Worm-Gearing Computer Design Algorithm

GHELASE DANIELA, DASCHIEVICI LUIZA
Faculty of Engineering Braila
"Dunarea de Jos" University
47, Domneasca St., Galati
ROMANIA

Abstract: - The paper presents an assisted design algorithm of the worm-gearing depending on the optimization criteria: medium rigidity criterion and rigidity variation criterion. The optimum dimensions of the gearing were determined using the two rigidity criteria, well suited to such problems especially because of their robustness and their ability to detect global extremes. The computerized algorithm can be adopted for any kind of worm-gearing and cylindrical or conical gears. To this end, the study presents the calculus examples for optimal design for worm-gearing.

Key-Words: - Optimal Design, Worm-gearing, Numerical Method, Amplitude, Rigidity, Computer Simulation.
Received: June 9, 2021. Revised: March 26, 2022. Accepted: April 21, 2022. Published: June 9, 2022.

1 Introduction

The proposed algorithm is based on research findings made by the authors of the paper :

- selection of a high value for diametral quotient q;
- increasing radius of profile curvature R;
- adoption of low value for worm profile angle α;
- reduction of number of gear teeth z_2.

b) from the viewpoint of the influences on the increase of medium rigidity, the hierarchy of the geometrical parameters is the following:

1) diametral quotient q;
2) radius of profile curvature R;
3) number of gear teeth z_2;
4) worm profile angle α.

c) The methodology regarding the achievement of amplitude as low as possible is the following:

- increasing number of gear teeth z_2;
- adoption of minimum diametral quotient q;
- increasing radius of profile curvature R;
- reduction of worm profile angle α.

d) The hierarchy of geometrical parameters, to achieve an amplitude as low as possible, is the following:

- radius of profile curvature has an independent value:

1) number of gear teeth z_2;
2) radius of profile curvature R;
3) profile angle α;
4) diametral quotient q.

- radius of profile curvature depends on axial module m_x:

1) radius of profile curvature R;
2) number of gear teeth z_2;
3) profile angle α;
4) diametral quotient q.

2 Design Algorithm

In case of the optimization using medium rigidity criterion, having as goal achievement of the medium

a) Generally, the methodology regarding the achievement of rigidity as high as possible is:

rigidity as high as possible for the worm-gearing tooth with circular profile (Fig. 2), we recommend the algorithm presented in Fig. 1.

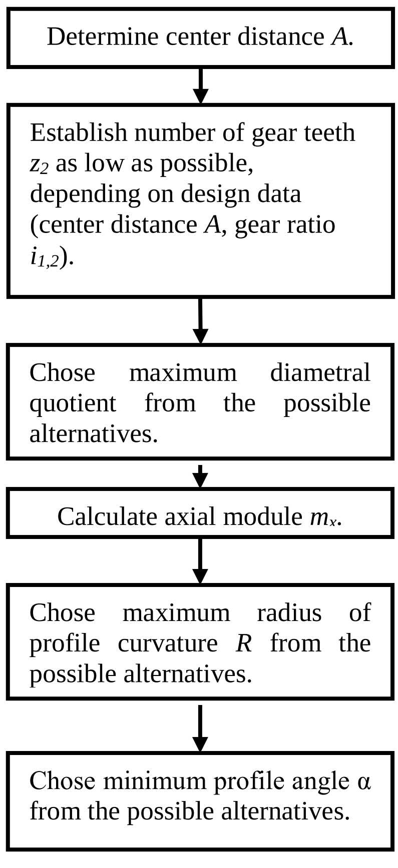

Fig. 1 Optimal design algorithm using medium rigidity criterion

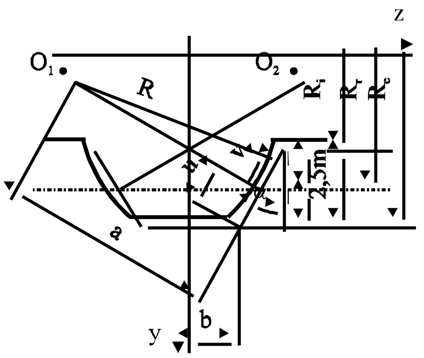

Fig. 2 Worm flank geometry

Figure 2 presents the axial section of the worm with constant pitch, having a circular arch profile with center O_1 for the right flank and O_2 for the left flank, where:.

$$
\begin{aligned}
& u = 1{,}25 \cdot m / \cos\alpha; \\
& p = m / 2; \\
& b = \pi \cdot m / 4 - 1{,}25 \cdot \mathrm{tg}\,\alpha \\
& R = \sqrt{a^2 + u^2}
\end{aligned} \qquad (1)
$$

To obtain a rigidity characteristic with amplitude as low as possible for the worm-gearing tooth with circular profile we recomand using the following algorithm:

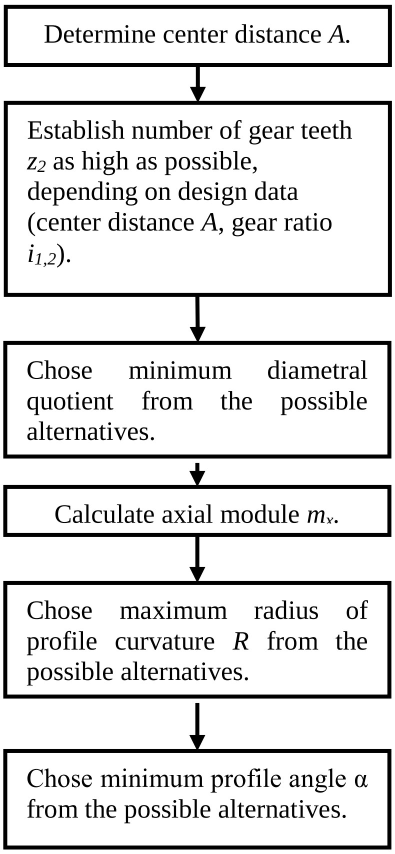

Fig. 3 Optimal design algorithm using rigidity variation criterion

3 Calculus Exemple For Optimal Design

3.1 Design of worm-gearing teeth with circular profile, having medium rigidity as high as possible

The known design data are:
- center distance A=315mm;
- gear ratio $i_{1,2}$=114/1.

So, results z_2=114.

We follow the algorithm established at **2.** (Fig. 1): Chose diametral quotient as high as possible q =16; Calculate axial module with the formula:

$$m_x = \frac{2 \cdot A}{q + z_2} \qquad (2)$$

$$m_x = \frac{2 \cdot 315}{16 + 114} = 4.846\text{mm}$$

The axial module is low enough, what doesn't help us to increasing rigidity. The design of worm-gearing imposes the adoption of standardized axial module. So, m_x=5mm.

Referring to the design data can't modify, we adopt q=12, what will attract a reduction of medium rigidity, from 2305.841kN/mm, what is in the case of q =16, to 1713.359kN/mm, the deference being 592.484kN/mm (Table 1).

Table 1. Influence of diametral quotient on rigidity

q	$R=3 \cdot m_x$ [mm]	Maximum Rigidity [kN/mm] for α=10°	Minimum Rigidity [kN/mm] for α=10°	Medium Rigidity [kN/mm] for α=10°
7	15.61	1087.027	647.259	867.143
8	15.49	1268.945	805.068	1037.007
9	15.36	1453.498	977.226	1125.362
10	15.24	1627.640	1114.246	1387.443
11	15.12	1792.178	1303.839	1548.008
12	15.00	1974.295	1452.423	1713.359
13	14.88	2136.667	1604.346	1870.006
14	14.76	2290.487	1759.527	2025.007
15	14.65	2414.695	1913.697	2164.196
16	14.53	2552.447	2059.235	2305.841

Another possibility would be, after the adopting of axial module m_x=5mm and diametral quotient q =16, the reduction of number of gear teeth z_2 (A=315mm), but the gear ratio deviation must be less than 3% ($\Delta i_{1,2}<$ 3%). So, if: m_x=5mm, q=16, z_2=110 $\Rightarrow$ $\Delta i_{1,2}$=3.5% (doesn't agree);

m_x=5mm, q=15, z_2=111 $\Rightarrow$ $\Delta i_{1,2}$=2.63%.

The last solution would allow to obtain higher rigidity than in the case of q=12 and z_2=114.

Table 2 Influence of diametral quotient and radius of profile curvature on amplitude

q	m_x [mm]	Amplitude of rigidity [kN/mm] for $R = 3 \cdot m_x$, $\alpha = 10°$	Amplitude of rigidity [kN/mm] for $R = 4 \cdot m_x$, $\alpha = 10°$	Amplitude of rigidity [kN/mm] for $R = 3 \cdot m_x$, $\alpha = 20°$	Amplitude of rigidity [kN/mm] for $R = 4 \cdot m_x$, $\alpha = 20°$
7	5.20	439.768	300.690	537.490	449.511
8	5.16	463.877	320.888	580.426	486.240
9	5.12	476.271	326.327	617.484	508.882
10	5.08	480.393	343.617	631.263	515.258
11	5.04	488.338	345.394	635.026	523.156
12	5	521.871	365.301	640.248	546.103
13	4.96	532.208	354.449	656.227	581.251
14	4.92	530.959	361.054	689.048	611.675
15	4.88	500.997	381.046	716.582	618.388
16	4.84	493.212	446.091	722.285	609.272

Don't forget about the possibility of the design of *x*-toothing gear drive (with negative or positive addendum modification).

Radius of profile curvature must be as large as possible, especially the value for axial module is low:

R=4m_x=4·4.846=19.38mm.

That is way, we recommend to adopt a larger radius of profile curvature, limited by the technological procedures of the circular profile manufacturing.The value for profile angle must be as low as possible, α=10°.

3.2 Design of worm-gearing teeth with circular profile, having constant rigidity

The known design data are:

- center distance A = 315 mm;
- gear ratio $i_{1,2}$ = 114/1.

So, results z_2 = 114.

We follow the algorithm established in this paper, at **2.** (Fig. 3):

Chose diametral quotient as low as possible, q = 7;

Calculate axial module:

$$m_x = 2A / (q + z_2) \tag{2}$$

$$m_x = 2 \cdot 315/(7 + 114) = 5.2066 \text{ mm}$$

We adopt standardized axial module, m_x = 5 mm.

If A = 315 mm and z_2 = 114, then diametral quotient becomes q = 12. Anticipating, this change of diametral quotient will attract the increasing of the rigidity amplitude (Table 2) from 439.768 kN/mm (q = 7, R = 3m_x) to 521.871 kN/mm (q = 12, R = 3m_x), the deference being 82.103 kN/mm.

If R = 4m_x, the amplitude increasing will be lower, from 300.690 kN/mm to 365.301 kN/mm (Table 2). In this case the deference is 64.61 kN/mm.

Another possibility would be, after the adopting of axial module m_x = 5 mm and diametral quotient q = 7, the increasing of number of gear teeth z_2 (A = 315 mm), but the gear ratio deviation must be less than 3% ($\Delta i_{1,2}$ < 3%).

So, if: m_x = 5 mm, q = 7, z_2 = 119 $\Longrightarrow$ $\Delta i_{1,2}$ = 4.3% (doesn't agree);

m_x = 5 mm, q = 8, z_2 = 118 $\Longrightarrow$ $\Delta i_{1,2}$ = 3.5%;

m_x = 5 mm, q = 9, z_2 = 117 $\Longrightarrow$ $\Delta i_{1,2}$ = 2.63%.

The last solution would allow to obtain higher reduction of the amplitude than in the case of q = 12 and z_2 = 114, because z_2 increases and q reduces.

Radius of profile curvature must be as large as possible, especially the value for axial module is low: R = 4m_x = 4·5.2066 = 20.82 mm.

We recommend that in the case of the low value for axial module, to adopt a larger radius of profile curvature, limited by the technological procedures of the circular profile manufacturing.

The value for profile angle must be as low as possible, α = 10°.

4 Conclusion

Based on the computerized simulation of the meshing and the influence of geometrical parameters on the rigidity of worm-gearing tooth, two algorithms have been developed for design of the worm gear drives:

- optimal design algorithm using medium rigidity criterion;
- optimal design algorithm using rigidity variation criterion.

The basic idea of the new approach to obtain high rigidity is to take into account the medium rigidity criterion and the rigidity variation criterion in the design phase.

The study presents the main steps of the proposed algorithms, which improve the performance of worm-gearing, and calculus examples.

References:

[1] D. Ghelase, *Rigidity of Worm-Gearing Tooth*, Ceprohart Publishing House, ISBN 973-85057-8-X, Brăila, 2002.
[2] N. Oancea, G. Frumuşanu, G. Dura, *Algorithms for Representation by Poles as a Way to Approximate Wrapping Curves Associated to Rolling Centrods*, Proceedings of ICMaS 2006, ,"Politehnica" University of Bucharest, pp. 319-322, 2006.
[3] D. Ghelase, L. Tomulescu, *Computerized Determination of the Elasticity of the Worm-Gearing Tooth for the Machine-Tools and Robots*, Proceeding of the “Machine-Building and Technosphere of the XXI Century” Donetsk National Technical University, pp. 262-266, 2003.
[4] D. Ghelase, *Influence of the Geometrical Parameters on Rigidity of the Worm-Gearing Tooth*, The Annals of "Dunărea de Jos" University of Galaţi, Fascicle XIV, pp. 45-48, 2003.
[5] D. Ghelase, L. Daschievici, *Optimal Design of the Worm-Gearing with Circular Profile Using Medium Rigidity Criterion*, Proceeding of the International Conference on the Manufacturing Systems ICMaS, "Politehnica" University of Bucharest, pp. 453-456, 2006.
[6] F. L. Litvin, *Gear Geometry and Applied Theory*, Prentice Hall, New Jersey, 1994
[7] D. Ghelase, L. Daschievici, *Computerized Design-Generation of the Worm-Gear Flank*, The Archive of Mechanical Engineering, Politechnika Warszawska, Polonia, Vol. LIII, Nr. 2, pp. 165-177, 2006.
[8] D. Ghelase, L. Daschievici,I. Gorlach, *Numerical Results of Simulation of Meshing of Worm-Gear Drive*, Abstracts of Sixth South African Conference on Computational and Applied Mechanics SACAM08, Cape Town, South Africa, 2008.
[9] H. Oliveira, A. Sousa, P. Moreira, P. Costa, *Dynamical Models for Omni-Directional Robots with 3 and 4 Wheels*, Proceeding of Fifth International Conference on Informatics in Control, Automation and Robotics ICINCO2008, Funchal, Madeira-Portugal, pp. 189-196, 2008.