Evaluation of the influence of lubricant quality on dynamic performance and resource of involute gears

S A Polyakov\textsuperscript{1,2}, E M Kuleshova\textsuperscript{1}, L I Kuksenova\textsuperscript{2,1} and A V Medovshchikov\textsuperscript{1}

\textsuperscript{1}Bauman Moscow State Technical University, Moscow, 105005 Russia
\textsuperscript{2}Institute of Machines Science, named after A A Blagonravov, Russian Academy of Sciences, Moscow, 101990 Russia

serpol50z@rambler.ru, lkukc@mail.ru, kuleshova.em@mail.ru, alexmed@bmstu.ru

Abstract. The article deals with the relationship between the occurrence of self-oscillations and the wear process in involute gears. The dependence of the amplitude of the torque fluctuations on the high-speed shaft of the involute gears on the type of lubricant is presented. The calculation of the transmission service life is given, taking into account the rate of growth of the dynamic coefficient.

1. Introduction
The service life limitation of involute gears can occur due to wear, due to the fact that wear leads to a change in the shape of the teeth, an increase in gaps and a significant deterioration in the dynamic performance of the transmission [1]. Therefore, the consideration of involute gears for wear resistance is an urgent task.

Dynamic parameters of involute gears will be defined in this article to assess the causes of wear and its permissible limits when using different compositions of lubricants.

2. Methodology
The service life of an involute gear according to the wear criterion can be determined using the formula [2]

$$T = \frac{h_{\text{lim}}}{\left(\Delta h_1 + \frac{\Delta h_2}{U}\right)}$$

where $h_{\text{lim}}$ — maximum permissible wear, $U$ — gear ratio, $\Delta h_1, \Delta h_2$ — values of gear and wheel wear.

The key value among the listed values is the maximum permissible wear $h_{\text{lim}}$, for which there are no methodological grounds for choosing a specific value for involute gears. This value is limited by the growth of the dynamic coefficient $K_d$, which is traditionally used to evaluate the dynamism of drives. The coefficient of dynamism depends, all other things being equal, on the gap in the engagement, which is associated with the wear process almost linearly. In this paper, we will show how the introduction of a film-forming additive to a lubricant affects the change in the dependence of the dynamic coefficient on the load.

The dynamic coefficient can be determined in accordance with [3] using the formula
\[ K_d = 1 + \Delta T / T_n = 1 + \left( C_f / T_n \right) \sum A_i, \quad (1) \]

where \( T_n \) is the nominal torque, \( \Delta T \) is the excess of the nominal torque, \( C_f \) is the coefficient of torsional stiffness of the transmission, and \( A_i \) is the amplitude of torsional vibrations of the transmission. The traditional understanding of this value is that the increment of the torque \( \Delta T \) occurs as a result of changes in the external load. However, such an overload should generally be considered as a random short-term deviation, and the nominal torque should be selected as the maximum permissible long-term current.

The most interesting is the behavior of the torque on the high-speed shaft, which plays the role of the friction force in relation to the braking torque. The dynamics of the torque on a high-speed shaft will depend significantly on the friction force in contact, thus on the internal dynamics of the transmission.

The dynamic coefficient for a high-speed shaft is proposed in [4]. It can be entered as the dynamic coefficient of a high-speed shaft, represented as

\[ K_d = 1 + \Delta T_{em}, \quad (2) \]

where \( \Delta T_{em} \) is the absolute value of the increase in the tensile moment during vibrations, which, as shown in [4], is formed mainly due to self-oscillations, correlated with the value of \( T_{tor} \), which is the current value of the braking (loading) moment at the reached loading stage. Formula (2) is a generalization of (1) for the case of a changing value of the braking torque and shows how the amplitude of vibrations can change due to an increase in the braking torque, including in the case of exceeding the nominal torque. The proposal of this formula is related to the phenomenon of self-oscillations that occur and may increase due to an increase in the friction force in the transmission contact due to an increase in the braking torque. As was shown in the study of self-oscillations, it is convenient to use the fast Fourier transform method to analyze the frequency response function (FRF).

The use of the relative amplitude (\( \Delta T_{em}, \% \)), that determines the dynamic coefficient of the high-speed shaft in accordance with the formula (1) not only evaluates the dynamism of the transmission with different lubricants, but also ranks these materials by their extreme pressure action, depending on the value of the loading moment, with a given maximum dynamic coefficient.

3. Results

Thus, we obtained distributions of the amplitude of the high-speed shaft moment vibrations over the vibration frequencies when using different oils (figures (1-4)).

**Figure 1.** Dependence of the amplitude of the moment fluctuations of the high-speed shaft of the involute gear on the frequency when using mineral oil \( (T_{tor} = 2 \text{ N·m}) \).

**Figure 2.** Dependence of the amplitude of the moment fluctuations of the high-speed shaft of the involute gear on the frequency when using mineral oil \( (T_{tor} = 4 \text{ N·m}) \).
Figure 3. Dependence of the amplitude of the moment fluctuations of the high-speed shaft of the involute gear on the frequency when using mineral oil with the Striboil alloying element ($T_{tor} = 2 \text{ N} \cdot \text{m}$).

Figure 4. Dependence of the amplitude of the moment fluctuations of the high-speed shaft of the involute gear on the frequency when using mineral oil with the Striboil alloying element ($T_{tor} = 4 \text{ N} \cdot \text{m}$).

Analysis of the dependence of the amplitude of the moment fluctuations on the high-speed shaft on the braking moment allowed us to construct the graph shown in figure 5.

Figure 5. Analysis of the dependence of the amplitude of the moment fluctuations on the high-speed shaft on the braking moment.

$$T_{em}' = \frac{T_{em}}{T_{max}} \times 100\%,$$

where $T_{max}$ corresponds to the maximum value of the magnitude of the moment oscillation.

The graph shows, firstly, that the growth of the brake torque on the fast shaft, expressed in relative units, is growing. As the load increases, the influence of tangential slippage and sliding friction in the gearing grows, hence this phenomenon is consistent with the Hercy–Striebek diagram due to the fact that the $T_{em}$ corresponds to the friction force [5] (as the load increases, the sliding friction coefficient grows exponentially under boundary lubrication conditions). Secondly, as the antifriction properties of the lubricant increase, the $T_{em}$ amplitude decreases, so, in accordance with the formula (2), the dynamic coefficient of the high-speed shaft decreases, which, in turn, leads to an increase in the maximum permissible wear and an increase in the service life according to the wear criterion.

The data presented in figure 5 allowed us to construct analytical dependences of the dynamic coefficient on the load (figure 6).

Figure 6. Dependence of the dynamic coefficient of the high-speed shaft on the braking torque.
We can calculate the service life of an involute gear using the dependencies shown in figure 6, taking into account the growth of the dynamic coefficient. Due to the fact that the value of the maximum allowable wear varies inversely with the growth of $K_d$, we present the proposed dependence

$$T = \frac{h_{\text{lim}}}{\left(\Delta h + \frac{\Delta h_1}{U}\right)\nu K_d}$$

Here are examples of calculating the service life of an involute gear with a braking torque of 2.47 N·m according to the standard method and the method that takes into account the growth of $K_d$ when using clear mineral oil and mineral oil with the Striboil alloying element. Typical values of the parameters of an involute gear: number of teeth $z_1=20$; $z_2=66$; modulus $m=1.5$; $h_{\text{lim}}=0.6\cdot m=0.6\cdot 1.5=0.9$ mm. Wear rate when using mineral oil $J=10^{-8}$, when using additives $J=10^{-10}$.

The calculation results are shown in table 1.

| Calculation formula | $T$, hour (mineral oil) | $T$, hour (mineral oil with the Striboi alloying element) |
|---------------------|-------------------------|--------------------------------------------------------|
| $T = \frac{h_{\text{lim}}}{\left(\Delta h + \frac{\Delta h_1}{U}\right)\nu K_d}$ | 3453 | 345300 |
| $T = \frac{h_{\text{lim}}}{\left(\Delta h + \frac{\Delta h_1}{U}\right)\nu}$ | 302 | 265615 |

Calculations show that the actual value of the service life is slightly lower using the method that takes into account changes in the rate of growth of the dynamic coefficient than using the standard method. The service life is low without the use of a film-forming additive, and it may not meet regulatory requirements, due to the fact that for such transmissions, the service life is usually at least 10 thousand hours. The service life is significantly increased due to the introduction of additives. This phenomenon is a result of an increase in $K_d$, which leads to a proportional increase in the actual load and a subsequent increase in the wear rate.

4. Conclusions
The dependence of the amplitude of the torque fluctuations on the high-speed shaft of an involute gear on the type of lubricant is revealed. It is shown that as the antifriction properties of the lubricant increase, the magnitude of the amplitude decreases, that leads to a decrease in $K_d$ and, consequently, to an increase in the service life according to the wear criterion, this is confirmed by calculations.

References
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