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CROSSFLOW COUNTERCURRENT HEAT EXCHANGER WITH INNER AND OUTER-TUBE SECTIONS MADE UP OF CLOSELY PACKED COAXIALLY NESTED LAYERS OF HELICOIDALLY WOUND TUBES

Filed April 29, 1966 3 Sheets-Sheet 1

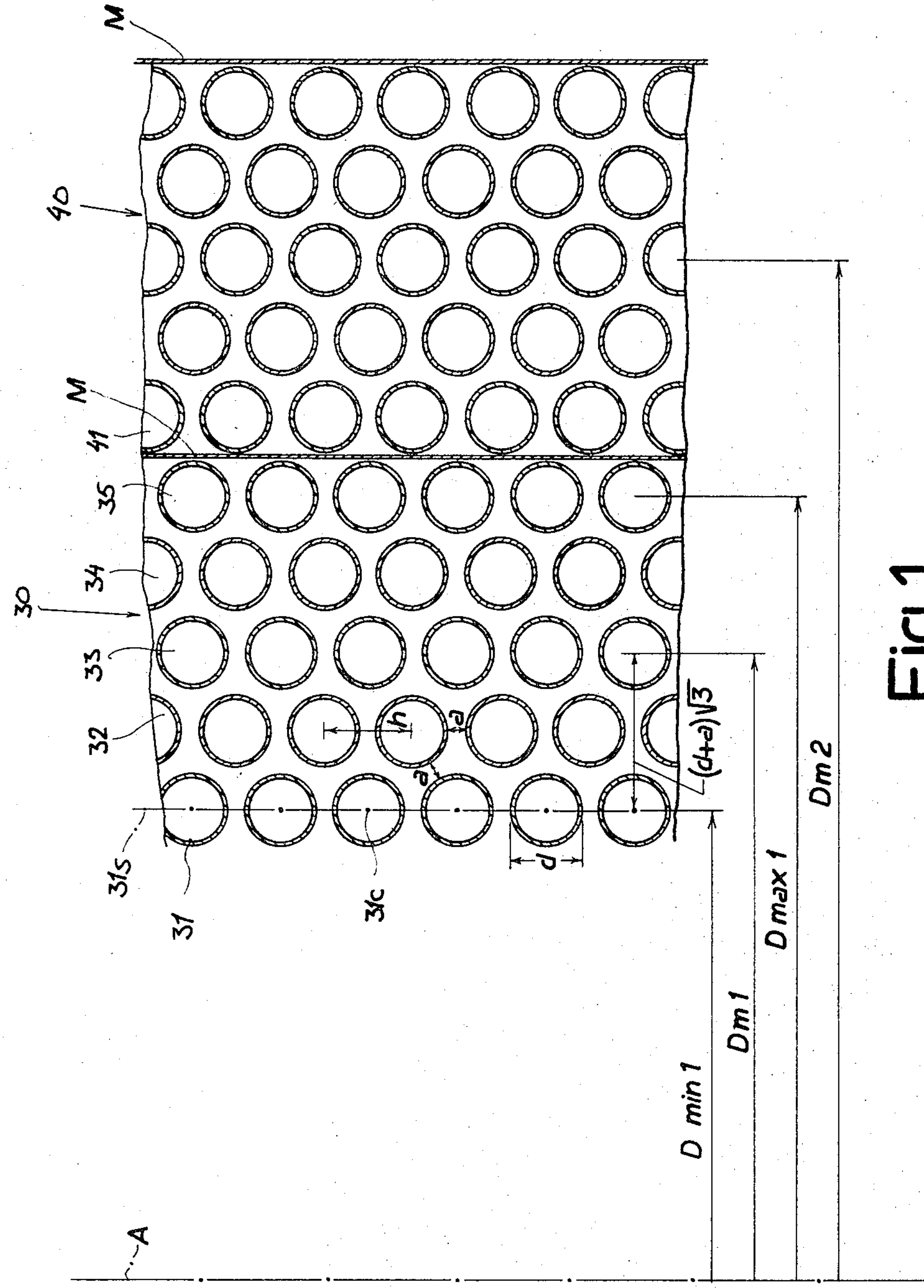

Fig. 1

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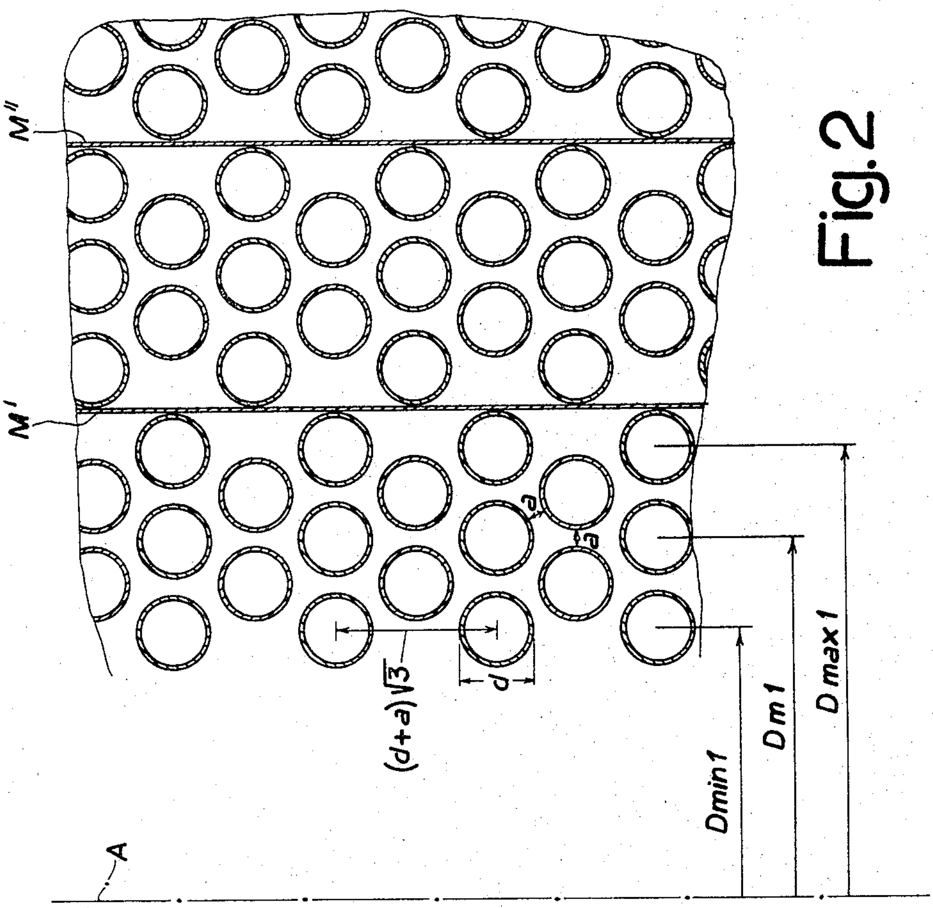

Fig. 2

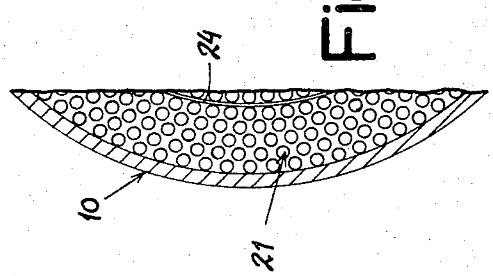

Fig. 4

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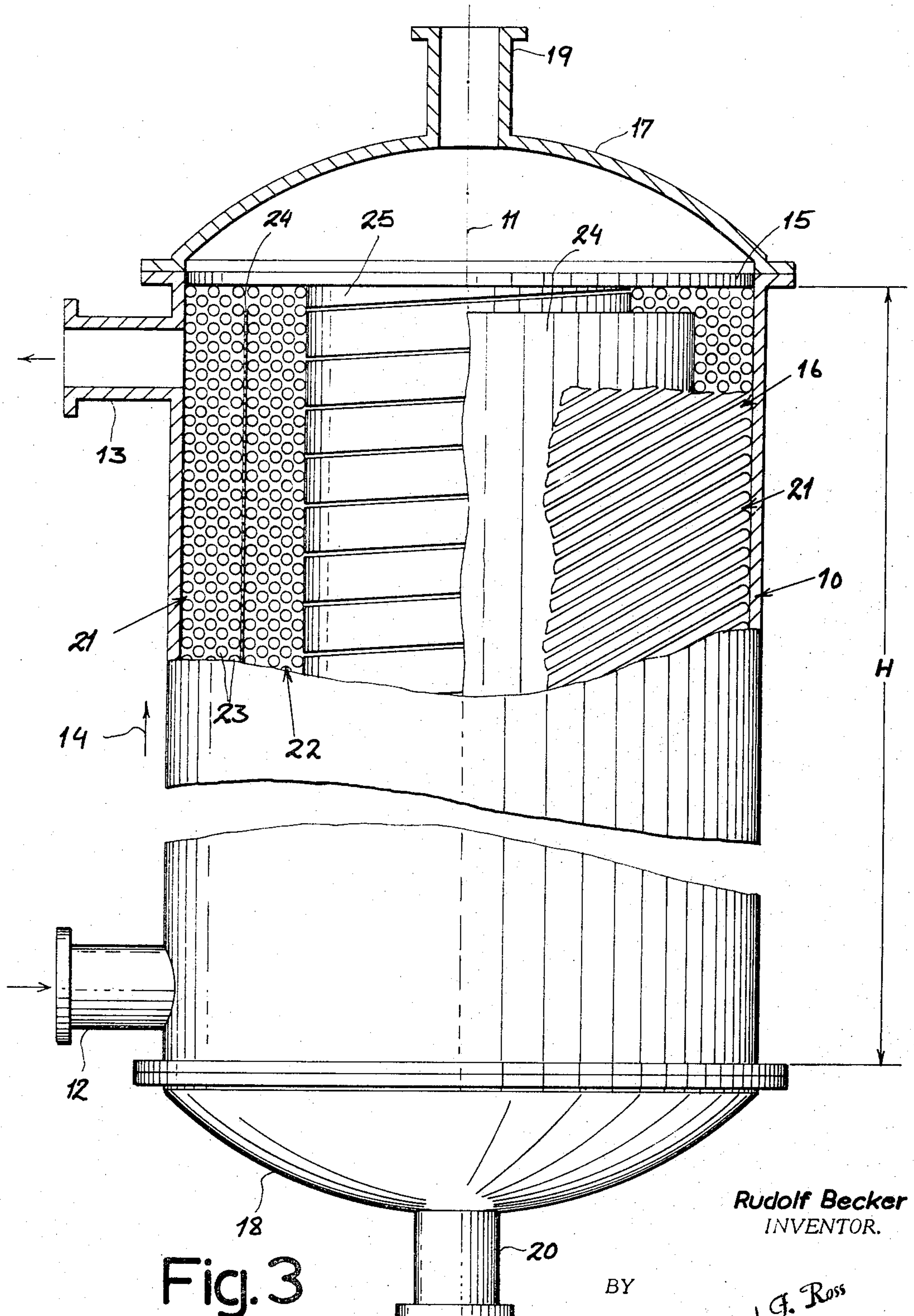

Fig. 3

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CROSSFLOW COUNTERCURRENT HEAT EXCHANGER WITH INNER AND OUTER-TUBE SECTIONS MADE UP OF CLOSELY PACKED COAXIALLY NESTED LAYERS OF HELICOIDALLY WOUND TUBES

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ABSTRACT OF THE DISCLOSURE

Crossflow countercurrent heat exchanger having a pair of coaxially nested tube sections separated by a thin sheet-like cylindrical mantle closely surrounding the inner-tube section and interposed between it and the outer-tube section, the sections each consisting of coaxially nested, closely packed layers of circumferentially spaced parallel-wound tubes, all turning in the same sense about a common axis, the closely packed relationship ensuring that the tubes of each layer are received between a pair of tubes of an adjoining layer so that the centers thereof form an equally lateral triangle whose side is only slightly greater than the diameter of the tubes.

The present invention relates to a crossflow countercurrent heat exchanger of the type in which two fluids are passed generally in countercurrent with respect to one another for heat exchange therebetween via a partition separating the fluids from one another and in which, at each point, the fluids can be considered to move transversely to one another. More particularly, the present invention relates to an improved spiral (or helicoidal) tube heat exchanger structure of this general type adapted to be used effectively in cryogenic processes (e.g. for the rectification of air).

In the theory of heat exchange between two flowing media, it has been determined that the transfer of heat from the warmer fluid to the cooler fluid is a function, considering the heat exchanger as a whole, of the mean temperature difference between these fluids, the nature and thickness of the intervening wall (which is generally constructed of a material of high thermal conductivity), the surface areas across which heat exchange can occur, the flow-rate ratio and the heat-transfer coefficient which, to a large measure, is determined by the relative flow directions. For the most part, heat exchangers are characterized by the relative flow directions of the tube media and thus can be classified as unidirectional heat exchangers, countercurrent heat exchanges and crossflow heat exchangers. In the unidirectional type of heat exchanger, the two fluids are passed generally parallel to one another in a common direction through the heat exchanger so that, when the exchanger is an elongated unit, both fluids are introduced at one end and removed at the other. In countercurrent heat exchanges, however, the flow of the fluid along opposite sides of the intervening walls is effected generally in opposite directions whereas, in crossflow heat exchangers, the flows of the fluid media on opposite sides of the walls are intended to be generally transverse to one another.

In countercurrent heat-exchange techniques, by contrast with unidirectional and crossflow heat exchange methods, the average temperature differential between the two fluids (for given input and exit fluid temperatures) is relatively large whereas the temperature differential at the hot and cold ends of the heat exchanger, depending upon its use, is relatively low although the heat-transfer efficiency (e.g. in terms of the heat-transfer coefficient) is relatively poor. Crossflow heat exchangers, by contrast, are characterized by the highest heat-transfer coefficient and heat-transfer efficiency of all of the various types of heat exchangers under discussion. For this reason, it has been proposed in heat-exchanger technology to combine characteristics of crossflow and countercurrent heat exchangers in so-called crossflow-countercurrent or "spiral-tube" heat exchangers. Heat exchangers of the latter type generally comprise a substantially cylindrical shell or housing receiving a nest or bundle of generally coaxial helicoidally or spirally coiled tubes. One of the heat-exchange fluids is fed through the axial extending cylindrical shell or housing from end to end while the other fluid flows through the tubes in the opposite direction and it can be observed that at any location along the tubes within the heat-exchanger shell, i.e. at any point along the partition walls of conductive material separating the fluids, the fluid passing through the outer chamber is flowing along the partition surface transversely and approximately perpendicularly to the fluid passing along the inner surface of the tube. At each point, therefore, the outer fluid flows generally perpendicularly to the fluid flow through the tube so that the fluid flow overall is both countercurrent (insofar as the axial component of flow is concerned) and perpendicular or crosscurrent.

In conventional crossflow-concurrent exchangers having multiple coaxial layers of coiled tubes, the tubes of each layer or "sheet" are substantially parallel to one another and spaced apart along a respective imaginary cylindrical surface surrounding the axis of the heat exchanger; the tubes of the coils can be considered to be coiled alternately in right hand and in left hand senses about the axis. To hold the tubes in place, rods or wires are disposed between the individual layers or tube sheets, these supporting elements having cross-sectional areas corresponding to the requirements for proper physical support of the tubes of the respective coils. In fact, the rods or wires have only a limited utility because of the nature of the supporting function; consequently, the supporting wires or tubes cannot augment adequately the turbulence within the heat-exchanger chamber and the heat exchange efficiency—which is related to the degree of turbulence—nor can they be so designed as to obtain an optimal flow velocity. It follows, therefore, that conventional spiral-tube heat exchangers have been severely limited in practical respects by the prior methods employed for positioning and supporting the spaced-apart tubes of the helicoidal arrays or layers and were not amenable to improvement with respect to the heat-transfer efficiency, the heat-transfer rate and the uniform channeling of the low-pressure gas stream constituting the outer heat-transfer fluid in cryogenic processes.

It has been found, moreover, that to effect a satisfactory degree and uniformity of heat transfer between the fluids, a uniform distribution of the gas or other fluid to the tubes and within the tubes must be ensured and, consequently, it has been recognized in connection with countercurrent heat exchangers that all of the tubes must be approximately and preferably precisely uniform and optimally arranged. In heat exchangers of the character described, however, the diameter of the coil varies from layer to layer and, when the diameter of the coil (the diameter of the imaginary cylindrical surface surrounding the axis of the heat exchanger upon which the tubes are centered) decreases and the number of turns over the length of the coil remain the same, the length of each tube increases from layer to layer. While this is a substantially immaterial matter when the number of concentric layers of helicoidal arrays or layers of tubes is small, the matter becomes of principal importance wih respect to the nature of the heat exchange process, the efficiency, the uniformity of gas distribution, and the economy as the number of layers increases. In conventional crossflow-countercurrent heat exchangers, it has been possible heretofore to approach the required uniformity of the length of the tubes of the coils, in spite of the diameter of the respective layer, by increasing the number of turns per coil layer and tube of each layer in proportion to the increase in the diameter of the respective layer. Correspondingly, the number of turns of each layer of parallel-wound tubes is reduced and the pitch increased.

The relationship between the various parameters of the tube coils can be expressed generally in terms of H, z, w and h where:

H is the axial length or height of the crossflow countercurrent heat exchanger (i.e., the axial length of the outer chamber from end to end and the axial length of each helicoidal-coil layer);

z is the number of mutually parallel and circumferentially spaced helicoidally coiled tubes of each cylindrical tube sheet or layer;

w is the number of turns per tube of each layer; and

h is the perpendicular distance between the center lines of the tubes of each layer.

It can be seen that:

$$H = z \cdot w \cdot h \tag{1}$$

and, for small pitches of the coiled tubes:

$$L \approx w \cdot 2\pi r \tag{2}$$

where L is the actual tube length and r is the radius of a turn.

The pitch N of each coil is determined by the ratio of the axial height or length H of the heat exchanger to the tube length L:

$$N = H/L \tag{3}$$

H, the axial length (or height if the exchanger is vertical) of the crossflow countercurrent heat exchanger and h, the perpendicular distance between the center lines of adjacent tubes of each layer should be substantially constant and are so selected that, on the one hand, the pressure drop through the heat exchanger is not excessively large and, on the other hand, the turbulence in the interests of a high heat-exchange rate is not too small.

L should, as has already been proposed, be maintained substantially constant over all the layers and it will be evident from Equation 2 supra that such a uniformity of the length L of the tubes can be attained with an increasing radius r from layer to layer if w is reduced; w is, moreover, determined in accordance with Equation 1 so that the product $z \cdot w$ must remain constant and there must, accordingly, be a corresponding increase in the value of z from layer to layer outwardly of the axis of the heat exchanger.

It is, therefore, an important object of the present invention to provide a crossflow-countercurrent heat exchanger having a plurality of generally coaxial layers of helicoidally coiled tubes and a cylindrical shell or housing surrounding these layers, in which a high heat-transfer rate and efficiency is maintained, a uniform flow of the fluid through the various passages of the heat exchanger is produced, and a general improvement of the operating characteristics of the heat exchanger is achieved without material increase in the complexity of the heat exchanger or its cost.

A further object of the present invention is to provide a crossflow countercurrent heat exchanger characterized by a relatively high turbulence along the heat-exchange surfaces and partitions.

Another object of the present invention is to provide a crossflow-countercurrent heat exchanger of the general character described in which the packing of the coiled tubes and their mutual spacing is not determined by the type of support members hitherto required and a general improvement of the structural integrity of the coil layers can be obained a relatively small intertube spacings.

According to an essential feature of the present invention, a crossflow-countercurrent heat exchanger comprises a multiplicity of generally coaxial arrays or layers of helicoidal and circumferentially spaced tubes centered essentially upon respective imaginary coaxial cylinders is provided with a generally cylindrical thin sheet-metal mantel centered upon the axis of the helicoidal coils intermediate a coil section surrounding at least one tube layer.

The term "shell" as used hereinafter will be assigned exclusively to the outer cylindrical casing in which the tube bundle is housed and to designate the enclosure forming the outer fluid-flow chamber. When the heat exchanger is used in cryogenic processes, in which such crossflow countercurrent units are particularly advantageous since cryogenic processes require thermodynamic reversibility with small temperature differences, the high-pressure gas is passed through the tubes while the low-pressure gas flows through the shell outside the tubes.

The "tube bundle" is defined as the entire assembly of tubes and it will be understood that the tubes themselves can be of conventional construction; more particularly, it will be understood that the tubes are composed of metals with high thermal conductivity (e.g., copper, stainless steel or other steel alloys).

For the purposes of the present description, the expression "tube sections" will be used to identify annular groups of a plurality of cylindrical tube sheets of helically coiled and mutually parallel spaced-apart tubes. The sheets of tubes or "layers" of each "section" are coaxial and have center lines lying along helices upon a respective imaginary cylindrical surface centered upon the axis of the heat exchanger. The heat exchanger is also provided in the conventional manner common with so-called spiral-tube exchangers, with manifolds or bonnets at the opposite ends of the shell and communicating with one or more tube sections or individual layers and adapted to supply the high-pressure fluid to the tube layers at one end and removing the high-fluid at the other end of the shell. The shell is provided at axially spaced locations which an inlet and an outlet in the usual manner.

Thus the present invention provides a crossflow-countercurrent heat exchanger which comprises at least one helicoidal layer of spirally (helicoidally) wound tubes surrounded by a spiral (helicoidal)-tube section advantageously comprising a plurality of spiral (helicoidal)-tube layers and a sheet-metal mantle disposed between the inner layer of coil tubes and the outer tube section. This intervening sheet-metal layer serves to strengthen the coils so that the strength and thickness of any support and spacing members can be reduced to a minimum. Especially when the inner annular helicoidal tube section, in accordance with this invention, comprises only a single layer of the tubes, it has been found to be advantageous to wind around the tubes wires whose thickness alone serves to maintain the desired spacing between the sheet-metal mantle and the tube. Through the provision of a multiplicity of single-layer sections each separated by intervening sheet-metal mantles or partitions whose thickness may be a fraction of the diameter of the tube and advantageously is equal to the spacing between cylindrical surfaces tangent to the tubes of the adjoining layers which it contacts at their confronting faces, it is found that a dense or close packing of the tubes can be maintained and the closeness of the packing can be greater than any which have been obtainable heretofore with alternately right-handed and left-handed layers of coils.

According to a further feature of this invention, a coil section surrounds an inner section consisting of a plurality of tube layers and this surrounding coil section has a multiplicity of layers of turns coiled spacedly in the same sense and such that the distance between the coils of each layer and between the surrounding coils of the adjacent layers is substantially identical. Thus each tube of an intermediate layer of the surrounding tube section has approximately six closest-neighbor tubes at the vertices of a hexagon (in the maximum spacing) centered upon the axis of the respective central tube in, substantially, a hexagonal close-packed relationship such that the perpendicular distance between each tube and all of its closest neighbors is the same. It will be understood that this arrangement is only possible when all of the tubes of each coil section (constituted of a plurality of layers) is wound in the same sense as the remaining tubes so that adjacent layers, with respect to the cross-sections of the tube in a plane transverse to the turns are staggered. When reference is made hereinafter, therefore, to a "close-packed tube section" it will be understood to identify a multiplicity of generally coaxial helicoidal layers of correspondingly coiled spaced-apart tubes with the layers so offset angularly with respect to one another that each tube lies essentially between a pair of adjoining tubes of an adjacent layer and the tubes of adjacent layers at their furthest distance from one another are disposed at the corners of an equilateral triangle in section along a plane through the tubes. The resulting arrangement is, in section, a lattice of spaced-apart tubes coiled in the same sense about a common axis. It will be seen that the closeness of the packing of the tubes is further augmented by the staggered arrangement of the tubes of the arrays and in a preferred system, it is possible for adjacent arrays to overlap or interleave as will become apparent hereinafter in practice. Moreover, such close packing fixes the positions of the tubes and eliminates the need, hitherto apparent in earlier spiral-coil arrangements in which the layers were wound in the same sense, for relatively thick support elements or rods. The only limit for the spacing between the layers is, therefore, the desired pressure drop so that the rods or support elements need have a thickness only sufficient to maintain this distance. In fact, in accordance with the principles of the present invention, the tubes of each layer have their centerlines lying upon respective imaginary cylindrical surfaces and a radial spacing between the imaginary cylindrical surfaces of each layer is at most equal to twice the outer radius or the diameter of a tube and may be appreciably less until, in the limiting sense, the tubes contact one another. The lattice-like close packing of each tube section does not permit a solution to the problem of the constancy of the length of the tubes in the conventional manner since it is not possible to reduce the number of tubes per layer in dependence upon the radius of the latter. Consequently, it is a feature of the present invention that the number of the tubes per layer of each annular tube section remains constant in spite of the fact that each layer is at a different distance from the center of the heat exchanger. It has, however, been found to be possible to substantially completely obviate any difficulties this solution might, at first blush, be considered involve if the mean tube length L from inner layer to outer layer of the tube section does not increase beyond a predetermined limit which has been found to range between 10–15% in excess of the length of the innermost tube.

Thus a multiplicity of closely packed and possibly interleaved tube layers of the same coiling sense together form a tube section, in accordance with the present invention. Within each annular tube section, the actual tube length increases from layer to layer with increasing diameter. Furthermore a minor reduction of the pitch of the turns from their innermost layer to the outermost layer also results. The average tube length, however, throughout the counterflow exchanger remains practically constant. When the difference between the length of the innermost tubes and the length of the outermost tubes of a tube section becomes so great that the distribution of the gas flow no longer can be substantially uniform, the tube section terminates and a thin-wall sheet-metal mantle is interposed between this outermost tube layer and the next outer layer which, according to this invention, then constitutes the innermost layer of the next tube section. The number of tubes per layer of each of the layers of this next section is also constant but, according to a specific feature of the invention, is increased in dependence upon the layer diameter or radius of the coil. The number of turns, as has already been demonstrated by the requirement for constancy of the product $z \cdot w$ of this latter tube section requires that the pitch of the coils decrease. The average pitch of the successively more outwardly tube section is, however, maintained substantially equal to the average pitch of the more inwardly tube sections.

This arrangement ensures that each tube section having at least two layers of similarly coiled tubes in close-packed relationship and surrounding at least two inner layers of similarly wound tubes in close-packed relationship, has a number of tubes per layer which is greater than the corresponding number of tubes per layer of the more inwardly set while the number of tubes per layer of each group of close-packed layers is constant. The number of tubes per layer in each case is such that the average length from innermost layer to outermost layer of the tubes does not exceed by more than 10 to 15% the length of the respective inner tube. Preferably, the distance a between each tube and its nearest-neighbor tubes should advantageously be between 0.2 and 2 mm. The sheet-metal mantle, in accordance with a more specific feature of the present invention, can be composed of one or more relatively wide bands in mutually parallel and axially extending relationship, of a cylindrical member with or without openings to permit the outer fluid to pass between the tube sections, or of one or more bands helicoidally surrounding the axis of the heat exchanger and wound in the same or opposite sense with respect to the winding of the tubes of the annular section which it surrounds. Between the individual bands, a clearance is maintained to ensure a uniform gas distribution over the interior of the shell and throughout the coil sections. It is, however, desirable under certain circumstances to constitute the mantle as a cylinder with a continuous or non-perforated surface and open axially at its extremities.

The above and other objects, features and advantages of the present invention will become more readily apparent from the following description, reference being made to the accompanying drawing in which:

FIGS. 1 and 2 are diagrammatically cross-sectional views through the tube bundle of a spiral-tube heat exchanger of the crossflow-countercurrent type in accordance with the principles of this invention;

FIG. 3 is an elevational view of a heat exchanger embodying the invention with parts broken away and in cross section; and

FIG. 4 is a fragmentary cross-sectional view from above a portion of the heat exchanger.

Referring first to FIGS. 3 and 4 it will be seen that a crossflow-countercurrent heat exchanger, in accordance with the principles of the instant invention, suitable for use in cryogenic processes such as air rectification, can include a generally cylindrical outer shell **10** of cast iron (advantageously lined with stainless steel) which surrounds an axis **11** and is provided with an inlet fitting **12** and an outlet fitting **13** for the throughflow of a low-pressure gas in axial direction (arrow **14**). At opposite extremities of the shell **10**, there are provided the usual plates (one of which is shown at **15**) to which the tube bundle **16** are anchored in the usual manner. Stationary heads or channels **17** and **18** are mounted as bonnets on opposite ends of the shell **10** and form manifolds communicating with the tube bundle **16**. The head **17** is provided with an inlet fitting **19** for the high-pressure fluid while a further fitting **20** is formed on the head **18** to lead the high-pressure gases away. The tube bundle **16** comprises, in accordance with the principles of the present invention, a plurality of annular tube sections two of which can be seen at **21** and **22**, each section being constituted of a plurality of layers **23** of helicoidally wound and mutually parallel but spaced-apart tubes. The tubes may be fabricated from drawn copper tubing. Between the tube sections **21**, **22**, etc., there are provided sheet-metal mantles two of which can be seen in FIG. 3. Thus a continuous cylindrical sheet-metal mantle **24**, opening into the interior of shell **10** at its opposite ends, is disposed between the tube sections **21** and **22** and extends substantially the full axial length H of the heat exchanger (i.e. the overall axial length of the coils); the mantle **24** is free from openings or discontinuities and is coaxial with the tube layers and the shell **10**. As shown at **25**, a sheet-metal mantle can consist of a band of sheet metal helicoidal wound about at least one inner layer of tube coils and surrounded by the outer section **22** thereof. The turns of the mantle **25** can be spaced apart as seen at **25'** in FIG. 3 or may abut one another in the manner described earlier.

The details of the dimensioning and construction of the tube sections and their relationship to one another are indicated in the diagrams of FIGS. 1 and 2.

Example

In accordance with the principles of this invention, a practical embodiment of the heat exchanger of the improved type adapted to be used in an air-rectification installation comprises 705 helicoidal or coil tubes in the spiral-tube bundle with a diameter d of 15 mm. Each tube is spaced from the proximal tubes of the next adjacent layer by a distance a of about 2 mm. while the center-to-center distance between the tubes of each pair of mutually overlying layers is given by $(d+a)\sqrt{3}$. The heat exchanger comprises six tube sections of which the innermost can be considered to be illustrated in FIGS. 1 and 2. Each of these sections comprises five helicoidal layers of tubes ($z_L=5$) and the number z of mutually parallel and correspondingly bound tubes of each layer varies from tube section to tube section, thus the number of tubes in the layers of the first tube section is given as $z_1=21$, in the second tube section as $z_2=22$, in the third tube section as $z_3=23$, in the fourth tube section as $z_4=24$, in the fifth tube section as $z_5=25$ and in the sixth or outermost tube section as $z_6=26$. Between each five layers of a respective tube section and the layers of the next adjacent tube section there is provided a sheet-metal mantle M. Within each tube section the length of the tubes from the innermost to the outermost layer is reduced and, accordingly, the pitch of the respective helices or coils and the respective pitch angles from the innermost layer to the outermost layer are reduced so that the inclinations of the tube sections are flattened.

In FIG. 1, which represents a vertical cross-sectional view through half of two adjacent coil sections of a heat exchanger of the general type illustrated in FIGS. 3 and 4, it will be seen that the inner section **30** comprises five layers designated generally at **31**, **32**, **33**, **34** and **35**, respectively. In general terms, the center lines of each tube of each layer, represented at center points **31***c* of FIG. 1, run helicoidally along a cylindrical surface represented by the dot-dash line **31***s* and centered upon axis A of the heat exchanger. The cylindrical surfaces **31***s* etc. are, of course, imaginary and are common to the tubes of each layer. While the tubes of each layer are shown in circular cross section in FIGS. 1 and 2 for convenience, it will be understood that a proper cross-sectional plane to the heat exchanger will intercept the tubes along ellipses and that the tubes of each array terminate in the support plate **15** along respective circles which can be considered the intersection of cylinder **31***s* with the support plate. In FIG. 1, there is shown the maximum spacing of the successive layers **31**, **32**, etc. from one another and it will be evident that the spacing between the nearest-neighbor tubes from one another at the point of closest approach is represented by the dimension a and may be 2 mm. in accordance with the principles of this invention as set forth earlier. The center-to-center distance between the tubes of each array is represented by the dimension h while the outer diameter of each tube is represented at d. The diameter (half shown) of the innermost helical coil **31** of the tube section **30** is represented at $D_{min\,1}$ and represents the minimum diameter of the tube section **30** while the outermost layer **35** of the first tube section here illustrated is represented at $D_{max\,1}$ and the mean diameter of the first tube section is designated at D_{m1}. The spacing between each layer and the next overlying layer is given by $(d+a)V3$. The center-to-center distances between each tube and its nearest-neighbor tubes can be represented by $(d+a)$ and, since $h=(d+a)$ in this case, the nearest neighbors lie at the vertices of an equilateral triangle or hexagon in hexagonal close-packed relationship.

Since within each tube section the tube length increases from the innermost layer to the outermost layer, the pitch of the turns is somewhat reduced and the pitch angle flattened from the innermost to the outermost layer of each tube section. The pitch of the innermost layer of the tube sections decreases from innermost section to outermost section while the pitch of the outermost layer of each section increases from innermost section to outermost section so that the average pitch of the turns of each section, corresponding to the pitch of the intermediate tube layer and the average tube length of each section remains constant over the entire tube bundle of the heat exchanger. If the system of FIG. 2 is now considered, it will be seen that the first tube section **50** is separated from the second tube section **60** by a sheet-metal partition M' of the type previously described and that the tube section **60** is, in turn, separated from an outer section **70** by another sheet-metal mantle M''. In this arrangement, however, the layers of each array are interleaved so that, as demonstrated with respect to the first layer **51** of the first tube section, the intertube spacing $h=(d+a)V\sqrt{3}$ and a is at its minimum value of say 0.2 mm.

Generalizing for both extremes, it will be seen that the ratio of the difference between the tube length of the outermost layer L_{max} and the tube length of the innermost layer L_{min} of each tube section to the average or mean tube length L_m thereof can be determined in the following manner:

The mean diameter of the first tube section D_{m1} bears a relationship to the mean diameter D_{m2} of the second section which is determined by the number of turns of the respective section and, since $z.w=$constant, from Equation 1, it can be seen that:

$$D_{m2}=\frac{z_2}{z_1}D_{m1} \quad (4)$$

Furthermore, the mean diameter D_{m2} of the second tube section differs from D_{m1} in accordance with the product of the number of layers Z_L and the spacing $(d+a)\sqrt{3}$ of these overlying layers (FIG. 1). It is evident that:

$$D_{m2}=D_{m1}+Z_L(d+a)\sqrt{3} \quad (5)$$

From Equations 4 and 5 it follows that:

$$D_{m1}=\frac{z_1}{z_2-z_1}Z_L(d+a)\sqrt{3} \quad (6)$$

The difference between D_{max}, the diameter of the outermost coil layer of each section, and the diameter D_{min}, of the innermost layer of the respective section is given by:

$$D_{max}-D_{min}=(Z_L-1)(d+a)\sqrt{3} \quad (7)$$

Since z_2-z_1 in general has the value of unity for the purposes of the present invention, Equations 6 and 7 demonstrate:

$$\frac{D_{max\,1}-D_{min\,1}}{D_{m1}}=\frac{Z_L-1}{Z_L z_1} \quad (8)$$

Since D and L are proportional to one another in accordance with Equation 2, it is evident that

$$\frac{L_{\max 1}-L_{\min 1}}{L_{m1}}=l=\frac{Z_L-1}{Z_L z_1} \qquad (9)$$

As will be apparent from the following table, the value of z_1 should be selected, in accordance with the principles of the present invention, such that the difference between the length of the tubes of each tube section does not exceed more than 10% the length of the smaller tube so that l is no greater than 0.1; Z_L will then have the values indicated in the table:

Table

$Z_L=3$	$Z_L=4$	$Z_L=6$
$z_1=6(l=0.111)$	$z_1=7(l=0.107)$	$z_1=8(l=0.104)$
$z_1=7(l=0.095)$	$z_1=8(l=0.094)$	$z_1=9(l=0.094)$

The lowest value of z_1 gives a length difference of somewhat more than 10% ($l>0.1$) while the higher values of z_1 yield length differences of somewhat less than 10% ($l>0.1$).

As can be demonstrated from Equation 9, for a constant Z_L, the length difference l falls with an increasing z and rises, at a constant z, with increasing Z_L. When one forms the first tube section in accordance with the left-hand section of the foregoing table ($l<0.1$, $z_1=7$ and $Z_L=3$), the remaining tube sections are easily determined. Since preferably $z_2-z_1=1$, the number of tubes per layer of the second section (z_2) is selected as 8 and the central portion of the table becomes pertinent. With $z_2=8$, Z_L must be 4 if $l<0.1$. For the third tube section, $z_3=9$ and $Z_L=6$. It is thus not necessary to maintain the value of Z_L constant throughout the heat exchanger and from section to section; it is, in fact, more advantageous to provide a value of Z_L increasing with increasing diameter as has been demonstrated. When the limits set forth above as to the length difference do not permit the value of Z_L to increase, it may be necessary in accordance with Equation 9 to increase the number of tubes outwardly from section to section by more than one.

Since the tube length must remain substantially constant, the value of w must increase in accordance with Equation 2 with increasing values of r. In the interest of satisfactory heat exchange, however, it is also necessary that height H of all the tube sections be constant. From Equation 1, therefore, it is clear that a distinct value of w yields a distinct value of z. When the variation of w requires an increase in the value of z, less than one, since the values of the z for each section must be integer, the requirement for a constant H cannot be maintained of the coils have a constant pitch within each layer.

This discrepancy can be eliminated and the tube length equalized by increasing the perpendicular center to center distances of the turns of the tubes (h) between the range $h_{\min}=(d+a)$ (FIG. 1) to $h_{\max}=(d+a)\sqrt{3}$ (FIG. 2). In the same way, the value of h can be reduced when it is intended to round off the value of z downwardly rather than upwardly. It is also possible, in accordance with a feature of this invention, to maintain the height of the counterflow exchanger constant over all of the layers by providing the tubes in the intermediate portion of the heat exchanger (i.e. somewhat between the extremities thereof) with one or more stretches of increased or decreased pitch angle or steepness. Thus the coils will be of uniform pitch at least at their axial extremities and may be of steep or shallower pitch at an intermediate location to permit the value of z to be rounded off upwardly or downwardly.

The formula for l associated with $h_{\min}$ has been given in Equation 9 and it will be readily seen that the corresponding value of l for the value of $H_{\max}$ can be derived in a similar manner and is:

$$l'=\frac{Z_L-1}{(Z_1+1)z_1} \qquad (9)$$

In the system of FIG. 2, it is apparent that the spacing between the tubes of each turn is greater than that corresponding to the hexagonal-close packed arrangement described above so that the successive layer of tubes can be interleaved or interfitted. In this case, the distance between the tubes of each layer is given by $h=(d+a)\sqrt{3}$ while the transverse or horizontal distance between a tube of each layer and the nearest-neighbor tube of the next layer is shown at a and may have a value of, say 0.2 mm. In this arrangement, one is able to provide a greater number of tube layers in a heat exchanger of a given diameter.

The advantages of a same-sense winding of the tubes of each of the multi-tube section thus lies in the fact that the intertube spacing can be made smaller than that characterizing earlier crossflow countercurrent heat exchangers in which the tube layers are wound alternately in opposite senses. Furthermore, from almost all theoretical and practical viewpoints, a marked increase in the heat-exchanging efficiency of the system results. This fact can be ascertained from the following equation:

$$F=\frac{1}{w_0}\cdot K_1 \qquad (10)$$

From this equation, it can be seen that the free cross section F, if the heat exchanger is inversely proportional to th gas speed w_0 while the constant K_1 includes a term related to the gas quantity to be treated per unit time (i.e. the desired volume rate of flow). The pressure drop Δp is related to the gas speed w_0, the number z of parallel-wound tubes of each of the layers and the number w of turns of this parallel-tubes in accordance with the following equation:

$$\Delta p=z\cdot w\cdot w_0^2\cdot K_2 \qquad (11)$$

The pressure drop thus increases with the square of the gas velocity and, when the perpendicular distance h between the tube centerpoints is constant, increases also with the axial height H of the heat exchanger, see Equation 1. Furthermore, the heat-transfer surface which remains free to effect thermal transfer between the fluids when all tubes are horizontally spaced apart from one another by the distance a is equal to the value $R\cdot L\cdot a$ where R is a function of the total number of tubes:

$$(R=z_1Z_{L1}+z_2Z_{L2}+\ .\ .\ .\ +z_nZ_{Ln}) \qquad (11a)$$

and L is the tube-bundle length. It is possible to determine the surface area also from the product $F\cdot z\cdot w$, i.e. the product of the free cross section and the number of tube spaces of each layer given by $z\cdot w$. Thus the following equation can be obtained by forming an identity by these relationships for the surface area:

$$R\cdot L\cdot a=F\cdot z\cdot w \qquad (12)$$

Finally the relationship between the surface area $R\cdot L$ and the gas speed w_0 is given by:

$$w_0^{0.609}=K_3/RL \qquad (13)$$

In comparing a countercurrent heat exchanger of the conventional type with the heat exchanger in accordance with the present invention, it can be seen that the intertube distance a can be made smaller than in any conventional configuration of the tubes. In order to demonstrate this further, Equations 10–12 can be rendered as follows:

$$R\cdot L\cdot a=\frac{z\cdot w}{w_0}K_1 \qquad (12')$$

$$z\cdot w=R\cdot L\cdot a\cdot w_0\cdot\frac{1}{K_1} \qquad (12'')$$

Inserting Equation 12″ into Equation 11, one obtains

$$\Delta p=R\cdot L\cdot a\cdot w_0^3\cdot\frac{K_2}{K_1} \qquad (11')$$

From this and Equation 13 it follows that

$$w_0^3=[K_3/RL] \qquad (13')$$

From a value where Δp=constant, it can be seen from Equations 11, 12'' and13' that

$$R\cdot L\cdot a\cdot w_0^3=\frac{R\cdot L\cdot a}{(R\cdot L)^{4.93}} \quad (14)$$

$$a\frac{}{(R\cdot L)^{3.93}}=\text{const.} \quad (15)$$

$$R\cdot L\cong\sqrt[4]{a} \quad (16)$$

The heat transfer surface is thus proportional to the 4th root of the intertube distance a and it follows for example that, for a reduction of the tube spacing from a to $a/2$, the surface requirement $R\cdot L$ becomes $RL/1.19$ and thereby falls by 20%.

From a consideration of Equation 12'', it can be seen that with a constant pressure drop, the reduction of the heating surface RL and a reduction of the tube spacing a is coupled with an increase in the gas velocity w_0. When one desires, therefore, a conventional heat exchanger and one embodying the principles of the present invention to have the same gas velocity w_0 it can be seen from Equation 10 (with w_0 held constant) and from Equation 12 with F constant that

$$\frac{R\cdot L\cdot a}{z\cdot w}=\text{constant} \quad (17)$$

For the same heating surface RL for both heat exchangers, the equation becomes a/zw=constant. A reduction in the value of a requires, therefore, a reduction in the value of zw and, according to Equation 11, a decrease in the pressure drop Δp, in accordance with Equation 11. Furthermore, if it is also required that the perpendicular distance h between the center points remain unaltered, a reduction in the axial height of the heat exchanger results. A value of F=constant ensures that the heat exchanger according to the invention, which has a smaller value of the intertube distance a and correspondingly more tube layers, has a larger diameter. For the same gas speed, therefore, a heat exchanger according to the invention, has a larger diameter, a reduced axial height and a reduced pressure drop.

The invention described and illustrated is believed to admit of many modifications within the ability of persons skilled in the art, all such modifications being considered within the spirit and scope of the appended claims.

I claim:

1. A crossflow-countercurrent heat exchanger comprising:

a generally cylindrical shell having an inlet and an outlet for the throughflow of the first heat-exchange fluid in generally axial direction through said shell;

a helicoidal-tube bundle within said shell having inlet and outlet means for the passage of a second heat-exchange fluid through said bundle in a generally axial direction counter to the direction of flow of said first fluid, said helicoidal-tube bundle comprising an annular inner tube section constituted of a plurality of coaxially nested inner layers each comprising a plurality of circumferentially spaced parallel-wound tubes having a plurality of helicoidal turns surrounding a common axis, and an annular outer tube section constituted of a coaxially nested plurality of outer layers surrounding said inner tube section, each of said outer layers comprising a plurality of circumferentially spaced parallel-wound tubes

having helicoidal turns surrounding said axis, the tube length of the tubes of each section being substantially constant and the axial length of said layers being substantially equal, the layers of each of said sections being wound in the same sence about said axis with the tubes of each section in a close-packed relationship with a tube of one layer disposed between a pair of adjacent tubes of an adjoining layer and spaced therefrom by a distance not substantially greater than the distance between said adjacent tubes;

And at least one relatively thin sheet-like mantle closely surrounding said inner tube section and interposed between said inner and said outer tube section.

2. A heat exchanger as defined in claim **1** wherein the number of tubes in each layer of each section is substantially constant but wherein the layers of said outer section each have a greater number of tubes that the layers of said inner section.

3. A heat exchanger as defined in claim **1** wherein the tube length increases from the innermost layer to the outermost layer of each tube section by a value not more than 10 to 15% of the length of tubes of the innermost layer.

4. A heat exchanger as defined in claim **1** wherein the maximum perpendicular center-to-center spacing between the tubes of each layer ranges between $h_{min}=(d+a)$ and $h_{max}=(d+a)\sqrt{3}$, where d is the outer diameter of the tubes and a is the spacing between each tube and a nearest-neighbor tube of another layer.

5. A heat exchanger as defined in claim **4** wherein the spacing a between each tube and the nearest-neighbor tubes of another layer ranges between substantially 0.2 and 2 mm.

6. A heat exchanger as defined in claim **5** wherein said tubes have central portions of a pitch angle different from the pitch angle of the tubes at their ends to maintain the length of each coil substantially constant.

7. A heat exchanger as defined in claim **1** wherein said mantle includes at least one sheet-metal band wound around said inner section.

8. A heat exchanger as defined in claim **7** wherein the turns of said band are spaced apart.

9. A heat exchaner as defined in claim **1** wherein said mantle is a cylindrical shell open at its extremities.

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