

Jan. 15, 1963

W. BAILEY ETAL  
ROTARY COMPRESSORS

3,073,514

Original Filed Nov. 14, 1957

3 Sheets-Sheet 1

Fig. 1

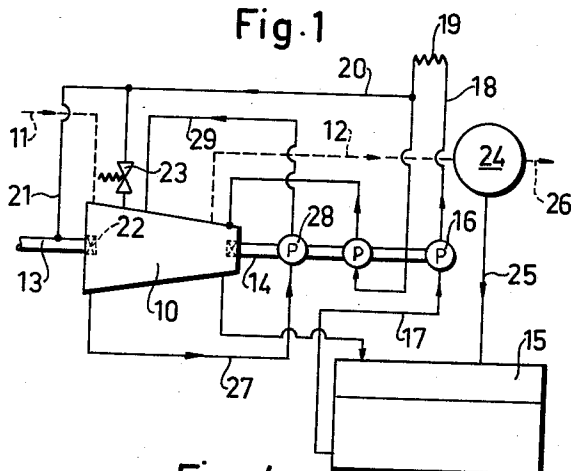


Fig. 4

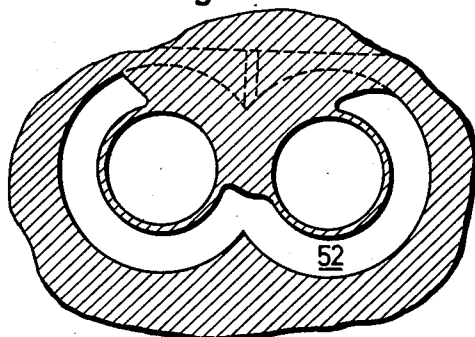
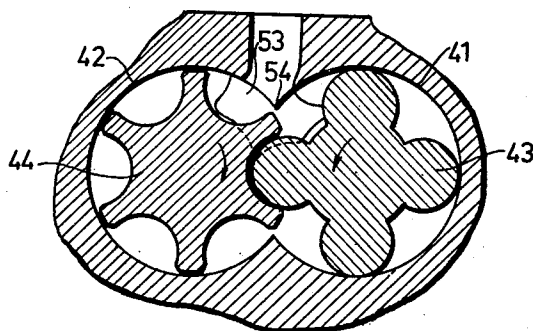


Fig. 5



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Fig. 2

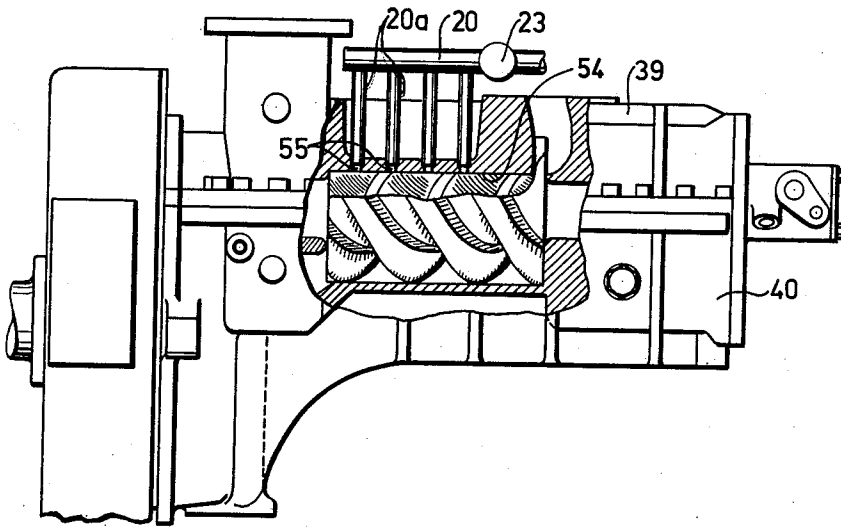
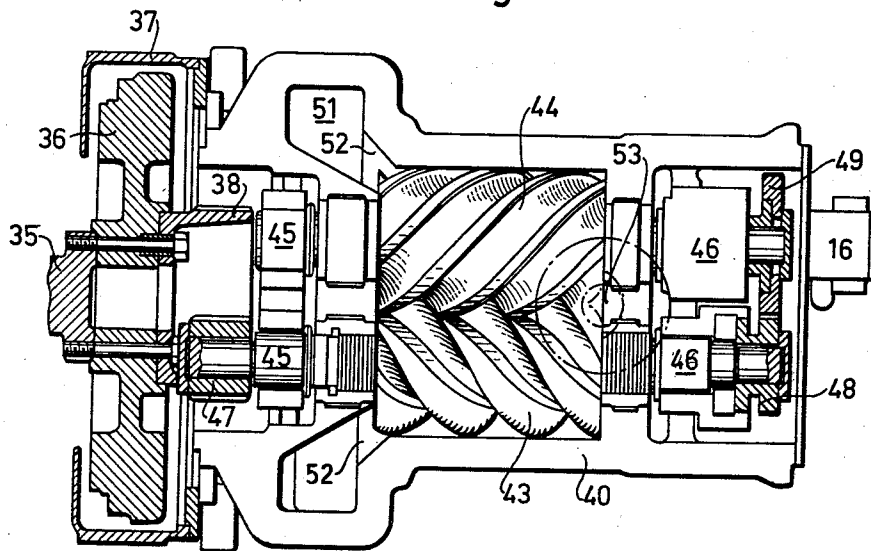


Fig. 3



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FIG. 6.

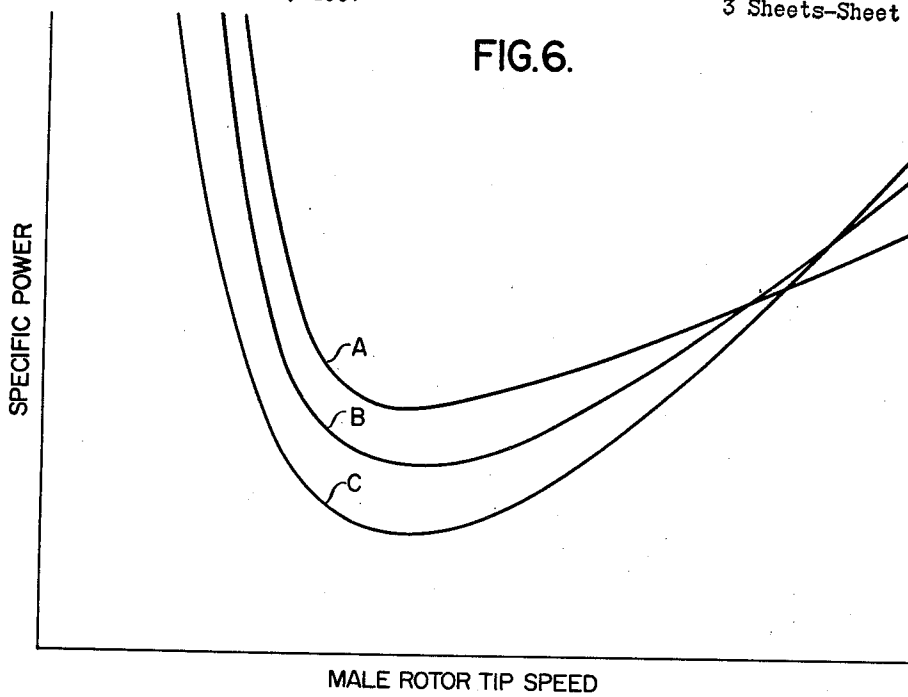
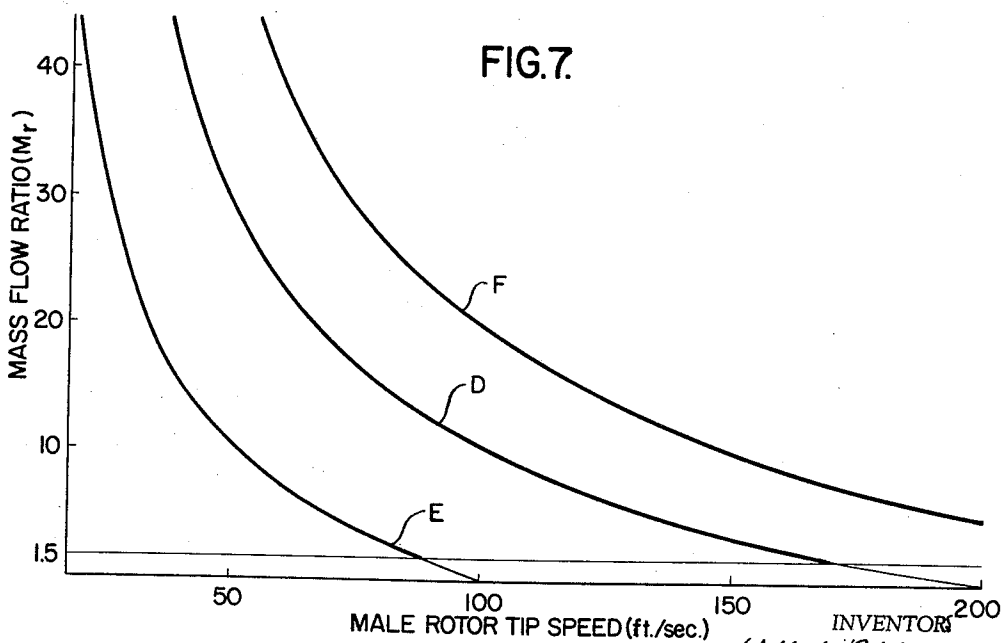


FIG. 7.



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3,073,514

## ROTARY COMPRESSORS

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Continuation of abandoned application Ser. No. 696,349, Nov. 14, 1957. This application Feb. 20, 1962, Ser. No. 174,207

Claims priority, application Great Britain Nov. 14, 1956 7 Claims. (Cl. 230—205)

This application is a continuation of our copending application Serial No. 696,349, filed November 14, 1957, now abandoned and relates back thereto, as to all common subject matter, for all dates and rights incident to the filing thereof and corresponding British application Serial No. 34,810/56, filed November 14, 1956.

The present invention relates to rotary piston, positive displacement compressors including two or more rotors disposed within an outer housing and formed with intermeshing helical lands and grooves, which in prior forms of this type of compressor have been operated dry (that is, except for appropriate lubrication of the bearings for the rotor shafts and for the gearing heretofore employed) and with the rotors not in physical contact with each other or with the housing, small clearances being maintained between the parts by employing suitable dimensions and providing time gears for connecting the rotors.

With such dry compressors, in which the compression chambers are sealed only by the so-called "space packing" provided by the close running clearances, high rotational speeds have been employed, which may be up to the order of 12,000 to 15,000 r.p.m., corresponding to tip speeds up to about 400 ft. per second for rotors six or about six inches in diameter, a size typical of the range of compressors usually drive by diesel or other usual types of internal combustion engines, or by normal speed electric motors. For such compressors to be used with such power, which operates normally in the 1500—2000 r.p.m. speed range, requires the use of step-up transmission gears.

Furthermore, with dry compressors, the maximum compression ratio practically obtainable in a single stage at acceptable efficiency is of the order of 4 to 1, even with externally cooled casings, the usual desirable upper limit being generally considered more nearly 3 to 1. When high pressures, such for example as for "shop" air or air for portable pneumatic tools at approximately 100 lbs./sq. in. gage pressure, representing compression ratio of approximately 8 to 1, is required, two stage compression with an intercooler has uniformly been resorted to.

It is accordingly a primary object of the present invention to provide a new and improved form of compressor of the general type under consideration which shall be capable of compressing air or other gaseous fluids efficiently to higher pressure ratios in a single stage than heretofore, to effect such compression at materially lower operating speeds enabling the compressor to be directly driven by internal combustion engines or normal speed electric motors, to enable timing gears to be dispensed with and to otherwise improve upon such compressors as presently developed.

For the attainment of the above and other and more detailed objects the nature of which will become apparent as this specification proceeds, the invention contemplates the introduction of liquid into the compressor for the dual purpose of providing a liquid seal closing the space packed clearance spaces characteristic of a dry compressor and for directly cooling the fluid being compressed to such material extent that compression can be effected to provide usual shop air pressure in a single stage at

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acceptable efficiency and at materially lower rotor speeds than has heretofore been possible. Further the invention contemplates such introduction to be made in such manner and also quantitatively in such relation to the speed of operation that optimum practical operating efficiency may be attained for any given desired set of conditions to be met.

The preferred manner in which the primary and other and more detailed objects hereinafter appearing may best be carried into effect, together with the advantages to be derived from use of the invention, will become apparent as the ensuing portion of this specification, taken in conjunction with the accompanying drawings forming a part hereof, proceeds.

In the drawings:

FIG. 1 is a diagram illustrative of one organization of apparatus suitable for carrying the invention into effect;

FIG. 2 is a side elevation, partly broken away in section, showing a suitable example of compressor for use in a system of the kind illustrated in FIG. 1;

FIG. 3 is a plan view taken on the line 3—3 of FIG. 2;

FIG. 4 is a section taken on line 4—4 of FIG. 2;

FIG. 5 is a section taken on line 5—5 of FIG. 2;

FIG. 6 is a diagram showing the effect at different rotor speeds of the introduction of different quantities of liquid; and

FIG. 7 is a diagram showing the relation of the quantity of liquid introduced into the compressor, in terms of the mass flow of the liquid to the mass flow of the gas being compressed, in relation to the tip speed of operation of the compressor rotors embodying the principles of the invention.

As will be more or less obvious, lubricating oil of the nature suitable for the lubrication of shaft bearings, is an appropriate liquid for use as the sealant and coolant employed in carrying out the present invention, and since both for such purposes as well as for bearing lubrication it is desirably supplied under pressure, it is convenient to use a common oil circulating system for all requirements, and such a system is illustrated by way of example but without limitation in FIG. 1. Referring now to the diagram of this figure, the compressor, as hereinafter more fully described, is shown at 10. Air entering the compressor is supplied through duct 11, and the compressed air is delivered through duct 12. One of the compressor rotors is driven by shaft 13, the other rotor being driven directly or indirectly from the first, and one of the rotor shafts 14 carrying pumps hereinafter referred to. For supplying the oil, a pressurized tank 15 is provided from which pump 16 draws oil through pipe 17, delivering it through pipe 18, a cooler 19, and pipe 20 to the interior of the compressor housing in a manner to be more fully described in detail later. A branch pipe 21 leading from pipe 18 conducts oil to the inlet end bearing 22. The oil supplied through pipe 20, the quantity of which is advantageously controlled by a suitable regulating valve 23, is discharged with the compressed air through the duct 12 to a separator 24, the separated oil being returned to tank 15 through pipe 25 while the air is delivered for use through conduit 26. Oil draining from the inlet end bearings 22 is delivered through pipe 27 to a pump 28 from which it is discharged in the example shown through pipe 29 to the compressor casing. This feature, however, may be omitted and the oil returned directly to the supply tank 15.

Referring now to FIGS. 2 to 5 there is illustrated by way of example but without limitation a compressor suitable for carrying the invention into effect. The example shown is suitable for and for purposes of discussion may be considered as a portable compressor having a single stage of compression of a ratio of the order of 8 to 1, to deliver air at rates (determined by the rotor size) within

the usual capacity range for such compressors, which are ordinarily furnished in a range of sizes varying in capacity from 150 to 1200 cu. ft. per minute and usually being powered by internal combustion engines operating in the medium speed range of from 1500 to 2500 revolutions per minute.

In the structure illustrated, the output or crankshaft of the prime mover, such as an internal combustion engine, is shown at 35, carrying a flywheel 36 encased in a housing 37 and also a driving gear 38. The compressor housing comprises separable upper and lower parts 39 and 40, respectively. Within the housing parts which together provide intersecting bores 41 and 42, male and female rotors 43, 44, respectively, are mounted to rotate in suitable shaft bearings 45 at the inlet end of the compressor and 46 at the outlet end, these bearings being carried by the lower housing part 40. The shaft at the inlet end of the male rotor 43 carries a gear 47 meshing with the driving gear 38, while the shafts of the rotors at the outlet end carry the meshing timing gears 48, 49 carried in overhung relation outside the bearings 46, for driving the female rotor from the male rotor and for holding the rotors in phase relation to each other. The rotors are provided with intermeshing helical lands and grooves, those of the male rotor being substantially outside the pitch circle of the rotor and those of the female rotor being substantially inside the pitch circle of the rotor. As shown, the lands have a total wrap angle of less than 360° and the profile of the lands and grooves advantageously may be of the form disclosed in Nilsson U.S. Patent No. 2,622,987, granted December 23, 1952, although insofar as the present invention is concerned the specific profile of the rotors is not controlling and rotors of other form may be employed.

The housing provides an inlet conduit 50 communicating with an inlet chamber 51 which in turn communicates with the bores 41, 42 through the inlet port 52, which as will be seen from the drawings is partly in the inlet end wall of the housing and partly in the barrel portion thereof, to provide combined axial-radial flow of air into the rotor grooves. As will be seen from FIG. 4, the major portion of port 52 is on the low pressure side of the plane through the rotor axes. At the outlet end the housing part 39 provides an outlet or discharge port 53, which as will be seen from FIGS. 2 and 5 is partly in the high pressure end wall of the rotor chamber and partly in the bore or barrel portion of the housing. Also, as will be seen from FIG. 5, port 53 is located entirely on the high pressure side of the plane through the rotor axes.

The operation of the apparatus in effecting compression is well known. The rotors are turned in the directions noted by the arrows in FIG. 5, the grooves in the rotors being filled as the inlet ends thereof pass the port 52, the inlet phase of the cycle terminating when the grooves pass out of registration with that port. The entrained air is carried upwardly in the grooves until a land of one rotor starts to enter a cooperating groove of the other rotor at the line of intersection 54 between the rotor bores on the upper or high pressure side thereof to initiate the compression phase of the cycle by forming chevron-shaped compression chambers comprised of two rotor groove portions delimited at one end by the high pressure end wall of the housing and at the other by the intermesh of the rotors. The latter point travels toward the high pressure end wall as the rotors continue to revolve, the compression chamber progressively shortening in length and decreasing in volume until it runs out to zero volume with its contents being discharged through the outlet port 53 during the final or delivery phase of the cycle which may be considered a continuation of the compression phase.

As will readily be understood, the inbuilt compression ratio obtained will be primarily determined by the size and configuration of the outlet port, which will determine the volume of the compression chamber when it is brought

into registry with the port. It will further be realized that this ratio can be altered by suitable valve controls of known nature but such expedients are not critical to the present invention. Suffice it to say that experience has proved that use of the principles of the present invention enables the desirable compression ratio of approximately 8 to 1 required for shop air to be obtained readily with a single stage compressor.

For supplying liquid to the compressor for sealing and cooling purposes, as described in connection with FIG. 1, pump 16, driven by an extension of the female rotor shaft, supplies oil through a suitable connection (not shown) to the pipe 20 (FIG. 2) having a series of branches 20a leading to a series of orifices or nozzles 55 situated in axially spaced relation along the line of intersection 54. The quantity of oil supplied is controlled by the regulating valve 23 which may be of any appropriate form. While the place of injection of the liquid is not critical insofar as the broader aspects of the present invention are concerned, it is of advantage in most instances particularly when rotors having profiles such as those disclosed in the aforesaid Nilsson patent or profiles of that general type are used. Such advantages and the advantages to be derived from the use of such profiles in compressors into which liquid is introduced are fully discussed in the pending Nilsson et al. application Serial No. 161,576, filed December 22, 1961.

In dry compressors of the general type under consideration, the efficiency obtained has primarily been dependent upon the closeness of the clearances which it has been practical to obtain and even more importantly, to maintain in service under the conditions, sometimes extremely rugged, under which many compressors, particularly portables, must operate. Such clearances have proved to be a very sensitive factor in dry compressor operation and it would appear only logical to assume, on the basis of prior experience with liquid handling gear pumps, wet rotary sliding vane type compressors and other forms of positive displacement fluid handling devices, that many of the difficulties encountered with dry compressors of the kind here under consideration might readily be ameliorated or entirely overcome by the expedient of introducing sufficient liquid, such as a thin oil or even water, to provide a seal for closing the clearance spaces providing the space packing for the chambers of a dry compressor. Not only should this action provide a more effective seal for preventing leakage from the compression chambers, but because of the liquid rather than gaseous nature of the sealant it should be possible to materially open up the clearance while still maintaining a seal.

Actual test experience, however, has provided such apparently logical conclusions to be wholly erroneous. Initial tests showed that marked loss in efficiency accompanied introduction of liquid, as compared with dry operation, which was reflected in an increase in the power input requirements for a given delivery of compressed air.

Analysis of such initial results indicated that introduction of liquid (hereinafter for convenience referred to as liquid injection) required complete revision of the nature of the concept of operation of the compressor. Because of the very much greater density and viscosity of even the thinnest and lightest of liquids as compared with gas, the losses due to the churning of a liquid injected into the compression chambers in the presence of closely fitting rotors turning at high speed, as compared with the losses incident to the turbulence of the gas in space packed dry chambers, far outweighed any improvement obtained by virtue of improved sealing due to filling the clearance spaces with liquid.

With such initial results it would further appear only logical to assume that increasing the quantity of liquid would only aggravate the losses, but contrary to that assumption, it has been found that the opposite is true when the factor of quantity of liquid injected is properly related

to the speed of operation of the rotors as well as to the quantity of air being compressed.

We have found that in the first place, injection of liquid at a rate far greater than that required only to provide a liquid seal, coupled with operation of the rotors in a wholly different range of tip speeds as compared with dry operation, is required if dry compressor performance is even to be matched, much less improved upon. We have further found that the desired improved results require these basic factors to be employed in a definitely related fashion.

In view of the very wide range of capacities, pressure ratios, absolute pressures and pressure differences, and qualities such as densities, specific heats, viscosities etc. of the many gases for which compressors of the kind under consideration are applicable, it is obviously not possible to discuss herein all of the combinations and permutations of such factors that may enter into any one specific compressor design embodying the present invention, but for the purpose of explaining its principles and enabling them to be utilized so as to obtain its benefits, it is believed that the following, particularly applicable to the large segment of the compressor field represented by portable air compressors, will be sufficient.

Tests of a single stage compressor for delivering 100 lb. per square inch gage (shop) air from atmospheric inlet pressure show that satisfactory results are obtained with tip speeds between 60 and 145 ft. per second and further show this speed range represents approximate limits for the best performance range. Also it appears that when liquid is injected, a higher inbuilt compression ratio than the theoretical ratio of 7.8 to 1, for example 9 to 1, may be required for optimum results.

Such tests have further demonstrated that while the most desirable quantity of liquid to be introduced will vary depending upon different specific compressor design features, the results obtained by varying the quantity introduced follow a definite and consistent pattern, as illustrated for example but without limitation, in the diagrams of FIGS. 6 and 7.

Referring now more particularly to FIG. 6, the effect on efficiency of operation by the injection of liquid at different rates is shown in relation to the speed of operation expressed in terms of tip speed of the rotors. For indicating efficiency the "specific power," that is, the horsepower required to deliver 100 cu. ft. per minute of shop air (an index commonly used in industry), is employed.

From the curves of this figure, several important factors are immediately apparent. Among these it is evident that there is a very definite optimum speed and also that within the useful speed range, increasing the quantity of liquid added results in increased efficiency. This latter factor may be explained by the fact that variations in the quantity of liquid injected do not appear to produce commensurate variations in the resultant churning losses, which remain relatively much more constant, while increase in the quantity of liquid added operates to increase efficiency due to the increased direct and efficient cooling effect. From the diagram, it might appear that adding liquid at a still higher rate would further increase optimum efficiency, and such might be the case. However, practical considerations become controlling when considering the maximum of liquid to be introduced, since the quantities involved become so large that the size and cost of the liquid pumping equipment outweigh any further advantage to be gained in efficiency. Likewise, when considering the minimum of liquid to be used, another practical limitation in addition to the factor of efficiency must be taken into consideration. In substantially all cases, the liquid employed will be of a combustible nature, ordinarily a hydrocarbon such as lubricating oil or the like, and in such cases the quantity employed must be sufficient to keep the temperature due to the heat of compression below that at which ignition might occur. With ordinary reciprocating compressors requiring the usual lubri-

cants, approximately 250° F. is considered the safe upper temperature limit.

Further from FIG. 6 it will be apparent that for optimum results the quantity of liquid introduced must be related to the speed of operation. As might be expected, when tip speed is increased, churning losses increase and if the quantity of liquid supplied is too great for the speed employed, decreased efficiency results. This factor is very clearly evidenced by the nature of the test curves shown in the figure, in which the rate of injection shown by curve B is 160% of that shown by curve A and that shown by curve C is 267% of curve A.

Tests of the kind of which those shown in FIG. 6 are typical show that the objects of the present invention are achieved with the amount of liquid introduced varying, on a volumetric basis, from 0.24% liquid to gas at maximum speed to 1.1% at low speed. These values correspond, under the assumed conditions, to ratios of from 1.5 to 1 to 10 to 1 on a mass flow basis, which is the more convenient basis to employ when considering the relationship under different pressure conditions between an elastic gaseous fluid and an incompressible liquid.

In the above noted range of mass flow ratios, we have found that for compressors for the service specifically under consideration here by way of example a mass flow ratio of approximately 4 to 1 is productive of substantial optimum results when the compressor is operated at the upper end of its speed range.

As previously noted, the relationship of the quantity of liquid introduced to the speed of operation is inverse. With the optimum quantity of liquid being reduced as the speed is increased. We have determined that for optimum results the relationship between the mass flows follows a definite pattern and from numerous tests of the kind productive of the curves shown in FIG. 6, we have found that the optimum relationship between the speed of operation and the quantity of liquid injection may be represented by a curve of hyperbolic characteristics such as that illustrated by curve D in FIG. 7, in which mass flow ratio is plotted against tip speed.

Curve D may be expressed in the terms of the following formula

$$M_r = \frac{C}{S} - 10$$

wherein  $M_r$  represents mass flow ratio,  $C$  represents a constant, and  $S$  represents rotor tip speed expressed in feet per second.

For optimum results such as are represented by curve D the value of the constant  $C$  is approximately 2,000. It will be understood however that commercially satisfactory operation is obtainable with the use of mass flow ratios other than that which is precisely productive of maximum efficiency and in many instances it may be found that other commercial factors may outweigh in importance the obtaining of maximum efficiency of operation. Accordingly it will be understood that the concept of the present invention is not limited in its scope to adherence to the specific value noted above, but includes the range of values within which satisfactory performance may be obtained. In terms of the above formula the satisfactory range is one in which the lower limit for the constant  $C$  is approximately 1,000 and the upper limit approximately 3,000. This range represents at the lower limit, denoted by curve E, values at which the specific power input becomes sufficiently high so that multiple stage compressions with intercooling becomes desirable rather than single stage compression, and at the upper limit, denoted by curve F, represents values at which the quantities of liquid involved become so large that the size and cost of the liquid pumping equipment outweigh any further advantages to be gained in the efficiency field. Further it will be understood that the above noted formula is considered applicable only down to a minimum value for the mass flow ratio of 1.5 to 1,

which will insure the maintenance of maximum temperatures not exceeding accepted safety standards.

From the foregoing it will be apparent that the benefits of the invention may be attained through the use of a wide variety of specific combinations of the major factors involved and the invention is accordingly to be understood as embracing all apparatus falling within the scope of the appended claims.

What we claim is:

1. A rotary piston, positive displacement, elastic fluid compressor having a housing structure including a barrel portion comprised of intersecting bores with coplanar axes providing a working space extending longitudinally of said barrel portion, said structure having a low pressure port communicating with one end of said space to provide an inlet a major portion of which is located at one side of the plane of said axes and a high pressure port spaced from said low pressure port to provide a discharge from said space, the major portion of which is located at the opposite side of said plane, rotors provided with helical lands and grooves having an effective wrap angle of less than  $360^\circ$  rotatably mounted in said bores and comprising a male rotor having lands provided with convexly curved flanks and intervening grooves the major portions of which lie outside the pitch circle of the male rotor and a female rotor having lands provided with concavely curved flanks 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 portions of said housing structure chevron shaped compression chambers each comprised of communicating portions of a male rotor groove and a female rotor groove, said chambers being defined at their base ends by an axially fixed transverse plane at which said high pressure port is located and at their apex ends by the intermeshing lands of the rotors and said apex ends moving axially toward said fixed plane and said chambers coming into communication with said high pressure port as the rotors revolve to cause said chambers to run out to zero volume at said fixed plane, and means for supplying liquid to said working space for sealing the perimeters of said chambers and cooling the contents thereof at a rate such that the ratio of mass flow of said liquid to the mass flow of the elastic fluid being compressed lies within a range of which the lower limit is substantially 1.5 to 1 and the upper limit is substantially 10 to 1.

2. A compressor as defined in claim 1, in which the tip speed of the male rotor lies within a range of which the lower limit is approximately 60 and the upper limit approximately 145 feet per second.

3. A compressor as defined in claim 2, in which the rotor speed is at the upper end of said range and liquid is supplied at a rate such that the mass flow ratio between the liquid and gaseous fluids is substantially 4 to 1.

4. A rotary piston, positive displacement, elastic fluid compressor having a housing structure including a barrel portion comprised of intersecting bores with coplanar axes providing a working space extending longitudinally of said barrel portion, said structure having a low pressure port communicating with one end of said space to provide an inlet a major portion of which is located at one side of the plane of said axes and a high pressure

port spaced from said low pressure port to provide a discharge from said space, the major portion of which is located at the opposite side of said plane, rotors provided with helical lands and grooves having an effective wrap angle of less than  $360^\circ$  rotatably mounted in said bores and comprising a male rotor having lands provided with convexly curved flanks and intervening grooves the major portions of which lie outside the pitch circle of the male rotor and a female rotor having lands provided with concavely curved flanks 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 portions of said housing structure chevron shaped compression chambers each comprised of communicating portions of a male rotor groove and a female rotor groove, said chambers being defined at their base ends by an axially fixed transverse plane at which said high pressure port is located and at their apex ends by the intermeshing lands of the rotors and said apex ends moving axially toward said fixed plane and said chambers coming into communication with said high pressure port as the rotors revolve to cause said chambers to run out to zero volume at said fixed plane, means for turning said rotors, and means for supplying liquid to said working space for sealing the perimeters of said chambers and cooling the contents thereof at a rate such that the minimum value of the ratio of the mass flow of the liquid to the mass flow of the elastic fluid is 1.5 to 1 and changes in inverse proportion to changes in the tip speed of the rotors.

5. A compressor as defined in claim 4, in which said change in inverse proportion is substantially in accordance with a hyperbolic function.

6. A compressor as defined in claim 4, in which said ratio varies in accordance with the formula

$$M_r = \frac{C}{S} - 10$$

in which

$M_r$  is the mass flow ratio

$S$  is the male rotor tip speed in feet per second

$C$  is a constant the value of which is in a range of which the upper limit is approximately 3,000 and the lower limit is approximately 1,000

7. A compressor as defined in claim 6, in which the value of  $C$  is approximately 2,000.

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