Adaptation of the semi-Hertzian method to wheel/rail contact in turnouts

M. Sebès & H. Chollet
Université Paris-Est, IFSTTAR, COSYS GRETTIA, Marne la Vallée, France

E. Monteiro
PIMM, Arts et Métiers ParisTech, CNRS, Paris, France

M. Laurent & C. Muller
Vossloh Cogifer, Technology Centre, Reichshoffen, France

ABSTRACT: A procedure is described in order to assess loads applied on a turnout due to track-train interaction. Co-simulation is used between a finite element method (FEM) model of the turnout and a multibody system (MBS) of the vehicle. Wheel/rail contact forces are computed in the MBS and applied to the rails of the turnout modelled as FEM beams. FEM displacements under the wheel are accounted in the MBS in the next time step. A modification has been applied to the semi-Hertzian (SH) method used to assess wheel/rail forces. This adapted SH method is designed to take in account the relative flexibility of the components of the turnout, like the stock rail and the switch rail. Such parts have their own degree of freedom and may in some extent behave independently: the proposed method takes it in account in the contact search. The co-simulation has been first used in a referenced case-study.

1 INTRODUCTION

Because of reliability issues and monitoring, turnouts are parts of the railway network which require a special attention. Therefore a precise assessment of wheel/rail forces in these devices is necessary. From a mechanical point of view, there are 2 main differences between an ordinary rail and a turnout: a) the variable geometry of the cross-sections, b) the variation of the track stiffness along the distance. Point a) is partly handled by adding a dimension to look-up tables describing wheel/rail contact: this method, or an alternate one avoiding pre-calculated tables, is implemented in multi-body-systems (MBS) of commercial packages, like SIMPACK, VAMPIRE (Sun et al. 2011), NUCARS (Shu et al. 2006) or GENSYS (Kassa & Nielsen 2008). Point b) requires a structural model of the track. As the design of a turnout is a complex one, the usual method consists in using the finite element method (FEM) in order to model the turnout. Wheel/rail forces act as a coupling term between MBS and FEM: They are the applied loads on the FEM, and by a feedback process, their value depends also of the vibration of the flexible track. Vehicle-track interaction in turnouts has been addressed by Alfi & Bruni (2009). It has also been implemented in the in-house code DIFF3D (Kassa 2004).

In MBS, normal wheel/rail contact is usually modelled with the Hertzian theory where the shape of the contact patch is elliptical. An alternate method allowing non-Hertzian contact has been introduced by Kik & Piotrowski (1996) and further developed by Ayasse & Chollet (2005). This so called semi-Hertzian method (SH) has been applied by Sebès et al. (2006) in the case of turnouts. In order to be coupled with FEM, it has been here necessary to slightly adapt it: this topic is addressed in section 3. This modified SH method has been implemented in the research MBS software VOCO. A co-simulation procedure between VOCO and the commercial FEM package ANSYS has been developed in order to handle dynamic track-train interaction. This topic will be addressed in section 4. Finally the procedure is benchmarked in section 5. The UIC60-760-1:15 turnout in Härad, measured in the INNOTRACK project (2009), will be studied in the diverged route at a facing-point. The track model is described in the next section.

2 TRACK MODEL

2.1 *Description of the FEM model*

A view of the ANSYS model of the UIC60-760-1:15 turnout is shown in Figure 1. Rails and sleepers are modeled with beam finite elements, here displayed with shapes determined from the section definition. 118 sections are used to define the whole turnout. They are imported from a computer-aided design (CAD) model of the turnout. There are 3 beam elements between 2 successive sleepers. Rail pads are modelled with 6 degrees-of-freedom (dof) spring-dampers (not shown in Figure 1). The ballast is modelled by a viscously damped Winkler foundation. An effective ballast contacting area is assigned under each rail above a sleeper. The value of this area depends of the location in the turnout. For instance, the area is smaller in the crossing zone, where the ballast is less compacted. Dofs are coupled at both ends of the diverging route. The total number of dofs is 56,000. The presented model doesn't include nonlinearities but could be nonlinear.

2.2 *Nodal Loads*

Wheel loads including moments (see section 3.2) are applied to the rails. For the special case of a two-point contact, with one contact on the stock rail and the other contact on the switch rail, both beams shown in Figure 1 will be loaded by a component of the total wheel load (see section 3.3).

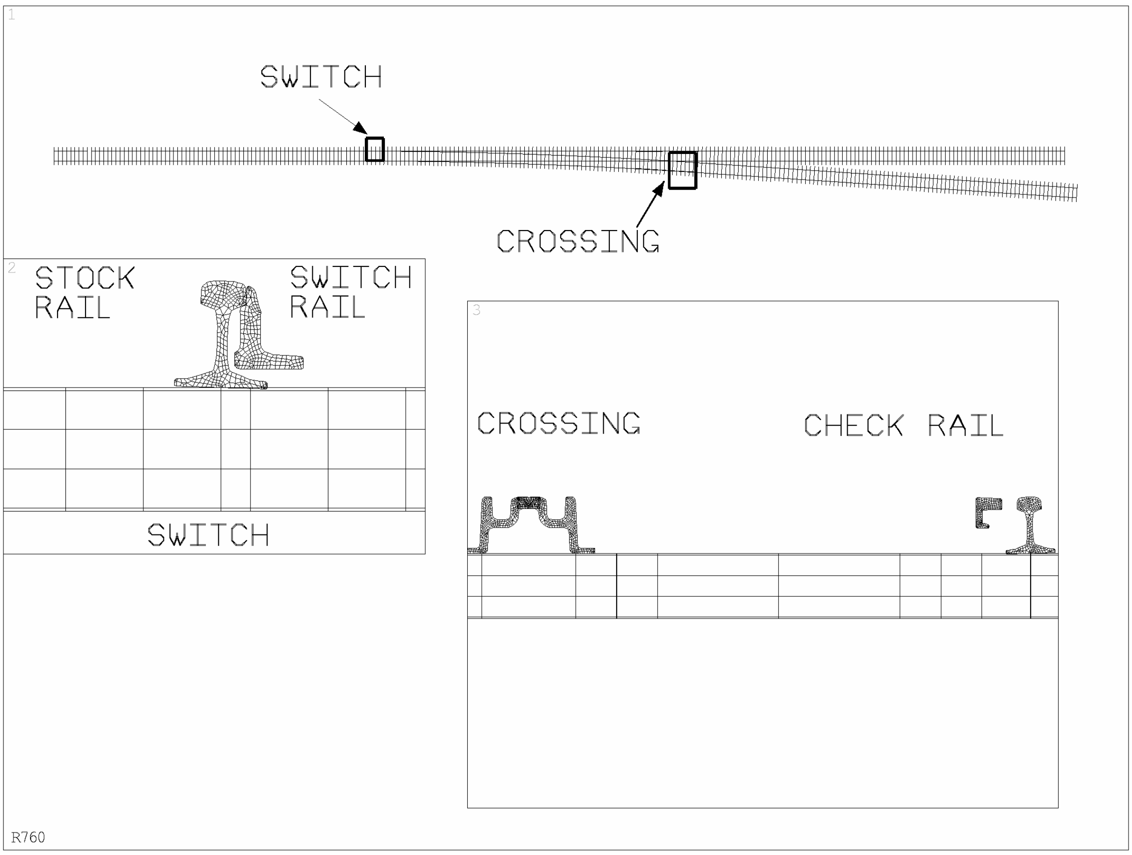

Figure 1. Model of the UIC60-760-1:15 turnout – Zoom on sections of the switch and the crossing

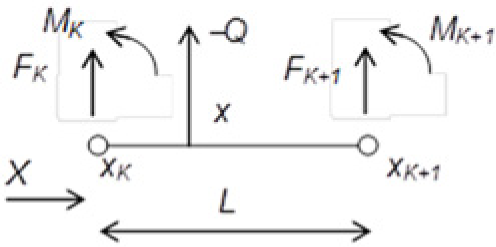

Figure 2. Distribution of the wheel vertical load Q to the FEM nodes

For each wheel, forces and moments are distributed to adjacent nodes around the position of the wheel. Hermitian cubic polynomials are chosen as they are continuously differentiable.

$$
\begin{aligned}
H_1 &= \frac{1}{4}(1-\xi)^2(2+\xi) \\
H_2 &= \frac{1}{8}(1-\xi)^2(1+\xi) \\
H_3 &= \frac{1}{4}(1+\xi)^2(2-\xi) \\
H_4 &= -\frac{1}{8}(1+\xi)^2(1-\xi)
\end{aligned}
\tag{1}
$$

ξ is the normalized coordinate:

$$
\xi = \frac{x - x_K}{L} + \frac{x - x_{K+1}}{L} \tag{2}
$$

where x = position of the wheel; x_k and x_{k+1} = positions of the adjacent nodes; and L = length of the element.

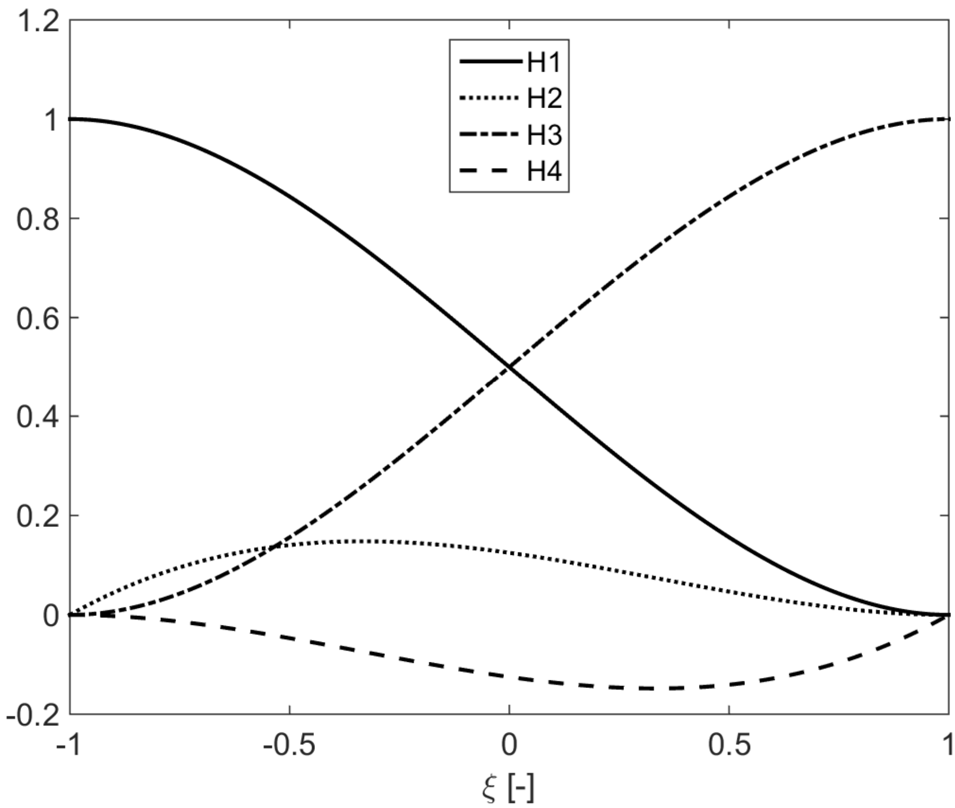

Figure 3. Hermitian cubic shape functions

For instance, the vertical wheel load Q is equivalent to 2 nodal loads $F_{z,k}$ and $F_{z,k+1}$ and 2 moments $M_{y,k}$ and $M_{y,k+1}$ (Figure 2).

$$\begin{aligned} F_{z,k} &= -H_1 \cdot Q \\ M_{y,k} &= -H_2 \cdot QL \\ F_{z,k+1} &= -H_3 \cdot Q \\ M_{y,k+1} &= -H_4 \cdot QL \end{aligned} \tag{3}$$

2.3 *Rail displacements under the wheel*

In the following sections, lateral and vertical displacements under the wheel are accounted in the MBS. Their value is also derived from the Hermitian shape functions. For instance, the vertical displacement u_z under the wheel has the following expression.

$$u_z = H_1 u_{z,k} + H_2 L\theta_{y,k} + H_3 u_{z,k+1} + H_4 L\theta_{y,k+1} \tag{4}$$

where $u_{z,k}$ and $u_{z,k+1}$ = vertical nodal displacements of the adjacent nodes; and $\theta_{y,k}$ and $\theta_{y,k+1}$ = nodal rotations around y.

In the special case of a two-point contact with one contact on the stock rail, and the other on the switch rail, there will be 2 vertical displacements, $u_{z,sw}$ and $u_{z,st}$ respectively associated to the switch rail and the stock rail.

3 WHEEL-RAIL CONTACT

3.1 *Short overview of the semi-Hertzian method*

Wheel-rail contact is handled with the semi-Hertzian (SH) method defined by Ayasse & Chollet (2005). It is based on a discretization of the wheel and rail profiles in strips and has been first introduced by Kik & Piotrowski (1996). The method is based on a virtual interpenetration of wheel and rail profiles. It enables to handle multiple contacts. It can also predict non-elliptical contact patches and has proved to give results very similar to FEM ones in non-Hertzian conditions (Quost et al. 2006). The SH method is implemented in the MBS research software VOCO, developed by IFSTTAR (formerly INRETS) and is about 3 times slower than the Hertzian methods usually used in MBS (Chollet et al. 2013).

As far as turnouts are concerned, the SH method has been applied by Sebès et al. (2006). In VOCO, contact parameters are preprocessed in look-up tables. In an ordinary rail, these tables are only function of one variable: the lateral position of the wheel relatively to the rail, t_Y. In turnouts, another variable is added: the longitudinal position of the wheel. In turnouts, at every time step, contact parameters are interpolated values of a function of these 2 variables. An important contact parameter is the relative vertical distance between the wheel and the rail.

3.2 *Derivation of moments from wheel/rail loads*

As stated in section 2.2, forces and moments are applied on the FEM model. In order to derive these latter ones from wheel/rail forces, it is necessary to assess the offsets of the contact location with respect to the nodal coordinates, Δy and Δz (Figure 4). Offset values depend on the location of the contact zone. They are expressed as a weighted sum of the strip coordinates.

$$\Delta y = \frac{\sum_i N_i \Delta y_i}{\sum_i N_i} \qquad \Delta z = \frac{\sum_i N_i \Delta z_i}{\sum_i N_i} \tag{5}$$

where Δy_i and Δz_i = coordinate of the strips with respect to the node location; and N_i = normal force on strip # i.

For instance, the moment of torsion M_x applied on the rail is given by the following expression.

$$M_x = -\Delta y \cdot Q + \Delta z \cdot Y \tag{6}$$

where Y = lateral force applied on the wheel; and Q = vertical force applied on the wheel.

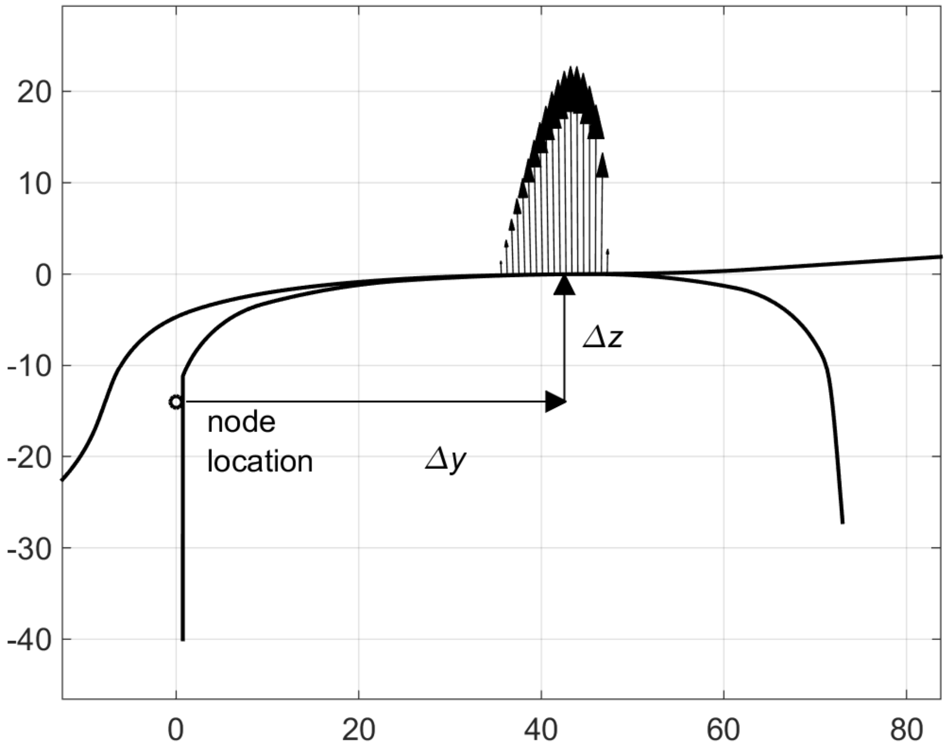

Figure 4. Offset between node coordinates and wheel/rail contact location

3.3 *Semi-Hertzian model of the UIC60-760-1:15 turnout*

Figure 5 shows the SH model of the UIC60-760-1:15 turnout identical to the one studied in the INNOTRACK project (2009). Each section is divided in 600 strips. 154 sections are required in order to model the whole turnout. In the switch zone, the spacing between sections is 300 mm. In the crossing zone, it is 50 mm. Theoretical profiles are imported from a CAD model of the turnout. Only the railhead is required.

Each component of the turnout is associated to a subset of the strips: e.g. strips # 301-450 are linked to the switch rail; strips # 451-600 are linked to the stock rail. This subdivision of the profiles in several components is an automated task performed during the preprocessing of the look-up tables.

In the special case of a two-point contact with one contact point on the stock rail and the other on the switch rail, it is necessary to apply specific forces on the finite elements associated to these components (Figure 1). With this partition, this can be done readily: e.g. forces on the switch rail are the sum of the forces associated to strips # 301-450. Moments are also deduced by considering only a subset of the strips in Equation 5.

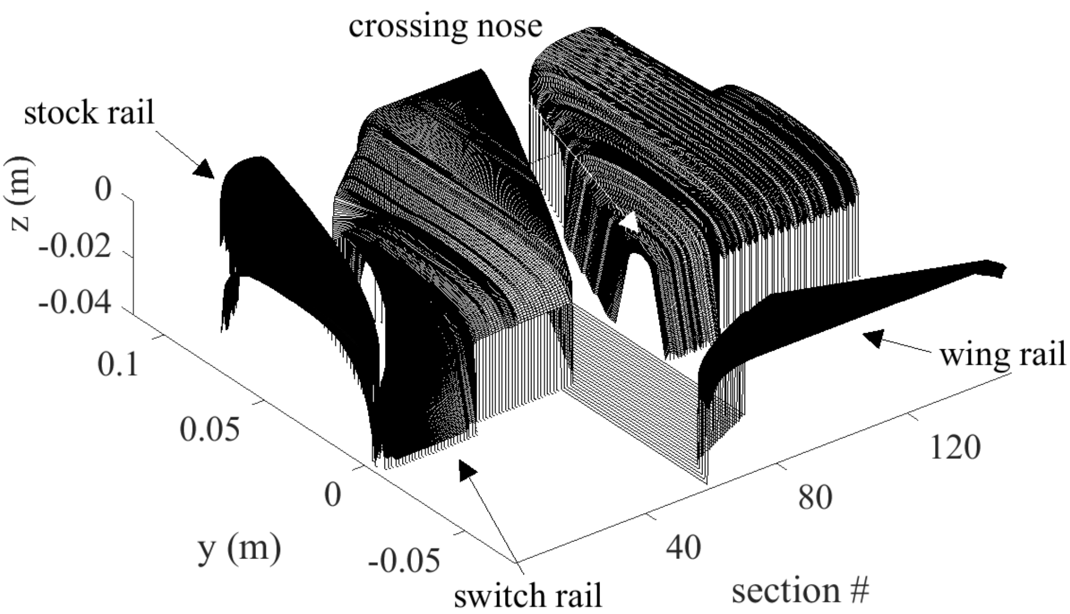

Figure 5. Semi-Hertzian model of the UIC60-760-1:15 turnout

3.4 *Purpose of the adapted SH method*

The suggested adaptation of the SH method is intended to better model contact in parts of turnouts with flexible components like a switch rail (Pålsson 2013), or a moveable frog (Pouligny et al. 2008). In such a device, a single case of simulation cannot pretend to predict a mean behavior. Most of the MBS packages find an instantaneous jump between say the stock rail and the switch rail, while a simple observation in situ shows a large zone of two-point contact. To overcome this discrepancy, Pålsson (2013) and Pouligny et al. (2008) take in account various input parameters in order to get a load collective which leads to a more realistic rolling contact zone. In both studies, it is found that the key parameter is the variability of the wheel profiles. The aim of the presented model is to take in account the relative flexibility of the components, instead of considering the turnout profile as a one-piece device. This is nothing more than trying to keep a self-consistency with the track model where the stock rail and the switch rail have their own dof. As a result it is expected to find a larger two-point contact zone in a single simulation. However it seems still necessary to consider a load collective. The proposed method is intended to supply more realistic loads, which could be helpful in the phase of design of turnouts. This model may also be useful in order to study derailment cases in switches, which may be caused by the opening of the switch blade (Ayasse et al. 2002).

3.5 *Adaptation of the SH method in a two-point contact configuration in turnouts*

On the top of Figure 6, the rail is considered as a one-piece section as it is usually assumed in MBS. A single contact is detected. The same configuration is shown on the bottom of Figure 6, the one-piece configuration being in dashed style. The profile shown in solid style is deduced from the dashed one by applying a different offset to each component, this offset being the vertical displacement given par Equation 4. In this configuration, the vertical displacement, $u_{z,st}$, is bigger than $u_{z,sw}$. A two-point contact is thereby detected. As a result the transition zone between the stock rail and the switch rail will be larger.

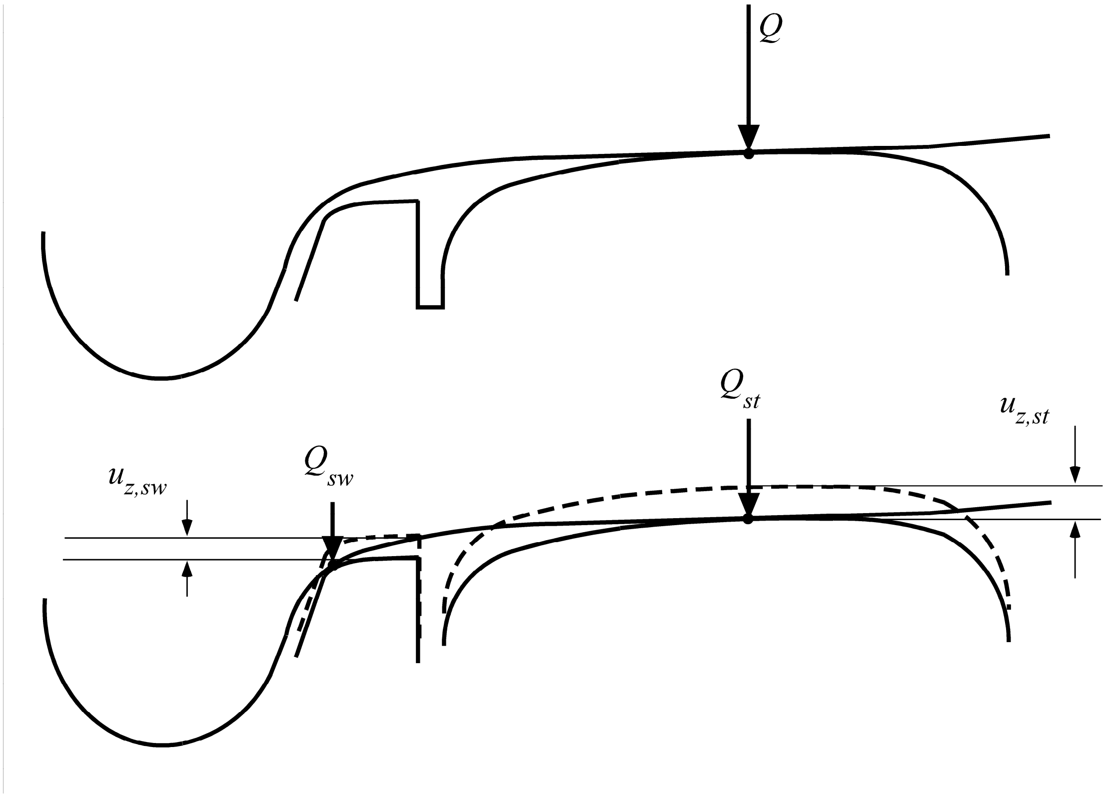

Figure 6. Configuration with a one- piece profile (top) – Proposed method (bottom)

As stated in section 3.1, wheel/rail vertical distance is interpolated from look-up tables. In the proposed method, the modification consists in adding an offset to the distances associated to each strip of a component, this offset being the vertical displacement of the component. It is also necessary to predefine a vertical wheel lift for each component.

A similar approach may be used in the lateral direction (Figure 7). The modification consists here in assigning to each component its own t_Y (see the definition of this variable in section 3.1). Contact parameters of a given component are then interpolated at the t_Y of this component.

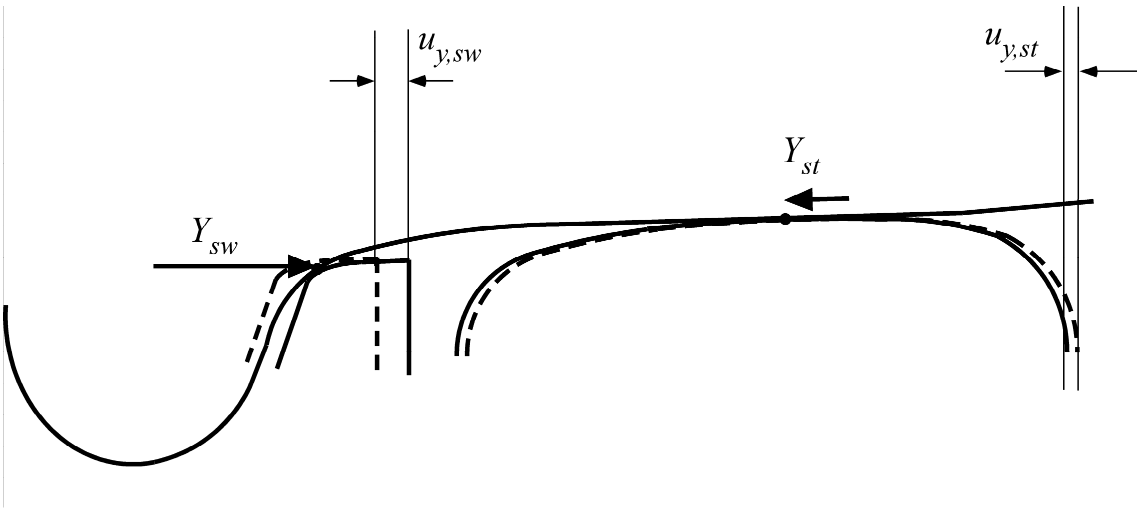

Figure 7. Two-point contact configuration – Proposed method in the lateral direction

4 COUPLING OF FEM WITH MBS

Figure 8 shows a flowchart of the co-simulation. At a given time step, MBS computes the wheel/rail forces and the associated moments (see section 3.2). They are applied to the FEM model and a transient analysis is performed for one time step. Afterwards displacements under the wheel are derived according to Equation 4. These displacements are taken in account in the next time step of the MBS solution process in two distinct ways: a) in the contact search as described in the previous section, b) as an added irregularity to the track. The added excitations are the displacements under both wheels in lateral and vertical directions. Each component (stock rail, switch rail…) may have a different excitation.

From a practical point of view, co-simulation is implemented by linking VOCO as an external command (ANSYS 2012).

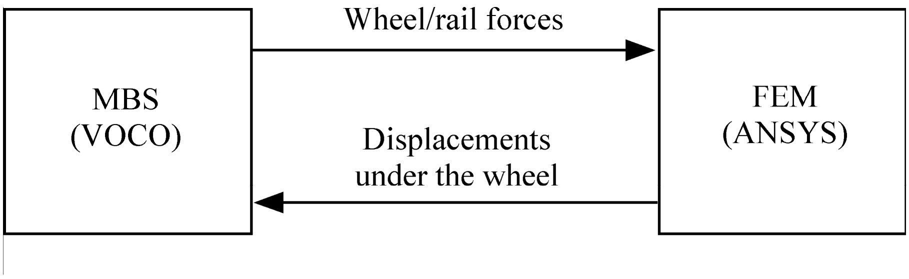

Figure 8. Flowchart of co-simulation

5 CASE-STUDY

5.1 *Summary of running conditions*

The UIC60-760-1:15 turnout in Härad, measured in the INNOTRACK project (2009), will be studied in the diverged route at a facing-point. The running conditions are intended to be identical to the ones described by Kassa & Nielsen (2008). A similar model of vehicle is the loaded freight wagon Sgns with Y25 bogies, built during the DynoTRAIN project (Polach & Böttcher 2014). The axle load is 25 tons. Wheel profile is S1002. Rail profiles are theoretical ones which is a major difference with Kassa & Nielsen (2008), as they had measured profiles at their disposal. Friction coefficient is 0.3. The track is a curve without cant, of radius 760 m with a short transition of 0.4 m. Vehicle speed is 80 km/h. Contact between the back of the wheel and the check rail is accounted by an equivalent spring-damper.

5.2 *Results without co-simulation*

Results without co-simulation are shown on the top of Figure 9. Due to the poor quality of the copy, measures taken from Kassa & Nielsen (2008) are not here reproduced. However comparison of simulation and measurement shows they are in pretty good agreement. Vertical Q force and lateral Y force are both taken on the outer wheel of the first axle. Flange contact on the switch begins at a distance of 3.7 m from the front of the turnout. The effect of the check rail is visible on the lateral force between 44 and 48 m. A peak at the crossing nose is visible on the vertical force at 47 m. It seems a bit underestimated.

5.3 *Results with co-simulation*

Results with co-simulation are shown on the bottom of Figure 9. Although they look globally the same as previous results, there are some differences: a) the sleeper passing frequency is visible on the vertical force with a standard deviation similar to measures b) the peak at the crossing nose is higher, but is a bit overestimated with respect to the measures.

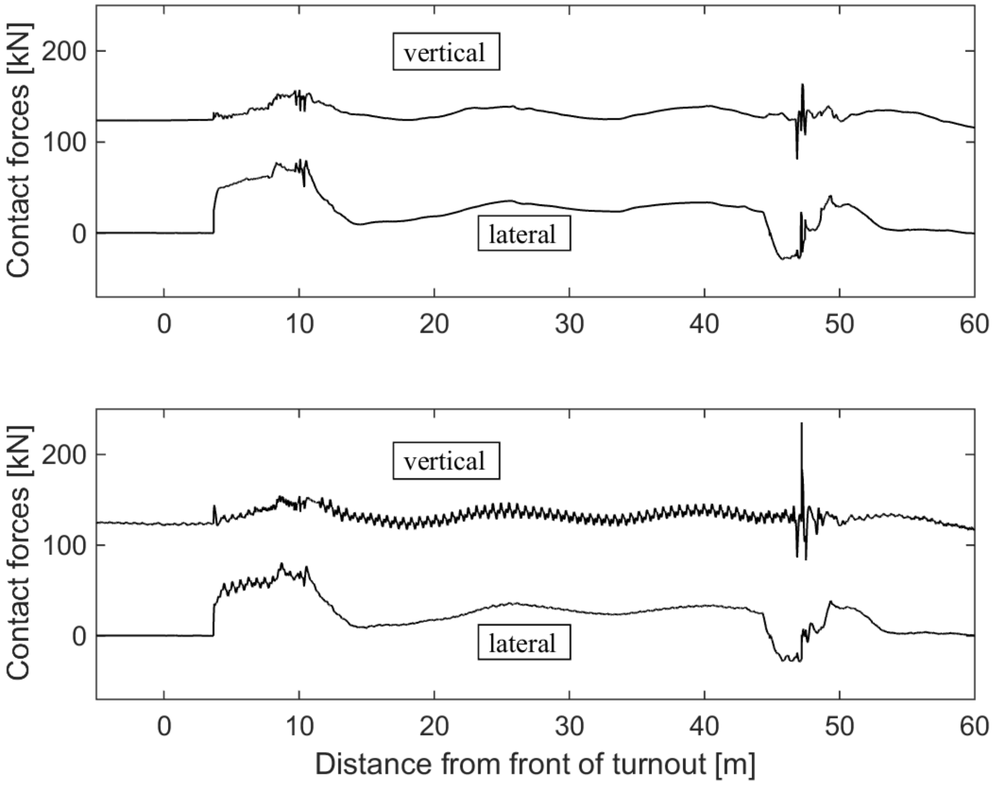

Figure 9. Wheel/rail forces on the outer wheel of first axle: VOCO (top), VOCO+ANSYS (bottom)

6 CONCLUSION

Design optimization of turnouts should take in account the flexibility of the track. This problem has already been addressed in the case of turnouts (Alfi & Bruni 2009, Kassa 2004). The proposed method aims to address vehicle-track interaction through coupling of a multibody system and a finite element model. The procedure of co-simulation is described and demonstrated in a referenced case-study (Kassa & Nielsen 2008). To the best knowledge of the authors, the presented procedure is a first attempt with a non-Hertzian method. The so-called semi-Hertzian method has been adapted in order to predict more realistic transition zones in flexible zones of turnouts. This aspect still needs to be validated by further analysis and suitable measurements.

ACKNOWLEDGMENT

This research was partly funded and supported by the CERVIFER project.

REFERENCES

Alfi S. & Bruni S. 2009. Mathematical modelling of train-turnout interaction. *Vehicle System Dynamics*, Vol. 47, No. 5, 551-574.

ANSYS 2012. Programmer's reference for the Mechanical APDL and Mechanical Applications, Ansys Release 14.5

Ayasse J.B., Chollet H., Fleuret J.F. & Levêque E. 2002. CAF, a generalised conicity criteria for the wheel rail contact – example of a switch blade safety study, in: *Proceedings of the 8th Mini Conference on vehicle system dynamics, identification and anomalies*, Budapest

Ayasse J.B. & Chollet H. 2005. Determination of the wheel–rail contact patch in semi-Hertzian conditions. *Vehicle System Dynamics* 43(3) , pp. 161–172.

Chollet H., Sebès M., Maupu J.L. & Ayasse J.B 2013. The VOCO multi-body software in the context of real-time simulation. *Vehicle System Dynamics*, Vol. 51, No 4, 570-580.

INNOTRACK 2009. D3.1.4 Summary of results from simulation and optimization of switches

Kassa E. 2004. Simulation of dynamic interaction between train and turnout, *Thesis for the degree of licentiate of engineering*, Göteborg, Chalmers university of technology.

Kassa E. & Nielsen J.C.O. 2008. Dynamic interaction between train and railway turnout: full-scale field test and validation of simulation models, *Vehicle System Dynamics* Vol. 46, Supplement, 521-534.

Kik W.& Piotrowski J. 1996. A fast, approximate method to calculate normal load at contact between wheel and rail and creep forces during rolling, in: *Proceedings of the 2nd Mini Conference on Contact Mechanics and Wear of Rail/Wheel Systems*, Budapest

Pålsson B. A. 2013. Design optimisation of switch rails in railway turnouts, *Vehicle System Dynamics*, Vol. 51, No. 10, 1619-1639

Polach O. & Böttcher A. 2014. A new approach to define criteria for rail vehicle model validation. *Vehicle System Dynamics*, Vol. 52, Supplement 1 – Special Issue: IAVSD Proceeding Supplement, pp. 125-141

Pouligny P, Nierengarten S., Plu, J., Grunchec J.Y., Pauchet J.M., Sebès M., Boulanger D., Mugg P., Winiar L. & Schöch W. 2008. Investigation and field results for high speed moving crossings, *Proceedings of the 8th World Congress on Rail Research*. Seoul

Quost X., Sebès M., Eddhahak A., Ayasse J.B., Chollet H., Gautier P.E. & Thouverez F. 2006. Assessment of a semi-Hertzian method for determination of wheel-rail contact patch. *Vehicle System Dynamics*, 44: 10, 789 — 814.

Sebès M., Ayasse J. B., Chollet H., Pouligny P. & Pirat B 2006. Application of a semi-Hertzian method to the simulation of vehicles in high-speed switches*, Vehicle System Dynamics*, Vol 44, Supplement, 341-348

Shu X., Wilson N., Sasaoka C. & Elkins J. 2006. Development of a real-time wheel/rail contact model in NUCARS® and application to diamond crossing and turnout design simulations, *Vehicle System Dynamics*, Vol. 44, Supplement, 251–260

Sun Y.Q., Cole C. & Boyd P. 2011. A numerical method using VAMPIRE modelling for prediction of turnout curve wheel–rail wear, *Wear* 271 482–491.