

[54] **HEAT EXCHANGE**

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165/172, 165/183

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A, 367 C, 225

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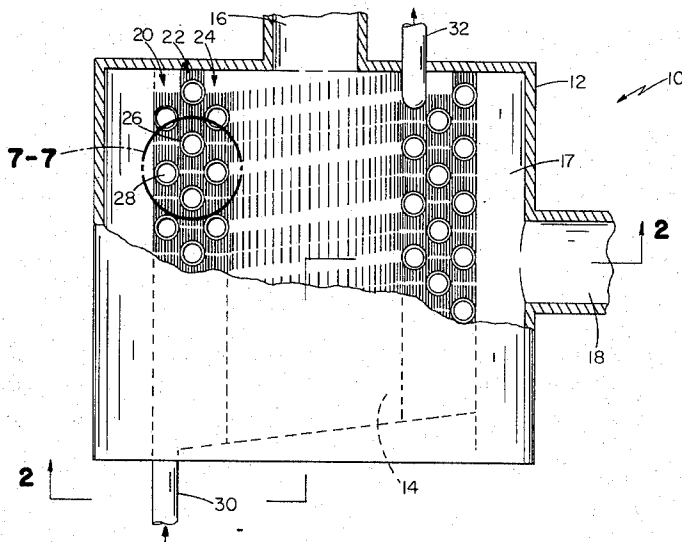
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[57] **ABSTRACT**

A thermoconductive, fluid-confining tube has a plurality of thermoconductive elements each having opposed major surfaces, in thermoconductive contact with the outer surface of the tube, each element having a major dimension extending outwardly from the outer tube surface and a major dimension extending substantially parallel to the axis of the tube, and a smallest dimension, and is adapted for heat exchange with a second fluid directed along the surface of the tube, in an overall flow direction substantially parallel to the smallest dimension of the elements; the thermoconductive elements are spaced apart in the flow direction and define open areas perpendicular to the flow direction with the total per cent of open area perpendicular to the flow direction being less than the total per cent of open area in the plane perpendicular to the tube axis.

7 Claims, 7 Drawing Figures



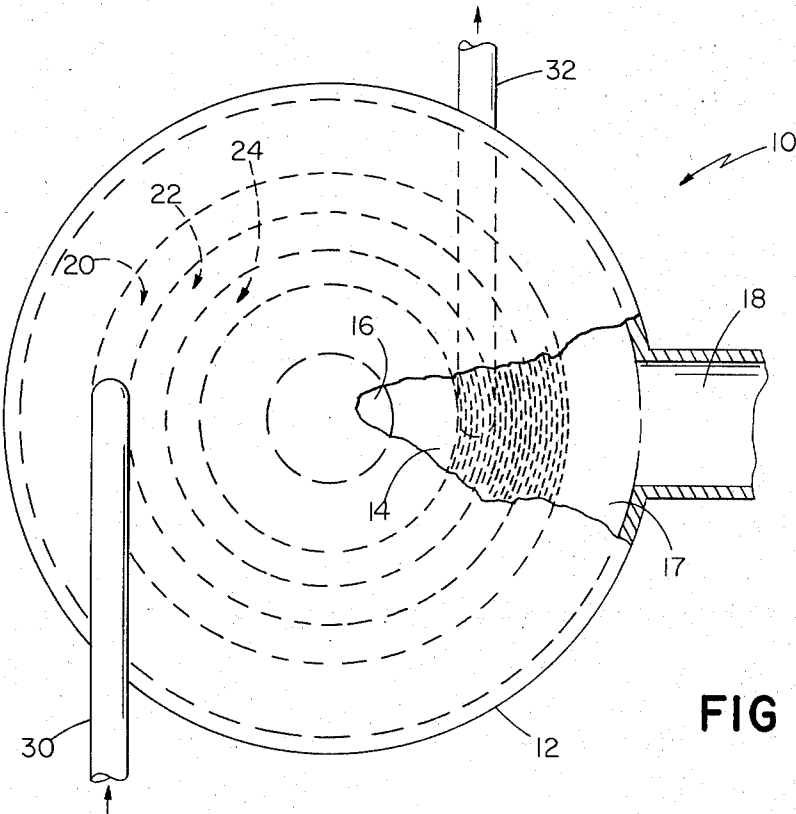
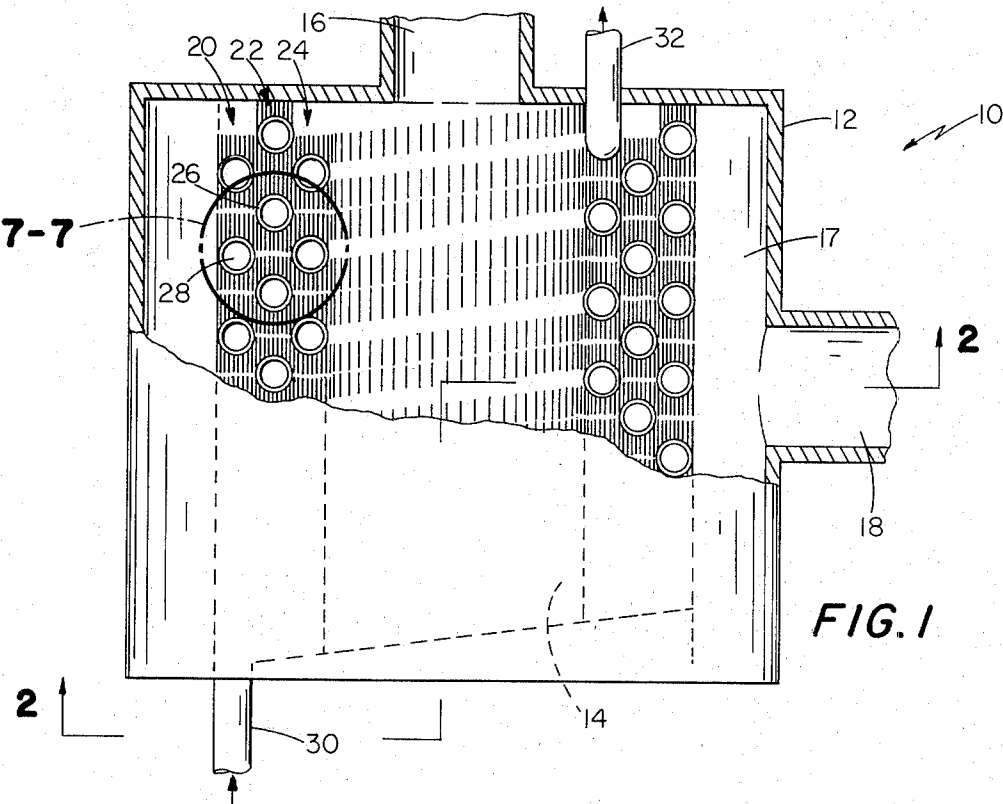


FIG 3

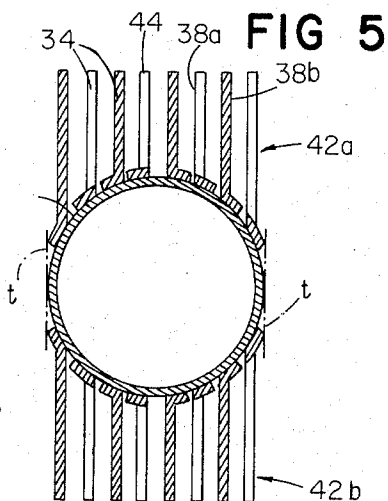
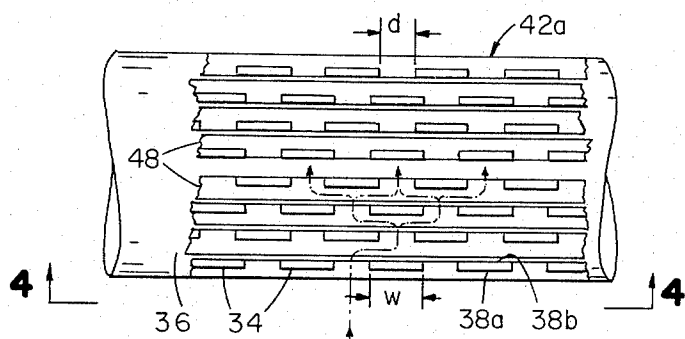


FIG 4

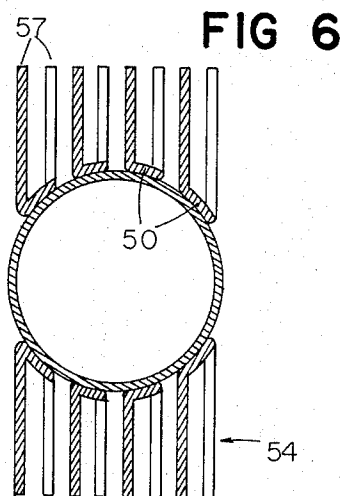
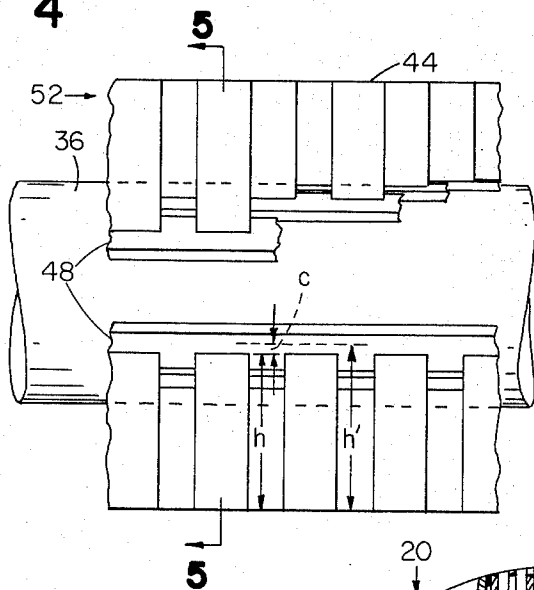
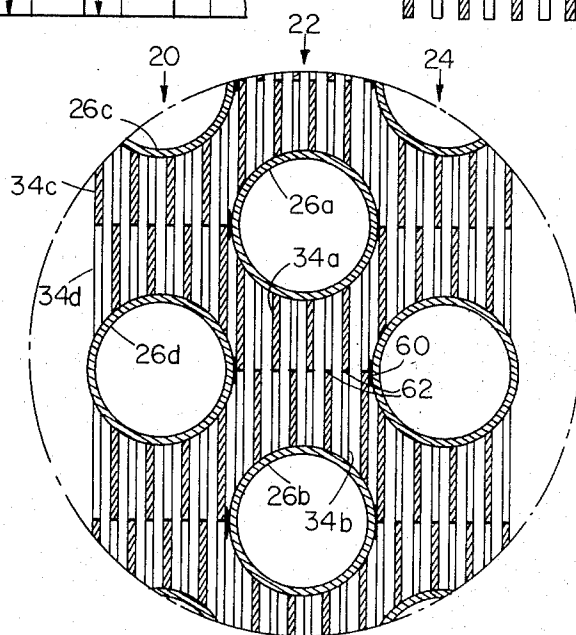


FIG 7



HEAT EXCHANGE

This invention relates to heat exchangers, and particularly to heat exchanger configurations useful for gas-fluid heat exchange, such as in vapor (e.g. steam) generators producing superheated vapor.

Some forms of heat exchanger configurations for steam generators conventionally employ circular tubing, having helically wound fins wrapped around the periphery of the tubing to form an extended heat exchange surface, with the tubing coiled into a geometrical shape such as a cone or cylinder. A first fluid flowing through the tube is heated by second hot fluid (typically combustion gases) flowing generally radially of the geometrical shape. Several concentric coils are commonly employed, although occasionally a single coil or bank is employed.

While there are many applications for which conventionally finned tubing is ideally suited, there are applications for which it is unsuitable or inefficient. It frequently occurs that a heat exchanger is to be designed which has a relatively large frontal area. Such is the case for instance, in the design of direct fired heat exchangers in which the heat exchanger is to surround the combustion volume. In small sizes, the volume required for complete combustion dictates that the surrounding heat exchanger frontal area be relatively large, thereby causing a low gas velocity through that area. Since the coefficient of heat transfer is dependent on gas velocity, a relatively low heat transfer coefficient results, which must be compensated for by a larger heat transfer surface. The requirement for large heat transfer surface necessitates utilizing more banks of finned tubing, which causes the heat exchanger to be quite large and heavy.

Similarly, the low gas velocity results in a low pressure drop. This pressure drop is frequently much below that which ordinarily would be tolerated. The need exists, therefore, for a compact heat exchanger extended surface which promotes high coefficients of heat transfer at low gas velocities, even though the increased heat transfer may be provided at the expense of higher pressure drop.

Another disadvantage of conventional finned tubing concerns the difficulty with which it is coiled. Because of the danger of crushing the fins on the winding mandrel, it must be specially made to support the fins during winding. The radius of the coil is also limited by the tips of the inner fins touching if the winding radius is small or if the fin height is large.

A further problem is encountered when only a single bank of tubes is employed. In this case, the minimum gas flow resistance occurs at the fin tips between adjacent tubes. Therefore a larger portion of the gas flows through this region, essentially by-passing the bulk of the extended surface. This has the effect of reducing the heat transfer performance proportionally below what it would be if there were no by-passing.

It is therefore an object of this invention to provide heat exchangers of increased efficiency, which are compact, and of conventional geometrical shapes.

Another object is to provide novel improved fluid carrying heat exchange tubing which not only has efficient extended surfaces, but is also easier to manipulate, and forms more compact heat exchangers.

A further object is to provide vapor generators of improved efficiency and compactness, which are suitable for a wide range of industrial applications, ranging from water heaters to vapor generators for external combustion engines used in land, water or air conveyances.

The invention features, in one aspect, a heat exchanger comprising a thermoconductive tube for confining a first fluid, a heat exchange matrix comprising a plurality of thermoconductive elements each having opposed major surfaces, in thermoconductive contact with the outer surface of the tube, each element having a major dimension extending outwardly from the outer tube surface and a major dimension extending substantially parallel to the axis of the tube, and a smallest dimension, and structure directing a second fluid along the surface of the tube in an overall flow direction substantially parallel to the smallest dimension of the elements, the thermoconductive elements spaced apart in the flow direction and the matrix having open areas perpendicular to the flow direction, with the total per cent of open area perpendicular to the flow direction being less than the total per cent of open area in the plane perpendicular to the tube axis. In a preferred embodiment, two sets of oppositely extending thermoconductive elements have elements arranged in parallel to one another with elements extending in the same outward direction from the tube terminating in the same plane.

In another aspect, the invention features a thermoconductive tube for confining a fluid for heat exchange with a second fluid external to the tube. The tube has a plurality of thermoconductive elements in thermoconductive contact with its outer surface, having opposed major surfaces, the elements being arranged with all the major surfaces in parallel and in two sets, each set extending in an opposite outward direction from the tube, and the elements being spaced apart in the axial direction to define therebetween open areas, the total per cent of open area in the plane of the major surfaces being less than the per cent of open area in the plane perpendicular to the tube axis. In a preferred embodiment, the elements are spaced apart and staggered in the flow direction so that upstream elements overlap downstream elements. Preferably, both ends of each upstream element overlaps a downstream element; and the major dimensions of the elements extending in the same direction outwardly of the tube terminate in the same plane. In a preferred embodiment, the elements in each set are arranged in axially extending rows, each row being coplanar with a row in the opposite set; the elements are secured to the outer tube surface through axially extending thermoconductive base strips secured to the outer tube surface, each base strip supporting at least one row; base strips supporting only one row do not extend all the way to the next row; and, a base strip supporting two adjacent rows is formed integrally with the member from a substantially U-shaped thermoconductive structure, the side walls of which are periodically slotted to define a row of elements (the slots between adjacent elements being preferably less than the width of the elements between slots).

In accordance with the present invention, the second fluid is therefore made to flow the "hard way" through the thermoconductive elements, inducing intense tur-

bulence and eddies which greatly increase "j factors" over those for more streamlined flow (the "j factor" being a measure of the intrinsic heat transfer capability of a surface). As a result, heat exchangers of increased efficiency are achieved which are still compact and hence economical.

In a further aspect, the invention features a heat exchanger in which a tube having two sets of thermoconductive elements as described is arranged in a coiled configuration with elements on adjacent windings extending toward one another and touching or, at most, having a very narrow passage between them. The fluid is thereby forced to follow a sinuous route along the extended surfaces and between adjacent elements, leading to a high heat exchange efficiency with a not intolerably large pressure drop across the coiled tube. In one preferred configuration, the tube is (preferably, helically) coiled to form one or a series of concentric nested members of circular cross section (e.g., cones or cylinders), with the flow direction of the second fluid substantially radial to the member, and the tube defining a continuous first fluid flow path within which, for example, an entering feed liquid may be heated to exit as superheated vapor (e.g., water to superheated steam).

Other objects, features and advantages will be apparent to one skilled in the art from the following description of a preferred embodiment of the present invention, taken together with the attached drawings thereof, in which:

FIG. 1 is an elevational view partially broken away, of a heat exchange apparatus embodying the present invention;

FIG. 2 is an end view, partially in section, of the apparatus of FIG. 1;

FIG. 3 is a plan view of a segment of heat exchange tube embodying the present invention;

FIG. 4 is a plan view of the heat exchange tube of FIG. 3, taken at 90° to the view of FIG. 3;

FIG. 5 is a sectional view of the tube of FIG. 3, along line 5—5 of FIG. 4;

FIG. 6 is a sectional view of another tube configuration embodying the present invention; and,

FIG. 7 is a magnified view of a portion 7—7 of the apparatus of FIG. 1.

The figures show a heat exchange apparatus 10, including a housing 12, an inlet manifold 14 (which may enclose a suitable burner (not shown)), a fluid inlet 16 to manifold 14, an outlet manifold 17, and a fluid outlet 18. Concentric heat exchange members 20, 22, 24 (illustratively, cylindrically shaped) are formed of a continuous helically wound thermoconductive tube 26, which has an interior fluid confining chamber 28 of cylindrical cross section. Member 20 includes a fluid inlet 30 which is in communication through tube 26 with a fluid outlet 32 in member 24. Where apparatus 10 is a steam generator, water enters as a liquid at inlet 30 and exits as superheated vapor at outlet 32.

Referring particularly to FIGS. 3-6, tube 26 has a plurality of sheetlike thermoconductive fins 34, in thermal contact with its outer surface 36. Each fin has opposed major surfaces 38a, 38b, (illustratively shown as planar and providing the two major dimensions of the fin) parallel to the axis of tube 26, and a smaller dimension perpendicular to surfaces 38a, 38b. The fins are ar-

ranged, in coplanar parallel rows, in two sets 42a, 42b which extend in opposite directions from the tube. The axial distance, d , (FIG. 3), between adjacent fins is less than the width, w , of major surfaces 38, and the heights, h , (FIG. 4), of fins in each row are adjusted so that all fins of each set have coplanar end walls 44. For any single row, therefore, the per cent of open area between the end walls 44 and tube surface 36, between adjacent fins 34, for fluid flow parallel to the smallest dimension of the fins, is less than 50 per cent of the total area including the fins and the spacing between adjacent fins. This is considerably less than the per cent of open area in the direction parallel to the tube axis (i.e., in a plane perpendicular to the tube axis). The spacing between adjacent rows is dependent, inter alia, on the tolerable pressure drop across the fins, as well as the heat exchange efficiency of the individual fins (a high efficiency limiting the required number), and weight considerations. As seen in FIG. 5, the outer two rows of fins are advantageously located as close to the tangents t of the tube parallel thereto as is consistent with equal spacing between rows.

The thermoconductive fins 34 may be secured to the outer surface 36 of tube 26 by conventional methods, such as welding, brazing and the like. A particularly rapid fabrication process is afforded if the fins are formed integrally with a base strip, which may support one (base strips 48 of FIG. 4) or two (base strips 50 of FIG. 6) or even more rows of fins. The integrally axially extending thermoconductive structures 52, 54 defined thereby may be formed of an L-shaped plate, as in structure 52, having a height h' , slotted at spaced equal intervals along its length to define a plurality of spaced thermoconductive fins of height h connected by a base wall of height c , c being most preferably equal to the plate thickness. For the structure 54 of FIG. 6, a U-shaped plate is similarly slotted along its length, except that the slots in each upstanding side 57 are staggered to produce adjacent rows of overlapping thermoconductive fins.

A tube formed as described is then wound about an axis approximately perpendicular to the tube axis and parallel to surfaces 38a, 38b, so that the end walls 44 of the opposed thermoconductive fins of adjacent coils are touching or nearly touching. These end walls may be secured to one another, again by welding, brazing, or the like, at selected points or throughout the apparatus, to lend integrity to the apparatus. As seen in FIG. 7, which is a magnified view of the encircled portion of the heat exchanger of FIG. 1, the parallel arrangement of the fins on the tube allows concentric windings to be located very close together, even touching, so as to increase the compactness of the heat exchanger and prevent by-passing of the gases over the tips of the fins. The tube or fins of one winding may be selectively spot welded (e.g., at selected positions 60, 62) to the thermoconductive fins of an adjacent winding if desired. As shown in FIG. 7, a winding 26a of member 22 has its fins 34a spot welded to the fins 34b of adjacent winding 26b, and to the fins 34c, 34d of the windings 26c, 26d of the member 20.

In operation, which will be described for illustrative purposes for a steam generator, fuel and air are fed through inlet 16 and are combusted in a suitable interior burner; the resultant hot fluids traverse members 20,

22, 24 in an overall flow direction parallel to the smallest dimension of the fins, and thus substantially perpendicular to the surfaces 38a, 38b but in a localized sinuous flow pattern, as shown diagrammatically by the dotted lines in FIG. 3. The actual flow pattern is more complicated than that shown, but the diagram illustrates the prolonged contact between hot gas and heat exchange surfaces throughout the traverse of the gas through the cylindrical heat exchange members. The cooled gas passing to manifold 17 exits through gas outlet 18.

Feed liquid enters at fluid inlet 30, of the outer member 20, gradually becoming hotter as it contacts hotter gas zones radially inwardly of the heat exchanger, and exits as superheated vapor through outlet 32.

Other embodiments, within the following claims, will occur to those skilled in the art. For example, two sets of thermoconductive fins, each only a few fins wide, could be arranged each on a semi-cylindrical base wall corresponding to the outer diameter of the tube, and the tips of the fins joined to form an integral structure in which fins extend between opposed semi-cylindrical end walls. A plurality of such structures could then be fitted in series onto a precoiled tube.

What is claimed is:

1. A heat exchanger comprising a coiled thermoconductive tube for confining a first fluid,
 - a heat exchange matrix comprising a plurality of thermoconductive elements, each having substantially planar and substantially parallel opposed major surfaces, in thermoconductive contact with the outer surface of said tube, each element having a major dimension extending outwardly from said outer tube surface, and a major dimension extending substantially parallel to the axis of said tube, and a smallest dimension, and
 - structure directing a second fluid along the outer surface of said tube in an overall flow direction substantially parallel to the said smallest dimension of

said elements,

said thermoconductive elements being spaced apart in the axial direction to define open areas therebetween perpendicular to the flow direction, being spaced apart and staggered in the flow direction such that upstream elements overlap downstream elements, and being arranged with all said planar surfaces in parallel in two sets having elements extending in opposite outward directions from said tube with elements extending in the same outward direction from said tube terminating in the same plane, said plane being parallel to the flow direction, and

thermoconductive elements of adjacent windings of said coiled tube extending toward and terminating close to one another to define therebetween at most a narrow passage.

2. The device of claim 1 wherein said coiled configuration defines a hollow heat exchange member of circular cross section.

3. The device of claim 2 wherein at least two said hollow heat exchange members are arranged in concentric, nested relationship, with their forming tubes connected to define a continuous flow path for said first fluid through said members.

4. The device of claim 3 wherein said members are of cylindrical configuration, and said second fluid is directed substantially radially outwardly of said cylinders.

5. The device of claim 1 wherein said tube is helically coiled.

6. The device of claim 1 in the form of a liquid heater, wherein said second fluid is hot gases, and said first fluid is an entering liquid heated by said second fluid in passage through said tube.

7. The device of claim 6 in the form of a vapor generator, wherein said coiled tube has a feed liquid inlet at one end and a superheated vapor outlet at the opposite end.

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