July 14, 1942. A. LYSHOLM ET AL 2,289,371

ROTARY SCREW APPARATUS

Filed Feb. 28, 1939

Fig. 1.

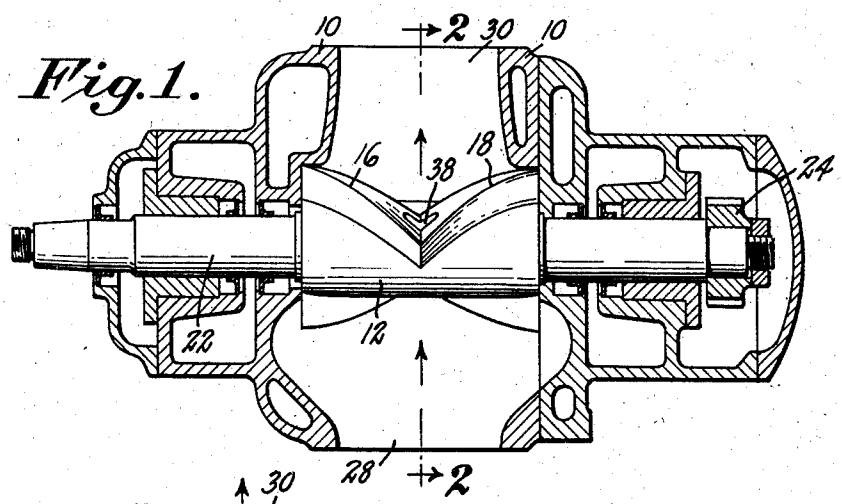


Fig. 2.

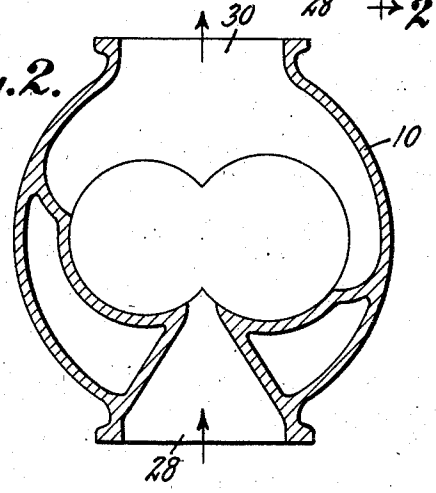


Fig. 5.

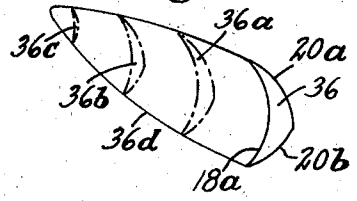
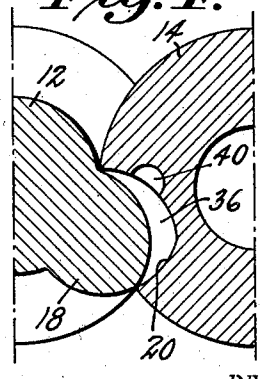


Fig. 4.



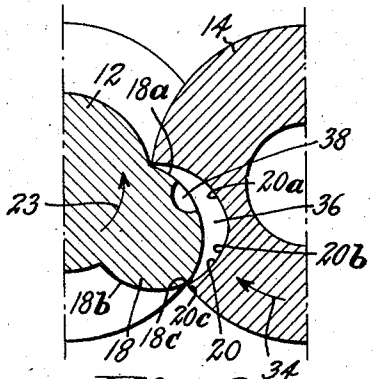


Fig. 3.

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ROTARY SCREW APPARATUS

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Application February 28, 1939, Serial No. 258,866
In Sweden March 1, 1938

8 Claims. (Cl. 103—128)

The present invention relates to rotary screw apparatus applicable for use either as a pump or a compressor, or as a motor, depending upon its direction of rotation and has particular reference to rotary screw machines of the kind in which the cooperating rotors are formed with threads or lobes of the double spiral type and commonly referred to as the herringbone type. Still more particularly, the invention relates to apparatus of the above kind in which cooperating rotors of unlike form are employed having intermeshing surfaces which on one rotor are generally convex in section and on the cooperating rotor are generally concave in section. In this type the pitch circles of cooperating rotors are such that the convex threads or lobes of one of two cooperating rotors lie entirely or substantially entirely outside of the pitch circle of the rotor, this rotor being conveniently referred to as the male rotor and the cooperating rotor referred to as the female rotor.

The invention further refers particularly to apparatus of the kind under consideration which is adapted to be operated at high speed with very small clearance between the intermeshing surfaces, such clearance being considered as space packing rather than contact packing, since with space packing high peripheral speeds may be employed because of the absence of friction between relatively moving surfaces and moreover, if desired, the apparatus may be run dry for handling gaseous media.

For apparatus of the character under consideration, it is highly important from the standpoint of efficiency of operation that the line of minimum clearance, which for convenience herein may be considered as the line of contact, between the intermeshing surfaces of the rotors be maintained as short as possible, since these lines of contact form boundaries defining in part the working spaces of the apparatus, and the less their length, the less the area for leakage of fluid from such spaces.

With apparatus of the above described character, the advantages to be gained by the use of respectively convex and concave intermeshing rotor surfaces are accompanied by a disadvantage which it is the primary object of the present invention to substantially eliminate in order to most usefully secure the advantageous characteristics provided by this kind of structure.

The nature of the disadvantage which the present invention eliminates, and the manner in which this disadvantage is overcome by application of the principles of this invention, may best be understood from a consideration of the following description of apparatus in which the invention is embodied in conjunction with the use of a preferred form of thread profile of the kind forming the subject matter of the co-pending application of Alf Lysholm Serial No. 183,664, filed January 6, 1938 (now Patent No. 2,174,522, dated October 3, 1939), although it is to be understood that the present invention is not limited in its broader aspects to use with this specific kind of thread profile, but may find application with other specific profiles having the same general characteristics.

In the accompanying drawing, in which such apparatus is illustrated:

Fig. 1 is a central longitudinal section through apparatus embodying the invention, which is designed to be employed as a fluid pump;

Fig. 2 is a section of the casing structure taken on the line 2—2 of Fig. 1;

Fig. 3 is a partial section taken on the line 2—2 of Fig. 1 and showing the intermeshing rotor structure;

Fig. 4 is a view similar to Fig. 3 showing an alternative form of rotor structure; and

Fig. 5 is a perspective diagrammatic view illustrating the form of closed or trapped clearance space produced in the apparatus by rotor structure of the type under consideration.

Referring now to the drawing, the apparatus comprises a casing 10 in which are rotatably mounted two intermeshing rotors 12 and 14, of which only the rotor 12 is shown in Fig. 1. Rotor 12 is provided with convex threads or lobes 16 and 18 meeting at an apex midway of the length of the rotor to form a lobe or thread of herringbone form. The rotor 14, as will be observed from Fig. 3, is provided with concave grooves with which the lobes of rotor 12 cooperate in intermeshing relation. The grooves 20, like the lobes on rotor 12, are of double helical form joining at an apex midway of the length of the rotor.

The shaft 22 of rotor 12 carries a gear 24 intermeshing with a gear (not shown) on the shaft of rotor 14 to cause the two rotors to revolve in properly timed relation with respect to each other.

As will be observed from Fig. 3, the lobes 18 of rotor 12, of which there are two in the embodiment illustrated, are formed with dissimilarly curved convex flanks, the profiles of which are indicated by lines 18a and 18b in Fig. 3.

The grooves of the rotor 14 have dissimilarly curved concave surfaces 20a and 20b.

Surfaces 18*a* and 20*a* are, in the embodiment illustrated, of arcuate form in cross-section and of substantially the same radius. Surfaces 18*b* and 20*b* are surfaces each respectively generated by a line on the other rotor, surface 18*b* being generated by the line forming the edge 20*c* of the groove 20, and surface 20*b* being generated by the line 18*c* forming the summit of the lobe 18.

The root circle of the convex lobe coincides or substantially coincides with the pitch circle of rotor 12 which may for convenience be considered as the male rotor, while the outer cylindrical surface of rotor 14, which for convenience may be considered as the female rotor, coincides with the pitch circle of this rotor.

In the embodiment illustrated, the apparatus provides a displacement pump capable of handling incompressible fluid media, the fluid entering through the inlet port 28 and being discharged through the outlet port 30 in the direction indicated by the arrows of Figs. 1 and 2.

In the operation of the apparatus, working spaces are formed by the spaces between adjacent rotor lobes and the casing, and as the rotors are turned in a direction to cause the apexes of the convex lobes to enter the apexes of the cooperating grooves, the volumes of the working spaces are progressively decreased.

By suitably choosing the areas of the inlet and outlet ports, the working spaces may be opened up to communication with the outlet port before the volume of the closed working space is diminished after passing out of communication with the inlet port, so that then the apparatus may operate as a displacement pump.

By diminishing the area of the outlet port so that threads progressively enter cooperating grooves after a given work space has passed out of communication with the inlet port, and before the space has been brought into communication with the outlet port, the apparatus may be employed as a compressor for producing within itself, and independent of any back pressure at the outlet, compression of a gaseous media. Conversely, if the direction of rotation of the apparatus is reversed from that for which it is utilized as a pump or compressor as the case may be, the apparatus may be operated as a motor which may be of either displacement or expansible chamber type. In such event the port 30 would become the inlet port and the port 28 the outlet port so that in so far as the present invention is concerned, these ports may have either of these functions.

Let it be assumed, for the purpose of explanation, that the apparatus is to be used as a pump or compressor as indicated in the drawing.

In such case the herringbone lobes, entering their cooperating grooves, displace fluid from the latter progressively toward the apex plane of the herringbone at the center of the apparatus. However, before the apex of any given lobe enters completely into the corresponding groove, a position is arrived at as illustrated in Fig. 3, which from the figure will be observed results in the formation of a clearance space 36 between the flank 18*a* of rotor 12 and the sides 20*a* and 20*b* of rotor 14, which space is of generally crescent-shaped form in cross-section.

Because of the contact line between the surfaces of the intermeshing lobe and groove, this space 36 assumes the form to one side of the central transverse plane of the apparatus, like that shown in perspective in Fig. 5. The cross-sectional area of this space diminishes to sections such as those indicated by dot and dash lines at 36*a*, 36*b*, and 36*c*, the contact line being of the general form indicated at 36*d* and being formed by the edges of the groove in substantially contacting relation with the surface of the flank 18*a*. A similar condition will obviously exist to the other side of the central transverse or apex plane so that the shape of the clearance space formed when the rotors are in the position of relative rotation shown in Fig. 3 will be generally V-shaped, with each arm of the V having the general shape of the space shown in Fig. 5.

With the rotors moving in the direction of arrows 23 and 34 in Fig. 3, it will be apparent that the volume of the clearance space which is trapped between the flanks of the lobes 16 and 18 and the walls of the cooperating groove, will be decreased to an ultimate volume of substantially zero by the time the rotors have moved from the position shown in Fig. 3 to a position wherein the profile of the flank 18*a* at the apex coincides with the side 20*a* of the groove at its apex.

This condition would of course produce destructively high pressure of any fluid trapped in this clearance space, even though such fluid were of compressible nature, if the desired conditions of operation were to be employed, which conditions are high speed of rotor operation and minimum practical clearance or space packing between the rotors. Obviously, under such conditions, even with a compressible fluid, the fluid could not escape through the space packing at sufficiently rapid rate to avoid building up pressures of destructively high value.

In order to avoid this condition, a relief chamber is provided by recessing the forward face of the male rotor lobe or the forward face of the female rotor groove, or both, at the apex zone of the rotors, that is, the area immediately adjacent to the apex plane and preferably on both sides of that plane.

When the rotors are in the position shown in Fig. 3 communication between the clearance space and the discharge side of the rotors has been cut off by the entry of the summit 18*c* of the male lobe into the cooperating groove. As the rotors revolve beyond this cut off position the volume of the clearance space is progressively decreased to its theoretical zero value, which in the case of the profile illustrated occurs when the point on the summit 18*c* lying at the apex of the lobe moves to a position in which it is in the plane coinciding with and joining the parallel axes of the rotors. When the rotors are in this position the lobe surface 18*a* and the groove surface 20*a* have moved into substantial contact with each other at the apex plane and the clearance space volume has been reduced to substantially zero.

If the apparatus is operating to pump or compress a gaseous medium, the gas trapped in the clearance space can flow into the relief chamber 38 as the volume of the clearance space diminishes, even if the relief chamber is not in communication with the inlet side of the rotors so as to act as a by pass. This however is not desirable, since if the volume of gas in the clearance chamber is forced into a closed relief chamber in addition to the gas already there, it results in compression of the gas in the relief chamber to relatively high pressure, which pressure is dissipated when the movement of the rotors brings the relief chamber into communication with the inlet side of the rotors. Such compression and

subsequent expansion represents useless work and a loss and is therefore to be avoided.

Preferably the construction is as shown in the drawing, whereby the relief chamber is adapted to come into communication with the inlet side of the rotors when (but only when) the clearance space is closed, so that as the clearance space is thereafter decreased in volume the fluid in it can pass to the inlet side of the rotors. From inspection of Fig. 3 it will be seen how this can be accomplished with the relief recess shaped and located as shown in Figs. 1 and 3. In Fig. 3 the clearance space has been closed, and while in the plane of this figure (which is the apex plane) the chamber **38** is not yet in communication with the inlet side, it must be remembered that the arm of chamber **38** below the plane of the paper curves upwardly as viewed in Fig. 3. This will became apparent by reference to the shape of the relief chamber as shown in Fig. 1. Furthermore the outer edge of the groove surface **20***a* curves downwardly in Fig. 3 below the plane of the paper, so that somewhere below this plane the edge of the groove crosses the relief chamber to provide an opening establishing communication between the relief chamber and the inlet side of the rotors.

The exact position of rotation of the rotors at which the groove edge crosses the relief chamber may vary, but if the apparatus is to be used to pump liquids, this crossing must be established coincidentally or substantially coincidentally with the closing of the clearance space, or in other words, just when the rotors reach the position shown in Fig. 3. This is necessary in order to vent to the inlet side the incompressible contents trapped in the clearance space when the latter is closed and thereafter begins to decrease in volume. For reasons previously explained the relief chamber can vent to the inlet side at a later time in the cycle if the medium in the clearance space is of a compressible nature, but it will be apparent that regardless of the nature of the fluid the relief chamber should not vent to the inlet side before the rotors reach the position of Fig. 3, since if that were permitted, there would be an open path for the blowthrough of an indeterminate amount of fluid from the discharge to the inlet side before the rotors reached the cut-off position of Fig. 3.

The nature of the relief chamber is thus such that there is returned to the inlet side only the relatively very small quantity of fluid trapped in the closed clearance space, and this quantity, in comparison with the total quantity handled by the apparatus, is negligible.

It will be evident that the same result attained by the use of the groove in the flank of the convex lobe may be obtained by making the relief chamber in the form of a recess in the face of the concave groove. Such an arrangement is illustrated in Fig. 4 where the relief chamber is formed by the groove **40** which takes the same general wedge-shaped character as the groove **38**, as shown in Fig. 1. It will further be evident that the relief chamber may be formed partially in each of the rotor structures to attain the same result.

The relief chamber is also useful in case the apparatus is used as a motor in order to avoid the production of undesirably high vacuum as the clearance spaces **36** are progressively formed by reverse rotation of the rotors and increase from substantially zero volume to the volume attained when the rotors are in the position shown in Fig. 3.

It will be apparent from the foregoing that the present invention is readily applicable to numerous different specific forms of apparatus of the general type under consideration, whether such apparatus be used as pump, compressor, or motor, and it is accordingly to be understood that the scope of the invention contemplates all structures falling within the terms of the appended claims when they are construed as broadly as may be permissible in the light of the prior art.

What we claim is:

1. In apparatus of the character described, a casing, cooperating rotors mounted for rotation about parallel axes in said casing and having herringbone lobes of generally convex profile and grooves of generally concave profile, respectively, said lobes and grooves intermeshing and cooperating with each other and said casing to provide working spaces for fluid, the surfaces of each of said lobes and its cooperating groove forming a clearance space extending longitudinally of the lobe and groove for a limited distance on either side of the apex plane of the herringbone, said clearance space being closed by the entry of the summit of the lobe into the cooperating groove adjacent to said apex plane and being reduced in volume to a theoretical zero value as the rotors turn thereafter to bring the lobe fully into the groove at said apex plane, and recess means formed in a wall defining a part of said clearance space to provide a relief chamber located at the apex zone and in comunication with the clearance space, said recess means being brought into communication with the inlet side of the rotors by rotation of the rotors but only after said clearance space has been closed by the entry of the lobe summit into the groove, whereby to prevent undue increase in pressure of the fluid trapped in said clearance space as the latter is reduced to substantially zero volume and to prevent blow-through from the discharge side to the inlet side of the rotors due to the fact that said relief chamber is brought into communication with said inlet side only after communication between said clearance space and said discharge side has been cut off by the entry of the summit of the lobe into the groove as aforesaid.

2. Apparatus according to claim 1 in which the recess is of generally wedge-shaped form having arms extending axially to each side of the apex of the lobe and having maximum cross-sectional area at its axially central part.

3. Apparatus according to claim 1 in which the recess is located in the apex zone of the forwardly facing convex flank of the male rotor which constitutes one of the walls of said clearance space.

4. Apparatus according to claim 1 in which the recess is formed in the concave surface of the apex zone of the forward side of the groove which forms a part of the wall of said clearance space.

5. A rotor for use in rotary screw apparatus, said rotor having a plurality of herringbone lobes, said lobes having convexly curved flanks, and each of said lobes being provided with a recess in the forwardly facing one of its flanks at the apex zone of the rotor, said recess being located between the root and summit circles of the lobe and having arms extending to each side of the apex plane of the lobe, said arms extending only

a minor portion of the distance from said apex plane to the respective ends of the rotor.

6. A rotor for use in rotary screw apparatus, said rotor having a plurality of herringbone lobes, said lobes having convexly curved flanks and each of said lobes being provided with a recess in the forwardly facing one of its flanks at the apex zone of the rotor, said recess being located between the root and summit circles of the lobes, and the cross-section of said recess being greatest at the apex plane of the rotor and diminishing axially in both directions away from said plane.

7. A rotor for use in rotary screw apparatus, said rotor having a plurality of herringbone grooves, said grooves having concavely curved sides, and each of said grooves being provided with a recess formed in the forward one of said sides between the root and top circles of the groove and having arms extending to each side of the apex plane of the groove, said arms extending only a minor portion of the distance from said apex plane to the respective ends of the rotor.

8. A rotor for use in rotary screw apparatus, said rotor having a plurality of herringbone grooves, said grooves having concavely curved sides and each of said grooves being provided with a recess formed in the forward one of said sides between the root and top circles of the groove, and the cross-section of said recess being greatest at the apex plane of the rotor and diminishing axially in both directions away from said plane.

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