United States Patent [19]

Koch

[11] **3,773,444**

[45] **Nov. 20, 1973**

[54] **SCREW ROTOR MACHINE AND ROTORS THEREFOR**

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[73] Assignee: **Fuller Company,** Catasauqua, Pa.

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[52] **U.S. Cl.** **418/201**

[51] **Int. Cl.** **F01c 1/16,** F04c 1/10, F04c 17/12

[58] **Field of Search** 418/197, 201, 202, 418/203

[56] **References Cited**

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2,174,522	10/1939	Lysholm	418/201
2,473,234	6/1949	Whitfield	418/201
2,622,787	12/1952	Nilsson	418/202
3,414,189	12/1968	Persson	418/201
3,423,017	1/1969	Schibbye	418/201

Primary Examiner—Carlton R. Croyle
Assistant Examiner—John J. Vrablik
Attorney—Jack L. Prather et al.

[57] **ABSTRACT**

A screw rotor machine and rotors therefore wherein the machine includes a casing defining a working space having a pair of parallel intersecting bores with a fluid inlet port and a fluid outlet port. A pair of cooperating rotors are mounted within the working space with each having helical lands and grooves with a wrap angle of less than 360°. The rotors are positioned to cooperate with each other and with the sidewalls of the working space to define chambers for working fluid which move from the fluid inlet to the fluid outlet and decrease in volume as the rotors are rotated to thereby compress fluid confined in the chambers. One of the rotors is of the female type and includes helical lands and intervening grooves which lie substantially completely inside the pitch circle of the rotor. The other rotor is of the male type and includes helical lands and intervening grooves which lie substantially completely outside the pitch circle of the male rotor. In a plane transverse to the longitudinal axes of the rotors, the grooves in the female rotor and the lands on the male rotor are a novel assymetrical design. One flank of the groove in the female rotor and one flank of the land on the male rotor are generated. The other flank of the groove in the female rotor is defined by a circular arc. The major portion of the other flank of the land on the male rotor is defined by a circular arc and a minor portion is line generated. Thus, the leading flank of the male rotor land is substantially circular and the trailing flank is generated.

16 Claims, 12 Drawing Figures

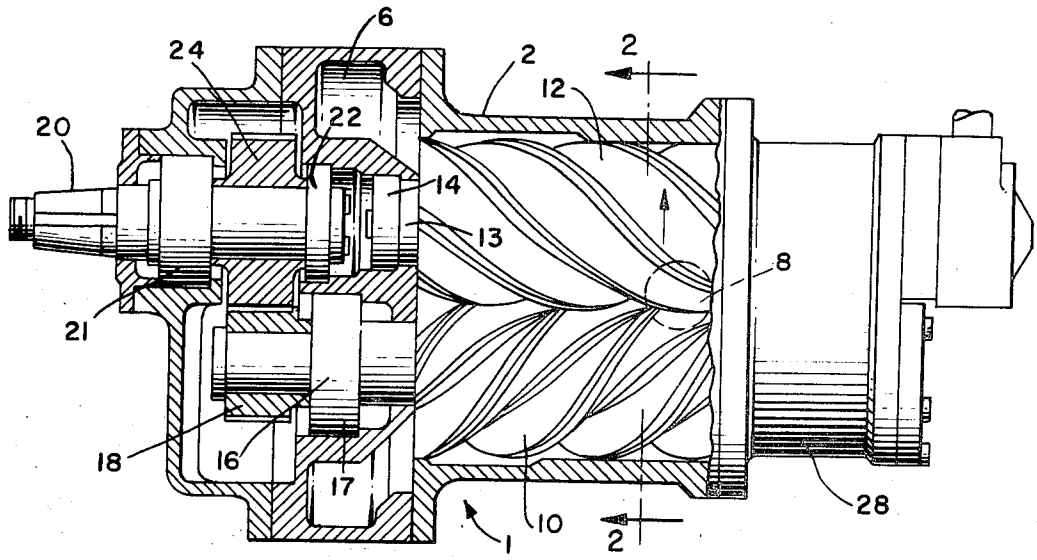


FIG. 1.

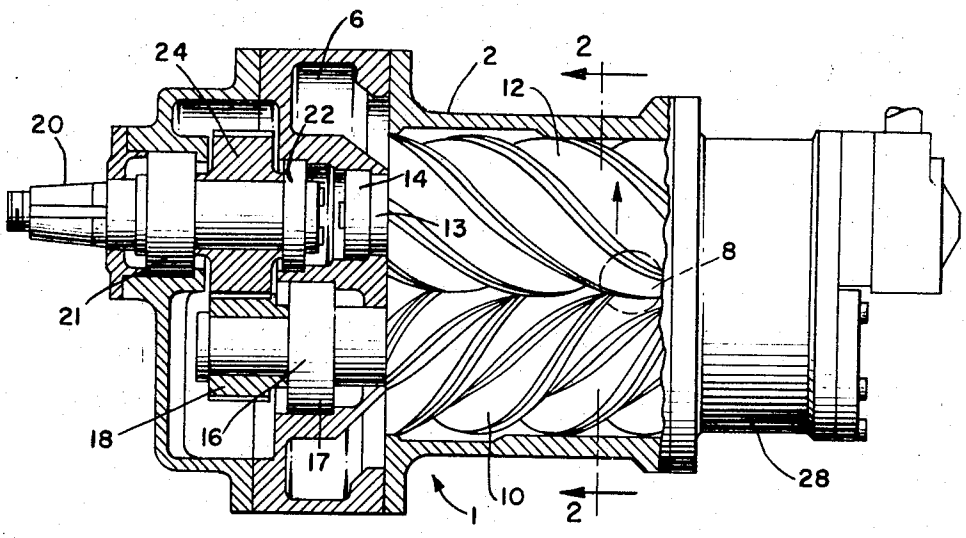

6
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FIG. 2.

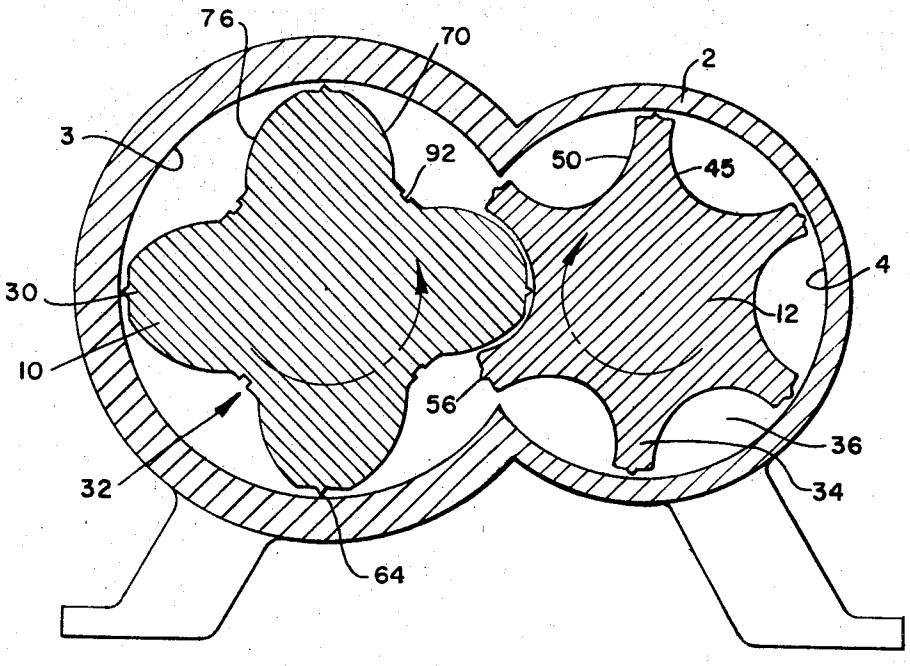

76
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2
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92
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34
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FIG. 3.

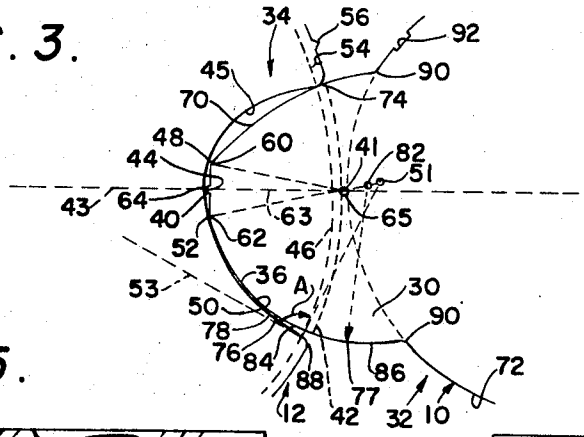


FIG. 5.

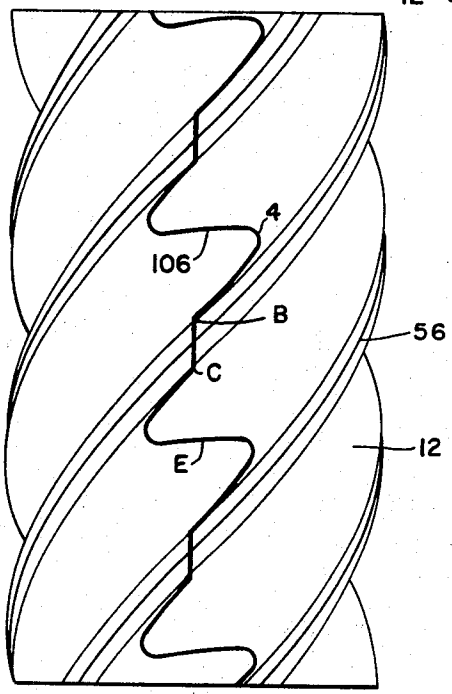


FIG. 6.

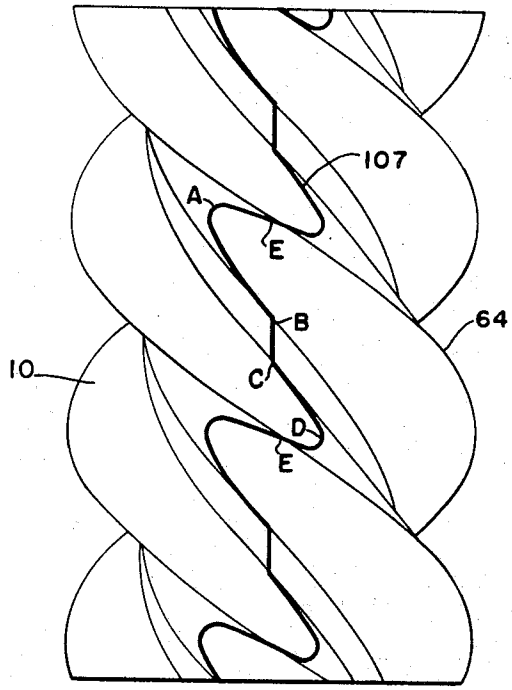


FIG. 5A.

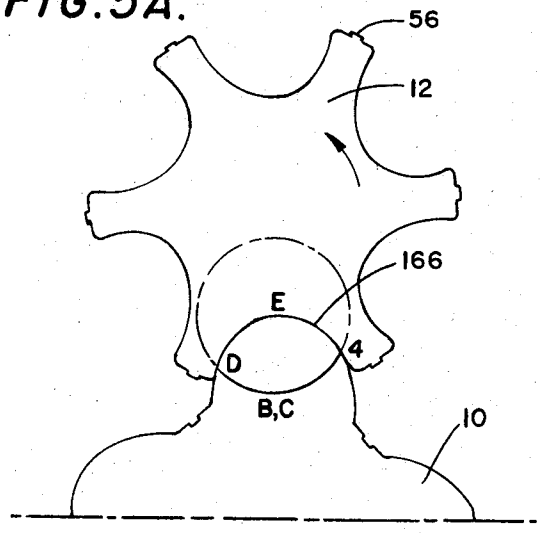


FIG. 6A.

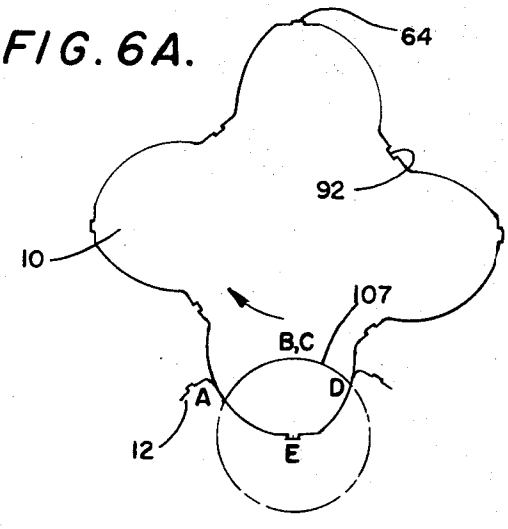


FIG. 4a.

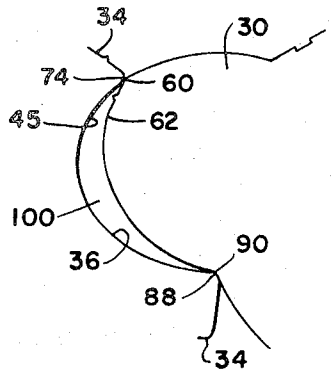


FIG. 4b.

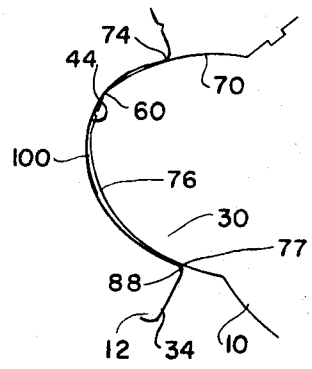


FIG. 4c.

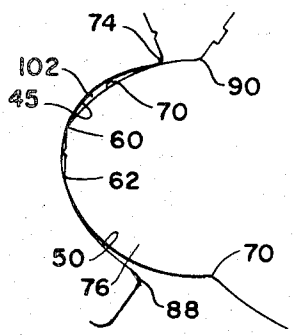


FIG. 4d.

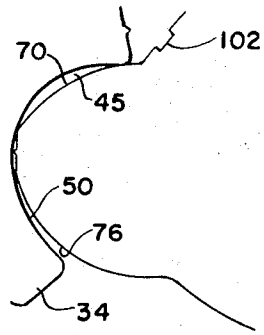
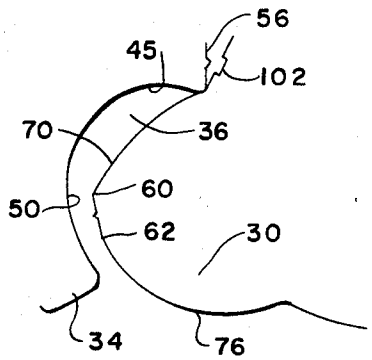


FIG. 4e.



SCREW ROTOR MACHINE AND ROTORS THEREFOR

BACKGROUND OF THE INVENTION

This invention relates to screw rotor machines particularly adapted for use as a fluid compressor such as an air compressor. The invention is particularly directed by a novel rotor profile which is believed to improve machine efficiency.

More particularly, this invention relates to rotary machines which include a housing having at least one pair of intersecting bores therein. Inlet and outlet ports are provided at opposite ends of the casing bores. A rotor is mounted for rotation with each of the bores. One of these rotors may be of the male type which includes a plurality of helical lands and intervening grooves which lie substantially completely outside the rotor pitch circle with the flanks of the lands having a generally convex profile. The other rotor is of the female type and formed so that it includes a plurality of helical lands and intervening grooves which lie substantially completely inside the rotor pitch circle with the flanks of the grooves having a generally concave profile. The lands on the male rotor cooperate with the grooves of the female rotor and the walls of the casing to define chambers for fluid. These chambers may be considered to be chevron shaped. Fluid to be compressed enters the casing bores through the inlet port and is trapped in the chambers formed between the grooves of the female rotor and the wall of the associated casing bore. As the rotors rotate, these chambers move from the inlet port toward the outlet port and the volume of the chambers decreases to thereby compress the gas in the chamber. When communication is established with the outlet port, compressed gas is discharged from the casing.

The construction and design of rotor profiles for the type of machine to which the present invention relates has been the subject of a great deal of consideration. The rotor profile is considered to be the configuration of the rotor in a plane transverse to the longitudinal axis of the rotor. Of particular concern is the configuration of the lands and grooves on the male and female rotors. This work has concentrated on efforts to design a machine with a large displacement and high volumetric efficiency. It is believed that with the design of the present invention, an optimum rotor profile design has been achieved.

Generally, there are considered to be three basic rotor profile designs. These may be classified as the generated profile, the circular profile and the assymetrical profile. The present invention is directed to the assymetrical design.

U.S. Pat. No. 2,287,716 issued to J. E. Whitfield is representative of a generated rotor profile. The details of the generated design need not be considered here as they are generally known to those skilled in the art and may be obtained from the above mentioned U. S. patent. The primary advantage of the generated profile is that this design permits a large displacement volume. The generated profile has the further advantage that no "blow holes" are formed as the rotors rotate. A blow hole allows communication between adjacent volumes being compressed. The fluid being compressed will flow from the high pressure volume to the low pressure volume which will result in a reduction in compressor

efficiency. The lack of such blow holes adds to the efficiency of the generated profile.

The generated profile does, however, have its disadvantages. The generated profile has a long sealing line between the male and female rotors. This long sealing line means that there is a large area through which fluid may leak from the working space directed to the low pressure side of the machine. This leakage will reduce the volumetric efficiency of the machine. An additional disadvantage of this design is that large clearances must be used between the two rotors in order to prevent damage to the rotors and the entire machine in the event the two rotors are not properly timed in relation to each other. Because of the long sealing line, these large clearances will increase the losses due to leakage and effect volumetric efficiency. A further disadvantage of the point generated profile is that large closed pockets are formed between the lands on the male rotor and the grooves in the female rotor. These pockets trap fluid thereby reducing volumetric efficiency of the machine. In addition, as the rotors rotate, this trapped fluid is compressed and produces a negative torque counteracting the rotation of the machine and creating a bending moment on the female lands. This requires that the thickness of these lands be increased thereby reducing the displacement volume of the machine.

U.S. Pat. No. 2,622,787 to H. R. Nilsson is representative of the circular profile design. The circular profile design is generally well known and in popular use in air and gas compressors. The circular profile design has the advantage that no closed pockets are formed and no fluid is trapped in such closed pockets. This permits the lands on the female rotor to be reduced in thickness because negative torque is not created. Because the female rotor lands can be reduced in thickness, the displacement of the machine for any given size can be increased. This design has the further advantage that the sealing line is much shorter than in the generated design. The reduction in length of the sealing line reduces losses and increases volumetric efficiency.

The primary disadvantage of the circular profile design is that it has a small displacement volume when compared with the generated profile. The circular profile has the further disadvantage that large blow holes are formed permitting communication between adjacent volumes being compressed. This reduces the volumetric efficiency of the machine and virtually offsets the gain made by the reduction in the length of the sealing line and the absence of closed pockets.

The assymetrical profile combines the advantages of both the circular profile and the generated profile. In the assymetrical design, one of the flanks of the lands on the male rotor is generated and the other flank is circular. Likewise, one of the flanks of the groove in the female rotor is generated and one of the flanks is circular. The assymetrical profile has the advantage that there is a reduction in the length of the sealing line as compared with the generated profile thereby reducing losses due to friction and leakage associated with a long sealing line. In addition, this profile reduces the size of the trapped pocket as compared with the generated profile and thereby reduces the losses and difficulties associated with a large trapped pocket. With respect to the circular profile, the assymetrical profile has the advantage that there is a substantial reduction in the size of the blow hole and the losses associated with such a

large blow hole. In addition, the displacement volume is substantially larger than with the circular profile although it is smaller than with the generated profile.

The assymetrical profile is per se generally well known and disclosed in U.S. Pat. Nos. 2,174,522 issued to A. Lysholm, 2,473,234 issued to J. E. Whitfield, 3,414,189 issued to J. E. Persson and 3,423,017 issued to L. B. Schibbye. These last two patents are useful in comparing the various rotor profile designs.

It is believed that the assymetrical profiles known prior to the present invention do not fully utilize the advantages of the assymetrical design. For example, in the design shown in U.S. Pat. No. 3,423,017, one embodiment is shown having a groove in the female rotor which includes one flank which is defined by a circular arc and a second flank which is partly point generated by the land on the male rotor lobe and partly a straight line. It is believed that the use of such a straight line portion will cause interference between the lobe on the male rotor and this portion of the groove in the female rotor.

Further, it is stated in U.S. Pat. No. 3,423,017 that by making the angle between the pitch circle and the non-circular flank of the groove in the female rotor less than 90° will increase the size of the blow hole. However, it is believed that the size of the blow hole is controlled by the width or thickness of the lobe on the female rotor and the width and depth of the lobe on the male rotor.

In prior designs, the pockets formed between the land on the male rotor and the groove in the female rotor are believed to be larger than necessary. It is believed that with the design of the present invention, the size of the blow hole and pockets formed between the male and female rotors has been reduced to a minimum. In addition, it is believed that the design of the present invention is such that the sealing line has been reduced in length to reduce friction and fluid losses due to the length of the sealing line. The length of contact between the male rotor and the female rotor has been reduced to a minimum in order to reduce the time that fluid is trapped within a pocket formed between the two rotors to thereby reduce the amount of negative torque. This permits a reduction in the thickness of the female rotor land which permits a greater displacement for a given size machine.

According to the present invention, the diameter of the male rotor is larger than the diameter of the female rotor. Although this concept is known per se, it should be pointed out that this results in a larger displacement volume for a given size machine. There are, of course, limits on the difference in size between the male and female rotor and these are controlled by the thickness of the land on the female rotor and the distance between the centers of the two rotors. The latter is important because of the size of the bearings which must be used to support the rotors. Characteristic is that the land on the male rotor is dimensioned so that if the distance from the male rotor pitch circle to the radially outermost point of the land on the male rotor is designated *b*, and from a line the center of the male rotor passing through the radially outermost point of the male rotor land, the distance to the circumferential edge land on the male rotor is designated *h*, the ratio of *b* to *h* should be less than one and preferably less than 0.9.

SUMMARY

It is the principal object of this invention to provide a screw rotor machine which includes a pair of cooperating rotors designed to provide a rotary machine which is more efficient than rotary machines employing rotor designs of the prior art.

It is a further object of this invention to provide a screw rotor machine which overcomes the disadvantages of prior screw rotor machines.

It is a still further object of this invention to provide a pair of cooperating rotors for a screw motor machine wherein the rotors have a profile which is designed for optimum volumetric efficiency.

It is still another object of this invention to provide a pair of cooperating rotors for a screw rotor machine which reduces the losses in the machine due to leakage and friction caused by contact between the two rotors.

In general, the foregoing and other objects will be carried out by providing a pair of cooperating rotors each having helical lands and intervening grooves with a wrap angle of less than 360° and adapted for rotation about parallel, spaced apart axes within the working space of a screw rotor machine, said working space including inlet and outlet ports and at least two parallel, intersecting bores whereby the rotors cooperate to define with the working space closed chambers for a working fluid moving from the inlet port to the outlet port as the rotors are rotated; one of said rotors being of the female type and formed so that a major portion of its lands and grooves lie inside its pitch circle and the flanks of the grooves have a generally concave profile; the other of said rotors being of the male type and formed so that a major portion of its lands and grooves lie outside its pitch circle and the flanks of the lands have a generally convex profile; (a) the grooves on the female rotor being formed such that in a plane transverse to the longitudinal axes of the rotors, each groove includes: (i) a central portion defined by a circular arc having its center outside the outside diameter of the female rotor and on a radial line from the center of the female rotor through the radially innermost point of the groove, (ii) a first flank extending from said central portion to approximately the pitch circle of the female rotor and being defined by a curve generated by the male rotor, and (iii) a second flank extending from said central portion to approximately the pitch circle of the female rotor and defined by a circular arc having its center outside the outside diameter of the female rotor on a line extending from the radially innermost point of the second flank through the center of the central portion and on the side opposite the second flank of a line from the center of the female rotor through the radially innermost point of said groove; (b) the lands on said male rotor being formed such that in a plane transverse to the longitudinal axes of the rotors, each land includes: (i) a first flank extending from substantially the root diameter of the male rotor toward the outside diameter of the male rotor which is defined by a curve generated by the female rotor, (ii) a second flank extending from substantially the root diameter of the male rotor toward the outside diameter of the male rotor and having a major portion defined by a circular arc having a radius of curvature shorter than the radius of curvature of the second flank of the groove in the female and having its center inside the root diameter of the male rotor on a line extending from the radially out-

ermost point of the second flank through the center of an arc connecting the radially outermost points of the first and second flanks, and (iii) a minor portion of the second flank from the radially innermost point of the major portion of the second flank substantially to the male rotor root diameter being defined by a curve generated by the female rotor.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be described in connection with the annexed drawings wherein:

FIG. 1 is a sectional view of a screw rotor machine according to the present invention;

FIG. 2 is a sectional view on an enlarged scale of the machine taken on the line 2—2 of FIG. 1;

FIG. 3 is a diagrammatic view on an enlarged scale of a portion of the male and female rotors showing the design of the present invention;

FIGS. 4*a* to *e* are diagrammatic views showing a portion of the rotors in different positions of rotation;

FIG. 5 is an elevational view of a female rotor having the sealing line shown thereon;

FIG. 5*a* is a transverse sectional view of the female rotor of FIG. 5;

FIG. 6 is an elevational view of a male rotor having the sealing line shown thereon; and

FIG. 6*a* is a transverse sectional view of the male rotor of FIG. 6.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to the drawings, there is shown a screw rotor machine generally indicated at 1 which includes a casing 2 having a pair of parallel intersecting bores 3 and 4. The casing includes an inlet port 6 for low pressure fluid and an outlet port 8 for high pressure fluid. A pair of rotors generally indicated at 10 and 12 are mounted in the bores 3 and 4, respectively for rotation therein. As shown in FIG. 2, in the preferred form, the bore 3 and rotor 10 have a larger diameter than the bore 4 and rotor 12. The rotor 12 includes a shaft 13 at each end which is suitably mounted in bearings such as that shown at 14 in FIG. 1 to rotatably mount the rotor 12 in the bore 4. The rotor 10 is provided with a long shaft 16 at the inlet end which is rotatably mounted in a bearing 17 to rotatably mount the rotor 10 in the bore 3.

A spur gear 18 is mounted on the shaft 16. A drive shaft 20 adapted to be connected to a source of power such as an electric motor (not shown) extends into the casing 2 at the inlet end and is mounted in bearings 21 and 22. A spur gear 24 is mounted on the shaft 20 and adapted to mesh with the gear 18 so that the rotary movement of the motor drives the rotor 10 through the gears 18 and 24. The machine may be of the oil flooded type which does not employ synchronizing gears, and the rotor 12 may be driven by the rotor 10 through the intermeshing of the two rotors. If the machine employs synchronizing gears, suitable gearing may be provided in the discharge end 28 of the housing 2.

The rotor 10 is of the male type and includes a plurality of spaced apart helical lands 30 and intervening grooves 32 each having a wrap angle of less than 360°. The lands 30 are generally convex in configuration. The rotor 10 is such that the lands 30 and grooves 32 lie substantially completely outside the pitch circle of the rotor 10.

The rotor 12 is of the female type and includes a plurality of helical lands 34 and intervening grooves 36 with a wrap angle of less than 360°. The grooves 36 are generally concave in configuration. The lands 34 and grooves 36 of the rotor 12 lie substantially completely inside the pitch circle of the rotor.

The rotors 10 and 12 mesh so that the lands and grooves of the rotors define with each other and with the inside walls of the casing, chambers which move from the inlet 6 to the outlet 8. These chambers which are generally chevron shaped decrease in volume as they move to thereby compress fluid trapped therein. This is similar to prior screw rotor machines.

Referring to FIG. 3, the rotor profile design of the present invention will be described. The female rotor groove 36 includes a central portion 40 which is defined by a circular arc having its center at 41 which is located outside the outside diameter 42 of the female rotor 12 and on a line 43 from the center of the female rotor through the radially innermost point 44 of the groove 36.

The groove 36 also includes a first flank 45 which extends from the end point 48 of the central portion 40 to approximately the pitch circle 46 of the female rotor 12 and is defined by a curve which is generated by a point 60 on the male rotor land 30.

The female rotor groove 36 also includes a second flank 50 which extends from the end point 52 of the central portion 40 to approximately the pitch circle 46 of the female rotor 12 and is defined by a circular arc having its center 51 outside the outside diameter of the female rotor 12 and on a line extending from the radially innermost point 52 of the second flank 50 through the center 41 of the central portion 40. The center 51 of the second flank 50 lies on the side opposite the second flank 50 of the line 43 extending from the center of the female rotor through the radially innermost point 44 and female groove 36. The center 51 should be located such that a line from the center 51 to the intersection of the flank 50 and the female rotor pitch circle 46 will be at an angle A of 90° or less to a line 53 from the center of the female rotor 12 to the intersection of the flank 50 and the female rotor pitch circle 46. The radius of curvature of the flank 50 is selected so that the flank 50 blends smoothly with the central portion 40 of the groove 36.

The lands 34 of the female rotor include addendums 54 which extend from the pitch circle 46 to the outside diameter 42 of the female rotor 12. From the pitch circle 46 to the outside diameter 42, these addendums 54 are defined by a circular arc having a radius designed to join smoothly with the flank 45 and the flank 50. The arc adjacent the flank 45 has been designated 74 and the arc adjacent the flank 50 has been designated 88. Thus, from approximately the pitch circle 46 to the outside diameter 42 of the female rotor 12, the first flank 45 and the second flank 50 of the groove 36 in the female rotor 12 are defined by a circular arc. These arcs aid in the machining of the groove. The addendums 54 further include a sealing strip 56 best shown in FIG. 5 and adapted to seal against the walls of the bore 4. The addendums 54 are such that the outside diameter of the female rotor is approximately equal to the pitch diameter plus twice the addendums.

The male rotor 10 includes lands 30 each of which include a pair of points 60 and 62 which, when the land 30 is aligned with a groove 36 so that a line connecting

the center of the female rotor **12** and the center of the male rotor **10** extends through the radially innermost point **44** of the female groove **36**, these points **60** and **62** are aligned with the points **48** and **52** on the adjacent female groove **36**. The distance between the points **60** and **62** is selected by the designer taking into consideration various manufacturing functions and the width of the sealing line required. A sealing line **64** is positioned between the points **60** and **62** and is best shown in FIG. **6**. The points **60, 64** and **62** may be connected by an arc having its center **65** which coincides with the center **41** when the male land **30** and female groove **36** are aligned. Thus, the point **65** is located outside the outside diameter of the female rotor on a radial line **63** extending from the center of the male rotor **10** through the radially outermost point **64** of the land **30**. The point **65** is also located inside the male rotor root diameter **72**.

The male rotor land **30** includes a first flank **70** which extends from the point **60** substantially to the root diameter **72** of the male rotor **10**. This flank **70** is defined by a curve which is line generated by the first flank of the female groove. More specifically, the flank **70** is line generated by a portion and the arcuate surface **74** on the addendum **54** of the female rotor **12**.

The land **30** on the male rotor **10** includes a second flank **76** which extends from the point **62** to approximately the root diameter **72** of the male rotor. The major portion **78** of the flank **76** is defined by a circular arc, and a minor portion **86** is defined by a curve which is line generated by the female groove. From the point **62** to approximately the point **77**, the major portion **78** of the flank **76** is defined by a circular arc having its center inside the root diameter **72** of the male rotor **10** and on a line extending from the radially innermost point **62** of the second flank **76** through the center **65** of an arc connecting the points **60, 64** and **62**. When the male rotor land **30** and the female rotor groove **36** are aligned as shown in FIG. **3**, the center **82** is positioned on a line connecting the point **65** and the center **51** and is positioned between these two points. The point **82** lies on the side opposite the flank **76** of a line **63** connecting the center of the male rotor and the radially outermost point **64** of the male rotor **10**. The radius of curvature of the arc **78** is shorter than the radius of curvature of the second flank **50** of the female groove **36**. This shorter radius serves to provide a clearance **84** between the second flank **76** of the male land **30** and the second flank **50** of the female groove **36**. The radius of curvature of the major portion **78** is selected so that this major portion blends smoothly with an arc connecting the points **62, 64** and **60**.

From the point **77** to approximately the root **72** of the male rotor **10**, the second flank **76** is line generated. This minor portion **86** is line generated by a portion of the arcuate surface **88** of the addendum **54** adjacent the second flank **50** of the female groove **36**.

Where the male land **30** joins the root **72** of the male rotor **10**, a circular arc is used. The radius of curvature of this arc is preferably approximately 80 percent of the radius of the circular arcs **74** and **88** of the addendums **54**. By rounding off the edges of the addendums **54** on the female rotor, the requirement of machining a complex and expensive undercut in the root of the male rotor is avoided.

The root of the male rotor **10** is provided with helical grooves **92** to receive the sealing strip **56** on the female rotor lands **34**.

The advantages of the profile design of the present invention is best seen by referring to FIGS. **4***a* to *e*. In FIG. 4a, it will be seen that there is contact between the male rotor land **30** and the female rotor groove **36** at points **60** and **90** on the male land **30** and **74** and **88** on the female groove **36**, respectively. This forms a pocket **100** of trapped fluid which like prior designs, will be compressed as the rotors rotate from the position of FIG. **4***a* to the position of FIG. **4***b*. The formation of the pocket **100** is detrimental to compressor efficiency. This pocket creates excessive pressure, bearing loads, noise and reduces machine capacity if not kept to a minimum. In addition, this pressure will create a negative torque and produce a bending moment on the female rotor land **34**. However, it is believed that with the design of the present invention, this pocket will be vented before the fluid therein is compressed to the extent that it affects the machine and will be such that the lands **34** can be made thinner thereby permitting an increase over prior designs in fluid displacement for a given size machine. If desired, the negative torque can also be reduced by changing the pitch diameter of the rotor.

As the rotors **10** and **12** rotate from the position of FIG. **4***a* to the position of FIG. **4***b*, contact will be made between the point **74** on the female rotor and the first flank **70** of the male land **30**. Contact is established between the point **60** on the land **30** and the first flank **45** of the female rotor groove **36**. In this position, there is also contact between the surface **88** and the second flank **76** of the male rotor land **30** at approximately the point **77** on the flank **76**. FIG. **4***b* is approximately the last point of contact between the surface **88** and the flank **76**. As a result, this is the maximum closing of the pocket **100** and the maximum compression of air trapped in the pocket **100**.

As the rotors rotate from the position of FIG. **4***b* to the position of FIG. **4***c*, there is no longer contact between the second flank **50** of the female groove and the second flank **76** of the male land, and the pocket **100** is vented to release any fluid trapped therein. Contact is lost between surface **88** and flank **76** about 3° of male rotor rotation after contact is made between point **62** and flank **50**. A pocket **102** is formed between the first flank **45** of the female rotor groove and the first flank **70** of the male rotor land, as defined by contact between the surfaces **74** and **70** and **60** and **45**. As the rotors **10** and **12** continue to rotate, the pocket **102**, initially formed in the position of FIG. **4***b*, will increase in size until it is at a maximum as shown in FIG. **4***d*. This pocket **102** will result in a small "blow hole" but it is believed that with the design of the present invention, it will be considerably smaller than with prior designs. In FIG. **4***d*, the sole remaining contact between the male and female rotors is at the first flank of each. This sole contact reduces the sealing line length and associated losses due to leakage and friction. In FIG. **4***e*, contact is being lost by the male rotor **10** and female rotor **12** and the cycle will begin anew with another set of lands and grooves.

The reduction in the sealing line length of the present invention is shown in FIGS. **5**, **5***a*, **6** and **6***a* where the sealing line is shown at **106** and **107**, respectively. The letters A, B, C, D and E represent various points along

the sealing lines. From these figures, it will be seen that there are substantial reductions in sealing line length if they are compared with FIGS. 6*a* and *b* of the U.S. Pat. to Persson, No. 3,414,189. However, the saeling line length has not been reduced to such an extent that volumetric efficiency is reduced by a decrease in the size of the displacement of the machine.

With the design of the present invention, good volumetric efficiencies are obtained while volume of the trapped pocket **100** is substantially reduced to reduce the losses associated therewith. The sealing line length has been shortened to reduce losses associated with the sealing line. The blow hole size has been reduced to a minimum to reduce losses associated with a blow hole. The displacement of the machine is large due to the design of the rotor profile. In view of these advantages, it is apparent that the objects of this invention have been carried out.

It is intended that the foregoing description be merely that of a preferred embodiment and that the invention be limited solely by that which is within the scope of the appended claims.

I claim:

1. A pair of cooperating rotors each having helical lands and intervening grooves with a wrap angle of less than 360° and adapted for rotation about parallel, spaced apart axes within the working space of a screw rotor machine, said working space including inlet and outlet ports and at least two parallel, intersecting bores whereby the rotors cooperate to define with the working space closed chambers for a working fluid moving from the inlet port to the outlet port as the rotors are rotated; one of said rotors being of the female type and formed so that a major portion of its lands and grooves lie inside its pitch circle and the flanks of the grooves have a generally concave profile; the other of said rotors being of the male type and formed so that a major portion of its land and grooves lie outside its pitch circle and the flanks of the lands have a generally convex profile;

a the grooves on the female rotor being formed such that in a plane transverse to the longitudinal axes of the rotors, each groove includes:

i. a central portion defined by a circular arc having its center outside the outside diameter of the female rotor and on a radial line from the center of the female rotor through the radially innermost point of the groove,

ii. a first flank extending from said central portion to approximately the pitch circle of the female rotor and being defined by a curve generated by the male rotor, and

iii. a second flank extending from said central portion to approximately the pitch circle of the female rotor and defined by a circular arc having its center outside the outside diameter of the female rotor on a line extending from the radially innermost point of the second flank through the center of the central portion and on the side opposite the second flank of a line from the center of the female rotor through the radially innermost point of said groove;

b. the lands on said male rotor being formed such that in a plane transverse to the longitudinal axes of the rotors, each land includes:

i. a first flank extending from substantially the root diameter of the male rotor toward the outside diameter of the male rotor which is defined by a curve generated by the female rotor,

ii. a second flank extending from substantially the root diameter of the male rotor toward the outside diameter of the male rotor and having a major portion defined by a circular arc having a radius of curvature shorter than the radius of curvature of the second flank of the groove in the female and having its center inside the root diameter of the male rotor on a line extending from the radially outermost point of the second flank through the center of an arc connecting the radially outermost points of the first and second flanks, and

iii. a minor portion of the second flank from the radially innermost point of the major portion of the second flank substantially to the male rotor root diameter being defined by a curve generated by the female rotor.

2. A pair of cooperating rotors according to claim **1** wherein when the relative rotational position of the rotors is such that a line connecting the centers of the male and female rotors passes through the radially innermost point on a groove in the female rotor and the radially outermost point of an adjacent land on the male rotor, the center of the major portion of the second flank of the land on the male rotor is on a line extending from the radially innermost point of the second flank of the groove in the female rotor to the center of the second flank of the groove in the female rotor and positioned between the center of the central portion of the groove in the female rotor and the center of the second flank of the groove in the female rotor.

3. A pair of cooperating rotors according to claim **2** wherein the lands on the female rotor include an addendum which lies outside the pitch circle of the female rotor and said addendum from the pitch circle to the outside diameter of the female rotor is defined by a circular arc so that from the pitch circle of the female rotor to the outside diameter of the female rotor, the first and second flanks of the grooves in the female rotor are defined by a circular arc.

4. A pair of cooperating rotors according to claim **3** wherein the first flank of the land on the male rotor is line generated by a portion of the circular arc extending from the pitch circle to the outside diameter of the first flank of the groove in the female rotor.

5. A pair of cooperating rotors according to claim **4** wherein the first flank of the groove in the female rotor is generated by the radially outermost point of the first flank of the land on the male rotor.

6. A pair of cooperating rotors according to claim **5** wherein the remaining portion of the second flank of the land on the male rotor is line generated by a portion of the circular arc extending from the pitch circle to the outside diameter of the second flank of the groove in the female rotor.

7. A pair of cooperating rotors according to claim **6** wherein the radially outermost points of the first and second flanks of the land on the male rotor are spaced apart a distance such that they substantially coincide with the radially innermost points of the first and second flanks of the groove in the female rotor and a sealing strip is positioned between the radially outermost points of the first and second flanks of the land on the male rotor and an arc connecting the radially outermost points will contact the sealing strip.

8. A pair of cooperating rotors according to claim 7 wherein the first and second flanks of the land on the male rotor include a circular arc connecting the flanks and the root of the male rotor and such arc has a radius of curvature approximately 80 percent of the radius of curvature of the circular arc defining the area from the female rotor pitch circle to the female rotor outside diameter of each flank of the groove in the female rotor.

9. A screw rotor machine apparatus for compressing an elastic working fluid comprising:

a. a casing defining a working space including at least a pair of intersecting bores having parallel axes and a fluid inlet port and a fluid outlet port; and

b. at least a pair of cooperating rotors each adapted to rotate in one of the bores of said casing and each having helical lands and intervening grooves with a wrap angle of less than 360°;

c. said rotors being positioned to cooperate with each other and with said working space closed chambers for the working fluid moving from the inlet to the outlet as the rotors are rotated;

d. one of said rotors being of the male type and formed so that a major portion of its lands and grooves lie outside its pitch circle and the flanks of the lands have a generally convex profile;

e. the other of said rotors being of the female type and formed so that a major portion of its lands and grooves lie inside its pitch circle and the flanks of the grooves have a generally concave profile;

f. the grooves of the female rotor being formed such that in a plane transverse to the longitudinal axis of the female rotor, each groove includes:

i. a central portion defined by a circular arc having its center outside the outside diameter of the female rotor and on a radial line from the center of the female rotor through the radially innermost point of the groove,

ii. a first flank extending from said central portion to approximately the pitch circle of the female rotor and being defined by a curve generated by the male rotor, and

iii. a second flank extending from said central portion to approximately the pitch circle of the female rotor and defined by a circular arc having its center outside the outside diameter of the female rotor on a line extending from the radially innermost point of the second flank through the center of the central portion and on the side opposite the second flank of a line from the center of the female rotor through the radially innermost point of said groove;

g. the lands on said male rotor being formed such that in a plane transverse to the longitudinal axis of the male rotor, each land includes:

i. a first flank extending from approximately the root diameter of the male rotor toward the outside diameter of the male rotor which is defined by a curve generated by the female rotor,

ii. a second flank extending from approximately the root diameter of the male rotor toward the outside diameter of the male rotor and having a major portion defined by a circular arc having a radius of curvature shorter than the radius of curvature of the second flank of the groove in the female and having its center inside the root diameter of the male rotor on a line extending from the radially outermost point of the second flank through the center of an arc connecting the radially outermost points of the first and second flanks, and

iii. a minor portion of the second flank from the radially innermost point of the major portion of the second flank substantially to the male rotor root diameter being defined by a curve generated by the female rotor.

10. A screw machine apparatus according to claim 9 wherein when the relative rotational position of the rotors is such that a line connecting the centers of the male and female rotors passes through the radially innermost point on a groove of the female rotor and the radially outermost point on an adjacent land on the male rotor, the center of the major portion of the second flank of the land on the male rotor is on a line extending from the radially innermost point of the second flank of the groove in the female rotor to the center of the second flank of the groove in the female rotor and positioned between the center of the central portion of the groove in the female rotor and the center of the second flank of the groove in the female rotor.

11. A screw machine apparatus according to claim 10 wherein the lands on the female rotor include an addendum which lies outside the pitch circle of the female rotor and said addendum from the pitch circle to the outside diameter of the female rotor is defined by a circular arc so that from the pitch circle of the female rotor to the outside diameter of the female rotor, first and second flanks of the grooves in the female rotor are defined by a circular arc.

12. A screw machine apparatus according to claim 11 wherein the first flank of the land of the male rotor is line generated by a portion of the circular arc extending from the pitch circle to the outside diameter of the first flank of the groove in the female rotor.

13. A screw machine apparatus according to claim 12 wherein the first flank of the groove in the female rotor is generated by the radially outermost point of the first flank of the land on the male rotor.

14. A screw machine apparatus according to claim 13 wherein the remaining portion of the second flank of the land on the male rotor is line generated by a portion of the circular arc extending from the pitch circle to the outside diameter of the second flank of the groove in the female rotor.

15. A screw machine apparatus according to claim 14 wherein the radially outermost points of the first and second flanks of the land on the male rotor are spaced apart a distance such that they substantially coincide with the radially innermost points of the first and second flanks of the groove in the female rotor and a sealing strip is positioned between the radially outermost points of the first and second flanks of the land on the male rotor and an arc connecting the radially outermost points will contact the sealing strip.

16. A screw machine apparatus according to claim 15 wherein the first and second flanks of the land on the male rotor include a circular arc connecting the flanks and the root of the male rotor which has a radius of curvature approximately 80 percent of the radius curvature of the circular arc defining the area from the female rotor pitch circle to the female rotor outside diameter of each flank of the groove in the female rotor.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,773,444 Dated November 20, 1973

Inventor(s) Hans-Juergen Koch

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 9, line 38, "land" should read --lands--.

Column 11, line 19, after "space" insert --to define--.

Signed and sealed this 23rd day of April 1974.

(SEAL)
Attest:

EDWARD M.FLETCHER,JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents