Worm-Type Gear with Steel Gearwheel

Evgenii S. Trubachev

Abstract The manuscript presents a study of a variety of a worm gear having a truncated gear rim as compared to the common worm gear. This gear has been originally invented by Egorov and Iofik and, as shown here, it has a number of favorable geometrical and kinematic, power, strength, layout, and manufacturing properties. It is offered to name it "QN-gear" and to use hardened steel as the material of its gear rim that sharply raises its strength. The quantitative and qualitative comparison with the nearest analogues—worm and spiroid gears has shown the perspective of applying such a gear design, first of all, for the case of low rotation speeds. Equations are obtained for calculation of the worm axial module and dependences for prevention of tooth undercutting and sharpening, which are, in fact, limitations at optimization design. Proposals are made to improve the performance and production characteristics of QN-gears with hardened steel gearwheel rims by gear optimization and contact localization. To improve the economic efficiency of gear production, it is proposed to produce gearwheels with curvilinear and asymmetric profiles by means of standard involute hobs and multi-cutter running heads. An example of a gearbox based on the steel QN-gear is given showing high load characteristics in testing.

Keywords Worm gear · Spiroid gear · Load capacity · Strength

1 Introduction

The problem of applying hardened steel to produce worm gearwheels [1] is one of promising and nowadays insufficiently used possibilities to improve the durability and economic performance of worm gears. The preventing property that forces to use antifriction materials enabling relatively fast running-in is the poor contact conditions in the pitch point of a cylindrical worm gear, because this point is always a regular node one [2]; the speed of its movement along the surface of the worm thread is equal to zero, the contact line is located extremely unfavorably, the angle between its

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tangent and the vector $\mathbf{v}_s$ of the relative speed is also almost equal to zero. A radical way to eliminate this disadvantage is known—to remove a part of the gearwheel rim adjacent to the pitch point. Perhaps, Grubin and Lytskhier were the first Russian (Soviet) researchers to propose this method [3]—see Fig. 1.

This idea was developed in gears, which were named by authors (Egorov and Iofik) as cylindrical worm ones [4, 5] having larger gearwheel diameters in comparison with the Grubin-Litskhier gearwheel—see Fig. 2.

The gear version in accordance with the scheme in Fig. 2a has exterior benefits: the total width of the gear rim is greater, respectively, the total length of contact lines is greater, and the load transmitted by their segments is lower. However, there still remain two other disadvantages in this version of the worm gear:

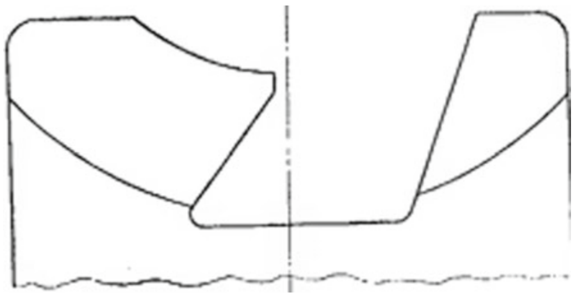


Fig. 1 The axial profile of the Grubin-Litskhier worm gearwheel

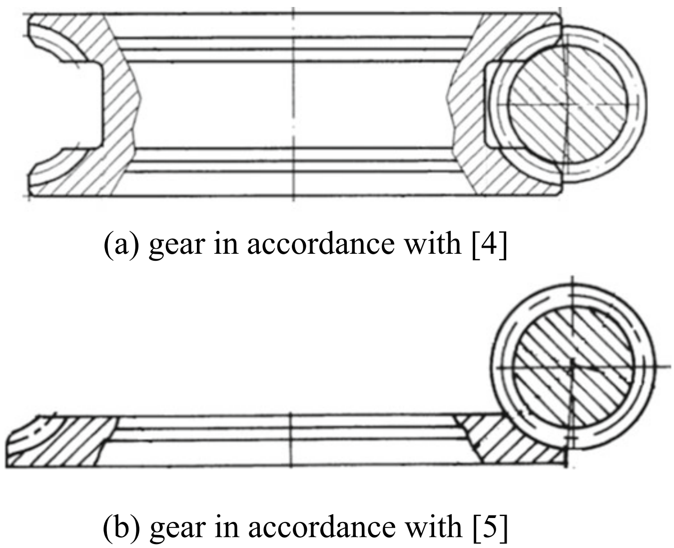


Fig. 2 Worm gears by Egorov and Iofik with the removed central part of the gearwheel rim

– the repeated contact of worm threads [6], which causes their higher heat load;
– the high sensitivity of the gear to the error of the gearwheel axial arrangement.

Probably, it is caused by the fact that at practical implementation of this gear in gearboxes for general purpose mechanical engineering [7] its gearwheel, like a gearwheel of a common worm gear, was made of bronze—the antifriction material that provides fast running-in.

In [8], a version of solving the problem of applying hardened steel for worm gears is presented for a non-orthogonal worm gear without the so-called meshing axes, and the result of practical implementation of this solution in a serially produced low-speed heavy-loaded gearbox for pipeline valves is shown. However, the non-orthogonal arrangement of axes is not a "great pleasure" for production. Therefore, the version for the scheme in Fig. 3b seems to become an effective solution to the above problem at the orthogonal arrangement of axes, because it provides:

– elimination of the repeated contact;
– possibility of a simple and easily controlled contact localization along the tooth length;
– lower sensitivity to the action of errors.

Exactly this version is a subject of consideration in this manuscript, particularly, the features of selecting gear parameters, obtained gear properties in comparison with its analogues, and results of practical cutting and testing. The presented results confirming and developing the initial conclusions made by authors of the gear are proving the gear originality and significant difference from its analogues. Therefore,

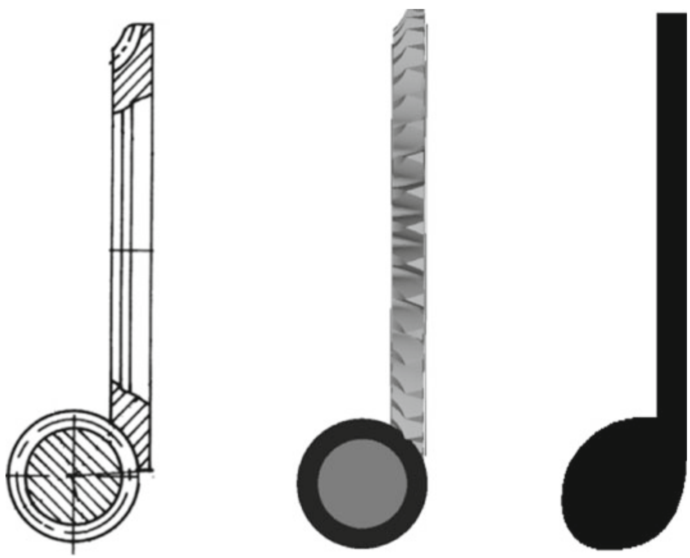


Fig. 3 QN-gear and the note "Quarter note"

it seems that the gear is worth giving its own name. Further in the manuscript the name “QN-gear” will be used for short (Quarter-note gear—see Fig. 3).

2 Some General Comments

The presence of the meshing axis in the meshing zone (in the orthogonal unshifted worm cylindrical gear it passes parallel to the gearwheel axis contacting the pitch cylinder of the worm) forces the contact lines to draw along teeth; this situation is fundamentally kept for almost any profile of the worm and gear ratio [9]. Figure 4a shows as an example the lines of the conjugated contact for an orthogonal Archimedean worm gear with the gear ratio equal to 40 and interaxial distance equal to 100 mm (the lines are extended beyond the meshing zone to demonstrate their shape evolution). It can be seen that as you move away from the middle plane of the gearwheel (plane $y = 0$), angles between the vector of relative velocity (it practically coincides with the circumferential direction of the worm) and the tangent to the contact line are increasing. This well-known fact is the main motive for removal of the central part of the gear rim. Having left, in fact, one third of the gear rim, Egorov and Iofik, developed the gearwheel for greater use of the meshing surface area with favorable properties—they moved the face of the gearwheel towards greater values of y coordinate, increased the maximum diameter, and introduced a negative shift of the worm (Fig. 4b). These techniques allow to increase the overlap factor and, as a consequence, to compensate the negative effect of the acquired reduction of the worm gearwheel face width. In accordance with the classification by Georgiev and Goldfarb [10], the worm gear is thus transferred to a higher class, and its gearing zone is shifted relative to the interaxial line along the gearwheel axis.

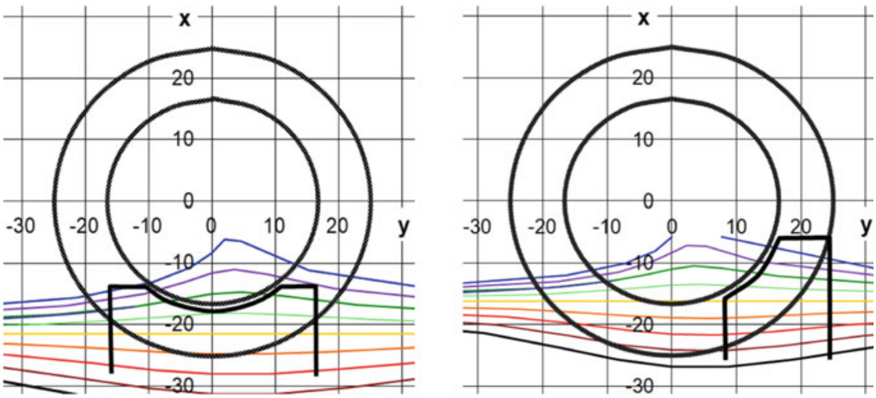


Fig. 4 Meshing surfaces of worm gears with Archimedes worms

3 Gear Scheme

As it is usually done [11, 12], it is reasonable to start the calculation of the QN-gear by choosing the gear scheme, i.e. by defining the pitch surfaces. The first question here is: which cylinder of the worm is reasonable to be chosen as the initial one? In the case of common worm cylindrical gears it is the pitch cylinder, and for spiroid ones it is the outer cylinder, because in this case:

- greater completeness of contact is provided (contact lines are stretched along the worm profile);
- conditions of tooth undercutting are adequately evaluated.

In the QN-gear the situation is almost the same: contact lines can be evaluated from Fig. 4; and undercutting, as it will be shown below, is one of the restrictions for parameter selection. Therefore, it is suggested to choose the outer cylinder with a diameter of d_{a1} as the worm initial cylinder in the QN-gear. The initial surface of the gearwheel in the orthogonal case will be a torus surface contacting with this cylinder and limited by two end faces, the position of which is set by the parameters B_2 and b_2—see Fig. 5, while contacting of initial surfaces takes place in the plane $z = 0$ along the arc of the circumference $x^2 + y^2 = (0.5d_{a1})^2$.

Reasons for selecting the worm diameter of the QN-gear are almost the same as for a conventional worm cylindrical gear. The issue of optimal ratio of the gearwheel rim displacement from the interaxial line, the width and the largest diameter of the gearwheel rim remains open in many respects and may depend, for example, on the required ratio of tooth contact and bending strength. Based on the experience in gear

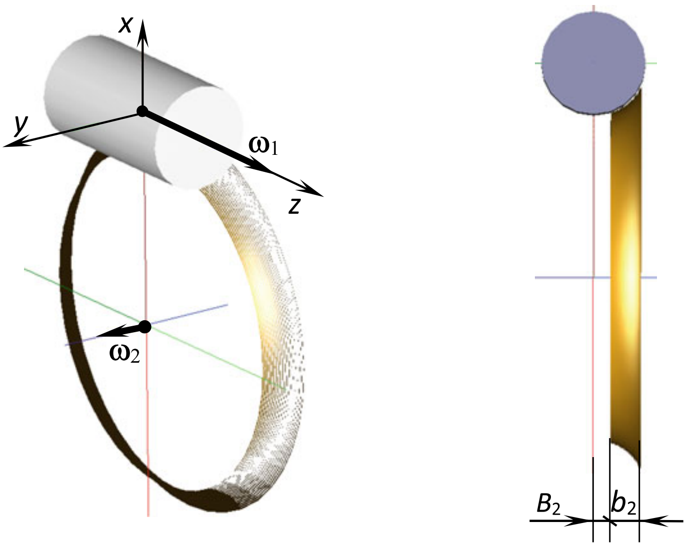


Fig. 5 Initial surfaces and scheme of QN-gear

design and testing, it is possible to accept:

$$B_2 = 0.15d_{a1}. \tag{1}$$

It allows providing the minimum angle between the tangent to a contact line and the vector of relative speed not <11°…15°, and the average one—bout 40°…45°, and it is well agreed with this recommendation in [4, 5] (0.2…0.0.4 from outer radius of the worm). For obvious reasons, the following inequalities should also be kept:

$$(B_2 + b_2) \to (\leq)0.5d_{a1}, \tag{2}$$

$$d_{M2} \to (\leq)2\left[a_w - \sqrt{(0.5d_{a1})^2 - (B_2 + b_2)^2}\right] \tag{3}$$

Selection of the worm axial module m_x should be made from the condition of collinearity of the helical line and the relative velocity vector, in the general case it gives the expression [13]:

$$m_{x\ id} = \frac{2}{z_{(1)}} \frac{(0.5d_{a1})^2(x + a_w)\sin\Sigma}{[(0.5d_{a1})^2(i - \cos\Sigma) - zy\sin\Sigma - xa_w\cos\Sigma]}, \tag{4}$$

where $z_{(1)}$ is the number of worm threads; a_w, Σ are the distance and angle between the axes; i is the gear ratio; x, y, z are the Cartesian coordinates of the point of contact of the pitch surfaces in the coordinate system shown in the Fig. 5.

In the orthogonal case:

$$m_{x\ id} = \frac{2(x + a_w)}{z_{(2)}} \tag{5}$$

or—for the convenience of correlating the variable coordinate with the assigned parameters B_2 and b_2 and taking into account $x^2 + y^2 = (0.5d_{a1})^2$:

$$m_{x\ id} = \frac{2\left(a_w - \sqrt{(0.5d_{a1})^2 - y^2}\right)}{z_{(2)}} \tag{6}$$

As you can see, the value of $m_{x\,id}$ depends on the selection of the point position on the line of the initial surfaces contact (x or y coordinates).

4 Restrictions of Parameter Selection—Tooth Undercut and Sharpening

The practice of calculations shows that the QN-gear is characterized by undercutting of the 1st type (Fig. 6) caused by the appearance of the so-called singular line of the enveloping surface in the meshing zone. It is the geometrical place of singular points of the enveloping surface. To predict and prevent this phenomenon at the stage of selecting parameters of the gear scheme, we use the method and algorithm thoroughly described in [14]. The main analytical expression here is a system of equations solved at points of the worm pitch surface:

$$\begin{cases} \mathbf{n}\mathbf{v}_s = F_m(x, y, z) = 0, \\ \mathbf{N}\mathbf{v}_2 = F_u(x, y, z) = 0. \end{cases} \tag{7}$$

Here, the first equation is the meshing equation ($\mathbf{n}(x, y, z, p_\gamma, \alpha_{lim})$ is the limiting contact normal, $p_\gamma = 0.5m_x z_{(1)}$ is the helical parameter of the worm, α_{lim} is the sought limiting axial angle of the worm profile), and the second one is the condition of intersection of the normal $\mathbf{N} = \{\partial F_m/\partial x, \partial F_m/\partial y, \partial F_m/\partial z\}$ to the meshing surface with the gearwheel axis ($\mathbf{v}_2$ is the speed of the contact point at joint motion with the gearwheel).

The solution (α_{lim}) for which the mentioned singular points and, consequently, the tooth undercut appear in the meshing is usually not single. The resulting set of solutions is used to select maximum and minimum values, and the profile angles of the right and left flanks of worm threads should be selected as their largest and smallest (taking into account the sign) values, respectively. The main factors that affect the values of angles α_{lim} are:

- removal of the gearwheel B_2;
- worm diameter d_{a1};
- worm axial module m_x;
- gear ratio i.

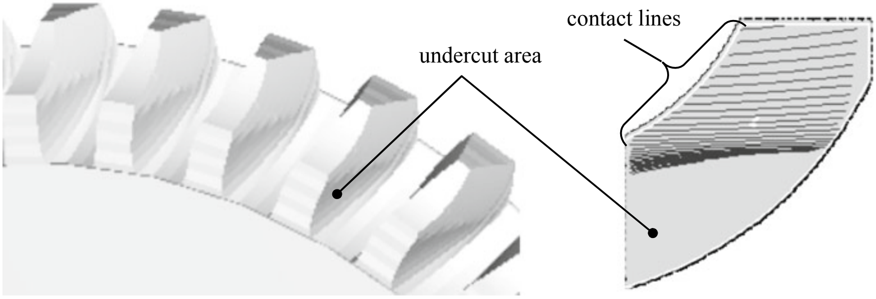
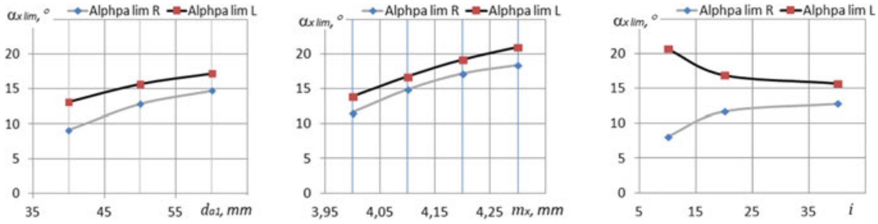


Fig. 6 Undercutting of gearwheel teeth in the QN-gear



(a) worm diameter (b) worm axial module (c) gear ratio

Fig. 7 Main factors of tooth undercutting in the QN-gear

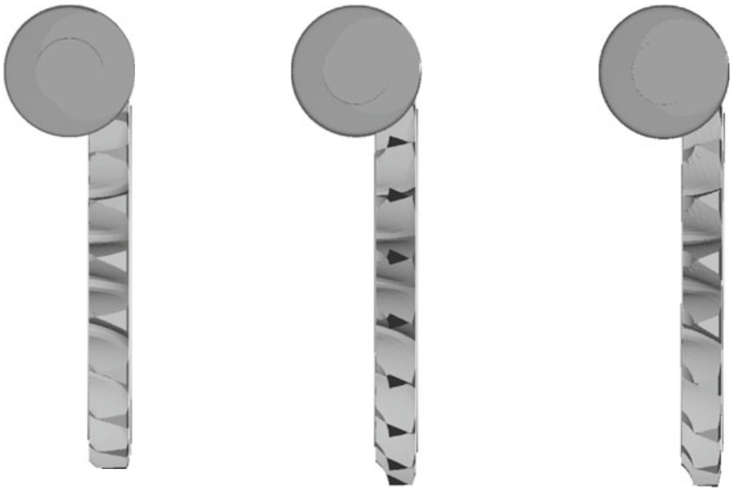
Figure 7 shows the dependencies of limiting angles on the last three factors (B_2, taken to be equal to $0.15d_{a1}$) for the orthogonal gear with the interaxial distance of 100 mm. Evidently, the risk of undercutting goes up with increasing the worm diameter, its axial module and decreasing the gear ratio, and it is higher for one of the flanks (the left one facing the positive values of z coordinate—for the coordinate system shown in Fig. 6). Considering this fact, one can recommend the introduction of an asymmetric worm thread profile to avoid undercutting if necessary, and the less the gear ratio is, the larger the asymmetry should be.

The second limitation is the tooth sharpening that occurs at the intersection of the torus and cylindrical sections of the tooth vertex edge. Of course, this sharpening can be eliminated by an additional chamfer on this tooth vertex, as it is shown in the original patent [4, 5]. However, this technique reduces the area of contact surfaces and the length of contact lines, so we will consider further how it can be done by choosing the gear parameters. The main influencing factors here are as follows:

– the thickness s_x of the worm thread;
– the axial module m_x of the worm;
– the largest diameter d_{M2} of the gearwheel.

Due to the obvious influence of the first of the factors, the main interest is to improve the gear by two latter ones. And here lies one of the main contradictions for the designing engineer: on the one hand, the gearwheel diameter should be increased to use the gear possibilities more completely—to increase the overlap factor and length of contact lines, and to reduce contact stresses. On the other hand, the risk of tooth sharpening is increased in this case (in Fig. 8b it can be recognized by the intersection of opposite flanks). The general thoughts here are as follows:

– to increase the diameter d_{M2} one should choose larger values m_x (larger values y in (6)), see Fig. 8c;
– this increases the risk of tooth undercutting (see Fig. 7b) which may require an increase in axial profile angles above the commonly accepted value of 20°;
– there is some limit of the gear diameter d_{M2} in terms of increasing the gear performance.



a) no sharpening   b) d_{M2} is increased – the sharpening is present   c) d_{M2} and m_{x0} are increased – no sharpening

Fig. 8 Tooth undercutting sharpening in the QN-gear (please, replace the word "undercutting"with "sharpening"

In summary, one can recommend the range $d_{M2} = (1.8\ldots1.9)a_w$ and the value $y = (0.30\ldots0.43)d_{a1}$ to calculate the axial worm module by (6) which results in the expression:

$$m_x = \frac{2a_w - (0.5\ldots0.8)d_{a1}}{z_{(2)}} \tag{8}$$

The values m_x obtained by this expression turn to be a little larger than worm modules for the common worm cylindrical gear with the same gear ratio, inter-axial distance and worm diameter, and they proximately correspond to the negative coefficients of its worm shift in the range –0.5…–2.5. This increase of the module (reduction of the shift coefficient) leads to an increase in the risk of undercutting and reduction of contact stresses in the meshing.

5 Contact Localization

The manuscript's title focuses on the fact that it is reasonable to use a steel gearwheel subjected to thermal or chemical thermal hardening treatment in the QN-gear. This, to our opinion, is not obligatory, but gives a better implementation of the gear potential in terms of tooth strength and economy. In this regard, the issue of providing the contact localization is more relevant than for common worm gears with gearwheels allowing for relatively fast running-in.

In conventional worm gears, the contact is usually localized in the central area of the tooth flank with adverse meshing conditions. For gears shown in Fig. 3a and b, the localized contact can only be achieved by cutting each of the half-rim individually (at least if the traditional method of forming by the generating worm is used): each rim should have the assigned design point and properly chosen number of machine-tool settings. In general, this results in at least two different sets and two gear machining operations. In this context, the QN-gear has the advantage: contact localization along both flanks can be achieved per one setting of the cutting tool. For this purpose, the method is applicable which has been first proposed in [15] and stated in detail in [16] and which involves the following basic steps:

– selection of design points on opposite tooth flanks;
– specifying a number of machine-tool setting parameters—the machine-tool interaxial angle, number of threads, axial module and the radii of the curvature of the generating worm profile;
– calculation of the remaining parameters—the machine-tool interaxial distance, diameter, angles and thicknesses of the generating worm profile—for conditions assigned at the design points [15, 17]:

$$\begin{aligned} \mathbf{n}\mathbf{v}_{01} &= 0 \\ \tan\ \gamma &= -\frac{\mathbf{n}\mathbf{e}_t}{\mathbf{n}\mathbf{k}}, \\ \tan\ \alpha_x &= -\frac{\mathbf{n}\mathbf{e}_r}{\mathbf{n}\mathbf{k}}, \end{aligned} \tag{9}$$

where $\mathbf{v}_{01}$ is the velocity of the generating worm relative to the operating worm in their joint virtual meshing at the design point, γ is the helix angle, $\mathbf{e}_t$, $\mathbf{e}_r$ and $\mathbf{k}$ are unit vectors of circumferential, radial and axial directions with respect to the axis of the generating worm.

Contact localization by the tooth height (profile) is provided mainly by the selection of the proper ratio of profile curvatures for the operating and generating worm, and by the length—by the selection of the module and the number of threads for the generating worm, and the machine-tool angle.

Similar to common worm gears, it is reasonable to provide contact localization in QN-gears by applying:

– a single- or double-thread hob (in particular, a standard involute hob) [18];
– an assembled running-in tool cutter head with 2, 3 or 4 carbide cutters [16]

The question still remains: What is the best place for contact localization—placing the design point in the middle of the tooth or shifting it to one of its face ends, apex or root? The answer to this question should take into account the asymmetry of the modification field (Fig. 11), gearwheel deformations during heat treatment and under load in the gearbox, and the necessity to provide the preferable initial load concentration within the area of more favorable contact conditions.

6 Comparison of the QN-Gear with Classical Worm and Spiroid Gears

The QN-gear can be competitive within a wide range of operating conditions, but if we talk about the gear with a steel gearwheel, to our opinion, the most promising implementable application here is the case of low speeds and heavy loads. Figure 9

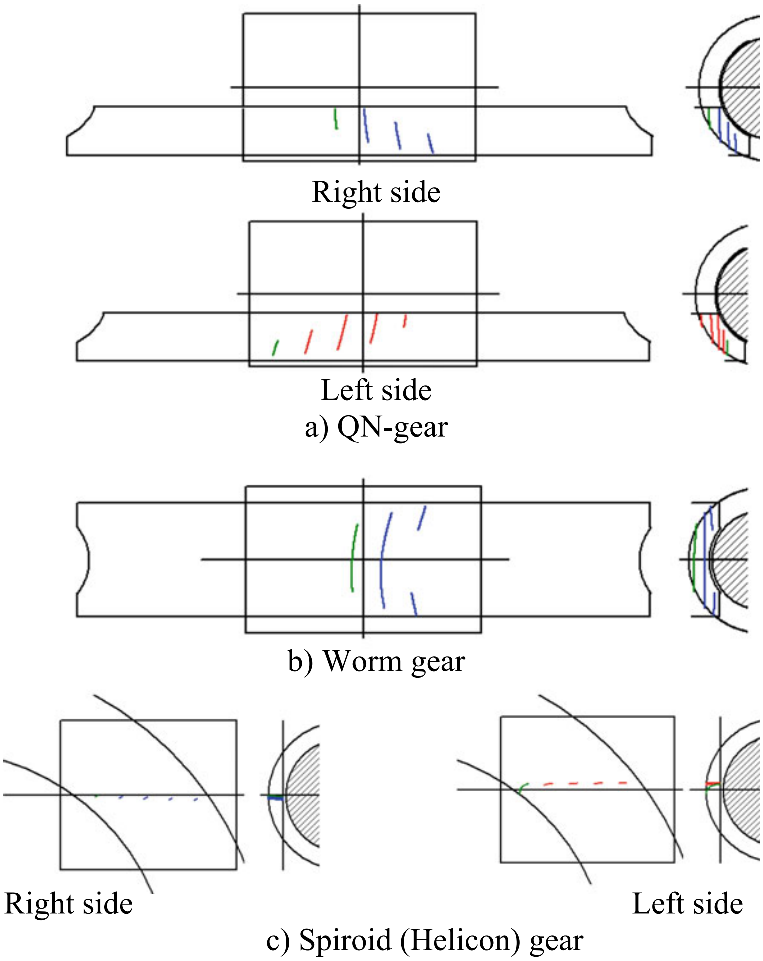


Fig. 9 Lines of conjugated contact in compared gears

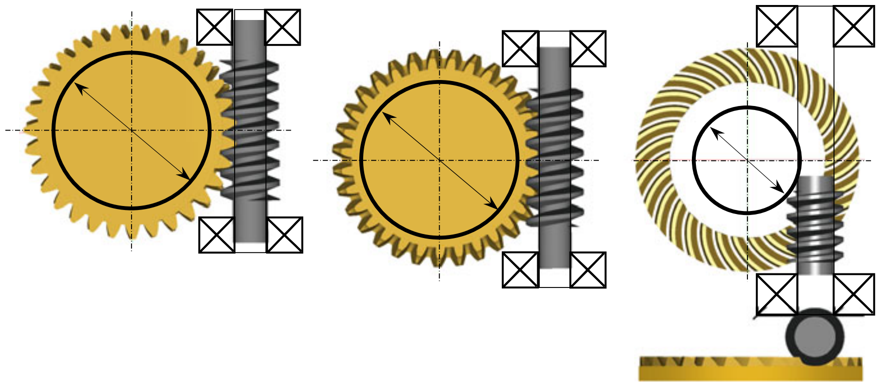
and Table 1 show some results for such a gear as compared with analogues of the same dimension (plan view areas taken by gears are said to be equal)—classical worm cylindrical and spiroid gears. Calculations are made by means of the program complex "SPDIAL+" [12]. Pairs of assessments are given for the QN-gear and spiroid one: for right flanks—in numerator, for left flanks—in denominator.

The main conclusions of gear comparison are as follows:

1. Indicators that determine the gear resistance to scuffing.

    1.1 Contact stresses in the classical worm gear are lower than in the QN-gear and spiroid one (right flanks are meshing that are usually chosen as the main operating ones due to higher efficiency, lower forces that deform the structure and cause the stress concentration; see also [19]).

    1.2 Gears are comparable in the sliding speed: at a relatively large gear ratio, its main component—the worm speed—linearly depends on the diameter of the latter, and it is selected almost equal in all three gears.



a) worm gear b) QN-gear c) spiroid gear

Fig. 10 To comparison of layout features of gears

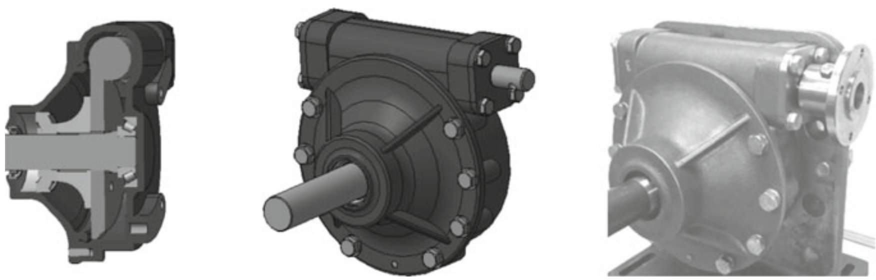


Fig. 11 The gearbox based on the QN-gear

Table 1 Main parameters and performance of compared gears

		QN-gear	Worm gear	Spiroid gear
Frequency of worm rotation, rev/min		35		
Torque at the gearwheel, Nm		5000		
Gear ratio		50		
Interaxial distance, mm		105	110	60
Maximum diameter of the gearwheel, mm		195	191	211
Outer diameter of the worm, mm		50	50	48
Axial module of the worm, mm		3.48	3.55	2.50
Gearwheel face width, mm		16	38	30.5
Overlap factor		4.2/4.7	2.75	5.7/5.8
Total length of contact lines, mm		30.3/35ю6	55.9	25.3/29ю1
Sliding speed, m/s		0.082	0.082	0.074
Average angle between the tangent to the contact line and the vector of relative velocity, °		43/38	23	82/77
Average speed of contact points displacement, m/s	Worm	0.051/0.050	0.031	0.078/0.076
	Gearwheel	0.0023/0.0018	0.0023	0.0047
Contact stresses, MPa (maximum in phases)		1600/1700	1150	1650/850
Efficiency (friction coefficient 0.1)		0.420/0.425	0.428	0.402/0.370
Axial force at the worm, H		57,000	56,300	72,000
Axial force at the gearwheel, H		5800/21,300	9300	13,300/39,600
Sensitivity to the error of the gearwheel axial position		Low	High	Low
Possibility to adjust the backlash by the gearwheel axial displacement		Yes	No	Yes

1.3 The QN-gear and the classical worm one have a higher efficiency (see also [18]), which determines less heat generation in the contact.

1.4 With regard to the arrangement of contact lines and velocities of contact point displacement along flanks (related parameters), the QN-gear actually takes an intermediate position between the classical worm gear and the spiroid one; in this case, contact points with these parameters equal to zero are excluded at meshing zones of the QN-gear and spiroid one.

Therefore, in the QN-gear there are objective reasons to obtain a higher resistance to tooth scuffing than in the classical worm one, and when added to the reduced sensitivity to the action of errors, it is the argument for applying the hardened steel as the QN gearwheel material.

2. Parameters that determine the tooth strength.

   2.1 Worm axial modules for the QN-gear and classical worm one are almost one and a half as much as that for the spiroid worm.
   2.2 The overlap factor of the QN-gear and the spiroid one are considerably higher than that of the worm gear.
   2.3 Forces acting in the meshing of the QN-gear and the classical worm one are more than a quarter less than those acting in the spiroid gear (see also [19]).

   Therefore, there are objective reasons to obtain a higher resistance to tooth breakage in the QN-gear than in both other compared gears.

3. With regard to gear layout properties, the QN-gear also has an intermediate position between the classical worm and spiroid gears. Thus, the following is true for this gear (Fig. 10):

   3.1 Similar to the classic worm gear, there are better options for positioning a large central hole in the gearwheel than in the spiroid gear.
   3.2 Similar to the spiroid gearwheel, the QN-gearwheel is flatter than the classical worm gearwheel, which allows saving its cost, and, additionally, reducing the corresponding size of the casing.
   3.3 Worm bearings arranged on either side of the gearwheel can be placed closer to the meshing area than in the case of a spiroid gear, and their dimensions can be reduced due to lower meshing forces.
   3.4 Gearwheel bearings are also a little less loaded with axial forces than in the case of a spiroid gear.

4. The influence of the worm profile. An additional benefit can be obtained if the QN-gear worm profile is made asymmetrical and convex (i.e., the risk of undercutting is reduced). Table 1 shows the parameters of a symmetrical QN-gear with worm profile angles 18.2°, and contact stresses in meshing the left flanks are higher than in meshing of the right ones by 7% (1710 and 1600 MPa, respectively). If we make the profile asymmetrical and convex (profile angles are 23° and 20°, profile radii are 80 mm each), it is possible to equalize and reduce contact stresses to the value of 1580 MPa. Thus, along with the varying m_x, d_{M2}, B_2 the change of the worm thread profile for the QN-gear provides an effective tool for the gear optimization.

7 Example of Practical Implementation: Cutting and Testing Results

The QN-gear with the parameters listed in Table 1 was implemented in the low-speed (worm speed 35 rpm) gearbox[1] (Fig. 11) of the agricultural machine bogie

[1] The design is made by eng. A. I. Shutkina.

drive. The worm and the gearwheel were made of 40X steel with volume hardening to the hardness of HRC 45–50. A grease with the addition of 10% mass fraction of lubricated graphite and 5% molybdenum disulfide (MoS_2) was used in the gearbox. The contact in the gear was localized, and the gearwheel was cut with a standard involute hob with a normal module of 3.5 mm. The actual contact pattern generally showed a good match with the calculated one (Fig. 12). After a short (20 revolutions of the gearwheel in each direction) running-in with a gradual increase in the load, the gear was subjected to peak torques 5000 Nm acting during one quarter revolution at a relative duty factor of 25% within 20 cycles. Then a long testing was carried out (the total time was about 1000 h) for the output torque smoothly varied within 1200…2400 Nm with the period of 8 h. The contact pattern increased during the test, taking up finally most of the active part of the tooth flank. In the steady heat mode, the gearbox heating reached 68°…70°, with an average efficiency of 0.40–0.43 (the resulting test diagrams of the efficiency variation versus torque are shown in Fig. 13). Next, the gear was subjected to peak loads of 5300 Nm acting throughout

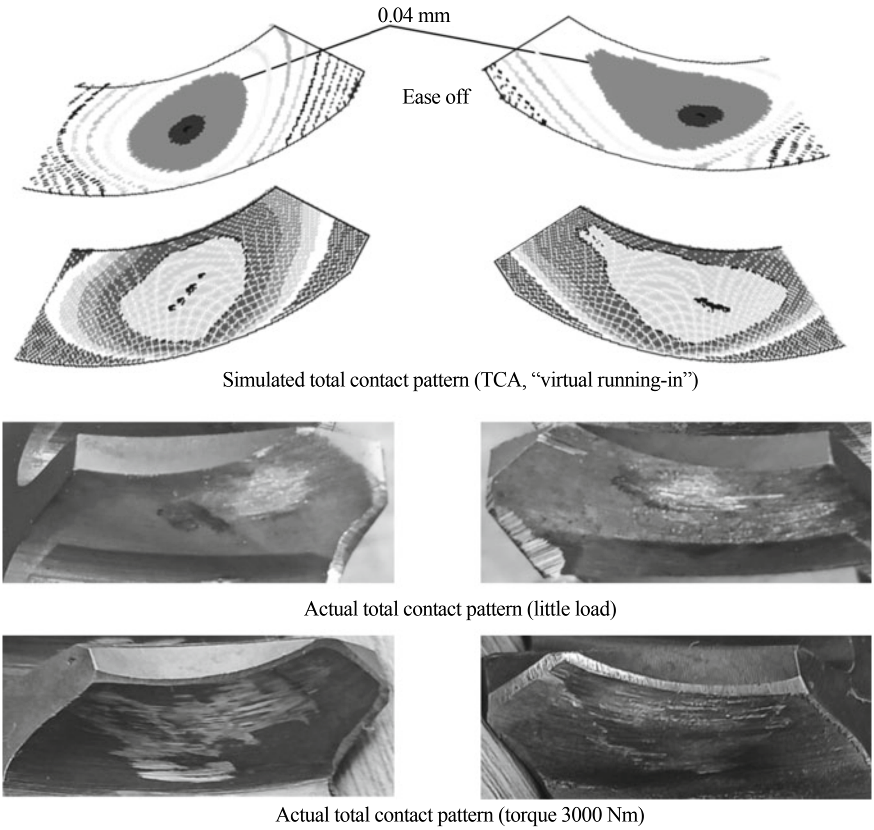


Fig. 12 Contact localization in the sample QN-gear

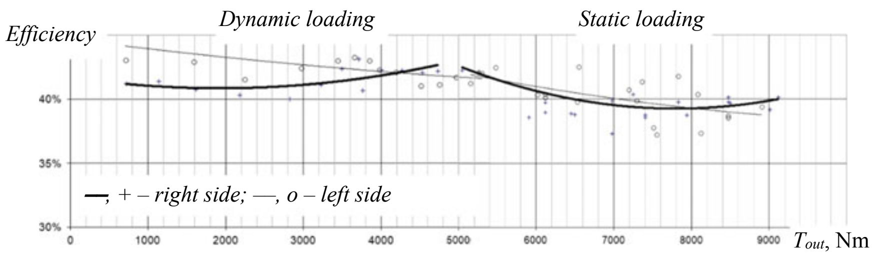


Fig. 13 Experimental values of efficiency for the gearbox with QN-gear

the gearwheel turning by 20°, with the total number of reversible loading of about 1000 cycles. No critical damage of the gear appeared. Finally, the gear was brought to failure at reversible static loading. The breakage occurred at 9000 Nm, and it was the gearbox shaft that fractured, mainly because of torsion stresses.

Of course, such a short first experience cannot be the basis for the calculated evaluation of the gear load capacity, but it has shown that the QN-gear with a steel gearwheel can become an effective alternative to the classical worm and spiroid ones. The obtained torques of 5300 Nm at short-term loading, and not <9000 Nm at static loading are not inferior (probably, surpass) to torques of the best samples of gearboxes for controlling the pipeline valves with similar dimensions [20–22].

8 Conclusion

The analysis made in the manuscript has shown the perspective application of a worm gear with a truncated gear rim made of hardened steel, which is proposed to be called "QN-gear". The perspective concerns, at least, the case of low rotational speeds, although this gear can be considered as an alternative to the classic worm and spiroid ones in other cases, as it combines many favorable geometric and kinematic, force, strength, layout, and manufacturing properties.

The obtained dependences for calculation of the worm axial module and dependences for prevention of tooth undercutting and sharpening are, in fact, limitations at the optimization design. Besides, in comparison with the basic solution of Egorov and Iofik, the following improvements allowing to increase operational and manufacturing characteristics of the gear are offered:

- to apply the hardened steel for manufacturing of QN-gearwheels, which provides the increased tooth strength;
- to go beyond worms with ruled surfaces and, moreover, apply a curvilinear and asymmetrical profile for gear optimization;
- to use standard involute hobs or multi-cutter running heads for gearwheel cutting to localize contact and increase the gear production efficiency.

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