

Oct. 10, 1961

F. L. TARLETON
REFRIGERATION APPARATUS

3,003,684

Filed May 29, 1957

4 Sheets-Sheet 1

Fig. 1.

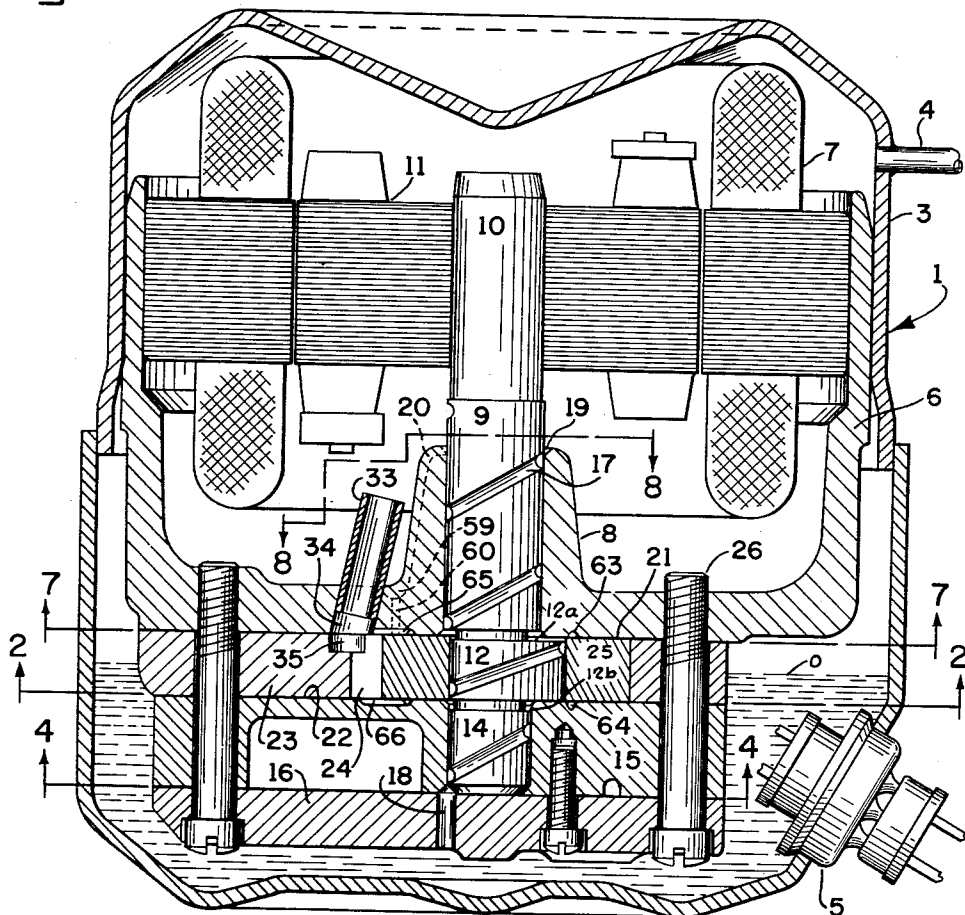
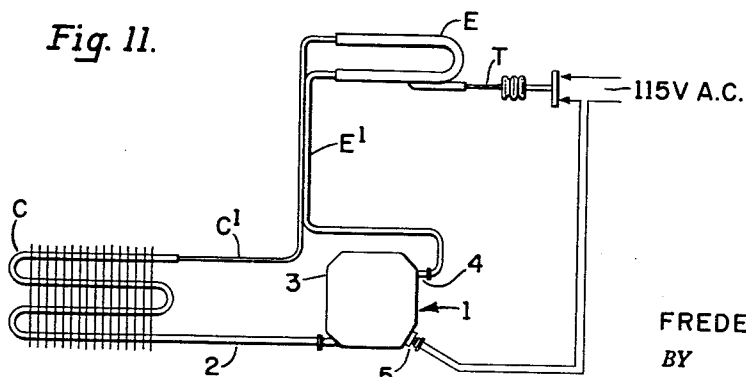


Fig. 11.



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Fig. 2.

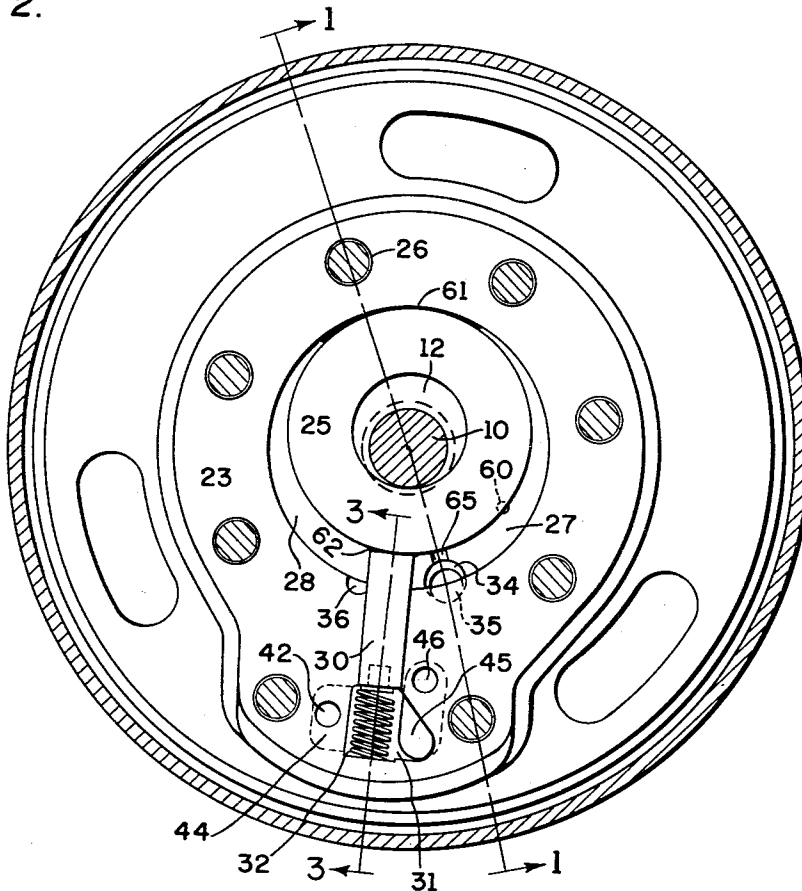
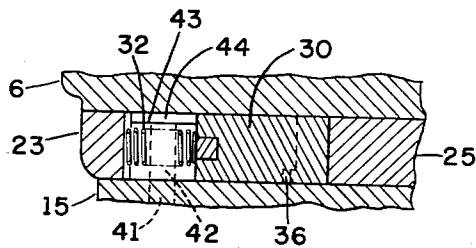


Fig. 3.



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Fig. 4.

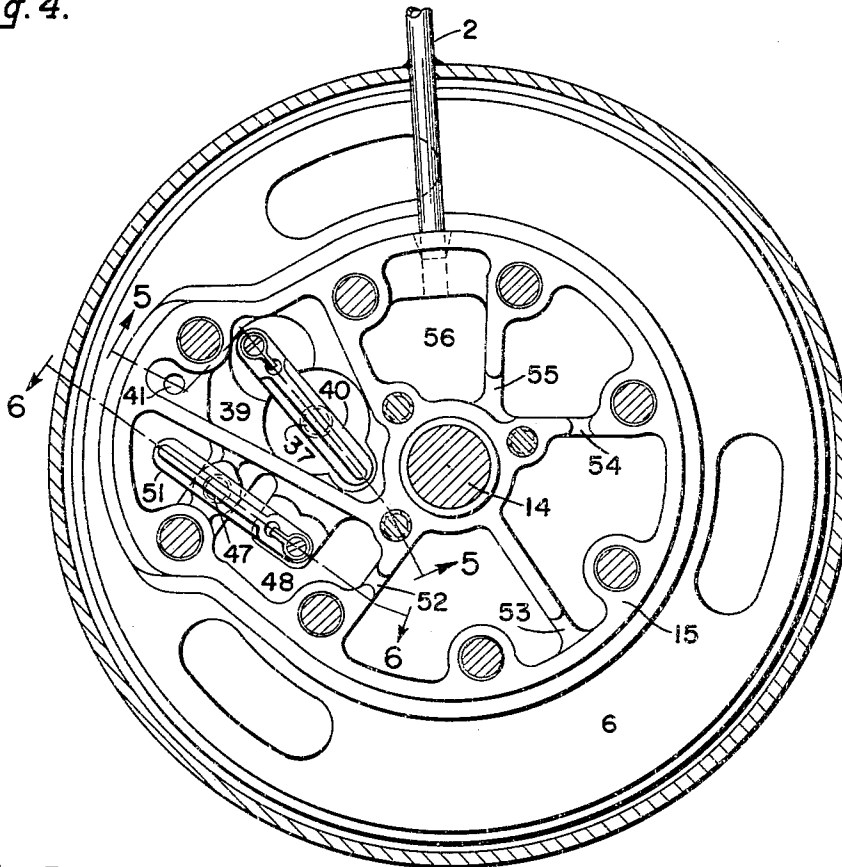


Fig. 5.

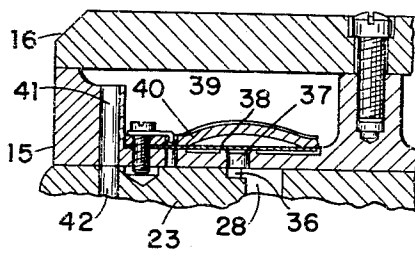
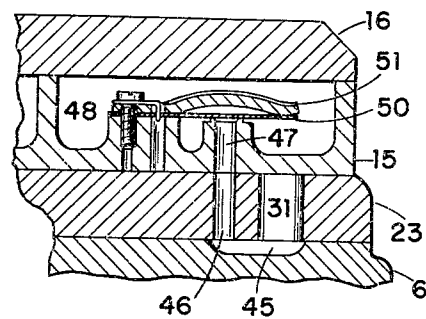


Fig. 6.



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Fig. 7.

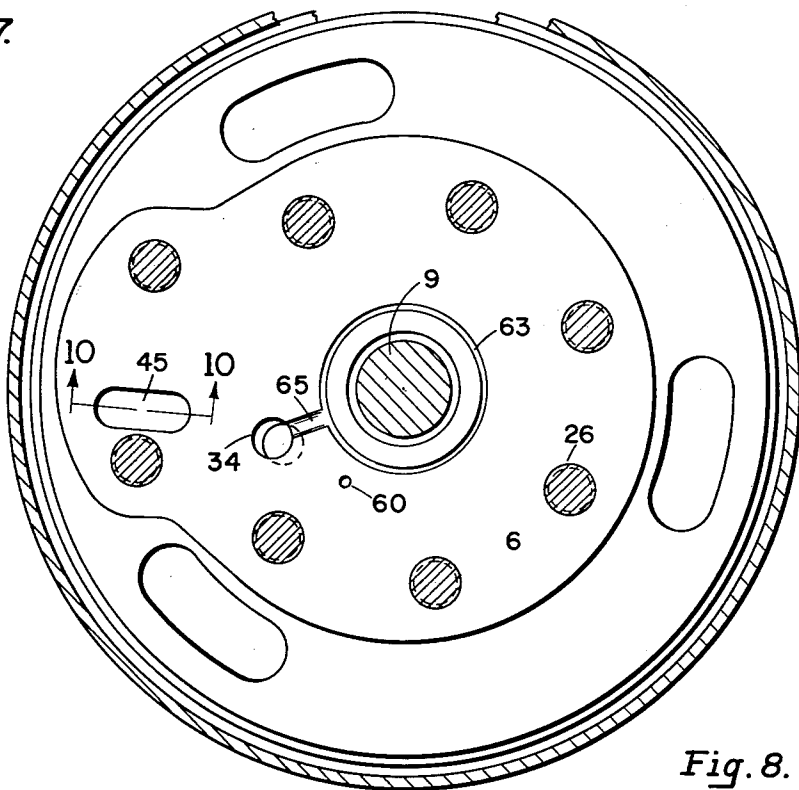


Fig. 8.

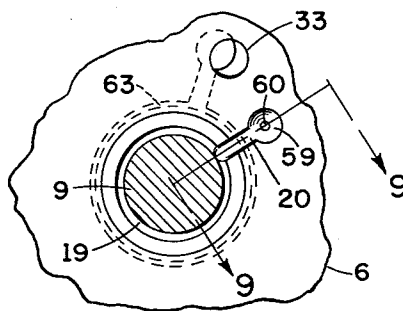


Fig. 9.

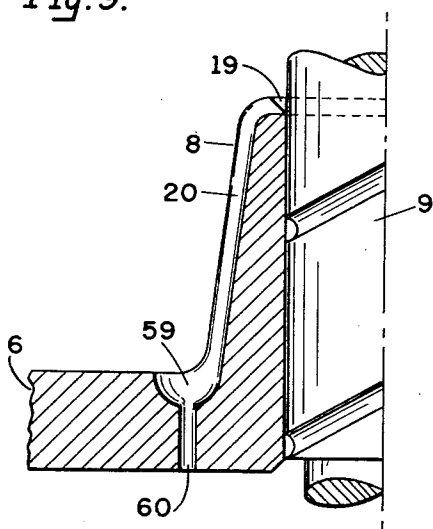
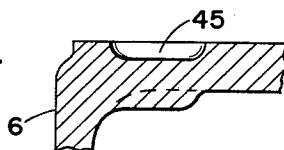


Fig. 10.



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3,003,684

REFRIGERATION APPARATUS

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Filed May 29, 1957, Ser. No. 662,441
7 Claims. (Cl. 230-207)

This invention relates to refrigeration apparatus and, in particular, to a refrigeration compressor of the rotary piston type, in which the compressor is housed in a hermetically sealed casing which forms a part of the low pressure side of the refrigeration system.

In the art, compressors in which the sealed casing is part of the low pressure refrigeration system are often referred to as "low pressure crankcase" compressors. Such compressors have a very substantial advantage over the high pressure crankcase type—in which the casing is a part of the high pressure side of the refrigeration system. Among other things, charging the system with refrigerant is a much less critical operation in the low pressure crankcase system than in the high pressure system.

For example, the dichloro-difluoromethane material sold under the trademark "Freon 12," one of the most commonly used refrigerants in domestic refrigeration, is completely miscible in oil. The crankcase pressure and temperature determine the amount of Freon 12 which can be dissolved in the oil, in accordance with the well-known laws of solubility of gases in liquids. In the low pressure crankcase compressor, the crankcase pressure is substantially the same as the evaporator pressure. Assuming a conventional domestic electric refrigeration system, an evaporator temperature of 10° F. corresponds to a gas pressure within the evaporator of 15 pounds per square inch gauge; and it may be assumed that the crankcase pressure is substantially the same. At 15 pounds per square inch gauge pressure, the solubility of Freon 12 in the oil is about 3% at an oil temperature of 200° F., and 8% at an oil temperature of 100° F. These oil temperatures are typical of those prevailing at the normal high and low ambient temperatures of domestic refrigeration operation.

In a high pressure system, in which the compressor discharges into the casing, the solubility of Freon 12 in the oil may range from 30% to 50% over the same ambient temperature range.

Assuming for purposes of easy example that each refrigeration system (that is to say, the system using a low pressure crankcase compressor and a system using a high pressure crankcase compressor) requires 10 ounces of Freon 12 for proper operation, it will be seen that at the low ambient temperature condition, only .8 ounce of refrigerant will be dissolved in the oil in the low pressure crankcase compressor; whereas in the high pressure crankcase compressor 5 ounces will be dissolved. At the high ambient temperature, only .3 ounce will be dissolved in the oil in the low pressure compressor; whereas 3 ounces of refrigerant will be lost to the refrigeration system in the high pressure crankcase compressor. In the high pressure crankcase compressor, therefore, if the amount of refrigerant for satisfactory operation is 10 ounces net, the amount charged into the system must allow for the 5 ounces dissolved into the oil at low ambient temperatures, and therefore a total of 15 ounces must be charged into the system. However, during a subsequent high ambient temperature operation, two of those five ounces will leave the oil, resulting in an overcharge of refrigerant in the system. Such refrigerant may flood the evaporator and spill over into the suction tube, an undesirable condition. On the other hand, if the refrigeration system is charged with an amount producing the proper net volumes for high ambient temperature operations, the

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system will be undercharged for low ambient temperature operations.

In the low crankcase compressor systems, the variation in the amount of refrigerant is only a matter of one-half of an ounce, and there is neither serious undercharging nor overcharging for proper operation through the normal ambient temperature operating range.

These conditions are well known, and apply to both rotary and reciprocating compressors. Rotary compressors have better capacity characteristics than reciprocating compressors, in that the rotary loses less capacity at low back pressures; and although the reciprocating compressor may have greater capacity than the rotary at high back pressures, the capacity requirement at the least favorable operating condition must determine the compressor capacity. Therefore the capacity of the reciprocating compressor must be increased to place it on a par with the rotary compressor at low back pressures; and such increase gives the compressor an altogether unnecessary capacity at the higher back pressures. The fact remains, however, that although low crankcase pressure reciprocating compressors are well known in the art, the extreme difficulty of lubricating the piston, eccentric shaft, and cylinder divider blade of a rotary compressor, and of maintaining a proper oil seal between the rotating piston and the cylinder and between the divider blade and the cylinder walls has in the past effectively limited the rotary compressor to the high crankcase pressure type.

The problem which previous workers in the low pressure rotary compressor art had not been able to solve in a simple and effective manner is that of compressor lubrication and sealing. The sealing of the rotary piston as it traverses the cylinder is entirely by means of an oil film. Although this film is really of microscopic thickness, the total amount of oil used during continuous compressor operation for one hour, even in the comparatively small size domestic refrigerator compressors, amounts to from twenty to sixty cubic centimeters. This is a substantial quantity and indicates the necessity for having a steady flow of lubricating oil for sealing purposes in addition to that required for normal lubrication.

In view of its structure and method of operation, the sealing and lubrication of a rotary compressor are directly related to the pressure differential between the average cylinder pressure and the crankcase pressure. In a high pressure crankcase rotary compressor, the crankcase pressure is substantially the same as the pressure in the cylinder at the instant of the discharge of gas therefrom. The average cylinder pressure is, of course, lower than the crankcase pressure. It is accepted practice to lift oil from the crankcase sump by means of a spiral groove pump on the eccentric shaft. This oil will be at the crankcase pressure and as it reaches the interface between the rotating piston and the bottom end plate of the cylinder, the pressure differential will cause some of that oil to flow radially outwardly to seal the interface and to lubricate the relatively rotating parts. Oil which continues to travel along the eccentric shaft will enter the second interface between the rotating piston and the upper end plate of the cylinder, whereupon some of this oil will flow radially outward to seal the upper surface of the roller against gas leakage and also to lubricate the rotor. There is a continuous flow of sealing oil and it will reach the side wall of the rotor to produce the necessary seal between the rotor and the cylinder wall. Oil which reaches the periphery of the rotor will coat the sliding surfaces of the reciprocating blade which in rotary compressors divides the cylinder into its low pressure and high pressure portions.

These favorable pressure relationships do not prevail in low pressure crankcase compressors because except for

the cylinder portion at the actual inlet thereof, the average cylinder pressure is greater than the crankcase pressure, and there can be no radially outward flow of oil from the shaft to the periphery of the roller. Obviously, the flow must be in the other direction. This pressure relationship introduces problems not only in the lubrication and sealing of the cylinder walls and rotary piston, but also in the sealing and lubrication of the blade.

It is, therefore, an object of the invention to provide an improved means for lubricating and sealing the piston and other operating components of a low pressure crankcase rotary refrigerant compressor.

It is still another object of the invention to provide a low pressure crankcase type of rotary refrigerant compressor having means whereby sufficient oil may be introduced into the compressor for lubricating and adequately sealing the relatively moving parts of the compressor without the danger of introducing excessive oil into the refrigeration system.

It is a further object of the invention to provide a low pressure crankcase type of refrigerant compressor having means whereby compressed gas which may penetrate the oil seal will be returned to the low pressure side of the cylinder, thus permitting the use of larger manufacturing tolerances without loss of compressor efficiency.

In the attainment of these and other objects, the present invention provides for introducing oil into the cylinder at the low pressure area thereof; specifically, a metered amount of oil is brought into the cylinder at a location such that it will coat the periphery of the rotor to seal the cylinder between the side wall of the rotor and the cylinder wall and will flow radially inward along the top and bottom faces of the rotor to seal the clearance between said faces and the cylinder end plates. Specifically, the inflowing oil is arranged directly to flow upon the upper face of the rotor for radially inward travel between said upper face and the upper cylinder end plate, and to flow along the side wall of the rotor. The reciprocating blade is sealed and lubricated by oil which is carried with the compressed gas into a chamber in back of the blade. A small quantity of this oil accumulates within this chamber and is distributed over the side walls of the blade by the rapid reciprocation thereof, whereupon all surfaces of the blade become coated with a lubricating and sealing oil film.

Although rotary compressors are manufactured to extremely close tolerances, it is good practice to make the tolerances as "easy" as are consistent with the necessary tightness against "blow-back" or other undesirable pressure loss conditions within the cylinder. Relaxed tolerances, even though the amount of relaxation is very small, increase the spaces between the respective piston and cylinder parts and require a necessarily larger quantity of oil for sealing. Merely to increase the introduction of oil into the cylinder would have the adverse effect of circulating too much oil within the refrigeration system.

In a presently preferred embodiment of the invention, I provide each of the cylinder end plates with a groove concentric with the cylinder and provide a passage between the groove and the low pressure side of the cylinder. The grooves make it possible to increase the oil needed for sealing and also provide a path whereby high pressure gas which may leak between the end plate and the rotor will be returned to the low pressure side of the cylinder. An excess of oil which may reach the grooves will not be discharged into the refrigerant system with the high pressure gas. The grooves, therefore, permit the increase in the amount of oil in the cylinder to that necessary to compensate for increased clearances and thus improve the performance of the compressor.

It must be understood, of course, that the expression "excessive oil" does not infer that oil is introduced into the cylinder in disregard of the basic consideration that

excessive oil is disadvantageous in a refrigeration system. According to the invention, I introduce oil into the cylinder in a manner in which the piston or rotor acts as a metering device. Therefore the oil inlet passage may be made large enough to insure a free flow of oil into the cylinder notwithstanding the extremely low pressure differential between the low pressure portion of the cylinder and the crank case.

As will later appear from an inspection of the drawings and description of operation related thereto, the relationship of the oil passage to the rotary piston is such that oil is deposited on the outer upper portion of the piston during the greater part of its rotation, and into the cylinder intake portion by flow along the side wall of the piston during the instant of lowest cylinder intake pressure. An oil film is thus created which spreads through the clearance between the piston and the cylinder end plates as the cylinder pressure quickly exceeds the crankcase pressure. This film provides lubrication and sealing. An oil film is also established between the adjacent side walls of the cylinder and piston. Although the clearance between the piston and the cylinder side walls is minute, the oil film provides an effective seal.

The cylinder divider blade is in continuous engagement with the rotating piston and such engagement is maintained by a spring, plus the high gas pressure in the chamber which is immediately behind the blade, as previously explained. A portion of the oil carried about the cylinder on the piston wall is removed from the piston by the blade, and will be conveyed by the high pressure gas into the chamber behind the blade, where it separates from the gas and becomes available for the lubrication and sealing of the blade.

Other features and advantages of the invention will best be understood by the following detailed description of a presently preferred embodiment, read in connection with the accompanying drawings in which

FIG. 1 is a side sectional elevation of a refrigerator compressor of the rotary piston, low pressure crankcase, type embodying the present invention;

FIG. 2 is a transverse sectional view of the compressor looking in the direction of the arrows 2—2 of FIG. 1 and showing the underside of the cylinder and the rotor;

FIG. 3 is a vertical sectional view of the blade and blade chamber taken on lines 3—3 of FIG. 2;

FIG. 4 is a transverse sectional view looking in the direction of the arrows 4—4 of FIG. 1 and showing the valve arrangement and the muffler chambers communicating with the high pressure discharge tube;

FIG. 5 is a transverse section taken on lines 5—5 of FIG. 4, showing the discharge valve communicating between the cylinder and a chamber in the valve plate;

FIG. 6 is a transverse section taken on lines 6—6 of FIG. 4, showing the valve and passages communicating between the blade chamber and the first muffler chamber of the valve plate;

FIG. 7 is a bottom plan view taken in section on lines 7—7 of FIG. 1, showing the underside of the frame member;

FIG. 8 is a fragmentary top plan section of the frame member taken on lines 8—8 of FIG. 1;

FIG. 9 is an enlarged fragmentary vertical elevation of the frame member, taken on lines 9—9 of FIG. 8;

FIG. 10 is a fragmentary elevation of the frame member in section on lines 10—10 of FIG. 7; and

FIG. 11 is a schematic refrigerator circuit diagram.

General description

The hermetically sealed, low pressure crankcase rotary compressor 1 is shown in FIG. 11 in its operating environment in a conventional compression-condensation-expansion type refrigeration system. Said system is shown schematically as including a condenser C connected directly to the discharge conduit 2 of the compressor and connected by means including the so-called

"capillary" tubing C' to an evaporator E. An expansion occurs when the liquid refrigerant leaves the small bore tubing C' and enters the substantially larger tubing of the evaporator as is well known in the art. From the evaporator suction tube E' communicates directly with the interior of the hermetic casing 3 of the compressor entering said casing by means of a suitable inlet connection 4. The compressor is under the control of a thermostat T having suitable switch means in series with the motor of the compressor and a suitable power source. A hermetic electrical connection 5 is provided at the casing 3. The high pressure side of the refrigeration system comprises the discharge portion of the compressor, the condenser C and the capillary tubing C'. The low pressure side is represented by the evaporator E, the suction tube E', the inlet connection 4 and the interior of the casing 3. In this important aspect, the refrigerator system embodying the present invention differentiates over conventional hermetically sealed rotary compressor refrigeration systems in which the compressor casing is an element of the high pressure side of the system rather than of the low pressure side. It will be understood, of course, that the refrigeration system has been schematically shown and does not include certain aspects of the evaporator, such as headers, accumulators, and the like, which are well known in the art.

The compressor 1, FIG. 1, includes a rigid main frame 6 which is secured to the upper part of the hermetic casing 3 and serves to mount the stator 7 of the electric motor. The main frame is formed with a bearing 8 which rotatably receives the eccentric shaft 9 of the compressor. An extension 10 carries the rotor 11 of the motor.

The shaft 9 is formed with an eccentric 12 and an extension 14 which is journaled in the valve plate 15 and rests upon the thrust plate 16. The lower portion of the casing 3 may be designated the "crankcase" and it contains a substantial quantity of suitable lubricating oil O, as indicated in FIG. 1. The eccentric shaft 9 is formed with a continuous spiral oil groove 17. The thrust plate 16 has a passage 18 communicating directly with the oil sump and as the shaft rotates, the groove 17 pumps oil from the sump to a receiver or reservoir of very small capacity, as represented by the pocket 19 at the top of bearing 8, the channel 20 communicating therewith, and the pocket 59 at the base thereof. It will be obvious, therefore, that the entire interior of the crankcase 3 is at the low suction pressure of the refrigeration system and that the lubricating oil being transported by the pumping groove 17 is likewise at the low pressure. Excess of oil reaching the pocket 59 merely spills over into the crankcase.

The lower surface 21 of the frame 6 and the upper surface 22 of the valve plate 15 serve as end plates for the cylinder structure 23 confined therebetween. Said cylinder is formed with a very accurately machined cylindrical chamber 24 which is concentric with the motor shaft 10 and forms the actual cylinder space within which operates the rotary piston or rotor 25. As best appears in FIG. 2, the piston 25 is substantially smaller in diameter than the chamber 24 and it is in rotary engagement with the eccentric 12, whereupon as the shaft 10 rotates in a counterclockwise direction, as viewed in FIG. 2, the piston 25 sweeps or oscillates about the inner wall of the cylinder 23 with very minute clearance between the otherwise tangential contact of the piston with the cylinder wall. As shown in FIG. 1, the respective components are secured together in face-to-face relationship by any suitable number of fastening means, such as the machine screws 26.

The cylinder is divided into a low pressure side 27 and a high pressure side 28 (FIG. 2) by means of a reciprocating blade 30 which operates in a slot provided for it in the cylinder 23. Immediately behind the blade, the cylinder is formed with a small volume chamber 31 which houses a spring 32. It will be understood that the end

portion of the blade will enter the chamber 31 during each revolution of the piston 25. The blade 30 is maintained in contact with the wall of the piston 25 by means of the spring 32 and a high pressure gas condition within the chamber 31, as presently explained.

The low pressure refrigerant vapor which enters the casing by way of the connection 4 is introduced into the low pressure portion of the cylinder by means comprising an inlet tube 33 which is fitted within an angularly directed gas intake passage 34 in the frame 6. The passage 34 directly communicates with a substantially semi-circular slot 35 provided in the wall of the cylinder, as best shown in FIGS. 1 and 2. As the piston 25 rotates in a counterclockwise direction, as viewed in FIG. 2, it compresses the gas ahead of it; and the compressed gas passes from the high pressure portion 28 of the cylinder, through a discharge port 36, and then (see FIGS. 4 and 5) through an opening 37 in the valve plate 15, and past the normally closed reed valve 38 into the chamber 39 formed by said valve plate and the thrust plate 16. The valve backstop 40 stiffens the reed valve 38 and prevents its excessive distortion. In view of the fact that the chamber 39 is not in communication with any of the other chambers in the valve plate, the high pressure gas passes directly through a port 41 which registers with a passage 42 extending through a step 43 formed in the chamber 31, as best shown in FIG. 3. The step 43 terminates below the frame member 6, and thus provides an alcove 44 through which the gas enters the chamber 31. The high pressure gas, therefore, undergoes an abrupt change in its direction of flow as it enters the chamber 31. As presently explained, the gas will contain a quantity of oil, some of which will be separated from the gas by reason of the abrupt change in direction and will accumulate within the chamber 31. The underside of the frame 6 is formed with an elongated passage 45 which communicates between the chamber 31 and a passage 46 in the cylinder 23. Passage 45 is, of course, above the chamber 31, and oil which was carried by the gas into the chamber 31, therefore, remains within the chamber. Passage 46 registers with a passage 47 leading into a chamber 48 in the valve plate. As best appears in FIG. 6, the passage 47 terminates in a valve seat with which cooperates the reed valve 50. It will be noted that said reed valve comprises part of a structure including the backstop 51 which, similar to the backstop 40, stiffens the valve reed and guards it against excessive distortion. The valve plate is subdivided by means of walls, as illustrated in FIG. 4, into a plurality of muffler chambers which are interconnected by means of variously arranged passages 52, 53, 54, and 55, which are traversed by the high pressure gas until it enters the final chamber 56 from which the high pressure gas leaves the compressor through the outlet connection 2.

Lubrication and sealing

It has previously been noted that rotary compressors are made to exceedingly fine tolerances and that the minute clearances between the rotary piston and the cylinder wall and between the piston and the upper and lower end plates thereof are lubricated and sealed by means of a film of lubricating oil.

Pursuant to the present invention, a metered amount of lubricating oil is introduced from the crankcase into the low pressure portion of the cylinder. Specifically, the lubricating oil inlet port is so located that for at least a large portion of each rotation of the piston, the port will be covered, wholly or in part, by the rotating piston. As shown in FIG. 9, the oil groove 20 terminates in the preferably hemispherical pocket 59 having in its bottom the oil passage 60. Excess of oil received by the pocket 59 will merely overflow the rim thereof and rejoin the oil in the crankcase. The passage 60 enters the cylinder at a location in which the surface of the piston 25 will sweep across the bottom of the passage. In this arrangement, the rotating piston exposes a pre-

determined area of the end of the passage 60 during a portion of each rotation of the piston; and during each rotation the periphery of the piston will briefly pass beneath the oil passage, whereupon oil flowing through the passage 60 will coat the side wall of the piston. Because the piston is made and located within the cylinder with extreme accuracy, it is used as a precise valving means to control the inlet of oil. This is very advantageous, for the oil passage itself may be made slightly oversize to insure steady flow of oil under the small pressure differential existing in the system, and yet the actual quantity of oil entering the cylinder is accurately metered. The top of the piston continuously wipes a very thin film of oil from the bottom of the oil passage during the rotation of the piston. This wiping action also takes place in varying degree as the eccentrically located piston covers and uncovers the oil inlet port.

The effect of this oil inflow metering operation is to provide an oil film between the upper surface of the piston and the bottom of the frame, and an oil film about the wall of the piston. This latter oil film provides the necessary sealing of the piston with respect to the wall of the cylinder. As indicated by the portion 61 in FIG. 2, a slight excess of oil accumulates ahead of the piston. Some of this excess oil, plus a small quantity of that which is introduced directly into the cylinder from the port 60, will flow between the bottom of the piston and the upper wall of the valve plate 22. This oil movement must occur, because the cylinder pressure exceeds the crankcase pressure except in the immediate gas inlet area of the cylinder; in other words, the average cylinder pressure is greater than the crankcase pressure, and the pressure differential will cause a flow of oil toward the crank shaft. A film of oil will also remain on the side wall of the piston. This film forms the seal between the end of the blade 30 and the piston wall. Oil which may be stripped from the piston wall by the action of the blade will be carried with the high pressure gas through the outlet port 36. Much of the oil which is entrained in the high pressure gas will separate therefrom as the gas enters the chamber 31, as previously stated.

The oil which separates out of the gas in the chamber 31 by reason of the abrupt directional change of the gas flow will, of course, tend to accumulate in the bottom of the chamber because of the elevation of the escape passage 45. Because of the rapid reciprocation of the rear portion of blade 30 within the chamber, and the high gas pressure condition therein, the blade will be thoroughly lubricated and sealed relative to the cylinder walls. The upper portion of the blade will receive oil by splash lubrication and the accumulation of oil on the blade as it separates from the oil. Therefore, although one face of the blade within the cylinder is exposed to high pressure, and the other face to low pressure, the combination of the high pressure condition and the oil separation within the chamber 31 insures proper and thorough lubrication and sealing.

It has previously been stated that it is desirable to make the manufacturing tolerances as "easy" as possible without detrimentally affecting the seal between the relatively moving parts of the piston and cylinder portions. In order to do this without introducing an undesirable excess of oil into the system, I provide the bottom surface of the frame 6 with a circular groove 63, and the upper surface of the valve plate 15 with a similar circular groove 64. As shown in FIG. 1, the respective grooves are always covered by the adjacent faces of the piston. Each groove, however, has a branch passage which affords communication from the groove to the low pressure portion of the cylinder. Groove 63 has a passage 65 communicating between groove 63 and gas intake passage 34, as best shown in FIGS. 1 and 7, whereas groove 64 has a similar passage 66 as indicated in FIG. 1. Be-

cause the respective grooves are thus always in communication with the low pressure side of the cylinder, any excess of oil which may be needed to provide proper sealing for the actual tolerance will always be returned to the cylinder rather than to the refrigeration system. This would not be true if the excess of oil were permitted to flow to the outlet port 36 of the compressed gas. By locating the respective grooves at a suitable radius relative to the axis of rotation, an effective area of oil seal can be assured. For example, in a small size domestic refrigeration compressor in which the inside diameter of the cylinder is 2.15 inches and the rotary piston diameter is 1.90 inches (thus producing a pumping stroke of .25 inch) the operational advantages of the grooves 63 and 64 have been fully realized by forming the grooves of V-shaped cross section, .045" wide, .030" deep, and at a .630" radius from the cylinder axis. The vent passages 65 and 66 may be .125" wide and .030" deep. It will be apparent that the rotary piston sweeps eccentrically over the grooves, thus aiding the maintenance of a film of oil in the radially innermost portions of the piston and cylinder end plates.

Although the illustrated embodiment produces superior results because of the oscillation of the piston over the grooves and the resulting distribution of oil throughout the minute space between the piston and the cylinder end walls, the grooves 63 and 64 may be formed in the faces of the rotary piston, arranged concentrically of the piston on radii which will place the grooves centrally between the inner and outer peripheries of the piston. In this latter arrangement, the bleed-off channels 65 and 66 will remain in the cylinder end walls, as previously described.

The grooves 63 and 64, and the therewith cooperating channels 65 and 66, prevent high pressure gas and oil from reaching the pumping groove 17 on the eccentric shaft, as would be possible if the high pressure gas could flow across the top and bottom of the piston to enter the chambers or pockets 12a and 12b, respectively, above and below the eccentric. It must be remembered that the groove 17 is at crankcase pressure, which is considerably lower than the average pressure within the cylinder. If high pressure gas were permitted to flow into the groove 17, the pumping action of said groove would be interrupted and the compressor starved for oil, with obviously objectionable results. In contrast, the interruption of the flow of oil and gas by the grooves 63 and 64, and the diversion thereof to the low pressure side of the cylinder, establishes a pressure relationship about the eccentric which is beneficial to the pumping of oil and the lubrication and sealing of the piston and cylinder in that area. It will be noted that the upper passage 65 is spaced from the oil inflow port 60 by a substantial distance; the pressure existing in the passage 65 is vented through the intake passage 34 and therefore cannot reach the oil inflow port to interfere with the flow of oil into the cylinder.

In contrast to the high-pressure crankcase compressors, in which the hot compressed gas discharges into the compressor housing, the low gas pressure within the housing of the present invention is relatively cool. This condition simplifies the cooling of the motor-compressor mechanism.

It will be seen, therefore, that the invention provides a low pressure crankcase rotary piston compressor in which the essential sealing and lubrication of the mechanism is assured.

While there has been described what is at present considered to be the preferred embodiment of the invention, it will be understood that various modifications may be made therein, and it is intended to cover in the appended claims all such modifications as fall within the scope of the invention.

I claim:

1. A motor-compressor mechanism comprising a cas-

ing having an inlet for gas to be compressed, a crankcase openly communicating with said casing and containing a quantity of lubricating oil, whereby said crankcase and oil are at substantially the pressure of the incoming gas, a motor and a compressor within said casing and including a cylinder and an annular rotor therein to provide a rotary piston, said cylinder having end plates overlying the top and bottom surfaces of the piston and a blade slidably mounted within said cylinder and engaging the peripheral wall of said piston to divide the cylinder into high and low pressure sides, a shaft driven by said motor, an eccentric on said shaft in continuous engagement with the inner wall of said piston for oscillating said piston about said cylinder in wall to wall tangential relationship, said eccentric shaft being continuously exposed to the pressure condition within said crankcase, means including a pump groove on the exterior wall of said eccentric shaft and said eccentric and openly communicating between said crankcase and the wall of said piston engaging said eccentric to lubricate said wall, means for introducing a predetermined quantity of oil into said cylinder at the low pressure side thereof during a short interval of each rotation of said piston, whereby the higher average cylinder pressure relative to the crankcase pressure will effect a flow of oil radially inwardly between the piston and the adjacent surfaces of the cylinder to lubricate and seal the same, means including a circular passage concentric with said eccentric shaft disposed between the piston and each of the end plates of the cylinder to intercept oil and high pressure gas during the passage thereof toward said eccentric shaft, passage means in said end plates communicating between the said circular passages and the low pressure side of the cylinder to conduct thereto the intercepted high pressure gas and oil, and a discharge conduit connected to said compressor and extending through said casing for the discharge of the compressed gas.

2. A motor-compressor mechanism comprising a casing having an inlet for gas to be compressed, a crankcase openly communicating with said casing and containing a quantity of lubricating oil, whereby said crankcase and oil are at substantially the pressure of the incoming gas, a motor and a compressor within said casing, said compressor including a cylinder and an annular piston rotatably confined therein, said cylinder having end plates overlying the top and bottom surfaces of said piston and a blade slidably mounted within said cylinder and engaging the peripheral wall of said piston to divide the cylinder into high pressure and low pressure sides, gas inlet means communicating between the casing and the low pressure side of said cylinder, a shaft extending through said cylinder to a position below the level of oil in said crankcase, means for rotating said shaft by said motor, an eccentric on said shaft rotatably within said piston in engagement with the wall thereof for oscillating said piston about said cylinder in wall to wall tangential relationship, said eccentric and eccentric shaft having an external groove serving to conduct oil from said crankcase to an open receiver within said casing above said cylinder, said groove openly traversing the inner wall of said rotary piston, an oil conduit communicating between said receiver and the low pressure side of said cylinder, said conduit opening into said cylinder at a location at which said rotary piston covers at least a substantial portion of said conduit opening during each rotation of said piston, whereby said piston comprises means for metering the flow of oil into said cylinder and the higher average cylinder pressure relative to the crankcase pressure will effect a movement of oil radially inwardly between the piston and the adjacent surfaces of the cylinder to seal and lubricate the same, and means for intercepting a portion of the oil and gas in its said radial movement and returning the same to the low pressure side of said cylinder, said means comprising a con-

tinuous circular groove formed in each of said end plates and openly facing said piston, said grooves being concentric with said cylinder and disposed closer to the eccentric shaft than to the wall of said cylinder and a passage in each of said end plates communicating between said groove and said low pressure side.

3. A motor-compressor mechanism comprising a casing having an inlet for gas to be compressed, a crankcase openly communicating with said casing and containing a quantity of oil, whereby said crankcase and the oil therein are at substantially the low pressure of the gas entering said casing, a compressor within said casing and including a cylinder and a rotary piston therein, a blade slidably mounted within said cylinder and bearing against said piston for dividing said cylinder into low pressure and high pressure portions, means including an eccentric and an eccentric shaft for rotating said piston to cause a wall thereof to sweep about said cylinder in tangential relation therewith, the eccentricity of said piston effecting a reciprocation of said blade, a chamber in said cylinder behind said blade and entered by the rear portion of said blade during the reciprocation thereof, a first duct from said casing to the low pressure portion of said cylinder, a second duct from the high pressure portion of said cylinder to said chamber, said second duct effecting an abrupt change in the direction of flow of the gas as it enters said chamber, a third duct opening into said chamber at a high point thereof and communicating with a high pressure discharge conduit extending to the exterior of said casing, motor means within said casing for rotating said eccentric shaft, means on said eccentric shaft for conveying the oil from said crankcase to a point of discharge within said casing above said compressor, a small capacity receiver disposed above said cylinder, said receiver being subject to the low pressure in said casing, means for conveying the oil from said point of discharge to said receiver, and a passage communicating between said receiver and the low pressure portion of said cylinder, said passage entering above said piston at a location where it will be alternately covered and exposed by said piston as said piston rotates eccentrically within said cylinder, whereby the oil is made available within said cylinder for the lubricating and the sealing of said piston within said cylinder by the radially inward flow of the oil induced by the higher average pressure within said cylinder as respects the low pressure in said crankcase, and whereby a portion of the oil is also moved about said cylinder in advance of said piston by the tangential relationship of said piston to said cylinder, so that the oil in said cylinder is expelled therefrom with the compressed gas through said first duct and enters said chamber behind said blade for the sealing and lubricating of said blade.

4. A motor-compressor mechanism comprising a casing having an inlet for gas to be compressed, a crankcase openly communicating with said casing and containing a quantity of oil, whereby said crankcase and the oil therein are at substantially the low pressure of the gas entering said casing, a compressor within said casing and including a cylinder and a rotary piston therein, a blade slidably mounted within said cylinder and bearing against said piston for dividing said cylinder into low pressure and high pressure portions, means including an eccentric and an eccentric shaft for rotating said piston to cause a wall thereof to sweep eccentrically about said cylinder in tangential relation therewith, the eccentricity of said piston effecting a reciprocation of said blade, a chamber in said cylinder behind said blade and entered by the rear portion of said blade during the reciprocation thereof, a first duct from said casing to the low pressure portion of said cylinder, a second duct communicating between the high pressure portion of said cylinder and said chamber, a third duct from said chamber communicating with a high pressure discharge conduit extending to the exterior of said casing, means for rotating said eccentric shaft at

high speed, means on said eccentric shaft for conveying the oil from said crankcase to a small capacity receiver disposed above said cylinder, said receiver being subject to the low pressure in said casing, a passage communicating between said receiver and the low pressure portion of said cylinder, said passage entering above said piston at a location where it will be exposed by said piston only as said piston attains a rotational position within said cylinder affecting minimal pressure with said low pressure portion thereof, whereby the oil is made available within said cylinder for the lubricating and sealing of said piston within said cylinder by the radially inward flow of the oil induced by the higher average pressure within said cylinder as respects the low pressure in said crankcase, and whereby a portion of oil is also moved about said cylinder in advance of said piston by the tangential relationship of said piston and said cylinder and flows with compressed gas into said chamber behind said blade, and means for separating the oil from the gas within said chamber and for accumulating the oil for the sealing and lubricating of said blade.

5. A motor-compressor mechanism comprising a sealed casing having an inlet for gas to be compressed, a crankcase openly communicating with said casing, a quantity of lubricating oil in said crankcase, structure within said casing providing a cylinder having upper and lower end plates in covering relationship thereto, an annular rotor of less diameter than said cylinder disposed therein in slidable engagement with said end plates, a divider blade slidably mounted in a wall of said cylinder in slidable engagement with the peripheral wall of said rotor and with the respective end plates, said blade dividing said cylinder into an intake portion and a compression portion, gas intake means communicating between said casing and said intake portion, discharge means for conducting compressed gas from said compression portion to the exterior of said casing, a drive shaft concentric with said cylinder journaled in said end plates, an eccentric on said shaft disposed rotatably within said rotor to effect a planetating movement of said rotor about said cylinder in tangential relation thereto to cause compression of said gas, a motor for rotating said shaft, a reservoir for lubricating oil within said casing above said upper plate, means including a groove openly communicating with said crankcase and extending spirally around the exterior of said shaft and said eccentric for conducting oil from said crankcase to said reservoir upon rotation of said shaft, the groove on said eccentric being in direct communication with said rotor and with the groove on said shaft, passage means communicating between said reservoir and said cylinder intake portion for introducing oil from said reservoir onto an end wall of said rotor during the greater portion of the planetation thereof, said passage means being uncovered by said rotor only during an instant of lowest pressure within said intake portion to introduce oil thereinto, whereby upon compression of gas within said cylinder the differential between the cylinder pressure and the crankcase pressure within said oil conducting groove will effect a radially inward movement of oil along surfaces of said rotor to lubricate and seal the same relative to said end plates, means including a passage disposed about said eccentric and between at least one end of the rotor and the facing end plate to intercept said oil and any high pressure gas entrained therewith, and passage means to return said intercepted gas and oil to the intake side of said cylinder.

6. A motor-compressor mechanism comprising a sealed casing having an inlet for gas to be compressed, a crankcase openly communicating with said casing, a quantity of lubricating oil in said crankcase, structure within said casing providing a cylinder having upper and lower end plates in covering relationship thereto, an annular rotor of less diameter than said cylinder disposed therein in slidable engagement with said end plates, a divider blade

slidably mounted in a wall of said cylinder in slidable engagement with the peripheral wall of said rotor and with the respective end plates, said blade dividing said cylinder into an intake portion and a compression portion, gas intake means communicating between said casing and said intake portion, discharge means for conducting compressed gas from said compression portion to the exterior of said casing, a drive shaft concentric with said cylinder journaled in said end plates, an eccentric on said shaft disposed rotatably within said rotor to effect a planetating movement of said rotor about said cylinder in tangential relation thereto to cause compression of said gas, a motor for rotating said shaft, a reservoir for lubricating oil within said casing above said upper plate, means on said shaft and said eccentric in open communication with said crankcase, said reservoir and said rotor for pumping oil from said crankcase to said reservoir upon rotation of said shaft, passage means communicating between said reservoir and the intake portion of said cylinder for gravity flow of oil onto an end wall of said rotor during planetation thereof and to admit oil to said cylinder during that portion of said planetation effecting minimal pressure condition within said intake portion, whereby upon compression of gas within said cylinder the differential between the cylinder pressure and the crankcase pressure will effect a radially inward movement of oil along surfaces of said rotor to lubricate and seal the same relative to said end plates, and means for interrupting said oil movement after it has traversed substantially the entire radial dimension of said rotor and for returning oil and any high pressure gas entrained therewith to the intake side of said cylinder.

7. Refrigerating apparatus comprising a casing having an inlet for gaseous refrigerant to be compressed, said casing having a crankcase openly communicating therewith for containing a quantity of lubricating oil, whereby the gaseous refrigerant and the lubricating oil are at substantially the same low casing pressure, a refrigerant compressor arranged in said casing and having a cylinder, upper and lower wall members in covering relation to said cylinder for sealing the same, an intake passage openly communicating between said casing and said cylinder for the admission of gaseous refrigerant at low casing pressure from said casing into said cylinder, a valved discharge passage communicating between said cylinder and the exterior of said casing, a rotary piston of smaller diameter than said cylinder disposed in said cylinder, motor driven shaft means rotatably mounted in concentric relation to said cylinder, an eccentric on said shaft means in driving relation to said piston for rotating said piston within said cylinder in a planetating manner in which said piston and said cylinder are in tangential relationship with minute clearance with respect to the respective facing walls of said piston and cylinder, divider means disposed in a wall of said cylinder between said intake and discharge passages and continuously engaging said piston to establish high pressure and low pressure portions of said cylinder, pump means supplied with lubricating oil from said crankcase for affecting a continuous flow of lubricating oil to a receiver of small capacity relative to said cylinder, said receiver being wholly above said cylinder and the lubricating oil level in said crankcase, whereby the lubricating oil in said receiver is exposed to the pressure within said casing, said receiver having an overflow rim externally of said cylinder and in close vertical proximity to the interior thereof, and a passage through the upper covering wall of said cylinder for conducting lubricating oil from said receiver to the low pressure portion of said cylinder, said piston being in covering relation to said passage except for that portion of the planetation of said piston deriving minimum pressure within said low pressure portion of the cylinder.

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