(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 4 431 741 A1

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
18.09.2024 Bulletin 2024/38

(21) Application number: **23425010.8**

(22) Date of filing: **14.03.2023**

(51) International Patent Classification (IPC):
F04C 2/08 *(2006.01)* ***F04C 2/16*** *(2006.01)*
F04C 2/18 *(2006.01)*

(52) Cooperative Patent Classification (CPC):
F04C 2/084; F04C 2/18; F04C 2250/20

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC ME MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA
Designated Validation States:
KH MA MD TN

(71) Applicant: **Settima Meccanica S.R.L.**
20123 Milano (IT)

(72) Inventor: **ROSSI, Manuele**
29028 Ponte dell'Olio (PC) (IT)

(74) Representative: **Botti & Ferrari S.p.A.**
Via Cappellini, 11
20124 Milano (IT)

(54) ENHANCED BI-HELICAL TOOTHED WHEEL WITH VARIABLE HELIX ANGLE AND NON-ENCAPSULATING TOOTH PROFILE FOR HYDRAULIC GEAR APPARATUSES

(57) Bi-helical toothed wheel (1) with non-encapsulating profile (4) for hydraulic gear apparatuses (2), of the type bound to a support shaft (5) to form a driving or driven wheel of said hydraulic apparatus and comprising a plurality of teeth (6) extending with variable helix angle with a composite function in the longitudinal or axial direction of the tooth (6), wherein the teeth profile (4) keeps a shape continuity in each cross section thereof, wherein each tooth (6) in the longitudinal direction is divided into five zones: initial (A), proximal intermediate (B), central (C), distal intermediate (D) and terminal (E), where said initial (A), central (C) and terminal (E) zones have a variable helix angle (ψ) and said proximal intermediate (B) and distal intermediate (D) zones have a constant helix angle (ψ).

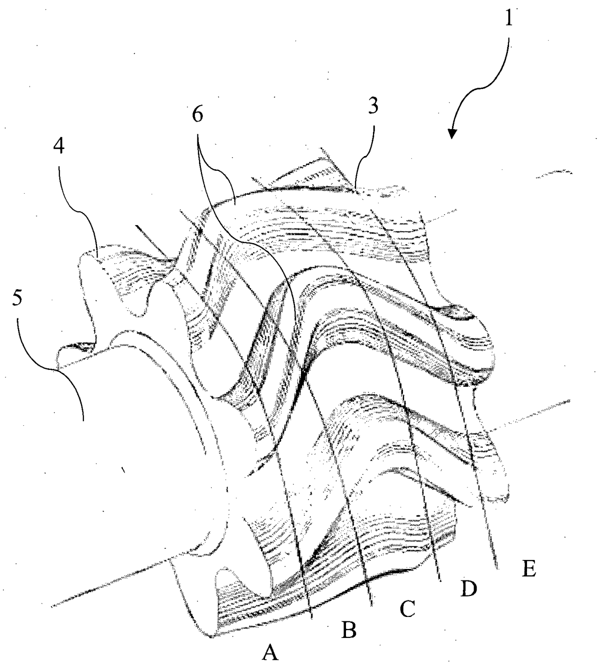


Fig. 5

EP 4 431 741 A1

Description

Field of application

[0001] The present invention relates to a bi-helical toothed wheel with non-encapsulating profile, adapted to mesh in a hydraulic gear apparatus.

[0002] More particularly, the invention relates to a toothed wheel intended to mesh without encapsulation with a toothed wheel of the same type in a hydraulic gear apparatus.

[0003] Typical examples of hydraulic gear apparatuses, where the toothed wheels of the present invention find an optimal application, and to which specific reference will be made below in the present description, are rotary volumetric gear pumps. However, the toothed wheels of the present invention can be also similarly applied to hydraulic gear motors and/or all hydraulic apparatuses operating through a pair of gears, which are therefore intended to be included within the scope of the present invention.

Prior art

[0004] As is well known in this technical field, rotary volumetric gear pumps generally comprise two toothed wheels, in most cases of the straight-teeth type, one of them, called driving wheel, being connected to a control shaft and driving in rotation the other wheel, called driven wheel.

[0005] In the toothed wheels with straight teeth each pair of teeth simultaneously meshes over the whole axial width of the toothed portion and similarly unmeshes. This type of coupling mechanically causes vibrations and noises due to the load variation on the tooth and to access and return shocks.

[0006] Another disadvantage which is particularly felt in the above gear pumps of the traditional type is due to the fact that the pumped fluid is encapsulated, i.e. entrapped and compressed, or anyway subjected to volume variations in the spaces enclosed between the teeth profiles in the meshing zone, thereby leading to detrimental and uncontrolled local stress peaks which cause a direct hydraulic operation noise.

[0007] A known technical solution to obviate the direct mechanical operation noise consists in adopting toothed wheels with helical teeth. The teeth of these helical wheels, instead of being parallel to the wheel axis, are oriented according to cylindrical helices.

[0008] In the toothed wheels with helical teeth, due to the slope, each pair of teeth gradually meshes and similarly unmeshes, leading to a more noiseless and regular transmission.

[0009] Although advantageous in many respects and substantially meeting the purpose of reducing operation noise, these toothed wheels introduce other issues due to the peculiar structure thereof. Indeed, due to the teeth slope, the transmitted force divides into a tangential component, needed for the transmission of the twisting moment, and an axial component, which tends instead to displace the wheel.

[0010] In order to obviate this problem either thrust bearings or two opposite helices with complementary angles are used, suppressing the induced axial thrust.

[0011] The invention aims at obviating the use of thrust bearings or any other device for compensating the axial forces generated inside and it joins instead the trend of opposite helices, intending moreover to suppress the hydraulic noise due to fluid encapsulation.

[0012] Here-attached figure 1 shows a known example of a toothed wheel with opposite helices, normally referred to as herringbone gears, where the two opposite helices connect in a cusp point.

[0013] The herringbone gears of figure 1 are used as rotors for hydraulic pumps in low-speed and high-power applications.

[0014] Despite the fact that this type of toothing has been used for several years, the accuracy of the teeth profile and the hardness thereof are limited by the construction difficulties due to machining at the cusp.

[0015] Indeed, the machines for manufacturing this type of toothed wheels are slotting machines in which the two opposite helices are machined simultaneously with a reciprocating motion of blades which interfere with each other at the cusp.

[0016] A limitation to this process is the impossibility of manufacturing wheels with a large size and high hardness, since the machining near the cusp point is very delicate and complicated to the extent that usually gears with materials having a hardness which is higher than 35 Rockwell C cannot be obtained.

[0017] In order to improve hardness properties, these gears can be treated for example with thermal nitridation treatments after the tooth machining. However, the tooth distorsion following the thermal treatment forces the designer to use wider tolerances to avoid damages to the tooth surface obtaining lower efficiencies.

[0018] An alternative solution is shown in figure 2 in which an interspace is provided between the two helices, allowing a variety of machine tools to be used for gear manufacture and allowing optimal accuracies to be achieved even on high hardnesses, which are for example higher than 58-60 Rockwell C. However, these gears cannot be used for pumping applications.

[0019] Although other alternative solutions have been suggested for manufacturing pumps with bi-helical gears having a high hardness, disclosed for example in US 2004/0031152 A1 and US 7,040,870 B1, they specifically relate to pumping molten plastic material - that is they work at low speeds - and use accordingly involute tooth profiles, without considering the noise due to fluid encapsulation which is negligible in these applications.

[0020] In order to at least partially solve the above-lamented problems, providing bi-helical gears with non-encapsulating profile which can be easily machined even reaching high hardnesses, the above European application no. 17 181 600.2 in the name of the Applicant has suggested a profile of these gears in which the helix angle is variable along the development thereof. In a preferred variant, depicted in attached figure 3, the development divides into three zones: in a first zone of the helix development the helix angle is constant, in a second zone the angle is variable and in a third zone, the angle is constant again.

[0021] Although substantially meeting the field requirements, the pumps manufactured with the toothed wheels identified in the above-recalled application have nevertheless some critical issues which are unsolved to date.

[0022] In particular, in this new type of known pumps the sudden break of a fragile portion or edge occurs at the ends of the gear tooth in certain operating conditions, with typical chipping depicted in figure 28b. This phenomenon, observed by the Applicant in the machines manufactured according to the teaching of the above European application no. 17 181 600.2, can limit the use of the device in some applications.

[0023] Moreover, the reduction of the vibrations still appears not to be optimal due to a discontinuity in the function of the fluid volume displaced over time due to the profile geometry.

[0024] The technical problem underlying the present invention is hence to devise a new type of bi-helical toothed wheel for hydraulic gear apparatuses, which has such structural and functional features as to simultaneously allow the mechanical and hydraulic operation noise to be suppressed and the generation of axial thrusts requiring any force compensation to be avoided, while proving to be easy to manufacture by means of numerically controlled machines of the substantially conventional type and suppressing the presence of a fragile portion or edge at the ends of the rotors themselves which causes limitations to the pump use.

[0025] A further object of the present invention is to manufacture a gear for volumetric pumps and other types of hydraulic apparatuses which is completely devoid of encapsulation.

Summary of the invention

[0026] The solution idea underlying the present invention is to obtain a bi-helical toothed wheel for hydraulic gear apparatuses, of the type bound to a support shaft to form a driving or driven wheel of said hydraulic apparatus and comprising a plurality of teeth extending with variable helix angle with a continuous function in the longitudinal or axial direction of the tooth, wherein each tooth has a central zone with variable helix angle wherein a helix transition from right-handed to left-handed occurs; wherein the teeth profile keeps a shape continuity in each cross section thereof, wherein each tooth comprises at least two initial and terminal zones with variable helix angle at the two opposite lateral ends of the toothed wheel, wherein in these initial and terminal zones the helix angle decreases when approaching the lateral end of the toothed wheel.

[0027] Continuous function means a function which is devoid of discontinuity, this definition also encompassing composite functions, composed for example of a plurality of arcs of circumference possibly connected by straight lines.

[0028] Preferably, said continuous function has neither angular points nor cusps; in particular, preferably the central zone has a curvilinear transition from right-handed helix to left-handed helix, with a transition point with null helix angle.

[0029] In a preferred variant of the invention, each tooth further comprises a proximal intermediate zone and a distal intermediate zone with constant helix angle which connect the initial and terminal zones to the central zone.

[0030] According to a possible alternative, the initial and terminal zones and the central zone with variable helix angle are directly connected by inflection points, so that the helix angle is variable along the whole tooth profile. In this case, the helix angle can develop for example according to a cosinusoid.

[0031] Preferably, the helix angle is null at the opposite lateral ends of the toothed wheel, that is the profile of each single tooth joins at a right angle the opposite lateral faces of the toothed wheel.

[0032] Due to the above-suggested expedient, the axial component of the force exchanged by said wheel meshed during use with another identical wheel is null at the lateral ends, which minimizes the risks of edge chipping even in unfavourable conditions of use.

[0033] Preferably, the teeth profile is mirrored with respect to a centre plane passing through the transition point between right-handed and left-handed helix.

[0034] Preferably, the teeth profile is parametrized and sized so that the helix contact ratio parameter is comprised between 0.6 and 1, preferably between 0.6 and 0.8, more preferably equal to 0.65.

[0035] Preferably, the teeth profile of the toothed wheel is a non-encapsulating profile.

[0036] This non-encapsulating profile can be defined by two arcs of circumference or elliptical crest and bottom portions connected by an involute profile comprised between two part-off diameters: a lower truncation diameter where the transition between bottom portion and involute profile occurs and an upper truncation diameter where the transition between involute profile and crest portion occurs.

[0037] Preferably, the lower involute truncation diameter is selected equal to: Oitr = Op - Øp * p1 , with Op pitch diameter and the parameter p1 comprised between 9.7% and 9.9%, preferably equal to 9.8%.

[0038] Preferably, the upper involute truncation diameter is selected equal to: Oetr = Op + Op * p2, with Op pitch diameter and the parameter p2 comprised between 12.1% and 12.3%, preferably equal to 12.2%.

[0039] Preferably, the top of each tooth has a cutting edge, defined by a limited thickness projecting with respect to the profile, mainly intended for running in the pump body.

[0040] The above-identified technical problem is also solved by a hydraulic gear apparatus comprising a pair of toothed wheels according to the above suggestions.

[0041] In particular, said apparatus can be a volumetric pump or a hydraulic gear motor.

[0042] The technical problem is also solved by a method for manufacturing a bi-helical toothed wheel with non-encapsulating profile, for hydraulic gear apparatuses, by means of an automatic numerically controlled machine powered by an appropriate software, comprising the following steps:

determining the equations of the rotations of all profile sections in the axial direction calculated on the pitch diameter obtaining a series of coordinates representing the helix path;

making the rotor front profile slide to manufacture the solid model according to the helix path with a 3D software;

transferring the solid model to CAD-CAM;

finishing the inter-tooth space on the working station.

[0043] Optionally, the method can comprise a thermal hardening treatment.

[0044] Further features and advantages of the present invention will be more apparent from the description of an embodiment thereof, made herebelow with reference to the attached drawings, given by way of non-limiting example.

Brief description of the drawings

[0045]

Figure 1 shows a perspective and schematic view of a herringbone toothed wheel manufactured according to the prior art;

Figure 2 shows a perspective and schematic view of a bi-helical toothed wheel with helices spaced apart from each other, manufactured according to the prior art;

Figure 3 shows a perspective view of a bi-helical toothed wheel with variable angle helices, manufactured according to the prior art;

Figure 4 shows a plan view of a pair of bi-helical toothed wheels according to the present invention coupled to each other in a hydraulic gear apparatus, for example a volumetric pump;

Figure 5 shows a perspective view of a bi-helical toothed wheel with teeth developed along variable angle helices, manufactured according to the present invention;

Figure 6 shows a lateral schematic view of a section of the wheel according to the invention;

Figure 7 shows a geometric construction of the involute segment of the gear tooth profile;

Figure 8a shows an enlarged detail F of figure 6;

Figure 8b shows a lateral schematic view of two toothed wheels according to the present invention meshed with each other, with identification of the pressure straight line;

Figure 8c shows an enlarged detail G of figure 8a;

Figure 8d shows a geometric schematization of the contact angle of the teeth on the involute profile in the gear composed of toothed wheels according to the present invention;

Figure 9a shows a front schematic view of the section of two adjoining teeth during meshing;

Figure 9b shows a lateral schematic view of the gear in figure 9a;

Figures 10a and 10b show respective geometric schematizations related to a general helix;

Figures 11a and 11b show the development of a turn of the helix for a known herringbone gear;

Figures 12a - 12c show respective schematic views of the geometry of the helix development in the toothed wheel according to the invention;

Figures 13 and 14 show geometric schematizations of the helix development in the toothed wheel according to the present invention;

Figures 15a and 15b show respective geometric diagrams to describe the kinematic analysis of the contact point between the helices of the gears on the pitch diameter as the rotation speed thereof varies;

Figures 16a to 19c show respective geometric schematizations for the calculation of the equations of the motion of the contact point between the two helices in the axial direction in three different zones;

Figures 20a and 20b show schematic lateral views of the fluid trapped in the chambers created by the teeth of the gears according to the present invention;

Figures 21a to 24c show respective geometric schematizations for the analysis of the fluid volumes trapped in the chambers created by the teeth of the gears;

Figure 25a shows a lateral schematic view of two toothed wheels according to the present invention meshed with each other, with identification of the pressure straight line;

Figure 25b shows an enlarged H of figure 25a;

Figure 26 shows a geometric diagram for the analysis of the forces acting on the teeth of the gears;

Figure 27a shows the orientation of the forces acting along the development of the teeth of a toothed wheel in a first gear according to the prior art;

Figure 27b shows the orientation of the forces acting along the development of the teeth of a toothed wheel in a second gear according to the prior art;

Figure 27c shows the orientation of the forces acting along the development of the teeth of a toothed wheel according to the present invention;

Figure 28a shows a perspective schematic view of a gear according to the prior art with identification of the orientation of the exchanged forces;

Figure 28b shows typical chipping in the toothed wheel profile according to the prior art of figure 28a;

Figure 29 shows a perspective schematic view of a gear according to the present invention with identification of the orientation of the exchanged forces;

Figure 30a shows the development of the tooth in a toothed wheel according to the prior art identifying the tooth angle with respect to a plane transverse to the lateral end thereof;

Figure 30b shows the development of the tooth in a toothed wheel according to the present invention identifying the tooth angle with respect to a plane transverse to the lateral end thereof.

Detailed description

[0046] With reference to figure 5, a toothed wheel manufactured in accordance with the present invention and of the type with a bi-helical profile is globally and schematically indicated with 1.

[0047] The toothed wheel 1 is particularly, but not exclusively, intended for hydraulic gear apparatuses and the following description will refer to this specific field of application to simplify the exposition thereof.

[0048] For a better understanding of all the aspects of the present invention "cylindrical helix " is defined as a curve described by an animated point of continuous circular motion and, simultaneously, of various motion having a direction which is perpendicular to the rotation plane.

[0049] Moreover, "helix pitch" is defined as the distance travelled by the helix generator point in a complete turn in the axial direction.

[0050] Moreover, "non-encapsulating profile" is defined as a rotor tooth profile which lets the pumped fluid flow between the meshed teeth of the rotors, without trapping it or compressing it or subjecting it to volume variations when sliding inside the pump. In order to obtain this effect, the non-encapsulating profile is able to perfectly match a corresponding profile, without defining cavities interposed between the two meshed profiles. A non-encapsulating profile can be made with heads and feet manufactured as arcs of circle and connected to each other by flanks with circle involute.

[0051] The invention aims at manufacturing a bi-helical toothed wheel which can be used with a wheel of the same type in a gear for a volumetric pump which uses rotors with opposite helix. Advantageously, according to the invention, the wheel 1 has a non-encapsulating profile and such a helix shape as to suppress the angular point in the centre of the traditional herringbone gears manufactured according to the prior art and to suppress at the ends of the rotors a fragile edge which is present in the rotors manufactured according to the prior art.

[0052] The issues related to the machining of rotors having such a profile by means of machine tools are thereby suppressed at source.

[0053] Figure 4 shows the view from above of the toothed wheel 1 which, coupled to a corresponding toothed wheel 1', defines a gear 2 of the bi-helical non-encapsulating type of a hydraulic apparatus, for example a volumetric pump.

[0054] The toothed wheel 1 is conventionally bound or fitted onto a support shaft 5 to form a driving or driven wheel depending on the role it was assigned in the hydraulic apparatus.

[0055] In the exemplary embodiment described herein by way of non-limiting example, the toothed wheel 1 has a front profile 4 with seven teeth, but nothing prevents to use a different plurality of teeth.

[0056] Advantageously, according to the invention, the bi-helical development 3 of the toothed wheel 1 varies with a continuous function and an arcuate pattern along the axial direction of the tooth, while keeping the shape continuity of the cross section thereof, which coincides with the front profile 4.

[0057] In the present invention, continuous function means a function which is devoid of discontinuity points. This function can be a unique function - for example a cosinusoid - or a composite function - for example three arcs of circle, possibly connected by rectilinear transitions.

[0058] In other words, the gear 2 has neither any cusp, but nor any acute angle in the central zone thereof. Each corresponding tooth 6 is continuous and devoid of undercuts. Moreover, each corresponding tooth 6 has no cusp in the central zone and this geometry is defined both in the central zone of the gear and at the ends thereof, where the external sections end with a helix angle which is equal to 0°.

[0059] This peculiar helix development, which will be further detailed below, allows a pair of rotors to be obtained, in which the pitch and helix angle vary with mathematical regularity and, in particular, in the preferred profile illustrated in the enclosed figures, a transmission continuity with a contact ratio which is equal to 0.65 is ensured.

[0060] In essence, this means that: before two teeth 6 are abandoned other two teeth 6 simultaneously begin to mesh. The contact is continuous and reversible and, depending on the right-handed or left-handed rotation and on the arrangement of the helices, it moves from the centre of the rotor outwards or vice versa.

[0061] Moreover, it should be noted that the teeth profiles are conjugated over the whole length of the rotor, that is the tangents to the profiles in the contact point coincide and the common normal passes through the centre of instantaneous rotation.

[0062] Referring to figure 5, a rotor covered by the present invention is shown in detail, which comprises several segments in which the helix angle is variable or constant, in particular:

- segment A: with variable helix angle;
- segment B: with constant helix angle;
- segment C: with variable helix angle;
- segment D: with constant helix angle;

- segment E: with variable helix angle.

[0063] In essence, the longitudinal development of the tooth of the rotor can be divided into five zones: initial, proximal intermediate, central, distal intermediate and terminal, where the segment A corresponds to the initial zone, the segment B corresponds to the proximal intermediate zone, the segment C corresponds to the central zone, the segment D corresponds to the distal intermediate zone and the segment E corresponds to the final zone.

[0064] It is noted that the division into five segments or zones is not essential for implementing the present invention, being it possible for the initial and distal segments to be directly connected to the central segment by inflection points, without any transition with constant helix angle. In this case, the profile of the tooth crest developed on a flat surface can be a cosinusoid, whereas in the preferred embodiment the terminal and distal segments can follow an arc of circle - or other plane curve - and the intermediate segments are rectilinear.

[0065] The lengths of the various segments A, B, C, D and E of the rotor are adjusted according to mechanical considerations and vary as the rotor band varies following geometric rules which will be described in detail below.

[0066] The pattern of the profile 4 of the single tooth 6 of the gear 2 is now described.

[0067] Referring to figure 6, the profile 4 of the teeth 6 of the gear 2 is formed by a segment with circle involute 8 comprised between two truncation diameters Øitr, Øetr which connects two arcs of circumference 7, 9, at the tooth crest and bottom.

[0068] Referring to figure 7, the involute segment is characterized by the following formulas:

$$\begin{cases} \overline{OP}\cos\gamma = R(\cos\beta + \beta\,\mathrm{sen}\,\beta) \\ \overline{OP}\,\mathrm{sen}\,\gamma = R(\mathrm{sen}\,\beta - \beta\cos\beta) \end{cases}$$

$$\overline{OP}^2 = R^2(1+\beta^2)$$

$$\tan\gamma = \frac{\mathrm{sen}\beta + \beta\cos\beta}{\cos\beta - \beta\,\mathrm{sen}\beta}$$

$$\overline{OP} = R\sqrt{1+\beta^2} = R(\cos\beta + \beta\,\mathrm{sen}\,\beta)\cos\gamma + R(\mathrm{sen}\,\beta - \beta\cos\beta)\,\mathrm{sen}\,\gamma$$

$$\begin{cases} \mathrm{R}\left(\sqrt{1+\beta^2}\right)\cos\gamma = \mathrm{R\,x} \\ \mathrm{R}\left(\sqrt{1+\beta^2}\right)\sin\gamma = \mathrm{R\,y} \end{cases}$$

[0069] The involute parameters used in the preferred embodiment of the invention are identified below, with reference to the aforementioned figure 6.

[0070] The lower involute truncation diameter is selected equal to:

$$Øitr = Øp - Øp * p1$$

with p1 comprised between 9.7% and 9.9%, preferably equal to 9.8%.

[0071] The upper involute truncation diameter is selected equal to:

$$Øetr = Øp + Øp * p2$$

with p2 comprised between 12.1% and 12.3%, preferably equal to 12.2%.

[0072] In both cases, Op is the pitch diameter of the wheels 1.

[0073] With regard to the crest and bottom diameters, referring to figures 8a, 8b and 8c, the following relations apply:

$$Øext = Øp + Øp * p3$$

with p3 comprised between 19.5% and 20.5%, preferably equal to 20%;

$$Øint = Øp - Øp * p4$$

with p4 comprised between 19.5% and 20.5%, preferably equal to 20%.

[0074] Referring in particular to figure 8a, the top of each tooth 6 has a cutting edge 10, defined by a limited thickness projecting with respect to the profile 4, in order to easily run in the pump body and have narrower size tolerances on the external diameter of the gears.

[0075] In figure 8b the meshing between the two toothed wheels is illustrated, identifying with Op the pitch diameters, with Cd the deferent circumferences and with Rp the pressure straight line. In the detail G of figure 8c the pressure angle φ is depicted, that is the angle between the pressure straight line and the tangent to the pitch diameters Op.

[0076] Preferably, the parameters of the profiles 4 according to the present invention are such that the pressure angle φ is comprised between 28° and 32°, preferably equal to 30°.

[0077] From the above-listed design parameters, referring in particular to figure 8d, it results that the contact angle α for two teeth engaging on the involute segment is equal to about 29°.

[0078] With regard to the helix contact ratio, the profile 4 of the rotors 1 is, as already said, such as to avoid the fluid encapsulation between the groove bottom and crest of two meshing teeth. However, it results therefrom that the circumferential contact ratio Rc, defined as the ratio between the involute contact angle and the circumferential pitch subtended angle, is lower than 1; in this specific case equal to about 0.5.

[0079] This substantially means that the teeth which are in contact and transmit the motion detach before the following ones mesh, involving the need to manufacture the gear in the helical shape thereof.

[0080] Referring now in particular to figures 9a and 9b, two adjoining teeth 6 in perpendicular section to the axis of rotation of the rotors 5 are indicated with I and II. The same teeth 6 in perpendicular section to the axis of rotation, at the distance L, are then indicated with I' and II'.

[0081] In order that the length of the contact line between the teeth 6 keeps constant along the axial direction during the mutual rotation of the rotors it is necessary that I and II' are at a distance Lf respectively rotated by: $2.\pi/(n° \text{ teeth})$

[0082] In this case it occurs that the helix contact ratio Re is equal to 1.

[0083] This is an ideal operating condition, since the hertzian contact stresses always discharge on the same surface.

[0084] However, based on the design specifications of the pump - which in the preferred case is designed to work at pressures of 250 bar and at speeds of 3600 rpm - and on the calculations performed on the strength of the gears, it is necessary not to have helix angles which are higher than 45° and hence a helix contact ratio Re was established, which is equal to 0.65 and anyway not lower than 0.6, in particular comprised between 0.6 and 0.8.

[0085] The calculation of the coordinates of the helix development in the three-dimensional space is now described, which allows to determine the inter-tooth space of the rotor by means of a 3D software.

[0086] With reference to figures 10a and 10b, some geometric definitions are recalled beforehand.

[0087] A helix is a curve in the three-dimensional space, depicted by a constant angle line wound around a cylinder.

[0088] The development of a turn of the helix is a straight-line segment corresponding to the hypotenuse of the right triangle having the pitch P and the length of the helix circumference $\pi\cdot d$ as catheti. The slope is determined by the angle α comprised between the hypotenuse and the cathetus corresponding to the helix circumference. Hence obtaining: $\tan(\alpha) = P/(\pi\cdot d)$.

[0089] Referring to figures 11a and 11b, it is known that the development of a turn of the helix for a traditional version of a herringbone gear is a straight-line segment having a slope α. Considering a half gear, a helix contact ratio which is equal to 1 is obtained.

[0090] A similar construction is suggested, in figures 12a and 12b, for the helix development suggested by the present invention. Obviously, in this case the hypotenuse is replaced by a curved line with a variable slope α. Figure 12c illustrates the pattern of this curve in space.

[0091] For the sake of simplicity only half a gear with helix contact ratio which is equal to 1 is considered hereafter, this in order to simplify formulas and explanations. The right triangle in which the helix development is depicted is used as a base for the following description.

[0092] In the following schematizations, in the depiction of the helix developing triangle, in order to obtain a contact ratio 1, the respective variables p and $(\pi\cdot dp)$ for the horizontal and vertical catheti are replaced by new variables P/n° teeth and $\pi\cdot dp$/n° teeth, with P helix pitch and dp pitch diameter used for the calculation of the average helix angle.

[0093] With these new variables, it goes from the graph of figures 11a and 11b to the graph of figures 12a-12c.

[0094] Given the band length, which determines the pump displacement, it occurs that the helix angle deriving from the prefixed geometric construction does not exceed the set design parameter.

[0095] In the construction according to the invention, all the sections which are perpendicular to the axis of rotation are equal to each other.

[0096] The choice of using, for manufacturing the helix, two identical arcs of circle for the development thereof in the rotor central and final part is mainly linked to making all the mathematical relationships deriving therefrom simpler; however, any other curve can also be used; moreover, also the rectilinear segment can be replaced by a curve which connects the two ends. In this specific case, the helix is formed by a straight line tangent to two arcs of circumference having the same radius comprised in an angle which is equal to the design one.

[0097] A variant providing the replacement of the rectilinear segment by a curved segment can be defined for example by a single cosinusoid, which will define in this case both the initial and terminal zones and the central zone, keeping the helix angle always variable along the whole tooth extension.

[0098] Referring to figure 13, the helix angle is obtained with the following relationship:

$$\frac{F}{4}\,sen\alpha = \left[\left(\frac{2\pi}{n°teeth}\,R_e\right)\,r_p\right]\frac{1}{2}$$

[0099] In this specific case, R_e=1 ; n° teeth = 7 are used

$$\alpha = arcsen\left(\frac{r_p\,4\,\pi}{7\,F}\right)$$

[0100] Referring to figures 15a and 15b, the kinematic analysis of the contact point between the helices of the gears on the pitch diameter upon variation of the rotation speed thereof is now described.

[0101] It is assumed that the contact is punctual and at the pitch diameter.

[0102] The helix development in the plane is depicted by dividing the graph into three distinct zones respectively:

a first zone z1 with variable helix angle;

a second zone z2 with constant helix angle;

a third zone z3 with variable helix angle.

[0103] Replacing on the x-axis and y-axis:

X → arc of circumference travelled by the generator point of the helix [C]

Y → space travelled in the axial direction by the generator point of the helix [F]

[0104] When rotating, the two gears keep in contact in a point up to travel the distance F which is equal to the band length.

[0105] The equations of the motion of the contact point between the two helices in the axial direction are written as a composite function of three clearly distinct: zones z1, z2 and z3.

[0106] For the first zone z1, referring to figures 16a and 16b, the following relationships apply:

$$\frac{C}{2} = \left[\left(\frac{2\pi}{n°teeth}\,R_e\right)\,r_p\right]\frac{1}{2} \quad \text{with } Re{=}1 \rightarrow \frac{C}{2} = \left[\left(\frac{2\pi}{7}\right)\,r_p\right]\frac{1}{2} \rightarrow \frac{C}{2} = \left[\left(\frac{\pi}{7}\right)\,r_p\right]$$

[0107] Indicating with $k = \left(\frac{\pi}{7}\right)$

$$\frac{C}{2} = \left[k\,r_p\right]$$

$$C1 = \theta 1\, r_p$$

[0108] It can be written:

$$\begin{cases} 1.\ F = \left[k\, r_p\, sen\varphi\right] \\ 2.\ C = \left[k\, r_p - k\, r_p\, cos\varphi\right] \end{cases} \quad \text{with} \rightarrow 0<\varphi<\alpha$$

$$C = \theta\, r_p$$

by replacing in 2:

$$\theta\, r_p = \left[k\, r_p - k\, r_p\, cos\varphi\right]$$

$$\theta\, r_p = k\, r_p\, (1 - cos\varphi)$$

$$3.1 \qquad \theta = k\,(1 - \cos\varphi) \rightarrow \theta = k - k\, cos\varphi$$

$$cos\varphi = \frac{k - \theta}{k}$$

$$\varphi = arccos\, \frac{k - \theta}{k}$$

by replacing in 1

$$3.2 \qquad F1 = k\, r_p\, sen\ [\arccos\left(\frac{k-\theta}{k}\right)]$$

[0109] Function which is valid for $0<\varphi<\alpha$ with $\theta max \rightarrow \varphi=\alpha$

[0110] Recalling 3.1:

$$\theta = k - k\, cos\varphi \text{ replacing } \varphi=\alpha v$$

$$\theta max = k - k\, cos\alpha$$

$$3.3 \qquad \theta max = k\,(1 - cos\alpha)$$

the time law of the motion of the contact point between the two meshing helices is now obtained by replacing in 3.2:

$$\theta = \omega t$$

$$3.4 \qquad F1(t) = k\, r_p\, sen\ \left[\arccos\left(\frac{k-\omega t}{k}\right)\right]$$

[0111] Another important relationship which can be obtained from 3.3 by replacing $\theta max = \omega * tmax$ with

$$\omega\, tmax = k\,(1 - cos\alpha)$$

$$3.5 \qquad tmax = \frac{k\,(1-cos\alpha)}{\omega}$$

[0112] Referring to figure 17, by deriving 3.4, 3.6 is obtained, which is the speed at which the point translates parallel to the rotor axis

$$3.6 \qquad V1(t) = \frac{\partial}{\partial t}\left\{k\, r_p\, sen\ \left[\arccos\left(\frac{k-\omega t}{k}\right)\right]\right\} = \frac{r_p\,\omega\,(k-\omega\,t)}{k\sqrt{\frac{\omega\,t\,(2k-\omega\,t)}{k^2}}}$$

[0113] By deriving 3.6, 3.7 is obtained, which is the equation of acceleration undergone by the fluid along the axial travel thereof towards the centre or the ends of the gear depending on the rotation direction.

$$3.7 \qquad a1(t) = \frac{\partial}{\partial t}\left\{\frac{r_p\,\omega\,(k-\omega\,t)}{k\sqrt{\frac{\omega\,t\,(2k-\omega\,t)}{k^2}}}\right\} = -\frac{r_p\,\omega^2}{k\sqrt{\frac{\omega\,t\,(2k-\omega\,t)}{k^2}}} - \frac{r_p\,\omega\,(k-\omega\,t)\left(\frac{\omega\,(2k-\omega\,t)}{k^2} - \frac{\omega^2 t}{k^2}\right)}{2k\sqrt{\left(\frac{\omega\,t\,(2k-\omega\,t)}{k^2}\right)^3}}$$

[0114] With regard to zone z2, referring to figures 18a- 18c, it can be written:

$$\begin{cases} 1.\ \tan\alpha = \frac{C}{F} \\ 2.\ C = r_p\,\theta \end{cases}$$

[0115] From 1

$$F = \frac{C}{\tan\alpha} \quad \text{by replacing 2}$$

$$F = \frac{r_p\,\theta}{\tan\alpha} \qquad \text{con } \theta = \omega t$$

$$3.7 \qquad F2(t) = \frac{r_p}{\tan\alpha}\,\omega t$$

$$\theta_2 = \theta_t - 2\,\theta_{max} \quad \text{con } \theta_t = \frac{2\pi}{7} = 2\,k$$

$$\theta_2 = 2k - 2\,k\,(1 - cos\alpha)$$

$$\theta_2 = 2k\,[1 - (1 - cos\alpha)]$$

$$3.8 \qquad \theta_2 = 2k\,cos\alpha$$

$$3.9 \qquad F2 = \frac{r_p}{\tan\alpha}\,2k\,cos\alpha$$

[0116] From 3.8:

$$\omega\,tmax = 2\,k\,cos\alpha$$

$$4.0 \qquad tmax = \frac{2\,k\,cos\alpha}{\omega}$$

[0117] By deriving 3.7:

$$4.1 \qquad V2(t) = \frac{\partial}{\partial t}\left[\frac{r_p}{\tan\alpha}\,\omega t\right] = \frac{r_p}{\tan\alpha}\,\omega$$

[0118] With regard to zone 3, referring to figures 19a- 19c, the same relationships as zone 1 apply, in particular it can be written:

$$\frac{c}{2} = \left[\left(\frac{2\pi}{\text{n° teeth}}\,R_e\right)r_p\right]\frac{1}{2} \quad \text{with} \quad Re{=}1 \rightarrow \frac{c}{2} = \left[\left(\frac{2\pi}{7}\right)r_p\right]\frac{1}{2} \rightarrow \frac{c}{2} = \left[\left(\frac{\pi}{7}\right)r_p\right];$$

$$k = \left(\frac{\pi}{7}\right);$$

$$\frac{c}{2} = [k\,r_p];$$

$$C3 = \theta 3\,r_p;$$

$$\begin{cases} 1.\ \ F = k\, r_p\, sen\alpha - k\, r_p\, sen\varphi \\ \\ 2.\ \ C = k\, r_p\, cos\varphi - k\, r_p\, cos\alpha \end{cases} \quad \text{with} \rightarrow 0<\varphi<\alpha$$

[0119] From 2.

$$\theta\, r_p = k\, r_p\, cos\varphi - k\, r_p\, cos\alpha \quad \rightarrow \quad \theta\, r_p = k\ \ cos\varphi - k\ \ cos\alpha \quad \rightarrow cos\varphi = \frac{\theta + k\ \ cos\alpha}{k} \rightarrow$$

$$4.2 \qquad \varphi = arcos\left(\frac{\theta + k\ \ cos\alpha}{k}\right)$$

[0120] By replacing in 1.

$$F = k\, r_p\, sen\alpha - k\, r_p\, sen\left[arcos\left(\frac{\theta + k\ \ cos\alpha}{k}\right)\right]$$

with with $\theta = \omega t$

$$4.3 \qquad F3(t) = k\, r_p\, sen\alpha - k\, r_p\, sen\left[arcos\left(\frac{\omega t + k\ \ cos\alpha}{k}\right)\right]$$

[0121] By deriving 4.3 4.4 is obtained

$$4.4 \qquad V3(t) = \frac{\partial}{\partial t}\left\{k\, r_p\, sen\alpha - k\, r_p\, sen\left[arcos\left(\frac{\omega t + k\ \ cos\alpha}{k}\right)\right]\right\} = \frac{r_p\, \omega\, (k\ \ cos\alpha + \omega\, t)}{k\sqrt{1 - \frac{(\omega\, t + k\ \ cos\alpha)^2}{k^2}}}$$

[0122] Completing with the equation of acceleration 4.5:

$$4.5 \qquad 3(t) = \frac{\partial}{\partial t}\left\{\frac{r_p\, \omega\, (k\ \ cos\alpha + \omega\, t)}{k\sqrt{1 - \frac{(\omega\, t + k\ \ cos\alpha)^2}{k^2}}}\right\} = \frac{r_p\, \omega}{k\sqrt{1 - \frac{(\omega\, t + k\ \ cos\alpha)^2}{k^2}}} + \frac{r_p\, \omega^2\, (k\ \ cos\alpha + \omega\ t)^2}{k^3\sqrt{1 - \left(\frac{(k\ \ cos\alpha + \omega\ t)^2}{k^2}\right)^3}}$$

[0123] The fluid dynamics of the pump is now described to prove the efficiency of the solution described for attenuating vibrations.

[0124] The above geometric considerations point out how a development of the modified helix angle allows vibrations on the fluid to be reduced with respect to the prior art.

[0125] The fluid volumes trapped in the chambers created by the teeth of the gears are brought from the suction zone to the delivery one. Once arrived in the highpressure zone, these volumes are ejected, being subjected to the compression they undergo because of the interpenetration of the solid tooth with the fluid itself. Being the teeth with helix, what is occurring is some sort of "helical extrusion" of the fluid.

[0126] Referring to figure 20, in order to explain the phenomenon, a section can be considered, which moves in the axial direction before being radially ejected from the pump. The section S moves in the axial direction with the same speed as the contact point of the helices of the two gears on the pitch diameter. Always assuming that the contact is punctual and only on the pitch diameter, (theoretical assumptions to explain the phenomenon), consideration is that the contact point on the helices of the gears continuously moves along the axial direction over an indefinite time: this occurs since at the end of the contact on the helix of a pair of meshed teeth the following one immediately conjugates.

[0127] Referring to figures 21a-c which show in a graph the functions of the fluid section displacement in the axial direction with the related speed and to figure 22, how the described solution improves the management of the vibrations induced on the pumped fluid is shown.

[0128] Considering a tank in which the liquid is pumped through a constant-section pipe. The pump flow rate formula can be written:

[0129] With $\dot{Q} = \frac{m^3}{s}$ and $\rho = \frac{kg}{m^3}$ the maximum flow rate $Q = \frac{kg}{s}$ is obtained

[0130] From the above-obtained relationships, the speed of the section S being considered is not constant and thus the flow rate is variable over time.

[0131] In order to understand the phenomenon linked to the vibrations induced on the fluid by the pump, consideration can be to have an infinitesimal mass of liquid m1 in the tank shown in figure 22, hit by a mass m0 exiting the pump, the mass m1 will change the speed thereof and will be able to do it in two ways: subjected to a small force for a long period of time or subjected to a great force for a short period of time.

[0132] Considering the second Newton's law in its simplest form:

$$F = m\,a$$

by multiplying both members by t, $Ft = mat \rightarrow Ft = mV \rightarrow I=p$ is obtained

with: I=impulse and p=momentum.

[0133] With reference to the two theorems of physics, i.e. the impulse theorem (which applies in the case of a single body) where the momentum variation is equal to the impulse of the force acting on a body and the momentum conservation principle where the total momentum of a system comprised of two particles only subjected to the mutual iteration thereof remains constant, it can be written:

$$p = p_0 + p_1 = m_0V_0 + m_1V_1$$

[0134] Obtaining in the following instant t':

$$p' = p'_0 + p'_1 = m_0V'_0 + m_1V'_1$$

[0135] Obtaining, according to the momentum conservation principle:

$$p' = p = cost$$

[0136] These considerations lead to the following observation: an interaction between two particles produces a momentum exchange, so that the momentum lost by one of the two particles is equal to the momentum gained by the other particle.

$$p = p_0 + p_1 = cost$$

$$p' = p'_0 + p'_1 = cost$$

[0137] By doing a member-to-member subtraction:

$$\Delta p_0 = -\Delta p_1$$

[0138] Knowing that:

$$\frac{dp}{dt} = \frac{d(mV)}{dt} = m\frac{dV}{dt} = ma = F$$

$$F_0 = -F_1$$

[0139] When two particles interact, the force acting on a particle is equal and opposite to the force acting on the other one (which coincides then with the third principle of dynamics).

[0140] From figures 21a-c and from the above formulas it is noted that the single particle is subjected to an abrupt acceleration by colliding with a following one having the same acceleration, with respect to the state of the art where instead the single particle is subjected to an abrupt acceleration by colliding with a following one having a low and constant speed.

[0141] Referring to figure 23, the contact line which is created between the rotating gears is now described, assuming the punctual contact on the pitch diameter, and considering that the contiguity occurs on a line which is created for the whole involute arc diagonally with respect to the gear band.

[0142] The calculation of the contact line length for a helical gear with constant helix is now described.

[0143] Indicating with:

Rp: length of the line described by the contact on the involute of the rotating gears on the pressure straight line (direction along which pressure forces are exchanged).

Rs: contact start radius on the involute

Rf: contact end radius on the involute

α_s: helix angle calculated for Rs

θ_{ev}: contact angle on the involute.

$$4.6 \quad F_p = \frac{R_s \vartheta_{ev}}{tg\alpha_s}$$

$$4.7 \quad LC = \sqrt{{R_p}^2 + {F_p}^2}$$

[0144] By analysing the contact line for a gear with constant helix with helix contact ratio which is equal to 1 it occurs that it always keeps the same total length, indeed, when a pair of teeth gradually unmeshes the line shortens by the same amount which meshes on the pair of following teeth, as shown in figures 24a-c.

[0145] The above description applies for a gear with helix contact ratio which is equal to 1, the case in which this value becomes lower than 1 is now analysed.

[0146] The contact line falls proportionally to the offset of the tooth following the meshed tooth at the opposed end of the gear band.

[0147] The value of Re selected for the present invention: Re=0.65 is considered.

[0148] This means that the gear front section rotates along the band by an angle which is equal to:

$$Re = 0.65 = \frac{13}{20}$$

[0149] The offset of the tooth which is adjacent to the opposed end of the band will be thus equal to:

$$Sf = \frac{20}{20} - \frac{13}{20} = \frac{7}{20}$$

$$\gamma = \frac{360°}{7}\,\frac{7}{20} = 18°$$

[0150] How the contact line length changes with respect to the case of Re=1 is now analysed

$$F_p = \frac{R_s(\vartheta_{ev} - \gamma)}{tg\alpha_s}$$

[0151] It must be $\vartheta_{ev} - \gamma > 0$ to have a meshing continuity and always a meshed tooth.

[0152] For the present invention ϑ_{ev} = 29° was selected, so it is $\vartheta_{ev} - \gamma$ = 11°

[0153] The above equations can be used to see an important benefit brought by the following invention with respect to the prior art.

[0154] By recalling the equation 3.4 of the segment with variable helix angle and the equation 3.7 for the segment with constant helix angle:

$$3.4 \qquad F1(t) = k\, r_p\, sen\ \left[\arccos\left(\frac{k-\omega t}{k}\right)\right]$$

$$3.7 \qquad F2\,(t) = \frac{r_p}{\tan\alpha}\,\omega t$$

[0155] By replacing the value of 11° in the equations 3.4 and 3.7 respectively it is obtained:

$$3.4 \qquad F1(t) = 5.85\,mm \rightarrow \text{Variable helix segment}$$

$$3.7 \qquad F2\,(t) = 3.05mm \rightarrow \text{Constant helix segment}$$

[0156] As it is noted from the obtained numerical values, in the segment with variable helix angle the contact line has a length which is higher than in the segment with constant helix angle.

[0157] Otherwise it can be written that for the segment with variable helix angle the relationship applies:

$$3.3 \qquad\qquad \theta max = k\,(1 - cos\alpha)$$

[0158] 3.4 and 3.7 are rewritten as:

$$3.4 \quad F1(t) = k\, r_p\, sen\ \left[\arccos\left(\frac{k-\theta max}{k}\right)\right]$$

$$F1 = k\, r_p\, sen\ \left[\arccos\left(\frac{k-k\,(1-cos\alpha)}{k}\right)\right] \quad \text{by simplifying} \quad \rightarrow F1 = k\, r_p\, sen\ \left[\arccos\left(\frac{k-k-k\,cos\alpha)}{k}\right)\right]$$

$$F1 = k\, r_p\, sen\ \left[\arccos\left(\frac{k-k-k\,cos\alpha)}{k}\right)\right] \rightarrow F1 = k\, r_p\, sen\ [\arccos(cos\alpha)]$$

$$F1 = k\, r_p\, sen\alpha$$

$$3.7 \quad F2 = \frac{r_p}{\tan\alpha}\,\theta max$$

$$F2 = \frac{r_p}{\tan\alpha}\,k\,(1 - cos\alpha)$$

thus :

$$F1 > F2 \rightarrow 4.8 \quad k\,r_p\,sen\alpha > \frac{r_p}{\tan\alpha}\,k\,(1 - cos\alpha)$$

[0159] Formula 4.8 proves that, for a gear with a certain helix angle α and with the selected geometric construction, the segment with variable helix angle has a longer contact line.

[0160] This allows the distribution of the efforts to be improved when the contact line takes the minimum length value thereof (since the helix contact ratio is lower than 1) with respect to the prior art.

[0161] The forces acting on the gear teeth are now analysed.

[0162] Starting with the formula to determine the power transferred to the fluid, from a hydraulic point of view it can be written:

A.

$$P = \frac{\Delta P\,\dot{Q}}{\eta_t}$$

from a mechanical point of view:

B.

$$P = C\,\omega$$

[0163] Note from A. the power supplied to the fluid, the twisting torque transmitted to the shaft can be obtained with B:

$$C_T = \frac{\Delta P\,\dot{Q}}{\omega\,\eta_t}$$

[0164] The torque transmitted from the driving wheel to the driven wheel will be equal to $C_r = \frac{C_T}{2}$

[0165] Referring to figures 25a and 25b, the total force F directed according to the pressure angle φt is schematically seen. In this view only a component of the total force F is depicted since the three-dimensional projection thereof sloping perpendicular to the helix angle ψ is not visible.

$$4.9 \quad C_r = F_t\,\frac{D_p}{2} \rightarrow F_t = \frac{2\,C_r}{D_p}$$

[0166] Referring to figure 26, a complete three-dimensional depiction of the forces acting on the tooth of a toothed wheel with helical teeth can be seen. The force F acts in the direction which is normal to the contact point.

[0167] Indicating with:

F: total force

Ft: tangential component (force component which is useful for power transmission)

Fr: radial component

Fa: axial component

[0168] Between the total force and the components thereof the following relationships are obtained:

$$5.0 \qquad F_t = F\, cos\varphi_n\, cos\psi$$

$$5.1 \qquad F_r = F\, sen\varphi_n$$

$$5.2 \qquad F_a = F\, cos\varphi_n\, sen$$

[0169] Ft can be usually obtained directly from design data from 4.9 and the other forces can be obtained by the following expressions:

$$5.3 \qquad F_r = F_t tan\,\varphi_t = F_t\,\frac{tan\,\varphi_n}{\cos\psi}$$

$$5.4 \qquad F_a = F_t\, tan\psi$$

$$5.5 \qquad F = \frac{F_t}{\cos\varphi_n \cos\psi}$$

[0170] Referring to figures 27a-c, the direction of the total force in the various points of the gear is evident, in the case of traditional herringbone rotors, prior art rotors with variable helix angle in three zones and the rotors according to the present invention respectively.

[0171] Referring to figure 28a and 28b, it is evident how the direction of the total force has an axial component in the contact point at the ends of the gear, assuming a punctual contact on the pitch diameter, it is evident how this zone is subjected to cyclical stresses and efforts which in certain conditions of use can lead to the chipping ch of this edge.

[0172] At point a there are thus all the components of the total F, whose module is equal to:

$$F = \frac{F_t}{\cos\varphi_n \cos\psi}$$

[0173] Referring to figure 29, it is instead noted how this edge b is subjected to a total force F which, besides having a lower module: $F = \frac{F_t}{\cos\varphi_n\, 1}$ has not even the axial component: $F_a = F_t\, tan0 = 0$.

[0174] Moreover, it is evident how the section of the tooth 6 at the ends has a right-angle shape, whereas it is acute in the prior art, as also shown in comparative figures 30a and 30b.

[0175] Furthermore, recalling the kinematic equations of the speed (at the ends) on the contact point of the helices of the gears on the pitch diameter (always using the punctual contact assumption).

$$V1(t) = \frac{\partial}{\partial t}\left\{k\, r_p\, sen\,\left[\arccos\left(\frac{k-\omega t}{k}\right)\right]\right\} = \frac{r_p\,\omega\,(k-\omega\, t)}{k\sqrt{\frac{\omega\, t\,(2k-\omega\, t)}{k^2}}}$$

$$\lim_{n\to\infty} V1(t) = \frac{r_p\, \omega\, (k-\omega\, t)}{k \sqrt{\frac{\omega\, t\, (2k-\omega\, t)}{k^2}}} = \infty$$

[0176] Tending to infinity represents a paradox since dealing with a speed, but this is due to the fact that, in that segment, the gear has a helix development with angle being equal to 0°. In the initial segment the rotor can be therefore considered in all respects with "straight" teeth and the contact in that zone will no longer appear to be punctual but will be distributed on a line whose length can be calculated and is moreover determined by the mechanical features of the material.

[0177] The method for manufacturing the above-described toothed wheel 1 comprises the following steps.

[0178] With reference to figures 12a-12c,

Step 1:

[0179] Determining the equations of the rotations of all the profile sections in the axial direction calculated on the pitch diameter. A series of coordinates (x_i; y_i; z_i) representing the helix path is obtained.

Step 2:

[0180] Making the rotor front profile slide to manufacture the solid model according to the helix path, shown in figures 8b and 8c, with a 3D software.

Step 3:

[0181] Transferring the solid model to CAD-CAM

Step 4:

[0182] Possible thermal hardening treatment and creation of the inter-tooth space using the numerically controlled working station, for example a five-axis machine.

[0183] The invention brilliantly solves the technical problem and achieves several advantages, the first of which is given by the fact that it was allowed to manufacture gears with opposite helix with partially or totally variable helix angle, with non-encapsulating profile and such a shape as to suppress the cusp in the centre of the rotors.

[0184] Moreover, the accurate and continuous opposite slope of the teeth does not generate any axial force which can tend neither to displace the wheel which can thus be incorporated in gears which are devoid of axial compensation, nor to break terminal parts of the teeth.

[0185] In short, the invention allows to manufacture rotors with opposite helix, non-encapsulating profile and such a helix shape as to suppress the angular point in the centre of the rotors themselves and accordingly all the issues related to the machining thereof by means of machine tools.

[0186] Moreover, the invention allows to manufacture gears for hydraulic apparatuses with opposite helix with a partially or totally variable helix angle.

[0187] Obviously, in order to meet contingent and specific requirements, a person skilled in the art will be allowed to bring several modifications and variants to the above-described invention, all falling however within the scope of protection of the invention as defined by the following claims.

Claims

1. Bi-helical toothed wheel (1) for hydraulic gear apparatuses (2), of the type bound to a support shaft (5) to form a driving or driven wheel of said hydraulic apparatus and comprising a plurality of teeth (6) extending with variable helix angle (ψ) with a continuous function in the longitudinal or axial direction of the tooth (6), wherein each tooth (6) has a central zone (C) with variable helix angle (ψ) wherein a helix transition from right-handed to left-handed occurs; wherein the profile (4) of the teeth keeps a shape continuity in each cross section thereof, **characterized in that** each tooth (6) comprises at least two initial (A) and terminal (E) zones with variable helix angle (ψ) at the two opposite lateral ends of the toothed wheel (1), wherein in these initial (A) and terminal (E) zones the helix angle decreases when approaching the lateral end of the toothed wheel (1).

2. Bi-helical toothed wheel (1) according to claim 1, wherein each tooth (6) further comprises a proximal intermediate zone (B) and a distal intermediate zone (D) with constant helix angle (ψ) which connect the initial (A) and terminal (E) zones to the central zone (C).

3. Bi-helical toothed wheel (1) according to claim 1, wherein the initial (A) and terminal (E) zones and the central zone (C) with variable helix angle are directly connected by inflection points.

4. Bi-helical toothed wheel (1) according to one of the previous claims, wherein the helix angle (ψ) is null at the opposite lateral ends of the toothed wheel (1), that is the profile (4) of each single tooth joins at a right angle the opposite lateral faces of the toothed wheel (1).

5. Bi-helical toothed wheel (1) according to claim 4, wherein the axial component (Fa) of the force (F) exchanged by said wheel meshed during use with another identical wheel is null at the lateral ends.

6. Bi-helical toothed wheel (1) according to one of the previous claims, wherein the profile (4) of the teeth (6) is mirrored with respect to a centre plane passing through the transition point between right-handed and left-handed helix.

7. Bi-helical toothed wheel (1) according to any one of the previous claims, wherein the helix contact ratio parameter (Re) is comprised between 0.6 and 1, preferably between 0.6 and 0.8, more preferably equal to 0.65.

8. Bi-helical toothed wheel (1) according to any one of the previous claims, wherein the profile (4) of the teeth of the toothed wheel (1) is a non-encapsulating profile.

9. Bi-helical toothed wheel (1) according to claim 8, wherein the non-encapsulating profile (4) is defined by two arcs of circumference or elliptical crest and bottom portions (7, 9) connected by an involute profile (8) comprised between two truncation diameters (Øitr, Øetr).

10. Bi-helical toothed wheel (1) according to claim 9, wherein the lower involute truncation diameter is selected equal to: Øitr = Øp - Øp * p1 , with Øp pitch diameter and the parameter p1 comprised between 9.7% and 9.9%, preferably equal to 9.8%.

11. Bi-helical toothed wheel (1) according to one of claims 9 or 10, wherein the upper involute truncation diameter is selected equal to: Øetr = Øp + Øp * p2, with Øp pitch diameter and the parameter p2 comprised between 12.1% and 12.3%, preferably equal to 12.2%.

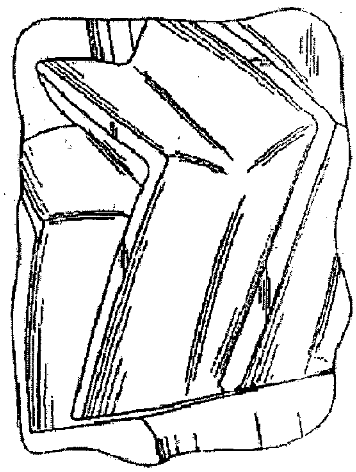
12. Bi-helical toothed wheel (1) wherein the top of each tooth (6) has a cutting edge (10), defined by a limited thickness projecting with respect to the profile (4)

13. Hydraulic gear apparatus comprising a pair of toothed wheels according to any one of the previous claims.

14. Apparatus according to claim 13, wherein said apparatus is a volumetric pump or a hydraulic gear motor.

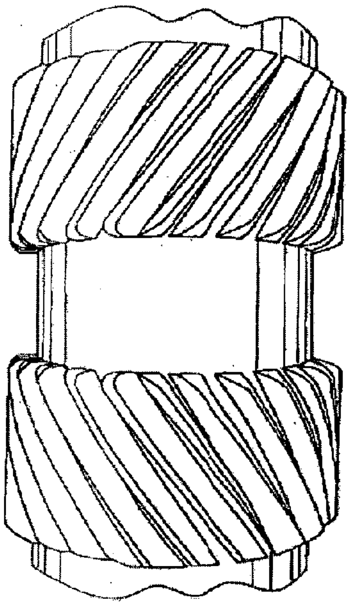
15. Method for manufacturing a bi-helical toothed wheel (1) with non-encapsulating profile (4), for hydraulic gear apparatuses (2), by means of an automatic numerically controlled machine powered by an appropriate software, comprising the following steps:

    determining the equations of the rotations of all the profile sections in the axial direction calculated on the pitch diameter obtaining a series of coordinates (x_i; y_i; z_i) representing the helix path;
    making the rotor front profile slide to manufacture the solid model according to the helix path with a 3D software;
    transferring the solid model to CAD-CAM;
    finishing the inter-tooth space on the working station.



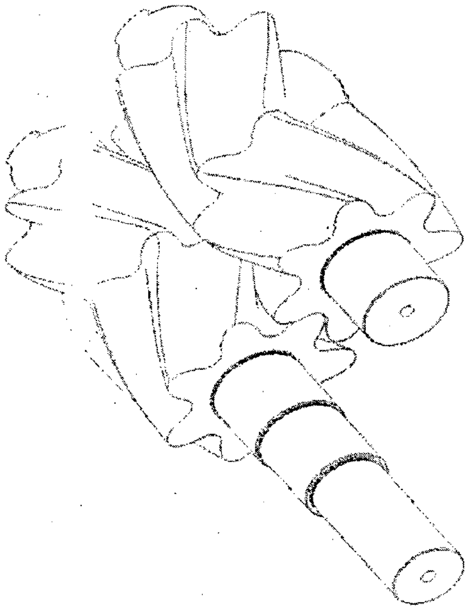
PRIOR ART

Fig. 1



PRIOR ART

Fig. 2



PRIOR ART

Fig. 3

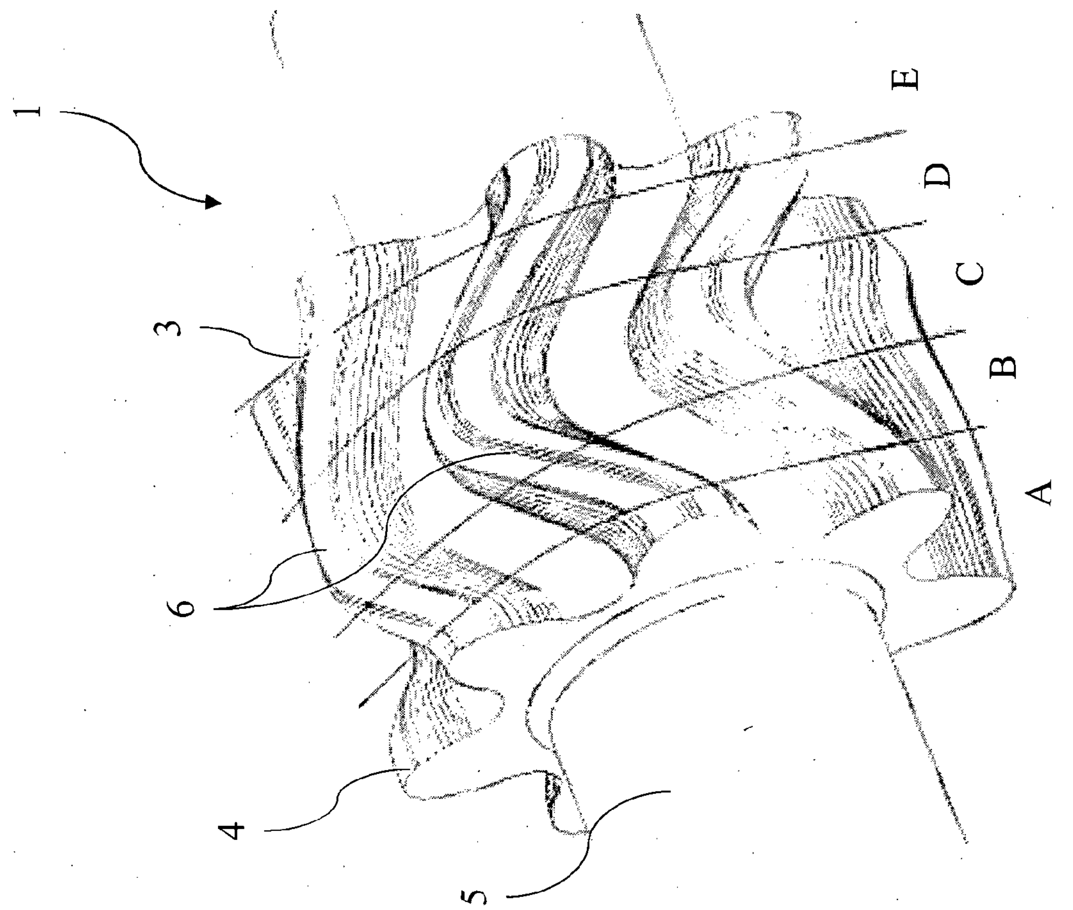


Fig. 5

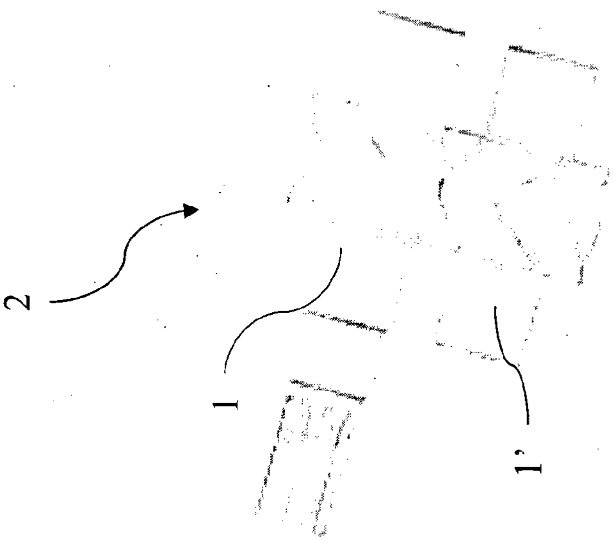


Fig. 4

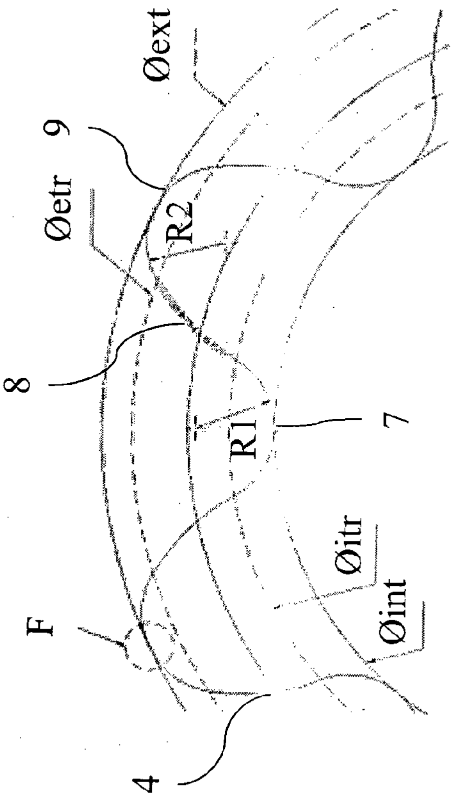


Fig. 6

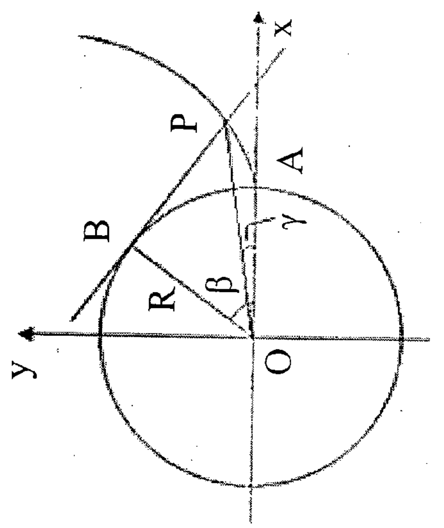


Fig. 7

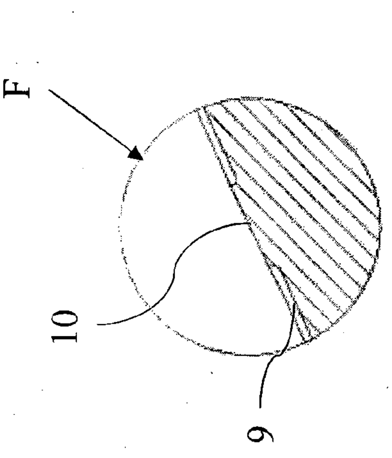


Fig. 8a

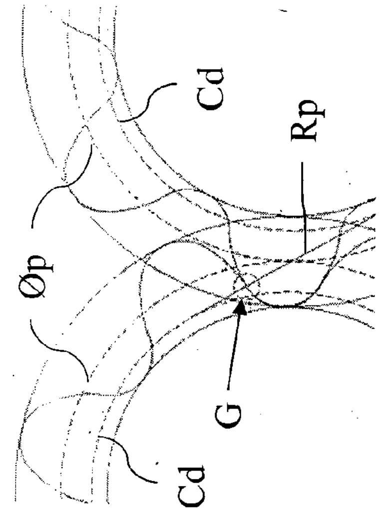


Fig. 8b

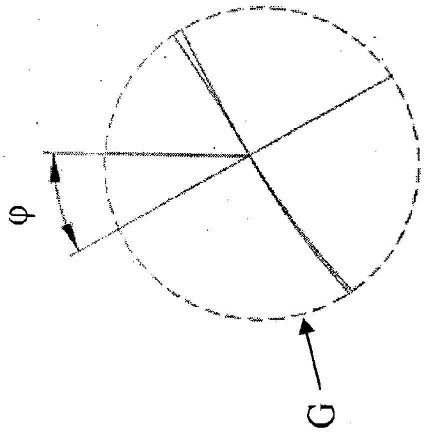


Fig. 8c

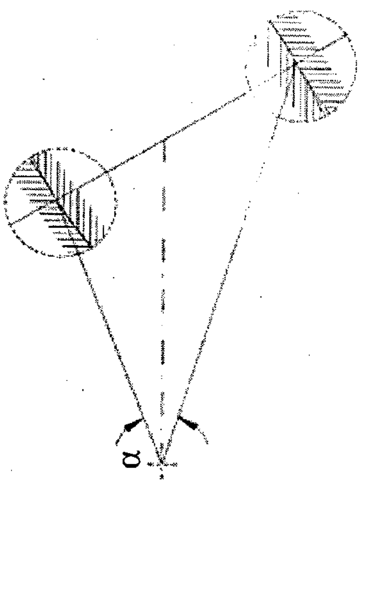


Fig. 8d

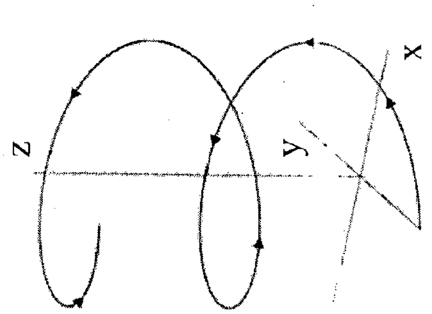


Fig. 10a

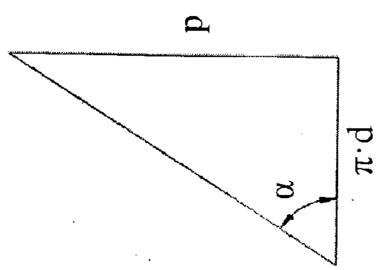


Fig. 10b

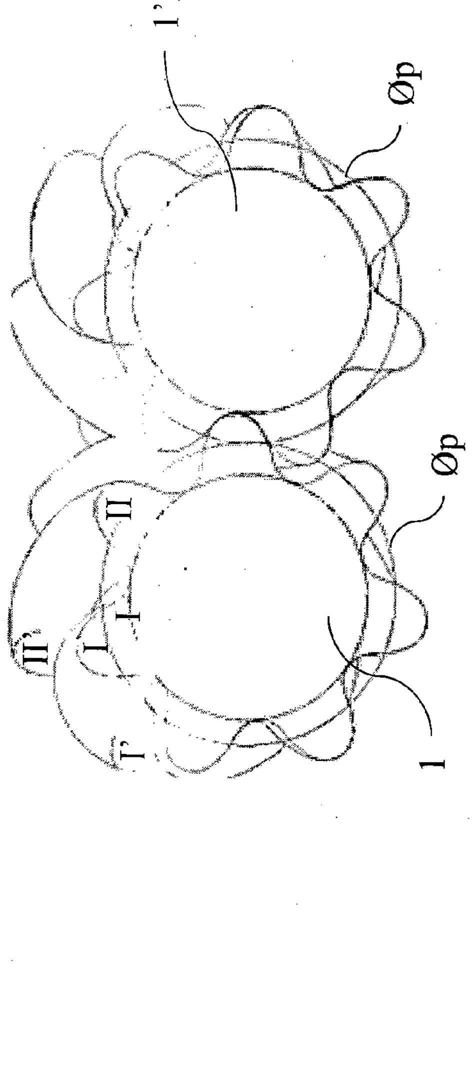


Fig. 9a

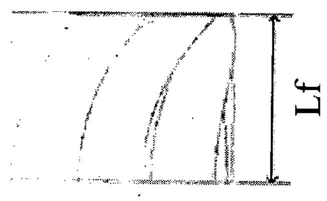
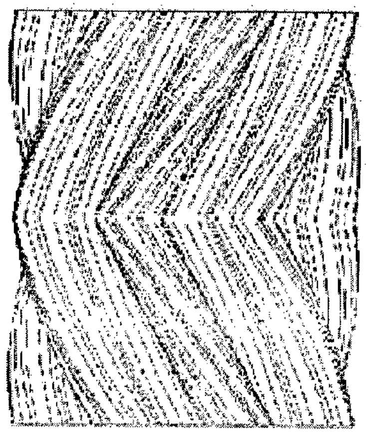


Fig. 9b



PRIOR ART

Fig. 11a

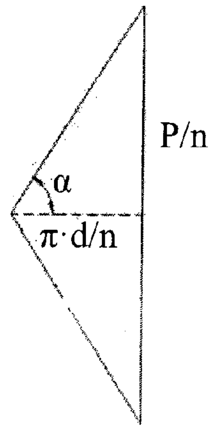


Fig. 11b

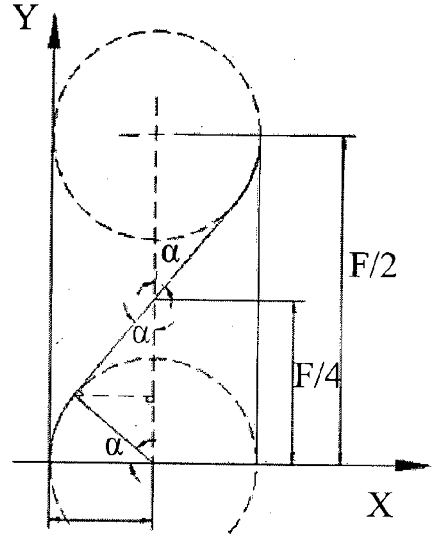


Fig. 13

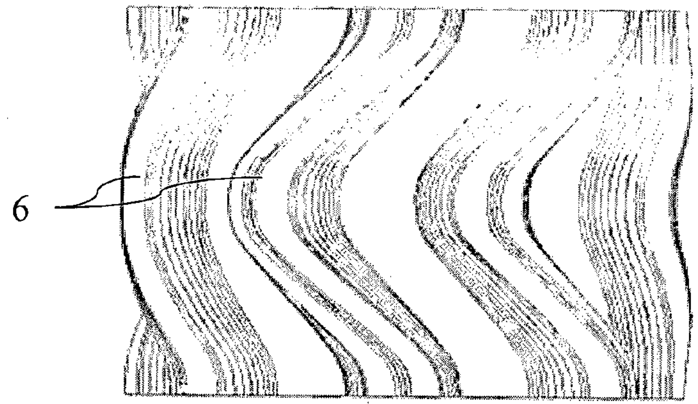


Fig. 12a

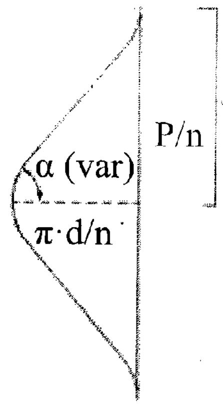


Fig. 12b

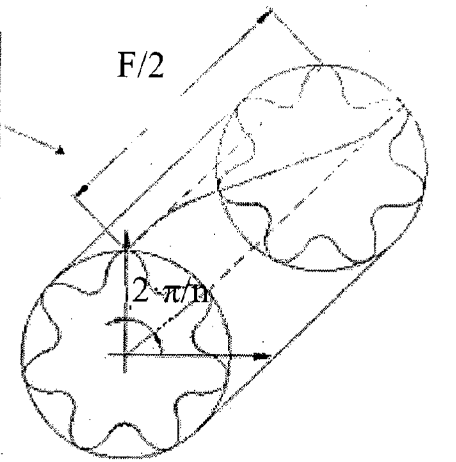


Fig. 12c

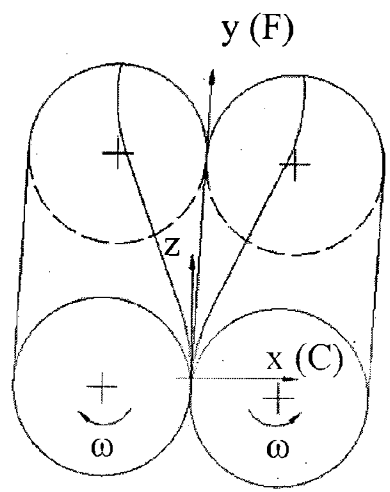


Fig. 14

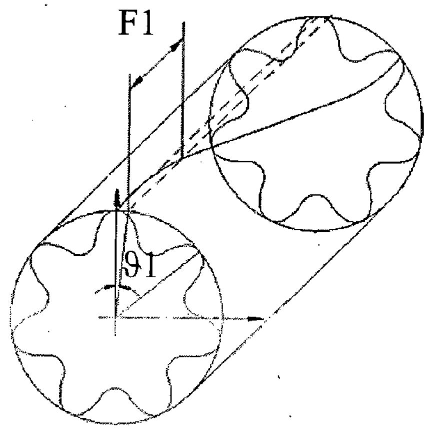


Fig. 16a

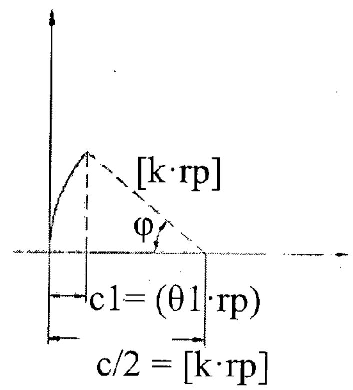


Fig. 16b

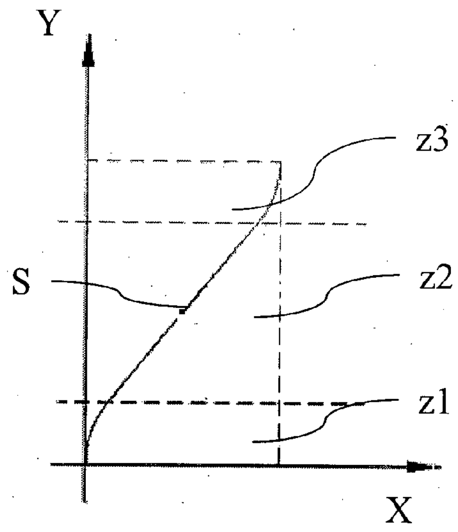


Fig. 15a

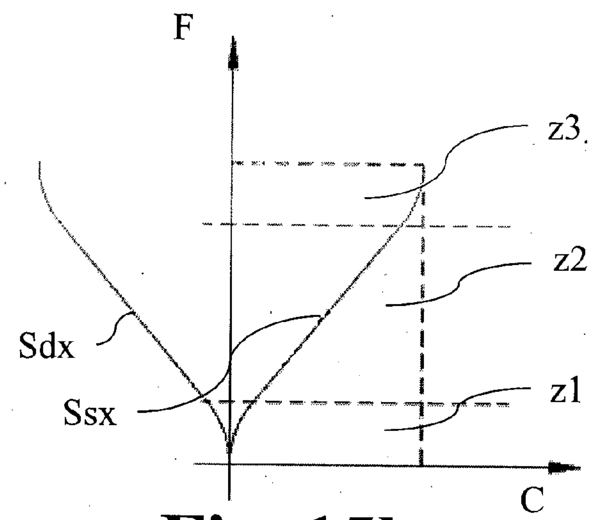


Fig. 15b

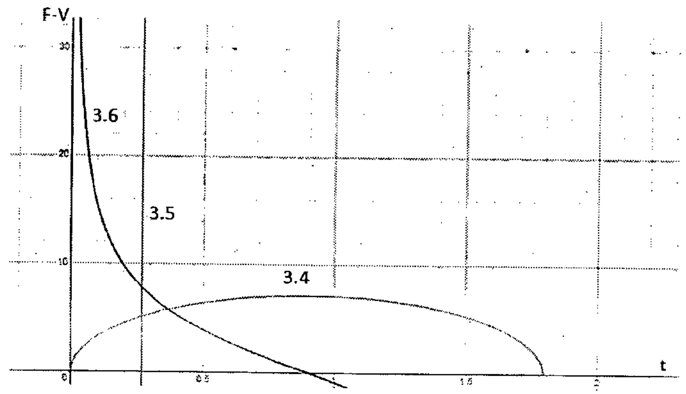


Fig. 17

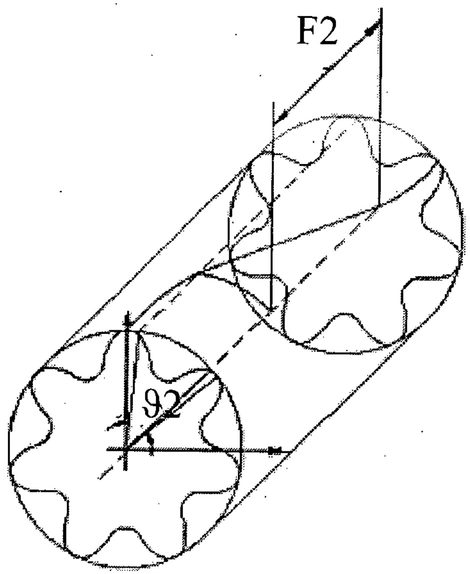


Fig. 18a

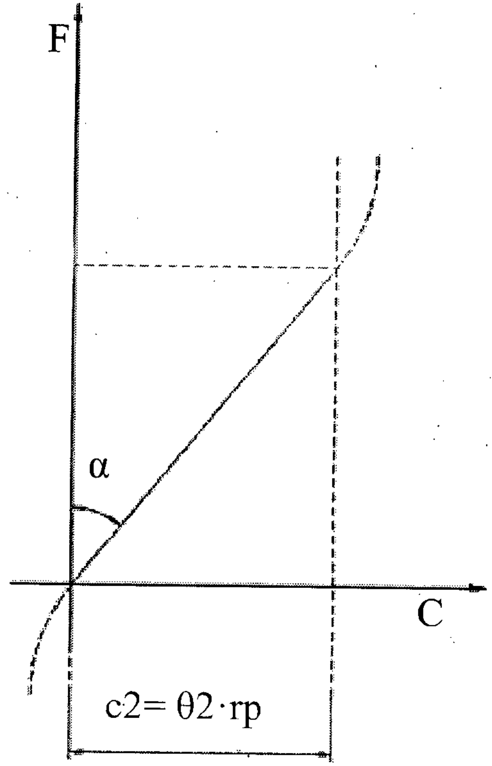


Fig. 18b

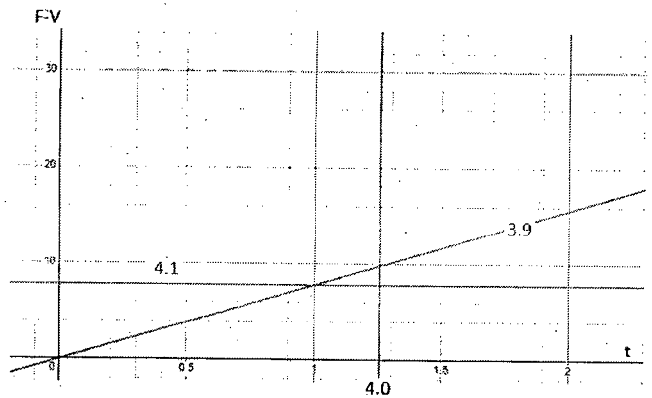


Fig. 18c

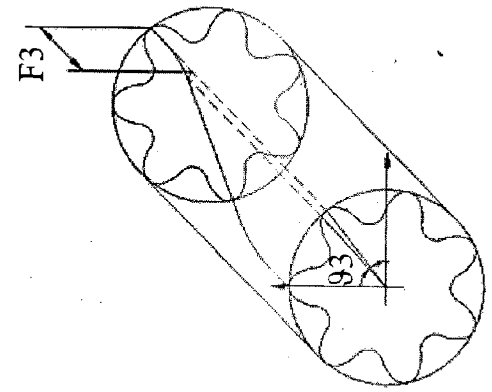


Fig. 19a

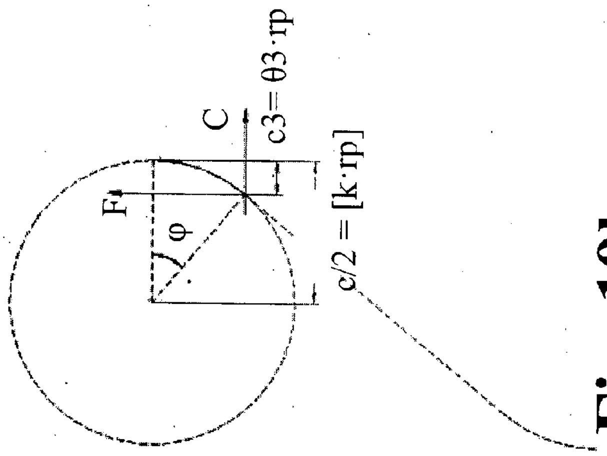


Fig. 19b

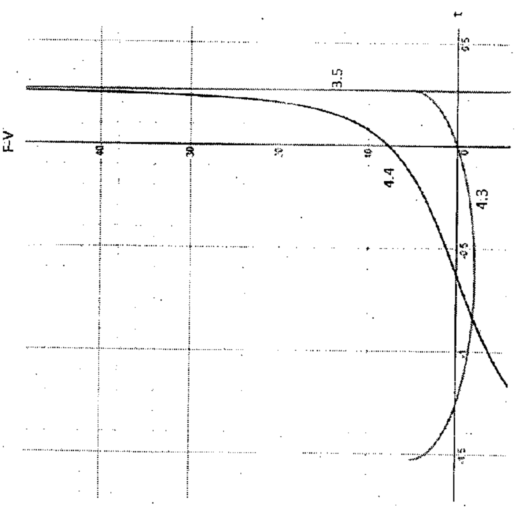


Fig. 19c

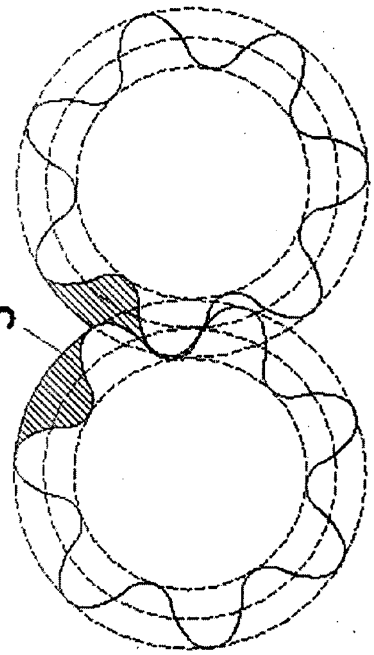


Fig. 20a

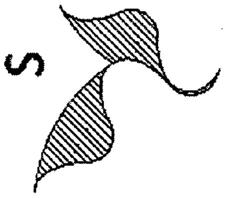


Fig. 20b

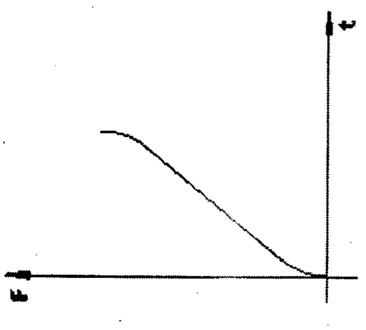


Fig. 21a

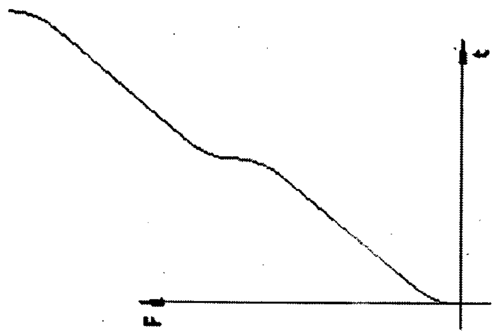


Fig. 21b

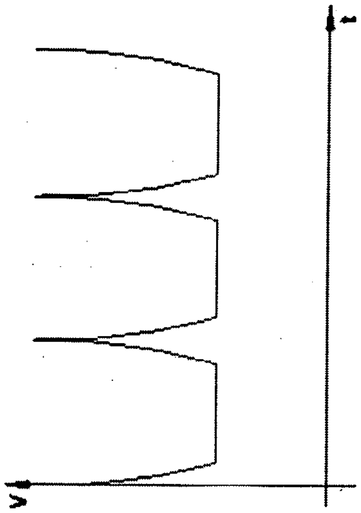


Fig. 21c

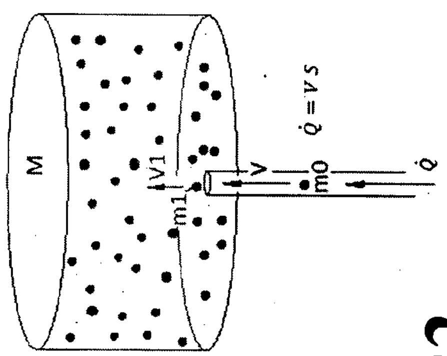


Fig. 22

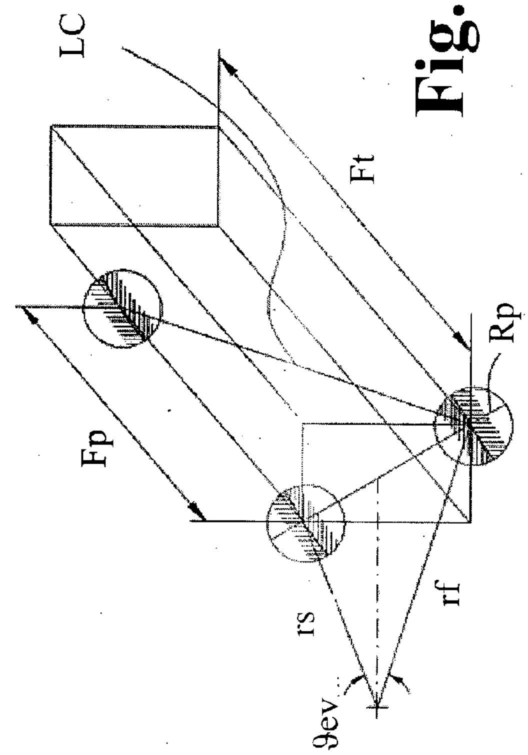


Fig. 23

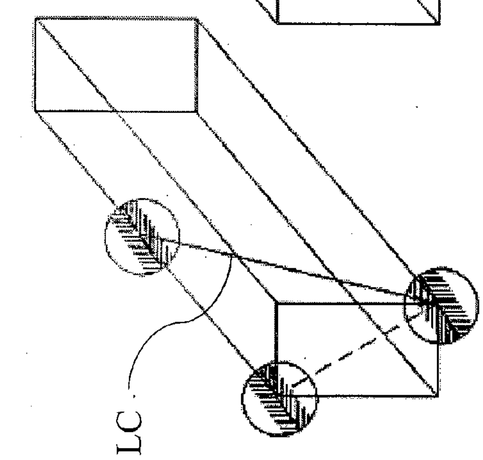


Fig. 24a

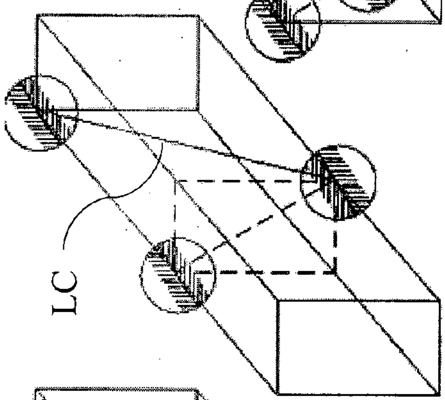


Fig. 24b

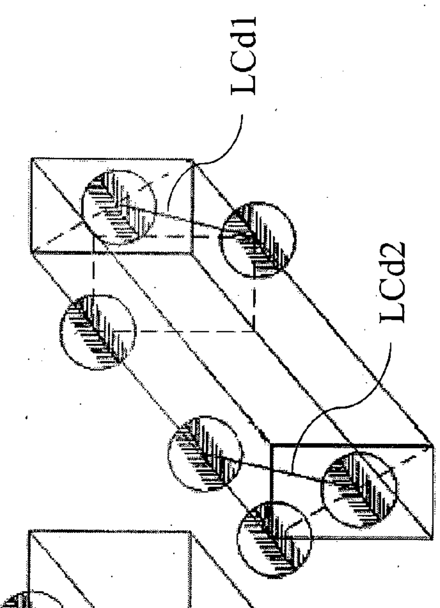


Fig. 24c

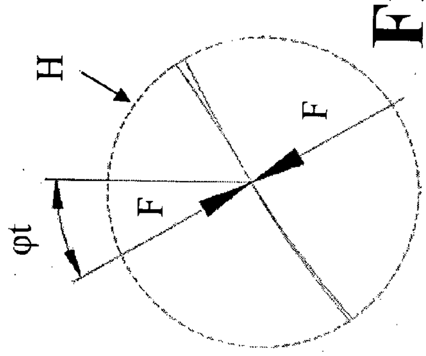


Fig. 25b

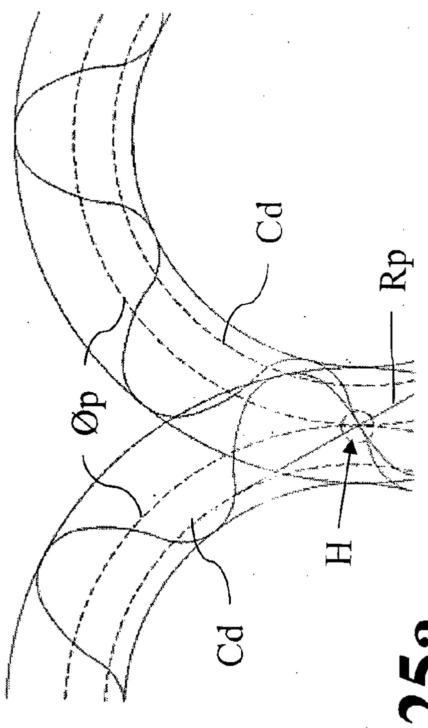


Fig. 25a

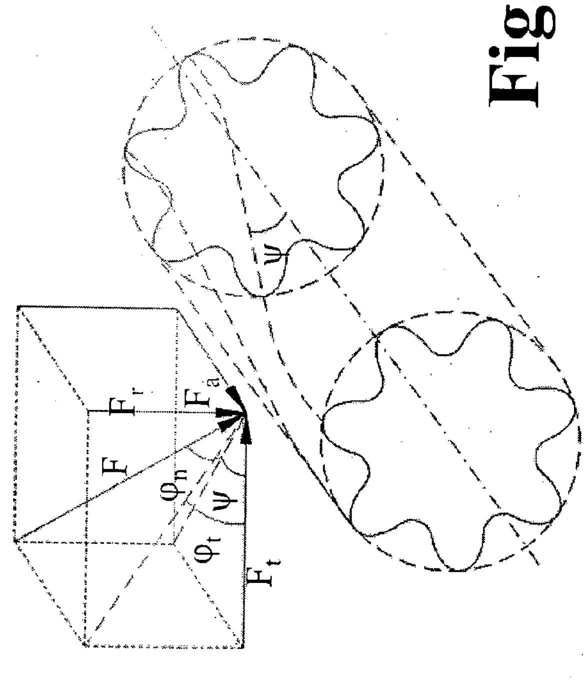
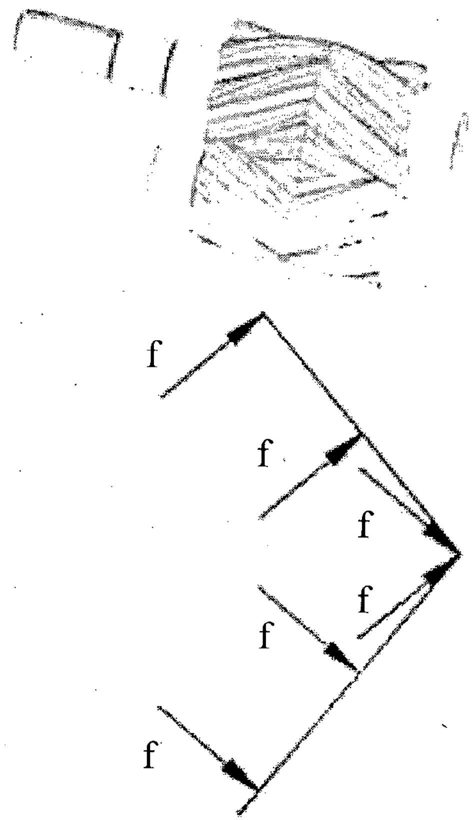
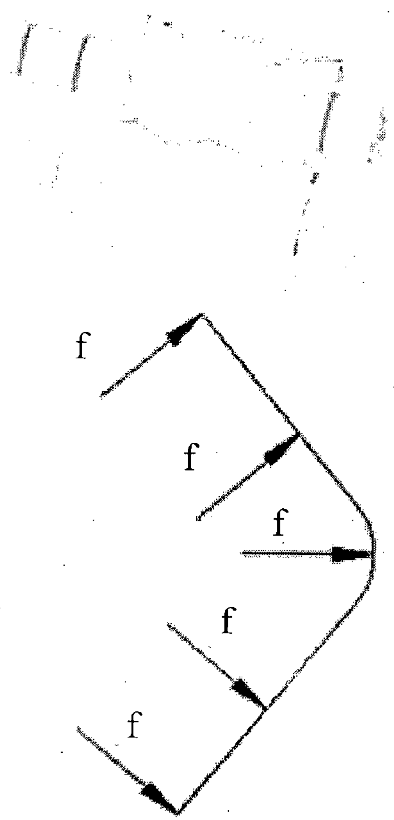


Fig. 26



PRIOR ART

Fig. 27a



PRIOR ART

Fig. 27b

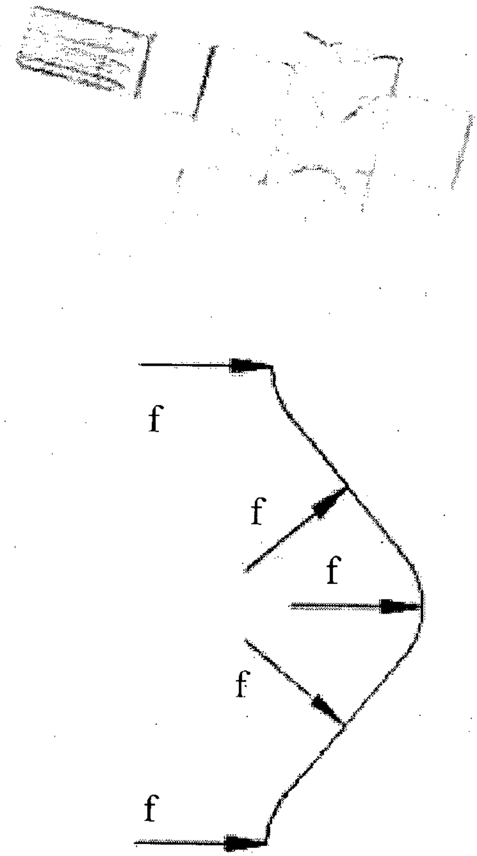
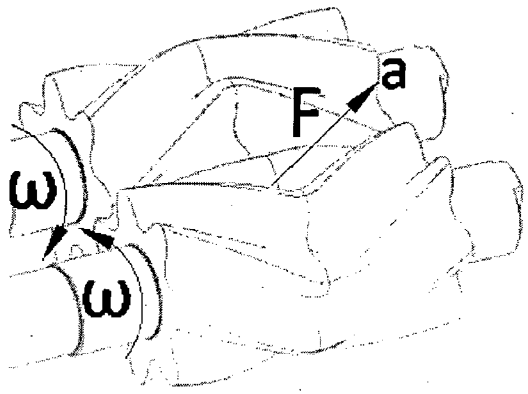
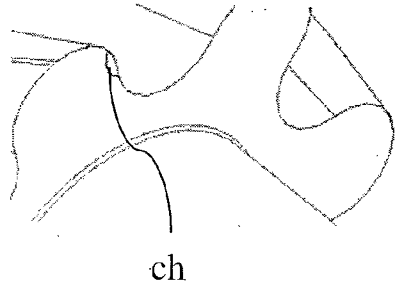


Fig. 27c





PRIOR ART

Fig. 28a

PRIOR ART

Fig. 28b

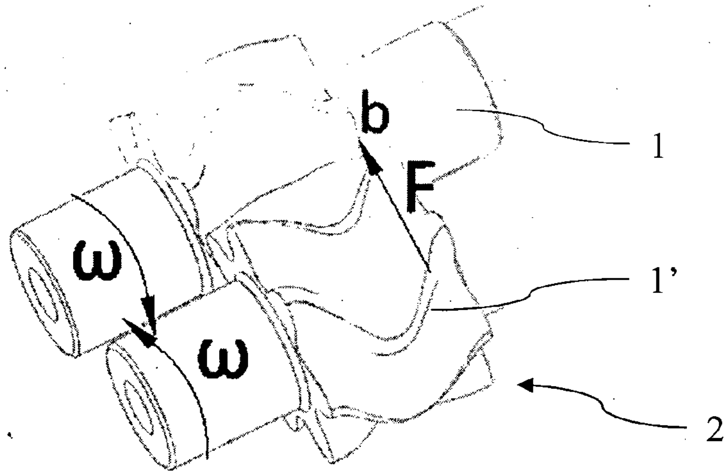


Fig. 29

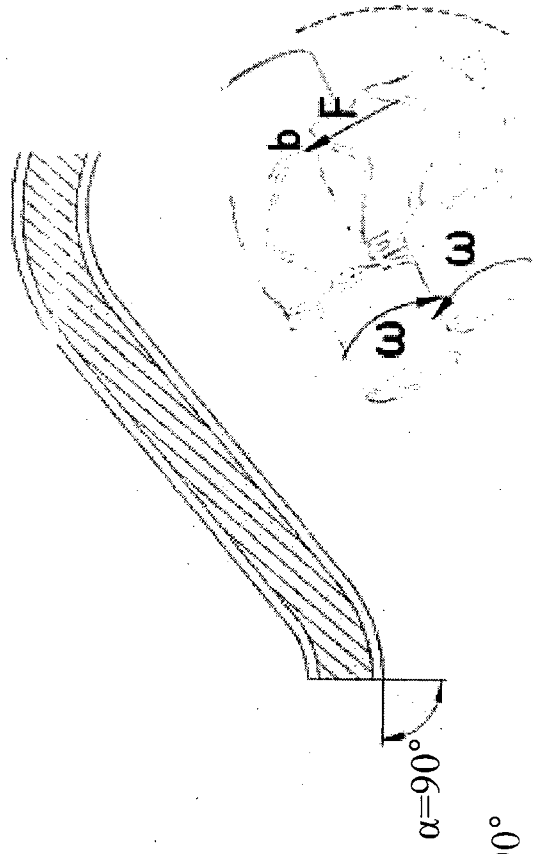
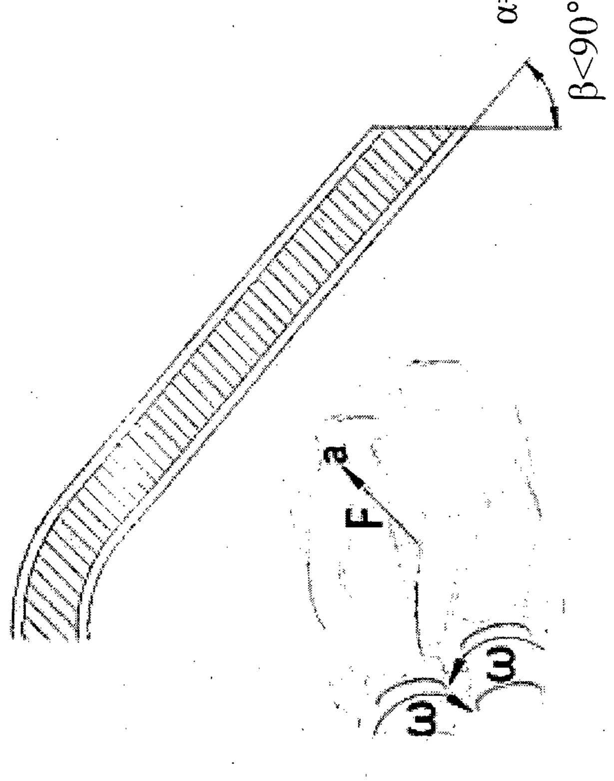


Fig. 30b



PRIOR ART

Fig. 30a

Europäisches Patentamt
European Patent Office
Office européen des brevets

EUROPEAN SEARCH REPORT

Application Number
EP 23 42 5010

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A,D	EP 3 272 999 A1 (SETTIMA MECC SRL - SOCIETÀ A SOCIO UNICO [IT]) 24 January 2018 (2018-01-24) * the whole document * -----	1-14	INV. F04C2/08 F04C2/16 F04C2/18
A	WO 2020/086699 A2 (THE GOVERNMENT OF THE UNITED STATE OF AMERICA AS REPRESENTED BY THE SE) 30 April 2020 (2020-04-30) * the whole document * -----	1-14	
A	DE 10 2014 109914 A1 (UNIV STUTTGART [DE]) 21 January 2016 (2016-01-21) * the whole document * -----	1-14	
A	US 5 415 041 A (FORAN JR CHARLES D [US] ET AL) 16 May 1995 (1995-05-16) * the whole document * -----	1-14	
A	DE 75 506 C (GEBR. PINTSCH) 12 June 1892 (1892-06-12) * the whole document * -----	1-14	TECHNICAL FIELDS SEARCHED (IPC) F04C F01C

~~The present search report has been drawn up for all claims~~

2

Place of search	Date of completion of the search	Examiner
Munich	23 August 2023	Alquezar Getan, M

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons
& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

Europäisches Patentamt
European Patent Office
Office européen des brevets

Application Number

EP 23 42 5010

CLAIMS INCURRING FEES

The present European patent application comprised at the time of filing claims for which payment was due.

☐ Only part of the claims have been paid within the prescribed time limit. The present European search report has been drawn up for those claims for which no payment was due and for those claims for which claims fees have been paid, namely claim(s):

☐ No claims fees have been paid within the prescribed time limit. The present European search report has been drawn up for those claims for which no payment was due.

LACK OF UNITY OF INVENTION

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

see sheet B

☐ All further search fees have been paid within the fixed time limit. The present European search report has been drawn up for all claims.

☐ As all searchable claims could be searched without effort justifying an additional fee, the Search Division did not invite payment of any additional fee.

☐ Only part of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the inventions in respect of which search fees have been paid, namely claims:

☒ None of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims, namely claims:

1-14

☐ The present supplementary European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims (Rule 164 (1) EPC).

Europäisches Patentamt
European Patent Office
Office européen des brevets

LACK OF UNITY OF INVENTION SHEET B

Application Number
EP 23 42 5010

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

1. claims: 1-14

   Relating to a bi-helical toothed wheel provided with a variable helix angle wherein in the initial and terminal zones the helix angle decreases

   ---

2. claim: 15

   Relating to a method for manufacturing a bi-helical toothed wheel with non-encapsulating profile

   ---

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO.

EP 23 42 5010

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
The European Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

23-08-2023

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
EP 3272999 A1	24-01-2018	CN 107642592 A	30-01-2018
		DK 3272999 T3	06-05-2019
		EP 3272999 A1	24-01-2018
		ES 2726026 T3	01-10-2019
		TR 201907186 T4	21-06-2019
		US 2018023561 A1	25-01-2018
WO 2020086699 A2	30-04-2020	EP 3870877 A2	01-09-2021
		US 2020124047 A1	23-04-2020
		WO 2020086699 A2	30-04-2020
DE 102014109914 A1	21-01-2016	DE 102014109914 A1	21-01-2016
		DE 202014007647 U1	22-01-2015
US 5415041 A	16-05-1995	NONE	
DE 75506 C	12-06-1892	NONE	

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- US 20040031152 A1 **[0019]**
- US 7040870 B1 **[0019]**
- EP 17181600 A **[0020] [0022]**