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LOAD DISTRIBUTION ALONG THE LINE OF CONTACT TO WORM GEARING

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ABSTRACT

The paper presents a study on the load distribution along the line of contact between the worm tooth and worm wheel tooth. The load distributed unequally and with discontinuities will lead to a variable deformation of the teeth, the appearance of transmission error, which means the change of the transmission ratio.

KEYWORDS: worm gearing, deformation, contact line, contact pattern, stress

1. INTRODUCTION

The rigid worm gearing with errors, due to instantaneous linear contact, has a high sensitivity to the errors from processing and assembly. In this regard, the localized contact concept appeared which wants to achieve a perfect contact pattern with low sensitivity to errors. It practically aims to reduce the instantaneous classic linear contact to one point, yielding improved operating parameters of the worm gearings. Most research ([1], [2], [3], [4], [5]) aimed the study of this type of contact and their results are outstanding: computerized design, computer simulation of the meshing and contact.

Today, it is possible to produce a computer simulation of the contact pattern before reaching the design final phase and to processin the worm gearings.

2. CONTACT OF THE RIGID WORM GEARING WITH ERRORS

Using a numerical tool [6], we can simulate the technological surface and we can study the radial and tangential toothing or one with flying knife, taking into account the kinematic error. If the contact pattern is not expected it can change the toothing parameters so that the quality of the contact pattern improves, yielding an optimization of the teeth parameters for both worm and worm wheel. There are two approaches to get a preferentially localized contact pattern:

a) by modifying the geometry of the worm gear teeth; In Figures 1 [7], 2 [2] and 3[2], there are a few examples of the influence of the geometry modification on the contact pattern. Thus, in the figure 1 it can be seen that:

- The R_A parameter (rolling radius of the worm wheel) has an important influence on locating contact. Changing the R_A from 128 to 127 and then 126, the distance between the flanks doubles or triples and contact pattern and cinematic errors decrease with radius R_A;

- If parameter z_A (contact point position toward the median plane of the wheel) has low values (0.8% $\cdot d_1$), it practically does not affect the size of the contact pattern and no cinematic errors. For zA = 16% ·d1 the cinematic errors increase and the contact pattern is placed in the third quarter of the length of the wheel tooth, where the contact is made most favourably;

- Increasing the diameter of the gear hob k_A, which increases the distance between the axes of the technological gear $\Delta A = 0{,}5 \cdot k_A \cdot d_1$, significantly influences the size of the contact pattern, for the parameter k_A =(3-9%) ·d1;

- If the wheels are without correction profile ($\delta = 0$), the kinematics error is zero, regardless of the values of other parameters. If the tooth flank is curved ($\delta \neq 0$), the kinematics error is present and the shape of the contact pattern changes.

Assembly errors cause the contact pattern translation ([8], [2] and [3]). After the changing geometry, the contact pattern becomes stable and it is located in a central position (Figures 2 and 3).

b) Numerical methods for designing a localized contact. The simulation of the meshing and contacts, called "Tooth Contact Analysis" (TCA) is an important problem that was solved using computer techniques and it drew the attention of many researchers. The program combines numerical programming with graphical illustration. A special attention is given to worm gearing synthesis, but the focus is on locating on the contact pattern and noise reduction. The contact pattern consists of more instantaneous contact ellipses (Figure 4).

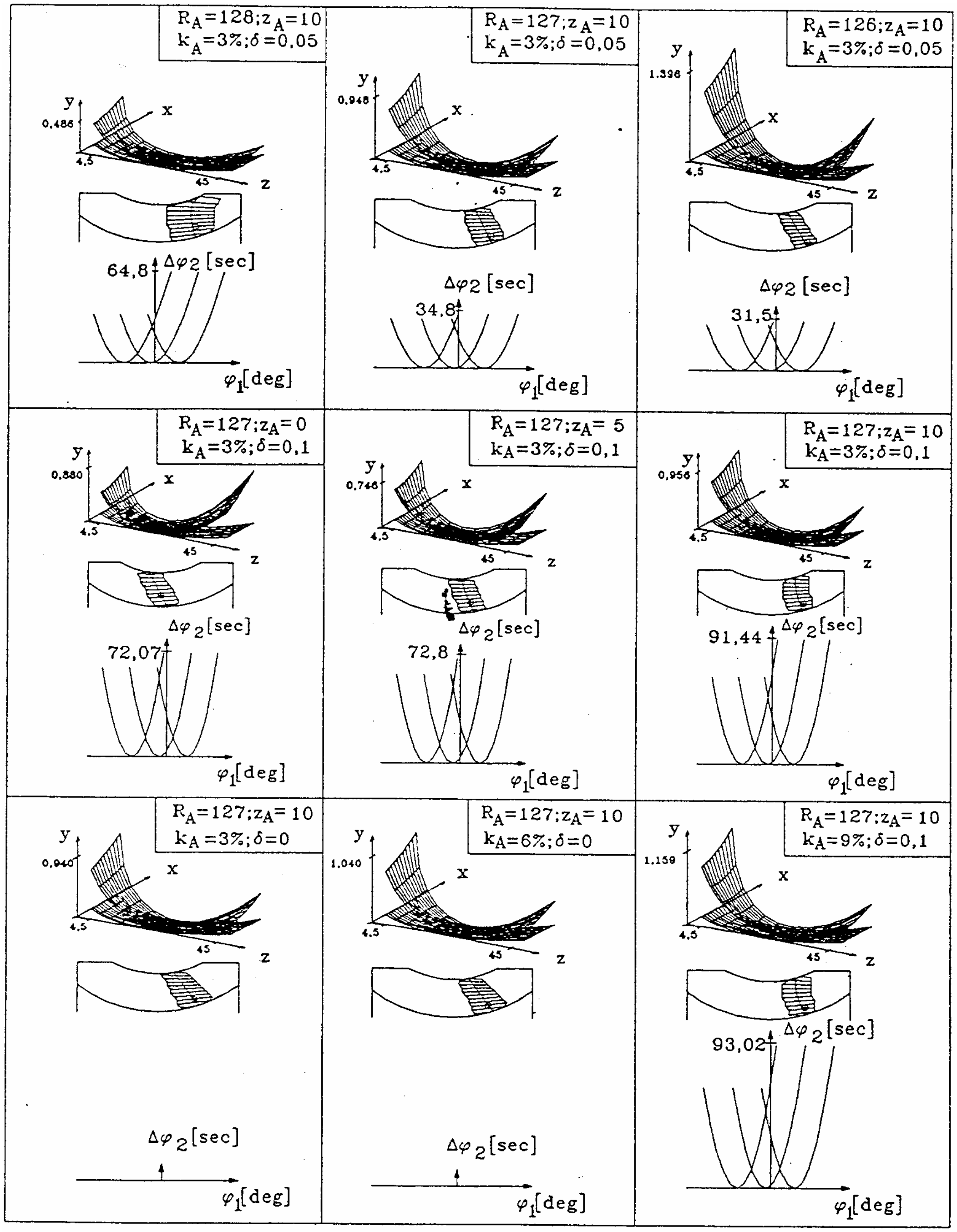

Fig. 1 [7] Numerical results related to the location of contact (contact pattern for a distance between the flanks Δ = 15μm) for a cylindrical worm gearing with the following parameters: A=160mm; m=5mm; u=51/2; ZK1- gearing type; d_1=12.5·5=62.5mm; d_{a1}=73.5mm; d_2=255mm; d_{a2}=268.5mm; d_{aM}=275mm; b_2=45mm.

- Top: the influence of the wheel rolling radius R (for z_A = 10; k_A = 3%; δ = 0.05);
- Middle: the influence of the contact point position, z_A (for R_A = 127; ka = 3%; δ = 0.1);
- Below: the influence of the increased distance between axes of the technological gearing ΔA; ΔA = $0.5 \cdot k_A \cdot d_1$ (for R_A = 127; z_A = 10; δ = 0 ... 0.1).

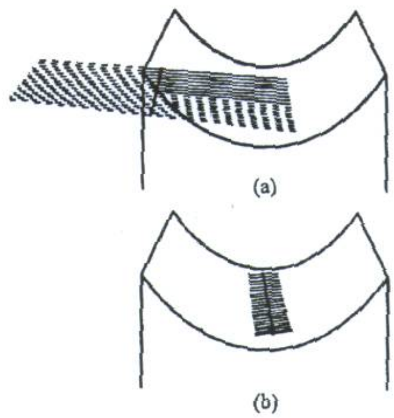

Fig. 2 [2] Contact pattern for the worm gearing ZE:

a) Unchanged worm gearing;

b) Changed worm gearing; ΔE=0.1mm; $\Delta\gamma$=3min (ΔE is the variation of the distance between axes, $\Delta\gamma$ is the variation of the gear

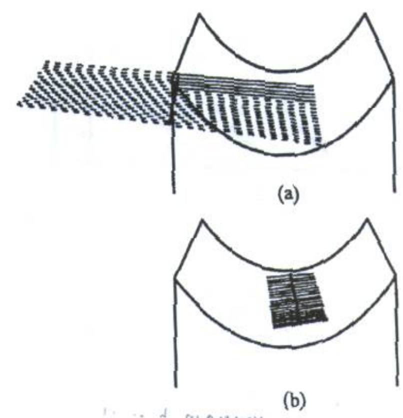

Fig. 3 [2] Contact pattern for the worm gearing ZK:

a) Unchanged worm gearing;

b) Changed worm gearing; ΔE=0.1mm; $\Delta\gamma$=1min

To determine the orientation and dimensions of these ellipses is necessary to know among other things, the size of the elastic deformation of the teeth.

3. LOAD DISTRIBUTION ALONG THE LINE OF CONTACT

The methodology for determining the load distribution along the line of contact between the worm tooth and worm wheel tooth is based on the following relation:

$$w(z_D) = \int_{L_{it}} K_d(z_D, z_F) \cdot p(z_F) \cdot dz + K_c(z_D) \cdot p(z_D) = \\ = \Delta y_n - e_n(z_D) + s(z_D) \qquad (1)$$

where:

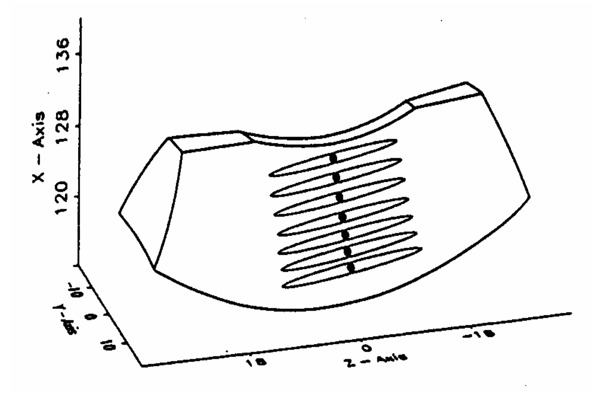

Fig. 4 [3] Contact pattern on the surface of the worm wheel

L_{it} is the length of the line of contact between the worm tooth and worm wheel tooth;

$K_d(z_D, z_F)$- influence factor of the load action on the tooth in the point F of the worm gearing tooth surface to the deformation of the worm tooth and wheel tooth there are in contact point D. Factor K_d includes the bending and shear deformations of the worm tooth and wheel tooth, the bending and torsion deformations of the wheel, the axial deformation of the worm and support shaft deflections;

$K_c(z_D)$- influence factor of the contact between the worm tooth and worm wheel tooth, including the contact deformation of both teeth in point D under the load action in the same point;

$p(z_F)$, $p(z_D)$ – loads on the tooth in point F, respectively point D;

$w(z_D)$- total deformation in point D;

Δy_n- projection of the wheel rotational delay to the worm on the normal to gearing teeth surface;

$e_n(z_D)$ – error in point D, which is the sum of the processing and assembling errors;

$s(z_D)$- relative geometric separation between worm tooth and worm wheel tooth in point D.

The calculus relations of K_d, K_c influence factors have been determined based on the analysis of the worm tooth and wheel tooth stresses, using finite element analysis. The variables w, e_n and s are measured along the normal to the conjugated surfaces.

In the case of theoretical worm gearing, the contact is linear, so that the relative separation of the geometric surfaces of the worm tooth and wheel tooth is zero. But the contact of the unconjugated worm gearing is punctual due to the introduction of changes to the surfaces of teeth or because processing the cutter wheel with a greater diameter, [9].

The investigations have shown that this contact transforms into a linear contact under load, but not along the whole width of the tooth wheel.

To be able to achieve a load distribution of the worm gearing with the geometric errors we must know the separations of the surfaces in the contact along these potential lines. In [4] it is presented the method of determining the line with minimum separations on the contact surfaces, so this line is entirely or a partially contact line. The method is based on a minimizing function which determines the separation of the conjugated surfaces.

The separation is defined as the distance between the points on the corresponding surfaces, obtained as points of the intersection of the common normal to the conjugated surfaces. Mathematically, it means minimizing function

$$s = \sqrt{(x_g^{(g)} - x_w^{(w)})^2 + (y_g^{(g)} - y_w^{(w)})^2 + (z_g^{(g)} - z_w^{(w)})^2}. \quad (2)$$

The torsion moment is given by the equation:

$$T = \sum_{i_t=1}^{i_t=N_t} \int_{L_{it}} r_F \cdot [\bar{p}(z_F) \cdot \bar{t}_{0F}]dz, \quad (3)$$

where:

r_F is the distance from the load application point F on the wheel axis;

$\bar{t}_{0F}$ - unit vector tangent to the circle of radius r_F, passing through the point F in the transverse plane of the wheel;

Nt - the number of teeth of the wheel are instantly in contact.

The distribution of the load on each line of contact is determined by solving a nonlinear system of equations (1) - (3).

Some results of the program running developed in [9] are shown in the figure 5 which presents the distribution of the load on the meshing teeth at a time, in the case of the grinding worm.

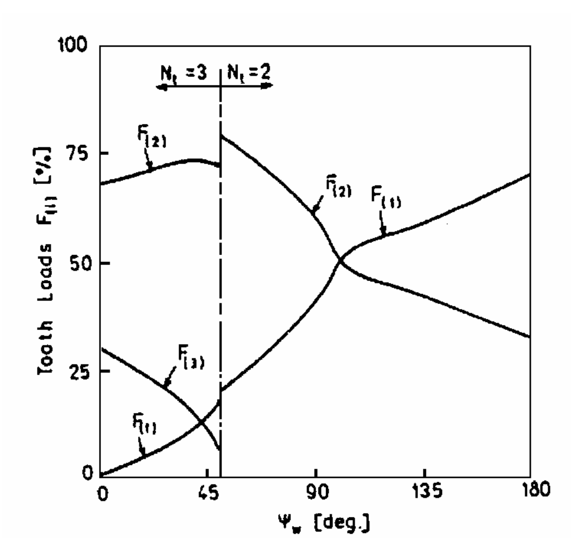

Fig. 5 [9] The distribution of the load on the meshing teeth at a time

As it may be seen, during meshing, if the teeth are in contact, the load is distributed unequally and with discontinuities by changing the number of pairs of teeth that are in the meshing. This will lead to a variable deformation of the teeth, the appearance of transmission error, which means the change of the transmission ratio. Important studies regarding reduction of these transmission errors were already made in the field of the plan gearing gear. In this regard, there have been analyses on the load distribution depending on the size of the load, the type of teeth (spur or bevel teeth), the width of the teeth, the position of a contact point (initial or final) [10].

In order to obtain a certain distribution of the load on the contact area, to ensure error reduction a methodology was realized for changing the profile teeth [10]. It was to develop a profile modification calculation maintaining constant rigidity of the teeth during meshing [11].

4. CONCLUSIONS

1. There are two approaches to get a preferentially localized contact pattern: a) by modifying the geometry of the worm gear teeth; b) Numerical methods for designing a localized contact (Tooth Contact Analysis- TCA).
2. During meshing, if the teeth are in contact, the load is distributed unequally and with discontinuities by changing the number of pairs of teeth that are in the meshing.
3. In order to obtain a certain distribution of the load on the contact area, to ensure error reduction a methodology was realized for changing the profile teeth.

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