CONDITIONS OF OCCURRENCE OF SHOCKS

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Abstract: Shocks in cases of cylindrical cogwheel gears appear because of the departure of the basic step. Related to the departure of the basic steps and the elastic distortion of the cogwheel's pair of teeth that are not involved in the collision, the contact between the cogwheel's pair of teeth that are in the engagement occurs wither outside the engagement line or on the engagement line. If the percussive force has comparable values with those of other forces that appear in the process of engagement, they can not be neglected. For studying the collision phenomena, a special interest is shown by the analysis of the behavior of the kinematic coupling elements when the instant modification of forces occurs.

Dynamic Errors

For studying the collision phenomena, a special interest is shown by the studying of the behavior of the kinematic coupling elements when the instant modification of forces occurs.

We assumed that in a certain position of the mechanism, determined by the angle α_{e2}, the external force vector value changes "instantly" when the force makes an additional relative movement, admitted by the couple movement.

In this position of the mechanism, until the instant movement of the external force, the relative position of the couple's kinematic elements is given by the angle, $\theta h_1 = a t g \mu$, μ being the coefficient of friction from the couple. The proposed model is shown in Figure 1.

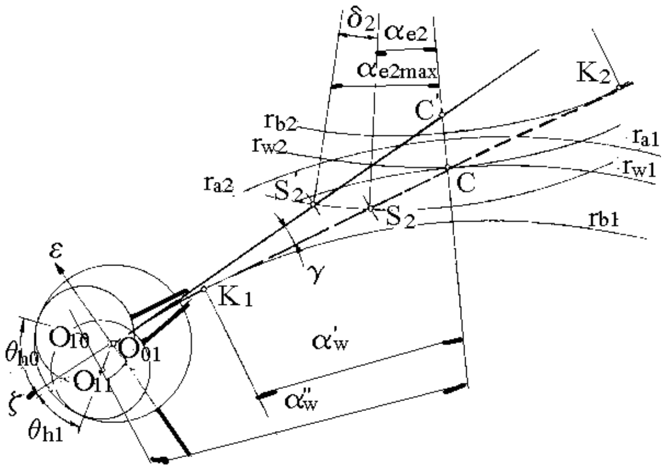

Figure 1 Collision within the rotation couple (Edge collision)

Also for simplifying, we accept that in this position $\Delta h = \frac{j}{2}$, j representing diametric movement and low radial force, F_N, balances with element 1's reaction over element 2:

$$F_N = F_N^0 \sqrt{1+\mu^2} \qquad (1)$$

Where

$$F_N^0 = \frac{M_{t1}}{r_{b1} - \mu\left(r_{b1} tg\alpha_w^{'} - r_{a2}\frac{\sin\alpha_{e2}}{\cos\alpha_w^{'}}\right)} \qquad (2)$$

Reduced tangential force:

$$F_T = F_N \cdot \cos.\alpha'_w \tag{3}$$

Imposes the relative movement of element 1 in relation to item 2.

Supposing that because of the instant modification of the force vector, the radial reduced force has changed its direction and is directed towards the center O_0.

This radial force of the kinematic coupling is determined by the relation:

$$F_{N1} = F_{N1}{}^{0}\sqrt{1-\mu^2} \tag{4}$$

Where

$$F_{N1}{}^{0'} = \frac{M_{t1}}{r_{b1} - \mu\left(r_{b1}\,tg\alpha''_w - r_{a2}\dfrac{\sin\alpha_{e2max}}{\cos\alpha''_w}\right)} \tag{5}$$

It causes a greater normal acceleration of the point O'_1 in relation to O_0. The radius of point's O_{11} curved trajectory shrinks instantly in relation to O_0 and becomes practically zero. Point O_1 starts its movement on the right LL, which coincides with the direction of resultant forces, F_{N1} and F_T, respectively in the direction of force:

$$P_1 = \sqrt{F_{N1}^2 + F_T^2}\,. \tag{6}$$

This way the element 2 will move freely in relation to element 1 and comes in contact with it again at the point O_{10} characterized by the coordinates (Δh, θ_{h0}). The direction of movement of the point O_1 in relation to O_0 forms with the axis $\varepsilon_k\varepsilon_k$ the angle γ, having the value:

$$\gamma = \theta_{h1} + \beta - \pi/2 \tag{7}$$

Where,

$$tg\,\beta = \frac{F_{N1}}{F_T} \tag{8}$$

Also, between the coordinates ζ_h, ε_h there is the following relationship:

$$\zeta_h = \varepsilon_h\, tg\,\gamma \tag{9}$$

We set the current position of the point O_1 on the right LL in report with it's initial position through the l_h segment.

The reduced mass of the system will be:

$$m^{red} = \frac{m_1^{red} m_2^{red}}{m_1^{red} + m_2^{red}} \tag{10}$$

Where:

$$m_1^{red} = \frac{J_1}{r_{b1}^2}\text{; and } m_2^{red} = \frac{J_2}{r_{b2}^2} \tag{11}$$

represents the mass of the cogwheel gears, reduced to the basic circles.

Due to the size of segment l_h, one can consider that the reduced mass does not dependent on l_h. Given this condition, the movement speed of the point O_1 on the right LL is determined by [3]:

$$\frac{d\left(m^{red} \cdot v_1\right)}{d\alpha} = \frac{1}{\omega \cdot P_1} \tag{12}$$

Follows:

$$v_1 = \frac{1}{m^{red}}\int \frac{d\alpha}{\omega \cdot P_1} - \frac{m_1^{red}}{m^{red}} v_{11} \tag{13}$$

Where

m_l^{red} and v_{11} are reduced mass and the speed of the point O_1 on the direction LL, in that position of the crank where the exterior forces system has changed "instantly".

This speed can be easily calculated knowing the speed of additional coordinates, known for this position:

$$v_{11} = \frac{\omega_1 r_{b1} \omega}{\cos\gamma} \quad (14)$$

The position of O_1 point on the line MM, for different values of the α angle is determined from the equation:

$$l_h = \int \frac{v_1 \cdot d\alpha}{\omega} \quad (15)$$

The elements of the kinematic couple come back into contact when point O_1 reaches the position O_{1o}, which is determined by the equation:

$$\theta_{h0} = \frac{\pi}{2} - (\gamma + \beta) \quad (16)$$

The total shift of the point O_1 on line LL is equal to:

$$L_h = 2\Delta h \cdot \sin\frac{\theta_{h1} + \theta_{h0}}{2} = 2\Delta h \cdot \cos\beta \quad (17)$$

From the equation (15) it can be determined the value α_{e2max} of the basic coordinateα, when elements 1 and 2 are in contact:

$$2\Delta h \cdot \cos\beta = \int \frac{v_1 \cdot d\alpha}{\omega} \quad (18)$$

From the equation (13), we determine the speed v_{1o} of point O_1, for this moment:

$$2\Delta h \cdot \cos\beta = \int \frac{v_1 \cdot d\alpha}{\omega} \quad (19)$$

$$v_{10} = \frac{1}{m_0^{red}} \int \frac{d\alpha}{\omega \cdot P_1} - \frac{m_1^{red}}{m_0^{red}} v_{11} \quad (20)$$

Due to the reduced size of the movement, α_{e2} and α_{e2max} angles slightly differ from one another, and the modification of the reduced mass and force, between these two positions of the crank, is so small that it can be considered a constant. We can accept as a calculating measure, the value at $\alpha = \alpha_{e2}$. If we consider the fact that in the interval, $\alpha_{e2} < \alpha < \alpha_{e2max}$, the angular velocity of the crank is constant, then the speed v_{1o} value, when items 1 and 2 are in contact, is determined by:

$$v_{10} = \sqrt{\frac{4P_1}{m^{red}} \Delta h \cdot \cos^2\beta + v_{11}^2} \quad (21)$$

The speed of the supplementary coordinates at the point where the elements are in contact is calculated with relation:

$$\dot{\varepsilon}_{10} = v_{10} \cdot \cos\gamma \text{ and } \dot{\xi}_{10} = v_{10} \cdot \sin\gamma \quad (22)$$

The calculation of the supplementary coordinate's acceleration does not involve special difficulties in this position of the mechanism.

Angular position error α_{e2max} can be determined by the relation:

$$\alpha_{e2max} = \frac{S_2^{'}S_2 + r_{a2} \cdot \alpha_{e2}}{r_{a2}} \quad (23)$$

The value of the engagement arch ($S^{'}_2 S_2$) is determined in relation to the basic step errors (Δp) and distortion(δ):

$$S_2^{'}S_2 = \delta \cdot r_{a2} \qquad (24)$$

Where:

$$\delta = \sqrt{\frac{2(f_0 + \Delta p)}{a \cdot \sin\alpha_w^{'}}}\sqrt{\frac{a \cdot \sin\alpha_w^{'} - \left(r_{a1}^2 - r_{b1}^2\right)^{\frac{1}{2}}}{\left(r_{a1}^2 - r_{b1}^2\right)^{\frac{1}{2}}}} \qquad (26)$$

Tooth deflection, $f_0=f_{i1}+f_{i2}+f_h$, consists of flexural strain($f_{i1,i2}$) and Hertz strain (f_h)

The parameter α_{e2} is calculated using the formula:

$$\cos\left(\alpha_w^{'} + \alpha_{e2}\right) = \left(1 - \frac{a - \frac{r_{b1}}{\cos\alpha_w^{'}}}{r_{a2}}\right) \cdot \cos\alpha_w^{'} \qquad (27)$$

Where

$$\Delta a = a_0\left(\frac{\cos\alpha_0}{\cos_w^{'}} - 1\right)$$

$$\sin\left(\alpha_{w1}^{''} + \gamma\right) = \frac{r_{b1} \cdot \cos\gamma}{r_{a1} \cdot \sin\alpha_{e2\max}} \qquad (28)$$

$$ctg\gamma = \frac{\sin\alpha_{e2\max}}{\frac{a}{r_{a2}} - \cos\alpha_{e2\max}}$$

The instant transmission ratio will be:

$$i_\omega = \frac{z_{1+z2}}{z_1}\left(1 + \frac{2\Delta a}{m \cdot z_{1+z2}}\right) \cdot \frac{\cos\alpha_{w2}^{'}}{\cos\alpha_0} \qquad (29)$$

The instant transmission ratio modifies with the approachment of the contact point near the engagement line when it becomes equal to the nominal one (Figure.2)

Figure 2 Actual transmission ratio variations

The results obtained with the help of the computer-aided simulation programmer are confirmed by the measurements made on the test stand. The program for data processing and vibration analysis in the frequential field has drawn up in the "MATLAB" environment language. Generally there is a concordance between the experimental (Figure 3), and theoretical results (Figure 4).

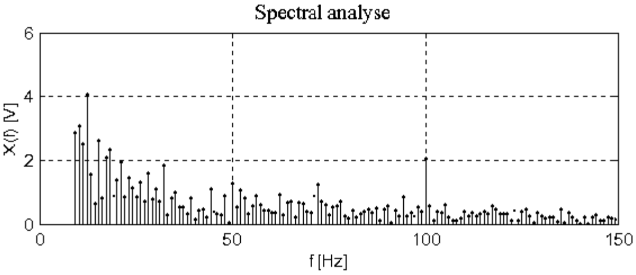

Figure 3 Experimental diagrams

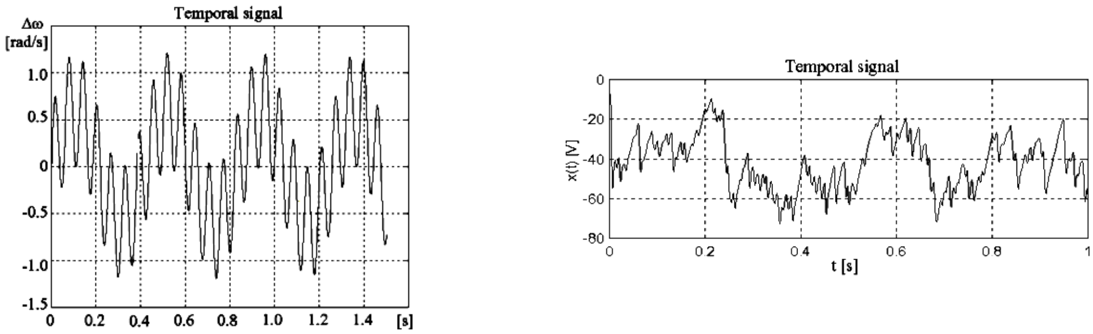

Figure 4 Theoretical diagram

Conclusions

• The theoretical and experimental research of the kinematics errors of gears under the impact of tensions and effective speeds represents a complex and up-to-date problem whose solution allow the evaluation of the functional qualities of a gear.
• The internal sources of the gear transmission error are very important for the gear durability. Some of them include deviations from the tooth processing precision, setting and operation errors, and the periodical variation of the gearing rigidity.
• The kinematics precision in running, as a global criterion of evaluating the gear quality, is influenced by the eccentricity of the basic circles of the toothed wheels.
• The eccentricity of a toothed wheel will tend to give a basic vibration amplitude modulation, generated at its tooth and harmonics gearing frequency, with and envelope period that correspond to the shaft rotation.
• The interpretation of the results obtained from the measurements has led to the formulation of some conclusions regarding the identification of the vibration sources.

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