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

(11) EP 2 317 144 A2

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
04.05.2011 Bulletin 2011/18

(51) Int Cl.:
F04C 2/14 *(2006.01)*

(21) Application number: **10189404.6**

(22) Date of filing: **29.10.2010**

(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 MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA ME

(30) Priority: **30.10.2009 IT BO20090714**

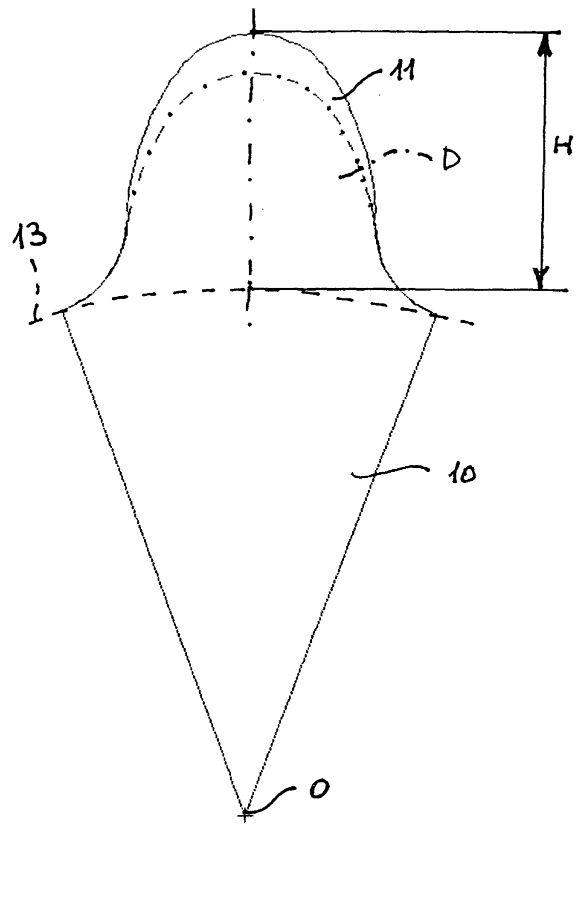
(71) Applicants:
- **Morselli, Mario Antonio**
41100 Modena (IT)
- **Settima Meccanica S.R.L.**
29020 Settima (Piacenza) (IT)

(72) Inventor: **Morselli, Mario Antonio**
41121 Modena (MO) (IT)

(74) Representative: **Provvisionato, Paolo**
Provvisionato & Co S.r.l.
Piazza di Porta Mascarella 7
40126 Bologna (IT)

(54) **Gear for an hydraulic gear machine**

(57) A hydraulic gear apparatus comprises a pair of gears which mesh with each other with semi-encapsulation. Each gear has a plurality of teeth having a profile which falls within a tolerance band of ± 1/15, more preferably ± 1/20, and even more preferably ± 1/30 with respect to the height of the tooth, with respect to a profile homothetic to a profile defined by a predetermined spline function passing through a plurality of node points having predetermined coordinates {X,Y} with their origin on the axis of rotation.



EP 2 317 144 A2

Description

[0001] The present invention relates to a gear of the type having a profile suitable for meshing with semi-encapsulation in a hydraulic gear apparatus. Typical examples of hydraulic gear apparatuses in which the gears of the present invention can be used in an optimum manner, and to which specific reference will be made in the course of the present description, are positive-displacement rotary gear pumps, but the gears of the present invention may also be used analogously in hydraulic gear motors, which are therefore regarded as being included in the scope of the present invention.

[0002] Positive-displacement rotary gear pumps are generally composed of two gears which in the majority of cases are of the type having straight teeth and one of which, referred to as the driving gear, is connected to a control shaft and drives the other gear, referred to as the driven gear, in rotation. A particular disadvantage of the above-mentioned gear pumps of the traditional type, which generally have an involute tooth profile, is the fact that the pumped fluid is encapsulated, that is to say, trapped, and compressed or at any rate subjected to variations in volume in the enclosed spaces between the profiles of the teeth in the meshing zone, thus giving rise to detrimental and uncontrolled local stress peaks which cause direct operating noise.

[0003] Also known in addition to the direct operating noise indicated above is the problem resulting from the phenomenon of irregularity, or ripple, in the transfer of fluid which involves indirect operating noise, known as ripple noise, associated with the flow pulsation, and therefore pressure pulsation, in the consumer circuit. In other words, oscillations in the flow of fluid generate a pulsating wave which, through the fluid itself, is transmitted to the surroundings and, in particular, to the walls of the pump, to the tubing and to the delivery ducts. The noise induced may even reach unpredictable levels if the above-mentioned bodies start to resonate at the frequency of oscillation or ripple. A series of studies and experiments has demonstrated that those oscillations are intrinsically due to the configuration of the rotors or gears of the above-mentioned pumps which, in subsequent stages of their meshing, give rise to a discontinuity in the variation of the volume which brings about the transport of the fluid from suction to delivery. In other words, the ripple is due to the discontinuity in the variation of that volume with respect to time or, rather, with respect to the angular position of the rotors relative to each other. The above-mentioned phenomena are clearly and fully described in the articles MORSELLI Mario Antonio, "Rumorosità meccanica e idraulica delle pompe ad ingranaggi " (Mechanical and hydraulic noise in gear pumps), Oleodinamica Pneumatica, January 2005, pp. 54-59*, and February 2005, pp. 42-46,* which also appeared in Fluides & Transmissions (Fluids & Transmissions), n°75, April 2005, pp. 34-3 7 *and n°77, May 2005, pp.20-26.*

[0004] Some solutions which have dealt with the problems illustrated above, with greater or lesser success, are known.

[0005] Some of those solutions relate to pumps with traditional toothing, with mostly, but not necessarily, involute tooth flank profiles, with straight, or much more rarely, helical toothing, with clearance (that is to say, with single contact of a tooth of one gear with a corresponding tooth of the other gear) or theoretically without clearance (that is to say, with double contact, where both of the tooth flanks are theoretically always in mesh, as in the pump manufactured by Bosch Rexroth AG which is known by the trade name SILENCE, or the pump manufactured by Casappa S.p.A. which is known by the trade name WHISPER). In those solutions, the fluid trapped between the teeth is at least in part "discharged", that is to say, evacuated, through suitable outlets or pockets or ducts formed on the faces of the lateral shims, otherwise referred to as supports or bushes, of the gears, that is to say, on the walls which face the flat lateral ends of the gears, and which enable the encapsulated volume of fluid to be discharged (or admitted) towards the appropriate opening or port, at high or low pressure, respectively.

[0006] The production of the pockets on the faces of the lateral shims is, however, much more complex when it is desired to produce helical gears for reducing the problem of ripple noise. Furthermore, the adoption of helical gears has, per se, a series of additional problems because in that case the volume of each area of entrapment of the fluid likewise extends, like the teeth of the gears, in the manner of a helical thread over the entire width of the gear, thus representing a potential communication route or by-pass between suction and delivery if special precautions are not taken. In practice, either there is a limitation to small helix angles of the gears, or solutions that are very complex and expensive from the point of view of construction are adopted, like that described in the document EP-0769104 of Brown David Hydraulics Ltd in which the gears have, for each of their cross-sections, at least two teeth which are simultaneously in mesh. However, those solutions are complex and, in essence, not very efficient because they have been developed on the basis of concepts closer to mathematical abstractions than to practical and technologically feasible possibilities; in practice, the geometry of the pockets is always a not entirely satisfactory compromise.

[0007] In any case, all of the known pump solutions, with either straight teeth or helical teeth and with single or double contact, which use discharge pockets on the lateral shims, nevertheless have a residual trapped volume which is subject to variations which cannot be eliminated and which therefore generate a certain residual noise, in addition to having a significant and detrimental ripple.

[0008] Other solutions of known type to the problems of direct and indirect noise indicated above relate to pumps having toothing with an unconventional profile, which we may define as "with continuous contact", which do not trap fluid between the tooth head and base. In practice, the intermeshing gears have profiles with a rounded appearance in the head of the tooth and a single theoretical contact point which moves from one flank to the other of the gear with continuity, in such a manner as never to produce a closed area of fluid entrapment during meshing, over the entire width of the gears. That principle, broadly outlined in theory and in a very general manner in the documents US-2159744, US-3164099, US-3209611, which has never, however, found any practical application, has been fully developed and described in the documents EP-A-1132618, EP-B-1371848, US-6769891 of the same joint applicant and inventor of the present application, as well as in the technical articles mentioned above, and has found practical application in the pump known by the trade name *Continuum®* and manufactured by Settima Flow Mechanisms. The toothings developed by the present inventor do not have a by-pass between the suction and delivery of the pump, and they have a minimum pulsation of the fluid and a substantial meshing quietness. This last solution, although it has been demonstrated to be markedly superior from the point of view of quietness compared with traditional pumps, nevertheless has the disadvantage of a volumetric efficiency slightly lower than that of the known pump solutions in which fluid entrapment occurs. The main reason for this resides in the modest tooth height producible with a profile configured in accordance with the concept of "non-encapsulation", and therefore a corresponding modest effective flow per volume unit, for the same number of teeth. In order to be able to have effective unit flow rates, comparable to those of pumps with encapsulation, the inventor, unlike the traditional bibliography, has identified an optimum range for this solution of from 5 to 10 teeth, with an optimum of 7 teeth, which is a low tooth number per se but which involves greater volumetric losses owing to the lesser sealing between delivery at high pressure and suction at low pressure, because the teeth also act as labyrinth seals.

[0009] All of the problems discussed above are heightened in the case of hydraulic apparatuses which are to operate with high pressure differentials, for example in the case of gear pumps for pressure differentials greater than a few tens of bar, and even more so for pressures greater than 80-100 bar, as for low-viscosity fluids.

[0010] International patent application WO 2008/111017 belonging to the same applicants, the content of which is fully incorporated herein by reference, relates to an improved hydraulic gear apparatus comprising a pair of meshing gears mounted to be rotatable relative to each other in a casing between an inlet side and an outlet side for a fluid having, in use, a substantially transverse flow with respect to the axes of rotation of the gears, the meshing gears producing, in their rotation relative to each other, progressive meshing configurations between respective co-operating teeth, there being defined in at least one of the said progressive meshing configurations, in at least one cross-section of the gears, at least one closed area between respective teeth for fluid entrapment, the said closed fluid-entrapment area decreasing until it substantially disappears at or in the vicinity of at least one other, different, progressive meshing configuration between the above-mentioned respective cooperating teeth.

[0011] In summary, the behaviour of the gears appearing in patent application WO 2008/111017 belonging to the same applicants is such that there is formed between teeth of the two gears which mesh, a zone of fluid entrapment or encapsulation which gradually, during the rotational movement of the gears, decreases until it substantially disappears when the head of a tooth of one gear touches the base of a tooth of the other gear. That behaviour, for the purposes of the present description, will be referred to as "semi-encapsulation".

[0012] Experiments carried out by the applicants on various gears to be used in the hydraulic apparatuses mentioned above have demonstrated that there is a limited range of tooth profiles capable of being simultaneously effective in reducing the noise of the pump and, at the same time, ensuring the possibility of relatively simple manufacture, which can help to contain the production costs of hydraulic apparatuses and, in particular, of positive-displacement pumps which adopt the principle of "semi-encapsulation". In addition, this specifically identified series of profiles has the advantage of enhanced reliability in use, which makes the use thereof particularly advantageous in high-pressure positive-displacement pumps. In this series of profiles, the higher proportion of teeth than in the previously known solutions enables markedly improved efficiency to be achieved.

[0013] In order to achieve the objects indicated above, the invention relates to a gear having a plurality of teeth suitable for meshing with the teeth of another, corresponding, gear, the profile of each tooth of the gear, in cross-section, being defined in the claims which follow.

[0014] In particular, the profile of at least one tooth of one of the two rotors is defined by a spline function passing through a plurality of node points having predetermined coordinates, with a tolerance of $\pm$ 1/15, more preferably $\pm$ 1/20, and even more preferably $\pm$ 1/30 of the height of the tooth of the gear on the theoretical profile defined by the plurality of preferred node points. The node points are defined by a pair of values {X', Y'} expressed in a system of cartesian coordinates having their origin in the centre of the pitch circle of the gear. Although evident from the following description, it should be specified that the origin of the system of x, y coordinates is the line of the axis of rotation of the gear on a plane perpendicular to that axis, which precisely coincides with the centre of the pitch circle of the said gear.

[0015] In the present description, the expression "spline function" refers generally to any spline function which does not introduce errors, or to a smoothing spline having a sufficiently small smoothing parameter not to introduce significant errors with respect to the node points.

[0016] In a preferred but non-limiting embodiment of the present invention, the spline function adopted is a natural cubic spline function, that is to say, a third-degree interpolating natural spline function.

[0017] Although the natural spline provides some theoretical advantages, the choice of the type of spline is nevertheless not binding because, depending on the case and, for example, on the data format required by the machine tools, the person skilled in the art may fmd it more convenient to use different spline functions, or also smoothing splines, also because some of those spline functions are normally present and used in CAD and CAD-CAM. systems.

[0018] The gears are advantageously helical and the face contact ratio of the helical toothing is from 0.4 to 1.2, preferably from 0.5 to 1.2, more preferably from 0.6 to 1.2, more preferably from 0.7 to 1.1, more preferably from 0.8 to 1.1, and even more preferably from 0.9 to 1. In a preferred but non-limiting embodiment of the present invention, the face contact ratio of the helical toothing is equal to or close to unity.

[0019] Advantageously, a gear according to the present invention has a ratio of the sizes of the face width and the pitch diameter of from 0.5 to 2, preferably from 0.6 to 1.8, more preferably from 0.65 to 1.5, and even more preferably from 0.7 to 1.25. In a preferred but non-limiting embodiment of the present invention, the ratio of the sizes of the face width and the pitch diameter is close to unity.

[0020] The present invention relates also to a hydraulic gear apparatus comprising a pair of meshing gears having a tooth profile of the type indicated above. In particular, this hydraulic apparatus may be a hydraulic pump or a hydraulic motor.

[0021] Further features and advantages of the invention will emerge from the following description of a preferred embodiment, with reference to the single appended Figure, which is given purely by way of non-limiting example and which illustrates the profile of a tooth of a gear according to the present invention, compared with the profile of a tooth of the prior art for a gear without encapsulation.

[0022] Although the description which follows is given with reference to a pump, the same arguments and considerations may be applied to the analogous hydraulic motors.

[0023] Referring now to the single Figure, a gear 10 according to the present invention (only one sector of which is illustrated in the Figure) is to mesh with another, corresponding, gear (not shown) for use in a positive-displacement rotary pump, preferably of the type for high operating pressures, where the pressure differentials between suction and delivery are greater than a few tens of bar, more particularly greater than approximately 50 bar, and even more particularly greater than approximately 80-100 bar.

[0024] The gear 10 comprises a plurality of teeth 11 having a height H and a profile suitable for meshing with semi-encapsulation with the teeth of the other, corresponding, gear. The profile of the teeth 11 cannot be described as a succession of simple geometric curves but can be defined by a natural cubic spline function (although it is possible, in the terms already indicated above, to use other spline or smoothing spline functions) passing through a plurality of node points 12 defined by pairs of values expressed in a system of cartesian coordinates having their origin in the centre O of the pitch circle 13 of the gear 10.

[0025] At any rate, the resultant profiles must be conjugate, if not exactly from an analytical point of view, at least from a practical point of view, that is to say, the profiles must be suitable for meshing correctly in actual use in the hydraulic apparatuses to which the present invention relates. It is worth pointing out in this connection that, even in the current art, traditional "involute" gears are often not produced in accordance with "pure" involute geometry but with small differences or variations with respect thereto, giving rise to profiles which are variously referred to as K profile, tip relief, etc.

[0026] Experiments carried out by the applicants have led to the identification of a series of tooth profiles especially suited to the production of gears with seven, eight, nine or ten teeth each. The actual profile of the teeth 11 may fall within a tolerance band whose width is ± 1/15, more preferably 1/20, and even more preferably ± 1/30 of the height H of the tooth of the gear.

[0027] The single Figure shows a comparison between the profile of the tooth 11 of a gear produced in accordance with the invention, and the profile of a tooth D of the prior art which is drawn with a dot-dash line and which is designed in accordance with the concept of "non-encapsulation". It can be seen immediately that the tooth 11 is markedly taller than the tooth D of the prior art and it will therefore be appreciated that a gear having teeth 11 produced in accordance with the principle of "semi-encapsulation" of the present invention leads to a greater volumetric efficiency than do the gears produced in accordance with the principle of "non-encapsulation", if for no other reason than that it is possible to adopt, for the same flow rate and space requirement, a larger number of teeth.

[0028] Given below are some examples relating to gears of the present invention having different numbers of teeth.

<u>Example 1</u>

[0029] A gear having a number of teeth equal to seven has a theoretical tooth profile defined by a natural cubic spline function (which, if necessary, may be replaced by another spline or smoothing spline function) passing through a plurality of node points defined by a pair of values {X',Y'} expressed in a system of cartesian coordinates having their origin in the centre O of the pitch circle P of the gear. The coordinates of the node points are homothetic to the pairs of values {X,Y} of the list reproduced in the following Table 1.

Table 1

X	Y	X	Y	X	Y	X	Y
-5.29	10.99	-3.52	13.62	-3.36	15.79	-2.09	18.28
-4.94	11.21	-3.51	13.84	-3.30	16.04	-1.79	18.51
-4.71	11.37	-3.52	14.06	-3.21	16.38	-1.46	18.70
-4.49	11.54	-3.55	14.35	-3.13	16.62	-0.93	18.92
-4.28	11.74	-3.56	14.61	-3.06	16.79	-0.75	18.98
-4.10	11.98	-3.55	14.78	-3.00	16.94	-0.57	19.03
-3.94	12.24	-3.54	14.95	-2.93	17.09	-0.38	19.06
-3.81	12.53	-3.51	15.12	-2.76	17.41	-0.19	19.07
-3.69	12.86	-3.44	15.46	-2.56	17.71	0.00	19.08
-3.58	13.25	-3.40	15.63	-2.35	18.01		

Example 2

[0030] A gear having a number of teeth equal to eight has a theoretical tooth profile defined by a natural cubic spline function (which, if necessary, can be replaced by another spline or smoothing spline function) passing through a plurality of node points defined by a pair of values {X',Y'} expressed in a system of cartesian coordinates having their origin in the centre O of the pitch circle P of the gear. The coordinates of the node points are homothetic to the pairs of values {X,Y} of the list reproduced in the following Table 2.

Table 2

X	Y	X	Y	X	Y	X	Y
0.00	19.08	2.42	17.52	3.07	15.62	3.45	13.12
0.30	19.06	2.53	17.26	3.10	15.44	3.54	12.94
0.61	19.01	2.60	17.09	3.13	15.26	3.70	12.68
0.91	18.93	2.66	16.92	3.17	14.99	3.86	12.45
1.20	18.81	2.73	16.75	3.19	14.81	4.05	12.24
1.46	18.64	2.84	16.50	3.20	14.63	4.28	12.06
1.70	18.44	2.90	16.33	3.20	13.99	4.66	11.84
1.91	18.23	2.96	16.15	3.23	13.76	4.86	11.72
2.11	18.01	3.00	15.98	3.29	13.53		
2.29	17.77	3.04	15.80	3.37	13.29		

Example 3

[0031] A gear having a number of teeth equal to nine has a theoretical tooth profile defined by a natural cubic spline function (which, if necessary, can be replaced by another spline or smoothing spline function) passing through a plurality of node points defined by a pair of values {X',Y'} expressed in a system of cartesian coordinates having their origin in the centre O of the pitch circle P of the gear. The coordinates of the node points are homothetic to the pairs of values {X,Y} of the list reproduced in the following Table 3.

Table 3

X	Y	X	Y	X	Y	X	Y
-4.47	12.27	-2.91	14.38	-2.68	16.34	-1.82	18.22
-4.34	12.33	-2.89	14.57	-2.62	16.51	-1.64	18.41
-4.09	12.47	-2.89	14.76	-2.55	16.68	-1.4	18.58
-3.85	12.62	-2.88	15.08	-2.48	16.85	-1.22	18.73
-3.64	12.79	-2.86	15.26	-2.41	17.02	-1.00	18.86
-3.45	12.98	-2.85	15.44	-2.34	17.19	-0.77	18.97
-3.19	13.37	-2.83	15.62	-2.28	17.36	-0.52	19.05

(continued)

X	Y	X	Y	X	Y	X	Y
-3.03	13.77	-2.80	15.80	-2.21	17.53	-0.26	19.06
-2.98	13.96	-2.77	15.98	-2.13	17.70	0.00	19.08
-2.95	14.14	-2.73	16.16	-1.97	18.01		

Example 4

[0032] A gear having a number of teeth equal to ten has a theoretical tooth profile defined by a natural cubic spline function (which, if necessary, can be replaced by another spline or smoothing spline function) passing through a plurality of node points defined by a pair of values {X',Y'} expressed in a system of cartesian coordinates having their origin in the centre O of the pitch circle P of the gear. The coordinates of the node points are homothetic to the pairs of values {X,Y} of the list reproduced in the following Table 4.

Table 4

X	Y	X	Y	X	Y	X	Y
-4.16	12.80	-2.84	14.03	-2.52	16.15	-1.54	18.41
-4.02	12.86	-2.75	14.33	-2.46	16.41	-1.38	18.57
-3.89	12.92	-2.73	14.44	-2.39	16.66	-1.19	18.72
-3.70	13.03	-2.70	14.65	-2.30	16.92	-0.99	18.83
-3.52	13.15	-2.69	14.75	-2.20	17.16	-0.78	18.93
-3.41	13.24	-2.68	14.96	-2.09	17.40	-0.56	19.00
-3.25	13.38	-2.67	15.19	-1.97	17.64	-0.34	19.06
-3.12	13.53	-2.65	15.37	-1.86	17.88	-0.12	19.07
-3.01	13.68	-2.61	15.63	-1.79	18.02	0.00	19.08
-2.92	13.83	-2.56	15.89	-1.67	18.22		

[0033] Once the centre distance between the meshing gears of the positive-displacement pump, or one of the characteristic circles of the gears, for example the pitch diameter or the head diameter, is known or defined, it is possible to obtain the coordinate values {X',Y'} from the pairs of values {X,Y} mentioned above using simple transformation calculations. Representative values of points of the gear tooth profiles are thus obtained and can be used in conjunction with a machine for cutting gears of known type, in particular for controlling the trajectory of a tool of a numerically controlled machine.

[0034] The production tolerance (and the design) of the gears must be such as to ensure that the profile of the cut teeth is within a tolerance band of ± 1/15, more preferably 1/20, and even more preferably ±1/30 of the height of the gear tooth.

[0035] Naturally, the principle of the invention remaining the same, the details of construction and the forms of embodiment may be varied widely with respect to those described and illustrated without thereby departing from the scope of the present invention.

Claims

1. A gear having a plurality of teeth suitable for meshing with the teeth of another, corresponding, gear, **characterized in that** the profile of each tooth falls within a tolerance band of t 1/15 of the height of the tooth of the gear with respect to a theoretical profile homothetic to a profile defined by a spline function passing through a plurality of node points which have predetermined coordinates {X,Y} expressed in a system of cartesian coordinates having their origin on the axis of rotation of the gear, and which correspond to Tables 1 to 4, also shown hereinafter, for gears having a number of teeth equal to seven, eight, nine and ten, respectively:

Table 1

X	Y	X	Y	X	Y	X	Y
-5.29	10.99	-3.52	13.62	-3.36	15.79	-2.09	18.28

(continued)

X	Y	X	Y	X	Y	X	Y
-4.94	11.21	-3.51	13.84	-3.30	16.04	-1.79	18.51
-4.71	11.37	-3.52	14.06	-3.21	16.38	-1.46	18.70
-4.49	11.54	-3.55	14.35	-3.13	16.62	-0.93	18.92
-4.28	11.74	-3.56	14.61	-3.06	16.79	-0.75	18.98
-4.10	11.98	-3.55	14.78	-3.00	16.94	-0.57	19.03
-3.94	12.24	-3.54	14.95	-2.93	17.09	-0.38	19.06
-3.81	12.53	-3.51	15.12	-2.76	17.41	-0.19	19.07
-3.69	12.86	-3.44	15.46	-2.56	17.41	0.00	19.08
-3.58	13.25	-3.40	15.63	-2.35	18.01		

Table 2

X	Y	X	Y	X	Y	X	Y
0.00	19.08	2.42	17.52	3.07	15.62	3.45	13.12
0.30	19.06	2.53	17.26	3.10	15.44	3.54	12.94
0.61	19.01	2.60	17.09	3.13	15.26	3.70	12.68
0.91	18.93	2.66	16.92	3.17	14.99	3.86	12.45
1.20	18.81	2.73	16.75	3.19	14.81	4.05	12.24
1.46	18.64	2.84	16.50	3.20	14.63	4.28	12.06
1.70	18.44	2.90	16.33	3.20	13.99	4.66	11.84
1.91	18.23	2.96	16.15	3.23	13.76	4.86	11.72
2.11	18.01	3.00	15.98	3.29	13.53		
2.29	17.77	3.04	15.80	3.37	13.29		

Table 3

X	Y	X	Y	X	Y	X	Y
-4.47	12.27	-2.91	14.38	-2.68	16.34	-1.82	18.22
-4.34	12.33	-2.89	14.57	-2.62	16.51	-1.64	18.41
-4.09	12.47	-2.89	14.76	-2.55	16.68	-1.44	18.58
-3.85	12.62	-2.88	15.08	-2.48	16.85	-1.22	18.73
-3.64	12.79	-2.86	15.26	-2.41	17.02	-1.00	18.86
-3.45	12.98	-2.85	15.44	-2.34	17.19	-0.77	18.97
-3.19	13.37	-2.83	15.62	-2.28	17.36	-0.52	19.05
-3.03	13.77	-2.80	15.80	-2.21	17.53	-0.26	19.06
-2.98	13.96	-2.77	15.98	-2.13	17.70	0.00	19.08
-2.95	14.14	-2.73	16.16	-1.97	18.01		

Table 4

X	Y	X	Y	X	Y	X	Y
-4.16	12.80	-2.84	14.03	-2.52	16.15	-1.54	18.41
-4.02	12.86	-2.75	14.33	-2.46	16.41	-1.38	18.57
-3.89	12.92	-2.73	14.44	-2.39	16.66	-1.19	18.72
-3.70	13.03	-2.70	14.65	-2.30	16.92	-0.99	18.83
-3.52	13.15	-2.69	14.75	-2.20	17.16	-0.78	18.93
-3.41	13.24	-2.68	14.96	-2.09	17.40	-0.56	19.00

(continued)

X	Y	X	Y	X	Y	X	Y
-3.25	13.38	-2.67	15.19	-1.97	17.64	-0.34	19.06
-3.12	13.53	-2.65	15.37	-1.86	17.88	-0.12	19.07
-3.01	13.68	-2.61	15.63	-1.79	18.02	0.00	19.08
-2.92	13.83	-2.56	15.89	-1.67	18.22		

2. A gear according to claim 1, wherein the tolerance band is ± 1/20 of the height of the tooth.

3. A gear according to claim 1, wherein the tolerance band is ± 1/30 of the height of the tooth.

4. A gear according to any one of the preceding claims, **characterized in that** the spline function is a natural cubic spline function.

5. A gear according to any one of the preceding claims, wherein the gear has helical toothing.

6. A gear according to claim 5, wherein the face contact ratio of the helical toothing is from 0.4 to 1.2, preferably from 0.5 to 1.2, more preferably from 0.6 to 1.2, more preferably from 0.7 to 1.1, more preferably from 0.8 to 1.1, and even more preferably from 0.9 to 1.

7. A gear according to claim 6, wherein the face contact ratio of the helical toothing is equal to or close to unity.

8. A gear according to any one of the preceding claims, wherein the gear has a ratio of the sizes of the face width and the pitch diameter of from 0.5 to 2, more preferably from 0.6 to 1.8, even more preferably from 0.65 to 1.5, and even more preferably from 0.7 to 1.25.

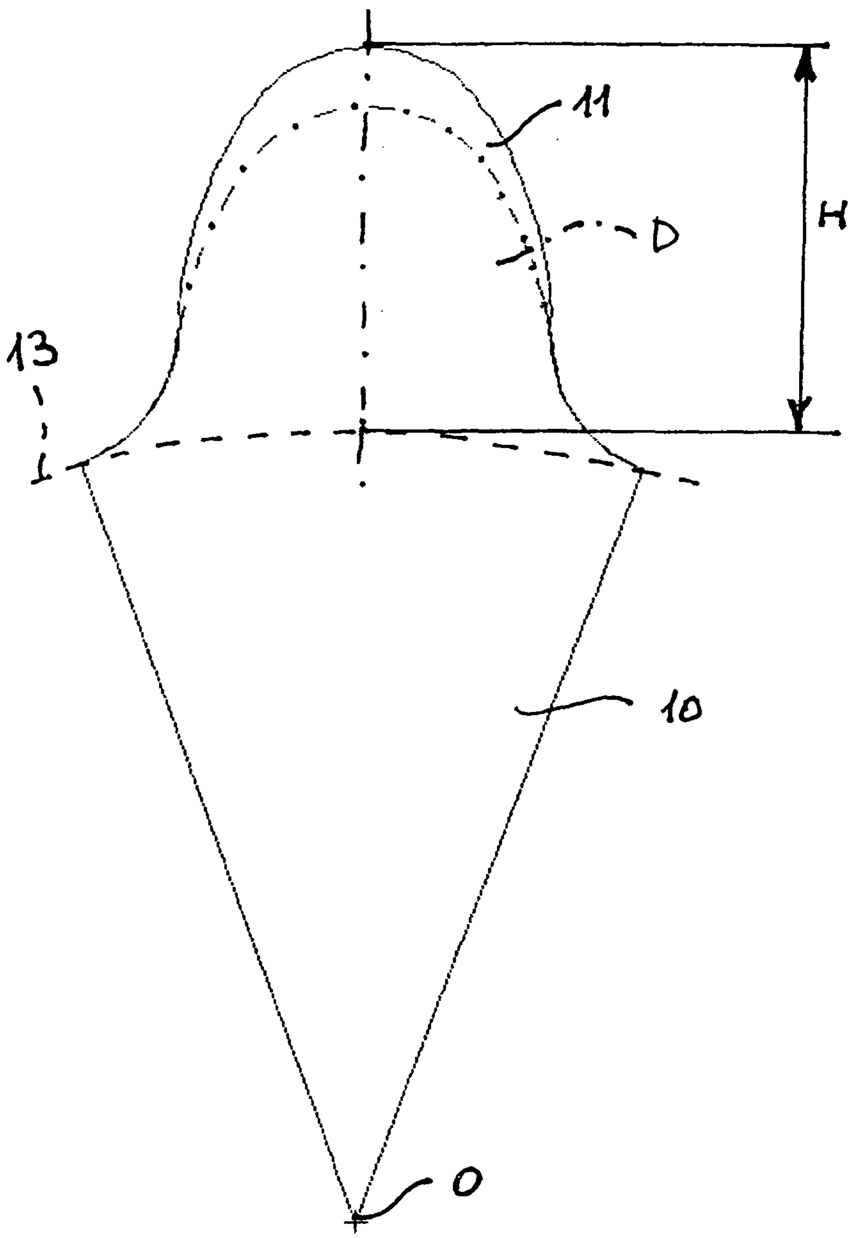
9. A gear according to any one of the preceding claims, wherein the gear has a ratio of the sizes of the face width and the pitch diameter of close to unity.

10. A hydraulic gear apparatus, **characterized in that** it comprises two gears according to any one of the preceding claims, the gears meshing with each other with semi-encapsulation.

11. A hydraulic apparatus according to claim 10, wherein the hydraulic apparatus is a hydraulic pump.

12. A hydraulic apparatus according to claim 10, wherein the hydraulic apparatus is a hydraulic motor.

1/1



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

- EP 0769104 A **[0006]**
- US 2159744 A **[0008]**
- US 3164099 A **[0008]**
- US 3209611 A **[0008]**
- EP 1132618 A **[0008]**
- EP 1371848 B **[0008]**
- US 6769891 B **[0008]**
- WO 2008111017 A **[0010] [0011]**

Non-patent literature cited in the description

- **MORSELLI Mario Antonio.** Rumorosità meccanica e idraulica delle pompe ad ingranaggi. *Oleodinamica Pneumatica,* January 2005, 54-59 **[0003]**
- *Fluides & Transmissions,* April 2005, (75), 34-3 7 **[0003]**