April 27, 1965 R. A. BIELEFELD 3,180,569

COOLED ROTARY PUMP

Filed May 21, 1962 5 Sheets-Sheet 1

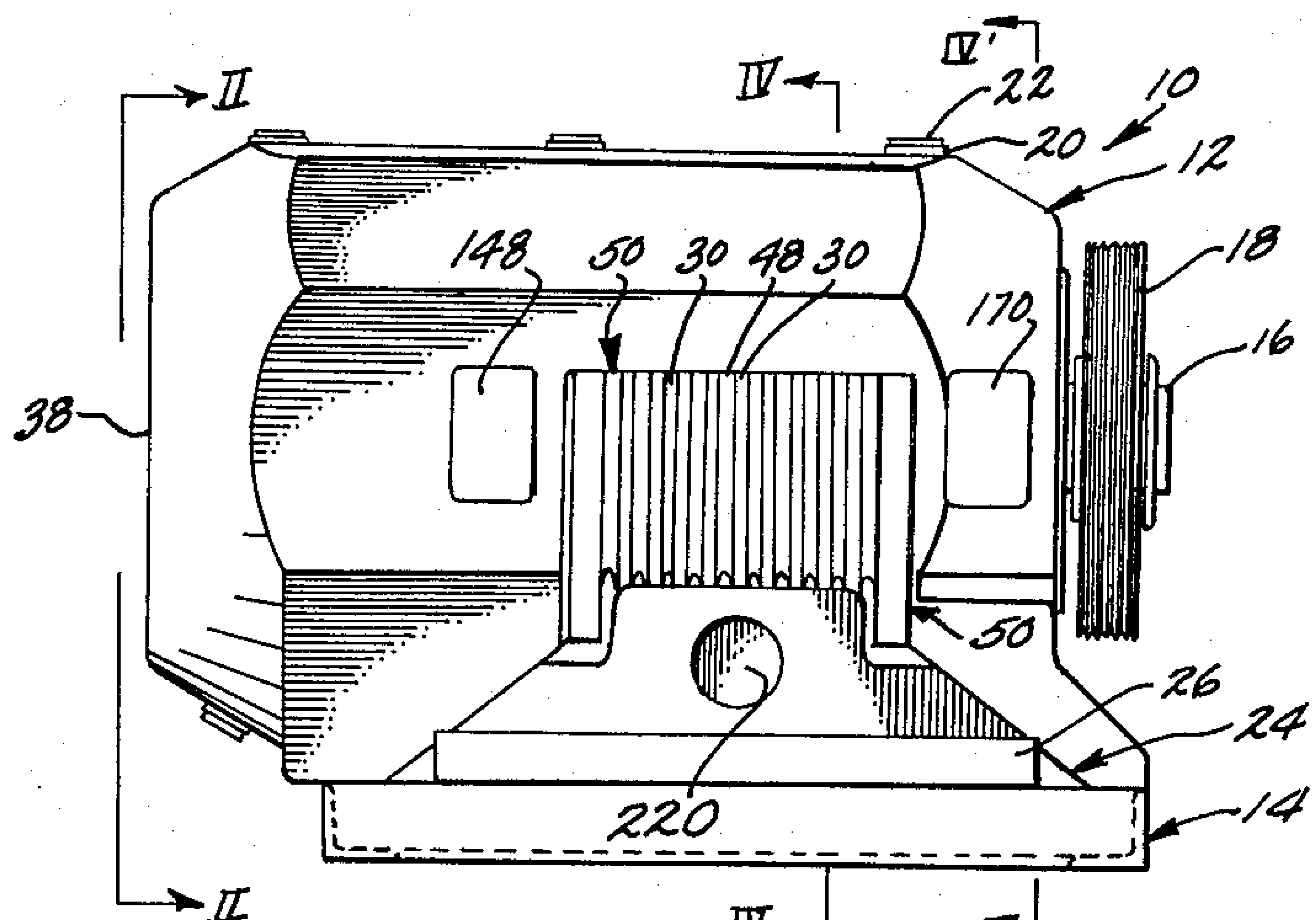

FIG. 1

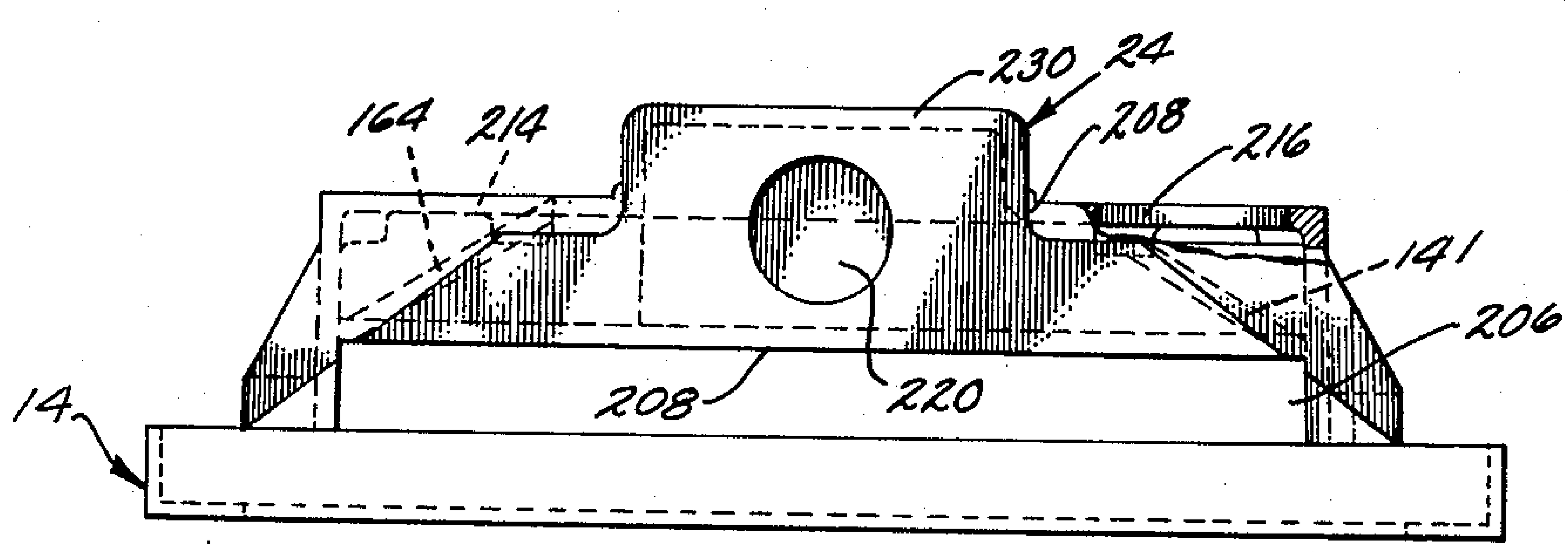

FIG. 7

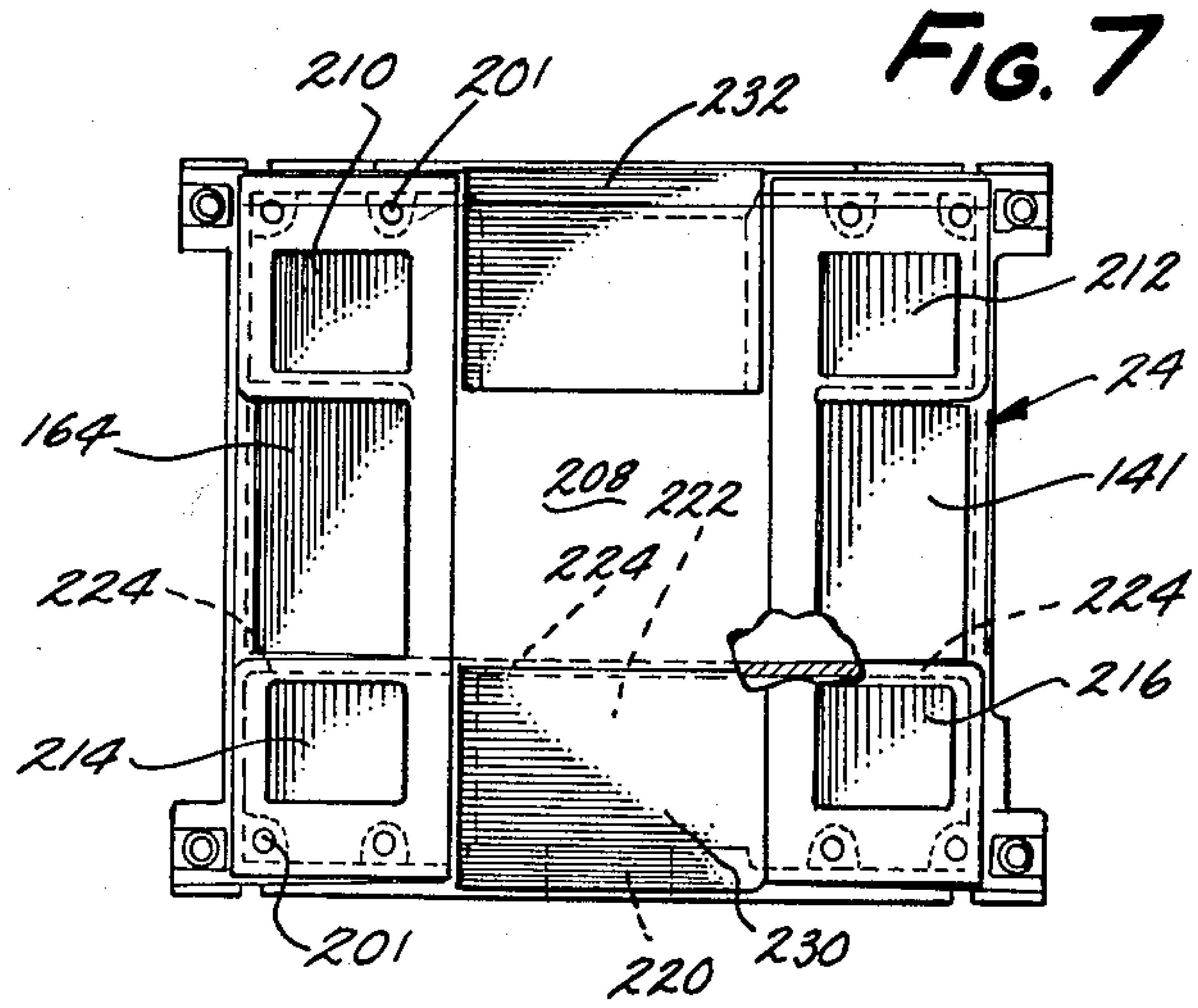

FIG. 6

INVENTOR.
ROLF A. BIELEFELD

BY Price & Heneveld

ATTORNEYS

April 27, 1965 R. A. BIELEFELD 3,180,569

COOLED ROTARY PUMP

Filed May 21, 1962 5 Sheets-Sheet 2

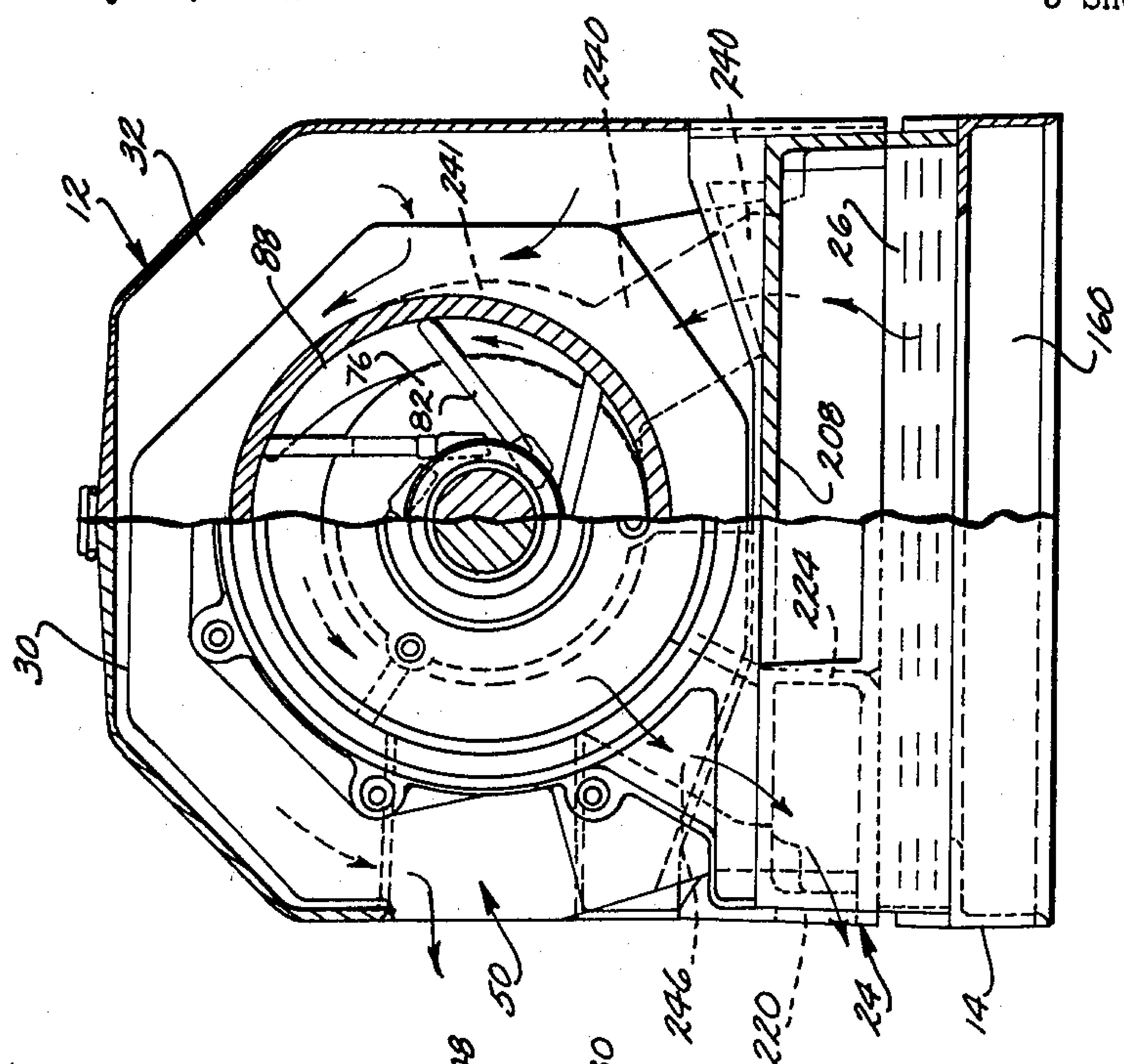

FIG. 4

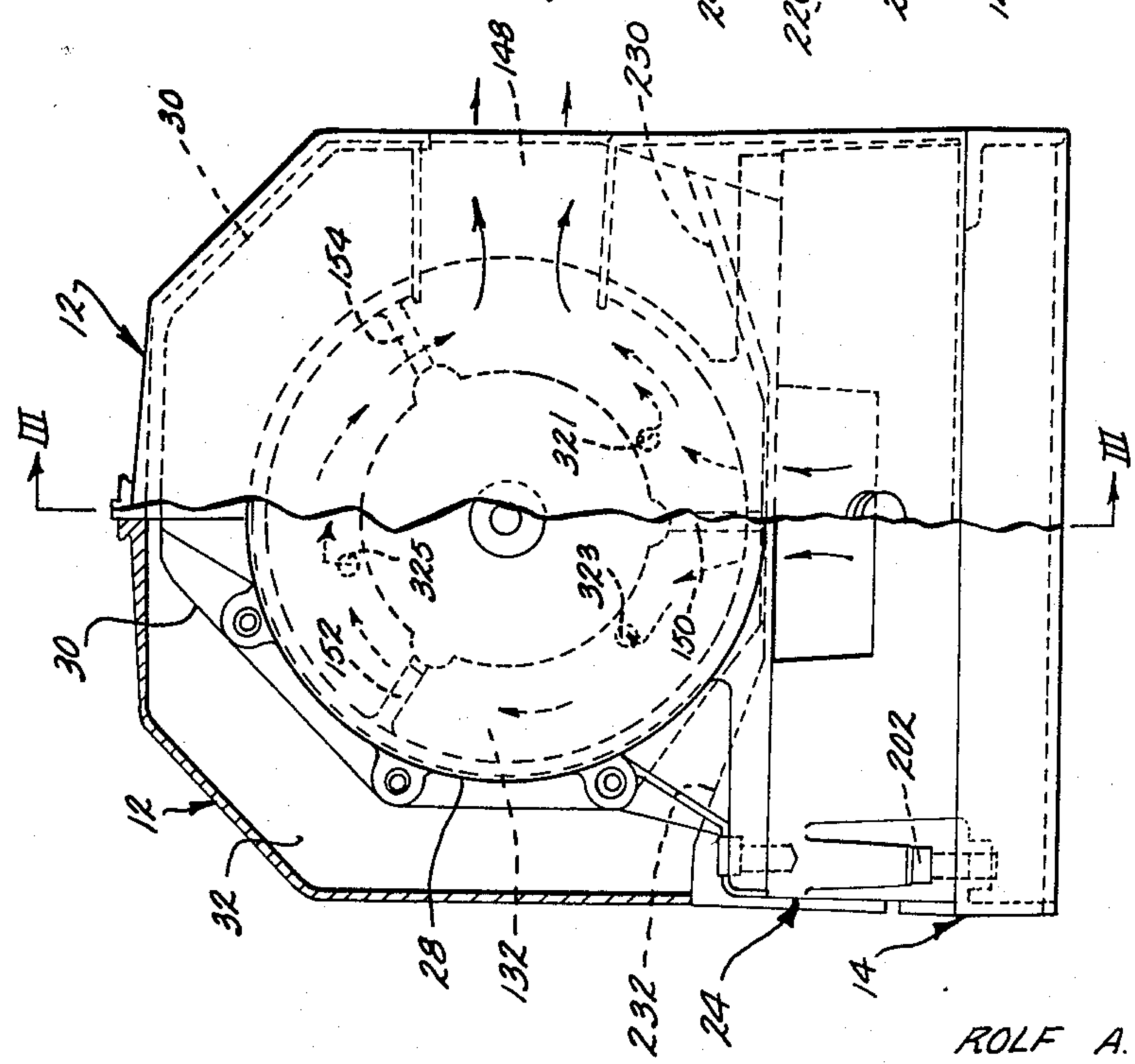

FIG. 2

INVENTOR.
ROLF A. BIELEFELD

BY Price & Heneveld

ATTORNEYS

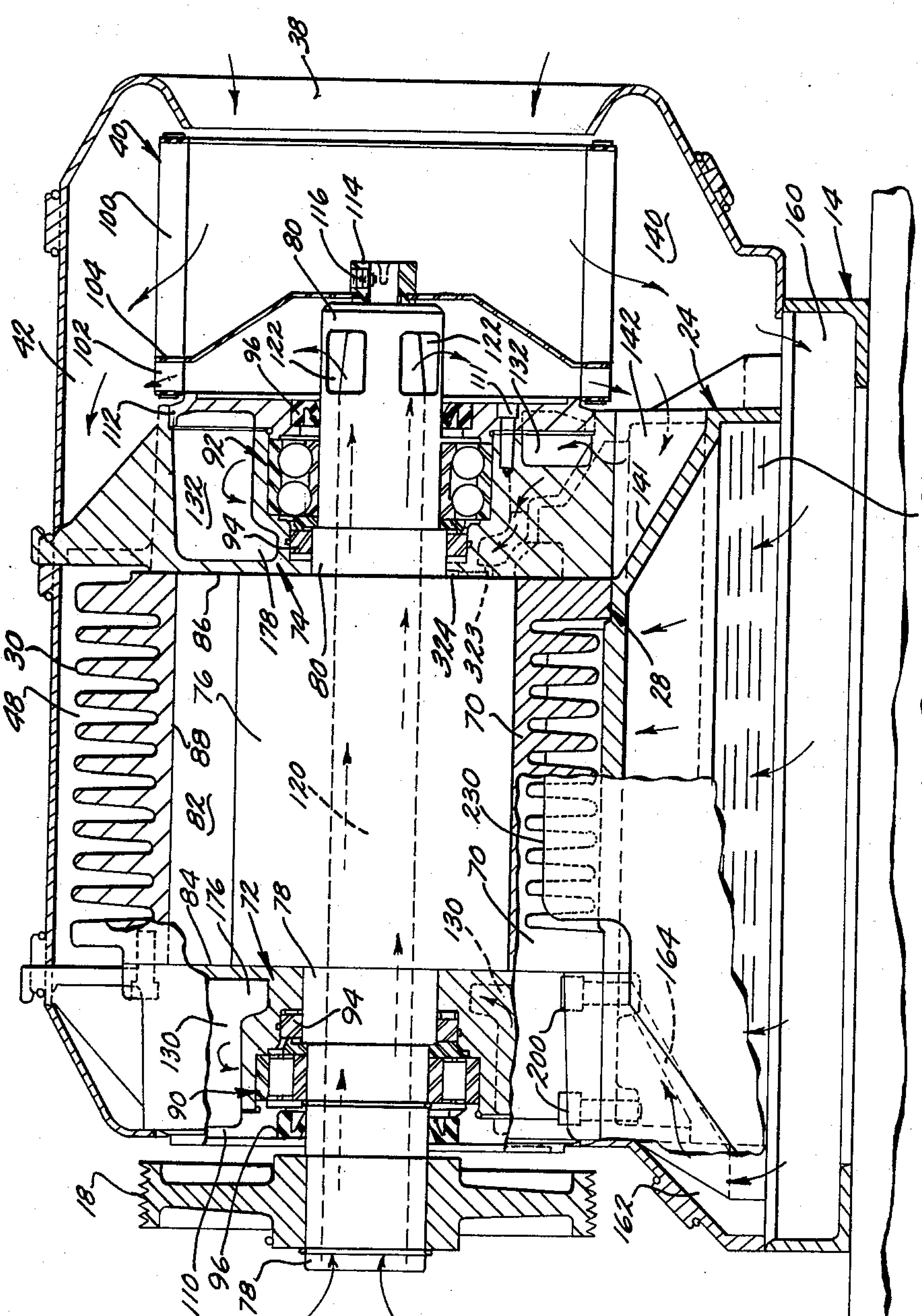

FIG. 3

INVENTOR.
ROLF A. BIELEFELD

BY Price & Heneveld

ATTORNEYS

April 27, 1965 R. A. BIELEFELD 3,180,569

COOLED ROTARY PUMP

Filed May 21, 1962 5 Sheets-Sheet 4

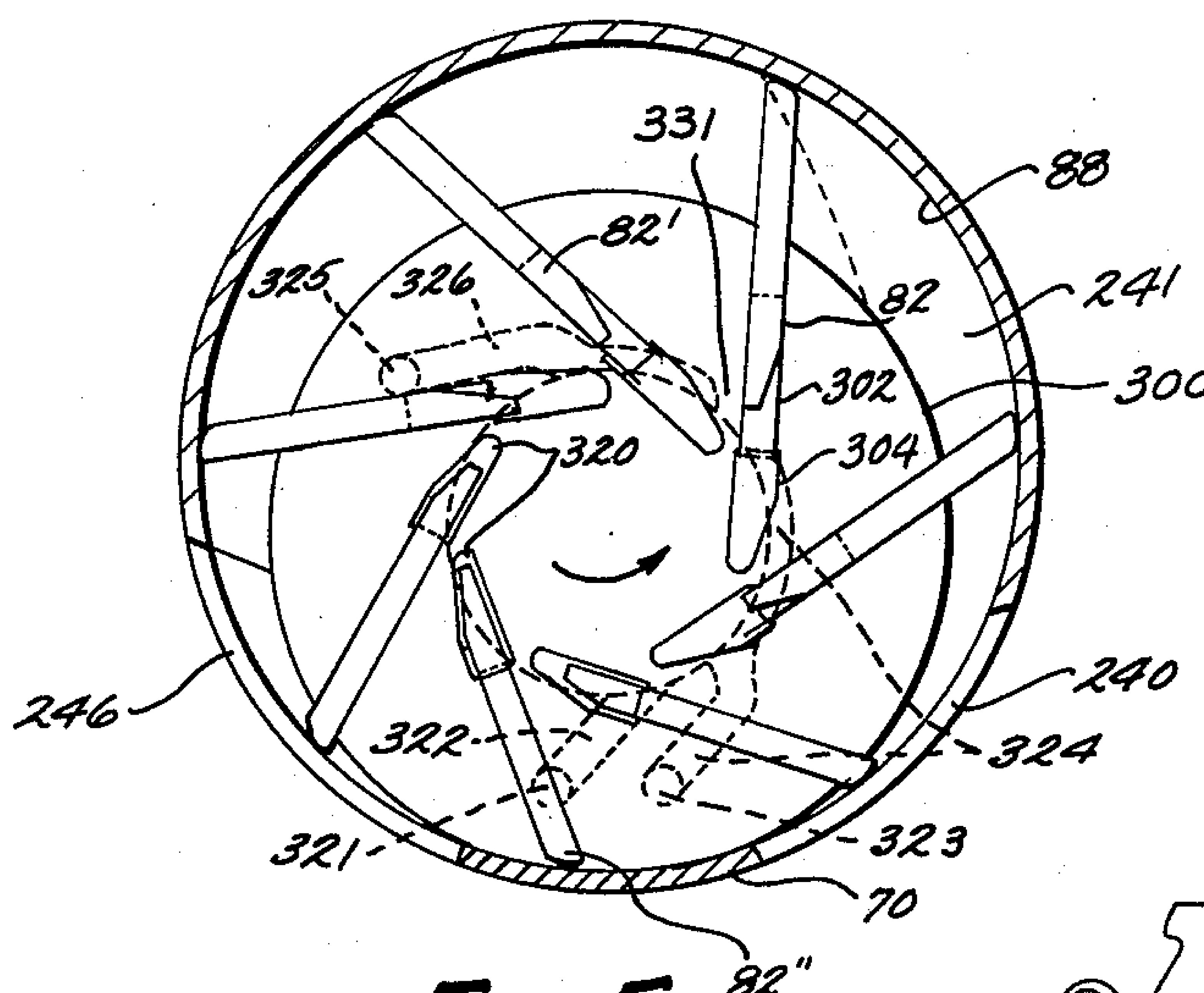

FIG. 5

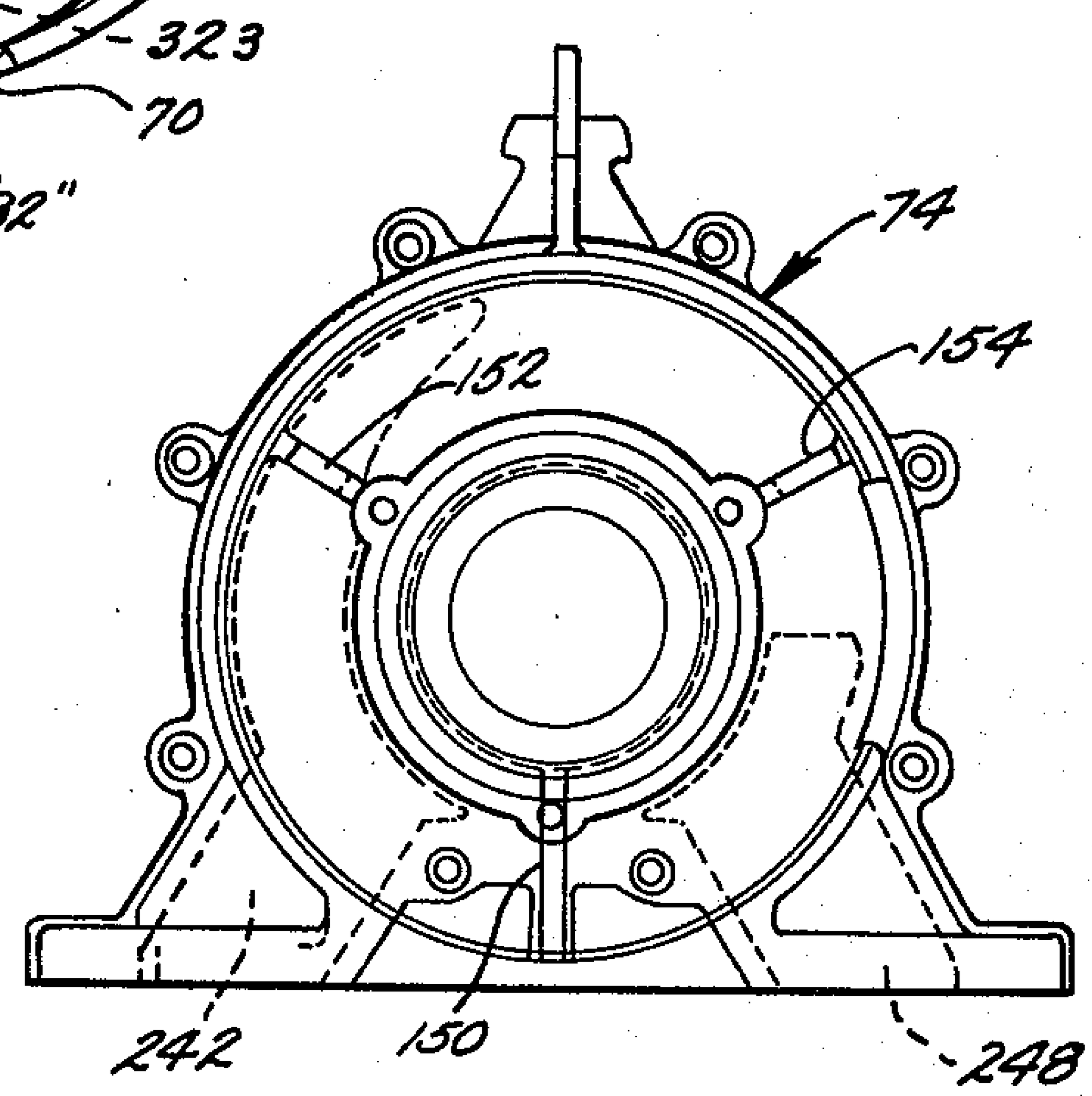

FIG. 8

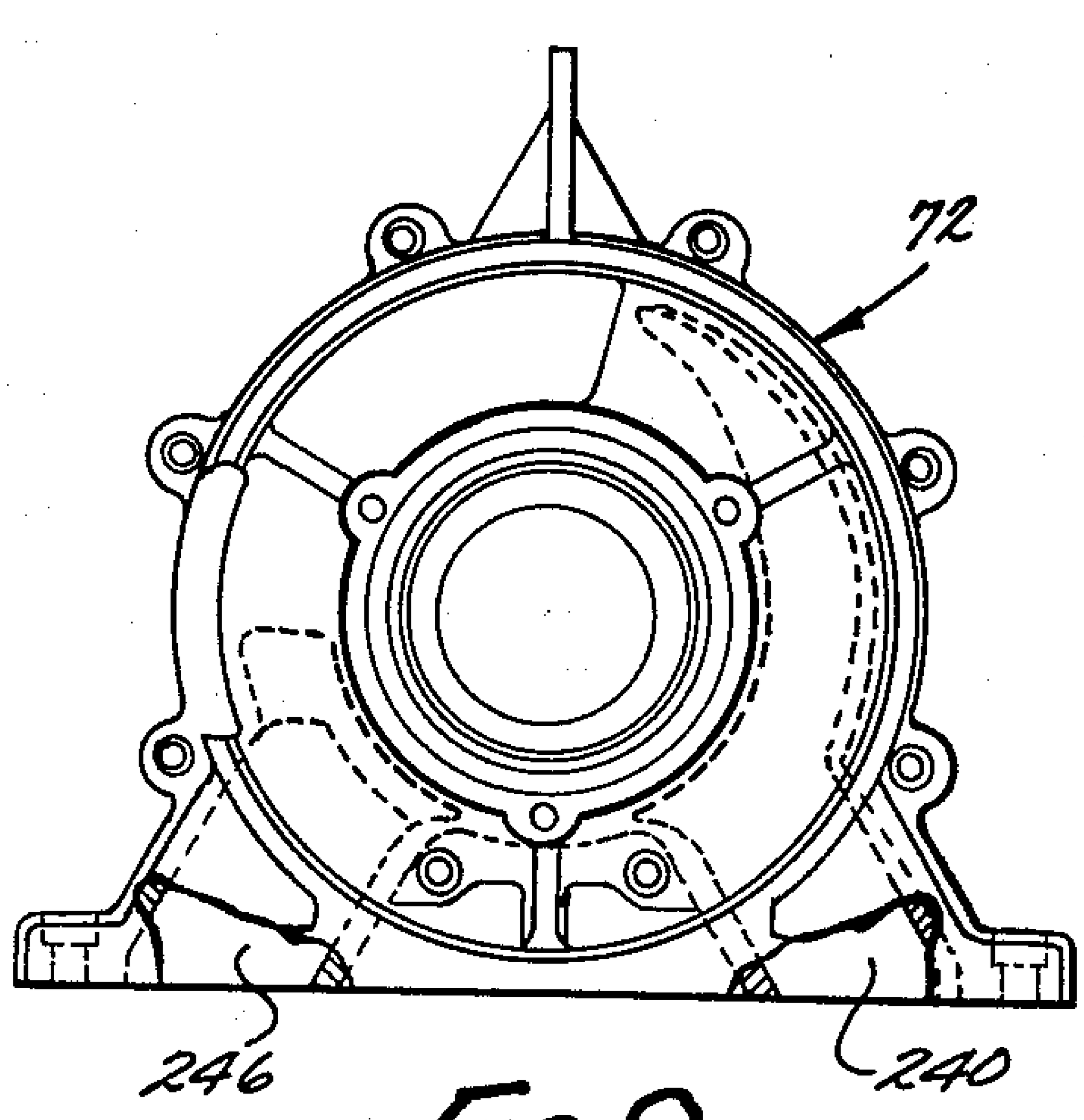

FIG. 9

INVENTOR.
ROLF A. BIELEFLD
BY Price & Heneveld
ATTORNEYS

April 27, 1965 R. A. BIELEFELD 3,180,569

COOLED ROTARY PUMP

Filed May 21, 1962 5 Sheets-Sheet 5

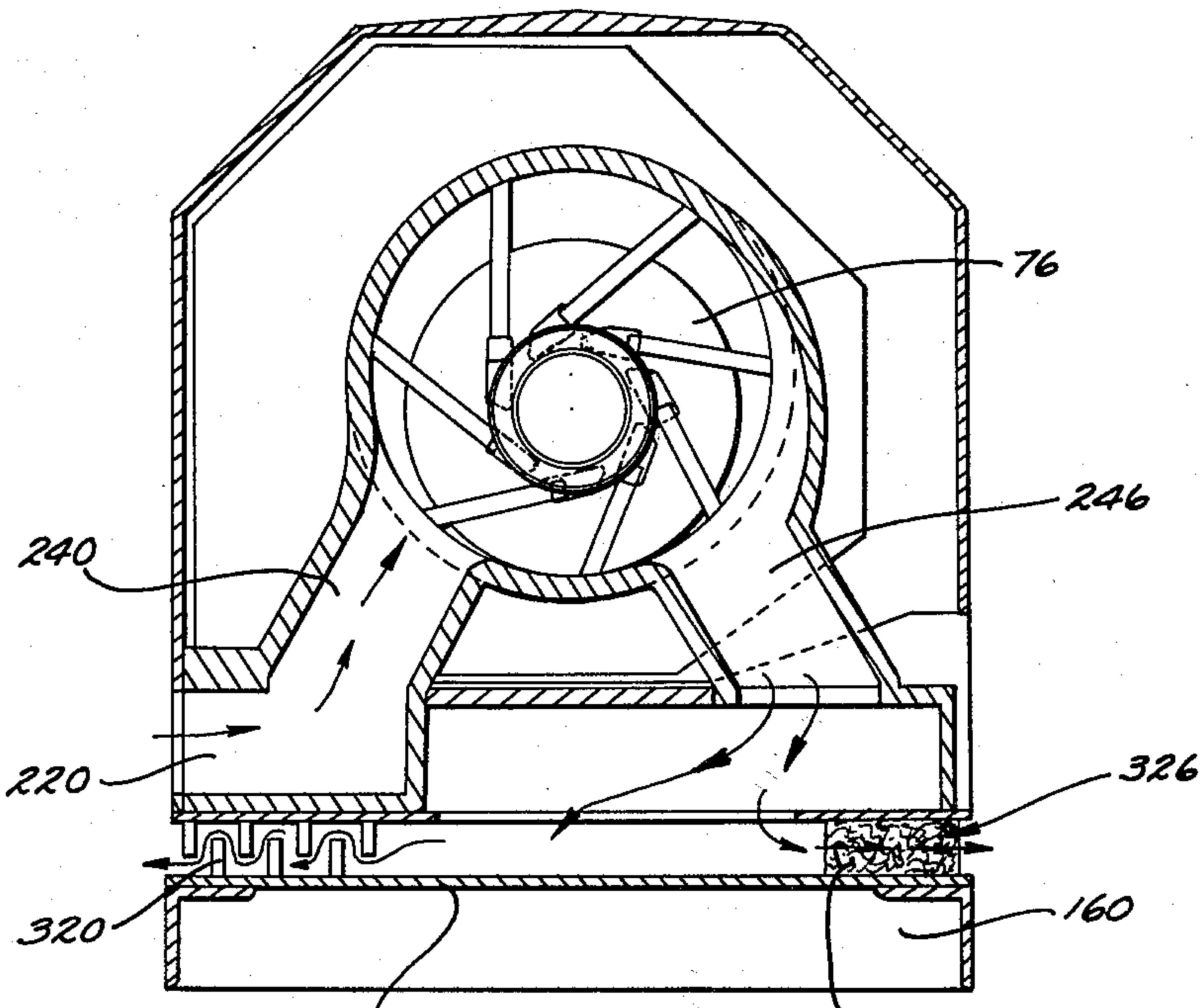

FIG. 11

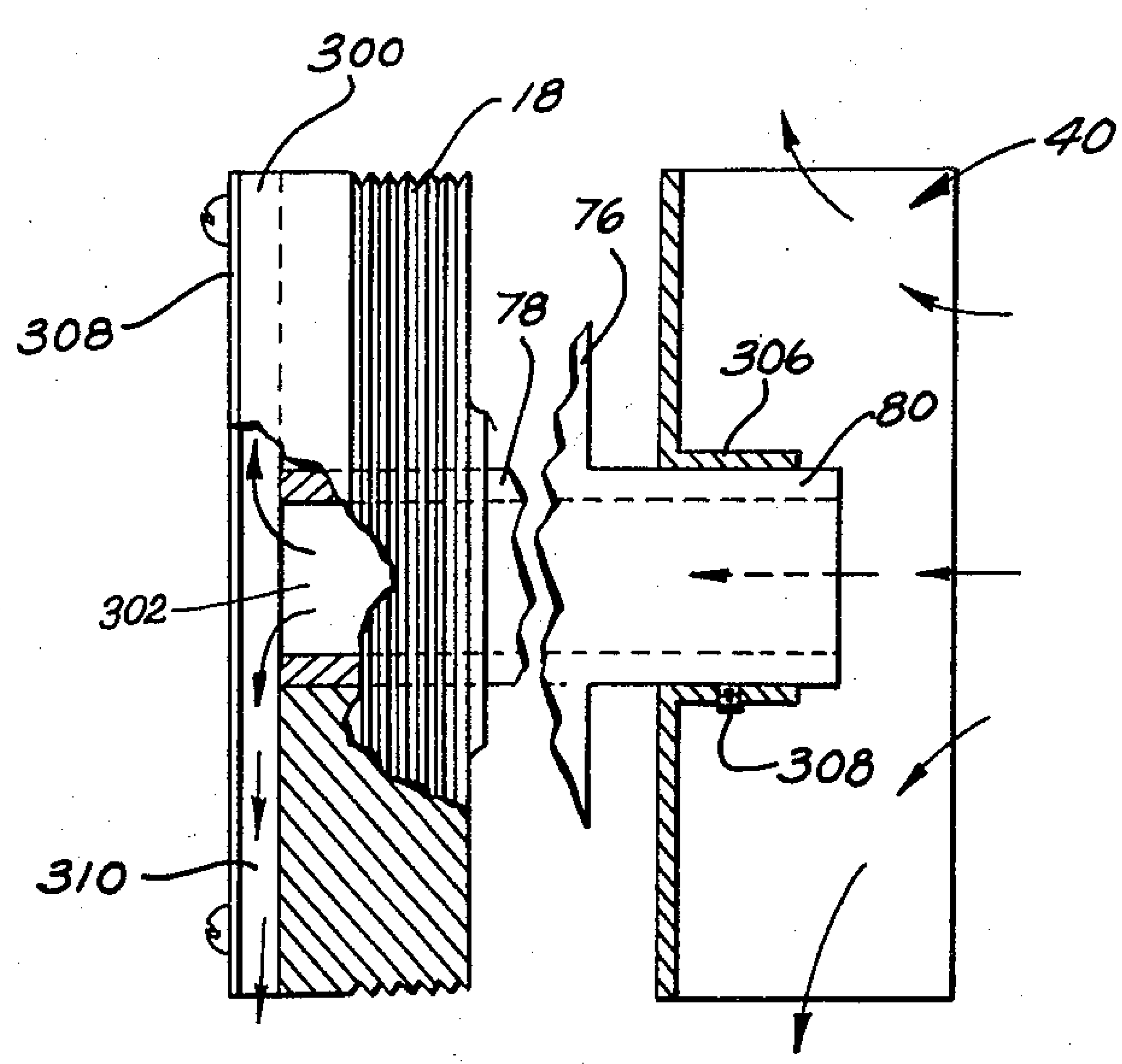

FIG. 10

INVENTOR.
ROLF A. BIELEFELD
BY Price & Heneveld
ATTORNEYS

United States Patent Office

3,180,569
Patented Apr. 27, 1965

3,180,569

COOLED ROTARY PUMP

Rolf A. Bielefeld, Saint Joseph, Mich., assignor to Gast Manufacturing Corporation, Benton Harbor, Mich., a corporation of Michigan

Filed May 21, 1962, Ser. No. 196,267
11 Claims. (Cl. 230—209)

This invention relates to rotary pumps, and more particularly to air-cooled, rotary vane pumps for gaseous media having controlled gas inlet and exhaust features.

The efficiency and dependability of a rotary pump, especially a rotary vane pump, depend to a large extent upon low operating temperatures of the pump. It is well known to those in the field that rotor hub temperatures often reach undesirably high values because of the inability of the hub to effectively dissipate its heat to the outside. Only a limited amount of the heat in the hub is conducted out through the end shafts. Moreover, this heat conducted through the end shafts tends to overheat the bearings supporting the end shafts in the pump housing end plates.

The bearings mounted in the end plates are further subjected to additional heat caused by frictional contact of the high speed rotating pumping vanes against the inner walls of the end plates. The adverse effect of these factors is further magnified by the present lack of a satisfactory housing structure to achieve proper cooling of the exterior of the entire pump in an economical manner.

Another shortcoming of present rotary pumps is the lack of a simplified positive control system of the pump inlet and exhaust gases to enable proper filtering without substantially detracting from the pumping efficiency, or to enable rapid operational changeover of the unit from a filtered compressor to a muffled vacuum pump.

Another feature of conventional rotary vane pumps which detracts from efficiency and output is the high pressure gaseous blow-by or "slippage" which passes beneath the lower edges of the vanes between the end plates and the ends of the rotor hub. This slippage creates operational losses and raises internal operation temperatures considerably.

To obtain optimum efficiency, rotary vane pumps must not only make an effective seal against the end walls, but also must seal effectively against the peripheral housing wall. The pressure of the individual vanes against the housing wall must at all times be great enough to keep the vanes in contact with the peripheral wall. Yet, the pressure of the vane against the peripheral housing wall must not be so great as to cause undue wear of the vanes. Various methods of pneumatically balancing or cushioning the vanes are presently utilized in efforts to accomplish this controlled peripheral pressure. However, these conventional methods generally lack a really accurate or controlled vane cushion since the gaseous pressures under the vane tend to vary largely.

It is an object of this invention to provide a rotary pump with greatly improved efficiency and reliability over an extended period of operating time due to a uniquely cooled structure. The interior of the pump which includes the rotor hub and the end shafts is effectively cooled. Further, the bearings in the end plates around the end shafts are maintained in a cool state by special end plate cooling. Moreover, the exterior housing of the pump is efficiently and economically cooled by a unique flow pattern of cooling air. In addition, the internal cooling, end plate cooling and exterior housing cooling are all effected by simultaneously blower means operated directly on the end shafts of the pump. The end plates not only maintain the bearings in a cooled condition, but also have a heat flow restricting "dam" structure substantially reducing the conduction of frictional heat from the inner walls of the end plates to the portions of the end plates mounting the bearings.

It is another object of this invention to provide a pump having a base construction which constitutes a manifold for directing pump inlet and outlet gases, for filtering or muffling gases, and for directing cooling air. The manifold is reversible with respect to the pumping unit to enable a special filter-receiving chamber (when the unit serves as a compressor) to alternately constitute a muffler-receiving chamber for use with the unit when operating as a vacuum pump. The manifold receives and directs air propelled by the blower on the end shaft.

It is another object of this invention to provide a vane type pump having an accurately controlled air cushion under the base of each vane to provide optimum constant contact of the outer vane edge against the inner housing peripheral wall. The controlled air cushion enables accurate cushion pressures under the vane even though the cushioned gas changes in temperature during the cycle.

These and other objects of this invention will be apparent upon studying the following specification in conjunction with the drawings in which:

FIG. 1 is a side elevational view of the novel pumping apparatus;

FIG. 2 is an end view of the apparatus as viewed from plane II—II of FIG. 1;

FIG. 3 is a side elevational sectional view taken on plane III—III of FIG. 2;

FIG 4 is a sectional view of the apparatus taken partly on plane IV—IV and partly on plane IV′—IV′ of FIG. 1 showing the apparatus adapted to operate as a compressor;

FIG. 5 is a fragmentary sectional view showing the pump rotor and vanes in the housing shell and showing the novel vane cushioning means;

FIG. 6 is a top plan view of the manifold portion of the base of the apparatus;

FIG. 7 is a side elevational slightly enlarged view of the complete base;

FIG. 8 is an end elevational view of one of the end plates adjacent the fan of the apparatus illustrated in FIG. 3;

FIG. 9 is an end elevational view of the end plate adjacent the drive pulley illustrated in FIG. 3;

FIG. 10 is a fragmentary partially sectioned view of a slightly modified double blower arrangement which can be used instead of the single blower illustrated in FIG. 3; and

FIG. 11 is a fragmentary sectional view of the apparatus with the pump reversed on the base to enable operation as a vacuum pump.

Basically, the invention comprises a rotary pump including a hub and attached end shafts having an elongated cooling fluid passageway therethrough. Cooling air is drawn through the passageway by a blower or fan means mounted on the end shaft means. The blower means may be of the centrifugal type. It can be mounted over radial outlet openings in one end shaft, or may be mounted to suck air out an annular passageway in an end shaft.

The pump has an outer shroud over the pump housing, including an end air inlet or one end adjacent a fan, and enclosed transverse fins on the housing to direct the cooling air flow. The fins are spaced from the shroud on the cooler, low pressure zone of the pump to allow air entering into the end shroud opening to flow into the intermediate transverse spaces between the fins. The fins are in close proximity to the shroud adjacent the hotter, high pressure zone of the pump housing to uniformly cool it by forming substantially closed conditions between the fins.

The housing end plates include circumferential, cooling-fluid passageways around the outer periphery of the bearings mounted in the end plates. The circumferential passageways receive cooling air from the fan on the end shaft. The inlet to the circumferential passageway is divided by a baffle to form separate cooling air paths causing high-velocity, low-resistance cooling-air flow over the hotter, high pressure zone of the pump, and causing separate higher-resistance, cooling-air flow over the cooler low pressure zone of the pump.

Each circumferential passageway in the end plates includes an annular, radially-inwardly-directed recess or cavity portion between the inner wall of the end plate and the portion of the end plate in which the bearing is mounted. This provides a heat-flow restricting "dam" substantially lessening the conduction of heat from the inner wall of the end plate to the bearing.

A unique combination supporting base and manifold is removably attached to the bottom of the pump and includes ports and passageways to direct cooling air to the end plates, to direct and filter inlet air or gas to the pump, and to direct compressed outlet air or gas. The manifold is readily reversible to enable rapid conversion of the pump from a compressor to a vacuum pump, with the filter-receiving chamber then comprising a muffler-receiving chamber for the vacuum pumping operation.

A unique series of ports or orifices communicable with the base of the rotor slots beneath the vane provide accurately controlled air cushioning supports especially when used with wide "swells" at the base of the slot. These may communicate with the circumferential end plate passageways or if desired, directly to the atmosphere. They preferably include a cool-air inlet port allowing entry of air under the extended vanes due to their position adjacent the air inlet portion of the pump, a bleed port located part-way around the pump and communicating with the slots under partly depressed vanes, and an exhaust port spaced operationally beyond the air outlet for the pump to communicate with the slots under completely depressed vanes to exhaust the heated, compressed cushioning air. These allow the influx of cool cushioning air during each cycle, and accurate control of the air cushions.

External cooling

Referring now to the drawings, pumping apparatus 10 as externally viewed includes a combination manifold and base 24, enclosing shroud 12, and a drive pulley 18 mounted on an end shaft 16. The term "entire base" is intended to normally include the manifold portion. The shroud 12 may comprise pre-formed thin material such as sheet metal. It is shown formed of two half shells secured together by bands 20 (FIG. 1) around pairs of facing semi-circular projections forming posts 22 (see FIGS. 1 and 4). It may be formed of one continuous piece instead of two, and may include a portion serving as a guard over pulley 18. It fits down around the manifold part of the base as in FIG. 4 to enclose the apparatus. The shroud has a plurality of openings for air inlet and outlets as explained hereinafter.

The base includes a chamber for receiving a filter 26 when the unit is used as a compressor (FIG. 4) or a muffler 326 connected together when the unit is used as a vacuum pump (FIG. 11). The base may be formed of two parts by suitable means such as studs 202 (FIG. 2), or the complete base may be of an integral construction.

Enclosed within shroud 12 and mounted upon the base as by studs 200 is the main pump housing 28 (FIG. 3). (The unit will first be described as a compressor.) It has a plurality of transverse fins 30 on its outer periphery. On the low pressure air intake side of the pump (the left side in FIG. 2 and the right side in FIG. 4), fins 30 are substantially spaced from shroud 12 to provide an air flow space 32 through which cooling air entering the axial end opening 38 (FIG. 3) in the shroud, may flow after passing through the outer portion 100 of centrifugal fan 40. Air drawn through opening 38 passes through fan 40, through space 42, then space 32. Here a change of direction from longitudinal to transverse occurs. The air then flows between the fins in spaces 48 (FIG. 3). Fins 30 are in close proximity to shroud 12 on the opposite side of the pump where the air pressures inside the pump are greater (as explained hereinafter) and thus the temperature is higher. The spaces between adjacent fins 30, the pump housing external surface and the inner surface of the shroud combine to form substantially closed conduits for uniformly distributing the cooling air over the hot portion of the pump. After the air passes between the fins 30 under the bottom and around the top of the pump (FIG. 3), it flows out outlet port 50 in the side of the shroud (FIGS. 1 and 4).

Internal cooling

Referring to FIG. 3, pump housing 28 includes a generally cylindrical central housing portion 70, a first end plate 72 on the drive end, and a second end plate 74 on the fan end of the pump. Within the housing is rotatably mounted a rotor means including central rotor hub 76, integral end shafts 78 and 80, and slidably mounted vanes 82 (FIGS. 3 and 4). The vanes 82 fit in generally radially oriented slots extending the length of the hub, such that the ends of the vanes contact the inner walls 84 and 86 of the end plates 72 and 74. The vanes also contact the inner peripheral wall 88 of the cylindrical portion of the housing during operation of the pump.

The end shafts are rotatably mounted in the end plates by suitable bearings 90 and 92 with adjacent seals 94 and 96. The end plates include caps 110 and 112.

In the form of the invention illustrated in FIG. 3, on end shaft 80 is attached a centrifugal fan 40 which includes an outer section 100 and an inner section 102, divided by a partition 104. The fan has a centrifugal action. Fan 40 is secured to end shaft 80 by a collar 114 and set screw 116, or other equivalent means. In order to provide proper cooling of the internal structure of the pump, the end shafts 78 and 80 and the hub 76 are provided with a central passageway 120 and radial outlet ports 122 in end shaft 80. These ports communicate with inner section 102 of fan 40 to enable air to be drawn through passageway 120, exit at ports 122, and then into passageway 42 between the housing and the shroud to mix with air introduced at port 38 of the shroud. This interior passageway thus cools both end shafts adjacent bearings 90 and 92 and also conducts heat away from the central rotor hub for optimum efficiency. It will be noted that the cooling occurs simultaneously with operation of the pump since the fan is mounted directly upon the end shaft, and that the same fan cools both the interior and the exterior.

Instead of the single fan shown in FIG. 3 and described above, it is sometimes desirable to utilize fan 40 of a conventional construction (see FIG. 10) on end shaft 80 to direct air around the outside of the unit, and to utilize a separate centrifugal fan 300 formed integrally with shiver 18 to suck air through passageway 120 in the opposite direction. In this instance an axial inlet 302 in end shaft 80 will suffice instead of radial openings 122. Fan 300 may include an end plate 308 to close off radial outlets 310. Fan 40 may in this instance be mounted to end shaft 80 by a suitable hub 306 and set screw 308. Pulley and fan 300 may be mounted to end shaft 78 by suitable key means (not shown).

Cooled end plates

The end plates 72 and 74 include circumferential passageways 130 and 132 respectively, encircling bearings 90 and 92. A portion of the cooling air entering at 38 flows through these circumferential passageways to cool the end plates. These passageways are enclosed by end caps 110 and 112 attached to the end plates as by bolts 111. More specifically, air entering at 38 (FIG. 3) passes downwardly into space 140 and is propelled by the outer section 100 of the fan 40 into passageway 142 over the deflecting diagonal partition 141 (FIG. 3) on the manifold base. It then flows into passageway 132 and out the lateral exhaust port 148 (FIG. 2). It will be noted from FIG. 2 that a baffle 150 divides the inlet to this circumferential passageway 132 into two portions. Part of the air flows (to the right) through a short, low flow-resistance, radial segment comprising about 90° of the circumferential passageway and out through the outlet port 148. Another portion passes (to the left) around the longer, higher-resistance segment comprising about 270° of the passageway and out the outlet port 148. The low resistance segment is adjacent the high compression zone of the pump which is the hottest. This zone demands more cooling than the lower compression zones which are cooled by the second flow path. Openings close to the heat dam are provided in the radial support flanges or baffles 152 and 154 to allow the cooling air to flow through them in the long segment.

Not only is end plate 74 cooled in this fashion but also end plate 72. The cooling air entering at port 38 (FIG. 3) and flowing into space 140 passes partially through open space 160 between the supporting bracket or stand 14 and lower base portion 24. It flows clear under the pump and through space 162 over the diagonal deflecting partition 164 on the manifold base. It then flows into circumferential passageway 130 where the flow is divided into two paths, one being a short, low flow resistance path over the high compression hot zone of the pump, and the other being a higher resistance longer path over the cooler low compression portions of the pump just as in the opposite end plate. The air then emerges at exhaust port 170 (FIG. 1). The walls of the exhaust port 170 may be integral with either the end plate or shroud.

To further protect bearings 90 and 92 from heat created by frictional contact of the ends of the sliding vanes 82 with the inner walls 84 and 86 of the end plates 72 and 74, a heat-flow restricting "damming" structure is provided in the end plates as shown in FIG. 3. This comprises annular, radially-inwardly directed cavities or recesses 176 and 178 in end plates 72 and 74, respectively extending a substantial distance between the bearing-receiving recess in the end plates and the inner wall of the end plate. Cooling air circulating through the circumferential passageways 130 and 132 thus effectively conduct the heat away from the inner end plate walls. This constitutes in effect a heat-flow damming structure since only a small metal partition 93 exists through which heat can be conducted.

Combination manifold base

The base not only comprises a removable mounting means for the pump, but it also constitutes a manifold for directing air flow and for receiving a filter or muffler. The detailed structure of the manifold base can be seen more clearly in FIGS. 6 and 7. It comprises essentially a generally rectangular, lower portion 14 and an upper portion 24 thereabove and having an open bottom. It includes a chamber 206 into which a filter 26 or a muffler 326 can be slidably inserted from the side. The area above the chamber is substantially closed by a central covering partition 208. The entire area beneath this partition is closed except for four openings 210, 212, 214 and 216 adjacent the corners. Two of these openings, 210 and 212, constitute ports leading from the filter chamber to the pump inlets 240 and 242 (FIGS. 8 and 9) in the end plates when the unit is adapted to operate as a compressor as shown in FIG. 4. This allows air or gas passing through the filter to pass into the pump to be compressed. The other two openings 214 and 216 in the base constitute outlet ports adapted to communicate with the pump outlet passageways 246 and 248 in the end plates. This allows compressed air to pass from the pump into a small end chamber 222 in the base (FIG. 6) and then out through compressed gas-exhaust port 220 to which a suitable conduit (not shown) may be attached. To separate compressed gas in chamber 222 from the uncompressed inlet gas in the filter chamber 206 under partition 208, an elongated vertical partition 224 is provided to extend across the width of the base.

The portion 230 of partition 208 adjacent outlet port 220 and that portion 232 on the opposite side are raised at the edge, and they tilt downwardly on an angle toward the center (see FIGS. 2 and 7). Surfaces 230 and 232 provide smooth deflecting surfaces for the cooling air passing beneath the housing between fins 30 as explained heretofore.

Ports 210 and 212 in the base and ports 214 and 216 cooperate respectively with the inlet and outlet passageways in the end plates of the pump housing, in the following manner. Referring to FIGS. 8 and 9 end plates 72 and 74 are shown without their end caps 110 and 112 so that radial bracing flanges 150, 152 and 154 are shown clearly. The end plates 72 and 74 have inlet passageways 240 and 242 respectively, and outlet passageways 246 and 248 respectively.

When air is to be compressed, it is supplied to the pump by being drawn through port 38 by fan 40 and downwardly through space 140 and space 160 (FIG. 3) and then up through filter 26. It then kows through openings 210 and 212 in the base 24, up through inlet ducts 240 and 242 in the end plates and between the vanes 82 inside peripheral wall 88. It is then compressed by the rotor means and discharged through ducts 244 and 246 in the end plates, down through openings 214 and 216 in the into chamber 222, and out through discharge port 220.

When a vacuum pump operation is desired (see FIG. 11) the novel manifold structure enables the unit to be quickly changed from the filtered compressor to a muffled vacuum pump unit. This may be effected by simply removing bolts 200 out of the threaded base openings 201, rotating the pump 180° with respect to the manifold base, and replacing bolts 200. The filter chamber 206 thereby becomes a muffler-receiving chamber by sliding out filter 26 and inserting muffler 326.

In the operation of the unit as a vacuum pump, it will noted that the rotor 76 and vanes are reversed with respect to the base so that port 220 becomes an inlet port. The chamber or line to be evacuated is thus attached over port 220. The evacuated gas then passes into port 220, up passageways 240 and 242, around rotor 76 in a clockwise direction as viewed in FIG. 11, down passageways 246 and 248 and into the chamber above the muffler 326. This muffler may contain conventional resonance baffles 320 or absorption structure 322 such as steel wool. The muffler includes a lower partition 324 preventing the hot evacuated gases from passing into chamber 160 where cooling air is flowing to end plate 72 (FIG. 3). Thus the evacuated hot gases pass out the sides of the muffler as indicated by the arrows in FIG. 11.

Controlled vane cushion

In FIG. 5, the fragmentary sectional view shows the pump housing (without its cooling fins) with the inner peripheral wall 88 defining the cylindrical chamber in which rotor 300 is eccentrically mounted. The rotor axis is parallel to the axis of the cylindrical chamber 88. Formed in the rotor over the length thereof is a plurality of generally radially positioned vane-receiving slots 302. Each of these slots preferably has an enlarged cavity 304 at the base thereof as claimed in my co-pending application Serial No. 181,618, filed March 22, 1962, now Patent No. 3,138,321. This cavity 304 may, in this invention, be of the same dimensions as the remainder of slot 302, but the wide cavity or well is preferred.

Since vanes 82 are slidably mounted in slots 302, and since rotor 300 is eccentrically mounted with respect to the main housing portion 70, the vanes constantly move between an extended position as shown by vane 82' to to a completely depressed position as shown by vane 82''. When the depressed vane begins to extend again, it passes the air inlet port 240 including recess 241 in the end plate. The vane continues to draw in air until it reaches the position of vane 82'. As these vanes pass around with the rotor hub, they are slowly depressed into their slots as the air is compressed between the rotor hub and the wall 88. When the exhaust port 246 is reached, compressed air is expelled while the vane continues to be depressed into the slots to its maximum depressed position as at vane 82'', and the cycle then begins again.

During operation, the movement of each vane in its respective slot causes sequential compression and evacuation of the air or gas trapped under the vane. The pressure under the vane tends to exaggerate vane tip pressure against the housing wall. Vacuum tends to keep the vane from continuous contact with the wall. Various methods have been denied in attempts to dynamically balance the vane including arcuate channels in the end plates or rotor ends, and including radial passageways in or adjacent the vane. At best, these allow constant pressure equalization, but do not provide accurate vane tip pressure control over the critical portions of the vane movement cycle. More specifically, it has been found that the vanes tend to move out of contact with the peripheral housing wall at and adjacent (just beyond) the exhaust port of the pump. Consequently, it has been found to be greatly advantageous to achieve larger vane tip contact pressure in this area. This necessitates a unique venting construction since this greater pressure caused by hot compressed gases in the vane cushion should be relieved right after this area has been passed. Thus, referring to FIG. 5, it will be noted that the bases 320 of the vane cavities are not exposed to a vent as the vane tip approaches and passes exhaust port 246 (this exhaust port may be either in the end plates, the peripheral housing wall, or both). Since the vane is compressing gases in the slot base 320, a pressure cushion is created forcing the vane tip into a greater pressure contact with the wall 88. This continues as the vane passes the exhaust port until it is adjacent the inlet port i.e. where it begins to extend out of its slot. At that time, slot base 320 (preferably wall 304) communicates with channel 322 and outlet port 321 to exhaust the hot compressed cushion air into the end plate passageway (or if desired to the atmosphere). If the unit is operating as a vacuum pump, on the other hand, port 321 allows entry of atmospheric air into the slot base which will have been largely evacuated by escape of the cushion alongside the vanes. Thus the new air provides pressure for continuous vane tip contact.

As the vane continues to move with the rotor with the unit operating as a compressor, slot base 320 next communicates with arcuate channel or passageway 324 which has an outlet port 323 to the end plate passageway in the cool-air portion thereof. The gradually extending vane thereby sucks in or aspirates fresh, cool air to form the next cushion. This maintains the vanes in a relatively cool condition, and enables an accurate control of the cushion pressure.

Preferably a third or bleed porting means comprising port 325 and passageway or channel 326 is provided part way around as the vane is in the initial depression stages with the slot base chamber being smaller. The exact location of the last portion of this channel 326 enables an exact adjustment of the volume of gases in the vane cushion. This is caused by a limited bleeding or venting. The small amount of air exchange enables an exact adjustment of the cushion to provide the optimum cushioning in the high compression zone of the pump. As stated, the exact location of one or more bleed ports determines the final cushion pressure. These three ports thus cooperate to first take in cool air, accurately bleed off a small percentage of the warm, partially compressed air to set the air cushion at the exact desired amount, and then exhausting the compressed hot cushion air to enable it to again receive cool cushion air. In the pump illustrated, each of the ports 321, 323 and 325 cooperating respectively with the passageways 322, 324 and 326 is preferably formed in the end plate as in end plate 74 in FIG. 3. They communicate through the annular recess 178 to the circumferential passageway 132. The other end plate 72 can be provided with similar parts. As shown more specifically in FIG. 2, the cool air inlet 323 is adjacent the start of the 270° passageway, the bleed port 325 is part way around the 270° passageway, and the exhaust port 321 is associated with the 90° flow path in the circumferential passageway. If desired, a pump may utilize this porting feature without end plate passageways by porting to the atmosphere.

Conceivably channels 324 and 326 can be joined into one continuous passageway to both provide cool inlet cushion air and bleed it to the proper volume before the vane causes cushion compression. However, it has been found desirable to provide a non-ported area 331 between the passageways 324 and 326, and located such as to eliminate slot base venting just before the vane passes through its most extended position. The vane, as it extends the final amount, thereby causes a small vacuum to occur in the slot base, tending to reduce vane tip contact pressure at this point and thereby reducing wear considerably. Thus, by using one or more ports at specific locations to vent the vane cushions only at crucial moments when they are under specific different pressure potentials, excellent control of vane tip pressure is achieved. The results achieved with the unit operating as a compressor are (1) a pressure cushion when the vane passes is most depressed in the slot to keep it in firm contact with the wall thereby preventing "slippage," (2) accurate bleeding and control of this pressure cushion, (3) complete aspiration of cushion cool air with each cycle, and (4) a vacuum cushion when the vane is in its most extended position to minimize wear.

In operation, fan 40 serves to propel cooling air between fins 30 to cool the pump exterior, can cause air flow through passageway 120 to cool the rotor hub and the end shafts (alternately fan 300 can cause this latter air flow), propels air through the circumferential passageways 130 and 132 in both end plates to cool the ends of the pump and the exterior of the bearings, and supplies air to be compressed by passing it through filter 26 and up through the manifold openings. The manifold directs the compressor inlet and outlet air or gas through filter 26 and openings 210, 212, 214 and 216, it directs the end plate air flow over surfaces 141 and 164, and it creates a smooth flow under the pump exterior with surfaces 230 and 232. Reversal of the manifold allows operation of the compressor as a vacuum pump during which the manifold base directs the evacuated gas through port 220, into the pump chamber, and through the muffler 326. The unit is capable of remarkable efficiency due to its cool operation, and also due to the controlled vane-contact pressure. It is relatively compact in spite of its elaborate cooling system and manifolding characteristics.

Certain obvious modifications of this apparatus within the inventive principles taught may be made without departing from the scope of the invention. Such modifications are deemed part of this invention, which is to be limited only by the appended claims and the reasonable equivalents thereto.

I claim:

1. A rotary compressing pump compressing: a housing including end walls; a rotatable rotor means within said housing; said pump having a low compression pumping zone and a high compression pumping zone; a plural-

ity of transverse cooling fins and intermedaite spaces on the housing exterior; a shroud generally around said housing and including cooling air inlet and outlet means, and said shroud and fins being relatively spaced from each other adjacent said low compression zone to allow air to pass from said shroud inlet to the spaces between said fins; said shroud and fins being closely positioned with respect to each other adjacent said high compression zone to cause said spaces to form substantially closed passageways whereby air can be uniformly directed over the heated high compression area for effective cooling; and cooling fan means adapted to propel air over said fins within said shroud.

2. A rotary compressing pump comprising: a housing including end walls; a rotatable rotor hub within said housing including slidable vanes and having end shafts; said rotor being eccentrically mounted within said housing to create a low compression pumping area and a high compression pumping area; a plurality of transverse fins and intermediate spaces on the housing exterior; a shroud generally around said housing and including cooling air inlet means on one end, and outlet means adjacent said high compression area; said transverse fins being displaced toward the side of said housing having said high compression area into close relationship with said shroud to define a plurality of substantially closed cooling passageways between said fins, shroud, and housing to uniformly conduct cooling air over said high compression area; and a cooling fan adapted to propel air over said fins within said shroud.

3. A rotary compressing pump comprising: a housing including end walls; a rotatable rotor hub within said housing including slidable vanes and having end shafts; said rotor being eccentrically mounted within said housing to create a low compression pumping zone and a high compression pumping zone; a plurality of transverse films and intermediate spaces on the housing exterior; cooling fan means mounted on at least one of said end shafts; a shroud around said housing; an air inlet in said shroud adjacent said fan; a passageway through said end shafts and said hub including outlet means in one end shaft adjacent said fan means; said shroud and transverse fins being relatively spaced from each other adjacent said low compression zone to allow air down through said passageway and through said air inlet to pass into said spaces between said fins; said shroud and fins being closely positioned with respect to each other adjacent said high compression area to cause said spaces to form substantially closed conduits; and outlet means in said shroud adjacent said conduits.

4. A rotary pump comprising: a pump housing; rotatable pumping means in said housing including rotor means and end shafts on said rotor means; cooling fan means mounted on at least one of said end shafts and adapted to rotate with said shaft; shroud means spaced from and around said housing including air inlet and outlet means; cooling fins on said housing to direct air flow around and conduct heat from said housing; an air flow passageway through said end shafts and said rotor means including outlet means adjacent said fan means; said housing including end plates; each of said end plates having a circumferential cooling fluid passageway including an inlet and outlet within the enclosure of said shroud; and said fan means being operably associated with said fins, said air flow passageway, and said circumferential passageway to simultaneously cool the pump housing exterior, the pump interior, and the end plates, while the pump is in operation.

5. A rotary pump comprising: a housing means including an inner peripheral wall and a pair of end plates; rotor means mounted in said housing and including end shafts supported by and extending into said end plates; bearing means mounted in said end plates for said end shafts; circumferential cooling fluid passageways around said bearing means in said end plates; air directing shroud means around said pump; and blower means mounted on at least one of said end shafts and adapted to propel cooling air directed by said shrouds into said circumferential passageways to maintain cool bearings.

6. A rotary pump comprising a housing including two end plates; eccentrically mounted rotor means in said housing including end shafts rotatably mounted in said end plates; said pump including a pump fluid inlet adjacent the relatively large low compression zone of the pump and including a compressed fluid outlet adjacent the relatively small high compression zone of the pump; a cooling fluid passageway in each of said end plates substantially encircling said end shaft; a cooling fluid inlet means to each of said passageways between said high and low compression zones; and each of said cooling fluid inlet means being divided to create a high velocity, low-flow-resistance coolant flow path past said relatively small, high compression zone, and to create a separate, lower velocity, higher-flow-resistance coolant flow path past said relatively large, low compression zone.

7. A rotary pump comprising: a housing means including an inner peripheral wall and a pair of end plates; rotor means mounted in said housing and including end shafts supported by and extending into said end plates; bearing means mounted in said end plates for said end shafts; circumferential cooling fluid passageways around said bearing means in said end plates; each of said end plates having an inner wall in rubbing contact with portions of said rotor means; and each of said passageways including a circumferential, inwardly radially directed cavity extending between said bearing means and said inner wall thereby forming a heat-flow restricting dam.

8. A combination manifold base for a rotary compressor pump having inlet and outlet ports adjacent the bottom thereof, comprising: a base housing including means for connecting and aligning said base to a pump; an air inlet in said base; a filter-receiving chamber in said base adjacent said air inlet; first air porting means in the upper portion of said base adapted to communicate with the inlet port of a rotary pump and with said filter-receiving chamber; second air porting means in said base adjacent the upper portion thereof and adapted to communicate with the outlet port of a rotary pump; and passageway means and an exhaust outlet in said base communicating with said second air porting means to control the pumped air.

9. The base in claim 8 wherein said first and second air porting means are generally symmetrically arranged, wherein said base is removably attachable to the pump, and wherein said filter retaining chamber is adapted to retain a muffler such that said base may be rotated to enable said pump to operate as a vacuum pump.

10. A rotary pump comprising a housing including two end plates; pumping rotor means mounted in housing between said end plates and having a pair of end shafts in bearing contact with said end plates; inlet and exhaust ports in said housing adjacent the bottom thereof; a generally flat manifold under said pump and removably secured thereto; said manifold having a lower air inlet and a filtering chamber communicating therewith; a pumping fluid outlet in the top of said manifold communicating with said filtering chamber and said inlet port in said pump housing; an exhaust entry port in the top of said manifold communicating with said pump exhaust port in said housing; an exhaust chamber in said manifold communicating with said exhaust entry port and having a pressure port leading outside said manifold; cooling air directing surfaces on said manifold associated with said end plates of said housing to direct cooling air thereto; and blower means capable of propelling air to said cooling air directing surfaces and said manifold lower inlet.

11. A rotary pump comprising a housing including two end plates; pumping rotor means mounted in said housing between said end plates and having a pair of end shafts in bearing contact with said end plates; inlet and exhaust

ports in said housing adjacent the bottom thereof; a generally flat manifold under said pump and removably secured thereto; said manifold having a lower air inlet and a filtering chamber communicating therewith; a pumping fluid outlet in the top of said manifold communicating with said filtering chamber and said inlet port in said pump housing; an exhaust entry port in the top of said manifold communicating with said pump exhaust port in said housing; an exhaust chamber in said manifold communicating with said exhaust entry port and having a pressure port leading outside said manifold; cooling air directing surfaces on said manifold associated with said end plates of said housing to direct cooling air thereto; a shroud around said housing and having an air entry port; and a blower means mounted on at least one of said end shafts adjacent said shroud air entry port, whereby said blower means propels air to enable said shroud and air directing surfaces to direct cooling air to said end plates and propels air to said manifold lower inlet.

References Cited by the Examiner

UNITED STATES PATENTS

1,895,816	1/33	Pfeiffer	230—210
2,454,371	11/48	Berges	230—211
2,489,887	11/49	Houghton	230—210
2,611,248	9/52	Ahlen et al.	230—211 X
2,808,813	10/57	Lindhagen et al.	230—211 X
2,831,631	4/58	Petersen	230—152
2,876,948	3/59	Hockel et al.	230—210
2,892,584	6/59	Briscoe	230—152
2,933,239	4/60	Blackman	230—211 X

FOREIGN PATENTS

335,735	10/30	Great Britain.
435,612	9/35	Great Britain.

LAURENCE V. EFNER, *Primary Examiner.*

ROBERT M. WALKER, *Examiner.*