United States Patent [19]

Levy

[11] **3,773,106**

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[54] **ROTARY HEAT EXCHANGERS**

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[21] Appl. No.: **180,733**

[52] **U.S. Cl.** **165/39**, 165/86
[51] **Int. Cl.** **B60h 1/00**
[58] **Field of Search** 165/86, 88, 1, 2, 165/39, 40

[56] **References Cited**

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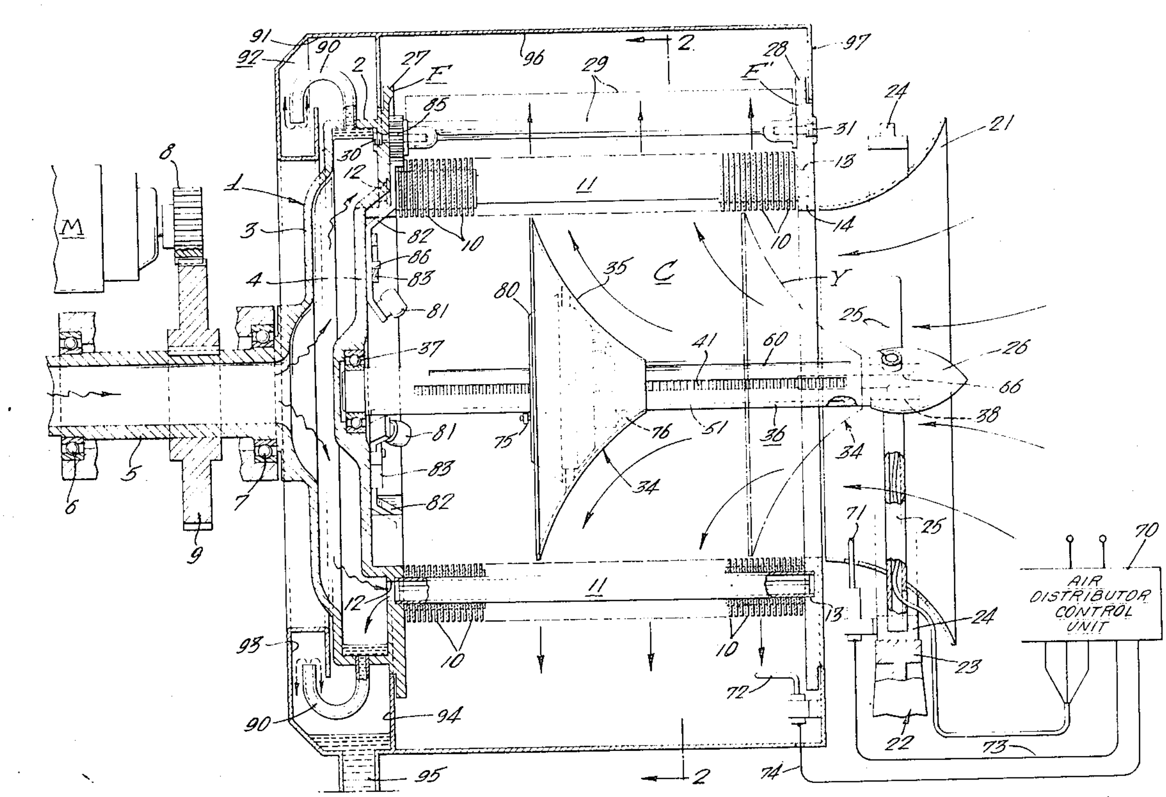
Primary Examiner—Charles Sukalo
Attorney—Charles H. Howson, Jr. et al.

[57] **ABSTRACT**

A rotary heat exchanger comprising an array of closely spaced parallel annular thermally conductive fins mounted coaxially for rotation as a unit and a plurality of thermally conductive heat exchange tubes extending through the fins. The fins are arranged and rotated at a speed to convey a fluid outwardly between the fins and provide optimum heat exchange between said fluid and another fluid in the heat exchange tubes. The heat exchange tubes are constructed to provide substantially maximum area of fluid contact surface interiorly of the tubes to minimize resistance to heat flow between the surface and the fluid in said tubes, and means are provided operable in conjunction with the fins to effect a substantial increase in fluid flow outwardly between the fins. Fluid distributing means is also provided to limit fluid flow through only the number of fins in the array required to provide optimum utilization of the heat exchanger according to the load requirement.

35 Claims, 18 Drawing Figures



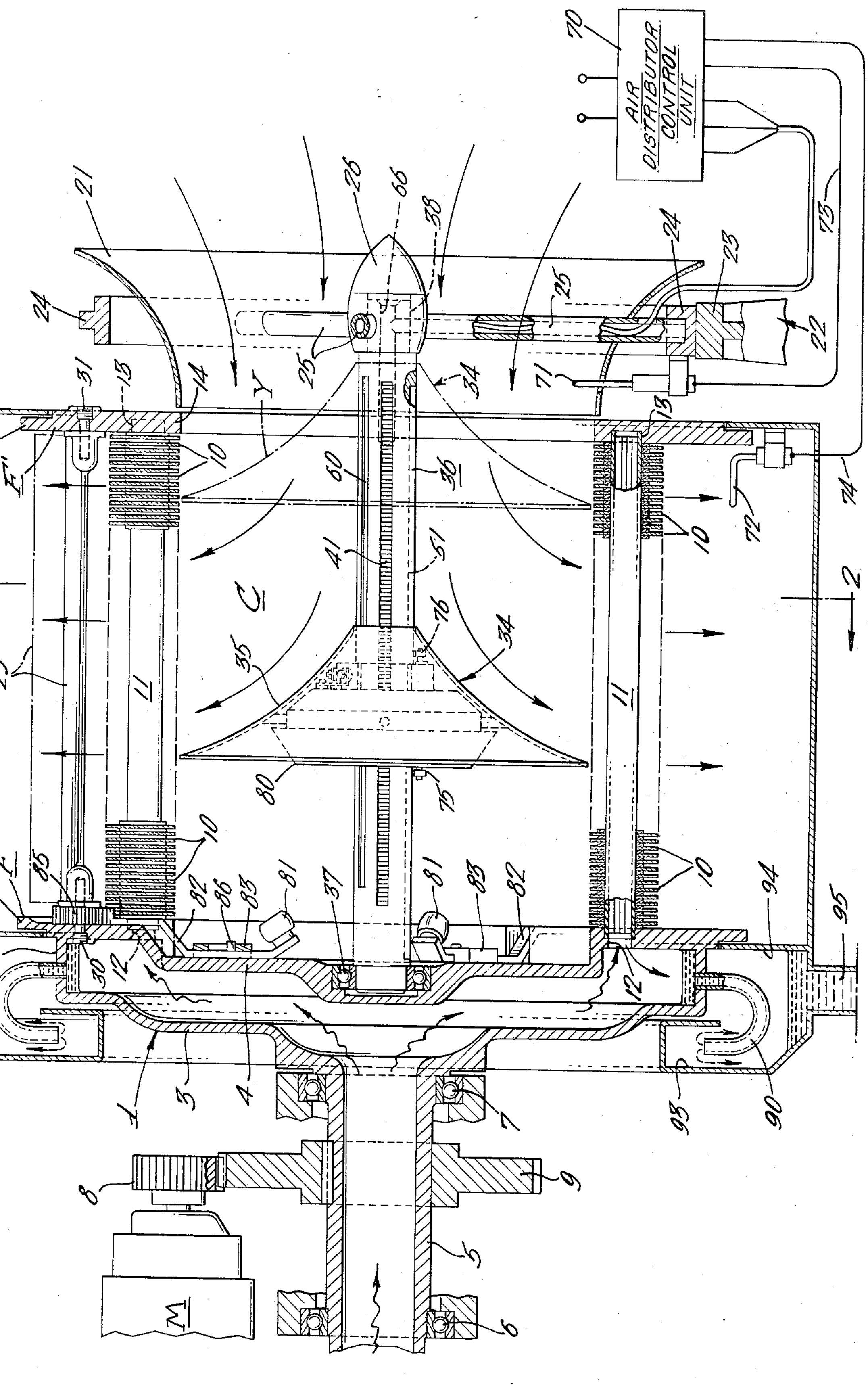


FIG. 1.

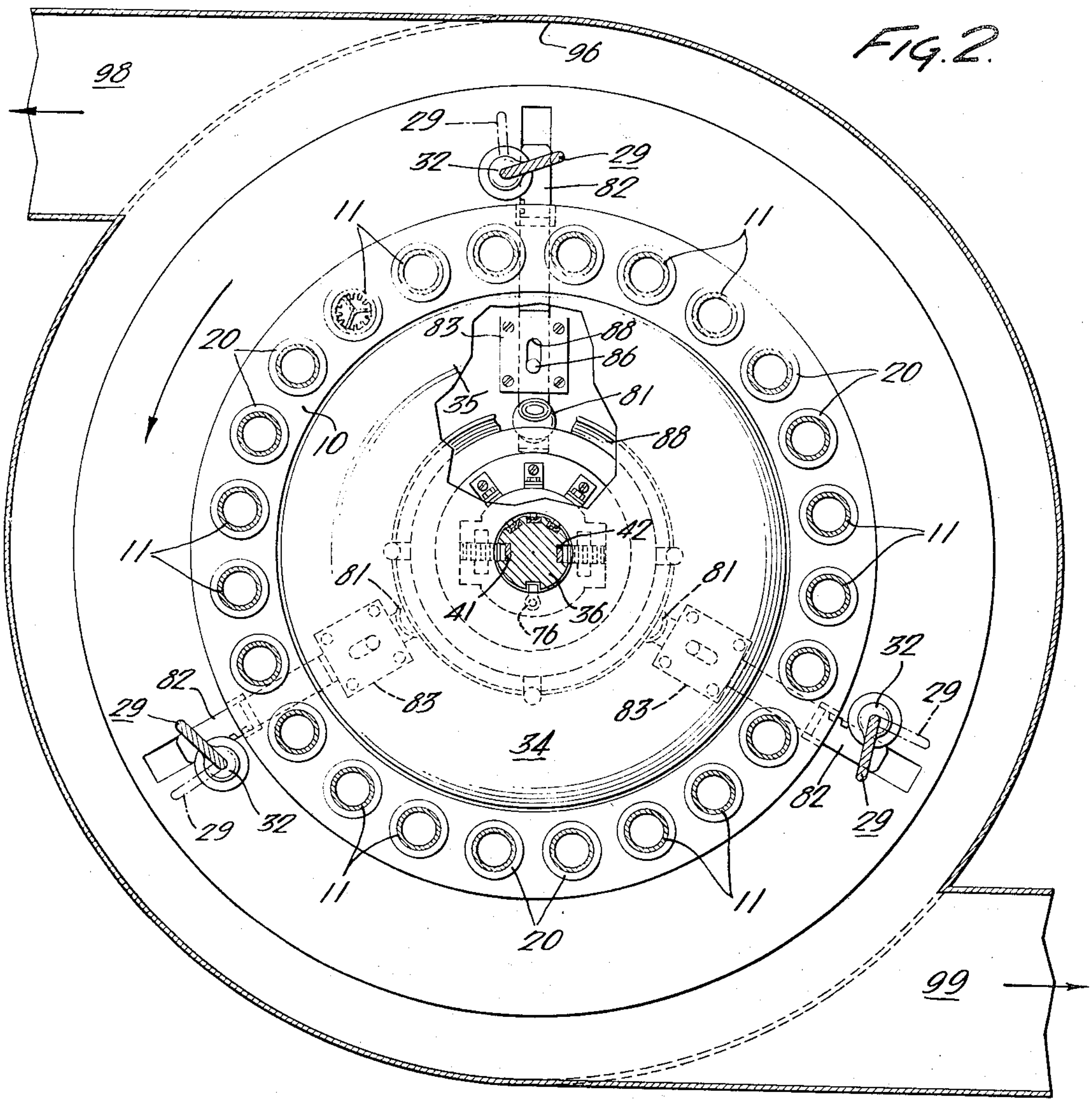

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FIG. 2.

FIG. 3.

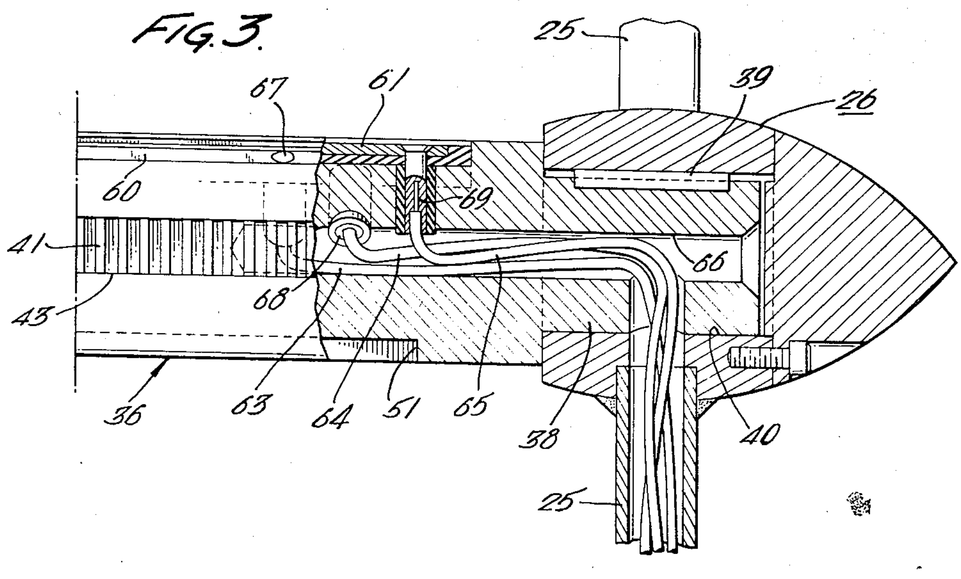
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FIG. 4.

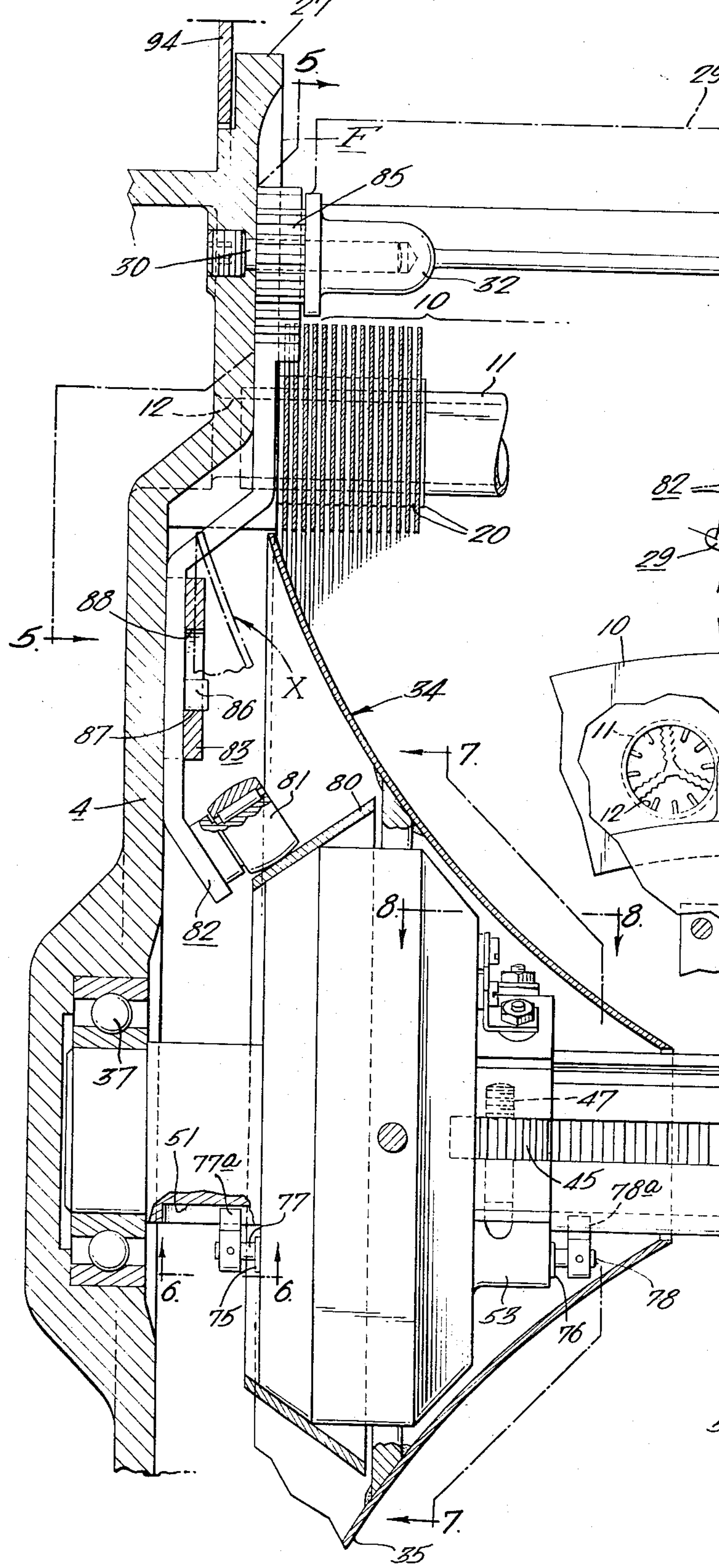


FIG. 5.

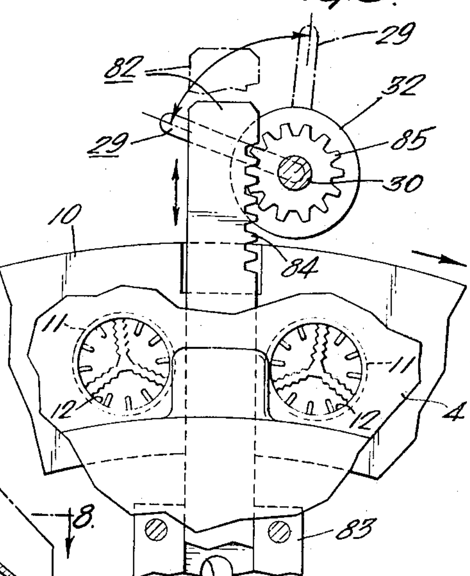


FIG. 6.

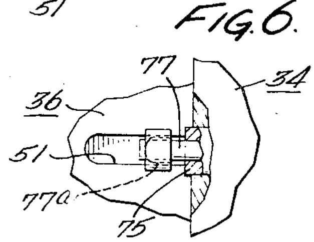


FIG. 7.

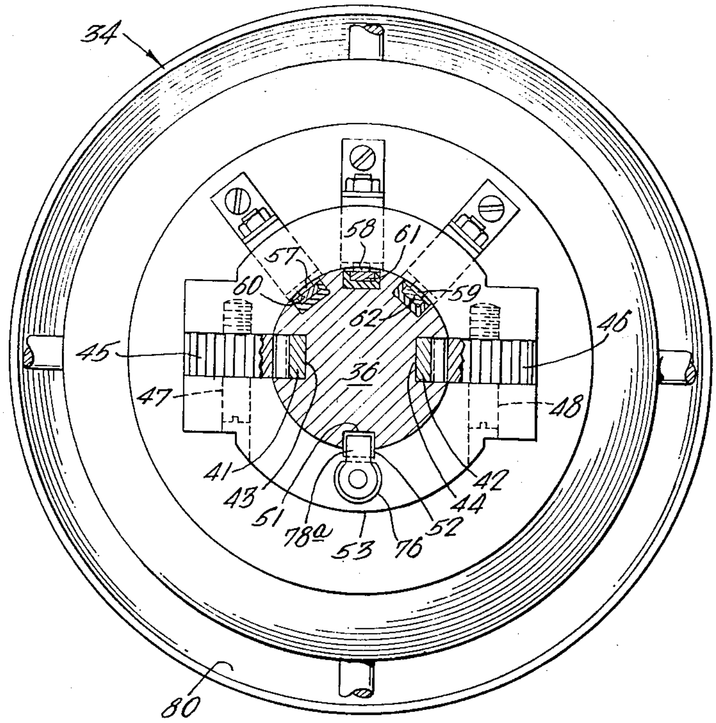


FIG. 8.

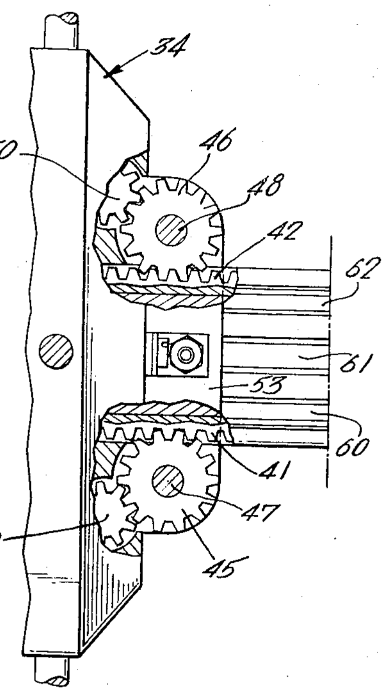


FIG. 13.

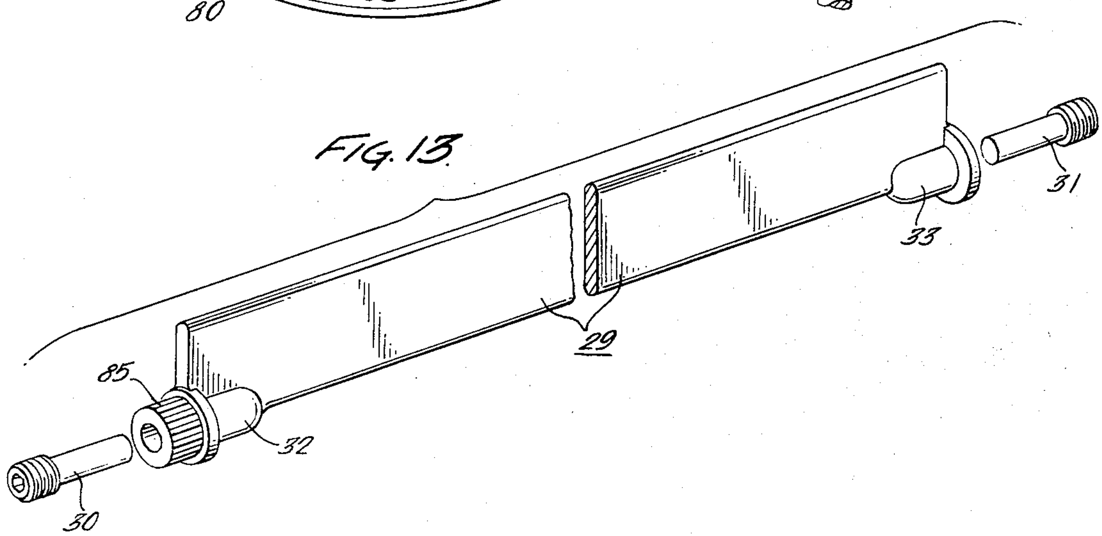


FIG. 14.

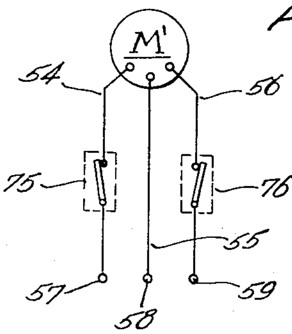


FIG. 9.

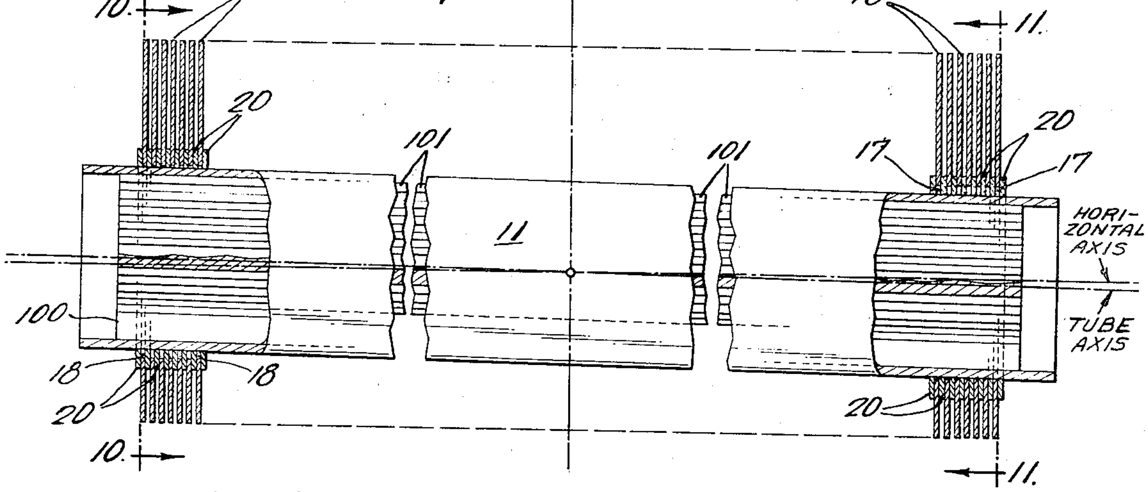


FIG. 10.

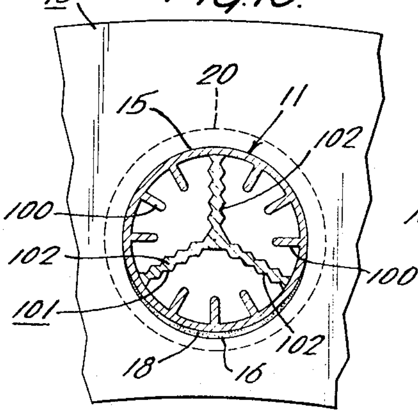


FIG. 11.

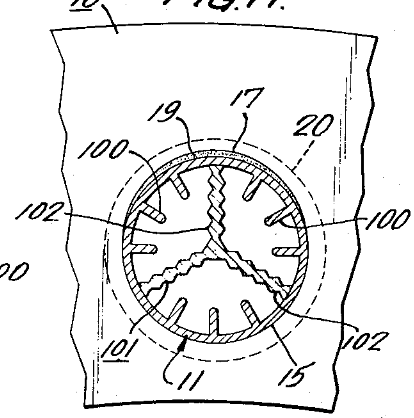


FIG. 12.

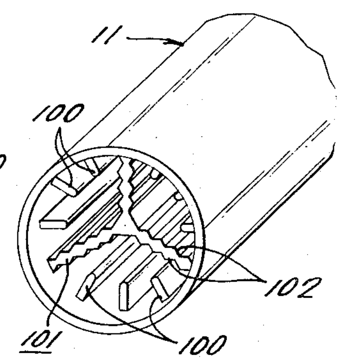


FIG. 15.

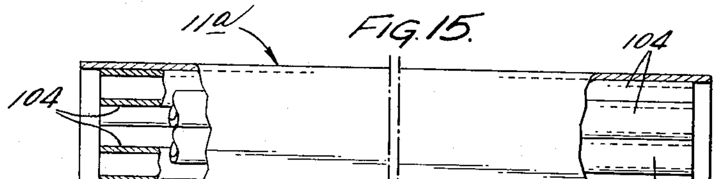


FIG. 16.

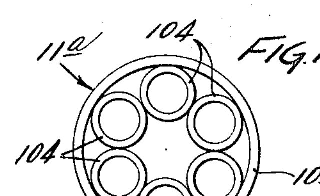


FIG. 17.

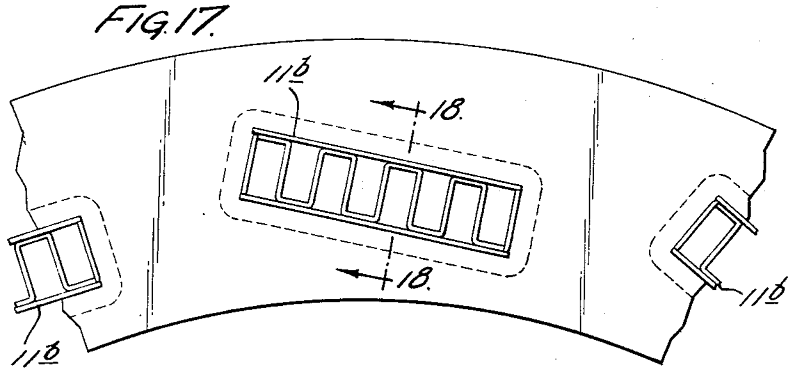
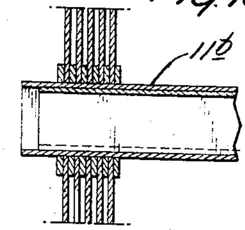


FIG. 18.



ROTARY HEAT EXCHANGERS

The present invention relates to improvements in rotary heat exchangers and more particularly to improvements in rotary heat exchangers of the type having an array of rotating annular fins through which a fluid is entrained and accelerated primarily by viscosity shear forces to the velocity providing efficient heat exchange between said fluid and another fluid in heat exchange tubes having thermal contact with the fins.

In rotary heat exchangers of the type described there are certain categories of resistance to heat flow that reduce the total heat exchange between the two fluids and consequently lower the thermal effectiveness of the heat exchanger. A major resistance to heat flow is between the exterior fin surface extending from the heat exchange tubes and the fluid discharged outwardly between the rotating fins. Another substantial resistance to heat flow is between the fluid in the heat exchange tubes and the interior wall surfaces of the tubes. A third resistance is the customary resistance to heat flow through the walls of the heat exchange tubes but this can be minimized simply by the use of heat exchange tubes having walls of a high heat conductive material as thin as possible that is structurally adequate.

According to the present invention it has been discovered that the resistance to heat flow between the exterior fin surfaces extending from the heat exchange tubes and the fluid discharged outwardly between the rotating fins can be greatly reduced by substantially increasing the volume of fluid flow outwardly through the rotating fins of the heat exchanger.

In addition, it has been discovered that the flow of fluid outwardly between the rotating fins of a heat exchanger of the present type is not directly proportional to the number of fins or the axial length of the fin array, but is attenuated according to the number of fins, the axial spacing thereof and the inner diameter of the fins. Consequently there is a marked decay or reduction in fluid flow through the fins per unit length of the heat exchanger as the ratio of the sum of the fin spaces to the inner diameter of the fins is increased and the length to diameter ratio of the fin array in the heat exchanger is increased. However, the heat exchange area increases with increasing length. Hence, a design point may be determined that balances the flow attenuation effect with the area increase effect to materially increase the volume of fluid flow outwardly toward the fins and thereby yield optimum total heat exchange between the fluid discharged outwardly through the array of fins and the fluid in the heat exchange tubes.

According to the present invention it has been discovered that the resistance to heat flow between the fluid in the heat exchange tubes and the interior wall surface of the tubes can be greatly reduced and effectively minimized by maximizing the area of the interior fluid contact surface of the tubes commensurate with pressure drop considerations. In addition, when the invention is employed as a condenser it has been found that canting or sloping the heat exchange tubes at a small angle to the axis of rotation of the heat exchanger to assist in draining the condensate, functions to reduce the thickness of the liquid film in the tubes and thus further decreases the resistance to heat flow through the condensed liquid to the inner surfaces of the walls of the heat exchange tubes. Also, if the heat exchange

tubes are canted or sloped sufficiently to offset fabrication errors so that no condensate can accumulate in the tubes, a source of unbalance and vibration is eliminated.

Furthermore, during operation of the heat exchanger at partial loads, and/or when the fluid inlet temperature to the heat exchange tubes is below design temperature, it has been found desirable to reduce fluid flow outwardly through the array of fins to match the reduced load requirement and/or the improved heat transfer rate of the heat exchanger and thereby reduce the pumping power input required to rotationally drive the heat exchanger. In accordance with the present invention this may be accomplished through modulation of the fluid flow, for example, by directing the fluid outwardly through only the number of fins in the array required to provide optimum utilization of the heat exchanger at any given load. This also minimizes the length/diameter ratio effect and resulting flow decay at low loads.

With the foregoing in mind an object of the present invention is to provide a rotary heat exchanger of the type described having heat exchange tubes of novel construction and arrangement operable to minimize resistance to heat flow between a fluid in the tubes and the interior wall surfaces of the tubes.

Another object of the invention is to provide a rotary heat exchanger of the stated type embodying a novel construction and arrangement of parts operable to minimize resistance to heat flow between the exterior fin surfaces extending from the heat exchange tubes and a fluid discharged outwardly between the rotating fins.

Another object of the invention is to provide a rotary heat exchanger as set forth embodying novel means and construction operable substantially to augment the fluid shear force action of the array of rotating fins and substantially increase the flow of fluid outwardly through the array of fins of the heat exchanger.

A further object of the invention is to provide a rotary heat exchanger of the type described having novel flow distributing means operable to cause outward discharge of fluid through only the number of fins in the array that is required to provide optimum utilization of the heat exchanger at any given load.

A still further object of the invention is to provide a heat exchanger as set forth having means for automatically controlling the aforesaid fluid flow distributing means in response to selected operating conditions in the heat exchanger.

These and other objects of the invention and the various features of the construction and operation thereof in accordance with the invention are hereinafter set forth and described with reference to the accompanying drawings, in which:

FIG. 1 is a sectional view diametrically of a rotary heat exchange condenser made according to and embodying the present invention;

FIG. 2 is a sectional view on line 2—2, FIG. 1 with a portion of the flow distributor cone broken away to show certain structural details hereinafter described;

FIG. 3 is an enlarged fragmentary view, partially in section, showing structural details of the flow distributor support shaft at the fluid inlet end of the heat exchange condenser;

FIG. 4 is an enlarged fragmentary view, partially in section, of a portion of the disclosure of FIG. 1, illustrating the flow distributor disposed at the inner end of the array of annular fins and one embodiment of means for actuating a plurality of fluid flow augmentation blades and for limiting traverse of the flow distributor in the inward direction;

FIG. 5 is a fragmentary sectional elevation on line 5—5, FIG. 4;

FIG. 6 is a fragmentary inverted plan view, partially in section, from line 6—6, FIG. 4;

FIG. 7 is a traverse sectional view on line 7—7, FIG. 4, with the fluid flow distributing cone removed to illustrate certain structural details hereinafter described;

FIG. 8 is a fragmentary plan view, partially in section, viewed from line 8—8 FIG. 4, showing one embodiment of actuating means for traversing the flow distributor;

FIG. 9 is an enlarged fragmentary diametrical view of a portion of FIG. 1 showing structural details of the annular fins and heat exchange tubes, exaggerated to show the inclination or slope of the tubes relative to the axis of rotation of the condenser;

FIGS. 10 and 11 are sectional views taken on lines 10—10 and 11—11, respectively, of FIG. 9;

FIG. 12 is a fragmentary perspective view illustrating the end construction of the heat exchange tube shown in FIG. 9;

FIG. 13 is a detached perspective view of one of the fluid flow augmentation blades hereinafter described;

FIG. 14 is a schematic view illustrating the arrangement and operation of means for limiting traverse of the flow distributor;

FIG. 15 is a detached elevation view, partially in section, showing another embodiment of heat exchange tube;

FIG. 16 is an enlarged end elevation of the heat exchange tube shown in FIG. 15;

FIG. 17 is a fragmentary end view of the annular fin showing still another embodiment of heat exchange tube; and

FIG. 18 is a fragmentary sectional view taken on line 18—18, FIG. 17.

Referring now more particularly to FIG. 1 of the drawings, there is illustrated one embodiment of a rotary heat exchange condenser made according to the present invention. As shown, the condenser comprises a cylindrical body or casing 1 of selected diameter and relatively short axial length having a continuous circumferentially extending wall 2 and axially spaced end walls 3 and 4, respectively. Extending coaxially outward from the casing side wall 3 is a tubular shaft 5 that is in communication with the interior of the casing 1. The shaft is rotatably mounted in bearings 6 and 7 and said shaft 5 and casing 1 are rotationally driven at the desired speed by means of an electric motor M driving a gear 8 which in turn drives a gear 9 on the shaft 5.

Mounted outwardly adjacent the opposite wall 4 of the casing 1 for rotation therewith is an array of annular fins 10 arranged coaxially of the casing 1 in predetermined equally spaced parallel relation. The fins 10 consist of separate or independent annular disk elements supported and secured in the desired closely spaced parallel relationship with respect to one another and the casing 1 by means of a plurality of heat exchange tubes or pipes 11 that extend longitudinally through the array of fins as shown.

As shown in FIG. 1, the outer radius of all of the fins 10 is the same and the inner radius of all of the fins is also the same. The inner peripheral edges of the fins 10 define internally thereof a coaxial inlet chamber C for the heat exchanger fluid to be discharged outwardly by and between the rotating fins 10 as hereinafter set forth.

The heat exchange tubes 11 are arranged in equally spaced relation circumferentially of the fins 10 and casing 1 as shown in FIG. 2 of the drawings. The inner ends of the tubes are mounted and secured in corresponding openings 12 provided through the casing end wall 4 so that the interiors of the tubes 11 are in communication with the interior of the casing 1. The other ends of the tubes 11 are disposed in recesses 13 provided in an annular end ring 14 that is disposed coaxially of the condenser adjacent the outermost of the fins 10. The end ring 14 effectively closes the outer ends of the tubes 11 and also supports them in the desired relationship, and the inner diameter of the end ring 14 is substantially the same as the inner diameter of the adjacent array of fins 10 so as not to restrict flow of fluid into the chamber C.

The fins 10 and heat exchange tubes 11 are fabricated of metal having high thermal conductivity such as, for example, copper or aluminum and the fins preferably are bonded to said tubes 11 by brazing, soldering or the like, to provide maximum thermal conductivity therebetween.

In accordance with the present invention, the heat exchange tubes 11 preferably are disposed at a small angle or slope relative to the axis of rotation of the condenser so that the tubes diverge slightly inwardly from the inlet end of the fluid chamber C to the casing 1 as more particularly illustrated in FIG. 9 of the drawings. This angular disposition or slope of the heat exchange tubes 11 relative to the rotation axis aids in draining the condensate from the tubes so that the thickness of the liquid film in the tubes is reduced and the resistance to heat flow from the fluid in the heat exchange tubes 11 to the interior wall surface of the tubes is materially decreased. In practice the angle of slope or divergence of the heat exchange tubes 11 relative to the rotation axis can be quite small and it has been found that a slope, for example, of the order of about 1/32 inch per foot of length of the array of fins is sufficient to provide the desired reduction in resistance to the heat flow through adequate drainage of the liquid condensate from the heat exchange tubes into the casing 1.

In the illustrated embodiment of the invention, the divergence or slope of the heat exchange tubes 11 relative to the rotation axis and through the array of fins 10 is accomplished by stamping or punching tube openings 15 in the fins 10 that are radially elongated or out-of-round to the extent of the overall slope of the tubes 11 as indicated at 16 and 17 in FIGS. 10 and 11, respectively, of the drawings. By this construction a single jig or fixture can be employed for forming similar openings 15 in all of the fins 10 and the small spaces or voids between the edges of the openings 15 and the heat exchange tubes 11 are completely filled with the brazing or soldering metal as indicated at 18 and 19, respectively, in FIGS. 10 and 11, to insure maximum thermal conductivity between the tubes 11 and fins 10. Uniform equal spacing of the fins 10 on the heat exchange tubes may be provided by employing spacer rings 20 between adjacent fins 10 as shown in FIG. 9.

An outwardly flared or bell-shaped fluid intake member 21 is disposed coaxially adjacent the outer face of the end ring 14. As shown in the drawings, the member

21 does not rotate with the heat exchange and is fixedly mounted in the described position by means of a stationary support structure 22 comprising a base 23, ring 24 and radical spokes 25 that extend through the member 21 and terminate at their inner ends and are secured in a coaxially disposed hub member 26. The smaller end of the intake member 21 adjacent the ring 14 has a diameter the same as the inner diameter of the ring and fins 10 to provide smooth, uninterrupted flow of fluid inwardly through the member 21 and ring 14 to the chamber C.

The axial spacing or distance between the adjacent fins 10 is determined with relation to the rotational speed at which the fins are driven and to the inner and outer radii of the said fins so as to utilize the viscous properties of the fluid and the shear forces exerted thereon by the rotating fins 10 to pump the fluid radially outward between said fins. Thus, upon rotation of the fins 10 at the predetermined speed related to the spacing of the fins 10 and their radii, fluid is caused to flow inwardly through the intake member 21 and ring 14 to the chamber C and enter radially into the spaces between the fins 10 where it is accelerated by the shear forces generated by the velocity difference or slip between the fins and the fluid. As the fluid is accelerated and forced outwardly between the fins, the fluid is pressurized and ultimately discharged at the outer edges of said fins. During passage between the fins the fluid particles follow a spiral trajectory. For optimum results the axial spacing of the fins, their speed of rotation and their inner and outer radii are correlated so that the fluid passing between the fins is accelerated to a velocity substantially less than the outer peripheral velocity of the fins in order to retain the fluid between the fins 10 the longer time required to provide the optimum total heat flux or total heat exchange between the fluid passing between the fins and another fluid in the heat exchange tubes 11.

The axial spacing between the fins 10 and the relationship of the inner radius of the fins to their outer radius may vary within predetermined ranges or limits for any given range of speeds of rotation of the condenser. The nature of the flow for rotational shear force devices is completely described by the Taylor number, N_{Ta}, wherein:

$N_{Ta} = d^2w/\nu$

d = distance between fins

w = angular velocity

ν = kinematic viscosity

It is known that most efficient pumping of a fluid occurs when $N_{Ta} = 3.25$. However, efficient fluid pumping does not lead to an efficient heat exchanger. Efficient pumping occurs when the energy transfer to the fluid is maximized whereas efficient heat exchange depends upon both the fin area and the difference in velocity between the fins and the fluid flowing between them. Thus, for heat transfer, the Taylor number is not adequate alone to completely describe an optimum configuration, and it has been determined that for various combinations of inner radius (Ri) and outer radius (Ro) of the fins the Taylor number for an efficient heat exchanger is always greater than that for most efficient pumping and is normally in the range of from about 5 to about 7. Thus, for example, for air the combination of Taylor number and geometrical terms given by $N_{Ta}(Ri/Ro)^2$ will be in the neighborhood of 4 for an efficient heat exchanger.

As previously pointed out, the resistance to heat flow between the surfaces of the fins 10 and the fluid discharged outwardly between the fins can be materially decreased by substantially increasing the volume of fluid flow through the rotating array of fins of the heat exchanger. Also, not only does such an increase in the volume of fluid flow through the array of fins 10 decrease resistance to heat flow between the fin surfaces and the fluid, but such an increase in fluid flow also functions to offset or reduce the aforesaid decay or reduction in fluid flow through the fins due to attenuation by reason of the effect of the length/diameter ratio of the array of fins 10.

According to the present invention, it has been discovered that fluid flow outwardly through the rotating array of fins 10 can be augmented and increased with a substantial increase in heat transfer by providing at opposite ends of the array of fins continuous, annular, outwardly projecting flanges or flange portions that extend a predetermined distance outwardly beyond the outer peripheral edges of the fins 10 and rotate as a unit with an array of fins. Preferably, the flanges or flange portions are disposed in the radial position parallel to the fins 10 for maximum flow augmentation.

In the embodiment of the invention shown in FIG. 1, such a radially projecting flange or flange portion is provided at the inner end of the fin array by constructing tne end wall 4 of the casing 1 of greater radius than the outer radius of the fins 10 and providing thereon a radially projecting annular flange portion 27. At the outer end of the fin array the flange or flange portion is provided by constructing the ring 14 of an outer radius greater than the outer radius of the fins so that the peripheral portion 28 of the ring 14 projects or extends radially beyond the arrays of fins as illustrated. The outer radius of the casing flange 27 and ring portion 28 is the same.

Thus, flange 27 together with the adjacent portion of the casing wall 4 extending outwardly beyond the fins 10 provide a fluid flow augmentation flange F at the inner end of the array of fins 10 and the projecting peripheral portion 28 of the ring 14 provides a similar flow augmentation flange F′ at the outer end of the array of fins. In functioning to substantially increase fluid flow through the array of fins, the flanges F and F′ operate as a radial diffuser and conserve the static pressure of the heat exchange fluid in that the flanges F and F′ not only prevent abrupt axial expansion of the fluid to the atmosphere but contain the fluid and effect a gradual radial deceleration thereof without disrupting tangential flow.

The ratio of the diameter of the flanges F and F′ to the outer diameter of the fins 10 may vary according to the dimensional limitations required for a particular heat exchanger, and good results have been obtained within the ratio range of about 1.2 to a maximum of about 1.5, it having been found that a diminishing improvement in results is obtained at ratios greater than about 1.5. In the illustrated embodiment of the invention, the ratio is about 1.3 and provides an increase in fluid flow volume of about 15 percent with an accompanying heat transfer increase of from about 7 percent to about 10 percent.

It has also been discovered that the volume of fluid flow outwardly through the array of fins 10 can be further increased about an additional 50 percent in the peak or full load operating conditions of the heat exchanger by providing circumferentially about the array of fins 10 and between the flanges F and F' a plurality of axially extending radially disposed flow augmentation blades 29. The blades 29 rotate with the array of fins 10 as a unit, and in the illustrated embodiment of the invention three blades 25 are provided and disposed in equally spaced relation circumferentially about the array of fins in spaced relation radially outward therefrom, for example, as shown inFIGS. 1 and 2 of the drawings. The blade construction is shown more clearly in FIG. 13.

The blades 29 are each pivotally mounted upon a pair of axially aligned pintles 30 and 31 that are fixedly mounted, respectively, in the end flangesF and F' and received within the bores of socket portions 32 and 33 formed at opposite ends of each of the blades 29 (see FIG. 13). Pivotal movement of the blades 29 is limited between the neutral or feathered position shown in solid lines in FIG. 2 and the true radial position shown in dotted lines, and suitable mechanism, hereinafter described, is provided for pivotally actuating the blades 29 between their neutral and radial positions as required. In the event that in a particular heat exchanger it may be desired for some reason to provide the blades 29 but eliminate the flanges F andF', it will be obvious that the blades can be supported and pivotally mounted as described in the outer ends of axially aligned pairs of radially projecting fingers or spokes (not shown).

Depending on the operating conditions of the heat exchanger, the blades 29 normally are rendered inoperative and disposed in the feathered or neutral position in or parallel to the streamline of flow of the heat exchange fluid discharged outwardly between the array of fins 10. The blades 29 are actuated to their radial position and rendered operative to augment pumping of the fluid only at peak or full load operating conditions of the heat exchanger. The additional flow augmentation provided by the blades 29 at full load conditions enables the heat exchanger to be made smaller and thereby more compact than would otherwise be possible.

At less than full load operation of the heat exchanger or on occasions, for example, when the temperature of the heat exchange fluid entering the chamber C is below the design temperature, it is desirable, as previously pointed out, to reduce fluid flow outwardly through the array of fins 10 to match the reduced load requirement or heat transfer rate and thereby reduce the pumping power input required to rotationally drive the heat exchanger. In accordance with the present invention, this is effected by modulating the flow of the heat exchange fluid and directing the fluid outwardly through only the number of fins in the array required to provide optimum utilization of the heat exchanger at any given load. As previously stated, this flow modulation also minimizes the length/diameter ratio effect of the fin array and the resulting flow decay at low loads.

In the illustrated embodiment of the invention, modulation of the flow of heat exchange fluid is accomplished by means of a flow distributor 34 that is movable coaxially within the inlet chamber C of the rotary heat exchanger to the axial position lengthwise of the array of fins 10 that provides the optimum operating condition of the heat exchanger at any given load. The flow distributor 34 is of generally conical configuration and is provided with a curved or arcuate surface 35 that has its peripheral edge disposed closely adjacent the inner edges of the fins 10. The arcuate surface 35 faces the fluid inlet to the chamber C and functions to guide and direct the heat exchange fluid entering chamber C smoothly to and between the fins 10 in the portion of the array of fins that extends outwardly or endwise from the peripheral edge of the flow distributor 34.

The flow distributor 34 is not rotatable with the heat exchanger and is slidably mounted for coaxial movement, interiorly of the fluid inlet chamber C, upon a non-rotating coaxial shaft 36. The shaft 36 has its inner end secured in a bearing 37 mounted in the adjacent end wall 4 of the rotationally driven casing 1. The other or outer end of the shaft 36 is provided with a reduced end portion 38 that is secured by a key 39 in the bore 40 of the aforesaid stationary member 26 that is fixedly supported coaxially of the heat exchanger by the radial spokes 25 of the stationary support structure 22.

Movement of the flow distributor 34 axially along the shaft 36 may be accomplished, for example, by means of a rack and pinion arrangement. In the illustrated embodiment of the invention, axially extending toothed rack members 41 and 42 are fixedly mounted in elongated grooves or recesses 43 and 44 formed in the shaft 36 at diametrically opposite sides thereof as more clearly shown in FIG. 7 of the drawings. Meshed with the racks 41 and 42 are pinions 45 and 46, respectively, pivotally mounted in the flow distributor 34 by means of stub shafts 47 and 48. The pinions 45 and 46 are driven through suitable gear trains 49 and 50, respectively, by one or more reversible electric motors as may be required, one such motor M' being shown schematically in FIG. 14 of the drawings. To further stabilize the flow distributor 34 on the shaft 36 for sliding movement therealong, an axially extending slot or keyway 51 is provided therein and engaged by a key member 52 carried by the hub portion 53 of the distributor 34, for example, as illustrated in FIG. 7.

The racks 41 and 42 extend continuously substantially the length of the shaft 36 so that the flow distributor 34 is movable along the shaft 36 between the inner limit position indicated by the broken line designated X shown in FIG. 4, and an outer limit position indicated by the broken line designated Y in FIG. 1 of the drawings.

The reversible motor M' for driving the pinions 45 and 46 is conveniently mounted within the hub portion 53 of the flow distributor and electrical energy is supplied thereto by conductors 54, 55 and 56 leading to the motor M' from contact brushes 57,58 and 59, respectively, mounted in and carried by the flow distributor 34. The brushes 57, 58 and 59 are in sliding electrical contact or engagement, respectively, with axially extending bus bars 60, 61 and 62 that are mounted in suitable recesses or grooves formed in the shaft 36.

Electrical current is supplied to the bus bars 60, 61 and 62 by conductors 63, 64 and 65, respectively, that lead through one of the stationary support spokes 25 and an axial bore 66, provided in the shaft 36, to suitable connectors 67 68 and 69. At their outer ends the conductors 63, 64 and 65 are connected to the output terminals of a suitable control unit 70 that operates automatically as hereinafter described to effect traverse of the distributor axially along the shaft 36 in accordance with variations above and below a predetermined differential between the temperature of the incoming heat exchange fluid entering the inlet chamber C as sensed by a thermocouple 71 and the temperature of the fluid after it has been discharged through the fins 10 as sensed by a thermocouple 72. The thermocouples 71 and 72 are connected to the control unit 70 by suitable conductors 73 and 74, respectively.

Limit switches 75 and 76 are mounted on and carried by the flow distributor 34 to open the circuits and deenergize the motor M' at the inner limit position X of the distributor 34 and at the outer limit position Y thereof, respectively, for example, as shown in FIG. 4 of the drawings. The limit switch 75 is connected in the conductor 54 to the motor M' and the switch 76 is similarly connected in the motor conductor 56 (FIG. 14). These limit switches are each normally spring-biased to the closed circuit position and are actuated to open positions by inwardly displaceable axial plungers 77 and 78, respectively, having radial fingers 77*a* and 78*a* that project into and move longitudinally within the aforesaid keyway 51 in the shaft 36. The fingers 77*a* and 78*a* are adapted to engage the opposite end surfaces of the keyway 51 to effect inward actuation of the switch plungers 77 and 78 in the described limit position of the distributor 34.

The switch 75 and its actuating plunger 77 are mounted on the distributor 34 and arranged so that as the distributor 34 reaches the inner limit position X shown in FIG. 4 of the drawings, the plunger 77 engages the adjacent inner end surface of the keyway 51 thereby actuating the plunger to open the switch 75 and prevent further energization of the motor M' in the inward traverse direction of the distributor 34 at least until the distributor has been moved out of the inner limit position X by operation of the motor M' in the opposite direction. The other limit switch 76 is similarly mounted and arranged to cause the plunger 78 thereof to engage the outer end surface of the keyway 51 and open said switch 76 as the flow distributor 34 reached its outer limit position Y, thereby preventing further energization of the motor M' in the outward traverse direction of the distributor 34 at least until the distributor has been moved inwardly from the outer limit position Y by operaton of the motor M' in the opposite or inward direction.

In the illustrated embodiment of the present invention use is made of the inward traverse of the flow distributor 34 on the shaft 36 to effect actuation of the augmentation blades 29 to their radial position upon movement of the distributor 34 to the inner limit position X shown in FIG. 4. To this end a cam 80 of annular conic section is coaxially mounted on and carried by the flow distributor 34 and disposed so that as the distributor moves inward from the solid line position shown in FIG. 4 to the limit position X, the cam 80 is caused to engage a plurality of cam rollers 81 rotatably mounted at the inner ends of a plurality of radially extending push rods 82.

The push rods 82 are mounted for limited radial sliding movement in brackets 83 secured to the adjacent surface of the casing end wall 4. On the outer end of each push rod 82 is formed a toothed rack section 84 (FIG. 5) that is meshed with a pinion 85 fixedly mounted on, or formed integrally with, the socket portion 32 at the inner end of each blade 29 coaxially with respect to the pinion 30 as illustrated in FIG. 13 of the drawings.

The arrangement is such that upon inward movement of the distributor 34 to the limit position X shown in FIG. 4, the cam 80 engages the cam rollers 81 and actuates the push rods 82 radially outward to cause the rack section 83 thereon to rotate the pinions 85 and pivot the blades 29 from the neutral positions shown in solid lines in FIG. 2 to the radial positions shown in broken lines. Upon outward traverse of the flow distributor 34 away from the inner limit position X the pressure of the discharged heat exchange fluid operates automatically to return the blades 29 to the neutral position and return the push rods 82 to the inward position thereof determined by engagement of a pin 86 on each rod 82 with the inner end 87 of a slot 88 provided in the brackets 83.

In the condenser embodiment of the invention shown in FIG. 1 the vapor to be condensed enters the casing 1 through the tubular shaft 5 and then passes into the heat exchange tubes 11 where the vapor is condensed by heat exchange with a cooling fluid, such as ambient air, discharged outwardly between the spaced fins 10 as herein described. The condensate thus formed in the tubes 11 flows back into the casing 1 from which it is discharged radially by centrifugal force generated by rotation of the condenser. In the arrangement shown, the condensate is discharged from the casing 1 through a plurality of U-shaped tubes 90 that form liquid traps which prevent discharge of the vapor directly from the casing and cause the vapor to be diverted into the heat exchange tubes 11. Upon leaving the U-shaped tubes 90 the vapor condensate is discharged radially outward against the inner surface of the cylindrical peripheral wall portion 91 of a stationary annular compartment 92 that circumscribes the rotating casing 1 and has spaced apart end wall portions 93 and 94 which lie closely adjacent the peripheral portions of the casing end wall 3 and flange F, respectively. Condensate collecting in the compartment 92 discharges therefrom through a drain 95.

The condensate compartment 92 may be formed as an integral part of a plenum chamber for the heat exchange fluid discharged outwardly through the array of fins 10, comprising a stationary cylindrical wall portion 96 circumscribing the rotating fin array and blades 29 and enclosed by the spaced apart end wall portion 94 and an opposite outer wall portion 97 that lies closely adjacent the rotating end flange F', as shown in FIG. 1. As shown in FIG. 2 of the drawings, the plenum chamber preferably is provided with a pair of diametrically disposed tangentially extending fluid outlets 98 and 99, respectively.

In addition, in accordance with the present invention and in order further to decrease and minimize the resistance to heat flow from fluid within the heat exchange tubes 11 to the interior wall surface of the tubes, the heat exchange tubes 11 are constructed to provide, within the limits of practicability, a maximum area of fluid contact surface within the tubes 11. To this end, each of the heat exchange tubes 11 in the embodiment of the invention shown in FIGS. 1 and 2 is constructed to provide interiorly of the tubes the cross-sectional configuration, for example, illustrated in FIGS. 10, 11 and 12 of the drawings. As shown, each of the tubes 11 comprises an extruded circular tube having a plurality of inwardly extending radial fins 100 disposed circumferentially thereabout in three groups that are spaced apart to receive between each group of fins 100 one leg of an insert 101 having a plurality of radial leg portions. The integral fins 100 and insert 101 are continuous and extend substantially the full length of the tubes 11 as shown in FIGS. 9 and 12, and the inserts 101 are press-fitted into the tubes 11 to provide good thermal conductivity between the outer ends of the insert legs and the inner wall surface of the tubes. Also, the opposite wall surfaces of the legs of the insert 101 are configurated to provide a plurality of longitudinally extending flat angular surface portions 102 thereby materially increasing the fluid contact surface area of the insert 101.

The invention is not limited to heat exchange condenser tubes 11 of the construction shown and described with reference to FIGS. 10 and 11 of the drawings, and other heat exchange tubes providing extended or increased internal fluid contact surface may be effectively employed. For example, a heat exchange tube 11*a* constructed as shown in FIGS. 15 and 16 may be employed comprising an outer circular tube 103 and a plurality of smaller inner circular tubes 104 arranged circumferentially therewithin and bonded to one another and to the outer tube 103 to provide maximum thermal conductivity therebetween.

Another embodiment of heat exchange tube is shown in FIGS. 17 and 18 of the drawings, wherein the tube 11*b* comprises a longitudinally corrugated continuous inner member 105 sandwiched between and bonded to longitudinally extending inner and outer flat plate members 106 and 107. When use is made of heat exchange tubes 11*b* of the construction shown in FIGS. 17 and 18, it is desirable that the tubes be disposed with the long axis of their cross-sectional configuration disposed substantially parallel to or in the direction of the flow streamline of the fluid passing between the fins 10, for example, as illustrated in FIG. 17 of the drawings, so as to provide minimum resistance to fluid flow between the fins.

As previously stated, during operation of the heat exchanger at partial loads, it has been found desirable to reduce fluid flow outwardly through the array of fins to match the reduced load requirement by directing the fluid outwardly through only the number of fins in the array required to maintain the condensing front in the heat exchange tubes 11 adjacent the outer ends of the tubes adjacent the fluid discharge sensor thermocouple 72 and provide optimum utilization of the heat exchanger at any given load. In the illustrated embodiment of the invention this is accomplished by inward and outward movement of the flow distributor 34 on shaft 36 in response to operation of the control unit 70. The specific components and circuitry of the control unit 70 form no part of the present invention and are fully within the knowledge and skill of the art relating to electrical control devices to accomplish the operation described.

For example, this may be accomplished by sensing the voltage generated by the thermocouples 71 and 72 representing the difference between the temperature of the heat exchange fluid entering the condenser and the temperature of said fluid after it has been discharged from the fins 10 at the point longitudinally along the condenser determined for the optimum location of the condensing front in the tubes 11. The voltage generated by the thermocouples 71 and 72 is fed into a feed back control system that operates, when the voltage is below a predetermined range, to energize the motor M' to cause outward traverse of the flow distributor 34 on shaft 36 and, at a voltage above said predetermined range, to drive the motor M' in the opposite direction and cause traverse of the flow distributor 34 in the opposite or inward direction on the shaft 36.

Thus, where the generated voltage is low and below the predetermined voltage range because the condensing front in the tubes 11 has not reached the optimum position relative to the sensor 72, the differential between the inlet and discharge temperatures of the heat exchange fluid is also substantially small, indicating the need for less condenser capacity, and the control unit 70 will operate to drive the motor M' and cause the flow distributor 34 to move outwardly on shaft 36 thereby reducing the number of fins being utilized and decreasing the effective condenser length. Similarly, where the generated voltage is high and above the predetermined voltage range because the condensing front in the tubes 11 has moved past the sensor thermocouple 72 and the discharge temperature of the heat exchange fluid is substantially higher than the inlet temperature thereof, indicating a need for increased condenser capacity, the control unit 70 operates to drive the motor M' in the opposite direction to cause inward traverse of the distributor 34 on the shaft 36 thereby increasing the number of fins 10 through which the heat exchange fluid is directed and discharged and the effective operating length of the condenser. Within the predetermined voltage range intermediate the two operating conditions just described, the condensing front in the heat exchange tubes 11 will be in the optimum position just at the sensor thermocouple 72 and so long as this condition pertains, the control unit 70 will not operate to traverse the distributor 34 in either direction.

A rotary heat exchanger made according to the present invention is characterized by its comparatively small, compact size and lightweight construction and the minimum power that is required to rotationally drive the heat exchanger at the designed speed. For example, in the case of an air-cooled condenser made in accordance with the present invention and having an array of fins 10.25 inches long consisting of 244 fins having an outer radius of 7.00 inches and driven at a speed of 2500 r.p.m., optimum results are obtained where the inner radius of fins is 5.25 inches and the spacing between the adjacent fins is 0.032 inches. The total number of fins 10 employed is directly proportional to the heat load of any given installation as determined by the axial position of the flow distributor 34 in response to operation of the control unit 70, as previously described.

Also, the use of viscosity shear forces to convey the heat exchange fluid between the array of fins 10 with the inherent absence of flow separation produces a very low operating noise level, free of cavitation such as frequently occurs when conventional lift forces are employed to accelerate a fluid. The heat exchange of the present invention is further characterized by a minimum resistance to heat flow between the exterior fin surfaces and the heat exchange fluid discharged outwardly therebetween as well as a minimum resistance to heat flow between a fluid in the heat exchange tubes and the interior wall surfaces of the tubes, and by the provision of novel means operable to substantially augment and increase the flow of fluid outwardly through the array of fins.

By reason of these characteristics and advantages, a rotary condenser embodying the present invention is particularly suited for use in high-performance closed

Rankine cycle power systems having a rotary boiler to which the condenser can be directly mounted or coupled coaxially for rotation with the boiler as a unit.

The user of a rotary heat exchanger embodying the present invention is not limited to use as a condenser, as previously described. The invention may be employed for heat exchange purposes generally, including other condenser applications as well as the cooling or heating of liquids, gases, and vapors where high-performance optimum total heat exchange is desired along with the accompanying advantages of compact lightweight construction, minimum power requirement and substantially noise-free operation. Rotary heat exchangers of the present invention can be used as condensers for closed Rankine cycle engines in vehicles for land use, such as the automobile, in which case air would be the preferred exterior fluid. Also, air-cooled engines using the present invention are useful in total energy systems wherein the heat rejected by the condenser in the form of hot air could be used to heat homes, shops, and other buildings, and the shaft energy can be simultaneously converted to electric power by means of a generator. In the case of a total energy system in which hot water is the preferred means of rejecting heat, the rotary heat exchanger of this invention can be optimized for silence, high efficiency, and lower power consumption and thus could be used for total energy on land and in boats. Other applications of the present invention include its use for assisting in the cooling of other types of engines, such as the internal combustion engines, for refrigeration cycle condensers and evaporators, and for chemical process cooling.

The cross-sectional shape and configuration of the heat exchange tubes, according to the present invention, is not limited to the tubes 11, 11*a* and 11*b* shown and described herein, and the particular tube shape and configuration will vary for optimum results in the particular usage or application of the heat exchanger just referred to. In some instances it may be desirable to provide the interior fluid contact surfaces of the heat exchange tubes with a coating of sintered or porous metal.

While certain embodiments of the present invention have been illusrated and described, it is not intended to limit the invention to such disclosures and changes and modifications may be made and incorporated as desired within the scope of the claims.

I claim:

1. A rotaty heat exchanger comprising,

an array of a plurality of annular fins each having predetermined inner and outer radii disposed coaxially in predetermined spaced parallel relation for rotation as a unit about their common axis and defining interiorly thereof a fluid chamber having a fluid inlet at least at one axial end thereof,

a plurality of heat exchange tubes each extending through said plurality of fins axially of the heat exchanger and disposed in radially spaced relation to the common axis circumferentially thereabout,

means for introducing and withdrawing a first fluid into and from said heat exchange tubes;

means for rotationally driving said plurality of fins and tubes at a predetermined speed of rotation about said common axis correlated to the axial spacing between the fins and the inner and outer radii thereof to convey and accelerate a second fluid from said fluid chamber substantially by viscosity shear forces spirally outward between the adjacent fins substantially to the velocity providing optimum total heat exchange between said second fluid and said first fluid in said heat exchange tubes,

and fluid flow augmentation means disposed outwardly of said array of fins and rotatable therewith operable to substantially increase the flow of the second fluid outwardly through the fin array and substantially reduce the resistance to heat flow between the exterior wall surfaces of the heat exchange tubes and said second fluid.

2. A rotary heat exchanger as claimed in claim 1 wherein the fluid flow augmentation means includes annular outwardly projecting flanges at opposite ends of the rotating fin array extending a predetermined distance outwardly beyond the fins.

3. A rotary heat exchanger as claimed in claim 2 wherein the annular projecting flanges are disposed radially parallel to the fins.

4. A rotary heat exchanger as claimed in claim 3 wherein the ratio of the diameter of the radial flanges to the outer diameter of the fins is within the range of about 1.2 to about 1.5.

5. A rotary heat exchanger as claimed in claim 1 wherein the fluid flow augmentation means includes a plurality of axially extending blades arranged in equally spaced relation circumferentially about the fins.

6. A rotary heat exchanger as claimed in claim 5 wherein the axial blades are pivotable between an inoperative netural position and an operative position radially of the fins.

7. A rotary heat exchanger as claimed in claim 6 wherein the axial blades normally are disposed in the neutral position and comprising mechanism operable to pivot the blades to their radial operative position under full load operating conditions of the heat exchanger.

8. A rotary heat exchanger as claimed in claim 2 wherein the fluid flow augmentation means also includes a plurality of axially extending blades mounted in equally spaced relation circumferentially about the fins.

9. A rotary heat exchanger as claimed in claim 8 wherein the axial blades normally are disposed in the neutral position and comprising mechanism operable to pivot the blades to their radial operative position under full load operating conditions of the heat exchanger.

10. A rotary heat exchanger as claimed in claim 1 comprising flow distributor means in the fluid chamber operable to direct the second fluid from said chamber outwardly through a predetermined number of the fins in the array.

11. A rotary heat exchanger as claimed in claim 1 comprising flow distributor means movable coaxially within the fluid chamber lengthwise of the array of fins to intermediate positions beteen an inner limit position in which the second fluid is directed thereby outwardly through all of the fins and an outer limit position adjacent the fluid inlet in which the second fluid is directed thereby outwardly through a predetermined minimum number of fins.

12. A rotary heat exchanger as claimed in claim 7 comprising flow distributor means movable coaxially within the fluid chamber lengthwise of the array of fins to intermediate positions between an inner limit position in which the second fluid is directed thereby out-

wardly through all of the fins and an outer limit position adjacent the fluid inlet in which the second fluid is directed thereby outwardly through a predetermined minimum number of the fins.

13. A rotary heat exchanger as claimed in claim 9 comprising flow distributor means movable coaxially within the fluid chamber lengthwise of the array of fins to intermediate positions between an inner limit position in which the second fluid is directed thereby outwardly through all of the fins and an outer limit position adjacent the fluid inlet in which the second fluid is directed thereby outwardly through a predetermined minimum number of the fins.

14. A rotary heat exchanger as claimed in claim 12 wherein the mechanism to pivot the blades is actuated by the flow distributor means upon movement thereof to the inner limit position.

15. A rotary heat exchanger as claimed in claim 13 wherein the mechanism to pivot the blades is actuated by the flow distributor means upon movement thereof to the inner limit position.

16. A rotary heat exchanger as claimed in claim 11, comprising,

power actuated drive means for moving the flow distributor in opposite directions coaxially lengthwise of the array of fins between said inner and outer limit positions,

and control means for actuating said drive means operable automatically in response to variations in the temperature difference of the second fluid before and after discharge through the fins to position the flow distributor to direct the second fluid outwardly through only the number of fins that provide optimum operation of the heat exchanger at the existing load.

17. A rotary heat exchanger as claimed in claim 11 wherein the flow distributor is slidably mounted for movement coaxially within the array of fins upon a shaft fixedly supported coaxially in the fluid chamber.

18. A rotary heat exchanger as claimed in claim 12 wherein the mechanism to pivot the blades comprises a radial push rod associated with each blade having geared connection therewith,

and cam means is carried by the flow distributor operable upon movement thereof to the inner limit position to engage and actuate said push rods radially outward thereby pivoting the blades from the neutral position to the radial position thereof.

19. A rotary heat exchanger as claimed in claim 12 comprising,

power actuated drive means for moving the flow distributor in opposite directions coaxially lengthwise of the array of fins between said inner and outer limit positions,

and control means for actuating said drive means operable automatically in response to variations in the temperature difference of the second fluid before and after discharge through the fins to position the flow distributor to direct the second fluid outwardly through only the number of fins that provide optimum operation of the heat exchanger at the existing load.

20. A rotary heat exchanger as claimed in claim 19 wherein the mechanism to pivot the blades comprises, a radial push rod associated with each blade having geared connection therewith,

and cam means is carried by the flow distributor operable upon movement thereof to the inner limit position to engage and actuate said push rods radially outward thereby pivoting the blades from the neutral position to the radial position thereof.

21. A rotary heat exchanger as claimed in claim 1, wherein the heat exchange tubes are each constructed to provide interiorly thereof a substantially extended and maximized heat exchange surface area for contact by the first fluid in said tubes to substantially minimize resistance to heat flow between said fluid and the internal surfaces of the tubes.

22. A rotary heat exchanger as claimed in claim 21, wherein each of the heat exchange tubes comprises a circular tubular member having therein a plurality of longitudinally extending circumferentially spaced inwardly projecting radial fins.

23. A rotary heat exchanger as claimed in claim 22, wherein each of the heat exchange tubes also comprises an insert extending longitudinally therein having leg portions equally spaced and extending radially outward between groups of the fins in the tubes with their outer ends in good thermal conductive engagement with the inner wall surface of the tube.

24. A rotary heat exchanger as claimed in claim 1, wherein the heat exchange tubes are disposed at a small acute angle to the rotation axis of the array of fins and are divergent with respect to said axis in the direction axially inward from the inlet to the fluid chamber defined by said fins.

25. A rotary heat exchanger as claimed in claim 2, comprising flow distributor means in the fluid chamber operable to direct the second fluid from said chamber outwardly through a predetermined number of the fins in the array.

26. A rotary heat exchanger as claimed in claim 2, comprising flow distributor means movable coaxially within the fluid chamber lengthwise of the array of fins to intermediate positions between an inner limit position in which the second fluid is directed thereby outwardly through all of the fins and an outer limit position adjacent the fluid inlet in which the second fluid is directed thereby outwardly through a predetermined minimum number of fins.

27. A rotary heat exchanger as claimed in claim 26, comprising,

power actuated drive means for moving the flow distributor in opposite directions coaxially lengthwise of the array of fins between said inner and outer limit positions,

and control means for actuating said drive means operable automatically in response to variations in the temperature difference of the second fluid before and after discharge through the fins to position the flow distributor to direct the second fluid outwardly through only the number of fins that provide optimum operation of the heat exchanger at the existing load.

28. A rotary heat exchanger as claimed in claim 10, wherein the fluid flow augmentation means includes annular radial flanges at opposite ends of the rotating fin array projecting a predetermined distance outwardly beyond the fins.

29. A rotary heat exchanger as claimed in claim 16, wherein the fluid flow augmentation means includes annular radial flanges at opposite ends of the rotating

fin array projecting a predetermined distance outwardly beyond the fins.

30. A rotary heat exchanger as claimed in claim 18, wherein the fluid flow augmentation means includes annular radial flanges at opposite ends of the rotating fin array projecting a predetermined distance outwardly beyond the fins.

31. A rotary heat exchanger as claimed in claim 29, wherein the heat exchange tubes are disposed at a small acute angle to the rotation axis of the array of fins and are divergent with respect to said axis in the direction axially inward from the inlet to the fluid chamber defined by said fins,

and said heat exchange tubes are each constructed to provide interiorly thereof a substantially extended and maximized heat change surface area for contact by the first fluid in said tubes to substantially minimize resistance to heat flow between said fluid and the internal surfaces of the tubes.

32. A rotary heat exchanger as claimed in claim 30, wherein the heat exchange tubes are disposed at a small acute angle to the rotation axis of the array of fins and are divergent with respect to said axis in the direction axially inward from the inlet to the fluid chamber defined by said fins,

and said heat exchange tubes are each constructed to provide interiorly thereof a substantially extended and maximized heat exchange surface area for contact by the first fluid in said tubes to substantially minimize resistance to heat flow between said fluid and the internal surfaces of the tubes.

33. A rotary heat exchanger as claimed in claim 1, wherein the Taylor number determined by the formula $N_{Ta} = d^2 w/v$ is in the range from about 5 to about 7.

34. A rotary heat exchanger as claimed in claim 1, wherein each of the heat exchange tubes comprises a longitudinally corrugated continuous inner member sandwiched between and bonded to longitudinally extending inner and outer flat plate members.

35. A rotary heat exchanger as claimed in claim 34, wherein the planes of the inner and outer flat plate member of each of said exchange tubes are disposed parallel to and in the direction of the flow streamline of the second fluid passing outwardly between the fins.

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