Dec. 28, 1948. A. LYSHOLM 2,457,314

ROTARY SCREW WHEEL DEVICE

Filed Aug. 12, 1943

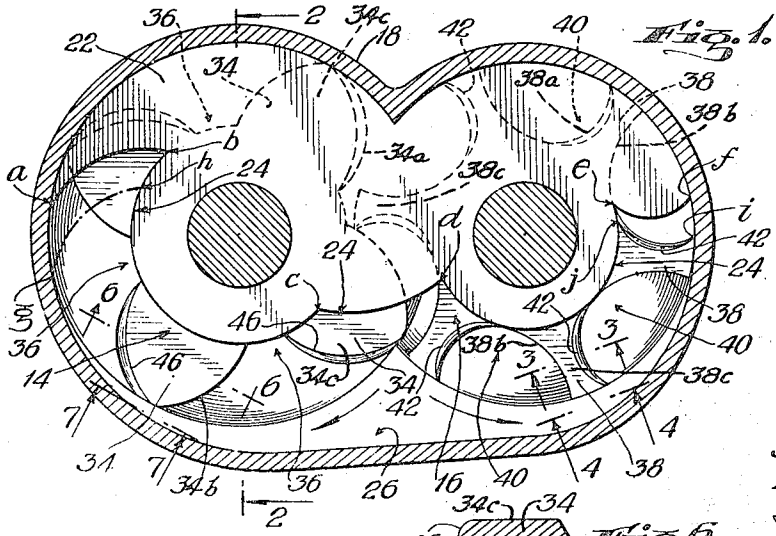


Fig. 1.

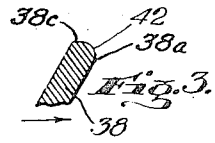


Fig. 3.

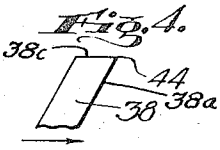


Fig. 4.

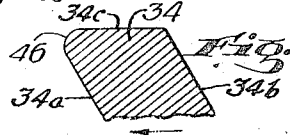


Fig. 6.

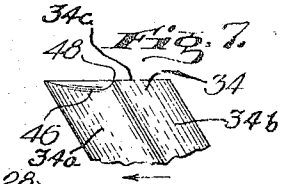


Fig. 7.

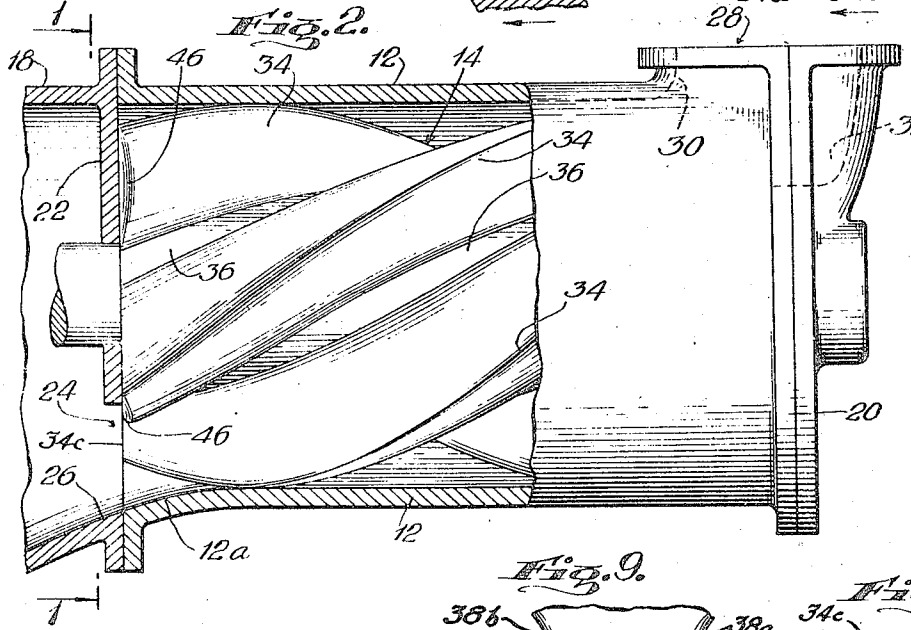


Fig. 2.

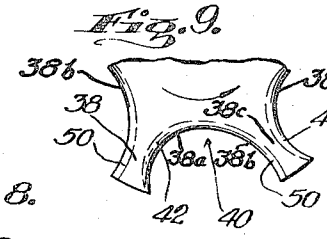


Fig. 9.

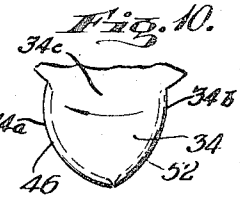


Fig. 10.

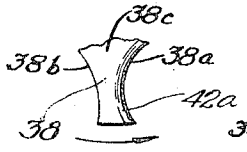


Fig. 5.

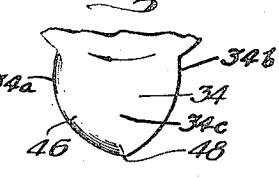


Fig. 8.

INVENTOR
Alf Lysholm
BY
Jarvis C. Marble
his ATTORNEY

Patented Dec. 28, 1948 2,457,314

UNITED STATES PATENT OFFICE

2,457,314

ROTARY SCREW WHEEL DEVICE

Alf Lysholm, Stockholm, Sweden, assignor, by mesne assignments, to Jarvis C. Marble, New York, N. Y., Leslie M. Merrill, Westfield, N. J., and Percy H. Batten, Racine, Wis., as trustees

Application August 12, 1943, Serial No. 498,321

7 Claims. (Cl. 230—143)

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The present invention relates to rotary screw wheel devices of the kind in which intermeshing rotors having helical lands and grooves cooperate with each other and with an enclosing casing to form working spaces or chambers which vary in volume as the rotors revolve. More particularly, the invention relates to such devices in which the size and location of the inlet and outlet ports are so related to the number, length and helix angle of the lands and grooves that the working spaces change in volume after passing out of communication with the inlet port and before coming into communication with the outlet port. Still more particularly, the invention relates to such devices of the type disclosed in my prior Patents Nos. 2,174,522 and 2,243,874, granted October 3, 1939, and June 3, 1941, respectively, especially but not necessarily when operated as compressors.

Devices of the character under consideration are ordinarily operated at high peripheral speeds with a small clearance or "space packing" between the peripheries of the rotors and the interior of the casings and with the rotors geared together by suitable timing or synchronizing gears to maintain clearance between the walls of the intermeshing lands and grooves. Operation at such high speeds, which are required in order to reduce to an acceptable percentage the losses due to leakage through the "space packing," involves numerous forms of dynamic losses, which must be minimized if acceptable volumetric and adiabatic efficiencies are to be attained.

Heretofore much attention has been devoted to port locations and configuration in order to reduce such losses but there previously has apparently been a lack of appreciation of the existence and magnitude of certain losses which it is the general object of the present invention to reduce, nor has there been an appreciation of the manner in which such losses can most effectively be reduced.

The losses with which the present invention is concerned are reduction losses, that is, the losses incident to the flow of fluid through and from the inlet passage into the rapidly moving grooves of the rotors, and in order to minimize such losses this invention contemplates the provision of novel rotor structure, preferably combined with inlet porting constructed and arranged in accordance with certain principles hereinafter more fully pointed out.

Since the invention is particularly advantageous in its use when applied to compressors of the type disclosed in my aforementioned patents,

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it will be described herein, by way of example but without limitation, as incorporated in such a device, and for a better understanding of the more detailed nature and objects of the invention, reference may best be had to the ensuing portion of this specification, taken in conjunction with the accompanying drawings, in which suitable embodiments of the invention are described.

In the drawings;

Fig. 1 is an end view taken on the line 1—1 of Fig. 2 of a device embodying the invention;

Fig. 2 is a side view partly in section and partly in elevation of the device seen in Fig. 1;

Figs. 3 and 4 are fragmentary sections taken on the respectively numbered section lines of Fig. 1;

Fig. 5 is a fragmentary end view showing a different form of one of the rotors shown in Fig. 1;

Figs. 6 and 7 are fragmentary sections taken on the respectively numbered section lines of Fig. 1;

Fig. 8 is a fragmentary view similar to Fig. 5 showing another form of one of the rotors shown in Fig. 1; and

Figs. 9 and 10 are fragmentary views showing respectively still other forms of rotors of the kind shown in Fig. 1.

Referring now more particularly to Figs. 1 and 2, the compressor comprises a housing consisting of a central barrel portion 12 providing parallel intersecting cylindrical bores for the reception of rotors 14 and 16. At its ends the barrel part of the casing is closed by end cover members 18 and 20, these latter members providing suitable bearings (not illustrated) for rotatably mounting the rotors within the casing. The rotors are interconnected by suitable timing gears for maintaining them in proper phase relation. Since this gearing forms no part of the present invention it is not illustrated.

The end cover 18 at the inlet end of the compressor provides an end wall 22 for closing the ends of the grooves in the rotors which form the working spaces. The inlet end wall 22 is cut away to provide an inlet port 24, a suitable outline for this port being indicated in Fig. 1 by the line *a—b—c—d—e—f*. Also, the lower portion of the member 18 and the adjacent portion 12*a* of the barrel 12 are shaped to provide a smoothly contracting inlet surface 26 for admitting the fluid to be compressed to the ends of the working spaces.

The inherent nature of the operation of the compressor is such that the discharge or outlet

port is located diagonally of the casing with respect to the inlet port and in the present instance the outlet is indicated at **28**, the port communicating with the working spaces partially in radial direction as indicated at **30**, and partially in axial direction as indicated at **32**. Insofar as the present invention is concerned, the specific outline of the exhaust port is not critical and has not been shown. For a suitable outline of exhaust port, which it is to be noted is usually preferably of the combined radial and axial outlet form, reference may be had to my previously granted patents hereinbefore referred to.

The general form of compressor just described and its mode of operation are well known and need not be described herein in detail except insofar as the following factors of the operation are involved. The fluid inducted, which may be assumed to be air, enters the working spaces formed by the grooves through the inlet port and fills these grooves from end to end. The grooves are enclosed at their ends by the respective end walls of the casing, and as the rotors revolve so that the filled grooves pass out of communication with the inlet port, the trapped volumes of air are carried peripherally, while as rotation of the rotors continues the intermeshing lands and grooves progressively decrease the volumes of the spaces or chambers formed in the grooves between the discharge ends of the rotors and the places where the respective grooves are closed by the cooperating lands. The places where the grooves are blocked or closed by the lands move progressively toward the discharge end of the rotors as the latter revolve and compression is effected in the spaces until they come into communication with the outlet port. The number and pitch of the lands and grooves and the size and location of the outlet port will determine the compression ratio. In the example shown, one rotor has four lands while the other has six, the pitch of the lands on the two rotors being correspondingly different and the rotors being geared to rotate at correspondingly different speeds. Insofar as this invention is concerned the number of lands on the two rotors may be the same or may have a different relationship. Ordinarily it is desirable to effect compression within the working chambers, but if desired, the extent of the inlet and outlet ports for a given set of rotors may be such that the grooves communicate with the outlet port at substantially the same time that they pass out of communication with the inlet port so that the device will operate merely as a positive displacement blower.

In the operation of a device of this kind, it is, of course, highly desirable from a volumetric efficiency standpoint to fill the grooves with the maximum weight of fluid, and to this end, when the device is of the high speed type involving high induction velocities, it is desirable to provide for essentially axial induction of air into the grooves. For this reason, an inlet port providing for substantially axial flow of air to the rotors such as herein disclosed is preferably employed. Also, it is to be noted that with substantially axial admission and high speed operation, which results in relatively high inlet velocity, a definite "ramming" effect is obtainable since the high velocity axially flowing air flows through the grooves until it is stopped by impact against the end wall at the discharge end of the compressor. This sets up pressure waves which travel back along the lengths of the respective grooves, and since this takes an appreciable length of time, the inlet port **24** is advantageously made of such extent peripherally that the port remains in communication with the grooves for a predetermined length of time after they have been fully opened from end to end by the rotation of the rotors. In order to illustrate this, the dotted line *g*—*h* indicates the position which the end edge of the inlet port would have if the port closed the grooves of the rotor **14** at the time when the grooves had been fully opened to the inlet port in a typical design, the port area *a*—*b*—*h*—*g* representing the amount of port area constituting what may be termed overlap for ramming purposes. Similarly, the area *e*—*f*—*i*—*j* represents the port area for ramming overlap of the grooves of rotor **16**.

While heretofore the fact has been recognized that throttling and related losses at the inlet due to acceleration to high velocity and change in the direction of flow of the fluid into the grooves of the rotors are encountered, the magnitude of such losses has not heretofore been fully appreciated, nor has the nature of the causes of this loss heretofore been fully understood. Such losses have been reduced to some extent by proper formation of the channel leading to the inlet port, but I have discovered that even with the most favorable configuration of inlet port passages from a flow standpoint, a relatively large induction loss remains which can be materially reduced by properly modifying the inlet ends of the rotor lands.

In the type of device illustrated herein, in which axial compression against an end wall is effected, two basic rotor characteristics are involved. The first is that the major portion of the lands of one rotor, which is conveniently referred to as the male rotor, lie outside the pitch circle of the rotor while the major portion of the depth of the lands of the other rotor, which is conveniently referred to as the female rotor lie inside the pitch circle of the rotor, the terms male and female being employed to distinguish these types of rotors from the twin rotors of the Roots type, or the like, which are characterized by pitch circles lying midway between the tips and roots of the lands. Also, in the present type device, if unnecessary and useless rotor lengths of given rotor diameter are to be avoided, the helix angle of the lands and grooves has a substantial value ordinarily lying within the range of which the upper limit is of the order of 45° and the lower limit of which is of the order of 25°. Also, the higher the speed of operation of the device, the higher the helix angle is more likely to be the most suitable design.

The substantial helix angle employed creates a dynamic induction loss due to the flow of fluid from the inlet into the rotor grooves which apparently can be affected only to a certain extent by design of the inlet port passage. I have discovered, however, that this loss can be very substantially reduced, and to an entirely unexpected extent, by chamfering the edges of the rotor lands at their inlet ends, preferably in the manner now to be described.

In the example illustrated, the male rotor **14** is provided with lands **34** defining between them grooves **36**, while the female rotor is provided with lands **38** defining between them grooves **40**. The lands **34** comprise convexly curved sides or flanks **34***a* and **34***b*, and have transversely extending inlet end surfaces **34***c*, while the lands **38** have concavely curved flanks **38***a* and **38***b* with transverse end surfaces **38***c*. The profiles of the

lands 34 and 38 are preferably, but not necessarily, as disclosed in my prior Patent No. 2,174,522, previously mentioned. In accordance with this invention the lands of the female rotor are chamfered along the lines of juncture between the leading flanks 38a and the inlet end surfaces 38c, as indicated in Figs. 1, 3 and 4. Preferably these chamfers are formed by rounding to a convex cross section as indicated at 42, and further the chamfering is not necessarily carried to the apex of the land, so that the apex edges 44 of the leading flanks extend without change of helix angle up to the inlet end surface of the land as seen in Fig. 4. The reason for this is that if the chamfer is continued to the apex of the land so that the edge 44 is substantially rounded off or beveled at the inlet end, an opening introducing a leakage loss will be opened up in certain positions of the rotors.

In other instances where extremely high speed operation is involved, the chamfering of the inlet ends of the rotor lands may be carried to the apex of the land, as indicated at 42a in Fig. 5, but in order to avoid losses of undesirable magnitude the apex edges of the leading flanks of the lands should extend to or closely adjacent to the inlet end surfaces of the lands without change of helix angle.

As will be seen from Fig. 1 the preferred construction results in the formation of generally crescent shaped chamfers having less lateral extent at the apexes and roots of the lands than at intermediate points.

Advantageously, the lands 34 of rotor 14 are also chamfered as shown in Figs. 1, 6 and 7, by rounding, as shown at 46 in Fig. 6, with the chamfering reduced toward the apex of the land so that the apex edge 48 extends without change of helix angle up to the inlet end surface of the rotor. As in the case of the chamfering of the female rotor, the specific conditions of operation of the device may be such that it will be desirable to carry the chamfering 46 to the apex line 48 of the land as shown in Fig. 8.

Also in some instances it may be desirable to chamfer along the lines of juncture of both the leading and the trailing flanks of the lands with the inlet end surfaces thereof as shown in Figs. 9 and 10 respectively. In Fig. 9 the lands 38 of the female rotor are shown as having chamfers 42 joining the leading flanks with the inlet end surfaces and chamfers 50 joining the trailing flanks with the inlet end surfaces. In Fig. 10 the male land 34 is shown as with its leading flank chamfered at 46 and with its trailing flanks chamfered at 52. Experience has shown that from the standpoint of the extent to which induction loss is reduced by chamfering at the inlet ends of the rotor lands the maximum effect is obtained by chamfering of the leading flanks of the female rotor lands, which is the most important modification of the form of the land to make. Chamfering of the leading flank of the male lands has a further loss reducing effect, but this appears to be not of the same order of magnitude as the effect of chamfering the leading flank of the female land at the inlet.

As noted above, the chamfering may be carried around the perimeter of the grooves so that both the leading and trailing flanks of the lands are chamfered, but the chamfering of the trailing flanks of the lands appears to be relatively unimportant as compared with the chamfering of the leading flanks.

While but one form of device has been described in detail by way of example for the purpose of explaining the nature of the invention, it will be apparent to those skilled in the art that the invention may be applied to advantage in other forms of structures, and it is accordingly to be understood that the scope of the invention includes all structures falling within the terms of the appended claims.

What is claimed:

1. A rotary device comprising a casing and intermeshing male and female rotors mounted for rotation in said casing, said rotors being provided with helical lands and grooves, the lands of the male rotor having convexly curved flanks at least the major portions of which lie outside the pitch circle of the rotor, the lands of the female rotor having concavely curved flanks at least the major portions of which lie inside the pitch circle of the rotor and the lands of both of said rotors having inlet end surfaces extending transversely of the axes of the rotors, said casing having inlet and outlet ports and including a ported inlet end wall cooperating with said inlet end surfaces of the lands to alternately open and close the inlet end of each of said grooves as the rotors revolve, the inlet end surface and the leading flank of each land of at least one of said rotors being joined by a chamfer extending generally transversely of the axis of rotation of the rotor along the curved line of juncture of said inlet end surface with said leading flank and the apex edge of said leading flank extending without change of helix angle from a point on the rotor remote from the inlet end of the rotor to or closely adjacent to the inlet end surface of the land.

2. A rotary device as set forth in claim 1 in which said chamfers are provided in the female rotor.

3. A rotary device as set forth in claim 1 in which said chamfers are provided in both the male and female rotors.

4. A rotary device as set forth in claim 1 in which said chamfers are of generally crescent shape having less lateral extent at the root and apex portions of the lands than at the radially intermediate portions of the lands.

5. A rotary device as set forth in claim 1 in which said casing is provided with an inlet passage for directing fluid through said ported inlet end wall into said grooves essentially in axial direction and said inlet port has a peripheral extent providing communication between said passage and said grooves until after the rotors have reached positions of rotation in which the grooves are opened up to their outlet ends.

6. A rotor for the kind of device described having helical lands and grooves and transversely extending inlet and outlet end surfaces, said lands having concavely curved flanks at least the major portions of which lie inside the pitch circle of the rotor and inlet end surfaces forming portions of the transversely extending inlet end surface of the rotor, the inlet end surface and the leading flank of each land, considered in the direction in which the rotor is intended to turn, being joined by a chamfer extending generally transversely of the axis of rotation of the rotor along the curved line of juncture of said inlet end surface with said leading flank and the apex edge of said leading flank extending without change of helix angle from a point on the rotor remote from the inlet end of the rotor to or closely adjacent to the inlet end surface of the land.

7. A rotor for the kind of device described having helical lands and grooves and transversely extending inlet and outlet end surfaces, said lands having convexly curved flanks at least the major portions of which lie outside the pitch circle of the rotor and inlet end surfaces forming portions of the transversely extending inlet end surface of the rotor, the inlet end surface and the leading flank of each land, considered in the direction in which the rotor is intended to turn, being joined by a chamfer extending generally transversely of the axis of rotation of the rotor along the curved line of juncture of said inlet end surface with said leading flank and the apex edge of said leading flank extending without change of helix angle from a point on the rotor remote from the inlet end of the rotor to or closely adjacent to the inlet end surface of the land.

ALF LYSHOLM.

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