

April 27, 1948.

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2,440,593

RADIAL VANE PUMP MECHANISM

Filed Oct. 23, 1946

4 Sheets-Sheet 1

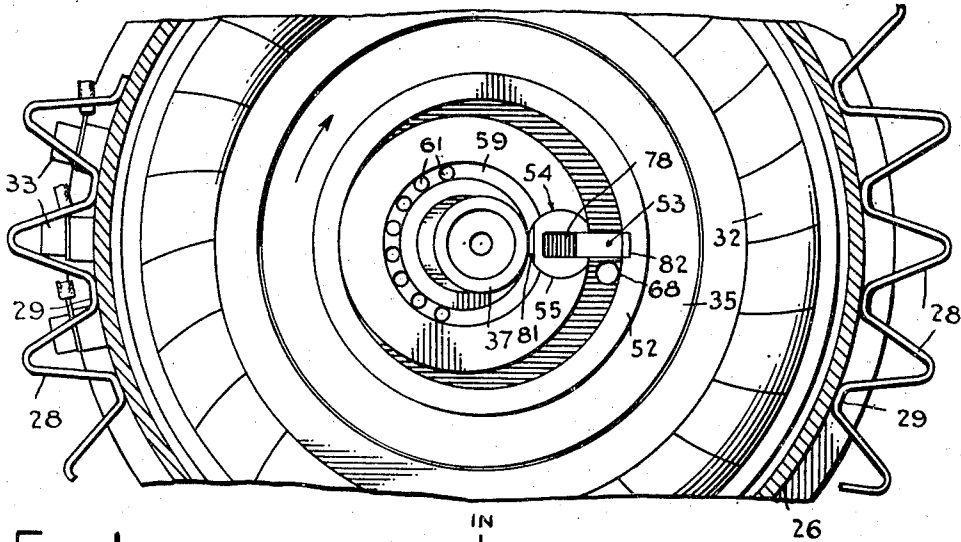


FIG. 1.

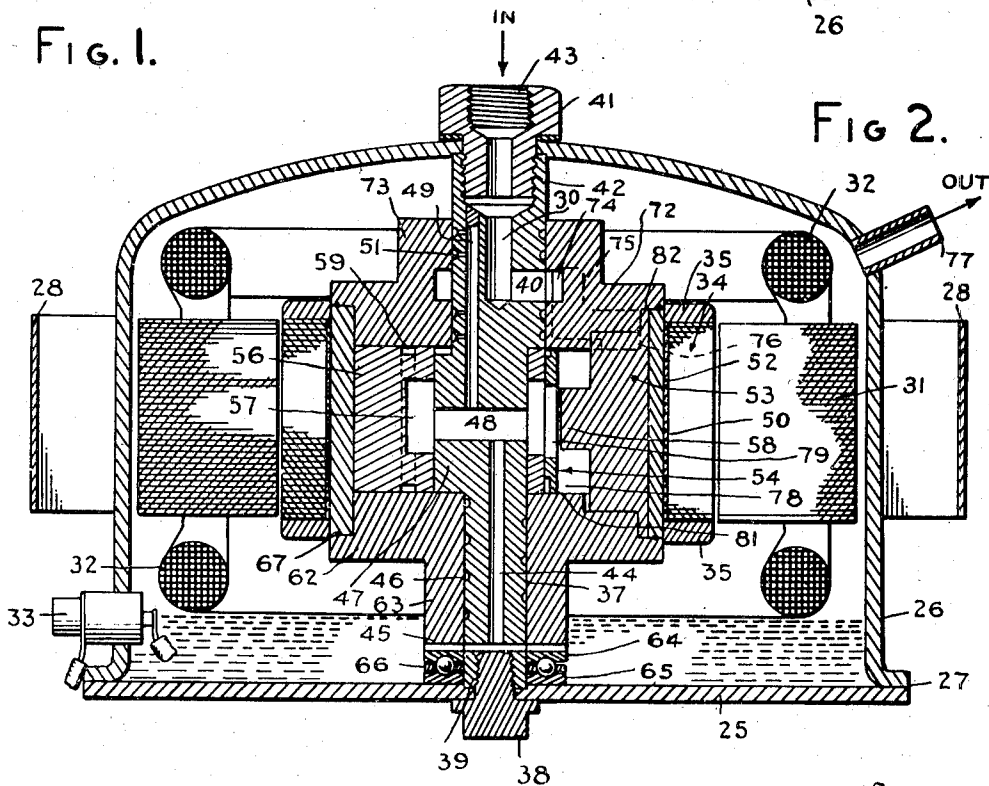


FIG 2.

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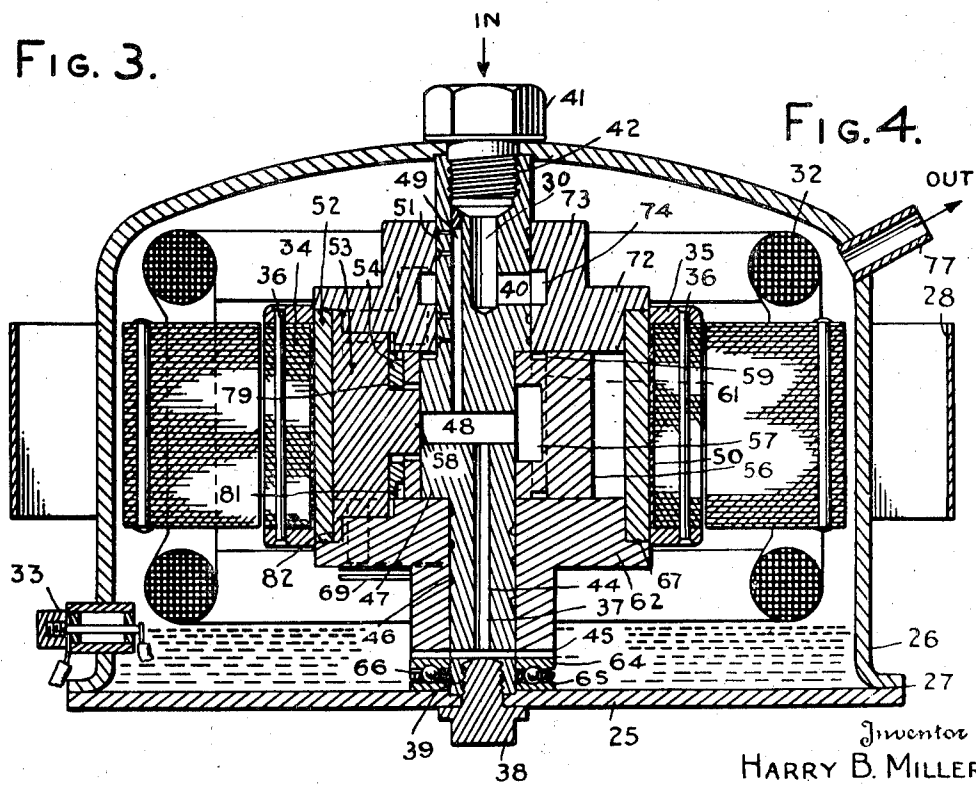
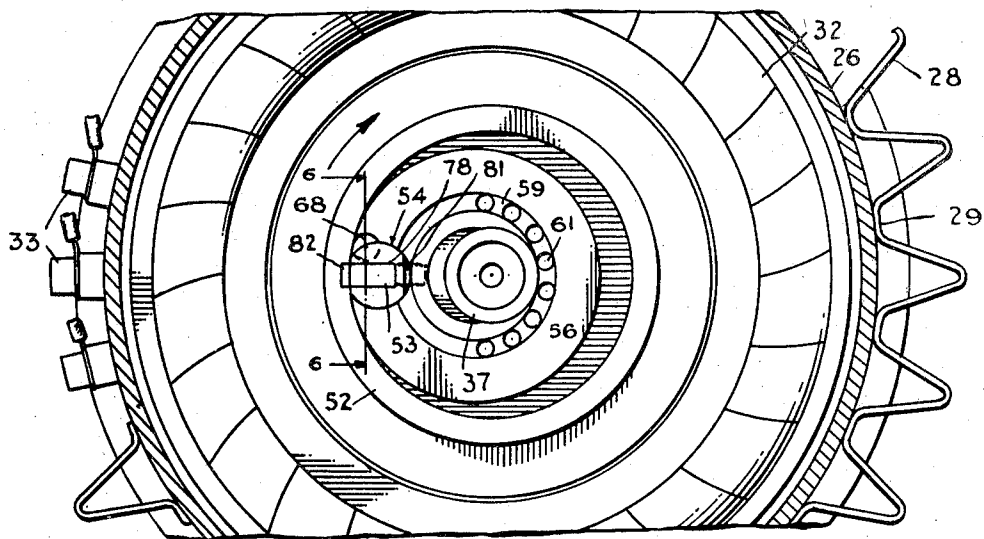
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FIG. 5.

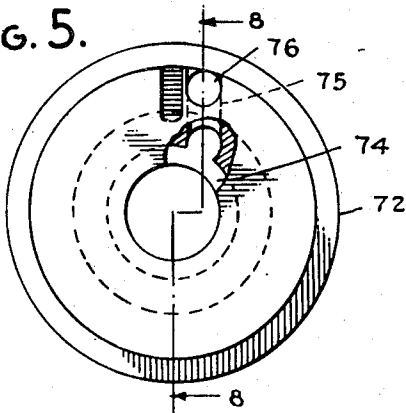


FIG. 8.

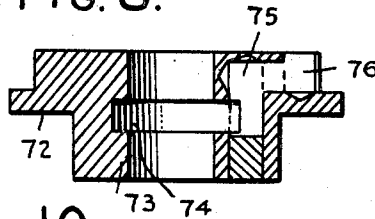


FIG. 10.



FIG. 6.

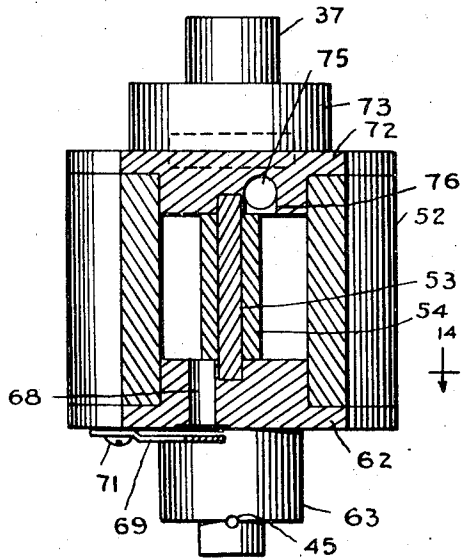


FIG. 9.

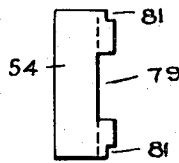


FIG. 11.

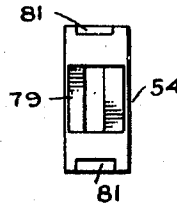


FIG. 13.

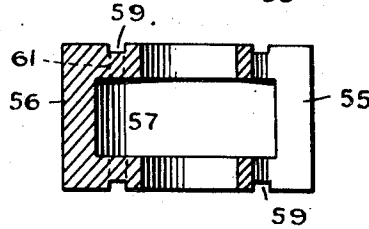
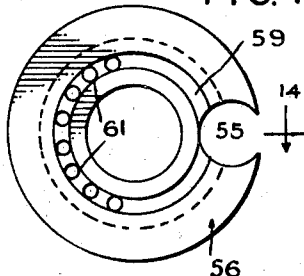


FIG. 14.

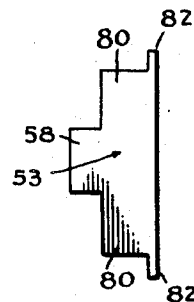
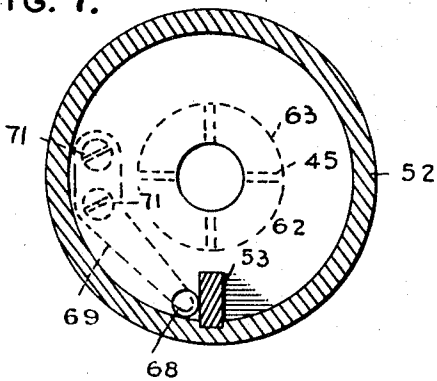


FIG. 12.

FIG. 7.



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FIG. 17.

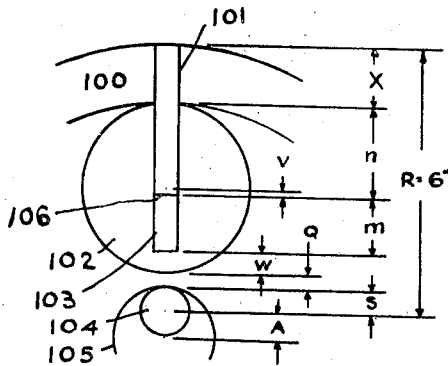


FIG. 18.

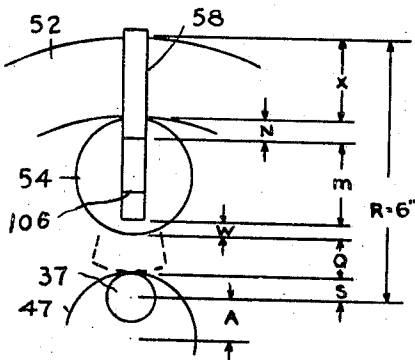


FIG. 19.

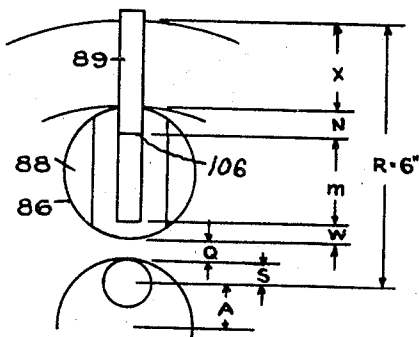


FIG. 15.

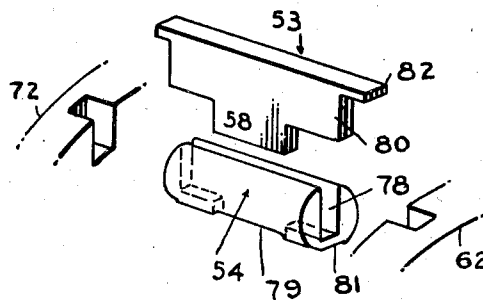
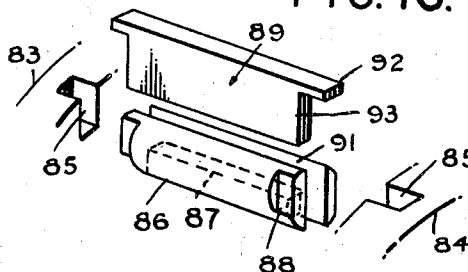


FIG. 16.



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UNITED STATES PATENT OFFICE

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RADIAL VANE PUMP MECHANISM

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Application October 23, 1946, Serial No. 705,180

2 Claims. (Cl. 230-140)

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This invention relates to rotary pumps of the positive displacement type in which a radial vane separates the intake and pressure chambers of the pump. More particularly, the invention relates to a pump in which the outer member or cylinder rotates, and the inner member or rotor also rotates, each about its own center but in eccentric relationship to each other, so as to form a crescent-shaped pumping chamber between the two members. This chamber is divided into intake and pressure portions by the radial vane, and the sides of the crescent-shaped pumping chamber are enclosed by suitable end plates.

It is well known that pumps of this character are advantageous because of their freedom from vibration and the absence of sliding friction between the piston and cylinder. Since the outer cylinder and the rotor both rotate, each about its own center, there is freedom from vibration and inertia effects, and, since the two parts are in tangential relation to each other, and the slight contact, if any, is a rolling one, the wear is reduced to a minimum. It has been the usual practice in the prior art to cause actual rolling contact between the outer cylinder and the rotor, but according to the present invention it is proposed to maintain a very slight clearance between these members and to seal this clearance by a thin oil film. In this way not only are frictional effects eliminated, but adequate sealing is accomplished as an incident to the lubrication of the pump and wear due to the usual contact is non-existent.

In pumps of the type under consideration where both the outer cylinder and the rotor rotate together, it is the usual practice to secure the radial vane to the outer cylinder and to connect it in driving relation with the rotor through a vane bearing which is cylindrical in cross section and contains a longitudinal slot which embraces the faces of the vane, producing substantial frictional contact during the operation of the pump. In known constructions the vane bearing rocks back and forth as the parts move so that the sides of the vane, as well as the slot in which it moves, are subject to serious wear with continuing tendency to leakage. Inasmuch as it is desirable to reduce this frictional contact to a minimum, it is considered to be good engineering practice to reduce this rocking action by so arranging the parts that the vane is never withdrawn radially outward beyond the geometrical center of the vane bearing during the operation of the pump. Should it go be-

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yond this center, the bearing is canted so as to rapidly increase the frictional contact and consequently the wear between the faces of the vane and their area of contact with the bearing. This required relation of parts automatically imposes a serious handicap on machines of this type because it limits the amount of pump displacement which can be secured without unduly weakening the metal sections of the pump. This is particularly true of the radial metal section existent in the rotor between its bearing and the vane bearing in which the vane moves. As will be pointed out hereinafter, the present construction maintains the faces of the vane in parallelism with the surfaces of the vane bearing with which it cooperates, and an arrangement is provided whereby rocking is completely eliminated and the vane partakes of straight line movement only with respect to the vane bearing. By the provision of a vane of peculiar shape and a cooperating bearing of appropriate shape, the displacement of the pump may be materially increased over that which is possible with prior art constructions. Whereas in prior art constructions the displacement of the pump was determined by the amount of radial movement of the vane in the vane bearing, practice of the present invention removes that limitation. Consequently the capacity of a pump of given outside dimensions may be increased as indicated. It is also proposed to provide the working parts with adequate lubrication and to provide a pressure seal for the rotor of the pump as an incident to the normal operation of the parts without the addition of any extraneous means. This pressure seal produced in this manner permits pump operation at pressures or vacua greatly in excess of those possible with existing apparatus, and with the simplification indicated.

Another proposal according to the present invention is to so construct and arrange the parts that the outer cylinder of the pump constitutes the rotor of an electric driving motor so as to economize in the size of parts and also to permit the substitution of different pump elements while utilizing the same electric motor construction. Because of the relatively high speed of operation of pumps of this character, an exceptionally great increase in capacity may be obtained by a relatively small change in the size of the pump parts, where this would not be possible in prior art pump constructions, particularly those of the reciprocating type. It goes without saying that apparatus of this type can operate either as a pump or as a motor, depending upon whether it

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is used as a source of pressure, or whether pressure is supplied to it.

It will be apparent from the preceding description that an important object of the present invention is to produce a simple compact device made up of a minimum number of parts and hence capable of manufacture at a minimum cost. Only a few of the advantages of the invention have been mentioned but many others will be apparent from the following description when it is read in conjunction with the accompanying drawings in which:

Figure 1 is a horizontal sectional view of a portion of the assembly showing the pump in end elevation;

Fig. 2 is a central vertical section through the complete assembly showing the pump parts in the position in Fig. 1;

Fig. 3 is a view similar to Fig. 1 with the pump rotor displaced 180 degrees from the position of Fig. 1;

Fig. 4 is a central vertical section through Fig. 3;

Fig. 5 is a view in elevation of one of the end plates showing the intake port;

Fig. 6 is a horizontal sectional view through the pump showing the intake and pressure ports with reference to the radial vane and its bearing, the section being taken substantially on the line 6-6 of Fig. 3;

Fig. 7 is an inside end view of the end plate carrying the exhaust valve, this valve being shown in dotted lines;

Fig. 8 is a horizontal section through the end plate of Fig. 5 substantially on the line 8-8 of that figure;

Fig. 9 is an elevation of a preferred form of vane bearing;

Fig. 10 is an end view of the vane bearing of Fig. 9;

Fig. 11 is a plan view showing the slot in the bottom of this bearing;

Fig. 12 is a side view of the preferred form of vane which forms an important feature of the invention and which cooperates with the bearing shown in Figs. 9, 10 and 11;

Fig. 13 is a detailed plan view of the pump rotor showing the oil ducts and the support for the vane bearing of Figs. 9, 10, and 11;

Fig. 14 is a section substantially on the line 14-14 of Fig. 13;

Fig. 15 is a detailed perspective view of the preferred form of vane showing its relation to the novel type of vane bearing, and to the end plates of the machine;

Fig. 16 is a perspective view of a modified form of vane bearing and vane showing the manner of supporting the bearing in the end plates of the machine and its relation to the vane;

Figs. 17, 18, and 19 are diagrammatic views showing the mathematical relation of the parts of apparatus embodying the present invention and comparing the relation of those parts to apparatus of the same general character taken from the prior art.

Referring now to the drawings and particularly to Fig. 2, this figure shows a complete unit in which the pump is enclosed within the rotor of an electric motor, preferably of the induction type adapted to operate on ordinary single phase alternating current. It is to be understood that although the invention has been illustrated in the form of a combination pump and electric motor, the novel features of the pump are capable of general application, and may be applied in

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other relations using other driving means, and further that the structure can be utilized as a motor as well as a pump if pressure medium is supplied to it in proper manner.

In Fig. 2 reference character 25 designates a flat base carrying an inverted bell-shaped housing 26 which may be set over it and preferably secured to it by welding peripherally as at 27. As here shown, the housing 26 carries a series of heat dissipating elements 28 secured to the casing at spaced intervals indicated 29 in the drawings.

Mounted within the casing is a stator 31 of an electric motor made up as is usual of laminations designed to provide minimum magnetic reluctance and freedom from eddy current losses. Disposed around the stator laminations 31 are stator windings 32 which will usually be made up in two sections to provide for starting and running conditions. While the particular type of motor employed is not important, as several types will be found useful in the present relation, it is preferred to use a motor of the capacitor type operating on the squirrel cage induction principle. A motor of this type lends itself readily to the present application because of the absence of movable mechanical parts, such as commutator short circuiting means, and its ability to utilize the metal structure of the pump cylinder as a support for the induction bars. For commercial installations where polyphase alternating current is available, a motor of axial air gap construction will be found applicable in simplifying the arrangement.

In the drawing the rotor of the electric motor is indicated at 34 as made up of a plurality of stacked laminations, supported between two spaced parallel end rings 35, preferably secured in position by induction heating, using some low melting point conducting metal such as silver. An arrangement of this character simplifies the construction because it reduces the time and cost of manufacture and provides for complete freedom from damaging the operating parts of the motor by the application of excessive heat. The conductor bars which join the end rings 35 are indicated in Fig. 4 by the reference character 36, and it will be understood that they are spaced circumferentially around the rotor in accordance with usual practice in order to provide adequate torque while insuring operating efficiency. The circuit for the stator windings has not been indicated because it forms no part of the present invention, being merely arranged to provide for the conduction of current to the windings 32 through binding posts or terminals such as are indicated at 33 in Fig. 4.

The rotating parts of the machine are carried by a central shaft 37 which is attached to the base plate 25 of the housing by a stud bolt 38 passing through a central aperture in the plate 25 and having a reduced threaded section 39 threaded into a correspondingly threaded portion of the lower end of shaft 37. The upper end of the shaft 37 terminates with the housing 26 to which it is secured by a hollow bolt 41 having an external threaded portion 42 which cooperates with an internally threaded bore in the upper end of the shaft. The bolt 41 contains a bore 43 which is threaded at its outer end to receive a suitable pipe fitting, and this bore serves as a gas inlet for the pump, delivering this gas to the pump through means later to be described.

The shaft 37 contains a central bore 44 at its lower end terminating in radial passages 45 com-

municating with the oil space in the bottom of the casing. The shaft adjacent its lower end carries a set of spiral grooves 46 serving for oil distribution in the usual manner. The central portion of the shaft 37 carries an eccentric 47 containing a transverse opening 48 connecting the bore 44 with an eccentric longitudinal bore 49 closed at the top but containing transverse openings 51, also serving for oil distribution and including peripheral grooves at the terminal ends of these openings. The parts 51 when filled with oil not only perform a lubricating function but also provide a sealing action between these moving parts to prevent pressure leakage from the pump chamber.

As indicated above, the shaft 37 carrying the eccentric 47 is stationary and serves as a bearing for the rotating parts of the pump. Referring especially to Fig. 1, the outer rotating cylinder comprising an annular metal member is designated 52. This cylinder is preferably of non-magnetic material such as brass or the like, and carries projecting radially inward from its inside surface and rigidly secured to it a vane 53. This vane, as indicated in detail in Fig. 12, is roughly of T-shaped formation and comprises a body 53 and an extension 58. The body 53, as shown in detail in Fig. 15, is wider than the extension and the extension projects centrally of the width of the body and from the inner end of the vane to make the vane symmetrical. The body carries mounting lugs 82 to hold it stationary and the body portions 80 extend into the end plates to assist this action. This vane is especially designed as indicated above for cooperation with a vane bearing 54 mounted in an opening 55 in the outer rim of rotor member 56. The rotor 56 is journaled to rotate about the eccentric 47 with which it is longitudinally coextensive as indicated in Fig. 2. This rotor is hollow and provides a lubricant chamber as indicated at 57 in Fig. 14. In addition, this hollow portion of the rotor member reduces the weight, and consequently the inertia, and provides clearance for the motion of the extension 58 provided on the vane 53. In other words, making the rotor hollow performs a three-fold function, namely, it forms an oil reservoir, reduces the weight, and avoids the necessity of machining operations to provide clearance for the movement of the extension 58 on the radial vane. This rotor, which is shown in detail in Fig. 3, also carries on each of its lateral faces an annular groove 59 serving for oil distribution and joining the opening 55 which carries the vane bearing 54. Inasmuch as it is highly desirable that this rotor be dynamically balanced, it contains a series of openings 61 conveniently cut in the top groove 59 so as to compensate for the removal of metal in forming the opening 55. Consequently when the vane bearing 54 is in position in the opening 55, the rotor with this bearing is dynamically balanced for smooth and efficient operation.

The cylinder 52 is attached to the rotor member 34 of the electric motor as herein shown with an interposed blanket 50 of heat insulating material. The electric rotor 34 during motor operation generates heat. Since the pump during the compression of gas necessarily generates heat, and it is desirable to keep the heat in the gas down to a minimum, the blanket 50 which may be of any poor heat conductor, with or without an air gap, will serve as a wall to isolate the motor rotor and the cylinder from each other. In some instances an air gap alone may serve this pur-

pose without the addition of other heat insulating means. In this way the operating efficiency of the pump may be kept at a maximum so far as the accumulation of heat is concerned. This cylinder is closed at its lower end by an end plate 62 shown in detail in Figs. 6 and 7. This end plate may be sealed to the cylinder 52 and carries a downwardly extending projection 63 which serves as a journal for the pump in cooperation with the shaft 37 and the oil groove 46 thereon. The extension 63 contains the radial grooves 45 on its lower face and is supported by a thrust bearing comprising upper and lower races 64 and 65 with interposed ball bearings 66. This end plate, when secured to the cylinder 52 to enclose the rotor as by welding at 67, provides the lower portion of the pump chamber seal except for the lubricant passageway formed by the grooves 46 and the oil duct 44 within the shaft 37. As indicated in the face view of Fig. 7, end plate 62 contains a discharge orifice 68 closed by a spring pressed discharge valve 69 which is secured to the end plate as at 71. Reference to Fig. 6 will indicate how this valve controls the discharge from the pressure chamber of the pump, and how the bias of the spring maintains the valve closed until the pump pressure has built up to a predetermined value. Consequently this valve also serves as a check valve to prevent backflow of gas from the dome to the pressure chamber of the pump.

The top side of the pump chamber is closed by an end plate 72 shown in side elevation in Fig. 5 and in section in Fig. 8. This end plate fits closely within the cylinder 52 and may conveniently be welded to it in sealing relation as indicated in respect to the end plate 62. This end plate contains a journal portion 73 adapted to embrace the upper portion of shaft 37 above the eccentric 47 and an annular chamber 74. The bore in this end plate communicates with the duct 43 arranged for the intake fitting. Communication between the duct 43 and the intake chamber of the pump is provided by bore 30 in the upper end of shaft 37 and a lateral port 40 which registers with the annular chamber 74 in the end plate 72. This annular chamber, communicating in turn with a longitudinal bore 75 and a second bore 76 terminating at the inner face of the end plate, provides communication between the gas inlet and the intake chamber of the pump. The path for the gas will be apparent from Fig. 6 of the drawing. It will be seen, therefore, from Figs. 4 and 6 that gas entering the inlet 43 can pass through duct 30 and port 40 through the end plate to the intake chamber of the pump, and after compression there will reach the pressure chamber and discharge through the port 68 and the valve 69 into the dome provided in the housing 26 and thence to a discharge pipe indicated at 77.

A characteristic of pumps of the present type has been that during rotation of the cylinder and rotor there is a rocking movement of the vane and its bearing with attendant wear. The present invention avoids this by providing the lugs 82 on the vane 53 which are set in sockets in the end plates and then having the ends of the vane also project into openings in the end plates so that the cylinder, the vane, and the two end plates constitute one unitary structure. This not only gives rigidity to the structure but avoids wear which would occur if the vane and its bearing were allowed to rock in the usual manner.

As indicated above, an outstanding feature of the present invention is the construction of the

vane 53 and its cooperating vane bearing 54. Through the peculiar formation and cooperation of these elements, it is possible to materially increase the displacement of the pump over prior art constructions of this same general type. For example, as suggested above, the vane 53 is generally of T-shaped form having a central extension 58. The bearing 54, on the other hand, is roughly cylindrical in form in order to fit closely in the opening 55 in rotor 56. During the operation of the pump the rotor 56 rocks on the vane bearing 54. This cylindrical member 54 contains a longitudinal slot or groove 78 extending throughout its length on the side through which the vane 53 projects into it with the parts assembled in the pump. The floor of the groove 78 contains a slot 79 long enough to accommodate the extension 58 on the vane 53 when the two parts are assembled as indicated in Fig. 15 of the drawing. The slot 79 is disposed centrally between the ends of the groove 78 and, as shown, closely engages the lateral faces of the vane 53 so as to provide the required sealing action. The ends of the slot do not engage the ends of the extension because oil must pass freely through this space as the extension moves into the slot. This relation of parts is clearly shown in Figs. 3 and 4 of the drawings. In addition, the bottom ends of the bearing 54 are cut away as at 81 to form lubricant pockets in cooperation with the walls of opening 55.

It will be apparent from an inspection of Fig. 15 that the body portion of the vane 53 is longer than the cylindrical bearing 54 so as to project into the end plates. The body portion 53 completely fills the groove 78, whereas the extension 58 works in the slot 79 and extends through it as indicated in dotted lines in Fig. 3 with the end clearance indicated in Fig. 4. Rigidity of the vane 53 is assured by its being securely mounted in the slotted portion of the cylinder 52, and by the lugs 82 which are supported in the end plates 72 and 82 as described.

As indicated above, an important feature and one which is outstanding with respect to the prior art, is the provision of a construction in which not only is the vane and the cooperating vane bearing prevented from any rocking movement, but the volumetric capacity of the pump is considerably enhanced over what constructions of the prior art permit. As set forth above, this may be accomplished by the use of peculiar vane and vane bearing construction such as that which has already been described. While that arrangement effectively prevents any rocking or canting of the vane and its bearing, an alternative manner in which this may be accomplished is that shown in Fig. 16 of the drawing.

Referring now to Fig. 16, the reference characters 83 and 84 designate the two end plates which are attached to the cylinder to seal the lateral sides of the pump chambers. These plates on their inner faces contain peripheral slots 85 adapted to receive and rigidly support lugs 86 carried by a vane bearing 86. As shown, each end of this bearing carries two spaced lugs 88, the spacing being such as to form a groove 91 and accommodate with a close fit the body portion of a vane 89 having a length sufficient to extend between the projecting bosses 88 and guided in a rigid straightline movement. The portions of the vane that are coextensive with lugs 88 are designated 93 in the drawing. The top edge of the vane 86 carries lugs 92 which fit into the cylinder and specifically into slots cut there-

in similarly to the lugs 82 on vane 53. In this way, as in the former construction, the vane and the bearing are held rigidly in position so that there is no possibility of canting or rocking movement of these parts during the operation of the pump. It will be noted, furthermore, that the vane bearing 86 is not only supported at its two ends in the sockets 85 in the end plates 83 and 84, but the floor of the groove 91 which receives the body of the vane 89 is solid as indicated by the dotted lines at 87. As will be pointed out hereinafter with respect to the diagrammatic views of Figs. 17, 18 and 19, this construction permits increase in capacity of the pump without reducing the metal sections to an undesired value. Accordingly, this arrangement offers all of the advantages which have been set forth in connection with the preferred construction already described, and even provides for a further increase in capacity over that construction.

It has been indicated that practice of the present invention greatly increases the amount of pump displacement which can be obtained with a structure of given size. This will be apparent from detailed consideration of Figs. 17, 18 and 19 wherein diagrammatic views are used to indicate the critical factors in the design of a pump of this type, both according to prior art teaching and according to the novel structure of the present invention.

For purposes of comparison Fig. 17 shows diagrammatically a radial vane pump employing a rectangular vane in which the vane in its retracted position always remains slightly below the geometrical center of the vane bearing in order to give it stability. In this figure 100 represents the maximum clearance between the rotor and cylinder when the vane 101 is retracted to its greatest extent. 102 represents the vane bearing having a slot 103, the diameter of the vane bearing being indicated N. V represents the distance which the vane 101 extends below the center of the vane bearing in the fully retracted position of the vane. M represents the space between the bottom of the vane 101 in its fully retracted position, and the bottom of the slot 103. W represents the section of metal which is left in the vane bearing after cutting the slot 103, this being one of the critical factors involved in producing a strong structure of the present type. 104 represents the shaft carrying the eccentric 105. The section of metal between the vane bearing 102 and the shaft is indicated Q. S represents the radius of the shaft and A the space between the center of the eccentric and the center of the shaft. R represents the radius of the pump cylinder.

Fig. 18 is a diagrammatic view of the principal form of the invention which has been described in Fig. 15. The reference characters used correspond to those used in the description and the letters correspond to those just given in connection with Fig. 17.

Fig. 19 is a diagrammatic view of the arrangement of Fig. 16 and similar reference characters are used to correlate the two figures.

By mathematical calculation, it may be shown in the arrangement of Fig. 17, if it is assumed that the radius of the pump cylinder is six inches, that sound engineering practice is followed in never retracting the vane 101 outwardly beyond the center of vane bearing 102, and that adequate metal sections W and Q are maintained, that the maximum pump displacement for a pump having a chamber length of 4 inches would

be 168.8612 cubic inches per revolution. In this calculation on Fig. 17, the following dimensions in inches are assumed:

R=6	X=1.25
S=0.5	V=0.25
Q=0.5	A=0.625
W=0.375	M=1.25
N=0.5	

With the T-shaped vane shown in Fig. 18, following the practice indicated above and never allowing the inner end of the extension 58 of the T-shaped vane to be withdrawn beyond the point 106, the displacement would be 235.9626 cubic inches per revolution. Here the assumed dimensions are:

R=6	X=1.850
S=0.5	V=absent
Q=0.925	A=0.925
W=0.375	M=1.850
N=0.5	

It is to be noted in this connection that the metal section W is the same in both constructions of Figs 17 and 18, and, even though structurally different, the effective amount of metal in section Q is substantially the same in both. Hence the construction of Fig. 18 not only increases the pump capacity by 67.1 cubic inches per revolution, but it positively eliminates canting of the vane bearing with respect to the vane and greatly enhances the life of the parts while insuring adequate sealing over long periods, and smoothness of operation which cannot be achieved when canting occurs.

Reference to Fig. 19 shows the displacement which can be obtained using the construction of Fig. 16 where the pump cylinder again has a radius of 6 inches and the longitudinal dimension of the pump chamber is 4 inches. The displacement here is 257.5620 cubic inches per revolution or an increase over the construction of Fig. 17 of 88.3 cubic inches per revolution. The dimensions assumed in Fig. 19 are:

R=6	X=2.062
S=0.5	V=absent
Q=0.5	A=1.031
W=0.375	M=2.062
N=0.5	

Here the metal sections W and Q are the same as those indicated in Fig. 17, and the vane 89 can be withdrawn beyond the center of the vane bearing 86 because this bearing is rigidly supported by the end lugs 88 in the end plates of the pump chamber. This manner of mounting prevents any possible canting of the vane bearing and hence again insures smooth, precise, and wear-free operation of the parts over long periods.

It will be clearly apparent and can be substantiated by mathematical proof that the embodiment set forth in the present specification provides a highly effective mechanism for obtaining maximum pump displacement with a minimum of structural parts, and with complete freedom from the usual pump leakage which is encouraged by rocking action of a vane with respect to its bearing. The space within the housing 26 serves as a receiver for pressure which is built up and which leads through the outlet 77 to the mechanism to be supplied with fluid under pressure. The casing contains oil which may usually be kept at the level indicated by dotted lines in Figs. 2

and 4 so that the inherent operation of the pump will supply the lubrication.

When the cylinder 52 is rotated as by supplying current to the stator windings of the electric motor, this cylinder rotates, carrying with it the vane 53 and the rotor 56, with the vane 53 moving out and in the groove 78 in the vane bearing 54. Gas will enter the intake 43 and after passing through the bore 30 and the lateral passage 40 will be delivered to the intake chamber on the upper side of vane 53 as indicated in Fig. 1 of the drawing, and then as the cylinder rotates in a clockwise direction this gas will be compressed until it escapes through the discharge valve 68 to the space within the pump chamber and then to the discharge line 77. The crescent-shaped chamber will always be divided into two parts by the tangential contact between the rotor 56 and the inside face of cylinder 52, and also by the vane 53. As has already been indicated, whenever the pressure in the pressure chamber of the pump exceeds the tension of spring 69, the discharge valve 68 will open, allowing the escape of gas under pressure, but preventing return flow into the pressure chamber of the pump. When, on the other hand, the pressure in the pressure chamber of the pump is less than the tension of spring 69, this valve remains closed.

It will be obvious that as the pump cylinder rotates, the pressure on the intake side of the pump will be less than the dome pressure, that is, the pressure inside of the housing. Hence, there will be a constant tendency for this pressure differential to cause flow of oil through the lateral openings 45 in the shaft extension 63, and through the duct 44 and space 48 to the interior of the rotor. Oil will also travel from the space 48 through the duct 49 and thence through the lateral ports into the spaces 51 between the journal 73 of the top end plate and the shaft extension. It will be noted that the extension 58 on the vane never is withdrawn beyond the center of the vane bearing 54, thus further assuring maintenance of straightline movement.

When the vane is moved from the fully retracted position of Figs. 1 and 2 to the position shown in Figs. 3 and 4, the extension 58 will force oil back into the oil spaces, insuring a continuous flow to the parts to be lubricated and also displacing oil from the top of the rotor into the groove 59 where it forms a fluid seal between the sides of the rotor and the two end plates 62 and 72. Thus it is possible to both lubricate and seal the pump by the inherent action of the pump parts and without the provision of any extraneous pumping means whatsoever. It therefore becomes possible to utilize the flow resulting from the differential between so-called dome pressure and intake pressure to fully lubricate and seal the working parts of the pump. It is therefore unnecessary to provide pumping means to increase the pressure above dome pressure.

It will be observed that the insulating means 50, be it in the form of a blanket or an air gap or both, will prevent any heat generated in the laminated structure 34 of the motor rotor from passing into the pump in order to increase the temperature of the gas which is under compression there. This freedom from heating up will likewise be enhanced when the pump cylinder 52 is made of non-magnetic metal in which no eddy currents can be produced.

It has been suggested above that the present construction makes it readily possible to utilize

pumps of different capacities without modifying the mechanical construction of the electric motor. This will be readily apparent from Fig. 2 if it is realized that the thickness of the rotor 52 may be varied in order to vary the diameter of the pump chamber. Likewise, variation in the thickness of the two end plates 62 and 72 will permit a substantial change in the longitudinal dimension of the pump without in any way altering the size of the mechanical parts of the electric motor. Since the pump operates at high speed, a very small change in longitudinal or radial dimensions will produce a substantial change in the pump capacity. Thus it will be possible by a single casing and motor structure to provide for a considerable number of pump capacities. By suitable design of the rotor and stator laminations a single laminated structure may be utilized for several sizes of pump. If the pump capacity is to be increased beyond the capacity of the windings, a simple change in the winding construction utilizing the same laminations may be provided so as to accomplish the results sought, namely, a universal motor and pump structure in which various sizes of pumps may be substituted directly in a given laminated construction without making it necessary for a great number of different part sizes to be carried.

Although only a few modifications have been described, it will be obvious that various changes and modifications of the invention may be made within the scope of the appended claims without departing from the spirit and scope of the invention. Having thus described the invention, what is claimed is:

1. A positive displacement radial vane pump comprising a fixed shaft, a hollow pump cylinder rotatable on said shaft, a pump rotor mounted eccentrically with respect to said shaft so as to maintain a point of tangency between its periphery and the inner surface of the cylinder as the cylinder and rotor rotate, said rotor having an internal annular groove forming an annular chamber about said shaft and said rotor also having a vane bearing opening extending through the periphery of the rotor at one point and communicating interiorly with said annular groove, a cylindrical vane bearing mounted in said rotor with a portion of the periphery of the bearing exposed through said one point, said bearing containing an exposed longitudinal groove throughout its length and a central elongated slot in the bottom of said groove, and a vane secured to said cylinder and extending radially inward therefrom, said vane having a wide body portion movable in the groove in said vane bearing, and a central extension projecting through the slot in the bottom of the groove with its sides in close sealing contact with the sides of said slot but with its ends spaced from the ends of the slot, said central extension being freely movable into the annular groove in said

rotor when the wide portion of the vane approaches the bottom of the groove in the vane bearing.

2. A pump of the radial vane positive displacement type comprising a symmetrically rotatable hollow cylinder having a radial vane extending inwardly therefrom, said vane having a wide body and a central projection, a symmetrically rotatable dynamically balanced hollow rotor having a cylindrical bearing receiving portion exposed at the periphery of the rotor and communicating with the hollow interior of the rotor, said rotor being mounted within said cylinder in eccentric relation to said cylinder, a stationary shaft for supporting said rotor and cylinder, said shaft having a bore therein communicating with the hollow interior of the rotor, a cylindrical vane bearing mounted in said bearing receiving portion of the rotor, said bearing having an exposed longitudinal groove to receive the wide body of the vane and a central elongated slot in the bottom of the groove to receive the projection on said vane, the sides of said slot closely engaging the sides of said vane but the ends of the slot being spaced from the edges of said vane, the projection on said vane being adapted to move in and out of said hollow rotor through said central slot as the cylinder and rotor rotate, a housing enclosing said cylinder, rotor and parts before mentioned, and connections for causing the pressure differential between the interior of said rotor and the interior of the housing to supply oil from the bottom of said housing through the bore in said shaft to said hollow rotor, said projection in its movement into and out of said hollow rotor serving to displace oil therefrom.

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