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THE MODERN WHEELSET DRIVE SYSTEM AND POSSIBILITIES OF MODELLING THE TORSION DYNAMICS

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Abstract. The article deals with the view of modern wheelset drive construction design in the first part. Next part deals with dynamics of the wheelset individual drive torsion system in electric traction vehicles, explained by the drive model in several variants. The basics of time simulation of the transition dynamics effects have been explained and the frequency analysis of the system has been shown. Identification of transition dynamics effects in the wheelset drive enables to evaluate precisely the loading on individual parts during the design stage, to apply the results in anti-skid protection and probably also to disclose some effects due to wear in the wheel-rail contact.

Keywords: railway wheelset, locomotive driving system, torsion dynamics, transition dynamics phenomena, adhesion problem, simulation, calculation.

1. Introduction

This paper shows briefly the dynamics of torsion system simulation of railway driving vehicles. Most attention is given to the cooperation cases of adhesive characteristics with drive motor characteristics under downgrade adhesive conditions, their simulation and influence on all system dynamics are analyzed. Dynamic system of wheelset drive is working by rotation of masses (wheelset, gearbox, clutch, rotor of traction motor, etc.) linked by elastic and dampened bindings and interface to vehicle's other parts as well by means of bindings. A dynamic transition process originating at the moment of rise, continuation and extinction of wheelset slip is described.

2. View of modern wheelset drive construction design

Drive systems of electric driving vehicles can be distributed, for example, according to the construction design:

- group or central wheelset drive,
- individual wheelset drive,

and according to orientation of rotation axis:

- wheelset drive by traction motor with longitudinal orientation of rotation axis,
- wheelset drive by traction motor with transversal orientation of rotation axis.

Next, the paper deals with individual wheelset drive with transversal orientation of rotation axis, because this design is most often used in modern locomotives and driving vehicles of high speed trains. Individual wheelset drives (Fig. 1) are the following:

- wheelset drive by axle-mounted traction motor,
- wheelset drive by hollow shaft hugging axle
- wheelset drive by joint shaft.

The drives by joint shaft can be distributed according to construction design of joints and construction and location of joint shaft: joint shaft saved inside the rotor of traction motor, joint shaft saved outside the rotor and joint shaft hugging axle.

The axle-mounted traction motor is beared on wheelset and unsprung vertical forces react on wheelset and track, but new – asynchronous traction motors have a lower mass and this force effect is not as large (it results from cooperation measuring between Siemens locomotive Eurosprinter 127 series (joint shaft hugging axle) and cargo locomotive 152 series (axle-hung traction motor, Fig. 2). Axle hung traction motor is used in cargo locomotives for smaller speeds. The dynamic influence in the case of modern AC motor is not that big as in the older DC motor, because AC motor is much lighter and smaller than the DC motor.

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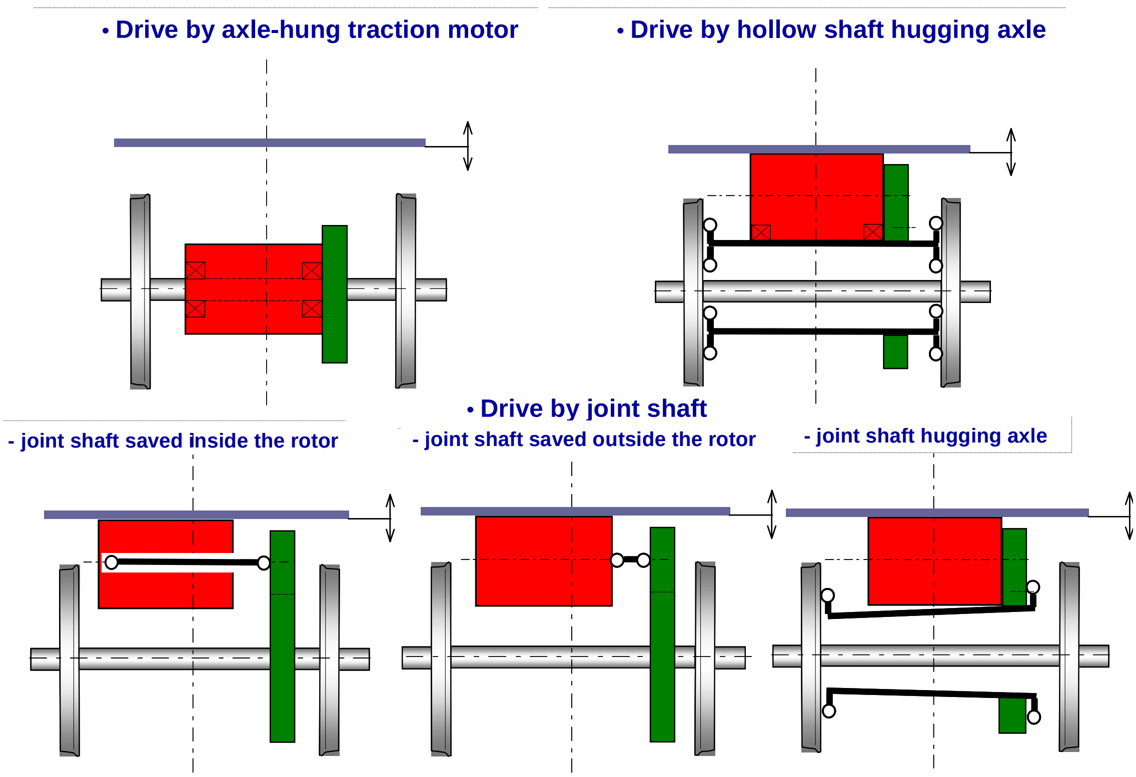


Fig. 1. The basic scheme of construction types of wheelset drive systems

The driving by hollow shaft hugging axle has a complicated mechanism between hollow shaft and wheels, that must execute the drive moment and vertical deformation of bogie-wheelset at the same time. This construction design is the historical design and is no longer used in modern railway vehicles.

Driving by joint shaft makes it possible to execute the drive moment and full joint of traction motor on bogie frame. All dynamic vertical forces of traction motor are sprung to wheelset.

Joint shaft saved inside the rotor is used at DC motors, that have large proportions and are used, for example, in locomotives SLM (Bombardier), or SKODA (Fig. 3).

At new constructions the asynchronous (AC) motor is used, and it has smaller proportions. Therefore we can locate joint shaft outside, for example in (Fig. 4), or (Fig. 5), where important design of joint shaft inside hollow pinion of traction gear is illustrated. This technical design is used for new, modern vehicles, for example system RHA (Siemens), 2016 series (Hercules) dieselelectric locomotives.

Drive by hollow joint shaft hugging axle is used at older and new vehicles too.

Driving by hollow join shaft hugging axle is an older construction design, but very good and still used in modern high speed locomotives. Original design is BBC, used at locomotion of DB 120 series, DB 101 (Bombardier), Eurosprinter series (Siemens) or at high speed trains ICE 1, ICE 2. Drive is composed from these single parts: traction motor, box of traction gear, hollow shaft hugging axle and joints - at one side between traction gear and hollow shaft, and at the other side between hollow shaft and one wheel of wheelset. This type of drive is

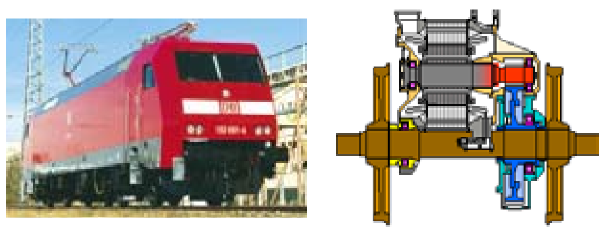


Fig. 2. A cargo locomotive Siemens 152 series and wheelset drive by AC motor and by axle-mounted traction motor

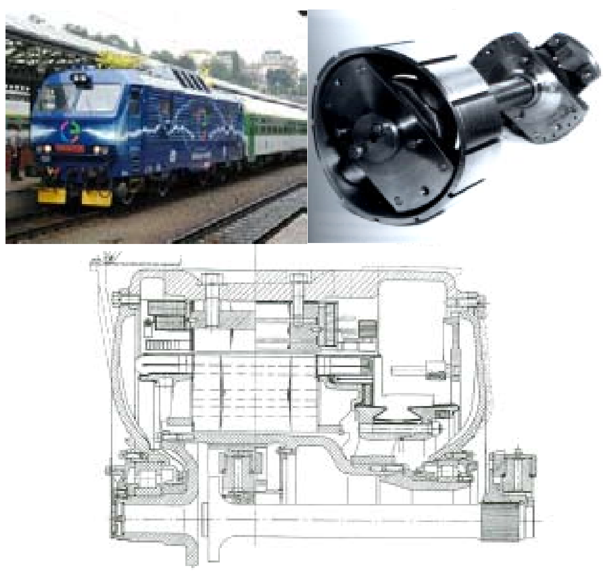


Fig. 3. Electric locomotive SKODA with jount shaft inside the hollow rotor and a detail section

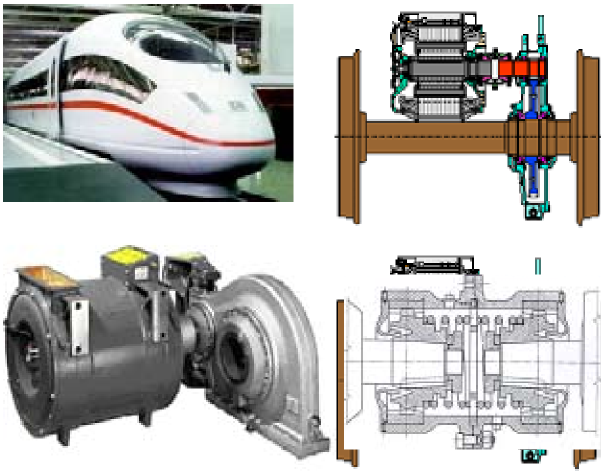


Fig. 4. High speed train ICE 3 – drive by outside location of cardan shaft and special tooth clutch

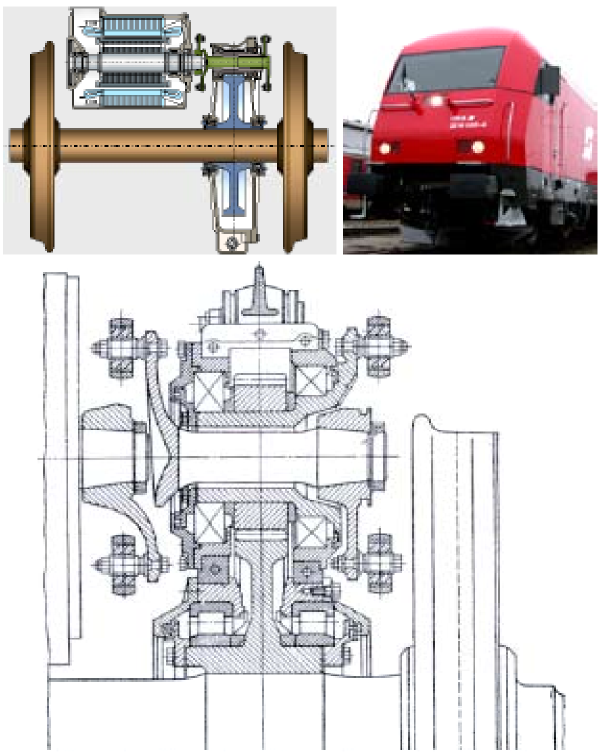


Fig. 5. System RHA – joint shaft inside hollow pinion loc. 2016 series (Hercules) and a detail section

designed mostly as integrated unit. Fig. 6 shows integrated unit IGA, used at older German locomotive DB 120.

Using brake discs directly at drive construction is necessary for high and very high speeds. In Fig. 7 we can see integrated unit with auxiliary hollow shaft with brake discs, used at ICE 1, ICE 2. Next technical design (Fig. 8) with additional brake shaft (HAB system) is used at electric locomotives Siemens 1016/1116 – Taurus or two linked shafts (Fig. 9), one of which is stationary beared on axle and the braking discs are mounted, and the second shaft functions as cardan shaft and is bent to gear.

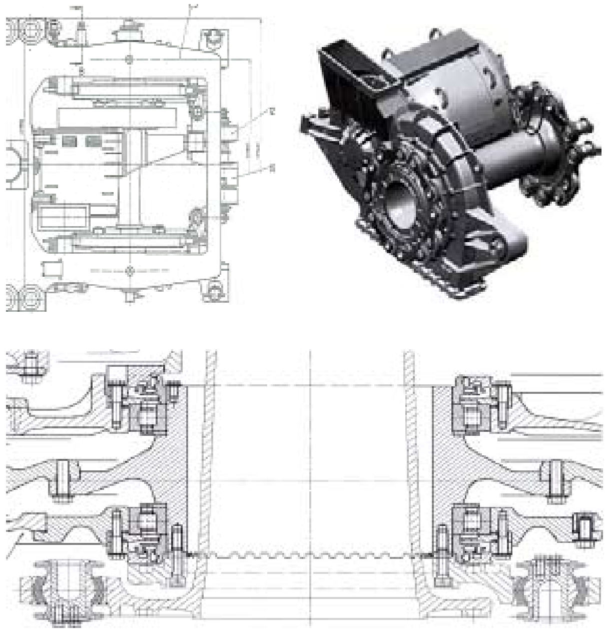


Fig. 6. Integrated unit IGA and its installation in bogie frame of locomotive DB 120; a detail section of complicated joint of hollow joint shaft with hollow toothwheel

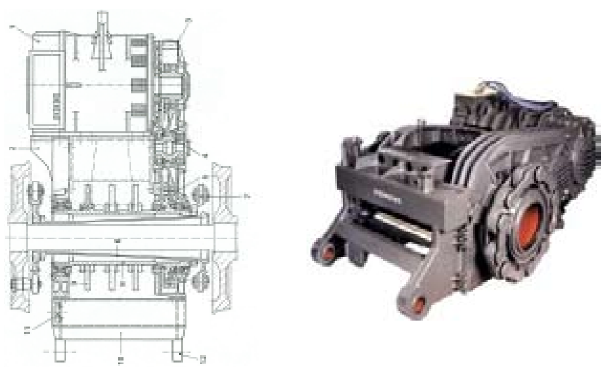


Fig. 7. Integrated unit with auxiliary hollow shaft with brake discs (ICE 1, ICE 2)

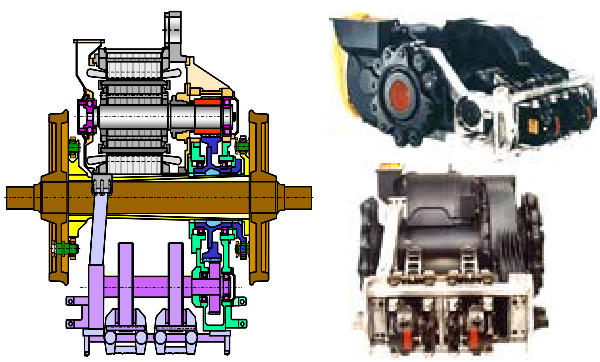


Fig. 8. Integrated unit HAB with additional brake shaft (Siemens 1016/1116 – Taurus)

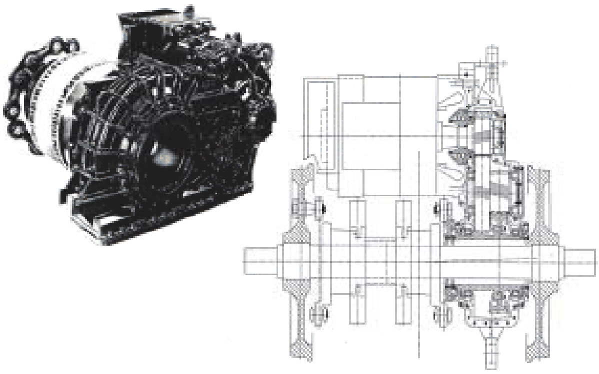


Fig. 9. Integrated drive QIH 5018 (Bombardier), used at locomotives 412 and 101 series

Fig. 10. Electric DC locomotive Skoda class 150

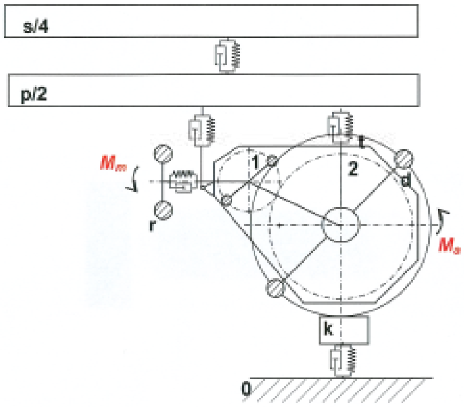


Fig. 11. The model of wheelset drive by hollow rotor

Fig. 12. Electric AC locomotive class Eurosprinter

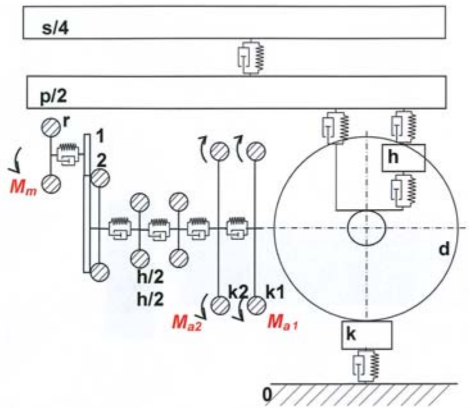


Fig. 13. The model of wheelset drive by hollow joint shaft

3. The dynamics model of wheelset drive

For the modelling two construction cases were selected: wheelset drive by hollow rotor and modern wheelset drive by hollow joint shaft.

The first example, i. e. electric locomotive Skoda class 150 (Fig. 10) is a DC locomotive for supply voltage 3000 V, with maximum power 3 MW, maximum speed 140 km/h, with DC series motor, regulated by resistances. The drive of wheelset is by joint shaft inside hollow rotor of tractive motor. Stator is stationary mounted on bogie frame. Gearbox is beared on the axle and hanged on bogie frame. The model (Fig. 11) has 6 degrees of freedom and equations of motion are the make-up of Lagrange equation 2nd type method.

For the second model a locomotive (Fig. 12), made by KM Siemens, was used, a modern electric locomotive with maximum power 6,4 MW, maximum speed 230 km/h, with asynchronous traction motor. This class is a typical representative of locomotive classes with wheelset drive by hollow joint shaft. The hollow joint shaft is a modern, quality solution and is used at construction of mechanical parts of many modern and high speed driving vehicles. The precise solution and optimization of dynamics parameters (moments of inertia, torsion stiffness, torsion damping) is very significant. For the optimization of these parameters we can use theoretical methods – simulation calculations at time area or analysis at frequency area. The dynamics model of torsional system of wheelset drive is made from rotation masses – traction motor armature, gearbox (big toothwheel), hollow joint shaft (linked by rods clutch) and mass of wheelset (Fig. 13). The precision of the model is shown in respect of torsion deformation of hollow shaft, or torsion deformation of wheelset (oscillation "wheel-to-wheel"), because only one wheel is driven.

The model has 10 degrees of freedom. In this case the equations of motion are the make-up of release methods.

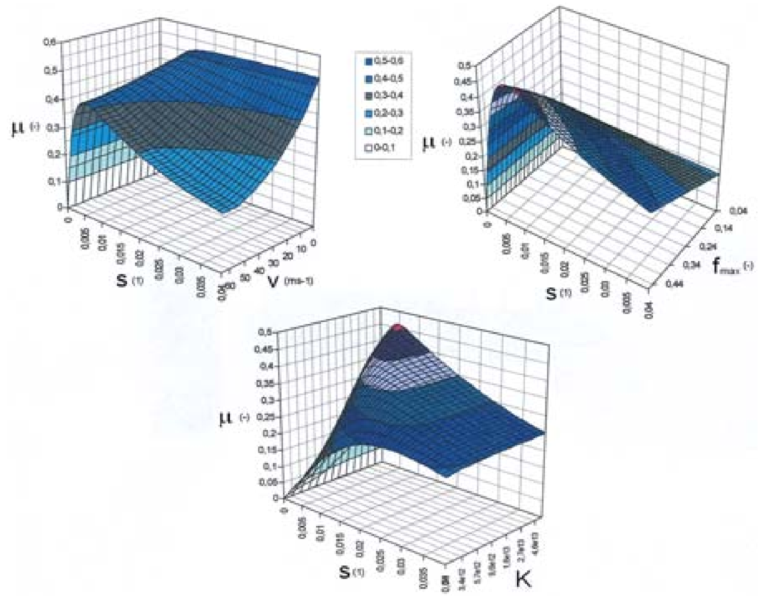


Fig. 14. The 3D-dependences of adhesive characteristics on drive speed, friction coefficient and steepness

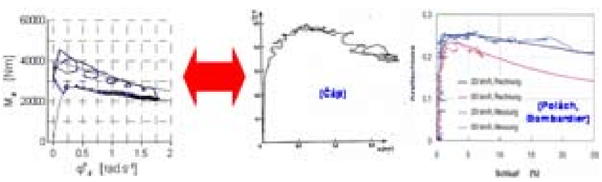
The release method is very quick method, good for not very complicated systems, with not very complicated kinematic bindings. Lagrange equation of 2nd type – method good for systems with many degrees of freedom and mainly for systems which consider complicated kinematic bindings.

On one side, on traction motor rotor, there is a function of tractive moment and on the other side, on wheelset – function of adhesive moment. This moment is non–linear and the exact modelling is very difficult. The tractive moment is independent on electrical parameters of motor and its regulation is nonlinear. The precision modelling of adhesion moment is very difficult and the shape of adhesion characteristics is strong non-linear.

The adhesive non-linear model, used for torsion modelling, makes use of theory presented by Freibauer (1983) and experimental work of Čáp (1988). The distribution of stress is not ellipsoid, according to classical Hertz contact theory, but a composite of 2 bodies: section of roller and section of ball.

The hypothesis about the connection between adhesion curve and friction curve was experimentaly validated by many authors. The shape of friction curve can be approximated by non-linear experiental function presented by Čáp (1988).

Downgrade adhesive conditions are shown as adhesive waveform deformation of running speed dependence on relative creeping and next parameters: especially non-linear friction coefficient, characteristics of steepness or combination of these influences. Time alterations exhibit adhesive characteristics at simulation (decline and resulting return to original conditions).



a b

Fig. 15. The comparison of adhesion characteristics shape

The 3D-dependences of adhesive characteristics on drive speed, friction coefficient and steepness are presented in Fig. 14.

Picture in Fig. 15a shows working adhesive characteristics course of moment M_a. Depiction serves, among other things, for check on rightness of algorithm function. The course is comparable with measured real characteristics shape (Fig. 15 b) according to authors Čáp (1988) and Polách (2002) with typical unstability bifurcations in zone of slip. Very significant in adhesion model is friction coefficient, that is not constant but dependent on creep velocity and adhesion conditions of adhesive surfaces.

The traction moment functions at rotor of traction motor. We consider functional dependence at angular speed of rotor.

Traction moment is also dependent on electric parameters of motor, power supply and regulation. Model of electric part of drive is very dificult. For simple solving it is possible to use the linearization around of working point, because transitional actions orginate at relatively small intervals of characteristics. The linear dependence is very real in case of direct-current motor with separate excitation.

For precision simulations we must perform modelling of real shape of characteristics and look at dependence of traction moment on electric parameters and electrical network dynamics. If it is required to exactly describe the dynamics of motor near to interaction with loading moment, it is necessary to specify the model by the influence of oscillating current in dependence on angular speed and influence of inductivity, that is analogous to damping in mechanics system, and is bound up with current time difference. The equation of circumference engagement of direct series motor is used according to literature, for example presented by Meźyk (2002) or Novák (2002).

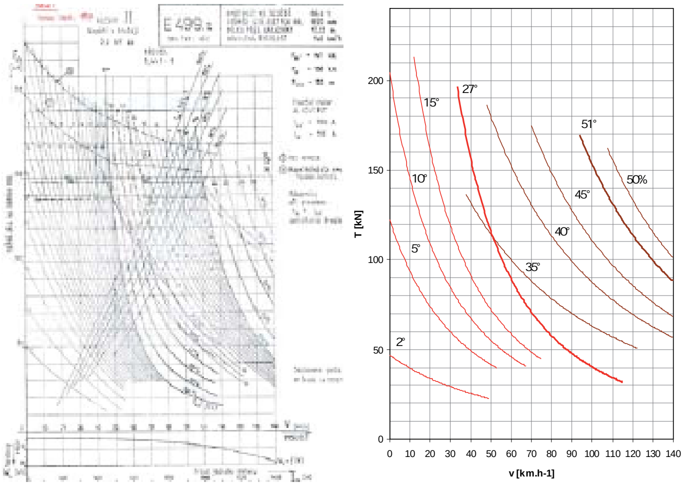


Fig. 16. The real tractive characteristisc and modelling courses

The real tractive characteristisc and modelling courses are presented in Fig. 16.

The adjustment of equation is necessary, so that we obtain the record of equation, formally the same as with Newton´s form of equation of motion by reason of using numerical methods. We present the dependence of electromotor moment on rotor angular speed and electric parameters. We can see, in Fig. 17, that around of working point, there is a range of transitional dynamic actions, the shape of characteristics approached to lines and we can linearize them.

The modelling of asynchronous (AC) motor is very difficult and a simplified solution was used here. According to Buscher (1995) and some other authors it is possible to express the independence of moment of asynchronous motor on relative creeping of motor, respectively, dynamic angular speed of rotor as Kloss's equation. The electrical parameters (see Buscher (1995)) for AEG motor of electric locomotive class 127 Eurosprinter were used.

Figs. 18 and 19 illustrate testing the modelling of characteristics. It is necessity to inform that presentation of dependence is correct for static, stationary state. The approximation to real state is possible by method of superponce higher harmonics, that at real electrical motor exists, for example the simple polyharmonics function.

4. Transitional action at wheelset torsion dynamics

At torsional driving system (Fig. 20) of railway driving vehicles, the dynamic transitional process of mechanic oscillation exists. The dynamics of transitional actions occurs in these cases:

- Influence of real vertical rail deviations on torsion dynamics – influence of kinematic bindings between vertical and torsional relative motions.
- Cooperation cases of adhesive characteristics with drive motor characteristics under downgrade adhesive conditions.
- Cooperation cases of adhesive characteristics with drive motor characteristics in the course of transition between traction running and braking.

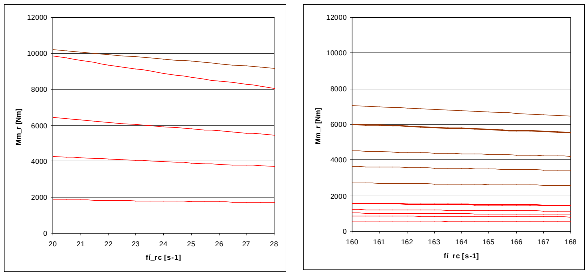


Fig. 17. Moment characteristics of one motor at rotor coordinates for the assumed range of simulated transitional actions

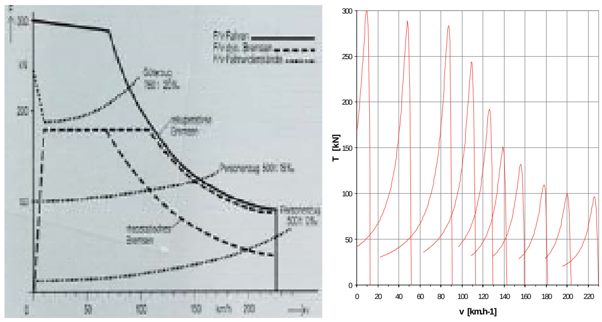


Fig. 18. The real tractive characteristisc and modelling courses

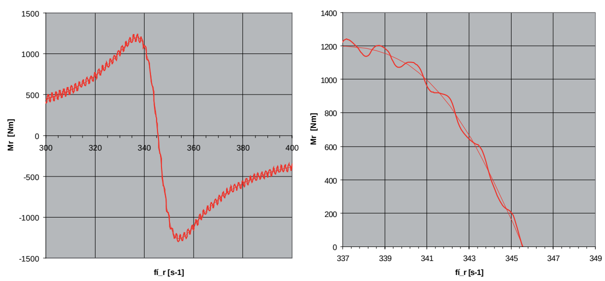


Fig. 19. Testing moment characteristics of rotor coordinates before implementation to computing; left: complete characteristics for speed 140 km/h; right: useful section of characteristics in the range of presumptive transitional actions; 1st harmonics with parameters: amplitude, frequency, is superponed

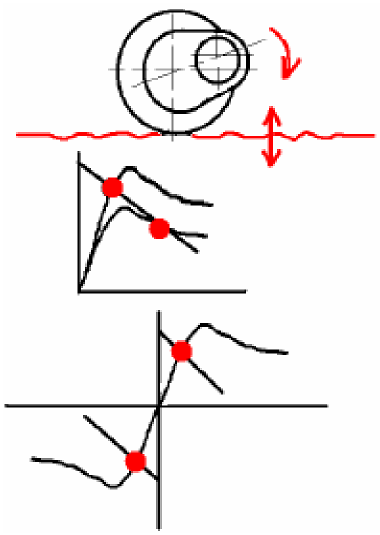


Fig. 20. Illustrative figures of three basic types of excitation of wheelset torsion dynamics

The vertical rail deviations have influence mainly on drive constructions, where bindings between rotation motion and vertical straight motion exist. Typical representative is the construction, where gearbox is beared on the axle and hanged on bogie frame. It is the case of electric locomotive Skoda. In the drive constructions with hollow joint shaft these bindings do not exist and influence is very small. In the vehicle running on real vertical rails, deviations oscillate vertical wheel force and through adhesion mechanism oscillate tangential forces and moments at drive. This influence is small on good track.

Keynote of dynamic transitional action simulations of rise, continuation and extinction (handhold) of wheelset slip consists in model cooperation of adhesive characteristics with drive motor characteristics. Characteristics are secant in the working point. Several cases of characteristics cooperation exist. The next figures show subcritical and supercritical curve cooperation:

- Subcritical cooperation: Engine moment is crossing new adhesive characteristics in its effective parts, or arising slip is handhold by downward branches of new characteristics.
- Critical cooperation: Engine moment curve is lying on the adhesion characteristic curve. It is a limiting condition.
- Supercritical cooperation: Drive motor torque characteristics has so small steepness, that lies over new adhesive characteristics and is not crossing it. Systems behaviour depends on a short time, in which arising slip is handhold by restoration of adhesive conditions. It also depends, if it is before or under the point, where engine graph crosses the original adhesive waveform. As far as restoration of adhesive conditions occurs before thereby point, there is a surplus of adhesive moment – the system slows down and converges into stable condition. As far as the system gets behind this point, amplitudes trend to henceforth growth.

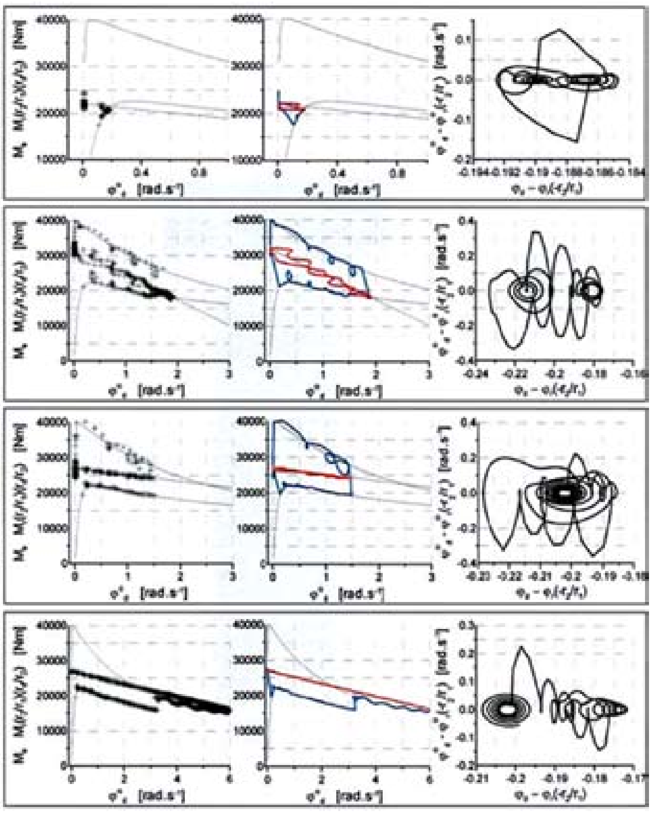


Fig. 21. Three basic types of excitation of wheelset torsion dynamics (output from simulation calculation)

Change of drive moment was designed at simulation between 4th and 7th seconds. Return to original engine characteristics followed after 7th second. Two cases are shown: decrease of driving moment to zero (M_m = 0) and the second case demonstrates the simulation of transition action between traction and brake sections of adhesion characteristics.

It is presumed that clearance in torsional system has considerable influence on these transitional modes. There is clearance in gear and joint of cardan shaft. If the clearance in gear and joint clutch is about 2 mm, then transformed angle clearance in axle axis is about 0,003 rad.

The simulation of decrease of driving moment to zero (M_m = 0) was designed as change of drive moment characteristics between 4th and 7th seconds of simulation calculation. The graphs show a high frequency of oscillating which is followed by clearances in torsional system. The working characteristics of binding is shown on the right of this graph (Figs 21–23). In reality we can interpret it as a high frequency beating tooth of gear.

We can analyse the bifurcations of working points. The rotor coordinate after start of transition process forms traction to braking "ask" and converges on the new stability of working point. After start of opposite transition process from braking to traction there comes a return to the original working point.

The precision of torsional clearance of gear modelling has significant influence. According to Meżyk (2002) it was used for interpolation of linear binding characteristics by 6 degrees polynom.

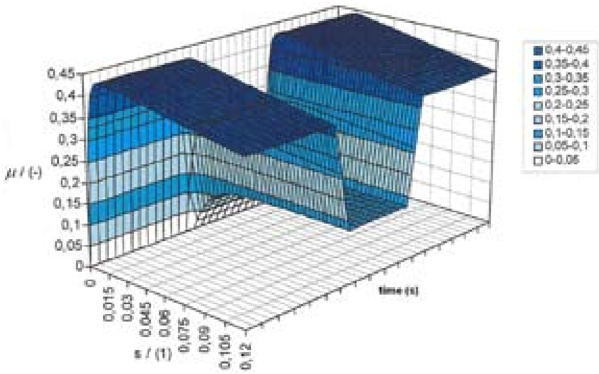


Fig. 22. Time simulation of downgrade adhesion and return

5. Solution at frequency area

Solution at frequency area presents the frequency analysis, or modal analysis, i. e. eigenvalue and eigenform problem. The algorithm of modal analysis of torsion dynamics system is necessary to use for obtaining the eigenvalues. First, we can use the very simple Holtzer´s method and next method, which we show, is a more complicated method in the system for complex roots solution.

Holtzer´s method is useful for conservative dynamics system (system without, or with very small damping). The first mass amplitude is selected e.g. 0, 1, 10. For every eigenfrequency (that is evaluated or computed by other simplified methods) we compute the single binding deformations, moments in the bindings and amplitudes of single rotation

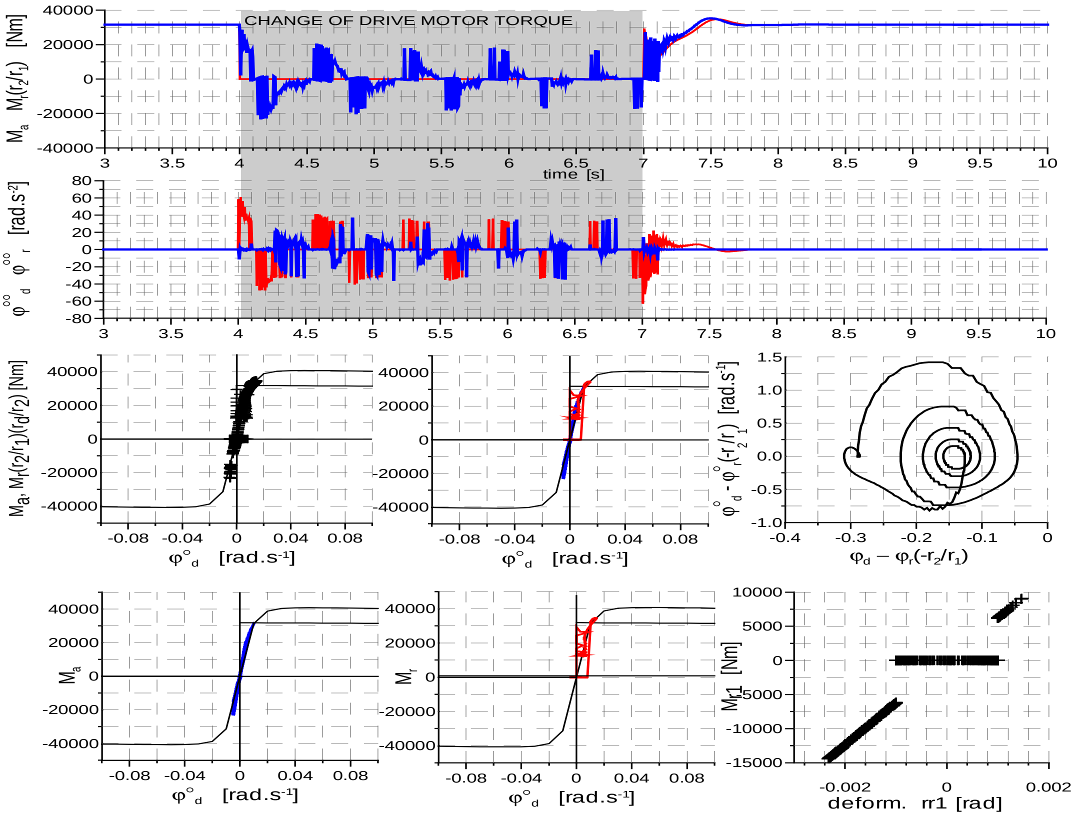


Fig. 23. Stop of traction moment function Mm = 0

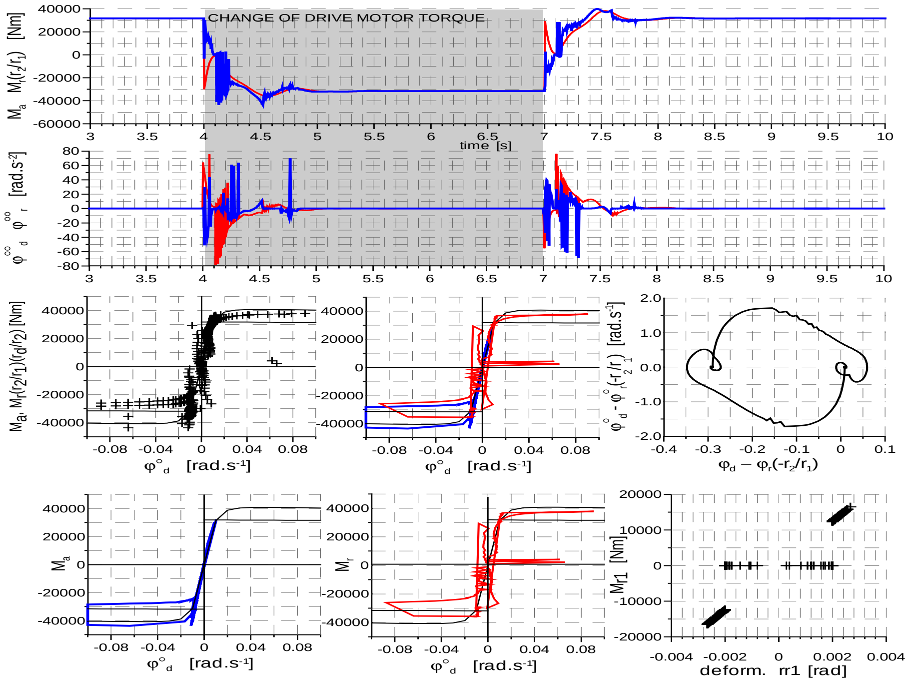


Fig. 24. Transition action between traction work and motor braking

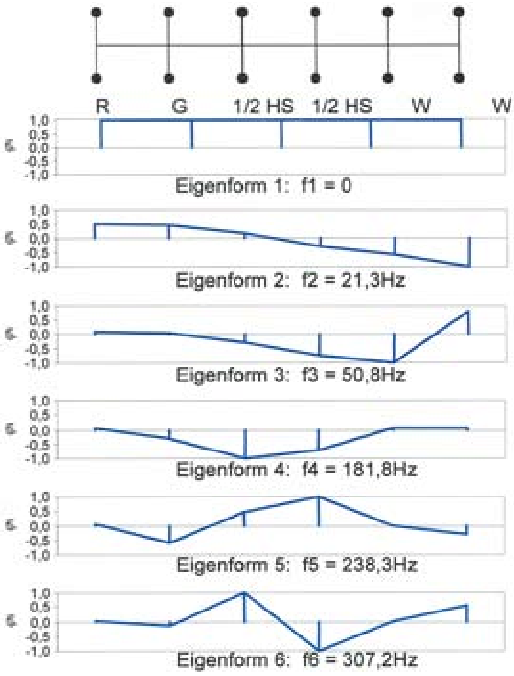


Fig. 25. Eigenform values for 6 degrees of freedom of torsional models of Eurosprinter wheelset drive

masses. Eigenform amplitudes can be normated – at mass with largest amplitude, or at first at any mass.

We can turn attention to nodal points. How much is a nodal point of a single eigenform? The first eigenform of oscillation does not have the nodal point - it is free rotation of motions of all masses. The 2nd eigenform of oscillation has 1 number of nodal points, the 3rd eigenform of oscillation has 2 numbers of nodal points, etc. The *N*-eigenform of oscillation has the (N-1) nodal points.

Now we turn attention to method of eigenvalues solution. Dynamical model can be mathematically described by system of linear (or linearized in parts), homogenous equations. We can make up the systemmatrix. This matrix is nonsymmetric and is transformed to the symmetric form and then at next steps is transformed by *QR*-algorithm (presented by Press *et al.* (1996), where Matrix $\mathbf{Q}_k^H$ is hermitian conjugate (tj. transponed and complex conjugate) regarding $\mathbf{Q}_k$ orthogonal matrix. After iteration running eigenvalues are located at main diagonal of matrix $\mathbf{R}_k \cong \mathbf{A}_k$ and they are complex conjugate $\lambda_j = Re_j \pm \mathrm{i}.Im_j$. Real part represents damping and imaginary part represents eigenfrequency oscillation. Around of adhesion characteristics peak eigenfrequency is retuned.

The knowledge of eigenforms has the practical use for basic tuning of torsional stiffness and dampings of the system and their optimization (Figs 25–26).

6. Conclusions

Full report knowledge of objective dynamic transition phenomenon is necessary for solving the problem of drive regulation and to predicate the load of its main parts. It allows as well to judge the dynamism influence on interaction with the track if need be brought out in the special cases of attrition. Research was successful in compiling algorithm which makes it possible to judge the drive in a few characteristic transitional actions. It makes possible, for example, to evaluate reaction of the system to changes of input parameters. It also makes possible to perform e.g. simulation of slip-resistant regulation. Accuracy of the function of algorithm was judged by partial verifications of adhesive waveform shapes and engine characteristics, namely with measured characteristics according to several authors. Very important is the verification of simulation records by experimental records (as far as they are

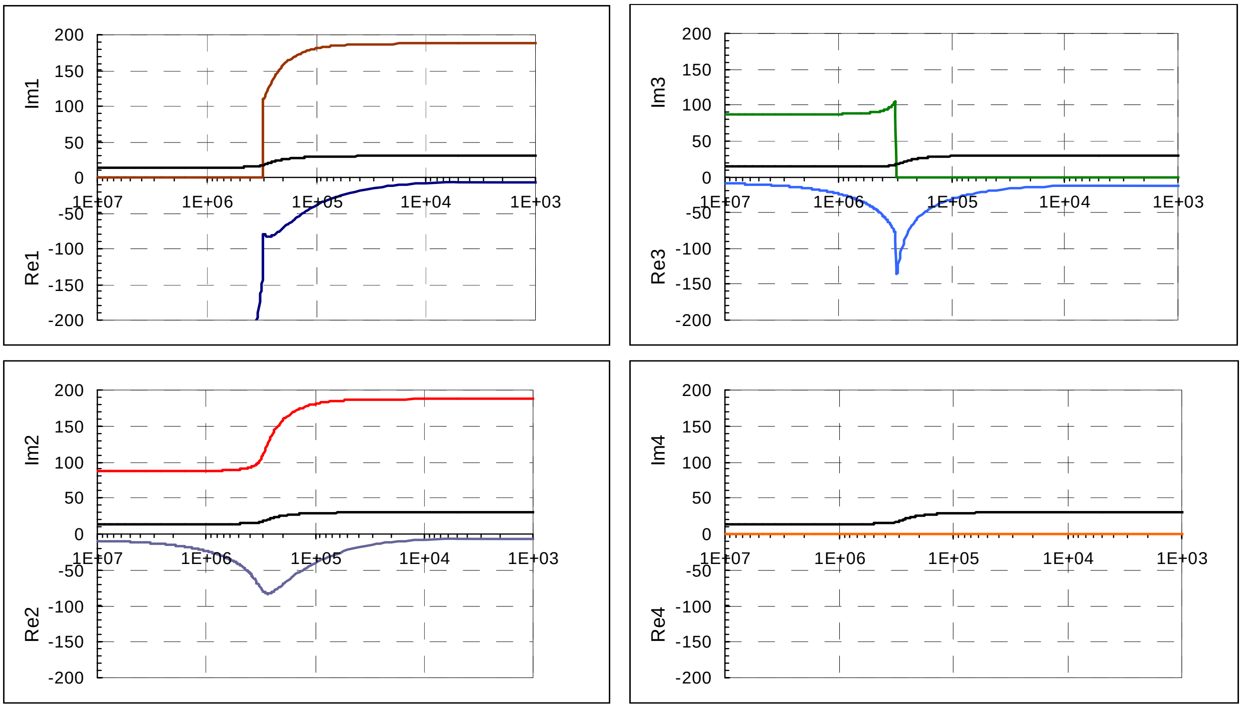


Fig. 26. Courses of eigenvalue roots of system before and after steep start (crossing through the adhesion characteristics peak)

at disposal for the given case or their fulfilment is real). The verification was carried out by using not only confrontation record but also a single part of algorithm. Most often depiction into graph helps judge the deformation and force in bindings, namely before early in initiation of final simulation reckonings. Introduced problems are being solved by Experimental centre of railway vehicles at The Jan Perner Transport Faculty, University of Pardubice, Czech Republic and research results are published in the proceedings of some conferences and colloquia, for example Lata, Čáp, Pokorný (2003), Lata (2005) and Lata (2006). In our faculty the testing adhesive stand of wheel-rotary rail is built (see Lata, Čáp (2007)).

References

Buscher, M. 1995. Radschlupfregelung zur maximalen Kraftschlussausnutzung bei elektrischen Traktionsantrieben [The regulation of wheelset creeping in exploitation of maximum drive force for electric driving vehicles]. Technische Hochschule Darmstadt. Shaker Verlag, Aachen (in German).

Čáp, J. 1988. Některé otázky vztahu třecí a adhezní charakteristiky ve styku kola s kolejnicí [New railway technique. Some questions of relation between friction and adhesion characteristics at contact wheel–rail], Prague, Czech Republic, *Žel. technika* 2: 66-69 (in Slovak).

Freibauer, L. 1983. Adheze kola na vozidlové dráze. 7. ved. Konference VŠDS [The adhesion of wheel on track], in *7th Scientific Conf.* VŠDS Žilina, Slovak Republic, 214–219 (in Slovak).

Meżyk, A. 2002. Analiza i ksztaltowanie cech dynamicznych napedów elektromechanicznych [The analysis and evaluation of the dynamic features of electromechanical drives]. Gliwice (in Polish).

Lata, M.; Čáp, J.; Pokorný, M. 2003. The influence of adhesive characteristics on dynamics of wheelset drive torsion system under downgrade adhesive conditions, in *XVI mezinárodní konference Súčasné problémy v koľajových vozidlách ProRail2003*. Žilina, Slovensko, 10: 55–65.

Lata, M. 2005. The transition dynamics phenomena of drive torsion system in the course of transition between traction running and braking, in *Proc. of the 22nd International Colloquium "Advanced manufacturing and repair technologies in vehicle industry"*. May 18–20, Czestochowa, Orle Gniazdo, Poland, 2005, 4–8.

Lata, M. 2006. The modeling of dynamics of the hollow joint shaft wheelset drive, in *Proc. of the 23rd Int. Colloquium "Advanced manufacturing and repair technologies in vehicle industries"*. May 10–12, Köln, Germany, 137–142.

Lata, M.; Čáp, J. 2007. The reconstruction and experimental possibility at the individual drived tram wheel stand, in *Proc. of the 24th International Colloquium "Advanced manufacturing and repair technologies in vehicle industries"*. May 22–24, Svitavy-Sindleruv haj, Czech Republic, 2007, 153–158.

Novák, J. 2002. *Elektromechanické systémy v dopravě a ve strojírenství* [The electromechanics systems in transport and machine industries]. Publisher ČVUT Praha (in Czech).

Polách, O. 2002. Optimierung moderner Lok-Drehgestelle durch fahrzeugdynamische Systemanalyse [Optimization of modern locomotive bogies by way of vehicle systems dynamic analysis], *EI-Eisenbahningenieur*, Publishing Eurailpress, Germany 53(7): 50–57 (in German).

Press, W. H.; Flannery, B. P.; Teukolsky, S. A.; Vetterling, W. T. 1996. *Numerical recipes in Pascal. The art of scientific computing*. Cambridge University Press. Cambridge, USA.