Article
The Individual Drive of a Wheelset and the Problematics of Its Electromechanical Phenomena
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Abstract: This paper deals with the phenomenon of torsion oscillations in railway vehicle drive systems. The main goal is to reduce the risk of presence of torsional oscillations in wheelset drive, eventually to propose systems to effectively identify and eliminate torsional oscillations of wheelsets. Therefore, a simulation wheelset drive model including a detailed model of the asynchronous traction motor control was built. The results of computer simulations show that the torsional oscillations can be effectively eliminated by avoiding the resonance states between the excitation frequencies given by pulse width modulation of the inverter and the eigenfrequencies of the mechanical part of the drive. Furthermore, it was found that the presence of torsional oscillations of the wheels can be detected based on the traction motor current ripple. The wheelset drive model was subsequently implemented in a simulation model of a four-axle locomotive. A new algorithm of an anti-slip protection system that utilizes motor currents was implemented in the model. Simulations show that such an anti-slip protection system can prevent the occurrence of undesired large amplitude of wheelset torsional oscillations. The models and simulation results are presented in detail in the paper.

Keywords: wheelset; inverter; torsion oscillation; pulse-width modulation; electromagnetic torque; railway vehicles; dynamics; MATLAB Simulink; SIMPACK

1. Introduction

In terms of development of traction drives for modern rail vehicles, efforts have generally been aimed at more efficient and powerful systems. Increasing the power up to theoretical physical limits (meaning especially adhesion limits) also reveals new negative physical phenomena. Torsional oscillations of traction drive components, which are discussed in this paper, play an important role among these undesirable effects.

Recently, problems with Deutsche Bahn locomotives class 145 (Bombardier TRAXX family) have surfaced. The relative rotation of the wheel disc with respect to the wheelset axle has been observed within locomotive maintenance (see Figure 1) [1]. Such slip indicates failure of the press-fitted joint that holds the wheel and the axle together. This means the friction between two parts is reduced enough to allow the wheel to freely rotate or shift along the axis for a short time.

This condition can be dangerous, especially in situations when the vehicle passes through a curve or railroad switch. At that moment, significant transversal forces which act in contact with a wheel flange and rail head [2] may cause a shift in the wheel along the axis, thus pushing the wheels closer to each other. This leads to a decrease in the lateral distance between wheels, e.g., back-to-back dimension, as schematically presented in Figure 2. If the distance between wheels decreases under limit values, the vehicle can derail.

As the most probable reason for such shifts, the phenomenon of torsion oscillations was assumed. An effort to investigate the occurrence of torsional oscillations motivated the research in the area of traction drives of modern locomotives and other high-power rail vehicles. Mathematical simulation models are usually used as the principal research...
tool. Complex simulation models of a whole railway vehicle were presented by Markovic et al. [3] and Trimpe et al. [4]. The research in the field of torsion oscillations in traction drives of railway vehicles focuses on the development of control systems, which are used for an elimination of undesired behaviour of traction drives during the vehicle operation, for example, anti-slip control systems [5]. However, many other control systems can be utilized in modern traction drive systems. Efforts in this research area include controllers and control systems for solving dynamic issues with an adhesion loss and re-gain [6,7], control systems for an active reduction of vibrations and noise [8–10], predictive slip control systems [11] or systems applying advantages of fuzzy logic for anti-slip control against classic Proportional-Integral-Derivative (PID) controllers [12]. Another area of research of torsion oscillations of traction drives focuses on the investigation of dynamic loading of mechanical parts of drive trains of railway vehicles.

Figure 1. Relative motion of the wheel and the axle. See the mismatch in yellow mark [1].

Figure 2. Schematic representation how wheel distance can decrease due to one-side wheel shift. In this case, the loss of connection occurs for the left wheel.

This research focuses especially on dynamic loading of wheelsets and its relationship to electrical parameters of the drive train. In [13], self-excited torsion oscillations of the wheelset were analysed. The same problematics from the perspective of an adhesion loss were analysed by Liu et al. [14], utilizing a simulation model of a wheelset drive with two degrees of freedom. Qi and Dai in [15] focused on the electro-mechanical coupling between harmonic components of the traction motor torque and the wheel wear, whereas in [16], the influence of traction dynamics on the wheel polygonization was analysed. The negative impacts of electro-mechanical coupling in the asynchronous motor drive systems were investigated by Winterling [17], who examined complex dynamic behaviour of drive systems of locomotives. Szołc et al. [18] focused on dynamic electro-mechanical coupling...
effects in machine systems driven by asynchronous motors. The system analysis presented by Schneider [19,20] describes the effects of a wheelset overloading caused by the combined load from torsion oscillations and acting lateral forces. The importance of the research of the harmonic components was also pointed out by Takahashi et al. [21], who focused specifically on the efficiency of traction motors for railway vehicles, or in the work of Lu et al. [22], which proves, that pulsating motor torque has significant influence on the wheel-rail contact forces and longitudinal and vertical vibrations within the vehicle bogie.

The work presented in this paper utilizes a complex model of the drive system of a locomotive. Unlike the works published so far, special emphasis is placed on two innovative ideas: elimination of torsion system resonances via torsional stiffness optimization and improving the sensitivity of the traction motor control system to the detection of presence of torsional oscillations using observation of traction motor phase currents.

2. Object of Interest

The parameters of the mathematical models in this study are based on the design of modern four-axle locomotives. These locomotives are equipped with a pair of traction bogies, both with two individually powered wheelsets. The asynchronous traction drives provide overall power output between 6 and 7 MW. Typical representatives of this category of locomotives are Siemens ES64U4, Siemens Vectron, or Skoda 109E (Figure 3).

Mathematical modelling was realized in two steps. In the first step, simulations were performed with a wheelset drive model containing a torsional-flexible system of a single wheelset and its electrical drive. In the second step, a model of a whole vehicle was utilized. The vehicle model uses the wheelset drive models and moreover includes a detailed model of the mechanical system of the whole vehicle with implemented real external conditions such as adhesion, drive resistances, and track irregularities. These two mathematical models, namely the wheelset drive model and vehicle model, respectively, are described in the sections below.

2.1. Simulation Model of the Wheelset Drive

A complete drive of any railway vehicle is an extensive system. The model of the wheelset traction drive is split into four main groups which define four main sub-models—a mechanical part, an electrical-control part, a wheel-rail contact and a train drive dynamics model.

The connection of these four sub-models creates the whole wheelset model. A scheme representing sub-models and their relations (outputs/inputs) is shown in Figure 4. This model of a single drive was created within Matlab Simulink R2017a software.
2.1.1. Mechanical Part

A fully suspended hollow shaft drive (see Figure 5) is addressed in this study. Such a design allows decreasing unsprung masses and forces resulting from track irregularities, which is important for higher velocities of railway vehicles within tracks of common quality. The mechanical part of the wheelset drive consists of these main parts:

- rotor of the traction motor,
- gearbox with a pinion and a gearwheel,
- hollow shaft with elastic articulations/clutch/coupling,
- wheelset consisting of an axle and wheels.

The fully suspended wheelset drive is schematically presented in Figure 6. For the purpose of mathematical description, the mechanical part of wheelset drive was divided into seven rotational masses mutually connected by torsion springs designated as $k_l$. Thus, it forms a torsion system with seven degrees of freedom.
2.1.2. Electrical Part—Generation and Control of Torque

The electrical part is focused on the traction motor and its control. An asynchronous motor is considered. The mathematical description corresponds to voltage-flux Equations (1)–(8):

\[
\begin{align*}
    u_{1a}(t) &= R_1 i_{1a}(t) + \frac{d\Psi_{1a}(t)}{dt} \\
    u_{1\beta}(t) &= R_1 i_{1\beta}(t) + \frac{d\Psi_{1\beta}(t)}{dt} \\
    u_{2a}(t) &= R_2 i_{2a}(t) + \frac{d\Psi_{2a}(t)}{dt} + p_p \omega_m(t) \Psi_{2\beta}(t) = 0 \\
    u_{2\beta}(t) &= R_2 i_{2\beta}(t) + \frac{d\Psi_{2\beta}(t)}{dt} - p_p \omega_m(t) \Psi_{2a}(t) = 0
\end{align*}
\]

where \( u_1 \) is stator voltage, \( i_1 \) is stator current, \( \Psi_1 \) is stator flux, \( R_1 \) is stator ohmic resistance, \( u_2 \) is rotor voltage, \( i_2 \) is rotor current, \( \Psi_2 \) is rotor flux, \( R_2 \) is rotor ohmic resistance, \( p_p \) is number of pole pairs, and \( \omega_m \) is rotor mechanical angular speed. Voltage equations, particularly (3) and (4), contain angular speed of the rotor, \( \omega_m \), which means there is a relation between electrical and mechanical values in the motor.

Stator and rotor flux are given by currents and inductances, as defined in Equations (5)–(8):

\[
\begin{align*}
    \Psi_{1a}(t) &= L_1 i_{1a}(t) + L_h i_{2a}(t) \\
    \Psi_{1\beta}(t) &= L_1 i_{1\beta}(t) + L_h i_{2\beta}(t) \\
    \Psi_{2a}(t) &= L_2 i_{2a}(t) + L_h i_{1a}(t) \\
    \Psi_{2\beta}(t) &= L_2 i_{2\beta}(t) + L_h i_{1\beta}(t)
\end{align*}
\]

where \( L_1 \) is stator inductance, \( L_2 \) is rotor inductance, and \( L_h \) is coupled inductance.

Resulting torque Equation (9) is derived from the previous Equations (1)–(8):

\[
M(t) = \frac{3}{2} p_p (i_{1\beta}(t) \Psi_{1a}(t) - i_{1a}(t) \Psi_{1\beta}(t))
\]

where \( M \) is the motor torque.

The input for the motor control is the required torque of the motor. The output from the motor control is the mechanical torque of the motor. The control is based on the vector control scheme which is shown in Figure 7. Such an approach allows us to control alternate current machine with direct current values (currents) similar to direct current motors, where torque is proportional to the direct current within a certain range of rotor speed.
2.1.3. Wheel-Rail Contact—Adhesion Model

The required value of torque is determined from the actual requirement for velocity of the vehicle, which is also controlled via an anti-slip protection block. This block permanently checks angular velocities and angular accelerations of all wheelsets (in fact, rotors) and via mutual comparison determines the occurrence of excessive slippage or loss of adhesion. This control is included in the controller’s block of the electrical model (see brown rectangle in Figure 7). The other parts of the electrical model in Figure 7 are: decoupling block (black rectangle), coordinate systems transformation block (orange rectangle), pulse-width modulation (PWM) block (green rectangle), supply voltage block (yellow rectangle), inverter block (blue rectangle), asynchronous motor (ASM) block (red rectangle) and ASM model block (violet rectangle).

The required value of torque is determined from the actual requirement for velocity of the vehicle at which the traction motors are able to develop maximum torque. At higher running speeds, the torque of the traction motors is limited by the maximum power output.

The maximum tangential force transmissible in a wheel-rail contact \( T \) is given by Equation (10):

\[
T = \mu \cdot Q
\]  

where \( \mu \) is the adhesion coefficient and \( Q \) is the vertical force, i.e., the wheel force.

The calculation of the adhesion coefficient \( \mu \) is based on the adhesion theory of Polach [27], considering minor updates allowing wider variability for the setting of friction parameters. The resulting adhesion coefficient \( \mu \) applied within the wheelset drive model in dependence on the relative slip and vehicle velocity is presented in Figure 8.

The longitudinal slip \( s_x \) is defined as a division of slip speed and vehicle speed according to Equation (11), where \( r_k \) is the rolling radius of the wheel, \( \omega_k \) is the angular speed of the wheel, and \( v \) is the velocity of the vehicle:

\[
s_x = \frac{r_k \cdot \omega_k - v}{v}
\]

The range of velocities of 5–20 m·s\(^{-1}\) corresponds to the range of running velocities of the vehicle at which the traction motors are able to develop maximum torque. At higher running speeds, the torque of the traction motors is limited by the maximum power output.

\[\text{Figure 7. Vector control scheme [26].}\]
2.1.4. Train Drive Dynamics—Velocity Calculation

This submodel calculates analytical formulas from the traction mechanics to provide the velocity of a simulated vehicle run. The calculations are based on the balance between accelerating forces and resistance forces. The difference between these forces determines the resulting longitudinal force acting on the vehicle, which in the relation to the mass of the vehicle results in acceleration, \( a \). The final product of these calculations is the speed of the modelled train, \( v \) (Equation (12)), which is the result of the time integration of the initial speed \( v_0 \) and acceleration \( a \).

\[
v = v_0 + \int_0^t a \, dt
\]  

(12)

2.2. Simulation Model of the Vehicle

Using the wheelset drive model, the vehicle model was built (Figure 9). The vehicle model contains all important components reflecting the drive scheme in Figure 5 and other vehicle features such as suspension, bogie frame, carbody etc. The mechanical part of the vehicle model was built within SIMPACK 2019 software. This software also considered an adhesion model and other outer conditions (drive resists, track irregularities, track curvatures etc.). Electrical and control parts of the model were taken from the wheelset model.

Figure 9. Models of the bogie (a) and the whole vehicle (b) as shown in MBS software SIMPACK.
3. Simulation Results

3.1. Electromagnetic Excitation and Torsion Vibrations

For purposes of the research explained in this section, the wheelset model of the single drive was utilized. The fundamental behaviour of the three-phase asynchronous traction motor supplied from the inverter could be explained on its one winding phase. The phase current is rippled. The essence of the traction motor current ripple $\Delta i_d$ (see Figure 10 and Equations (13)–(17)), is described in the literature, e.g., [28].

![Figure 10. Pulse inverter scheme and the course of the output voltage and the current.](image)

When the switch is ON, the supply voltage $U_1$ is put on the L-R-Ui load, and in the circuit there is an immediate direct voltage $u_d$ and direct current $i_d$ increase according to Formulas (13) and (14):

$$R \cdot i_d + L \cdot \frac{di_d}{dt} + U_i = U_1$$

$$i_d = I_{01} e^{-\frac{t}{\tau_d}} + \frac{U_1 - U_i}{R} \left(1 - e^{-\frac{t}{\tau_d}}\right)$$

where $R$ is ohmic resistance, $L$ is inductance, $t$ is time, $\frac{di_d}{dt}$ is a time derivative, $I_{01}$ is the initial value of direct current, $I_{02}$ is the final value of direct current, and $\tau_d$ is the machine time constant. The induced voltage $U_i$ is induced in the load as one of its characteristics.

When the switch is OFF there is no supply voltage $U_1$ on the load. In the circuit, there is no immediate direct voltage $u_d$, and the direct current $i_d$ decreases according to Equations (15)–(17). Only the induced voltage $U_i$ in the load is present:

$$R \cdot i_d + L \cdot \frac{di_d}{dt} + U_i = 0$$

$$i_d = I_{02} e^{-\frac{t}{\tau_d}} - \frac{U_i}{R} \left(1 - e^{-\frac{t}{\tau_d}}\right)$$

$$\Delta i_d = I_{02} - I_{01}$$

A real asynchronous motor supplied from an inverter has a rippled current by many harmonics, including the switching frequency and its multiples, the first harmonic of the supplying voltage and its multiples, frequency sidebands etc. Harmonic components are the source of the excitation of torsion vibrations in the mechanical part of the traction drive. More specifically, the current ripple causes the ripple of the electromagnetic torque. The excitation frequencies of the inverter were determined in accordance with [29] by Equations (18) and (19):

- For odd multiples of the switching frequency $f_{PWM}$:

$$f_i = k \cdot f_{PWM} \pm 2 \cdot l \cdot f_1$$

(18)
where: $f_1$ is the first harmonic of the supply voltage, ($i = 1, 2, 3, \ldots$); ($k = 1, 3, 5, \ldots$); ($l = 1, 2, 3, 4, \ldots$);

- For even multiples of the switching frequency $f_{PWM}$:

$$f_i = m \cdot f_{PWM} \pm (2n + 1) \cdot f_1$$

where: ($i = 1, 2, 3, \ldots$); ($m = 2, 4, 6, \ldots$); ($n = 0, 1, 2, 3, \ldots$)

The basic simulation and modal analysis of the torsion system provide essential information about eigenfrequencies, content of harmonics (excitation frequencies involved in the electromagnetic torque of the traction motor) and their resulting potential resonance states. This complex knowledge is presented via Campbell’s diagram (Figure 11), which shows six eigenfrequencies of the torsion system (from the second the seventh, as the first eigenvalue corresponds to the natural rotation of the torsion system), ten excitation frequencies (from the third multiple of the first harmonics $f_1$ through multiples of switching frequencies and their side bands) and five potential resonances (from I to V). Further simulations were focused on the pinion and its respective resonance states:

- III—resonance of the fifth eigenfrequency with the frequency sideband of the switching frequency $f_{PWM}$.
- V—resonance of the seventh eigenfrequency with the frequency sideband of the third multiple of the switching frequency $f_{PWM}$.

![Campbell diagram](image)

**Figure 11.** Campbell diagram.

To study these resonance states, simulations of the acceleration of the vehicle from 0 kmh$^{-1}$ up to the speed of 86 kmh$^{-1}$ (approx. 23.8 ms$^{-1}$, $\omega_R = \omega_p \approx 165$ rads$^{-1}$) were performed. During the simulations, the following limitations of traction drive were applied: the maximal change in the traction motor drive torque over time was 1000 Nms$^{-1}$, with maximal traction motor torque of 8000 Nm in the whole speed range. Figure 12 presents the course of the pinion driving torque within this simulation with an excited resonance in the time around 27 s.
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During the drive through the speed spectrum, very significant torsion vibrations were excited. The FFT analysis revealed that these enormous vibrations have a frequency of 568 Hz, which is close to the fifth eigenfrequency of the torsion system. The excitation frequency of the sideband is 570 Hz, which provides evidence of excitation of the supposed resonance state III.

To investigate the harmonics with respect to speed, simulations were carried out for vehicle speeds of 10 kmh\(^{-1}\), 25 kmh\(^{-1}\), 52 kmh\(^{-1}\), 75 kmh\(^{-1}\) and 86 kmh\(^{-1}\) (Figure 14). The figure shows increasing magnitudes, which were expected. The most dominating component is the third multiple of \(f_1\).

![Simulated dependence of magnitudes of harmonic components of electromagnetic torque](image)

**Figure 14.** Harmonic components magnitude function [30].

In the context of Figure 14, the first way of elimination is to optimize the fifth eigenfrequency via change in torsion stiffness \(k_{tR}\)—which represents stiffness of the shaft between the rotor and pinion. The excitation magnitudes decrease with decreasing speed, and therefore, the potential of harmonic components which could excite a resonance state decreases as well. This leads to the increase in \(k_{tR}\) and the eigenfrequency in this case. The resonance state excitation can also be eliminated by moving the specific eigenfrequency into lower values, i.e., out of the excitation frequency range.

The set of simulations for modified \(k_{tR}\) and respective fifth eigenfrequency was carried out as presented in Figure 15. The stiffness ratio is defined towards the nominal value applied in the first simulation (Figure 13).
The approach to eliminate the resonance for the fifth eigenfrequency via moving it out of the range of excitation frequencies is supported by the Campbell diagram, which indicates that the fifth eigenfrequency should be under 560 Hz. As the asynchronous motor works with a slip, the stiffness was set up for $k_{R1}$ up to 85%.

The second way of elimination of resonance states was tested via a set of simulations for the torsion stiffnesses from 105% to 205%. It was assumed to reduce the potential of excitation harmonic components by increasing the eigenfrequencies. Figure 16 shows the result of this set of simulations and the fact that the excited resonance state of the pinion loading torque was significantly reduced by that approach. Equivalent results were reached by the reduction in torsion stiffness (from 85% to 90%). The Figure 17 summarizes results of FFTs from these simulations with respect to maximum magnitudes of excited resonance states.

It is clear that with increasing of the stiffness $k_{R}$, the magnitude of the studied harmonic component decreases, as well as when the eigenfrequency is moved out of the range of excitation frequencies via decreasing of the stiffness $k_{R}$. This is the conclusion which confirms the initial assumption for carrying out the simulations.

Figure 15. Set of torsion stiffnesses.

Figure 16. Driving torque of the pinion for increased torsion stiffness $k_{R}$ up to 205%.

Figure 17. Summary of FFT results from simulations with respect to maximum magnitudes of excited resonance states.
3.2. Phenomena Occurring during Loss of Adhesion

Another situation with a high potential for torsional oscillations is running a wheelset on the limit of adhesion and sudden decrease in friction between wheel and rail. Two cases of such a situation can be distinguished:

a. the wheelset loses its adhesion instantly (prestressed components such as an axle may quickly release its torsion energy);
b. the wheelset loses its adhesion gradually (energy accumulated in the wheelset is gradually released).

The first situation usually leads to rapid acceleration of the wheelset (likewise the rotor of traction motor) and relatively quick reaction of the anti-slip protection. The second situation is considerably harder to recognize. The axle usually starts to oscillate on one end, but the other wheel still holds the adhesion, so the wheelset as a whole does not accelerate to higher speeds. Absence of rapid change then results in undesirable delays in the reaction time of the anti-slip protection system. Such a situation is shown in Figure 18. At time 15 s, the adhesion of the right wheel W1 (see Figure 7) decreases from 0.4 to 0.2, and the wheelset subsequently starts to oscillate within maximum torsion amplitudes up to 0.02 rad (corresponding to approximately 1 degree). The anti-slip protection system recognizes the non-standard behaviour of the drive around time 17 s and decreases engine torque, which leads to gradual disappearance of wheelset oscillations.
Such oscillations are observable via phase currents of the asynchronous motor. Figure 19 shows time course of phase current $i_{1q}$ both as an absolute value (solid line) and as a filtered value (dashed line). It is noticeable that the current starts to ripple significantly between the time of 16 and 16.5 s. The torsional oscillations of the wheelset increase undisturbed up to the time of 16.8 s, when anti-slip protection reacts and decreases torque of the motor, which also causes the disappearance of the oscillations.

The important fact is a correlation between frequencies of wheelset oscillations and oscillations of phase current $i_{1q}$, which is the result of the presence of angular velocity in the voltage equation of asynchronous motors (4). Frequencies are identical as can be seen in Figure 20, which means that the wheelset oscillations can be identified via observation of electrical values. This allows us to check actual state of the wheelset (whether it is oscillating or not).

This approach allows us to implement an additional block into the control scheme of the drive that monitors the phase currents of the motor in order to detect the presence of oscillations in the wheelset. This block is able to recognize the wheelset oscillations earlier than a standard anti-slip protection system.

A set of simulations was performed in order to prove the relation between mechanical and electrical oscillations in the drive. The simulations were focused on the situation of the loss of adhesion on the left or right wheel considering different velocities of the vehicle and different amounts of motor torque. The range of motor torques was between 60 and 100% of the maximum value.
oscillations in the wheelset. This block is able to recognize the wheelset oscillations earlier than the anti-slip protection does not detect them. The system is able to recognize a situation when the wheelset oscillations detected by the oscillations of the motor currents are harmful or not.

Comparison of mechanical and electrical oscillations allows us to determine the relation between them, which is shown in Figure 21. It can be seen that the relation between the amplitudes of the motor current and the wheelset oscillations is almost linear. This applies to slippage on both (left and right) wheels. Using these linear dependencies, it is possible to assess whether the wheelset oscillations detected by the oscillations of the motor currents are harmful or not.

\[ f_i = \frac{q}{l} \]

\[ y = 4091x \]

\[ y = 1480,5x \]

3.3. Simulations of Vehicle Model

The wheelset model was utilized in the vehicle model. The control structure containing standard anti-slip protection shown in Figure 4 was upgraded with a part measuring and evaluating values of motor phase currents. This was done in order to recognize the appearance or evolution of mechanical oscillations into higher values. These oscillations cause enormous load to the axle and other components of the drive, but the anti-slip protection does not detect them. The system is able to recognize a situation when the wheelset has completely lost adhesion and starts to accelerate together with the rest of the drive, but not the situation when the wheelset oscillates. This is due to fact the anti-slip protection compares angular speeds and angular accelerations of all rotors. When one of
the rotors rotates much faster than the others, the control block evaluates this as a slippage and reduces torque of the motor. A similar situation occurs when one of the rotors starts to accelerate faster than the others, or when the acceleration exceeds physically real values.

Regardless of whether the wheelset starts to accelerate or not, oscillations of one wheel always transfer through the drivetrain into the rotor and influence electrical values of the motor itself. Observing the motor current may help recognize situations when the wheelset oscillates. This may be a source of the information for a quicker reaction of the anti-slip protection and a reduction in oscillations.

A comparison between standard and updated controls is shown in Figure 22. This figure shows time courses of original and updated settings, considering the same bounding conditions of a vehicle running through the curve. The wheelset starts to oscillate between 15.5 s and 17.75 s. Additional information about the presence of oscillations allows decreasing the torque faster by approximately 0.2 s, so oscillations amplitudes cannot evolve to higher values. Therefore, it is possible to reduce significantly high amplitudes of oscillations that occur in the wheelset—approximately to 20% of the original value, similarly to time duration of the phenomenon.

![Figure 22. Comparison of different time courses—standard torque control and the improved torque control that assumes currents in the motor.](image)

The important aspect is that such an upgrade of drive control is mainly a software task without the necessity for any design changes in mechanical and power parts of the wheelset drive. Therefore, this approach may be relatively easily implemented on already running vehicles that are operated frequently and thus spare significant costs connected with the presence of torsion oscillations.

4. Conclusions

Wheelset torsional oscillations are an undesirable and potentially very dangerous phenomenon. Present anti-slip systems are not capable of effectively detecting and eliminating such torsional oscillations.

Computer simulations realized with the wheelset drive simulation model revealed that harmonic components of the electromagnetic torque of the asynchronous traction motor can excite significant torsion vibrations, which cause torsion overloading of the
mechanical components. Such problems can be eliminated by optimizing the eigenfrequencies of mechanical parts of the wheelset drive by modifying torsion stiffnesses of coupling elements. Furthermore, it was found that the presence of torsional oscillations of the wheels can be detected based on the traction motor current ripple. This finding was used in the design of a new algorithm of anti-slip protection system. This algorithm was implemented on a four-axle locomotive model.

The simulations proved that wheelset torsional oscillations can be observed in advance within currents of the asynchronous traction motor. Our results showed that the improved anti-slip protection, implementing identification of torsion vibrations via motor currents, reacts faster and enables preventing the loss of traction and development of large amplitudes of torsion oscillations jeopardizing a wheelset. This new algorithm for anti-slip protection can also be implemented in software of vehicles that are already in operation and for which any changes in their mechanical design are very difficult, if not impossible.


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