University of Pardubice
Faculty of Transport Engineering

EVALUATING METHODOLOGY FOR DYNAMIC EFFECTS OF RAIL VEHICLE ON TRACK

DISSERTATION THESIS - ANNOTATION

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Contents

1. State of the Art

The evaluation of the dynamic effects of a rail vehicle on a track is a common part of standards ensuring running safety, methodologies for evaluation of running comfort or damaging effects caused by forces acting in the wheel-rail contact. Some methodologies are also used by railway infrastructure managers to determine the damaging effects of running vehicle on a track. Then they can compare different types of vehicles depending on track damage and set the appropriate amount of fees for using the track.

From point of view of the damaging effects of a rail vehicle on a track, two basic and most important damage mechanisms are considered: rolling contact fatigue (*RCF*) and wear of rails and wheels ([1], [2]). The wear of wheels and rails can be modelled and predicted using a method based on dissipated energy in wheel-rail contacts. This method defines and uses the Wear Number $T\gamma$ (Eq. 1.1, where T is creep force, γ is creepage) that represents the specific work of the creep forces.

$$T\gamma = |T_x\gamma_x| + |T_y\gamma_y| + |M_z\varphi_z| \qquad [\mathrm{Nm/m}] \tag{1.1}$$

In some works, the Wear Number can be calculated by an equation without spin φ_z and spin moment M_z (see Eq. 1.2). Spin is usually neglected or already taken into account in the calculation of creep forces.

$$T\gamma = |T_x\gamma_x| + |T_y\gamma_y| \qquad [\mathrm{Nm/m}] \tag{1.2}$$

This method assumes wheel and rail wear is a function of the Wear Number, but does not predict a specific amount of material wear.

RCF damage prediction can be also provide by Wear Number values. Models for the prediction are defined based on laboratory measurements and the dependence of *RCF* on the Wear Number varies according to material properties (see Fig. 1.1). So the dissipated energy method allows to deal with both mechanisms of damage.

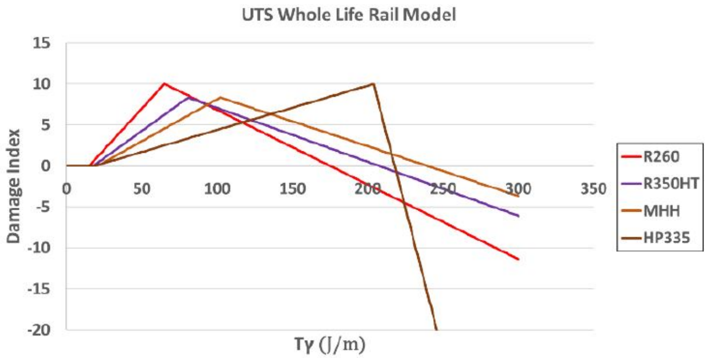


Figure 1.1: *RCF* damage index function depending on the Wear Number for different rail materials. [3]

There are other methods for predicting wheel and rail wear and *RCF*. The Archard's model is widely used. It directly predicts the loss of material volume. The Shakedown theory is another option for evaluating the possibility of *RCF* formation, but the theory does not take wear into account.

Sophisticated methodologies for evaluating the influence of rail vehicle on the track that take rail damage into account are used by Swiss [4] and British [5] railway infrastructure managers. This motivates transport operators to use so-called "track-friendly" vehicles and create a more sustainable railway system. Both mentioned infrastructure managers use the Wear Number as a parameter to evaluate the damaging effect of rail vehicle in a curve. Since it is currently not possible to obtain the values of the Wear Number by measuring on real vehicles (due to its dependence on creepages), it is necessary to perform a simulation calculation of vehicle running to evaluate these methodologies.

The Wear Number is also mentioned in standard EN 14363 Annex K [6]. The standard states that this parameter shows good agreement with rail damage and is suitable for its evaluation. Based on the Wear Number values, the Rail surface damage parameter T_{qst} is defined.

2. Aims of the Dissertation

The aims of the dissertation are set as follows:

- Analysis of the influence of wheel-rail contact modelling on the dynamic effects of a rail vehicle when passing through a curve, focusing on different solutions of the tangential problem.
- Defining the parameters of the rail vehicle that have a major influence on the dynamic effects when passing through a curve based on sensitivity analysis.
- Proposal of a methodology for evaluating the dynamic effects of a rail vehicle when passing through a curve and its comparison with existing methodologies.
- Assessment of the contribution of technical solutions to reduce the damaging effects of a rail vehicle on a track when passing through a curve.

Based on the previous chapter, the Wear Number is a widely used parameter for evaluating the damaging effects of a vehicle on a track. The evaluation of the Wear Number values is dependent on multi-body simulation of vehicle running that requires the creation of a detailed virtual mathematical model of the rail vehicle. The motivation is to find an equivalent parameter whose values (under the given conditions) correspond to the values of the Wear Number. The parameter can also be used for early, basic prediction of the damaging effects of a rail vehicle. The aim is not to calculate the specific worn volume of rail material, but to be able to compare different vehicle designs.

It is very difficult or impossible to change design parameters on a real rail vehicle, therefore, for the purpose of sensitivity analysis, simulation calculations are used, where the virtual vehicle model can be easily changed.

Running through a curve is considered a quasi-static phenomenon, but it is also commonly mentioned in publications dealing with vehicle dynamics [7]. In this thesis, the damaging effects of the vehicle on the track are evaluated using quasi-static values of quantities. In the field of rail vehicles, quasi-static values are commonly included in the description of vehicle running dynamics.

3. Methods of Solution

The method that was used in the solution of this work is multi-body simulations (*MBS*) of rail vehicle running. This method is based on the creation of a virtual vehicle model described by a system of differential equations that are solved in the time-domain. The reference rail vehicle model, which represents an electric multiple unit passenger car, was defined and the influence of selected parameters was analysed. For this part of the thesis solution, commercially available and commonly used SIMPACK software was used (see Fig. 3.1 on the left). The software also has various options for setting the wheel-rail contact modelling.

The validity of the reference vehicle model was verified using the analysis of natural frequencies. These eigenvalues correspond to the eigenvalues of the regular passenger car defined for the Manchester Benchmark (table 17.1 in [8]). Furthermore, a specific *MBS* of vehicle running was carried out to evaluate safety against derailment using method 1 according to the EN 14363 standard.

Support programs were created to solve the sensitivity analyses, which enable the setting of simulation calculation parameters and the automatic launch of simulations with new settings.

The analysis of the output data from the *MBS* was processed using the MATLAB program (see Fig. 3.1 on the right). The data were converted into a suitable format and loaded into the author's own program for sensitivity analysis evaluation. An important part of this program is the curve fitting tool which apply the least squares method to find the most appropriate functional dependence for the obtained data.

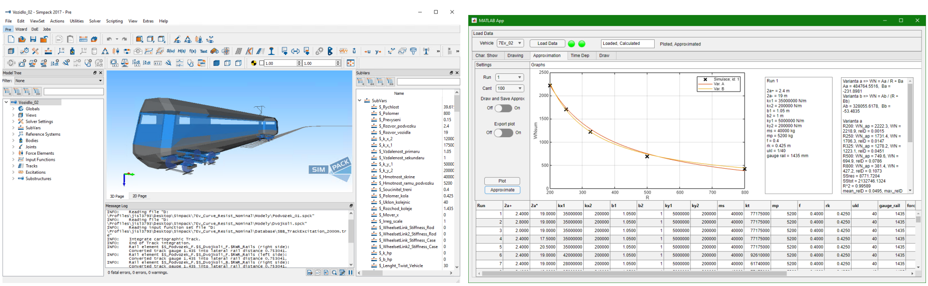


Figure 3.1: SIMPACK model (left) and MATLAB app (right).

4. Achieved Results

4.1 Relationship of Wear Number and Curve Resistance

MBS allow to simulate vehicle running under the condition of the action of only the curve resistance and without other resistances. Then, in order to maintain a constant running speed of the vehicle, it is necessary to apply such an external traction force F that balances the curve resistance R_c. The vehicle's equation of motion for this case is:

$$m_{red}\ddot{x} = F - R_c. \tag{4.1}$$

When the running speed is constant, the acceleration is zero, the equation becomes:

$$0 = F - R_c \quad => \quad F = R_c. \tag{4.2}$$

The point is that the curve resistance and the required traction force is equal to the specific work of the creep forces in all wheel-rail contacts, i.e. the sum of the Wear Number $\Sigma T\gamma$. This is proven by the simulation results (see Fig. 4.1). Then this equation holds:

$$F = R_c = \Sigma T\gamma. \tag{4.3}$$

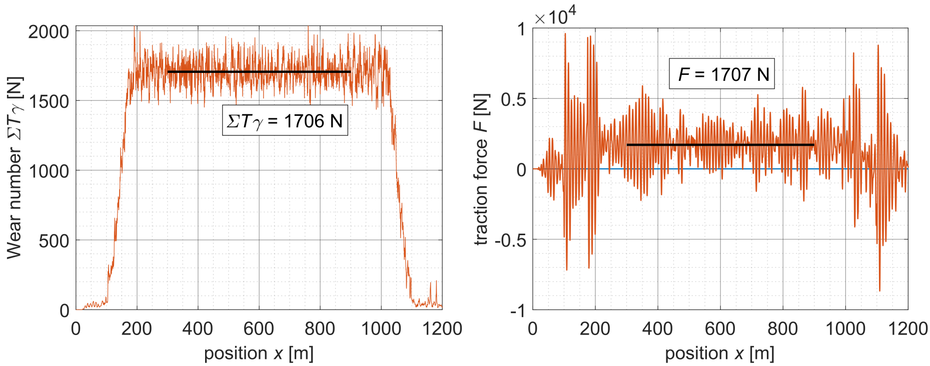


Figure 4.1: Values of the sum of the Wear Number in all wheel-rail contact (left) and values of the external traction force maintaining a constant running speed (right).

The Wear Number is widely used for wheel-rail contact damaging effects evaluation. This results show that the damaging effects are directly connected to the curve resistance. The added energy of the train needed to pass through the curve is used to damage the wheels and rails.

4.2 Influence of Tangential Problem Solving Methods

The wheel-rail contact solution can be divided into 3 parts: description of the relative movement of the wheel to the rail, a normal problem dealing with the formation of a contact area, a tangential problem that defines tangential (creep) forces arising from creepages and friction conditions. These parts must be solved in the order mentioned. Various theories and methods for their computational solution are created (see [8]). This chapter shows examples of how the results obtained may differ when different methods are used.

4.2.1 Equivalent and Discrete Elastic Contact Solutions

In SIMPACK, there are two contact area solution models.

The equivalent elastic solution uses an equivalent elliptical contact area defined based on the intersection of two linearly elastic bodies. This model allows the use of multiple methods of solving the tangential problem.

The discrete elastic solution works with more precise shape of the contact area (it is not replaced by an ellipse). The contact area is divided into several longitudinal strips. The normal and tangential problems are solved in each strip separately. In SIMPACK, only solution using the FASTSIM algorithm (tangential problem solution) can be used. Then this algorithm divides each strip into small elements (see Fig. 4.2).

In both solutions, Hertz's theory is used for the normal problem.

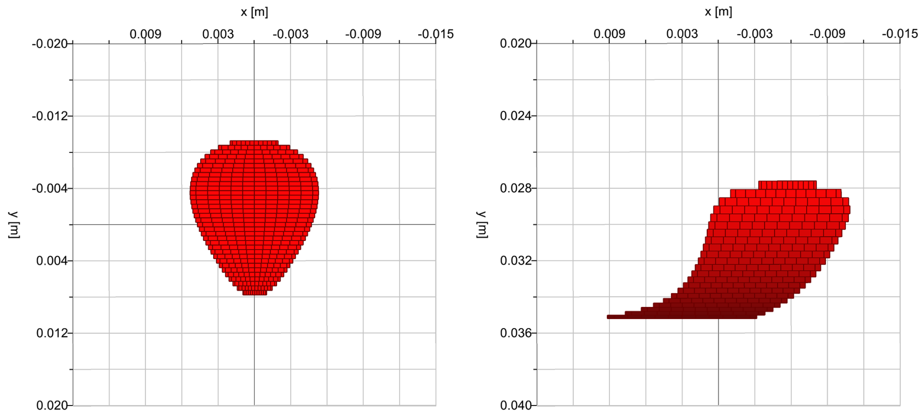


Figure 4.2: Contact area solved by the discrete elastic solution - in a straight track (left) and in a small curve radius (right).

When comparing the mentioned models, the biggest differences between the values of the Wear Number occur on the leading wheels (the outer wheel of the first wheelset). The differences increase with decreasing curve radius (see Fig. 4.3 on the left). The same effect can also be observed in the dependency for the sum of the Wear Numbers for the entire vehicle $\Sigma T\gamma$ (see Fig. 4.3 on the right).

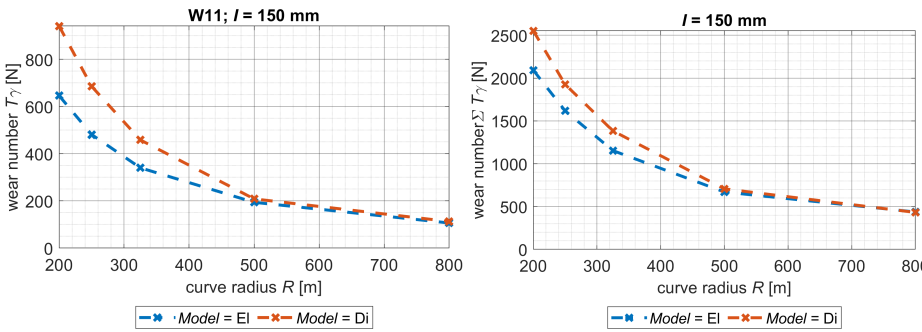


Figure 4.3: Comparison of Wear Number values for equivalent and discrete elastic models of the contact area solution - values on the leading wheel (left) and the sum for the entire vehicle (right).

4.2.2 Tangential Problem Solutions

The FASTSIM algorithm [9] is probably the most widely used method for solving the tangential problem in multi-body simulations. This method is based on the so-called Simplified Theory created by Kalker. It divides the contact area into small elements in which creep forces are calculated. Global quantities are obtained by summing over the contact area.

Another possible method of solving the tangential problem is Polach's method [10]. This simplified method is derived in order to reduce the computation time. Equations for calculating creepage forces are defined, but only as global quantities.

A comparison of Wear Number values for both mentioned method is shown in Fig. 4.4, where the equivalent elastic contact solution and Hertz theory are used. The largest differences occur on the leading wheel and increase with decreasing curve radius.

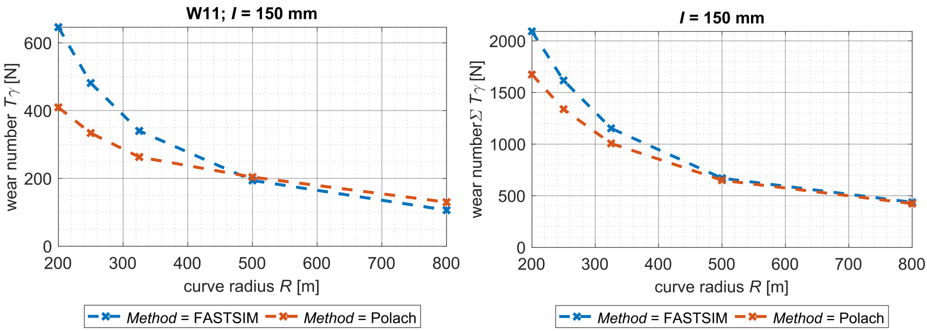


Figure 4.4: Comparison of Wear Number values for FASTSIM and Polach's methods of solving the tangential problem - values on the leading wheel (left) and of the sum for the entire vehicle (right).

4.2.3 Summary of this Section

This section does not aim to show which solution is more accurate. The choice of models and methods of solving the wheel-rail contact is very important when evaluating Wear Number values. Any methodology that works with the Wear Number should define a way to solve the wheel-rail contact for comparability of results. For further evaluation in this work, the equivalent elastic contact model and the solution of the tangential problem using FASTSIM were chosen.

4.3 Sensitivity Analysis

This section presents the influence of selected parameters on Wear Number values. All results shown are obtained under conditions of 100 mm cant deficiency.

4.3.1 Curve Radius

When analyzing the dynamic and damaging effects of a vehicle on the track passing though a curve, the curve radius R is probably the most important parameter. According to the usual formulas expressing the dependence of the curve resistance on the curve radius, a hyperbolic dependence can also be expected for Wear Number values (see Fig. 4.5).

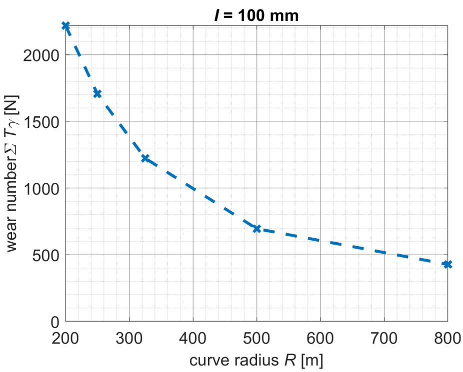


Figure 4.5: Dependence of the sum of the Wear Number on the curve radius.

The increase in Wear Number values for very small curve radii is mainly caused by lateral creepage on the leading wheel, which is related to the increase in the angle of attack of the first wheelset with decreasing curve radius.

4.3.2 Wheelset Distance

The wheelset distance $2a_1$ is the longitudinal distance between axles (wheelsets) in one bogie. It is known that bogies with a shorter wheelset distance are more suitable for tracks with small curve radii. In the same way, as the wheelset distance decreases, the sum of the Wear Number values also decrease (see Fig. 4.6 on the right) with a linear dependence.

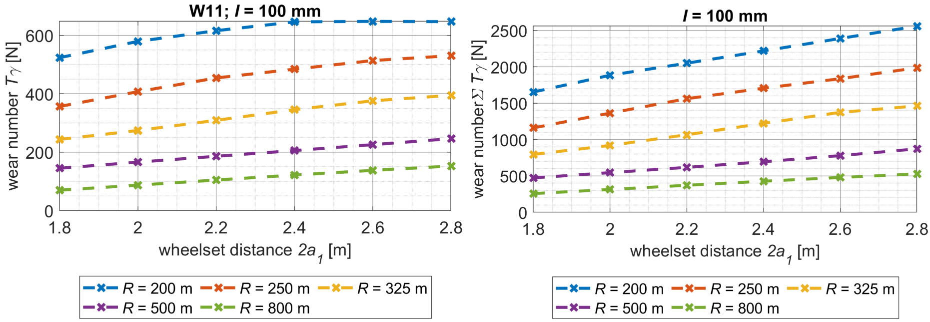


Figure 4.6: Dependence of Wear Number values of the leading wheel *W11* (left) and the sum of the Wear Number (right) on the wheelset distance for individual curve radii.

The dependencies of the Wear Number on the leading wheel on the wheelset distance for small curve radii do not show this linearity (see Fig. 4.6 on the left). In this case, the Wear Number values of the inner wheel of the second wheelset are increased. This is due to the diagonal position of the bogie.

4.3.3 Primary Yaw Stiffness

Another very important parameter is the primary yaw stiffness k_{t1}, which is defined using the longitudinal primary stiffness k_{x1} and the lateral distance of the primary suspension b_1 (see Eq. 4.4). These parameters can be modified to affect the primary yaw stiffness. As the value of this stiffness increases, the Wear Number values also increase (see Fig. 4.7 on the right). The shape of the dependence is very similar to the natural logarithm, which means that for very small stiffness the Wear Number values decreases rapidly. The reduction in the stiffness is manifested by a change in the longitudinal position of the contact area and a change in the angle of attack on the leading wheel *W11* (see Fig. 4.7 on the left, which leads to a decrease in the Wear Number.

$$k_{t1} = 2k_{x1}b_1^2. \tag{4.4}$$

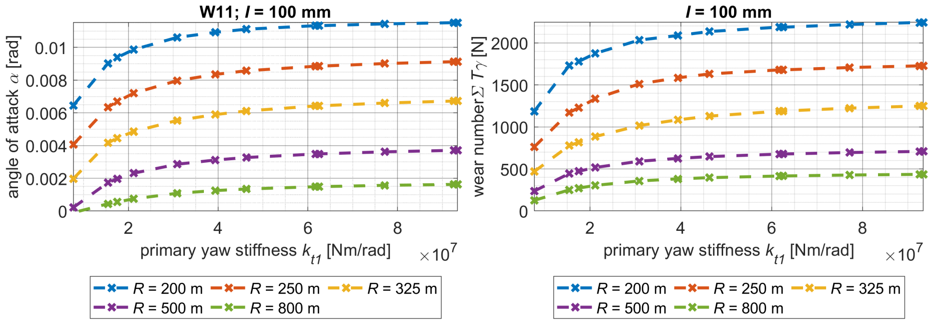


Figure 4.7: Dependence of the angle of attack on the leading wheel $W11$ (left) and the sum of the Wear Number (right) on the primary yaw stiffness for individual curve radii.

For most vehicle designs, it is possible to deal with the values of longitudinal primary stiffness, but its significant reduction leads to unstable running at high speed and a straight track. Another solution can be a bogie design with an internal frame, where the lateral distance of the primary springs and the weight of the bogie frame are significantly reduced.

4.3.4 Bogie Distance and Secondary Yaw Stiffness

The effects of bogie distance $2a_2$ and secondary yaw stiffness k_{t2} on the sum of the Wear Number for entire vehicle is similar and very small (see Fig. 4.8).

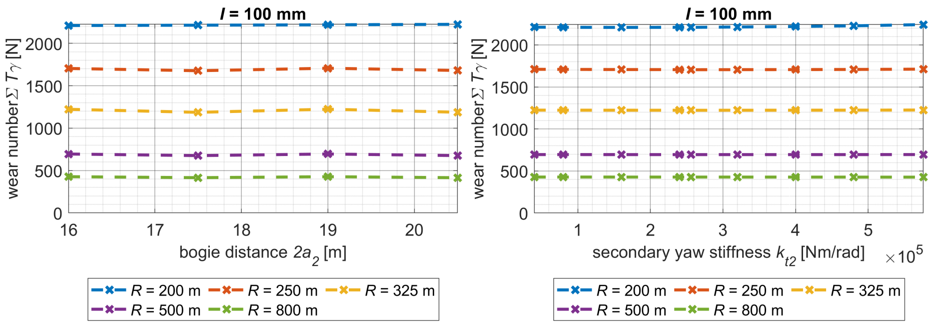


Figure 4.8: Dependence of the sum of the Wear Number on the bogie distance (left) and on the secondary yaw stiffness (right) for individual curve radii.

For both of these parameters, when their value is increased, the Wear Number on the front bogie also increases and conversely, decreases on the rear bogie. A longer bogie distance leads to a greater rotation angle of the bogie and a higher stiffness value results in a greater torque required for this rotation. This is positive for the rear bogie because the torque acts in the direction of the curve and the bogie takes a more suitable position from point of view of creep effects.

If the secondary yaw stiffness were increased further, there would be an increase in the Wear Number on the front bogie due to achieving a diagonal position of the bogie.

4.3.5 Vehicle Weight

Dependence of the sum of the Wear Number on the weight of vehicle m_v has a linear character for all curve radii (see Fig. 4.9 on the left). The effect of increased weight is mainly manifested in the values of creep forces, but the effect on the creepages is very small. This means that the weight does not have a significant effect on the change in the wheel-rail contact areas position and the angle of attack (see Fig. 4.9 on the right).

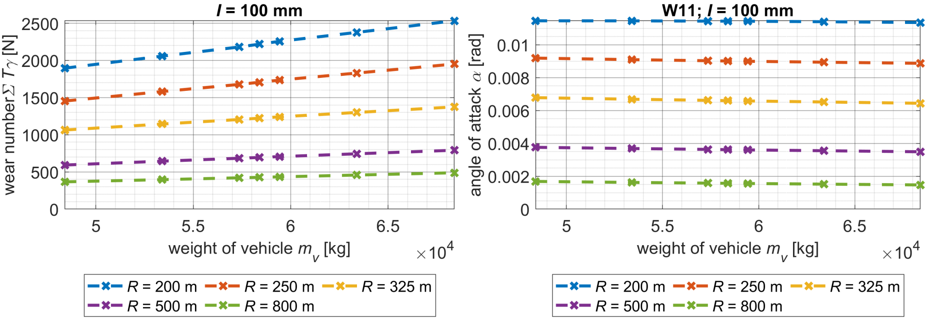


Figure 4.9: Dependence of the sum of the Wear Number (left) and the angle of attack of the first wheelset (right) on the weight of the vehicle for individual curve radii.

4.3.6 Summary of this Section

The following parameters have the greatest influence on the sum of Wear Number values for the entire vehicle: curve radius R, wheelset distance $2a_1$, primary yaw stiffness k_{t1}, weight of vehicle m_v. The parameters of the secondary suspension have an effect on the Wear Number for individual bogies, however, when evaluating the entire vehicle, this effect is eliminated (for the considered range of parameters values).

Then the values of the secondary suspension parameters have an impact on the evaluation of the Swiss methodology and the Rail surface damage parameter T_{qst} mentioned in Chapter 1. From the point of view of the overall effects on the track, these methods are not set correctly.

4.4 Design of Methodology for the Assessment of Dynamic Effects

In this section, the models based on the sum of the Wear Number values are designed. This parameter represents damaging effects (wear) and has a quasi-static character, but is commonly mentioned in publications dealing with vehicle dynamics [7], [8]. In this context, it is considered as a dynamic effect.

4.4.1 Strategy of Inserting the Influence of Parameters

The basic model is based on the well-known hyperbolic dependence of the sum of the Wear Number (as well as the curve resistance) on the curve radius. The effect of the cant deficiency is not directly included in the model, but it is possible to define a set of model coefficients for specified values of cant deficiency. Selected parameters based on the sensitivity analysis are included using functions. The output value of these functions for the reference vehicle model is equal to 1. If the output values are greater than 1, the vehicle has worse properties in terms of the sum of the Wear Number and also the damaging effects than the reference vehicle.

4.4.2 Obtained Models

Two variant models are considered. The new parameter is called Equivalent Sum of Wear Number $T\gamma_{ekv}$, which is defined based on the sum of Wear Number values $\Sigma T\gamma$ obtained from simulations. A curve fitting tool (based on the least squares method) was used to find functional dependencies (f and r) of individual parameter (p_i)s. Variant A of this model is as follows:

$$\Sigma T\gamma \approx T\gamma_{ekv,a} = \frac{4.85 \cdot 10^5 \prod_i f_a(p_i)}{R} - 232 \prod_i r_a(p_i), \tag{4.5}$$

where $p_i = \{2a_1; k_{t1}; m_v\}$. The functional dependencies of f_a and r_a for the condition of cant deficiency of 100 mm are given by the relations:

$$f_a(2a_1) = 0.358 \cdot 2a_1 + 0.134 \qquad r_a(2a_1) = -0.349 \cdot 2a_1 + 1.82 \tag{4.6}$$

$$f_a(k_{t1}) = 0.151 \cdot \ln(k_{t1}) - 1.72 \qquad r_a(k_{t1}) = -0.176 \cdot \ln(k_{t1}) + 4.21 \tag{4.7}$$

$$f_a(m_v) = 1.45 \cdot 10^{-5} \cdot m_v + 0.153 \qquad r_a(m_v) = 1.65 \cdot 10^{-5} \cdot m_v + 0.0369 \tag{4.8}$$

Variant B of this model is as follows:

$$\Sigma T\gamma \approx T\gamma_{ekv,b} = \frac{3.28 \cdot 10^5 \prod_i f_b(p_i)}{R - 53.5 \prod_i r_b(p_i)}, \tag{4.9}$$

where $p_i = \{2a_1; k_{t1}; m_v\}$. For the cant deficiency of 100 mm, the following applies:

$$f_b(2a_1) = 0.689 \cdot 2a_1 - 0.659 \qquad r_b(2a_1) = -0.887 \cdot 2a_1 + 3.17 \tag{4.10}$$

$$f_b(k_{t1}) = 0.289 \cdot \ln(k_{t1}) - 4.23 \qquad r_b(k_{t1}) = -0.504 \cdot \ln(k_{t1}) + 10.1 \tag{4.11}$$

$$f_b(m_v) = 1.32 \cdot 10^{-5} \cdot m_v + 0.229 \qquad r_b(m_v) = 0.348 \cdot 10^{-5} \cdot m_v + 0.796 \tag{4.12}$$

The developed algorithm makes it possible to define coefficients for other values of the cant deficiency in these mentioned equations. These coefficients are also defined based on the selected reference vehicle model. If it is necessary to use another reference vehicle model, it is possible to perform only selected simulation scenarios and calibrate the coefficients.

4.4.3 Evaluation of Obtained Models

The obtained models contain several approximation functions, which cause the inaccuracy on the outputs from these models compared to the simulation results. Verification of the accuracy of the obtained models is implemented using the coefficient of determination R^2. The coefficient of determination values were calculated for all dependencies (used to derive the models) of the sum of the Wear Number on the curve radius and divided into intervals of size 0.005 (see Fig. 4.10). Variant B shows better agreement with the sum of Wear Number values from the simulations than variant A. For both variants, except for one approximation, the coefficient of determination values are always greater than 0.97, which indicates a good agreement of the created models with the original simulation data.

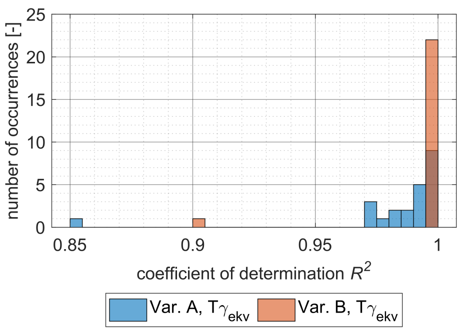


Figure 4.10: Graph of number of occurrences in the coefficient of determination intervals by size 0.005.

Another part of the verification is the analysis of the absolute and relative deviation of the obtained models from the simulation data. This analysis shows the accuracy of the models at specific curve radii (see Fig. 4.11 and Fig. 4.12 on the left). Variant A shows larger deviations for the smallest and largest curve radii considered. This effect is more noticeable in the relative deviation for large curve radii due to the small absolute values of the Wear Number. On the right side of these figures, the obtained values of deviations are summarized in graphs of the number of occurrence.

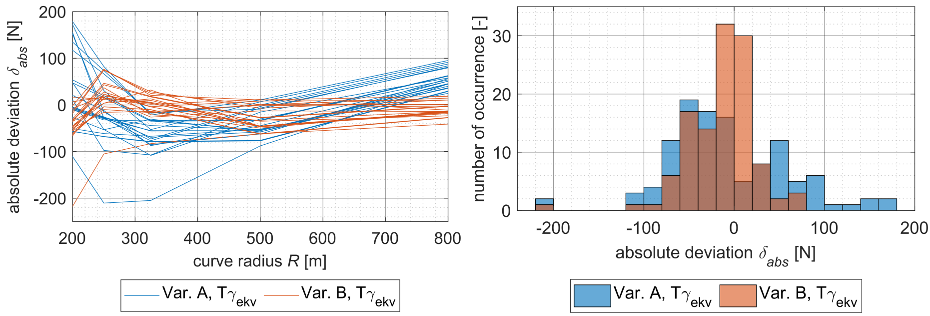


Figure 4.11: Absolute deviations of $T\gamma_{ekv}$ from values obtained by simulations depending on the curve radius R (left) and the graph of the number of occurrence of absolute deviation values with an interval of 20 N (right).

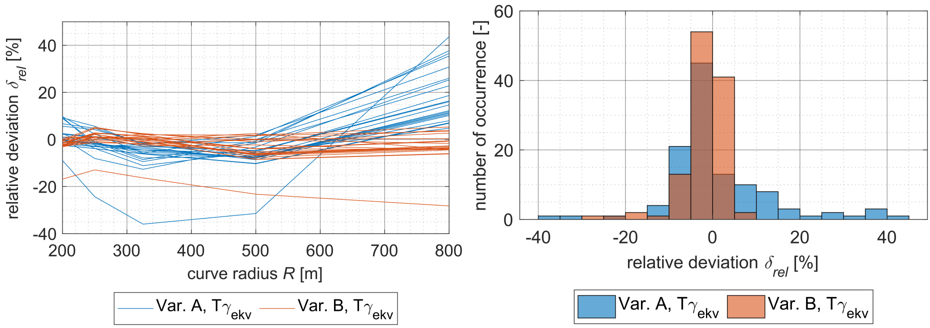


Figure 4.12: Relative deviations of $T\gamma_{ekv}$ from values obtained by simulations depending on the curve radius R (left) and the graph of the number of occurrence of relative deviation values with an interval of 5 % (right).

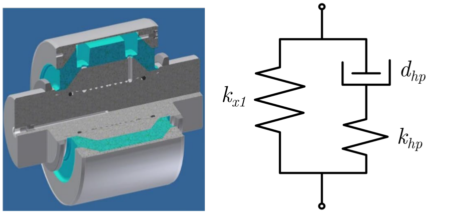
The considered model $T\gamma_{ekv}$ of variant B shows better values of the coefficient of determination and the values of absolute and relative deviation are in a narrower interval than for variant A. In conclusion, variant B can be recommended as more accurate and suitable.

4.5 Technical Solutions to Reduce Damaging Effects

Reducing the longitudinal stiffness of the primary suspension is one way to reduce the damaging effects of the vehicle on the track when passing through a curve, but this leads to problems with the running stability of the vehicle.

4.5.1 Hydraulic Bushing

The hydraulic bushing [11] is a specific element for primary suspension with a stiffness that depends on the frequency of its deformation (see Fig. 4.13). In curves, the stiffness of the element is low and at high speed (critical for stability) the stiffness is high.



Model	**Run1**	**Run31**	**Run77**	**Run78**
k_{x1} [N/m]	$35 \cdot 10^6$	$3,5 \cdot 10^6$	$3,5 \cdot 10^6$	$3,5 \cdot 10^6$
k_{hp} [N/m]	0	0	$31,5 \cdot 10^6$	$38,5 \cdot 10^6$
b_{hp} [Ns/m]	0	0	$6,5 \cdot 10^6$	$6,5 \cdot 10^6$

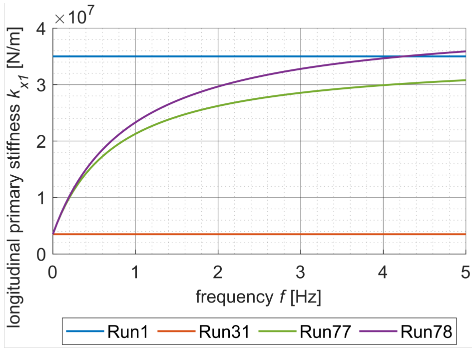


Figure 4.13: Image [11], model of the hydraulic bushing, considered stiffness setting listed in the table and depending on the frequency shown in the graph.

The considered models of the hydraulic bushing (see Fig. 4.13) show a low sum of Wear Number values in the wheel-rail contact when the vehicle passes through a curve (see Fig. 4.14 on the left). In these situations, the frequency of deformation is low. The stability analysis shows that the critical speed will increases and approaches the case with higher stiffness when this element is used (see fig. 4.14 on the right).

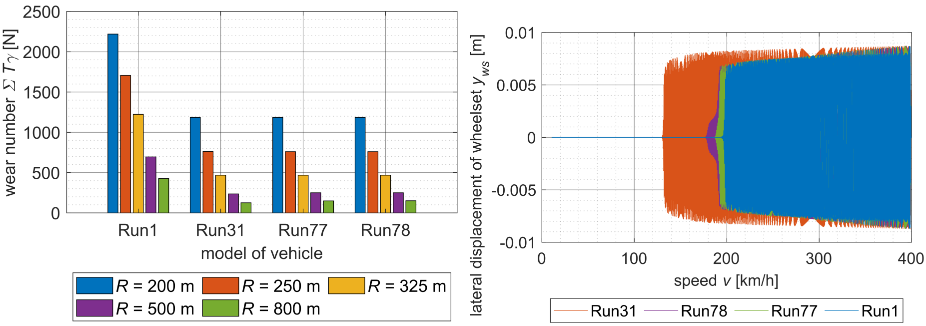


Figure 4.14: Dependence of the sum of the Wear Number for individual models and curve radii (left) and dependence of the lateral displacement of the wheelset on the running speed in a straight track for individual models (right).

4.5.2 Wheelset Coupling

To increase the vehicle stability at high speeds, it is possible to use wheelset couplings, which allow to use a lower longitudinal stiffness of the primary suspension [8], [12]. There are several variants of this connection, 3 types were considered for this work (see Fig. 4.15) with very low longitudinal stiffness of the primary suspension.

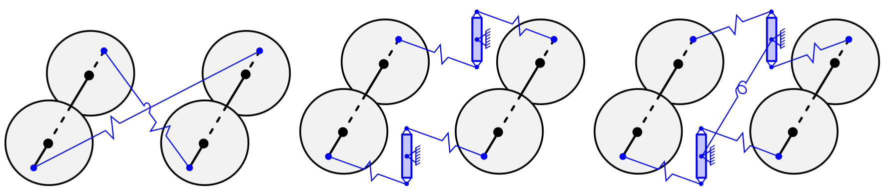


Figure 4.15: Overview of considered wheelset couplings (from the left - Run67, Run76, Run79).

For very small curve radii, the wheelset couplings do not affect the values of the sum of the Wear Number. As the curve radius increases, the Run67 solution (diagonal coupling) increases Wear Number values compared to the others (see Fig. 4.16). This occurs due to the Wear Number values on the rear wheelset and its angle of attack, which is the effect of the mechanism used, which connects wheelsets in the lateral direction as well. The Run67 solution significantly improves vehicle running stability and critical speed. The remaining solutions (Run76 and Run79) show a higher critical speed than the case with the same low longitudinal stiffness but without wheelset coupling (Run31).

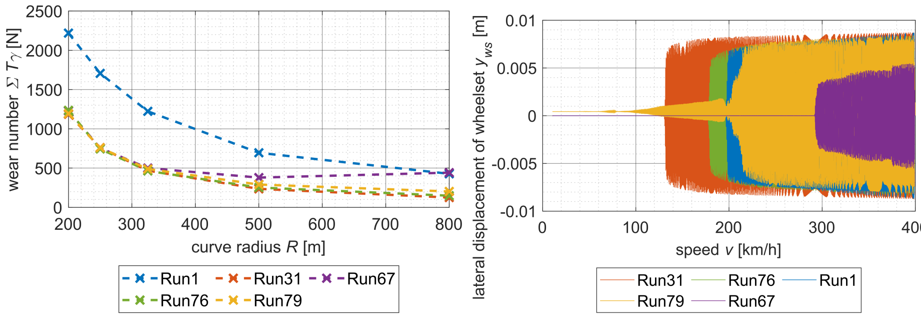


Figure 4.16: Dependence of the sum of the Wear Number for individual models (left) and dependence of the lateral displacement of the wheelset on the running speed in a straight track for individual models (right).

Thanks to these wheelset couplings, it is possible to use lower longitudinal stiffness in the vehicle design (reduce the damaging effects on the track in the curve) and maintain stable running at higher speeds.

4.5.3 Friction Coefficient Modifier

A direct way to reduce the damaging effects of the vehicle on the track in a curve and not significantly affect the vehicle design is to use a friction modifier. Since the friction coefficient in the wheel-rail contact is also an important parameter for traction and braking, it is necessary to modify it locally only in the flange area of the wheel profile (see Fig. 4.17 on the left).

The dependence of the sum of the Wear Number on the value of the friction coefficient on the flanges of the leading wheels of both bogies has a linear character (see Fig. 4.17 on the right). This effect can be compared to the effect of vehicle weight, as the characters are very similar. The friction modifier could be introduced into the track damage prediction calculations as a reduction in the vehicle weight.

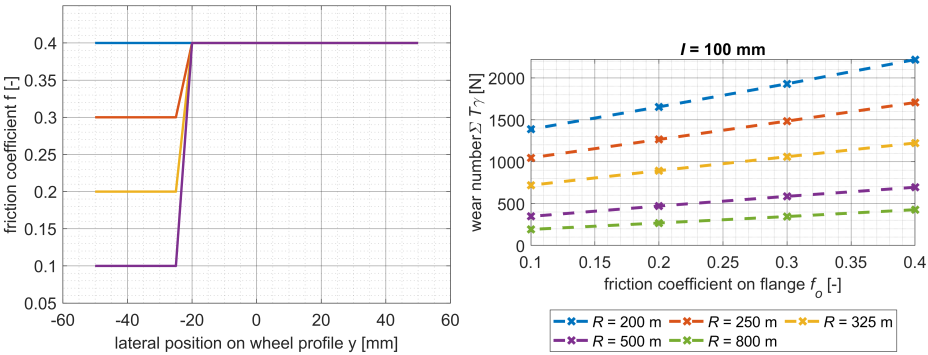


Figure 4.17: Dependence of the friction coefficient values on the lateral position on the wheel profile (left) and dependence of the sum of the Wear Number on the friction coefficient on flange for individual curve radii (right).

The reduction in the friction coefficient on the leading wheel leads to a better (from the point of view of damaging effects) position of the wheelset. This also affect the position of the wheel-rail contact on the inner wheel *W12* and the forces acting on it (see Fig. 4.18). Thus, by using the friction modifier, damage reduction can be achieved on the entire wheelset.

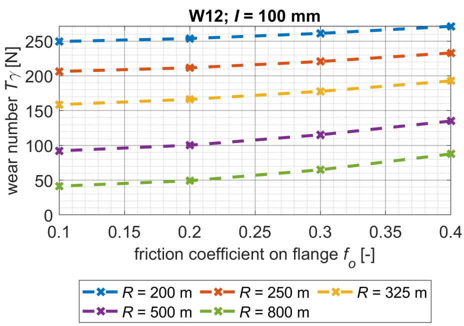


Figure 4.18: Dependence of $\Sigma T\gamma$ of inner wheel *W12* on the values of friction coefficient on flange of leading wheels of both bogies.

5. Contribution of the Dissertation Thesis

The main result of the dissertation thesis is the creation of a method for evaluating the dynamic effects that cause damaging effects of a vehicle on a track in a curve. This method defines the parameter $T\gamma_{ekv}$ that can compare the vehicles design in terms of their damaging effects on the track. This parameter is derived based on sensitivity analysis of vehicle parameters performed using multi-body simulations. The parameter makes it possible to use the known dependencies of the Wear Number and wheel-rail damage, but compared to the Wear Number, it is not necessary to carry out further simulations for its evaluation. The sensitivity analysis shows the essential parameters that have an influence on the damaging effects of the vehicle on the track. These parameters are entered into the created method in such a way that they are not dependent on each other. It allows to enter them gradually or to add other parameters.

Further contribution of the dissertation can be summarized in the following points:

- A new procedure has been created for performing a large number of simulation calculations in the SIMPACK program, which includes a program for automatically starting simulation calculations. This new skill speeds up working with the SIMPACK program.
- For easier and faster work with data, programs for processing output data from the SIMPACK program were created.
- The results show the agreement of the sum of the Wear Number for the entire vehicle when passing through a curve with the force that needs to be applied to the vehicle to maintain a constant running speed. This means that the energy supplied to the vehicle (under constant speed condition) is converted into wheels and rails damage due to creepage effects. This phenomenon can be called curve resistance.
- According to the sensitivity analysis, the curve resistance (as well as Wear Number values and damaging effects of the vehicle on the track) depends on the design parameters of the vehicle and not only on the curve radius and

the weight of the vehicle, as is currently considered for the needs of traction calculations. This work found that other significant parameters are: wheelset distance, primary yaw stiffness, cant deficiency (vehicle speed).

- Methodologies dealing with the evaluation of the damaging effects of the vehicle on the track, which use the results of simulation calculations, especially the Wear Number values, should define the parameters and the method of solving the wheel-rail contact. The infrastructure manager methodologies mentioned in the first chapter do not define this. The choice of the method of solution can significantly change the results.

- To evaluate the damaging effects of the vehicle on the track, it is important to consider the effects of the entire vehicle. Some vehicle design parameters affect the Wear Number values of the leading wheel, which is often considered representative of the evaluation of the vehicle's dynamic effects on the track (for example in the Swiss methodology and the Rail surface damage parameter mentioned in Chapter 1), however, this effect is reflected in the sum of the Wear Number and the proposed model in this work $T\gamma_{ekv}$ for the entire vehicle. An example is the secondary suspension parameters (considered in certain intervals of values), which increase the value of the Wear Numbers of the front bogie, but reduce these values on the rear bogie.

- Selected technical solutions to reduce damaging effect were tested. These solutions are mostly aimed to improving running stability at high speed, which makes it possible to design bogies with softer suspensions and reduce the damaging effects on the track. These characteristics have been confirmed.

- A very effective solution for reducing damaging effects that does not require major changes in bogie design can be the use of a friction modifier. This locally adjusts the contact conditions in the flange area of the wheel profile. Its effect is comparable to reducing the weight of the vehicle. For vehicle damaging effects evaluation, the use of this technology can be compensated by an equivalent reduction in the computational weight of the vehicle.

The knowledge gained can help railway infrastructure managers develop methodologies for comparing vehicle designs and their damaging effects on the track. Comparisons can be made without the need to perform simulation calculations, but more accurately than currently used methods. This can help motivate transport operators to use track-friendly vehicles, leading to a more sustainable rail transport.

During doctoral studies, the author participated in research tasks of the Department of Transport Means and Diagnostics that were closely related to the topic of this thesis. They were assigned and solved with the railway infrastructure manager and also with the rail vehicle manufacturers. This means that practice and industry are interested in solving the problem of damaging effects of vehicles on th track and simulation calculations of vehicle running.

For future work, the agreement of the curve resistance with Wear Number for the entire vehicle opens the possibility of experimental verification. The curve resistance can be measured as the tractive force required to maintain a constant speed and compared to running on a straight track. Furthermore, the obtained model does not include *RCF* and active suspension element analyses, which may be future topics.

Bibliography

[1] TUNNA, J.; SINCLAIR, J. and PEREZ, J. A review of wheel wear and rolling contact fatigue. In: *Procceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit.* 2007. ISSN 0954-4097. p. 271-289. Available from: https://doi.org/10.1243/0954409JRRT72.

[2] DIRKS, S.; ENBLOM, R. and BERG, M. Prediction model for wheel profil wear and rolling contact fatigue. In: *Wear.* 2011, p. 210-217. ISSN 00431648. Available from: https://doi.org/10.1016/j.wear.2010.10.028.

[3] WOODHEAD, Daniel Herbert. Investigating the performance of rail steels. In: *Fields: journal of Huddersfield student reasearch.* 2021. ISSN 2057-0163. Available from: https://doi.org/10.5920/fields.810.

[4] BAV-242.9-00001/00003/00025/0020. *Base Price Wear in the train-path pricing system 2017: Instructions for determining vehicle prices.* Bern: Federal Office of Transport (BAV), Finance departmanet, 2017.

[5] NETWORK RAIL. *Control Period 6 (CP6) Variable Usage Charge (VUC) guidance document - the calculation and approval of new rates.* Version 2. 2020.

[6] EN 14363 +A2. *Railway applications - Testing and Simulation for the acceptance of running characteristics of railway vehicles - Running behaviour and stationary tests.* Brussels: European Committee for Standardization, 2022.

[7] KNOTHE, Klaus and STICHEL, Sebastian. *Rail Vehicle Dynamics.* Cham: Springer International Publishing AG, 2017. ISBN 978-3-319-45376-7.

[8] IWNICKI, Simon; SPIRYAGIN, Maksym; COLE, Colin a MCSWEENEY, Tim. *Handbook of Railway Vehicle Dynamics.* Second Edition. CRC Press, 2019. ISBN 978-1-1386-0285-4.

[9] KALKER, J. J. A Fast Algorithm for the Simplified Theory of Rolling Contact. In: *Vehicle System Dynamics.* 2007, p. 1-13. ISSN 0042-3114. Available from: https://doi.org/10.1080/00423118208968684.

[10] POLACH, Oldrich. A Fast Wheel-Rail Forces Calculation Computer Code. In: *Vehicle System Dynamics.* 1999, p. 728-739. ISSN 0042-3114. Available from: https://doi.org/10.1080/00423114.1999.12063125.

[11] ZHOU, Yichang; TIAN, Qiuyong a HECHT, Markus. Application of Hydraulic Arm Bushings with Frequency-Dependent Stiffness to Compromise Hunting Stability and Curve Negotiation Performance for a Passenger Coach. In: *Journal of Vibration Engineering & Technologies.* ISSN 2523-3920. Available from: https://doi.org/10.1007/s42417-024-01393-1.

[12] POLACH, O. Curving and stability optimisation of locomotive bogies using interconnected wheelsets. In: *Vehicle System Dynamics.* 2004. Available from: https://www.polach.ch/data/object_4/Paper_IAVSD-2003.pdf

List of author's publications

- JENIŠ, F.; MICHÁLEK, T.; KUBÍK, M.; STRECKER, Z.; ŠLAPÁK, J. and MAZŮREK, I. The influence of semi-actively controlled magnetorheological bogie yaw dampers on the guiding behaviour of a railway vehicle in an S-curve: Simulation and on-track test. In: *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit.* 2024. p. 1-10.
Available from: https://doi.org/10.1177/09544097241282728

- MICHÁLEK, T.; KOHOUT, M.; ŠLAPÁK, J.; VÁGNER, J. and PULDA, J. Curving and running resistance of freight trains: current experience with on-track measurements. In: *Vehicle System Dynamics.* 2024. p. 1-15. ISSN 0042-3114.
Available from: https://doi.org/10.1080/00423114.2024.2398003

- ŠLAPÁK, J.; LOKVENC, M.; MICHÁLEK, T.; HÁBA, A. and BÍLEK, A. Provoz elektrických jednotek v podmínkách ČH(E)Ž. In: *Zborník prednášok XXVI. medzinárodnej konferencie Súčasné problémy v kolajových vozidlách – PRORAIL 2023, Diel II.* Žilina, Slovensko, 2023, p. 313-322. ISBN 978-80-89276-62-2.
Available from: https://doi.org/10.26552/spkv.Z.2023.2.36

- MICHÁLEK, T.; KOHOUT, M.; ŠLAPÁK, J.; PULDA, J.; VÁGNER, J. et al. Experimental measurement and simulation of running resistance of container trains. In: *Proceedings of the 12th International Conference on Railway Bogies and Running Gears.* Budapest: Scientific Society of Mechanical Engineers, Department of Rolling Stock, 2022, p. 137-147. ISBN 978-963-9058-45-3.

- ŠLAPÁK, J. and MICHÁLEK, T. Comparison of selected parameters for evaluation of rail surface damage intensity. In: *Acta Polytechnica CTU Proceedings: Proceedings.* Praha: České vysoké učení technické v Praze, 2022, p. 11-12. ISSN 2336-5382.
Available from: https://doi.org/10.14311/APP.2022.35.0042

- ŠLAPÁK, J. Behaviour of the rail load parameter B defined by the standard EN 14363. In: *Proceedings of Computational Mechanics 2021: 36th conference with international participation.* Srní: University of West Bohemia, 2021, p. 224 - 227. ISBN 978-80-261-1059-0.

- MICHÁLEK, T.; JAROŠ, P.; VÁGNER, J.; KOHOUT, M.; ŠLAPÁK, J. et al. Zkoušení prvků tažného a narážecího ústrojí na dynamickém zkušebním stavu Dopravní fakulty Jana Pernera Univerzity Pardubice. In: *Současné problémy v kolejových vozidlech 2021: sborník příspěvků z XXV. konference s mezinárodní účastí.* Pardubice. Univerzita Pardubice, 2021, p. 199-206. ISBN 978-80-7560-377-7.

- ŠLAPÁK, J. and MICHÁLEK, T. Vehicle/track interaction under the conditions of high speed railway operation. In: *Acta Polytechnica CTU Proceedings.* 2021, p. 45-52. ISSN 2336-5382.
  Available from: https://doi.org/10.14311/APP.2021.31.0045

- ŠLAPÁK, J.; MICHÁLEK, T.; ŠPALEK, P. and HORA, M. Rozbor švýcarské metodiky pro hodnocení poškozujících účinků jízdy kolejového vozidla na kolej. In: *Vědeckotechnický sborník Správy železnice, státní organizace: č. 2/2020.* Praha: Správa železnic, státní organizace, 2020, p. 1-16. ISSN 2694-9172.

- MICHÁLEK, T.; ŠLAPÁK, J.; ŠPALEK, P. and HORA, M. Hodnocení poškozujících účinků jízdy kol kolejových vozidel na kolej. In: Súčasné problémy v kolajových vozidlách - PRORAIL 2019: zborník prednášok II. Žilina: VTS pri ŽU v Žiline, 2019, p. 65-73. ISBN 978-80-89276-59-2.

Abstract / Souhrn

The aim of the dissertation is the analysis of the dynamic effects of a vehicle running on a track when passing through a curve, which cause damage in the wheel-rail contact. Approaches to the assessment of damage effects and their application in methodologies for determining track usage charges by railway infrastructure managers are summarized. The so-called Wear Number, which expresses the specific work of the creep forces in the wheel-rail contact, was chosen as an evaluation parameter. The thesis uses multi-body simulation calculations of rail vehicles, as the Wear Number cannot currently be obtained by means of measurements, and the thesis contains an extensive sensitivity analysis of the vehicle parameters. The sensitivity analysis is devoted to the setting of the computational model of wheel-rail contact, the design parameters of the vehicle, the curve radius and the cant deficiency. Based on this analysis, a model for calculating the equivalent wear number $T\gamma_{ekv}$ is created. The model can be used to estimate the values of the sum of the Wear Numbers for the entire vehicle and to refine the calculation of the curve resistance. At the end, attention is paid to technical solutions for reducing the curve resistance.

Cílem disertační práce je analýza dynamických účinků jízdy vozidla na kolej při průjezdu obloukem, které způsobují poškození kola a kolejnice v jejich kontaktu. Jsou shrnuty přístupy k hodnocení poškozujících účinků a jejich aplikace v metodikách pro stanovení poplatků za použití dopravní cesty správci železniční infrastruktury. Jako hodnotící parametr zvoleno tzv. číslo opotřebení, které vyjadřuje měrnou práci skluzových sil v kontaktu kolo-kolejnice. V práci je využito simulačních výpočtů jízdy kolejového vozidla, jelikož číslo opotřebení nelze v současné době získat pomocí měření a práce obsahuje rozsáhlou citlivostní analýzu parametrů vozidla. Citlivostní analýza se věnuje nastavení výpočetního modelu kontaktu kolo-kolejnice, konstrukčním parametrům vozidla, poloměru oblouku a nedostatku převýšení. Na základě této analýzy je vytvořen model pro výpočet ekvivalentního čísla opotřebení $T\gamma_{ekv}$, pomocí kterého lze odhadnout hodnotu součtu čísel opotřebení na všech kolech vozidla a zpřesnit výpočet odporu z jízdy obloukem. Na závěr je věnována pozornost technickým řešením pro snížení odporu z jízdy obloukem.