Applied Computer Science, vol. 16, no. 1, pp. 33–40
doi:10.23743/acs-2020-03

Submitted: 2020-02-26
Revised: 2020-03-10
Accepted: 2020-03-20

modeling of mechanisms, working surface meshing

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MODELING TRANSMISSION MECHANISMS WITH DETERMINATION OF EFFICIENCY

Abstract

Continuing previous studies, reviewed by modeling transmission mechanisms with the definition of one of the major criteria of transmission efficiency – the efficiency of the gearing, which depends on the geometry and kinematics of the working surfaces of the engagement and position of the contact point on the engaging surface in the object of the simulation model the designed transmission.

1. INTRODUCTION

Every modern machines and mechanisms are in their design different gear mechanisms that are parts with a complex profile. Using the full-scale simulation to automate the optimal design make it possible to transfer the process of testing actually made arrangements for testing and analysis of the simulation model, which significantly saves material and time resources for the preparation and introduction of modern machinery or equipment and guarantee their quality and reliability in the process. Using a simulation model is possible in the construction of adequate mathematical models that represent the working process of engagement and allow a comparative analysis of efficiency designed for transmission of the transmission device.

In performance gear machines usually estimated quality indicators (Gribanov, 2003) – criteria characterizing locally kinematic and hydrodynamic phenomena in the tooth contact area. Geometrical parameters of transmission loss significantly affect meshed (Gribanov, 2003; Gribanov & Ratov, 2011). The spatial gears

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longitudinal sliding provides additional frictional losses in gearing. Therefore, an important criterion for evaluating the performance of a transmission criterion characterizing the losses in engagement. Such losses can be estimated efficiency engagement coefficient which depends on the geometry and kinematics of the working surfaces of the engagement position of the point of contact on the engagement surface.

The aim of the article is to develop a refined model for calculating efficiency spatial transmission operatively engaged with the contact interactions and side surfaces of the teeth, taking into account the rolling speed and the slip contact pads.

2. MODELING THE OBJECT OF RESEARCH

To study the efficiency of spatial transmission using a mathematical model of the process of formation of teeth on the primary hyperboloidal surfaces (Ratov & Balitska, 2007; Gribanov & Ratov, 2009). The model is based on the mutual bending of producing and manufactured surfaces. For a description of the model have been met:

1. Calculation of initial engagement circuit (Ratov & Balitska, 2007).
2. Calculation of theoretical initial spatial transmission surfaces, which are one-sheeted hyperboloids of rotation (Gribanov & Ratov, 2009; Nosko, Shyshov & Ratov, 20140.
3. Preparation of the radius-vector form of the equations describing the lateral surface of the tooth (Gribanov, 2003; *Instructional "Gleason" company materials*, 2001).
4. Model working engagement spatial wheels (Gribanov, 2003; Gribanov & Ratov, 2009). Figure 1 shows a computer implementation of working engagement simulation.

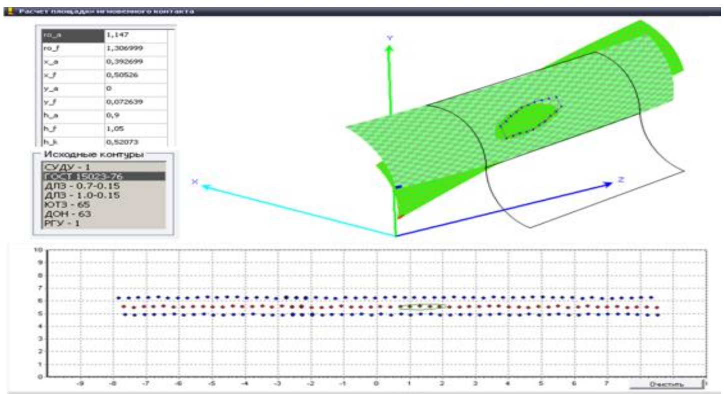

Fig. 1. Computer implementation working engagement simulation

Based on the simulation-object modeling mathematical model obtained in the system of the hybrid parametric solid modeling was performed SolidWorks working engagement teeth contacting spatial transmission and a model of the worm gear and the gear transmission with spatial developed (Fig. 3). The structure and steps of construction in the modeling system is shown in Figure 2.

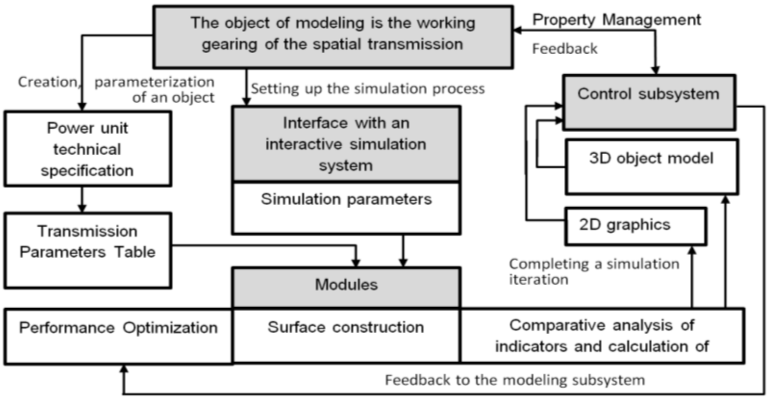

Fig. 2. Structure – modules working gears meshing spatial modeling system

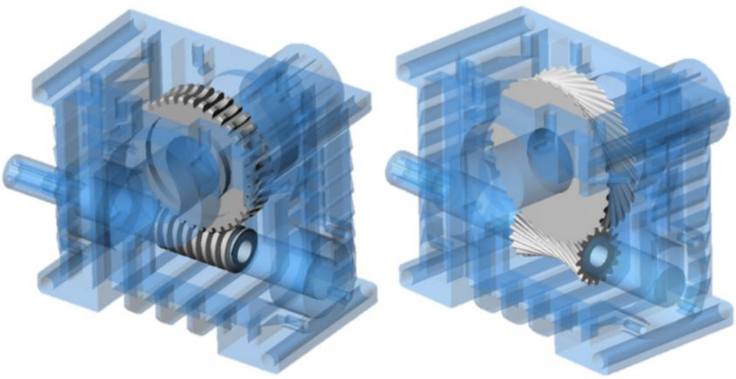

Fig. 3. Worm gearbox and gearbox with developed spatial gear

3. MODEL FOR DETERMINING EFFICIENCY COEFFICIENT

For evaluation and **comparative** analysis of the synthesized helical gear form the model for calculating the efficiency of the spatial transmission. We express this ratio as the ratio of the elementary works on the driven and the driving gear:

$$\eta = \frac{A_1}{A_2} \tag{1}$$

where: $A_1 = F_1' \cdot V^{(1)} A_2 = F_2' \cdot V^{(2)} V^{(1)} V^{(2)}$ – the elementary work on the driven and driving gears, respectively, $V^{(1)}, V^{(2)}$ – the circumferential speed of the wheels (Figure 4b), F_1', F_2' – the projection of the normal force F_n on $V^{(1)}$ and $V^{(2)}$.

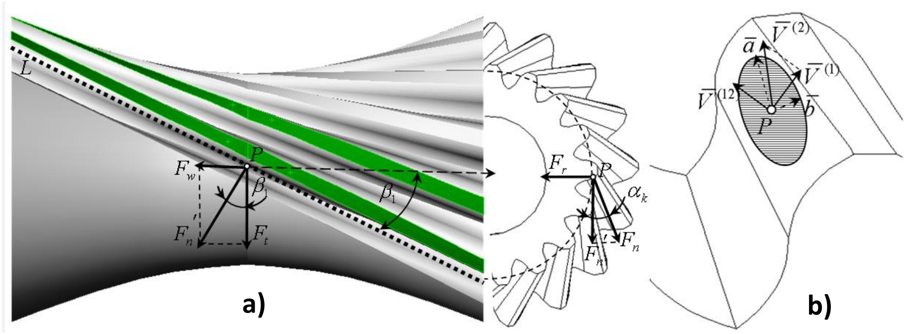

Fig. 4. a) The forces acting in the engagement F_n normal force, F_t circumference force, F_r – radial force, F_w – axial force; b) vectors peripheral speeds

Construct a mathematical model for determining the efficiency of the spatial transmission is made in consideration of contact interactions side surfaces of the teeth, taking into account the rolling speed and the slip contact pads.

We define the forces at the wheel and pinion helical gear (Fig. 4a). The normal force F_n between the surfaces of the gear teeth and pinion assumed applied to the nominal point of contact P, and its projection F_n' on a common tangent plane to the primary hyperboloidal aksoids at point P coincides with the direction normal to the teeth and is equal to:

$$F_n' = F_n \cos \alpha_k \tag{2}$$

where: α_k – in nominal pressure angle of the teeth the point of contact.

Hoop forces F_t at the wheel (all parameters with index 1) and gear (all parameters with index 2) is equal to the projection F_n' (2) in the direction perpendicular to the generator hyperboloid L (Figure 4a):

$$F_{t1} = F_n' \cos \beta_1;\ F_{t2} = F_n' \cos \beta_2 \tag{3}$$

where: $\beta_1,\ \beta_2$ – angles of generators.

Fig. 4b shows the circumferential velocity vectors point P, lying in the tangent plane. The vector of relative moving speed of the contact points of the active surfaces of the teeth (vector relative sliding velocity) will be:

$$\bar{V}^{(12)} = \bar{V}^{(2)} - \bar{V}^{(1)} \tag{4}$$

The magnitude of this vector:

$$V^{(12)} = V^{(2)} \sin\beta_2 - V^{(1)} \sin\beta_1 \tag{5}$$

We take into account the condition of equality of vectors $V^{(1)}$ and $V^{(2)}$ projections on the normal direction:

$$V^{(2)} = V^{(1)} \frac{\cos\beta_1}{\cos\beta_2} \tag{6}$$

Longitudinal sliding of the presence of the normal force between the teeth causes the frictional force directed oppositely sliding velocity and direction of the deflection force.

Consequently, the normal force on the projection plane is also tangential to change its direction, deviating at angle of friction ρ (Gribanov, 2003; Shishov, Velichko & Karpov, 2009):

$$f = \mathrm{tg}\rho \tag{7}$$

where: f – coefficient of sliding friction at the contact points the working surfaces of gear pair.

The projection of the normal force F_n on the velocity $V^{(1)}$ and $V^{(2)}$ direction are of the form:

$$F_1' = F_n \cos\alpha_k \cos(\beta_1 - \rho);\; F_2' = F_n \cos\alpha_k \cos(\beta_2 - \rho) \tag{8}$$

Substituting (6) and (8) into (1) we obtain:

$$\eta = \frac{F_1' \cdot V^{(1)}}{F_2' \cdot V^{(2)}} = \frac{F_n \cos\alpha_k \cos(\beta_1 - \rho) \cdot V^{(1)}}{F_n \cos\alpha_k \cos(\beta_2 - \rho) \cdot V^{(2)}} = \frac{\cos(\beta_1 - \rho) \cdot \cos\beta_2}{cos(\beta_2 - \rho) \cdot \cos\beta_1} \tag{9}$$

After standard identity transformations, taking into account (7), the efficiency of spatial transmission (9) takes the form:

$$\eta = \frac{(\cos\beta_1 \cdot \cos\rho + \sin\beta_1 \cdot \sin\rho) \cdot \cos\beta_2}{(\cos\beta_2 \cdot \cos\rho + \sin\beta_2 \cdot \sin\rho) \cdot \cos\beta_1} = \frac{1 + \tan\beta_1 \cdot f}{1 + \tan\beta_2 \cdot f} \tag{10}$$

Rolling bodies slidably most complete linkage geometry and the force interaction in contact bodies considered in determining the coefficient of sliding friction in (Shishov, Velichko & Karpov, 2009):

$$f = \frac{0.09 \cdot q_n^{0.1} \cdot \left(10 + \lg\frac{HB \cdot R_a}{E_r} |\chi_r|\right)}{\left(V^{(12)}\right)^{0.35} \cdot \left(V^{(\Sigma)}\right)^{0.1} \cdot \nu^{0.07}} \cdot |\chi_r|^{0.25} \tag{11}$$

where: q_n – the contact load per unit length of the line (N/cm),
HB – hardness of less solid bodies in contact (N/cm^2),
$V^{(12)}$ – normal to the contact line relative sliding velocity (cm/s),
$V^{(\Sigma)}$ – total rolling velocity (cm/s),
E_r – reduced modulus (N/cm^2),
R_a – the roughness value of the most solid (cm),
ν – oil viscosity (cSt),
χ_r – reduced curvature.

Load per unit length of the line of contact can be found from the relationship:

$$q_n = \frac{T}{r \cdot L \cdot \varepsilon} = \frac{T \cdot \cos\beta_1}{r \cdot \varepsilon \cdot b_w} \tag{12}$$

where: T – torque on the input shaft,
r – the distance from the contact point to the axial line of the shaft,
L – length of the line of contact,
ε – the overlap coefficient.

In view of formula (12) and the resistance to scoring coefficient represented in (Gribanov, 2003; Gribanov & Ratov, 2009; Ratov, & Lyfar, 2019), where it is expressed as a ratio of the relative sliding velocity teeth surfaces to the total rolling rate of contact points:

$$K_v = \frac{V^{(12)}}{V^{(\Sigma)}} \tag{13}$$

coefficient of sliding friction (11) finally becomes

$$f = 0.09 \cdot \left(\frac{T \cdot \cos\beta_1}{r \cdot \varepsilon \cdot b_w}\right)^{0.1} \cdot |\chi_r|^{0.25} \cdot \frac{\left(10 + \lg\frac{HB \cdot R_a}{E_r} \cdot |\chi_r|\right)}{(K_v)^{0.35} \cdot \left(V^{(\Sigma)}\right)^{0.45} \cdot \nu^{0.07}} \tag{14}$$

The turns of the worm in the worm gear slip when driving on the wheel teeth. Large slip is the cause of the wear and seizing of such transfers, reducing their efficiency. Model for calculating efficiency worm gear similar construct, the calculation of the efficiency screw pair, since the friction conditions they are identical (Shishov, Velichko & Karpov, 2009):

$$\eta = 0.95 \frac{\mathrm{tg}\gamma}{\mathrm{tg}(\gamma + \rho)} \tag{15}$$

where: 0.95 – factor takes into account the energy loss in mixing oil for lubrication dipping,
γ – lifting pitch helix angle ($5° - 20°$),
ρ – the friction angle.

Considering equation (7), the expression for calculating the efficiency a worm gear (15) takes the form:

$$\eta = 0.95\frac{\mathrm{tg}\gamma\cdot(1-\mathrm{tg}\gamma\cdot f)}{\mathrm{tg}\gamma+f} \tag{16}$$

4. THE METHODOLOGY AND THE RESULTS OF RESEARCH

Using the equations of the surfaces in the radius-vector form $\bar{r}\,(\nu,\varphi)$ (Gribanov, 2003; *Instructional "Gleason" company materials*, 2001), the equation for K_ν (Gribanov, 2003; Gribanov & Ratov, 2009; Ratov, & Lyfar, 2019), $V^{(\Sigma)}$ (Gribanov & Ratov, 2009), which are defined using the coefficients of the quadratic forms of the surface *E*, *F*, *G*, equation given curvature χ_r, the resultant sliding friction coefficient (14), taking into account the efficiency of the equation spatial transmission (10) and worm gear (16) in Computer Algebra System Wolfram Mathematica 8 construct a mathematical model for calculating efficiency spatial transmission operatively engaged (at the angle of engagement) and worm gear (Fig. 5). In this case T = 23000 Ncm; HB =2800 N/cm^2; E_r = 203.9·10^3 N/cm^2; R_a= 4·10^{-5} cm; ν = 20 cSt.

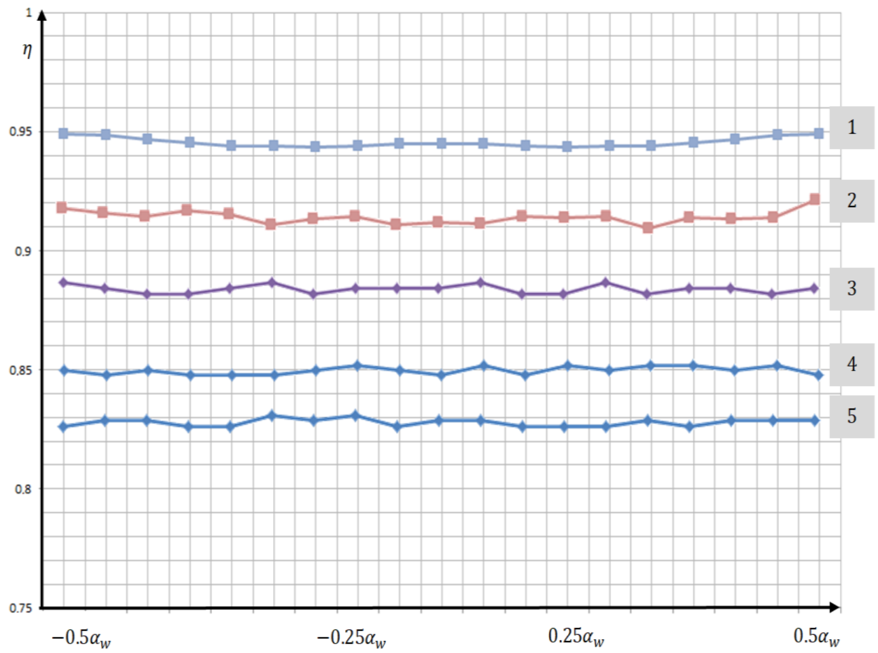

Fig. 5. The efficiency of the test gear

Fig. 5 is a graph 1, 2 – efficiency spatial transmission schedule 3–5 – efficiency worm gear. The average efficiency of the theoretical value spatial transmission №1 – $\eta_{\text{cp}} = 0.945$, spatial transmission №2 – $\eta_{\text{cp}} = 0.9114$, worm gear №3 – $\eta_{\text{cp}} = 0.8845$, worm gear №4 – $\eta_{\text{cp}} = 0.8538$, №5 worm gear $\eta_{\text{cp}} = 0.8334$.

Efficiency worm gear increases with the number of turns of the worm (increase divider angle γ). In the worm gear №5 number of turns is 1, y transmission №3 – 4. Increasing the number of turns γ decreases the rigidity of the divider angle of the worm, and increasing the number of teeth increases the transmission distance between the supports. Increase efficiency spatial transmission №1, № 2 caused a decrease in friction coefficient f of the contact surfaces (friction angle decreases ρ), which depends on the reduction ratio K_v (reduction of relative sliding velocity $V^{(12)}$ and the total velocity $V^{(\Sigma)}$ increase lateral rolling surfaces).

5. CONCLUSIONS

1. Developed a refined model for calculating efficiency engagement with the contact interactions and side surfaces of the teeth, taking into account the rolling speed and the slip contact pads.
2. Conducted implementation of simulation and object modeling. The results of computing experiment, which showed an increase of the value of efficiency gireboloidnoy investigated gearbox transmission.

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