

June 28, 1960

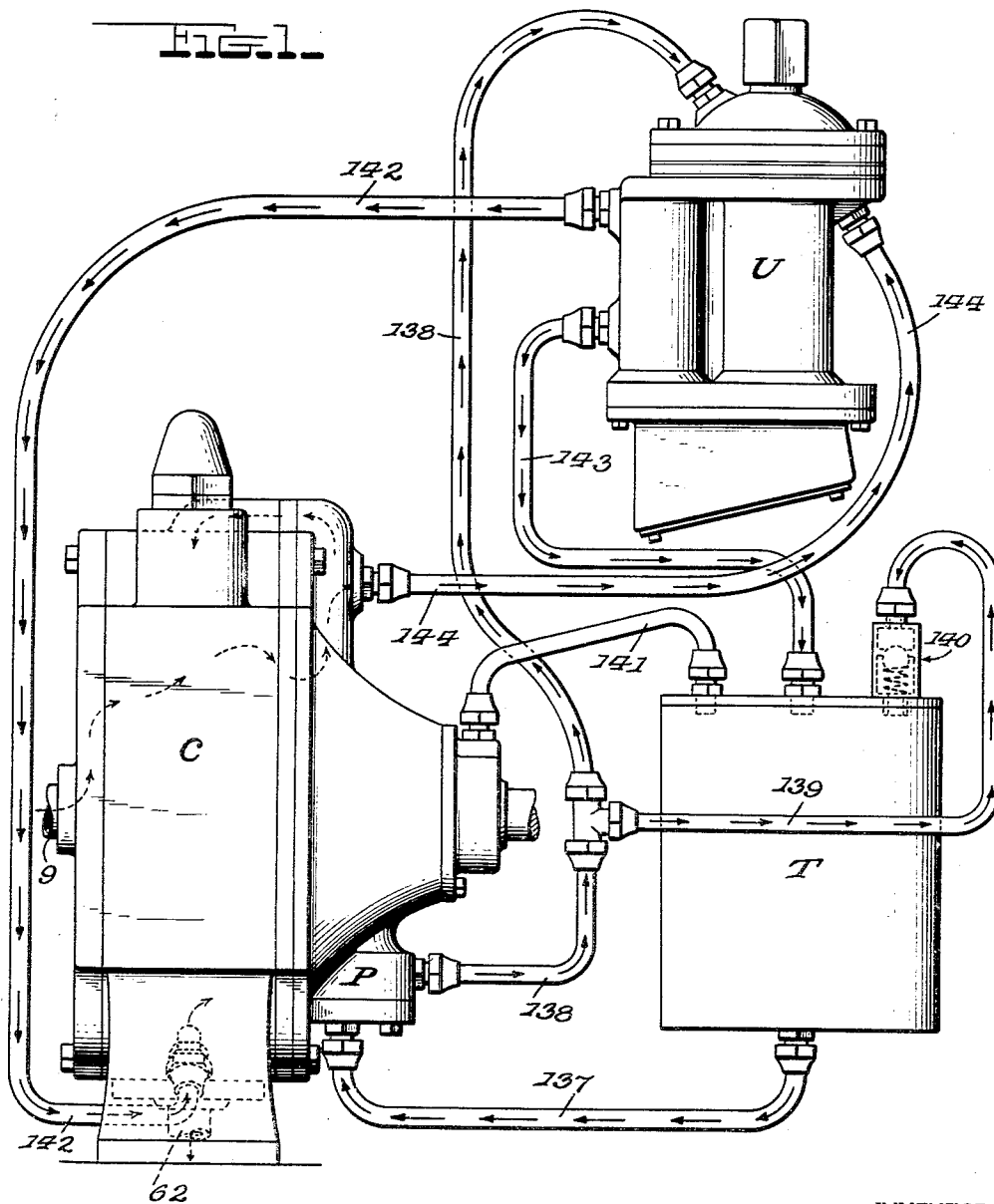
A. BLACKMAN

2,942,774

COMPRESSOR AND CONTROLLING MEANS THEREFOR

Filed March 2, 1956

7 Sheets-Sheet 1



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June 28, 1960

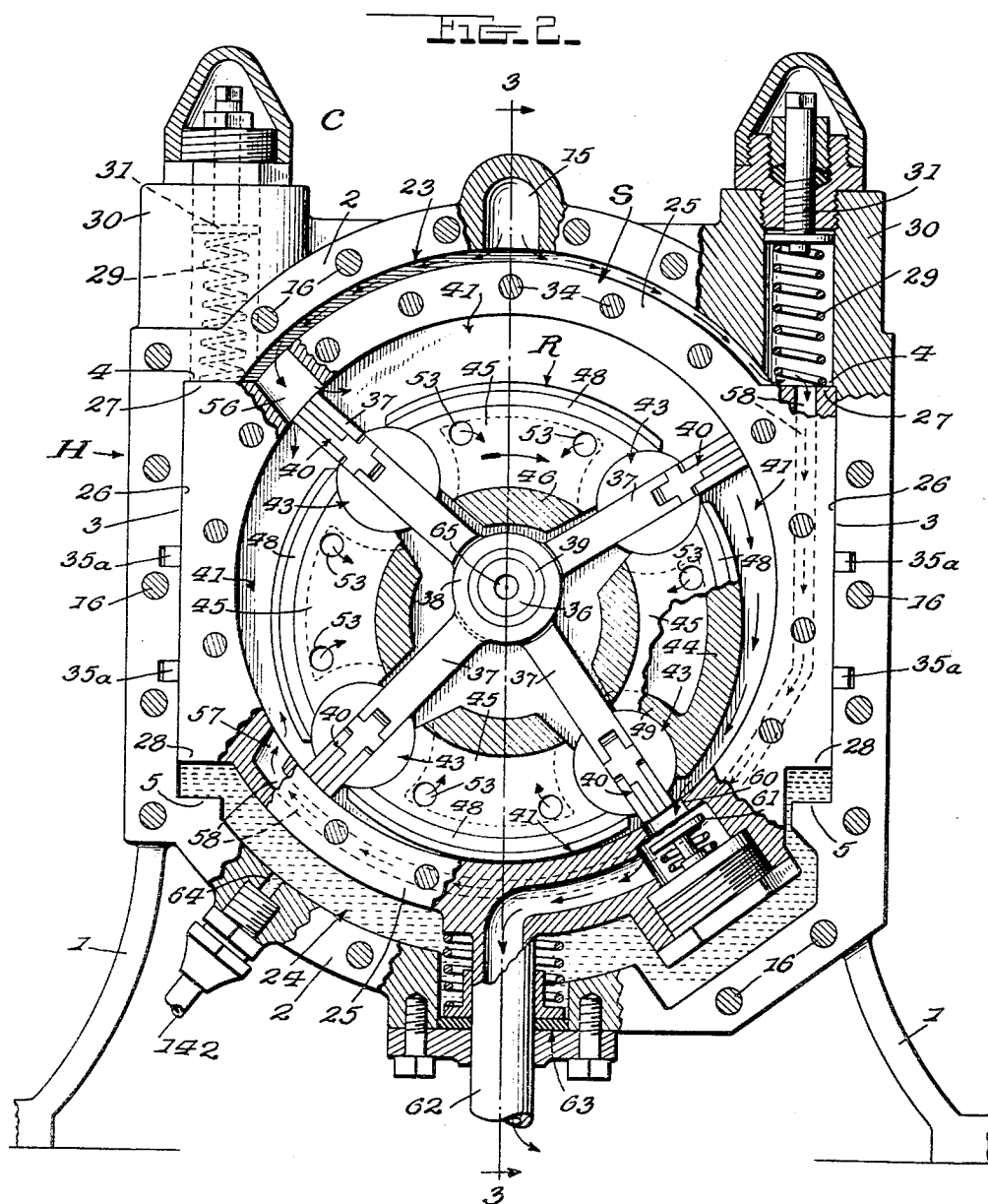
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7 Sheets-Sheet 2



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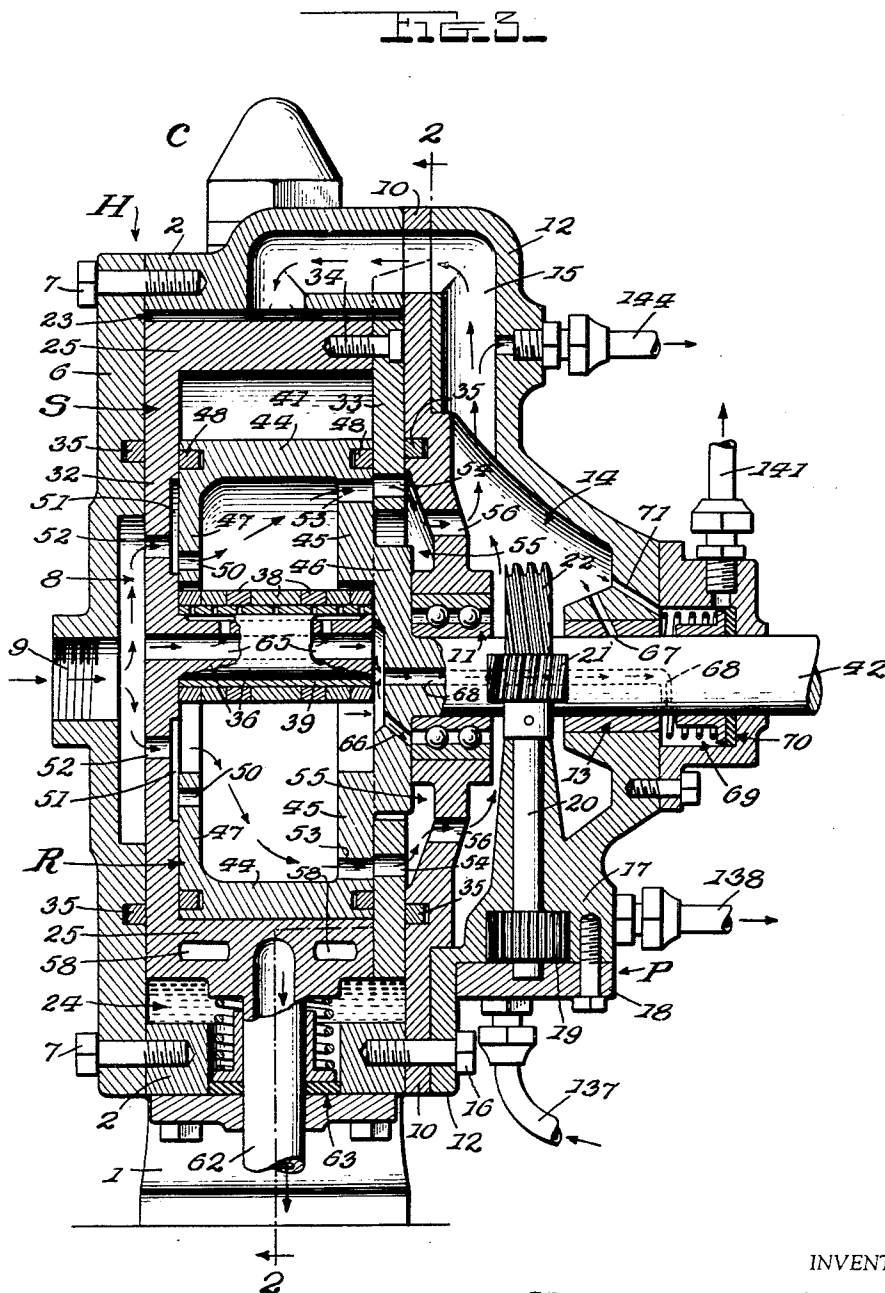
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COMPRESSOR AND CONTROLLING MEANS THEREFOR

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7 Sheets-Sheet 3



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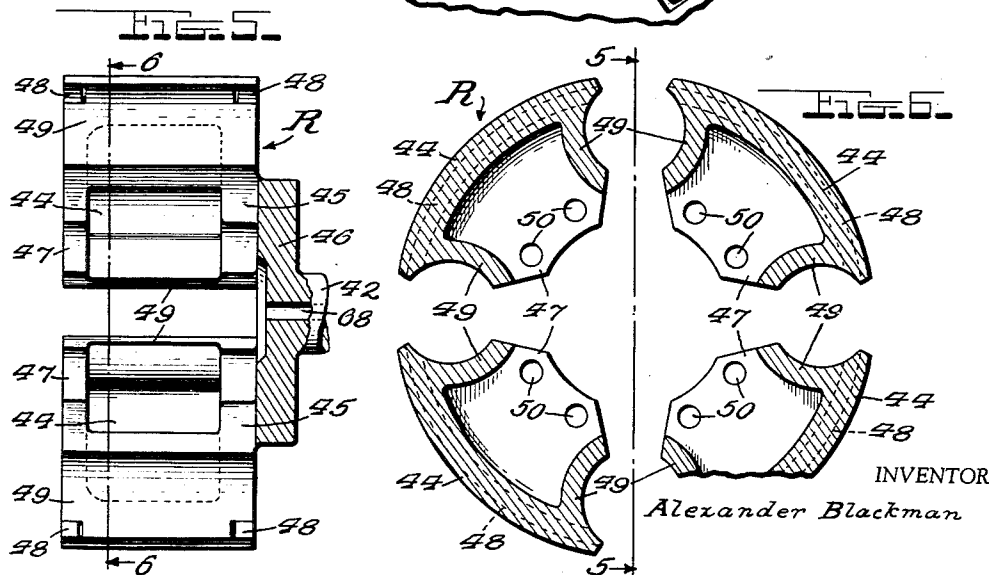
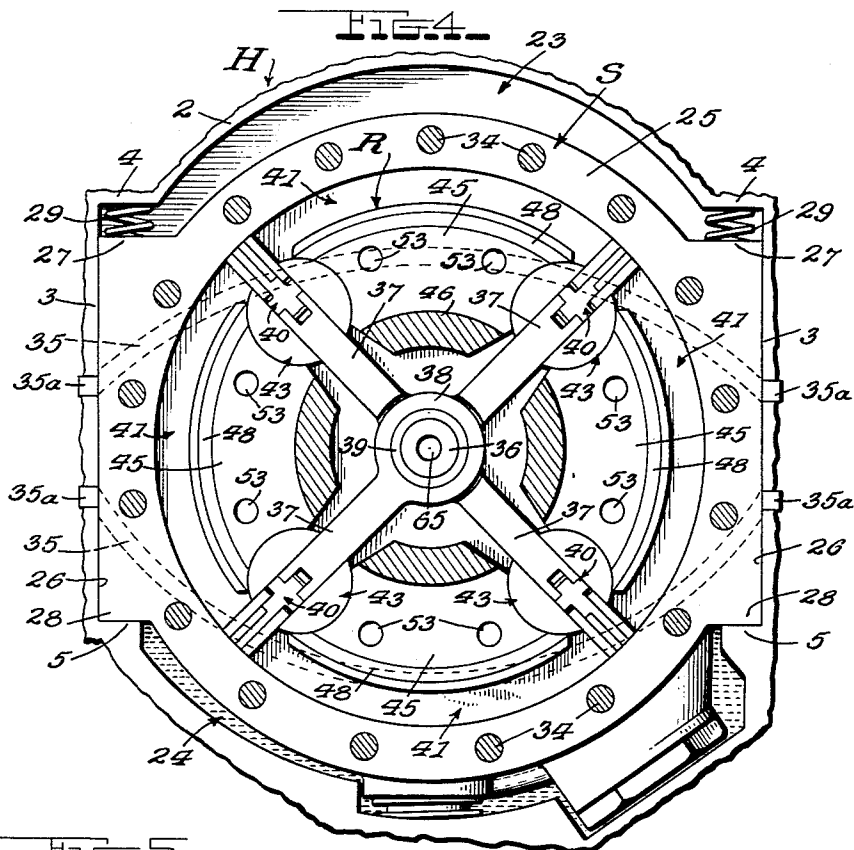
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COMPRESSOR AND CONTROLLING MEANS THEREFOR

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7 Sheets-Sheet 4



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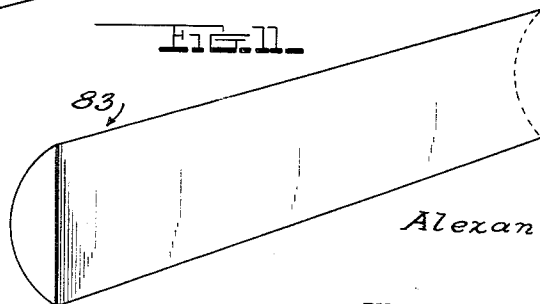
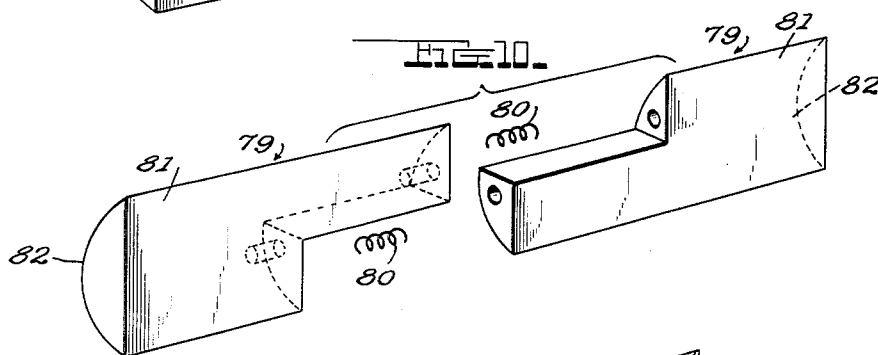
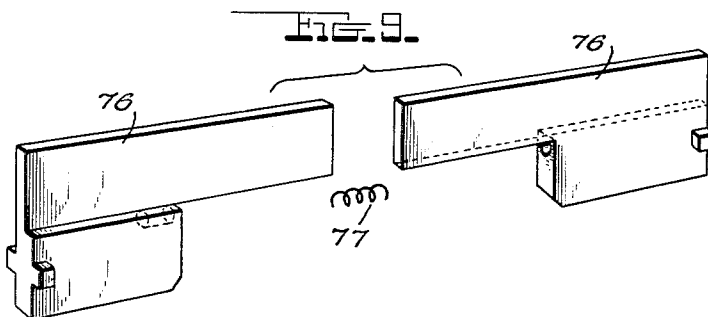
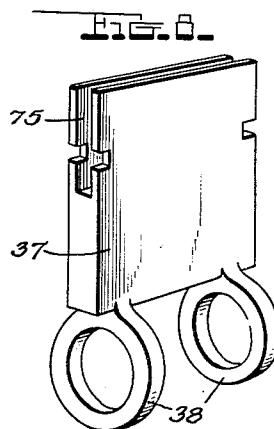
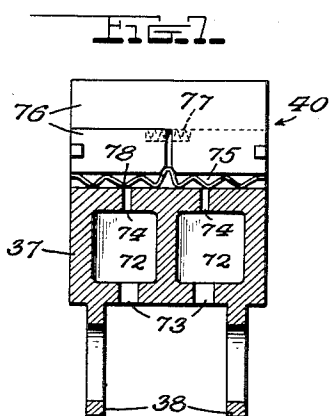
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COMPRESSOR AND CONTROLLING MEANS THEREFOR

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7 Sheets-Sheet 5



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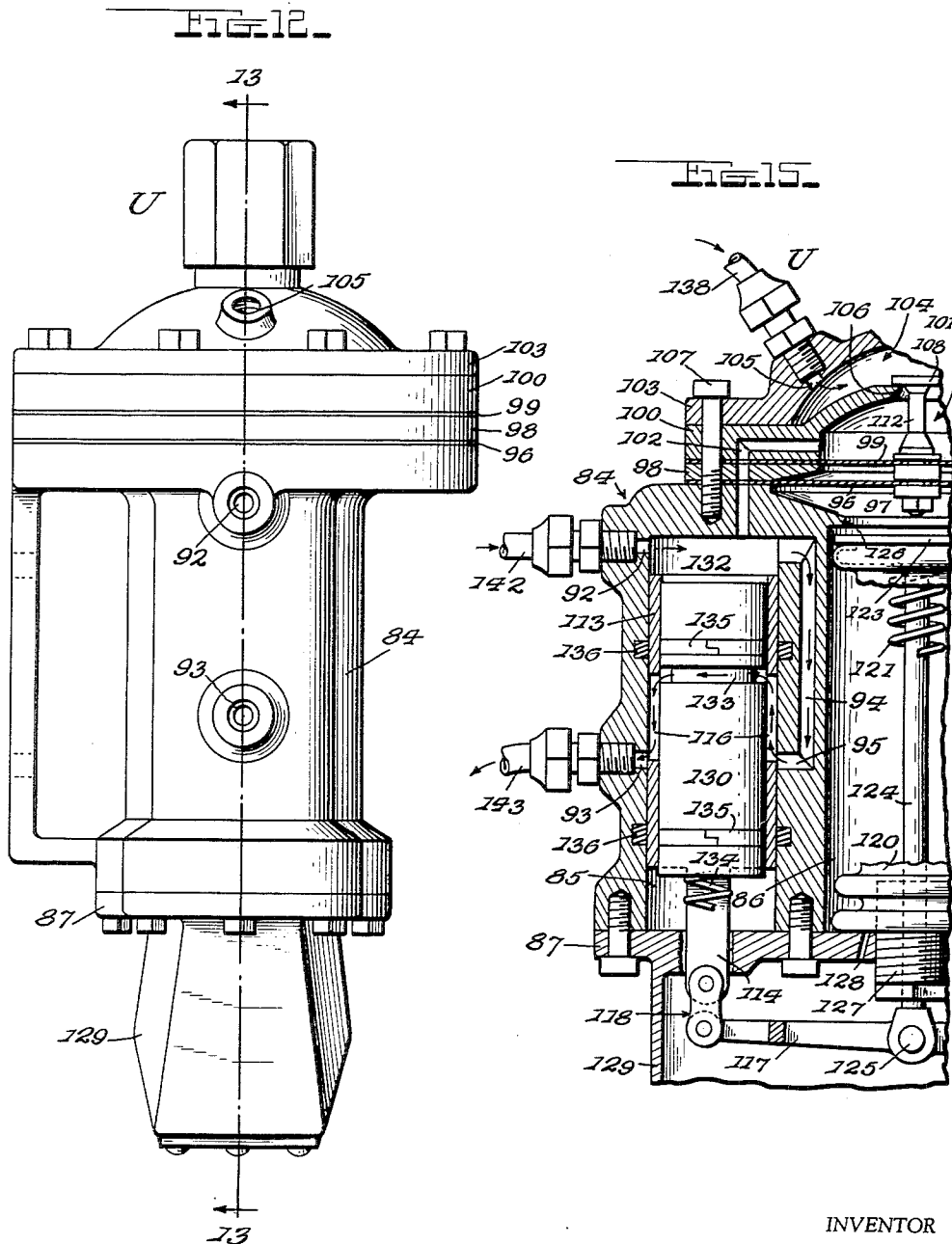
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COMPRESSOR AND CONTROLLING MEANS THEREFOR

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7 Sheets-Sheet 6



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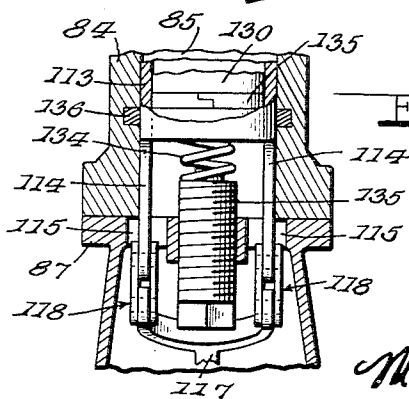
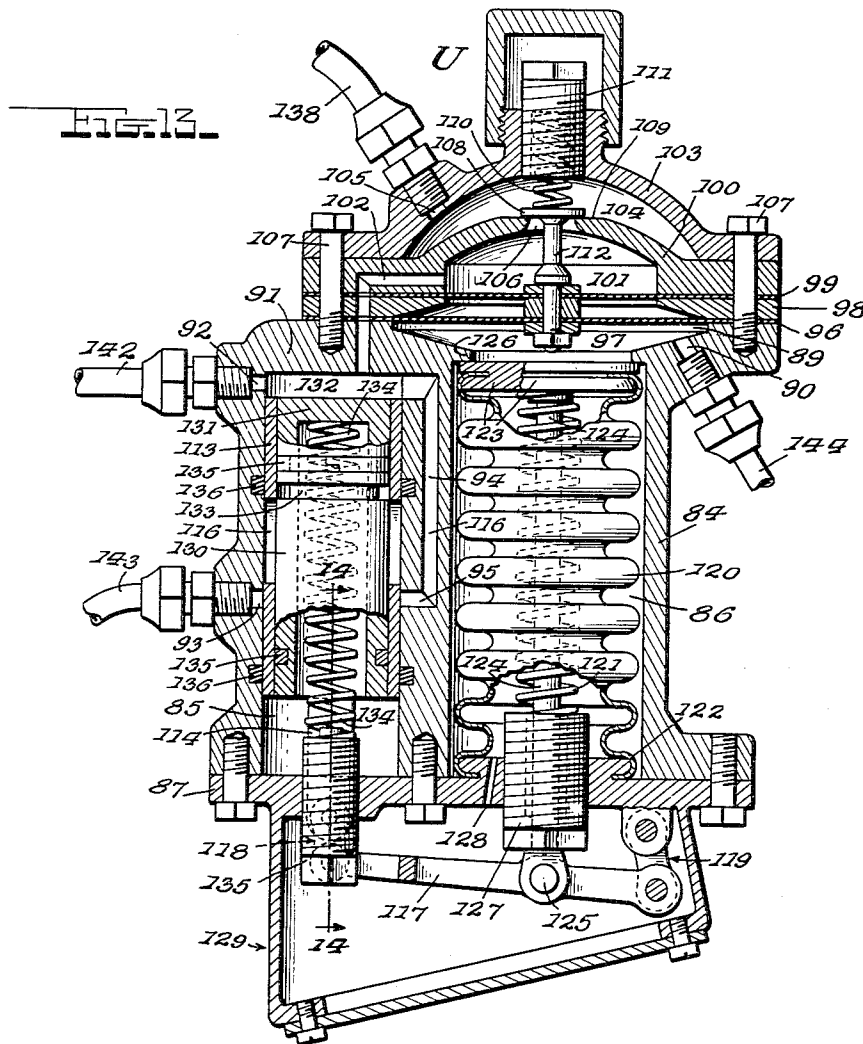
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COMPRESSOR AND CONTROLLING MEANS THEREFOR

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7 Sheets-Sheet 7



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2,942,774

## COMPRESSOR AND CONTROLLING MEANS THEREFOR

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3 Claims. (Cl. 230—138)

This invention relates generally to refrigeration or air conditioning apparatus in which the expansion of a compressed gas is utilized to produce temperature reduction. Specifically, it relates to a novel rotary gas compressor and to novel means for varying the compressor output as required.

The compressor is somewhat similar to prior compressors in that a stator, an eccentric rotor and rotor-driven piston blades coact in forming gas receiving and compressing chambers. In prior compressors with which I am familiar, the piston blades are slidable in and disposed radially of the rotor. Thus, each blade occupies a position radial to the stator, only when it reaches a plane common to both the rotor axis and the stator axis. At each side of this plane, the angular relation of the blades with respect to the stator wall constantly changes during rotor rotation. Therefore, sharp corners cannot be maintained on the outer ends of the blades and said outer ends must either be initially rounded or will become rounded by wear. As these rounded ends necessarily have only line contact with the stator wall, it is practically impossible to provide a gas-tight seal between blades and stator wall.

The present invention aims to overcome the above difficulty by providing a novel well sealed construction in which the piston blades remain invariably in strict radial relation with the stator and in which necessary sliding and rocking movements of said blades with respect to the eccentrically positioned rotor are permitted by novel rocker means rockably mounted on said rotor and slidably contacting with said blades. This construction also causes succeeding blades to swing away from each other during half of each rotor revolution and to swing toward each other during the remainder of each rotor revolution. These movements result in circumferential expansion and contraction of the gas receiving and compressing chambers in addition to the radial expansion and contraction caused by the relative eccentricity of the rotor and stator.

Another object of the invention is to provide an improved construction in which provision is made for relatively shifting the stator and rotor to change the relative eccentricity thereof and thereby vary the sizes of the gas receiving and compression chambers to vary the compressor output.

Yet another object is to provide an improved construction in which provision is made for relatively shifting the stator and rotor from operative eccentric relation with each other, into coaxial relation with each other to terminate compressor output when required.

A further object is to provide a novel construction in which the stator is shiftable with respect to the rotor to vary and terminate compressor output as required.

A still further object is to provide an improved construction in which the stator and rotor are yieldably biased toward maximum eccentric relation with each other, and in which an expansible hydraulic chamber is utilized to relatively shift said stator and rotor toward or to axial alignment as required.

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Yet another object is to provide a novel construction in which the stator is shiftable with respect to the rotor, and in which said stator forms one wall of the aforesaid expansible chamber.

5 A further object is to provide a novel construction in which a pump is driven by the compressor and utilized to supply a hydraulic fluid to the aforesaid expansible chamber to effect relative shifting of the stator and rotor.

10 A still further object is to provide a novel construction in which the ingoing gas is utilized for compressor cooling.

Yet another object is to provide a novel control unit for automatically effecting relative shifting of the rotor and stator as required, to vary compressor output or terminate said output, in accordance with the needs of the system in which the compressor is used.

Further objects are to form each blade engaging rocker of two stepped and slidably overlapped sections which are acted upon by spring means to hold their ends in fluid tight contact with the stator end walls; to provide each piston blade with a pair of slidably overlapped L-shaped sealing members which are held by spring means in fluid tight contact with the stator side wall and end walls; to provide the ends of the rotor with arcuate sealing members contacting fluid tightly with the stator end walls and also fluid tightly abutting the rockers; and to make novel provision whereby, when a gas having lubricating qualities is being compressed some of this gas may be utilized to lubricate the above mentioned L-shaped sealing members.

With the above and other objects in view that will hereinafter appear, the nature of the invention will be more clearly understood by reference to the following detailed description, the appended claims and the several views illustrated in the accompanying drawings.

In the drawings:

Figure 1 of the accompanying drawings is a side elevation showing the preferred compressor, the hydraulic pump, a hydraulic fluid tank and the control unit operatively connected.

Figure 2 is a vertical sectional view of the compressor substantially on the dot and dash line 2—2 of Figure 3, the stator being shown in its uppermost position for maximum output.

Figure 3 is a vertical sectional view substantially on line 3—3 of Figure 2, the stator being again shown in its uppermost position.

Figure 4 is a fragmentary view similar to a portion of Figure 2 but showing the stator in its fully lowered position for output termination.

Figure 5 is a diametrical sectional view of the rotor on line 5—5 of Figure 6.

Figure 6 is an axial sectional view of the rotor on line 6—6 of Figure 5.

Figure 7 is a longitudinal sectional view of one of the piston blades showing the preferred sealing means therein.

Figure 8 is a perspective view of one of the piston blades.

Figure 9 is a perspective view of the sealing means shown in Figure 7.

Figures 10 and 11 are perspective views showing two forms of the rockers which connect the piston blades with the rotor.

Figure 12 is a side elevation of the control unit.

Figure 13 is a vertical sectional view of the control unit, substantially on line 13—13 of Figure 12.

Figure 14 is a fragmentary sectional view substantially on line 14—14 of Figure 13.

70 Figure 15 is a fragmentary sectional view similar to a portion of Figure 13 but showing different positions of parts.

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Preferred construction has been disclosed in the drawings and will be rather specifically described but attention is invited to the possibility of making variations within the scope of the invention.

In Figure 1, the compressor C, its pump P, the hydraulic fluid reservoir or tank T, and the control unit U are shown operatively connected by various gas and hydraulic fluid lines. These lines will be later explained after description of the compressor C, pump P, and control unit U. The compressor construction will first be described and attention is directed to Figures 2 to 6.

#### Compressor C

A housing H is provided having supporting legs 1 whereby it may be stationarily mounted. This housing includes a continuous side wall 2 having opposed, parallel, vertical, internal surfaces 3 (Figures 2 and 4), and stop shoulders 4 and 5 at the upper and lower ends of said surfaces, respectively. A vertical end wall 6 (Figure 3) is secured by screws 7 to one edge of the side wall 2. This end wall 6 is formed with an internal recess constituting a receiving chamber 8 for the gas to be compressed, and a gas inlet 9 is provided for said chamber 8. A second vertical end wall 10 is secured against the other edge of the side wall 2 and is provided with a ball bearing 11. A cap plate 12 is secured against the outer side of the end wall 10 and is provided with a bearing 13 aligned with the bearing 11. The space at the interior of the cap plate 12 provides a gas-conducting chamber 14, and a gas port 15 extends from this chamber 14 to the top of the housing H for conducting gas into the upper end portion of this housing. The cap plate 12 and end wall 10 are secured to the side wall 2 by means of screws 16.

A recessed portion 17 of the cap plate 12 (Figure 3) coacts with a cover plate 18 in forming a housing for the pump P. This pump is preferably of gear type and one of its gears is shown at 19 in Figure 3, said gear being on the lower end of a pump drive shaft 20. The upper end of this shaft 20 has a worm gear 21 driven by a worm 22 on the rotor drive shaft later described.

A stator S is mounted for vertical sliding in the housing H and segregates the upper and lower end portions of said housing from each other to provide a gas chamber 23 over said stator and a hydraulic fluid chamber 24 under said stator. This stator has a continuous side wall 25 which is cylindrical internally. Externally, this side wall 25 has flat vertical surfaces 26 (Figures 2 and 4) which contact slidably with the surfaces 3 of the housing side wall 2. At the upper and lower ends of the surfaces 26, the stator side wall 25 is provided with stop shoulders 27 and 28, respectively. The upper shoulders 27 coact with the shoulders 4 of the housing side wall 2 to limit the upward movement of the stator S (see Figure 2). Similarly, the lower shoulders 28 coact with the shoulders 5 of the housing side wall 2 in limiting the downward movement of the stator S (see Figure 4).

Compression springs 29 are mounted in bosses 30 on the upper end of the housing H to bias the stator S downwardly, and hydraulic fluid supplied to the lower chamber 24 is utilized to upwardly shift said stator. The lower ends of the springs 29 abut the shoulders 27 of the stator wall 25, and suitable spring-adjusting screws 31 abut the upper ends of said springs 29.

The stator S is provided with one complete end wall 32 (Figure 3) which is preferably integral with one edge of the stator side wall 25 and contacts slidably with the housing end wall 6. An end ring 33 is secured by screws 34 to the other edge of the stator side wall 25 and contacts slidably with the housing end wall 10. Suitable seals 35 (Figures 3 and 4) are associated with these contacting end walls 6, 32 and 10, 32, and other seals 35a (Figures 2 and 4) are associated with the contacting surfaces 3, 26 of the housing H and stator S.

The end wall 32 of the stator S rigidly carries a bear-

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ing stub 36 (Figures 2, 3 and 4), said stub being coaxial with the stator side wall 25. The piston blades 37 (Figures 2, 3 and 4) have bearings 38 on their inner ends, those bearings 38 being rotatably mounted on the stub 36 and thus maintained in strictly radial relation with the stator side wall 25. A bushing 39 is preferably interposed between the stub 36 and the bearings 38. The blades 37 extend outwardly to the stator side wall 25 and are of a width for contact with the stator end wall 32 and the end ring 33, and said blades are equipped with suitable sealing means 40 contacting with said side wall 25, end wall 32 and end ring 33. As the blades 37 remain strictly radial to the stator S, the outer ends of these blades need not have rounded corners and will therefore have extensive contact with the stator side wall 25 and better compression sealing will result.

A hollow rotor R (Figures 2, 3, 4, 5 and 6) coacts with the stator S and the piston blades 37 in forming expansible gas receiving and compression chambers 41 (Figures 2 and 4), said rotor being carried by a motor driven shaft 42 which is rotatably mounted in the bearings 11 and 13 of the housing H. This shaft is provided with the pump-driving worm 22 above mentioned. The rotor R is fluid tightly connected with the piston blades 37 by rocker means 43 rockably mounted on said rotor and slidably contacting with said blades. Thus, the rotor R will drive the blades 37 and permit necessary relative rocking and sliding.

When the axis of the stator S is above the axis of the rotor R (see Figure 2 for example), the chambers 41 gradually expand during half of each rotor revolution, for gas reception, and gradually contract during the remainder of each revolution, for gas compression. When the stator S is downwardly shifted to place said stator in coaxial relation with the rotor R (see Figure 4), the chambers 41 remain of uniform size during rotor rotation and no gas compression will occur. The gas to be compressed enters the chambers 41 and the compressed gas discharges from the chambers in ways which can best be explained after describing the rotor construction. For this purpose, reference is again made to Figures 2 to 6.

The rotor R has arcuate side wall sections 44 between the blades 37. Substantially segmental end flanges 45 are integrally joined at their outer edges to one edge of the side wall sections 44, respectively, the inner edges of said end flanges 45 being integrally joined to a hub 46 which is integral with or otherwise secured to the shaft 42. Additional substantially segmental end flanges 47 are formed integrally with the other edges of the wall sections 44. These end flanges 47 contact slidably with the stator end wall 32, the end flanges 45 similarly contact with the stator end ring 33 as seen in Figure 3, and suitable arcuate seals 48 are provided. Transversely arcuate bearing members 49 extend from end to the end of the rotor R and are integral with the end flanges 45 and 47 and with the side wall sections 44. The bearing members 49 are disposed in pairs and the members of each pair are curved about a common line. These bearing members rockably receive the rocker means 43 for the blades 37.

The construction above described provides a hollow formation for the rotor R and provision is made whereby the gas to be compressed is directed through this hollow rotor for compressor cooling. To this end, the rotor end flanges 47 have ports 50 communicable with an annular recess 51 in the stator end wall 32, and this end wall has ports 52 extending to said recess 51 from the gas receiving chamber 8 of the housing end wall 6. Thus, the gas entering the compressor through the inlet 9 is directed into the hollow rotor R. The rotor end flanges 45 and the stator end ring 33 have communicable ports 53, 54 and the ports 54 open into a recess 55 in the inner side of the housing end wall 10. Ports 56 extend from this recess 55 to the gas chamber 14 at the interior of the cap plate 12. Thus, the ingoing gas reaches this chamber 14 and flows

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from the latter through the port 15, above described, into the chamber 23 over the stator S. One gas admission port 56 extends directly from this chamber 23 into the stator S. Another gas admission port 57 is provided in advance of the port 56, and said port 57 communicates with the chamber 23 by way of ports 58 in the stator side wall 25. The gas flowing through the ports 58 aids in compressor cooling and the two ports 57, 56 adequately supply gas to the chambers 41 during expansion of the latter.

A discharge port 60 (Figure 2) is provided for the gas compressed in the chambers 41. A check valve 61 is mounted in the stator S and controls communication of the discharge port 61 with a gas discharge pipe 62 which is integral with or otherwise secured to the stator side wall 25. The pipe 62 extends slidably through suitable sealing means 63 to the exterior of the housing H and is to be hose-connected to the usual condenser of the refrigeration or other system in which the compressor is used.

The expansion and contraction of the chambers 41 is the result of two contributing factors, first the eccentric relation of the rotor R and stator S, and second relative swinging movements of the piston blades 37. Due to the construction employed, succeeding blades swing away from each other during half of each rotor revolution (during the intake cycle), and said blades swing toward each other during the remainder of each rotor revolution (during the gas compression cycle). This expansion and contraction of the chambers 41 in two directions, results in high efficiency of the compressor.

When the compressor is to be operated at maximum output, the stator is fully raised as shown in Figure 2, to dispose said stator at maximum eccentricity to the rotor R. As this eccentricity is decreased by raising the stator S, the output is correspondingly decreased and when stator and compressor are coaxial (Figure 4) there is no output. It will be recalled that the springs 29 bias the stator S downwardly and that hydraulic fluid pressure is utilized in the chamber 24 to raise said stator. A port 64 is shown in Figure 2 for admitting the hydraulic pressure to the chamber and for relieving pressure from this chamber as required.

In addition to the general structure above described, the compressor preferably includes numerous details. Assuming that Freon or other gas having lubricating qualities, is to be handled by the compressor, some of this gas may be conducted to the contacting surfaces of the stub bushing 39 and blade bearings 38 through suitable ports including a supply port 65 (Figure 3) in the stub 36 and communicating with the chamber 8. Ports 66 and 67 may also be provided for conducting some of the gas to the bearings 11 and 13, respectively. Another port 68 is shown in the rotor shaft 42 for conducting some of the gas to a chamber 69 in which sealing means 70 for said shaft is located. Another port 71 is shown directly placing the chambers 14 and 69 in direct communication with each other.

In Figure 7, one of the piston blades 37 is shown as having chambers 72, gas ports 73 leading to said chambers 72, and gas ports 74 extending from said chambers into the recess 75 which receives the sealing means 40. This sealing means (see Figures 7 and 9) preferably comprises two stepped and overlapped sealing members 76, and a spring 77 for holding them against the end members of the stator S. Also, a spring 78 (Figure 7) is shown to hold the sealing members 76 against the stator side wall 25.

In Figure 10, a rocker is shown consisting of two stepped interengaging sections 79 and springs 80 for exerting outward longitudinal pressure thereon. Each rocker section 79 has a flat inner side 81 for slidable contact with one of the piston blades 37, and a transversely arcuate outer side 82 for contact with one of the bearing members 49 of the rotor R. One of these rockers is dis-

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posed at one side of each blade 37 and one at the other side thereof, and the arcuate outer sides 82 then curve about a common center. These arcuate sides 82 gas-tightly contact with the bearing members 49 of the rotor, and the outer ends of the sealing members 79 gas-tightly contact with the end members of the stator.

In Figure 11, a one-piece rocker 83 is shown which could be employed instead of the rocker of Figure 10 but with less efficiency.

#### Control unit U

The construction of the control unit U is shown in Figures 12, 13, 14 and 15 to which attention is now directed. A casing body 84 is provided, said body having two parallel vertical bores 85 and 86 which both open through the lower end of said body. An end plate 87 is secured to the body 84 by screws 88 and spans the lower ends of the bores 85, 86. The upper end of the body 84 is provided with a shallow recess 89 at the upper end of the bore 86 and with a gas inlet 90 into said recess. The other bore 85 has a closed upper end 91, an upper hydraulic fluid outlet port 92 near said upper end, and a lower hydraulic fluid outlet port 93 between the ends of the bore. A vertical port 94 is formed in the body 84 between the bores 85 and 86, the upper end of said port 94 being in communication with the upper end of the bore 85. The lower end 95 of the port 94 opens into the bore 85 opposite the lower hydraulic fluid outlet port 93.

A lower diaphragm 96 lies upon the upper end of the body 84 and spans the recess 89 to coact with the latter in forming a gas-receiving chamber 97. The diaphragm 96 has a relatively large effective area. A spacer ring 98 lies upon the peripheral portion of the diaphragm 96, and an upper diaphragm 99 lies upon said spacer ring, said upper diaphragm 99 having a relatively small effective area. A lower cap 100 lies upon the peripheral portion of the diaphragm 99, and this cap and diaphragm coact in forming a lower hydraulic fluid chamber 101. A port 102 places this chamber 101 in communication with the upper end of the bore 84. An upper cap 103 lies upon the cap 100 and coacts with the latter in forming an upper hydraulic fluid chamber 104. This chamber 104 has an inlet 105, and a central aperture 106 is provided from said chamber 104 to the chamber 101. Screws 107 secure all of the elements 96, 98, 99, 100 and 103 to the body 84.

A valve 108 is provided for closing and opening the aperture 106, said valve being downwardly closeable upon a seat 109 on the cap 100. A spring 110 biases the valve 108 toward closed position and is provided with an adjusting screw 111. A stem 112 extends downwardly from the valve 108 and is secured to both of the diaphragms 96 and 99.

A sleeve valve 113 is slidable in the bore 85 and has arms 114 projecting downwardly through openings 115 (Figure 14) in the end plate 87. Longitudinal slots 116 are formed through opposite sides of the sleeve valve 113, and the lower ends of said slots are cooperable with the port 93 and the lower end 95 of the port 94, respectively.

A substantially horizontal lever 117 is disposed under the end plate 87, and one end of this lever is linked at 118 to the lower ends of the sleeve valve arms 114, the other end of said lever being fulcrumed to the end plate 87 by linkage 119. A vertically elongated bellows 120 and a spring 121 therein, are employed to operate the lever 117. The bellows 120 is disposed within the bore 86 and has its lower end fluid-tightly secured at 122 to the end plate 87. The upper end wall 123 of the bellows 120 is secured to the upper end of a rod 124, the lower end of this rod being pivoted at 125 to the lever 117. The spring 121 surrounds the rod 124 and biases the bellows toward extended position, and bellows extension is limited by a suitable stop 126. The lower end

of the spring 121 abuts an adjusting screw 127 through which the rod 124 extends slidably. The bellows may vent through this screw or a separate vent 128 may be provided. This vent extends through the end plate 87 and communicates with a protecting housing 129 in which the lever 117 is located. The bellows 120 is subjected to gas pressure in the chamber 97 and when said pressure acts to contract said bellows, the rod 124 and lever 117 slide the sleeve valve 113 downwardly.

A piston valve 130 is slidable in the sleeve valve 113 and has a head 131 coacting with the closed end of the bore 85 in forming a hydraulic fluid chamber 132. The piston valve 130 has a peripheral groove 133 for cooperation with the upper ends of the slots 116 of the sleeve valve 113. A spring 134 acts upwardly on the piston valve 130 and reacts against an adjusting screw 135 threaded through the end plate 87. Piston rings 135 are provided for sealing between the piston valve 130 and the sleeve valve 113, and sealing rings 136 are provided for sealing between said sleeve valve and the wall of the bore 85.

#### *Connections between compressor C, control unit U and tank T*

The compressor C, pump P, control unit U and hydraulic fluid tank T are connected by various lines as seen in Figure 1. However, Figures 2, 3 and 13 showing portions of the lines should also be kept in view.

A hydraulic fluid line 137 extends from the lower end of the tank T to the inlet of the pump P, and another hydraulic fluid line 138 extends from the pump outlet to the admission port 105 (Figure 13) of the hydraulic chamber 104. A branch line 139 extends from the line 138 to the tank T and includes a relief valve 140 for relieving excess hydraulic pressure and returning it to said tank. A line 141 is provided to equalize the pressure in the top of the tank T with that in the gas chamber 14 of the compressor C. In the present disclosure, this line 141 communicates with the seal-containing chamber 69 (Figure 3), which chamber communicates with the chamber 14 through the port 71. Another hydraulic line 142 connects the port 92 (Figure 13) of the control unit U with the port 64 (Figure 2) of the hydraulic chamber 24 under the stator S of the compressor C, and yet another hydraulic line 143 extends from the port 93 of said control unit U to the tank T.

A gas line 144 places the gas admission port 90 (Figure 13) of the control chamber 97 in communication with the gas chamber 14 (Figure 3) of the compressor C. In the present disclosure, this line 144 is connected with the port 15 which in turn communicates with the chamber 14.

#### *Operation*

Due to the line 144, the diaphragm 96 is subjected to the pressure of the gas flowing through the chamber 14 on its way to the compressor chambers 41 (Figures 2 and 4) for compression. Also, due to the line 144, the bellows 120 is subjected to said gas pressure. Thus, this gas pressure is instrumental in operating the diaphragm 96 to open the valve 108 and is also instrumental in operating the bellows 120 to thereby operate the sleeve valve 113. When the valve 108 is open, hydraulic fluid supplied to the chamber 104 by the pump P flows through aperture 106 into chamber 101, from this chamber through port 102 into chamber 132, and from this chamber through line 142 to chamber 24 (Figures 2 and 4) of the compressor C. As the hydraulic pressure increases, it acts on diaphragm 99 to cause movement of the valve 108 toward closed position. In view of the fact that the piston valve 130 is subjected to the hydraulic pressure in chamber 132 and the bellows 120 which operates the sleeve valve 113 is subjected to the pressure of the gas entering the gas chamber 97, said sleeve valve and piston valve coact to regulate the hydraulic pressure

in the chamber 132 and therefore in the compressor chamber 24 as required. Here, it should be recalled that the hydraulic pressure in chamber 24 controls the position of the stator S with respect to the rotor R, and that the position of said stator dictates the compressor output.

As the gas leaves the low pressure side of the system for recompression into the high pressure side, the pressure of said gas varies from a minimum to a maximum of around 40 pounds, and it is estimated that the hydraulic pressure supplied by the pump P should double the pressure of the gas, from minimum to maximum. Thus, the relative effective areas of the diaphragms 96 and 99 are such as to properly utilize these gas and hydraulic pressures to automatically actuate the control unit U and thereby automatically control compressor output in accordance with the needs of the system.

Assuming that the compressor is pumped down, there would be no appreciable hydraulic pressure in the chamber 24 and the stator S would be in axial alignment with the rotor R (Figure 4). Thus, upon compressor starting, the compressor chambers 41 would neither expand nor contract and no gas compression would occur. When the switch of the compressor driving motor is turned on, the customary solenoid valve of the system opens and releases the liquid refrigerant to the usual expansion valve. In passage through the expansion valve, the refrigerant is transformed as usual into a gas which is received in the low pressure side of the system. Thus, the gas pressure in the compressor C and in the chamber 97 of the control unit U will gradually increase. At the same time, the pump P is building up hydraulic pressure to the limit in the chamber 104 of the control unit U. The built up hydraulic pressure in chamber 104 is trapped therein by the valve 103 and this valve does not open until the gas in chamber 97 reaches the pressure for which the control unit U is set. In the meantime, the compressor C has been operating without output due to the axial alignment of rotor R and stator S and ample time has been allowed for the chambers to clear of any accumulated oil and liquid which might otherwise cause slugging.

As soon as valve 108 opens, the previously trapped hydraulic pressure is released from the chamber 104 to the chamber 101, thence through port 102 to chamber 132 and thence through line 142 to the chamber 24 under the stator S. Consequently, this stator is moved upwardly to start compressor operation. However, due to the sleeve valve 113 and piston valve 130, the control unit U only admits sufficient hydraulic pressure to chamber 24 to cause the compressor output to meet the demands of the gas pressure in the low pressure side of the system. As above stated, it is estimated that the hydraulic pressure should double the gas pressure. Therefore, if the gas in the low pressure side of the system and in chamber 97 of the control unit U has a relatively low pressure of say 10 pounds, only 20 pounds of hydraulic pressure will be admitted to the chamber 24 and this is sufficient to only partially raise the stator S (about one-fourth of its travel) to the required position to cause the compressor to handle the 10 pounds of gas pressure. Moreover, as the gas pressure rises or falls, the control unit U insures a corresponding rise or fall of the hydraulic fluid pressure in chamber 24 and the position of the stator S is thus changed to properly relate the compressor output with the pressure of the gas to be compressed.

Assuming that the control unit is set to open the valve 108 at 10 pounds gas pressure, this amount of pressure acts on the diaphragm 96 to open said valve 108, and this valve will remain open until the hydraulic pressure increases to 20 pounds, whereupon this hydraulic pressure will act on the diaphragm 99 to again close said valve.

Relief of hydraulic pressure from chamber 24 to permit lowering of the stator S correspondingly to any lowering of the gas pressure is effected by the sleeve valve 113 and piston valve 130. In this connection, it should

be recalled that the head 131 of the piston valve 130 is subjected to the hydraulic pressure in the chamber 132, and the bellows 120 which operates the sleeve valve 113 is subjected to the gas pressure in chamber 97. The above mentioned 10 pounds of gas pressure compresses the bellows 120 sufficiently to move the sleeve valve 113 downwardly about one fourth of its travel. Also, the above mentioned 20 pounds of hydraulic pressure presses the piston valve 130 down about one-fourth of its travel. In these one-fourth-down positions, the sleeve valve 113 and piston valve 130 do not cause any relief of hydraulic pressure from the chambers 132 and 24, and the stator S thus remains in the one-fourth-up position to which it has been moved by the 20 pounds of hydraulic pressure. However, if the gas pressure decreases to say 7 pounds, the bellows 120 expands sufficiently to raise the sleeve valve 113 to a position in which its slots 116 communicate with the groove 133 of the piston valve 130. At this time, the lower end of one slot 116 is in communication with the lower end 95 of the port 94 and the lower end of the other slot 116 is in communication with the lower end of the port 93. Thus, hydraulic pressure is released through the line 143, the pressure in the compressor chamber 24 is correspondingly lowered, and the stator S moves downwardly to adjust the compressor output for handling the seven pound gas pressure.

If the gas pressure to be handled by the compressor increases to 40 pounds, the control unit U will supply hydraulic pressure at 80 pounds to the chamber 24, thereby raising the stator S to the position of Figure 2, for maximum compressor output. The sleeve valve 113 and piston valve 130 then occupy fully lowered positions and the slots 116 are then out of communication with the groove 133 to trap the 80 pounds of hydraulic pressure in said chamber 24 until the gas pressure again lowers, whereupon said valves effect lowering of the hydraulic pressure in accordance with the gas pressure and change the position of the stator S accordingly.

From the foregoing, it will be seen that novel and advantageous provision has been disclosed for attaining the desired ends. However, attention is again invited to the possibility of making variations within the scope of the invention.

#### I claim:

1. A rotary compressor comprising a stationary housing having a side wall and two end walls secured to said side wall, one of said end walls having a gas-receiving chamber in its inner side and a gas inlet for this chamber, the other of said end walls being provided with a shaft bearing; a cap plate secured against the outer side of said other end wall and having a second shaft bearing aligned with the aforesaid shaft bearing, the interior of said cap plate constituting a gas-conducting chamber; a stator within said housing, spaced downwardly from the upper end of said housing and segregating the upper and lower portions of said housing from each other, said stator being vertically shiftable in said housing and comprising a side wall, a complete end wall contacting with said one end wall of said housing, and an end ring contacting with said other end wall of said housing; means

for raising and lowering said stator; a hollow rotor within said stator and disposed eccentrically thereof, said hollow rotor having a drive shaft extending through the aforesaid shaft bearings, said hollow rotor having end members contacting with said complete end wall and said end ring of said stator respectively; piston blades disposed radially of said stator and extending from the interior to the exterior of said rotor; bearing means mounting the inner ends of said blades on said complete end wall of said stator, said bearing means being coaxial with the stator side wall; and rocker means slidably and rockably connecting said blades with said rotor; the aforesaid end wall and end ring of said stator, the aforesaid end members of said rotor, and the aforesaid other end wall of said housing having passages for conducting gas from the aforesaid gas-receiving chamber into the hollow rotor and from this hollow rotor into the aforesaid gas-conducting chamber, a gas passage from this gas-conducting chamber to the space between the upper end of the aforesaid housing and the stator, said stator having means for admitting this gas to its interior for compression, said stator also having a compressed gas outlet.

2. A structure as specified in claim 1; said means for admitting gas to the interior of the stator comprising one port extending directly from said space to the stator interior, a second port spaced circumferentially from said one port, and an arcuate port in the stator side wall and extending from said space to said second port.

3. A structure as specified in claim 1; together with spring means acting on said stator and reacting on said housing for biasing said stator downwardly into eccentric relation with said rotor, and means for admitting fluid under pressure into the lower end of said housing to raise said stator into axial alignment with said rotor.

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