

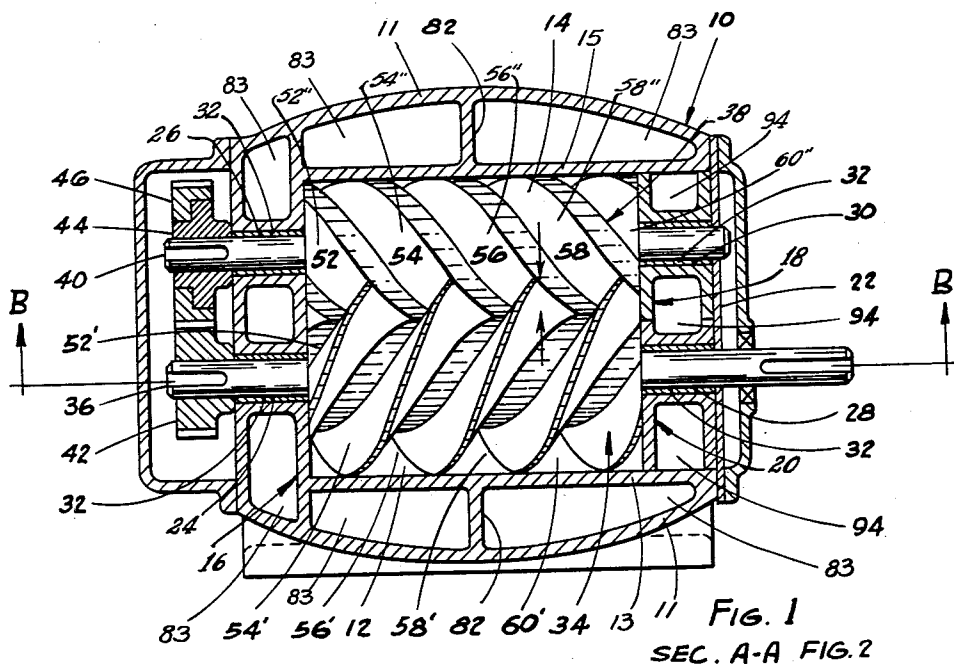
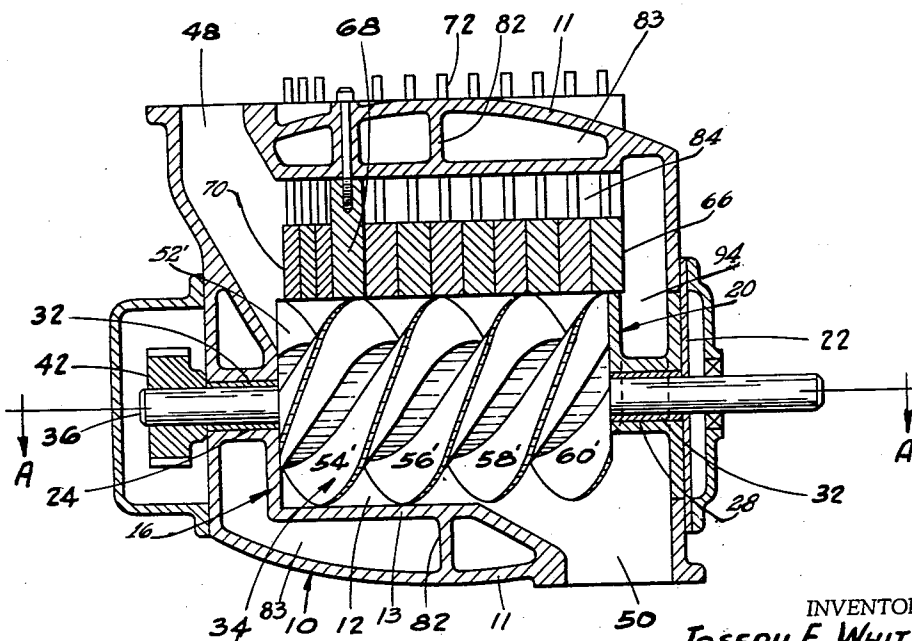
Oct. 6, 1964

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SCREW TYPE COMPRESSOR HAVING VARIABLE VOLUME
AND ADJUSTABLE COMPRESSION

3,151,806

Filed Sept. 24, 1962

5 Sheets-Sheet 1



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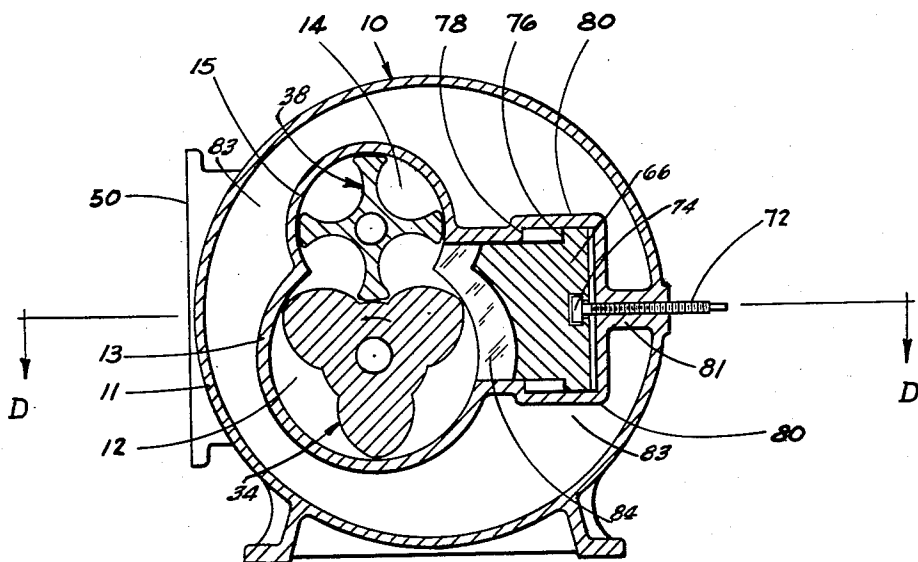


FIG. 3
SEC. C-C FIG. 4

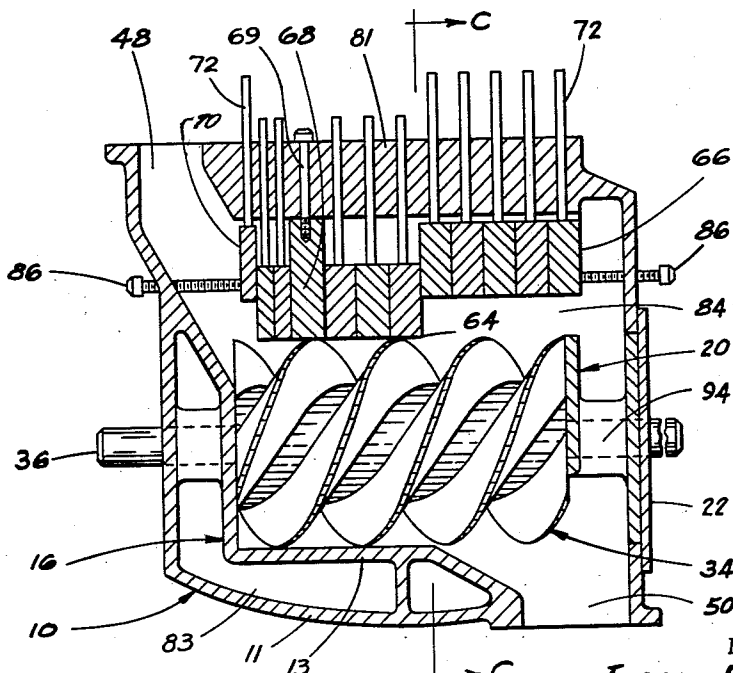


FIG. 4
SEC. D-D FIG. 3

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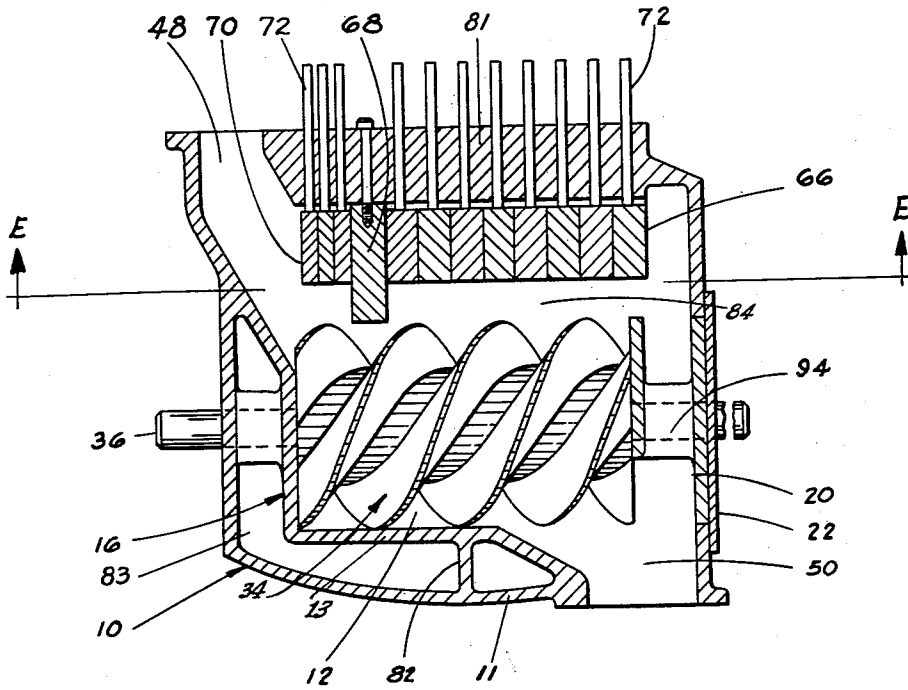


FIG. 5

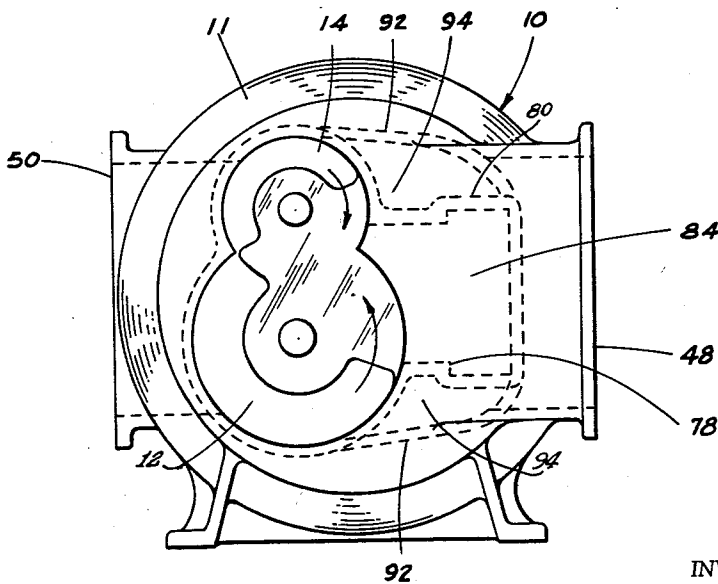


FIG. 6

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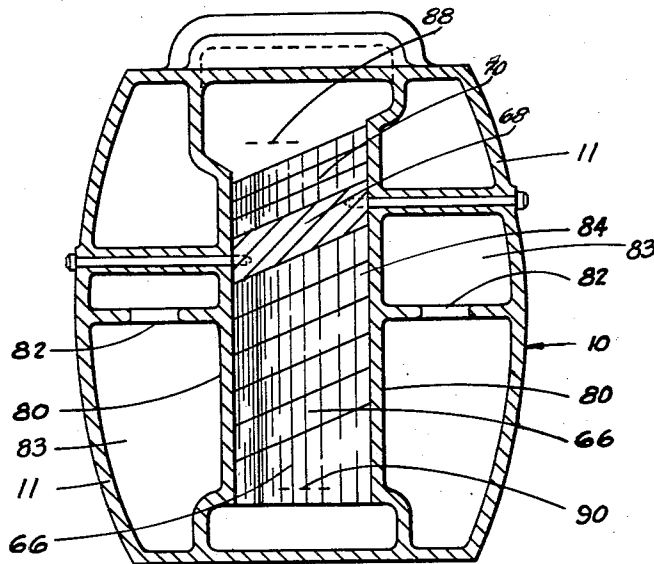


FIG. 7
SEC. E-E FIG. 5

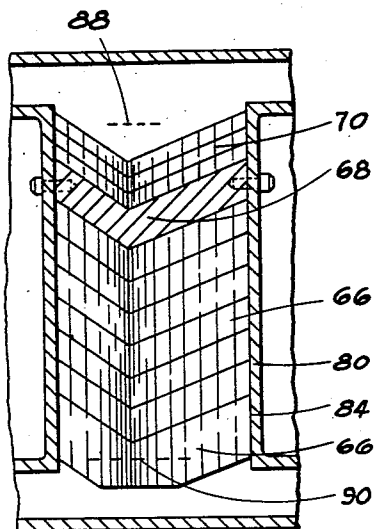


FIG. 9
SEC. F-F FIG. 8

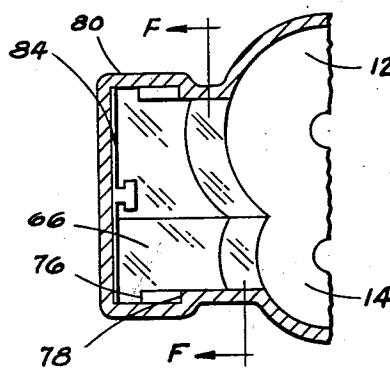


FIG. 8

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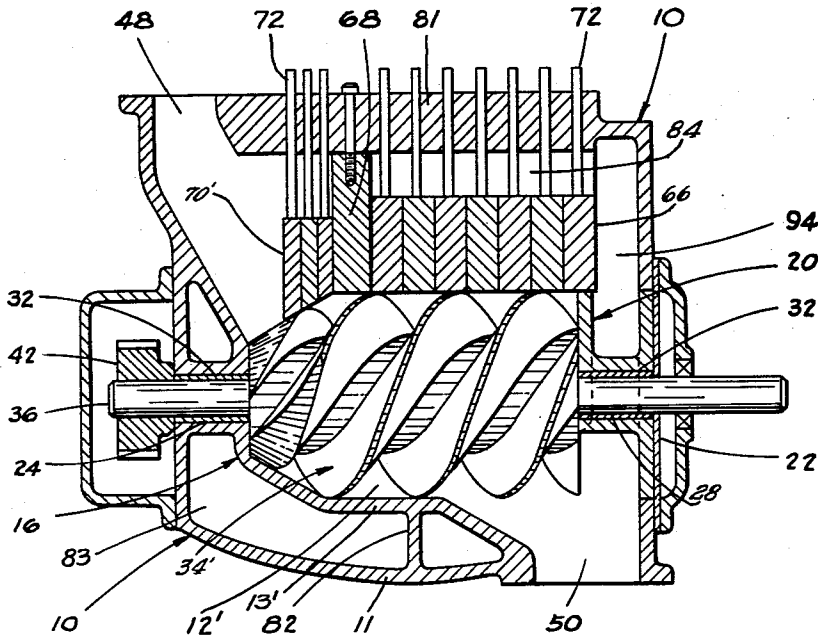


FIG. 10

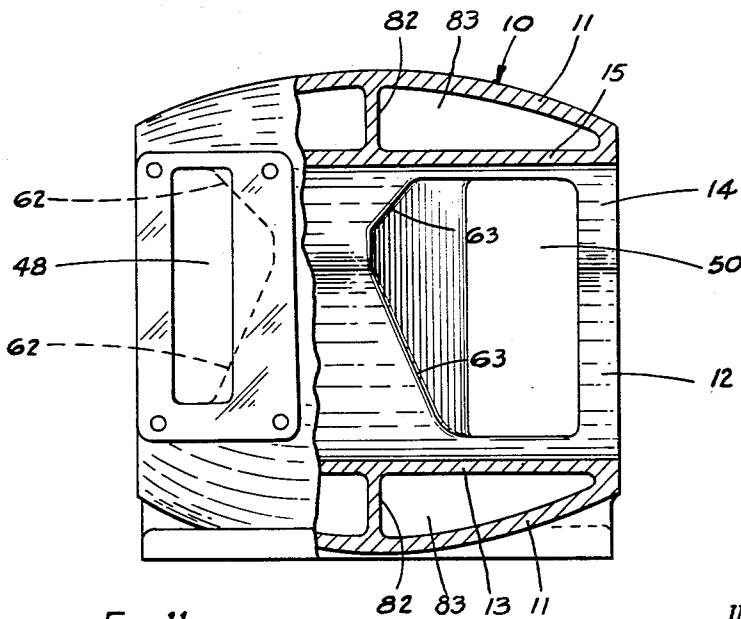


FIG. 11

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3,151,896

SCREW TYPE COMPRESSOR HAVING VARIABLE VOLUME AND ADJUSTABLE COMPRESSION

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Filed Sept. 24, 1962, Ser. No. 225,657

11 Claims. (Cl. 230-138)

This invention relates generally to fluid pumps, motors, compressors, blowers and similar devices in which interengaging rotary members are provided with intermeshing helical threads, and more particularly to a novel form of housing for such devices and to the incorporation therein of novel means for adjusting their volume and internal compression.

Screw type devices of this general nature have two or more helically threaded members rotatably supported with their axes substantially parallel and with their complementary threads intermeshing to form a working seal between the rotors. A housing encloses the rotary members and the chambers of said housing form a working seal with the perimetric tips of the rotor threads. Thus, in operation, the fluid being pumped from the suction end of the housing to the discharge end is confined in individual pockets, the fluid being separated into individual slugs. As the rotors rotate the pockets develop continuously at the suction end and fill with fluid as they increase in size because of the vacuum thus created. The pockets progress axially along the rotors and diminish progressively in size at the discharge end until they are reduced to zero or run out, thus expelling the fluid being pumped through the discharge port.

The threads must be complementary to form satisfactory seals and spaces and to permit rotation of the members. Thus the rotors are dissimilar in cross section and one has right-hand threads while the other has left-hand threads.

While more than two rotors can be used without departing from the spirit of this invention, only two will be shown and described herein for reasons of simplicity. The threads on one rotor lie wholly, or almost wholly, outside its pitch circle and, as this rotor absorbs almost all the input power, it is termed the main rotor. The threads on the mating rotor lie wholly, or almost wholly, within its pitch circle and, as this rotor forms a gate across the path of the main rotor threads, it is termed the gate rotor. If a gate rotor or some other device were not provided across the path of the main rotor threads, the fluid being pumped would simply revolve with the main rotor and no pumping effect would be produced.

The main rotor is sometimes termed the male rotor and the gate rotor is sometimes termed the female rotor. However, in this disclosure the terms "main" and "gate" rotors will be used because they are more accurately descriptive of the members to which they are applied.

As the rotors are rotated the threads on the main rotor, in effect, act as a continuous series of pistons which progress endwise through the troughs formed by the gate rotor threads, and form a continuous series of compression spaces, or pockets, which convey the fluid from the suction end to the discharge end of the housing in the form of separate, distinct slugs. The opening at the inlet end of the housing is called the suction port and the opening at the outlet end of the housing is termed the discharge port.

While this invention is not directed to any particular rotor form, it is appropriate to explain the action of screw type rotors in general as they apply to this invention. The rotors may be of any of the three well-known forms exemplified by the generated rotors shown in Whitfield Patents Nos. 2,486,770, 3,922,377 and 2,287,716, the asymmetrical rotors shown in Lysholm Patent No. 2,174,-

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522, and the arcuate rotors shown in Nilsson Patent No. 2,622,787. While these three general types of helical rotors differ slightly in cross section, their operating characteristics are practically the same in that, as the rotors revolve, they continuously form fluid pockets at the suction end and carry the fluid axially along the rotors to the discharge end. At the discharge end the pockets run out and the fluid is discharged.

When the rotors are in operation, the pockets formed by the rotor threads develop from zero size to a definite, fixed, maximum size and simultaneously are filled with fluid. After reaching maximum size and being filled with fluid, they are sealed by the coaction of the threads with one another and with the walls of the rotor chambers in the housing. The fluid is now trapped in the pockets and the entire amount is delivered to the discharge port by the rotation of the rotors. Since the capacity of the device is thus determined by the size of the pockets formed by the rotors and the speed of operation, it may be termed a fixed capacity or positive displacement device. Although there are several factors that affect capacity to a minor degree, such as discharge pressure, intake pressure, slip, temperature and overall efficiency, these factors will usually be ignored in this description for reasons of simplicity.

In service, however, it is often desirable to provide a variable delivery of fluid because of changeable demands. Variable delivery can be attained in one of two ways, or by a combination of two ways. First, the speed of operation can be subject to variation. This is usually expensive, inefficient and unreliable mechanically. The second, and best, method is to provide a device which is capable of delivering a variable volume of fluid at constant speed. Since these devices usually have a fixed built-in ratio of internal compression, this ratio changes as the volume changes. Consequently, in order to vary the volume and maintain the same pressure of internal compression, the ratio of compression must also be adjustable. To maintain high efficiency, the internal compression pressure inside a pocket just before the pocket registers with the discharge port should be substantially the same as the discharge pressure under all operating conditions. Should the internal compression be too high, the pressure in the pockets will be higher than the discharge pressure and there will be a drop in pressure when the pockets register with the discharge port. This excess in internal compression represents power lost through re-expansion, producing inefficiency and excess noise. Likewise, should the internal compression pressure be lower than the discharge pressure, when the pockets register with the discharge port there will be a slight reversal of fluid flow as the fluid flows backward into the pockets to balance the pressure. This inadequacy of internal compression also represents power lost through re-expansion, likewise producing inefficiency and excess noise.

Constant speed, variable volume blowers are numerous in the prior art, but there are certain basic objections to all those heretofore disclosed, including the following: they may reduce the volume only to a very small extent; they may not produce a power reduction in relation to the decrease in volume; they may be difficult to construct and control; they may not affect the internal compression; they may not affect the volume independently of the internal compression; they may create excess noise; they may affect the construction of the housing to such an extent that the housing will not maintain its proper shape when in service; or they may not provide for a variable volume control that can be adjusted while the device is in operation.

The present invention is intended to eliminate these inadequacies and to attain the following objectives:

The principal object of the invention is to provide a

device of the character described with a variable volume of delivery wherein the efficiency will remain substantially the same throughout its range of operation.

A second important object is to provide such a device with a variable volume control wherein the input power will decrease substantially in the same proportion as the delivery volume decreases.

Another object is to provide a device wherein the delivery volume can be regulated without changing the internal compression ratio.

Another object is to provide a device wherein the internal compression ratio can be regulated without changing the delivery volume.

Another object is to provide a device wherein the internal compression can be adjustably maintained while changing the delivery volume.

Another object is to provide a device with a variable volume control which has enough capacity to bypass the surplus fluid with a minimum of pressure rise.

Another object is to provide a device with a variable volume control wherein the surplus volume is bypassed before its pressure is increased, or before work is expended on it.

Another object is to provide a device with a variable volume control wherein the opening of the pockets to the discharge port, by the action of the rotors, may be adjustably regulated.

Another object is to provide a device with a novel delivery volume control and a housing of such design that it will normally maintain its proper shape under severe service conditions.

Another object is to provide a device of the character above described with a housing wherein the rotor chamber walls are maintained in proper shape by a barrel-shaped outside wall.

Other objects and advantages will appear upon consideration of the following description, reference being had to the several embodiments illustrated in the accompanying drawings in which:

FIG. 1 is a longitudinal sectional view of one form of screw type compressor embodying the present invention, the section being taken through the axes of the rotor members on the line A—A in FIG. 2, the rotors not being in section.

FIG. 2 is a longitudinal sectional view taken through the axis of the main rotor on the line B—B in FIG. 1 perpendicular to the plane of FIG. 1 and showing the suction and discharge ports of the compressor, the volume control valves being shown closed for maximum volume and the internal compression control valves being shown closed for maximum compression, the rotor not being shown in section.

FIG. 3 is a cross sectional view, perpendicular to the rotor axes, taken on the line C—C in FIG. 4.

FIG. 4 is a longitudinal sectional view taken on the line D—D in FIG. 3, the volume control valves being shown in a position to deliver substantially one-half of the maximum volume of the device and the internal compression control valves being positioned to produce a reduced discharge pressure.

FIG. 5 is a view similar to FIG. 4 except that all the valves are open, a condition in which the device would pump no fluid if operated.

FIG. 6 is an end view of the open end of the barrel-shaped housing intended to show the two integral end walls which close the valve chamber at both ends and maintain the accuracy of the housing bores, the discharge end of the housing being shown as the open end in this view only.

FIG. 7 is a longitudinal sectional view taken on the line E—E in FIG. 5 showing the angular disposition of the valve which conforms substantially with the helix angle of the main rotor, overlapping slightly on the wide lands of the gate rotor, only the fixed valve spacer being shown in section.

FIG. 8 is a fragmentary cross sectional view of an optional design.

FIG. 9 is a longitudinal sectional view taken on the line F—F in FIG. 8, only the fixed valve spacer being shown in section.

FIG. 10 is a view similar to FIG. 2 except that the internal compression control valves are shown as applied to a compressor having a rotor with a conical end.

FIG. 11 is a partially sectioned side view of the housing per se showing the suction and discharge ports, and indicating in broken lines the approximate position of the cut-off edge of the discharge port when all of the internal compression control valves are closed.

In FIGS. 1, 2, 4, 5, 10 and 11 the suction end is shown as being the open end of the housing, while in FIG. 6 the discharge end is shown as being the open end as explained later.

In order to make clear the meanings of certain terms used in the specification and claims, such as "spaces," "pockets," "run-out," "internal compression," "variable volume" and "variable internal compression," the following definitions are set forth:

Spaces.—This is the term applied to the cavities between the threads of each individual rotor, in the same manner as the grooves between gear teeth are called spaces.

Pockets.—Pockets are formed by the merging together of two complementary spaces as the rotors revolve. The two rotors and the rotor chamber walls coact to complete the sealed working pockets.

Run-out.—This is the term used in connection with the pockets as they reach the discharge end of the rotors and decrease in size to zero, i.e., vanish, or "run-out." At the suction end these pockets continue to fill with the fluid being pumped until they reach maximum size. They are then disconnected from the suction port by the rotation of the rotors and begin to decrease in size to compress the fluid therein. Compression continues until the edge of the discharge port is reached when discharge of the fluid takes place. After passing the edge of the discharge port the pockets are no longer sealed because of the absence of the chamber walls, but they continue to advance to the discharge end where they vanish or "run-out."

Internal compression.—Internal compression is the increase in pressure of the fluid which results from the reduction in volume of the pockets between the times of disconnection from the suction port and registry with the discharge port. The size of the suction port is always such that it is disconnected from the pockets when the pockets have reached maximum size. To make the suction port adjustable would be of no advantage. The amount of internal compression is determined by the location of the edge of the discharge port. When the pockets are disconnected from the suction port by rotation of the rotors they are of maximum size; further rotation of the rotors then starts to decrease the volume of the pockets. This decrease in volume is termed internal compression and continues until the pockets reach the edge of the discharge port. Thus the internal compression ratio can be controlled by shifting the position of the discharge port edge.

Variable volume.—Compressors of the general type shown are normally fixed volume devices. In other words, a certain pair of rotors will normally displace the same specific volume of fluid during each revolution, exactly the same as a piston operating in a cylinder. To reduce the displacement per revolution of a given set of rotors, it is necessary to render a certain portion of the length of the rotors ineffective so far as pumping is concerned, just as if the rotors were cut off to a length that would produce only the amount of displacement desired. This is comparable to reducing the length of the stroke of a piston in a cylinder.

Variable internal compression.—As stated above, the amount of internal compression is determined by the

location of the edge of the discharge port. With a fixed port edge and a given quantity of fluid the internal compression is fixed. To vary the internal compression an adjustable port edge is required.

Referring now to FIGS. 1, 2 and 3, wherein the invention is shown, by way of example, as applied to a screw type of pump or compressor, the housing 10 of the device contains two cylindrical chambers 12 and 14 disposed side by side in parallelism and merging into each other to form a common rotor chamber, the cross section of which is somewhat in the form of a figure 8. One end of the housing (the discharge end in the embodiment illustrated in FIGS. 1 and 2) is provided with an integral end wall 16 having spaced inner and outer portions, the inner portion of which forms one end wall of the chambers 12 and 14. The other end of the housing is closed by a removable end wall, also having inner and outer portions, which enables insertion of a pair of mating helical rotors 34 and 38 arranged to operate within the rotor chambers 12 and 14, respectively. Rotor 34 is termed the main rotor and has left-hand threads in the device shown. Rotor 38 is termed the gate rotor and has threads which are complementary to those of the main rotor, i.e., right-hand threads.

As above mentioned, this invention may be used with various types of screw type rotors, but for reasons of simplicity only the fully generated type will be shown. However, it is not intended that the invention be limited to use with generated rotors. It can also be used to vary the volume of liquid pumps when used without internal compression.

The removable end wall of the housing is made in two parts in the form of re-entrant heads which are self-centering in the rotor chambers. The re-entrant head 18 for the gate rotor chamber 14 is continuously circular on its outside diameter and fits into its chamber without clearance. The re-entrant head 20 for the main rotor chamber 12 is also circular on its outside diameter except it has a concave niche formed in one side to allow it to be assembled into the housing beside the gate rotor re-entrant head. The re-entrant heads 18 and 20 are held in proper relation to the housing by the head plate 22. The housing end wall 16 is provided with two cylindrical bearing bores 24 and 26 which are concentric with the main rotor chamber 12 and the gate rotor chamber 14, respectively. Likewise, the re-entrant heads 18 and 20 are provided with cylindrical bearing bores 28 and 30 respectively, which are centrally located in their respective re-entrant heads.

Each of the cylindrical bearing bores 24, 26, 28 and 30 is provided with a bearing bushing 32. The main rotor 34 is fixedly attached to a driving shaft 36 and is centrally located in the main rotor chamber 12 by the bushings 32 in bearing bores 24 and 28. The gate rotor 38 is likewise fixedly attached to a shaft 40 and is centrally located in the gate rotor chamber 14 by the bushings 32 in bores 26 and 30.

The main rotor shaft 36 has a timing gear 42 fixedly attached thereto while the gate rotor shaft 40 has a hub 44 fixedly attached thereto on which a gate rotor gear 46 is adjustably mounted. This provision for adjustment allows the rotor threads to be timed in spaced relationship at assembly.

There is a lateral discharge port 48 in the gear end of the housing and a lateral suction port 50, diagonally opposite port 48, in the drive end of the housing. As viewed in FIG. 1, with the rotors rotating as indicated by the arrows, the discharge port 48 would be at the gear end of the housing on the front side, while the suction port 50 would be diagonally opposite on the rear side at the drive end.

The rotor chambers 12 and 14 and the re-entrant heads 18 and 20 are slightly larger in diameter than the rotors 34 and 38 so that the rotors are centrally located in a positive manner and may operate with minimum clear-

ance. Ordinary bearing bushings 32 are shown for reasons of simplicity, but it will be understood that suitable bearings may be supplied to carry both the radial and the thrust loads which are developed in such a device.

To produce high efficiency, the rotors must operate with very close clearances between each other and with the housing chamber walls. Heretofore, when portions of the chamber walls were cut away to install variable control means, the strength and rigidity of the housing was greatly reduced, making it difficult to maintain close running clearances with the rotors. The design of housing 10 shown in the drawings is an important part of this invention in that the barrel-shaped outer wall 11 of the housing is so formed that it has great strength and permanent rigidity. The cylindrical inner walls 13 and 15 which form the rotor chambers 12 and 14, respectively, are supported by the outer wall 11 through suitable transversely extending ties or ribs 82, as shown best in FIG. 1. Furthermore, as indicated in the drawings, one end of the housing is completely closed except for the rotor shaft bearing bores 24 and 26, while the open end is open only enough to allow assembly of the rotors and re-entrant heads 18 and 20. Such a design, in conjunction with the barrel-shaped outer wall 11, provides a maximum of rigidity.

The open spaces 83 between the barrel-shaped outer wall and the cylindrical inner walls of the housing 10, and between the inner and outer portions of end wall 16, are used as passages for cooling fluid, such as water. Since the gear end of the housing is the discharge end in this instance, it is also the hot end. Provisions are thus made to cool the entire housing and the end wall 16 in such a thorough manner that expansion will be substantially equal throughout the unit, which is highly important in a device of this nature. While the re-entrant heads 18 and 20 are at the suction end and do not normally become heated by the discharge, it will be apparent that they can be easily cooled in the same manner as end wall 16 by a minor change in design. Under certain operating conditions when there is a major change in temperature between the suction and discharge ends of the compressor, it may be necessary to maintain a uniform temperature at all points by providing complete cooling of the housing and both end walls. Under severe conditions the rotors may also be liquid-cooled. However, since liquid-cooled rotors are old in the compressor art, such cooling is not shown in the drawings.

As before stated, screw type rotors of the character used in practicing the present invention have a fixed displacement, depending on size and design, and, when rotating in chambers having fixed walls, deliver a constant measured amount of fluid per revolution. In fact, they are often used to measure the flow of fluids, as, for example, when used as fluid meters.

The spaces between the threads on the gate rotor and the spaces between the threads on the main rotor combine, or merge, in pairs, to form common spaces which are termed pockets. For example, it can be seen in FIG. 1, wherein the front half of the housing 10 is cut away, that spaces 52' and 52'' merge together at the intersection of the rotors to form a common pocket 52; likewise spaces 54' and 54'', 56' and 56'', 58' and 58'', 60' and 60'', also merge together to form common pockets 54, 56, 58 and 60, respectively.

Referring to FIG. 1 and assuming the housing is complete rather than cut away as shown in the drawing, upon rotation of the rotors the spaces 60' and 60'' will merge together and form a common pocket 60 as compression begins. The fluid in the pocket is confined as soon as the threads forming the trailing edge of the pocket have passed the edge 63 (FIG. 11) of the suction port 50 on the opposite side of the housing. This pocket 60 will become smaller as it advances toward the discharge port until the leading edge of the pocket passes the cut-off edge

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of the discharge port, whereupon the fluid is discharged from the pocket into the discharge port 48.

In order to enable variation of either the volume of delivery or the internal compression ratio of the compressor, or both, the housing 10 is provided with a novel arrangement of independently adjustable valves which extend along one side of the common rotor chamber for substantially the full length of the rotors and are movable in directions perpendicular to the rotor axes.

As shown in FIGS. 2-5 and 7, portions of rotor chamber walls 13 and 15 adjacent their line of intersection are cut away along the side of the device at which the discharge port 48 is located, and said walls are extended laterally outwardly away from the rotors to form the walls 80 of a valve housing enclosing a chamber 84 wherein are mounted a plurality of adjustable volume control valves 66 and internal compression control valves 70 separated by a fixed valve spacer 68, which spacer prevents compressed fluid from leaking back to the suction end of the compressor. The walls 80 of valve chamber 84 may be formed integrally with the rotor chamber walls 13 and 15, and are held in rigid relationship with the outer wall 11 of housing 10 both by any suitable number of transverse ribs 82 and by a longitudinally extending rib 81. Valve spacer 68 is fixedly mounted in chamber 84 by screws or bolts 69 which pass through rib 81 and are threaded into the outer end of the spacer.

As indicated in FIG. 3, the inner edge of each of valves 66 and 70 and valve spacer 68 is curved to conform to the inner surfaces of those portions of chamber walls 13 and 15 which are cut away to form the valve chamber 84, while FIG. 7 shows that each of the valves and the valve spacer are set in the valve chamber at an angle substantially equal to the helix angle of the threads of the main rotor 34. When the valves and valve spacer are flat, as shown in FIG. 7, the curved edge portions which cooperate with the gate rotor 38 are of limited extent in order to prevent blow-by. Alternatively, the valves and valve spacer may be V-shaped, as illustrated in FIGS. 8 and 9, so as to substantially match the helix angles of both rotors and thereby effectively prevent blow-by. This alternate valve form provides larger valves and a larger valve chamber, but is more difficult to construct and is not normally necessary.

In the embodiment illustrated in FIGS. 2, 4 and 5, each of volume control valves 66, of which eight are shown, and the fixed valve spacer 68 has a width or thickness in a direction parallel to the rotor axes which is slightly more than one-third of the width of the space between the tips of two adjacent threads of the main rotor 34. However, each compression control valve 70, of which three are shown, is only about half as thick as the volume control valves and the valve spacer. As indicated in FIG. 7, wherein the broken lines 88 and 90 represent the locations of the discharge ends and the suction ends, respectively, of the rotors relative to the valves, the volume control valves 66 extend over approximately 75% of the length of the rotors measured from their suction ends, while the compression control valves 70 cover only about 15% of the length of the rotors adjacent their discharge ends.

For the purpose of selectively adjusting the valves 66 and 70 to provide the desired variations in delivery volume and internal compression ratio, each valve is provided with an operating stem in the form of a screw 72 which passes through rib 81 in threaded engagement therewith as shown in FIG. 3, the outer end of the screw projecting outwardly from housing 10 and being adapted to receive a suitable tool by which it may be turned. The inner end of each screw 72 is provided with an enlarged head 74 which fits loosely in a complementary T-shaped slot in the outer end of the valve. Each valve is also provided with a pair of shoulders 76 adapted to abut against complementary shoulders 78 in the valve chamber walls 80 and to serve as stops which, when the valve

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is in its innermost or closed position, accurately position the curved inner edge portions of the valve with respect to the inner surfaces of the rotor chamber walls 13 and 15 and the tips of the threads of rotors 34 and 38. Since the walls 80 of valve chamber 84 and rotor chamber walls 13 and 15 are rigidly fixed to the outer wall 11 of housing 10 by the ribs 81 and 82, the barrel shape of the outer wall holds the rotor chamber walls to their precise size even though parts of these chamber walls are cut away to form the valve chamber 84. As indicated in FIG. 7, both ends of the valve chamber are closed by portions of the end walls of housing 10, cutter reliefs being provided at the ends of the chamber for machining purposes.

When the valves are closed, it is necessary to hold them firmly in their closed position without distorting the housing in any manner. The force which holds the valves closed is applied by the heads 74 of screws 72 against the bottoms of the T-shaped slots in the valves, this force being transmitted through the shoulders 76 on the valves to the shoulders 78 in the valve chamber walls 80. The valve closing force and resistance are thus applied closely together so that distortion of the rotor chambers will not occur as a result of the valve closing pressure applied by screws 72. In order to eliminate chatter of the valves when in open position, screws 86 may be provided which, as shown in FIG. 4, are threaded through the end walls of housing 10 and are adapted to engage the outer side surfaces of those valves 66 and 70 which are located at the ends of the valve chamber 84.

In some instances it may be necessary to liquid-cool the valves 66 and 70 and valve spacer 68. This can easily be accomplished by making the valves and their operating stems hollow. However, it is not necessary for any of the valves to fit neatly in the valve chamber 84 except the valve spacer 68, and since the latter is fixedly attached it can easily be cooled. The valves 66 and 70 may have sufficient side clearance with the valve chamber walls 80 to provide for expansion. If the valve shoulders 76 and the chamber wall shoulders 78 are properly located, there will be no change in clearance between the rotors and the inner edges of the valves due to expansion, thus eliminating any need for cooling of the valves. The valve stems 72 need not be arranged in a straight line as shown, but may be staggered to provide more space for larger stems, and to avoid weakening of the housing in this critical area.

If, in effect, parts of the rotor chamber walls are removed by lifting the first five volume control valves 66 from close proximity to the rotors as shown in FIG. 4, compression is delayed until the pockets become sealed at the point 64 by the valves 66 which remain in closed position. The fluid in the pockets not sealed may then return freely to the suction side of the compressor through the portion of valve chamber 84 which was previously occupied by the lifted valves and through by-pass passages 94. Thus, that part of the rotors not sealed by the valves 66 would be rendered ineffective for pumping purposes and the delivery volume would be affected accordingly. The valves 66 serve a double purpose when pulled away from the rotors; they are only break the seal of the rotor pockets and allow fluid to escape therefrom, but also provide a passage in valve chamber 84 through which the escaping fluid may return to the suction side of the compressor. In order to be efficient and practical the chamber 84 must be large enough to allow the fluid to escape from the pockets with a minimum of pressure increase, because any pressure required to create flow through the chamber is power lost and cannot be recovered. The input driving power will therefore decrease almost as fast as the volume of fluid delivered decreases; in other words, the overall efficiency should be very nearly as good at partial volume as at full volume delivery. Further, when operating at full

capacity the efficiency will be as good as an equivalent device without volume control.

Either the suction end or the discharge end of the housing 10 may be the open end to receive the rotors and re-entrant heads 18 and 20. In FIGS. 1, 2, 4, 5, 10 and 11, the suction end is shown as the open end. However, in order to illustrate the by-pass walls 92 and by-pass passages 94 more clearly, FIG. 6 shows the discharge end of the housing as being the open end. As will be evident from the drawings, any excess fluid being pumped and not retained in the pockets by the valves 66 may be by-passed through passages 94 to the suction side of the rotors. Actually, under most operating conditions, very little fluid may return to the suction side due to any of the valves 66 being open. The fluid in the pockets not confined by the walls of the rotor chambers 12 and 14 because of open valves 66 may simply by-pass to the trailing pockets, so that there would be a reduced flow into the suction port 50 from outside but very little back-flow to it from the pockets.

When the trailing edge of a pocket has advanced to position 64 (FIG. 4), the pocket has been reduced in length due to the rotation of the rotors in conjunction with the end wall 16, and its volume has been reduced accordingly. Had all of valves 66 been closed, some internal compression would then have occurred. However, since the first five valves 66 are shown as being open, instead of internal compression having taken place, the fluid has escaped into the valve chamber 84 where it is picked up by the trailing pockets. Thus, only enough fluid is retained in the pockets to fill that part of their length between point 64 and the discharge end of the rotors. The delivery volume is therefore reduced because only a part of the rotor length is effective in pumping fluid.

This variable volume feature of the rotary compressor of the present invention can be compared with a variable volume piston type compressor. For example, if a piston compressor were provided with an adjustably controlled, mechanically operated suction valve, the volume of delivery could be controlled by changing the time of closing of the suction valve. If full volume were desired, the suction valve would close at the end of the suction stroke when the cylinder is full, and a maximum volume would be delivered. For partial volume, the suction valve would remain open until the piston has returned part way of the compression stroke, the excess air being exhausted through the suction valve until it is closed, after which compression would begin and a reduced amount of air would be delivered through the discharge valve. Changing the time of closing of the suction valve would thus change the delivery volume. Of course, the delivery volume could also be changed by "blowing off" part of the discharge air, but such a procedure would waste the work expended in compressing blown-off air. To be efficient, the volume must be reduced to the desired amount before compression begins.

If the housing 10 were to be provided with a discharge port 48 of a fixed size and shape, as is customary in the prior art, the device would have "built-in" or non-adjustable internal compression ratio determined by the position of the cut-off edge of the port relative to the discharge ends of the rotors. However, due to the provision of compression control valves 70, the effective edge of the discharge port may be adjusted, and the internal compression ratio thereby changed independently of the delivery volume, by moving valves 70 toward or away from the rotors.

For example, with all of compression control valves 70 in closed position, as shown in FIG. 2, the effective cut-off edge of the discharge port occupies a position indicated approximately by line 62 in FIG. 11, and the maximum internal compression will be produced. Furthermore, since all of volume control valves 66 are also closed, the maximum volume of fluid will be delivered to the discharge port. If, however, the first five volume control valves 66 and the last compression control valve 70 are opened,

as shown in FIG. 4, the compressor will deliver only about half of its maximum capacity and the internal compression will be less than the maximum due to a shifting to the right of that portion of the cut-off edge of the discharge port formed by the valve 70. On the other hand, if all of valves 66 and 70 are moved away from the rotors to the open positions indicated in FIG. 5, no fluid will be delivered to the discharge port 48 against pressure because, in order to pump fluid, a sufficient number of valves must be closed to seal at least one of the spaces between two adjacent threads of the rotors. In this connection, it will be seen from FIGS. 2 and 4 that the combined width of valve spacer 68 and the two adjacent volume control valves 66 is sufficient to seal one such space. Consequently, if all of valves 66 were to be opened except the two immediately adjacent to valve spacer 68, and if all of valves 70 were open as shown in FIG. 5, the device would pump fluid at its minimum capacity and the internal compression would be minimal, i.e., that established by the position of the cut-off edge of the discharge port formed by fixed valve spacer 68. In the disclosed embodiment of the invention, valve spacer 68 is so located that, with all of valves 70 open, the fluid is discharged from the pockets with the minimum desired increase in pressure.

Referring now to FIG. 10, the compressor illustrated therein is of basically the same construction and mode of operation as that of FIGS. 1-7 and 11 except that the discharge ends of the rotors and the cooperating portions of the rotor chamber walls are conical in form, as disclosed in Whitfield Patent No. 2,922,377. In view of this modification in shape of the discharge ends of the rotors, the compression control valves 70' are similarly modified insofar as their lengths and the shapes of their inner edges are concerned. As indicated in FIG. 10, wherein only the main rotor 34' is shown, the inner edges of valve 70' are beveled at the same angles as the conical end portions of the rotor so that, when the valves are in closed position, their inner edges are parallel to the tips of the rotor threads and constitute continuations of the inner surfaces of the rotor chamber walls.

While the invention has been disclosed as applied to a compressor, it will be obvious that it may be embodied in other types of fluid handling devices and used with rotor forms other than those specifically illustrated. It will also be understood that the invention is not limited to the particular structures described and shown in the accompanying drawings, but that various changes, which will now suggest themselves to those skilled in the art, may be made in the form, details of construction and arrangement of the parts without departing from the inventive concept. Reference is therefore to be had to the appended claims for a definition of the limits of the invention.

What is claimed is:

1. In a fluid pump device of the type having

(a) a housing provided with intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing, and

(b) a plurality of complementary helically threaded rotors rotatably supported within said rotor chambers and cooperating with each other and the chamber walls to form pockets at the suction end of the rotors for receiving fluid through said suction port, to advance said pockets and the fluid therein axially along the rotors and to discharge the fluid from said pockets through said discharge port as said rotors revolve about the axes thereof,

the improvement which comprises:

(1) a substantially barrel-shaped outer wall for said housing surrounding and spaced from the walls of said rotor chambers, said outer wall being curved both circumferentially and longitudinally and having its maximum diameter at a point substantially midway between the ends of said housing,

- (2) radially extending strengthening members rigidly connecting said rotor chamber walls to said barrel-shaped outer wall,
 - (3) portions of said rotor chamber walls being cut away and extended outwardly away from the rotor axes to form the axially extending side walls of a valve chamber housed within said barrel-shaped outer wall and in communication with said rotor chambers,
 - (4) radially extending walls formed integrally with said barrel-shaped outer wall and closing the end of said valve chamber, and
 - (5) valve means in said valve chamber movable radially toward and away from the rotors for controlling the volume and internal compression of the fluid discharged from said pockets through said discharge port.
2. The improvement defined in claim 1 which is further characterized by
- (1) a rib member extending substantially parallel to the rotor axes and rigidly connecting said valve chamber walls to said barrel-shaped outer wall.
3. In a fluid pumping device of the type having
- (a) a housing provided with intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing, and
 - (b) a plurality of complementary helically threaded rotors rotatably supported within said rotor chambers and cooperating with each other and the chamber walls to form pockets at the suction end of the rotors for receiving fluid through said suction port, to advance said pockets and the fluid therein axially along the rotors and to discharge the fluid from said pockets through said discharge port as said rotors revolve about the axes thereof,
- the improvement which comprises:
- (1) a valve chamber having walls integral with the walls of said rotor chambers and extending outwardly away from the rotor axes, said valve chamber being in communication with and extending axially along a substantial portion of the length of said rotor chambers,
 - (2) a plurality of valve members mounted in said valve chamber intermediate the ends of the rotors adjacent the suction end thereof for radial movement toward and away from said rotors,
 - (3) each of said valve members having a radially inner edge which in the closed position of the valve member constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads, and
 - (4) means for moving said valve members in radial directions away from closed position to provide space within said valve chamber between the rotor threads and the radially inner edges of said valve members through which fluid may pass from said pockets to the suction end of said rotors.
4. In a fluid pumping device of the type having
- (a) a housing provided with intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing, and
 - (b) a plurality of complementary helically threaded rotors rotatably supported within said rotor chambers and cooperating with each other and the chamber walls to form pockets at the suction end of the rotors for receiving fluid through said suction port, to advance said pockets and the fluid therein axially along the rotors and to discharge the fluid from said pockets through said discharge port as said rotors revolve about the axes thereof,
- the improvement which comprises:
- (1) a valve chamber having walls integral with the walls of said rotor chambers and extending outwardly

- away from the rotor axes, said valve chamber being in communication with and extending axially along a substantial portion of the length of said rotor chambers,
- (2) a plurality of valve members mounted in said valve chamber intermediate the ends of the rotors adjacent the discharge port for radial movement toward and away from said rotors,
 - (3) each of said valve members having a radially inner edge which in the closed position of the valve member constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads so as to prevent the passage of fluid from said pockets into the discharge port, that valve member closest to the discharge end of the rotors which is in closed position forming the effective cut-off edge of said discharge port, and
 - (4) means for moving said valve members in radial directions away from closed position to enable the passage of fluid from said pockets into the discharge port and thereby vary the position of the effective cut-off edge of said discharge port.
5. A fluid compressor comprising:
- (1) a housing having intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing,
 - (2) a pair of complementary helically threaded rotors rotatably supported within the rotor chambers of said housing and each having a plurality of threads and troughs,
 - (3) the threads and troughs of said rotors intermeshing and cooperating with one another and with the chamber walls to form pockets communicating at different times with said suction port and said discharge port, which pockets are advanced axially along the rotors to effect compression of the fluid therein as the rotors revolve about the axes thereof, compression of the fluid in each pocket commencing when said pocket moves out of communication with said suction port, and
 - (4) means movable relative to said housing perpendicularly to the axes of said rotors for varying the volume of said pockets at the commencement of compression therein, said means including
 - (5) a plurality of individually operable valves mounted in said housing for radial movement relative to the rotor axes and extending along a substantial portion of the length of the rotors from the suction end thereof toward the discharge end thereof, each of said valves having a radially inner surface which in closed position of the valve constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads, and
 - (6) means for selectively moving said valves in radial directions away from closed position to provide space within the rotor chambers between the rotor threads and said radially inner surfaces of the valves through which fluid may escape from said pockets to the suction end of said rotors.
6. A fluid compressor comprising:
- (1) a housing having intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing,
 - (2) a pair of complementary helically threaded rotors rotatably supported within the rotor chambers of said housing and each having a plurality of threads and troughs,
 - (3) the threads and troughs of said rotors intermeshing and cooperating with one another and with the chamber walls to form pockets communicating at different times with said suction port and said discharge port, which pockets are advanced axially along the rotors to effect compression of the fluid

therein as the rotors revolve about the axes thereof, compression of the fluid in each pocket terminating when said pocket comes into communication with said discharge port, and

- (4) means movable relative to said housing perpen- 5
dicularly to the axes of said rotors for varying the axial position at which said pockets come into communication with said discharge port, said means including
- (5) a plurality of individually operable valves mounted 10
in said housing for radial movement relative to the rotor axes intermediate the ends of said rotors adjacent the discharge port for controlling communication between said pockets and said discharge port, each of said valves having a radially inner surface 15
which in closed position of the valve constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads so as to prevent the passage of fluid from said pockets into the discharge port, that valve closest to the discharge end of the rotors which is in closed position 20
forming the effective cut-off edge of the discharge port, and
- (6) means for selectively moving said valves in radial 25
directions away from closed position to enable the passage of fluid from said pockets into said discharge port and thereby vary the degree of internal compression of the fluid in said pockets independently of the volume of fluid delivered to said discharge port.
7. A fluid compressor comprising: 30
- (1) a housing having intersecting cylindrical rotor chambers and suction and discharge ports arranged diagonally opposite each other in the rotor chamber walls near the ends of said housing,
- (2) a pair of complementary helically threaded rotors 35
consisting of a main rotor and a gate rotor each having a plurality of threads and troughs,
- (3) said rotors being rotatably supported within the rotor chambers of said housing and cooperating with each other and the chamber walls to form pockets 40
at the suction end of the rotors for receiving fluid through said suction port, to advance said pockets and the fluid therein axially along the rotors and to discharge the fluid from the pockets through the discharge port as the rotors revolve about the axes 45
thereof,
- (4) a valve chamber having walls integral with said housing extending substantially the full length of said rotors in communication with said rotor chambers, 50
- (5) a first group of valves in said valve chamber intermediate the ends of the rotors and cooperating with the suction end thereof for controlling the volume of the fluid delivered to said discharge port,
- (6) a second group of valves in said valve chamber 55
intermediate the ends of the rotors and cooperating

with the discharge end thereof for controlling the internal compression of the fluid delivered to said discharge port,

- (7) each of said valves being individually movable in a radial direction relative to the rotor axes toward and away from said rotors and having a radially inner edge which, when the valve is in its closed position, constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads so as to prevent the escape of fluid from said pockets,
- (8) means for selectively moving said valves in radial directions away from closed position to provide space within said valve chamber through which fluid may escape from said pockets, and
- (9) a valve spacer fixedly mounted in said valve chamber between said groups of valves, said valve spacer also having a radially inner edge which constitutes substantially a continuation of the inner surface of the rotor chamber walls and seals with the rotor threads.
8. A fluid compressor as defined in claim 7 wherein
- (1) the width of the radially inner edge of each of said valves in a direction parallel to the rotor axes is less than the axial width of the space between the tips of two adjacent threads of the main rotor.
9. A fluid compressor as claimed in claim 7 wherein
- (1) the width of the radially inner edge of the fixed valve spacer in a direction parallel to the rotor axes is less than the axial width of the space between the tips of two adjacent threads of the main rotor so that, when all of said valves are moved to open position, rotation of the rotors does not deliver any fluid to the discharge port against pressure.
10. A fluid compressor as defined in claim 7 wherein
- (1) the radially inner edges of said valves and said valve spacer are substantially parallel to the tips of the threads of at least the main rotor.
11. A fluid compressor as defined in claim 7 wherein
- (1) the radially inner edge of each of said valves and said valve spacer is substantially V-shaped so that side edge is substantially parallel to the tips of the rotor threads.

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