TEKA. COMMISSION OF MOTORIZATION AND ENERGETICS IN AGRICULTURE – 2014, Vol. 14, No.1, 312-319

A theoretical evaluation of locomotive wheelsets tires wear rate

Ilya Tsyganovskiy

Volodymyr Dahl East-Ukrainian National University,
Molodizhny bl., 20a, Lugansk, 91034, Ukraine, e-mail: ilyats@list.ru

Received January 21.2014: accepted February 10.2014

S u m m a r y A theoretical estimation of locomotive wheelset tires' wear intensity with the use of different contact models is presented in the paper. The received difference in the results of modeling varies from 40% to 100% depending on initial pre-conditions of contact model.
K e y w o r d s : wheel, rail, wear

INTRODUCTION

Today with the purpose of cost effectiveness on model experiments it is expedient to create the virtual prototypes of locomotives. The so-called "multibody dynamics systems" are used for this purpose. Except solving dynamics tasks, the obtained data about position and stress state of wheel - rail contact patch can be used in a further analysis for prognostication of wear (evolution) of wheel and rail profiles and for the modeling damages caused by rolling contact fatigue .

OBJECTS AND PROBLEMS

The most reliable theoretical results can be obtained only when taking into account the complexity of locomotive wheels – railway tread contact process in real-life environment. Because of importance of this task many scientific works are devoted to it [9, 12, 14, 25 etc.]. Two approaches can be marked out for adhesion force calculation.

With the first approach adhesion forces are calculated from the approximate expressions, received by authors [17, 18, 19, 23], depending on the number of factors. Main factors are vertical load, vehicle speed, relative slippage, frictional conditions. The advantages of this approach is its simplicity and high calculation speed. The disadvantage is that it is not possible to obtain force and temperature characteristics of contact patch.

With the second approach a mathematical modeling of wheel and rail contact is performed [10, 11, 20, 21, 22]. Since in general it presents significant difficulties, some simplifying assumptions are often made, that allow to separate initial problem into two less complex problem: normal and tangential. The typical assumption are that contacting bodies' dimensions are much greater than contact zone (this assumption allows to treat contacting bodies as elastic half-spaces) and that the difference between normal elastic displacements, caused by tangential tractions in contact zone, is zero.

The aim of normal contact problem is to find a contact patch shape and normal pressure distribution within it. Then the obtained data is used as initial data for tangential contact problem, that aims to find a tangential traction distribution and creep forces.

The normal problem is usually solved analytically with a use of Hertz theory [8], but it calls for bodies to have constant curvatures in contact zone. This limitation is often violated for worn wheel and rail profiles. In this case numerical non-hertzian or semi-hertzian solution should be used, and most of them are based on Bousinesque – Cerutti solution for unit force acting on elastic – half space [9, 13, 14].

There are two different approaches for solving tangential problem.

With the first approach it is assumed that contact is separated in slip (sliding) zone and stick (adhesion) zone. The most popular solution for this approach is Kalkers' FASTSIM algorithm and it's various modifications [11, 12, 20, 30].

Since that stick zone can exist only with a very small (elastic deformation order of magnitude) relative wheel – rail slippage values [28, 31], with the second approach the possibility of it's existence is ignored and it is supposed that the whole contact area is covered by slip zone. An example of this approach is developed by Golubenko school semi-empirical method of solving the tangential problem using experimental dependence of friction coefficient on the temperature in contact zone for different frictional conditions [14]. It should be noted that unlike FASTSIM, this model provides the correct results when modeling locomotive motion in traction (breaking) regime.

According to the results [29, 32], the approximation of the contacting bodies with elastic half-spaces and separation the contact problem into normal and tangential problems for wheel-rail contact in rail gauge corner zone, where lateral curvature radius is 15 mm, can bring significant error in calculations. Besides, existing contact models do not consider the lateral bending of railhead. As it was studied by author, even 0.2 mm relative displacement between rail foot and railhead leads to significant changes in the contact patch shape, and according to JSC "VNIIZhT" (Railway Research Institute) researches, the difference between rail head and foot displacements can exceed 60 mm.

At the Railway Vehicles department of Volodymyr Dahl East Ukrainian National University a mathematical model of quasistatic contact of wheelset and track was developed, that takes into account rail lateral bending, load redistribution in case of two-point contact, increases the reliability when modeling contact in rail gauge corner zone, and takes into account the friction coefficient dependence on the contact temperature [2]. After a model discretization, a computer program called *VDEUNU CONTACT* (*Volodymyr Dahl East Ukrainian National University Contact model*) was developed to study wheel-rail contact problems. Since in problem solution the iterative algorithm is used, the program can't be used directly during the vehicle dynamics simulation. Therefore it was used for compilation of so called "contact look-up tables". Then during the vehicle dynamics simulation the data about contact patch parameters and creep forces is read directly from these pre-calculated tables. The program FASTTAB was also developed, specified for data reading from look-up tables during the dynamics simulation.

The verification of *VDEUNU CONTACT* program was carried through several steps [15].

On the first step, to compare the solution of normal problem to other existing theoretical solutions, the *VDEUNU CONTACT* program was used for passing through the Manchester contact benchmark. The aim of the test is to provide for the end user the choice of the contact model for the particular modeling situation.

Two cases are suggested for benchmark. Case A is supposed for modeling of single wheelset contact with rails with prescribed motion conditions. Case B is supposed for modeling railway vehicle (freight wagon with two bogies) for a study of its dynamical behavior. This case is now under revision.

Case A aiming to solve normal (Case A-1) and tangential (Case A-2) contact problems. In turn, case A-1 is divided on two sub-cases A-1.1 and A-1.2, that differ from each other with input parameters.

After the specification for Case A was developed, an invitation to take part in it was in open access during the 2006 year. Total 10 software packages developers confirmed their participation. The results of Cases A-1 and A-2 can be found in paper [27].

The size and form of contact patch are very important factors, influencing traction, wear and rolling contact fatigue. On Fig. 1 is shown the dependence of contact patch area from lateral displacement for right wheel-rail pair. It can be seen, that before the flange clearance is exceeded, the results for all programs are near similar. Once the flange clearance is exceeded (lateral displacement more than 6 mm) the largest contact size is four times that of the smallest. The reason for that can be the difference in contact points detection and methods of solving normal contact problem. That's why it makes sense to compare the results from VDEUNU CONTACT to experimental data using the results from [16, 24].

The experimental set- up for ultrasonic detection of wheel- rail contact parameters consists of focusing transducer, ultrasonic pulser – receiver (UPR), a digital oscilloscope, a control PC, a scanning table (automated with x, and y-stepper motors) and a load frame. Two wheel - rail pairs were used for experiments, the ne ones and the worn profiles taken from heavily worn in service components. The profiles were digitized with MiniProf device. The wheel and rail specimens were cut from actual wheel and rail sections. The rail specimen is fixed to the upper plate, and the wheel specimen to lower plate. The specimens are moving relatively to each other as in the real wheel – rail contact conditions. To fulfill this requirements a grid was marked on the lower plate, that was used to set up lateral displacement and yaw angle. The load frame consists of fixed upper plate and moving lower plate The wheel and rail specimens are loaded with hydrocylinders , and focusing transducer is located above them in the reservoir filled with distilled water. The control signal is coming from PC, and UPR brings the focusing transducer to excitation. Then focusing transducer sends ultrasonic wave and gets the reflected signal form contact area. The sent and reflected signals are then shown on the digital oscilloscope and stored in memory. After finishing the measurements in prescribed point, with the scanning table focusing transducer goes to the next point until the whole contact area is not scanned (in paper [16] 0,25 step was used in and directions).

As it can be seen from paper [15], the numerical solution from VDEUNU CONTACT and experimental data from [16, 24] has good agreement.

For case A2 (tangential problem) of Manchester Contact Benchmark the creepages are input parameters, and for VDEUNU CONTACT they are output parameters. That's why case A-2 study can't be preformed.

The verification of VDEUNU CONTACT was performed by means of comparing creepage – friction coefficient curves, built with different methods. The most common contact models (FASTSIM (J. Kalker), ADH (O. Polach), T. Muller and D.Minov) were used as alternative to VDEUNU CONTACT.

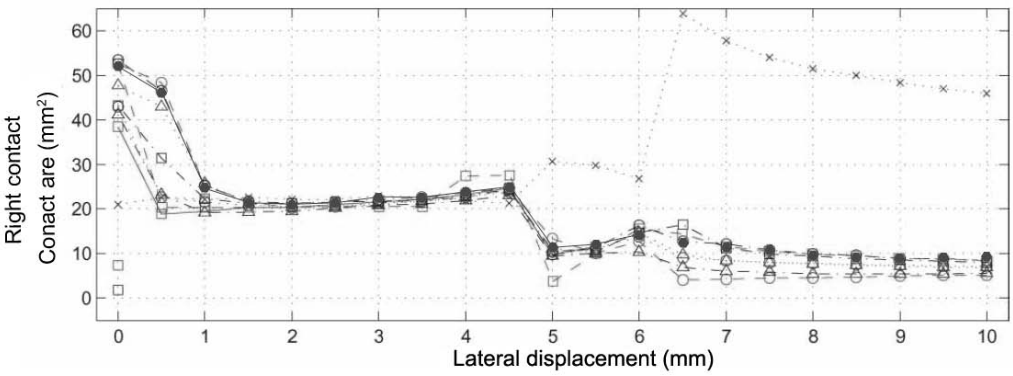

Fig. 1. The dependence of contact patch area on lateral displacement for right wheel – rail pair for different codes [15].

The next initial conditions are accepted. Wheel and rail profiles are new according to Ukrainian State Standards. Lateral displacements and yaw angles are zero. Vehicle speed is 20 m/s. Wheel vertical load - 100 kN. Two frictional cases are studied. For the first case the friction coefficient is 0.38 (clean, dry surfaces), for the second case – 0.07 (surface, coated with grease). For VDEUNU CONTACT program corresponding experimental friction coefficient dependencies on temperature were used.

For high friction coefficient values, a critical creep value, calculated with VDEUNU CONTACT, ADH, Muller and Minov is 0.02…0.025, and there is no major differences in ascending branch (no more than 7%). The results of FASTSIM, coincides with others only at very small (near 0.001) creeps. The reason is that FASTSIM was developed at the base of Kalker’s linear theory, the basic assumption of which is vanishingly small slip between wheel and rail. The falling friction branch for FASTSIM, ADH и Mullers models are absent.

The experimental studies shows the under the bad frictional conditions the critical creep is increasing. But as it was found out, the critical creep, calculated with FASTSIM, ADH и Muller, as it can be predicted, shifted to the zone of microcreep (up to 0.001), for Minov program left the same, and only for VDEUNU CONTACT increased to 0.04.

Taking into account so substantial difference in the modeling results, the questions raised about the ability of using VDEUNU CONTACT program for railway vehicle dynamics modeling. The verification was carried through the Manchester Dynamics benchmark [7].

On the international symposium «Computer Simulation of Rail Vehicle Dynamics», that took place at Manchester Metropolitan University 23 and 24 June of 1997, were agreed the etalon benchmarks for railway vehicles dynamics simulation. The aim of benchmarks is to provide for researchers, studying railway vehicles dynamics, evaluation the compatibility of different software packages. The initial results, calculated in the most popular software packages, were shown at the special meeting on 15 December, 1997. There were some corrections in the vehicle and track models, and in the results presentation to avoid misconstrues. The full description of benchmarks initial data can be found in [7]. The etalon models were chosen to cover the most common vehicle and track models, that are used in vehicle dynamics simulation. Despite the both vehicle models are simplified, they include examples of complex elements. The detailing level of models is chosen to provide every software developers to build the model. There is no restrictions in the documents about exact models usage, including wheel – rail contact model.

For modeling the «Universal Mechanism» software was used. VDEUNU CONTACT model was integrated in Universal Mechanism as dynamic link library. As alternative to VDEUNU CONTACT FASTSIMA (FASTSIM modification) was used.

As it was cleared up [15], for all cases simulation results do not depend on the chosen contact model. It can be explained with two facts. First of all, the friction coefficient for Manchester dynamics benchmarks simulations is chosen to be 0.4. Secondly, the maximal stated creep value during simulations, was 0.005. In other words, the section, called “zone of FASTSIM validity” was used on traction curve. In this zone there is no major difference which contact model is used. The influence of contact model used to be seen while modeling locomotive motion in traction regime. The choice of correct contact model is very important in prediction of wheel wear, as the wear of locomotive wheelset tires flanges is one of the main problems on ex - USSA railways [3, 26].

According to GOST 2823 – 94, the *wearing* is defined as a process of material loss from the friction surface of rigid body and (or) expansion of its residual deformation in friction conditions, shown in progressive change of body sizes, form and (or) body mass, and *wear* – as the results of wearing, expressed in prescribed units of length, volume, mass and so on. At the former USSA railways the wear rate of wheelset locomotive tires is calculated, which is measured in mm on 10000 km of haulage.

In the fundamental work of Sakalo an Kossov [25] was made a review of theoretical approaches for calculation of wheel and rail rolling surfaces wear. It is said, the most of comprehensively grounded and widespread wear models, widely adopted in railway vehicle branch, are based on the preposition, that the loss of material on the section of surface profile, is proportional to the material constant and the sum of local friction works. The wear model supposes a proportional dependence between the wear volume V_e an friction work A_r :

$$V_e = kA_r , \tag{1}$$

where k - wear index and its value, determined with experiments, is $10^{-4} < k < 10^{-2}$ mg/ (Nm). The value of k used for calculation must be selected to satisfy the operational conditions [1, 19].

Also in the paper [25] is admitted that one of the most adverse type of wear is wear of wheel flange and rail side face of outer rail in curves. This wearing is especially intensive in case of wheel- rail two – point contact The experimental test on 2ТЭ116 locomotive allowed to ascertain that lateral force can reach 62 kN.

In the book [5], which summarizes up-to-date international experience on the issues of wheel and rail interaction, is also admitted, that the wear magnitude is proportional to energy, that was dissipated during the process of overcoming the resistance to rolling with sliding of wheel on rail. The wearing of wheel and rail is determined by relative slip and pressure on contact areas. In turn, the relative slip and pressure depend on dynamical parameters of wheel and rail interaction. The wear is substantially determined by the characteristics of third body, that depend on presence of lubrication, weather conditions (humidity, rain, snow) and use of sending.

The aim of this paper is to compare the results of theoretical evaluation of locomotive wheelsets tires wear rate, received with the use of different contact models in a traction regime

RESULTS AND DISCUSSIONS

To succeed the research, the FASTSIM algorithm, original Golubenko school model and the model developed by author were integrated as contact models in mathematical model of ТЭ116 locomotive motion, which was developed in Railway department of Volodymyr Dahl East Ukrainian National University [4]. The next premises were made before the construction of the model:

- All bodies of the system (locomotive body, bogies’ frames, traction motor, wheelsets and wheel treads are considered perfectly rigid.
- Nonlinearities in axleboxes during the lateral run of wheelsets, in pivot units according to the lateral displacements of the bogies, in the support of the locomotive body during the yawing are considered.
- The action of the hydraulic and frictional oscillation dampers in axlebox suspension and in the bodie – bogie links.
- Train and locomotive running resistance forces are considered.
- The simulation is performed in the locomotive traction, braking and stopway regimes.
- A traction force value is determined for each wheel separately, depending on the linear velocity of the vehicle, sliding speed of the contacting bodies, frictional condition, wheel –rail profiles and their orientation.
- The longitudinal velocity of the locomotive is determined in the process of the motion differential equations integration and no limitation is put on it.
- A railway track is considered as discrete inertial beams of infinite length, which are laying on the elastic – dissipative or visco - elastic foundation and are under the influence of the vertical and lateral forces, applied at the wheel- rail contact points.
- Wheel tread and rail can have new or worn profiles.

• A wheel flange – rail friction is considered when the once flangeway clearance is exceeded.

• The electrodynamical processes in the engine action are considered.

• During the running process the longitudinal vibrations of the train are considered.

• Torsional stiffness of the wheelset axle is considered.

In the present paper, for the evaluation of wheelset tires wear was used an approach, presented in paper [6]. It is said there, that the intensity of wear can be judged by the power of friction. As during the vehicle dynamics simulation at every time point the relative slip speeds and friction forces between the whole interacting elements are known, than instant power of friction is calculated by multiplication of friction force on slip speed and is stored on PC as function of time or distance passed.

For the wear intensity determination the next expression is used in present paper:

$$I_{ij} = \frac{k_{lin}}{L}\int_0^T \iint_{S_C^{ij}} \left(V\cdot\varepsilon_x^{ij}\cdot\tau_x^{ij} + V\cdot\varepsilon_y^{ij}\cdot\tau_y^{ij}\right)dxdydt, \tag{2}$$

where: i – wheelset number, j – wheel number (1 – left, 2 – right), T – simulation time, V – vehicle speed, ε_x^{ij}, ε_y^{ij} – slip vector projections on x, y axis correspondently, τ_x^{ij}, τ_y^{ij} – tangential tractions on x, y axis correspondently, S_C^{ij} – contact area(s) on j–s wheel of i wheelset in t time point, k_{lin} – linear wear coefficient, L – distance passed.

The locomotive motion was studied in traction regime on dry rails with vehicle speed 60 km/h in 300 m curve. The wear intensity was calculated for tread and flange separately, as the new wheel and rail profiles were used, and when flangeway clearance is exceeded, two – point contact occurs. For the tread k_{lin} is set to 10^{-6} mm/kJ [1], and for the flange - $6{,}4\cdot 10^{-6}$ mm/kJ [19].

The modeling results are introduced in Fig. 2 and Fig. 3.

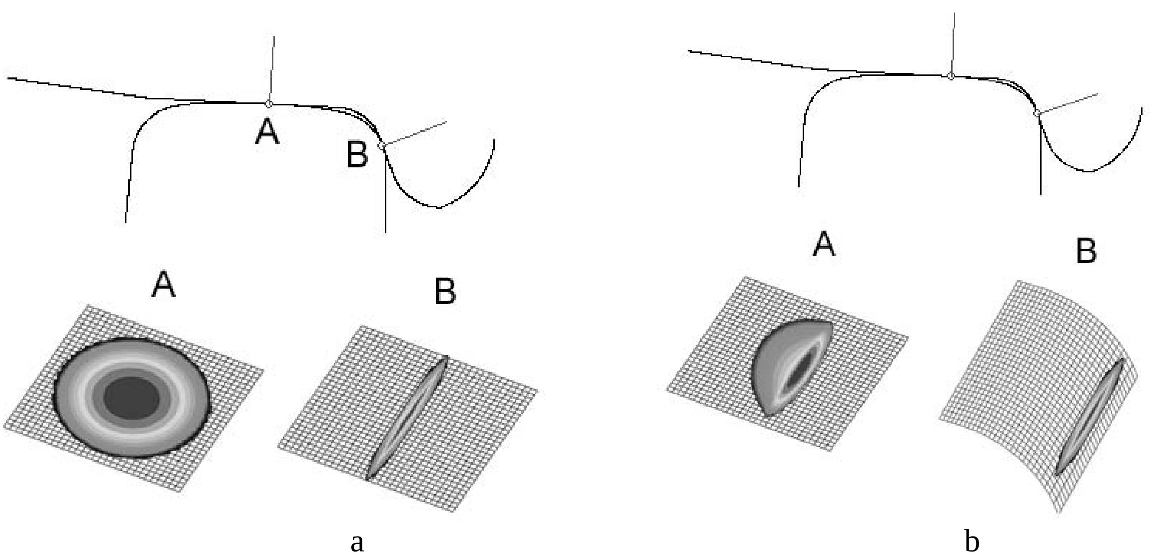

Fig. 2. Contact patches when flangeway clearance is exceeded with the use of original Golubenko school model (a) and developed model (b)

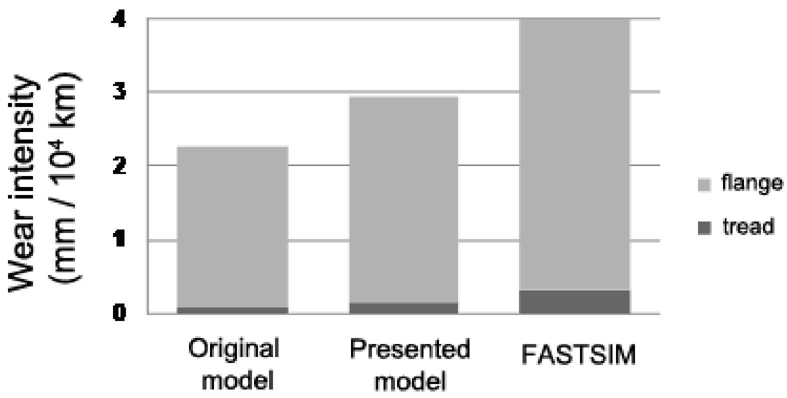

Fig. 3. The results of first wheelset tires wear intensity calculations with the use of different contact models

CONCLUSIONS

1. The difference in results obtained with the use of original model and FASTSIM is about 100% (see Fig. 2). It can be explained by the difference in definition of tribological behavior of contact.

2. The difference between the results of original model and developed model is about 40%. It can be explained by the introduction to the model additional parameters, that significantly changes the position and stress state of contact patch (see Fig.1).

3. The most reliable wheel tires wear prediction results can be obtained only with developed wheel-rail contact model, because it includes the largest number of operational conditions.

REFERENCES

1. **Braghin F., Bruni S., Resta F., 2001.:** Wear of railway wheel profiles: a comparison between experimental and a mathematical model, 17th IAVSD Symposium Dynamics of Vehicles on Road and Tracks , 43–45.
2. **Golubenko A., Kostyukevich A., Tsyganovskiy I., 2012.:** A wheel – rail frictional contact model, Bulletin of V. Dahl East - Ukrainian National University, № 5(176), 7-12. (in Russian).
4. **Golubenko A., Malohatko A., Klyuev S., Klyuev A., 2011.:** The application review on the rolling stock of devices for turn of wheel pairs in the horizontal plane, TEKA Commission of Motorization and Power Industry in Agriculture, V. XIA, 5-12.
5. **Gorbunov N., Golubenko A., Kostyukevich A., Kashura A., 2002.:** A prediction of the tractive and dynamical features of locomotives with simulation technique method. Luhanks, V. Dahl East - Ukrainian National University, 104.
6. **Harris W., Ebersöhn W., Lundgren J., Tournay H., Zakharov S., 2001.:** Guidelines To Best Practice for Heavy Haul Railway Operations (Wheel and Rail Issues), International Heavy Haul Association, USA, 408.
7. **Husidov V., Zaslavskiy L., 1995.:** Numerical modeling of coach vibrations when moving on straight and curved track. Bulletin of VNIIZT, Vol. 3, 18-25. (in Russian).
8. **Iwnicki S.,1998.:** Manchester Benchmarks for Rail Vehicle Simulation., Vehicle System Dynamics, 30:3-4, 295-313.
9. **Johnson K.L., 1985.:** Contact Mechanics. Cambridge University Press, Cambridge (UK).
10. **Kalker J.J, 1967.:** On the rolling contact of two elastic bodies in the presence of dry friction, Doct. Thes. Delft Universiti, 160.
11. **Kalker J.J., 1973.:** Simplified Theory of Rolling Contact, Delft Progress Report. University of Technology, The Netherlands, Vol. 1. 1973, 1-10.
12. **Kalker J.J., 1982.:** A fast algorithm for the simplified theory of rolling contact (FASTSIM program), Vehicle Systems Dynamics, Vol. 11, 1-13.
13. **Kalker J.J.,1990.:** Three Dimensional Elastic Bodies in Rolling Contact, Dordrecht. London.: Kluwer Academic Publishers, 314.
14. **Kalker, J.J., Piotrowski, J., 1989.:** Some new results in rolling contact, Vehicle System Dynamics, Vol. 18, 223-242.
15. **Kostyukevich A. I., 1991:** A numerical and experimental identification of the locomotive wheel – rail adhesion process, Dis. candidate of technical sciences, Luhansk, 230.
16. **Kostyukevich A., Tsyganovskiy I., 2013.:** Testing developed wheelset – track interaction model [E-resource]: http://nbuv.gov.ua/j-pdf/Nvdu_2013_10_17.pdf.

17. Marshall M. B., Lewis R., Dwyer-Joyce R. S., 2006.: Experimental Characterization of Wheel-Rail Contact Patch Evolution, Journal of Tribology, 128(3), 493-504.
18. **Minov D., 1965.:** The enhancement of tractions efforts of electric locomotives and diesel locomotives with electrical transmission, Moscow, 268. (in Russian).
19. **Menshutin N., 1960.:** The investigation of slippage of electric locomotives' wheelset under the tractive effort in operational conditions, Works of VNIIZT, Vol.188, 113-132. (in Russian).
20. **Masliev V., 2008.:** The dynamics of diesel trains with devices that reduce wheelset tires' wear, Kharkov, 288. (in Russian).
21. **Piotrowski J., Kik W.,2008.:** A simplified model of wheel/rail contact mechanics for non-Hertzian problems and its application in rail vehicle dynamic simulations, Vehicle System Dynamics, Vol. 46:1, 27-48.
22. **Piotrowski, J. , Chollet, H., 2005.:** Wheel-rail contact models for vehicle system dynamics including multi-point contact, Vehicle System Dynamics, Vol. 35, 361-407.
23. Polach O., 1999.: A fast wheel–rail forces calculation computer code, Vehicle System Dynamics, Vol.33, 728-739.
24. **Polach O., 2001.:** Influence of locomotive tractive effort on the forces between wheel and rail. Vehicle System Dynamics Supplement, Vol. 35, 7-22.
25. Rovira A., Roda A., Marshall M.B., 2011.: Experimental and numerical modelling of wheel–rail contact and wear, Wear 271, 911-924.
26. **Sakalo V., Kossov V., 2004.:** Contact problems of railway transport. Moscow: Mashinostroenie, (in Russian).
27. Sapronova S., 2010.: Modeling of locomotive wheel profile form, TEKA Commission of Motorization and Power Industry in Agriculture, V. XC, 270-278.
28. **Shackleton P., Iwnicki S.D., 2008.:** Comparison of wheel–rail contact codes for railway vehicle simulation: an introduction to the Manchester Contact Benchmark and initial results. Vehicle System Dynamics, vol.46(1), 129-149.
29. **Spiryagin M., Polach O., Cole C., 2013.:** Creep force modeling for rail traction vehicles based on the Fastsim algorithm, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 51:11, 1765-1783.
30. **Vollebregt E., Segal G., 2013.:** Solving conformal contact problems, Proceedings of the 23rd International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD2013), Qingdao, China.
31. **Vollebregt, E., 2011.:** FASTSIM2: a second-order accurate frictional rolling contact algorithm./ Vollebregt, E., Wilders, P. // Computational Mechanics, Vol. 47, Issue 1, 105.
32. **Vollebregt E., 2013.:** Numerical modeling of measured railway creep versus creep-force curves with CONTACT. Wear, doi 10.1016 / j.wear.2013.11.030.
33. **Yakovlev V., 1963.:** About applicability of Hertz – Belyaev theory for the calculation of contact stresses in railhead gauge corner and wheel flange, Works of LIIZT, Vol. 210, 121-123 (in Russian).

ТЕОРЕТИЧЕСКАЯ ОЦЕНКА ИНТЕНСИВНОСТИ ИЗНОСА БАНДАЖЕЙ КОЛЕСНЫХ ПАР ЛОКОМОТИВОВ

Илья Цыгановский

Аннотация: В статье при помощи различных контактных моделей проведена теоретическая оценка интенсивности износа бандажей колесных пар локомотивов. Полученная разница в результатах моделирования составляет от 40% до 100% в зависимости от того, какие исходные предпосылки заложены в контактную модель.
Ключевые слова: колесо, контакт, износ