Influence of locomotive traction drive design on main forms of self-oscillations during spinning

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Abstract. The problem of the effect of the locomotive traction drive design on the main forms of self-oscillations during spinning is considered. It is established that in most cases it is sufficient to carry out a simulation for 2-3 prevailing self-oscillation modes, which simplifies the analysis of the results and selects the best variant of the traction drive.

1. Introduction
The competition of modes (vibration modes) - the suppression of some modes by others in self-oscillating systems - is due to the fact that competing modes draw energy to cover dissipative costs from a common source. As a result, some modes create additional nonlinear damping for others. This phenomenon manifests itself in the frictional self-oscillations of the locomotive traction drive in the form when maximum amplitude of self-oscillations is reached only when one of the forms predominates, and an increase in the wheel sliding speed along the rail can lead to a change in one prevailing form of oscillation to another. The predominance of this or that form of oscillation is due to a combination of random conditions of the beginning and development of spinning, the effects of shock vibration in the drive, the presence of resonances and external perturbation.

2. Materials and methods
The effect of vibration damping of a wheel pair self-oscillations in a traction gear with a gear wheel directly located on the wheel pair axis can be estimated through a conditional parameter, representing the ratio of the energy lost per cycle to the maximum possible energy stored in the system in the absence of damping [1-4]:

\[ K_b = \frac{1 - R^2}{2} \left( \frac{V}{V_s} \right)^2 \frac{J_p}{J_k + J_p}, \]  

(1)

where \( J_k \) – wheel center moment of inertia; \( J_p \) – moment of inertia of the drive gear and the input shaft with the coupling half, brought to the axis of the wheel pair; \( V \) – relative velocity of bodies at the moment of collision; \( V_s \) – sliding speed; \( D \) - wheel center diameter; \( R \) – coefficient of recovery upon impact.

For a particular drive, when the critical sliding speed in the steady-state mode is reached, \( V/V_s = \text{const} \), one can take:
\[ K_b = C_k \frac{J_b}{J_k + J_b}, \]  

(2)

where \( C_k \) – a constant, depending on the design parameters of the drive.

In this case, the vibro-impact regime occurs under the condition

\[ V_s \geq \frac{M_D}{2\omega I_b}, \]  

(3)

where \( M \) – driving torque of the engine reduced to the wheel pair axis; \( \omega \) - circular frequency of self-oscillations. This means that in drives with axial reduction gears in which the gear wheel is removed from the oscillation unit, it is necessary to increase the moment of inertia of the parts on the high-speed side of the transmission (for example, a half-coupling flange) at a high level of tangential stresses in the axis.

The phenomenon of subharmonic resonances was discovered during testing of the rubber-metal traverse of the Research and Design Institute of Rolling Stock (VNIKTI) construction of the diesel locomotive 2TE116 [2]. If in the considered frequency range of traction drive oscillations damping in the rubber element is slightly dependent on the frequency, then energy loss per oscillation cycle will be [5-9]:

\[ T = C_r \sum_{i=1}^{n} \Delta_i, \]  

(4)

where \( C_r \) - loss of energy in the rubber damper element for a cycle of oscillations with a strain amplitude equal to unity, determined experimentally or by calculation; \( i \) – harmonic oscillations' number; \( \Delta_i \) - the rubber element amplitude of deformation for oscillations at the frequency \( f_i = f_0/i \); \( n \) - the number of considered partial vibration frequencies of the damping mass.

The presence of an external perturbation with a frequency close to the frequency of one of the forms of self-oscillations increases the amplitude of self-oscillations and raises the probability of the development of this particular self-oscillation form. Fig. 1 shows the results of the simulation (obtained by the authors) of the external perturbation effect on self-oscillations of a two-mass system— the dependence of the relative amplitude of the oscillations \( A/A \) as a function of the detuning amount \( \xi \) (\( A \) is the oscillation amplitude under a harmonic perturbation action, \( \alpha \) is the amplitude of frictional self-oscillations, \( \xi \) is the quantity determining closeness of the frequency of self-oscillations to harmonic perturbation frequency). Points D-D’ denote transitions from the synchronization area of self-excited oscillations to the biharmonic regime (beat mode).

Types of self-oscillations in traction drives can be classified according to their shape and nature (Fig. 2). The greatest number of forms of self-oscillations can be detected during the development of spinning at the moment of starting the locomotive from the place where the friction coefficient decreases most rapidly, and, accordingly, during the period of oscillations, the energy entering the oscillatory system prevails over the energy scattered in the oscillatory system.

Self-oscillations in the traction drive system and the carriage part of the locomotive can arise wherever there is a single- or multi-mass oscillatory system, in which it is possible to isolate the mass and the elastic link, the magnitude of the elastic link deformation in closely linked to, besides other factors, the instantaneous value of the wheel frictional force in contact with a rail. On the basis of this, the block of engineering analysis of self-oscillations in the ICAD (intelligent CAD) should contain a procedure for searching for possible forms of self-oscillations by identifying elastic links experiencing a significant deformation when implementing the calculated thrust force by the locomotive and determining the estimated frequencies of these oscillations, on the basis of the experience [10-15 etc.] can be assumed in the first approximation to be equal to the natural frequencies of oscillations of the considered systems under the condition of the wheel and the rail contact mobility.
Figure 1. Dependences of the relative amplitude of the oscillations $A/a$ on the detuning $\xi$; 1 – is calculated using the analytical method; 2 – is obtained by numerical integration; 3 – taking into account the ascending part of adhesion coefficient.

Figure 2. The main types of frictional self-oscillations in the traction drive.

As practice shows, for a particular design of the traction drive, most of the found forms of vibration are not realized. For the obtained forms of oscillations, a procedure for subsequent verification based on a nonparametric description of the traction drive design is necessary. It is proposed to arrange such a check on the basis of a simplified grouping of the most common constructive schemes into the groups shown in Fig. 3a-d, selecting a separate system for the supersystem, for which two forms of self-oscillations are characteristic: vertical and longitudinal oscillations of the bogie. Carriages in which there is a significant redistribution of vertical load along the axes with the application of traction force tend to develop vertical self-oscillations of the bogie which can be checked by calculations. Carriages in which traction force is transmitted to the body through horizontal or longitudinal thrusts with shock absorbers in the form of elastic washers tend to develop longitudinal self-oscillations.

All groups of drives, except for the direct drive with the rotor on the axis (Fig. 3d), are prone to the development of self-oscillations of the wheel pair (antiphase oscillations of the wheels on the compliance of the axis). Their frequency for broad gauge locomotives is usually in the range of 50-100 Hz, and the maximum amplitude depends on the moment of inertia of the wheels (i.e., the diameter) and the attenuation produced by the traction transmission.
Figure 3. Grouping of the basic designs of traction drives: a – support-axial; b – support-frame with axle reducer; c – support-frame; d – direct; e – of locomotives with hydraulic transmission; 1 – a wheel pair; 2 – a traction motor (TM); 3 – a bogie frame; 4 – a traction gear; 5 – an elastic element in the cogwheel; 6 – a suspension; 7 – an axial reducer; 8 – a drive shaft; 9 – a stator; 10 – a rotor; 11 – a hydraulic transmission.

This attenuation depends on the distance of the point of torque application to the wheel pair axis from the vibrating unit and the presence of elastic links between the gear and the wheel pair [1]. Thus, on locomotives operated on Russian Railways, with a wheel diameter of 1050 mm, a support axle drive and the presence of an USC with a progressive stiffness characteristic, the tangential stresses in the axis are several times lower than the endurance limit (40-60 MPa), and with a wheel diameter of 1250 mm and one-sided gear transmission tangential stresses in the axis can reach unacceptable values (160 MPa). With a rigid gear transmission and a support-axial drive, there is no significant level of shearing stresses in the axis during self-oscillations. In the drives with axial reduction gears and rigid gears (including those with hydraulic transmission), when the large gear wheel is located close to the wheel pair, the stresses did not exceed the permissible values of 100 ... 115 MPa. In the support-frame traction drives (Fig. 3c), high tangential stresses in the axis (up to 160 MPa) can develop at low torsional stiffness of the hollow cardan shaft couplings, which implies that the design algorithm should take into account the preferred choice of couplings with greater torsional stiffness. The so-called support-center traction drives, in which the traction motor (TM) is supported on the axis through elastic elements, can be considered as similar to a support frame with a symmetrical torque transmission to both wheels when examining self-oscillations of a wheel pair.

Direct traction drives with a rotor located on the wheel pair axis are not inclined to the development of this form of self-oscillations, due to high rigidity of the axis with the rotor. The direct traction drives with the support frame engine suspension or the support of the rotor on the axle through the elastic elements differ from the support-frame drives with traction gear, since there is no energy dissipation in the gear teeth. Support frames, including group traction drives (Figure 3 b, c, d), for which torsional stiffness of the traction drive shafts is close to linear, tend to develop traction drive self-oscillations, in which the wheels of the wheel pair oscillate in phase. The level of dynamic torque in the shafts reaches a two-fold magnitude of the moment before breaking into a clutch. In the multimass systems of group drives (Fig. 2e), the forms of self-oscillations predominantly develop, for which the hydraulic transmission (or the reducer connecting with the TM) is located at the node of
oscillations. The development of this form of oscillation is possible in direct traction drives with TM support or a frame suspension or an elastic support of the TM on the wheel pair axis. In support-axial drives (Fig. 3a) with a spring traverse, self-oscillations of the TM core on the traverse can develop. Dynamic moments can be 2.5 times higher than the design moment for the conditions of adhesion. Almost the same dynamic moments can arise in the support-axial drives with a rigid gear wheel with self-oscillations of the supersorbing structure.

3. Conclusions and suggestions
The development of this or that form of self-oscillations in case of the traction drive spinning is affected by a combination of random conditions of the beginning and development of spinning, the effects of shock vibration in the drive, the presence of subharmonic resonances and external disturbances at frequencies close to the frequencies of self-oscillations.

In connection with the fact that self-oscillations of a wheel pair are the most typical form of self-oscillations for known drive structures, to reduce self-oscillations to a safe level, it is proposed to use energy dissipation at impacts in geared transmission.

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