Features of the Relationship Between Vibration, Lubrication and Noise of Gears

V.L. Basinyuk, V.E. Starzhinsky, A.I. Mardasevich, S.V. Shil'ko and E.M. Petrokovets

Abstract The realization of a method of a complex estimation of the manufacture and assembly quality of gear-based mechanical drives for parameters of vibration, noise, and lubricating film thickness using the amplitude spectra analysis is described.

Keywords Mechanical drive · Gears · Quality criteria · Vibration · Noise

1 Introduction

Up to now, gears have been one of the most common elements of mobile transmission technology and process equipment. Moreover, they are constantly being improved in almost all industrialized countries. This is due to, on the one hand, the technological and methodological software computation that has been developed, design, manufacture, production quality control and monitoring of the technical state of gears during operation, and on the other hand, to the relatively low cost of manufacturing a sufficiently large resource. The use of modern materials and hardening technologies, as a rule, allows us to provide the required parameters of reliability and technical and economic indexes of gears.

The existing approaches to assessing the quality of mechanical drives include the monitoring and verification of compliance with the technical requirements of the results of design, manufacturing and assembly of the basic components. The complex criteria of the drive quality are:

V.L. Basinyuk · A.I. Mardasevich
Joint Institute of Mechanical Engineering NASB, Minsk, Belarus
e-mail: vladbas@mail.ru

V.E. Starzhinsky (✉) · S.V. Shil'ko · E.M. Petrokovets
Metal-Polymer Research Institute NASB, Gomel, Belarus
e-mail: star_mpri@mail.ru

V. Goldfarb et al. (eds.), *Advanced Gear Engineering*,
Mechanisms and Machine Science 51, DOI 10.1007/978-3-319-60399-5_10

– the vibration generated with the drive, characterized not only by the correctness of the design, precision of the manufacturing and assembly of the transmission, but also, to a certain extent, its real dynamic loading [1, 2];
– the noise emitted from the drive, the parameters of which characterize the above indicators, as well as the correctness of the choice of material, the housing arrangement, and the parameters of the related structural elements and compliance requirements of environmental safety.

In some cases, the noise parameters characterize the conditions of the frictional interaction in terms of gear meshing [3], including the thickness of the lubricating film, which has a direct impact on the gear's lifetime [4–6] and the noise that it generates. It is worth noting the following [3]:

– the vibration and noise are interconnected, but the influence of tooth friction on them is different;
– the vibration parameters are most sensitive to longitudinal oscillations, including resonant ones, taking into account the transmission of vibrations of the shaft, bearing units and housing in the primary transducer (usually the piezoelectric);
– the parameters of sound pressure are most sensitive to the longitudinal and torsional oscillations, including the resonant ones, which are seen in very limited numbers in the vibration, but have a significant impact on the frictional conditions of the teeth in some cases, taking into account the effect of the internal environment on the gearbox, the housing walls, bearing units and external air, or another medium;
– the assessment of the conditions of the gears' interaction on the lubricating film thickness [4, 7–9] does not fully reflect the lubrication effect on the gearing resource, since the ratios of the lubrication thickness to the geometric mean roughness in surfaces in contact [10] and surface topography [11–13] are important;
– the analysis of vibrations and noise usually does not so assess the important parameters for the evaluation of resources in compliance with the required structure and properties of raw materials, as well as the hardening treatment. However, the deviations of these parameters can be characterized by the elevated change gradients of vibration and noise in operation, relative to the predictable ones, to some extent, caused by higher wear, compared with a predictable one and/or contact chipping friction surfaces of the teeth in operation [3].

The goal of the paper is to establish the features and the relationship of parameters of the amplitude spectra of the vibration, noise and lubricating film thickness, as well as the effect of the functioning modes of the gear on the thickness of the lubricating film in the process of engagement.

2 Research Method

A testing unit with an open power circuit was used for the experimental studies. Its design ensures the smallest possible number of elements of the kinematic chain and components, vibrations of which could distort the amplitude spectra of the parameters studied. The test had a spur gear transmission ratio equal to the unit, i.e., $z_1 = z_2 = 40$, and the module $m = 3$ mm, which allowed us to reduce the variations in the dynamic loading of the individual pairs of teeth due to the random nature of the combination of the pinion and gear base pitch errors at each subsequent complete revolution. Gears were mounted on shafts by cylindrical surfaces with a torque transmission by keyed joint. The shafts were installed in the housing with sufficient hard-row tapered roller bearings.

The casing of the test gear was carried out using a window covered with a thin plate made of PMMA. It provided a low level of noise absorption and distortion, resulting in the propagation of sound from the gear to the measuring microphone.

The research into the amplitude spectra was performed at angular velocities of rotation $\omega = 50$, 100, 150 and 200 rad/s and a torque $T = 0$, 100 and 200 Nm. The following signals of primary transducers with reference to real time scale were recorded:

– the voltage of the piezoelectric sensor U_{H} (mV) by which the vibration accelerations of the gear bearing support were recorded;
– the voltage of the microphone U_{L} (mV) by which the sound waves were recorded. They characterize the noise generated by the gear and stand elements;
– the voltage U_{h} (mV) (for the stabilized current $I = 1.5$ A), starting from which, using the calibration curves obtained previously, the lubricating film thickness was recorded in the gear meshing.

Since these voltages and their corresponding physical quantities are connected by linear relations, the frequency content of their spectra is identical and can be used to analyze the results.

The typical character of the relative change in the thickness of the lubricating film on the engagement phase (ratio of the current value of the lubricating film thickness h to the lubricating film thickness in the pitch point h_0) is shown in Fig. 1 at various speeds and loading modes of the tested gear, and the amplitude spectra of the thickness fluctuations of the lubricating film, vibrations and noise—in Fig. 2 [3].

3 Analysis of the Results

The thickness of the lubricating film, except for the idle run mode, raises the power function of 0.6–0.8 by increasing the circumferential speed and the associated rolling speed. This fact correlates well with the results of studies [8, 11–13]. At the same time, the certain laws of change h were established as being slightly different from the results in these studies. For example, the dynamic interaction of the teeth at engage and disengage is significantly affected as a destabilizing lubricating film of the edge or median impacts when entering or leaving the teeth out of engagement on the real thickness h in the gearing. Variations h reach the maximum values

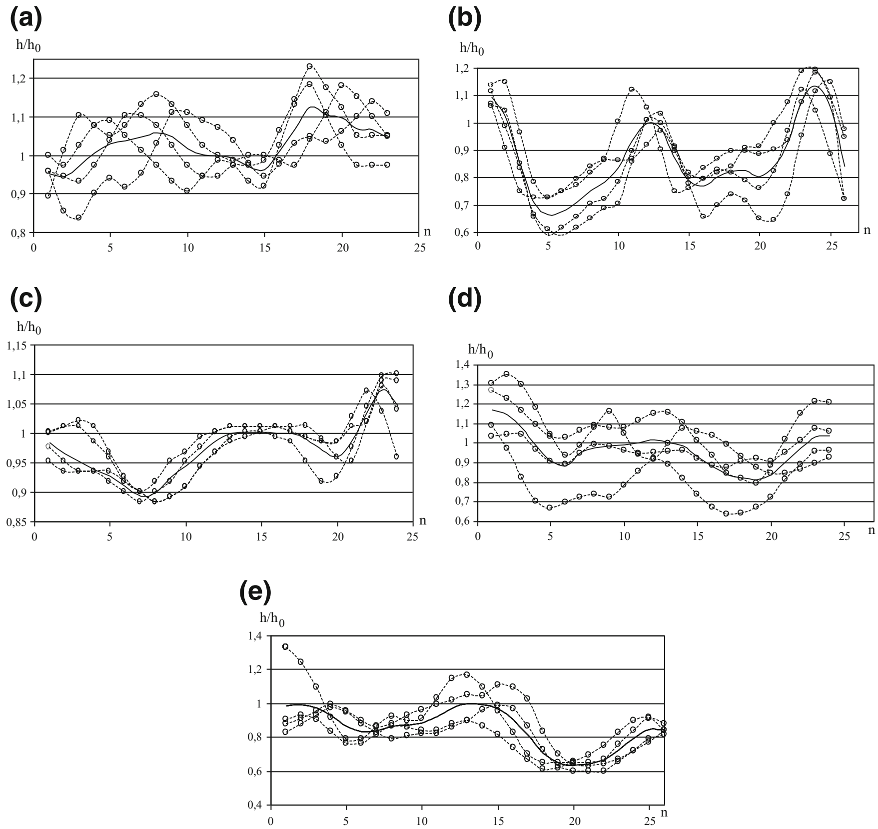

Fig. 1 The typical character of the relative change in the thickness of the lubricating film on the engagement phase at different speeds and load conditions of functioning of the test gear pair: **a** V_0 = 3 m/s, idling; **b** V_0 = 3 m/s, T = 100 Nm (σ_{H1} = 800 MPa); **c** V_0 = 3 m/s, T = 200 Nm, (σ_{H1} = 1150 MPa); **d** V_0 = 6 m/s, T = 100 Nm (σ_{H1} = 800 MPa); **e** V_0 = 9 m/s, T = 100 Nm (σ_{H1} = 800 MPa)

(Fig. 1a–d) in areas of tooth changeover. While, as noted in [8], the lubricant comprised between the teeth is not squeezed out instantaneously when entering, the new pair of teeth engages or disengages the upstream rotational pair of teeth.

This not only provided an acceptable mode of lubrication of the contact surfaces, but also contributed to the damping of the edge and middle impacts to some extent. In general, changes in the lubricating film thickness in these regions and along the line of engagement can be characterized by the following features.

At lightly loaded slow-speed transmission mode for large value *h*, its variation along the line of engagement was of a pronounced stochastic character. The minimum values *h* were recorded in tooth changeover and pitch point after a change in the opposite direction of the sliding speed of working surfaces of the teeth. The maximum value *h* corresponds to the engagement portion of the line between these

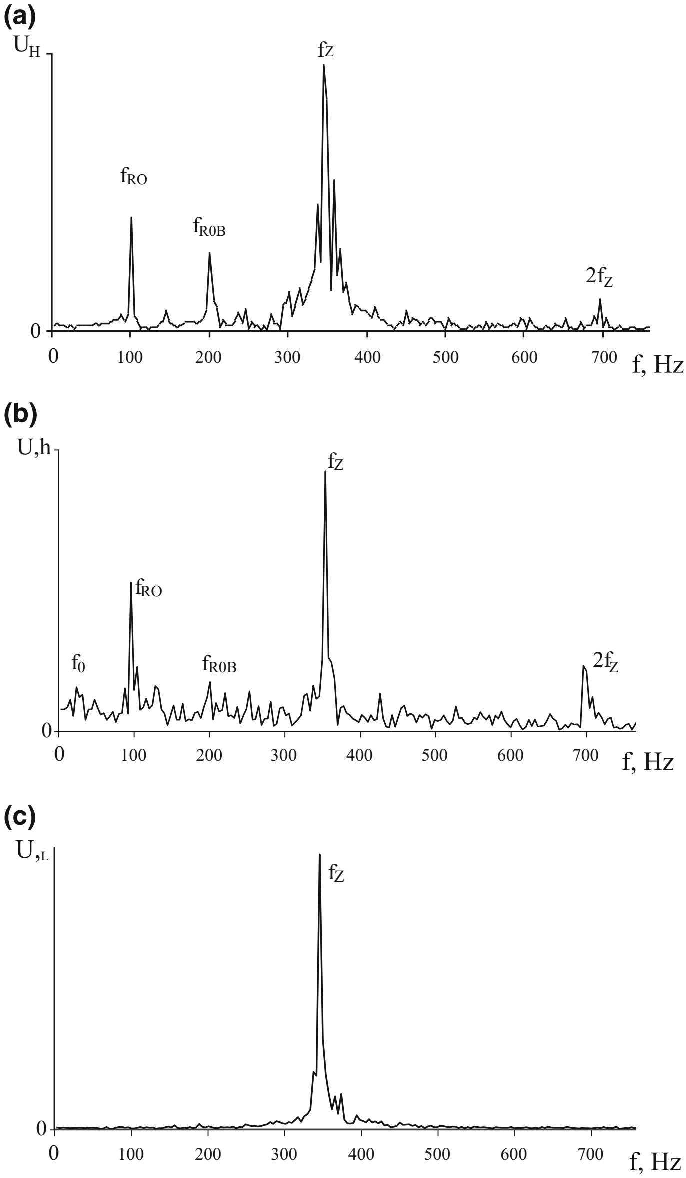

Fig. 2 Typical types of amplitude spectra of the noise oscillation (**a**), the lubricating film thickness (**b**) and the vibrations (**c**). f_0 is the negotiable rate, f_{R0} is the natural frequency of torsional vibrations, f_{R0B} is the natural frequency of the stand elements associated with gears, f_Z is the tooth mesh frequency

two points, which is characterized by a relatively stable interaction of the teeth. Minimum average values h were different from the lubricating film thickness at the pitch point by only 5–10%.

Change in the thickness of the lubricating film was close to the deterministic calculated value with respect to the loaded and heavy-loaded conditions in double tooth meshing [3]. The greatest value h was achieved at the pitch point, as well as at the entry and exit zones of tooth meshing. The "extrusion" time of the excess lubricant from the meshing zone was t = (0.06–0.08) t_T = (0.1–0.12) t_p.

The value h is greater than the value of the lubricating film thickness h_0 at the pitch point by 1.1–1.2 times at tooth changeover. Zones of minimum lubricating film thickness with these modes are arranged between the pitch and the entrance and exit points of engagement of the teeth, where the average values h were smaller than those of the lubricating film thickness at the pitch of 30–40%.

It should be noted that the change in the thickness of the lubricating film differed somewhat from that calculated by increasing the circumferential speed and the associated rolling velocity. This fact was also observed in [8]. The value h was first increased and then stabilized at a certain level and even slightly decreased, while it began to rise again at $V_0 \geq 12$ m/s with increasing circumferential speed at contact stresses σ_{H1} = 800 MPa and σ_{H2} = 1150 MPa. This effect can be explained by the influence on the lubrication thickness h of dynamic and static loads, as well as the high-speed and thermal modes of interaction between the contacting surfaces. Theoretical analysis of the influence of these factors on the value h in [5, 6] showed the possibility of such a change in the thickness of the lubricating film at a level similar to that when changing gear operation modes.

The value h along the lines of engagement was varied stochastically with minor deviations from the mean at high circumferential speeds (V_0 > 10 m/s). Standard deviations σ_h and coefficients of variation v of the lubricating film thickness were non-linearly dependent on the circumferential speed and the contact pressures. In all cases, an increase in the speed V_0 contributed to the increase of these indicators. At idling, the value σ_h increased from 0.2–0.28 to 0.35–0.44 µm, v from 7–10 to 18–22%, and the value v increased in power function at a nearly linear increase σ_h. The deviation σ_h increased from 0.22–0.3 to 0.4–0.5 µm, but up to V_0 = 6 m/s, the value v decreased from 20–23 to 16–21%, and then increased to 22–34% (T = 100 Nm) in the gears loaded by torque (T = 100 Nm). Trends of change σ_h and v were close to the above.

Standard deviation σ_h and coefficient of variation v were monotonically increased, respectively, from 0.09–0.16 to 0.32–0.36 µm and from 9–0.16 to 17–0.20% in heavy-loaded transmissions (T = 200 Nm), stabilizing at the top level. This is associated with a stationary mode of contact interaction and consistency in the thickness of the lubricating film. The increase in amplitude values of forced oscillation and natural vibrations of gears [3] is significantly affected by the values σ_h and v. This is due to the increase in dynamic loading level at the edge or median base impacts.

The nature of the change in the thickness of the lubricating film on the engagement phases is significantly changed by varying the circumferential speed and gear loading. Variations in the thickness are reduced with increasing circumferential speed and loading, but the process of their change comes closer to that of the stochastic one.

Thus, the methodological approaches, built on the use of deterministic algorithms for determination of h, are limited. They do not take into account the variations of combinations of base pitch errors in transmission ratios, being unequal to the unit, the variations within manufacturing deviations in the confines of tolerances on total profile error, the stochastic nature of the conditions of the dynamic interaction between the teeth and the amplitudes of the oscillations generated, or the influence of the topography of the interacting surfaces. This makes it difficult to use the dependences obtained on the basis of these approaches for the detailed calculation of the thickness of the lubricating film in the absence of comprehensive experimental data on specific groups of transmission, modes of operation and characteristics of lubricants.

The existing analytical dependences and calculation methods of the thickness of the lubricating film can be used to determine the approximate values of the lubricating film thickness h at the pitch point, as well as the evaluation of the viscosity of lubricants, speed, load and thermal modes of gearing on the thickness of the lubricating films and durability of the tooth-working surfaces. in many cases in the absence of such data. It is advisable in some cases to introduce the coefficient k_g, taking into account the effect of the dynamic nature of the tooth interaction, the natural and forced oscillations of the system and changes in the conditions of tooth interaction by moving the contact zone along the line of engagement [2].

Analysis of the amplitude spectra of noise, vibrations and oscillations of the lubricating film thickness showed the following.

The forced vibrations with a tooth frequency f_Z, caused by errors in the manufacture of the gear according to the smoothness norms and related fluctuations in stiffness along the engagement phase, were not only decisive under almost all speed and load conditions of the test gear pair, but also have the most significant effect on the spectra of noise, vibrations and changes in the lubricating film thickness. The only exception was the idle run, in which the greater amplitude was the second harmonic.

In the spectrum of noise, the most significant amplitude corresponds to tooth frequency f_Z at a relatively low angular speed of rotation and heavy loaded operation mode with a thin lubricating film. This confirms a correlation between the noise and frictional interaction of gears, to some extent.

In other cases, the forced longitudinal oscillations, as well as the mechanical torsional vibrations with the natural frequency of torsional vibrations which arise mainly due to a kinematic error in gear manufacturing, strongly influence the composition of the noise amplitude spectrum.

The tooth mesh frequency f_Z was basically the second largest and only under certain conditions—the first or third after frequency of oscillations having the frequency value f_{R0} of natural torsional vibrations [3] in the amplitude spectrum of the oscillation of the lubricating film. This fact indicates a significant influence of the dynamic processes in the teeth contact on the torsional vibrations and confirms the appropriateness of the account of this factor when choosing a lubricant and evaluating its impact on gear lifetime. Furthermore, the most similar compositions were the amplitude spectra of the noise and vibrations of the lubricating film, indicating the correlation between the noise and friction conditions in the gearing.

The forced vibrations with a tooth mesh frequency are due to an error primarily in the gear manufacturing for smoothness standards and the engagement rigidity periodically changing with a frequency f_Z. They have the most significant impact on all drive parameters, including the frictional characteristics, as well as the noise and vibrations generated in the meshing process. The degree of this effect was not only dependent on the accuracy of the gears, but also the mode of their operation and the inertial and stiffness parameters of the mechanical system in radial and circumferential directions, as well as the associated natural torsional vibrations.

4 Conclusions

Mechanical vibrations and noise generated by gears, as well as the value of and change in the thickness of the lubricating film, have a significant impact on the resource. This is advisable to take into account when we develop the systems for monitoring the manufacture and assembly quality, as well as the residual lifetime to operation. The most effective one includes a comprehensive assessment of the quality of manufacturing and mechanical drive monitoring systems based on analysis of the parameters of noise, vibrations and lubricating film thickness. Tribotechnical parameters of the gear may be determined through comparative analysis of the vibration and noise parameters [3], in some cases, in connection with the use of non-standardized equipment for registration of the thickness of the lubricating film.

Acknowledgements The authors would like to acknowledge the Belarusian Republican Foundation for Fundamental Research (project T16R-202) for their support.

References

1. Basinyuk, Ya.V., Ishin, N.N., Basinyuk, V.L., Mardasevich, A.I.: Vibration monitoring of internal dynamic loading, accuracy parameters and wear of individual pairs of teeth by gearing. Mech. Eng. Autom. Comput. **4**(10), 48–53 (2001) (Vestnik of Brest State Technical University) (in Russian)
2. Starzhinsky, V.E., Shalobaev, E.V., Shil'ko, S.V., et al.: Elements of Drive Devices. Calculation, Design, Technology. Belaruskaya Navuka, Minsk (2012) (in Russian)
3. Basinyuk, V.L.: Dynamics, Lubrication and Noise of Gears. MPRI NASB, Belarus (2006) (in Russian)
4. Petrusevich, A.I.: The Main Conclusions of the Contact and Hydrodynamic Lubrication Theory. Publishing House AS USSR, DTS, vol. 2, pp. 209–233 (1951) (in Russian)

5. Van, K.L., Chzhen, Kh.S.: Numerical determination of dynamic stress, lubricating film thickness and surface temperature for spur gears. Part I. Research. In: Proceedings of American Society of Mechanical Engineers (Series: Design and Engineering Technology). Mir, vol. 1, pp. 81–92 (1981) (in Russian)
6. Van, K.L., Chzhen, Kh.S.: Numerical determination of dynamic stress, lubricating film thickness and surface temperature for spur gears. Part II. Results. In: Proceedings of American Society of Mechanical Engineers (Series: Design and Engineering Technology). Mir, vol. 1, pp. 92–99 (1981) (in Russian)
7. Kodnir, D.S., Ratner, I.D.: Contact and hydrodynamic calculation of gears. Reliab. Qual. Gears. **18-67-73**, 2–12 (1967) (in Russian)
8. Raiko, M.V.: Lubrication of Gears. Tekhnika (1970) (in Russian)
9. Skurka, I.S.: Elastohydrodynamic lubrication of rolling bearings. In: Proceedings of American Society of Mechanical Engineers (Series: Problems of Friction and Lubrication). Mir, vol. 2, pp. 110–121 (1970) (in Russian)
10. Kragelsky, I.V., Alisin, V. (eds.) Friction, Wear and Lubrication: Handbook. Mashinostroenie, Russia (1979) (in Russian)
11. Chzhu, D., Chzhen, Kh.S.: Influence of surface roughness on egd lubrication of point contacts. In: Proceedings of American Society of Mechanical Engineers (Series: Problems of Friction and Lubrication). Mir, vol. 4, pp. 36–43 (1988) (in Russian)
12. Tonder, K.: Effect of one-dimensional roughness of the inclined orientation of the microscopic irregularities in the hydrodynamic lubrication. part 2: moving rough surface. In: Proceedings of American Society of Mechanical Engineers (Series: Problems of Friction and Lubrication). Mir, vol. 3, pp. 103–107 (1988) (in Russian)
13. Tonder, K.: The Consequences of a number of common theories of roughness in the case of lubrication of surfaces with parallel ridges. In: Proceedings of American Society of Mechanical Engineers (Series: Problems of Friction and Lubrication). Mir, vol. 2, pp. 16–20 (1986) (in Russian)