

[54] COMPRESSOR CONTROL

[76] Inventor: **Arthur A. Jamieson**, 2010 E. Coalspring Ave., Michigan City, Ind. 46360

[22] Filed: **Apr. 26, 1973**

[21] Appl. No.: **354,757**

**Related U.S. Application Data**

[63] Continuation-in-part of Ser. No. 191,598, Oct. 22, 1971, abandoned.

[52] U.S. Cl. .... **417/281, 417/282, 417/295**

[51] Int. Cl. .... **F04b 49/00**

[58] Field of Search ..... 417/281, 295, 13, 23, 24, 417/26, 27, 28, 29; 418/84, 282

[56] **References Cited**

**UNITED STATES PATENTS**

1,786,278 12/1930 Wilson ..... 417/295 X

|           |         |                   |          |
|-----------|---------|-------------------|----------|
| 2,991,002 | 7/1961  | Quertier.....     | 417/295  |
| 3,105,630 | 10/1963 | Lowler et al..... | 417/295  |
| 3,582,233 | 6/1971  | Bloom.....        | 417/26 X |

*Primary Examiner*—William L. Freeh

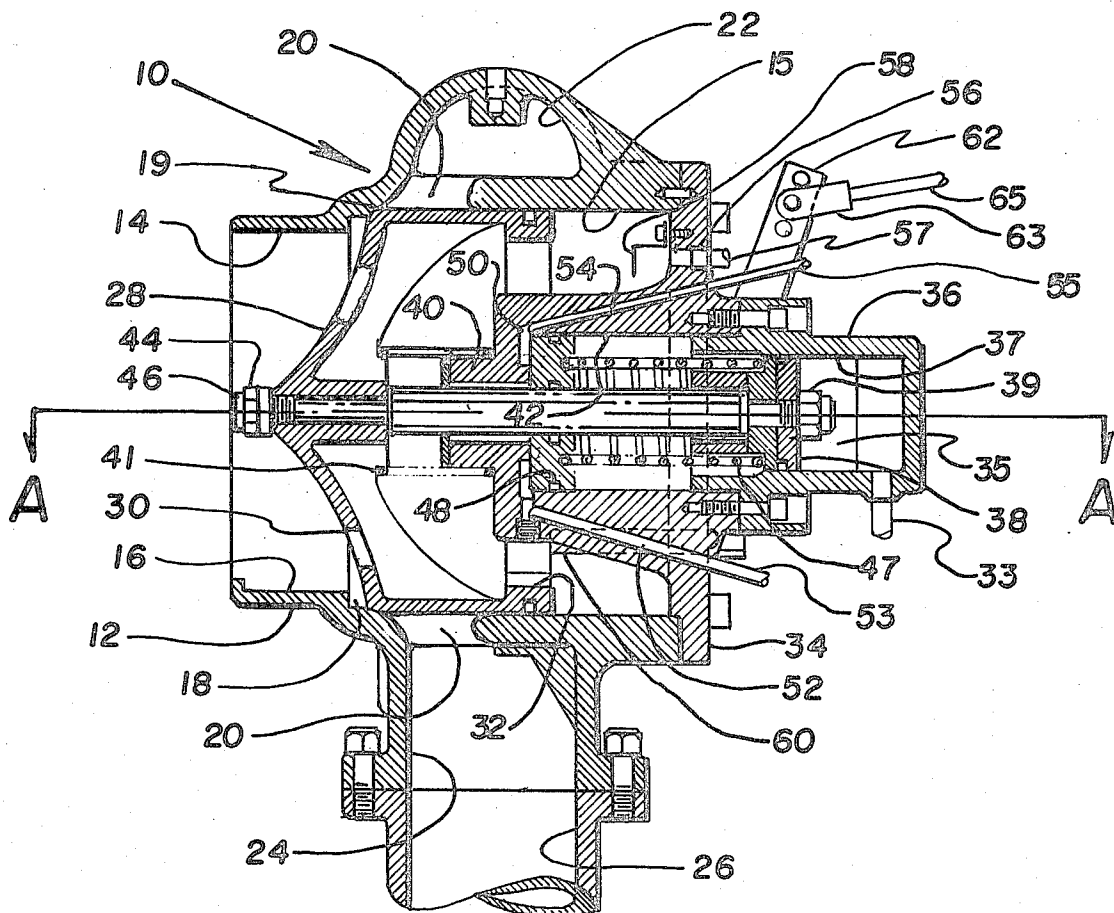
*Assistant Examiner*—Gregory P. La Pointe

[57]

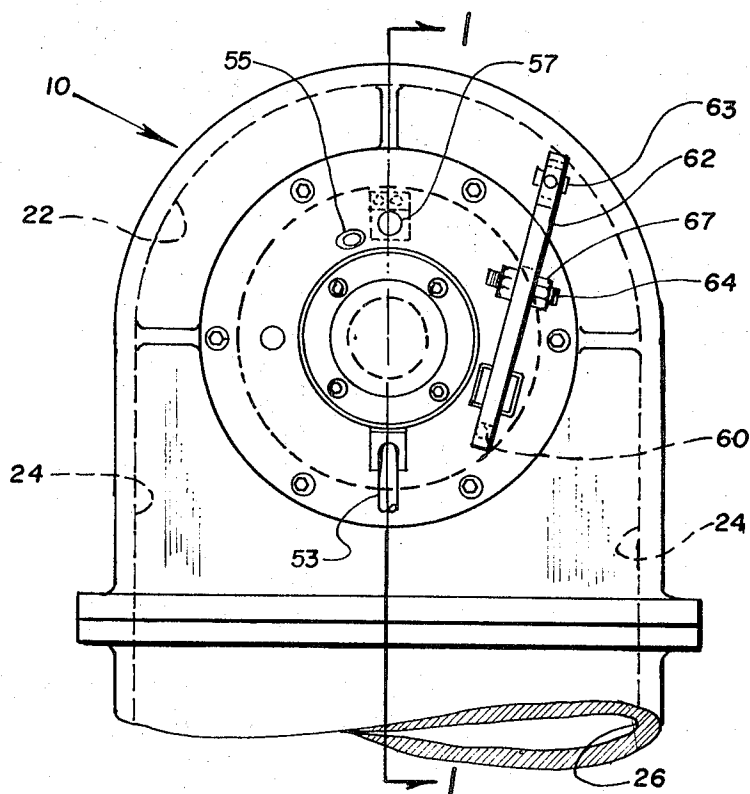
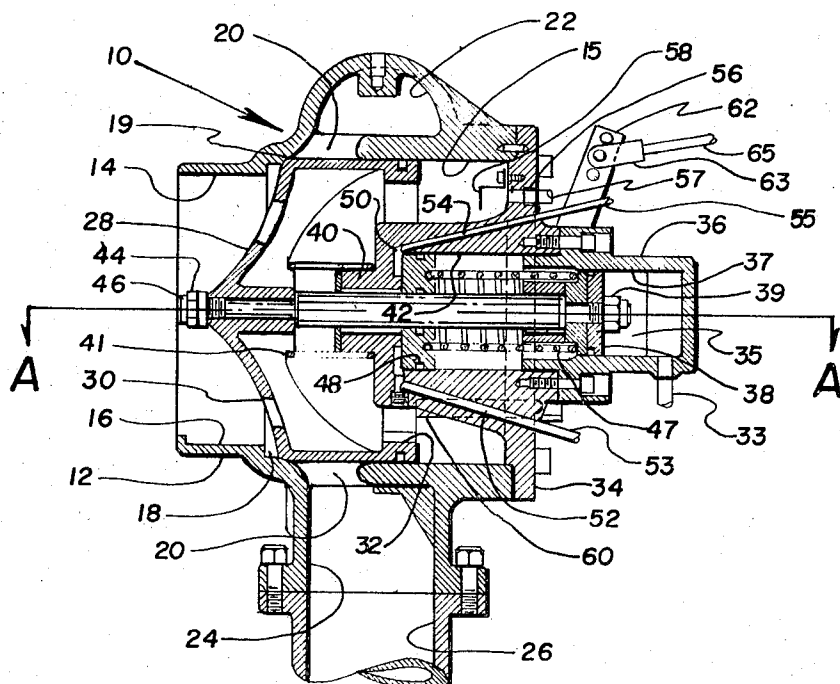
**ABSTRACT**

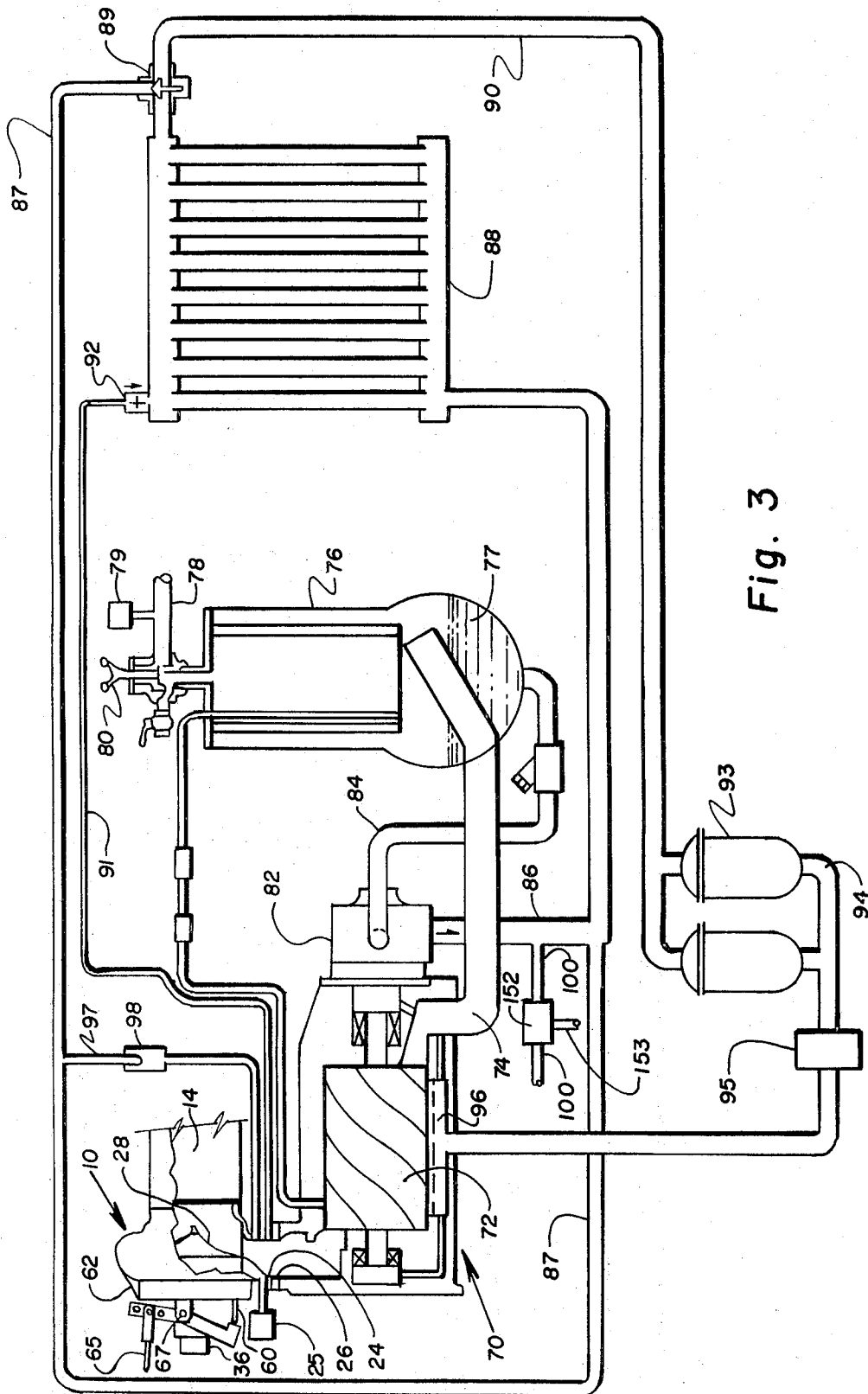
Apparatus and method for control of the operation of a liquid flooded compressor using an inlet valve operated by a combination of biasing forces including the force of the pressure of the compressed gas to provide unloaded startup and warmup periods and modulated operation from full capacity compression at receiver pressures below a preselected minimum to fully unloaded operation at preselected maximum receiver pressure.

**11 Claims, 7 Drawing Figures**



SHEET 1 OF 5





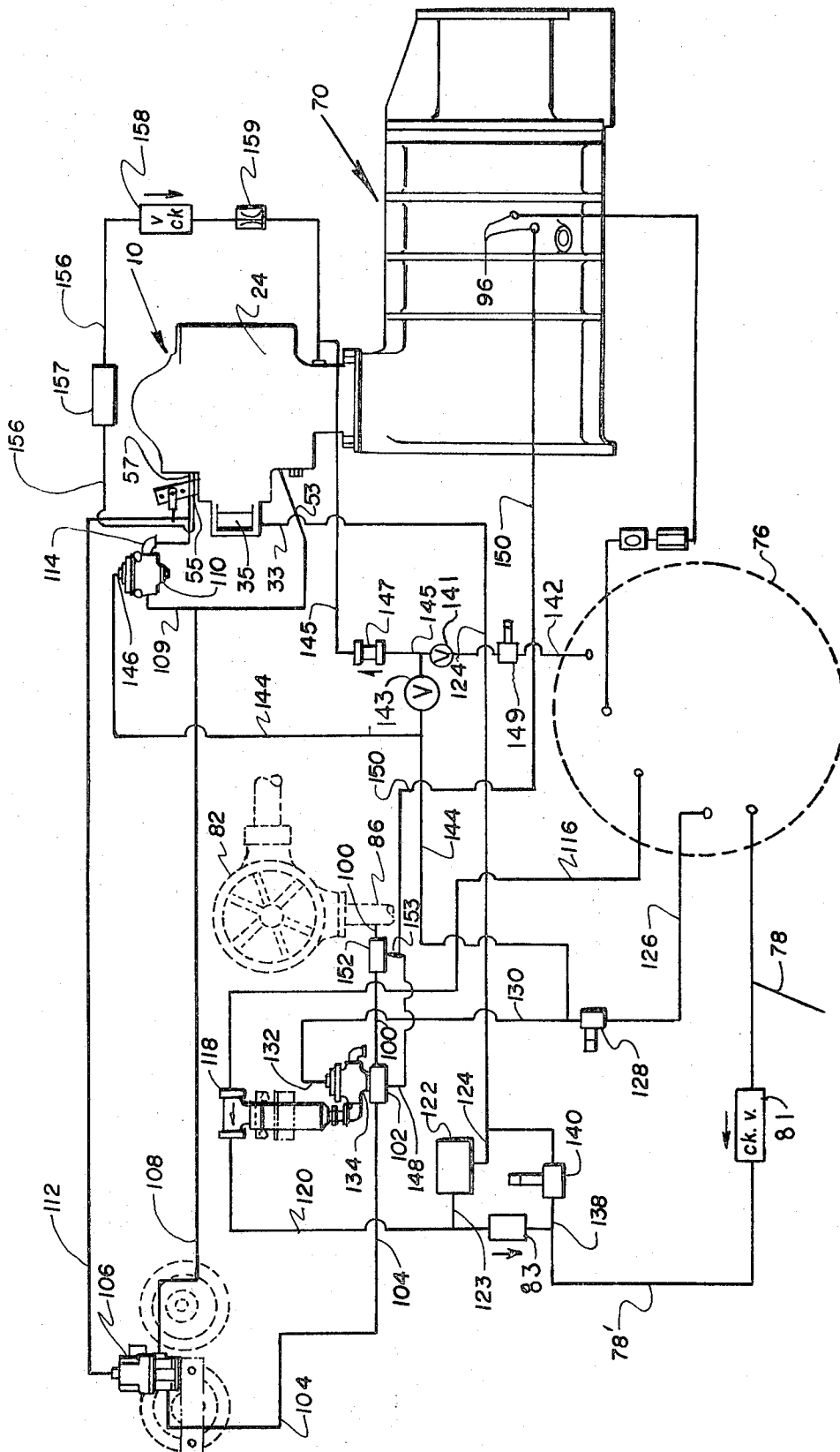


Fig. 4



FIG. 6

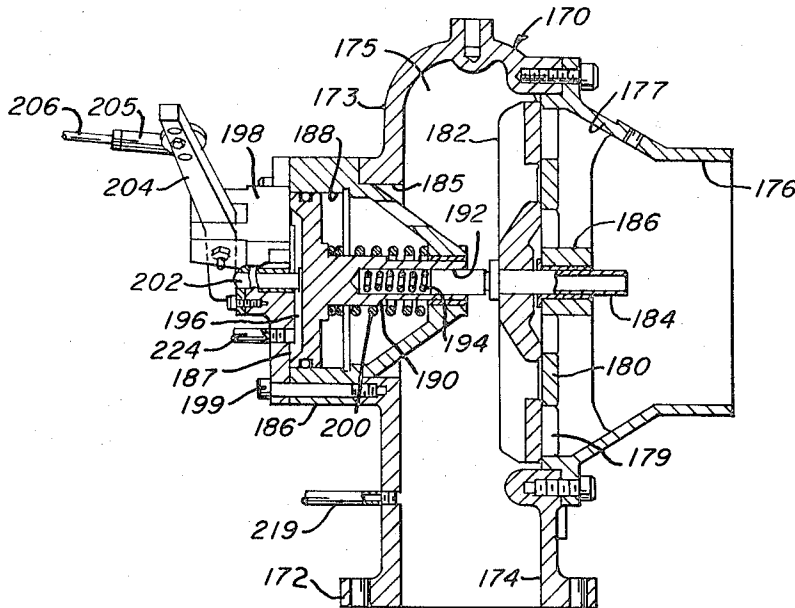
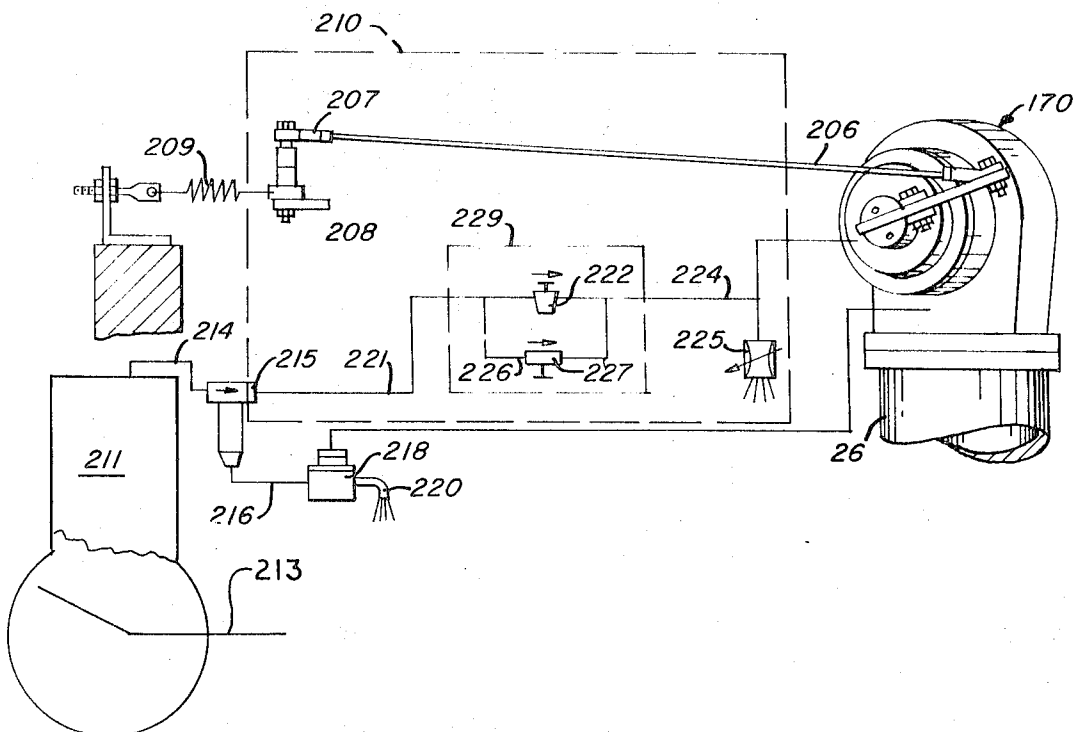


FIG. 7



## COMPRESSOR CONTROL

This application is a continuation-in-part of application Ser. No. 191,598 filed Oct. 22, 1971, now abandoned.

The apparatus and method of this invention applies to liquid flooded screw compressors including water flooded and oil injected screw compressors wherein the flooding liquid is used as a sealant and coolant within the compression chamber and in the case of oil injected compressors the liquid also serves as a lubricant for all pressure lubricated elements of the compressor. Since the preferred embodiment of this invention has been applied to an oil injected air compressor with the lubricating oil pump of the compressor supplying the liquid pressure necessary to the method and apparatus of this invention the liquid employed will hereinafter be referred to as oil and the pump supplying the liquid pressure will be referred to as the oil pump but only illustratively and not to be taken as limiting upon the scope of this invention. Operation of a compressor using other flooding liquids including water, fluorocarbons and the like and for compressing gases other than air is within the scope of this invention.

It is well known to those versed in the art of compressor operation that screw compressors have often been unloaded by closing the inlet valves thereof but this has normally occurred only at the point of maximum preselected pressure in receiver while starting up procedures have been followed with the intake valve open. Such prior art apparatus and method leads to high starting up torque with overloading effects on the compressor prime movers; initial compression of air without sufficient cooling oil giving danger of fires or explosions and because the prior art valves were unbalanced high control pressures and large control forces give rise to possible inlet valve or control element failure and consequently highly loaded compressor operation at times when such loading is either undesirable or dangerous.

The apparatus of this invention provides an inlet valve readily controlled in infinitely small steps through a range of conditions from full inlet closure to wide open and any point in between by moderate control pressures and forces. Such modulation of the control valve is produced by a combination of spring force or oil pressure and air pressure operating in separate chambers and connected in a circuit as hereinafter described so that at normal startup the inlet valve remains closed until the compressor oil pump develops a preselected pressure which will then open the inlet valve. The compressor will now operate normally until it reaches a preselected pressure in the receiver at which time the air pressure will begin to close the inlet valve but only as a throttling process more and more restricting the intake of the compressor until at a preselected maximum receiver pressure the inlet valve is completely closed and the compressor operates unloaded until the pressure in the receiver drops sufficiently to cause the inlet valve to open.

Thus, at least one form of the compressor control method and apparatus of this invention provides a completely closed inlet valve for unloaded startup for a compressor, maintaining of the unloaded position until compressor oil pressure, temperature and oil flow reach desired values, modulated opening of the inlet valve to give loaded operation under influence of the compressor oil pressure, low forces for operation of the

inlet valve during compressor operation because of the balancing feature of the inlet valve design; and modulated throttling of the inlet air to the compressor as the receiver pressure approaches a preselected maximum and complete closing of the inlet valve in response to the attainment of maximum desired receiver pressure.

These and other features and advantages of the compressor control method and apparatus of this invention will be more readily understood upon consideration of the following description and drawings in which:

FIG. 1 is a sectional view of a balanced inlet valve constructed according to the principles of this invention and taken substantially on line 1—1 of FIG. 2 looking in the direction indicated by the arrows;

FIG. 2 is an end elevational view of the apparatus of FIG. 1 taken from the right hand side of that figure looking to the left;

FIG. 3 is a schematic representation of an oil injected air compressor together with the air handling elements and the general circuit for oil injection and cooling of the compressor incorporating the balanced inlet valve of this invention;

FIG. 4 is a schematic representation of a compressor control circuit constructed according to the principles of this invention;

FIG. 5 is a median sectional view of a second embodiment of a balanced inlet valve constructed according to the principles of this invention;

FIG. 6 is a median sectional view of third inlet valve embodiment of the principles of this invention; and

FIG. 7 is a schematic representation of a compressor control circuit incorporating the inlet valve of FIG. 6.

In FIGS. 1 and 2 there is shown an inlet valve assembly generally indicated at 10 comprising a hollow body of generally flattened cylindrical shape having a stepped cylindrical bore 14 therethrough along a horizontal axis A—A. The bore 14 has a large diameter portion 15 to the right as viewed in FIG. 1 and a smaller diameter portion 16 forming the leftward open end of the body 12 with a shoulder 18 formed by the transition between the larger diameter portion 15 and the smaller diameter portion 16. Slightly to the right of shoulder 18 as viewed in FIG. 1 the large bore portion 15 is intersected by a plurality of passages 20 which communicate between the bore 14 and a semi-encircling passageway 22 which in turn communicates with a downwardly extending passageway or valve throat 24 suitably mated to and mounted upon a tubular flanged inlet member 26 of a screw compressor to be hereinafter described.

Slidably sealingly received within the bore portion 15 of the body 12 is a hollow cylindrical cup-shaped valve member 28 with its cylindrical outer portion of an axial dimension greater than the axial extent of the passages 20 so that in the position shown in FIG. 1 the valve member 28 closes off the passages 20 to interrupt communication between the open end 16 of the valve body 12 and the throat 24 and inlet member 26 by completely closing the passages 20. As will be obvious from inspection of the drawings, valve member 28 when moved axially to the right in FIG. 1 will open communication between the open end 14 of the valve body 12 and the throat 24 (see FIG. 3) in a manner familiar to those versed in the field of air compressing. The valve member 28 is provided with a plurality of openings 30 in its left hand face portion so that air pressure within and outside of the member 28 will be the same and

there will be no tendency to open or close the valve by differences in pressure between the bore portion 16 and the throat 24. There will be of course be forces acting upon the inside of the cylindrical body portion of the valve member 28 but these forces will be equally applied in all directions and merely push outward on the material of the valve member 28 in a radial direction not interfering with the opening and closing of the valve 28 in the body 12.

It is to be noted that the valve body 28 forms an interference fit with a left hand end portion of the bore portion 15 and does not normally impinge against the shoulder 18 but rather forms a sealing fit against a sloping portion 19 of the bore 14 along the left hand edge of the openings 20 as seen in FIG. 1. The right hand end of the body valve member 28 has a large diameter opening therein which actually leaves only a thickened rim 32 around the open end of the cup-shaped valve member 28.

Covering the open right hand end 15 of the bore 14 is a broadly flanged hollow cylindrical inwardly extending closure member 34 which as a central axis coinciding with axis A—A and is composed of a short hollow cylindrical portion extending to the right of the flange for the mounting of a cup-shaped end cap member 36 thereon while a leftward extending portion of the closure member 34 of slightly larger outside diameter extends axially to the left as viewed in FIG. 1 to a point within the rim portion 24 of the valve member 28 with a shoulder forming the transition to a much smaller diameter leftward extending cylindrical portion 40 which forms a guide for a compression spring 41 extending between the closure member 34 and the interior of the valve member 28 for a purpose to be described.

The closure member 34 has, extending axially there-within, a stepped cylindrical bore 42 having a small diameter leftward portion slidably receiving an elongated axially extending shouldered mounting shaft 46 extending through the central axis of the valve member 28 and secured therein as by threaded fasteners or lock nuts 44. Use of a pair of locknuts 46 provide for loose mounting of valve member 28 on the shaft 46 so that a certain amount of radial float allows valve member 28 to make better sealing contact with a sealing surface 19 near the left hand end of bore portion 15 adjacent the shoulder 18 and reduces wear by relieving shaft 46 of any misalignment effects.

The cap member 36 has an axially extending blind bore 37 therewithin which slidably sealingly receives the outer periphery of a modulating piston 38 therewithin. The central shaft 46 extends through the larger diameter portion of the bore 42 and through the modulating piston 38 to be rigidly secured therewithin as by a threaded fastener such as a nut 39 so that movement of the piston 38 along the bores 37 will cause the shaft 46 and consequently the valve member 28 to move therewith. The modulating piston 38 is provided on its leftward surface with a suitable shoulder to receive and guide an end of a compression spring 47 suitably compressed with a proper preload between the modulating piston 38 and a floater piston 48 slidably sealingly received within the bore 42 so that the spring 47 holds the floater piston 48 against a shoulder portion of the bore 42 surrounding the central shaft 40 to prevent the floater piston 48 from resting against the bottom of the bore portion 42. There is thus formed an expansible chamber 50 between the leftward face of the floater

piston 48 and the bottom of the bore portion 42 which communicates with a source of pressurized fluid by way of a passage 52 formed in the closure member 34 and a pipe connection 53 engaged in the outer end of the passage 52 in fluid tight relationship thereto. At a similar location but above the axis A—A and penetrating from the outside of the closure member 34 into the chamber 50 is a passage 54 having a pipe connection 55 engaged in the outer end of passage 54 to provide communication with chamber 50 for a purpose to be described. Above connection 55 and extending through the flange portion of the closure member 34 into the hollow interior of the body 12 is a bore 56 having a pipe connection 57 sealingly secured therein and having a fluid deflector 58 secured to the inner surface of the closure member 34 in a position to obstruct liquid entering the bore portion 15 from the bore 56 for the purpose of deflecting a stream of liquid before it travels too far through the valve assembly with undesirable effects.

A further feature of the inlet assembly 10 is a throttle control pin 60 slidably mounted in a bore portion of the closure member 34 in a position parallel to the axis A—A and so located as to have its inner end in contact with the right hand annular surface of the rim 32 on the valve member 28 and its outer end in contact with a lever 62 best seen in FIG. 2. The lever 62 is pivoted on a pin 64 pivotally mounted in a pair of bosses 67 extending outwardly from the surface of the closure member 34 for that purpose. The lever 62 has a plurality of through holes at its upper end for adjustably and pivotally mounting a crotch type clevis 63 rigidly but adjustably secured to a throttle control rod 65 (only partially shown) which extends to the throttle control mechanism of a prime mover such as a gasoline or diesel engine (not shown) being used to provide power for operating the compressor with which the control system and method of this invention is operatively associated.

It is to be noted that the modulating piston 38 forms with the blind bore 37 an expansible chamber 35 which communicates with a source of pressurized gas by way of a pipe connection 33 which is a part of the control circuit hereinafter described.

It is to be noted that the spring 41 is suitably preloaded to hold the valve member 28 in the closed position as shown in FIG. 1 until a suitable amount of pressure applied through the connection 53 to the chamber 50 moves the floating piston 48 to the right far enough to compress spring 47 to a greater tension than that of the spring 41 at which time the valve member 28 and the shaft 46 carrying the modulating piston 38 will move to the right as seen in FIG. 1 bringing the valve member 28 into the full open position seen in FIG. 3. At this time the floating piston will be moved to a position reducing the chamber 35 to a small portion of the blind bore 37 connected as shown by the pipe 33 to a source of compressed air. When the pressure of the compressed air in the chamber 35 becomes great enough the modulating piston will move to the left against the tension of spring 47 and moving the shaft 46 with the valve member 28 mounted thereon will move the valve member 28 into the closed position again as shown in FIG. 1 but with the spring 47 compressed to a maximum extent by the positioning of the floater piston 48 against the inner end of the cap member 36.

It is further to be noted that whenever the pressure of liquid in the chamber 50 falls below a preselected minimum the valve 28 will close whether or not there is any pressure in the chamber 35. It is also to be noted that the valve member 28 will be in the closed position whenever the pressure in the chamber 35 is great enough to overcome the force of the spring 47 regardless of how much pressure there is in the chamber 50.

In the air oil flow diagram labeled FIG. 3 there is shown the inlet valve assembly 10 mounted on the inlet member 26 (as has heretofore been described) of a screw compressor unit 70 of a type well known in the art wherein air or other compressible gas admitted through the intake body 12 past the valve member 28 shown in the open position in FIG. 3 is compressed in the screw compressor unit 70 by action of at least a pair of screw rotors 72 which compress the air and deliver it through an outlet connection 74 to an air oil separator and receiver 76 from whence it is delivered through an air outlet connection 78 controlled by a discharge valve 80 to a compressed air storage tank or compressed air operated unit as desired.

At a suitable point in the discharge line or air outlet connection 78 (see FIG. 3) a pressure switch 79 is mounted in communication with the discharge or service air. This pressure switch is normally closed but opened by any pressure above a set minimum for example 102 pounds per square inch for a purpose to be set forth.

Also shown in FIG. 3 is a vacuum switch 25 communicating with the interior of valve assembly throat 24. This vacuum switch is normally closed but opens at any vacuum greater than a selected value, for example a vacuum of 22 inches of mercury or more for a purpose to be set forth.

The screw rotors 72 being driven by a gasoline or diesel engine or an electric motor or other prime mover (not shown) in turn are connected to and drive an oil pump 82 of any well known type in a liquid conditioning circuit wherein the oil pump 82 has an inlet connection 84 leading from a sump portion 77 of the oil air separator 76 wherein the oil having been separated from the compressed air gathers and is recirculated accordingly through the circuit to the following description. Oil from the sump 77 through the inlet connection 84 is driven by the pump 82 through a discharge connection 86 to and through a suitable heat exchanger such as an oil cooler 88. After flowing through the oil cooler 88, an oil line 90 carries the oil to and through a pair of oil filters 93 having a discharge connection communicating, through an oil line 94, with an oil gallery or manifold 96 which suitably connects through the body of the compressor unit 70 to a compression chamber within the compressor unit 70 to provide cooling, sealing and lubrication for the screw rotors 72 in a manner well known in the art. As is well known, the oil circulating through the above described circuit absorbs the heat of compression from the compressed air; falls into the sump 77 and, by way of the oil pump 82 and cooler 88, extracts the heat from the system to be carried away by circulated air or water according to the conditions present.

Special features of the oil circulating portion of the compressor cooling system of this invention as seen in FIG. 3 are the by-pass oil line 87 communicating from discharge line 86 to a thermal by-pass valve 89 situated in the cooler discharge line 90 wherein the thermal by-

pass valve 89 is the heart of the compressor temperature regulating system in that discharge air temperature is held in direct relation to the compressor oil temperature. The thermal valve 89 is designed to allow full oil by-pass flow through the by-pass line 87 into the cooler discharge line 90 until the oil temperature reaches a preselected value for instance 100°F. At temperatures ranging above that value the valve 89 gradually throttles the by-pass flow as the temperature increases and becomes completely closed to by-pass oil when the oil temperature reaches a preselected maximum value for instance 130°F. Thus, with the compressor operating and the temperature of the oil below 100°F most of the oil from the oil pump 82 will pass through the by-pass line 87 and not through the cooler 88. With the thermal valve 89 set to the above-mentioned temperatures there will be assured a minimum oil injection temperature approximately 115°F to the compressor. The valve 89 also has a built-in pressure relief feature so that a pressure difference of more than a selected value, for instance, 50 pounds per square inch from line 87 to line 90 will open the thermal valve 89 regardless of temperature. Thus, if the cooler should become plugged, the thermal valve will open regardless of temperature to provide lubrication to the compressor until the safety circuit shuts down the machine because of high discharge temperature.

Another feature of the oil cooling system resides in a check valve 92 connected by an anti-vacuum line 91 from the inlet upper end of the cooler 88 to the throat 24 for the purpose of breaking any vacuum that should form when the compressor has been shut down and the oil is draining back through the line 86 to the sump 77 so that pressure of the atmosphere in sump 77 will not prevent proper draining of the cooler 88.

Also connected to the outlet side of the oil pump 82 as by connection to the by-pass line 87 is a line 97 communicating with the interior of the inlet valve throat 24 by way of relief valve 98 set to open at a desired maximum value say 150 pounds per square inch to provide safety control against overpressuring the cooler or the oil filters in the case of either of these elements becoming completely plugged up.

Some of the advantages of such a system are the provision for low pressure drop between the oil sump and the oil pump inlet which is of considerable benefit during cold starting or operation at sub-zero weather conditions and insures faster delivery of oil at the compressor injection area; line 97 and relief valve 98 provide for oil dumping as a safety feature and allows an extended time period for the compressor high temperature safety shut-down switch to operate; and of course the anti-vacuum line 91 provides for good draining of the cooler 88 so that starting will not find the cooler 88 completely filled with cold oil with the attendant disadvantages.

In the oil line 94 there is shown an oil flow switch 95 which is normally open with no oil flowing and electrically closed whenever oil flow through switch 95 exceeds a preselected minimum. Flow switch 95 may have an on-off feature providing for manual or electrical opening of the switch to provide unloaded operation when the engine and compressor are running.

At a point in the oil pump discharge line 86 a control oil supply line 100 communicating therewith connects the oil pump into the control oil circuit as shown in FIG. 4. The line 100 provides pressurized oil to the

inlet end of a control oil pressure regulator 102 which is any suitable device which can be set to control the oil pressure to the inlet valve assembly 10 to not exceed a maximum pressure of 30 to 35 psi. This connection of the regulator 102 to the inlet valve assembly 10 is by way of a line 104 through a 3-way solenoid valve 106 and from thence by way of a line 108 to the connection 53 on the inlet valve assembly 10 leading to the floating piston chamber 50 or alternatively by a branch line 109 to a control oil exhaust valve 110 for a purpose hereinafter made plain. The third connection of the 3-way solenoid valve 106 communicates by a line 112 with the connection 57 on the inlet valve assembly 10 which leads into the interior of the assembly as a convenient disposal place for oil released by the solenoid valve 106 or by the control oil exhaust valve 110 through a connection 114 also communicating with pipe connection 57 of the inlet valve assembly 10.

It is to be noted that, by suitable reconnections, 3-way solenoid valve 106 can be replaced by a 2-way solenoid valve (not shown) and line 112 eliminated.

The air connections of the control circuit of the inlet valve assembly 10 begin with an inlet air line 116 leading from an upper portion of the air oil separator to an upper connection on a control air condensate trap 118 through which the control air from the receiver 76 passes with suitable elimination of any oil and water condensate or solid contaminants which might interfere with the operation of components further along the line. From the trap 118 by way of a control air line 120 the air from the receiver 76 enters a control air regulator 122, by way of a connection line 123, this is an adjustable regulator designed to produce outlet pressures by way of a discharge connection line 124 of a pressure value varying from 0 to 50 psi proportional to a variation in supply pressure totaling 15 lbs. per square inch i.e., from a reading of 100 lbs. per square inch to 115 lbs. per square inch if this is the selected range of pressures. It is, of course, possible to use other pressure ranges as found to be desirable merely by using a different setting of regulating valve 122. In other words when the pressure at connection 123 is less than 100 lbs. the pressure at connector 124 will be zero approximately. And when the pressure at connection 123 is at or near 115 lbs. per square inch the outlet pressure at connection 124 will be of the order of 50 lbs. per square inch.

The line 124 will be seen in FIG. 4 to connect with the pipe connection 33 leading to the chamber 35 of the inlet valve assembly. Another air connection from a point on the receiver 76 is a line 126 conducting air through a normally open solenoid valve 128 and a further line 130 to a pressure control connection 132 on a condensate blow-down valve 134 which is normally maintained in a closed position by pressure applied through the condensate trap 118. A bypass line 138 with a normally open solenoid valve 140 leads from connection 123 on line 120 to a connection on line 124. Solenoid valve 128 also connects by way of an air line 144 to the control diaphragm of the control oil exhaust valve 110 by way of a connection 146.

A center connection 148 on the control oil regulator 102 leads by way of line 150 to a point 96 in the screw compartment of the compressor unit 70.

A 3-way bypass valve 152 inserted in control oil line 100 has a third connection 153 controlled by a temperature sensitive element so that the bypass valve 152 is

maintained in a condition to connect control oil line 100 with bypass line 150 until the oil temperature has reached approximately 50°F. When this condition has been reached the bypass valve 152 operates into the condition of passing oil through the line 100 with the bypass 153 shut off.

The only electrical connections necessary to the explanation of the operation of this control system will be the valve energizing circuit comprising, the electrical connection to solenoid valves 106, 128 and 140 in such a manner that the valves 128 and 140 will be energized and closed whenever the control system is energized and the unit 70 is operating at a speed sufficient to provide oil flow to close the flow switch 95 in series with solenoid valves 106, 128 and 140 which will be energized and electromagnetically closed. Energizing the solenoid valve 106 will provide fluid flow connection from the line 104 to the line 108. Deenergization will open the valve 106 so that the connection is from line 108 to line 112 and thence to the atmospheric oil dump 57 in the inlet valve assembly 10 with line 104 blocked at the port of valve 106.

The pressure switch 79 and the vacuum switch 25 are electrically connected in parallel to each other and in series with the three solenoid valves 106, 128 and 110 as well as in series with the flow switch 95 in the valve energizing electrical circuit.

Operation of the control system will begin with the energization of the compressor electrical system closing valves 128 and 140 as above mentioned and connecting lines 104 and 108 by way of 3-way solenoid valve 106 if oil temperature and flow have been established. Next the prime mover and the compressor connected thereto are operated in the starting mode and because of the spring 41 within the valve member 28 the valve member 28 will remain in a completely closed position with no inlet air allowed into the compressor so that the compressor will be operated unloaded. With the valve member 28 in a closed position the control pin 60 will be just disengaged from the lower end of the lever 62 and the rod 65 biased by a spring (not shown) into the farthest right hand position as seen in FIG. 1 will hold the engine speed control at idle or thereabouts so that the engine can warm up in an unloaded condition without undue stress in the starting mode.

When the oil temperature reaches approximately 50° the temperature sensitive element will begin to operate bypass valve 152 into the mode to pass oil through line 100 to the regulator 102 and shut off the bypass connection 153 to the line 150. With this operation some oil under pressure will pass through the regulator 102 and the line 104 and through the valve 106 in its energized mode to the line 108 and thence to the connection 53 on the inlet valve assembly 10. From the connection 53 the oil pressure is transferred to the float piston chamber 50 by way of the passage 52 and as the pressure builds up through operation of the oil pump 82 float piston 48 is moved to the right as seen in FIG. 1 until the force of the spring 47 pushing to the right becomes greater than the force of the spring 41 pushing to the left and modulating piston 38 together with the shaft 46 and the valve member 28 are moved to the right.

As soon as control oil pressure from connection 53 is applied to chamber 50 some of the chamber oil begins to bleed off through the passage 54, connection 55, and a line 156 to the interior of throat 24 as shown

in FIG. 4. Line 156 communicates through a filter or oil strainer 157, a check valve 158 and a restrictive orifice 159 to maintain a small flow of oil through chamber 50 at all times during compressor operation so that, once proper oil temperature has been achieved, the oil in chamber 50 will be kept warm enough for proper operation.

As the valve member 28 moves to the right the passageways 20 are opened gradually to a greater and greater degree so that air entering through the inlet end of bore 14 will traverse the inlet valve throat 24 and inlet member 26 into the compressor unit 70 wherein it is compressed and mixed with oil and passed on through the separator receiver assembly 76 where the oil having been separated from the air falls to the bottom portion 77 of the air oil separator 76 while the service air passes on out through the valve 80 and the line 78 to a place of use of the compressed air.

When the valve member 28 moves to the right the control pin 60 is biased in the same direction by the right hand edge of the valve member 28 and in turn the control pin 60 moves the bottom end of the lever 62 to the right as seen in FIG. 1 moving the control rod 65 to the left and thus opening the throttle on the engine (not shown) as hereinabove explained. It is to be noted that as long as the oil pressure is below normal the valve member 28 will not be in full open position and as a matter of fact the passageways 20 are opened only gradually as the oil pressure increases and as the engine gradually picks up speed. Thus there is no snap open, sudden loading of the compressor or sudden explosive increase in sound level as the engine picks up speed. Thus even in start-up the engine is only gradually loaded and gradually accelerated by a slowly opening throttle under control of the valve assembly 10.

The compressor now operating at full throttle and full delivery will continue in this mode until the pressure in the receiver 76 exceeds the set minimum as for example 100 lbs. per square inch desired for working air.

When the pressure in the separator 76 reaches 100 lbs. per square inch air passing through the line 116 and the condensate trap 118 will be applied through lines 120 and 123 to the air pressure regulator 122 and as hereinbefore stated when the pressure exceeds 100 lbs. per square inch at the connection 123 the regulator 122 will begin to transmit air at control pressure through the line 124 and the connection 33 to the chamber 35 of the inlet valve assembly 10. As the pressure increases above 100 lbs. per square inch on the connection 123 the regulator will increase the pressure in line 124 and consequently in the chamber 35 from 0 to a control air maximum pressure of 50 lbs. per square inch. In the range of control air pressure from 0 to 50 lbs. per square inch corresponding to delivery air pressure between 100 lbs. per square inch and 115 lbs. per square inch the modulating piston will be moved to the right as seen in FIGS. 3 and 4 or to the left as better shown in FIG. 1 and by axially moving the shaft 46 will gradually close the valve member 28 across the passages 20 gradually reducing the amount of inlet air into the compressor 70 and at the same time control rod 60 will follow the right hand edge of the valve member 28 allowing the lever 62 to rotate in a clockwise direction as seen in FIG. 1 under influence of a spring (not shown) which is part of the throttle equipment of the engine (also not shown) so that as the

passageways 20 become progressively narrower, the fuel supply to the engine will be decreased and there will be a gradual approach to, and final achievement of, fully unloaded compressor operation at pressures below atmospheric and engine operation at idle speed with again no sudden change in noise level which is so annoying to people in the vicinity.

As may be well appreciated, the above action from start-up through pressure build-up to full receiver pressure with compressor unloaded and idling achieved will be essentially repeated whenever a demand for air reduces the pressure in the separator 76 and consequently through the regulator reduces the pressure in the chamber 35 below the 50 lbs. per square inch control air pressure necessary to hold the valve member 28 in the closed position. The inlet valve will open under the force of spring 47 and of the oil pressure in the chamber 50 and concomitantly open the throttle of the engine to furnish the power necessary for the required compression in the unit 70.

For normal shut-down of the compressor the compressor is first unloaded by increasing the receiver pressure to 115 lbs. per square inch with the resultant closing of the valve member 28 as hereinabove described so that the compressor is running in a completely unloaded state with air pressure below atmospheric in the throats 26 and 24. It is to be noted that during this unloaded operation the floater piston 48 is being held against the left hand end of the cap member 36 as during all normal operation of the compressor. The engine is then stopped and the power shut-off from the solenoid valves 106, 128 and 140 with the result that the line 108 is now connected by the 3-way valve 106 to the line 112 whence it communicates with connection 57 into the interior of the valve assembly housing 12. The release of oil pressure from line 108 through the valve 106 to the line 112 and thence to the atmospheric dump inside the valve assembly 10 allows the floater piston 48 to move to the left in the chamber 50 (FIG. 1) and reduce the rightward bias of the spring 47 so that the spring 41 can hold the valve member 28 in the closed position. It is to be noted that the oil dumped through port 56 in the valve housing will lie in the valve housing and form a good lubricant for the unloader piston and associated parts and will be sucked into the compressor when the compressor is restarted.

As the 3-way solenoid 106 is being deenergized so also are the two solenoid valves 128 and 140 allowing receiver air to pass through the solenoid valve 140 to the modulating piston chamber 35 via lines 138 and 124 producing a pressure greater than 50 lbs. per square inch in the chamber 35 thus insuring good sealing of the unloader piston 28 over the passages 20 as the compressor becomes pressurized. As solenoid 128 is deenergized receiver air is passed along the line 126 and 130 to the blow-down valve 134 and through the lines 126 and 144 to the blow-down valve 110 which are both opened by this control pressure through connections 132 and 146 respectively. With the blow-down valve 134 at the bottom of the trap 118 opened, any moisture collected by the trap 118 will be blown out through the blow-down valve 134. A branch line 145 extending from the blow-down control line 144 by way of a check valve 147 communicates with the throat 24 of the inlet valve assembly 10 and causes the receiver through the line 126 to pressurize the inlet of the compressor to prevent or reduce the reverse rotation of

the compressor as the pressure at the discharge leaks back to the inlet and tends to drive the compressor as an expander after normal shutdown.

With the engine approaching idle speed and the inlet valve 28 almost closed a vacuum will be developed within valve throat 24. When this value exceeds the set value of vacuum switch 25 that switch will open causing valves 106, 128 and 140 to be deenergized since the service air pressure in connection 78 is above 102 psi and pressure switch 79 is open. This operation of valves 106, 128, 140 will cause the same action as normal shut-down above described resulting in sump blow-down to atmospheric pressure.

Thus, under full service pressure and inlet vacuum not only will the engine be idling but the compressor will be pumping only against atmospheric forces thus greatly reducing the horsepower expended in unloaded operation. Obviously, as soon as line pressure drops below 102 psi the pressure switch 79 will close and re-energize the electrical valve circuit reestablishing full power compressor operation as above described.

Whenever unloaded operation is desired switch 95 may be opened manually or electromagnetically to de-energize solenoid valves 106, 128 and 140 to provide unloaded operation as hereinbefore described for normal shutdown except that the engine and compressor will continue to run with the inlet valve 28 closed and the sump 76 will be depressured to reduce torque on the engine crankshaft at shut-down.

Under emergency shut-down, where the compressor is working at full speed and load, the machine safety circuit starts to function say due to overheating and the electrical circuit will be broken and compressor and prime mover will be stopped with the inlet valve at that particular instant completely open and the prime mover speed lever in full speed position since there will be zero secondary air pressure at modulating piston chamber 35 when the compressor stops. However, at the instant of electrical supply cutoff the solenoid valve 140 will be deenergized and being open will pass receiver air directly to the modulating piston chamber 35 by way of lines 116, 118, 120, 138 and 124 leading to the chamber 35. This will immediately force the modulating piston to the left (in FIG. 1) completely closing the inlet and through the throttle spring on the engine (not shown) move the prime mover throttle lever into idle speed position. Note also with the system deenergized the solenoid valves 128 and 106 will again perform their function as above described exhausting the oil in chamber 50 to the atmospheric connection 57 and through the bore 56 into the interior of the valve assembly 10.

The novel features of the control as described above include (1) a compressor inlet valve completely pressure balanced; (2) the compressor inlet remains closed completely during starting and warm-up periods; (3) the opening of the inlet valve depends on compressor oil pressure which keeps it closed during start-up until oil pressure is sufficient and immediately closes the inlet valve bringing the compressor to unloaded operation should the oil pressure fail; (4) opening of the inlet valve depends also on compressor oil temperature and flow rate, that is during cold starting the inlet valve will remain closed until the oil temperature has achieved a satisfactory operating temperature and satisfactory rate of flow to provide good bearing lubrication and sealing within the compressor and adequate cooling to absorb

the heat of compression thus avoiding the danger of un-cooled compression developing high temperature and possibly causing fires; (5) the prime mover will start and warm-up at idle speed without having to manually make any adjustment to the control linkage.

It is to be noted that the above description of the construction and operation of the inlet valve assembly and the control circuit therefore is applicable throughout, to the operation of a compressor by an electric motor with the exception of reference to the control rod 60 and lever 62 and the control rod 65 actuating the throttle mechanism of the engine. Therefore, the description is to be taken as applicable to electric motor operation except for the above noted differences.

It is further to be noted that the construction of the valve assembly 10 could be simplified by elimination of the floater piston 48 and the springs 41 and 47 with the addition of a spring similar to 47 in the chamber 35 to the right of the modulating piston 38 as seen in FIG. 1 so that the oil pressure in the chamber 50 would act directly on the backside of the modulating piston 38 while the control air pressure in the chamber 35 was acting in the opposite direction against the right hand face of the modulating piston 38. With this arrangement the spring in the chamber 35 would close the valve member 28 whenever the oil pressure in the chamber 50 was below a preselected minimum value. Then, by a proper selection of control oil pressure and control air pressure values, operation of the inlet valve member 28 could be accomplished without the interposition of the floater piston 48 as seen in FIG. 1.

The advantages resident in the valve assembly described in the immediately preceding paragraph would reside in the simpler design being less expensive and possibly more durable because of fewer moving parts.

In FIG. 5 there is shown a second embodiment of the balanced inlet valve structure necessary for the proper operation of the control circuit and method of this invention. In FIG. 5 some of the two-figure numbers are plain which shows that these numbers are the same elements in every respect as those shown under the same numbers in FIG. 1, other of the two-figure numbers bear the prime indicating that these elements are functionally equivalent to the elements of FIG. 1 carrying the same number without the prime. Other numbers in FIG. 5 are numbers from 160 upward indicating that these are new parts not found in FIG. 1.

In the showing of FIG. 5 the inlet valve assembly of FIG. 5 is generally indicated at 10' with the housing of the inlet valve indicated at 12'. A bore 14' extending inwardly from the left hand end of the body 12' as seen in FIG. 5 conducts inlet air into the interior of the body 12' under control of a double-headed balanced valve 28' operating vertically to control the entrance of inlet air into the passage 20' of substantially cylindrical nature which in turn leads to the inlet throat 24' mounted upon and communicating with the inlet member 26 of the compressor unit 70 as hereinbefore described. The valve member 28' is mounted upon a vertically extending shaft 46' and secured thereon by a nut 44'. The shaft 46' as in the earlier embodiment extends through a floater piston 48, a compression spring 47 and a modulating piston 38 secured to the upper end of the shaft 46'. An inverted cup-shaped valve plate 160 slidable on shaft 46' is held against upper sealing surfaces 162 of the opening between the upper portion of the passage 20' and the valve chamber 14'. The valve member 28'

is held against lower sealing surfaces 161 by its own weight. The valve plate 160 is held against the upper seal face 162 by a spring 41' interposed between the valve plate 160 and a cover member 34' upon which in turn is mounted a cup-shaped closure member 36' enclosing the modulating piston 38 and the air pressure chamber 35'.

Similarly to the operation of the valve assembly 10, oil pressure through the connection 53, of valve assembly 10', transferred into the float piston chamber 50' moves the float piston 48 in an upward direction until biasing upwardly of the spring 47 overcomes the weight of the valve member 28' and lifts it off the seal face 161 by moving the modulating piston 38 in an upward direction. It is to be noted that the outside pressure even though much higher than that in the throat 24' during operation of the compressor unit 70 in the unloaded mode will be acting equally in the upward and downward direction on the oppositely facing surfaces of valve member 28' in the usual design of a balanced valve.

After a short amount of motion upwardly of the valve member 28' by motion of the modulating piston 38 the valve member 28' comes into contact with the underside of the valve plate 160 and lifts it off the seal face 162 for a full opening of the valve 28' in the same manner as described for the valve assembly 10. The pressure equalizing holes 30' in the valve plate 160 serve to prevent development of pressure differential between the upper and lower sides of the valve plate 160 which might unbalance the valve action. The connection 33 as seen in the first description provides compressed air under controlled pressures to close the valve 28' by pressure exerted upon the modulating piston 38 through the chamber 35'. The atmospheric dump connection 57 communicating through the side of the body 12' by a bore 56' allows dumping of oil into the body 12' in the same manner as described for the oil dump bore 56 into the body 12 of the valve assembly 10.

The advantages of the valve assembly 10' over the valve assembly 10 reside almost entirely in the vertical positioning of the shaft 46' so that all of the movable elements within the body 12' have a greater tendency to remain in alignment than the horizontally disposed movable elements of the valve assembly 10. It is also to be noted that the openings at the seal faces 161 and 162 can be smaller than the opening within the shoulder 18 of the valve assembly 10 because in the case of the valve assembly 10' there are two openings acting through nearly all of the period during which the valve 28' is open. A further advantage resident in valve assembly 10' is to be found in the two step opening, first a lower seal face 161 and then an upper seal face 162 which refines the capacity regulation with more sensitive throttling of the compressor inlet air.

It should be noted that as shown the valve assembly 10' is suitable only for electric motor driven machines since there is no control pin or control lever shown as part of the valve assembly 10'. It is to be appreciated however that a comparatively small amount of redesign would suffice to incorporate a control pin and control lever into the valve assembly 10' should such components be desired to provide valve assembly 10' with an engine throttle control.

FIG. 6 illustrates a third embodiment of the throttling inlet valve of this invention generally indicated at 170 comprising a flange 172 suitable for mounting on the

inlet member such as member 26 of the screw compressor 70 (see FIG. 3) which flange 172 is the bottom supporting portion of a hollow valve body 173 the interior of which communicates with the interior of the inlet member 26 by way of a throat portion 174. A vertical passageway 175 in the valve body 173 communicates between the inlet member 26 and large horizontal bore 176 which forms the outer portion of the valve inlet passageway connected by an expanding conical portion 177 to the vertical inlet passageway 175 by way of a plurality of openings or ports 179 in a valve body plate 180 extending across the large end of the conical portion 177. The body plate 180 forms a seat for a movable valve plate 182 of a size and shape to completely cover the openings or ports 179 in the body plate 180, with valve plate 182 centrally supported on a horizontal shaft 184 slidably received within a bore in a central boss 186 of the valve body plate 180. The valve plate 182 is removably secured to shaft 184 and movable therewith toward and away from the body plate 180 for closing and opening the inlet ports 179 in a well known manner.

A horizontal bore 185 in the left hand side of the valve body 173 and coaxial with bore 176 removably sealingly receives and supports a hollow cylindrical valve cylinder member 186 having therein a cylinder portion 188 slidably sealingly receiving a modulating piston 187. A stem portion 190 of the piston 187 extends rightwardly as seen in FIG. 6 and has a portion slidably received in a rightward extension of the cylinder member 186, with the stem portion 190 having a central bore 192 extending axially inwardly therein and coaxial with the bores 176 and 188 to slidably receive a leftward extension of the shaft 184 with the shaft bearing upon a compression type spring 194 contained within the bore 192. The spring 194 is strong enough to push the valve plate 182 into closed position when pressure in the bore 176 and the passageway 175 are equal but is light enough to allow the valve plate 182 to move leftwardly to an open position as soon as a slight pressure differential is developed by operation of the compressor 70 moving air out of the passageway 175 and thus reducing the pressure slightly below ambient.

The piston 187 forms a closed cylinder chamber 196 within the cylinder 188 in conjunction with a cylinder head 198 sealingly secured to the left hand end of the cylinder member 186 as by cap screws 199 or other retaining elements to hold the cylinder head 198, the cylinder member 186 and the valve body 173 in abutting engagement. Upon the stem portion 190 of the piston 187 there is a compression type spring 200 engaged with an interior shoulder at the right hand end of the cylinder member 186 and the underside of the head portion of the piston 187 to bias the piston 187 to the left and maintain the piston in that position until sufficient pressure differential exists between the chamber 196 and the portion of the cylinder 188 to the right of piston 187. With sufficient pressure differential existing the piston 187 will be moved to the right as seen in FIG. 6.

Abuttingly engaged with the central portion of the left hand face of the piston 187 is an elongated cylindrical operating pin 202 slidably received in a central bore of the cylinder head 198 and abuttingly engaged with a valve lever 204 pivotally mounted on the outside of the cylinder head 198 so that axial motion of the pin

102 is transmitted from the lower end of lever 204 with which the pin is engaged to the upper end of the lever 204 to which is pivotally connected a rod end 205 threadedly engaged with a threaded portion of an elongated control rod 206 extending from the inlet valve 170 to a pivotal connection with a speed control or throttle lever 208 by way of a rod end 207. The lever 208 is a throttle member control of a prime mover 210 such as a diesel engine or gasoline engine suitably connected to the screw compressor 70 for powered rotation thereof in a well known manner. The engine throttle member 208 is biased toward the throttle closed for idling position by an extension spring 209 connected between any stationary portion of the engine and the free end of a lever 208 so that when piston 187 moves to the right as seen in FIG. 6 lever 204 will be urged to rotate in a counter-clockwise direction pushing the pin 202 to the right as seen in FIG. 6 maintaining contact with the piston 187 as it moves to the right. This action will bring engine speed down to a predetermined idle speed.

FIG. 7 is a schematic representation of the compressor control circuit of the third embodiment wherein an air oil separator 211 mounted upon and communicating with an air receiver (not shown) is connected via a compressor discharge line 213 to the screw compressor 70 and is otherwise connected by a control air conducting line 214 to a moisture filter and trap 215 having a drain line 216 communicating with a blow-down valve 218 connected by an air conducting line 219 to the throat 174 (see FIG. 6) for the purpose of controlling the action of the blow-down valve as will hereinafter be made plain. It is to be seen that the blow-down valve also has a drain spout 220 as is well known.

The control air line 214 communicates through the filter of element 215 with a main control air line 221 in turn communicating with a manually adjustable pressure regulating valve 222, mounted on an instrument panel 229, which controls air flow into a line 224 leading to the head 198 and communicating therethrough with the cylinder chamber 196 as seen in FIG. 6. The regulator 222 is designed to prevent passage of air therethrough from the line 221 to the line 224 until a preselected minimum pressure has been reached as for instance 100 lbs. per square inch in line 221 before any pressure is added to line 224. Thereafter the pressure in line 224 hereinafter designated secondary or control pressure will build up rapidly as for instance 102 lbs. of primary pressure giving 5 lbs. per square inch secondary pressure and 115 lbs. per square inch primary pressure giving 60 lbs. secondary pressure.

Also connected to the line 224 is an adjustable venting orifice 225 which connects line 224 with the ambient atmosphere for a purpose to be made plain. A gas conducting line 226 is arranged to bypass the regulating valve 222 and connect lines 221 and 224 through a manually operated unloading valve 227 mounted on the instrument panel 229 for a purpose to be described. It is to be noted that the unloader valve 227 is a simple open and shut valve wherein the open position of the handle is marked "start" and the closed position of the valve handle is marked "run."

In the following description of control operation of the third embodiment above described it will be assumed that the compressor is to be operated within a 15 psi pressure differential between 100 and 115 lbs. per square inch discharge pressure but it is to be noted

that any maximum pressure from 70 psig to 125 psig can be obtained by a simple adjustment of the pressure regulator mounted on the instrument panel 229. For a description of the ordinary operation it is to be assumed that the compressor is already warmed up but that the receiver upon which the air oil separator 211 is mounted and with which it communicates has been blown down to ambient pressure.

The prime mover 210 is started up and beings to rotate the compressor 70 at full speed throttle setting with the various parts related as shown in FIG. 6 at the very beginning of the rotation. Very soon thereafter the pressure in the passageway 175 falls below atmospheric and the difference in pressure from bore 176 to throat 174 causes the valve plate 180 to be biased to the left against the biasing of the light spring 194 so that the valve ports 179 are open and air begins to move through the valve body 173 in to the compressor and on through into the air oil separator 211 through the discharge line 213. As the compressor continues to rotate at full speed pressure in the line 213 and consequently in the air oil separator 211 builds up rapidly to and beyond the minimum primary pressure for control, namely 100 lbs. per square inch.

As soon as the primary pressure in the line 221 exceeds the preselected 100 lbs. per square inch the regulator 222 begins to furnish air to the line 224 and therethrough to the cylinder chamber 196. When sufficient pressure has been developed in the chamber 196 the piston 187 begins to move to the right against the force of modulator spring 200 allowing pin 202 to move to the right as seen in FIG. 6. The corresponding movement to the left of control rod 206 in turn reduces the fuel supplied to the prime mover 210 by motion of the throttle 208 activated in the direction of reduced fuel supply by the spring 209.

As long as the amount of compressed air being produced by the compressor 70 exceeds the use of compressed air the pressure in the separator 211 and consequently in the line 221 will continue to rise and as the pressure reaches for instance 110 lbs. primary the regulator 222 will furnish 20 lbs. per square inch secondary pressure to the line 224 and the chamber 196. With this amount of pressure piston 187 will be moved to the right far enough to partially close the valve plate 182 over the ports 179 thus throttling the air supplied to the compressor and partially unloading the compressor while, at the same time, motion of the piston 187 to the right allows the rod 206 to move to the left as seen in FIG. 6 and FIG. 7 with a further reduction in the fuel supply and hence in the speed of the prime mover 210.

As the pressure in the separator 211 approaches the preselected maximum of 115 lbs. per square inch the piston 187 moves farther and farther to the right with greater and greater throttling of the intake air of the compressor passing through the ports 179 with further and further unloading of the compressor 70 as well as further slowing of the prime mover 210 by further restriction of the fuel supplied thereto. When the separator pressure reaches the preset maximum of 115 lbs. per square inch the regulator will supply 60 lbs. per square inch to the secondary air line 224 and the chamber 196 so that the piston 187 is moved as far to the right as necessary to completely close the ports 179 and the compressor is now operating in the unloaded state as well known in the art. With the compressor completely unloaded the rod 206 will have moved the lever

208 to the preselected idle position of the prime mover which will then be turning over without any substantial loading until demand for air lowers the pressure in the separator 211 and the lines 214 and 215 and 221 with the result that the regulator will reduce the pressure in the secondary air line 224 below the 60 lbs. per square inch maximum control pressure and with the further result that the piston 187 will be moved to the left from its extreme rightward position by the force of the spring 200 overcoming the reduced pressure in the cylinder chamber 196.

As the piston 187 moves to the left the control rod 206 will be moved to the right with the result that the lever 208 is moved to the right to a position of greater fuel supply for the prime mover 210 whereupon the speed of the prime mover will be increased. If air use continues to exceed that of the rate of air supply from the compressor, motion of the piston 187 to the left will be continued because of the drop in pressure in line 224 reflecting the drop of pressure at the separator 211 with the result that the valve plate 182 having been first slightly moved to the left by the first motion of the piston 187 allowing the pressure difference to act on the plate 182 will further move to the left until the ports 179 are completely open and the compressor is completely reloaded with the prime mover 210 operating at full speed under full throttle, until the pressure in the separator 211 again exceeds 100 lbs. per square inch and the above described control actions are repeated.

It is to be particularly noted that there are no sudden changes in the throttle settings or the loading of this embodiment and the valve plate 182 is only gradually moved except for the immediate action upon starting up when it rapidly moves from completely closed to completely open. But all times during normal operation of this apparatus the valve plate 182 is gradually moved from closed to open position and vice versa under the influence of the secondary air pressure variation controlled by the separator 222.

The adjustable orifice 225 is a vent which makes control of the pressure in the line 224 and the chamber 196 much easier to maintain than it would be with a dead end configuration since it will provide rapid pressure reduction in the line 224 when the regulator 222 shuts off the supply of air because of reduced pressure in the line 221. The orifice 225 also provides for venting any air leaking past the piston 187 during the unloaded operation and further the orifice 222 provides for venting any condensate moisture that can collect in the chamber 196.

The unloader valve sometimes called the cold start valve 227 is simply opened up at cold start so that primary pressure in the line 221 can be applied directly to the piston 187 which thereby begins to move to the right as soon as the compressor begins to furnish compressed air to the separator 211 with the result that by the time the compressor has reached 60 lbs. per square inch primary pressure the inlet valve 182 will be completely closed and the compressor operating unloaded until such time as the warm-up has been completed at which time the valve 227 will be set on run and will be actually closed and the regulator 222 will take over to keep the pressure in line 224 at zero until the pressure in line 221 exceeds 100 lbs. per square inch. At the moment when the cold start valve 227 has been closed if there is less than 100 lbs. per square inch in the line 221 the regulator 222 will be closed and the orifice 225 will

rapidly bleed off the pressure in the cylinder chamber 196 with the result that the compressor will be immediately brought up to full load condition where it can supply the air necessary to bring the system up to selected pressure.

When the compressor is stopped the pressure in the throat 174 rapidly builds up to somewhat above atmospheric pressure and through the line 219 acts upon the control surface of the blow-down valve 220 to open the blow-down valve 220 and allow the air in the receiver and the separator 211 to pass through the moisture trap portion of the element 215 and rid the trap of the condensate gathered therein by exit through the spout 220.

The advantages of the third embodiment of FIGS. 6 and 7 over the earlier described embodiments reside in the simplicity of the mechanism since only a single piston is employed and no liquid pressures are involved in the operation of the valve 170 with consequent reduction in cost and some increase in reliability of operation.

Preferred embodiments of the valve assembly necessary for the air sealed unloading operation of the method of this invention having been hereinabove shown and described it is to be realized other embodiments of the same principles are possible and envisioned within the scope of the present invention. Therefore it is respectfully requested that this invention be interpreted as broadly as possible limited only by the appended claims.

What is claimed is:

1. A method for controlling the operation of a liquid flooded compressor provided with a normally completely closed gas inlet valve means, a circuit means for delivering flooding liquid under pressure to said compressor and a compressed gas receiving means comprising the steps of: providing a complete closing of such inlet valve means before start up, maintaining said complete closing of such inlet valve means during a start-up period, gradually opening such inlet valve means during a subsequent warm-up period when the temperature of said liquid has increased to a preselected minimum value, maintaining such inlet valve means fully open subsequent to said warm-up period whenever the pressure in such receiving means is below a preselected minimum pressure value; gradually closing such inlet valve means as said gas pressure increases through a preselected range from said minimum value to a predetermined maximum pressure value, completely closing such inlet valve means when said gas pressure reaches said maximum value, and gradually opening such inlet valve means as said gas pressure decreases within said range.

2. The method of controlling compressor operation as specified in claim 1 including the additional step of maintaining said last mentioned complete closing during shut-down of such compressor operation.

3. The method of controlling compressor operation as specified in claim 1 wherein said gradual opening during warm-up is produced by increasing pressure provided by such liquid supplying means.

4. The method of controlling compressor operation as specified in claim 1 including the additional step of reducing pressure in said gas receiving means substantially to ambient pressure after said last mentioned complete closing while maintaining such inlet means completely closed.

5. The method of controlling compressor operation as specified in claim 1 wherein said gradual closing is produced by reaction between liquid pressure from such circuit means and increasing gas pressure in such gas receiving means.

6. Apparatus for controlling operation of a liquid flooded compressor having a compressed gas receiving means and circuit means for delivering flooding liquid under pressure to said compressor, comprising; controllable inlet valve means operable from fully closed to fully open for supplying gas to such a compressor, said inlet valve means being operable at start up of such compressor to open from said fully closed position through various partially open stages during warm up of such compressor to fully open at the termination of said warm up; biasing means for biasing said inlet valve means in a direction to hold said inlet valve means completely closed, force applying means communicating with said liquid delivery circuit means acting on said inlet valve means in opposition of said biasing means for opening said inlet valve means by force of said liquid under pressure, and force modulating means communicating with said gas receiving means and producing a force, on said inlet valve means opposing said

force of said liquid on said inlet valve means for gradually closing said inlet valve means in response to increasing pressure in said gas receiving means.

7. Apparatus as specified in claim 6 additionally comprising circuit means for delivering pressurized gas to said force modulating means at a pressure related to the pressure in said gas receiving means.

8. Apparatus as specified in claim 6 additionally comprising restricted fluid conducting means communicating between said force applying means and a throat portion of said inlet means.

9. Apparatus as specified in claim 6 wherein a valve member of said inlet means is movable by said force applying means and said force modulating means for opening and closing said inlet means.

10. Apparatus as specified in claim 9 wherein said valve member is force balanced under any pressure differential.

11. Apparatus as specified in claim 6 wherein said liquid delivery circuit means comprises flow sensing means for maintaining said inlet means completely closed whenever liquid flow through said circuit is less than a predetermined minimum flow rate.

\* \* \* \* \*

25

30

35

40

45

50

55

60

65

UNITED STATES PATENT OFFICE  
CERTIFICATE OF CORRECTION

PATENT NO. 3,873,239  
DATED March 25, 1975  
INVENTOR(S) Arthur A. Jamieson

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

In column 19, line 19 ... after "means" add -- and --.

Signed and sealed this 27th day of May 1975.

(SEAL)  
Attest:

RUTH C. MASON  
Attesting Officer

C. MARSHALL DANN  
Commissioner of Patents  
and Trademarks