March 7, 1950 A. E. BIERMANN 2,500,143

ROTARY ABUTMENT COMPRESSOR

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Fig. 1

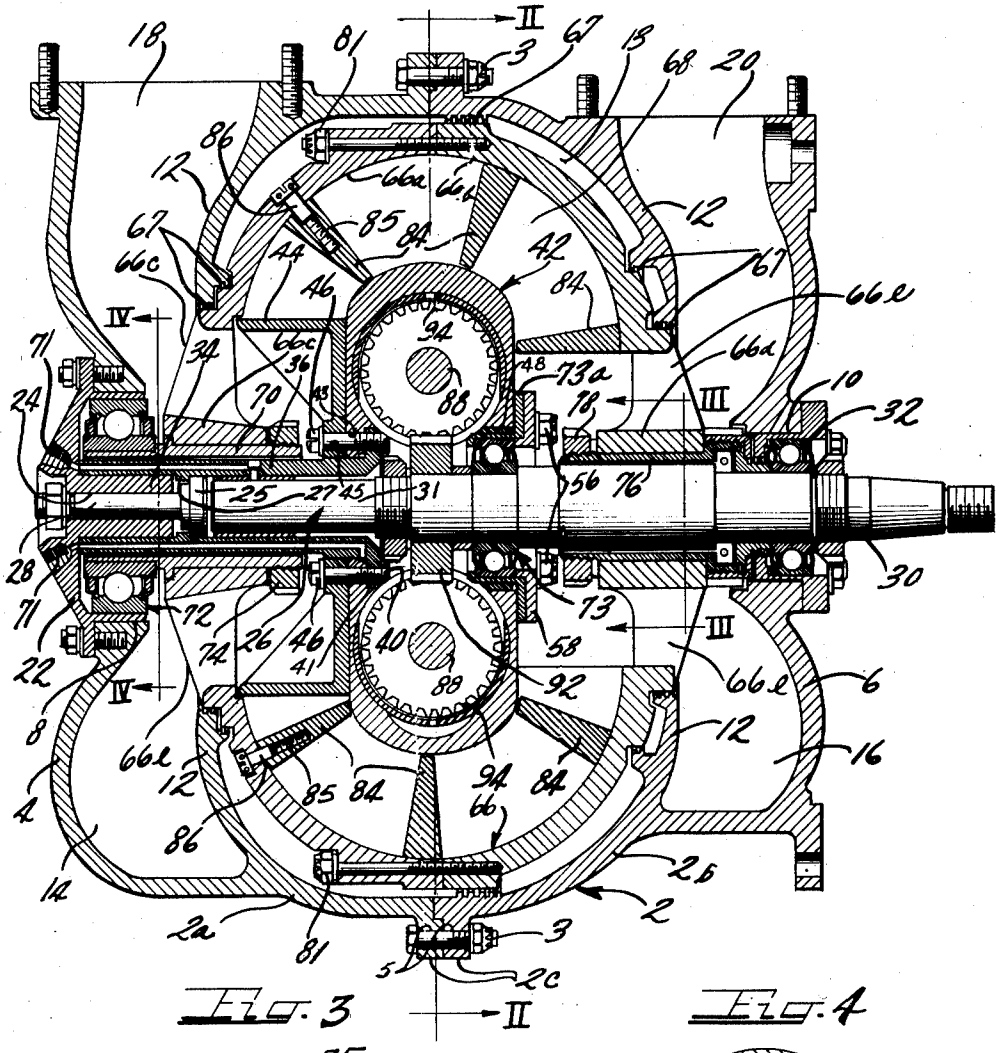


Fig. 3

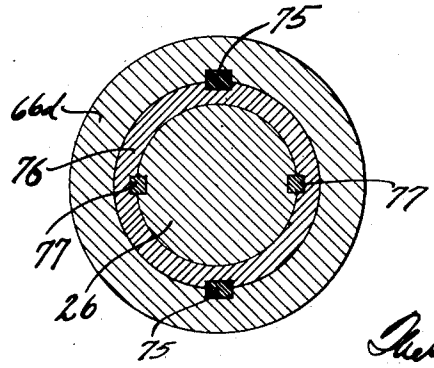
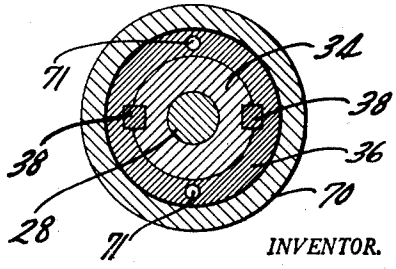


Fig. 4



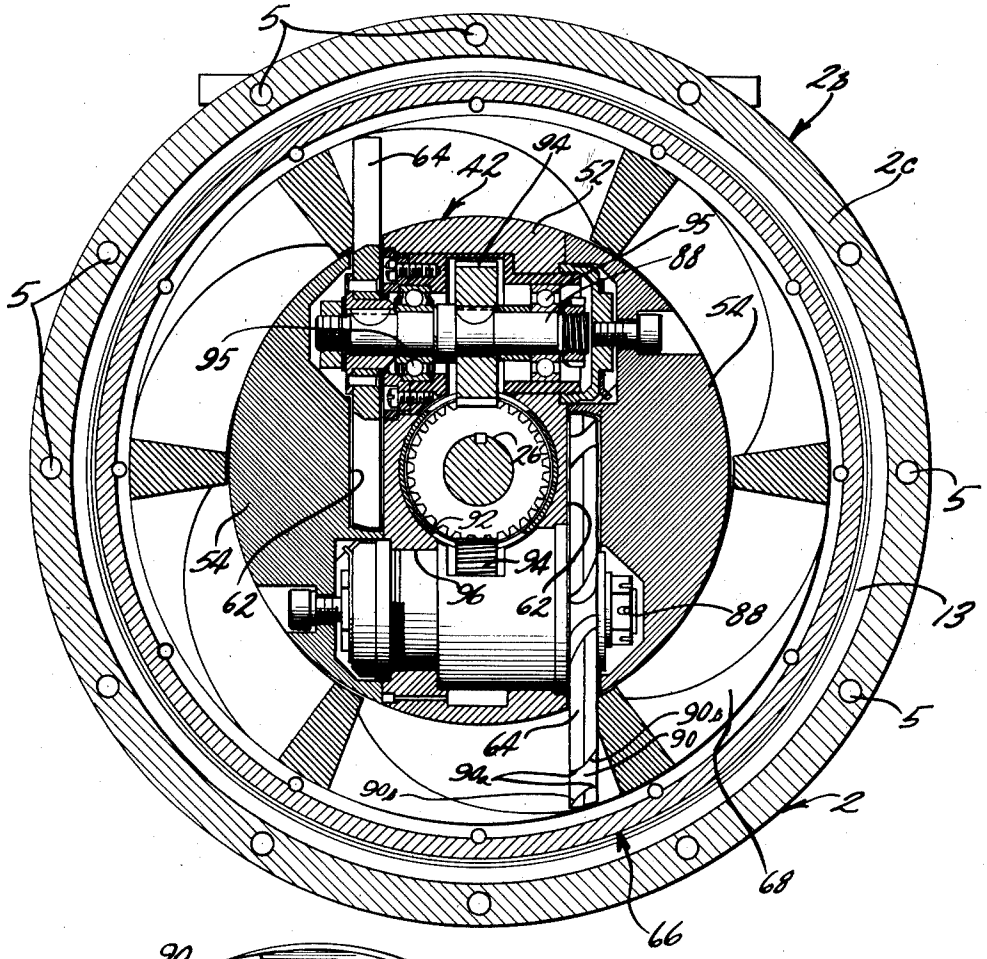
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March 7, 1950 A. E. BIERMANN 2,500,143

ROTARY ABUTMENT COMPRESSOR

Filed Sept. 26, 1946 9 Sheets-Sheet 2

Fig. 2



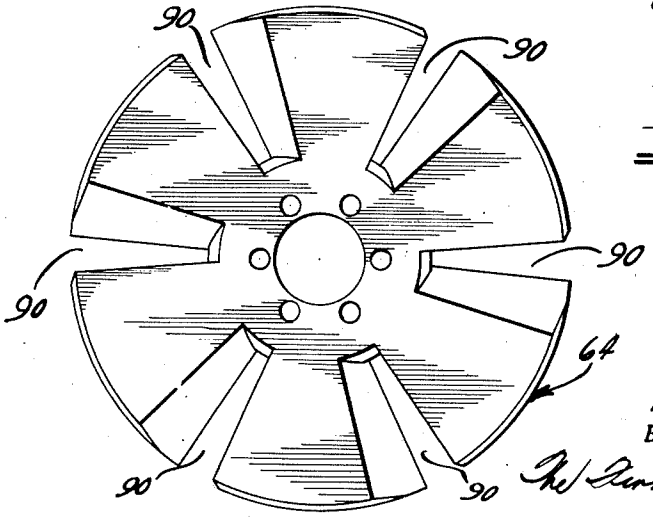


Fig. 7

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ROTARY ABUTMENT COMPRESSOR

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Fig. 5

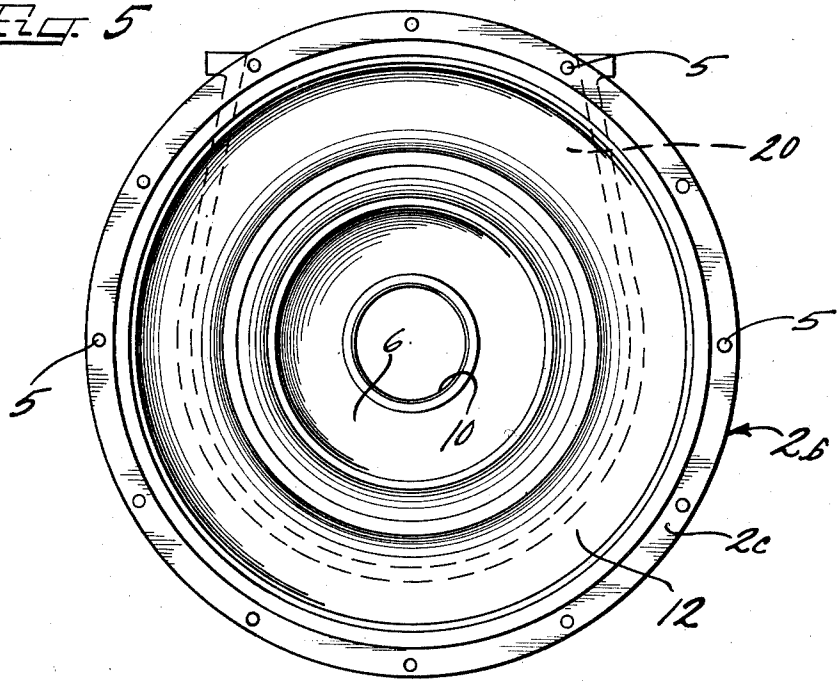
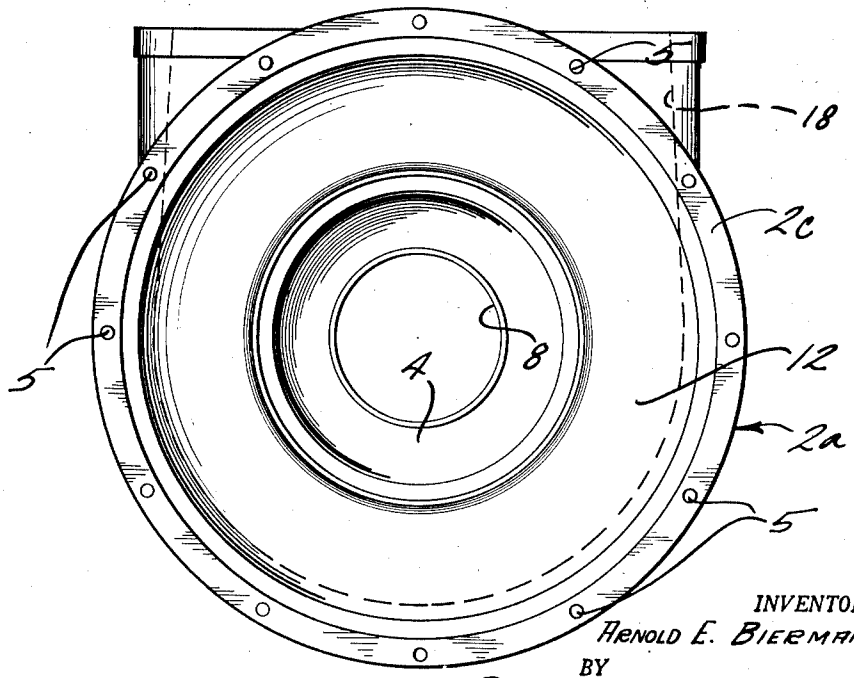


Fig. 6



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March 7, 1950 A. E. BIERMANN 2,500,143

ROTARY ABUTMENT COMPRESSOR

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Fig. 8

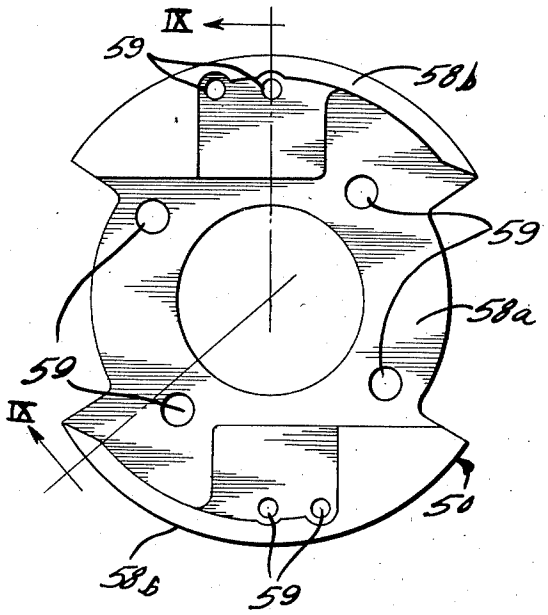
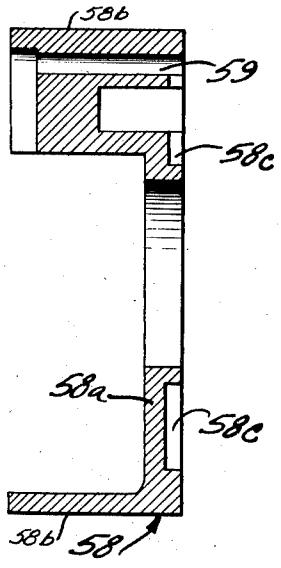


Fig. 9



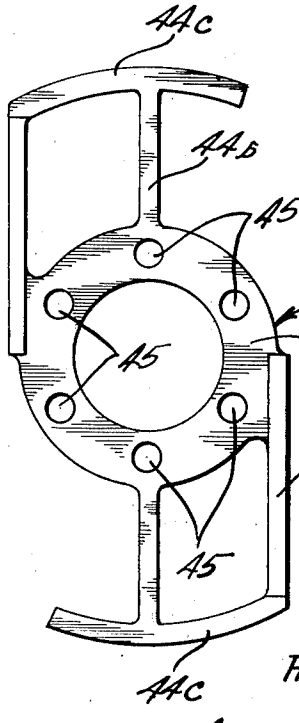


Fig. 10

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March 7, 1950 A. E. BIERMANN 2,500,143

ROTARY ABUTMENT COMPRESSOR

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Fig. 11

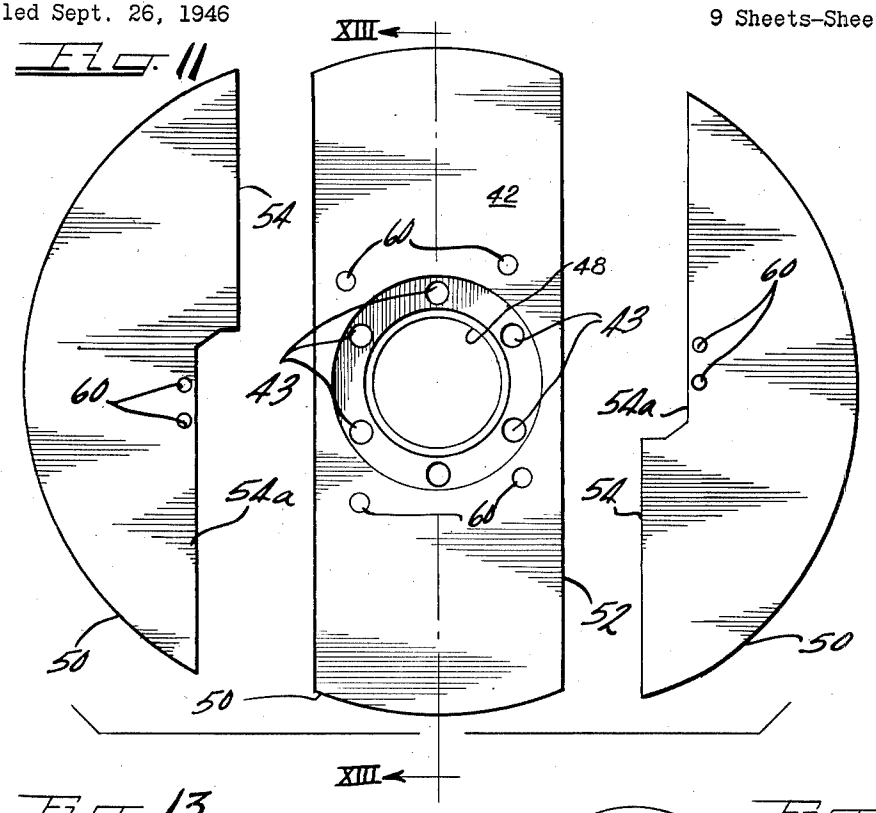


Fig. 13

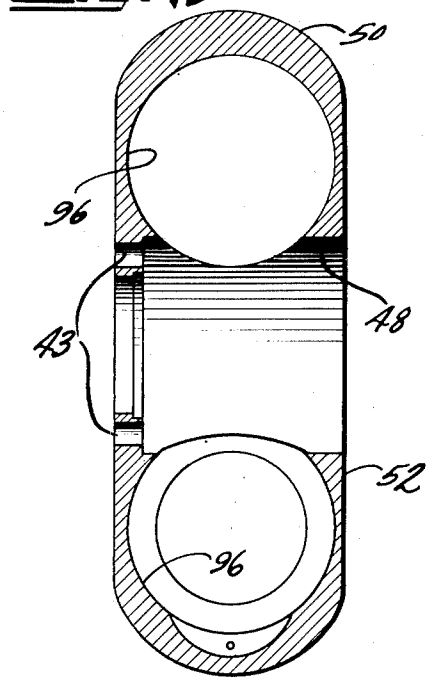
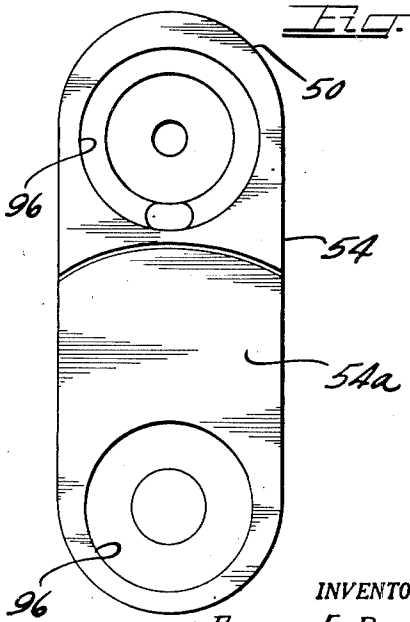


Fig. 12



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ROTARY ABUTMENT COMPRESSOR

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Fig. 20

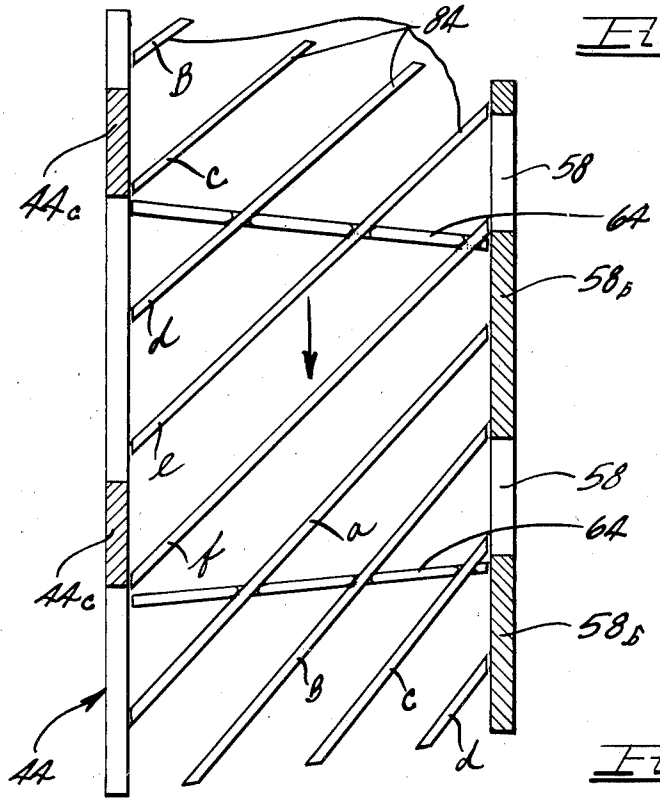
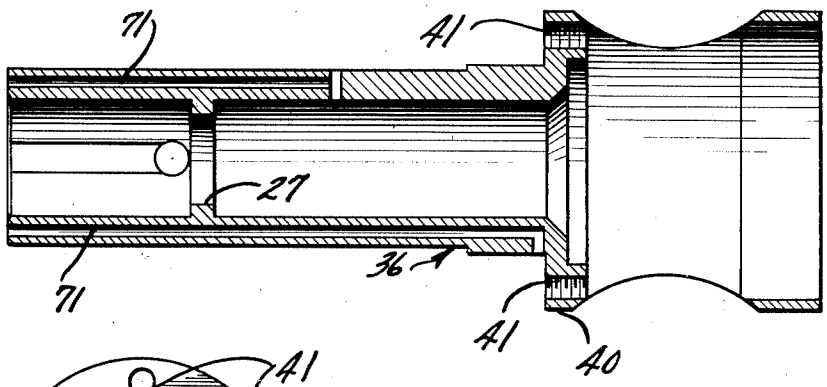


Fig. 14



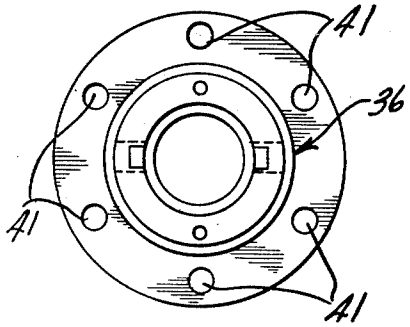


Fig. 15

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ROTARY ABUTMENT COMPRESSOR

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Fig. 16

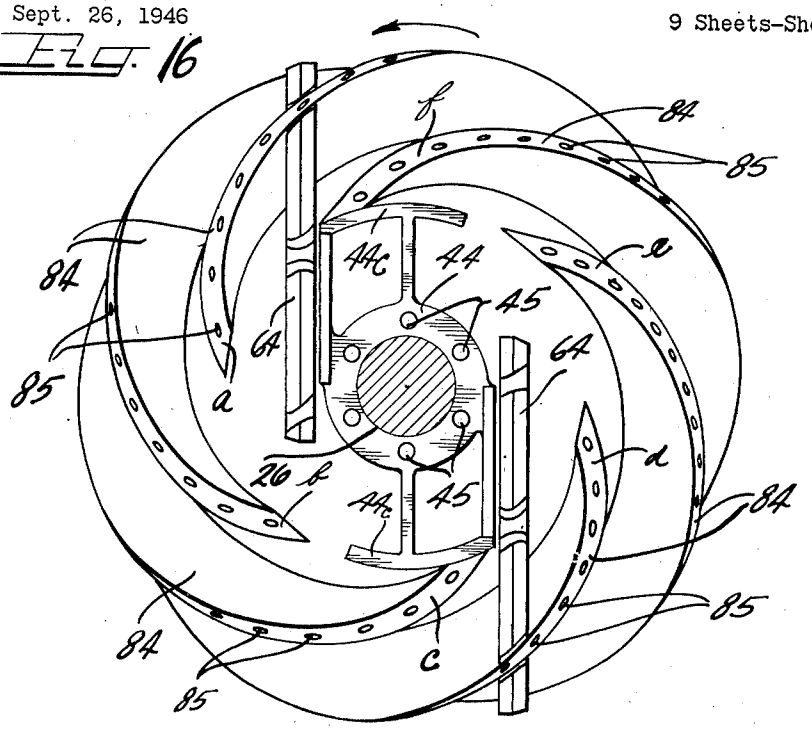
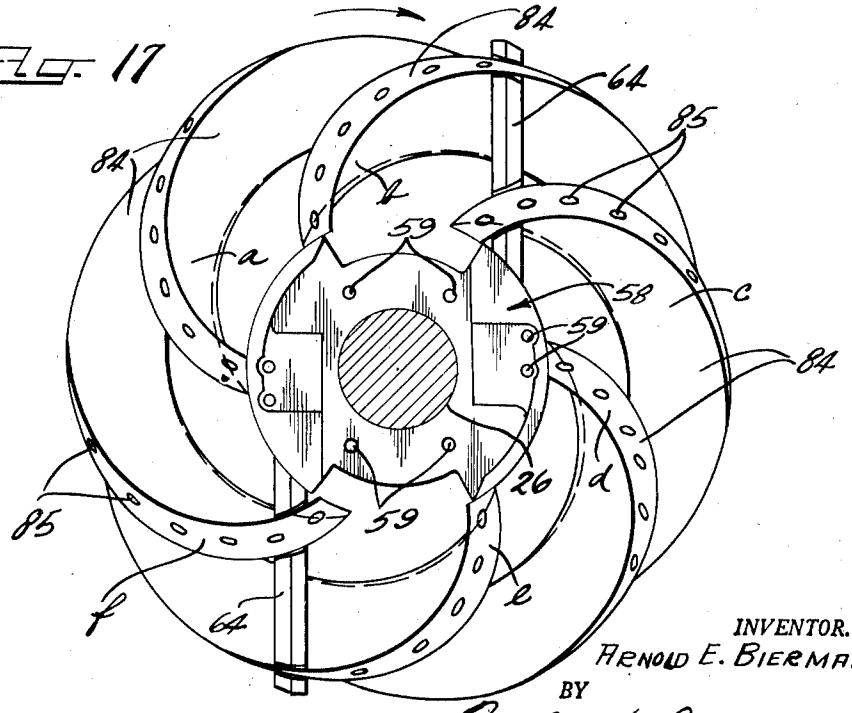


Fig. 17



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ROTARY ABUTMENT COMPRESSOR

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Fig. 19

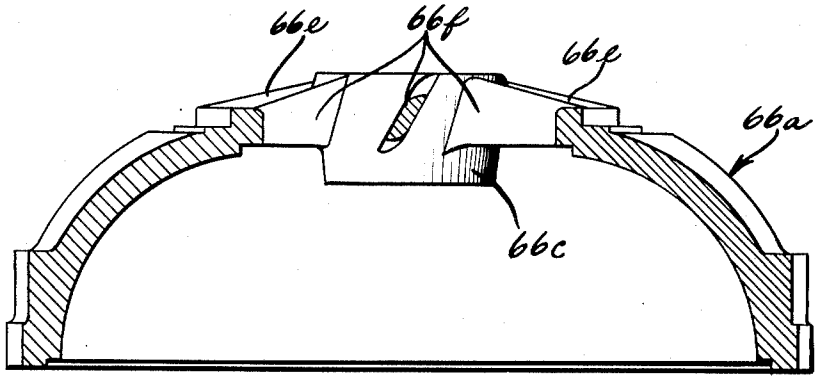
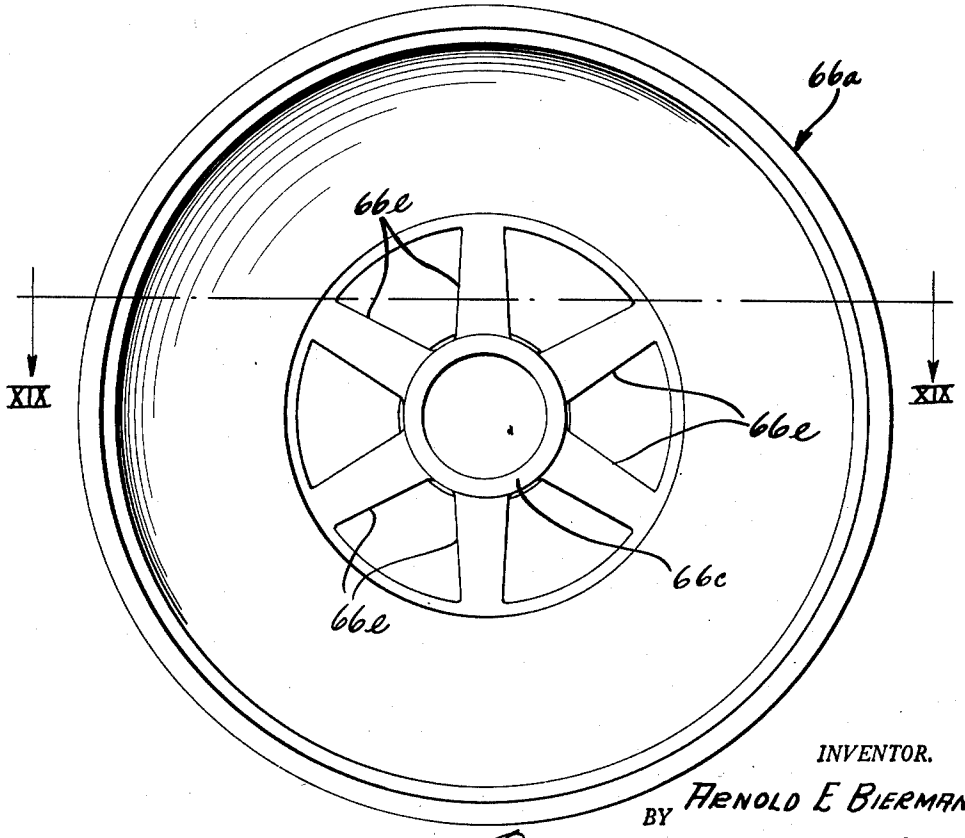


Fig. 18



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ROTARY ABUTMENT COMPRESSOR

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Fig. 21

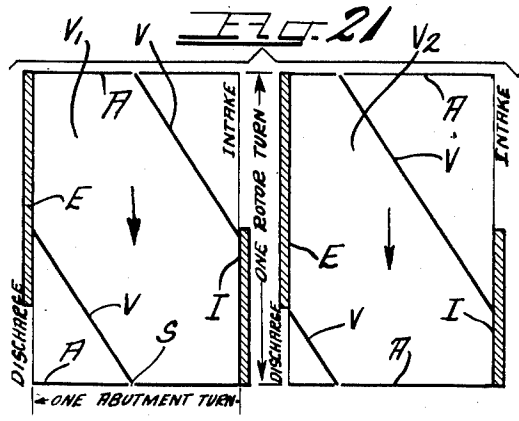


Fig. 22

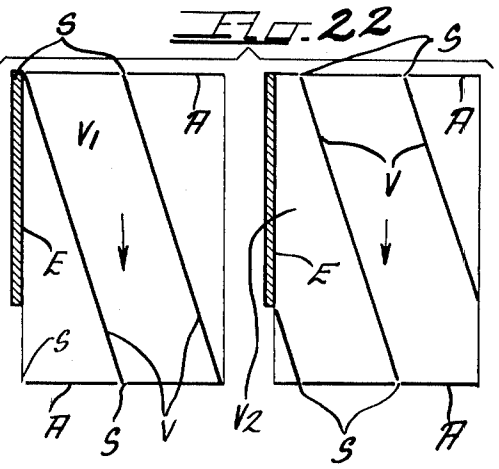


Fig. 23

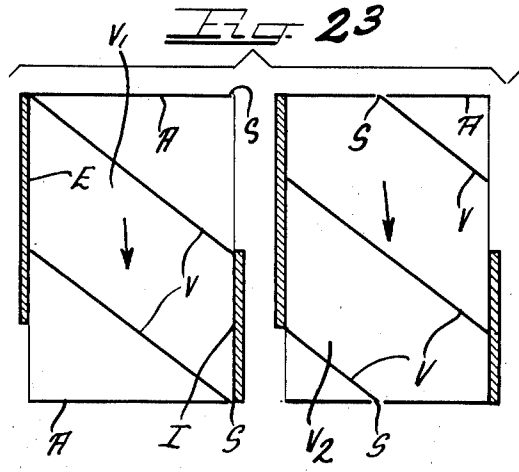


Fig. 24

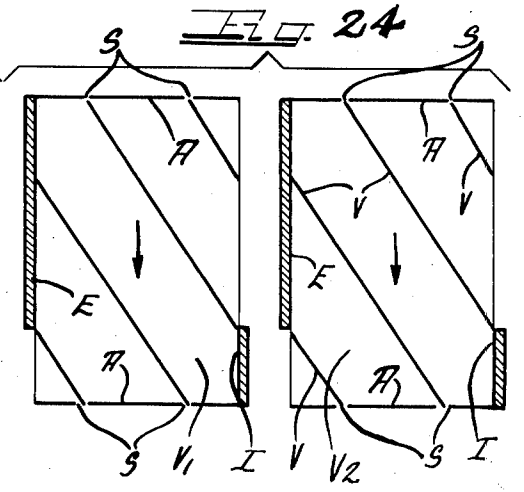
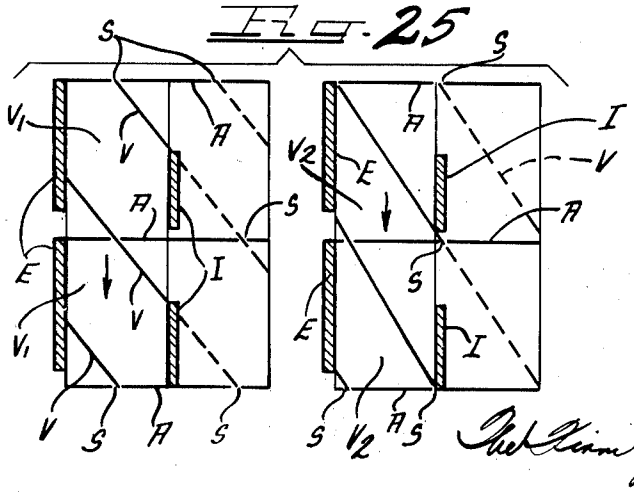


Fig. 25



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Patented Mar. 7, 1950

2,500,143

UNITED STATES PATENT OFFICE

2,500,143

ROTARY ABUTMENT COMPRESSOR

Arnold E. Biermann, Rocky River, Ohio

Application September 26, 1946, Serial No. 699,457

19 Claims. (Cl. 230—150)

1

This invention relates to a fluid compressor or a power generating fluid engine of the rotary type.

More particularly, this invention relates to an improved form of rotary fluid compressor or engine of the type wherein a rotor element cooperates with a stator element to define an annular chamber and helical vanes are mounted on the rotor element to traverse such chamber and to co-operate with the slots of peripherally slotted barrier members mounted on the stator element to define compartments variable in volume as the rotor rotates relative to the stator. As to certain features, this application constitutes a continuation-in-part of my copending application Serial No. 424,335, filed December 24, 1941, now Patent No. 2,411,707, issued November 26, 1946.

It is an object of this invention to provide an improved compressor or fluid operated engine characterized by unusual simplicity and ease of manufacture of its components and by the high efficiency and unusual performance characteristics of its operation.

A further object of this invention is to provide an improved rotary compressor of the general class referred to above, wherein the rotor element of the compressor surrounds the stator element to define an annular chamber therewith and the rotary barrier or abutment members are mounted in recesses in the stator periphery. This construction permits of a direct gear drive between the rotor shaft and the rotary abutments and has the further advantage of substantially decreasing the over-all size of the machine over constructions heretofore known for a given volume of the annular displacement chamber.

A particular object of this invention is to provide an improved stator construction for a rotary compressor or fluid engine wherein the stator constitutes an assemblage of the central body portion and peripheral portions, such components, when assembled, defining peripheral recesses in the stator suitable for mounting of the rotary abutment members therein.

Still another object of this invention is to provide an improved construction for mounting the stator and rotor elements of a rotary compressor or fluid engine within a housing, characterized by the mounting of the stator element in surrounding relationship to the main power shaft and the further mounting of the rotor element in surrounding relationship to the stator element.

A specific object of this invention is to establish, for a rotary compressor or fluid engine of the above referred to class, the optimum relationships between relative speeds of rotation of the rotor and the rotary abutments, the helical angle of the rotor vanes, and the number of rotary abutments, rotary abutment slots, and rotor vanes, to achieve the most desirable performance characteristics at highest efficiency.

2

The specific nature of the invention as well as other objects and advantages thereof will become apparent to those skilled in the art from the following detailed description of the annexed sheets of drawings which, by way of preferred example only, illustrate one specific embodiment of the invention.

On the drawings:

Figure 1 is a sectional view of an assembled rotary compressor embodying this invention, taken along a plane passing through the axis of the main power shaft;

Figure 2 is a sectional view taken along the plane II—II of Figure 1;

Figure 3 is an enlarged scale sectional view taken along the plane III—III of Figure 1;

Figure 4 is an enlarged scale sectional view taken along the plane IV—IV of Figure 1;

Figure 5 is a side elevational view of one half portion of the housing of the machine of Figure 1;

Figure 6 is a side elevational view of the other half portion of the housing of the machine of Figure 1;

Figure 7 is an elevational view of one of the slotted rotary abutment members;

Figure 8 is an elevational view of the discharge port defining member;

Figure 9 is a sectional view taken on the plane IX—IX of Figure 8:

Figure 10 is an elevational view of the intake port defining member;

Figure 11 is an exploded elevational view illustrating the major components from which the stator of the machine in Figure 1 is assembled;

Figure 12 is a side elevational view of one of the peripheral components of the stator assembly;

Figure 13 is a sectional view of the central stator component taken along the plane XIII—XIII of Figure 11;

Figure 14 is a longitudinal sectional view of the sleeve element by which the stator element is supported in the housing:

Figure 15 is an end elevational view of the sleeve element;

Figure 16 is an elevational view, with parts omitted for clarity, illustrating the co-operation of the rotor vanes, the rotary abutments, and the

intake port defining member, as viewed from the intake side of the main power shaft;

Figure 17 is an elevational view, with parts omitted for clarity, showing the relationship between the rotor vanes, the rotary abutments, and the exhaust port defining member as viewed from the exhaust side of the main driving shaft;

Figure 18 is an elevational view of one of the rotor halves;

Figure 19 is a sectional view taken on the plane XIX—XIX of Figure 18;

Figure 20 is a schematic developed view illustrating the operation of the machine of Figure 1;

Figures 21 through 25 are schematic developed views of various rotary compressors of the class to which this invention relates illustrating the effect of various combinations of rotary abutment-rotor speed ratios, helical angle of the vanes and number of abutments, slots and vanes.

As shown on the drawings:

In the description of the specific embodiment of this invention illustrated in the drawings, such machine will be referred to as a compressor. However, it will be understood by those skilled in the art that the principles of this invention are equally applicable to a fluid operated rotary engine for producing rotation of a power shaft in response to the application of fluid pressure to the machine.

A rotary compressor embodying this invention comprises a housing **2** which may be conveniently formed by bolting together two half portions **2***a* and **2***b* respectively by bolts **3** which pass through holes **5** in circular flanges **2***c* provided respectively adjacent the mating faces of each of the housing half portions. The opposed end walls **4** and **6** of the housing **2** respectively define aligned bearing apertures **8** and **10** for a purpose to be described. Each of the housing halves is provided with internally projecting wall portions **12** which co-operate to define a pumping chamber **13** which is preferably concentric about the axis of the bearing apertures **8** and **10**. Wall portions **12** are respectively spaced inwardly from the end walls **4** and **6** and define therebetween inlet and discharge chambers **14** and **16** respectively. A suitable radial opening **18** is provided to communicate with inlet chamber **14** and likewise a radial opening **20** communicates with discharge chamber **16** and permits the machine to be connected to suitable fluid conduits (not shown).

A cap **22** is suitably bolted in the bearing aperture **8** on the inlet side of the housing **2** and provided with a stem portion **34** having an axial bore **24** which receives a bolt **28**. A stator supporting sleeve **36** surrounds stem portion **34** and is keyed thereto by keys **38** (Fig. 4). The head **25** of bolt **24** engages an integral inwardly projecting flange **27** in the bore of stator sleeve **36** and secures such sleeve to cap **22** against axial movement.

A power shaft **26** has its one end **31** journalled in the inner end of sleeve **36** adjacent bolt head **25**. The other end **30** of shaft **26** projects outwardly through housing wall **12** and is journaled in bearing aperture **10** by a ball bearing assembly **32**.

Sleeve **36** is provided at its inwardly projecting end with a radial flange portion **40**, and a stator element **42** as well as an inlet valve member **44** are rigidly mounted on sleeve **36** (Figure 1) by a plurality of bolts **46** which pass through holes **45** in inlet valve **44**, holes **43** in stator **42** (Figure 11) and threadably engage in suitable holes **41** in flange **40** (Figure 14).

As is best shown in Figures 11, 12 and 13, stator element **42** comprises a circular member with a large axial bore **48** therethrough and having a circular periphery **50** which is arcuately shaped in cross section. Stator element **42** may be conveniently manufactured by utilizing a generally doughnut-shaped body portion **52** to which are secured a plurality of segmental peripheral portions **54**. As will be shown, the various components of stator element **42** are secured in assembled relationship by suitable bolts **56** (Figure 1) which pass through holes **59** in an exhaust port defining member **58** (Figure 8) and engage in threaded holes **60** in stator body portion **52** and peripheral portions **54**.

Each of the peripheral portions **54** are identical. Along the chordal edge of peripheral portions **54** at which such members are joined to body portion **52**, the peripheral portions **54** are cut away as indicated at **54***a* so that when assembled to body portion **52**, the stator element **42** is effectively provided with oppositely extending, non-radial recesses **62** (Figure 2) in which rotary abutment members **64** may be rotatably mounted in a manner that will be described.

A rotor **66** is provided which is mounted in the concentric chamber **13** of housing **2** in such manner as to co-operate with the periphery of stator element **42** to define an annular generally toroidal recess **68**. Specifically, the inside contour of rotor **66** is such as would be generated by revolution of the outer half of a rotary abutment **64** about shaft **26**. Rotor **66** is preferably formed from two half portions **66***a* and **66***b* which are joined by suitable bolts **81** along a radial plane. The rotor half portion **66***a* which is disposed on the inlet side of housing **2** includes a spider-supported, integral bearing hub portion **66***c* which is secured on a bearing sleeve **70** which in turn freely surrounds the external cylindrical surface of stator sleeve **36**. Bearing sleeve **70** is rotatably journaled in bearing aperture **8** of housing **2** by a suitable ball bearing unit **72**. The rotor hub portion **66***c* is secured onto bearing sleeve **70** by a nut **74**. The rotor half portion **66***b* which is disposed adjacent the discharge side of housing **2** is provided with an integral spider-supported hub portion **66***d* which is secured to a bearing sleeve **76** by a nut **78** and keys **75**. Bearing sleeve **76** is in turn keyed to the main power shaft **26** by keys **77** (Figure 3). It should be particularly noted that the periphery of stator element **42** is completely enclosed by rotor **66** and that the rotor hub portions **66***c* and **66***d* are disposed on opposite sides of stator element **42**.

Lubrication of the various bearings associated with the shaft and rotor is accomplished in any conventional manner, such as by the provision of axially extending oil passages **71** in the stator sleeve **36** which communicate with suitable radial passages to supply lubricant to shaft **26**. An additional ball bearing unit **73** is provided in the stator bore **48** to support the central portion of power shaft **26** and to axially position the rotor **66** with respect to the stator **42**.

Secured to the inside surface of rotor element **66** in equispaced relationship are a plurality of generally helical vanes **84**. Vanes **84** are of generally wedge-shaped cross section and disposed with their wide face portion abutting the inner surface of rotor **66**. Vanes **84** may be conveniently secured to rotor **66** by bolts **86**

which pass through suitable holes in rotor 66 and thence into threaded holes 85 in vanes 84.

By reference to Figures 16 and 17, it will be seen that vanes 84 are preferably of both a spiral and helical configuration. Each individual vane starts at the inlet side of the rotor and moves in a generally helical direction around the rotor for approximately 180 degrees, terminating at the discharge side of the rotor. In particular, the vanes are of such shape as would be generated between two surfaces of revolution when such surfaces of revolution revolve in a plane substantially parallel to the axis of the rotor and simultaneously revolve about the axis of the rotor, and with the axes of such surfaces of revolution angularly displaced one from the other. The small ends of the wedge-shaped blades thus lie closely adjacent to the external surface of stator element 42 and, in effect, maintain a substantially sealing engagement with the stator surface to minimize leakage of fluid therethrough. Likewise, the internal surface of housing 2 which surrounds rotor 66 is provided with a plurality of sets of sealing grooves 67 which reduce fluid leakage around the rotor to a negligible amount.

As was heretofore mentioned, a plurality of rotary abutments 64 are mounted in the peripheral recesses 62 provided in stator element 42. The rotary abutments 64 constitute generally disk-shaped barrier members which are rotatably mounted on shafts 88 suitably supported in chordal relationship in the interior of stator element 42. The peripheral portions of rotary abutments 64 traverse the annular toroid-like space 68 defined between rotor element 66 and stator element 42. To permit passage of the vanes 84 through the rotary abutments 64, the rotary abutments 64 are provided with a plurality of radial slots 90 which are suitably shaped to snugly receive the vanes 84 respectively as the rotor rotates.

The opposed edges of the slots 90 are preferably formed as an arc of a surface of revolution, indicated at 90*a* in Figure 2, which connect with plane surfaces 90*b* tangent to the surfaces of revolution 90*a*. In addition, the axes of the surfaces of revolution on either side of the slots are displaced toward the side of the rotary abutments which bears an obtuse angle with respect to the spiral vane passing through the respective slot. This construction permits using similarly shaped milling cutters or grinding wheels in the manufacture of the spiral vanes. Obviously the vanes must be generated by moving a cutting tool through the same path followed by a slot.

As was heretofore mentioned, an inlet valve member 44 is secured to stator element 42 on the inlet side thereof while an exhaust valve member 58 is secured to stator element 42 on the exhaust side. Referring particularly to Figure 10, it will be observed that inlet valve 44 comprises a hub portion 44*a* which surrounds stator sleeve 36 and carries suitable holes 45 for the bolts 46 by which the inlet valve 44 and stator element 42 are secured to stator sleeve 36. Extending from hub portion 44*a* are arm portions 44*b* which support arcuate segment portions 44*c* which co-operate with the vanes 84 to provide an inlet valve controlling action. Similarly, the outlet valve member 58 comprises an annular hub portion 58*a* which surrounds the rotor shaft 26 and carries the bolt holes 59 to receive the bolts 56 which not only secure the exhaust valve 58 to the stator element 42 but in addition serve to secure the peripheral portion 54 to the body

portion 52 of the assembled stator element. Hub portion 58*a* has formed thereon opposed arcuate segmental surfaces 58*b* which co-operate with the vanes 84 to provide an outlet valve control action. In addition, the inner face of exhaust valve member 58 is provided with an annular recess 58*c* to accommodate a flange portion of a bearing cage retainer member 73*a* (Figure 1) which co-operates with a central ball bearing unit 73.

It should be noted that the angular distribution of the inlet valve arcuate surfaces 44*c* and the outlet valve arcuate surfaces 58*b* is such as to provide a successive arrangement of inlet and outlet valve surfaces around the axis of the rotor. Furthermore, in the illustrated example, the angular extent of the arcuate inlet valve surfaces 44*c* is substantially equal to the angular displacement of successive vanes 84, while the angular extent of the arcuate outlet valve surfaces 58*b* is substantially equal to twice the angular spacing of successive vanes 84. Generally speaking, the angular distribution of the arcuate surfaces of the inlet and outlet valves is determined by the speed and compression ratio for which the machine is designed. Those portions of the inlet and outlet valve members which lie in between the arcuate valve surfaces thereof are sufficiently cut away so as to provide unimpeded fluid passage therethrough into the particular rotor compartment between the vanes 84 which happens to be aligned therewith.

In Figure 16 there is shown in elevation, the assembled relationship of the inlet valve member 44, the rotary abutments 64 and the rotor vanes 84, while in Figure 17 there is shown the assembled relationship of the exhaust valve member 58 with respect to the rotary abutments 64 and the rotor vanes 84. For convenience in identification, letters *a* through *f* have been applied to corresponding vanes in these two views. The rotor and stator elements have not been shown for purposes of clarity.

From the construction thus far described, it is apparent that when rotor 66 is rotated, the rotary abutments 64 must rotate in fixed relationship therewith due to the fact that the rotor vanes 84 are threading through the slots 90 of the rotary abutments. I preferably provide a positive driving mechanism for rotating the rotary abutments 64 and hence maintain the rotation of such elements independent of frictional contact with the rotor vanes 84. In the construction embodying this invention, such positive drive is conveniently accomplished by the provision of a main driving gear 92 which is keyed to rotor shaft 26 within the hollow central portion of the stator element 42. Co-operating gears 94 are keyed to each of the shafts 88 which support the rotary abutments 64. Shafts 88 are preferably supported in ball bearing units 95 within a transverse bore 96 (Figures 12 and 13) provided within the body portion 52 and adjacent peripheral portion 54 of the stator assembly.

Referring now particularly to Figure 20, there is shown a schematic, developed view illustrating the co-operation of the rotor vanes 84, the rotary abutments 64, and the inlet and outlet valve members 44 and 58 respectively. The operation of a machine embodying this invention will be apparent from a study of this diagram. As the rotor rotates, the rotor vanes *a* through *f* respectively move downwardly as viewed in Figure 20. Since the opposite end faces of the vanes 84 are respectively disposed closely adjacent the interior wall

of rotor 66 and the external periphery of stator element 42, it follows that fluid compartments are defined between each pair of adjacent vanes. By virtue of the provision of the rotary abutments, such fluid compartments vary in volume as the rotor rotates.

For example, the compartment defined between vanes *e* and *f*, hereinafter referred to as compartment *e—f*, is, in the position shown in Figure 20, about to be sealed off by the arcuate sealing surface 44*c* of the inlet valve 44. Since further movement of the rotor brings the vane *f* into traversing relationship with one of the slots 90 of the rotary abutment 64, the compartment *e—f* is gradually reduced in volume as the rotor rotates. Such compartment remains substantially enclosed except for a slight amount of leakage through the junctures of the various relatively moving surfaces defining the compartment. As a result, when the compartment *e—f* moves into the position shown as that occupied by compartment *a—b* of Figure 20, the volume of the compartment has been substantially reduced. Accordingly, it follows that the fluid trapped therein has been subjected to compression approximately proportional to the reduction in volume of the compartment. Further movement of the rotor will bring the trailing edge of vane *f* off the arcuate sealing surface 58*b* of the discharge valve 58 and hence permit the fluid contents of the compartment *e—f* to discharge into the discharge sump portion 16 of housing 2. It should be further noted that the co-operation of vane *e* with the rotary abutment has the effect of sweeping substantially all of the fluid initially contained in compartment *e—f* into the discharge sump.

The performance of the aforedescribed compressor may be further improved by shaping the spider arms which support the main portion of the rotor half 66*a* or 66*b* on the hubs 66*c* or 66*d* respectively to function as propellers. As best shown in Figures 18 and 19, the spider arms 66*e* are provided with propeller-like surfaces 66*f* to produce a further pumping action on the fluid. In the case of the inlet rotor half 66*a*, the surfaces 66*f* force fluid into the rotor compartments while on the outlet rotor half 66*b* the surfaces 66*f* withdraw fluid from the rotor compartments.

It should be noted that the unique arrangement provided by this invention wherein the rotor surrounds the stator and in effect defines a toroid-like pumping chamber about the stator permits the design of a machine of greatly increased fluid capacity for the same overall dimensions as compared with constructions heretofore known. By locating the pumping chamber near the periphery of the machine, a greatly increased volume of such chamber is obtained relative to the overall displacement of the machine. Furthermore, the utilization of a generally toroidal pumping chamber further increases the available fluid displacement volume without increasing the diameter of the entire machine.

In a machine of the class described, it is apparent that the performance characteristics are dependent upon a large number of variables. It is therefore extremely desirable to obtain an optimum combination of the number of helical vanes, the number of rotary abutments, the total number of rotary abutment slots, and the ratio of rotary abutment rotational speed to rotor rotational speed. From a performance standpoint, maximum flow capacity combined with minimum leakage and port losses is desired. Minimum leakage losses are obtained when a few vanes, slots and rotary abutments are used. On the other hand, flow volume increases rapidly when the number of rotary abutments employed is increased, provided the vanes are also increased in number. The relationship of the aforementioned variables, and the determination of an optimum combination of such variables, can best be understood by referring to schematic developed diagram of machines embodying various combinations of such variables, such as shown in Figures 21 through 25.

Figure 21 shows a schematic development of the most fundamental arrangement comprising a compressor having one rotary abutment A with one slot S and one vane V. The inlet valve sealing surface is represented by I and the exhaust valve sealing surface by E. In all of Figures 21 through 25, the left-hand diagram represents the position of the rotor when one of the compartments defined by the rotor vane is at its maximum volume V_1. The diagram on the right-hand side of Figure 21 shows the same compartment's size when the rotor has advanced to the point where the trailing end of the vane V is about to pass off the sealing surface E of the exhaust valve. The volume of the compartment at this time is represented by V_2. In order to obtain maximum efficiency, it is desirable to compress the air to the final discharge pressure before opening the discharge port. The ratio of the volumes V_1/V_2 thus constitutes a measure of the compression ratio possible for a machine utilizing such a combination of variables. For the present study, the discharge port area has arbitrarily been assumed to cover 25% of the wall on that side of the structure and this proportioning of the discharge port area has been carried through throughout all combinations of variables illustrated in Figures 21 through 25.

Simple calculations will show that for the fundamental arrangement illustrated in Figure 21, the ratio V_1/V_2, and hence the maximum compression ratio possible, is approximately 1.09. This indicates that such a compressor would be efficient only at very low pressure ratios.

Another criterion of compressor performance is the effective displacement per revolution of that part of the compressor that limits the compressor speed. The particular type of compressors under consideration are limited in speed by stresses in the rotor. The utilization of the available volume of the pumping chamber in the compressor as effective displacement may be conveniently expressed in terms of a displacement factor. Such a displacement factor is defined as the ratio of the volume displacement effected per revolution of the rotor to the volume available for displacement, which generally is the volume of the annular pumping chamber if the relatively small volumes of the vanes, abutments, etc., are disregarded. Considering the fundamental arrangement of Figure 21, it is apparent that such an arrangement yields a displacement factor of 1, inasmuch as substantially all of the available volume of the annular pumping chamber is swept toward the discharge side of the compressor once during each revolution of the rotor.

The first of the variables to be considered is the pitch of the rotor vanes. It is believed obvious that the angle of inclination of the rotor vanes V must be definitely correlated with the relative rotational speed of the rotor with respect to the rotary abutments and with the number of vanes and slots. In the arrangement shown in Figure 21, a 1 to 1 ratio of rotor speed to rotary

abutment speed is utilized and the pitch of the vane there employed requires only one vane and one slot for operation of the machine. Referring now to Figure 22 wherein the vanes are illustrated as having an extremely low pitch such that the single vane traverses the rotor twice, it is then necessary to have a ratio of rotary abutment speed to rotor speed of .5 and two slots S must be provided in the single rotary abutment A. In this construction, two complete turns of the rotor are required to sweep the available displacement volume into the discharge side of the compressor, and the displacement factor is reduced to .5. The pressure ratio of V_1/V_2 becomes 1.39.

Figure 23 illustrates an opposite extreme condition of rotor vane pitch and speed ratio of the rotary abutment with respect to the rotor. In the arrangement of Figure 23, a high pitch of the rotor vanes is utilized and a ratio of rotary abutment rotational speed to rotor rotational speed of 2.0 is provided. With such an arrangement two individual vanes V are required and one rotary abutment slot S. While the displacement factor is again unity and the pressure ratio of V_1/V_2 is 1.14, such slight improvement is not justification for use of such proportioning when compared with the arrangement of Figure 21. It is obviously undesirable to utilize twice the number of vanes and rotate the rotary abutments at twice the speed to utilize the arrangement of Figure 23 in place of the arrangement of Figure 21. The pitch of the vanes and the speed ratio employed in the arrangement of Figure 21 is therefore an optimum arrangement.

As will be recalled, the arrangement of Figure 21 utilized a ratio of rotary abutment speed to rotor speed of 1 to 1. Accordingly, the optimum angle of pitch of the vane can be defined as that resulting when the angle of turn of the rotary abutment in traversing one vane equals the angle of turning of the rotor. When a gear drive is utilized between the rotor and the rotary abutments, such as that heretofore described, a further premium is derived from utilizing a ratio of rotary abutment speed to rotor speed of 1 to 1. Reference to Figures 1 and 2 will readily show that the rotary diameter of the abutment gears controls the size of the stator assembly and therefore is an important factor in determining the maximum possible displacement volume for a machine of given size. Furthermore, the rotor gear determines the extent of displacement of the rotary abutment from the center of the stator element. An off center position causes distortion of the vanes and indirectly causes a loss in displacement volume. It is also important to obtain as great a displacement between shaft centers as possible for a given size of gear. This is most conveniently accomplished when all the gears are the same diameter, which is the condition which results when a 1 to 1 ratio of rotary abutment speed to rotor speed is employed.

A comparison of the arrangement of Figure 24 with that of Figure 21 indicates how the pressure ratio of machines may be conveniently increased by adding vanes. In the arrangement shown in Figure 24 the optimum pitch angle of the vanes is employed as well as the optimum speed ratio of rotary abutment to rotor of 1 to 1. However, two vanes V are utilized and a single abutment A having two slots S. While the displacement factor remains at unity with such

an arrangement, it is readily observable that the pressure ratio, or V_1/V_2 is substantially increased over that available from the arrangement of Figure 21. Specifically, V_1/V_2 in this arrangement equals 1.75.

It will be apparent to those skilled in the art that further increases in the number of vanes will increase the pressure ratio of V_1/V_2 up to a theoretical limit of 4 when a 25% discharge port opening is assumed. A smaller discharge opening will permit still greater volume ratios to be obtained.

In the arrangements illustrated in Figures 21 through 24, it was assumed for simplicity that the vanes extend completely around the hypothetical toroid which would be defined by rotation of the rotary abutments about the axis of the rotor. In a practical design of a machine, the arcuate extent of the vanes in the plane of a rotary abutment cannot be much more than 180 degrees and the illustrated embodiment of the invention such arcuate extent is actually somewhat less than 180 degrees. The effect of this factor on the schematic diagrams heretofore considered is illustrated in the diagrams of Figure 25 wherein the rotor vanes V are shown to extend merely for 180 degrees of the angular extent of the rotary abutments and the hypothetical position of the remaining 180 degrees of the rotor vanes which is not actually utilized in a practical embodiment of the machine is illustrated by the dotted line extensions. The optimum relationship heretofore indicated for ratio of rotary speed to rotor speed and vane pitch angle are maintained identical to that utilized in the arrangement of Figure 21.

The arrangement of Figure 25 also illustrates the effect of the addition of annother abutment to the machine. Adding another abutment of course requires that an additional set of inlet and outlet valve surfaces be provided. In the arrangement of Figure 25, two vanes are provided and the ratio of rotary abutment speed to rotor speed is unity. It therefore follows that the total number of rotary abutment slots required for the machine is four. The pressure ratio or V_1/V_2 is equal to 1.09. The displacement factor, however, is increased to 2.0 due to the fact that the displacement volume of the machine is swept into the discharge side of the machine twice during each revolution of rotor by virtue of the fact that two rotary abutments are provided. In the specific embodiment of this invention heretofore described, which is represented schematically by Figure 20, the aforementioned optimum relationships are incorporated. Thus the ratio of rotary abutment speed to rotor speed is unity and two rotary abutments and six vanes are utilized. This number of vanes provides a pressure ratio of V_1/V_2 equal to 2.12 and of course the displacement factor is 2.0 by virtue of the utilization of two rotary abutments.

For purposes of the following discussion relating to optimum relationship of the variables involved, the letters T, H and Q will be used to denote the following dimensions in Figures 21 and 25:

T, one complete abutment turn.

H, that part of an abutment turn required to traverse one vane.

Q, that part of a rotor turn required to traverse one vane. In the foregoing description it has been shown that the vane angle was optimum when, H=Q. Furthermore, it was shown that the relationship of variables was optimum when

the ratio, abutment speed/rotor speed, which is hereafter denoted by the letter M, is equal to 1. From Figure 25 it will be observed that the abutment rotates through H/T fraction of a rotation as the rotor rotates from one abutment to the next. Therefore, for each rotation of the rotor the abutment will rotate AH/T revolutions, where A is equal to the number of abutments in the machine. Therefore,

$$M=AH/T=1 \quad (1)$$

and

$$A=T/H \quad (2)$$

These relationships state that an optimum condition exists when the number of abutments is equal to the reciprocal of that fraction of one abutment turn required to traverse one vane.

Further reference to Figure 25 reveals that the number of impeller slots, S, is a function of the number of vanes that actually pierce the abutments in addition to the vanes that would pierce the abutments if the vanes were extended around the abutment as shown by the dotted lines in the diagrams. The number of additional slots resulting from extension of the vanes, V, is a function of T/H, therefore,

$$S=VT/H \quad (3)$$

By substituting Equation 3 in Equation 1 we have,

$$M=AV/S=1 \quad (4)$$

or,

$$S=AV \quad (5)$$

Equation 5 states that an optimum condition of operation exists when the number of slots is equal to the product of the number of abutments and the number of vanes.

It will, of course, be understood that various details of construction may be varied through a wide range without departing from the principles of this invention, and it is, therefore, not the purpose to limit the patent granted hereon otherwise than necessitated by the scope of the appended claims.

I claim as my invention:

1. In a rotary compressor, a generally annularly shaped stator defining peripherally recessed portions therein, a rotor mounted for rotational movement relative to said stator, said rotor having a portion thereof surrounding said stator and cooperating therewith to define an encircling annular chamber, slotted barrier means mounted for rotation in said peripherally recessed portions of said stator on an axis transverse to the rotor axis and traversing said chamber, said barrier means conforming to the cross section of said chamber, and a generally helical vane secured to said rotor in radially inwardly projecting relation and disposed in said chamber, said vane cooperating with said stator and with the slot of said barrier means to define compartments variable in volume as said rotor rotates, and inlet and outlet port defining means successively communicating with said compartments as said rotor rotates.

2. In a rotary compressor, a stator, a rotor mounted for rotational movement relative to said stator, said rotor having a portion thereof surrounding said stator and co-operating therewith to define an annular chamber, slotted barrier means rotatably mounted in said stator and transversing said chamber, generally helical vanes secured to said rotor and disposed in said chamber, said vanes co-operating with said stator and with the slots of said barrier means to define compartments variable in volume as said rotor rotates, inlet and outlet port defining means successively communicating with said compartments as said rotor rotates, power means for rotating said rotor, and means for rotating said barrier means in fixed relation with said rotor, including a gear train having a driven connection with said power means and being housed by said stator.

3. In a rotary compressor, a stator, a rotor mounted for rotational movement relative to said stator, said rotor having a portion thereof surrounding said stator and co-operating therewith to define an annular chamber, slotted barrier means rotatably mounted in said stator and transversing said chamber, generally helical vanes secured to said rotor and disposed in said chamber, said vanes co-operating with said stator and with the slots of said barrier means to define compartments variable in volume as said rotor rotates, inlet and outlet port defining means successively communicating with said compartments as said rotor rotates, a shaft for driving said rotor, said shaft having a portion thereof passing through said stator, and a gear train housed in said stator drivingly connecting said rotating barrier means and said shaft, whereby said barrier means is rotated in fixed relation with said rotor.

4. In a rotary compressor, an annularly shaped stator having a curved peripheral surface, a rotor substantially in the form of an annulus having wall portions of generally semi-circular cross-sectional configuration and being mounted for rotational movement relative to said stator, said rotor surrounding said stator and co-operating therewith to define an annular chamber, said stator having a plurality of recesses extending inwardly from the periphery thereof, a circular disk-like barrier member rotatably mounted in each of said recesses on an axis transverse to said rotor axis and having a peripheral portion thereof conforming to and traversing said chamber, each of said barrier members having peripherally spaced slots therein, a plurality of generally helical vanes secured to said rotor in radially inwardly projecting relation and disposed in said chamber, said vanes being constructed and arranged to respectively traverse said barrier slots and to cooperate with said peripheral surface of said stator to thereby define compartments variable in volume as said rotor rotates, and inlet and outlet port defining means successively communicating with said compartments as said rotor rotates.

5. In a rotary compressor, a stator, a rotor mounted for rotational movement relative to said stator, said rotor having a portion thereof surrounding said stator and co-operating therewith to define an annular chamber, said stator having a plurality of non-radial recesses extending inwardly from the periphery thereof, a disk-like barrier member rotatably mounted in each of said recesses having a peripheral portion thereof traversing said chamber, each of said barrier members having peripherally spaced slots therein, a plurality of generally helical vanes secured to said rotor and disposed in said chamber, said vanes being constructed and arranged so as to respectively traverse said barrier slots and to co-operate with the periphery of said stator to define compartments variable in volume as said rotor rotates, a shaft for driving said rotor, said shaft having a portion thereof passing through said stator, and gearing mounted in

said stator and connecting said rotating barrier means and said shaft, whereby said barrier means are rotated in fixed relation with said rotor.

6. In a rotary compressor, a stator having a circular periphery and an axial bore, a shaft rotatable in said stator bore, a rotor secured to said shaft and having a portion thereof surrounding the periphery of said stator and co-operating with said stator to define an annular chamber, said stator having a plurality of non-radial recesses extending inwardly from the periphery thereof, a disk-like barrier member rotatably mounted in each of said recesses and having a peripheral portion thereof traversing said chamber, each of said barrier members having peripherally spaced slots, a plurality of generally helical vanes secured to said rotor and disposed in said chamber, said vanes being constructed and arranged to respectively traverse said barrier slots and to co-operate with the periphery of said stator to define compartments variable in volume as said stator rotates, gearing mounted in said stator and connecting said rotating barrier means and said shaft, whereby said barrier means are rotated in fixed relation with said rotor, and inlet and outlet port defining means successively communicating with said compartments as said rotor rotates.

7. The combination defined in claim 4 wherein said stator comprises an assembly of a body portion and a plurality of peripheral portions, said stator recesses being defined respectively by the junctures of said body portion and said peripheral portions.

8. The combination defined in claim 6 wherein said stator comprises an assembly of a body portion and a plurality of peripheral portions, said stator recesses being defined respectively by the junctures of said body portion and said peripheral portions.

9. The combination of a housing, a rotor mounted in said housing, a stator element supported by said housing and disposed entirely within the periphery of said rotor, slotted rotary abutments mounted for rotation on said stator element, generally helical vanes carried by said rotor and co-operating with said stator element and said slotted rotary abutments to form compartments adapted to vary in volume as said rotor is rotated, an intake valve secured to one axial side of said stator element, and a discharge valve secured to the opposite axial side of said stator element, said intake and discharge valves being arranged to successively communicate with said compartments as said rotor rotates, said intake and discharge valves being in respective communication with a suitable inlet and outlet formed in said housing.

10. The combination of a housing, a rotor mounted in said housing, a stator element supported by said housing and disposed within the periphery of said rotor, slotted rotary abutments mounted for rotation on said stator element, generally helical vanes carried by said rotor co-operating with said stator element and said slotted rotary abutments to form compartments adapted to vary in volume as said rotor is rotated, an intake port defining member secured to one axial side of said stator element, a discharge port defining member secured to the opposite axial side of said stator element, said intake and discharge ports being arranged to successively communicate with said compartments as said rotor rotates, said intake and discharge port defining members having peripherally disposed wall portions arranged to substantially seal the adjacent compartments and recessed portions intermediate said wall portions arranged to provide fluid communication with the adjacent compartments.

11. In a rotary compressor having a housing with a pumping chamber and an inlet and an outlet communicating therewith, an annularly shaped stator element having a curved peripheral surface, a rotor mounted for rotational movement relative to said stator element, said rotor having a portion thereof surrounding said stator element and co-operating therewith to form an annular chamber, said stator element having a plurality of recesses extending inwardly from the periphery thereof, a disk-like barrier member rotatably mounted in each of said recesses and having a peripheral portion thereof traversing said chamber, each of said barrier members having peripherally spaced slots, a plurality of generally helical vanes secured to said rotor and disposed in said chamber, said vanes being constructed and arranged to respectively traverse said barrier slots and to co-operate with the periphery of said stator element to define compartments variable in volume as said rotor rotates, an intake port defining member secured to one axial side of said stator element, a discharge port defining member secured to the opposite axial side of said stator element, said intake and discharge ports being arranged to communicate with the housing inlet and outlet and to successively communicate with said compartments as said rotor rotates.

12. The combination defined in claim 11 wherein said stator comprises an assembly of a body portion and a plurality of peripheral portions, said stator recesses being defined respectively by the junctures of said body portion and said peripheral portions, and fastening means carried by said exhaust port defining means for securing said stator components in assembly.

13. In a rotary compressor of the class described, a housing having a peripheral wall defining a chamber concentric about an axis of the housing, a shaft rotatably mounted in said housing coaxially with said axis, a sleeve surrounding said shaft secured to said housing, an annular stator element freely surrounding said shaft, means securing said stator element to said sleeve to position said stator element concentrically in said chamber, thereby defining an annular space between the external periphery of said stator element and the peripheral wall of said housing, and a rotor element disposed in said annular space, said rotor element having a first hub portion surrounding said sleeve and rotatably journaled in said housing and a second hub portion axially spaced from said first hub portion and keyed to said shaft at a point on the opposite side of said stator element than said first hub portion.

14. The combination defined in claim 13 wherein said rotor element comprises an assembly of two half portions joinable on a radial plane, one of said half portions including said first hub portion and the other half portion including said second hub portion.

15. In a rotary compressor, a housing having a peripheral wall defining a chamber concentric about an axis of the housing, a shaft rotatably mounted in said housing coaxially with said axis, a sleeve surrounding said shaft and secured to said housing, an annular stator element freely surrounding said shaft, means securing said

stator element to said sleeve to position said stator element concentrically in said chamber, thereby defining an annular space between the external periphery of said stator element and the peripheral wall of said housing, a rotor element disposed in said annular space, said rotor element having a first hub portion surrounding said sleeve and rotatably journaled in said housing and a second hub portion axially spaced from said first hub portion and keyed to said shaft at a point on the opposite side of said stator element than said first hub portion, slotted rotary abutments mounted for rotation on said stator element, generally helical vanes carried by said rotor co-operating with said stator element and said rotary abutments to form compartments variable in volume as said rotor element rotates, inlet and outlet port defining means successively communicating with said compartments as said rotor element rotates, and gearing mounted in said stator element connecting said rotary abutments and said shaft, whereby said rotary abutments are rotated in fixed relation with said shaft.

16. In a rotary compressor, a housing defining a chamber concentric about an axis of the housing, a shaft rotatably mounted in said housing coaxially with said axis, a sleeve surrounding said shaft secured to said housing, an annular stator element freely surrounding said shaft, means securing said stator element to said sleeve to position said stator element concentrically in said chamber, thereby defining an annular space between the external periphery of said stator element and the housing, a rotor element disposed in said annular space, said rotor element having a first hub portion surrounding said sleeve and rotatably journaled in said housing and a second hub portion axially spaced from said first hub portion and keyed to said shaft at a point on the opposite side of said stator element than said first hub portion, slotted rotary abutments mounted for rotation on said stator element, generally helical vanes carried by said rotor element cooperating with said stator element and said slotted rotary abutments to form compartments variable in volume as said rotor element rotates, an intake port defining member and a discharge port defining member respectively secured to opposite sides of said stator element and co-operating with said compartments, said intake port defining member being secured to said stator element by said means securing the stator element to said sleeve.

17. In a rotary compressor, a housing defining a chamber concentric about an axis of the housing, a shaft rotatably mounted in said housing coaxially with said axis, a sleeve surrounding said shaft secured to said housing, an annular stator element freely surrounding said shaft thereby defining an annular space between the external periphery of said stator element and the housing, said stator element having a plurality of recesses extending inwardly from the periphery thereof, said stator element comprising an assembly of a body portion and a plurality of peripheral portions, such recesses being defined respectively by the junctures of said body portion and said peripheral portions, a rotor element disposed in said annular space, said rotor element having a first hub portion surrounding said sleeve and rotatably journaled in said housing and a second hub portion axially spaced from said first hub portion and keyed to said shaft at a point on the opposite side of said stator element than said first hub portion, a disk-like barrier member rotatably mounted in each of said recesses and having a peripheral portion thereof traversing said annular space, each of said barrier members having peripherally spaced slots, a plurality of generally helical vanes secured to said rotor element, said vanes being constructed and arranged to respectively traverse said barrier slots and to co-operate with the periphery of said stator element to define compartments variable in volume as said rotor element rotates, an intake port defining member and a discharge port defining member respectively secured to opposite sides of said stator element and co-operating with said compartments, said intake port defining member being secured to said stator element by said means securing the stator element to the sleeve, and said discharge port defining member including fastening means for securing said peripheral portions and said body portion of said stator element in assembly.

18. The combination defined in claim 17 plus gearing means mounted in said stator element and connecting said barrier members to said shaft, whereby said barrier members are rotated in fixed relation with said rotor element.

19. In a device of the class described, the combination of a housing having radially spaced inner and outer portions defining an annular recess therebetween, an annularly shaped hollow rotor mounted for rotation in said recess of said housing, rotary abutments conforming to the cross-section of said hollow rotor mounted in the inner portion of said housing for rotation in said hollow rotor, generally helical vanes in said hollow rotor to move in close association with respect to said inner portion of said housing and said rotary abutments to form compartments adapted to vary in volume as the rotor is rotated, and intake and discharge ports for communication with said compartments.

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