June 17, 1941. W. W. DAVIDSON 2,246,276

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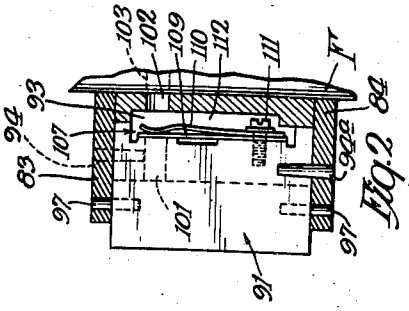


Fig. 2

Fig. 1

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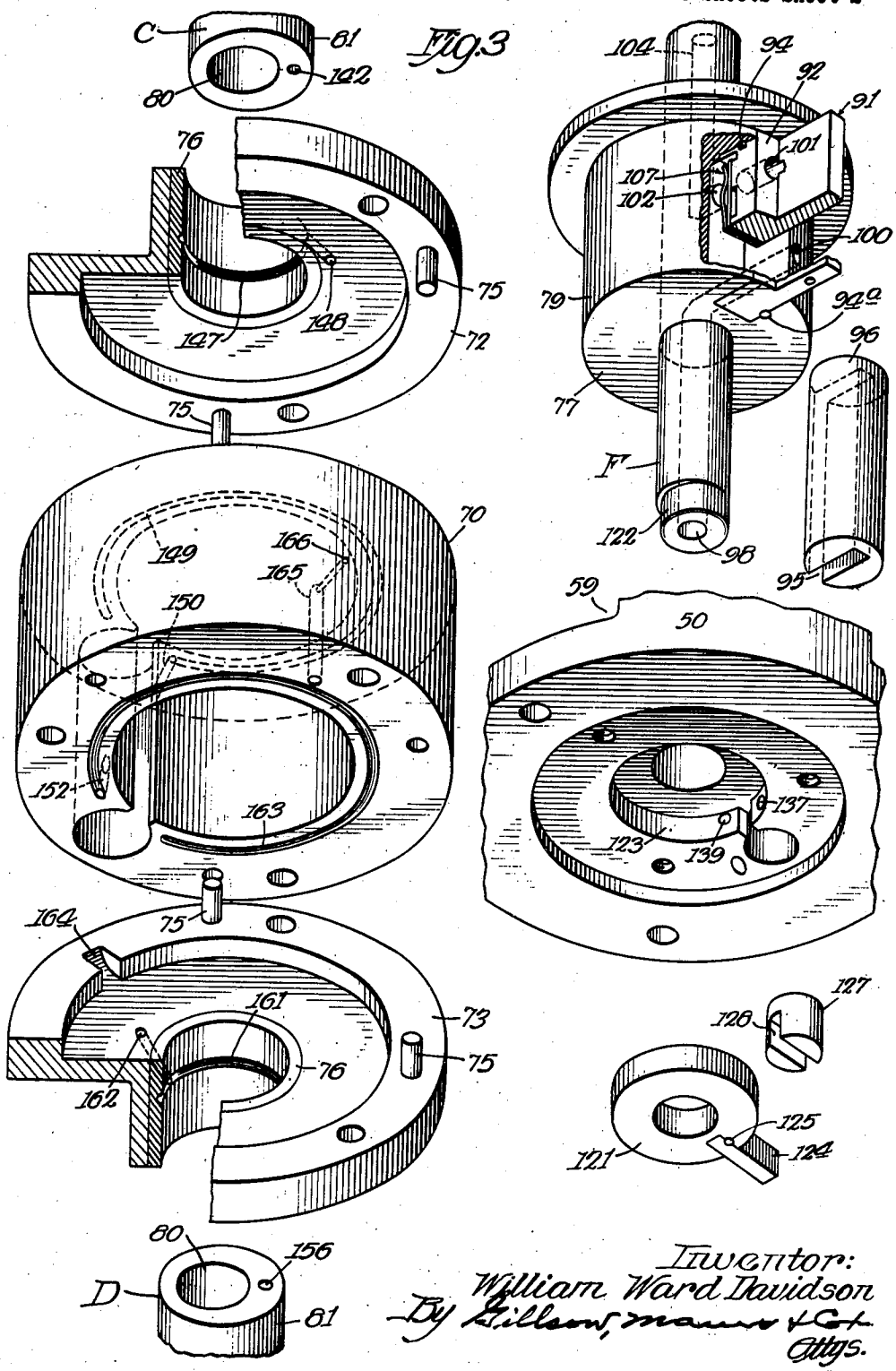


Fig. 3

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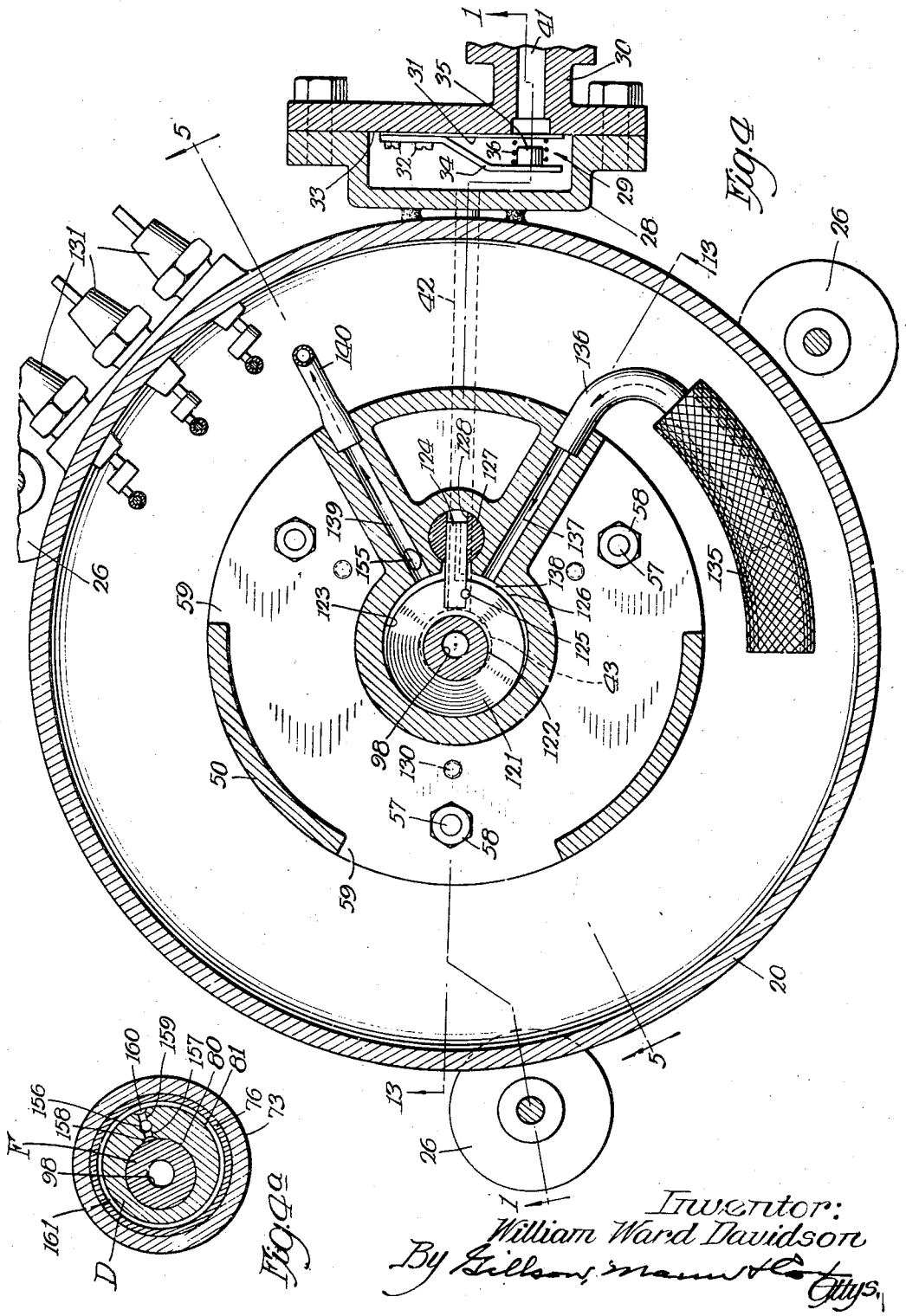


Fig. 4

Fig. 4a

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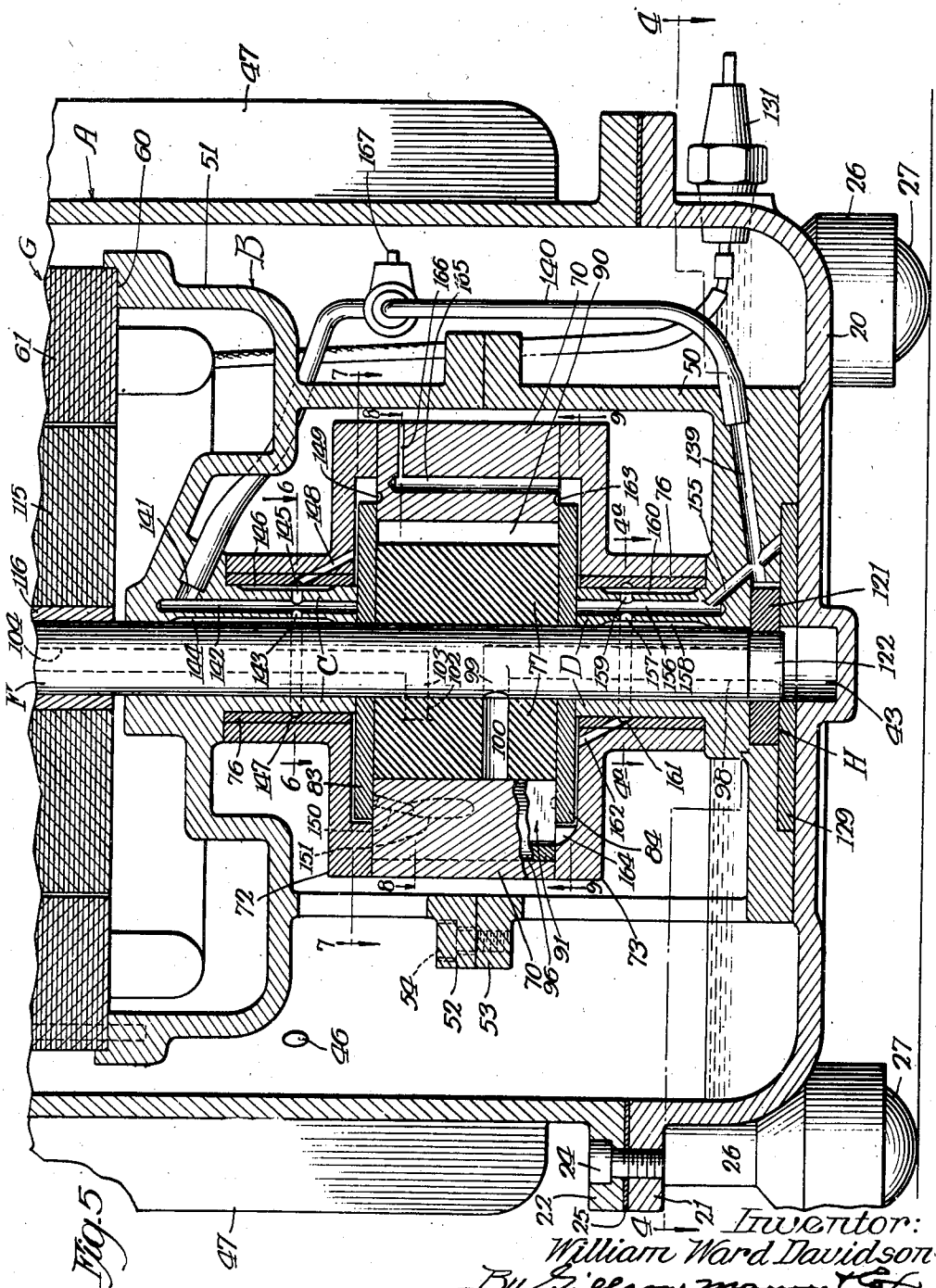


Fig. 5

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June 17, 1941. W. W. DAVIDSON 2,246,276

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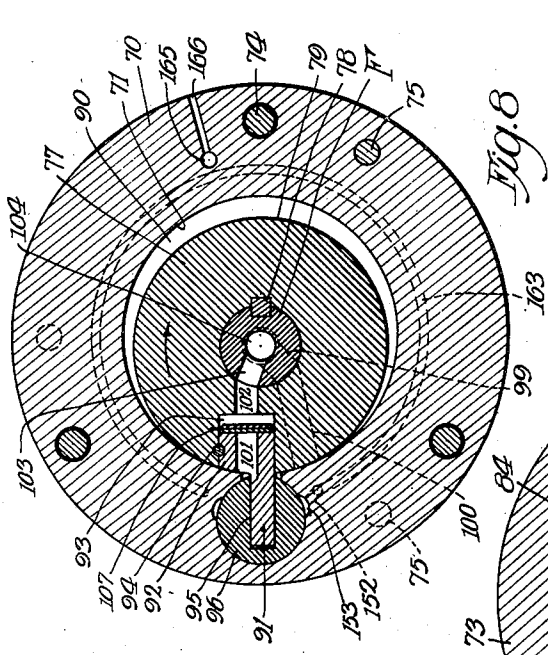


Fig. 8

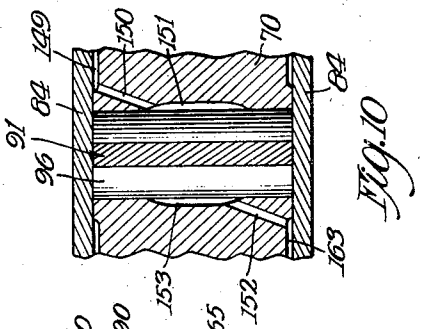


Fig. 10

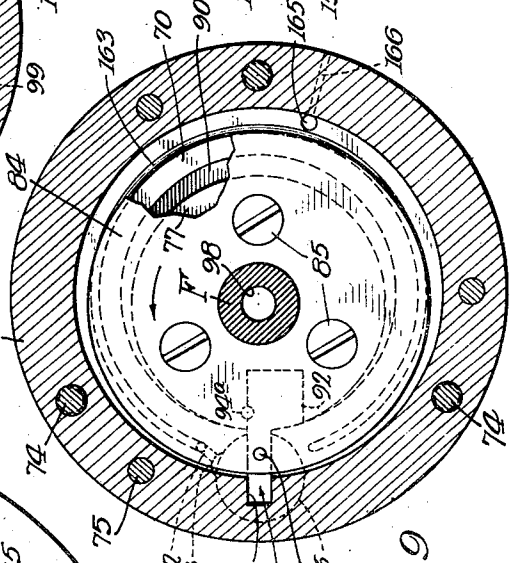


Fig. 9

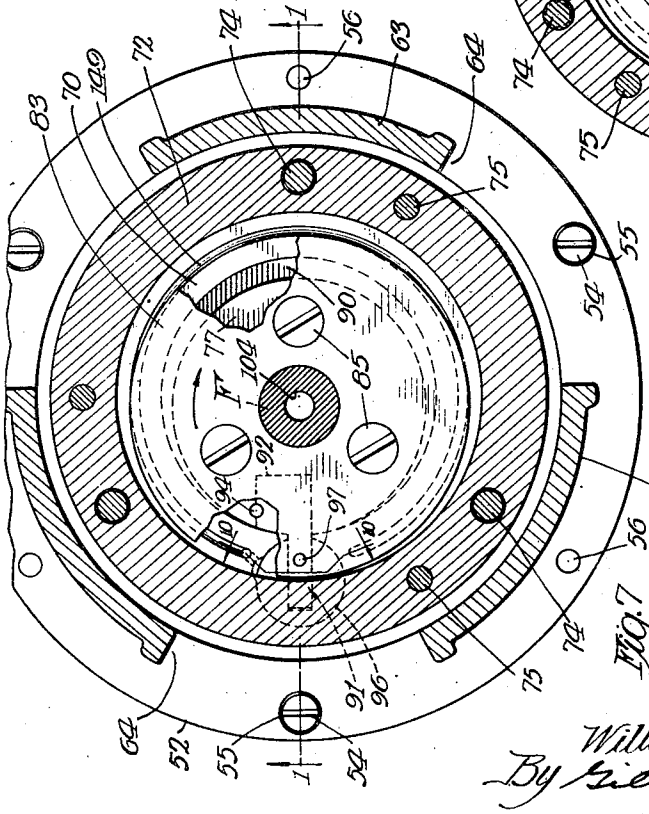


Fig. 7

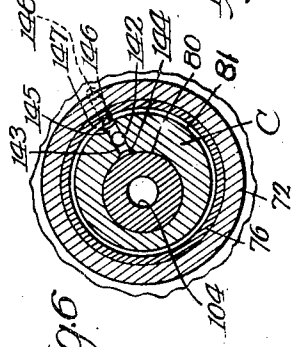


Fig. 6

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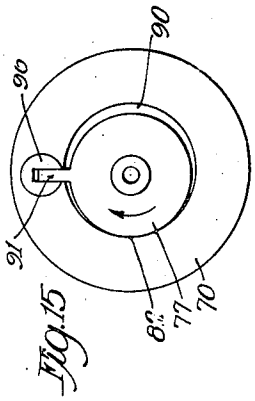


Fig. 15

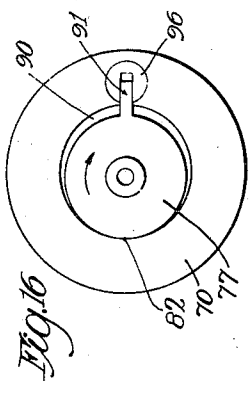


Fig. 16

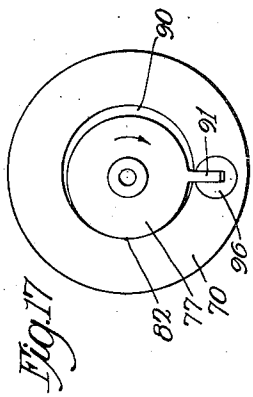


Fig. 17

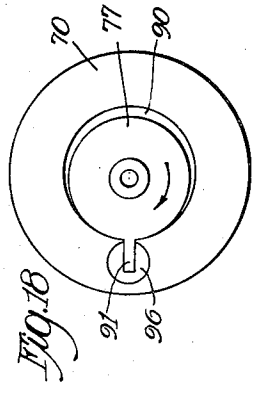


Fig. 18

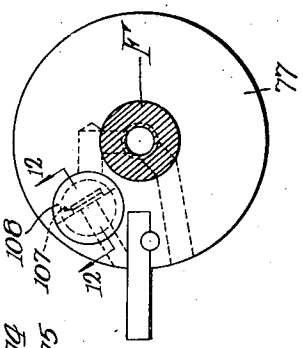


Fig. 11

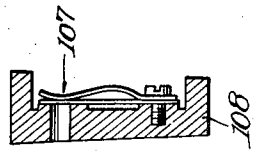


Fig. 12

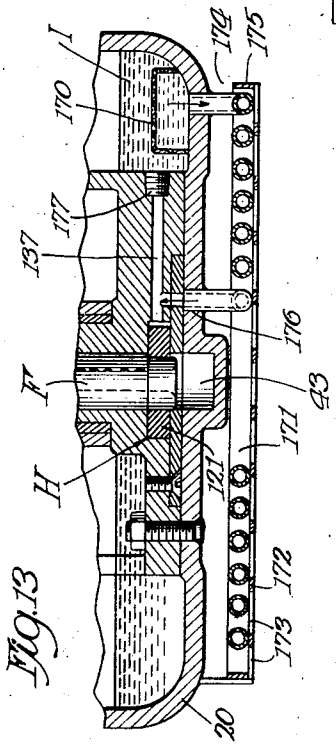


Fig. 13

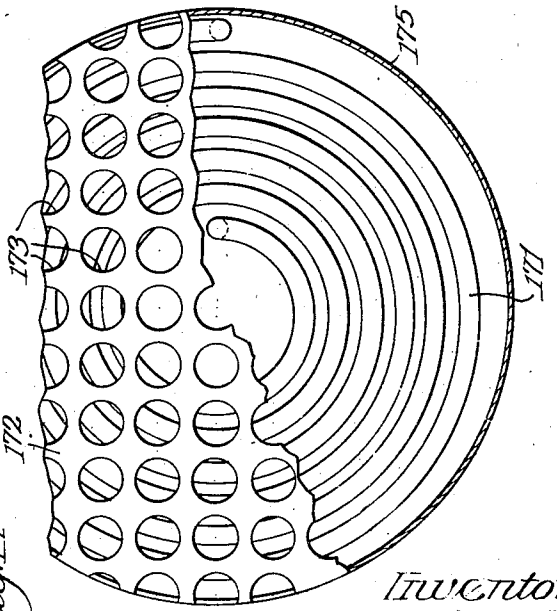


Fig. 14

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Patented June 17, 1941 2,246,276

UNITED STATES PATENT OFFICE

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PUMP

William Ward Davidson, Evanston, Ill.

Application January 20, 1938, Serial No. 185,830

16 Claims. (Cl. 103—121)

A rotary pump, and for that matter any other type of a pump, to be commercially successful must meet certain rather well defined standards, and this is particularly true for moderate size refrigerant compressors, the market for which is highly competitive.

Generally speaking, these necessary attributes come under five headings, to-wit:

1. The pump must be able to pull a satisfactory vacuum when operating against either no head pressure or a head pressure corresponding to operating conditions.

2. The pump must meet certain requirements with respect to overall efficiency, i. e. a pump of given displacement must stay within specified power limits when operating at various head pressures and, in the case of refrigerant compressors, the ratio of B. t. u. output to power input must stay within certain limits.

3. The pump must substantially maintain its overall efficiency under high temperature operating conditions, and it must be so constructed that after it has been run at the high temperatures for a considerable period of time, and then stopped, it will be able to start again within a reasonable period even though there are forces acting which tend to oppose starting, such as lower motor efficiency, smaller clearances between the relatively movable pump parts, high head pressures, static friction, etc. Furthermore, when the pump is started again under these conditions, the oil sealing and lubricating system should positively supply oil to the clearance spaces almost simultaneously with the starting of the pump, so that the pump will at all times have its desirable operating efficiency.

4. The pump should be so constructed that it is as nearly dynamically balanced as possible, and all relatively movable parts should be so arranged and lubricated that the pump as a whole maintains its high efficiency and reliable performance over a long period of time.

5. The pump must have a minimum number of parts, all of which are easily machined and readily assembled.

The manner in which the pump of this invention achieves all of the above mentioned desirable qualities of a commercially successful pump will become apparent as the disclosure proceeds and the description is read in conjunction with the accompanying drawings, in which

Fig. 1 is a vertical, cross-sectional view taken on the lines 1—1 of Figs. 4 and 7, (the latter figure, however, omitting the outer casing);

Fig. 2 is an enlarged detail view showing the check valve for the discharge passage of the pump;

Fig. 3 is an enlarged exploded view of the principal pump members, all of which are shown in perspective;

Fig. 4 is a horizontal sectional view looking down on the oil pump, the section being taken on the lines 4—4 of Figs. 1 and 5;

Fig. 4a is a transverse-sectional view through the lower pump bearing, the section being taken on the line 4a—4a of Fig. 5;

Fig. 5 is a vertical, sectional view taken on the line 5—5 of Fig. 4 and showing particularly the sealing and lubricating system, the remainder of the pump structure being shown somewhat diagrammatically.

Fig. 6 is a transverse section through the upper pump bearing, the section being taken on the line 6—6 of Fig. 5;

Fig. 7 is a horizontal, sectional view through the inner assembly, the section being taken on the line 7—7 of Fig. 5;

Figs. 8 and 9 are horizontal, sectional views taken on the lines 8—8 and 9—9, respectively, of Fig. 5;

Fig. 10 is a fragmentary sectional view taken on the line 10—10 of Fig. 7;

Fig. 11 is a horizontal, sectional view showing a slightly different arrangement for the check valve on the discharge side of the pump;

Fig. 12 is a sectional view taken on the line 12—12 of Fig. 11;

Fig. 13 is a fragmentary, vertical sectional view taken on the line 13—13 of Fig. 4, showing the manner in which an oil cooling system may be incorporated into the pump;

Fig. 14 is a fragmentary bottom plan view of the oil cooling system; and

Figs. 15–18 inclusive are diagrammatic views illustrating the operation of the main pump E.

It should be understood that the description and illustration of what is now considered a preferred form of invention is merely for the purpose of complying with sec. 4888 of the Revised Statutes, and that the appended claims should be construed as broadly as the prior art will permit.

The pump of this invention, while it has many uses, is primarily intended as a refrigerant compressor, and it peculiarly satisfies all of the practical requirements for a commercial product.

General organization

The pump in general comprises an outer casing or dome A within which the entire pump assembly is hermetically sealed. The inner assembly consists of a frame B having upper and lower bear-

ings C and D, respectively, supporting a main pump unit E. This unit is driven through a shaft F by a standard electric motor G mounted above the pump E. The principal moving parts of the pump E are effectively sealed against gas leakage as well as being lubricated by a sealing and lubricating system which includes an oil pump H operated from the lower end of the shaft F and submerged in an oil reservoir I at the bottom of the casing A.

The outer casing or dome

The outer casing A consists of a base **20** which may either be cast, or stamped from a sheet of cold rolled steel of suitable thickness. In either case, the base preferably has a flange **21** to which a corresponding flange **22** of the dome cover **23** is rigidly and hermetically sealed. The union may be made by bolting the dome cover to the base, as indicated at **24** (in which case a gasket **25** is preferably interposed between the flanges), or, if desired, the bolts and gasket may be omitted and the juncture of the two flanges may be welded. The base is supported on a plurality of feet **26**, each of which preferably includes a resilient element, such as a rubber ball **27**, for preventing vibration and sound from being transmitted from the casing to the part upon which the pump as a whole is supported.

On one side of the base **20**, a small housing **28** is spot welded within which an intake check valve, generally designated **29**, is located, the latter being mounted on a coupling member **30** and comprising (as best shown in Figs. 1 and 4) a flap valve **31** formed of spring steel and fixed at one end by screws **32** to the face **33** of the coupling member **30**. The screws **32** also secure a spring retainer and stop **34** to the coupling member **30**, the outer end being bent outwardly as shown in Fig. 4 and provided with a stud **35** to receive a spring **36** which bears against the flap valve **31** and tends to hold it in closed position. The stud **35**, in addition to supporting the spring **36**, limits the opening of the flap valve **31**.

To the opposed face **37** of the coupling member **30**, a conventional compressor shut off valve **38** is secured by bolts **39** and the feed line to the compressor includes a tube **40** leading to the shut off valve, a passageway **41** through the coupling member **30**, and a copper tube **42** connecting the housing **28** with a recess **43** in a depressed portion of the base **20**. Preferably, the tube **42** is silver-soldered to the housing **28** and the base **20** in a manner well known to the art, to form a gas tight seal.

On one side of the dome cover **23** a second compressor shut off valve **44** is mounted, and the discharge line from the compressor includes a tube **45** which communicates through the compressor shut off valve **44** with the interior of the dome through an opening **46**. Preferably, the dome cover **23** is equipped with a plurality of radially extending vertical fins **47** to assist in dissipating the heat from the dome.

Supporting frame for pump and motor

The supporting frame B, for the main pump E, motor G and the oil pump H is preferably a two part semi-steel casting consisting of a lower half **50** and an upper half **51**. Since the lower bearing D for the pump E is carried by the lower casting **50**, and the upper bearing C is carried by the upper casting **51**, it is important to provide means for securing the two parts of the frame B together in a predetermined angular relationship. This is accomplished by providing mating flanges **52** and **53** on the upper and lower castings, respectively, the two flanges being held together by three or more screws **54**, the heads of which are countersunk into the flanges **52** and have clearances **55** to permit the true angular alinement of the castings to be determined by two or more dowel pins **56** which have a tight press fit in one of the flanges **52** or **53**, and a light press fit in the other. The dowel pins are accurately located so that when the two castings are secured together, the bearings C and D which support the main pump E are in exact alinement.

The lower casting **50** has its lower face machined to rest upon a slightly raised machined portion of the base **20**, and the casting is held in fixed relationship to the base by upstanding studs **57** welded into the base and having nuts **58** which clamp the casting **50** in place. To save weight and to provide for additional space for oil, portions of the side wall of the lower casting **50** are cut away as indicated at **59**, as best shown in Figs. 1 and 4.

The upper casting **51** of the frame B has an annular ledge **60** adapted to support the stator **61** of the motor G. The lower portion of the winding of the stator extends down into an annular trough **62** in the casting **51**, and the connecting wall **63** between the trough **62** and the flange **52** is preferably cut away at intervals as indicated at **64**, to conserve weight. The stator **61** is held to the frame B by bolts **65** as shown in Figs. 1 and 5.

The main pump

The main pump E is preferably of the rotating cylinder type in which two eccentrically mounted pump members rotate on their own centers in a manner to produce a pumping action.

In the embodiment of the invention shown, the pump comprises a cylinder ring **70** preferably made of semi-steel having an accurately machined inner cylindrical surface **71**, the ring being supported for rotation about an axis defined by the outer bearing surface of the bearings C and D by cylinder heads **72** and **73** which are clamped to the ring by elongated machine screws **74** having slight marginal clearances with the upper head **72** and the ring **70**, in order that the angular position of the ring **70** with respect to the bearings C and D may be accurately determined by three or more dowels **75** extending between the ring **70** and two cylinder heads **72** and **73**. The dowels have a tight press fit in the cylinder heads and a light press fit in the cylinder ring for convenience of manufacture and assembly.

Since the cylinder heads **72** and **73** are preferably made of semi-steel, it is desirable to drive hardened steel bushings **76** into the hubs of the heads to assure long life for the bearing surfaces.

Within the cylinder ring **70** is an eccentrically mounted rotor **77** which is also preferably made of semi-steel, the rotor being keyed to the shaft F as indicated at **78** and having its outer surface **79** concentric with the shaft. The shaft, which is preferably made of hard steel is mounted in the bearings C and D so that the rotor may revolve about an axis defined by the inner bearing surface **80** of the upper and lower bearings C and D, and the axis of this bearing surface bears such relationship to the axis of the outer bearing surface **81** that the rotor has a point of tangency **82** (see Figs. 15–18 inclusive) with the inner cylindrical surface **71** of the ring **70**.

Hardened steel end plates **83**, **84** are fastened

by screws 85 to the ends of the rotor and these plates are accurately machined to provide a close sliding fit with the correspondingly machined planar surfaces 86 flanking the ring 70. The plates bear against the bearings C and D, but have slight clearances with the cylinder members 72 and 73 as best shown in Fig. 1.

The crescent-shaped space 90 between the ring 70 and rotor 77 is divided into intake and discharge sides by a vane generally designated 91, preferably of hard steel, which has an enlarged portion 92 drive fitted into a recess 93 in the rotor 77. The vane is securely held in the rotor by taper pins 94 and 94a which are driven tightly into place.

The portion of the vane which projects beyond the rotor is carefully machined to have a close sliding fit in a slot 95 of a rocker 96, the latter being of cylindrical form and rotatably mounted in the cylinder ring 70. The vane may, therefore, be said to have a tight fit with respect to one of the pump members and a rocking and sliding engagement with the other member.

The vane has a gas tight fit with the end plates 83 and 84, and this fit is assured by small dowels 97 which are driven through the plates into the vane. If desired, the portion of the vane which projects from the rotor may be made slightly wider and fitted within slots in the end plates in order to more perfectly effect a gas tight seal between the vane and the end plates.

The fluid intake into the pump E includes in addition to the feed pipe 40, tube 42 and recess 43, an axial passage 98 which communicates with a radial bore 99 in the shaft and another bore 100 in the rotor which leads into the intake side of the pump chamber. The discharge passageway from the pump chamber includes a bore 101 in the enlarged portion 92 of the vane, the recess 93 and bore 102 in the rotor, a radial bore 103 in the shaft and an axial bore 104 in the shaft which extends upwardly and connects with the chamber 105 of a combination lock nut and oil separator, generally designated 106.

A check valve, generally designated 107, is interposed in the discharge passageway, either at the base of the vane, as shown in Figs. 1 and 2 particularly, or preferably in a separately formed member or plug 108, as shown in Fig. 11. The valve in both cases consists of a flap 109 which is normally held on its seat by a leaf spring 110, the ends of the flap 109 and spring 110 being secured in place by a screw 111. When the check valve is located at the base of the vane, as shown in Figs. 1 and 2, a clearance space 112 is provided in the adjacent portion of the rotor to permit opening of the valve.

There is an advantage in placing the check valve 107 in a plug which is completely disassociated from the vane, as shown in Fig. 11. With this modification, the vane can be machined in a cross-sectionally rectangular shape of the simplest form. The discharge passage for the pump in this embodiment comprises two chordal bores in communication with each other through an intermediate longitudinal bore running parallel to the axis of the rotor. The remote ends of the chordal bores terminate, one upon the cylindrical surface 79 proximate the vane, and the other in a radial bore leading to the axial passageway 104. The plug 108 is driven into the longitudinal bore with the valve 107 opening inwardly away from the pump chamber 90. The plug 108 is preferably tapered, as shown in Fig. 12, and the rear half of the plug is cut away to provide the necessary room for the check valve.

It will be understood that as the shaft F is rotated in the direction shown by the arrows in Figs. 15–18 inclusive, the vane 91 sweeps through the crescent-shaped pump chamber and displaces fluid to effect a pumping action. Simultaneously, the intake side of the pump is drawing in additional fluid to be compressed on the next cycle of operation.

Motor drive

The motor G which drives the shaft F is preferably a standard alternating current hermetic motor of the capacitor start, induction run type. The rotor 115 is mounted on a sleeve 116 which has an instanding flange 117 adapted to rest on a shoulder 118 of the shaft, so that the rotor is supported in appropriate relation to the stator 61. The motor rotor 115 is clamped in place by the lock nut 106 which is screw-threaded onto the threaded end 119 of the shaft and preferably, though not necessarily, a lock washer 120 is used to lock the cap on the shaft.

The leads for the motor G are brought into the dome through gas tight terminals 131 (Fig. 4) of conventional make, the terminals preferably being located in the base 20 of the dome or outer casing.

Sealing and lubricating system

In order to have a highly efficient pump, it is necessary to effectively lubricate the moving parts and seal them against gas leakage. In the present invention, this is accomplished by providing a positive displacement oil pump which forces the oil taken from a reservoir I through a series of passageways into the bearing surfaces and the various clearance spaces where there is a gas pressure differential which tends to cause gas leakage.

The oil pump is best shown in Figs. 1 and 4, and it comprises a rotor 121 which is mounted on an eccentrically formed reduced portion 122 of the shaft F. The rotor 121 travels in a planetary manner within a cylindrical recess 123 formed in the base of the lower casting 50 of the supporting frame B and a vane 124 fixed by a dowel 125 in the rotor 121 has sliding and rocking engagement with the casting 50 to divide the crescent-shaped pump chamber 126 into intake and discharge stages. The sliding and rocking engagement of the vane with the casting 50 is accomplished by means of a cylindrical rocker 127 which is recessed into the base of the casting and has a slot 128 to receive the vane. The rear of the slot 128 communicates with the intake side of the oil pump by means of a small passage, shown in dotted lines in Figs. 1 and 4, so that oil cannot be trapped behind the vane as it reciprocates within the rocker 127.

The bottom of the pump chamber is closed by a plate 129 which is suitably secured to the casting, as by means of screws 130. Preferably the rotor 121, the vane 124, the rocker 127 and the end plate 129 are all made of hardened steel.

In the form of the invention shown in Fig. 4, oil is taken from the reservoir or sump I through a screen 135, tube 136 and bore 137 in the lower casting 50 in to the low pressure side 138 of the oil pump. The planetary movement of the rotor 121 forces the oil through a series of connecting passageways to the parts to be lubricated and sealed against gas leakage. The pump H being a positive displacement pump will force the oil through these passages and drive it into the spaces

where lubrication is necessary and where the continual supply of a film of oil to relatively movable parts across the surface of which gas has a tendency to escape due to pressure differentials will make gas leakage impossible and thereby increase the volumetric efficiency of the pump.

The upper part of the main pump is lubricated and sealed by oil passing successively through a discharge passage 139 in the lower casting 50, a tube 140 which connects the passage 139 with the bore 141 in the upper casting 51, bore 142 extending vertically through the bearing C and various branch bores and passageways, one of which 143 communicates with a vertical groove 144 to lubricate and seal the upper portion of the shaft F and another bore 145 communicating with a vertical groove 146 and an annular groove 147 to lubricate and seal the bearing which supports the upper cylinder head 72.

A diagonal bore 148 in the hub of the cylinder head 72 connects the grooves 146 and 147 with the interior of the cylinder head and thus constantly fills the space between the upper cylinder head 72 and the end plate 83 and ring 70 with oil under pressure. A circular groove 149 in the upper face of the ring 70 assures that the underside of the plate 83 is properly lubricated and sealed against gas leakage, and this circular groove communicates through a diagonal bore 150 with a recess 151 to supply oil under pressure to the rocker 96.

The lower part of the pump E is lubricated and sealed by a series of passageways which include a portion of the discharge passageway 139, a diagonal bore 155 in the lower casting 50, a vertical bore 156 in the bearing D, and various branch passageways which communicate with this latter bore. One such passageway 157 connects the bore 156 with a vertical groove 158 which lubricates and seals the lower end of the shaft. Another such passageway 159 connects the bore 156 with a vertical groove 160 and an annular groove 161, both of which serve to lubricate and seal the lower supporting bearing for the lower cylinder head 73.

A diagonal passageway 162 connects the annular groove 161 with the space between the lower cylinder head 73 and the lower plate 84 and ring 70, and this space is kept full of oil under pressure by the operation of the oil pump H.

An annular groove 163 in the lower surface of the ring 70 makes certain that the top surface of the plate 84 is effectively lubricated and sealed against gas leakage, and this groove communicates with the rocker 96 through passageways 152 and 153.

Oil is permitted to flow freely into the rear portion of the slot 95 of the rocker 96 to lubricate and seal the vane 91, and to avoid setting up a damaging back pressure in the lubricating system, the slot 95 being in free communication with the space between the lower cylinder head 73 and adjacent end plate 84 by means of a passage 164 which is cut through the upstanding flange of the lower cylinder head 73 (see Figs. 3 and 5).

To further avoid the possibility of creating undesirable back pressures in the lubricating and sealing system which would not only damage the overall efficiency of the pump, but also tend to make the pump noisy, the space between the lower cylinder head 73 and the end plate 84 is connected by a vertical bore 165 in the ring 70 with a restricted relief passage 166 communicating with the interior of the dome. It has been found that when the size of this passage is correctly proportioned for the displacement of the oil pump and the size of the storage passageways, the overall efficiency of the pump may be maintained at a high standard and still have a quietly operating pump. The amount of oil to be discharged through the opening 166 must be determined empirically for a pump of given size and proportions, but it is sufficient for this disclosure to know that the relief passageway must be small enough to maintain the desired pressure in the oil system, and yet large enough to avoid setting up undesirable back pressure and causing the pump to operate noisily.

For the same purpose, the tube 140 is provided with a relief vent 167, the size of which is also empirically determined to suit conditions.

The cross sectional areas of the relief vents 166 and 167 are more or less complementary, for their total cross sectional area determines the oil pressure in the lubricating and gas sealing system.

Under some circumstances, it may be desirable to provide an oil cooling system, and this may be conveniently done in the manner shown in Figs. 13 and 14. As there shown, oil is taken from the reservoir I and passed downwardly through a screen 170 into a spiral copper tubing coil 171 which is supported beneath the base 20 by a cup-shaped brass or copper plate 172 which is soldered, or otherwise secured by a metal bond to the tubing 171 to assist in the heat dissipation, and is preferably provided with perforations 173 to facilitate the circulation of air over the tubing. A space 174 is left between the top of the marginal flange 175 around the plate 172 so that air may freely circulate through the openings 173 over the tubing 171 and out through the space 174.

The coiled tubing 171 re-enters the dome through an opening 176 where it communicates with the intake passage 137. A plug 177 closes the passageway 137 to the space within the dome.

Assembly and accessibility of parts for repair and replacement

In the manufacture of the pump and its assembly, the preferred routine is as follows: First the lower casting 50 with the oil pump H fitted into place is secured to the base 20 of the dome. The main pump E is then built up by taking the shaft F, keying the rotor 77 and the lower end plate 84 to the shaft with the vane 91 in place. The cylinder ring 70 is then laid on the plate 84 with the vane engaging the rocker 96, and the top end plate 83 is then screwed into place (it may here be noted that the projecting end plates 83 and 84 when secured to the rotor 77 may conveniently be used as a gauge in obtaining the correct depth of the ring 70 during its machining). The upper cylinder head 73 is then slipped into place and the cylinder heads are clamped together by the elongated machine screws 74.

The pump assembly with the shaft is then assembled on the lower frame by passing the reduced eccentrically turned end of the shaft F into the opening of the oil pump rotor 121 (the end of the shaft being slightly rounded, as indicated at 168 to facilitate this part of the assembly) and the top frame including the upper casting 51 and the stator is then clamped to the lower frame by the screws 54, the dowels 56 determining the proper angular relationship between the upper and lower frame. The rotor is then slipped over the top of the shaft and the oil separator and retaining nut 106 is screwed into place.

From this brief explanation of the preferred method of assembly of the pump, it is obvious that when necessity demands, the casing may be opened and the motor readily replaced, or even the entire pump unit E may be removed for inspection and repair.

Summary of advantages

The following advantages are present in the pump structure which has been described and shown in the drawings:

1. The eccentric 122 on the shaft F may bear such angular relationship with the point of tangency 82 between the rotor 77 and the cylinder ring 70 that the oil pump may be made to deliver its maximum pressure during a given cycle at any predetermined part of the cycle of the main pump. In other words, if desired, the moment when the oil pump H delivers its maximum pressure during a given cycle of operation may be made to coincide with (or slightly precede) the moment when the main pump E develops its greatest pressure in the pump chamber, so that maximum sealage is obtained.

2. Centrifugal force aids in closing the check valve 107 in the discharge passage of the pump.

3. By driving the inner pump member and arranging the vane 91 so that it has a rocking and sliding engagement with the outer cylindrical member, it is possible to construct a pump of given theoretical displacement with less material than if the outer member were driven and the vane had rocking and sliding engagement with the inner member.

4. Simplicity of pump structure is obtained by having the intake passage for the main pump extend axially through the shaft in one direction and the discharge passage extend axially through the shaft in the other direction.

5. By placing the oil pump below the oil level, the pump is always full of oil so that whenever the pump starts, it immediately begins to pump oil. This is accomplished without actually having either the oil pump or the main pump rotate in the oil reservoir.

6. The physical separation of the gas and oil pump makes it impossible for gas to leak into the oil pump and thereby interfere with lubrication and sealing.

7. The main pump E can be used as a rotary engine with little or no modification. This is in contradistinction to many prior art rotary engines which cannot be made to operate as pumps without radical reorganization.

Moreover, the particular combination of pump parts, their relationship to one another—all in combination with an efficient lubricating and gas sealing system—constitute an important part of this invention, and failure to describe any such combination or relationship of parts, although shown in the drawings, should not be interpreted as revealing a lack of appreciation of their importance or advantage. Although certain preferred embodiments of the invention have been shown and described herein, it will be apparent to those skilled in the art that various and further uses, modifications and changes may be made without departing from the spirit and substance of the invention, the scope of which is commensurate with the appended claims.

I claim:

1. In a rotary pump or the like, pump members comprising a cylinder, a rotor mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the rotor and having sliding engagement with the cylinder, a vane extending between the rotor and the cylinder dividing the crescent-shaped space therebetween into intake and discharge stages having inlet and oulet ports respectively, said vane being rigidly fixed in one of the members and having rocking and sliding engagement with the other, bearing means for the cylinder extending laterally therefrom, bearing means for the rotor, and means for producing a pumping action between the rotor and the cylinder.

2. In a rotary pump of the type having a crescent-shaped pump chamber, an outer pump member having an inner cylindrical surface flanked with planar surfaces normal to the axis of the cylindrical surface, an inner pump member of cylindrical form mounted in eccentric tangential relation to the inner surface of the first mentioned pump member, a vane extending between the two members dividing the pump chamber into intake and discharge stages having inlet and outlet ports respectively, said vane being rigidly fixed in one of the members and having rocking and sliding engagement with the other, end walls rigidly secured to the inner pump member having sealed engagement with the vane and sliding engagement with the planar surfaces of the outer pump member, bearing means for the outer pump member extending laterally therefrom, bearing means for the inner pump member, and means for producing a pumping action between the pump members.

3. In a rotary pump or the like, pump members comprising a cylinder, a rotor mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the rotor and having sliding engagement with the cylinder, a vane rigidly secured in sealed engagement to the end walls and the rotor and having rocking and sliding engagement with the cylinder to divide the crescent-shaped space therebetween into intake and discharge sections having inlet and discharge ports respectively, and means for producing a pumping action between the rotor and the cylinder.

4. For use in a pump of the class described, a rotor having secured thereto flanking end walls of greater diameter than the rotor itself, and a vane rigidly fixed relative to the end walls and extending radially outward from the rotor.

5. In a rotary pump or the like, pump members comprising a cylinder, a rotor mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the rotor and having sliding engagement with the cylinder, a vane rigidly secured to the rotor and end walls in sealed engagement therewith and extending between the rotor and the cylinder for dividing the crescent-shaped space therebetween into intake and discharge stages having inlet and outlet ports respectively, and means including the vane for positively rotating both the rotor and the cylinder to produce a pumping action between the rotor and the cylinder.

6. In a rotary pump of the type including an outer rotatable member having planar surfaces at its ends, a cylindrical rotor eccentrically mounted in tangential relation to an inner cylindrical surface of the outer rotatable member, the combination therewith of end walls rigidly fixed to the rotor and arranged for sliding engagement with the planar surfaces on the outer member, a vane rigidly fixed to the rotor having a tight fit with the end walls and having rocking

and sliding engagement with the cylindrical surface, inlet and outlet ports upon opposite sides of the vane, and means for driving the rotor.

7. In a rotary pump of the class described, a sealed casing, a pump unit in the casing comprising an oil pump having a stationary outer cylinder, a gas pump having a rotatable outer cylinder, and an electric motor rotor all rotatable on a common vertical axis with the latter two surmounting the oil pump, a quantity of oil in the casing sufficient to submerge the oil pump enough to render it promptly effective to deliver oil directly to the gas pump on starting but not sufficient to submerge a substantial part of the rotating cylinder of the gas pump, said gas pump including a rotor eccentrically mounted in tangential relation with the cylinder and having a vane extending between the rotor and the cylinder to divide the crescent-shaped space between the rotor and the cylinder into intake and discharge stages, means for driving the rotor of the gas pump from the electric motor rotor, and shaft means extending downwardly from said pump rotor to drive the oil pump, said oil pump being constructed and arranged to deliver oil to the gas pump at a pressure greater than the pressure in the casing.

8. For use in a pump of the class described, a rotor having separately formed but rigidly attached flanking end walls of greater diameter than the rotor itself, and a vane rigily secured to the rotor and the end walls and extending outwardly from the rotor.

9. In a rotary pump of the class described, an outer pump member having an inner cylindrical surface flanked by planar end walls normal to the axis of the cylinder surface, cylinder support heads attached to the outer pump member and having alined hub bearings, an inner pump member comprising a rotor mounted within the outer pump member in eccentric tangential relation to the inner cylindrical surface, means for driving at least one of said pump members for producing a pumping action between the rotor and the outer pump member, end plates rigidly attached to the rotor having sliding engagement with the end walls of the outer pump member and being spaced from the adjacent inner faces of the cylinder support heads, means for dividing the crescent-shaped space between the rotor and the outer pump member into intake and discharge chambers having inlet and discharge ports respectively, and a lubricating and gas sealing system associated with the pump including means for supplying oil under pressure to the space between the rotor end plates and the cylinder support heads whereby said space is constantly filled with oil under pressure.

10. In a rotary pump of the class described, an outer pump member having an inner cylindrical surface flanked by planar end walls normal to the axis of the cylinder surface, cylinder support heads attached to the outer pump member and having alined hub bearings, an inner pump member comprising a rotor mounted within the outer pump member in eccentric tangential relation to the inner cylindrical surface, means for driving at least one of said pump members for producing a pumping action between the rotor and the outer pump member, end plates rigidly attached to the rotor having sliding engagement with the end walls of the outer pump member and being spaced from the adjacent inner faces of the cylinder support heads, a rocker mounted in the outer pump member, a vane fixed to the rotor and having a sealed sliding engagement with said rocker in the outer pump member for dividing the crescent-shaped space between the rotor and the outer pump member into intake and discharge chambers having inlet and discharge ports respectively, and a lubricating and gas sealing system associated with the pump including means for supplying oil under pressure to the space between the rotor end plates and the adjacent cylinder support heads and to that portion of the space behind the vane within said rocker whereby said spaces are constantly filled with oil under pressure, and a relatively large passageway connecting the two spaces.

11. In a rotary pump of the class described, an outer pump member having an inner cylindrical surface flanked by planar end walls normal to the axis of the inner cylindrical surface, cylinder support heads attached to the outer member and having alined hub bearings, a rotor mounted within the outer member in eccentric tangential relation to the cylindrical surface having laterally extending trunnion bearings projecting through the cylinder support heads, end plates rigidly secured to the rotor and having sliding engagement with the planar end walls of the outer pump member to form a crescent-shaped pump chamber between the rotor and the outer member, a vane rigidly secured to the rotor and having rocking and sliding engagement in the outer pump member to divide the pump chamber into intake and discharge sides having inlet and discharge ports respectively, bearing means for cooperating with the trunnion bearings of the rotor and the hub bearings of the cylinder support heads, and means for applying a driving force to the rotor to rotate the rotor and vane to produce a pumping action within the pump chamber.

12. A pump comprising a shaft, a rotor secured to the shaft, a cylinder, laterally extending bearings supporting said shaft, and cylinder eccentrically to each other, with the rotor tangential to the cylinder, a vane and end walls rigidly carried by said rotor in mutually sealed relation, the end walls and the vane having a sliding engagement with the cylinder, inlet and outlet ports on opposite sides of the vane, and means carried by said shaft for driving the shaft, said shaft and rotor having a passageway interconnecting the pump compartment and an isolated compartment having a fluid therein.

13. In a rotary pump or the like, pump members comprising a cylinder, a piston mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the piston and having infinitesimal clearances with the cylinder, a vane rigidly secured to one of the pump members, sliding in a rocker in the other pump member, and having sealing engagement with the end walls for dividing the work chamber between the pump members into intake and discharge sections, said sections having inlet and outlet ports respectively, means for rotating the pump members for producing a pumping action between them, and an oil pump and conduits for delivering oil at a pressure higher than the pressure of discharge from the work chamber to the clearances between the end walls and the cylinder and to the sliding surfaces of the vane and the rocker.

14. In a rotary pump or the like, pump members comprising a cylinder, a piston mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the piston and having infinitesimal clearances with the cylinder, a vane rigidly secured to one of the pump members, sliding in a rocker in the other pump member, and

having sealing engagement with the end walls for dividing the work chamber between the pump members into intake and discharge sections, said sections having inlet and outlet ports respectively, means for rotating the pump members for producing a pumping action between them, and an oil pump and conduits for delivering oil at a pressure higher than the pressure of discharge from the work chamber to the clearances between the end walls and the cylinder and to the sliding surfaces of the vane and the rocker, including a groove extending substantially throughout the circumference of each of said clearance spaces approximately as close to the work chamber as the difference between the diameter of the piston and the diameter of the cylinder.

15. In a rotary pump or the like, pump members comprising a cylinder, a piston mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the piston and having infinitesimal clearances with the cylinder, a vane rigidly secured to one of the pump members, sliding in a rocker in the other pump member, and having sealing engagement with the end walls for dividing the work chamber between the pump members into intake and discharge sections, said sections having inlet and outlet ports respectively, means for rotating the pump members for producing a pumping action between them, and an oil pump and conduits for delivering oil at a pressure higher than the pressure of discharge from the work chamber to the clearances between the end walls and the cylinder and to the sliding surfaces of the vane and the rocker, including a groove extending substantially throughout the circumference of each of said clearance spaces approximately as close to the work chamber as the difference between the diameter of the piston and the diameter of the cylinder, said oil pump being positioned within a reservoir of oil below the oil level therein, and said cylinder being positioned above said oil level.

16. In a rotary pump or the like, pump members comprising a cylinder, a piston mounted in the cylinder in eccentric tangential relation thereto, end walls rigidly secured to the piston and having infinitesimal clearances with the cylinder, a vane rigidly secured to one of the pump members, sliding in a rocker in the other pump member, and having sealing engagement with the end walls for dividing the work chamber between the pump members into intake and discharge sections, said sections having inlet and outlet ports respectively, means for rotating the pump members for producing a pumping action between them, bearings for the cylinder on both sides thereof, and cant-preventing bearing means for the piston, and an oil pump and conduits for delivering oil at a pressure higher than the pressure of discharge from the work chamber to the clearances between the end walls and the cylinder and to the sliding surfaces of the vane and the rocker, including a groove extending substantially throughout the circumference of each of said clearance spaces approximately as close to the work chamber as the difference between the diameter of the piston and the diameter of the cylinder.

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