July 12, 1966 N. LAING 3,260,306

HEAT EXCHANGERS

Filed Jan. 2, 1963 2 Sheets-Sheet 1

Fig. 1.

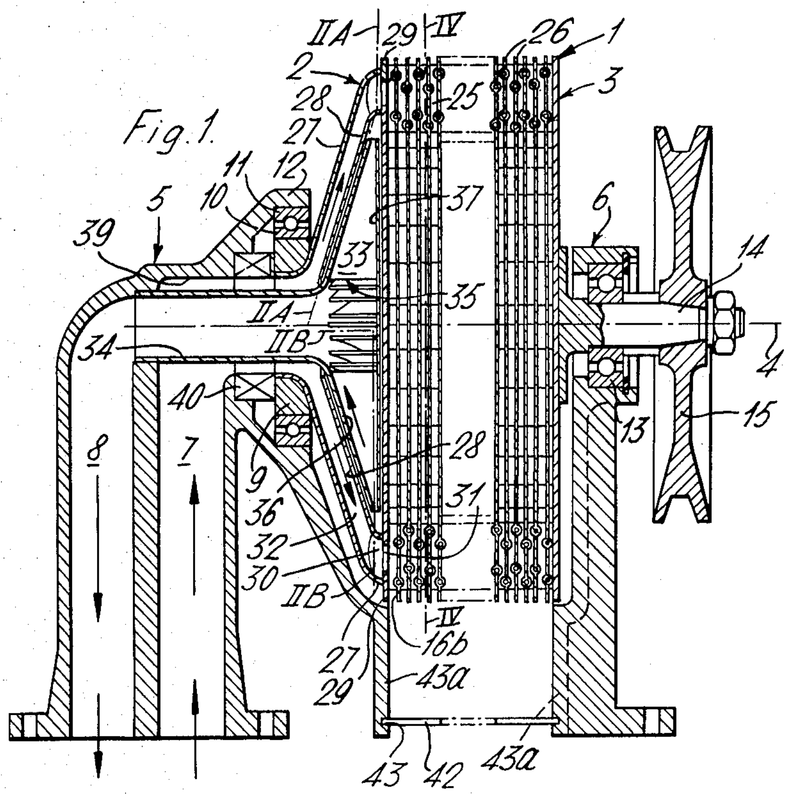


Fig. 2.

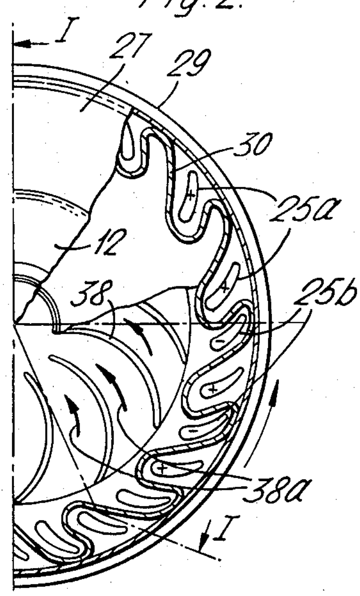
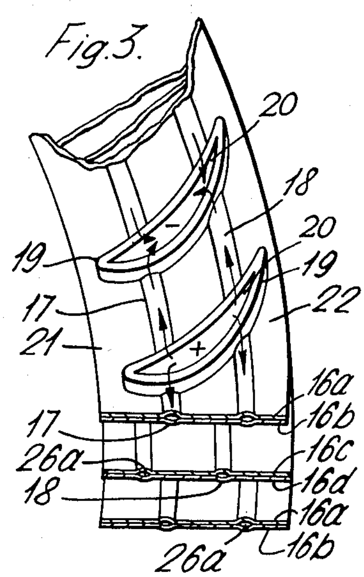


Fig. 3.



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HEAT EXCHANGERS

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Fig. 3a.

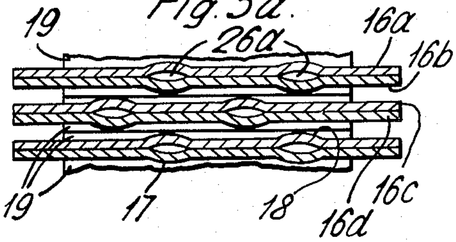


Fig. 4.

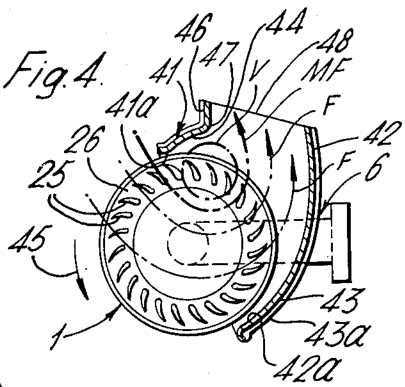


Fig. 6.

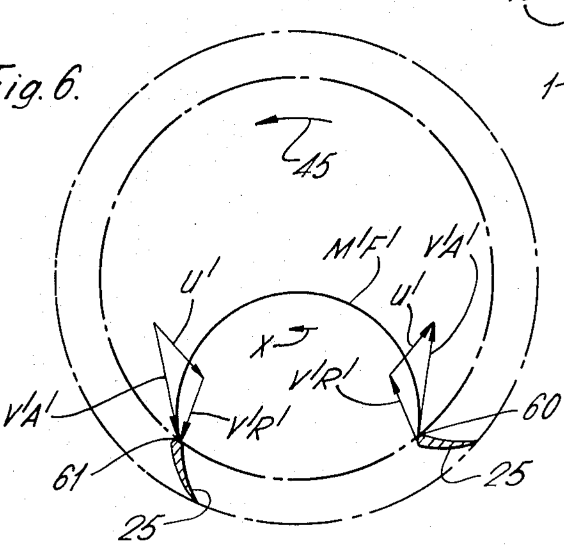


Fig. 7a.

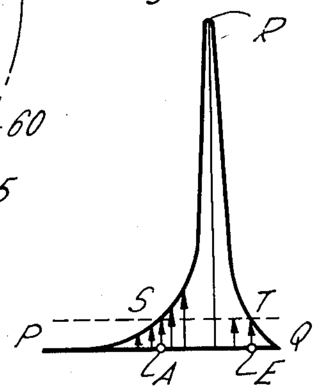
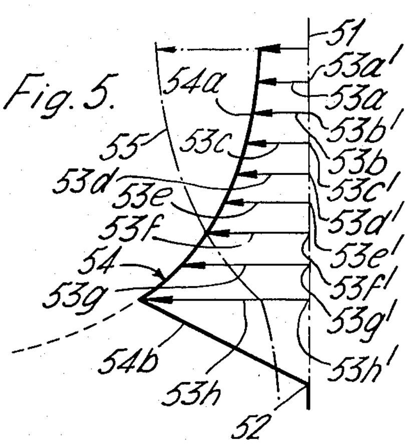


Fig. 5.



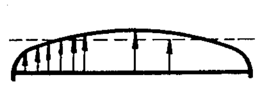


Fig. 7b.

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HEAT EXCHANGERS
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9 Claims. (Cl. 165—89)

This is a continuation-in-part of application Serial No. 701,600, filed December 9, 1957, now abandoned.

This invention relates to improvements in heat exchangers for exchange of heat between two fluids wherein the fluids are kept separate.

The apparatus according to the invention employs walls of heat-conducting material to keep the fluids apart, and heat is transferred between the fluids through this wall as they move relatively to it. One problem affecting all forms of apparatus where heat is transferred between a wall and a fluid passing over it is the thin more or less stationary layer of fluid which forms adjacent the wall, and which acts as an efficient heat insulator. Merely to increase the speed of relative movement between the main body of fluid and the wall with which it is in contact does not break up this insulating layer of stagnant fluid. The normal consequence of this layer is to make the apparatus more bulky than it would otherwise have to be, since the area of the heat conducting wall has to be extended to obtain a given performance.

The main object of the invention is to provide apparatus for the exchange of heat between two fluids which substantially reduces the stagnant fluid layer on one side of the wall separating the fluids, thus lending a reduction in the bulk of the apparatus for a given performance.

The invention depends on the appreciation that in a flow machine of the kind sometimes referred to as a "cross-flow" machine, which machine has a cylindrical rotor with blades arranged in a ring about and extending generally longitudinally of the rotor axis, and means to guide fluid flow from a suction side of the machine through the path of the rotating blades into the interior of the rotor and thence once again through the path of the rotating blades to a pressure side of the machine, the fluid in fact flows over each blade first in one direction (in its first passage through the path of the blades) and then in the other direction (in its second passage through the path of the blades), this reversal of flow occurring twice in every revolution of the rotor. The invention provides for a rotary heat exchange apparatus comprising a cross-flow machine constructed for flow of one fluid (e.g., air) through the rotor from the suction to the pressure side of the machine, and for the flow of the other fluid (e.g., hot water) through the blades of the machine, which are made hollow. The reversal of flow over the blades substantially reduces the stagnant fluid layer which would build up if the flow did not change in direction. For efficient heat exchange the invention provides further that the apparatus operates efficiently as a flow machine for the first fluid, since this efficiency helps to reduce the insulating fluid layer referred to. The blades are preferably concave facing in the direction of rotation and have their outer edges leading their inner edges. The blades and guide means are preferably such as to cooperate together to set up and stabilize a vortex of Rankine type interpenetrating the path of the rotating blades. For reasons which will appear below, this flow gives particularly good results at low Reynolds numbers and in this connection it should be appreciated that heat exchangers according to the invention will commonly be designed to work under conditions of low Reynolds numbers.

While efficient flow is desirable, other considerations may sometimes dictate a construction which is less than ideal considered simply as a flow machine. In particular it is sometimes desirable to obstruct flow in the passages

between the blades to improve heat transfer, as by annular radially extending fins in heat-conducting relation with the blades. The same reversal of flow takes place over the fins as over the blades, and minimizes the accumulation of stagnant fluid on the fins. Because of this dissipation of stagnant fluid, the fins can be set closer, and the apparatus made smaller, than would be practical if the flow over the fins were unidirectional.

A major advantage of the flow principles utilized in carrying out the present invention lies in the ease of control, and the fact that when throughput of the first fluid is reduced by throttling the power taken to drive the rotor is also reduced.

Embodiments of the invention will now be described by way of example with reference to the acompanying drawings in which:

FIGURE 1 is a longitudinal sectional view of one form of rotary heat exchanger according to the invention, the section line being indicated as I—I in FIGURE 2;

FIGURE 2 is a composite view illustrating one end member of a rotor forming part of the FIGURE 1 apparatus, the upper sector of this view showing a partial end elevation, and, the next sector showing a partial section of the member on the line IIA—IIA in FIGURE 1, and the remaining sector showing a further partial section of the member on the line IIB—IIB in FIGURE 1;

FIGURE 3 is a cut-away perspective showing on an enlarged scale six of a series of rings from which the rotor of the FIGURE 1 apparatus is made up;

FIGURE 3*a* is a partial longitudinal section on an enlarged scale of the FIGURE 1 rotor showing a few of the rings;

FIGURE 4 is a transverse section on the line IV—IV of the FIGURE 1 apparatus;

FIGURE 5 is a diagram illustrating velocity distribution in a Rankine vortex;

FIGURE 6 is a diagram showing blades of the FIGURE 1 rotor and the velocity triangles thereat of a maximum-velocity flow tube passing through the rotor; and

FIGURES 7*a* and 7*b* are velocity profiles.

Referring now to FIGURES 1 to 3*a*, the rotary heat exchanger there shown comprises a rotor designated generally **1** and having end members **2**, **3**. The rotor is mounted for rotation about an axis indicated at **4** upon stationary supports designated generally **5**, **6** one at either end of the rotor. The support **5** is formed with inlet and outlet passages **7**, **8** enabling liquid to be cooled to pass into and out of the rotor **1** through the end member **2**, which carries a ring **9** mounting the inner race **10** of a ball bearing the outer race **11** of which is secured within a bell-mouthed extension **12** of the support **5** directed towards the rotor. The other support **6** mounts a ball bearing **13** through which extends a stub-shaft **14** secured to the rotor end member **3** and carrying at its free end a pulley wheel **15** by which the rotor can be driven by a belt (not shown).

The rotor **1** is built up from a multiplicity of thin, generally flat, rings **16*a***, **16*b***, **16*c***, **16*d***, stamped from heat conducting material such as sheet copper, for example, of 0.50" thickness, which rings may be secured together by brazing between the end members **2**, **3**. The rings are of four kinds, each of the reference numerals **16*a***, **16*b***, **16*c***, **16*d*** denoting one kind of ring. All rings however have similar interior and exterior diameters including a pair of annular dishings **17**, **18** and upset flanges **19** about similar apertures **20** spaced regularly about the rings. The pairs of rings **16*a***, **16*b*** and **16*c***, **16*d*** are similar except that the dishings **17**, **18** of one pair are radially displaced relative to those of the other. The rings are of alternate left hand and right hand character and secured together so that inner and outer rim portions **21**, **22** of each pair are sealed together in abutting relationship with the flanges **19** directed outwardly, the flanges of one pair of rings interengaging with and being sealed to the flanges of the adjacent pairs. The flanges **19** can be alternately belled and coned to facilitate interengagement. A brazing composition can be applied to interengaging surfaces, the rings **16**a, **16**b, **16**c, **16**d stacked between the end members **2, 3** and brazed together as a unit in a furnace with all operations being performed by automatic machinery. In the completed rotor, the flanges **19** are aligned to provide in effect a series of lengthwise-extending hollow blades designated **25** as best seen in FIGURE 4. The pairs of rings **16**a, **16**b, and **16**c, **16**d form annular radially-extending fins **26** on these blades, each fin providing, by means of the dishings **17, 18**, a pair of annular liquid channels **26**a communicating with the interiors of the blades.

The rotor end member **2** comprises a pair of spaced frusto-conical sheet metal wall members **27, 28** telescoped one within the other and secured to a stout disc **29** which in turn is secured to the extreme ring **16**b. The outer wall member **27** has its larger end of circular configuration and secured adjacent the outer periphery of the disc **29** while its smaller end is secured within the mounting ring **9**. The inner frusto-conical wall member **28** has its inner end free and its outer end corrugated as indicated at **30** in FIGURE 2 and secured to the disc **29**. The corrugations **30** are such that alternate blades **25**a have their interiors communicating, through aligned holes **31** in the disc **29**, with the annular-section jacket space **32** between the wall members **27, 28** while intermediate blades **25**b have their interiors communicating through holes **31** with the frusto-conical interior space **33** within the inner member **28**. A fixed tube **34** projects axially from the end support **5** and carries within the space **33** a fixed guide assembly designated generally **35** to assist in conveying liquid from the blades **25**b to the tube **34**. This guide assembly **35** comprises an end portion **36** of the tube **34** belled to frusto-conical shape to lie closely within the inner frusto-conical member **28**, a disc **37** disposed radially in closely overlying relationship with the disc **29**, and a series of curved vanes **38** secured between the disc **37** and the tube end portion **36** and holding their peripheries spaced apart.

It will be seen that the tube **34** has its interior in communication with the outlet passage **8** and extends through the bell-mouthed extension **12** of the support **5** in spaced relation to the interior walls **39** thereof to allow for flow from the inlet passage **7** about the exterior of the tube **34** to the jacket space **32**.

The liquid circulation through the heat exchanger is as follows: liquid entering the jacket space **32** through inlet passage **7** and about the exterior of tube **34** is subjected to centrifugal action and gains kinetic energy before entering blades **25**a. The liquid passes along the blades **25**a, which form a "flow" system and are labelled "+" in the drawing, but at each fin **26** some of the liquid flows into the channels **26**a thereof and thence into the blades **25**b, which form the "return" system and are labelled "—" in the drawing. The liquid entering interior space **33** from blades **25**b encounters the vanes **38**, where the flow velocity decreases and pressure increases, the spaces **38**a acting as diffusers. The liquid finally issues through tube **34** and outlet passage **8**. Flow between the interior space **33** to the jacket space **32** is substantially prevented by the close spacing of tube end portion **36** and wall member **28**, while a rotating liquid seal shown diagrammatically at **40** and acting between the mounting ring **9** and the support **5** prevents leakage to the exterior, thus flow and return are both completely sealed to the exterior by a single seal, while minimal leakage between flow and return is regarded as permissible. Heat is thus transferred between the fluid passing through the hollow blades to the surrounding air through means of the walls of the blades and radially extending fins **26**. The transfer of heat from the blade and fin surfaces to the air surrounding the blades is increased over that normally obtained in conventional heat exchangers in that the relative direction of flow of air over the blade and fin surfaces is reversed upon a complete rotation of the rotor thus reducing the build up of any boundary layer of stagnant air where such a boundary layer would tend to serve as insulation and reduce transfer of heat.

The end supports **5, 6** are interconnected by a pair of sheet metal guide walls or guide means **41, 42** (FIG. 4) having end edges received in and located by grooves **43** in end walls **43**a integral with and forming extensions of the supports. At their lines **41**a, **42**a of nearest approach to the rotor **1**, which are diametrally opposite, the walls **41, 42** are well spaced therefrom, preferably by at least half the radial depth of the blades **25**. The wall **41** includes a curved portion **44** converging with the rotor **1** in the direction of intended rotation thereof indicated by the arrow **45** and a rear portion **46** merging with the portion **44** in a rounded nose or end **47**, the wall portions **44, 46** defining an angle of a little over 90°. The wall **42** diverges steadily from the rotor going from its line **42**a of nearest approach and defines with the wall portion **46** an outlet **48**.

It is to be noted that the blades **25** are profiled to airfoil shape with their larger edges directed inwardly. The inner and outer edges of the blades are located on coaxial cylindrical envelopes, the blades being concave facing in the direction of rotation indicated by arrow **45** and having their outer edges leading.

Quite apart from considerations of liquid flow through the interior of the blades **25**, the rotor **1**, end supports **5, 6** and interconnecting guide walls **41, 42** form a cross-flow blower, the operation of which will now be discussed, with particular reference to FIGURES 4 to 7b.

On rotation of the rotor **1** in the direction of arrow **45** a vortex of Rankine type is set up, the core region of which is eccentric to the rotor axis and indicated by the flow lines shown chain dotted at V (FIGURE 4); the whole throughput flows twice through the rotor blades **25** in a direction always perpendicular to the rotor axis as indicated in general direction only by the chain dotted flow lines F, MF.

FIGURE 5 shows the distribution of velocity in the vortex. The chain dotted line **51** represents a diameter of the rotor taken through the axis **52** of the vortex core V. Velocity of fluid at points on the line **51** by reason of the vortex is indicated by the horizontal lines **53**a, **53**b, etc., the length of each line **53**a, **53**b, etc., being a measure of the velocity at the point **53**a′, **53**b′, etc., respectively. The envelope of these lines is shown by the curve **54**, which has two portions, one **54**a approximately a rectangular hyperbola and the other, **54**b, a straight line. The curve **54**a relates to the field region of the vortex and the curve **54**b to the core region.

The core V of the vortex is a whirling mass of air with no translational movement as a whole, and velocity diminishes going from the periphery of the core to its axis **52**. Static pressure along the line **51** is shown by the curve **55**. It will be seen that the vortex core V is a region of low pressure. The location of the core region can be discovered by investigation of pressure distribution within the rotor.

It will be understood that the curves in FIGURE 5 are those of an ideal or "mathematical" Rankine vortex and actual flow conditions will be only approximate to those curves. Although for convenience the vortex core V has been shown circular in FIGURE 4 and has been regarded as possessing an axis, the core will usually not be truly circular.

The velocity profile of the air at the second entrance thereof to the rotor blades will be that of the vortex. In the ideal case of FIGURE 5 this profile will be that of the Rankine vortex there shown by curves 54a, 54b; in an actual case the profile will still have the general character of a Rankine vortex. Thus there will be in the region of the periphery of the core V a flow tube of high velocity indicated at MF in FIGURE 4 by the heavier chain dots while the flow tubes remote from the periphery of the core will have a very much smaller velocity. Because of the vortex there is at the exit from the rotor **1** a velocity profile such as shown in FIGURE 7*a*, where the line PQ represents the arc of exit from the rotor **1** and the ordinates represent velocity. The curve exhibits a pronounced maximum at R which is much higher than the average velocity represented by the dotted line (the average being obtained by dividing the total throughput per unit of time by the area represented by the exit arc).

It will be appreciated that much the greater amount of air flows in the flow tubes in the region of maximum velocity. It has been found that some 80% of the flow is concentrated in the portion of the output represented by the line ST corresponding to the points A and E on the line PQ, which is less than 30% of the total arc. A much smaller arc is in fact all that needs to be considered. A normal velocity profile for fluid flow in a defined passage is shown by way of contrast in FIGURE 7*b* and those skilled in the art will regard this as an approximately rectangular profile. This is the type of profile which, following the principles generally adhered to in the art hitherto, a designer of a flow machine would aim for in the outlet thereof.

The maximum velocity R as shown in FIGURE 7*a* appertains to a maximum-velocity flow tube indicated in FIGURE 4 by the heavier chain dots and designated MF. With a given construction the physical location of the flow tube MF is fairly closely defined. Therefore in the restricted zone of the rotor blades **25** through which this flow tube MF passes, the relative velocity between blades and fluid is much higher than it would be in a flow machine which, following the principles adhered to hitherto in the art, was designed for a rectangular velocity profile and uniform loading of the blades in the zones thereof where fluid passes.

The Reynolds numbers at a particular fluid flow condition represents the ratio of the product of flow velocity and a characteristic linear dimension of the part under observation to the kinematic viscosity of the fluid and for a given bladed rotor the Reynolds number (Re) can be defined as

$$Re=\frac{d.c.}{\gamma}$$

d being the blade depth in the radial direction, c the peripheral speed of the rotor, and γ the kinematic viscosity, the latter being equal to the quotient of dynamic viscosity and density. A Reynolds number is considered herein to be low if, as above defined, it is less than 5×10^4. It will be appreciated that the air passing the rotor **1** of FIGURES 1 to 4 will be heated by the heat developed in operation of the apparatus which increases the kinematic viscosity and reduces the Re number.

In general, the designer will be forced to operate the rotor under conditions of low Reynolds numbers despite the fact that it is well known that separation losses are much greater when flow takes place under those conditions.

One main advantage of the Rankine vortex flow will now be appreciated. As the velocity of the flow tube MF is several times greater than the average velocity (see FIGURE 7*a*), in the restricted blade zones through which the flow tube MF passes, there will be much less separation loss than if that tube flowed at the average velocity of throughput so that in the flow tube MF transfer of momentum to the air occurs under excellent conditions. The transfer of momentum in the flow tubes travelling below the average velocity will be poorer, but on balance there is a substantial gain because by far the greater proportion of the throughput is associated with the flow tube MF.

It follows that the throughput is greater than with a comparable rotor and guide means as heretofore known operating under the same conditions. This gives rise to improved heat exchange, partly by reason of the increased throughput but partly also because the high velocity flow tubes exert a more pronounced scouring action on the blades than a slower flow would do and thus more powerfully reduce the tendency to set up a stagnant layer of air on the blades.

FIGURE 6 shows the maximum velocity flow tube MF intersecting the inner envelope of the rotor blades **25** at **60**, **61**. It will be seen that ideally the maximum velocity flow tube MF undergoes a change of direction of about 180° in passing through the interior of the rotor, and that the major part of the throughput (represented by the flow tube MF) passes through the rotor blades where they have a component of velocity in a direction opposite to the main direction of flow within the rotor indicated in FIGURE 6 by the arrow X.

Vector diagrams are shown in FIGURE 6 for the velocities at the points **60** and **61**. In the diagrams U′ is the velocity of the inner edges of the blades at the points **60** and **61**, respectively, V′A′ the absolute velocity of the air in the flow tube MF at the point **60**, and V′R′ the velocity of that air relative to the blade as found by completing the triangle. The rotor blade angles can be designed for shock-free flow of the maximum flow tube MF since, as shown above, it flows through narrowly defined zones of the rotor.

It is considered that the blade angles and blade curvature determine the character of the vortex while the position of the vortex core is determined by means of the guide wall **41**.

It is considered that in a given case the particular blade angles and blade curvature, depend on the following parameters among others: the diameter of the blades, the depth of a blade in radial direction, the density and viscosity of the fluid, the dispositions of the external guide body, the rotational speed of the rotor, as well as on the ratio between overall pressure and back pressure. These parameters must be adapted to correspond to the operating conditions ruling in a given case. Blade curvature is in this connection to be understood to mean not only the curvature of a blade of uniform thickness, but also the curvatures of the contours of profiled blades. Whether or not the angles and curvatures have been fixed at optimum values is to be judged by the criterion that the flow tubes close to the vortex core should be deflected by approximately 180°.

While a preferred vortex flow pattern has been decribed, the invention comprises other flow patterns where the air is induced to flow twice through the path of the rotating blades of the rotor in a generally transverse direction. Again, while a preferred construction for the formation of a vortex has been described, other constructions are also contemplated.

I claim:

1. A heat exchanger to effect exchange of heat between a first fluid stream and a second fluid stream comprising a rotor including hollow blade portions and support means mounting the blade portions to extend generally longitudinally and in a ring about the rotor axis and providing inlet and outlet ducts intercommunicating with the interiors of said hollow blades; means mounting the rotor for rotation about an axis; non-rotating means providing inlet and outlet passages and cooperating with said support means for flow of said first fluid stream through the inlet passage to the inlet duct of the support means and thence through the hollow blade portions to the outlet duct of the support means and to the outlet passage; non-rotating guide means cooperating with the blade portions to induce on rotation of the rotor a flow of said second fluid from one side of the rotor through the path of the rotating blade portions to the interior of the rotor and thence once again through the path of the rotating blade portions to another side of the rotor.

2. A heat exchanger as claimed in claim 1, wherein the blade portions have inner and outer edges on coaxial envelopes, are generally concave facing in the direction of rotation, and have their outer edges leading their inner edges.

3. A heat exchanger as claimed in claim 1, wherein said blade portion support means provides said inlet and outlet ducts at one end of the rotor and said rotor mounting means includes a fixed support forming said inlet and outlet passage-providing means, said ducts and passages including communicating annular parts surrounding communicating axial parts, and a rotary seal for preventing escape of said first fluid to the exterior between the annular part of the duct and the annular part of the passage.

4. A heat exchanger as claimed in claim 3 having a fixed wall diverging from the axis and a wall forming part of the end member and closely overlying the fixed wall to provide a seal between said inlet duct and passage and said outlet duct and passage.

5. A heat exchanger as claimed in claim 1, wherein the support means provides said inlet and outlet ducts at one end of the rotor with said inlet duct communicating with the interiors of certain of said blade portions and said outlet duct communicating with the interiors of other blade portions, and wherein the blade portion support means provides an end member at the other end of the rotor having conduits providing intercommunication between the interiors of said blade portions to provide inlet and outlet passages therein.

6. A heat exchanger as claimed in claim 1, wherein said blade portion support means includes an end member having an outer annular element and an inner annular element, said elements defining an annular space between them and said inner element defining a central space within it, one end portion of said inner element being corrugated with a wave length equal to the circumferential spacing of the blade portions, said end member being secured with respect to the blade portions with the corrugations of the inner annular element oriented with respect to the blade portions whereby the interiors of alternate such portions communicate with said annular space and the interiors of other portions communicate with said central space, said inlet and outlet ducts including said annular and said central spaces.

7. A heat exchanger as claimed in claim 6, said central space forming part of said outlet duct and including stationary guide vanes mounted within said central space to guide said first fluid towards a fixed axial conduit supporting the guide vanes and forming part of the outlet passage.

8. A heat exchanger to effect exchange of heat between a first fluid stream and a second fluid stream comprising a rotor; means mounting the rotor for rotation about an axis and including a fixed support providing coaxial inlet and outlet passages for said first fluid; said rotor including hollow blade portions and support means therefor, said support means mounting the blade portions to extend generally longitudinally and in a ring about the axis and including a member at one end providing coaxial inlet and outlet ducts communicating respectively with the inlet and outlet passages in said fixed support with the inlet duct communicating with the interiors of alternate blade portions and the outlet duct communicating with the remaining blade portions to provide flow and return portions, the rotor further having channels interconnecting said flow and return portions, and said blade portions being concave facing the direction of intended rotation with their outer edges leading their inner edges; a rotary seal acting between the rotor end member and the fixed support to prevent escape of said first fluid from the outer ones of said ducts and passages; and non-rotating guide means cooperating with the blade portions to induce on rotation of the rotor a flow of said second fluid from one side of the rotor through the path of the rotating blade portions to the interior of the rotor and thence once again through the path of the rotating blade portions to another side of the rotor.

9. A heat exchanger comprising a generally cylindrical rotatable hollow member, a hollow drive shaft axially supporting and secured to said member, a plurality of blades mounted peripherally on the drive shaft and extending parallel to the axis thereof, said blades defining between themselves channels providing communication between interior and exterior portions of said rotatable member, said blades having passages formed therein for passage of heated fluid therethrough, said passages extending parallel to the axis of said shaft, said passages being arranged in pairs with the passages in each pair being connected together at one end thereof so that fluid flowing therethrough passes first in one direction and then in an opposite direction, said member being closed at opposite ends thereof, one end of said member having inlet openings for admission of said fluid into said passages and having outlet openings for emergence of said fluid from said passages, the channels formed between the blades extending in such manner that cooling air enters the channels radially on one side of the rotating member over its entire length inside the member and is diverted to emerge radially over the entire length of said member, the air being struck twice by the blades while absorbing therefrom heat transferred from the heated fluid in said passages, and a housing surrounding said member and providing an inlet for the cooling air, said housing extending parallel to the axis of rotation of said member over the entire length thereof, said housing further providing an outlet for the cooling air extending parallel to the axis of rotation of said member; said housing including a guide body having a rounded end past which the cooling air moves upon entrance between the blades, and a guide wall forming a convergent passage at the outer periphery of said member to guide the cooling air after emergence from said blades.

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