July 29, 1969 N. P. THEOPHILOS ET AL 3,457,990

MULTIPLE PASSAGE HEAT EXCHANGER UTILIZING NUCLEATE BOILING

Filed July 26, 1967 4 Sheets-Sheet 1

Fig. 1.

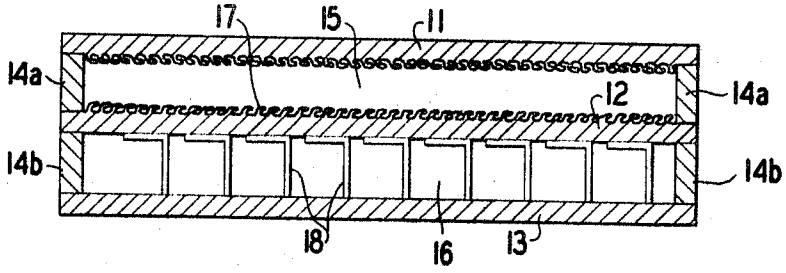


Fig. 2.

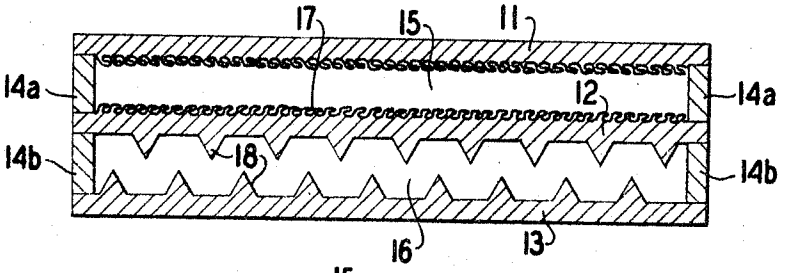


Fig. 3.

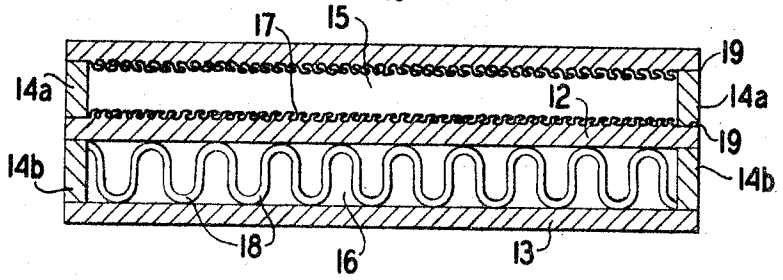
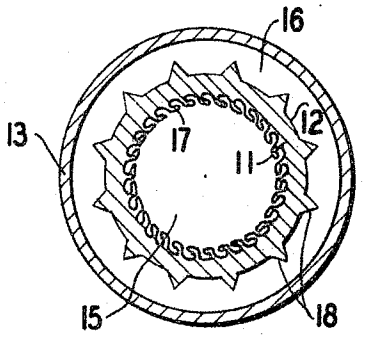


Fig. 5.



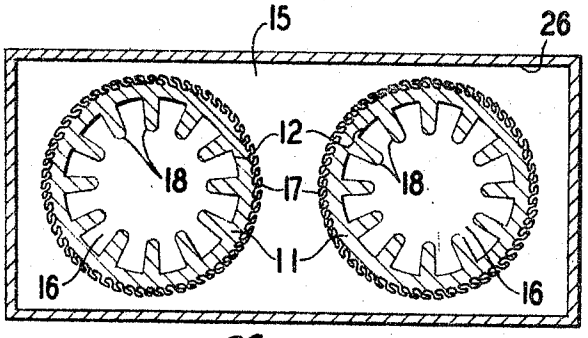


Fig. 6.

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MULTIPLE PASSAGE HEAT EXCHANGER UTILIZING NUCLEATE BOILING

Filed July 26, 1967 4 Sheets-Sheet 2

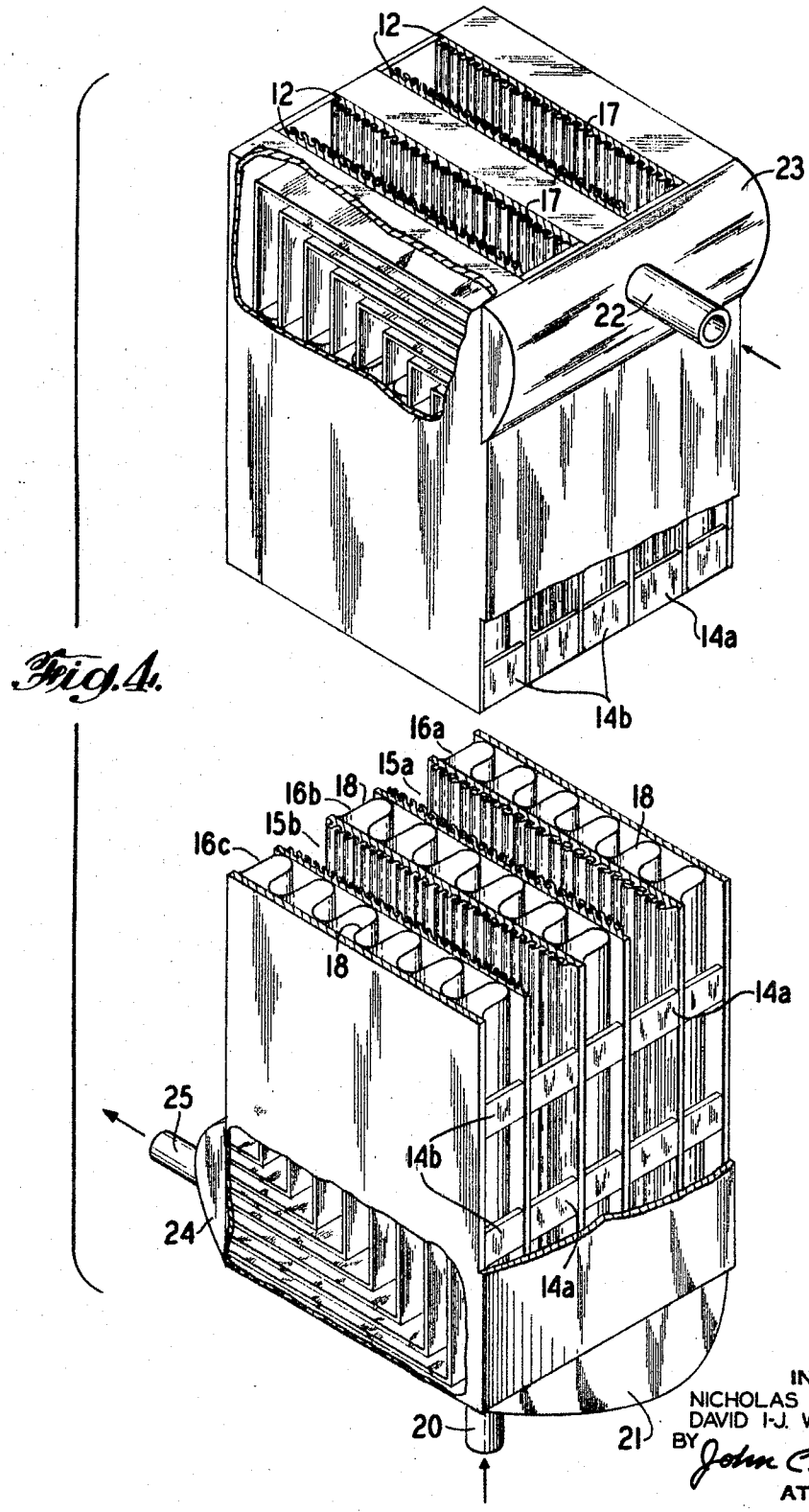


Fig. 4.

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MULTIPLE PASSAGE HEAT EXCHANGER UTILIZING NUCLEATE BOILING

Filed July 26, 1967 4 Sheets-Sheet 3

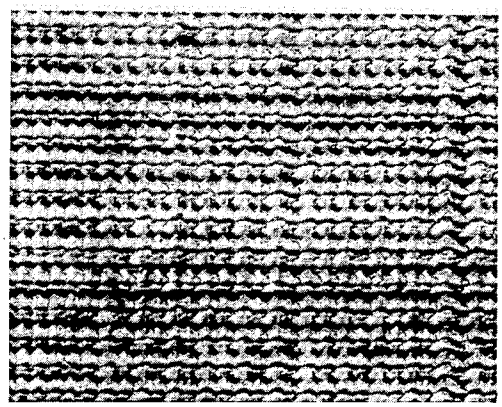


Fig. 7.

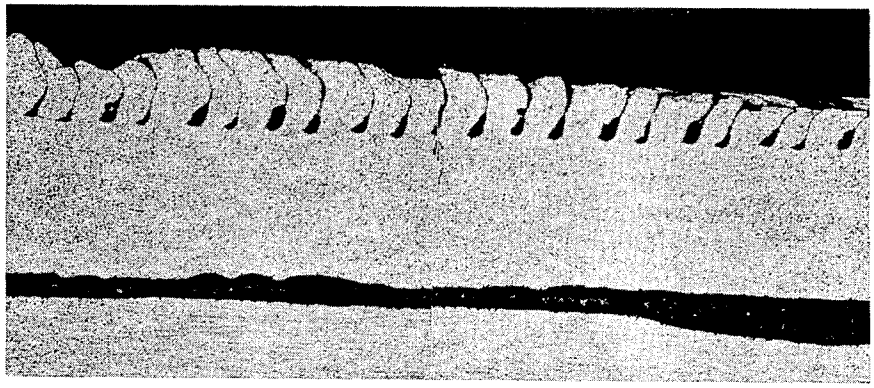


Fig. 8.

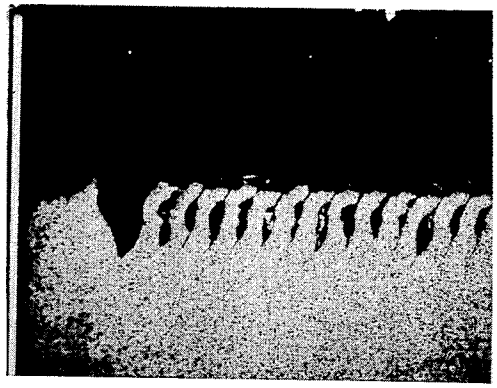


Fig. 9.

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MULTIPLE PASSAGE HEAT EXCHANGER UTILIZING NUCLEATE BOILING

Filed July 26, 1967 4 Sheets-Sheet 4

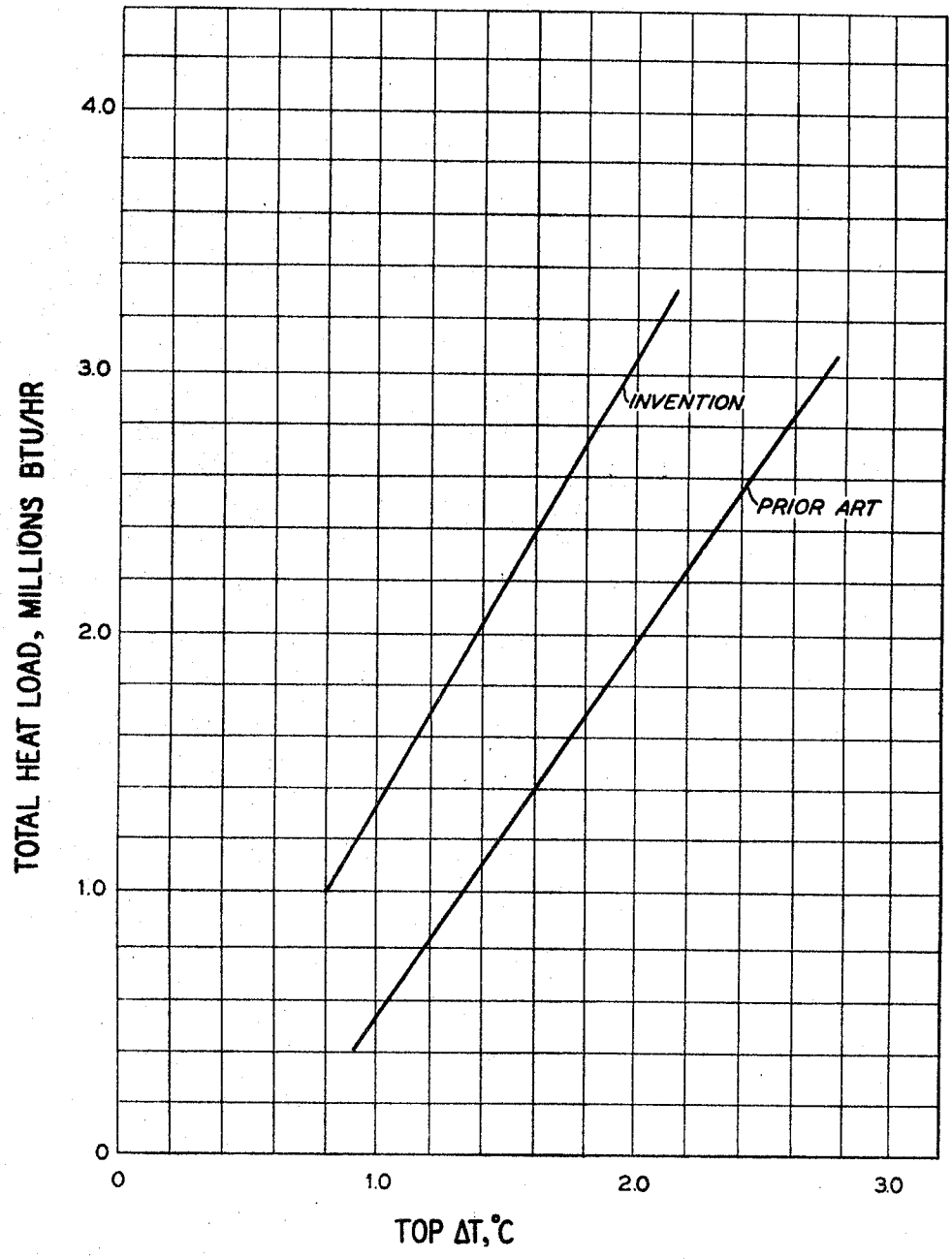


Fig. 10.

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United States Patent Office 3,457,990
Patented July 29, 1969

3,457,990

MULTIPLE PASSAGE HEAT EXCHANGER UTILIZING NUCLEATE BOILING

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Filed July 26, 1967, Ser. No. 656,234
Int. Cl. F28f 19/02, 13/18, 7/00
U.S. Cl. 165—133 13 Claims

ABSTRACT OF THE DISCLOSURE

A heat exchanger having a finless first passageway for boiling liquid and a finned second passageway for warmer condensing fluid provided with an intermediate wall having contoured layers on the boiling side including ridges, grooves, and reentrant cavities to hold vapor bubbles.

BACKGROUND OF THE INVENTION

This invention relates to improved apparatus for transferring heat from a warmer condensing fluid to a colder fluid capable of being boiled by such warmer fluid to form a disengaging vapor.

It is well known that apparatus provided with separate but thermally associated passageways each containing extended surfaces, may be used to exchange heat between condensing and boiling fluids. However, the need for extended surfaces in each passageway makes the material and fabrication costs of such heat exchangers extremely high. Moreover, their efficiency is limited by the characteristically long and indirect heat transfer path through the extended surfaces. Another limitation is the thermal conductivity of the metal used to form the extended surfaces. In some instances the selection of metals is limited to relatively low thermal conductivity materials by the corrosive nature of the fluids to be processed. Also, the use of extended surfaces on the boiling side of a heat exchanger may provide pockets for accumulation of contaminating and even combustible nonvolatile constituents of the boiling fluid.

Extended surfaces on the boiling side are particularly undesirable when the passageway is used to process boiling fluid under sub-atmospheric pressure e.g. sea water desalination. The high pressure drop characteristic of extended surface passageways requires high temperature differences to achieve the desired heat exchange, and high ΔT represents inefficient operation.

It is an object of this invention to provide heat exchange apparatus for boiling colder fluids and condensing warmer fluids which affords higher overall heat transfer coefficients than heretofore attained.

Another object is to provide such highly efficient heat exchange apparatus at significantly lower material and fabrication costs than presently available boiling-condensing equipment.

Still another object is to provide highly efficient heat exchanger apparatus for boiling and condensing fluids which does not require extended heat transfer surfaces on the boiling side.

Other objects and advantages of this invention will be apparent from the ensuing disclosure and the apparatus claims.

SUMMARY

This invention relates to a heat exchanger having spaced metal walls to form a first fluid passageway for colder boiling fluid and a second fluid passageway for warmer condensing fluid, with the first and second passageways separated by an intermediate wall for heat flow between the fluids.

In particular, contoured layers are integral with and may be formed from the intermediate wall bounding the first fluid passageway as the sole heat transfer surface therein. That is, extended heat transfer surfaces are not provided in such passageway. The contoured layers themselves are described and claimed in copending application Ser. No. 634,403 filed Apr. 7, 1967 in the name of L. C. Kun et al., now Patent No. 3,392,780 incorporated herein to the extent pertinent.

The contoured layers have a plurality of ridges separated by grooves provided at density greater than about 20 grooves per inch, with outer sections of such ridges partially deformed into the grooves such that a plurality of sub-surface cavities are formed therein to entrap bubbles which provide boiling nucleation sites. These cavities communicate with the outer surface of the contoured layers through restricted openings having smaller cross-sectional area than the largest cross-sectional area of the cavity interior. In this manner the restricted openings permit vapor egress from the cavities to the outer surface of the contoured layer and/or liquid ingress. The resulting vapor bubble rises through the liquid bath of the colder boiling fluid to the liquid-vapor interface for disengagement into the gas space above the boiling liquid in the first passageway, and is discharged therefrom.

At any particular moment when vapor is being emitted from certain cavities, other cavities are receiving liquid from the bath through their restricted openings to the outer surface of the contoured layers. Sub-surface openings are also provided in these contoured layers between the interiors of at least some adjacent cavities, thereby affording fluid communication between such cavities.

A multiplicity of spaced metal fin members are provided in the second fluid passageway. They are joined to the intermediate wall and extend transversely into the passageway.

To use this heat exchanger, a relatively warm condensible vapor is introduced in the second passageway and a relatively cold liquid is introduced in the first passageway. This liquid is either at its boiling temperature or sufficiently close to such temperature that boiling will be initiated and sustained by the heat introduced thereto through the intermediate wall from the condensing vapor. This vapor is channeled by the metal fin members in intimate flow relation to the intermediate wall, and the vapor-condensate flows along the fin members thereby transferring its heat of condensation to the fins as well as directly to the intermediate wall. The fins are joined to the intermediate wall, so that the condensing vapor's heat is transferred through such wall to the contoured layer in the first passageway. This heat sufficiently warms the liquid within at least some of the sub-surface cavities to form vapor bubbles therein. These bubbles in turn serve as boiling nucleation sites, and the boiling continues at such sites despite the release of bubbles. The latter emerge through the restricted openings, rise through the liquid bath in the first passageway and are released in the overhead disengaging space within the first passageway for discharge as discussed previously.

It has been unexpectedly discovered that remarkably high overall heat transfer coefficients are afforded by the heat exchange apparatus of this invention, despite the absence of extended surfaces in the first passageway where boiling occurs. These coefficients are on the order of 1.5 times the coefficients achievable with conventional heat exchangers employing extended surfaces on the boiling side.

This great improvement is achieved with uniform vapor distribution of across the first passageway, despite the elimination of extended surfaces which tend to break up discrete channels of liquid and vapor in boiling liquid

passageways. This uniformity is believed due to the formation of vapor bubbles in the contoured layers which are too small for agglomeration, so that a homogeneous liquid-vapor mixture emerges from the warm end of the first passageway.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is a schematic taken in cross-sectional elevation of a heat exchanger constructed according to the invention with flat walls.

FIG. 2 is a schematic taken in cross sectional elevation of another flat-walled embodiment.

FIG. 3 is a schematic taken in cross-sectional elevation of still another flat-walled embodiment.

FIG. 4 is an isometric view taken in cross-section of a multiple passageway heat exchanger embodying this invention.

FIG. 5 is a schematic taken in cross-section of the end of a cylindrical embodiment with the contoured layer on the inner side of the intermediate wall.

FIG. 6 is a schematic taken in cross-section of the end of another cylindrical embodiment with the contoured layer on the outer side of the intermediate wall.

FIG. 7 is a photomicrograph, magnification 20 fold, of the top surface of a cross-grooved contoured layer suitable for use in the boiling fluid passageway of this heat exchanger.

FIG. 8 is a photomicrograph, magnification 50 fold, of a cross-section of the FIG. 7 contoured layer taken in a vertical plane approximately along the line **8—8** in FIG. 7.

FIG. 9 is a photomicrograph taken in a vertical plane, magnification 40 fold, of a single-direction grooved contoured layer suitable for this invention.

FIG. 10 is a graph comparing the performance of this heat exchanger with the prior art for oxygen liquid boiling-nitrogen vapor condensing.

Corresponding items have been identified by the same numerals in the various figures for simplicity and cross-reference.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings, FIG. 1 shows a simple two-pass flat plate-type heat exchanger having first wall **11**, intermediate wall **12** and third wall **13**. The first and intermediate walls **11** and **12** are spaced apart by separator members **14***a* to form first fluid passageway **15** for the boiling fluid. The intermediate and third walls **12** and **13** are spaced apart by separator members **14***b* to form second fluid passageway **16** for the condensing fluid. Contoured layers **17** are preferably formed from and integral with intermediate wall **12** bounding first fluid passageway **15** as the sole heat transfer surface in this passageway. These layers are shown schematically and will be hereinafter discussed in greater detail in connection with FIGS. 7–10.

Metal fin members **18** are provided in the second fluid passageway **16** of the FIG. 1 heat exchanger, as strips bent in the longitudinal direction to form an included right angle. One section of each strip is joined to intermediate wall **12** and the other section is joined to the third wall **13**, the strips being preferably oriented with their longitudinal axes parallel to the direction of fluid flow through second passageway **16**. In the FIG. 1 embodiment, the condensable fluid is then introduced to and withdrawn from passageway **16** in the same direction as viewed. The condensate flows downwardly along the section of each strip **18** joining third wall **13** and collects over such wall for flow to the outlet end of passageway **16**.

At the same time the condensable fluid is being introduced to second passageway **16**, the liquid to be boiled is introduced to passageway **15** in the same direction as viewed and flows over contoured surface **17** toward the discharge end. Although both first and intermediate walls **11** and **12** are illustrated with contoured surfaces **17**, it should be understood that for a simple two-pass heat exchanger, only the intermediate wall **12** need be provided with surface **17**. This is because heat is transferred through intermediate wall **12** to the boiling liquid flowing over this wall in first passageway **15**. The vapor disengaged from the liquid-gas interface in first passageway **15** flows through the overhead gas space and is discharged therefrom.

The FIG. 2 embodiment is similar to FIG. 1, differing only in the nature of metal fin members **18**. Here they are sections raised from the intermediate and third walls **12** and **13** bounding second passageway **16**. The raised sections are illustrated with a triangular cross section, although other configurations would be suitable, e.g. rounded. These sections **18** are preferably longitudinally oriented in the direction of fluid flow with adjacent fins at spaced intervals in the lateral direction. It is also preferred to laterally space the raised sections on intermediate and third walls **12** and **13** in alternating sequence to provide maximum extended surface per unit volume of heat exchanger passageway. As illustrated in FIG. 2, adjacent sections **18** raised from intermediate wall **12** are spaced between adjacent sections **18** raised from third wall **13**. These sections are illustrated as being integral with and formed from the walls **12** and **13**, but alternatively may comprise separate pieces individually positioned and secured to such walls as by metal bonding. It should be understood that for a simple two-pass heat exchanger, only the intermediate wall **12** need be provided with fin members **18** attached thereto.

FIG. 3 illustrates another type of suitable finned surface **18**, in the form of a corrugated or continuous wave-type contour with adjacent crests and troughs joined to the intermediate and third walls **12** and **13**. Finned surfaces **18** may for example be joined to these walls using conventional furnace dip brazing procedures.

Spacers **14***a* and **14***b* such as the illustrated end bars are preferably brazed to the edges of walls **11**, **12** and **13**. It has been discovered however that difficulties arise when one attempts to braze these spacers directly to the contoured surfaces **17**. The brazing flux and filler metal migrate out of the joint due to capillary action of the adjacent contoured surface, and thus contaminate the contoured surface and may structurally weaken the brazed joint. Such migrating metal also tends to plug the subsurface cavities of the adjacent portions of contoured surfaces **17** and prevents their functioning as boiling nucleation sites. These problems are overcome by either omitting the contoured layers from the edges of walls **11** and **12** or alternatively by removing such layers from the immediate area of the joint. Either approach provides flat non-contoured sections **19** under and adjacent to the brazed joints. The flat sections should extend a small distance, e.g. ⅛-inch away from the joint edge to effectively prevent migration of braze or filler metal from the joint to the contoured surfaces.

It should be understood that the FIGS. 1–3 embodiments may be used to exchange heat between a warmer condensible fluid and a colder boiling fluid using either parallel or counter-current flow relation through first and second passageways **15** and **16**, the latter usually being preferred for greater efficiency. Another possible flow relation is to pass the fluids at right angles to each other, as for example by introducing the boiling liquid into passageway **15** through openings between adjacent spacers **14***a*.

FIGS. 1–3 illustrate simple two-pass heat exchangers, but in actual practice many more passageways are employed to process relatively large fluid flow throughputs. FIG. 4 illustrates a suitable multiple pass heat exchanger using the corrugated fins **18** of the FIG. 3 embodiment in the second passageways **16***a*–**16***c* processing the warmer condensible fluid. Each of these second passageways **16***a*–

16c is in heat exchange relation with at least one first passageway 15a–15b having contoured layers 17 on walls 11 and 12. For such larger multiple pass heat exchangers, intermediate separator members 14a may be used in the boiling pass if desired for structural support. In the FIG. 4 multiple passageway flat-walled heat exchanger the colder boiling liquid is introduced through inlet conduit 20 to manifold 21 at the lower end, and directed to first passageways 15a and 15b for upward flow therethrough. The resulting partially vaporized fluid is collected in a manifold at the upper end (not illustrated) and discharged from the exchanger for further processing. At the same time the warmer condensing fluid is supplied through conduit 22 to upper side manifold 23 which may for example be provided with distributor fins to uniformly divide the fluid among second passageways 16a, 16b and 16c. This fluid flows downwardly through the aforementioned passageways for cooling and condensation by heat exchange with the colder boiling fluid in first passageways 15a and 15b, employing the heat transferring intermediate walls 12. The resulting partially or completely condensed fluid at the lower end is collected in lower side manifold 24 and withdrawn through discharge conduit 25.

This invention is also suitable for use in cylindrical shaped heat exchangers, as for example the well-known finned tube variety illustrated in FIGS. 5 and 6. In FIG. 5 the first passageway 15 for colder boiling fluid is formed by tube 11 provided with contoured layer 17 on the inside surface of intermediate wall 12. The latter has fins 18 spaced from each other and attached to the outer surface of tube 11 around its periphery. These fins may for example have any of the configurations illustrated in FIGS. 1–3, and extend longitudinally from one end to the other end of the tube 11. Outer tube 13 is positioned around smaller tube 11 with the intervening space forming second passageway 16 for the condensing warmer fluid. In operation, the multiple tube assembly of FIG. 5 may be positioned vertically or inclined from the horizontal with the warmer condensing fluid introduced at the upper end and withdrawn as droplets at the lower end after downward flow along the extended fin surfaces. The colder boiling fluid may be introduced to inner tube 11 at the lower end and withdrawn as disengaged vapor at the upper end of the heat exchanger.

FIG. 6 illustrates another heat exchanger embodiment of this invention in which the contoured surfaces 17 are provided on the outer side of multiple tubes 11, and the first passageway 15 for boiling colder fluid is the space between these surfaces and the inner wall of surrounding casing 26. Fins 18 extend radially inward from the inner side of tubes 11 and are spaced around the tube perimeter to form the second fluid passageway 16 for condensing warmer fluid. In the FIG. 6 construction heat is transferred through the walls of tube 11 as the intermediate wall 12.

The contoured layers used in the heat exchanger of this invention are preferably formed by scoring the surface of the desired heat exchange wall such that the wall material is substantially displaced normal from the direction of scoring into adjacent ridges rather than being removed. When a scoring tool is used to form the contoured layer, the tool tends to displace the wall material upward from the wall surface and outward away from the tool as the latter moves across the wall surface such that grooves separated by ridges are formed in the wall material. The scoring tool is provided with a sharp projection that slices through the surface with the leading edge making a shallow cut becoming progressively deeper to the trailing edge. The latter makes the deepest cut into the wall and thereby establishes the groove depth.

If the desired heat exchange wall is to be cylindrical, a flat wall may be scored and then formed into a cylinder. Flat plates and sheets are conveniently scored by holding them firmly down against the flat bed of a planing mill and mounting the tool in conventional fashion on a fixed gantry above the work. The surface is scored as the bed moves the work horizontally under the tool. After each scoring stroke of the machine, the tool is indexed laterally a few thousands of an inch into position for forming the next adjacent groove. An alternative machine for scoring flat sheets is a shaper, on which the work is held firmly against the fixed bed and the tool moves horizontally over the work.

Alternatively the contoured layers may be formed by other cutting methods such as milling, a grooving procedure by which at least a portion of the metal is removed from the groove in the form of chips for shavings. Milling may be accomplished with either a rotary cutter, or with a tool pushed through the metal on a shaper or planer. In either case, the face of the tool which bites into the metal is blunt and angled upward with increased clearance provided towards the rear so as to remove the metal from the groove as cleanly as possible. The leading edge of the milling tool makes the deepest desired cut thereby establishng the groove depth, with the trailing edge becoming progressively shallower.

The contoured layers preferably comprise a plurality of substantially parallel ridges with adjacent ridges separated by first grooves, and a second plurality of depressions or grooves superimposed on the ridges at an angle to the latter's orientation, most suitably 90 degrees. The outer section of the ridges are partially deformed into adjacent groove such that the aforementioned cavities are formed in the grooves. This surface has a cross-grooved appearance.

The second set of depressions or grooves may be prepared by deforming techniques other than scoring or milling.

In some instances the first set of grooves may be cut without restricted openings from the cavity interiors to the outer surface of the wall, or alternatively the openings may not have as restricted cross-sectional areas as desired for effective vapor bubble entrapment. In either situation, a second deforming step may be used to partially smash in the ridge top surfaces and thereby reduce the cross-sectional area of the openings. This second deforming step may for example be performed by rolling a smooth member of circular cross-section across the surface at 90 degree orientation to the first set of grooves and ridges. Still anther technique for forming the desired vapor reentry cavity contour is by knurling the metal wall containing the first set of grooves and ridges, preferably at 90 degrees orientation thereto.

It may also be possible to prepare the contoured layers by methods other than metal deformation and removal. For example, multiple layers of screening or wires perhaps of different diameters and orientations might be applied to a substrate. Another possibility is to position a series of parallel spaced strands of corrodible or decomposable material on the substrate, followed by casting a layer of non-corrodible metal over and between the aligned strands. The latter could then be removed as by leaching, leaving cavities surrounded by the non-corrodible metal layer.

The preparative method must produce contoured layers having the previously described essential characteristics if the remarkably high boiling heat transfer coefficients are to be obtained. There must be a plurality of ridges separated by grooves provided at density greater than about 20 grooves per inch to form cavities adapted to entrap vapor bubbles and provide boiling nucleation sites. The cavities must communicate to the outer surface of the layers through restricted openings having smaller cross-sectional area than the largest cross-sectional area of the cavity interiors. Finally there must be sub-surface openings between at least some adjacent cavities providing fluid communication therebetween.

When boiling fluid is contacted with the contoured layer, vapor bubbles are permanently trapped within the cavities which serve continuously as nuclei for the for-

mation of vapor. It is believed that a thin liquid layer is maintained between a trapped vapor bubble and the adjacent metal surface defining the cavity constituting a very low heat transfer resistance between the metal and the liquid-vapor interface. This liquid film is replenished to sustain growth of the entrapped vapor bubbles as vapor escapes from the cavities through the restricted openings of the cavities. The extreme thinness of the liquid film within the cavities formed within the grooves is thought to account in large part for the very high boiling heat transfer coefficients achieved with the contoured layer.

An important variable of the contoured layers is the groove density. A relatively high groove density aids in the formation of smaller cavities which function better in boiling liquids having relatively low surface tensions such as liquid oxygen and nitrogen. For these two liquids, and for liquids having similar surface tensions, a groove density of between 120 and 250 grooves per inch is preferred. A relatively low groove density aids in the formation of larger cavities which function better in boiling liquids having relatively high surface tensions such as water. For this liquid, and for liquids having similar surface tensions, a groove density of between 20 and 120 grooves per inch is preferred. Cavities for boiling liquids having surface tensions intermediate water and liquid oxygen and nitrogen would be preferably formed in surfaces having groove densities intermediate the foregoing values.

The metal fin members may be molecularly integral with the intermediate wall (FIG. 2) or thermally metallurgically bonded thereto and formed by extrusion rolling, casting or machining. For relatively viscous fluids such as water, at least 10 fins per inch (measured parallel to flow of condensing fluid) are preferred to match the performance of the contoured layers on the boiling side of the heat exchanger. That is, with less than about 10 fins per inch, the overall performance of the heat exchanger will be limited by the condensing heat transfer coefficient. With less viscous fluids such as oxygen and nitrogen, at least 20 fins per inch are preferred for the same reason. The latter's higher fin density is for the reason that fluids of lower surface tension will penetrate into and drain from smaller spaces.

The contoured layers of FIGS. 7–9 were prepared from flat aluminum sheets held down on the bed of a planer with adhesive tape. Multiple circular scoring tools with a 30° included tip angle were mounted on an arbor bar and advanced transversely across the work by 0.006–0.008 inch before each stroke of the planer bed. The crosswise grooves are formed by repositioning the work 90° to the original position and repeating the scoring operation.

A portion of the contoured layers was impregnated in a plastic resin to more clearly show the sub-surface structure. This portion was vertically cross-sectioned at an angle of 5° to the normal of the first grooving, and one of the sectioned edges polished and microphotgraphed. The resulting FIG. 8 depicts the structure in different vertical planes, and permits one to construct a mental image of the interrelated groove-cavity—restricted opening—ridge structure by mentally arranging each ridge over the adjacent ridge from left to right.

It is significant to note that the ridges in the left side of FIG. 8 are inclined toward the left. In this manner each ridge is deformed over the groove to its left thereby forming the restricted opening to the outer surface. In the center right side of FIG. 8 the ridges and thus the restricted openings are more nearly vertically aligned over the grooves with less ridge deformation; also the cross-sectional area of the restricted openings is relatively larger. The contoured layers in the extreme right side of the figure are characterized by ridge deformation toward and over the groove to its right, again producing relatively small restricted openings.

In the FIGS. 7–8 contoured layer, the first and cross-grooves were scored into the sheet at a nominal depth of 0.007 inch. That is, the scoring tools were mechanically set to cut groves of this depth whereas the actual depth may have been slightly more or less depending on tool-work "play" and vibration. The scoring tools were positioned to cut 140 grooves per inch in both the first and cross directions. It should be appreciated however that in the cross-grooved embodiment of the contoured layers, the groove depth need not be the same in both directions.

Although the individual sections of the contoured layers are clearly recognized across the entire length of FIG. 8, this is an idealized model and all cavities need not be capable of retaining bubbles as long as some of the cavities communicate with the outer surface through restricted openings and communicate with other cavities through sub-surface connections.

The single-direction scored layer of FIG. 9 was prepared in the same general manner as the cross-scored layer of FIGS. 7–8. However, the grooves were cut at 0.0125 inch nominal depth and at nominal density of 230 grooves per inch. The contoured layer was formed by the scoring tool advancing from right to left with the last groove being formed at the left. The scoring tool deformed the near ridge (toward the direction of advancement) of an adjacent preceding groove into that preceding groove during the scoring of the next succeeding groove.

The differences in appearance between various contoured layers are due to such variables as groove density, method of forming, groove depth, speed of movement between tool and work, angle of tool inclination, the type of lubricant (if used), and tip configuration of the tool. For example, scoring to a greater depth at a particular groove density will result in a greater degree of deformation. Likewise, inclining the scoring tool over the first-scored material or alternatively over the next material to be scored at a greater angle, will also result in a greater degree of deformation. Further, a higher boiling coefficient can be obtained by cross-grooving than single direction grooving due to more metal deformation and re-entrant cavities produced with the former.

The advantages of this invention were illustrated in tests employing a heat exchanger using a contoured layer having the same specifications as the layer of FIGS. 7 and 8 and the corrugated finned surface of FIG. 3. This particular heat exchanger was formed of 0.03 inch thick aluminum sheets separated by spacers as illustrated in FIG. 4 so that adjacent sheets were 0.20 inch apart. Each first boiling and second condensing passageway was 0.20 inch thick, about 11 inches wide and 60 inches long, with both the first and intermediate walls having the contoured layers (140 grooves per inch at 0.007 inch depth, 90° cross-grooved at 140 grooves per inch). Each second condensing passageway was provided with 25 corrugated-shaped fins per inch width (perpendicular to the direction of fluid flow). The 0.008 inch thick aluminum fins extended between and were bonded to both the intermediate and third walls. The fins were perforated with $\frac{1}{32}$ inch diameter holes uniformly spaced to provide openings comprising 10% of the fin area for drainage and uniform pressure drop.

The assembled exchanger had a total of five passageways comprising three condensing and two intervening boiling passageways. It was vertically positioned in a casing with a surrounding space such that liquid oxygen flowed into the lower open end of the first passageways from a liquid bath partially immersing the passageways. Oxygen liquid-vapor mixture was discharged through the upper end of the first passageways into a disengaging space. The oxygen vapor portion was withdrawn and metered at about 6 p.s.i.g., and the oxygen liquid portion returned to the bath by gravity. The nitrogen vapor at about −178° C. and 65 p.s.i.g. was introduced to the upper end of the second finned passageways and flowed

downwardly countercurrent to the upwardly flowing oxygen for partial vaporation of the latter. The resulting nitrogen liquid was discharged at the heat exchanger lower end and withdrawn at about —178° C. Auxiliary heat exchange equipment was provided so that the nitrogen and oxygen were continuously recirculated in a closed circuit.

The heat exchanger was operated at various liquid oxygen levels in the vertically aligned first passageways between 45% and 75% of the passageways length. It was found that the liquid level did not have a pronounced effect on the heat transfer coefficients. The heat exchanger was also operated at various heat loads between about 1.0 and 3.3 million B.t.u. per hour, and the temperature difference (ΔT) necessary to maintain such loads was measured at the top end.

The resulting data is summarized in the performance curve of FIG. 10, and may be compared with the corresponding curve for a widely used commercially available heat exchanger construction. The latter comprised aluminum sheets spaced 0.20 inch apart with corrugated finned surfaces extending across both the boiling oxygen and condensing nitrogen passageways at density of 12 and 25 fins per inch width, respectively.

Comparing the performance of these two heat exchangers at a heat load of 1.0 million B.t.u. per hour, the present construction requires a 0.8° C. ΔT whereas the prior art unit requires a 1.3° C. ΔT, or about 60% greater temperature difference. At 2.0 million B.t.u. per hour heat load the present construction requires a 1.4° C. ΔT whereas the prior art unit requires a 2.0° C. ΔT—about 40% greater temperature difference. Another comparison may be made by comparing the heat loads at the same temperature differential, e.g. 1.3° C. ΔT. The prior art unit affords a heat load of 1.0 million B.t.u. per hour whereas the instant heat exchanger provides a heat load of 1.85 million B.t.u. per hour—an 85% improvement. This means that the heat exchanger of the invention may be operated at a significantly lower temperature difference with attendant lower power costs to achieve a desired heat transfer rate. Stated otherwise, the improvement may be reflected as a substantially higher heat transfer rate for a given temperature difference.

The unexpected nature of the improvement demonstrated in the aforedescribed test is emphasized by comparing the heat exchange surface areas in the boiling passageways of large size heat exchangers scaled up from the FIG. 10 performance curves. A flat sheet type unit with the contoured layers having the overall dimensions of 25 inches thick x 25 inches wide x 60 inches long would have a first passageway surface area of 790 square feet. The same size heat exchanger without the contoured layers but with 12 corrugated parallel fins per inch width would have a first passageway surface area of 2750 square feet. Based on the prior art teaching that higher surface area affords greater heat transfer efficiency, one would expect the prior art unit to be superior. It is surprising that the heat exchanger of this invention transfers at least 1.5 times as much heat with only 29% as much surface area.

Another test unit comprising two fluid passageways was constructed using a 3/16-inch thick copper plate, enclosed on both sides to form a first passageway for boiling water and a second passageway for condensing steam. Each passageway was 2½-inches wide and 12 inches long. The boiling side of the copper plate was first scored with 100 grooves per inch at 0.012 inch depth, using a tool having 30° included angle inclined at 5° away from the direction of feed (toward previously cut grooves). The same boiling side was then cross-scored at 100 grooves per inch at 0.012 inch depth at 90° angle to the first direction of grooving. On the condensing side of the same copper plate, longitudinal grooves 0.125 inch deep x 0.015 inch wide were milled into the plate at 0.028 inch pitch. This milling produced integral, parallel and rectangular shaped fins 0.010–0.013 inch thick x 0.125 inch high and spaced 35 fins per inch. Also on the condensing side, cross channels about 3/32 inch wide were milled approximately 1 inch apart and inclined at 45° angle with the main grooves to provide drain channels for the condensate to flow over to one side of the passageway. Overall heat transfer coefficients of about 1000 B.t.u./hr./ft.^{2}/°F. were obtained. For example at a heat flux of 10,000 B.t.u./hr./sq. ft., the overall ΔT was 11.2° F. (6.2° C.). At a heat flux of 5,000 B.t.u./hr./sq. ft., the overall ΔT was 5° F. (2.8° C.).

These tests demonstrate that the apparatus of this invention affords superior heat transfer between fluids having both low surface tension (oxygen and nitrogen) and high surface tension (steam and water), and the exchanger may be constructed of different metals as for example copper and aluminum.

Although certain embodiments have been described in detail, it will be appreciated that other embodiments are contemplated along with modifications of the disclosed features, as being within the scope of the invention.

What is claimed is:

1. In a heat exchanger having spaced metal walls to form a first fluid passageway for colder boiling fluid and a second fluid passageway for warmer condensing fluid with the first and second passageways separated by an intermediate wall for heat flow between the fluids, the improvement comprising:

(a) contoured layers integral with said intermediate wall extending into said first fluid passageway as its sole heat transfer surface, having a plurality of ridges separated by grooves provided at density greater than about 20 grooves per inch, with outer sections of said ridges partially deformed into said grooves such that a plurality of sub-surface cavities are formed therein to entrap vapor bubbles to provide boiling nucleation sites, said cavities communicating with the outer surface of said contoured layers through restricted openings having smaller cross-sectional area than the largest cross-sectional area of the cavity interiors providing communication between the interiors of said cavities and the outer surface of said contoured layers for vapor egress and liquid ingress, said grooves and cavities being formed such that sub-surface openings are provided between at least some adjacent cavities providing fluid communication therebetween; and

(b) a multiplicity of spaced metal fin members in said second fluid passageway being joined to said intermediate wall and extending transversely into the passageway.

2. A heat exchanger according to claim **1** in which said metal walls including said intermediate wall are flat plates.

3. A heat exchanger according to claim **1** in which said metal walls including said intermediate wall are flat plates, and separator members are positioned against noncontoured sections of said intermediate wall to form said first and second passageways.

4. A heat exchanger according to claim **1** in which said metal walls are flat plates, and separator members are positioned against and brazed to outer noncontoured edges of said intermediate wall to form said first and second passageways.

5. A heat exchanger according to claim **1** in which said ridges of contoured layers *a* comprise metal displaced from said grooves and sub-surface cavities.

6. A heat exchanger according to claim **1** in which a plurality of second grooves at density of greater than about 20 grooves per inch are superimposed on said ridges of contoured layers *a* at an angle to the orientation of said ridges.

7. A heat exchanger according to claim **1**, in which said grooves are generally parallel to each other and provided at density between about 120 and 250 per inch, a plurality of second grooves at density of between about 120 and 250 per inch are superimposed on said ridges of contoured layers *a* parallel to each other and at about

90 degrees orientation to said ridges and the ridges comprise metal displaced from the grooves and subsurface cavities.

8. In a heat exchanger having at least three flat metal walls spaced apart by separator members to form a first fluid passageway bounded by a first wall and an intermediate wall, and a second fluid passageway bounded by said intermediate wall and a third wall for heat transfer between a colder boiling fluid in said first passageway and a warmer condensing fluid in second passageway, the improvement comprising:

(a) contoured layers formed from and integral with said intermediate wall extending into said first fluid passageway as the sole fluid flow obstruction in such passageway other than said separators, having a plurality of ridges separated by first grooves generally parallel to each other provided at density between 120 and 250 grooves per inch with the ridges composed of metal displaced from said grooves and with outer sections of said ridges partially deformed into said first grooves, and a plurality of second grooves generally parallel to each other provided at density between 120 and 250 grooves per inch being superimposed on said ridges at an angle of about 90 degrees to the orientation of said ridges, said ridges and first and second grooves being shaped such that a plurality of sub-surface cavities are formed in said first grooves adapted to entrap vapor bubbles to provide boiling nucleation sites, said cavities communicating with the outer surface of said contoured layers through restricted openings having smaller cross-sectional area than the largest cross-sectional area of the cavity interiors providing communication between the interior of said cavities and the outer surface of said contoured layers for vapor egress and liquid ingress, said grooves and cavities being formed such that sub-surface openings are provided between at least some adjacent cavities providing fluid communication therebetween;

(b) as said separator members, a multiplicity of solid metal bars positioned against and brazed to flat non-contoured edges of said three metal walls with adjacent bars between said first and intermediate walls, and between said third and intermediate walls being aligned with each other perpendicular to said three metal walls; and

(c) a multiplicity of corrugated metal fin members in said second fluid passageway having a continuous wave type contour with adjacent crests and troughs brazed to said intermediate and third walls, being longitudinally aligned parallel to condensing fluid flow with at least 20 fins per inch.

9. A heat exchanger according to claim 1, in which said grooves are generally parallel to each other and provided at density between about 20 and 120 per inch, and a plurality of second depressions at density of between about 20 and 120 per inch are superimposed on said ridges of contoured layers *a* parallel to each other and at about 90 degrees orientation to said ridges and the ridges comprise metal displaced from the grooves and sub-surface cavities.

10. A heat exchanger according to claim 1, in which said spaced metal walls including said intermediate walls are cylindrical.

11. A heat exchanger according to claim 1 in which said spaced metal walls including said intermediate wall are cylindrical, said contoured layers *a* are inside the cylinder and the spaced metal fin members are outside said cylinder.

12. A heat exchanger according to claim 1 in which said spaced metal walls including said intermediate wall are cylindrical, said contoured layers *a* are outside the cylinder and the spaced metal fin members are inside said cylinder.

13. In a heat exchanger having at least three flat metal walls spaced apart by separator members to form a first fluid passageway bounded by a first wall and an intermediate wall, and a second fluid passageway bounded by said intermediate wall and a third wall for heat transfer between a colder boiling fluid in said first passageway and a warmer condensing fluid in second passageway, the improvement comprising:

(a) contoured layers formed from and integral with said intermediate wall extending into said first fluid passageway as the sole fluid flow obstruction in such passageway other than said separators, having a plurality of ridges separated by first grooves generally parallel to each other provided at density between 20 and 120 grooves per inch with the ridges composed of metal displaced from said grooves and with outer sections of said ridges partially deformed into said first grooves, and a plurality of second depressions generally parallel to each other provided at density between 20 and 120 grooves per inch being superimposed on said ridges at an angle of about 90 degrees to the orientation of said ridges, said ridges and first grooves and second depressions being formed such that a plurality of sub-surface cavities are formed in said first grooves adapted to entrap vapor bubbles to provide boiling nucleation sites and sub-surface openings are provided between at least some adjacent cavities providing fluid communication therebetween, said cavities communicating with the outer surface of said contoured layers through restricted openings having smaller cross-sectional area than the largest cross-sectional area of the cavity interiors providing communcation between the interior of said cavities and the outer surface of said contoured layers for vapor egress and liquid ingress;

(b) as said separator members, a multiplicity of solid metal bars positioned against and brazed to flat non-contoured edges of said three metal walls with adjacent bars between said first and intermediate walls, and between said third and intermediate walls being aligned with each other perpendicular to said three metal walls; and

(c) a multiplicity of spaced metal fin members in said second fluid passageway being joined to said intermediate wall and extending transversely into the passageway and longitudinally aligned parallel to condensing fluid flow at density of at least 10 fins per inch.

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