United States Patent

[11] 3,632,239

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[31] 181,008

[54] **ROTATABLE WORM FLUID COMPRESSION-EXPANSION MACHINE**
8 Claims, 9 Drawing Figs.

[52] U.S. Cl. **418/150,** 418/195, 29/156.4
[51] Int. Cl. **F01c 1/24,** F04c 17/04, F04c 17/16
[50] Field of Search 418/150, 195; 29/156.4; 74/458

[56] **References Cited**

UNITED STATES PATENTS

711,083	10/1902	Taylor	418/195
2,279,414	4/1942	Scott	74/458
2,603,412	7/1952	Chilton	418/195
3,180,565	4/1965	Zimmern	418/195
3,181,296	5/1965	Zimmern	418/195

FOREIGN PATENTS

334,636	2/1920	Germany	74/458

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ABSTRACT: A fluid compression-expansion machine comprising a worm rotatable inside a casing and at least one toothed pinion meshing with the worm and rotatable about an axis transversal to the axis of the worm, wherein the dimensions and relative positions of the worm and the pinion are so selected that there exists at least one axis of insertion for the pinion which satisfies the equation:

$$d_ih_i/r_i \quad d_jh_j/r_j$$

in which:

d_i is the algebraic distance between the axis and a point M_i of contact between one flank of a pinion tooth and a worm thread, the flank in question being on the opposite side of the tooth from the axis,

h_i is the algebraic distance from this point M_i of the perpendicular dropped from the center of the pinion onto the tangent at point M_i to the flank of the tooth,

r_i is the absolute value of the distance from point M_i to the projection of the axis of the worm on the plane of contact between the worm and the pinion,

d_j, h_j, r_j are analogous quantities relating to a point M_j of contact between a flank of a tooth and a thread, this flank being on the same side of the tooth as the axis.

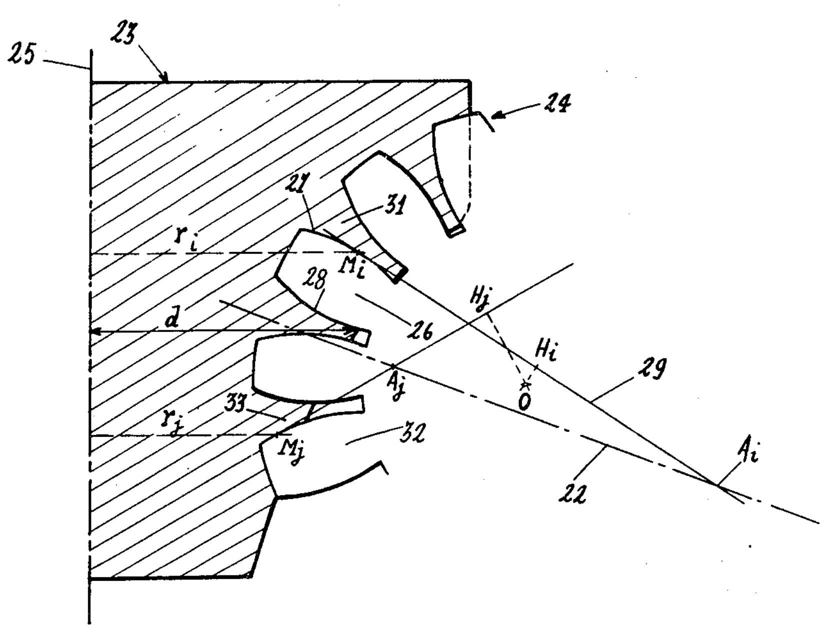


Fig. 1

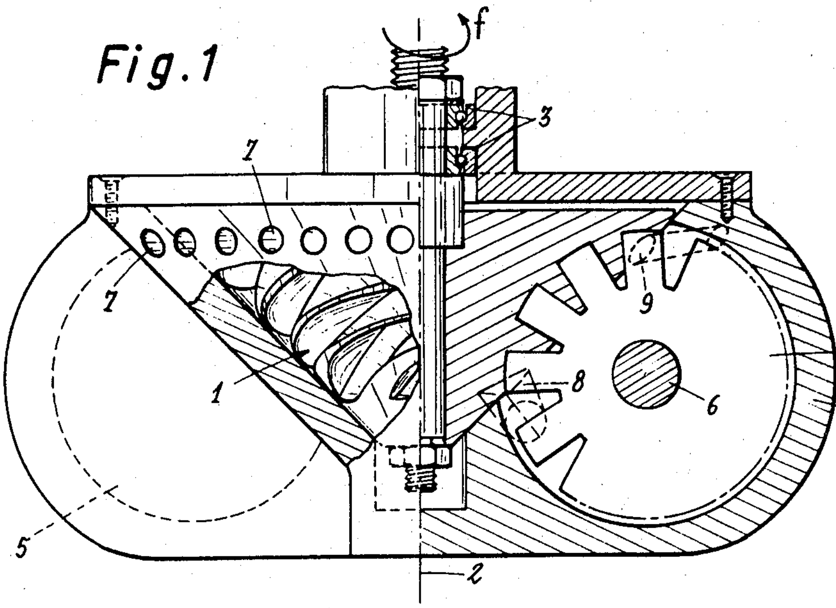


Fig. 2

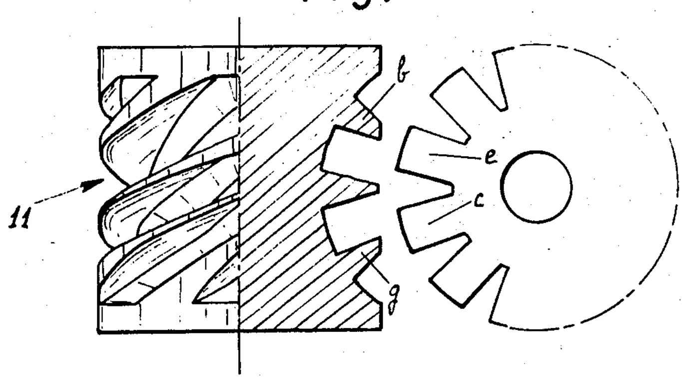
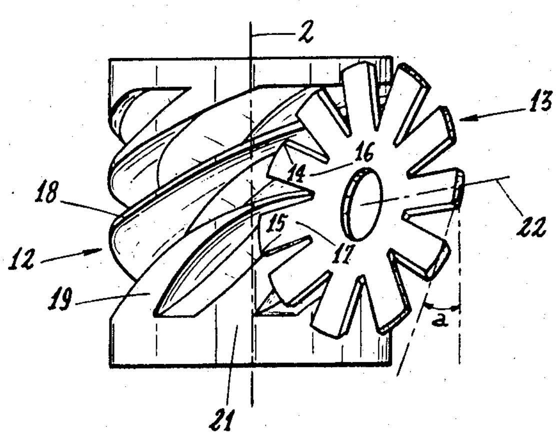


Fig. 3



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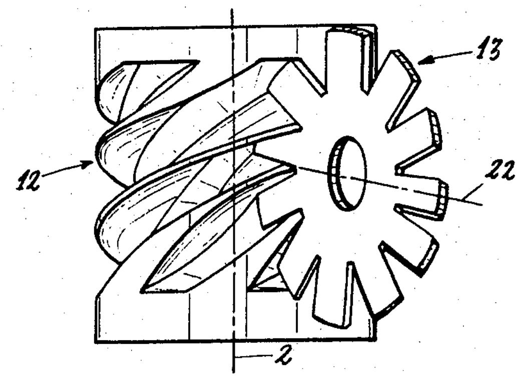

Fig. 4

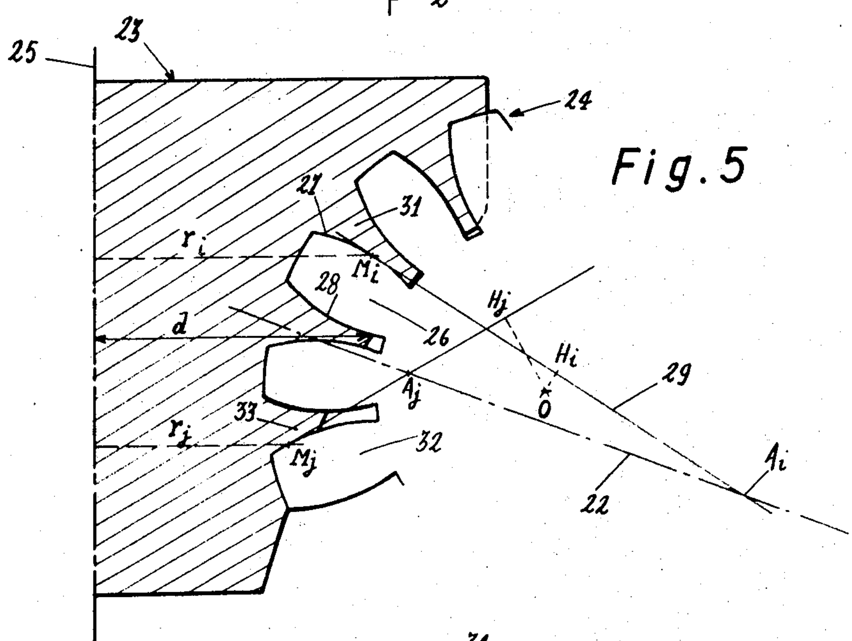

Fig. 5

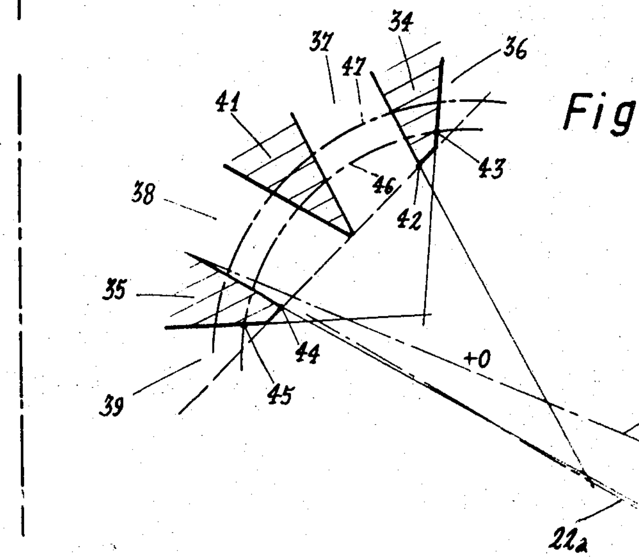

Fig. 6

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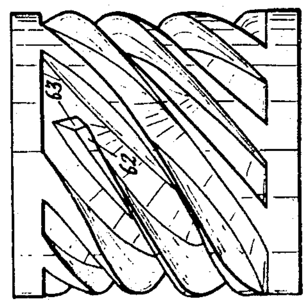

Fig. 9

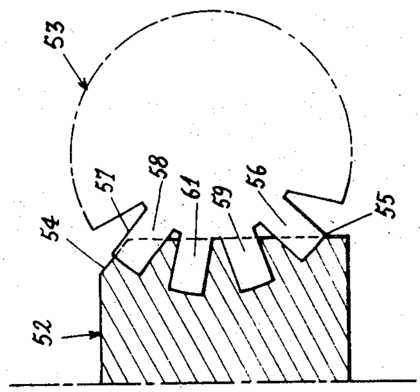

Fig. 8

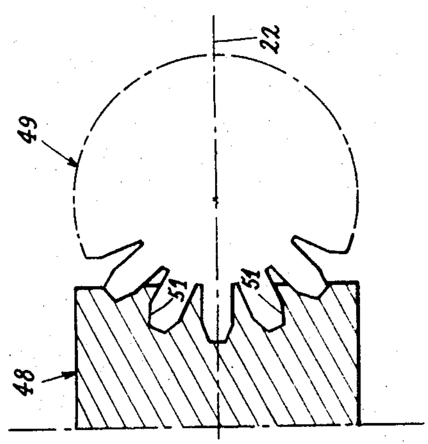

Fig. 7

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ROTATABLE WORM FLUID COMPRESSION-EXPANSION MACHINE

This invention relates to a machine of the kind comprising a worm rotatable about a first axis with respect to a casing, and at least one toothed pinion meshing with the worm and rotatable about an axis, with respect to said casing, which is transverse to said first axis, the crests of the worm threads lying on a common surface of revolution about said first axis, the casing having a portion making sealing engagement with a least a portion of said common surface of revolution, extending between circumferentially adjacent sides of two circumferentially adjacent said pinions, or between opposite sides of said pinion where a single pinion is used, the casing sealingly engaging said pinion sides.

The form of the worm and of the or each pinion is such that the portions of the pinion teeth within said common surface of revolution make fluidtight contact with the worm teeth so that fluidtight chambers are defined between the worm teeth, the casing and the pinion teeth, the casing having an inlet and an outlet and the arrangement being such that as the worm and pinion rotate in mesh one such chamber communicating with the outlet increases in volume to draw fluid into the chamber, is subsequently sealed off from the inlet by a pinion tooth, is transferred between pinion teeth to the outlet, is communicated with the outlet and has its volume decreased to discharge its fluid through the outlet. A machine of the above kind is hereinafter referred to as being "of the kind specified". A worm having the form necessary to fulfill the conditions set out above is hereinafter referred to as a "globoid worm".

A machine of the kind specified may be used as a compressor or as a pressure reducer.

The pressure variation ratio (compression or expansion ratio) of a machine of the kind specified depends on the number of worm threads meshing simultaneously with the teeth of the pinicn. The higher this number is, the lower is the residual volume of the compression or expansion chambers at the moment at which they are connected to the high-pressure orifice, (i.e., the outlet in the case of a compressor, or the inlet in the case of a pressure reducer) and the higher, therefore, is the pressure variation ratio.

In some known machines of the kind specified, the number of threads simultaneously meshing with the pinion teeth is high. In these machines, however, the worm and pinion cannot be fitted together without splitting either the worm or the pinion into several parts. Since the worm and pinion rotate at a high speed they must be manufactured with great precision. However accurately the various parts constituting the worm or pinion are made individually, some "division faults" in the gearing will inevitably be introduced when they are assembled. Also, the considerable heating to which the machine is subjected may produce expansion of the components which is not evenly distributed over the different parts of a split worm or pinion element, so that the division faults are aggravated during operation. Rapid wear may result, very soon making the machine unserviceable.

In other known machines of the kind specified, the number of threads simultaneously in mesh with the pinion is high and the worm and pinion can be fitted together without splitting either of these elements. To achieve this, however, the threads of the worm must have profiles such that the volume between adjacent, threads is very small, with the result that the delivery rate and the power-to-weight ratio are low. Such a machine is described in French Patent specification No. 1,331,998.

French patent specification No. 1,331,998 also describes compressors in which the outer profile of the worm, (i.e., the envelope of the thread crests) is cylindrical, and by means of which fairly high compression ratios can be obtained without splitting the worm or pinion. However, in such compressors the number of pinion teeth simultaneously in engagement with the worm cannot exceed three, and in certain angular positions of the worm only a single tooth of the pinion has both of its flanks in contact with adjacent thread flanks of the worm. The compression ratio is thereby limited to a value of the order of six, so that when such a compressor is used to compress air, compressed air at the standard pressure of 7 bars can be obtained. This compression ratio is too low for some purposes, for example for refrigerator compressors.

The French patent specification already mentioned also describes compressors in which the crests of the worm threads define a conical or plane surface of revolution about the worm axis. These compressors can give a compression ratio better than six, with the same number of teeth in engagement as the compressors with cylindrical worms mentioned above. They can therefore be assembled without splitting either the worm or the pinion. However, these compressors produce considerable axial thrust, and it is necessary to arrange two compressors in opposition in order to obviate such thrust, especially in the case of plane worms. Also, the sector of engagement of the pinion with the worm must be restricted, and in many cases the worm must be truncated on the low-pressure side, which reduces the delivery rate.

It is an object of the present invention to provide a form of machine of the kind specified which can be made to have a high delivery rate and a high compression or expansion ratio and can be assembled without splitting the worm or pinion.

According to the present invention there is provided a machine of the kind specified in which the dimensions and the dispositions of the worm and pinion are so selected that, for at least one angular position of the worm and for at least one straight line situated in the plane containing the lines of contact of the pinion with the worm, the following equation is satisfied, for at least part of that portion of the teeth which is in contact with threads of the worm:

$$d_i h_i / r_i \geqslant d_j h_j / r_j$$

in which:

- d_i is the algebraic distance between the straight line and a point M_i of contact between one flank of a pinion tooth and a worm thread; this distance being measured from the M_i along the tangent at this point the flank of the tooth, the flank in question being on the opposite side of the tooth from the straight line;
- h_i is the algebraic distance from this point M_i to the perpendicular dropped from the center of the pinion onto the said tangent, this distance being measured from the point M_i;
- r_i is the absolute value for the distance from the point M_i to the projection of the axis of the worm on the plane of contact between the worm and pinion;
- d_j, h_j, r_j are analogous quantities relating to a point M_j of contact between a flank of a tooth and a thread, this flank being situated on the same side of the tooth as the straight line.

It has been found that, if these geometrical conditions are satisfied, it is possible without splitting either the worm or the pinion, to assemble machines of the kind specified in which at least three teeth on the pinion mesh simultaneously with the worm and in which the number of teeth which are in contact with the worm threads along both their flanks is never less than two, whatever the angular position of the worm.

Under these conditions, the worm and pinion can be assembled by making the latter carry out a composite displacement combining translation and rotation about said straight line which will hereafter be termed "insertion axis."

The optimum conditions obtain when there is only one axis for which the equation set out above is satisfied. The maximum pressure variation ratios and the maximum deliveries compatible with the use of one-piece worms and pinions can therefore be obtained.

According to a preferred embodiment of the invention the worm and pinion are formed so that insertion axis exists only in a number of ranges of relative angular positions of the worm and pinion said number being equal to the number of worm threads, these zones being each of only a few degrees extent and being obtained by appropriate truncation of the worm on the low-pressure side.

Alternatively the worm and pinion may be formed so that insertion axis exist only in a single range of relative angular position of the worm and pinion, said range being of only a few degrees extent and being obtained by truncation of a single thread on the low-pressure side.

The maximum pressure variation ratio and the maximum delivery are therefore obtained, though the worm and pinion can only be assembled in certain angular positions of the worm, possibly only one angular position.

Further features of the invention will be clear from the following detailed description with reference to the accompanying drawings, wherein;

FIG. 1 is an elevation view partly in section and partly cutaway, of a first embodiment of the invention, with a globoid worm having a conical outer surface;

FIG. 2 is an elevation view, partly in section, of a globoid worm having a cylindrical outer surface a pinion adapted to cooperate with this worm, before it is assembled with the worm in a machine according to the invention;

FIGS. 3 and 4 are elevation views illustrating respectively the initial and final phases in assembly of the pinion and worm of FIG. 2;

FIG. 5 is a diagram illustrating the geometrical relationships between a worm and pinion of a machine according to the invention;

FIG. 6 is a diagram illustrating a method of obtaining the relative dimensions of a worm and pinion for a machine according to the invention;

FIG. 7 is a diagrammatic section illustrating one embodiment of worm and pinion for a machine according to the invention;

FIG. 8 is a diagrammatic section illustrating another embodiment of a worm and pinion for a machine according to the invention; and

FIG. 9 is a perspective view illustrating yet another embodiment of a worm for a machine according to the invention.

A pump according to the invention for use as a compressor or pressure reducer is illustrated in FIG. 1. This machine has a globoid worm 1, the crests of whose threads define a conical surface of revolution about the rotary axis of the worm. The worm 1 is mounted so that it is rotatable about its axis 2 and is supported by ball bearing 3 in a casing 4. Two symmetrically disposed toothed pinions 5 are rotatable about their axes 6 and mesh with the worm 1. The casing 4 has a series of low-pressure orifices 7, which are spaced around its periphery and form the fluid intake in the case of a compressor or the fluid outlet in the case of a pressure reducer. On the high-pressure side, the casing 4 contains two orifices of triangular cross section, forming an outlet for the discharge of compressed fluid in the case of a compressor or an inlet for the intake of fluid for expansion in the case of a pressure reducer. One if these orifices is indicated by broken lines at 8 in FIG. 1, this orifice being in fact situated in front of the plane of section.

Fluidtightness of the compression or expansion chambers, each of which is defined by two adjacent threads on the worm 1 the pinion teeth and the internal surface of the casing 4, is ensured in a conventional manner by liquid seals. Sealing liquid is injected into the machine on the intake side for the fluid being compressed or expanded, near the pinions 5. Injectors 9 are provided on the low-pressure side of the intake of this sealing fluid if the machine is operating as a compressor. If the machine is being used as a pressure reducer, these injectors are housed in the fluid intake orifice 8.

The worm 1 turns in the direction of the arrow f if the machine is operating as a compressor and in the opposite direction if it is operating as a pressure reducer.

A worm having a cylindrical surface, such as the worm 11 shown in FIG. 2, may be substituted for the conical worm 1. It should be noted that, for both the conical worm 1 and the cylindrical worm 11, there are, for all angular positions of the worm, at least two teeth on the pinion 5 which are simultaneously in contact along both their flanks with the threads on the worm. It is therefore impossible to insert the pinion in the worm merely by means of translation, unless, as indicated above, the teeth and threads are tapered, which considerably reduces the delivery and power-to-weight ratio of the machine. If the worm and pinion are brought together by mere translation it will be seen from FIG. 2 that the tooth e comes to bear on the tread b, and the tooth c, on the tread g. In conventional machines of this type, the problem of assembling the worm and pinion is resolved by splitting either the worm or the pinion into a plurality of parts. However, as already stated, anisotropic deformations then arise during operation and soon make the machine unserviceable.

In the embodiments illustrated in the drawings, one-piece worms and pinions are used for which, in any angular position of the worm, at least two teeth on the pinion are simultaneously in contact along both their flanks with the threads on the worm.

It has been found that, if certain geometrical conditions, which will be stated below, are satisfied, there is at least one angular position of the worm in which such worms and pinions can be assembled without splitting them.

The manner of assembly is illustrated in FIGS. 3 and 4 for a cylindrical worm 12 and a pinion 13. The pinion 13 (FIG. 3) is inclined so that its plane forms an angle a with the axis of the worm 12, and the ends 14, 15 of the teeth 16, 17 which must ultimately be in contact along both their flanks with the threads 18, 19 and 21 of the worm, are brought into contact with the two end threads 18, 21 respectively. The pinion 13 is then inserted along an axis 22, which will be defined below, while subjecting this pinion to rotation about this axis 22 in such a way as to reduce the angle of inclination a progressively, until the pinion 13 is in position (FIG. 4). Hereafter an axis such as 22 will be termed an "insertion axis".

The geometrical conditions which the worm 23 and pinion 24 must satisfy in order to ensure that at least one insertion axis 22 exists, will now be stated with reference to FIG. 5.

The plane of FIG. 5 is the plane containing the lines of contact of the pinion 24 with the worm 23 when the pinion is in its operating position. This plane does not necessarily contain the axis of the worm 23. However, it is assumed that the distance from this axis to the plane of the Figure does not exceed 20 percent of the external diameter d of the worm 23 in the center of the sector of cooperation between the worm and pinion, or that the angle formed by the worm axis with the plane of the Figure does not exceed 20°. The projection of the axis of the worm 23 onto the plane of the Figure is indicated at 25.

An axis 22 is traced in this plane, and the conditions under which this axis forms an insertion axis, as defined above, will now be defined.

In relation to this axis 22, each tooth 26 has a flank 27 situated on the opposite side of the tooth from the axis 22 and a flank 28 situated on the same side of the tooth as the axis 22. For the sake of simplicity, flanks such as 27 will be termed "remote flanks" and flanks such as 28 "adjacent flanks".

The tangent 29 to the flank 27 at a point M_i has been traced from the point M_i on a remote flank 27, the point M_i being a point of contact between the tooth 26 and a thread 31 on the worm. This tangent intersects the axis 22 at a point A_i. A perpendicular OH_i is dropped from the center O of the pinion 24 onto the tangent 29. The distances M_iA_i and M_iH_i measured algebraically along the tangent 29 from the point M_i will be designated d_i and h_i respectively. The absolute value for the distance from the point M_i to the projection 25 of the worm axis will be termed r_i.

Take a point M_j of contact between another tooth 32 and another worm thread 33, this point M_j being situated on a flank facing the axis 22 relative to the tooth 32. The relative quantities at the point M_j, respectively homologous to the quantities d_i, h_i and r_i just defined for point M_i, will be termed d_j, h_j and r_j.

The axis 22 is an insertion axis in the sense defined above if the following condition is satisfied for every pair of points such as M_i and M_j:

$$d_i h_i/r_i \geqslant d_i h_j/r_j \qquad (1)$$

In the particular case in which the teeth of the pinion 24 have parallel flanks, the condition (1) stated above need only be verified for two pairs of points belonging respectively to the two extreme threads **34, 35** (FIG. **6**) of the worm which are in contact with the teeth of the pinion.

FIG. **6**, for example shows a pinion whose teeth, which have parallel flanks, mesh with a worm having a conical outer surface. For at least one angular position of the worm, the pinion has four teeth **36** to **39** which are in contact with three threads **34, 41** and **35** on the worm. The intermediate teeth **37, 38** are in contact with the threads along both their flanks simultaneously whereas the extreme teeth **36, 39** come into contact with the threads along one of their flanks only.

In this case, the condition (1) must be verified for the following two pairs of points. The first pair is formed by two points **42, 43** on the extreme thread **34**, point **42** being situated on the tooth **37** at the periphery of the worm, on that flank remote from the axis **22**, and point **43** being situated on the adjacent flank of the tooth **36** at the periphery of the pinion. The second pair is formed by points **44, 45** homologous to the points **42, 43** and situated on the other extreme thread **35**.

If the worm has a cylindrical outer surface and the cross section of this surface through the plane of the pinion is symmetrical relative to the perpendicular dropped from the center of the pinion onto the projection of the worm axis on this plane, the possible insertion axes, if they exist, must comprehend this perpendicular, which forms an axis of symmetry.

The condition (1) imposes certain limits on the dimensions of the worm and pinion and on their relative positions.

FIG. **6**, therefore, illustrates how to determine, by means of the condition (1), the maximum diameter of the pinion such that at least one insertion axis exists.

In this example, the generatrix of the conical outer surface of the worm is assumed to have an inclination of 35° relative to the worm axis. The pinion has **11** teeth, each 16 millimeters wide. The distance from the center of the pinion to the worm axis is 80 millimeters and the distance between this center and the generatrix of the cone is 30 millimeters. The maximum diameter of the pinion for which at least one insertion axis exists may be determined graphically as follows, proceeding by means of successive approximations:

An initial diameter of the pinion is assumed, such that the profile of this pinion corresponds to the circle **46** having the center **0**. A first axis **22** is traced, such that the condition $d_i h_i/r_i = {}_j h_j/r_j$ is satisfied for the pair of points **42, 43** situated on the extreme thread **34**. This can be carried out in an infinite number of ways, since only the quantities h_i, r_i, h_j, r_j, are fixed in the equation given above.

The position of the axis **22** is then modified so that, with the above-mentioned equation still satisfied, the quantity *dh/r*, relating to the point **44** and the point **45** defined above, has a value respectively greater than and lower than the common value relating to points **42** and **43**.

If a position of the axis **22** which satisfies this condition can be found, this signifies that a plurality of possible insertion axes exist and that the value assumed for the pinion diameter is less than the maximum value.

The diagram is then begun again for a greater value of the pinion diameter, until a diameter, corresponding to a pinion profile **47**, is found for which a position **22*a*** of the axis **22** exists such that the quantity *dh/r* has the same value for the four points **42** to **45**. This position **22*a*** then represents the only possible insertion axis, and the profile **47** corresponds to the maximum pinion diameter. With the numerical values given above, this maximum diameter is found to be of the order of 91 millimeters. Beyond this diameter, the worm and pinion cannot be assembled without splitting one or the other of these two elements.

Obviously, this determination of the maximum diameter may be carried out in other ways. In particular, an electronic computer could be used.

Similarly, it is found (FIG. **2**) that, for a cylindrical worm symmetrical relative to the perpendicular dropped from the center of the pinion onto the worm axis, and for the following values:

worm diameter: 100 mm.
number of teeth in pinion: 11
center-to-center distance from worm to pinion: 80 mm.
width of teeth: 16 mm.

the maximum diameter for the pinion is of the order of 94 mm. The only insertion axis is then the axis of symmetry, i.e. the perpendicular dropped from the center of the pinion onto the worm axis.

In this example, and those below, all the dimensions may, of course, be multiplied by any given factor without the conditions of existence of the insertion axis being affected.

Similarly, it is found that, for a cylindrical worm, there is no insertion axis for the following condition:

worm diameter: 100 mm.
pinion diameter: 100 mm.
number of teeth: 15
width of teeth: 11 mm.
center-to-center distance from worm to pinion: 80 mm.
teeth having parallel flanks.

Under these conditions, the worm and pinion cannot be assembled without splitting one or other of these elements. For an insertion axis to exist, the pinion diameter must be reduced to 84 mm. In this case, the pinion has five teeth in engagement with the worm, three of which are, in the insertion position, in contact with the worm threads along both their flanks.

The geometrical relation (1) given above was established for ideal conditions, i.e. for teeth which mate perfectly with worm threads. In practice, some play must be provided between the flanks of the teeth and the flanks of the threads. Experience shows that the total play (i.e. the sum of the play on one side of a tooth and the play on the other side) may be up to 1 to 2 percent of the width of the teeth without substantially diminishing the delivery rate.

This play facilitates assembly of the worm and pinion and makes it possible to exceed the theoretical limits imposed by the condition (1).

In practice, this condition (1) need only be satisfied over at least 70 percent of the effective height of the tooth, in the case of teeth having parallel flanks. The effective height of a tooth is defined as that length of the flank of the tooth which is in contact with the threads of the worm when the pinion is fully inserted.

Take, for example, cylindrical worms with six or eight threads, having a diameter of 100 mm., and cooperating with pinions whose teeth have parallel flanks and which have 11 and 15 teeth respectively. It has been found that, for a center-to-center distance of 80 mm., the effective height of the teeth can be increased by a factor of the order of 45 percent over the theoretical limit, if the total play equals 2 percent of the width of the teeth.

This effective height of the teeth can be further increased if the flanks are not parallel over the entire height of the tooth.

FIG. **7** shows a cylindrical worm **48** with a diameter of 100 mm. cooperating with a pinion **49** having 15 teeth. The width of the teeth is 11 mm., and the center-to-center distance between the worm and pinion is 80 mm. It has been demonstrated above that, with these characteristics, there is no insertion axis for a pinion diameter greater than 84 mm. if the teeth have parallel flanks. In the embodiment in FIG. **7**, the flanks of the teeth are parallel over a height of 15 mm. from the base of the teeth, and then have a convergent portion **51** forming an angle of the order of 30° with the tooth axis. Under these conditions, the pinion diameter may be up to 100 mm. There is then an insertion axis **22**, which is the axis of symmetry.

If, for this pinion diameter of 100 mm., the flanks of the teeth were parallel over their entire height, the delivery would be very slightly greater than in the embodiment shown in FIG. 7. This increase in delivery would not exceed a few percent. However, this very slight advantage would be very largely compensated by the disadvantages (already mentioned) arising from splitting of the worm or pinion.

It is very important to note that it is by no means essential that the worm and pinion can be assembled in all angular positions of the worm. On the contrary, the worm and pinion may have profiles and dimensions such that assembly is only possible in a limited number of angular positions of the worm, possibly even only one of these positions.

FIG. **8**, therefore, shows a cylindrical worm **52**, having an external diameter of 100 mm. and cooperating with a pinion **53** having 13 teeth with parallel flanks. The center-to-center distance between the worm and pinion is 80 mm., and the effective height of the teeth is 20 mm.

If the worm **52** is cylindrical over its entire height, the pinion **53** has three teeth simultaneously in engagement along both their flanks with the worm threads whatever the angular position of the worm. It is found that, for this effective tooth height, no insertion axis which fulfills the condition (1) exists. Either the tooth height must be reduced, or the worm or pinion must be split.

However, it is possible to make the profile of the worm **52** such that at least one insertion axis exists for a limited number of angular positions of the worm.

For example, as FIG. **8** shows, the worm **52** may be truncated on the low-pressure side by means of a conical surface **54**, which is coaxial with the worm. It will be noted that, for a certain number of worn positions such as that shown in FIG. **8**, when the end **55** of the tooth **56**, which is in contact with the worm on the high-pressure side, is flush with the cylindrical contour of the worm, the flank **57** of the tooth **58**, which is in contact with the worm on the low-pressure side, is free on account of the truncation. In this position only two teeth, **59** and **61** are in contact with the worm threads along both their flanks, and the worm and pinion can be assembled without splitting either. The length of engagement of the worm with the pinion, measured at the periphery of the pinion, extends substantially over a whole number of teeth. In this embodiment the number of positions of the worm which permit assembly of the worm and pinion is equal to the number of worm threads.

Instead of truncating all the threads on the worm, as FIG. **8** shows, a single thread may be truncated, as indicated in FIG. **9**. In this embodiment only thread **62** is truncated at **63** on the low-pressure side. There is then only one angular position of the worm for which the worm and pinion can be assembled, and in this position only two teeth of the pinion are in contact along both their flanks with the worn threads.

Solutions such as those illustrated in FIGS. **8** and **9** give both a maximum compression or expansion ratio and a maximum delivery. In particular, machines may be constructed in which the pinion has only two teeth in contact with the worm along both of their flanks in at least one angular position of the worm, and at most four teeth in engagement with the threads in all positions of the worm. Also, machines may be built in which the pinion has, in at least one angular position of the worm, only three teeth in engagement along both their flanks with the worm threads, the number of teeth in engagement being at most five for all positions of the worm.

It should be noted that the condition (1) is independent of the number of threads on the worm. This number can therefore be selected in such a way that each thread extends, in a known fashion, over an angular sector of the worm substantially equal to the angular interval separating two successive pinions. This last condition, of course, permits optimum use of the volume between two adjacent threads. The exact angular extent of each thread may be determined while taking well-known considerations into account, for example the fact that the number of worm threads and the number of teeth on the pinions are preferably incommensurable.

Machines according to the invention can give high deliveries and compression ratios, while using one-piece worms and pinions which do not deform during operation and which ensure long service.

With a cylindrical or conical worm having six threads and cooperating with a pinion having 11 teeth, the delivery and compression or expansion ratio are increased by more than 10 percent over conventional machines having the same dimensions.

Above all, however, the solution provided by means of the invention makes it possible to make machines of the kind specified having cylindrical worms with compression or expansion ratios greater than 10. For example, it is possible to use a worm having eight threads and a pinion having 15 teeth, five of which are simultaneously in engagement with the worm threads, and three of these teeth being in contact with the worm threads along both the flanks of the teeth when in the assembly position. Using conventional techniques, such high compression ratios can be obtained only with conical worms, which, as already stated, have the disadvantage of producing considerable axial thrust, or with worms or pinions made in a plurality of parts, which deteriorate rapidly. These high compression ratios are particularly desirable in the refrigeration industry.

I claim:

1. A fluid compression or expansion machine comprising a casing, a worm rotatable in said casing about a first axis, at least one toothed pinion rotatable in said casing about a second axis which is transverse to said first axis, said worm having a plurality of threads in meshing engagement with the teeth of said pinion, said casing having passageways for the fluid at lower and higher pressure respectively and having a portion in sealing engagement with the crests of the worm threads and defining fluidtight chambers between the casing and the threads of the worm, said chambers coming successively into communication with the higher pressure passageway on the high-pressure side of the worm and being successively sealed off from the lower pressure passageway by a pinion tooth, the number of the teeth of one pinion which simultaneously mesh with threads of the worm being greater than three for at least one angular position of the worm about said first axis, the first of said simultaneously meshing teeth having at least part of its flanks inclined at an angle with respect to the flanks of the last of said simultaneously meshing teeth and the profile of the worm threads and pinion teeth and the dimensions of the worm and the pinion being such that for at least one straight line situated in the plane which contains the lines of contact of the pinion with the worm and for at least one angular position of the worm about said first axis, the following condition is satisfied for at least part of that portion of the flanks of the teeth which is in contact with the worm threads, and for any pair of points M_i and M_j located on the flanks of the teeth as defined herebelow:

$$d_i h_i / r_i \geqslant d_j h_j / r_j$$

in which:

M_i is a point of contact between a worm thread and one flank of a pinion tooth, said flank being on the opposite side of said tooth from said straight line,

M_j is a point of contact between a worm thread and one flank of a pinion tooth, said flank being on the same side of the tooth as said straight line,

d_i is the distance between point M_i and said straight line, said distance being algebraically measured from point M_i along the tangent at point M_i to said flank of the tooth,

h_i is the distance from point M_i to the perpendicular dropped from the center of the pinion to said tangent, said distance being algebraically measured from point M_i along said tangent,

r_i is the absolute value of the distance from point M_i to the projection of said first axis on the plane of contact between the worm and the pinion,

d_j, h_j, r_j, are the same as d_i, h_i, r_i, but relative to point M_j.

2. A machine as claimed in claim 1, in which at least one angular position of the worm, only two teeth of the pinion are in contact along both their flanks with the threads of the worm, the pinion having in some positions of the worm four teeth meshing with the threads of the worm.

3. A machine as claimed in claim 1, in which each tooth of the pinion has parallel flanks, and in which said condition is satisfied over at least 70 percent of that portion of the teeth which is in contact with the threads of the worm.

4. A machine as claimed in claim 1, in which in at least one angular position of the worm, only three teeth of the pinion are in contact along both their flanks with the threads of the worm, the pinion having, in some positions of the worm, five teeth meshing with the threads of the worm.

5. A machine as claimed in claim 4, in which both the flanks of each tooth are parallel over part of their length and converge towards their outer ends.

6. A machine as claimed in claim 1, in which the meshing length of the worm and pinion, measured at the periphery of the pinion, extends substantially over a whole number of teeth.

7. A machine as claimed in claim 6, in which on the low-pressure side, the worm is truncated along a conical surface coaxial with the worm.

8. A machine as claimed in claim 6, in which on the low-pressure side of the worm the end of a single thread on the worm is truncated.

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