May 28, 1968 C. R. KILGORE 3,385,513

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 1

Fig. 1

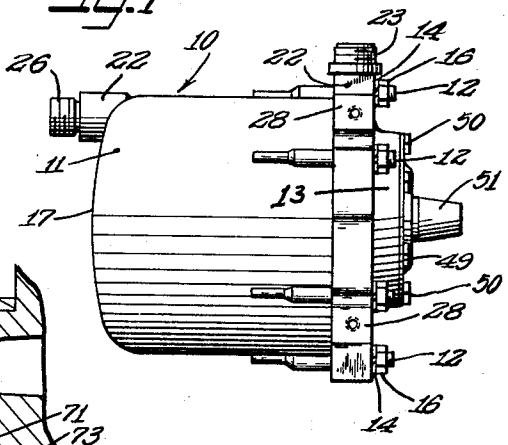


Fig. 6

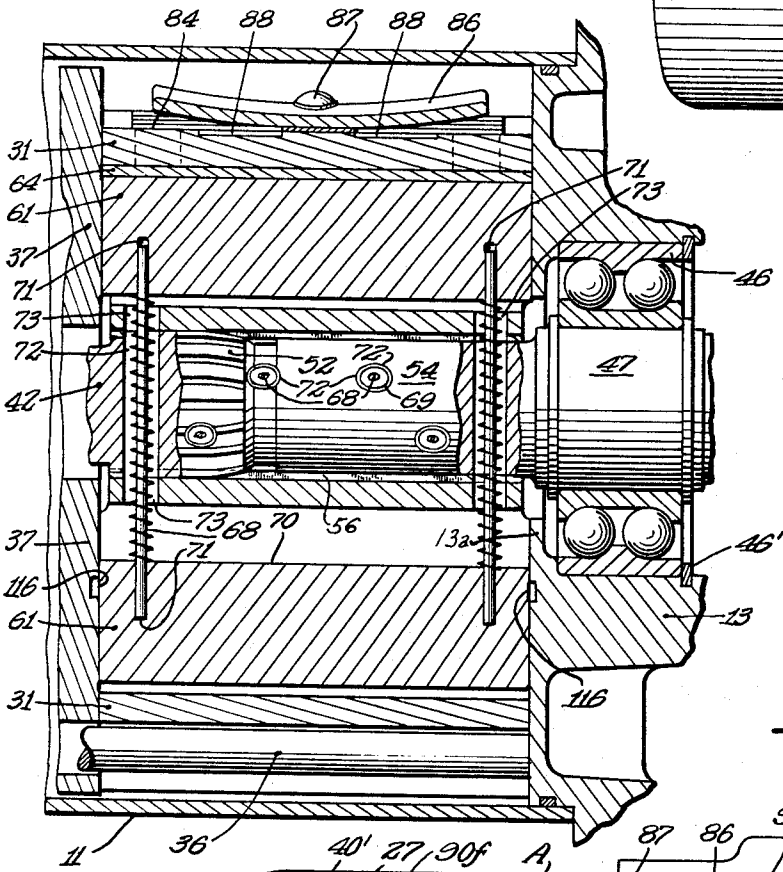
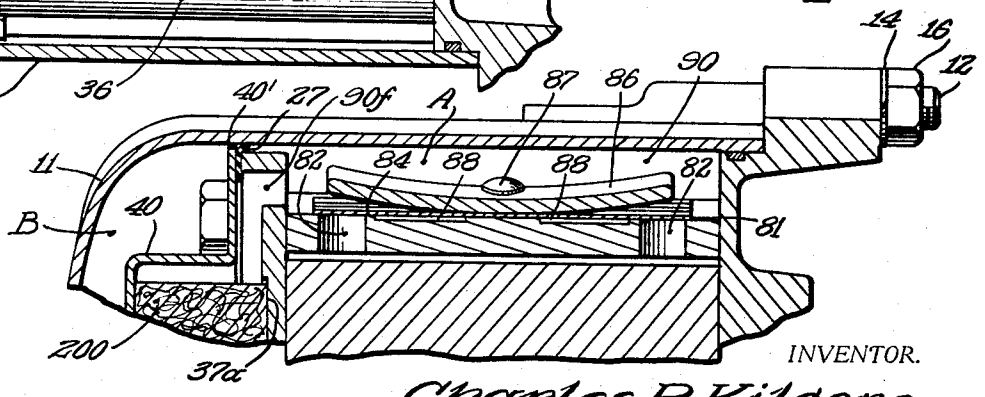


Fig. 7



INVENTOR.

Charles R. Kilgore

BY Hill, Sherman, Meroni, Gross & Simpson ATTORNEYS

May 28, 1968 C. R. KILGORE 3,385,513

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 2

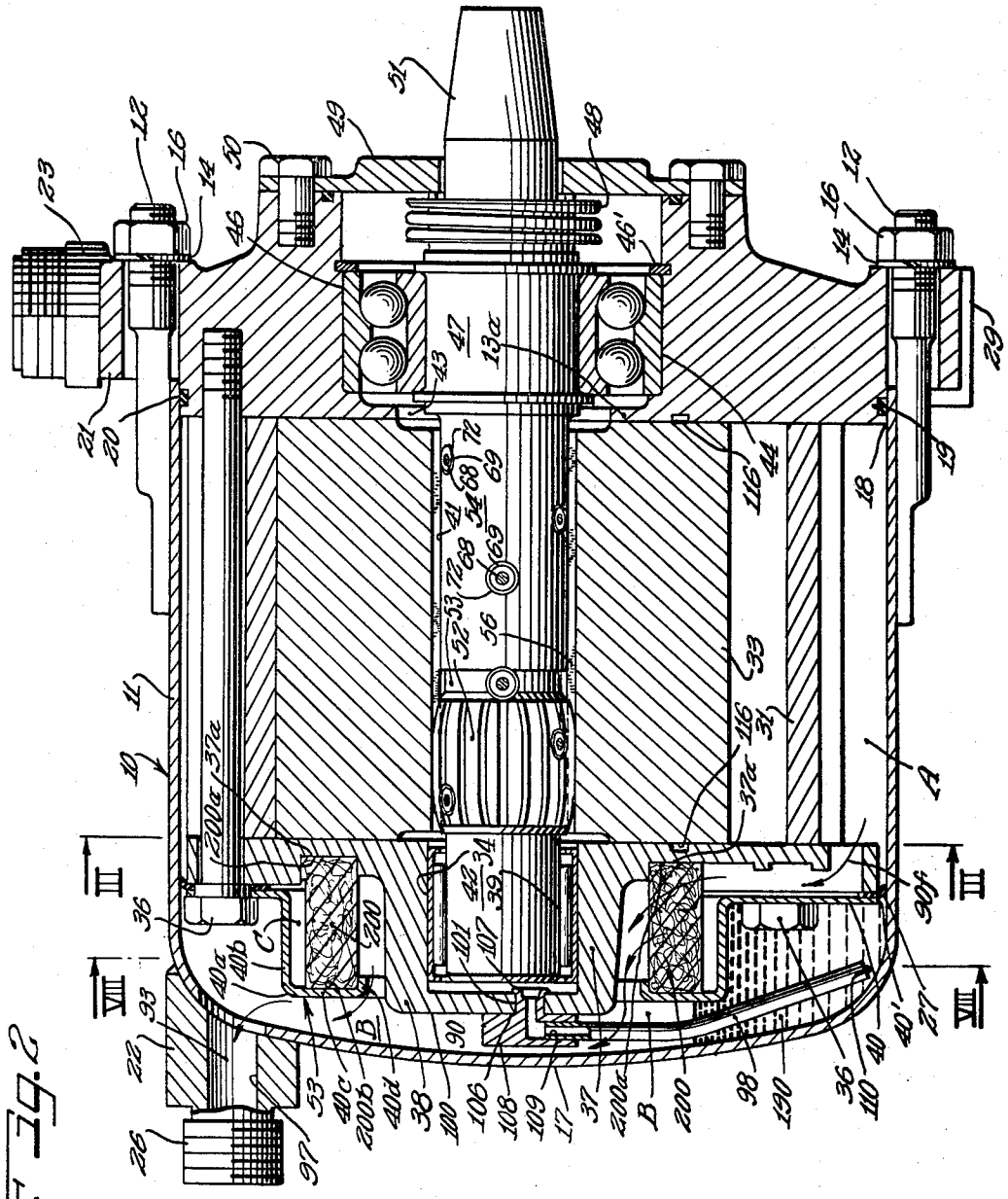


Fig. 2

INVENTOR.

Charles R. Kilgore

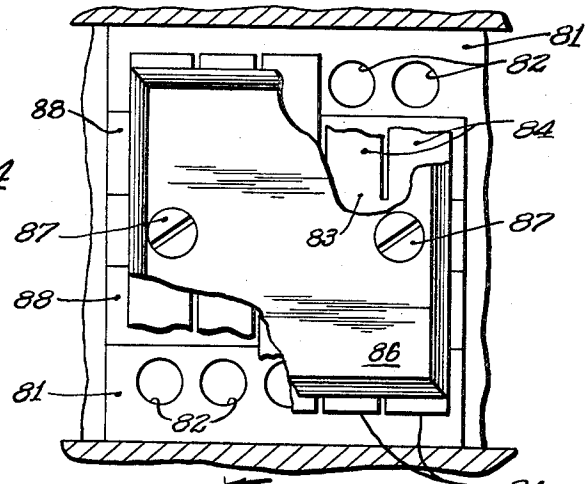
BY Hill, Sherman, Meroni, Gross & Simpson ATTORNEYS

May 28, 1968 C. R. KILGORE 3,385,513

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 3

Fig. 4



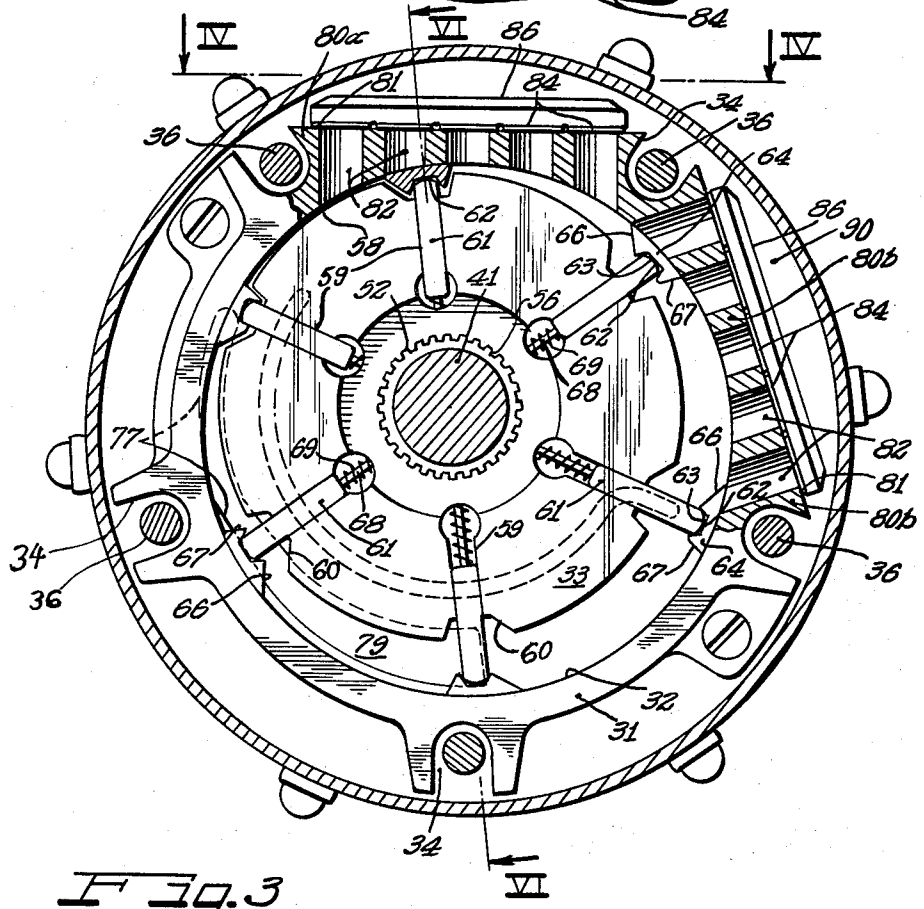


Fig. 3

INVENTOR.

Charles R. Kilgore

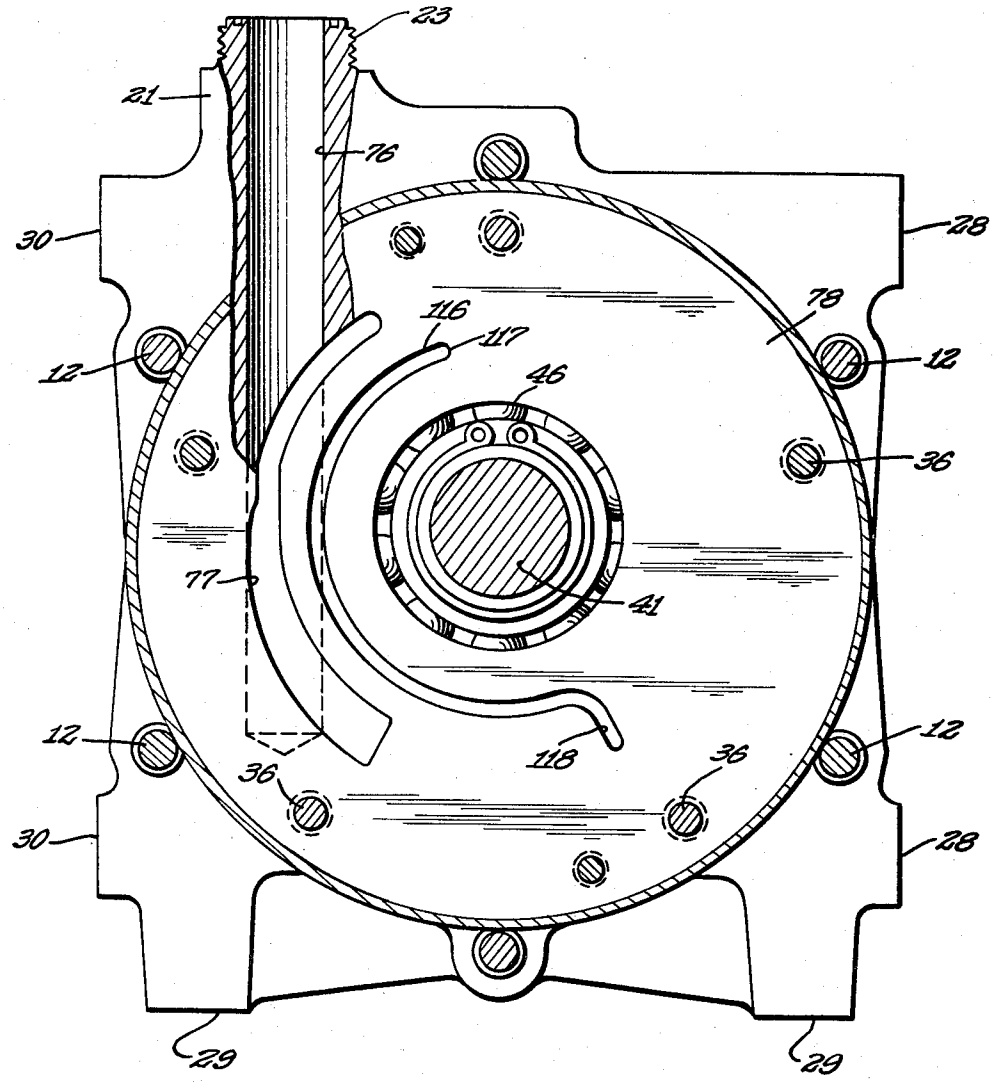
BY Hill, Sherman, Meroni, Gross & Simpson ATTORNEYS

May 28, 1968 C. R. KILGORE 3,385,513

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 4

Fig. 5



INVENTOR.
Charles R. Kilgore

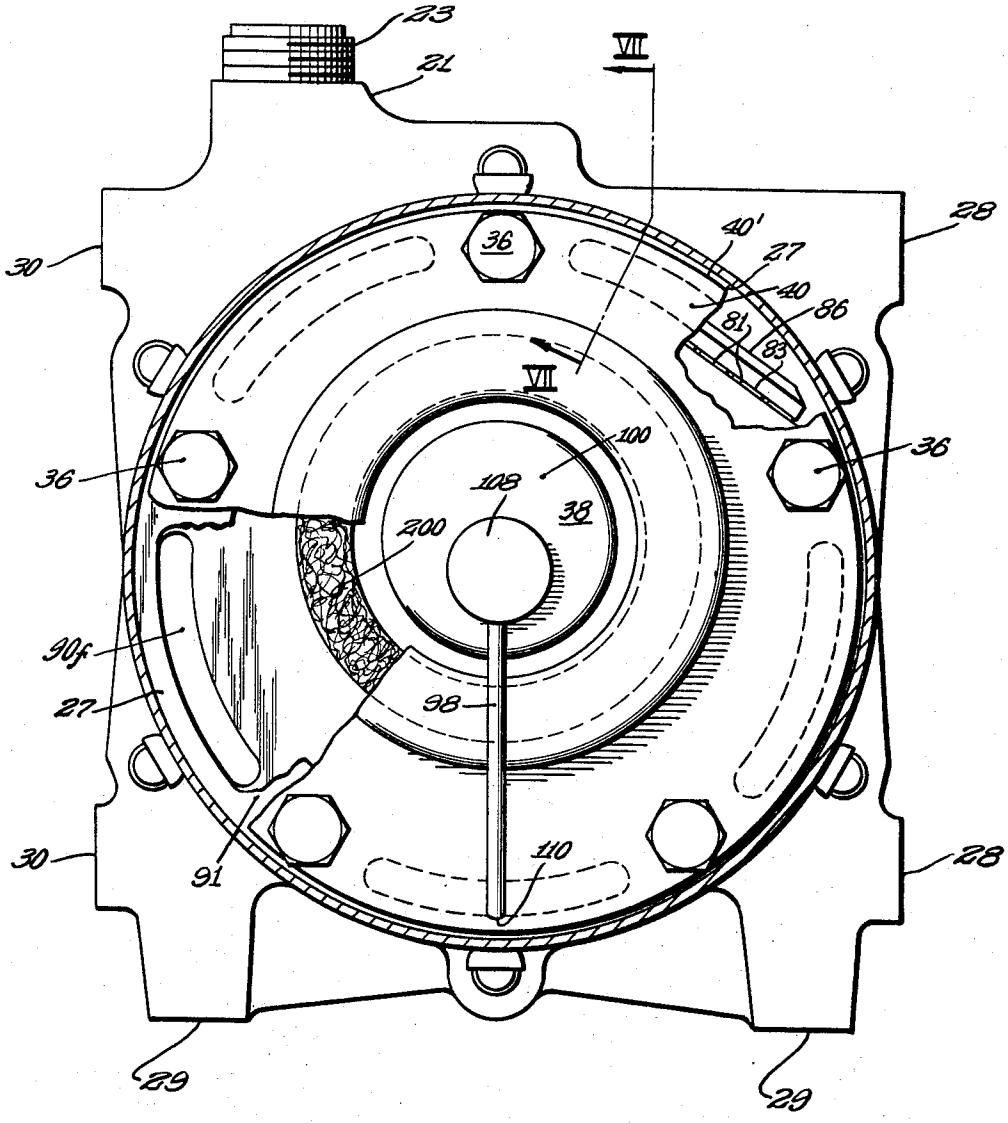
BY Dill, Sherman, Meroni, Gross & Simpson ATTORNEYS

May 28, 1968 C. R. KILGORE 3,385,513

REFRIGERANT VAPOR COMPRESSOR

Filed April 11, 1966 5 Sheets-Sheet 5

Fig. 8



INVENTOR.
Charles R. Kilgore

BY Hill, Sherman, Meroni, Gross & Simpson ATTORNEYS

United States Patent Office 3,385,513 Patented May 28, 1968

3,385,513

REFRIGERANT VAPOR COMPRESSOR

Charles R. Kilgore, Willowick, Ohio, assignor to TRW Inc., Cleveland, Ohio, a corporation of Ohio

Filed Apr. 11, 1966, Ser. No. 541,836
5 Claims. (Cl. 230—152)

This invention relates to an improvement in 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 minimize leakage into the suction side of the compressor. And, in accordance with the present invention a demister element is provided between the discharge ports and the reservoirs for materially aiding separation of the oil lubricant from the refrigerant.

The invention has the object and advantage of improving overall refrigeration system efficiency by retaining lubricant in the compressor.

On the drawings:

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

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

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

FIG. 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 FIG. 3;

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

FIG. 6 is a fragmentary cross-sectional view taken on line VI—VI of FIG. 3;

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

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

The rotary vane vapor compressor of the present invention is shown generally at **10** and comprises a 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** (FIG. 2).

The bearing plate **13** is characterized by the formation thereon of a boss **21**. The boss **21** receives suitable coupling means shown generally at **23** (FIG. 5) for connection thereto of an inlet service valve.

The casing **11** has a boss **22** which also has suitable coupling means **26** connected thereto by means of which a discharge service valve may be connected to the compressor.

As is most clearly shown in FIGS. 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** (FIG. 3) 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**. 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** from the outside of the front bearing plate **13** to receive a double roll ball bearing assembly shown generally at **46**, the inner race of which engages against a bearing portion **47** on the shaft **41**. In constructing, the assembly **46** is inserted from the front of the plate **13** and is held therein by restraining means **46'**. Installing the assembly **26** through the front or outside surface of the bearing plate **13** permits use of a radial inward lip **13***a* extension to the inner surface of the front bearing plate **13**. This increases the length of the leakage path across the annulus defined by the inside surface of the front bearing plate and the front inside surface of the rotor **33**. 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 section **43** and reduced diameter section **54**. The splined section **52** is co-rotatable and 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 **46'** is used to hold the ball bearing assembly **46** in place in the counterbore **44** of the front bearing plate **13**.

As shown in FIG. 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**. A 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 FIG. 6, the rotor is also formed with passages **73** for a similar purpose.

Referring now more particularly to FIGS. 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 entering the compressor 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 a first boss **80***a* adjacent the tangent contact area **58** and having formed thereon a flat valve-seating surface **81** (FIG. 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 FIG. 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 outer periphery **91** of the rear bearing plate **37** is continuous (FIG. 8), and serves to define a number of flow channels **90***f* of a curving slot configuration each of which is defined between adjacent circumferentially spaced boats **36** and lie about equal length and commonly centered arc segments.

As shown in FIGS. 2 and 8, a swivel mounted lubricant pick-up tube is provided at **98**. The tube **98** is positioned in FIGS. 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 **98** has a portion at **109** which extends within hub **106** and communicates therethrough to the bearing **39**. The tube **98** depends gravitationally towards the lowermost portion of the pressure reservoir **90** opening at **110**.

In operation, counterclockwise rotation of the rotor **33** and the attached vanes **61** and shoes **64**, as viewed in the orientation of FIG. 3, causes pressure in the suction side of the compressor to be reduced. This, in turn, causes gaseous refrigerant and minute quantities of lubricant in mist form to flow into the suction passage **76** and pass through the suction kidney **77** to fill the cavity between vanes **61, 61**. In FIG. 3, the vane at the bottom of the drawing is shown at the point where its leading edge has just cut off the working chamber **79** from the suction kidney **77**. Thus, compression is about to commence in the corresponding portion of the working chamber between the bottom vane and the lower right vane.

The relative position of the rotor **33** and any particular vane **61** at the time the discharge reed valves **84** first operate depends upon the discharge pressure. At very low discharge pressures, nearly all of the reed valves **84** operate. At maximum pressure only the reed valves **84** between the uppermost vane and the point of tangent contact **58** would operate. 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 cutoff, thus preventing liquid lock.

It is contemplated 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.

One advantage of the compressor is that 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.

Another advantage of the compressor 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 FIG. 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 annular 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.

As best shown in FIGS. 2 and 8, in accordance with the present invention, the compressor **10** is provided with

means designated generally by the numeral **53** for demisting the refrigerant-lubricant mixture that passes without the compressor **10**. These means for demisting **53** comprise a zone or compartment defining member **40** which is formed in a unitary structure generally affixed in a plane transverse to the axis of the rotor **33**. The member **40** divides the sections of the casing into a front chamber or zone A in communication with the outlets from the reed valves and a rear chamber or zone B in communication with the exit port passage **93**. The member **40** has an outer edge **40'** of conforming but slightly smaller size than the inside size of the casing **11** at its point of affixture. Thus, the outer edge **40'** is of circular configuration. About the outer edge **40'** is affixed a seal **27** of a flexible material such as rubber. The seal **27** extends to the interior surface of the housing **11** and there makes sealing contact. The member **40** has an outstanding annular extension **40*a*** of general L-shape in cross-section positioned eccentrically to conform with a bearing housing extension **38** of the rear bearing plates **37**. The extension **40*a*** comprises an outstanding circularly curved wall **40*b***, centered about the shaft portion **42**, and a planar portion **40*c*** extending from the end of wall **40*b*** and terminating in an inwardly curving lip **40*d*** of a circular shape. The lip **40*d*** is positioned about but spaced from the curved outer surface of the extension **38**. The bearing plate **37** has an annular shaped bore **37*a*** formed therein about the extension **38** and opposite the portion **40*c*** of the member **40**. The member **40** is affixed to the plate **37** about the periphery thereof by means of the bolts **36** which pass through conforming holes in the member **40**. Positioned between the front facing portion **40*c*** of the member **40** and the annular base **37*a*** of the plate **37** is a demisting element **200** of a generally annular shape. The demisting element **200** has a generally rectangular cross-section with a planar forward facing surface **200*a*** abutting against the interior surface of the base **37*a*** and a rearward facing surface **200*b*** abutting against the frontward facing surface of portion **40*c*** of the member **40**.

During compressor operation a small pressure differential exists between chambers A and B shown in FIG. 2. This pressure differential forces the mixture of superheated refrigerant and lubricant suspended in particle form in chamber A to flow through channels **90*f***, into chamber C surrounding the demister element **200**, through the demister element **200** and exit around the rear bearing plate hub **38**. The demister element **200** consists of a suitably porous material which has the property of causing fine particles of lubricant dispersed in refrigerant to collect and exit in droplet form. The lubricant droplets are sufficiently heavy to fall out of the refrigerant stream and drop into the lubricant reservoir **190**.

One main advantage of reducing the percent of lubricant circulated around the air conditioning system is that under certain operating conditions the lubricant tends to become trapped in the evaporator and condenser which reduces the heat transfer efficiency of those units. The demister apparatus **53** thus serves to improve overall refrigeration system efficiency because of the ability of the compressor to retain lubricant as well as to improve the efficiency of the compressor.

Since the gas pressure inside of the rotor **33** is less than the discharge pressure inside of the casing **11**, lubricant flows up the lubricant pick-up tube **98** and passes through the wall **109** 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 vanes and the ball bearing 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. Upon entering the pumping chamber, the shoes **64** are lubricated.

It is contemplated to provide scavenging grooves on the suction side of the compressor one of which is most clearly shown in FIG. 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 FIG. 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 pickup **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 pickup 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 pickup 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 I wish to embody within the scope of the patent warranted hereon all such modifications as reasonably and properly come within the scope of my contribution to the art.

I claim as my invention:

1. 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 journaled in said bearing plates and extending through said bore on an 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 a plurality of 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 leakage into the suction side of the compressor,

outlet means for conducting pressurized fluid from the working chamber to a point of utilization, 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 outlet side of said working chamber through said valve seating surface,

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, said boss having a recess spaced adjacent each row of openings to vent the underside of said reeds,

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

means for demisting comprising a demisting element for causing said lubricant to form in larger sized droplets than normal, said element being positioned in said casing in said reservoir and defining therewith a lubricant reservoir, and said means for demisting including a generally planar circular casing dividing member for affixing to said rear bearing plate and defining with said plate an annulus-receiving cavity, said member dividing said casing into front and rear chambers being in communication with said reed valves and said rear chamber being in communication with said compressor output means and forming said lubricant reservoir, said demisting element being annular in shape and postioned between said member and said rear bearing plate in said annulus-receiving cavity so that said fluid in flowing from said reed valves to said outlet passes through said demisting element,

a tube carried by said rear bearing plate for swivel movement and depending gravitationally towards the bottom of the lubricant reservoir for conducting liquid lubricant to said rotor shaft, said tube being swivelly adjustable in response to the orientation of the compressor, said front bearing plate having plural mounting pads for effecting left, right and vertical mounting of said compressor, 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,

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, 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, and

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.

2. A compressor comprising:

inner and outer casings forming a pumping chamber in said inner casing and a pressure reservoir between said casings,

inlet means communicating with said pumping chamber,

outlet means between said pumping chamber and said pressure reservoir,

means for demisting comprising a demisting element for causing said lubricant to form in larger sized droplets than normal, said element being positioned in said casing in said reservoir, said means for demisting being further characterized by the inclusion of an annular member having an outer periphery generally conforming to the interior shape of said casing for dividing said reservoir into a first and second zone, said demisting element being affixed in relation to said annular member so as to cause fluid flowing from said first zone to said second zone to pass therethrough,

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

a rotary fluid displacement means journaled for rotation in said inner casing.

3. The compressor for a refrigerant system comprising:

a casing cylinder having a bore formed therein,

front and rear bearing plates on opposite sides of said casing cylinder, said rear bearing plate having a rotor shaft receiving boss outstanding therefrom,

a rotor shaft journaled in said bearing plates and extending through said bore on an 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,

outlet means for conducting pressurized fluid from the working chamber,

a casing surrounding said casing cylinder and forming a pressure reservoir for receiving the fluid pressurized in said pump and having an exit port formed therein,

means for demisting comprising a compartment defining member which is formed in a unitary structure generally affixed in a plane transverse to the axis of the rotor and which divide the pressure reservoir into a front chamber and a rear chamber in which the front chamber is in communication with the outlet means from said working chamber and the rear chamber is in communication with said exit port, and the member furthermore has an edge conforming in configuration to the interior of said casing about the outer edge of which is affixed a seal constructed of a flexible material which extends to the interior surface of the casing and makes contact therewith, the member having an outstanding annular extension of generally L-shape in cross-section positioned to conform with the rear bearing plate shaft receiving boss, the extension comprising an outstanding circular curved wall centered about the shaft boss and a planar portion extending from the wall and terminating in an inwardly curving lip of circular shape which is positioned about but spaced from the outer surface of said boss; and

an annular shaped demisting element positioned in said annular extension of said member and extending between said rear bearing plate and said member so that fluid from said front chamber to said rear cham-

ber must pass therethrough, said element being generally porous and having the property of causing droplets of lubricant to form in larger droplets than otherwise.

4. The improvement as claimed in claim 3 in which said edge of said member is circular in shape, and said seal is of rubber.

5. The compressor as claimed in claim 2 further including a tube swivelly mounted in 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.

References Cited

UNITED STATES PATENTS

1,580,337	4/1926	Schutten	222—321
2,827,226	3/1958	McCormack	230—152
2,962,977	12/1960	Klessig et al.	103—220
3,015,222	1/1962	Wellborn et al.	230—152 X
3,208,667	9/1965	Boettcher	230—207
3,070,032	12/1962	Rhode	103—220
3,184,157	5/1965	Galin	230—152 X
3,258,198	6/1966	Harlin	230—207
3,270,955	9/1966	Huyser et al.	230—152

FRED C. MATTERN, JR., *Primary Examiner.*

W. J. KRAUSS, *Assistant Examiner.*