May 28, 1968 C. R. KILGORE ETAL 3,385,514

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 1

Fig. 1

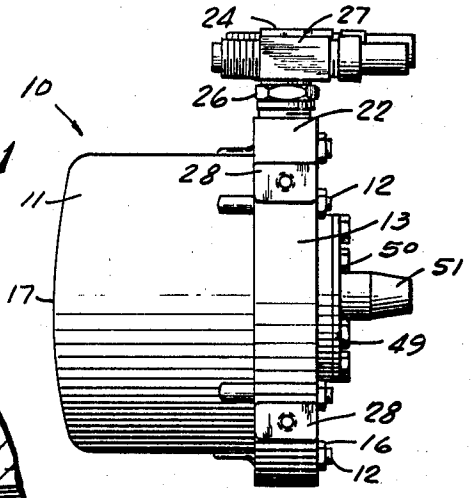


Fig. 6

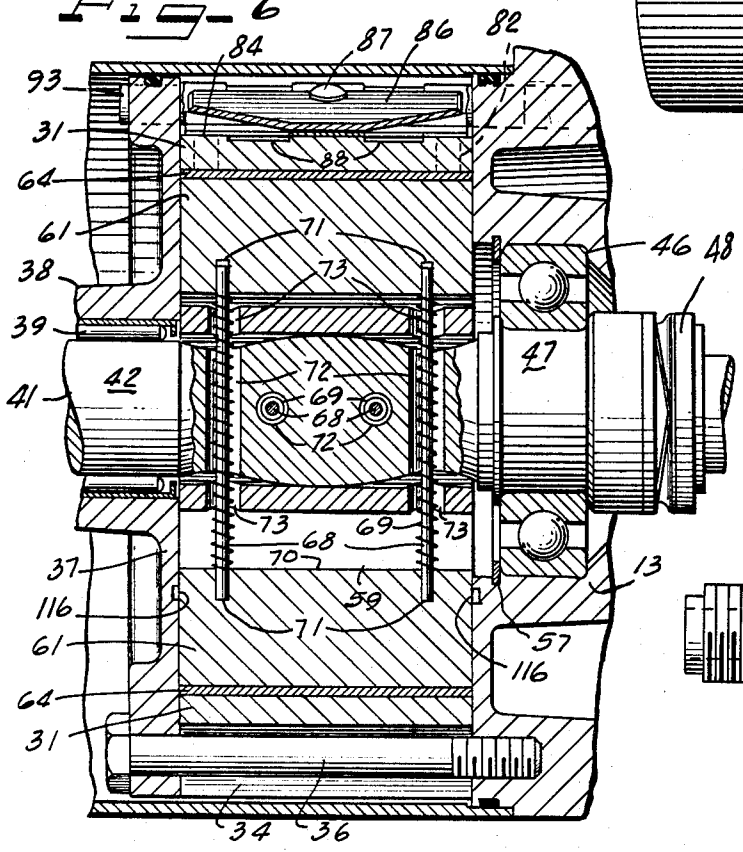
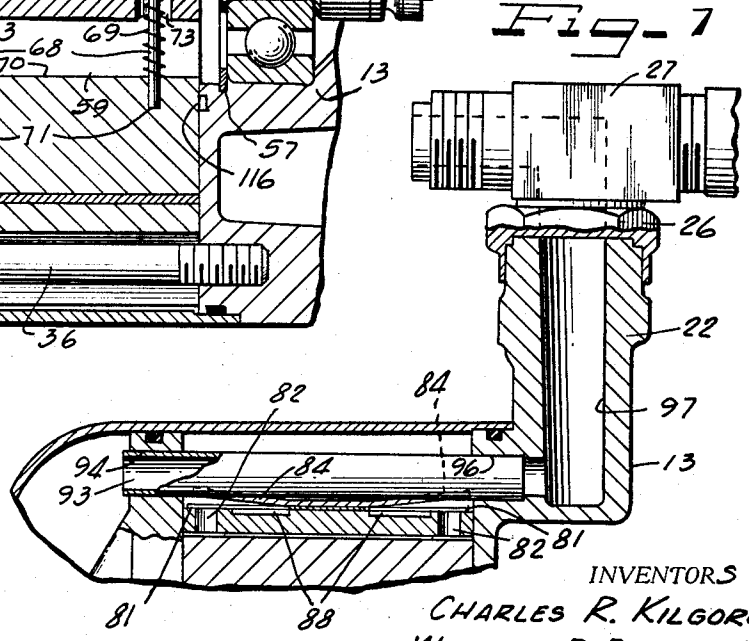


Fig. 7



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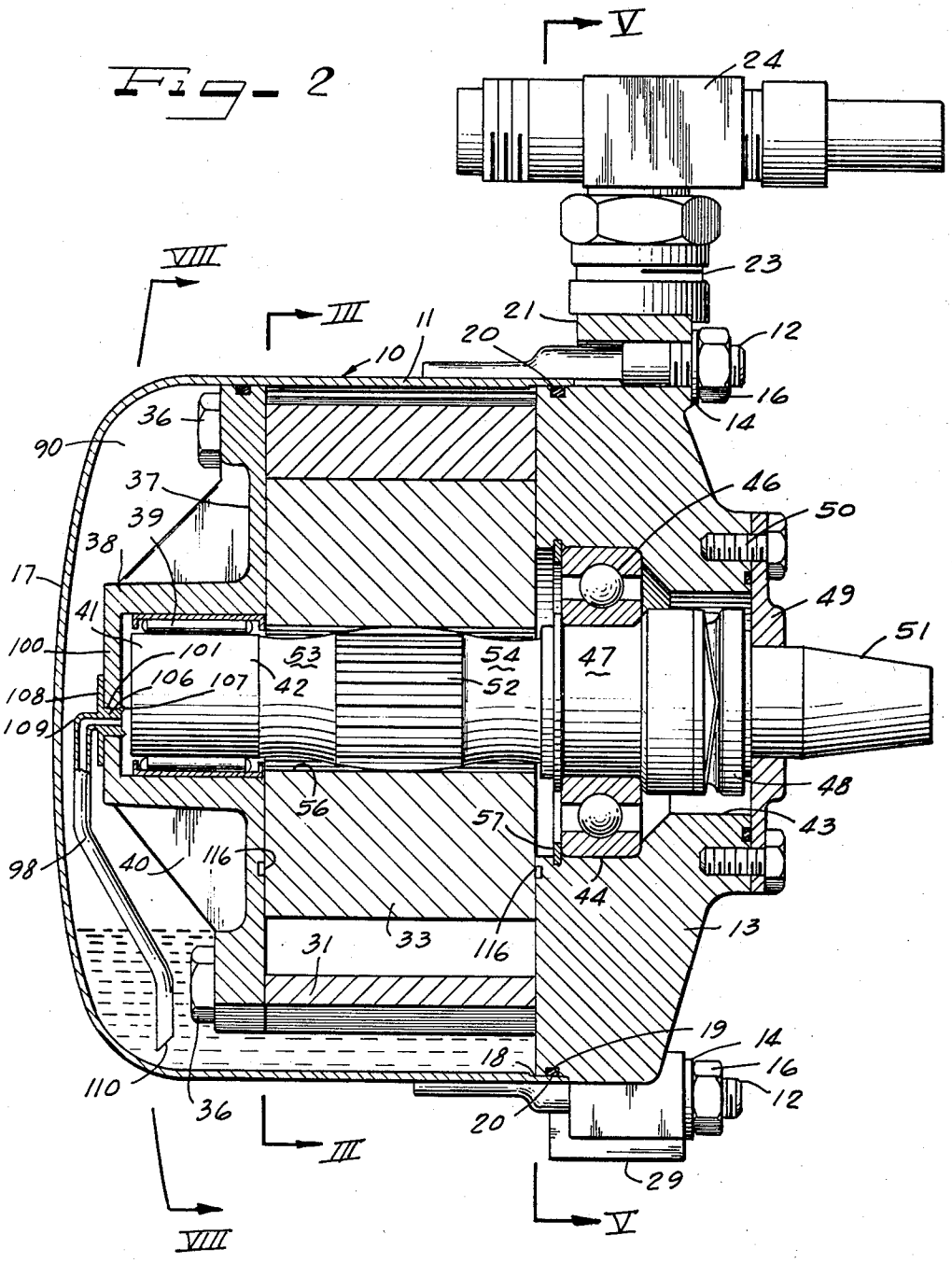
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May 28, 1968 C. R. KILGORE ETAL 3,385,514

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 2

Fig. 2



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May 28, 1968 C. R. KILGORE ETAL 3,385,514

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 3

Fig. 4

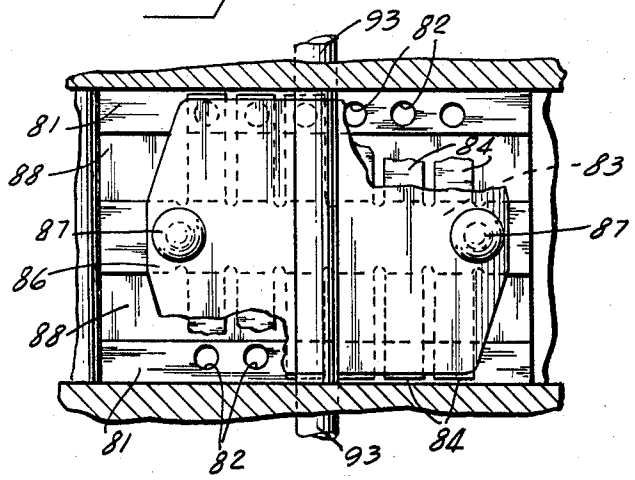
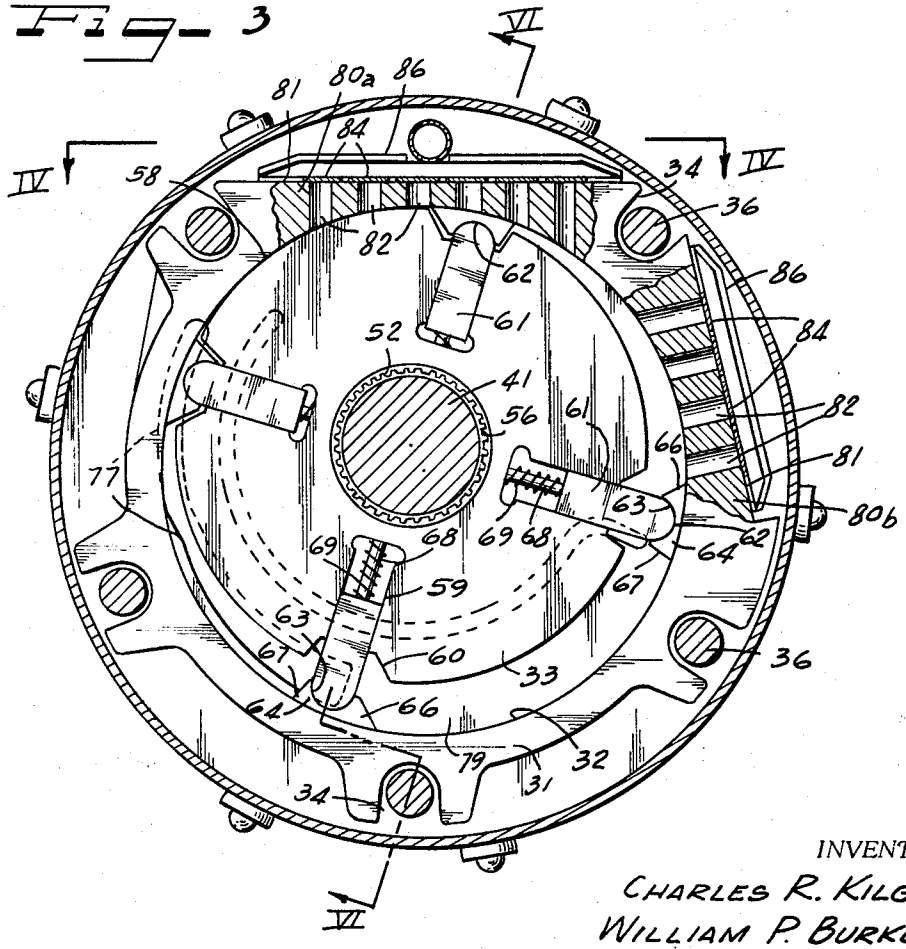


Fig. 3



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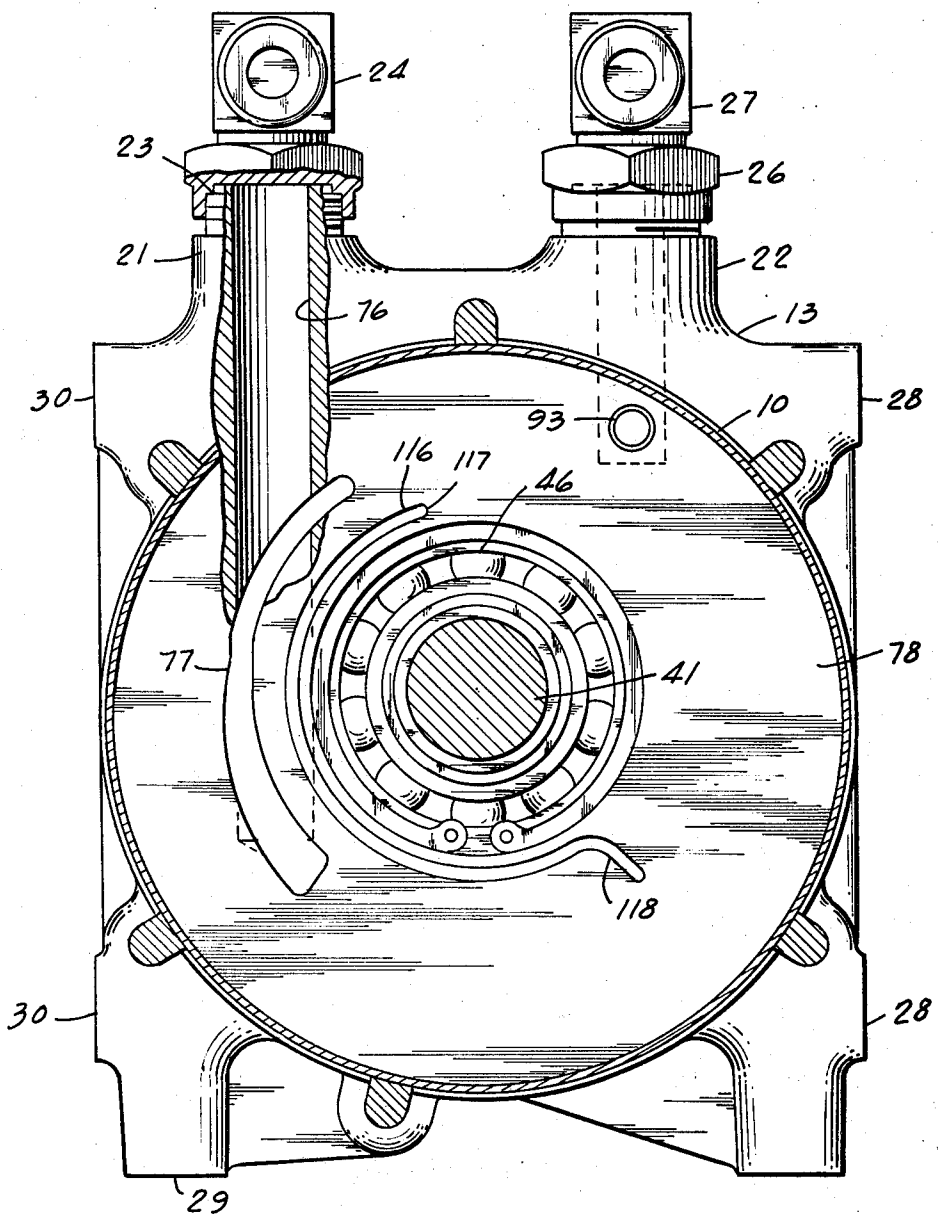
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May 28, 1968 C. R. KILGORE ETAL 3,385,514

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 4

Fig. 5



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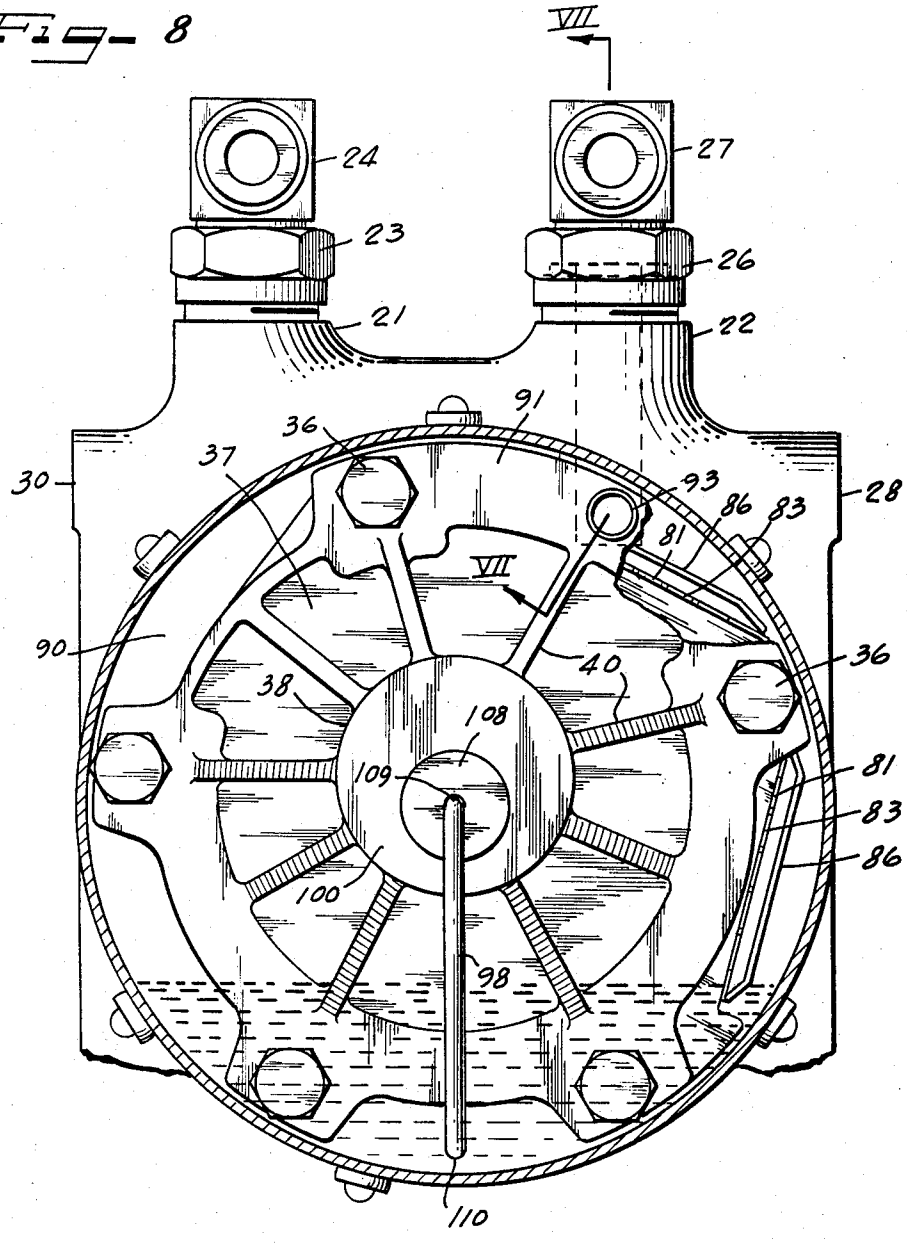
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REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 5

Fig. 8



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United States Patent Office 3,385,514
Patented May 28, 1968

3,385,514

REFRIGERANT VAPOR COMPRESSOR

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Filed Apr. 11, 1966, Ser. No. 541,837
17 Claims. (Cl. 230—152)

This invention relates to a refrigerant vapor compressor characterized by a double row of circumferentially spaced discharge ports controlled by reed valve means so that the ports open in unison and close in pairs in discharging to a pressure reservoir. A swivel tube dips into the lower part of the pressure reservoir and carries liquid lubricant to the inside of the pump, regardless of the pump orientation in left, right or vertical mounting positions, and special scavenging grooves in the faces of the bearing plates minimizes leakage into the suction side of the compressor.

On the drawings:

FIGURE 1 is a reduced scale overall outside view of the rotary vane vapor compressor of the present invention adapted for external drive;

FIGURE 2 is a longitudinal cross-sectional view of the compressor of FIGURE 1;

FIGURE 3 is an axial cross-sectional view taken on line III—III of FIGURE 2;

FIGURE 4 is a fragmentary cross-sectional view with parts broken away to illustrate additional details of the reed valve means and discharge port arrangement and is taken substantially on line IV—IV of FIGURE 3;

FIGURE 5 is an inside view of the front bearing plate taken on line V—V of FIGURE 2;

FIGURE 6 is a fragmentary cross-sectional view taken on line IV—IV of FIGURE 3;

FIGURE 7 is a fragmentary cross-sectional view taken on line VII—VII of FIGURE 8; and

FIGURE 8 is an axial cross-sectional view taken on line VIII—VIII of FIGURE 2.

As shown on the drawings:

The rotary vane vapor compressor of the present invention is shown generally at **10** and comprises an outer casing **11** connected by means of a plurality of lug bolts **12** to a front bearing plate **13**, the lug bolts **12** cooperating with suitable washers **14** and nuts **16**, thereby to form an integral housing assembly. The casing **11** is generally cup-shaped in configuration and has an end wall **17**. A recessed lip **18** overlies a correspondingly recessed portion **19** formed in the front bearing plate **13** and in which is seated an O ring sealing member **20**.

The bearing end plate **13** is characterized by the formation thereon of two separate bosses **21** and **22**. The boss **21** receives suitable coupling members shown generally at **23** for connection thereto an inlet vent valve **24** (FIGURE 5).

The boss **22** also has suitable coupling means **26** connected thereto by means of which a discharge vent valve may be connected to the compressor, as shown at **27**.

As is most clearly shown in FIGURES **1** and **5**, the front bearing plate **13** has plural mounting pads formed thereon disposed to afford right, left and vertical mounting. The separate mounting pads are arranged in different planes on plural sides so that a first set of mounting pads is shown at **28, 28**, while a second set of mounting pads is shown at **29, 29** and a third set of mounting pads is shown at **30, 30**.

A casing cylinder is shown at **31** and has formed therein a bore **32** in which is received a rotor **33**. The casing cylinder **31** has a plurality of circumferentially spaced slots **34** which engage fastening bolts **36** used to secure a rear bearing plate **37** in assembly with the front bearing plate **13**.

The rear bearing plate **37** has an axially projecting boss **38** forming a bearing housing in which is received a needle bearing assembly **39**. Circumferentially spaced stiffening webs **40** are situated between the housing **38** and the radially extending portion of the bearing plate **37**. A rotatable shaft **41** has a bearing portion **42** journaled in the needle bearing assembly **39** and projects through the bore **32** of the casing cylinder **31** and through an opening **43** formed in the front bearing plate **13**. The opening **43** is counterbored at **44** to receive a ball bearing assembly shown generally at **46**, the inner race of which engages against a bearing portion **47** on the shaft **41**. Thus, the shaft **41** is journaled in the front and rear bearing plates **13** and **37**. A rotary seal assembly **48** is provided to effect a shaft seal with the shaft **41** and a seal end plate **49**. The plate **49** is attached to the end of the front bearing plate **13** by a plurality of bolts **50**. A coupling portion **51** on the shaft **41** projects outwardly for connection to a suitable source of power for the compressors.

Between the bearing portions **42** and **47**, the shaft **41** has a splined portion **52** and disposed intermediate reduced sections **53** and **54**. The splined section **52** is corotatable connected with a correspondingly splined internal bore **56** formed in the rotor **33**, thereby permitting the rotor to move axially in a self-centering action between the front and rear bearing plates **13** and **37**, while insuring a good rotatable drive connection between the rotor **33** and the shaft **41**.

A snap-type retainer ring **57** is used to hold the ball bearing assembly **46** in place in the counterbore **44** of the front bearing plate **13**.

As shown in FIGURE 3, the rotor **33** is disposed within the bore **32** of the casing cylinder in eccentrically offset relation relative to the bore axis and a portion of the rotor **33** is in tangent seal contact with the casing cylinder as shown at **58**.

The rotor **33** has formed in the peripheral surface thereof a plurality of circumferentially spaced vane slots each identified at **59**, which slots are enlarged at their outer ends to form recesses **60**. Received within each vane slot is a slidable vane **61** having a rounded outer end **62** which engages into the socket **63** of a vane shoe **64**. The toe or leading end **66** of each shoe **64** is somewhat longer than the heel or trailing end **67** of each respective shoe **64**, thereby permitting the shoe to develop a planing action with respect to the adjoining bore wall.

A pair of pins **68** extend between oppositely disposed vanes **61** and each pin **68** is surrounded by a coil spring **69** bottomed against an end wall **70** of a corresponding vane, in order to bias the vanes in outward direction. The ends of the pins **68, 68** are received in corresponding recesses **71, 71**. The shaft **41** is provided with transverse passages **72** through which the pins **68, 68** and corresponding springs **69** extend. As shown in FIGURE 6, the rotor is also formed with passages **73** for a similar purpose.

Referring now more particularly to FIGURES 3 and 5, it will be noted that the boss **21** has extending therethrough an inlet passage **76** which communicates with a kidney-shaped inlet recess **77** formed in a radial face **78** of the front bearing plate **13**. The kidney-shaped inlet recess **77** is located adjacent the crescent-shaped working chamber **79** which extends around from the point of tangent contact **58** through an inlet side circumferentially towards an outlet side.

Fluid from the inlet vent valve **24**, therefore, passes through the passage **76** and the recess **77** to fill the expanding inlet area between adjacent vanes **61** as they move through the working chamber **79**.

On the discharge side of the compressor, the casing cylinder is particularly characterized by the formation of

boss 80(*a*) adjacent the tangent contact area 58 and having formed thereon a flat valve-seating surface 81 (FIGURE 4). A double row of openings extends from the working chamber 79 outwardly and intersects the valve-seating surface 81 at opposite sides thereof, thereby forming two axially separated rows of circumferentially spaced discharge ports each indicated at 82.

A reed valve means is provided to form plural valves for independent regulation of the discharge ports 82 and in this regard, the valve means comprises a main body member 83 having a plurality of digital projections or reeds, each of which is shown at 84. Each reed 84 comprises a spring finger for overlying a corresponding port 82, thereby to engage the valve-seating surface 81 and regulating the opening and closing of a corresponding opening or port 82.

A holddown and travel-limiter clamp is shown at 86 and is fastened to the boss by a pair of spaced fasteners 87, 87. The reed valve element 83 can be made of a suitable resilient material such as spring metal. Referring to FIGURE 4, specifically, it will be noted that a recess groove 88 is located adjacent each row of openings or ports 82, thereby to insure venting of the underside of the individual reeds 84 and further insuring that the reeds 84 will seat against the valve seating surface 81 in register with a corresponding opening or port 82.

A second boss 80(*b*) is provided on the casing cylinder 31 to provide a second series of openings or ports extending circumferentially towards the inlet side of the compressor so that there is an upper set of valves and a lower set of valves. Since the construction is otherwise identical, like reference numerals have been applied to like parts.

Fluid discharged through the ports 82 is directed into a pressure reservoir 90 formed inside of the casing 11 and outside of the casing cylinder 31. It will be noted that the rear bearing plate 37 has a plurality of circumferentially spaced outwardly extending turbulence barriers 91, which also serve to provide openings for the bolts 36.

In order to conduct pressured fluid from the pressure reservoir 90 to a point of utilization, a discharge tube is provided which is shown at 93. An inner end of the tube 93 is located in an opening 94 formed in the rear bearing plate 37. The tube 93 extends axially of the pump and is received in an opening 96 formed in the front bearing plate 13. A passage 97 formed in the boss 22 conducts the fluid to the discharge vent valve 27.

As shown in FIGURES 2 and 8, a swivel-mounted lubricant pick-up tube is provided at 98. The tube 98 is positioned in FIGURES 2 and 8 as it would dwell with the compressor mounted in the vertical position. Thus, it will be noted that a radial end wall 100 formed in the housing 38 of the rear bearing plate 37 has an opening 101 formed therein through which extends a hub 106. The hub 106 is offset at 107 to retain the tube 98 in relatively rotatable assembly in the wall 100. A flange 108 is formed on the hub 106 in spaced relation to the offset portion 107 and overlies the wall 100. The tube has an L portion 109 and depends gravitationally towards the lowermost portion of the pressure reservoir 90 opening in an inclined end wall 110.

In operation, counterclockwise rotation of the rotor 33 and the attached vanes 61 and shoes 64, as viewed in the orientation of FIGURE 3, causes pressure in the suction side of the compressor to be reduced. This, in turn, causes gaseous refrigerant and lubricant in droplet form to flow through the suction vent valve 24 and to enter the suction passage 76 and pass through the suction kidney 77 to fill the cavity between the vanes 61, 61. In FIGURE 3, the vane at the lower left-hand side of the drawing is shown at the point where its leading edge of the shoe 64, as shown at 66, has just about cut off the working chamber 79 from the suction kidney 7. Thus, compression is about to commence in the corresponding portion of the working chamber between the vanes.

The relative position of the rotor 33 and the vane 61 at the time the discharge reed valve 84 first operates depends upon the discharge pressure. At very low discharge pressures, nearly all of the reed valves 84 are open, as for example, when the vanes, shown in the right-hand section of FIGURE 3, are in the position therein depicted. At maximum pressure, only the reed valves 84 between the uppermost vane and the point of tangent contact 58 would open. Both sets of reed valves 84 in both of the bosses 80(*a*) and 80(*b*) open upon start-up. This purges the space swept by the vanes 61 of lubricant and liquid refrigerant and promotes rapid pressure build-up.

The lowermost reed valves in the secondary valve set, i.e., that valve set farthest from the point of tangent contact 58, are located so that upon startup they will open shortly after suction kidney cut-off, thus preventing liquid lock.

It is contemplated by the present invention that each of the openings or discharge ports 82 will be sized to have a diameter somewhat smaller than the circumferential width of a corresponding shoe 64. Such relationship accomplishes two objectives. First of all, compressed gas will not by-pass across the ports 82 from the leading to the trailing cavity as the shoe crosses the port openings 82. Secondly, the individual reed valves 84 are given time to close before each respective shoe 64 passes. If the shoe 64 were to pass a valve port 82 before the reed valve 84 for that port 82 closed, compressed refrigerant would flood through the valve port 82 into the trailing cavity. This would raise discharge temperatures and horsepower required to drive the compressor.

In accordance with the present invention, rapid valve opening and closing is promoted by the low mass and short travel of the individual reed valves 84 and by venting the under surface of the reed valves 84 with the relief grooves 88, 88. Such grooves 88, 88 prevent the reed valves 84 from clinging to the casing cylinder 31 at any point other than the valve seating surfaces 81, 81. The grooves 88, 88 also assist in venting the discharged fluids from the valve area.

One of the features of the compressor of the present invention resides in the low level of valve noise which occurs because all of the valves, or a percentage of them, depending on the discharge pressure, open simultaneously but close in pairs. Further, the low valve mass and the short valve travel also promotes quiet operation.

It will be noted upon referring to FIGURE 3, that the inner end of the last pair of discharge ports 82, 82 is located very near to the line of tangency 58. This is specifically arranged to minimize the volume of fluid which would otherwise become trapped between the line of tangency and the shoe 64 after the shoe 64 passes the last pair of discharge ports 82, 82. Accordingly, vane thumping is eliminated which might occur if the pressure is relieved by inward displacement of the vanes because of trapped fluid.

The valve arrangement of the present invention also affords greatly improved reduction in over-compression, thereby reducing drive torque, discharge temperature, vibration and leakage through the tangency clearance. It also reduces leakage through the annual clearances between the adjoining faces of the bearing plates 13 and 37 and the rotor 33. Fluids which thus leak into the suction side of the compressor tend to throttle or subtract from the charge of incoming gas and thus reduce volumetric efficiency.

The lubricant in aerosol and droplet form that enters the compressor lubricates the vanes 61, the shoes 64, the cylinder walls 32 and the rotor 33. Once the lubricant and the gaseous refrigerant are discharged into the pressure reservoir 90, however, the gas velocity is reduced and the lubricant particles settle into a refrigerant saturated pool of lubricant in the bottom of the pressure reservoir 90, as shown in FIGURES 2 and 8.

The turbulence barrier 91 reduces the turbulence inside the casing 10 and minimizes the percent of lubricant mist which is transported out of the pressure reservoir 90 through the discharge tube 93 with the gaseous refrigerant.

Since the gas pressure inside of the rotor **33** is less than the discharge pressure inside of the casing **11**, lubricant migrates up the lubricant pick-up tube **98** and passes through the wall **100** to lubricate the needle bearings **39**. Upon passing through the needle bearing assembly **39**, part of the lubricant enters the rear annular clearance between the rear end plate **37** and the adjoining side face of the rotor **33** and part of the lubricant travels through the splines **52, 56** and across the inner ends of the vanes **61** to lubricate the ballbearing assembly **46** and the rotary seal **48**. The remaining lubricant passes through the front annular clearance between the front bearing plate **13** and the adjoining side face of the rotor **33**. Lubricant in the two clearance annuluses serves the dual purpose of lubrication and retarding the rate of refrigerant flow from the discharge to the suction side of the compressor.

It is contemplated in accordance with the principles of the present invention to provide scavenging grooves on the suction side of the compressor one of which is most clearly shown in FIGURE 5 of the drawings at **116**. It will be noted that the scavenging groove has an end **117** spaced inwardly of the inlet recess **77** and the groove **116** extends circumferentially towards the discharge side of the pump terminating in an outwardly directed leg identified at **118**. Such a scavenging groove **116** is located in both the radial face of the front bearing plate **13** as well as the radial face of the rear bearing plate **37** and such grooves trap refrigerant and lubricant migrating towards the suction side of the compressor through the clearance spaces between the adjoining side faces of the bearing plates and the rotor. Fluids trapped by the scavenging grooves **116, 116** travel counterclockwise and are discharged through the discharge ends **118, 118** communicating with the working chamber **79** at a location such that the portions **118** are opened immediately after the suction kidney is closed, as shown in FIGURE 3. The pressure in the scavenging grooves **116**, therefore, fluctuates depending upon the position of the rotor **33** and the vane **61**. Although flow reversal in the grooves **116, 116** can occur under some conditions for a fraction of the cycle, the net effect is for counterclockwise flow. By preventing most of the gas and lubricant from leaking into the suction side of the compressor, the maximum charge of refrigerant can enter the suction side of the compressor and high volumetric efficiency and reduced discharge temperatures result.

It may be noted that the shoes **64** are designed to minimize wear and torque required to move them since the planing action of the shoes causes the shoes to ride up on the lubricant, thereby providing a tendency for the shoes **64** to float. During continuous operation, therefore, the shoes **64** will rarely touch the cylinder walls **32**.

Since the lubricant pick-up **98** is swivelly mounted, gravity causes the tube **98** to extend into the lubricant pool in the casing **11** regardless of the plane or orientation in which the compressor is mounted. Thus, any given compressor can be used in many different system applications and it is not necessary to have a large inventory of assembled compressors for different types of installations.

Once the compressor is put into operation, the lubricant pick-up tube **98** is rigidly held in place by the differential of pressure occurring between the pressure reservoir **90** and the interior of the pump since the flange **108** is pressure-biased against the wall **100**. In addition to holding the lubricant pick-up tube **98** in place, the flange **108** serves as a seal to prevent gaseous refrigerant from leaking into the interior of the pump around the flange and through the clearance between the outside of the hub **106** and the opening **101** in the wall **100**.

Although minor modifications might be suggested by those versed in the art, it should be understood that we wish to embody within the scope of the patent warranted hereon all such modifications as reasonably and properly come within the scope of our contribution to the art.

We claim as our invention:

1. A compressor comprising

means forming a pumping bore,

a rotary fluid displacement means eccentrically offset in said bore to form therewith a crescent-shaped working chamber and having circumferentially spaced pumping members mounted for rotation on a shaft and movable in following the bore contour to develop a pumping action in said chamber,

inlet means communicating with one side of said chamber,

and outlet means comprising a row of circumferentially spaced openings forming ports extending outwardly from said working chamber,

and plural valve means corresponding in number to said openings, said plural means being axially disposed with respect to said shaft and being exposed sequentially to working chamber outlet pressure for opening said ports in unison and closing said ports separately during rotary operation of the compressor.

2. A compressor comprising

inner and outer casings forming a pumping chamber in said inner casing and a pressure reservoir between said casing, said inner casing having a rear bearing plate at one end thereof,

inlet means communicating with said pumping chamber,

outlet means between said pumping chamber and said pressure reservoir, said outlet means comprising a row of circumferentially spaced openings forming ports extending outwardly from said working chamber,

and discharge means extending outwardly from said pressure reservoir for connection to a point of utilization,

a rotary fluid displacement means journalled for rotation on a shaft in said inner casing, plural valve means corresponding in number to said space openings, said plural valve means being axially disposed with respect to said shaft and being exposed sequentially to pumping chamber outlet pressure for opening said ports in unison and closing said ports separately during rotary operation,

and a tube swivelly mounted in the rear bearing plate on one end of said inner casing and depending into said pressure reservoir and forming a passage for conducting pressurized liquid lubricant from the lowermost portion of the reservoir into the inner casing for lubricating the fluid displacement means.

3. A compressor as defined in claim **2** and further characterized by

said outer casing having mounting pads disposed on plural sides for left, right and vertical mounting,

whereby said tube will always gravitationally swivel into depending relation towards the lowermost portion of the reservoir.

4. In a compressor as defined in claim **1**,

each of said ports being sized to be of smaller circumferential extent than the circumferential extent of the associated pumping member.

5. In a compressor as defined in claim **1**,

said plural valve means comprising a reed-type valve having a body portion and a plurality of digital projections each overlying a corresponding port.

6. In a compressor as defined in claim **5**,

said reed-type valve being made of spring metal and said digital projections comprising resilient spring fingers.

7. A compressor comprising

a casing cylinder having a bore formed therein, front and rear bearing plates on opposite sides of said casing cylinder,

a rotor shaft journalled in said bearing plates and ex-

tending through said bore on axis eccentrically offset relative to the bore axis,

a rotor on said shaft having a tangent seal contact with the bore wall and forming therewith a crescent-shaped working chamber,

splined coupling means between said shaft and said rotor affording self-centering movement of said rotor between said bearing plates,

said rotor having plural circumferentially spaced peripheral vane slots formed therein,

a vane in each said slot,

pins extending through said shaft between oppositely disposed vanes and having coil springs around said pins and engaging said vanes to pressure-bias said vanes outwardly against the adjoining bore wall,

said front bearing plate having an inlet valve and a discharge valve and being formed with an inlet kidney-shaped recess adjacent one side of said working chamber,

at least one of said bearing plates having formed therein a scavenging groove formed axially opposite said rotor and extending circumferentially on the inlet side of the compressor and spaced radially inwardly of said inlet recess,

said scavenging groove having a discharge end extending radially outwardly into the working chamber to minimize gas and lubricant leaking into the suction side of the compressor,

and outlet means for conducting pressurized fluid from the working chamber to a point of utilization.

8. A compressor as defined in claim **7** and further characterized by said outlet means comprising

a boss formed on said casing cylinder adjacent said tangent seal contact and forming a flat valve seating surface,

and said casing cylinder having a double row of parallel circumferentially spaced openings forming ports extending outwardly from the oulet side of said working chamber through said valve seating surface,

and a valve means on said boss comprising plural resilient reeds corresponding in number to said openings and engaging said valve seating surface to regulate said ports,

said valve reeds opening in unison and closing said ports in pairs.

9. A compressor as defined in claim **8** and further characterized by said boss having a recess spaced adjacent each row of openings to vent the underside of said reeds.

10. A compressor as defined in claim **8** and further characterized by

a casing surrounding said casing cylinder and forming a pressure reservoir for receiving the fluid pressurized in said pump,

said rear bearing plate having turbulence barrier portions extending outwardly to engage said casing.

11. A compressor as defined in claim **10** and further characterized by said outlet means including a tube extending from said front bearing plate to said pressure reservoir to receive pressurized fluid therefrom.

12. A compressor as defined in claim **11** and further characterized by

a tube carried by said rear bearing plate for swivel movement and depending gravitationally towards the bottom of the pressure reservoir for conducting liquid lubricant to said rotor shaft.

13. A compressor as defined in claim **12** and further characterized by said front bearing plate having plural mounting pads for effecting left, right and vertical mounting of said compressor,

said tube swivelly adjusting in response to the orientation of the compressor.

14. A compressor as defined in claim **13** and further characterized by said swivel tube having a flange overlying said rear bearing plate to form a pressure loaded seal with the adjoining surface during operation of the compressor.

15. A compressor as defined in claim **8** and a second boss formed on said casing cylinder providing a second series of valve controlled discharge ports extending towards the inlet side of the compressor for low pressure discharge operating conditions.

16. A compressor as defined in claim **8** and further characterized by the size of each said opening being smaller than the circumferential width of a vane to prevent by-passing across the ports.

17. A compressor as defined in claim **8** and further characterized by a shoe pivotally carried on the end of each said vane and each shoe having a leading end longer than the trailing end thereof to develop a planing action relative to the lubricant and the adjoining bore wall.

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