

[54] **ROTARY CONTROL VALVE FOR SCREW COMPRESSORS**[75] Inventors: **Michael G. Herschler; Robert R. Anderson; John C. Shoop**, all of Quincy, Ill.[73] Assignee: **Gardner-Denver Company**, Quincy, Ill.[22] Filed: **Nov. 12, 1973**[21] Appl. No.: **415,279**[52] U.S. Cl. **418/87, 418/97, 418/201**[51] Int. Cl. **F01c 21/04, F04c 29/02, F01c 1/16**[58] Field of Search..... **418/87, 97, 159, 201-203; 417/310, 440**[56] **References Cited****UNITED STATES PATENTS**

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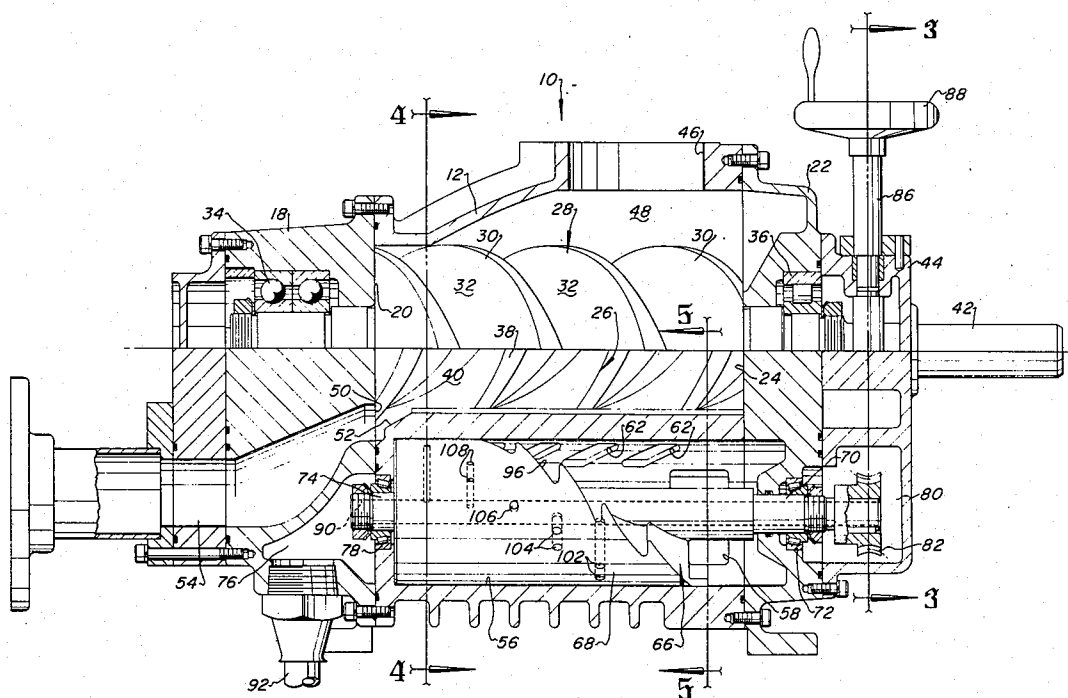
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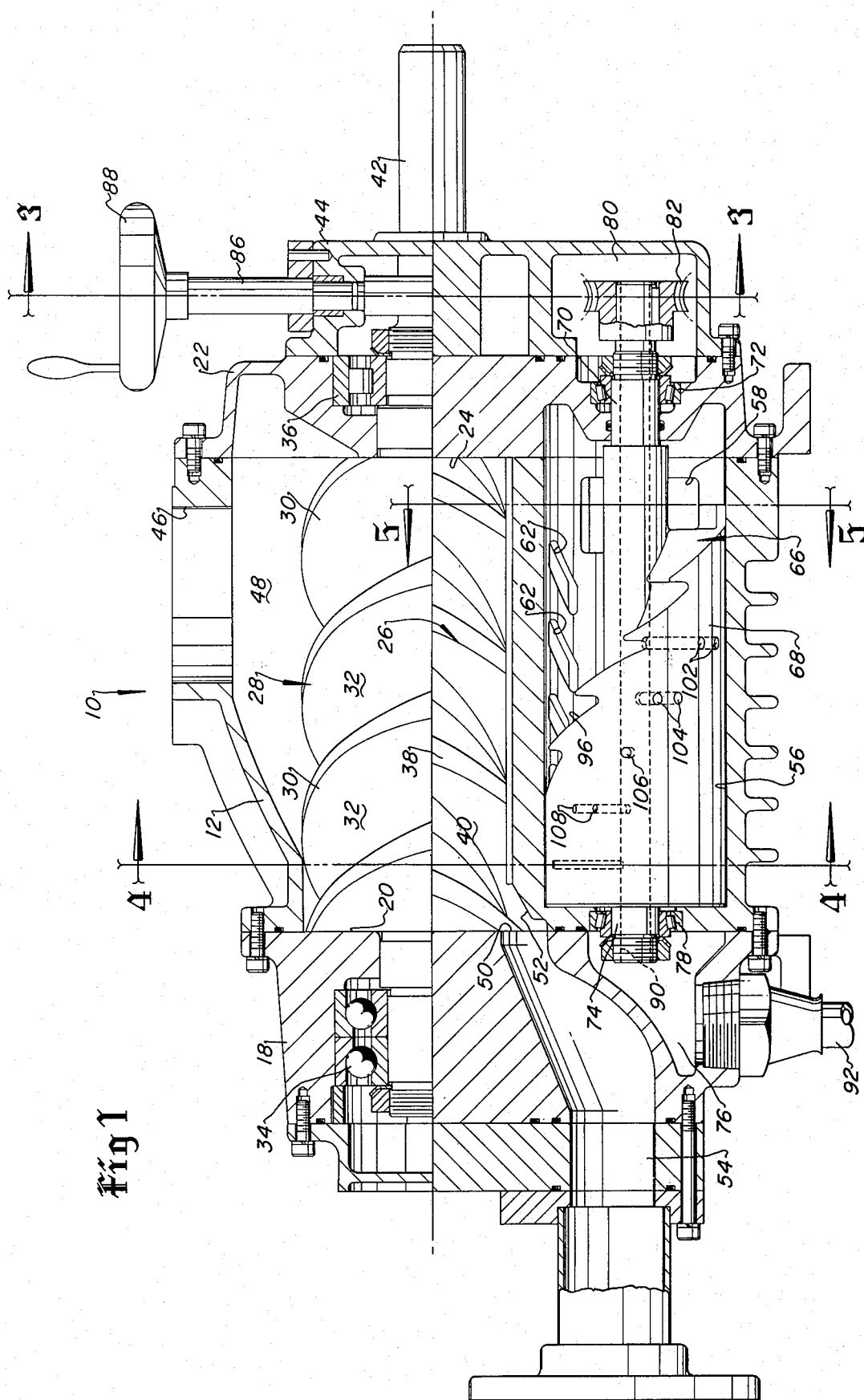
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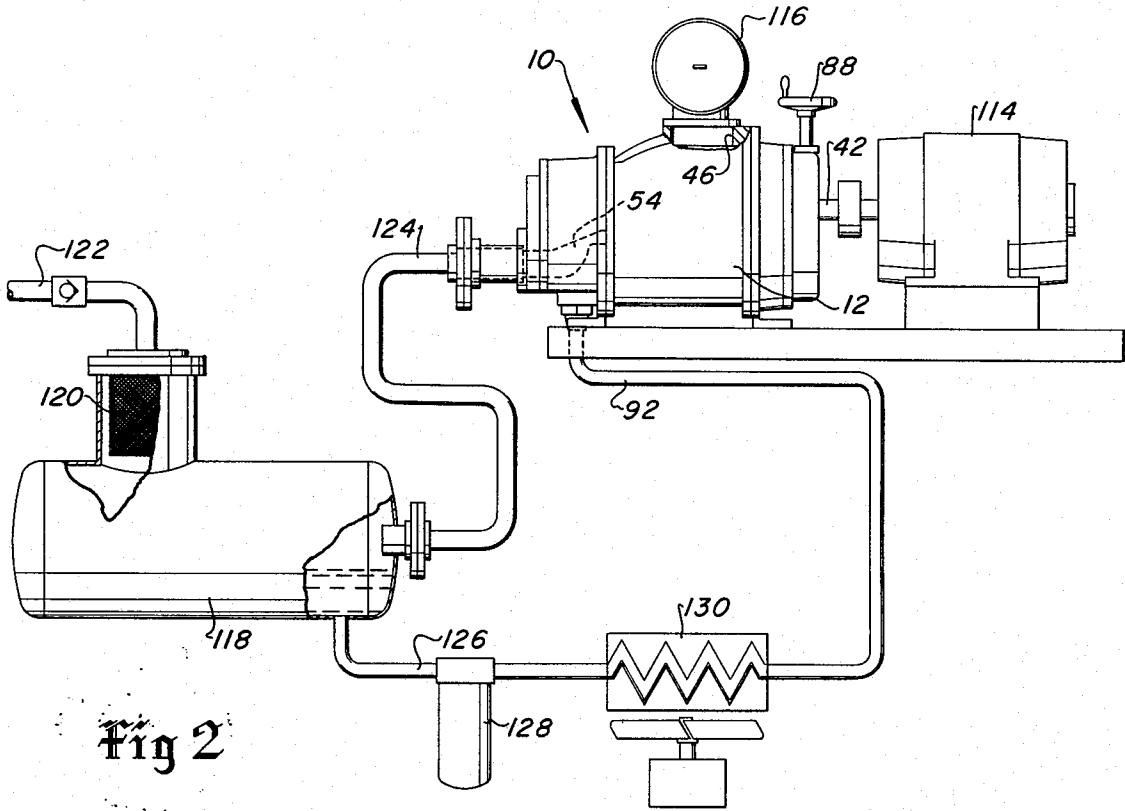
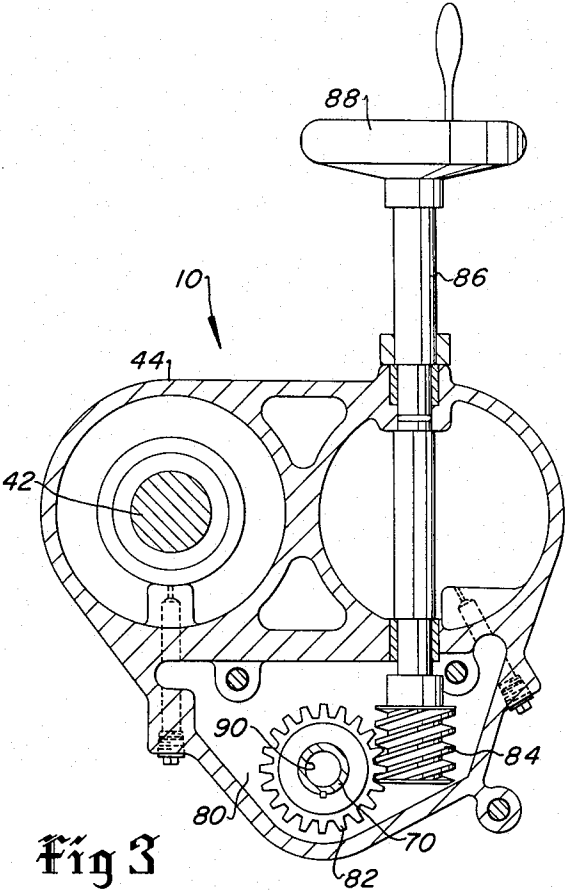
Primary Examiner—John J. Vrablik
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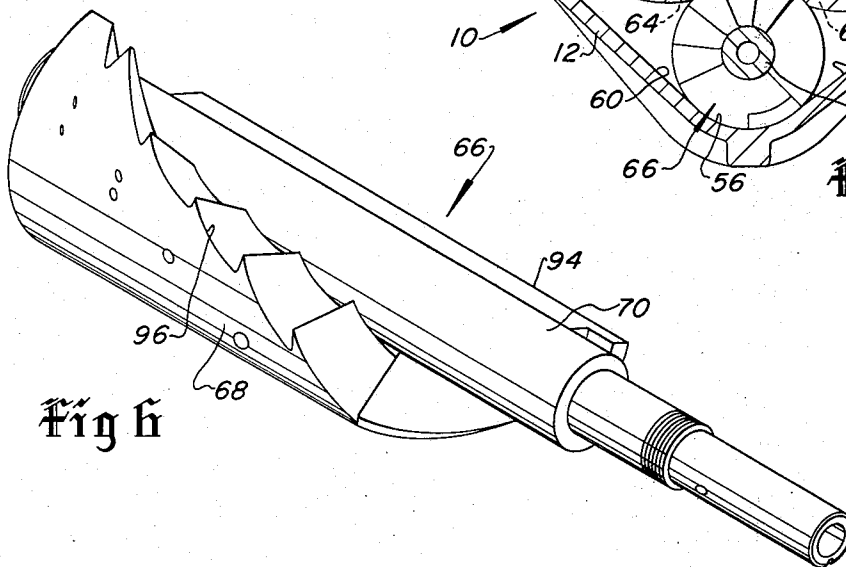
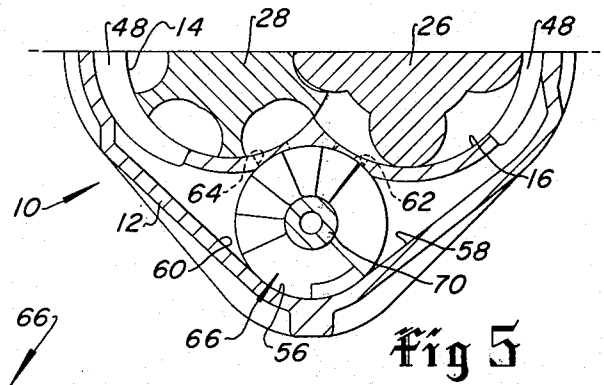
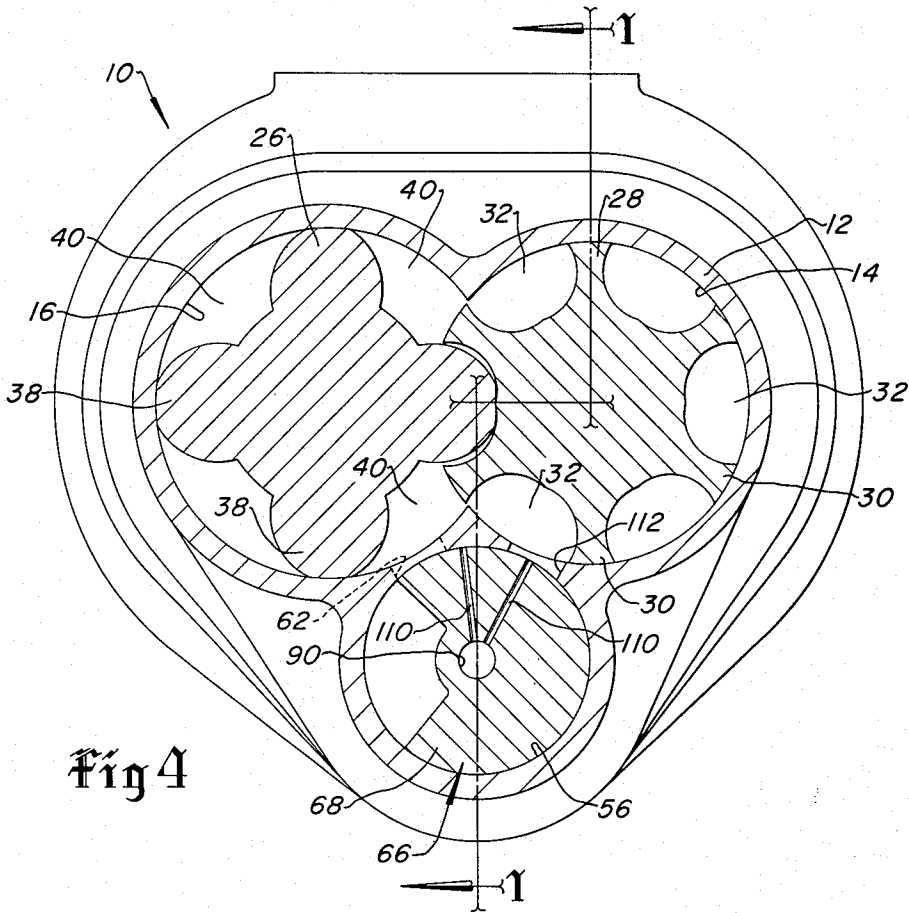
[57] **ABSTRACT**

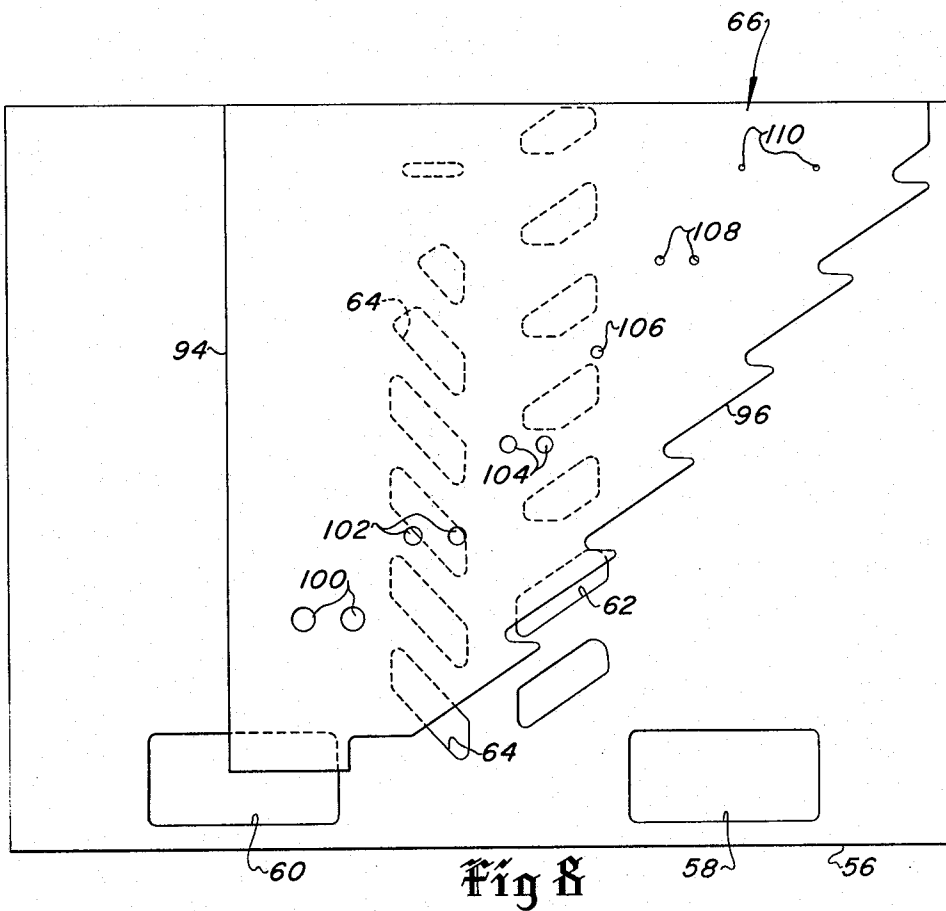
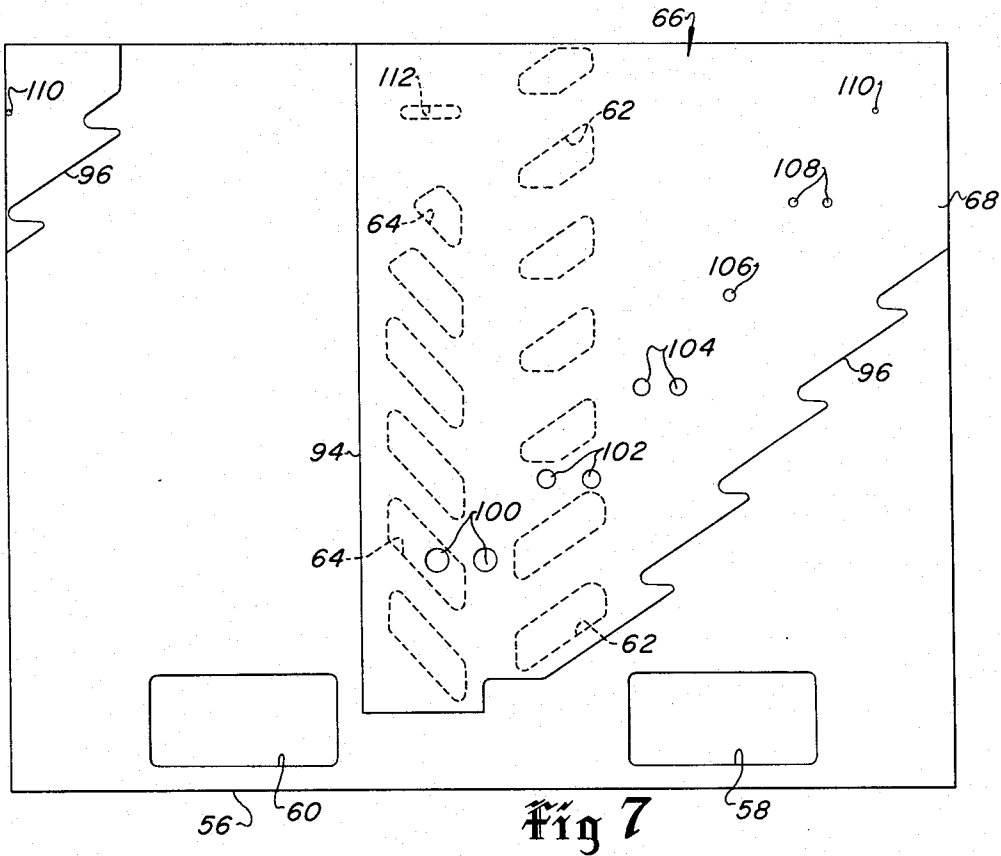
A liquid injected helical screw gas compressor having a capacity control valve comprising a cylindrical plug member fitted in a chamber which is adjacent to and in communication with the compressor working chambers by way of a series of auxiliary ports. The control valve member includes a control edge coating with the auxiliary ports for controlling the gas throughput and the built-in volume ratio of the compressor. The control valve member also includes a series of liquid injection passages spaced along and around the perimeter of the cylindrical plug. The liquid injection passages are selectively placed in communication with the auxiliary ports in accordance with the control valve position for controlling the position and rate of injection of liquid into the working chambers.

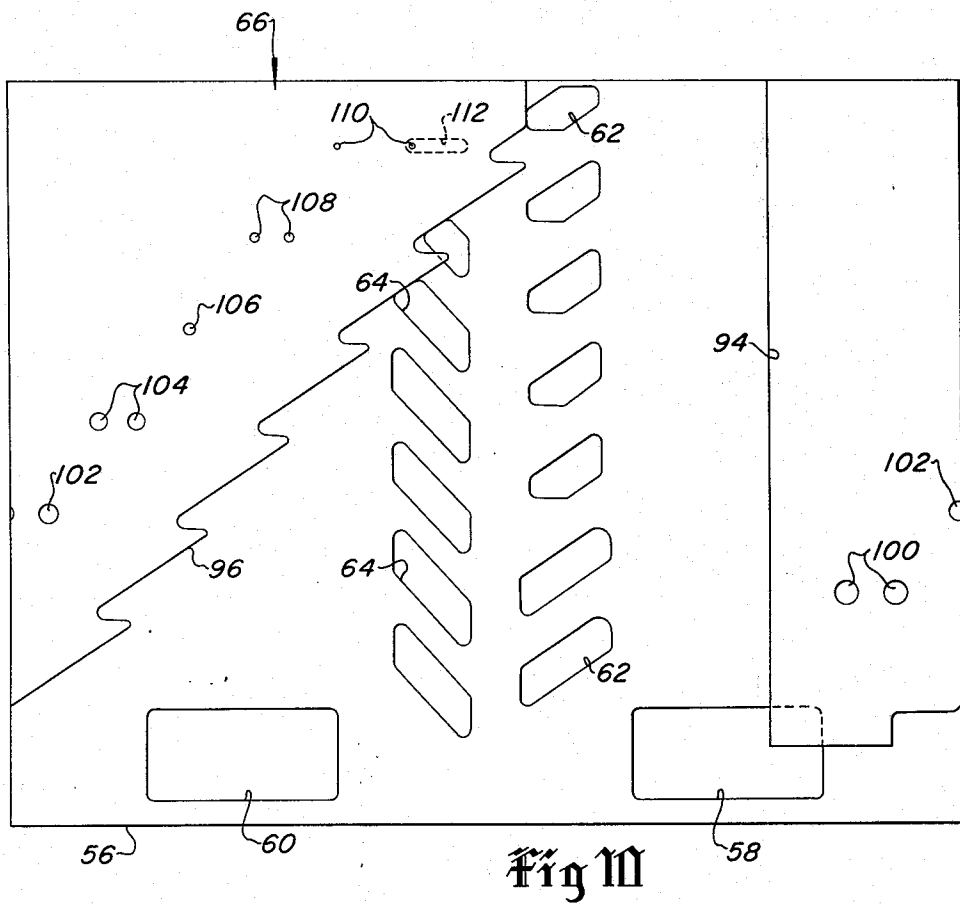
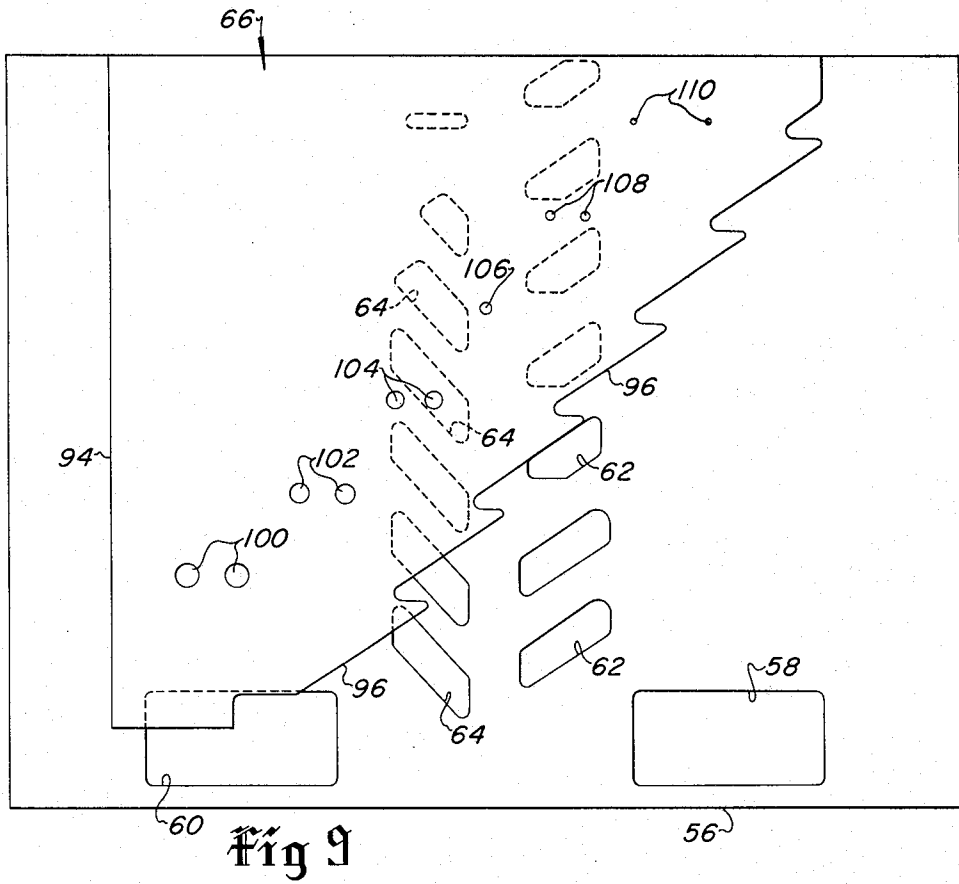
11 Claims, 15 Drawing Figures

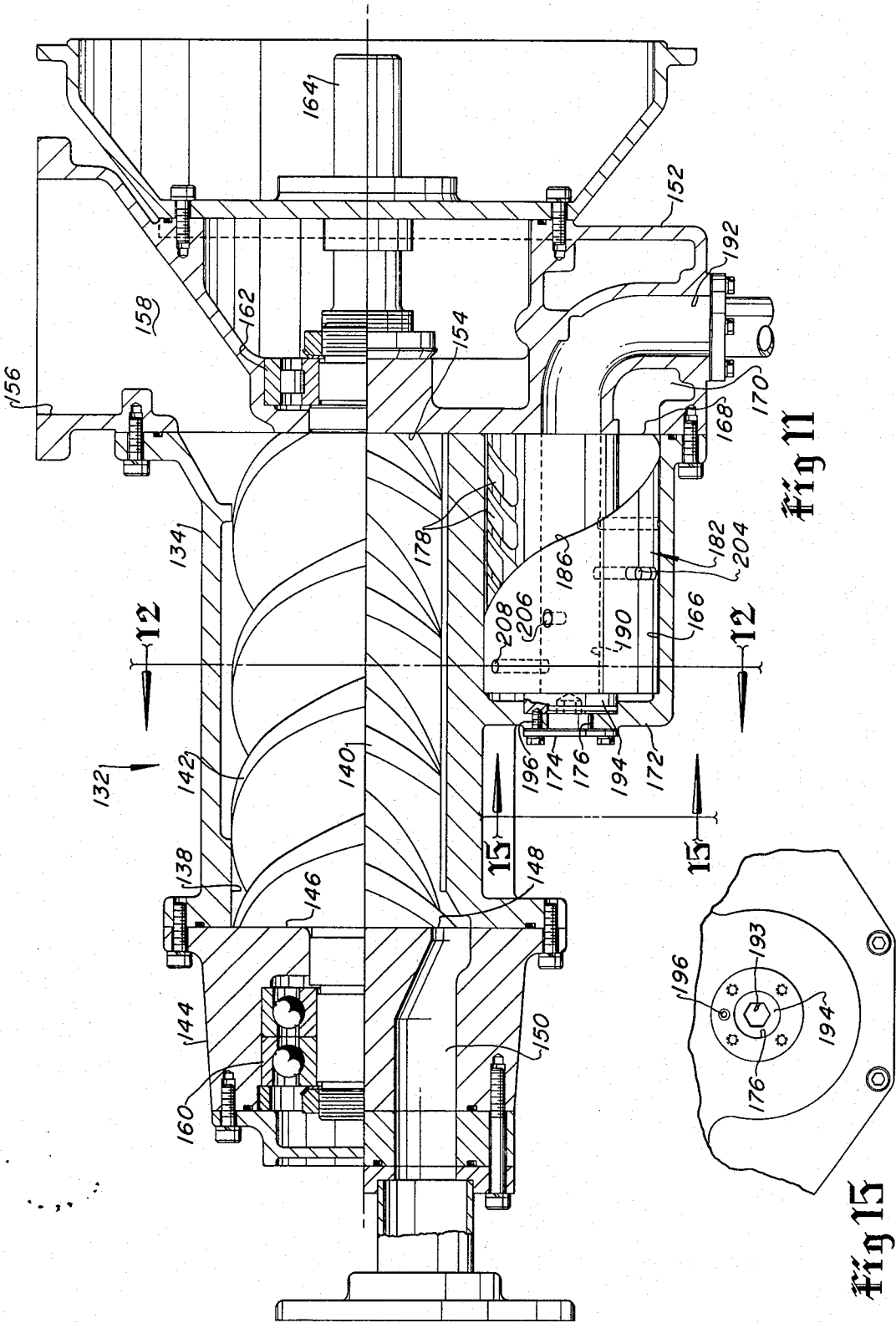












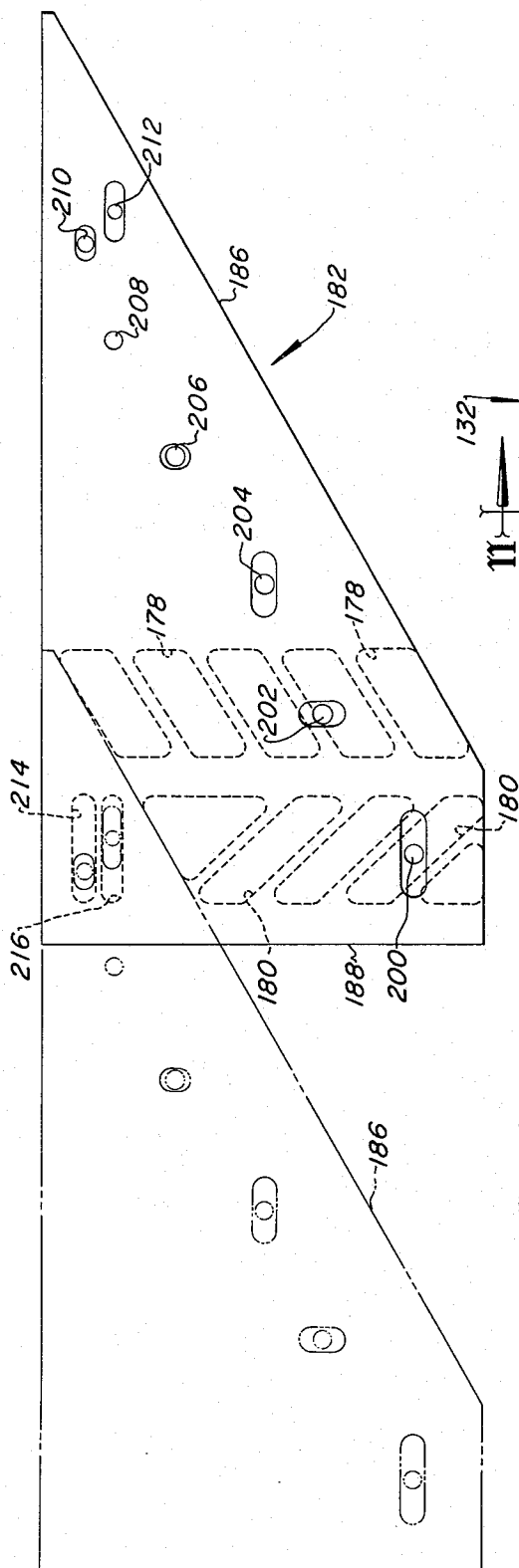


Fig 14

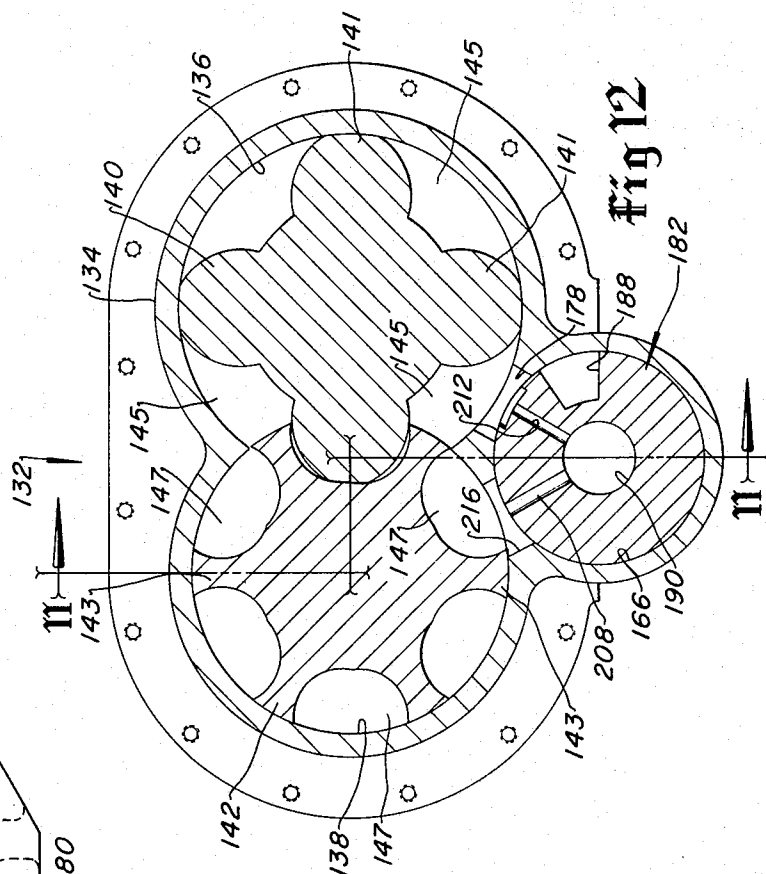


Fig 12

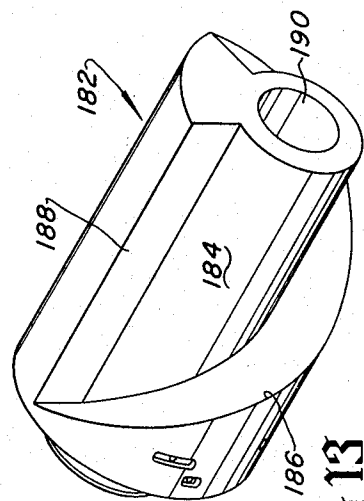


Fig 13

ROTARY CONTROL VALVE FOR SCREW COMPRESSORS

BACKGROUND OF THE INVENTION

In the art of helical screw rotor compressors it has been suggested to provide for control of the gas throughput of the machine and the built-in volume ratio by opening and closing auxiliary ports in the bore walls of the working chambers in which the cooperating screw rotors are disposed. U.S. Pat. No. 3,088,658 to H. B. Wagenius discloses a rotary regulating valve device which is disposed in a cylindrical chamber adjacent to the screw rotor bores. A control edge on the rotary valve member sequentially uncovers a series of auxiliary ports for bypassing back to the compressor inlet port certain portions of the compressor working fluid entrapped in the rotor grooves. Other forms of capacity control devices have been suggested including the axial slide valve devices disclosed in U.S. Pat. Nos. 3,088,659 and 3,314,597. Generally, the arrangement including the rotary type capacity control valve has been found to be more economical to manufacture and does not require the precision with which the axial slide valve must be supported in the compressor casing.

The rotary regulating or capacity control valve has generally not proven to be as suitable as the axial slide valve when considering the overall operating efficiency of certain types of helical screw rotor machines because of the unavoidable unswept volume formed by the auxiliary ports which open into the compressor working chambers. However, in accordance with the present invention this problem has been substantially overcome by the use of the rotary capacity control valve in a helical screw gas compressor in which liquid is injected directly into the compressor working chambers. Moreover, improvements realized with the present invention concerning the amount of liquid injected and the location of the injection passages with respect to the working chambers have also contributed measurably to improving the efficiency of compressors of the subject type.

SUMMARY OF THE INVENTION

The present invention provides for an improved liquid injected helical screw rotor gas compressor which includes a rotary capacity control valve characterized by a cylindrical plug member having a control edge which coacts with a series of ports in the working chambers to effect the sequential opening and closing of said ports to regulate the gas throughput and built-in volume ratio of the compressor. With the use of a rotary capacity control valve in a helical screw gas compressor in accordance with the present invention improvements in compressor operating efficiency have been realized thereby overcoming the disadvantages which previously burdened helical screw rotor machines with rotary type capacity regulating devices.

Further, in accordance with the present invention, a rotary capacity control valve is provided in a helical screw gas compressor which is capable of regulating the amount of liquid injected into the compressor working chambers as well as the location of the liquid injection passages in accordance with the rotative position of the valve member itself. In this way the desired flow rate of liquid and the location of injection into the compressor working chambers can be conveniently

controlled in accordance with the gas throughput and the working pressure of the compressor.

The present invention also provides a liquid injected gas compressor which includes a structurally simple and reliable capacity control device which may be provided in a form for use with compressors where only infrequent adjustment of compressor capacity is required. Accordingly, with the present invention a capacity controlled liquid injected gas compressor is realized which is not only economical to manufacture but is more efficient than certain prior art screw compressors with attendant capacity regulating devices.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal section view of a helical screw compressor in accordance with the present invention and is taken along line 1—1 of FIG. 4;

FIG. 2 is a schematic view of a liquid injected compressor system including the compressor of FIG. 1;

FIG. 3 is a section view taken along line 3—3 of FIG. 1;

FIG. 4 is a section view taken along line 4—4 of FIG. 1;

FIG. 5 is a section view taken along line 5—5 of FIG. 1;

FIG. 6 is a perspective view of the control valve member of the compressor of FIG. 1;

FIGS. 7 through 10 are planar developments of the cylindrical control valve chamber of the compressor of FIG. 1 and illustrating the auxiliary ports and various angular positions of the control edge and liquid injection passages of the control valve member;

FIG. 11 is a longitudinal section view of a second embodiment of a liquid injected helical screw gas compressor in accordance with the present invention and is taken along line 11—11 of FIG. 12;

FIG. 12 is a transverse section taken along line 12—12 of FIG. 11;

FIG. 13 is a perspective view of the control valve member of the compressor of FIG. 11;

FIG. 14 is a planar development of the cylindrical surface of the control valve member of the compressor of FIG. 11 illustrating the auxiliary ports and two positions of the control valve member; and,

FIG. 15 is a view taken from the line 15—15 of FIG. 11.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIGS. 1 and 4 of the drawings, a liquid injected helical screw gas compressor in accordance with the present invention is generally designated by the numeral 10. The compressor 10 includes a main casing 12 having parallel intersecting bores 14 and 16, a bearing support member 18 forming an end wall 20, and a bearing support member 22 forming an end wall 24. The bores 14 and 16 form working chambers together with a pair of cooperating main and gate helical screw rotors 26 and 28. The gate rotor 28 is provided with six helical lobes 30 and intervening grooves 32 and includes oppositely projecting shaft end portions which are supported in bearings 34 and 36. The main rotor 26 is provided with four helical lobes 38 and intervening grooves 40 and is similarly supported by bearings, not shown, disposed in the housing members 18 and 22. The main rotor 26 also includes a shaft portion 42 which extends from an end cover 44 for connecting the

compressor 10 to a rotary power source such as a motor, turbine, internal combustion engine or the like. In a known way the rotors 26 and 28 cooperate with each other and with the casing 12 and end walls 20 and 24 for compressing fluid trapped in the grooves 32 and 40 as the rotors rotate together. In the compressor 10 the main rotor 26 drivingly engages the gate rotor 28 although synchronizing gears can be used to form the driving engagement between the rotors as is known in the art of helical screw rotor machines. The lobes and grooves of the main rotor 26 have a wrap angle of about 300° and the lobes and grooves of the gate rotor 28 have a wrap angle of about 200°. Other combinations of lobes and wrap angles may be used with the advantageous features of the present invention.

The casing 12 includes an opening 46 which is in communication with a low pressure or inlet port 48 formed in the casing and partly in the member 22 whereby gas may be admitted to the bores 14 and 16 for compression by the rotors 26 and 28. The compressor 10 also includes a high pressure or discharge port 50 formed substantially in the end wall 20 and also having a portion 52 disposed in the casing 12. Gas compressed by the intermeshing rotors 26 and 28 is discharged through the port 50 and into a passage 54 in a substantially continuous and pulsation free flow when the compressor 10 is in operation.

The compressor 10 is suitably adapted for the injection of liquid into the bores 14 and 16, as will be explained in greater detail herein, for sealing the spaces formed between the rotors 26 and 28 and the walls of the bores 14 and 16 as well as the end walls 20 and 24. The liquid injected into the working chambers formed in the bores 14 and 16 also absorbs at least part of the heat of compression of the gas being compressed and is pumped through the discharge port 50 with the compressed gas to be separated therefrom by suitable means downstream of the compressor proper. The injection liquid is preferably an oil which is also used as a lubricant for the intermeshed rotors and the bearings of the compressor.

As shown in FIGS. 1, 4, and 5 the casing 12 also includes an elongated cylindrical chamber 56 formed adjacent and substantially parallel to the bores 14 and 16. The casing 12 further includes passages 58 and 60 which are in communication with the chamber 56 and with the inlet port 48 formed in the casing. The chamber 56 is in communication with the bores 14 and 16 by a series of auxiliary ports 62 opening into the bore 16 and a series of auxiliary ports 64 which open into the bore 14. The configuration and relative positions of the auxiliary ports 62 and 64 are more clearly shown by the planar developments of the bore wall of the cylindrical chamber 56 shown in FIGS. 7 through 10. The ports 62 and 64 are generally formed as spaced apart oblique slots disposed along and on each side of the line of intersection of the bores 14 and 16. The ports 62 and 64 are formed to have their longer sides substantially parallel to the radial outermost tips of the lobes 38 and 30 of the respective main and gate rotors. In other words, the sides of the slots 62 and 64 respectively form angles with the longitudinal axis of the chamber 56 which correspond substantially to the helix angles of the rotors 26 and 28.

The compressor 10 includes a capacity control device characterized by a rotatably turnable valve member 66 having a cylindrical plug portion 68 closely fit-

ted in the cylindrical chamber 56. The member 66 includes a hollow shaft portion 70 supported by a bearing 72 disposed in the support member 22. The opposite end 74 of the valve member 66 extends into a chamber 76 in the bearing support member 18 and is supported in a bearing 78 disposed in the casing 12. The shaft 70 extends into a substantially closed space 80 formed in the cover member 44. A gear 82 is keyed to the distal end of the shaft 70 and, as may be seen from viewing FIG. 3, is meshed with a helical screw or worm gear 84 fastened to the end of a control shaft 86. The opposite end of the control shaft 86 has a hand wheel 88 connected thereto for manually turning the shaft and the rotatable valve member 66. As may be appreciated by those skilled in the art, suitable motorized actuating means may be used for turning the valve member 66 in place of the manual operating means disclosed.

The valve member 66 is also characterized by a longitudinal passage 90 extending from the end portion 74 through the shaft 70. A conduit 92 is connected to the chamber 76 for conducting liquid to the passage 90 in the valve member 66. Referring to FIG. 6 also the cylindrical plug portion of the valve member 66 extends only partially around the shaft 70 and is delimited by a longitudinal edge 94. The cylindrical portion 68 is further delimited by a stepped helical control edge 96 which is operable in response to turning the valve member 66, to sequentially open and close the auxiliary ports 62 and 64. In this way, as is known from U.S. Pat. No. 3,088,658, the capacity or quantity of gas or working fluid passed through the compressor can be varied. The main portion of the helical edge 96 is formed to be substantially parallel to the longitudinal sides of the ports 62 for quick opening and closing of the ports in response to minimal rotational turning of the valve member. As the valve member 66 is turned to progressively uncover the ports 62 from the inlet end of the bores 14 and 16 toward the high pressure or discharge end delimited by the end wall 20 the ports 64 are progressively opened also to provide additional area for gas entrapped in the grooves 32 of the gate rotor to be bypassed with minimum throttling losses back to the compressor inlet port by way of the chamber 56 and passages 58 and 60. The arrangement of auxiliary ports 62 and 64 and the configuration of the valve member 66 provides for control of the capacity or gas throughput of the compressor 10 from the maximum displacement provided by the rotors 26 and 28 to substantially zero flow.

A compressor of the type disclosed herein and operating with liquid injected into the bores 14 and 16 has been found to have improved efficiency over prior art nonliquid injected helical screw rotor compressors with rotary type regulating valves. Such improvements are seen to be realized from a reduction in the clearance volume of the ports 62 and 64 between the respective bores 16 and 14 and the cylindrical plug portion 68 of the control valve member 66. As may be seen in FIGS. 4 and 5, even though the cylindrical valve chamber 56 is placed as close to the rotor bores as is practical the auxiliary ports 62 and 64 occupy a finite volume or space in which compressed gas is trapped and reexpanded as the grooves of the rotors 26 and 28 move progressively past the ports. By the use of liquid injected into the bores 14 and 16, the clearance volume of the ports 62 and 64 which are unopened is substantially reduced because of a filling up of the ports with

liquid. Therefore, as the rotors 26 and 28 compress gas in the chevron shaped chambers formed by the grooves and intermeshing lobes of the rotors, a negligible amount of gas is trapped in the ports 62 and 64 and lost to reexpansion as the lobe tips of the rotors pass over the ports. Moreover, with the improved liquid injection means associated with the rotary capacity control valve of the present invention the efficiency of liquid injected helical screw compressors is further improved.

The control valve member 66 of the present invention is further characterized by a plurality of passages which extend radially with respect to the longitudinal turning axis of the valve member from the longitudinal passage 90 to the cylindrical surface of the plug portion 68. The passages are spaced longitudinally along the valve member 66 and are also spaced apart in angular increments around the circumference of the cylindrical plug portion 68. FIGS. 7 through 10 illustrate planar developments of the cylindrical chamber 56 with a planar development of the cylindrical plug portion 68 of the control valve member 66 superimposed on the chamber development to show various positions of the control edge 96. The developments of FIGS. 7 through 10 also show pairs of liquid injection passages 100, 102, 104, a single passage 106, and further pairs of passages 108 and 110. All of these passages are in communication with the longitudinal passage 90. The pairs of passages 100, 102, 104, 108, and 110 are progressively smaller in diameter and effective flow area, and accordingly provide for progressively smaller flow rates of liquid into the bores 14 and 16 as they respectively communicate with the ports 64 on turning of the valve member 66. The single passage 106 is smaller in diameter than the passages 104 but larger than the passages 108. In the embodiment of FIGS. 1 through 10 most of the liquid injection passages communicate only with the ports 64 and some of the ports 62 are modified in shape to prevent unwanted registration with the liquid injection passages.

FIG. 7 illustrates the maximum capacity position of the control valve 66 wherein all of the ports 62 and 64 are closed. In this position one passage 100 of the largest diameter is in communication with a port 64 which assures that an adequate amount of liquid is injected into the bore 16 for cooling and sealing purposes. The location of liquid injection shown by FIG. 7 is also into a chamber formed by the rotors 26 and 28 which is only slightly reduced in volume but is no longer in communication with the inlet port 48 as is true for all liquid injection positions in the embodiment of FIGS. 1 through 10. In this way the liquid injected into the compressor does not mix with the gas flowing into the grooves 32 and 40 of the rotors until after a working chamber formed by the casing bores 14 and 16 and a pair of grooves of the rotors 26 and 28 is no longer in communication with the inlet port 48. It is desirable to avoid injecting liquid into the rotor grooves which are still in communication with the inlet port in order to prevent unwanted reduction in filling of the grooves with inflowing gas. However, it is also desirable to inject liquid at a position in the bores which will assure that liquid flows into all of the ports 62 and 64 which are unopened to thereby fill up the clearance space formed by the ports.

FIG. 8 illustrates a position of the valve member 66 in which two of the ports 62 are open as well as one port 64. This position represents a reduction in working

fluid capacity of approximately 25 percent for the compressor 10. Accordingly, portions of both of the smaller passages 102 are in communication with a port 64 and are the only passages not blocked and in a no-flow condition by the close fit of the plug portion 68 in the chamber 56. Since the compressor throughput capacity is now reduced, a smaller quantity of liquid is required for adequate cooling. Moreover, excessive amounts of liquid cause unwanted pumping losses as well as overcooling of the working fluid. Overcooling of the working fluid can cause condensation of water vapor in air compressors working in ambient atmosphere. Accordingly, with the control valve 66 of the present invention the rate of liquid injection and location of injection with respect to the rotor bores 14 and 16 is automatically controlled in accordance with the rotary position of the valve member itself.

FIGS. 9 and 10 show further reduced capacity positions of the control valve member 66. In FIG. 9 the valve member is in a position in which the throughput capacity is reduced to approximately 50 percent of the maximum capacity of the compressor 10. One passage 104 is in communication with a port 64 and the rate of liquid injection is further reduced from the rate of the positions of the valve member in FIGS. 7 and 8. In FIG. 10, the control valve member 66 has been turned to open all of the ports 62 as well as all of ports 64 and the capacity of the compressor has been reduced to substantially zero flow. One passage 110 is in communication with a slot 112 in the casing which opens into the bore 14 whereby a still further reduced quantity of liquid is injected into the working chambers. In the zero capacity position of the valve member 66 it is desirable to provide some lubrication for the intermeshed rotors and to cool working fluid which is disposed in the discharge port 50 and the discharge passage downstream thereof and which is unavoidably expanded and recompressed by the intermeshing rotors 26 and 28. Although FIGS. 7 through 10 illustrate four positions of the rotary control valve member 66, including the maximum and zero capacity positions, it will be appreciated that the valve member may be turned to any position between the limit positions for controlling the working fluid capacity of the compressor 10. Moreover, as the valve member 66 is turned to reduce the capacity of the compressor 10 the rate of liquid injection is also reduced as the injection passages are sequentially placed in communication with the ports 64.

The arrangement of liquid injection passages in the control valve member 66 is also advantageous for certain applications of the compressor 10 wherein the discharge working pressure is increased and the compressor throughput capacity is proportionally reduced to prevent overloading the compressor driving motor. Referring to FIG. 2, the compressor 10 is shown in a system including a drive motor 114 which is drivably connected to the shaft 42 of the rotor 26. The compressor 10 also includes an inlet filter 116 for filtering air admitted to the compressor inlet opening 46. The compressor system of FIG. 2 also includes a combination compressed air receiver and liquid separator and reservoir tank 118 having a liquid separator element 120 therein and a final discharge conduit 122 for conducting liquid-free pressure air to its end use. A conduit 124 connects the discharge passage 54 of the compressor 10 to the tank 118 for conducting the liquid-gas mixture from the compressor. A conduit 126 is connected

to tank 118 for conducting liquid through a filter 128 and to a heat exchanger 130 from which the liquid is conducted back to the compressor through conduit 92. The liquid is forced to recirculate back to the compressor 10 by the pressure differential which exists between the tank 118 and the particular injection passage which is in communication with the working chambers of the compressor.

In certain applications of the compressor system of FIG. 2, or a similar system, it may be desirable to increase the working pressure of the system including the pressure in the tank 118. Therefore, in order to avoid overloading the motor 114 the capacity of the compressor may be required to be reduced to limit the power absorbed by the compressor to what the motor can provide. This may be conveniently accomplished by adjusting the valve member 66 to reduce the compressor throughput capacity by opening the required number of ports 62 and 64. Moreover, as the valve member 66 is rotated to a new position, a progressively smaller diameter passage or pair of passages will become operable for injecting liquid into the bore 16. Since the pressure differential between the tank 118 and the point of liquid injection has been increased due to increasing the working pressure in the conduit 92 the smaller diameter passages will limit the flow of liquid to a rate in accordance with the needs of the compressor and will prevent excessive amounts of liquid from being injected into the bore 16 through one or more of the larger passages. In other words, for any position of the valve member 66 the flow of injection liquid is automatically adjusted thanks to the arrangement and size of the passages 100, 102, 104, 106, 108, and 110. If the capacity is reduced while working pressure is held substantially constant the rate of liquid injection is reduced in accordance with reduced gas throughput. If capacity is reduced and working pressure is increased, the liquid injection rate normally will remain substantially constant thanks to the reduced size of the injection passages operating at a higher pressure differential. Moreover, if it is desired to turn the valve member 66 to reduce the capacity and the built-in volume ratio for operation of the compressor at low working pressures in order to conserve power absorption by the compressor, then the rate of injection of liquid will be decreased because the smaller liquid injection passages will be operable and at lower working pressures. This also will be in accordance with the liquid injection needs of the compressor for efficient operation.

A second embodiment of the present invention is illustrated in FIGS. 11 through 15. Referring to FIGS. 11 and 12, a liquid injected helical screw rotor gas compressor is illustrated and generally designated by the numeral 132. The compressor 132 includes a casing 134 having a pair of parallel intersecting bores 136 and 138 in which are disposed respectively a pair of intermeshing helical screw rotors 140 and 142. The rotors 140 and 142 are of substantially the same configuration as the rotors 26 and 28 of the compressor 10 and have, respectively, helical lobes 141 and 143 and intervening grooves 145 and 147.

The compressor 132 also includes a bearing support member 144 fastened to the casing 134 and forming a high pressure end wall 146 of the compressor working chambers formed within the bores 136 and 138. A discharge port 148 is formed in the end wall 146 for communicating the bores 136 and 138 with a discharge pas-

sage 150 for the compressor working fluid. The compressor 132 also includes a bearing support member 152 fastened to the casing 134 and forming a low pressure end wall 154. The member 152 further includes an inlet opening 156 and an inlet port 158 for admitting working fluid such as air to the bores 136 and 138 for compression and delivery to the discharge passage by the rotors 140 and 142 in a known way. The inlet port 158 is basically of the axial configuration as opposed to the primarily radial inlet port of the compressor 10. The members 144 and 152 include respective bearings 160 and 162 for rotatably supporting the gate rotor 142. The main rotor 140 is similarly supported by bearings, not shown, disposed in the members 144 and 152. The main rotor 140 also includes an extended shaft portion 164 for drivably connecting the compressor 132 to a suitable motive power means.

The casing 134 is provided with a cylindrical chamber 166 formed closely adjacent and parallel to the bores 136 and 138. The chamber 166 has an open end 168 which is in communication with a portion 170 of the inlet port 158 and a closed end formed by a wall 172. A cover plate 174 is removably fastened to the wall 172 and covers an opening 176. The casing 134 also includes a series of axially spaced oblique ports 178 opening from the bore 136 into the chamber 166. The ports 178 are similar in configuration to the auxiliary ports 62 of the compressor 10. The casing 134 further includes a series of ports 180, as shown by the planar development of FIG. 14, which open into the chamber 166 from the bore 138. A rotary capacity control valve member 182 is provided in the compressor 132 and is formed as a substantially cylindrical plug having a radially inwardly relieved portion 184, FIG. 13, which is delimited by a helical control edge 186 and a longitudinal edge 188. The valve member 182 is closely fitted in the chamber 166 and includes an axial passage 190 which is in communication with a liquid inlet passage 192 formed in the support member 152. A hub portion 194 on the valve member 182 is fitted in a recess in the wall 172. A set screw 196 threaded into the wall 172 is engaged with the hub 194 for locking the valve member 182 in various rotative positions in accordance with the desired capacity of the compressor 132. As shown in FIG. 15, in which view the cover member 174 is removed, a flat sided socket 193 is provided in the end face of the hub 194 for insertion of a suitable wrench for turning the valve member 182.

The valve member 182 is provided with a plurality of liquid injection passages designated by numerals 200, 202, 204, 206, 208, 210 and 212 which are of progressively decreasing diameter from the passage 200 to the passage 212. The passages 200 through 212 are spaced axially along the valve member 182 and also are disposed spaced apart around the circumference of the cylindrical plug portion of the valve member. Each passage is in communication with the passage 190 for conducting liquid to the bores 136 and 138 by way of the auxiliary ports 178 and 180. The passages 208, 212 and 210 are operable to be in registration with slots 216 and 214, FIG. 14, which open into the bore 138. With the exception of passage 208 the passages 200 through 212 are respectively in communication with elongated grooves of various lengths and position on the cylindrical surface of the valve member 182. The arrangement of the passages 200 through 212 and their respective associated grooves is such as to be cooperable with the

ports 178 and 180 and the slots 214 and 216 to vary the amount and location of liquid injected into the bores 136 and 138. This is in accordance with the position of the control edge 186 and, accordingly, the capacity or gas throughput of the compressor 132.

FIG. 14 illustrates a planar development of the cylindrical control valve member 182 superimposed on a planar development of the ports 178 and 180. The development of the valve member 182 represented by the solid lines shows the position of the valve with respect to the ports when all ports are closed and the compressor is operating at the maximum displacement volume or capacity of the rotors 140 and 142. In this position the two largest diameter passages 200 and 202 are in registration with the ports 180 and 178, respectively. The position of the valve 182 represented by the dashed lines is the maximum reduced capacity position in which all ports 178 and 180 are open and in communication with the chamber 166 and the inlet port 158. In this position the passages 210 and 212 of smallest diameter are in registration with slots 214 and 216 for injecting a reduced quantity of liquid in accordance with the reduced capacity or gas throughput of the compressor 132.

As may be appreciated from the foregoing description and illustrations of the compressor 132 the ports 178 and 180 do not extend toward the high pressure end of the casing 134 sufficiently to provide for reduction of compressor capacity to zero flow. The arrangement of the valve member 182 is also provided for compressors wherein only infrequent adjustment of capacity is required. Such a compressor as the compressor 132 might be used where it would be required to operate the compressor continuously at reduced capacity perhaps together with increased inlet and/or discharge working pressures. In this way a compressor such as the compressor 132 could be operated continuously at reduced capacity more efficiently than by the use of certain other capacity reduction means such as throttling the inlet gas flow. Inlet throttling could be used, however, for short term reduction of capacity to zero flow, if needed.

What is claimed is:

1. In a helical screw gas compressor:
 - a casing having two parallel intersecting bores; an inlet port and a discharge port opening into said bores;
 - interengaged helical screw rotors disposed in said bores and characterized by cooperating helical grooves and lobes forming working chambers operable to entrap and compress gas admitted to said bores through said inlet port;
 - a cylindrical chamber formed in said casing and having an opening in communication with said inlet port;
 - a plurality of auxiliary ports formed in said casing and opening to at least one of said bores and to said chamber, said auxiliary ports being spaced apart axially with respect to the longitudinal axis of said one bore;
 - a rotatably turnable control valve member disposed in said chamber characterized by a portion forming a closure for said auxiliary ports delimited by a control edge coactable with said auxiliary ports for opening and closing said auxiliary ports in response to turning said valve member whereby the gas throughput of said compressor is controlled in ac-

cordance with the opening and closing of said auxiliary ports by said valve member;

means for turning said valve member; and,

passage means for injecting liquid into said bores for mixing with said gas being compressed by said rotors, said passage means being operably associated with said valve member whereby the effective fluid flow area of said passage means is increased in response to the turning of said valve member to close said auxiliary ports.

2. The invention set forth in claim 1 wherein:

said passage means for injecting liquid into said bores is disposed in said casing and is operable to control the axial position of liquid injection with respect to the longitudinal axis of said bores in accordance with the opening and closing of said auxiliary ports by said valve member.

3. The invention set forth in claim 2 wherein:

said passage means for injecting liquid into said bores is movably disposed in said casing for injecting liquid into a working chamber formed by a pair of grooves of said rotors only after said working chamber is no longer in communication with said inlet port.

4. The invention set forth in claim 2 wherein:

said valve member comprises a cylindrical plug portion closely fitted in said chamber and delimited by said control edge, and said passage means comprises plural passages in said cylindrical plug portion operable to be in communication with said auxiliary ports in accordance with the rotative position of said valve member for injecting liquid into said bores.

5. The invention set forth in claim 4 wherein:

said passages in said valve member are spaced apart axially with respect to the rotational axis of said valve member and said passages are spaced apart around and open to the circumference of said cylindrical plug portion whereby one or more of said passages are in communication with said auxiliary ports in accordance with the rotative position of said valve member.

6. The invention set forth in claim 4 wherein:

said compressor includes a conduit in communication with said discharge port, a liquid separator and reservoir tank connected to said conduit, and conduit means connected to said reservoir tank and in communication with said passage means for conducting liquid to said passage means for injection into said bores at a flow rate in accordance with the differential pressure between said reservoir tank and said auxiliary port in communication with said passage means.

7. The invention set forth in claim 4 wherein:

said valve member includes a shaft portion and said means for turning said valve member includes a gear connected to said shaft portion, gear means meshed with said gear, and means for reversibly moving said gear means for causing the reversible rotation of said valve member.

8. The invention set forth in claim 4 together with: means for locking said valve member in a predetermined rotative position, said means comprising a set screw disposed in said casing and engageable with said valve member.

9. In a helical screw gas compressor:

a casing having two parallel intersecting bores;

an inlet port and a discharge port opening into said bores;
 interengaged helical screw rotors disposed in said bores and characterized by cooperating helical grooves and lobes forming working chambers operable to entrap and compress gas admitted to said bores through said inlet port;
 a cylindrical chamber formed in said casing and having passage means in communication with said inlet port;
 a plurality of auxiliary ports formed in said casing and opening to at least one of said bores and to said chamber, said auxiliary ports comprising a series of oblique slots spaced apart axially with respect to the longitudinal axis of said one bore, the longer sides of said slots forming an angle with respect to said longitudinal axis which corresponds substantially to the helix angle of the rotor disposed in said one bore;
 a rotatably turnable control valve member comprising a cylindrical plug portion closely fitted in said chamber and forming a closure for said auxiliary ports delimited by a control edge coactable with said auxiliary ports in response to turning of said valve member whereby the gas throughput of said compressor is controlled in accordance with the opening and closing of said auxiliary ports;
 means for turning said valve member;
 a longitudinal passage disposed in said valve member and in communication with a source of liquid; and,
 a series of radially extending passages in said valve member for injecting liquid into said bores for mixing with the gas being compressed by said rotors and for filling up the clearance volume formed between said chamber and said one bore, said passages being spaced apart axially toward said discharge port and with respect to the rotational axis of said valve member and connected to said longitudinal passage, said passages being spaced apart around and opening to the circumference of said cylindrical plug portion, and said passages including respective grooves formed on the circumference of said cylindrical plug portion and proportioned to be cooperable with said auxiliary ports to provide progressively smaller effective flow areas for injection liquid flowing to said bores in accordance with the turning of said valve member to progressively reduce the gas throughput of said compressor, the amount of and axial position of injection of said liquid being in accordance with the rotative position of said valve member.

10. In a helical screw gas compressor:

a casing having two parallel intersecting bores;
 an inlet port and a discharge port opening into said bores;
 interengaged helical screw rotors disposed in said bores and characterized by cooperating helical grooves and lobes forming working chambers operable to entrap and compress gas admitted to said bores through said inlet port;
 a cylindrical chamber formed in said casing and having passage means in communication with said inlet port;
 a plurality of auxiliary ports formed in said casing and opening to at least one of said bores and to said chamber, said auxiliary ports comprising a series of oblique slots spaced apart axially with respect to the longitudinal axis of said one bore, the longer sides of said slots forming an angle with respect to said longitudinal axis which corresponds substantially to the helix angle of the rotor disposed in said one bore;
 a rotatably turnable control valve member comprising a cylindrical plug portion closely fitted in said chamber and forming a closure for said auxiliary ports delimited by a control edge coactable with said auxiliary ports in response to turning of said valve member whereby the gas throughput of said compressor is controlled in accordance with the opening and closing of said auxiliary ports;
 means for turning said valve member; and,
 a plurality of passages in said valve member spaced apart axially toward said discharge port and with respect to the rotational axis of said valve member and having progressively smaller effective flow areas toward said discharge port, said passages being spaced apart around and opening to the circumference of said cylindrical plug portion whereby one or more of said passages are in communication with said auxiliary ports in accordance with the rotative position of said valve member for injecting liquid into said bores for mixing with the gas being compressed by said rotors and for filling up the clearance volume formed by said auxiliary ports between said chamber and said one bore, the amount of and axial position of injection of said liquid being in accordance with the rotative position of said valve member.

11. The invention set forth in claim 10 wherein: said passages are formed as plural pairs of passages, each pair spaced axially closer to said discharge port having a progressively smaller combined effective flow area.

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