

Jan. 28, 1969

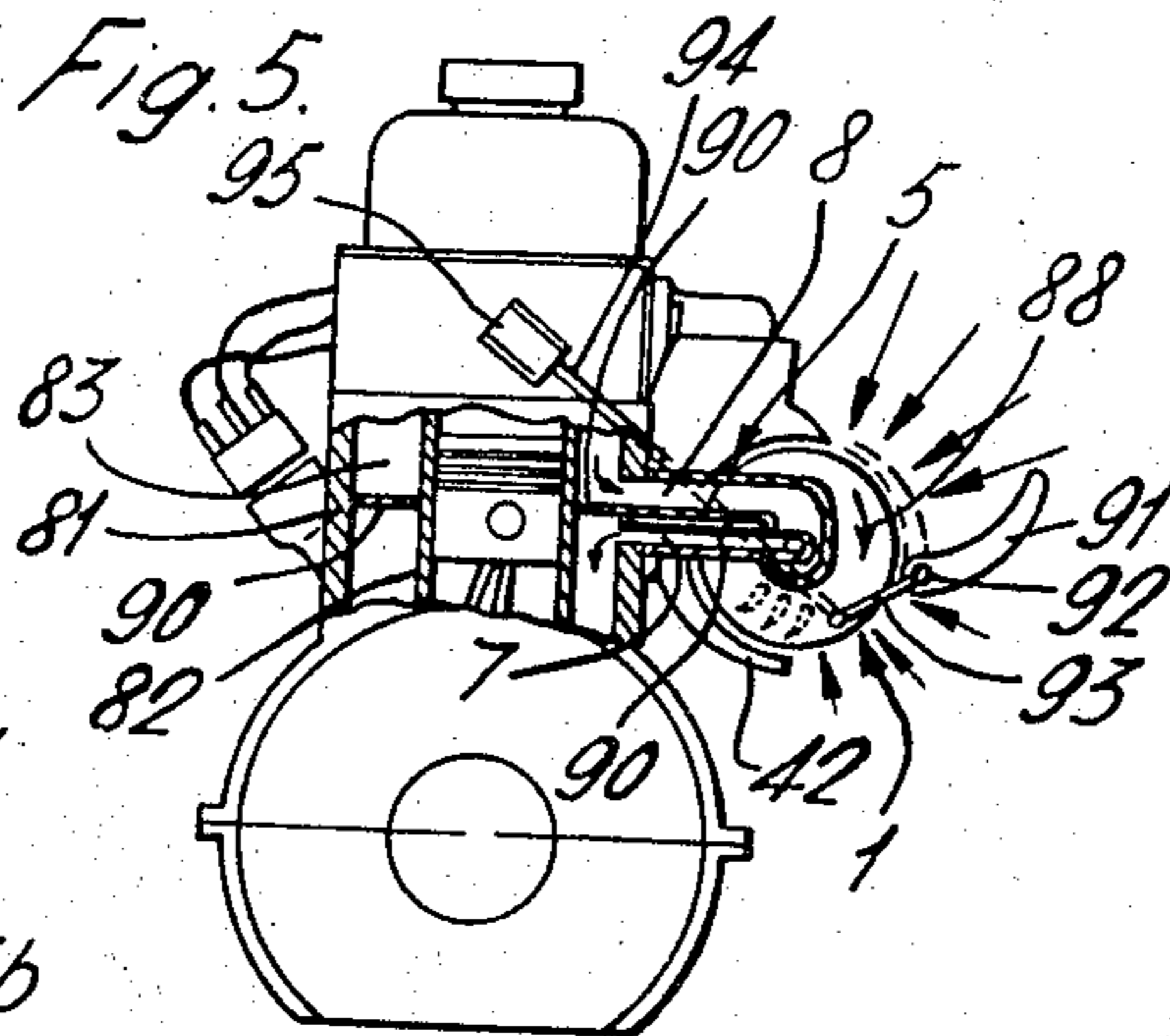
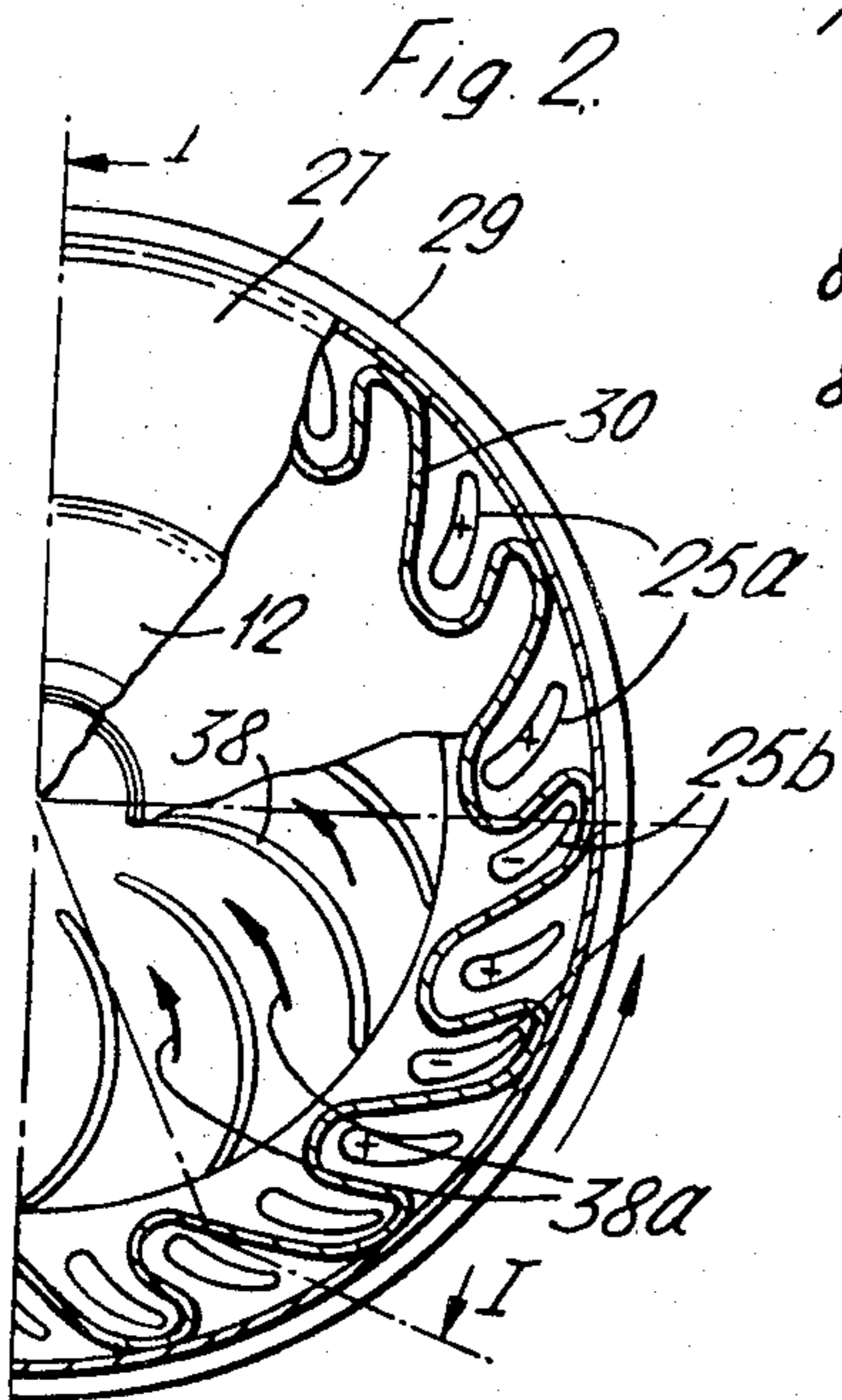
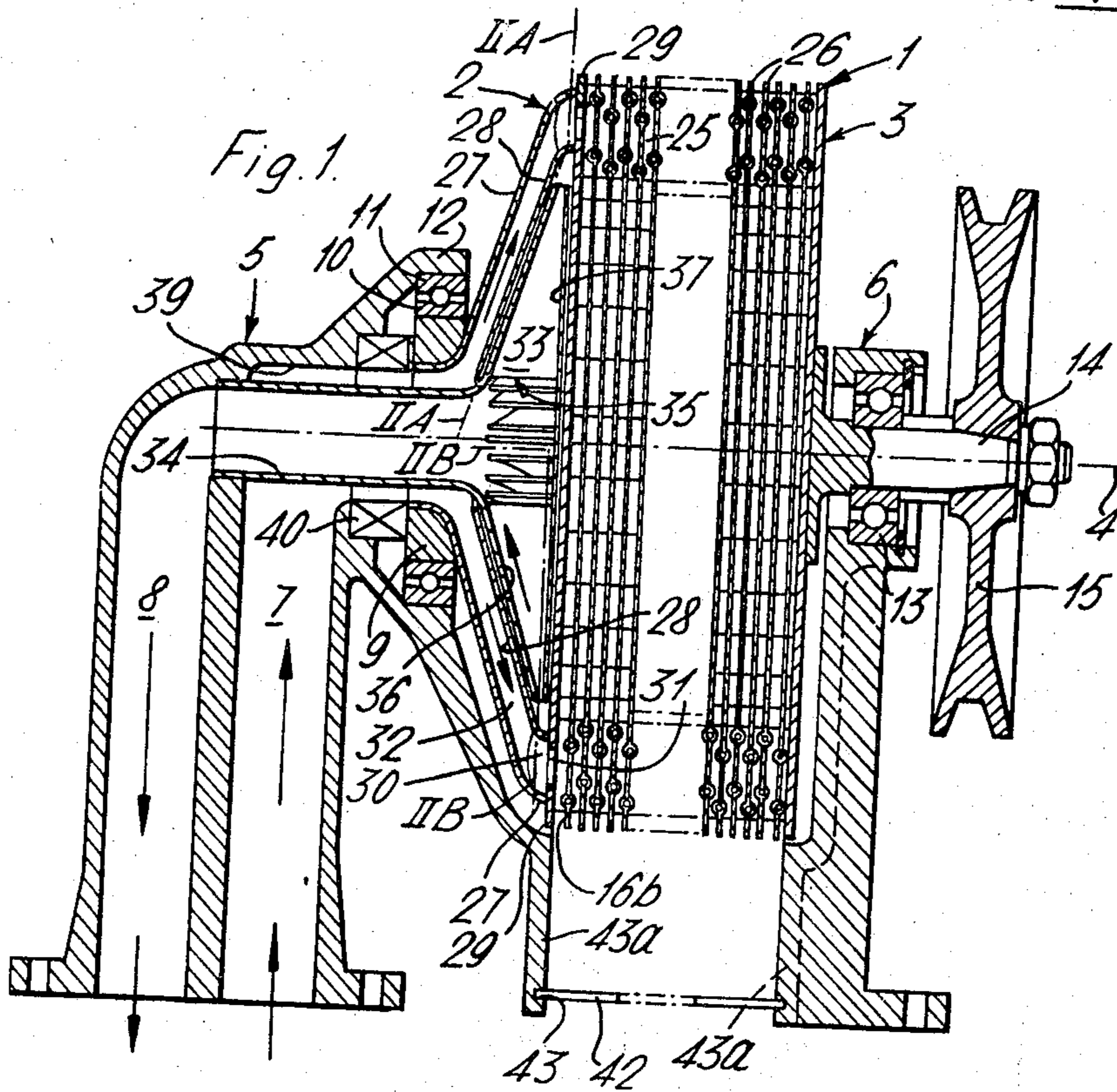
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3,424,234

ROTARY HEAT EXCHANGERS

Original Filed Jan. 2, 1963

Sheet 1 of 3



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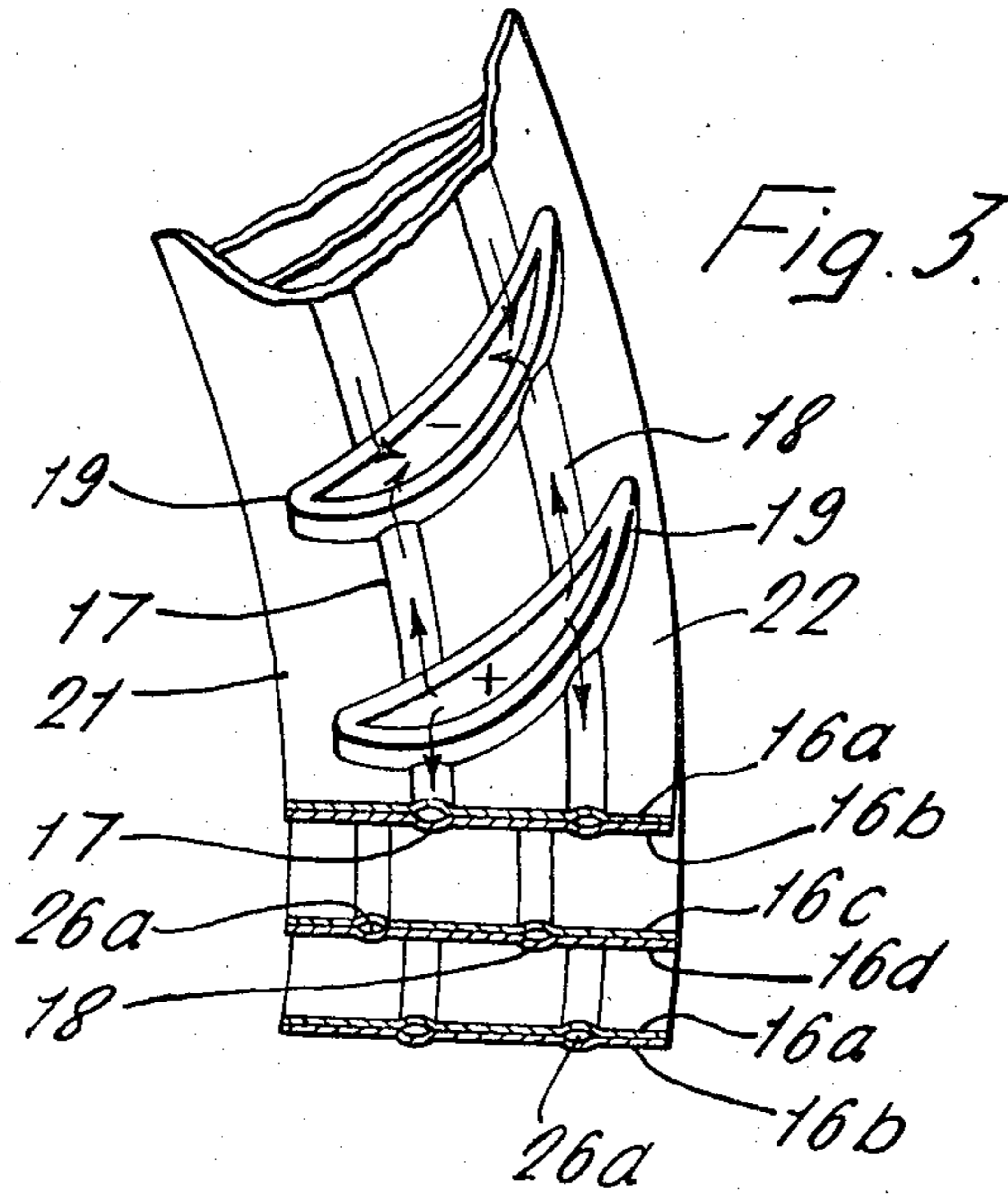


Fig. 3.

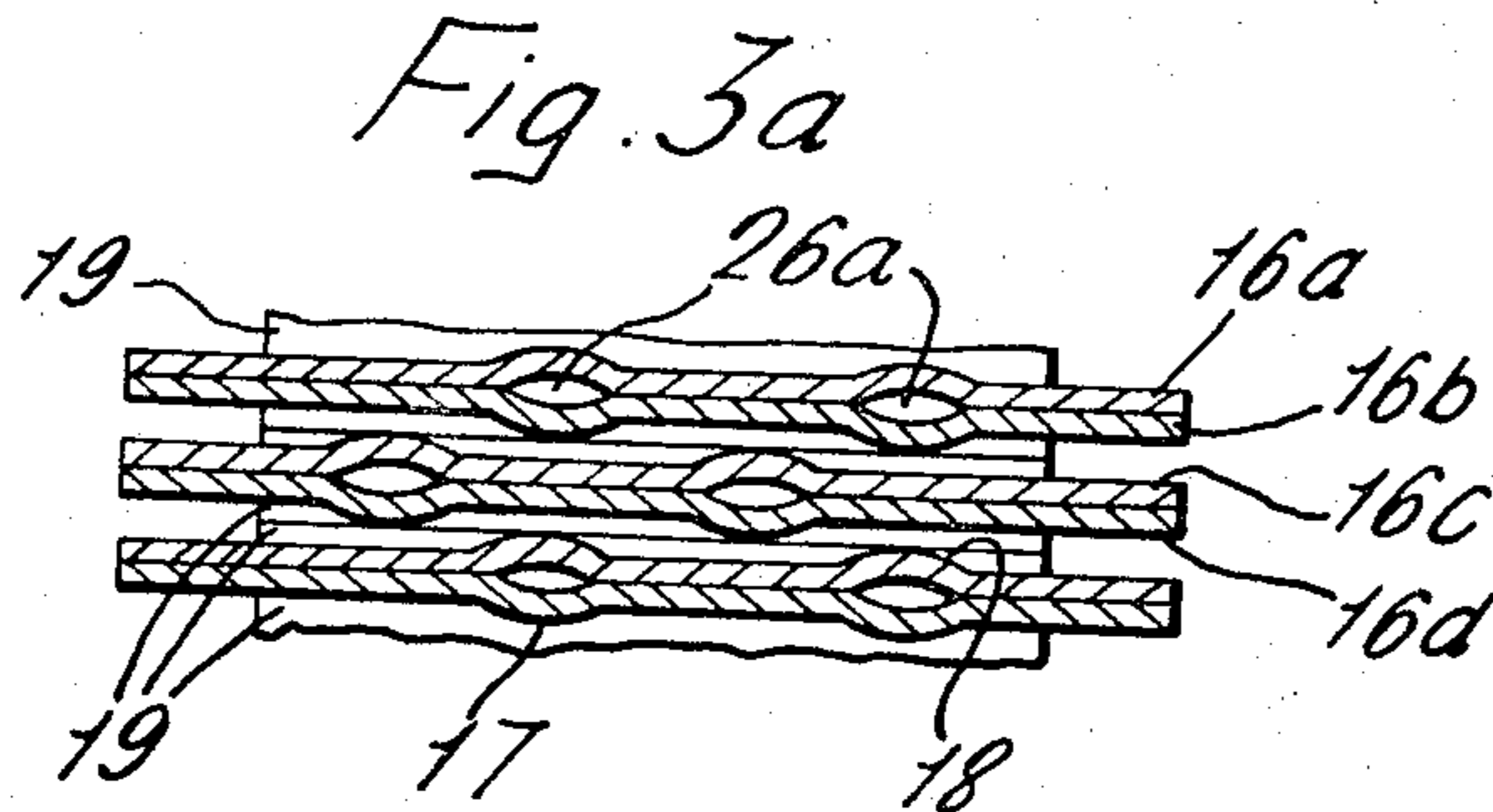


Fig. 3a

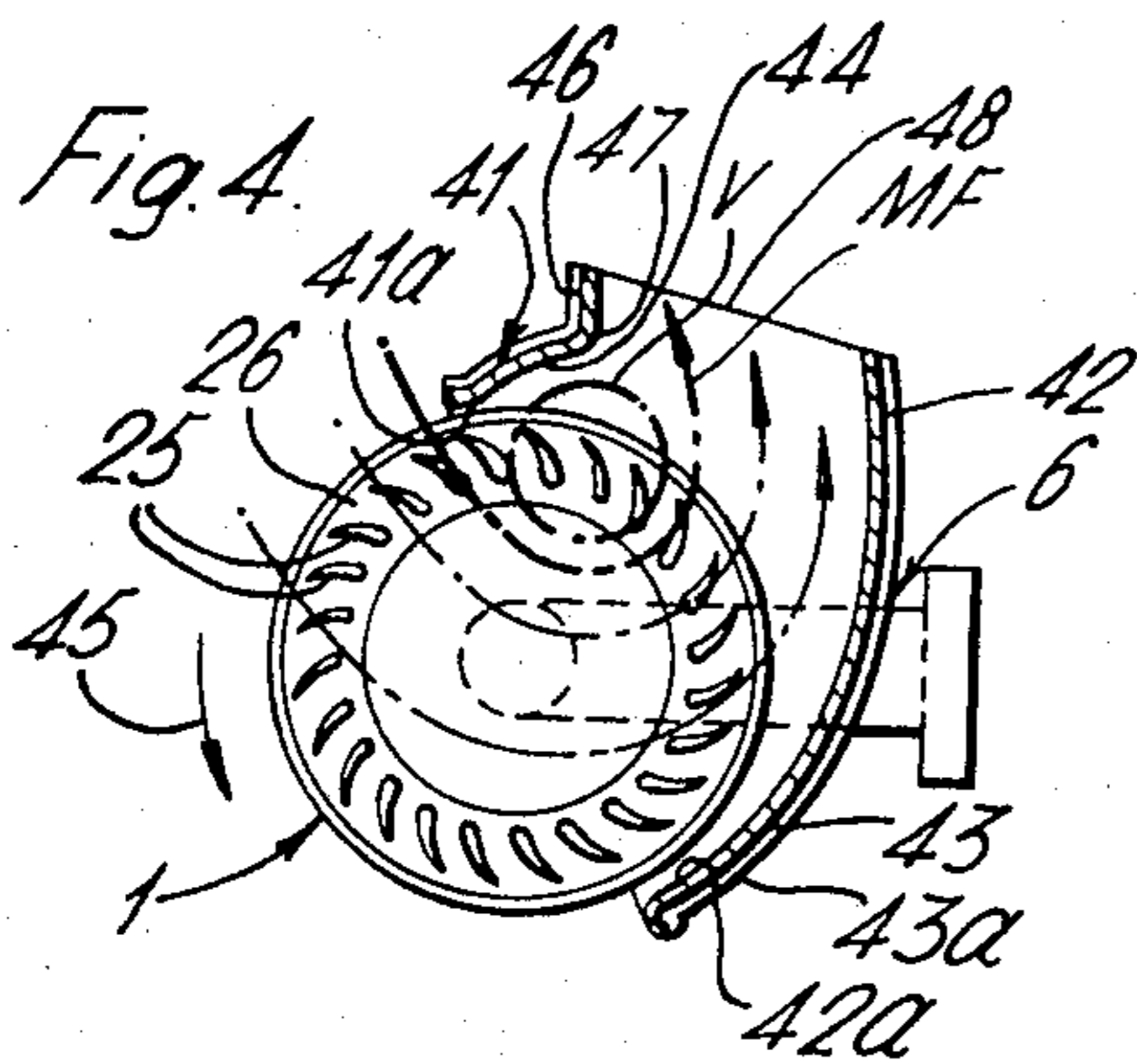


Fig. 4.

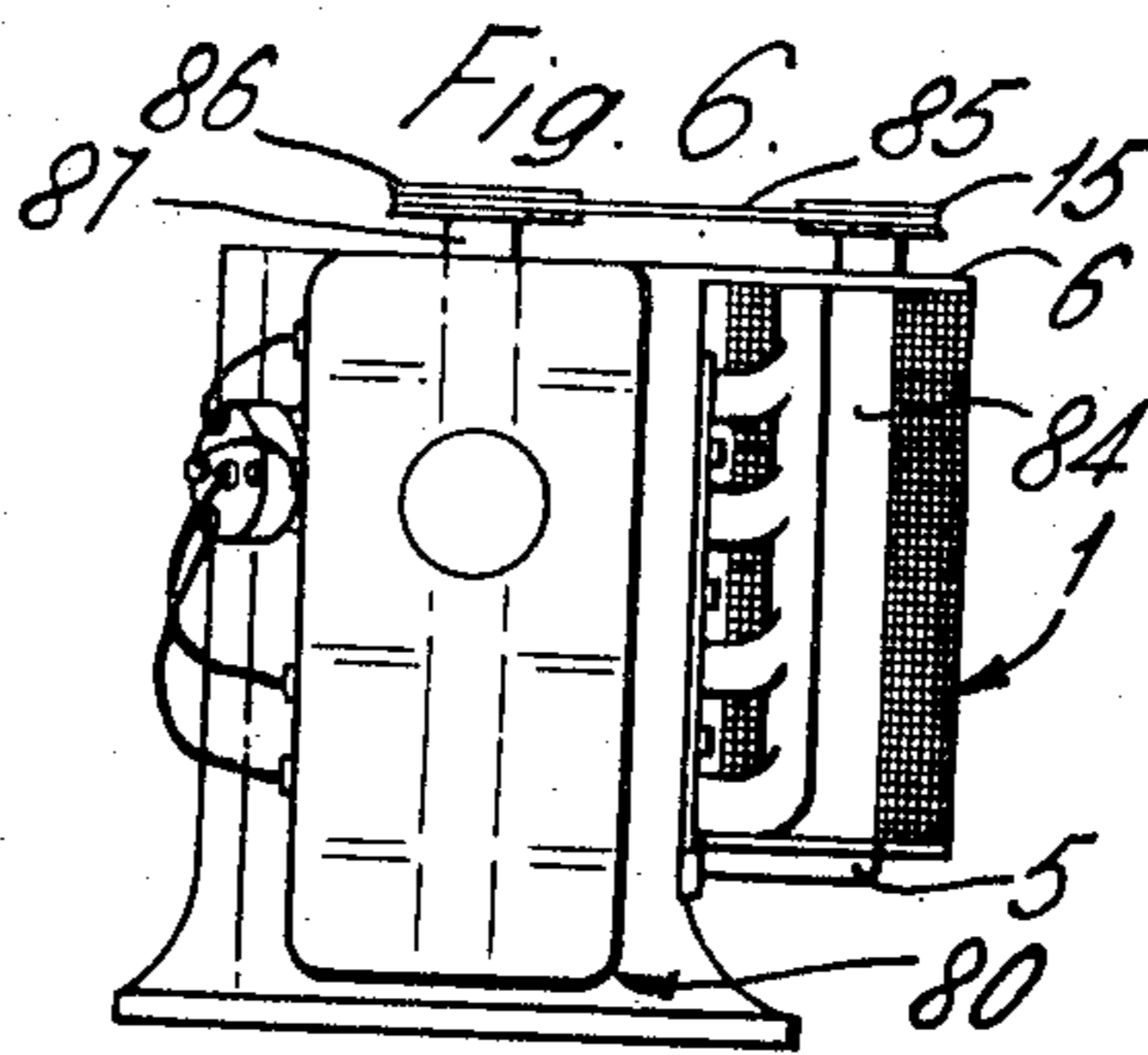


Fig. 6.

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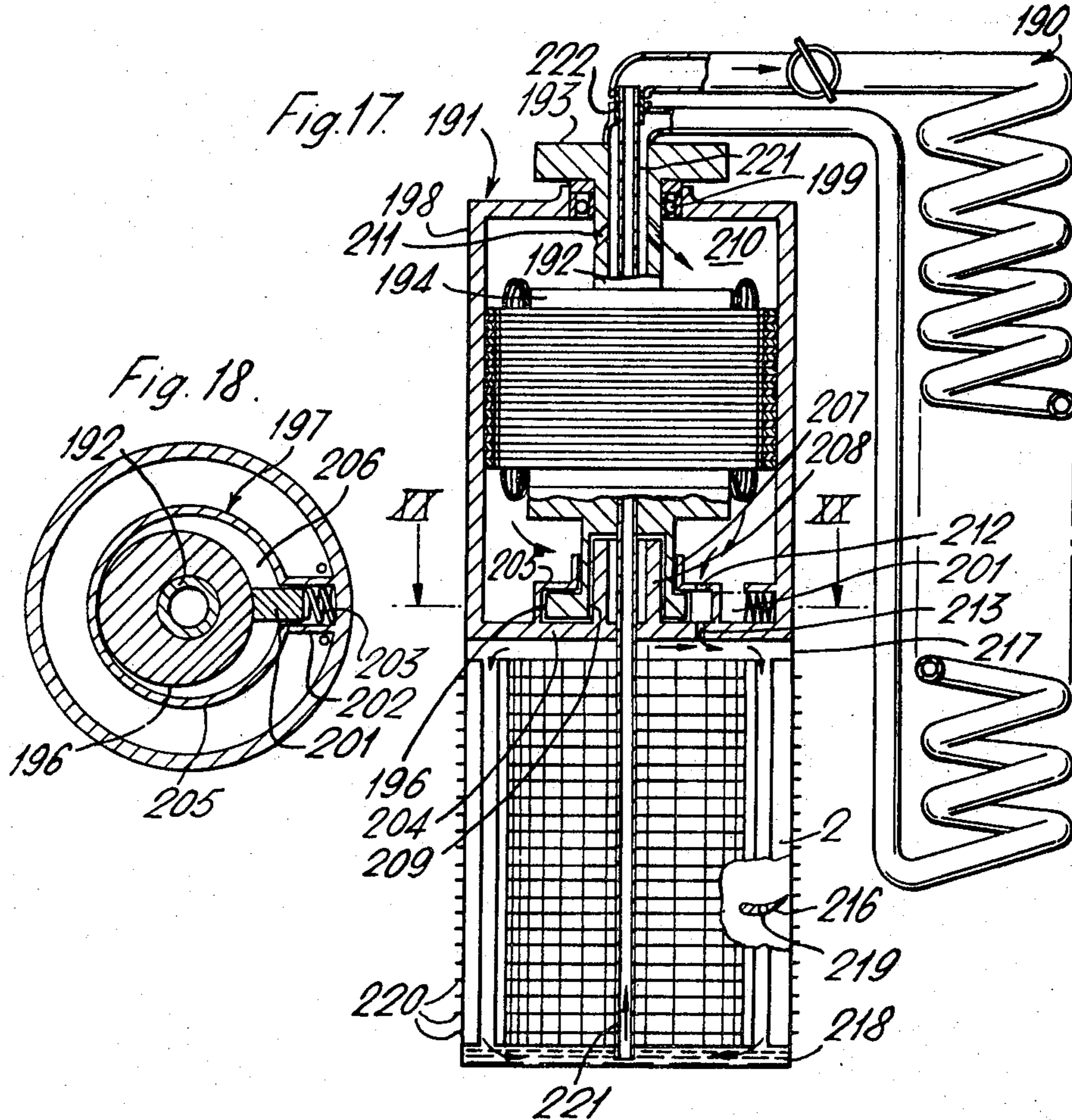
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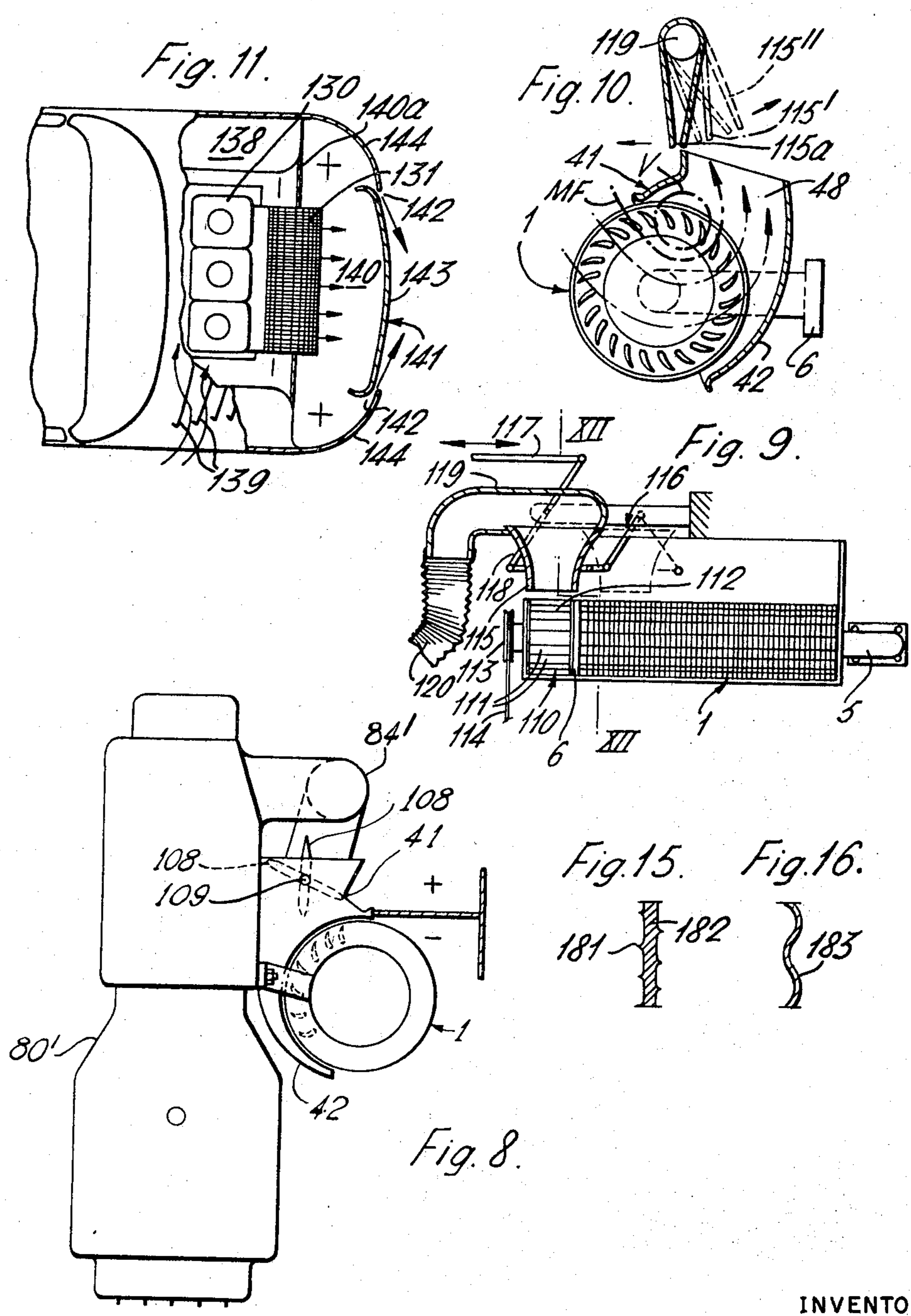
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ROTARY HEAT EXCHANGERS

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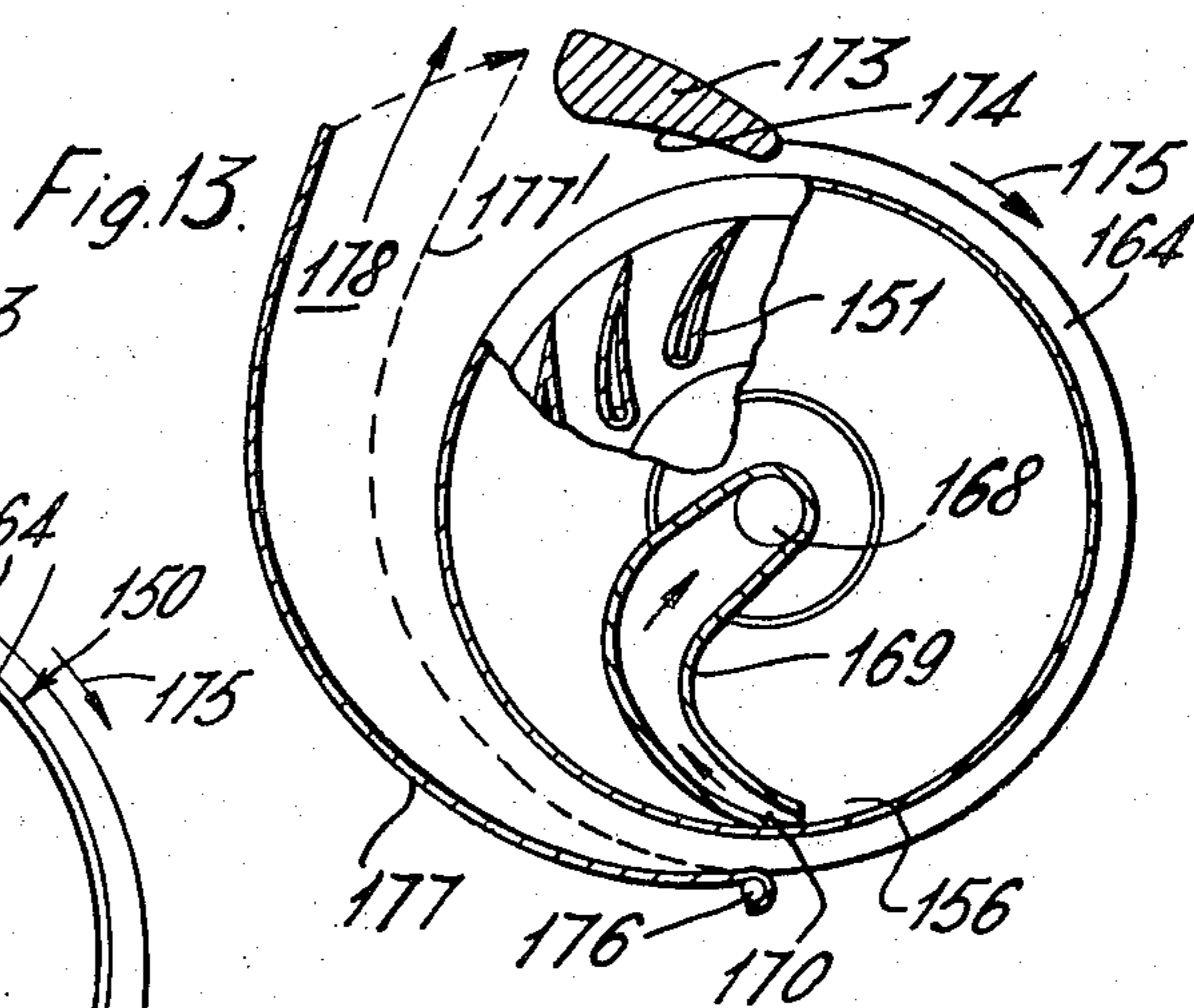
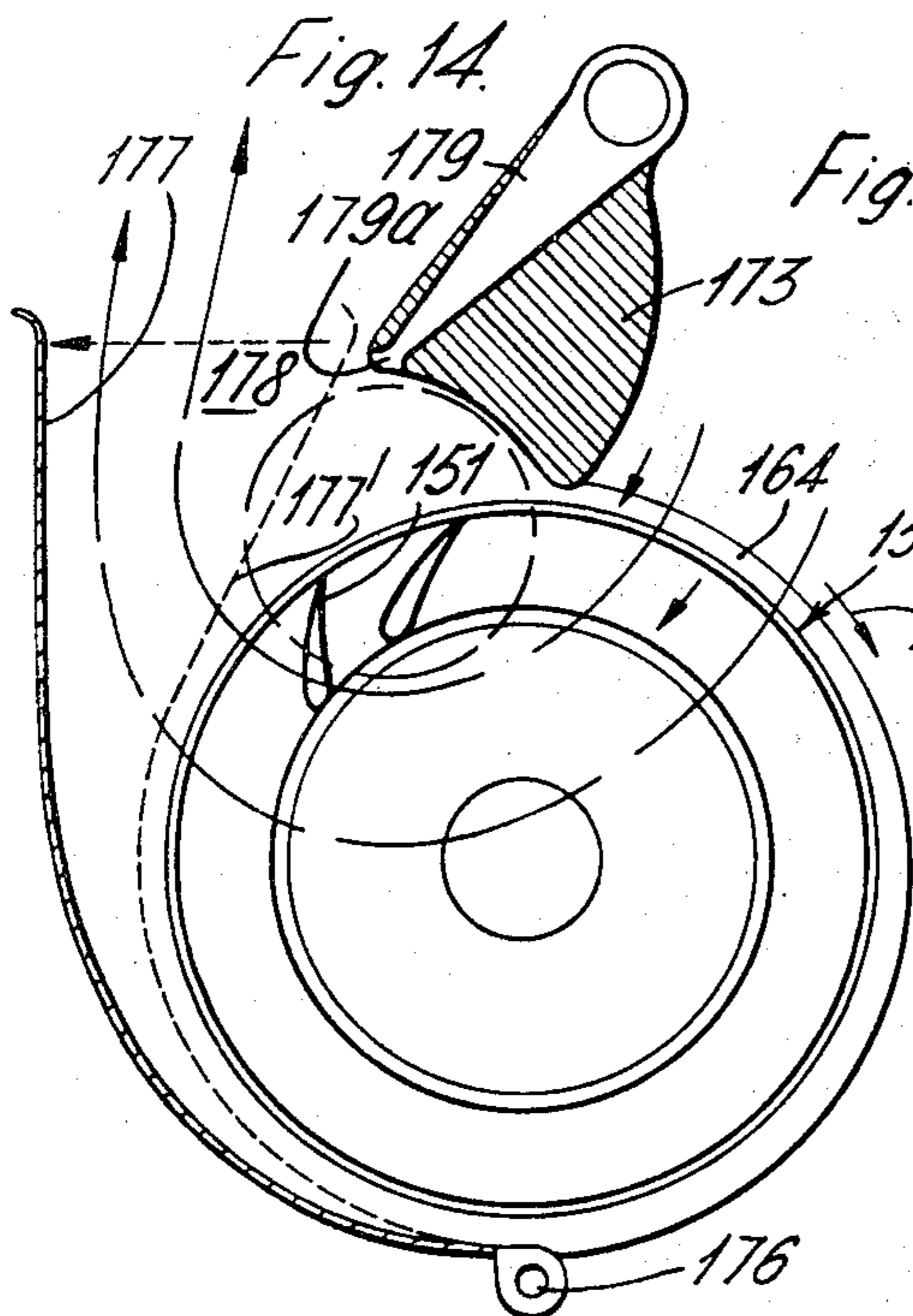
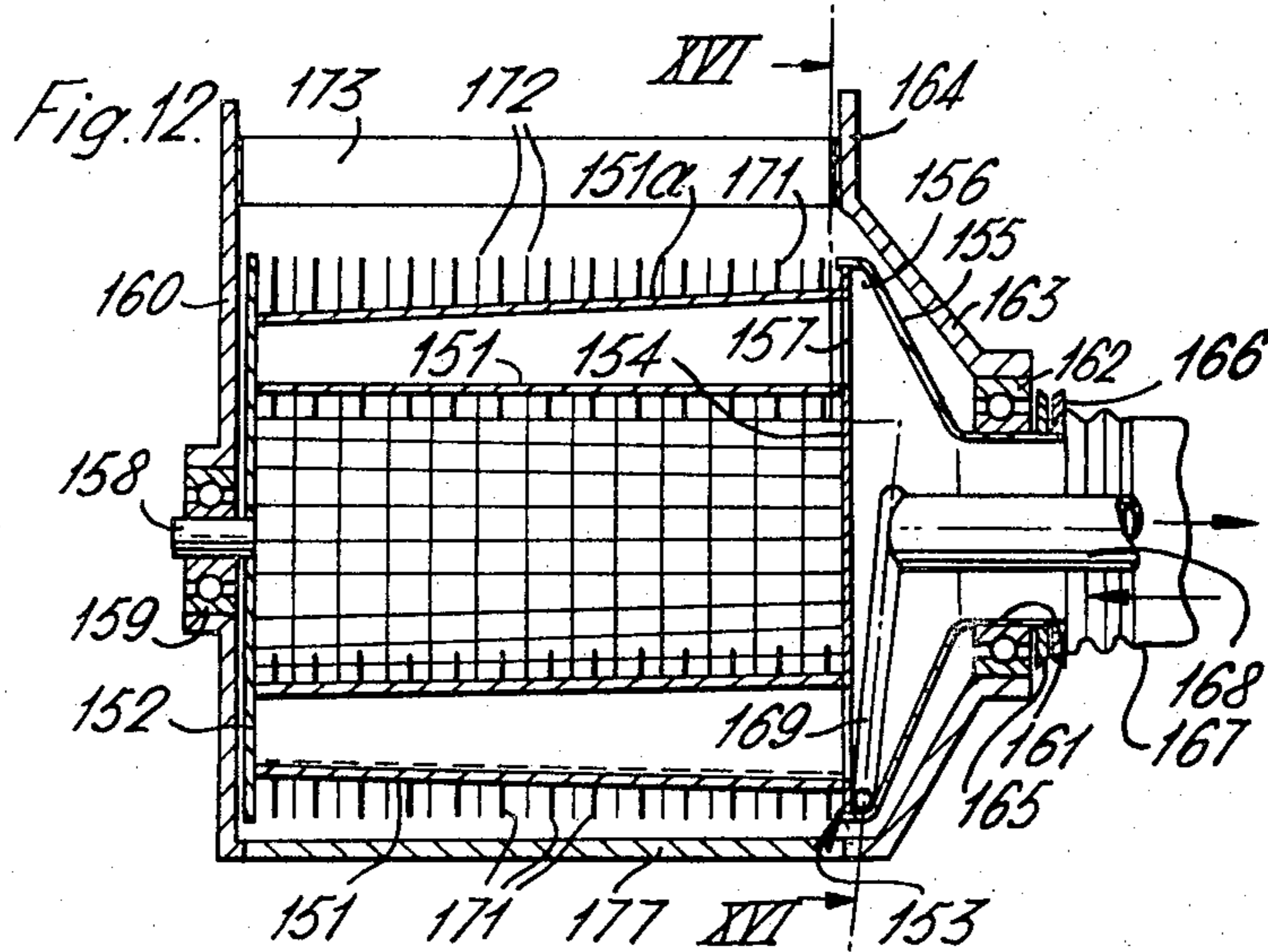
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ROTARY HEAT EXCHANGERS

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Original application Jan. 2, 1963, Ser. No. 249,064, now
Patent No. 3,260,306, dated July 12, 1966. Divided and
this application May 26, 1966, Ser. No. 553,225

U.S. Cl. 165—89

9 Claims

Int. Cl. F28f 5/02; F28b 9/04; F28d 11/08

ABSTRACT OF THE DISCLOSURE

A rotary heat exchanger having a rotor comprising a plurality of longitudinally extending hollow fluid passages and manifold means including means therein for guiding fluid into and out of said passages upon rotation of the rotor.

This invention relates to rotary heat exchangers for exchange of heat between two fluids which are kept separate: the application is a division of my copending application Ser. No. 249,064 filed Jan. 2, 1963, and now Patent No. 3,260,306, granted July 12, 1966, itself a continuation-in-part of my application Ser. No. 701,600 filed Dec. 9, 1957 and now abandoned.

A rotary heat exchanger comprises essentially a rotor having interior passages through which a first fluid stream is passed, with the second fluid flowing past the outside of the rotor.

One main object of the invention, according to a principal aspect thereof, is to provide a rotary heat exchanger wherein rotation of the rotor itself causes flow of the first fluid through said interior passages, whereby to eliminate the need for a separate pump for this purpose.

With this main object in view, in the rotary heat exchanger according to this aspect of the invention, the rotor has its interior passages spaced from the axis, and comprises manifold means providing inlet and outlet ducts coaxial with said axis and communicating with said interior passages. Non-rotating flow guide means are provided having inlet and outlet passages coaxial with said axis and co-operating with said inlet and outlet ducts for flow of fluid through the inlet passage to the inlet duct and thence through said interior passages of the rotor to the outlet duct and to the outlet passage. The manifold means and co-operating stationary flow guide means may be such that flow takes place into the rotor at one end out at the other, but preferably, as will appear, the manifold and flow guide means are at one end only of the rotor, so that flow in and flow out takes places at the same end. According to the invention, the outlet duct includes a collector space receiving fluid from the interior passages of the rotor with flow in the collector space having a radially inward component, and stationary guide vanes within the collector space to convert at least a portion of the momentum of the fluid from a circumferential to a radially inward direction. In this way use is made of the circumferential momentum imparted to the fluid to drive it out of the rotor against the centrifugal force acting, so that the rotor acts as a pump. The spaces between the guide vanes are preferably formed as diffusers, to reduce the velocity and increase the pressure of the fluid. So far as the present aspect of the invention is concerned, it is immaterial how the second fluid is induced to flow past the outside of the rotor. However, it is preferred that the rotor have blade portions enabling it to act as a fan or pump upon the second fluid. Preferably though not necessarily the rotor is such as to operate as a cross flow fan or pump, i.e. the second fluid, on rotation of the rotor, is induced to flow from an inlet region through the path of the rotating blade portions to the in-

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terior of the rotor and thence again through the path of the rotor to an outlet region. Stationary guide means may be provided to co-operate with the rotor for this purpose.

The invention has also other aspects, and includes a combination of rotary heat exchanger and liquid cooled internal combustion engine wherein the engine cooling liquid is circulated through the rotary heat exchanger. Preferably, though not necessarily, the rotor operates in cross flow manner (as above explained). The rotary heat exchanger may be driven by the engine and the air which it circulates may be used to cool the exhaust manifold. Thus if the engine has a bank of cylinders in line, the exhaust manifold will run parallel to the bank. The rotor may extend over the length of the manifold, with its axis parallel to the engine crankshaft. The heated air may be used for warming a vehicle passenger compartment, or it may be discharged to the exterior in a manner to reduce air drag on the vehicle at speed, as will be described.

Other aspects of the invention relate to the incorporation of a rotary heat exchanger in a steam condenser, and in a refrigerator.

Embodiments of the invention will now be described by way of example with reference to the accompanying 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 3a 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;

FIGURES 5 and 6 are respectively a cross-sectional and a plan view of an internal combustion engine combined with a rotary heat exchanger as shown in FIGURES 1 to 4;

FIGURE 7 is a graph illustrating the operation of the combination of FIGURES 5 and 6;

FIGURE 8 is a vertical section showing a further internal combustion engine-rotary heat exchanger combination;

FIGURES 9 and 10 are respectively a side elevation, partly sectioned, and a transverse section on the line XII—XII of a rotary hat exchanger and ducting adapted for ventilating or heating a vehicle;

FIGURE 11 is a plan view partly cut away and sectioned of the rear compartment of a rear-engined vehicle, showing an air arrangement of internal combustion engine-rotary heat exchanger combination to effect boundary layer suction;

FIGURES 12 and 13 are respectively a longitudinal section and a transverse section of a steam condenser employing a rotary heat exchanger, the latter section being taken on the line XVI—XVI of FIGURE 12 with a cut-away showing blades;

FIGURE 14 is a transverse section of a modified form of steam condenser;

FIGURES 15 and 16 are broken cross-sections showing fin surfaces alternative to those of FIGURES 1 to 4, and

FIGURES 17 and 18 are respectively a longitudinal section and a transverse section of a refrigerator in-

corporating a rotary heat exchanger, the latter section being taken on the line XX—XX of FIGURE 17, certain guide means are omitted and a broken section forming part of FIGURE 17 illustrating a blade profile.

Referring now to FIGURES 1 to 3a, 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 16a, 16b, 16c, 16d, stamped from heat conducting material such as sheet copper, for example, of .050" 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 16a, 16b, 16c, 16d 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 16a, 16b and 16c, 16d 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 16a, 16b, 16c, 16d 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 length-extending hollow blades designated 25 as best seen in FIGURE 4. The pairs of rings 16a, 16b, and 16c, 16d 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 26a 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 16b. 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 25a 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 25b 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 25b 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 communicating 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 40 is subjected to centrifugal action and gains kinetic energy before entering blades 25a. The liquid passes along the blades 25a, which form a "flow" system and are labelled "+" in the drawing, but at each fin 26 some of the liquid flows into the channels 26a thereof and thence into the blades 25b, which form the "return" system and are labelled "-" in the drawing. The liquid entering interior space 33 from blades 25b encounters the vanes 38. The vanes 38, acting rather like stationary turbine blades, receive the liquid with minimum shock and change its direction from circumferential to radial, with little loss of momentum. In addition the flow velocity decreases and pressure increases, the spaces 38a between the vanes 38 acting as diffusers. The liquid finally issues through tube 34 and outlet passage 8. By reason of the vanes 38 in the space 33 the rotor functions as a pump to circulate liquid.

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.

The end supports 5, 6 are interconnected by a pair of sheet metal guide walls 41, 42 (FIGURE 4) having end edges received in and located by grooves 43 in end walls 43a integral with and forming extensions of the supports. At their lines 41a, 42a of nearest approach to the rotor 1, which are diametrically 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 47, the wall portions 44, 45 defining an angle of a little over 90°. The wall 42 diverges steadily from the rotor going from its line 42a of nearest approach and defines with the wall portion 45 an outlet 48.

It is to be noted that the blades 25 are profiled to air-foil 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.

The parent application, Ser. No. 249,064, describes in detail how flow takes place in the vortex.

While a preferred vortex flow pattern has been described, 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.

FIGURES 5 and 6 show one way in which the rotary heat exchanger can be used. A four-cylinder water-cooled gasoline engine designated generally 80 comprises the usual block 81 providing cylinders 82 and water cooling passages 83 surrounding the cylinders, while an exhaust manifold 84 running horizontally lengthwise of the block is secured to one side thereof near the top. As thus far described, the engine is of well-known type and no further description of its conventional features will be required. The engine departs from the conventional in utilizing the rotary heat exchanger previously described for removing heat from the engine cooling water. The end supports 5, 6 of the heat exchanger (see FIGURE 1) are secured at opposite ends of the block 81 on the same side as the manifold 84 so as to mount the rotor 1 directly below it for rotation about a horizontal axis, and the air flow guide walls 41, 42 (of which only the latter is shown in FIGURE 5) are arranged so that the rotor discharges air upwards past the manifold over its whole length. The pulley 15 of the heat exchanger is driven by a belt 85 from a pulley 86 on the engine crankshaft 87, the direction of rotation of the rotor being shown by the arrow 88. The belt 85 may be driven over the usual dynamo (not shown).

The water cooling passages 83 are connected with the inlet and outlet passages 7, 8 in the end support 5 of the heat exchanger. A horizontal wall 89 is disposed in the water cooling passages 83 in approximate alignment with the divisions 90 (see FIGURE 1) between the inlet and outlet passages 7, 8 to ensure proper distribution of cooling water within the block 81, the direction of flow being indicated by arrows.

A control flap 91 (shown only in FIGURE 5) is pivoted at 92 to extensions of the heat exchanger end supports 5, 6 and is movable between the open position shown and a closed position shown dotted in which it closely overlies that arc of the rotor 1 where the fastest air flow enters (see FIGURE 4). Although the control flap 91, when closed, does not cover the whole arc of entry to the rotor, it nevertheless effects virtually complete throttling by reason of its effect on the fast stream tubes. A lever 93 rigid with the flap 91 is movable to and fro by a push rod 94 actuated by a bellows 95 disposed in the engine (e.g. in association with the oil circulation) thereby to expand and contract in response to changes of engine temperature. The arrangement is such that increase of engine temperature moves the control flap 91 from the closed to the open temperature thus progressively reducing throttling of air flow through the heat exchanger rotor 1.

In the graph of FIGURE 7, curves 100, 101 and 102 represent respectively the brake horsepower required to drive the heat exchanger rotor 1, the total air pressure at the output side of the rotor, and the efficiency of the rotor considered purely as a blower, all plotted against a fixed scale of throughput of air through the rotor. These curves indicate that when the throughput is low, which is the case when control flap 91 occupies its dotted position, the brake horsepower absorbed by the rotor is small, little more in fact than is required to overcome friction losses. When the control flap 91 is in open or partially open position, the throughput will increase to a value determined by the operating speed of the engine such that at a given engine speed the throughput will depend on the degree of opening of the flap 91. The optimum operating conditions for the engine correspond to a throughput indicated by the line 103, where the efficiency of the rotor passes through a maximum. In the normal operating range, the brake horsepower taken is substantially linearly related to the throughput and hence to the cooling. If the engine speed becomes excessive, the horsepower increases very steeply so that the rotor tends to act as a brake.

From a study of FIGURES 5 and 6 in conjunction with FIGURE 7, it will be understood that on starting the engine 80 from cold, the control flap 91 is closed so that, though the heat exchanger rotor 1 is driven and operates as a pump to circulate water through the cooling passages 83 of the block 82, little air passes the rotor and the water rapidly heats. As the engine approaches its operating temperature, the control flap 91 opens to some extent so that air is caused to pass through the rotor 1 and cooling of the circulating water begins. The rotor throughput discharges over the exhaust manifold 84 and helps to keep its temperature down. During operation, the control flap 91 reacts to engine temperature to change the rotor throughput in a direction to bring that temperature to a predetermined value. Indirectly the flap reacts also engine speed, since this also affects throughput and thus cooling. The engine and rotary heat exchanger are designed so that sufficient cooling is just obtained with the flap 91 fully open under high-load, low-speed conditions; that is, the worst possible conditions from the cooling point of view. Under other conditions, adequate cooling is obtained with the flap partially closed. Under all normal conditions the brake horsepower used to drive the rotor depends on the amount of cooling needed as will be understood from FIGURE 7. However, if the engine races, the rotor will act as a brake.

The engine-rotary heat exchanger combination can be used in a vehicle to obviate the need for the usual radiator, thus enabling the exterior form of the vehicle to be designed to produce less drag. The combination mentioned has the advantage over a radiator that since the latter has to be designed for the worst possible conditions; that is, high engine output at low vehicle speed, and its power consumption varies as a power of the vehicle speed, it must under normal conditions, and especially at high speed, waste a considerable amount of power. By contrast with the combination of engine and rotary heat exchanger above described, the power consumed is always adjusted to the cooling requirements, as has been pointed out.

It is to be understood that the curves of FIGURE 10 are typical of rotary heat exchangers according to the invention and not restricted to the embodiment of FIGURES 5 and 6. It is to be further understood that there are many ways of throttling besides using a control flap such as 91 in FIGURE 5. Several other ways are described below.

FIGURE 8 shows one alternative method of throttling. This figure illustrates an engine similar to the engine 80 of FIGURE 5 and designated 80' and having an exhaust manifold 84'. A rotary heat exchanger similar to that in FIGURES 1 to 4 is mounted on the engine block underneath the manifold 84' in the same way as described with reference to FIGURE 5 and the engine cooling water is circulated through the rotary heat exchanger also as described previously. In the figure, only the rotor 1 and guide walls 41, 42 are shown and in the air outlet defined by walls 41 and 42, there is mounted a butterfly valve designated generally 108 and capable of pivoting on an axis 109 running parallel to the axis of rotor 1. In the vertical position shown, the valve 108 presents negligible obstruction to air flow upward against manifold 84'. In the dotted position of the valve 108, designated 108', the throughput of the rotor 1 is completely throttled. The throttling can be varied by moving the valve 108 to intermediate positions.

The warm air produced by an engine and rotary heat exchanger combination such as shown in FIGURES 5 and 6 can be used to heat the interior of a vehicle, as shown in FIGURES 9 and 10. In these figures, details of the combination not directly relevant to the air-supply arrangements illustrated are omitted since they can readily be supplied by those skilled in the art by reference to previous figures.

FIGURES 9 and 10 illustrate the rotor 1, end supports

5 and 6, and guide walls 41, 42 of the heat exchanger, which parts are substantially the same as in FIGURE 1. The end supports 5, 6 are mounted on the block of the engine (not shown) as in FIGURES 5 and 6. Unlike previous constructions there is mounted for rotation with the rotor 1 an impeller 110 comprising a series of blades 111 running parallel to the axis of rotation and arranged in a ring thereabout between end discs 112 and co-operating with guide walls (not shown) to induce a flow of air twice through the path of the rotating blades in a direction transverse to the axis. The impeller 110 and guide walls have the same characteristics as regards air flow as are described with reference to FIGURE 4. The orientation of the guide walls for rotor 1 and impeller 110 respectively is similar, so that both rotor and impeller discharge generally upwardly. The end disc 112 of the impeller 110 away from the support 6 carries a pulley 113 whereby impeller 110 and rotor 1 are rotated by a belt 114 from the engine crankshaft (not shown).

An air collector 115 in the shape of a diffuser with a narrow slit-like inlet opening 115a of the same axial length as the impeller 110 is mounted on a parallelogram linkage designated generally 116 for movement parallel to the rotational axis under the control of an actuating lever 117 connected to an extension of one linkage member 118. The air collector 115 leads into a short length of ducting 119 running parallel to the axis and capable of angular movement as illustrated by the full and dotted lines in FIGURE 10. The end of the ducting 119 is angled towards the axis and connected to large diameter flexible tubing 120 leading to the interior of the vehicle.

In operation, the rotor 1 produces a stream of warm air and the impeller 110 a stream of cold air. Both these air streams discharge in the same upward direction, and both include a narrowly defined group of stream tubes (indicated by the tube MF) which have a velocity far above the average taken across the whole exit duct 48.

With the air collector 115 in its upper, full-line, position, no air will enter the duct 119. A small angular movement of the air collector 115 to the dotted position shown at 115' brings the opening 115a into alignment with the high-velocity flow tube MF and in its passage through the air collector 115, a good part of the velocity energy of the air stream is transformed into pressure. The position 115' gives the maximum quantity of air to the vehicle interior. A further angular movement of the air collector 115 to the position 115'' brings the inlet opening 115a out of alignment with the high-velocity flow tube MF and positions it to receive only slower flow tubes, so that the quantity of air reaching the interior of the vehicle is diminished. Reference to the discussion relative to FIGURES 9 and 10 will indicate that angular movement of the air collector 115 across the air stream from the position 115' will progressively reduce the amount of air delivered to the vehicle interior.

With the air collector 115 positioned opposite the impeller 110 only cold air will be delivered to the vehicle. If the lever 117 is actuated to shift the collector 115 longitudinally to the right (as seen in FIGURE 9) until it is wholly opposite the heat exchanger root 1, then only warm air will be delivered. In intermediate positions, the collector 115 will be opposite both the impeller 110 and the rotor 1 and will receive some hot and some cold air.

It will be seen that the lever 117 (which can be extended, for example, to the vehicle dashboard) is effective as a temperature control, while angular movement of the duct 119 and air collector operates to control the volume of the air delivered to the vehicle interior. The flexible tubing accommodates the various movements of the duct 119 and collector 115.

The air passing through a rotary heat exchanger according to the invention can be used, when the heat exchanger is combined with the engine of a rear-engined vehicle, to reduce the aerodynamic drag on the vehicle when it is travelling at speed. There is normally a region

of break-away and violent random eddying at the rear of a fast moving vehicle which is responsible for much of the so-called air resistance thereon. In aircraft similar conditions can occur near the trailing edge of an airfoil at the suction side, and it is known that the break-away can be controlled to some extent by applying suction to the surface where it tends to occur or by jets directed along the surface. The present invention according to one aspect thereof, makes use of the air blown through the rotary heat exchanger for the application of similar measures to the rear of a vehicle and two different schemes of such are illustrated in FIGURE 11.

In FIGURE 11, the vehicle engine is shown at 130 and an associated heat exchanger rotor at 131. The rotary heat exchanger is in each case similar in essence to that described with reference to FIGURES 1 to 4 and associated with engine 130 as described with reference to FIGURES 5 and 6.

In FIGURE 11, the heat exchanger rotor 131 takes air from a suction chamber 138 containing the motor 130 through louvres 139 in the top of this chamber. The suction chamber 138 is divided from a pressure chamber 140 by a bulk-head 140a the rotor 131 delivering to the pressure chamber. The rear of the vehicle, designated 141, provides one wall of the pressure chamber 139 and is formed with symmetrically disposed vertical slit-like nozzles 142 inwardly directed over a central portion 143 of the vehicle rear wall. The pressurized air within the chamber 142 issues through the nozzles 142 in the form of relatively long narrow jets over the surface central wall portion 143 and tends to stabilize flow against break-away over at least the rounded rear corners 144 of the vehicle.

The rotary heat exchangers so far described in detail have all been designed for the cooling of hot water flowing through the blades by an air flow past them. It will be evident that other liquids could be cooled in the same way and alternatively water or other liquids could, if desired, be heated by a flow of hot air, such as furnace gases. A further possibility is for a flow of a gas or vapor through the blades to be heated or cooled by a flow of liquid past them. A rotary heat exchanger according to the invention can also be used to effect a change of phase, as will be shown by the embodiments of the invention next to be described.

FIGURES 12 and 13 illustrate a condenser which includes a rotor designated generally 150 mounted for rotation about a vertical axis and comprising a series of hollow blades 151 of airfoil profile arranged in a ring about the axis and extending generally parallel thereto between upper and lower end members 152, 153. The upper end member 152 is a plain flat disc to which the blades 151 abut and to which they are secured by welding or brazing depending on the temperatures at which the condenser is to be used. The lower end member 153 comprises a flat disc 154 to which is secured (also by welding or brazing) a dished annular support element 155. The disc 154 and support element 155 are secured at their peripheries, which lie radially outward of the outer envelope of the blades 151, and which define an interior annular collection space 156. The blades 151 are secured to the disc 154 in the same way as to the end member 152 except that the disc 154 is apertured at 157 over the whole cross section of each blade for unobstructed communication between the space 156 and the interior of the blade while the member 152 closes off the upper end thereof. A stub shaft 158 is secured axially to the upper rotor end member 152 and is journaled in a bearing 159 mounted upon a stout stationary end wall 160 extending in a radial plane and closely overlying the end support. The support element 155 of the lower rotor end member 153 is formed with a downwardly projecting large diameter tubular extension 161 journaled in a bearing 162 mounted in an outwardly belled portion 163 of an otherwise planar stout stationary end wall 164 which is in approximate radial

alignment with the disc 154, the belled portion 163 accommodating the lower end members 153. The tubular extension 161 projects beyond the bearing 162 and has welded thereto a radial flange 165 co-operating with a similar stationary flange 166 which is flexibly supported by a stationary bellows 167 and urged thereby against the flange 165 to form a rotating seal. A stationary pipe 168 extends coaxially within the support element 155 and carries a curved stationary scoop 169 presenting an inlet opening 170 in a generally tangential direction within the collecting space 156.

It is to be noted that the blades 151 differ from those denoted 25 in previous figures in that each is an integral tubular structure instead of being built up from a multiplicity of flanges and in that their outer edges 151 lie on an upwardly tapering frusto-conical envelope. The semi-vertical angle of the cone is, however, very small, as FIGURE 15 shows, and therefore from the point of view of air flow, the rotor can still be regarded as essentially cylindrical. The inner blade envelope is cylindrical as before.

The blades 151 extend through a series of interleaved relatively thick rings 171 and relatively thin discs 172 lying in axially spaced radial planes. These rings 171 and discs 172 rigidify the rotor construction, reinforce the blades 151 against the effect of pressure and centrifugal force, and extend the area for dissipation of heat into the air traversing the rotor.

The stationary end walls 160, 164 are inter-connected by a guide body 173 extending the length of the rotor and defining a guide surface 174 converging therewith in the direction of rotor rotation indicated by the arrow 175. Also interconnecting the end walls 160, 164 is a stout rod 176 diametrically opposite the guide body 173 and serving as a pivot for a guide wall 177 extending between the end walls and having one end curled over the rod. In its open position, the guide wall 177 diverges steadily from the rotor 150 and defines with the guide body 173 an outlet 178 for air which has passed the rotor. The guide wall 177 can however be pivoted about the rod 176 to the position shown dotted at 177' where its end remote from the rod lies close against the guide body 173 and effectively closes the outlet 178. It will be understood that in positions of the guide wall 177 intermediate the full and dotted positions shown the throughput of the rotor will be throttled to a greater or lesser extent.

As regards air flow through the condenser of FIGURES 12 and 13, the rotor 150 and guide surfaces 174, 177 function (with the latter in open positions and the rotor rotating in the direction 175) in the same manner as the corresponding parts discussed with reference to FIGURES 4. The throughput of air at a given rotor speed, and thus the cooling applied to the condenser, can be varied or derived by movement of wall 177. This can be effected manually or by a thermostat.

In operation steam is led through the annular space between bellows 167 and pipe 168 into the interior of the rotor end member 153 and thence into the blades 151. There the steam condenses and the resulting water is thrown by centrifugal force towards the outer edges 151a of the blades. The water is then subjected both to gravity and to centrifugal force due to the frusto-conical envelope of the outer edges 151a of the blades 151 and flows down along these outer edges until it collects in the annular space 156 whence it is removed by the scoop 169 and discharged through the pipe 168. Velocity energy in the water in the space 156 is converted into pressure energy in the scoop 169.

By reason of the air flow over the blades heat transfer is better than in constructions, whether stationary or moving, where a relatively stationary insulating boundary layer of air builds up about the parts to be cooled. It is to be noted that due to the rotation of the blades there will moreover be a negligible layer of water over the blade interiors except over a comparatively small area at

the outer edges of the blades. The insulating water layer that builds up in the tubes of a conventional stationary condenser, and can attain a thickness of several tenths of a millimetre, will accordingly be virtually absent and the conditions for heat transfer from steam to metal will be correspondingly improved. The water layer within the blades will be thinner the higher the speed of rotation; in general a thickness of some hundredths of a millimetre will not be exceeded while at high speeds the water layer thickness may be reduced to only a few microns.

FIGURE 14 illustrates a construction similar in general to that of FIGURES 12 and 13; similar parts are given similar reference numerals and will need no further description. The FIGURE 14 construction differs from that just described in that the guide body 173 is formed at its side remote from the rotor 150 with a diffuser 179 having a narrow slit-like inlet 179a which receives warm air whether the wall 177 is open or closed. The diffuser 179 can be used to supply warm air for some heating purpose and will be substantially unaffected by regulation of the rotor throughput. Naturally in the condenser of FIGURES 12 and 13, the whole throughput could be used for some heating purpose, but amount of the warm air supplied would depend on the setting of wall 177 and could be reduced to zero, whereby this can not happen in the FIGURE 14 arrangement.

FIGURES 15 and 16 show two ways in which a heat exchanging surface can be rendered more effective. As shown in FIGURE 15 ridges 181 can be rolled in a metal sheet 182. FIGURE 16 shows a portion of a corrugated sheet 183. In the construction of FIGURES 12 and 13, the rings 171 and discs 172, as well as the interior surfaces of the blades 151 can be formed as in either of FIGURES 15 and 16. Surfaces having projections as there shown are capable of other applications within the scope of the present invention and can, for example, be used as exterior surfaces in contact with moving air where the projections tend to increase the degree of turbulence of the boundary layer.

FIGURES 17 and 18 illustrate a refrigerator comprising an evaporator coil 190 and a compressor unit designated generally 191 which latter includes a fixed vertical hollow shaft 192 supported on a mounting member 193 at one end and carrying the internal stator 194 of an electric motor designated generally 195, and an eccentric disc 196 forming part of a rotating disc compressor shown generally at 197. A cylindrical casing 198 is rotatably mounted on the shaft 192, one of the bearings being shown at 199. The casing 198 encloses the motor 195 and compressor 197 and carries within it the external rotor 200 of the motor. The casing 198 also mounts a slider 201 forming part of the compressor 197, which is radially reciprocable within guides 202 provided by an inward extension of the casing, and which is urged inwardly by a compression spring 203 against the stationary disc 196. The casing 198 further provides, supported on a transverse wall 204, a concentric cylindrical housing 205 enclosing the eccentric disc 196 and defining the working chamber 206 of the pump 197, the slider 201 extending into this chamber through a slot in the housing. A tubular extension 207 of the pump housing 205 embraces the fixed shaft 192 to seal the pump chamber 206 against leakage therefrom upwardly about the shaft and provide a steadying bearing thereon.

A tubular sealing element 208 extends upward from the wall 204 of the casing 198 into a deep recess 209 in the shaft 192 so as to provide a labyrinth seal against leakage downwardly from the pump chambers.

The interior of the casing 198 forms a suction region 210 drawing refrigerant in vapor form from the evaporator coil 190 through the hollow shaft 192 and passages 211 therein. Refrigerant enters pump chamber 206 from the suction region 210, through a hole 212 in one side of the pump housing 205 and leaves the chamber through a hole 213 in the transverse wall 204.

The rotating casing 198 carries, extending downwardly from the transverse wall 204, the rotor designated generally 215 of a rotary heat exchanger. This rotor 215 comprises a series of hollow blades 216 arranged in a ring and extending parallel to the rotational axis between upper and lower end members 217, 218 in the form of hollow discs. The blades 216 have a profile of air foil section as shown by the broken section forming part of FIGURE 19. The blades 216 include within their profile tubes 219 whose interior communicates with the spaces within the end members 217, 218. The blades extend through a series of flat rings 220 arranged in spaced radial planes and, as in the condenser of FIGURES 12 and 13, these rings rigidify the construction and extend the area for dissipation of heat to the air passing through the rotor. The end member 217 forms a unit with the transverse wall 204 of the casing 198. The end member 218 has its interior in communication with a relatively narrow bore tube 221 extending axially through the rotor 215, fixed shaft 192, and support member 193. The tube 221 co-operates with a rotating pressure seal arrangement indicated diagrammatically at 222 and communicates with the inlet end of the evaporator 190. The tube 221, as will be understood, extends through the end member 217 which is sealed about its periphery. The tube is of substantially smaller outside diameter than the interior diameter of the shaft 192 so as not to impede flow therethrough.

The rotor 215 co-operates with guide means which may be such as described with reference to FIGURES 12 and 13. These guide means are not illustrated in FIGURES 17 and 18 but can be supplied by analogy with the previous figures. The rotor 215 and guide means co-operate as previously described to induce a flow of air transversely through the rotor.

It will be understood from the foregoing that refrigerant in vapor form is drawn from evaporator 190 into space 210 and thence to the rotating vane pump 197 where it is compressed and delivered through upper rotor end member 217 to the rotating blades 216. Here the compressed vapor condenses due to the cooling air stream past the blades, and collects as a liquid in the lower rotor member 218 whence the pressure causes it to rise in the tube 221 and return to the evaporator 190.

I claim:

1. A centrifugal flow rotary heat exchanger comprising support means, a rotor mounted upon the support means for rotation in a predetermined direction about an axis the rotor having interior passages extending generally lengthwise of the rotor spaced from the axis and manifold means providing an inlet distributor space and an outlet collector space communicating with said interior passages; and non-rotating flow guide means providing inlet and outlet passages coaxial with said axis and co-operating with said inlet distributor space and said outlet collector space for flow of fluid through the inlet passage to the inlet distributor space and thence through said interior passages of the rotor to the outlet collector space and to the outlet passage, with flow in the outlet collector space having a radially inward component, and stationary guide vanes within the outlet collector space to convert at least a portion of the momentum of the fluid flowing therein from a circumferential to a radially inward direction whereby the rotor on rotation itself causes flow of said fluid through said interior passages.

2. A rotary heat exchanger as claimed in claim 1, wherein the spaces between the guide vanes form diffusers.

3. A rotary heat exchanger as claimed in claim 1, wherein the inlet distributor space and outlet collector space, and said non-rotating flow guide means, are all situated at one end of the rotor.

4. A rotary heat exchanger as claimed in claim 3, wherein the inlet passage is annular and surrounds the outlet passage, and the inlet distributor space surrounds

the outlet collector space, and the stationary guide vanes are supported on a tube providing said outlet passage and projecting into the collector space.

5. A rotary heat exchanger as claimed in claim 3, wherein the manifold means comprises an end disc, an outer annular end wall secured to the end disc and connected to an end bearing on the support means, and an inner annular end wall also secured to the end disc, the inner and outer end walls being spaced to form the inlet distributor space and the inner end wall and said disc defining the collector space.

6. A rotary heat exchanger as claimed in claim 5, wherein the inner end wall is substantially frusto-conical whereby the collector space has an axial dimension which increases in the direction of flow through said collector space.

7. A rotary heat exchanger as claimed in claim 6, wherein the tube providing the outlet passage has a belled extension within the collector space and overlying the inner end wall whereby to provide a seal therewith.

8. A condenser comprising a rotor mounted for rotation about an axis and including hollow blades and support means mounting the blades to extend generally parallel to the rotor axis and in a ring thereabout, the outer edges of the interior blade surfaces lying on a frusto-conical envelope and the blade support means including a hollow end member at the end of the envelope where the diameter of the conical envelope is greater with the interior of said end member communicating with the interiors of all the blades and defining an annular collection space of greater diameter than the maximum envelope diameter, a plurality of radially extending cooling fins attached to said blades to increase the effective heat exchange area of said blades subject to contact by a cooling fluid; fixed means to introduce a vapor phase into the hollow end member; a fixed scoop extending into the collection space to remove liquid phase therefrom and convey it out of the end member; said blades on rotation of the rotor in a given direction inducing a flow of cooling fluid through the rotor past the rotating blades, the cooling fluid during its passage through the rotor extracting heat through the material thereof from said vapor phase in said blades.

9. A condenser comprising a rotor mounted for rotation about an axis and including hollow blades and support means mounting the blades to extend generally parallel to the rotor axis and in a ring thereabout, the outer edges of the interior blade surfaces lying on an envelope spaced from the rotor axis and the blade support means including a hollow end member at the end of the envelope where the diameter of the envelope is greater with the interior of said end member communicating with the interiors of all the blades and defining an annular collection space of greater diameter than the maximum envelope diameter, a plurality of radially extending cooling fins attached to said blades to increase the effective heat exchange area of said blades subject to contact by a cooling fluid; fixed means to introduce a vapor phase into the hollow end member; a fixed scoop extending into the collection space to remove liquid phase therefrom and convey it out of the end member; said blades on rotation of the rotor in a given direction inducing a flow of cooling fluid through the rotor past the rotating blades, the cooling fluid during its passage through the rotor extracting heat through the material thereof from said vapor phase in said blades.

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