

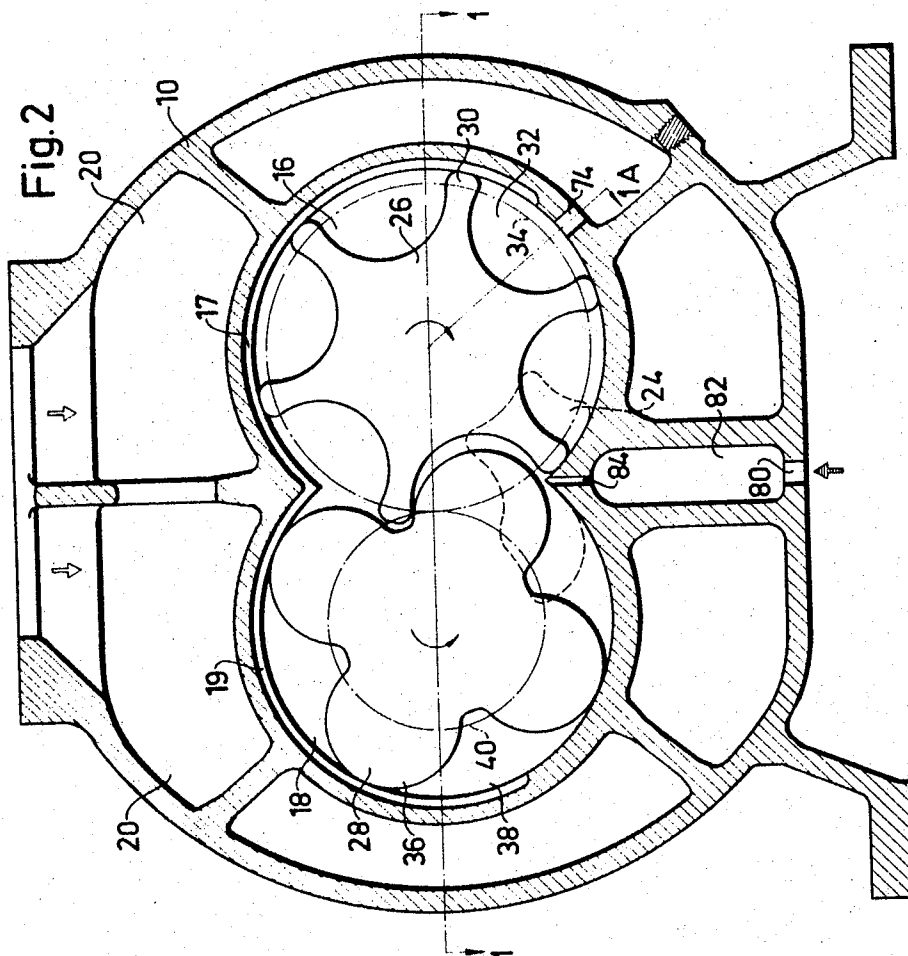
Aug. 19, 1969

L. B. SCHIBBYE
SCREW ROTOR MACHINE

3,462,072

Filed May 2, 1968

5 Sheets-Sheet 2



INVENTOR
Lauritz Benedictus Schibbye
by *Green Marshall Jr.*

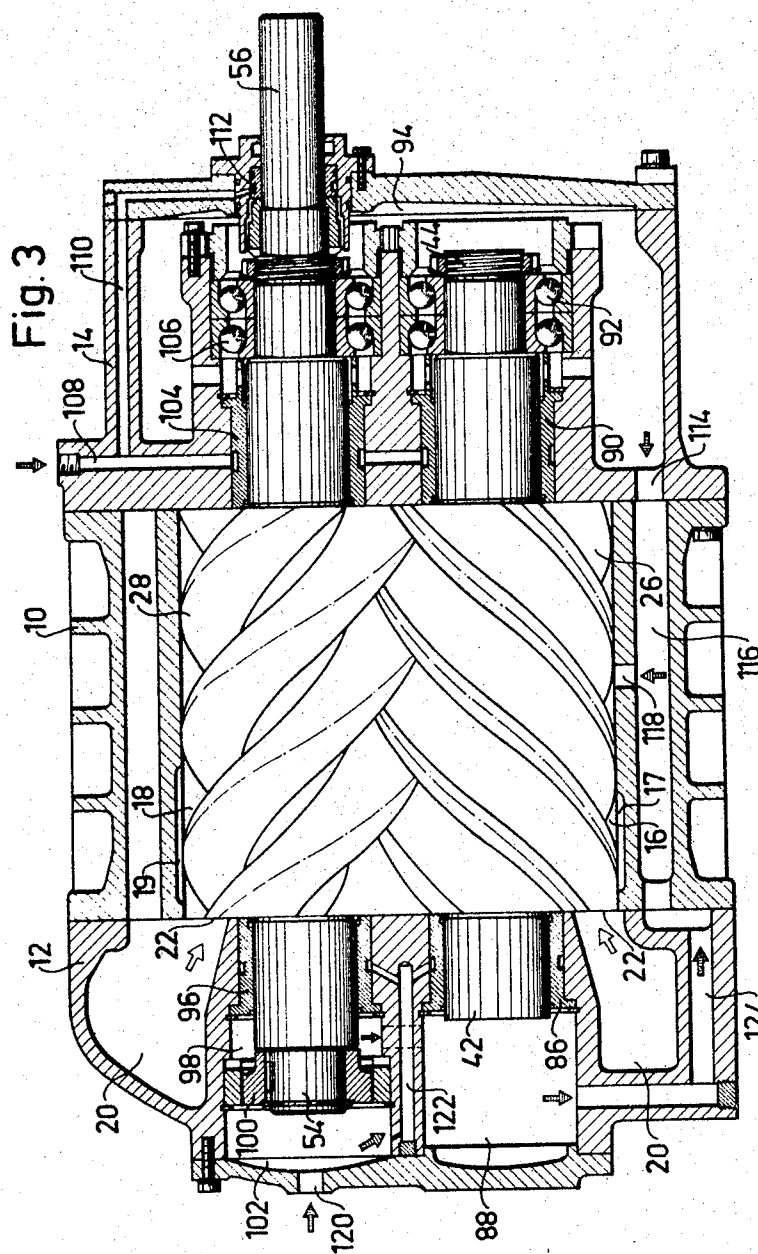
Aug. 19, 1969

L. B. SCHIBBYE
SCREW ROTOR MACHINE

3,462,072

Filed May 2, 1968

5 Sheets-Sheet 3



INVENTOR
Lauritz Benedictus Schibbye

by *Green Marichal Jr.*

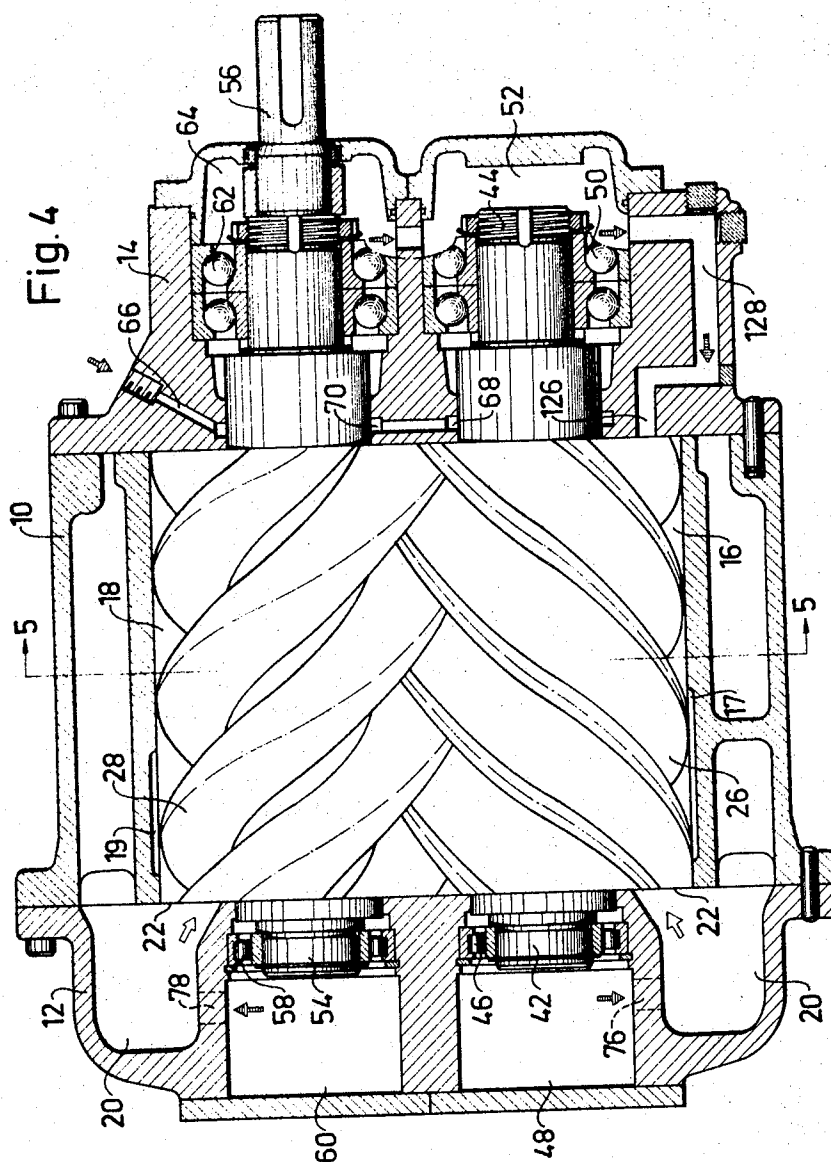
Aug. 19, 1969

L. B. SCHIBBYE
SCREW ROTOR MACHINE

3,462,072

Filed May 2, 1968

5 Sheets-Sheet 4

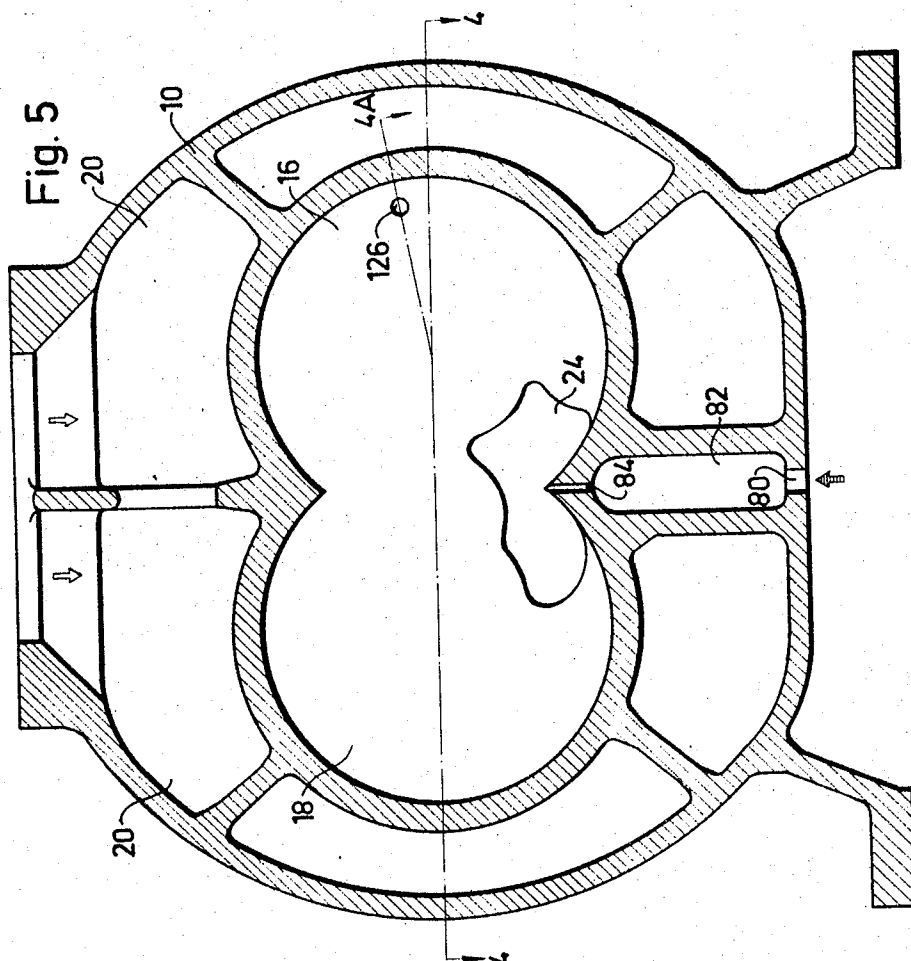


INVENTOR
Lauritz Benedictus Schibbye
by Greer Marichal Jr.

Filed May 2, 1968

SCREW ROTOR MACHINE

5 Sheets-Sheet 5



INVENTOR
Lauritz Benedictus Schibbye
by Green Marshall Jr.

1

3,462,072

SCREW ROTOR MACHINE

Lauritz Benedictus Schibbye, Saltsjo-Duvnas, Sweden, assignor to Svenska Rotor Maskiner Aktiebolag, Nacka, Sweden, a corporation of Sweden

Filed May 2, 1968, Ser. No. 726,135

Claims priority, application Great Britain, May 3, 1967, 20,454/67

Int. Cl. F04c 17/12, 27/00, 29/02

U.S. Cl. 230—143

7 Claims

ABSTRACT OF THE DISCLOSURE

An oil drainage system for a screw rotor compressor with supply of oil to the working space, in which oil from at least one chamber enclosing rotor bearings is drained to an opening in the walls of the working space.

The present invention relates to a screw rotor compressor for a gaseous working fluid of the type having means for introducing a liquid into the working space.

A screw rotor compressor of the actual type comprises at least two meshing rotors cooperating in pairs and provided with helical lands and intervening grooves having a wrap angle of less than 360 degrees. One rotor of a pair is of the male rotor type having at least the major portion of its lands and grooves located outside the pitch circle of the rotor and provided with generally convex flanks. The second rotor of a pair is of the female rotor type having at least the major portion of its lands and grooves located inside the pitch circle of the rotor and provided with generally concave flanks. The compressor further comprises a casing composed of a barrel section, a low pressure end plate section and a high pressure end plate section. The casing provides a working space generally in the form of intersecting bores with coplanar axes surrounded by barrel and end walls, and low pressure and high pressure channels for admittance and discharge of the working fluid to and from the working space, respectively. The casing is further provided with means for introducing a liquid into the working space. Each of the bores encloses one of the rotors. The low pressure and high pressure channels communicate with the working space through low pressure and high pressure ports, respectively, provided in the walls of the working space. Each of the ports is defined by edges in portions of the walls cooperating with the confronting rotor with only a running clearance therebetween. Each of the rotors is provided with coaxial shaft extensions rotationally mounted in bearings in said end plate sections and extending into chambers therein. Means for supply of a sealing liquid around the shafts are located at least in the high pressure end plate section for blocking leakage of working fluid from the working space to the chamber in the end plate section.

In compressors of the actual type the oil supplied to the chambers in the end plate sections for bearing lubrication and other purposes have up to now been drained to the low pressure channel of the casing, as shown for instance in Swedish patent application 3,444/64, which corresponds to United States Patent No. 3,314,597.

As the oil drained from the chambers in the end plate sections circulates within the compressor plant and gets a maximum temperature corresponding to the temperature of the working fluid in the high pressure channel it has to be cooled down before recirculation into the compressor. However, owing to the temperature of the available cooling fluid and the practically possible size of the cooler the oil introduced into the compressor will have a considerably higher temperature than the temperature

2

of the working fluid to be compressed. The contact between the working fluid and the oil of the higher temperature during the inflow phase results in a heating of the working fluid and thus in a decrease of the volumetric efficiency. There is also a considerable power required for the inflow of the oil from the low pressure channel through the low pressure port into the working space. Furthermore a certain amount of the oil flows through the bore of the male rotor and has to be accelerated to the high speed of the tips of the lands thereof.

A special problem arises in compressors forming a part of a refrigeration cycle using a working fluid of the type being dissolvable to a considerable extent in the oil, such as fluids of the type normally referred to as Freon, and commercially known for instance as R-12 and R-22. The oil supplied to the chambers in the end plate sections for bearing lubrication, shaft sealing, thrust balancing and similar purposes, normally has a pressure exceeding the pressure in the high pressure channel of the compressor and the amount of working fluid solved therein is considerable. When the chambers are drained to the low pressure channel most of the working fluid is evaporated out of the oil as the solubility decreases with decreasing pressure. The amount of working fluid in this way supplied to the low pressure channel is so large that it will need a very considerable portion of the displacement volume of the compressor. The same amount of working fluid is during the compression solved in the oil. Owing to this fact the amount of working fluid passing through the compressor and circulating within the complete cycle will be much less than the nominal capacity of the compressor or in other words the volumetric efficiency of the compressor will be low.

All the factors mentioned above will be more accentuated the smaller the dimensions of the compressor are as the amount of oil supplied to the chambers in the end plate sections can not be reduced in the same proportion as the reduction of the amount of working fluid passing through the compressor.

The purpose of the present invention is to reduce the disadvantages mentioned above by a new type of draining system the details of which will be apparent as this specification proceeds.

The invention will now be described more in detail in conjunction with a number of preferred embodiments shown in the accompanying drawing in which:

FIG. 1 shows a longitudinal section through a compressor according to the invention taken along line 1—1 in FIG. 2 and fractionally along line 1—1A in FIG. 2, FIG. 2 shows a cross section taken along line 2—2 in FIG. 1,

FIG. 3 shows a section similar to FIG. 1 through another compressor according to the invention,

FIG. 4 shows a longitudinal section through a third compressor according to the invention taken along line 4—4 in FIG. 5 and fractionally along line 4—4A in FIG. 5 and

FIG. 5 shows a cross section taken along line 5—5 in FIG. 4.

The compressor shown in FIGS. 1 and 2 is especially intended for working fluids which are unsolvable in oil and comprises a casing composed of a barrel section 10, a low pressure end plate section 12 and a high pressure end plate section 14 providing a working space generally composed of two intersecting bores 16, 18 with parallel axes. The casing further provides a low pressure channel 20 and a high pressure channel, not shown, for the working fluid which low and high pressure channels communicate with the working space 16, 18 through a low pressure port 22 and a high pressure port 24, respectively. The barrel walls of the working space 16, 18 are within an area adjacent to the low pressure port 22 provided

with relieved portions 17, 19 in the way described for instance in Swedish Patent 203,185. The ports 22, 24 are disposed in unrelieved portions of the walls of the working space and defined by edges therein.

In the working space 16, 18 two cooperating rotors, one female rotor 26 and one male rotor 28, are located with their axes coaxial with the axes of the bores 16 and 18, respectively. The female rotor 26 is provided with six helical lands 30 with intervening grooves 32, each having a wrap angle of about 200 degrees. The major portions of the lands 30 and grooves 32 are located inside the pitch circle 34 of the rotor 26. The flanks of the grooves 32 are shaped such that in a plane transverse to the axis of the rotor 26 the portions thereof lying inside the pitch circle 34 are concave and follow a circular arc having its centre on the pitch circle 34 whereas the portions thereof lying outside the pitch circle 34 are convex and follow circular arcs, each having its centre located adjacent to the pitch circle 34. The male rotor is provided with four helical lands 36 with intervening grooves 38, each having a wrap angle of about 300 degrees. The major portions of the lands 36 and grooves 38 are located outside the pitch circle 40 of the rotor 28. The flanks of the lands 36 are generally convex and shaped such that a continuous sealing line is provided between the rotors 26, 28 when the lands 30, 36 of one rotor passes into and out of mesh with a cooperating groove 38, 32 of the second rotor. The walls of the working space surrounding the rotors are shaped in such a way that there is only a running clearance between the unrelieved portions thereof and the confronting rotor lands.

The female rotor 26 is provided with axial shaft extensions 42, 44, one 42 being mounted in a radial bearing 46 of the roller bearing type in the low pressure end section 12 and extending out into a chamber 48 therein and the other one 44 being mounted in a combined radial and thrust bearing 50 of the ball bearing type in the high pressure end plate section and extending out into a chamber 52 therein. The male rotor 28 is also provided with axial shaft extensions 54, 56, one 54 being mounted in a radial bearing 58 similar to the bearing 46 in the low pressure end plate section 12 and extending out into a chamber 60 therein and the other one 56 being mounted in a combined radial and thrust bearing 62 similar to the bearing 50 in the high pressure end plate section 14 and extending through a chamber 64 therein and projecting outside the end plate section 14 to form the external shaft of the compressor.

Oil having a pressure exceeding the pressure in the high pressure channel is supplied from a pressure oil source, not shown, through a channel 66 in the high pressure end plate section 14 to annular grooves 68, 70 therein, each of those grooves 68, 70 surrounding one of the shafts 44, 56, which oil blocks the flow of working fluid along the shafts and thus acts as a sealing liquid between the working space 16, 18 and the chambers 52, 64. Most of the oil supplied to the grooves 68, 70 flows along the shafts 44, 56 to the chambers 52, 64 and lubricates the bearings 50, 62. The chambers 52, 64 are drained through a channel 72 in the casing to an opening 74 in the barrel wall of the bore 16 enclosing the female rotor 26. The opening 74 is located within an area of the wall of the bore 16 which faces a groove 32 in the female rotor 26 which groove 32 sometimes is in communication with and sometimes by the trailing land 30 thereof is brought out of communication with the low pressure channel 20.

The bearings 46, 58 in the low pressure end plate section 12 are lubricated by oil flowing along the shafts 42, 54 from the working space 16, 18 to the chambers 48, 60. The chambers 48, 60 are drained to the low pressure channel 20 through channels 76 and 78, respectively.

Oil from the pressure oil source for supplying to the channel 66, mentioned above, is also supplied through an

opening 80 in the casing to a chamber 82 therein from where it is introduced into the working space 16, 18 through nozzles 84.

The compressor acts in the following way. The male rotor 28 is brought into rotation by a prime mover, not shown, connected to the shaft 56. The female rotor 26 is brought into a corresponding rotation by direct contact between the rotors 26, 28. Working fluid is then sucked in through the low pressure channel 20 and the low pressure port 22 to a pair of rotor grooves 32, 38. The working fluid flows continuously along and circumferentially around the axes of the rotors 26, 28. After a certain angle of rotation the grooves 32, 38 are brought out of communication with the low pressure port 22 and a land 30, 36 of one rotor enters a groove 32, 38 of the cooperating rotor thus the compression phase commences. During the continuing compression, oil is injected through the nozzles 84 for cooling, sealing and lubrication purposes. When the rotors 26, 28 reach the angular position in which the grooves 32, 38 are brought into communication with the high pressure port 24 the working fluid and the oil enclosed in the grooves begin to flow out through the port 24 through the high pressure channel to an oil separator, not shown. The working fluid is then brought forward to the place where it is needed whereas the oil by means of a pump is circulated through a cooler back to the compressor. A certain amount of the oil thus circulated back to the compressor is supplied through the channel 66 to the grooves 68, 70 and blocks the flow of working fluid along the shafts 44, 56. A fraction of this oil flows along the shafts 44, 56 into the working space 16, 18 and is rapidly brought away through the high pressure port 24. Most of the oil, however, flows along the shafts 44, 56 through the bearings 50, 62, which are lubricated thereby, to the chambers 52, 64. From the chambers 52, 64 the oil is drained through the channel 72 and the opening 74 to the bore 16 enclosing the female rotor 26. Owing to the location of the opening 74 the oil is supplied to the working space during the last period of the filling and first period of the compression when the pressure rise is practically negligible so that the pressure in the chambers 52, 64 will be practically the same as the pressure in the low pressure channel 20. Owing to the location of the opening 74 the oil cannot influence on the conditions in the low pressure port and thus not decrease the volumetric efficiency of the compressor in the same way as when oil is drained to the low pressure channel as discussed in the introduction of the specification. The inflow losses of the oil when entering the working space will also be considerably decreased in relation to those arising from inflow from the low pressure channel. Furthermore all the drained oil will be accelerated by the female rotor 26 which has a very considerably lower tip speed than the male rotor 28 so that the acceleration losses are brought down to a real minimum. The oil supplied through the opening 74 will during the compression act in the same way as the oil injected through the nozzles 84 so that the amount of injected oil can be reduced correspondingly and under certain conditions be completely eliminated. The bearings 46, 58 in the low pressure end plate section 12 will be lubricated by oil flowing along the shafts 42, 54 to the chambers 48, 60. Those chambers are drained to the low pressure channel 20 which is of no practical importance owing to the very small amount of oil in relation to the amount of working fluid passing through the channel 20.

The compressor shown in FIG. 3 is generally of the same type as the compressor shown in FIGS. 1 and 2 and the corresponding details have been denoted by the same reference numbers but it is especially designed for handling working fluids dissolvable in oil. The difference between the two compressors is essentially restricted to the bearings and the oil system. The shaft 42 of the female rotor 26 is mounted in the low pressure end plate section by a radial bearing 86 of journal bearing type and extends out into a chamber 88 therein. The shaft 44 of

5

the female rotor 26 is mounted in the high pressure end plate section 14 by a radial bearing 90 of journal bearing type and a thrust bearing 92 of ball bearing type and extends out into a chamber 94 therein. The shaft 54 of the male rotor 28 is mounted in the low pressure end plate section 12 by means of a radial bearing 96 of journal bearing type and extends through a chamber 98 therein and carries a balancing piston 100 located in a pressure chamber 102. The shaft 56 of the male rotor 28 is mounted in the high pressure end plate section 14 by means of a radial bearing 104 of journal bearing type and a thrust bearing 106 of ball bearing type and extends through the chamber 94 therein and projects outside the high pressure end plate section 14 to form the external shaft of the compressor. Oil is supplied from the pressure oil source through a channel 108 to the journal bearings 90, 104 in the high pressure end plate section 14 and also via a branch channel 110 to a shaft sealing 112 surrounding the projecting portion of the shaft 56. The chamber 94 in the high pressure end plate section 14 is drained through a channel 114 in the end plate section 14, through a chamber 116 in the barrel section 10 and through an opening 118 in the barrel wall to the bore 16 enclosing the female rotor 26. Oil is further supplied through an opening 120 in the low pressure end plate section 12 to the pressure chamber 102 and from there through a channel 122 to the journal bearings 86, 96. The chambers 88, 98 in the low pressure end plate section 12 are drained through a channel 124 in the end plate section, through the chamber 116 and through the opening 118 to the bore 16. The opening 118 is located within an area of the barrel wall facing a groove 32 in the female rotor 26 which groove is blocked from communication with the low pressure port 22 by the trailing land 30 of the groove 32.

The compressor shown in FIG. 3 acts generally in the same way as the compressor shown in FIGS. 1 and 2. However, the compressor shown in FIG. 3 is designed for a higher pressure difference and is particularly intended to be used in a refrigeration cycle using a working fluid that is to a considerable degree dissolvable in oil. The oil supplied through the channel 108 to the journal bearings 90, 104 not only lubricates those bearings but also acts as a sealing liquid blocking the flow of working fluid along the shafts 44, 56 from the working space 16, 18 to the chamber 94. After the passage of the journal bearings 90, 104 the oil lubricates the thrust bearings 92, 106 and flows to the chamber 94. The oil supplied through the channel 110 lubricates a mechanical seal which blocks leakage of working fluid along the shaft 56 from the chamber 94 to the outside of the compressor. The pressure of the oil supplied through the opening 120 acts upon the piston 100 and thus reduces the thrust of the male rotor 26 acting upon the thrust bearing 106. A certain amount of the oil flows along the piston 100 to the chamber 98. The remaining oil flows through the channel 122 to the journal bearings 86, 96 where it lubricates the bearings and acts as a sealing liquid blocking flow of working fluid along the shafts 42, 54, after which it flows to the chambers 88, 98. The oil in the chambers 88, 98 is drained through the channel 124 to the chamber 116 where it meets the oil drained from the chamber 94 and then continues into the bore 16 through the opening 118. Owing to the location of the opening 118 in relation to the low pressure port 22 and the rotors 26, 28 the pressure in the groove 32 into which the oil is introduced will be lower than the pressure in the high pressure channel. A certain amount of the working fluid solved in the oil supplied through the channel 108 and the opening 120 will thus be evaporated out of the oil. However, as this working fluid will never be brought into communication with the low pressure channel 20 the inflow conditions through the low pressure port 22 will not be affected by this further introduction of working fluid. As the pressure of the working fluid increases during the compression the amount of working fluid solved in the

6

oil will increase and the mass of gaseous working fluid leaving the compressor plant will be the same as the mass of gaseous working fluid supplied through the low pressure channel. The working fluid alternately solved in and evaporated from the oil will thus just circulate within the compressor plant and does not affect the inflow conditions of the compressor.

The compressor shown in FIGS. 4 and 5 differs from the compressor shown in FIGS. 1 and 2 only with regard to the location of the drainage opening 126 and the form of the draining channel 128. The opening 126 is thus located in the high pressure end wall of the working space and located correspondingly to the opening 118 in FIG. 3 with regard to the low pressure port 22 and the rotors 26, 28. The action will also be practically the same as that of the compressors described above.

What I claim is:

1. Screw rotor compressor for a gaseous working fluid comprising at least two meshing rotors cooperating in pairs and provided with helical lands and intervening grooves having a wrap angle of less than 360 degrees, one rotor of a pair being of the male rotor type having at least the major portion of its lands and grooves located outside the pitch circle of the rotor and provided with generally convex flanks, the second rotor of a pair being of the female rotor type having at least the major portion of its lands and grooves located inside the pitch circle of the rotor and provided with generally concave flanks, and further comprising a casing composed of a barrel section, a low pressure end plate section and a high pressure end plate section, said casing providing a working space generally in the form of intersecting bores with coplanar axes surrounded by barrel and end walls, and low pressure and high pressure channels for admittance and discharge of the working fluid to and from the working space, respectively, said casing being further provided with means for introducing a liquid into said working space, each of said bores enclosing one of said rotors, said low pressure and high pressure channels communicating with said working space through low pressure and high pressure ports, respectively, provided in the walls of the working space, each of said ports being defined by edges in portions of said walls cooperating with the confronting rotor with only a running clearance therebetween, each of said rotors being provided with coaxial shaft extensions rotationally mounted in bearings in said end plate sections and extending into chambers therein, and means at least in said high pressure end plate section located between said chamber therein and the working space for supply of a sealing liquid around said shafts, characterized in that at least one of said chambers communicates with an opening in said walls of the working space for drainage of liquid from the chamber to the working space.

2. Screw rotor compressor as defined in claim 1, in which said opening is located within an area of said walls facing a rotor groove adjacent to a land of the rotor when said land is in a position in which it directly blocks the communication between one of said adjacent rotor grooves and the low pressure port.

3. Screw rotor compressor as defined in claim 2, in which said area faces the groove preceding said land.

4. Screw rotor compressor as defined in claim 3, in which said opening is so located that the communication between the facing rotor groove and the low pressure port is all the time blocked by said rotor land.

5. Screw rotor compressor as defined in claim 1, in which said opening is located in the wall of the working space surrounding the female rotor.

6. Screw rotor compressor as defined in claim 5, in which said opening is located in the barrel wall.

7. Screw rotor compressor as defined in claim 5, in which said opening is located in the high pressure end

7

wall of the working space cooperating with the high pressure end of the rotor.

8

3,265,292 8/1966 Schibbye ----- 230—143
3,265,293 8/1966 Schibbye ----- 230—143

References Cited

UNITED STATES PATENTS

1,409,868 3/1922 Kien ----- 230—207
2,082,412 6/1937 Morton ----- 230—207
3,161,349 12/1964 Schibbye ----- 230—205

DONLEY J. STOCKING, Primary Examiner

5 WILBUR J. GOODLIN, Assistant Examiner

U.S. Cl. X.R.

230—205, 207, 210