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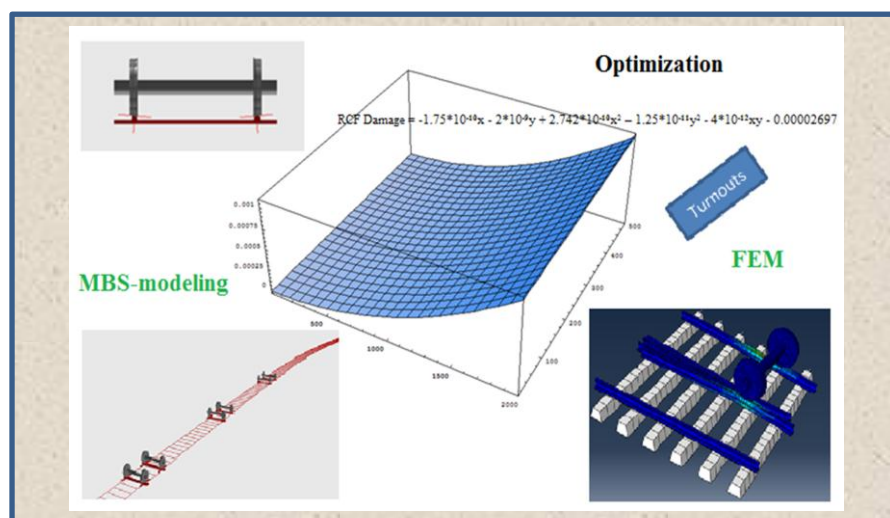
FATIGUE LIFE OPTIMIZATION OF SIMPLE TURNOUT IN BALLASTED TRACK

A Thesis in Railway Civil engineering

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A Thesis

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ABSTRACT

Turnouts are very important components in railway infrastructure for facilitating changes in direction and shift in track that trains are moving. Therefore, various discontinuities are introduced at turnouts which affect the dynamic interactions and increase the damage severity on the turnout and on the trains using the turnout. The annual investment of funds to construct and maintain a viable track system is enormous. Then optimum use of these funds is a challenge which demands the best technology available. Therefore, research on fatigue life optimization of simple turnout in ballasted tracks, which are the very critical elements in railway infrastructure, worth researching a lot.

In this research, multiple cases of virtual experiments, based on design of experiment approach have been done to come up with an optimal parameter combination which gives maximum fatigue life or minimum cumulative damage. Numerical mechanical dynamic simulation in an MBS environment SIMPACK on a computer run and then a finite element simulation finally done on ABAQUS based on the optimum parameter combination obtained in the MBS simulations to determine fatigue life. And lastly results have been compared between the MBS and FE dynamic simulation then interpretations made after that.

The results show that an optimum fatigue life will be obtained if the rail-pad vertical stiffness is 300MPa and effective sub-structure stiffness is 110MPa. And the optimum fatigue life is calculated to be 1.1years as per the MBS result and 1.2years from the FE result. Possible reasons for the discrepancy in life prediction have been forwarded with ways in maximizing the accuracy and precision.

Key word: Simple Turnout, Multi-Body-System, finite element, modeling and simulation, fatigue life, optimization.

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NOMENCLATURE

The nomenclatures in this report include abbreviations and symbols used in equations, longer naming's, acronyms, representation of quantities and parameters.

Abbreviations

AALRT	Addis Ababa light rail transit
3D	Three dimensional
CREC	China Railway engineering corporation
DEM	Discrete element model/method
DOE	Design of experiment
dof	Degree of freedom
FE	Finite element
FEM	Finite element method/ modeling
MBS	Multi-body system
ODE	Ordinary differential equation
RCF	Rolling contact fatigue
RS	Response surface
S2R	Shift2rail
S-N	Stress-Number of stress reversals

Symbols/ Notations

M	Mass matrix
D	Damping matrix

G	Gyroscopic matrix
k	Stiffness matrix
N	Circulatory matrix
a,b,c,d,f,g	Regressive coefficients of the surrogate model

CHAPTER 1 INTRODUCTION

Railway is one of the oldest transportation systems in the world after marine transportation and it is growing exponentially in sophistication, technology and innovations. It has a profound implication in the economy of a country and ease of life to its citizens. In Ethiopia, the system has been laid since the end of 19th century with a little crack for some less than two decades, recently, but now it has been revived to the better journey of the country's renaissance and in the near future the enlightenment is waiting with grace. All over the globe, there is an elevated interest in rail transport, especially due to tremendous concerns to the global ecosystem and environmental friendliness issues. Even, there are scientific research projects, initiatives and strategies to transfer the road, water born and air passengers to the promising rail system for many reasons and with a better safety and economy. Projects and initiatives like shift2rail (S2R), D-rail and MAINLINE are some of those that manifest these endeavors.

One of the special characteristics of rail transportation system is about being a guided system on land whether in the form of contact-based or contact-less way and having a restricted path to move. Contact-based rail transportation is the major way but recently things are evolving in high speed trains so contact-less railway systems are growing, like the Mag-Lev train technologies from Germany's Trans-rapid and Japanese high speed Mag-Lev trains. Also there are some other innovative train technologies besides the Mag-Lev system, like the aero-train. In general, the platform that trains move-on, in railway transportation system, is called railway track and there are many different kinds which are suitable for various natural as well as artificial conditions and/ or requirements including economic considerations.

Railway turnouts are the fundamental elements, used in railway track structures, for trains to change direction and at the end of the day get the job done with flexibility, of rail transportation. However, this comes with cost in increased damage and reduced comfort at this parts of a railway line. These are caused by unavoidable and built in for purpose discontinuities that exist within the turnout area. Therefore, the discontinuities cause a critical stress environment due to a higher dynamics going on in the turnouts and result in elevated damages like the mostly observed ones of plastic damage, wear and rolling contact fatigue (RCF). Studying turnouts is very complex but things can be

simplified and researched to get a very good deal of information and knowledge, and that is the one done in this master's thesis research by studying turnouts from the perspective of fatigue life optimization, with manageable parameters and modeling, in order to find out the best possible parameters for a simple ballasted turnouts. Knowing this will enlighten railway track engineers into whether ballasted turnouts could still have room for innovation in maintenance free paradigm, which is the one we look forward in railway engineering, or is it better we find a new track form and/or new designs to fulfill that.

1.1 Background

A fundamental characteristic of railways is the one degree of freedom in their movement in contrast to roads which are of two degree of freedom that make them change direction easily. However, as railways are one degree of freedom in nature, then switching component is required to change direction of train movement without interruption (V.A.PROFILIDIS 1995).

Recognizing the shortcomings of the existing design approach, new methods for the design and evaluation of turnout geometry are now being used. Turnout design has to be considered as vehicle/ track interaction problem. The method is to use a vehicle dynamics simulation to predict forces and accelerations throughout the turnout. The optimum geometry usually is a double clothoid or vertex clothoid geometry. One method to further reduce the forces that are created when a car enters a turnout and the wheel makes the transition from stock rail to switch blade is called kinematic gauge optimization (KGO) which make use of the rolling radius difference of the two wheels on an axle when it enters a turnout by bending the stock rail out. And calculations have predicted an advantage compared to conventional turnout design (Esveld 2001). The numerical simulation results in (Matjaz, Janko and Miha 2015) show that the elastic properties of rail-pad have biggest influence on the wheel impact forces and application of soft rail-pads gives a dynamic force reduction of 21% in the rail impact force (P1), 67% in sleeper forces and 28% in ballast forces is observed.

The railway track system is an important part of the transporting infrastructure of a country, and plays a significant role in sustaining a healthy economy. The annual investment of funds to construct and maintain a viable track system is enormous. And the

optimum use of these funds is a challenge which demands the best technology available (Selig and Waters 1995).

Track maintenance is the total process of maintenance and renewal required to ensure that the track meets safety and quality standards at a minimum possible cost which could be done manually, using human labor, or mechanically, using machines. It is in principle scheduled on the basis of control data from measuring systems, visual observation and financial-economic data and bearing in mind local conditions (Esveld 2001). In recent years more thought has been given to maintaining both the railway infrastructure and the rolling stock with the use of components which can be replaced after an estimated lifespan rather than seeking to mend or refurbish such components, leading to a maintenance free track (Matjaz, Janko and Miha 2015).

Many researchers working on turnout dynamics have strived to understand and solve the working principles and mechanisms of turnout damages and the way to optimize them from many perspectives like optimizing the crossing nose geometries, turnout stiffness and the overall geometry. However, conducting a study on the fatigue life optimization is the key to remedy the problem of switches being the weak links in a railway track infrastructure; as well provide a platform to come up with the best maintenance and operating schemes in rail transport. Therefore, the coming subsection discussed the purpose and significance of studying this.

1.2 Purpose of the study

In railway track infrastructure, the cost comes from construction expenses, maintenance expenses and operating expenses as in any other engineering facility. However, cutting or minimizing the maintenance cost without or with a very small increment in the other expenses, that are the construction and operating expenses, is very beneficial when the total cost is seen in a life cycle cost perspective. So, one of the main ultimate goals and recently sought concept in railway infrastructure maintenance is to come up with a minimum maintenance railway track, through a means of reducing damage in a track or finding a track form either with a minimum damage or maintenance free track within the design life/ period of the infrastructure. Of all the elements in railway track, the turnout is the very critical element with respect to maintenance frequency as well as maintenance cost due to its unique and dramatic dynamics.

Turnouts are very critical in the sense that damage happens, which might be due the following reasons:

- ✓ There are tighter curves because of restricted available area;
- ✓ Curves with transition and super-elevation are hardly possible;
- ✓ Existence of discontinuities at the switch points and crossing noses in addition to the nominal irregularities like any other section of railway networks.

Therefore, being the critical part and having the potential of being the bottle neck within any railway network system, turnouts require big attention and research for improvisations with various innovations ahead.

Fatigue life study is very important and this research has been based on its theories. And it is because, through fatigue study, it is possible to predict expected service life of turnouts and specifically the rail in this research; since it is located directly at the source of huge stresses due to wheel/rail interactions and irregularities, corrugations and discontinuities existence at their interface which induce even more stresses. As well many researches are done and also being made in the field of life estimations through fatigue theories and designs using fatigue theory are also evolving this days. Therefore, due to these facts, this research has got the impetus to be initiated in the form of service life optimization – to get the best thing possible – at the critical part of railway networks, which are the turnouts.

Therefore, the basic motivation of this study is the need for a better understanding on the identification of the best simple turnout possible, which accounts major maintenance cost, in ballasted track forms through fatigue life optimization study. This give insight into whether ballasted track forms could serve the purpose of maintenance free paradigm or not. This showed the limiting life of ballasted track forms at simple turnout section of railway infrastructure.

Therefore by doing this research, multitude of benefits and advantages are realized through the better understanding of whether the ballast type of turnouts could serve the purpose or not, with regard to the track being maintenance free or with a very minimal maintenance, which is the recently sought concept in railway infrastructure and technologies. Therefore, the research has the significance of being useful in:

- ✓ Better understanding of the turnout parameters effect on service life of simple turnouts and suggesting the best parameter combinations in turnout design and specification;
- ✓ Determining the limiting service life of ballasted simple turnouts so as to decide whether a maintenance free concept is possible or not during the design life of railway infrastructure;
- ✓ Paving the way to better design experiments, which are too expensive to do without prior knowledge in the outcomes, that validate the results and seek further unaccounted phenomenon;

1.3 Objective of the study

The study of fatigue life optimization of simple turnout ballasted track-forms need to answer the need for proper dynamic track models in the MBS and FE environment and incorporating near to reality fatigue life prediction models. In doing so, the following general and specific research objectives are set:

i. General objectives

The main objective of this study is to identify the parameters affecting the fatigue life and determine the optimized fatigue life of simple turnouts in order for deciding whether ballasted track-forms are feasible or not in satisfying the maintenance free paradigm.

ii. Specific objectives

The specific objectives in this research include:

- ✓ Identification of parameters affecting fatigue life of simple turnouts;
- ✓ Development of MBS model,
- ✓ Developing FE model
- ✓ Optimized life determination of simple turnout in ballasted track forms.

1.4 Methods

To perform this research, multiple cases of virtual experiments has been done, based on design of experiment approach to come up with an optimal parameter combination, using

numerical mechanical dynamic simulation in an MBS environment using lumped parameter modeling on a computer and then a finite element simulation finally been done based on optimum parameter combinations obtained in the MBS simulations, to compare the results from the MBS dynamic simulation and make interpretations after that on fatigue life of rails. The procedures to follow and resources to use are discussed in detail in the following paragraphs.

i. Material and Geometric properties

- ✓ Gather secondary data to be used in the MBS and FE modeling process;
- ✓ Use virtual FE experiments to find some material parameters when needed, based on some other already known material properties;

ii. Methodologies

The methodology to implement the research and come up with good result was by developing and numerically simulating 3D MBS dynamic model of simple turnout with different parameter combination from design of experiment approach, which was to represent various possible turnout scenarios, and identification of the best parameter combination. Then FE simulation has run based on the optimum parameter combination to validate the MBS simulation based optimization results and rigorously determine the fatigue life.

The step-by-step procedure of the approach used is outlined as follow:

a) MBS modeling

- ✓ Identifying the influential parameters;
- ✓ Modeling simple turnout in an MBS environment;
- ✓ Running the MBS model by varying the parameters based on the design of experiment approach;
- ✓ Extracting the important outputs from the MBS analysis including fatigue life and RCF calculations which has been used as objective functions in the optimization process;
- ✓ Finding the parameter combination with maximum fatigue life which is the properties of optimized simple turnout under possible constraints.

b) FE modeling

- ✓ Modeling the MBS based optimized turnout in FE environment i.e. ABAQUS;
- ✓ Run the simulation;
- ✓ Extract the important FE analysis outputs including the fatigue life through post processing;
- ✓ Compare the results from the MBS simulation results with the ones obtained in the FE simulation procedure and make interpretation.

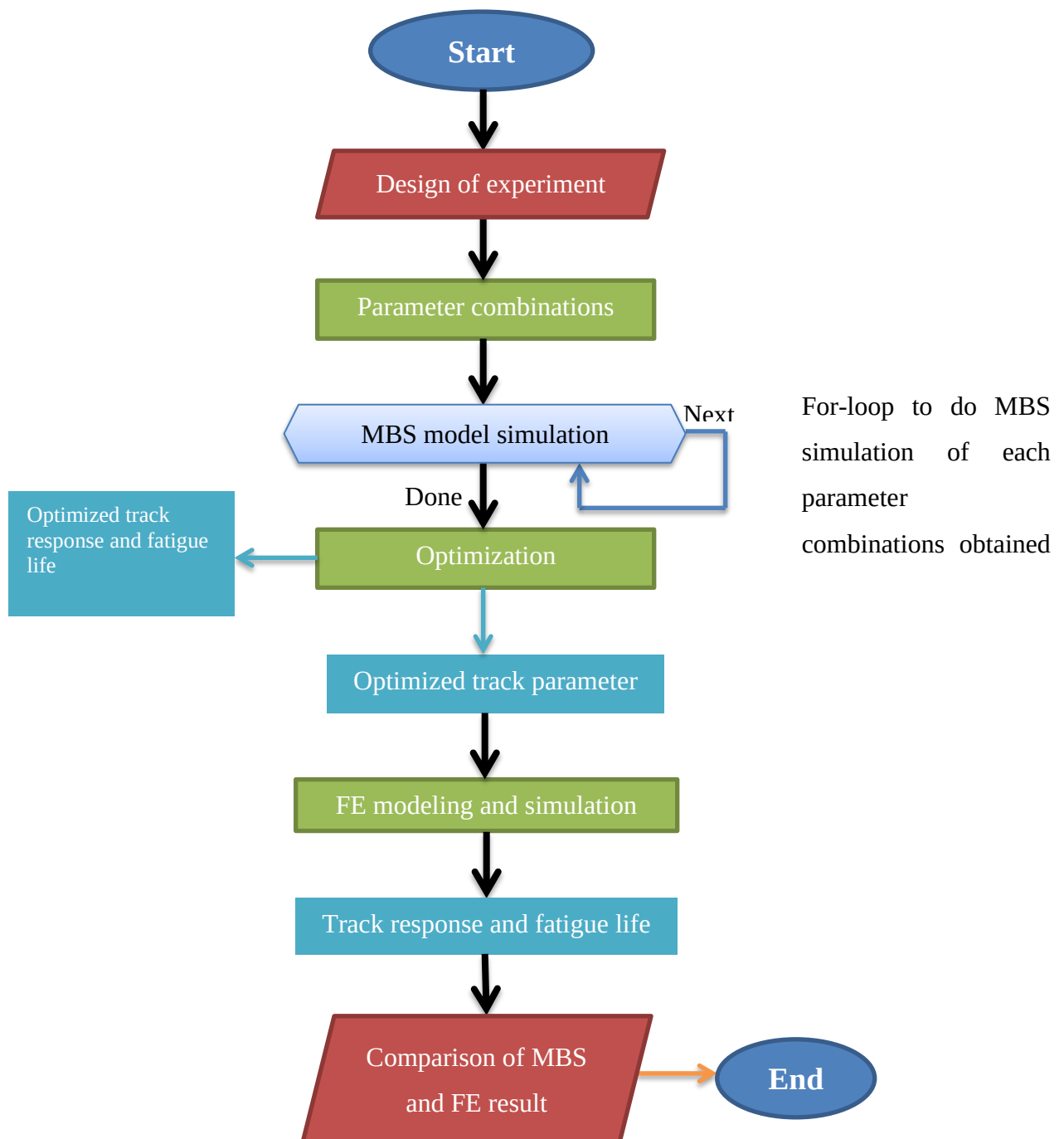


Figure 1-1: Flow chart- Algorithm for conducting fatigue life optimization

iii. Materials/ resources

The materials and/ or resources been used include hardware and software for the purpose of doing the virtual experiments of the developed model, both MBS and FE approaches, either in the case of the conceptual/ preliminary models or the final in detail established models. The resources required in achieving this research included:

- ✓ Computing hardware/ computer;
- ✓ Computing software which are ABAQUS for FE simulation and SIMPACK for MBS simulation, academic versions;

1.5 Scope and limitation

This research is mainly done to be useful and advance some of the knowledge in the domain of ballasted turnouts specifically for simple turnouts, so the scope of the study was applicable to such a domain of turnouts. And also the study is only made based on the rail fatigue life rather than considering the whole physical components of the turnouts like sleepers, ballasts and so on. Therefore, the research is done accordingly to successfully acquire the advancement of the knowledge required in this regard.

The work is completely based on computer simulation of turnout models in ballasted track forms, therefore experimental investigations were not conducted either to validate or gain an understanding on the behavior of such a complicated dynamical system. Even though it would have been better, but limitations in resource and available research duration make it an impossible undertaking. So the work is solely drawn from numerical simulation outcomes and their analysis.

The above scopes and limitations presented, at least, make the research outcomes to be useful for Ethiopian context, although not the only possibility, because those within the constraints and domains stated above could also make use of the results successfully, of the railway infrastructure built and operated for the better transportation system of anyone without particulars, utilizing the resources and time constraints of the research project of this thesis to the best.

1.6 Outline of the research

There are four main chapters with subsections in each, for the purpose of classifying the chunks into easily manageable contents.

In chapter one, it was mainly about introducing the research and giving the right way through the study. It discusses some background and provides answer for a question of why the research is required to be conducted. And also, objectives are set and methods layout.

In the second chapter, surveys of literatures on the area of turnout dynamics and life prediction are presented in order to lay down the bases for the research work of optimization of the fatigue life of turnouts after insightful discussion on tracks in general and turnouts in specific.

Then, the third chapter focuses on the modeling issues of vehicle-track interaction and goes through a discussion of the models proposed in the research for the fatigue life optimization of simple turnouts. In addition, modeling approaches and theories are straightened out alongside, at the beginning of this section.

The fourth chapter starts with some discussion on simulation theory and continued presenting the simulation results of the models and analysis outcomes for the presentation of parametric correlations and determination of the optimum life including the optimum parameter combinations which give the best possible life of rails in simple turnouts of ballasted track-forms. And finally, a summary has been given at the end of this section.

The last chapter finally concluded by giving conclusion and forwarding some further works to be made for the better fidelity outcomes we seek for the turnout technology in railway track engineering.

CHAPTER 2 LITERATURE REVIEW

In this section, presentations of literatures about railway turnouts and the related issues of fatigue life prediction theories, of the main theme of the research, are made with further topics of discussions on the design of experiment, optimization theories and surrogate modeling principles and procedures. Therefore, to that end, some introductory subsections have been given to railway tracks and turnouts discussion.

2.1 Railway track

The classical railway, basically, consist of flat framework made up of rails and sleepers, connected by fastenings, which are supported on ballast. And the ballast bed rests on a sub-ballast layer which forms the transition layer. These components and other structures such as switches and crossings are considered as part of the track (Esveld 2001)

Track guidance of the wheels is achieved in principle by making the following two provisions (Esveld 2001):

- The wheels conicity- a centering force exerted on the wheel-set when slight lateral displacement occurs and this promote a better radial wheel-set adjustment which leads to more rolling and less wear.
- Wheel flanges on the inside of the track- prevent derailment in case the above provision no longer sufficient in restricting lateral displacements adequately but when the wheel flange touch the rail head face, it result in high lateral forces and wear.

In addition to the provision of only conicity of wheel-rail and the wheel flange, for the purpose of wheel guidance, it is mostly provided with guard (check or wing) rails as a means of constraining the wheel movement in cases and situations of vehicle lateral undesirable movements during train motion. This circumstance happens mostly in the area of switches and crossings, in general called turnouts and especially in the switch panel due to a very high dynamic interaction and excitation. Therefore in places where guard rails are installed there might be contact between the back of wheel to guard (check or wing) rail's inside.

Due to the conical shape of the wheels, the effective rolling radius for each wheel will change as the contact point location changes. Here the difference in rolling radius between the left and right wheels is illustrated as a function of lateral wheel-set displacement. As the wheels in a standard wheel-set are rigidly connected via an axle, the rolling radius difference provides a counteracting steering effect as the wheel with the larger rolling radius will travel faster for a given rotational speed of the wheel-set. A non-dimensional measure of the influence of lateral wheel-set displacement on the difference in rolling radius, and thus steering, is obtained by the concept of equivalent conicity as shown in the equation below (Bjorn 2014):

$$\lambda_{eq} = \frac{r_r - r_l}{2\Delta y} = \frac{\Delta r_r - \Delta r_l}{2\Delta y} \quad \text{Equation 2-1}$$

Here r_r and r_l are the rolling radii of the left and right wheels and Δ indicates a change. Note that the rolling radius difference is typically a non-linear function of the lateral displacement which makes a linearized measure, that is only valid for a given lateral displacement amplitude (Bjorn 2014).

Early rails were made of wood, cast iron or wrought iron. Unlike some other uses of iron and steel, railway rails are subject to very high stresses and are made of very high quality steel. It took many decades to improve the quality of the materials, including the change from iron to steel. Minor flaws in the steel that may pose no problems in other applications can lead to broken rails and dangerous derailments when used on railway tracks. And all modern rails are hot rolled steel of a specific cross sectional profile. Typically the cross section (profile) approximates an I-beam but is asymmetric about a horizontal axis. The head is profiled to resist wear and to give a good ride, the foot is profiled to suit the fixing system (Rail profile - Wikipedia, the free encyclopedia 2015). There are many different rail profiles developed in the history of railway which are listed below and described as it is taken from (Rail profile - Wikipedia, the free encyclopedia 2015):

- Strap rail- The earliest rails were simply lengths of timber. To resist wear a thin iron strap was laid on top of the timber rail. This saved money as wood was cheaper than metal. The system had the flaw that every so often the passage of the wheels on the train would cause the strap to break away from the timber.

- Flange rail- was an early type of rail and had an 'L' cross section in which the flange kept an un-flanged wheel on the track. The flanged rail has seen a minor revival in the 1950s, as guide bars, with the Paris Metro (Rubber-tyred metro or French Métro sur pneus) and more recently as the Guided bus.
- Baulk rail- The rail was continuously supported by a baulk of timber running longitudinally. Because of the continuous support, the rail did not need to behave as a girder and so could be lighter.
- Barlow rail- was invented by William Henry Barlow in 1849. It was designed to be laid straight onto the ballast, but the lack of sleepers meant that it was difficult to keep it in gauge.
- Flat-bottomed rail- is the dominant rail profile in worldwide use.
- Flanged-T rail (T-section)- is the name for flat bottomed rail used in North America. Iron-strapped wooden rails were used on all American railways until 1831. Col. Robert L. Stevens, the President of the Camden and Amboy Railroad, conceived the idea that an all-iron rail would be better suited for building a railroad. There were no steel mills in America capable of rolling long lengths, so he sailed to England which was the only place where his flanged T rail (also called T-section) could be rolled. Railways in England had been using rolled rail of other cross sections which the iron masters had produced.
- Vignol rail- is the popular name of the flat-bottomed rail, in recognizing engineer Charles Vignoles who introduced it to Britain. Charles Vignoles observed that wear was occurring with wrought iron rails and cast iron chairs upon stone blocks, the most common system at that time. In 1836 he recommended flat-bottomed rail to the London and Croydon Railway for which he was consulting engineer. His original rail had a smaller cross section to the Stevens rail, with a wider base than modern rail, fastened with screws through the base.
- Double-headed rail- . The supposed advantage was that, when the head became worn, the rail could be turned over and re-used. In practice, this form of recycling was not very successful as the chair caused dents in the lower surface, and double-headed rail evolved into bullhead rail in which the head was more substantial than the foot.
- Bullheaded rail- Bullhead rail was the standard for the British railway system from the mid-19th until the mid-20th century. Bullhead rail is similar to double-headed

rail except that the profile of the head of the rail is not the same as that of the foot. Bullhead rail evolved from double-headed rail but, because it did not have a symmetrical profile, it was never possible to flip it over and use the foot as the head. Therefore, because the rail no longer had the originally-perceived benefit of reusability, it was a very expensive method of laying track. Heavy cast iron chairs were needed to support the rail, which was secured in the chairs by wooden (later steel) wedges or "keys" which required regular attention.

- Switch rail- the tips of the switch blades have to be planed down to fit snugly against the fixed or stock rail. Switch rail has to be planed, fitted against the stock rail and thicker base used to raise the tops level with each other. Tangential switch rail has less steel planed away, and the mid part of the switch "scallops" into the web of the stock rail for greater strength. The higher baseplate also supports the switch rail better. This makes a stronger switch which can be curved, reducing the jolt to the train and allowing higher speeds. However, tangential turnouts still lack a smooth transition where the switch blade contacts the main rail, so there is still some jolt as the train passes over. It's a much smaller jolt than with the old-style turnouts.
- Grooved rail- The flange-way has the railhead on one side and the guard on the other. The guard carries no weight, but may act as a checkrail. Grooved rail was invented in 1852 by Alphonse Loubat, a French inventor who developed improvements in tram and rail equipment, and helped develop tram lines in New York City and Paris. The invention of grooved rail enabled tramways to be laid without causing a nuisance to other road users, except unsuspecting cyclists, who could get their wheels caught in the groove. The grooves may become filled with gravel and dirt (particularly if infrequently used or after a period of idleness) and need clearing from time to time, this being done by a "scrubber" tram. Failure to clear the grooves can lead to a bumpy ride for the passengers, damage to either wheel or rail and possibly derailing.
- Girder guard rail- This rail is a modified form of flanged rail and requires a special mounting for weight transfer and gauge stabilization. If the weight is carried by the roadway subsurface, steel ties are needed at regular intervals to maintain the gauge. Installing these means that the whole surface needs to be excavated and reinstated.

- Block rail- is a lower profile form of girder guard rail with the web is eliminated. In profile it is more like a solid form of bridge rail with a flange-way and guard added.
- LR55 rail- is a special rail section, promoted by Tram Power of the UK, designed for use in embedded tramway track installations. The rail itself is like a lop-sided "V". It is a top-suspended rail: the load is transmitted by the rail head, rather than through a web to the rail foot.

When it became possible to preserve wooden sleepers with mercuric chloride (a process called Kyanizing) and creosote, they gave a much quieter ride than stone blocks and it was possible to fasten the rails directly using clips or rail spikes. Their use spread worldwide. The joint where the ends of two rails are connected to each other is the weakest part of a rail line. The earliest iron rails were joined by a simple fishplate or bar of metal bolted through the web of the rail. Stronger methods of joining two rails together have been developed. When sufficient metal is put into the rail joint, the joint is almost as strong as the rest of the rail length. The noise generated by trains passing over the rail joints, described as "the clickity clack of the railroad track", can be eliminated by welding the rail sections together. Continuously welded rail has a uniform top profile even at the joints (Rail profile - Wikipedia, the free encyclopedia 2015).

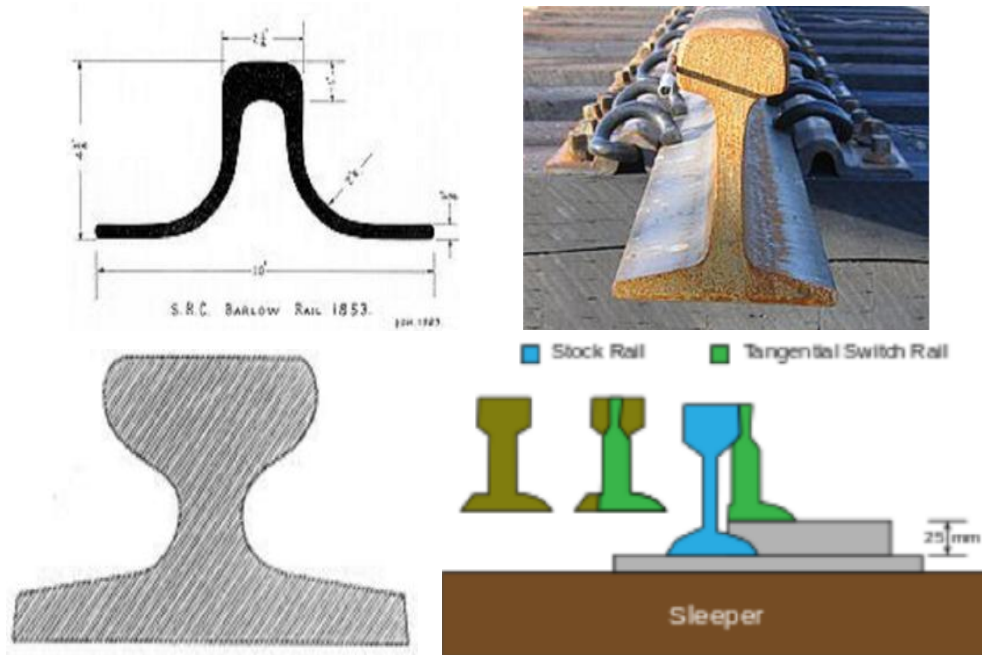


Figure 2-1: Typical rail profiles - Barlow, Flat-Bottomed, Vignol and switch rails - Source: (Rail profile - Wikipedia, the free encyclopedia 2015)

2.2 Turnouts

Turnouts are used to divide a track into two, sometimes three tracks, if a train is to pass from one track to another while moving and without being subdivided; and also essential in the absence of turntables or traverser (Esvelt 2001).

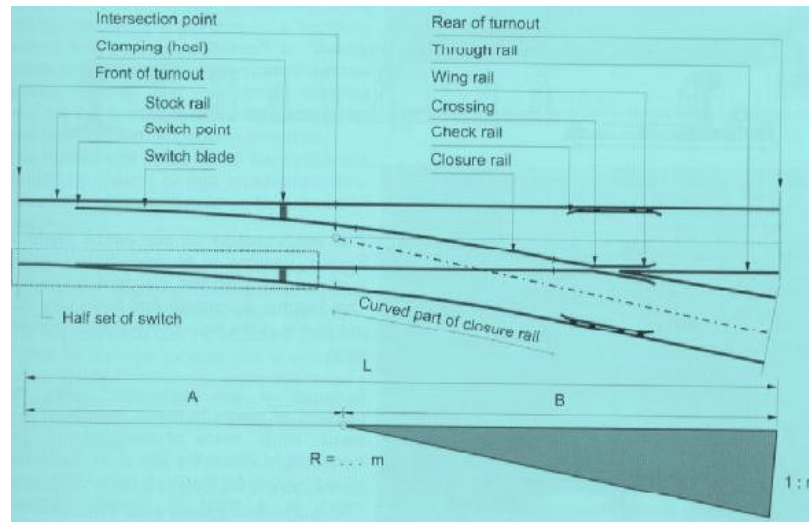


Figure 2-2: Standard right-handed turnout (Source: (Esvelt 2001))

Turnout features a straight section called the through route and a curved deviating part called the diverging route. The front of the turnout is defined as the start of the deviating curve in the switch panel. The switching function is realized by switching machines or actuators that position the switch rails according to the desired traffic route. The closure panel connects the switch and crossing panels, whereas the crossing panel allows for wheels to travel along both intersecting paths. Opposite to the crossing, and next to the adjacent through (stock) rails, are the check rails that enforce a constraint on the lateral position of passing wheel-sets. This is to avoid interference contact between wheel and crossing nose (Bjorn 2014).

There are two different types of turnout entry: tangential entry and radial entry. It describes the track route before the switch. In the tangential entry the switch tongue builds up as a tangent, parallel to the straight track before the radius of the diverging track begins. And in the radial entry the switch rail starts to build up within the curvature. High lateral contact forces are present between the wheel and rail as the train negotiates the diverging track (ANDERSSON 2012).

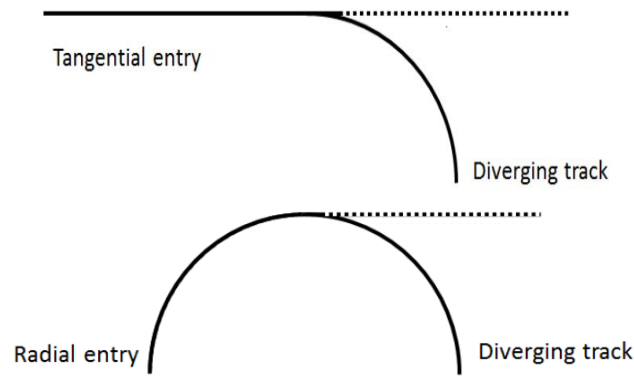


Figure 2-3: Turnout entry types (Source: (ANDERSSON 2012))

The variations and discontinuities in rail profiles, present in the turnout to achieve the function of the switch and crossing, result in an increased dynamic loading during wheel passage and thus increased degradation of these components compared to regular track. Further, turnouts are often built without transition curves, causing high vehicle jerk (time derivative of lateral vehicle acceleration) at entry and exit of the diverging route. Due to the planar nature of a turnout, track inclination (cant) to compensate for the lateral acceleration is not possible (Bjorn 2014).

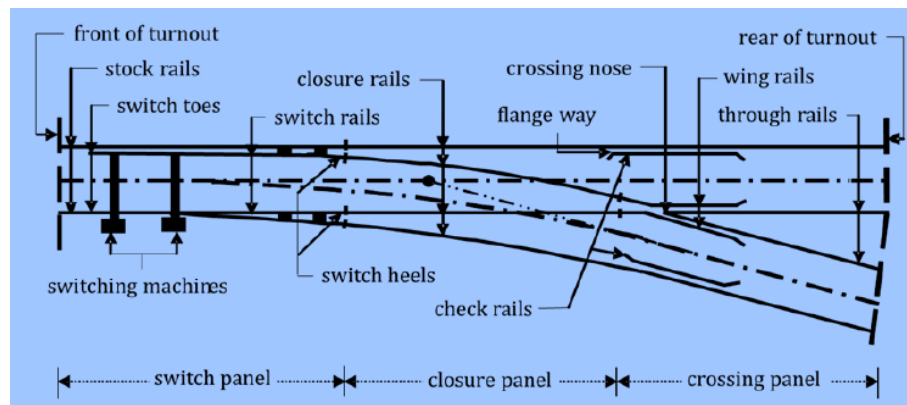


Figure 2-4: Schematic of turnout detail (Source: (Bjorn 2014))

Because it is impractical to plane down a rail to give a tangential entry to the turnout direction an entry angle is unavoidable. This is followed by the plane radius, which is effective over the length of switch rail which is planed away, i.e. that part of the switch rail which is touching the stock rail. The radius then changes to the switch radius which is a tangential curve to the through straight track, and finally onto the turnout radius, which is chosen to give a crossing angle within a range of standard values and is often equal to the switch radius. The radius of the turnout curve can then begin (A. HUNT 1986). Therefore, due to impracticality in laying out a perfect tangential geometry in turnouts, two main kinks are created which are the entry angle and crossing angle

(related to frog number). Then these kinks and rail discontinuities, resulting from rail cross section variability in turnouts, lead to very high dynamic loading that bring a higher rate of deterioration/ damage.

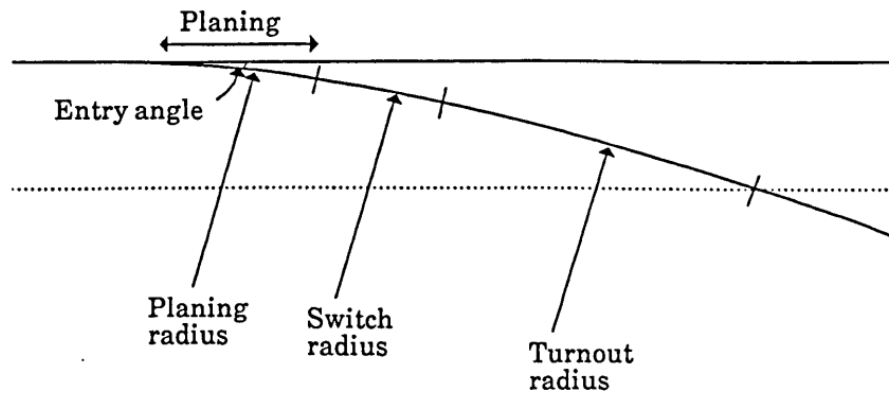


Figure 2-5: Typical turnout geometry (Source: (A. HUNT 1986))

2.3 Contact mechanics

The subject of contact mechanics may be said to have started in 1882 with the publication by Heinrich Hertz of his classic paper on the contact of elastic solids. At that time Hertz was only 24 and was working as a research assistant to Helmholtz in the University of Berlin. His interest in the problem was aroused by experiments on optical interference between glass lenses. The question arose whether elastic deformation of the lenses under the action of the force holding them in contact has a significant influence on the pattern of interference fringes. It is easy to imagine how the hypothesis of elliptical area of contact could have been prompted by observations of interference fringes. His knowledge of electrostatic potential theory then enabled him to show, by analogy, that an ellipsoidal Herzian distribution of contact pressure would produce elastic displacements in the two bodies which were compatible with the proposed elliptical area of contact (Johnson 2003).

Hertz presented his theory to the Berlin physical society in January 1881 when members of the audience were quick to perceive its technological importance and persuaded him to publish a second paper in technical journal. However, developments in the theory did not appear in the literature until the beginning of this century, stimulated by engineering developments on the railways, marine reduction gears and rolling contact bearing industry (Johnson 2003).

During simulation, an active constraint can become inactive or an inactive one becomes active. An active contact can be in the state of either sliding contact or rolling contact. Further it is possible to have transitions between the contact states (Peng 2002):

- (a) From rolling to sliding;
- (b) From sliding to rolling; and
- (c) From contact to no-contact

2.4 Fatigue theory

Fatigue does include some sort of exhaustion of the material which is manifested by material failure at fluctuating stress levels far below those leading to fracture at static loading. The provision is that the material is exposed to these fluctuating stress levels for a “long time”, or rather, for many cycles (Ekberg and Bjarnehed 1995).

In fatigue theory literatures, it is possible to find two main classes of fatigue life calculation approaches which are based on uniaxial fatigue theory and the multi-axial fatigue theory, which includes biaxial and tri-axial cases of loading. The uniaxial fatigue theory provides the bases for the whole theory of fatigue in structures, by extending it into those of the multi-axial theories of fatigue, in some way or another. And uniaxial fatigue theory is the simplest to deal with fatigue in mechanical structures like railway track in general and turnouts in specific. However, multi-axial fatigue theory is the realistic and accurate representation of fatigue mechanisms, surface fatigue or body fatigue, in structures such as turnouts which involve rolling contact at the wheel-rail interface and this contact phenomenon is best captured through multi-axial fatigue theory calculation. Since wheel-rail interaction in vehicle-track coupled dynamics simulations involve surface interactions between the rail and the wheel, and then tribological approach to deal with the fatigue life estimation can be practiced in a more computationally efficient perspective through surface fatigue studies using only handful of surface properties rather than the bulk material properties of the rail. And internal fatigue layers could be studied in a more detailed way through FEM in a virtual experimentation and/or in an X-Ray Fourier analysis of field experimentations/measurement campaigns.

Even though wear is the most frequent cause behind wheel re-profiling, this is a fairly harmless (although costly) form of deterioration. Fatigue failure on the other hand, are more violent in nature and may lead to the break-off of a large part. Consequences of such failures include damage to rails and sleepers and to train suspensions and bearings. In rare cases, even derailments may result. These failures may be very costly in terms of economics and human injuries as well as in reliability of train operations (Anders).

The behavior of solid materials responses to cyclic rolling contact loads has four phases of character (Johnson 2003):

- Perfectly elastic – rolling contact of solids up to the yield point. The stresses in rolling contact are given by Hertz theory provided the two contacting bodies are elastically similar.
- Yield onset – onset of yield in free rolling is the same as frictionless normal contact.
- Repeated rolling - Shakedown – When elastic limit is exceeded first, some plastic deformation will take place and introduce residual stresses then in the next passage of load the material is subject to the combined action of the contact stresses and previous residual stresses. It is possible that after a few passes of the load result in entirely elastic deformation called shakedown under repeated loading, whereby initial plastic deformation introduces residual stresses which make the steady cyclic state purely elastic.
- Repeated rolling – cumulative deformation – when the load exceeds shakedown limit, orthogonal plastic shear occur in the subsurface elements

Static collapse represents unlimited plastic deformation while the loads remain constant in time. If the loads are applied in a cyclic manner, without ever reaching the static collapse condition, other forms of collapse may occur. It may happen that plastic deformation in each cycle accumulates so that after enough cycles, the displacements are large enough to be equivalent to collapse; this is called incremental collapse. On the other hand, it may happen that no further plastic deformation occurs after one or a few cycles — that is, all subsequent unloading-reloading cycles are elastic. In that case the body is said to have experienced shakedown or adaptation. It is obvious that for bodies subject to repeated loading, shakedown is more relevant than static collapse, and that criteria for shakedown are of great importance. If the strain increments change sign in

every cycle, with yielding on both sides of the cycle, then alternating plasticity is said to occur; the net plastic deformation may remain small, but weakening of the material may occur nevertheless — a phenomenon called low-cycle fatigue — leading to breaking of the most highly stressed points after a certain number of cycles (Lubliner 2005). Shakedown concept is a branch of plasticity theory which deals with the irreversible response of structure to repeated loading (Fekadu 2014). In elasticity, the corresponding collapse mode to the low cycle fatigue is called high cycle fatigue and the number of cycles resisted before failure are in the order of 10^6 and above; that of the low cycle fatigue is below 10^6 cycles before collapse. The fatigue life estimation principles for the high and the low cycle fatigue are different; for the high cycle fatigue the S-N approach could be used and in the case of low cycle fatigue phenomenon rather the ϵ -N approach is applicable. However, for a very accurate work the critical plan approach and that of the fracture mechanics approach is recommended.

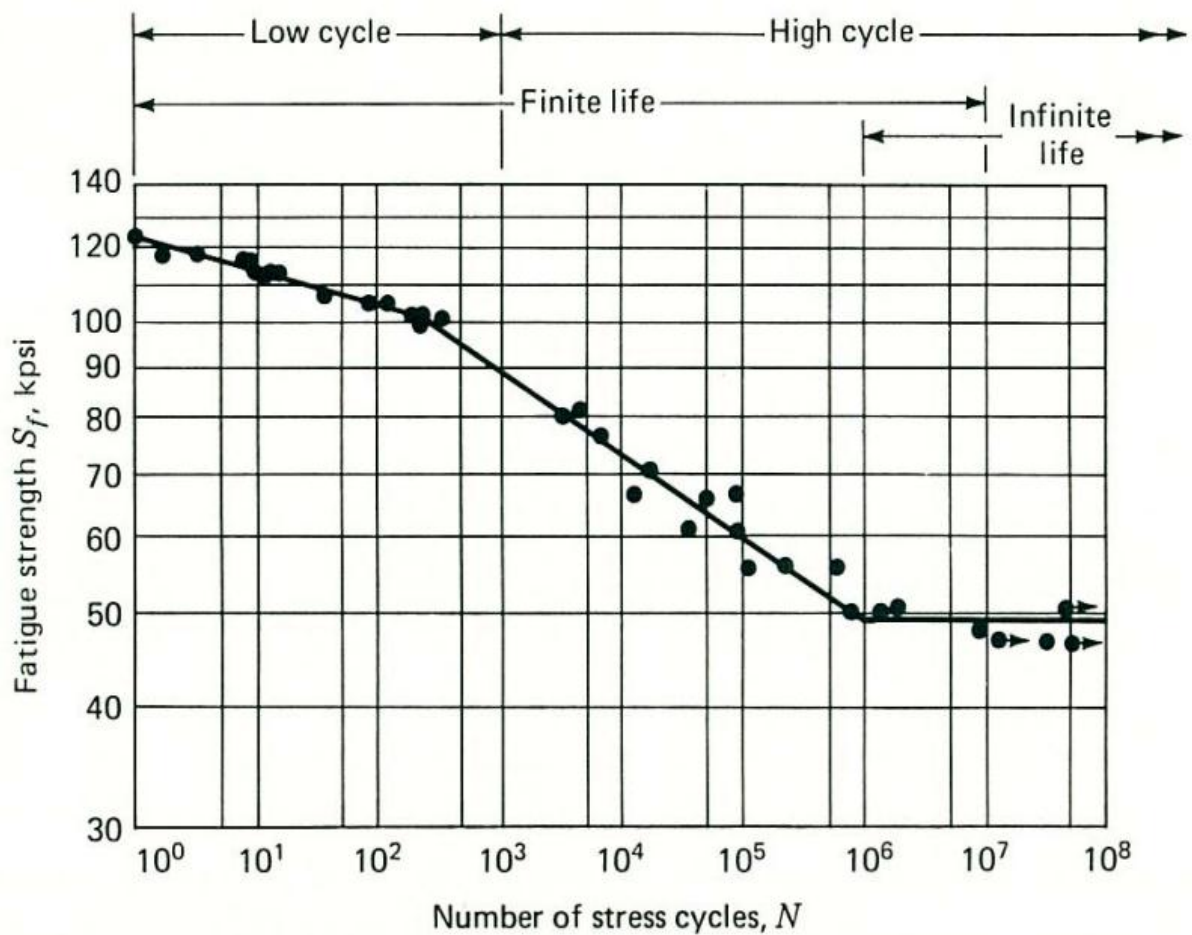


Figure 2-6: Approximate synthetic S-N relationship (Source: (Stone))

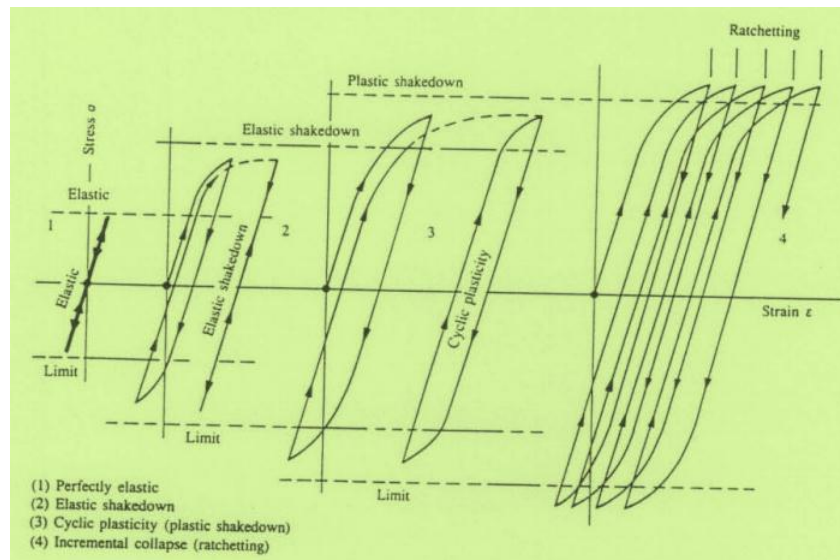


Figure 2-7: Rolling contact phases

In all mathematical modeling of physical processes, there is a need for experimental testing in order to develop, verify and calibrate the models. In the case of fatigue modeling, this need is especially high since the underlying physical phenomena are difficult to understand. Besides, several different phenomena are interacting, which often calls for an advanced statistical treatment of the collected test results (Ekberg and Bjarnehed 1995).

Computer aided fatigue evaluation of engineering components/structures consists of two main steps: dynamic stress computation and fatigue life prediction. Dynamic stress histories can be obtained either from experiments (mounting sensors or transducers on a physical component) or from computer simulation. The simulation-based approach is usually done by performing finite element analysis of the component under the specified set of applied loads. Then fatigue life prediction is carried out as a post-processing step of finite element output results (Andrea, Nanuel and Andrea 2003).

2.5 Optimization

Optimization is central to any problem involving decision making, whether in engineering or in economics. The task of decision making entails choosing between various alternatives. This choice is governed by our desire to make the "best" decision. The measure of goodness of the alternatives is described by an objective function or performance index. Optimization theory and methods deal with selecting the best

alternative in the sense of the given objective function. The area of optimization has received enormous attention in recent years, primarily because of the rapid progress in computer technology, including the development and availability of user-friendly software, high-speed and parallel processors, and artificial neural networks (K.P.Chong and H.Zak 2001).

In the theory of optimization, two major classes of optimizations could be identified, based on the number of objective functions used for optimization, as single-objective optimization and multi-objective optimization. In a single-objective optimization, simply searching for the best will be made based on a single objective function, where as in multi-objective optimization the searching is needed to be supported by Pareto dominance. And in the sense of mechanics, structural engineering or civil engineering, there are two main categories of optimizations which are those of structural optimizations and non-structural optimizations. In general however, both the categories involve searching algorithms, in one way or another, operating on a collection of design spaces. Further, structural optimization is classified into shape-optimization and topology-optimization. Each of these concepts are present in standard structural optimization literatures.

2.6 Design of experiment

Design of experiments was invented by Ronald A. Fisher in the 1920s and 1930s at Rothamsted Experimental Station, an agricultural research station 25 miles north of London. In Fisher's first book on design of experiments he showed how valid conclusions could be drawn efficiently from experiments with natural fluctuations such as temperature, soil conditions, and rain fall, that is, in the presence of nuisance variables. The known nuisance variables usually cause systematic biases in groups of results (e.g., batch-to-batch variation). The unknown nuisance variables usually cause random variability in the results and are called inherent variability or noise. Although the experimental design method was first used in an agricultural context, the method has been applied successfully in the military and in industry since the 1940s. Besse Day, working at the U.S. Naval Experimentation Laboratory, used experimental design to solve problems such as finding the cause of bad welds at a naval shipyard during World War II. George Box, employed by Imperial Chemical Industries before Coming to the

United States, is a leading developer of experimental design procedures for optimizing chemical processes. W. Edwards Deming taught statistical methods, including experimental design, to Japanese scientists and engineers in the early 1950s at a time when “Made in Japan” meant poor quality. Genichi Taguchi, the most well-known of this group of Japanese scientists, is famous for his quality improvement methods. One of the companies where Taguchi first applied his methods was Toyota. Since the late 1970s, U.S. industry has become interested again in quality improvement initiatives, now known as “Total Quality” and “Six Sigma” programs. Design of experiments is considered an advanced method in the Six Sigma programs, which were pioneered at Motorola and GE (Telford 2007).

The fundamental principles in design of experiments are solutions to the problems in experimentation posed by the two types of nuisance factors and serve to improve the efficiency of experiments. Those fundamental principles are, as it is shown and described in (Telford 2007):

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- Randomization
- Replication
- Blocking
- Orthogonality
- Factorial experimentation

Randomization: is a method that protects against an unknown bias distorting the results of the experiment.

Replication: increases the sample size and is a method for increasing the precision of the experiment.

Blocking: is a method for increasing precision by removing the effect of known nuisance factors.

Orthogonality: in an experiment results in the factor effects being uncorrelated and therefore more easily interpreted.

Factorial experimentation: is a method in which the effects due to each factor and to combinations of factors are estimated.

2.7 Surrogate modeling

surrogate models can be regarded as approximation models for the cost function (s) and state function (s), which are built from sampled data obtained by randomly probing the design space (called sampling via Design of Experiment (DoE)). Once the surrogate models are built, an optimization algorithm such as Genetic Algorithms (GA) can be used to search the new design (based on the surrogate models) that is most likely to be the optimum. Since the prediction with a surrogate model is generally much more efficient than that with a numerical analysis code, the computational cost associated with the search based on the surrogate models is generally negligible. Surrogate modeling is referred to as a technique that makes use of the sampled data (observed by running the computer code) to build surrogate models, which are sufficient to predict the output of an expensive computer code at untried points in the design space. Thus, how to choose sample points, how to build surrogate models, and how to evaluate the accuracy of surrogate models are key issues for surrogate modeling (Zhong-Hua and Khe-Shi).

CHAPTER 3 NUMERICAL METHODS AND CONDITIONS

This section of the report presented the modeling issues after background discussion on modeling theories and procedures. Modeling approaches for turnout dynamics have been clearly presented. In addition, presentations on modeling of materials, interactions, boundary conditions and model reduction approaches have been made with final subsections dedicated to the experimental design and model development of simple ballasted turnout which is the first stage of this thesis's research, simple ballasted turnout model development in MBS and FE frameworks.

3.1 Background

The first step in conducting a significant simulation project is to ensure that adequate attention has been directed toward understanding what is to be accomplished by performing the study. These activities are known as the problem formulation process and consist of (Christopher 2004):

- A formal problem statement
- Orientation of the system
- Establishment of specific project objectives

There are two common tools available to the practitioner for assisting with the problem statement. These are not actually problem-solving tools but rather are problem identification tools. Although these tools were originally developed for the manufacturing environment, they can easily be adapted to other sectors of industry. These tools are (Christopher 2004):

- ✓ Fishbone chart
- ✓ Pareto chart

The Fishbone chart- the purpose of this chart is to identify the cause of the problem or effect of interest. The Fishbone chart looks similar to the bones of a fish. The head of the fish is labeled with the problem or effect. Each major bone coming out of the spine is a possible source or cause of the problem (Christopher 2004).

The Pareto chart-It may turn out that there are several sources or causes of the problem or problems of interest. The concept of the Pareto chart was developed by the Italian economist Wilfredo Pareto, who postulated that only a few factors are the cause of many problems. This is frequently referred to as the 80–20 rule. The 80–20 rule states 80% of the problem is caused by 20% of the factors. The 80 and the 20 are not magic numbers but seem to be approximately observed in many instances (Christopher 2004).

Once the problem has been properly formulated, the next step in simulation modeling is to direct the attention toward defining the system that should be modeled (Christopher 2004). The objective of the preceding system definition and model formulation phase was to determine which components of the system should be included in the model and how the model should flow. Then model translation phase is used to translate the system into a computer model that can be used to generate experimental data. This is a two-step process. The first part of this process requires that the practitioner decide what type of computer program to utilize to model the system. The second part of the model translation phase is to actually perform the programming/modeling of the simulation model (Christopher 2004).

During the model translation phase, it will be naturally interesting to ensure that the simulation model has all the necessary components and that the model actually runs. In reality, we are interested in getting the model not just to run but to run the way we want it to. In other words, we are interested in ensuring that the model operates as intended. This process is known as model verification. Another way to look at the verification processes is to consider it as: “Building the model correctly”. For some reason, even very experienced practitioners confuse model verification with model validation. Whereas verification is the continuous process of insuring that the model operates as intended, validation is the process of insuring that the model represents reality. Validation can be considered as: “Building the correct model”. And it is pointless, perhaps impossible, to attempt to see if the model represents reality if the model does not even operate as intended. In other words, you should not attempt to validate a model that has not successfully undergone the verification process (Christopher 2004).

The validation of a model is the process through which one ensures that the model yields a valid representation of the real system. Absolute standards for validation that would be applicable to all models under all circumstances cannot be set. However, whether or not

a particular model is valid depends on how and where it is to be used. Model validation is an open-ended question. For example, a model may be considered to be valid for a preliminary design if it shows the correct trend; however, it may not be suitable for a final design. Likewise, a model may be valid under one set of operating conditions but not under another. The validation criteria are a uniquely defined set of considerations used to decide whether the agreement between the model predictions and test results is reasonable enough to consider the model to be valid. The validation criteria must be selected before one proceeds with the validation process. These selection criteria must be based on the model's purpose and a full understanding of the dynamic behaviors of the vehicle to be tested (Rao and Vijay 1984). The selection of the validation criteria involves four elements, which must be followed in sequence:

1. Which dynamic response variables to compare,
2. Which measurements (or statistics) of each variable to examine,
3. What tolerance level to apply to each, and
4. What mathematical form to use to combine the individual measurements.

The dynamic response variables should be those that are most important for the ultimate use of the model (i.e., the primary outputs) and those intermediate outputs that are capable of revealing the model's deficiencies (Rao and Vijay 1984).

Railway models are so numerous and diverse that it is difficult to generalize about them. The complexity and level of detail vary widely from model to model, depending on the purpose and requirements of the model. However, because of the dynamic processes represented by models and the analytical techniques used in their solutions, it is possible to categorize the railway models within each of these categories. The models can vary greatly in their levels of details, such as the number of degrees of freedom, the nonlinearities that are included, and so on. There are generally three types of analytical techniques often used to solve railway dynamic models. These techniques include the following (Rao and Vijay 1984):

- *Quasi-static techniques*: are the simplest to validate and often used in the steady state models designed to predict the dynamic equilibrium performance.
- *Frequency domain analysis techniques*: often based on linear assumptions; however, they can be applied to nonlinear systems by using quasi-linearization

techniques. Eigenvalues and eigenvectors are used to determine the natural frequencies, damping ratios, and mode shapes of the vehicle's response.

- *Time domain techniques*: are the most difficult to validate and very often used for both linear and nonlinear dynamic systems. Time domain models produce a sampled time history of each response variable and any auxiliary variables.

3.2 Modeling approaches in turnout dynamics

Dynamic systems have an important property that they have MEMORY; which means their output at a given time is dependent on past inputs (M. Turner ,Spring Semester). In railway mechanics there are two major modeling tools in studying the dynamics of tracks and their interaction with trains or vehicles. These modeling approaches are the MBS modeling approach and the FE modeling approach. And in the next sub-section discussion on their theory and modeling is given.

3.2.1 MBS modeling

Multi-body dynamics is based on classical mechanics and has a long and detailed history. The simplest multi-body system is a free particle which can be treated by Newton's equations published in 1686. D'Alembert considered a system of constrained rigid bodies where he distinguished between applied and reaction forces. A systematic analysis of constrained mechanical systems was established by Lagrange. Modern methods for the dynamic analysis of constrained multi-body systems fall into two main categories: differential algebraic equations (DAEs) and ordinary differential equations (ODEs). DAEs employ a maximal set of variable to describe the motion of the system and use multipliers to model the constraint forces. Pre-multiplying the constraint reaction-induced dynamic equations by the orthogonal complement matrix to the constraint Jacobian results in the governing equations as ODEs. Numerous advances have been made during the last couple of centuries in theory and in methods of formulating the equations of motion. Among these, Kane and Levinson introduced the concept of partial velocity which leads to a compact description of multi-body systems represented by a minimal set of first order ODEs (Peng 2002). In general, there are three formalisms of classical mechanics to represent any classical mechanics system which are the Newtonian formalism, Lagrangian formalism and Hamiltonian formalism in their chronological order discovery.

The dynamical analysis of multi-body systems is closely related to vibration theory. The dynamical phenomena are classified according to the linear and nonlinear equations of vibration (Schiehlen 2006). For engineering applications mechanical vibrations of holonomic, rheonomic systems are most important. For small vibrations the equation of motion can be linearized from the generic non-linear equation of motion as (Schiehlen 2006):

$$M(\mathbf{y}, t) \ddot{\mathbf{y}} + \mathbf{k}(\mathbf{y}, \dot{\mathbf{y}}, t) = \mathbf{q}(\mathbf{y}, \dot{\mathbf{y}}, t) \quad \text{Equation 3-1}$$

- For the holonomic systems

$$\dot{\mathbf{y}} = \dot{\mathbf{y}}(\mathbf{y}, \mathbf{z}, t) \quad M(\mathbf{y}, \mathbf{z}, t) \dot{\mathbf{z}} + \mathbf{k}(\mathbf{y}, \mathbf{z}, t) = \mathbf{q}(\mathbf{y}, \mathbf{z}, t) \quad \text{Equation 3-2}$$

- For the non-holonomic systems; explicit non-holonomic constraints are required

In classical mechanics a system may be defined as holonomic, if all constraints of the system are holonomic which means it could be expressible as a function and depends only on the position coordinates $\mathbf{x}_j$ and time t but not the velocities. A constraint and/ or a system that is not holonomic is called non-holonomic constraint and/ or system (Holonomic constraints -Wikipedia, the free encyclopedia 2015). Constraints are further classified according to whether the equations of constraint contain the time as an explicit variable called Rheonomous or are not explicitly dependent on time called scleronomous. Generally constraints introduce two difficulties in the solution of mechanical problems. The first being the coordinates, $\mathbf{r}_i$, are no longer all independent since they are connected by the equations of constraint; hence the equations of motion are not all independent. The second being the force of constraint is not furnished a priori. They are among the unknowns of the problem and must be obtained from the solution we seek. Indeed, imposing constraints on the system is simply another method of stating that there are forces present in the problem that cannot be specified directly but are known rather in terms of their effect on the motion of the system (Herbert, Charles and John 2000)

A system may feature parametrically excited vibrations due to the time-varying, often periodic matrices. However in the case of time-invariant matrices, one gets after decomposition into symmetric and skew-symmetric parts:

$$M\ddot{y} + (D + G)\dot{y} + (k + N)y = h(t) \quad \text{Equation 3-3}$$

The system will be a forced vibration if the external force excitation is different from zero and otherwise it becomes free vibration system. If the damping matrix D , the gyroscopic matrix G , and the circulatory matrix N are missing, a conservative system with free un-damped vibrations is found as (Schiehlen 2006):

$$M\ddot{y} + ky = 0 \quad \text{Equation 3-4}$$

On the other hand, nonlinear time-variant mechanical systems even with one dof may show chaotic vibrations (Schiehlen 2006). Chaotic processes are deterministic in nature for a non-linear systems; however they are very sensitive to changes or variation to inputs and produce a very unpredictable output, such as the famous butterfly-effect in climate models, which makes them look like random but they are not, because randomness is the characteristics of stochastic processes and not of a deterministic process or system.

3.2.2 Finite element modeling approach

Finite element procedures are at present very widely used in engineering analysis, we can expect this use to increase significantly in the years to come. The procedures are employed extensively in the analysis of solids and structures and heat transfer and fluids, and indeed, finite element methods are useful in virtually every field of engineering analysis. Its development for the solution of practical engineering problems began with the advent of digital computer. That is, the essence of finite element solution of governing algebraic equations is established and solved, and it was only through the use of digital computer that this process could be rendered effective and given general applicability. These two properties – effectiveness and general applicability in engineering analysis- are inherent in the theory used and have been developed to a high degree for practical computations, so that finite element methods have found wide appeal in engineering practices. As is often the case with original development, it is rather difficult to quote an exact date of invention but the roots can be traced back to three separate research groups: from an applied mathematicians, physicists and engineers. However the method obtained its impetus from the developments of engineers like J.H.Argyri, S.Kelsey, M.J.Turner, R.W.Clough, H.C.Martin, and L.J.Topp. Since the early 1960s, a large amount of research has been devoted to the technique. The method

in engineering was initially developed on a physical basis for the analysis of problems in structural mechanics. However, it was soon recognized that it could be applied equally well to the solutions of many other classes of problems like heat and fluid flow problems (Bathe 1996).

3.2.3 Hybrid modeling approach

Due to some limitations in computational resources, there is a need in combining the two most prevalent modeling techniques in such a way that the total computational effort required to do the necessary simulations and obtained the important outcomes. In addition, large displacement simulations can be made in the MBS approach and further detailed stress analysis run in a FEM. Therefore, to this effect simulation data in one of the two modeling methods is required to be used in the other and so standard co-simulation files are developed like the SID file in SIMPACK which could be used to import the super-elements of the modal analysis from FEM code for the incorporation of non-rigid modes into the MBS simulation run of flexible multi-body system models.

3.3 Material modeling

In material engineering materials are modeled at various scales and different frameworks in time and space. However, it might be sometimes important to model materials at a multi-scale level; micro, meso and macro scales; with respect to spatial and temporal dimensions. And the different frameworks/theoretical approaches used in material modeling included, but not limited, are continuum approach, rheological approaches, molecular dynamic approaches, dislocation approaches and quantum mechanics approaches. The figure underneath depicts the multi-scale material modeling.

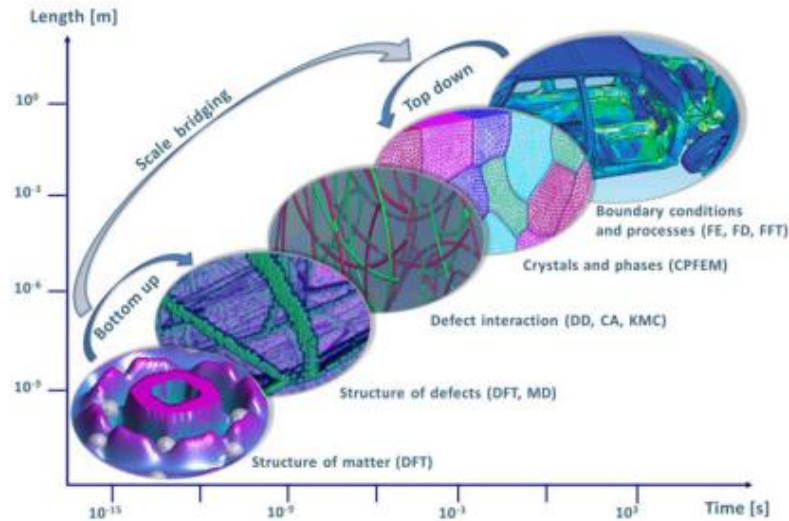


Figure 3-1: Material modeling at various scales (Source: (Raabe, et al.), DFT: *density functional theory*; MD: *molecular dynamics*; DD: *dislocation dynamics*; CA: *cellular automata*; KMC: *kinetic Monte Carlo*; CPFE: *crystal plasticity finite elements*; FE: *finite elements*; FD: *finite difference*; FFT: *Fast Fourier transforms*)

The continuum and rheological frameworks are the mostly utilized material modeling approaches in mechanical dynamic modeling and simulation. In the continuum approach materials could be modeled as elastic, plastic, elastic-plastic and some other linear and non-linear materials characters. And in rheological approach materials could be modeled as spring and damper systems of many kind and character or combination of these two, in various ways.

In a macro-scale, materials could be modeled through continuum approach; in meso scale through dislocation approach; and in micro scale molecular dynamic approaches and quantum mechanics approaches could be utilized. However, in most cases of railway track modeling, the continuum way of material modeling at a macro-scale is deployed. And in this research the material modeling in the MBS framework utilizes the macro-scale rheological material models through spring and damping of various natures and in the FE framework materials are modeled through linear-elastic behavior.

3.4 Boundary condition modeling

Boundary conditions are constraints imposed on a dynamic system model that is formulated as a boundary value problem in its mathematical modeling. And in a physical

sensation they could represent joints of various kinds such as free end, pin, fixed and others. However, there are also many more boundary conditions that could capture specific and special boundaries like infinity boundary conditions which have a character of no stress wave reflection. Such special boundary condition modeling is very difficult but with some compromises on accuracy of results the modeling can be achieved in a relatively simple way.

The boundary conditions for the turnout model developed in the MBS environment is a free end at the rail extremes, in the model entry and exit, with some extension beyond the turnout area to find the dynamic responses at the extremities negligible and dissipate the waves present in the rails, beyond the turnout region. The extension extent could be found by using simplified BOEF track model response mathematical expressions or from previous studies on the extent of a track where the structural responses become negligible. The detailed explanations and their calculations could be found in the appendices. On the other hand the boundary condition in the FE model used an infinite boundary condition using infinite elements. Infinite boundary conditions are radiating or silent boundary conditions where the waves coming to the boundary are not reflected rather absorbed. In general, many mechanisms have been developed to capture the idea of radiating boundaries which is the general class of boundary with wave non-reflecting character and among the many proposed approaches found in literature include PML (perfectly matching layers) technique, absorbing layer and infinite element. Even though many attempts are made, perfect radiating capabilities are hard to achieve and some percentage of the waves coming to the boundary are reflected which will act like if an external energy source exist to act upon the system; some of the techniques are better than others in some cases than others and finding a perfect one recipe for all cases is hard to find.

3.5 Interaction modeling

Wheel-rail and other Contact which could be contact or contact-less interaction; however the conventional railway track system of those of contact based interactions are much prevailed in our railway civilization than the contact-less ones such as the Mag-Lev guidance system which are installed in few locations in the world like in one of the Chinese busiest cities, Shanghai.

Contact between fixed components such as sleep-rail, rail pad-rail, ballast-sub ballast, and sub ballast-roadbed could be modeled as normal-tangential interactions or interaction coordinate constraints through kinematics.

Interactions could be between moving–moving, moving-nonmoving or nonmoving-nonmoving components of vehicle/ track interaction models; for instance, wheel-rail contacts are moving-nonmoving component interaction type. In general, moving-nonmoving and moving-moving component interactions could be modeled as normal-tangential interaction in the form of Herzian or non-Hertzian contact interaction which might consider frictional and adhesive effects as well in one way or another; and that of the nonmoving-nonmoving components interaction could be modeled as spring-dashpot, normal-tangential interaction or as a Winkler foundation if the second fixed component is a foundation like the roadbed of railway track structures.

In wheel to rail interaction, due to contact input data and solution method sensitivity, a non-Hertzian multi-point contact theory particularly the Kalker's contact theory is deployed for a better and successful overall simulation in turnouts which involves geometric non-linearity. Multi-point contact behavior is natural contact phenomenon in turnouts due to rail cross section variability. The detailed kalker's coefficients calculation and its theoretical background could be found in the appendices. Interaction between layers and multi-body systems is made through spring-dashpot interaction and Winkler foundation for track foundation/ subgrade modeling. However, in some cases constraint interactions between layers could be used to minimize the computational effort requirement.

3.6 Model reduction theory

An important tool in the design of complex high-tech systems is the numerical simulation of predictive models. However, these dynamical models are typically of high order, i.e. they are described by a large number of ordinary differential equations. This results from either the inherent complexity of the system or the discretization of partial differential equations. Model reduction can be used to find a low-order model that approximates the behavior of the original high-order model, where this low-order approximation facilitates both the computationally efficient analysis and controller design for the system to induce desired behavior (Besselink, et al. 2013).

In the context of full simulation, numerical effort is an important issue. Due to the long simulation times when computing durability or comfort loading, the number of degrees of freedom needs to be kept small. In this field, multi-body simulation methods are extensively used and well established. By modeling most parts as rigid bodies, the behavior of a vehicle can be described with very few degrees of freedom. However, some applications require the inclusion of flexible components (Herkt, Drebler and Pinnau 2009). In this case, a hybrid simulation modeling, involving the multi-body technique in combination with that of the FE technique, will be used in order for solving the simulation problem effectively and efficiently with less computational effort requirement to do the job done.

3.7 Design of experiment

This part of the work is concerned about identifying causal parameters that affect the fatigue life of rails in turnout area and it involves in developing fish-bone diagram for making the work manageable in parameter identification. And then the most influential parameters are selected according to their Pareto dominance in a Pareto chart and the focus was on turnout related parameters because the interest of this thesis work is on the turnout best property determination in relation to fatigue damage; the parameters used for the development of design of experiment are the three Pareto dominant turnout related parameters and they are discussed in the following subsections.

3.7.1 Parameter identification

In addition to be specific in its scope, this particular research, there are some setout/ fixed parameters, in order to reduce the number of parameter combinations and then the possible simulation runs required to deduce on the optimization of fatigue life, that are due to reasons of limited resource and the available research duration, which included the following in this research:

- Speed limitation of 20 km/hr;
- AALRT Train to be used for modeling;
- Rail of 50 Kg/m U71Mn (Tensile strength=880Mpa from Chinese standard);
- For a specific sleeper, TYPE-II pre-stressed concrete sleeper spaced at 0.6m, matching the Ethiopian railway case;

- Standard gauge, 1435mm;
- Turnout type to be optimized is #9 which is used for speeds less than or equal to 30Km/h like 20Km/h speed limit in the AALRT turnout system;
- Right branching Turnout
- Under the assumption of equivalent traffic proportion in both routes;

Table 3-1: Turnout #9 geometric parameters from Chinese railway standard

Frog angle (deg.)	Lead curve central radius (m)	Distance between beginning and center of turnout (m)	Distance from center of turnout to heel of frog (m)	Total turnout length (m)	permissible speed (Km/h)
6°20'25"	180	13.839	15.009	28.848	30

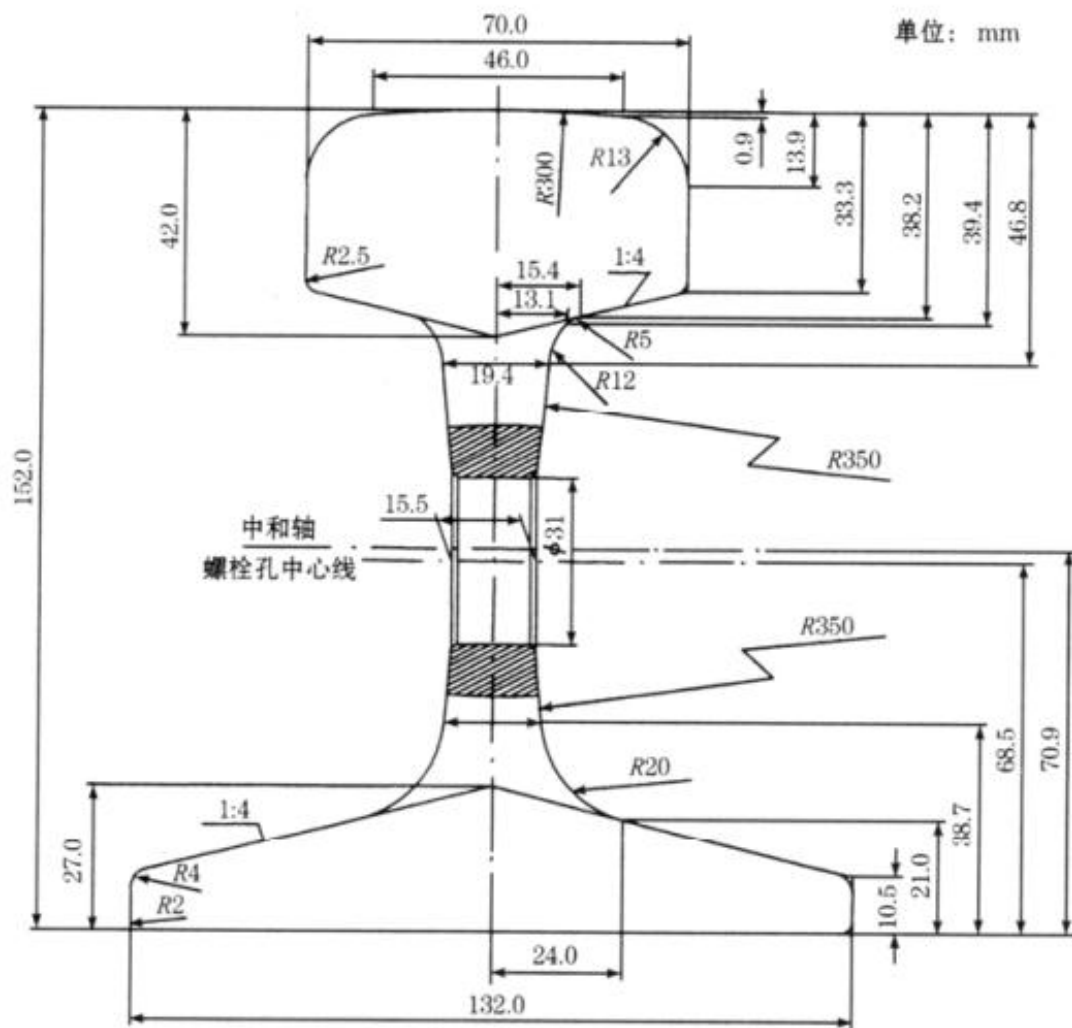


Figure 3-2: 50Kg/m rail cross-sectional geometry [mm, one digit precision] (source: B17)

Table 3-2: Vehicular geometric data of AALRT system from CREC

Length of car body:	≤30000 mm
Height of vehicle roof from top of rail (excluding pantograph):	≤3700 mm
Maximum width of car body:	2650 mm
Height of vehicle floor from top of rail (low floor area, new wheels and empty load)	≤380 mm
Height of vehicle floor from top of rail (exit and entry areas, new wheels and empty load)	≤350 mm
Height of vehicle floor from top of rail (raised floor area, new wheels and empty load)	≤900 mm
Wheelbase (powered bogie)	1900 mm
Wheelbase (unpowered bogie)	1800 mm
Clear height of passenger compartment	≥1980 mm
Wheel diameter (new wheel)	≤660 mm
Wheel diameter (Max. wear)	≤600 mm
Side doors of passenger compartment	four pairs per side
Clear opening of passenger compartment door (width x height):	≥1300×1860 mm

Table 3-3: Vehicular weight data of AALRT system from CREC

Loads	Car body weight	Passenger weight	Total weight
Empty vehicle (ton)	44	0	44
Rated passenger capacity (ton)	44	15.24	59.24
Overload (ton)	44	19.02	63.02

- Type-II pre-stressed concrete sleeper - 276Kg (Based on calculation of its simplified trapezoidal dimension and 24KN/m³ unit weight assumption)
- Bogie mass (including bolster and central pivot) – 6,800Kg (for conventional bogies)
- Wheel-set mass – 1,191.64Kg (From mass property calculation on Abaqus)
- Wheel-set mass moment of inertia (From mass property calculation on Abaqus):
 - ✓ $I_{xx} = 543.62 \text{ Kg.m}^2$
 - ✓ $I_{yy} = 45.52 \text{ Kg.m}^2$
 - ✓ $I_{zz} = 543.62 \text{ Kg.m}^2$

✓ $I_{xy} \approx I_{xz} \approx I_{yz} \approx 0$

- Therefore the total mass on the axle = $63,020/6 + 6,800/2 + 2,100 = 16,003.33\text{Kg}$
- And on a wheel the static Load became $16,003.33\text{Kg}/2 = 8,001.67\text{Kg}$
i.e. the static vertical load = $P_o = 80\text{KN}$ ($P_o = mg$)
- Lateral track stiffness – 32MN/m

Linear elastic theory is used and no non-linearity/ plasticity considered, except in some spots like in the case of wheel-rail contact identification, to minimize the computational effort during simulation and to gain a first order understanding of the systems behavior; as well the use of parameters are limited and made to be not more than three for same reason as before, which was the computational effort reduction due to minimal availability of computing resources/ capability.

The first thing to do in parameter identification is to find a cause-effect relationship which could be realized through the utilization of fish-bone diagrams, in a very smart way. The fish bone diagram for the parameter identification in this research is shown in Fig.3-3. Once the possible collections of parameters are found through fish-bone diagram, which organized the way we work on parameter identification, we proceed through Pareto chart development which helped the selection of those very influential parameters and narrow down the parameters to use in the design of experiment.

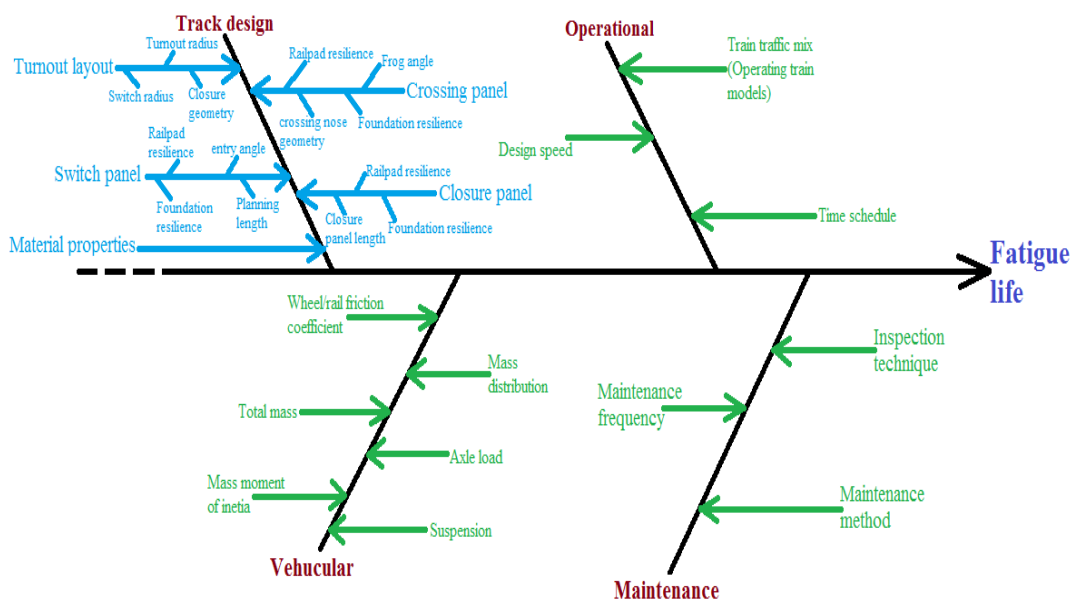


Figure 3-3: Fish-bone diagram

The Pareto dominance analysis could be reached upon by using previous studies and their corresponding results, experimental database from measurement campaigns, logical reasoning and expertise suggestions.

In developing a Pareto chart, in order for producing Pareto dominance, there are some challenges facing researchers in the area of turnout which might include but not limited:

- ✓ The turnout area is categorized into three panels with distinct behavior and influential factors. For instance the switch panel is less affected positively in reducing dynamic stress than in the area of crossing panel. And some parameters (factors) are only applicable in some specific panel like crossing nose geometric parameters which have higher localized effect on manipulating the dynamics in the crossing panel. Another instance could be the rail-pad resilience having a magnificent dynamic load reduction in crossing than switch panel.
- ✓ Some parameters only have an influential effect in one dynamic response and will have a negligible impact on other responses; such research outcomes are available in the works of vehicle-turnout interaction study by Elias and Nielson as it is presented in the reference (Kassa and Nielsen 2008). In this research normal contact force is much affected by axle load and wheel profile; wear index is much affected by axle load, friction coefficient, wheel profile, interaction between axle load and friction coefficient, rail profile and probably the train speed; surface rolling contact fatigue index from Ekberg, which was developed from shakedown theory, is much affected by running direction and friction coefficient; subsurface rolling contact fatigue index is rather much affected by the wheel profile majorly.

These mentioned dependences are correlations and there might not be causal relationship such that further in depth study on causal relationship of the observed correlational relation, in each case, is required. The collections of parameters (factors) could include others, which might have impact on the responses discussed, like gauge widening, track stiffness, crossing nose geometry, switch blade geometry and so on.

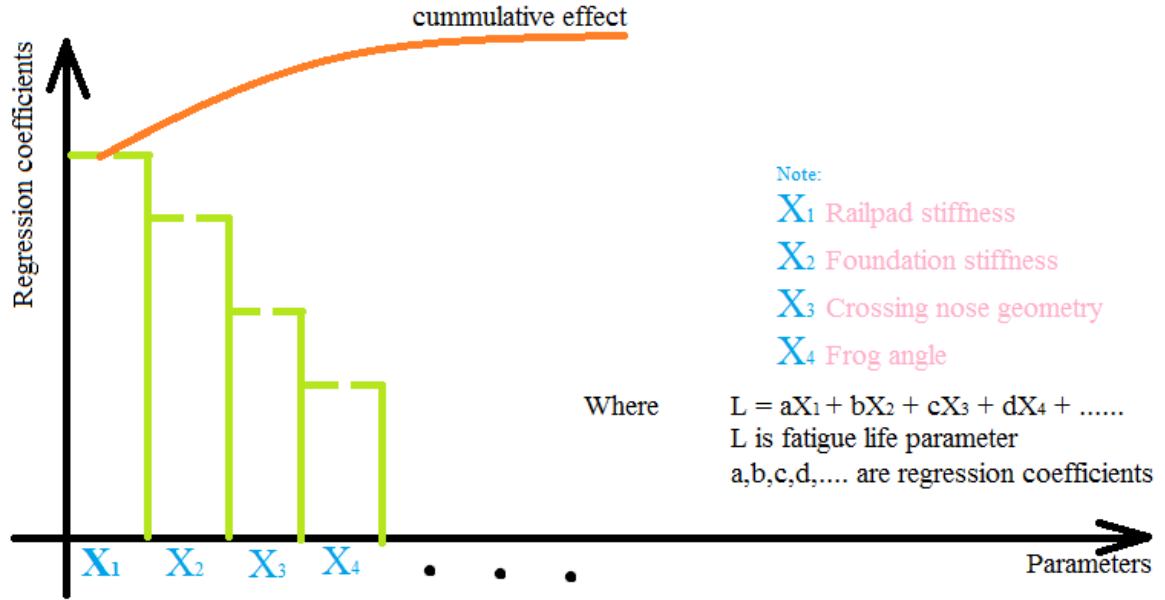


Figure 3-4: Pareto dominance of parameters for turnout fatigue

Even though parameters related to the vehicular, operational and maintenance has great impact on the outcomes of fatigue life of rails in turnout structure of a railway infrastructure, the main focus of this study, however, is more on the fixed/permanent component of the railway system which is the track; therefore the parameters under the track design in the fish-bone diagram are emphasized for the development of Pareto chart and, of these parameters, the geometric ones are not focused because of the limitation posed in this study by the assumption that the #9 turnout is the turnout which is going to be optimized. So the #9 turnout geometries are applied and the optimization is performed.

In the Pareto chart above, parameters with higher coefficients are indicated to have bigger effect in affecting the rail fatigue (RCF) that can be shown by introducing a very small change in magnitude of the parameters which can actually be demonstrated by summation of the product of coefficients and their corresponding parameters, under main effect consideration only:

$$\text{Response} = \sum_{i=1}^n a_i p_i \quad \text{where}$$

a_i – regression coefficients

$$p_i \text{ – parameters}$$

$$a_i p_i = a_i p_{oi} + a_i \Delta p_i \quad \text{Where } p_{oi} \text{ – operating mean parameter magnitude}$$

$$\Delta p_i \text{ – small change in parameter magnitude}$$

This shows that for small variation of the parameters, there is a great effect on rail fatigue response due to those parameters with higher corresponding coefficients. From turnout optimization studies by (Kassa and Nielsen 2008), the rail-pad and track resilience are the two most critical parameters selected for the design of experiment and their vertical stiffness characteristics is only used in this study.

Table 3-4: Range of parameters

Parameters	Stiffness		Damping	
	Higher	Lower	Higher	Lower
Rail-pad (Medium Stiffness)	600MN/m	300MN/m	Operating point 75KNs/m	Operating point 75KNs/m
Sub-structure (Under the sleeper)	170MN/m	110MN/m	Operating point 20KNs/m	Operating point 20KNs/m
Track (serial spring-dashpot connection)	(+,+) 132.5MN/m	(-,-) 80.5MN/m	15.79KN.s/m	15.79KN.s/m

Note:

- For the combination of (+,-), the track stiffness = 92.96MN/m
- For the combination of (-,+), the track stiffness = 108.51MN/m

3.8 Experimental design

The design of experiment involves the generation of combination of parameters which actually help in capturing the information in a system under study in a better way in a given study level like two-level design of experiment (doe) for this particular research. The design of experiment for the turnout optimization is shown in the table below.

Table 3-5: Design of experiment

S.N ^o	Rail-pad stiffness (A)	Track foundation stiffness (B)	Code
1	+	-	A ₁ B ₁
2	+	+	A ₁ B ₂
3	-	-	A ₂ B ₁
4	-	+	A ₂ B ₂

Train movement in the facing direction only has been considered to minimize the number of simulation runs required to develop correlations and compute the optimization works. This is due to the limited computational resource and restricted time availability.

Randomization was supposed to be required to avoid any unknown bias but because the experiment is a virtual experimentation then such bias is almost not present due to our possible control on the inputs and parameters in the dynamic system simulations; so the simulations can run in any order.

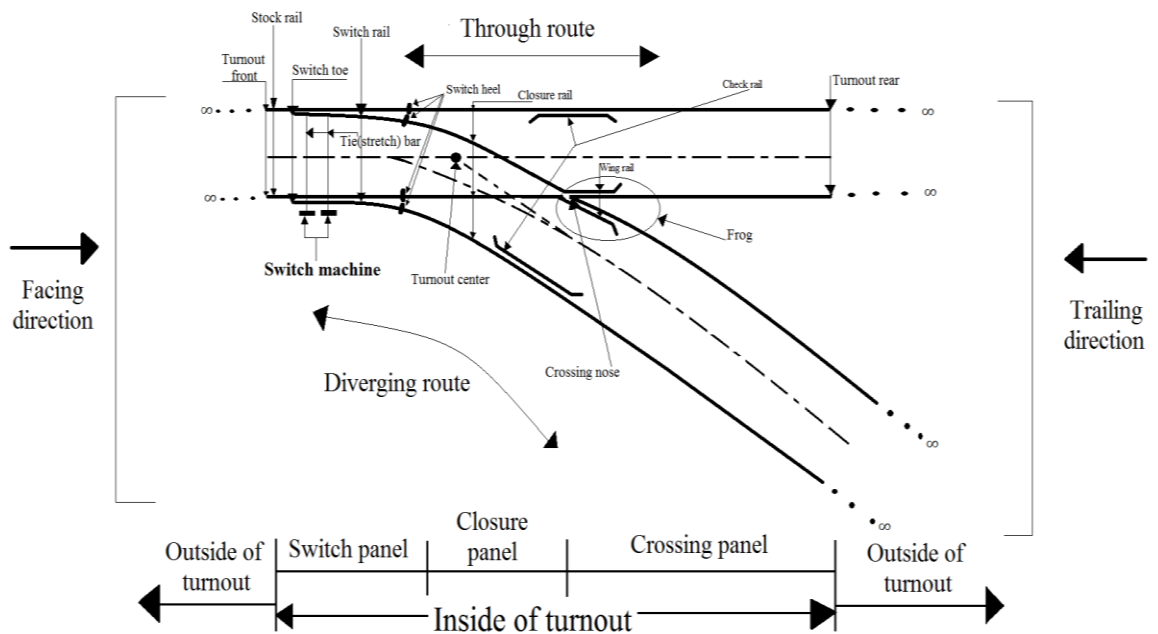
Another important point is the route that is to be very critically damaged due to traffic passage. In knowing that, a random selection of parameter combination from the design of experiment table above has been made to avoid bias in selecting the combinations and the simulation ran in both routes, which are the through route and divergent route. Random selection of the combination of parameters has been made through random number generation using pseudorandom number generator of a scientific calculator Casio and representing combination codes as quadrants which equally divide the plan area into four. Then each quadrant represents a range of 0.25, as the calculator generates pseudorandom numbers between 0 and 1; and it is shown as follow:

- Quadrant 1 (0 – 0.25)
- Quadrant 2 (0.26 – 0.5)
- Quadrant 3 (0.51 – 0.75)
- Quadrant 4 (0.76 – 1)

The random number generated was 0.179 which is in quadrant one and therefore combination one was used for the identification purpose of the critical route, which is (+, -). Then this combination was used to clear-out the question of which route be the critical route in a turnout. Therefore, identification of the criticality has been decided from the RCF result in both scenarios. And the outcome has revealed that the divergent route is the critical route under the assumption of equal traffic proportion in the two routes. This is because the RCF critical damage in the tangent route is smaller than in the divergent route where a detailed damage calculation could be found in the appendix.

3.9 Simple turnout modeling

In modeling turnouts, there are many ways of approaching the problem and of these approaches Multi-body system (MBS) and Finite element (FE) techniques are the very important ones. And actually there are pros and cons for each one of the modeling approaches. However, MBS modeling has computational efficiency character and that of the FE modeling has a character of producing refined responses with better accuracy if done properly.



Note:- There are four possible ways a train could travel in a Turnout, generally, which are:

- * Facing direction - Through route (→)
- * Facing direction - Diverging route (↘)
- * Trailing direction - Through route (←)
- * Trailing direction - Diverging route (↙)

Figure 3-5: Turnout nomenclature

The figure above shows the different areas of turnout section and train movement directions as well as routes. The three major areas within a turnout section which are the switch panel, crossing panel and closure panel with their detailed nomenclature is depicted above which helped for modeling purpose and focus oneself into the turnout section.

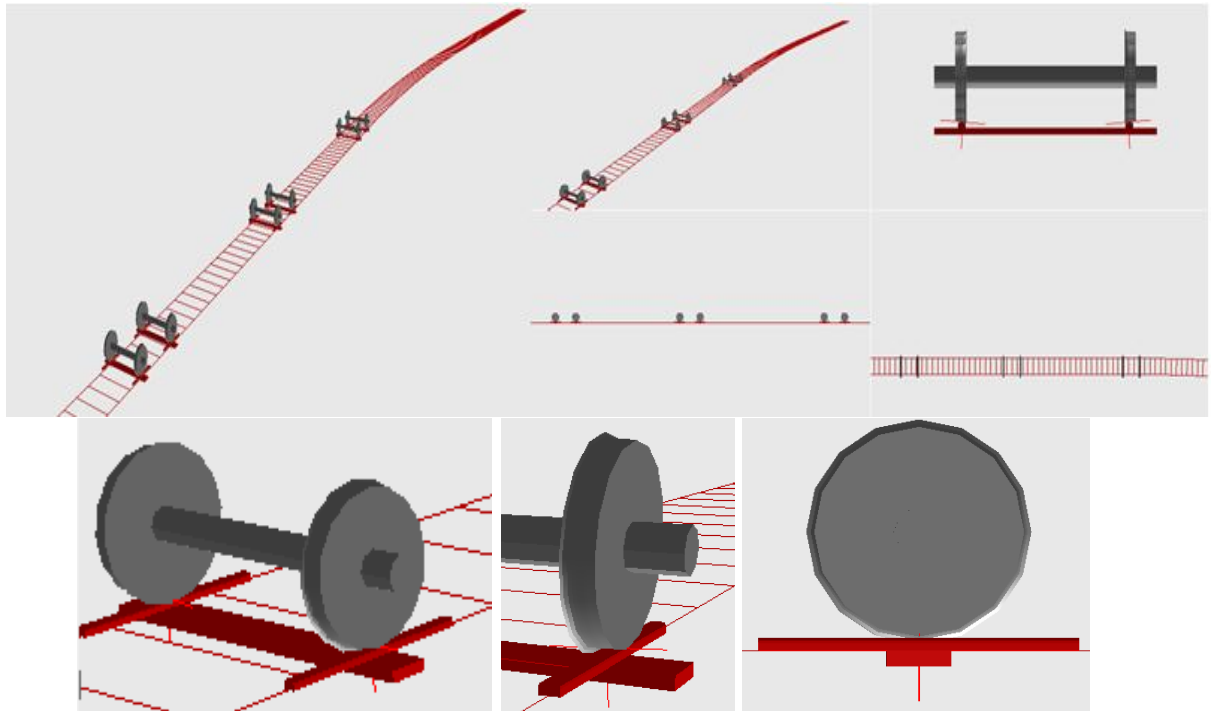


Figure 3-6: Developed MBS model on Simpack as seen from different directions – diverging route

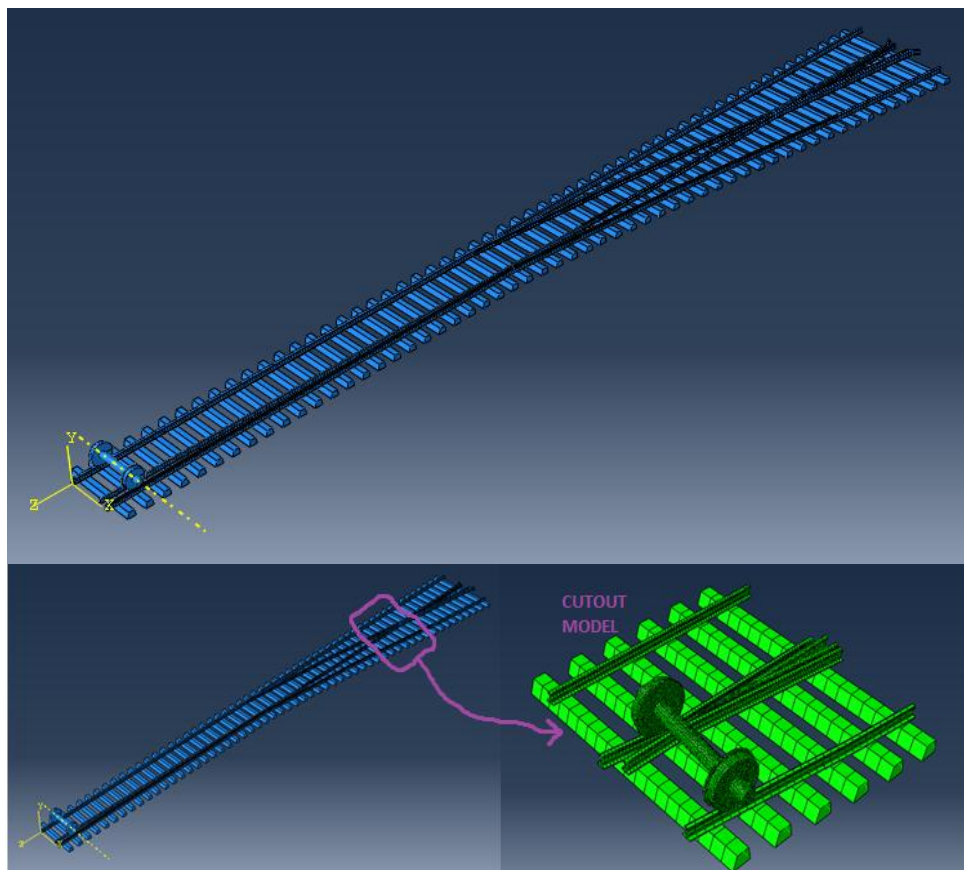


Figure 3-7: Developed FE model on Abaqus software

CHAPTER 4 RESULTS AND DISCUSSIONS

This is the core section of the report, apart from the previous section being the root for this section, in that, simulation and the analysis of the simulation runs were conducted, in order for finding correlations between parameters and determining the optimum fatigue life of simple ballasted turnouts. So, initially some discussions on simulation theory and presentation of simulation results of the dynamic responses of the turnout model were made, to give ideal insight into the outcomes of the simulation result analysis, which was followed by correlation development and optimum fatigue life determination. And finally summary has been given about the discussions made on this section.

4.1 Background on simulation theory

Simulation which interchangeably called numerical simulation, computer simulation or virtual experiment is one of the ways that the working principles of nature are studied in detail besides the other ways of doing that, which are physical experimentation and analytical (theoretical) approach. It is the last approach developed recently in the history of science with the deployment of computers and advances in algorithms and numerical techniques. And either it could be utilized all alone, in some cases, or in combination with physical experiment within a theoretical framework.

Numerical Simulation is a scientific technique which gained a magnanimous acceptance and with further innovations still being made in the field to have a clear and reliable wisdom of a dynamic system under study in almost every discipline of science and engineering. It gives very highly reliable and high fidelity results that could have been obtained from physical experiments with huge resource investment, time wastage and dangerous encounters. There are many advantages which are on the side of numerical simulations and even could be the only possible way in some situations to study a system, like in the case of research in the unknown territories of science. In these circumstances, simulation would be the technique used to get the job done successfully.

Simulation technology becomes more and more important to industry. Simulation systems can be seen as virtual test rigs that allow for replacement of long and costly experiments on a real test rig with faster, cheaper, and more detailed simulations on a computer. And historically, two main types of simulation of mechanical systems have been dominating (Iakov 2002):

- Multi-body dynamics simulations typically deal with systems of interconnected rigid bodies. In many cases the connections are limited to idealized joints (e.g. revolute, prismatic, etc.) and traditional force elements (e.g., spring, damper, external load). Multiple dynamic contacts are often important parts of the model and a relatively simple contact model is most often employed for performance reasons. Multi-body systems are mostly used for time domain dynamic simulations.
- Finite element tools are mostly used for static and quasi-static (with constant velocities) analysis of systems and components as well as for modal analysis of structures. They are also used for detailed simulations of very small and specific parts of a system, e.g., a single contact. Finite element analysis generally requires more computational effort than multi-body analysis.

An industrial simulation system is always the result of interdisciplinary research where contributions from different fields are integrated into a single general useful system. In the case of flexible multi-body simulation system contributions from mechanics, numerical analysis, and computer science are necessary. Here, mechanics provide the mathematical models of physical phenomena in the form of equations; numerical analysis comes up with the methods and algorithms for the solution of the equations, and computer science gives the design guidelines for the efficient implementation of the model on a computer (Iakov 2002)Simulation response

The simulation has used both the MBS and FE modeling approaches in order for solving the problem on fatigue life optimization by efficiently utilizing the best part of each simulation techniques in the perspective of mechanic as well as computational efficiency. Therefore in this subsection dynamic responses found from simulation in the two modeling techniques are presented.

I. MBS simulation response

The multi-body system simulation sample results are shown in the diagrams below and discussion made on the responses observed.

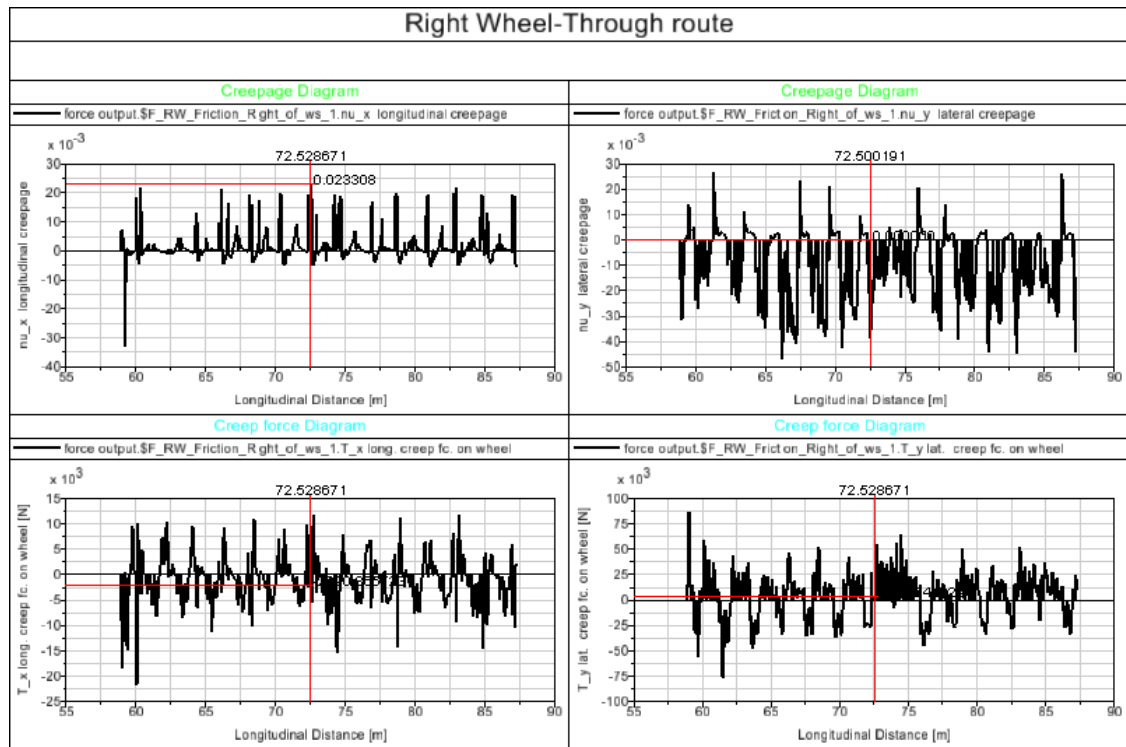


Figure 4-1: Creep force and creepage response due to wheel-set 1 right wheel

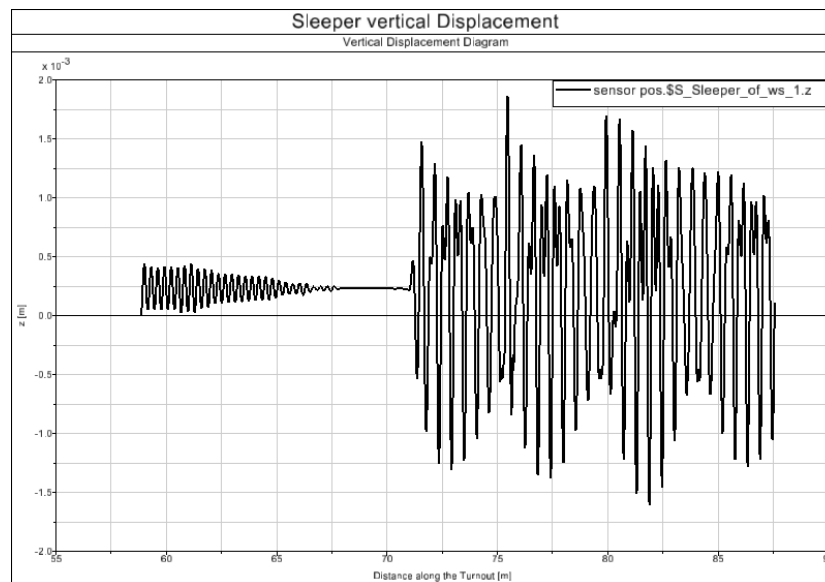


Figure 4-2: Sleeper vertical displacement due to wheel-set 1 in scenario 3 (diverging route)

From the displacement diagram in the above figure, it is possible to observe that the sleepers within the range of the crossing panel get a pattern of oscillatory motion.

II. FE simulation response

Due to a high impact occurrence at the crossing nose, it is very logical to be convinced that the stresses at the crossing panel to be higher than the rest of the turnout area. And this is visible on the von-mises stress contour and the stress histories in the plots below. Then this will make the fatigue life at the crossing nose to be the life determining area in turnouts.

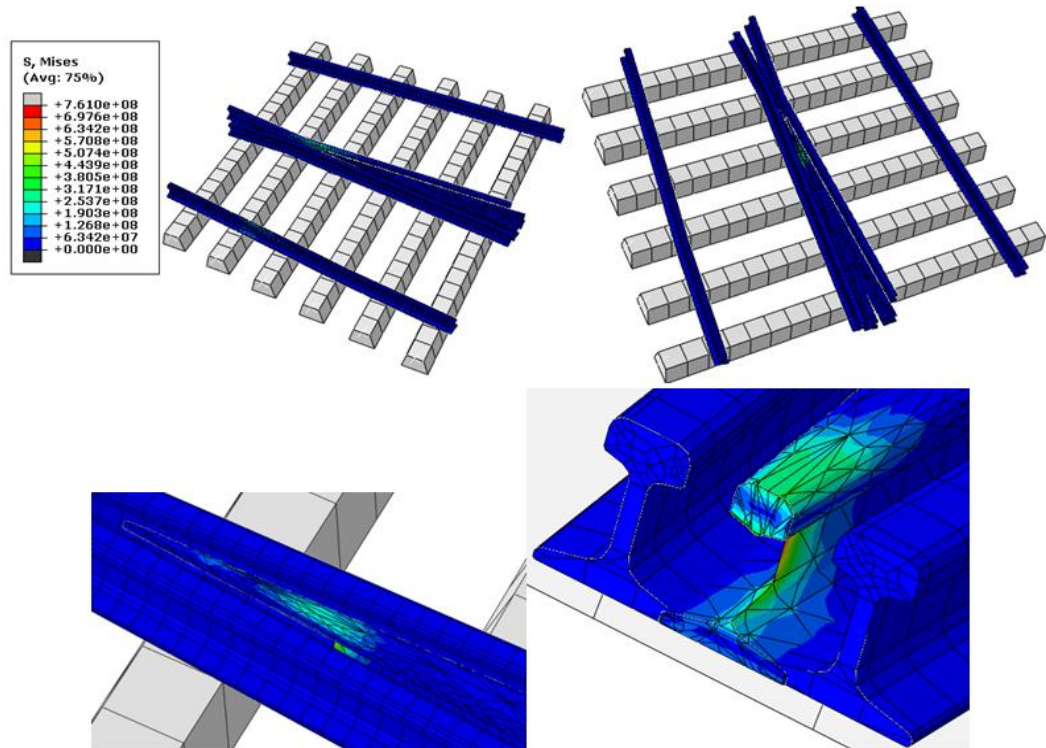


Figure 4-3: Von-mises stress contour from different perspectives and details

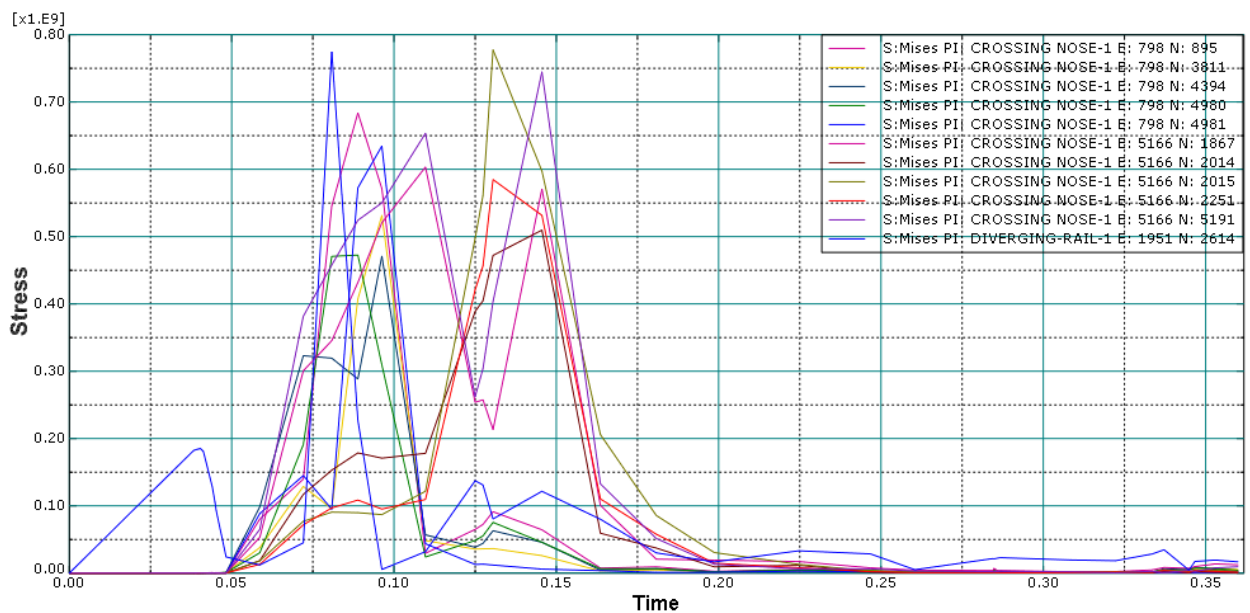


Figure 4-4: Von-mises stress histories

4.2 Parametric correlation

From the simulation output post-processing, it was possible to develop surrogate models in order for predicting damages on rails of the turnouts – for the assumed turnout type – throughout the range of application of the surrogate models. This helped for computing the parameters that produce local optimum damage – implying the minimum damage in this particular study. Therefore in the following tables and paragraphs discussions on the cumulative damage and fatigue life determination of rails has been made. Therefore planners and designers could utilize the surrogate models put in this work in various possible ways.

Table 4-1: Damage calculation outputs due to Right Rails

Combination scenario	Rail-pad stiffness	Substructure Stiffness	Critical Accumulated RCF only Damage	Critical Accumulated Wear only damage	Critical Accumulated Net Damage
1	600	170	2.77272E-05	-2.02327E-02	-2.02049E-02
2	600	110	1.32727E-05	-1.95917E+00	-1.95916E+00
3	300	170	8.49885E-06	-4.73162E+00	-4.73161E+00
4	300	110	1.23829E-05	-2.90264E-03	-2.89026E-03

Table 4-2: Damage calculation outputs due to Left Rails

Combination scenario	Rail-pad stiffness	Substructure Stiffness	Critical Accumulated RCF only Damage	Critical Accumulated Wear only damage	Critical Accumulated Net Damage
1	600	170	1.98829E-05	-4.10713E-05	-2.11884E-05
2	600	110	2.16088E-05	-3.39161E-06	1.82172E-05
3	300	170	2.27406E-05	-4.73162E+00	-4.73159E+00
4	300	110	1.35745E-05	-3.99508E-03	-3.98150E-03

Table 4-3: Damage calculation outputs considering both Rails [$\text{Damage}_{\text{left}} + \text{Damage}_{\text{right}}$]

Combination scenario	Rail-pad stiffness	Substructure Stiffness	Critical Accumulated RCF only Damage	Critical Accumulated Wear only damage	Critical Accumulated Net Damage
1	600	170	4.76101E-05	-2.02737E-02	-2.02261E-02
2	600	110	3.48816E-05	-1.95917E+00	-1.95914E+00
3	300	170	3.12394E-05	-9.46324E+00	-9.46320E+00
4	300	110	2.59574E-05	-6.89772E-03	-6.87176E-03

In addition the developed surrogate models for the various cases and situations has the advantage of estimating the damage mode and quantified damage for the purpose of maintenance scheduling, maintenance planning and design of turnouts for rails damage.

At this point, the estimated optimum fatigue life of turnout rails is determined – considering both rails – from the surrogate models which are paraboloid response surfaces that are shown in below. Paraboloid RS is used because the study focuses on determining global single optimum fatigue life and the corresponding parameters. The detailed derivation and development of the surrogate models is found in the appendix.

- **RCF Damage** = $f(x,y) = -1.75 \cdot 10^{-10}x - 2 \cdot 10^{-9}y + 2.742 \cdot 10^{-10}x^2 - 1.25 \cdot 10^{-11}y^2 - 4 \cdot 10^{-12}xy - 0.00002697$
- The parameters for the optimum damage – both for an RCF as well as for Wear criteria at the boundaries are: $K_{\text{rail-pad}} = 300\text{MN/m}$
 $K_{\text{sub-structure}} = 110\text{MN/m}$

However this parametric combination has safety issues with regard to derailment and riding comfort; therefore the next best would be wiser to be considered as the optimal scenario:

- ✓ For an RCF criteria – the third scenario with $K_{\text{rail-pad}} = 300\text{MN/m}$ and $K_{\text{rail-pad}} = 170\text{MN/m}$
- ✓ For a Wear criteria – the first scenario with $K_{\text{rail-pad}} = 600\text{MN/m}$ and $K_{\text{rail-pad}} = 170\text{MN/m}$

Then here comes calculus part where the critical parameters are determined and tested for local optimum:

$$D = f_{xx}(x,y) f_{xx}(x,y) + f_{xy}(x,y)^2$$

Critical parameters are determined by setting $f_x(x,y) = 0$ and $f_y(x,y) = 0$

$$f_x(x,y) = 5.484 \cdot 10^{-10}x - 4 \cdot 10^{-12}y - 1.75 \cdot 10^{-10} \quad \text{and} \quad f_y(x,y) = -4 \cdot 10^{-12}x - 2.50 \cdot 10^{-11}y - 2 \cdot 10^{-9}$$

Solving the above linear simultaneous equations will give $(x_o, y_o) = (65.92, -90.81)$ but the substructure stiffness parameter obtained is not physically feasible; therefore the parameters with optimum damage in fatigue are those obtained on the boundary which is $(300\text{MN/m}, 170\text{MN/m})$ as per the discussion in the previous discussion.

Based on the optimum parameters input the average life of the rails in turnouts of the type assumed in this study is:

$$1/\text{cumulative RCF Damage} = \frac{1}{\left(\frac{3.12394\text{E-}05}{2}\right)} = 64,021.72$$

This means that the design car – AALRT train single car – can make 32,010 numbers of passes. These much cycles could be around:

$$\frac{64,021}{(365 \times 16 \times 60 / 6 + 1)} = 1.10 \text{ years} \quad \text{under an assumption of a 6min headway and 16hr. daily operation.}$$

4.3 Optimized fatigue life determination

The basic approach in the determination of fatigue life in the form of number of cycles till failure from FEM simulations follow the following simple procedure or algorithm:

1 – Prepare stress time history from the FEM simulation; in a multi-axial stress condition an equivalent stress like a von mises stress would be necessary to work just like a uniaxial stress problem of fatigue life calculation which is a matter of single stress time history data. Once the stress history is found then rotate it 90 degree to make it look like a pagoda roof.

2 – Perform cycle counting of the stress time history data or signal; the method of cycle counting could be rain-flow counting, peak flow counting, reservoir counting or any other counting technique or algorithm however the rain-flow counting method is the most popular and recommended in various international standards. The stress counting in rain-flow counting method looks like a rail flowing on a pagoda roof.

3 – Determine an effective alternating stress for each stress ranges using various synthetic relationships like Goodman's relation, Soderberg relation, Marrow relation or other relation.

4 – Develop an S-N curve either using direct experimentation or synthetic approximations like Goodman line, Soderberg line or other appropriate approximate S-N diagrams.

5 – Determine the numbers of cycles till failure from the S-N curve developed and convert it into any required form of life like number of years till failure from information or assumption on train schedule.

In this study a cycle counting technique used was the famous rain-flow method and the approximate synthetic method for S-N curve determination was the Goodman's line; as well the effective alternating stress is determined from Goodman's relation. At this point it would be instructive to comment on the way that S-N curves developed from a direct experimental determination would be very much exact, reliable but too expensive; and therefore a less expensive way of determining S-N diagrams could be chosen in a resource scarce research situation like this particular study as can be observed from the discussions made in the previous and this paragraphs.

Once the stress ranges and number of cycles with the corresponding information on the S-N curve is determined then calculation of fatigue damage and the damage accumulation be done using either linear or non-linear damage accumulation theories; however linear damage accumulation theories are simple and give a good first hand understand of the damage accumulation process therefore in this study the linear damage accumulation theory using Palgrem-Miner damage accumulation has been used.

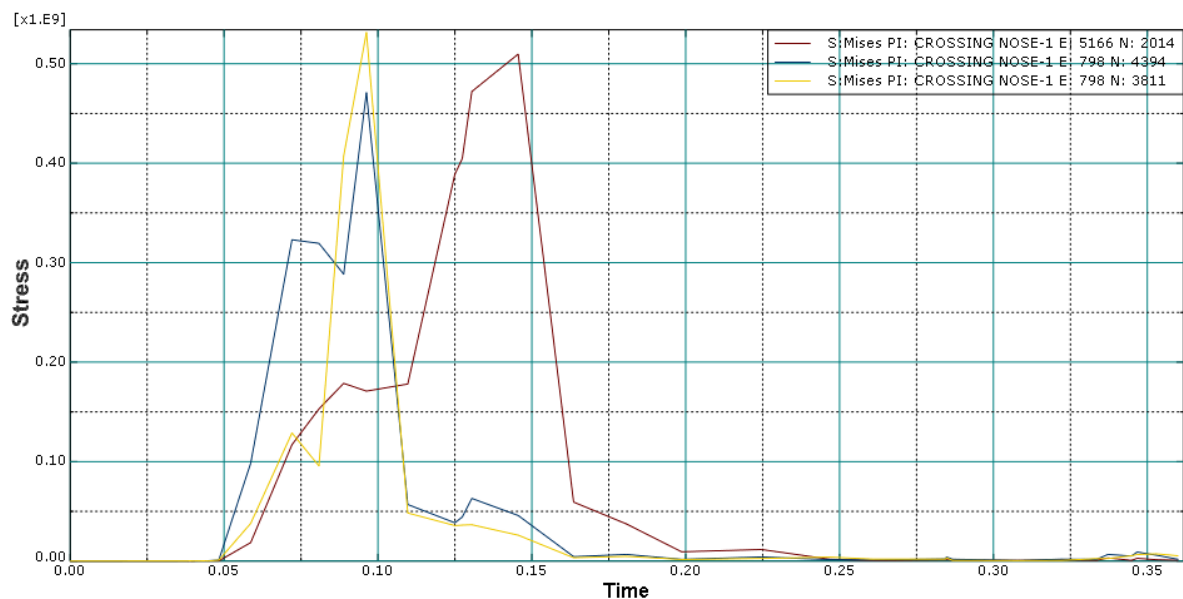


Figure 4-5: Von-mises stress histories filtered for further Fatigue life calculation

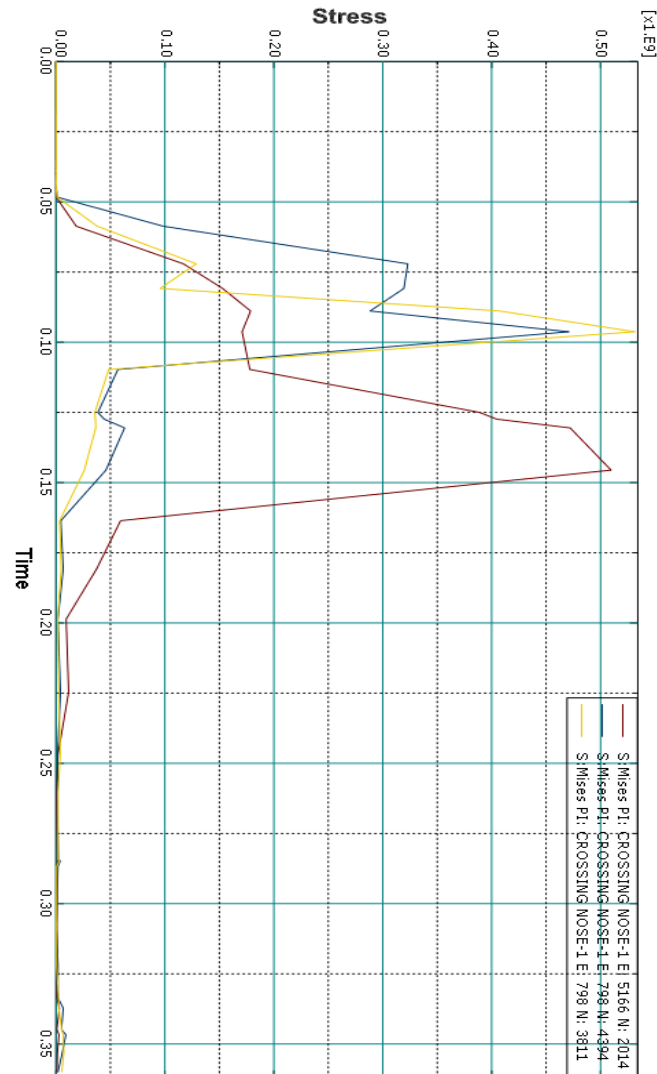


Figure 4-6: Von-mises stress histories rotated at 90 degree for rain-flow analysis

Table 4-4: Sample Stress cycle counting and the corresponding damage determination

Stress range	Mean stress	Alternating stress	Effective alternating stress	Number of cycles	Number of cycles till failure from S-N curve	Damage
0.000 – 0.000	0.0000	0.0000	0.000	0.5	Infinite	0.00E+00
0.035 – 0.010	0.0225	0.0250	0.026	0.5	Infinite	0.00E+00
0.010 – 0.000	0.0050	0.0100	0.010	0.5	Infinite	0.00E+00
0.325 – 0.280	0.3025	0.0450	0.069	0.5	Infinite	0.00E+00
0.475 – 0.000	0.2375	0.4750	0.651	0.5	4.02E+05	1.24E-06
0.070 – 0.035	0.0525	0.0350	0.037	0.5	Infinite	0.00E+00
* The analysis has been made for element Crossing Nose-1 E:798 N:4394						
* Stresses are in GPa				Cumulative Damage		1.24E-06

Table 4-5: Calculated Damage at various locations

Location	Damage
Crossing Nose-1 E:798 N:895	5.00E-01
Crossing Nose-1 E:798 N:3811	0.00E+00
Crossing Nose-1 E:798 N:4394	1.24E-06
Crossing Nose-1 E:798 N:4980	1.24E-06
Crossing Nose-1 E:798 N:4981	5.00E-01
Crossing Nose-1 E:5166 N:1867	5.00E-01
Crossing Nose-1 E:5166 N:2014	2.37E-06
Crossing Nose-1 E:5166 N:2015	5.00E-01
Crossing Nose-1 E:5166 N:2251	1.15E-02
Crossing Nose-1 E:5166 N:5191	5.01E-01
Diverging Rail-1 E:1951 N:2614	5.00E-01

Damage is an evolutionary mechanism in that there is progression rather than an immediate surprise except for very highly discontinued situation like totally overloaded structure. Therefore, from the above result it could be possible to predict damage starts in some elements and evolve throughout the rail till it led to failure. And this would be an intermediate damage level rather than the very extremes, low or high, from those damages on the elements. Upon the discussions made, the rail actual overall damage for the purpose of its fatigue life became **2.37E-06**.

The total damage on the rail due to the whole wheel-sets on a single car would be found by a mere linear superposition of damages due to each wheel-set. This implies multiplying the damage obtained from each single wheel-set by six, because there are six Wheel-sets on one car, to determine the damage resulted from a single car passing a turnout of the type assumed:

$$\text{Damage}_{\text{total}} = 6 * 2.37\text{E-}6 = 14.22\text{E-}6$$

⇒ The optimal fatigue life of rails on turnouts of the assumed type is:

$$\frac{(1/ \text{Damage}_{\text{total}})}{(365 * 16 * 60 / 6 + 1)} \approx 1.2 \text{Years} \quad \text{under an assumption of 16hr operation and 6 min. headway times.}$$

One thing to observe from the calculation above is that non-linear accumulation of damage due to the multiple wheel-sets is ignored even though material hardening behavior of the rail could bring non-linear damage accumulation into play. However, the following reasons could be given why it has been done the way it is observed:

- ✓ The simulation was based on elastic analysis and it would make sense to approximate the reality by assuming that elastic shakedown would not be passed even if the reality could show some plastic shakedown and even ratcheting character during impact at the crossing nose. However, the analysis would help understand the mechanisms and behaviors with simple calculations.

Behaviors beyond the elastic limit could be accumulated by including non-linearity through hardening characterization and using strain fatigue life approach with a combination of non-linear damage accumulation theories. Or it might be needed to model the whole turnout and simulate all the six wheel-sets running on the turnout for some longer time, so that strain life calculations be done and total damage due to a single car with six wheel-sets be determined keeping the material non-linearity into the model and with linear damage accumulation. In general, after all damage accumulation theories are required because of the following reasons:

- ✓ At least damages due to various effective alternating strains/stresses have to be cumulated.
- ✓ As well it is almost impossible to run a simulation for the whole life time of the rail (like beyond one year) in turnouts which might minimize the use of damage accumulation theories.

In general, according to the results from the FEM simulation, the optimal fatigue life of rails in turnouts of the type assumed is 1.2 Years.

4.4 Outcome comparison between modeling approaches

The focus of the study was on the optimal fatigue life prediction and corresponding track parameters. In doing that MBS and FE simulation modeling techniques has been used. However fatigue life estimations through post processing of structure responses from the two mechanical models show some difference. The reasons that might contribute to the

fatigue life prediction discrepancy between the two simulation modeling approaches includes but not limited to the following:

- I. Some non-linearity in the wheel/rail interaction has been considered with in the MBS but not in the FE. This is due to computational load that make the simulation impossible unless linearization and simplification made based on the available hardware for simulation.
- II. Elastic material behavior has been used and this makes stress not to redistribute throughout the members. The estimations then brought higher stresses in some areas and low stresses in most parts of the members. Therefore this led to higher fatigue life prediction.
- III. If plasticity would have been included into the model, then it was possible to use strain life prediction instead of stress life calculation. Stress life calculations work for higher cycle fatigue phenomenon which predicts higher life than the strain life technique. Strain life technique is much more accurate for low cycle fatigue and members with plastic deformation; this would be more accurate fatigue life calculation at the crossing nose which involves impact and plastic phenomenon. However, this accuracy comes with higher computational cost because it requires plastic simulation which requires higher computational capacity than its elastic counterpart.
- IV. If the whole model have been plotted with damage contour, then it might be possible to accurately predict the life with various failure modes identified. This would have given a possibility of zero discrepancy and identification of the failure modes. However, it would have required huge computational capacity and fracture experimental setup, sample cut-out model fracture simulation or fracture simulation of the original model by inserting various cracks which could be obtained from the damage contour.

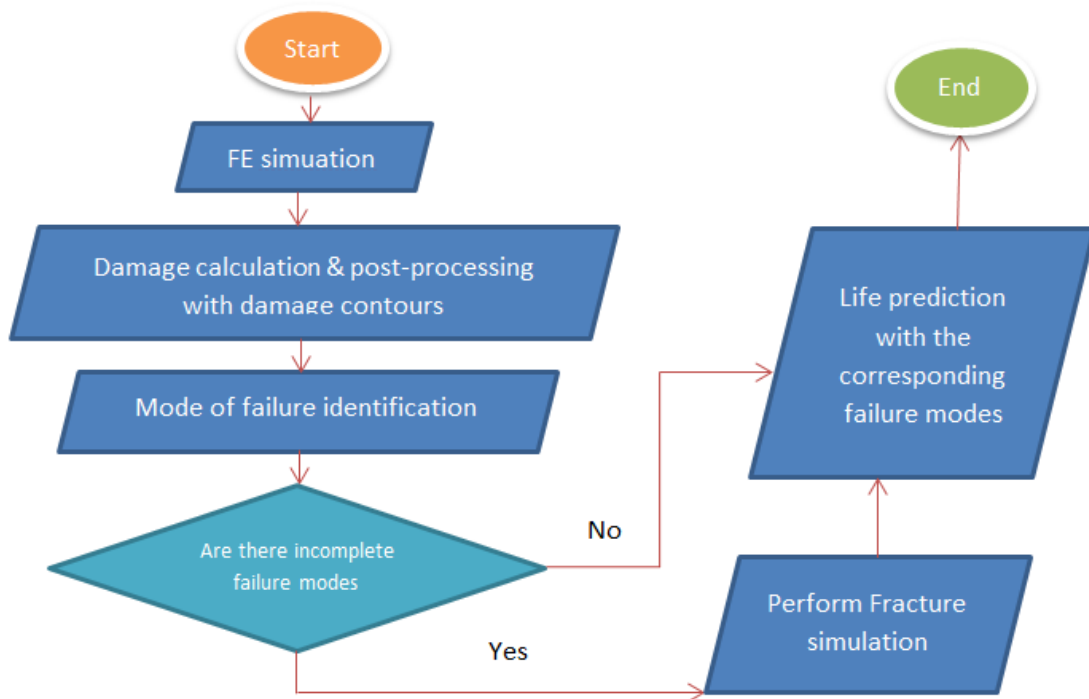


Figure 4-7: Flow chart – Algorithm for better accuracy Fatigue life prediction

This however, would have necessitated immense resources beyond the resource constraints for this research.

4.5 Summary

In this section discussions on the outcomes of the simulations from the MBS and FEM techniques have been made and fatigue life of rails determined. The fatigue life determinations from the two simulation approaches have been compared and interpretations as well been made on the results obtained.

CHAPTER 5 CONCLUSIONS, RECCOMENDATIONS AND FUTURE WORKS

This section is the final part of this report and gave a concluding remark at the beginning and then followed by propositions of some further works to be made that are recommended by the author for the better understanding and advancement of wisdom/ knowledge in turnout dynamics and technological innovations.

CONCLUSION

In conclusion, the work had performed mechanistic dynamic simulation of turnouts on MBS and FE environment and post-processed the responses for further fatigue life prediction. Then the predictions have been used for fatigue life optimization and the accompanying optimal parametric identification.

Furthermore, comparison between the fatigue life predictions from the response outputs post-processing of the two simulation techniques has been made and possible reasons forwarded for the discrepancy observed on the life estimations. As well, techniques for improving accuracy have been put forward, in order for minimizing the error of fatigue life predictions.

However, in general it was not very difficult to estimate the optimal fatigue life of rails, within the assumed turnout design which is turnout #9, would be in the range between 1.1 and 1.2 years only. Therefore either it would be required to upgrade the existing designs or search totally new designs/ track-forms which suit the maintenance free paradigm that we seek for and wait in the near future.

RECOMMENDATIONS

From the analysis results and discussions made it is recommended that the rail-pads stiffness be 300MN/m and sub-structure stiffness be 170MN/m such that the tracks in turnout sections could live optimally. And in general, using the fourth scenario parameters are not recommended for use in a real world since there are safety issues that need to be sought.

FUTURE WORKS

This work has been made with simplifications and assumptions which had made the research manageable in the time frame of this thesis work as well as due to computational capability limitations posed. However, in order for maximizing the accuracy and precision of the research and gain in an in depth understanding of the phenomenon, further works are in demand and includes the following but not limited to:

- Design of experiment at a higher level than only at two levels; like four level design of experiment;
- Investigation of more parameters for the optimization work;
- Experimental campaigns shall be conducted to validate and calibrate the models for better precision and accuracy outcomes;
- Robustness analysis should be included in order for accounting variability and uncertainties in input values like vehicles operating on the turnout, material properties, surface characters, etc.;
- Inclusion of damage evolutions in the models simulation, like rail and wheel wear;
- Using non-linear damage accumulation models;
- Using multi-physics modeling approach in a single turnout model, like utilization of continuum model for the solid components and discrete element model (DEM) for the granulated components such as the ballast.

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APPENDICES

APPENDIX A

Surrogate model regressive derivation

Surrogate models are parameters-response relationship models which are developed from small number of simulation runs or physical experiment findings data space and used as if we have got them from full data space of the system though statistical approaches. In this study response surface (RS) technique has been utilized and a paraboloid (3D version of parabolic) response surface developed using regressive statistical approach, based on the simulation parameter – response data space obtained. The derivation of the RS is shown below:

$$z = ax + by + cx^2 + dy^2 + fxy + g$$

$$\sum e^2 = \sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]^2$$

$$\text{Equation A-1} \quad \frac{de^2}{da} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum x$$

$$\rightarrow 0 = \sum z \sum x - a \sum x \sum x - b \sum y \sum x - c \sum x^2 \sum x - d \sum y^2 \sum x - f \sum xy \sum x - ng \sum x$$

$$\text{Equation A-2} \quad \frac{de^2}{db} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum y$$

$$\rightarrow 0 = \sum z \sum y - a \sum x \sum y - b \sum y \sum y - c \sum x^2 \sum y - d \sum y^2 \sum y - f \sum xy \sum y - ng \sum y$$

$$\text{Equation A-3} \quad \frac{de^2}{dc} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum x^2$$

$$\rightarrow 0 = \sum z \sum x^2 - a \sum x \sum x^2 - b \sum y \sum x^2 - c \sum x^2 \sum x^2 - d \sum y^2 \sum x^2 - f \sum xy \sum x^2 - ng \sum x^2$$

$$\text{Equation A-4} \quad \frac{de^2}{dd} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum y^2$$

$$\rightarrow 0 = \sum z \sum y^2 - a \sum x \sum y^2 - b \sum y \sum y^2 - c \sum x^2 \sum y^2 - d \sum y^2 \sum y^2 - f \sum xy \sum y^2 - ng \sum y^2$$

$$\text{Equation A-5} \quad \frac{de^2}{df} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum xy$$

$$\rightarrow 0 = \sum z \sum xy - a \sum x \sum xy - b \sum y \sum xy - c \sum x^2 \sum xy - d \sum y^2 \sum xy - f \sum xy \sum xy - ng \sum xy$$

$$\text{Equation A-6} \quad \frac{de^2}{dg} = 2 * [\sum [z - (ax + by + cx^2 + dy^2 + fxy + g)]] \sum (1)$$

Where $\sum(1) = n$

$$\rightarrow 0 = n \sum z - an \sum x - bn \sum y - cn \sum x^2 - dn \sum y^2 - fn \sum xy - n^2 g$$

$$\therefore 0 = \sum z - a \sum x - b \sum y - c \sum x^2 - d \sum y^2 - f \sum xy - ng$$

Where: - **x** is Rail-pad vertical stiffness
y is substructure vertical stiffness
and **z** is Damage (the RCF, Wear or Combined Damage)

The above systems of equations are then transformed/ converted into matrix form for the purpose of computational simplicity and efficiency i.e. $[P] = [\text{matrix}][a \ b \ \dots \ g]^T$

$$\Rightarrow [a \ b \ \dots \ g]^T = [\text{matrix}]^{-1}[P] \quad \text{where:- } [\text{matrices}]^{-1} \text{ is } \textbf{inverse of the matrix (6x6)}$$

and $[P]$ is a **Vector**

The components of the matrix and the vector include:

$$P = \begin{bmatrix} \sum z \sum x \\ \sum z \sum y \\ \sum z \sum x^2 \\ \sum z \sum y^2 \\ \sum z \sum xy \\ \sum z \end{bmatrix} \quad \text{and} \quad \text{matrix} = \begin{bmatrix} \sum x \sum x & \sum y \sum x & \sum x^2 \sum x & \sum y^2 \sum x & \sum xy \sum x & n \sum x \\ \sum x \sum y & \sum y \sum y & \sum x^2 \sum y & \sum y^2 \sum y & \sum xy \sum y & n \sum y \\ \sum x \sum x^2 & \sum y \sum x^2 & \sum x^2 \sum x^2 & \sum y^2 \sum x^2 & \sum xy \sum x^2 & n \sum x^2 \\ \sum x \sum y^2 & \sum y \sum y^2 & \sum x^2 \sum y^2 & \sum y^2 \sum y^2 & \sum xy \sum y^2 & n \sum y^2 \\ \sum x \sum xy & \sum y \sum xy & \sum x^2 \sum xy & \sum y^2 \sum xy & \sum xy \sum xy & n \sum xy \\ \sum x & \sum y & \sum x^2 & \sum y^2 & \sum xy & n \end{bmatrix}$$

Components of the matrix are found from the parameter combinations obtained from the design of experiment and that of the MBS simulation response post-processing through T_x-RCF damage emperical relationship.

Notice that: - **z₁** is RCF only Damage

z₂ is Wear only Damage

z₃ is combined RCF and Wear Damage

n = 4 (four data points obtained from four MBS simulations)

Table A-1: Design of experiment data space when only Right Rails considered

x	y	z_1	z_2	z_3	x^2	y^2	xy
600	170	4.761007E-05	-2.027373E-02	-2.022612E-02	360000	28900	102000
600	110	3.488156E-05	-1.959173E+00	-1.959138E+00	360000	12100	66000
300	170	3.123945E-05	-9.463235E+00	-9.463204E+00	90000	28900	51000
300	110	2.595738E-05	-6.897720E-03	-6.871762E-03	90000	12100	33000
1800	560	1.39688E-04	-1.14496E+01	-1.14494E+01	900000	82000	252000

Table A-2: Design of experiment data space when only Left Rails considered

x	y	z_1	z_2	z_3	x^2	y^2	xy
600	170	1.988286E-05	-4.107127E-05	-2.118842E-05	360000	28900	102000
600	110	2.160882E-05	-3.391611E-06	1.821720E-05	360000	12100	66000
300	170	2.274060E-05	-4.731618E+00	-4.731595E+00	90000	28900	51000
300	110	1.357450E-05	-3.995075E-03	-3.981501E-03	90000	12100	33000
1800	560	7.780678E-05	-4.735657E+00	-4.735579E+00	900000	82000	252000

Table A-3: Design of experiment data space when both Rails considered

x	y	z_1	z_2	z_3	x^2	y^2	xy
600	170	4.761007E-05	-2.027373E-02	-2.022612E-02	360000	28900	102000
600	110	3.488156E-05	-1.959173E+00	-1.959138E+00	360000	12100	66000
300	170	3.123945E-05	-9.463235E+00	-9.463204E+00	90000	28900	51000
300	110	2.595738E-05	-6.897720E-03	-6.871762E-03	90000	12100	33000
1800	560	1.39688E-04	-1.14496E+01	-1.14494E+01	900000	82000	252000

- Surrogate model for RCF only Damage due to Right wheels:

$$\text{RCF Damage} = 3.4907 \times 10^{-22}x - 4.67721 \times 10^{-20}y + 2.79376 \times 10^{-10}x^2 + 2.66352 \times 10^{-22}y^2 - 6.35378 \times 10^{-23}xy - 0.0000279376$$

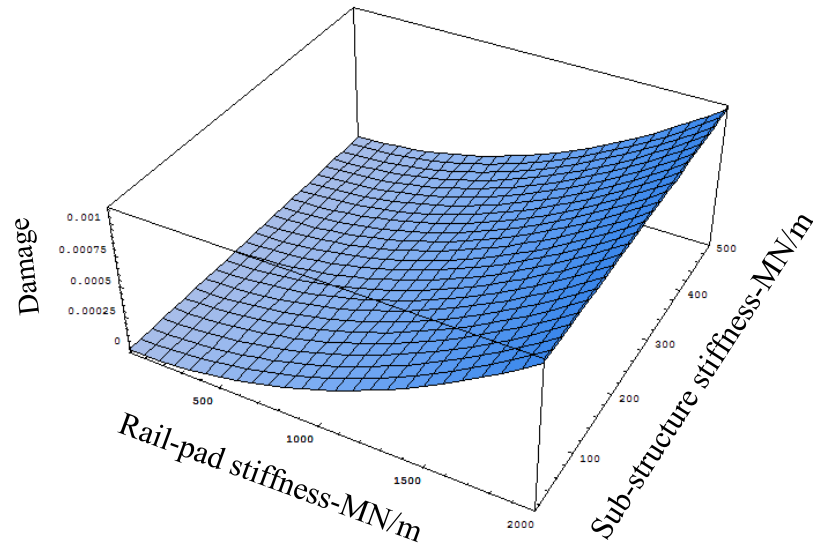


Figure A-1: RCF Damage response surface due to right wheels in a divergent route

- Surrogate model for wear only Damage due to Right wheels:

$$\text{Wear Damage} = 0.000505x - 0.00194y - 0.00002114x^2 + 9.7 \times 10^{-6}y^2 + 2.0 \times 10^{-7}xy + 1.727$$

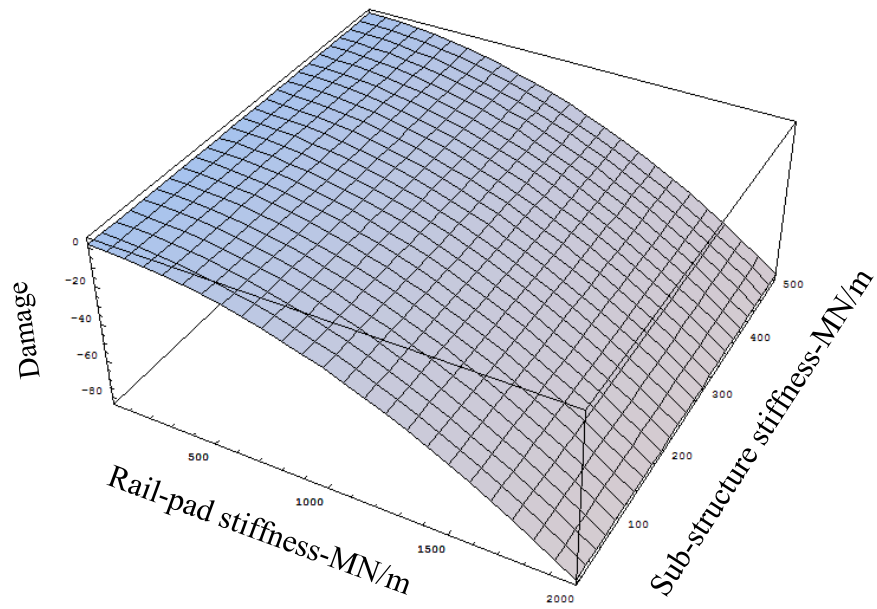


Figure A-2 Wear Damage response surface due to right wheels in a divergent route

- Surrogate model for a combined RCF and Wear Damage due to Right wheels:

$$\text{Combined RCF \& Wear Damage} = 0.00001x - 0.00014y - 0.00002256x^2 + 7 \cdot 10^{-7}y^2 + 2.0 \cdot 10^{-7}xy + 2.227$$

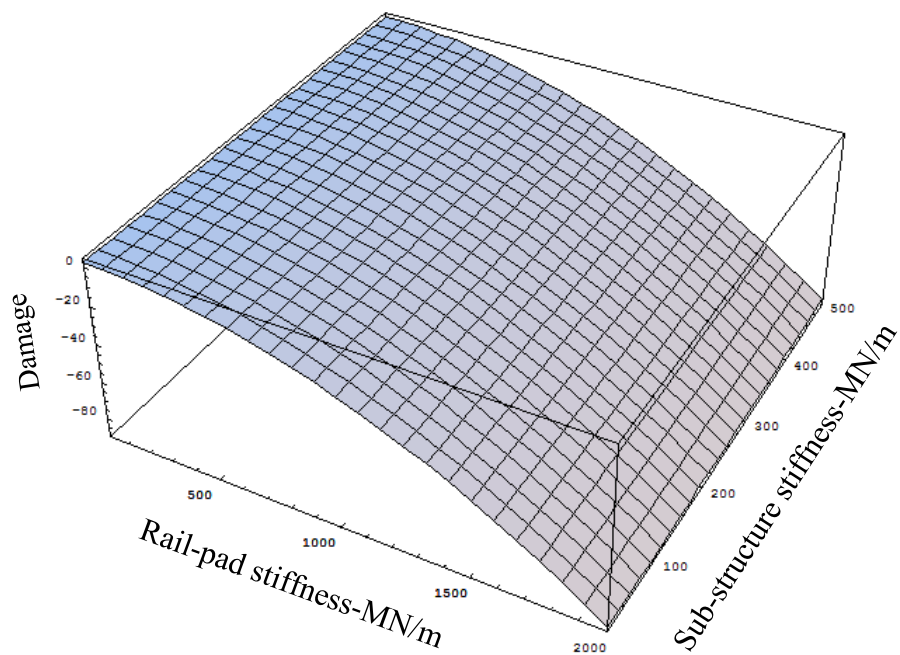


Figure A-3: Combined RCF & Wear Damage RS due to right wheels in a divergent route

- Surrogate model for RCF only Damage due to Left wheels:

$$\text{RCF Damage} = 1 \cdot 10^{-11}x - 1 \cdot 10^{-10}y + 1.5586 \cdot 10^{-10}x^2 + 2.37091 \cdot 10^{-22}y^2 - 1 \cdot 10^{-13}xy - 0.000015601$$

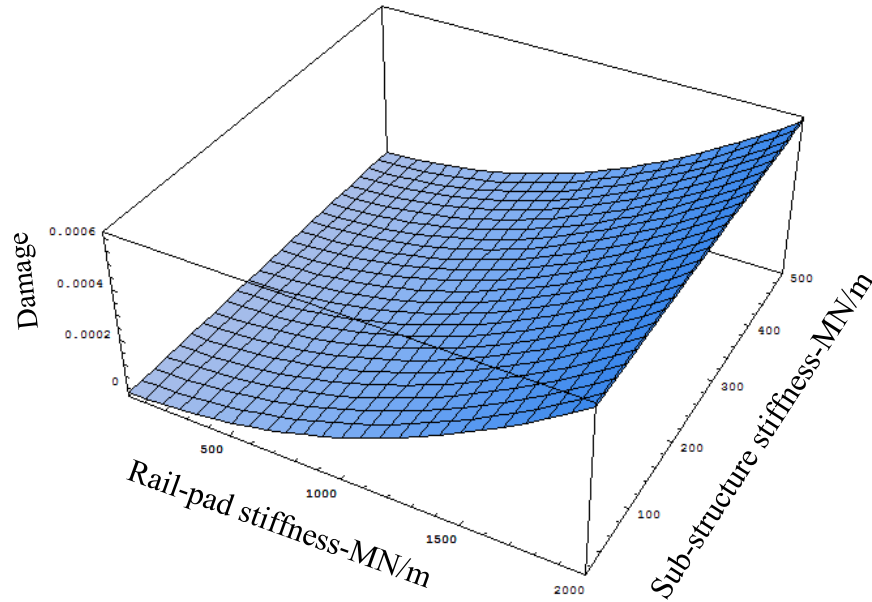


Figure A-4: RCF Damage response surface due to Left wheels in a divergent route

- Surrogate model for wear only Damage due to Left wheels:

$$\text{Wear Damage} = -1.5 \cdot 10^{-6}x + 0.00001y - 9.479 \cdot 10^{-6}x^2 - 5 \cdot 10^{-8}y^2 + 2.0 \cdot 10^{-8}xy + 0.9479$$

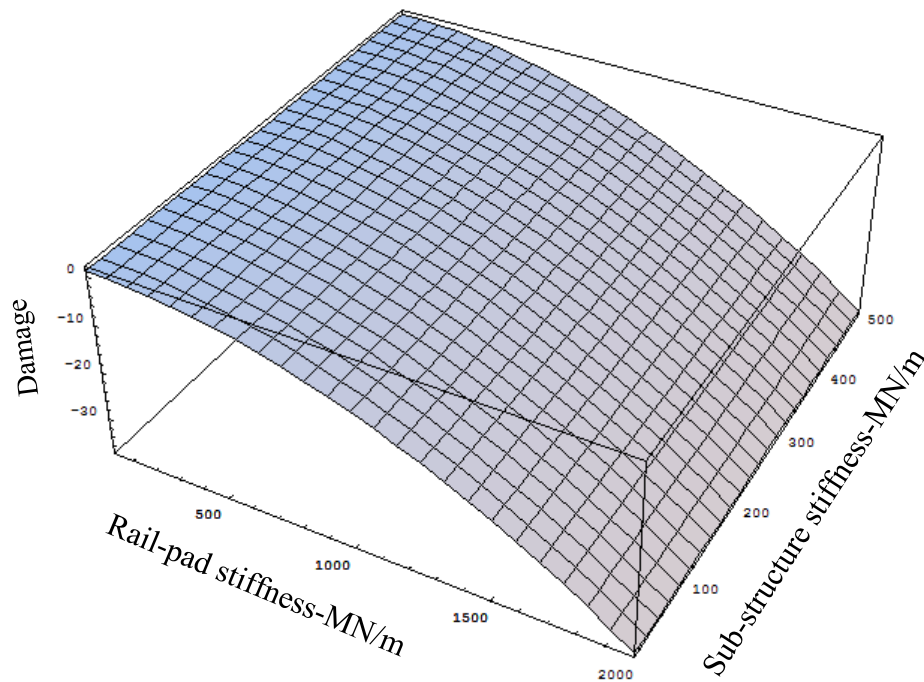


Figure A-5 Wear Damage response surface due to left wheels in a divergent route

- Surrogate model for a combined RCF and Wear Damage due to Left wheels:

$$\text{Combined RCF \& Wear Damage} = 0.00001x - 0.00014y - 0.00002256x^2 + 7*10^{-7}y^2 + 2.0*10^{-7}xy + 2.227$$

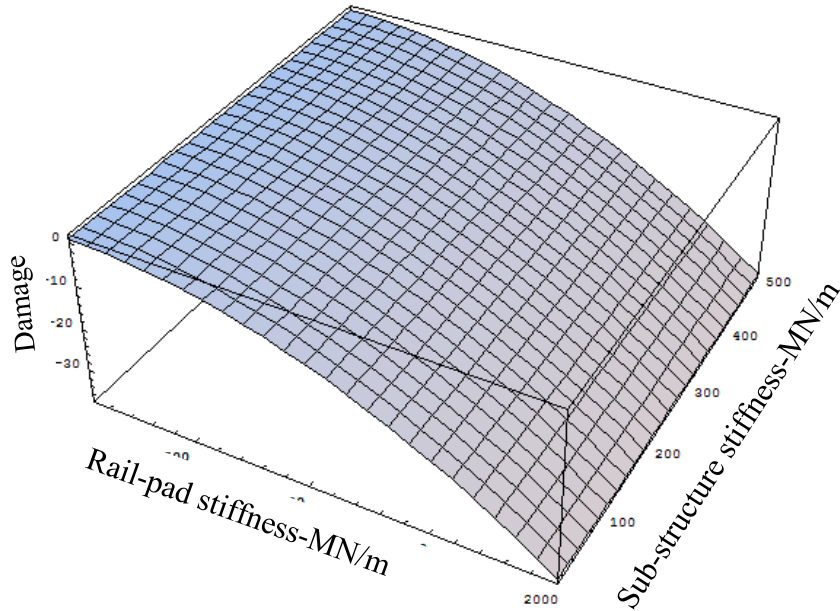


Figure A-6: Combined RCF & Wear Damage RS due to Left wheels in a divergent route

- Surrogate model for RCF only Damage Both wheels considered:

$$\text{RCF Damage} = -1.75*10^{-10}x - 2*10^{-9}y + 2.742*10^{-10}x^2 - 1.25*10^{-11}y^2 - 4*10^{-12}xy - 0.00002697$$

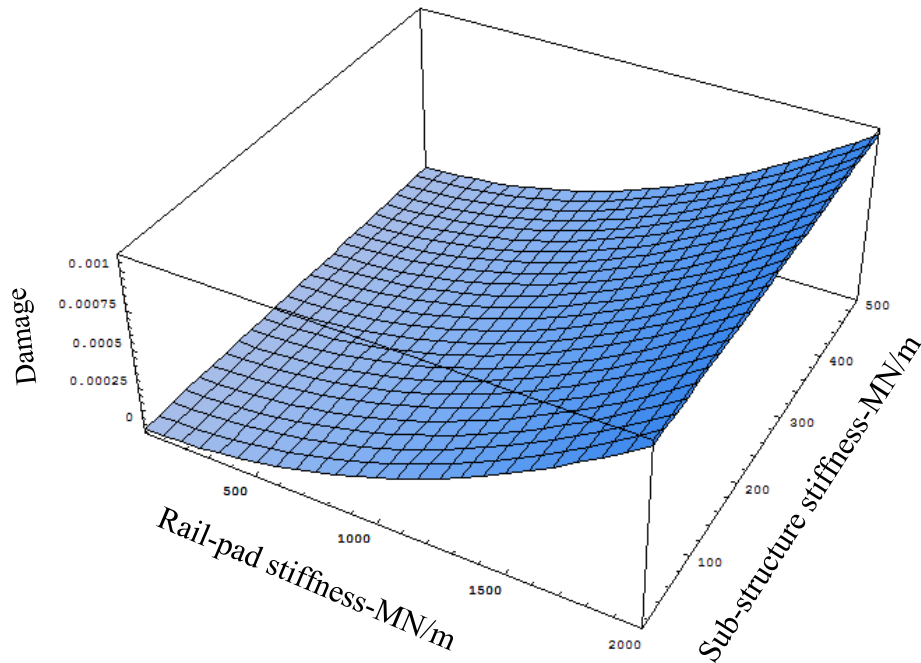


Figure A-7: RCF Damage response surface both wheels considered in a divergent route

- Surrogate model for wear only Damage Both wheels considered:

$$\text{Wear Damage} = -12.5*10^{-6}x + 0.00013y - 23.21*10^{-6}x^2 - 6.5*10^{-7}y^2 + 2.0*10^{-7}xy + 2.348$$

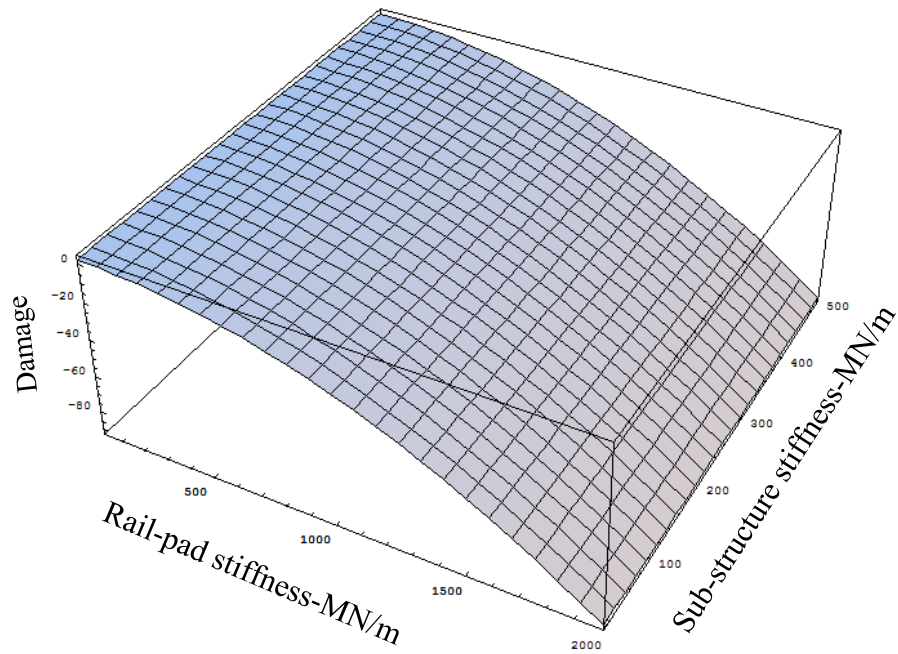


Figure A-8 Wear Damage response surface both wheels considered in a divergent route

- Surrogate model for a combined RCF and Wear Damage Both wheels considered:

$$\text{Combined RCF \& Wear Damage} = 0.00001x - 0.00014y - 0.00002256x^2 + 7 \cdot 10^{-7}y^2 + 2.0 \cdot 10^{-7}xy + 2.227$$

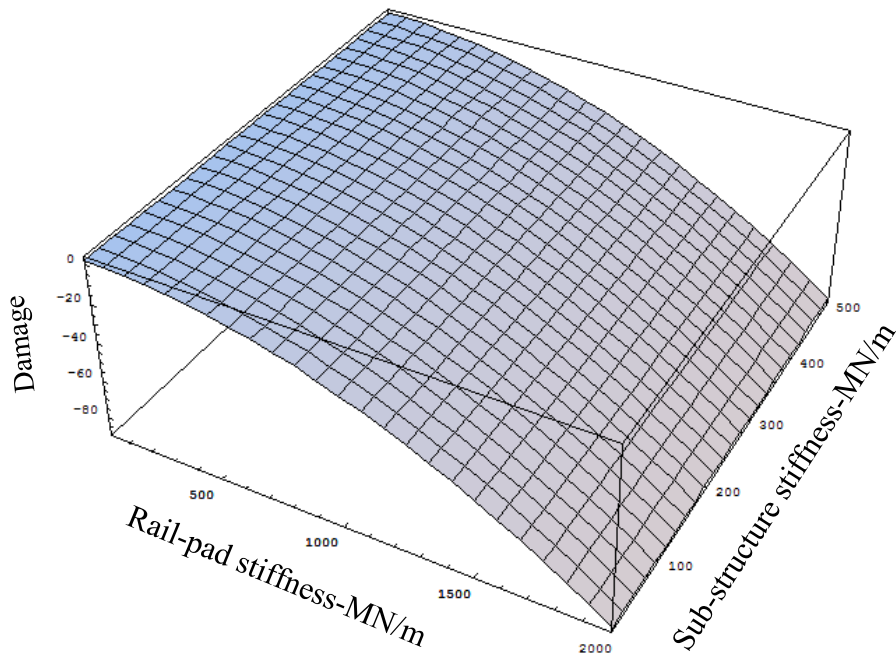


Figure A-9: Combined RCF & Wear Damage RS both wheels considered in a divergent route

APPENDIX B

RCF damage calculation from MBS simulation based on empirical T_r -RCF damage relationship

In lieu of describing the phenomenon of damage in rails of railway track, a parameter called T_r -index (Contact patch energy or Wear number) has been developed from huge experimental data, by researchers, on the damage quantification and response measurements of railway tracks. This parameter captures information about RCF and Wear damages; the parameter requires only the creepage and creep forces in the transverse and longitudinal direction to predict damage on rails which is given as: $T_r = T_x \gamma_x + T_y \gamma_y$ and empirically determine the damage from a graph which can be transformed into an equation (the graph shown below).

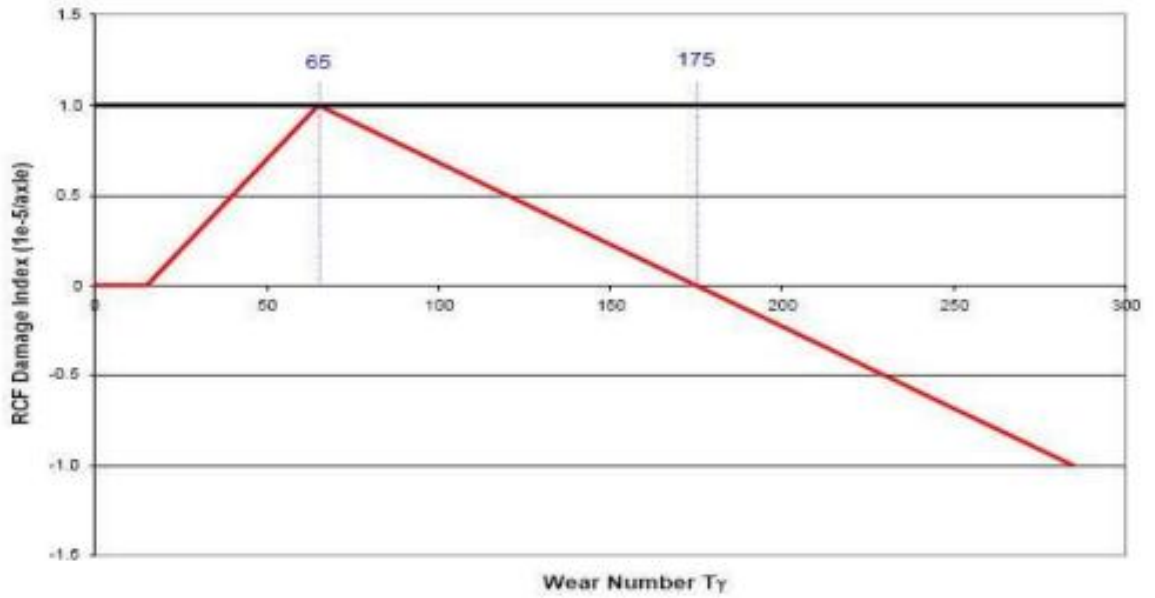


Figure B-1: T_r -RCF damage relationship (Source: [19] iwnicki)

$$T_r = \begin{cases} 0 & T_r \leq 15 & \text{no RCF Damage} \\ 2 \times 10^{-7} T_r - 3 \times 10^{-6} & 15 \leq T_r \leq 65 & \text{RCF only damage} \\ 1.591 \times 10^{-5} - 9.091 \times 10^{-8} T_r & T_r \geq 65 & \text{Wear dominated Damage} \end{cases} \quad \text{Equation B-1}$$

A. Through route

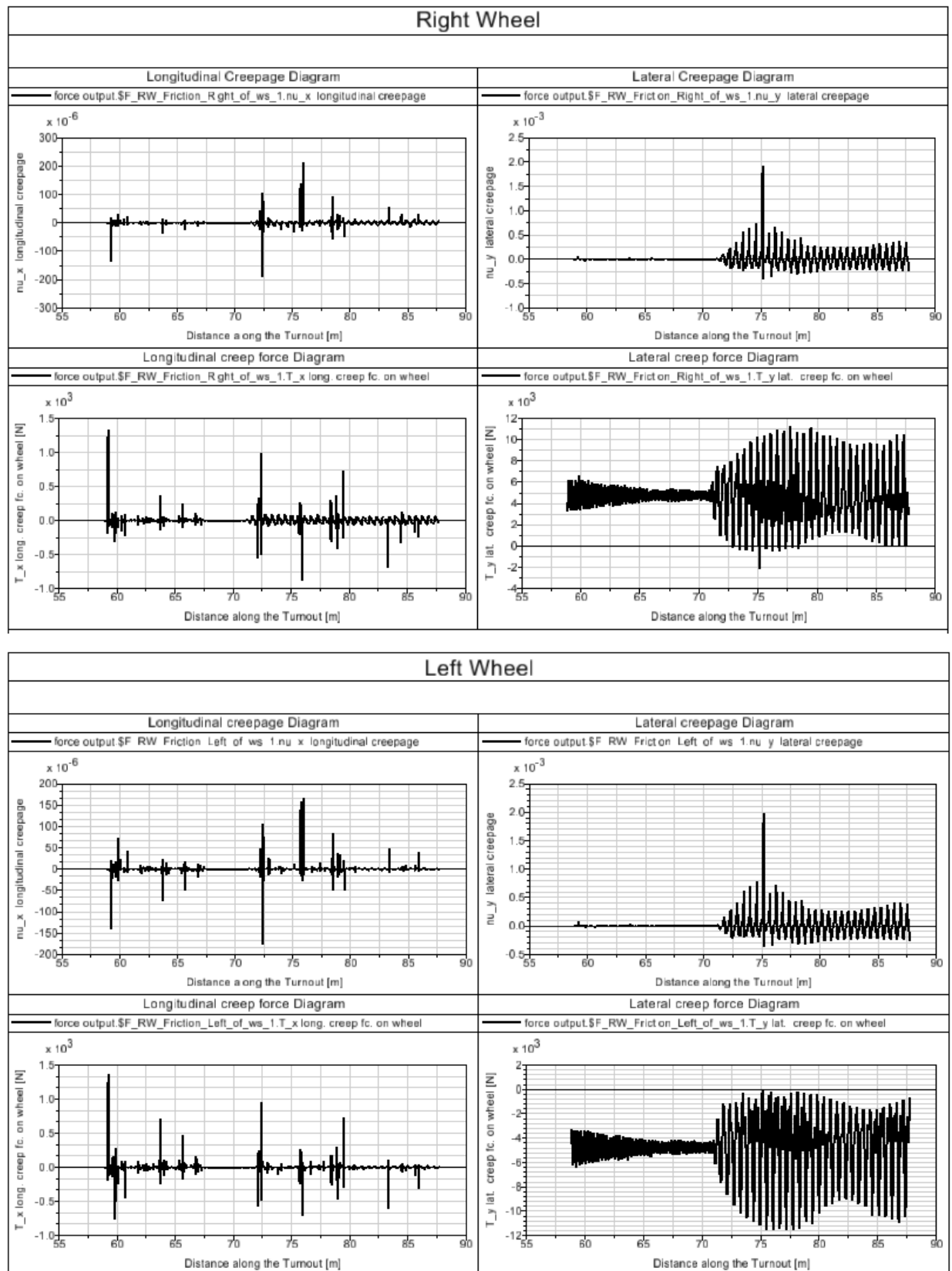


Figure B-2: Creepage and creep force response due to wheel-set 1 in the through route

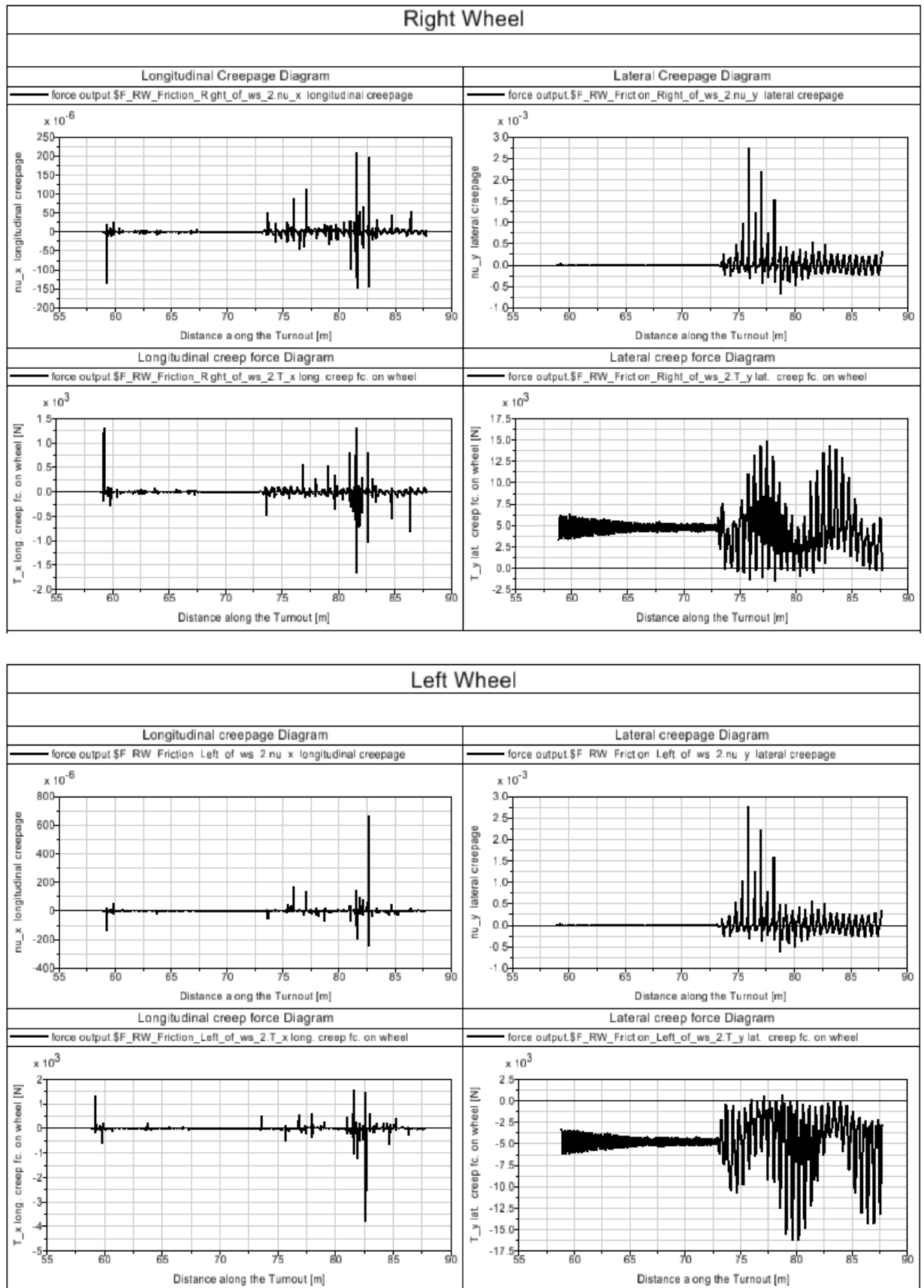


Figure B-3: Creepage and creep force response due to wheel-set 2 in the through route

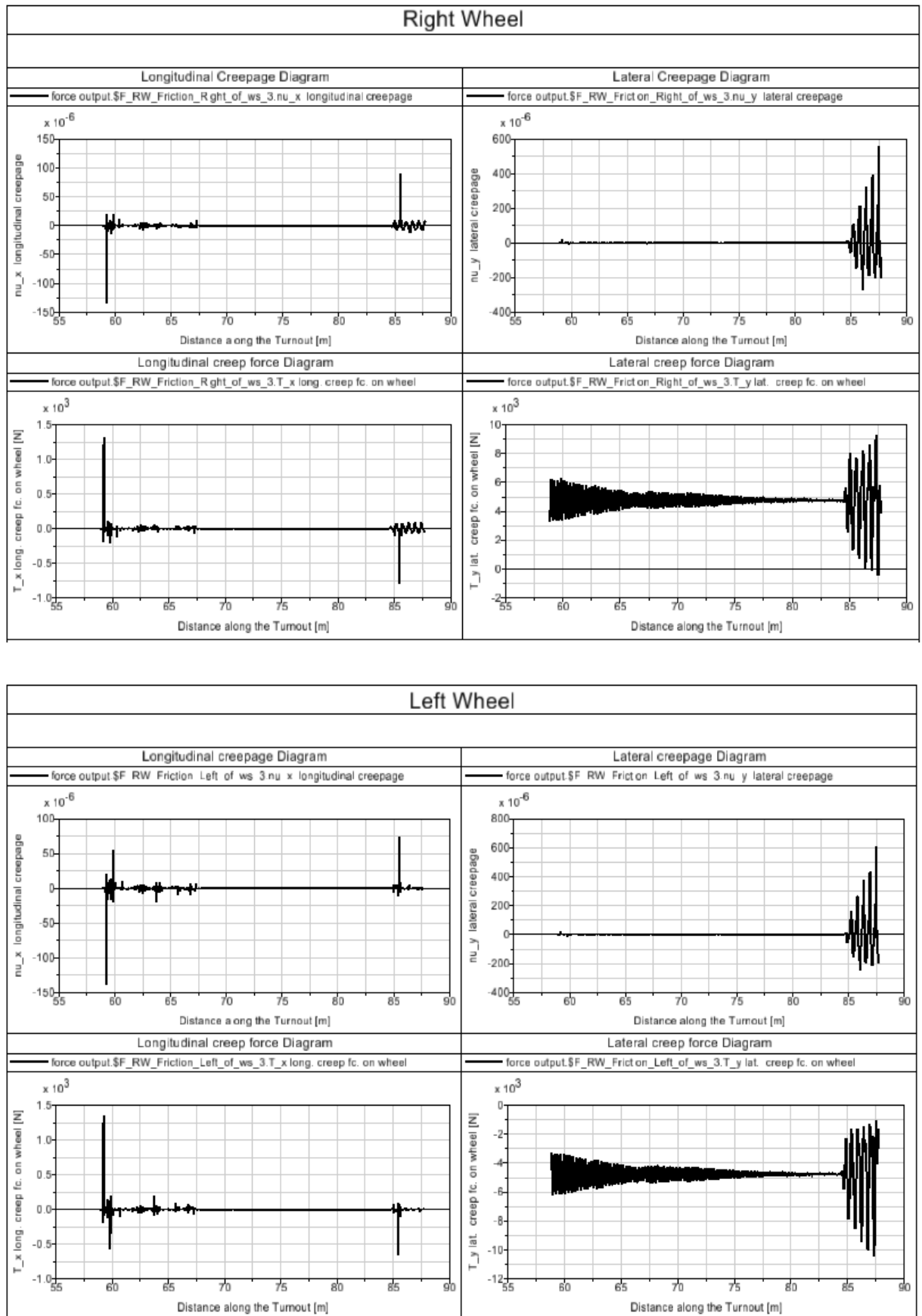


Figure B-4: Creepage and creep force response due to wheel-set 3 in the through route

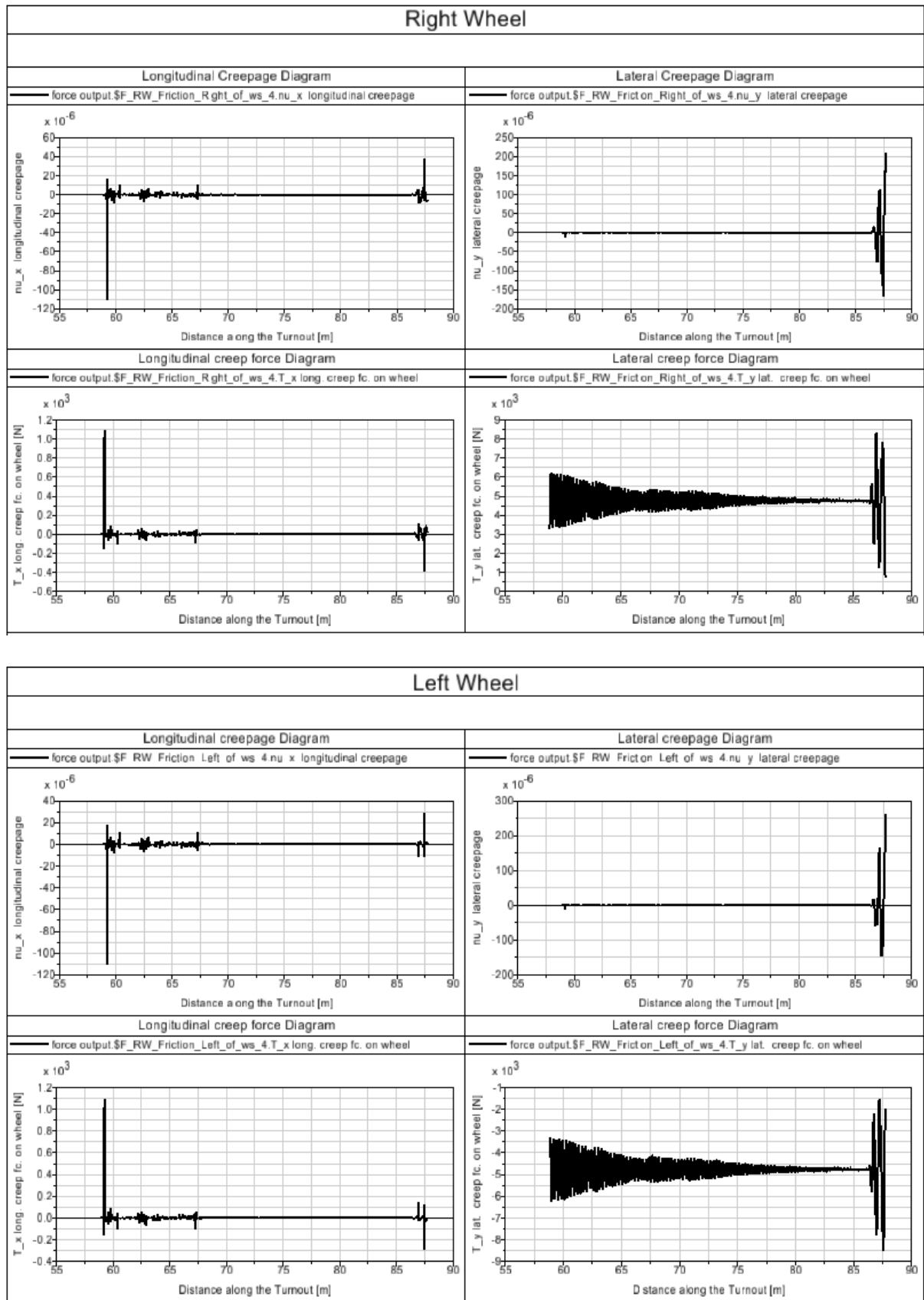


Figure B-5: Creepage and creep force response due to wheel-set 4 in the through route

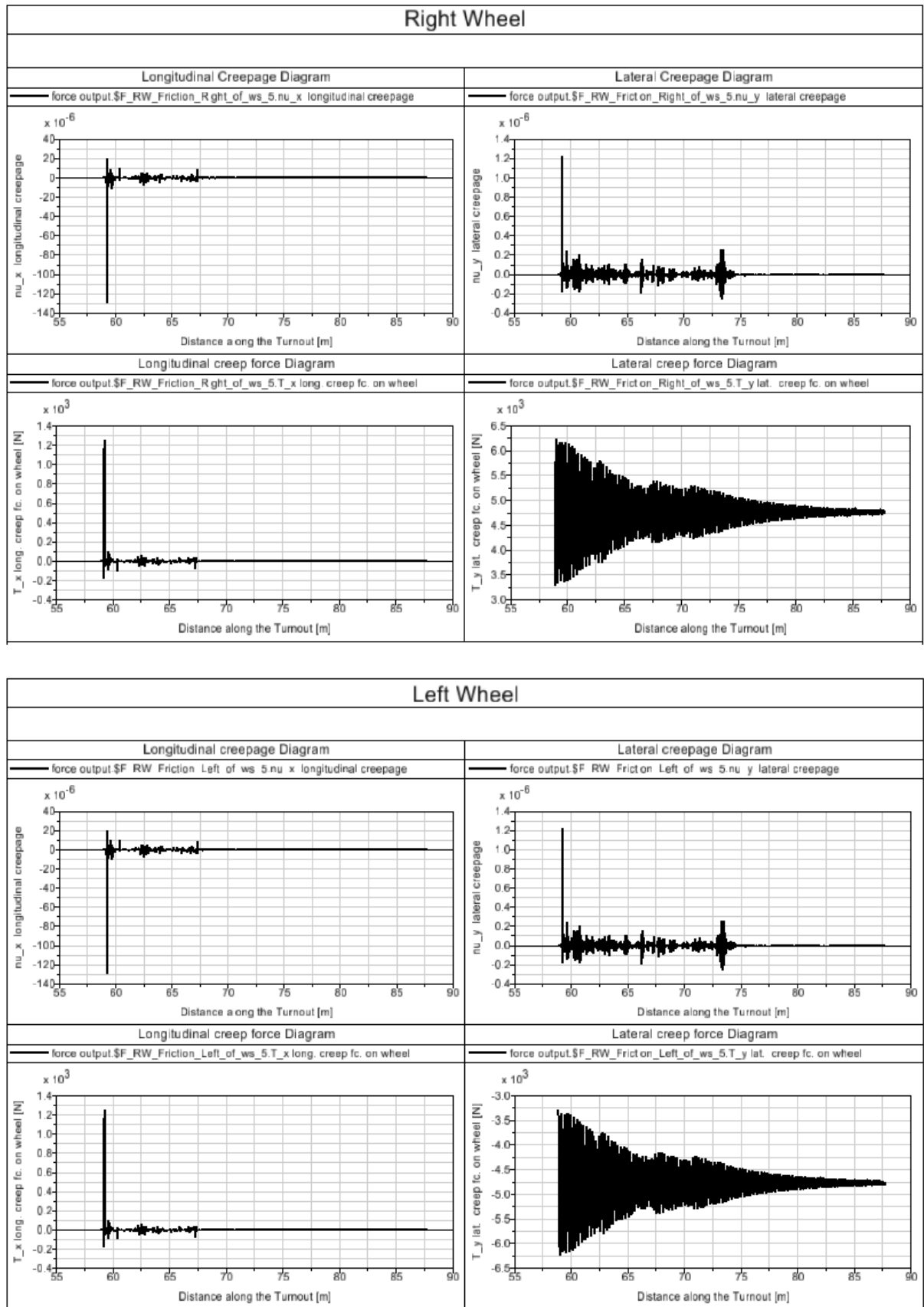


Figure B-6: Creepage and creep force response due to wheel-set 5 in the through route

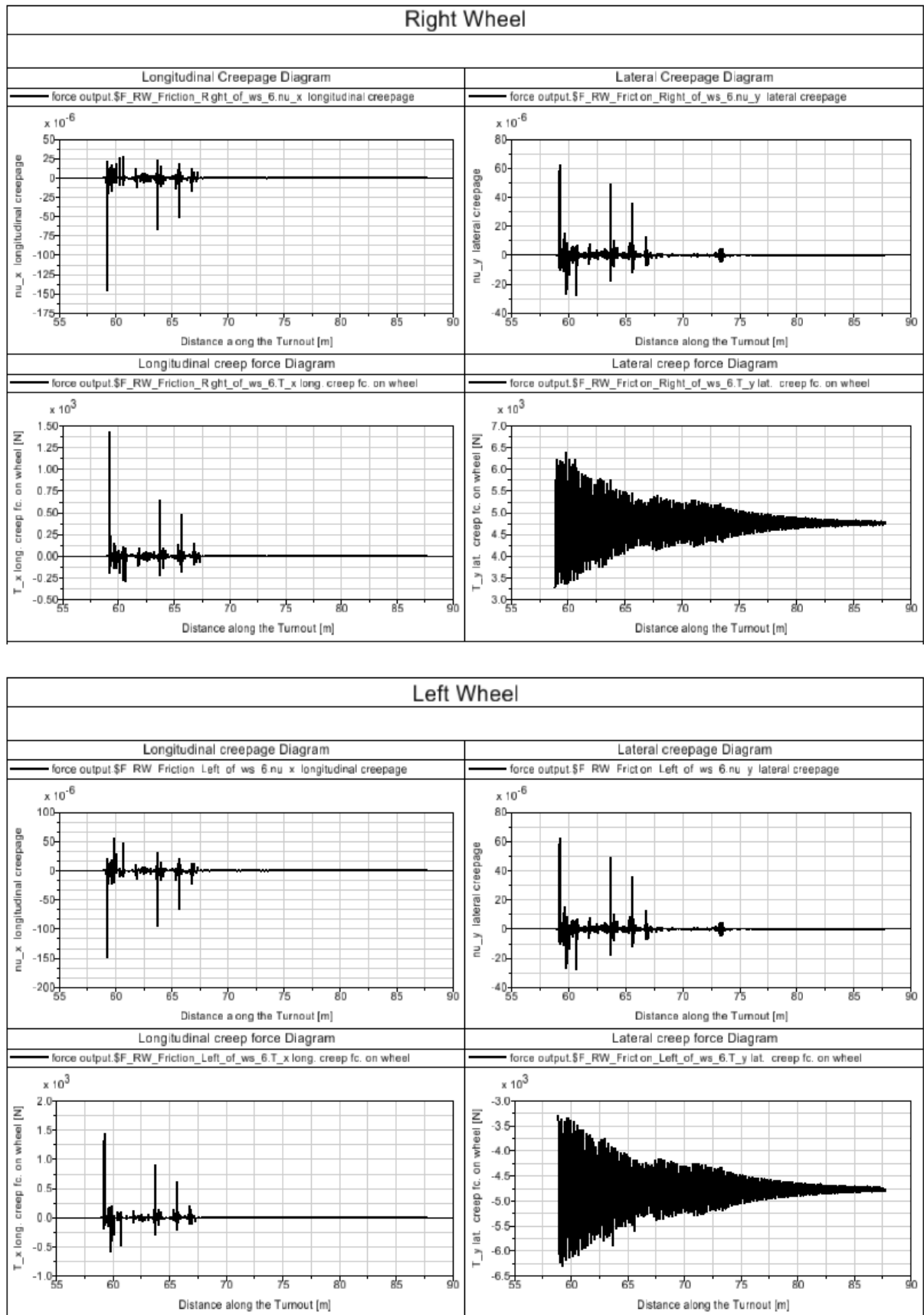


Fig.B-7: Creepage and creep force response due to wheel-set 6 in the through route

Table B-1: Damage calculation in the through route on the left rail

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	Ty	RCF Damage	Remark
1	58-64	61	1.33E-04	7.60E-05	1.33E+03	6.51E+03	0.6724489	0	No Damage
	64-70	67	3.00E-05	1.00E-05	2.58E+02	5.30E+03	0.06074	0	No Damage
	70-76	73	2.11E-04	1.90E-03	9.99E+02	1.03E+04	19.818789	9.63758E-07	RCF Damage
	76-82	79	9.70E-05	6.63E-04	7.30E+02	1.12E+04	7.49641	0	No Damage
	82-88	85	6.30E-05	3.65E-04	6.94E+02	1.05E+04	3.887172	0	No Damage
2	58-64	61	1.35E-04	4.80E-05	1.31E+03	6.42E+03	0.485175	0	No Damage
	64-70	67	7.00E-06	3.00E-03	7.67E+01	5.41E+03	16.230537	2.46107E-07	RCF Damage
	70-76	73	1.13E-04	2.75E-03	4.77E+02	1.09E+04	29.979401	2.99588E-06	RCF Damage
	76-82	79	2.09E-04	2.23E-03	1.68E+03	1.50E+04	33.755684	3.75114E-06	RCF Damage
	82-88	85	2.00E-04	2.30E-04	1.06E+03	1.43E+04	3.50529	0	No Damage
3	58-64	61	1.33E-04	4.00E-06	1.32E+04	4.67E+03	1.77827	0	No Damage
	64-70	67	7.00E-06	1.00E-06	4.73E+01	5.11E+03	0.0054408	0	No Damage
	70-76	73	0.00E+00	2.00E-06	8.16E-01	5.09E+03	0.01017	0	No Damage
	76-82	79	0.00E+00	0.00E+00	1.70E-02	4.81E+03	0	0	No Damage
	82-88	85	8.89E-05	5.60E-04	8.02E+02	9.15E+03	5.1969778	0	No Damage
4	58-64	61	1.10E-04	1.30E-05	1.10E+03	6.19E+03	0.20147	0	No Damage
	64-70	67	5.00E-06	0.00E+00	7.93E+01	5.50E+03	0.0003965	0	No Damage
	70-76	73	1.00E-06	0.00E+00	9.97E-01	5.27E+03	9.97E-07	0	No Damage
	76-82	79	0.00E+00	0.00E+00	8.00E-03	4.91E+03	0	0	No Damage
	82-88	85	3.70E-05	2.08E-04	4.00E+02	8.30E+03	1.7412	0	No Damage
5	58-64	61	1.10E-04	1.22E-06	1.26E+03	6.25E+03	0.1466589	0	No Damage
	64-70	67	7.00E-06	2.00E-07	1.00E+02	6.41E+03	0.001982	0	No Damage
	70-76	73	0.00E+00	2.50E-07	1.22E+00	5.28E+03	0.0013193	0	No Damage
	76-82	79	0.00E+00	0.00E+00	0.00E+00	4.95E+03	0	0	No Damage
	82-88	85	0.00E+00	0.00E+00	0.00E+00	4.86E+03	0	0	No Damage
6	58-64	61	1.48E-04	6.20E-05	1.44E+03	6.42E+03	0.611036	0	No Damage
	64-70	67	5.00E-05	3.60E-05	4.85E+02	5.52E+03	0.223063	0	No Damage
	70-76	73	0.00E+00	4.00E-06	1.45E+00	5.31E+03	0.021236	0	No Damage
	76-82	79	0.00E+00	0.00E+00	5.00E-02	5.00E+03	0	0	No Damage
	82-88	85	0.00E+00	0.00E+00	1.50E-02	4.86E+03	0	0	No Damage
Location on the Route			61	67	73	79	85		
Cummulative Damage			0	2.46107E-07	3.95964E-06	3.75114E-06	0		
Cummulative Damage due to RCF only			0	2.46107E-07	3.95964E-06	3.75114E-06	0		
Cummulative Damage due to Wear only			0	0	0	0	0		

Table B-2: Damage calculation in the through route on the right rail

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	T _y	RCF Damage	Remark
1	58-64	61	9.00E-05	7.60E-05	1.30E+03	3.42E+03	0.37692	0	No Damage
	64-70	67	5.00E-06	5.00E-06	4.70E+04	4.80E+03	0.259	0	No Damage
	70-76	73	1.75E-04	2.00E-03	9.50E+02	1.15E+04	23.24625	1.64925E-06	RCF Damage
	76-82	79	1.67E-04	7.00E-04	5.60E+02	1.15E+04	8.17152	0	No Damage
	82-88	85	5.00E-05	4.00E-04	5.11E+02	1.08E+04	4.34555	0	No Damage
2	58-64	61	1.33E-04	4.80E-05	1.33E+03	5.63E+03	0.44713	0	No Damage
	64-70	67	7.00E-06	4.00E-06	3.96E+01	4.09E+03	0.016637	0	No Damage
	70-76	73	1.56E-04	2.75E-03	6.00E+02	1.06E+04	29.3261	2.86522E-06	RCF Damage
	76-82	79	2.00E-04	2.25E-03	1.58E+03	1.62E+04	36.6985	4.3397E-06	RCF Damage
	82-88	85	6.67E-04	5.00E-04	3.83E+03	1.43E+04	9.68461	0	No Damage
3	58-64	61	1.38E-04	4.00E-06	1.35E+03	3.27E+03	0.198705	0	No Damage
	64-70	67	1.10E-05	1.00E-06	9.84E+01	5.40E+03	0.006482	0	No Damage
	70-76	73	0.00E+00	2.00E-06	1.10E+00	5.00E+03	0.010008	0	No Damage
	76-82	79	0.00E+00	0.00E+00	1.97E-02	4.69E+03	0	0	No Damage
	82-88	85	7.50E-05	6.00E-04	6.68E+02	1.04E+04	6.3192	0	No Damage
4	58-64	61	1.10E-04	1.30E-05	1.10E+04	3.27E+03	1.24811	0	No Damage
	64-70	67	1.00E-05	3.00E-06	1.12E+02	4.04E+03	0.01324	0	No Damage
	70-76	73	1.00E-06	1.00E-06	1.80E+00	4.54E+03	0.004545	0	No Damage
	76-82	79	0.00E+00	0.00E+00	7.60E-03	4.60E+03	0	0	No Damage
	82-88	85	2.90E-05	2.50E-05	2.80E+02	8.43E+03	0.21887	0	No Damage
5	58-64	61	1.30E-04	1.20E-06	1.26E+03	6.25E+03	0.171709	0	No Damage
	64-70	67	6.71E-06	2.00E-07	4.04E+01	5.52E+03	0.001375	0	No Damage
	70-76	73	0.00E+00	2.33E-07	1.16E+00	5.28E+03	0.00123	0	No Damage
	76-82	79	0.00E+00	0.00E+00	0.00E+00	4.98E+03	0	0	No Damage
	82-88	85	0.00E+00	0.00E+00	0.00E+00	4.85E+03	0	0	No Damage
6	58-64	61	1.50E-04	6.40E-05	1.45E+03	6.25E+03	0.61795	0	No Damage
	64-70	67	6.70E-05	3.60E-05	6.35E+02	5.49E+03	0.240152	0	No Damage
	70-76	73	0.00E+00	4.00E-06	1.02E+00	4.96E+03	0.01984	0	No Damage
	76-82	79	0.00E+00	0.00E+00	0.00E+00	5.00E+03	0	0	No Damage
	82-88	85	0.00E+00	0.00E+00	0.00E+00	4.84E+03	0	0	No Damage
Location on the Route			61	67	73	79	85		
Cummulative Damage			0	0	4.51447E-06	4.3397E-06	0		
Cummulative Damage due to RCF only			0	0	4.51447E-06	4.3397E-06	0		
Cummulative Damage due to Wear only			0	0	0	0	0		

As it can be seen from the results on the damage calculation, there is no any wear rather the through route is dominated by RCF Damage mechanism. Therefore, it is wiser to improve the RCF susceptibility of the rails and able to optimize in view of RCF minimization; such that the RCF dominance throughout the turnout is observable in this case.

B. Divergent route – scenario 1

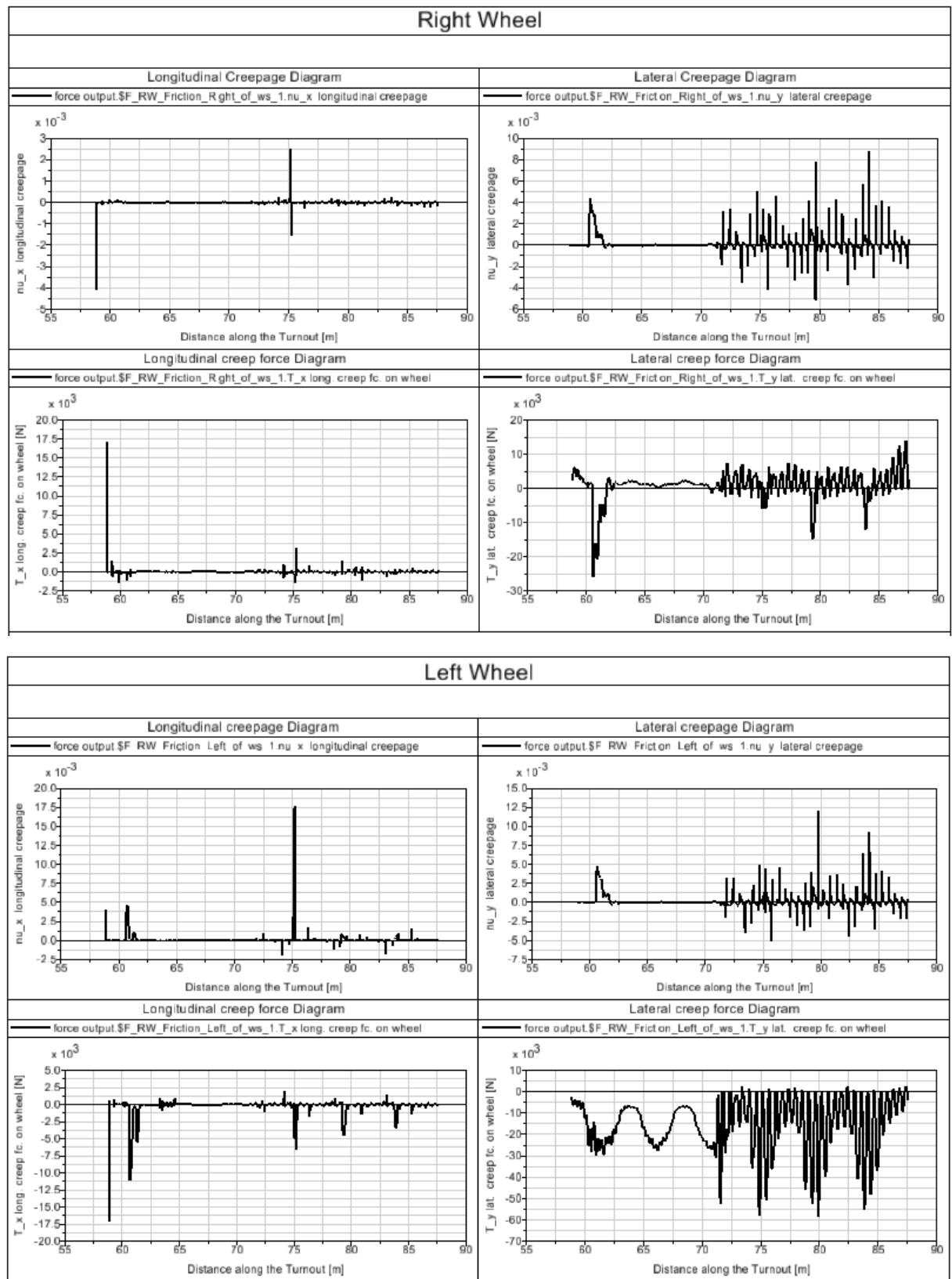


Figure B-8: Creepage and creep force response due to wheel-set 1 in the diverging route- case 1

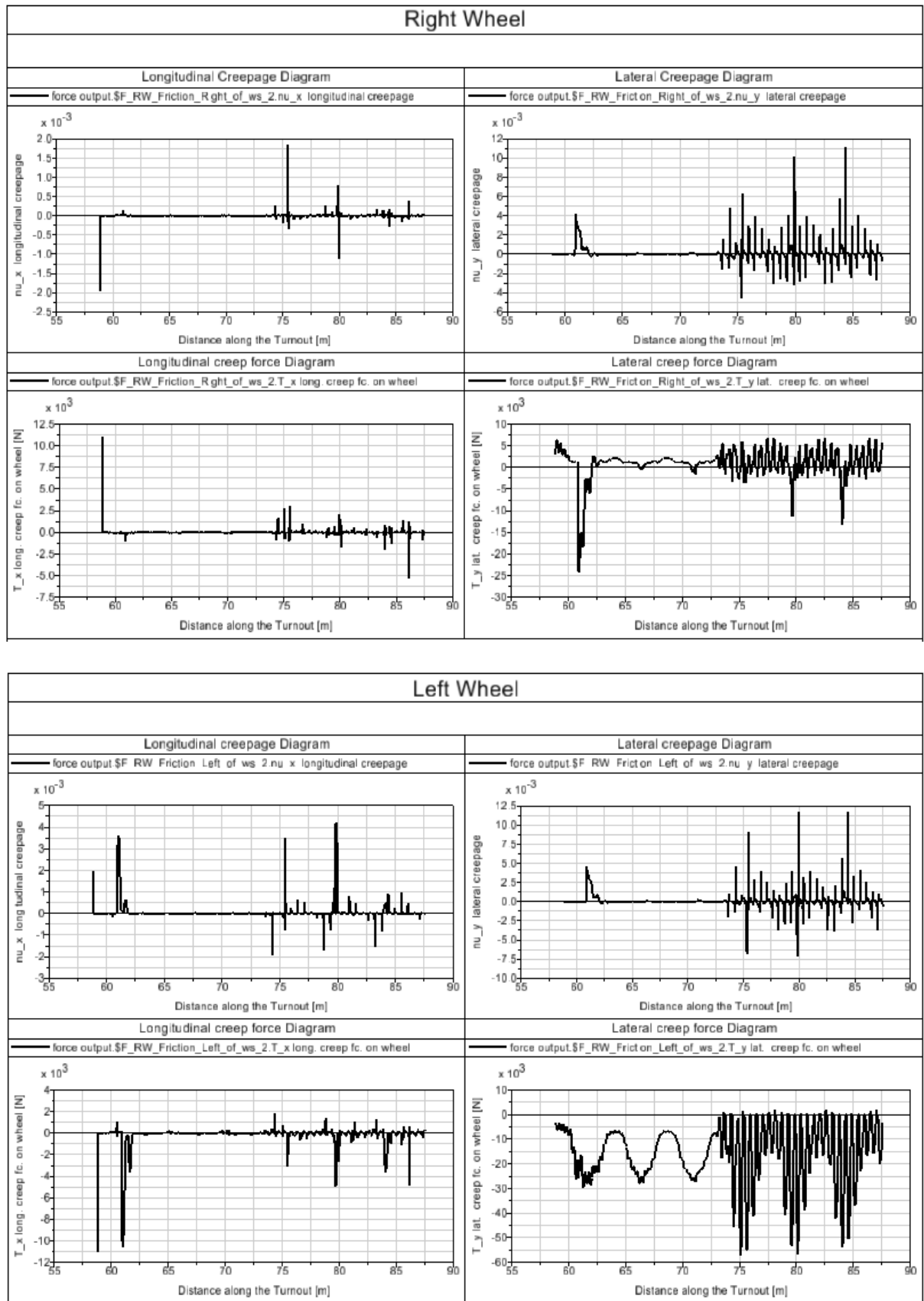


Figure B-9: Creepage and creep force response due to wheel-set 2 in the diverging route-
case 1

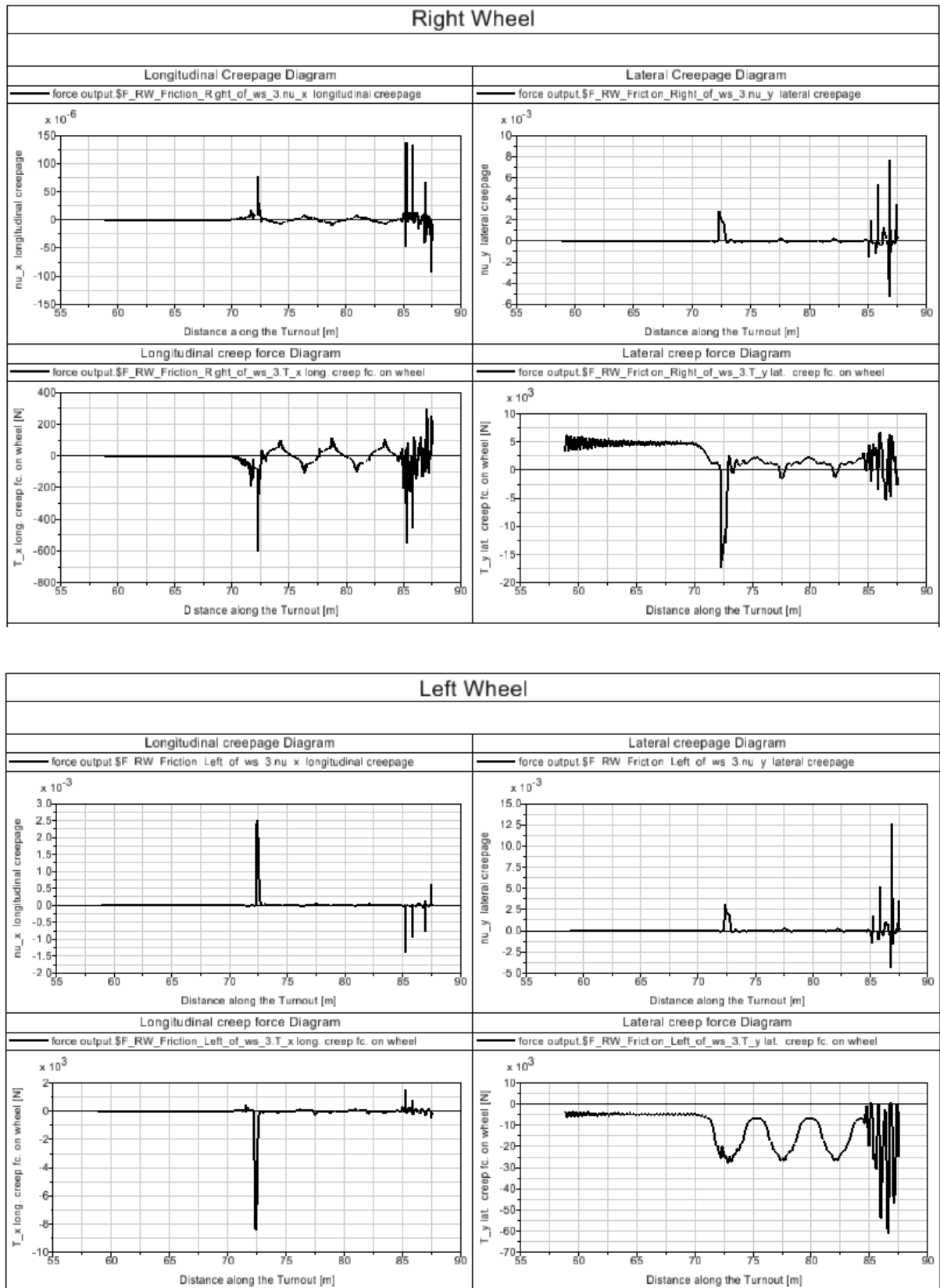


Figure B-10: Creepage and creep force response due to wheel-set 3 in the diverging route-
case 1

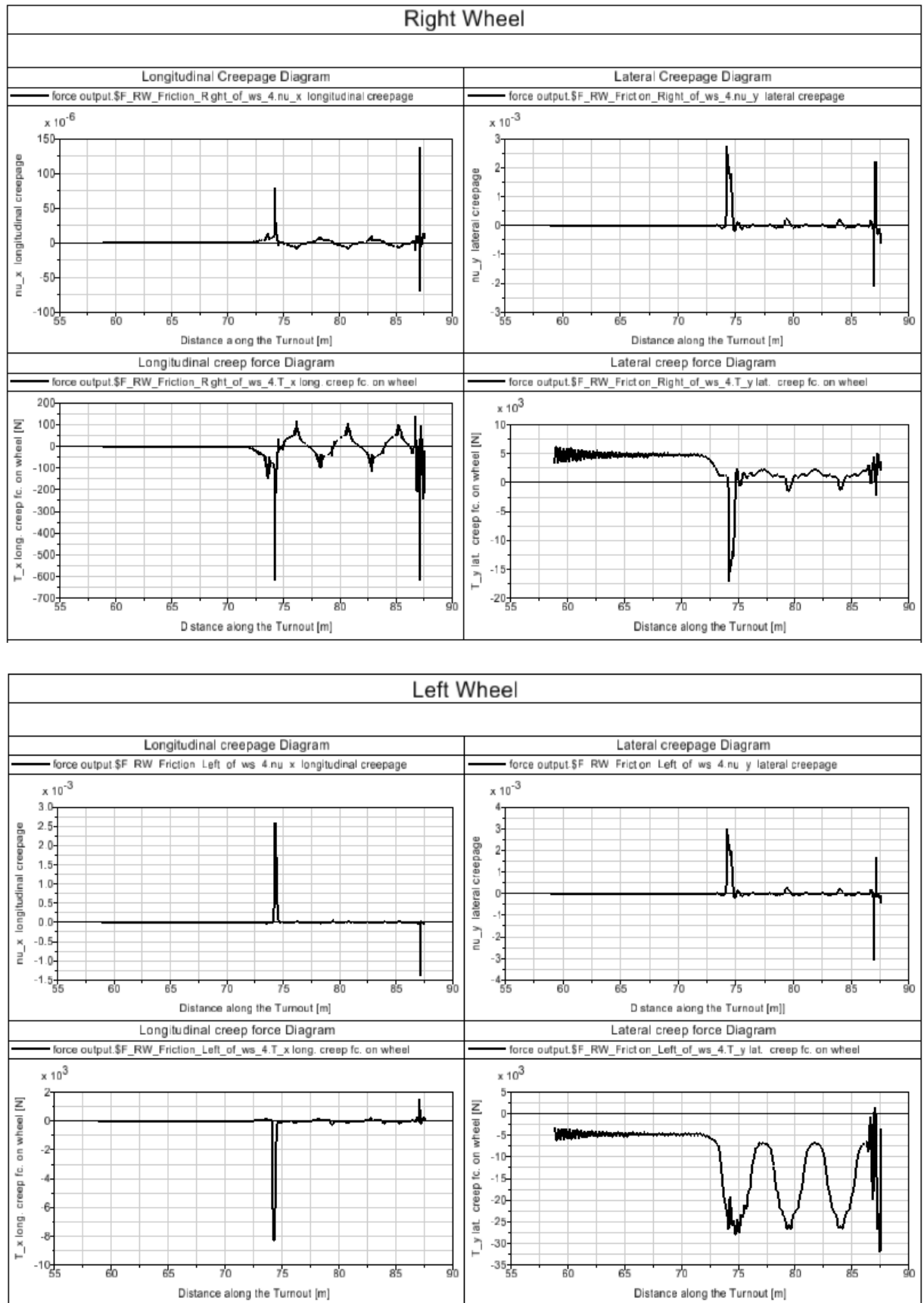


Figure B-11: Creepage and creep force response due to wheel-set 4 in the diverging route- case 1

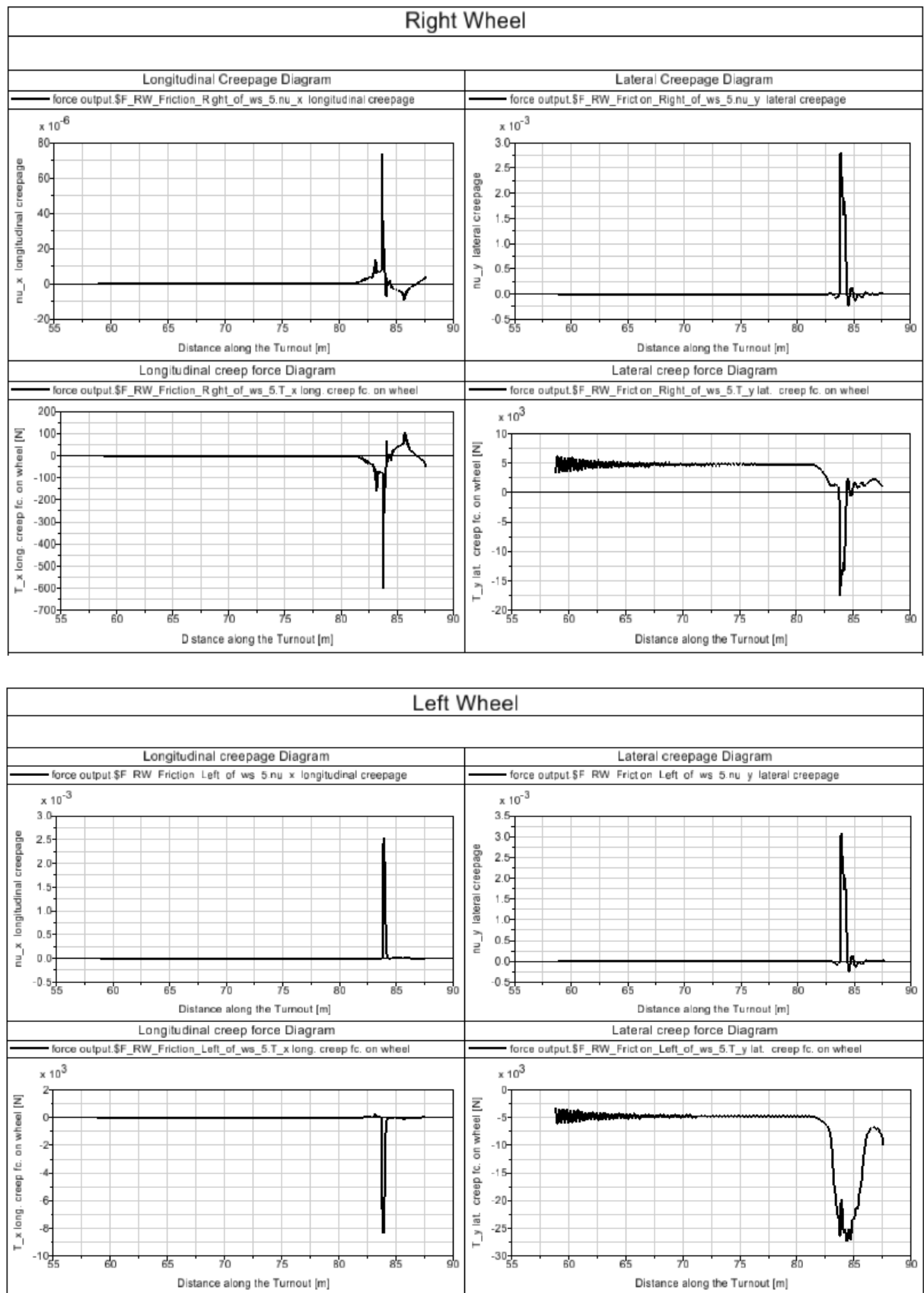


Figure B-12: Creepage and creep force response due to wheel-set 5 in the diverging route- case 1

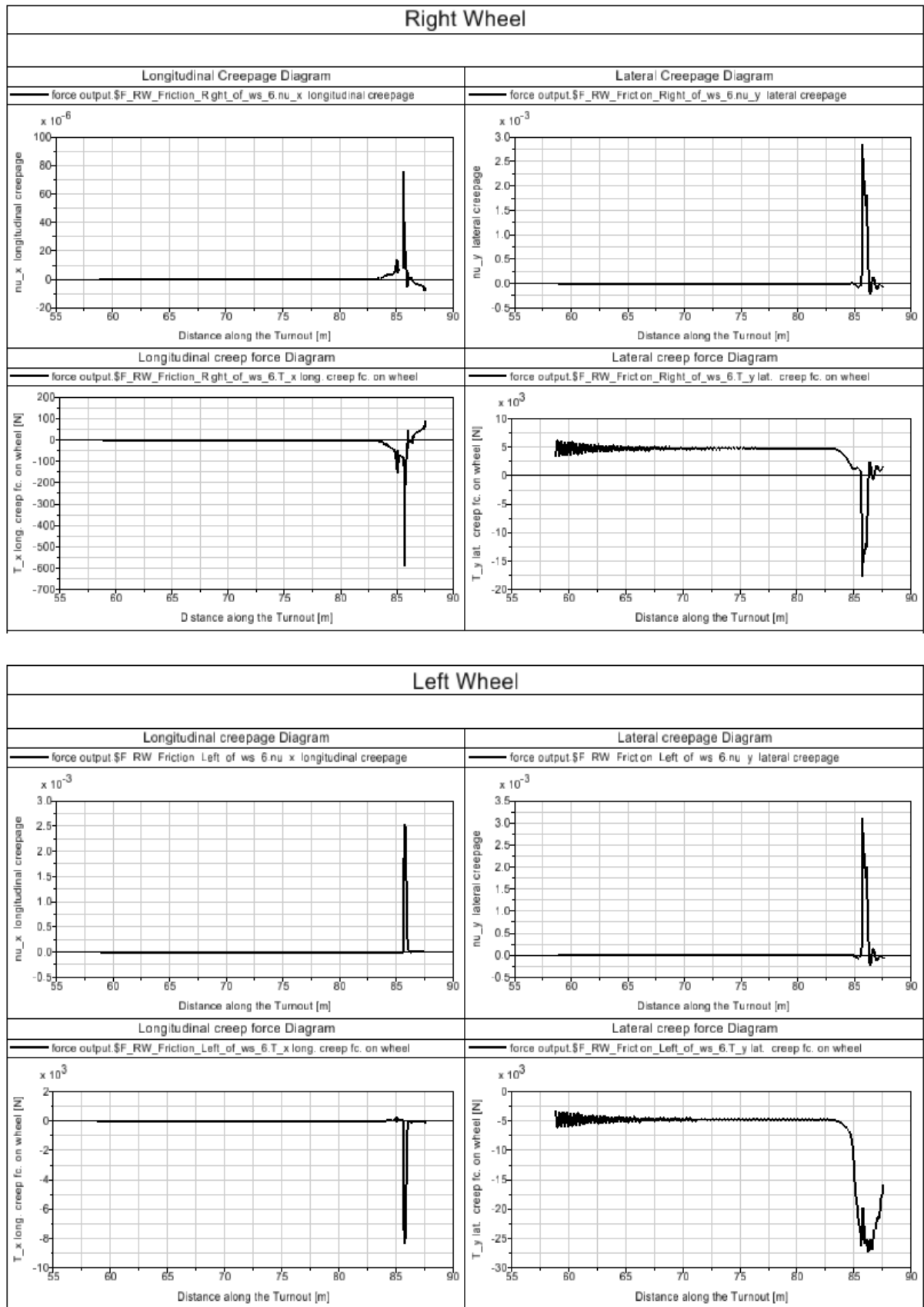


Figure B-13: Creepage and creep force response due to wheel-set 6 in the diverging route-case 1

Table B-3: Damage calculation in the diverging route on the right rail- Case 1

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	Ty	RCF Damage	Remark
1	55-62	58.5	4.00E-03	2.84E-03	1.72E+04	2.55E+04	141.1388	3.07907E-06	RCF Damage
	62-69	65.5	8.00E-06	1.17E-04	8.96E+01	2.14E+03	0.2514481	0	No Damage
	69-76	72.5	2.50E-03	5.00E-03	3.01E+03	7.45E+03	44.775	0.000005955	RCF Damage
	76-83	79.5	1.78E-04	7.90E-03	1.51E+03	1.36E+04	107.84219	6.10607E-06	RCF Damage
	83-90	86.5	2.50E-04	8.76E-03	5.65E+02	1.36E+04	119.43493	5.05217E-06	RCF Damage
2	55-62	58.5	1.98E-03	3.73E-03	1.11E+04	2.41E+04	111.85685	5.74109E-06	RCF Damage
	62-69	65.5	9.00E-06	2.68E-04	1.05E+02	2.24E+03	0.6001903	0	No Damage
	69-76	72.5	1.83E-03	6.00E-03	3.01E+03	5.72E+03	39.80898	4.9618E-06	RCF Damage
	76-83	79.5	1.12E-03	1.00E-02	2.06E+03	1.08E+04	110.14608	5.89662E-06	RCF Damage
	83-90	86.5	4.00E-04	1.10E-02	5.39E+03	1.26E+04	140.844	3.10587E-06	RCF Damage
3	55-62	58.5	0.00E+00	0.00E+00	6.89E-02	6.18E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	1.98E-02	5.36E+03	0	0	No Damage
	69-76	72.5	7.50E-05	2.56E-03	6.07E+03	1.71E+04	44.32371	5.86474E-06	RCF Damage
	76-83	79.5	6.92E-06	2.27E-04	1.13E+02	1.47E+03	0.3337882	0	No Damage
	83-90	86.5	1.35E-04	7.90E-03	5.52E+02	6.58E+03	52.024893	7.40498E-06	RCF Damage
4	55-62	58.5	0.00E+00	0.00E+00	0.00E+00	5.64E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	0.00E+00	5.42E+03	0	0	No Damage
	69-76	72.5	6.50E-05	2.38E-03	5.99E+02	1.70E+04	40.479863	5.09597E-06	RCF Damage
	76-83	79.5	8.36E-06	2.24E-04	1.15E+02	2.21E+03	0.4955517	0	No Damage
	83-90	86.5	1.19E-04	2.08E-03	6.20E+02	2.37E+03	4.9991605	0	No Damage
5	55-62	58.5	0.00E+00	0.00E+00	1.28E-01	5.57E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	6.20E+01	5.43E+03	0	0	No Damage
	69-76	72.5	0.00E+00	0.00E+00	1.01E+02	4.91E+03	0	0	No Damage
	76-83	79.5	7.00E-06	1.40E-05	4.95E+01	4.84E+03	0.0681487	0	No Damage
	83-90	86.5	7.40E-05	2.76E-03	6.01E+02	1.77E+04	48.78604	6.75721E-06	RCF Damage
6	55-62	58.5	0.00E+00	0.00E+00	1.28E-01	5.64E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	1.10E-05	5.49E+03	0	0	No Damage
	69-76	72.5	0.00E+00	0.00E+00	1.10E-05	4.73E+03	0	0	No Damage
	76-83	79.5	0.00E+00	4.60E+01	1.39E+00	4.84E+03	222732	-0.020232656	Wear Damage
	83-90	86.5	7.60E-05	2.35E-03	5.94E+02	1.79E+04	42.034944	5.40699E-06	RCF Damage

Location on the Route	58.5	65.5	72.5	79.5	86.5
Cummulative Damage	8.82E-06	0	2.18775E-05	-0.020220653	2.77272E-05
Cummulative Damage due to RCF only	8.82E-06	0	1.09168E-05	1.20027E-05	2.77272E-05
Cummulative Damage due to Wear only	0	0	0	-0.020232656	0

Table B-4: Damage calculation in the diverging route on the Left rail- Case 1

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	T _y	RCF Damage	Remark
1	55-62	58.5	4.51E-03	4.46E-03	1.72E+04	3.00E+04	211.1214	-3.28305E-06	Wear Damage
	62-69	65.5	6.50E-05	1.25E-04	9.67E+02	2.77E+04	3.529355	0	No Damage
	69-76	72.5	1.75E-05	5.00E-03	6.69E+03	5.74E+04	287.3071	-1.02091E-05	Wear Damage
	76-83	79.5	1.60E-03	1.20E-02	4.55E+03	5.80E+04	703.788	-4.80714E-05	Wear Damage
	83-90	86.5	1.61E-06	9.20E-03	3.35E+03	5.53E+04	508.7378	-3.03394E-05	Wear Damage
2	55-62	58.5	3.60E-03	3.75E-03	1.11E+04	2.93E+04	149.8818	2.28425E-06	RCF Damage
	62-69	65.5	1.70E-05	3.41E-01	1.65E+02	2.73E+04	9322.943	-0.000831639	Wear Damage
	69-76	72.5	3.46E-03	8.80E-03	3.10E+03	5.72E+04	514.4889	-3.08622E-05	Wear Damage
	76-83	79.5	4.20E-03	1.18E-02	4.89E+03	5.68E+04	688.0975	-4.66449E-05	Wear Damage
	83-90	86.5	1.00E-03	1.18E-02	4.89E+03	5.34E+04	631.872	-4.15335E-05	Wear Damage
3	55-62	58.5	0.00E+00	0.00E+00	7.52E-02	5.69E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	2.80E-01	4.73E+03	0	0	No Damage
	69-76	72.5	2.50E-03	2.89E-03	8.37E+03	2.72E+04	99.4413	6.86979E-06	RCF Damage
	76-83	79.5	4.70E-05	2.38E-04	2.49E+02	6.80E+03	1.629151	0	No Damage
	83-90	86.5	1.45E-03	1.25E-02	1.51E+03	6.15E+04	771.527	-5.42295E-05	Wear Damage
4	55-62	58.5	0.00E+00	0.00E+00	1.28E+02	6.18E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	3.18E-03	4.39E+03	0	0	No Damage
	69-76	72.5	2.58E-03	2.98E-03	8.29E+03	2.79E+04	104.6265	6.3984E-06	RCF Damage
	76-83	79.5	5.00E-05	2.37E+02	2.42E+02	2.68E+04	6359658	-0.5781406	Wear Damage
	83-90	86.5	1.25E-03	3.00E-03	1.49E+03	3.18E+04	97.12375	7.08048E-06	RCF Damage
5	55-62	58.5	0.00E+00	0.00E+00	1.28E+02	6.09E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	5.70E+01	4.39E+03	0	0	No Damage
	69-76	72.5	0.00E+00	0.00E+00	1.02E-04	5.05E+03	0	0	No Damage
	76-83	79.5	5.00E-06	8.13E-05	1.97E-02	9.47E+03	0.76983	0	No Damage
	83-90	86.5	2.48E-03	3.10E-03	8.28E+03	2.72E+04	104.7924	6.38332E-06	RCF Damage
6	55-62	58.5	0.00E+00	0.00E+00	1.28E-01	6.09E+03	0	0	No Damage
	62-69	65.5	0.00E+00	0.00E+00	4.50E-05	5.48E+03	0	0	No Damage
	69-76	72.5	0.00E+00	0.00E+00	6.40E-05	5.05E+03	0	0	No Damage
	76-83	79.5	0.00E+00	0.00E+00	5.28E+03	4.76E+03	0	0	No Damage
	83-90	86.5	2.50E-03	3.08E-03	8.29E+03	2.72E+04	104.3994	6.41905E-06	RCF Damage

Location on the Route	58.5	65.5	72.5	79.5	86.5
Cummulative Damage	-9.99E-07	-0.000831639	-2.7803E-05	-0.578235316	-0.000106219
Cummulative Damage due to RCF only	2.28E-06	0	0	0	1.98829E-05
Cummulative Damage due to Wear only	-3.28E-06	-0.000831639	-4.1071E-05	-0.578235316	-0.000126102

In this scenario the form of damage mechanism shifts from a Wear mode into the RCF damage mechanism; from the damage contributions due to the first two wheel-sets to the following wheel-sets and the RCF domination caused by the last few wheel-sets location is found to be away from the crossing nose localities. Rather the crossing nose, in this scenario, is dominated by a net cumulative wear on the part of the rail and wear exist every-where.

C. Divergent route – scenario 2

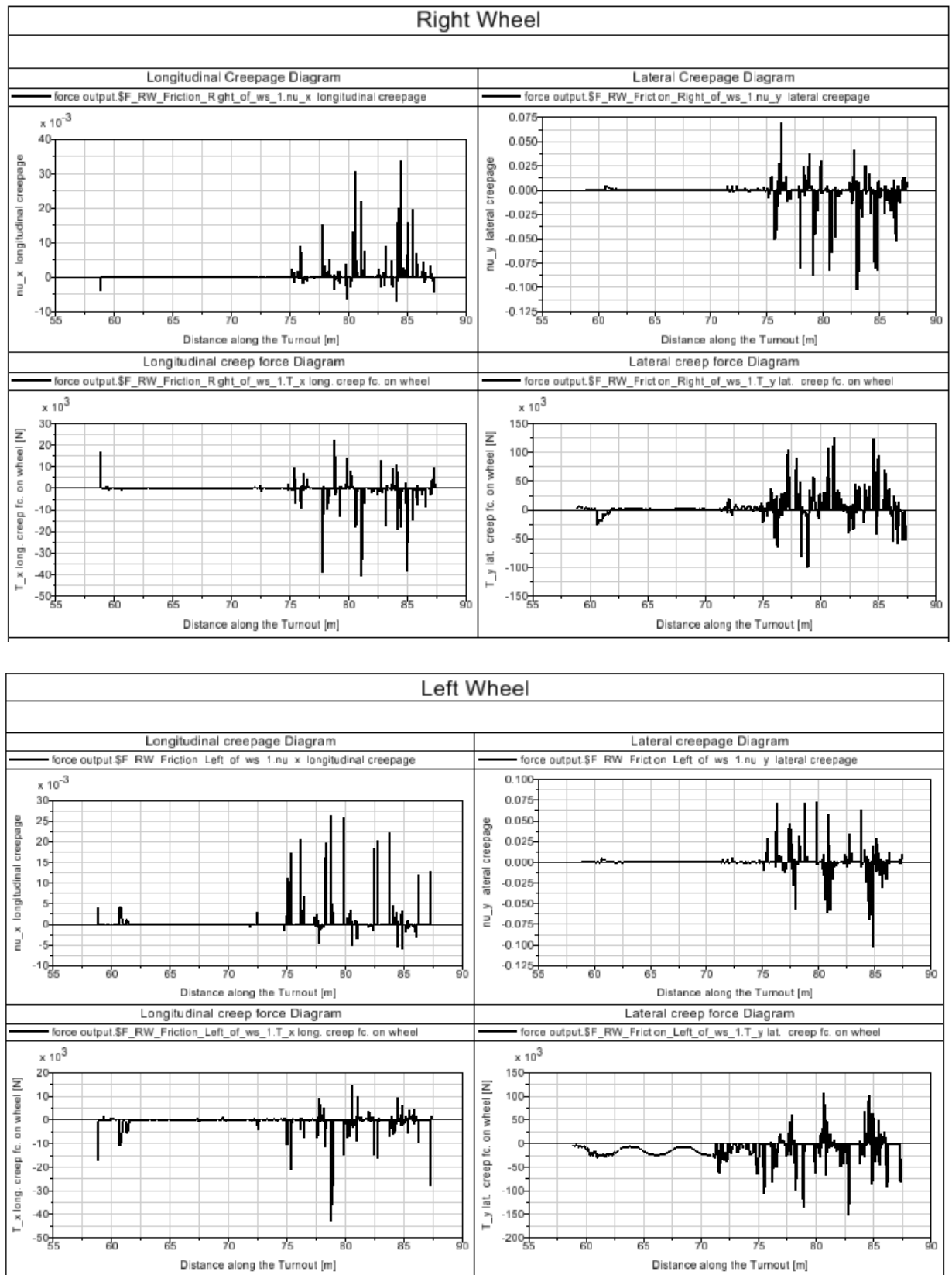


Figure B-14: Creepage and creep force response due to wheel-set 1 in the diverging route-case 2

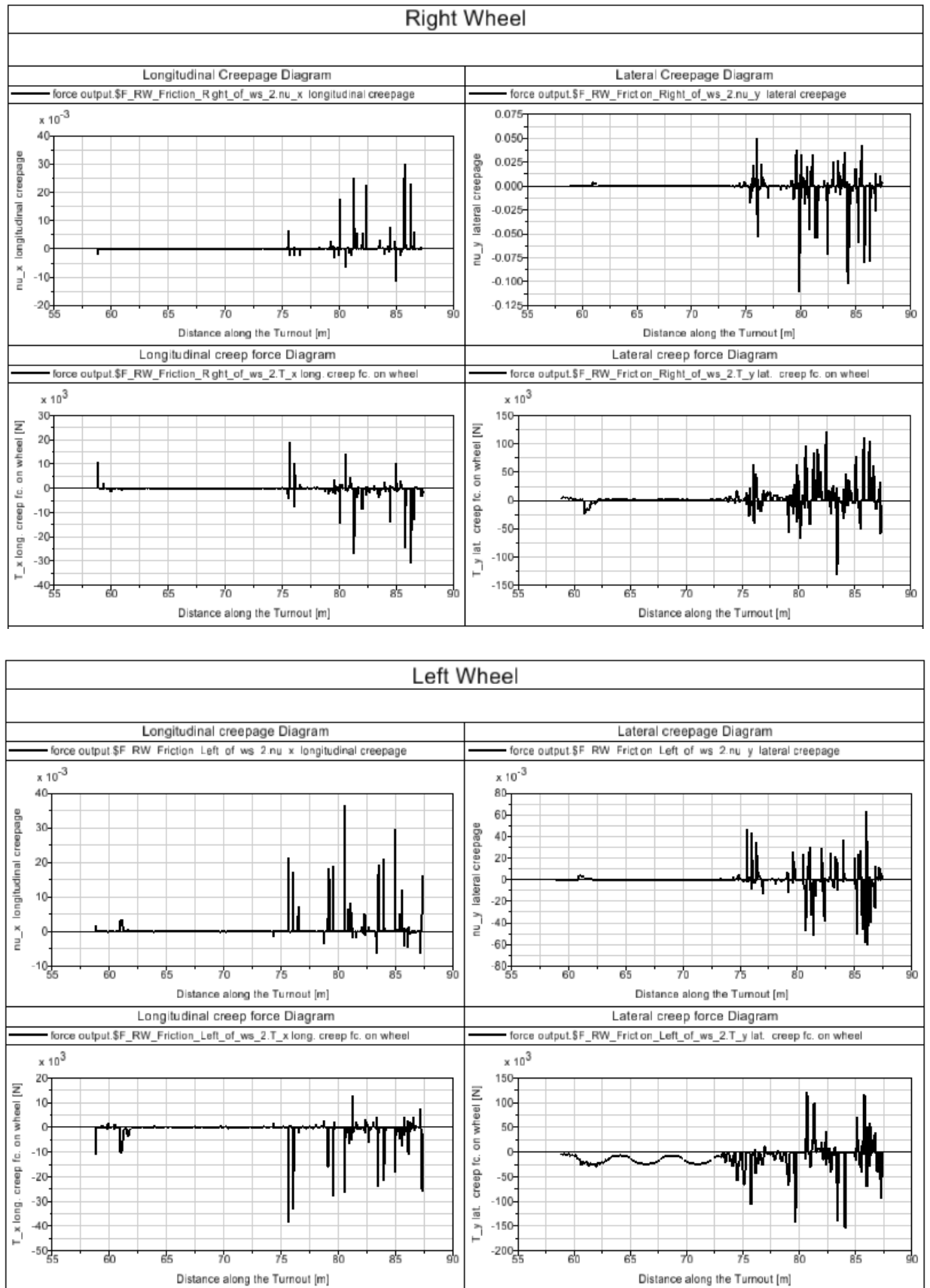


Figure B-15: Creepage and creep force response due to wheel-set 2 in the diverging route-case 2

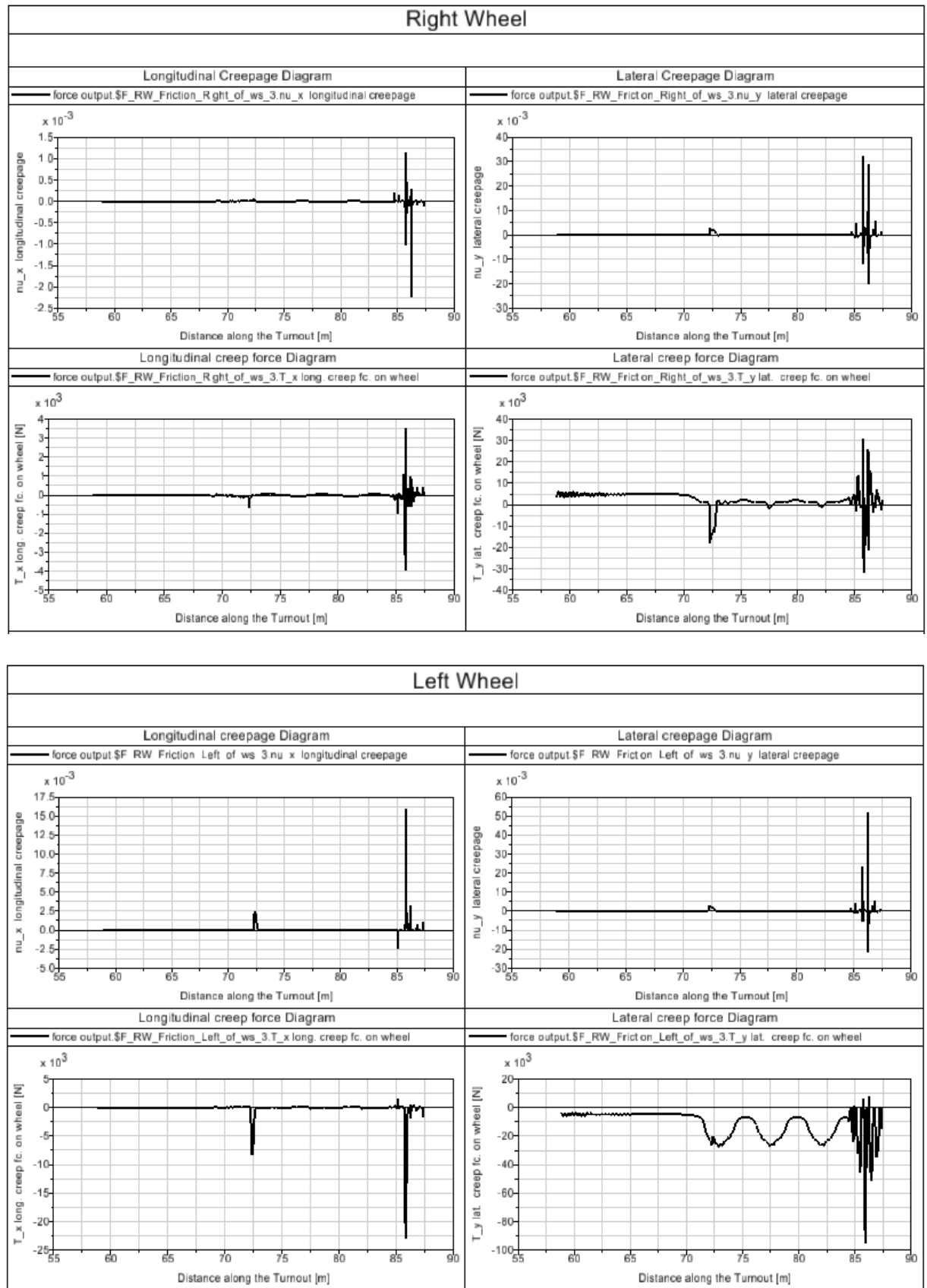


Figure B-16: Creepage and creep force response due to wheel-set 3 in the diverging route-case 2

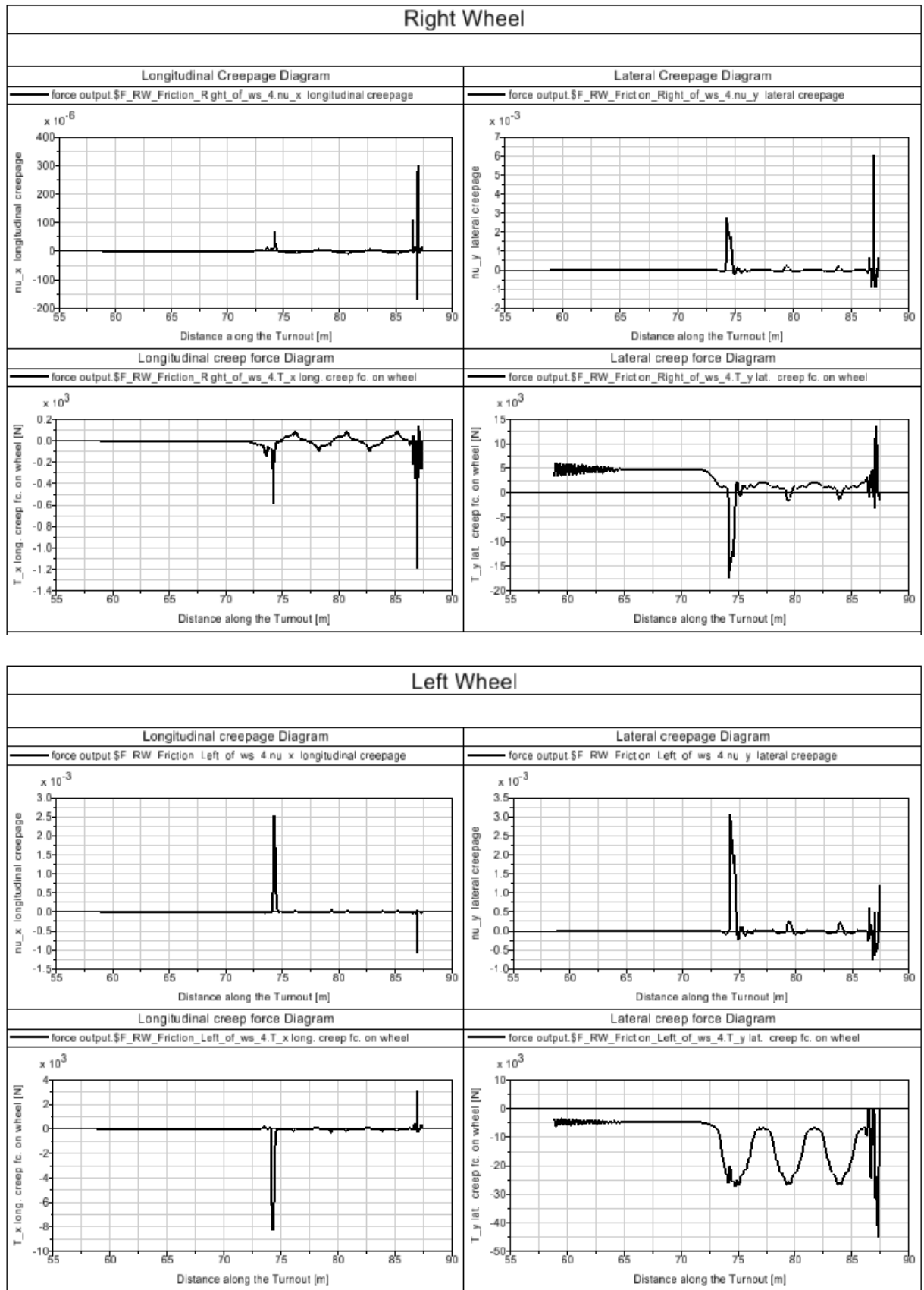


Figure B-17: Creepage and creep force response due to wheel-set 4 in the diverging route-case 2

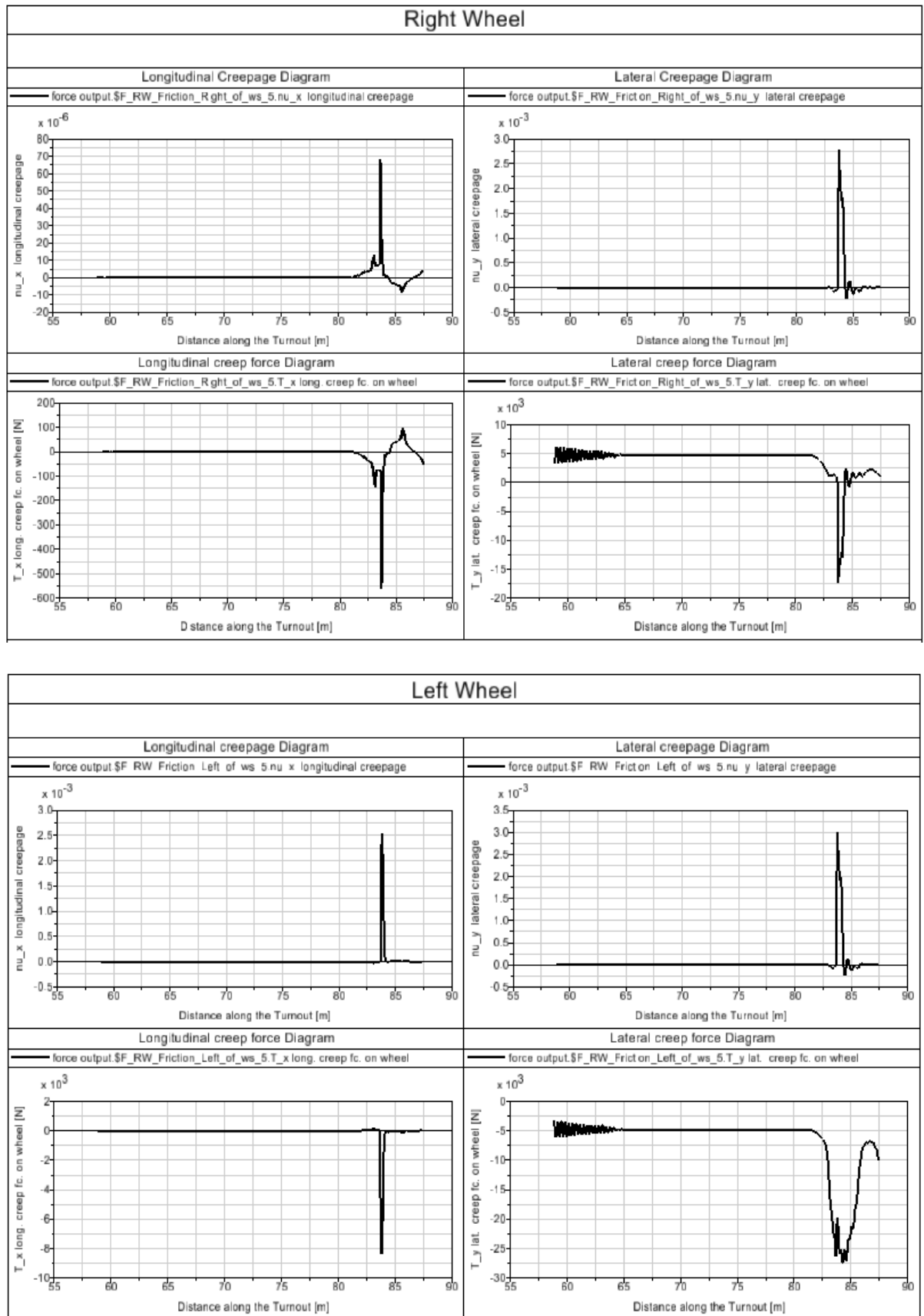


Figure B-18: Creepage and creep force response due to wheel-set 5 in the diverging route-case 2

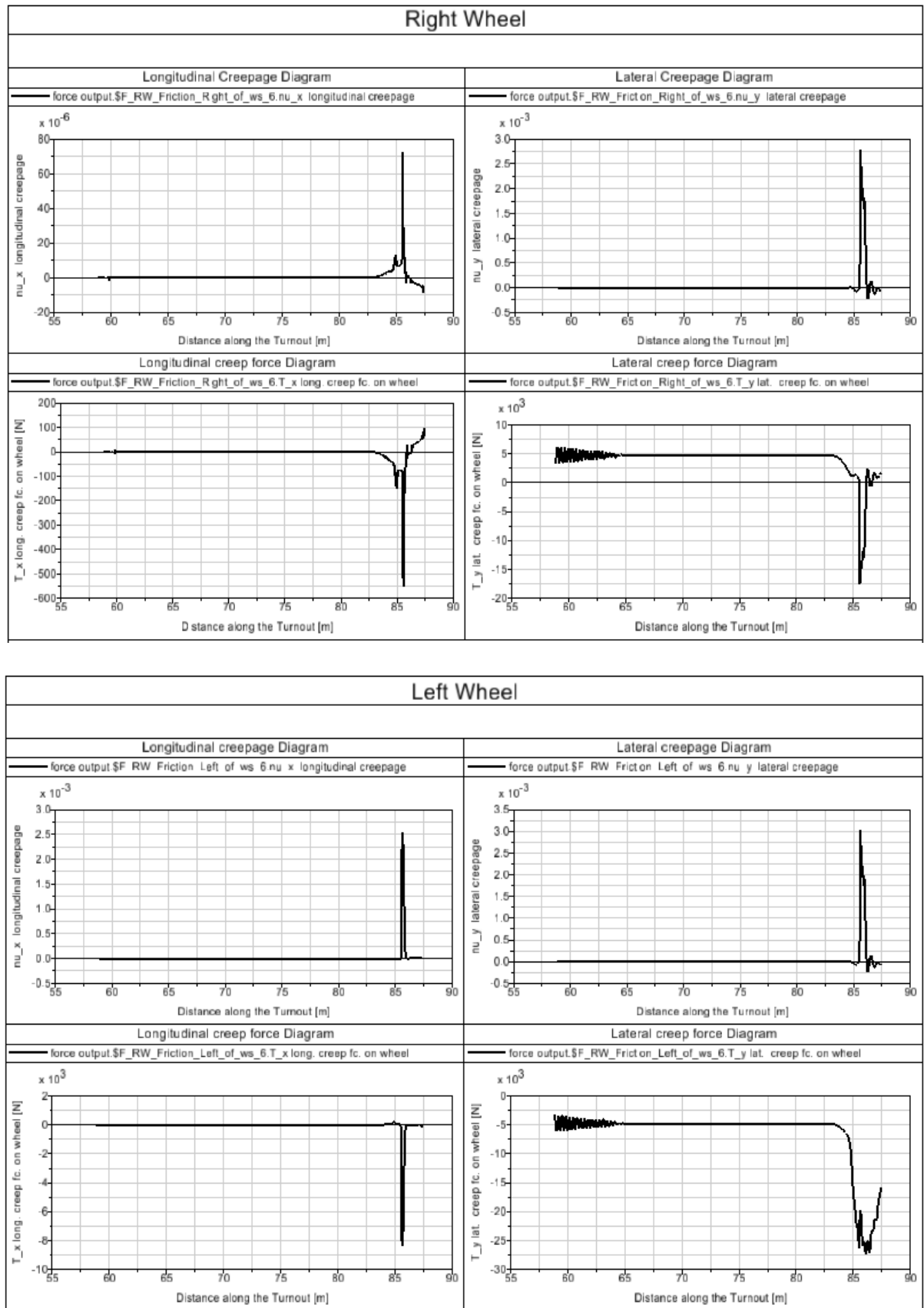


Figure B-19 Creepage and creep force response due to wheel-set 6 in the diverging route-
case 2

Table B-5: Damage calculation in the diverging route on the right rail- Case 2

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	Ty	RCF Damage	Remark
1	57-63	60	4.11E-03	4.26E-03	1.72E+04	2.63E+04	182.42644	-6.74388E-07	Wear Damage
	63-69	66	7.00E-06	6.20E-05	9.18E+01	2.14E+03	0.1331366	0	No Damage
	69-75	72	5.90E-05	4.99E-03	1.75E+03	2.34E+04	116.89414	5.28315E-06	RCF Damage
	75-81	78	3.01E-02	8.80E-02	3.91E+04	1.07E+05	10565.877	-0.000944634	Wear Damage
	81-87	84	3.34E-05	1.02E-01	4.10E+04	1.23E+05	12581.026	-0.001127831	Wear Damage
2	57-63	60	1.95E+03	3.73E-03	1.11E+04	2.38E+04	21550813	-1.959168469	Wear Damage
	63-69	66	7.00E-06	1.23E-04	9.58E+01	2.22E+03	0.2739766	0	No Damage
	69-75	72	1.30E-05	1.34E-04	2.85E+02	1.70E+04	2.285055	0	No Damage
	75-81	78	1.80E-02	1.12E-01	1.90E+04	9.38E+04	10842.764	-0.000969806	Wear Damage
	81-87	84	3.00E-02	1.00E-01	3.10E+04	1.32E+05	14161.85	-0.001271544	Wear Damage
3	57-63	60	0.00E+00	0.00E+00	3.31E-01	5.39E+03	0	0	No Damage
	63-69	66	8.00E-06	2.00E-06	5.42E-01	5.38E+03	0.0107563	0	No Damage
	69-75	72	5.90E-05	2.19E-03	5.50E+02	1.79E+04	39.33857	4.86771E-06	RCF Damage
	75-81	78	1.00E-05	2.32E+02	1.01E+02	2.19E+03	508312	-0.046194734	Wear Damage
	81-87	84	2.25E-03	3.40E-02	3.99E+03	3.19E+04	1094.3483	-8.35772E-05	Wear Damage
4	57-63	60	0.00E+00	0.00E+00	4.90E+02	5.64E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	8.86E-02	5.38E+03	0	0	No Damage
	69-75	72	7.00E-05	2.74E-03	5.85E+02	1.75E+04	48.10055	6.62011E-06	RCF Damage
	75-81	78	8.00E-06	2.25E-04	9.36E+01	2.18E+03	0.4921488	0	No Damage
	81-87	84	3.00E-04	6.05E-06	1.19E+03	1.35E+04	0.4386352	0	No Damage
5	57-63	60	0.00E+00	0.00E+00	5.40E+00	5.61E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	1.50E+02	5.36E+03	0	0	No Damage
	69-75	72	0.00E+00	0.00E+00	1.43E+02	4.77E+03	0	0	No Damage
	75-81	78	0.00E+00	0.00E+00	4.93E+01	4.77E+03	0	0	No Damage
	81-87	84	6.80E-05	2.75E-03	5.63E+02	1.75E+04	48.034034	6.60681E-06	RCF Damage
6	57-63	60	0.00E+00	0.00E+00	3.20E+00	5.64E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	2.35E+02	5.40E+03	0	0	No Damage
	69-75	72	0.00E+00	0.00E+00	2.08E+02	4.77E+03	0	0	No Damage
	75-81	78	0.00E+00	0.00E+00	5.50E+03	4.77E+03	0	0	No Damage
	81-87	84	7.15E-05	2.75E-03	5.55E+02	1.76E+04	48.329683	6.66594E-06	RCF Damage

Location on the Route	60	66	72	78	84
Cummulative Damage	-1.95917	0	1.6771E-05	-0.048109174	-0.002469679
Cummulative Damage due to RCF only	0	0	5.28315E-06	0	1.32727E-05
Cummulative Damage due to Wear only	-1.95917	0	0	-0.048109174	-0.002482952

Table B-6: Damage calculation in the diverging route on the left rail- Case 2

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	T _y	RCF Damage	Remark
1	57-63	60	4.48E-03	4.62E-03	1.71E+04	2.93E+04	212.3156	-3.39161E-06	Wear Damage
	63-69	66	1.12E-04	5.70E-05	7.57E+02	2.43E+04	1.472278	0	No Damage
	69-75	72	2.50E-03	4.80E-03	9.94E+03	6.32E+04	328.2747	-1.39335E-05	Wear Damage
	75-81	78	3.80E-02	7.30E-02	4.28E+04	1.33E+05	11360.303	-0.001016855	Wear Damage
	81-87	84	2.25E-02	1.03E-01	2.77E+04	1.52E+05	16244.8885	-0.001460913	Wear Damage
2	57-63	60	3.58E-03	3.84E-03	1.11E+04	2.79E+04	146.85218	2.55967E-06	RCF Damage
	63-69	66	3.80E-05	6.30E-05	3.03E+02	2.48E+04	1.571898	0	No Damage
	69-75	72	1.67E-03	5.66E-04	2.08E+03	5.20E+04	32.887572	3.57751E-06	RCF Damage
	75-81	78	2.15E-02	4.80E-02	3.84E+04	1.41E+05	7612.5545	-0.000676147	Wear Damage
	81-87	84	3.60E-02	6.10E-02	2.63E+04	1.54E+05	10339.839	-0.000924085	Wear Damage
3	57-63	60	0.00E+00	0.00E+00	2.86E+00	5.80E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	3.68E-01	5.33E+03	0	0	No Damage
	69-75	72	2.51E-03	2.57E-03	7.94E+03	2.68E+04	88.85662	7.83204E-06	RCF Damage
	75-81	78	5.00E-05	2.53E-04	1.53E+02	2.65E+04	6.721258	0	No Damage
	81-87	84	1.61E-02	5.20E-02	2.31E+04	9.50E+04	5313.94778	-0.000467181	Wear Damage
4	57-63	60	0.00E+00	0.00E+00	2.95E+00	5.96E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	1.20E-01	5.38E+03	0	0	No Damage
	69-75	72	2.50E-03	3.05E-03	7.93E+03	2.72E+04	102.8302	6.56171E-06	RCF Damage
	75-81	78	5.20E-05	2.50E-04	2.16E+02	2.63E+04	6.5814612	0	No Damage
	81-87	84	1.08E-03	1.20E-03	3.16E+03	4.46E+04	56.9316	8.38632E-06	RCF Damage
5	57-63	60	0.00E+00	0.00E+00	5.36E+00	6.07E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	1.49E+02	5.27E+03	0	0	No Damage
	69-75	72	0.00E+00	0.00E+00	1.38E+02	4.77E+03	0	0	No Damage
	75-81	78	0.00E+00	0.00E+00	6.30E-01	4.77E+03	0	0	No Damage
	81-87	84	2.50E-03	3.00E-03	8.31E+03	2.72E+04	102.4575	6.59559E-06	RCF Damage
6	57-63	60	0.00E+00	0.00E+00	5.15E+00	6.07E+03	0	0	No Damage
	63-69	66	0.00E+00	0.00E+00	2.23E+02	5.40E+03	0	0	No Damage
	69-75	72	0.00E+00	0.00E+00	2.08E+02	4.77E+03	0	0	No Damage
	75-81	78	0.00E+00	0.00E+00	5.48E+03	4.77E+03	0	0	No Damage
	81-87	84	2.50E-03	3.00E-03	8.22E+03	2.72E+04	102.113	6.62691E-06	RCF Damage

Location on the Route	60	66	72	78	84
Cummulative Damage	-8.31943E-07	0	4.03781E-06	-0.001693002	-0.00283057
Cummulative Damage dure to RFC only	2.55967E-06	0	3.57751E-06	0	2.16088E-05
Cummulative Damage dure to Wear only	-3.39161E-06	0	-1.3933E-05	-0.001693002	-0.002852179

In the sense of damage mode sensitivity, wear damage mechanism is the prior mode of damage almost throughout the turnout except the around the entrance to the crossing nose which is not harmed that much or at all. However, in a global sense the wear damage dominates as well the RCF exist with an immediate removal of the RCF crack with the wear damage due to the first few wheel-sets. And the left rail wears much more than the right rail.

D. Divergent route – scenario 3

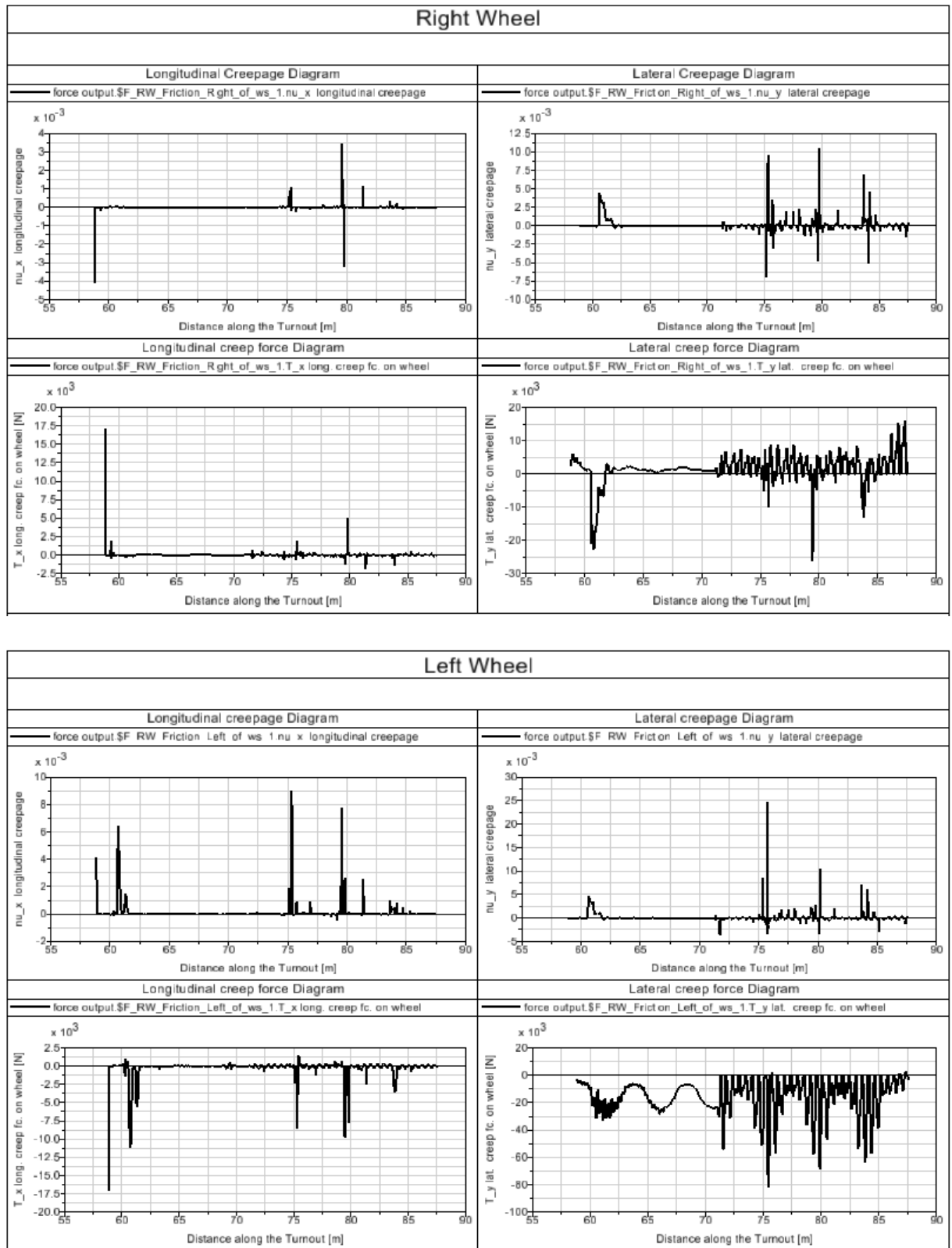


Figure B-20 Creepage and creep force response due to wheel-set 1 in the diverging route- case 3

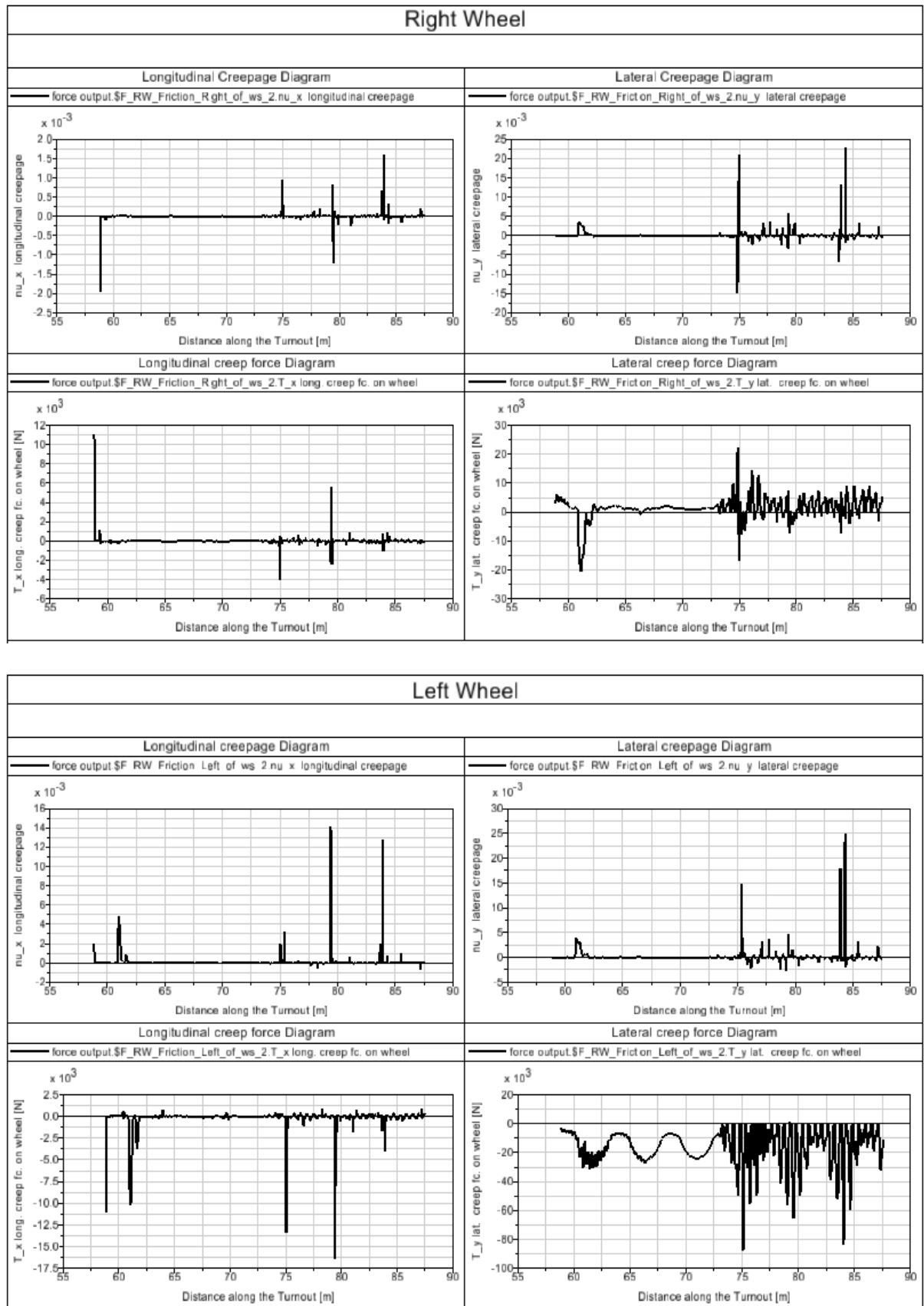


Figure B-21 Creepage and creep force response due to wheel-set 2 in the diverging route-case 3

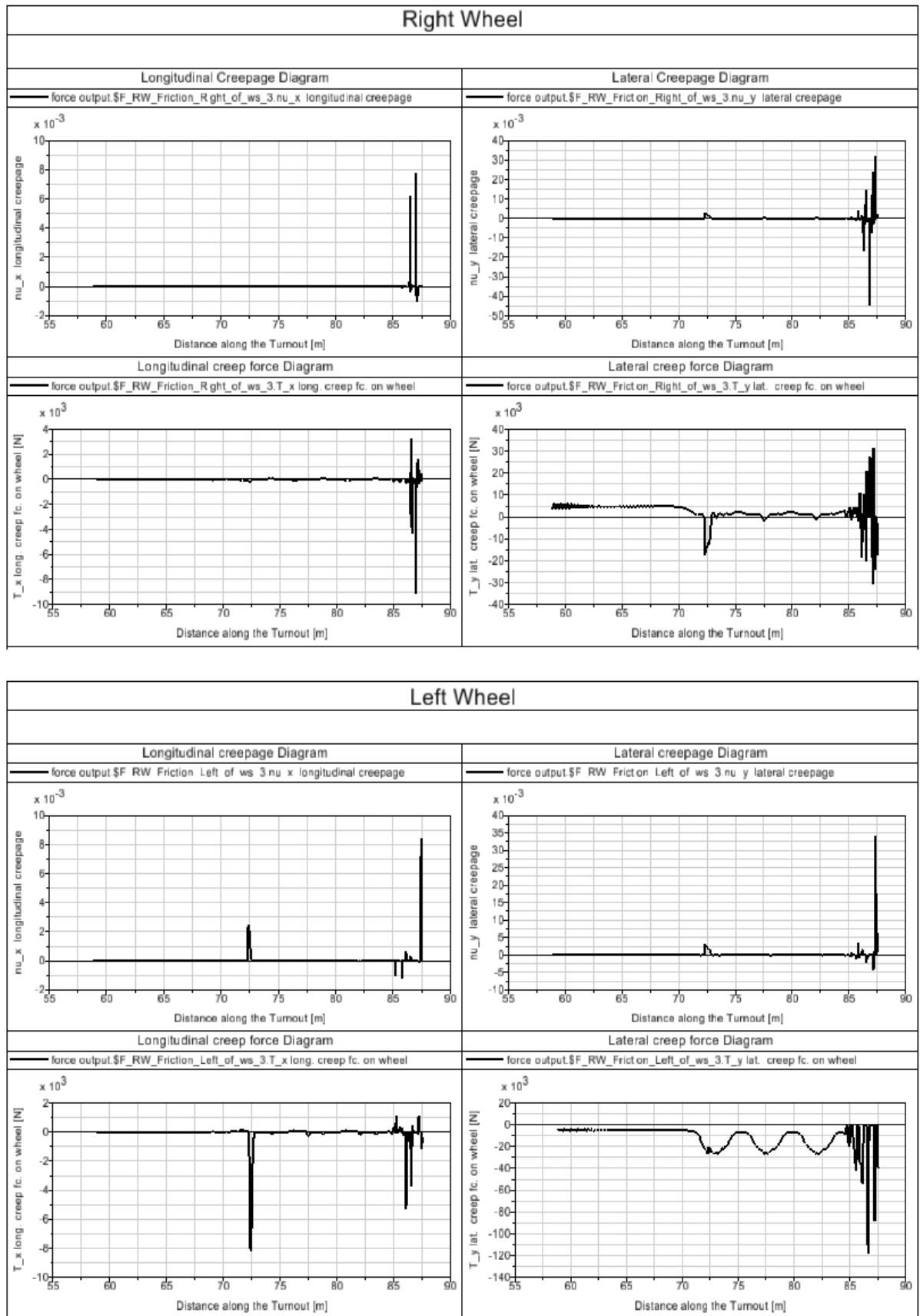


Figure B-22 Creepage and creep force response due to wheel-set 3 in the diverging route- case 3

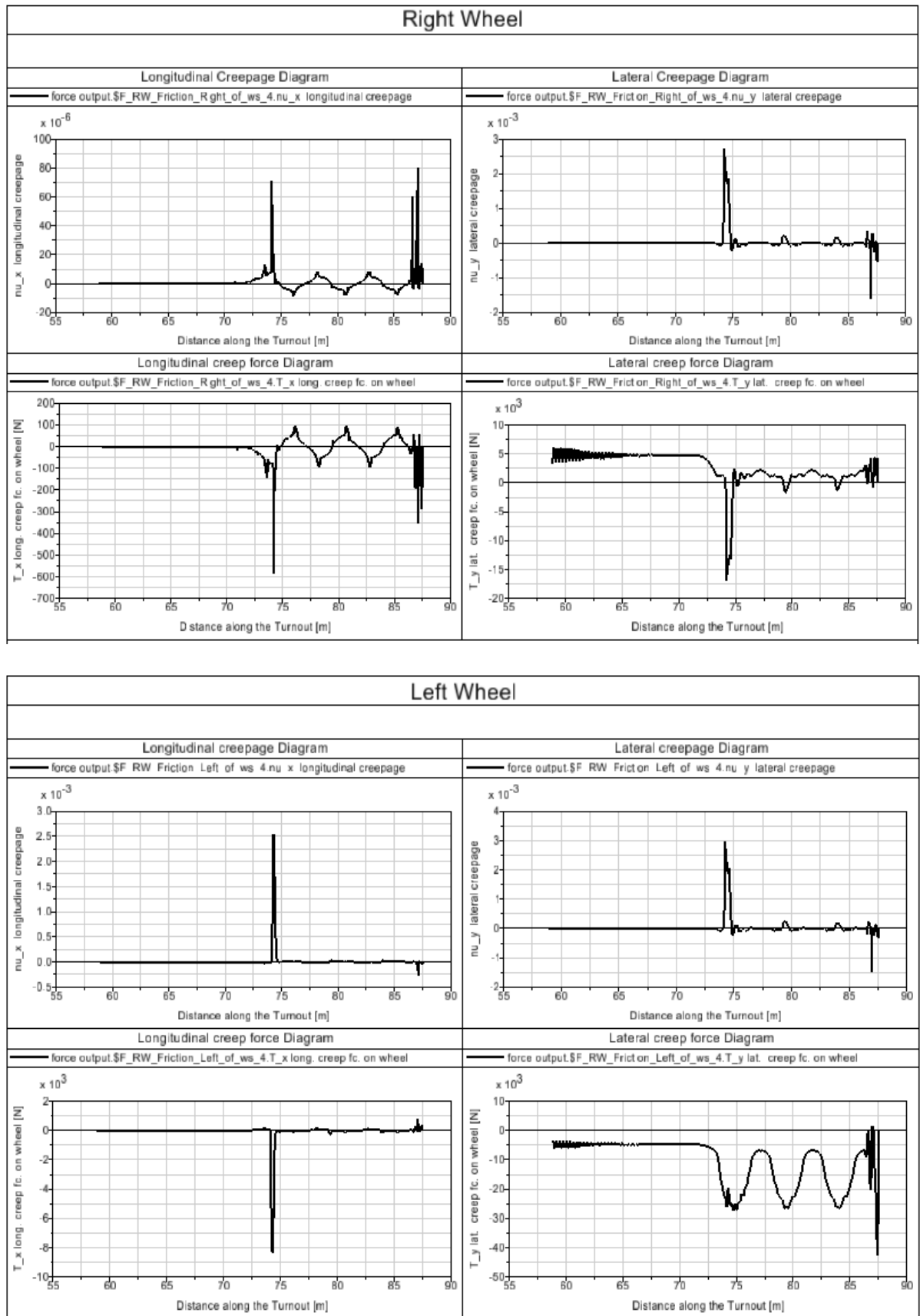


Figure B-23 Creepage and creep force response due to wheel-set 4 in the diverging route- case 3

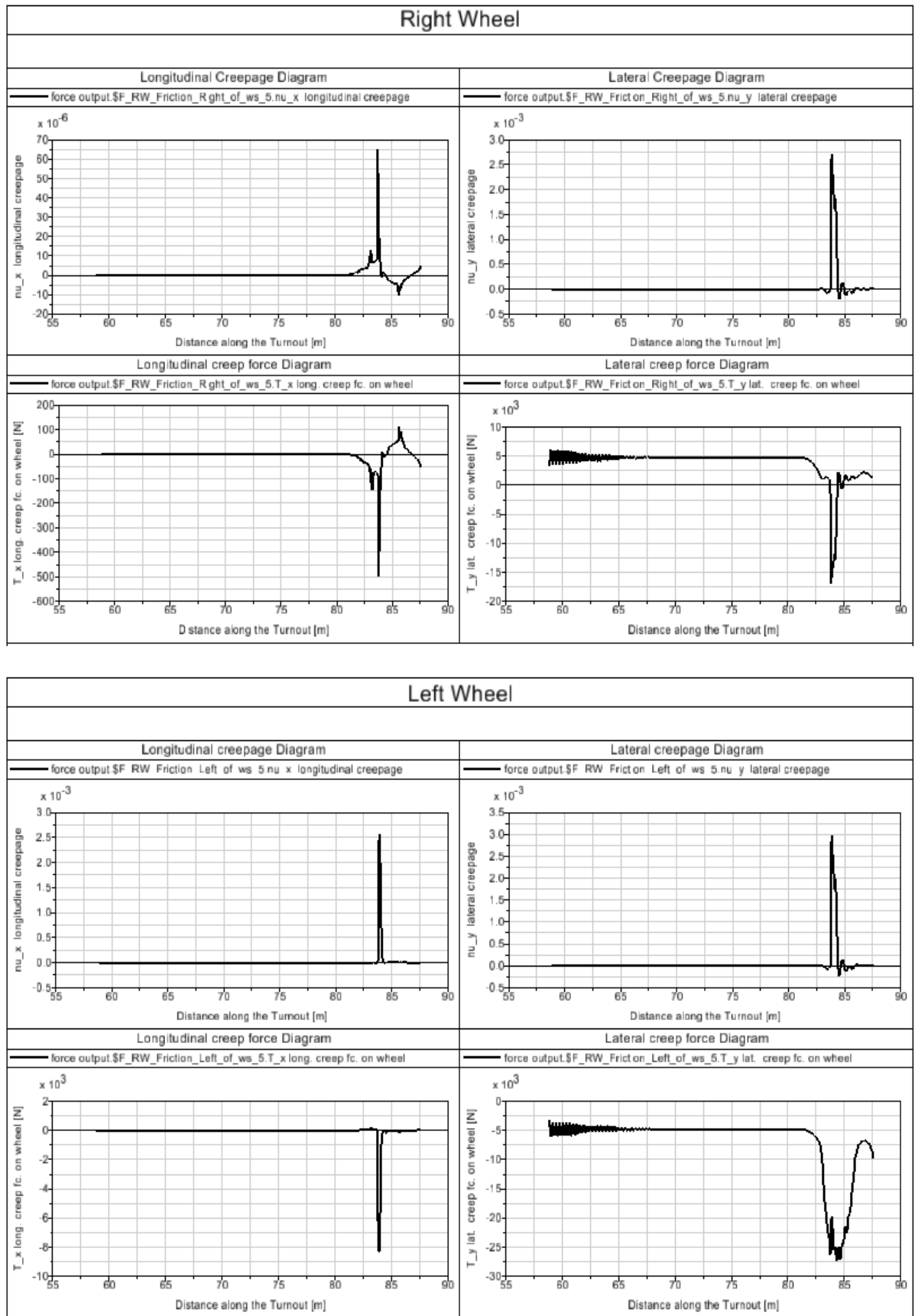


Figure B-24 Creepage and creep force response due to wheel-set 5 in the diverging route-
case 3

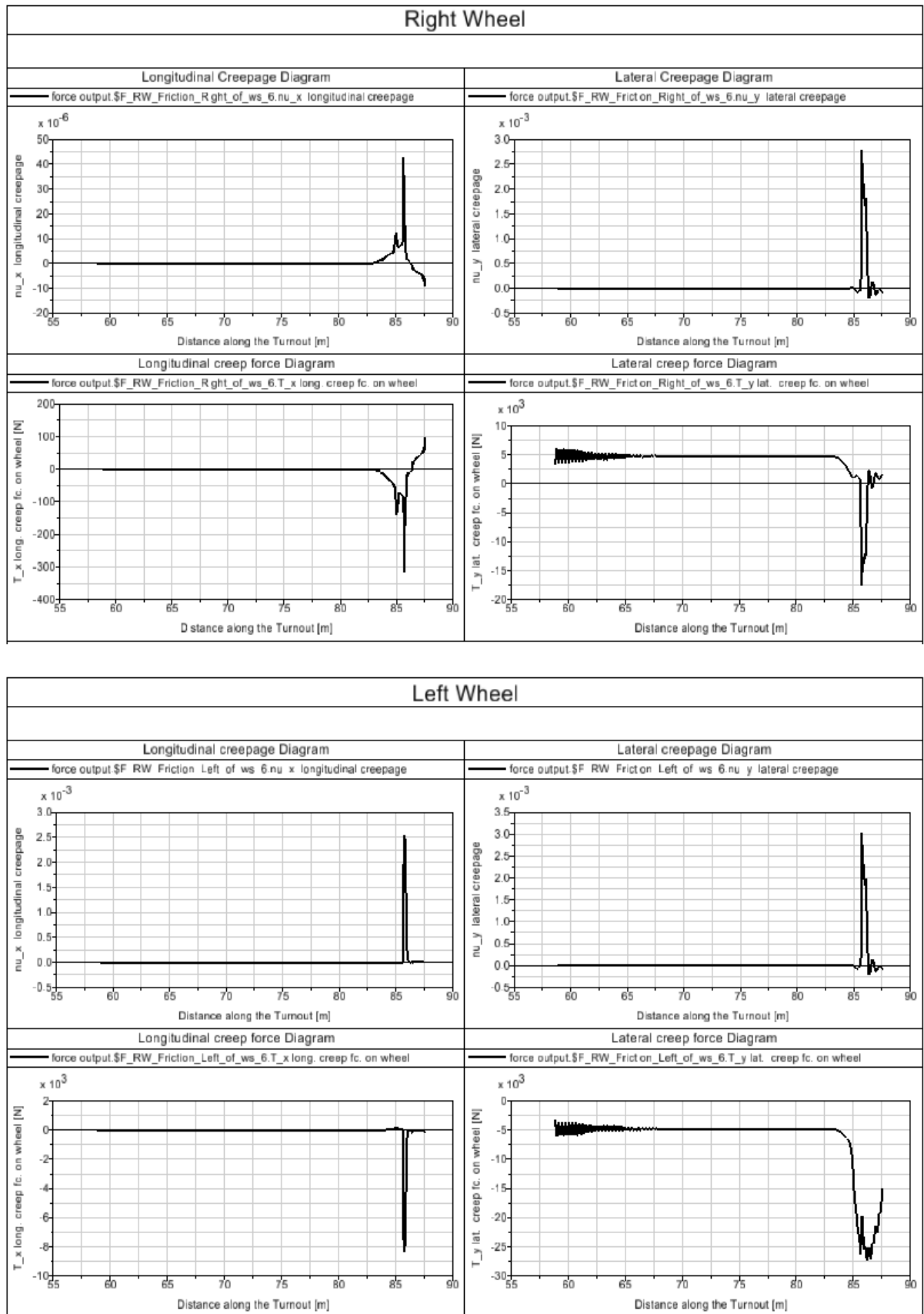


Figure B-25 Creepage and creep force response due to wheel-set 6 in the diverging route-
case 3

Table B-7: Damage calculation in the diverging route on the right rail- Case 3

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	Ty	RCF Damage	Remark
1	58-64	61	4.11E-03	3.83E-03	1.72E+04	2.22E+04	155.65094	1.75977E-06	RCF Damage
	64-70	67	8.00E-06	6.30E-05	9.29E+01	2.06E+03	0.1307752	0	No Damage
	70-76	73	1.05E-03	9.63E-03	1.88E+03	1.03E+04	100.9233	6.73506E-06	RCF Damage
	76-82	79	3.50E-03	1.04E-02	4.97E+03	2.62E+04	289.7954	-1.04353E-05	Wear Damage
	82-88	85	3.66E-04	5.00E-03	1.90E+03	1.62E+04	81.521864	8.49885E-06	RCF Damage
2	58-64	61	1.95E-03	3.43E-03	1.11E+04	1.95E+04	88.356662	7.8775E-06	RCF Damage
	64-70	67	7.00E-06	1.12E+02	9.20E+01	2.31E+03	258272.001	-0.023463598	Wear Damage
	70-76	73	9.66E-04	2.10E-02	4.02E+03	2.26E+04	477.451584	-2.74951E-05	Wear Damage
	76-82	79	1.49E-03	5.95E-03	5.56E+03	1.44E+04	93.6405307	7.39714E-06	RCF Damage
	82-88	85	1.62E-03	2.29E-02	1.05E+03	9.00E+03	207.71829	-2.97367E-06	Wear Damage
3	58-64	61	0.00E+00	0.00E+00	5.46E+03	5.08E+03	0	0	No Damage
	64-70	67	7.00E-06	2.00E-06	6.20E+01	5.21E+03	0.01085092	0	No Damage
	70-76	73	3.80E-05	2.45E-03	2.28E+02	1.76E+04	43.0097844	5.60196E-06	RCF Damage
	76-82	79	9.00E-06	2.26E+02	9.49E+01	2.18E+03	493132.001	-0.04481472	Wear Damage
	82-88	85	7.80E-03	4.46E-02	9.15E+03	3.11E+04	1458.68423	-0.000116699	Wear Damage
4	58-64	61	0.00E+00	0.00E+00	1.28E-01	5.78E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	3.90E-02	5.21E+03	0	0	No Damage
	70-76	73	7.00E-05	2.72E+03	5.89E+02	1.70E+04	46305976.5	-4.209660417	Wear Damage
	76-82	79	8.00E-06	2.35E+02	9.91E+01	2.22E+03	522221.701	-0.047459265	Wear Damage
	82-88	85	8.00E-05	1.61E+03	3.57E+02	4.03E+03	6475673.31	-0.58868755	Wear Damage
5	58-64	61	0.00E+00	0.00E+00	1.28E-01	5.78E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	2.95E+02	5.26E+03	0	0	No Damage
	70-76	73	0.00E+00	0.00E+00	1.02E-04	4.77E+03	0	0	No Damage
	76-82	79	2.00E-06	1.00E-06	1.67E+01	4.77E+03	0.00479942	0	No Damage
	82-88	85	6.50E-05	2.70E+03	5.00E+02	1.69E+04	45562500	-4.142070968	Wear Damage
6	58-64	61	0.00E+00	0.00E+00	1.28E-01	5.78E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	4.11E+02	5.21E+03	0	0	No Damage
	70-76	73	0.00E+00	0.00E+00	7.00E-06	4.77E+03	0	0	No Damage
	76-82	79	0.00E+00	0.00E+00	1.90E-02	4.77E+03	0	0	No Damage
	82-88	85	0.2	0.5	3.16E+02	1.65E+04	8307.354	-0.000739312	Wear Damage

Location on the Route	61	67	73	79	85
Cummulative Damage	9.63727E-06	-0.0234636	-4.20967558	-0.092277023	-4.731609004
Cummulative Damage due to RCF only	9.63727E-06	0	6.73506E-06	7.39714E-06	8.49885E-06
Cummulative Damage due toWear only	0	-0.0234636	-4.20968791	-0.09228442	-4.731617503

Table B-8: Damage calculation in the diverging route on the left rail- Case 3

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	T _y	RCF Damage	Remark
1	58-64	61	6.46E-03	4.39E-03	1.71E+04	3.26E+04	253.58798	-7.14368E-06	Wear Damage
	64-70	67	7.30E-05	7.40E-05	6.16E+02	2.71E+04	2.0533651	0	No Damage
	70-76	73	9.00E-03	2.48E-02	8.70E+03	8.21E+04	2112.823	-0.000176167	Wear Damage
	76-82	79	7.74E-03	1.05E-02	9.39E+03	6.69E+04	772.51039	-5.43189E-05	Wear Damage
	82-88	85	8.37E-04	7.12E-03	2.44E+03	6.41E+04	458.22904	-2.57476E-05	Wear Damage
2	58-64	61	4.88E-03	3.99E+03	1.11E+04	3.20E+04	127701770	-11.60935199	Wear Damage
	64-70	67	4.70E-05	1.32E-04	5.63E+02	2.76E+04	3.6654229	0	No Damage
	70-76	73	3.21E+03	1.49E+04	1.31E+04	8.82E+04	1.353E+09	-123.0410272	Wear Damage
	76-82	79	1.40E-02	4.62E+03	1.63E+04	6.55E+04	302131276	-27.46673835	Wear Damage
	82-88	85	1.28E+04	2.50E-02	3.98E+03	8.36E+04	51058433	-4.641706241	Wear Damage
3	58-64	61	0.00E+00	0.00E+00	5.17E-03	5.79E+03	0	0	No Damage
	64-70	67	7.00E-06	2.00E-06	6.20E-05	5.26E+03	0.010522	0	No Damage
	70-76	73	2.45E-03	3.06E-03	8.06E+03	2.73E+04	103.27132	6.5216E-06	RCF Damage
	76-82	79	5.00E-05	2.47E-04	2.72E+02	2.66E+04	6.59123	0	No Damage
	82-88	85	3.40E-02	3.40E-02	5.25E+03	1.20E+05	4251.7	-3.71E-04	Wear Damage
4	58-64	61	0.00E+00	0.00E+00	1.28E-01	5.93E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	4.14E-02	5.26E+03	0	0	No Damage
	70-76	73	2.50E-03	2.97E-03	8.09E+03	2.67E+04	99.319753	6.88084E-06	RCF Damage
	76-82	79	5.10E-05	2.56E-04	2.81E+02	2.66E+04	6.8345857	0	No Damage
	82-88	85	2.50E-03	1.48E-03	7.00E-04	4.27E+04	63.090057	9.61801E-06	RCF Damage
5	58-64	61	0.00E+00	0.00E+00	1.28E-01	6.13E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	2.97E-04	5.21E+03	0	0	No Damage
	70-76	73	0.00E+00	0.00E+00	1.02E-04	4.77E+03	0	0	No Damage
	76-82	79	2.00E-06	1.00E-06	1.46E+01	4.77E+03	0.0047967	0	No Damage
	82-88	85	2.56E-03	3.00E-03	7.95E+03	2.72E+04	102.01521	6.6358E-06	RCF Damage
6	58-64	61	0.00E+00	0.00E+00	1.28E-01	6.13E+03	0	0	No Damage
	64-70	67	0.00E+00	0.00E+00	4.12E-04	5.26E+03	0	0	No Damage
	70-76	73	0.00E+00	0.00E+00	7.00E-06	4.77E+03	0	0	No Damage
	76-82	79	0.00E+00	0.00E+00	1.97E-02	4.77E+03	0	0	No Damage
	82-88	85	2.50E-03	3.03E-03	8.34E+03	2.73E+04	103.65427	6.48679E-06	RCF Damage

Location on the Route	61	67	73	79	85
Cummulative Damage	-11.60935913	0	-123.04119	-27.46679266	-4.64207986
Cummulative Damage due to RCF only	0	0	0	0	2.27406E-05
Cummulative Damage due to Wear only	-11.60935913	0	-123.041203	-27.46679266	-4.6421026

The mode of rail damage mechanism involved almost completely wear due to the first few wheel-sets then the following wheel-sets actually damages less and the form of damage involved due to this last few wheel-sets is RCF mode. The net cumulative damage is totally dominated by the wear damage in these particular parametric combinations.

E. Divergent route – scenario 4

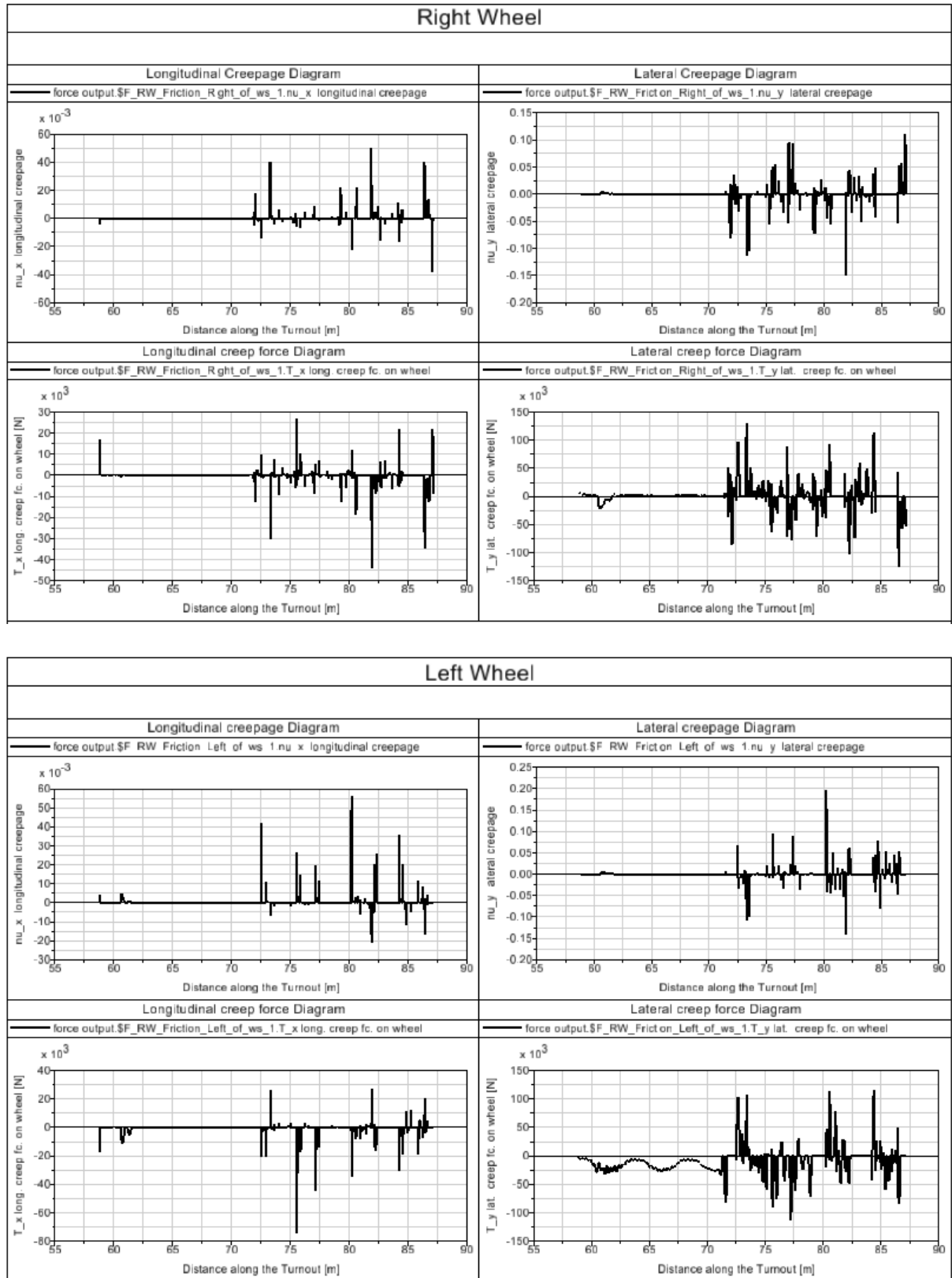


Figure B-26 Creepage and creep force response due to wheel-set 1 in the diverging route- case 4

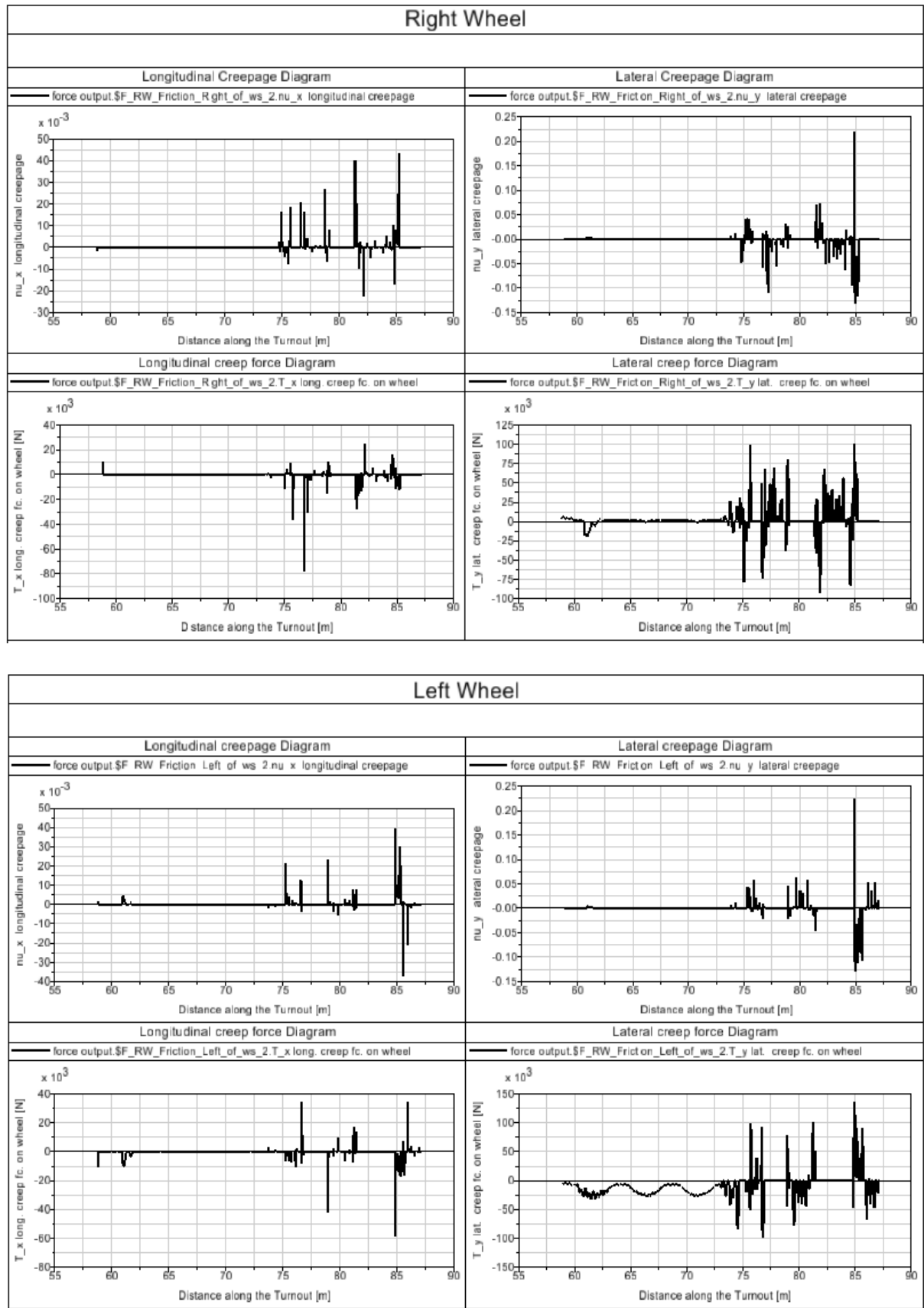


Figure B-27 Creepage and creep force response due to wheel-set 2 in the diverging route- case 4

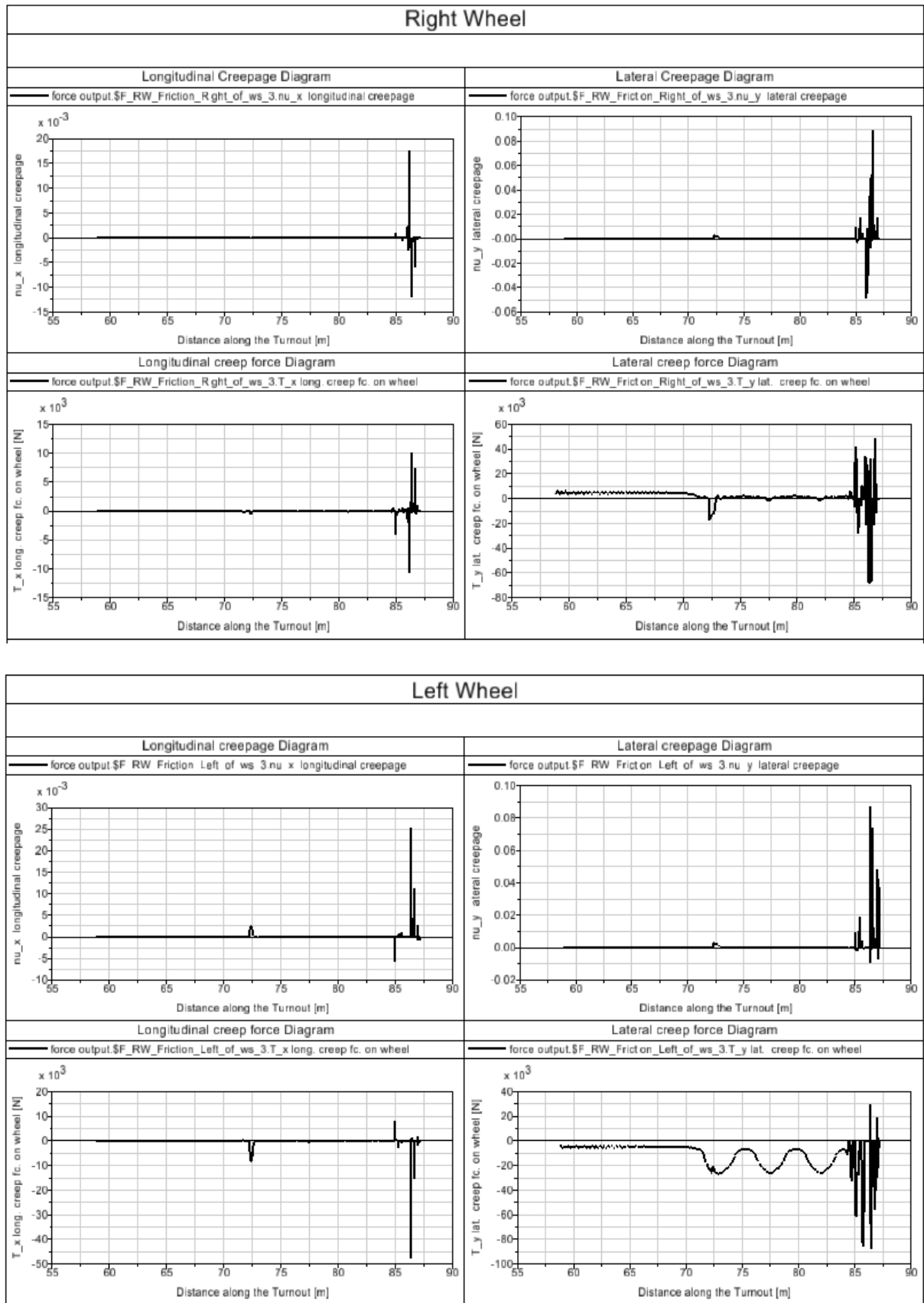


Figure B-28 Creepage and creep force response due to wheel-set 3 in the diverging route

-case 4

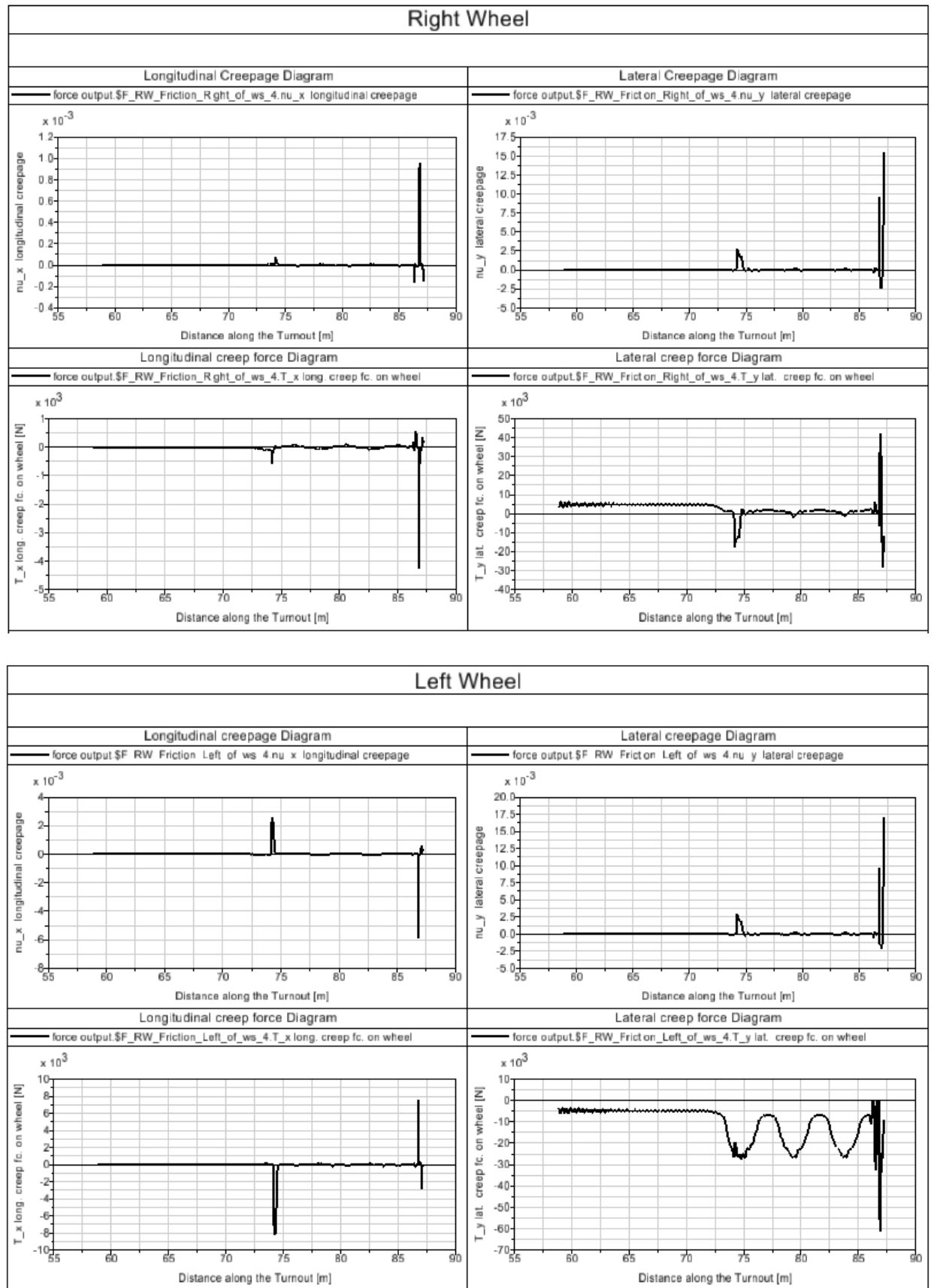


Figure B-29 Creepage and creep force response due to wheel-set 4 in the diverging route- case 4

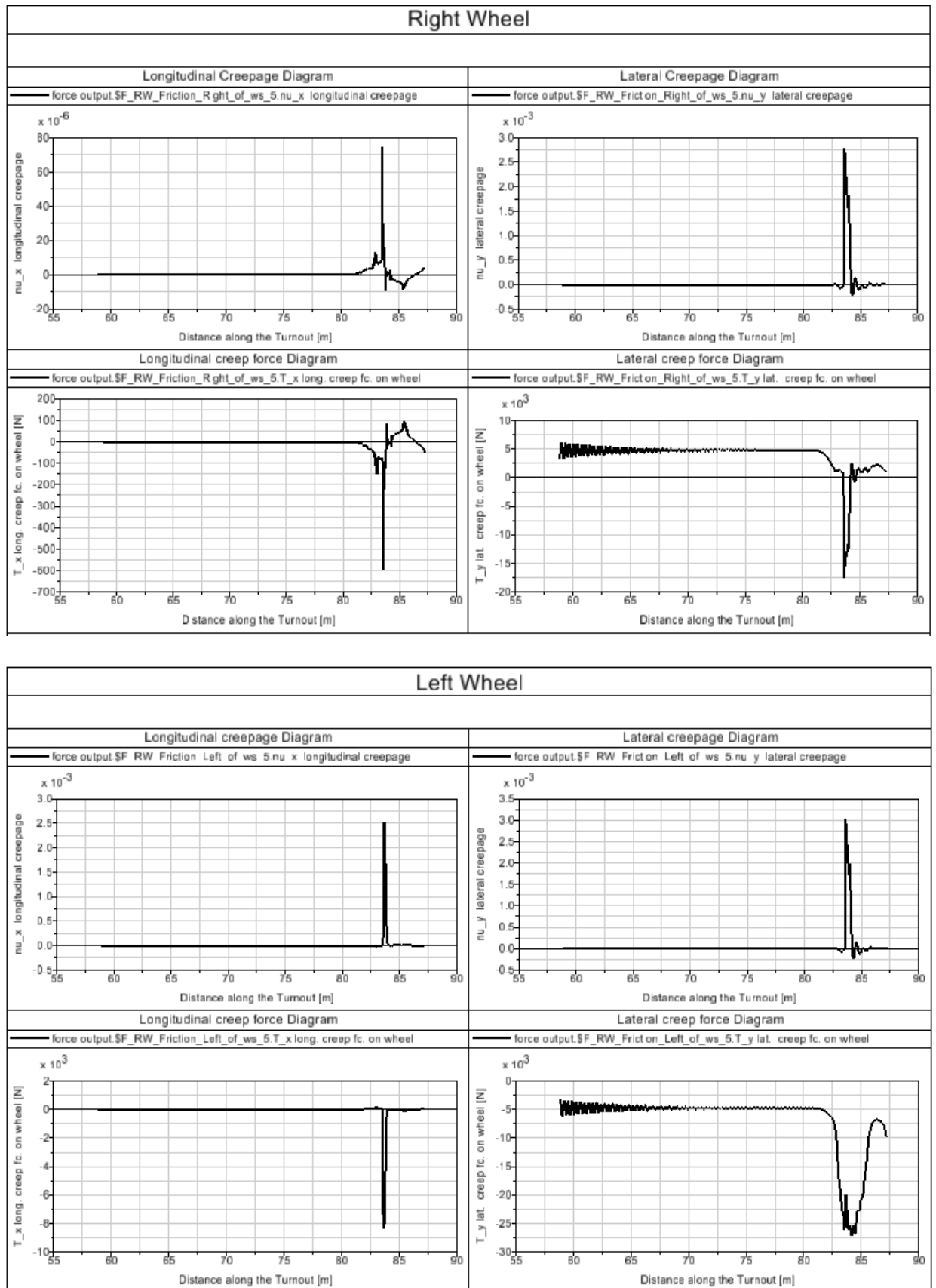


Figure B-30 Creepage and creep force response due to wheel-set 5 in the diverging route-
case 4

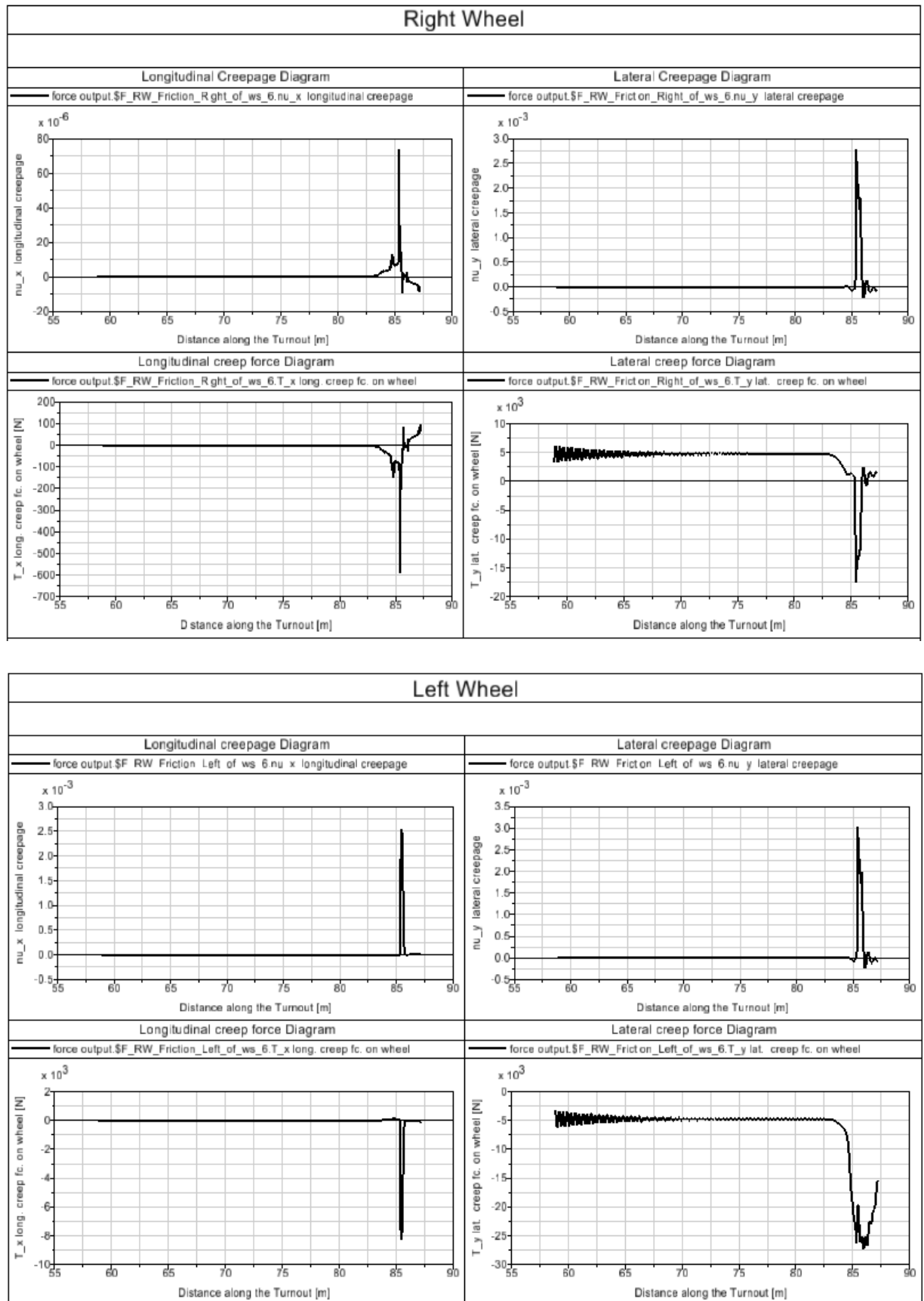


Figure B-31 Creepage and creep force response due to wheel-set 6 in the diverging route- case 4

Table B-9: Damage calculation in the diverging route on the right rail- Case 4

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	T _y	RCF Damage	Remark
1	70-74	72	4.00E-02	1.07E-01	3.01E+04	1.61E+05	18458.5588	-0.00166216	Wear Damage
	74-78	76	9.50E-03	9.71E-02	2.68E+04	8.91E+04	8908.77995	-0.00079399	Wear Damage
	78-82	80	5.00E-02	1.35E-01	4.42E+04	9.34E+04	14845.0331	-0.00133365	Wear Damage
	82-86	84	1.50E-02	4.47E-02	2.20E+04	1.14E+05	5416.4622	-0.0004765	Wear Damage
	86-90	88	4.00E-02	1.07E-01	3.49E+04	1.26E+05	14867.5605	-0.0013357	Wear Damage
2	70-74	72	3.10E-05	3.55E-04	1.82E+03	2.66E+04	9.49919117	0	No Damage
	74-78	76	2.00E-02	1.06E-01	7.83E+04	9.96E+04	12127.07	-0.00108656	Wear Damage
	78-82	80	4.00E-02	7.12E-02	2.80E+04	9.13E+04	7616.38371	-0.0006765	Wear Damage
	82-86	84	4.30E-02	2.16E-01	2.52E+04	100937	22866.2991	-0.00206287	Wear Damage
	86-90	88	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0	0	No Damage
3	70-74	72	1.50E-05	2.01E-03	2.15E+02	1.57E+04	31.5900055	3.318E-06	RCF Damage
	74-78	76	8.00E-06	2.34E-04	9.41E+01	2.20E+03	0.5157868	0	No Damage
	78-82	80	8.00E-06	8.40E-05	8.40E+01	2.17E+03	0.183288	0	No Damage
	82-86	84	8.33E-01	1.76E-02	4.12E+03	4.22E+04	4171.03091	-0.00036328	Wear Damage
	86-90	88	1.75E-02	8.88E-02	1.05E+04	6.79E+04	6207.96377	-0.00054846	Wear Damage
4	70-74	72	1.20E-05	7.40E-05	1.22E+02	4.98E+03	0.3699434	0	No Damage
	74-78	76	9.00E-05	2.37E-03	5.89E+02	1.71E+04	40.6968048	5.13936E-06	RCF Damage
	78-82	80	9.00E-06	2.30E-04	7.30E+01	2.19E+03	0.504794	0	No Damage
	82-86	84	8.00E-06	1.98E-04	8.06E+01	2.28E+03	0.452263	0	No Damage
	86-90	88	9.60E-04	1.55E-02	4.30E+03	4.23E+04	657.97203	-4.3906E-05	Wear Damage
5	70-74	72	0.00E+00	0.00E+00	7.00E-06	4.98E+03	0	0	No Damage
	74-78	76	0.00E+00	0.00E+00	1.42E-03	4.87E+03	0	0	No Damage
	78-82	80	2.00E-06	1.00E-06	2.48E+01	4.82E+03	0.00486666	0	No Damage
	82-86	84	7.40E-06	2.75E-03	6.00E+02	1.69E+04	46.44919	6.28984E-06	RCF Damage
	86-90	88	5.00E-06	1.90E-05	9.85E+01	2.22E+03	0.0425986	0	No Damage
6	70-74	72	0.00E+00	0.00E+00	1.60E-05	4.98E+03	0	0	No Damage
	74-78	76	0.00E+00	0.00E+00	1.01E-04	4.87E+03	0	0	No Damage
	78-82	80	0.00E+00	0.00E+00	2.49E-02	4.82E+03	0	0	No Damage
	82-86	84	7.20E-05	2.76E-03	5.73E+02	1.65E+04	45.4652086	6.09304E-06	RCF Damage
	86-90	88	9.00E-06	1.09E-04	9.60E+01	2.24E+03	0.24524191	0	No Damage

Location on the Route	72	76	80	84	88
Cummulative Damage	-0.00166	-0.00187541	-0.00201015	-0.002890261	-0.001928062
Cummulative Damage due to RCF only	3.32E-06	5.13936E-06	3.318E-06	1.23829E-05	0
Cummulative Damage due to Wear only	-0.00166	-0.001880549	-0.00201015	-0.002902644	-0.001928062

Table B-10: Damage calculation in the diverging route on the left rail- Case 4

Wheelset	Location bin	Location	Creepage-x (Longitudinal)	Creepage-y (Transverse)	Creep Force- Longitudinal	Creep Force- Transverse	Ty	RCF Damage	Remark
1	70-74	72	4.21E-02	1.08E-01	2.61E+04	1.08E+05	12757.91998	-0.00114391	Wear Damage
	74-78	76	2.65E-02	9.51E-02	7.44E+04	1.11E+05	12560.24102	-0.00112594	Wear Damage
	78-82	80	5.62E-02	1.97E-01	3.49E+04	1.14E+05	24346.2642	-0.00219741	Wear Damage
	82-86	84	3.58E-02	8.12E-02	3.07E+04	1.14E+05	10341.18407	-0.00092421	Wear Damage
	86-90	88	1.58E-02	3.70E-02	2.07E+04	8.25E+04	3380.291674	-0.00029139	Wear Damage
2	70-74	72	3.50E-05	7.97E-04	3.74E+02	2.46E+04	19.601756	9.20351E-07	RCF Damage
	74-78	76	2.14E-02	5.43E-02	3.47E+04	9.95E+04	6140.252308	-0.0005423	Wear Damage
	78-82	80	2.40E-02	6.30E-02	4.20E+04	9.93E+04	7263.43019	-0.00064441	Wear Damage
	82-86	84	4.00E-02	2.20E-01	5.90E+04	1.37E+05	32464.79153	-0.00293546	Wear Damage
	86-90	88	1.09E-04	5.30E-02	1.78E+03	5.30E-02	0.19671788	0	No Damage
3	70-74	72	2.29E-03	2.64E-03	7.94E+03	2.65E+04	88.1428665	7.89693E-06	RCF Damage
	74-78	76	5.30E-05	2.54E-04	2.97E+02	2.63E+04	6.699243	0	No Damage
	78-82	80	3.80E-05	1.30E-04	2.15E+02	2.60E+04	3.3816077	0	No Damage
	82-86	84	5.50E-03	1.91E-02	8.06E+03	8.50E+04	1664.439198	-0.0001354	Wear Damage
	86-90	88	2.50E-02	8.75E-03	4.76E+04	8.76E+04	1956.527271	-0.00016196	Wear Damage
4	70-74	72	3.40E-05	7.90E-05	1.68E+02	2.26E+04	1.7949446	0	No Damage
	74-78	76	2.57E-03	2.33E-03	8.06E+03	2.77E+04	85.14687596	8.1693E-06	RCF Damage
	78-82	80	5.20E-05	2.50E-04	2.71E+02	2.65E+04	6.644092	0	No Damage
	82-86	84	4.00E-05	2.15E-04	2.20E+02	2.62E+04	5.63965	0	No Damage
	86-90	88	5.95E-03	1.71E-02	7.64E+03	6.08E+04	1085.08745	-8.2735E-05	Wear Damage
5	70-74	72	0.00E+00	0.00E+00	1.50E-05	4.98E+03	0	0	No Damage
	74-78	76	0.00E+00	0.00E+00	1.41E-03	4.87E+03	0	0	No Damage
	78-82	80	2.00E-06	1.00E-06	1.46E+01	5.62E+03	0.00565066	0	No Damage
	82-86	84	2.50E-03	3.00E-03	8.27E+03	2.70E+04	101.58394	6.675E-06	RCF Damage
	86-90	88	1.00E-05	1.70E-05	7.78E+01	9.89E+03	0.1689166	0	No Damage
6	70-74	72	0.00E+00	0.00E+00	2.20E-05	4.98E+03	0	0	No Damage
	74-78	76	0.00E+00	0.00E+00	1.01E-04	4.87E+03	0	0	No Damage
	78-82	80	0.00E+00	0.00E+00	2.86E-02	4.82E+03	0	0	No Damage
	82-86	84	2.50E-03	2.97E-03	8.31E+03	2.64E+04	99.11451645	6.8995E-06	RCF Damage
	86-90	88	2.20E-05	2.25E-04	1.58E+02	2.72E+04	6.11945312	0	No Damage

Location on the Route	72	76	80	84	88
Cummulative Damage	-0.001135	-0.001660073	-0.00284182	-0.003981501	-0.000536086
Cummulative Damage due to RCF only	8.817E-06	8.1693E-06	7.89693E-06	1.35745E-05	0
Cummulative Damage due to Wear only	-0.001144	-0.001668242	-0.00284182	-0.003995075	-0.000536086

As it can be seen from the analysis outcomes, the first few wheel-sets cause much of the damage through wear mechanism than the RCF - just fatigue form of damage; however the following wheel-sets cause less damage as well as the mode of damage shifts into that of RCF damage.

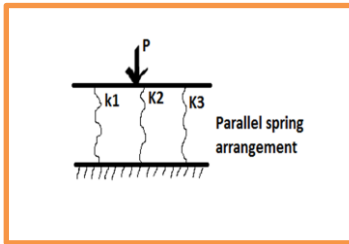
APPENDIX C

Equivalent stiffness and damping

In MBS modeling and simulation, we could encounter serial, parallel or combination of serial and parallel arrangement of springs and damper; in such a case it is possible to simply compute a single stiffness and damping parameter and this is called equivalent stiffness and/or damping. Equivalent stiffness and/or damping have same effect on the dynamic system under study. In the next paragraphs their derivation is explained:

i. Equivalent stiffness

Parallel arrangement



Δx – displacement and it is constant

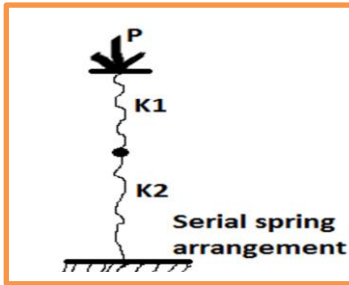
$$\Delta x_1 = \Delta x_2 = \dots = \Delta x$$

$$P = K_1 \Delta x_1 + K_1 \Delta x_2 + \dots = K_e \Delta x$$

$$\therefore K_e = K_1 + K_2 + \dots$$

Serial arrangement

P – Applied force and it is constant in each spring



$$P = K_1 \Delta x_1 = K_1 \Delta x_2 = \dots = K_e \Delta x$$

$$\Delta x_1 + \Delta x_2 + \dots = \Delta x$$

$$\frac{P}{K_e} = \frac{P}{K_1} + \frac{P}{K_2} + \dots$$

$$\therefore K_e = \frac{1}{\frac{1}{K_1} + \frac{1}{K_2} + \dots}$$

ii. Equivalent damping

Doing same thing and differentiating the displacement we get similar form equivalent damping for parallel as well as serial arrangements:

Parallel arrangement

$$\Delta \dot{x}_1 = \Delta \dot{x}_2 = \dots = \Delta \dot{x}$$

$$P = C_1 \Delta \dot{x}_1 + C_1 \Delta \dot{x}_2 + \dots = C_e \Delta \dot{x}$$

$$\therefore C_e = C_1 + C_2 + \dots$$

Serial arrangement

P – Applied force and it is constant in each damper (dashpot)

$$P = C_1 \Delta x_1 = C_2 \Delta x_2 = \dots = C_e \Delta x$$

$(\Delta x_1 + \Delta x_2 + \dots = \Delta x)$ Differentiating displacement to get velocity information

$$\Delta \dot{x}_1 + \Delta \dot{x}_2 + \dots = \Delta \dot{x}$$

$$\rightarrow \frac{P}{C_e} = \frac{P}{C_1} + \frac{P}{C_2} + \dots$$

$$\therefore C_e = \frac{1}{\frac{1}{C_1} + \frac{1}{C_2} + \dots}$$