Thermo-metal-cladding of working surfaces of closed friction units of mobile systems

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Abstract. Modeling methods play a significant role in the operation, diagnostics, and prediction of the current state of various mechanical systems. The authors have developed fundamentally new approaches to solving problems of optimization, increasing the efficiency and competitiveness of nonlinear technical systems. Test facility investigations of closed friction units of mobile systems using a lubricant with the aluminum powder insertion were carried out. Test bench allows simulating a reciprocating motion. The analysis of research demonstrates the stabilizing effect of such a lubricant on the tribothermodynamics of the friction unit.

1. Introduction
The usage of modern technological equipment, the application of cutting-edge methods and methods for the production of structural and consumables, as well as the optimization of technological processes, significantly affect the development of modern industries. Modeling methods play a significant role in the operation, diagnostics, and forecasting of the current state of various mechanical systems. The authors of the article used the basis of available theoretical and experimental data in the field of dynamics of friction processes and research methods. They have developed fundamentally new approaches to solving problems of optimization, increasing the efficiency and competitiveness of nonlinear technical systems [1-6].

Thus, samples of the spline joint of the tail rotor drive transmission coupling of the Mi-26 helicopter were investigated using a test bench that allows simulating the reciprocating motion with the given load and velocity parameters.

2. Examination of model samples of splined connection of transmission coupling
In accordance with the method of physical and mathematical modeling [5…10], the load-velocity conditions for carrying out a model experiment were determined with a simplex of diameters of a full-scale splined joint and its model Cd = 1.62 and a scale of the loading moment CM = 196.8. The load during the reciprocating motion of the spline joint model Nm = 2100 N and the frequency of the reciprocating motion ωm = 44.73 Hz, which corresponded to the tractive effort torque of the full-scale system.

For research of model samples of the spline joint of the helicopter tail rotor drive transmission coupling, a friction pair (full-scale pair) (Figure 1) was chosen. There was a cup made of 38Kh2MYuA steel (Figure 1, a) and a tip made of 12Kh2N4A (Figure 1, b). There was a model pair (Figure 2) with
When the frequency of the reciprocating motion and the load moment on the friction unit were changed, the dynamics of frictional interaction was modeled using a laboratory test bench with normal pressure of 100 MPa when working with different lubricants: in the oil bath with a standard HG oil lubricant and in the oil bath with a lubricant added to aluminum powder.

During the research, we recorded the values of the trends in the friction coefficient both in statics and in dynamics, as well as the energy losses of the dynamic friction coefficient, dissipative losses of the dynamic friction coefficient and the dimensionless value of the damping coefficient in the most significant octave frequency ranges, and the dynamic quality criterion of a nonlinear technical system.

Figure 3 shows the trends in the stationary (steady-state) value of the friction coefficient and its value during natural oscillations which occur under the influence of forced oscillations.

The most significant value is the dynamic friction coefficient of the elastic and inertial interaction 1 (Figure 3), since it takes into account additional inertial disturbances from the frictional mechanical system. The frictional mechanical system significantly affects the approach of the contacting friction surfaces to each other, as well as the increase in normal tension and temperature.

The dynamic friction coefficient of the inertial impact 2 (Figure 3) significantly affects stability of the frictional bonds, stability of the frictional mechanical system, and deviation of the characteristics transient in time from the stationary motion trajectory.
Figure 3. Comparative analysis of the friction coefficient in the statics and dynamics of the friction unit functioning with samples

The dynamic friction coefficient of resistance forces 3 (Figure 3) determines the speed of operation of the transient characteristics in time, when the vector of resistance forces is oppositely directed to the velocity vector of the relative sliding of the contacting friction surfaces.

The dynamic friction coefficient, caused by the frictional self-oscillations 4 formed in the friction contact (Figure 3), significantly affects both the stability of the frictional bonds, the stability of the frictional mechanical system, and the deviation of the characteristics transient in time from the stationary motion trajectory.

The test data show (see Figure 3) that with an increase in the relative sliding speed within 0 to 22.5 Hz at 1350 min – 1, the inertial components of the dynamic friction coefficient are absent, and the dissipative ones are insignificant. Due to frictional self-oscillations, the components of the dynamic friction coefficient increase significantly with a rise in the relative sliding speed to the nominal values. The inertial impacts on the friction unit significantly increase when reaching the rated load-speed operating mode and before the completion of transient oscillations in the range from 360 s till 680 s. From 680 s till 1000 s of observations, the stationary friction mode takes place, which is characterized by a decrease in all comprising values of both the dynamic friction coefficient and its value after the end of transient oscillations.

At 1020 s of observation period, aluminum powder is inserted into the oil bath with HG oil lubricant. At the same time, an increase in low-frequency vibration and a significant decrease in the volumetric temperature of the oil bath are noticed. In the range of experiment time from 1020 s till 1500 s, transient forced vibrations occur in the frictional mechanical system that are associated with the restructuring of the tribo-characteristics of the friction unit functioning. From 1500 s till 1590 s of the experiment, the stabilization of all components of the friction coefficient is recorded, and in the time range from 1590 s till 1630 s, braking and stopping are implemented.

Figure 4 shows the trends in energy losses of the dynamic friction coefficient. Indeed, in the ranges from 280 s till 600 s, and from 1020 s till 1480 s of observation periods during transient oscillations, we
record significant losses of the dynamic friction coefficient. The stationary friction mode is characterized by the coincidence of the gradients of change in both the friction coefficient in stationary motion and the integral estimate of the energy losses of the dynamic friction coefficient. Further, there is a change in the dissipative losses of the dynamic friction coefficient of the mechanical system in octave frequency ranges [4,8].

Figure 5 shows the trends of dissipative losses of the dynamic friction coefficient in the most significant octave frequency ranges. The significance of the frequency ranges was assessed by the value of the Pearson correlation coefficient and testing the null hypothesis using the Student's test. It was revealed that the frequency ranges with geometric mean frequencies of 36 Hz (Cxy = 0.77); 72 Hz (Cxy = 0.54) and 18 Hz (Cxy = 0.5) are the most significant in terms of the degree of impact on the tribo-characteristics of the system. The frequency range of dissipative energy losses with a geometric mean of 128 Hz has almost no correlation with the friction coefficient in stationary motion (Cxy = 0.19), but has a significant effect during free oscillations caused by external influences.

![Figure 4. Comparative analysis of the friction coefficient f in static, and the integral value of energy losses fdyn](image)

When reaching the nominal friction mode from 0 s till 360 s of observation time, dissipation energy losses in the frequency range 11.2 – 22.4 Hz (1) decrease, since equilibrium roughness is formed. In a wide frequency range of 22.4 – 177.8 Hz, on the contrary, dissipation losses increase as the relative slip speed goes up.

Upon reaching the rated load-speed operating conditions, dissipation losses are almost stationary in 11.2 – 22.4 Hz frequency range (1); in 22.4 – 177.8 Hz frequency range, they decrease monotonically, which indicates the stabilization of friction processes. According to the graph in Figure 5, the stationary friction regime is fixed in the time range from 760 s till 800 s of observation time, which correlates with the results of data analysis in Figure 3 and 4.

It should be noted that the insertion of aluminum powder into the oil reservoir for 800 s of the experiment stabilizes the dynamics of the frictional mechanical system in 11.2 – 22.4 Hz (1) and 22.4 – 44.7 Hz frequency ranges (2), and it is characterized by the values of dynamic friction coefficient
0.000125 ± 0.00003 (0.09 ± 0.015). In frequency ranges 44.7 – 89.1 Hz (3) and 89.1 – 177.8 Hz (4), we observe a decrease in dissipation energy losses.

Figure 5. Comparative analysis of the friction coefficient f in static, and dissipative components of the dynamic coefficient of friction fdyn in octave frequency ranges

The analysis of the experiment shows that the insertion of aluminum powder into the HG oil lubricant improves the tribothermodynamics of the friction unit.

The results of determining the dimensionless value of the damping coefficient $\xi$ are presented in the most significant octave frequency bands (Figure 6). The graphs display that upon completion of transient oscillations, the damping coefficient $\xi$ is stationary in 0.71 - 22.4 Hz frequency ranges (1); in the frequency ranges 22.4 - 44.7 Hz (3) and 89.1 - 177, 8 Hz (4), it has a positive gradient of physical and mechanical properties, which indicates the stabilization of dynamic characteristics and an increase in the stability of the frictional mechanical system.

The insertion of aluminum powder into the HG oil lubricant reduces the speed of the system in 22.4 - 44.7 Hz frequency range (3). However, upon completion of transient oscillations, the value of the damping coefficient $\xi$ in all frequency bands is characterized by more stationary values.

3. Organization of monitoring of friction-mechanical system functioning

It is more convenient to monitor the functioning of a frictional mechanical system on the basis of some dynamic quality criterion which characterizes the simultaneous change in frequency, time and integral quality criteria, as well as elastic-dissipative characteristics [6, 7,10]. Figure 7 shows the dependence of the dynamic quality criterion of the frictional mechanical system on the load-speed operating conditions, tribo characteristics and analyzed parameters of the system.

The maximum permissible values of the dynamic quality criterion correspond to one and are determined by the established load-speed modes – the so-called "warning" threshold 1 (Figure 7). If the values of the dynamic quality criterion are less than the "warning" threshold, the frictional mechanical system is operated in stationary-stable modes.
A dynamic quality criterion from 1 to 1.15 informs about the advent of critical operating modes, and the criterion above 1.15 signals about the danger of the formation of abnormal friction modes which can lead to a failure of the frictional mechanical system.

**Figure 6.** Comparative analysis of the friction coefficient $f$ in statics and dimensionless damping coefficient $\xi$ in octave frequency ranges

**Figure 7.** Trends of the dynamic criterion of system quality
A dynamic quality criterion from 1 to 1.15 informs about the advent of critical operating modes, and
the criterion above 1.15 signals about the danger of the formation of abnormal friction modes which can
lead to a failure of the frictional mechanical system.

In the interval from 0 s till 320 s, the nominal load-velocity friction mode is reached, which is
characterized by low values of the dynamic quality criterion, and this determines the normal operating
mode of the technical system. According to observations, transient friction regimes from 360 s till 600
s and from 1015 s till 1180 s are characterized by a significant value of the dynamic quality criterion,
which exceeds the established threshold of "danger" by 1.15 times. Consequently, the investigated
frictional mechanical system is subject to significant inertial and elastic-dissipative influences that
exceed the permissible ones which may be due not only to the operation of the friction unit itself but
also to additional dynamic influences from the drive.

It should be underlined that the stationary friction mode of the frictional mechanical system is
characterized by the coincidence of the gradients of the friction coefficient and the dynamic quality
criterion.

4. Conclusion

Thus, the use of the methods of tribospectral identification of friction processes and dynamic monitoring
of changes during observation time of the parameters of the frictional mechanical system makes it
possible to implement the problems of diagnostics of friction units and to reveal the influence of
changing load-speed operating modes, wear, tribological characteristics of lubricants on the
tribodynamic characteristics of the friction unit.

It should be noted that the introduction of aluminum powder into the HG oil lubricant has a stabilizing
effect on the dynamic characteristics of the friction unit, reduces the speed of the system, shows more
stationary values of the damping coefficient, reduces the bulk temperature of the lubricant in the friction
unit by 20-30 ° C and, therefore, significantly improves tribothermodynamics of the friction unit.

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