

Aug. 27, 1957

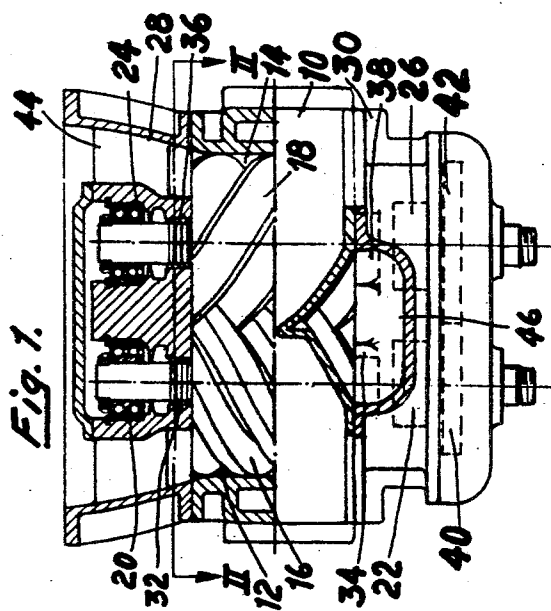
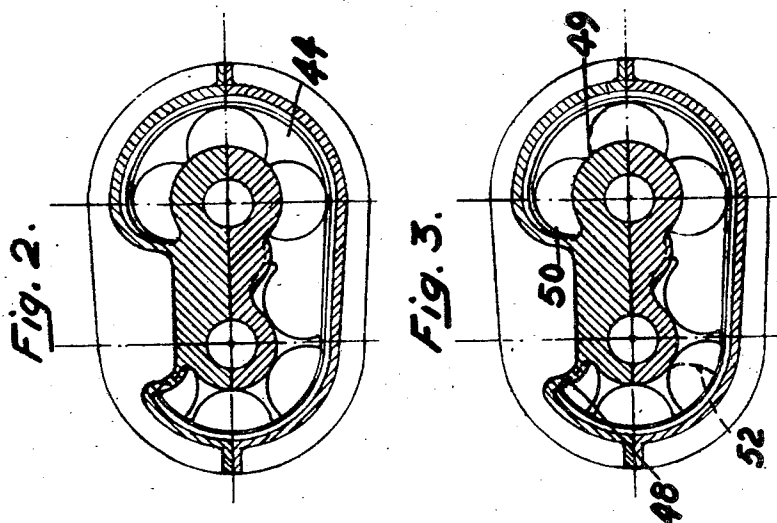
H. R. NILSSON ET AL

2,804,260

ENGINES OF SCREW ROTOR TYPE

Filed May 27, 1950

6 Sheets-Sheet 1



INVENTORS
Hans Robert Nilsson
Teodor Emanuel Lindhagen
BY
Jens L. Markes
ATTORNEY

Aug. 27, 1957

H. R. NILSSON ET AL
ENGINES OF SCREW ROTOR TYPE

2,804,260

Filed May 27, 1950

6 Sheets-Sheet 2

Fig. 4.

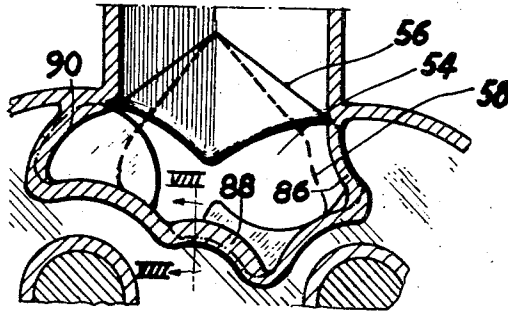


Fig. 7.

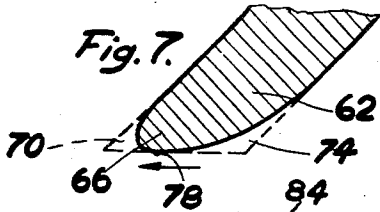


Fig. 6.

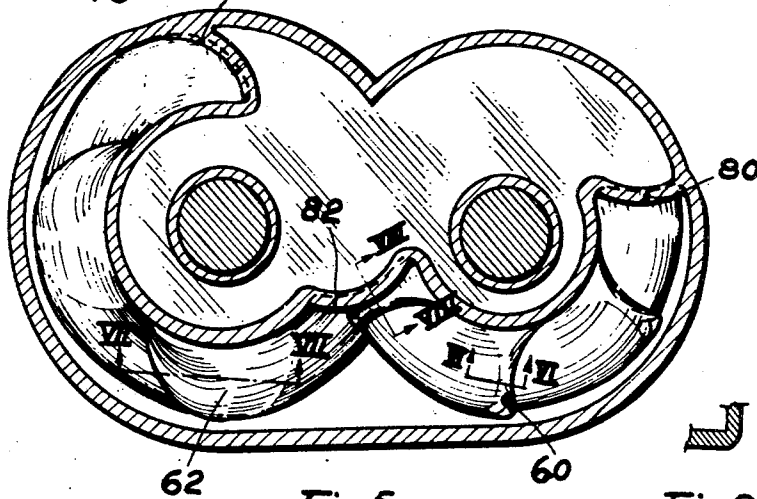
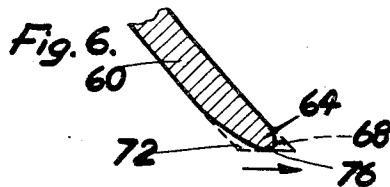


Fig. 5

Fig. 8.

INVENTORS
Hans Robert Nilsson
Teodor Immanuel Lundhaga
BY
Jawist. Mante
ATTORNEY

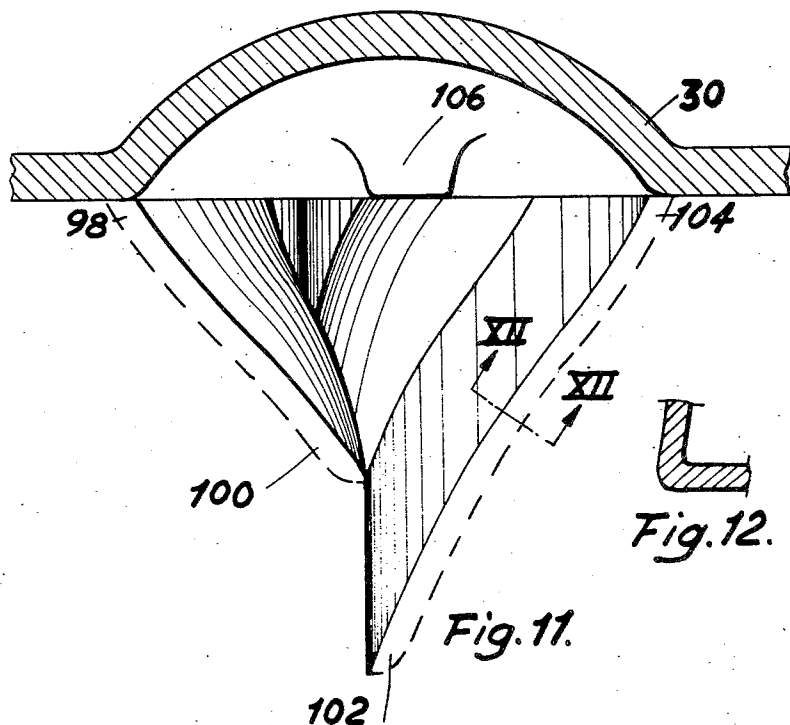
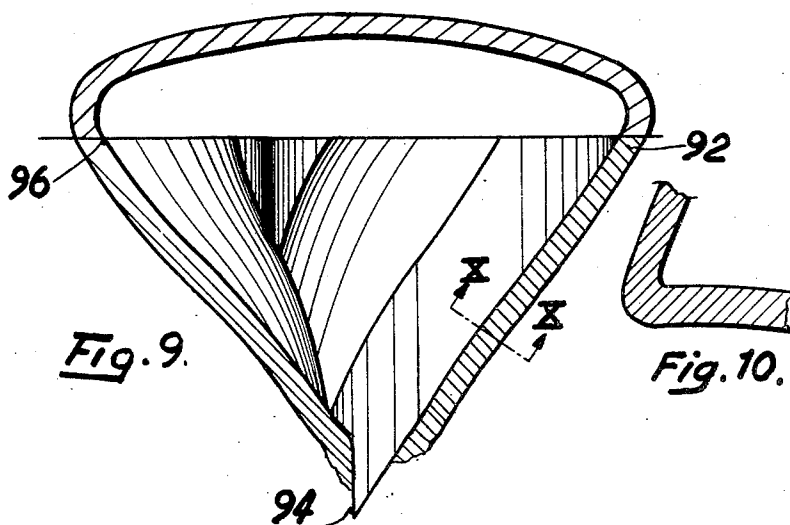
Aug. 27, 1957

H. R. NILSSON ET AL
ENGINES OF SCREW ROTOR TYPE

2,804,260

Filed May 27, 1950

6 Sheets-Sheet 3



INVENTOR
Hans Robert Nilsson
Theodor Emanuel Lindgren
BY
Jarvis C. Marklee
ATTORNEY

Aug. 27, 1957

H. R. NILSSON ET AL
ENGINES OF SCREW ROTOR TYPE

2,804,260

Filed May 27, 1950

6 Sheets-Sheet 4

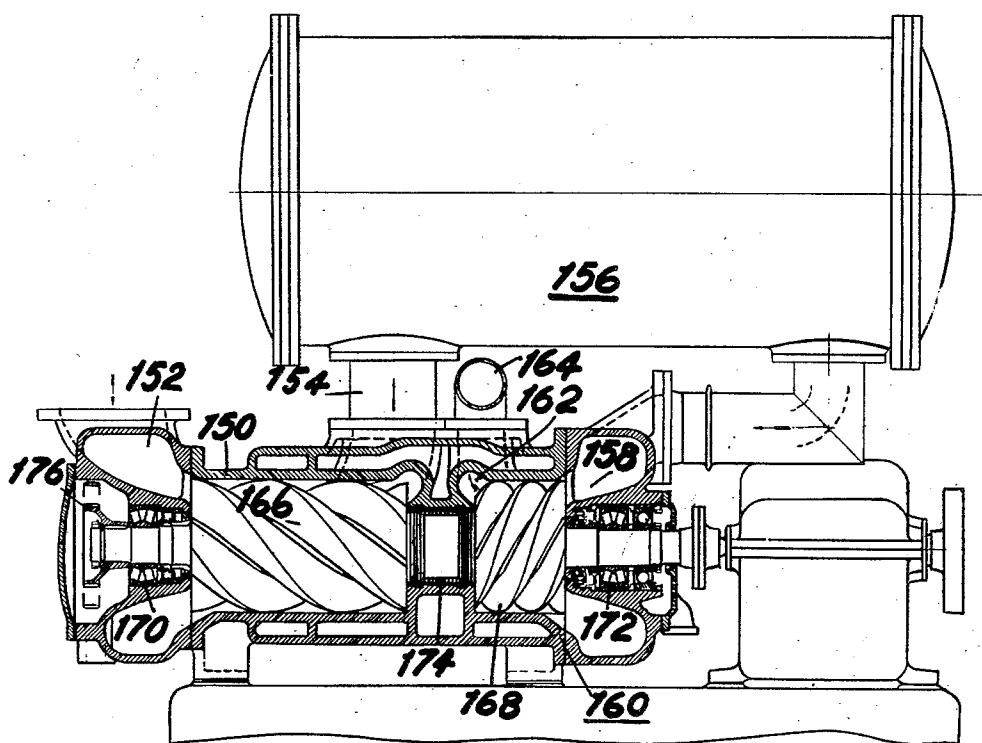


Fig. 13.

INVENTOR
Hans Robert Nilsson
By Immanuel Lindhagen
J. W. C. Moore
ATTORNEY

Aug. 27, 1957

H. R. NILSSON ET AL
ENGINES OF SCREW ROTOR TYPE

2,804,260

Filed May 27, 1950

6 Sheets-Sheet 5

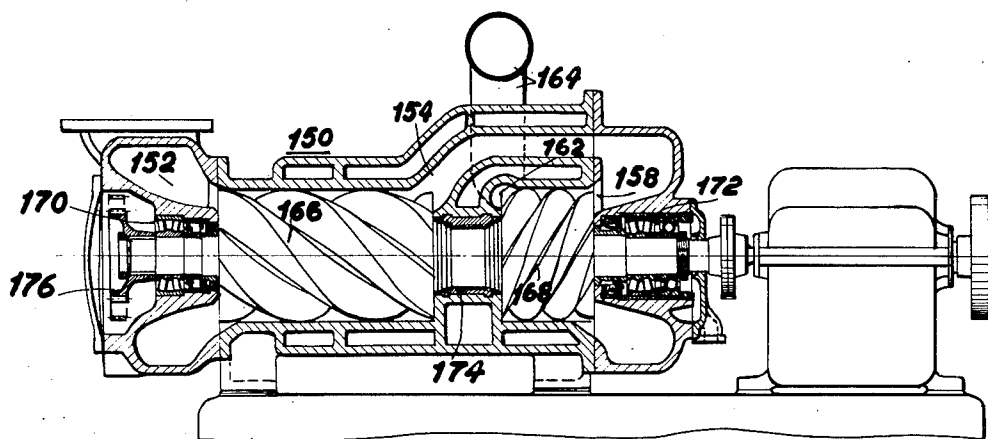


Fig. 14.

INVENTORS
Hans Robert Nilsson
Torsten Gunnar Lindgren
BY
Jens E. Mathiasen
ATTORNEY

Aug. 27, 1957

H. R. NILSSON ET AL

2,804,260

ENGINES OF SCREW ROTOR TYPE

Filed May 27, 1950

6 Sheets-Sheet 6

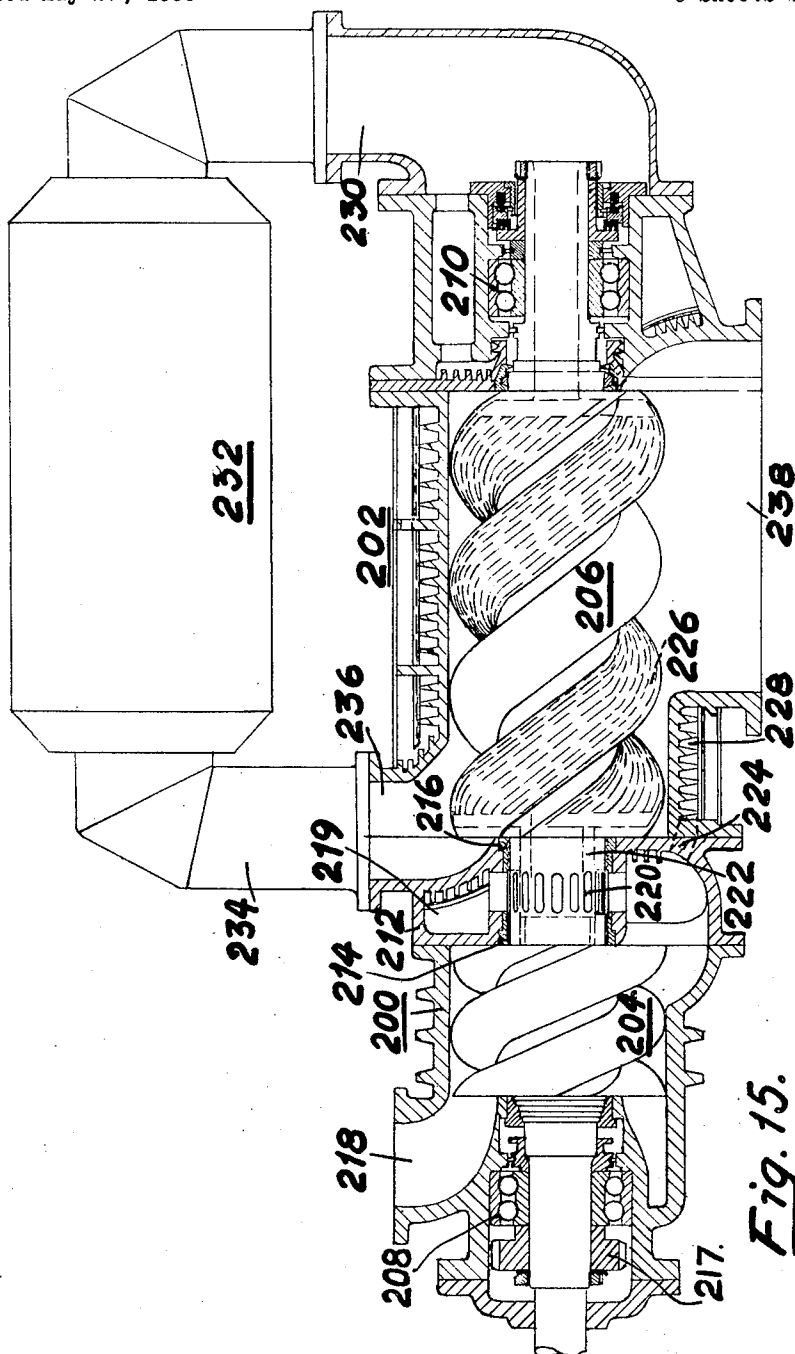


Fig. 15.

INVENTOR
Hans Robert Nilsson
Teodor Gunnar Lundhagen
BY
Jensiel, Wolfe
their ATTORNEY

1

2,804,260

ENGINES OF SCREW ROTOR TYPE

Hans R. Nilsson and Teodor Immanuel Lindhagen, Stockholm, Sweden, assignors, by mesne assignments, to Svenska Rotor Maskiner Aktiebolag, Nacka, Sweden, a corporation of Sweden

Application May 27, 1950, Serial No. 154,642

Claims priority, application Sweden July 11, 1949

21 Claims. (Cl. 230—139)

This invention relates to displacement engines of the rotary screw wheel type having two or more rotors formed with intermeshing helical lobes and grooves with working chambers enclosed between the flanks of the lobes and the casing surrounding the rotors, the working medium being admitted to and discharged from the working chambers during their rotation past inlet and outlet openings formed in the casing. In such engines the lobes of the two cooperating rotors may have the same form of profile, but in general they are formed with different profiles, one rotor, referred to as the male rotor, being provided with lobes of substantially convex cross section lying substantially or for the major portion thereof outside the pitch circle of the rotor, while the mating rotor, hereinafter referred to as the female rotor, is provided with lobes the flanks of which are of substantially concave cross section and lie substantially or for the major portion thereof within the pitch circle of the rotor. Rotors of the latter type are, for example, described in the specification of U. S. Patent No. 2,622,787 granted December 23, 1952, on the application of Hans Robert Nilsson.

In the case of rotors having axially straight or nearly straight lobes, that is, having a helix angle providing a displacement of the lobes along the length of the rotor of up to approximately 60°, an engine is obtained in which the working chambers open to the outlet without change of the volume of the chambers having taken place therein after their being filled through the inlet port. Rotors of this type, therefore, are used as pumps for pumping liquids and as pumps (or blowers) for gaseous fluids for displacement thereof in accordance with the principle of the Roots Blower.

If, on the other hand, rotors are formed with a wrap angle greater than approximately 60°, the engine may in practice work as a displacement pump if the inlet and outlet openings are dimensioned particularly for this purpose. Primarily, however, such rotors are used in compression and expansion engines for elastic fluids adapted to produce either an internal compression or an internal expansion to an extent desired for the working process, that is, a change of volume in the working chambers while communication is cut off both from the inlet and from the outlet of the device. The type disclosed in U. S. Patent No. 2,243,874 may be mentioned as an example of such an engine.

However, the helix or wrap angle alone is not a criterion of a sufficient volume change being obtained in the working chamber during compression and/or expansion phase of the cycle, since the number of lobes plays an important part in the operation of the device. The male rotor must be provided with at least three lobes to obtain any significant internal volume change before the outlet opens to provide a suitable discharge area in the case of the compressor. From the practical standpoint the lobe combinations 3/3, 4/4 and 4/6, wherein the first figure indicates the number of lobes of

2

the male rotor and the second figure the number of lobes of the female rotor, have proved most advantageous for engines having low, average, and high-pressure ratios, respectively. At the same time the helix or wrap angle has been selected within the limits of from approximately 150° to 213°, the lower value normally being used for engines having a low-pressure ratio with the lobe combination 3/3 while the higher value applies to engines of relatively high-pressure ratio and the lobe combination of 4/6. By providing the rotors with a small number of lobes the depth of the grooves may be made relatively great and the greatest possible part of the volume of the rotor may be utilized as working chamber volume.

For reasons of strength in order to obtain a sufficiently outlet opening at higher pressure ratios, the number of lobes as well as the wrap angle have been increased. The change in internal volume requires a greater and greater part of the wrap angle, the area of the outlet opening being accordingly decreased. On the other hand, in prior constructions it was not considered desirable to increase the wrap angle beyond the aforementioned values given in relation to the number of lobes because, in doing so, the compression phase of the cycle will begin before the inlet or filling cycle is completed. Furthermore, the leakage losses increase with increasing wrap angle.

In view of the length of the leakage space between the rotors and the casing and between the rotors themselves, engines of the type under discussion have been made with a length "L" of the rotor which in relation to its diameter "D" is such that in general the ratio L/D amounts to about 1.5 or more, with which dimensions the leakage area between the different working chambers is a minimum for a given capacity of the engine.

Tests made with rotors of the design disclosed in U. S. Patent No. 2,457,314 show that, contrary to expectation, even very small deviations from sharp edges, which may conveniently be designated as convexly chamfered edge portions, result in an improvement in efficiency of operation of as much as two percent. Such results have led applicants to a new approach to the problem of increasing the efficiency of operation by introducing modifications in the mode of operation directed to reduction in dynamic losses, instead of proportioning the apparatus with the principal object in view of obtaining the minimum internal leakage loss. This approach has resulted in radically changing the hitherto considered optimum values of rotor proportions and wrap angle.

The principal object of the present invention is to provide an improved construction of the above described type of positive displacement engine wherein the dynamic losses at the inlet and outlet openings of the casing are substantially reduced as compared with prior forms of construction. According to the present invention in an engine of the kind under discussion the male rotor is provided with at least three lobes, the lobe angle of which exceeds 50°, and the wrap angle of which results in an angular displacement of a lobe along the effective length of the rotor of between 200° and 325° and in which the male rotor is so proportioned that the ratio between its length "L" and diameter "D" lies within the limits of 0.5 and 1.5.

A further object of the invention is the provision of an improved embodiment of the end planes of the rotor lobes and of the edges defining the inlet and outlet openings of the casing, in order still further to reduce the dynamic losses occurring at the inlet and the outlet. Additionally, a further object is a practical application of an engine of the type under discussion in the form of

aggregates composed of several engines of the screw rotor type.

The term "lobe angle" as used in this specification means the smaller angle between the direction of the crest of the top of a lobe of a male rotor and a line parallel to the axis of the rotor.

The principles of the invention will be further discussed in the following portion of this specification, taken in conjunction with the accompanying drawings, it being understood that the scope of the invention is not to be considered as limited to the embodiments hereinafter described by way of example.

Fig. 1 is a plan view, partly in section, of an engine embodying the present invention;

Fig. 2 is an end view of the engine shown in Fig. 1 looking toward the inlet end of the engine;

Fig. 3 illustrates the purely axial inlet end of the compressor having a materially greater wrap angle as compared with the inlet construction of prior engines having a relatively smaller wrap angle;

Fig. 4 illustrates a combined axial-radial outlet for a compressor having a considerably greater wrap angle than that utilized with prior forms of construction;

Fig. 5 illustrates the axial inlet of a compressor having an inlet port in the casing, with chamfered edges, as well as having chamfered edges on the rotor lobes;

Fig. 6 illustrates in section on the line VI—VI of Fig. 5 the lobe of a female rotor constructed in accordance with the present invention;

Fig. 7 shows in section on the line VII—VII of Fig. 5 the lobe of a male rotor constructed in accordance with the present invention;

Fig. 8 shows a section on the line VIII—VIII of Fig. 5 through the curved edges of the ports at the inlet and outlet of the casing;

Fig. 9 shows the radial portion of the outlet port of a compressor embodying the invention;

Fig. 10 is a section taken on the line X—X of Fig. 9;

Fig. 11 shows the outlet of a compressor having rotor lobes with generated profiles;

Fig. 12 is a section taken on the line XII—XII of Fig. 11;

Fig. 13 shows the application of the invention to a unit having compression in two stages, with intercooling between the stages;

Fig. 14 shows a variation of the device shown in Fig. 13, with the intercooler omitted; and

Fig. 15 illustrates a combustion gas operated power plant employing structure embodying the invention.

Figs. 1 and 2 illustrate an engine of the rotary screw wheel type embodying the principles of the present invention, which may be utilized either as a compressor or as a motor, depending upon the direction of rotation of the rotors, but for purposes of the present description the engine will be hereinafter described on the assumption that it is to operate as a compressor.

The casing 10 is provided with bores 12 and 14 forming intersecting barrel portions in which the male rotor 16 and the female rotor 18, respectively, are rotatably mounted, the male rotor being carried by bearings 20 and 22 and the female rotor by the bearings 24 and 26, located in the end walls 28 and 30. Shaft seals are provided at 32 and 34 and at 36 and 38 to reduce the leakage outwardly between the shaft journals of the rotors and the end walls.

The rotors are provided respectively with intermeshing synchronizing gears 40 and 42 which operate to maintain the rotors in their proper relative rotational positions and thereby insure a predetermined clearance between the cooperating rotor lobes. The rotors also rotate completely freely relative to the casing but with the smallest practical clearance to reduce leakage losses.

The casing is provided with an inlet 44 and an outlet 46.

In the embodiment shown the rotors are of the kind described in U. S. Patent No. 2,622,787. With this kind of rotor, the outer part of the profile of the lobe of the male rotor, which when in full engagement with the female rotor is located within the pitch circle of the female rotor, is substantially arcuate, while the remaining part of the profile of the male lobe is generated by points on or substantially on the pitch circle of the female rotor.

The groove of the female rotor has a profile corresponding to the profile of the lobe of the male rotor, that is, of substantially arcuate cross-section, so that with full engagement between the outer part of the lobe of the male rotor and a groove in the female rotor, the groove will be practically completely filled by the outer portion of the lobe of the male rotor.

A characteristic feature of this type of rotor is that no trapped volumes of fluid are formed either at the inlet or outlet of the compressor, which trapped volumes are detrimental to efficiency, as is the case in prior forms of rotor construction. At the same time, a shorter sealing edge length and less leakage area between the working chambers formed by the rotor lobes are obtained, and therefore the internal leakage losses are less than in prior forms of construction.

By utilizing arcuate rotor profiles in an engine embodying the present invention the following advantages, among others, are obtained: The leakage area is reduced, there are no trapped pockets of working fluid, and the dynamic losses are low. The invention is obviously not restricted to the specific form of rotor profile previously discussed and disclosed in U. S. Patent No. 2,662,787, although such form of profile is to be preferred.

The scope of the invention is such that it may be with advantage applied also to engines having other lobe profiles, such as those known in the prior art which embody either completely or partially generated profiles. Even with such profiles, increase in efficiency can be obtained by the present invention by reduction of dynamic losses.

In accordance with the principles of the present invention, by changing the dimensions of the compressor rotors to provide a smaller ratio L/D for a given capacity and a given wrap angle of the engine, a greater lobe angle is obtained as will be clear from Fig. 1. This lobe angle will be increased by increasing the wrap angle to a value approaching 325° . The reduction in the ratio L/D permits an increased area of inlet for the compressor, relative to the volume of fluid inducted. Therefore, in most cases, the inlet 44 may be made for purely axial flow without increasing the inlet velocity of the fluid to an undesirably high value and without causing unnecessarily large inlet losses. Since the filling of a groove takes place in a wholly axial direction, the working medium or fluid will continue in its natural axial direction of motion during the entire filling or charging period and during the induction period will be disturbed only by the end planes of the rotor lobes when the latter move across the inlet opening.

Fig. 3 illustrates the influence of the wrap angle upon the size of the axial inlet in a compressor of given rotor dimensions. In an embodiment of the invention having a wrap angle of 300° the suction phase does not end until the edges 48 and 50 are reached, while in earlier constructions with a wrap angle of 200° , or thereabout, the filling of the working chambers during the suction phase is ended along the dotted lines 49 and 52.

During the ensuing compression phase the trapped gaseous fluid is displaced peripherally-axially toward the outlet 46 which, also due to the increased lobe angle of the rotors with a small L/D ratio, in accordance with the principles of the present invention, can be made with a greater area of opening but, more important, with a greater length of opening edge in relation to the volume of the working fluid handled. If the wrap angle is further increased it becomes possible to increase the area of

the inlet and outlet openings still further. As a result the peripheral displacement of the mass of the working fluid is reduced on its passage from inlet to outlet, this applying particularly to that part of the volume of the working fluid last inducted into the working chamber, such part flowing almost directly axially through the compressor without being checked by the end wall 30 of the outlet, as is the case with the portion first inducted, which after impact is moved toward the outlet 46 upon rotation of the rotors. The velocity of the straight or purely axial flow of the gas is, however, reduced additionally through the increased axial cross-section area, when the rotors are made with a small ratio L/D , and therefore the dynamic losses which occur when the axial movement of the particles of the working fluid are decelerated and converted into peripheral movement, are also reduced.

The outflow losses from the compressor depend, to a great extent, on the backflow which at the point of highest efficiency takes place between the narrow space between the rotor lobes and the contour of the outlet opening during the first moment of opening while the pressure in the rotor grooves is still lower than the counter-pressure in the outlet passage. However, the partial vacuum becomes a pressure above atmospheric in the working chambers due to the fact that the change in volume of the working chambers takes place more rapidly than the increase of the outlet area. The above process increases the required work of compression beyond the theoretical value and thus also increases the power required to operate the compressor.

By proportioning the compressor with a relatively small ratio of L/D a large wrap angle in accordance with the principles of the invention, the power required is reduced because reduction in volume of the trapped volumes of working fluid takes place more slowly while the length of the opening edge of the outlet, and thus the outlet opening area, increases in relation to the quantity of working fluid passing through the compressor.

Fig. 4 illustrates the influence of the wrap angle on a compressor having a combined axial-radial outlet 54 with given rotor dimensions. To facilitate explanation and understanding of the construction, the radial portion of the port has been swung up from a plane at right angles to the axial portion of the outlet into the same plane as that of the axial portion of the outlet.

With a wrap angle of 300° , exhaust begins along the outlet edge 56, shown in full lines, while with a wrap angle of 200° opening does not take place until the dash line 58 has been passed. The longer length of the edge of the opening obtained with the greater wrap angle affords a greater opening area during the initial or opening period and also affords a greater total outlet area, so that the working chambers are maintained open at the outlet during the greater part of a revolution, all of which contributes to the reduction of the outlet losses.

The above described flow conditions and their influence upon the compressor efficiency will best be understood by a comparison between two compressors having the same displacement or stroke volume, one being constructed in accordance with prior known principles and the other being constructed in accordance with the principles of the present invention. The prior construction is one having a ratio L/D of 1.5, the rotor length being 300 mm., the rotor diameters being 200 mm. with a wrap angle of the lobe crests of 200° . The construction embodying the present invention is one having a ratio L/D of 0.75, the rotor length being 189 mm., and the rotor diameter being 252 mm., with the wrap angle of the lobe crests being 250° .

Both compressors were operated under the same pressure conditions and at the rotational speed corresponding to the values giving optimum efficiency for the compressor constructed in accordance with the earlier design. The capacity of the two engines was thus the same, every regard having been paid to giving the compressor of the

earlier design as favorable operating conditions as possible.

Tests under such conditions have demonstrated that a compressor with a small ratio L/D and a great pitch angle gives a considerably higher efficiency even under such selected operating conditions. This result must obviously depend primarily on reduced dynamic losses during the filling or suction and the exhaust periods.

An increase of the wrap angle from 200° to 250° does not produce any reduction in the volumetric efficiency because of the fact that the compression stroke begins prior to the filling of the working chambers being completed. This is because the working chambers, formed by the rotor grooves, due to the improved inlet conditions, will be filled with gaseous fluid of a higher static pressure and thus a greater quantity of the working medium per unit volume of the working chambers than in the case of a compressor constructed in accordance with earlier concepts. Furthermore, the compressor benefits by this improved filling or charging of a higher static pressure because less work of compression is required to obtain a desired ratio of pressure increase. Thus, the improved filling or charging contributes to a higher overall efficiency.

The ratio between the area of the inlet opening, the length of the edge of the opening at the outlet and the outlet opening area and the respective gas velocities in the two compressors will be seen from the comparative figures in the following table:

Compressor	$L/D=1.5$	$L/D=0.75$
Total area of inlet.....	1	2
Velocity of working medium.....	1	0.5
Velocity height.....	1	0.25
Opening edge length.....	1	1.6
Exhaust velocity during the first moments of the opening period.....	1	0.4
Velocity height.....	1	0.16
Total area of outlet.....	1	2.2
Outlet velocity.....	1	0.45
Velocity height.....	1	0.20

That the exhaust velocity during the first moments becomes so low for the compressor having the smaller ratio L/D and the greater wrap angle depends not only on the longer opening edge but also on the fact that the volume reduction of the working chambers is taking place more slowly, so that per unit time a smaller quantity is exhausted to the outlet while at the same time the opening is taking place more rapidly in the engine with lower ratio L/D due to its greater rotor diameter and greater peripheral speed of the lobe crests of the rotors.

The test results show that through the constructional improvements according to the invention it has become possible to improve the compressor efficiency by 5 to 8% compared with earlier designs.

Besides the abovementioned advantages with a small ratio of L/D and great wrap angle it is also pointed out that the sealing distance between the rotor lobes can be less. Wear of the synchronizing gear or a distortion of the rotors, computed in terms of percentage, has less influence upon the clearance and thus upon the reliability of service of the engine.

As already mentioned, the screw compressor obtains still further improved properties by rounding the end edges of the lobes at the inlets and outlets, which results in reduced flow losses to the rotor grooves. However, a further improvement is obtained by designing these rounded portions in such a manner that the end planes of the male rotor, and of the female rotor respectively, obtain a more advantageous profile form in the direction of flow than what is shown in the abovementioned U. S. Patent 2,457,314, and at the same time those edges of the inlet, and the outlet opening respectively, are convexly chamfered between which and the convexly chamfered side edges of the rotors the working fluid is inducted and exhausted, respectively.

Fig. 5 shows the inlet of a compressor having purely

axial inflow, the rotors and inlet opening of which are convexly chamfered according to the principles of the invention. Thus, the end planes of the female rotor lobes 60 have been convexly chamfered so that in cross section as shown in Fig. 6 a stream-lined profile is obtained which extends from the root portion of the lobes to the upper part thereof but which leaves the outermost portion free so that the sealing remains unchanged.

The lobes 62 of the male rotor at their inlet ends are provided with profiles as shown in Fig. 7 in order to offer the least possible resistance to the inflowing working fluid.

In order that the sealing between the rotor and the casing shall remain unchanged, it is necessary that the crest of the lobe adjacent to the bore in which the latter is located, be maintained and that at least a narrow sealing surface extend from the lobe root to the lobe crest at the end surface of the lobe.

If the sealing between the rotors is to remain unchanged, those parts of the flanks of the female rotor which generate the root portion of the male rotor must be left unchanged while the convex chamfering of the end surfaces of the lobes of the male rotor is extended only from the upper or crest portion of the lobe to the root portion thereof which is generated by the points on the lobes of the female rotor. In other words, only the substantially arcuate portions of the flanks of the lobes of the male rotor are chamfered at the end surfaces of the rotor.

The characteristic feature of the design of the profile of the lobes of the rotors shown in Figs. 6 and 7 is that the nose portions 64 and 66, respectively, are directed in the direction of rotation of the rotors. This is accomplished by rounding the leading edges 68 and 70, respectively, considered in the direction of rotation, with a smaller radius of curvature than the respective trailing edges 72 and 74 and possibly displacing the remaining end sealing surfaces 76 and 78, respectively, toward the end wall of the casing in the direction toward the leading flank of the lobes, considered in their direction of rotation.

The inlet opening in the casing should, at the same time, be convexly chamfered along those edges where the inflow to and the outflow from the working chambers commences and terminates.

As shown in Fig. 5, the edges 80, 82 and 84 are thus convexly chamfered, as shown more clearly in Fig. 8, to produce progressive opening and closing of the working chambers which has a favorable effect on the flow of working fluid during the filling or inlet phase of the cycle.

The outlet or outflow losses are reduced in a similar manner by utilizing a combined axial-radial outlet, as shown in Figs. 4, 9 and 10, having convexly chamfered opening edges. The axial part of the outlet is accordingly rounded at 86, 88 and 90, while the radial opening shown in Fig. 9 is convexly chamfered in like manner along the entire triangular contour 92, 94, 96, to the form shown in enlarged section in Fig. 10.

The above described axial-radial outlet is intended for a compressor having lobes of circular profile but, for the sake of completeness, Figs. 11 and 12 show the form of an outlet embodying the principles of the present invention but applicable to an engine of prior known form in which the lobes of the male rotor have a profile, one flank of which is substantially arcuate, while the other is generated by a point on the pitch circle of the female rotor. In such construction at least the edges of the radial opening along the line 98—100 and lines 102—104 should be convexly chamfered as well as that part of the end wall 30 which projects into the outlet to form a seal between adjacent working chambers which are, respectively, being filled and which are discharging. Furthermore, as regards the axial portion of the outlet, those edges should be convexly chamfered to correspond to the edges indicated by 86, 88 and 90 in Fig. 4.

In order to further reduce the dynamic losses the edges of the rotor lobes adjacent to the outlet should also be slightly convexly chamfered although not necessarily to the extent as indicated in Figs. 6 and 7, since in this case only a question of effecting gradual opening of the communication between the working chamber and the outlet port is involved.

In Figs. 1 to 12 the inlet and outlet openings have been designed on the assumption that the engine is to operate as a compressor. However, if the engine is to operate as a power producing motor, by expanding a pre-compressed and possibly heated elastic fluid medium, the design of the inlet and outlet of the engine and the shaping of the convexly chamfered portions of the openings, as described in Figs. 4 to 12, are similar, since also in this case the inlet can be made wholly axial while the outlet should preferably be made as a combined axial-radial port which, in the case of such a power producing engine, the rotor lobes profiled at their ends, as shown in Figs. 6 and 7, are located at the inlet end, that is, in such case the high-pressure end or side of the engine.

In view of the advantages in making the compressor rotors with a small ratio of L/D or, in any event, with a relatively great wrap angle, it is possible to construct units comprising tandem engines of the kind shown in Figs. 13 to 15.

Fig. 13 shows a two-stage compression unit with intercooling between the stages. Air is inducted into the low-pressure compressor 150 through the inlet 152 and is discharged from the outlet 154 to the intercooler 156. From the intercooler the air is delivered through the inlet 158 to the high-pressure compressor 160 and leaves the unit through outlet port 162 and conduit 164. It is a characteristic feature of this unit that the male rotors 166 and 168, as well as their cooperating female rotors, form a single rotor supported at two points only by the bearings (shown only for the male rotor shaft) 170 and 172 without any intermediate bearing being provided between the two rotor parts 166 and 168, or between the corresponding female rotor parts meshing with the male rotor parts 166 and 168. Seals, such as those indicated at 174, are provided to reduce leakage between the two compression stages. In order to enable such a construction to be made the two male rotor parts and the cooperating female rotor parts must have the same cross-sectional form and pitch diameter and the axial separation between the male rotor parts and the female rotor parts must be the same. As will be seen a single synchronizing gear is all that is required, one of the synchronizing gears being shown at 176.

The practical solution for making such an engine lies in making the high-pressure compressor part with relatively short rotors having an L/D ratio of from 0.5 to 0.7, with a moderate wrap angle, while the rotors of the low-pressure compressor part are made relatively long with an L/D ratio of from 1.5 to 2.0 and with a relatively great wrap angle, preferably greater than 250° . With such an arrangement the bearing distances are relatively short in comparison to the core diameter of the rotors and the dynamic losses are relatively low for both compressor stages, since in both cases a very substantial wrap angle is utilized in accordance with the principles of the invention as hereinbefore stated.

A further advantage of the unit shown in Fig. 13 is that to a large extent the unit is balanced axially. Furthermore, the shaft leakage is relatively small as compared with designs in which the two stages are completely separated.

Fig. 14 shows a modification of the above described two-stage unit in which the two compressor stages communicate directly with each other without the interposition of the intercooler.

Fig. 15 shows a gas power unit comprising a compressor 200, of the screw wheel type, and an expander or motor 202, of the screw wheel type, these components

being assembled in accordance with the same principles as shown in the embodiments illustrated in Figs. 13 and 14. In the present instance the rotors with respect to the selection of the ratio L/D and/or the wrap angle, are proportioned within the limits hereinbefore discussed in this application.

Thus, the rotor parts 204 and 206, in this embodiment, form parts of a common male rotor supported only at its ends by the shaft journals 208 and 210. The intermediate or common end wall 212 serves only as an air distributing chamber and carries no bearing for the supporting of the rotors. The sealing devices, such as are shown at 214 and 216, are intended primarily only to reduce the amount of leakage air passing from the compressor side to the working chambers of the motor. Synchronizing between the male rotor parts and their cooperating female rotor parts is obtained by means of usual synchronizing gears, one of which is shown at 217 at the compressor end of the unit.

In this case also, the male rotor parts and their cooperating female parts are of similar cross-sectional form, but the length and wrap angle of the different parts of the unit are different. The rotors forming the compressor part of the unit have a relatively small ratio of L/D and a relatively moderate wrap angle, while the rotors of the expander or motor part have a relatively high ratio of L/D and a relatively greater wrap angle. In both cases a relatively large lobe angle is utilized to secure favorable inlet and outlet flow conditions and thus maintain relatively low dynamic losses.

In the unit illustrated, air is inducted through the inlet 218, is collected after compression in the distribution chamber 219, and is led from the latter through apertures 220 in the intermediate shaft portion 222 and apertures 224 in the casing of the motor part of the unit into cooling systems 226 and 228 of the motor, formed in part in the rotors as well as in the casing. Having passed through the cooling system the air is led by way of conduit 230 to the combustion chamber 232 where it is heated to the desired motive fluid temperature and is then delivered to the motor part 202 through conduit 234 and inlet 236. After expansion in the motor the spent gases are discharged through outlet 238.

The high-pressure ends of both the compressor and motor parts of the unit are adjacent to each other and the inlets are located on the same side of the rotors so that the axial forces developed by the compressed air and the gaseous working fluid are almost completely balanced, thus contributing to the balancing of the axial and radial forces acting on the rotors.

We claim:

1. A rotary device comprising a casing structure having inlet and outlet ports, and at least one rotor set comprising a plurality of rotors mounted to rotate in said structure, said rotors in each set having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of the male rotor of at least one rotor set being within the range from 0.5 to 1.1.

2. A rotary device comprising a casing structure providing intersecting barrel portions and having inlet and outlet ports, the inlet port being arranged at least axially and the outlet port being arranged axially-radially, and a plurality of rotors mounted to rotate in said barrel por-

tions, said rotors having at least three helical lobes and intervening grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the lobes of said male rotor having generally convex profiles and the lobes of said female rotor having generally concave profiles and said male rotor intermeshing with said female rotor to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the axially extending part of said outlet port including convexly chamfered edge portions facing said rotor grooves, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1.

3. A rotary device comprising a casing structure providing intersecting barrel portions and having inlet and outlet ports, the inlet port being arranged at least axially and the outlet port being arranged axially-radially, and a plurality of rotors mounted to rotate in said barrel portions, said rotors having at least three helical lobes and intervening grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the lobes of said male rotor having generally convex profiles and the lobes of said female rotor having generally concave profiles and said male rotor intermeshing with said female rotor to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the axially extending part of said outlet port including convexly chamfered edge portions facing said rotor grooves, the inlet end portions of the flanks of said lobes being rounded to provide a streamlined profile for the inlet ends of said lobes, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1.

4. A rotary device comprising a casing structure providing intersecting barrel portions and having inlet and outlet ports, the inlet port being arranged at least axially and the outlet port being arranged axially-radially, and a plurality of rotors mounted to rotate in said barrel portions, said rotors having at least three helical lobes and intervening grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the lobes of said male rotor having generally convex profiles and the lobes of said female rotor having generally concave profiles and said male rotor intermeshing with said female rotor to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the axially extending part of said outlet port including convexly chamfered edge portions facing said rotor grooves, the inlet end portions of the flanks of said lobes being rounded to provide a streamlined profile for the inlet ends of said lobes, said inlet ends having narrow radially extending portions at the apices of said streamlined profiles for sealing against the adjacent end of the casing structure, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1.

5. A rotary device comprising a casing structure providing intersecting barrel portions and having inlet and outlet ports, the inlet port being arranged at least axially and the outlet port being arranged axially-radially, and a plurality of rotors mounted to rotate in said barrel portions, said rotors having at least three helical lobes and intervening grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the lobes of said male rotor having generally convex profiles and the lobes of said female rotor having generally concave profiles and said female rotor intermeshing with said male rotor to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the ro-

tors rotate, the axially extending part of said outlet port including convexly chamfered edge portions facing said rotor grooves, the inlet end portions of the flanks of said lobes being rounded to provide a streamlined profile for the inlet ends of said lobes, the apices of said profiles being offset with respect to the radial center lines of the lobes toward the acute angled edges of the lobe ends, and said apices providing narrow radially extending lobe ends for sealing against the adjacent end of the casing structure, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1.

6. A rotary device comprising a casing structure providing intersecting barrel portions and having inlet and outlet ports, the inlet port being arranged at least axially and the outlet port being arranged axially-radially, and at least one rotor set comprising a plurality of rotors mounted to rotate in said barrel portions, said rotors in each set having helical lobes and grooves having a wrap angle within the range from 200° to 325° and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1.

7. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of the male rotor of at least one of the sections being within the range from 0.5 to 1.1.

8. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of said male rotor being within the range from 0.5 to 1.1 and said ratio being larger for one section than for the other section.

9. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted

to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° , the male rotor of one section having a smaller ratio of length to diameter than the male rotor of the other section and said smaller ratio being within the range from 0.5 to 1.1.

10. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of the male rotor of at least one section being within the range from 0.5 to 1.1, coaxial male rotors forming a combined male rotor common to both sections and coaxial female rotors forming a combined female rotor common to both sections.

11. A rotary device as claimed in claim 10 in which sections have different ratios of length to diameter for the male rotors thereof.

12. A rotary device as claimed in claim 10 and provided with two sections of different lengths of which the shorter section compresses and delivers an elastic medium and the larger section expands a high temperature medium to provide power for operating the shorter section.

13. A rotary device as claimed in claim 10 in which the coaxial male rotors have the same cross sectional area but different lobe angles and the coaxial female rotors have the same cross sectional area but different lobe angles.

14. A rotary device as claimed in claim 11 in which the male rotor of the section having the larger ratio of length to diameter has a wrap angle exceeding 250° .

15. A rotary device as claimed in claim 11 for two stage pressure change of a working fluid, in which the section with the larger one of said ratios constitutes the low pressure stage and the section with the smaller of said ratios constitutes the high pressure stage.

16. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communi-

cating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and one section having a smaller ratio of length to diameter of the male rotor than the other section, the smaller one of said ratios being within the range from 0.5 to 1.1, the section with the larger one of said ratios constituting the low pressure stage and the section with the smaller one of said ratios constituting the high pressure stage.

17. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle or said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of the male rotor length to diameter of at least one of the sections being within the range from 0.5 to 1.1, coaxial male rotors being connected to form a combined male rotor common to both sections and coaxial female rotors being connected to form a combined female rotor common to both sections, and one of said sections operating as a motor driving the other of said sections as a compressor.

18. A rotary device as claimed in claim 17 in which the high pressure ends of the male coaxial rotors and of the female coaxial rotors, respectively, of the two sections are directed axially toward each other, the main parts of the inlet ports of the two sections being located on the same side of a plane common to the rotor axes.

19. A rotary device comprising a casing structure divided into two succeeding sections, each section having inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50° and the ratio of length to diameter of the male rotor of at least one section being within the range from 0.5 to 1.1, coaxial male rotors forming a combined male rotor common to both sections and coaxial female rotors forming a combined female rotor common to both sections, the value of said ratio being different for the different sections and the lobes of the male rotor of the section with the larger

ratio having a wrap angle exceeding 250°, said section with the larger ratio constituting the low pressure stage and the section with the smaller ratio constituting the high pressure stage.

20. A rotary device comprising a casing structure divided into two succeeding sections, each section having an inlet and outlet ports, and a plurality of rotors mounted to rotate in each of said sections having helical lobes and grooves having less than 325° wrap angle and comprising at least one male rotor and at least one female rotor, the male rotor being provided with at least three convex lobes and intervening grooves located substantially outside the pitch circle of said male rotor and the female rotor being provided with concave lobes and intervening grooves located substantially inside the pitch circle of said female rotor and said male rotor intermeshing with the corresponding female rotor of the set to provide working chambers each including communicating groove portions of two intermeshing rotors and varying in volume as the rotors rotate, the lobes of said male rotor having a lobe angle exceeding 50°, the male rotor of one section having a larger ratio of length to diameter than the male rotor of the other section, the smaller one of said ratios being within the range of 0.5 to 1.1 and the male rotor having the larger ratio having lobes with a wrap angle exceeding 250°, the section with the larger one of said ratios constituting the low pressure stage and the section with the smaller one of said ratios constituting the high pressure stage.

21. A rotary device as claimed in claim 15 in which the high pressure ends of the male coaxial rotors and of the female coaxial rotors, respectively, of the two sections are directed axially toward each other, the main parts of the inlet ports of the two sections being located on the same side of a plane common to the rotor axes.

References Cited in the file of this patent

UNITED STATES PATENTS

Re. 22,201	Lysholm	Oct. 13, 1942
1,701,198	Tift	Feb. 5, 1929
1,930,403	de Bijle	Oct. 10, 1933
2,100,560	Kennedy	Nov. 30, 1937
2,174,522	Lysholm	Oct. 3, 1939
2,248,639	Miksits	July 8, 1941
2,287,716	Whitfield	June 23, 1942
2,369,539	Delamere	Feb. 13, 1945
2,381,695	Sennet	Aug. 7, 1945
2,441,771	Lysholm	May 18, 1948
2,457,314	Lysholm	Dec. 28, 1948
2,459,709	Lysholm	Jan. 18, 1949
2,460,310	Rathman	Feb. 1, 1949
2,477,004	Paget	July 26, 1949
2,481,527	Nilsson	Sept. 13, 1949
2,578,196	Montelius	Dec. 11, 1951

FOREIGN PATENTS

108,467	Sweden	Sept. 14, 1943
264,422	Switzerland	Jan. 3, 1950
384,355	Great Britain	Dec. 8, 1932
588,287	Great Britain	May 19, 1947
626,444	France	May 9, 1927
723,315	Germany	Aug. 3, 1942
853,166	France	Nov. 18, 1939
898,637	France	July 10, 1944