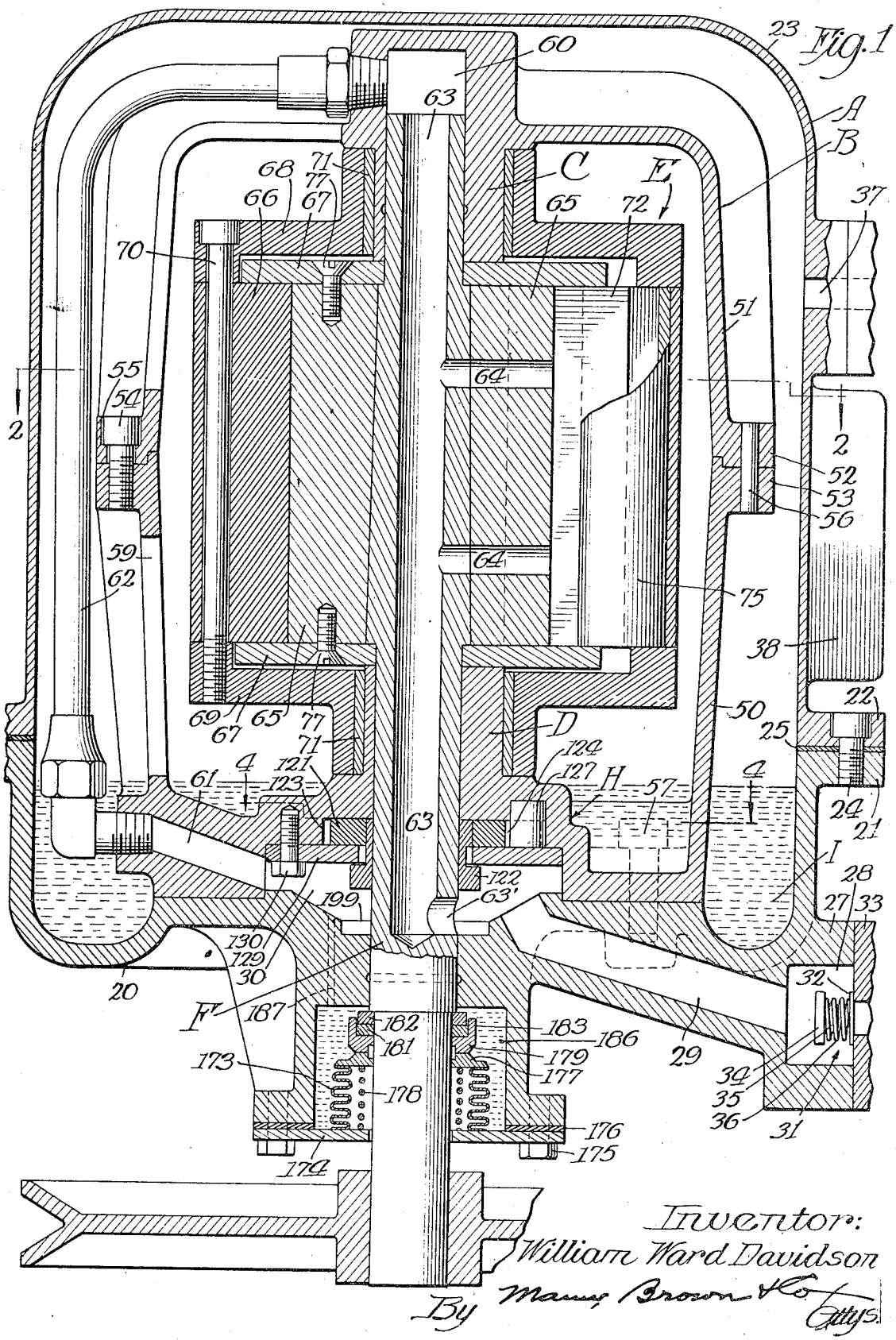
Jan. 26, 1943. W. W DAVIDSON 2,309,577

ROTARY COMPRESSOR

Filed Oct. 1, 1938 3 Sheets-Sheet 1



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ROTARY COMPRESSOR

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Fig. 2

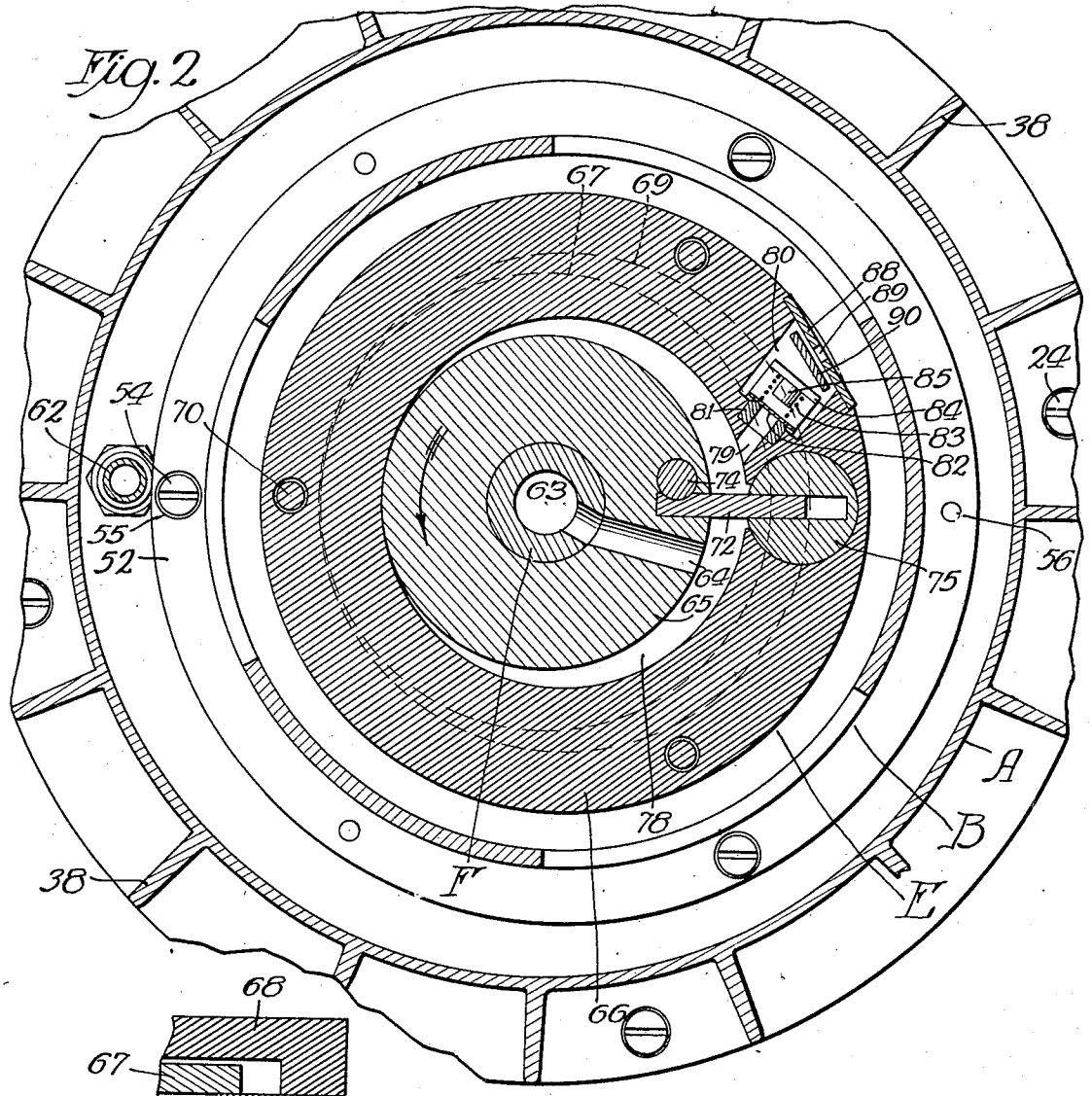
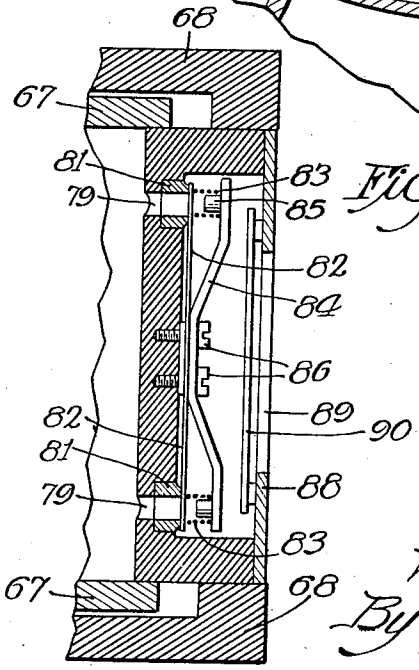


Fig. 3



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Fig. 4

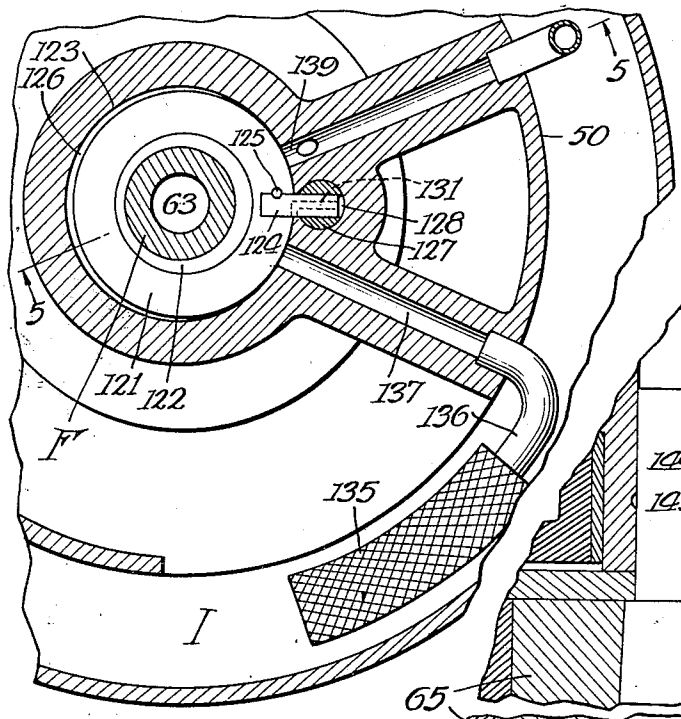


Fig. 5

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UNITED STATES PATENT OFFICE

2,309,577

ROTARY COMPRESSOR

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Application October 1, 1938, Serial No. 232,840

7 Claims. (Cl. 230—140)

Although it is quite desirable for domestic refrigeration units to have a compressor and motor hermetically sealed within a cylindrical dome, experience has taught that larger units, particularly those having upwards of two or three tons capacity, should have the driving motor removed from the dome which houses the compressor. There are several reasons for this. In the first place, it is difficult to dehydrate a large electric motor and unless the dehydration is accomplished thoroughly, the moisture introduced into the system is certain to cause trouble. Then too, the larger sized refrigeration units are used in commercial installations where power sources vary greatly in kind, and it is, therefore, desirable to be able to supply to the user a compressor of given capacity and furnish with it a motor that is suitable for the power source available without carrying numerous complete units in stock. Obviously, also, repair of the motor is greatly facilitated by having it outside of the hermetically sealed dome.

With the motor outside of the dome, the problem of sealing the drive shaft as it emerges from the dome becomes extremely important, and many efforts have been made towards solving this particular problem.

It is an accepted fact that a mechanical seal can be used successfully to seal a shaft where either the speed of the shaft or the pressure on the seal is relatively low, but no seal has yet been devised to my knowledge which will perform satisfactorily with a shaft running at high speed and subjected to relatively high pressure, particularly where that pressure varies between rather wide limits.

It is a well recognized fact that it is highly advantageous to have the dome on the high pressure side of the pump rather than on the low pressure or suction side because of the way it facilitates the conservation of the oil supply for the compressor lubrication system. Since the drive shaft must pass through the dome for its communication with an external motor and since the shaft is preferably driven at high speed to give the pump maximum capacity, the condition which makes for an unsatisfactory use of a mechanical seal is inherently present, viz. a high speed shaft subjected to a varying high pressure gas.

The principal object of this invention is to provide a desirable and effective way for using a mechanical seal with a so-called open type refrigeration unit in which the compressor is sealed within a dome that is on the high pressure side of the compressor system. Other seals, of course, can be used besides mechanical seals with corresponding advantage.

A further object of the invention is to provide a pump structure which has adequate inlet and discharge passages without unduly increasing the size of the compressor base.

Another object of the invention is to provide an improved arrangement of the discharge check valve so that even though this valve be positioned in the outer pump cylinder it will not be unduly affected by centrifugal force.

Further objects and advantages 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 sectional view showing a preferred embodiment of my invention.

Fig. 2 is a horizontal sectional view taken on the line 2—2 of Fig. 1.

Fig. 3 is a detail view showing the dual discharge valve in the pump cylinder.

Fig. 4 is a fragmentary horizontal sectional view taken on the line 4—4 of Fig. 1.

Fig. 5 is a fragmentary sectional view taken on the line 5—5 of Fig. 4 showing particularly the lubrication system and the seal.

It should be understood that the description and illustration of what is now considered to be a preferred form of the 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 having 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, exclusive of the motor, is hermetically sealed. The inner assembly consists of a frame B having upper and lower bearings C and D, respectively, supporting a main pump unit, generally designated E. The unit is driven through a shaft F by a suitable electric motor, not shown, either by means of a direct connection in which the motor shaft is in alinement with the shaft F, or through a pulley, gear or other power transmission means. The principal moving parts of the pump E are properly lubricated and effectively sealed against gas leakage by a sealing and lubricating system which includes an oil pump H operated from the lower portion of

the shaft F and submerged in an oil reservoir I formed by 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 by a suitable stand which may be bolted or otherwise secured to the bosses 26 (Fig. 5) provided on the base 20.

On one side of the base 20, an enlargement 27 is provided having a recess 28 which communicates through a passageway 29 with an intake chamber 30 formed in the dome beneath the oil pump H. The recess 28 houses a suitable intake check valve, generally designated 31, which comprises a flap valve 32 formed of spring steel fixed at one end by screws to the face of the coupling member 33. The screws also secure a spring retainer and stop 34 to the coupling member 33, the outer end being bent outwardly and provided with a stud 35 to receive a spring 36 which bears against the flap valve 32 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 32. It should be understood that a conventional compressor shut-off valve is normally secured to the opposed face of the coupling member 33 to positively disconnect the compressor from the suction line of the compressor system.

On one side of the dome cover 23, a second compressor shut-off valve (not shown) is mounted to positively disconnect the pressure side of the compressor system from the external apparatus. A passageway 37 which leads from the dome to the shut-off valve constitutes a discharge passage for delivering the high pressure gas to the evaporator or other apparatus to which compressed gas is being furnished.

Preferably the dome cover 23 is equipped with a plurality of radially extending vertical fins 38 to assist in dissipating the heat from the dome.

Supporting frame for the pump

The supporting frame B for the main pump E 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 flanges, 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 relation to the base by studs or bolts 57 and may be accurately positioned by dowels if desired. To save weight and provide for additional space for oil, portions of the side wall of the lower casting 50 are cut away, as indicated at 59.

The intake chamber 30 at the base of the dome communicates with an auxiliary intake chamber 60 at the top of the casting 51 by means of a passageway 61 formed in the lower casting 50 and a tube 62. Both of the intake chambers 30 and 60 communicate with a passage 63 which comprises a longitudinal bore in the shaft F. The passage 63 communicates with a plurality of passages 64 extending approximately radially through the wall of the shaft F and through an inner cylinder 65 which is carried by the shaft F and keyed thereto.

The main pump E comprises in addition to the inner cylinder 65 an outer cylinder 66, end plates 67, and upper and lower cylinder heads 68 and 69, which may be clamped to the outer cylinder or ring 66 by elongated machine bolts 70. The cylinders 65 and 66 are preferably made of semi-steel and have accurately machined contacting surfaces, the outer cylinder 66 being supported for rotation in eccentric tangential relationship to the inner cylinder 65 by the external bearing surfaces of the bearings C and D. The heads 68 and 69 are also preferably made of semi-steel and therefore it is desirable to drive hardened steel bushings 71 into the hubs of the heads to assure long life of the bearing surfaces. In order to assure accurate angular positioning of the ring 66 with the heads 68 and 69, a plurality of dowels are preferably provided, such dowels having a tight press fit in the cylinder heads and a light press fit in the cylinder ring for convenience of manufacture and assembly.

A vane or partition 72 is rigidly carried by the rotor 65 at a position adjacent the inlet passages 64 and preferably being secured in position by a dowel 74. This vane has a sliding fit with a slot in a rocker 75 carried by the ring 66 so that in effect the vane 72 has a pivoting and radially sliding fit with the ring 66.

The end plates 67 are preferably formed of hardened steel and are rigidly secured to the rotor 65 as by screws 77. The ring 66 is machined to fit smoothly between the end plates 67. The vane 72 is machined at its ends to fit tightly against the end plates 67 to form a gas-tight joint therewith, or they may be countersunk into the end plates if preferred.

The inner cylinder or rotor 65 has an outer surface which is concentric with the shaft F. Accordingly, the two cylinders have a point of tangency which remains fixed as seen at the left-hand side of the rotor in Figs. 1 and 2 forming a crescent-shaped pumping chamber 78. The cylinders are rotated in the direction of the arrow seen in Fig. 2 and, accordingly, the vane 72 in effect sweeps through the crescent-shaped pumping chamber 78 and draws gas into the chamber behind it while compressing the gas in the portion of the chamber in front of it. The compressed gas escapes through a discharge passage 79 in the ring 66. The discharge passage 79 is enlarged at its outer end

to form a valve chamber 80 in which the check valve is positioned. The check valve preferably includes a seat ring 81, a Swedish spring steel flap valve 82 which is urged to the closed position by a coil spring 83 which is held in position by a bracket 84 having a lug 85 thereon which helps to hold the spring and also limits movement of the valve 82. As seen best in Fig. 3, the flap valve 82 and the bracket 84 may be secured in position by screws 86.

Fig. 3 also shows that there are preferably two discharge passages 79. As previously explained, there are also two inlet passages 64 and two inlet chambers 30 and 60 by which gas may flow from the chamber 30 directly into the passage 63 through the radial bore 63' and may also flow from the chamber 30 through the tube 62 to the chamber 60 and from there to the bore 63. Accordingly, gas entering the main inlet passage 29 may flow into and out of the main pump through two different routes, thus furnishing adequate cross-sectional flow area for the gas. This is a very valuable feature inasmuch as the efficiency of the pump depends in part upon the extent to which the pumping chamber is filled up with the uncompressed gas in each revolution.

Although it is desirable from the standpoint of simplicity to have the discharge passages open into the dome A, it has heretofore been thought best to have the check valve and hence the discharge passages positioned in the inner cylinder so that centrifugal force would tend to close the check valve instead of tending to open it and also because it is desirable to have the discharge port as close to the vane as possible. According to the present invention the check valve is positioned in the outer cylinder but is positioned at an angle with respect to the radius thereof so that the passage 79 will extend toward the vane and the effective centrifugal force on the flap valve 82 will be minimized.

The discharge passage 79 has also been provided with a muffler so that the sound of the escaping gas and of the closing valve will not be transmitted too readily into and through the dome. This muffler comprises a discharge cover plate 88 provided with a slot 89, on the inner side of which is positioned a baffle 90 preferably considerably larger than the slot 89 so that the sound waves are trapped within the valve chamber 80.

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 the 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 eccentric bushing 122 rigidly keyed or otherwise secured on 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. A vane or partition 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 124 with the casting 50 is accomplished by means of a cylindrical rocker 127 which is recessed into the base of the casting and is slotted at 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 131 through the vane 124 so that oil cannot be trapped behind the vane 124 as it reciprocates within the rocker 127. The bottom of the oil pump chamber is closed by a plate 129 which is suitably secured to the casting as by means of the screw 130. Preferably the rotor 121, the vane 124, the rocker 127 and the end plate 129 are all made of hardened steel. As seen best in Fig. 4, oil is taken from the reservoir I through a screen 135, a tube 136 and bore 137 into the crescent-shaped pump chamber 126, being drawn therein by the planetary movement of the rotor 121. This planetary movement also forces oil through a discharge port 139 and 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 will drive it into the spaces where oil is necessary. The main pump is lubricated and sealed by a plurality of passages which need not be described in detail but to which the tube 140 and the passage 141 conduct the oil from the discharge port 139. It should be specifically noted that the system of lubricating passages includes annular passages 143 and connecting vertical passages 144 for lubricating and sealing the shaft F where it passes through the bearings C and D.

To avoid the possibility of creating undesirable back pressures in the lubricating and sealing system, a relief valve 165 and, if desired, a restricted relief passage 166 are provided.

It will be observed that the high pressure chamber within the dome A and the high pressure gases within the pump E are sealed from the low pressure intake chambers 30 and 60 by the oiled bearings D and C respectively. The oil pump H will also help to seal the intake chamber 30 from the high pressure gases. It is not vital that this seal be absolutely perfect because, although minute amounts of oil will be forced through the bearings into the intake chambers, this will be harmless inasmuch as these minute quantities will merely be pumped back into the dome A and the oil reservoir I by the main pump E so that there will be no loss of oil or gas at these points from the dome A. Where the shaft F passes out through the dome A, however, it is highly desirable to completely prevent leakage since leakage at this point would constitute a loss of oil or gas from the dome A with the result that in the course of time the quantity available would be insufficient.

To prevent such leakage it has been common in domestic compressor units to locate the motor within the dome A so that the dome could be hermetically sealed and there would be no possibility of loss through the dome. As previously stated, it is disadvantageous in larger units to have to seal the motor within the dome. With larger units it is therefore desirable to have the shaft pass through the dome and to provide a mechanical rotary seal between the shaft and the dome as seen in Fig. 1. However, such seals have not to my knowledge been satisfactory in systems having high speed and high pressure compressors. According to the present invention, however, the seal is made satisfactory by

virtue of its being freed from the high pressure produced by the compressor.

The rotary mechanical seal may be of any suitable type, one form being illustrated in Fig. 1. According to this form, a bronze bellows 173 is soldered to a cover plate 174 which in turn is sealed to the bottom casting 20 of the dome A by means of screws 175 and a gasket 176. A rotary seal ring 177 is carried by and soldered to the bellows 173 and is urged by a steel spring 178 against a companion seal ring 179. The seal ring 179 is sealed to the shaft F in any suitable manner as by a washer 181 which is preferably made of "Nuprene" or some other suitable form of rubber and which bears against a steel washer 182. The pressure of spring 178, urging the ring 179 toward the washer 182, will squeeze the washer 181 against the shaft F, thus sealing the ring 179 to the shaft. A flange 183 on the ring 179 will prevent the washer 181 from spreading outwardly. It will be understood of course that the relative rotation takes place between the rings 177 and 179 whose engaging surfaces are ground flat and polished. The ring 179 is preferably made of steel and the ring 177 may be made of bronze. There will be no appreciable leakage between these rings providing that they are kept lubricated and providing that the pressure of the ring 177 against the ring 179 is kept within a suitable range.

In the past when oil has been supplied to such seals it has been supplied from the recervoir I or equivalent reservoir in such manner that it was subjected to the full pressure of the high pressure gas within the dome A. It should be understood that the space above the oil level constitutes a high pressure chamber into which the gas is compressed by the pump. Accordingly, the oil in the reservoir I being unprotected from this pressure is likewise subjected to this pressure. This pressure is advantageous in the case of lubricating the parts of the pump inasmuch as it supplements the pressure developed by the oil pump H and aids that pump in forcing the oil into the spaces where it is required but from which the pressure developed by the pump would tend to exclude the oil. In the case of the rotary seal, indicated generally by the letter J, however, this pressure would tend to collapse the bellows 173 and hence let oil escape between the seal rings 177 and 179. If the spring 178 were made sufficiently strong to overcome the highest pressure thus encountered, it would be excessive when the pressure on the oil was not so high and would cause some other trouble such as the excessive wearing of the sealing surfaces of the seal rings 177 and 179.

According to the present invention these difficulties are avoided by submerging the rotary seal J in a bath of oil and supplying oil to this bath in minute quantities while drawing off any excess so as to prevent the building up of an oil pressure in the vicinity of the seal J. To this end the rotary seal J is preferably positioned within the well or seal chamber 186 at the bottom of the casting 20, and a passage 187 provides free communication between the chamber 186 and the intake chamber 30 so that any excess of oil will be drawn into the main pump E and pumped by it back into the dome A and the oil reservoir I. Of course, an excessive supply of oil to the chamber 186 would be objectionable in burdening the main pump E with pumping an appreciable quantity of oil. Accordingly, means is provided so that only minute quantities of oil will be supplied to the chamber 186 such as the quantities which would seep from a bearing to which oil is supplied under pressure. A special passage with a minute opening could be provided, but according to the illustrated form of the invention a third bearing 190 is provided to which oil is supplied through the passage 191 in casting 50 and passage 192 in casting 20, as well as the annular passage 193 in the bearing 190. This bearing is desirable in any event for adequately journaling the shaft F and it serves the additional purpose of providing an edequate supply of seepage oil for the rotary seal J, any excess supply passing up through the passage 187 to be drawn into the main pump E.

The supply of oil to the seal chamber 186 is so slow that, if the pump were started up while the chamber 186 was empty, it would be a long time before the oil reached the level of the seal rings 177 and 179. Accordingly, a passage 196 and a valve 197 may be provided for rapid filling of the chamber 186. When the pump is initially started, the valve 197 may be opened so that oil may pass freely through passage 196. After a few seconds or when the change in sound indicates that the chamber 186 has been filled, the valve 197 may be closed and the slight seepage of oil between the bearing 190 and the shaft F will keep the chamber 186 filled with oil. Because the oil in sealed chamber 186 has free access to the intake chamber 30 and hence to the intake passage 63, the pressure of the oil is limited to the pressure in the intake chamber 30. This pressure will usually ze quite low and will always be much lower than the maximum pressure within the dome A. Being subjected only to this low pressure, the seal J will be satisfactory even though the shaft F is driven at high speed.

It is desirable that the oil which enters the intake chamber 30 through the passage 187 or by seepage from the oil pump H pass into the inlet passage 63 instead of flowing down the intake passage 29. To this end a recess 199 is formed in the bottom of the intake passage 30 with the peripheral walls of the recess extending up above the bottom of the port 63' of passage 63. Of course, to make certain that the oil is drawn upwardly into the passage 63 the peripheral wall of recess 199 may extend entirely above the port 63', but this has not been found necessary heretofore.

I claim:

1. In a pump, a dome having a high pressure chamber and a low pressure chamber vertically spaced and sealed from each other, a main pump in the high pressure chamber having its outer wall rotating in said chamber and surrounded by gas, a vertical drive shaft extending downwardly through the low pressure chamber, an oil reservoir in the high pressure chamber, an oil pump adapted to take oil from the reservoir and distribute it under pressure to various passages for lubricating and sealing the relatively movable parts, a bearing for the shaft adjacent the low pressure chamber, means for sealing the shaft outwardly beyond said bearing, said means including an oil chamber receiving limited quantities of oil from one of said passages to provide an oil level therein, and a mechanical seal for the shaft in the oil chamber including a pair of flat relatively rotating seal rings below the oil level and flooded with oil and pressed toward one another by means opposing the oil pressure thereon; said oil chamber being in open communication with said low pressure chamber whereby ex-

cess oil may flow from the oil chamber to the low pressure chamber and an oil pressure on the seal substantially as low as the pressure in the low pressure chamber is assured.

2. Pumping apparatus including a main pump of the type having two cylindrical pumping members, one within the other in tangential relationship, a shaft for rotating the inner pumping member, an intake chamber at each end of the shaft, a passage in the shaft communicating with each intake chamber and extending therefrom toward the inner cylindrical member, and a pump housing enclosing the intake chamber, said apparatus including means for supplying fluid to be pumped to each of the chambers, and a passage through the inner cylindrical member for supplying fluid from the shaft to the space between the cylindrical members, and outlet means for said space.

3. A rotary pump including two cylindrical members, one within the other in tangential relationship to form a pumping chamber therebetween, a partition between the cylindrical members, a cylindrical rocker in the outer cylindrical member receiving the partition in sliding engagement, and fluid inlet and outlet passages for the pumping chamber including a valve chamber within the outer cylindrical member, and a short substantially straight passage leading directly from the pumping chamber to the valve chamber and inclined with respect to an axial plane of the outer cylindrical member to extend from the valve chamber toward the partition.

4. A rotary pump including two rotatable cylindrical members, one within the other in tangential relationship to form a pumping chamber therebetween, a partition between the cylindrical members, a cylindrical rocker in the outer cylindrical member receiving the partition in sliding engagement, and fluid inlet and outlet passages for the pumping chamber including a valve chamber within the outer cylindrical member proximate said engagement, a passage from the pumping chamber to the valve chamber and inclined with respect to an axial plane of the outer cylindrical member and extending from the valve chamber toward the partition to open upon the inner surface of the outer member at a point immediately adjacent said partition, and a valve member within the valve chamber movable in a direction longitudinal with respect to the passage.

5. A rotary pump including an outer rotating cylinder substantially free of circumferential projections which tend to resist its rotation by friction with a surrounding fluid and having an outlet recess extending inwardly from the outer surface thereof forming a valve chamber communicating with the inside of the cylinder, a check valve in the valve chamber, an outlet wall for the chamber secured within the recess and having an opening therethrough and a baffle larger than the opening slightly spaced from the inner side of the wall for muffling the sounds within the valve chamber.

6. In a pump, a dome having a high pressure chamber and a low pressure chamber sealed from each other, a main pump in the high pressure chamber having relatively movable parts including a downwardly extending drive shaft extending through the low pressure chamber, passages leading to points between the relatively movable parts requiring lubrication and sealing an oil reservoir in the high pressure chamber, an oil pump adapted to take oil from the reservoir and distribute it under pressure to said passages for lubricating and sealing the relatively movable pump parts, a bearing for the shaft adjacent the low pressure chamber, means for sealing the shaft outwardly beyond said bearing, said means including an oil chamber receiving limited quantities of oil from one of said passages, and a mechanical seal for the shaft in the oil chamber; said oil chamber being in open communication with said low pressure chamber whereby excess oil may flow from the oil chamber to the low pressure chamber and an oil pressure on the seal substantially as low as the pressure in the low pressure chamber is assured, said shaft being substantially sealed between the main pump and the low pressure chamber by closely fitting parts supplied with oil from said oil pump at a pressure in excess of the pressure in the high pressure chamber.

7. Pumping apparatus including an outer housing having intake and outlet connections therein and having a vertically disposed shaft passing through a bottom wall thereof, said housing having an oil supply therein with an oil level above the bottom of said housing, partition means within the housing to cooperate with the shaft to divide the housing into high and low pressure chambers communicating with the intake and outlet connections respectively, a fluid pump mounted on and driven by the shaft having its intake communicating with the low pressure chamber and its outlet communicating with the high pressure chamber, an oil pump driven by the shaft and located on the shaft below the oil level and supplying oil to said fluid pump, a rotary seal between the shaft and the bottom wall exposed to a space in open communication with the low pressure chamber, and means for initially charging said seal with oil on the side thereof toward said fluid pump supplied from the high pressure chamber and thereafter supplying the seal with minute quantities of oil from the high pressure chamber; said fluid pump operating to pump any excess of oil from the low pressure chamber to the high pressure chamber whereby the pressure on the seal is substantially the same as the pressure in the low pressure chamber and whereby any excess oil is fed by the main pump back to the high pressure chamber.

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