July 12, 1966 R. F. WILLIAMS ET AL 3,260,444

COMPRESSOR CONTROL SYSTEM

Filed March 30, 1964 3 Sheets-Sheet 1

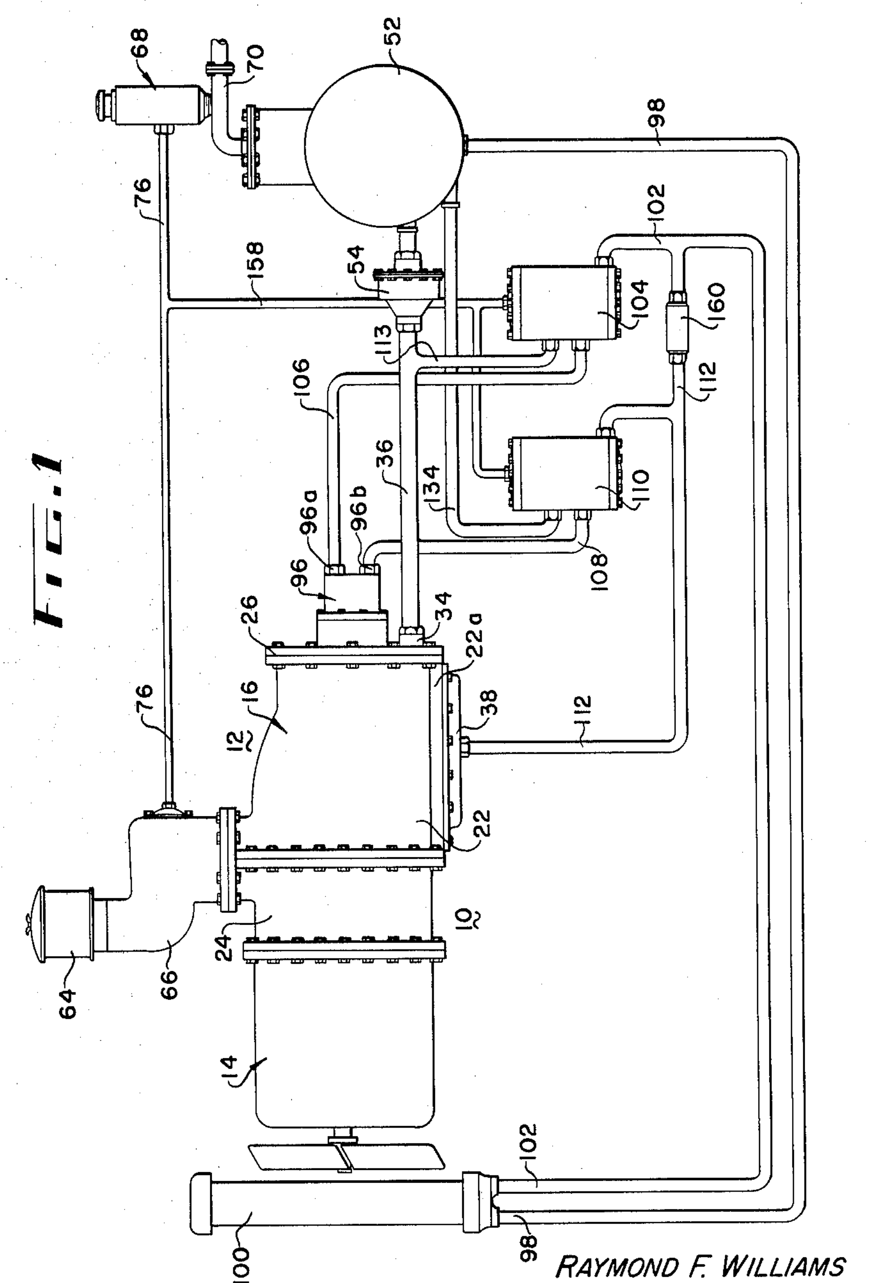

FIG. 1

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COMPRESSOR CONTROL SYSTEM

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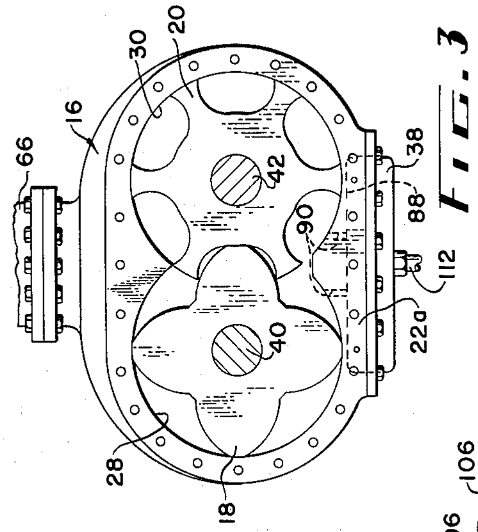

FIG. 3

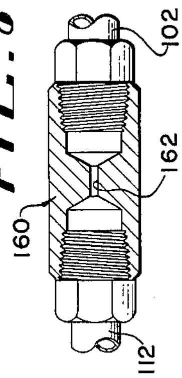

FIG. 8

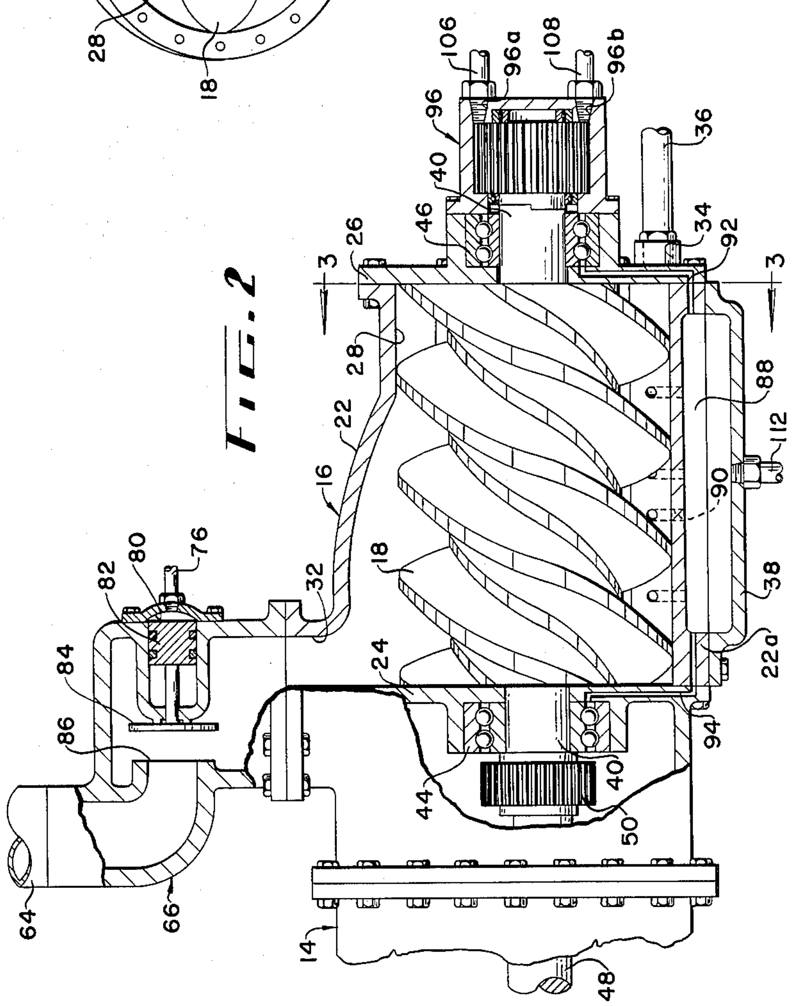

FIG. 2

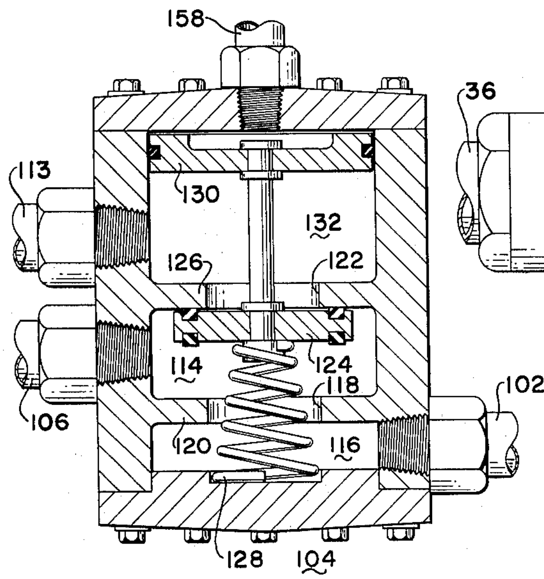

FIG. 6

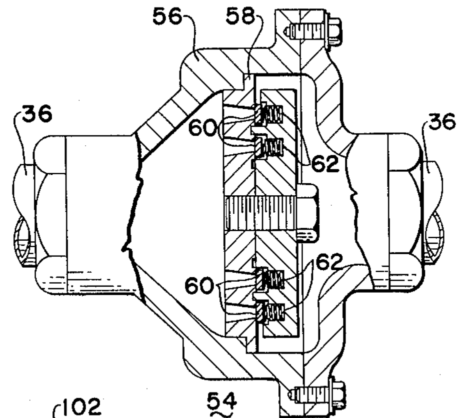

FIG. 4

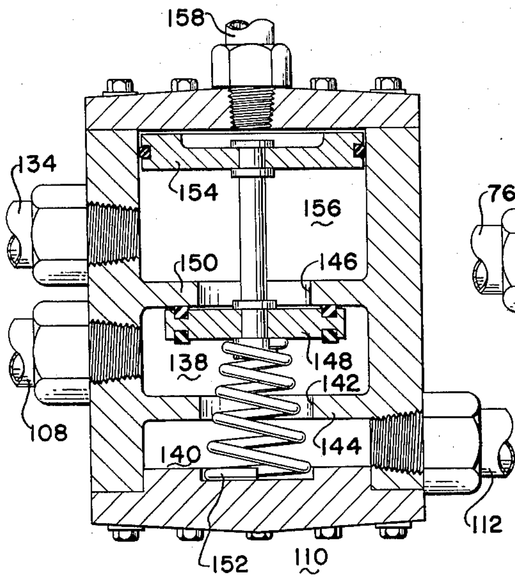

FIG. 7

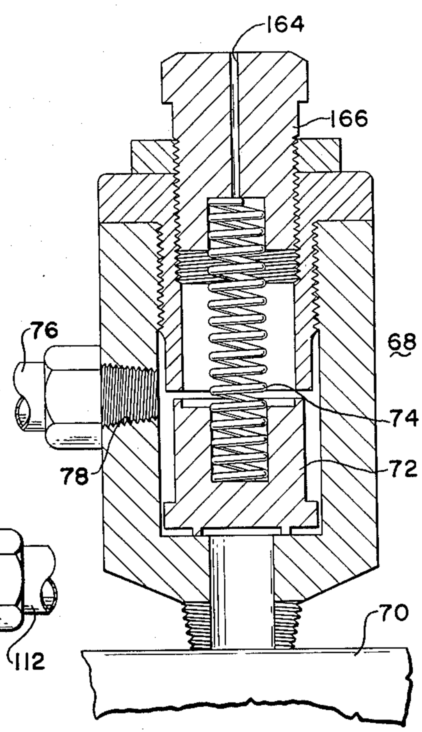

FIG. 5

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COMPRESSOR CONTROL SYSTEM
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Filed Mar. 30, 1964, Ser. No. 355,658
6 Claims. (Cl. 230—138)

This invention generally concerns control means for rotary gas compressors and more particularly concerns an improved unloading system for compressors of the aforesaid type whereby the present cost of running a compressor unloaded may be substantially reduced.

In accordance with conventional practice, positive displacement gas compressors of the various rotary types are unloaded by closing the compressor inlet opening thereby shutting off the flow of gas into the compression chambers of the compressor. The efficiency of this method for reducing unloaded power consumption and attendant operating costs is inherently low due to cyclic expansion and recompression of trapped gas remaining inside the compressor system after closure of the inlet port. Thus, with the inlet port closed and the compressor running, the trapped gas inside the compressor system exhibits a pressure gradient increasing from inlet pressure to discharge pressure and such a gradient in pressure causes the trapped gas to circulate from the compressor discharge back toward the compressor inlet through internal leakage paths within the compressor. The trapped gas which leaks or circulates back toward the compressor inlet expands to a lower pressure approaching inlet pressure and is thereafter recompressed to discharge pressure only to again leak back toward the compressor inlet. The work performed in recompressing such internally circulating gas is wasted; and, even though rotary compressors display several well-known advantages over reciprocating piston compressors, the latter type of compressor has attained general acceptance as a means for supplying gas under conditions of fluctuating demand in factories, shops and the like where power costs are of paramount concern to users. This user preference is understandable since the gas inlet valve of piston compressors is usually conditioned to vent the compression chamber during unloaded operation so that power is not consumed in recompressing trapped gas in the compression chamber. This difference in unloading methods is believed to account for the fact that the power consumed by a prime mover in driving an unloaded piston compressor is substantially less than that consumed in driving an unloaded rotary compressor of the same capacity.

Therefore, the principal object of this invention is to provide means for unloading a positive displacement rotary compressor so that the power consumed during unloaded operation can be reduced sufficiently to render such compressors superior, or at least competitive, in this regard to reciprocating piston compressors. For the attainment of the above generally stated object and other more detailed objects which will herein after appear, this invention contemplates the provision of an improved compressor control system, effective upon compressor output demand being met, for reducing, or ideally eliminating, the pressure gradient between the inlet and the discharge of a rotary compressor thereby eliminating or substantially reducing the power consumed by recompression of internally circulating gas.

Another object is to eliminate or reduce the aforementioned pressure gradient by evacuating trapped gas from the interior of the compressor.

Another object is to effect evacuation of the compressor as aforesaid by means of a pump device which is driven by the compressor prime mover.

Yet another object is to evacuate the compressor as

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aforesaid by a pump device which normally functions during loaded operation to supply liquid under pressure to the interior of the compressor to seal and cool the same.

Still another object is to provide a compressor unloading system of the aforedescribed type which is characterized by efficiency and economy of operation and by simplicity and low cost of construction.

These and other objects and advantages will become apparent upon reading the following description and claims in connection with the attached drawings in which:

FIG. 1 is a schematic illustration of a gas compressing system embodying the present invention;

FIG. 2 is an enlarged fragmentary view of the motor-compressor unit shown in FIG. 1 having a portion of the casing broken away;

FIG. 3 is a view taken along lines 3—3 of FIG. 2;

FIG. 4 is an enlarged view of a check valve shown in FIG. 1 having a portion of the valve housing broken away;

FIG. 5 is an enlarged sectional view of a pilot valve shown in FIG. 1;

FIG. 6 is an enlarged sectional view of a control valve shown in FIG. 1;

FIG. 7 is a view similar to FIG. 6 of another control valve shown in FIG. 1; and

FIG. 8 is an enlarged sectional view of a choke device shown in FIG. 1.

FIG. 1 of the drawings illustrates a gas compressor control system which, in accordance with this invention, is adapted to load and unload a motor-compressor unit **10** which comprises a screw type rotary compressor **12** driven by an electric motor **14**. As shown in FIGS. 2 and 3, the illustrative compressor **12** comprises a stationary housing, indicated in its entirety by numeral **16**, and a pair of complementary, meshing rotors **18** and **20** which are rotatively journalled within the housing. The plural-part housing **16** comprises a central cylinder or casing **22** which is closed at opposite ends by removable end walls or heads **24** and **26**. The casing and the heads are provided with abutting flanges and suitable fasteners for securing the parts in place; and, a gear pump **96** is removably secured to the head **26**. The central casing member **22** is provided with intersecting parallel bores **28** and **30** which are in open communication with an inlet port **32**, opening substantially radially through casing **22** and the inlet-end head **24**, and with a discharge port **34** opening substantially axially through the head **26** to a gas discharge pipe **36**. The base portion **22*a*** of casing **22** is provided with certain cavities and passageways, to be hereinafter described, and is closed by a removable base cover plate **38**.

As best illustrated in FIGS. 2 and 3, the compressor operating means comprise complementary helically threaded rotors **18** and **20** which are rotatively journalled in the parallel bores **28** and **30**. Rotor **18** has four generally convex lobes and is commonly denominated the main rotor; and, rotor **20** has six generally concave grooves and is commonly denominated the gate rotor. The rotors are provided with reduced diameter shaft extensions **40** and **42** which are formed integrally and coaxially with the helical body portions of the respective rotors. The extending rotor shafts **40** and **42** are rotatively journalled at their opposite ends by antifriction bearing assemblies mounted in the heads **24** and **26**. A bearing assembly **44** is mounted in the inlet head **24** to support and journal the inlet end of the main rotor shaft **40**; and, a bearing assembly **46** is mounted in the discharge head **26** to support and journal the discharge end of the main rotor shaft in a similar manner. The main rotor shaft **40** extends through the inlet head **24** into driving engagement with the drive shaft **48** of the electric motor **14**. Preferably the cooperating rotors are synchronized or timed to provide interrotor clearance by means of suitable gearing 50 mounted on the inlet ends of the rotor shafts 40 and 42.

In the illustrative compressor, a plurality of compression chambers are formed by the mating main rotor lobes and gate rotor grooves as the rotors 18 and 20 rotate in opposite directions and in approximate contact with the walls of the casing bores 28 and 30. As the rotors rotate, the compression chambers sequentially open to their full volume and fill with gas while in communication with the inlet port 32. Thereafter, due to entrance of the main rotor lobes into mating gate rotor grooves, the volume of gas contained in the compression chambers is progressively reduced by the shortening of the length of each chamber from its inlet end to its discharge end; and, with a decrease in volume, the internal pressure in each chamber is progressively raised from inlet pressure to discharge pressure. Thus, the compression chambers are axially displaced toward the discharge port 34 and are sequentially brought into gas-delivering registration with the discharge port. The discharge gas is conducted from the compressor discharge port 34 to a receiver tank 52 by means of the discharge pipe 36, the latter having a check valve 54 therein to prevent the backward flow of receiver gas to the compressor when the compressor is unloaded or is stopped. As best seen in FIG. 4, the check valve 54 is conventionally constructed and generally comprises a two-part housing 56; a plural port seat 58, and closure members 60 which are biased to a seated or closed condition by springs 62. It will be understood that the bias of the springs 62 is overcome by the pressure of the compressor discharge gas operating on the closure members 60.

As the rotors 18 and 20 rotate, gas is drawn into the interior of the compressor housing 16 through a gas filtering device 64 and an inlet valve assembly 66; and, the gas is thereafter compressed and discharged to the receiver tank 52 in the manner hereinbefore described. In a conventional manner, the flow of gas into the inlet port 32 of compressor 12 may be interrupted by the inlet valve 66 when the gas pressure in the receiver tank 52 reaches a rated value. To this end, a pressure sensitive pilot valve 68 is connected between a final delivery pipe 70 and the inlet valve 66; and, upon receiver pressure having reached rated value, a pilot valve closure member 72 will be unseated against the adjustable bias of a spring 74 thereby permitting pilot gas to flow from the delivery pipe 70 through a control line 76 communicating between the pilot valve outlet 78 and a piston chamber 80 of inlet valve 66. A fluid-actuated piston member 82 and an attached closure member 84 of the inlet valve 66 are urged to the position shown in FIG. 2 by the pressure created by the flow of intake gas through a valve seat 86 when the compressor 12 is running loaded; however, when pilot gas is communicated to the piston chamber 80 as aforedescribed, the piston 82 and valve closure member 84 will be forcibly shifted to seat the latter on valve seat 86 thereby placing the inlet valve 66 in a closed condition.

Although a screw-type compressor has been described in detail and this type of compressor is particularly well adapted to be loaded and unloaded by the hereinafter described control system, it will be apparent to those skilled in the art that the present invention can be employed to similar advantage with other compressors such as the vane type or eccentric rotor type, for example.

In compressor apparatus of the type presently under consideration, a suitable liquid medium may be supplied under pressure to a gallery 88 relieved in the casing base 22*a* for injection into the interior of the housing 16 through a plurality of spaced apertures 90 for the purpose of sealing clearances between the meshing rotors and between the rotors and the housing 16 and for removing the heat of compression generated during loaded operation. If the selected liquid medium has good lubricating properties, it may be desirable to employ the sealing and cooling liquid for lubricating certain moving parts of the compressor. In the exemplary compressor construction, oil is employed and, as shown by FIG. 2, the discharge end of gallery 88 communicates with the aforementioned bearing assembly 46 by means of a passageway 92 defined in the casing base 22*a* and the discharge head 26. Likewise, oil from gallery 88 is supplied to the main rotor bearing assembly 44 through a passageway 94 defined in the casing base 22*a* and the inlet head 24. The lubricating system for the gate rotor bearing assemblies, not shown, is substantially identical in structure and operation to that described above in connection with the main rotor bearing assemblies 44 and 46. If the sealing and cooling liquid should comprise water, for example, it would, of course, be desirable to provide a separate lubricating system. When the compressor is running loaded, the oil which is injected from gallery 88 into the housing 16 is entrained in the dischrage gas and is expelled therewith through the discharge pipe 36 and the check valve 54 into the receiver tank 52. In a well understood manner, the oil separates from the gas and collects in a bottom or sum portion of the tank 52 while the gas discharges from the top of the tank through the air delivery pipe 70. A positive displacement gear pump 96 of conventional construction and suitable capacity is drivably connected directly to the discharge end of the main rotor shaft 40 and is provided with an inlet 96*a* and an outlet 96*b*. During loaded operation of compressor 12, the gear pump 96 draws separated oil from the bottom or sump portion of tank 52 through a pipe 98, a radiator type heat exchanger 100, a pipe 102, a pump inlet valve 104, a pipe 106 and pump inlet 96*a*. The oil pump discharge is communicated to the gallery 88 through pump outlet 96*b*, a pipe 108, a pump discharge valve 110 and a pipe 112. The controlling functions and construction of valves 104 and 110 will be hereinafter more fully described.

As thus far described, the compressor unloading system is generally conventional in structure and operation and, without more, would display the aforedescribed disadvantages occasioned by trapping gas within the housing 16 and discharge pipe 36 and thereafter cyclically recirculating and recompressing the same. However, as noted above, an object of this invention is to reduce or eliminate such recirculation and recompression created by the undesirable gas pressure gradient between the inlet and discharge of compressor 12 by evacuating the trapped gas from the interior of the compressor housing and discharge pipe 36. In accordance with this feature of the invention, a pumping device is employed for this purpose; and, by means of a novel arrangement of valves and pipes, the aforedescribed oil circulating pump 72 is advantageously connectable in circuit with the compressor discharge opening 34 to evacuate gas and oil from the interior of the compressor 12. Generally, this aspect of the invention is accomplished by means of the aforementioned pump control valves 104 and 110. Control valve 104 is operable to connect the pump inlet 96*a* and pipe 106 interchangeably to pipe 102 leading to tank 52 or to a pipe 113 which joins with the discharge pipe 36 at a point between the compressor discharge port 34 and the check valve 54. With compressor 12 running loaded, pipe 106 is in communication with pipe 102 by means of valve chambers 114 and 116 which are in open communication through a port 118 in an internal wall 120. Another port 122 opening from chamber 114 is closed by a valve closure member 124 which seats on an internal wall 126 under the bias of a spring 128. In a manner to be described, the closure member 124 is shiftable by an attached valve-actuating piston 130 to close port 118 and to open port 122 thereby communicating pipes 106 and 113 through chamber 114, port 122 and a chamber 132. Control valve 110 is operable to connect the pump outlet 96*b* and pipe 108 interchangeably to pipe 112 leading to the gallery 88 or to a pipe 134 leading to the gas receiver-oil sump tank 52. With the compressor running loaded, pipe

108 is in communication with pipe 112 by means of valve chambers 138 and 140 which are in open communication through a port 142 in an internal wall 144. Another port 146 opening from chamber 138 is closed by a valve closure member 148 which seats on an internal wall 150 under the bias of a spring 152. In a manner to be described, the closure member 148 is shiftable by an attached valve-actuating piston 154 to close port 142 and to open port 146 thereby communicating pipes 108 and 134 through chamber 138, port 146 and a chamber 156. The upper portions of control valve-chambers 132 and 156 comprise cylinders for pistons 130 and 154, respectively; and, the outlet 78 of pilot valve 68 is connected to these cylinders by branches of a control line 158 which communicates with control line 76.

The following is a brief description of the operation of the aforedescribed control system. With the electric motor 14 operating to drive the compressor 12 for loaded operation, the inlet valve 66 and the pump control valves 104 and 110 will be in the condition shown in the drawings. Thus the closure member 84 of the inlet valve 66 will be open thereby permitting the rotors 18 and 20 to draw gas into the housing 16 and to compress the gas to a final discharge pressure which may conveniently be assumed to be 100 p.s.i.g. The discharge gas enters the receiver tank 52 through pipe 36 and check valve 54, and, after entrained oil is separated therefrom, the gas is delivered through the pipe 70 to any desired gas supply system. The oil pump 96 is driven by the main rotor shaft 40 and has its inlet 96*a* connected to the bottom sump portion of tank 52 by means of pipe 106, control valve 104, pipe 102, oil cooler 100, and pipe 98. The pump outlet 96*b* is connected to the gallery 88 by pipe 108, control valve 110 and pipe 112; and, by this means oil is supplied under pump discharge pressure to the rotor bearings and to the compression chambers of the compressor. When the demand for receiver gas decreases for one reason or another, the gas pressure in tank 52 and delivery pipe 70 will increase to rated output pressure of 100 p.s.i.g. making it possible to unload the compressor to reduce the power consumed by the motor 14. In some applications of this invention, such as in an engine-driven portable compressor, it may be feasible to decrease the speed of operation of the prime mover as an incident to unloading the compressor; however, it will be assumed that the illustrative electric motor 14 is of the constant speed type and that the compressor 12 is operated at constant speed to meet a fluctuating demand. An increase in receiver pressure will be sensed by the pilot valve 68; and, when receiver pressure reaches 100 p.s.i.g. or above, valve-actuating gas will be passed by the pilot valve and communicated simultaneously to the inlet valve 66 and to the pump control valves 104 and 110. The pilot gas communicated through control line 76 to the piston chambre 80 of the inlet valve 66 will forcibly shift the piston 82 and the valve closure member 84 to the left, as viewed in FIG. 2, thereby seating the closure member upon seat 86 to shut off the supply of gas being drawn into the compressor inlet port 32. Obviously, a quantity of gas will be trapped within the compressor housing 16 and the discharge pipe 36 when the inlet valve 66 closes; and, such trapped gas will be compressed to compressor discharge pressure of approximately 100 p.s.i.g. by the action of rotors 18 and 20. At the inlet end of the housing 16 the compression chambers defined by the rotors continue to open periodically to their full volumes and in doing so create a suction or vacuum. Thus, it will be appreciated that, with the inlet valve 66 closed and with the rotors revolving, the trapped gas within the housing 16 would conventionally be subjected to a pressure gradient ranging from approximately 100 p.s.i.g. at the compressor discharge to approximately zero p.s.i.g. or less at the compressor inlet. Such a pressure differential of approximately 100 pounds would be sufficient to cause the trapped gas to flow back from compressor discharge toward compressor inlet through internal clearances between the rotors 18 and 20, between the walls of bores 28 and 30 and the rotors, and between the ends of the rotors and the housing end walls 24 and 26 As this leakage gas flows toward the compressor inlet to points of diminishing pressure, it expands and again fills the compression chambers toward the low pressure end of the rotor screws. The expanded gas is then recompressed to discharge pressure. Thus, the pressure gradient between the opposite ends of the compressor 12 causes trapped gas to circulate internally and to be cyclically compressed and expanded with a consequent waste of the power expended in recompressing the trapped gas.

In accordance with this invention, the aforedescribed oil pump 96 is employed to draw gas from the compressor discharge port 34 to reduce substantially or to eliminate the undesirable pressure gradient within the compressor housing. Thus, when rated delivery pressure is reached in the receiver tank 52 and pilot valve 68 opens to actuate inlet valve 66, pilot gas is simultaneously supplied to pump control valves 104 and 110 through lines 76 and 158. The piston member 130 of valve 104 will be shifted downwardly against the bias of spring 128 to open port 122 and to close port 118. This valving action effectively disconnects the pump inlet 96*a* from the sump portion of tank 52 and connects the pump inlet 96*a* to the discharge port 34 of the compressor housing 16 by means of the pipe 113 and the discharge pipe 36. At the same time the piston member 154 of control valve 110 will be shifted downwardly against the bias of spring 152 to open port 146 and to close port 142 thereby disconnecting the pump discharge port 96*b* from pipe 112 and the gallery 88 and connecting the pump discharge port 96*b* to the receiver tank 52 by means of line 134.

Since the compressor 12 may run unloaded for substantial periods of time with the oil pump functioning as an evacuating pump, alternate means are provided to supply oil to the rotor bearings 44 and 46. To this end gas pressure acting on the oil in the sump portion of tank 52 will force oil through pipe 98, the heat exchanger 100, pipe 102, a choke 160 and pipe 112 to the gallery 88. Since the gas pressure in the tank is much greater than that in the gallery, the valving action of pump control valve 104 and 110 is employed to connect the choke 160 between pipes 102 and 112 thereby metering the quantity of oil which enters the gallery 88 to that quantity which will properly lubricate the bearing assemblies 44 and 46 without deleterious flooding of the interior of compressor housing 16. It will be appreciated that the reduction in oil flow into the compressor housing during unloaded operation will correspondingly reduce the power expended by the pump 96 in evacuating the oil from the housing 16. However, the reduced amount of oil which is evacuated through the pump 96 during unloaded operation is effective to seal the pump gears, thereby enabling the pump to function more efficiently in drawing gas from the compressor housing.

When the demand for receiver gas again increases and the delivery pressure within tank 52 and delivery pipe 70 decreases below the minimum rated delivery pressure, the pilot valve closure member 72 will seat, thereby interrupting the supply of pilot gas to the inlet valve 66 and the control valves 104 and 110 and causing the piston chambers of these valves to be vented to atmosphere through a passage 164 in the spring adjusting member 166 of the pilot valve 68. The valve 66 will then open to admit gas to the housing 16 thereby restoring the compressor 12 to its loaded condition; and, simultaneously, the control valves 104 and 110 will disconnect the pump 96 from the compressor discharge port 34.

From the foregoing, it will be understood that the valving action of valves 104 and 110 connects the pump 96 between the housing 16 and the tank 52 to effect evacuation of trapped gas from the interior of the compressor. Ideally the pump 96 should remove all trapped gas from the housing 16 and discharge pipe 36 and should create a vacuum therein so that the rotors **18** and **20** perform no work in compressing trapped gas; however, the degree to which this ideal may be achieved is limited in practice by the efficiency of the pump **96**. Where the invention has been carried out with a gear-type oil pump of conventional construction having standard clearances, the pressure at the discharge of a compressor having a rated discharge pressure of 100 p.s.i.g. has been reduced to approximately 1.5 p.s.i.g. Such a reduction in discharge pressure and in the pressure gradient within the compressor housing has been shown to be effective to reduce the unloaded power consumption to approximately one-third of what would normally be expected from the use of known systems for unloading rotary compressors. This unexpected reduction in power required to run an unloaded rotary compressor renders rotary compressors more efficient in this regard than piston type compressors. Therefore, as a result of this invention, users may for the first time enjoy the many advantages provided by rotary compressors and yet operate the same at costs which are lower than the operating costs of a piston compressor.

While the aforedescribed embodiment of the invention utilizes to particular advantage the availability of the oil pump **96** to evacuate the compressor housing **16**, the present invention in its broadest aspects contemplates evacuation of trapped air from a compressor housing by any effective means. For example where a compressor of the dry type is used, a suitable auxiliary pump could be driven directly by the compressor prime mover or by a rotary shaft of the compressor with the expenditure of very little power. Alternatively, such an evacuating pump could be driven by means independent of the compressor or its prime mover. Furthermore, it will also be understood that any readily available source of vacuum can be substituted for an independent evacuating pump. Moreover, it will be apparent to those skilled in the art of compressor control that various changes in construction, proportion and arrangement of the elements of the described control system may be made without sacrificing any of the advantages of the invention or departing from the scope of the appended claims.

Having fully described the invention, we claim:

1. A gas compressor having inlet and discharge openings; closure means for said inlet opening; pump means

connectable with the interior of said compressor for removing gas trapped therein upon closure of said inlet opening; a gas receiver; a source of liquid; and control means for said pump means operable to connect the same interchangeably between said source of liquid and the interior of said compressor and between the interior of said compressor and said gas receiver.

2. The combination according to claim **1** wherein said control means comprises first fluid conducting means operable to connect the inlet of said pump interchangeably to said source of liquid and to said compressor discharge opening; and second fluid conducting means operable to connect the outlet of said pump interchangeably to the interior of the compressor and to said gas receiver.

3. The combination according to claim **2** wherein said first and second fluid conducting means include fluid-actuated valves having valve actuating members responsive to changes in pressure in said gas receiver.

4. The combination according to claim **3** together with pressure sensitive valve means communicating with said gas receiver and operable to supply actuating fluid to said fluid-actuated valves.

5. The combination according to claim **1** together with conduit means communicating said source of liquid to the interior of the compressor when said pump is connected to the interior of the compressor.

6. The combination according to claim **5** wherein said conduit means includes a choke for restricting fluid flow therethrough.

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