

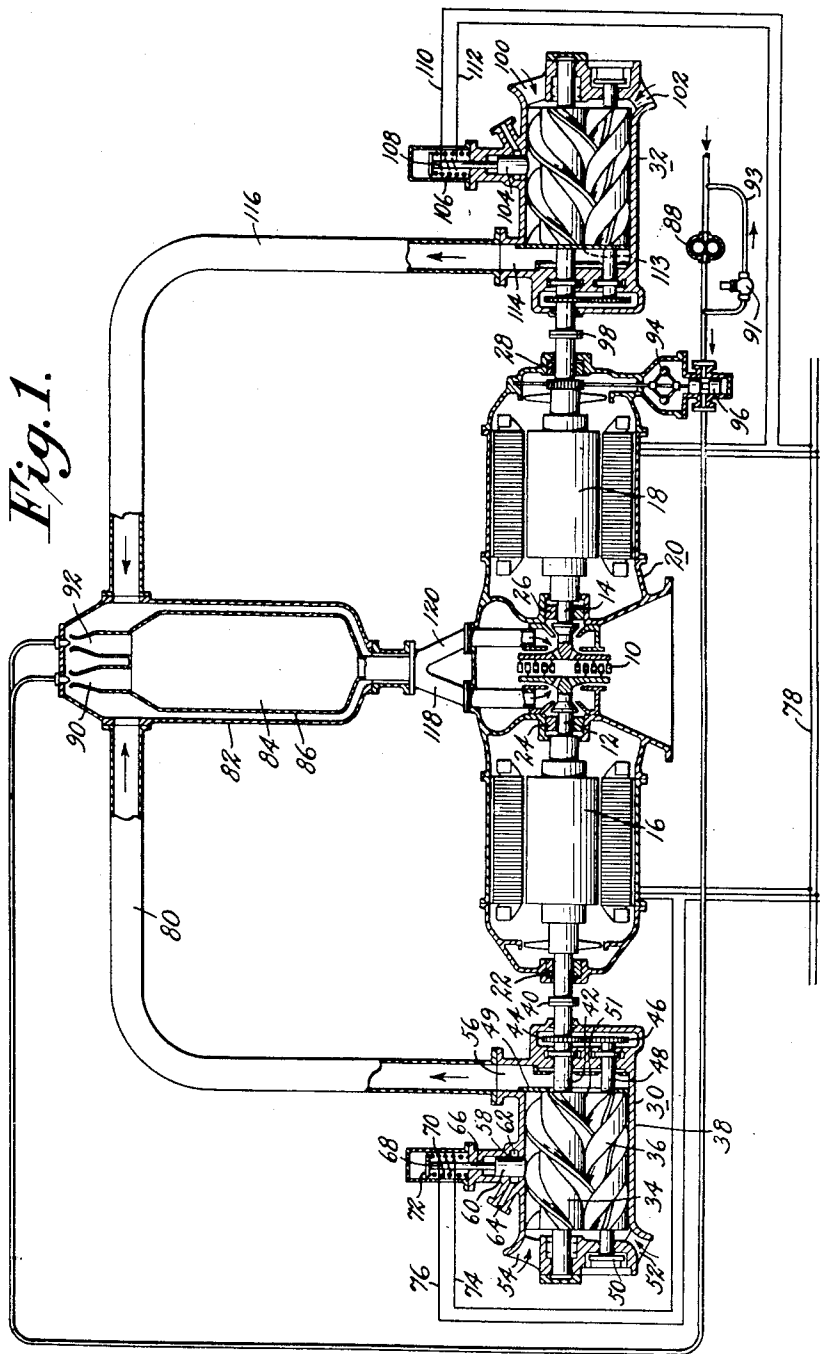
Sept. 26, 1944.

A. LYSHOLM

2,358,815

COMPRESSOR APPARATUS

Original Filed March 28, 1936 3 Sheets-Sheet 1



BY

INVENTOR
A. Lysholm

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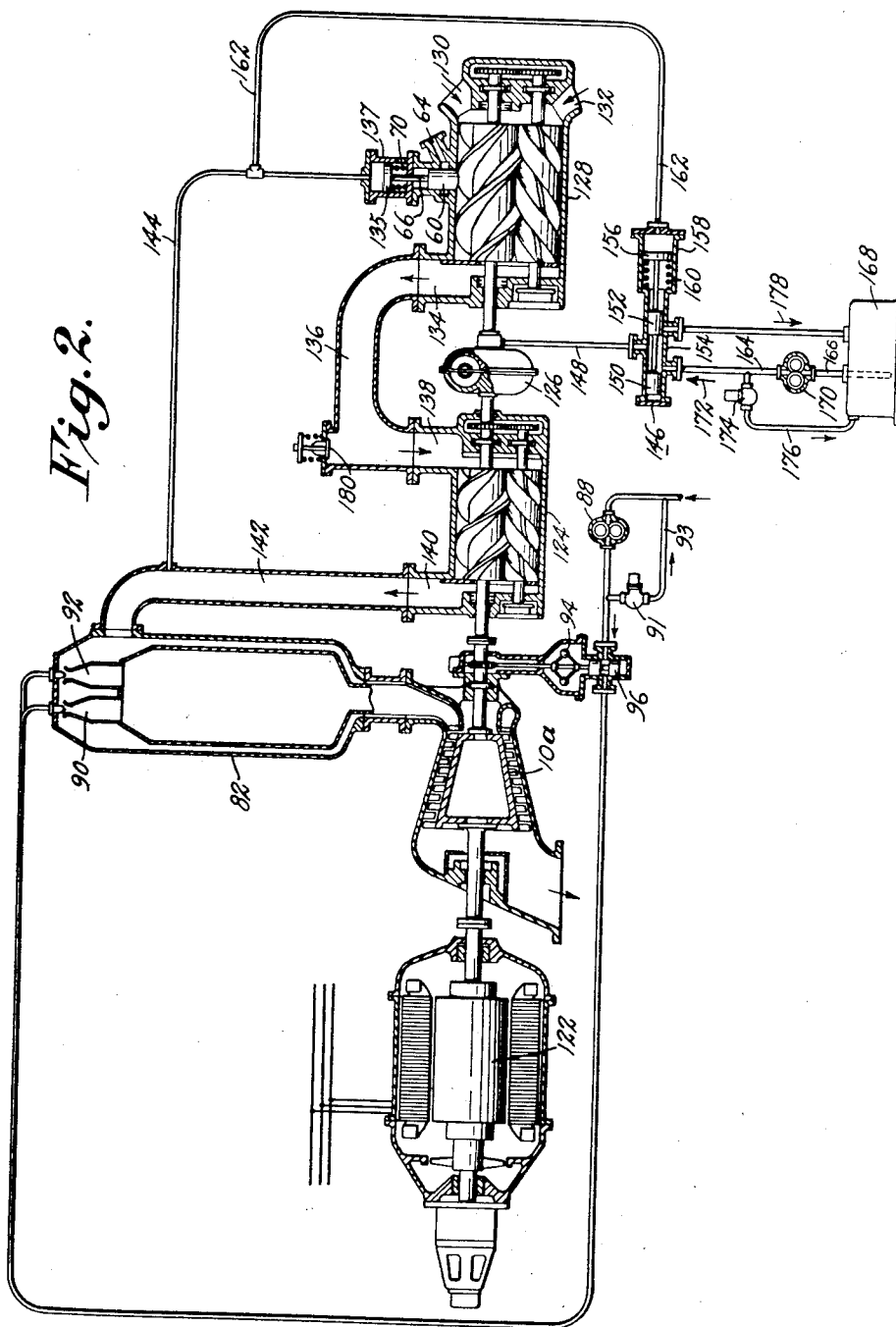
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COMPRESSOR APPARATUS

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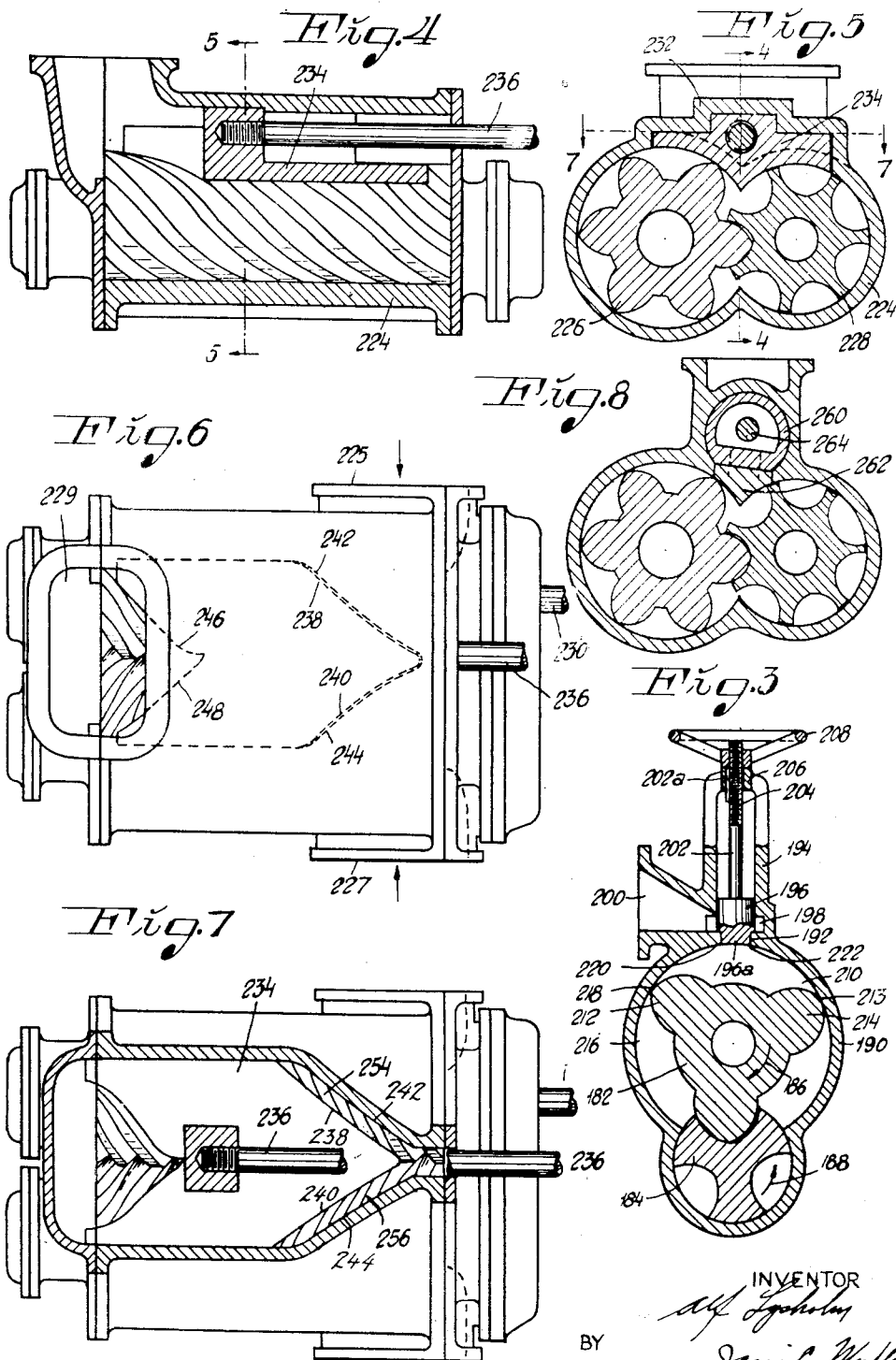
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COMPRESSOR APPARATUS

Original Filed March 28, 1936 3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,358,815

COMPRESSOR APPARATUS

Alf Lysholm, Stockholm, Sweden, assignor, by
meane assignments, to Jarvis C. Marble, Leslie
M. Merrill, and Percy H. Batten, as trustees

Application March 28, 1936, Serial No. 71,363.
Renewed February 8, 1940. In Sweden March
28, 1935

24 Claims. (Cl. 230—143)

The present invention relates to compressor apparatus and has particular reference to rotary compressor apparatus of the screw or worm type. Still more particularly the invention relates to compressor apparatus for operation at constant or relatively constant speed and variable load, particularly as incorporated in gas turbine systems of the continuous combustion type.

Because of the particularly advantageous utility in continuous combustion gas turbine systems of compressor apparatus embodying the principles of the present invention, the invention will be considered and apparatus embodying it will be described, in connection with gas turbine systems of the above mentioned type, but it is to be understood that the invention is not limited in its use to compressor apparatus embodied in this type of system.

In the case of a gas turbine plant where the compressor is driven by a turbine which is mechanically independent of the turbine producing power for external use, the compressor capacity may be controlled within wide limits by changing the speed of operation of the compressor independently of the speed of operation of the useful power turbine, the latter, particularly in cases where it is employed to drive an electric generator, being usually operated at constant speed. This kind of compressor control is however not possible if the compressor is driven by or mechanically connected to the useful power turbine and if the useful power turbine can not be operated at variable speed because of the character of the power output therefrom. In the latter instance, the quantity of air delivered by the compressor can be controlled by throttling to compensate for varying loads. This kind of control is however undesirable because it introduces into the system unavoidable throttling losses. It has also been proposed heretofore to admit suction air to different stages of rotary compressors in a manner such that the suction air is admitted to a pressure stage higher than the normal low pressure stage when the load on the plant, and consequently the required output from the compressor, is reduced. With this method also, the efficiency of the compressor drops materially from its normal efficiency when the compressor is operated under loads materially different from the normal load for which the compressor is designed. Control of the compressor apparatus in order to take care of variable loads has also been suggested, in which the work of compression is divided up among a

number of independently operable compressors, one or more of which may be disconnected, depending upon the amount of change in the load imposed on the system. This last-mentioned method of control is effective only to a certain extent, since by changing the number of compressors in operation it is possible to effect only a very coarse control of the quantity or pressure of the air delivered and in order to obtain the requisite nicety of control, resort must be had in addition to throttling, which involves the introduction of the undesirable and unavoidable throttling losses.

Amongst the major objects of the present invention are: to provide improved compressor apparatus capable of operating with acceptable efficiency over a wide load range while at the same time operating at constant speed or with variations in speed which are relatively minor as compared with the degree of variation of the load; to provide improved compressor apparatus of the screw type in which variable output may be obtained at constant speed of operation without involving undue losses; to provide improved screw compressor apparatus in which variations in capacity of the apparatus are obtained at constant speed by means of novel bleeding arrangements which permit a desired quantity of air to be bled from the compression spaces of the compressor before compression commences; and to provide improved gas turbine system apparatus of the continuous combustion type embodying compressor apparatus and controls therefore enabling the system to be operated efficiently at variable load and with constant or substantially constant speed of operation of the compressor apparatus constituting a part of the system.

For a better understanding of the nature of the present invention and the advantages to be derived from its use, reference may best be had to the ensuing portion of this specification in which is described several embodiments of apparatus for carrying the invention into effect and to the accompanying drawings forming a part hereof, in which such embodiments are illustrated.

In the drawings:

Fig. 1 shows more or less diagrammatically in longitudinal cross section a gas turbine system of the continuous combustion type comprising apparatus embodying the invention;

Fig. 2 is a section similar to Fig. 1 showing another arrangement of apparatus embodying the invention;

Fig. 3 is a transverse section through com-

presser apparatus having hand operated bleeding means embodying the invention;

Fig. 4 is a longitudinal section partly in elevation of another form of compressor apparatus embodying the invention;

Fig. 5 is a section taken on the line 5-5 of Fig. 4;

Fig. 6 is a plan view of the compressor shown in Fig. 4;

Fig. 7 is a view similar to Fig. 6 but with a portion of the compressor casing broken away; and

Fig. 8 is a fragmentary section similar to Fig. 5 of a modification of the compressor apparatus shown in Figs. 4 to 7.

Referring now more particularly to Fig. 1, 10 designates generally a double rotation radial flow gas turbine of known kind having shafts 12 and 14 upon which are mounted the armatures of electric generators 16 and 18. The turbine and generators are mounted in a common frame or casing 20 having bearings 22, 24, 26 and 28. Shaft 12 is arranged to drive a compressor 30 and shaft 14 drives a compressor 32.

Compressor 30 is of the screw type and, in the embodiment illustrated, comprises two rotors 34 and 36 mounted in suitable bearings in housing 38. As will be observed from Fig. 1 and more particularly from Figs. 3, 5, and 8, a characteristic feature of the rotors is that they are dissimilar in cross-sectional contour, consisting of a male rotor having lobes projecting wholly or substantially wholly beyond the pitch circle of the rotor, and a female rotor having grooves lying wholly or substantially wholly within the pitch circle of the rotor. This construction produces working spaces having materially different characteristics from those obtainable with intermeshing rotors of twin form such as are provided by intermeshing gears or rotors of the Roots type and as employed in the appended claims, it will be understood that the terms male rotor and female rotor define rotors of this dissimilar cross-sectional nature.

The rotors are preferably "space packed." That is, clearance is provided between the rotors and the walls of the casing and between the intermeshing portions of the rotors. The rotor construction may advantageously be in accordance with the principles disclosed in my co-pending application, Serial No. 44,935, filed October 14, 1935. Because of the utilization of space packing, the rotors may be operated at the very high speed resulting from direct connection of the rotors to the turbine. This connection is made through the coupling 40 connecting shaft 12 with the shaft part 42 of rotor 34. The gear 44, fixed on the rotor shaft 42 meshes with gear 46 fixed to the shaft part 48 of the rotor 36 and provides drive for the latter rotor and maintains the peripherally spaced relation of the intermeshing parts of the two rotors. Rotor 36 is provided with a pressure equalizing piston 50 for equalizing axial pressure. If the shaft 12 of the turbine is assumed to rotate clockwise when viewed from the left, air which is drawn in through the inlet openings 52 and 54 is compressed between the lobes of the rotors forming working spaces the volumes of which diminish towards the right hand side of the compressor. The compressed air enters the outlet 56 on that side of the compressor which lies in front of the plane of the drawings and which may be termed the pressure side of the compressor as distinguished from the suction side located behind the plane of the

drawings. In order to prevent direct communication between the inlet and the outlet of the compressor, the end surfaces at the right of the rotors are partly closed by an end wall member 48 which on its front side is provided with an opening 51 for the flow therethrough of the compressed air. The compressor casing 38 is provided with a lateral opening 58 providing a cylinder for a valve member such as the plunger 60 which when it is in the position indicated in the figure closes the opening 58. An annular space 62 communicating with an outlet 64 provides communication between the interior of the compressor casing and the atmosphere when the valve member 60 moves outwardly beyond the space 62. It will thus be evident that the opening afforded at 64, which may conveniently be termed a bleeder opening, may be brought into full or partial communication with the interior of the compressor, depending upon the position of the valve member 60.

The compression spaces formed in the compressor will be in communication with the bleeder opening at the moment when normal compression in such spaces commences and if the bleeder valve is opened, compression in such spaces will commence only after the time when the thread or lobe limiting the posterior boundary of the compression space has passed the bleeder opening. This will be at a time subsequent to the normal commencement of compression with the bleeder valve closed.

As indicated in the drawings, it is advantageous to have the bleeder valve extend, in its closed position, inwardly so that the inner end of the valve is flush with the inner surface of the casing and preferably the inner face of the valve is made to conform to the curvature of the casing. The reason for this is to avoid any large leakage space at this point, through which the air being compressed in one compression space may leak past the posterior limiting lobe, to the next succeeding space on the other side of such lobe.

The bleeder valve member 60 is provided with a stem 66 forming the armature of a solenoid 68 and is also encircled by a spring 70 located between the housing and an abutment plate 72. Spring 70 tends to move the valve member to open position and this tendency is opposed by the action of the solenoid winding, which when finally energized has sufficient strength to overcome the action of the spring and maintain the valve member in the closed position shown in the drawings. Solenoid 68 is energized by current supplied through wires 74 and 76 from the generator mains indicated generally at 78.

The outlet 56 of the compressor 30 is connected by means of conduit 80 to a combustion chamber 82 having a combustion space 84 surrounded by a jacket 86. Fuel such for example as oil is supplied by a pump, such as that indicated at 88, to the burner nozzles indicated at 90 and 92. A governor 94 operated from the shaft 14 actuates a fuel control member 96 to control the amount of fuel fed to the combustion chamber under different conditions of load. A pressure relief valve 91 connected to the discharge pipe of the pump 88 permits return of fuel oil to the supply pipe of the pump through the return pipe 93 when the control member 96 throttles or closes the passage of flow. The fuel control apparatus per se forms no part of the present invention and is diagrammatically illustrated. Any suitable apparatus for controlling the amount of fuel in

response to variations in the load on the system may be employed.

The compressor 32, which is shown with its suction side in front of the plane of the drawings, is connected by means of coupling 98 to shaft 14 and is similar in construction to the compressor 30 previously described and need not be described in detail. Compressor 32 has air inlet openings at 100 and 102 and is provided with a bleeder valve 104 urged toward open position by spring 106 and closed by the solenoid 108, the latter being energized by current derived through wires 110 and 112 connected into the electrical system 78. The outlet 114 of compressor 32, which is placed in communication with the compression spaces of the compressor by means of the port 113 which lies behind the plane of the paper, is connected by means of conduit 116 to the combustion chamber 32.

Insofar as air flow is concerned, it will be seen that compressors 30 and 32 are arranged in parallel, both delivering to the outer chamber of the combustion apparatus, part of the air entering the inner chamber 84 to supply the oxygen necessary for combustion of the fuel and part flowing through the jacket space around the inner chamber. Motive fluid consisting of the combustion gases from the inner chamber and the jacketing air is conducted through the twin pipes 118 and 120 to the turbine 10 for expansion in the turbine.

As indicated, the generators 16 and 18 feed into a common electrical system and, as is usual in cases of this kind, operate synchronously.

With normal load on the system, the requisite quantity of fuel is pumped to the combustion chamber and with the motive fluid generated therein by combustion of the fuel with the air compressed in the compressors, the turbine generates the power required to operate the generators and also the power required to in addition drive the compressors. Under such normal load conditions, the bleeder valves of the two compressors are closed and the compressors operate under what may be termed full load compression.

When the load on the plant changes, the fuel supply is governed so that an increased quantity of fuel is supplied upon increased load and a reduced quantity of fuel is supplied with decreasing load. Assuming constant normal full load, the quantity of air delivered by the compressor remains constant. Upon decrease in load, the closing effect exerted by the solenoids which hold the bleeder valves closed is decreased and these valves will open to an extent governed by the degree of the reduction in the load, so as to open more or less the bleeder openings. Consequently, the greater the decrease in load on the plant, the greater will be the quantity of air which is discharged through the bleeder openings and the smaller will be the quantity of air which is compressed in the compressors to the final pressure for delivery to the combustion chamber. When the bleeder valves are opened, the final pressure of the compressed air as well as the quantity of air compressed per unit of time will be reduced, because the ratio of the volume of the compression spaces at the commencement of the compression period to the volume thereof at the end of the compression period is constant. If a constant final pressure is desired regardless of variations in load or if the final pressure is desired to be varied with variations in load in a manner other than that effected with this apparatus, this may be accomplished by means to be here-

inafter described and adapted to vary the ratio of compression space volume at the commencement and end of the compression periods.

The bleeder valve control need not necessarily be made dependent upon the electrical load on the plant but may be controlled by any other suitable factor which changes with and is indicative of changes in the value of the load.

An example of such other control is illustrated in the plant shown in Fig. 2. In this embodiment, the gas turbine 10a is indicated as being of the axial flow type connected for direct drive of a generator 122 which delivers the power output of the system. Turbine 10a also drives directly a compressor 124 similar in type to the compressors previously described except for the omission of the bleeder valve arrangement shown on the compressor illustrated in Fig. 1. Compressor 124 is a high pressure compressor and is connected by means of a hydraulic coupling 126 of known construction to a low pressure compressor 128 which is also of the same screw type as that previously described. Both compressors are viewed from the pressure side. Compressor 128 draws in low pressure air through inlet openings 130 and 132 and has an outlet 134 connected by means of conduit 136 to the inlet 138 of compressor 124. The latter has an outlet 140 connected by conduit 142 to a combustion chamber 32 similar in construction to that described in connection with Fig. 1. A governed fuel supply from pump 88 is supplied to the burner nozzles 90 and 92 as previously described. Motive fluid formed in the combustion chamber from the fuel which is burned with the air supplied from the compressors is expanded in the turbine to produce the power required for driving the generator and the compressors.

The low pressure compressor 128 is provided with an opening 64 and a bleeder valve 60 controlling this opening, the construction being similar to that previously described with reference to Fig. 1. In this instance however, the control of the bleeder valve is different from that previously described. In this embodiment the stem 66 of the bleeder valve is provided with a piston 135 working with a tight fit in cylinder 137 and acted on by a spring 70 tending to move the piston and bleeder valve to a position in which the bleeder passage is opened. The upper side of piston 135 is placed in communication with the compressed air conduit 142 by means of the pipe 144.

The hydraulic coupling 126 is placed in communication with a governor indicated generally at 146 by means of a pipe 148. The governor comprises two plungers 150 and 152 connected to each other and working in a cylinder 154. A piston 156 is rigidly connected to the plungers 150 and 152 and works in a cylinder 158. A spring 160 acts on one side of piston 156 while the other side of the piston is exposed to the pressure existing in conduit 142 and pipe 144 by means of the connection 162. Pipes 164 and 166 connect the interior of the governor with a liquid supply vessel 168. The vessel 168 is located at a lower level than that of the hydraulic coupling so that the working liquid may under certain conditions of operation flow by gravity from the coupling to the supply vessel. A gear pump 170 is located in the pipe 164, the suction side being connected to the supply vessel and the discharge side to the governor. The pump feeds working liquid in the direction of the arrow 172. A pressure relief valve 174, on the discharge side of pump 170, permits return of working liquid from the pump to

the supply vessel through the return pipe 176 when the discharge end of pipe 166 is closed by the governor.

Operation of the system is as follows. Under full load operating conditions, the bleeder valve 60 is closed, as shown in the drawings, the strength of spring 70 being such that the spring is compressed by application of normal full load air pressure to the upper side of piston 135. Governor 146, under the assumed full load condition, is as shown in the drawings, with the pipes 164 and 148 connected and with the inlet end of pipe 166 closed. The working chamber of the hydraulic coupling is filled with working fluid and the two compressors are connected so that both are operated by the turbine. If it is now assumed that the load on the generator is reduced, the fuel supply is correspondingly reduced through the action of the governor and associated control mechanism. This results in a reduction of the temperature of the motive fluid as supplied to the turbine because of the proportionately lower amount of fuel as compared with air supplied to the combustion chamber. The reduced temperature of the motive fluid results in a reduction in the pressure prevailing in the combustion chamber and in the pipe 142 communicating therewith. This will be easily understood from the following calculation. The quantity of motive fluid flowing through a turbine can be expressed by the equation:

$$F = K\sqrt{\frac{P}{v}}$$

where F is the flow of motive fluid in pounds per unit of time, P the initial absolute pressure of the motive fluid in pounds per square inch, v the specific volume of the motive fluid at the pressure P , and K a constant. The value of v may be derived from the well-known thermodynamic equation $Pv = BT$, in which T is the absolute temperature and B a constant. Substituting the value of v in the first equation,

$$F = K\sqrt{\frac{P}{BT}} = K_1\sqrt{\frac{P}{T}}$$

At the first moments after reduction of the fuel supply, the amount of motive fluid flowing through the turbine is approximately the same as before, the reduction in quantity due to the decreased fuel supply being of negligible order as compared with the quantity of air delivered by the compressor. Thus, if P and T represent initial pressure and initial temperature respectively at normal load, and P_1 and T_1 represent initial pressure and initial temperature respectively shortly upon reduction of the fuel supply,

$$F = K_1\sqrt{\frac{P}{T}} = K_1\sqrt{\frac{P_1}{T_1}}$$

or:

$$P_1 = P\sqrt{\frac{T_1}{T}}$$

For instance, if at normal full load $P=80$ pounds per square inch, $T=1800$ degrees F. absolute, and if the temperature due to the reduced fuel supply is decreased to $T_1=1600$, the pressure in the combustion chamber and in the pipe 142 will immediately be reduced to

$$P_1 = 80\sqrt{\frac{1600}{1800}} = 75.4 \text{ pounds per square inch}$$

Due to this reduction in the pressure the bleeder

valve 60 commences to open under the influence of spring 70. This opening or partial opening of the bleeder valve 60 permits air to flow to a greater or lesser extent through the bleeder opening 64 to atmosphere. Thus, a reduced quantity of air, corresponding to the reduced load and the reduced fuel supply, will be compressed and supplied to the combustion chamber. Reduction of the quantity of air compressed will tend to bring the temperature of the motive fluid back to about normal value, the quantity of motive fluid produced per unit of time now being reduced as compared with full load conditions and the pressure of the motive fluid being reduced also because of the reduced pressure at which the air is finally delivered from the compressor system because of the bleeding of some of the air. It will be seen that for every value of part load, there is a definite position of the bleeder valve 60 and consequently for each such value of part load, there is a definite quantity of air discharged from the bleeder opening.

As the load on the system falls below the normal load value and as a consequence the pressure in conduit 142 is reduced, the pressure acting on piston 156 of the governor will also be reduced. This piston then commences to move toward the right, as viewed in the figure, under the influence of spring 160 and this movement is transmitted to the plungers 150 and 152. Initial movement to the right from the position shown in the figure will have no effect on the hydraulic coupling but after the load and the pressure of the motive fluid have fallen below a predetermined value of part load, for example, idling load, the plunger 150 will close the outlet of pipe 164, and simultaneously the plunger 152 will open the inlet of pipe 166. Movement of the governor to this position closes the connection between pump 170 and the coupling, which is instead connected through pipes 148 and 166 to the supply vessel 168 to which the working fluid in the coupling will flow by gravity, in the direction indicated by the arrow 178. Draining of the working fluid from the hydraulic coupling renders this coupling inoperative and the compressor 128 ceases to operate. At this time a spring loaded check valve 180 in the air conduit 136 connecting the two compressors is opened by the suction created by the compressor 124 and air is now compressed only in this compressor.

If the load increases from the value at which compressor 128 is rendered inoperative, the accompanying increase in the amount of fuel supplied will result in rising temperature and pressure of the motive fluid. The increased pressure of the motive fluid acting on piston 156 of the governor moves the plunger valves of the latter to the left and when a predetermined value of load is reached, the original connections are established, that is, the pump is again connected with the hydraulic coupling and the drain pipe 166 is closed. The pump refills the coupling with working liquid and the low pressure compressor 128 is brought back into operation. Reestablished operation of the low pressure compressor produces pressure in the connecting conduit 136 and as a consequence valve 180 closes. With this valve closed, the compressors again operate in series. Further increase in load causes the bleeder valve 60 to be brought more and more nearly to closed position, progressively cutting down the amount of air bled from the low pres-

sure compressor and, when normal full load has been reached, the parts are again in the position shown in the drawings.

While automatic control of the bleeder valve or valves is advantageously employed, manual control may be resorted to and in Fig. 3 a manually controlled valve is illustrated. In this figure, which shows a compressor in transverse section, rotors 182 and 184 operate in the direction indicated by arrows 186 and 188 in casing 190. The casing 190 is provided with a bleeder opening 192 at the lower end of a cylinder 194 in which the bleeder valve 196 of the plunger type operates. A recess 198 communicates with the bleeder discharge passage 200.

Bleeder valve 196 is adapted to seat against the casing in the closed position of the valve shown in the figure and is advantageously provided with an extension or plug portion 196a, the lower surface of which is curved to form a continuation of the cylindrical inner surface of the casing. A space 199 surrounding the valve communicates with the bleeder outlet passage 200. A stem 202 is attached to the valve and is provided with a key 202a working in a suitable slot in the yoke 206 to prevent rotation of the bleeder valve. The upper portion of the stem is threaded at 204 through the hub of a hand wheel 208, the rotation of which on the threaded stem acts to lift the valve.

In the position of the apparatus shown, the bleeder valve is closed and compression of trapped air takes place in the compression space 210 between the threads or lobes 212 and 214 of the rotor 182. Space 210 extends helically toward the suction side of the compressor to a point where a cooperating thread or lobe of the rotor 184 projects into the space so as to progressively reduce its volume upon rotation of the rotors in the directions indicated. The succeeding compression space 216, looking in the direction of rotation of the rotor, is still in communication with the suction side and remains so until the edge 218 of the lobe 212 reaches the left hand limiting edge 220 of the bleeder opening. At this moment, compression commences in space 216. If the bleeder valve 196 is then opened, air may escape through the bleeder opening without its having been appreciably compressed. From the foregoing it follows that, for instance, space 210 is in communication with the bleeder valve 196 during the period from the instant the edge 218 of the lobe 214 reaches the left hand limiting edge 220 of the bleeder opening to the instant the edge 218 of the lobe 212 reaches the right hand limiting edge 222 of the bleeder opening.

Advantageously the bleeder opening 192 is shaped so that communication of the compression space with the atmosphere takes place with minimum throttling losses. To this end, it is preferable to make the limiting edges 220 and 222 of the bleeder opening parallel, or approximately parallel, to the helical edge 218 of lobe 212 and consequently in the same relation with respect to the corresponding edges of the remaining lobes of the rotor. With the limiting edges of the bleeder opening parallel to the outer edges of the rotor lobes, maximum speed of opening and closing of the bleeder opening with respect to the compression spaces, will be obtained.

By making the lower surface of the bleeder valve curved to conform to the curvature of the inner surface of the casing, leakage back from the compression space 210 to the space 216 is prevented when the edge 218 passes the bleeder

opening 192 under full load operating conditions when the bleeder valve is closed.

As previously mentioned, the relation of the variation in final compression pressure to the variation in the quantity of air compressed at part loads can be varied as desired by changing the compression ratio which is effected under different conditions of load.

Apparatus for accomplishing this is shown in Figs. 4 to 7.

Referring now to these figures, the compressor comprises a casing 224 in which rotors 226 and 228 are mounted, these rotors being geared together as in the compressors previously described and being adapted to be driven by means of a power input shaft 230. In this embodiment, six lobes are shown on one rotor and seven on the other in contrast to the three lobes shown for example in Fig. 3. It will be understood that the number of lobes per rotor may be varied as desired within the scope of the present invention.

The casing 224 is provided with inlet openings 225 and 227 and an outlet 229. In this instance, it will be observed that the inlet ports in the casing provide for both radial and axial admission of air to the compression spaces and both radial and axial discharge from the compression spaces.

Casing 224 has a portion 232 which does not follow the contour of the rotors but is spaced therefrom and in the space provided between the rotors and this portion of the casing, a slide 234 is provided, the inner surface of which is formed to the same radius as the radii of the rotors. This slide is axially movable as indicated in Fig. 4, through the medium of a control rod 236 which may be either manually operated or operated automatically in response to load variations on the system in which the compressor is incorporated, as has already been described in connection with Figs. 1 and 2.

The end of the slide at the inlet end of the compressor is preferably provided with curved edges 238 and 240 adapted to abut against correspondingly curved edges 242 and 244 when the slide is moved to the right hand end of its travel as seen in the drawings. At the outlet end of the compressor, the slide is advantageously provided with curved edges 246 and 248, these latter edges determining the area for radial exhaust from the compression spaces and consequently affecting the total exhaust area from these spaces and the discharge pressure of the compressed air.

If we now assume the slide to be moved to the extreme right hand position with edges 238 and 240 abutting respectively against edges 242 and 244, compression will commence at the moment intended for normal full load compression. This will be at the time when the edges of the rotor lobes, which cooperate to form a given compression space, pass the limiting edges of the inlet ports.

If it is desired to decrease the quantity of air delivered by the compressor, the control member or slide 234 is moved to the left from its extreme right hand position, to a position such for example as that shown in Fig. 7 and this movement opens up passages 254 and 256 between the end edges 238 and 240 of the slide and the respectively cooperating casing edges 242 and 244. It will be evident that with the slide moved to this position, commencement of compression in any given compression space will be delayed until the edges of the lobes defining the posterior boundary

of such space pass beneath the slide edges 238 and 240.

If for the moment we disregard the effect on the outlet opening of the movement of the slide to the left from its full load position, it will be evident that the compression ratio will be altered by such movement to the left since the volume of the compression space at the moment compression commences is decreased as compared with full load adjustment, while the volume of the compression space at the moment when the exhaust port opens remains constant. This would result in decreased pressure at the moment of discharge from the compressor space.

Obviously however, movement of the control slide toward the left to a part load position will cause the area of the outlet opening to be decreased and, consequently, as the volume of the compression space at the instant compression commences is decreased by movement of the slide, the volume when exhaust occurs is decreased and the final compression volume is also decreased. It will be evident that the change in the relation of the volume of the compression space at the moment when compression commences to the volume of the compression space at the moment when exhaust or discharge commences can be varied as desired by suitably relating the contours of the edges of the slide which control the commencement of compression and the termination of compression in the various positions of the slide.

From the foregoing description of this embodiment, it will be evident that the control slide constitutes a bleeder valve for delaying the normal time of commencement of compression as compared with full load compression. By use of the slide form of bleeder valve, it is however possible, if desired, to accomplish the additional function with this single valve member of obtaining such constancy or variation in the compression ratio with change in compressor capacity as may be desired.

Fig. 8 shows a modification of the apparatus illustrated in Figs. 4 to 7. The construction is similar to that described above except the form of the control member or slide which in this instance is in the form of a cylinder 260 the lower portion of which is cut off and connected with a member 262 the inner surface of which is formed to the same radius as the radii of the rotors. As in the example previously described, the control member is axially movable by means of a control rod 264. As will be easily understood from the drawings, the shape of the control member 260 facilitates manufacturing of the apparatus as compared with the shape of the control member according to Figs. 4 to 7.

From the foregoing description, it will be evident that many specific different forms of structure may be employed within the scope of the invention and it is to be understood that the invention embraces all that falls within the scope of the appended claims when they are construed as broadly as is consistent with the state of the prior art.

Certain subject matter disclosed but not claimed in this application forms the claimed subject matter of my copending application Serial Number 500,010, filed August 25, 1943.

I claim:

1. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors hav-

ing spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, a bleeder opening in said casing located so that compression can commence in any one of said spaces only after the rotor lobe defining a posterior boundary of the compression space has passed said opening and means for controlling flow through said opening.

2. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, a bleeder opening in said casing located so that compression can commence in any one of said spaces only after the rotor lobe defining a posterior boundary of the compression space has passed said opening, said opening having opening and closing edges disposed substantially parallel to the edges of the rotor lobes passing the opening and means for controlling flow through said opening.

3. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, a bleeder opening in said casing for permitting escape from the compressor of fluid in said spaces, a valve for controlling said opening, said valve having a smooth inner surface which in closed position of the valve provides substantially a continuation of the inner surface of the casing and means for actuating said valve to open or close said opening.

4. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, and means movable relative to said casing to alter the volume of the spaces at the beginning of compression therein.

5. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having

spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, and means movable relative to said casing for altering the volume of the spaces at the commencement of compression therein and simultaneously altering the volume of the compression spaces at the termination of compression therein.

6. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall and a member movably mounted between said casing and said rotors and cooperating with the rotors and with portions of the casing to vary, upon movement of the member, the volumes of the spaces at the commencement of compression and the volumes of the spaces at the termination of compression, the variation in said volumes due to such movement of the member effecting a change in the compression ratio of the compressor.

7. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall to compress and discharge fluid adjacent said end wall, an axially slidable member interposed between said rotors and a portion of said casing, said member having an end arranged to move away from cooperating portions of the casing to provide passage for escape of fluid from the spaces upon movement of said member toward the discharge end of the compressor, whereby to delay commencement of the compression period and said member having an opposite end portion for reducing the area of the outlet opening of the compressor upon movement of the member toward the discharge end of the compressor, and means operable from the exterior of the compressor for adjusting the position of said member.

8. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rota-

tion of the rotors to come into communication with said end wall, and means for varying the capacity of the compressor without changing the speed of operation thereof comprising a bleeder valve member movable to vent said spaces until their volume is less than the volume at the moment of commencement of full load compression, whereby to delay the commencement of compression in such spaces as compared with the commencement therein at full load.

9. In the operation of a rotary compressor of the type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor upon rotation of the rotors so as to come into communication with an end wall of the casing to effect compression, that improvement which consists in varying the capacity of the compressor while maintaining substantially constant speed of operation thereof by altering the effective initial volume for compression of said spaces.

10. In the operation of a rotary compressor of the type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor upon rotation of the rotors so as to come into communication with an end wall of the casing to effect compression, that improvement which consists in reducing the capacity of the compressor while maintaining substantially constant speed of operation by decreasing the effective initial volume for compression of said spaces, and modifying the decrease in delivery pressure which would result from such decrease in effective initial volume by decreasing the effective final compression volume of said spaces.

11. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, and means including a bleeder valve member adjustable to cause initiation of part load compression in any given compression space with said rotors in any one of a series of different angular positions of rotation.

12. A compressor of the rotary screw type including a casing having an inlet and an outlet for fluid and an end wall, a plurality of rotors mounted in said casing, each of said rotors having spiral lobes and intervening spiral grooves of uniform height and depth respectively along their lengths, the lobes and grooves intermeshing and cooperating with each other and with the

casing to form spaces registering at different times with said inlet and said outlet and which are displaced axially of the compressor by rotation of the rotors to come into communication with said end wall, and control means having adjustable portions cooperation with said rotors to vary the relation of the angular position of rotation of the rotors with respect to both the initiation and the termination of compression in any given one of said spaces.

13. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, a bleeder opening in said casing located so that compression can commence in any one of said spaces only after the rotor lobe defining a posterior boundary of the compression space has passed said opening and means for controlling flow through said opening.

14. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, a bleeder opening in said casing located so that compression can commence in any one of said spaces only after the rotor lobe defining a posterior boundary of the compression space has passed said opening, said opening having opening and closing edges disposed substantially parallel to the edges of the rotor lobes passing the opening and means for controlling flow through said opening.

15. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, a bleeder opening in said casing for permitting escape from the compressor of fluid in said spaces, a valve for controlling said opening, said valve having a smooth inner surface which in closed position of the valve provides sub-

stantially a continuation of the inner surface of the casing and means for actuating said valve to open or close said opening.

16. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and means movable relative to said casing to alter the volume of the spaces at the beginning of compression therein.

17. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and means movable relative to said casing for altering the volume of the spaces at the commencement of compression therein and simultaneously altering the volume of the compression spaces at the termination of compression therein.

18. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and a member movably mounted between said casing and said rotors and cooperating with the rotors and with portions of the casing to vary, upon movement of the member, the volumes of the spaces at the commencement of compression and the volumes of the spaces at the termination of compression, the variation in said volumes due to such movement of the member effecting a change in the compression ratio of the compressor.

19. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their

lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, an axially slidable member interposed between said rotors and a portion of said casing, said member having an end arranged to move away from cooperating portions of the casing to provide passage for escape of fluid from the spaces upon movement of said member toward said outlet whereby to delay commencement of the compression period and said member having an opposite end portion for reducing the area of the outlet opening of the compressor upon movement of the member toward the outlet, and means operable from the exterior of the compressor for adjusting the position of said member.

20. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their length and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and means for varying the capacity of the compressor without changing the speed of operation thereof comprising a bleeder valve member movable to vent said spaces until their volume is less than the volume at the moment of commencement of full load compression, whereby to delay the commencement of compression in such spaces as compared with the commencement therein at full load.

21. In the operation of a rotary compressor of the type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotor after passing out of communication with said inlet and before coming into communication with said outlet to effect compression, that improvement which consists in varying the capacity of the compressor while maintaining substantially constant speed of operation thereof by altering the effective initial volume for compression of said spaces.

22. In the operation of a rotary compressor of

the type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet to effect compression, that improvement which consists in reducing the capacity of the compressor while maintaining substantially constant speed of operation by decreasing the effective initial volume for compression of said spaces, and modifying the decrease in delivery pressure which would result from such decrease in effective initial volume by decreasing the effective final compression volume of said spaces.

23. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and means including a bleeder valve member adjustable to cause initiation of part load compression in any given compression space with said rotors in any one of a series of different angular positions of rotation.

24. A compressor of the rotary screw type having a casing providing an inlet and an outlet for fluid, a plurality of rotors mounted for rotation about parallel axes in said casing including a male rotor and a female rotor having intermeshing spiral lobes and grooves respectively, the intermeshing lobes and grooves being of substantially uniform height and depth respectively along their lengths and cooperating with each other and with the casing to form compression spaces registering at different times with said inlet and said outlet and being decreased in volume by contraction in a direction parallel to the axes of said rotors after passing out of communication with said inlet and before coming into communication with said outlet, and control means having adjustable portions cooperating with said rotors to vary the relation of the angular position of rotation of the rotors with respect to both the initiation and the termination of compression in any given one of said spaces.

ALF LYSHOLM.