

Jan. 28, 1969

J. W. GARDNER

3,424,373

VARIABLE LEAD COMPRESSOR

Filed Oct. 28, 1966

Sheet 1 of 5

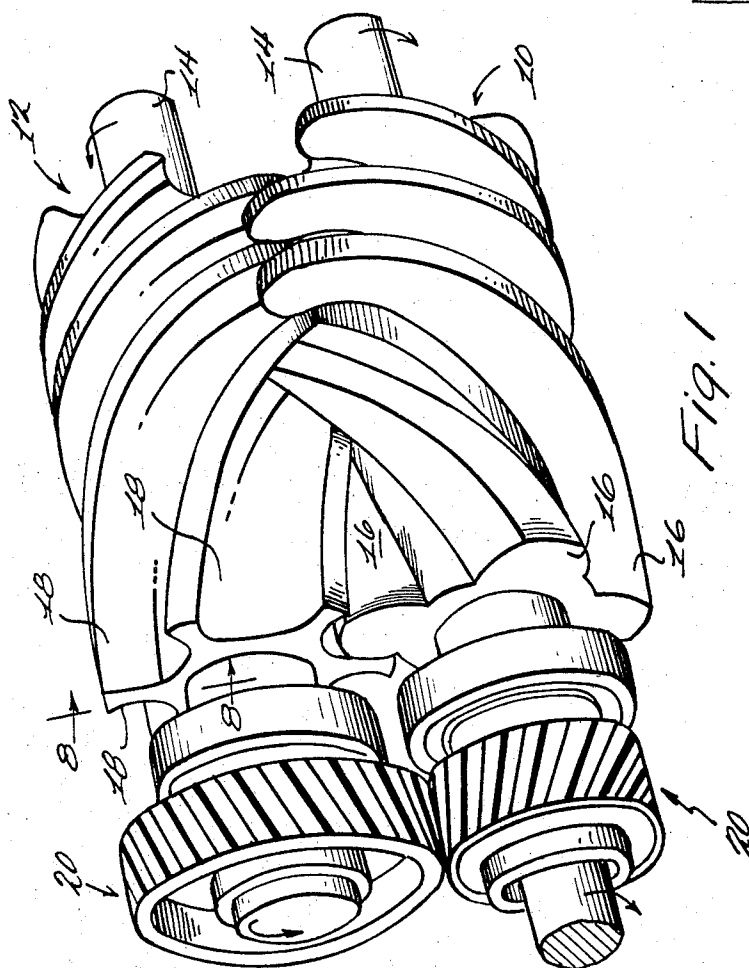


Fig. 1

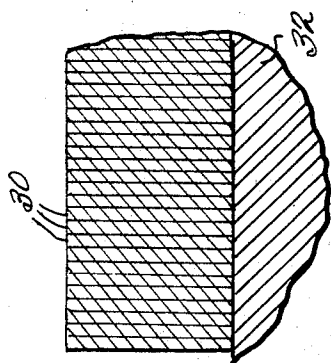


Fig. 8

INVENTOR
JOHN W. GARDNER

BY *Cushman, Doyle & Cushman*
ATTORNEYS

Jan. 28, 1969

J. W. GARDNER

3,424,373

VARIABLE LEAD COMPRESSOR

Filed Oct. 28, 1966

Sheet 2 of 5

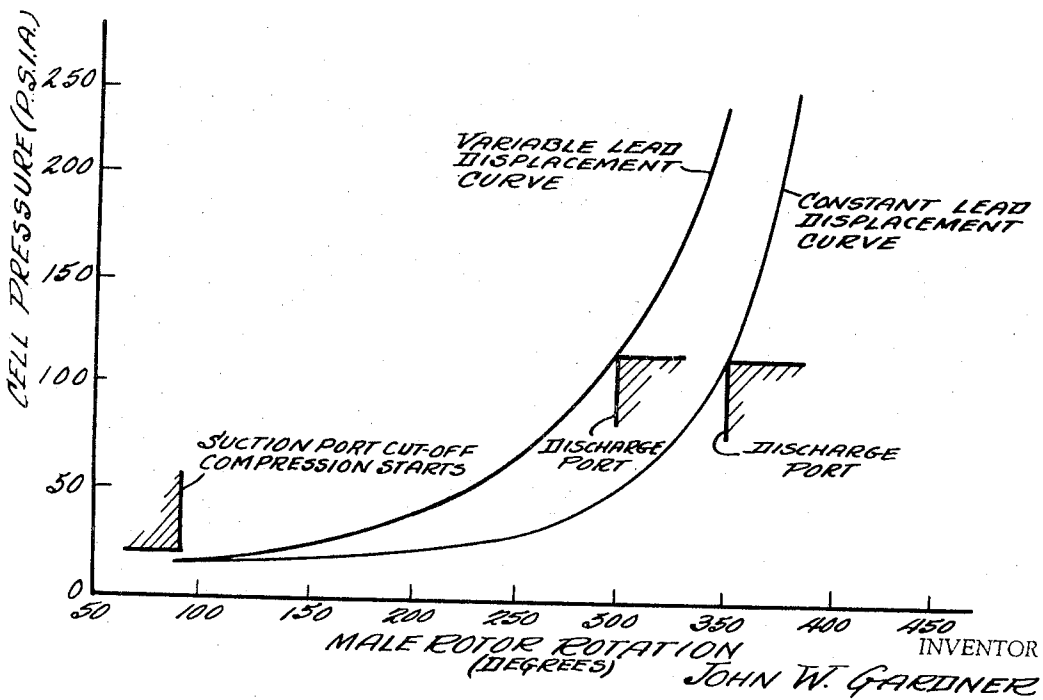
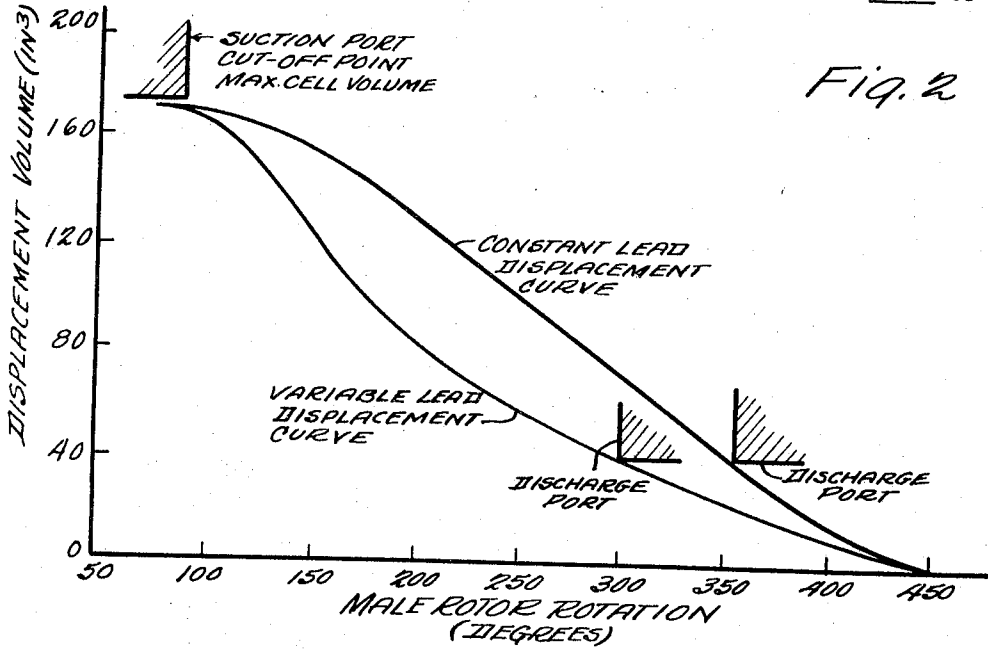


Fig. 3

INVENTOR
JOHN W. GARDNER
BY
Cushman, Kelley & Cushman
ATTORNEYS

Jan. 28, 1969

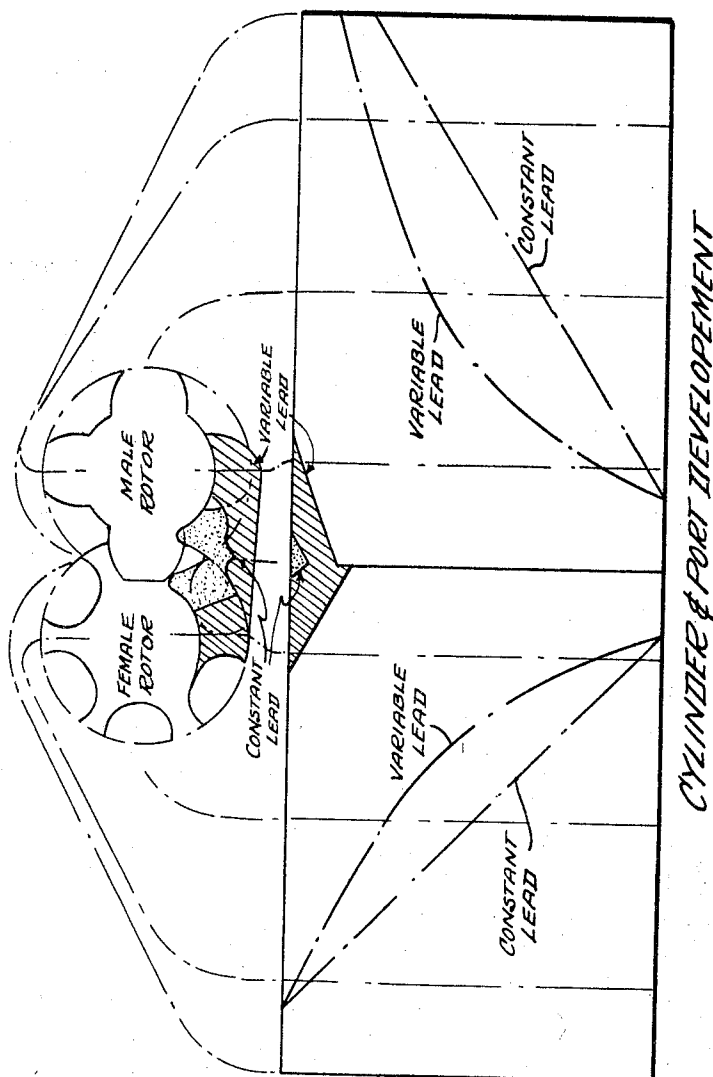
J. W. GARDNER

3,424,373

VARIABLE LEAD COMPRESSOR

Filed Oct. 28, 1966

Sheet 3 of 5



INVENTOR

JOHN W. GARDNER

BY

Cushman, Doherty & Cushman
ATTORNEYS

Jan. 28, 1969

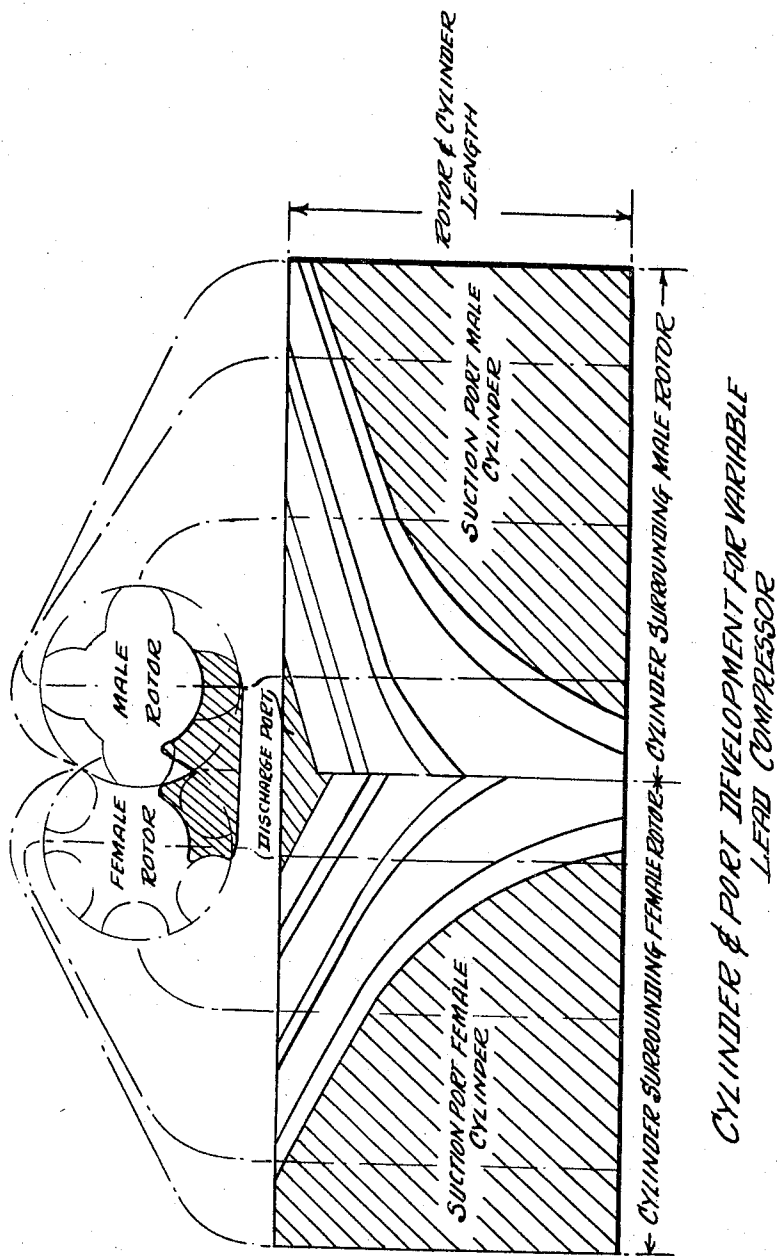
J. W. GARDNER

3,424,373

VARIABLE LEAD COMPRESSOR

Filed Oct. 28, 1966

Sheet 4 of 5



CYLINDER & PORT DEVELOPMENT FOR VARIABLE LEAD COMPRESSOR

Fig. 5

INVENTOR

JOHN W. GARDNER

BY

Cushman, Darby & Cushman

ATTORNEYS

Jan. 28, 1969

J. W. GARDNER

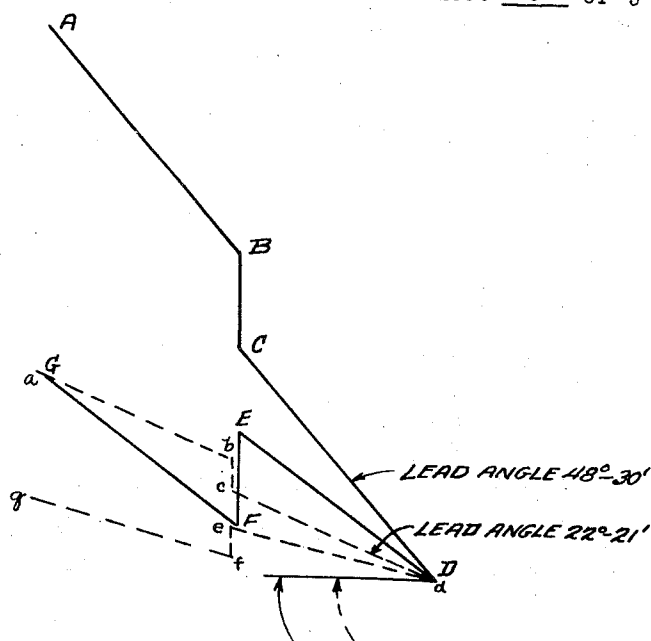
3,424,373

VARIABLE LEAD COMPRESSOR

Filed Oct. 28, 1966

Sheet 5 of 5

Fig. 6



(A-B = FEMALE ROTOR LEADING EDGE SEALING ON MALE ROTOR FLANK
(B-C = FEMALE O.D. SEALING ON MALE ROTOR ROOT DIAMETER
(C-D = FEMALE ROTOR LAGGING TIP SEALING ON MALE ROTOR FLANK
(D-E = MALE ROTOR LEADING EDGE SEALING ON FEMALE ROTOR GROOVE
(E-F = MALE ROTOR O.D. SEALING IN FEMALE ROOT DIAMETER
(F-G = MALE ROTOR LAGGING TIP SEALING IN FEMALE ROTOR POCKET

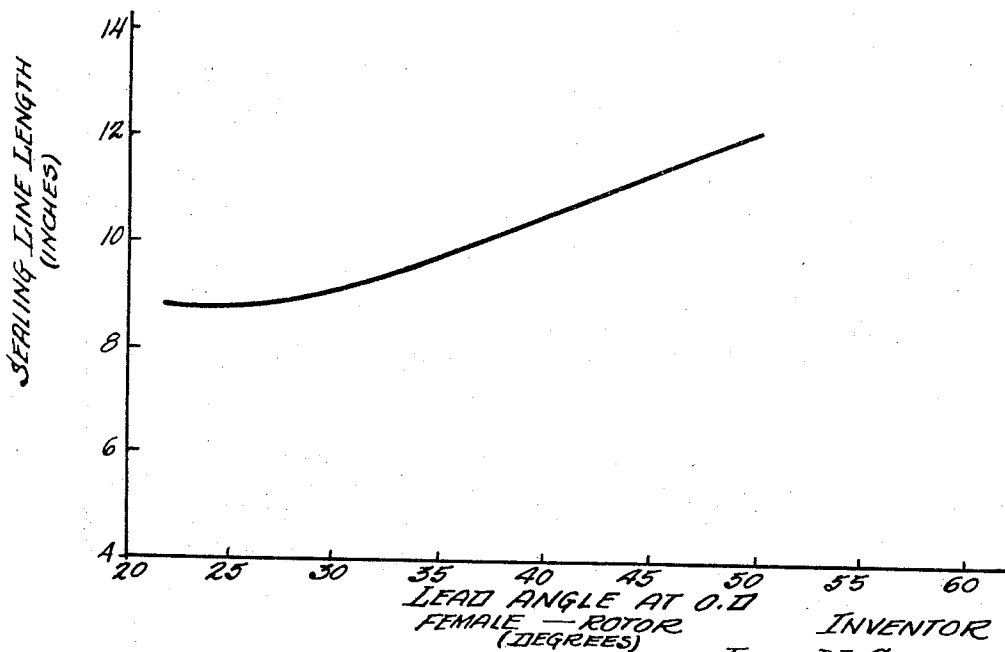


Fig. 7

INVENTOR
JOHN W. GARDNER

BY
Cushman, Daley & Cushman
ATTORNEYS

1

2

3,424,373

VARIABLE LEAD COMPRESSOR

John W. Gardner, 1 York Lane, Quincy, Ill. 62301

Filed Oct. 28, 1966, Ser. No. 590,322

U.S. Cl. 230-143

Int. Cl. F04c 17/04, 17/12

6 Claims 5

ABSTRACT OF THE DISCLOSURE

A fluid compressor having a pair of intermeshing rotors is provided with a continuously variable lead for the lobes and gates of the intermeshing rotors. The variable lead extends from an inlet end of the compressor to an outlet end of the compressor, and the lobes of the male rotor, of the pair of rotors in the compressor, have wrap angles of less than 360 degrees. The intermeshing relationship between the pair of rotors is such that a sealing line is formed with a decreasing length from the inlet end of the compressor to the outlet end of the compressor. A method of assembling a variable lead compressor is described.

This invention relates to an improvement in helical screw type compressors, and in particular, it relates to a compressor of the type in which the helix has a continuously variable pitch or lead which results in improved compression characteristics.

Rotary type screw compressors are well known in the art, and commonly include two or more intermeshing rotors positioned within a housing and provided with some driving means for rotating the intermeshed rotors relative to one another. Such a structure is shown in Lys-holm et al. Patent 2,111,568, Mar. 22, 1938. Compressors of this type are used for compressing gases and an inlet is provided for introducing gas into one end of the housing for compression and movement axially to an outlet at an opposite end of the housing. Such rotary screw compressors may be operated dry where no liquid is injected into the compression chamber, or wet where a liquid is injected for use in the compression chamber. Dry compressors rely upon close tolerances between intermeshing lobes and gates of the rotors to effect a seal between the rotors, and such a sealing arrangement is commonly referred to as "space sealing." The use of liquids in a compression chamber of screw compressors provides for additional sealing between the rotor clearances, and further provides for removal of the heat of compression. A compressor which is operated dry typically has had a maximum compression ratio of 4:1, whereas the wet operated compressor, such as described in Bailey Patent 3,073,514, Jan. 15, 1963, may have a compression ratio of up to 9:1. Such prior art screw compressors are constructed with constant lead lobes and gates on intermeshing rotors, which construction has necessarily limited the efficiency and performance of the compressors.

Some of the difficulties in obtaining acceptable efficiencies at higher compression ratios in prior constant lead machines have resided in throttling losses which occur because of small discharge ports and in high leakage losses in the high compression zones because the leakage line lengths are fixed. The leakage factor of a helical screw compressor may be related to the length of sealing line which is formed between intermeshed lobes and gates of two rotary members. By adjusting the length of the sealing line for a given pair of rotors in a compressor, a control of compressed fluid leakage can be accomplished. This invention has found that the angles of a sealing line provide for a critical control of the leakage in a compressor, and a construction having a con-

tinuously variable lead or pitch for the lobes and for the gates of a pair of rotors provides a desired leakage control. As a result of this improved construction, it has been found that compression ratios above 9:1 in a wet operating compressor, and above 4:1 in a dry operating compressor, can be efficiently attained in a machine having similar length and diameter ratios to prior machines having constant pitch ratios.

The compressor of this invention employs but two rotors, thus obviating the known disadvantages attendant constructions which employ more than two. Moreover, the wrap angle of a compressor embodying this invention is less than 360°, again obviating greater manufacturing costs attendant greater wrap angles.

In addition to the above advantages, the compressor of this invention provides greater capacities for a unit of given size by increasing the discharge area for the unit. Prior devices that have been designed to attain high compression ratios have necessarily included limited area discharge ports which result in throttling losses within the compressor. In the present device of this invention, a higher compression ratio and capacity can be attained with a given size of discharge port, and this accounts for substantially improved efficiencies.

A method of assembly will also be described with reference to the invention, and the method includes the formation of thin, plated metal discs which are formed to provide the cross sectional configurations of the compressor rotors. A plurality of the discs are stacked together and arranged with a template to set the desired variable lead in the rotor components. After the desired pitch is attained, with each disc being slightly offset relative to the next one, the entire assembly can be fused together by heating to a temperature which will melt the platings on adjoining surfaces of the discs. Finally the entire assembly may be finished and coated with a suitable material.

These and other advantages of the present invention will become apparent in the more detailed discussion which follows, and in that discussion, reference will be made to the accompanying drawings in which:

FIGURE 1 illustrates a perspective view of a pair of compressor rotors having variable lead intermeshing elements;

FIGURE 2 is a graph showing the improved displacement characteristics of the compressor of this invention as compared to a constant lead rotor construction;

FIGURE 3 is a graph showing the improved compression characteristics of the present invention as compared to a constant lead compressor;

FIGURE 4 is a cylinder and port development for a rotor pair;

FIGURE 5 is a complete cylinder and port development for a variable lead compressor;

FIGURE 6 is a depiction of sealing line lengths as lead angle changes from one end of a rotor to another;

FIGURE 7 is a graphic description of sealing line length as compared to the lead angle of a female rotor; and

FIGURE 8 illustrates a vertical cross section of a portion of a rotor which has been assembled by a plurality of stacked discs.

Compressors having interengaging screw rotors are well known in the art, of compressing gases, and the theory for forming a pocket of gas which is compressed and carried from an inlet end of a unit to an outlet end of the unit is also well known. In the usual compressor design, the interengaging rotors have wrap angles of less than 360 degrees and length over diameter ratios (L/D) ranging from 1.0 to 2.0. Such conventional compressor units include rotors having a constant lead or pitch to their

3

respective lobes and gates. It is also known in the prior art to provide for two stage compressors having two separate compressor elements combined together to form a single rotor wherein each of the compressor elements has a different constant lead to its lobes or gates. Such constructions require an assembly of four separate elements to provide for a two rotor compressor having two separate stages of compressing capacity.

The present invention provides for compressor rotor structures wherein the rotors have continuously variable leads which result in an improved compression efficiency for the unit.

FIGURE 1 illustrates a perspective view of a set of rotor units embodying this invention for use in a compressor having a continuous single stage for compression. The rotor units in a given set comprise a male rotor 10 and a female rotor 12, and the two rotors are mounted on parallel shafts 14 so that they may be driven in counter-rotational directions to each other. When the rotors are driven, the lobes 16 of the male rotor 10 fit into the gates or grooves 18 formed within the female rotor 12, and the intermeshing of the two rotors causes a compression of a gas pocket trapped between intermeshing lobes and gates. The illustrated rotors are of the type wherein the male rotor 10 has four lobes and wherein the female rotor 12 includes six gates 18. Further, the rotors are of the well known geometry providing for the tips of the lobes 16 to lie outside of the pitch circle of the male rotor and for the bases of the gates 18 to lie within the pitch circle of the female rotor. The lobe and gate configurations can be circular or generated, or any of the well known combinations or variations therefrom. The compressor unit includes a conventional housing (not shown) that surrounds the two rotors so as to confine gas which is admitted into an inlet end of the compressor unit. The housing structure and means for confining gases, such as the use of end plates in conjunction with the housing, are well known in the art and do not form a separate part of this invention. The set of rotors illustrated in FIGURE 1 may include gears 20 which engage with each other to provide for a timed rotation of one rotor relative to the other. The gears 20 are normally incorporated in compressors which are operated dry, however, it has been found that gears may be entirely omitted from wet operating compressors.

The lobes and gates of the rotors are formed in continuous variable leads which provide for a wrap angle of less than 360° for a lobe of a male rotor. Of course, the wrap angle of the female rotor corresponds to the wrap angle of the male together with a consideration of the number of lobes and gates used in a particular rotor set. When mounted in a conventional casing for use as a compressor, an inlet for the set of rotors would be at the left end of the FIGURE 1 drawing, and an outlet would be at the right end of the drawing. Accordingly, gas would enter the set of rotors at the end having the lesser lead for the lobes and gates, and the gas would be exhausted in a compressed state at the end of the set where the lobes and gates intermesh at a greater lead.

Substantial benefits are obtained from a compressor having rotors of a variable lead, as shown in FIGURE 1. The compressor of this invention provides for improved volumetric efficiencies in gas compression as a result of (a) increased compression of a gas pocket due to the ever increasing lead for the lobes and gates as the gas pocket moves from an inlet to an outlet of the compressor, (b) less throttling losses as a result of a larger discharge area which is available, and (c) decreased leakage losses due to an improved sealing line characteristic. The invention provides for a compressor having an increased capacity for a given length over diameter ratio (L/D) of a unit, and also increased compressions are efficiently attainable within such a compressor unit. The above advantages and benefits will be discussed with reference to the graphic illustrations of FIGURES 2 through 7,

4

FIGURES 2 and 3 compare displacements and cell pressures for compressors having constant lead rotors and variable lead rotors of a type made in accordance with this invention. The FIGURES 2 and 3 analyze only one pressure cell in a compressor having a four lobe male rotor and a six gate female rotor. There are four separate compression cycles per revolution of the male drive rotor, but of course it is understood that the number of compression cycles per revolution would vary with the rotor ratio (male rotor lobes to female rotor gates) and/or the particular rotor which is utilized for driving. Referring to FIGURE 2 there is shown a displacement-time diagram of one cell of a conventional constant lead screw rotor set as compared to a variable lead rotor set of the type constructed in accordance with this invention. The rotor sets selected for illustrating the displacement-time diagram of the variable and constant lead compressors are dimensionally similar in that the rotor lengths, rotor diameters, rotor wrap angles, theoretical displacement and rotor profile configurations are substantially identical. With the FIGURE 2 graph the port locations for the compared compressors can be determined in a well known manner (as is known for compressors of the fixed port type wherein there is a built-in internal volume ratio which may be defined as the ratio of suction port cut off volume to discharge port expelled volume). For example, FIGURE 2 shows the total cell (male and female) theoretical displacement volume for constant lead and variable lead rotor sets to be 172 cubic inches. Assuming that gas being compressed is air ($n=1.4$) and that the compression cycle follows very nearly to the adiabatic compression cycle, the volume required at the discharge port to obtain a discharge pressure of 114.7 p.s.i.a. or 100 p.s.i.g. would be 39.6 cubic inches. This relates to a volume ratio of 4.3 to 1. By a well known thermodynamic formula this can be illustrated:

$$\frac{P_2}{P_1} = \left(\frac{V_2}{V_1}\right)^n \quad P_2 = 14.7 \left(\frac{172}{39.6}\right)^{1.4} = 114.7 \text{ p.s.i.a.}$$

Applying this formula to the theoretical displacement diagram of FIGURE 2, pressure time cards for the constant lead and variable lead rotor sets can be constructed and compared as shown in FIGURE 3.

By analyzing FIGURES 2 and 3, it is readily apparent that the variable lead rotor set cell will reach the required volume ratio and its related discharge pressure of 114.7 p.s.i.a. approximately 52 degrees earlier than the standard lead rotor set cell. Since the total wrap angles of both rotor sets (constant lead and variable lead) are identical, the time remaining to expell the gas is greater. This time difference is directly related to the discharge port size. FIGURES 2 and 3 have been marked to show discharge port locations for 114.7 p.s.i.a. for the two types of rotor constructions when operating on air ($n=1.4$) and following an adiabatic compression cycle. It is clear from the FIGURES 2 and 3 that a greater discharge port area for higher discharge pressures are available in the variable lead rotor set built in accordance with the present invention; and as will be discussed below, throttling losses are reduced for a compression unit utilizing this arrangement.

In order to further illustrate the benefit of increased discharge port area which is obtained with the present invention, FIGURE 4 illustrates a typical diagrammatical layout of a cylinder which surrounds a rotor set and a discharge end plate for a compressor. The FIGURE 4 diagram is a cylinder and end plate port development for constant and variable lead rotor sets having axial and radial discharge port locations. Of course, it is understood that the rotor sets may be in compressor units having only axial ports or only radial ports, as are well known in the art. The FIGURE 4 development includes an identification of the suction and discharge ports which are determined from the information of FIGURES 2 and 3. Again, the illustrated rotor pairs are of the type wherein the male rotor

5

includes four lobes and the female rotor includes six gates or grooves, however, similar developments could be made for other well known rotor configurations and rotor ratios. In the FIGURE 4 comparison, the exhaust port areas for a conventional constant lead rotor set is identified by the stripped sections shown on the end plate development and on the cylinder development immediately below the end plate illustration. The comparable discharge port area for a variable lead rotor set is identified by all of the stippled area together with the cross hatched section which is shown. It is readily apparent that the variable lead rotor set will provide a substantially greater discharge port area, and this results in lower throttling losses in a compressor unit. Throttling loss, as is well known in the art, considers the amount of pressure backup within a compressor unit which results from the inability of compressed gas to be easily and quickly discharged. As a larger discharge port area is made available, the throttling or backup pressures will be diminished and there will be less throttling leakage within the compressor unit.

FIGURE 5 shows a complete cylinder and port development of the type shown in FIGURE 4, but illustrating a variable lead rotor set only. The development of FIGURE 5 also shows the locational relationships of the male and female rotor outside diameters with respect to the surrounding housing.

FIGURE 6 illustrates a further advantage obtained from the variable lead construction of this invention. In prior rotor type compressors of the constant lead type, the compression efficiencies have been depended upon the geometry of the set of intermeshed rotors used in a given compressor unit. The geometry includes such variables as the number of lobes and gates, length and diameter of the individual rotors, and the wrap angle which is utilized with respect to the lobes and gates formed on individual rotors. Assuming much of the geometry of the present device to be comparable to prior art types of rotor sets, a further variable is adjusted in the present invention to improve the efficiency of a rotor pair in a compression operation. This variable is concerned with the compression cell sealing line length, and the length of sealing line is directly related to the amount of leakage loss which may occur within a compressor unit. The sealing line of a rotor set can be considered the line of closest proximity between intermeshed lobes and gates in the set. Since the rotors are not in actual contact with each other at any time, the sealing line represents the closest point of contact and is determinative of the amount of leakage which will occur between intermeshed rotors in a compressor construction. The heavy line A-G of FIGURE 6 represents a sealing line configuration at the intake end of a rotor pair made in accordance with this invention; and the dashed line a-g represents a sealing line configuration at an exhaust end of a rotor pair made in accordance with the present invention. It can be seen by comparing the sealing line configuration at the intake end of the rotor pair to the sealing line at the exhaust end that there is a substantial reduction in sealing line lengths as the lead angle of a rotor pair varies. Thus, in accordance with the present invention there is provided an ever decreasing sealing line length between intermeshed rotors from one end of the compressor unit to the other. The reduction in sealing line length takes place toward the critical end (that is, the exhaust end) of the unit where greater pressures are developed within the compressor cells, and therefore, gas leakage between intermeshed rotors at this critical portion of a compressor unit is substantially reduced. The length of sealing line is a critical factor in the performance and efficiency of a unit, regardless of whether the compressor is operated wet or dry, and the reduction in sealing line length toward an exhaust end of a unit results in an improved efficiency of the compressor unit. The FIGURE 6 comparison is taken at the extreme ends of a rotor set constructed in accordance with the present invention, but it is to be understood that the sealing line length is a

6

continuously changing factor because of the continuously variable change in lead angle of the rotor pair built by the present invention.

Sealing line lengths in a conventional constant lead compressor remain constant throughout the length of a given compressor unit, and of course, this means that there is no reduction in leakage between intermeshed rotors as the gas pocket is compressed from the intake end to the exhaust end of a unit. Such is not the case with the present invention, since the sealing line length reduces as the gas pocket is compressed toward the exhaust end of the unit. FIGURE 7 further illustrates the sealing line length relationship to varying lead angles of a typical female rotor construction as the lead angle changes from one end of the rotor to the other. It is apparent from the FIGURE 7 graph that there is a considerable and continuous reduction in the sealing line length from the intake end of the rotor to its exhaust end. Of course, it is understood that the illustrations of FIGURES 6 and 7 are by way of example only, and that numerous rotor profiles exist which have different numbers of sealing lines and different configurations for sealing lines. Also, it is possible to plot the sealing line as a function of the male rotor lead angle instead of the illustrated female rotor lead angle.

Having described the constructional features of the present invention, and having depicted the benefits obtained thereby, it can be seen that the provision of a continuously variable lead for a rotor pair provides substantially improved compression results. The continuously variable lead for intermeshing rotors provides for a more rapid compression of a gas pocket, resulting in a capability of discharging the compressed gas in a higher state of compression. Also, the improved construction of this invention provides for a larger discharge port area for a given compressor unit, and this results in less throttling losses within the unit from a backup of compressed gas between intermeshed rotors. Finally, the construction of this invention provides for an improved sealing line characteristic which becomes shorter as a gas pocket is moved from an inlet end to a discharge end of a compression unit. The shortening of the sealing line results in less gas leakage between intermeshed rotor pairs.

As an example of an operating construction for a compressor unit of this invention, rotor sets can be made which include length over diameter ratios within the usual range of 1.0 to 2.0. The male rotor can be rotated at a lobe tip speed within a range of 60 to 125 feet per second, and the female rotor is operated at a speed to accommodate the male rotor speed. In such a unit, the compression reached approximately 230 p.s.i.a. as compared to approximately 115 p.s.i.a. on a conventional unit of similar dimension but having constant lead lobes and gates. This difference in compression capacity can be attributed to a greater progressive compression of a fluid pocket and also to improved efficiency resulting from lower leakage losses in the unit of this invention.

The rotor set of this invention may be formed by any suitable method such as by casting or cutting the required lobes and gates into the separate rotors, or by machining the same. By way of example the individual rotors may be formed from an assembly of stacked plates which are mounted face to face along the central longitudinal axis of the rotor to be formed. Each succeeding plate in the assembly is slightly offset relative to other plates, and the amount of offset determines the change in lobe or gate pitch which is required for the construction. FIGURE 8 illustrates a vertical section taken on line 8-8 of FIGURE 1, and this section illustrates a series of adjoining plates 30 which are assembled side by side to form the desired structure for a rotor. The stacked plates may be initially mounted on an arbor and cut to provide a desired profile and lead angle, as would be formed in a standard compressor rotor having a constant lead rotor construction. Existing cutting devices can be used for

forming such a constant lead and profile in a rotor surface. Then the discs can be removed from the arbor and re-mounted on cores 32 where they are displaced into a continuously variable lead of the required pitch. A template may be used for displacing or setting the stacked discs into the desired variable helix. Then the entire assembly can be heated to the melting temperature of the surface of the individual discs, and the entire assembly will be fused together into a solid structure. In order to facilitate brazing or melting of adjoining surfaces of individual discs, it has been found that an improved unit can be formed by coating the assembly, prior to heating, with a silver base alloy. The silver base alloy may be applied in a fluid form so that capillary action will carry the alloy between adjoining discs of the assembly. This provides a coating on the adjoining surface of the disc, and the heating step then results in a solid bonding of adjacent disc to one another. Finally, the assembled rotor may be coated with molten tin, or other material, to provide a smooth surface on the rotor configuration which is formed. It has been found that such coating materials as epoxy resins and metal alloys can be added in a fluid state while the rotor pair is being slowly rotated together and intermeshed. This method has resulted in an improved fitting of the lobes and gates of the rotor pair to one another in the finished set.

Alternatively, the variable lead can be formed from stacked discs which are not staggered in a progressive fashion relative to one another but wherein each disc has a slightly different helix angle in its profile. The assembly of such discs would result in a predetermined variable lead angle, and the bonding techniques could be of the type described above.

Although the term "gas" has been used throughout the above description, such language contemplates a change of state of certain gases which may take place during compression, and therefore the term may include liquid condensation or liquids generally. Also, other changes and obvious variations may be made in the invention and such variations are intended to be included within the scope of this invention.

What is claimed is:

1. A fluid compressor having improved compression characteristics and an improved availability of an increased size for an outlet port leading out of the compressor, said fluid compressor being of the screw type having only a pair of complementary intermeshing cylindrical rotors mounted in a housing having cylindrical chambers for receiving said pair of rotors and for controlling the transfer and compression of fluids from an inlet into one end of said housing to an outlet from an opposite end of said housing, the improvement in said rotor pair which comprises:

a male rotor having a plurality of equally spaced lobes projecting from its cylindrical surface, said lobes extending from one end of the rotor cylinder to the other end of the cylinder along equally spaced paths

which continuously vary in lead as they progress along and around the surface of the rotor, said lobes having wrap angles of less than 360 degrees and a female rotor having a plurality of equally spaced gates or grooves formed in its cylindrical surface, each of said gates being formed to complement and receive a lobe from said male rotor when said rotor pair is mounted in intermeshing relationship in a compressor unit, said intermeshing relationship being such that a sealing line formed between the intermeshed rotor pair decreases in length from the inlet end of the compressor to the outlet end thereof.

2. The fluid compressor of claim 1 wherein the variable lead of the lobes and gates for each of said rotors progresses from a relatively slight angular relationship to a central longitudinal axis of an associated rotor to a relatively steep angular relationship to said associated central longitudinal axis as the lead varies from an inlet end of said rotor pair to an outlet end of said rotor pair, whereby pockets of fluids are progressively compressed as they are transferred from the inlet end to the outlet end upon rotation of the intermeshed pair of rotors.

3. The fluid compressor of claim 1 wherein the male rotor is rotated at a lobe tip speed within the range of approximately 60 feet per second to approximately 145 feet per second.

4. The fluid compressor of claim 1 wherein the lobes of the male rotor lie outside of the pitch circle of said rotor and wherein the gates or grooves of said female rotor lie within the pitch circle of said female rotor, said lobes and gates intermeshing to form compression chambers which compress and confine fluid pockets in a progressive movement of the fluid from an inlet end of the housing to an outlet end of the housing, and said cylinder chambers within said housing further confining said pockets of fluid to compression chambers formed by the rotation of said rotor pair.

5. The compressor of claim 4 wherein each lobe of said male rotor has a convex surface configuration which includes two apexes which generate two spaced thread points relative to the concave base of a complementary gate.

6. The compressor of claim 1 wherein said rotors are constructed with a ratio of length to diameter (L/D) within a range of 1.0 to 2.0.

References Cited

UNITED STATES PATENTS

630,648	8/1899	Brewer	91—84
2,148,205	2/1939	Kiesskalt	103—128
2,804,260	8/1957	Nilsson et al.	230—143
3,314,597	4/1967	Schibbye	230—143

DONLEY J. STOCKING, *Primary Examiner*.

WILBUR J. GOODLIN, *Assistant Examiner*.

PO-1050
(5/69)

**UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION**

Patent No. 3,424,373 Dated January 28, 1969

Inventor(s) John W. Gardner

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

┌

Column 3, line 57, "lesser" should be corrected to read --greater--;

line 59, "greater" should be corrected to read --lesser--.

**SIGNED AND
SEALED
JAN 27 1970**

(SEAL)

Attest:

Edward M. Fletcher, Jr.

Attesting Officer

**WILLIAM E. SCHUYLER, JR.
Commissioner of Patents**

└