Open gear train as a source of oscillations of the construction machines mechanisms

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Abstract. The article considers kinematic, force and parametric characteristics of an open cylindrical gear train of the rotation mechanism of an excavator as sources of oscillations of mechanisms with a tooth frequency. The author builds a two-mass dynamic model with an internal perturbation in the form of a variable gear ratio and taking damping into account. The paper shows that the main perturbations can be the difference between the basic pitches of the gear and the wheel and the error of the involute profile of the side surface of the teeth. These errors occur during the manufacture of a gear pair and due to wear of the teeth. The errors in the relative position of the gear and wheel teeth, the variable tooth hardness with an overlap ratio greater than one, friction in the engagement for open gear train of a large module and manufactured by 8 ... 9 degrees of accuracy, play a significantly smaller role in the excitation of oscillations. The effect of the considered sources of oscillations should be taken into account in the resonant modes of the mechanism operation.

1. Introduction

Open gear trains are widely used in the mechanisms of rotation of construction excavators and cranes, as well as in some models of concrete mixers, mills and other similar machines. Questions of the dynamics of construction machines and their mechanisms remain relevant [1]. The peculiarity of open gear trains is that usually they do not have a casing that isolates the transmission from the external environment; the gear ratio is usually more than recommended for one pair of wheels of closed gear circuits; the engagement module is relatively large: it can be much larger than 4 ... 5 mm (for powerful machines up to 60 ... 80 mm); wheel diameter over 1000 mm; degree of accuracy (kinematic, smoothness of work and contact patch) is usually no better than 8 ... 9th; It is difficult to ensure normal transmission lubrication conditions. Most often, open gears are made with an involute tooth profile, which is important due to the constructive absence of a single frame for gear and wheel bearings (low sensitivity of the involute gear to small errors in the center distance). All these features are largely interrelated and quite often one follows from the other. It is especially relevant for the accuracy and wear of open gear trains. In a closed drive of construction machines and equipment, gear trains may also have a gearing module of at least 5 mm [2]. The issues of this article may be interesting for such mechanisms as well.
There are many analytical and experimental works [3, 4, 5, 6, 7, 8, 9, 10] on the dynamics of gear trains which describe and identify main causes of dynamic loads. These include: the presence of the difference between the basic pitches of the gear teeth and the wheel, the error of the involute tooth profile due to manufacturing errors and wear, variable stiffness of the gearing with one-pair and two-pair gearing, friction in the gearing, the variability of the instantaneous gear ratio. It is also possible to note the studies indicating the nonlinearity of oscillations of gears and their role as an internal vibration exciter of oscillations of the entire mechanism.

This paper aims to study an open gear train as an internal source of oscillations in the drive mechanisms of machines of the construction industry.

2. The main part
We conducted this study on the example of the mechanism of rotation of the excavator on a two-mass dynamic model (Figure 1) with elasticity, damping and internal disturbance $i$: kinematic, force or parametric, in different modes of motion. This model is a non-free non-conservative holonomic system with a stationary kinematic connection. Let us explain the possible internal perturbations for the system considered in the study.

We took into account the kinematic perturbation from the action of the difference between the basic pitches of the gear and wheel, the error of the involute tooth profile, non-parallelism and skewing of the relative position of the axes of the gear and the open gear train wheel. Force perturbation was investigated by the influence of friction force in the gearing. Parametric perturbation was represented by a variable stiffness of the link when the overlap coefficient is greater than one.

2.1. Kinematic perturbation

Figure 2 shows:

- $\phi_1$ - edge gearing angle, counted on the angle of rotation of the gear $\phi_1$; $\phi_{1e}$ - the end of the rotation on the angular pitch of the gear teeth; $f_{a1}$ - the greatest error of the gear ratio at edge gearing; $f_{a2}$ - gear ratio error when contacting involute profile sections; $\tau_1$ - angular tooth pitch; $i$ - gear ratio; $U_c$ - average per wheel rotation gear ratio; the dotted line of the graph shows the change in gear.
ratio at \( f_{pb} < 0 \), the solid line – at \( -f_{pb} > 0 \). Figure 2 graphs are built without regard to compliance of the elements involved in the transfer of load.

The measurements of the basic pitch of the gear wheels of a large module of 10 and 36 mm showed that for this wheel the error of the main pitch almost always retained its positivity or negativity.

![Figure 2](image.png)

**Figure 2.** The change in gear ratio due to the difference of the basic pitch of gear teeth and wheels \( f_{pb} \).

The error of the involute teeth profile also leads to a variable instantaneous gear ratio and a corresponding perturbation of the system. For excavators and other machines operating in a dusty atmosphere, the distortion of the side surface profile of the teeth of open gears due to wear may be significantly greater than manufacturing tolerances. We can note that the edges of the teeth wear out first. This leads to an increase in the basic pitch of the wheels and a corresponding disturbance. Experiments show that fluctuations can increase by 1.5 ... 2 times with worn teeth of gearing wheels compared to the new teeth. At the same time, there is evidence that the wear of the teeth has almost no effect on the magnitude of oscillations [11]. The effect of the gear manufacturing errors on the smoothness of the transmission is considered in [12].

We assessed the non-parallelism and skewness of the relative position of the axles of the gear and the wheel of the open gear train using the error of the gear ratio relative to the gear number of the transmission. Numerical estimates of this error for 12 degrees of accuracy give values less than 0.0002. This value is significantly less than the relative error in the gear ratio resulting from the difference in the basic pitch of the gear teeth and the wheel. We made the similar conclusion when evaluating other violations of the relative position of the wheels of an open gear pair: displacement of the wheels along and across the center line, non-perpendicularity of the plane of the wheels to their axes and non-parallelism of the teeth of the spur gears to their axes.

The force perturbation of the friction force in the gear happens due to two factors: 1) in an involute spur transmission, the friction force in the engagement changes its direction when the point of contact of the teeth along the engagement line passes through the engagement pole; 2) during the movement of the point of contact of the teeth along the engagement line, the shoulder of the friction force action changes relative to the supports.

For excavator rotation mechanisms, the reduced flexural stiffness of the output gear shaft, taking into account the compliance of the supports, is usually an order of magnitude higher than the torsional stiffness of the mechanism. Therefore, we can consider the influence of friction forces only on torsional oscillations. We also assume that the coefficient of friction in the process of teeth engagement on the excitation of oscillations with a tooth frequency does not change significantly, and therefore we take the coefficient of friction \( f = \text{const} \) for this analysis. Another assumption is that the moment of friction in the gear supports is almost constant and quite small and can be neglected.

The effect of friction was taken into account by the relation of the gear ratio of an open gear pair by the moment \( i_M \) to the gear number of this pair \( U \). The gear ratio of the moment \( i_M = M_2 / M_1 \), where \( M_1 \)
and $M_2$ – moments occurring on the gear and the wheel, with the friction forces. We found these moments from the equilibrium condition of the gear and the wheel. We should also note that we performed an experiment on an excavator with two drives of the turning mechanism, when one gear and wheel were well lubricated and the second pair was without lubrication. As a result, there was practically no difference in the oscillations of the lubricated and non-lubricated teeth on the recording of moments on the shafts. A similar result is noted in [13].

Variable stiffness at one-pair and two-pair engagements can lead to parametric oscillations. In this case, the oscillations of open gears were described by the Hill equation. For the considered mechanisms, there is a sufficient level of the natural decrement of oscillations in the system and it does not lose stability. Numerous studies, for example [4], conclude that the average specific stiffness of a single-pair link is estimated by the value of $C_0 = 1390$ MPa and two-pair – $C_d = 2700$ MPa. However, these values and the corresponding calculations show that the stiffness of the teeth is more than 10 times greater than the stiffness of the shafts of the turning mechanism. Therefore, in the reduced stiffness $C$ of the calculated dynamic system (Figure 1), the stiffness of the engagement plays a small role and is not a source of oscillation.

Preliminary studies of the dynamics of the rotation mechanism of the excavator show that it is characterized by a repetitive mode of acceleration-braking-stopping. In this mode, when a rotary platform accelerates, a transition resonance may occur in the rotation mechanism. A resonance, if it occurs, does not have time to develop to significant values. And noticeable fluctuations in the turning platform accelerates, a transition resonance may occur in the rotation mechanism. A resonance, if it occurs, does not have time to develop to significant values. And noticeable fluctuations in the turning mechanism occur only at resonance. Therefore, the main studies were performed for the resonant mode, for which the single-frequency model of small oscillations is justified with obligatory consideration of damping. The perturbation from the gear transmission was represented by the variable gear ratio according to figure 2. We assumed that the disturbance amplitude from the difference of the basic pitch is equal to the disturbance amplitude from the teeth profile errors and these disturbances were represented by the two-term decomposition of the curve in figure 2 in a Fourier series. The nature of the equations in dimensionless coordinates was:

$$
\begin{align*}
\ddot{x}_1 + b\dot{x}_1 + x_1 &= -\mu \cdot \phi(x_1, x_2, \dot{x}_1, \dot{x}_2, \gamma, b) + a_1 \\
\ddot{x}_2 &= \mu a_2 \cdot \phi(x_1, x_2, \dot{x}_1, \dot{x}_2, \gamma, b) + a_3,
\end{align*}
$$

where $\phi(x_1, x_2, \dot{x}_1, \dot{x}_2, \gamma, b)$ - a function depending on the coordinates $x_1$ and $x_2$, their velocities, the damping properties of the system and the phase of the perturbation $\gamma$, $\mu$ – relative amplitude of kinematic perturbation, $a_1, a_2, a_3$ – constant components depending on the inertial, stiffness and force (driving moments and resistance) system parameters.

We studied the nonlinear equations of motion by graphic methods in the stroboscopic phase plane [14].

3. Results

The results of studies of the adopted dynamic model are quite logical and consistent with the results of field tests. For example: 1) in relation to the mechanism of rotation of the excavator with internal kinematic perturbation, the oscillations decrease after reaching a maximum; 2) the maximum amplitude of the arising oscillations depends on the amplitude of the internal kinematic perturbation, the ratio of the oscillating masses and the magnitude of the attenuation coefficient 3) depending on the magnitude of the attenuation coefficient, the oscillations can take the form of beats or be almost periodic (as one of the masses tends to infinity, the oscillations become periodic); 4) resonant modes of operation should be avoided: the main forced oscillations occur with the tooth frequency of an open gear; 5) with existing norms on the kinematic accuracy of large-module tooth wheels, the main kinematic perturbation is the difference between the basic pitch of the engaging wheels and the error in the profile of the side surface of involute teeth; 6) if we consider the difference between the basic pitch of the engaging wheels as the main cause of the kinematic perturbation, then the practical overlap ratio of the open gear pair must
exceed one by at least one on the relative angle of rotation of the wheels with the edge gearing; 7) studies have shown that for single-bucket excavators, when the reduced value of the moment of inertia of the turntable is more than twice the inertia moment of the rotor of the engine, the excitation of resonant oscillations only by friction forces is unrealistic (the estimate was carried out with a friction coefficient of not more than 0.23, a gear ratio of 10 and logarithmic decrement of oscillations on average 0.23). Design-wise, the reduction of possible resonant oscillations can be obtained by using special dampers, vibration absorbers or due to the characteristics of the engine.

4. Conclusion

Usually, the issues of the dynamics of gear wheels are solved by modeling methods, for example, in the works [6, 7, 15, 16]. In this article, we chose an original model, which along with simplicity has certain universal properties for studying oscillations of mechanisms with gear wheels. Recommendations to reduce resonant oscillations relate to improving the accuracy of manufacturing gears, damping oscillations in one way or another, and changing the shape of teeth, for example, using flanking, non-involute tooth profile and non-circular gear wheels [17].

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