

Jan. 3, 1967

F. O. BELLMER

3,295,752

ROTARY VANE COMPRESSOR

Filed April 4, 1966

3 Sheets-Sheet 1

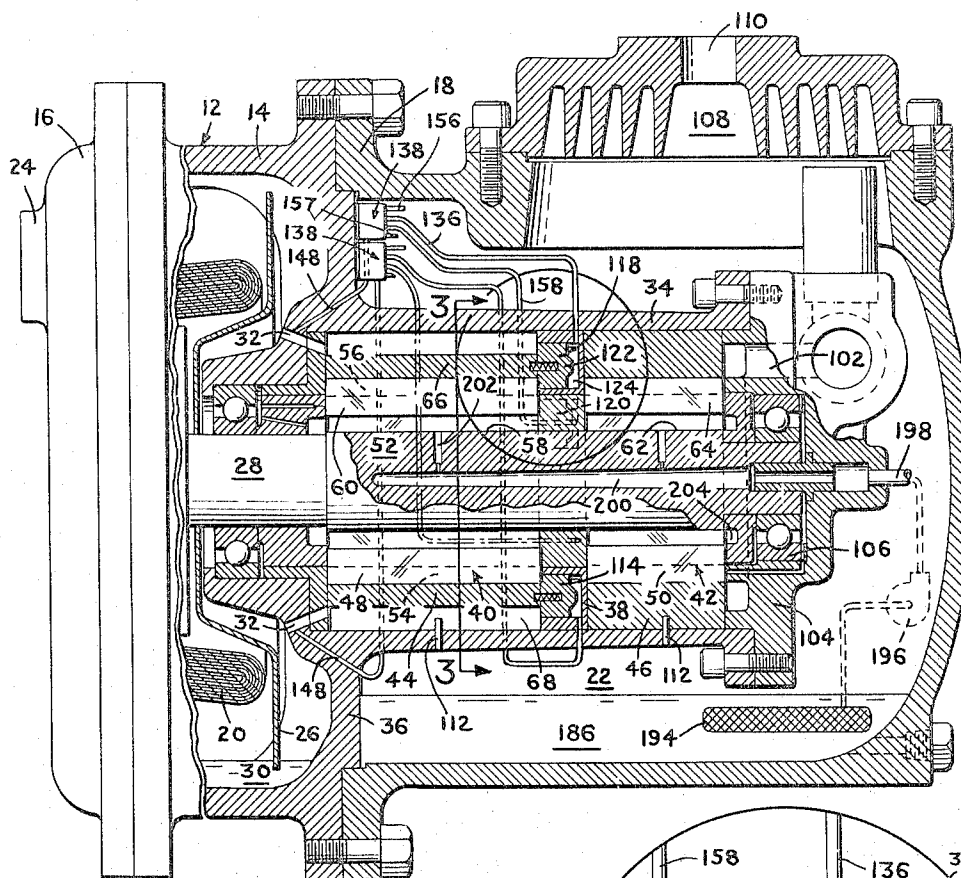


FIG. 1

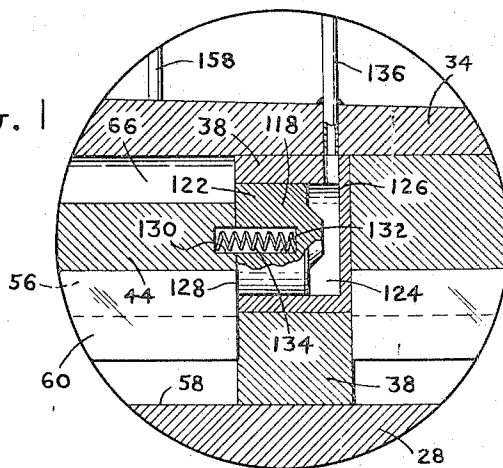


FIG. 2

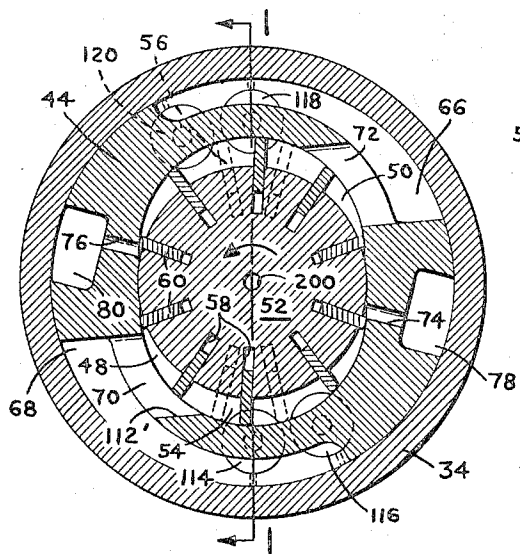


FIG. 3

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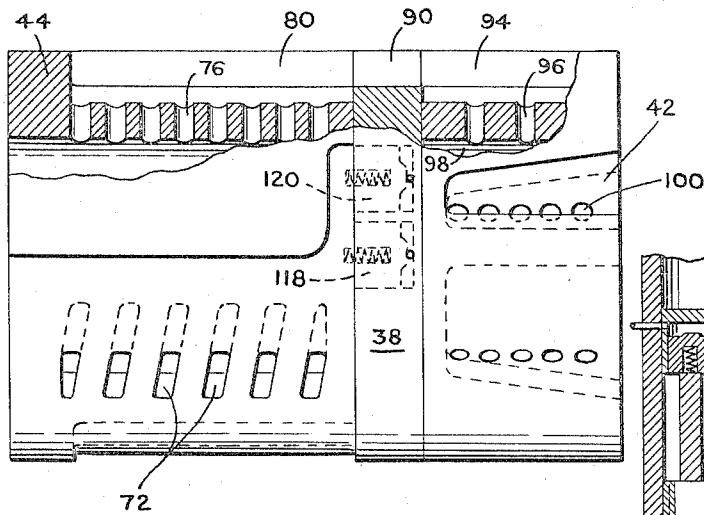


FIG. 4

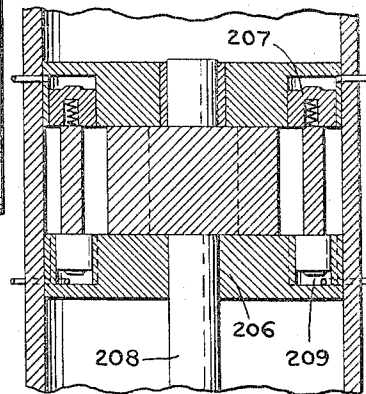


FIG. 5

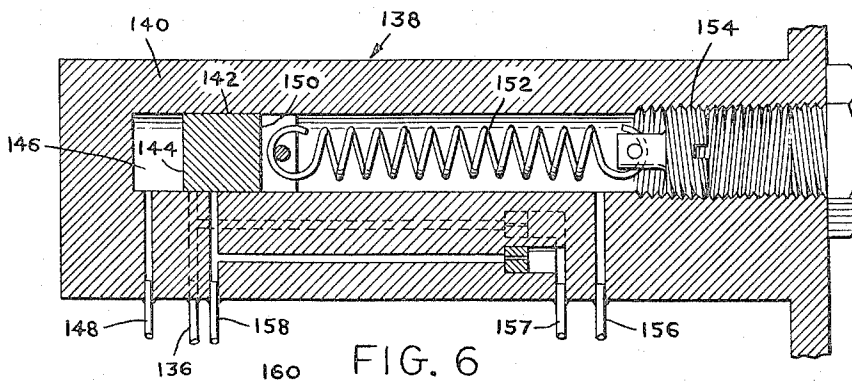


FIG. 6

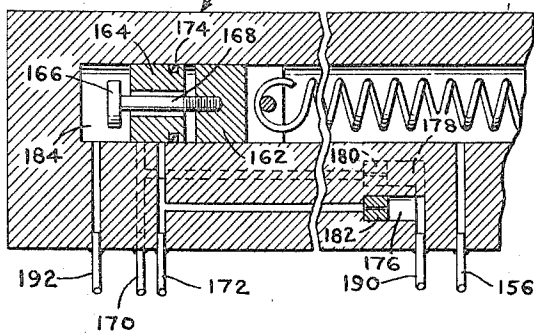


FIG. 7

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FIG. 8

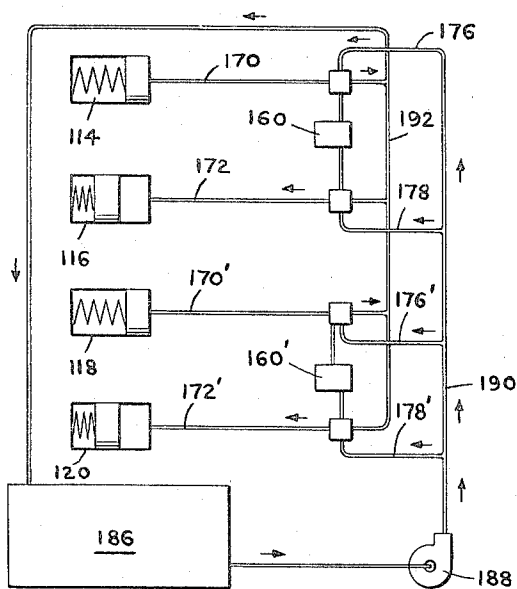
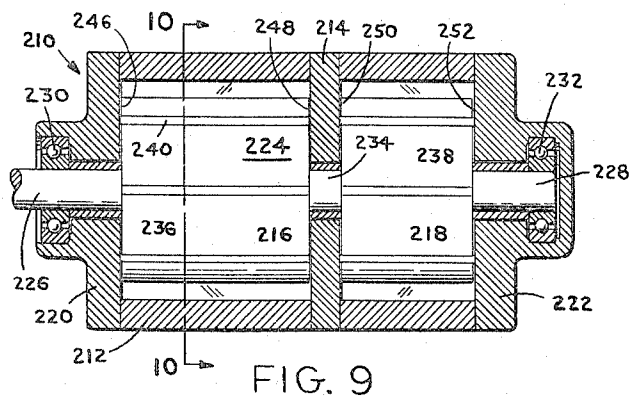


FIG. 9

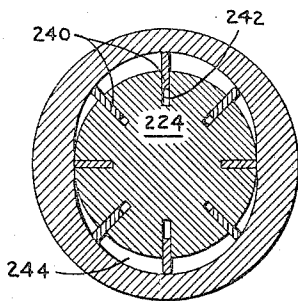


FIG. 10

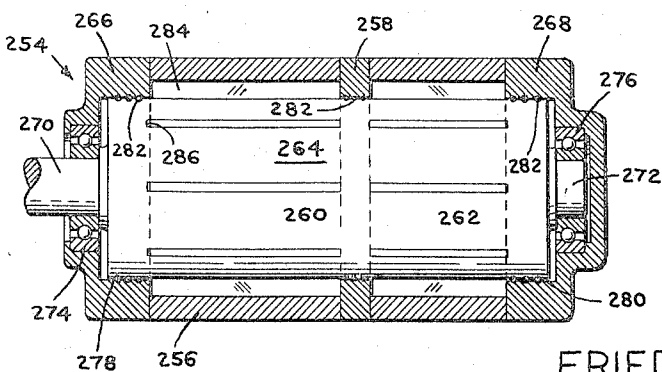


FIG. 11

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3,295,752

## ROTARY VANE COMPRESSOR

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16 Claims. (Cl. 230-138)

In general, this invention relates to new and improved rotary vane compressors and improvements thereon. More particularly, it relates to a better and more easily manufactured rotary sliding vane compressor with improved unloading and loading control features for varying the capacity of the machine.

This application is a continuation-in-part of my co-pending U.S. Patent application Serial No. 363,555 filed April 29, 1964, now abandoned, for "Capacity Control for Rotary Vane Compressors."

Capacity controlled rotary vane compressors have, in the past, utilized the principle of throttling the discharge for suction flow, or relieving the discharge pressure to suction. These systems have always been objectionable because they involve energy losses and, accordingly, even when the compressor was operating in an unloaded state, it would require as much energy as in the loaded condition. Other types of capacity controls utilized in the past included variable taps or lifting devices for the vanes. However, these required complicated mechanical structures and were extremely expensive to manufacture.

There has been some indications in the prior art that a better means for varying the capacity of a compressor would be to change the location of the closing edge of the suction openings in such a way that more or less gas will be compressed by the vanes if more or less capacity is required. However, these systems were also difficult to control and not amenable to automatic operation.

Another problem which arises specifically when the compressor forms part of a refrigeration system is the possibility that liquid will find its way into the compressor and cause over compression. This would be undesirable as, in some cases, it can cause a breakdown of the compressor.

Rotary sliding vane compressors having more than one stage of compression have generally been the type wherein a rotor was driven from a single source and was divided, by suitable partitions, to form the two stages. However, the manufacture of these types of compressors involves certain inherent problems. For example, to obtain clearances between the portions of the rotors in each stage of the compressor and the partition walls forming the stages, required the compounding of at least four tolerance deviations and, in addition, there was, by necessity, additional tolerance variations which had to be compensated for in forming the space for positioning the bearings and the placement of the grooves on the rotor within which the vanes would be positioned. Additionally, the vanes themselves created tolerance problems in their position within each stage. These tolerance variations had to be further compensated for heat expansion and axial loads in order to obtain desired clearances. Therefore, using tolerances which would not necessitate high machining costs created larger gaps than desired and resulted in a decrease of volumetric efficiency. If extremely accurately machining were utilized, production costs would rise to impracticably high levels.

In order to avoid high start up torques, it is also desirable to start operation of a compressor in its unloaded condition and then bring it up to the loaded condition as quickly as possible. In this way, there will be avoided any heavy drains on the motor driving the compressor. However, this stepped loading at start up should be done automatically and quickly in order to be effective.

It is therefore the general object of this invention to avoid and overcome the foregoing and other difficulties of prior art practices by the provision of a new and better rotary, sliding vane, compressor.

Another object of this invention is the provision of a new and better rotary sliding vane compressor having controlled loading and unloading achieved without energy losses.

Another object of this invention is the provision of a new and improved rotary sliding vane compressor which, with a single unit, achieves control of loading and unloading and, additionally, compression overload sensing.

A further object of this invention is the provision of a new and more economical multi-stage rotary sliding vane compressor which can be manufactured with a minimum of machining to achieve minimum deviation in tolerances between moving parts.

A further object of this invention is the provision of a new and improved rotary sliding vane compressor operative to start up with the unloaded position, and automatically, to build up to the loaded condition.

A still further object of this invention is the provision of a new and better rotary sliding vane compressor forming a part of a refrigeration system which will automatically control the load on the compressor in accordance with the load on the refrigeration system.

Other objects will appear hereinafter.

For the purpose of illustrating the invention, there are shown in the drawings forms which are presently preferred; it being understood, however, that this invention is not limited to the precise arrangements and instrumentalities shown.

The objects of the present invention are achieved by utilizing a rotary vane compressor, preferably multi-stage, in which unloader pistons are placed in the side walls of the chambers adjacent the leading edge of the inlet opening to the chambers. These unloader pistons, when opened, allow gases from a portion of the compression chamber to be returned to the inlet passageway before any compression of these gases has taken place. Accordingly, there is no energy lost during the unloading process and the unloader pistons act to move the leading edge of the inlet opening closer to the discharge outlet of each compressor chamber. The loader pistons themselves are spring biased to an open position and are maintained in a closed position by way of pressurized fluid which, in the preferred embodiment, is supplied from the discharge outlet of the compressor itself. In this way, on starting up, since the discharge pressure is essentially the same as the suction inlet pressure, the unloader pistons will be opened and the compressor will operate unloaded. As the pressure builds up, the unloader pistons will be closed and the compressor will run loaded. Should any liquid be passed through the suction inlet into one of the compression chambers, the unloader pistons will automatically open as the compressor starts to overload thus returning the liquid back to the inlet.

The multistage rotary sliding vane compressor of the present invention is manufactured with a single rotor of a constant diameter with the housing being partitioned into chambers in a manner whereby the rotor extends beyond the opposite end walls of the stages in the compressor. The vanes are positioned in slots slightly longer than the vanes, whereby the length of the rotor is removed as a critical factor in manufacturing the compressor to close tolerances.

The present invention may be embodied in other specific forms without departing from the spirit or essential attributes thereof and, accordingly, reference should be made to the appended claims rather than to the foregoing specification as indicating the scope of the invention.

FIGURE 1 is a segmentary view in cross-section of one embodiment of the compressor of the present invention.

FIGURE 2 is a segmentary view on an enlarged scale of a portion of the compressor shown in FIGURE 1.

FIGURE 3 is a cross-sectional view taken along lines 3—3 in FIGURE 1.

FIGURE 4 is a top elevational view of a portion of the compressor with walls broken away to show internal members.

FIGURE 5 is a cross-sectional view of an alternate embodiment of the present invention showing unloader means at opposed ends of each compression chamber.

FIGURE 6 is a partial cross-sectional view of the unloader control shown in FIGURE 1.

FIGURE 7 is a partial cross-sectional view of a second type of unloader control built in accordance with the teachings of the present invention.

FIGURE 8 is a schematic showing of another type of control for the unloader system of the present invention.

FIGURE 9 is a schematic representation of a basic prior art compressor with walls removed.

FIGURE 10 is a cross-sectional view of the compressor of FIGURE 9 shown along lines 10—10.

FIGURE 11 is a schematic view of a compressor built in accordance with the principles of the present invention shown in the manner similar to the compressor of FIGURES 9 and 10 so as to emphasize the differences therebetween.

In FIGURE 1, there is shown a compressor built in accordance with the principles of the present invention and generally designated by the numeral 12. The compressor 12 includes an elongated casing 14 divided into two compartments 16 and 18. The compartment 16 houses a motor 20 and compartment 18 houses the compressor 22. Refrigerant fluids enter through inlet 24 into motor compartment 16 and aids in cooling the motor 20 during its operation. Thus, during normal usage, the compressor is incorporated into an enclosed fluid cycle in a refrigeration system. Thus, the motor compartment 16 inlet 24 is connected to the low pressure side of the refrigeration system receiving vaporized refrigerant therefrom. The refrigerant vapors pass from the motor compartment 16 into the compressor compartment 18 in a manner to be discussed below. In the compressor compartment 18, the refrigerant vapors are first passed through a first stage of compression, thence to a second stage of compression, and then discharged to an oil separator and thence to the condenser or other appropriate portion of the refrigeration system.

The compressor unit is hermetically sealed and, under normal operating conditions, there will be a substantial pressure imbalance between the motor compartment 16 and the compressor compartment 18. This pressure imbalance can result in unequal distribution of lubricant oil which accumulates in the low pressure motor compartment 16. To assure a return of lubricant to the compressor compartment, a flinger 26 or similar rotating device is attached to the shaft 28 of motor 20 to splash into the oil bath 30 in motor compartment 16. The oil sprayer mist caused by splashing of the flinger 26 into the oil bath 30 will then be carried by the refrigerant gas into the compressor 22 through the compressor inlet openings 32.

The compressor 22 includes an elongated casing 34 which is an extension of the casing 14. The compressor casing 34 has an upright partition wall 36 at one end thereof which forms a partition between the motor compartment 16 and the compressor compartment 18. Centrally on the compressor casing 34 there is provided an inwardly directed partition wall 38 which divides the compressor casing into a first or low pressure stage 40 and a second or high pressure stage 42. The first stage 40 has its discharge outlets connected to the inlet passageways of the second stage in a manner to be discussed below.

The first and second stages 40 and 42 are formed by first and second compressor stage liners 44 and 46 having their outer surface contiguous with the inner surface of the casing 34 and being separated by partition wall 38.

Liners 44 and 46 have ellipsoidal cavities 48 and 50 each having a minor diameter substantially equal to the diameter of a cylindrical rotor 52 driven by shaft 28. Rotor 52 forms, with the liner 44, two compression chambers 54 and 56, arcuate in shape, within the ellipsoidal cavity 48. Similarly, the rotor 52 forms with in the liner 46, two compression chambers in the second stage 42 within the cavity 50.

The rotor 52 has ten equally spaced longitudinally extending radial slots 58 within which are slidably mounted vanes 60. The vanes 60 reciprocate within the slots 58 and have their outer edges forced against the surface of the cavity 48 by centrifugal force.

Similarly, there are ten slots 62 and associated sliding vanes 64 on the portion of the rotor 52 within the cavity 50.

As stated previously, refrigerant fluid enters through the inlet 24 into the motor compartment 16. These low pressure vapors, after passing across the heated motor parts for cooling the latter, pass from the motor compartment by way of inlets 32 into the compressor compartment 18. The low pressure vapor are mixed with oil tossed by the flinger 26.

The inlet openings 32 connect to compressor inlet passageways 66 and 68 formed in liner 44. The inner passageways 66 and 68 extend arcuately from respective points on opposite sides of the minor diameter of the ellipsoidal cavity 48 to respective points substantially more than halfway around the compression chambers 54 and 56. The liner 44 separates the inlet passageways 66 and 68 from their respective compression chambers 54 and 56. However, adjacent the minor diameter portion of the ellipsoidal cavity 48 there are provided slots 70 and 72 respectively connecting the inlet passageways 66 and 68 to the compression chambers 54 and 56.

At the opposite ends of compression chambers 54 and 56 from the slots 70 and 72 there are provided a second group of slots 74 and 76 respectively which are in communication with outlet passageways 78 and 80 respectively. The outlet passageways 78 and 80 are connected through a suitable connecting passageway 90 through partition wall 38 to the inlet passageways 94 of the second stage 42 of the compressor. The inlet passageways 94 are connected through the suitable slots 96 to the compression chambers 98 of the second stage 42, and, thence through suitable outlet slots 100 to a common outlet passageway 102 formed in the end cover 104 for the casing 34. The end cover 104 also supports the bearings 106 holding the shaft 28 and rotor 52 in place.

The outlet passageway 102 directs the hot compressed refrigerant gases into an oil separator 108 from which oil free gas may pass through an outlet 110 into the refrigeration system.

The liners 44 and 46, as shown, guide the rotor vanes 60 and 64. The liners 44 and 46 are themselves pinned to the casing 34 by a plurality of fasteners 112.

Although the cavities 48 and 50 have been shown as ellipsoidal to define two diametrically opposed compression chambers, it is understood however that any number of such chambers may be incorporated into the cavity configuration to provide a multi-chamber compressor structure.

The walls of the cavities 48 and 50 are machined to provide a smooth sliding fit and substantially fluid tight seal with the moving vanes. Compression of gases entering the inlet passageways 66 and 68 is achieved by allowing the gases to enter through slots 70 into the space between adjacent vanes 60 as best shown in FIGURE 3. As the trailing vane 60 passes by the leading edge 112' of the slot 70, there is formed a closed space which slowly has its volume decreased until the leading vane 60 reaches

the slot 74. Thus, the lead edge of 112' should be at the maximum volume point in compression chamber 54.

Each of the compression chambers 54 and 56 in the first stage 40 has two unloader assemblies 114, 116, and 118, 120 respectively. Only the unloader assembly 118 will be described in detail, it being understood that the other unloader assemblies 116, 118, and 120 are substantially similar in construction.

The unloader assembly 118 includes a reciprocally movable plunger 122 guided in a cylindrical chamber 124 having a back wall 126. One face 128 of the piston 122 is disposed flush with the rubbing face of partition wall 38 to contact the edge surface of vanes 60, the edge surface of liner 44, and a portion of inlet 66. The liner 44 has a recess 130 axially aligned with a recess 132 extending inwardly from the face 128 of plunger 122. Within the recesses 130 and 132 there is disposed a coil spring 134 for normally biasing the plunger 122 to an open position. An inlet conduit 136 is utilized to provide fluid behind the plunger 122 to maintain it in its closed or loaded position.

It will be noted, as shown in FIGURE 3, that the unloader assembly 118 is positioned immediately down stream of the leading edge 112' of slots 70. Thus, the plunger 122 straddles the portion of the liner 44 between inlet passageway 66 and compression chamber 54. With the piston or plunger 122 in the closed position as shown in FIGURE 2, compression of gases within the chamber 54 is as has been described previously. However, if plunger 122 is retracted to an open position, a bypass channel is formed in the compression chamber permitting flow of gas from the compression chamber 56 through the chamber 124 into the inlet passageway 66. This is because the space between adjacent vanes 60 is greater than the distance between unloader assembly 118 and the leading edge 112'. If this were not the case, the gases would start to compress before being bypassed by the unloader assembly 118 and, accordingly, energy would be lost.

The second plunger 120 is spaced down stream from the first plunger 118 a distance less than the distance between adjacent vanes so that, in effect, the leading edge 112' of the passageway 70 has been moved down stream to the unloader assembly 120. Accordingly, a smaller volume of gas will be compressed within the compression chamber 56 thus decreasing the load on the compressor or "unloading" the compressor.

Each pair of unloader assemblies 114, 116, and 118, 120 are connected to a sensing unit 138. The sensing unit 138 uses the pressure difference between the suction and discharge pressures to control a pair of unloader assembly. The operation of the sensing unit 138 can best be shown with respect to FIGURE 6 wherein the sensing unit 138 shown as comprising a cylinder 140 within which is mounted a piston 142 for slidable movement therein. The piston 142 has a low pressure face 144 forming a chamber 146 with the closed end of the cylinder 140. The chamber 146 is in communication with the inlet passageway 32 through conduit 148. The opposite side face 150 of piston 142 is connected to a suitable spring 152 whose tension is controlled by a screw plug 154. This high pressure side of the piston 142 is connected through a suitable passageway 156 to the interior of the compressor housing 18 which is, of course, at the discharge pressure of the compressor. Alternatively, the outlet 156 could be connected to the discharge from the first stage 40 of the compressor.

Spaced between passageways 148 and 156 are two passageways 158 and 136 which are connected respectively to the unloader assemblies 120 and 118.

In operation, if for example, the suction pressure through inlet 148 is 60 lbs. and the discharge pressure 120 lbs., the difference of 60 lbs. on the opposed faces 144 and 150 of piston 142 has to be balanced by the

spring 152 so that the piston 142 just closes passageway 136 and 158.

If the suction pressure goes down from 60 to 58 lbs., for example, and the discharge pressure thus decreases from 120 to 116 lbs., the pressure difference decreases from 60 to 58 lbs. This means the required capacity for the refrigeration system is too high. The piston 142 then moves, pulled by the spring 152, and opens passageway 136. Opening of passageway 136 causes plunger 122 to move to the unloaded position thus dropping the capacity of the compressor. This first step of capacity control decreases the capacity of the compressor a certain amount to a new balance position. If the change in pressure is still higher, the piston 142 will travel opening passageway 158 to unload piston 120 and accordingly decrease the capacity further. Then a new balance is established. If the pressure increases, the discharge inlet 156 pressure will cause the plunger 142 to move to the left closing passageways 158 and 136 returning discharge pressure from passage 157 to the plungers forming parts of the unloaders 118 and 120 and placing them in the loaded position again. It will be seen that on startup of the compressor, the compressor discharge pressure is equal to the inlet pressure and, accordingly, the piston 142 will be moved to the right allowing the compressor to start unloaded. The compressor will gradually increase its capacity as pressure is built up in the compressor.

If in certain cases the capacity control for the unloader assemblies cannot operate too often because of undesirable wear on the parts or, if greater differential is required, a deviation from the design of FIGURE 6 may be utilized, as shown in FIGURE 7. In addition, if oil pressure is to be utilized to control the unloader assemblies as shown in FIGURE 8, then the capacity control must also be modified. For this reason, the capacity control device 160 is shown in FIGURE 7.

The control 160 is substantially similar to the control 138 shown in FIGURE 6 except that changes in pressure which move the piston 162 back and forth will not affect a delay piston 164 so long as the length of travel of the piston 162 is smaller or equal to the distance between the head 166 and one face of the delay piston 164 or the distance between the other face of the delay piston 164 and the piston 162. The piston 162 is connected to the head 166 through a rod 168 passing through a central bore in the delay piston 164. Thus, only when changes in pressure become excessive, will the piston 162 push the delay piston 164 back or forth so as to open or close channels 170 and 172. The adjustable width of the gap between head 166 and delay piston 164 or between delay piston 164 and piston 162 gives a degree of steering delay. In case of slight friction of the delay piston 164 is desirable to prevent any self-movement, a piston ring 174 is provided.

The unloaders 114, 116, 118 and 120 are adapted to be controlled by oil pressure with each control device 160 controlling a pair of unloaders 114, 116 or 118, 120. The control device 160 has channels 170, 172 connected respectively to unloaders 118, 120. The oil pressure is connected to the control device 160 by the control channels 176, 178. The operational speed (switching from one step to the other) is chosen by setting the orifice size of bleeders 180, 182 in channels 176, 178. The oil, if channels 170 and 172 are opened, will bleed into the cylinder chamber 184 within which piston 164 is mounted and return to the oil sump 30 joining there the lubrication oil. With channels 170, 172 closed, any leakage oil will lubricate the piston 164 and then be returned to the oil sump.

The only sliding part, and, therefore, the only spot of friction, is the piston. Thus, the greatest operating accuracy can be expected. If necessary, to eliminate any gas leakage through the piston, the cylinder cham-

ber 184 may stay under the oil level and be filled with oil instead of gas.

In FIGURE 8, the oil pressure control unloader system is illustrated schematically. Thus, a source of oil is utilized as the pressurizing medium. Such oil can be found in the reservoir 186, which reservoir is connected to a suitable pump 188 which pumps the oil through a manifold 190. Manifold 190 is connected to two sets of oil pressure inlets 176, 178 and 176', 178' which are connected to their respective control devices 160 and 160'. Pressurized oil is then pumped through manifold 190 and control device 160, 160' to the unloader devices 114, 116 and 118, 120 respectively. The control devices 160 and 160' are further connected by return manifold 192 to the pressurized medium source 186. With the piston 164 in the position shown in FIGURE 7, pressurized medium is being supplied through conduits 170, 172 and 170', 172' to the rotor assemblies 114, 116 and 118, 120. However, when the piston 164 is moved to the right uncovering passageway 170 and, in control device 160', the passageway 170' then the spring biased unloader assemblies 114 and 118 will be unloaded as shown in FIGURE 8 and the oil will be removed by passageway through manifold 192 back to sump 186. Further movement to the piston 164 will cause unloading of unloader assemblies 116 and 120.

Whether oil pressure or gas pressure is utilized to bias the unloader pistons 122, that pressure is set at a value which when subtracted from the bias pressure of spring 134 will cause the unloader assembly to open when pressure in an individual compression pocket between adjacent vanes 60 adjacent a particular unloader assembly exceeds a predetermined value. Thus, this act as an automatic safety feature to prevent damage to the compressor should there be over compression for any reason. Such over compression could be caused by a large amount of liquid being supplied to the inlet of the compressor, variations in speed of the compressor, and for other causes which are well known. Thus it can be seen that the unloader assemblies have a double function. That is, they are controlled unloaders and, additionally, are safety valves to prevent damage to the compressor due to over compression.

In the closed refrigerant system, hot compressed vaporized refrigerant is passed from the compressor discharge 110 and is subsequently returned to the motor compartment 16 for cooling the latter. Since a certain amount of the liquid lubricant is carried through the system by vaporous refrigerant, such lubricant will tend to accumulate in the motor compartment 16 in the pool 30.

An oil strainer 194 placed in the oil sump 186 filters oil which is pumped by pump 196 into an inlet 198 in the shaft 28. Thereafter, oil under pressure is directed through an axially extending passageway 200 and radially extending slots 202 into the elongated guide slots 58 guiding the sliding vanes 60 to lubricate the interface between the vane surfaces and the guiding slots.

Some lubricant from the grooves 58 is thrown by centrifugal force outward against the housing walls for lubricating the latter and improving the gas tight seals between said walls and the vane upper edge. Excess oil flows laterally to end grooves 202 and 204 respectively. In said grooves, the oil is bled to the shaft bearings 106 to assure operation thereof with minimum maintenance.

As shown in FIGURE 5, an alternate embodiment of the present invention resides in an arrangement for incorporating unloading means into opposed ends of a rotary vane compressor. As shown in FIGURE 5, the compressor assembly includes a casing 206 which supports a shaft 208 on which the sliding vanes are operably carried. Unloader assemblies 207 and 209 are disposed in opposite ends of the compressor casing in communication with the compressor chambers formed by the rotating vanes.

Unloader assemblies 207 and 209 are similar in construction to the unloader assembly herein described. However, in the present embodiment, in loader assemblies being disposed at opposite ends of the compressor casing, is simultaneously function or alternately function in sequence to effect a desired degree of unloading in the compressor stage. Thus, a greater number of unloader assemblies may be incorporated into a single unit to provide a lesser increment of capacity differential for purposes of finer control. It will be noted that the unloaders 207 and 209 are not in the same radial position and, in fact, are not aligned with one another so as to effect this incremental variation.

In FIGURE 9, there is shown a conventional compressor 210 stripped of many of the features shown in FIGURES 1-8 in order to more clearly describe this present embodiment. That is, the FIGURE 9 and FIGURE 10 compressor 210 includes an outer casing 212 having a centrally disposed partition wall 214 to divide the compressor 210 into two stages 216 and 218. Stage 216 is formed within wall 212 between partition wall 214 and a second end wall 220 within the casing 212. A rotor 224 is mounted between shaft extensions 226 and 228, which shaft extensions 226 and 228 are supported by bearings 230 and 232 mounted respectively within end walls 220 and 222. The rotor 224 has a reduced diameter portion 234 which is in rotary sealing engagement with the annular partition wall 214. The rotor 224 is thus divided into sections 236 and 238 in compressor stages 216 and 218. Each of the rotor sections 236 and 238 has longitudinally extending parallel vanes 240 along the surface thereof 242. The casing 212 has an ellipsoidal cavity 244 to form a pair of compression chambers as has been taught by the prior art.

In constructing and assembling the compressor 210, provision must first be made to limit variations in length at the gaps 246 and 248 between rotor section 236 and end wall 220 and rotor section 236 and partition wall 214 respectively; and additionally to minimize the variation between partition wall 214 and rotor section 238 forming gap 250 and the space between rotor section 238 and end wall 222 forming gap 252. The tolerance variations between gaps 246, 248, 250, and 252 are accumulative. These gaps should have a maximum width of about .0005 inch secure a sufficient gas seal between discharge and suction within and between the stages 216 and 218. This requirement cannot be achieved by easy means with the structure shown in FIGURES 9 and 10 because of the following reasons:

(1) From the production side, to obtain a clearance of .0005 inch for gap 252 where nine tolerances are accumulated from bearings, rotor parts, and housing parts would result in extremely high production costs. Fitting of the parts together is thus out of the question.

(2) Variations in length because of heat expansions and axial loads have to be compensated for by increased clearances at 246, 248, 250, and 252. Therefore, using tolerances which would not necessitate high machining costs would end up with a gap 252 of about .007 inch wide, fourteen times wider than the required width as per the above example. This increase in the maximum tolerance results in a decrease of volumetric efficiency.

In FIGURE 11, there is shown a new compressor built in accordance with the teachings of the present invention which enables the compressor to be manufactured in a far more simple manner. That is, the compressor 254 includes an outer casing 256 divided by a partition wall 258 into two stages 260 and 262. The stages 260 and 262 are formed between a rotor 264 having uniform outer diameter, the casing 256, partition wall 258, and oppositely disposed end caps 266 and 268. The rotor 264 is mounted on shafts 270 and 272 supported by bearings 274 and 276 mounted respectively within end caps 266 and 268. The rotor 264 has a uniform outer diameter and extends a length greater than the overall length of

the two stages 260, 262 so that the ends of the rotor 264 extend within cup-shaped cavities 278 and 280 formed respectively in end caps 266 and 268. Thus the inner diameter of the annular partition wall 258 is slightly greater than the diameter of the rotor 264. The end face seals of the rotor are thus eliminated. The only seal gaps are, therefore, the gap between the rotor 264 and the cup-shaped cavities 278 and 280 and the gap between the partition wall 258 and the rotor 264. These clearances are similar to those of the ordinary shaft seals found in the configuration of FIGURE 9. A minimum of clearance of .0005 inch or smaller between the rotor 264 and the cup-shaped cavities 278, 280 and the partition wall 258 is neither a cost nor a production problem. Deviations in the length of the rotor 264 caused by bearing, rotor, and housing tolerances and also by heat expansions and axial loads do not have an effect on these clearances. Rather, the seals between the cup-shaped cavities 278 and 280 and the shaft 264 can be enhanced by the use of a suitable O-ring seals 282 on the outer surface of the rotor 264. Any individual fitting on the parts is thereby eliminated. With respect to the vanes 284 and their associated slots 286, the slots 286 can be made slightly longer than the vanes 284 with the end spacings between the vanes and slots being filled with the lubricant for the vanes so as to prevent gas losses through the rotor 264. Accordingly, the only tolerance required for the vanes are that they are accurately spaced between the partition wall 258 and their respective end wall end caps 266 and 268.

Thus, the compressor can be easily manufactured while maintaining minimum tolerances and all of the length variations in the rotor have been eliminated as source of possible error.

The present invention may be embodied in other specific forms without departing from the spirit or essential attributes thereof and, accordingly, reference should be made to the appended claims rather than to the foregoing specification as indicating the scope of the invention.

I claim as my invention:

1. A sliding vane rotary compressor comprising:

- (a) a casing having an elongated cavity formed therein,
- (b) said casing including end walls on opposite ends thereof,
- (c) a rotor positioned within the cavity and rotably mounted in the end walls, said rotor having axially extending vanes slidably carried in the outer surface thereof to engage the walls of said cavity,
- (d) said end walls, said rotor, and walls of said cavity forming at least one compression chamber in said cavity, said sliding vane extending radially outwardly from the axis of said rotor to form compression pockets between adjacent rotor vanes,
- (e) an inlet passage in said casing having an opening in said compression chamber at the portion of maximum volume of said compression chamber, an outlet passage through said casing having an opening to said compression chamber at the portion of minimum volume of said compression chamber, said compression chamber decreasing in volume from the inlet passage opening to the outlet passage opening,
- (f) at least one bypass passageway having its inlet opening between said inlet and outlet passage openings in said compression chamber, said bypass passageway having its outlet opening connected to said inlet passage,
- (g) said bypass passageway inlet opening being spaced from said compression chamber inlet opening a distance less than the distance between the tips of adjacent vanes forming a compression pocket,
- (h) and valve means in said bypass passageway, said valve means being responsive to the pressure within an individual compression pocket adjacent to said

bypass passageway inlet opening and, additionally, responsive to a separate control signal, said valve means being operative to be opened when the pressure in the individual compression pocket exceeds a predetermined value.

2. The sliding vane rotary compressor of claim 1 wherein said bypass passageway includes a piston receiving cavity in said end wall; a portion of said piston receiving cavity forming said bypass passageway inlet opening, said valve means including a piston slidably mounted in said piston receiving cavity with one face thereof being adapted to cover said bypass passageway inlet opening.

3. The sliding vane rotary compressor of claim 2 including spring biasing means for biasing said piston to an open position to open said bypass passageway inlet opening and connect said compression chamber to said inlet passage.

4. The sliding vane rotary compressor of claim 2 wherein said valve means includes a fluid inlet means connected to said piston receiving cavity at the end thereof opposite from said inlet opening to allow fluid pressure to be applied to the side of said piston opposite from said inlet opening.

5. The sliding vane rotary compressor of claim 4 including fluid pressure supply means operative to supply fluid under pressure to said fluid inlet means to force said piston to close said inlet opening, said fluid supply means supplying fluid at a pressure greater than the desired pressure within compression pockets adjacent said inlet opening whereby when pressure in compression pockets adjacent said inlet opening exceeds a predetermined amount, said piston will open to bypass fluid in compression pockets adjacent the inlet opening to the inlet passage.

6. The sliding vane rotary compressor of claim 1 including control means operative to control said valve means; said control means including pressure differential responsive means, said pressure differential responsive means having a first inlet responsive to the pressure in said inlet passage and a second inlet responsive to the pressure in said outlet passage, said pressure differential responsive means being operative in one position to supply a first control signal to said valve means to open said bypass passageway and, in the second position to supply a second control signal to said valve means to close said bypass passageway.

7. The sliding vane rotary compressor of claim 6 wherein said pressure differential responsive means is a piston, said control device first inlet supplying a pressure signal to one face of said piston, said control device second inlet supplying pressure to the other face of said piston, and biasing means for biasing said piston with a force additive of the force on said one face of said piston.

8. The sliding vane rotary compressor of claim 7 including delay means associated with said piston, said delay means being operative to delay operation of said first and second control signals by reason of movement of said piston until said piston has traveled a predetermined distance.

9. The sliding vane rotary compressor of claim 7 including a second bypass passageway having its inlet opening between said first mentioned bypass passageway inlet opening and said outlet passage opening in said compression chamber, said second bypass passageway having its outlet opening connected to said inlet passage, said second bypass passageway inlet opening being spaced from said first bypass passageway inlet opening a distance less than the distance between the tips of adjacent vanes forming a compression pocket, and second valve means in said second bypass passageway responsive to the pressure within an individual compression pocket adjacent to said second bypass passageway inlet opening and additionally responsive to a separate control signal, said second valve means being operative to be opened when the pressure in the individual compression pocket adjacent to said second

bypass passageway inlet opening exceeds a predetermined value.

10. The sliding vane rotary compressor of claim 9 wherein said pressure differential responsive means is operative to provide a third and fourth control signal, said third control signal being operative to cause said second valve means to open said second bypass passageway, and said fourth control signal being operative to cause said second valve means to closed said second bypass passageway.

11. The sliding vane rotary compressor of claim 10 wherein said piston is normally biased to a position wherein said first and third control signal are operative, said piston being operative, when said pressure at said outlet passage increases with respect to the pressure at said inlet passage to provide pressure against the other side of said piston to cause said piston to move towards said first control device inlet to cause actuation sequentially of said fourth and second control signals.

12. The sliding vane rotary compressor of claim 1 including a second bypass passageway having its inlet opening between said inlet and outlet passage openings in said compression chamber, said second bypass passageway having its outlet opening connected to said inlet passage, said bypass passageway being formed in a portion of one end wall in sliding engagement with the side edges of said vanes, said second bypass passageway being formed in a portion of said second end wall in sliding engagement with the other side edges of said vanes, said second bypass passageway inlet opening being radially spaced from said first bypass passageway inlet opening with respect to the axis of the rotor, and second valve means in said second bypass passageway, said second valve mean being responsive to the pressure within an individual pressure pocket adjacent to said second bypass passageway inlet opening and additionally responsive to a separate control signal, said second valve means being operative to be opened when the pressure in the individual compression pocket adjacent to said second bypass passageway inlet opening exceeds a predetermined value.

13. The sliding vane rotary compressor of claim 1 wherein said casing includes the third end wall parallel to said first and second end walls; said first end wall being spaced between said second and third end walls to define between said first and second end walls a first compression stage and between said first and third end walls a second compression stage; said outlet passage extending through said first end wall and being connected to a second stage inlet passageway; said rotor, said first and third end walls, and the walls of said cavity forming at least one compression chamber in said second stage; said rotor having vanes slidably carried on the outer surface thereof to engage the walls of said cavity in said second pressure stage, said inlet passage opening into said second stage compression chamber at the portion of maximum volume of said compression chamber, an outlet passage through said third end wall to said second stage compression chamber at the point of minimum volume of said compression chamber, said rotor having a constant outer diameter along the length thereof and extending beyond the ends of said first and second compression stages into recesses formed in said second and third end walls, and rotor sealing means operative to seal said rotor during rotation thereof in cooperation with said first, second and third end walls.

14. The sliding vane rotary compressor of claim 13 wherein said vanes in said first and second compression stages are positioned within radially axially extending grooves in the surface of said rotor, each of said grooves being slightly axially longer than the vane slidably positioned therein to compensate for variations in the length of said rotor.

15. A sliding vane rotary vane compressor comprising:  
(a) a casing having an elongated cavity formed therein,

(b) said casing including end walls on opposite ends thereof,

(c) a rotor positioned within the cavity and rotatably mounted in the end walls, said rotor having axially extending radially positioned vanes slidably carried in the outer surface thereof to engage the walls of said cavity,

(d) said end walls, said rotor, and walls of cavity forming at least one compression chamber in said cavity, said sliding vanes extending radially outwardly from the axis of said rotor to form compression pockets between adjacent rotor vanes,

(e) an inlet passage in said casing having an opening into said compression chamber at the portion of maximum volume of said chamber,

(f) an outlet passage in said casing having an opening into said compression chamber at the portion of minimum volume of said compression chamber,

(g) said compression chamber decreasing in volume from the inlet passage opening to the outlet passage opening,

(h) a first unloader assembly adapted to selectively unload compression pockets adjacent said inlet opening in accordance with a control signal,

(i) a control signal device, said control signal device being operative to transmit either a first signal to said first unloader assembly to maintain the loaded condition in said compression pockets adjacent said first unloader assembly or a second control signal to said first unloader assembly to unload compression pockets adjacent said first unloader assembly,

(j) said control signal device being responsive to the difference in pressures between the pressure at said inlet passage and the pressure at said outlet passage, said first control signal being transmitted when said pressure differential is above a predetermined amount, said second control signal being transmitted when said pressure differential drops below said predetermined value.

16. The sliding vane rotary compressor of claim 15 including a second unloader assembly adjacent to said first unloader assembly to further unload said compression chamber, said second unloader assembly being operative from a third control signal transmitted by said control device to maintain compression pockets adjacent thereto in a loaded condition and further operative from a fourth control signal transmitted by said control device to unload compression pockets adjacent thereto, said control signal device being operative to transmit said third control signal when said pressure differential is above a predetermined value and to transmit said fourth control signal when said pressure differential drops below said predetermined value, said fourth control signal being operative only after said first unloader assembly has been unloaded.

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