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Towards Adaptive Bogie Design

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ABSTRACT

Suspension components play a crucial role in bogie dynamics behavior. In this regard, passive and active systems are developed to meet various design requirements. Adaptive suspension systems can adjust the bogie dynamics with respect to different operational scenarios and as a result improve the vehicle performance. Therefore, there is a great demand on design of adaptive suspension systems for high speed train bogies. Sensitivity analysis, optimization, and active vibration control techniques are the most important steps towards adaptive bogie design. In this thesis, the target is to cover these steps partly by formulating and solving the prescribed problems for some example railway vehicle models. Therefore, hierarchical levels of vehicle modelling are considered. The overall performance of a vehicle can be evaluated from different perspectives. In this thesis, the dynamics behaviour of railway vehicles is reflected by ride comfort, wear, safety, and in particular running stability, track shift force, and risk of derailment objective functions. The mathematical representation of the prescribed objectives as well as the evaluation procedure are described thoroughly. As an example on optimization problem with application in railway vehicles, the comfort/safety multiobjective optimization of a one car vehicle lateral dampers is considered. The genetic algorithm routine is used to solve this problem. In order to have a better wear estimation, a theoretical contact search approach is applied to calculate the creepages and wear. The optimization problem of primary dampers towards wear showed that one might achieve better wear performance by using active technology in bogie primary suspension components. Therefore, different on/off semi-active control strategies are integrated together with the magnetorheological dampers in bogie primary suspension and the corresponding effects on wear is explored on different operational scenarios. Finally, to have a better insight into adaptive bogie design, the global sensitivity analysis of bogie dynamics behavior with respect to suspension components is considered. The multiplicative version of the dimension reduction method is employed to provide the sensitivity indices. The result of such analysis can narrow down the number of input variables for the optimization and adaptive bogie design problems and improve the computational efficiency. All in all, this thesis deals with formulating and solving some example problems on the sensitivity analysis and optimization of railway vehicles. Furthermore, as an introduction to active bogie systems, the application of semi-active vibration control strategies in bogie primary suspension is also considered. The results of the current thesis can provide useful hints in design of adaptive suspension system for high speed train bogies.

KEYWORDS: Optimization, sensitivity analysis, bogie, suspension system, active control.

To my dear mom, dad, and sisters

PREFACE

This work has been accomplished during November 2011 until June 2013 at the department of Applied Mechanics, Chalmers University of Technology, Gothenburg, Sweden. The current licentiate thesis is part of the SD9 project defined at the Chalmers railway mechanics center of excellence (CHARMEC). This project is financially supported by the Ekman family foundation which is gratefully acknowledged.

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Gothenburg, May 2014

Seyed Milad Mousavi Bideleh

THESIS

This thesis consists of an extended summary and the following appended papers:

Paper A Mousavi Bideleh S.M., and Berbyuk V., Multiobjective optimization of a railway vehicle dampers using genetic algorithm, *Proceedings of the ASME 2013 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference IDETC/CIE 2013*, August 4-7, Portland, Oregon, USA, 2013.

Paper B Mousavi Bideleh S.M., and Berbyuk V., Optimization of a bogie primary suspension damping to reduce wear in railway operations, *Proceeding of the ECCOMAS Thematic Conference on Multibody Dynamics*, July 1-4, Zagreb, Croatia, 2013.

Paper C Mousavi Bideleh S.M., and Berbyuk V., Application of semi-active control strategies in bogie primary suspension system, *Proceeding of the 2nd International Conference on Railways*, April 8-11, Ajaccio, France, 2014.

Paper D Mousavi Bideleh S.M., and Berbyuk V., Global sensitivity analysis of bogie dynamics towards suspension components, *submitted for international publication*.

Appended papers were prepared in collaboration with the co-author. The author of this thesis is responsible for the major progress of the work in papers A-D. This includes the planning, theory development, numerical simulations, analysis and writing.

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PART I

EXTENDED SUMMARY

The general outlines of this thesis are described in this section. After a brief overview concerning the aims and background of the project, the vehicle models developed for the analysis are introduced and a set of objective functions are defined to reflect the vehicle's dynamics performance in mathematical terms. Finally, an overview on optimization, semi-active control, and global sensitivity analysis problem formulation and methodology is given.

1 INTRODUCTION

Railways are known as one of the most prominent ways of transportation, nowadays. Reasonably cheap, clean, and fast transport are some of the most important aspects of railway industry which makes it competitive with road vehicles, ships and airplanes. The cost efficiency in railway operations can be scrutinized from different points of view. Maintenance cost, track access charges and design requirements due to the passenger ride comfort, safety, and wheel/rail contact wear improvement are some of the most significant challenges in this field. On the other hand, it is often desired to run the vehicle as fast as possible. This could not only shorten the journey time, but also reduce the track access charges. However, higher speeds can deteriorate the safety, ride comfort and wear conditions. Therefore, it is inevitable to formulate and solve several multiobjective optimization problems to keep the different objective functions within the satisfactory limit and achieve a higher efficiency. In this thesis, ride comfort, safety and wear are chosen as the main objective functions to evaluate the vehicle's performance. Furthermore, the vehicle speed is usually set to the maximum admissible value corresponding to each particular operational scenario.

There are several possibilities to improve the vehicle performance with respect to the abovementioned objective functions. For example, using new material properties as well as optimized design for wheel and rail could be an option. However, the wheel and rail profiles are subject to change during the operation and there are some uncertainties concerning the proficiency of such solutions. Another alternative is to make the vehicle's performance better by improving the bogie primary and secondary suspension components. In this regard, passive, semi and fully active suspension components are developed. Each of these systems has some advantages and disadvantages. Passive components are quite simple, cheap, and easy to use. However, the ultimate efficiency that could be extracted from such systems is limited and there is no control on the corresponding suspension force. Semi and/or fully active systems on the other hand can adapt the bogie dynamics with respect to different operational scenarios and yield a higher performance. However, the design and maintenance costs are more remarkable, additional power supply is needed and there is a risk of failure.

In general, high speed train's bogie is a complex nonlinear dynamical system with highly interconnected elements. A change in suspension components to improve the ride comfort for instance, might worsen the wear and safety conditions. Consequently, as an initial stage in adaptive bogie design, it is important to study the effects of different suspension components on the vehicle's dynamics as well as formulate and solve several multiobjective optimization problems to attain trade-off solutions that provide the optimized suspension components characteristics. Such an analysis not only yields informative data concerning the passive suspension design, but also gives insight into adaptive bogie design.

In this regard, several researches on the bogie suspension optimization problem are done. Eberhard *et al.* proposed an efficient optimization algorithm with application in multibody dynamics analysis using the deterministic gradient-based as well as stochastic methods [1]. A switching between those algorithms during the optimization can help to find the global minima in a more efficient way. The genetic algorithm based multiobjective optimization routines are widely used in railway applications, see e.g. [2, 3]. Johnsson *et al.* considered the multiobjective optimization problem of bogie suspension components with respect to safety and ride comfort [4]. Results for a half car model developed in MATLAB as well as a three car railway vehicle model in GENSYS proved the efficiency and reliability of such optimization routines. The same approach is followed here to solve the optimization problems required for the sensitivity analysis and semi-active control of bogie suspension, see **PAPERS A-D**.

In the case of passive suspension, it is vital to utilize the optimized values of design parameters to achieve satisfactory level of improvement in dynamics behaviour. However, optimization is usually a time costly process especially when it comes to the complex systems with large number of degrees of freedom (DOFs) and design variables. Consequently, a proper choose of the objective functions combination and design parameters can ameliorate the computational efficiency of the optimization. This could be done by means of sensitivity analysis. Some studies are already done on the local sensitivity analysis with application in railways, see e.g. [5-7]. However, the efficiency of such approaches is highly depends on the initial values considered for the analysis and there are some uncertainties correlated to the reliability of the obtained results. Therefore, the global sensitivity analysis of the bogie dynamics behaviour with respect to the suspension components is an interesting problem to solve.

Monte Carlo simulation is one of the most widely-used procedures in sensitivity analysis of mechanical systems. However, the computational effort for Monte Carlo simulation is costly (especially for a complex nonlinear system like a high speed train bogie) and it is necessary to use a more time efficient algorithm for that particular purpose. Based on the multiplicative version of dimension reduction method (M-DRM), Zhang *et al*. proposed a closed form solution for the global sensitivity indices [8]. This method can dramatically reduce the computational efforts required for the global sensitivity analysis compared to Monte Carlo simulation, while the results obtained are within the same order of accuracy. The same methodology is applied in this thesis to solve the global sensitivity analysis problem of bogie dynamics towards suspension components. Based on the results attained from such an analysis, one can detect those suspension components that have the most important influence on bogie dynamics behaviour. This is particularly useful in narrowing down the number of input design parameters required for optimization and design of adaptive bogies which can attenuate the computational efforts in simulations, significantly. This is the subject of **PAPER D**.

Application of the semi and/or fully active components in bogie suspension is another important issue in adaptive bogie design criterion. Such components are mostly used in the secondary suspension system and many researches are done to investigate the effects of active suspension technology on the ride comfort in passenger trains, see e.g. [9-11]. One of the main targets of the current research is to explore the applicability of different control techniques in bogie primary and secondary suspension system and examine the changes in the bogie dynamics behaviour with respect to the corresponding passive system. This could be usually done by means of a set of sensors and actuators and a proper control strategy. Magnetorheological (MR) dampers have a simple construction and are widely-used as actuators in different applications. As an introduction to active bogie design, integration of such dampers in a semi-active control with passive elements is subject to study in this project, see **PAPER C**. This could give insight into future researches on the design criterion of fully active control systems for the high speed train bogies.

All in all, optimization, sensitivity analysis, and semi-active vibration control of bogie suspension are the main aims and objectives in this thesis. The vehicle models developed in each case are introduced in the subsequent section. Ride comfort, safety, and wear are the main objective functions that exhibit the vehicle's dynamics performance on different operational scenarios. Finally, the results obtained from the current research can provide informative data for design and optimization of passive and active suspension systems for railway vehicles.

2 VEHICLE MODELS

As a preliminary stage in adaptive bogie design, it is necessary to choose a proper vehicle model for the analysis. Many researches are done on vehicle modelling especially suspension system, see e.g. [12-18]. As an example, in the case of ride quality, trains with tilting mechanism can increase the passenger ride comfort level, significantly [9, 19, 20]. In this regard, hierarchical levels of modelling such as a single wheelset attached to a fixed bogie frame, different half car as well as one car vehicle models have been developed and the system dynamics response in each case has been compared with respect to the most advanced vehicle model. In the following, three types of the vehicle models used in this thesis are introduced.

2.1 Vehicle model A

The first model is a one car vehicle model shown in Fig. 1. This model is used in **PAPER A** for the comfort/safety multiobjective optimization problem with respect to the primary and secondary lateral dampers.

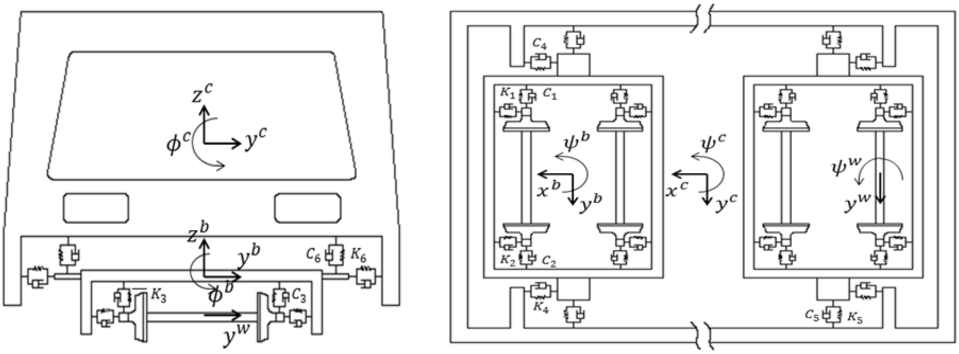

Fig. 1: One car vehicle model A.

The model is composed of a one carbody, two bogie frames, and four wheelsets. All these elements are rigid and each has a total DOFs equal to six in space, except for the wheelsets that each has two DOFs: a lateral (y^w) and a yaw (ψ^w) motion. It should be noted that the roll motion (ϕ^w) of the wheelset is considered as a dependent generalized coordinate as follows [21]:

$$\phi^w = (\lambda/a)\, y^w, \tag{1}$$

where, λ, and a are the wheel conicity angle and one-half the gauge, respectively. The primary and secondary suspension components of this vehicle model consist of a set of linear spring and dampers in the longitudinal, lateral, and vertical directions. The Lagrange formalism is employed to derive the equations of motion and 4th order Runge-Kutta method is used to solve the corresponding ordinary differential equations and yield the system response.

This type of model is simple, easy to implement, and the computation time is efficient (especially for optimization problems). However, there are some drawbacks within this model. For instance, the wheel profile is conical which is not realistic and as a result the creepages, contact forces and the vehicle dynamics response might show inappropriate values. In addition, the wheelsets have two DOFs, which makes it difficult to investigate the effects of vertical track irregularities on this vehicle model. Therefore, more advanced models are also subject to study in this thesis and introduced in the following subsections.

2.2 Vehicle model B

In order to surmount the wheel profile problems associated with the model described in the previous section, a more advanced one car vehicle model is developed in the multibody dynamics software SIMPACK. The overall model configuration is the same as Fig. 1, but the wheel and rail profiles are created based on the real unworn *S1002*, and *UIC60* profiles, respectively. Each wheelset has a total six DOFs in space and eight single DOF axle boxes (journal box) are added to the model to connect the wheelsets to the bogie frames through a set of primary suspension components. Again all the model elements except for the suspension components are rigid bodies.

Similar to the prior model, the suspension system in vehicle model B composed of a set of linear primary and secondary springs and dampers in the longitudinal, lateral and vertical directions and the only difference is that a lateral bump-stop is added to the secondary suspension components. Since the wheel and rail profiles are more advanced in this case, the creepages, contact forces and as a result the dynamics response of the system are more realistic. Furthermore, SIMPACK provides the possibility to use the measured data as the track irregularities in different directions and from this point of view more pragmatic operational scenarios can be considered for the analysis. This vehicle model is used in **PAPER C** of this thesis.

2.3 Vehicle model C

Similar to the previous case, vehicle model C is also developed in SIMPACK and composed of a one carbody, two bogie frames, four wheelsets, and eight journal boxes with the same DOFs as the vehicle model B. The difference here is that a more advanced suspension system with components listed in Table 1 is considered for the Vehicle model C. The superscripts *p* and *s* indicate primary and secondary suspension, respectively. l=1,2,3,4 is the wheelset number, m=x,y,z denotes the longitudinal, lateral and vertical directions, respectively. n=R,L represents the right or left hand side suspension component and h=L,T shows leading or trailing bogie, respectively. It should be noted that within this suspension system configuration the suspension elements corresponding to the right and left hand side of the wheelsets as well as the leading or trailing bogies are not necessary the same and could take different values. As an example, one could have $k^{p}_{1xR} \neq k^{p}_{1xL}$, which means the right and left hand side primary springs of the leading axle are not the same. To take into account the bushing effects, all dampers are modeled as serial spring-damper elements.

Table 1: Suspension system components for vehicle model C.

Suspension	Springs	Dampers	Anti-Roll bar	Traction-Rod	Bump-Stop
Primary	k^{p}_{lmn}	c^{p}_{lmn}	-	-	-
Secondary	k^{s}_{hmn}	c^{s}_{hmn}	k^{AR}_{h}	k^{TR}_{h}	k^{BS}_{h}

The Bogie and corresponding suspension system associated with the vehicle model C is shown in Fig. 2.

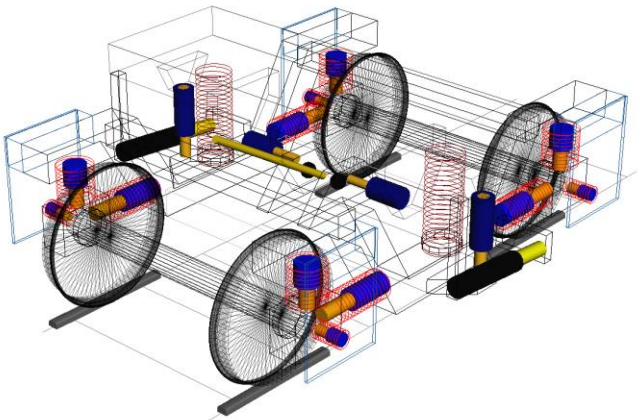

Fig. 2: Bogie corresponding to the vehicle model C developed in SIMPACK.

This vehicle model gives the possibility to explore the effects of different suspension components on the dynamics response of the vehicle on various operational scenarios and is particularly used for the sensitivity analysis problem in **PAPER D**.

3 OBJECTIVE FUNCTIONS

The ultimate goal of this research is to improve the vehicle's dynamics performance by modification of the bogie suspension components that leads to the adaptive bogie design concept. Within this framework, optimization, semi-active vibration control, and sensitivity analysis are the main problems subject to study in this thesis. As aforementioned earlier, the bogie dynamics performance is evaluated based on the vehicle's safety, ride comfort, and wear status on different operational scenarios. Therefore, in order to formulate the optimization, semi-active vibration control and sensitivity analysis problems, it is necessary to express the prescribed objective functions in mathematical terms. In most of the cases, this could be done by means of the railway standards. In the following the mathematical representation of the objective functions are introduced.

3.1 Ride comfort

Passenger comfort is one of the most prominent design issues in high speed trains. This criterion could be analyzed from different points of view such as ride comfort, passenger personal space, cabin temperature and ventilation system, noise and harshness isolation, boarding feasibilities, etc. However, in the present study, the focus is put on the effects of unpleasant vibrations felt by the passengers during the operation or simply the ride comfort quality. The carbody accelerations in different directions are the basis of the ride comfort evaluation in most of the cases. In the following, some of the common procedures regarding the high speed train's ride comfort evaluation are introduced.

3.1.1 Root mean square of accelerations

The carbody accelerations can be measured at different points. However, it is often customary to measure the accelerations on the floor of each carbody. The root mean square (RMS) of the carbody accelerations $a(t)$ could be considered as a ride comfort objective:

$$\hat{a}^{i}_{rms} = \sqrt{\frac{1}{t_f - t_0}\int_{t_0}^{t_f} \left(a^i(t)\right)^2 dt}\ ,\ i=1,2,\ldots,m. \tag{2}$$

Where, m is the total number of the measurement points. Based on this objective function, an increment in the RMS values of the carbody acceleration decreases the ride comfort quality.

Since, human body is sensitive to vibrations within a specific frequency range, it is better to use the frequency weighted accelerations in ride comfort evaluation. This could be done using special filters designed for that particular purpose in railway standards.

3.1.2 Wertungszahl (Wz)

Based on Wz criterion, the accelerations have to be measured at the floor plane of the vehicle carbody. Wz is then calculated as follows [22]:

$$Wz = 4.42\left(a^{wrms}\right)^{0.3}, \tag{3}$$

where a^{wrms} is the RMS value of the frequency weighted acceleration $a^w(t)(m/s^2)$. The corresponding frequency weighted functions for ride comfort evaluation in railway applications can be found in [21]. Based on Wz, human body is most sensitive to lateral and vertical excitations with frequencies around 3-7 Hz.

3.1.3 ISO 2631

In contradict to the previous criterion which directly uses the carbody accelerations, in ISO 2631 the focus is on the vibrations transmitted to the human body through the supporting surfaces. The frequency range is around 0.5-80 Hz for ride comfort and 0.1-0.5 Hz for motion sickness. The RMS values of the frequency weighted accelerations in the longitudinal, lateral, and vertical directions are used to evaluate the ride comfort quality based on ISO 2631. The corresponding frequency weighting functions can be found in [22].

3.1.4 ENV 12299

One of the most widely-used ride comfort criteria in railway application is evaluated based on the CEN standard ENV 12299 [23]. According to this standard, the ride comfort (N_{MV}) is evaluated in terms of the carbody frequency weighted accelerations in the longitudinal, lateral and vertical directions as follows:

$$N_{MV} = 6\sqrt{\left(a_{XP95}^{W_{ad}}\right)^2 + \left(a_{YP95}^{W_{ad}}\right)^2 + \left(a_{ZP95}^{W_{ab}}\right)^2}\ , \tag{4}$$

where, $W_{ad}=W_a \times W_d$ and $W_{ab}=W_a \times W_b$ are the weighting functions. $a_{XP95}^{W_{ad}}$, $a_{YP95}^{W_{ad}}$, and $a_{ZP95}^{W_{ab}}$, represent the 95% of the RMS value of the frequency weighted accelerations measured at the floor of carbody in the longitudinal, lateral and vertical directions, respectively.

The required transfer functions to calculate the frequency weighted accelerations in this case are given as follows [23]:

$$H_A(s) = \frac{s^2 4\pi^2 f_2^2}{\left(s^2 + \frac{2\pi f_1}{Q_1}s + 4\pi^2 f_1^2\right)\left(s^2 + \frac{2\pi f_2}{Q_1}s + 4\pi^2 f_2^2\right)}\ , \tag{5-a}$$

$$H_B(s) = \frac{(s+2\pi f_3)\left(s^2 + \frac{2\pi f_5}{Q_3}s + 4\pi^2 f_5^2\right)}{\left(s^2 + \frac{2\pi f_4}{Q_2}s + 4\pi^2 f_4^2\right)\left(s^2 + \frac{2\pi f_6}{Q_4}s + 4\pi^2 f_6^2\right)} \frac{2\pi K f_4^2 f_6^2}{f_3 f_5^2}\ , \tag{5-b}$$

$$H_D(s) = \frac{(s+2\pi f_3)}{\left(s^2 + \frac{2\pi f_4}{Q_2}s + 4\pi^2 f_4^2\right)} \frac{2\pi K f_4^2}{f_3}\ , \tag{5-c}$$

where, $H_A(s)$, $H_B(s)$, and $H_D(s)$ are the transfer functions associated with the band-pass filter W_a, vertical weighting filter W_b, and lateral weighting filter W_d, respectively. All the parameters required to evaluate the transfer functions in Eq. (5) can be found in [23].

The ride comfort should be evaluated leastwise at three points, in particular at the center of the carbody and above each bogie. Table 2 gives the ride comfort classification based on the CEN standard ENV 12299.

Table 2: Ride comfort classification.

N_{MV}<1	Very comfortable
1≤N_{MV}<2	Comfortable
2≤N_{MV}<4	Medium
4≤N_{MV}<5	Uncomfortable
N_{MV}≥5	Very uncomfortable

3.2 Safety

One of the most important criteria that must always be within the admissible design range is safety. This objective function can be scrutinized from different points of view. Track shift force, running stability and risk of derailment are the most prominent safety parameters that are considered in this thesis.

3.2.1 Track shift force

Track shift force (Y) is measured as the difference between the lateral forces acting on the left and right wheels of a wheelset. High track shift force might worsen the track irregularity condition and as a result increase the maintenance cost. According to the CEN standard EN-14363 [24], the track shift force limit is expressed (in kN) as follows:

$$\sum Y_{20Hz,2m,mean,99.85\%,\lim} \leq K_1\left(10+2Q_0/3\right), \tag{6}$$

where, K_1 is a constant (K_1=1, for passenger trains), and $2Q_0$ is the mean static axle load of the vehicle defined as:

$$2Q_0 = \frac{m_{veh}g}{n}, \tag{7}$$

m_{vel} is the total mass, and n is the number of axles of the vehicle. The final track shift force is equal to the 99.85% of the value obtained from the forces with a sliding mean over 2m window in 0.5m increments and subject to a 20 Hz low-pass filter. The track shift force objective function (Γ_{TS}) is then defined as:

$$\Gamma_{TS} = \max\left(\sum Y_{20Hz,2m,mean,99.85\%}\right)_l, \tag{8}$$

where, l=1,2,3,4 is the axle number. Indeed, Γ_{TS} is a scalar denotes the maximum filtered track shift force among all the wheelsets of the vehicle.

3.2.2 Running stability

Another important safety criterion is running stability that is particularly important at velocities near the critical hunting speed. The lateral guiding force (Y) defined in the previous subsection, can also be used as a measure of the running stability in railway applications.

According to the CEN standard EN-14363 [24], the limit condition for a vehicle to run stable, is expressed as:

$$\sum Y_{rms,100m,\lim} = \frac{\sum Y_{\max,\lim}}{2} = \frac{K_1\left(10+2Q_0/3\right)}{2}, \tag{9}$$

A sliding root mean square (rms) of the band-pass filtered guiding force in combination with a 100m window is applied to attain the final value. The running stability objective function (Γ_{St}) is then defined as:

$$\Gamma_{St} = \max\left(\sum Y_{rms,100m}\right)_l,\ l=1,2,3,4 \tag{10}$$

Similar to the previous case, the maximum value among all the axles is chosen as the objective function.

3.2.3 Risk of derailment

The final safety issue considered here is the risk of derailment, which is particularly important for vehicles navigating on curve operational scenarios at high speeds. Based on the CEN standard EN-14363 [24], the derailment coefficient is defined as the ratio of the lateral (Y) to vertical (Q) forces acting on each wheels of the vehicle. The safety condition to avoid derailment is then defined as:

$$\left(Y/Q\right)_{20Hz,2m,mean,99.85\%} \le 0.8, \tag{11}$$

in fact, the final derailment coefficient is calculated as 99.85% of the sliding mean over a 2m window of a low-pass filtered signal (with cut-off frequency 20Hz). The risk of derailment objective function (Γ_{RD}) is then defined as:

$$\Gamma_{RD} = \max\left(Y_t/Q_t\right)_{20Hz,2m,mean,99.85\%}, \tag{12}$$

in which, t=1,2,…,8 is the wheel number.

3.3 Wear

High speed and poor track quality can increase the wheel-rail contact wear and maintenance cost, significantly. Suspension components on the other hand can affect the wheelset dynamics behavior and resulting wear. In order to improve the bogie performance from wear point of view, it is interesting to explore the effects of different suspension components on wear. Therefore, wear is one of the main objective functions in this study. There are several procedures to estimate wear in railway operations, see e.g. [25-27]. In the following two main approaches in this field are introduced.

3.3.1 Archard's number

According to Archard, the material loss (V_w) is expressed as follows [28, 29]:

$$V_w = k\frac{N}{H}s, \tag{13}$$

where, N is the normal force, s the sliding distance, H material hardness, and k is the wear coefficient which depends on the governing wear regime, running environment, and wheel material properties.

Application of Archard equation in wear estimation of the railway vehicles on different operational scenarios has showed good agreement with the experimental data for both the flange and thread parts of the wheels, see e.g. [30].

3.3.2 Energy dissipation

The energy dissipation at the contact patch is another important measure for wear which is widely-used in railway applications and is defined as follows:

$$\bar{E} = F_{\xi}\upsilon_{\xi} + F_{\eta}\upsilon_{\eta} + M_{\xi\eta}\phi_{\xi\eta} \ , \tag{14}$$

where, υ_{ξ}, υ_{η}, and $\phi_{\xi\eta}$ are the longitudinal, lateral, and spin creepages and F_{ξ}, F_{η}, and $M_{\xi\eta}$ are the corresponding contact forces and contact moment. These creepages are defined as follows [31]:

$$\left.\begin{aligned} \upsilon_{\xi} &= \frac{\left(\dot{\mathbf{r}}^{w} - \dot{\mathbf{r}}^{r}\right)^{T} \mathbf{t}_1^r}{V} \\ \upsilon_{\eta} &= \frac{\left(\dot{\mathbf{r}}^{w} - \dot{\mathbf{r}}^{r}\right)^{T} \mathbf{t}_2^r}{V} \\ \phi_{\xi\eta} &= \frac{\left(\boldsymbol{\omega}^{w} - \boldsymbol{\omega}^{r}\right)^{T} \mathbf{n}^r}{V} \end{aligned}\right\} , \tag{15}$$

here, the subscripts w, and r denote wheel, and rail, respectively. $\dot{\mathbf{r}}$, and $\boldsymbol{\omega}$ are the global velocity vector of the contact point, and angular velocity vector, respectively. V is the reference velocity. $\mathbf{t}_1^r$, $\mathbf{t}_2^r$ are the longitudinal and lateral unit vectors and $\mathbf{n}^r$ is the normal unit vector on the rail profile at the contact point. It is clear that the creepages are a function of the contact point position on the wheel and rail. Therefore, as an initial step in wear calculation, it is necessary to identify the contact point location on the wheel and rail surfaces. Most of the multibody dynamics softwares like GENSYS [32], and SIMPACK [33] utilize look-up tables for this specific purpose. However, to have more accurate wear estimation, a theoretical approach known as the elastic contact formulation using algebraic equations is also considered here. This method is described in the next subsection.

Once the creepages attained, one can compute the respective contact forces through a proper contact theory. In **PAPER A**, the nonlinear heuristic theory is employed which works based on the linear Kalker's theory. One of most efficient procedures in this field is FASTSIM algorithm and is utilized in **PAPERS B**, **C**, and **D** of this thesis. It should be noted that Polach contact theory is also one of the most well-known algorithms in this field [34, 35]. Finally, the wear objective function (Γ_W) is defined as:

$$\Gamma_W = \sqrt{\frac{1}{t_f - t_0}\int_{t_0}^{t_f} \left(\left|F_{\xi}\upsilon_{\xi}\right| + \left|F_{\eta}\upsilon_{\eta}\right| + \left|M_{\xi\eta}\phi_{\xi\eta}\right|\right)^2 dt} \ , \tag{16}$$

3.3.3 Contact search problem

As aforementioned, the contact point positions on the wheel and rail surfaces play an important role in evaluation of the wheelset dynamics behavior; in particular the contact creepages, contact forces, and as a result wear. Therefore, it is vital to consider the contact search problem in advance to determine the contact point position on wheel and rail during the operation. Shabana *et al*. developed four main theoretical approaches suitable for computer implementation to solve the contact search problem in railway application [31, 36]. All these methods work based on the parameterization of the wheel and rail surfaces.

The elastic contact formulation using algebraic equations (ECF-A) approach is chosen for the analysis here. The first step is to describe the wheel, and rail geometries based on the surface parameters. This makes it easier to express the contact point positions in different coordinate systems. Therefore, four surface parameters $\mathbf{s}=[s_1^r \quad s_2^r \quad s_1^w \quad s_2^w]^T$ are introduced as shown in Fig. 3.

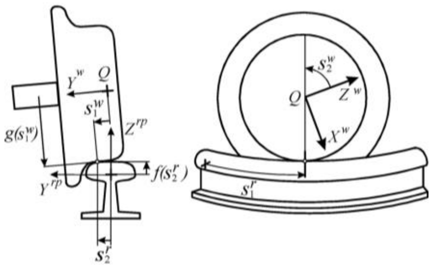

Fig. 3: Wheel and rail surface parameters [36].

s_1^r, s_2^r are the rail surface parameters. Indeed, s_1^r indicates the rail arch length, s_2^r denotes the lateral position of the contact point and $f(s_2^r)$ is the rail profile function which determines the vertical position of the contact point with respect to the rail profile coordinate system (X^{rp},Y^{rp},Z^{rp}). Similarly, s_1^w, s_2^w are the wheel surface parameters that determine the lateral and angular position of the wheel contact point with respect to the wheel local coordinate system (X^w,Y^w,Z^w), and $g(s_1^w)$ is the wheel profile function. A useful aspect of such formulation is that the effects of different wheel and rail profiles on the contact point positions can be easily taken into account by using the measurement data as the wheel and rail profile functions.

The coordinate systems required to specify the track geometry are shown in Fig. 4. Therefore, the contact point position on the rail surface can be expressed in global coordinate system as follows:

$$\mathbf{r}^r=\mathbf{R}^r+\mathbf{A}^r(\mathbf{R}^{rp}+\mathbf{A}^{rp}\bar{\mathbf{u}}^{rp}), \tag{17}$$

where, $\mathbf{R}^r$ and $\mathbf{A}^r$ are the position vector and transformation matrix that determine the position and orientation of the rail reference coordinate system with respect to the global coordinate system, respectively. In a similar manner, the position and orientation of the rail profile coordinate system with respect to the rail reference coordinate are expressed using $\mathbf{R}^{rp}$

(position vector) and $\mathbf{A}^{rp}$ (transformation matrix), respectively. $\bar{\mathbf{u}}^{rp}$ is the contact point position in the rail local coordinate system given by Eq. (18), see Fig. 3.

$$\bar{\mathbf{u}}^{rp} = \begin{bmatrix} 0 & s_2^r & f(s_2^r) \end{bmatrix}^T \tag{18}$$

It should be noted that for each specific value of the rail arch length s_1^r, the position and orientation of the profile coordinate system can be uniquely determined. In other words, the position and orientation of the rail profile coordinate system are merely a function of the track arch length. As a result, a track pre-processor could be utilized for online evaluation of the profile coordinate position vector and orientation matrix on each time step.

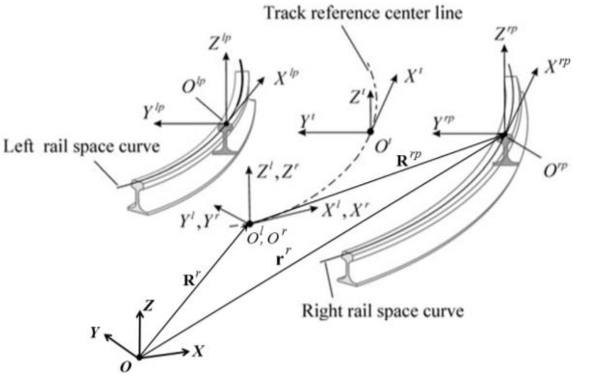

Fig. 4: Track coordinate systems [31].

The coordinates required to specify the wheelset geometry are shown in Fig. 5. Similar to the previous case, the position of an arbitrary point on the wheel profile with respect to the global coordinate system is expressed as:

$$\mathbf{r}^w = \mathbf{R}^w + \mathbf{A}^w \bar{\mathbf{u}}^w , \tag{19}$$

where, $\mathbf{R}^w$ and $\mathbf{A}^w$ determine the position vector and orientation matrix of the wheelset coordinate system (X^w,Y^w,Z^w) with respect to the global coordinate system.

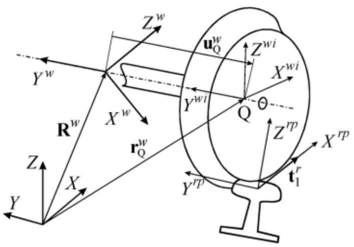

Fig. 5: wheel coordinate systems [36].

$\bar{\mathbf{u}}^w$ is the position vector of an arbitrary point on the wheel surface towards the wheelset coordinate system. For example, one could have the following for the left hand side wheel:

$$\bar{\mathbf{u}}^w = \left[g\left(s_1^w\right)\sin s_2^w \quad s_1^w + L \quad -g\left(s_1^w\right)\cos s_2^w \right]^T , \tag{20}$$

where, L is the distance between the wheelset coordinate system and the origin of the wheel local coordinate system (point Q in Fig. 5.).

Euler angels can be utilized to attain the transformation matrix requires to determine the orientation of a coordinate system with respect to a reference frame, see Eqs. (17, 19). As an example, the transformation matrix (A^w) associated with the wheelset coordinate system in Eq. (19) is given by Eq. (21) for the following rotation order: 1) a yaw rotation (ψ) around Z^w; 2) a roll rotation (ϕ) about X^w; 3) a pitch rotation (θ) around Y^w, see e.g. [31] for more details.

$$A^w = \begin{bmatrix} \cos\psi\cos\theta - \sin\psi\sin\phi\sin\theta & -\sin\psi\cos\phi & \cos\psi\sin\theta + \sin\psi\sin\phi\cos\theta \\ \sin\psi\cos\theta + \cos\psi\sin\phi\sin\theta & \cos\psi\cos\phi & \sin\psi\sin\theta - \cos\psi\sin\phi\cos\theta \\ -\cos\phi\sin\theta & \sin\phi & \cos\phi\cos\theta \end{bmatrix}, \tag{21}$$

Similar approach can be used to evaluate the transformation matrices correlated to the rail coordinate system. It is clear that the global position vector of the wheel and rail contact points can be expressed in terms of the four surface parameters described earlier. Based on the ECF-A method, for potential contact point candidates, the following conditions have to be satisfied [31, 36-38]:

$$\left.\begin{aligned} \mathbf{t}_1^r \cdot \mathbf{r}^{wr} &= 0 \\ \mathbf{t}_2^r \cdot \mathbf{r}^{wr} &= 0 \\ \mathbf{t}_1^w \cdot \mathbf{n}^r &= 0 \\ \mathbf{t}_2^w \cdot \mathbf{n}^r &= 0 \end{aligned}\right\} , \tag{22}$$

where, $\mathbf{r}^{wr}=\mathbf{r}^w-\mathbf{r}^r$ is the displacement vector between the points on wheel and rail surfaces that might come into contact, $\mathbf{t}_1^w$, $\mathbf{t}_2^w$ are the tangential vectors to the wheel profile at the contact point. The physical interpretation of the four algebraic equations (22) can be stated as follows: in order to two points on the wheel and rail profiles come into contact, the tangential vectors of the rail ($\mathbf{t}_1^r$, $\mathbf{t}_2^r$), must be perpendicular to the vector that connects those two points ($\mathbf{r}^{wr}$). In addition, the tangential vectors of the wheel profile at the wheel contact point ($\mathbf{t}_1^w$, $\mathbf{t}_2^w$), must be perpendicular to the normal vector acting on the rail surface at the rail contact point ($\mathbf{n}^r$).

The system of nonlinear algebraic equations (22) can be solved using the numerical methods such as Newton-Raphson to yield the four unknown surface parameters, i.e. $\mathbf{s} = [s_1^r \quad s_2^r \quad s_1^w \quad s_2^w]^T$ that are necessary to evaluate the wheelset dynamics.

Finally, the penetration condition given by Eq. (23) must be checked to see if the two potential contact points obtained are in contact or not, see e.g. [37] for more details.

$$\delta = \mathbf{r}^{wr} \cdot \mathbf{n}^r \leq 0 \tag{23}$$

where, δ is known as indentation. It should be noted that however the wheel and rail profiles are rigid, in contact search problem using the ECF-A approach the two bodies are allowed to penetrate. Once δ and the contact point positions achieved, the normal contact force and the dimensions of the contact ellipse can be obtained using the Hertzian contact theory [31].

3.3.4 Wear post-processor

The computer implementation of the contact search problem introduced in the previous section as well as wear calculation is described in this section. The explicit expression for the global velocity vector of the contact point is given by Eq. (24), see e.g. [31] for more details. In which, a_k, and r_k are the lateral position of the contact point and the instantaneous rolling radius of the wheel, respectively. K=R,L denotes the right or left hand side wheel.

$$\dot{\mathbf{r}}^w = \dot{\mathbf{R}}^w + \begin{bmatrix} -\dot{\psi}(a_k\cos\phi + r_k\sin\phi)\cos\psi - \dot{\theta}r_k\cos\psi - \dot{\phi}(r_k\cos\phi - a_k\sin\phi)\sin\psi \\ -\dot{\psi}(a_k\cos\phi + r_k\sin\phi)\sin\psi - \dot{\theta}r_k\sin\psi - \dot{\phi}(r_k\cos\phi - a_k\sin\phi)\cos\psi \\ \dot{\phi}(r_k\sin\phi + a_k\cos\phi) \end{bmatrix} \tag{24}$$

In a trajectory coordinate system (that is widely used in multibody dynamics formulation in railway applications), the position of wheelset coordinate is described as follows:

$$\mathbf{q} = \begin{bmatrix} s & y & z & \psi & \phi & \theta \end{bmatrix}^T , \tag{25}$$

where, s is the arch length coordinate along the track, y and z are the lateral and vertical displacements of the center of mass, ψ, ϕ, and θ are the yaw, roll, and pitch angles, respectively. These parameters together with the corresponding time derivatives can be extracted from the vehicle dynamics trough the time integration process. The spin creepages for the right and left wheels are given by the following relations [31]:

$$\left.\begin{aligned} \phi_{\xi\eta}^R &= \frac{1}{V}\left\{\left(\dot{\theta} + \dot{\psi}\sin\phi\right)\sin\delta_R + \left(\dot{\psi}\cos\phi\right)\cos\delta_R\right\} \\ \phi_{\xi\eta}^L &= \frac{1}{V}\left\{-\left(\dot{\theta} + \dot{\psi}\sin\phi\right)\sin\delta_L + \left(\dot{\psi}\cos\phi\right)\cos\delta_L\right\} \end{aligned}\right\}, \tag{26}$$

where, δ_R and δ_L denote the right and left contact angles, respectively. Consequently, the contact search procedure described earlier together with Eqs. (15,24,26) are required to calculate the creepages.

It is unfeasible to accomplish the prescribed contact search formulation during the time integration process in SIMPACK. Therefore, wear calculation is implemented in a post-processor stage as shown in Fig. 6.

The wear post-processor works as follows: the SIMPACK solver provides the dynamics response of the system ($\mathbf{q},\dot{\mathbf{q}}$) that can be stored on a file. Based on the wheel and rail profiles and the track geometry, one can construct the system of nonlinear algebraic equations (22). Then, a proper initial guess for the four unknown surface parameters $\mathbf{s} = [s_1^r \quad s_2^r \quad s_1^w \quad s_2^w]^T$ should be considered.

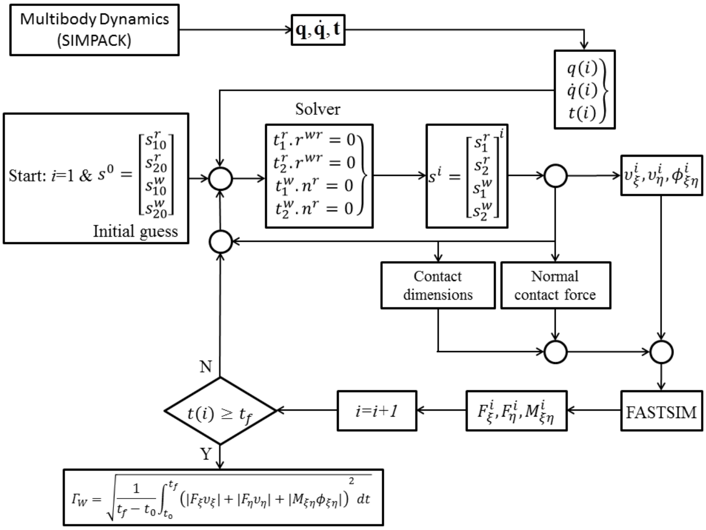

Fig. 6: Wear post-processor algorithm.

The Newton-Raphson method is employed to solve the system of equations (22). Therefore, it is necessary to evaluate the Jacobian of the system of equations. In order to have more computationally efficient algorithm, the solver algorithm is implemented in FORTRAN. The solution of the contact search problem is used as the initial guess for the next iteration. The dynamics response of the system and the surface parameters are used to evaluate the creepages by means of Eqs. (15,24,26). The normal contact force, and the dimension of the contact ellipse, can either be obtained using the SIMPACK output, or using the Hertzian contact theory. FASTSIM algorithm is then utilized to attain the contact forces and moment. This process is repeated until the end of the time signal. Finally, the wear objective function is calculated using Eq. (16). The wear post-processing stage described here is used in **PAPER D** of this thesis.

4 PROBLEM FORMULATION AND METHODOLOGY

As discussed earlier, the ultimate goal of the current thesis is to contribute with knowledge and approaches that lead to adaptive bogie design. In this regard, a brief overview on the problem formulation and methodology for optimization, semi-active control and global sensitivity analysis of bogie suspension components considered in **PAPERS A-D** of this thesis is given in this section.

4.1 Optimization

In general, the optimization problem of m design parameters $\mathbf{d}=[d_1,d_2,...d_m]^T\in\mathbf{\Omega}$ (where, $\mathbf{\Omega}$ is the domain of design variables) with respect to a vector of objective functions $\mathbf{\Gamma}=\mathcal{F}(\mathbf{d})$, that is evaluated from the vehicle's dynamics response, can be states as:

$$
\left.\begin{array}{c}
\mathcal{F}(\mathbf{d}^*,\mathbf{x}^*(\mathrm{t}))=\min\mathcal{F}(\mathbf{d},\mathbf{x}(\mathrm{t})),\ \mathbf{d}^*\in\mathbf{\Omega} \\
\\
\Gamma_1 \le \Gamma_1^{\max} \\
\Gamma_2 \le \Gamma_2^{\max} \\
. \\
. \\
\Gamma_n \le \Gamma_n^{\max}
\end{array}\right\}, \tag{27}
$$

in which, $\Gamma_j^{\max}$, $j=1,2,\ldots,n$ denote the threshold values, $\mathbf{x}$(t) is the state vector of the railway vehicle estimated by using the computational model. Therefore, based on the objective functions introduced in section 3, and suspension system components several multiobjective optimization problems can be formulated. Genetic algorithm based multiobjective optimization routine in MATLAB is utilized to solve the optimization problems specified in **PAPERS A-D**. The procedure can be described as follows: on each iteration, MATLAB updates the design parameters file as an input to the vehicle model developed in SIMPACK or MATLAB, the dynamic response of the system is then obtained using the time integration solvers and the respective objective functions are attained using a post-processing stage and the threshold conditions are checked to make sure if all the objective functions are within the admissible limit. This procedure continues until convergence or the maximum number of generations achieved. In the case of multiobjective optimization problems, the results can be plotted in terms of Pareto-set and Pareto-front graphs as shown in **PAPER A**.

4.2 Semi-active control

As an introduction to active control of railway vehicles that is an important stage in adaptive bogie design, application of the on/off semi-active control strategies in bogie primary suspension is considered in **PAPER C**. In this regard, instead of the conventional passive dampers, MR dampers are integrated with the vehicle dynamics through the SIMPACK-SIMULINK co-simulation environment to apply the control force to the system as shown in Fig. 7.

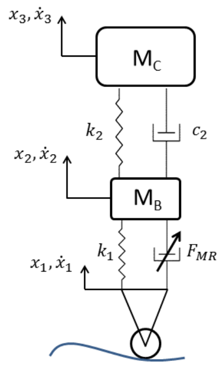

Fig. 7: Application of MR dampers in bogie primary suspension.

The MR damper control current is decided based on the control strategy. One of the most popular on/off switching control laws (skyhook) is defined as:

$$\begin{cases} F_{MR}(I_{\max}) & \dot{x}_2\left(\dot{x}_2-\dot{x}_1\right)\geq 0 \\ F_{MR}(I_{\min}) & \dot{x}_2\left(\dot{x}_2-\dot{x}_1\right)< 0 \end{cases}, \tag{28}$$

where, $\dot{x}_2$, and $\dot{x}_1$ are the velocity of the second and first carts (bogie frame, and axle box in this case), respectively. Based on various combinations of displacement and velocity of the first and second cart, several semi-active control laws can be introduced. The efficiency of such on/off switching control strategies in wear reduction of a railway vehicle running on different operational scenarios is subject to study in **PAPER C**. The SIMPACK-SIMULINK co-simulation environment can be used as the main tool in exploring the application of different active control techniques in the subsequent studies of this project.

4.3 Global sensitivity analysis

Sensitivity analysis is one of the most prominent steps in design and optimization of bogies that can provide informative design insights. In this section, some basic concepts on the global sensitivity analysis formulation that is the basis of **PAPER D** are given. In general, different objective functions specified in section 3 can be expressed as functions of a set of *m* independent random variables, *i.e.* design parameters $\mathbf{d}=[d_1,d_2,...d_m]^T\in\mathbf{\Omega}$, through the respective deterministic functional relationship $\Gamma=\mathscr{F}(\mathbf{d})$. The mean ($\mu$) and variance ($V$) of Γ are defined as [8]:

$$\begin{cases} \mu_\Gamma = E_{\mathbf{d}}[\Gamma]=\int_{\mathbf{d}}\mathscr{F}(\mathbf{d})\,f_{\mathbf{d}}(\mathbf{d})\delta\mathbf{d} \\ V_\Gamma = E_{\mathbf{d}}[(\Gamma-\mu_\Gamma)^2]=E_{\mathbf{d}}\{[\mathscr{F}(\mathbf{d})]^2\}-\mu_\Gamma^2 \end{cases}, \tag{29}$$

where, $E[.]$ is the expectation operator, and $f_{\mathbf{d}}(\mathbf{d})$ is the joint density of $\mathbf{d}$. Assume that $\mathbf{d}_{-i}$ is a m-1 dimensional sub-vector of $\mathbf{d}$, in which contains all the elements of $\mathbf{d}$ except d_i. Therefore, one can define the following conditional expectation:

$$E_{-i}[\Gamma\mid d_i]=\int_{\mathbf{d}_{-i}}\mathscr{F}(\mathbf{d}_{-i},d_i)\,f_{\mathbf{d}_{-i}}(\mathbf{d}_{-i})\delta\mathbf{d}_{-i} \tag{30}$$

The primary (S_i) and total (S_{Ti}) sensitivity indices are defined as follows, see e.g. [8, 39, 40]:

$$S_i=\frac{V_i[E_{-i}(\Gamma\mid d_i)]}{V_\Gamma},\ S_{Ti}=\frac{E_{-i}[V_i(\Gamma\mid \mathbf{d}_{-i})]}{V_\Gamma} \tag{31}$$

It is clear that in order to achieve the global sensitivity indices, multilayer integrals have to be evaluated. This process requires a heavy computational effort. Therefore, it is vital to apply an efficient algorithm to increase the computational proficiency. The M-DRM method can approximate the global sensitivity indices in an efficient and accurate manner and is employed in **PAPER D** to solve the global sensitivity problem of bogie dynamics with respect to suspension components.

5 SUMMARY OF APPENDED PAPERS

5.1 Paper A

The comfort/safety multiobjective optimization problem of lateral primary and secondary dampers of a one car vehicle model with 26 DOFs developed in MATLAB is considered in this paper. The main target is to examine the applicability and computational efficiency of the genetic algorithm based multiobjective optimization routines developed in MATLAB on a simple vehicle model. The contact points are estimated based on the conical wheel profiles and two DOFs motion of the wheelsets. The results obtained from this paper showed that genetic algorithm can be considered as a useful tool in solving different multiobjective optimization problems formulated in the subsequent parts of this project. Furthermore, lateral primary and secondary dampers can affect the comfort and safety during the operation and as a results semi-active switching or fully active control techniques can be applied to improve the vehicle's dynamics performance from those perspectives.

5.2 Paper B

In order to investigate the effects of primary dampers on wear, a single wheelset with five DOFs attached to a fixed bogie frame is subject to study in this paper. A more advanced wheel/rail contact formulation is considered here. Actual wheel and rail profiles are used and the elastic contact formulation is employed to attain the contact point locations. The target is to formulate and solve the wear optimization of the primary dampers on different operational scenarios, while safety and comfort objectives are taken as thresholds. The same optimization routine used in **PAPER A** is considered for the analysis. The results of this paper showed that to some extents, wear is sensitive towards the primary dampers. As a result using on/off switching semi-active control or fully active control techniques in bogie primary suspension might reduce wear on different operational scenarios.

5.3 Paper C

As a continuation of **PAPER B** and in order to check the applicability and efficiency of semi-active control strategies in bogie primary suspension a one car vehicle model developed in multibody dynamics software SIMPACK is considered for the analysis. The vehicle navigates on different operational scenarios including tangent and curved tracks with different radius of curvatures. The primary springs and dampers are optimized with respect to wear on different operational scenarios (while safety and comfort are considered as thresholds) to yield a suitable reference model for comparison the efficiency of different semi-active control strategies. A proper mathematical model for magnetorheological dampers (MR) is implemented in MATLAB-SIMULINK and 13 different on/off switching control strategies are considered for the analysis. All the primary longitudinal dampers are replaced by MR dampers and the efficiency of the prescribed semi-active controls in wear reduction is explored. The same procedure is followed for the primary dampers in the lateral and vertical directions. Finally, the results for two control strategies that work based on the displacement and relative velocity of the bogie frame and wheelset showed up to 20% wear reduction on the most critical operational scenario with very small radius of curvature.

5.4 Paper D

As preliminary stage in adaptive bogie design, it is necessary to study the influence of different primary and secondary suspension components on vehicle's dynamics on various operational scenarios. Such an analysis narrows down the number of input variables for optimization and design problem of adaptive bogie systems and increases the computational efficiency. In this regard the global sensitivity analysis problem of bogie dynamics behavior towards suspension components is considered in **PAPER D**. A one car vehicle model with passive primary and secondary suspension components developed in multibody dynamics SIMPACK is considered for the analysis. The vehicle runs on different operational scenarios with various radius of curvature of the track ranging from very small radius curves up to tangent track. For each particular scenario, measurement data are applied to the model as the track irregularities and the vehicle runs at maximum admissible speed on each case. The multiplicative version of the dimension reduction method (M-DRM) together with Gaussian quadrature integrals is an efficient method for global sensitivity analysis. The same approach is followed in this paper to solve the global sensitivity analysis problem of the bogie system. Wear, ride comfort, and safety are considered as the objective functions and the effects of each particular suspension elements on these functions are investigated thoroughly for two configurations of the suspension system. The results of this paper provide informative data that can be used in optimization and design of adaptive bogies.

6 CONCLUDING REMARKS AND FUTURE WORK

This thesis deals with some of the mathematical formulation and computational algorithms required for adaptive design of bogie suspension. In this regard, four main issues are considered:

- First, it is significant to choose a proper model. This includes the engineering, mathematical and computational models required for the analysis. Therefore, hierarchical levels of engineering models are scrutinized to detect the most convenient models for different problems considered in this thesis. Furthermore, proper definition and evaluation of different objective functions were other important issues that have been explored thoroughly in this thesis.
- Second important task was to formulate and solve several optimization problems. This is not only the basis for the analysis of passive suspension systems, but also required for the active control of bogies as well as sensitivity analysis formulation of vehicle's dynamics behavior. In this regard, the genetic algorithm based multiobjective optimization routines in MATLAB have been selected to solve the optimization problems formulated in different parts of this thesis.
- Third, as an introduction in active control of bogie system, application of MR dampers in on/off switching semi-active vibration control of primary dampers of a one car vehicle has been considered. This study showed that different semi-active vibration control techniques can be integrated with different vehicle models developed in SIMPACK through the algorithms implemented in SIMPACK-SIMULINK co-simulations environment.
- Finally, the global sensitivity analysis problem of bogie dynamics with respect to the suspension stiffness and damping components has been considered as one of the key

stages in adaptive bogie design. In this regard, a reliable and computationally efficient algorithm has been used for the analysis. This analysis can give beneficial insights into formulating and solving different optimization and design problems for passive and active suspension components of bogie systems. Global sensitivity analysis results obtained in this thesis can reduce the design and computational costs associated with adaptive bogie design.

The future steps of the current research can be laid within the following three main subjects:

- Formulating and solving different global sensitivity analysis and multiobjective optimization problems to investigate the interaction between railway vehicle speed, track irregularities and structural parameters that could be used in adaptive bogie design.
- Advances in flexible multibody dynamics formulation of the vehicle model can also be an interesting problem that might lead to more accurate results especially on wear, and ride comfort.
- Application of semi and/or fully active control systems in bogie suspension to have adaptive railway vehicle.

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PART II

APPENDED PAPERS A-D

PAPER A: Multiobjective optimization of a railway vehicle dampers using genetic algorithm, *Proceedings of the ASME 2013 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference IDETC/CIE 2013*, August 4-7, Portland, Oregon, USA, 2013; doi: 10.1115/DETC2013-12988

PAPER B: Optimization of a bogie primary suspension damping to reduce wear in railway operations, *Proceeding of the ECCOMAS Thematic Conference on Multibody Dynamics*, July 1-4, Zagreb, Croatia, 2013; http://eccomas.fsb.hr/proceedings.php

PAPER C: Application of semi-active control strategies in bogie primary suspension system, *Proceeding of the 2nd International Conference on Railways,* April 8-11, Ajaccio, France, 2014; doi: 10.4203/ccp.104.318

PAPER D: Global sensitivity analysis of bogie dynamics towards suspension components, *submitted for international publications.*