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54 **Apparatus and process for heat transfer.**

57 Apparatus and a process of heat transfer employ a fluid flow passage in which the flow consists of non-turbulent boundary-layers adjacent the surfaces and a non-turbulent core-layer between the boundary-layers and interfacing therewith. An interrupter-structure is disposed within the flow passage and interrupts the full development of the boundary-layers at a multitude of spaced spots, leaving the heat transfer surfaces unaltered, unmodified and uninterrupted, so that the boundary-layers cannot increase in thickness but will partially separate from the surfaces and mix non-turbulently with the core-layer to effect the required heat transfer between the walls and the fluid. The interrupter-structure preferably consists of a plurality of spheres, or a sheet in which spherical, elliptical, drop-shaped or egg-shaped protrusions have been formed, all of which can be obtained readily on the market place at very low cost. This is to be contrasted with prior art apparatus which increase the heat exchange by rendering the fluid turbulent e.g. by turbulence promoters or roughening, ridging or interrupting the heat transfer walls. The increased turbulence increases heat transfer but produces a proportionally much greater increase in pumping power that makes these proposals uneconomic for many applications.

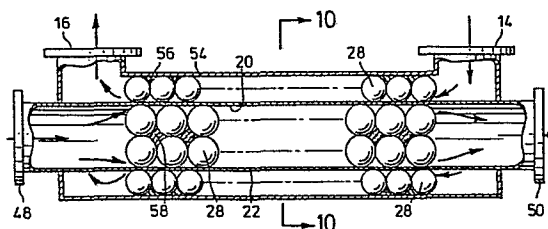


FIG. 11

APPARATUS AND PROCESS FOR HEAT TRANSFERField of the Invention

This invention is concerned with new apparatus for heat transfer and with a new process for heat transfer, as employed in such apparatus.

5 Review of the Prior Art

It is a constant endeavour in the field of heat transfer to improve the efficiency of heat transfer processes in order to improve the efficiency and also if possible lower the cost of the apparatus employing the improved process. To this end a number
10 of prior proposals have been made among which are:

a) reducing the thickness of the boundary-layers of the fluid flowing in a passage by the promotion of turbulence in the fluid flow, for example by roughening the flow passage walls and/or the provision of turbulence promoters in the passage.

5 b) the induction of boundary-layer separation from the heat transfer surfaces by use of curved or wavy heat transfer surfaces and

c) the interruption of the heat transfer surfaces, as with the so-called "split-fin" apparatus.

20 A frequent serious problem with such proposals is that, although the promotion of turbulence or the interruption of the heat transfer surfaces in laminar flow do increase the heat transfer per unit area, they also cause a disproportionate increase in the pumping power per unit area required to maintain the fluid
15 flow at the required rate, because of the increased turbulence and inefficient laminar flow diffusion mixing, and a consequent considerable increase in manufacturing cost, with the result that the overall economy of the system is reduced. In

commercial practice therefore the undesirable results prevent the adoption of such proposals in many cases unless there is an overriding need, for example, for compactness in size.

As an example of proposal a) British Patent Serial No:
5 1,172,247 issued to Hugh Eddowes and Peter Ernest Goss discloses a heat exchange apparatus in which a flow passage formed between parallel plates is provided with a structure consisting of crossed rods or woven wire mesh in order to promote turbulence in the flow. U.S. Patent 1,862,219 issued to J.M. Harrison
10 discloses another structure in which expanded metal is used as a fluid deflector to thin out the boundary-layer. A large number of other so-called "turbulence promoters" have been proposed hitherto for this purpose.

As an example of proposal b) the publication "Heat Transfer Handbook" by A.P.V. Company Inc. of Tonawanda, New York, provides pumping power vs. heat transfer data and describes the way in which an improvement can be achieved in heat transfer by
5 induction of turbulent fluid flow even at low Reynolds numbers, this induction being produced by use of curved or wavy heat transfer plates which are stacked together with interposed gaskets to constitute the so-called "plate and frame" exchangers. Other examples are described at pages 216 and 217 in the publication by
.0 W.M. Kays and A.L. London "Compact Heat Exchangers" 2nd edition, McGraw Hill Series in Mechanical Engineering, New York, 1964.

As an example of proposal c) U.S. Patent Serial No: 2,360,123 issued to George W. Gerstung and Hiram Walker discloses a heat transfer apparatus employing split corrugated fins, as
5 does also the above-mentioned "Compact Heat Exchangers" at page 212. With this arrangement very high coefficients of heat transfer are obtained while the flow is laminar by keeping the fin length very short through slitting and off-setting. This allows the

maximization of heat transfer within the developing boundary layers, since these are very thin and close to the leading edges of the split fins, the splitting and offsetting preventing development of thick boundary layers. The mixing with the core-
5 layers occurs mainly by conduction through the fluid and therefore extremely small hydraulic radii are necessary for acceptable mixing efficiencies. Quite frequently slight burrs develop when the fins are cut, making the flow turbulent at quite low Reynolds numbers with an associated higher friction drag. The results in
10 both cases are high pumping power requirements per unit of heat transfer surface and very high cost of manufacture.

Definition of the Invention

It is therefore an object of the invention to provide a new process for heat transfer by which the heat transfer can be
15 increased without a corresponding disproportionate increase in pumping power.

It is also an object of the invention to provide heat transfer apparatus of a new type in which the heat transfer can be increased without a corresponding disproportionate increase in
20 pumping power.

More specific objects are to provide new heat transfer apparatus and processes in which the heat transfer is increased with avoidance of turbulence in the presence of laminar wake-interference flow.

25 In accordance with the present invention there is provided a process of heat transfer including establishing in a fluid flow passage, comprising at least one heat transfer surface, a non-turbulent fluid flow consisting of a non-turbulent boundary-

layer immediately adjacent each passage surface, and a non-turbulent core-layer interfacing with the resultant boundary layer or layers;

interrupting non-turbulently the full development of the boundary-layer of each heat transfer surface at a plurality of spaced interruption spots to cause non-turbulent separation of parts of the boundary-layer from the respective heat transfer surface at those interruption spots and their mixing with the core-layer, to effect heat transfer between the heat transfer surface, its respective boundary-layer and the core-layer.

Also in accordance with the invention there is provided apparatus for heat transfer between two fluids comprising:

a body providing a fluid flow passage comprising at least one flat heat transfer surface;

the parameters of said passage in relation to the fluid flowing therein being such that the said fluid flow will take the form of a non-turbulent boundary layer or layers immediately adjacent to the respective surface or surfaces, and a non-turbulent core-layer interfacing with the boundary-layer or layers; and

a fluid flow interrupter structure disposed within the flow passage, interrupting non-turbulently the full development of at least the boundary-layer of the flat heat transfer surface at a plurality of spaced interruption spots, whereby parts of the interrupted boundary-layer will separate non-turbulently from the heat transfer surface between the said interruption spots and mix with the core-layer to effect heat transfer between the heat transfer surface, its respective boundary-layer, and the core-layer.

Description of the Drawings

Apparatus and processes which are particularly preferred embodiments of the invention will now be described, by way of example, with reference to the accompanying diagrammatic drawings wherein:

FIGURE 1 is a perspective view of a heat transfer apparatus, wherein the heat transfer is from a fluid flowing within the apparatus to the ambient atmosphere surrounding the apparatus;

FIGURE 2 is a similar view to Figure 1, but with the fins, side and end walls removed to show one preferred form of fluid flow interrupter structure within its interior;

FIGURE 3 is a section taken on the line 3-3 of Figure 2 and showing the fluid flow obtained inside the apparatus;

FIGURES 4a, 4b, 4c and 4d are plan views of a small portion of different interruption structures to show the respective forms that can be taken thereby;

FIGURE 5 is a schematic cross-section to illustrate the preferred form of the interrupter structure;

FIGURE 6 is a view similar to Figure 2 to show another preferred form of fluid-flow interrupter structure;

FIGURES 7a and 7b are respective perspective views to illustrate the adaption of the structures of the invention to a heat exchanger employing two confined fluid flow paths in heat exchange relation with one another;

FIGURE 8 is a perspective view of an embodiment comprising a flow passage of square transverse cross-section and wherein the flow interruption structure is a plurality of spheres;

FIGURES 9a and 9b are transverse cross-sections through two further embodiments wherein the flow passages are of circular cross-section and showing two different arrangements of interrupter structure comprising a plurality of spheres;

5 FIGURE 10 is a transverse cross-section through a single tube-in-shell heat exchanger of the invention;

FIGURE 11 is a longitudinal cross-section on the line 11-11 of Figure 10;

10 FIGURE 12 is a longitudinal cross-section on the line 12-12 of Figure 13 through a multiple tube shell-and-tube heat exchanger of the invention in which the tubes are of circular cross-section;

FIGURE 13 is a transverse cross-section on the line 13-13 of Figure 12;

15 FIGURE 14 is a longitudinal cross-section similar to Figure 12 and on the line 14-14 of Figure 15, but with a different heat exchange structure surrounding the tubes;

FIGURE 15 is a transverse cross-section on the line 15-15 of Figure 14; and

0 FIGURE 16 is a graph to show the relative performance ranking of heat exchanger surfaces of the invention as compared with surfaces from prior art tubulus and plate heat exchangers.

Description of the Preferred Embodiments

The simple convection-type heat transfer apparatus 10 of Figure 1 is intended for the transfer of heat carried by a liquid fluid, such as oil or water, to the gaseous ambient atmosphere; such apparatus is commonly used for example as an oil or water cooler. The apparatus consists of a hollow body 12 providing a parallel-walled flow passage containing a fluid flow interrupter structure to be described below. The liquid fluid is fed into the apparatus via an inlet pipe 14 and discharged there-
10 from via an outlet pipe 16. The exterior of the body may be provided in known manner with spaced parallel fins 18 for more efficient heat transfer to the ambient air.

The interior of the body 12 provides a non-turbulent fluid flow passage comprising two spaced parallel facing heat-transferring wall surfaces 20 (Fig. 3) provided by the walls 22, between which wall surfaces the liquid fluid flows. The passage is completed by two side walls 24 and the enclosure is completed by two transition pieces 26 which progressively change the circular cross-section of the pipes 14 and 16 to the rectangular
20 cross-section of the flow passage. In this embodiment a fluid flow interrupter structure disposed within the passage consists of a plurality of densely packed spheres 28 of a material that will be unaffected by the fluid, such as metal, glass or porcelain, the packing being such that the spheres contact one another. The
25 diameter of the spheres is such that they are each in point contact with the opposed heat transferring wall surfaces 20. Since in this embodiment the spheres are touching one another they may be joined to each other at their points of mutual contact to

form a unitary structure. In other embodiments they may be packed at a lower density at which they are spaced from one another, for example, by an interposed apertured plate having the spheres disposed in the apertures thereof. Other variations will
5 be described below.

It is known to those skilled in the art that fluid flowing within a passage has boundary-layers 30 immediately adjacent the surfaces 20, which act to insulate the wall surfaces from the main body of the fluid flowing in a core layer 32
0 between and interfacing with the boundary layers 30, and which therefore reduce the heat transfer between the surfaces 20 and the core layer 32. Corresponding boundary layers 30 are also present on the surfaces of the spheres 28. It is also known that an unobstructed boundary layer increases progressively in thickness
5 in the direction of fluid flow, which will increase its insulating effect. As described above, proposals have been made hitherto to disrupt the boundary layers by roughening or ridging the surface over which they flow, but such proposals have the effect of also increasing to a disproportionately greater extent the pumping
0 power required.

In apparatus of the invention the boundary layers 30 are interrupted in a "spot-wise" manner at spaced spots 34 by means of the fluid flow interrupter structure interposed between the heat transfer surfaces, while maintaining a non-turbulent fluid
5 flow in the core 32. In the apparatus of the invention not only are the heat transfer surfaces 20 not roughened, etc., but on the contrary they are made as smooth as is economically possible, to the extent that in many embodiments the surfaces 20 will be

polished to the desired degree of smoothness. The disruption of the boundary layers 30 at the multitude of spaced spots 34 ensures that they stay thin, while the manner of their disruption ensures that turbulence is avoided that would cause unduly high friction drag. The polishing of all surfaces including those of the spheres also assists in the desired minimizing of the friction drag.

Thus, the invention may be regarded as comprising a fluid flow system for improving the ratio of convective heat transfer to friction power per unit heat transfer surface area by sandwiching specially shaped interrupting and mixing-structures of low friction drag immediately adjacent a smooth heat transfer surface using hydraulic radii that guarantee total laminar flow. The mixing structures contain cellular voids, which are connected with one another, in each of which the fluid rotates spiral-like as a single laminar eddy. These eddies are very efficient means of mixing laminar streams, and preferably are obtained by coinciding a wake eddy downstream of an interruption point with an advance eddy upstream of a subsequent interruption point so as to produce wake-interference flow, which provides the highest efficiency. The boundary layers 30 on the curved surfaces of the mixing-structures are fairly thick, whereas the boundary layers of the flat heat transfer surfaces, situated opposite the mixing-structure surfaces, remain very thin on average because they are reduced regularly and spotwise at the large number of contact points between the surface of the mixing-structure and the heat transfer surface, and are in addition exposed to the highest local velocities which occur predominantly very close to the flat heat

transfer surface. This allows rapid heat flow through the heat transfer surface.

It is also believed that efficiency is improved because the velocity gradients adjacent to the flat heat transfer surfaces are much larger than over the curved surfaces of the interrupting mixing structure. These velocity gradients are, moreover, maintained largely at the heat transfer surfaces by virtue of the regularly spaced flow interruptions occurring at the large number of contact points between the mixing structure and the heat transfer surface. These interruptions also cause the flow to swirl at high velocity toward the heat transfer surface. Since it is well known from numerous experiments and theoretical analysis (see, e.g. pp 422-423, Principles of Heat Transfer, 3rd edition by F. Kreith, Publishers Harper and Row, New York, 1976) that temperature gradients are proportional to velocity gradients, the interrupting mixing structure which produces repeated steep velocity gradients similar to so-called "entrance effects", also promotes increased heat transfer while the flow remains laminar.

The general direction of flow of the fluid is indicated by arrows 36 and the flow interrupter structure causes the production of laminar flow eddies 38 of shape and rotational frequency that depend upon the geometry of the structure. Wake-eddies will be produced around the spots of interruption downstream of the flow, while advance eddies will be produced upstream of the flow. If the spacing of the interruption spots 34 is made such that the advance- and wake-eddies of immediately successive spots coincide, then wake-interference flow is obtained whereby, in the absence of turbulent friction-drag, very efficient non-turbulent

mixing is obtained between the interrupted boundary-layers 30 and the adjacent core layer 32. A turbulent flow, which is to be avoided, may be distinguished from an eddy in that the former is irregular and there is no observable pattern as with an eddy.

5 Eddies and swirls therefore do not constitute turbulence. Again a laminar eddy or vortex is confined by solid boundaries or by laminar fluid flows, while a turbulent eddy or vortex will be surrounded by other eddies and vortices which interact with the turbulent eddy or vortex. The conditions for maintenance of
10 laminar flow with a particular structure can be observed for example by providing suitable windows in an experimental structure and adding visible fluids to the fluid flow if required.

Conveniently, as illustrated, the diameter of the spheres is equal to the spacing between the surfaces 20 and the
15 spots of interruption 34 by the spheres 28 are their points of contact with the walls. The connections provided by these contact points gives high heat transfer at the points and also serve to support the walls against external pressures greater than the internal pressure. These effects can be increased even more by
20 making the connections solid, e.g. by brazing or otherwise fixing the structure to the surfaces at the contact points. It is not necessary for the interposed structure to touch the passage walls as long as it is sufficiently close thereto to provide the necessary extent of interruption to the boundary layers. Thus, in
25 the illustrated embodiment the portion of each spherical surface around the actual point of contact and submerged in the boundary layer will also be effective in this interrupting function. The interrupter structure may therefore be suspended within the

enclosure and not actually touch the walls, or touch the walls at fewer points than there are interruption points.

Figure 4a shows in plan view the profile of spherical interrupter structure elements of the structure of Figures 1 to 3, taken in the direction of flow of the fluid in the passage; the profile is of course a circle. Other profiles can be used and should be such as to present a smoothly contoured surface to the fluid flow, so as to reduce friction losses to a minimum and also to ensure the maintenance of laminar flow. Figure 4b shows for example an ellipsoidal profile, while Figure 4c shows an egg-shaped profile and Figure 4d shows a drop-shaped profile; in the latter two profiles the face of largest radius faces upstream.

Figure 5 illustrates the statement above that the elements of the interrupter structure do not necessarily contact the heat exchange surface and a chain-dotted profile 40 is illustrated in which this is not the case, the highest point of the profile being spaced a minimum distance d from the surface 20; in the case of a convex curvilinear surface spaced from a flat surface 20 this distance d should not be more than about 10% of the effective diameter of the curved surface.

A pyrimidal surface 42 is also illustrated in broken lines terminating at the contact point 34 and this is unsatisfactory for use in flow interrupting structures of the invention, principally because there is a drastically reduced opportunity for the establishment of high fluid flow velocities at the boundary layer, with consequent less disrupting of the layer and much less effective heat exchange at the surface; the preferred form of the profile may be characterised as being convex curvilinear and this

is arranged to provide the maximum possible velocity as close as possible to the flat and smooth heat transfer surface while maintaining laminar flow.

Special situations arise for example when the fluid is very viscous, such as a viscous oil that is to be heated. When the fluid is of high viscosity the spacing apart of the parallel walls of the passage can be increased considerably without the establishment of turbulent flow, but such a fluid is usually of low thermal conductivity and a thermal boundary layer will be established immediately adjacent to the heat transfer surface that is much thinner than the respective boundary layer. The interposed structure must be arranged to interrupt this thinner thermal boundary-layer irrespective of the thickness of the boundary-layer. The principal factor in the determination of the thickness of the thermal boundary layer is the Prandtl number, which is high when the viscosity is high and the thermal conductivity is low.

One of the principal parameters to be considered in determining whether a particular fluid flow will be laminar and non-turbulent is the Reynolds number which is obtained by the relation:

$$R = \frac{\text{Fluid Mass Velocity} \times \text{Passage Equivalent Diameter}}{\text{Fluid Viscosity}}$$

Classically it was believed that with a Reynolds number less than about 4,000 the flow must be laminar, while if it was greater than about 6,000 it would become turbulent. It is not possible in the apparatus of the invention to determine the fluid velocities in the interrupter structure but only the overall

velocity and the only proof that the flow will be laminar is to plot the so-called J-factor curve which will show an abrupt change in slope at the onset of turbulence. The existence of a J-factor curve of constant slope is therefore proof that laminar flow is occurring and this can occur with Reynolds numbers as high as 15,000.

Figure 6 illustrates another form that may be taken by the interrupter structure consisting of a sheet 44 in which alternate convex and concave profiles 28 and 46 respectively have been formed, so that the convex profile 28 on one side forms the concave profile 46 on the other side. Such a "convex-concave" sheet can be manufactured inexpensively by roll-forming from a thin metal sheet to a thickness such that it can just be slid into the space between the two walls 22. It will be noted that the convexities on one side in successive rows are staggered transversely of the direction of flow so that there are no channels permitting straight-through flow without interruption of the respective adjacent boundary layer 30.

Figure 7a illustrates the application of the interrupter structure of Figure 6 to a two-path, two-fluid heat exchanger in which the heat exchange takes place through the material of the sheet 40. Thus, inlet 14 and outlet 16 are for the path on one side of the sheet, while inlet 48 and outlet 50 are for the path on the other side of the sheet. In a more complex structure not illustrated a number of parallel passages will be formed by a stack of spaced sheets 22, each passage containing a sheet 44, the heat exchange between adjoining paths taking place through the walls 22 as well as the sheet 44.

Figure 7b serves to illustrate the application of the interrupter structure of Figures 1 to 3 to such a two-path, two-fluid heat exchanger, the heat exchange taking place through the wall 22 between the two paths.

5 The invention has been described above principally in connection with heat exchange apparatus of the kind known generally as of plate type, since the heat exchange takes place through a plate. The invention is also applicable to heat exchangers of the equally well known shell and tube type. For
10 example, in a further form of the apparatus illustrated by Figure 8 the flow passage is a tube of square cross-section having two pairs of opposed parallel wall surfaces 20 and the interrupter structure is a row of closely packed spheres 28 touching both pairs of surfaces at spaced interrupter spots.
15 Such a structure will exhibit improved heat exchange through all four of the tube walls 22. In the form of apparatus illustrated by Figures 9a and 9b the flow passage is a tube 22 of circular cross-section and the interrupter structure spheres must be of smaller diameter than its internal diameter so as not to block
20 the passage. The minimum diameter for the spheres is about 75% of the tube internal diameter, more preferably from about 90% to about 95% thereof. In practice the difference should be as small as possible consistent with obtaining the necessary flow capacity and ensuring that the flow path cannot be blocked by
25 retained solid material in the fluid. Figure 9a shows a preferred configuration in which the spheres are distributed helically around the longitudinal axis 52, while Figure 9b shows a less satisfactory structure in which all of the spheres are to the same side of the

axis. The structures of Figures 8, 9a and 9b can be used as the basis for a heat exchanger of the so-called bayonet type.

Figures 10 and 11 illustrate a single tube-in-shell exchanger in which one fluid path with inlet and outlet 14 and 16 respectively is formed by the annular space between an outer shell 54 and the inner circular cross-section tube 22, while the other fluid path with inlet 48 and outlet 50 is of course formed by the tube 22. The annular shell space is of radial dimension just sufficient to receive the spheres 28 and the spaces between the spheres and the inner wall of the outer shell are completely filled with a suitable cementitious material 56 to prevent fluid flow therethrough that would be wasted. The interrupter system employed within the tube 22 can be that shown in Figures 9a or 9b, but preferably is as shown in Figures 10 and 11 in which rows of smaller spheres are used to provide the necessary flow capacity with a sufficiently large number of interrupting points 34 both along the length of the tube and also around its circumference. As with the shell-side interrupter system the useless space between the rows is filled with a cementitious or other suitable material 58, such as concrete or ceramic cement.

Figures 12 and 13 illustrate a multiple tube in shell heat exchanger in which a plurality of parallel tubes 22 are disposed within a single outer shell 54.

Each tube 22 is surrounded by spheres 28 in rows, circles or helixes thereof with some of the spheres contacting two adjacent tubes, so that it disrupts the boundary layers of both tubes. The tubes 22 thus take the place of the cement 58 of the structure of Figures 10 and 11 and only the cement 56 is

required between the outermost spheres and the shell 54. The interrupter structure within the tubes 22 can be of the form illustrated by Figures 9a or 9b.

Figures 14 and 15 illustrate a particular form of tube-in-shell exchanger in which the shell-side interrupter structure is provided by a plurality of outer rods 60 of hemispherical cross-section and a plurality of inner rods 62 of circular cross-section that are assembled into the form of stackable grids of square shape in end elevation. When so stacked the grids form a plurality of parallel passages 64 of square cross-section through each of which a respective tube 22 of circular cross-section is threaded with the convex curvilinear faces of the rods 60 and 62 in contact with the tubes external walls to provide the necessary interrupter structure; these passages 64 thus form the shell side flow path and the unwanted space is filled by the cementitious material. In such a structure the necessary pointwise interruption occurs at the intersection of two curved cylindrical surfaces and illustrates that a spherical or similar surface is not always required. It will be apparent that the passages 64 can instead be of some polygonal cross-section other than square, i.e. with less or more than four sides, for example triangular or hexagonal, a space-filling configuration being chosen to ensure that all of the available space is utilised.

The invention can also be applied in the apparatus of Figure 1 to the air fluid flow paths constituted by the spaces between the fins 18, these spaces being provided with any of the flow

interrupter structures of the invention, such as those specifically illustrated. The interposed structure can be made to increase the efficiency of the heat transfer between the fins and the air over the fins. Such an arrangement can be made to be more efficient than the "split fin" structures proposed hitherto in which the fins are split and staggered in order to disrupt the boundary layers.

The evaluation of the performance of heat exchanger surfaces is a difficult subject because of the large number of variables involved, but one method that has gained acceptance is described in the Transactions of the Society of Mechanical Engineers, Vol. 100, August 1978 in a paper by J.G. Soland, W. M. Mack, Jr. and W. M. Rohsenow entitled "Performance Ranking of Plate-Fin Heat Exchanger Surfaces". Figure 16 is a plot of the ranking of surfaces in accordance with this method, comparing surfaces of the invention with a surface provided by a tube of 1.2 cm diameter and a plate heat exchanger of 0.5 cm plate pitch. Thus the vertical plot indicates the number of heat transfer units (NTU) per unit volume of the heat exchanger core (V), while the horizontal plot indicates the pumping power (E) required to move the fluid through the core per unit volume of the heat exchanger core (V). An improvement in heat exchanger performance is indicated by the line being higher on the vertical plot, and the increase in performance can be measured along any vertical line.

The test fluid was water and the lowest chain-dotted line A is for heat transfer in a tube of 1.2 cm diameter, using data obtained from the above-mentioned paper of Soland, Mack and Rohsenow. The broken line B is for a "APV" plate heat exchanger of 0.5 cm plate pitch, using data obtained from the "APV Heat Transfer Handbook, 2nd Edition, published by APV Inc. of Tonawanda, New York, U.S.A." It will be seen that line B represents an improvement of 28% in performance over line A. The lower solid line C plots the performance of a heat exchanger of the invention employing closely packed spheres of 6.35 mm diameter between plates of that spacing, while the higher solid line D plots the maximum performance so far obtained with a heat exchanger of the invention. It will be seen that line C represents an improvement of respectively 100% and 52% over lines A and B, while line D represents an improvement of respectively 415% and 290%.

1. Apparatus for heat transfer between two fluids comprising:

a body (10) providing a fluid flow passage comprising at least one heat transfer surface (20) characterised in that

5 the parameters of said passage in relation to the fluid flowing therein being such that the said fluid flow will take the form of a non-turbulent boundary layer or layers (3) immediately adjacent to the respective surface or surfaces, and a non-turbulent core-layer (32) interfacing with the

10 boundary-layer or layers; and

a fluid flow interrupter structure (28) disposed within the flow passage, interrupting non-turbulently the full development of at least the boundary-layer of the heat transfer surface at a plurality of spaced interruption spots (34), whereby parts of the interrupted boundary-layer will separate non-turbulently from the heat transfer surface between the said interruption spots and mix with the core-layer to effect heat transfer between the heat transfer surface, its respective boundary-layer, and the core-layer.

20 2. Apparatus as claimed in claim 1, characterised in that the spacing of immediately successive spaced interruption spots (34) in the direction of flow is such that wake-interference flow is established between the said
25 successive spots.

3. Apparatus as claimed in claim 1 or 2, characterised in that surfaces (20) are polished to decrease the friction of the fluid flowing thereover.

30 4. Apparatus as claimed in any one of claims 1 to 3, characterised in that the surfaces of the said fluid flow interrupter structure (28) are polished to decrease the friction of the fluid flowing thereover.

35

5. Apparatus as claimed in any one of claims 1 to 4, characterised in that the body is provided with at least one pair of spaced external parallel fins (12) and a gaseous fluid flow interrupter structure is provided between the said pair
5 of fins.

6. Apparatus as claimed in any one of claims 1 to 5, characterised in that the said fluid flow passage has two parallel facing heat transfer surfaces (20), wherein the
10 fluid flow interrupter structure is of a thickness to fit between the facing surfaces and wherein the structure engages both of the surfaces at respective interruption spots (34).

7. Apparatus as claimed in any one of claims 1 to 7,
15 characterised in that the said fluid flow interrupter is a plurality of spheres (28).

8. Apparatus as claimed in claim 7, wherein the said spheres are packed at maximum density so as to touch the
20 immediately adjacent spheres.

9. Apparatus as claimed in any one of claims 1 to 7, characterised in that the said fluid flow interrupter structure is a plurality of ellipsoids (Fig. 4b).
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10. Apparatus as claimed in any one of claims 1 to 7, characterised in that fluid flow interrupter structure provides a plurality of drop-shaped shapes (Fig. 4c) having the face of maximum radius facing into the direction of flow
30 of fluid in the passage.

11. Apparatus as claimed in any one of claims 1 to 7, characterised in that the said fluid flow interrupter structure provided a plurality of egg-shaped shapes (Fig. 4d)
35 having the face of maximum radius facing into the direction of flow of fluid in the passage.

12. Apparatus as claimed in any one of claims 1 to 11, characterised in that the said fluid flow interrupter structure comprises a sheet having a plurality of convex curvilinear shapes formed therein, the sheet being spaced from
5 the respective heat transfer surface with the said shapes interposed between the surface and the sheet.

13. Apparatus as claimed in claim 12, wherein the said
10 fluid flow passage has two parallel facing heat transfer surfaces and the said sheet fluid flow interrupter structure has a plurality of said convex curvilinear shapes extending from each face thereof so as to have them interposed between the respective surface and the sheet.

14. Apparatus as claimed in any one of claims 1 to 13, characterised in that the said fluid flow passage comprises two spaced facing heat transfer surfaces, one of which is flat and smooth and the other of which has a plurality of convex curvilinear shapes formed therein, the said shapes
20 constituting the fluid flow interrupter structure and having their apicies immediately adjacent to or contacting the flat and smooth heat transfer surface to form the said boundary-layer interruption spots.

15. Apparatus as claimed in any one of claims 1 to 8, characterised in that the fluid flow passage is of square transverse cross-section and the fluid flow interrupter structure is a plurality of spheres of diameter to just fit
25 within the passage.

16. Apparatus as claimed in any one of claims 1 to 8, characterised in that the fluid flow passage is of circular transverse cross-section and the fluid flow interrupter structure is a plurality of spheres of diameter not less than
30 75% of the diameter of the passage.

17. Apparatus as claimed in claim 16, wherein the fluid flow interrupter spheres are of diameter between about 90% and 95% of the diameter of the passage.
- 5 18. Apparatus as claimed in claim 16 or 17, wherein the spheres are disposed in a helical formation about the longitudinal axis of the passage.
- 10 19. Apparatus as claimed in any one of claims 16 to 18, and of shell and tube type, wherein the said fluid flow passage is provided by a tube (22), a shell (54) surrounds and encloses the tube and provides a corresponding shell side fluid flow passage, and the shell side fluid passage surrounding the tube contains a fluid flow interrupter structure (28) surrounding and contacting the external surface of the tube.
- 15 20. Apparatus as claimed in claim 19 and characterised by a plurality of fluid flow passages provided by respective tubes, and wherein a shell side passage fluid flow interrupter structure contacts the external surface of more than one tube.
- 20 21. Apparatus as claimed in claim 19, characterised in that the shell side passage fluid flow interrupter structure comprises a plurality of spheres surround and contacting the tube.
- 25 22. Apparatus as claimed in claim 20, characterised in that the said shell side passage fluid flow interrupter structure comprises a plurality of spheres surrounding and contacting the tube, some of the spheres contacting more than one tube.
- 30 23. Apparatus as claimed in claim 21, characterised in that the space between the shell side passage structure spheres and the inner wall of the surrounding shell is filled with a space-filling material.
- 35

24. Apparatus as claimed in claim 19, characterised in that the fluid flow passage is of circular transverse cross-section and the fluid flow interrupter structure is a plurality of parallel rows of spheres contacting the inner wall of the passage, the space between the rows being filled with a spece-filling material.

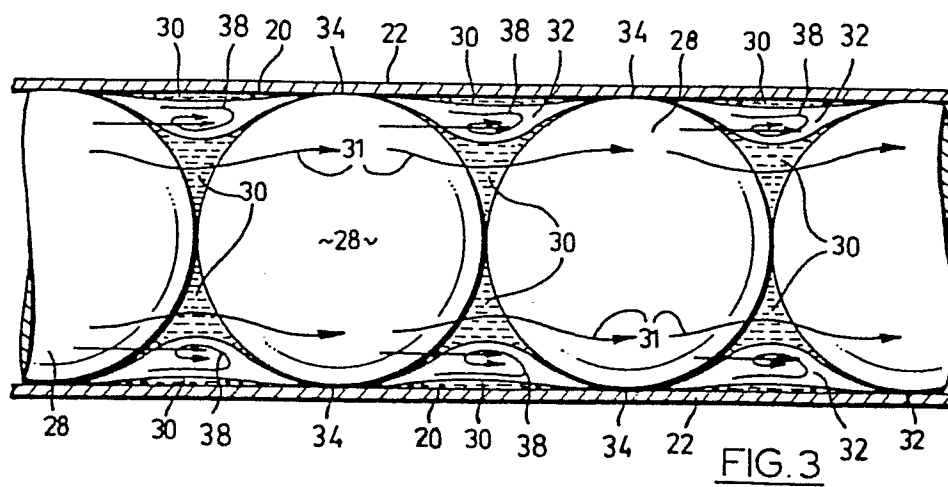
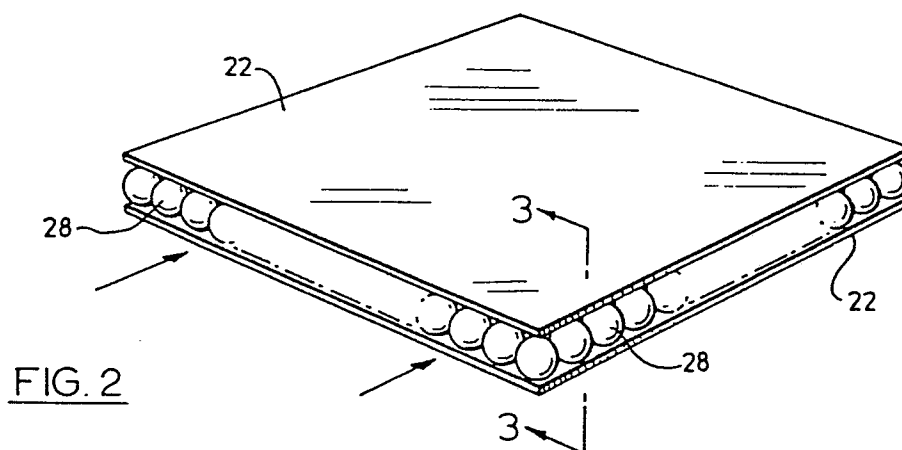
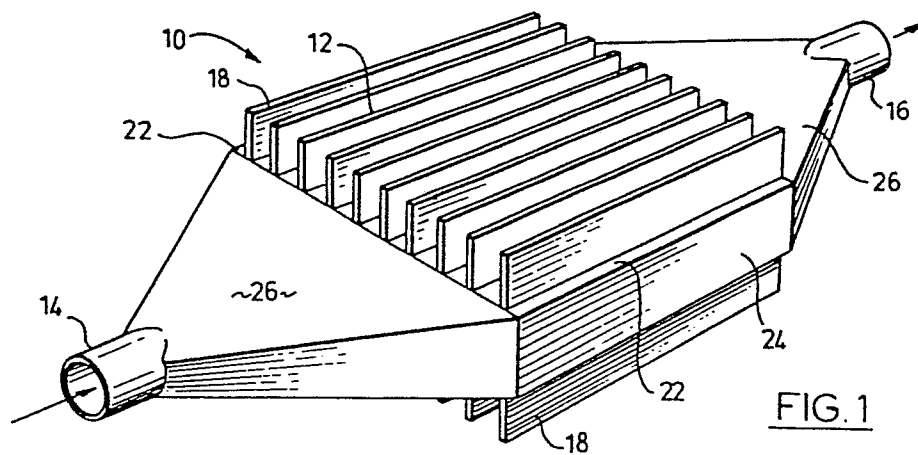
25. Apparatus as claimed in claim 19, characterised in that the said shell side interrupter structure comprises a plurality of rods of hemispherical transverse cross-section disposed with their convex curvilinear surfaces in contact with the tube external surface.

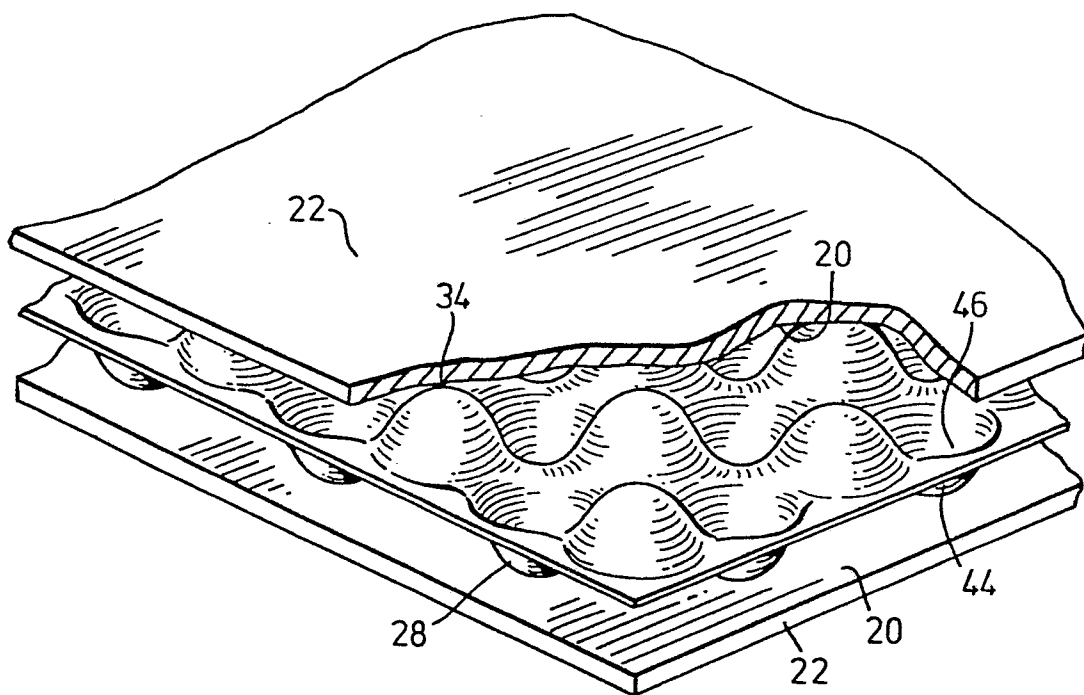
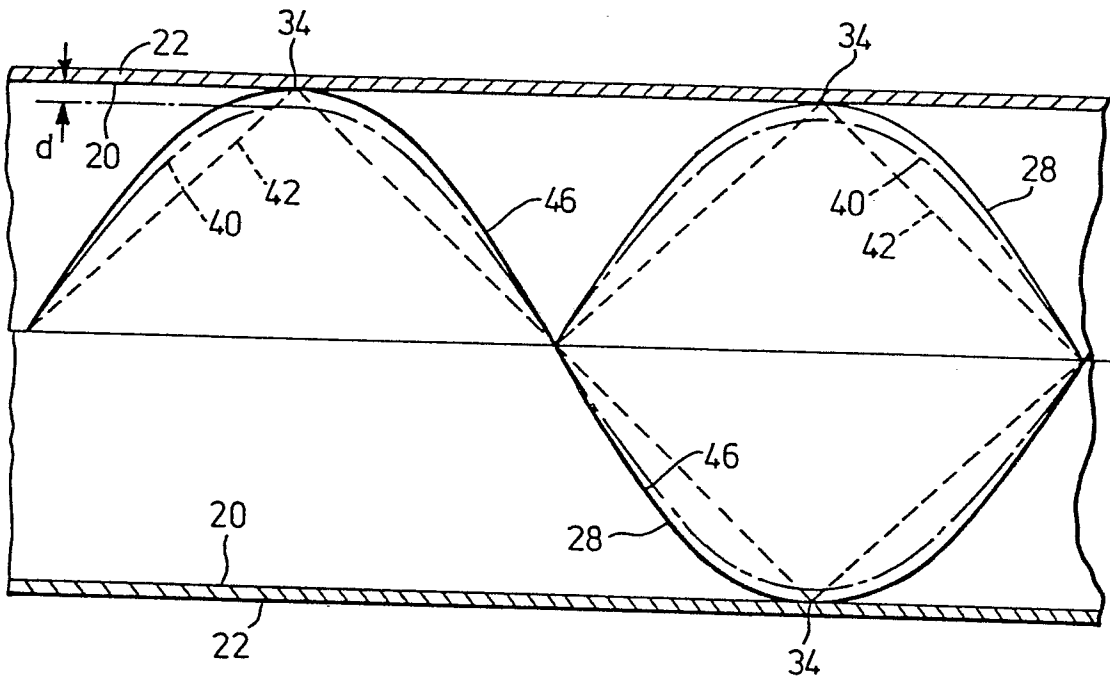
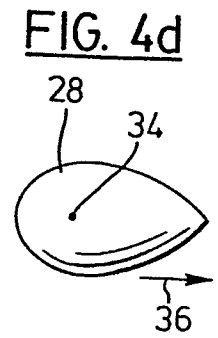
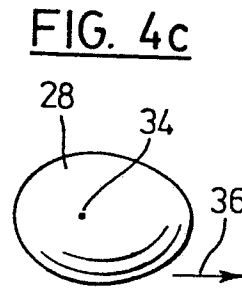
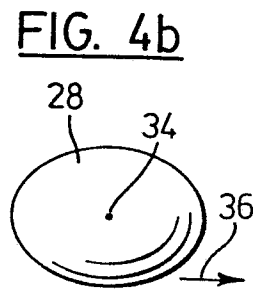
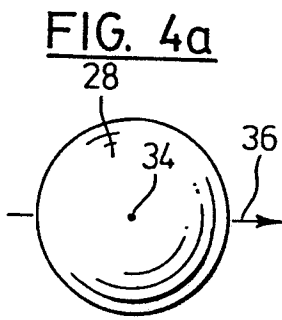
26. Apparatus as claimed in claim 24, characterised in that the shell side interrupter structure has the form of a multi-aperture grid having a tube passing through each aperture.

27. Apparatus as claimed in any one of claims 1 to 26, characterized in that the interrupter structure is connected to the respective flow passage wall at the spaced interruption spots to increase the heat transfer between the fluid, the structure and the wall.

28. A process of heat transfer including establishing in a fluid flow passage comprising at least one heat transfer surface a non-turbulent fluid flow consisting of a non-turbulent boundary-layer immediately adjacent each passage surface, and a non-turbulent core-layer interfacing with the resultant boundary layer or layers;

interrupting non-turbulently the full development of the boundary-layer of each heat transfer surface at a plurality of spaced interruption spots to cause non-turbulent separation of parts of the boundary-layer from the respective heat transfer surface at those interruption spots and their mixing with the core-layer, to effect heat transfer between the heat transfer surface, its respective boundary-layer and the core-layer.





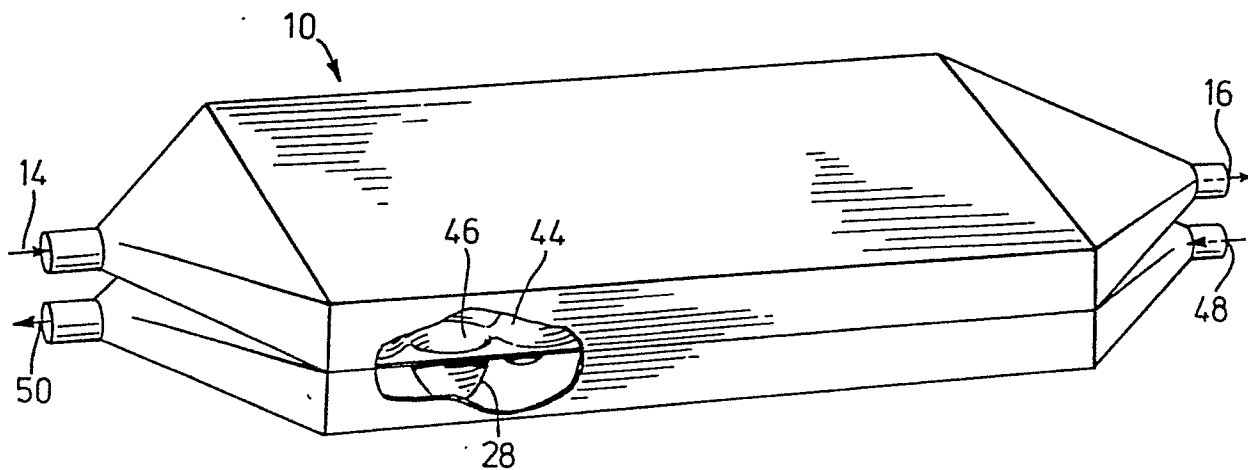


FIG. 7a

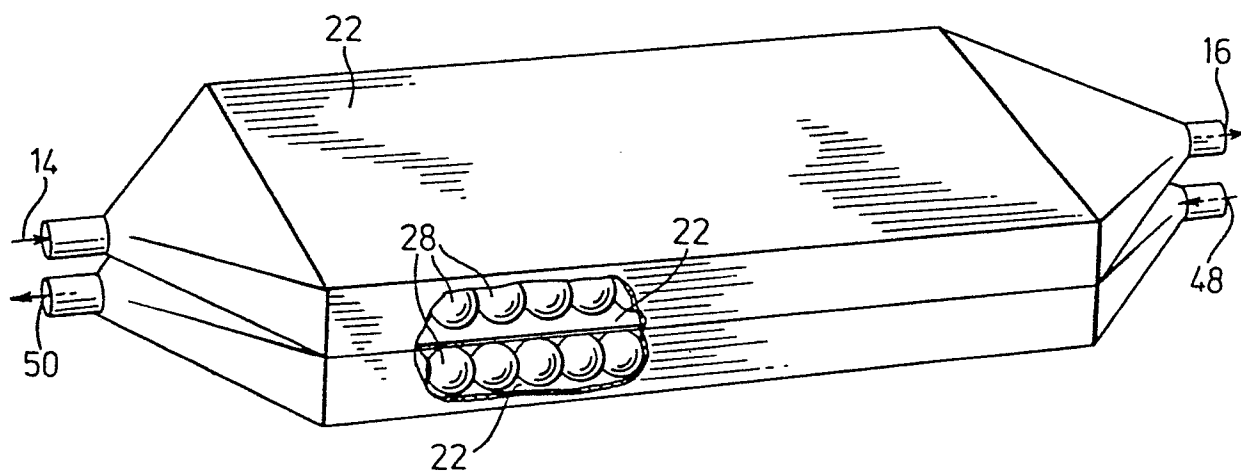


FIG. 7b

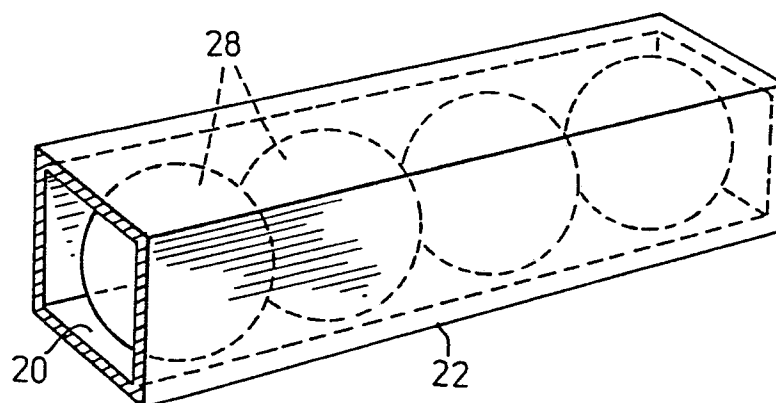


FIG. 8

4/2

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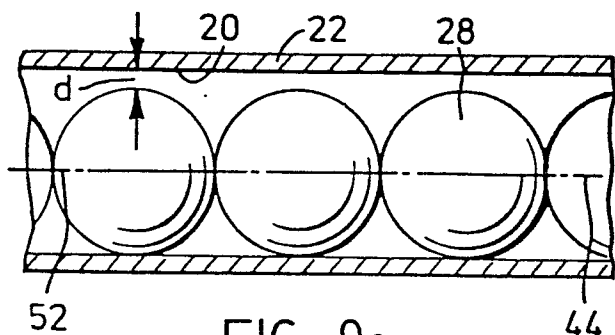


FIG. 9a

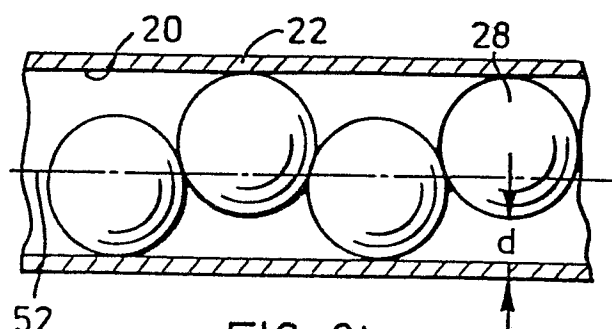


FIG. 9b

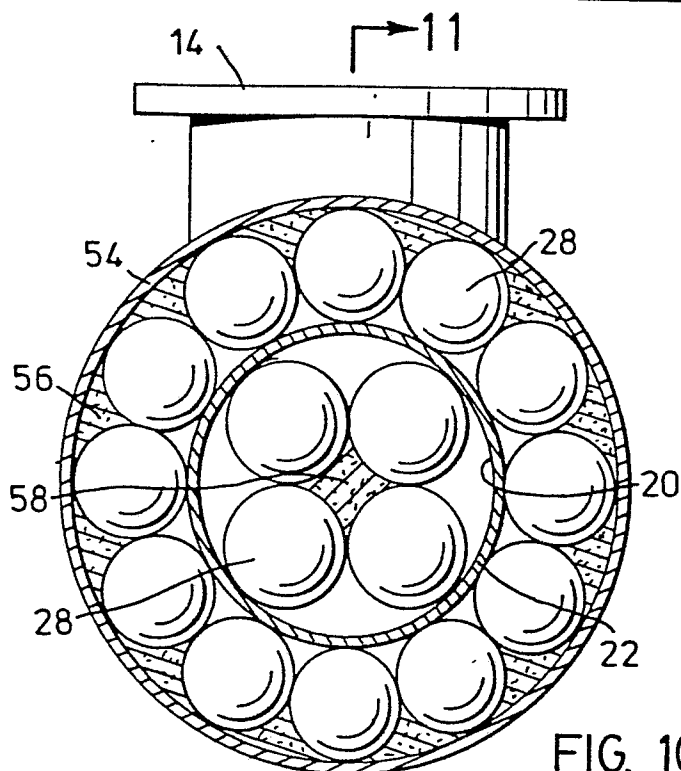


FIG. 10

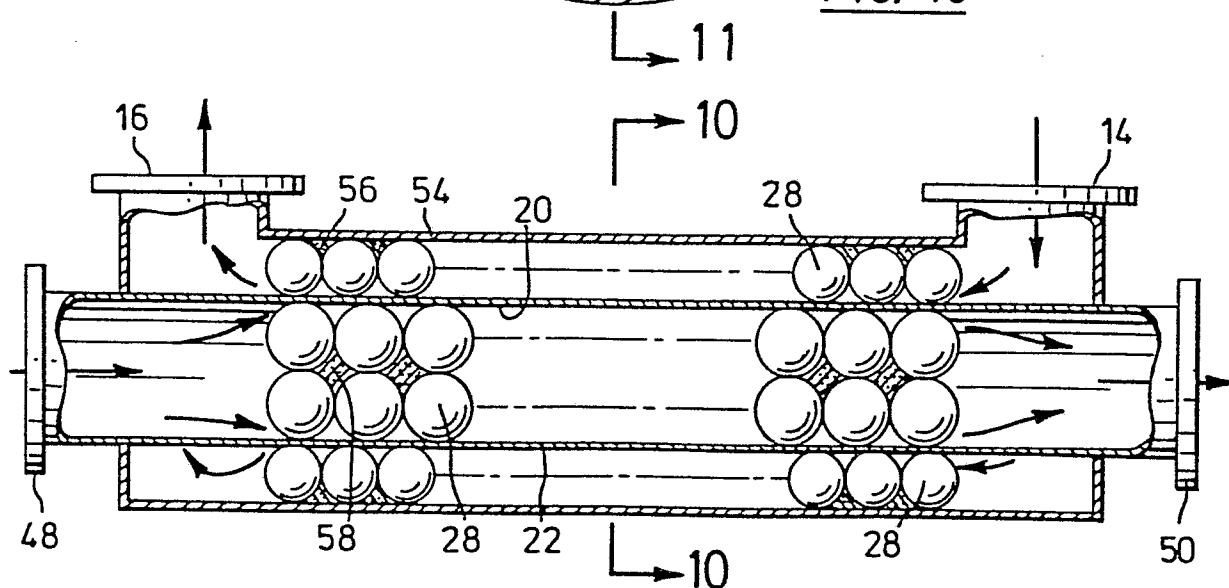
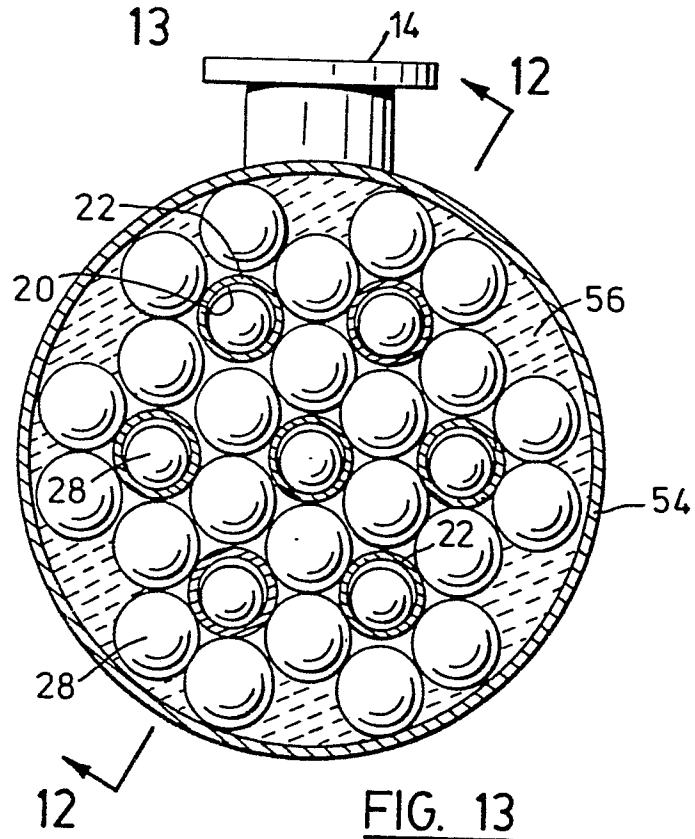
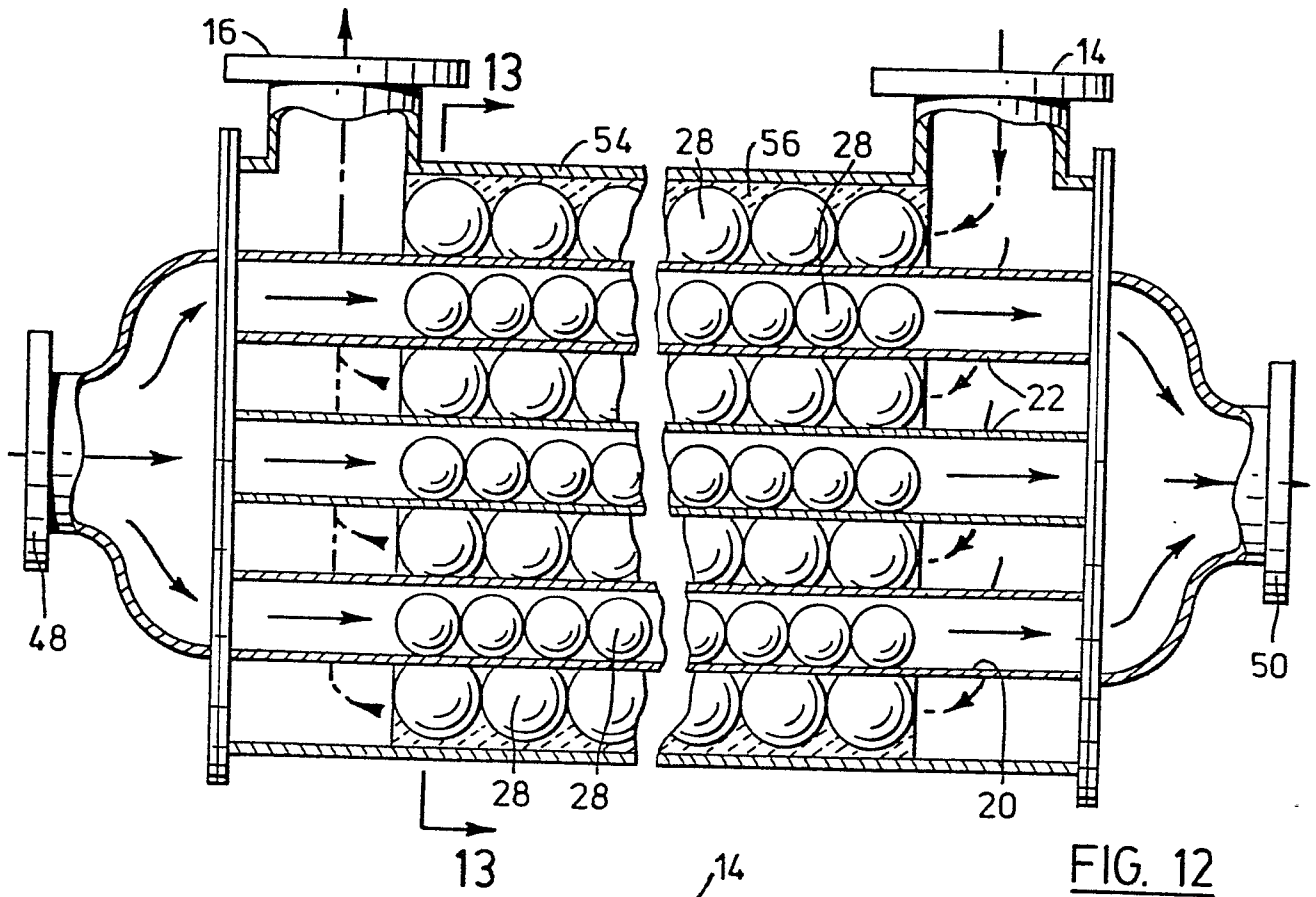


FIG. 11

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6/7

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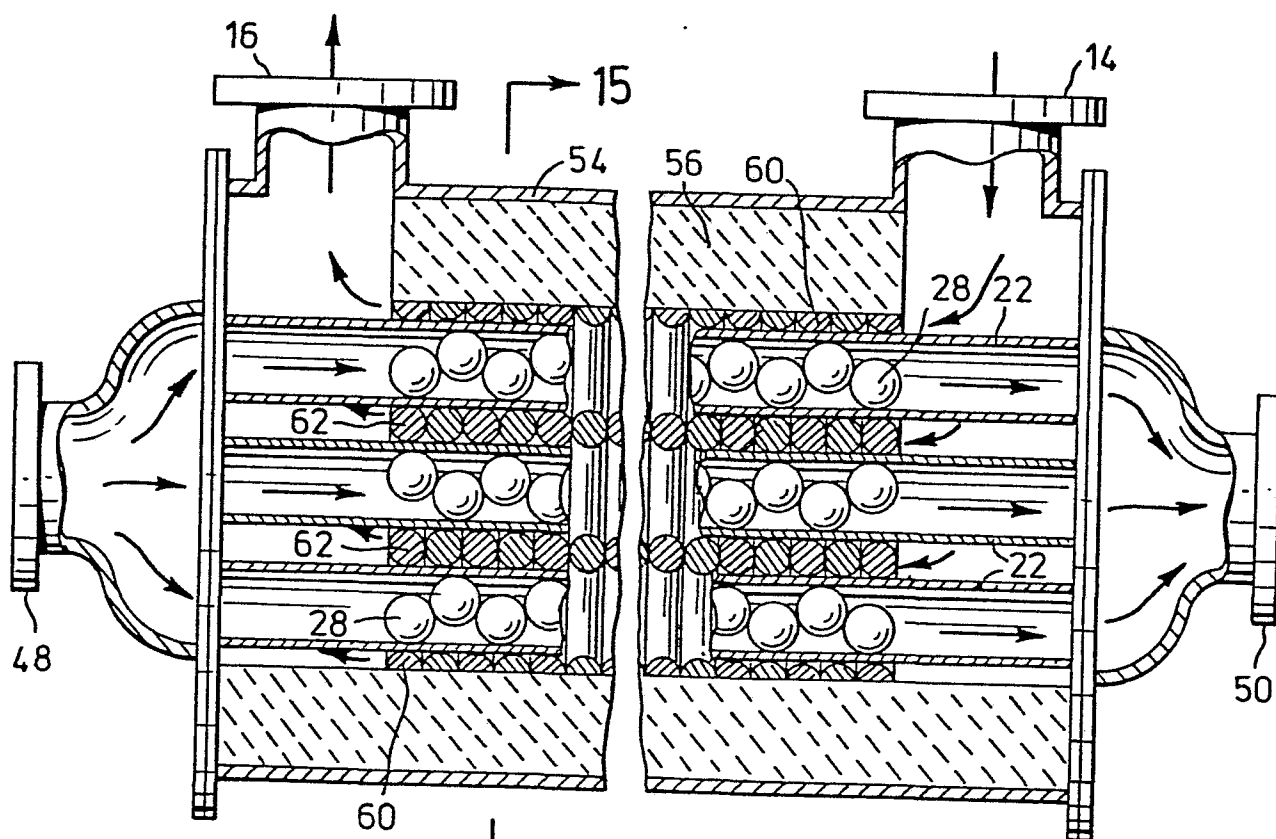


FIG. 14

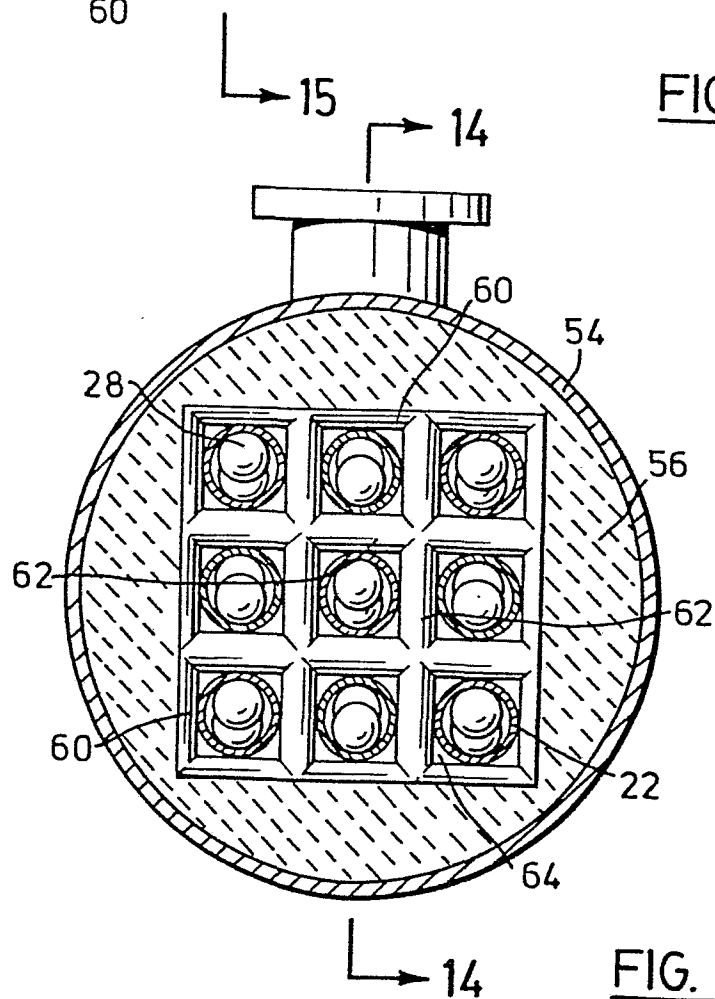


FIG. 15

FF

NUMBER OF HEAT TRANSFER UNITS
VOLUME OF HEAT EXCHANGER CORE

