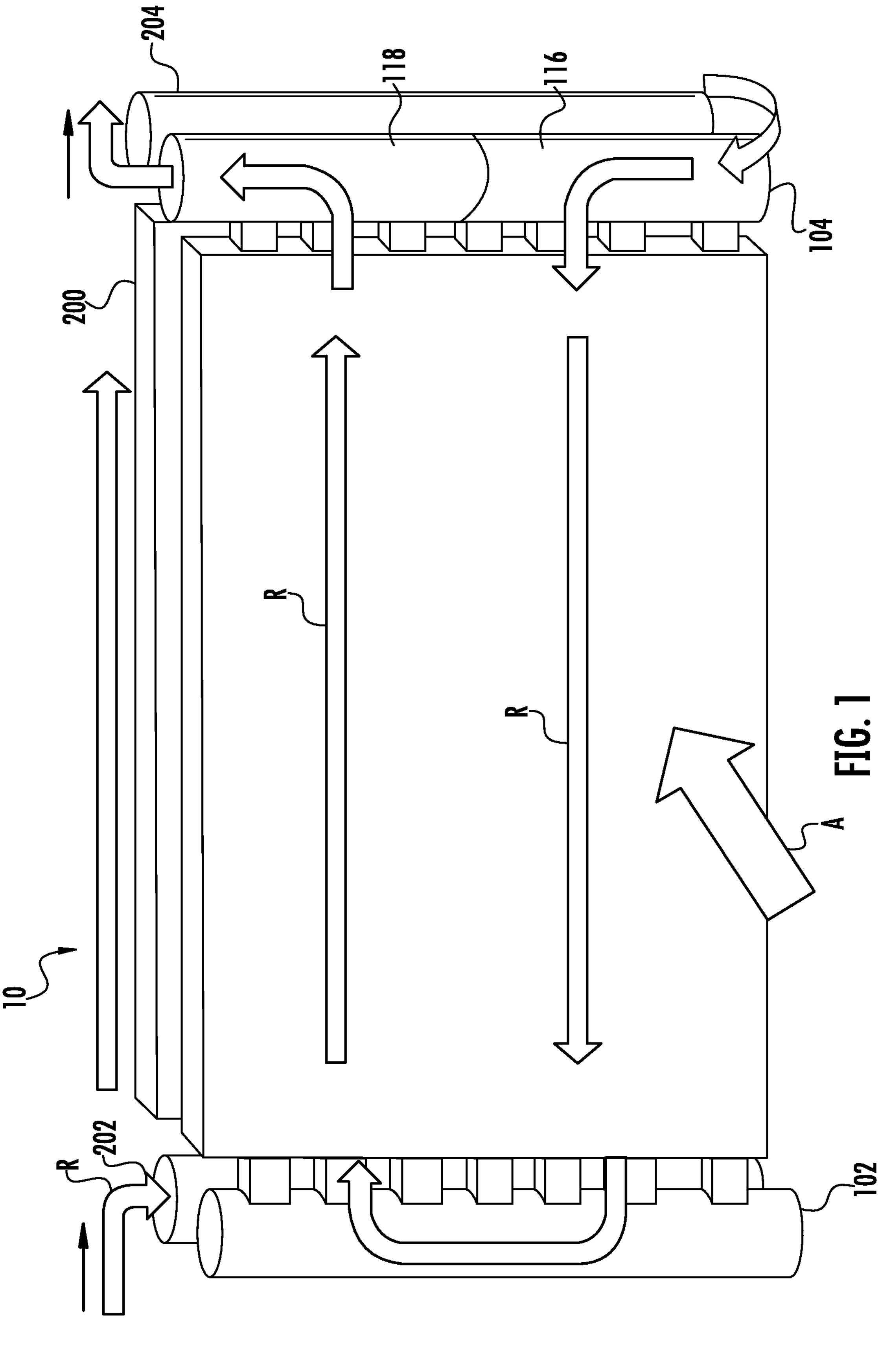


FIG. 1

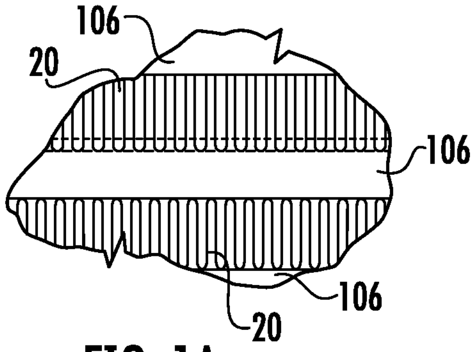


FIG. 1A

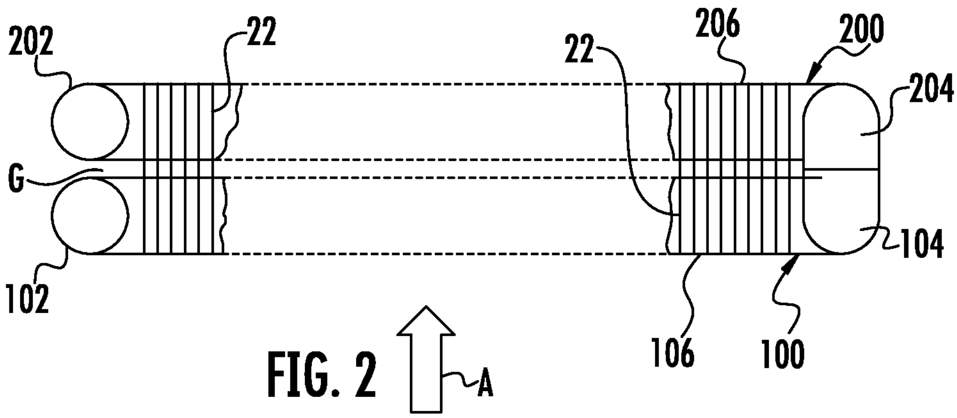
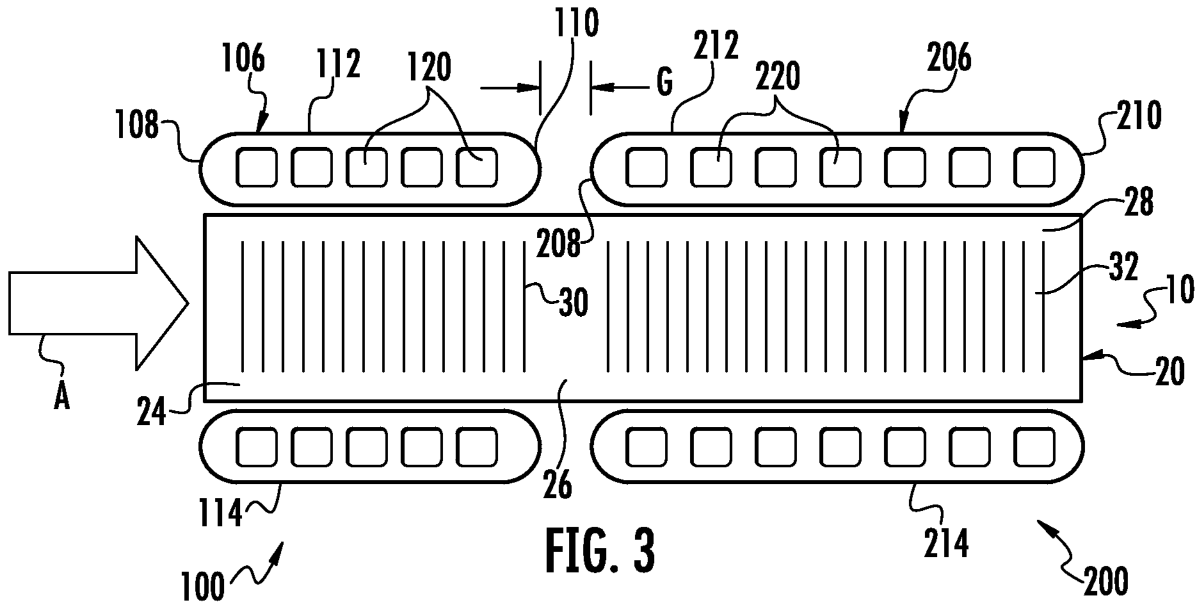


FIG. 2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200

FIG. 3

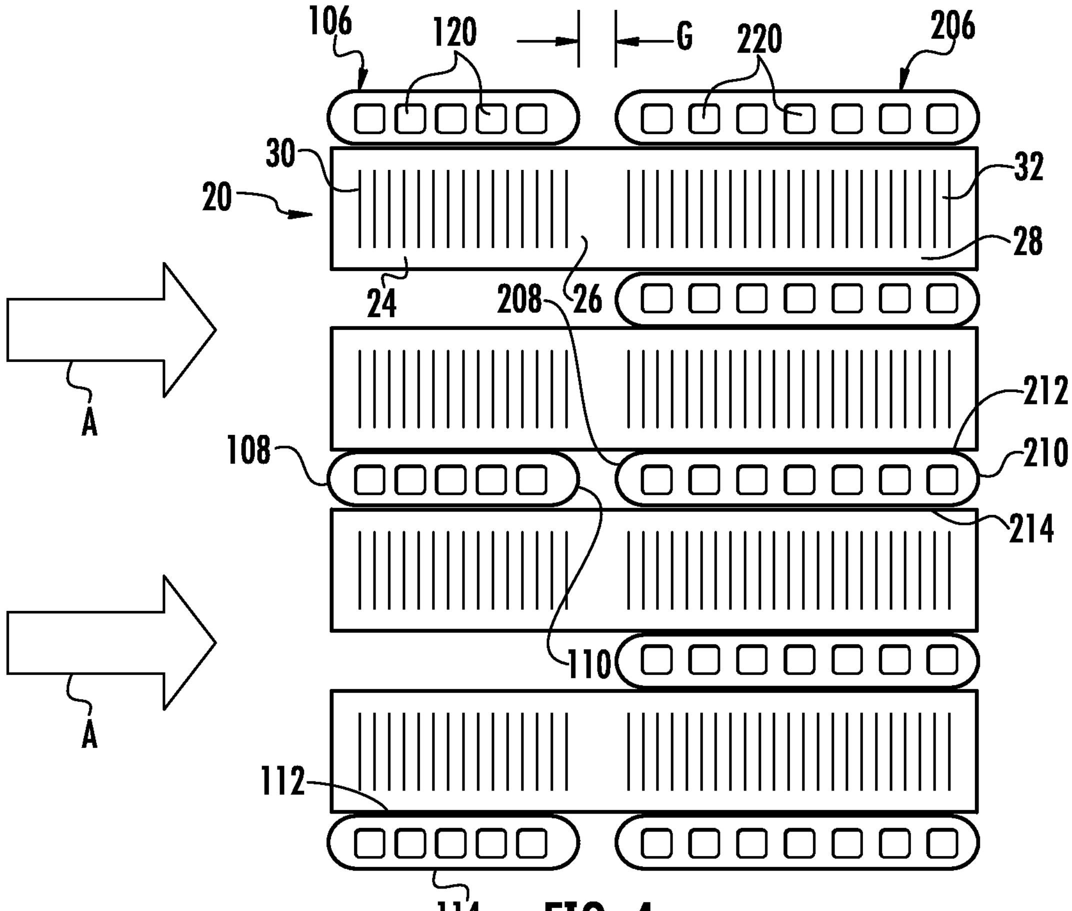
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FIG. 4

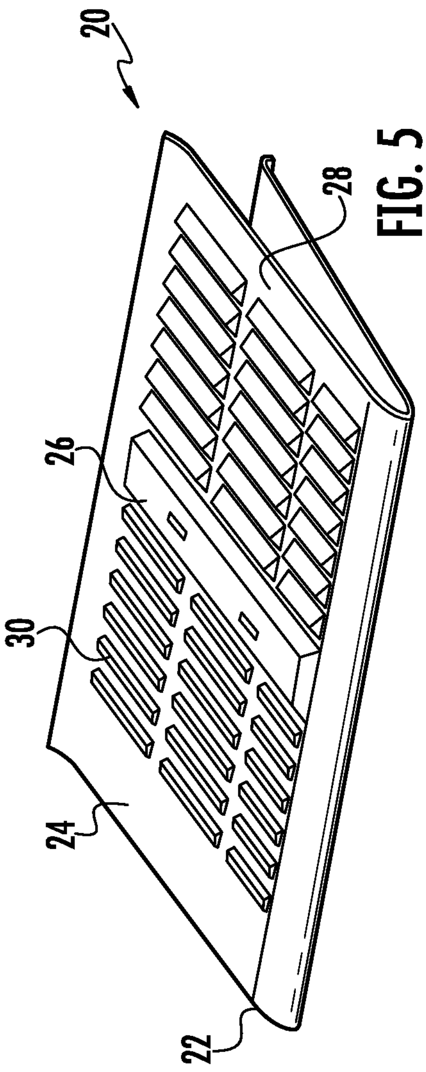


FIG. 5

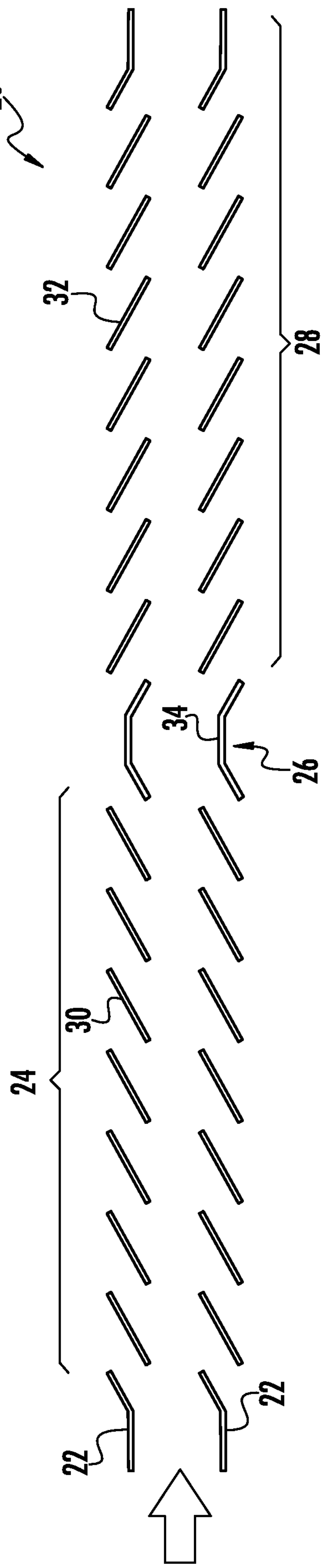


FIG. 6

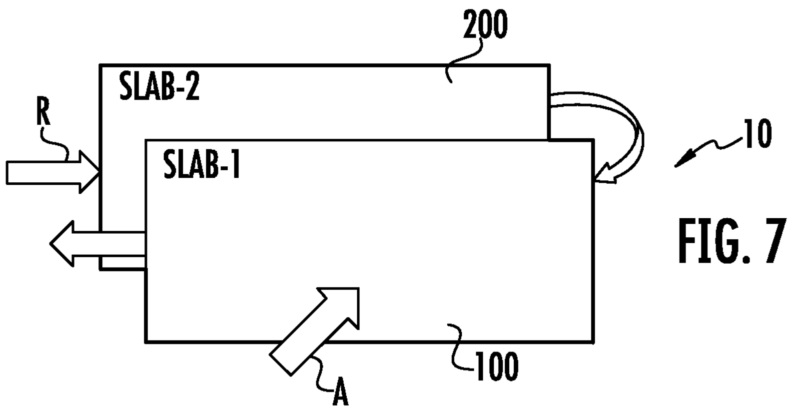


FIG. 7

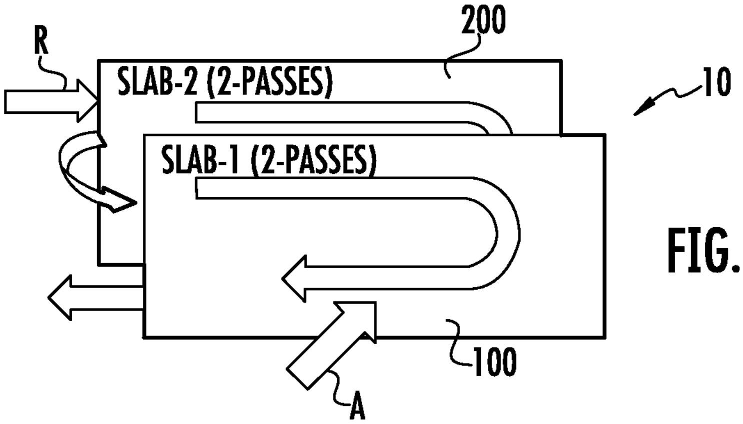


FIG. 8

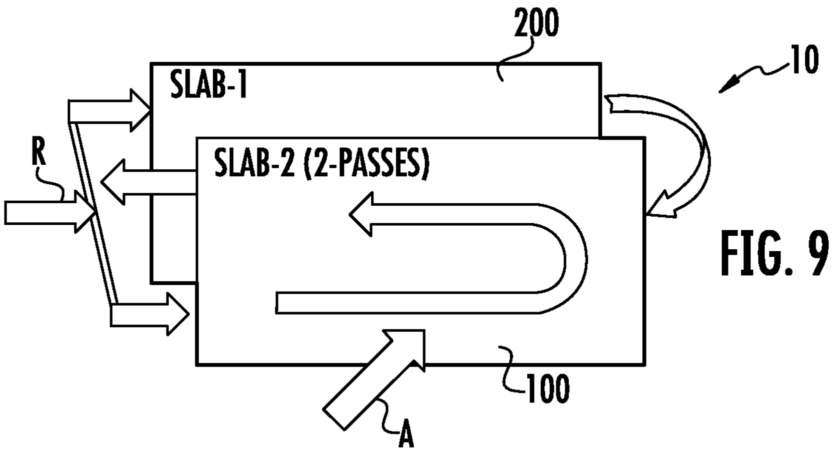


FIG. 9

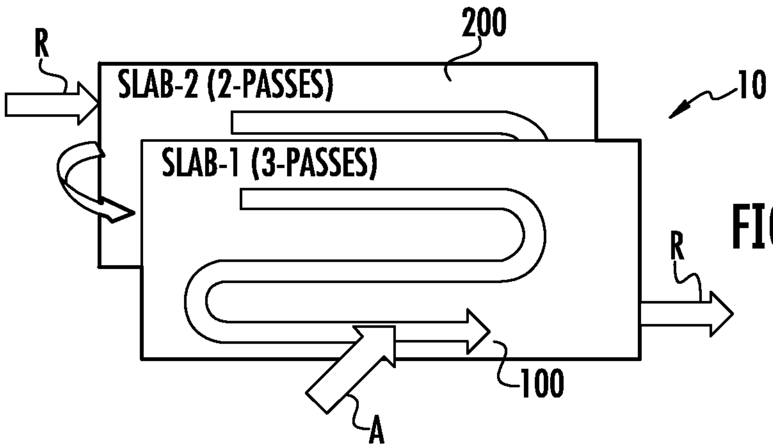
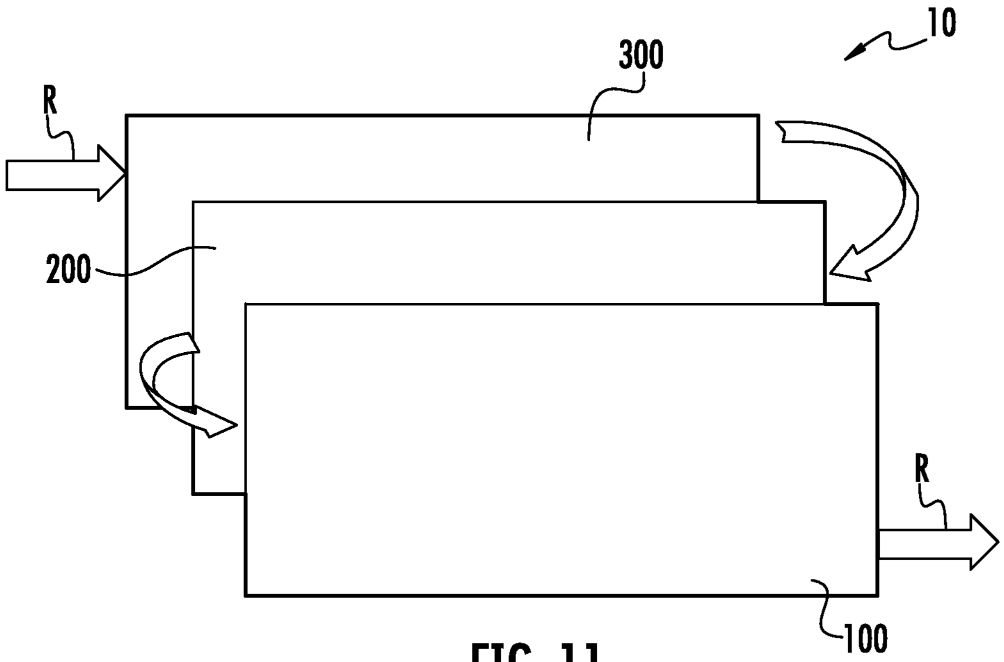


FIG. 10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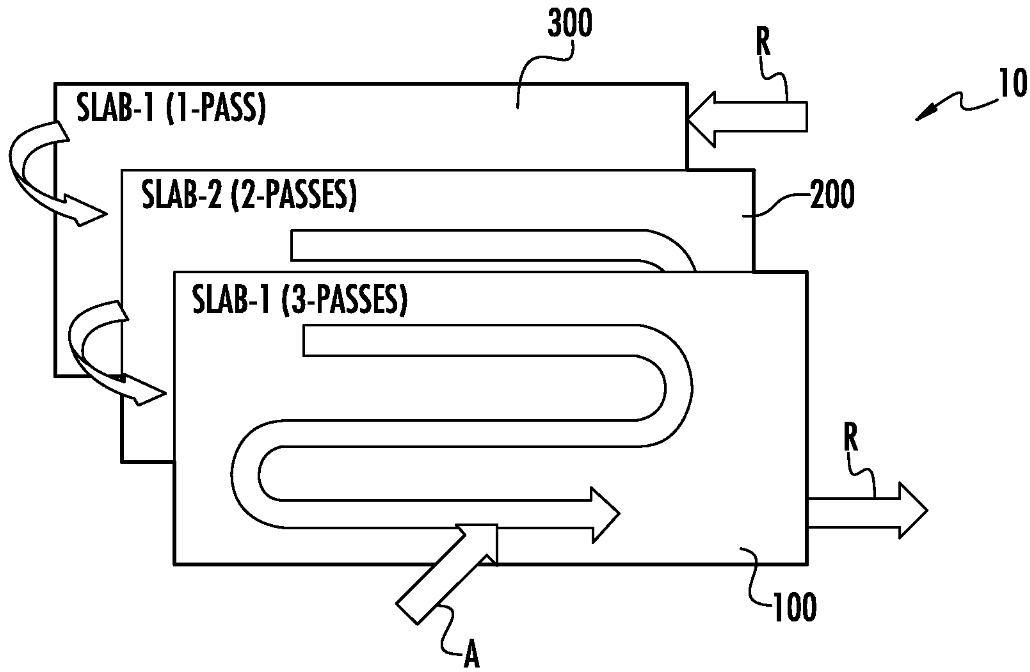
300
R
200
R
100

FIG. 11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R
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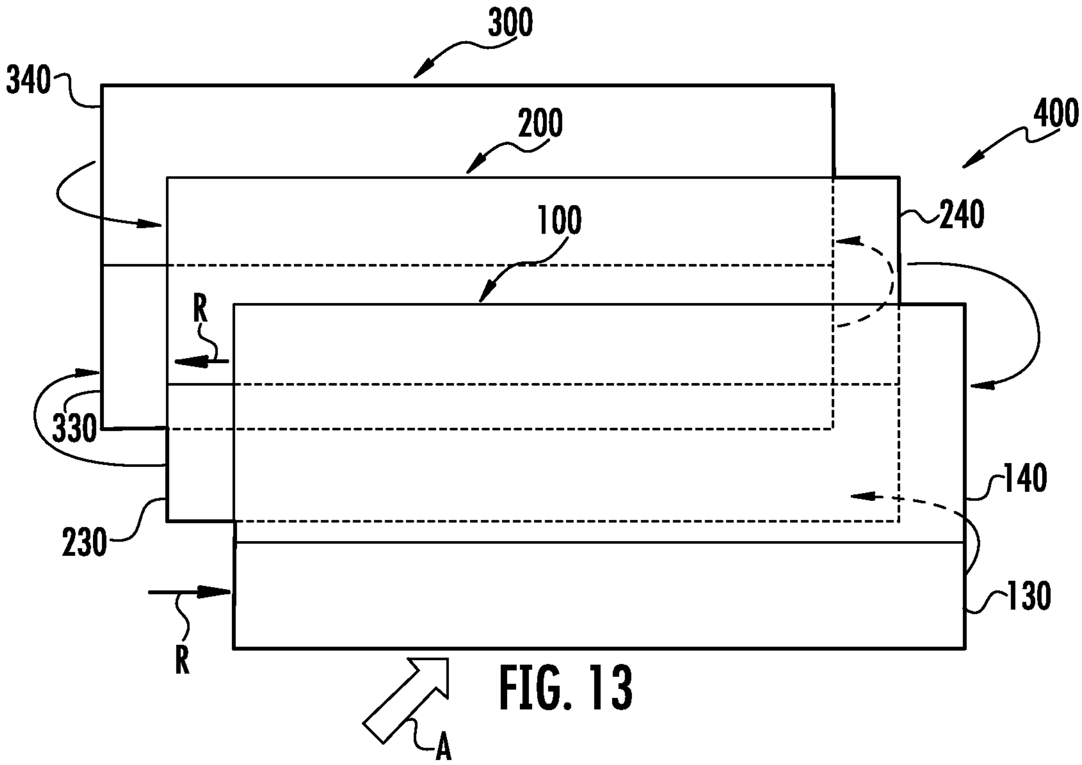
SLAB-1 (1-PASS)
SLAB-2 (2-PASSES)
200
SLAB-1 (3-PASSES)
R
100
A

FIG. 12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400
240
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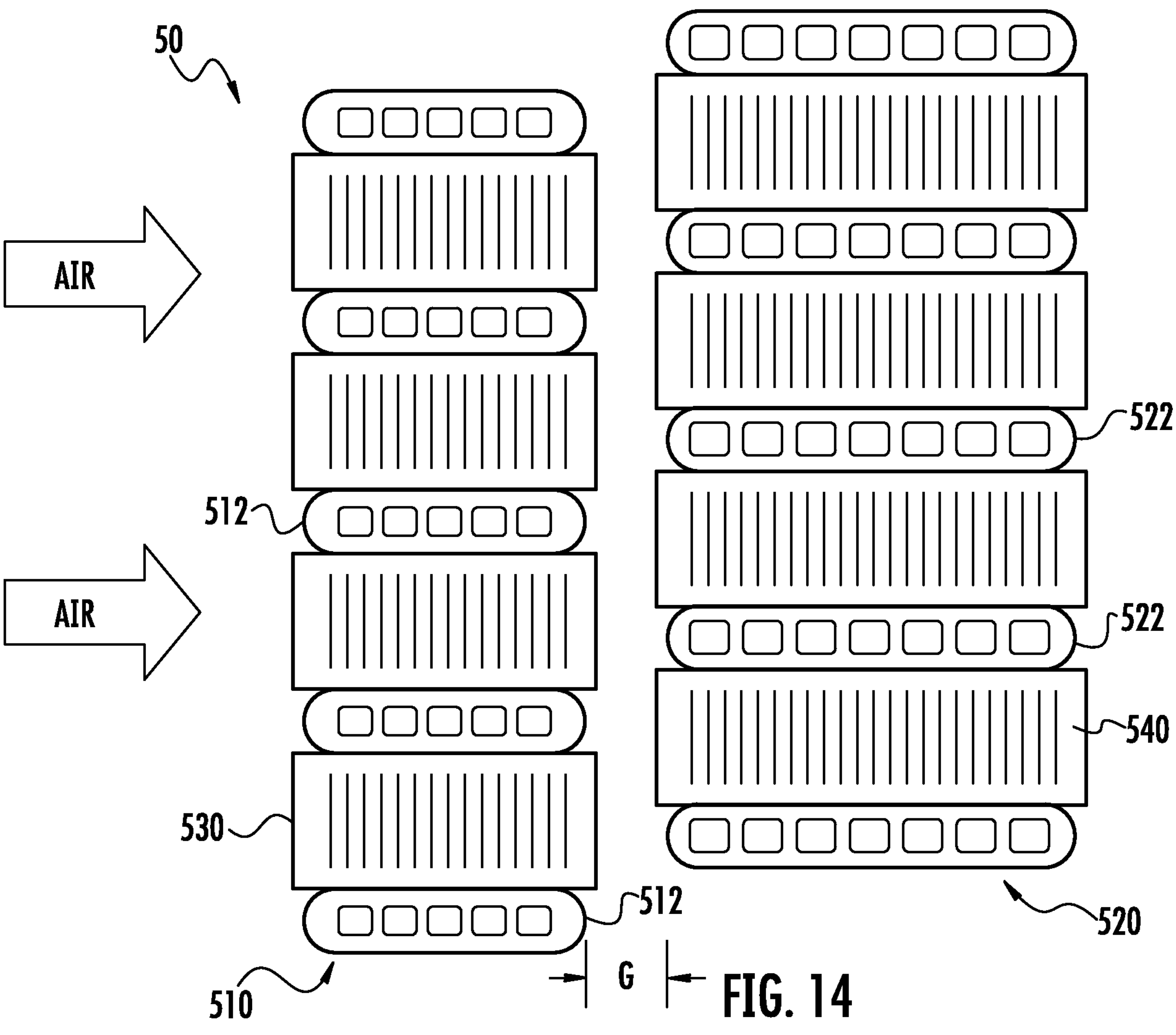
R
330
140
230
130
R
A

FIG. 13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AIR
AIR
522
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522
540
530
512
520
510
G

FIG. 14