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Laing

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[54] **ROTARY HEAT EXCHANGERS IN THE FORM OF TURBINES**

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[52] **U.S. Cl.** ..... **165/85,** 60/39.51 H, 165/92, 165/105, 165/107, 165/125

[51] **Int. Cl.** **F28d 11/04**

[58] **Field of Search** ..... 165/85, 86, 92, 110, 125, 165/105; 60/95 R, 39.51 R, 39.51 H

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[57] **ABSTRACT**

A rotary heat exchanger, in which an internally circulating heat-carrying fluid thermally interacts with an external fluid of different temperature, is set in rotary motion by the kinetic force of either of these fluids acting upon a set of turbine blades on a rotary member carrying the heat-exchanging surfaces. In one embodiment, the rotary member forms a header centered on the axis of rotation and carries a set of axially extending tubes which communicate with the header at one end and are closed at the opposite end; the tubes are embraced by annular fins at locations remote from the header and are arrayed in a plurality of concentric groups, their lengths decreasing progressively from the outermost to the innermost group.

8 Claims, 6 Drawing Figures

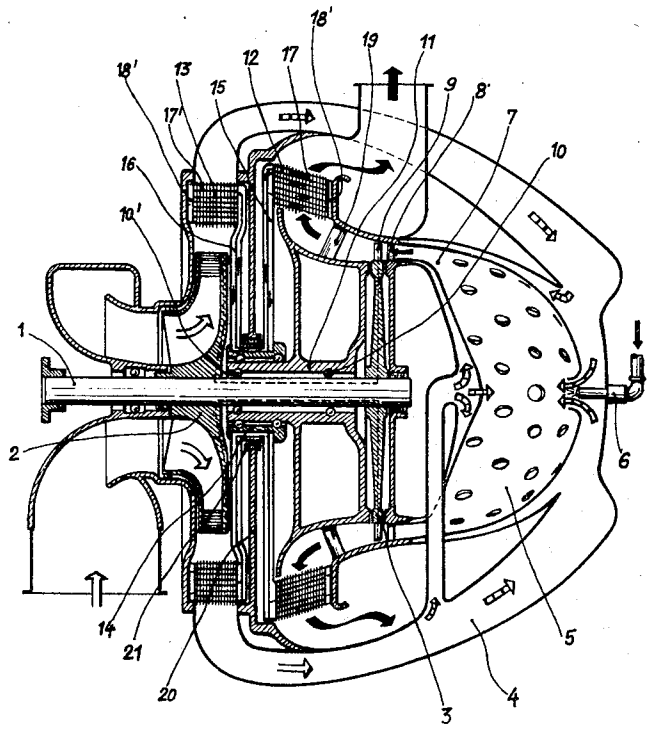
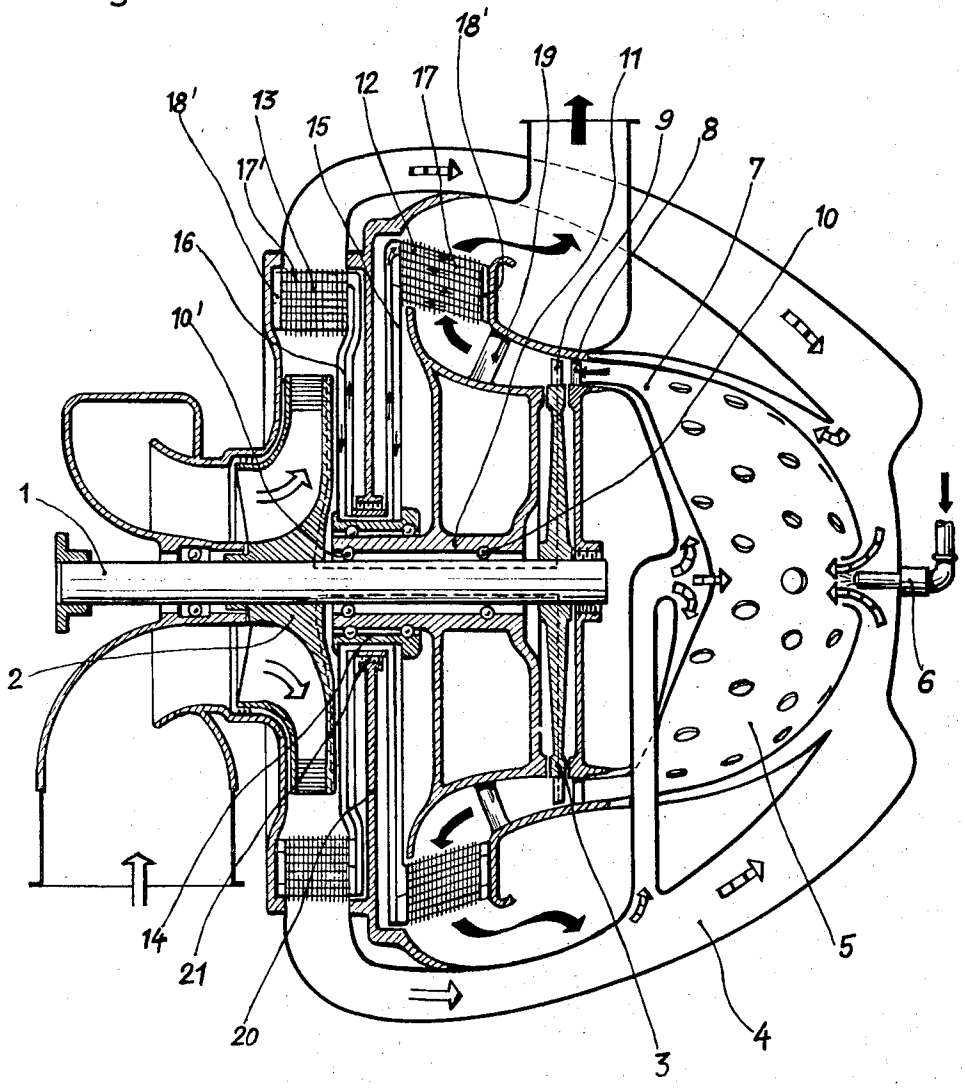
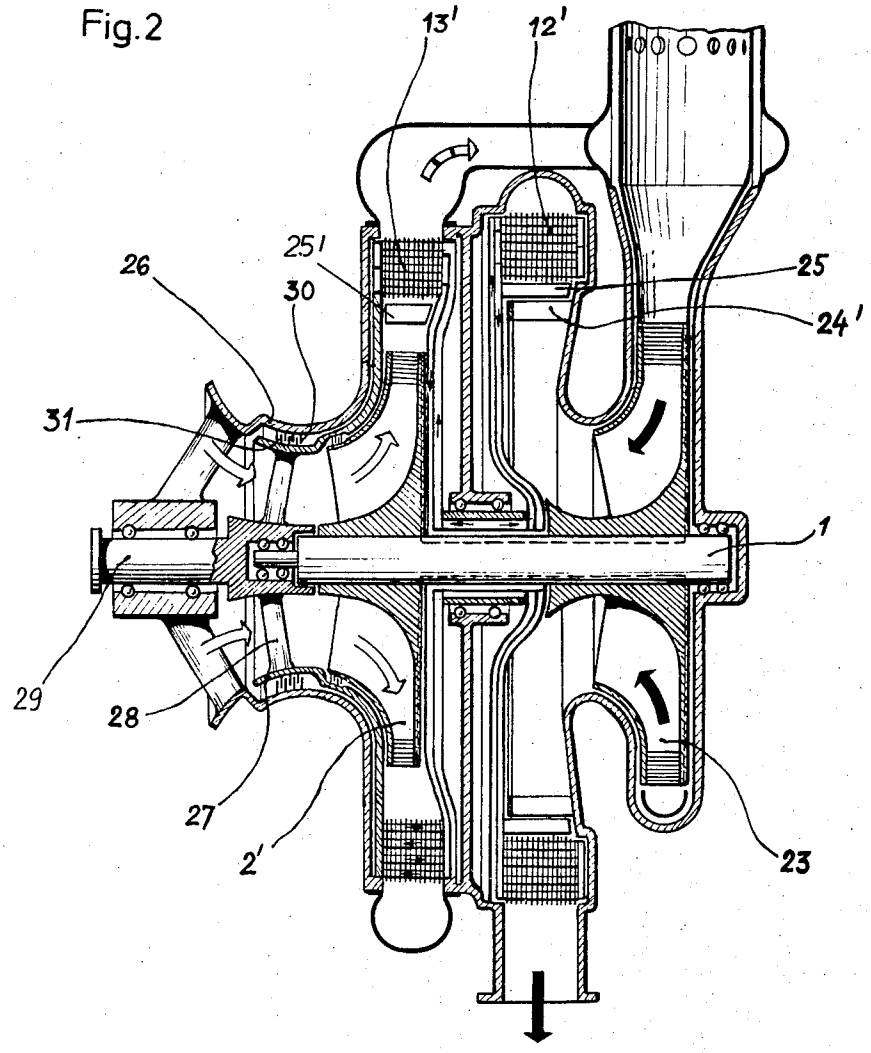


Fig.1



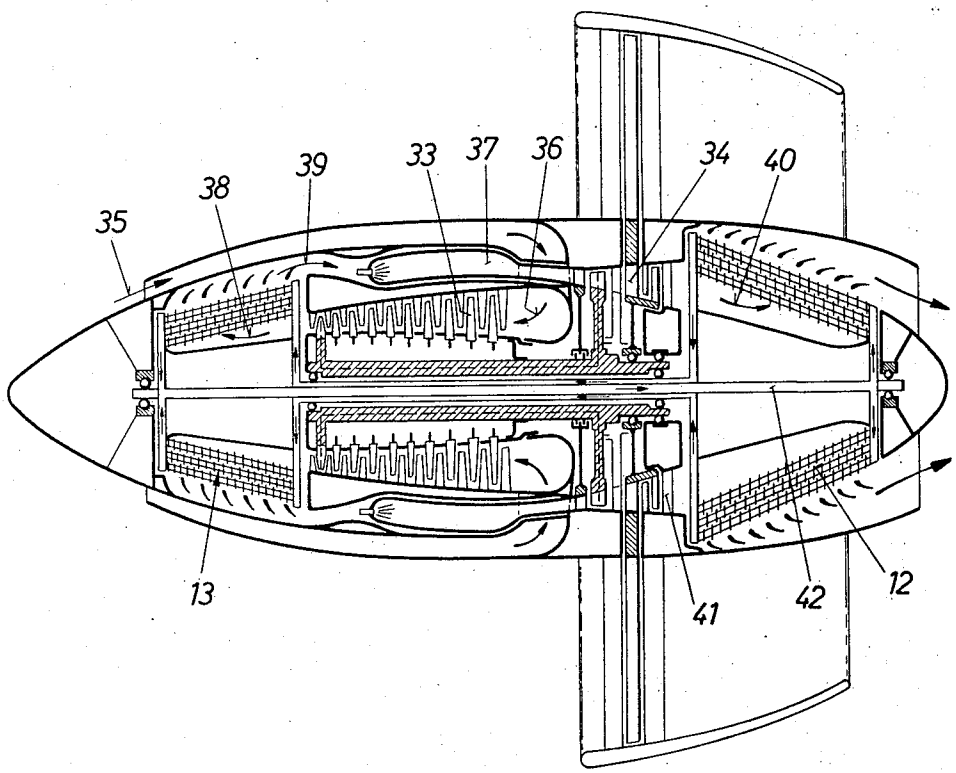
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Fig.2



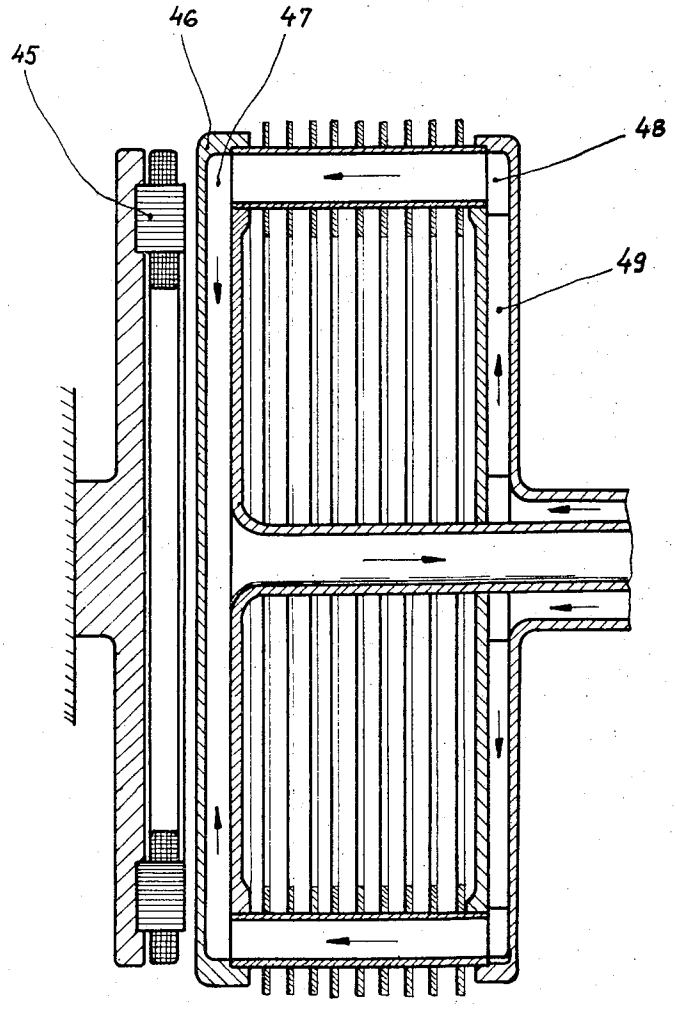
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FIG. 3



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Fig. 4



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FIG. 5

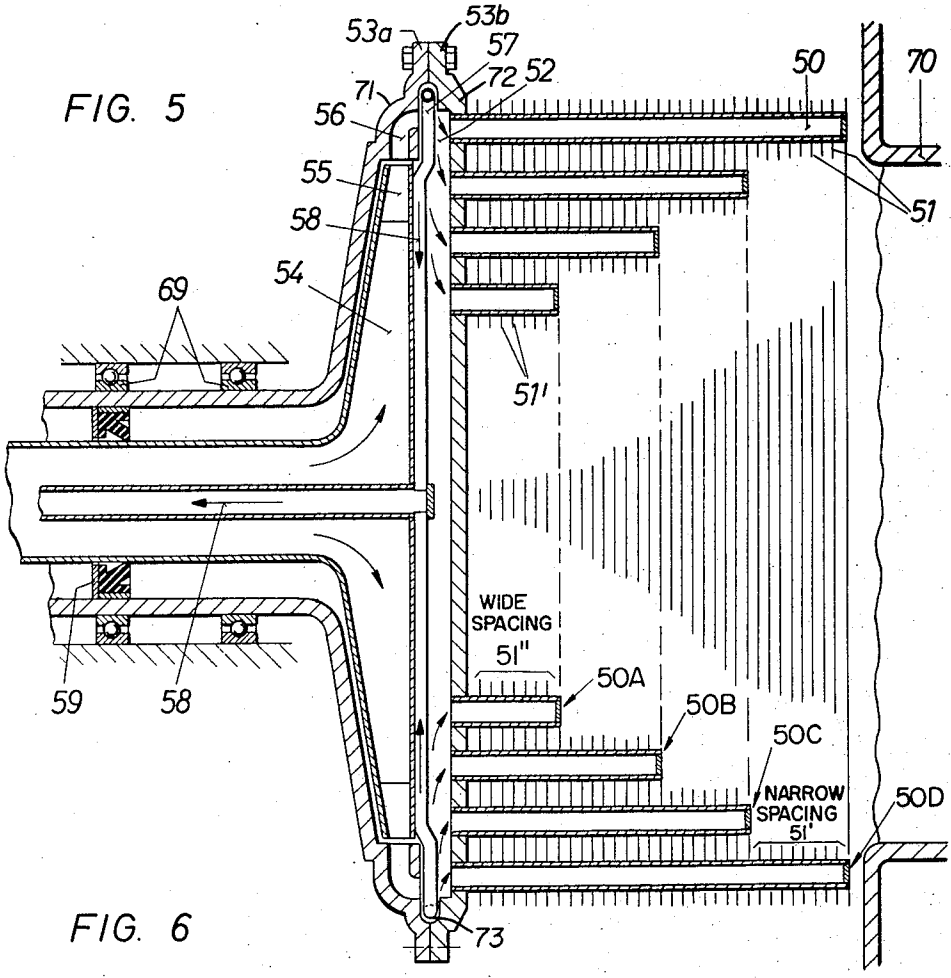
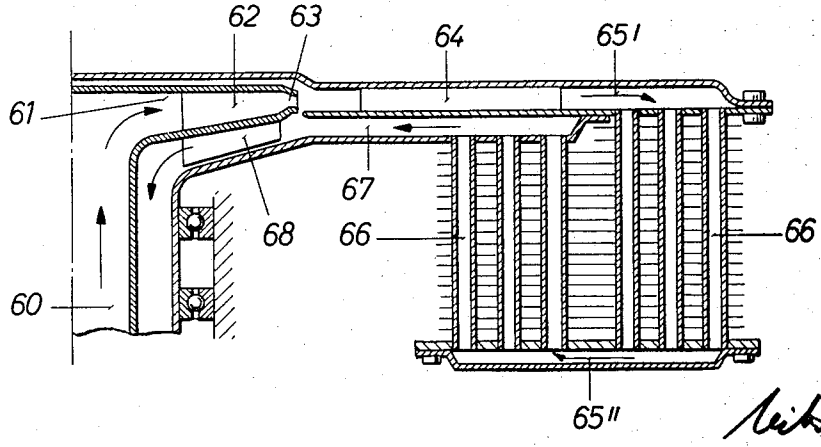


FIG. 6



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ROTARY HEAT EXCHANGERS IN THE FORM OF TURBINES

FIELD OF THE INVENTION

My present invention relates to a rotary heat exchanger of the type wherein two heat-carrying fluids, one of them circulating in a channel system which rotates about an axis, thermally interact through the walls of a rotating member forming the channels.

BACKGROUND OF THE INVENTION

Conventional rotary heat exchangers may act as impellers for the external, usually gaseous heat carrier. In some instances they operate simultaneously as a pump and therefore also convey the internal heat carrier. If the heating effort is to be reduced, the rotational speed must be lowered, since driving power, which varies as the cube of the speed, would otherwise be wasted. This entails expensive control drives.

A further disadvantage of rotary heat exchangers resides in the fact that the rotary system of channels which is filled with a heat carrier has a large moment of inertia, defined as the product of mass times the square of its radius which militates against acceleration. This has a particularly adverse effect on the cooling of internal-combustion engines, which inherently permit an extremely high acceleration, so that the drives for the heat exchanger, e.g., V-belts, have to be overdimensioned by several factors.

SUMMARY OF THE INVENTION

According to the invention, the rotary heat exchanger is designed as of a turbine and driven by the internal or the external heat carrier. The driving energy can be supplied to the heat carrier by means of a blower or a pump; very frequently, however, driving energy is available from one of the fluids, as where a hot stream of gas is to be cooled by means of mains water. The interior of the heat exchanger is then provided with turbine blading in which the mains-water pressure is converted into driving power for the rotary heat exchanger. Such an arrangement has the advantage that the temperature reduction for the hot gas is approximately constant, since the greater the amount of cooling water flowing through the heat exchanger, the higher is its speed and the greater is the amount of gas which is inducted by it.

Heat exchangers frequently find application in installations of machines in which one of the heat carriers still has usable kinetic energy. Thus, for example, the exhaust gas from a gas turbine still has an appreciable energy content which, in accordance with the invention, is utilized for driving the rotary heat exchanger. Thus, in this case, the outer heat-exchange surface of a rotary heat exchanger is constructed as a gas-driven turbine, whereas a liquid heat carrier in the interior of the channels is conveyed by pumps which obtain their driving energy from the rotation of the heat exchanger.

Other features of my invention relate to thermodynamically and hydromechanically advantageous constructions of rotary heat exchangers.

BRIEF DESCRIPTION OF THE DRAWING

The invention will now be further explained with reference to the accompanying drawing as regards its principle of operation and the improvements which it provides. In the drawing

FIG. 1 shows a gas turbine with a heat exchanger filled with a liquid and arranged between a compressor and a turbine rotor;

FIG. 2 shows a comparable gas turbine in which the heat exchanger is used to provide the entire torque;

FIG. 3 shows a gas turbine in which the heat exchanger is arranged fore and aft of a compressor-turbine unit;

FIG. 4 shows a rotary heat exchanger for electrically conductive liquid heat carriers with eddy-current-producing magnets;

FIG. 5 shows a steam-driven rotary condenser; and

FIG. 6 shows a liquid-driven rotary heat exchanger which conveys air.

SPECIFIC DESCRIPTION

FIG. 1 shows a gas turbine according to the invention. The shaft **1** carries the radial-compressor wheel **2** and the axial-turbine wheel **3.** The coaxial and approximately rotationally symmetrical combustion chamber **5** is supplied with compressed air via the conduits **4** and with fuel via the nozzle **6.** The hot gases are expanded in the annular chamber **7.** Thereafter the combustion gases flow through a guide-blade ring **8** and then traverse the axial blades **9** of the turbine wheel **3.** The shaft **1** is carried in antifriction bearings **10** and **10'** in the housing portion **11,** and a heat-exchanger unit, consisting of a first heat exchanger **12** constituting the heat sink and a second heat exchanger **13** through which the compressor air flows, is also carried by the same stationary housing portion **11** by means of hub **14.** Channels **15** and **16** lead to the hub **14.** The heat exchangers **12** and **13** consist of tubes **17** and **17'** having substantially parallel axes, the tubes communicating with one another at their outlying ends by rotary annular chambers **18** and **18',** and with the channels **15** and **16** at their opposite ends. The entire channel system is filled with a thermally stable liquid, preferably a eutectic of sodium and potassium. A guide-blade ring **19** gives the stream of exhaust gas emerging from the axial blading **8, 9** a whirl whose thrust serves the purpose of driving the heat-exchanger unit **12** and **13** by its passage through the heat exchanger **12.** As a result of the rotation of the heat exchangers and the heating up of the wall of the channels facing the turbine, the internal liquid heat carrier is set in circulation in the direction of the arrows, since in the larger diameter heat exchanger **12** the inwardly flowing liquid stream is of lower density than the portion of liquid flowing outwardly from the interior. In this way the heat which is withdrawn from the exhaust gas in the heat exchanger **12** is transferred to the compressor air via the heat exchanger **13.** A partition wall **20** which is made up of two halves and carries a shaft seal **21** separates the heat exchangers. The power take-off is via the shaft **1.**

FIG. 2 shows a similar arrangement in which the heat exchangers **12'** and **13'** assume not only the function of recuperation but also that of a fluid transmission. The high-speed machine consisting of a radial-compressor wheel **2'** and a radial turbine **23** serves exclusively the purpose of generating hydraulic energy. This hydraulic energy drives the rotary heat exchangers **12'** and **13',** a guide-blade ring **24** and rotor-blade ring **25** being so dimensioned that they are capable of transforming this hydraulic energy into mechanical energy on the shaft.

Between the housing wall 26 and the radial-compressor wheel 2' a ring 27 extends which seals in both directions by means of a labyrinth seal 30, 31 and which forms a unit with the driving shaft 29 via spokes 28. The arrangement shown makes considerable reduction ratios possible.

FIG. 3 shows diagrammatically an arrangement of a fan drive, in which the heat exchangers 12 and 13 are arranged outside the gas turbine consisting of a compressor 33 which leads to the heat exchanger 13 and a turbine stage 34 leading to the heat exchanger 12. The air enters in the direction of the arrow 35, is conducted to the compressor 33 in the direction of the arrow 36 via deflectors which are distributed over the circumference and which cross the annular combustion chamber 37, thence flows in the direction of the arrow 38 through the heat exchanger 13, thence in the direction of the arrow 39 into the combustion chamber, thereafter through the turbine stage 34, finally to be conducted in the direction of the arrow 40 through the heat exchanger 12; the latter is caused to rotate by the whirl component produced in the guide wbeel 41. The internal channels of heat exchanger 12 communicate with thereof the heat exchanger 13 via an axially entering twin tube 42. FIG. 4 shows diagrammaticaly a rotary heat exchanger, particularly for hermetically sealed heat-carrier circuits and a heat carrier consisting of molten metal. By means of magnets 45 an eddy current is produced within the revolving liquid metal in the region 47, via the non-magnetic wall 46, the eddy current causing braking of the circumferential velocity of the liquid metal, Since no retardation takes place in the opposite annular channel 48, which may be provided with blades 49 as shown, a circulation commences in the direction of the arrows. Permanent magnets may also be used in place of electromagnets.

FIG. 5 shows a heat-exchanger arrangement in which the internal heat carrier, which flows through the channels 50 is used for driving purposes in place of the external heat carrier. The heat exchanger shown serves as a condenser. The channels 50, which are provided with annular fins 51 lying in the plane of rotation, communicate at one end with an annular chamber or header 52 and are closed at their other ends.

A stationary hollow body 54 through which steam is conducted is provided in the annular chamber 52 which is created by bolting two shells 71, 72 together at flanges 53*a*, 53*b* thereof body 54 has radially oriented discharge nozzles 55. The steam jets being discharged traverse the blading ring 56 which is provided with turbine blading. Thereafter the steam is distributed to the channels 50, from which the condensate flows back into the chamber 52 and collects at its periphery. Scooping pipes 57, which are bent over in the direction opposite to the direction of rotation, extend into this annulus of water condensate. These pipes convert the velocity of the revolving water annulus into pressure and convey the condensate back in the direction of the arrows 58. The shaft seal 59 prevents leakage of steam from the rotary heat exchanger which is supported by the anti-friction bearings 69. The radial distance of the annular fins 51 from the axis of rotation increases with the distance from the rotary header 52 while their external diameter is preferably kept constant; the spacing between the fins 51 varies inversely with the internal fin diameter so as to be at locations 51' remote from header 52, where their inner radius is large, than at locations 51'' proximal to the header, where that radius is small.

The relative staggering of the closed ends of the tubes 50 (remote from header 52), together with the progressive increase of the inner radii of their annular fins 51 in conformity with this longitudinal staggering, results in a more efficient heat exchange between the circulating internal fluid (steam) and the external medium (air) surrounding the tubes. Thus, as the external medium rotationally entrained by the fins and tubes is driven radially outwardly by centrifugal force, it simultaneously cools the projecting outer ends of all the tubes of the four concentric groups 50A, 50B, 50C, 50D illustrated in FIG. 5, ranging from the shortest innermost group 50A to the longest outermost group 50D. At the same time, the fins closest to the header 52 are cooled most intensely since they have the largest surface area exposed to the external gas flow. Moreover, the longer outer tubes give up more heat to the gas flow than the shorter inner tubes so that condensation is more effective near the outer periphery of the rotating tube carrier 71, 72, i.e. in the vicinity of the trough 73 which is swept by the scooping pipes 57. The aforedescribed inverse relationship between the spacing of fins 51 and their inner radius establishes a more uniform flow resistance for the stream of cooling air throughout the axial length of the rotor.

FIG. 6 shows a rotary heat exchanger for cooling liquids. The liquid heat carrier enters under pressure through the stationary tube 60 and is distributed in the annular chamber 61 and thereafter flows through the guide-blade ring 62 and leaves the stationary part through the nozzle ring 63. The pressure energy which is converted into velocity by the nozzle ring 63 causes rotation of the rotary heat exchanger by impinging on the blades 64. The liquid heat carrier then passes through the channels 66 of the heat exchanger in the direction of the arrows 65' and 65'' and flows through the annular channel 67 in a centripetal direction. In the stationary blading 68 the circumferential velocity component is converted back into pressure.

I claim:

1. A rotary heat exchanger comprising a hollow member mounted for rotation about an axis; conduit means for the circulation of a fluid through said body in heat-exchanging relationship with an external medium differing in temperature from said fluid, said body forming a rotary header for said fluid centered on said axis, said conduit means including a set of tubes substantially parallel to said axis extending outwardly from said header, said tubes being arrayed in a plurality of concentric groups with the length of said tubes decreasing progressively from the outermost group to the innermost group; and a multiplicity of annular fins centered on said axis, embracing said tubes and physically interconnecting same at locations remote from said header, said annular fins having inner radii increasing progressively in axial direction in conformity with the staggered length of said tubes.

2. A heat exchanger as defined in claim 1 wherein said tubes have closed ends remote from said header.

3. A heat exchanger as defined in claim 2 wherein said fluid is a vapor liquefiable by heat exchange with said external medium, said header extending radially beyond said tubes and forming a peripheral trough for

collecting the liquefied fluid, said conduit means including discharge means terminating at said trough.

4. A heat exchanger as defined in claim 3, wherein said discharge means comprises at least one stationary pipe with an inlet in said trough facing in a peripheral direction opposite the direction of rotation of said member for scooping up the liquefied fliud collected therein.

5. A heat exchanger as defined in claim 1 wherein said annular fins have identical outer radii.

6. A heat exchanger as defined in claim 1 wherein the spacing of said fins along said axis decreases axially outwardly with the increase of their inner radii.

7. A heat exchanger as defined in claim 1 wherein said header is provided with turbine blades, further comprising inlet means for said fluid terminating within said body for training a stream of said fluid onto said turbine blades to impart rotation to said member.

8. A heat exchanger as defined in claim 7 wherein said inlet means comprises a stationary body provided with a peripheral array of nozzles confronting said turbine blades.

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