April 21, 1964 H. R. NILSSON ETAL 3,129,877

ROTARY PISTON, POSITIVE DISPLACEMENT COMPRESSOR

Original Filed April 29, 1957

Fig. 1

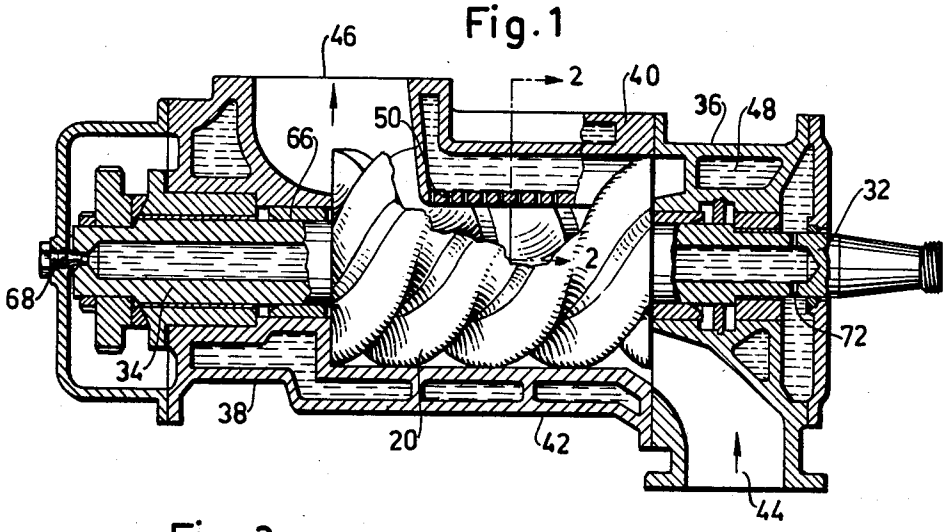


Fig. 2

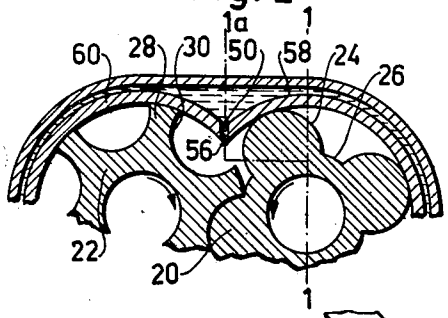


Fig. 4

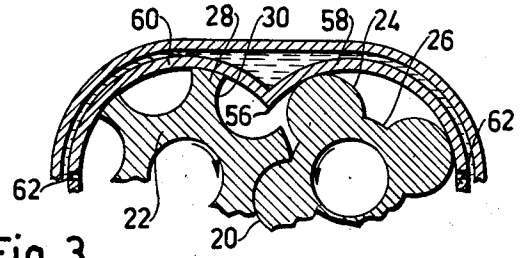
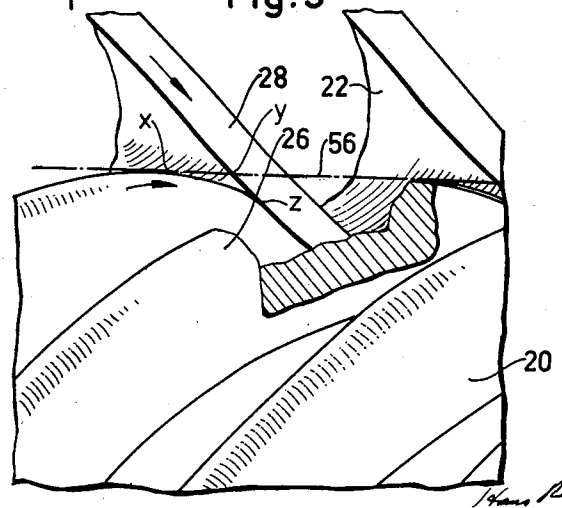


Fig. 3



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ROTARY PISTON, POSITIVE DISPLACEMENT COMPRESSOR

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Continuation of application Ser. No. 655,843, Apr. 29, 1957. This application Dec. 22, 1961, Ser. No. 161,576
Claims priority, application Sweden May 17, 1956
4 Claims. (Cl. 230—143)

This application is a continuation of our prior application Serial No. 655,843, filed April 29, 1957, and now abandoned, and relates back thereto for all dates and rights incident to the filing thereof and of our corresponding Swedish application Serial No. 4,723/56, filed May 17, 1956, and from which priority is claimed.

The present invention relates to rotary piston, positive displacement compressors of the kind in which at least two dissimilar rotors, male and female (as hereinafter defined) and provided with helical lands and grooves, are mounted to rotate in intermeshing relation about coplanar axes in intersecting bores in a casing structure the walls of which cooperate with portions of the rotor grooves to form compression chambers diminishing in volume as the rotors revolve to compress an elastic fluid contained therein.

This general type of compressor is well known in the art, the first practical design of which, embodying an inbuilt or internal compression ratio, is disclosed in U.S. Patent No. 2,243,874, granted to Alf Lysholm June 3, 1941, the original Lysholm design being modified to provide an improved form of rotor by the invention disclosed in U.S. Patent No. 2,622,787, granted December 23, 1952, to Hans Nilsson, the latter form of rotor being employed in the present disclosure, by way of example but without limitation, to illustrate the principles and advantages of the present invention.

Compressors of the kind here under discussion have attained a considerable measure of commercial success, but heretofore have without exception been operated as "dry" compressors, that is, without lubricant or other liquid being introduced into the working spaces of the machine and with the rotors being maintained out of contact with the casing walls by suitable clearance and out of contact with each other by appropriate clearance through the use of timing or synchronizing gears. In the case of dry compressors, leakage from the working chambers is retarded only by the "space packing" afforded by the relatively close clearances obtained through very accurate manufacture, but even with the maintenance of very close clearances, acceptable efficiencies are obtainable with dry compressors only by operating them at relatively high speeds so that the volumes of fluid handled are very large in relation to the size of the machine and consequently of the dimensions of the chambers from which the leakage occurs through the space packing.

With such compressors, the tip speed of the male rotor may be as high as of the order of from 150 to 200 meters per second, as noted in the aforesaid Nilsson patent, which may entail rotor speeds of as much as 10,000 to 12,000 r.p.m. for compressors of the sizes usually driven by electric motors or internal combustion engines. Such speeds necessitate the use of step-up gears between the

prime mover and the compressor which are undesirable from the standpoint of weight bulk and service as well as cost.

A further practical disadvantage of the high speed of operation of the dry compressor is the relatively high noise level resulting from the high frequency pulsating admission to and discharge from the working chambers of the gaseous medium being compressed, which requires in most installation the added cost, weight and bulk of silencing equipment.

Still more important are the practical limitations imposed on dry compressors by the factor of the heat generated by the work of compression. Such heat obviously and unavoidably creates dimensional changes due to expansion and contraction of the casing, rotor and other parts controlling the clearances, which are probably the most important as well as the most sensitive factor determinative of the performance of the dry compressor. To keep such changes to an acceptable minimum, requires as a practical matter that at least the casings of dry compressors be liquid cooled if any substantial inbuilt compression ratio, say three to one or better, is incorporated in a single stage of compression from atmospheric inlet pressure, and even with liquid cooling, practical experience has demonstrated that a ratio of five to one represents the desirable upper limit for single stage compression.

Thus while dry compressors of the kind under discussion have proved to be of practical commercial value and have gone into extensive use, they are subject to several important limitations, particularly in the very large field requiring so-called "shop air," that is, air at about one hundred pounds gage pressure, and moreover requiring that such air be provided by portable compressors desirably driven directly by internal combustion engines and effecting compression in a single stage without an intercooler.

It is therefore the primary object of the present invention to provide a new and improved compressor of the general type under consideration which will eliminate the above noted and other limitations and deficiencies of such compressors as heretofore employed.

To this end the invention contemplates the injection of liquid into the working chambers of the compressor to effect two major purposes, the first being to provide a liquid seal instead of dry space packing for the clearance spaces between the rotors and the casing and between the intermeshing portions of the rotors and the second being to effect direct cooling of the gaseous fluid while it is being compressed, so that the compression cycle deviates from the adiabatic to more nearly approach the isothermal. Preferably such injection is confined to or substantially to the compression phase of the cycle for reasons hereinafter more fully appearing, and for a better understanding of other and more detailed objects of the invention together with the preferred manner of carrying the invention into effect in order to obtain the maximum benefit therefrom, reference may best be had to the ensuing portion of this specification, taken in conjunction with the accompanying drawings, in which:

FIG. 1 is a longitudinal section, partly in broken away elevation, taken along the lines 1—1 and 1*a*—1*a* of FIG. 2;

FIG. 2 is a partial section taken along line 2—2 of FIG. 1;

FIG. 3 is a fragmentary elevation, in part broken away and on much larger scale, of portions of the rotors shown in FIGS. 1 and 2; and

FIG. 4 is a section similar to FIG. 2 showing a modified form of construction.

The compressor shown in FIGS. 1 and 2 consists of two co-operating rotors 20 and 22 of which one, the male rotor 20, is provided with convex lands 24 and intervening grooves 26 and co-operates with the other or female rotor 22 which is provided with concave lands 28 and intervening grooves 30. As will be obvious from the drawings, the rotors are not only dissimilar in form, but also embody the basically characteristic feature of compressors of the kind to which the invention is directed, that the major portion of the lands and intervening grooves of the male rotor lie outside the pitch circle of the rotor while the major portion of the lands and grooves of the female rotor lie inside the pitch circle of the female rotor. In fact as in usually the preferred case, the outer circumference or envelope of the female rotor substantially coincides with its pitch circle while the lands of the male rotor lie substantially entirely outside its pitch circle. Also as will be evident from FIG. 2, the convex profile of the lands of the male rotor provide an outer or crest portion that is substantially circular in cross section, being of the form disclosed in the aforesaid Nilsson patent, which in connection with the use of the present invention has certain definite advantages as will hereinafter more fully appear. The rotors are provided with end shafts 32 and 34 which are located in end walls 36 and 38 of the compressor casing 40 which further includes a jacketed barrel portion 42 enclosing the rotors. The end wall 36 comprises a separate element whilst the end wall 38 is made integral with the barrel portion 42.

The casing is provided with an inlet 44 and an outlet 46 for the gas, for example air, which is to be compressed in the machine. In the example shown inlet 44 is located at the bottom and outlet 46 at the top, each being within or adjacent to the corresponding end wall of the casing, respectively. However, the invention is not restricted to this placing or design of inlet and outlet which may be varied according to well-known principles.

The barrel portion 42 and the end walls 36 and 38 are double-walled to provide jacket spaces 48 to which liquid under pressure is supplied from a source thereof (not shown) from which said liquid is injected via supply openings into the working chambers of the machine.

As previously noted, the profile of the rotors shown is that disclosed in the Nilsson patent aforesaid, characterized by male rotor lands the crest portion of the profile of which are circular in outline. This specific form of profile is particularly advantageously employed in a compressor into which liquid is introduced in substantial quantity for the purpose of obtaining appreciable cooling effect as well as for sealing purposes, and conversely, injection of liquid in the preferred manner cooperates, as hereinafter explained, in a particularly advantageous way to enhance the effectiveness of operation of rotors having the circular form of profile.

The particular advantage of the use of rotors characterized by the circular profile, in a compressor into which liquid is introduced, is perhaps best understood from a consideration of the Nilsson patent, which for the purpose of giving a detailed explanation of the nature and advantages of the circular profile, may be considered as incorporated herein by reference and further may be considered in conjunction with FIG. 3 of the disclosure. As explained in the Nilsson patent, the early forms of devices of the character under consideration employed male rotors, one or both of the flanks of the lands of which were generated curves, usually generated by a point on or closely adjacent to the pitch circle of the cooperating female rotor. The

purpose of this construction was to insure continuity of the sealing line defining the perimeter of each working chamber at the point of entry of the crest of a male land into a female groove to create the chamber. This was accomplished by such construction, but it was also accompanied by the unavoidable and inherently disadvantageous formation of closed or trapped pockets in the interspace between fully meshed lands, which trapped pockets run out to zero volume as the rotors revolve to cause the compression chambers also to run out to zero volume as they deliver their contents to the outlet of the compressor. The creation of these trapped pockets, their disadvantages, the necessity for venting them if excessive pressures and undesirable losses are to be avoided, and different expedients for venting them, are recognized and discussed in Lysholm Patent No. 2,243,874, granted June 3, 1941, and Whitfield Patent No. 2,287,716, granted June 23, 1942. One of the major advantages achieved by the circular profile disclosed in the aforesaid Nilsson patent is the elimination of the trapped pockets characteristically produced by generated profiles and while this advantage is of importance in the field of dry compressors, it will be evident that it is of even greater importance for compressors into the compression spaces of which liquid is introduced, particularly if the quantity introduced is substantial, as must be the case if a material cooling effect is to be secured. With an incompressible liquid present in the chambers, it is self evident that effective venting means must be provided for venting the trapped pockets if destructive pressures are to be avoided, and experience has shown that the required venting of the trapped pockets, through the medium either of excessively large clearance spaces or the provision of venting spaces or passages, can be effected only at the expense of undesirable loss in the overall efficiency of operation of the machine.

With dry compressors, experience has demonstrated that timing gears for maintaining the intermeshing rotors in properly spaced phase relation to insure clearance between them are a practical necessity. It is to be noted, however, that in the case of generated profiles the necessity for timing gears is more pronounced than in the case of circular profiles since in the former case the trapped pockets produce torque forces tending to make one rotor overrun or underrun its mating rotor while in the latter case such torque forces are not produced. In any case it is highly desirable if not essential to use timing gears with dry compressors, whereas with compressors into which liquid is introduced, such gears may be dispensed with, particularly when the liquid used is a lubricant such as ordinary lubricating oil, which is quite suitable for the purpose. This is particularly true in the case where the preferred form of circular profile herein disclosed is employed and no fluid pressure generated torques are developed tending to make the rotors turn at different speeds. Thus it will be evident that compressors constructed in accordance with the present concept afford the additional advantage of enabling the extra bulk, weight and cost of timing gears to be eliminated.

While as previously pointed out, the introduction of liquid into the compression chambers results in obtaining important advantages, such as improved sealing, direct cooling of the fluid being compressed, elimination of gearing and the like, such introduction involves other factors of a less favorable nature that must be taken into consideration and which the present invention operates to greatly ameliorate if not entirely eliminate.

In the first place there is the matter of dynamic losses due to agitation of the fluid in the compressor resulting from the movement of the rotors. These losses have heretofore commonly been referred to as "ventilation" losses, engendered by the turbulence of the elastic fluid constituting the sole content of the compression chambers, and the gaseous fluid friction developed as a result of the flow through the space packed clearance spaces. These losses have been of substantial magnitude, particularly in view

of the high peripheral or tip speeds required for acceptable operation of a dry compressor.

The introduction of liquid, by providing a solid fluid closure instead of the space packed clearances of a dry compressor, reduces the rate of leakage loss from the compression chambers to such an extent that the tip speeds of the rotors can be reduced, while still maintaining acceptable efficiency, to a minor fraction of the speed necessary for operation of a dry compressor. Such reduction is in fact of such magnitude that it is possible to reduce the speed to a value such that compressors of a type and size heretofore requiring step-up gears when operated dry by normal speed electric motors or by internal combustion engines, may be and are normally operated directly connected to such power plants when operated with liquid introduced, thus achieving one of the major objects of the invention.

Not only is major reduction in speed of operation made possible by the introduction of liquid, but experience has shown it to be necessary if undue losses are to be avoided. Basically, of course, the reason for this is that the very much greater density and viscosity of even the thinnest liquids (water or very light hydrocarbons could be used if desired), as compared with that of the gaseous fluids usually compressed, results in what may be termed "churning" losses developed by the liquid (as distinguished from turbulence or ventilation losses developed by a gas) of such magnitude that the high speeds characteristic of dry operation could not be tolerated.

In spite of major reduction in speed as compared with dry operation, wet operation, that is, operation with the introduction of liquid, entails substantial churning losses, and in one of its more specific aspects the present invention contemplates introduction of the liquid into the compression chambers in a particular fashion the effect of which is to minimize churning losses as well as other losses of a nature to be dealt with later.

In order best to understand this, it is appropriate first to briefly review the fundamental nature of the operation of compressors of the kind disclosed. When operating as a compressor, the rotors are turned in the directions indicated by the arrows appearing in FIGS. 2 and 4, this action serving to draw air into the rotor grooves through the inlet 44 and to discharge the compressed air through the outlet 46, in the directions noted by the arrows in FIG. 1. As will be noted from this figure, the helices of the lands and grooves of the rotors have a total wrap angle of less than 360°, in accordance with standard practice for this type of compressor, so that the working chambers are not delimited axially by successive intermeshes of the same lands and grooves, which would produce chambers moving axially at constant length and volume as transport chambers rather than as compression chambers decreasing in length and in volume as the rotors revolve. With the present construction the rotor grooves first fill with fluid during the suction phase of the cycle as their right hand ends (as seen in FIG. 1) pass the port 44. Following the point of cutoff from this port, the grooves carry their fluid contents upwardly at constant volume in a so-called transport phase of the cycle until the right hand end of a male or female rotor land commences entry, respectively, into the right hand end of a cooperating female or male rotor groove. The place of such entry is at the line of intersection 56 between the two bores 58 and 60 in the barrel portion of the casing, and such entry commences the compression phase of the cycle. It is well to note at this point that the entry of a land into a groove to initiate compression may occur before the groove involved has moved to a position in which it has been cut off from the inlet, so that there is an overlap between the inlet and compression phases of the cycle and no intervening constant volume transport phase. All this, however, is well known in the art, as exemplified by Lysholm Patent No. 2,410,172, granted October 29, 1946.

At the time the compression phase commences each compression chamber is formed by two cooperating grooves, one male and one female, and is axially delimited at one end by the fixed transverse plane of the delivery end wall 38 of the casing and at the other end by the place of intermesh of the rotors. This results in the formation of what may be termed chevron-shaped chambers extending between an axially fixed end plane, which remains fixed, and a moving place of intermesh which moves toward that plane as the rotors revolve, to shorten the chamber axially until it runs out to zero volume. When the chamber comes into communication with the outlet port 46, the compression phase may technically be said to terminate, to be followed by the final delivery phase at constant pressure. In practice, however, the latter phase may for practical purposes be considered as a final part of the compression phase since the volume of the compression chamber is continuing to decrease and the final pressure may increase to a value above that existing when registration with the outlet port occurs. This is fully recognized and discussed in Lysholm Patent No. 2,519,913, granted August 22, 1950, and for the purposes of the present disclosure and specification, the term compression phase as hereinafter used is to be considered as including the delivery phase. While for the sake of simplicity the present invention has been described and illustrated as applied to a compressor having rotors with helical lands and grooves of single hand, the invention is in no wise limited to such structures but is equally applicable to compressors having herringbone rotors of the kind shown in Lysholm Patent No. 2,289,371, granted July 14, 1942, such compressors, insofar as the present invention is concerned, being the full equivalent of the example herein illustrated.

With the foregoing in mind, it will be apparent that, insofar as the factor of sealing the compression chambers is concerned, only the compression phase of the cycle, and that portion of the cooperating parts defining the perimeter of a chamber in which a pressure differential above inlet pressure has been established, need be taken into account. The perimeters of the chambers in which elevated pressures are present define areas very considerably less than the total internal area of the bores and casing ends enclosing the rotors, the total of what may be termed the area of the pressure surface or pressure area being in fact less than half, or the minor portion, of the total of the internal casing surface. It is only over this area of the surface that sealing against leakage from the compression chambers is required, and in accordance with one particular aspect of the invention, the churning losses incident to the use of a liquid sealant are minimized by confining the introduction of the liquid to, or substantially to, the pressure area of the casing. It will be understood, of course, that the liquid sealant will necessarily be eventually distributed over the entire internal surface of the casing, but it will equally be evident that if a sufficient quantity is injected to provide a material cooling effect the relative concentration of liquid and the resultant churning loss will be greatest in the vicinity of the place of introduction which should therefore be held to a minimum area.

Furthermore from the standpoint of the desired cooling effect to be produced by the liquid, it is obvious that no such effect is required until the compression zone is reached, where heat is first generated by the work of compression. On the other hand, it is highly desirable that compression be commenced without preheating of the inlet air. Since the liquid used for sealing and cooling is heated in the process and must for reasons of economy be recirculated in the system after having been separated from the air delivered by the compressor, its admission temperature to the compressor may well be above that of the incoming air unless additional and undesirable cooling equipment is provided. By confining the introduction of liquid to or approximately to the compression phase of the cycle, undesirable preheating of the inlet air by recircu-

lated liquid at higher than inlet temperature is with certainty avoided.

Reverting again to the preferred form of rotor with lands of circular profile, the advantages to be derived from it are accompanied by one unfavorable characteristic which, however, in accordance with another of the more specific aspects of the present invention, is substantially eliminated as a practical consideration. This characteristic is the inherent formation of a gap or interruption in the sealing line of uniform clearance defining the perimeter of each compression chamber, due to the non-generated character of the male rotor lands at the crests thereof. The nature of this gap and the cause of its formation are disclosed and explained in detail in the aforesaid Lysholm Patent No. 2,622,787, to which reference may be had for such detailed explanation, but for the purposes of this disclosure and an understanding of the special relationship between a rotor of circular profile, with its numerous advantages hereinbefore noted, and the particular preferred mode of introducing liquid in accordance with the present invention, reference to FIG. 3 hereof is sufficient. As will be seen from this figure, the circular crest of the male rotor land **26** reaches the line of intersection **56** between the two rotor bores on the high pressure side of the plane of the rotor axes, at a place axially displaced from the adjacent edge of the female rotor land **28**. This inherently produces a small and substantially isosceles triangular gap indicated at *x*—*y*—*z* in the figure, resulting in a break in the sealing line between that portion formed between one casing bore and the crest of the female land and the contiguous portion formed between the other bore and the crest portion of the cooperating male land. To effect compression the rotors revolve in the directions indicated by the arrows in the figure, and with such directions of rotation, the gaps travel axially along the line of intersections **56** from right to left as viewed in the figure, from the position at the right hand end of the rotors where the gaps are formed as the cooperating lands come into mesh to establish the compression chambers, to the position where the gaps disappear by virtue of the fact that the place of intermesh of the rotors has come into communication with the outlet port of the compressor. These gaps constitute undesirable leakage or blow holes through which compressed working fluid can leak from any given compression chamber to the next succeeding chamber of the series, which is always at a lower pressure.

Obviously, anything that will tend to reduce the leakage through the blow holes is highly desirable and accordingly the injection of the liquid for the broad general purposes of the invention is availed of to provide a further improvement, in the preferred embodiment of apparatus, by effecting the injection through a series of orifices or nozzles **59** distributed axially along the length of the line of intersection **56** between the casing bores, along which the gaps travel. Such injection, even though it is desirable to inject in the form of a spray to secure the maximum cooling effect from a given amount of liquid, obviously provides a concentration of liquid at the line along which the gaps travel sufficient to very materially reduce their size and the leakage rate of the fluid therethrough.

While for more or less self-evident reasons, it is preferable in order to secure the maximum benefit to inject along the line of intersection on the high pressure side, the beneficial effects derived from this arrangement are not critically dependent upon this precise location. As will be apparent from the direction of rotation of the rotors as seen in FIGS. 2 and 4, liquid injected into the barrels at any place along the high pressure side of the plane of the rotor axes will be carried with relatively short paths of travel to the line of intersection where the gaps occur, tending to fill them. Also, under some circumstances it may be desirable to include a larger area on the high pressure side over which newly injected liquid is distributed to the chambers, and instead of injection exactly along the high pressure line of intersection, injection may for example be made through series of orifices **62** opening through the walls of the respective barrels at places peripherally spaced from the high pressure line of intersection but still preferably on the high pressure sides of the barrels, as shown in the modification illustrated in FIG. 4.

From the foregoing it will be apparent that the principles of the invention are applicable to a wide variety of specific compressor applications involving widely varying compression ratios, different gaseous fluids to be compressed, and different liquids to be utilized for sealing and cooling. The specific design and mode of operation will naturally be governed by the particular conditions to be met and may require other specific expedients than those herein illustrated and described by way of example, and the invention is accordingly to be considered as embracing all forms of structure falling within the scope of the appended claims.

We claim:

1. A rotary piston, positive displacement, elastic fluid compressor comprising a casing structure providing a barrel portion having intersecting bores with co-planar axes and further providing spaced apart ports communicating with said bores and comprising a high pressure port at least the major portion of which is located to one side of the plane of said axes and a low pressure port at least a major portion of which is located on the opposite side of said plane, rotors provided with helical lands and grooves having an effective wrap angle of less than 360° rotatably mounted in said bores and comprising a male rotor having lands and intervening grooves the major portions of which lie outside the pitch circle of the male rotor and a female rotor having lands and intervening grooves the major portions of which lie inside the pitch circle of the female rotor, the lands and grooves of said rotors intermeshing to form with the confronting portion of the casing structure chevron-shaped working chambers each composed of communicating portions of a male rotor groove and a female rotor groove joining at the apex end of the chamber and said apex end moving axially towards said high pressure port as the rotors revolve to decrease the volume of the chamber and effect compression of the fluid contents thereof, the lands of said male rotor having convexly curved profiles the outer portions of which lie peripherally within the lines of a land having a root of the same peripheral width and flanks generated by points on the pitch circle of said female rotor and the profile of the grooves of said female rotor being the envelopes developed by the convex lands of said male rotor in passing into and out of mesh with said female rotor, whereby a series of gaps is created in the sealing lines of normal clearance between the cooperating parts defining the perimeters of said compression chambers, said gaps extending axially of the line of intersection between said bores on the high pressure side of the bores and said gaps traveling axially of said line of intersection toward said high pressure port as the rotors revolve, and means for sealing the clearance between the parts defining the perimeters of said chambers inclusive of said gaps and for cooling the fluid being compressed therein comprising a plurality of openings in the barrel portion of said casing structure for injecting liquid into said chambers, said openings being distributed lengthwise of said barrel portion of the casing structure in the area thereof adjacent to and inclusive of said line of intersection.

2. A compressor as defined in claim 1, in which the convexly curved flanks of the lands of the male rotor have crest portions which are of generally circular profile and in which the grooves of the female rotor are generally circular and the profiles of which are the envelopes developed by the convex lands of the male rotor in passing into and out of mesh with the female rotor.

3. A compressor as defined in claim 1, in which said openings for injecting liquid into said chambers are located along said line of intersection.

4. A compressor as defined in claim 2 in which said openings for injecting liquid into said chambers are located along said line of intersection.

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