Friction self-oscillation decrease in nonlinear system of locomotive traction drive

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Abstract. The problems of the friction self-oscillation decrease in a nonlinear system of a locomotive traction drive are considered. It is determined that the self-oscillation amplitude decrease in a locomotive wheel pair during boxing in traction drives with an elastic linkage between an armature of a traction electric motor and gearing can be achieved due to drive damping capacity during impact vibro-damping in an axle reduction gear with a hard driven gear. The self-oscillation amplitude reduction in a wheel pair in the designs of locomotive traction drives with the location of elastic elements between a wheel pair and gearing can be obtained owing to the application of drive inertial masses as an anti-vibrator. On the basis of the carried out investigations, a design variant of a self-oscillation shock absorber of a traction electric motor framework on a reduction gear suspension with an absorber located beyond a wheel-motor unit was offered.

1. Introduction
In a locomotive boxing mode, the dynamic loads in a traction drive are caused by friction self-oscillations. Friction self-oscillations arise in systems with nonlinear dry friction in the presence of a falling area in friction characteristics, to which the dependence of the cohesion between a wheel and a rail upon a slip velocity corresponds. As applied to train crews, the friction oscillations were investigated in a number of works ([1, 2] and others). As appears from these works, the self-oscillations in a locomotive traction drive differ in:
- the multi-modiness (there are known five basic forms of oscillations: oscillations of a wheel pair, oscillations of an engine of the suspension, oscillations of a traction drive, vertical and horizontal oscillations of a bogie);
- the competition of modes (during boxing, one of the oscillation forms dominates which can be changed by other forms when changing external conditions).
- the hard predictability of self-oscillation development during each separate case of boxing, in view of the dependence of a constant of friction upon many external conditions subjected to a spontaneous change (rail pollution, contact temperature of a wheel and a rail and so on).

It is emphasized in [1, 2] that self-oscillations can result in breakage of traction drive parts; at the same time, the most dangerous failure is the breakage of a wheel-set axle which can result in a train crash. The experience of the operation of the DS3 electric locomotive shows that the application of traction induction motors (TIM) does not exclude a boxing mode. Thus, the problem of data systematization of methods for the decrease of self-oscillation amplitudes and criteria for the choice of
these methods during locomotive designing arises. This paper is an attempt to solve this problem.

It is experimentally established that the process of the self-oscillation onset and development with increasing a slip velocity of wheels on a rail can be divided into the following phases: the clutch failure and slippage beginning; the self-oscillation development with the self-oscillation amplitude increase as the velocity increases; the beginning of the breakdown of self-oscillations when their amplitude ceases to grow; the breakdown of self-oscillations when the further velocity increase results in the amplitude decrease.

These phases are characteristic for any forms of friction self-oscillations during boxing which could be seen in Figures 1a and b, where the process of self-oscillation development during boxing of the 2TE121 type diesel locomotive according to two forms is shown: self-oscillations of a wheel pair as a rigid solid, in case of shafting compliance of a traction drive (Figure 1 a) and self-oscillations of a wheel pair in case of the compliance of its axle, when wheels oscillate in an anti-phase (Figure 1 b). It is possible to see in both figures that the basic influence upon the limit value of the self-oscillation amplitude is performed by a speed at which the power growth input to an oscillating system per one cycle of oscillations stops and the further increase of the wheels slip on a rail results in a quick decrease of the amplitude.

Figure 1. Examples of torque diagrams of a high-speed shaft of an axial reducing gear at self-oscillations of a traction drive (a) and tangential stresses in the axle of a wheel pair (b) in the boxing mode of diesel locomotive 2TE121: EC – variant with an elastic cog-wheel, RC – variant with a rigid cog-wheel.

A conclusion appears from the fact that for the most critical unit of a diesel locomotive traction drive, a wheel pair, tangential stresses during its self-oscillations in the mode of boxing could be decreased considerably at the expense of power diffusion in a traction drive which is also shown in work [1]. For self-oscillation absorbing, it is possible to introduce an additional oscillatory unit with an elastic-dissipative connection in a dynamic system as it is offered in [3], or to use the existing design elements of a drive. Let us consider first a possibility of self-oscillation power dispersion in a traction drive owing to the use of its design elements.

The appearance of new locomotives and road machines with an elastic link between a traction motor armature and a wheel pair (elastic cog-wheel, elastic clutch, torsion) poses a problem of the creation of a friction self-oscillation absorbing theory by elements of a traction drive and a crew part of a locomotive. Let us consider factors affecting the capacity of a drive to damp self-oscillations of a wheel pair and required dynamic properties of a traction drive ensuring a sufficient damping capacity.

2. Materials and methods
The main purpose of the work is the analysis of existing and search for new options for locomotive traction drive design in order to reduce impact of dynamic loads. In work use analytical methods to research the dynamics of locomotives, the results of which were confirmed by field experiments.
3. Impact oscillation damping in the axial reducing gear

During the testing time of a traction drive of diesel locomotive 2TE121 with a rigid cog-wheel of an axial reducing gear, during boxing, the stress in an axle of a wheel pair reached 85...90 MPa, and in case of an elastic cog-wheel – 140...160 MPa. A dynamic constituent of a torque on an input shaft of a reducing gear in both cases amounts to 6...7 kN/m. Inasmuch as in this case, the self-oscillation frequency in a wheel pair during boxing (49 Hz) is several times higher of the frequency of armature oscillations of a traction motor (TM) making 4-6 Hz, the armature oscillations in drive 2TE121 did not affect considerably dynamic processes in a drive, and the observable effect could be explained only by an impact damping of oscillations in a drive with a rigid cog-wheel. In the absence of self-oscillation damping in a wheel pair (a driven wheel is located in the oscillation unit [1], or a wheel pair is separated from a gearing by an elastic clutch [2]), stresses in an axle reach 160 MPa. Assuming 160 MPa as a level of stresses reached in the absence of considerable power scattering in elements of a drive, it is possible to draw a conclusion that the effect of power scattering in shock-absorbers of an elastic cog-wheel, elastic joints of the reducing gear suspension and friction in gearing is negligible in comparison with the effect of vibroimpact damping except when strong resonance oscillations develop.

As a rule [1], dynamic properties of a traction drive with non-linear elastic-dissipative properties were estimated on the basis of linearized models. At the same time, it was considered that ‘presence of considerable non-linearity (random kind) of drive elastic elements does not affect in any way the changes in the development and stabilization of friction self-oscillations’.

For a more specific problem of the analysis of power waste in a vibroimpact system we will consider not the process of self-oscillation itself but pulse interactions of system joints accompanying it. As it is shown in [4], in a symmetric two-mass dynamic system (Figure 2) at one concussion, the amount of released energy $T$ is equal to

$$T = \frac{(1 - R^2)}{2} \frac{Mm}{M + m} \cdot V^2,$$

where $R$ – the coefficient of restoration at a blow; $M$ – the mass of the deformable body (of a wheel center with a cog-wheel); $m$ – the mass of an absorber (gear with a flange or a rotor of a traction engine); $V$ – the relative velocity of bodies at the moment of the concussion.

![Figure 2. The scheme of the two-mass vibro-impact system of the traction drive.](image)

For the traction drive of diesel locomotive 2TE121 in the mode of boxing it was revealed that on the body of a traction gear box close to a suspension bracket, the intensive accelerations arise both with the frequency of a basic tone of wheel pair self-oscillations and with double frequency. The phenomenon is noted both for the variant with a rigid rim, and for that with an elastic one of a cog-wheel of the axle reducing gear and it indicates that during one cycle of wheel pair self-oscillations in gearing, two concussions of teeth occur. A rigid rim of a cog-wheel impacts directly the axle blows, and an elastic one weakens them and, in this way, decreases the efficiency of impact vibro-absorbing. The connection rigidity of a cog-wheel hub in a reducing gear with the closest wheel center (for a drive with spur gears) is many times higher than that with other wheel center, and a moment of inertia of a driven cog-wheel does not exceed 20...25% of the moment of inertia of the wheel center. The results of experimental researches did not reveal a considerable influence of the cog-wheel place upon the location of an oscillation unit as the moment of inertia of wheel masses of a wheel pair is several times higher than the moment of inertia of masses of a cog-wheel (for diesel locomotive 2TE121 – 5 times higher). It substantiates the reason to suppose that an oscillation system of a wheel pair represents a symmetric two-mass system with an oscillation unit in the middle of an axle. On the basis of this, in [5], the authors offered a way to estimate efficiency of vibro-absorbing through a conditional parameter being the
ratio of power used in the course of a cycle to the maximum possible power stored in the system in the absence of damping:

$$K_b = \left(1 - R^2\right) \frac{D}{2} \left(\frac{V}{V_s}\right)^2 \frac{J_b}{J_k + J_b},$$  \hspace{0.5cm} (2)

where $V_s$ – slip velocity; $D$ – the diameter of the wheel center; $J_k$ – the moment of inertia of the wheel center, $J_b$ – the moment of inertia of the driving gear and the input shaft with half coupling moved to the axle of the wheel pair.

A practical value of parameter $K_b$ consists in the definition with its aid the design measures which allow decreasing stresses in the axle during boxing in the course of developmental testing. For a specific drive, when reaching a critical slip velocity in set mode $V/V_s=\text{const}$, it can be assumed that:

$$\left(1 - R^2\right) \frac{D}{2} \left(\frac{V}{V_s}\right)^2 = C_k,$$  \hspace{0.5cm} (3)

where $C_k$ – constant depending on design parameters of the drive.

A vibro-impact mode begins to function at the fulfillment of the condition

$$V_s \geq \frac{M_t \omega}{2 \omega_0 b},$$  \hspace{0.5cm} (4)

where $M_t$ – the motor traction moment applied to the axle of a wheel pair; $\omega$ – circular frequency of self-oscillations.

Thus, the shown inertia moment of a half-coupling flange defines a critical velocity at which a vibro-impact dissipation of self-oscillation power occurs. To increase the vibro-damping efficiency and the dynamic stress decrease in the axle of a wheel pair, it is necessary to increase ratio $J_b/(J_k+J_b)$, which could be carried out at the expense of the half-coupling inertia moment increase at the input shaft of a reducing gear.

In the mentioned designs of the drive, we did not observe large-scale failures of wheel pairs in contrast to, for instance, industrial electric locomotive 14KR1 [1], in spite of the boxing mode appearance during starting off. The presence of the rigid gearing in a traction drive, as a rule, ensures a sufficiently low level of tangential stresses during torsional self-oscillations of a wheel pair without the additional increase of the inertia moment in a half-coupling of a driving shaft.

It is confirmed also by the experience of the development and operation of electric locomotives ‘RM’ and ‘RC4’ of ‘ASEA’ Co., which have a support-frame drive with an axial reducing gear and the rigid gearing [6]. It is well-known that a large-scale appearance of cracks in wheel axles is observed at the amplitude of alternating stresses equal to 160 MPa [1, 8]. In [7], it is recommended for shafts of a traction drive to admit a lower limit of a safety margin equal to 1.4…1.6, which allows assuming tangential stresses in an axle defined during tests, 100…115 MPa, as a limit. In narrow-gauge locomotives MD54 and TU6, the tangential stresses in an axle during the tests reached 115 MPa [1], and a fact of a reliable operation of axles in these locomotives confirms the correctness of safety margin chosen.

4. The use of inertia masses of the drive as a shock absorber

Elastic elements in a cog-wheel reduce sharply the efficiency of impact vibro-absorbing. The installation of an elastic self-installing cog-wheel (IESCW) in a traction drive of diesel locomotive 2TE116 with a support-axle drive did not cause large-scale failures in axles. It was defined experimentally that tangential stresses in an axle during self-oscillations do not exceed 40…60 MPa (the drive of diesel locomotive 2TE10L with a rigid cog-wheel – 30 MPa [1]).

For the IESCW of a diesel locomotive, the rigidity characteristic during the realization of traction efforts close to utmost ones on coupling is non-linear [9], which can result in the appearance of resonance phenomena at frequencies several times less than excitation frequency. During the tests of the support-axial traction drive with an elastic cog-wheel and a rubber-metal cross-arm [9, 10], self-oscillations of a wheelset during boxing with frequency $f_0=80…85$ Hz were followed by intensive subharmonic oscillations of an engine on a rubber-metal cross-arm with frequency $(19…20 \text{ Hz}) \sim f_0/4$, \hspace{0.5cm} (1)
which was not observed in the course of testing the traction drive of diesel locomotive 2TE10L and diesel locomotives with hydrotransmission.

The phenomenon revealed cannot be explained by the highest harmonics of traction engine self-oscillations around the axle of a wheelset since it is typical that for this form of self-oscillations, the armature current fluctuations with frequency 8…12 Hz were not in tests. The appearance of sub-harmonic fluctuations can be caused only by fluctuations in the system of gyrating masses. It allows using the armature of a driving motor or a flange of a half-coupling of an axial reducing gear in a traction drive with an axial reducing gear as a resonance absorber (antivibrator) since the basic frequency of self-oscillations in a wheelset in these drives is usually far from frequencies of characteristic oscillations of drive elements. Sub-harmonic oscillations can become excited in case a twisting rigidity of shaftings of a support-frame drive with an axial reducing gear has a significant non-linear characteristic, for instance, in case a driving motor is connected with an axial reducing gear through a rigid compensating coupling (gear clutch, cam clutch, hinged-actuating coupling and so on), and shafting twisting compliance is ensured by an elastic cog-wheel of an axial reducing gear with a rigidity non-linear characteristic.

If in the considered range of oscillation frequencies of a traction drive the damping in a rubber element depends slightly upon frequency, then power waste during an oscillation cycle will amount to

$$T = C_r \sum \Delta_i$$

where $C_r$ – power waste in the rubber element of the absorber in the course of a cycle of oscillations at the amplitude of deformation equal to a unit determined experimentally or in a computation way; $i$ – the number of oscillation harmonic; $\Delta_i$ – the amplitude of rubber element deformation at oscillations at frequency $f_i = f_0/\sqrt{i}$; $n$ – the number of analyzable partial frequencies of oscillations of a damping mass.

The efficiency of self-oscillation damping in a wheelset during sub-harmonic resonance depends upon the amplitude of relative deformation of elastic rubber elements, the value of which can be large enough in a resonance operating mode of an absorber. The growth of the deformation amplitude in rubber elements of a drive during boxing results in the decrease of their life. From the practical point of view, the most convenient way of damping with the aid of structure elements of a drive should be shock vibro-damping in the gearing. The use of inertia masses of a drive as a shock absorber (antivibrator) is useful only when an elastic joint in shaftings of a drive according to design ideas cannot be located just between a wheelset and a cog-wheel.

5. Search of design solutions for additional devices of self-oscillation damping

The use of special absorbers to reduce self-oscillation amplitudes in a traction drive is difficult in connection with the fact that the elements of a traction drive are located in a limited area of a bogie. In a diesel locomotive with a hydrotransmission, it is possible in principle to use silicon absorbers for the dissipation of oscillation power in the shafting, but in practice, it is impossible to carry out for diesel locomotives with power transmission and electric locomotives. Besides, during the influence of buffing loadings upon hydraulic absorbers arising in a traction drive, these buffing loadings can pass to a bogie frame.

In connection with this, the authors have offered the solution for a self-oscillation absorber of a driving motor (DM) on a traverse suspension with the absorber located beyond the dimensions of a wheel-motor unit and combined with a suspension unit (Figure 3).

With this arrangement, the unit of the driving motor suspension has suspension link 1 and some rubber or rubber-metal elements 2 arranged in alignment and preliminarily tightened. At the same time, the end of the suspension link has non-self-braking thread 3 on which there is nut 4 connected with the axle of speed regulator 5, the body of which is connected with the frame of bogie 6 through rubber elements 7.

During deformation of rubber elements 5 during driving motor oscillations on surface imperfections of a track or during self-oscillations when locomotive boxing suspension link 1 moves upright, thereupon nut 4 turns on non-self-braking thread 3 and rotates the axle of speed regulator 5. During oscillation, the amplitude of the velocity of axle rotation in speed regulator 5 increases that
results in the friction increase in speed regulator 5, in the increase of resistance to the nut 4 rotation on non-self-braking thread 3, which increases the resistance of a driving motor to oscillations, reduces the oscillation amplitude in case of the resonance or self-oscillations at boxing at the expense of the power dissipation increase in speed regulator 5, preventing in this way the appearance of large loads in parts. Rubber elements 7 protect speed regulator 5 from damages at shocks when passing track imperfections.

Figure 3. The version of friction self-oscillation absorber arrangement.

To solve effectively the problem of tangential stresses in the axle during self-oscillations is possible through the introduction of traction drive designs integrated with systems excluding a boxing mode at the expense of the control of a physical friction coefficient by means of the effect of the contact of a wheel and a rail of electric current and/or a magnetic field upon the area, and the creation of systems of boxing warning forecasting changes into a physical friction coefficient. This type of a drive allows also decreasing considerably power waste during wheel slipping on a track, which reaches 10-30% of power on traction which allows characterizing this sort of the drive as an energy-saving one [5].

6. Conclusion
1. The self-oscillation amplitude decrease in a wheelset during boxing in traction drives with an elastic link between the armature of a driving motor and gearing can be obtained at the expense of drive damping capacity during shock vibro-absorbing in an axial reducing gear with a rigid driven cog-wheel. At the same time, the efficiency of shock vibro-absorbing is small, which is critical to the change in design parameters of a wheel-motor unit and can be increased by the growth of an inertia moment of masses on a speed shaft of gearing.

   2. In case of the elastic element location between a wheelset and gearing, the decrease of the self-oscillation amplitude in the wheelset can be obtained at the expense of using drive inertia masses as an anti-vibrator, in particular, at the expense of sub-harmonic resonance in a non-linear dynamic system of a drive, the presence of which was revealed in a diesel locomotive support-axial drive with an elastic cog-wheel.

   3. Practically, on the basis of the rolling-stock operation, we can suppose that the value of tangential stresses in an axle during boxing must not exceed 100…115 MPa.

   4. This paper offers a design of a self-oscillation absorber of the frame of the driving motor on the suspension of a reducing gear with an absorber located beyond the wheel-motor unit.

   5. It is possible to solve the problem of the dynamic load decrease during self-oscillations through the realization of drive designs integrated with systems excluding boxing at the expense of control of a physical coefficient of friction with the effect upon the area of a wheel and a track contact of electric current and/or a magnetic field and to create boxing preventing systems, forecasting changes in the physical coefficient of friction.

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