Occurrence of Torsional Oscillations in Railway Wheelsets

TOMÁŠ FRIDRICHOVSKÝ and BOHUMIL ŠULC
Faculty of Mechanical Engineering
Czech Technical University in Prague
Technická 4, 166 07 Prague 6
CZECH REPUBLIC
tomas.fridrichovsky@fs.cvut.cz http://www.fs.cvut.cz

Abstract: - The latest achievements in semiconductor electro technology allowed to operate railway locomotives with very high tractive power. Transmission of high torque values to wheels reveals failures in the press-fitted joint between the wheel and the axle that were almost unknown until these days. One of the specific phenomena participating in the creation and developing of fatigue failures are periodically repeating oscillations of running wheelsets, which may appear and compromise the safety of railway vehicles. The article is devoted to simulation of dynamical processes influencing development of such phenomenon that may occur in drives of the vehicles. Models of such processes involve both dynamics in the transmission of the torque to the wheels and the variable external influences (adhesion conditions, track irregularities, variation of wheel forces, etc.), which the running wheel is exposed to. A procedure of creating mathematical model based on the idea of simplified description is proposed and described in the article. Consequent use of such mathematical modelling is described when a Simulink model of a real railway vehicle is designed. In the simulation experiments the Simulink model serves for the parametric analysis of individual components. The model is ready to make an effective design of control for suppression unwanted impacts.

Key-Words: - railway wheelset, torsional oscillations, adhesion, creep, modelling dynamics

1 Introduction

Modern railway vehicles require high travel velocities and high tractive efforts for their operation. This brings demand on high tractive power of those vehicles which has to be produced and transmitted from the traction motor to the wheels. But a usage of traction motors with powers 1 MW and higher brings new problems that may have serious consequences.

In 2009, a slightly small relative rotation between the wheel and the axle has been discovered on one of the DB145 type locomotive of German Railways (DB) locomotives during the maintenance. This issue has been then discovered on some other locomotives of the same type. Discovered rotation was relatively small, but it meant a big security risk due to its seriousness, especially when this problem has appeared on more vehicles. The problem is following – the wheelset holds and guides the vehicle along the track so it is irreplaceable. A standard railway wheelset is composed from the axle and two wheels. Due to design and economic purposes the wheels are press-fitted on the axle

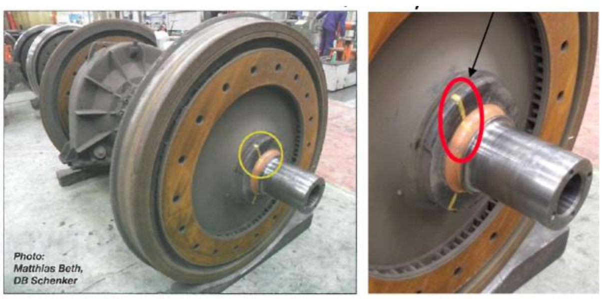

Fig. 1 locomotive wheelset with marked relative rotation (see yellow indication mark in the red ellipse on the right part of the picture) [1]

Fig. 2 DB145 locomotive TRAXX [2]

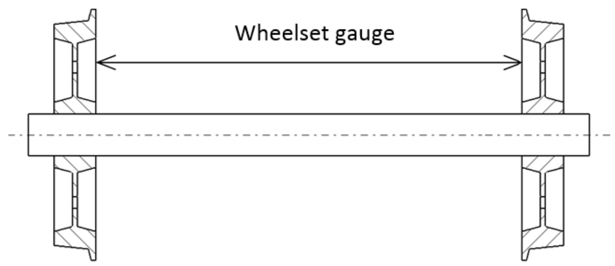

Fig. 3 a standard wheelset composed from one axle and two wheels

(Fig. 3) so the components are held together via friction between them. The main risk is a fact that the relative rotation between the wheel and the axle means losing friction (or rapid decrease) and the failure of the whole press-fitted joint.

It means that a wheel can move almost freely along the axle in transversal direction in that moment, during interaction of a guiding force. In such situation the distance between both wheels (wheelset gauge) can decrease under limits and make vehicle derailed.

Investigation has been made when the problem has been discovered, but there were not found any manufacture problems or failures [1]. Thanks to this, attention was turned to the phenomenon called “torsional oscillations of wheelsets“. Torsional oscillation is a situation when both of the wheels start to oscillate one against the other (in an opposite phase). This event leads to slightly twisting of the axle.

Torsional oscillations require an impulse to appear. This can happen when the adhesion on one of the wheels is lost or significantly decreased. In case of a locomotive this may also happen, when the vehicle runs with a high tractive effort and the adhesion force between a wheel and a rail is exceeded. Another reason is a situation when the vehicle passes through a small radius curvature. Oscillations with opposite phase may also merge during different slip velocities when the vehicle runs without tractive effort or breaking force [3].

Long lasting or periodically repeating oscillations may lead to the origin and developing of fatigue failures in the press-fitted joint (Fig. 1). Another problem is a fact that the axle is bending and twisting at the same time. This may lead to creation of fatigue cracks in the axle and decrease its durability.

2 Analysis of torsional oscillations and finding a way of their reduction

Because of the seriousness of this problem, intensive research on oscillation sources and ways to reduction has been carrying out during last two decades. The research of this issue has two possible ways of performing – via simulation and experimental measurement.

Simulation methods are the major tool of the research because of experimental measurements on a real vehicle are very expensive and require a lot of time as well. Another advantage of simulation methods is a possibility to consider the wheelset as a part of the whole drive, which can be considered as a controlled dynamic system. This allows to apply design procedures that are based on the theory of system control. This approach has been used by Böcker, Amann and Schulz [4] for presenting design of the active reduction of torsional oscillations in the drivetrain of a car. This uses an estimation regulator with help of the Kalman filter. The research was based on a simplified and idealized interpretation of flexible and inertial properties of the used components.

Situation in railway vehicles is much more difficult. Railway vehicles work with proportionally higher values of the torque, which is influenced by much higher values of inertia momentum. External influences are specific for operation of railway vehicles (effect of wind, uncompensated centrifugal forces…), especially during creation of the model of torque transmission on the tractive effort.

One of the first publications focused on torsional oscillations was the article written by authors Kaderavek and Pernicka [1]. In this article, the phenomenon was described on a general level. There was mentioned history of this phenomenon and its general specification, namely verbal description and some of the necessary conditions. More detailed description, objected on physical principle of this phenomenon, was presented by authors Benker and Weber [5] who paid attention to this phenomenon in some of their articles. These articles were mainly aimed on a wheelset that was presented as a force excited mechanical oscillator. Similar approach have chosen Yao, Zhang and Luo [6] and Müller and Kögel [7] in their articles.

The ways of regulation of the torque, where the fuzzy control was used, described Javadi and Nabizadeh [8]. Authors demonstrated the contribution of this approach that reduces occurrence of slips, whose risk described Mei, Yu and Wilson in [9]. Another usage of the Kalman filter for slip detection can be found in Mei, Yu, and Wilson [9]

Possibility of detection of the oscillations has been described by authors Markovic, Kostic and Bojovic [10]. Authors made a simplified model of the class 444 locomotive, propelled via DC motors. The model was created as a scheme of transmission of the torque, using the set of parts connected via torsional elastic elements, representing the wheelset, the gearbox and the traction motor. Results of the simulations showed that in the moment when the adhesion is decreased enough and the wheelset starts to slip, the voltage of the motor increases above its nominal value. Closer look at the voltage course shows that the voltage oscillates with a small amplitude and a specific frequency. This frequency corresponds to the natural frequency of the wheelset and appears only in the moment of losing the wheelset adhesion.

Possibility of reduction of the oscillations has been described by authors Bieker, Dede, Dörner, Klein and Pusnik [11]. Authors have dealt with an idea of using of brake discs elastically connected to the wheels that may serve as additional oscillators. These oscillators, suitably configured, should start to oscillate instead of the wheelset when the adhesion is lost. This means the axle should not twist at all. Then brake discs can serve as absorbers of the wheelset torsional oscillations. This may work only on condition when all the parameters are well set. This may be problematic because of diameter of the wheel that may differ during the vehicles lifetime, which causes changes of the wheelset natural frequency. Another problem is a fact that the natural frequency depends on the value of tangential forces between the wheel and the rail, the concept of the drive, and velocity of the vehicle as described Kolar [3], [12], [13]. Another theoretical possibility of reduction described Mei, Yu and Wilson [14] using the feed forward control. Combination of simulations and experiments on the real vehicle were performed by Allota, Conti, Meli, Pugi and Ridolfi [15]. Authors objected on a situation when the vehicle accelerates and decelerates during different adhesion conditions.

3 Approach to modelling used in simulation

The problem of modelling seems to be related only to the wheelset, but there are more factors that may have influence on it. These factors are necessary to be considered because of complexity of the model, in simplified form with certain degree of idealization or, because of difficulty of mathematical description, fully neglected. For example a vehicle body may interact with an elastically connected two-axle bogie [14], [16].

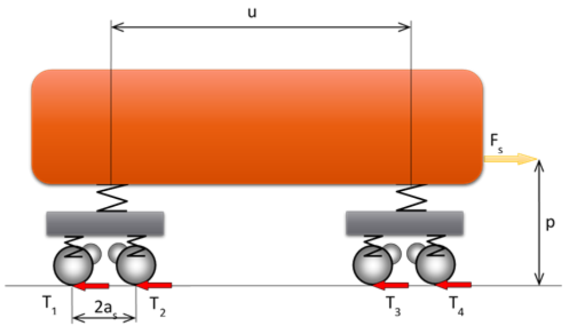

Fig. 4 schematic depicting of a four axle locomotive

According to the different positions of interacting forces F_s and T_i (see Fig. 4), the vehicle body may slightly bank and start to oscillate. Thanks to this, the wheel force, interacting between the wheel and the rail, and the pulling force may then differ during the operation.

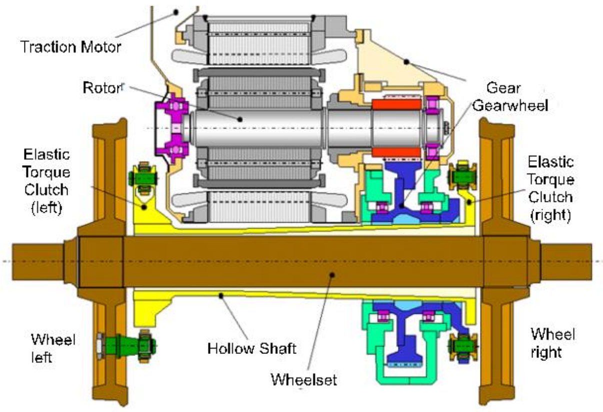

Fig. 5 construction of fully suspended drive chain

These influences are not considered in the presented model that is used for further analysis. The major attention has been aimed on the model of the whole drive chain and the adhesion between the wheel and the rail.

The design scheme of this drive chain containing cross section view is shown in Fig. 5. The locomotive drive is mainly composed from a converter fed motor (traction motor), a gearbox and clutches. The torque from the rotor passes through the gearbox (the torque increases and the speed decreases here) and through the clutches passes to the wheels. Modern locomotives have usually partially or fully suspended drive. The fully

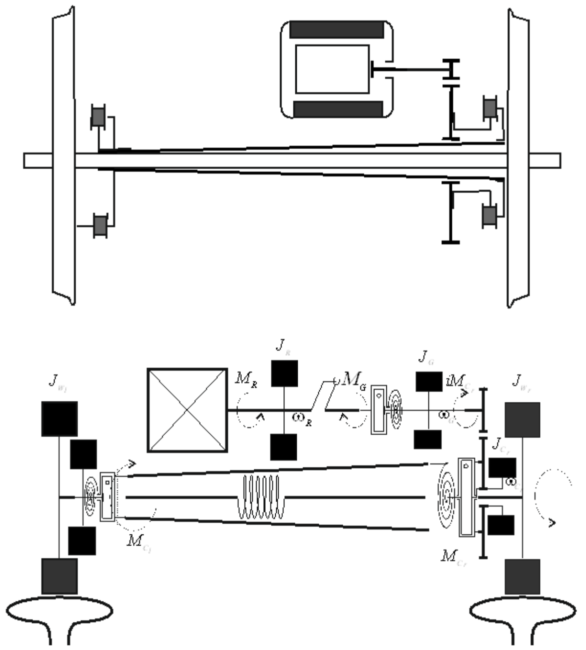

Fig. 6 two forms of the wheelset – simplified concept and representation used for mathematical description

suspended drive, which is considered for purposes of simulation, is shown in Fig. 5. This construction protects the motor from vibrations and decreases undesirable vertical forces interacting between the wheel and the rail.

3.1 Model of the drive chain

As a support of the models creation, the decomposition into the base elements has been made as it can be seen in the upper part of Fig. 6. The drive chain is here divided into fundamental parts, namely a traction motor, a gearbox, clutches, a hollow shaft, an axle and wheels.

In the lower part of Fig. 6, a schematically depicted concept of substitution via idealized elements participating on the torque transmission is shown. This is a fundamental part in creating a mathematical model of the transmission. The components, which are passing a rotary motion, are idealized and their mechanical properties are concentrated into fundamental elements. These elements represent then in an idealized way distributed dynamic properties of components such as moment of inertia, torsional stiffness and torsional damping. The two rails on the bottom of the picture represent only a requirement to include an adhesion model.

The model of drive chains dynamics is composed of three connected subsystems:

- model of transmission of the torque on the wheelset including the model of the gearbox
- model of transmission of the toque on the adhesion force between the wheel and the rail
- model of the traction motor

The creation of the model was based on the idea that the real mechanical components schematically depicted in Fig. 6 are massless and rigid and are unable to resist to the rotary motion. All these properties are concentrated in elements linked to each specific component, i.e. all these mechanical resists are born via these mutually interconnected fictive elements that than replace dynamics of these real components.

These elements are resolved with different line thickness or bordered with the dashed line in Fig. 6. Continuously distributed properties (moment of inertia, torsional stiffness, torsional damping) are replaced with three idealized elements: massless spiral spring with a constant torsional stiffness *c*, rotary hydraulic dumper with an incompressible fluid with the dumping coefficient *b*, rotating disc with the moment of inertia *J*. These three elements create a fictional component that is added to the depicted modelled component of the drive, so it is not disrupting the functionality of the transmission of the rotary motion. The respective component is then considered as ideally rigid and massless.

Numerical values of the mechanical parameters of each component used in this replacement are based on the linear theory of Hook's law. All the shafts are considered as parts made from cylindrical element. The torsional stiffness coefficient of the whole shaft k_t is given by Young's module *G*, polar section modulus J_p, length l divided into segments li and diameter *d* via following equation

$$k_{t_i} = \frac{G_i J_{p_i}}{l_i} \tag{1}$$

The torsional stiffness coefficient k_t of the whole shaft is given by the sum of element stiffness via following equation

$$\frac{1}{k_t} = \sum_{1}^{n} \frac{1}{k_{t_i}} \tag{2}$$

The polar section modulus is calculated via equation

$$J_{p_i} = \frac{\pi d_i^4}{32} \tag{3}$$

Inertia momenta are specified by means of a calculation in 3D CAD system where all the components have been designed.

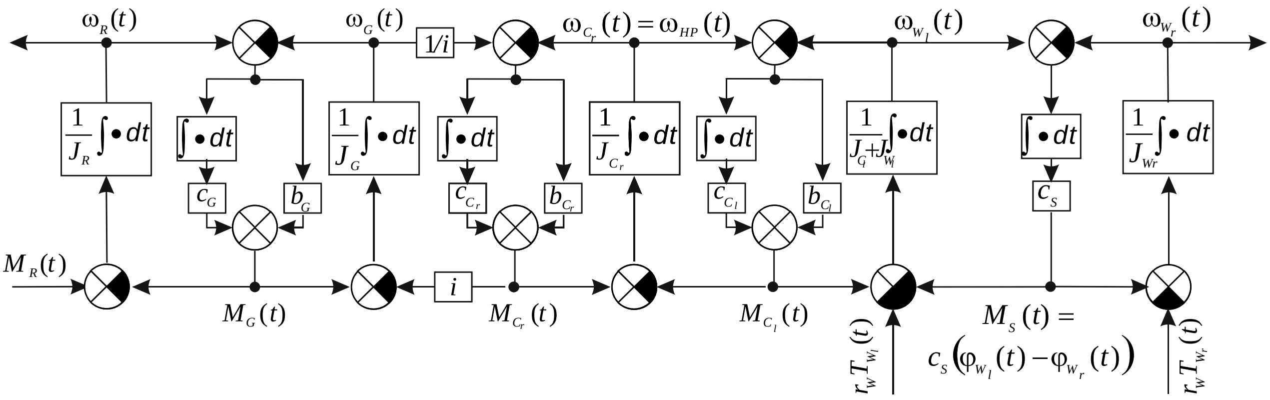

Fig. 7 Simulink block scheme of the drive chain

The equation description, which is based on the mentioned substitute concept, is a shortened description of the increment interpretation of all variables in the symbolic labelling of all the values that are characterizing the rotary motion (the angle of rotation $\varphi(t)$, the rate of its change $\omega(t) = \dot{\varphi}(t)$ (angle velocity). It means all the values are the increments – deviations from the nominal values. The usually used symbol Δ has been suppressed because of shortening of the description.

The increment model of the drive is described with following equations, where the symbols in brackets explain way of indexing from which it is clear where the value is belonging to – rotor (*R*), gearbox (*G*) right torsional clutch (C_r) hollow shaft (*HS*), left wheel (W_l), the axle (*S*) and right wheel (W_r). Every single component has its own equation of motion balancing torques on this component:

- equation of the rotor

$$J_R \dot{\omega}_R(t) = M_R(t) - M_G(t) \tag{4}$$

- equation of the gearbox (the gear ratio is given by coefficient i)

$$M_G(t) = c_G\left(\varphi_R(t) - \varphi_G(t)\right) + b_G\left(\omega_R(t) - \omega_G(t)\right) \tag{5}$$

$$J_G \dot{\omega}_G(t) = M_G(t) - iM_{C_r}(t) \tag{6}$$

- equation of the clutches

$$M_{C_r}(t) = c_{C_r}\left(\frac{1}{i}\varphi_G(t) - \varphi_{C_r}(t)\right) + b_{C_r}\left(\frac{1}{i}\omega_G(t) - \omega_{C_r}(t)\right) \tag{7}$$

$$J_G \dot{\omega}_G(t) = M_G(t) - iM_{C_r}(t) \tag{8}$$

$$\varphi_{HP_r}(t) = \varphi_{HP_l}(t) = \varphi_{HP}(t) \quad \omega_{HP_r}(t) = \omega_{HP_l}(t) = \omega_{HP}(t) \tag{9}$$

$$\varphi_{HP_r}(t) = \varphi_{HP_l}(t) = \varphi_{HP}(t) \quad \omega_{HP_r}(t) = \omega_{HP_l}(t) = \omega_{HP}(t) \tag{10}$$

$$M_{C_l}(t) = c_{C_l}\left(\varphi_{HP}(t) - \varphi_{W_l}(t)\right) + b_{C_l}\left(\omega_{HP}(t) - \omega_{W_l}(t)\right) \tag{11}$$

- equations representing the wheelset modelled (axle + wheels)

$$(J_{W_l} + J_{C_l} + \frac{1}{2}J_S)\dot{\omega}_{W_l}(t) = M_{C_l}(t) - M_{W_l}(t) \tag{12}$$

$$M_S(t) = c_S\left(\varphi_{W_l}(t) - \varphi_{W_r}(t)\right) \tag{13}$$

$$(J_{Wr} + \frac{1}{2}J_S \dot{\omega}_{W_r}(t) = M_{C_l}(t) - r_w T_{W_r}(t) - M_S(t) \tag{14}$$

Dynamic changes of the wheel angular velocities make changes of the tangential forces which depend on them. The tangential forces have to be evaluated in an adhesion model. Therefore, introduction of an adhesion model reflecting changeable weather conditions is an important next step.

3.2 Adhesion model

The model describing transmission of the tangential forces between the wheel and the rail is based on Polach theory [17]. According this theory, the tangential forces are transmitted via the friction which is bounded with a slight slip of the wheels on the rails. The amount of friction and the tangential forces as well is depending on the friction coefficient μ. The slip *s* is defined as a dimensionless ratio of the perimeter speed of the wheel and the vehicle velocity *v(t)* and is defined by the following equation

$$s(t) = \frac{r_w \omega_w(t) - v(t)}{r_w \omega_{w0}} \tag{15}$$

This means that the wheel is revolving a bit faster than the vehicle moves. This difference between the

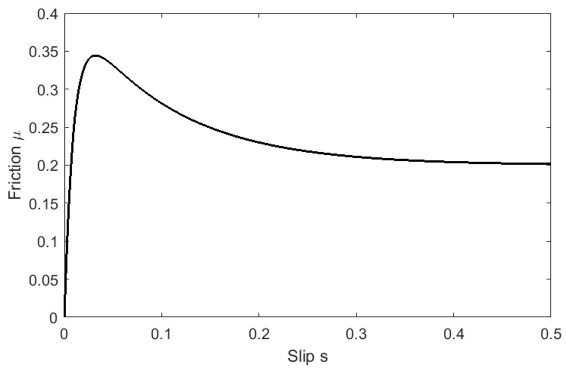

Fig. 8 slip characteristics showing relation between the slip and the friction on the rail with good adhesion conditions. The characteristics is described by (17)

velocity of the vehicle and the perimeter velocity is called "slip velocity" and it is given by the numerator of (15).

$$v(t) = r_w \omega_w(t) - v(t) \tag{16}$$

The relation between the slip and the friction coefficient is expressed by the slip characteristics. The experimentally obtained characteristics we approximated by analytically expressed formula in the form of a sum of two exponential functions. This way of replacement of experimental characteristics is quite sufficient and better for the simulation purposes.

$$\mu(s) = \mu_{MAX}\left(1 - e^{\frac{-s}{\tau_1}}\right) - \mu_{RED} h\left(s - s_{0_{MAX}}\right)\left(1 - e^{-\frac{\left(s - s_{0_{MAX}}\right)}{\tau_2}}\right) \tag{17}$$

Equation (17) uses the filter function $h(t)$. This function distinguishes the area, where the friction is decreasing according to the increasing slip.

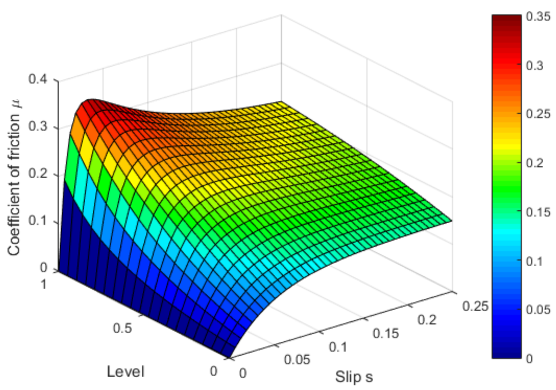

Fig. 9 slip characteristics for different adhesion conditions according to the their quality (parameter Level)

$$h\left(s - s_{0_{MAX}}\right) = \begin{cases} 0 & \forall s \le s_{0_{MAX}} \\ 1 & \forall s > s_{0_{MAX}} \end{cases} \tag{18}$$

The variability of the slip conditions for a dry or wet rail may be well expressed with appropriate chosen parameters μ_{MAX}, μ_{RED}, τ_1, τ_2 and s_{0MAX}. These parameters can be changed during the simulation via the special parameter called "Level", value of which is between 0 (wet rail) and 1 (dry rail). This parameter enables to set certain starting adhesion condition and fluently changes these conditions. Courses of the adhesion conditions depicted as changes of the friction coefficient μ are displayed in Fig. 9.

This adhesion model is based on an assumption that the vehicle is in a steady state of movement. This means that the vehicle moves with a constant velocity v_0, the pulling force of the vehicle, divided on single wheels is constant as well as the tangential forces T_{w0}, which correspond with the engine torque M_{w0}. Then the slip must have such a steady value s_0 so the friction coefficient $\mu_0 = g\text{-}1T_{w0}/m_w$ corresponds with the requested value of the tangential force T_{w0} coinciding with the vehicle mass divided onto single wheels mw.

The angular velocity of the wheel differs by an increment in the non-steady state. This velocity increment is bounded with an angular acceleration and a moment of inertia J_w during imbalance of the dynamic change of the torque on the wheel $\Delta M_w(t)$, transmitted through the drive, and change of the torque that occurs during changes of the slip $m_w g \Delta s(t)$.

3.3 Calculation of drive resistance

It is necessary to complete calculation of the slip differences with the dynamics of velocity changes that depend on the drive resists. The drive resist $O(v)$ (the roll and the aero dynamical resist) can be expressed with a parabolic relation for a single vehicle or the whole train with the equation $O(v)=m_s g(a+bv(t)+cv^2(t))$ expressing dependence on the velocity v(t). The equation takes into account the mass of the train m_s, and empirically given parameters a, b and c that correspond to type of locomotive and wagons.

3.4 Model of the traction motor

The model of the asynchronous motor used is based on a real traction motor type ML 4550 K/6 whose parameters are as follows

Power output	1 600 kW
Nominal speed	1 825 min-1

Maximal torque 10 000 Nm

In the model of the traction motor is considered a constant torque control. In its derivation the subsequent equations have been used. The supplied torque *M(t)* is calculated according the formula

$$M(t)=\frac{3}{2}p_p\left(i_{1\beta}(t)\Psi_{1\alpha}(t)-i_{1\alpha}(t)\Psi_{1\beta}(t)\right) \tag{19}$$

The motor currents i_{xy} and magnetic fluxes Ψ_{xy} are given by voltage and flux equations of asynchronous motor [18].

$$\begin{aligned}
u_{1\alpha}(t) &= R_1 i_{1\alpha}(t)+\frac{d\Psi_{1\alpha}(t)}{dt}\\
u_{1\beta}(t) &= R_1 i_{1\beta}(t)+\frac{d\Psi_{1\beta}(t)}{dt}\\
u_{2\alpha}(t) &= R_2 i_{2\alpha}(t)+\frac{d\Psi_{2\alpha}(t)}{dt}+p_p\omega_m(t)\Psi_{2\beta}(t)=0\\
u_{2\beta}(t) &= R_2 i_{2\beta}(t)+\frac{d\Psi_{2\beta}(t)}{dt}-p_p\omega_m(t)\Psi_{2\alpha}(t)=0
\end{aligned} \tag{20}$$

$$\begin{aligned}
\Psi_{1\alpha}(t) &= L_1 i_{1\alpha}(t)+L_h i_{2\alpha}(t)\\
\Psi_{1\beta}(t) &= L_1 i_{1\beta}(t)+L_h i_{2\beta}(t)\\
\Psi_{2\alpha}(t) &= L_2 i_{2\alpha}(t)+L_h i_{1\alpha}(t)\\
\Psi_{2\beta}(t) &= L_2 i_{2\beta}(t)+L_h i_{1\beta}(t)
\end{aligned} \tag{21}$$

The harmonic courses of the currents are recalculated to the quasistatic values via equations

$$\begin{aligned}
i_{1d}(t) &= i_{1\alpha}(t)\cos\gamma(t)+i_{1\beta}(t)\sin\gamma(t)\\
i_{1q}(t) &= -i_{1\alpha}(t)\sin\gamma(t)+i_{1\beta}(t)\cos\gamma(t)
\end{aligned} \tag{22}$$

This allows to control the asynchronous motor analogously to a DC motor according to the equation

$$M(t)\approx k\cdot i_{1q}(t) \tag{23}$$

The current i_{1q} is part of the total currents that is used in setting the desired torques. The desired value of a required torque is input to a PI controller.

Importance of the traction motor presence in the simulation has proven mainly in generating torque changes when the motor own dynamics does not play the most important role.

4 Results

From the experiments that can be made in simulation in a big number, we present the experiments on a situation when the vehicle accelerates with the maximal torque and then adhesion is rapidly lowered from its nominal state. This corresponds to the situation when the vehicle suddenly enters section of the rail which is polluted, e.g. with oil, grass, leaves or snow and ice. The change of adhesion conditions has been done in time t=10 seconds.

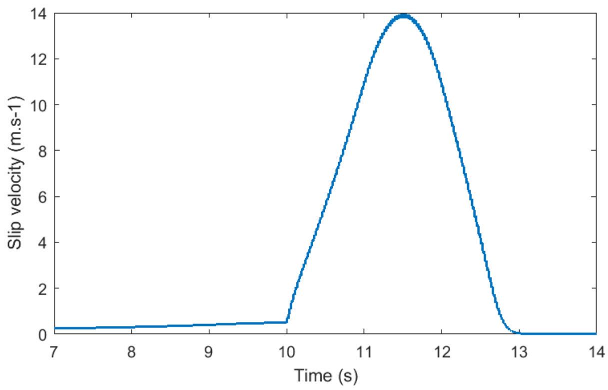

Fig. 10 Slip velocity during the loss of adhesion

Due to this, the wheelset lost its adhesion (operating point translates into the area of higher slip) so the torque is used for acceleration of the wheelset instead for acceleration of the vehicle. In this moment the slip velocity rapidly increases till the stick slip protection recognizes slip of the wheelset and makes an intervention.

During this intervention the torque of the traction motor is rapidly decreased. This allows the operating point on the slip characteristics to move back to the area of a low slip and to renew the adhesion. This is important, because high slip velocities increase the wear of the wheel and the rail. The loss of adhesion was also able to damage the traction motor in older types of locomotives.

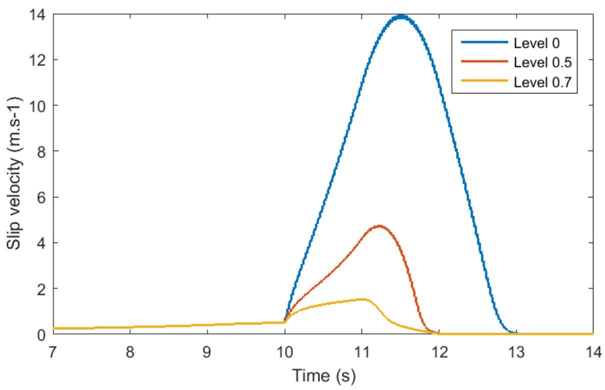

Fig. 11 slip velocities on one of the wheels according to the level of adhesion (1 – best adhesion, 0 – worst adhesion)

The amount of slip velocity depends on the torque and actual adhesion conditions. The worse the adhesion condition are, the higher the slip velocity is. Fig. 11 displays courses of the slip velocity according to the level of adhesion.

In the moment when the adhesion is lost, the wheelset starts to oscillate. This is consequence of a slight axle twist by an angle φ during the torque change. This twist accumulates energy in the axle. When the adhesion is lost, the (we imagine the axle as a torsion spring with stiffness k_t) energy is released so the wheels can start to oscillate with a specific frequency.

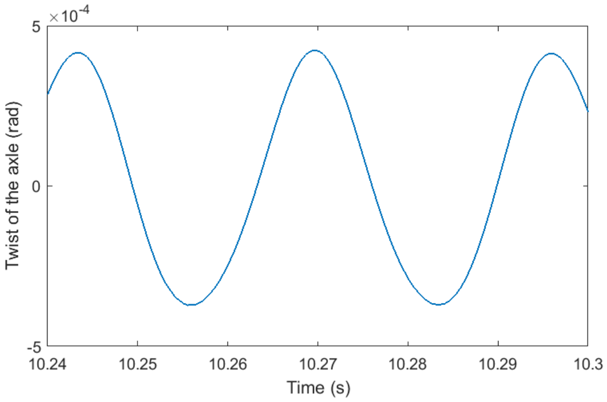

Fig. 12 detailed view at torsional oscillations of running wheelset

These oscillations depend on the actual torque and adhesion conditions and in normal conditions would be damped. Unfortunately, this damping is decreased during worse adhesion conditions. Thanks to this, the oscillations are damped very slowly or in no way. Another problem is supply of a right motor torque. Amount of the torque depends on reaction time of the stick slip protection and on the ratio of decrease. The torque decreases much slowly than the adhesion changes. During this situation the oscillations may by slightly amplified. When the torque is small enough and the slip velocity is decreased close to the zero, the oscillations are not amplified anymore, so they disappear.

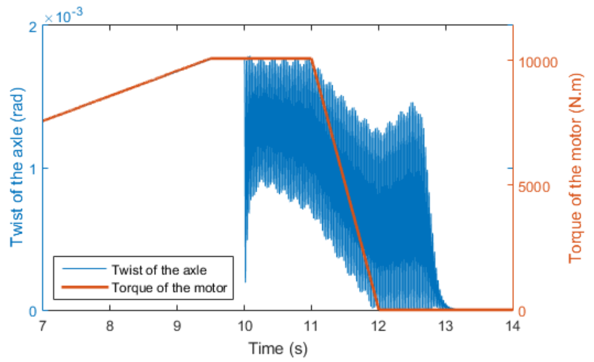

Fig. 13 time course of the torque and wheelsets oscillations

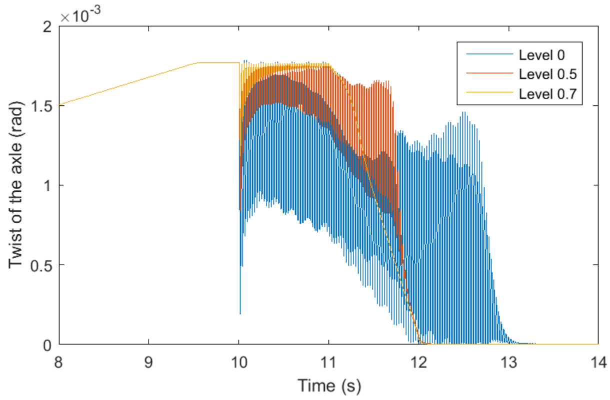

Fig. 14 torsion oscillations of the wheelset during different adhesion conditions

The time courses of the oscillations that were calculated with respect to the level of adhesion, have conditions similar to the slip velocities (Fig. 11). Higher the slip velocity is, longer the duration of oscillations is. Therefore it is important to make an intervention as fast as possible.

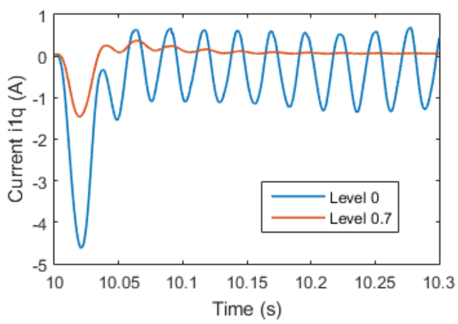

Fig. 15 oscillations of the current that are reflecting oscillations of the wheelset

The oscillations from the wheelset are transmitted through the drive chain to the traction motor so the angular velocity of the rotor is slightly influenced. This is important because the angular velocity of the motor is an input of the model of asynchronous motor (19) so the oscillations may show in the time courses of electrical values of the motor [10]. This is displayed in Fig. 15, where the current i_{1q} is oscillating according to oscillations of the wheelset. Important is that the frequency of the oscillations are in coincidence with the frequency of the wheelset. This allows to analyse the courses of the current and assume if the wheelset is in the state of slip or if the wheelset oscillates in real time.

5 Conclusion

From a series of experiments that have been carried out are presented those focused on the torsional oscillations of powered wheelset. These oscillations can be expected as a possible consequence of changes in adhesion. Generally, an online motor torque adaptation to the state of the slip is very important. The faster the slip is recognized and the torque is reduced, the smaller the oscillations and their duration are. Therefore any ways of an early slip loss detection is important, mainly those hat can be done indirectly by measuring currents in the traction motor. For such further investigations the presented model offers a good opportunity for advanced control algorithm design, or for assessment of efficiency if some special construction details should be implemented.

Assuming calculated courses of the currents, there is a possibility to recognize emerging oscillations and protect the wheelset against these influences or at least reduce negative consequences on the vehicle.

Acknowledgement

The paper was created with the financial support SGS grant No. SGS14/184/OHK2/3T/12 and of the Technology Agency of the Czech Republic, project No TE01020038 "Competence Centre of Railway Vehicles".

References:

[1] P. Kaderavek, J. Pernicka, Torsion oscillations of powered wheelsets, *Railvolution*, 34-37 (2013).

[2] DB145 In. Railcolor [online]. © 2000. Available: http://www.railcolor.net/index.php?nav=1405310#.V9Fy2TVAb9A.

[3] J. Kolář, Design of a Wheelset Drive, *Transaction on Electrical Engineering*, 4, 11-19 (2015).

[4] J. Böcker, N. Amann, B. Schulz, Active Suppression of Torsional Oscillations, *EEE/ASME Transactions on Mechatronics*, 9, 6 (2004).

[5] T. Benker, T. Weber, Torsionsschwingungen von Raddsätzen - eine Herausforderung?, *Eisenbahn-ingenieur*, 47-52 (2015).

[6] Y. Yao, H. Zhang, S, Luo, An analysis of resonance effects in locomotive drive systems experiencing wheel/rail saturation adhesion (2012).

[7] S. Müller, R. Kögel, The influence of mechanical design parameters on roll-slip oscillations in locomotive drives. *International ADAMS User Conference*: 7 (2000).

[8] S. Javadi, E, Nabizadeh, Fuzzy logic slip control design for railway vehicles. Recent research-es in system science. *Proceedings of the 15th WSEAS International Conference on Systems*: 202-206 (2000).

[9] T. X. Mei, J. H. Yu, D. A. Wilson. A mechatronic approach for anti-slip control in railway traction. *Proceedings of the 17th IFAC World Congress*: 8275-8280 (2008).

[10] P. Markovic, D. Kostic, N. Bojovic, One Method for Detection of Torsional Oscillations of Driving Axles of Electrical Locomotives, *Proceedings of the 22nd international conference on Current problems in Rail vehicles – PRORAIL 2015*, 1, 27-35 (2015)

[11] G. Bieker, J. Dede, D. Dörner, H. Klein, A. Pusnik, Bremsscheibe als Tilger für Radsatz-torsionsschwingungen, *ZEV Rail*, 381-387 (2014).

[12] J. Kolář, Problems with modelling of the influence of the vertical track irregularities on the dynamics of the vehicle drive, *Proceedings of the 22nd international conference on Current problems in Rail vehicles – PRORAIL 2015*, 1, 275-286 (2015).

[13] J. Kolář, Dynamics of Wheelset with Axle Gearbox, *Proceedings of the 22nd international conference on Current problems in Rail vehicles PRORAIL 2015*, vol. 2, 139-148 (2011).

[14] R. Allota, R. Conti, E. Meli, L. Pugi, A. Ridolfi, Development of a HIL railway roller rig model for the traction and braking testing activities under degraded adhesion conditions, *International Journal of Non-Linear Mechanics 57*: 50-64 (2013).

[15] R. Allota, R. Conti, E. Meli, L. Pugi, A. Ridolfi, Development of a HIL railway roller rig model for the traction and braking testing activities under degraded adhesion conditions, *International Journal of Non-Linear Mechanics 57*: 50-64 (2013)

[16] S. Iwnicky, *Handbook of Railway Vehicle Dynamics*, SBN 9780849333217, Taylor & Francis Group (2006)

[17] O. Polach, Creep forces in simulations of traction vehicles running on adhesion limit, *Proceedings of the 6th International Conference on Contact Mechanics and Wear of Rail/Wheel Systems (CM2003)*, 279-285 (2003)

[18] M. Kaźmierowsky, M. Piotr, H. Tunia. *Automatic control of converter-fed drives. Amsterdam: Elsevier*, (C). XIII, 559 s. Studies in electrical and electronic engineering; 46. ISBN 0-444-98660-X (1994)