Improving the efficiency of the path - rolling stock system based on the implementation of anisotropic frictional bonds

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Abstract. Unlike the materials widely represented in domestic and foreign scientific, technical and reference literature, the article theoretically substantiates the need to take into account in laboratory and bench tests of mechanical systems with friction units the presence and interconnection of dynamic processes occurring in a frictional contact and in a mechanical quasilinear subsystem; ensure identical: parameters of macro- and micro-roughness of contacting surfaces, frequencies and forms of natural vibrations, physical and mechanical properties of tribocontacts, as well as dynamic (integral) characteristics of a natural object and its physical model. The article presents the methods of physical and mathematical modeling and tribospectral identification of friction processes. The features of these methods and an example of dynamic monitoring of a mobile mechanical system using the example of a friction system: “way - rolling stock” are considered

1. Introduction

Issues of increasing the volume and stability of locomotive traction force play a key role in improving the efficiency of the railway industry, which is also reflected in the above concept. In order to increase traction, a train with two locomotives located in front and behind the train is used; quartz sand is used as a friction activator. Possible misalignment of traction forces when using a locomotive-pusher leads to an increase in resistance to movement of the train, creates additional forces in the “wheel flange-rail” contact, increases the intensity of the transverse creep forces and the wear intensity of the wheels and rails [1-3]. The use of sand, which is an active abrasive, leads to a sharp increase in the wear rate of wheels and rails, as well as to losses of up to 5% of traction power associated with the energy required to destroy sand particles falling under the wheels. On prolonged slopes, due to the active supply of sand, ballast sanding occurs, which reduces the repair time of railway maintenance. The way. A single refueling of locomotive with sand provides a turnover shoulder of not more than 1.5 thousand km, with the necessary shoulder reaching up to 8 thousand km.

To solve these problems, a technology has been developed for controlling the frictional contact of the wheels of a locomotive. An on-board automatic control and management system (ACMS) with friction surface modifier (FSM) briquette feed devices installed on a locomotive predicts the onset of the slip level with the aim of preliminarily turning on the modification system and eliminating traction implementation modes at an increased slip speed level. FSM briquette feeders with ACMS systems are installed on all wheelsets of a locomotive, which implements autonomous (axial) control of power units [1-4].
2. Organizations of dynamic monitoring of processes wheel-rail interaction

The quality criteria of the friction-mechanical system are set up on the basis of frequency characteristics. Frequency characteristics do not consider the type of transient process, are based on some frequency properties of the system and are designed to evaluate parameters intended to limit certain parameters [2-6].

The following characteristics of the friction-mechanical system can be identified by the frequency characteristics [2,3,7-10]:

1) stationary value of the transmission coefficient

\[ W_{xy}(0) = A(0) = f_{tr} = \frac{F_{tr}}{N} \approx \frac{C_x x}{C_y y} \]  

where \( W_{xy}(0) \) is the value of the complex transfer coefficient at a value of the oscillation frequency \( \omega \), equal to zero, which corresponds to the transfer coefficient in a stationary steady motion of the friction-mechanical system;

\( A(0) \) is amplitude value on the graph of the amplitude-frequency curve;

\( f_{tr} \) is the value of the friction coefficient in a stationary steady motion;

\( F_{tr} \) is the friction force, N;

\( N \) is the normal load, N;

\( C_x \) is the coefficient of elasticity of frictional bonds in the direction of relative displacement, N/m;

\( C_y \) is coefficient of elasticity of friction bonds in the direction of relative approach of the contacting friction surfaces, N/m;

\( x \) is the amplitude of tangential displacement (deformation of active volumes) of the friction surface;

\( y \) is the amplitude of approach of the contacting friction surfaces.

2) the magnitude of the stability margin in amplitude [2, 10], characterizing the possibility of limiting the inertial effects that contribute to breaking of the friction coupling in tribocontact:

\[ L = 20 \log \frac{1}{|W(i\omega_{c2})|}, \]  

where \( \omega_{c2} \) is cutoff frequency corresponding to the point of intersection of the hodograph of the amplitude phase function \( W(i\omega_{c2}) \) dynamic coefficient of friction of the real axis, at which the frequency transfer coefficient of the open system is equal to unity, and the phase shift is \(-180^\circ\);

\( W(i\omega_{c2}) \) is amplitude phase function of the dynamic coefficient of friction;

\( 1 \) is the critical value of the amplitude at which the amplitude of the tangential displacement of the friction-mechanical system is equal to the amplitude of the load;

3) the value of the phase stability margin \( \Psi \) [2, 3, 10], allowing to limit the inertial and dissipative components of the frictional interaction forces that contribute to the convergence of the contacting surfaces of friction, increase contact stresses and temperature - a value determined at a frequency at which the complex coefficient of friction is equal to unity, i.e. \(|W(i\omega)| = 1\).

4) the frequency indicator of the oscillation \( M \), characterizing the tendency of the friction-mechanical system to oscillate, and its value is limited in the range from 1.1 to 1.5

\[ M = \left[ \frac{\Phi(i\omega)_{\text{max}}}{\Phi(0)} \right] = \frac{W(i\omega)_{\text{max}}}{1 + W(i\omega)_{\text{max}}} = \frac{A(\omega)_{\text{max}}}{A(0)} = m^2 + \frac{1}{2m}, \]  

where \( \Phi(i\omega) \) is amplitude phase frequency curve of a closed system using unit feedback via tribocontact.
\[ \Phi(i\omega) = \frac{W(i\omega)}{1 + W(i\omega)} ; \]

\( W(i\omega) \) is original open-loop frequency transfer function;
\( A(\omega)_{\text{max}}, A(0) \) is accordingly, the maximum amplitude of the lowest frequency harmonic and the amplitude of the steady state;
\( m \) is root indicator of system stability margin.

5) cutoff frequency \( \omega_{c3} \) at \( A(\omega_{c3}) = A(0) \), determining the duration of the transition process. The higher the cutoff frequency, the shorter the transition process and the greater the speed

\[ \omega_{c3} = \omega \bigg|_{A(\omega) = A(0)} , \]

where \( A(\omega) \) is amplitude frequency function;
\( A(0) \) is the value of the coefficient of friction in the steady (stationary) state.

6) transmission band frequencies \( \omega_0 \) at \( A(\omega_0) = 0.707 \cdot A(0) \), which evaluates the noise immunity of the friction-mechanical system. The transmission band should not be too wide

\[ \omega_0 = \omega \bigg|_{A(\omega) = \frac{\sqrt{2}}{2} A(0)} . \]

7) the integral quality criterion \( I \) of the Heaviside function \([2, 3, 10]\), which characterizes the dynamic error that occurs when the system responds to the input master action of the type of the Heaviside function \( 1(t) \). The estimate is calculated in the frequency domain based on the Parseval theorem and the Rayleigh formula. Based on these methods, the integration of the square of the time function ranging from zero to infinity is replaced by the integration of the square of the module of the complex amplitude of the closed system, and the Fourier image of the deviation under study determines the error of the transient dynamic process

\[ I = \int_0^\infty (h_{st} - h(t))^2 dt \equiv \frac{1}{\pi} \int_0^\infty \frac{[\Phi(0) - \Phi(i\omega)]^2}{\omega^2} d\omega \]

where \( h_{st} \) is steady-state (stationary) value of the Heaviside transition function (with time \( t = \infty \)).

8) integral quality criterion \( I' \) of Dirac function \([2, 3, 10]\), which characterizes the dynamic error that occurs when the system responds to the input defining action of the type of the Dirac function \( \delta(t) \), that is, an impulse effect. The estimate is also calculated based on the Parseval theorem and Rayleigh formula:

\[ I' = \int_0^\infty |w(t)|^2 dt \equiv \frac{1}{\pi} \int_0^\infty |\Phi(i\omega)|^2 d\omega , \quad (4) \]

where \( w(t) \) is values of Dirac impulse function at time \( t \).

9) the integral value of the elastic-inertial components \( I_C \), contributing to an increase in potential energy. The lower value \( I_C \), the less inertia of the system has on the dynamic approach of the contacting surfaces of friction to each other, an increase in contact stresses and surface temperature, the development of plastic deformations, and also on the stability of the dynamics of the processes of friction \([2, 3, 10]\).

\[ I_C = \int_0^\infty P(\omega) d\omega , \text{if } P(\omega) \geq 0, \]

where \( P(\omega) \) is real frequency function.

10) the integral value of the inertial components \( I_m \)\([2, 3, 10]\), contributing to an increase in kinetic energy, loss of stability of the frictional bond, decrease in system stability. The less the value of \( I_m \), the lower the inertia of the system and the higher the stability of the friction processes.
\[ I_m = \int_{0}^{\infty} |P(\omega)| d\omega, \text{ if } P(\omega) < 0. \]

11) the integral value of resistance to vibrations \( I_{\text{con}} \) [2, 3, 10], characterizing the dissipative properties of the system. The larger the integral estimate, the higher the amplitude of the oscillations caused by the combination of inertial and dissipative components of the frictional interaction forces. The lower \( I_{\text{con}} \) value, the less kinetic energy is spent on friction processes in the frictional bonds of the friction-mechanical system, the lower the temperature fluctuations

\[ I_{\text{con}} = \int_{0}^{\infty} |Q(\omega)| d\omega, \text{ if } Q(\omega) \leq 0, \]

Where \( Q(\omega) \) is imaginary frequency function.

12) the integral magnitude of the forces caused by frictional self-oscillations [2, 3, 10], \( I_{\text{fr.k}} \), characterizes the dissipative properties of the system when the vector of the resistance to motion is co-directional with the velocity vector. Its increase contributes to a decrease in stability and stability, breaking friction bonds. A significant increase in this estimate may indicate a significant increase in the relative sliding rates of friction surfaces while overcoming local or local resistances caused by the transition from elastic to elastoplastic deformations or from elastoplastic to plastic deformation of contacting surfaces, wear, and thermal damage to the friction-mechanical system. The lower the \( I_{\text{fr.k}} \) value, the more stable the friction-mechanical system

\[ I_{\text{fr.k}} = \int_{0}^{\infty} Q(\omega) d\omega, \text{ if } Q(\omega) > 0. \]

13) Integral damping coefficient estimation \( I_\xi \), characterizing the ratio of the dissipative and conservative energy of the friction-mechanical system and approximately corresponds to its linear value \( \xi \)

\[ I_\xi \approx \frac{1}{\sqrt{1 + \left(\frac{I_{\text{con}} + I_{\text{fr.k}}}{I_c + I_m}\right)^2}}, \quad \xi = \frac{n}{\omega_0} = \frac{\beta}{\sqrt{4mC}} = \frac{\beta}{\beta_0}. \]

Where \( I_{\text{con}} \) is the integral assessment of resistance to vibrations;
\( I_{\text{fr.k}} \) is the integral assessment of forces caused by frictional self-oscillations;
\( I_c \) is the integral assessment of elastic-inertial components;
\( I_m \) is the integral assessment of inertial components;
n is the vibration damping coefficient, \( \text{s}^{-1} \);
\( \omega_0 \) is the natural frequency, \( \text{s}^{-1} \);
\( \beta \) is the equivalent vibration resistance coefficient, N\( \cdot \text{s/m} \);
m is the reduced mass of contacting surfaces involved in friction, kg;
C is the reduced coefficient of elasticity of contacting surfaces, N/m;
\( \beta_0 \) is the equivalent critical value of the coefficient of resistance to vibrations, at which the oscillatory character is replaced by a monotonically damping (aperiodic).

Having determined the frequency characteristics of the friction-mechanical system, according to the real function and integral V.V. Solodovnikov the transition function is calculated that characterizes the free component of the vibrational states of the system.
3. Conclusion
The temporal characteristics in the form of a transition function make it possible to evaluate the nature of the change in the coefficient of friction in the friction-mechanical system from the impact on the object of study (mathematical model) of the standard Heaviside function 1 (t). If the system meets the specified quality requirements, then it will satisfy these requirements under any other arbitrary impact.

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