

[54] **COMPOSITE VARIABLE OIL PRESSURE
RELIEF AND COMPRESSOR UNLOAD
VALVE ASSEMBLY**

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[75] Inventors: **Donald D. Schaefer**, Farmington,
Conn.; **James R. Morin, Sr.**,
Springfield, Mass.

Primary Examiner—Carlton R. Croyle

Assistant Examiner—Richard Sher

Attorney—Richard C. Sughrue et al.

[73] Assignee: **Dunham-Bush, Inc.**, West Hartford,
Conn.

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[57] **ABSTRACT**

[21] Appl. No.: **234,123**

An axially shiftable piston is pressure biased by screw compressor oil pressure to a position closing off communication between the intake and the discharge sides of an axial screw compressor but is biased to the unload position by the compressor discharge and a biasing spring acting on the same. The piston carries a fixed or variable oil pressure relief valve to vary the compressor oil pressure with change in compressor thrust load which, in turn, varies with the compressor discharge pressure.

[52] U.S. Cl. **417/281, 417/282, 417/299,
417/310, 418/84, 418/203**

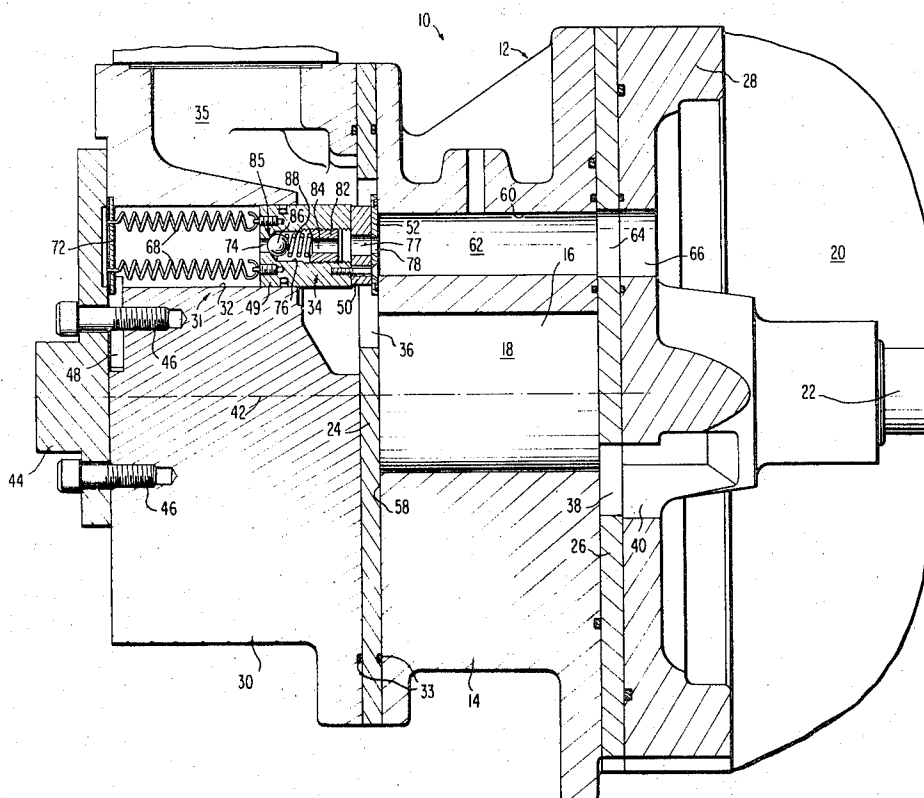
[51] Int. Cl. **F04b 49/02, F04b 49/08**

[58] Field of Search **417/281, 282, 310,
417/299; 418/84, 203**

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13 Claims, 5 Drawing Figures



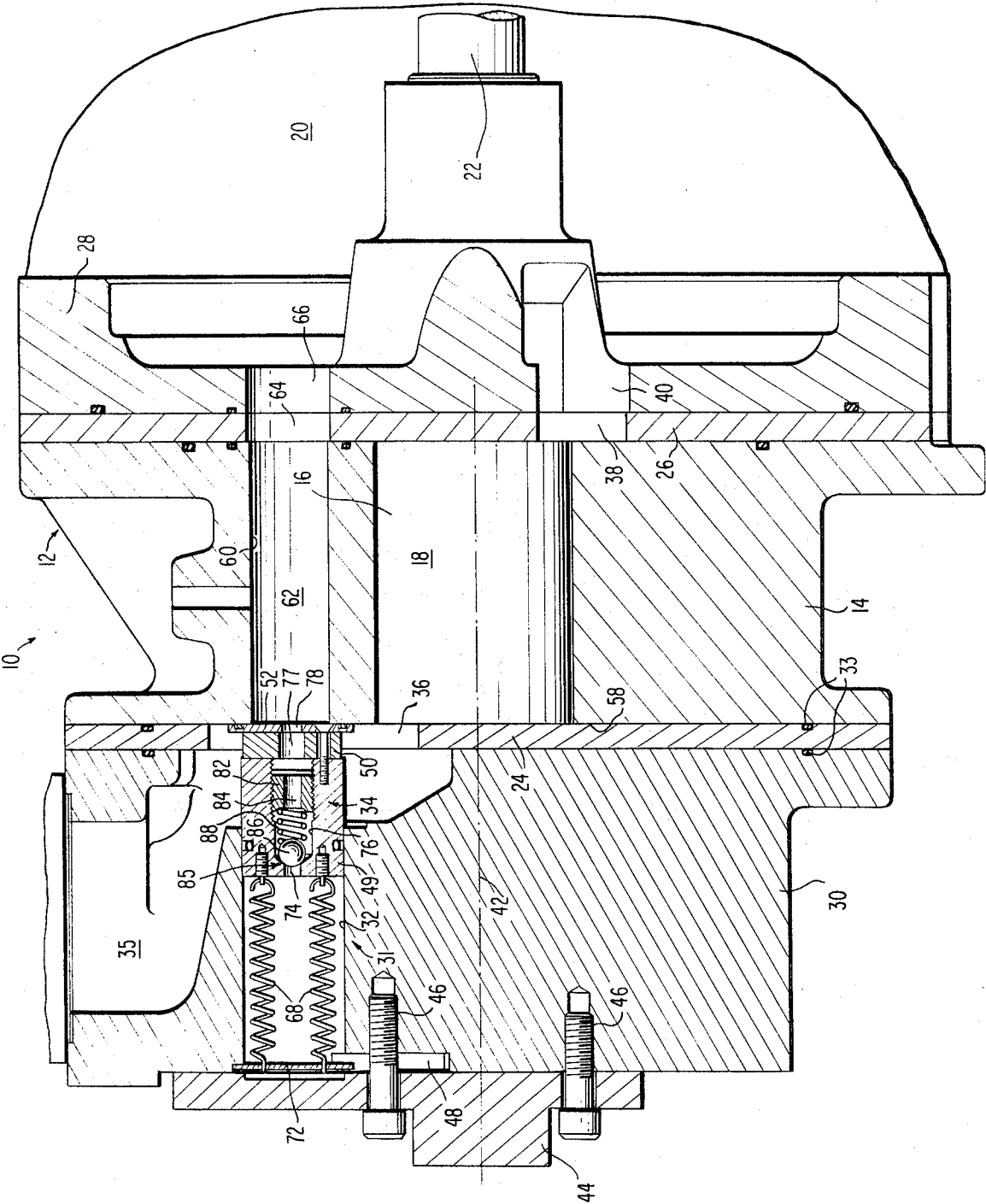


FIG. 1

FIG. 2

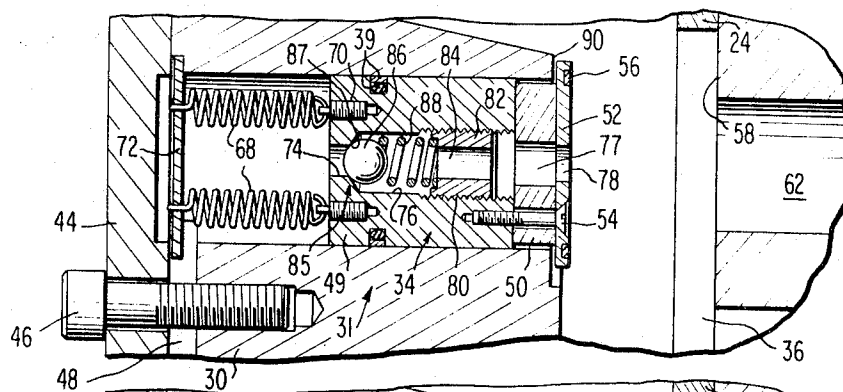


FIG. 3

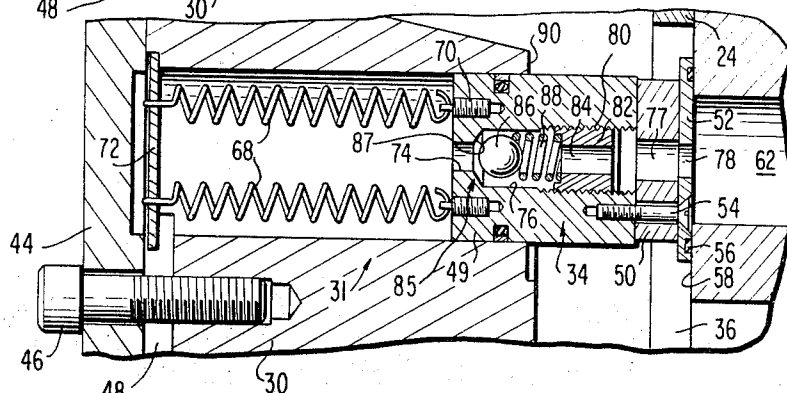


FIG. 4

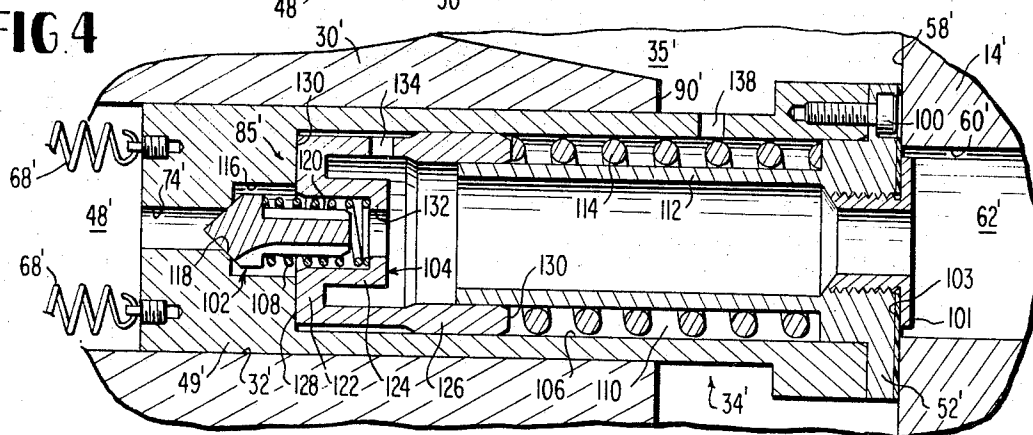
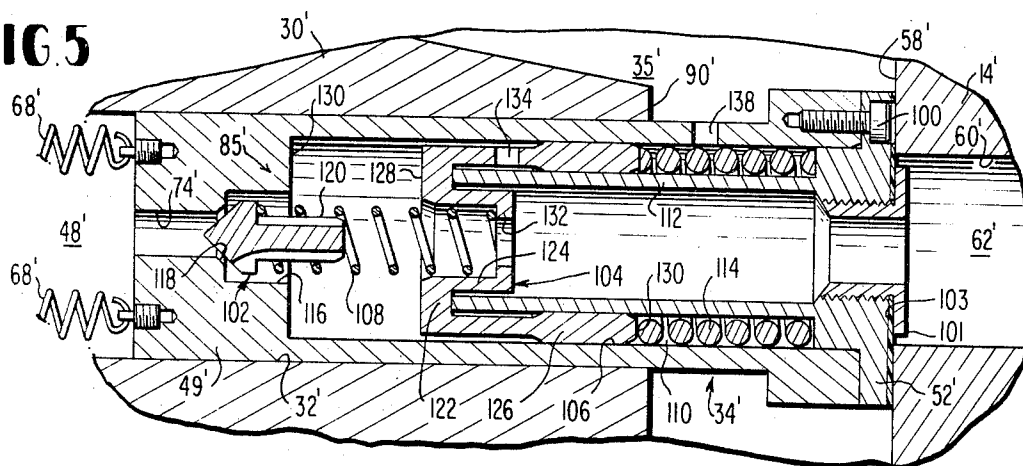


FIG. 5



COMPOSITE VARIABLE OIL PRESSURE RELIEF AND COMPRESSOR UNLOAD VALVE ASSEMBLY

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to axial screw compressors and, more particularly, to such compressors wherein the compressor system oil pressure is employed to create a counterthrust to balance the axial thrust created by the compressor screws acting on the fluid to be compressed. The invention is particularly directed to axial screw compressors for compressing refrigerants.

2. Description of the Prior Art

In axial screw compressors, the proper management of axial forces, generated during the compression process, is mandatory if a successful design is to be achieved. This axial force is the sum of:

A. the resultant gas force due to the pressure difference across the machine, and

B. the contact forces which are the result of one rotor driving the other.

The resultant gas force is the sum of the increased gas pressure in the rotors interlobe spaces, and the gas pressure acting on the end faces of the rotor blades and the supporting and driving shaft ends.

In compressors currently being manufactured, these axial forces are balanced by:

1. the use of matched ball bearing sets on each rotor "to absorb" the thrust;

2. the use of balance pistons, which are grooved oversized shaft ends, which have the compressor fluid discharge plus oil pressure acting on one side and the suction pressure acting on the other, wherein the pressure differential across the balance pistons generates the necessary counterthrust; or

3. the use of a cover overlying the outlet shaft ends to permit the gas pressure to be kept at suction pressure and thus lessen the thrust; or

a combination of all three.

In the current modes of balancing or compensating for the thrust forces, there are unfavorable features to each of these three approaches. For instance, where ball bearings are employed to absorb the thrust, a fixed high side rotor end clearance is provided so that the rotors do not come in contact with the housing. This clearance results in fluid leakage from the high pressure side to the low pressure side of the compressor. This, in turn, causes a loss in compressor capacity, etc. Further, anti-friction bearings are noisy and comparatively expensive.

Where balance pistons are employed, they require a large amount of oil because of their poor sealing ability and, if too much oil is "consumed" by the pistons, the suction gas will be displaced in the compressor by oil, thus the compressor will be flooded with oil and will be unable to pump an efficient amount of refrigerant, if the working fluid constitutes a refrigerant, greatly lowering the efficiency of the compressor. While covers which overlie the shaft ends may be suitable for machines which are being driven from the suction side, this approach at overcoming the thrust created by the operation of the screws is complicated when the screws are driven from the compressor discharge side, since the covers must employ dynamic seals which are expensive and somewhat unreliable.

SUMMARY OF THE INVENTION

The present invention is directed to an axial screw compressor which employs oil pressure acting on balance pistons to provide the counter-thrust necessary to balance out the thrust forces created by the pressure differential across the compressor screws, and by one of the screws being driven by the other. The invention itself is directed to a composite valve assembly which performs the functions of:

1. insuring that the compressor starts unloaded;

2. if for any reason the oil pressure is insufficient, unloading the compressor; and

3. providing an oil pressure relief valve which may be selectively preset or which may vary with the compressor thrust load.

Specifically, the invention is directed to a screw compressor which includes a housing defining a working chamber which carries a pair of intermeshed screws for compressing fluids entering the chamber from an inlet at low pressure, to a higher pressure for discharge at the outlet, and wherein an axial thrust force is developed within the screws during fluid compression. The invention is further directed to such a screw compressor in which oil pressure thrust balancing means counteract this thrust and wherein an oil pump responsive to compressor operation provides the thrust balancing means with pressurized compressor oil.

The improvement resides in a valve assembly responsive to compressor start up and/or low oil pressure for automatically unloading the compressor during compressor start up and at any time the oil pressure drops below a preset value. An unload or bypass passage connects the compressor outlet to the inlet and a shiftable main valve piston is operatively positioned within the passage and is subjected to oil pressure acting on one end of the piston to maintain the unload passage closed, while spring biasing means acts in conjunction with the pressure of the compressor discharge on the other end to open the valve.

The main valve piston carries an oil pressure relief valve which in one form is adjustably set to prevent the oil pressure from rising above a preset maximum level, the pressure relief valve opening to permit the oil to enter the unload passage when the main piston is in valve closed, compressor load position, whereupon the oil mixes with the compressor working fluid discharge. Alternatively, the main piston which itself is shiftable to permit or shut off fluid communication between the discharge side of the compressor and the intake side may be bored and counterbored to receive a floating piston which is spring biased toward the oil pressure end of the main piston by a main power spring. A plunger which closes the bore and is subjected to compressor oil pressure, reciprocates within a central bore of the floating piston, being spring biased by means of a calibration spring towards valve closed position. The oil pressure acting on the seated area of the plunger is opposed by the compressive force of the calibration spring and the discharge pressure of the compressor, while the same discharge pressure acts on the floating piston carrying the plunger and is opposed by the bias of the compression force of the main power spring and the intake suction pressure of the compressor acting on the opposite end of the floating piston. This arrangement permits the control of oil pressure inversely to

working fluid pressure differential across the compressor.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of a portion of an axial screw compressor embodying the improved, combined compressor unload and oil pressure relief valve assembly of the present invention, in one form, with the main piston in valve closed, compressor load position.

FIG. 2 is a sectional view of the combined compressor unload and oil pressure relief valve assembly of FIG. 1, with the main piston in valve open, compressor unload position.

FIG. 3 is a sectional view of the combined compressor unload and oil pressure relief valve assembly of FIGS. 1 and 2, with the main piston in valve closed, compressor load position, but with the oil pressure relief valve in open position.

FIG. 4 is a sectional view of a portion of a rotary screw compressor incorporating a second embodiment of the combined compressor unload and oil pressure relief valve assembly of the present invention in compressor load position and at relatively low compressor working fluid pressure differential.

FIG. 5 is a sectional view similar to that of FIG. 4 of the valve assembly, with the floating piston shifted and the plunger of the oil pressure relief valve in valve open position under high compressor working fluid pressure differential conditions.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Turning to FIG. 1 of the drawings, a first embodiment of the present invention is illustrated as being applied to a rotary screw compressor of which only a portion is shown, indicated generally at 10. Compressor 10 comprises a sealed outer housing including a generally cylindrical casing or housing section 14 carrying at the center thereof a configured working space or working chamber 16 supporting a pair of intermeshed rotors screws such as the female screw 18 which, in this case, is directly driven by means of an electric motor (not shown) fitting space 20 whose rotor is directly coupled to screw 18 by means of an integral connecting shaft 22. The intake and discharge ends of the compressor housing section 14 are sealed by means of end plates 24 and 26 respectively, housing 12 further including an additional plate or section 28 at the discharge side which forms, in this case, an end plate portion of the electric drive motor 20. At the intake or inlet side of the screw compressor 10, there is further provided a relatively thick valve block 30 which is bolted or otherwise fixed to the exposed face of compressor end plate 24, the components 30, 24, 14, 26 and 28 of the compressor carrying appropriate recesses which receive O-rings 33 to insure fluid sealing between these sections of the compressor housing 12. The means for coupling the sections of the compressor housing together are not illustrated, but may in fact constitute bolts or the like which extend through multiple sections. Valve body 30 in addition to carrying within bore 32, the main valve body or piston 34 of a combined compressor unload and oil pressure relief valve assembly 31 which reciprocates therein, is provided with a relatively large intake or inlet passage 35, which constitutes the compressor inlet permitting, in the illustrated embodiment, a refrigerant gas which forms the working fluid for the com-

pressor to pass through openings 36 within end plate 24 and enter compressor working chamber 16 for compression between the female screw 18 and the intermeshed driven male screw (not shown) which is laterally offset from screw 18. The compressed working fluid is discharged from the screw compressor through discharge opening 38 of end plate 26 at the right hand end of the housing 14. Opening 38 is aligned with an opening 40 within the motor end plate 28, allowing the compressed gas to pass over the rotor and stator of the drive motor 20 to cool the same in conventional fashion. This general arrangement of elements is conventional and forms no part of the present invention. Further, while the bearings supporting the screws are not illustrated, screw 18 rotates about its axis and centerline 42, being positively driven by a direct connection with the electric rotor shaft 22 which is in axial alignment therewith. Valve block 30 is further provided at the left hand end with an end plate or cover 44 which overlies bore 32 and is sealably affixed to the outer side wall of block 30 by means of plurality of mounting screws 46. Block 30 is further provided with a recess 48 which carries oil under pressure from a compressor driven oil pump (not shown) which may, for instance, be directly connected to the electric motor rotor shaft 22. Recess 48 extends to bore 32 so that pressurized oil fills bore 32 to the extent permitted by a slidable and sealable cylindrical main valve body or main piston 34. Multiple ring seals 39 fill a peripheral recess within the main piston 34. Main piston 34 constitutes a composite cylindrical valve body including a first section 49 having a diameter slightly less than that of bore 32, a second section 50 of slightly less diameter, and a third section 52 in the form of a relatively thin plate whose diameter is in excess of both sections 49 and 50. Sections 50 and 52 are coupled to the longer section 49 by several screws 54, the section 52 constituting the headed end of the main piston and being provided with an annular O-ring 56 constituting a peripheral seal for contact with the main piston valve seat defined by left hand end wall 58 of the compressor housing section 14 surrounding bore 60. Bore 60 within housing section 14 forms a compressor unload or bypass passage 62 and acts in conjunction with openings 64 and 66 within right hand end plate 26 and the electric motor housing end wall 28, respectively, to permit fluid connection between the intake and discharge sides of the screw compressor. With the main piston 34 in valve closed position as illustrated in FIG. 1, the working fluid for the compressor such as a refrigerant gas, may enter compressor inlet passage 35 within block 30 and pass through the opening 36 within the compressor end plate 24 for compression by the screws and discharge at a higher pressure downstream of the screws via opening 38 within end plate 26. The unload passage 62 is cut off by the main piston 34 since its right hand end abuts the left hand wall 58 of the compressor housing surrounding bore 60. The main left hand section 49 of the main piston 34 has coupled thereto, a pair of tension coil springs 68 by means of mounting screws 70, while the springs at their opposite ends are fixed to a spring retainer or plate 72 which extends transversely across bore 32. The coil springs 68 thus tend to bias the main piston 34 into valve open or compressor unload position as illustrated in FIG. 2. Section 49 of the main piston 34 is centrally bored at 74 and is counterbored at 76, while sections 50 and 52 carry, respectively, axi-

ally aligned bores 77 and 78, thus forming a fluid passageway through the center of the main piston 34. A portion of the counterbore 76 is threaded as at 80 and receives a threaded, cylindrical adjusting screw 82 which itself is centrally bored at 84. Further, a ball 86 of a diameter in excess of bore 74 but less than counterbore 76 is provided within counterbore 76, and is biased to valve closed position against an inclined seat 87 defined by the wall connecting bore 74 with counterbore 76 within section 49 of the main piston 34. Ball 86 is biased by means of a compression coil spring 88, one end of which abuts hollow adjusting screw 82. Thus, the main piston 34 carries a compressor oil pressure relief valve 85 which may be set to open at a desirable oil pressure by axial shifting of the threaded adjusting screw 86 in the embodiment of FIGS. 1-3 inclusive.

As mentioned previously, the axial screw compressor which employs the combined unload and oil pressure relief valve assembly of the present invention, is the type in which oil pressure is employed to counter balance the axial thrust created by operations of the compressor screw compressing the working fluid. As illustrated in FIG. 1, this is the normal condition of the composite valve 31 during sustained operation of the compressor. In such case, the working fluid entering inlet 35 passes through opening 36 within end plate 24 and at intake or suction pressure is compressed conventionally by rotation of the intermeshed screw, as it passes axially from left to right for discharge through aligned openings 38 and 40 into the interior of the electric motor section of the screw compressor assembly downstream and to the right of the compressor itself. The high pressure refrigerant gas, assuming refrigerant is the working fluid, enters the unload passage 62 through aligned openings 64 and 66 and acts directly on the right hand end of the main piston 34 tending to unseat the main piston 34 and permit return of the pressurized working fluid to the intake side of the compressor. The force of the compressed gas acts jointly with the coil springs 68 tending to return the piston 34 to an extreme left hand position as defined by abutment 90 which stops the radially enlarged or headed end of the main piston 34. In contrast, the oil pressure acting on the left hand end of the main piston 34 tends to maintain piston 34 in the extreme right hand position or compressor load position.

Prior to reaching the condition illustrated in FIG. 1, at start up, the suction, discharge and oil pressure are all equal, since the axial thrust in the compressor is in part due to the difference in gas pressure across the compressor. The low intake pressure at the inlet may vary from zero to 100 psi and the high discharge pressure at the outlet of the compressor may range from 200 to 400 psi. It is seen, therefore, that it is undesirable to let this difference in pressure occur until the coil pressure has built up to a sufficient level to counteract the thrust load via the balance pistons (not shown) on the rotor ends.

The pressurized oil within recess 48 and bore 32 to the left of the main piston 34 is supplied by means of an oil pump (not shown) connected directly to the electric motor 20 and oil pressure is established almost instantaneously. However, the present invention prevents the gas and contact thrust from adding prior to oil pressure build-up to the point where the oil pressure establishes a counterthrust in unusual instances where the oil pressure is slow in building, where the compres-

sor is started after long shut downs or where there is high percent of refrigerant in the oil. Since the gas suction pressure is equal to the gas discharge pressure and oil pressure at start-up, the only force acting on the main piston 34 is that provided by coil springs 68. The coil springs therefore shift the piston 34 from the extreme right hand position where the headed end abuts end wall 58 of screw compressor housing 14 adjacent the left hand end of bore 60 to a position in which the headed end contacts abutment 90 unloading the compressor, with the compressor discharge recirculated to the compressor inlet, as seen in FIG. 2. However, almost immediately the oil pressure builds up sufficiently to force the piston to move from the position shown in FIG. 2 to the position shown in FIG. 1 with the oil pressure relief 85 in closed position in the absence of excessive oil pressure.

It is to be noted, that the motor horse power required to start and run the compressor is proportional to the pressure difference across the compressor at the time of starting and thus by eliminating the pressure difference at start up, this reduces the necessary horse power requirement at a time in which the power requirements are normally the highest. Thus, the compressor can start easier, since it is unloaded at start with less amperage requirement. Momentarily, the lack of oil pressure permits start up with the discharge gas being bypassed to the suction side of the compressor, but almost immediately thereafter, oil pressure rises sufficiently to shift the main power piston 34 from left to right to seal the bypass or unload passage 62 from the compressor intake passage 35.

The combined valve assembly 31 of the present invention permits the compressor to be unloaded automatically at any time during operation when the oil pressure falls below a preset value. For instance, if the oil pressure is normally in excess of 40 psi above compressor discharge pressure, but if it should fall below approximately 10 psi, the combination of the force created by the coil springs 68 and the compressor discharge gas acting on the right hand end of the main piston 34 causes the main piston 34 to move from right to left, again unloading the compressor and permitting recirculation of the compressed gas to the discharge side of the screws. Oil pressure may drop as a result of loss of oil in the sump, failure of the oil pump (not shown), blockage of the oil pump inlet, etc. As the bearing loads are partially a result of the pressure difference across the compressor, it is desirable to eliminate this loading if the relative oil pressure to the bearings falls below a safe level, for instance, 25 psi. In contrast to commercially available oil pressure safety devices of the type such as the externally mounted bellows operated electrical contacts, which have a 30 to 90 second delay built into them to eliminate nuisance stoppage of the compressor, the mechanism of the present invention immediately lessens the bearing loads in the unprotected first zero to 90 seconds before the external safety device shuts the compressor down. If there is an oil failure of short duration, that is from zero to 90 seconds, the combined valve assembly 31 of the present invention will partially protect the compressor without shutting it off by simply unloading the compressor but permitting the compressor to continue to operate in unloaded condition.

Turning to FIG. 3, the contrast in conditions for the valve assembly 31 and those shown in FIG. 1, illustrates

the operation of the oil pressure relief valve 85 which forms a portion of combined valve assembly 31. Rotary screw compressors, even where the pressurized oil system does not function to absorb or counteract the axial thrust set up in the rotary screws, employs a pressure relief valve to maintain the oil pressure at a preset level depending upon the design requirements of the compressor. In air conditioning and refrigeration use, rotary screw compressor oil pumps are usually oversized by 50 to 100 percent so sufficient oil and oil pressure will be available upon bearing wear, and where oil requirements go up, and also to insure adequate lubrication when the oil becomes less viscous under certain conditions that are outside of the design limits of the machine such as when the oil is extremely hot or where there is an excessive amount of refrigerant within the oil. The present invention is advantageously directed to the incorporation of oil relief valve 85 as a part of the combined valve assembly 31, and further, to a valve whose relief pressure is adjusted by increasing or decreasing the compression of the coil spring 88 which abuts the ball 86. Assuming that the maximum relative oil pressure desired for all oil functions such as lubrication, and in particular, oil pressure balancing of the axial thrust of the screws is in the neighborhood of 80 psi above compressor discharge. The screws 82 can be shifted axially to the left to an extent where the ball 86 will unseat from the end of the bore 74 at oil pressure of 100 psi for instance. During this time, the oil pressure acting on the left hand end of the main piston 34 maintains the valve in closed or compressor load position, that is, to its extreme right hand position, as illustrated in FIG. 1. If the oil pressure exceeds 100 psi, the ball 86 unseats to permit bleeding of the oil into the compressor discharge via unload passage 62 to the extent necessary to maintain the oil pressure at 100 psi, as evidenced in FIG. 3.

One of the characteristics of the axial screw compressor is that the amount of thrust varies as the working fluid pressure difference across the compressor changes, thus it is possible to set the oil pressure relief valve to provide the proper oil pressure to the balance pistons at a given pressure differential across the compressor to offset the thrust load. It is most desirable therefore to provide a relief valve that varies in its setting according to the change in pressure difference across the compressor and further to provide a pressure relief valve that provides the correct pressure to balance the thrust load effectively, regardless of the pressure difference across the compressor. It was determined that where a 330 psi pressure differential exists across the compressor, the oil pressure permitting the proper thrust balancing via the balance pistons would necessarily be of the order of 40 psi above compressor discharge pressure, while for a pressure differential across the compressor on the order of 110 psi, a relative oil pressure of 80 psi is required to permit the balance pistons to properly balance out the compressor thrust. Thus, the alternate form of the composite valve 31' of the present invention, illustrated in FIGS. 4 and 5, not only automatically varies the oil pressure to the needs of the hydraulic balance pistons, but because the oil pressure requirements are such that an increase in compressor pressure difference requires less relative oil pressure, the oil relief valve 85' in the alternate embodiment of the present invention operates inversely in terms of the pressure difference across the compressor.

Turning to FIGS. 4 and 5, it is noted that in terms of this embodiment, the casing or housing section of the axial screw compressor is identical to that of the prior embodiment. Like elements carry like primed numerical designations. Thus, compressor housing section 14' carries a bore 60' defining a compressor bypass or unload passage 62' which is sealed off at its left hand end by the headed end 52' of the main piston 34'. In this case, the main piston 34' is a composite body, formed of steel or like metal including joined sections 49' and 52', which is carried within bore 32' of a valve block 30'. Portion 48' of bore 32' receives the pressurized oil which creates a force acting on the left hand end of the main piston 34' in opposition to the compressor discharge which acts on the right hand end of piston 34'. Again, the main piston 34' reciprocates between a closed position in which its right hand end abuts a portion of the end face 58' of housing section 14' surrounding bore 60' to shut off the bypass or unload passage 62' from the compressor inlet passage 35', but upon loss of oil pressure, the coil springs 68' acting in conjunction with the compressor discharge move the main piston 34' to a position in which the shoulder formed by the headed end 52' of the valve body contacts the abutment 90' to limit further movement to the left and permit the compressor discharge to re-enter the intake side of the compressor.

Thus, insofar as unloading the compressor at start or at any time that the oil pressure decreases below a preset minimum figure, the embodiment illustrated in FIGS. 4 and 5 operates identically to that illustrated in FIGS. 1-3 inclusive. As mentioned previously, the main piston 34' consists of a relatively long section 49' to which is coupled the enlarged diameter right hand end section 52' by means of a plurality of screws 100 whose headed ends are recessed within section 52'. A hollow threaded plug 101 fixes a thin sealing disc 103 to the end of piston section 52'. The variable oil pressure relief valve 85' which is carried centrally of and within the main piston 34', consists essentially of four parts in addition to the main piston 34'. A plunger 102 is axially shiftable with respect to a floating piston 104 which supports the same, the floating piston in turn being seated within counterbore 106 of section 49' of the main piston 34', the plunger being biased to the left and to valve closed position by a calibration coil spring 108. In turn, the two part main piston 34' defines an annular cavity 110, which lies between an elongated inner tubular wall portion 112 of section 52' and counterbore 106 of section 49'. It is within this annular cavity 110 that the floating piston 104 which is irregularly double U-shaped in cross section is permitted to slide and may be positioned to the left by the main power spring 114 taking the form of a compression coil spring. Section 49' of the main piston 34' is provided with a small diameter bore 74' and a first counterbore 116 in addition to the second counterbore 106, the end of bore 74' being beveled adjacent the headed end of plunger 102 to define the valve seat 118 for the variable oil pressure relief valve 85'. The plunger 102 is provided with a reduced diameter portion 120 about which is concentrically positioned the forward end of the calibration spring 108. The floating piston 104 is provided with a base portion 122 which connects the spaced parallel tubular side walls 124 and 126, the outer side wall 126 being of extended length and of a thickness generally on the order of the annular recess 110 receiving the same, permit-

ting it to be sealable and slidable therein. The end of the inner tubular wall portion 112 of main piston 34' is positioned in the path of base 122 to limit the right hand movement of the floating piston 104 which compresses the main power spring 114 during movement from left to right. Further, face 128 of the base portion 122 of the floating piston abuts shoulder 130 defined by counterbores 116 and 106 of section 49' of the main piston 34' to limit movement of the floating piston to the left. The tubular wall 124 of the floating piston is provided with a flange portion 132 against which the calibration spring 108 abuts, the calibration spring 108 concentrically surrounding the plunger 102 at its forward end while the floating piston concentrically surrounds the calibration spring 108 at its rear to increase the compressive force acting on the plunger and to maintain it against the valve seat 118 as the floating piston shifts left from its extreme right hand position to the point where face 128 abuts the shoulder 130 of the main piston section 49'. The floating piston is provided with at least one radial opening 134 permitting the compressor discharge to act at all times on the left hand end face of the floating piston and in opposition to the compressive force of the main power spring 114 and the suction gas pressure acting on the right hand end face 130 of the outer tubular wall 126 of the floating piston. Further, section 49' of the main piston is provided with at least one radial opening 138 which permits gas inlet pressure to reach the annular cavity 110 carrying the main power spring 114 and the tubular outer wall 126 of the floating piston. Sufficient clearance is provided between the plunger 102 and the floating piston 104 which carries the same, such that when the piston is displaced to the right of valve seat 118, the pressurized oil may escape through bore 74' and enter the unload passage 62' through the hollowed interior of the floating piston 104.

FIG. 4 shows the valve assembly 31' of the present invention in its high relative oil pressure mode, while FIG. 5 shows the composite valve in its low relative oil pressure mode. Reference to FIG. 4 is made with the realization that under this mode of operation, the working fluid pressure difference across the compressor and acting on floating piston 104, is relatively small, in which case the main power spring 114 maintains the floating piston 104 at its extreme forward position, in which case, face 128 of the floating piston 104 abuts the shoulder 130, causing the calibration spring 108 to be compressed to the point where it is exerting its greatest spring force and thus the relative oil pressure needed to unseat the plunger 102 from seat 118 must be at its highest in terms of the setting for the valve assembly. In the illustrated embodiment, a spring force of 3.92 pounds maintains the plunger in oil pressure relief valve seated position and the oil pressure needed to unseat the plunger would necessarily be 80 psi above the gas discharge pressure. If it is assumed the working area of the plunger 102 is 0.0491 square inches ($80 \text{ psi} \times 0.0491 \text{ square inches}$) = to 3.92 pounds, which is equal to that supplied by the calibration spring 108. In the illustrated version, the mechanism works linearly, although a non-linear mechanism could be made by using non-linear springs or two or more linear springs with different spring rates.

In contrast, turning to FIG. 5, the pressure differential between the discharge pressure of the compressor and the inlet pressure is sufficiently high to shift the

floating piston 104 from its extreme left hand position to its extreme right hand position, wherein the end of the tubular wall 112 abuts the base 122 of the floating piston. In this case, the main power spring 110 is compressed to its fullest extent. At the same time, the calibration spring 108 lengthens and its spring force reduces to about 1.96 pounds. The working area of the plunger is again 0.0491 square inches for the illustrated embodiment, and oil pressure above 40 psi will unseat the plunger from its valve seat 108, relieving the oil, since the force acting against the plunger would be $0.0491 \text{ square inches} \times 40 \text{ psi}$ = 1.96 pounds and the forces on both sides of the plunger would be equal at that point. It is at this point that the plunger starts to relieve as shown, and a constant relative oil pressure is thus maintained at 40 psi, assuming that the pressure differential across the compressor remains the same and at a relatively high value. It is noted that the power spring 114 and the calibration spring have the same travel between extreme limits, although of course it is not necessarily to maintain this relationship.

From the above description, it is appreciated that, in particular, with the embodiment of FIGS. 4 and 5, the combined valve assembly permits the compressor to start unloaded, where the compressor has a built-in compression that is not relieved, unloads the compressor when oil pressure drops below a preset level, maintains the oil pressure setting in the compressor oil system and varies the oil pressure with change in compressor thrust load to provide the necessary varying counterthrust to the changing axial thrust experienced by the screws during compressor operation.

What is claimed is:

1. In an axial screw compressor including: a housing defining a working chamber carrying a pair of intermeshed screws for compressing a working fluid from a lower intake pressure to a higher discharge pressure with thrust forces being created within said screws during compressor operation, oil pressure thrust balancing means acting on said screws and pump means responsive to compressor operation for supplying pressurized oil to said thrust balancing means, the improvement comprising:

a valve assembly including means responsive to compressor discharge pressure and compressor oil pressure for automatically unloading said compressor during compressor start up, automatically unloading said compressor upon reduction in compressor oil pressure below a predetermined minimum value after start up, and automatically preventing the compressor oil pressure from rising above a predetermined maximum value during compressor operation.

2. The axial screw compressor as claimed in claim 1, wherein said means is responsive to variation in the pressure difference of the working fluid between compressor intake and discharge for automatically varying the compressor oil pressure.

3. The axial screw compressor as claimed in claim 1, wherein said means is responsive to variation in the pressure difference of the working fluid between the compressor intake and compressor discharge, for inversely varying the compressor oil pressure.

4. The axial screw compressor as claimed in claim 1, wherein said axial flow compressor includes a working fluid bypass passage connecting the discharge side of the compressor to the intake side, said valve assembly

includes a shiftable valve member, said valve member being operatively positioned with respect to said bypass passage and said pressurized oil such that the compressor discharge pressure tends to move said valve member to compressor unload position, and said compressor oil pressure tends to move said valve member to compressor load position, and said valve assembly further includes spring biasing means tending to move said valve member to compressor unload position in the absence of compressor oil pressure.

5. The axial screw compressor as claimed in claim 1, wherein: said compressor includes a bypass passage fluid connecting said discharge and intake sides of said compressor, and said valve assembly comprises: a valve block including an elongated bore, a cylindrical main piston slidably and sealably positioned within said bore and having one end in a first position closing off one end of said bypass passage to load the compressor with the compressor discharge acting on that end of the piston when in valve closed position, means subjecting the other end of said piston to said pressurized oil for maintaining said main piston in said first position and spring means acting on said main piston in opposition to said oil pressure to shift said piston to a second position within said bore to permit said discharge passage to be in fluid communication with the compressor intake, whereby; upon loss of oil pressure or during compressor start up, said spring means maintains said piston at said second compressor unload position.

6. The axial screw compressor as claimed in claim 3, wherein: said compressor includes a bypass passage fluid connecting said discharge and intake sides of said compressor, and said valve assembly comprises: a valve block including an elongated bore, a cylindrical main piston slidably and sealably positioned within said bore and having one end in a first position closing off one end of said bypass passage to load the compressor with the compressor discharge acting on that end of the piston when in valve closed position, means subjecting the other end of said piston to said pressurized oil for maintaining said piston in said first position and spring means acting on said main piston in opposition to said oil pressure to shift said piston to a second position within said bore to permit said discharge passage to be in fluid communication with the compressor intake, whereby; upon loss of oil pressure or during compressor start up, said spring means maintains said piston at said second compressor unload position.

7. The axial screw compressor as claimed in claim 5, wherein said main piston is centrally bored to fluid connect said bypass passage with the pressurized oil acting on the end of said piston opposite that subjected to compressor discharge, and said valve assembly further includes an oil pressure relief valve carried by said piston for closing off said bore, whereby the oil pressure is maintained below a certain predetermined maximum value by bleeding off compressor oil into the compressor discharge.

8. The axial screw compressor as claimed in claim 5, wherein: said main piston is centrally bored and counterbored, to effect fluid communication between respective ends of said piston, a ball of a diameter less than the counterbore but greater than the bore is carried within said counterbore, said counterbore is axially threaded, a hollow screw is threadably received by the threaded counterbore, and a coil spring is axially compressed between the ball and the screw to permit set-

ting of said valve to a predetermined maximum desired oil pressure.

9. The axial screw compressor as claimed in claim 5, wherein: said main piston is hollow for placing the ends of said piston in fluid communication, an oil relief valve plunger is slidably and sealably positioned within hollow main piston, a floating piston is slidably positioned within said main piston, a calibration compression spring is positioned coaxially between said floating piston and said plunger to bias said plunger and floating piston in opposite directions, a power spring is positioned between said floating piston and the main piston end wall remote from said plunger for biasing said plunger and said floating piston in a common direction to close off fluid communication between said main piston ends, and said valve assembly includes means permitting the working fluid pressure differential between the compressor inlet and the compressor discharge to act on the floating piston and to thereby control the extent of compression of said calibration spring, whereby the maximum oil pressure permitted by said oil relief plunger varies inversely with the working fluid pressure differential across the compressor.

10. The axial screw compressor as claimed in claim 9, wherein said main piston includes means defining an annular chamber at the end of said piston remote from said plunger, said floating piston includes a tubular wall portion slidably received within said annular chamber, and said main piston includes means permitting compressor working fluid discharge pressure to act on one end of said floating piston tubular wall portion to compress said power spring, and said main power piston further includes means for fluid communicating the compressor working fluid intake with said annular chamber to permit the compressor inlet pressure to act on the other end of said tubular wall portion of said floating piston and in conjunction with said power spring.

11. The axial rotary screw compressor as claimed in claim 10, wherein said power spring comprises a coil spring carried by said annular chamber having one end in contact with the end of said main piston, and the opposite end in contact with said floating piston tubular wall portion.

12. The axial screw compressor as claimed in claim 1, wherein: said compressor housing section carrying said intermeshed screws is bored axially the length of the same to define a bypass passage, a valve block is fixed to the inlet end of said axial screw compressor housing section and includes means defining a compressor working fluid intake passage, said block includes a bore coaxial with the bypass passage, acts as an extension of the same and intersects said compressor inlet, a main piston is slidably and sealably carried by said bore within said valve block and includes means for shutting off the end of said bypass passage in a first position but permitting fluid communication between said bypass passage and said compressor intake passage when in a second position, spring means coupled to said main piston acts to move said main piston to compressor unload position, means directs compressor oil to the end of said main piston remote from said bypass passage for producing a force in opposition to said springs and the compressor discharge acting on the opposite end of said main piston, said main piston is bored and counterbored to define a fluid passage connecting the ends of the same, means define an annular cavity

13

within said main piston, a floating piston including a tubular portion is slidably and sealably received within said annular cavity, means permit compressor discharge to act on one end of said tubular portion of said floating piston, means fluid communicate the compressor intake to said annular chamber to permit the working fluid at the compressor inlet pressure to act on the other end of said floating piston, a power compression coil spring is coaxially carried within said annular cavity to bias said floating piston away from said fluid by-pass passage end of said main piston passage, a plunger is slidably carried by said floating piston and adapted to seat within said bore and acts as a pressure relief valve, and a calibrating compression coil spring is concentrically carried by said plunger with one end in contact with the same and the other end abutting said floating piston, whereby at high compressor working fluid pressure differential, said floating piston compresses the power the to lower the biasing force acting on the plunger and thereby relieve the oil pressure at a relatively low pressure value, while at low working fluid pressure differential between the compressor intake and compressor discharge, the power spring forces the floating piston to compress the calibrating spring and

14

increase pressure at which the oil pressure relief valve operates and to thereby effectively control oil pressure thrust compensation in terms of the axial thrust forces on the compressor screw resulting from operation of the compressor.

13. The axial screw compressor as claimed in claim 12, wherein said main piston comprises first and second cylindrical members including spaced concentric tubular wall portions coupled at the end closest to said by-pass passage to define said annular cavity, and said floating piston comprises spaced inner and outer tubular wall portions coupled together by a base portion and said main piston includes a bore and two counterbores in that order and wherein the shoulder between the two counterbores defines a stop determining the maximum compression of said calibrating spring, while the end of inner tubular wall portion of said main piston contacts the base portion member coupling the inner and outer tubular wall portions of said floating piston to limit the extent of expansion of said calibrating spring compressively positioned between the floating piston and the plunger.

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