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USE OF *"CONTACT"* IN MULTI-BODY VEHICLE DYNAMICS AND PROFILE WEAR SIMULATION: INITIAL RESULTS

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Abstract

This paper reports the first results of a new interface between the multi-body simulation software SIMPACK Rail and Kalker's wheel-rail contact software CONTACT. The main benefit is in the more accurate distributions of shear stress and micro-slip in the contact area, that together form the primary inputs to wear calculations. Further benefits of the interfacing reside in the ability to perform detailed contact analyses within the multi-body framework, and to investigate the importance of fully non-Hertzian calculations in more extreme situations such as derailment and limit-cycle studies.

1. INTRODUCTION

Multi-body simulation is today the most feasible method for the prediction of the safety, wear, fatigue and noise behaviour of rail vehicles. A multi-body system is described by a limited number of interconnected rigid or flexible bodies [6]. The behaviour of the system is then obtained through analysis (e.g. time-integration) of the equations of motion: The multi-body software computes the dynamic movement of and the interactions between the different components of the train and of the track. An important aspect concerns the frictional interaction between wheels and rails. Because of computational efficiency, simplified models are generally used [8]. Due to the rapid increase of computational power and due to algorithmic speed-up as well, it is nowadays feasible to use more detailed rail-to-wheel contact models in vehicle system dynamics simulations as well.

This paper presents the initial results of a new interface between the multi-body software SIMPACK Rail and Kalker's contact mechanics software CONTACT. After a brief introduction of these two software packages, we show the way that the interface is set up and present the initial results. Then we present the conclusions that can be drawn from the experiments performed thus far, describe the steps to be taken to complete the interface, and the plans for refinement of the wheel-rail contact modelling in the coming years.

2. SOFTWARE

2.1 Multi-body software

SIMPACK Rail [3] is an advanced multi-body package for the simulation of the dynamic running behaviour of railway vehicle systems on the track. In order to achieve a calculation speed sufficient for dynamic simulations with actual vehicles, the rail-to-wheel contact locations and forces are determined by means of an approximate, non-iterative method, called *equivalent-elastic*. Its results are usually accurate enough for the daily work of vehicle manufacturers, engineering service providers and operators, i.e. predicting hunting, derailment and traction forces and providing the excitations needed for passenger comfort and component fatigue analyses.

The equivalent-elastic method originates from an approach by Kik and Piotrowski [5]. To reduce the calculation effort even more, the actual contact patch shape is converted into an equivalent ellipse whose width and length are set equal to the maximum width and length of the interpenetration area, see Figure 1. An equivalent penetration is determined as the penetration of a circle segment that has the same width and cross-section area as the actual interpenetration. The well-known fact that the contact area is smaller than the interpenetration surface area [2] is deliberately ignored. Instead, the equivalent penetration is artificially increased by a constant factor, as explained in [5], to make it consistent with the interpenetration patch area. Using the equivalent semi-axes and new penetration, the contact forces can be easily determined by means of the Hertzian formulas and using FASTSIM for normal and tangential directions respectively. The forces are applied at the so-called contact reference point, which is located at the area center of gravity of the actual interpenetration cross-section area, see Figure 1.

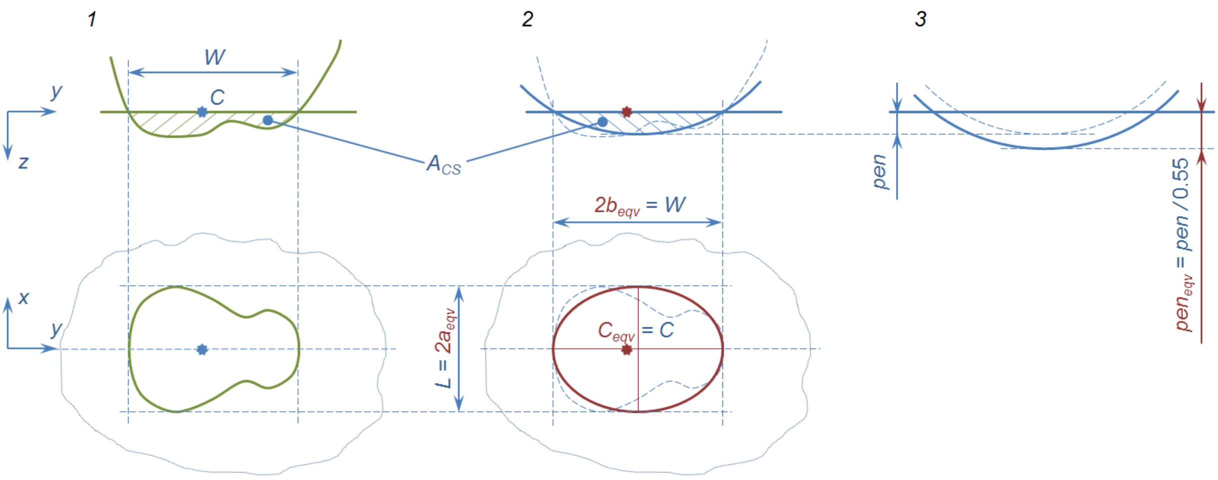

Figure 1 Equivalent-elastic contact method. W = width, L = max. length. A_{CS} = interpenetration cross-section area. C = contact reference point, pen = penetration or approach, a,b = semi-axes. The subscript "eqv" means "equivalent".

2.2 Kalker's software CONTACT

Over the years many different contact theories were presented, ranging from analytic and approximate approaches (Hertz' theory for the normal direction, Kalker's linear theory, Shen-Hedrick-Elkins, Polách, FASTSIM for the frictional behaviour in tangential direction) [8], via numerical half-space based approaches (CONTACT) [2] to full nonlinear finite element models [9].

CONTACT computes the size and shape of the contact patch that arises between two elastic bodies that are pressed together and are moving (rolling, sliding) tangentially. This involves the local geometry in the region near the initial contact point, which does not have to be Hertzian. Further the full tangential problem is solved too: for given relative motion (creepage) between the bodies, CONTACT determines the elastic deformations in the contact patch, the regions where local sticking and micro-slip occur, the frictional shear stresses in the contact interface, and the overall resulting forces that thus come about.

CONTACT is built on the theory of linear elasticity. The contact pressures are assumed to be concentrated in a small contact patch relative to the overall geometries, i.e. no sharp corners exist in and near the contact area. These assumptions allow for using the half-space approach. The deformation of a half-space due to a prescribed load on its boundary is known analytically. Using a superposition of such prescribed loads the problem is simplified. Instead of computing the elastic field in the wheel and rail interiors, as done in finite element models, the problem is restricted entirely to the contact interface.

CONTACT is considered an advanced simulation model for computing the frictional contact between wheels and rails. For instance it is considered to be a reference for the approximate models mentioned above, see, e.g., [8]. However, currently the element sizes that are feasible are much larger than the roughness that exists at the micro-scale. Therefore the true frictional processes are not resolved and are inserted in the model via the local friction law. For this Coulomb's law of dry friction is applied locally in each point of the contact patch.

2.3 Interfacing SIMPACK Rail and CONTACT

The so-called "CONTACT add-on" is an add-on module to SIMPACK Rail. It manages the transfer of the contact geometry and kinematics description from the multi-body core to CONTACT, as well as the transfer of CONTACT's results back into the solver and/or post-processing framework of the multi-body package.

Different possible usages of the interface are proposed:

- to use an approximate contact model in the dynamic simulation, and check its validity afterwards with CONTACT in post-processing mode;
- to inspect the detailed contact stresses in the multi-body framework, i.e. using SIMPACK for locating the contact point and preparing the inputs to CONTACT and then to inspect the CONTACT results;
- to compute the inputs for wear calculations without direct interaction to the dynamic simulation, i.e. using the post-processing mode;
- to use CONTACT directly in the dynamic simulation, particularly in extreme situations where more accurate contact forces are relevant.

In SIMPACK there is a separation between the actual time integration of the state vector (dynamic calculation) and the calculation of derived results (the so-called "measurements step", in this paper "post-processing mode"). In the integration a variable time step size is used, typically ranging from 10^{-6} to 10 s. The state variables are collected and are then used to evaluate the additional results at a user-defined fixed step size.

Currently CONTACT is made available only in the post-processing mode, see Figure 2: The shapes and relative orientation of the profiles, the positions of the contact reference points and the creepages are determined during the dynamic simulation with the equivalent-elastic contact method. These are then fed into CONTACT. The resulting contact forces and the (non-elliptic) shape of the contact patches are used for visualization only. In a later stage the interface will allow use of CONTACT in dynamics calculations too. Wear calculations are often implemented at the post-processing stage such that the current arrangement would be appropriate, but this coupling has not yet been realized.

The user prescribes for which wheels in the simulation CONTACT calculations are requested. Further configuration merely consists of the grid discretisation step size (e.g. 0.5 x 0.5 mm elements) and the configuration of the outputs. Also, a switch is provided between elliptical (Hertzian) and non-elliptical contact geometries. In the former case the semi-axes of the equivalent contact ellipse are used, in the latter case the full undeformed distance function is evaluated by SIMPACK and then used in the CONTACT case. Several output channels are available for each contact patch. These include all the expected values such as the total forces N, T_x, T_y and the creep torque M_z. Further the size of the contact patch and adhesion and slip areas are presented, as well as the maximum pressure and shear traction in the contact patch which are relevant for the prediction of rolling contact fatigue (RCF).

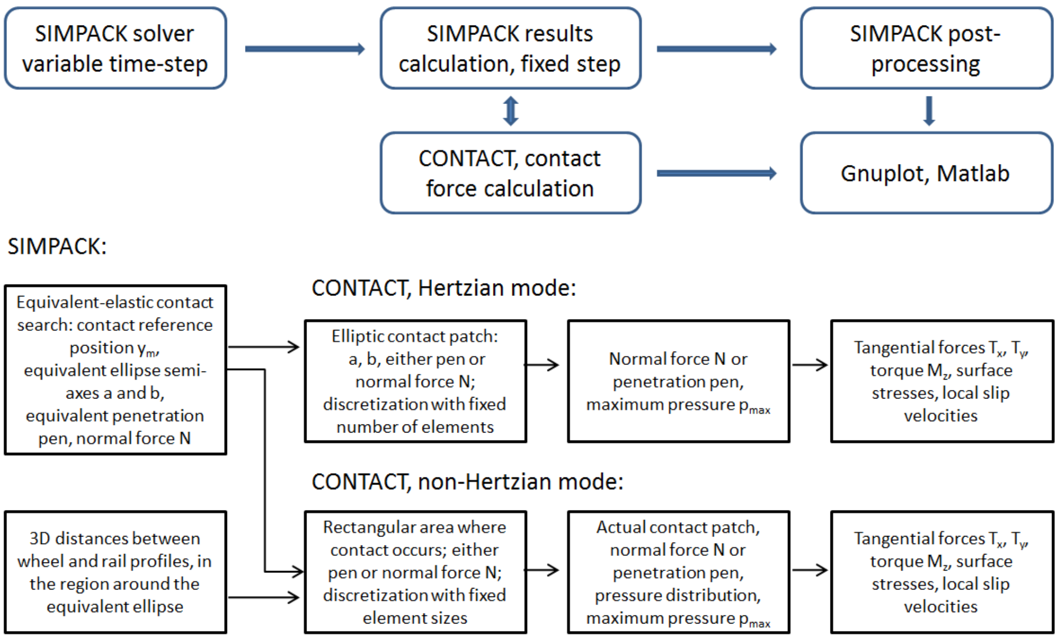

Figure 2 Interface multi-body core – CONTACT

3. TEST CASES

To assess the added value of CONTACT in the multi-body framework, several test cases have been analysed. For each case a comparison between the "multi-body" solution and the CONTACT solution was performed. The "multi-body" solution is the one that is currently used within the vehicle dynamics simulation, it is found by means of an equivalent Hertzian elliptic contact and the standard FASTSIM algorithm without extensions. The "CONTACT" solution is found by CONTACT, with the same input data as the "multi-body" solution.

- The first test case is derived from the Manchester Contact Benchmark [7], case A-1. This is a frictionless and thus more or less academic case but useful for a comparison of the fundamental solution processes.
- The second test case concerns a typical contact patch that appears on the flange, in a near-derailment situation. This case was also found in the Manchester Contact Benchmark, case A-2, but the wheelset load has been increased to 16 t to make the scenario more realistic. The profiles are the benchmark profiles.

- The third test case concerns a typical contact patch at the transition between tread and flange. This situation was found during quasi-static curving of a realistic 2-bogie local train vehicle model in a radius 150 m, 1450 mm gauge curve at 5 km/h. The profiles are S 1002 and UIC 60 (both new) with 1:40 rail cant. Besides RCF assessment, the shape of the contact patch at the transition may also be important for limit-cycle and running stability calculations.

Due to space limitations only test case 2 is shown here, which covers most of the findings of the other cases as well.

4. RESULTS FOR TEST CASE 2

In this case a wheelset is moved from a centered position to the right, with the yaw angle linearly increasing to 24 mrad. The wheelset y position is prescribed at the track level, in order to avoid the wheel-lift problem described by J.P. Pascal [4] when measuring y at the axle center height.. The overall results are shown in Figures 3 and 5 for the normal and tangential forces respectively. Figures 4 and 6 present the detailed results in the contact patch. In all cases the results concern the right-hand wheel of the wheelset at various lateral displacements y.

Two different approaches are provided for connecting CONTACT with the multi-body outputs. Either the (equivalent) penetration can be prescribed and the total normal force computed or the other way around, see Figure 3 (top). The results generally correspond well to each other, particularly in tread contact (y < 5mm) and flange contact (y > 7mm). A larger discrepancy is found in the transition in between. When CONTACT is used in the dynamic simulation, the contact forces are needed such that the option with penetration prescribed will be used. In the post-processing mode the alternative proves more convenient, because it allows to compare the effects of normal and tangential contact approaches separately. This is the method that is used in all following results.

Figure 3 (bottom-right) demonstrates one of the main reasons for a non-Hertzian contact simulation in the rail-to-wheel contact: The maximum pressures calculated with an equivalent Hertzian ellipse can be very different from the pressures that are found when the actual non-elliptic shape of the contact patch is taken into account. This is illustrated further in Figure 4, that shows the pressure distribution and contact patch shape at y = 1 mm computed with CONTACT. It must be said that in this test case they still result from a purely elastic calculation whilst the actual material behaviour is elastic-plastic, see, e.g., [1]. However, it is obvious that the Hertzian pressure will not predict the high stresses well that appear in the flange groove (y = 5 – 6 mm) or on the flange (y > 6 mm).

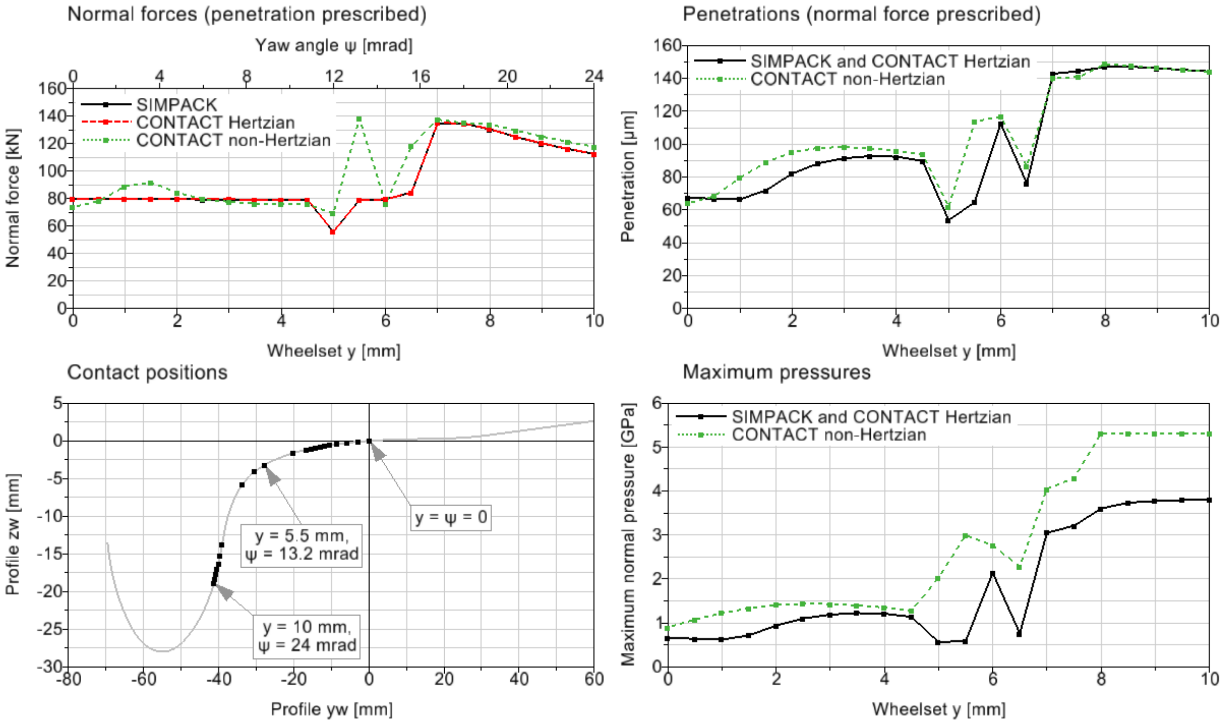

Figure 3 Top: comparison of normal forces and penetrations of the equivalent-elastic and CONTACT approaches. Bottom: contact positions and maximum pressures in the contact patch.

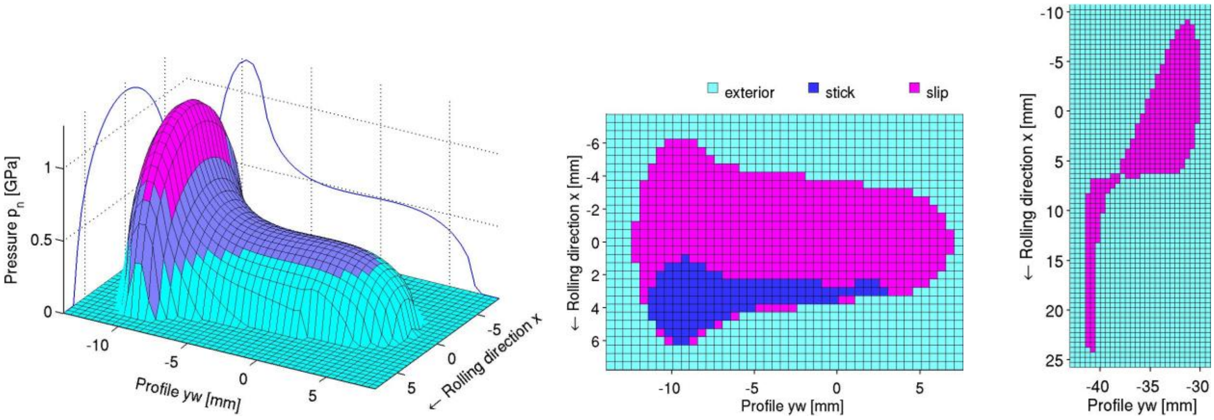

Figure 4 *Left: normal pressure distribution at y = 1 mm. Center: corresponding contact patch. Right: strongly non-elliptic contact patch, at y = 6.5 mm, ψ = 15.6 mrad.*

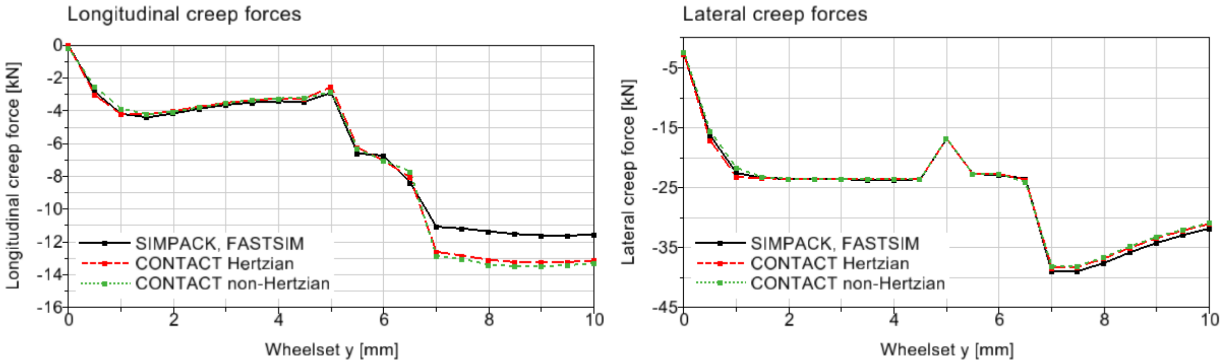

Figure 5 *Comparison of creep forces between SIMPACK and CONTACT.*

Figure 5 clearly shows that the tangential forces are predicted quite well even with an equivalent contact ellipse and using the FASTSIM approach. There is only a difference on the flange, where the spin creepage becomes very large. Here the longitudinal creep force does not coincide with the full solution of CONTACT anymore.

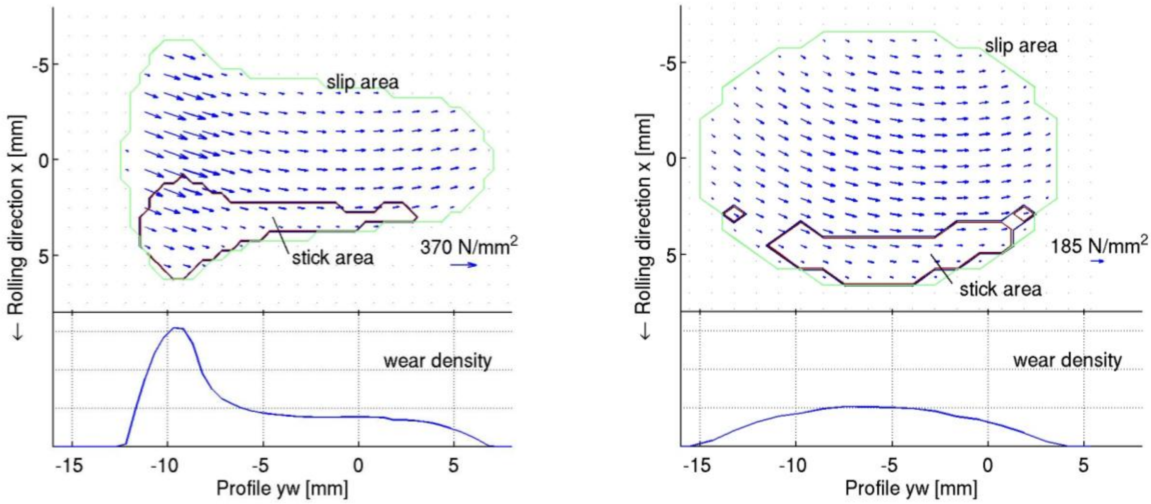

Figure 6 *Distribution of shear stresses in the contact patch for CONTACT (left) and equivalent-elastic/FASTSIM (right) at y = 1 mm, and corresponding wear density along the wheel/rail profile.*

Whereas the total creep forces computed by the original approach are generally sufficiently accurate, the distribution of these forces appears to be less adequate. Figure 6 shows the local surface tractions and the distribution of the stick and slip areas. The shape and location of slip and stick areas is quite consistent between the two approaches. Twice higher surface tractions occur in the CONTACT results compared to the original result, in the region where higher normal pressures are found compared to Hertz' theory (Figure 4). And this leads to a three times higher wear density, which is also much more shifted to one side of the contact patch.

5. CONCLUSIONS AND OUTLOOK

In this paper we presented the set-up and initial results for a new interface between the multi-body simulation software SIMPACK Rail and Kalker's wheel-rail contact software CONTACT. The main benefit is in the more accurate shear stress and micro-slip distributions in the contact area that together form the primary inputs to wear calculations. Further benefits of the interfacing reside in the ability to perform detailed contact analyses within the multi-body framework, and to investigate the importance of fully non-Hertzian calculations in more extreme situations such as derailment and limit cycle studies.

The initial results demonstrate that the interface works and delivers consistent results to the already existing contact theories in SIMPACK Rail. Deviations between the theories are found that are in line with earlier comparisons.

- In cases where strongly non-elliptic contact patches occur, the normal forces deviate up to 75%, e.g. in the flange groove. This changes the contact stiffness;
- The Hertzian approach leads to considerable under-estimation of the maximum pressure in the contact patch, e.g. 3.6 vs. 5.3 GPa, which has consequences for the study of RCF;
- The longitudinal and lateral creep forces of CONTACT and FASTSIM generally agree up to about 10%, except at larger lateral displacements of the wheelset where large spin creepage occurs. In that case the difference goes up to 20%;
- The distributions of shear traction and local slip velocities differ significantly between CONTACT and FASTSIM, with considerable differences in the wear distribution as a result.

Because of the more elaborate modelling strategy that CONTACT uses, e.g. based on non-elliptic, non-Hertzian geometry, we expect that the accuracy of the results is improved, but this needs to be validated.

Further work is needed to complete the interface: allow using CONTACT directly in the dynamic simulation too, and using the detailed CONTACT results for the computation of wheel and rail profile wear. A full version of the CONTACT add-on is expected to be completed by the end of 2011, including the use of the CONTACT results for wear calculations. After that, the authors are planning to extend the range of applicability of the CONTACT add-on in the coming years, namely to increase the resolution that can be used after a drastic speed-up of the CONTACT calculations, to improve the handling of conformal contact, and to extend the local friction laws used for the consideration of additional effects such as temperature, contamination or lubricants.

References

[1] K.L. Johnson. *Contact Mechanics.* Cambridge University Press, 1987.

[2] J.J. Kalker. Three-Dimensional Elastic Bodies in Rolling Contact. *Solid Mechanics and its Applications.* Kluwer Academic Publishers, 1990.

[3] N.N. Website of SIMPACK and SIMPACK Rail. www.simpack.com, 2011.

[4] J.P. Pascal. Multi-Hertzian method for simulating conformal wheel-rail pairs – application to S1002/UIC60 pair – proposal for a physical laboratory test for assessing the method. *Symposium of Advances in Contact Mechanics: a tribute to Prof. J.J. Kalker*, Delft, 2008.

[5] J. Piotrowski and W. Kik. A simplified model of wheel/rail contact mechanics for non-Hertzian problems and its application in rail vehicle dynamics simulations. *Vehicle System Dynamics*, 46(1-2):27–48, 2008.

[6] A.A. Shabana, K.E. Zaazaa, and H. Sugiyama. *Railroad Vehicle Dynamics: A Computational Approach.* CRC Press, Boca Raton, 2008.

[7] P. Shackleton and S.D. Iwnicki. Comparison of wheel-rail contact codes for railway vehicle simulation: an introduction to the Manchester Contact Benchmark and initial results. *Vehicle System Dynamics*, 46(1-2):129–149, 2008.

[8] E.A.H. Vollebregt, S.D. Iwnicki, G. Xie, and P. Shackleton. Assessing the accuracy of different simplified frictional rolling contact algorithms. *Vehicle System Dynamics*, to appear, also available as Memo EV/M10.035, VORtech, Delft, The Netherlands, 2010.

[9] P. Wriggers. *Computational Contact Mechanics*, 2nd ed. Springer, Heidelberg, 2006.