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Multiobjective Optimisation and Active Control of Bogie Suspension

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Cover:
Bogie suspension system of a one-car railway vehicle model developed in the multibody dynamics software SIMPACK.

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ABSTRACT

Railways provide fast, safe, clean, and cheap transportation service. The cost efficiency in railway operations can be scrutinized from different perspectives. Here, passenger ride comfort, wheel/rail contact wear, and safety (in particular running stability, track shift force, and risk of derailment) are considered as objective functions introduced to evaluate the dynamics behaviour of railway vehicles. Running speed also plays a key role in cost efficiency of railway operations. Higher speeds shorten journey time and make railways more competitive with other types of transportation systems. However, this might increase wear and deteriorate ride comfort and safety. To improve the performance in railway operations advanced designs and technologies are developed during the past decades. Bogie primary and secondary suspension systems of high speed trains can significantly affect the dynamics behaviour of the vehicle. Such components might have conflicting effects on different objective functions. It is important to have the optimum performance of suspension components. In this regard, one of the ultimate goals of this thesis is to improve the vehicle performance from different points of views by studying passive and active suspension systems and using multiobjective optimisation techniques to meet conflicting design requirements. Computational cost is one of the main challenges in multidisciplinary design optimisation. The computational efforts for optimisation can be significantly mitigated by narrowing down the number of input design parameters. Here, an efficient global sensitivity analysis is carried out to identify those suspension components that have prominent influences on different objective functions. Based on the global sensitivity analysis results obtained two multiobjective optimisation problems are formulated and solved. First, multiobjective optimisation of bogie suspension components with respect to safety to improve running speed on curves. Second problem is to reduce wear and improve ride comfort when the vehicle is operating with the enhanced speeds. Consequently, the vehicle runs secure and faster with higher ride comfort and less wear by means of the two optimisation problems solved. The optimisations are carried out using the genetic algorithm. In the case of safety optimisation problem, semi active control strategies are also applied using magnetorheological dampers and the effects on the dynamics behaviour are explored. The robustness of the bogie suspension Pareto optimised solutions against uncertainties in the design parameters is also studied. Active control technology is one of the main targets of this thesis. In this regard, a robust controller is designed using the H_∞ control technique to stabilize the wheel set motion and improve curving performance. The controller is robust against track irregularities. Finally, the actuator dynamics is considered and a compensation technique is applied to reduce the actuator's time delay and improve the performance.

KEYWORDS: Bogie, suspension system, global sensitivity analysis, multiobjective optimisation, robustness analysis, active control.

To my beloved family

PREFACE

This work has been accomplished during November 2011 until September 2016 at the department of Applied Mechanics, Chalmers University of Technology, Gothenburg, Sweden. The current PhD thesis is part of the SD9 project defined at the Chalmers railway mechanics center of excellence (CHARMEC).

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Gothenburg, September 2016

Seyed Milad Mousavi Bideleh

LIST OF PUBLICATIONS

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

Paper A Mousavi Bideleh, S.M. and Berbyuk, V., Global sensitivity analysis of bogie dynamics with respect to suspension components, *Multibody System Dynamics*, 2016, **37**(2): pp. 145-174, doi: 10.1007/s11044-015-9497-0.

Paper B Mousavi-Bideleh, S.M. and Berbyuk, V., Multiobjective optimisation of bogie suspension to boost speed on curves, *Vehicle System Dynamics*, 2016, **54**(1): pp. 58-85, doi: 10.1080/00423114.2015.1114655.

Paper C Mousavi Bideleh, S.M., Berbyuk, V., and Persson, R., Wear/comfort Pareto optimisation of bogie suspension, *Vehicle System Dynamics*, 2016, **54**(8): pp. 1053-1076, doi: 10.1080/00423114.2016.1180405.

Paper D Mousavi Bideleh, S.M., Robustness analysis of bogie suspension components Pareto optimised values, *To be submitted for international publications*.

Paper E Mousavi Bideleh, S.M., Mei, T.X., and Berbyuk, V., Robust control and actuator dynamics compensation for railway vehicles, *Accepted for publication in Vehicle System Dynamics,* 2016.

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

In addition to papers A-E the following conference papers have also been part of the research in this PhD project which are not included in this thesis:

1. Mousavi Bideleh, S.M. and Berbyuk, V. Global sensitivity analysis and multiobjective optimization of bogie suspension, in *The 24th International Congress of Theoretical and Applied Mechanics*, 21-26 August 2016, Montreal, Canada.
2. Mousavi Bideleh, S.M. and Berbyuk, V. Variance-based wheel/rail contact sensitivity analysis in respect of wheel set dynamics, in *The ASME 11th International Conference on Multibody Systems, Nonlinear Dynamics, and Control, Paper DETC2015-47342*. 2-5 August 2015. Boston, Massachusetts, USA. doi: 10.1115/DETC2015-47342.
3. Mousavi Bideleh, S.M. and Berbyuk, V. Application of semi-active control strategies in bogie primary suspension system, in *The Second International Conference on Railway Technology: Research, Development and Maintenance*. Vol 104, 2014. Ajaccio, France. Published in Stirlingshire, United Kingdom, paper 318: Civil-Comp Press, doi: 10.4203/ccp.104.318.
4. Mousavi Bideleh, S.M. and Berbyuk, V. Multiobjective optimization of a railway vehicle dampers using genetic algorithm, in the ASME 2013 International Design Engineering Technical Conferences & Computers and Information in Engineering

Conference IDETC/CIE, Paper DETC2013-12988. 4-7 August 2013. Portland, Oregon, USA. doi: 10.1115/DETC2013-12988.
5. Mousavi Bideleh, S.M. and Berbyuk, V. Optimization of a bogie primary suspension damping to reduce wear in railway operations, in *The ECCOMAS Multibody Dynamics, pp. 1025-1034*. 1-4 July 2013. University of Zagreb, Croatia.

CONTENTS

PART I

EXTENDED SUMMARY

The general outlines of the thesis are described in this section. After a brief overview of the main aims of the project, vehicle models developed for the analysis and objective functions defined to reflect the vehicle's dynamics performance are introduced. An overview on problem formulation and methodologies used to carry out global sensitivity analysis, multiobjective optimisation, robustness analysis and active control is given. The summary ends up with concluding remarks and outlines of future research.

1 INTRODUCTION

Railways are one of the most significant means of transportation. Freight and passenger trains provide reasonably cheap, clean, and fast service which make the railways highly competitive with other types of transport systems such as ground, marine and air services.

During the past few decades demands on improving the cost efficiency in railway operations has been increased. Bogie system of high speed trains transmits forces between the vehicle and the track. In this regard, bogie suspension design can significantly affect the vehicle dynamics behaviour. As a result, with increasing requirements on the performance of railway vehicles, the demands on the bogie suspension system improvement are also increased.

The cost efficiency in railway operations can be scrutinized from different perspectives. Maintenance cost, track access charges and design requirements due to the passenger ride comfort and safety improvement as well as wheel/rail contact wear reduction 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 (if those are set on the track occupation time). 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 set to the maximum admissible value corresponding to each particular operational scenario.

Here, one of the most significant challenges is that the suspension system components have conflicting effects on different objective functions, see Fig. 1. For instance, a new suspension design with the aim of improving ride comfort might reduce the vehicle safety. Such conflicting demands on the bogie suspension system make it difficult to meet different design requirements by means of optimisation of bogie suspension components with respect to a single objective function. As a result, multiobjective design optimisation problems for passive suspension elements have to be carried out or active control techniques should be employed.

As aforementioned, one of the common ways to make the vehicle's performance more efficient is to improve the bogie primary and secondary suspension components. In this regard, passive, semi active, and active suspension components are developed, see e.g. [1-9]. Each of these systems has some advantages and disadvantages. Passive components are quite simple, reliable, cheap, and easy to use. However, a limited performance can be achieved by means of the passive components. On the other hand, semi active and active systems can adjust the bogie dynamics with respect to different operational scenarios and provide higher performance. However, the design and maintenance are more costly, additional power supply is needed, and there is a higher risk of failure.

The ultimate goal of this thesis is to study the multiobjective optimisation and active control of bogie suspension components to be able to improve the vehicle performance with respect to the objective functions shown in Fig. 1.

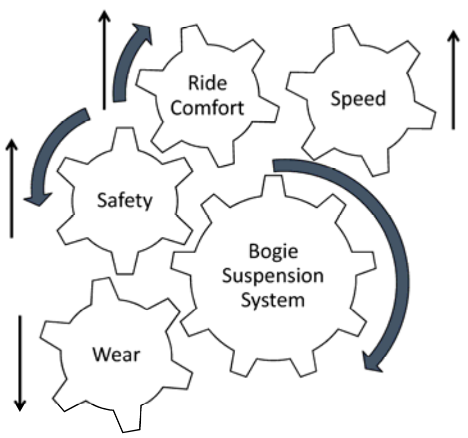

Fig. 1: Conflicting effects of bogie suspension system on vehicle performance.

One of the early stages in this project is to develop a suitable vehicle model for the multiobjective optimisation and active control design problems of the bogie suspension system. Some aspects have to be addressed when developing a vehicle model for such analysis. The model should not be too simple as it might fail to represent the realistic dynamics behaviour and it also should not be too complicated as it would be difficult and time costly to study. Here, a one-car railway vehicle model with 50 degrees of freedom (DOFs) developed in the multibody dynamics software SIMPACK is chosen for the analysis. The structural parameters used to develop the vehicle model are provided by the Bombardier Transportation, Västerås, Sweden. Since the abovementioned vehicle model is too complicated for the active control design, a simple half-car railway vehicle model with 7 DOFs is also considered in this thesis for that particular purpose.

The bogie system of high speed trains is a complex nonlinear mechanical system with highly interconnected elements. It is important to study the effects of different bogie suspension components on the vehicle's dynamics behaviour. In the case of passive suspension, it is necessary to formulate and solve several multiobjective optimisation problems to be able to attain the trade-off solutions that provide the optimised suspension characteristics.

In this regard, several researches on the multibody systems optimisation are done. Eberhard et al. proposed an efficient optimisation algorithm with application in multibody dynamics analysis using a deterministic gradient-based as well as stochastic methods [10]. Switching between those algorithms during the optimisation can help to find the global minima in a more efficient way. The genetic algorithm (GA) based multiobjective optimisation routines are widely-used in railway applications, see e.g. [11-17]. Johnsson et al. considered the multiobjective optimisation problem of bogie suspension components with respect to safety and ride comfort [18]. 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 optimisation routines. The same approach is followed here to solve the optimisation problems formulated in **papers A-C**.

In the case of the passive suspension, it is vital to utilise the optimised values of design parameters to achieve satisfactory level of improvement in the dynamics behaviour. However, optimisation is usually a time costly process especially when it comes to the complex systems with large number of DOFs and design variables. Consequently, smart selection of design parameters can significantly attenuate the computational efficiency of the optimisation. 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. [19-21]. However, the efficiency of such approaches highly depends on the initial values considered for the analysis and there are some uncertainties regarding 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 the global sensitivity analysis. However, the computational effort for the 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 efficient algorithm for that particular purpose. Based on the multiplicative dimension reduction method (M-DRM), Zhang et al. proposed a closed form solution for the global sensitivity indices [22]. This method can dramatically reduce the computational efforts required for the global sensitivity analysis in comparison with the Monte Carlo simulation, while the results obtained are within the same order of accuracy. Similar methodology is applied in this thesis to solve the global sensitivity analysis problem of bogie dynamics with respect to the suspension components. Based on the results achieved from such an analysis, one can identify 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 design optimisation of bogie suspension which can significantly reduce the computational efforts in simulations. This is the subject of study in **paper A**.

Application of the semi active and/or active components in bogie suspension is another important issue in bogie suspension design. Such components are mostly used in the secondary suspension system and many researches are done to investigate the effects of active suspension technology on ride comfort of passenger trains, see e.g. [23, 24]. 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 study the effects in the bogie dynamics behaviour with respect to the corresponding passive system. This is done by means of a set of sensors and actuators and a proper control strategy. Magnetorheological (MR) dampers and electromechanical actuators have a simple construction and are widely-used in different applications. Integration of such dampers and actuators with passive elements and respective effects on the dynamics behaviour of the vehicle is subject to study in this project, see **papers B** and **E**, respectively.

All in all, global sensitivity analysis, multiobjective optimisation, and active 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 which reflect the vehicle's dynamics performance on different operational scenarios that are introduced in mathematical terms in the following sections. Finally, a brief introduction about the methodologies used to solve various problems is also given.

The general outlines of this PhD study are:

1. To perform the global sensitivity analysis of different objective functions with respect to the bogie suspension components. Such an analysis helps to identify the important design parameters and provides a basis to solve the bogie suspension system multiobjective optimisation problems in a computationally efficient framework.
2. To improve safety and as a result running speed on curves by solving the multiobjective optimisation problem of bogie suspension with respect to different safety objective functions.
3. To improve passenger ride comfort and reduce wear when the vehicle runs with the enhanced high speeds by solving the wear/comfort Pareto optimisation problem of bogie suspension.
4. To carry out the robustness analysis of the Pareto optimised solutions achieved by the two multiobjective optimisation problems and study the effect of parameter uncertainties on the dynamics behaviour of the vehicle.
5. To design a robust controller for bogie primary suspension system using practical sensors and actuators.

2 VEHICLE MODELS

As a preliminary stage in formulating the global sensitivity analysis, multiobjective optimisation, and active control problems, it is necessary to choose a vehicle model with an appropriate level of complexity to carry out the analysis. As aforementioned, the vehicle model should neither be too simple as it might fail to reflect the realistic vehicle dynamics behaviour nor be too complicated as it might make the problem computationally inefficient. In the following, two types of the vehicle models used in this thesis are introduced.

2.1 Vehicle model A

A generic one-car railway vehicle model developed in the multibody dynamics software SIMPACK is chosen for the analysis as shown in Fig. 2. The model consists of a carbody, two bogie frames, four wheel sets, and eight axle boxes. All these body elements are rigid and have 6 DOFs except for axle boxes which merely allow a rotation around the wheel set axes. As a result, the model has a total 50 DOFs.

The Hertzian contact theory is employed to evaluate the normal contact force. Look up tables might be used for the contact search problem, see e.g. [25]. However, an equivalent-elastic contact search approach in SIMPACK is chosen. This method takes the intersection of the wheel and rail profiles and approximates each continuous intersection area by an equivalent contact ellipse. Indeed, the shape of the contact patch is converted into an equivalent ellipse which results in the force values relatively close to those obtained by the actual contact patch. The wheel/rail contact penetration is then approximated based on the distance between the equivalent contact ellipses [26]. It should be noted that a theoretical approach known as the elastic contact formulation using algebraic equations (ECF-A) is also used in some parts of this thesis to solve the contact search problem. In this method the wheel and rail profiles are parameterised using two wheel and two rail surface parameters. The physical contact constraints are then utilised to construct a system of four algebraic equations. The solution of this nonlinear system of equations yields the four surface parameters that represent the contact point positions on wheel and rail. More details on the ECF-A approach can be found in [27-29].

Fig. 2: One-car railway vehicle model developed in SIMPACK.

The wheel and rail profiles are created based on the nominal S1002 and 60 E1 profiles, respectively. However, in some parts of the thesis the worn S1002 and 60 E1 wheel and rail profiles are also used for the analysis.

2.1.1. Suspension system configuration

Many researches are done on model development of railway vehicles especially with focus on suspension system, see e.g. [30-36]. The suspension system of the vehicle model under study consists of a set of primary and secondary springs and dampers in the longitudinal, lateral and vertical directions as shown in Fig. 3. In addition to these springs and dampers a traction rod, a bump stop, and an anti-roll bar are also added to the bogie secondary suspension system to improve the dynamics behaviour of the vehicle. The aforementioned components are listed in Table 1.

Table 1: Suspension system components for the vehicle model A.

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

The superscripts p and s denote primary and secondary suspension elements, respectively. The variable l=1, 2, 3, 4 is the wheel set number, m=x, y, z denotes the longitudinal, lateral, and vertical directions, respectively. The variable n=R, L represents the right or left hand side suspension component and h=L, T indicates leading or trailing bogie, respectively. The components anti-roll bar ($k_{\mathrm{h}}^{\mathrm{AR}}$), traction rod ($k_{\mathrm{h}}^{\mathrm{TR}}$), and bump stop ($k_{\mathrm{h}}^{\mathrm{BS}}$) are modeled as linear springs, torsional springs, and nonlinear spring-damper sets, respectively. In practice, dampers are usually equipped with elastic bushings at both ends. In order to account for the corresponding effects, all the dampers are modelled as a spring and a damper in series. Furthermore, the damping of all the physical springs is also taken into account.

The numerical values of the design parameters listed in Table 1 are in-service values provided by the Bombardier transportation, Västerås, Sweden.

It should be noted that within this suspension system configuration the suspension elements corresponding to the right and left hand side of the wheel sets as well as the leading or trailing bogies are not necessary the same and could take different values. Indeed, two general suspension configurations are considered for the analysis. A symmetric and an asymmetric.

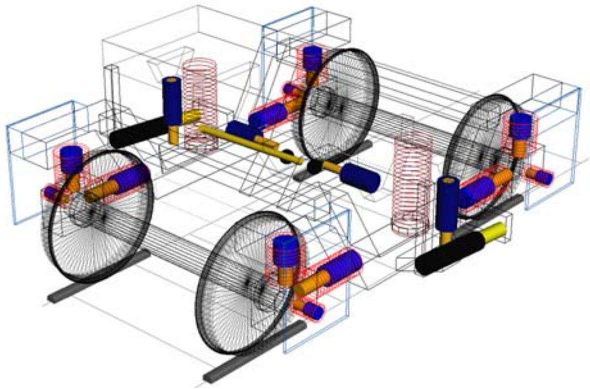

Fig. 3: Bogie system corresponding to the vehicle model A developed in SIMPACK.

In the symmetric case, the respective suspension components attached to the right and left hand sides as well as leading or trailing wheel sets (or bogies in the case of secondary suspension) take the same values. For instance, $k^p_{1xR} = k^p_{1xL} = k^p_{2xR} = k^p_{2xL} = k^p_{3xR} = k^p_{3xL} = k^p_{4xR} = k^p_{4xL}$ which means all the longitudinal primary springs have the same stiffness. This is the most practical setup and is the basis for the vehicle model considered in **papers A**-**E** for the analysis.

In contrast, one might have dissimilar stiffness and damping values for the corresponding suspension components on the right or left and leading or trailing wheel sets (or bogies in the case of secondary suspension) in an asymmetric setup. For example, the longitudinal primary springs on the right and left hand sides of the leading axle might have unequal stiffness values (i.e. $k^p_{1xR} \neq k^p_{1xL}$). In the most general asymmetric case, each suspension element could have a unique value which is different from the rest of the components. Such a model makes it possible to study the effects of every single suspension element on the dynamics behaviour of the vehicle. This is part of the study in the global sensitivity analysis carried out in **paper A**.

In addition to the vehicle model with general asymmetric suspension system, an asymmetric suspension configuration with diagonal symmetry is also considered for the analysis in this thesis. As an example on the asymmetric vehicle model with diagonal symmetry, one can have

$\left(k^{p}_{1xR}=k^{p}_{2xL}=k^{p}_{3xR}=k^{p}_{4xL}\right)\neq\left(k^{p}_{1xL}=k^{p}_{2xR}=k^{p}_{3xL}=k^{p}_{4xR}\right)$. This vehicle model is subject to study in **paper C**.

It should be noted that the number of design parameters for the vehicle model with symmetric, general asymmetric, and asymmetric with diagonal symmetric suspension configurations are 14, 76, and 27, respectively (excluding the bump stop). The number of input design parameters for multiobjective optimisation problems formulated in **papers B** and **C** are significantly reduced with the aid of the global sensitivity analysis as described in **paper A**.

2.2 Vehicle model B

The nonlinear 50 DOFs railway vehicle model introduced in the previous section is too complicated for the robust controller design purpose which is subject to study in **paper E**. Consequently, a simpler linear model with fewer DOF has to be chosen for the control design. Here, a linear half-car railway vehicle model is considered as shown in Fig. 4. The vehicle model B includes two wheel sets, one bogie frame, and a half carbody. All these components are rigid and allow a yaw and a lateral motion except for the half carbody which only allows a lateral motion. Therefore, the model has a total of 7 DOFs.

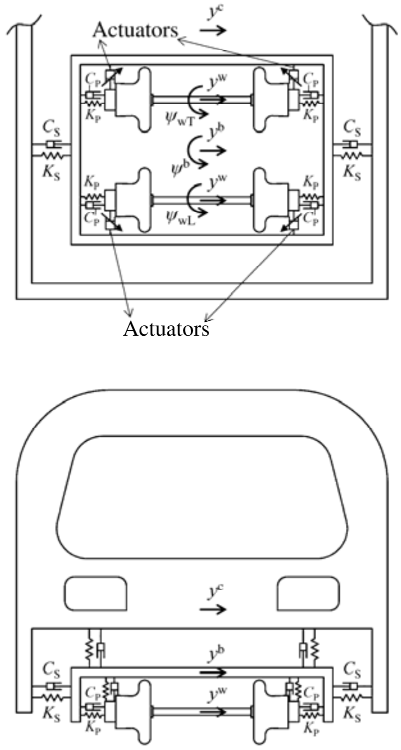

Fig. 4: Half-car railway vehicle model B.

The primary and secondary suspension components of the vehicle model B consist of a set of linear springs and dampers in the lateral direction as well as a set of actuators to apply the control torque to the wheel sets.

The Kalker's linear contact theory [37] is employed to yield the contact forces. It should be noted that the assumptions and theories used to develop the mathematical model of the vehicle such as rigid body elements, linear wheel profiles, two DOF wheel sets and bogie frame as well as the Kalker's linear theory are sufficient for the purpose of suspension control design of high speed bogies, see e.g. [38, 39]. This type of model is simple, easy to implement, and the computation time is efficient (especially for optimisation and active control design problems).

It should be noted that the robust controller designed based on this model is tested on the vehicle model A (which is more advanced and realistic) to check if the designed controller is practical.

3 OBJECTIVE FUNCTIONS

The ultimate goal of this PhD study is to improve the vehicle's dynamics performance from different perspective by working on the multiobjective optimisation and active control of bogie suspension components. Within this framework, global sensitivity analysis, multiobjective optimisation, semi active and active vibration control of bogie suspension are the main problems subject to study in this thesis. As aforementioned, the bogie dynamics performance is evaluated based on the vehicle's safety, ride comfort, and wear on different operational scenarios. Therefore, in order to formulate the sensitivity analysis, multiobjective optimisation, semi active and active control problems, it is necessary to express the prescribed objective functions in mathematical terms. In most of the cases, this could be done using the railway standards. In the following sections the mathematical representation of the objective functions are introduced.

3.1 Ride comfort

Passenger comfort is one of the most prominent design objectives in high speed trains. This criterion could be analysed 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 and power spectral density 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:

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

Where, t_0 and t_{f} indicate the initial and final simulation times, respectively and m is the total number of the measurement points.

Based on this objective function, higher RMS values of the carbody acceleration represent lower 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 introduced in railway standards.

Alternatively, the power spectral density (PSD) of the lateral carbody accelerations can also be used as a ride comfort measure index as described in **papers B** and **E** of this thesis.

3.1.2 Wertungszahl (Wz)

Based on the Wz index, the accelerations have to be measured at the floor plane of the vehicle carbody. The index Wz is calculated as follows [40]:

$$\mathrm{Wz} = 4.42\left(a^{\mathrm{wrms}}\right)^{0.3}, \tag{2}$$

where a^{wrms} is the RMS value of the frequency weighted acceleration $a^{w}(t)$(m/s^2). The corresponding frequency weighted functions for evaluation the ride comfort in railway applications can be found in [41]. 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 index which directly used 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 [40].

3.1.4 ENV 12299

One of the most widely-used ride comfort criteria in railway application is measured based on the CEN standard ENV 12299:2009-08 [42]. 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:

$$N_{\mathrm{MV}} = 6\sqrt{\left(a_{\mathrm{XP95}}^{\mathrm{W_{ad}}}\right)^2 + \left(a_{\mathrm{YP95}}^{\mathrm{W_{ad}}}\right)^2 + \left(a_{\mathrm{ZP95}}^{\mathrm{W_{ab}}}\right)^2}, \tag{3}$$

where, $\mathrm{W_{ad}}=\mathrm{W_a}\times\mathrm{W_d}$ and $\mathrm{W_{ab}}=\mathrm{W_a}\times\mathrm{W_b}$ are the weighting functions. Variables $a_{\mathrm{XP95}}^{\mathrm{W_{ad}}}$, $a_{\mathrm{YP95}}^{\mathrm{W_{ad}}}$, and $a_{\mathrm{ZP95}}^{\mathrm{W_{ab}}}$ 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 [43]:

$$H_{\mathrm{A}}(s) = \frac{s^2 4\pi^2 f_2^2}{\left(s^2 + \dfrac{2\pi f_1}{Q_1}s + 4\pi^2 f_1^2\right)\left(s^2 + \dfrac{2\pi f_2}{Q_1}s + 4\pi^2 f_2^2\right)}, \tag{4a}$$

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

$$H_{\mathrm{D}}(s)=\frac{(s+2\pi f_3)}{\left(s^2+\frac{2\pi f_4}{Q_2}s+4\pi^2 f_4^2\right)}\frac{2\pi K f_4^2}{f_3}, \tag{4c}$$

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 Eqs. (4) can be found in [43].

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. This ride comfort index is used in **paper A**.

Table 2: Ride comfort classification.

$N_{MV} < 1$	Very comfortable
$1 \le N_{MV} < 2$	Comfortable
$2 \le N_{MV} < 4$	Medium
$4 \le N_{MV} < 5$	Uncomfortable
$N_{MV} \ge 5$	Very uncomfortable

3.1.5 Continuous comfort

The CEN standard EN 12299:2009-08 [42] also proposed the continuous ride comfort index. The standard procedure is described in [42] in which the continuous comfort index (C_{Ci}, i=x, y, z) in the longitudinal, lateral, and vertical directions is evaluated as:

$$C_{\mathrm{Cx}}(t)=a_{\mathrm{XP}}^{\mathrm{W_d}}(t), \tag{5a}$$

$$C_{\mathrm{Cy}}(t)=a_{\mathrm{YP}}^{\mathrm{W_d}}(t), \tag{5b}$$

$$C_{\mathrm{Cz}}(t)=a_{\mathrm{ZP}}^{\mathrm{W_b}}(t), \tag{5c}$$

where, $a(t)$ is the carbody acceleration, W_d is the weighted frequency function in the lateral/longitudinal direction, and W_b is the weighted frequency function in the vertical direction. The subscript P denotes the floor level. Similar to the previous case, the continuous comfort index is evaluated leastwise at three points of the carbody floor; in the front, centre, and rear of the carbody. The ride comfort objective function (Γ_{C}) is defined as the maximum RMS value of the continuous comfort index measured in different directions at those three points as follows:

$$\Gamma_{\mathrm{C}} = \max \begin{bmatrix} \mathrm{RMS}\left(C_{\mathrm{Cx}}(t)\right) \\ \mathrm{RMS}\left(C_{\mathrm{Cy}}(t)\right) \\ \mathrm{RMS}\left(C_{\mathrm{Cz}}(t)\right) \end{bmatrix}_{\mathrm{h}} \text{, h=front, center, rear} \tag{6}$$

Here, the classification of ride comfort is determined as given in Table 3 [42]:

Table 3: Ride comfort classification (continuous comfort).

Continuous comfort index	Ride comfort feeling
$C_{Cy}(t)$, $C_{Cz}(t) < 0.20$ m/s^2	Very comfortable
0.20 m/s$^2 \leq C_{Cy}(t)$, $C_{Cz}(t) < 0.30$ m/s^2	Comfortable
0.30 m/s$^2 \leq C_{Cy}(t)$, $C_{Cz}(t) < 0.40$ m/s^2	Medium
0.40 m/s$^2 \leq C_{Cy}(t)$, $C_{Cz}(t)$	Less comfortable

The continuous ride comfort index is used in **papers B-E**.

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 safety objective functions that are considered in this thesis.

3.2.1 Track shift force

Track shift force (ΣY) is measured as the difference between the lateral forces acting on the left and right wheels of a wheel set. High track shift force might worsen the track condition and as a result increase the maintenance cost. According to the CEN standard EN-14363 [44] the track shift force limit is expressed (in kN) as follows:

$$\sum Y_{\text{20 Hz, 2 m, mean, 99.85\%, lim}} \leq K_1\left(10 + 2Q_0 / 3\right), \tag{7}$$

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:

$$2Q_0 = \frac{m_{veh}g}{n}, \tag{8}$$

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 2 m window in 0.5 m increments and subject to a 20 Hz low-pass filter. The track shift force objective function (Γ_{TS}) is then defined as:

$$\Gamma_{\mathrm{TS}} = \max\left(\sum Y_{\text{20 Hz, 2 m, mean, 99.85\%}}\right)_l, \tag{9}$$

where, l=1, 2, 3, 4 is the axle number. Indeed, Γ_{TS} denotes the maximum filtered track shift force among all the wheel sets 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 (ΣY) 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 [44], the limit condition for a vehicle to run stable, is expressed as:

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

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

$$\Gamma_{\text{St}} = \max\left(\sum Y_{\text{RMS,100 m}}\right)_l, \; l\!=\!1, 2, 3, 4 \tag{11}$$

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 objective function considered here is the risk of derailment, which is particularly important for vehicles operating on curved operational scenarios at high speeds. Based on the CEN standard EN-14363 [44] 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:

$$\left(Y/Q \right)_{\text{20 Hz, 2 m, mean, 99.85\%}} \le 0.8, \tag{12}$$

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

$$\Gamma_{\text{RD}} = \max\left(Y_t/Q_t \right)_{\text{20 Hz, 2 m, mean, 99.85\%}}, \tag{13}$$

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

3.3 Wear

High speed and poor track quality might significantly increase the wheel-rail contact wear and maintenance cost. Suspension components on the other hand can affect the wheel set dynamics behaviour and in particular wear. In order to reduce wear, it is interesting to explore the effects of different suspension components on it. 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. [45-47]. In the following, two main approaches are introduced.

3.3.1 Archard's number

According to Archard, the material loss (V_w) is expressed as follows [48, 49]:

$$V_w = k\frac{N}{H}s\ , \tag{14}$$

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

Application of Archard's number 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. [50]. However, Archard's number mostly deals with material properties (see Eq. (14)) and it might fail to reflect the wheel set dynamics effects on wear. Therefore, another index which better reflects the effect of suspension components on the wheel set dynamics and wear namely energy dissipation at the contact patch is employed in this thesis to estimate wear.

3.3.2 Energy dissipation

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

$$\bar{E} = \left|F_\xi \upsilon_\xi\right| + \left|F_\eta \upsilon_\eta\right| + \left|M_{\xi\eta}\phi_{\xi\eta}\right|\ , \tag{15}$$

where, variables 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 [29]:

$$\begin{cases} \upsilon_\xi = \dfrac{\left(\dot{\mathbf{r}}^{\mathrm{w}} - \dot{\mathbf{r}}^{\mathrm{r}}\right)^{\mathrm{T}} \mathbf{t}_1^{\mathrm{r}}}{V} \\ \upsilon_\eta = \dfrac{\left(\dot{\mathbf{r}}^{\mathrm{w}} - \dot{\mathbf{r}}^{\mathrm{r}}\right)^{\mathrm{T}} \mathbf{t}_2^{\mathrm{r}}}{V} \\ \phi_{\xi\eta} = \dfrac{\left(\boldsymbol{\omega}^{\mathrm{w}} - \boldsymbol{\omega}^{\mathrm{r}}\right)^{\mathrm{T}} \mathbf{n}^{\mathrm{r}}}{V} \end{cases}\ , \tag{16}$$

Here, the subscripts w, and r denote wheel, and rail, respectively. Vectors, $\dot{\mathbf{r}}$ and $\boldsymbol{\omega}$ are the global velocity vector of the contact point, and angular velocity vector, respectively. Variable V is the reference velocity. Vectors $\mathbf{t}_1^r$, $\mathbf{t}_2^r$ 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 position on the wheel and rail surfaces. Most of the multibody dynamics softwares like GENSYS [51] and SIMPACK [26] utilise look-up tables for this purpose. However, to have more accurate wear estimation, a theoretical approach known as the elastic contact formulation using algebraic equations is also considered in some parts of this thesis.

Once the creepages attained, one can compute the respective contact forces through a proper contact theory. In most of the cases the FASTSIM algorithm [52] is utilised for this particular purpose. However, in part of **paper E** the Kalker's linear theory is also used to simplify the equations required for the robust control design of the railway vehicle.

Finally, the wear objective function (Γ_{W}) is defined as:

$$\Gamma_{\mathrm{W}} = \sqrt{\frac{1}{t_{\mathrm{f}} - t_0}\int_{t_0}^{t_{\mathrm{f}}} \left(\left|F_{\xi}\upsilon_{\xi}\right| + \left|F_{\eta}\upsilon_{\eta}\right| + \left|M_{\xi\eta}\phi_{\xi\eta}\right|\right)^2 dt} \ , \tag{17}$$

here, t_0 and t_{f} indicate the initial and final simulation time, respectively.

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 multiobjective optimisation and active control of bogie suspension. In this regard, a brief overview on the problem formulation and methodologies for global sensitivity analysis, multiobjective optimisation, and active control of bogie suspension components considered in **papers A-E** of this thesis is given in this section.

4.1 Sensitivity analysis

Sensitivity analysis can be carried out either locally or globally. In the following a brief introduction about these two approaches is given.

4.1.1 Local sensitivity analysis

In the local methods the effects of the design inputs on the system response is approximated as the partial derivative of an objective function (Γ) with respect to the design parameter (x_i) which is taken around a fixed point x_0. Such methods only take into account the variation of an objective function with respect to a single design parameter at a time. Furthermore, the domain of the input design variables might not be appropriately scanned using the local methods. Therefore, for a large scale nonlinear railway vehicle model with many DOFs the local sensitivity analysis methods might lead to inappropriate results and a global approach should be considered instead.

4.1.2 Global sensitivity analysis

Sensitivity analysis is one of the most prominent steps in design and optimisation 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 A** 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 $\mathbf{d} = [d_1, d_2, \ldots d_m]^T \in \mathbf{\Omega}$, through the respective deterministic functional relationship $\Gamma = \mathcal{F}(\mathbf{d})$. Where, $\mathbf{\Omega}$ is the domain of input design variables. The mean (μ) and variance (V) of Γ are defined as [22]:

$$\begin{cases} \mu_\Gamma = E_{\mathbf{d}}[\Gamma] = \int_{\mathbf{d}} \mathcal{F}(\mathbf{d}) f_{\mathbf{d}}(\mathbf{d}) \delta\mathbf{d} \\ V_\Gamma = E_{\mathbf{d}}[(\Gamma - \mu_\Gamma)^2] = E_{\mathbf{d}}\{[\mathcal{F}(\mathbf{d})]^2\} - \mu_\Gamma^2 \end{cases} , \tag{18}$$

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

$$E_{-i}[\Gamma \mid d_i] = \int_{\mathbf{d}_{-i}} \mathcal{F}(\mathbf{d}_{-i}, d_i) f_{\mathbf{d}_{-i}}(\mathbf{d}_{-i}) \delta \mathbf{d}_{-i} \tag{19}$$

The primary (S_i) and total (S_{Ti}) sensitivity indices are defined by Eqs. (20) and (21), respectively. See e.g. [22, 53, 54] for more details.

$$S_i = \frac{V_i[E_{-i}(\Gamma \mid d_i)]}{V_\Gamma}, \tag{20}$$

$$S_{Ti} = \frac{E_{-i}[V_i(\Gamma \mid \mathbf{d}_{-i})]}{V_\Gamma}, \tag{21}$$

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 A** to solve the global sensitivity problem of bogie dynamics with respect to suspension components.

4.1.1 Sensitivity analysis research outcomes

Within and after carrying out the sensitivity analysis, the following outcomes are expected:

- An efficient algorithm suitable for global sensitivity analysis of complex multibody systems.
- Identification of important design parameters.
- Narrowing down inputs for optimisation.
- Improving computational efficiency of optimisation.

4.2 Optimisation

In general, the optimisation problem of m design parameters $\mathbf{d} = [d_1, d_2, ...d_m]^T \in \mathbf{\Omega}$ (where, $\mathbf{\Omega}$ is the domain of input 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 follows:

Determine $\mathbf{d}^*$ and $\mathbf{x}^*(t)$ such that

$$\mathcal{F}(\mathbf{d}^*, \mathbf{x}^*(t)) = \min \mathcal{F}(\mathbf{d}, \mathbf{x}(t)),\ \mathbf{d}^* \in \mathbf{\Omega}, \tag{21}$$

subject to

$$\Gamma_j(\mathbf{d}) = \mathcal{F}_j(\mathbf{d}) \leq \Gamma_j^{\max}, \tag{22}$$

in which, $\Gamma_j^{\max}$, j=*1*, *2*,…, n denote the threshold values, $\mathbf{x}(t)$ is the state vector of the railway vehicle model estimated by using the computational model. Therefore, based on the objective functions introduced in section 3 and suspension system components several multiobjective optimisation problems can be formulated. GA-based multiobjective optimisation routine in MATLAB is utilised to solve the optimisation problems specified in **papers A-C**. The procedure can be described as follows: in each iteration, MATLAB updates the design parameters file as an input to the vehicle model developed in SIMPACK, the dynamic response of the system is then evaluated using the time integration solvers and the respective objective functions are attained after a post-processing stage. At this step, the thresholds are checked to make sure if all the objective functions are within the admissible limits. If at least one of the objective functions violated the thresholds, the vector of the objective function is penalised to assure that all the Pareto optimised results satisfy the problem constraints. This procedure continues until convergence or the maximum number of generations achieved. In the case of multiobjective optimisation problems, the results can be plotted in terms of Pareto set and Pareto front graphs.

4.2.1 Optimisation research outcome

Within and after carrying out the optimisation the following outcomes are expected:

- An interface between SIMPACK and MATLAB SIMULINK to be able to run co-simulations for optimisation and active control purposes.
- Possibility of simultaneous optimisation of active and passive components.
- Safety multiobjective optimisation results of bogie suspension.
- Enhanced running speeds on curves.
- Reduce wear and improve ride comfort on curves.

4.3 Genetic algorithm

The GA is an optimisation technique which has biological origins and works based on the probabilistic searching. The GA is successfully applied to the multiobjective optimisation problem of a variety of complex nonlinear systems such as bogie suspension of high speed trains, see e.g. [1, 11-16].

A GA is generally consists of the following main steps [55]:

- chromosome encoding
- fitness
- selection
- recombination
- evolution

In contradict to the natural GAs which have to follow certain laws observed in nature, the characteristics of the abovementioned steps in a GA with optimisation applications are determined by the designer based on the design requirements.

The ordinary optimisation techniques such as Newton-Raphson and its variants are mostly suitable for convex optimisation problems. Such methods utilise local information and as a result might fail to find the global minima. The GA is known to be a method which attains reasonably good global solutions in many applications. However, the initial guess, number of generations, population size, and other settings have to be carefully selected to be able to achieve satisfactory results. In this regard, the initial guess for most of the optimisation problems formulated in this thesis comes from the in service data provided by the Bombardier transportation, Västerås, Sweden. In multistep optimisation problems, the Pareto optimised results obtained in the previous step are chosen as the initial guess for optimisation of the next steps, see **papers B** and **C**. Furthermore, the GA initial settings in MATLAB such as population size and number of generations are chosen close to those associated with similar problems solved earlier, see e.g. [18].

In a multiobjective optimisation problem, if no weighting coefficient act on the vector of the objective functions, the GA minimises every single objective function. Indeed, all the objective functions are treated the same way and there is no need to normalise the objective functions in that case. However, to better represent the Pareto optimised results, the objective functions are normalised with respect to their initial values in some parts of the thesis, see **paper C**.

When it comes to the complex nonlinear models, it is difficult to impose constraints to the optimisation algorithm using MATLAB routine. To overcome such a problem, in the case of a violation of the constraints a penalty factor is imposed to the objective functions to make sure that the Pareto optimised results remain within the constraints. More details can be found in **papers B** and **C**. The steps required to set up a GA are briefly introduced in the subsequent sections.

4.3.1 Chromosome encoding

After selecting the design parameters for optimisation (stiffness and damping values that are chosen with the aid of the global sensitivity analysis, see **paper A**), the GA encodes the design parameters. Each encoded design parameter is known as a gene. The complete set of genes (design parameters) that uniquely describe an individual is referred to as a chromosome. Indeed, gene is a particular position or locus in a chromosome.

The particular string representations used for a given problem is known as the GA encoding of the problem. Therefore, the encoded chromosomes are string representations of the solutions to a particular problem. The classical GA uses a bit-string representation to encode the solutions. Bit-string chromosomes composed of a string of genes which contains 0 or 1 characters, see e.g. [55] for more details.

4.3.2 Fitness

The quality of a chromosome as a solution to a particular problem is determined by the fitness function. Each objective function (such as ride comfort, safety, and wear introduced earlier) considered for the optimisation problem or a combination of the objective functions can be considered as the fitness function. It is necessary to evaluate the fitness of each particular chromosome. The fitness information are then used to bias the next generations based on the better genes.

4.3.3 Selection

Based on the fitness, the chromosomes should be selected for recombination and to construct the next generations. In general, the chromosomes which resulted in a better fitness function should have a higher chance to be selected. This might lead to a more highly fit solutions by the upcoming generations. It should be noted that highly fit chromosomes might have a chance to be selected twice or more or even recombined with themselves.

Fitness proportional also known as the roulette wheel is one of the common selection methods. The probability of parenthood (to be selected) in this method is proportional to the fitness. There are many different selection methods such as random stochastic selection, tournament selection, and truncation selection. More details on the selection schemes can be found in [55].

4.3.4 Recombination

In the recombination, the chromosome of a child is being created using the chromosomes of the parents selected earlier. The main operators of the recombination are known as crossover and mutation. An example of the crossover is shown in Fig. 5. The child properties depend on the crossover point. For instance, in Fig. 5 (a) the child has similar ears to the parent 1 and similar eyes to the parent 2, but in Fig. 5 (b) the child has similar ears to the parent 1 but different eyes from both parents.

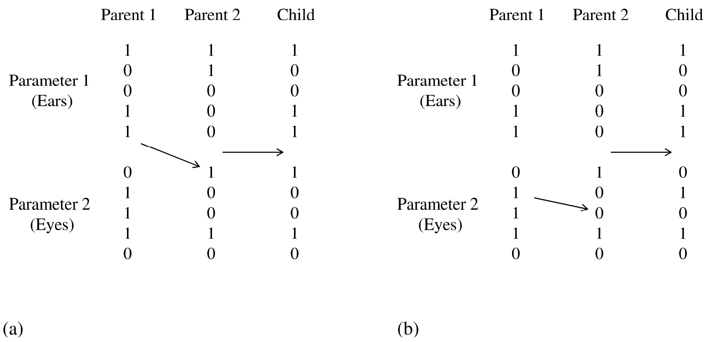

Fig. 5: The crossover example.

Once the child chromosome is generated by the crossover, the GA applies the mutation operator on the resulting chromosome to change one or more properties, see Fig. 6 for example.

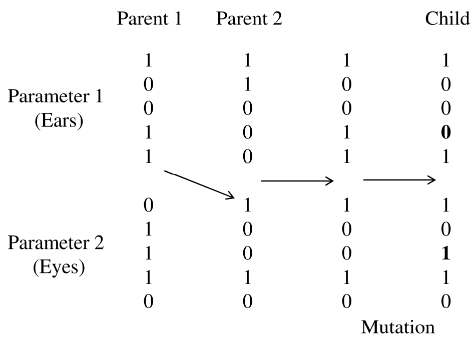

Fig. 6: The mutation example.

How to decide for the crossover and mutation methods to achieve more fitted results depends on the GA settings.

4.3.5 Evolution

The chromosomes obtained from the previous stages are diverted into the so called successor population. The selection and recombination steps are then repeated until a complete successor population achieved which is going to be considered as the next generation. The GA repeats this process through a number of generations until certain convergence to a best fitness solution or maximum number of iterations achieved.

The evolutionary schemes determine which chromosomes from the source population are eligible to remain unchanged when passing to the successor population. It is vital to employ an appropriate evolutionary scheme. This is usually decided based on the nature of the domain of the input design parameters being searched. One of the most well-known schemes is replacement with elitism. To create the successor population, this scheme preserves the best one or two individuals from the source population and generates the rest through selection and recombination. This method assures that solutions of the highest relative fitness will be appear in the next generation through the selection process, see [55] for more details. The GA flowchart is plotted in Fig. 7.

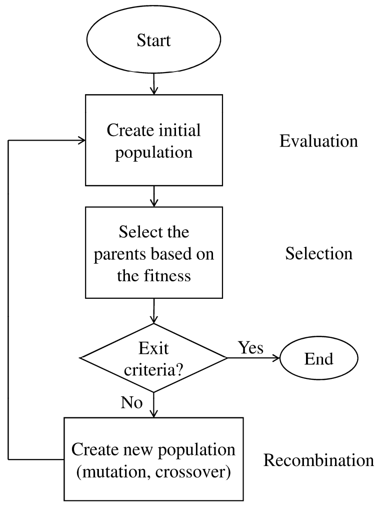

Fig. 7: The GA flowchart.

4.4 Robustness analysis

Uncertainties in the design parameters might happen due to manufacturing errors, environmental conditions and so on. It is vital to study the dynamics behaviour of the vehicle in the presence of uncertainties in the design parameters. One of the targets of this thesis is to check the robustness of the bogie suspension Pareto optimised values achieved by the war/comfort Pareto optimisation problem (carried out in **paper C**) against uncertainties in the design parameters. This is subject to study in **paper D**.

Monte Carlo simulation is one of the most well-known tools in reliability analysis. However, in a same manner to the global sensitivity analysis problem, high number of samples and heavy computational efforts make it difficult to apply the Monte Carlo simulation to nonlinear multibody dynamics systems with many DOFs like the vehicle model in question. Consequently, a more efficient algorithm has to be chosen for that particular purpose.

Zhang and Pandey [56] proposed an efficient algorithm for reliability analysis which works based on the entropy concept. The M-DRM method also utilised to convert the multivariable integrals (required to calculate the fractional moments) into univariate integrals. The proposed algorithm significantly improved the computational efficiency and has been successfully employed in some simple civil and mechanical engineering applications. The target here is to apply this algorithm for robustness analysis of the bogie Pareto optimised parameters with respect to the uncertainties in the design parameters.

The simplified maximum entropy problem with fractional moments can be formulated as [56]:

Find Lagrange parameters $\boldsymbol{\alpha} = [\alpha_1 \quad ... \quad \alpha_m]^{\mathrm{T}}$ and $\boldsymbol{\lambda} = [\lambda_1 \quad ... \quad \lambda_m]^{\mathrm{T}}$ which minimise the functional

$$\Upsilon(\boldsymbol{\lambda}, \boldsymbol{\alpha}) = \log\left[\int_Y \exp\left(-\sum_{k=1}^{m} \lambda_k y^{\alpha_k}\right) dy\right] + \sum_{k=1}^{m} \lambda_k M_Y^{\alpha_k} \ , \tag{23}$$

variable $M_Y^{\alpha_k}$ is the α_kth order fractional moment of the objective function defined as

$$M_Y^{\alpha} = \int_{\mathbf{X}} [\eta(\mathbf{X})]^{\alpha} f_{\mathbf{X}}(\mathbf{X}) d\mathbf{X} \, , \tag{24}$$

here, $f_{\mathbf{X}}(\mathbf{X})$ is the joint density of the random variables **X**.

The multi-dimensional integral in Eq. (24) can be simplified using the M-DRM. Indeed, by using this method an αth order fractional moment can be approximated by a product of αth order moments of the univariate functions which significantly simplifies the problem complexity.

Based on the maximum entropy principle and the fractional moments of the system response, the probability density function (PDF) of an objective function is estimated as:

$$\widehat{f}_Y(y) = \exp\left(-\sum_{k=0}^{m} \lambda_k y^{\alpha_k}\right), \tag{25}$$

where, α_0=0 and $\lambda_0 = \log\left[\int_Y \exp\left(-\sum_{k=1}^{m} \lambda_k y^{\alpha_k}\right) dy\right]$. More details on reliability analysis using maximum entropy, fractional moments, and M-DRM concepts can be found in [56, 57].

4.4.1 Reliability analysis research outcome

Within and after carrying out the reliability analysis the following outcomes are expected:

- An efficient algorithm suitable for robustness analysis of performance of multibody dynamics systems against design parameters uncertainties.
- Probability of failure of different objective functions.

4.5 Active control

The multiobjective optimisation problems formulated and solved in **papers A-C** concern with the passive suspension components. Application of semi active control strategies and MR dampers in bogie secondary suspension is also investigated in **paper B**. Indeed, the input electrical current to the MR damper together with the longitudinal and vertical secondary stiffnesses are optimised with respect to safety. The vehicle performance once different semi active control schemes are in use is then explored.

In addition to the passive and semi active control strategies, active control of bogie suspension is also one of the main targets of this research. In this regard, a robust controller is designed for active steering of the wheel sets. The controller is robust against sensor noise and track irregularities. This is subject to study in **paper E**. In this section, an introduction on the active control with application in railway vehicles is given and two common control theories are briefly reviewed.

4.4.1 Classical control strategy

Study the dynamics of a solid axle wheel set showed that it is possible to stabilize the wheel set with a simple feedback control applied either as a lateral control force or a control torque. However, the lateral control force might deteriorate ride comfort [58]. Pearson et al. [39] proposed a control yaw torque proportional to the axle's lateral velocity. A phase lead term has also been added to the feedback loop to improve the controller performance for different values of the wheel conicity as shown in Fig. 8.

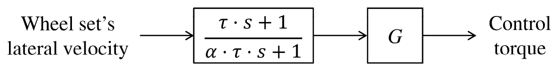

Fig. 8: The classical controller.

Where, G is the proportional gain and has to be tuned together with α and τ to yield a satisfactory performance.

4.4.2 Optimal control strategy

Another suitable possibility for the active control of wheel sets is the optimal control technique. The state space equations of the vehicle model can be written as follows:

$$\begin{cases} \dot{\mathbf{x}} = \mathrm{A}\mathbf{x} + \mathrm{B}\mathbf{u} + \Gamma\mathbf{w} \\ \mathbf{y} = \mathrm{C}\mathbf{x} + \mathrm{D}\mathbf{u} \end{cases}, \tag{26}$$

where, matrices A, B, C, and D are the matrices of the state space representation which specifies the vehicle's dynamics equations. Vector **x** is the state vector, **u** is the control input vector, **y** is the output measurement vector, and **w** is the vector of the track inputs.

The system equations of motion should be solved subject to minimization of the following cost function:

$$J = \int \left(\mathbf{y}^{\mathrm{T}} \cdot \mathrm{Q} \cdot \mathbf{y} + \mathbf{u}^{\mathrm{T}} \cdot \mathrm{R} \cdot \mathbf{u} \right) \mathrm{d}t \tag{27}$$

This means that the lateral motion of the wheel set (y) has to be minimised with minimum control effort. The gain matrices Q and R should be selected in an appropriate way to be able to lead to a satisfactory result. The solution of the formulated optimal control problem can be found using the MATLAB function CARE which solves the Riccati equation. The optimal feedback control can then be stated as follows:

$$\mathbf{u} = -K^{*}\mathbf{x} \tag{28}$$

Where, K^{*} is the optimal control gain achieved from the solution of the Riccati equation. The two control strategies introduced here are quite simple and easy to implement. However, they might fail to reduce the effects of the track and sensor disturbances on the dynamics behaviour of the vehicle. Consequently, a robust controller is designed using the μ-synthesis toolbox in MATLAB for this particular purpose which is the subject to study in **paper E**.

4.4.3 Active control research outcome

Within and after carrying out the active control design the following outcomes are expected:

- Usage of practical sensors and actuators to develop the controller.
- An active controller which is robust against track irregularities.
- An actuator compensation technique to improve the performance.

5 SUMMARY OF APPENDED PAPERS

A brief summary on the appended papers of the thesis is given in this section.

5.1 Paper A

Study the multiobjective optimisation problem of suspension system of a nonlinear multibody system with many DOFs is an elaborate task which requires high computational efforts. Furthermore, it is not practical to optimise every single element of the suspension components. Indeed, the number of input design parameters is one of the most critical issues which can significantly affect the computational burden of the optimisation. Therefore, the multiobjective optimisation has to be carried out with respect to a few suspension elements at a time. Now, the challenge is to select the suitable design parameters for multiobjective optimisation in a logical way.

As a preliminary stage in multiobjective optimisation and active control problem, it is necessary to study the influence of different primary and secondary suspension components on vehicle's dynamics behaviour on various operational scenarios. Global sensitivity analysis makes it possible to recognize the design parameters that have the most prominent effects on the system dynamics response and attenuate the number of inputs for optimisation. This significantly improves the computational efficiency. However, the global sensitivity analysis of nonlinear complex systems itself requires large number of samples and high computational efforts. Here, an efficient method for the global sensitivity analysis of multibody systems is followed. The methodology works based on the M-DRM and is applied for the global sensitivity analysis and optimisation of a railway vehicle dynamics with respect to suspension stiffness and damping components.

A one-car railway vehicle model with passive primary and secondary suspension components developed in multibody dynamics software SIMPACK is considered for the analysis. The vehicle runs on different operational scenarios with various curve radii ranging from very small radius curves up to straight track. For each particular scenario, measurement data are applied to the model as the track irregularities and the vehicle runs at maximum admissible speed. Wear, ride comfort, and safety are considered as the objective functions and the effects of each particular suspension element on these objective functions are thoroughly investigated for symmetric and asymmetric configurations of the bogie suspension system.

The results of this paper provide informative data that can be used in multiobjective optimisation and active control design of bogie suspension. This is subject to study in **paper A**.

5.2 Paper B

Another main target of this thesis is to boost the vehicle's maximum admissible speed on curves by improving the passive suspension components. To account the variations in the track curve radii, the simulations have been performed on various operational scenarios. The enhanced maximum admissible speeds have been calculated using the European railway standards according to the track plane accelerations of a_y=1.5 m/s^2. Increasing the speed on curves might negatively affect safety. Therefore, those suspension components that had a remarkable effect on different safety objective functions (i.e. running stability, risk of derailment, and track shift force) identified by the global sensitivity analysis were subject to multiobjective optimisation with respect to safety. To further improve the computational efficiency the secondary and primary design parameters have been classified into two levels and the multiobjective optimisation has been accordingly carried out.
As aforementioned, the vehicle model is developed in the multibody dynamics software SIMPACK. On the other hand, the multiobjective optimisation and active control techniques used in this thesis are implemented in MATLAB. Therefore, the SIMAT module in MATLAB-SIMULINK has been employed to carry out SIMPACK-SIMULINK co-simulations. It should be noted that the GA multiobjective optimisation routine in MATLAB has been used to solve the formulated optimisation problems.

The optimisation results represent the optimal values of the design parameters which allow higher running speeds on curves while guarantee a secure run. Up to this point only passive suspension systems have been considered. To study the vehicle's running safety when semi-active control strategies are in use, passive yaw dampers are replaced with MR dampers and the input electrical current to the MR dampers is subject to optimisation. The multiobjective optimisation results of this part guarantee a secure run once a combination of passive and semi-active components are in service. The abovementioned issues are subject to study in **paper B**.

5.3 Paper C

So far, the most significant design parameters are identified and part of those are optimised with respect to safety to improve the running speed and assure a secure run on different operational scenarios. Now, it is desired to improve ride comfort and reduce wear when running with the enhanced speeds. In this regard, the wear/comfort optimisation problem of bogie suspension is formulated in **paper C**. The design parameters are selected based on the global sensitivity analysis performed earlier. Furthermore, the results obtained from multiobjective optimisation with respect to safety (presented in **Paper B**) are utilised as the initial values of design parameters to develop the vehicle model and start the wear/comfort Pareto optimisation. Safety objective functions are introduced as thresholds to the multiobjective optimisation problem to guarantee that the results obtained in this part assure running safety. Similar to the previous problem, the design parameters have been classified into two groups and the optimisation has been performed in a multistep manner.

Furthermore, the effects of asymmetric suspension configurations on the dynamics behaviour of the vehicle have also been investigated and it has been shown that such configurations can significantly improve the dynamics performance of the vehicles on curves.

The wear/comfort Pareto optimisation results represent informative data about the suspension properties which lead to a better ride comfort and less wear on different operational scenarios while a satisfactory running safety is guaranteed. This is significant from practical point of view as the journey time is reduced by increasing the speed, passenger ride comfort is improved, and less wear resulted on the wheel and rail profiles. This is subject to study in **paper C**.

5.4 Paper D

After solving the two multiobjective optimisation problems the vehicle is able to securely run with the enhanced speeds while ride comfort is improved and wear is reduced. Uncertainties in the design parameters (suspension system components) might negatively affect the dynamics behaviour of the vehicle. As a result, the objective functions measured in practice might be different from the optimised values. Therefore, it is vital to check the effects of the design parameters uncertainties on the vehicle's dynamics response. In this regard, the robustness analysis of the bogie suspension Pareto optimised values achieved earlier is considered. The safety and wear/comfort Pareto optimised solutions obtained in **papers B** and **C** are considered as the reference case for the analysis. Different objective functions are then evaluated by considering a parameter variation which is applied according to a lognormal distribution of the design parameter with different coefficient of variations. A simplified method which works based on the maximum entropy, fractional moments, and M-DRM concepts is utilised to attain the probability distribution of the system response in different cases. Such an analysis is subject to study in **paper D**.

5.5 Paper E

At this stage, the running speeds on curves are boosted. For the vehicle model running at those speeds the ride comfort is improved and wear is reduced. Furthermore, the robustness of the bogie suspension optimised values is checked against uncertainties in the design parameters. Now the target is to explore the effects of active control techniques on the dynamics behaviour of the system. The first step here is to design an active controller which stabilises the wheel set using a realistic control effort. Since the track irregularities often reduce the controller performance, it is desired to have a robust controller. In this regard, an active controller is designed which is robust against track irregularities. Sensors and electromechanical actuators are employed to implement the active control scheme in a practical manner. Finally, a compensation technique is proposed to attenuate the actuator dynamics effects and improve the active control efficiency. This is subject to study in **paper E**.

6 CONCLUDING REMARKS AND FUTURE WORK

This thesis deals with the mathematical formulation and computational algorithms required for multiobjective optimisation and active control of bogie suspension system. In this regard, the following issues were in focus in this thesis:

- First, it is important to choose a proper model for multiobjective design optimisation and active control of railway vehicles. The model should not be too complicated as it might increase the computational efforts and it should not be too simple as it might fail to represent the realistic system response. Therefore, two generic models have been developed. The first one was a 50 DOFs one-car railway vehicle developed in SIMPACK and it was suitable for the multiobjective optimisation purpose. The second one was a half-car vehicle model with 7 DOFs and it was appropriate for the active control design purposes.
- The second major step was to determine the objective functions required to reflect the vehicle dynamics behaviour in mathematical terms. Ride comfort, wear, and three safety criteria including track shift force, running stability, and risk of derailment have been considered as the objective functions used to formulate different multiobjective optimisation problems. In most of the cases the railway standards have been used to evaluate the objective functions. However, there is no standard way to evaluate wear. Therefore, the energy dissipation at the contact patch has been chosen to reflect the wear objective function.
- Third, several multiobjective optimisation problems have been formulated to improve the vehicle performance from different perspectives. However, due to the system complexity and nonlinearities the computation time was a major challenge for optimisation. To improve the efficiency of the algorithm a global sensitivity analysis of the vehicle dynamics response with respect to the suspension components has been carried out using an efficient algorithm. The results of such an analysis provided informative data regarding the importance and weight of each particular design parameters on a specific objective function. The number of input design parameter for optimisation has been significantly mitigated with the aid of sensitivity analysis and as a result the computational efficiency of the optimisation has been improved.
- Fourth, once the desired design parameters have been recognized two multiobjective optimisation problems have been formulated. First to boost the running speed on curves by multiobjective optimisation of bogie suspension with respect to safety and second to reduce wear and improve comfort when the vehicle operates with the enhanced running speeds. The optimisations have been carried out using the GA in MATLAB. Furthermore, the effects of the semi active control strategies and the asymmetric suspension system configurations on the dynamics behaviour of the vehicle have been explored. The results of the asymmetric vehicle model revealed that significant improvement in the vehicle dynamics performance can be achieved with the aid of such configurations.
- Fifth, a robustness analysis has been carried out on the bogie suspension components Pareto optimised solutions achieved by the wear/comfort Pareto optimisation problem to check the effects of design parameters uncertainties on the dynamics behaviour of the vehicle.

- Finally, the active control problem of wheel set has been considered and a robust controller has been designed to reduce the effects of the track irregularities on the dynamics behaviour of the vehicle. The ball-screw electromechanical actuators have been employed to apply the control torque to the system and a compensation technique has been proposed to alleviate the actuator dynamics effect. The controller performance has been tested on a full scale railway vehicle model with realistic structural parameters and input track data.

In brief, the following aims and objectives in this PhD study have been achieved:

1. A global sensitivity analysis of bogie dynamics with respect to suspension components which provided a basis for solving the bogie suspension system multiobjective optimisation problems in a computationally efficient framework.
2. Multiobjective optimisation problems have been successfully formulated and solved for a multidimensional nonlinear dynamical system which represented the bogie model of a high speed railway vehicle.
3. The vehicle runs securely with enhanced running speeds on curves using the safety multiobjective optimisation results.
4. Passenger ride comfort is improved and wear is reduced when the vehicle runs with the enhanced high speeds using the wear/comfort Pareto optimisation results.
5. Robustness of the bogie Pareto optimised solutions is checked against uncertainties in the design parameters.
6. A robust controller for bogie primary suspension system is designed using practical sensors and actuators and the performance is checked on a full scale vehicle model.

The abovementioned stages are schematically shown in Fig. 9.

The future steps of the current research can be considered as follows:

- Formulating and solving different global sensitivity analysis and multiobjective optimisation problems to investigate the interaction between railway vehicle speed, track irregularities and structural parameters.
- Advances in flexible multibody dynamics formulation of the vehicle model could also be an interesting subject that might help to achieve more accurate results especially on wear and ride comfort.
- Active noise control of bogie.
- Application of semi active and/or active control systems in bogie suspension to simulate the asymmetric suspension system configurations in practice.

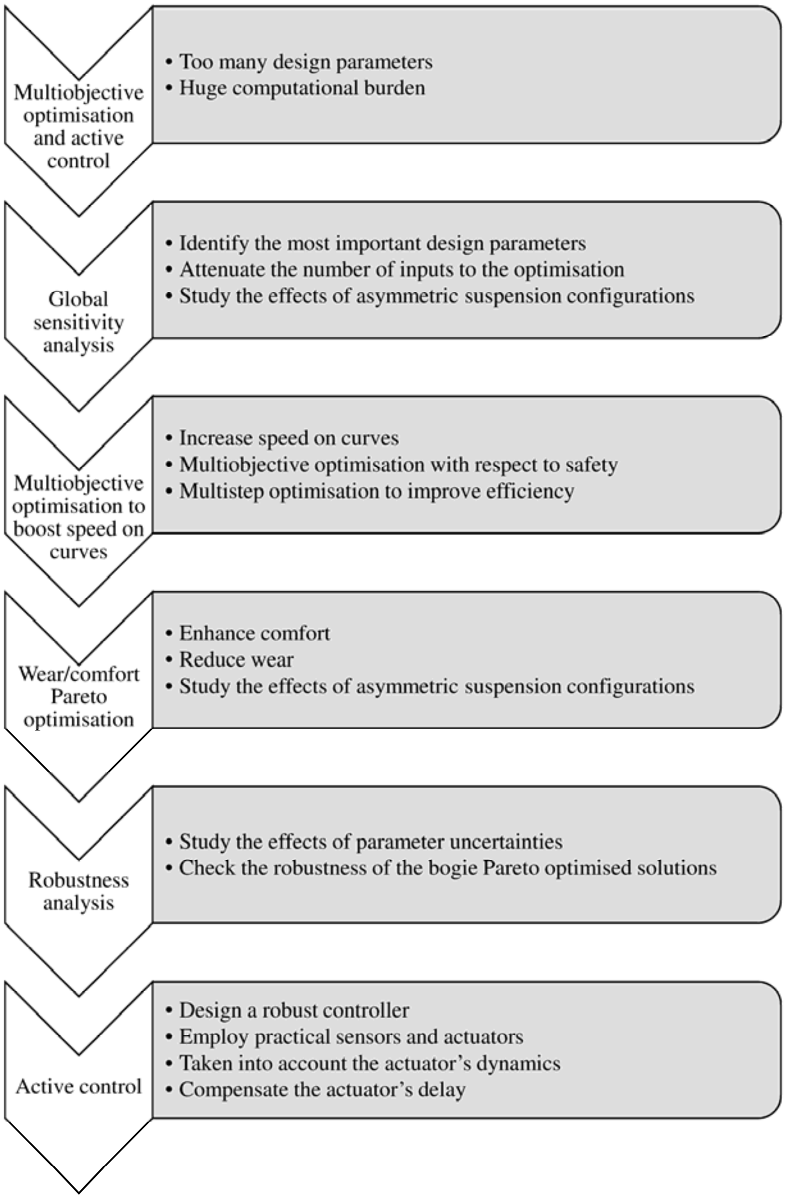

Fig 9: Main targets and stages of the PhD thesis.

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PART II

APPENDED PAPERS A-E

Paper A

Global sensitivity analysis of bogie dynamics with respect to suspension components

Published in Multibody System Dynamics, 2016, ***37****(2): pp. 145-174.*

http://dx.doi.org/10.1007/s11044-015-9497-0

Paper B

Multiobjective optimisation of bogie suspension to boost speed on curves

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http://dx.doi.org/10.1080/00423114.2015.1114655

Paper C

Wear/comfort Pareto optimisation of bogie suspension

*Published in Vehicle System Dynamics, 2016, **54**(8): pp. 1053-1076.*

http://dx.doi.org/10.1080/00423114.2016.1180405

Paper D

Robustness analysis of bogie suspension components Pareto optimised values

To be submitted for international publications

Paper E

Robust control and actuator dynamics compensation for railway vehicles

Accepted for publication in Vehicle System Dynamics, 2016