

Jan. 26, 1971

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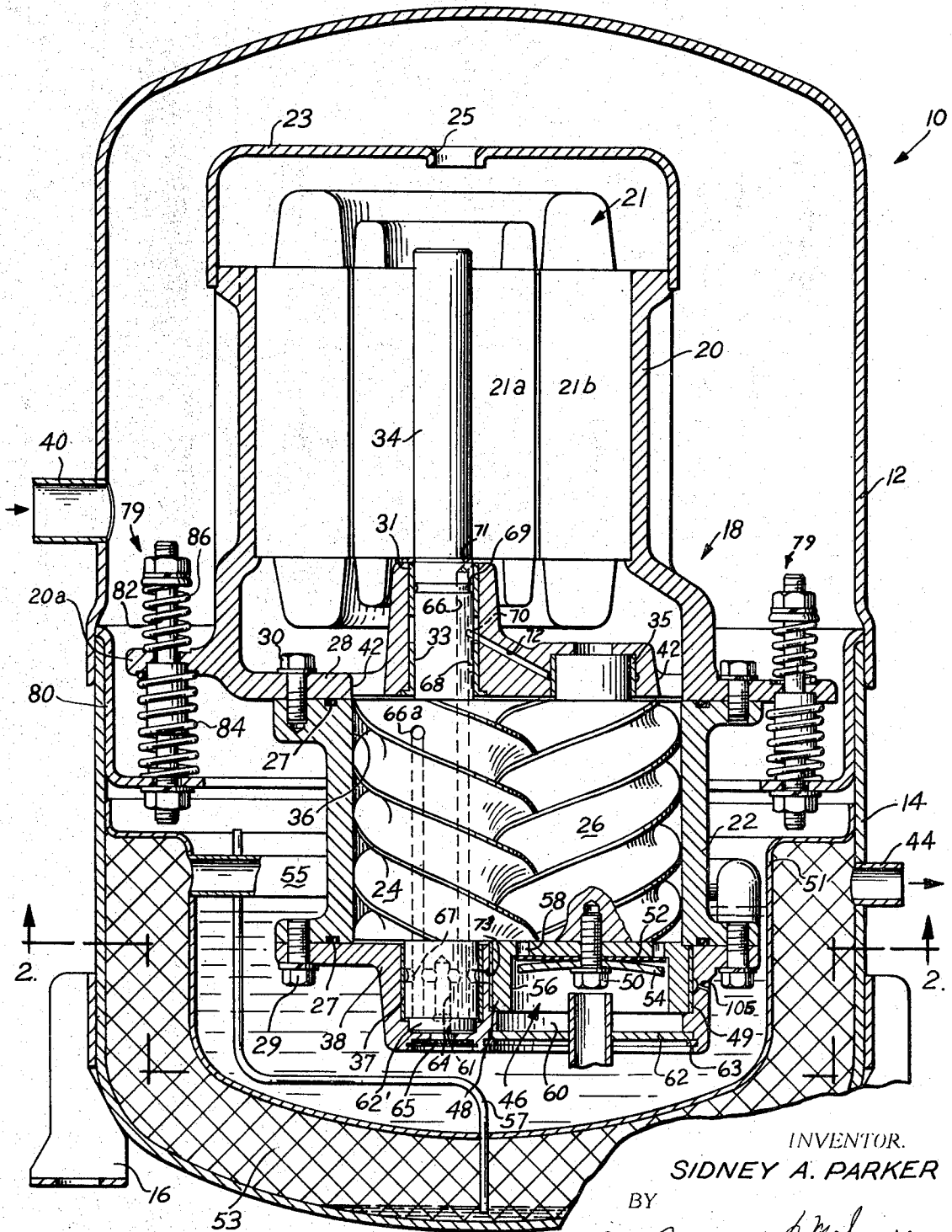
3,558,248

SCREW TYPE REFRIGERANT COMPRESSOR

Filed Jan. 10, 1969

5 Sheets-Sheet 1

Fig. 1



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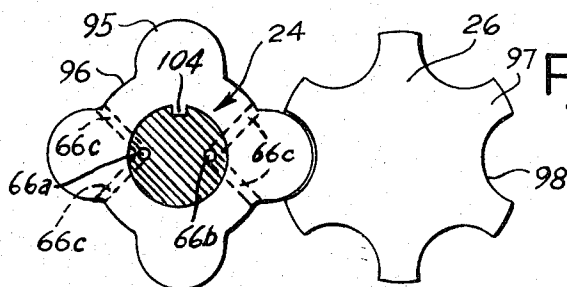
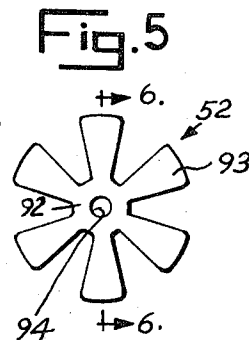
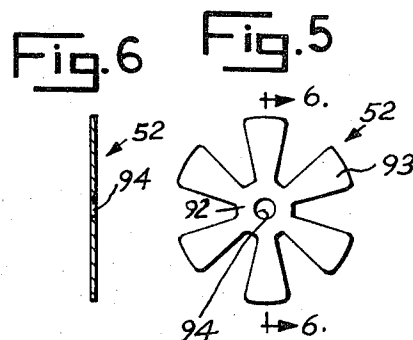
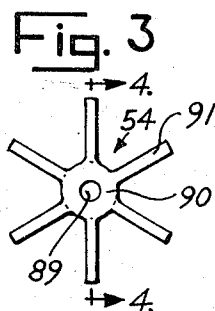
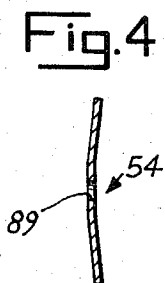
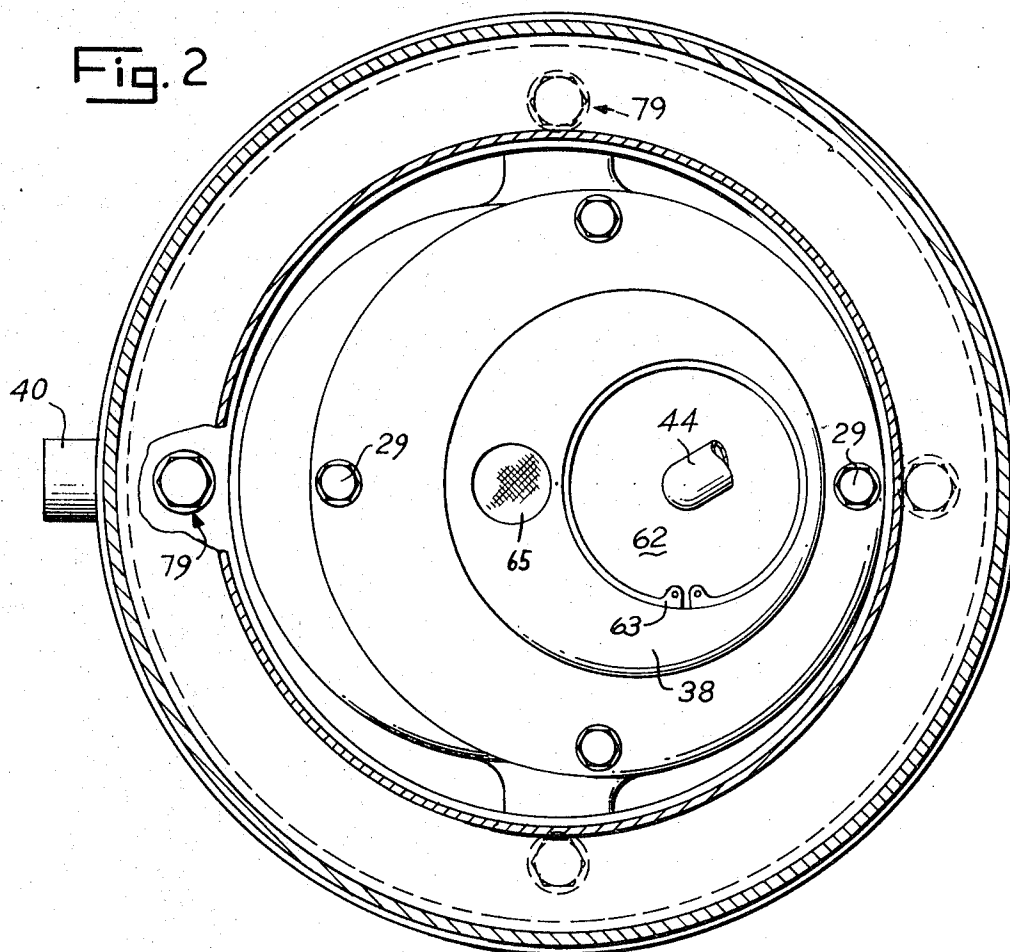
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SCREW TYPE REFRIGERANT COMPRESSOR

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5 Sheets-Sheet 2



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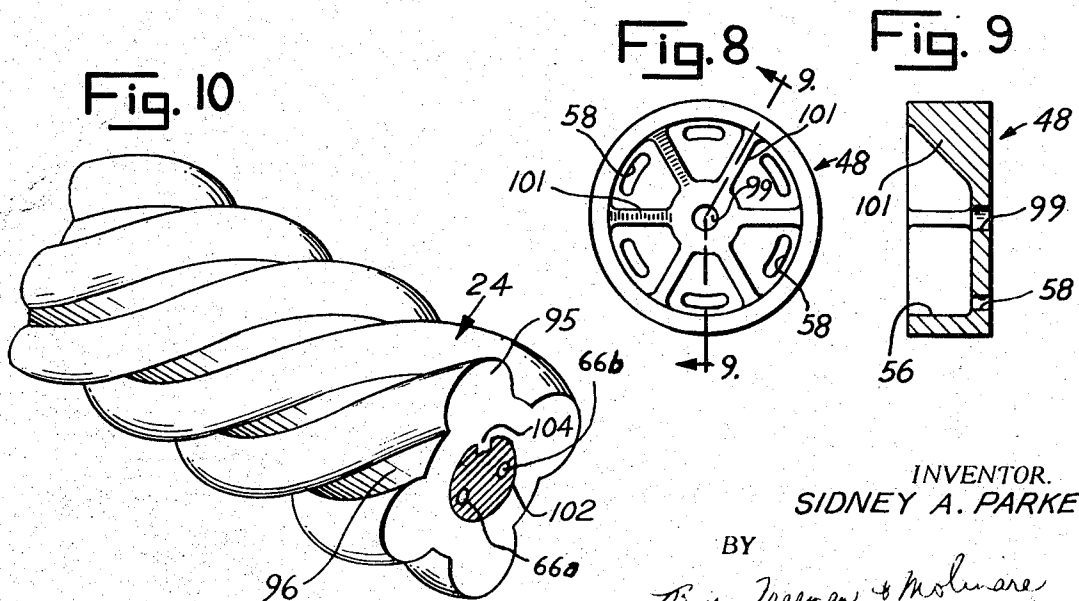
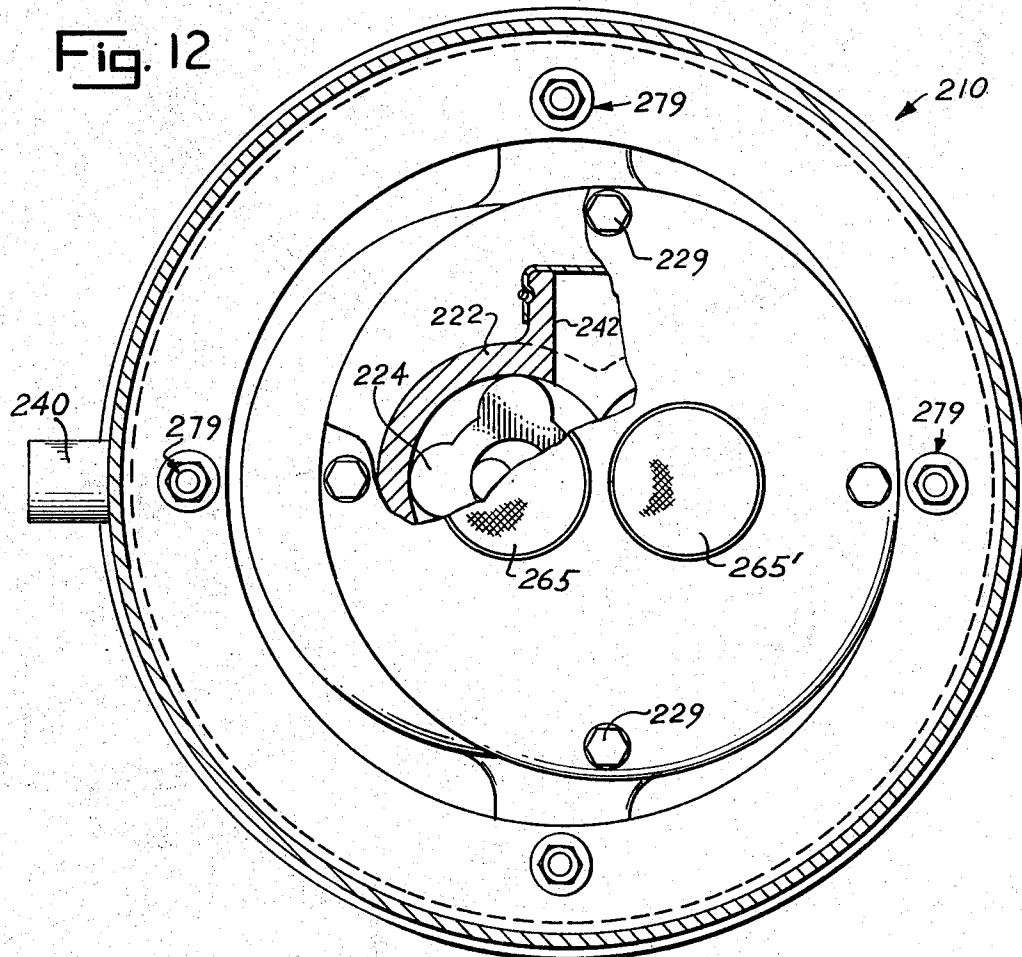
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SCREW TYPE REFRIGERANT COMPRESSOR

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5 Sheets-Sheet 3



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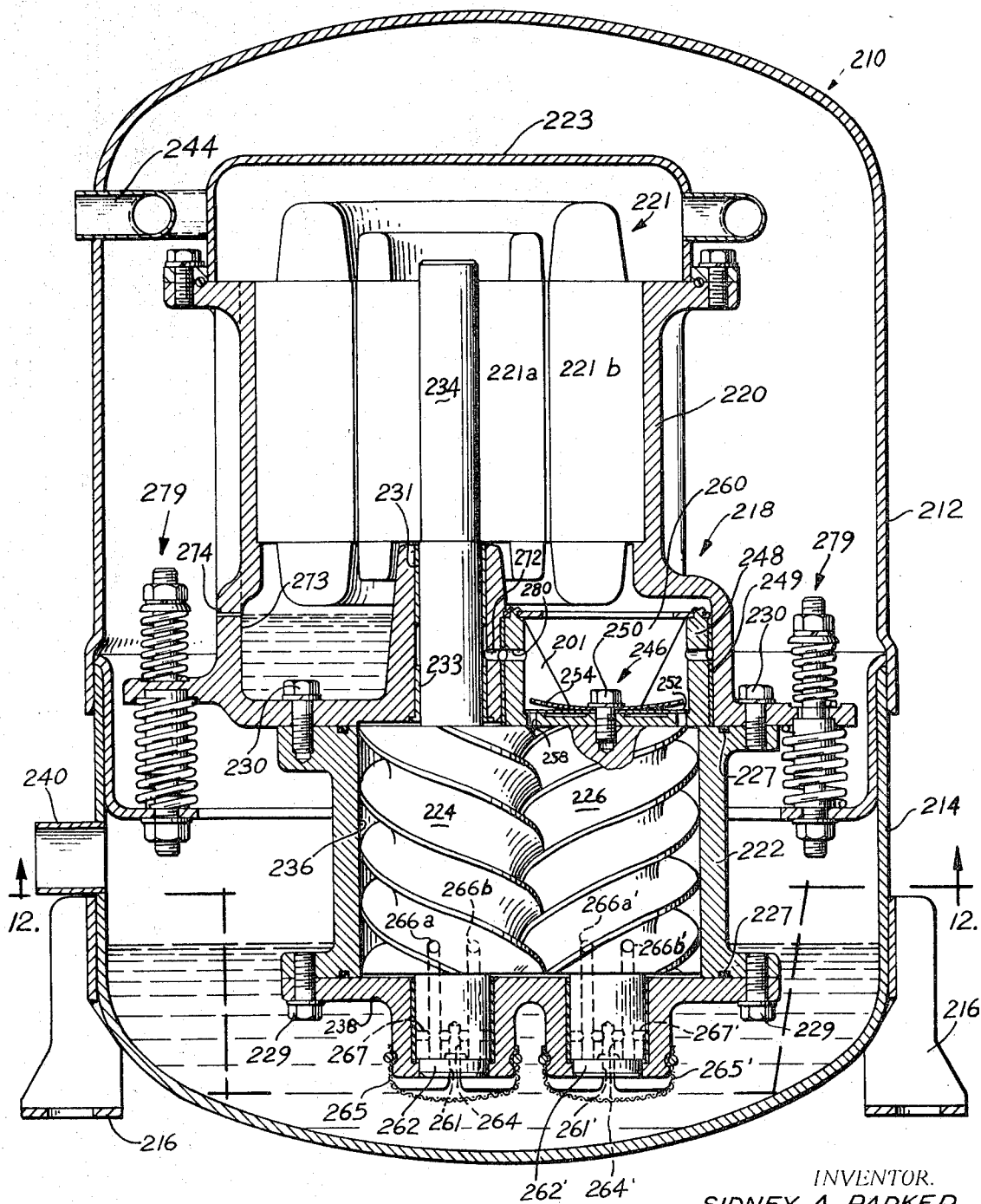
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SCREW TYPE REFRIGERANT COMPRESSOR

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Fig. 11



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SCREW TYPE REFRIGERANT COMPRESSOR

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Fig. 13

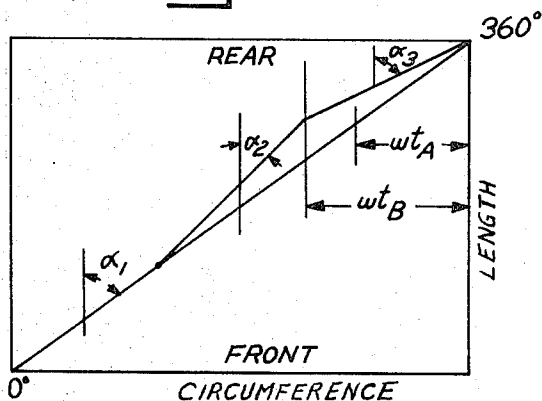


Fig. 14

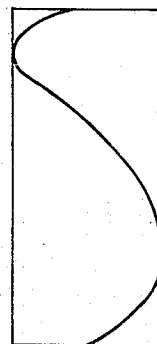


Fig. 15

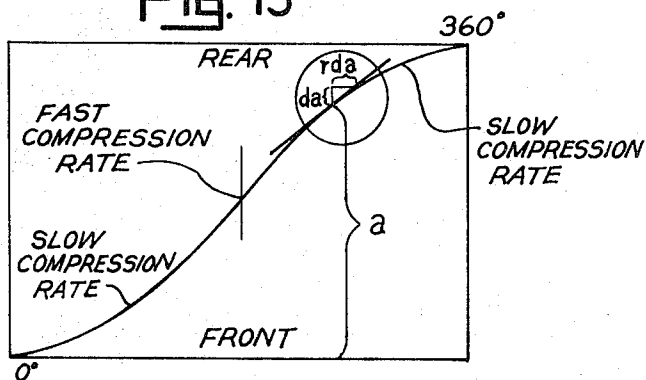


Fig. 16

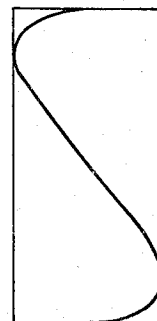


Fig. 17

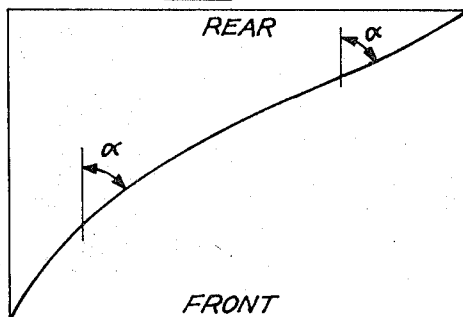


Fig. 18

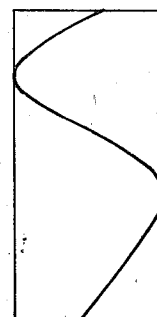
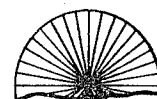


Fig. 19



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3,558,248

SCREW TYPE REFRIGERANT COMPRESSOR

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Filed Jan. 10, 1969, Ser. No. 790,212
Int. Cl. F04c 27/02, 29/02, 17/12
U.S. Cl. 418—88

11 Claims

ABSTRACT OF THE DISCLOSURE

A refrigerant screw compressor incorporating valve means constructed and arranged to rotate with the screws to provide a variable compression ratio responsive to conditions of the refrigeration system in which the compressor is used. The screws may be provided with a preselected variable skew for maintaining adequate fluid flow through the valve means with a minimum pressure drop. Lubricant is supplied to the working space within which the screws operate from a sump within the compressor casing by pump means within the screws for increasing screw compression performance.

BACKGROUND OF THE INVENTION

This invention relates to a screw compressor and, more particularly, to a screw compressor incorporating valve means rotatable with the screw means in the compressor for providing a variable compression ratio responsive to conditions of the refrigeration system in which the compressor is to be used. Another aspect of this invention is the provision of screws or rotors having a variable skew angle for maintaining adequate fluid flow through the valve means with minimum pressure drop. This invention also relates to a screw compressor of the type described employing pump means within the screws or rotors for supplying lubricant from a sump in the compressor to the working space within which the screws are disposed.

Screw compressors are finding increased use, but, are limited for certain applications because of the fact that such screw compressors have a fixed compression ratio. An advantage of a reciprocating compressor with pressure responsive valves is that it has a variable compression ratio and is responsive to conditions in the refrigeration system in which it is used.

One attempt to vary the built-in compression ratio of a screw compressor has been to vary the shape of the outlet port from the working chamber in the compressor housing, within which housing the screws are disposed. Such adjustments require clumsy and expensive external components as shown, for example, in Whitfield Pat. No. 2,287,716. In addition, in Whitfield, the amount of variance of compression ratio is limited.

An object of this invention is to provide an improved screw compressor having valve means cooperating with the interengaging screws for providing a wide range variable compression ratio responsive to conditions of the refrigeration system.

Another object of this invention is to provide an improved screw compressor having valve means rotatable with the screws for providing a variable compression ratio (as low as one to one) responsive to continuously changing conditions of the refrigeration system.

Still another object of the present invention is to provide an improved screw-type compressor having valve means rotatable with the cooperating screw means in the compressor for giving a screw compressor the variable compression ratio characteristic of reciprocating compressor with pressure responsive valves.

Further, it has been found desirable to control the rate of fluid flow through the valve means in a compressor embodying the present invention. In a reciprocating com-

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pressor, the piston is at O-velocity at the bottom of the stroke and again at the top of the stroke and varies during the discharge and suction strokes so that, in effect, a reciprocating compressor has a variable compression rate. This is desirable to provide adequate fluid flow through the valve means with a minimum pressure drop. The screws in a screw compressor conventionally have a fixed or constant skew angle, with a resultant constant compression rate.

Thus, another object of the present invention is to provide a rotary screw compressor having rotatable valve means with rotors having a preselected variable skew for providing a preselected variable compression rate during operation of the compressor.

Still another object of this invention is to provide an improved rotary screw compressor incorporating rotors having a variable skew characterized for desired compression and adapted to discharge compressed gas at a lower rate and at a lower exit velocity than a comparable compressor incorporating uniformly skewed rotors, whereby the pressure losses as the compressed gas leaves the valve means is reduced.

In a known screw compressor, as, for example, as disclosed in Bailey et al. Pat. No. 3,073,513, liquid is supplied to the working chamber from a source external of the compressor to seal the clearances between the screws and between the screws and the housing. Separate pump and pressurized tank means external of the compressor are employed.

Therefore, yet another object of this invention is to provide an improved refrigerant screw compressor having simplified means for supplying liquid to the working chamber disposed within the compressor for sealing the clearances between the screws and between the screws and the housing, the source for the liquid and the supplying means being within the compressor. A further object is to provide an improved screw compressor incorporating pump means in the screws for providing liquid for sealing within the working chamber. Other objects and advantages of the present invention will be made more apparent hereinafter.

BRIEF DESCRIPTION OF THE DRAWING

There is illustrated in the attached drawing presently preferred embodiments of the present invention.

FIG. 1 is a cross-sectional view of a screw compressor embodying the present invention;

FIG. 2 is a cross-sectional view of the compressor of FIG. 1 taken generally along the line 2—2;

FIG. 3 is a plan view of the retaining member for the valve means of the present invention;

FIG. 4 is a cross-sectional view of the retaining member taken generally along the line 4—4 of FIG. 3;

FIG. 5 is a plan view of the valve plate of the valve means for the compressor shown in FIG. 1;

FIG. 6 is a cross-sectional view of the valve plate taken generally along the line 6—6 of FIG. 5;

FIG. 7 is a cross-sectional view of a pair of cooperating rotors;

FIG. 8 is a bottom view of the valve housing for the valve means of the compressor shown in FIG. 1;

FIG. 9 is a cross-sectional view of the valve housing taken generally along the line 9—9 of FIG. 8;

FIG. 10 is a perspective view of the male rotor of FIG. 7;

FIG. 11 is a cross-sectional view of a modified compressor embodying the present invention;

FIG. 12 is a cross-sectional view of the compressor of FIG. 11 taken generally along the line 12—12 of FIG. 11;

FIG. 13 is a plot of circumference versus length for a variable skew rotor;

FIG. 14 is a skew layout for the rotor of FIG. 13;

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FIG. 15 is a plot of circumference versus length for a modified variable skew rotor;

FIG. 16 is a skew layout for the modified rotor of FIG. 15;

FIG. 17 is a plot of circumference versus length for a further modified variable skew rotor;

FIG. 18 is a skew layout for the rotor of FIG. 17; and

FIG. 19 is a partial end view of a rotor indicating the basis for the layouts of FIGS. 14, 16 and 18.

DESCRIPTION OF THE PREFERRED EMBODIMENT OF THE PRESENT INVENTION

Referring now to FIG. 1, there is illustrated a rotary screw compressor 10 embodying principles of the present invention. The compressor 10 comprises a sealed outer housing or casing having an upper casing portion 12 and a lower casing portion 14 joined thereto in sealed relationship, as, for example, by welding. Support members or legs 16 may be secured to the lower casing portion for retaining the compressor in an upright position where desired for use.

Resiliently supported within the outer housing in spaced relationship thereto is the compression mechanism 18. The compression mechanism includes a compressor block or body having an upper portion 20 enclosing an electric drive motor 21 and a lower portion 22 receiving a pair of cooperating screws or rotors 24 and 26. The upper housing portion 20 includes an end cap 23 secured to portion 20 in a suitable manner and a bottom wall 28 adapted to abut lower portion 22 and secured thereto by machine screws 30. Disposed in the bottom wall 28 are upper bearings 31 and 33 for receiving the drive shaft 34 to the rotor 21a of motor 21 is connected and upper bearing 35 for receiving an end of rotor 26. A lower bearing for journaling the rotor 24 is provided in the lower end cap 38 which is suitably secured to and closes the bottom housing portion 22 of the compression mechanism. For example, machine screws 29 may join the lower end cap 38 to portion 22. The journaling of the lower end of rotor 26 will be discussed more fully hereinafter.

The lower housing portion 22 is sealed with respect to the lower end cap 38 and the upper housing portion 20 by means of O-ring seals 27.

The cooperating male rotor 24 and female rotor 26 are located in the working chamber or space 36 defined in the lower housing portion 22. The axes of the rotors 24 and 26 are parallel to one another and in a common plane. The rotor 24 is adapted to be secured to the shaft 34, which may be integral with, or separately connected to the drive shaft of the electric motor 21. The male rotor 24 functions as a drive rotor for operatively driving the female rotor 26.

Suction gas enters the space between the compression mechanism 18 and the outer housing via a suction inlet fitting 40 adapted to be connected to a suction line in a refrigeration system. The suction gas passes through the opening means 25 in the end cap 23 and over the electric motor 21 for cooling same. Then, the suction gas flows through the openings or ports 42 in the end or bottom wall 28 of the upper housing portion 20 into the working space 36. The screws or rotors 24 and 26, each of which is provided with lands and grooves, cooperate with the housing portion 22 to define diminishing chambers extending from the inlet end of the working space 36 to the discharge end thereof. Since the successive volumes diminish in size, or stated another way, since the volume of each chamber is decreasing from the inlet and to the discharge end of working space 36, there is a compression of the gas as it is forced by the rotors 24 and 26 from the inlet end of the working space 36 at the top thereof to the discharge end at the bottom thereof. The compressed gas passes from the discharge end of the working space 36 into the discharge line

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44 from which it is conducted to a discharge line in a refrigeration system via baffle means 53 as will be described more fully hereinafter.

A feature of this invention is the provision of rotatable valve means 46 at the discharge end of the working space for providing a variable compression, as low as one to one, which is responsive to conditions of the refrigeration system in which the compressor is to be used. Heretofore, refrigerant screw compressors have commonly had a constant compression ratio built into the design of the device. Attempts have been made to vary the compression ratio of these rotary screw compressors. One manner of varying the ratio has been to vary the geometry of the discharge port means from the working space. A prior arrangement as shown, for sample in Whitfield, 2,287,716, is bulky, is limited in range and is incapable of providing a compressor characteristic similar to a reciprocating piston type device. Utilization of the valve means 46 of the present invention in a rotary screw compressor produces a rotary screw compressor having the variable ratio characteristics of a reciprocating compressor.

The valve means 46 comprises a valve block 48 secured to the lower end of rotor 26 for rotation with the rotor and journaled in bearing 49. The means for fastening valve block 48 to rotor 26 may comprise a machine screw 50 as shown. The machine screw 50 also serves to retain the discharge valve 52 and the discharge valve stop plate 54 in place within the recess 56 defined in the valve block 48. The valve plate 52 will cooperate with the openings 58 defined in the valve block 48 for controlling the egress of compressed gases from the working space 36. Gases passing through the openings 58 flow into a chamber or space 60 defined within the recessed portion of the valve block 48, a portion of the lower bearing head 38 and the closure plate 62 which is secured to the discharge line 44. Thus, gases entering the space 60 will pass into the discharge line 44 and through the discharge line from the compressor 10 to a refrigeration system.

Another feature of this invention is the provision of pump means in the drive shaft 34 for lubricating the bearing surfaces of drive shaft 34, and, also, for lubricating the upper bearing surface of the rotor 26 and the bearing surface of valve block 48 within bearing 49. The pump means may comprise an inlet opening 61 in the thrust plate 62' at the bottom of the drive shaft 34, which communicates with an inlet opening 64 in the lower end of the drive shaft 34. Openings 61 and 64 are aligned with one another and with an opening in the lower end cap 38 that communicates with the sump defined between the bottom portion of the compression mechanism 18 and the separator housing 51. The inlet opening 64 in the drive shaft communicates with an upright passage 66 that is offset from the rotational axis of the drive shaft. Transverse bores or passages 67, 68 and 69 are provided in the drive shaft 34 for communicating the main eccentrically disposed pump passage 66 with the bearings 37, 33 and 31, respectively. Vent passage 71 communicates the top of passage 66 with the motor compartment for venting refrigerant from the pump passage 66 at startup to assure bearing lubrication. A transverse opening 70 is provided in the drive shaft 34 which operatively connects with a passage 72 in the bottom wall 28 of the upper housing portion 20 of the compressor block for communicating lubricant to the bearing surface between bearing 35 and the upper stub end of rotor 26. Lubricant is supplied to the bearing surface between valve block 48 and bearing 49 via opening 73 in the lower end cap 38 that connects with transverse passage or bore 67.

Also provided in drive shaft 34 are longitudinally extending pump passage means 66a and 66b which communicate at the lower end with the transverse passage or passages 67 of the pump means and at the other

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end with the recesses or grooves in the rotor or screw 24. For example, in a rotor 24 having four grooves, there would be two passages 66a and 66b disposed 180° apart, with each supplying lubricant to two of the grooves in the rotor. Alternatively, there could be four separate pumps, each supplying lubricant to a groove. The purpose of injecting the lubricant into the grooves is to seal the clearance between the two rotors 24 and 26 and between the rotors and the housing in order to increase the efficiency of compression.

A relatively simple arrangement for separating the sealing lubricant from the compressed gas discharged from the compressor is shown in FIG. 1. Such separating arrangement includes a separator housing 51 which is secured fixedly within the outer casing and baffle means 53 between the separator housing 51 and lower shell 14. In use, compressed gas discharged from the discharge gas manifold passes through line 55 to the chamber between housing 51 and lower shell 14. The sealing medium, for example, oil, will adhere to the baffle means 53, which may be fabricated from mesh or loosely packed fibers made from metal or plastic and be separated from the compressed gas, which is then discharged from the compressor via discharge line 44.

It is desired that lubricant removed from the compressed gas be collected in the bottom of the lower shell 14 and then returned to the oil sump defined between the compression mechanism and separator housing 51. This is accomplished by means of a metering tube 57 fixed to the housing 51 and having a lower end extending closely adjacent the bottom of the reservoir defined in the lower shell 14 and with the upper end above the normal level of lubricant in the sump in separate housing 51. Preferably, the metering tube 57 will include therein a Dole-type valve comprised of a resilient restriction member that functions to maintain a constant flow volume under varying pressure differentials.

The means for resiliently mounting the compressor block within the outer housing or casing includes an annular mounting ring 80 suitably connected to the lower shell or lower casing portion 14 of the outer housing, pin or post 82 extending upwardly from the mounting ring 80 and through an opening in the flange 20a on the body portion and springs 84 and 86 carried about the pin 82. Spring 84 is disposed concentrically about the pin 82 between flange 20a and the flange 80 for biasing the compression block upwardly. The spring 86 which is also carried on the pin 82 is adapted to bias the compressor block in an opposite direction, so as to provide for resilient spring support of the compression mechanism 18 both upwardly and downwardly.

Turning to FIG. 2, there is shown a cross-sectional view of the bottom of the compressor 10 taken generally along the line 2—2 of FIG. 1. It is readily seen that there are four spring means 79 provided equidistantly spaced from one another within the compressor 10 to support the compressor block. Other numbers of springs may be utilized depending upon the relative weight of the compressor block within the outer housing. It is seen that the plate 62 for closing the bottom of the chamber 60 is retained in place by means of a retaining ring 63. Discharge line 44 is suitably secured to the plate to provide for sealing engagement therewith and extends from the plate 62. The lower end cap 38 is retained in position on the lower portion 22 of the compressor block by means of the machine screws 29.

Lubricant enters the pump means in the rotor 24 through an opening in the bottom of the lower end cap 38 that is covered by screen 65 which functions to prevent particles from entering the pump means.

In FIGS. 3 and 4, there is illustrated the retaining ring for the discharge valve. The retaining ring or stop member 54 takes the form of a spider having a hub portion 90 with a plurality of fingers 91 extending therefrom. The hole 89 formed centrally thereof is for reception of the

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machine screw 50 which is adapted to secure the stop member 54 in place in the valve means. As best indicated in FIG. 4, the stop member 54 is bowed or dished in configuration. When assembled, the retaining ring 54 functions to limit downward flexing of the discharge flexure valve plate 52 from the valve block openings 58.

In FIGS. 5 and 6, there is illustrated the discharge valve plate 52 of the valve means 46. The discharge valve plate 52 includes a hub portion 92 having a plurality of outwardly extending lobes or valves 93, which cooperate with the openings 58 in the valve block 48 for restricting fluid flow therethrough. As indicated from the cross-section of the discharge valve plate shown in FIG. 6, the valve plate is comprised of a substantially flat planar member which has a central opening 94 therethrough. The inherent resiliency of the material of the valve plate 52, for example, steel, will tend to maintain the valve plate in the closed position as indicated in FIG. 1. Furthermore, the centrifugal force will straighten the valve plate 52 during rotation and tend to urge it closed, thus obviating the need for a back-up spring.

There is shown in FIG. 7 a transverse cross-sectional view illustrating the cooperation between the two rotors 24 and 26. The male rotor 24 is provided with helically extending lands 95 and recesses or grooves 96 therebetween. The female rotor 26 is provided with lands 97 and recesses 98 therebetween. The intermeshing helical lands and grooves of the rotors 24 and 26 are provided with wrap angles up to 360°. The lands and grooves of the male rotor 24 lie substantially entirely without the pitch circle of the rotor 24 and the lands and grooves of the rotor 26 lie substantially entirely within the pitch circle of the rotor 26.

In prior known screw compressor constructions, the wrap angle on the screws was on the order of 180° in order to accommodate proper orientation of the inlet and outlet from the working space. Through utilization of the valve means of the present invention, discharge of compressed gases from each of the diminishing chambers between the cooperating screws and housing is not limited to a particular location of discharge outlet, but can occur at any time during rotation of the screws. Another advantage of the present arrangement is that there is considerable design flexibility, since capacity can be increased or decreased for a compressor employing a particular diameter and flute configuration of screws by merely increasing or decreasing the length of the screws.

In FIGS. 8 and 9, there is shown the valve block 48 of the present invention. The valve block 48 includes a plurality of discharge ports or holes 58 extending there-through for receiving compressed gases from the working space 36. As shown, there are six holes 58 which cooperate with the six lobe portions 93 of the valve 52. Centrally of the valve block 48, there is provided an opening 99 for reception of the bolt 50. A plurality of radially extending ribs 101 are provided in the valve blocks 48. The ribs cooperate with the valve members 93 to make sure that the gas and liquid (oil) emitted from the openings 58 is moving at the same radial speed as the valve lobes 93. Also, the ribs 101 act as a centrifuge for separating the oil from the gas.

The construction of the male rotor 24 is somewhat better seen in FIG. 10, wherein, note, that an opening 102 is provided within the rotor 24 for connection to the shaft 34. As seen in FIG. 10, a key 104 is provided in the rotor 24 for keying same to the drive shaft 34.

In order to improve the sealing of the small clearance between the cooperating rotors and between each rotor and the lower housing portion 22 of the compressor block, a liquid sealing medium, for example, oil, is injected into the working space through the injection passages 66a, 66b, 66c. Besides serving as a sealing medium in the clearance space between the rotors 24 and 26 and between the rotors and the cooperating wall of lower housing portion 22, the oil injected into the working space 36 serves as a

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medium for direct cooling of the gas, and, also, is a lubricant between the rotors 24 and 26. Some of the oil carried from the working space 36 into the space 60 is centrifuged and separated from the gas by means of the webs 101 on the rotating valve block 48. Such oil will find its way into the passage 105 in the lower end cap 38 and be returned to the sump defined between the separator housing 51 and the compression mechanism 18. Compressed gas containing oil is passed through baffle means 53 before discharge from the compressor to effect further separation of the gas from the oil. Oil carried over into the refrigeration system from the compressor can reduce the efficiency of the heat transfer of the heat exchange members in the refrigeration system.

Turning now to FIGS. 11 and 12, there is illustrated a modified hermetic screw compressor 210 incorporating features of the present invention. The basic design difference between the compressor illustrated in FIGS. 1 and 2 and that shown in FIGS. 11 and 12 is that in the design of FIGS. 1 and 2, refrigerant to be compressed is introduced to the top of the working space 36 and is discharged from the bottom of the working spaces whereas in the design of FIGS. 11 and 12, refrigerant is introduced to the working space 236 adjacent the bottom of the rotors 224 and 226 and is discharged from the working space 236 at the upper end of the rotors.

The compression mechanism 218 is resiliently supported within the outer housing comprised of upper shell 212 and lower shell 214 by the spring means 279. Spring means 279 are essentially the same as spring means 79 described previously. The compression mechanism 218 includes a housing comprised of upper portion 220 and lower portion 222 affixed to the upper portion by means of bolts 230. The portions 220 and 222 are suitably sealed one to the other, for example, by an O-ring seal. The top of the compressor block is closed by an end cap 223 suitably affixed to the top of the upper portion 220. The lower part of the compressor block is comprised of a lower end cap 238 secured to the lower portion 222 of the compressor block by means of machine screws 229.

An O-ring seal 227 is provided between the abutting surfaces of the lower housing portion 222 and the lower end cap 238. The rotor 221a of the motor 221 is connected to the upper end of the drive shaft 234 and inductively connected to stator 221b mounted within housing portion 220 for driving the male rotor 224 connected to the lower end of the drive shaft 234. The male rotor 224 operates within the working space 236 to drive the female rotor 226. The rotors 224 and 226 may be fabricated in the same fashion as the rotors 24 and 26 of the embodiment of the invention illustrated in FIGS. 1 and 2.

Suction gas is introduced into the space between the compression mechanism 218 and the outer casing or housing including upper and lower cup-shaped shell members 212 and 214 through the suction inlet 240 in lower shell member 214. The suction inlet 240 is adapted to be connected to the suction line of a refrigeration system. The gas then passes through an opening 242 (FIG. 12) in the side of the lower portion 222 of the compressor block into the working space 236. The refrigerant is compressed by the screws 224 and 226 and discharged from the upper end or discharge end of the working space 236 through the openings or discharge ports 258 in the valve block 248. The compressed gases passing from the valve means 246 will pass over the refrigerant compressor motor and will be discharged through the discharge line 244 from the compressor. The passage of compressed gases through the ports 258 in the valve block 248 is controlled by the valve plate 252 which is fabricated in a fashion similar to the valve plate 52. The flexing of the valve members or arms 252 away from the discharge ports 258 is restricted by the retaining member or valve plate stop 254. Bolt or machine screw 250 extends through the central aligned openings in the valve plate stop 254, the valve plate 252 and the valve block 248 to retain these elements in place

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on the upper end of the female rotor 226. Therefore, the valve means 246 are rotatable with the rotor 226.

Another feature of this invention is the method for injecting fluid into the working space 236 to seal the small clearance between the rotors 224 and 226 and between the rotors and the housing 222. Pump means are provided in each rotor 224 and 226 for drawing fluid, namely, oil, from the sump between the compression mechanism and the outer housing and forcing it into the working space 236. The pump means in the rotor 224 will be described and it will be understood that the components of the pump means in rotor 226 are similar. Accordingly, the like elements of the pump means in rotor 226 will be identified by the same reference numeral as the pump means in rotor 224, except that the reference numerals will be primed. At start-up, oil is drawn through filter screen 265 and inlet opening 261 in thrust washer 262 into opening or bore 264 in the bottom of the rotor shaft. The oil is forced through transverse passages 267 in the drive shaft 234 by the rotation of the shaft and up the upright pump passages 266a and 266b. The outlets from passages 266a and 266b are in the bottom of the grooves in the rotor 224. The suction or filling action in the working space between cooperating rotors may assist the oil injection.

Oil will be discharged from the upper end of the working space 236 through the openings 258 in valve block 248 into the space 260 above valve means 246. The valve means 248 rotates with rotor 226 and the webbing 201, valve means 246 will function as a centrifuge to separate the heavier particles of oil from the lighter gas and, also, keeps the gas and oil at the same radial speed of the rotor. This is important for proper valve action. The oil particles will be slung from the recessed chamber 260 defined in part by recess 256 in the valve block 248 through the sidewall openings 280 in the valve block 248 to lubricate the surface between the bearing 249 and the valve block 248. Aligned openings are provided in the bearing 249 and hub of the upper housing portion 220. Lubricant will pass from openings 280 into the aligned openings 272 to lubricate the bearings 231, 233. Some oil carried over from the valve means 246 will be accumulated in the reservoir 273 defined in the upper housing portion 220. Also, oil overflowing the bearing 231 will be accumulated in reservoir 273. Oil from the reservoir 273 will be permitted to return to the sump defined in the lower portion of the lower shell 214 through the constant flow rate orifice 274. The foregoing provides novel means for lubricating the journal bearing surfaces of rotors 224 and 226.

By another aspect of the present invention, it is contemplated that the cooperating rotors in the compressor will be provided with a variable skew, so as to change the rate of compression, and, specifically, to slow down the rate of compression at the discharge end of the rotors, so as to reduce the pressure drop across the valve means.

FIG. 13 illustrates a chart with the circumference of the rotor plotted against the length. In FIG. 14, there is a layout of a variable flute of a variable skew rotor.

Within a flute of a conventional screw compressor, the rate at which compression occurs is a direct function of the rate at which flute or groove volume is reduced. Further, flute volume, having fixed cross-sectional area, reduces at a rate proportional to axial velocity of the seal line. The axial velocity (V_a) is:

$$V_a = \frac{wr}{\tan \alpha}$$

From this, it is seen that as the helix angle α increases with an accompanying increase in $\tan \alpha$, the axial velocity and the rate of the compression or rate of fluid discharge decreases. Referring to FIG. 13, flute A, having a variable skew, is depicted as having three helix angles α_1 , α_2 , and α_3 . The flute B is one with a constant helix angle α . Both flutes A and B "wrap" the same on the rotor so that the

rotor's axial length and pumping capacity are equal. For simplicity, in FIG. 13, the axial length is shown as the total lead for the rotor flute. The word "lead" is a machining term which simply means the axial length that would develop if the flute at helix angle α were to wrap the full 360° of the rotor. In practice, the axial length usually amounts to 50-75 percent of the "lead" length however, by utilization of the valve means of the present invention it need not be so limited.

At some angle of rotor rotation, the gas must exit from the compressor. The remaining flute distance is used for discharging gas. "Discharging" rotation is shown in FIG. 13 as ωtA for flute A and ωtB for flute B. It is seen that the angle remaining (and the time) allowed for discharging gas with flute A is greater than the angle with flute B. It is also seen that flute A can discharge the same quantity of gas at a lower rate and lower exit velocity than flute B.

The main advantages of a lower rate of discharge of gas is that it will promote smaller pressure losses as the discharge gas passes through the valve means at the discharge end of the working space. Pressure losses will vary as the square of the axial velocity, V_a . Axial velocity is lower for flute A, than for flute B, in the discharge portion adjacent the valve means 48 due to its larger helix angle (α_3 and $\tan \alpha_3$ is greater and α_1 and $\tan \alpha_1$).

The reason for flute A's ability to discharge before flute B has to do with the intermediate helix angle α_2 . In the region known to have a "no flow" condition, compression to discharge pressure can be expedited with a helix angle of smaller slope (α_2 and $\tan \alpha_2$ is less than α_3 and $\tan \alpha_3$ and α_1 and $\tan \alpha_1$). The higher axial velocity will cause a higher change in pressure per unit time.

Variable helix angles can be obtained in several ways to meet the need. The laid out flute can take the shape of a trigonometric function as in FIGS. 15 and 16 or a parabolic function as in FIGS. 17 and 18 and other shapes that may be expressed mathematically.

Considering FIG. 15, it is seen that the flute is again carried the full "lead" length for simplicity. The axial position of the flute, a is related to angular position as follows:

$$a = \frac{a_{\max}}{2} \left[1 - \cos \frac{\theta}{2} \right]$$

and the derivative of this function, da , is:

$$da = \frac{a_{\max}}{4} \sin \left[\frac{\theta}{2} \right] d\theta$$

The small triangle within the encircled area of the curve in FIG. 15 shows a typical helix angle, which is instantaneous in value. By definition, the tangent of this helix angle is:

$$\begin{aligned} \tan \alpha &= \frac{r(d\theta)}{da} \\ &= \frac{r(d\theta)}{\frac{a_{\max}}{4} \sin \left(\frac{\theta}{2} \right) d\theta} \\ \tan \alpha &= \frac{4r}{a_{\max} \sin \left(\frac{\theta}{2} \right)} \end{aligned}$$

Also:

$$\begin{aligned} V_a &= \frac{\omega r}{\tan \alpha} \\ &= \frac{\omega r}{\frac{4r}{a_{\max} \sin \left(\frac{\theta}{2} \right)}} \end{aligned}$$

$$V_a = \frac{a_{\max} \omega \sin \left(\frac{\theta}{2} \right)}{4}$$

- 5 Now, assume that the dimensions of the rotor in FIGS. 15 and 16 are $a_{\max}=4.25$ inches and $r=$ one inch, and assume also a rotating speed of 3,450 r.p.m. or 361 rad/sec.

At $\theta=180^\circ$, where a "no flow" condition exists:

$$10 \quad V_a = \frac{(4.25)(361)(\sin 90)}{4} = 383.5 \frac{\text{inches}}{\text{sec.}}$$

At $\theta=270^\circ$, where gas begins to leave the compressor:

$$15 \quad V_a = \frac{(4.25)(361)(\sin 135)}{4} = 271 \frac{\text{inches}}{\text{sec.}}$$

Thus, when compressed gas begins to leave the compressor, the axial seal line velocity is about 29 percent lower than the highest "no flow" velocity. As stated above, the higher velocity with a "no flow" condition is desirable in that the rate of compression per unit time is higher and more of the flute's length can be used for gas flow. The lower velocity while gas is discharging from the machine is also desirable. Pressure losses through the valve ports will reduce as the square of the reduction in velocity. This arrangement coupled with the rotating valve means provides for increased capabilities in screw compressor design.

There has been provided by the present invention an improved rotary screw compressor incorporating valve means rotatable with the screws and constructed and arranged to provide a variable compression ratio responsive to conditions of the refrigeration system in which the compressor is used. The valve means include a valve block having openings therethrough and a valve plate rotatable with the valve block for closing such openings, the valve block having ribs therein for assuming that gas and liquid from the openings rotate at the same radial speed and to help centrifuge liquid from the gas. Another feature of this invention is the provision of such rotary screw compressor with screws that have a preselected variable skew for providing a preselected variable compression rate during operation. The advantage of this is that the velocity of discharging gases is reduced and pressure drop is reduced through the valving means at the discharge end of the working space. Pump means are provided in the rotors for injecting oil from the sump within the compressor into the working chamber without the necessity for components external of the compressor. Further, a separator for separating compressed gas and sealing medium is incorporated into the novel compressor.

While I have shown presently preferred embodiments of the invention, it will be understood that the invention is not limited thereto, since it may be otherwise embodied within the scope of the following claims.

55 What is claimed is:

1. In a screw compressor including a housing defining a working space having at least two cooperating screw rotors disposed therein for compressing fluid, the improvement characterized by valve means rotatable with the screw rotors including a rotatable discharge valve, said discharge valve being responsive to pressure in the working space so as to provide a variable compression ratio comparable to that of a reciprocating compressor.

2. A compressor as in claim 1 wherein the valve means comprises a valve block secured to a rotor for rotation therewith, said valve block having discharge port means therein, and said discharge valve controls the passage of gas from the working space through the discharge port means.

70 3. A compressor as in claim 2 wherein the discharge valve comprises a plate and the valve means include a stop member for restricting movement of the discharge valve plate from the discharge port means.

75 4. A compressor as in claim 3 wherein web members are provided in the valve block to assure that gas and

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liquid emitted from the discharge port means is moving at the same radial speed and to centrifugally separate the gas and liquid.

5. A compressor as in claim 2 wherein the discharge valve comprises a flat planar member, the flat planar member tending to remain flat and in discharge port means closing position due to centrifugal force during operation of the rotors.

6. A compressor as in claim 2 wherein liquid is injected into the working chamber and the valve block is cup-shaped and includes web members within a recessed chamber in the cup-shaped valve block for centrifuging and separating gas from said fluid.

7. A compressor as in claim 1 wherein the compressor includes an outer casing surrounding said housing and wherein a liquid sump is provided within the outer casing of the compressor, at least one of said rotors including a shaft portion having pump means therein communicating with said sump for ingesting fluid from said sump and discharging it into said working space for sealing the working clearances between the rotors and between the rotors and the housing.

8. A compressor as in claim 7 wherein said rotors include lobes and grooves and said pump means include eccentrically disposed passage means in said shaft portion for communicating liquid to the grooves of said one of said rotors.

9. A compressor as in claim 7 wherein separator means are provided within the outer casing for separating liquid from the compressed gas discharged through the valve means.

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10. A compressor as in claim 9 including a separator housing disposed in the outer casing and spaced therefrom, said sump being defined in said separator housing, baffle means in the space between the separator housing and the outer casing, and means for returning liquid separated from compressed gas in said space to said sump.

11. A compressor as in claim 1 wherein the rotors each have flutes with a variable skew angle for discharging gases through the valve means a predetermined velocity less than the velocity from a comparable rotor having a uniform skew.

References Cited

UNITED STATES PATENTS

1,890,205	12/1932	Andresen	230—139
1,930,403	10/1933	De Bijle	230—143
1,931,323	10/1933	McCormack	230—139
2,460,310	2/1949	Rathman	230—143
2,545,600	3/1951	Berry	230—139
2,578,196	12/1951	Montelius	230—143
2,656,972	10/1953	Rathman	230—143
2,883,101	4/1959	Kosfeld	230—139

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