Aug. 22, 1950 A. LYSHOLM 2,519,913

HELICAL ROTARY COMPRESSOR WITH PRESSURE AND VOLUME REGULATING MEANS

Filed Aug. 21, 1943 3 Sheets-Sheet 1

Fig. 1.

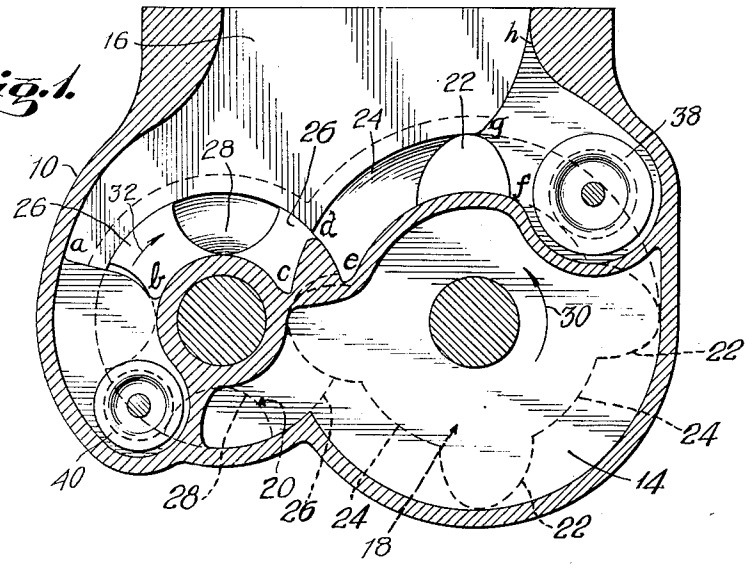
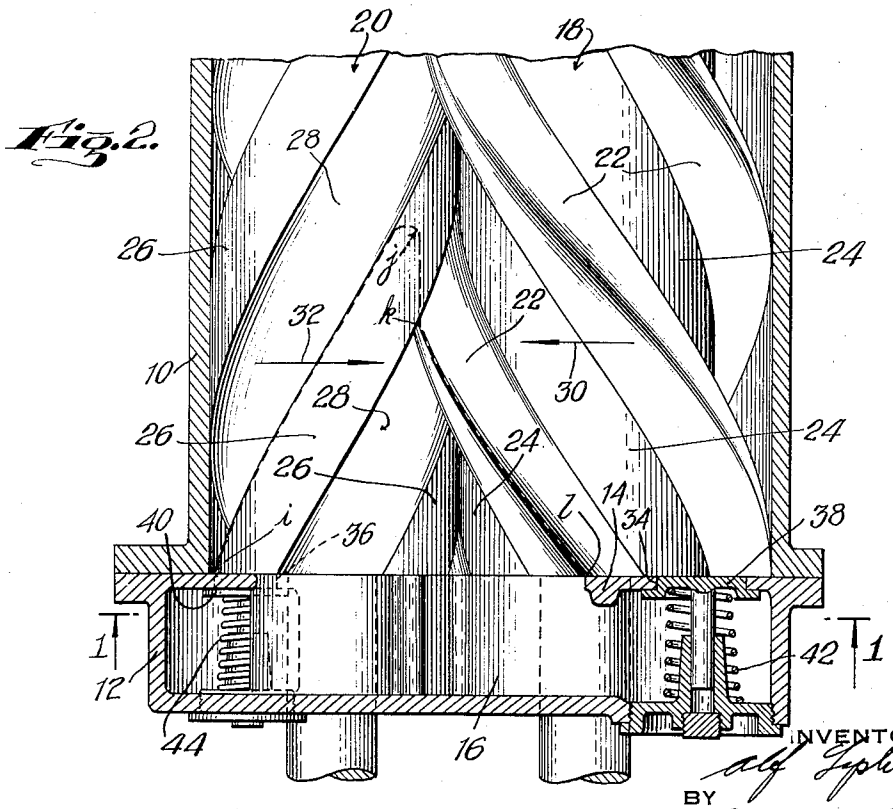


Fig. 2.



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Fig. 3.

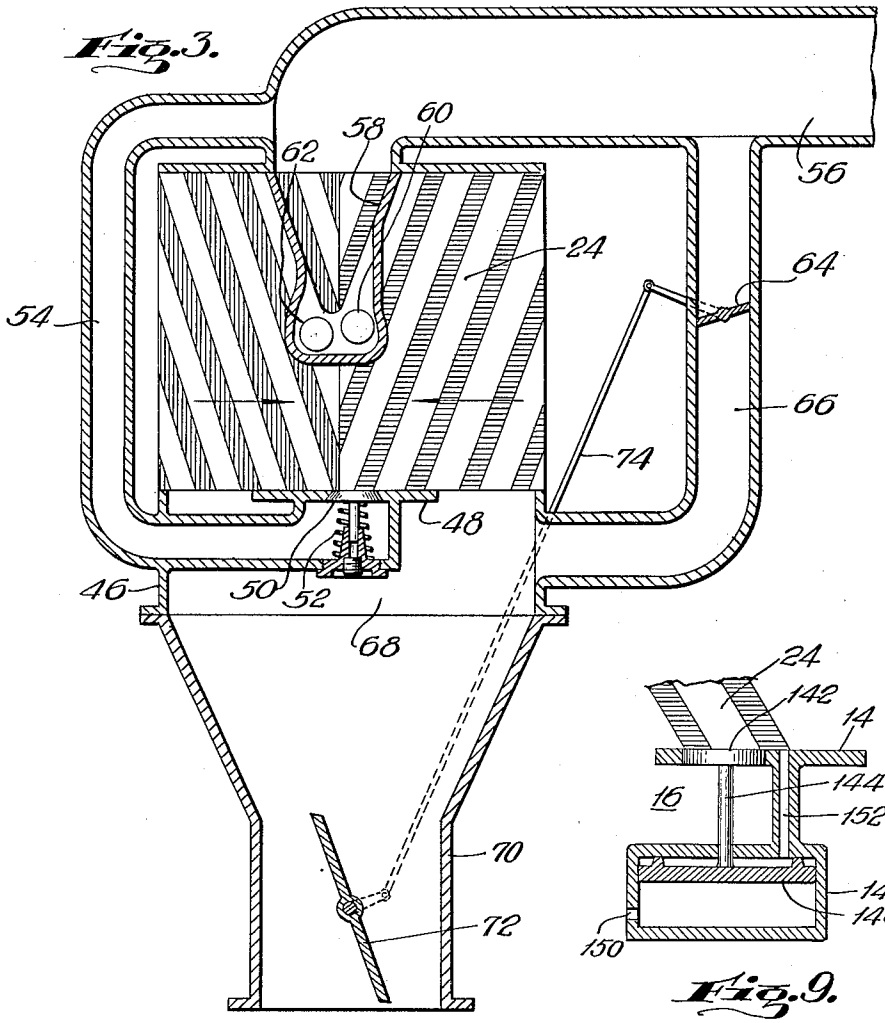
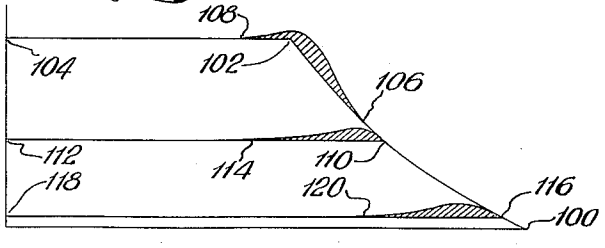


Fig. 9.

Fig. 6.



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Fig. 4.

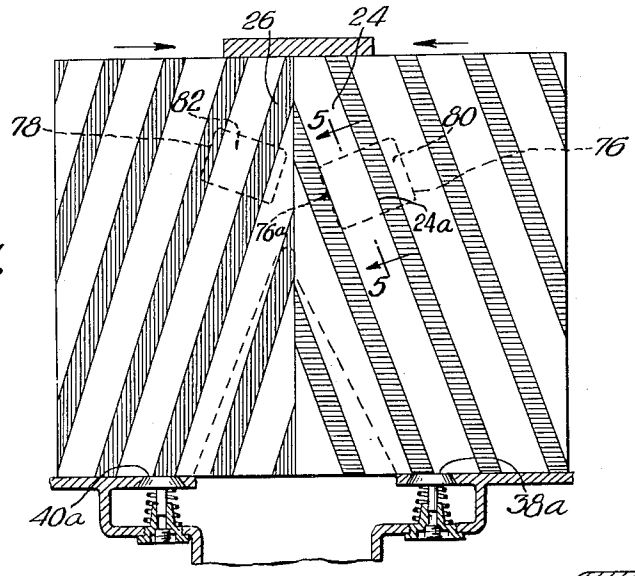


Fig. 5.

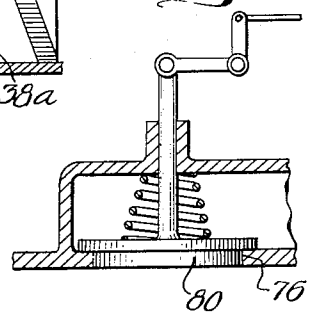


Fig. 7.

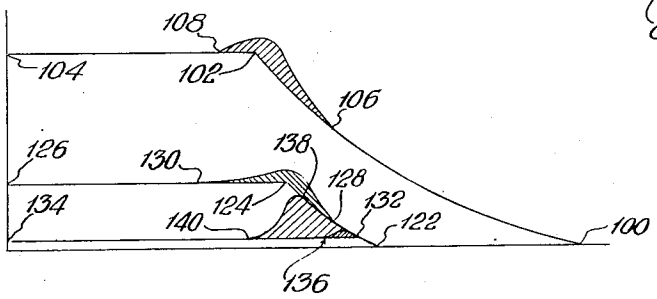
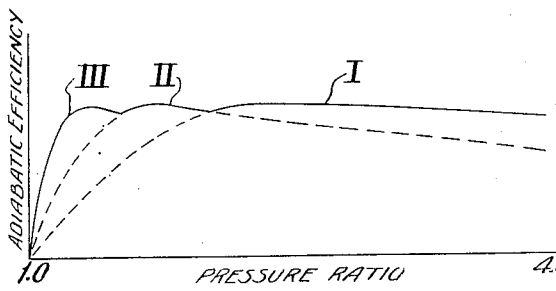


Fig. 8.



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Patented Aug. 22, 1950 2,519,913

UNITED STATES PATENT OFFICE

2,519,913

HELICAL ROTARY COMPRESSOR WITH PRESSURE AND VOLUME REGULATING MEANS

Alf Lysholm, Stockholm, Sweden, assignor, by mesne assignments, to Jarvis C. Marble, New York, N. Y., Leslie M. Merrill, Westfield, N. J., and Percy H. Batten, Racine, Wis., as trustees

Application August 21, 1943, Serial No. 499,467

2 Claims. (Cl. 230—143)

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This application is a continuation-in-part with respect to my copending application Serial No. 439,300, filed April 17, 1942 (now Patent No. 2,383,979 granted September 24, 1945) and relates back as to all common subject matter for all dates and rights incident to the filing of said application.

The present invention relates to compressors and has particular reference to rotary positive displacement compressors of the kind in which working spaces of variable volume are formed by the grooves and lands of intermeshing spirally grooved rotors operating in a suitable casing. More specifically, the invention relates to such compressors of the rotary screw wheel type in which the compression spaces are defined at the discharge or outlet end by a casing end wall, as disclosed in my prior Patents Numbers 2,174,522 and 2,243,874 granted October 3, 1939 and June 3, 1941, respectively. In compressors of the kind under consideration, certain volumetric and adiabatic efficiency characteristics require modification in order to secure the most efficient performance of the compressor in certain types of use where variable quantity and pressure characteristics of the air delivered are required. As an example of such variable delivery characteristics, there may be mentioned the use of such a compressor as a supercharger for an internal combustion aircraft engine which under different conditions of load will require different quantities of combustion air and which will require different ratios of compression to be effected depending upon the altitude at which the engine is operating in order to obtain a desired boost pressure.

The general object of the present invention, therefore, is the provision of novel control means for effecting both pressure and volume control in the operation of a compressor of the kind under consideration, to enable most efficient operation to be effected under conditions requiring either variation in quantity of air delivered, or variation in compression ratio, or both, while operating at substantially constant speed.

For a better understanding of the more detailed nature of the invention, its more detailed objects and the advantages to be derived from its use, reference may best be had to the ensuing portion of this specification, taken in conjunction with the accompanying drawings, which disclose by way of example but without limitation, suitable physical embodiments of apparatus for carrying the principles of the invention into effect.

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In the accompanying drawings,

Fig. 1 is a sectional view of a compressor embodying the invention taken on the line 1—1 of Fig. 2;

Fig. 2 is a fragmentary longitudinal section showing the discharge end portion of the compressor shown in Fig. 1;

Fig. 3 is a diagrammatic developed view illustrative of a different embodiment of the invention;

Fig. 4 is a view similar to Fig. 3 of still another embodiment;

Fig. 5 is a fragmentary section of a valve useable in the embodiment shown in Fig. 4;

Figs. 6 to 8 inclusive are diagrams illustrative of certain operating and efficiency characteristics of apparatus embodying the invention; and

Fig. 9 is a fragmentary view showing a different modification of valve useable in the apparatus shown in Figs. 1 through 4.

Referring now more particularly to Figs. 1 and 2 of the drawings, the compressor illustrated is of the type disclosed in my aforesaid prior patents and includes a casing having a barrel part 10 closed at both ends by suitable end cover members, the one at the discharge or outlet end of the compressor being indicated at 12 in Fig. 2. The structure 12 provides an end sealing wall 14 for closing the working spaces of the compressor in certain positions of the rotors, this end wall being cut away in part to provide an end wall port, the outline of which in the end wall follows the line *a—b—c—d—e—f—g—h* in outlet port 16.

Within the barrel portion of the casing there are mounted the intermeshing spirally grooved rotors 18 and 20 which advantageously but not necessarily are of the form disclosed in my prior Patent 2,174,522 and consist of a male rotor having spiral lands 22 between which are located spiral grooves 24 the bottoms of which substantially coincide with the pitch circle of the rotor, and a female rotor having lands 26 between which are located grooves 28, the pitch circle of the latter rotor substantially coinciding with its outer diameter.

As viewed in Fig. 2, the rotors revolve in the directions indicated by arrows 30 and 32 and by such rotation the compression spaces formed by cooperating grooves such as those indicated at 24 and 28 in Fig. 2 are decreased in length axially toward the discharge end wall 14 to effect compression until the grooves come into registry with the outlet port. As seen in Fig. 2, certain of the grooves 24 and 28 are just about to come

into such registry to effect discharge through the port. In order to effect simultaneous commencement of discharge in both radial and axial directions, the end point *i* of the radial portion of the port advantageously coincides with the end point *a* of the axial portion of the port while the end point *l* of the radial portion of the port coincides with the point *g* of the axial portion of the port.

Insofar as the specific configuration of the inlet port is concerned, this is not critical for the present invention and the inlet port can be either a combined radial axial port, or substantially entirely for axial admission, in accordance with known forms of design. Such porting has therefore not been shown in the figures under consideration.

In accordance with one aspect of the present invention, the end wall **14** is provided with two valve controlled ports **34** and **36** closed respectively by valves **38** and **40**, the discharge sides of which are in communication with the discharge port passage **16**. These valves are loaded respectively by springs **42** and **44** tending to keep them closed and the inner faces of the valves are preferably flat as shown and located so that when the valves are closed the valve faces are flush with the inner surface of the end wall **14** and form in effect a part thereof. If it is assumed for the moment that the compressor is without the valves **34** and **36**, it is characteristic of the operation of the compressor that given a substantially constant speed of operation, the compression ratio effected within the compressor will be constant, since the rotors will always cut off from the inlet port in the same positions of rotation and will always register with the outlet port in the same position of rotation. The compression ratio will depend upon the amount of rotational movement of the rotors between the time of passing out of registry with the inlet port and the time of coming into registry with the outlet port. Change in speed of operation will change the compression ratio only to the extent that the volumetric efficiency is affected by the change in speed, the percentage of leakage loss from the compression spaces becoming progressively greater as the speed decreases.

Now let it be assumed that the compressor is to be used as an engine supercharger for an aircraft engine and that a compression ratio of 1.5 to 1 is desired, in order to produce an engine manifold pressure of approximately 45 inches mercury, absolute. If the inlet and outlet ports are laid out so as to produce this compression ratio at ground level, then the absolute boost pressure will fall as altitude increases, due to the lower pressure of the air as admitted to the compressor. Consequently, for altitude operation, the compressor must be designed to give a compression ratio high enough to maintain an absolute delivery pressure of 45 inches mercury, utilizing rarefied inlet air. If the design is made so as to effect this function, then the final delivery pressure will become too high as altitude decreases, due to the denser inlet air, and if the engine is not to be injured due to too high supercharging pressure, the compressor must be either governed by a variable speed gear or a step gear so that the quanttiy which it will deliver will be decreased, or an air waste gate must be provided for permitting some of the compressed air to escape in order to bring the manifold pressure down to the permissible maximum. The latter expedient is wasteful since excess power is employed for supercharging. The former expedient,, while less wasteful of power, is not wholly satisfactory, since if a step gear is employed practical considerations usually limit it to two speeds which do not give sufficiently small increments in change of performance, while an infinitely variable speed arrangement requires for example, a hydraulic or equivalent drive which introduces substantial additional complication, weight and cost.

By utilizing an auxiliary discharge valve constructed in accordance with the present invention, the delivery pressure can be maintained automatically at the desired value without resort to either an air waste gate or a multiple or variable speed drive, and with the compressor operated at substantially constant speed to a final delivery pressure that may be either constant or variable. The reason for this is, as will be observed from Figs. 1 and 2, that the auxiliary discharge valve or valves is or are located so as to come into registry with the working spaces prior to the time when the latter come into registry with the fixed outlet port. Consequently, the auxiliary outlet valve comes into registry with the respective working spaces when the pressures in the latter are lower than the pressures attained therein by the time they register with the fixed or main dischare port. The auxiliary outlet valve is subject to and will automatically open in response to a differential pressure between that existing in a working space at the time the space is in registry with the valve and the pressure in the discharge conduiit. In order to avoid chattering of the auxiliary valve it is for practical reasons advantageous to spring load the valve as indicated in the present example, but usually this spring loading will be very light so that comparatively little differential pressure is required in order for the valve to open. If conditions so dictate, the spring load can be made of any desired value, so that the auxiliary valve will open only in the event a predetermined differential pressure of substantial value is established.

If now, one considers the action in a specific case such as that previously discussed, the action of the auxiliary valve will be as follows: If we assume that a maximum delivery or manifold pressure of 45 inches mercury is to be maintained and that the supercharger is to be capable of maintaining this pressure at 20,000 feet altitude, the compressor may be laid out to provide a compression ratio of approximately 3.3 to 1. When operating at this altitude with this compression ratio, a manifold pressure of approximately 45 inches mercury can be maintained and since the compressor is designed for a final compression pressure not in excess of this value, the auxiliary outlet valve will not operate since the pressure in the working chambers at the time they register with the auxiliary valve will be less than the outlet or manifold pressure acting on the discharge side of the valve. If the altitude at which the compressor operates decreases, the denser air will cause the absolute pressure in the working spaces to reach the desired manifold pressure of 45 inches mercury earlier in the cycle of operation and when the altitude has decreased sufficiently to cause the pressure to rise appreciably, then a differential pressure is set up which will operate automatically to cause the auxiliary valve to open, thereby limiting the pressure that can be obtained in the compression spaces to the value selected as the maximum value which is

to be produced in the discharge line from the compressor.

Since the working spaces provided by the grooves are spiral in form, the auxiliary outlet or relief valves may be placed in various positions in the compressor casing and by utilizing a suitable number and location of these valves, complete pressure control can be effected so that the device can be made to operate to not overcompress even though the desired discharge pressure may vary from the maximum desired down to inlet pressure. In the latter event, the device would operate substantially as a positive displacement blower without effecting compression. In Fig. 3 there is shown more or less diagrammatically an arrangement of valves whereby this may be effected. In this example the inlet end closure member **46**, which provides a ported inlet end wall **48**, carries an auxiliary pressure control valve **50** loaded by spring **52** and communicating at its discharge side with the passage **54** which leads to the discharge conduit **56** with which the compressor discharge port **58** also communicates. In this case, a groove such as the one shown at **24** comes into registry with this valve before a land from the other rotor has entered the groove to reduce its volume, and will have had its volume reduced only very slightly by the land of the other rotor by the time it passes out of registry with the valve **50**. Consequently, if the pressure in the outlet conduit **56** is very low, valve **50** will open before appreciable pressure or compression has been effected in the groove **24**. In the present example, a second auxiliary valve **60** is located in the barrel portion of the casing as well as a third valve **62**, these valves also being lightly spring loaded and communicating on their discharge sides with the outlet port **58**. By reference to the drawing it will be seen that before the groove **24** has passed out of registry with the relief valve **50** it will have come into registry with valve **60** and differential pressure will act to continue to limit the pressure which can be built up in the groove. Before the groove passes out of registry with valve **60** it comes into registry with the fixed main port **58**. Consequently the pressure attainable in the groove can be limited throughout the compression portion of its cycle. Operation to limit the pressure in the grooves of the companion rotor is also controlled by registry successively with valves **50**, **62** and the port **58**, the several valves and port providing what may be termed "overlapping" communication with the groove.

While in the examples described, different valves have been shown for cooperation with the grooves of each of the rotors, it will be apparent that since the grooves of the male and female rotors are in cooperative communication in pairs with each other during the major portion of the compression part of the cycle, an auxiliarly valve located to register with the groove of one rotor before the groove registers with the exhaust port will operate to limit the pressure not only in the groove with which it is in registry but also in the companion groove of the other rotor. The use of auxiliary valves registering with the grooves of each rotor or of a valve or valves registering only with the grooves of one rotor will be determined in different cases by the amount of air that will be by-passed, the number and size of the valves being chosen to enable the desired maximum amount of air to be by-passed without undue throttling losses occasioned by inadequate valve port area.

In addition to effecting maximum pressure control by means of auxiliary valves such as those described, certain types of installation, particularly supercharging installations, require a volumetric or quantity control, as for example in those cases where it is desired to cruise at a given altitude with a lower than maximum manifold pressure. Such control may be effected in different ways and in Fig. 3 there is shown one arrangement for effecting a satisfactory volumetric or quantity control. In this arrangement, when reduced discharge conduit pressure is desired, a by-pass valve **64** is opened which provides communication through the by-pass conduit **66** from the discharge conduit **56** to the inlet chamber **68** of the compressor.

Advantageously the by-pass control may be combined with a throttling control for reducing the quantity of air admitted to the compressor, as shown in Fig. 3, particularly in those cases where it may be desirable to reduce very substantially the quantity of air as compared with that which would otherwise be delivered. In such case the two controls are advantageously interconnected, and if interconnected are preferably so connected that the by-pass control opens to effect a certain degree of by-passing before the throttling control comes into action. It will be observed that in the arrangement shown in Fig. 3, this sequence is effected, since any movement of the link **74** from the position shown in the figure will operate to open the by-pass valve **64**, while a substantial movement will be required to move the throttle valve **72** from the position shown, across dead center position to a position where appreciable throttling will be effected. Obviously the air passing valve **64** does not necessarily have to be returned to the compressor inlet **70** but this is usually preferable if for no other reason than to reduce the noise produced by open exhaust from a compressor of the kind under consideration.

Since this arrangement will operate to reduce the discharge conduit pressure, the auxiliary pressure relief valves will come into play and the compressor will operate with the minimum of loss due to excess work of compression.

Another arrangement is indicated in Fig. 4, in which in addition to the auxiliary valves **38***a* and **40***a*, similar to the valves **38** and **40** of Fig. 2 and located in the outlet end wall of the compressor, the barrel part of the compressor casing is provided with volume control ports **76** and **78** controlled by valves **80** and **82**, respectively, the discharge sides of which are in communication with the atmosphere or with the inlet passage of the compressor by means of suitable connections. In Fig. 5 one of such volume control valves is indicated. The control of such a valve may be either manual or automatic in response to any condition requiring decreased volumetric capacity from the compressor at a given speed.

As will be apparent from Fig. 4, the air in a groove such as that indicated at **24** will, if the valve **80** is closed, commence to be compressed by the entry into the groove of the land **26** of the companion rotor as the rotors revolve from the positions indicated in the figure. If, however, the valve **80** is open, compression in this groove cannot commence until after the trailing edge **24***a* of the groove has passed the edge **76***a* of the port. From then on compression will ensue but a smaller quantity of air will be compressed. In discussing the above it is assumed that the valve **82** is closed. Obviously the two valves **80** and *82*

can be located in the casing so that they pass out of registry with the respective compression spaces at different times, or, in other words, overlap. Since during the compression cycle pairs of grooves are in communication, one volume control valve will be sufficient to vent each compression space consisting of intercommunicating grooves and as in the case of the pressure control valves, the number and location of the volume control valves will be dictated in individual instances by the volume of air which it may be desired to pass through the valves and the extent to which a series of overlapping valves may be desired in order to provide for varying the volumetric output to a minimum which may be a large or a small fraction of maximum capacity, such control being effected stepwise through the opening of different ones or all of the volume control valves.

In Fig. 6 there are shown compression diagrams of the operation resulting from different operating conditions requiring different final compressions. In this diagram the line 100—102—104 represents the theoretical or ideal compression line at maximum designed delivery pressure and maximum capacity. Point 105 represents the point in the cycle when the compression spaces first come into communication with the fixed or main outlet port, there being a slight rise of pressure in the compression spaces after such communication, resulting in a slight amount of overcompression represented by the shaded area between points 106 and 108. This represents normal full capacity full pressure operation without the pressure control valves coming into action.

The line 100—110—112 represents the ideal compression line at an intermediate delivery pressure with full volume operation, such for example as would occur with the valves 86—92 of Fig. 3 in operation but with the delivery pressure sufficiently high so that the valve 50 would remain closed. Under this condition, the excess compression would not only amount to that indicated by the shaded area between the points 110 and 114. If the delivery pressure is reduced to a value bringing all of the relief valves into action, the ideal curve would be for example that represented by the line 100—116—118 and the excess work of compression would amount only to that indicated by the shaded area between the points 116 and 120.

Thus, under substantially all conditions of operation, no extra work of compression in excess of that which is normally done at full pressure, full capacity operation will be done.

With both pressure and volume control employed, as for example by valves such as indicated in Fig. 4, the compression curves will be as indicated in Fig. 7. In this case the theoretical and actual compression curves will be the same as shown in Fig. 6 and if it is assumed that volume as well as delivery pressure are to be decreased, the ideal curve for an intermediate pressure and intermediate volume will be as indicated by the line 122—124—126, while the overcompression will be as indicated between the points 128 and 130. With reduced volume and very low delivery pressure, the ideal compression line will be 122—132—134 with the overcompression occurring between points 132 and 136. If the pressure control valve arrangement were omitted, the excess work of compression at very low delivery pressure and reduced volume would be that represented by the area 122—138—140.

It will thus be seen that under the most widely varying conditions of volume and pressure requirements, efficient operation is attainable by the means provided by the present invention.

In Fig. 8 typical adiabatic efficiency curves are shown, curve I indicating the character of such efficiency obtainable with normal or full pressure full capacity operation, curve II indicating the efficiency characteristic with volume control alone and curve III indicating the efficiency with combined volumetric and pressure control.

While in the several embodiments previously described the pressure control valves have been shown as governed directly by the discharge pressure and the pressure in the compressor grooves, it will be apparent that servo-motor or other remote controls operative in response to the desired differential pressure may be employed if desired and in Fig. 9 there is shown a form in which the pressure control valve 142 is of cylindrical form and attached by stem 144 to a control piston 146 operating in cylinder 148, one side of the piston being subject to delivery pressure through passage 150 and the other side being subject to rotor groove pressure through passage 152. With this or other remote controls, spring loading of the valve may be omitted.

From the foregoing it will be evident that within the scope of the invention many variations of specific design, combination and arrangement of parts may be made and that certain features may be employed to the exclusion of others. The invention is therefore to be understood as embracing all forms of operation thereof falling within the scope of the appended claims.

What is claimed is:

1. A positive displacement rotary compressor comprising a casing having a barrel portion and inlet and discharge end walls, intermeshing rotors mounted in said casing and forming with said casing chambers closed at their ends by said walls, a delivery space, a main outlet port providing communication between said chambers and said space, a pressure control port in said inlet end wall with which the chambers communicate respectively prior to communicating with said outlet port, a passage connecting said pressure control port with said delivery space and valve means for controlling flow of fluid through said passage, said valve means being constructed and arranged to open toward said delivery space in response to differential pressure between the pressures in said chambers and in said delivery space.

2. A positive displacement rotary compressor comprising a casing having a barrel portion and inlet and discharge end walls, intermeshing rotors mounted in said casing and forming with said casing chambers closed at their ends by said walls, a delivery space, a main outlet port providing communication between said chambers and said space, a first pressure control port in said inlet end wall, a second pressure control port in said barrel portion, said control ports being located to communicate with the respective chambers at different times and in the order named, prior to communication between the chambers and said outlet port, passages connecting said control ports with said delivery space and valve means for controlling said passages, said valve means being constructed and arranged to open in response to differential pressure between the pressures in said chambers and in said delivery space.

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