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Improving Worm-Gear Performance by Optimal Lubricant Selection in Accelerated Tests

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Abstract—In tests of worm gears, their reliability and load capacity may be assessed in terms of the braking torques corresponding to maximum efficiency. Different lubricants are used in the tests.

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The energy losses of worm gears exceed those of cylindrical gears by at least 10–15%. However, with the same gear ratios, worm gears are more compact than cylindrical gears and require less metal. Therefore, reduction in the frictional losses of worm gears is of great interest. This is associated with increase in their reliability, since reduction in the mechanical losses boosts the load capacity and hence the reliability, as confirmed by experiments.

Russian worm gears are marked with the permissible output torque—that is, the load capacity corresponding to the standard gear life [1]. Thus, for the 5Ch80 gear (with an axial spacing of 80 mm), which is tested in the present work, the rated (permissible) output torque is 260 N m; that may be regarded as the gear's load capacity. If this value is exceeded at slow shaft speeds, the standard life may be reduced by 90%. For the 5Ch80 worm gear, the standard life is 10 000 h, as against 25 000 h for cylindrical gears [1]. In addition, exceeding the permissible output torque may result in the most hazardous type of failure: jamming.

The simplest method of increasing the resistance to jamming and reducing the energy consumption, without changing the design of the worm gear, is by careful selection of the lubricant. Assessment of the possible changes calls for the development of the corresponding experimental method. Existing test methods have numerous deficiencies.

The wear resistance and load capacity of worm gears may be assessed by various methods. For example, the test procedure outlined in [2] is prescribed in State Standard GOST R 50801–96 [1]. This approach permits assessment of the wear rate of a worm gear under a steady rated load. However, it makes no provision for accelerated tests or for evaluation of the main factor responsible for gear failure, which, as we know, is jamming at impermissible loads.

A load increasing at a specified rate is employed in tests to assess the load capacity of worm gears by the method in [3]. The load capacity is assumed to be the load at which jamming begins. The problem with this method is that no account is taken of the running-in stage. Because the estimate of the jamming load is made for a gear without any running-in, it will be much too low.

In the present work, we create a test procedure by which changes in the structural materials and lubricants may be ranked in terms energy efficiency and reliability. Our starting point is the method used for slip bearings and described in State Standard GOST 23.224–86 and in [4, 5]; we adapt this method for worm gears.

In the proposed method, we estimate the maximum permissible load and the permissible loading rates by means of a loading system with negative feedback, which permits correction of the magnitude and growth rate of the load (braking torque) on the basis of the change in torque at the gear's input (high-speed) shaft.

The proposed method takes into account that the running-in process is slower for worm gears than for slip bearings. We also note that the time corresponding to significant reduction in the frictional force may be considerably greater than the time assumed in [4, 5]. Accordingly, the tests of the worm gears are conducted in three stages.

1. Accelerated testing. A contact spot on the teeth of the worm gear that is at least 60% of the rated value is obtained. To this end, ordinary mineral oil is used in lapping of the gear teeth at relatively low loads (no more than 50% of the rated load).

2. Determination of the maximum short-term load using negative feedback. The load increases not only on account of the reduced frictional force but also when there is no growth over the period determined by

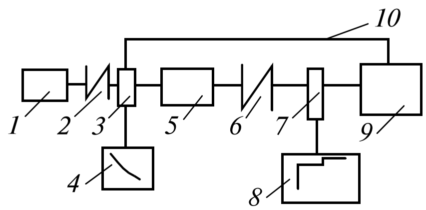

Fig. 1. System for accelerated tests of worm gears: (*1*) electric motor; (*2*) clutch; (*3*) instrument for torque measurement at the high-speed gear shaft; (*4*) device for real-time plotting of that torque; (*5*) gear system; (*6*) clutch; (*7*) instrument for torque measurement at the low-speed gear shaft; (*8*) device for real-time plotting of the torque at the loading system; (*9*) loading system and torque-control device; (*10*) feedback mechanism permitting torque regulation as a function of the torque on the high-speed shaft.

the method in [4]. The dependence of oscillation of the frictional force on the load is analyzed: with significant increase in amplitude of the frictional-force oscillation, further loading is stopped.

3. Determination of the long-term time dependences of the frictional force and wear rate. The frictional force is assessed on the basis of the torque on the electric motor. In loading, the torque on the slow-moving gear shaft is changed by means of an electromagnetic brake.

A special test bench has been developed for test simulations with kinematics similar to a worm gear and reproduction of the actual stress in the interaction of the frictional surfaces. This test bench includes not only components of the actual test bench for mechanical drives according to the procedure in [6] (components *1*, *2*, *5*, *6*, and *9* in Fig. 1) but also new components that ensure running-in of the gear in conditions at the edge of jamming (components *3*, *4*, *7*, *8*, and *10* in Fig. 1).

In the tests, we use the following characteristics: the frictional coefficient f; the efficiency η; the torques T_1 and T_2 at the high- and low-speed shafts, respectively; and the torque $T_{2\text{opt}}$ at the low-speed shaft corresponding to extrema of η and f.

For the worm gear, η consists of coefficients characterizing the operation of its components, including the bearings. Since these coefficients amount to only fractions of a percent of the total energy losses and are practically constant in the course of operation, we may determine the efficiency of the worm gear from the formula

$$\eta = T_2/(uT_1), \tag{1}$$

where u is the gear ratio.

Since the efficiency η basically determines the energy efficiency of gear operation, we assess η on the basis of the frictional angle φ for the given materials and lubricants

$$\eta = \tan\gamma/\tan(\gamma+\varphi),$$

where γ is the inclination of the worm gear's helical line. Thus, η is inversely proportional to f.

We now consider the test procedure in the second stage.

After lapping, the worm gear is placed in the gear system; oil is poured in; the electric motor is turned on; and the maximum permissible load on the gear system is determined by stepwise loading, with variation in magnitude and duration of the braking torque. The second stage of the test is concluded; the gear system is unloaded; and the maximum permissible long-term braking torque $T_{2\text{opt}}$ is determined from the maximum efficiency given by Eq. (1). That approach is equivalent to its determination from the minimum of the frictional coefficient according to State Standard GOST 23.224–86. This is the key characteristic of the gear system, since it expresses its energy efficiency relative to a standard gear with known reliability (260 N m, in the present case) and expresses its reliability in terms of jamming and the wear resistance determined by the critical load (the torque $T_{2\text{opt}}$). As a rule, when using a standard lubricant, $T_{2\text{opt}}$ is equal to the rated torque, as confirmed experimentally in [7]. The wear resistance of the worm gear may be assessed in tests at this load in the third stage.

In Fig. 2, we show the primary experimental data obtained in the second stage of the test. In the second stage, the dependence of η on the load torque T_2 is determined for three lubricants; the results are shown in Fig. 3 and in the table.

Energy efficiency and reliability of gear in tests with different lubricants

Lubricant	$T^*_{2\text{opt}}$, N m	$T_{2\text{opt}}/T_{\text{rat}}$, %	$\eta_{\max}$, %
TM 5-18 mineral oil, with additives (OOO Pushkinskii Zavod)	270	1.01	65
TM 5-18 synthetic oil with additives (OOO Tatneft'–Nizhnekamskneftekhim)	320	1.20	78
TM 5-18 mineral oil, with additives and Striboil nanocomponent (OOO Pushkinskii Zavod)	340	1.30	86

Note. The torque $T_{2\text{opt}}$ corresponds to $\eta_{\max}$.

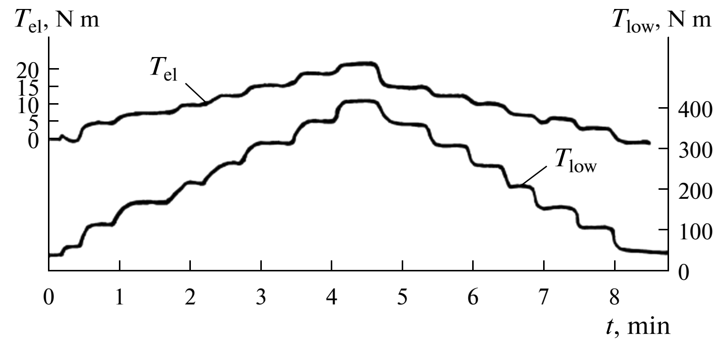

Fig. 2. Real-time variation of the torque T_{el} at the electric-motor shaft and the torque T_{low} at the low-speed gear shaft, in the second stage of the test.

For TM 5-18 mineral oil, the rated torque is equal to T_{2opt}, as noted in [7]. As we see in Fig. 3, the maximum efficiency η_{max} is shifted to higher T_{2opt} with improvement in lubricant quality. This confirms that the reliability of the gear increases with increase in its energy efficiency.

As we see in the table, the use of synthetic lubricant increases the gear's load capacity by 20% [8]. On adding nanomodifier, the total increase in load capacity is 30%. The efficiency η increases accordingly, to 0.86, which significantly exceeds the standard value of 0.78 [1].

The functionality and hence the reliability of the gear is indicated by the maximum permissible load, which is compared with the torque T_{2opt} at the low-speed shaft corresponding to maximum efficiency (Fig. 3). Therefore, we may conclude that the careful selection of up-to-date lubricants permits increase in the energy efficiency and reliability of worm gears.

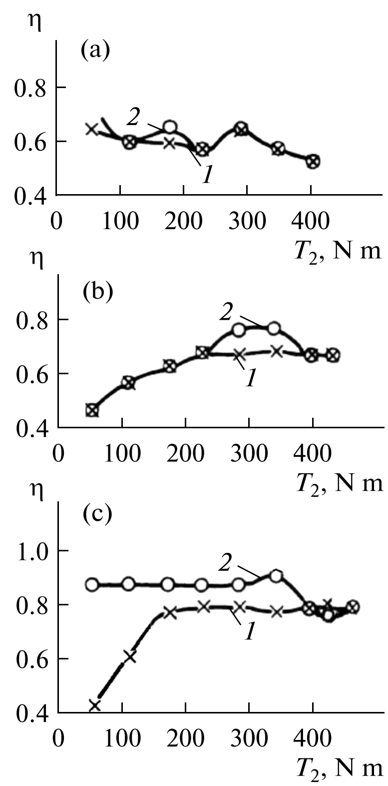

Fig. 3. Dependence of the efficiency η on the braking torque T_2 at the low-speed gear shaft with lubrication by mineral oil (a), synthetic oil (b), and mineral oil with Striboil nanomodified additive (c), in loading (*1*) and unloading (*2*) of the gear.

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