Dec. 2, 1941. J. C. BUCKBEE 2,264,616

ROTARY COMPRESSOR

Filed Sept. 21, 1938 3 Sheets-Sheet 1

Fig. 1

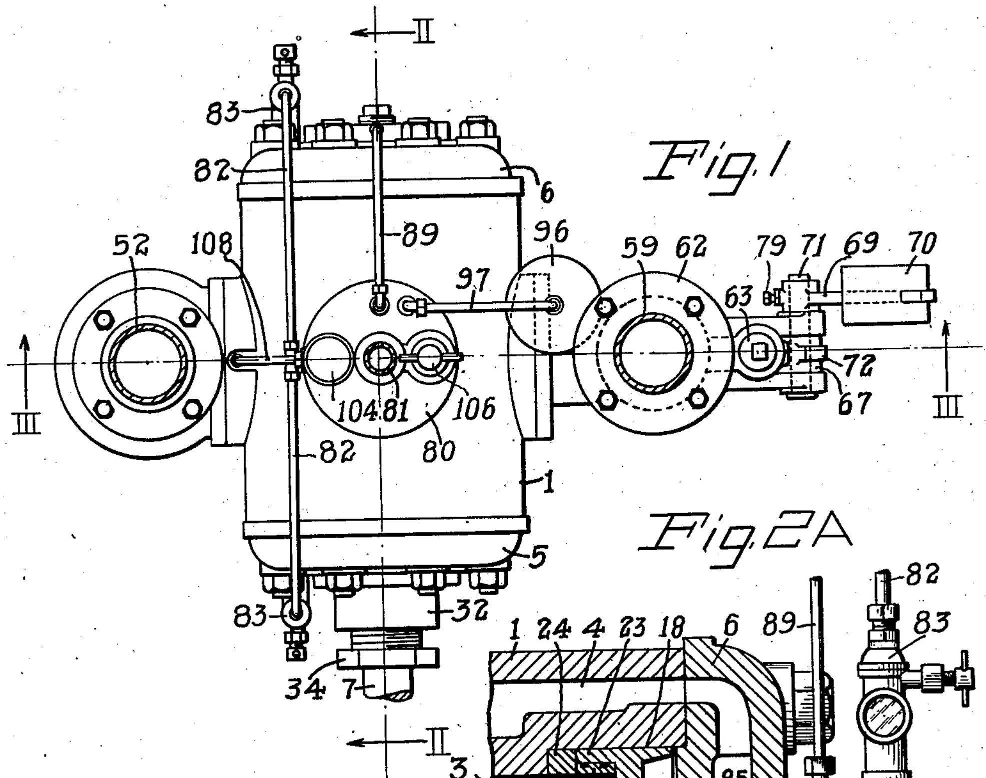


Fig. 2A

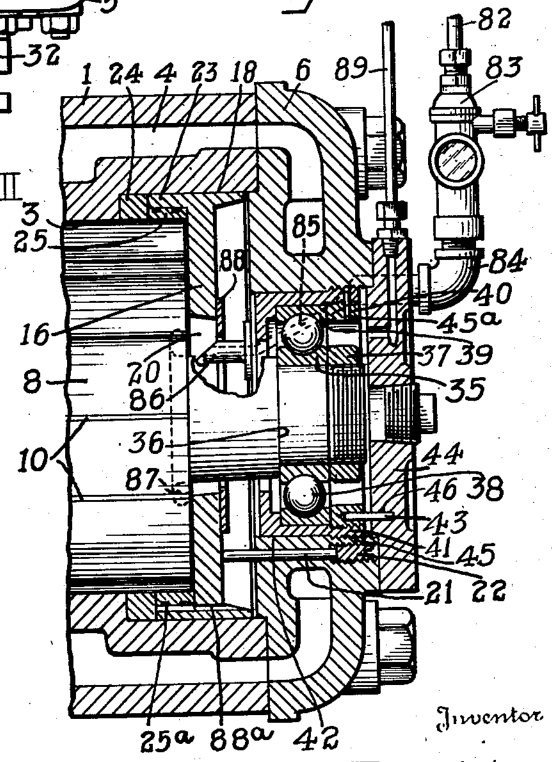
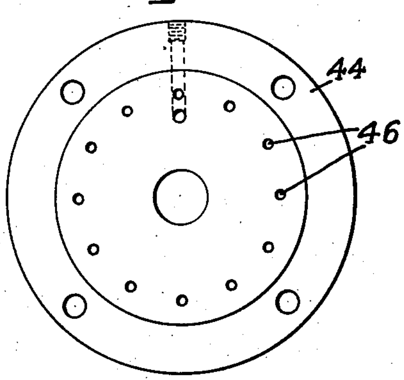


Fig. 4



Inventor

John C. Buckbee

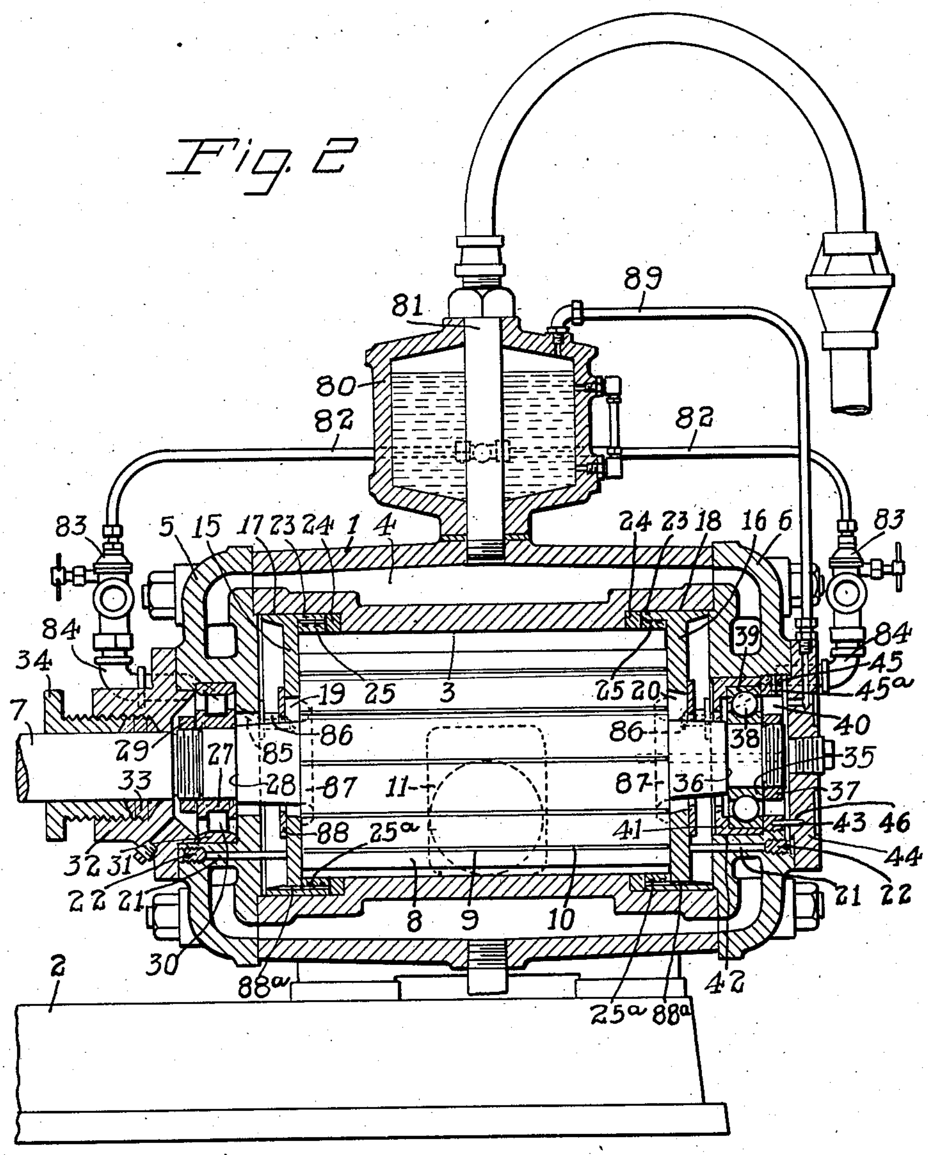
By Lyon & Lyon Attorneys

Dec. 2, 1941. J. C. BUCKBEE 2,264,616

ROTARY COMPRESSOR

Filed Sept. 21, 1938 3 Sheets-Sheet 2

Fig. 2



Inventor
John C. Buckbee

By Lyon & Lyon Attorneys

Dec. 2, 1941. J. C. BUCKBEE 2,264,616

ROTARY COMPRESSOR

Filed Sept. 21, 1938 3 Sheets-Sheet 3

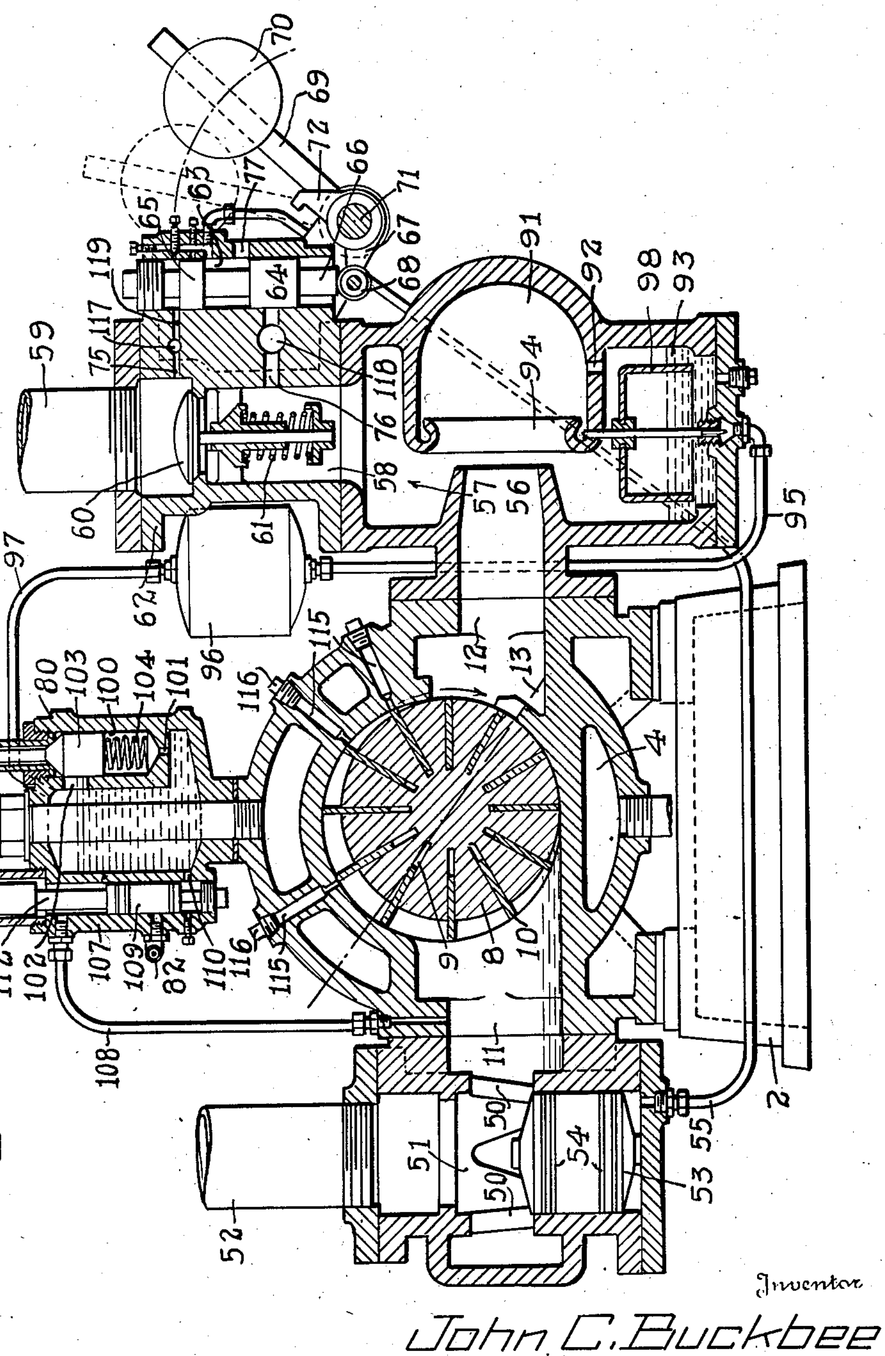


Fig. 3

Inventor

John C. Buckbee

By Lyon & Lyon

Attorneys

Patented Dec. 2, 1941

2,264,616

UNITED STATES PATENT OFFICE

2,264,616

ROTARY COMPRESSOR

John C. Buckbee, Los Angeles, Calif.

Application September 21, 1938, Serial No. 230,985

12 Claims. (Cl. 230—152)

This invention relates to rotary pumps and engines of the type in which a cylindrical rotor having radially slidable vanes is eccentrically mounted in a cylindrical casing. The particular design disclosed herein is adapted for use as an air pump but it will be understood that various features of the invention may also be employed in rotary engines and motors.

A broad object of the invention is to provide improved lubrication in a rotary machine of the type referred to.

Another object is to reduce vibration and promote smooth operation of such devices.

Another object is to provide a practicable rotary engine or pump structure of the vane type in which an effective fluid seal between the relatively movable parts can be readily produced and maintained.

Another object is to promote the efficiency of a rotary pump operating under varying load conditions.

The manner in which the foregoing objects are achieved will be explained by describing in detail, with reference to the drawings, the construction and operation of an air pump in accordance with the invention, it being understood that various changes and adaptations of the particular structure disclosed may be made without departing from the invention.

In the drawings:

Fig. 1 is a partial plan view showing in general exterior aspect an air pump constructed in accordance with the invention;

Fig. 2 is a vertical longitudinal section through the pump, the section being taken in the plane II—II of Fig. 1;

Fig. 2—A is a sectional view similar to Fig. 2, but on a larger scale and showing only one end portion of the pump;

Fig. 3 is a cross section through the pump taken in the plane III—III of Fig. 1; and

Fig. 4 is a detail inside face view of an adjusting end plate of the unit.

Referring to the drawings, the air pump therein disclosed comprises a casing member 1 of roughly cylindrical shape, which is stationarily supported on a suitable base 2 and is provided with an inner cylindrical surface 3 constituting one of the walls of a compression chamber. For convenience the member 1 will hereinafter be referred to as the stator. The stator 1 may be provided with a water passage 4 for cooling purposes.

The opposite ends of the stator 1 are closed by heads 5 and 6, respectively, which may be bolted to the stator 1 and support bearing members, which in turn rotatably support a shaft 7 on which a cylindrical rotor 8 is mounted, the rotor being positioned eccentrically within the casing 3. The heads 5 and 6 may also have passages 4 therein for the circulation of cooling liquid.

The rotor 8 is of smaller diameter than the inner surface 3 of stator 1 and is positioned eccentrically within the stator so that its periphery approaches very close to the inner surface 3 of the stator 1 at one point so as to provide for effecting a fluid seal between the rotor and casing along the line where they most nearly approach. The rotor 8 is provided with a plurality of radial slots 9 extending from end to end of the rotor, in which slots there are slidably mounted vanes 10. In operation, the rotor is rotated at relatively high velocity so that the vanes 10 are urged outward toward the inner surface 3 of the stator by centrifugal force. The stator 1 is provided with an inlet port 11 on one side and an outlet port 12 on the other side so that when the rotor is rotated in clockwise direction (referring to Fig. 3), the air or other gas to be pumped, which enters through the inlet port 11, is trapped in the pocket created between the vanes 10 and between the surface of the rotor and the surface 3 of the casing, and is carried around with the rotor and compressed as the rotor surface approaches the casing surface, and finally discharged through the port 12.

Insofar as the general structure so far described is concerned, it is in accordance with the general well-known principles of rotary pump design.

A common defect of older rotary pumps of the type described is that they tend to vibrate due to the fact that under varying conditions of load, varying radial forces are applied in different directions to the rotor. Thus under conditions of no load the chief force acting upon the rotor is gravity, which of course exerts a downwardly directed force. On the other hand when a pump of older design of the type described is working, new forces are developed which act radially on the rotor in directions other than downward, so that they oppose the force of gravity, some times exceeding it, and at other times being less than it. These varying radial forces tend to shift the rotor and shaft, throwing undesirable strain on the bearings and often causing severe vibration.

In accordance with the present invention, I greatly reduce the vibration resulting from the causes referred to by fixing the plane of eccentricity of the rotor and casing (as indicated by the line 13 in Fig. 3), at an angle to the horizontal, with the line of closest approach of the rotor to the casing closely adjacent to the lower part of outlet port 12. With this construction the forces resulting from the compression of gas between the upper surface of the rotor and the juxtaposed surface of the casing act in a general downwardly direction on the rotor, thereby holding the rotor shaft firmly in its bearings, avoiding vibration and chattering in the bearings, and giving a smooth, quiet-running machine.

The fact that the compression of the gas takes place on the upper side of the rotor also facilitates free entry of air or gas into and out of the machine, further reducing vibration of the machine by minimizing eddy currents of air or gas within the machine and contributing to its smooth operation and efficiency. Other advantages of the particular orientation of the rotor as shown will be discussed later in connection with the lubricating system of the machine.

In a machine of the type disclosed, it is desirable that the ends of the rotor and the vanes be liquid sealed with respect to the casing as well as that the outer edges of the vanes 10 be so sealed with the inner surface 3 of the casing. Heretofore, to the best of my knowledge, it has been the practice to carry the ends of the rotor out to contemplated running clearances with the heads of the casing, the actual clearance in operation being determined by the thickness of the gaskets placed between the ends of the cylindrical casing and the heads, and by the extent to which the bolts holding the heads in place were tightened. This was obviously a crude and uncertain method of construction where clearances measured in thousandths of an inch must be secured and maintained to secure efficiency.

In the machine disclosed, I permit of accurate adjustment of the clearance between the ends of the rotor 8 and vanes 10 and the abutting surfaces of the casing by providing separate inner head members 15 and 16, respectively, which are slidably mounted in counterbores 17 and 18, respectively, in the opposite ends of the casing member 3. These counterbores extend inwardly from the ends of the casing 3 substantially beyond and into overlapping relation with the ends of the rotor 8. The heads 15 and 16 are provided with apertures 19 and 20, respectively, for the passage of the rotor shaft 7 and for admitting lubricant to the vane slots 9, and are machined to closely approach the end faces of the rotor and the ends of the vanes 10 to provide for effecting a fluid seal therewith.

As previously indicated, the inner heads 15 and 16 are slidably mounted within the counterbores 17 and 18 and must therefore be retained in desired longitudinal positions with respect to the shoulders of the said counterbores and the rotor 8. I provide for accurate longitudinal adjustment of the inner heads 15 and 16 and the locking of the said heads against outward movement from the desired position of adjustment by means of several plungers 21 which are disposed at convenient circumferential intervals back of heads 15 and 16, and extend through longitudinal apertures provided therefor in the outer heads 5 and 6, respectively, into abutment with the outer faces of the inner heads 15 and 16. The outer ends of the apertures in the heads 5 and 6 which receive these plungers 21 are threaded to receive screws 22, the inner ends of which screws bear against the outer ends of the plungers 21. It will be obvious that by adjustment of screws 22 the heads 15 and 16 may be adjusted to desired longitudinal positions within said counterbores and maintained against outward movement from those positions.

To limit inward movement of the heads 15 and 16 from their positions of desired adjustment, each of the heads 15 and 16 is provided with a peripheral flange 23, the inner end face of which seats against an adjustment ring 24 which in turn seats against the shoulder at the end of the counterbore 17 or 18, as the case may be. By suitably choosing the longitudinal thickness of the rings 24 and adjusting the screws 22 to force the inner heads 15 and 16 up snugly against the rings 24, the desired accurate clearance between the ends of the rotor and the heads 15 and 16 may be readily obtained. The clearance necessary in any particular case will depend to a large extent upon the temperature variations in the pump, which cause varying expansion of the parts, and pumps constructed as described can be adapted for different conditions of operation involving different operating temperatures by merely changing the spacer rings 24.

It may be desirable in many instances to form the plungers 21 of a material having a greater rate of expansion for a given temperature rise than the stator material, so that after the machine has been started cold the increase in temperature during operation increases the pressure of the plungers 22 against the inner heads 15 and 16.

The adjustment rings 24 have their inner surfaces flush with the surface 3 of the stator. The spaces within the flanges 23 are provided with accurately machined floating rings 25 which receive the radial thrust of the vanes 10 due to centrifugal force, and rotate with the vanes. The floating rings 25 within the flanges 23 are bored slightly smaller in diameter than the stator surface 3 and the blades 10 are thereby held clear of the stator surface 2 when the machine is in operation, and wear between the vanes and the stator surface 3 avoided. Floating rings have been used before to limit outward movement of vanes in rotary machines of the type with which this invention is concerned, but to the best of my knowledge it has been the practice in the past to position such rings in counterbores in the stator some distance in from the stator ends, which construction requires the placing of stationary rings between the floating rings and the end heads to complete the stator bore from the floating rings out to the end heads. Such construction was objectionable in that it was expensive and in a measure uncertain as to the degree of clearances obtainable. By placing the floating rings 25 at the ends of the rotor 8 within the flanges 23 of the inner heads 15, I provide a very simple construction for obtaining very accurate and definite clearances. Thus it is merely necessary, prior to insertion of the heads 15 and 16 into the stator, to position the floating rings 25 within the flanges 23 and check the widths of the rings 25 by observing whether their end faces are clear of the end faces of the flanges 23. If they have the clearance required for successful operation, it is then known that when the inner heads and rings are assembled in the stator the longitudinal clearances at the edges of the rings will be correct.

In the pump as so far described, the inner end walls 15 and 16 are fitted to very close clearances with the ends of the rotor and vanes. However the interior end walls 15 and 16 are not intended to take any thrust that may be applied by the rotor, and I provide a readily adjustable bearing construction for accurately supporting the rotor against both radial and longitudinal displacement.

I prefer to employ antifriction bearings for supporting the rotor, and to facilitate adjustment and provide for longitudinal expansion of the rotor, I employ a longitudinally adjustable combined radial and thrust bearing at one end and a radial bearing only at the other end of the rotor.

Referring to Fig. 2, the radial bearing is positioned at the left end of the rotor 8 and is of the roller type; thus it comprises an inner race 27 mounted on the rotor and rigidly secured in position against a shoulder 28 on the shaft by a nut 29 threaded onto the shaft. The race 27 is provided with an annular recess, the sides of which bears against the ends of a plurality of cylindrical rollers 30 which may be confined in a cage in accordance with the usual practice. The rollers 30 bear against a stationary outer race 31 which is cylindrical on its inner surface so that the rollers 30 can move freely longitudinally with respect thereto. The race 31 is clamped in the outer head 5 of the pump by a closure member 32, which closure member also incorporates packing material 33 and a packing gland 34 for effecting a seal about the shaft 7. It will be observed that this bearing rotatably supports the shaft 7 against any radial movement but permits free longitudinal movement between the rollers 30 and the outer race 31.

The bearing at the other end of the rotor (the right end in Fig. 2) is adapted to resist both radial and longitudinal movement of the rotor while permitting free rotation thereof, and, as shown, is of the ball type. Thus it comprises an inner race 35 rigidly clamped on the shaft 7 against a shoulder 36 thereon by a nut 37. The outer surface of the inner race 35 is grooved to receive a ring of balls 38 which also fit in a juxtaposed groove in the inner wall of an outer race 39. By virtue of the grooves in the inner and outer races, the races are restrained against longitudinal as well as radial movement with respect to each other. The outer race 39 is rigidly secured by a threaded bushing 40 in a cage 41, which cage is provided with an outer cylindrical surface 42 ground to slidably fit in a cylindrical recess in the head 6. Adjacent its outer edge the cage 41 is provided with exterior threads 45 and cooperating threads are provided on the inner surface of the recess in the head 6 at its outer end. Therefore by rotating the cage 41 the cage and the ball bearing may be shifted longitudinally to adjust the rotor 8 to the desired position of clearance with respect to the inner heads 15 and 16, respectively.

To lock the cage 41 in a desired position of adjustment a pin 43 is provided in the bushing 40, this pin projecting longitudinally from the bushing into one of a plurality of recesses provided in an end plate 44 which closes the end of the head 6. It is to be understood that the bushing 40, after being screwed up tight against the outer race 39, is locked against further rotation with respect to the cage 41, as by passing a pin 45a through registering apertures in the cage and bushing. Hence the pin 43 restrains the bushing from rotation and the bushing in turn restrains the cage 41 from rotation.

Referring to Fig. 4, which is a view of the inner face of the closure plate 44, it will be observed that a plurality of apertures 46 are provided in the plate for reception of the pin 43 so that the cage 41 can be locked at such position as longitudinal adjustment of the rotor may require. Longitudinal adjustment of the rotor to the desired running position is effected by removing the cover plate 44 and rotating the cage 41 in one direction until the end of the rotor bears against one of the inner head members 15 or 16. The cage 41 is then rotated in the opposite direction until the rotor bears against the other inner head. Then either by micrometer gauge readings or through knowledge of the number of threads per inch on the cage and observing the angle through which the cake 41 has been turned in the above operation, the total clearance between the ends of the rotor and the faces of heads 15 and 16 may be readily determined as well as the relative position of heads 15 and 16 with respect to the rotor, and it is then merely a matter of again rotating cage 41 to move the rotor longitudinally until the desired running clearance at the rotor ends is obtained, after which cage 41 is locked through the medium of pin 43.

When a pump of the type described is rotated continuously to supply a varying demand, and the demand decreases, the pressure in the discharge port rises beyond the desired value and if the pump continues to work, power is needlessly wasted. I avoid this objectionable feature by providing a special valve system for cutting off the supply of air to the pump and venting the discharge side of the pump to the atmosphere whenever the pressure in the discharge port exceeds a desired value.

Thus referring to Fig. 3, the inlet passage 11, leading into the stator 1, communicates through a plurality of ports 50 with a central passage 51 which in turn is connected at its upper end to an intake pipe 52. A piston valve 53 is slidably fitted in the passage 51 and may be provided with piston rings 54 at its upper and lower edges for effecting a fluid-tight seal against the walls of the passage 51. When the valve 53 is in lowermost position, as shown in Fig. 3, its upper end is substantially flush with the lower edges of the ports 50 so that the pipe 52 is in free communication with the intake passage 11. However, when the valve 53 is in uppermost position in the passage 51, it is juxtaposed to the ports 50 and blocks the flow of air therethrough. The valve 53 normally remains in the lowermost position shown in Fig. 3, in which the ports 50 are open. When it is desired to close the ports 50 the valve 53 is raised to its uppermost position by introducing fluid under pressure to the passage 51 below valve 53. The fluid employed for lifting the valve 53 may be air under pressure from the discharge side of the pump, and is delivered through a pipe 55 under the control of a valve mechanism next to be described.

Referring to Fig. 3, the discharge passage 12 in the stator 1 communicates through an orifice 56 with a chamber 57 having a discharge passage 58 extending from the upper side thereof into communication with a discharge pipe 59. A check valve 60, which is urged into closed position by a spring 61, serves to cut the passage 58 off from the pipe 59 in response to any return flow of air. The passage 58 is incorporated in a block 62, which also has formed therein, to one side of the passage 58, a cylinder 63, which slidably contains a piston element 64. A second piston element 65 is attached to and spaced above the piston element 64, the two being formed integral with each other, and with a push rod 66 extending below the piston **64**. The pistons **64** and **65** are normally maintained in an upper position, as shown in Fig. 3, by a short arm **67** supporting a roller **68**, which bears against the lower end of push rod **66**. The short arm **67** is attached to a shaft **71** journaled in bearings in the block **62**, as shown in Fig. 1. The shaft **71** also has attached to it a long arm **69** carrying a weight **70**. A short stop arm **72** is attached to the shaft **71** for limiting downward movement of the short arm **67**.

A passage **75** connects the discharge pipe **59** permanently with the upper end of cylinder **63**, and a passage **76** permanently connects the passage **58** with the cylinder **63** at a point in the latter juxtaposed to the piston **64** when the latter is in uppermost position, as shown. A vent **77** communicates the central portion of the cylinder **63** with the atmosphere under normal conditions. A pipe **55** also communicates with the cylinder **63** at a point a short distance above the vent **77**.

Under normal conditions of pump operation, the pistons **64** and **65** occupy the positions shown in Fig. 3, and the arm **69** occupies the position shown in full lines in Fig. 3. Under these conditions pressure in the discharge pipe **59** is applied through passage **75** to the upper end of piston **65**, tending to force the piston down, against the force exerted thereon by the weight **70** acting through arm **69**, shaft **71** and the short arm **67**, and the piston **64** seals the outer end of the passage **76**. The condition described will obtain whenever the pressure in the discharge pipe **59** is insufficient to overcome the force of the weight **70**, and with the pistons **65** and **64** in the upper position shown, the passage **76** is sealed and the pipe **55** is connected to the vent **77** so that atmospheric pressure exists below the inlet valve **53**, permitting the latter to remain in its lowermost position and air to flow freely into the inlet **11**.

When the consumption of air supplied through the discharge pipe **59** is reduced so that the pressure rises sufficiently to force the pistons **65** and **64** downwardly, thereby swinging the arm **69** into the position shown in dotted lines in Fig. 3, the downward movement of the pistons carries the upper edge of the piston **65** below the passage communicating with the pipe **55**, and carries the upper edge of the piston **64** below the passage **76**. Therefore air under high pressure from the discharge pipe **59** flows through passage **75**, the upper end of the cylinder **63**, and into and through the pipe **55** to raise the intake valve **53** into uppermost position, in which it closes the ports **50**. This cuts off the supply of air to the intake passage **11** and the compressor ceases to pump air. At the same time the movement of the piston **64** communicates the passage **76** with the vent **77** so that any pressure in the outlet passages **12** and **56** and the chambers **57** and **58** is relieved. It is understood, of course, that as soon as flow of air from passage **58** to the discharge pipe **59** ceases, the check valve **60** closes to prevent return flow.

When the pressure in the discharge pipe **59** drops below an amount sufficient to overcome the force of the weight **70**, the weight **70** swings the arm **69** back into the full line position, raising the pistons **64** and **65** into uppermost position, in which the pipe **55** is vented to the atmosphere, and the passage **76** is closed. Thereupon the inlet valve **53** drops into lowermost position, opening the ports **50**, and the pumping operation is resumed.

It will be observed from Fig. 3 that the normal position of the long arm **69** extends upwardly and outwardly from shaft **71** at a relatively large angle from the vertical, but that in the dotted line position the arm **69** makes a much smaller angle with the vertical. Obviously the torque applied to the shaft **71** by the weight **70** is substantially greater when the arm **69** is in the full line position than when it is in the dotted line position. Hence a substantially greater pressure in the discharge pipe **59** is required to stop the pumping action than to resume it. This is desirable because it prevents unnecessarily frequent starting and stopping of the pumping action. Thus the above described controlling device may be adjusted to cut out the pumping action when the terminal air pressure in the discharge pipe **59** reaches, say, 35 pounds per square inch and not cut in again until the pressure falls to 30 pounds per square inch. Various desired differentials between the pressure required to stop the pumping action and that permitting resumption of the pumping action may be obtained by adjusting the radial position of the arm **69** on the shaft **71**. As shown in Fig. 1, the arm **69** is locked to the shaft **71** by a set screw **79**. By loosening the screw **79** the radial position of the arm **69** relative to the shaft **71** may be altered to vary the relative pressures at which the compressor cuts in and cuts out of service.

In a rotary compressor, particularly compressors operating at high speeds, it is essential to provide a positive supply of oil to the various working surfaces in order to reduce friction and in many parts to help maintain a seal.

The present invention incorporates a novel and particularly effective oiling system which will now be described.

Referring to Figs. 2 and 3, there is mounted on the upper side of the stator **1**, an oil reservoir **80** which, as shown, happens to be secured to the stator by a water pipe **81** which extends downwardly therethrough and communicates with the water passage **4** for cooling the compressor. However, there is no communication between the interior of the water pipe **81** and the reservoir **80**, the pipe **81** functioning only to retain the reservoir in position. The reservoir **80** supplies oil, through a valve mechanism, to be described later, and a pair of pipes **82**, to opposite ends of the compressor. Each pipe **82** is connected through a sight feed regulating valve **83** and an elbow **84** to a conduit **85** extending through the end members **32** and **44**, and the outer heads **5** and **6**. Thus the conduits **85** terminate in orifices **86, 86** which are positioned within the openings **19** and **20**, respectively, in the inner heads **15** and **16**, respectively. These orifices **86** at the discharge ends of the conduits **85** face in the direction of the windage stream created by the rotor and are so shaped that the windage created by the rotor tends to suck or draw oil from the conduits **85**.

When the compressor is in operation, oil flows by gravity from the reservoir **80** through the pipes **82** and the sight feed regulating valves **83** and through the conduits **85**, and is discharged through orifices **86** into the annular spaces **19** and **20** in heads **15** and **16**. The rotor is provided at each end with an annular groove **87** which extends to the bottoms of the vane slots **9**. The oil therefore flows from the annular spaces **19** and **20** into the annular grooves **87** and thence into the slots back of the vanes as the vanes move outwardly in the slots, and later, when the vanes move inwardly, the oil in the slots works outwardly between the side walls of the slots and the vanes (being assisted by centrifugal force) thereby thoroughly lubricating them. Sufficient oil also works outwardly on each end of the rotor to lubricate the ends of vanes 10 and to help seal the ends of the rotor and the surfaces of heads 15 and 16.

A baffle plate 88 is preferably positioned on the outer surface of each of the inner heads 15 and 16, these plates having openings through which the orifices 86 project into the annular spaces 19 and 20 in the heads 15 and 16, and through which the shafts 7 project. These baffles intercept the oil that may splash out of the rotor slots 9 when the vanes descend in the said slots and return most of it to the annular grooves 87 in the ends of the rotor, but some of the oil works outwardly along shafts 7 and lubricates the bearings at each end of the rotor, and some of the oil also drops down in the space between the inner head 15 and the outer head 5, and between the inner head 16 and the outer head 6, and flows through channels 88a in the lower edges of the heads 15 and 16 to the under sides of the floating rings 25. The floating rings 25 have numerous pockets 25a on their outer face and these pockets pick up oil received through channels 88a, as the rings 25 rotate in service, thereby materially improving the lubrication of said rings.

When the pump is being operated as a compressor. the pressure adjacent the ends of the rotor is considerably above atmospheric, due to unavoidable leakage past the ends of the rotor. Such pressure would interfere with free gravitational flow of oil from the reservoir 80 through the pipes 82. To equalize the pressure in the reservoir, I therefore provide an air tube 89 which communicates the space within the head 6 with the upper space of the reservoir 80. This insures equal pressures above and below the oil in the reservoir and uniform gravitational flow of oil from the reservoir through the conduits 85 into the compressor.

Referring now to Fig. 3, it will be noted that the inlet port 11 is so positioned with respect to the rotor that excess oil escaping from the rotor slots accumulates in the bottom of the inlet port 11 in the stator, creating a pool of oil therein. The lower edges of the ports 50, communicating the passage 51 with the port 11, are positioned a substantial distance above the bottom of the port 11 so as to permit a substantial pool of this oil to accumulate without overflowing into the intake passage 51. The bottom of the port 11 preferably extends substantially horizontally and tangentially with respect to the lowermost portion of the rotor. Therefore as the rotor rotates at high speed in a clockwise direction (the direction of rotation being taken with reference to Fig. 3), the outer edges of the vanes whip the oil in the pool in the port 11 and splash it into the intake passage so that the oil is finely broken up and carried into and through the compressor with the air stream. This flow of oil through the compressor not only helps to maintain proper lubrication between the moving parts but carries away dirt that might otherwise tend to accumulate within the pump. The mixture of air and oil is discharged from the pump proper through port 12, the lower inner edge of which is positioned well above the bottom of the port so spent oil and dirt can not flow back and foul the rotor.

It is very desirable to separate the oil from the air before the air is discharged from the compressor, not only to eliminate oil from the air delivered, but to recover as much of the oil as possible for reuse. To this end a pocket 91 is formed in the chamber 57, juxtaposed to the discharge orifice 56, so that the mixture of oil and air escaping from orifice 56 is directed into the pocket 91. In order for the air to escape from pocket 91 into chamber 57 it has to reverse its direction of flow and pass out between the rim of the pocket 91 and the edges of the orifice 56. As a result of this sharp change in the direction of the flow of the air within the pocket 91, most of the oil leaves the air and accumulates on the walls of the pocket, and runs down the walls to the bottom of the pocket where it escapes through a vent 92 into a sump 93 in the bottom of chamber 57. To reduce as much as possible the quantity of oil that tends to follow the air stream out of the pocket, the mouth of the pocket is preferably provided with an inturned flange 94 which traps excess oil and carries it to the bottom of the pocket and into the vent 92.

An oil return tube 95 extends from the bottom of the sump 93 up to an oil filter 96. Another tube 97 extends from the outlet of the oil filter to the top of the oil reservoir 80. Communication between the sump 93 and the tube 95 is regulated by a float valve 98 operated by a float. Whenever the oil rises in the sump above a predetermined value, the float lifts to open the valve 98, whereupon oil is discharged from the sump through the filter 96 to the reservoir 80 by virtue of the fact that the pressure existing in chamber 57 and sump 93 is greater than the pressure in the oil reservoir. When the oil drops to a desired minimum level in the sump 93 the float valve 98 stops the flow of oil therefrom.

By virtue of the fact that when the pump is in operation the pressure within the ends of the compressor and the oil reservoir 80 is usually above atmospheric, I find it desirable to provide an automatic valve through which the reservoir 80 may be refilled as necessary without danger of the hole through which it is filled being accidentally left open, as such an accident would result in failure of the oiling system to operate. This valve comprises a cylinder 100 formed within the reservoir 80 and in communication with the interior thereof through a vent 101 in the lower end and a port 102 in the side. A piston 103 is slidably mounted in the cylinder 100, which piston is normally maintained in uppermost position in which it blocks the port 102, by a spring 104 compressed between the piston and the lower end of the cylinder 100. The piston 103 is provided with a stem 105 extending up through and beyond the upper end of the cylinder. The upper end of the cylinder is normally closed by a screw cap 106 having a reentrant recess therein for clearing the valve stem 105. To replenish the oil supply in the reservoir, the cap 106 is removed, the stem 105 depressed to push the piston 103 clear of the port 102, and oil poured in. The stem 105 is then released, and the filler cap 106 replaced. Should the operator neglect to restore the filler cap 106, the piston valve 103 would prevent escape of air or oil from the reservoir through the filler opening.

Since oil feeds by gravity from the reservoir 80 to the compressor mechanism, flow would normally continue at all times regardless of whether or not the compressor was operating.

Obviously when the compressor is not operating there would be no pressure in the chamber 57 to restore the oil to the reservoir, with the result that the entire mechanism might become flooded with oil. I prevent possibility of such flooding by providing an automatic control valve interconnecting the reservoir 80 with the feed pipes 82. This valve construction is shown in Fig. 3, and comprises a cylinder 107 formed in the wall of the reservoir 80, which cylinder communicates, adjacent its lower end, with the reservoir 80 through port 110, and, adjacent its upper end through a pipe 108, with the intake passage 11 of the compressor. The cylinder 107 contains a weighted piston valve 109, which, when in lowermost position, covers the ports leading to the pipes 82, but has its lower end positioned above the port 110 leading to the reservoir 80. The piston valve 109 is connected by a stem 112 to a weight 113 for normally maintaining the valve in lowermost position. The upper end of the cylinder 107 is closed with a casing 114, which encloses the weight 113 and seals the upper end of the cylinder from the atmosphere.

When the pump is not in operation, the pressure within the reservoir 80 is substantially atmospheric. Likewise, the pressure in the intake passage 11 is substantially atmospheric. Therefore, the pressures acting on the upper and lower surfaces, respectively, of the piston 109 are substantially equal, and the piston remains in its lowermost position by virtue of its weight. In this position, the ports in the cylinder 107 connected with the pipes 82 are closed, so that oil can not flow from the rservoir to the compressor.

However, when the pump is placed in operation, the pressure in the intake passage 11 is reduced, and the pressure in the reservoir is increased by virtue of the fact that certain leakage of air occurs past the rotor into the end spaces of the compressor, increasing the pressure in such spaces above atmospheric. This increased pressure in the end space within the compressor is transmitted through the air tube 89 to the reservoir 80. As a result of the decreased pressure acting on the upper end of the piston 109 and the increased pressure acting on the lower end, the piston is lifted to uncover the ports leading to the oil pipes 82, whereupon normal gravity circulation of oil from the reservoir to the compressor commences and continues as long as the compressor is working.

When the compressor stops, the pressures within the ends of the compressor casing and the inlet passage 11 equalize, whereupon the piston 109 drops by its own weight to shut off the oil supply to the pipes 82. Test ports 115 for attachment of pressure gauges may be provided at circumferentially spaced intervals in the upper half of the cylinder 1, these ports being normally closed by screw plugs 116.

Cylindrical recesses 117 and 118 may be provided in the passages 75 and 76 respectively, for the insertion of cylindrical wire screens to keep foreign matter in the air stream from reaching and fouling the valve cylinder 63.

A valve screw 119, adjustable into and out of the passage 75, is preferably provided to regulate the rate of flow of air to the upper end of the cylinder 63 and thereby regulate the speed at which the pistons 64 and 65 move.

For convenience, the invention has been explained by describing in detail one specific form of pump incorporating the invention. It is to be understood, however, that many changes in the details of the particular structure shown can be made without departing from the invention, which is to be limited only to the extent set forth in the appended claims.

I claim:

1. In a rotary machine of the type described, a horizontally disposed cylinder, a rotor eccentrically positioned off vertical center within the lower part of said cylinder and having radially slidable vanes therein, bearing means for rotatably supporting said rotor, said cylinder having inlet and outlet ports upon opposite sides thereof so positioned that fluid entering the cylinder through the intake port and discharged through the outlet port is compressed between the upper side of the rotor and the upper side of the cylinder, the plane of eccentricity of the rotor and stator extending at such an angle from the vertical and the vanes being so arranged that the vertical resultant of the fluid pressure forces on the rotor acts downwardly whereby the compressed fluid acts in the same general direction as gravity to aid gravity in holding the rotor down in the bearings.

2. In a rotary machine of the type described, a stationary casing member, a rotor eccentrically positioned within said casing member and having a radially slidable vane nearly contacting said casing member with its outer edge, means for rotatably supporting said rotor in said casing member, said casing member having a curved surface for nearly contacting and sealing with the rotor vane, which surface is shorter axially than said vane and rotor, said surface merging into a counterbore portion of larger diameter at one end, an end casing member longitudinally slidable in said counterbore portion of said first mentioned casing member, for completing the compression chamber at the end of the rotor and vane, means for varying the longitudinal position of said end member in said counterbore to adjust the clearance between the end member and the rotor, and a removable filler ring interposed between said end member and the end of said counterbore, said filler ring effecting a seal between the end of said slidable end casing member and the end of said cylinder casing member, in which said means for varying the longitudinal position of said end member includes thermo-responsive pins bearing against and extending outwardly from the outer side of the slidable end casing member and anchored with respect to said casing member at their outer ends to compensate for expansion and contraction due to temperature changes and maintain a seal between said slidable end casing member and the end of said rotor in varying temperature conditions.

3. A rotary machine of the type described, comprising a stationary cylinder member, a rotor eccentrically positioned therein and having a plurality of radially slidable vanes adapted to nearly contact and seal with the inner surface of said cylindrical member, means for rotatably supporting said rotor in said cylinder member, the latter having a central interior surface shorter than said vanes and rotor and merging into a counterbored portion of larger diameter at each end, end casing members seating in said counterbored portions of said cylindrical member and completing the compression chamber at the ends of the rotor and vanes, said end casing members having inwardly extending flanges, the said flanges having inner bearing surfaces thereon of

larger diameter than the diameter of said interior surface of said cylindrical means, and a vane-retaining floating ring rotatably mounted within each of said flanges, the inner diameters of said rings being slightly less than the diameter of the said interior surface of said cylinder member.

4. A machine as described in claim 3, with means for supplying oil to said floating rings, the outer surfaces of the rings having circumferentially spaced recesses therein constituting oil pockets.

5. A rotary machine of the type described comprising casing means forming a rotor chamber having a continuous side wall, and end walls, at least one of said end walls being adjustable to vary the distance between the inner faces of the end walls, a rotor within said rotor chamber, and shaft means for rotatably supporting said rotor comprising shaft ends extending from said rotor through said end walls, bearing means for radially supporting one end of said shaft while permitting rotational and longitudinal movement of the shaft relative to the casing and bearing means for radially and longitudinally supporting the other end of the said shaft while permitting free rotation thereof, and means for adjustably supporting said last mentioned bearing means for longitudinal adjustment thereof relative to said casing, whereby said rotor can be positioned with respect to said end walls by longitudinal adjustment of said last mentioned bearing means and maintained in said position, said last mentioned bearing means comprising a cage threaded into said casing whereby said bearing can be adjusted longitudinally by rotating said cage, and means for fixing said cage in predetermined position of rotary adjustment.

6. In combination, a fluid pump having a fluid inlet passage and a fluid outlet passage, inlet valve means for closing said inlet passage in response to fluid pressure applied to said valve means, a check valve in said outlet passage for preventing return flow through said outlet passage to said pump, and a pressure responsive means responsive to predetermined pressure in said outlet passage beyond the said check valve therein for applying fluid pressure to said inlet valve means and actuating the latter to close said inlet passage, said pressure-responsive means comprising a cylinder, a piston reciprocable therein and having a face exposed to pressure in said outlet passage beyond the said check valve, means yieldably resisting movement of said piston in response to pressure on said face, a duct from said cylinder to said outlet passage at a point ahead of said check valve, a vent duct in said cylinder, said piston normally blocking said cylinder between said ducts but moving out of blocking position in response to said predetermined pressure.

7. A machine of the type described, comprising a cylinder, a rotor therein having a radial slot with a vane slidably mounted in the said slot, said vane nearly sealing with the cylinder, an inner end wall nearly sealing with one end of said vane and the outer portion of the end of said rotor, a shaft for rotatably supporting said rotor extending therefrom and through an aperture larger than said shaft provided therefor in said inner end wall, an outer end wall sealing with said cylinder and about said shaft, a closed oil reservoir above said cylinder, an oil feeding conduit extending from the lower portion of said reservoir to a point adjacent the end of said rotor and within said aperture in said inner wall, and a pressure equalizing conduit communicating the space between said inner and outer end walls with the upper portion of said oil reservoir, whereby oil flows by gravity from said reservoir through said oil feeding conduit to the rotor ends irrespective of the pressure between said end walls and rotor.

8. A machine as described in claim 7, in which the open end of said oil conduit adjacent to the rotor end is directed in the direction of rotation of said rotor, whereby windage created by the rotor tends to draw oil from the conduit.

9. A machine as described in claim 7, in which said rotor is provided with an annular recess in the end thereof below the end of said vane slot whereby oil from said conduit is directed below the vane into the vane slot in the rotor.

10. A machine of the type described comprising a cylinder, a rotor therein having a radial slot with a vane slidably mounted in said slot, said vane nearly sealing with the cylinder, an end wall sealing with one end of said vane and with the outer portion of the one end of said rotor, a shaft rotatably supporting said rotor extending therefrom and through an aperture larger than said shaft provided for the passage of the shaft in said end wall, means for supporting said shaft, an oil reservoir, an oil feeding conduit extending from said reservoir to a position adjacent the end of said rotor within said aperture in said wall, the outer orifice of said aperture being smaller than the inner orifice thereof, whereby the end of the rotor and the aperture wall form an annular trough for containing oil.

11. A machine of the type described, comprising a casing means forming an inlet passage and an outlet passage, and a mechanism therebetween which creates when in operation a higher pressure within said mechanism than in the inlet passage, a closed oil reservoir above said mechanism, an oil conduit extending from said reservoir to a point within said mechanism for delivering oil thereto by gravity flow to lubricate the same, a pressure equalizing conduit communicating said reservoir with the interior of said mechanism, and pressure responsive valve means for communicating said oil conduit with said reservoir only in response to pressure in said reservoir higher than the pressure in said inlet passage, in which said oil reservoir has a filler opening for replenishing the oil supply therein, and an automatic check valve in said opening for maintaining said filler opening closed at all times except when it is manually retained in open position.

12. In combination, a fluid pump having a fluid inlet passage and a fluid outlet passage, inlet valve means for closing said inlet passage in response to fluid pressure applied to said valve means, a check valve in said outlet passage for preventing return flow through said outlet passage to said pump, a pressure responsive means responsive to predetermined pressure in said outlet passage beyond the said check valve therein for applying fluid pressure to said inlet valve means and actuating the latter to close said inlet passage, said pressure-responsive means comprising a cylinder, a piston reciprocable therein and having a face exposed to pressure in said outlet passage beyond the said check valve, means yieldably resisting movement of said piston in response to pressure on said face, a duct from said cylinder to said outlet passage at a point ahead of said check valve, a vent duct in said

cylinder, said piston normally blocking said cylinder between said duct but moving out of blocking position in response to said predetermined pressure, in which said pressure-responsive means comprises piston means movable from a first position to a second position in response to fluid pressure in said outlet passage and applying fluid pressure from said outlet passage to said inlet valve means when in said second position, means opposing movement of said piston from said first to said second position, comprising a lever arm having a weight thereon, said weight being positioned upwardly and outwardly from the fulcrum point of said lever, whereby the force applied to said piston by said lever arm is greater when said piston element is in said first position than when in said second position, and a substantially higher pressure in said outlet passage is required to move said piston from said first to said second position than to retain the piston in the second position.

JOHN C. BUCKBEE.