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Faris et al.

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[45] **Sept. 17, 1974**

[54] **ROTATABLE HEAT EXCHANGER**

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[52] **U.S. Cl.** **165/86,** 62/426, 62/499

[51] **Int. Cl.** **F28d 11/00,** F28g 5/00

[58] **Field of Search** 165/86, 87, 122, 499; 415/54, 56, 52, 53, 89, 120; 62/325, 499

[56] **References Cited**

UNITED STATES PATENTS

1,829,753	11/1931	MacLean	165/87
3,347,059	10/1967	Laing	62/325
3,477,499	11/1969	Goetz	62/499

FOREIGN PATENTS OR APPLICATIONS

888,048	8/1953	Germany	415/89

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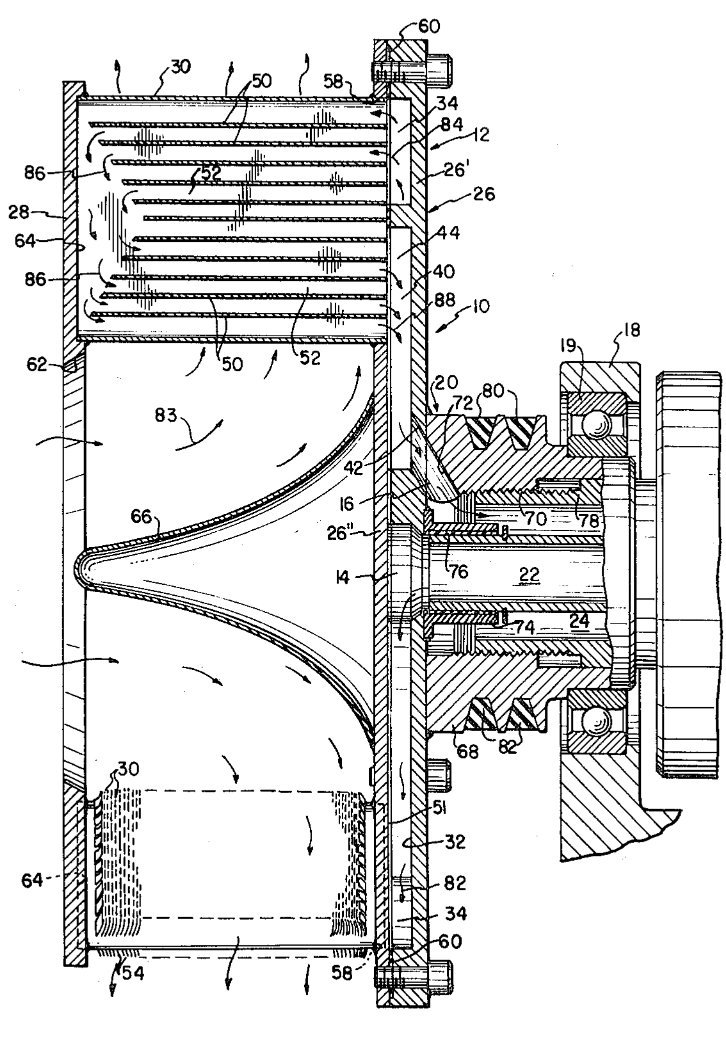
[57] **ABSTRACT**

A rotary heat exchanger including a plurality of hollow blades fixed to the periphery of a rotatably mounted member oriented to serve as means for providing forced circulation of ambient air in a radial direction upon rotation. Passages are formed in the rotatable member which receive the preheated or precooled thermal fluid from external equipment and distribute such fluid to the hollow blades within which the fluid circulates causing heat transfer to ambient fluid. The thermal fluid is then directed to collection passages in the rotatable member from which the fluid exits and returns to the external equipment. In one embodiment, the distribution and collection passages are formed to obtain a reduced pressure drop and a compact device. In another embodiment, a stator is provided enabling the heat exchanger to function as a pump for circulating the thermal fluid.

7 Claims, 3 Drawing Figures



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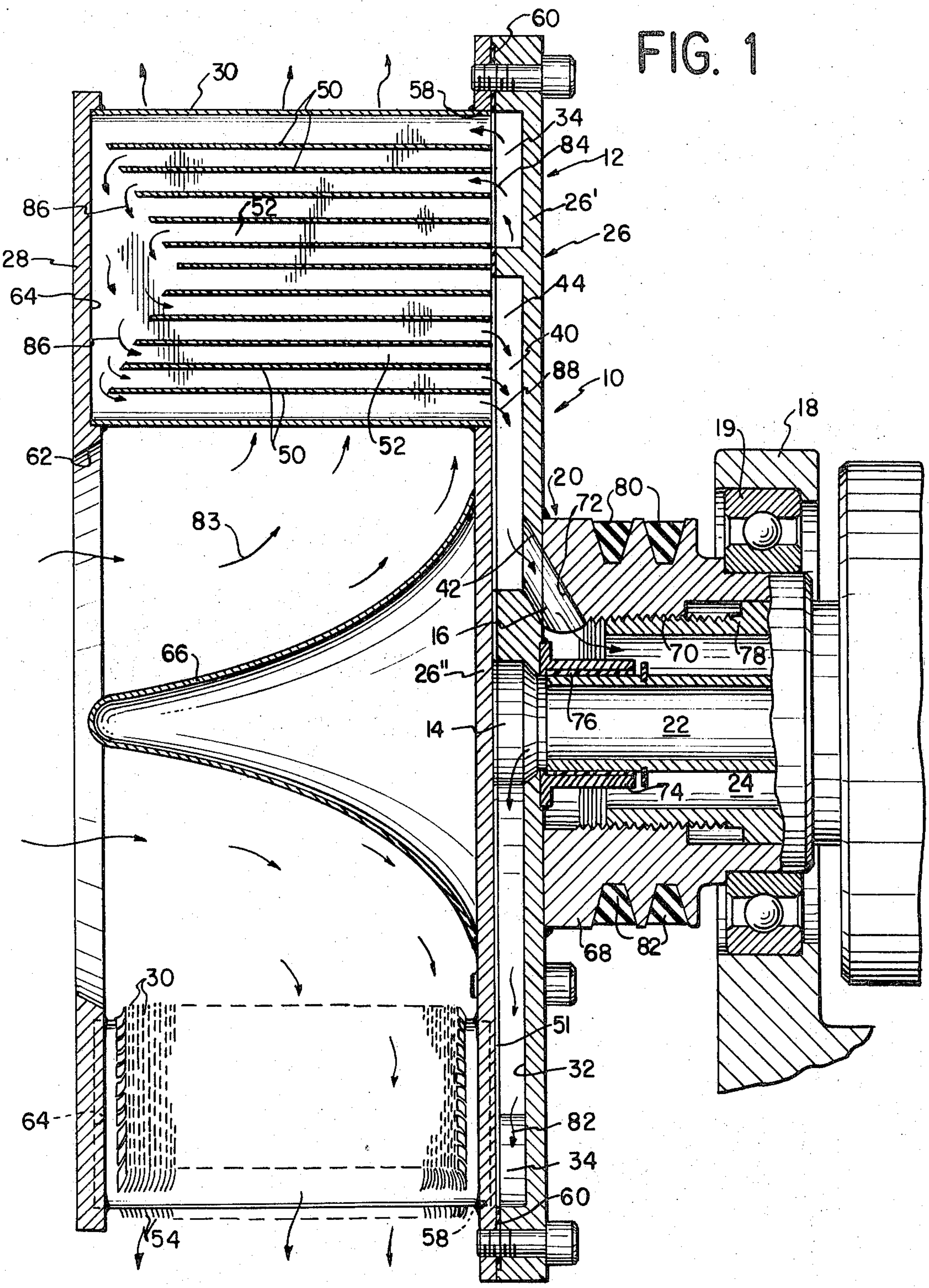

FIG. 1

FIG. 2

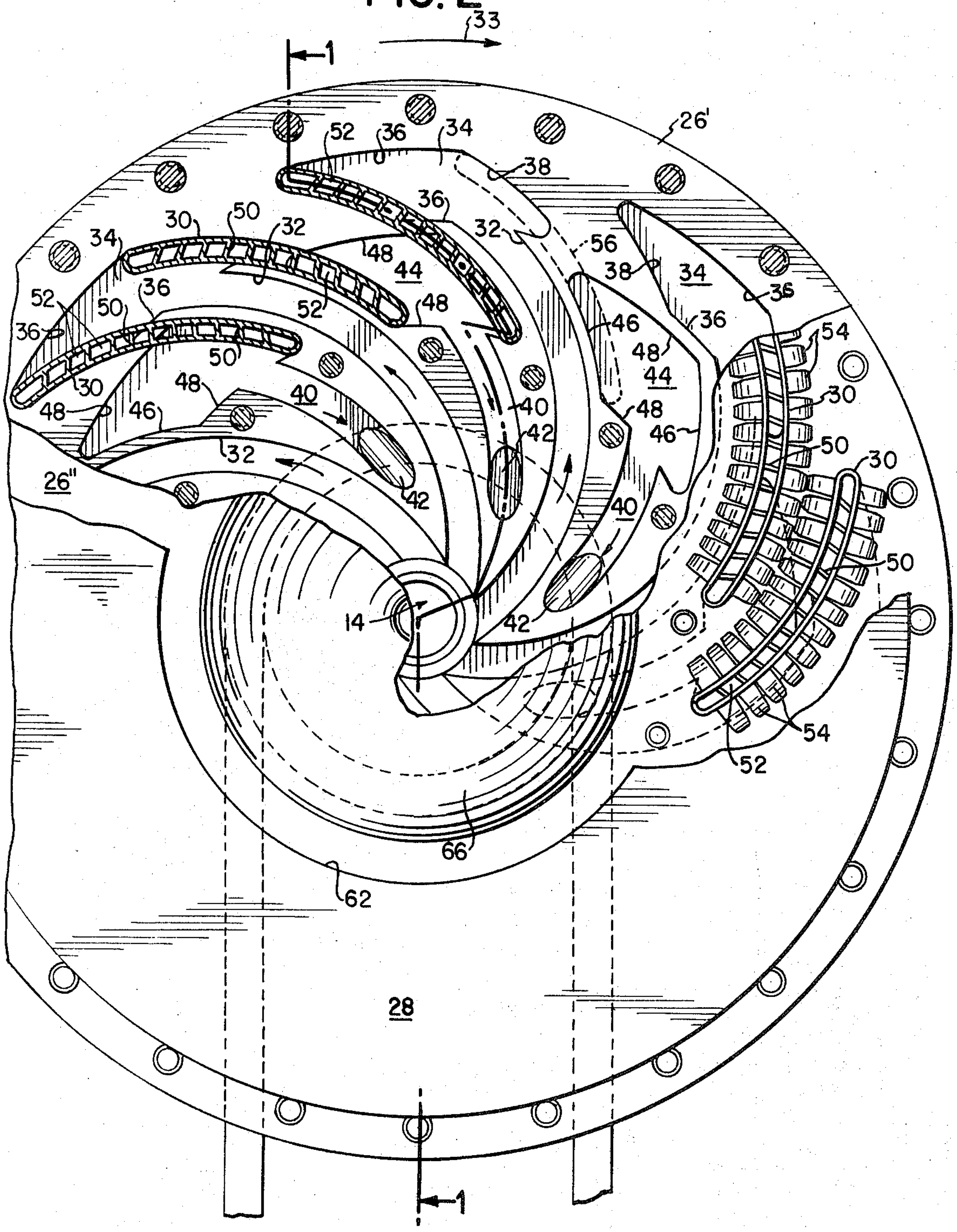
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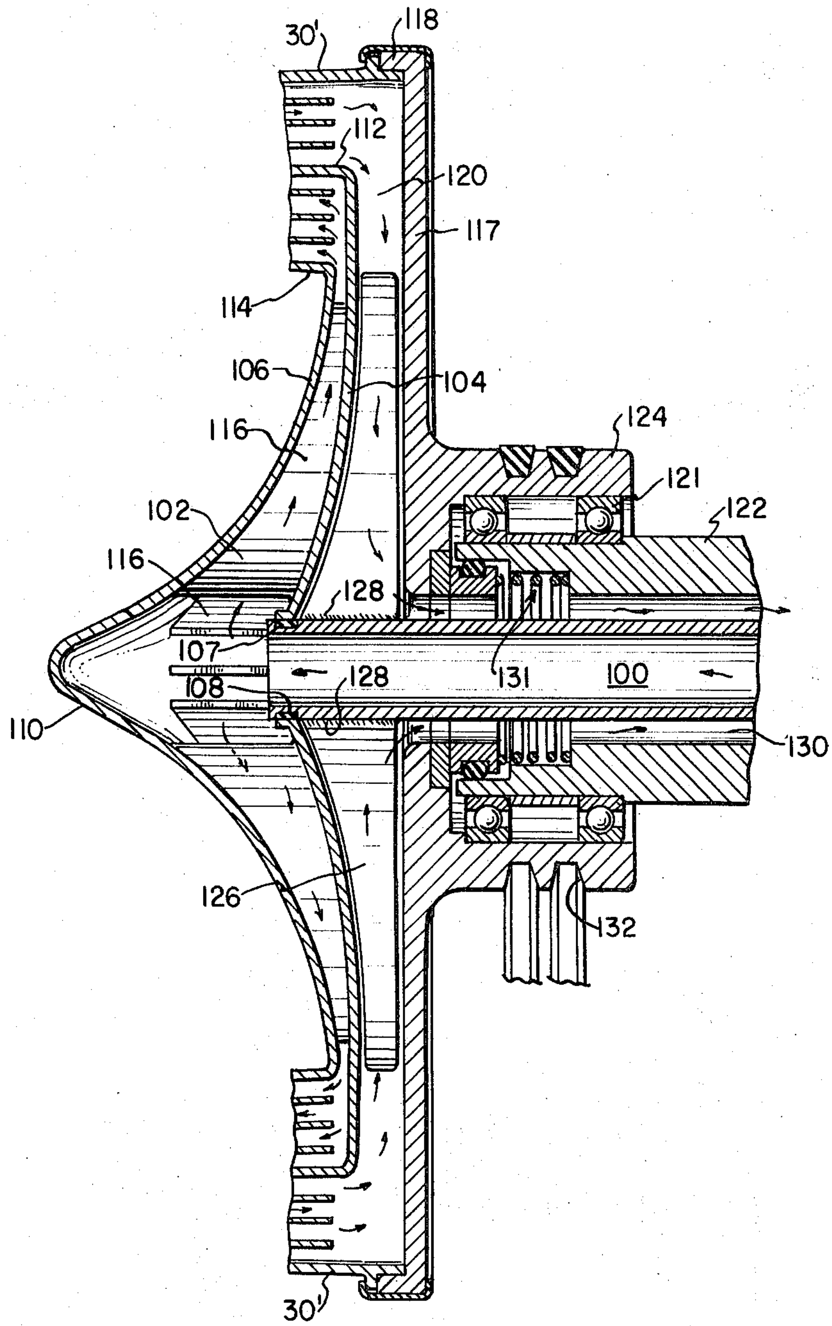
FIG. 3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ROTATABLE HEAT EXCHANGER

BACKGROUND OF THE INVENTION

This invention relates to heat exchangers and, more particularly, to heat exchangers of the rotary type.

Rotary heat exchangers have advantages due to their relatively high heat transfer efficiency and relatively compact design, in addition to the substantial decrease in the noise output of these devices relative to the conventional stationary heat exchangers equipped with air circulating fans. Further, the power requirements to operate such rotary heat exchangers are substantially smaller than the power required to run conventional heat exchangers.

By way of example, the use of the conventional rotating fan-stationary radiator combination in automobiles has been handicapped by a variety of factors, e.g., noise becomes intolerable at air velocities needed for high efficiency operation; efficiency is reduced due to the requirement of clearance space for the fans resulting in the impracticality of optimum air flow over the entire heat exchanger surface; power in amounts needed to drive the fan often detracts from performance; and the necessity of placement of the radiators in the airstream imposes a severe design limitation under the automobile hood. The low noise, high efficiency and compact design characteristics of rotary heat exchangers provide a feasible solution to these problems.

A variety of designs of rotary heat exchangers have been proposed. Essentially, most rotary heat exchangers include a rotor formed by a plurality of hollow blades annularly mounted on side plates. As the rotor rotates, ambient air is caused to pass over the outer surface of the blades while thermal fluid (refrigerant or heated fluid) is directed to and passes through the blade interiors thereby effecting heat transfer between the thermal and ambient fluids.

Although rotary heat exchangers provide the advantages set forth hereinabove, in the past several problems have decreased their effectiveness. For example, relatively substantial energy losses occur during the flow of the thermal fluid through the exchanger passages. These flow losses not only result in increased pressure drop within the apparatus but also increase the power requirements to circulate the fluid through the system. Secondly, in prior heat exchangers, the designs of both the interiors and exteriors of the thermal transfer blades have not optimized their heat transfer properties. Thirdly, the relatively complex fluid circulation systems of rotary heat exchangers often have required the use of bulky fluid transport apparatus which have naturally reduced the exchangers' compactness. Additionally, a rotary heat exchanger incorporating improvements of the nature described above is not available wherein the exchanger has the capacity to pump the thermal fluid through the circulation system, thus dispensing with the need for an external pump.

SUMMARY OF THE INVENTION

Accordingly, an object of this invention is to provide a new and improved rotary heat exchanger.

Another object of the invention is the provision of a new and improved fluid distribution and collection element for use in rotary heat exchangers which will reduce pressure drop and energy requirements.

A further object is to provide more efficient thermal transfer blades for use in rotary heat exchangers.

A still further object is to provide a new and improved compact rotary heat exchanger.

Yet another object is to provide a heat exchanger satisfying all of the above objects which also provides pumping action for the thermal fluid circulating therein.

Briefly, in accordance with the preferred embodiments of this invention, these and other objects are attained by providing an end plate in a rotary heat exchanger having passages which transmit the thermal fluid from external equipment to the blade interiors having a compact configuration wherein the energy necessary to move the fluid from the entrance of the exchanger to the blades is minimal relative to prior art apparatus. The configuration of the passages results in the force due to the rotation of the end plate having a minimum effect on the flow of the thermal fluid. Additionally, in order to improve the thermal transfer characteristics of the heat exchanger, the thermal transfer blades are formed with a plurality of parallel fluid circulation passages extending longitudinally within each blade and with a maximum number of integral heat transfer fins. To obtain pumping action, an embodiment is disclosed wherein a stator is provided in the fluid collection area.

DESCRIPTION OF THE DRAWINGS

A more complete appreciation of the invention and many of the attendant advantages thereof will be readily appreciated by reference to the following detailed description when considered in connection with the accompanying drawings, in which —

FIG. 1 is a side view partly in section of the rotary heat exchanger of the present invention;

FIG. 2 is a front view of the rotary heat exchanger partially broken away; and

FIG. 3 is a side view in cross section of another embodiment of a rotary heat exchanger.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings where like reference characters designate identical or corresponding parts throughout the several views, and more particularly to FIG. **1**, the rotary heat exchanger, generally denoted as **10**, includes a rotor assembly **12** having a fluid inlet passage **14** and a plurality of outlet openings **16** for transporting thermal fluid to and from the rotor assembly respectively from and to external equipment. By way of example, the fluid inlet may be coupled to the water line of an automobile engine to receive the coolant in its heated state while the outlets **16** return the coolant subsequent to the dissipation of its heat to the automobile engine. The heat exchanger is rotatably mounted on a suitable fixed support **18** by a bearing **19**. Completing the heat exchanger assembly, a hub assembly **20** is rigidly connected to the rotor and includes a fixed tubular passage **22** which communicates with the central fluid inlet **14** and an annular passage **24** which fluidly communicates with outlets **16** as will be described in greater detail hereinbelow.

For the sake of convenience, the system will be described as a thermal transfer unit for cooling the thermal fluid rather than as one which heats the fluid. However, it is understood that the heat exchanger may be used equally as well to transfer heat to a precooled thermal fluid which circulates through the exchanger.

The rotor assembly 12 includes a manifold plate 26 having an end of each of a series of thermal transfer blades 30 fixed thereto, such as by welding, soldering, brazing or any other suitable technique, in an annular configuration. An end plate 28 is provided at the other end of each of the blades having a central opening 62 which serves to direct ambient air drawn into the rotor. In some circumstances, such an end plate is not necessary, e.g., when the blades are relatively short and the rotor is operated at low r.p.m. In such cases, the blades may be cantilevered from the manifold plate.

The manifold plate 26 as best seen in FIG. 2 preferably comprises a channel plate 26′ having a central circular chamber 29 formed therethrough which comprises the central fluid inlet 14. Emanating from the central fluid inlet 14 are a plurality of distributor channels 32 which extend outwardly towards the periphery of the manifold plate assembly. As seen in FIG. 2, the distributor channels 32 are arcuate in shape, being curved rearwardly relative to the direction of rotation of the rotor indicated by arrow 33. In the preferred embodiment, the channels form segments of a circle. However, it is understood that arcs of other geometric configuration may be utilized within the scope of the invention. Circular segment arcs have been found particularly suitable due to the ease of machining the channels in their circular form. The distributor channels extend outwardly from the central fluid inlet 14 forming a curved spoke or "pinwheel" type pattern. The outer ends of the distributor channels terminate in enlarged pockets 34 having approximate trapezoidal shapes defined by two substantially concentric curved ends 36 having their centers at the channel plate center and opposed sides 38 having arcuate shapes which are oriented towards each other as they progress away from the periphery of the plate.

A plurality of collector channels 40 are formed in channel plate 26′ alternately spaced between adjacent pairs of distributor channels 32. Collector channels 40 assume a rearwardly curved shape similar to that of distributor channels 32. The inner ends of the collector channels 40 terminate in respective bores 42 opening onto the right side of channel plate 26′ as seen in FIG. 1. These openings are spaced around an inner annular zone of the channel plate 26′, each opening defining a thermal fluid outlet 16 through which the thermal fluid exits from the rotor assembly 12 into outlet passage 24 in the hub assembly 20. The outer ends of collector channels 40 are formed into enlarged pockets 44 having trapezoidal configurations similar in shape and size to distributor channel end pockets 34. As seen in FIG. 2, pockets 44 are formed inwardly of pockets 34 and include inwardly directed sides 46 and concentric parallel sides 48. Pockets 44, as will be seen, act as outlets from the thermal transfer blades and direct the thermal fluid from the blades to the outlets 16.

The manifold plate assembly 26 is completed by positioning a template plate 26″ over the channel plate 26′ and fastening it thereto by conventional means such as by bolts 42. The template plate 26″ has a plurality of openings 58 formed therethrough, each opening having a shape of a blade cross section so that a blade end may be received therein. These openings 58 are formed so that each aligns with an associated pair of end pockets 34, 44 in a manner described in detail below. The inner surface 51 of template plate 26″ overlies all the distributor and collector passages (except where openings 58 are present) so that the channels are closed defining enclosed passages in the manifold plate assembly 26. A substantially circular gasket 60 is provided between the channel and template plates to prevent leakage of thermal fluid from the passages during operation. A series of openings are formed in the gasket which align over the distributor and collector passages to prevent leakage therefrom in a conventional manner.

The structure of the thermal transfer blades 30 and their cooperation with the distributor and collector passages will now be explained in greater detail. Blades 30 preferably have a curved cross section as best seen in FIG. 2 and have a generally hollow interior within which the thermal fluid is received from the manifold plate assembly. As shown in the preferred embodiment, the interior of each blade 30 is formed having a plurality of partitioning walls 50 which extend longitudinally along the length of each blade defining a plurality of circulation passages 52 which conduct the thermal fluid through the blade in a manner to be described. The end of each wall 50 at the blade end remote from the manifold plate assembly 26 is recessed within the blade, the extent of each recess increasing incrementally towards the center of the blade as seen in FIG. 1. In this manner, the thermal fluid may exit from the left-hand end of those circulation passages communicating with distributor passages 32 and enter the left-hand ends of those circulation passages communicating with collector passages 40 as shown by arrows 86. The plurality of reduced size circulation passages provides a double advantage over conventional blades which usually have two adjacent passages. Firstly, the plurality of passages increases the heat transfer surface area by providing additional surfaces (those of walls 50) which conduct heat to the blade surfaces and, secondly, increases the structural integrity of the blade which is advantageous due to the relatively high pressure of the thermal fluid which circulates through the blade. Of course, any number of partitions and cells may be used so long as at least one passage is provided for the fluid to move away from the manifold plate and one is provided for return.

A plurality of heat transfer fins 54 are provided on each blade 30 in order to further improve the heat transfer characteristics of the system. In construction, the blades are preferably extruded with the walls 50. Subsequently, the outer surfaces of the blades are preferably skived to form a variable number of fins per unit length. The skiving technique has been found to produce finned blades having extremely good heat transfer characteristics because of the fins being an integral part of the tubes thus eliminating contact resistance as well as promoting greater turbulence within the airstream. Finally, the ends of walls 50 are trimmed as shown in FIG. 1 to allow for the thermal fluid flow described above.

Referring to FIG. 2, it is seen that the inwardly directed sides 38 of distributor passage terminal pockets 34 overlap the corresponding sides 46 of the collector passage terminal pockets 44 in the direction defined by the side edges 38, 46 respectively of the pockets. More specifically, referring to the direction of rotation of the manifold plate indicated by arrow 33 in FIG. 2, the leading edge 38 of each distributor passage terminal pocket overlies the trailing edge 46 of the associated collector passage terminal pocket in front of it whereas the trailing edge 38 of the same distributor passage terminal pocket overlies the leading edge 46 of another collector passage terminal pocket to its rear. Each pair of sides 38, 46 define the locus of a curved path, such for example as path 56 shown in dotted lines in FIG. 2. The curved configuration of the blade cross section is adapted to identically match with the curved configuration of the above-described locus of points. Hence, the configuration of blade 30 as seen in FIG. 2 is identical with the dotted path 56. In a similar manner, the openings 58 formed in the template plate 26'' have an identical configuration as that defined by dotted path 56. Each opening cooperates with a respective pair of pockets by overlying the curved path as shown.

By inserting the end of each blade 30 into a template plate opening 58 where it is subsequently bonded into place, such as by welding, a number of the circulation passages are positioned to communicate with a distributor passage pocket while others communicate with the collector passage pocket. Thus, the template plate serves to both position the ends of the blades over the respective distributor and collector passage terminal pockets and also support the blade ends for rotation with the manifold plate assembly 26.

The other ends of the thermal transfer blades 30 are supported by a cover plate 28 having a circular opening 62 formed centrally therethrough. The blade ends are received within recessed grooves 64 and are held in place by any conventional means, such as by welding or brazing. A cone 66 may be provided having its base attached to the template plate so that its tapered end extends to the vicinity of the plane of the opening 62. The cone serves to improve the flow of ambient air through the device as is explained in the description of the operation of the device.

The hub assembly 20 serves to transmit the thermal fluid to and from the rotor assembly from and to external equipment respectively. In the present embodiment, the hub assembly includes a pulley member 68 rigidly attached, as by welding, to the outer surface of the manifold plate assembly 26. Pulley member 68 is formed with a partially threaded stepped bore 70 formed centrally therethrough. A plurality of angularly oriented passages 72 (only one of which is shown in FIG. 1) are formed extending from that end of pulley member 68 which abuts the manifold plate assembly into bore 70. These passages 72 are adapted to align with the thermal fluid outlets 16 formed in the channel plate 26'. A collar 74 is also attached to the manifold plate assembly disposed within bore 70 and aligned with the central fluid inlet 14. A lubricated seal 76, made of a suitable material such as carbon-filled Teflon, lines the inner surface of collar 74 and forms a sleeve which receives fluid inlet tube 22 which is adapted to remain stationary. The seal 76 therefore acts as both a dynamic fluid seal and as a bearing.

A tubular fluid return pipe 78 having an externally threaded end is fastened to the interiorly threaded pulley 68 and defines, in cooperation with the outer surface of passage 22, annular fluid return passage 24. The pipe 78 is rotatably mounted in a conventional sealed fluid rotational connection, such as Model No. 755 "All-Purpose Union" manufactured by the Deublin Co. A pair of belts 80 are provided within grooves 82 formed on pulley member 68 for driving the hub assembly and attached rotor assembly. It should also be noted that the rotor assembly may be used for driving other equipment, such as an automobile generator, either through a belt drive or by direct coupling.

In operation, the rotor assembly is rotated by belts 80 from conventional driving means, such as an automobile engine. The blades 30 act as centrifugal fan blades thereby drawing ambient air through opening 62 and over the blade surfaces as depicted by arrows 83. Cone 66 insures that the air drawn into the center of the rotor assembly is substantially evenly distributed over the length of the blades by deflecting the air flow in a radial direction along the axis of the rotor. As the rotor assembly rotates, the thermal fluid enters the manifold plate assembly through passage 22 under pressure supplied by external equipment, such as an automobile water pump. In the present illustration, the thermal fluid is described as being preheated by the external equipment (such as an automobile engine). However, as mentioned above, the invention is equally applicable to cases where the thermal fluid has been precooled by external equipment. As the thermal fluid enters central fluid inlet 14, it is moved substantially radially towards the periphery of the manifold plate assembly in a relatively orderly manner through the distributor passages 32 partially under the action of centrifugal force. Such fluid flow is denoted by arrows 82 in FIG. 1. Upon reaching the terminal pockets 34 of distributor passages 32, the fluid is at a considerably higher pressure than at the inlet due to the rotation of the device. For example, for a one-foot diameter rotor having an angular velocity of 2,000 r.p.m., the fluid will be at a pressure of approximately 70 p.s.i. more at the blade entrance than at the inlet of the manifold plate assembly.

As best seen in FIG. 1, the thermal fluid enters the circulation passages 52 of the thermal tubes 30 which cooperate with pockets 34 as illustrated by arrows 84. The reduced cross sections of the tube circulation passages 52 minimize the effect of the higher pressure and increase the heat transfer to the blade surfaces as described above. The fluid circulates the length of each of the tubes 30 through the inlet passages 52 and is deflected by the inner surface of the cover plate 28 closing the blade end which causes the fluid to enter the return circulation passages 52 of the fins as indicated by arrows 86. The fluid completes the circuit exiting through the collector passage terminal pockets 44 as indicated by arrows 88. The thermal fluid exits from rotor assembly 12 via the thermal fluid outlets 16 which communicate with the tubular fluid return passage 24.

The curved spoke design for the inlet and return passages of the manifold plate assembly serves to reduce the power requirements necessary to circulate the thermal fluid through the rotor since the flow losses due to turbulence and other effects are reduced as is the rate of pressure increase of the fluid as it travels outwardly. Since the thermal fluid outlets 16 are annularly spaced around the central fluid inlet 14, the thermal fluid exiting from the rotor assembly is subject to an increased pressure head relative to that entering, thereby facilitating circulation. The design of the blade circulation passages optimizes both structural integrity and heat transfer.

In the present embodiment, the design of the manifold plate assembly wherein the inlet and outlet passages are in substantially the same plane serves to increase the compact character of the heat exchanger. In the past, the fluid inlet and outlet passages have been in different planes which not only made the fluid transfer apparatus more complicated and cumbersome, but also increased the bulkiness of the exchanger. The present invention maximizes the compact character of the device while minimizing the structural complexity of the design. Of course, the manifold plate structure may be different from that specifically described. For example, the manifold plate may be made in a unitary manner rather than having a multiple plate configuration. Still another advantage derived from the manifold plate construction is the large choice of fabrication techniques available. For example, the manifold plate may be formed by casting, machining (such as by milling), chemical etching, etc.

With respect to the orientation of the blades of the rotor, as seen in FIG. **2,** the blades are mounted in a rearward direction, i.e. sloping downwardly and inwardly in the direction of rotation of the rotor assembly. Such orientation provides a relatively high velocity of air over the surface of the blades relative to the surface of the blades while maintaining the power requirements at a relatively low value due to the decreased absolute velocity of the air relative to the stationary surroundings. Of course, the blades may be oriented otherwise (e.g. radially) to take advantage of other resulting characteristics, i.e. strength, air velocity, etc.

There are circumstances where a cover plate such as plate **28** is not required for efficient operation of the heat exchanger. For example, where the length of the heat transfer blades is reduced and where the speed of rotation of the rotor assembly is low, the blades may be cantilevered on the manifold plate without the need for a supporting cover plate. In such cases, of course, each blade must have an end cover to direct the flow of the thermal fluid from the inlet circulation passage to the outlet circulation passage.

The embodiment of the heat exchanger described above, although having a design wherein the pressure drop is reduced relative to prior art devices, does not act as a pump. By definition, pumping action results in an increase in pressure from the fluid inlet to outlet thereby allowing the fluid to circulate without an external pump which is usually necessary in heating and cooling systems of this type. Generally, by providing an annular chamber having stationary vanes for the thermal fluid to enter as it exits from the blade interiors, the high velocity and direction of the fluid is transformed with minimum shock, with little loss of momentum. The flow velocity will decrease with consequent increase in pressure thereby resulting in pumping action. This action is similar to that disclosed in U.S. Pat. No. 3,424,234 to N. Laing, granted Jan. 28, 1969.

Referring to FIG. **3,** a typical structural arrangement according to the present invention is illustrated, wherein the identical blade structure to that shown in FIGS. **1** and **2** is employed. An inlet tube **100** serves to direct incoming thermal fluids to a distribution chamber **102** defined between a pair of opposed dish-shaped wall members **104, 106** rotatably mounted over tube **100.** The inner wall member **104** is formed with a central opening **108** which receives the end of tube **100** and is in continual cooperation with a dynamic seal **107.** The central portion of the outer wall member **106** is closed and may be appropriately formed to function

as a cone **110** for directing air drawn into the rotor assembly by the blades. The outer periphery of inner wall **104** is turned inwardly to form a lip **112** which abuts the end of each blade **30'** so as to divide the blade substantially in half. In a similar manner, the outer periphery of the outer wall **106** is inwardly turned to form a lip **114** which cooperates with the blade end edge. The distribution chamber **102** thereby serves to direct the incoming thermal fluid to the lower half portion of the blade as seen in FIG. **3.** Preferably, the blade is formed with a plurality of longitudinally extending cells as was the case in the embodiment described previously. In this case, the distribution chamber **102** communicates with the lowermost cells as seen in FIG. **3,** and these cells will function as thermal fluid inlet circulation passages. Integrally provided within distribution chamber **102** are a plurality of evenly spaced impeller blades **116** which rotate with the distribution chamber as will be described to direct the incoming thermal fluid outwardly towards the rotating blades **30'**.

An annular thermal fluid collection chamber **120** is formed by an end member **117** having a lip **118** formed on its outer periphery in cooperation with the other edge of the blade end and the outer surface of inner wall member **104.** The end member **117** is rotatably mounted on a stationary tubular pipe **122** by means of bearings **121** positioned thereover in cooperation with a collar **124** centrally formed on end member **117.** A plurality of stator blades **126** have their ends **128** rigidly attached to the fluid inlet passage **100** so that they are stationary within the fluid collection chamber **120.** The inner surface of the stationary tubular pipe **122** cooperates with the outer surface of the inlet tube **100** to form an annular fluid passage **130** which serves to transmit the fluid back to the external equipment. An O-ring assembly **131** is provided to prevent escape of the thermal fluid from the fluid outlet passage **130.**

In operation, the blades are rotated via rotation of a pulley **132** formed on collar **124** which rotates end member **117** thereby causing the rotor blades **30'** to revolve. This in turn revolves the inner and outer opposed walls **104, 106** defining the fluid distribution chamber and integral impeller blades which are rigidly connected to the blades. The incoming thermal fluid is directed to the periphery of the distribution chamber and enters the inlet circulation passages in the blades. Upon circulating through the blades, the fluid enters the fluid collection chamber **120.** At this point, the circular motion of the fluid is interrupted by the stator blades **126** which convert the high circumferential velocity of the fluid to radial movement with little or no loss in momentum thereby causing a pressure increase of the fluid within the collection chamber **120.** The fluid is directed outwardly through the fluid transfer passage **130** at a higher pressure than when it entered through inlet passage **100.** Thus, the rotary heat exchanger functions as a pump. This enables external fluid pumps in the circulation system to be dispensed with.

What is claimed is:

1. Apparatus for effecting heat exchange between an ambient fluid and a thermal fluid comprising:

a rotatably mounted rotor assembly including,

a substantially circular manifold plate having fluid distribution passages formed therein defining a single plane, the two ends of each of said distribution passages terminating in the central area and the peripheral area of said plate respectively, and fluid collection passages located in the same single plane as said distribution passages, the two ends of each of said collection passages terminating in the central area and the peripheral area of said plate, respectively, and

a plurality of hollow thermal transfer blades extending substantially parallel to the axis of rotation of said rotor assembly, each having one end fixed to the peripheral area of said manifold plate, the interior of at least one of said thermal transfer blades communicating with the peripheral end of one of said fluid distribution passages and the interior of at least one of said thermal transfer blades communicating with the peripheral end of one of said fluid collection passages; and

a hub assembly contiguous to said manifold plate including

a fluid inlet passage communicating with the central end of each of said fluid distribution passages and a fluid outlet passage communicating with the central end of each of said fluid collection passages.

2. Apparatus as recited in claim **1** wherein each of said fluid distribution passages has an arcuate shape, said arcuate shapes all lying within said single plane.

3. Apparatus as recited in claim **1** wherein each of said fluid collection passages has an arcuate shape, said arcuate shapes all lying within said single plane.

4. Apparatus as recited in claim **1** wherein said collection passages are alternately interposed between said distributor passages and wherein each of said blades is divided into at least two separate chambers and mounted on said manifold plate with one of said chambers in communication with the peripheral end of a fluid distribution passage and the other of said chambers in communication with the peripheral end of the adjacent fluid collection passage.

5. Apparatus as recited in claim **4** wherein the peripheral ends of each of said collection and distribution passages are enlarged so that the peripheral ends of pairs of adjacent collection and distribution passages overlap along blade mounting loci.

6. Apparatus as recited in claim **1** wherein the interior of each of said thermal transfer blades is provided with partitions defining a series of parallel cells extending longitudinally within said blade and wherein the ends of each partition remote from the manifold plate are recessed, the extent of such recess in each partition increasing incrementally in those partitions in closer proximity to the center of said blade.

7. Rotary apparatus for effecting heat exchange between an ambient fluid and a thermal fluid comprising:

a rotor assembly including,

a rotatably mounted fluid distribution chamber having a central fluid inlet for receiving thermal fluid from external equipment and a peripheral fluid outlet for directing thermal fluid into the interior of hollow thermal transfer blades and a plurality of impeller blades fixed within said distribution chamber for rotation therewith;

a fluid collection chamber having a central fluid outlet and a peripheral fluid inlet for receiving thermal fluid from the interior of hollow thermal transfer blades, said fluid collection chamber having a plurality of stationary stator elements located therein,

a plurality of hollow thermal transfer blades, the interior of at least one of said blades communicating with the peripheral fluid outlet of said fluid distribution chamber and the interior of at least one of said blades communicating with the peripheral fluid inlet of said fluid collection chamber,

a hub assembly contiguous with said rotor assembly including

a fluid inlet passage communicating with the central fluid inlet of said distribution chamber and a fluid outlet passage communicating with the central fluid outlet of said collection chamber.

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