Oct. 14, 1969 A. E. BROWN 3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 1

FIG. I

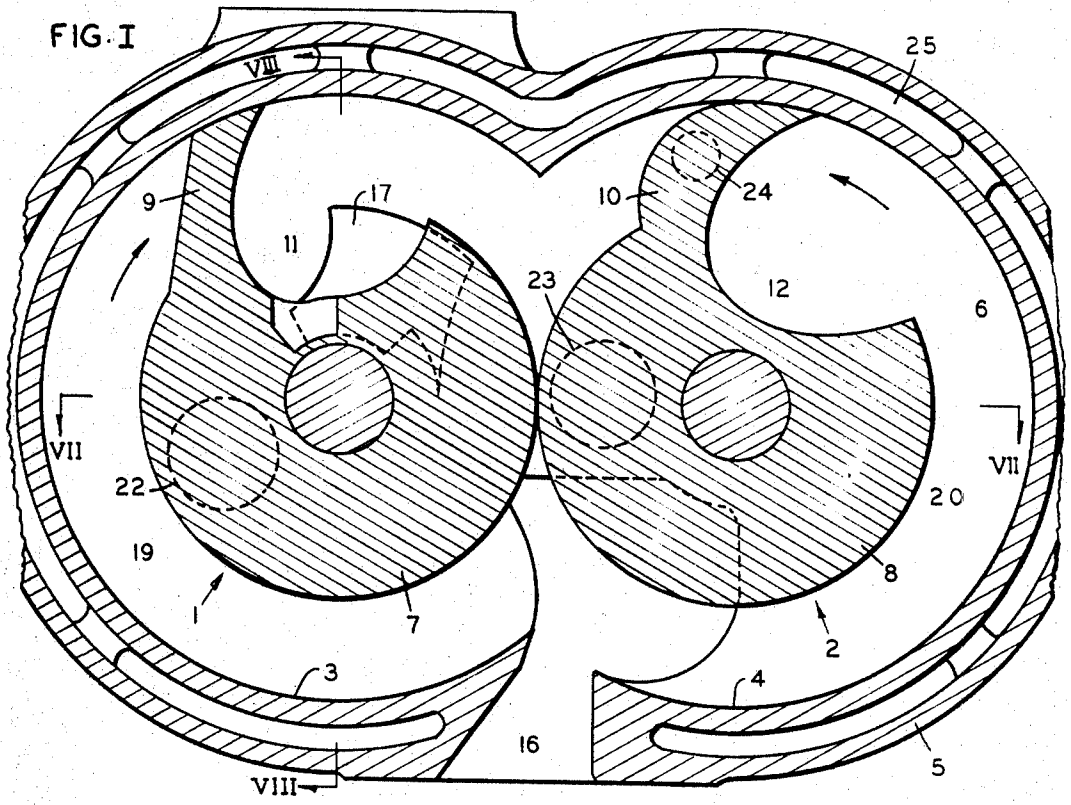
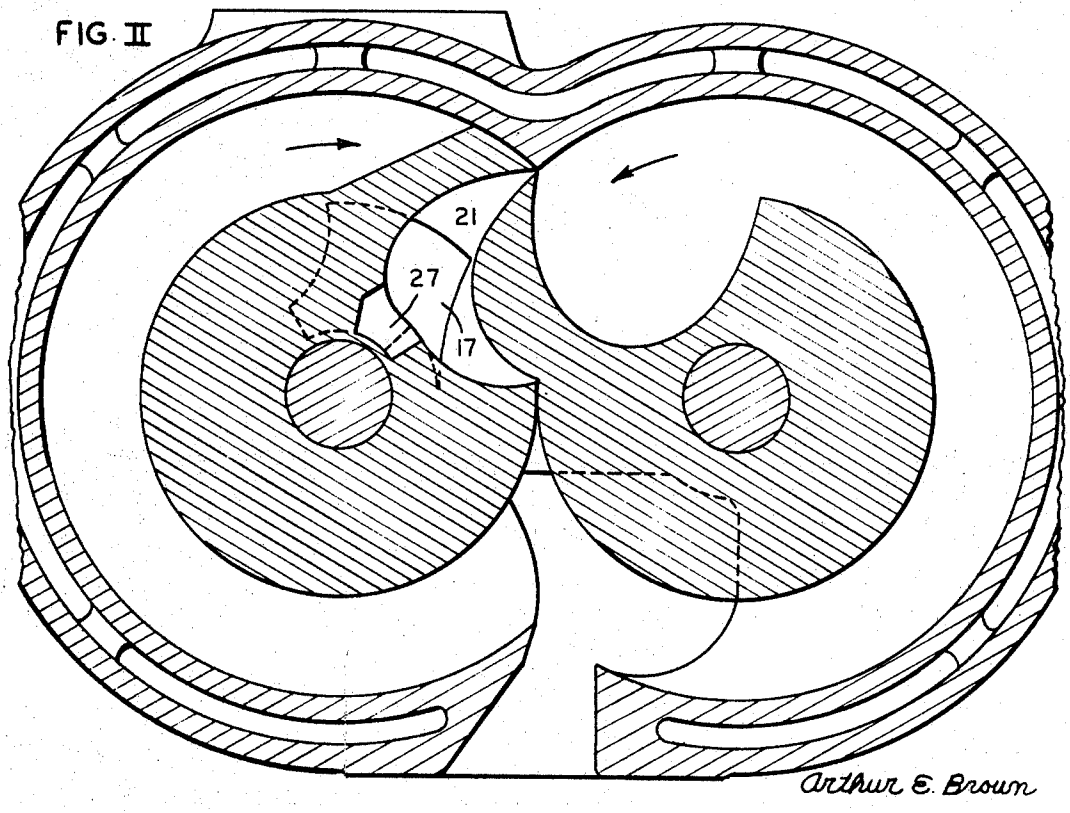


FIG. II



Arthur E. Brown

Oct. 14, 1969 A. E. BROWN 3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 3

FIG. III

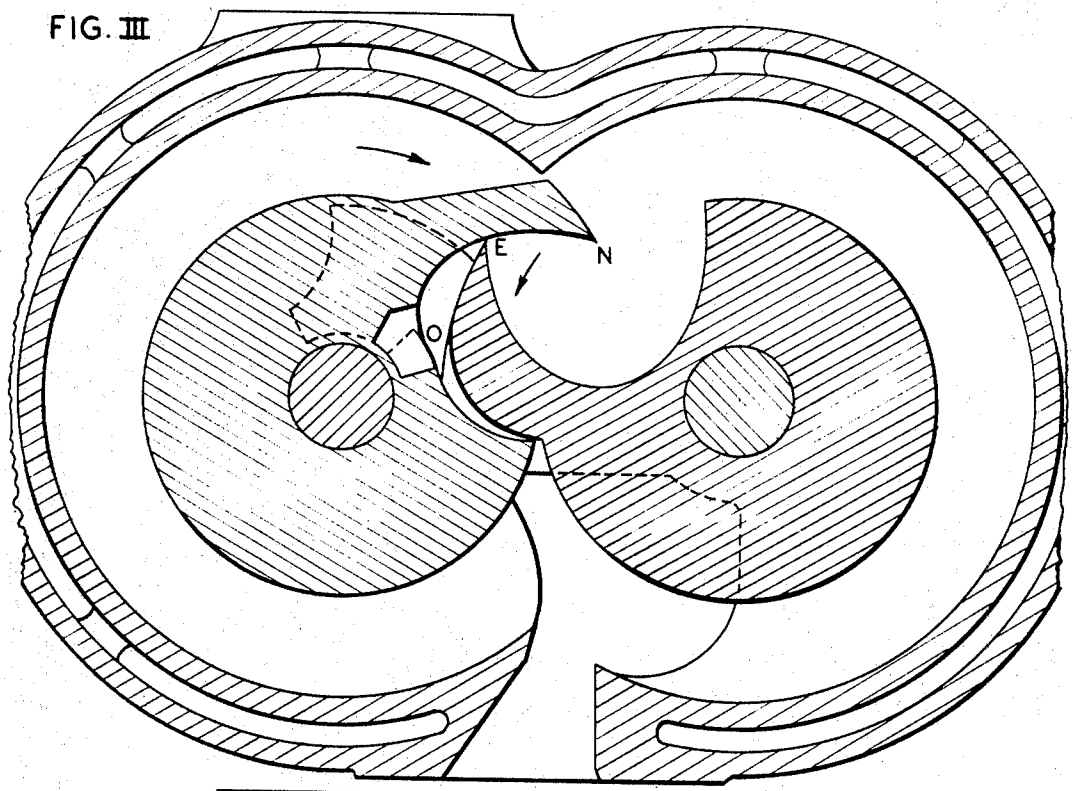
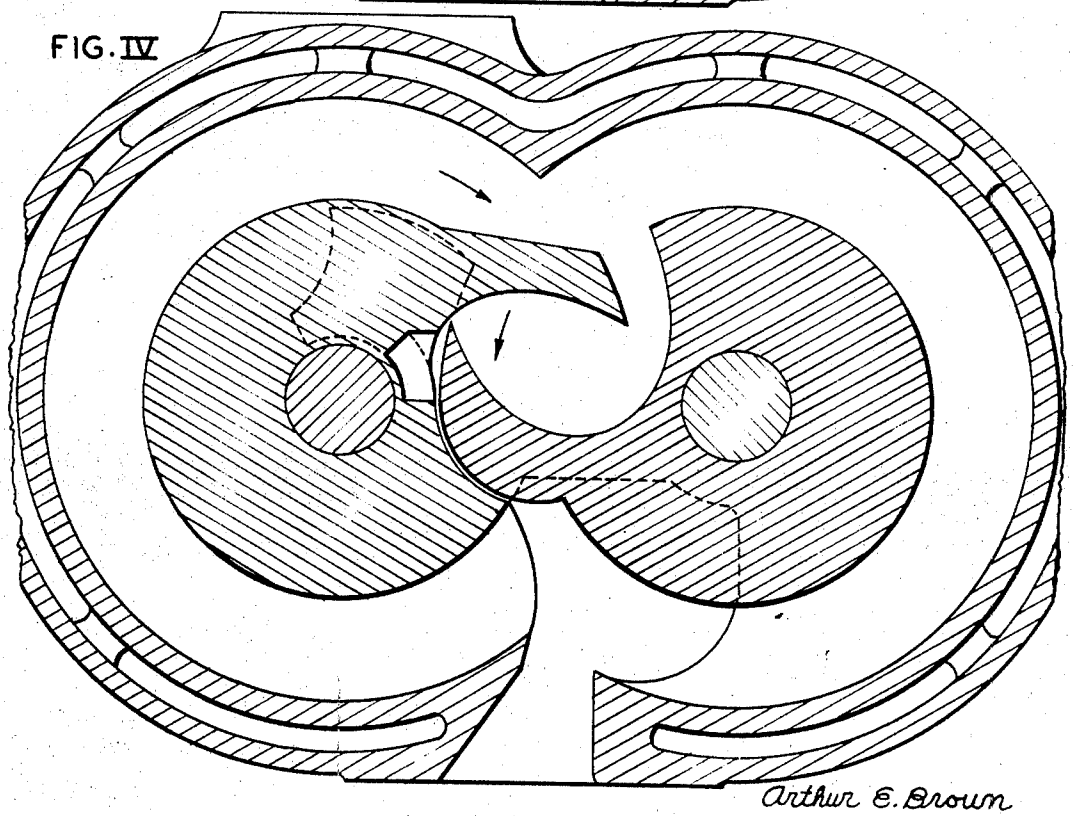


FIG. IV



Arthur E. Brown

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 3

FIG. V

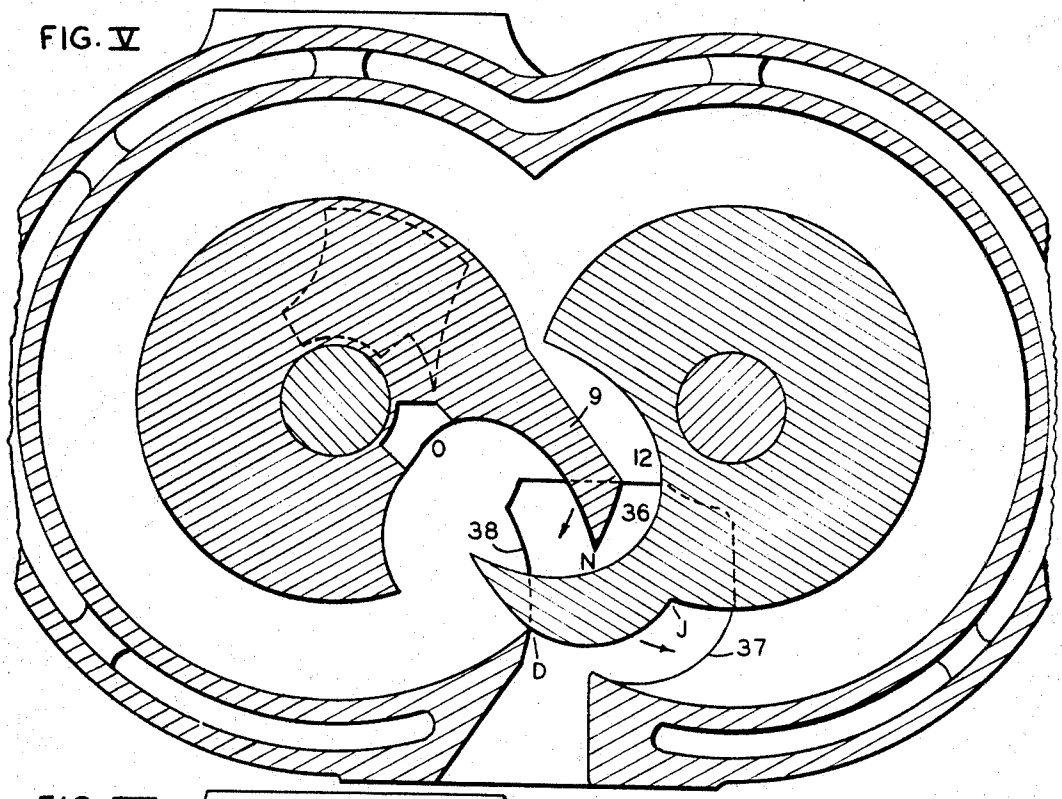
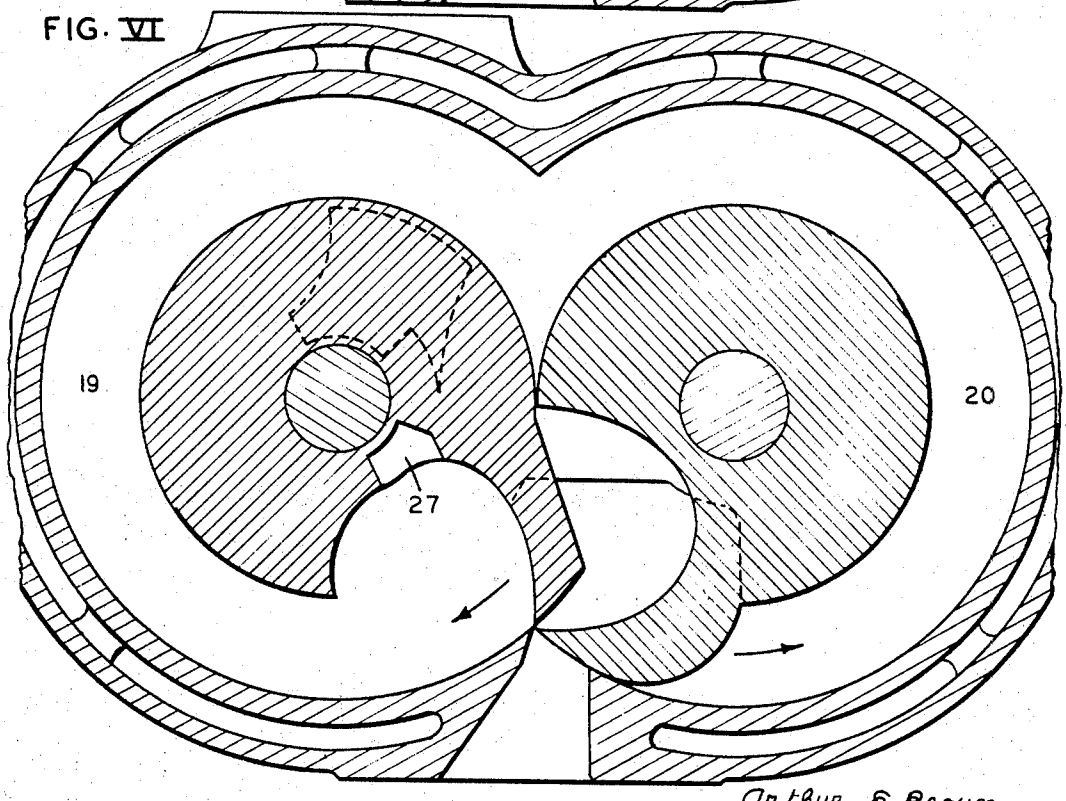


FIG. VI



Arthur E. Brown

Oct. 14, 1969 A. E. BROWN 3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 4

FIG. VIII

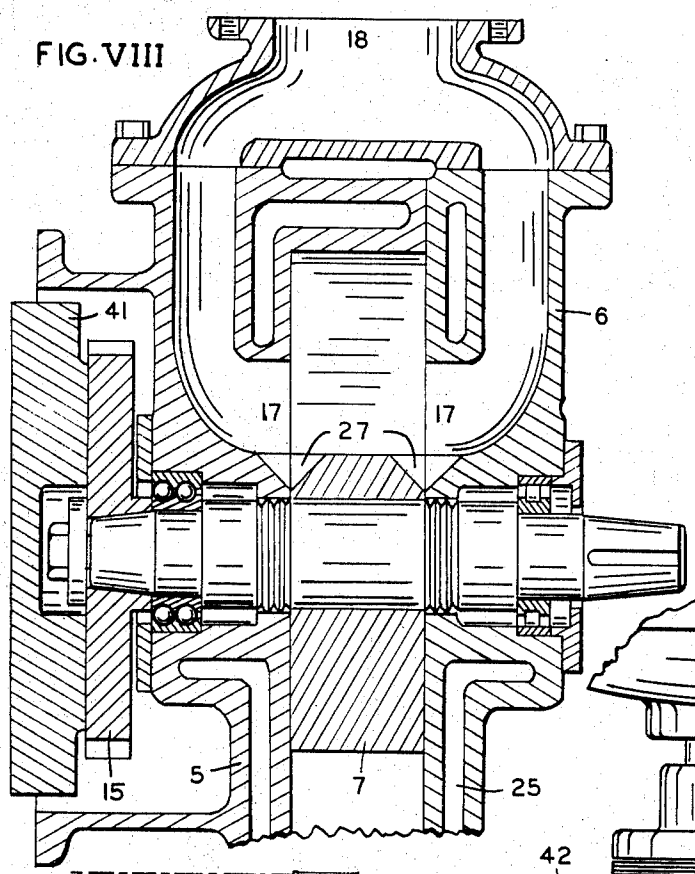
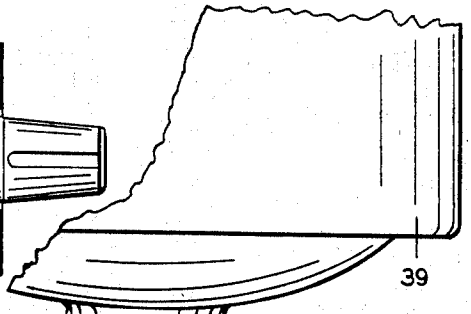
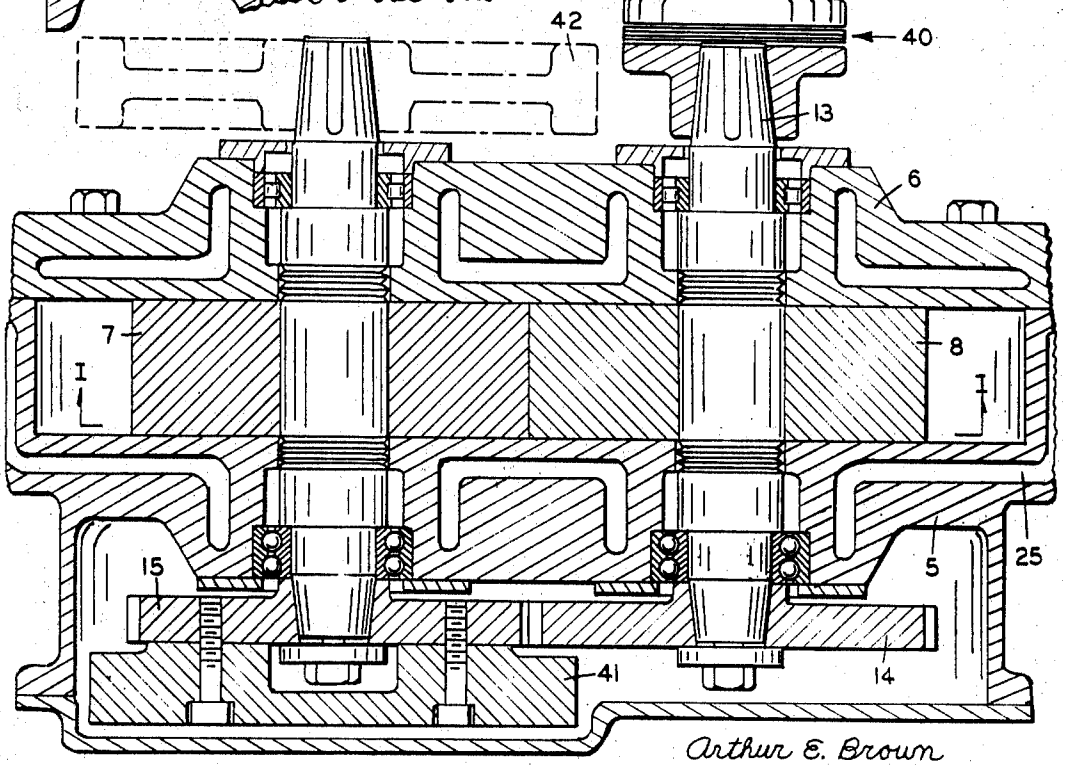


FIG. VII





Arthur E. Brown

Oct. 14, 1969 A. E. BROWN 3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 5

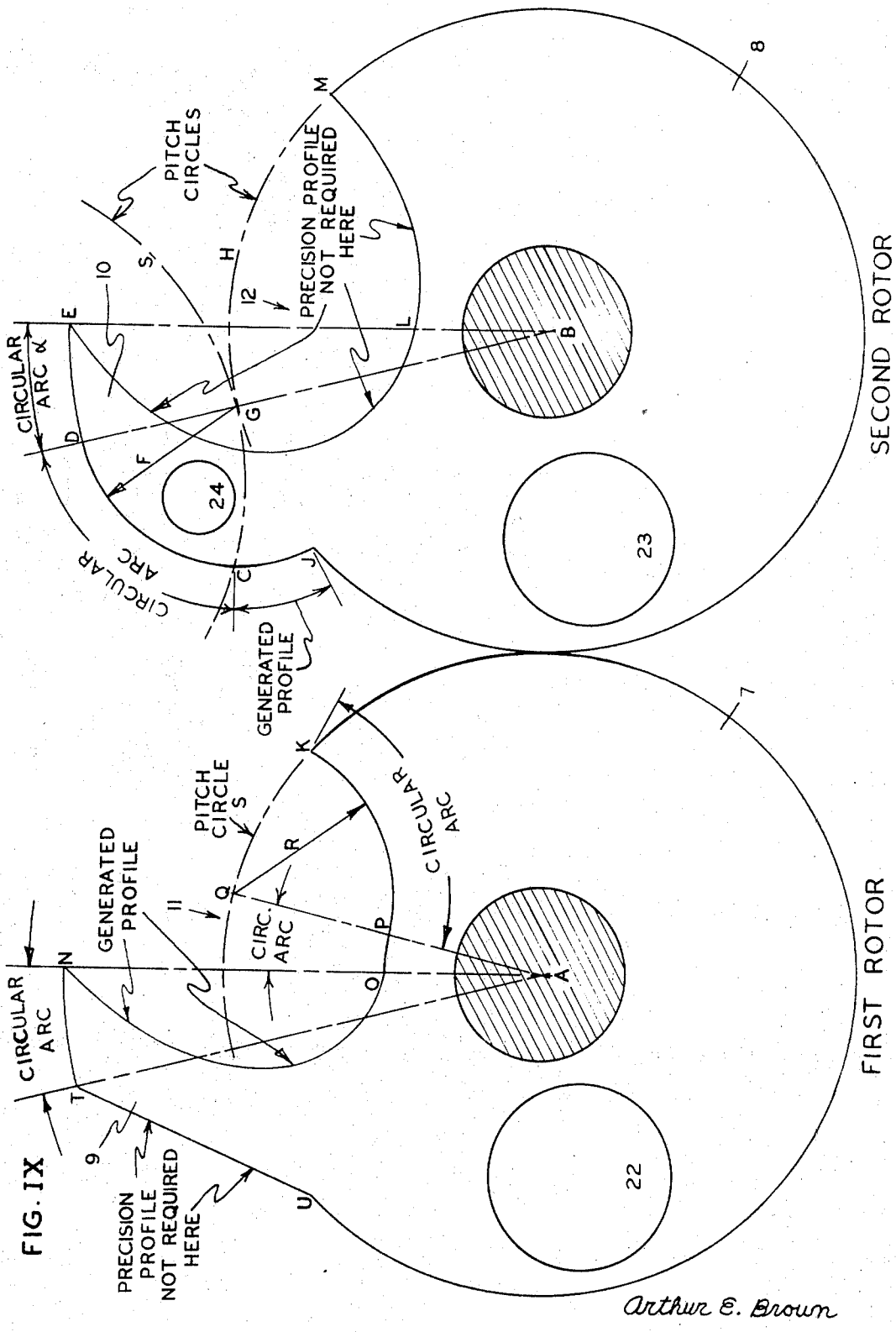


FIG. IX

Arthur E. Brown

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 6

FIG. X

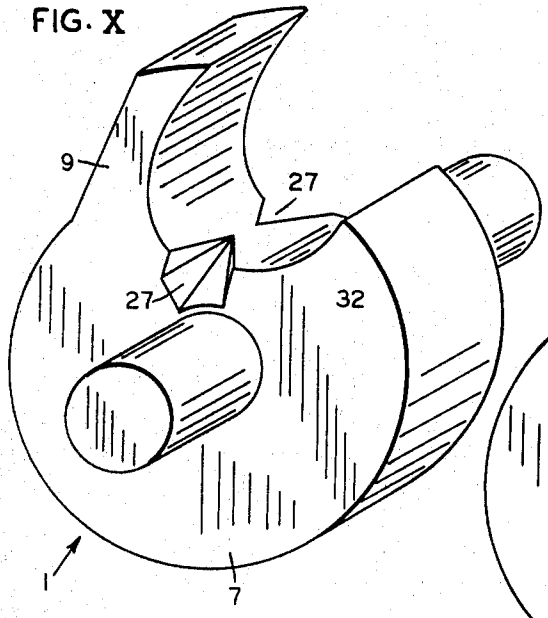


FIG. XXII

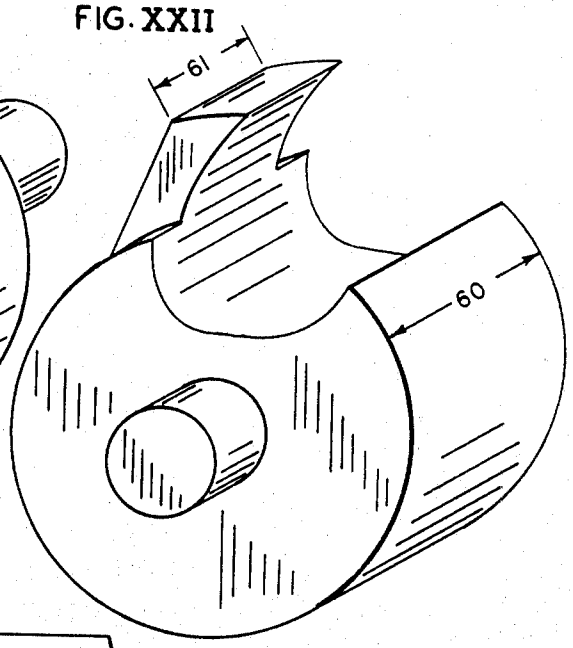
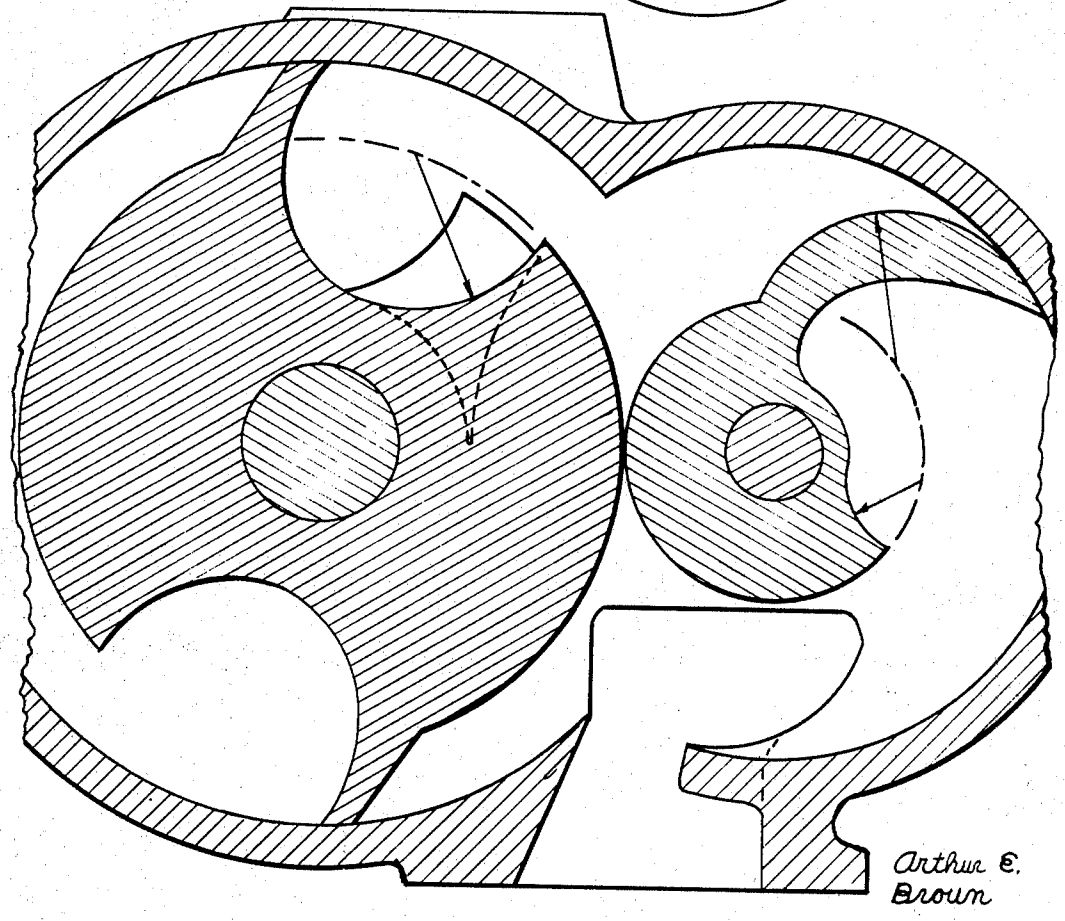


FIG. XXI



ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 7

FIG. XI

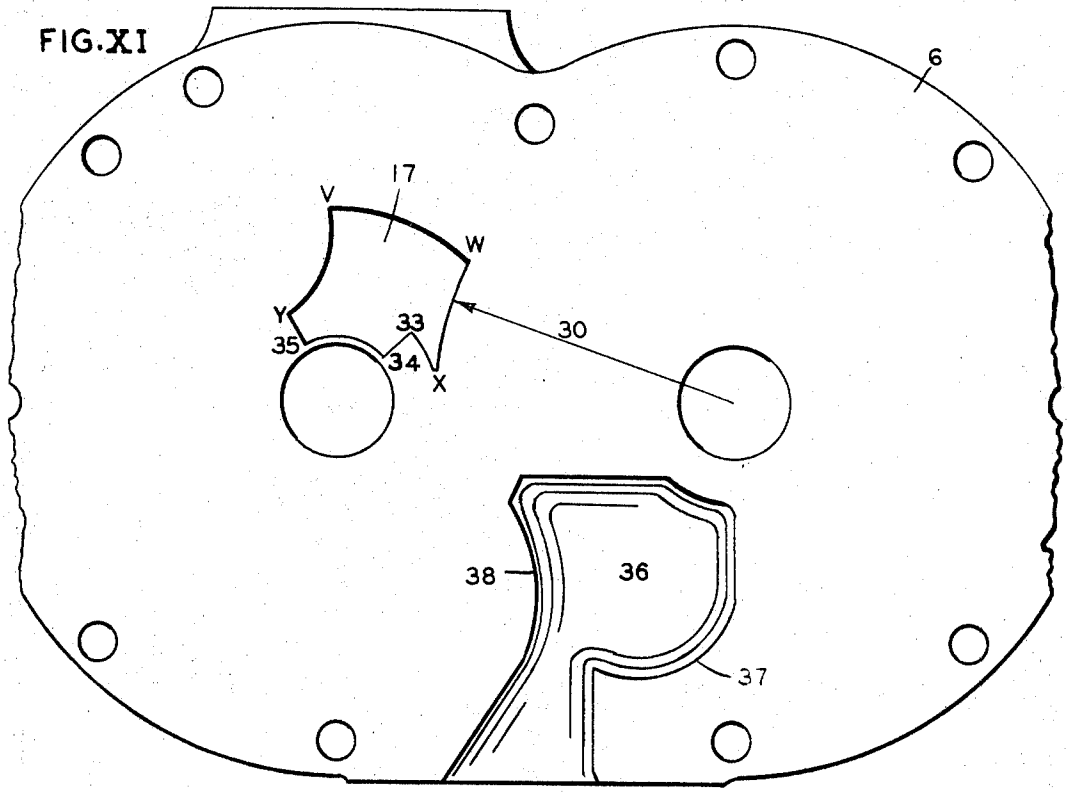
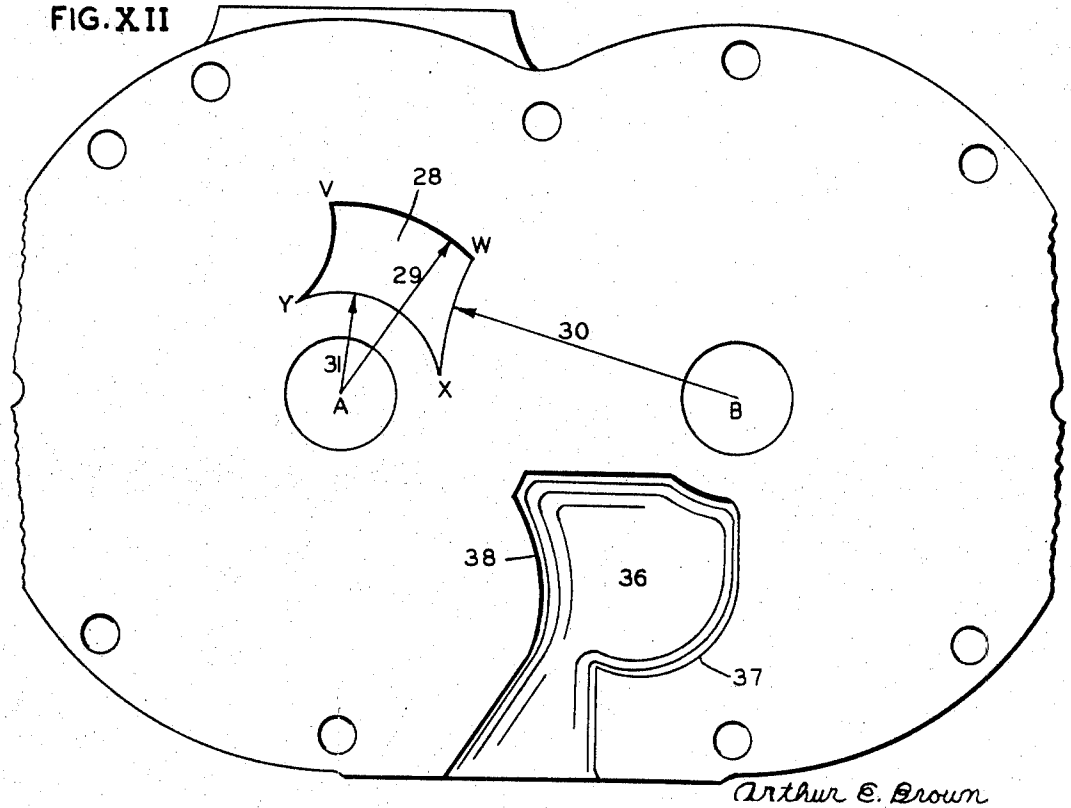


FIG. XII



Arthur E. Brown

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 6

FIG. XV

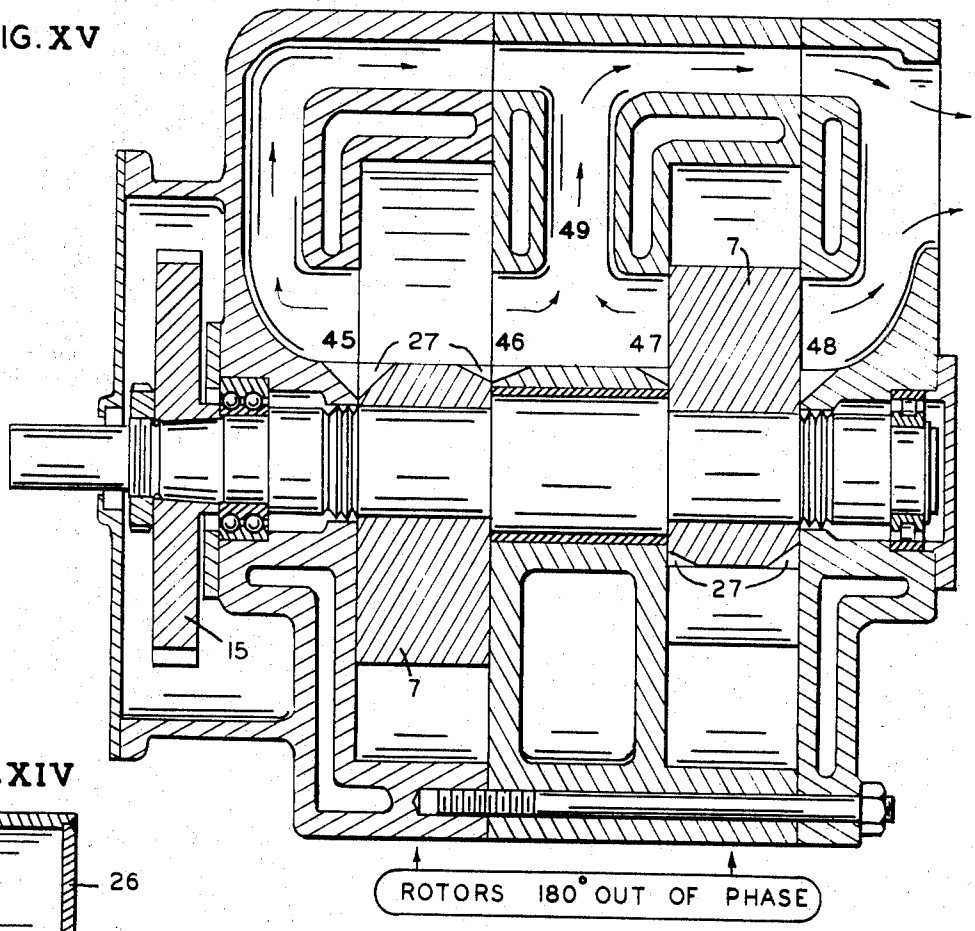


FIG. XIV

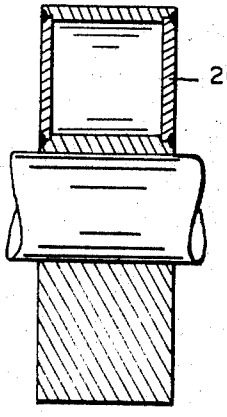
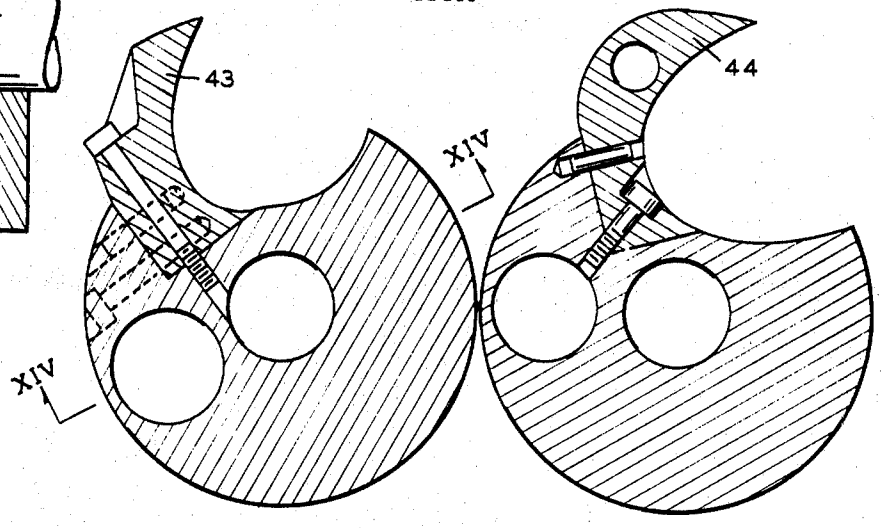


FIG. XIII



Arthur E. Brown.

FIG. XVI

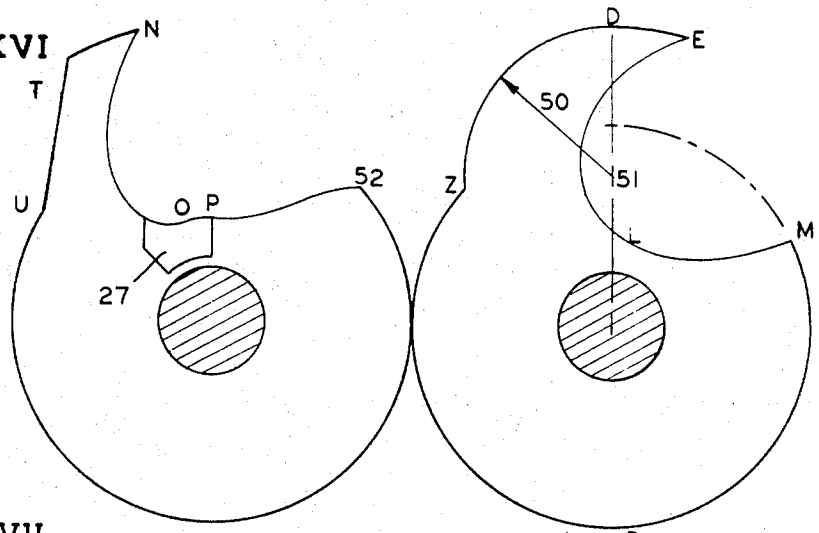


FIG. XVII

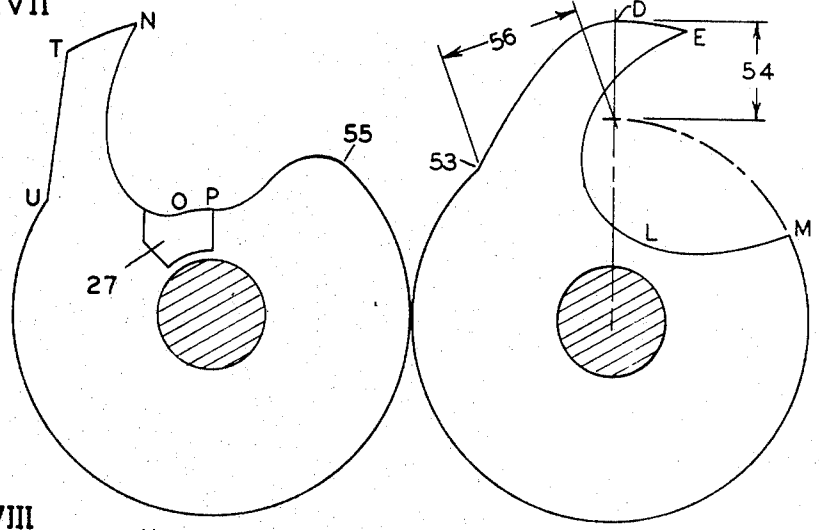
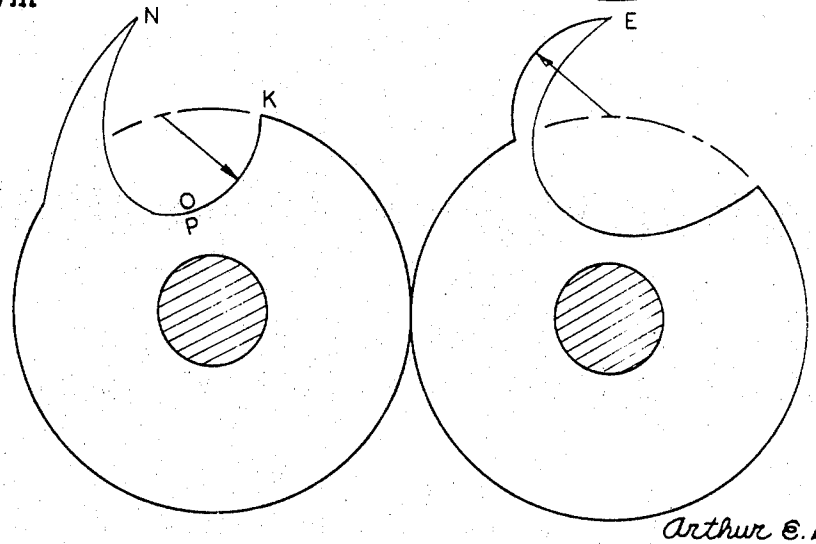


FIG. XVIII



Arthur E. Brown

Oct. 14, 1969 A. E. BROWN 3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 1

FIG. XIX

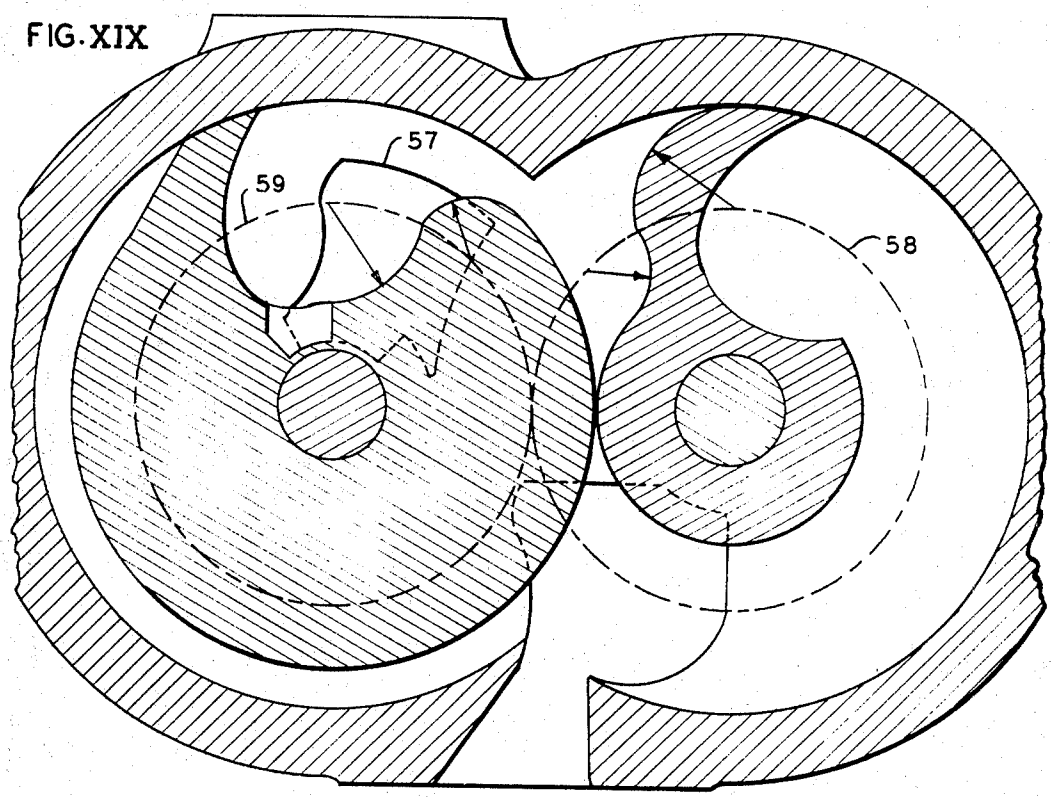
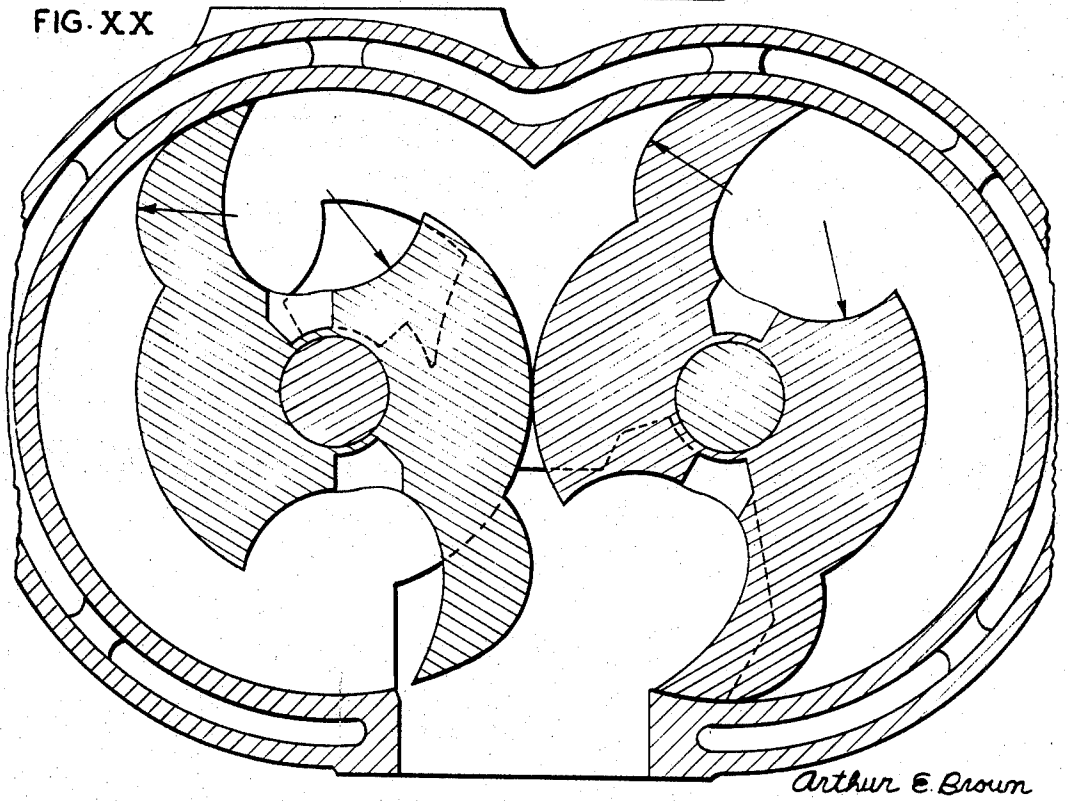


FIG. XX



Arthur E. Brown

ROTARY POSITIVE DISPLACEMENT MACHINES

Filed April 8, 1968 11 Sheets-Sheet 11

FIG. XXIII

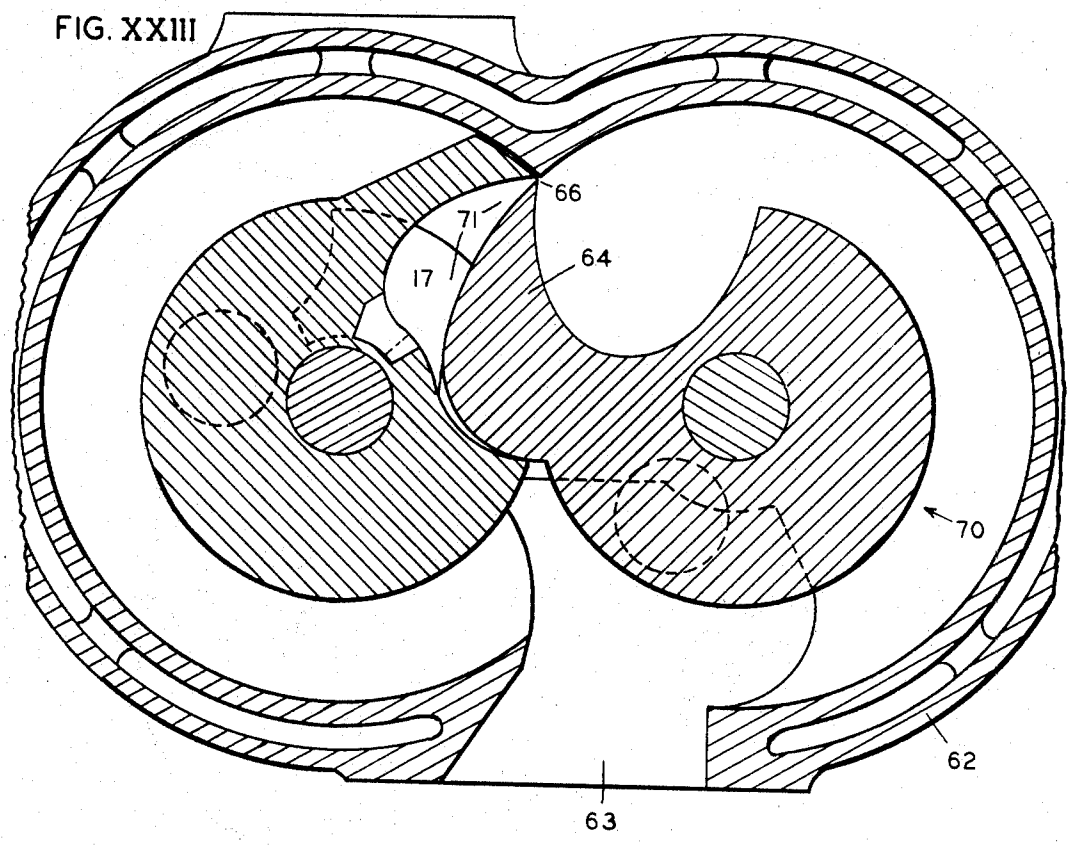
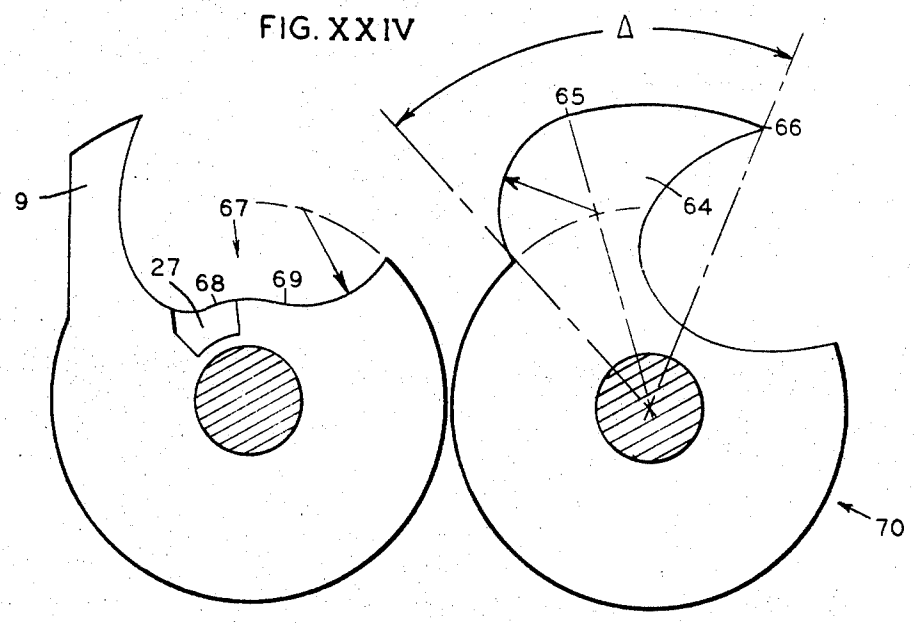


FIG. XXIV



Arthur E Brown

United States Patent Office 3,472,445
Patented Oct. 14, 1969

3,472,445

ROTARY POSITIVE DISPLACEMENT MACHINES

Arthur E. Brown, 117 E. 5th St., Corning, N.Y. 14830

Continuation-in-part of application Ser. No. 522,946, Jan. 25, 1966. This application Apr. 8, 1968, Ser. No. 719,360

Int. Cl. F04c 17/04

U.S. Cl. 230—141

21 Claims

ABSTRACT OF THE DISCLOSURE

A rotary positive displacement machine adapted to handle working fluid. A casing structure has a pair of intersecting bores with an inlet port and an outlet port communicating with the bores. A pair of rotors are rotatably mounted, one in each bore. Each rotor has a hub and a tooth projecting radially outward from each hub. The rotor profiles and the outlet port have such configuration that the clearance volume can be zero or near zero when operated as a compressor, so that nearly all of the gas in the compression chamber is delivered to the outlet port.

This application is a continuation-in-part of my application Ser. No. 522,946, filed Jan. 25, 1966, now abandoned. The drawings and specification in this continuation-in-part application are identical with those in said earlier application except that FIGS. XXIII–XXIV and the description thereof have been added, and "Discussion of Rotor Width and Diameter" has been deleted.

This invention relates to rotary positive displacement machines and has particular applications as a compressor, vacuum pump, or expander. Also shown is a novel method of coupling a flywheel and drive motor to this particular type of rotary machine so as to obtain smoother running operation.

DISCUSSION OF PRIOR ART

Screw type compressors are presently made commercially with several rotor profiles. In one particular type (Whitfield No. 2,287,716), a two thread helical male rotor engages with a four groove helical female rotor. In this machine, each thread space has a small end pocket of fluid or gas which cannot be delivered directly into the discharge port due to the geometry of the configuration. This pocket of gas becomes trapped and would undergo high compression if not for bleed-off schemes. These undelivered trapped pockets represent a clearance volume loss which reduces both capacity and overall efficiency. This loss increases in proportion to the compression ratio of the machine.

A problem with screw type compressors is that the sealing surfaces between the rotors are complex profiles and in addition are helical in shape. The screw type rotors must be held in precise timed relationship (with timing gears) throughout 360 degrees of rotation in order to avoid rotor contact. This means that the running clearances between these complex sealing surfaces cannot be reduced to a low value without greatly increasing the cost, or reducing the reliability against jamming. The screw rotors have end thrust and this requires thrust bearings which much hold the rotors in precise axial positions in order to avoid rotor contact.

Prior art Patent No. 2,097,097 shows the Northey rotary compressor which uses single tooth rotors and side outlet rotor controlled discharged ports. The rotors and discharge ports of the Northey machine are shaped in such a way that there is a substantial clearance volume. This means that a substantial volume of fluid is compressed but not delivered each cycle.

OBJECTS AND ADVANTAGES OF THIS INVENTION

(1) An object is to employ internal compression within the machine so as to avoid backflow throttling of fluid or gas into each successive tooth space.

(2) Clearance volume is defined as a volumetric space in a compressor machine into which fluid or gas is compressed but not delivered to the discharge line. In a rotary compressor it is very desirable to secure a minimum or zero clearance volume because the fluid or gas in the clearance space is generally throttled back to inlet pressure instead of being re-expanded and therefore the clearance volume causes a loss in thermal efficiency as well as a loss in capacity. Clearance volume is particularly undesirable in a high pressure ratio rotary machine because the clearance volume becomes a larger percentage of the delivered volume.

Therefore, a primary object of this invention is to provide novel rotor profiles and a novel discharge port configuration such that the clearance volume (as a compressor) can be reduced to zero (or near zero). Therefore, nearly all of the gas in the compression chamber can be delivered to the discharge ports. As explained, this improves thermal and volumetric efficiencies.

(3) An advantage of this invention is that there are no undesirable trapped pockets which reduce the overall efficiency and capacity of the machine, and cause momentary high loads on bearings and gears. A trapped pocket is one form of clearance volume described in item (2) above and is undesirable for the same reasons.

(4) Another object is to reduce leakage and this is accomplished because of the following reasons (a) to (i).

(a) The rotors conform to the casing structure along the arcs T–N and D–E of FIG. IX. Thus sealing between the casing and the rotors is accomplished with surfaces which are in close proximity along a substantial arc length instead of at just one line or edge. Thus, this leakage path has more resistance.

(b) Most (not all) of the sealing surfaces are cylindrical and straight and can be accurately formed on a lathe. In comparison, most of the sealing surfaces in a screw type compressor are complex, helical surfaces which require contoured milling cutters and complex milling machines to form the helical surfaces. It is easier to accurately form a cylindrical surface than it is a complex helical surface. More accuracy of the sealing surfaces reduces leakage.

(c) The rotor hubs 7 and 8 have the same peripheral velocity and therefore the clearance distance (and leakage) between the hubs can be very low (or even zero) and there is no danger of rubbing or scoring since rolling contact can take place.

(d) During 312 degrees of rotor rotation, sealing between rotors does not depend on timing gear accuracy. At the rotor positions shown in FIG. I, one rotor could rotate several degrees ahead or behind the other and there would be no effect on leakage or interference between rotors. It is only at the positions shown in FIGS. II, III, and IV that rotor timing must be accurate.

(e) Sealing is accomplished with cylindrical sealing surfaces during most of the rotor cycle. It is only during a 48 degree portion of the rotor cycle (FIGS. II, III, and IV) that the generated sealing surfaces are called upon to do a sealing job. This effect will reduce leakage.

(f) The rotors are straight (not helical) and therefore the length of the seal line between the rotors and between each rotor and the casing is equal to the width of the rotors. In comparison, note the much longer seal line 88 in FIG. 31 of the screw machine shown in Whitfield Patent No. 2,287,716. A shorter seal line means less leakage.

(g) As will be subsequently explained, the machine in this invention has 40 percent more displacement (for a given rotor diameter and rotor length) compared with one type of screw machine. Because of this, the length of the seal line is reduced and this will reduce leakage.

(h) The valving is accomplished solely by the rotor profiles and two fixed discharge ports **17** located in the end walls of the casing or housing. Thus, there are no separate valve surfaces to cause leakage.

(i) There are no broken seal lines or built-in leak paths due to the geometry of the rotors as is the case in some prior screw type machines.

(5) A specific object of this invention is to provide simple novel profiles such that the rotors are economical to manufacture and at the same time secure the advantages of zero or low clearance volume, no trapped pockets, and reduced leakage.

(6) Another specific object of this invention is to provide novel notches **27** in a rotor for the purpose of reducing throttling losses which occur during the last portion of each delivery stroke.

(7) Another object is to reduce certain throttling losses through ports **17** by means of the modified rotors shown in FIGS. XXIII and XXIV.

(8) Another object is to shape and locate the inlet port so as to minimize the dynamic losses through the inlet port and at the same time secure maximum displacement.

(9) Another specific object is to reduce the flow losses and secure smoother operation in a large capacity machine by employing two pairs of rotors in a novel paralleling arrangement as shown in FIG. XV.

(10) Another object is to internally balance the single tooth rotors.

(11) Another specific object is to provide a novel flywheel arrangement (FIG. VIII) so as to secure smoother running and reduce timing gear loads.

(12) Another object is to make the rotors more economical to fabricate by employing removable teeth as shown in FIG. XIII.

(13) Another object is to provide an alternate profile for the rotors (as shown in FIG. XIX) which permits additional discharge port area.

(14) An advantage of this invention is that the displacement per rotation substantially exceeds that of a screw type machine for a given rotor diameter and length. This is because of fewer and thinner teeth and the fact that both rotors rotate at the same r.p.m.

(15) Another advantage of this invention is that the valving is accomplished with simple fixed ports **17** located in the end walls of the casing or housing. This reduces cost, reduces leakage, and increases reliability against seizure. Furthermore, the undelivered clearance volume associated with a separate valve has been eliminated.

(16) Another advantage of this invention is that there is no axial thrust on the rotors due to gas pressure. This eliminates the need of larger thrust bearings and permits better control of clearance between the ends of the rotors and the end walls of the casing. This is in contradistinction to screw type compressors which have a substantial end thrust on the rotors.

(17) Another advantage of this invention is that there is less chance of scoring (and ultimate seizure) of the rotors because of the following reasons (a) to (e):

(a) The rotor hubs have the same peripheral velocity and therefore if the hubs should contact each other, the contact would be rolling instead of sliding.

(b) The rotors are larger in diameter and much shorter in length compared with the rotors in a screw type machine. Therefore the rotors are quite rigid and are deflected very little when employed in a high pressure machine. Less deflection means less chance of seizure.

(c) Each rotor has only one profile surface which is generated and this is a straight (not helical) profile. All other non-cylindrical profile surfaces are merely wide clearance surfaces. Therefore there is less chance of rotor seizure.

(d) There are no trapped pockets which would cause high forces tending to separate the rotors and cause them to rub against the housing.

(e) The rotor timing is less critical—item (19).

(18) Another advantage of this invention is that the rotors are lighter in weight and have reduced Wr^2 compared with the rotors in a screw type machine. This is because the rotors have fewer and thinner teeth.

(19) Another advantage of this invention is that the rotor timing is less critical compared to a screw type machine. Accurate timing is required only during 48 degrees of rotor rotation (instead of 360 degrees). This relaxes the cost and accuracy requirements for the timing gears.

(20) Another advantage of this invention is reduced cost compared with a screw type machine because of the following reasons:

(a) More compact for a given displacement (item (14).

(b) The rotors have simple profiles which are easy to fabricate (item (5)), and there is generally one tooth per rotor.

(c) No large thrust bearings are required.

(d) Reduced rotor weight.

(e) The rotors are narrow in width but large in diameter and are non helical. Therefore, several rotors can be stacked together and finish profiled on one machining set up. This saves cost.

(f) Rotor timing is less critical.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. I is a section view taken perpendicular to the rotor axes along the lines I—I in FIG. VII.

FIGS. II to VI are similar to FIG. I except they illustrate successive rotor positions.

FIG. VII is a plan view shown partly in section. The section is taken along the lines VII—VII in FIG. I.

FIG. VIII is a section view taken along the lines VIII—VIII in FIG. I.

FIG. IX is a larger scale view which illustrates in detail the profiles of the FIG. I rotors.

FIG. X is an isometric view of one rotor and its shaft for use in the machine shown in FIG. I.

FIG. XI is an elevation view of the inside face of the casing end plate **6**. This is the same end plate used on the machine shown in FIG. I.

FIG. XII is similar to FIG. XI except it shows the discharge port configuration **28** when no rotor notches are employed.

FIG. XIII is a section view illustrating how the FIG. I rotors may be constructed with removable teeth if desired.

FIG. XIV is a section view taken along the lines XIV—XIV in FIG. XIII.

FIG. XV illustrates another specie of the invention in which two rotors (similar to those in FIG. I) are mounted on one shaft for use in a larger capacity, smoother running machine.

FIGS. XVI to XVIII illustrate alternate rotor profiles operable in the FIG. I casing.

FIG. XIX illustrates an alternate construction which provides additional discharge port area if desired.

FIG. XX illustrates an alternate construction in which each rotor has two teeth instead of one.

FIG. XXI illustrates another specie of the invention.

FIG. XXII illustrates an alternate rotor (similar to FIG. I) but which has a step in it.

FIGS. XXIII and XXIV illustrate new matter in this continuation-in-part. FIG. XXIII is similar to FIG. II except the rotor profiles have been changed slightly and the port **63** has been increased in size. FIG. XXIV is a detail view of the rotors used in FIG. XXIII.

Like numbers indicate similar parts throughout the drawings.

DETAILED DESCRIPTION OF FIGS. I—VIII

Operation as a compressor will be first explained. A first rotor **1** and a second rotor **2** are rotatably mounted in the intersecting bores **3** and **4** in the casing structure **5**. The casing structure includes an end plate **6**. Each rotor has a hub **7–8** and a tooth or lobe **9–10** projecting radially outward from the hub. Each hub has a groove **11–12** located adjacent its respective tooth. A source of power is applied to the drive shaft **13** turning rotor **2** in the direction indicated. The timing gears **14** and **15** drive rotor **1** in proper timed relation. The fluid or gas to be compressed enters the inlet port **16**, is compressed internally within the machine, and is then delivered through the two discharge ports **17—17** which are located in both end walls of a bore in the casing. The compressed fluid is then conducted to a common outlet **18** as shown in FIG. VIII.

The rotor cycle will next be explained. The expandable working chambers **19–20** are filled with gas at inlet pressure during the preceding rotation. At the rotor positions shown in FIG. VI, the chambers are sealed off and internal compression begins. The discharge ports **17** are closed during this part of the cycle by the rotor hub **7**. After the gas in the chambers **19–20** has been compressed to a desired value, the discharge ports **17** are then uncovered by the groove **11** (as shown in FIG. I) so as to permit delivery. The first rotor thus serves as a rotary valve for controlling the flow of the gas through the discharge ports **17**. FIG. II illustrates the position at which the rotors are beginning to intermesh and there is a substantial volume of compressed gas still remaining in the chamber **21** formed between the rotors. It is very desirable that the gas in chamber **21** be delivered through the discharge ports **17**. Otherwise this gas would be wasted with a resulting loss in thermal efficiency and capacity. The profile surface N–O (FIG. III) is shaped such that the outer end E of the tooth on the second rotor sweeps in close sealing proximity across the surface N–O so as to force substantially all of the compressed gas in chamber **21** into the discharge port **17**. FIG. III illustrates the rotor positions as the tip E has swept part way across the face N–O.

FIG. IV illustrates the rotor positions when the delivery stroke has been nearly completed. It can be seen in FIGS. III and IV that substantially all of the gas in chamber **21** is delivered to the discharge ports **17** and there are no trapped pockets. This is accomplished by employing a particular rotor profile and discharge port combination as will be subsequently explained.

FIG. V illustrates the rotor positions during a portion of the cycle when delivery has been completed and the rotors are approaching the start of another compression portion of the cycle.

Balancing holes **22**, **23**, and **24** are shown dotted in FIG. I but have been omitted in FIGS. II to VI for clearness.

The passages **25** are for liquid coolant.

When operating as an expansion engine, the cycle is reversed with compressed gas entering at **17** and exhausting at **16** after expanding internally within the machine.

DESCRIPTION OF ROTOR PROFILES—FIG. IX

The rotors have profiles which permit the machine to have no trapped pockets, a zero (or very low) clearance volume, and an unbroken seal line. The two rotors can be identical in profile although it is preferable to make them of different profiles. The rotors are designated first rotor and second rotor for purposes of identification in the description and claims. The second rotor has a tooth or lobe **10** which has two circular arc profiles C–D and D–E. The arc D–E has its center at B and is a finite arc because it extends over a finite angle. If the angle α (and arc O–P) is reduced to zero, then rotors similar to those shown in FIG. XVIII will be obtained. It is preferable, however, to make arc D–E a finite arc because this: (a) reduces leakage past the tooth, (b) secures a stronger tooth, and (c) reduces pressure drop losses through the discharge ports when the rotors are near the end of each delivery stroke.

The arc C–D is tangent to the finite arc D–E and has a radius F whose center is at G on the pitch circle H. The pitch circle in this case is the same diameter as the hub **8**. The profile surface C–J (generated by the edge K on the first rotor) is not a true circular arc but is very close to being one. For many purposes profile surface C–J can be merely an extension of the arc C–D. The profile surface E–L–M is merely a wide clearance surface and has no sealing function and therefore it is not necessary to form it precisely. The edge E should be straight and accurately located since this edge must pass in close sealing proximity to the generated profile surface N–O on the first rotor.

The circular arc P–K has a radius R whose center is located at Q on the pitch circle S. The radius R may be slightly larger than the radius F so that the two rotors have a running clearance when at the rotor positions shown in FIG. IV.

The circular arc O–P has its center, at A (the rotor axis) and is tangent to P–K and N–O.

The profile surface N–O is generated or developed by the outer edge E of the second rotor. In other words, the edge E sweeps in close sealing proximity across the profile surface N–O as the two rotors rotate. The edge E may be rounded if desired, in which case the profile N–O would be modified accordingly. The circular arc N–T has its center at A. The profile surface T–U is merely a wide clearance surface, has no sealing function, and therefore it is not necessary to form it precisely. The surface T–U is made a straight line as shown instead of making it convex like profile D–C–J on the other rotor. In this way, the tooth **9** can enter and recede from the groove **12** with less displacing and throttling of the gas in the groove **12**.

The arcs C–D and P–K may, if desired have minor deviations from being true circular arcs and substantially the same results will be obtained. The terms employed in the appended claims are intended to include such deviations.

The rotors should be dynamically balanced about their axes to avoid vibration. The lightening holes **22**, **23**, and **24** (with a size and location as drawn) secure balancing of the rotors. This has been verified by calculation. The rotors are balanced internally within themselves and it is not necessary to add counterweights at outboard locations exterior of the rotors. This feature saves cost and prevents shaft deflection. If desired, the balancing holes may be provided with welded end plates **26** as shown in FIG. XIV.

DESCRIPTION OF DISCHARGE PORTS—FIG. XII

The location and geometry of the discharge ports (in combination with the particular rotor profiles) permits the machines to have no trapped pockets and zero (or very low) clearance volume. FIG. XII illustrates the discharge port configuration to be used when no notches **27** (FIG. X) are employed. The lines V–W–X–Y–V define the outline of a discharge port **28** located in an end wall of a bore in the casing structure. The circular arc V–W is the outer edge of the port and has its center at A and has a radius **29** which may generally be a fraction of an inch smaller than the radius of the rotor hub **7** so that the discharge port can be covered or controlled by the rotor hub. The circular arc W–X is termed the trailing edge of the port and has its center at B, the axis of the second rotor. The arc W–X has a radius **30** substantially equal to the outer radius of the tooth **10** on the second rotor. The circular arc X–Y is termed the inner edge of the port, has its center at axis A, and has a radius **31** which generally may be approximately equal to the radius of the arc O–P in FIG. IX so as to secure maximum port area. The shape of the leading edge Y–V is not critical and the machine would function if it were a straight

radial line instead of being a circular arc as shown. However, for optimum performance, it is desirable that the discharge port open as rapidly as possible once the gas being compressed is up to the desired pressure. For this reason therefore, the leading edge Y–V is made a circular arc as shown so as to coincide with the circular arc P–K on the first rotor profile (FIG. IX).

DESCRIPTION OF FIGS. X AND XI

These figures illustrate more clearly the form and function of the notches **27** which are located on each side of the first rotor. The notches interrupt the side wall **32** of the rotor and the groove **11** in the rotor. The use of the notches is optional but they are very desirable under certain operation conditions as will next be explained. As the rotors approach the end of their delivery stroke (FIGS. II and III), the uncovered area of the discharge ports becomes less and less. Therefore under some operating conditions the pressure drop through the discharge ports (near the end of each delivery stroke) would become too high and this would cause a loss in efficiency. Therefore, to overcome this problem, the notches provide more flow area through the discharge ports at a critical time when it is needed. The notches communicate with or overlap the added areas **33–34–35–Y** (FIG. XI) in the discharge ports. The shape of the discharge port in FIG. XI is similar to that in FIG. XII except for the area **33–34–35–Y** which has been added for the purpose of serving the flow of gas through the notches **27**. In FIGS. II and III, it is seen that the notches are overlapping the discharge ports. This means that gas can flow through the notches during these rotor positions. In FIG. IV, the notches are being cut off from the discharge ports since the rotors have nearly completed their delivery stroke.

The notches, of course, constitute a clearance volume and therefore the volume of the notches is made small in relation to their flow area. For this reason, the notches do not extend across the width of the rotor but are located one on each side as shown in FIG. X. The notches are also tapered or conical in shape so as to further reduce their volume but still retain their desired flow area to the discharge ports. As previously explained, a clearance volume is not desirable but it has been calculated that under many operating conditions, the notches do more good than harm. The notches are particularly desirable where the r.p.m. is high, when the rotors have a relatively wide width, and when the pressure ratio across the machine is low. When the machine is operating as a vacuum pump, the notches may be omitted.

It has been calculated that when compressing air from atmospheric inlet to a 4.5 pressure ratio using 15 inch diameter x 3.25 wide rotors running at 3600 r.p.m., the instantaneous pressure drop through the discharge ports when at the FIG. II rotor position is reduced from 6 p.s.i. to 2.5 p.s.i. by the addition of notches. Likewise the instantaneous pressure drop at the FIG. III position is reduced from 10 p.s.i. to 2.6 p.s.i. by the addition of notches. The calculated loss in capacity (due to the addition of notches) under the above conditions is 1.6% and the calculated power loss (because of clearance volume losses due to the notches) is less than 1%. In this case, the reduction in power loss, due to pressure drop through the discharge ports, more than makes up for the added power loss due to the addition of clearance volume.

DISCUSSION OF INLET PORTS

An object is to provide adequate inlet port area (so as to prevent throttling losses through the inlet port) but a conflicting objective is to minimize the angle occupied by the inlet port so as to secure maximum displacement in a given machine. Referring to FIGS. XI and XII, the casing end walls are recessed at **36** for the purpose of providing additional inlet port area. This is in addition to the radial inlet **16** shown in FIG. I. Referring to FIG. V, the inlet port edge **37** has the same shape as the leading edge D–J of one rotor tooth. Also, the other inlet port edge **38** has the same shape as the leading edge N–O of the other tooth. The port edges **37** and **38** are located such that when the rotors reach the position shown in FIG. VI (start of compression), the port edges coincide with the leading edges of the rotor teeth. Thus, a maximum inlet port area is secured without detracting from the maximum displacement of the machine.

When the rotors are at the positions shown in FIG. V, the tooth **9** on the first rotor is withdrawing from the groove **12** in the second rotor. At this position, the groove **12** has already begun to overlap the inlet port **36** in the end walls and therefore the groove can be filled (with low losses) from this source. Without the end wall porting, as shown, the gases would have to follow a tortuous path around the rotors to fill the groove **12** when at the FIG. V position and this would result in a loss in efficiency and capacity particularly at higher speeds.

DISCUSSION OF FLYWHEEL AND DRIVE MOTOR—FIG. VII

When operating as a compressor, it will be convenient to direct couple the machine to an electric drive motor **39** by means of a torsionally stiff shaft coupling **40** as shown in FIG. VII. A flywheel **41** is fastened to the timing gear **15**. It is particularly pointed out that this flywheel is not attached to the same shaft which is driven by the electric motor but is instead attached to the other shaft system. The flywheel **41** is made of a size such that it has about the same Wr^2 as the electric motor and coupling. Thus, the two oppositely rotating systems have substantially equal Wr^2. The purpose of the flywheel and the reason for locating and sizing it as described will next be explained. The rotors shown in FIGS. I and VII have only one tooth per rotor. This means that the torque required to turn the rotors against gas pressure is not smooth; in fact this torque varies from a maximum in FIG. I to zero in FIG. V. The electric drive motor puts out an average steady torque which changes very little between the FIG. I position and the FIG. V position because the rotative speed does not change that much. When the rotors are at the FIG. I position, the torque required to turn the rotors against gas pressure exceeds the steady average torque put out by the electric drive motor and therefore the additional needed torque must come from flywheel effect. Because the two rotor shafts turn in opposite directions and because they have equal Wr^2, there will be no torque reaction transmitted to the foundation. The torque reaction of one rotor shaft counteracts equally and oppositely the torque reaction of the other rotor shaft so that the net external torque reaction transmitted to the foundation is zero. This prevents unwanted vibration and facilitates smooth running.

The flywheel **41** has an added function and advantage in that it smooths out the load on the timing gears **14–15**. In FIG. I, part of the peak torque (required to turn the rotors against gas pressure) comes from the flywheel. Otherwise, part of this peak torque would have to be transmitted through the timing gears.

It is thus seen that the flywheel (sized and located as described) performs two functions: (a) It prevents vibration due to torque reaction and (b) it smooths out the load on the timing gears. An alternate flywheel location is shown at **42**.

DESCRIPTION OF FIG. XIII

This figure illustrates an alternate rotor construction in which the teeth **43** and **44** are separate pieces so as to facilitate manufacturing operations of both the hubs and the teeth. The bolting and doweling arrangements shown securely hold the teeth, yet, do not interfere with the sealing function of the profile surfaces.

DESCRIPTION OF FIG. XV

This figure illustrates a paralleling arrangement best suited for a large capacity machine. A problem is to secure adequate area for the discharge ports so as to avoid pressure drops through same. FIG. XV illustrates a simple method of doubling the discharge port area for a given rotor diameter. Four dischage ports are shown at **45, 46, 47,** and **48.** There is also another advantage to be gained at the same time which is smoother air flow. FIG. XV is a section view taken through the axis of the shaft upon which two first type rotors are mounted. The two rotors are mounted 180 degrees out of phase from each other. In this manner, the air or gas delivery from each pair of rotors does not occur at the same time but is instead spaced 180 degrees apart. This arrangement thereby facilitates a simple construction in which the passage **49** and adjacent discharge ports **46–47** serve both sets of rotors. The passage **49** need be no larger than if it served one set of rotors because the delivery cycles are out of phase. The inlet gas flow is also smoother with this arrangement.

DESCRIPTION OF FIG. XVI

This figure illustrates alternate rotor profiles operable in the casing shown in FIG. I. The profile surfaces U–T–N–O–P and D–E–L–M are identical with those of FIG. IX. The circular arc D–Z has a radius **50** whose center is at **51** (not on the pitch circle). The circular arc D–Z is tangent to the circular arc D–E. The profile surface **52**–P is an irregular curve and is the envelope (plus running clearance) developed by the surface D–Z as the two rotors rotate. These rotors (in combination with the discharge ports shown in FIG. I) will result in a machine with no trapped pockets, no clearance volume (except for the notches **27**) and no broken seal line.

The radius **50** is larger than the radius F in FIG. IX. The dimension F is equal to the radial height of the tooth outside the pitch circle. If the circular arc D–Z had a radius substantially smaller than the radial height of the tooth, then the rotors would not function properly because the seal line would be broken. If the circular arc D–Z is used, then its radius must be substantially equal to F or larger in order for the rotors to function properly.

DESCRIPTION OF FIG. XVII

This figure illustrates another pair of alternate rotor profiles which are operable in the FIG. I housing. The profile surfaces U–T–N–O–P and D–E–L–M are identical with those of FIGS. IX and XVI. The surface D–**53** is a convex non-circular curve which is tangent at D to the curve D–E. The dimension **54** denotes the radial height of the tooth. For best operation, the dimension **56** should be substantially equal to or greater than **54.** Otherwise the seal line may be broken or there may be a trapped pocket prior to completion of the delivery stroke. The profile surface **55**–P is an irregular curve and is the envelope (plus running clearance) developed by the curve **53**–D as the two rotors rotate.

DESCRIPTION OF FIG. XVIII

Referring first to FIG. IX, if the circular arcs T–N, O–P, and D–E were reduced to zero, then the rotor profiles shown in FIG. XVIII would be essentially obtained. The FIG. XVIII rotors may operate in the FIG. I casing and have zero clearance volume and no trapped pockets.

DESCRIPTION OF FIGS. XIX, XX, XXI, and XXII

FIG. XIX illustrates a method of increasing the discharge port area for a given rotor diameter. The discharge port has an outer radius **57** which is larger than the pitch circles **58** and **59.** The rotor profiles contain four circular arcs whose centers lie on the pitch circles as indicated.

The double tooth rotors shown in FIG. XX can provide aboout 20% more displacement than the FIG. I rotors but they also have several disadvantages.

In FIG. XXI, one rotor rotates twice the r.p.m. of the other. The clearance volume may be negligible and there are no trapped pockets.

Another alternate (but less preferred) embodiment of this invention (FIG. XXII) is to make the axial width **60** of the rotor hub slightly wider than the axial width **61** of the tooth. Then a corresponding diametral step in the casing bore is required to accommodate the axial extension of the rotor hub. Under this condition, the outer edge of the discharge port may have a radius equal to that of the pitch circle of the rotor. It is intended that the appended claims are not avoided merely through the use of this variation.

DESCRIPTION OF FIGS. XXIII AND XXIV

When the machine operates as a compressor, the uncovered flow area through discharge ports **17** becomes less and less as the rotors approach the end of their delivery stroke (FIGS. II and III rotor positions). Careful analysis and calculations have indicated that by employing the modified rotors shown in FIGS. XXIII and XXIV, it is possible to operate the machine at higher rotative speeds and/or with wider rotors without sustaining excessive throttling losses through ports **17.** The preceding statement applies to machines operating on pressure ratios of about 3.5 or less.

The casing **62** is similar to that in FIG. I except for the larger inlet port **63.** The rotors are identical with those in FIG. I except that the tooth **64** has a larger angle of arc **65–66** and the groove **67** has a larger angle of arc **68–69.** The second rotor **70** is able to complete its portion of the delivery stroke while there is a substantial uncovered flow area through ports **17** still available. At the rotor position shown in FIGS. XXIII, the second rotor **70** has nearly completed its portion of the delivery stroke. This means that (after this point in the rotor cycle) there is a reduced rate of displacement (and flow rate) and thus the pressure loss through the discharge ports **17** is reduced.

The outer edge E (or **66**) of the second rotor sweeps in sealing proximity across the face of the tooth on the first rotor and this portion of the rotor cycle is hereby defined as phase P. This definition of phase P is also employed in one or more annexed claims. When operating as a compressor, the beginning of phase P would be at the rotor positions shown in FIGS. II or XXIII and the end of phase P would be approximately at the rotor positions shown in FIG. III. The rotor positions shown in FIG. IV are not within the defined phase P.

The numerals **21** and **71** denote chambers formed between the rotors and the casing at the rotor positions shown in FIGS. II and XXIII. The volume of chamber **71** is substantially less than chamber **21,** yet the uncovered flow area through ports **17** is the same in both cases. This means that in FIG. XXIII there is less volume of gas to flow through the same uncovered port area; and thus there is less throttling loss during this critical portion of the rotor cycle.

A further advantage of the modified FIG. XXIV rotors is that the groove **67** has a larger angle of arc **68–69** and therefore there is a larger uncovered flow area through ports **17** at rotor positions prior to that shown in FIG. XXIII.

Also, such a modification of the rotors permits the inlet port **63** to be larger (less flow resistance) yet there is very little reduction (compared to FIG. I) in total displacement.

An advantage of the larger tooth **64** is that the leak path past the tooth is longer and hence the tooth has more resistance to leakage.

It is noted that the above advantages are brought about by making the tooth on the second rotor and the groove in the first rotor wider in angle of arc. It is not desirable

to increase the angle of arc of the tooth on the first rotor as this would reduce the total displacement of the machine.

For optimum results, the arc 65–66 should be large enough that the second rotor has nearly completed its displacement at the beginning of phase P. This is accomplished by making the total angle Δ equal to about 65 degrees. The angle Δ refers to the total angle occupied by all portions of the second tooth (as shown in FIG. XXIV).

The preceding discussion of FIGS. XXIII and XXIV applies to machines operating on pressure ratios of about 3.5 or less. For the higher pressure ratios, it will sometimes be preferable (depending on rotor speed and width) to employ the FIG. I proportions; otherwise the maximum area of the ports 17 is limited.

While the preferred embodiments of the invention have been described, it will be understood that the invention is not limited thereto since it may be otherwise embodied within the scope of the following claims.

In order to make the claims more clear, reference letters and numbers are used in the claims to identify certain parts and a mode of operation. Examples of such reference letters and numbers are: first rotor, second rotor, phase P, line C, etc. The claims are not to be avoided merely by labeling a part by a different reference letter or number or by omitting a reference letter or number from a part.

What is claimed is:

1. In a rotary positive displacement machine adapted to handle a working fluid, the combination of: a casing structure having a pair of intersecting bores, said casing structure having an inlet port for the entrance of said fluid into said bores, said casing structure having an outlet port for the exit of said fluid from said bores, at least one of said ports being located in an end wall of a said bore, a first rotor rotatably mounted in one said bore and a second rotor rotatably mounted in the other said bore, each rotor having a hub, a first tooth projecting radially outward from the hub of the first rotor, a second tooth projecting radially outward from the hub of the second rotor, each hub having a groove in it, each said groove being located adjacent its respective tooth, timing gears constraining said rotors to rotate at equal r.p.m. in opposite directions of rotation, said second tooth being adapted to interengage with said groove in the first rotor as the two rotors rotate, said first tooth having a concave face surface the profile of which is hereby designated line "A," said groove in the first rotor having a concave surface the profile of which is hereby designated line "B," said line "A" and said line "B" having profiles such that the outer end of said second tooth sweeps in close sealing proximity along lines "A" and "B" as the two rotors rotate, said second tooth having a face surface which is convex and the profile of which is hereby designated as line "C," said two rotors having pitch circles of equal diameter, said line "C" substantially intersecting the pitch circle of the second rotor at a point of intersection which is hereby designated point "D," the outer radial end of said second both being adapted to rotate in close sealing proximity to the wall of its respective casing bore, an imaginary circle passes through said outer radial end of the second tooth, said imaginary circle having its center at the axis of the second rotor, the outer radial end of said line "C" generally intersects said imaginary circle at a point which is hereby designated point "E," an imaginary straight radial line interconnects said point "E" and the axis of the second rotor and this line is hereby designated line "F," said line "F" intersects the pitch circle of the second rotor at a point hereby designated point "G," the distance from said point "G" to said point "D" being substantially equal to or greater than the distance from said point "G" to said point "E," and the purpose of shaping said rotors as specified in this claim being to prevent the formation of trapped pockets and also to substantially maintain a seal line between the rotors during a portion of the rotor cycle when said second tooth engages with said groove in the first rotor.

2. The combination defined in claim 1 wherein the said line "C" is substantially tangent at point "E" to the said imaginary circle.

3. The combination defined in claim 1 wherein the said second tooth has a profile at the outer radius of the tooth which is in the form of a finite circular arc, said finite circular arc having its center at the axis of the second rotor, said line "C" being substantially tangent to said finite circular arc.

4. The combination defined in claim 2 wherein the said line "C" is a circular arc of substantially constant radius.

5. The combination defined in claim 4 wherein line "C" has its center substantially on the pitch circle of said second rotor, said groove in the first rotor having a portion of its profile in the form of a concave circular arc, and said concave circular arc having its center substantially on the pitch circle of said first rotor.

6. The combination defined in claim 1 wherein the said groove in the first rotor has a portion of its profile in the form of an irregular curve, and said irregular curve being the envelope plus clearance developed by said line "C" as the two rotors rotate.

7. In a rotary positive displacement machine adapted to handle a working fluid, the combination of: a casing structure having intersecting bores, a first rotor rotatably mounted in one said bore and a second rotor rotatably mounted in the other said bore, said rotors and said bores bounding an expandable working chamber, said casing structure having an inlet port for the entrance of the working fluid into said expandable chamber, said casing structure having a discharge port for the exit of said fluid from said expandable chamber, each rotor having a hub, a tooth projecting radially outward from each said hub, each hub having a groove in it, each groove being located adjacent its respective tooth, said tooth on the second rotor being adapted to interengage with said groove in the first rotor as the two rotors rotate, at least one of said ports being located in an end wall of a said bore, said port in an end wall being closed during a portion of the rotor cycle by the hub of said first rotor, said port in an end wall being uncovered during another portion of the rotor cycle by said groove in the hub of the first rotor, said first rotor thus serving as a rotary valve for controlling the flow of said fluid through said port in an end wall, and the edges of said port in an end wall being defined as follows: (a) the outer edge of the port runs in a circumferential direction with respect to the axis of the first rotor, the outer edge of the port is located from the axis of the first rotor at a radial distance such that the port may be controlled by the hub on the first rotor; (b) the trailing edge of the port is substantially a circular arc having its center substantially at the axis of the second rotor, the trailing edge of the port is located in proximity to the circular path of the outer end of the tooth on the second rotor, (c) the inner edge of the port runs in a generally circumferential direction and connects the leading and trailing edges of the port, and (d) the leading edge of the port is a radially directed edge which connects the said inner and outer edges of the port.

8. The combination defined in claim 1 wherein the said port in the end wall of a bore has one of its edges in the form of a substantially circular arc which conforms substantially to the path of the tip of the tooth on the second rotor.

9. In a rotary positive displacement machine adapted to handle a working fluid, the combination of: a casing structure having a pair of intersecting bores, said casing structure having an inlet port and a discharge port, at least one of said ports being located in an end wall of a bore, a first rotor and a second rotor both mounted for rotation inside said casing, timing gears constraining the

two rotors to rotate in timed relation, each rotor having a hub, a first tooth attached to the hub of said first rotor, a second tooth attached to the hub of said second rotor, each hub having a groove in it, the hub of said first rotor serving as a rotary valve for opening and closing said port in an end wall, said second tooth being adapted to interengage with said groove in the hub of the first rotor, said first tooth having a profile on one face such that the outer end of said second tooth sweeps in sealing proximity across said face of the first tooth and this portion of the rotor cycle is hereby designated phase "P," said rotors and said casing bounding a certain chamber during said phase "P," said chamber being of variable volume as said phase "P" progresses, and said port in an end wall being shaped and located such that it is in open communication with said chambers throughout said phase "P" so that said working fluid may pass through said port in the end wall during said phase "P."

10. The combination defined in claim **9** wherein the said casing structure contains a second pair of intersecting bores, a third rotor and a fourth rotor rotatably mounted in said second pair of bores, said first and third rotors being coaxially mounted on one shaft, said second and fourth rotors being coaxially mounted on a second shaft, timing gears fastened to and interconnecting the two said shafts so as to constrain the shafts to rotate in opposite directions in proper timed relation, said casing structure having at least one inlet port for the entrance of said fluid into all four said bores, said casing structure having at least four discharge ports for the exit of said fluid from all four said bores, said four discharge ports being located in the end walls of said bores, said third and fourth rotors each having a hub, a third tooth projecting from the hub of said third rotor, a fourth tooth projecting from the hub of said fourth rotor, the said hubs of the third and fourth rotors each having a groove adjacent their respective teeth, each tooth being adapted to enter and recede from the groove in its coacting rotor as the rotors rotate, all four said teeth being adapted to displace and compress said fluid, the purpose and advantage of employing four motors (instead of two wider rotors of the same total displacement) being the gain in area of the discharge ports for a given rotor diameter, and the advantage of said gain in discharge port area being a reduction in throttling loss of the fluid passing through the discharge ports.

11. In a rotary positive displacement compressor machine adapted to compress a working fluid, the combination of: a casing structure having at least two intersecting bores, said casing structure having an inlet port for the entrance of said fluid into the interior of said bores, said casing structure having a discharge port for the exit of the fluid from the interior of said bores, said discharge port being located in an end wall of a said bore, a first rotor rotatably mounted in one said bore and a second rotor rotatably mounted in another said bore, each rotor having a hub, a tooth projecting radially outward from each said hub, each hub having a groove located adjacent its respective tooth, the tooth on said first rotor having a profile such that the outer end of the tooth on the second rotor sweeps in close sealing proximity across the leading face of the tooth on the first rotor as the two rotors rotate, the said groove in the first rotor having a profile such that the outer end of the tooth on the second rotor sweeps in close sealing proximity across one face of the groove as the two rotors rotate, said tooth on the second rotor being adapted to interengage with said groove in the first rotor as the two rotors rotate, said first rotor serving as a rotary valve for controlling the flow of said fluid through said discharge port, said first rotor having a notch interrupting the side wall of the rotor and the groove in the rotor, said discharge port being shaped so as to cyclically communicate with said notch as the rotor rotates, said notch being alternately in open and closed communication with said discharge port as the rotor rotates, said notch serving as a flow path for the flow of said fluid into said discharge port, and the purpose of said notch and shaped discharge port being to increase the uncovered discharge port area near the end of each delivery stroke so as to reduce throttling losses of the working fluid during this portion of the rotor cycle.

12. The combination defined in claim **11** wherein said notch is tapered and extends only part way across the width of its rotor, and the purpose of shaping said notch as just described being to reduce its undelivered volume, yet, retain its ability to pass fluid into the discharge port.

13. The combination defined in claim **9** wherein the total angle occupied by all portions of said second tooth is equal to a minimum of 55 degrees.

14. In a rotary positive displacement compressor machine adapted to compress a working fluid, the combination of: a casing structure having at least two intersecting bores, said casing structure having an inlet port for the entrance of said fluid into said bores, said casing structure having a discharge port for the exit of said fluid from said bores, said casing structure having internal end walls which bound the axial ends of said bores, said discharge port being located in a said end wall, a first rotor rotatably mounted in one said bore and a second rotor rotatably mounted in the other of said bore, each rotor having a hub, a first tooth projecting radially outward from the hub of said first rotor, a second tooth projecting radially outward from the hub of said second rotor, each hub having a groove in it, the hub of said first rotor being adapted to cover said discharge port during a portion of the rotor cycle, said groove in the hub of the first rotor being adapted to uncover said discharge port during another portion of the rotor cycle, said first rotor thus serving as a rotary valve for controlling the flow of said fluid through said discharge port, said second tooth being adapted to interengage with said groove in the first rotor as the two rotors rotate, said first tooth having a profile on its leading face such that the outer end of said second tooth sweeps in close sealing proximity across the leading face of said first tooth as the two rotors rotate, a first portion of said inlet port being located in a said end wall, and said first portion of the inlet port being intersected by both of said bores in the casing.

15. The combination recited in claim **14** wherein a second portion of said inlet port passes radially through the radial wall of said second bore.

16. The combination recited in claim **14** wherein one edge of said inlet port has a shape substantially similar to the profile of one face of said first tooth, wherein another edge of said inlet port has a shape substantially similar to the profile of one face of said second tooth, and the purpose of shaping said inlet port as just specified being to secure maximum inlet port area, yet minimize the angle occupied by the inlet port so as to secure maximum displacement of said compressor machine.

17. The combination defined in claim **9** wherein the diameter of said hub on said first rotor is substantially larger than the pitch diameter of said first rotor, the diameter of said hub on said second rotor is substantially smaller than the pitch diameter of said second rotor, the outer radius of said port in an end wall is substantially larger than the radius of the pitch circle of said first rotor, and said second tooth having a profile on its leading face which is convex outside the pitch circle and concave inside the pitch circle.

18. In a rotary positive displacement compressor machine adapted to compress a working fluid, the combination of: a casing structure having intersecting bores, said casing structure having inlet and discharge ports for the entrance and exit of said working fluid, a first rotor rotatably mounted in one said bore and a second rotor rotatably mounted in the other said bore, each rotor having a hub, a single tooth projecting radially outward from each said hub, each hub having a groove in it, each groove being located adjacent its respective tooth, each tooth

being adapted to enter and recede from the groove in the opposite rotor as the two rotors rotate, each said tooth being adapted to displace and compress the working fluid within the casing structure, the torque required to turn the rotors being cyclical in magnitude due to the fact that each rotor has only a single tooth, timing gears constraining the two rotors to rotate in opposite directions in proper timed relation, an electric drive motor coupled to one rotor so as to drive the rotor, a flywheel coupled to the other rotor, said flywheel being adapted to rotate in the opposite direction to that of said electric rotor, and the purpose and use of said flywheel being to cause the machine to be smoother running and to smooth out the loads transmitted by said timing gears.

19. The combination defined in claim **18** wherein the Wr^2 of all parts rotating clockwise is approximately equal to the Wr^2 of all parts rotating counterclockwise (when viewed from one end of the machine).

20. In a pair of rotors for use in a positive displacement machine, the combination of: each rotor having a hub, each hub being circular throughout a portion of its circumference, a single tooth attached to each hub and projecting radially outward therefrom, each hub having a groove extending across its width, each groove being located adjacent its respective tooth, each tooth being adapted to enter into and recede from the groove in the opposite rotor as the two rotors rotate, each hub having a balancing hole located therein, said balancing holes having a size and location such that both of said rotors are balanced about their axes, and each said balancing hole being located at an angular location (about its respective rotor axis) which is between its respective tooth and a point diametrically opposite its respective groove.

21. The combination defined in claim **9** wherein said second tooth has a face surface which is convex and the profile of which is hereby designated line "C," said two rotors having pitch circles of equal diameter, said line "C" substantially intersecting the pitch circle of the second rotor at a point of intersection which is hereby designated point "D," the outer radial end of said second tooth being adapted to rotate in close sealing proximity to the wall of its respective casing bore, an imaginary circle passes through said outer radial end of the second tooth, said imaginary circle having its center at the axis of the second rotor, the outer radial end of said line "C" generally intersects said imaginary circle at a point which is hereby designated point "E," an imaginary straight radial line interconnects said point "E" and the axis of the second rotor and this line is hereby designated line "F," said line "F" intersects the pitch circle of the second rotor at a point hereby designated point "G," the distance from said point "G" to said point "D" being substantially equal to or greater than the distance from said point "G" to said point "E," and the purpose of shaping said second tooth as specified in this claim being to prevent the formation of trapped pockets and also to substantially maintain a seal line between the rotors during a portion of the rotor cycle when said second tooth engages with said groove in the first rotor.

References Cited

UNITED STATES PATENTS

2,058,817	10/1936	Northey.	
2,097,037	10/1937	Northey	230—141
2,287,716	6/1942	Whitfield	230—143

FOREIGN PATENTS

341,324	1/1931	Great Britain.
661,749	11/1951	Great Britain.
900,881	7/1962	Great Britain.

DONLEY J. STOCKING, Primary Examiner

WILBUR J. GOODLIN, Assistant Examiner