IDENTIFICATION AND DEVELOPMENT OF A MODEL OF RAILWAY TRACK DYNAMIC BEHAVIOUR

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Abstract

The research presented in this thesis has identified and developed a sophisticated computer model for the analysis of railway track dynamic behaviour to be used by the Rail Cooperative Research Centre for Railway Engineering and Technologies (Rail CRC) in Australia.

To be competitive railway track owners need to extract as much performance as possible from their asset without serviceability or catastrophic failure. Railway track designers therefore need to develop more knowledge of the static and dynamic loadings that track may be subjected to in its lifetime. This would be best undertaken using computer modelling capable of quantifying the effects of train speed, traffic mix, wheel impact loading and distribution of vehicle loads into the track.

A comprehensive set of criteria for the selection of a model of track dynamic behaviour was developed. An international review of state-of-the-art models which represented the railway track structure under the loading of a passing train was undertaken. The models’ capabilities were assessed and a number of potential models identified.

A benchmark test was initiated to compare current models available throughout the international railway research community. This unique benchmark test engaged six researchers to compare their railway track models using a set of theoretical vehicle and track data. The benchmark results showed that significantly different results may be obtained by models, depending on the assumptions of the user in representing a particular track scenario. Differing complexities and modelling methods, the number of different input parameters required and the representation of the irregularities in the wheel and rail all have effect on the results produced.

As a result of these initiatives, the DARTS (Dynamic Analysis of Rail Track Structures) computer model was chosen for use by the Rail CRC. A user-friendly interface was created for DARTS by the writer, which was readily interpretable by railway design engineers. At the time of writing, DARTS was found to be suitable for detailed investigations planned by the Rail CRC for future research and was provided for use through an Intellectual Property agreement with its author.
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<td>Rail CRC</td>
<td>Cooperative Research Centre for Railway Engineering and Technologies</td>
</tr>
<tr>
<td>QR</td>
<td>Queensland Rail</td>
</tr>
<tr>
<td>QUT</td>
<td>Queensland University of Technology</td>
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<tr>
<td>UoW</td>
<td>University of Wollongong</td>
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<td>CQU</td>
<td>Central Queensland University</td>
</tr>
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<td>CRE</td>
<td>Centre for Railway Engineering</td>
</tr>
<tr>
<td>RIC</td>
<td>Rail Infrastructure Corporation</td>
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<tr>
<td>ARTC</td>
<td>Australian Rail Track Corporation</td>
</tr>
<tr>
<td>ARA</td>
<td>Australasian Railways Association Inc.</td>
</tr>
<tr>
<td>ROA</td>
<td>Railways of Australia</td>
</tr>
<tr>
<td>RSSB</td>
<td>British, Rail Safety and Standards Board</td>
</tr>
<tr>
<td>RTRI</td>
<td>Railway Technical Research Institute</td>
</tr>
<tr>
<td>DFG</td>
<td>German Research Council</td>
</tr>
<tr>
<td>CHARMEC</td>
<td>CHAlmers Railway MEChanics</td>
</tr>
<tr>
<td>AAR</td>
<td>American Association of Railroads</td>
</tr>
<tr>
<td>TTCI</td>
<td>Transportation Technology Center Incorporated</td>
</tr>
<tr>
<td>DARTS</td>
<td>Dynamic Analysis of Rail Track Structures</td>
</tr>
<tr>
<td>DIFF</td>
<td>Vehicle-Track Dynamic Analysis Model</td>
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<tr>
<td>NUCARSTM</td>
<td>New and Untried Car Analytic Regime Simulation</td>
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<td>SUBTTI</td>
<td>Subgrade Train-Track Interaction</td>
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<tr>
<td>TRACK</td>
<td>Track Design Software</td>
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<tr>
<td>VICT</td>
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<td>RAIL</td>
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<td>3D WTSD</td>
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<td>BOEF</td>
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Statement of Originality

The work contained in this thesis has not been previously submitted for a degree or diploma at any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made.

Signed: .................................  David Martyn Steffens

Date: .................................
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Accepted Papers

CHAPTER 1

Introduction

1.1 Background of the Research

Ever increasing axle loads and train speeds are pressing track owners to extract as much performance as possible from their asset without serviceability or catastrophic failure. Unfortunately, there is insufficient knowledge of the static and dynamic loadings that track may be subjected to, in its lifetime. There is also widespread suspicion that track components have reserves of strength that are untapped, especially concrete sleepers. Addressing these issues has the potential for substantial savings for track owners.

The Australian Standards CE/2 Committee requested the Australasian Railway Association Inc (ARA) in 1996 to initiate a review of Australian Standard AS1085.14 ‘Permanent way materials: Prestressed concrete sleepers’ (Standards Australia, 2003) for load distribution and dynamic impact allowance. The committee was responding to a widespread belief that the state of knowledge of track forces and their transmission to concrete sleepers, ballast and formation was unsatisfactory. A Brief prepared by the ARA noted the need for an approach, which would clarify loads and their distribution in track for axle loads up to 30 tonnes and speeds up to 160km/h, and be applicable to Australian heavy haul, freight and passenger operations.
In response to this request, a comprehensive review was conducted by Queensland University of Technology (QUT) (Murray & Cai, 1998) covering a wide range of literature relating to concrete sleepers from the major research and commercial rail organisations in USA, Canada, UK, France, Sweden, Germany, Japan, China, and Australia. The report reviewed theoretical models of static and dynamic analysis of track behaviour in detail. Laboratory and field tests of impact and load distribution together with the common design codes and manuals from a number of countries were also examined.

The review recommended that with further research a more cost effective appreciation of track performance could be realised. The research should include a more specific definition of the loading environment and an examination of: the relationships between the major wheel and vehicle parameters; the attenuation effects of resilient rail pads; and the flexural behaviour of the sleepers to impact loading.

It would be impractical and costly to accumulate a comprehensive set of measurements of track forces under the varied mix of traffic and track in Australia. Consequently, these force studies would be best undertaken by a sophisticated track analysis model. The model could be used to quantify the effects of train speed, traffic mix, and frequency spectrum of wheel loading, upon impact and distribution of vehicle loads into the track.

This research forms one part of a project in a range of railway research projects supported by the Cooperative Research Centre for Railway Engineering and Technologies (Rail CRC). The project is titled ‘Dynamic Analysis of Track and the Assessment of its Capacity with Particular Reference to Concrete Sleepers’. Queensland University of Technology (QUT) and the University of Wollongong (UoW) are working in collaboration during this project with the aim of developing in part a new limit state approach for the Australian Standard AS1085.14 ‘Permanent way materials: Prestressed concrete sleepers’ (Standards Australia, 2003).
1.2 Rail CRC Project Aim

The broad aim of this Rail CRC project is to help rail track owners make more cost-effective use of the asset through improved knowledge of track behaviour under static and dynamic loading and in particular through more realistic processes of analysis and design of concrete sleepers.

The project aims to yield the following:

1. A usable software package for the rigorous analysis of the dynamic behaviour of railway track in Australia;
2. A greatly improved knowledge of track and sleeper behaviour;
3. A more realistic design approach for Australian Standard AS1085.14; and
4. Significant savings flowing from increased confidence in the capacity of track and sleepers to carry increased traffic demands.

The research presented in this thesis specifically focuses on point one of the above project aims. The research aim is to provide an appropriate railway track dynamic analysis tool for the Australian railway industry and research community so that rigorous analysis of the railway track structure can be undertaken efficiently.

1.3 Scope of this Research

The scope of the research presented in this thesis includes:

1. Examination of the Australian railway system and present design procedures;
2. Identification of models of railway track dynamic behaviour that are suitable to assess Australian railway conditions;
3. Selection of an appropriate model of railway track dynamic behaviour; and
4. Development of the model into usable software available for rigorous analysis of railway track in Australia.
1.4 Objectives of this Research

In the current economic environment, it is important for railway organisations to be as competitive as possible. The major task for the railway track engineer is often that of analysis, determining the economic effect or allowable limit to increasing axle loads and vehicle speeds on existing tracks. By analysing the railway track structure using realistic track simulation models, more informed economic and design decisions could be made. The research presented in this thesis aims to make the calculation of dynamic forces more accessible to railway track engineers by providing a comprehensive analysis tool.

A comprehensive analysis tool will need to:

- Be validated against actual track data;
- Be readily accessible to Rail CRC partners;
- Have user-friendly, readily interpretable interfaces;
- Be cost effective in access and operation; and
- Be suitable for detailed investigations planned for future research.

The outcome of this research will be the development of a user-friendly model of track dynamic behaviour that is capable of establishing the full breadth of likely and extreme forces for the range of track and rollingstock characteristics found in Australia.

1.5 Structure of the Research

The research described in this thesis is separated into four main parts:

- Understanding the railway system, design and standards;
- Identification of railway dynamic analysis models;
- A benchmark comparison of railway dynamic analysis models; and
- Selection and development of a railway dynamic analysis model into a useful engineering tool.
Chapter 1 introduces the research and discusses how it fits into the range of research being undertaken through a Cooperative Research Centre for Australian Railways. It presents the broad Rail CRC project aim and then defines a specific scope for the research presented in this thesis. The objectives of the present research are detailed and the structure of the thesis document is discussed.

Chapter 2 provides a definition of the railway system and details the terminologies commonly used to describe the three main railway subsystems: the vehicle, the wheel/rail interface and the track. Railway track design is discussed and the main forces experienced by the track structure are examined. The limits placed by design standards on dynamic forces experienced by the railway track are detailed. Procedures used for railway track design are discussed and the current design tools available to the Australian railway industry are presented.

Chapter 3 details the relevant terminology and the methods typically used for modelling railway track dynamics. A discussion is provided on how the various components of the railway system are represented in analysis models. Criteria for the selection of a railway track dynamic analysis model are presented and models from around the world are reviewed with a focus on models relevant for use by the Australian railway track design engineer. The findings of a number of benchmarks for models of the railway system are also presented.

A new benchmark test of models of railway track dynamic behaviour undertaken by the writer is presented in Chapter 4. The instructions for the benchmark test are detailed and a comparison of the participating dynamic analysis models is presented. A discussion is provided on how the benchmark instructions were interpreted by the participants. The results are evaluated with regard to the selection and validation of an appropriate model for the Australian railway community.

Chapter 5 explores the benchmark test further by examining the models’ ability to analyse non-concrete sleepered railway track. The benchmark results are then also compared to traditional empirical methods for railway track design. Finally an appropriate model of track dynamic behaviour is selected.
Chapter 6 details the development of a user-friendly interface for the selected model of track dynamic behaviour. Further developments required before release to the Australian railway industry and research community are discussed.

Chapter 7 provides the main conclusion and recommendations presented throughout this thesis. The future direction of the Rail CRC project which this research forms part, is discussed.
CHAPTER 2

Review of Railway Terminology, Track Design & Standards

2.1 Introduction

Railways are complex systems having many varied characteristics. Railway design is typically separated into the design of trains or rollingstock and the design of the supporting track structure. This research is focussed on the design and analysis of the supporting track structure; however it is important to understand the system as a whole. This chapter will discuss the terminology used to describe the railway system, the present procedure for design of the track structure in Australia and the software available for use by track design engineers. A discussion of the standards placed on the railway track design is also provided so that an understanding of the limits placed on forces within the track structure can be established. It is important to understand the limitations of current standards and procedures in order to appreciate the need for more rigorous analysis of track behaviour, especially under dynamic actions. There is no single holistic review of all these matters able to be found in the literature.

2.2 Railway System Terminology

The word ‘system’ used in this thesis refers to several interacting parts, each of which performs a specific role. The railway system can be separated into three subsystems including:
• the vehicle;
• the wheel and rail interface; and
• and the track structure.

An understanding of the terminology used in these subsystems is important as these terms will be used throughout this thesis.

### 2.2.1 Vehicle Subsystem

Trains are typically referred to as rollingstock and consist of two types of vehicle: a locomotive or power-car that enables the train to operate; and wagons that carry goods of some kind. Modern locomotives are usually powered by electricity (in-rail or overhead power), or diesel (mechanical, hydraulic or electric).

For the purposes of investigating rail track forces, vehicles can be defined by the number of axles they have. Locomotives typically have triple axle bogies; wagons may have triple, double or single axle bogies. This research has concentrated on wagons with two double axle bogies. A typical freight wagon is shown in Figure 2.1 and consists of a car body and bogies.

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This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.
The Car Body is best defined as a container for the goods, whether they are human or material. Vehicle motions are defined in terms of the vehicle and track’s vertical, lateral and longitudinal axes. The basic modes of oscillation are:

- Pitch; forward and backward rotation about the transverse axis;
- Bounce; up and down movement along the vertical axis;
- Yaw; rotation about a central vertical axis;
- Lateral; movement side to side along the lateral axis; and
- Roll; tipping side to side about the longitudinal axis.

The Bogie is the part that guides the train on the rails and provides stable operation. There are two main types of bogies, one with and the other without primary suspension. Suspension is typically made of coil springs that minimise the impact and enhance the stability of wagon operations. The bogie incorporates various springs and dampers to cushion the ride. The terminology used to describe the components of a typical bogie is shown in Figure 2.2.

![Figure 2.2 Typical bogie components](image)

The most common type of bogie used to transport freight is termed a Three-piece Bogie because its frame has three basic parts: a bolster and two side frames as shown in Figure 2.3. Three-piece bogies are the cheapest to purchase and most economical to maintain (Skerman, 2001). However, they provide a low level of lateral stability and poor ride quality. This is mainly due to having only secondary suspension creating a higher unsprung mass. The Unsprung Mass is mass of the components which are not dynamically isolated from the track by suspension elements.
Figure 2.3 shows a typical three piece bogie. Load from the vehicle is applied through the **Centre Plate** or bowl, where longitudinal or lateral relative movement is restrained. The load is then transferred to the **Bolster** which spans between the two side frames and rest on secondary suspension. The **Secondary Suspension** which is usually a group of resilient coiled springs then takes the load and transfers it to the **Side Frame**. In three piece bogies, the side frames sit directly on top of the axle boxes or package bearing adaptors and tie the two wheelsets together longitudinally.

![Figure 2.3 Three piece bogie](image)

Bogies used for passenger trains usually have **Primary Suspension** such as a spring, airbag or rubber positioned between the axles of the wheelsets and the side frames. By including primary suspension the lateral stability of the wagons and the ride quality can be significantly improved. This is due to the lower unsprung mass as the size of the unsprung mass directly relates the amount of force transferred into the car body and track from wheel or track irregularities.

The **Wheelset** is the assembly consisting of two wheels and bearings on an axle. The two wheelsets at each end of the vehicle are fitted to the bogie which can yaw in order to negotiate curves.
The **Wheel** is the contact element connecting the vehicle to the track. Modern wheels are conical rather than cylindrical in shape. This promotes a centring effect that helps the wheelset along straight or tangent track and slight lateral displacements of the track (Esveld, 2001). The conical shape also assists steering through curves due to the rolling radius difference generated by the lateral displacement of the wheelset in curved track.

### 2.2.2 Wheel-Rail Interface Subsystem

The connection of the vehicle and track through the wheel-rail interface is critical for the successful operation of trains. If the connection is interrupted through breakdown of either system, a derailment could occur which would have significant consequences. Figure 2.4 (a) & (b) shows how the entire train load is distributed down into the track system through a very small contact area on each wheel.

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.
Hertz theory (1887) can explain what happens in the vertical or lateral direction: the elastic deformation of the steel of the wheel and rail creates an elliptic contact area. The dimensions of the contact ellipse are determined by the normal force on the contact area and the elasticity and hardness of the wheel and rail running surfaces, while the ratio of the ellipse axes depends on the main curvatures of the wheel and rail profiles. The shape of the contact ellipse changes in relation to the location of the wheel-rail contact area across the railhead. Inside the contact area, a pressure distribution develops which in cross section is shaped in the form of a semi-ellipse with the highest contact pressure usually occurring at the centre (Johnson, 1985).

2.2.3 Track Subsystem

The typical track structure used throughout Australia is ballasted track. Other types of track systems such as slab track are also used elsewhere; however this research will be focusing on the ballasted track structure (Figure 2.5).

The main components of ballasted track structures may be grouped into two categories: superstructure and substructure. The superstructure consists of the rails, fasteners and pads, and the sleepers. The substructure consists of the ballast, the subballast (capping layer) and the subgrade (formation).

![Figure 2.5 Typical ballasted track system](image)
**Rails** are the longitudinal steel members that directly guide the train wheels evenly and continuously. They must have sufficient stiffness to serve as beams that transfer the concentrated wheel loads to the spaced sleeper supports without excessive deflection between supports (Selig & Waters, 1994).

![Figure 2.6 Typical concrete sleeper fastening system](image)

**Fasteners** are typically required to retain the rails on the sleepers and to resist vertical, lateral, longitudinal and overturning moments of the rails. The force systems causing these movements are from the wheels and from temperature change in the rails.

**Rail Pads** or **Plates** are required between the rail seat and the sleeper surface to fulfil various functions. These include providing resilience for the rail-sleeper system, damping of wheel induced vibrations, and reduction of rail-sleeper contact attrition.

**Sleepers** are essentially beams that span across and tie together the two rails. They have several important functions including receiving the load from the rail and distributing it over the supporting ballast at an acceptable ballast pressure level, holding the fastening system to maintain proper track gauge, and restraining the lateral, longitudinal and vertical rail movement by anchorage of the superstructure in the ballast. In addition, sleepers provide a cant to the rails to help develop proper rail-wheel contact by matching the inclination of the conical wheel shape.
**Ballast** is the layer of crushed stone on which the sleepers rest. The ballast assists in track stability by distributing load from the sleepers uniformly over the subgrade. It anchors the track in place against lateral, vertical and longitudinal movement by way of irregular shaped ballast particles that interlock with each other. Any moisture introduced into the system can easily drain through the ballast away from the rails and sleepers. The course grained nature of ballast assists in track maintenance operations due to its easy manipulation. The rough interlocking particles also assist in absorbing shock from dynamic loads by having only a limited spring-like action (Hay, 1982).

**Subballast**, also known as the Capping Layer is usually a broadly graded material that assists in reducing the stress at the bottom of the ballast layer to a tolerable level for the top of the subgrade. The subballast is usually an impervious material that can prevent the inter penetration of the subgrade and ballast, thereby reducing migration of fine material into the ballast which effects drainage. This layer also acts as a surface to shed water away from the subgrade into drainage along the side of the track.

**Subgrade**, also known the Formation, offers the final support to the track structure. The subgrade bears and distributes the resultant load from the train vehicle through the track structure. The subgrade facilitates drainage and provides a smooth platform, at an established grade, for the track structure to rest upon.
2.3 Railway Track Design

The successful design of structural elements such as concrete railway track sleepers and track itself goes back to ancient times. For many centuries, buildings and other structures were designed using common sense, trial and error, and rules of proportion acquired through experience. Their effectiveness depended on the knowledge and skills of master craftsmen. Industrialization and the mass-production of iron and steel in the nineteenth century led to rapid changes in construction methods. This in turn provided a drive to replace the traditional trial-and-error approach for designing structures, which was slow to adapt to innovations by calculations based on scientific principles. The only scientific tools available at that time for designing structures were Newton's laws of motion and the theory of elasticity. As time went on, these scientific principles were developed into a unified, practical tool for structural calculations called allowable stress design (Allen, 1982).

In allowable stress design, the adequacy of a structural element is checked by calculating the elastic stresses in it due to the maximum expected loads, and comparing them with allowable stresses. The allowable stress is equal to the failure stress of the material divided by a safety factor. Safety factors were first determined by applying allowable stress design methods to successful structures existing at that time. The safety factors for new materials were estimated in comparison with those for traditional materials by taking into account the nature of failure for the new material and its uncertainty or variability. Allowable stress design formed the basis of structural codes and standards, including track design standards, for most of the twentieth Century.

The most common procedures used for designing railway track in Australia were presented by the Railways of Australia (ROA) in their Review of Track Design Procedure (Jeffs & Tew 1991). An allowable stress design flow chart for conventional ballasted track structure design (Prause et al 1974) is typically used by the track design engineer to undertake design of new railway track. This flow chart is show in Figure 2.7.
Figure 2.7 Track Design Procedure (after Prause et al 1974)
Hagaman (2001) notes that traditional track structure design has been empirical, with economic considerations frequently dictating component selection and size. The railway track in reality is a complex non-linear system, which is a characteristic that current design procedures typically ignore.

The traditional method for designing new railway track uses an ‘allowable stress’ design approach in determining the load applied to the track. This approach expresses the wheel load empirically as a function of the static wheel load with a dynamic factor to account for vehicle and track irregularities.

\[ P_D = \phi \cdot P_0 \]  

(2.1)

where:
- \( P_D \) = design wheel load (kN);
- \( P_0 \) = static wheel load (kN); and
- \( \phi \) = design dynamic increment (always > 1).

The following sections provide an examination of the typical forces experienced by the track structure, their usage in current design and analysis of the railway track. Sections 2.4 and 2.5 present a review of the current design standards, procedures and the tools used presently for rigorous analysis of the track structure.

### 2.3.1 Design Forces

Railway track is subjected to vertical, lateral and longitudinal forces which can be applied as static, dynamic or thermal forces. Lateral and longitudinal forces are mainly used for rail design as they give rise to significant rail stresses. These stresses may in-turn limit the allowable stress the rail can withstand from vertical loading. The present research focuses on vertical static and dynamic forces in the railway track structure when running trains at varying speeds and large axle loads.

A simplistic approach for railway track design would be that both vehicle and track are free from irregularities. However irregularities do exist and may take the form of
geometrical track irregularities (long wavelength) or discrete irregularities (short wavelength) in the wheel tread or rail running surface. These two classifications of track irregularities produce different magnitudes of forces due to the resonances they create within the track structure.

Vehicles moving on the track at speed exert certain forces on the track structure due to the behaviour of the vehicle body, bogie and other masses in response to geometrical irregularities in the track. These low frequency (below 20 Hz) forces are known as quasi-static or dynamic ride forces and are associated with vehicle movements. New railway track is typically designed with these forces in mind. With good maintenance practice track geometrical irregularities should be kept to a minimum.

Significant irregularities may occur during the life of the track structure and vehicle. These are usually discrete irregularities in the wheel tread or rail running surfaces and they create much higher frequency forces than quasi-static loads. These forces are known as dynamic wheel/rail forces and are much higher in magnitude than quasi-static (dynamic ride) forces.

### 2.3.2 Quasi-Static (Dynamic Ride) Forces

The Australian Standard for pre-stressed concrete sleepers (AS1085.14, 2003) defines the quasi-static load as the sum of the static load and the effect of the static load at speed. The standard also states that the load includes the effects of the geometrical roughness of the track on vehicle response and the effect of unbalanced superelevation (the effect of the train load not being distributed evenly over both rails).

Grassie (1994) comments that if both the wheel tread and rail surface are in good condition, the wheel/rail contact force would be similar to the static wheel load. Loads applied to the track from vehicle responses to track geometry and support conditions have frequencies of a few cycles per second (up to 20 Hz). This track
geometry includes design geometry (curvature and superelevation) and also incidental geometry such as track roughness. In Australia this load is termed quasi-static, however it would better be described by the term ‘dynamic ride force’ as the load is directly related to the speed of the vehicle and condition of the track. To reduce confusion however these loads will be referred to as quasi-static throughout this thesis.

The quasi-static force has been found to be typically between 1.4 and 1.6 times the static wheel load before unbalanced superelevation effects are included (AS1085.14, 2003). Forces greater than 1.6 times the static wheel load are found to be the result of more significant wheel/rail contact conditions.

An empirically based means of calculating the dynamic increment of the quasi-static force was proposed by Eisenmann (1972). Eisenmann proposed a formula that accounted for factors of vehicle speed, track condition and a value of statistical uncertainty in the parameters used. This formula was based on testing undertaken on the German railway system, which predominately ran light-axle load passenger trains at speeds up to 200 km/h.

Railway organisations throughout the world have developed their own methods to calculate the dynamic increment for quasi-static vertical load through extensive investigations by various researchers. These methods have varying degrees of complexity. Tew et al (1991) presents a comparison of dynamic increment formulae (see Appendix A). These empirical expressions have the common factor of vehicle speed but also include varied relationships of vehicle and track construction and maintenance.

The various forms of the Eisenmann formula are the most widely used methods for determining dynamic increment for Australian railway track. Broadley et al (1981) discusses the comparison of field testing, undertaken by the Railways of Australia (ROA) in 1979 and 1980, to various formulae for the calculation of dynamic increment. A modified Eisenmann formula was suggested to best represent Australian freight operations.
Eisenmann’s formula was intended as a means to describe elastic deflection of a rail under axle load. There was no mention of discrete wheel/rail irregularities, only reactions to variations in properties of the subsoil and track geometry. Broadley et al (1981) in a paper titled “The Dynamic Impact Factor”, attempted to relate Eisenmann’s formula to impact irregularities. Variations to the original formulae were suggested for speeds under 60 km/h, for loaded and empty vehicles and also for Australian track conditions. The modified formula was also only applicable up to 115 km/h as this was the highest speed during the field testing. Broadley et al (1981) redefined Eisenmann’s original track condition values based on geometry parameters of top of rail, twist, gauge and versine. They related five new track condition values to a Track Condition Index (TCI) parameter.

Although the Eisenmann approach provides reasonable results for most Australian new track design scenarios, it does have a limited range of applicability especially in relation to wheel/rail impact events. The variety of adjustment factors for the modified formula need to be calibrated against measured data to provide some degree of confidence in the output. Impact events represent an important factor in track behaviour and often cause significant problems to the track structure as a whole. Impact events often initiate problems that lead to an overall decline in condition. It is therefore important to explore these higher frequency events.

### 2.3.3 Dynamic Wheel/Rail Forces

Dynamic wheel/rail forces are principally caused by discrete changes in the wheel tread or running surface of the rail. The Australian Standard for prestressed concrete sleepers (AS1085.14, 2003) defines these forces as the load due to high frequency effects of the wheel/rail load interaction and track component response.

Irregularities causing high frequency reactions may be periodic loads such as rail corrugations or impact loads such as wheel flats, wheel burns, rail joints and welds. Literature (Jenkins et al, 1974; Newton & Clark, 1979; Grassie & Cox, 1984; Tunna, 1988; Clark et al, 1982) shows that periodic and impact loading of the track creates
forces that are not only dependent upon the characteristics of the vehicle and of the track but also the shape of the irregularity. The force on the rail arising from a typical dipped joint increases almost linearly with the product of train speed and the angle of the dip, whereas the force arising from a chordal wheel flat increases with increasing severity of the flat but not in proportion with the vehicle speed.

Jenkins et al (1974) termed the force peaks created by a wheel travelling across a dipped rail joint as P1 and P2. This notation for describing the frequency components of a force created by an irregularity was adopted by industry and is in common use today to describe limitations on forces applied to the track structure (Section 2.4).

The P1 force is a very high frequency (>100 Hz) force of less than half a millisecond in length. The force is due to the inertia of the rail and sleepers resisting the downward motion of the wheel and compression of the contact zone between the wheel and rail. Its effects are largely filtered out by the rail and sleepers, and do not directly affect ballast or subgrade settlement (Frederick and Round, 1985). However, they have a great influence on wheel/rail contact behaviour.

The P2 force occurs at a lower frequency (30 – 90 Hz) than the P1 force, although it is still classed as a high frequency force in comparison to quasi-static forces. The force is due to the unsprung mass and the rail/sleeper mass moving down together and causing compression of the ballast beneath the sleeper. P2 forces therefore increase contact stresses, contribute to the total stress range experienced by the rail and also increase the loads on sleepers and ballast. For this reason P2 forces are of great interest to the track design engineer.

**Dipped Joints and Welds**

Dipped joints are typically represented in literature as the sum of the angles of the dip between each rail and the horizontal (in milli-radians). In practice this angle consists of two components: one due to permanent deformation of the rail ends and the other due to the deflection of the joint under load. It is this total angle and not the
total dip of the joint which governs the magnitude of the track forces and stresses produced in the immediate vicinity of the joint (Jenkins et al, 1974).

Figure 2.8 shows a force/time history where a force peak occurs a quarter to a half of a millisecond after crossing a dip of $2\alpha$ mrad. The peak is high frequency and approximately equal to the vehicle unsprung mass and the track mass connected by the Hertzian contact stiffness. Jenkins et al (1974) comment that spreading of the rail head contact area can be observed close to the rail end at positions which correspond to the predicted P1 force peak. The P1 force is also responsible for rail batter just after the joint gap and production of high stresses in the rail web.

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library

The Figure 2.8 force/time history also shows that a second force peak called P2 occurs after several milliseconds. At 160km/h (100 mile/h) this peak would be in the vicinity of the first sleeper after a dipped joint. Unlike the P1 force peak which reacts mainly from the inertia of the rail and sleeper, the P2 force peak is transmitted into the ballast, producing the corresponding peak track deflection. The P2 force increases the sleeper loads at the joint and is the cause of ballast damage and track top deterioration.

Jenkins et al (1974) proposed a theoretical formula (2.1) to calculate P2 forces at dipped joints. The formula shown below was dependent on the vehicle speed, joint dip angle, vehicle unsprung mass, track stiffness and track mass.
\[ P_2 = P_0 + 2\alpha V \left[ \frac{M_u}{M_u + M_t} \right] \left[ 1 - \frac{C_t\pi}{4\sqrt{K_t(M_u + M_t)}} \right] \sqrt{K_t M_u} \]  

(2.2)

where:

\( P_2 \) = Dynamic rail force \hspace{1cm} kN
\( P_0 \) = Vehicle static single wheel load \hspace{1cm} kN
\( M_u \) = Vehicle unsprung mass \hspace{1cm} kg
\( 2\alpha \) = Total joint angle \hspace{1cm} rad
\( V \) = Speed of vehicle \hspace{1cm} m/s
\( K_t = 2K_{td} \) = Equivalent track stiffness \hspace{1cm} MN/m
\( C_t = \frac{3C_{td}}{2} \) = Equivalent track damping \hspace{1cm} kNs/m
\( M_t = \frac{3M_{td}}{2} \) = Equivalent track mass \hspace{1cm} kg
\( M_{td} \) = Rail + Sleeper mass per metre \hspace{1cm} kg/m
\( K_{td} \) = Ballast Stiffness per metre \hspace{1cm} MN/m/m
\( C_{td} \) = Ballast Damping per metre \hspace{1cm} kNs/m/m
\( \beta = \frac{1}{(K_{td} / (4EI))^{0.25}} \) = Effective track length \hspace{1cm} m

This formula is commonly used by railway organisation throughout the world to calculate theoretical \( P_2 \) forces. A discussion of limiting \( P_2 \) forces defined by railway standards is presented in section 2.4.

**Wheel Flats and Out-of-Round Wheels**

Other irregularities can also cause high frequency dynamic forces in the track. The highest loads which are normally encountered by the track are those due to irregularities on the wheel tread. These could be due to manufacturing faults, the wheel sliding on the rail, tread braking or faulty turning and grinding during maintenance and repair. Wheel irregularities are typically defined into three categories: out-of-roundness of the wheel, tread damage from loss of metal and flat zones (wheel flats) on the circumference caused by sliding.
The typical wheel flat, caused by wheelset lock-up, has a simple geometrical relationship between the depth and length. Tunna (1988) comments that after several kilometres the corners of this chordal type wheel flat become rounded due to the large contact stresses. Flats have been observed in service with lengths extending significantly around the wheel (out-of-round wheels) and it is thought that such flats are produced by the dynamic forces modifying the shape of the original flat (Lyon, 2002).

Literature (Ahlbeck, 1987; Johansson & Nielsen, 2003) has shown that out-of-round wheels also create significant forces in the track. This type of wheel damage has been termed: wheel tread runout error; long wavelength defect; or long local defects. This type of defect is difficult to detect through visual inspection and therefore most wheel removal criterion is based on the length of a supposed wheel flat.

Figure 2.9 below shows a force/time history of a reasonably significant chordal wheel flat strike. It is clear that soon after the wheel pivots on the leading corner of the flat zone, the wheel begins to free fall. The wheel/rail force drops to zero upon free fall of the wheel and the rail begins to move back toward the wheel and away from the sleeper due to its larger mass.

![This figure is not available online. Please consult the hardcopy thesis available from the QUT Library](image-url)

The force then peaks due to the wheel/rail contact upon landing. Very soon after the wheel/rail contact the combined wheel and rail masses move down back onto the
sleeper to cause a second peak. The wheel, rail and sleeper masses then all move downwards onto the ballast causing the third peak.

Tunna (1988) comments that the peak forces developed from wheel flats are different to those from dipped joints and occur in three distinct frequencies with corresponding modes of vibration of the track system. The dominant frequencies were:

- $P_1$ – the wheel mass contacting the rail – 1500Hz;
- $P_{1\frac{1}{2}}$ – the wheel and rail masses contacting the sleeper – 200Hz; and
- $P_2$ – the wheel, rail and sleeper masses contacting the ballast – 45Hz.

Unlike dynamic forces at dipped joints, wheel flat forces do not increase linearly with speed. Maximum force levels are produced when the wheel flat resonant frequency (which depends upon flat length and vehicle speed) and the frequency of the track response coincide. It could be said therefore that longer flats develop their peak forces at higher speeds. This is true only for chordal type wheel flats; however literature has shown that the flat depth is a much better guide to flat severity than their length.

Literature has shown that most wheel irregularity standards designate limits based on the length of wheel flats rather than the forces created. This is different to the standards placed on track irregularities where $P_2$ force limits are given based on the irregularity dimensions and track condition. Section 2.4 discusses the current standards and procedures used in Australia for calculation of design forces for railway track.
2.4 Design Standards and Procedures

Knowledge of the limits placed on impact forces and the procedures used in design are required for in-depth analysis of railway track. Section 2.3 Railway Track Design presented information regarding the importance of dynamic wheel/rail forces when designing railway track. These dynamic forces generated at the wheel/rail interface inevitably cause deterioration and damage to the whole track structure. Safe operation of the system as a whole requires that (Lyon, 2002):

- the maximum force levels are clearly defined;
- the design of vehicle and track are controlled so that the operation of one over the other produces forces that comply with prescribed limits; and
- the inspection, maintenance and renewal of track is sufficient to prevent the inevitable accumulation of damage and deterioration within the track from reaching levels where overall safety is jeopardised.

There are various standards, codes and procedures available to the track design engineers in Australia.

2.4.1 Code of Practice for the Defined Interstate Rail Network

The Australia Transport Council agreed to an Inter-Governmental Agreement for Rail Uniformity in November 1999. As a result of this agreement the Australian Rail Operations Unit (AROU) was established on 1 January 2000. The AROU was to work with industry to implement a Code of Practice for the Defined Interstate Rail Network for standard gauge railway linking the major cities of Australia. This new national code was in part to replace the Manual of Engineering Standards and Practices produced by the former Railways of Australia (ROA) Committee.

On 15 July 2003 the Commonwealth Government transferred ownership of the Code of Practice for the Defined Interstate Rail Network to the Australasian Railway
Association (ARA), and the AROU was formally closed. The ARA established a Code Management Company to own, manage and further develop the Code of Practice.

The Code of Practice currently consists of five volumes:

- Volume 1 – General Requirements and Interface Management;
- Volume 2 – Glossary;
- Volume 3 – Operations and Safe Working;
- Volume 4 – Track, Civil and Electrical Infrastructure; and
- Volume 5 – Rollingstock (Draft).

Volumes 4 and 5 are of interest to the track design engineer. Volume 4 Part 3: Infrastructure Guidelines (2002) details performance limits for various track components including rails, sleepers, fasteners and ballast. These performance limits include guidelines on rail discontinuities such as peaks, dips, vertical and horizontal steps in the running surface and gauge narrowing and widening. For dipped or peaked new welds a limit of 0.5 mm measured over 1 m is set. For existing track, weld limits for dips and peaks have been set to 2 mm over 1 m (total dip angle of 4.4 mrad). No dip limits are placed on jointed rail.

Volume 5 Part 2: Commissioning and Recommissioning (2003) details performance criteria for rollingstock design and sets limits for the forces that rollingstock may apply to the track. The code sets P2 force limits and states these forces shall not exceed the limits set by the access provider. For general interstate operations the P2 force should not exceed 230 kN for freight vehicles and 295 kN for locomotives. The code recommends the calculation of the vertical P2 force using Jenkins et al (1974) formula for comparative design purposes. A series of dynamic tests should also be performed to evaluate the ability of a vehicle to operate at any speed up to and including its nominal authorised speed, or to establish an acceptable maximum speed for a particular vehicle/bogie combination. Measurement of the P2 forces in-track is required for new vehicles with an unsprung mass of greater than 1.9 tonnes per axle or nominal axle loads greater than 25 tonnes. The theoretical P2 force is required to be calculated regardless of the axle load or unsprung mass and for standard gauge track must be under the limit specified. The P2 force should be determined for a
nominal 10 m rad angled ramp (total joint angle between 8 and 4 mrad). Other limiting values have been specified for dynamic loading including a wheel unloading period of no greater than 50 milliseconds and maximum body acceleration for the vehicle not exceeding +/- 0.8g (gravity).

Volume 4 Part 1: Identification and Classification of Wheel defects (2003), is also of interest to the track design engineer. Section 8: Skidded Wheels, details the limits set on wheel flats. Wheel flats of 40 to 60 mm in length or multiple flats of 25 to 40 mm in length require a speed restriction of 80 km/h. Wheel flats of 60 to 100 mm in length must be taken out of service for repair upon completion of journey, and wheel flats of greater than 100 mm shall not be moved until repair. It is interesting to note that there is no specification for depth of a wheel flat, however it is reported in literature that the depth of worn flat can be related approximately by the formula: 

\[
\text{depth} = \frac{\text{length}^2}{10 \times \text{wheel radius}}
\]

which results in the flat representing a cosine type curve.

2.4.2 Railway Track Asset Owner Standards and Specifications

Throughout Australia railway track asset owners have limits for rail dips and wheel flats similar to the Code of Practice discussed above. Both QR and the Rail Corp quote the Code of Practice standards in their rollingstock safety standards (RSU120, 2002; STD/0026/TEC, 2001).

Internationally similar limits are placed on track and rollingstock. The British, Rail Safety and Standards Board (RSSB) Railway Group Standard GM/TT0088: Permissible Track Forces for Railway Vehicles (1993) states that when a vehicle negotiates a vertical ramp discontinuity at its maximum design operating speed the total P2 force produced should not exceed 322 kN per wheel. This wheel force relates to the Jenkins et al (1974) research of the force generated by the Class 55 Deltic locomotive at 160 km/h. The current P2 force limit for vehicles travelling at 100 km/h is 250 kN which corresponds better to Australian traffic. Calculation of the P2 forces was undertaken using Jenkins et al (1974) formula with a dip angle of 20 mrad. Lyon (2002) comments that because the limit has been applied for 30 years
with apparent success, and because there is no scientific means as yet of defining such a limit in absolute terms, it has in effect become the sole means of defining an acceptable level of high frequency dynamic force. It does however represent a force level that occurs only infrequently, only from certain vehicles (mainly locomotives) and at significant dips, occurring on average once every 10 km.

The RSSB, Railway Group Standard GM/TT0089: Geometric Interfaces between Railway Wheelsets and Track (1994) provides dimension limits for wheel flats similar to the Australian Code of Practice. For vehicles with greater than 17.8 tonne axle loads a limit of 60 mm is given for immediate stop and attention. Flats of 40 to 60 mm length require attention at the end of a journey. In addition, the standard requires that where any defect such as a wheel flat produces a total force per wheel of more than 350 kN, the vehicle shall be examined upon end of journey. Lyon (2002) comments that no source is given for the figure of 350 kN and it is suggested that the most likely basis was the 322 kN P2 limit, rounded up to a generalised value.

In recent times forces created from out-of-round wheels have been given more attention. Testing presented by Johansson & Nielsen (2003) showed that a long local defect of 500 mm and a depth of 5.5 mm led to measured forces that exceeded the impact load limit given by the current Swedish wheel removal criterion. When the criterion was tested for the maximum allowed wheel flat of 40 mm in length the impact did not exceed the load level limit specified.

2.4.3 Australian Standards for Railway Track

The Australia Standard series AS 1085 specifies railway permanent way materials, manufacture, design and testing for various components of the track structure. Of particular interest to the Australian track design engineer are the codes provided for Prestressed Concrete Sleepers (AS1085.14, 2003) and Steel Sleepers (AS1085.17, 2003).

The AS1085.14 Prestressed Concrete Sleeper code states that the combined vertical design load factor (including quasi-static and dynamic loads) shall not be less than
2.5 times the static wheel load. The code also comments that combined loads equivalent to 2.5 and 3.0 times the static load have been used for balanced loads at speeds of 80 km/h and 115 km/h respectively. The code also comments that for a certain amplitude of rail head irregularity, the dynamic force between wheel and rail and the dynamic sleeper strains depend most significantly upon: the rail pad that has been placed between the rail and sleeper; the sleeper design; and how well the ballast damps the vibration of the sleeper. The code goes on to say that it would be desirable to consider the components together as a dynamic system in their design to withstand high frequency dynamic loading, and that specification of their performance should also be considered similarly.

The AS1085.17 Steel Sleeper code notes that steel sleepers are not usually designed for high frequency dynamic load effects, however where they are to be used in jointed track, dynamic effects may need to be considered. The standard goes on to state that all relevant track conditions shall be taken into account in determining the loading including dynamic effects. However the vertical wheel load formula provided is a quasi-static formula (Eisenmann, 1972; Broadley et al, 1981). The code comments that impact factors calculated using this formula do not include allowance for the effects of wheel flats, rail joints and other significant irregularities. Therefore another method needs to be used to define impact events not stated in the standard.

### 2.4.4 Limit States Design

The Australian Standard AS1085 series for railway permanent way materials, is an ‘allowable stress’ or ‘working stress’ design code. Standards Australia, in conformity with North American and European standards organisations, is committed to transforming all of its standards to the more rigorous and defensible philosophy of ‘limit states’, which is a probability based approach.

All structures have two basic requirements in common: safety from collapse and satisfactory performance of the structure for its intended use (Allen 1982). The limit states define the various ways in which a structure fails to satisfy these basic
requirements. Ultimate limit states relate to safety and correspond to strength, stability, fatigue and very large deformation due to fire or earthquake. Serviceability limit states relate to satisfactory performance and correspond to excessive deflection, vibration and local deformation. The steps involved in checking a structure or its components for any limit state are shown in Figure 2.10.

![Figure 2.10 Limit states design method (Allen, 1982)](image)

The first step in verifying the limit state of a structure is to determine the most adverse combination of loads that may occur in the lifetime of the structure. The Australian Standard load code AS/NZS 1170 (2002) does not specify loading criteria for railways. Thus a statistical/probabilistic analysis of loads would need to be undertaken determining magnitudes of loads and the load combinations so the structure has an acceptably low risk of failure or of unserviceability.

The load combinations are multiplied by a load factor ($\gamma$) determined on the basis of a probabilistic analysis of real life variabilities arising from (Gorenc et al, 1996):

- variabilities of loads in time and space;
- combination of loads and different types;
- modelling of the structure and loads; and
- analysis of the structure.

The nominal capacity of a material may also be reduced by a capacity factor ($\phi$) that takes into account variabilities arising from (Gorenc et al, 1996):

- variation in material strength, defects;
- deviations in section properties and manufactured sections;
- correlation between isolated member tests and behaviour of the member in the real structure;
- variation in accuracy of the fabricated components (tolerances);
- modelling of connection behaviour; and
- effects on strength of the fabrication defects and the construction process.

Of greatest interest to the track design engineer are the strength limit states for actions such as bending, compression and tension. The object of a design for strength is to ensure the track structure including all its components and connections have capacities in excess of the design action effects.

In the case of the serviceability limit states, a deflection, stress or acceleration due to the loads is compared with an allowable deflection, stress or acceleration. The latter are based on user acceptability and specific requirements such as for the operation of equipment.

<table>
<thead>
<tr>
<th>Load</th>
<th>Structural Analysis</th>
<th>Deflection, Stress or Acceleration</th>
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<tr>
<td></td>
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<td>Deflection ≤ Allowable Deflection</td>
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<td>Stress ≤ Allowable Stress</td>
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<td></td>
<td></td>
<td>Allowable Deflection, Stress or Acceleration</td>
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*Figure 2.11 Serviceability limit states design method (Allen, 1982)*

An important part of this research is to provide a tool for rigorous examination of the track structure. Future research will explore the provision of probability factors and rules for analysis and design of concrete sleepers and other components in the track.
2.5 Design tools used by the Australian Railway Industry

There are various tools available for track design engineers in Australia to assist in the design of railway track. These tools are typically based on quasi-static analysis of track (see Section 2.3.2). The tools usually incorporate an empirically calculated dynamic force factor such as the factor calculated by Eisenmann’s formula. However there are very few railway track dynamic analysis tools available for the designer to examine the track in depth, component by component. It is important to understand what tools are presently available and why they do not fulfill the needs of engineers to conduct rigorous analyses.

2.5.1 Quasi-static Design Tools

There are numerous quasi-static design tools available for Australian industry to use. BHP Research – Melbourne Laboratories (BHPR-ML) developed many of these for the Railways of Australia. Other programs have been developed in-house by various railway track owners and have specific purposes. Interviews were conducted with track design engineers at QR (Boyce & Hermann, 2003) and the Rail Corp (Ikaunieks, 2003); a summary is given below of some of the programs mentioned by the interviewees.

Enhanced Rail Section Module (RSM)

The Rail Selection Module (RSM) version 4 (Twiddle et al 1991), uses general equations for determining lateral and vertical force ratios acting on the rail, applied rail stresses, permissible wear limits, wear rates and rail life. The module uses imposed rail loads and resultant stresses calculated from existing finite element vehicle-track interaction models. The module is interactive and requires input of a number of design variables before delivering the design answer. Such inputs include gauge, axle load, vehicle speed, curvature of track, superelevation, wheel contact position on the rail head, track modulus, track condition factor and track maintenance
costs. The module is widely used in Australian organisations to aid in the selection of new rail and assessment for the need to replace worn and damaged rails in track relating to rail wear. The program was written in the Fortran programming language and has a straightforward user interface that provides an extensive selection of parameters. The program allows a choice of four empirical methods to calculate dynamic loads.

**Track Design and Maintenance Module (TDMM)**

The Track Design and Maintenance Module (TDMM) version 2 (Ravitharan & Tew, 1997) can be used for the selection of acceptable sleeper spacing, sleeper type and ballast depth for specific conditions during design of a new railway track. The module incorporates numerous quasi-static empirical and theoretical relationships. For the module to output reliable results, it requires selection of compatible theories for component design and analysis. The module was written in the Turbo Pascal program language and is noted to have problems with the current Microsoft Windows XP software. The module has a basic menu driven system that allows input of various parameters. The TDMM software was found to be in use by the Rail Infrastructure Corporation, however QR preferred to use the various in-house design tools developed before the TDMM was available.

**QR Track Design Spreadsheet**

The MB Design tool (Boyce, 1998) developed in-house at QR is an accumulation of various design tools for the calculation of rail bending stresses, timber sleeper stresses, sleeper-ballast contact pressures and formation pressures. Wheel-rail vertical and lateral quasi-static and P2 dynamic forces can also be calculated using these tools. QR like many other railway organisations has developed various smaller design tools to assist in various stages of the design process.

**Prestressed Concrete Sleeper Design (PCS Design)**

QR has developed a prestressed concrete sleeper design and analysis package (McCombe et al 2001) based on AS 1085.14 Permanent Way Materials – Prestressed
Concrete Sleepers, and AS3600 Concrete Structures. The spreadsheet package guides the user through the allowable stress design standard and includes detailed design of the sleepers.

**Other Design Software**

There are also other software design tools available including:

- **QR Rail Contact Stress** (Tew, 2000) a spreadsheet that calculates railhead Hertzian contact stresses and octahedral shear stresses for specified wheel and rail contact conditions;
- **CWR Buckle** version 2 (Kish, 1996) which assesses track stability assessment model for timber and concrete sleepers;
- **QBAM – Queensland Ballast Advisory Model** (Ravitharan, 1997): which is used to assess the geotechnical properties of ballast; and
- **ROA Rail Grinding Model** (Soeleiman et al 1990) which is able to calculate optimum grinding cycles for extending rail life.

**2.5.2 Dynamic Analysis Tools**

Of particular interest to this research are the available dynamic analysis tools for the Australian railway industry. The only commercially available track dynamic analysis model identified was the ‘Track Design’ software package, commonly known as TRACK (Grassie, 1994) (see also Section 3.3.9). Interviews conducted with track design engineers at QR (Boyce & Hermann, 2003) and the Rail Corp (Ikaunieks, 2003) showed that the general opinion of industry was that the TRACK software interface was confusing and the results provided were difficult to interpret. Although training in use of TRACK for both organisations had been undertaken, the interviewees were of the opinion that without continued use, the program became hard to use and the various parameters needed for inputs were not readily available.

Railway consulting organisations such as TMG and railway component manufacturers such as Pandrol, Rocla and Austrak have either developed their own
in-house analysis tools or have sought international expertise through research bodies such as CHARMEC in Sweden (see Section 3.3.6).

The American Association of Railroads (AAR), Transportation Technology Centre Inc (TTCI) produces a computer simulation program called NUCARS™ (New and Untried Car Analytic Regime Simulation) available to the international rail industry (NUCARSTM, 2003). The program is mainly used to analyse railway vehicle dynamics, but can also be used to analyse dynamic wheel-rail interaction forces, which could be used in track design calculations. TTCI has developed an extension to their NUCARS™ software to include a flexible track model for further analysis of the track structure; however this was only available to TTCI staff at the time of writing (see Section 3.3.10).

It is clear that there is a need for some software for the analysis of dynamic wheel/rail forces on the track structure. This software would need to be simple enough to understand its operation, but powerful enough to examine railway track structure interaction in detail.
2.6 Summary

A summary of the main outcomes of this chapter is provided below:

- Traditional railway track design uses an allowable stress design approach, with economic considerations frequently dictating component selection and size. The railway track in reality is a complex non-linear system, which is a characteristic that current design procedures typically ignore.

- Loads applied to the track having frequencies of a few cycles per second (up to 20 Hertz) are applied from a vehicle's response to track geometry including: design geometry, incidental geometry and support conditions. In Australia this load is termed quasi-static however it would be better described as a dynamic ride force as the load is directly related to the speed of the vehicle and condition of the track.

- Dynamic wheel/rail forces are principally caused by discrete changes in the wheel tread or running surface of the rail. Jenkins et al (1974) termed the force peaks created as a wheel travels across a dipped rail joint as P1 and P2.

- The P1 force is a very high frequency (>100 Hz) force of less than half a millisecond in length. Its effects are largely filtered out by the rail and sleepers, and do not directly affect ballast or subgrade settlement. P1 forces do have a great influence on wheel/rail contact behaviour.

- The P2 force occurs at a lower frequency (30 – 90 Hz) than the P1 force. The force is due to the unsprung mass and the rail/sleeper mass moving down together and causing compression of the ballast beneath the sleeper. P2 forces therefore increase contact stresses, contribute to the total stress range experienced by the rail and also increase the loads on sleepers and ballast.

- The highest loads normally encountered by the track are those due to irregularities on the wheel tread. Most wheel irregularity standards designate
limits based on the length of wheel flats rather than the forces generated by the wheel flat. This is different to the standards placed on track irregularities where force limits (P2 Force) are specified and calculated based on track irregularity dimensions and track condition.


- Australian railway track asset owners have set limits for rail dips and wheel flats very similar to the Code of Practice mentioned above. Both QR and Rail Corp quote the Code of Practice standards in their rollingstock safety standards.

- The Australian Standard AS1085.14 - Prestressed Concrete Sleepers (2003) states that the combined vertical design load factor (including quasi-static and dynamic loads) shall not be less than 2.5 times the static wheel load. The code comments that when designing individual track components it would be desirable to consider the track components together as a dynamic system.

- Standards Australia is committed to transforming all of its standards to the more rigorous and defensible philosophy of ‘limit states’, which is a probability based approach. The AS1170 (2002) limit state loading code does not specify loading criteria for railways. Thus a statistical/probabilistic analysis of loads would need to be undertaken determining magnitudes of loads and the load combinations so the structure has an acceptably low risk of failure or of unserviceability.

- There are various design tools available for track design engineers in Australia to assist in the design of railway track. These tools are typically based on quasi-static analysis of track. There are very few railway track dynamic analysis tools available for designer to examine the track in depth component by component. There is a need for a new approach.
CHAPTER 3

Review of Railway Track Dynamic Analysis using Models

3.1 Introduction

Knowledge of mid to high frequency dynamic wheel/rail forces is important when designing railway track, as these forces inevitably cause deterioration and damage to the whole track structure. Actual calculation of dynamic forces is however extremely complex and is by no means generally accessible to the track design engineer. To understand the railway track’s response to dynamic vehicle loading many engineering models have been established for the analysis of the track structure (see section 3.3). Very few of these analysis tools are available for use by the Australian track design engineer to examine the track in depth component by component.

The principal function of a railway track dynamic analysis model is to couple the components of the vehicle and track structure to each other so that their complex interaction is properly represented when determining the effect of traffic load on stresses, strains and deformations in the components of the railway system. Such a model provides a foundation for predicting the track performance and serves as a technical and economical device for track design and maintenance (Oscarsson & Dahlberg, 1998).

This chapter presents a review of models that allow railway track dynamic analysis and the considerations required for selection of an appropriate model. Modelling
methods and the modelling of system components is discussed. This chapter also identifies possible models of track dynamic behaviour that are available internationally and could be chosen to use as part of this research. It is important to understand the complexities and limitations of modelling railway track before choosing an appropriate model for use by the Australian Railway Industry.

3.2 Modelling the Railway System

3.2.1 Railway Dynamic Modelling Terminology

Esveld (2001) defines dynamics as the interaction between load and structure. Loads vary in time and the way this happens determines the character of the load. Structures such as railway track and rail vehicles are characterised by their frequency response function which is governed by mass elastic properties. These parameters determine the natural frequencies of the structure or those frequencies at which the structure is likely to vibrate. If the loads contain frequency components corresponding to the natural frequencies of the structure, large amplifications (resonances) may occur.

De Man (2002) comments that when considering the railway system as an assembly of structural components, mechanical properties should be appointed to the various parts of the assembly. For this purpose, several mechanical elements are available. In order to make the formulation of the vehicle and track structure not too complicated, some of the properties of the components are considered more important than others. A fundamental distinction is made between the components with mass and inertia properties and those with elastic properties.

The Mass of a component makes the track structure less sensitive towards changes in geometry, induced by dynamic forces. These changes generate in turn even higher dynamic forces and eventually damage the track. The amount and the type of load (quasi-static, impact or periodic) depends on the velocity at which the load is applied, which is often proportional to the velocity at which a vehicle is running.
The **Inertia** of the mass is the resistance to a change of velocity.

The **Elastic** properties of a component allow distribution of loads through elastic supports. The elasticity also reduces energy transfer so that force levels are lower and less harmful to underlying structures. Implicitly the energy absorption properties of elastic materials refer to the damping characteristics of the material, sometimes called the loss factor, and the elastic stiffness, all of which are present to a certain extent in every component of track.

The mechanical properties of the vehicle and track components form an essential starting point for analysing track behaviour. It is important to understand the mechanical characteristics of the main elements of the rail track structure when modelling the railway system.

Most of the equations in mechanics in use today can be traced to the work of Leonhard Euler. Bernoulli and Euler (1736) established the law that the bending moment on a thin elastic beam is proportional to a measure of the elasticity of the material and the second moment of inertia of a cross section, about an axis through the centre of mass and perpendicular to the plane of the couple.

The earliest modelling of the railway system was reported by Winkler (1867). In this model Bernoulli and Euler mechanics equations were used to define an infinitely long rail beam resting on a uniform elastic foundation representing all track components. A load was allowed to move on the rail beam to simulate the moving wheel. The deflection and bending moment of the rail beam due to the moving load were determined using this **Beam on Elastic Foundation (BOEF)** model. The BOEF model is still used in track design and analysis. Allowable rail bending stresses and allowable vertical deflections of the track are the parameters used in the design.

The range and applicability of the one-dimensional theory of beams can be extended by taking account of transverse shear deformations and, in the case of vibrating beams, rotary inertia. The equations which include these effects are generally
referred to as Timoshenko beam equations (Timoshenko, 1921 and 1922) and they have received considerable attention in literature.

The classical depiction of the track as a simple BOEF subjected to a single moving load provided an initial understanding of track vibration at the steady-state. It is nevertheless an oversimplification of track modelling when subjected to high frequency impact associated with modern-day railway tracks. Several deficiencies arise from the BOEF model:

- Sleeper mass and bending flexibility are ignored;
- The shear distortion and rotary inertia of the rail, that are important factors for high frequency vibration, and not accounted for by the simple Bernoulli-Euler theory;
- The discrete rail supports from individual sleepers are neglected and replaced by a uniform underlaying foundation;
- Impact or impulsive loading associated with high frequency vibration is not considered; and
- Detailed dynamic behaviours of track components are not obtained.

Railway track researchers have attempted to consider these aspects in the simulation of railway track dynamics. Section 3.3 presents a review of the most recent research into railway track dynamics.

The response of the dynamic system of vehicle and track can be found in either the **Frequency** or the **Time domain**. Frequency domain models calculate solutions by applying varying frequencies to the track at a fixed position. Time domain models calculate dynamic solutions by applying loads that move along the rail over a certain time period. Frequency domain calculations can often be completed quickly; however time domain solutions can take significant amounts of time for calculations depending of the size of the time step taken for analysis.

The track structure can be modelled as being either **Finite** or **Infinite** in length. The type of structure is closely connected to the solution technique. Track structures of infinite length are commonly used for frequency-domain solutions. Finite length
track structures are more appropriate for time-domain solutions, particularly if there are any non-linearities modelled. The main problem with a finite track model is that boundary conditions may introduce undesirable effects in studying the response to a moving load (Knothe & Grassie, 1993).

Excitation at the wheel/rail interface has been modelled in literature by various methods. The simplest method uses a Stationary Load and is appropriate for comparing the calculated and measured response of the track excited by a stationary periodic or momentary force (for example a vibrator or impulse hammer). An extension of this method is the Moving Load model (vertical only) that was used mainly in early theoretical investigations. A more realistic model of excitation uses a Moving Irregularity that relates the wheelset to the rail by keeping the wheelset in a fixed position on the rail, and effectively pulling a strip containing the irregularities on the railhead and wheel tread between the wheel and rail at a steady speed. It is relatively easy to study the dynamic response of a wheelset on discretely supported track using this type of model as frequency-domain calculations are straightforward (Knothe & Grassie, 1993). The most realistic model of vertical excitation of the wheel-rail contact is that of a Moving Mass (wheel) rolling on the track. The irregularities may be located on the rail or wheel surface. This type of model is also the most complex and is usually used in time-domain models.

Calculations undertaken in the frequency domain always assume that the model is completely Linear. However in reality there are several Non-linearities in the system. Components such as the rail pad and ballast have been shown in literature to be highly non-linear. Unloading of the wheel-rail contact (wheel ‘take-off’) is probably the most significant non-linear response. Thus a time-domain model is usually preferred when including non-linearities.

3.2.2 Considerations in Dynamic Modelling

Simulation of vehicle-track interaction requires complex mathematical modelling of numerous components and their connections. Some models have used mathematical
equations which mimic the track and vehicles components, others used discrete finite elements with characteristic boundary conditions. Some have lumped components together, others created very detailed models of each component.

Explanations of the mathematics involved in modelling a dynamic system are readily available in literature (Esveld, 2001; Meriam & Kraige, 1987). Many of the components in railway tracks are made of material with complicated or even unknown constitutive relationships, for example rail pads or ballast. The behaviour of these components could be load, time or frequency dependent. Because of such complexities, vehicle-track interaction models are usually not designed as ‘all-in-one’ applications, but more often ‘fit-for-purpose’ investigation tools (Andersson & Abrahamsson, 2002).

Popp et al 1999 states that the general rule for modelling is that, “Models should be as simple as possible and as accurate as necessary regarding the task they serve”.

The simplicity of a model is in regard to the need for computational efficiency and also to minimise parameter inputs. This need is typically fulfilled in linear models by using fast solution methods, which mostly work in the frequency domain. However, as previously discussed, refined models often have to account for non-linear effects (such as wheel lift-off) that require more time-consuming analysis, carried out in the time domain.

It is also important for models to have sufficiently small simulation errors. Such errors can be caused by neglecting important interaction effects throughout the vehicle-track system, incorrect assumptions regarding vehicle loading of the track, wrong assumptions regarding vehicle or track irregularities, or unrepresentative parameters for the systems various components. To avoid these errors, models are best validated against real experimental data; many models are compared against one another, but ultimately comparison to actual test data from real ‘on-track’ testing would be required.

Popp et al (1999) noted that until recently most models neglected all non-linearities and assumed a perfect periodicity of the track structure. In reality this periodicity is
disturbed because of the influence of preload, dispersed sleeper distances, voids between sleepers and ballast, and inhomogeneous subgrade characteristics. Popp et al note that there are three main reasons for the difficulties of track modelling:

1. A track is built out of materials with complicated constitutive equations, like rubber and ballast;
2. The number of contacts between the track components is very high. For all these contacts boundary conditions must be formulated; and
3. Some components have large dimensions such as the infinite length rails or infinite half-space subgrade, which causes problems for spatial discretisation of the structure.

Grassie (1994) commented that it was questionable whether the sophistication of some models was yet necessary. This was in view not only of the relative ignorance which exists about the physical behaviour of essential components such as ballast and rail pads, but also of the variability of typical track.

In general, there is not a single adequate model of a system, but many different ones depending on the different tasks they are created for. In complex dynamic systems such as the railway system, models for components such as a wheel, pad or sleeper are initially built separately. These component models are then combined to describe subsystems like the vehicle subsystem (section 3.2.4), or track subsystem (section 3.2.6). Finally, a combination of the subsystems models results in the entire system model.

### 3.2.3 Modelling Methods

Railway system components can be divided on the basis of their principal properties: either mass or elastic properties, or both. Together with the geometrical design (layout) of a track structure, a mechanical design or a model can be described. Such a model is basically formed by a set of relationships between all components with inertia properties. These relationships are influenced both by elastic properties and
by dimensions of the components. The set of relationships gives a mechanical model of a track structure, suitable for the analysis of the structural behaviour.

De Man (2002) comments that in order to combine properties and dimensions into models, two modelling methods may be used: analytical and numerical modelling. The latter is in fact a gathering of elementary analytical models.

**Analytical** models are preferably based upon homogenous situations, in which:

- continuous conditions apply with respect to support;
- there is a limited number of connections;
- there is a limited number of load positions; and
- there is a clear definition of the conditions at infinity and at symmetry axes.

These conditions facilitate the retrieval of (simple) analytical solutions. Zimmermann (1888), Euler, Bernoulli (1736) and Timoshenko (1926) have found mathematical solutions for an infinite beam on an elastic foundation. More comprehensive additions to dynamic behaviour, multiple beams, support conditions and other properties have been included and will be discussed in detail in section 3.2.6.

**Numerical** models are typically used for more refined stress analysis of track components and where retrieval of solutions in analytical models is difficult. Instead of finding a solution in a continuous input range, numerical methods search for the solution of a model, comprising of nodes, connecting elements and boundary conditions. All component properties and model restrictions have to be embedded in the definition of this numerical model, which is generally termed a finite element model (FEM).

Complete finite element modelling of the full track system is complicated due to the interface characteristics of the various track components. With the proper selection of the required parameter values and constitutive relationships, three dimensional and non-linear models should provide a better solution to track responses under wheel loads. However, due to their high degree of sophistication and requirement for
accurate constitutive relationships, these models lack convenient application to field problems and design.

Furthermore, numerical models are usually complicated and require a large amount of computer power and memory for calculations. Finite element modelling also requires sophisticated software that is often not very user-friendly. For these reasons this research will focus on the use of analytical models for the assessment of the dynamic characteristics of the railway systems.

Typically the entire system of railway components is divided into two subsystems, the vehicle and the track, between which there is a physical wheel/rail contact. It is common that the behaviours of these dynamic subsystems are investigated quite separately. Such investigations may include either a complex vehicle model together with a simple track model or a simple vehicle model with a complex track model.

Railway track dynamic behaviour occurs in a fairly wide band of frequencies that induce vibration in the system, as discussed in Chapter 2. The suspension system between the wheelset and bogie (primary suspension) is the first to reduce vibrations originating from wheel/rail interaction. The reduction of the vibrations of lower frequency is dealt with in the second stage between the bogie and car body (secondary suspension). Esveld (2001) suggested that this terminology can also be applied to the track structure, where the rail pad and fastener represent the primary suspension of the track and ballast/subballast layers represent the secondary suspension. Using this theory a model of masses, springs and dampers can be developed for the entire railway system.

### 3.2.4 Modelling the Vehicle

Vehicle submodels may be represented either by **Rigid or Elastic Multibody Models**. Rigid multibody models are appropriate in the low frequency range of quasi-static forces from 0 Hz to 50 Hz. Elastic multibody models are appropriate in the mid to high frequency range of about 50 Hz to 20 kHz and have been typically
developed to solve acoustic problems related to wheel resonances (De Man, 2002). For the investigation of railway track, rigid multibody models have been found by the majority of model creators to be adequate.

Vehicle submodels may be classified broadly into three categories as shown below:

![This figure is not available online. Please consult the hardcopy thesis available from the QUT Library](image)

The first category considers single wheel with static wheel load moving along a track. The static load representing the vehicles mass may be applied directly on the wheel or through a primary suspension element.

The second category considers one bogie or half a car body and consists of two wheelsets that are connected to the sideframes either directly or through primary suspensions. The sideframes support the mass either directly or through the secondary suspension.

Popp et al (1999) comments that in the mid to high frequency range the dynamic behaviour of the car body and the bogies is decoupled by the soft secondary suspension. Thus, very simple models of the car body are sufficient. Frequently it is
even replaced by its constant weight and a kinematical constraint. More detailed models would only be required for ride comfort analysis.

The third category in Figure 3.1 considers a single wagon with all components including a car body resting on the bolsters. The bolster may rest either on the secondary suspensions or directly on the side frames. The bogies may be modelled either with or without primary suspensions.

3.2.5 Modelling the Wheel/Rail Contact

The vehicle and track subsystems are excited by the same wheel/rail forces at the wheel-rail interface. Elkins (1992) surveyed literature regarding the wheel/rail contact and found that the Hertzian Contact Theory of two elastic bodies in was valid in most circumstances. Railway track dynamic models in the literature typically use the Hertzian Contact Theory to represent the wheel-rail contact as a linear or non-linear contact stiffness.

Most two dimensional models consider the wheel/rail contact as a point, thus the real shape of the contact patch is not accounted for. When modeling non-linear contact, the force is determined taking into account the varying size of the contact area, which is assumed to be an ellipse. As long as no irregularities of wheel or rail with wavelengths shorter than the contact dimensions are considered the Hertzian theory yields good results and no filters need to be applied.

3.2.6 Modelling the Track

Models of track dynamic behaviour may be broadly classified into two categories: those that represent the track as a continuously supported rail beam and those that represent the track as a discretely supported rail beam.
**Continuously Supported Models** of infinite length have been extensively investigated in literature. They are based on the beam on elastic foundation (BOEF) theory (Hay, 1982). The BOEF method was originally developed for longitudinally sleepered tracks, and has since been applied to transversely sleepered track. Tew *et al* (1991) commented that many studies have tried to develop methods for discrete elastic supports (sleepers), but the results do not differ significantly from those of the simple BOEF model. Hence, the BOEF model is generally regarded as the most acceptable method for the analysis of rail foot stress and rail deflections.

The stiffness of the whole track structure on an elastic foundation is known as the track modulus and is defined as the force per unit of deflection per unit of track length. The rail, pads, fastenings, sleepers, ballast, subballast and subgrade are components that determine the value of the modulus of track elasticity. The track modulus is a crude average or composite value of the individual stiffness values of the track’s components (Hay, 1983).

**Discretely Supported Models** are similar to the continuously supported models but allow for the discrete spacing of sleepers and often have multiple layers representing the rail pads, sleepers, ballast, subballast and subgrade. Grassie *et al* (1982) noted because the simple BOEF model ignores the discrete support of the rail and discrete mass of the sleepers, the model may become inadequate at high excitations. The reason is that at the higher frequency range up to 1500 Hz, corresponding to short wave corrugations of 40-80 mm wavelengths, the wavelength of bending waves in the rail becomes comparable with the sleeper spacing. In addition, continuously supported models cannot deal with non-linear effects, such as loss of contact, and the effects of rail pads in concrete-sleepered tracks (Zhang, 2000). Discretely supported models can be solved in both the frequency and time domains, but frequency domain techniques reach their limits if non-linearities inside the track structure are taken into account.

There is a large variety of types of discretely supported track model available because of variability in the way the following factors are represented in each model:

- the number of layers of the track structure;
the kind of load: forces or masses, both stationary or moving;
the spatial dimensions of the structure: finite or infinite;
the periodicity: perfect or disturbed;
a consideration of nonlinearities;
the direction of motion: vertical, lateral or longitudinal; and
the modelling of single components, for example the rails or the ballast.

A discussion of the mechanical characteristics considered in modelling track components is provided below.

**Rails** are linear elements characterised typically by an infinite length. This allows modelling the rail as beams. Rails have flexural stiffness in vertical and lateral directions and compression stiffness in the longitudinal direction. Rails also have a shear stiffness which is often neglected.

Rails can be modelled using either the Euler-Bernoulli or the Timoshenko (1926) beam theories. Shabana & Sany (2001) commented that several studies have shown that when the frequency of the vertical excitation force on the rail is less than 500Hz, the Euler-Bernoulli beam model leads to satisfactory results. However, in the case of higher frequencies, the shear deformation effect becomes increasingly important and Timoshenko beam models lead to accurate results.

The rail **Fastening System** commonly used on concrete sleepers comprises a resilient spring **Fastener**, acting essentially in parallel with a much stiffer **Rail Pad**. Knothe and Grassie (1993) comment that the load/deflection behaviour of the fastening system is non-linear; however since its behaviour when loaded by a wheel is of greatest interest, some linearization of the load/deflection behaviour can be justified. For vertical vibration a pad is usually modelled as a spring and viscous dashpot in parallel. The pad is represented similarly in models of the lateral dynamic behaviour of track. Rail pads are mainly loaded in compression, permanently by the fastening system and/or repetitively by the rail traffic. In two dimensional models a pad can be represented as acting at a point on the rail foot, however for three dimensional models a visco-elastic layer across the rail foot is often considered (Kumaran, 2003). The inclusion of rail pads is effective in reducing this force as it...
reduces the effective track mass acting on the sleepers and ballast (Newton & Clark, 1979). Thus the rail pad is the one component which can be most readily changed to influence the track’s dynamic behaviour and rate of degradation.

**Sleepers** are also linear elements, positioned in the horizontal plane just below the rails. Sleepers may be modelled as rigid beams or beams with flexural and shear stiffness or as rigid bodies. Knothe and Grassie (1993) comment that the most complete sleeper model is a Timoshenko beam of variable thickness. Sleepers may be represented by one element or a group of elements which can have changing dimensions to more closely represent the sleeper profile. At higher frequencies, the mass of a sleeper becomes increasingly important and it is essential to consider the sleeper as a dynamic component that has both mass and stiffness. Because of its distributed mass and stiffness, the sleeper resonates at a series of frequencies, the most significance for typical sleepers being around 200 Hz and 650 Hz (AS1085.14, 2003).

**Ballast** and **Subballast** is typically modelled as a load distributing material. Ballast material in particular deflects in a highly non-linear manner under load due to voids at the sleepers/ballast interface and also in the ballast itself. Despite this, ballast beds are often modelled by discrete or distributed linear springs and viscous dampers in the vertical direction. Some models include shear properties and elastic layers in three dimensions (Zhai *et al.*, 2001; Sun, 2002). The mass properties of a ballast bed are also important; however the amount of mass that should be incorporated in dynamic models is difficult to estimate. Another possible model represents the ballast and subgrade together as an elastic or visco-elastic half-space which considers the fact that track is supported over a number of sleepers rather than a single sleeper.

**Subgrade** material is similar to ballast material in its properties. The load distribution influences the effective mass and the stiffness and damping properties, so that modelling is considered as very complicated. Detailed three dimensional layer models or half space models would resolve such problems more accurately than simplified distributed linear springs and viscous dampers, but require more detailed input parameters.
3.2.7 Modelling of Irregularities

Most models mathematically represent wheel or rail irregularities as a one dimensional variation of the vertical surface profile. The profile of the wheel or rail head is taken into consideration in some three dimensional models concerned with wear in the two components. However for analysis of mid to high frequency dynamic loading in the vertical direction a variation in surface profile is adequate (such as when modeling wheel flats or peaks and dips in the track surface).

Models of railway track dynamic behaviour are typically focused on the effect of periodic or discrete irregularities in the wheel or rail profiles. Models will often be created for analysis of one type of irregularity, such as corrugations or wheel flats. Some include an ability to model a wide range including dipped or peaked welds, dipped joints, out-of-round wheels or arbitrary profiles of the rail or wheel surface.

Some models consider lateral displacements in the track’s geometry and the effects this has on the track dynamic behaviour. These models will be discussed section 3.3.

3.2.8 Outputs of Models

Models of dynamic railway track behaviour can produce a varying number of outputs depending on the purpose they serve. The dynamics of the railway system are typically measured as accelerations of the masses of individual components connected by elastic elements. These accelerations are converted into velocities and then into deflections of individual components by integration (Esveld, 2001). The deflections over time allow the calculation of forces (in Newtons) and then stresses and strains by accounting for the material properties of each component. The forces on components with linear extensions (such as rail and sleepers) also allow the calculation of bending moments.

Thus for any model many types of output can be calculated. For different situations different outputs are required, thus some models have limited outputs programmed. However these could be extended to a greater range of outputs depending on the results of interest.
3.3 Criteria for the Selection of a Model

This research aims to select and develop a user-friendly model of track dynamic behaviour that is capable of establishing the full breadth of likely and extreme forces or the range of track and rollingstock characteristics found in Australia. The comprehensive set of criteria for selection of a model is detailed below. The model selected must:

1. be readily available for evaluation and, if necessary, be able to be released by the owner for development in the project and ultimately for use by Rail CRC partners;
2. be able to be readily improved in applicability and scope (where necessary);
3. ultimately incorporate a user interface for ease of operation;
4. model the passage of single, double, or triple axle bogie assemblies;
5. fully specify the mass and operating characteristics of a vehicle;
6. fully specify rail section and material properties, rail pad and fastener properties, gauge, sleeper section and visco-elastic/dynamic material properties, spacing and number of sleepers, ballast and sub-ballast depth and properties and formation properties;
7. define a wide range of wheel and rail discrete defects;
8. produce outputs including dynamic and impact force histories for components throughout the track structure, including resonant frequencies of track and vehicle, dynamic compliances, force distributions, bending stresses in sleepers, ballast and formation pressures, and track deformations;
9. be upgradeable (where possible) in the future to include lateral and longitudinal loads and restraints, thermal actions, and track stability.

This is a comprehensive list of criteria and no one model will meet all of the points. However the best model for selection should meet as many of the criteria as possible.
3.4 Identification of an Appropriate Model

Various analytical and empirical models exist that have been developed to represent the railway track structure under the transient loading of a passing train. This section identifies and reviews some of the most recent developments in track dynamic analysis modelling throughout the world. A comparison of the models of particular interest to this research is provided in Appendix B. A focus is placed on comparing the capabilities of the models available for use by the Australian railway industry.

3.4.1 Australia

Centre for Railway Engineering

The Central Queensland University Centre for Railway Engineering is well known throughout Australia for research expertise in train dynamics, wagon and bogie dynamics, wagon/track system dynamics. Sun (2003) recently published his PhD thesis on a three dimensional wagon-track system dynamics (WTSD) model. The model can predict the lateral and vertical dynamic response of a wagon running at a constant speed on a tangent track. The program was developed with particular attention to wagons and track used for heavy haul transportation in Australia.

The model was very comprehensive and is shown in Figure 3.2 & 3.3. It has the flexibility to include as many components of the wagon and track as required for a specific analysis. The model consisted of three subsystems: wagon, wheel-rail interface and track.

The wagon subsystem consisted of a full wagon with 37 degrees of freedom (DoFs) including a freight wagon car body and two bogies. Each bogie consisted of two secondary suspension elements, two sideframes, four primary suspension elements and two wheelsets, as shown in Figure 3.2(b). All components were considered as rigid bodies with mass and inertial moments in three directions.
This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.

Figure 3.2 Schematic diagram of the wagon subsystem
Figure 3.3 Schematic diagram of the track subsystem

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.
The suspension elements were represented by linear springs and dampers, with the primary suspension allowing longitudinal, lateral and vertical viscoelasticity, and the secondary suspension allowing only lateral and vertical viscoelasticity. The wagon subsystem could be easily reduced to represent a traditional three-piece bogie resulting in a 15 DoF system.

The track subsystem model contained four layers represented by springs and dampers and was based on the discretely supported approach. The subsystem shown in Figure 3.3 is based on the traditional ballasted track structure and comprises two rails, sleepers, fastener and pad assemblies, ballast, subballast and subgrade. The lateral and the vertical bending and shear deformations of the rail beam were described using the Timoshenko beam theory extended by considering the torque of the rail beam. The sleepers were considered as rigid short Euler beams resting on an elastic foundation and were represented by their mass and viscoelastic properties at the rail seat location. The ballast and the subballast layers were represented by their mass and viscoelastic properties. These properties are determined from a pyramid model that was widely reported in literature (Ahlbeck et al. 1975, Zhai & Sun, 1993, Sun & Dhanasekar, 2002). The ballast and subballast were connected in the vertical and horizontal directions to depict continuity of these layers. The subgrade was represented by its viscoelastic properties without mass.

The normal contact force due to wheel-rail rolling contact was determined using a modified Hertzian static contact theory including a multipoint contact sub model first developed by Dong et al. (1994). Creep forces and moments were determined using Kalker's linear creep theory (Kalker, 1967).

The equations of motion for the vehicle and track subsystems are solved in the time domain using a modified Newmark-\(\beta\) method developed by Zhai and Sun (1993). The model was capable of predicting the natural frequency of a discrete defect-free and/or defect inclusive wagon-track system. The model could examine lateral and vertical dynamics of the vehicle and track system components to discontinuities at the wheel-rail interface that could be either symmetric or un-symmetric to the track geometry. The model could analyse uneven vehicle loading and includes the
potential to predict tangent track derailment due to wheel flange climb on the rail head.

The 3DWTSD model was capable of dealing with any possible discrete defect of the wheel and the rail including wheel flats, out-of-round wheels, indentations on the rail top surface (spalling and shelling of the rail), rail corrugations, and dipped rail joints. Defects in the wheel and/or rail could be defined by simple sinusoidal functions or any other complex function that described the shape, size and location.

Limitations of the 3DWTSD model included the lack of validation undertaken of the track submodel, and the length of time to undertake analysis (up to 3hrs). However, the vehicle submodel had been extensively tested using the facilities available at Centre for Railway Engineering (Sun & Dhanasekar, 2003).

**Cooperative Research Centre for Railway Engineering Technologies**

The Rail CRC’s mission is to deliver valuable research, knowledge and innovation to the Australian railway industry using an internationally collaborative approach. This research forms part of the Rail CRC as discussed in Chapter 1.

Sun *et al* (2004) presented an extension of the Centre for Railway Engineering work for the Rail CRC to examine the wheel/rail impact forces due to track geometry irregularities such as vertical surface profile, cross level defects, gauge variation and alignment variation. It was shown that the alignment irregularities contributed most to the lateral impact whilst vertical surface profile irregularities contributed most to vertical impact.

There are presently many other research projects being undertaken through the Rail CRC. Investigation of parameters to be used in models of track dynamic behaviour is of great interest to this research. Results from in depth examinations of rail pads, prestressed concrete sleepers and ballast should become progressively available.
**University of Queensland**

Daniel & Meehan (2003) and Meehan *et al* (2003) presented models developed through a Rail CRC project on analysis of the development of corrugations on rails. One of the key project aims was to develop integrated, analytical and numerical models for rolling contact instability. Various detailed models have been developed for a linear and non-linear wheelset and bogie.

The bogie model represented components through numerical means in the time domain. It predicted steady cornering of the bogie through various traction and creep relationships. Wheelset model had been developed in both the frequency and time domain. The non-linear wheelset model allowed the accumulation of wear and includes flexibility to model bending and twisting modes. Traction and creep properties at the wheel/rail interface permitted stick-slip behaviour at the contact.

The rail was modelled as Timoshenko beam elements with consistent mass matrices to represent the track. The sleeper was as a mass, connected to the rail by a spring and a viscous damper representing the rail pads. The ballast was also included as a spring and damper connected to each sleeper.

Research is ongoing at University of Queensland and the vehicle model is expected to be expanded to include the car body. A limitation of the existing model is its inability to model the sleepers’ flexural characteristic, however this could be incorporated into the finite element model of the track.

**3.4.2 Canada**

**Queen’s University & Royal Military College**

Cai (1992) presented a model with the primary objective of establishing an improved and comprehensive theoretical model of rail track dynamics for investigating the fundamental characteristics of railway track dynamic responses and wheel/rail interaction. This model was known as Dynamic Analysis of Rail Track Structures.
(DARTS). The two dimensional model includes three subsystems: a double-axle bogie, wheel/rail interface and track.

The vehicle was represented by a single double-axle bogie with four DoFs as shown in Figure 3.4. The model represented transversely symmetric vibration, thus limiting the bogie model to include only two wheels (unsprung masses) and a side frame on one rail. The bogie side frame and rotatory inertia were considered. Cai & Raymond (1992) noted that the vehicle components above the bogie would not contribute significantly to high frequency wheel/rail impact, due to the low resonant frequencies common to wagon dynamics; thus only the static car body weight was included.

The track was represented by rails discretely supported by flexible sleepers as shown in Figure 3.5. Both the rail and sleepers were modelled using the Timoshenko beam theory. The sleeper cross-section could be varied in three places and ballast properties along the sleeper length could be varied. Rail and sleeper interaction was modelled through a linear rail pad stiffness and damping elements. The underlaying track bed (including ballast, subballast and subgrade) was modelled as a continuous array of linear springs and viscous dampers. Ballast and subgrade layering effects were taken into account in the model by incorporating a method to estimate the track modulus developed by Cai, Raymond and Bathurst (1994).

The wheel/rail interface was modelled using the Hertzian contact theory and the calculation of wheel/rail interaction forces was undertaken in the time domain by employing the 4th order Runge-Kutta method (Cai & Raymond, 1992).
The model was capable of calculating the dynamic response of the track to many discrete irregularities including wheel flats, corrugated rail, randomly worn wheel profile, rail joint, uneven weld and other random rail profile shapes.

The model was capable of performing four types of analysis of the railway system:

1. Natural frequency characteristics; including natural frequency and mode shapes of individual track components and the entire track as an integral structure;
2. Dynamic track responses in the frequency domain; including frequency response characteristics of the entire track, and of the rail and the sleeper as integral components of the track;
3. Dynamic track responses under a stationary impact load; including dynamic responses for rail-sleeper displacement, acceleration, shear stresses, rail seat forces and ballast pressures at any wayside location along the track; and also rail and sleeper deflection and stress wave shapes across the track;
4. Dynamic track/wheelset responses to moving wheel-rail interaction for the input of excitation sources from any wheel or rail profile irregularities. The predicted dynamic responses included:
- On-board reactions; wheelset and car frame acceleration and displacement; and stress/strains under wheel-rail contact points; and
- Wayside reactions; dynamic stresses/strains in the rail and a chosen sleeper at any specified location; rail seat forces and ballast pressure for any specified sleeper; acceleration and displacement of rail and sleeper at any specified location.

The DARTS model has been used by its author for various commercial applications and is said to be validated. The model appears comprehensive allowing an extensive range of analysis. One limitation of the DARTS model is that it considers only transversely (or cross-track) symmetric track vibrations. It would also be advantageous if the track bed could be modelled as a layered material rather than just a Winkler foundation.

3.4.3 China

Train & Track Research Institute (Southwest Jiaotong University)

Zhai (1991) presented a detailed model for the investigation of vertical interactions between railway vehicles and track. Zhai & Sun (1994) extended this model to show the significance of mutual dynamic influence of the neighbouring wheelsets via the rail and the bogie. The vehicle-track vertical interaction model was established based on the coupling dynamics shown in Figure 3.6.

The vehicle model included two double axle bogies in a 10 DoF lumped mass system comprising the car body mass, two bogie frame masses, four wheelset unsprung masses and moments of inertia for each component. The side frame mass was linked with the wheelset through the primary suspension and linked with the carbody mass through the secondary suspension. The vehicle was assumed to move at a constant velocity.
The track model used an Euler-Bernoulli beam to represent the rail and is discretely supported at rail-sleeper junctions by a series of springs, dampers and masses. The three layers of discrete springs and dampers represented the elasticity and damping effects of the rail pads, ballast and subgrade. The two layers of discrete masses below the rail represented the sleepers and the ballast. The ballast mass blocks are interconnected elastically, so vertical deflection of one ballast block was spread via shear springs and dampers to neighbouring blocks. The system interaction between the vehicle and the track was described by the non-linear Hertzian contact theory.

Computer software named Vertical Interactions between Cars and Tracks (VICT) had been developed to calculate the dynamic responses in the time domain of both vehicle and track due to various wheel/rail geometric irregularities as well as track support stiffness variations in the longitudinal direction. Zhai (1996) developed a new fast time integration method known as the Newmark-\(\beta\) method which was adopted in VICT to increase the computational speed for large dynamic systems involving non-linearities. Sun (2003) also used this method in the 3D WTSD model.
VICT was verified by many field experiments conducted with a variety of vehicle and track conditions representative of Chinese trains and tracks (Zhai & Sun, 1994; Zhai & Cai, 1997). The software was capable of including various discrete geometrical irregularities on the surface of rail and wheels such as out-of-round wheels, wheel flats, corrugated rails, and sinusoidal rail deformations. The model was also useful for simulating varying track parameters, such as sleeper spacing or track stiffness, which may vary arbitrarily in the rail longitudinal direction. The model could also analyse special problems including the dynamic influence of damaged components in the track structure such as useless rail fastenings or pads, and hardened ballast blocks.

Zhai et al (1996) presented a model which considered the vehicle and track as a whole system and coupled vertical interaction with lateral interaction. The model was an extension of the model used in the VICT software and allows 3D railway system interaction. The vehicle model was represented by a multi-body system with 37 DoF, which run on the track at a constant velocity (Figure 3.7). The track

Figure 3.7 The coupling model of vertical and lateral vehicle-track interactions
substructure was extended to represent the lateral restriction of the fasteners and ballast, including the cross track shear connection of the ballast blocks.

The model has been used to investigate dynamic interactions between railway freight cars and tracks due to track twist, combined alignment and cross-level irregularities. This investigation was to provide some safety limits against derailments for China’s Railway Track Administration. The model was capable of predicting vehicle-track interactions from combined irregularities that couple the vertical with the lateral irregularities at the same place of the track.

Zhai & Cai (2002) described a numerical technique that was used to investigate dynamic train/track/bridge interactions. Two dynamic models were established to simulate the dynamic responses of a train running on bridges with ballasted track and non-ballasted slab track, respectively. The models investigated vertical dynamics only and allowed for multiple vehicles including a power car and a number of passenger cars. The simulations using the models demonstrated that the bridges under investigation are able to satisfy the demand of dynamic performance of high-speed transport. The dynamic indices were to be measured and then used to validate the simulation method.

The only limitation noted by the writer regarding the VICT model was that the sleepers could not be analysed as flexible beams allowing the calculation of bending moments at the rail seat and centre.

3.4.4 Germany

German Research Council

Popp & Schiehlen (2003) presented a collection of papers titled ‘System Dynamics and Long-Term Behaviour of Railway Vehicles, Track and Subgrade’. The work was produced as part of a program funded by the German Research Council (DFG), with a goal of better understanding of the dynamic interaction of vehicle and track, and the long-term behaviour of the components of the system. Germany like most
countries around the world has been aiming to increase traction, axle load and travelling speed. However, new developments had revealed new limitations:

- Settlement and destruction of the ballast and subgrade had led to deterioration of the track;
- Irregular wear of the wheels had caused an increase in overall load and deterioration in passenger comfort; and
- Damage of the running surfaces of the rail and the wheel were becoming more frequent.

**Technical University of Berlin**

Much of the rail research undertaken in Germany in the last 15 years was led by a branch of the Institute of Air and Space Travel (ILR) dedicated to rail transportation called the ‘Field of Activity Construction Computation’ at the Technical University of Berlin. Like the Rail CRC in Australia, the Institute aimed to improve the long-term behaviour of the railway system.

One of the first track dynamic models produced in Germany was a finite element track model used by Knothe and Ripke (1989) and by Hempelmann *et al* (1992) to study the deterioration of track components, the propagation of sound, and the formation of rail corrugations and other rail irregularities (Figure 3.8).
In the FE model, the railhead, web and foot were treated as stacked individual Timoshenko beams. The fastenings and rail pads were described by springs and dampers. The sleepers were considered rigid bodies, and the ballast was represented by a system of springs in parallel with dampers. Only one wheelset was considered.

Ripke and Knothe (1995) presented a time-domain method for simulating vertical vehicle-track interactions (SiRaGe). The track model could consider many irregularities and non-linearities. Modelling was restricted to vertical and longitudinal dynamics with symmetric loading of the track. Single non-linear effects such as non-linear suspension or voided sleepers could be considered by using non-linear forces. The vehicle model consisted of a rigid car body, up to two rigid bogies and up to four elastic wheelsets. The primary and secondary suspensions were modelled using springs and dampers. The track model consisted of at least 30 sleepers modelled either as rigid bodies with three DoF or as Timoshenko beams with six elements per sleeper. The rail was also a Timoshenko Beam with up to four elements between sleepers. The rail pad and ballast were modelled using linear springs and dampers, with an effective mass for the ballast. Ballast masses were connected with springs and dampers for consideration of wave propagation.

Knothe & Wu (1998, 1999) extended the frequency domain model developed by Ripke & Hempelmann (1994). Knothe & Wu investigated the vertical behaviour of railway track on an elastic half-space and layered half-space. The results were compared to Ripke & Hempelmann’s simpler model, where the ballast and subgrade were considered as a viscoelastic foundation. In the low and mid-frequency range up to 250Hz, great differences were observed between the two models. Contradictions observed in previous work were explained using Knothe & Wu’s half-space model. For frequencies higher than 250Hz, the influence of the subgrade was found to be negligible, thus the simpler viscoelastic foundation model could be used.

Gerstberger et al (2002) presented a frequency domain model for vertical and lateral dynamics and two time domains models for vertical dynamics of a ballasted track. The three models took into account the subgrade behaviour using half-space theory.
The frequency domain analysis was restricted to the mid-frequency range up to 600 Hz, which was relevant for the mechanisms of excitation considered. The ballasted track model was considered as an infinite, periodic structure placed on an elastic half-space. The discretely supported model is shown in Figure 3.9 and comprised of rail pads, sleeper, sleeper pads and ballast. The pads were modelled by elastic elements with visco-elastic or structural damping. The sleeper was modelled as a rigid mass with additional elastic modal DoF to account for the first symmetric and anti-symmetric bending mode of the sleeper. Further analysis modes could be accommodated in the model if necessary. No rotation of the sleeper around the vertical axis of the track was allowed. The ballast was modelled using discrete blocks beneath the area of contact of the sleeper and the ballast layer. The mid-span section of the sleeper was assumed to be a non-contact area. In the vertical direction the cross-sectional area of the block varied in accordance with the angle of load-distribution within the ballast material.

Gerstberger et al (2002) commented that the validity of frequency domain models in general was limited, due to the fact that the dynamic loads must not exceed the static loads. In the case of total unloading, lift-off of the wheel from the rail was most likely to occur and non-linear contact formulations had to be used to simulate the interaction of wheel and rail accurately.
Gerstberger et al (2002) also presented a general time domain model in order to account for a greater variety of non-linear mechanisms (Figure 3.10). The model allowed for mechanisms of excitation leading to non-linear behaviour such as voided sleepers and out-of-round wheels.

The non-linear components of the system included sleeper pads, ballast and ballast mats. The vehicle model consisted of a half-car body, single bogie and two wheelsets supported by longitudinal and vertical spring and dashpot elements. Non-linear Hertzian contact conditions defined the wheel-rail interface and wheel lift-off was possible. The rail was discretely supported, finite in length and rests on visco-elastic linear rail pads. The sleepers were represented by rigid masses resting on non-linear sleeper pads. The sleeper/ballast contact allowed for lift-off and ballast voids. The ballast was modelled as an elastic rod with mass and damping properties. A ballast mat was included that allowed lift-off.

**German Railways Comparative Study of Track-Subsoil Calculations**

A benchmark test (Ruecker et al, 2003) was arranged for models of ‘vehicle-track-subgrade’ amongst partners of the DFG-priority program “System dynamics and
long-term behaviour of vehicle, track and subgrade”. The benchmark was initiated in order to validate a range of models and the different calculation procedures that had been created by various research organisations in Germany. The report focussed on complete track systems including ballasted and slab track systems with homogeneous and layer half-space.

The following findings were made regarding the numerical models of track:

- The number of sleepers taken into account in a track model only marginally affected the static track flexibility. For frequencies higher than 20 Hz the flexibilities were almost identical independent of the number of sleepers;
- The type of material damping of the ballast (viscous, hysteretic) had no influence on the static value but affected the flexibility functions of the track with increasing frequency. A viscous damping element lead to smaller values of the flexibility functions;
- The coupling boundary conditions between ballast and subsoil was of importance with regard to the amplitudes of the flexibility functions up to approximately 50 Hz;
- The flexibility of the complete system was influenced mainly by the calculation of the stiffness matrix of the subsoil;
- All models in the benchmark described the behaviour of a layered half-space sufficiently. Reduced radiation damping in a layered half-space allowed the possibility of having resonances in block-ballast models; and
- The flexibility functions of adjacent sleepers were described sufficiently by all models. Therefore analytical models which included coupling through the subsoil were suitable for the modelling of extended sleeper systems.

Research on track dynamic behaviour in Germany is ongoing and the results of further advancements in the use of non-linear models will be very interesting.
3.4.5 India

Indian Institute of Technology (Department of Civil Engineering)

Kumaran et al (2003) presented a railway ‘system’ model to analyse the dynamic response of a typical prestressed concrete railway track sleeper due to wheel/rail interaction dynamics. Dynamic amplification factors were calculated for deflection, ballast pressure and bending moments at critical sections (rail seat and centre) for various exciting frequencies under different vehicle-track parametric conditions. The results were used for a basis of improved and rational design of sleepers. The ‘system’ included three sub-models for dynamic analysis: a vehicle model, track model and isolated sleeper model.

The train vehicle model adopted for the study was one conforming to typical vehicles on Indian railways. The analytical model consisted of a car body, two bogie frames and four wheel sets (Figure 3.11). The car body, bogies and wheelsets were modelled as rigid bodies with mass and inertia. The properties of the primary and secondary suspension was characterised by linear spring stiffness and damping. In total 17 DoFs were considered in the study for the vehicle model. Modelling of the wheel-rail contact conditions was not clearly explained, however it is mentioned that no loss of contact between the wheels and the rail was allowed.
The track structure was a numerical three dimensional finite element model. The model consisted of a rail, rail pad, sleeper, ballast, subballast subgrade and a track length encompassing 12 prestressed concrete sleepers. The rail was modelled as solid Timoshenko beam elements’ resting on discrete supports. The rail pads were represented as a spring element with stiffness and damping, while the spring behaviour of the rail fastenings was not considered. The sleeper, ballast and subballast were modelled as solid elements with elastic properties. Contact between the sleeper and the ballast was considered along the length of the sleeper. The subgrade was represented using linear boundary conditions as shown in Figure 3.12. The vehicle and track models were used to determine the rail-seat load history for further analysis of the sleepers.

To obtain the detailed dynamic response of the individual sleepers, a larger finite element analysis model was created for a single sleeper unit, supported on ballast and subgrade. Analysis was undertaken using the rail-seat load history, calculated as the sum of the reaction forces obtained in the nodes of the larger finite element track model. A parametric study was undertaken to assess the influence of different track parameters on the dynamic behaviour of the sleepers. Kumaran et al (2003) noted that based on their study an equivalent static model may be proposed to form a rational basis for improved design recommendations of prestressed concrete sleepers.
3.4.6 Japan

Railway Technical Research Institute

Ishida et al. (1996, 1997a & 1997b) presented extensive research on the dynamic effects of high-speed trains on track in Japan. The research included the influence of track stiffness, loose sleepers and bending fatigue of rail welds on track dynamics, and modelling of track irregularity growth and wheel flats using track dynamic analysis.

Ishida et al. (1997a) and Ishida & Ban (2001) presented a detailed track model with a very simple vehicle model represented by a single unsprung wheel mass with primary suspension (Figure 3.13). This model was specifically used for analysis of wheel flats and track stiffness where the authors commented that the frequency of the impact phenomena excited by wheel flats was so high that the mass of the vehicle above the primary suspensions, car body and so on, had almost no influence.

Ishida et al. (1999) and Ono et al. (2001) presented the same track model shown above, but introduced a more detailed vehicle model (see Figure 3.14). The reason for including a full bogie and half-car body was its suitability for the analysis of longer wavelength track irregularities.

The track dynamic model uses a continuous Timoshenko beam supported discretely at the fastening points where the sleepers, ballast and subgrade are elastically joined.
The ballast is divided up into three layers: upper, middle and lower ballast; a full explanation of the theory is available in Miura (1993) “Analysis of ballast vibration by layered model” (in Japanese).

Knothe & Grassie (1993) commented that the attraction of a model that included multiple layers representing the ballast and subgrade (such as that in Figure 3.14), was that it offered the possibility of obtaining better correlation between calculated and measured responses. In practice however, the insensitivity of the response to the many additional parameters would make it difficult to obtain satisfactory parameter values from experimental data.

Miura & Ishida (1991) stated that a simple model such as Figure 3.13 was adequate to analyse dynamic forces between rail and wheel vibrations and forces caused by trains running on track irregularities whose wavelengths were relatively short. The authors note that it was verified that the masses of car body and bogie, and the stiffness of the primary suspension did not significantly influence the dynamic behaviour of one wheel by another in a wheelset. Knothe & Grassie (1994) also support this theory that a single unsprung mass had enough precision to predict the phenomenon of track dynamic behaviour. However due to the recent increases in computer power, higher levels of model sophistication can now be used with relative ease, thus comprehensive model may allow the prediction of more realistic dynamic forces.
3.4.7 Netherlands

Delft University of Technology

Esveld and Kok (1998) presented a model that represented the entire train (5 passenger carriages). This model was developed at Delft University of Technology to assess dynamic behaviour of railway track due to moving railway vehicles (Oostermeijer & Kok, 2000). The research version of the time-domain model was named RAIL, however the commercial version of this software named *Dynamic Analysis of a Rail Track Structure* (Esveld, 2003) was available for EU$10,000 at the time of writing. De Man (2002) also presented software called *DynaTrack* that utilises RAIL and other numerical and analytical models developed at Delft University of Technology.

The RAIL model integrated the vehicle and track in the vertical direction through a time-integration solving method. A train was modelled consisting of five passenger coaches. These were modelled as rigid bodies pivoted at the bogies by secondary suspension represented by springs and dampers. The bogies and wheelset were also modelled as rigid masses connected via primary suspension represented by springs and dampers. The contact with the rail was modelled by a non-linear Hertz spring following Grassie (1984a).
The track model shown in Figure 3.15 was a numerical finite element model that allowed the simulation of various track structures (shown in Figure 3.16) including conventional ballasted track and a non-conventional embedded rail structure (ERS).

The ballasted track structure model comprised of the rail, rail pad, sleepers, ballasted bed and subgrade. The model allows discrete support of the Euler or Timoshenko rail beams by rigid sleeper mass including a rail pad stiffness and damping element. The softer ballast layer incorporated a vertical and longitudinal stiffness and damping to represent the continuity of the ballast layer. The contact between the rail/sleeper and ballast/subgrade elements could be released and thereby reflect reality. This criterion was represented by allowing no tensile forces.

The ballast-less track structure (embedded rail structure) was supported by a rigid concrete slab or a flexible concrete slab supported discretely by piles. Loading was assumed to be symmetric with only one rail being modelled.

De Man (2002) commented that a Fortran program formed the heart of the RAIL software. Pre and post-processing programs allowed the input and output accessible in computer systems like Microsoft Windows. The commercial software (Esveld, 2002) also included a library of European track and vehicle properties. The model allowed rail surface geometry input and purports to allow track incident management by modifying properties at an element level.

Esveld (2004) claimed that the commercial version of RAIL was the only model that allows the interaction between track and complete train set running at a particular
speed. The writer feels the model appears to be reasonably comprehensive, however it did not consider the sleeper as a flexible element which would be useful in the rigorous design of sleepers. There was also limited evidence in literature of the validation of the RAIL software. It appeared that the software was best suited to rigid track structures such as ERS.

3.4.8 Sweden

Chalmers Railway Mechanics (CHARMEC – Chalmers University)

CHARMEC is a centre of excellence in Railway Mechanics established at Chalmers University of Technology. CHARMEC has many industrial partners that fund ongoing research, including Abetong Teknik a very large concrete sleeper manufacturer. Research has been conducted in a number of areas including interaction of train and track.

Many mathematical models to simulate the vertical vehicle-track dynamic interaction have been developed at CHARMEC. Nielsen (1993) developed the original version of the most noted track dynamics model, DIFF. The program has been continuously improved by others at Chalmers University. The model was very similar to DARTS developed in Canada (Cai, 1994). The original DIFF, however, only included one wheel and a two layer discretely sleepered track. DIFF showed improved attention to the sleepers by allowing differing ballast-subballast stiffness and damping properties at the centre and ends of the sleepers as shown in Figure 3.17. Several parameters including the rail, the pads, the sleepers and the ballast were varied during research to minimise the maximum bending moments of the rail and the sleeper.

Nielsen & Igeland (1995) expanded on the DIFF model to investigate track-vehicle interaction including corrugations, wheel flats and unsupported sleepers. The model consisted of a single bogie on a discretely support track.
The bogie model (shown in Figure 3.18) comprised a six DoF system including a bogie side frame, primary suspension and two unsprung wheel masses. The loading was assumed symmetrical over the two rails, thus a half bogie was sufficient. Only vibrations in the vertical plane had been studied. The authors noted that as only the dynamic response of the track was of interest, a lumped model of the vehicle was adequate and the inertia of the sprung masses above the secondary suspension could be neglected. The wheel-rail interface was represented through a non-linear Hertzian contact spring that can handle loss of contact between the wheel and rail.

The track was represented using a linear finite element model that included one rail supported by rail pads and sleepers on ballast, symmetrical about the centreline. The rail and sleepers were modelled by use of undamped uniform Timoshenko beam elements with bending stiffness, shearing stiffness, mass and rotary inertia. To account for varying cross-sectional area, the sleepers were modelled as three beam
elements with different beam properties. The sleepers are supported on a viscously damped, massless elastic foundation modelling the ballast. The foundation consisted of distributed non-interacting springs and dampers. The rail pads were modelled as discrete massless spring-damper systems.

Dahlberg (1995) reported full-scale on-track testing to verify the DIFF program’s computational method. Measurements of strains and accelerations in the track were made simultaneously with measurements on the train of wheel-rail contact forces and accelerations. Test data was compared to simulated results from DIFF and showed good agreement.

Oscarsson & Dahlberg (1998) also presented work based on the DIFF model. Their model included a single wheelset comprising a wheel and half axle modelled as a rigid mass. This was due to the main interest of study being the behaviour of track. The finite element track model was varied by assuming rigid rather than flexible sleepers. The authors commented that the flexibility of the sleepers was not important as long as the bending moments in the sleepers were not to be calculated. The model permitted calculation of deflections, velocities, accelerations, and forces in various track components. This enabled the investigation of how parameters such as train speed, axle load, bogie wheelbase, rail corrugations and wheel flats influenced the track and vehicle components.
Oscarsson (2002a, 2002b) used the DIFF program to model the random behaviour of railway track. The author stated that through the use of a stochastic (statistical) track model, the prediction and estimation of the effect of varying track characteristics could be undertaken. It was well known that railway track properties of the rail pads, sleepers, ballast and subgrade showed strong variability. Oscarsson (2002a, 2002b) used a statistical process whereby the mean values and variations of selected track parameters were investigated. To obtain sufficient statistical information from different track structures full-scale field measurements were taken at two sites, complemented by laboratory measurements.

Andersson & Oscarsson (1999) presented work that extended the existing DIFF model to include non-linear track properties and flexible vehicle components. A linear finite element model was used to represent the train and the track modelled as separate non-linear components. The track model accounted for the load dependent behaviour of rail pads and ballast/subgrade.

Nielsen & Oscarsson (2004) used this state-dependent numerical model (without the complex vehicle model) and separated the track properties into linear contributions corresponding to an unloaded track and non-linear contributions that were dependent on the time-variant state of the different track components due to the dynamic loading from a moving train model. The state dependent contributions to the modal loads were determined by the displacement and velocity states of the rail pads and
ballast/subgrade. The method was validated with field measurements and a good agreement between calculated and measured responses was found.

Andersson & Abrahamsson (2002) developed this work further by enabling general three-dimensional motion of the train traversing the track. The method developed is called DIFF3D. A multibody dynamics method was used for the vehicle that allowed for flexible components. The track model accordingly was made three-dimensional in the sense that it allowed for deflections and forces in all three directions at the wheel/rail contact. The model was complex and models the wheel-rail contact in some detail. The model had mainly been used for rail wear studies and the track submodel was undergoing improvements at the time of writing.

Nielsen (2004) commented that the DIFF software had recently converted from the Fortran programming language to the mathematics software MatLab. The input to the program had changed dramatically and was more straightforward; however, no user manual exists. The various versions of the DIFF model appeared to be very comprehensive and there was ample validation of the models to warrant confidence in its outputs.

3.4.9 United Kingdom

British Rail and Cambridge University

Various models of railway track were presented by Grassie et al (1982) including a model with a simple elastic support, a continuous two layer support and a discrete support of the rail. Findings from this research prompted the widespread use of discretely support track using the Timoshenko beams to represent the rail for modelling track. Grassie & Cox (1984) expanded on this research to include sleepers as beams so that it was possible to calculate bending strains and thus ascertain which track parameters influence sleeper damage. In the model the sleepers were represented as a continuous layer of elemental beams which had significant flexural rigidity along the track length. Both the rails and sleepers were represented by
Timoshenko beams. The model was capable of modelling a single layer of sleepers for non-ballasted track, or a double layer of sleepers and ballast on subgrade for conventional track. The vehicle was modelled by a single wheelset (unsprung mass) with the appropriate fraction of the vehicle’s weight.

**Engineering Solutions Dynamics**

Grassie (1994) presented a software package ‘Track Design’ (referred to through this thesis as TRACK) that aimed to give the railway engineer a tool to calculate dynamic loads on track components. The software utilised the flexible sleeper model presented by Grassie & Cox (1984) and allowed the calculation of loads arising from track with particular irregularities including dipped welds and wheel flats. The primary focus of the TRACK software was to investigate corrugations; however it could investigate several mechanisms of track damage by dynamic loading, including:

- Corrugation by gross plastic flow, in curves and in tangent track;
- Fatigue of the rail in bending;
- Corrugation of the rail by yield in bending; and
- Cracking of concrete sleepers.

Deflections of rail pads and sleepers could also be calculated within TRACK. The software was able to calculate loads on components and to examine whether the specified strength of components were sufficient to avoid damage. As previously mentioned in section 2.5.2 the TRACK software was available to the Australian railway industry. Track design engineers had found it difficult to use TRACK as its input was fairly limited and results hard to interpret.

**Benchmark of Models of Railway Track at High Frequencies**

Two benchmark tests were undertaken by Knothe & Grassie (1995) to review models of railway track at low and high frequencies. The benchmark of high frequency models (Grassie, 1996) included models from the UK, Germany, Netherlands, Sweden and South Africa. The benchmark results included effects of forces, rail...
acceleration and bending moments of both quasi-static and dynamic loads. Several different sleeper and vehicle models were used.

The following findings were made:

- The dynamic contact force, the railhead acceleration and the rail pad force appeared to not be greatly affected by the type of sleeper model.
- The level of detail of the vehicle model also appeared to be relatively insignificant;
- Time domain models with discrete support of the rail at the sleepers gave significantly different results to frequency domain models for the contact force and the rail acceleration at the so called ‘pinned-pinned’ resonance, where the rail vibrates with a semi-wavelength equal to the sleeper spacing;
- To reliably calculate the effects of the pinned-pinned resonance it appeared to be essential to include excitation from a moving mass rather than from a moving irregularity.

Grassie (1996) commented that both types of models (time and frequency domain) had advantages and disadvantages, but as similar results were obtained (except at the ‘pinned-pinned’ resonance) it showed that either could be used with confidence. Grassie (2004) commented that it was important for the user of a model to be careful how results were derived and interpreted as great variations could easily occur.

**Institute of Sound and Vibration Research (University of Southampton)**

Wu & Thompson (1999) investigated track vibration with particular focus on railway rolling stock noise. The authors presented a model of track dynamics based on an infinite discretely supported Timoshenko beam including both a preloaded pad and dynamic stiffness of the ballast. The model used a single wheel load and considered the non-linear properties of the track foundation.

Wu & Thompson’s work formed part of ongoing investigation at The Institute of Sound and Vibration Research (ISVR). The current and recent research concerned a
number of aspects including: the reduction of rolling noise by wheel and track modifications; the modelling of ground vibration and ground-borne noise; noise from steel and concrete railway bridges; noise inside railway vehicles; models of noise from rail joints and wheel flats. The work concentrated on the development of state-of-the-art theoretical models together with the application of generic methods ranging from finite element and boundary element methods to statistical energy analysis.

3.4.10 United States of America

Transportation Technology Centre Incorporated (TTCI) and the Association of American Railroads (AAR)

Blader et al (1989) first presented the computer simulation program NUCARSTM (New and Untried Car Analytic Regime Simulation). The program predicted the dynamic response of a railroad vehicle to specified track conditions. The program was capable of predicting the response of any railway vehicle (locomotive, passenger, transit, or freight car) on any type of track geometry, including special track work such as turnouts and guard rails. Simulation results using the software had been well validated and had gained broad acceptance within the rail industry (Iwinicki, 1999).

Elkins et al (2002) discussed the recent addition of a flexible track model to NUCARSTM allowing analysis of the interaction of specific track components with the vehicle. The authors aimed to model the track with the same level of detail that was available in the vehicle model. Thus it would be necessary to include flexible rails and sleepers, together with the characteristics of the various types of rail fastener and pads. A finite length, flexible beam model had been chosen as a basis for the track model over a variety of other methods. The authors stated that this was a far more efficient method. The other methods investigated involved assembling the rail from a number of rail sub-elements, which were either rigid or flexible. These models had greater complexity and resulted in longer execution time. Elkins et al.
(2002) commented that the method of modelling chosen had been successfully implemented and used for several applications, mostly associated with tangent track. The authors noted that various issues were being addressed at the time and TTCI was continuing to develop the NUCARSTM track model to include:

- Stabilising wheel-rail contact calculations, especially in two-point contact and with rail roll;
- Along track variation and interpolation of rail shapes;
- More test versus model validations; and
- Further development of track receptance testing techniques.

Wilson & Shu (2004) described the NUCARSTM track model as including two flexible rails and 80 flexible sleepers. Each sleeper was connected to the left and right rails with lateral, vertical and roll springs. The ballast beneath each sleeper was represented by 8 vertical springs equally distributed along the sleeper. Both the rails and sleepers were modelled as Euler-Bernoulli beams. The flexible rails had roll, lateral and vertical rigid displacement DoFs. The rail flexible modes included 80 modes each of torsion, lateral bending and vertical bending. The sleepers were allowed vertical displacement and roll (around the longitudinal track centre line) rigid displacement DoFs and flexible vertical bending DoF with 7 modes. The roll mode DoF for the sleeper was required to accommodate simulations of the dipped rail joint and wheel flat implemented only on one side of the rail.

**Manchester Benchmark for Rail Vehicle Simulation**

Manchester Metropolitan University initiated a benchmark in 1997 to allow railway vehicle suspension designers and researchers investigating vehicle dynamic behaviour to assess the suitability of the various software packages (Iwnicki, 1998). The intention was not to provide accurate validation of the software packages, but to compare results for assessing the effect of the various techniques and approximations made.
Participants of the benchmark exercise included the software packages ADMS/Rail, MEDYNA, GENSYS, NUCARS, SIMPACK and VAMPIRE. The overall impression when reviewing the results was that the software packages had a generally good agreement. The benchmark allowed railway vehicle design engineers to gain an insight into the suitability of the different packages for their specific requirements (Iwnicki, 1999).

The treatment of the contact patch elasticity was identified as an area where further work would be required. There was no agreement about the methods used to establish the exact location of the contact patch, its shape and the point at which the tangential forces act. It could be argued that the results of the exercise showed that these variations did not lead to large differences in overall results and were therefore insignificant. However in some cases these small differences may have become very important and therefore warranted further research.

Iwnicki (1999) noted that all participants agreed that the benchmark exercise would be most useful if it was a continuous process with vehicle models and track cases developing as typical modelling requirements changed.
3.5 Summary

The principal function of a railway track dynamic analysis model is to couple the components of the vehicle and track structure to each other so that their complex interaction is properly represented when determining the effect of traffic load on stresses, strains and deformations in the components of the railway system.

The mechanical properties such as the mass, inertia and elasticity of the vehicle and track components form essential input for track behaviour. It is important to understand the mechanical characteristics of main elements of the rail track structure when modelling the railway system. Many of the components in railway tracks are made of material with complicated or even unknown constitutive relationships, for example rail pads or ballast. The behaviour of these components could be load, time or frequency dependent.

Popp et al (1999) states that the general rule for modelling is that, “Models should be as simple as possible and as accurate as necessary regarding the task they serve”. Grassie (1994) commented that it was questionable whether the sophistication of some models was yet necessary in view not only of the relative ignorance which exists about the physical behaviour of essential components such as ballast and rail pads, but also of the variability of typical track.

De Man (2002) comments that in order to combine the properties and dimensions of a structure into models, two modelling methods may be used: analytical and numerical modelling. The latter is in fact a gathering of elementary analytical models. In general, numerical models are complicated and require a large amount of computer power and memory for calculations. Finite element modelling also requires sophisticated software that is often not very user-friendly. For these reasons this research will focus on the use of analytical models for the assessment of the dynamic characteristics of the railway systems.

A review of the various analytical models that represent the railway track structure under the transient loading of a passing train, has been undertaken in this Chapter. A
The following findings were made regarding the capabilities of models that have been developed:

- Models are usually two-dimensional allowing symmetry along the centreline of the track and vehicle;
- Vertical forces are of particular interest for railway dynamic behaviour;
- A single bogie is generally agreed to be adequate for modelling of the vehicle;
- The primary suspension of the bogie is often included in vehicle models, however secondary suspension is typically ignored when examining track dynamics;
- The wheel/rail interface may adequately be represented by the Hertzian contact theory;
- Elements with long dimensions like rail and sleepers are best modelled using the Timoshenko beam theory as it incorporates the shear characteristics of the beam;
- The rail pad component is usually modelled as a spring and dashpot. The rail pad also usually includes the characteristics of the fastening system; and
- Ballast, subballast and subgrade are often modelled as spring and dashpot systems; however some models incorporate a ballast mass and shear connecting springs, or treat the whole substructure as a half space.

Various benchmark tests are available in literature that compare the merits of vehicle and track simulation models. Chapter 4 presents the results of a new benchmark undertaken as part of this research. The benchmark provides an opportunity to compare various model complexities and capabilities. The purpose of the benchmark test is to assist in the selection of a model of track dynamic behaviour for use by the Australian railway industry and research community.
4.1 Introduction

A number of railway track dynamic analysis models were identified in Chapter 3 that could be selected for use by Australian railway track design engineers. In order to assist in the assessment and comparison of the capabilities of these models, a benchmark test was initiated.

As previously discussed in Chapter 3 a benchmark test was undertaken in 1996 on behalf of the International Association of Vehicle System Dynamics (Grassie, 1996) to allow railway track design engineers to see whether the calculation of one model agreed with those calculated by others when the inputs were rigorously stipulated.

Since that time a significant number of models of railway track have been created that allow more in-depth analysis of the railway track, especially concerning its non-linear characteristics. This chapter presents a new benchmark test to compare the present models available throughout the railway research community.

This chapter details the instructions provided for the benchmark simulations, a comparison of participating model capabilities and a discussion of the results.
4.2 Benchmark Test Instructions

Research organisations from Australia, Canada, USA, Sweden, China, Netherlands, Germany, UK and Japan were approached to participate in the new benchmark test for models of railway track dynamic behaviour. The tests were specifically directed at models that could analyse the railway track in the vertical plane and in particular assess the flexural behaviour of sleepers.

The aim of the benchmark test was to compare model capabilities and simulation results using a standard set of parameters. The secondary objective was to provide a forum of discussion and information sharing among researchers developing models of railway track behaviour.

A set of instructions providing theoretical parameters for a vehicle travelling on two types of track structure was prepared by the writer. The document (Appendix C) was supplied to the participants for their comment and suggestions were received from a number of the participants. A second version of the document was then distributed and shortly after an amendment to this version was also distributed to participants to clarify some remaining concerns. The following sections briefly explain the format of the benchmark tests.

4.2.1 Requested Simulations

A passenger coach was chosen to run on two track structures with various contact irregularities. A passenger train scenario was chosen due to its common nature in most railway organisations throughout the world. As most European railways have dedicated high-speed passenger train lines, a scenario of a 160 km/h train travelling on concrete sleepered ballasted track was chosen. In Australia however timber sleepered ballasted tracks are very common for passenger services, thus a scenario of a 100 km/h train travelling on this structure was also chosen. Many of the models presented in literature have been designed specifically for concrete sleepered
ballasted railway tracks. The timber sleepered track scenario allows an opportunity to assess the limitation of a range of parameter values that the models can handle, especially in relation to the pad and sleeper properties.

A number of reasonably severe discrete irregularities in the track and wheel running surface were chosen to examine how the models would handle non-linear reactions throughout the track structure, such as allowance for non-linear ‘lift-off’ at the wheel/rail contact, rail pad/sleeper and sleeper/ballast interfaces.

Six runs in total were chosen comprising a passenger coach, two track structures and three track surface condition descriptions. Table 4.1 describes the six simulations requested from the participants.

Table 4.1 Benchmark Test Simulations

<table>
<thead>
<tr>
<th>Sim No.</th>
<th>Speed</th>
<th>Ballasted Track Structure</th>
<th>Wheel/Rail Contact Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>160 km/h</td>
<td>Concrete Sleepers</td>
<td>No Irregularities</td>
</tr>
<tr>
<td>2</td>
<td>160 km/h</td>
<td>Concrete Sleepers</td>
<td>Worn Wheel Flat</td>
</tr>
<tr>
<td>3</td>
<td>160 km/h</td>
<td>Concrete Sleepers</td>
<td>Dipped Rail Joint/Weld</td>
</tr>
<tr>
<td>4</td>
<td>100 km/h</td>
<td>Timber Sleepers</td>
<td>No Irregularities</td>
</tr>
<tr>
<td>5</td>
<td>100 km/h</td>
<td>Timber Sleepers</td>
<td>Worn Wheel Flat</td>
</tr>
<tr>
<td>6</td>
<td>100 km/h</td>
<td>Timber Sleepers</td>
<td>Dipped Rail Joint/Weld</td>
</tr>
</tbody>
</table>

4.2.2 Vehicle Parameters

For simplicity the passenger vehicle described in the 1998 Manchester Benchmarks (Iwnicki, 1998) discussed in Chapter 3 was used. The vehicle was a general passenger coach with two bogies and a simple primary and secondary suspension.

The vehicle model was symmetric and all bodies were assumed rigid. Details of the suspension characteristics, dimensions of the car body, bogies and wheel profile contact conditions were provided in the benchmark test instructions (Appendix C).
4.2.3 Track Parameters

Two typical track structures were chosen with different rail and sleeper types. Standard gauge (1435 mm) ballasted railway track was chosen that included a subballast (capping layer) over the formation (subgrade). As a European passenger coach was used in the benchmark tests, UIC 60kg/m rail was chosen to eliminate any conflicts at the wheel/rail interface profiles.

![Figure 4.1 Ballasted Track Structure](image)

Figure 4.1 shows the structure of the standard gauge ballasted railway track used in the benchmark tests. Details such as rail and formation cant were ignored for simplicity. Table 4.2 describes the track structures in more detail, further information was provided in the benchmark test instruction document.

<table>
<thead>
<tr>
<th>Component</th>
<th>Concrete Sleepered</th>
<th>Timber Sleepered</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail</td>
<td>UIC 60 kg/m</td>
<td>UIC 54 kg/m</td>
</tr>
<tr>
<td>Fastener</td>
<td>Pandrol ‘e’ 2003 clip</td>
<td>Pandrol ‘e’ 2003 clip</td>
</tr>
<tr>
<td>Rail Pad</td>
<td>7.5 mm HDPE</td>
<td>19 mm Steel sleeper plate</td>
</tr>
<tr>
<td>Sleeper</td>
<td>Concrete (30 tonne axle load rated)</td>
<td>Timber (Hard wood)</td>
</tr>
<tr>
<td>Ballast</td>
<td>250 mm (below the sleeper base)</td>
<td>250 mm (below the sleeper base)</td>
</tr>
<tr>
<td>Subballast</td>
<td>150 mm</td>
<td>150 mm</td>
</tr>
<tr>
<td>Formation</td>
<td>Medium Stiffness</td>
<td>Medium Stiffness</td>
</tr>
</tbody>
</table>

Table 4.2 Concrete and Timber Sleepered Ballasted Track
4.2.4 Wheel and Rail Irregularity Parameters

Three contact surface conditions were chosen for modelling (shown in Table 4.3), including rail that had no irregularities (for base-line assessment of quasi-static forces), a wheel with a worn flat spot, and a dipped rail joint along the rail surface. Participants were instructed that analysis should be undertaken at the point directly above the centre of “Sleeper A”. A detailed description of how each participant modelled the irregularities is presented in Section 4.3.6.

Table 4.3 Irregularity Types for Modelling

<table>
<thead>
<tr>
<th>Irregularity</th>
<th>Description</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Irregularity</td>
<td>No surface irregularities on the wheel or rail (for examination of the quasi-static condition).</td>
<td></td>
</tr>
<tr>
<td>Wheel Flat</td>
<td>The wheel flat should strike directly above the sleeper</td>
<td>a = 50 mm (length of flat area on wheel tread)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>d = 0.3 mm (depth of flat area on wheel tread)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>NB The width of the flat area on the wheel tread is assumed to be the width of the wheel/rail contact area</td>
</tr>
<tr>
<td>Dipped Rail Joint</td>
<td>Dip located in centre of sleeper bay</td>
<td>L = 1000 mm (total length of Irregularity, 500 mm either side of dip)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>d = 3.5 mm (depth of dip)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>NB The width of the dipped weld is assumed to be the width of the rail head</td>
</tr>
</tbody>
</table>
4.2.5 Requested Simulation Outputs

Participants were to calculate the quantities detailed in Table 4.4. A range of outputs from throughout the system were requested so the ability of the models to simulate the various components of the track subsystem could be assessed.

Table 4.4 Quantities to be calculated

<table>
<thead>
<tr>
<th>Code</th>
<th>Output Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Normal contact force between the wheels and rail</td>
<td>kN</td>
</tr>
<tr>
<td>B</td>
<td>Bending moment in the rail immediately above ‘Sleeper A’</td>
<td>kNm</td>
</tr>
<tr>
<td>C</td>
<td>Vertical acceleration of the rail immediately above ‘Sleeper A’</td>
<td>m/s²</td>
</tr>
<tr>
<td>D</td>
<td>Force in rail pad at rail seat on ‘Sleeper A’</td>
<td>kN</td>
</tr>
<tr>
<td>E</td>
<td>Bending moment in ‘Sleeper A’ at rail seat</td>
<td>kNm</td>
</tr>
<tr>
<td>F</td>
<td>Bending moment in ‘Sleeper A’ at centre</td>
<td>kNm</td>
</tr>
<tr>
<td>G</td>
<td>Summation of the force transmitted from the ballast to the sleeper OR The peak pressure at the ballast / sleeper interface</td>
<td>kN</td>
</tr>
</tbody>
</table>

A comparison of the participating model capabilities is detailed in Section 4.3. Results for Simulations 1 to 3 (as per Table 4.1) are presented in Section 4.4, while results for Simulations 4 to 6 are presented in Appendix E.
4.3 Comparison of Benchmark Model Capabilities

A wide variety of analytical track models with simulations undertaken in the time domain are included in the benchmark tests. This section provides a brief description of the participating model capabilities so that simulation results may be interpreted more readily. The historical development of the models presented in this section has been discussed in Chapter 3.

4.3.1 Benchmark Participants

Simulation results were received from six of the original ten invitees as shown in Table 4.5. The models have a range of complexity, from the TRACK model with one wheel on a symmetric railway track to the NUCARS™ model which allowed three-dimensional analysis of the whole vehicle and various layers of the track. Therefore the benchmark provided a good opportunity to compare the merit of having complex or simple models.

Table 4.5 Benchmark Test Participants

<table>
<thead>
<tr>
<th>Model Name</th>
<th>Developed at</th>
</tr>
</thead>
<tbody>
<tr>
<td>DARTS</td>
<td>Queen’s University, Canada</td>
</tr>
<tr>
<td>DIFF</td>
<td>CHARMEC, Sweden</td>
</tr>
<tr>
<td>NUCARS™</td>
<td>AAR, TTCL, USA</td>
</tr>
<tr>
<td>SUBTTI</td>
<td>Technical University of Berlin, Germany</td>
</tr>
<tr>
<td>TRACK*</td>
<td>Cambridge University, UK</td>
</tr>
<tr>
<td>VICT</td>
<td>Southwest Jiaotong University, China</td>
</tr>
</tbody>
</table>

*NOTE: TRACK is the acronym for ‘Track Design v3.4’*

The TRACK model was the only commercially available model of railway track dynamics that participated in the benchmark. The writer ran TRACK on behalf of its authors (Grassie & Saxon, 1995) using the ‘Track Design’ software version 3.4 and the user manual (Grassie, 1994) available. The DARTS model was also run by the
writer on behalf of its author (Cai, 1992) using the program code provided (Cai, 2004) and Absoft Pro Fortran (2003) to compile and run the software (See Chapter 5 for further details).

The NUCARSTM modelling was undertaken by the Transportation Technology Centre Incorporated (TTCI) (Wilson & Xinggao, 2004) using the standard commercially available NUCARSTM software for vehicle modelling, with an experimental track model extension presently under development.

The DIFF, SUBTTI and VICT modelling was undertaken by their respective authors (Nielsen, 2004; Gerstberger, 2004; Zhai, 2004) using various types of simulation and programming language software.

An examination of the capabilities of the various models is available in Appendix B “Comparison Table for Railway Track Dynamic Analysis Models”. All modelling was undertaken in the time-domain, aside from TRACK which undertakes analysis in the frequency domain and then utilises a Fast-Fourier transform to convert results into the time domain.

4.3.2 Interpretation of Benchmark Parameters

Even though a rigorously specified set of parameters was provided to the benchmark participants, each participant interpreted the information in a slightly different way. Due to the range of complexity in the models, parameters were provided for the most complex case. The simpler models were required to lump parameters together and where information was not available for the more complex model, various assumptions were made by the participants. For example parameters were supplied for ballast, subballast and subgrade; some models could handle this level of complexity whereas others represented these layers as a single element.

Due to the release of an initial set of benchmark instructions for discussion and the subsequent release of a second version and an amendment, some participants used
varying parameters. For example, the SUBTTI model used a sleeper spacing for the concrete sleepers of 600 mm whereas the other models used 610 mm as specified in the final benchmark instructions.

Analysis was requested to be undertaken at ‘Sleeper A’ as denoted in Table 4.3. However the actual location of ‘Sleeper A’ in each model was left to the participant to decide. Typically an analysis position should be placed in the centre of a model to eliminate boundary effects in the time domain models. This was the case in all but the NUCARS™ model where the analysis point was located very close to the beginning of the simulated track. An overall length of track to be simulated was not specified, however a minimum length of 20 m was recommended with an analysis time step of 0.1 milli-seconds. Table 4.6 summarises the participant’s decisions regarding the track submodel.

Table 4.6 Participant description of Track Submodel Characteristics

<table>
<thead>
<tr>
<th>Model Name</th>
<th>Simulated Track Length (m)</th>
<th>Location of Sleeper A (m)</th>
<th>Analysis Time Step (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>DARTS</td>
<td>23.18</td>
<td>9.15</td>
<td>0.050 or 0.100</td>
</tr>
<tr>
<td>DIFF</td>
<td>30.5</td>
<td>9.15</td>
<td>0.025</td>
</tr>
<tr>
<td>NUCARS™</td>
<td>32.94</td>
<td>2.44</td>
<td>0.100 or 0.200</td>
</tr>
<tr>
<td>SUBTTI*</td>
<td>18.30</td>
<td>8.40</td>
<td>0.100</td>
</tr>
<tr>
<td>TRACK</td>
<td>44.44</td>
<td>22.22</td>
<td>0.244</td>
</tr>
<tr>
<td>VICT</td>
<td>122.00</td>
<td>61.00</td>
<td>0.100</td>
</tr>
</tbody>
</table>

*SUBTTI has a 0.6 m sleeper spacing for concrete sleepered track

The only other significant difference in interpretation of the benchmark instructions was in the way the models represented the various irregularity types. A discussion of the methods used to describe the irregularities is provided in Section 4.3.6. The location of the irregularities was requested as per Table 4.3, however for worn wheel flat irregularity the location varied amongst models. The SUBTTI and VICT model placed the trailing edge of the worn wheel flat on the centreline of ‘Sleeper A’, however the other models placed the centre of the worn wheel flat over the centreline of Sleeper A as described in the benchmark instructions. This discrepancy can clearly
be seen in Figure 4.14 (b) of Section 4.4.3 where a discussion of the outcome of this difference is provided.

4.3.3 Vehicle Submodels

The TRACK model represented the vehicle as a two mass system (‘dynamic wheel mass’ and ‘dynamic bogie mass’ ) separated by a primary suspension element.

The DARTS, DIFF and SUBTTI models represented the vehicle by a single bogie with two wheel masses and a sideframe mass, with each wheel separated by a primary suspension element. No secondary suspension elements were used in any of these models. All three models assumed symmetry of loading about the track centreline.

The NUCARS\textsuperscript{TM} and VICT models represented the vehicle by using a carbody with two bogies, including primary and secondary suspension. The models assumed symmetry of loading about the track centre line however NUCARS\textsuperscript{TM} could allow full three-dimensional loading, where VICT only allowed symmetric loading.

All elements in the vehicle submodels were represented as rigid bodies with suspension springs being linear in nature.

4.3.4 Wheel/Rail Interface Submodels

The TRACK model undertook calculation in the frequency domain and was therefore incapable of simulating non-linear conditions. Thus TRACK did not allow ‘lift-off’ of the wheel from the rail and assumed a linear Hertzian contact between the two components.

The DARTS, DIFF, SUBTTI and VICT models undertook calculation in the time domain and allowed non-linear Hertzian contact conditions. The lift-off of the wheel
from the rail was allowed by the use of a compressive stiffness for each wheel-rail contact and no tensile stiffness according to the Hertz theory (Hertz, 1887). The wheel/rail contact in these models was represented by a single point and no filtering was undertaken at the contact to allow for consideration of the actual shape of the contact patch (see Section 3.2.6 for a description of filtering).

NUCARS™ used a ‘Real Time Wheel Rail’ contact model developed in-house at TTCI. The actual wheel/rail contact geometry was computed at each integration time step continuously during the simulation. The local deformation at the contact point of rail and wheel was taken into account for the contact geometry and contact forces.

### 4.3.5 Track Submodels

DARTS, DIFF, NUCARS™ and TRACK represented the track with a two-layer mass/elastic stiffness model including the rail and sleepers as the masses, and the rail pad and ballast elements with elastic stiffness. The SUBTTI also used a two-layer model however a half-space was used to represent the subgrade layer.

The VICT model used a three-layer model by incorporating a ballast mass. The ballast mass blocks were interconnected longitudinally so that vertical deflection of one ballast block would spread via shear stiffness and damping to neighbouring blocks (see Section 3.3.3).

DARTS, DIFF, NUCARS™ and TRACK allowed simulation of both flexible rails and flexible sleepers; SUBTTI and VICT allowed only flexible rails with rigid sleepers. DARTS, DIFF, SUBTTI and TRACK all used Timoshenko beams to represent the rail; NUCARS™ and VICT used Euler beams. DARTS, DIFF and TRACK also used Timoshenko beams to model sleepers; however NUCARS™ used Euler beams for the sleepers.

The SUBTTI and DIFF models represented their rails and sleepers using a series of finite beam elements. The DIFF model used eight beam elements for the rail per
sleeper bay and three beam elements for each half sleeper. The SUBTTI model used one beam element per sleeper bay and one element for the whole sleeper. SUBTTI was the only model that allowed sleeper lift-off from the ballast.

The SUBTTI model also used a ‘Ring’ type mode for simulation of the rail end boundary conditions. This allowed the model to represent a continuous loop simulating an infinite length of track. The other time domain models assumed fixed end boundary conditions.

### 4.3.6 Methods used for Irregularity Modelling

Although the irregularities required for the benchmark tests were detailed, each participant represented the irregularities in a different way. A comparison of the equations used for modelling the worn wheel flat and the dipped rail joint or weld is shown in Appendix D. The actual profiles of the irregularities used in the models are shown in Figures 4.2 (a) and (b) and Figure 4.3.

**Worn Wheel Flat**

The worn wheel flat was represented in most models as a track irregularity, shown in Figure 4.2 (a), rather than an actual variation in the wheel profile. The DARTS, DIFF, TRACK and VICT models all used a cosine function to represent the wheel flat as a variation along the track surface. This is the most common representation of wheel flats found in literature and was previously discussed in Section 2.3.3. The SUBTTI model represented a deeper flat with relatively sharp leading and trailing edges that are not typically representative of a worn flat.

The NUCARS™ model represented the flat as an actual variation in the wheel profile as shown in Figure 4.2 (b).
**Figure 4.2 Comparison of the Worn Wheel Flat Profiles used for Modelling**

The solid profile lines in Figure 4.2 represent the actual modelling methods used to describe the profile of the worn wheel flat. The dashed lines allow a comparison of the profiles for representation in the track or as a variation of the wheel shape.

It can be seen in Figure 4.2 (b) that an allowance has been made in most of the models for the effect of wear and metal flow at the leading and trailing edges of the

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*NUCARS profile is modelled as an actual variation of the wheel radius. All other models represent the wheel flat as a track irregularity.*

*NUCARS profile is modelled as a variation of the wheel radius not as a track irregularity as shown here.*
flat. Wilson and Shu (2004) commented that the wheel flat shape also affects the longitudinal point of contact along the track. Correctly simulating the change in longitudinal point of contact has a significant effect on the simulated impact force (Tunna, 1988). All participating models assume the longitudinal wheel/rail contact position is not altered by the change in wheel radius, and that point of contact between the wheel and rail remains below the axle centre line.

**Dipped Rail Joint**

The track dip irregularity has been typically represented in literature as a cusp shape described by a total joint angle (see Table 4.3 and Section 2.3.3). Railway industry in Australia however defines a dip in the rail by its depth over a one metre length when measured by a straight edge (see Section 2.4.1) Figure 4.3 shows the distinct variations in how the track dip was represented in each of the models by the benchmark participants.

*The TRACK model profile shown is only for comparison of the change of angle in the dip.*

**Figure 4.3 Comparison of Dipped Rail Joint or Weld Profiles used for Modelling**
The DARTS and TRACK models used the classical cusp representation of a dipped joint (Jenkins, 1974) with a cosine function to describe the irregularity. The TRACK model however did not describe the joint over a one metre length. As the TRACK model is a frequency domain model, the irregularity cannot be placed at a specific point along the track. Rather, the dip is described over the entire length of the model with the description of the change in angle at the dip being the most important characteristic. Figure 4.3 indicates that the joint angle for DARTS and TRACK is the same (14 milli-radians). The severity of the TRACK joint angle change is correct, however the gradual transition into the cusp dip creates a different overall force history to the other models as it lacks the significant change in shape over the specified one metre length of the irregularity. The shape of the transition into the dip also greatly affects the tendency for wheel lift-off which the TRACK model is unable to represent.

The NUCARS™ profile appears to be similar to the DARTS and TRACK profiles; however, a more complex exponential function was used to describe the cusp shape (see Appendix D). This created a higher change in joint angle than that described by DARTS and TRACK.

The VICT model represented the track dip as a shape that would typically be used to describe shelling of the rail. This shape was also used in the benchmark to describe the worn wheel flat condition in the rail surface profile. It differs significantly from the cusp shape that is typically used, as there is no change in angle at the centre point of the dip.

The DIFF and SUBTTI models used the specified dipped weld description for the irregularity profile. This profile has very distinct start, mid and end points that are significantly different to the flowing cusp shape used by the other models. Even though the DIFF and SUBTTI models used the same length and depth irregularity as the other models, the change in angle at the dip centre was significantly less due to the straight line description of the dipped weld approach and exit paths.
4.4 Evaluation of Benchmark Test Results

This section presents an evaluation of the benchmark outcomes by comparing the simulation results for the different forces experienced by the track structure. Evaluation of the models is undertaken with regard to selection and validation of an appropriate model for the Australian railway community.

Participants provided a varying number of output parameters depending on the capabilities of their model. Table 4.8 details the output parameters provided by each model for Simulations 1, 2 and 3 (described in Table 4.1).

Table 4.8 Output Parameters provided by Participants for Simulations 1 to 3

<table>
<thead>
<tr>
<th>Model Name</th>
<th>Output Parameter (see Table 4.5)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
</tr>
<tr>
<td>DARTS</td>
<td>✓</td>
</tr>
<tr>
<td>DIFF</td>
<td>✓</td>
</tr>
<tr>
<td>NUCARS™</td>
<td>✓</td>
</tr>
<tr>
<td>SUBTTI</td>
<td>✓</td>
</tr>
<tr>
<td>TRACK</td>
<td>✓+</td>
</tr>
<tr>
<td>VICT</td>
<td>✓+</td>
</tr>
</tbody>
</table>

✓ Simulations for these outputs were provided
+ Simulation 1 ‘No Irregularity’ results were not provided
* The TRACK model calculates a dynamic increment only

The SUBTTI and VICT models represented the sleepers as rigid beams and so were unable to provide an analysis of the sleepers’ bending characteristics. The SUBTTI model also had not implemented a calculation function for rail bending moment; the authors advised that this function was to be added for any future benchmarking. The VICT and TRACK models were unable to provide ballast/sleeper interface forces or pressures; however the VICT author did provide a single value for maximum sleeper ballast pressure. The TRACK model was only capable of providing dynamic forces above the quasi-static force (referred to as the Dynamic Increment), thus a direct comparison to other models was difficult.
The following sections of this chapter discuss results of the benchmark test. An explanation of how the benchmark results are presented is included in Section 4.4.1.

In section 4.4.2 there is a discussion of the results of Simulation 1 ‘No Irregularity’ for concrete sleepered ballasted track. The quasi-static forces produced by each model can be assessed through this ‘No Irregularity’ condition, enabling the basic workings of the models to be evaluated and compared.

Simulation 2 ‘Worn Wheel Flat’ results for concrete sleepered ballasted track are discussed in Section 4.4.3. This irregularity scenario allows evaluation of the models with regard to a repetitive high impact force. Wheel flats are often overlooked during design as discussed in Chapter 2. It is appropriate to compare the models with regard to this type of irregularity as a rigorous analysis of track and loading scenarios (for definition of limit state conditions) would include this type of irregularity.

Simulation 3 ‘Dipped Rail Joint’ results for concrete sleepered ballasted track are discussed in Section 4.4.4. This scenario allows evaluation of the models in relation to the classical impact scenario of a dipped joint as discussed in Chapter 2.

### 4.4.1 Understanding the Benchmark Results

The benchmark results are organised using the simulation numbers and output parameter codes described in Table 4.1 and Table 4.5. For example the results for Simulation 2 (passenger vehicle travelling at 160 km/h on a concrete sleepered ballasted track with a worn wheel flat irregularity) and output parameter D (rail seat force) would be termed ‘2D’. This coding system is used for all results presented in this thesis. There are also three distinct types of analysis used to present the results, described below as Types 1, 2 and 3.
Type 1 (Moving Point Analysis)

This type of analysis is used specifically for monitoring the moving contact point between the wheel and the rail or Output Parameter ‘A’ (see Table 4.5). ‘Type 1’ moving point analyses are presented in the following sections as graphs of distance (metres) versus output parameter “A”. Figure 4.4 shows an example of the ‘Type 1’ analysis for the dipped rail joint simulation. The dashed vertical grid lines represent the position of sleepers along the track in relation to ‘Sleeper A’.

![Graph showing wheel/rail contact force for bogie leading wheel](image)

**Figure 4.4 Example comparison of a ‘Type 1’ Moving Point Analysis**

The track dip is located in the sleeper bay prior to ‘Sleeper A’ as per Table 4.4. The wheel/rail contact force shown in Figure 4.4 can be described as follows: the contact force drops from the quasi-static contact force to zero upon entering the track dip; a peak force then occurs as the leading bogie wheel makes contact with the far side of the track dip; the wheel then proceeds to fly and bounce in the third and fifth sleeper bays from ‘Sleeper A’; the wheel/rail contact force eventually returns to the quasi-static contact force some ten sleeper bays after ‘Sleeper A’.
Type 2 (Fixed Point Analysis)

This type of analysis is used to describe the response in time at a single point on the track structure. For this benchmark the point was designated to be in line with ‘Sleeper A’. Output Parameters ‘B’, ‘C’, ‘D’, ‘E’, ‘F’ and ‘G’ (as defined in Table 4.5) are all ‘Type 2’ fixed point analyses and are presented in the following sections shown as time (milli-seconds) versus output parameters graphs. Figure 4.5 shows an example of the ‘Type 2’ analysis for the dipped rail joint simulation. The time marks are directly related to the point in time when a wheel passes over a sleeper.

![Figure 4.5 Example comparison of a ‘Type 2’ Fixed Point Analysis](image)

The force/time history shown in Figure 4.5 only applies to the single point in line with ‘Sleeper A’. The zero mark on the horizontal axis represents the point in time where the wheel passes a point located two sleepers before ‘Sleeper A’. The rail pad force shown in Figure 4.5 represents the response with time of the force in the rail pad on ‘Sleeper A’ and can be described as follows: on approach of the leading wheel to ‘Sleeper A’ the quasi-static force in the pad increases steadily; when the leading wheel enters the track dip the force in the pad falls to zero; when the wheel contacts the far side of the track dip a peak force occurs; this force falls to zero as the
leading wheel flies and bounces; the pad force is then seen to climb steadily again representing the approach to ‘Sleeper A’ of the trailing wheel in the bogie set.

**Type 3 (Fixed Point Analysis – Dynamic Increment Only)**

This type of analysis portrays the ‘Dynamic Increment’ which is the difference between the total dynamic wheel/rail force and the quasi-static force. Output Parameters ‘B’, ‘D’, ‘E’ and ‘F’ may also be shown as ‘Type 3’ fixed point analyses. ‘Type 3’ comparisons presented in the following sections are shown as time (milliseconds) versus output parameters graphs. Figure 4.6 shows an example of the ‘Type 3’ analysis for the dipped track simulation.

![Figure 4.6 Example comparison of a ‘Type 3’ Fixed Point Analysis](image-url)

*Figure 4.6 Example comparison of a ‘Type 3’ Fixed Point Analysis*
4.4.2 Quasi-Static Forces when there are No Irregularities

Wheel/Rail Contact Force for Bogie Leading Wheel

When the wheel and rail have no irregularities, the quasi-static forces produced by each model may be compared for the vehicle travelling at speed. The average wheel force for DIFF, SUBTTI and NUCARS™ shown in Figure 4.7(a) is 54.5 kN which is equal to the calculated static wheel load produced by the benchmark passenger vehicle. The vehicle in motion produces a slightly varying force as the wheel passes over the sleepers in the track. The system is stiffer when the wheel is over a sleeper than when between sleepers.

![1A - Wheel/Rail Contact Force for Bogie Leading Wheel](image)

In Figure 4.7 (a), the DARTS model calculated an average quasi-static wheel force of 42 kN significantly lower than the other models. DARTS author (Cai, 2004) was aware of the quasi-static force calculation inaccuracy and commented that it should not affect the calculation of the more significant dynamic wheel/rail forces. At the time of writing the DARTS author was addressing this problem, further details are provided in Chapter 5.
The VICT model is not included in Figure 4.7(a) because the author of VICT did not provide results for the ‘No Irregularity’ simulation due to time constraints.

The TRACK model is also not included in Figure 4.7(a) because the software does not calculate a quasi-static force as it is a frequency domain model. Therefore only the dynamic increment results are available to the user of TRACK, allowing assessment of the dynamic response of the track. The user is required to add a quasi-static force to the dynamic increment results to find the total wheel/rail force. To determine the quasi-static value of the wheel/rail contact force, the user simply adopts the static wheel load; however for other forces in the system such as rail pad force the quasi-static value is difficult to calculate accurately.

![1A - Wheel/Rail Contact Force for Bogie Leading Wheel](image)

*Distance marks = Position of sleepers*

**Figure 4.7 (b); 1A - Wheel/Rail Contact Force for the Bogie Leading Wheel (Type 1)**

(Vertical scale zoomed)

Figure 4.7 (b) provides a closer examination of the quasi-static forces from Figure 4.7(a). It can be seen that the NUCARS™ model is out of phase with the other models at the beginning of the simulation but slowly makes its way into phase further through the simulation. This would be a result of the system settling due to the initial boundary conditions. NUCARS™ authors (Wilson & Xinggao, 2004) commented that the variation of the quasi-static force was negligible in comparison with the dynamic forces created from the other irregularities and that the purpose of
the ‘1A’ analysis was to check that the average quasi-static force was correct. The writer suspects that the resulting dynamic forces that NUCARSTM calculated for other irregularities may be affected by this out of phase quasi-static force. It appears to make more sense to locate the analysis point of the system away from the ends of the simulated track where the boundary conditions can affect the results.

**Rail Bending Moment at Sleeper A**

The bending moment plot in Figure 4.8 shows a typical response of rail to an approaching bogie. When the leading wheel is still four sleeper bays away from ‘Sleeper A’ (at time 0ms), the rail at ‘Sleeper A’ initially develops an upward curvature (hogging) and then as the wheel passes over the analysis point the sleeper curvature changes downwards (sagging). The second set of rail bending moments (peaking at time 109.8 ms) would be due to the trailing wheel of the bogie set.

![Graph showing rail bending moment at Sleeper A](image)

*Figure 4.8; 1B - Rail Bending Moment at Sleeper A (Type 2)*

The three models show certain similarities in the shapes of the rail bending moment distribution. However, DIFF and DARTS differ in the magnitude (9 kNm and 4 kNm) of peak sagging moment as the wheel passes directly over ‘Sleeper A’, and the NUCARSTM model shows smaller and oddly changing hogging moments than the
other two where the wheel is between sleepers. This difference in NUCARSTM moments could be a result of ‘Sleeper A’ being relatively close to the starting point of the mode simulation and its related boundary conditions. However, the low values of hogging moment (1 kNm compared to 4 kNm) in the rail may also reflect a track structure that is stiffer or heavier than the other models. Furthermore, the DIFF and DARTS models used Timoshenko beams for the rail, whereas NUCARSTM used an Euler beam that does not take into account the shear characteristics of the rail.

The DIFF and NUCARSTM models show sharply peaked sagging bending moments in the rail as the wheels pass directly over ‘Sleeper A’ in Figure 4.8, whereas the DARTS model shows a lower more rounded peak. Sleepers in reality provide distributed support to the rail rather than a point support and so one would expect a rounded peak such as DARTS produced. DARTS may therefore be modelling reality more closely than the other two models.

Rail Acceleration at Sleeper A

Figure 4.9 shows rail accelerations (output parameter ‘C’) for four of the models. Peaked accelerations can be seen in Figure 4.9 for each model as the wheel passes over each sleeper location (ie. at times 13.725ms, 27.450ms, etc). The largest peaks are observed as the leading and trailing wheels pass over ‘Sleeper A’ as would be expected.

The regular peaks displayed by the SUBTTI and DIFF models relate to the Timoshenko beam finite elements used to describe the rail (Gerstberger, 2004). The SUBTTI model used one beam element per sleeper bay, thus significant peak accelerations are shown at each sleeper. By contrast, the DIFF model used eight beam elements per sleeper bay, thus a higher number of significant peak accelerations are shown.
The effect of the boundary conditions on the NUCARS™ can be seen in its burst of rapid close spaced peaks shortly after the zero time mark in Figure 4.9. The boundary condition causes a significant instability in the model and this effect does not settle out for some distance.
Rail Pad Force on Sleeper A

The rail pad forces in Figure 4.10 produced by the leading and trailing wheels from the bogie set are approximately half that of the wheel/rail contact force shown in Figure 4.7 (a), due to the attenuation by the rail mass and stiffness and by the rail pad.

The DIFF and SUBTTI force plots in Figure 4.10 show the rail pad force beginning to appear when the wheels are up to three or four sleeper bays from ‘Sleeper A’, in correspondence with the initial appearance of rail moments shown in Figure 4.8. However the NUCARS™ model displays a response starting abruptly from only two sleeper bays from ‘Sleeper A’. This could be due to NUCARS™ having a stiffer track structure than the other models. A stiffer or heavier track structure would be less likely to react to approaching wheels.

![1D - Rail Pad Force on Sleeper A](image)

Figure 4.10; 1D - Rail Pad Force on Sleeper A (Type 2)

A notable characteristic of the DARTS, DIFF and SUBTTI models is the distinct drop in force of around 2 kN, as the wheels pass directly over ‘Sleeper A’. This is an unusual response and could be due to the discrete sleeper spacing and the fact that
the models treat the sleepers as individual points supporting the rail rather than as elements with a defined width. The smooth response of NUCARS may indicate a more continuous support of the rail in that model.

The DARTS model shows a drop of 5 kN in rail seat force as the wheels pass over ‘Sleeper A’. Cai (1992) commented that the DARTS rail seat force exhibited a drop when the wheel was immediately above the sleeper, while the ballast force reached a high peak. This ballast force peak can be seen later in this Section in Figure 4.13. Cai (1992) noted that the drop in rail seat force was due to a faster vertical movement of the sleeper, as it was being excited by the oncoming wheel, than that of the rail above the sleeper. The faster downward moving sleeper offsets the compressive rail seat force and increased the ballast force.

The DARTS response of 12 kN at ‘Sleeper A’ is however, up to 50 % less than the other models, the response could be due to a problem with the way that DARTS handles the stiffness and damping properties of the rail pad. As shown later in Section 4.5.1, the difference between DARTS and the other models is even more evident in the results for the timber sleepered track simulations that had low sleeper mass and steel sleeper plates of very high stiffness, reinforcing the idea of there being a problem in DARTS.

**Bending Moment in Sleeper A at the Rail Seat**

Figure 4.11 shows the varying complexity by which sleepers are represented in each of the models. This complexity may be related to the number of bending modes modelled and also to number of discrete elements forming the sleeper representation, as described below.

The NUCARSTM model assumes one Euler beam element with consistent depth along the sleeper, so the response shown in Figure 4.11 is of a relatively simple shape, with sagging accompanied by a return to zero moment between the first and second wheel passes.
The DIFF model sleepers are represented as 6 finite Timoshenko beam elements that can be varied in depth; in combination with the rail finite elements, this could be a reason for the ‘noise’ apparent in the sleeper’s bending moment history in Figure 4.11. The drop in rail pad force after each wheel pass is reflected in the small drop in sleeper bending moment for DIFF at those points in the graph.

The DARTS model appears to provide a more complex sleeper response with a greater number of bending modes. The sleeper is modelled as three Timoshenko beam elements that can be varied in depth. The analysis point (rail seat) reacts to an approaching wheel up to five sleeper bays from ‘Sleeper A’ in Figure 4.11. The sleeper hogs slightly then sags at the rail seat, then partly returns to its normal state before sagging more dramatically at the rail seat as a wheel passes over ‘Sleeper A’. The response continues as the leading wheel passes and the trailing wheel approaches. This more complex response of the sleeper could indicate a more realistic representation of the sleeper’s behaviour. The effect of the drop in DARTS’ rail pad force is less apparent in the sleeper moment plot in Figure 4.11.
Bending Moment in Sleeper A at the Centre

The models show a similar but opposite responses for the quasi-static bending moment at the centre of ‘Sleeper A’. Figure 4.12 shows the centre of ‘Sleeper A’ experiencing hogging moments at the same times as Figure 4.10 shows sagging moments in the sleeper at the rail seat.

The intensity of the sleeper centre hogging moment in Figure 4.12 is approximately equal to that of the sleeper rail seat sagging moment for DARTS and DIFF (2.5 kNm). The NUCARSTM model however shows a much lower sleeper centre hogging moment (1 kNm), being less than half that of the sleeper rail seat sagging moment (3.8 kNm). In static situations, moments at the location of a point force (ie. at the rail seats) would be expected to be larger than any opposite moment between these points (ie. at the sleeper centre). It’s uncertain however, that NUCARSTM is better representing the situation, because under the rapid dynamic loading of the sleeper occurring in these simulations, modes of vibration of the sleeper can produce moment distributions quite different from static cases.

![Graph showing bending moment in Sleeper A at the Centre](image)

*Figure 4.12; 1F - Bending Moment in Sleeper A at the Centre (Type 2)*

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As with sleeper rail seat moments, the DARTS model again shows a complex bending response with hogging of the sleeper centre at and between wheel passes over ‘Sleeper A’. DARTS also shows small sagging moments when the wheels are around four sleeper bays away from ‘Sleeper A’, indicating perhaps a more flexible structure than the DIFF and NUCARS™ models. This type of bending response of the track could be seen to be more realistic especially for vehicles travelling at high speeds.

**Total Ballast Force on Sleeper A from Two Rails**

The total ballast force acting on the base of the sleepers shown in Figure 4.13 is approximately equal to twice that at the rail seat for one rail at any point in time, for all the models as one would expect.

![Graph of Total Ballast Force on Sleeper A from Two Rails](image)

*Figure 4.13; 1G - Total Ballast Force on Sleeper A from two Rails (Type 2)*

As mentioned previously the DARTS ballast force response has not been affected by the large drop in the DARTS rail pad force as the wheels pass over ‘Sleeper A’, shown in Figure 4.10. The lower values of peak ballast forces for DARTS in Figure 4.13 is a consequence of the lower value of quasi-static wheel/rail contact force identified earlier in Figure 4.7.
The DIFF and DARTS models also show a negative force pressure as the wheels approach and leave ‘Sleeper A’ (ie at times 0ms and 164.7ms). This of course would not occur in reality. This negative action is a result of the tensile forces developed in the stiffness/damping properties of the sleeper/ballast connection. The SUBTTI and NUCARS™ outputs in Figure 4.13 show no tensile force at the sleeper/ballast interface. However only the SUBTTI model actually allows lift off of the sleeper from the ballast layer, representing a more realistic simulation of sleeper/ballast interface forces.

4.4.3 Dynamic Wheel/Rail Forces from Worn Wheel Flats

A comparison of how the benchmark models assess dynamic wheel/rail forces created by wheel irregularities is important for the selection of a model that can simulate a broad range of conditions. Most discrete irregularities examined in existing track design methods relate to rail profile defects; wheel defects are often overlooked.

All the benchmark models were able to simulate wheel flats. However, it was previously discussed in Section 4.3.6 that most railway track dynamic analysis models found in literature represent wheel defects as rail surface profile defects. Only one model participating in the benchmark test (NUCARS™) represented the worn wheel flat as a variation of the wheel radius.

Wheel/Rail Contact Force for Bogie Leading Wheel

Wheel flats produce very short high impact forces on the track structure that are repeated with the passage of every wheel circumference. These severe impact forces hammer the track structure repeatedly and therefore are of great significance. Figure 4.14 (a) shows that wheel irregularities produce short period, high impact wheel/rail contact forces.
Figure 4.14 (b) allows a closer examination of the wheel/rail contact force responses. All models aside from the TRACK model allow a non-linear contact at the wheel/rail interface, so that no negative forces are created. The TRACK model has a linear connection and therefore a negative force is produced as the wheel is effectively pulled back down onto the rail during a lift-off event, as discussed in Section 4.3.6.

The discrepancy in the point where each model shows initial loss of contact between times 1.18ms and 1.22ms in Figure 4.14 (b) is due to the positioning of the worn wheel flat irregularity in the model. The SUBTTI and VICT model positioned the trailing edge of the wheel flat on the centre of ‘Sleeper A’, whereas the other models placed the centre of the wheel flat over the centre of ‘Sleeper A’. This produces the out of phase effect in the peak contact forces that appear after 1.22ms in Figure 4.14(b).

These peak contact forces in the graph are quite different among the models, varying from 4 times to 16 times the quasi-static force.
**Rail Bending Moment in Sleeper A**

Figure 4.15 (a) shows the bending moment in the rail as the leading wheel experiences impact from its wheel flat. The quasi-static motion of the rail is displayed in the same manner shown in Figure 4.8 as the wheel approaches ‘Sleeper A’; the additional effect of the high frequency impact force due to the wheel flat strike is clear in Figure 4.15(a).

The response shown by the NUCARSTM models in the expanded time scale shown in Figure 4.15 (b) is likely a result of NUCARSTM very high P1 wheel/rail contact force seen earlier in Figure 4.14. It appears that the rail above ‘Sleeper A’ experiences very high stress during the entire response to the impact event. The secondary peak in NUCARSTM rail bending moment plot relates to the second impact after the lift-off motion of the leading wheel. As the NUCARSTM model assumes an Euler beam for the rail, no shear effect in the rail has been considered.
Figure 4.15 (a); 2B - Rail Bending Moment in Sleeper A (Type 2)

Figure 4.15 (b); 2B - Rail Bending Moment in Sleeper A (Type 2)

(Horizontal scale zoomed compared to Figure 4.14 (a))

Figure 4.15 (c) displays only the dynamic increment component in the graphs of Figure 4.14 (b); that is, the quasi-static component has been removed to allow comparison with the TRACK model output. The negative rail bending moment peaks show in 4.15 (c) coincide with the peak contact events shown in Figure 4.14 (b).
As with the wheel/rail contact force plots shown earlier in Figure 4.14 (b), the DARTS and VICT model in Figure 4.15 (b) show responses that are in phase with each other, however have significantly different magnitudes. DARTS wheel/rail contact force is greater than VICT (330 kN compared to 275 kN), however the VICT rail bending moment is more than 50 % greater than DARTS.

The TRACK model maximum positive rail bending moment (10 kNm) is in agreement with the DIFF model. TRACK’s high hogging rail bending moment at around 27ms in Figure 4.15 (c) coincides with the negative wheel/rail contact force at the same time in Figure 4.14 (b). This response should however be ignored as it does not reflect reality and is an effect of the frequency domain model.

**Rail Acceleration at Sleeper A**

The rail accelerations produced by the models are plotted in Figure 4.16 and are typical for the short period, high impact event of a wheel flat. DARTS and VICT are again in general agreement in amplitude and wavelength. The NUCARS™ model has an unusually small acceleration (up to 3 times less) considering the magnitude of
its wheel/rail contact force in Figure 4.14 (up to 8 times greater). The DIFF model also has a significantly different acceleration to the other models.

Figure 4.16 (a); 2C - Rail Acceleration at Sleeper A (Type 2)

* Time marks = Time when wheel passes over a sleeper

Figure 4.16 (b); Rail Acceleration at Sleeper A (Type 2)
Horizontal scale zoomed compared to Figure 4.16 (a)
Rail Pad Force at Sleeper A

The models show a better agreement in their determination of the rail pad forces (30% variation) shown in Figure 4.17, than in the calculation of P1 wheel/rail contact forces (75% variation) in Figure 4.14. This better outcome probably relates to the models’ better abilities to represent the mid range frequencies of forces (known as ‘P2’, see section 2.3.3 for more details) than the higher P1 frequency forces. The P2 forces are transmitted further into the track structure than P1 forces and are the key factor in loads experienced by the sleepers and ballast.

![2D - Rail Pad Force at Sleeper A](image)

*Figure 4.17 (a); 2D - Rail Pad Force at Sleeper A (Type 2)*

The behaviour of the wheel shown in the models’ wheel/rail contact histories in Figure 4.14 is reflected in the rail pad force histories in Figures 4.17 (b) and (c). During lift-off of the wheel most of the models show a negative (tensile) force at the rail pad, which is representative of the fastener experiencing holding down forces. The tensile properties of the fastener however are different to the tensile properties of the pad, thus a single elastic element representing both components is not overly realistic. The pad and fastener probably should be modelled as two separate parallel elements with different properties. NUCARS™ has adopted this approach and represents the stiffness of the rail pad with only compressive properties and the
stiffness of the fastener with only tensile properties. This is the reason the NUCARS™ rail pad force is zero during lift-off of the rail from the sleeper as shown in Figure 4.17 (b).

Figure 4.17 (b); 2D - Rail Pad Force at Sleeper A (Type 2)

Figure 4.17 (c); 2D - Rail Pad Force at Sleeper A (Type 3)
(Vertical and horizontal scale zoomed compared to Figure 4.17 (a))
The TRACK model shows a larger negative (tensile) force than the other models due to the model not allowing lift off of the wheel; this effect diminishes the subsequent positive (compressive) rail pad force in Figure 4.17 (c). There are also two distinct peaks in the TRACK response which could reflect the resonant effect in the system.

**Bending Moment in Sleeper A at the Rail Seat**

In Figure 4.18 (a) all models except NUCARSTM show a repeated hogging and sagging of the rail seat after the wheel impact event, indicating numerous collisions between the rail and sleeper. This is in line with the theory put forward by Tunna, 1988 and supports the description of a P1½ force that occurs between the initial high impact P1 force and secondary lower frequency P2 force.

![Graph: 2E - Bending Moment in Sleeper A at the Rail Seat (Dynamic Increment Only)](image)

*Figure 4.18 (a); 2E - Bending Moment in Sleeper A at the Rail Seat (Type 3)*

Figure 4.18 (b) shows an initial upwards curvature (hogging) of the sleeper for all models. This would occur during the wheel lift-off before the peak contact force is experienced. When the wheel and the rail contact each other and move downwards during the impact event the sleeper would still be in an upward curvature (hogging)
momentarily. The wheel and rail mass combined then collide with the sleeper mass and the sleeper is forced into a downward curvature (sagging) as indicated by the DARTS and TRACK models. The system then recovers and slowly stabilises.

![2E - Bending Moment in Sleeper A at the Railseat (Dynamic Increment Only)](image)

*Time marks = Time when wheel passes over a sleeper

*Figure 4.18 (b): 2E - Bending Moment in Sleeper A at the Rail Seat (Type 3)

*(Horizontal scale zoomed compared to Figure 4.18 (a))*

**Bending Moment in Sleeper A at the Centre**

In discussion about the quasi-static simulation earlier, it was shown how the bending moment at the sleeper centre reacts in the opposite direction to that of the rail seat. When the rail seat is hogging the centre is sagging. It is interesting to note when the combined wheel and rail masses collide with the upwards curvature of the sleeper at the rail seat during a wheel flat impact, the sleeper centre experiences a large hogging moment as shown in the negative peaks in Figure 4.19 (a) and (b). This is particularly apparent in the NUCARSTM model where the magnitude of the negative moment is very large.
2F - Bending Moment in Sleeper A at the Centre (Dynamic Increment Only)

Figure 4.19 (a); 2F - Bending Moment in Sleeper A at the Centre (Type 3)

* Time marks = Time when wheel passes over a sleeper

NUCARS = 28

2F - Bending Moment in Sleeper A at the Centre (Dynamic Increment Only)

Figure 4.19 (b); 2F - Bending Moment in Sleeper A at the Centre (Type 3)

(Horizontal scale zoomed compared to Figure 4.19 (a))

130
Total Ballast Force on Sleeper A from two Rails

Unlike the quasi-static forces, the total ballast forces due to the worn wheel flat irregularity shown in Figure 4.20 (a) and (b) is not twice that of the rail pad force.

Figure 4.20 (a); 2G - Total Ballast Force on Sleeper A from two Rails (Type 2)

Figure 4.20 (b); 2G - Total Ballast Force on Sleeper A from two Rails (Type 2)
The NUCARS™ and SUBTTI models show the peak ballast force (80 kN and 75 kN) from two rails is roughly equal to the rail seat force from one rail. This indicates a significant loss of force from the rail/sleeper to sleeper/ballast interfaces at any one point in time.

The DIFF model however displays a high positive peak ballast force (185 kN) significantly different from the other models and coinciding in time with the rail and sleeper mass collision mentioned previously. This peak force is greater than twice the force at one rail seat, indicating that the P2 force determined by DIFF penetrates more deeply into the system than the other models determined.

The DARTS model plot in Figure 4.20 (b) shows two distinct and similar positive peak ballast forces (90 kN). The second peak could be in relation to the P1½ and P2 force explanation provided by Tunna (1988) (see section 2.3.3). The first ballast force peak coincides with the wheel and rail masses colliding with the sleepers mass and then the ballast. The second ballast peak appears to relate to when the wheel, rail and sleeper masses act together downwards on the ballast mass during the second wheel/rail contact.

4.4.4 Dynamic Wheel/Rail Forces from Dipped Rail Joints

The most commonly specified discrete irregularities accounted for in normal design of track are dipped joints. It is therefore relevant to evaluate the benchmark models’ responses to this type of irregularity. As discussed in Section 4.3.6 the participants interpreted the irregularity scenario in different ways, thus various responses to the defects are apparent in the simulation results.

Wheel/Rail Contact Force for Bogie Leading Wheel

Figure 4.21 (a) shows the output of the models’ wheel/rail contact force for a dipped joint. As with the worn wheel flat irregularity there’s a large range of the magnitudes
of peak wheel/rail contact forces in the diagram, from 4 times to 9 times the quasi-static force. The model force plots shown in the diagram do however agree regarding the general shape of the response of the leading wheel when it negotiates the track dip located in the centre of the sleeper bay before ‘Sleeper A’.

![Diagram of Wheel/Rail Contact Force for Bogie Leading Wheel](image)

*Figure 4.21 (a); 3A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1)*

All the models show the bouncing effect of the leading wheel occurring with a range of four to eight sleeper bays from the dip location. As the TRACK model calculates forces in the frequency domain, it is the only model that shows a linear response at the wheel/rail contact.

Figure 4.21 (b) allows a closer examination of the individual model responses. The models show a significant number of small but rapid force peaks superimposed onto the overall force response to the track dip. This high frequency motion represents the P1 forces in the system.
Figure 4.21 (b): 3A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1)
(Horizontal scale zoomed compared to Figure 4.21 (a))

The magnitudes of the peak contact forces appear to relate to some extent to the angle of the track dip profiles shown in Figure 4.3 (see Section 4.3.6). The NUCARSTM model has the greatest angle change at the joint shown in Figure 4.3 followed by DARTS, then DIFF and SUBTTI models. Although the VICT model did not represent the dip as a cusp shape the approaching slope of the joint is very similar to the DARTS model. This could be a reason for the similar peak force magnitude in Figure 4.21 (b). The TRACK model models the dip as a continuous function over the entire length of the simulated track, thus there is no free-fall motion of the wheel when entering the dip. This could explain the lower peak contact force.

**Rail Bending Moment in Sleeper A**

When the wheel makes contact with the rail after flying free across the cusp of the dip, the rail undergoes downward curvature creating a positive (sagging) bending moment, as shown in Figure 4.22. The DARTS and VICT model agreed on this impact wheel/rail contact force in Figure 4.21, but differ significantly for the rail bending moment in Figure 4.22.
The NUCARSTM model has a significantly different rail bending moment shape with multiple peaks in the above diagrams. This secondary response would most likely be due to the bouncing motion of the leading wheel before the trailing wheel has reached the track dip. The model’s very high moment of -185 kNm and the subsequent response is very unusual and very different from the other models.

![3B - Rail Bending Moment at Sleeper A](image)

*Figure 4.22 (a); 3B - Rail Bending Moment in Sleeper A (Type 2)*

**Rail Acceleration at Sleeper A**

The rail accelerations shown in Figure 4.23 for a dipped rail joint correspond to the trends of each model’s peak wheel/rail contact forces in Figure 4.21 as expected. As with the analysis of wheel flats, it is interesting to note that the NUCARSTM model has surprisingly small accelerations considering the high wheel/rail contact forces the model calculated.
Figure 4.23; 3C - Rail Acceleration at Sleeper A (Type 2)

Rail Pad Force at Sleeper A

The shape of the rail pad force responses shown in Figure 4.24 (a) for a dipped joint are in good agreement with each other. The NUCARSTM model however reports a force (220 kN) up to 2 times higher than the other models, in keeping with its high wheel/rail contact force. The VICT model calculates the lowest force (80 kN) but has the second highest wheel/rail contact force, indicating a greater amount of high frequency P1 forces in the system.

As with the worn wheel flat there are some negative (tension) forces produced by some models. The dynamic increments are plotted in Figure 4.24 (b) and show that the TRACK model produces the lowest rail pad force, as would be expected from its significantly lower wheel/rail contact force.
Figure 4.24 (a); 3D - Rail Pad Force at Sleeper A (Type 2)

Figure 4.24 (b); 3D - Rail Pad Force at Sleeper A (Type 3)
Bending Moment in Sleeper A at the Rail Seat

In the rail seat sleeper bending moments shown in Figure 4.25 (a) for a dipped joint, the sleepers undergo a significant sagging when the wheel strikes the track dip. The DARTS and DIFF models are in general agreement as to the magnitude of this positive rail seat bending moment. It is interesting to note that the rail pad force for DARTS in Figure 4.24 was slightly less than that for DIFF, but the sleeper bending moment at the rail seat is slightly greater. This may be due to DARTS’ more accurate simulation of the sleeper resonances discussed in Section 4.4.2.

![Bending Moment in Sleeper A at the Rail Seat](image)

Figure 4.25 (a); 3E - Bending Moment in Sleeper A at the Rail Seat (Type 2)

The TRACK model calculation of dynamic increments shown in Figure 4.25 (b) agrees with the DARTS and DIFF models. This is interesting considering that the TRACK rail pad forces were half that of the other two models.

The NUCARSTM model again produces results significantly different from the others in Figure 4.25. As with the rail bending moment the magnitude of the peak response is twice that of the other models. When considering the significant validation that the TRACK and DIFF models have undergone against real track data (discussed in Chapter 3) the writer of this research finds the NUCARSTM results questionable and
believes the discrepancy may be the result of the use by NUCARSTM of an Euler beam to describe the sleeper element as discussed previously.

**Figure 4.25 (b); 3E - Bending Moment in Sleeper A at the Rail Seat (Type 3)**

**Bending Moment in Sleeper A at the Centre**

The bending moments at the sleeper centre are shown in Figure 4.26 (a) and once again are opposite in direction to the moments at the rail seat as expected. The DARTS, DIFF and TRACK models are within 30% of each other for the magnitude of the centre bending moment.

The NUCARSTM model however calculates what appears to be a more complex response for the wheel striking the track dip with significant positive and negative moments. The NUCARSTM sleeper model allows longitudinal roll of the sleepers as well as vertical and lateral displacement, so a more complex result may be possible.
Figure 4.26 (a); 3F - Bending Moment in Sleeper A at the Centre (Type 2)

Figure 4.26 (b); 3F - Bending Moment in Sleeper A at the Centre (Type 3)
Total Ballast Force on Sleeper A from two Rails

The total ballast force plots in Figure 4.27 show the most consistent agreement among the models for ballast force calculation.

![Graph of Total Ballast Force on Sleeper A from two Rails](image)

Figure 4.27: 3G - Total Ballast Force on Sleeper A from two Rails (Type 2)

The SUBTTI model provides the only simulation that allows a non-linear sleeper lift-off from the ballast; the other models may simply truncate their outputs when the sleeper/ballast force goes negative. This characteristic of SUBTTI is similar to the NUCARSTM representation of the rail pad/fastener connection. The ballast is only represented by compressive properties and therefore no tension or pulling-down forces are produced.

The NUCARSTM response of ballast force at the point when the wheel lifts off the rail upon entering the track dip is also different to the other models. NUCARSTM indicates that there is still some downward pressure of the sleeper onto the ballast, whilst the other models indicate a zero force during this period. The NUCARSTM rail pad force is zero during the wheel lift-off, thus the motion of the sleeper would be away from the rail and towards the ballast. The response immediately after the initial wheel/rail contact also shows this.
4.5 Summary

The benchmark results presented in this chapter have shown that significantly different results may be obtained by models of track dynamic behaviour, depending on the assumptions taken by the user for a particular track scenario. Even though a substantial set of parameters was provided for each particular vehicle and track simulation, the participants were unable to produce the same results. This is an important finding regarding the how parameters are interpreted in models of track dynamic behaviour.

Some reasons why it would be difficult to obtain the same results from different models with different users could include the:

- level of experience of the user and therefore the detail sought by the user when undertaking a simulation;
- differing modelling methods and complexities;
- number of different input parameters required;
- interpretation of the irregularities in the wheel and rail;
- differences in analysis time step and the frequency range;
- flexible and rigid representation of the sleepers; and
- varying methods (Euler and Timoshenko) used for modelling rails and the sleepers which were found to have significant effects on results.

A summary of the major findings from the benchmark test for each of the participant models is provided below.

The DARTS model was consistently similar to responses provided by the DIFF and TRACK models for the rail seat forces and sleeper bending moments. There are obvious problems with the calculation of the quasi-static wheel/rail contact force; however these issues were being addressed by the models’ author at the time of writing.

The DIFF model provided responses that were consistently near the average of other model responses for most benchmark results. The DIFF results were typically within
15% of the TRACK model, which indicates some similarities in the calculation methods used. The sleeper is modelled by DIFF as 6 beam elements over its length, thus the ability to vary the support characteristics of each beam element is of benefit in representing realistic scenarios.

The NUCARSTM model produced varying results often inconsistent (up to 400% different) to the other models. More often than not NUCARSTM calculated responses were higher in magnitude than the other models. The model showed consistency between its own high results, so it cannot be ruled out as being incorrect. The great complexity of NUCARSTM with regards to its whole car body model and relatively complex track model may have been the underlying cause of the different results. It appears however that the use of an Euler beam to model the rail and sleeper has led to some of the higher results. Investigation of the use of Timoshenko beams for representation of the rail and sleeper would prove interesting. The range of frequencies allowed in the system may also need to be expanded to allow for higher resonances in the track structure, especially in the sleeper.

The SUBTTI model provided results typically with 20% of the other models for the various output parameters submitted. The use of a finite element half space to represent the substructure (ballast and subgrade) has produced results in line with the typical method of representing stiffness and damping properties adopted by the other models. The author of the model indicated to the writer that SUBTTI would be extended to include calculation of sleeper moment, stresses and strains. As the model has shown comparable outputs to the other models, the results for sleeper behaviour would prove useful in any future benchmarking.

The TRACK model although simplistic in its assumptions regarding the wheel/rail contact, did however give results within 25% of the other models. The comparisons made using dynamic increments indicated that TRACK was underestimating the forces experienced at the various component interfaces. Nevertheless, the moments calculated for the rail and sleepers were consistent with DARTS and DIFF indicating a degree of reliability.
The VICT model typically showed agreement to the DARTS model. This could indicate similar broad underlying theories used in each model. However, the more complex method to model the substructure in VICT, by using ballast masses and shear spring connections, appears to be the reason for different sleeper/ballast interface force outputs (shown in Table 4.9 through 4.12, in Section 4.6). Despite this, the complex substructure does not appear to have affected the wheel/rail contact or rail seat force calculations, however it does affect the calculations of the sleeper and ballast vibrations.

This chapter has compared the outputs of the benchmarked models against each other on the basis of a single type of train vehicle (passenger) and single type of track (concrete sleepers). It's clear from this single benchmark scenario that although some broad agreement was achieved between some of the force and moment outputs of the models, there is significant difference between them all in many instances. Chapter 5 compares the models further using the same vehicle travelling on timber sleepered track. A comparison of the models' outputs with parameter values obtained from traditional methods of track design is also included. From the results so far it is recommended that further research should be undertaken to develop additional benchmark scenarios which will allow comparison of the models' outputs against carefully measured real track force/time behaviour.
CHAPTER 5

Selection of a Model of Railway Track Dynamic Behaviour

5.1 Introduction

This chapter extends examination of the Benchmark Tests to explore the models' abilities to mimic the performance of timber sleepered track, to identify limitations of the models when applying them to different types of track structure, and to compare the overall outputs of the benchmarked models against the empirical track design approach widely used in Australia. This latter comparison allows assessment of the degree to which the benchmark model responses may be acceptable to railway track design engineers. Finally a brief review of potential models available for selection is undertaken. A model will be chosen according to the criteria established in Chapter 3.
5.2 Modeling of Timber Sleepers

It was established in the literature review presented in Chapter 3 that most models of railway track dynamic behaviour have typically been designed to simulate concrete sleepered ballasted track or slab track. In Australia a large amount of timber and steel sleepered railway track exists that will most likely never be replaced with new track structures. It is therefore important to assess the limitations of the benchmark models in simulating non-concrete sleepered track, to determine whether a range of track structures can be analysed.

Benchmark Simulations 4, 5 and 6 (see Table 4.1) represented timber sleepered ballasted track. Parameters for the rails, steel sleeper plates, sleepers and speed of the vehicle were changed compared to Simulations 1, 2 and 3. The vehicle, wheel and rail irregularities, rail gauge and ballast condition were not changed. Thus the effect of lighter rail, a stiffer pad element and highly flexible timber sleepers could be examined.

The results received from benchmark participants for the simulation runs are reviewed in the following sections. All results for Simulations 4, 5 and 6 are presented in Appendix E. For clarity of discussion, some results have been shown in the following sections.

5.2.1 Quasi Static Forces when there are No Irregularities

When comparing the various graphs from Simulation 1 (quasi-static forces from concrete sleepered railway track) to those for Simulation 4 (quasi-static forces from timber sleepered railway track) in Appendix E, the shapes of the graphs are very similar. The magnitudes of the peak forces for the various output parameters are typically less for the timber sleeper railway track, as would be expected for a lighter track structure and a vehicle travelling at a slower speed (see Table 4.1).
The following points detail the major differences between Simulations 1 and 4:

- The amplitude of the NUCARSTM wheel/rail contact force response shown in Figure E1.1 (b) (see Appendix E) has decreased in comparison to Figure 4.7 (b), and is also significantly different from the other model responses;
- The NUCARSTM rail seat forces (below the steel sleeper plate) in timber track have dropped in magnitude significantly (see Figure 5.1 below) and are now more comparable to the other models for the timber sleepered response;
- In Figure 5.1 the SUBTTI model shows a slightly different response in rail seat force when the wheels pass over ‘Sleeper A’ in comparison to the concrete sleepered results. This could be a result of the steel sleeper plate component being very stiff in comparison to the rail pad of the concrete sleepered scenario, thus there is a less significant force drop-off effect than seen in Figure 4.10;

![4D - Rail Pad Force on Sleeper A](chart.png)

*Figure 5.1; 4D - Rail Seat Force on Sleeper A (Type 2)*

- The DARTS rail seat force response shown in Figure 5.1 appears to be affected by the properties of the steel sleeper plate in a similar way to that noted previously in Section 4.4.2 for Figure 4.10. The high stiffness of the
steel sleeper plate for the timber sleepered track in comparison to the HDPE rail pad for the concrete sleepered track appears to have led to the more severe drop in force in Figure 5.1 as the wheels pass over ‘Sleeper A’;

- The ballast pressure for DARTS shown in Figure 5.2 is half that of the other models, as a result of the low rail pad force. It is interesting to note that when a defect is analysed with DARTS, as in Figure 5.5, the ballast pressure appears unaffected by the quasi-static force irregularity shown in Figure 5.1.

Figure 5.2; 4G – Total Ballast Pressure on Sleeper A (Type 2)

- In Figure E1.5 (see Appendix E) the NUCARSTM model response for rail seat sleeper bending moment was more inline with DIFF and DARTS than for the concrete sleepered track response (see Figure 4.11). It can be seen that the rail seat moment at ‘Sleeper A’ was affected when the wheel was up to three sleepers from ‘Sleeper A’ in comparison to the concrete sleepered simulation where the rail seat was only affected when the wheel was two sleepers from ‘Sleeper A’;

- Figure E1.6 (see Appendix E) shows the DARTS model undergoing an even more complex sleeper bending resonance than that shown in Figure 4.12 for the concrete sleepered track. This could be due to the highly flexible nature of the timber sleepers in the model.
5.2.2 Dynamic Wheel/Rail Forces from Worn Wheel Flats

The worn wheel flat simulation for timber sleepers helped identify a number of modelling discrepancies among the benchmark models. In particular the wheel/rail contact, rail seat and ballast force responses varied among the models for the timber sleepered track (Simulation 5) in comparison to the concrete sleepered track (Simulation 2). There were no notable differences in the models’ responses for rail bending moment and rail acceleration. However for the rail seat and centre bending moment responses the NUCARS™ model showed significant differences in comparison to those for the concrete sleepered track simulation.

The following points show the major differences between Simulations 2 and 5:

- Figure 5.3 below shows the DARTS and NUCARS™ models having a very short peak wheel/rail contact force of much greater magnitude than the other models for the timber sleepered track simulation;

![Figure 5.3; 5A - Wheel/Rail Contact Force from Bogie Leading Wheel (Type 1)](image)
• The DIFF, TRACK and SUBTTI models show a different response to the wheel flat striking the rail, with no subsequent lift-off of the wheel from the rail. This could indicate the structure flexing as the wheel makes contact and absorbing the bounce motion. DARTS and NUCARSTM however appear to simulate a stiffer structure that is more affected by the steel sleeper plate element. Thus there is some bouncing of the wheel for these two models;

• The SUBTTI peak wheel/rail contact force shown in Figure 5.3 occurs after the other models due to the SUBTTI assumption that the wheel flat strikes the rail slightly later than the other models. This has been discussed in section 4.4.3;

• All models except for SUBTTI have shown a significant attenuation of the wheel/rail contact force by the rail for the rail seat force as shown in Figure 5.4. The peak rail seat force response calculated by SUBTTI in Figure 5.4 could indicate that the wheel/rail contact force has a large component of mid frequency P2 force. This is significantly different to the concrete sleepered response shown in Figure 4.17 (b) where there was much better agreement among the models;

![Figure 5.4; 5D - Rail Seat Force on Sleeper A (Type 2)](image-url)
The magnitude of the DARTS peak rail seat force shown in Figure 5.4 was low (50 kN) in comparison to NUCARS™ (340 kN) but similar to the DIFF rail seat response (60 kN). This indicates that the force shown in Figure 5.3 was predominately a high frequency P1 force that was filtered out by the rail mass and stiffness before reaching the rail seat;

The ballast forces shown in Figure 5.5 indicate lift-off of the sleeper from the ballast. The SUBTTI model demonstrates this clearly by allowing actual lift-off of the sleeper from the ballast, whereas the other models experienced impractical negative forces at the sleeper/ballast interface. These negative forces would tend to pull the two masses back together in those models and not show a realistic response;

The NUCARS™ ballast force is up to 50% lower than the other models, this is interesting considering the very high peak in the NUCARS™ wheel/rail contact and rail seat forces shown in Figures 5.3 and 5.4;

Figure 5.5; 5G - Total Ballast Force on Sleeper A from two Rails

The DARTS and DIFF models show peak ballast responses greater than the peak rail seat force for two rails, whilst the NUCARS and SUBTTI models show peak ballast responses less than a single rail seat force. This is
significantly different to the concrete sleepered simulation shown in Figure 4.20 (b);

- The NUCARSTM bending moments for the rail and sleeper centre also show responses significantly different in comparison to the concrete sleeper simulation. The large peak moments shown in Figures E2.2 and E2.6 (see Appendix E) would most likely be due to the very high wheel/rail contact forces and rail seat forces shown in Figure 5.3 and 5.4 and the fact that the NUCARSTM model used Euler beams to represent the rail and sleepers as discussed previously.

### 5.2.3 Dynamic Wheel/Rail Forces from Dipped Rail Joints

The benchmark tests showed agreement in the response shape of track dip simulation for responses throughout the timber sleepered track structure. There are very few differences in magnitude between the concrete sleepered track (Simulation 3) and the timber sleepered track responses (Simulation 6) as can be noted in Appendix E3.

The following points detail the major differences between Simulations 3 and 6:

![Graph of Wheel/Rail Contact Force for Bogie Leading Wheel](image)

*Distance marks = Position of sleeper*

**Figure 5.6; 6A - Wheel/Rail Contact Force from Bogie Leading Wheel (Type 1)**
The slower vehicle speed and timber sleepered track appears to have limited the amount of bouncing experienced by the wheel as shown in Figure 5.6. DARTS is the only model that shows significant lift-off of the wheel from the rail. As was found with the worn wheel flat irregularity, the DARTS and NUCARSTM models show wheel/rail contact forces that are higher in magnitude than the other models;

- The peak rail seat and ballast force responses for the timber sleepered track simulation shown in Figure E3.4 and E3.7 (see Appendix E) are in very good agreement (within 10%), as they were for the concrete sleepered track simulation. This indicates that the models may handle the mid frequency P2 forces in the system in a similar manner;

- The NUCARSTM responses for rail bending moment, for sleeper rail seat and for sleeper centre bending moments are up to 2.5 times higher than the other models for the timber sleepered simulation. This was also found for the concrete sleepered simulations and would appear to relate to NUCARSTM using an Euler beam rather than a Timoshenko beam for the rail and sleepers.

### 5.2.4 Limitations of Models

From the comparison made in the previous section it would appear that most of the models were able to handle the timber sleepered simulation reasonably well. There were various discrepancies throughout the simulations; however the models appeared to be able to handle the flexible timber sleepered track structure and stiff steel sleeper plate.

It was evident that the DARTS model had issues with calculation of the quasi-static forces and this affected to some extent the way the model estimated the magnitudes of forces throughout the system. The model’s assumptions regarding the rail seat element also appeared to create issues regarding the calculation of rail seat forces. It is the writer’s opinion that the whole track response was affected to some degree by the very stiff properties of the steel sleeper plate element.
The DIFF and SUBTTI models appeared to be in relatively good agreement with each other, and modelled force responses in a very similar way. This agreement was particularly evident in the dipped rail joint simulation. The TRACK model also appeared to give good agreement with the force responses of the DIFF and SUBTTI models, but it produced more simplified responses due to its linear nature as can be noted in Appendix E.

The NUCARS™ model showed much better agreement with the other models in the timber sleepered simulation for the dipped rail joint simulation, than for the worn wheel flat simulation. The magnitudes of the forces calculated in the track structure for NUCARS™ were however in the majority of cases significantly higher than other models. The NUCARS™ wheel/rail contact and rail seat peak forces seemed unrealistically large, questioning the validity of the modelling methods.
5.3 Traditional Track Design Methods

The traditional methods to assess railway track forces in the design process often use formulas that determine peak track forces by applying various ‘dynamic impact factors’ that have been fine tuned on the basis of observation and experimentation on track structures.

Calculation methods for determination of the quasi-static forces and dynamic wheel/rail forces experienced at dished joints are readily available for track design engineers. A comparison of these peak empirical track forces to the peak force calculated by the benchmark models will been presented in Sections 5.3.2 and 5.3.3.

Dynamic wheel/rail forces experienced by wheel flats however are not readily calculated by empirical formulas. Thus comparisons against results from the benchmark model are not presented.

For the calculation of forces experienced by concrete sleepers in Australia, the Australian Standard for prestressed concrete sleepers (AS1085.14, 2003) was used. For the calculation of forces experienced by timber sleepers the generally accepted method is found in Jeffs & Tew (1991).

A comparison of the peak benchmark simulation results calculated by the analytical models for each output parameter (see Table 4.5) is provided in Appendix F.

5.3.1 Empirical Force Calculations

Concrete Sleepered Ballasted Railway Track

The ‘Dynamic Impact Factor’ described in Chapter 2 is the basis for determining peak forces applied to the track. Various methods are available to determine this factor; however for dished rail joint scenarios the wheel/rail contact forces are often
estimated using the empirical formula developed by Jenkins et al (1974) (see Section 2.3.3). AS1085.14 (2003) states that the ‘dynamic impact factor’, or the ‘Combined vertical design load factor’ (j) shall not be less than 2.5 times the static wheel load. In the case of calculation of quasi-static forces the ‘dynamic impact factor’ is taken as being 1 and therefore only the static wheel load is considered in calculations.

Once the dynamic impact factor has been determined, the rail seat load (Output Parameter D) may be calculated using the following formula from AS1085.14 (2003) Clause 4.2.3.

\[ R = jQ \cdot \frac{DF}{100} \] (4.1)

where:

- \( R \) = Rail seat load (kN)
- \( j \) = Combined vertical design load factor (\( \geq 2.5 \))
- \( Q \) = Static wheel load (kN)
- \( DF \) = Axle load distribution factor

The axle load distribution factor (DF) may be obtained from Figure 4.1 of AS1085.14 (2003) for rail sizes larger than 47 kg/m. For sleepers spaced at 610 mm the DF is equal to 52%.

The design rail seat and centre sleeper moments may be calculated from the rail seat load as per Clause 4.3.2 and 4.3.3 of AS1085.14 (2003). The formulas are relatively simplistic and are based on the calculation of static beams under load. The moments are based on an effective length under the rail seat, calculated by subtracting the rail gauge from the overall length of the sleeper.

Section 4.3.2 of AS1085.14 (2003) states that the maximum positive bending moment (Output Parameter E) shall be taken to occur at the rail seat, producing compressive stress at the top and tensile stress at the underside of the sleeper. The following formula can be used to calculate the maximum positive design bending moment at the rail seat for standard gauge railways.
\[ M_{R+} = \frac{RL - g}{8} \]  
\[ \text{(4.2)} \]

where:
- \( M_{R+} \) = Maximum rail seat positive moment (kNm)
- \( R \) = Rail seat load (kN)
- \( L \) = Length of the sleeper (m)
- \( g \) = Rail gauge (m) (Standard Gauge)

The maximum rail seat negative bending moment \( M_{R-} \) shall not be less than 67\% of the positive moment or 14 kNm, whichever is the greater.

Section 4.3.3 of AS1085.14 (2003) states the maximum negative bending moment (Output Parameter F) shall be taken at the centre of the sleeper, under partially or totally centre-bound conditions producing tensile stress at the top and compressive stress at the underside of the sleeper. The following formula may be used for calculation.

\[ M_{C-} = \frac{R(2g - L)}{4} \]  
\[ \text{(4.3)} \]

where:
- \( M_{C-} \) = Maximum rail seat negative moment (kNm)
- \( R \) = Rail seat load (kN)
- \( L \) = Length of the sleeper (m)
- \( g \) = Rail gauge (m) (Standard Gauge)

The maximum centre positive bending moment may be calculated using the following formula:

\[ M_{C+} = 0.05R(L - g) \]  
\[ \text{(4.4)} \]

where:
- \( M_{C+} \) = Maximum rail seat positive moment (kNm)

The maximum ballast pressure (Output Parameter G) may also be calculated from the rail seat load using the following formula from Clause 4.2.4 of AS1085.14 (2003).
\[ p_{ab} = \frac{R}{b(L-g)} \]  \hspace{1cm} (4.5)

where:

- \( p_{ab} \) = Maximum ballast pressure (kPa)
- \( R \) = Rail seat load (kN)
- \( b \) = Breadth of the sleeper (m)
- \( L \) = Length of the sleeper (m)
- \( g \) = Rail gauge (m) (Standard Gauge)

**Timber Sleepered Ballasted Railway Track**

Essentially the same procedure for the calculation of forces in concrete sleepered ballasted track is used for timber sleepered track. The only significant difference is in the method used for the calculation of the distribution factor DF in order to determine rail seat load. The following formula may be used for the calculation of the DF (Jeffs & Tew, 1991).

\[ DF = s \left( \frac{k}{4EI} \right)^{0.25} \]  \hspace{1cm} (4.6)

where:

- \( s \) = Sleeper spacing (m)
- \( k \) = Track modulus (MPa)
- \( E \) = Elastic modulus of rail steel (MPa)
- \( I \) = Moment of inertia of rail (mm\(^4\))

The formula above calculates a DF of 37% for timber sleepers; this is lower than the 52% distribution factor for concrete sleepers, which is to be expected since timber sleepered track should be more flexible than the stiff concrete sleepered track, and therefore distribute load amongst more sleepers.
5.3.2 Quasi-static Forces

Empirical results for quasi-static forces (dynamic impact factor of 1) calculated using the methods shown above in Section 5.3.1 are presented in Table 5.1 and Table 5.2.

The major difference for the concrete sleepered track calculations appears to be for the calculation of ballast contact pressure (Output Parameter G). This could be related to the method used for calculation of ballast pressure. The empirical method assumed the rail seat force should be distributed over an area the width of the sleeper multiplied by the length of the sleeper less the rail gauge. The analytical models assumed the force applied by both rail seats should be distributed over the total contact area of the sleeper/ballast interface.

Table 5.1 Comparison of Quasi-static Forces for Concrete Sleepered Track

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL based on AS 1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS™</th>
<th>SUBTTI</th>
<th>TRACK</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (kN)</td>
<td>54.5</td>
<td>41.4</td>
<td>54.5</td>
<td>54.5</td>
<td>54.5</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>D (kN)</td>
<td>29.4</td>
<td>14.8</td>
<td>23.5</td>
<td>28.7</td>
<td>23.6</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>E (kNm)</td>
<td>4.0</td>
<td>2.1</td>
<td>2.6</td>
<td>3.9</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>F (kNm)</td>
<td>-2.5</td>
<td>-1.8</td>
<td>-2.4</td>
<td>-0.9</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>G (kPa)</td>
<td>108.0</td>
<td>61.7</td>
<td>78.5</td>
<td>91.4</td>
<td>80.7</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

* Ballast pressures reported by dynamic models may not be maximum values

The DARTS static wheel/rail contact force (A) is lower than the correct value of 54.5kN (as discussed in section 5.2.2). This error in the model appears to have also affected the magnitude of the rail seat force (D), moment (E, F) and ballast pressure (G).
NUCARS™ has the closest rail seat force (D), rail seat moment (E) and ballast pressure (G) to the empirically calculated values. The DIFF model compares best with the empirical centre seat sleeper moment; and DARTS model is low for both the sleeper centre and rail seat moment.

The benchmark results presented in Table 5.2 show up to a 50% difference in comparison to the empirically calculated results. The rail seat bending moments (E and F) and ballast pressure (G) calculations show the most disagreement.

Table 5.2 Comparison of Quasi-static Forces for Timber Sleepered Track

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL (based on AS 1048.14)</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS™</th>
<th>SUBTTI</th>
<th>TRACK</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (kN)</td>
<td>54.5</td>
<td>41.3</td>
<td>54.5</td>
<td>54.4</td>
<td>54.6</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>D (kN)</td>
<td>19.9</td>
<td>13.7</td>
<td>22.1</td>
<td>21.0</td>
<td>25.4</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>E (kNm)</td>
<td>2.3</td>
<td>1.4</td>
<td>2.4</td>
<td>2.9</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>F (kNm)</td>
<td>-2.8</td>
<td>-1.3</td>
<td>-1.3</td>
<td>-0.7</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>G (kPa)</td>
<td>93.0</td>
<td>39.1</td>
<td>78.3</td>
<td>74.9</td>
<td>91.0</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Again the DARTS quasi-static results are lower and this clearly affects the calculation of some of the other parameters. The rail seat and centre sleeper bending moments are around half the empirical results. Both DIFF and NUCARS™ show a similar rail seat moment than the empirical results however calculated much lower centre moments.

The SUBTTI model calculated a ballast pressure very similar to the empirical results which could be a result of the model allowing the sleeper to lift off the ballast. The DARTS model indicated a much lower ballast pressure, which could be a result of the incorrect quasi-static wheel/rail contact force and poor handling of the high stiffness of the steel sleeper plate as discussed previously.
5.3.3 Dynamic Wheel/Rail Forces

Empirical calculation of dynamic wheel/rail forces required the calculation of the ‘dynamic impact factor’. Using the Jenkins Formula (1974) and the parameters presented in Appendix C for the benchmark simulations, the dynamic impact factor for the dipped rail joint scenario was calculated to be 3.6 times the static wheel load. This dynamic impact factor was then used in the process outlined in Section 5.3.1. The results are compared for the various Output Parameter forces calculated by the benchmark models in Tables 5.3 and 5.4.

No empirical values were calculated for Output Parameter A (wheel/rail contact force) as the Jenkins Formula only calculates the mid frequency P2 force which is of significance for below rail components only.

Table 5.3 Comparison of Dynamic Wheel/Rail Forces for Concrete Sleepered Track

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL based on AS 1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS™</th>
<th>SUBTTI</th>
<th>TRACK</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (kN)</td>
<td>338.9</td>
<td>293.3</td>
<td>456.5</td>
<td>259.7</td>
<td>194.2</td>
<td>338.4</td>
<td></td>
</tr>
<tr>
<td>D (kN)</td>
<td>105.0</td>
<td>126.1</td>
<td>140.3</td>
<td>227.2</td>
<td>118.5</td>
<td>73.0*</td>
<td>87.5</td>
</tr>
<tr>
<td>E (kNm)</td>
<td>14.3</td>
<td>16.8</td>
<td>15.6</td>
<td>36.1</td>
<td>-</td>
<td>16.0*</td>
<td>-</td>
</tr>
<tr>
<td>F (kNm)</td>
<td>-9.1</td>
<td>-15.2</td>
<td>-13.7</td>
<td>-22.3</td>
<td>-</td>
<td>-9.3*</td>
<td>-</td>
</tr>
<tr>
<td>G (kPa)</td>
<td>385.1</td>
<td>508.2</td>
<td>466.9</td>
<td>433.7</td>
<td>367.2</td>
<td>-</td>
<td>452.5</td>
</tr>
</tbody>
</table>

* Dynamic Increment only

The large variance in the wheel/rail contact forces (A) calculated by the benchmark models shown in Table 5.3 was explored previously in Section 4.4.4. The VICT and SUBTTI models calculated rail seat forces (D) closest to the empirical results, with the NUCARS model indicating a rail seat force (D) of over twice the empirical value. The SUBTTI model again produced a ballast pressure (G) closest to the empirical results.
The dynamic increment for TRACK (calculated by subtracting the quasi-static forces from the dynamic wheel/rail forces) shows the model having a very similar result as the empirical calculation.

The rail seat (E) and centre (F) sleeper moments predicted by the models however are all larger than the empirically calculated results, indicating that the dynamics of the sleeper component play a large role in determining the bending moments experienced. It would appear that the current empirical method may not take the dynamic movement of the concrete sleepers into account adequately.

The timber sleepered track results for the dynamic wheel-rail forces compared more favourably in most cases to the empirically calculated results as shown in Table 5.4.

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL based on AS 1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS™</th>
<th>SUBTTI</th>
<th>TRACK</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (kN)</td>
<td>-</td>
<td>220.0</td>
<td>160.3</td>
<td>269.3</td>
<td>176.2</td>
<td>128.1</td>
<td>-</td>
</tr>
<tr>
<td>D (kN)</td>
<td>51.0</td>
<td>67.8</td>
<td>60.7</td>
<td>95.8</td>
<td>79.2</td>
<td>27.3*</td>
<td>-</td>
</tr>
<tr>
<td>E (kNm)</td>
<td>6.0</td>
<td>7.9</td>
<td>7.0</td>
<td>14.2</td>
<td>-</td>
<td>5.0*</td>
<td>-</td>
</tr>
<tr>
<td>F (kNm)</td>
<td>-7.3</td>
<td>-5.2</td>
<td>-3.7</td>
<td>-14.4</td>
<td>-</td>
<td>-2.4*</td>
<td>-</td>
</tr>
<tr>
<td>G (kPa)</td>
<td>237.0</td>
<td>222.0</td>
<td>256.7</td>
<td>218.2</td>
<td>246.1</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

*Dynamic Increment only

The rail seat (E) sleeper moments are in better agreement than the concrete sleeper results, however the centre moments (F) for the empirical results appeared to be largely overestimated. The NUCARS™ sleeper bending moments predicted were over twice the empirical results and that of DARTS, DIFF and TRACK, questioning the validity of the modelling methods as discussed previously.
5.3.4 Findings regarding Empirical Track Design Methods

The comparison of the benchmark results to the empirical design methods currently used has shown that for the calculation of forces arising from discrete irregularities such as a dipped rail joint a detailed examination of the track structure may be in order. The results comparison showed an inconsistency in agreement among the benchmark model and empirical results.

A more detailed examination of a series of discrete irregularities and track scenarios would have to be undertaken before either method could be ruled out as inappropriate.

It does appear however that the calculation of dynamic forces experienced by the sleeper may be more realistically represented using analytical models rather than the simplistic empirical calculation methods which are based on the simple beam theory.

It also appears that the empirical methods are more suited to the calculation of dynamic forces experienced by timber sleepers. The concrete sleepered simulation results for the benchmark models were on the whole higher in magnitude than the empirical results, whilst the timber sleepered simulation results calculated by the benchmark models compared more favourably.
5.4 Selection of a Model

A brief review of potential models available for use by the Australian railway industry is provided below. Unfortunately not all models reviewed in Chapter 3 were available to participate in the Benchmark Tests; this however should not discount them from selection. The most significant criterion for selection of the models were availability, potential for release to the Australian Rail CRC for use and finally development to a user-friendly state.

The commercial version of Netherlands Delft University’s RAIL model was available for E$10,000 at the time of writing. The writer reviewed the model’s manuals that were available on the internet (Esveld, 2004) and found that the user-interface was relatively difficult to interpret. The model was not available for further development, but was supported by its author for instruction on use, though not for interpretation of results.

Central Queensland University’s 3DWTSD model created at the Centre for Railway Engineering (CRE) in Rockhampton showed promise as it was developed in Australia and has been used in other Rail CRC projects. Discussions undertaken with the model’s author (Sun, 2004) were helpful in understanding the model’s abilities and the model was initially available for evaluation. A large drawback of the model was its poor usability. The program code effectively had to be rewritten for each different type of simulation. Although the model was very detailed and various components and irregularity descriptions could be readily changed, a great deal of work would be needed to create an effective user-interface. The run-time for the model to undertake a single standard simulation was also up to three hours on a Pentium IV 1.6 GHz Personal Computer, which appeared excessive. At the time of writing the author of the model was undertaking another Rail CRC project and the subsequent lack of time prohibited him from participating in the benchmark test. Also the fact that the track model had not been well validated was of concern for the CRE in terms of its release for this research.
When initially inviting participants to take part in the benchmark test, the writer also questioned whether the models were available for purchase or use through some type of agreement. The DIFF model was found to be available on a consulting basis through CHARMEC. At the time of writing the Australian concrete sleeper manufacturer Austrak Pty Ltd was utilising this service, however it was said to be costly and communication with Chalmers University slow (Perera, 2004). Communications with the model author (Neilsen, 2004) throughout the benchmark were very positive and a good relationship was founded for continued cooperative research in the future.

The American Railroad Association’s, Transport Technology Centre (TTCI) joined the benchmark test late in the process due to the participant’s heavy workload. The NUCARS™ track model that was eventually used in the benchmark was still under development and was not available to licensed users of NUCARS™ at the time of writing. It was unclear whether the track package would be included with the normal licence for NUCARS™ or whether it would be an optional extra at a separate cost. The benchmark test indicated several significant inconsistencies in comparison to other models; therefore it will be interesting to see what changes are made to the model in the future.

The SUBTTI model was unfortunately not available for public use at the time of writing, and was described as the intellectual property (IP) of the Technical University Berlin, Germany. At the time of writing the model did not allow analysis of sleeper bending moments and stress and strains; however this functionality was supposed to be added for future benchmarking exercises.

The TRACK model was available at the time of writing and was used by some rail organisations in Australia. The author quoted the cost of an open license of the TRACK software to be £2000 at the time of writing. The TRACK software is reasonably limited in scope (as it models only a few irregularities) and the interface has been found hard to use and results difficult to interpret. The TRACK software however does serve a useful purpose of secondary validation of results calculated by the model chosen in this thesis.
The VICT model was also not available for public use at the time of writing and was the IP of Southwest Jiaotong University, China. Although the model was relatively comprehensive it did not allow the examination of sleeper bending moments and stresses and strains. The type of user-interface for the software is also unknown.

The DARTS software was made freely available to the writer for evaluation and use as part of the benchmark tests discussed in Chapter 4. The author of DARTS had cooperated with the writer and was keen to improve the program to a useable state.

A literature review regarding design and analysis of prestressed concrete sleepers (Murray & Cai, 1998) recommended the use of a sophisticated analysis package, such as DARTS to quantify more clearly the effects of train speed, traffic mix, and frequency spectrum of wheel loading, upon impact and distribution of load on concrete sleepers.

The writer examined DARTS in detail and undertook a number of simulations to test the program. These tests showed that DARTS had some issues that have been identified in the benchmark tests described in Chapter 4; however at the time of writing the author was confident that these problems could be readily fixed. The benchmark results provided a reasonable level of confidence in the program’s capabilities.

The DARTS program met the criteria set by the Rail CRC submission as follows:

1. The program was freely available for development and its author was willing to undertake an IP agreement with the Rail CRC for use;
2. The program code was available for further development either through its author or as part of future research in the Rail CRC;
3. The DARTS software was comprehensive and allowed a large range of simulations and use of analysis methods. The program required input via a sequence of parameters listed in a text file, thus conversion to user-friendly software would not be very difficult;
4. The model was limited to the passage of a double axle bogie at the time of writing;
5. Vehicle mass and operating speeds could be specified;
6. Rail section and material properties, rail pad/sleeper plate properties, gauge, sleeper section and visco-elastic/dynamic material properties, spacing and number of sleepers, ballast and formation properties could be defined;
7. A wide range of wheel and rail discrete defects were available as well as the ability to use an arbitrary profile for the rail or wheel surface;
8. Outputs including dynamic and impact force histories for components throughout the track structure, including resonant frequencies of track and vehicle, dynamic compliances, force distributions, bending stresses in sleepers, ballast and formation pressures, and track deformations were available; and
9. Longitudinal loads in the rail could be incorporated, however lateral movement of the vehicle was not allowed at the time of writing. Nevertheless, free access to the code and to support by DARTS author means that adding further functionality to DARTS should not be difficult.

The DARTS model has been shown to satisfy the greatest number of criteria of the models examined in this thesis. The program was supported by its author and looked as if it could be made reliable and accurate at the time of writing. DARTS showed promise to provide the output quantities required for future research and for these reasons was chosen for use by the Australian railway industry and rail research community.
CHAPTER 6

Development of a User-Friendly Model of Track Dynamic Behaviour

6.1 Introduction

This chapter details the development of the selected model of railway track dynamic behaviour for the Australian Railway Industry and Research Community. The program structure of the selected model is detailed and the development of a user-friendly interface for the Australian railway track design engineer is discussed. Further developments for the program and interface are also discussed.

6.2 DARTS Program

6.2.1 Program Origins and Background

The only detailed description found explaining the workings of DARTS (Dynamic Analysis of Rail Track Structures) was in the author’s PhD thesis “Modelling of Rail Track Dynamics and Wheel/Rail Interaction” (Cai, 1992). The program was not mentioned by its present name (DARTS); however an evaluation of the models’ capabilities was presented. No discussion was provided for the operation of the program, however some example inputs were analysed and results presented.
The model was capable of investigating four general areas of railway track dynamics:

1. Natural vibration characteristics.
2. Dynamic track responses in the frequency domain (dynamic compliance).
3. Dynamic track responses under a stationary impact load.
4. Dynamic track/wheelset responses to moving wheel/rail interaction.

A more detailed description of DARTS’ capabilities and formulation was presented in Chapter 3.

6.2.2 Program Structure

DARTS was provided by its author (Cai, 2004) as a set of un-compiled Fortran 77 files. No formal manual was available, however some limited notes were scattered throughout the program code. The original program code was written using NDP Fortran (a Fortran 77 compiler) developed by the company Microway (Microway, 2004). As NDP Fortran was no longer sold at the time of writing, Absoft Pro Fortran v8.2 (Absoft, 2003) for Microsoft Windows was used to compile the program code into an executable program.

The names of the files provided by the DARTS author are listed in Table 6.1 with a description of their general purpose. Two example input text files were also provided (see Figure 6.2 later) which showed the required input parameters for running the DARTS program.

Upon examination of the files provided it was found that the program called various subroutines through the DTRACK.F file. In particular the READAT.F file contained a subroutine that collected parameters from the DARTS.INP file. By analysing these two files the writer was able to deduce the various options available to the user when operating the DARTS program.
Table 6.1 DARTS Program code files

<table>
<thead>
<tr>
<th>File</th>
<th>Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>DTRACK.F</td>
<td>Main program structure</td>
</tr>
<tr>
<td>DYNCOM.F</td>
<td>Track Responses in the frequency domain (dynamic compliance)</td>
</tr>
<tr>
<td>DYNML.F</td>
<td>Moving wheel/rail interaction for one to three DoF vehicle</td>
</tr>
<tr>
<td>DYNML2.F</td>
<td>Moving wheel/rail interaction for four DoF vehicle (bogie)</td>
</tr>
<tr>
<td>DYNSL.F</td>
<td>Track responses under a stationary impact load</td>
</tr>
<tr>
<td>DYNSL2.F</td>
<td>Track responses under a stationary impact load</td>
</tr>
<tr>
<td>FMRSF.F</td>
<td>Uncertain</td>
</tr>
<tr>
<td>NATOM.F</td>
<td>Uncertain</td>
</tr>
<tr>
<td>NATY0.F</td>
<td>Natural vibration characteristics</td>
</tr>
<tr>
<td>NATYMC.F</td>
<td>Uncertain</td>
</tr>
<tr>
<td>READAT.F</td>
<td>Reads input parameters into program</td>
</tr>
<tr>
<td>TOEF.F</td>
<td>Beam on Elastic Foundation Calculation</td>
</tr>
<tr>
<td>DARTS.INP</td>
<td>Input parameter text file</td>
</tr>
</tbody>
</table>

The READAT.F file was the most useful in deciphering the program structure. Figure 6.1 shows a flow chart created by the writer to describe the selection of input parameters for the DARTS program. The figure focused on selection of the moving wheel/rail interaction simulation. The start and end points of Figure 6.1 designated as ‘B’ relate to Figures 6.7 and 6.8 which describe the user-interface that was created for the program which will be discussed in Section 6.4.
Figure 6.1 Selection of Input Parameters for DARTS (see Appendix H for further details)
6.2.3 Input Parameters

A number of the files listed in Table 6.1 contained details of the parameter names used in the DARTS program code. Appendix G lists all parameters used in the various types of DARTS simulations. The writer confirmed the list of parameters and the relevant units with the DARTS author.

An example of the DARTS.INP file is shown in Figure 6.2. The order of the parameters is very important, as the DARTS program code expects to find various parameters in particular locations of the DARTS.INP file. The units of the parameters are also of importance and the types of numbers described (integer or decimal) must be correct.

![DARTS.INP file example](image)

*Figure 6.2 Example DARTS.INP file for input of parameters into DARTS*

Appendix H describes the parameters which are required for each line of the DARTS.INP file. The flow chart shown in Figure 6.1 can be used to navigate Appendix H. Depending on the options selected different DARTS.INP files may be created.

Many of the parameter values shown in Figure 6.2 are dependent on other parameters in the input file. Thus some calculations must be undertaken whilst developing an
input file for a particular simulation. The following sections detail the major decisions required before any simulation may be undertaken.

**Run & Write Selection**

The first decision to be made is the type of analysis required to be undertaken. The choices are detailed in Section 6.1.1; however a user would normally undertake an analysis of moving loads (DYNML) as this provides the best representation of a vehicle-track scenario. For the initial run the whole program must always be executed, however if a user decides to complete subsequent runs the program run-time can be reduced. This can be undertaken by changing any parameters except those relating to the track structure. Thus an examination of various vehicle types, speeds and analysis criteria for the same type of track structure maybe undertaken.

![Figure 6.3 Track structure Input Parameters required by the DARTS model](image)

Figure 6.3 shows the major input parameters required by the DARTS track sub-model. The rail, pad, sleeper and track bed (representing the combined effect of the ballast, subballast and subgrade) data is collected at various stages along the flowchart shown in Figure 6.1.
Rail and Sleeper Selection

The rail and the sleepers may be represented as Euler or Timoshenko beams. The sleepers can also be represented as rigid beams. The sleeper cross-sections may be uniform, or non-uniform by specifying the dimensions of three different cross sections over the length of the sleeper. The author of DARTS recommended that Timoshenko beams be used for both rails and sleepers, and that the sleeper be represented as a non-uniform beam.

DARTS is capable of simulating a track of 37 sleepers in length. The rail ends are represented as fixed boundary conditions, thus analysis is best undertaken in the centre of the simulated track length where literature has shown that the boundary conditions have less effect. The rail may also have a longitudinal force applied which would represent thermal stresses in the track.

Rail Data & Data Points on Rail Section

A number of input parameters are required to describe the rail, these include: elastic modulus; Poisson’s ratio; Timoshenko coefficient; material density; moment of inertia; cross sectional area and the axial force applied. The spacing of the sleepers is also defined in this portion of the input file.

For the calculation of rail stresses and strains in the rail from loads applied, other details regarding the rail are required. These include: a distance from the rail’s neutral axis to the stress plane of interest, the section modulus of the rail head and the width of the web at the neutral axis.

Sleeper Data

Depending on the choice made regarding the uniformity of the sleeper, various input parameters need to be established. For a non-uniform, Timoshenko beam sleeper, the following input parameters are required: the half sleeper length; breadth; depth at centre and at shoulder; rail gauge; moment of inertia at the centre and the shoulder; material density; elastic modulus; Poisson’s ratio; Timoshenko coefficient and
flexural rigidity. A number of these parameters are to be calculated at the input stage and this is discussed further in Section 6.3.3.

**Rail Pad and Track Bed Data**

The rail pad and track bed are represented by their stiffness and damping properties. The inputs required include: rail pad stiffness and damping; track bed stiffness at the sleeper centre and under the shoulder; and track bed damping along the length of the sleeper.

Definition of the frequency range of analysis is also required for input. The maximum frequency and the frequency interval for rail free vibration may be defined. The maximum frequency and the frequency interval for the sleeper free vibration are also required to be set.

**Coupling Parameters and Method of Integration**

DARTS allows a number of mathematical methods for the calculations undertaken during simulation. This includes coupled and uncoupled modes; proportional and un-proportional damping; and a choice of integration methods including Gauss Quadrature, Runge-Kutta fixed step and varying step method. The DARTS author suggested the use of coupled modes with un-proportional damping and the Runge-Kutta fixed step method as being the optimal mix of choices.

**Rail and Sleeper Analysis Positions**

The position on the rail at which actions are to be analysed during the simulation, must be defined by the user. The sleeper of analysis must also be defined along with the definition of what position on the sleeper is to be used for calculations of sleeper actions. Figure 6.4 shows an example of the various analysis positions required for simulation. The position of the centre of the irregularity is also shown in Figure 6.4, and will be discussed later.
Vehicle Selection and Data

The vehicle may be represented by a range of mass/suspension combinations detailed in Appendix H. The DARTS author recommended the use of the ‘Type 4’ vehicle which represents a bogie via a side frame mass and mass moment of inertia, two primary suspension elements and two wheel masses. The 4 DoF (degrees of freedom) model is symmetrical about the centreline of the track. An example of the bogie is shown in Figure 6.5. The DARTS model assumes the secondary suspension properties have no effect on railway track dynamics, thus the quarter car body weight is applied directly onto the side frame through the bolster.
As mentioned in the Benchmark analysis in Chapter 4, the average quasi-static force calculated by the DARTS model was not correct. The writer found through examination of the DARTS input parameters that the average quasi-static force calculated by the DARTS model was always equal to half the static carbody weight applied to the side frame. There appears to be an inherent problem with the definition of the carbody weight applied to the side frame and the definition of actual wheel load applied to the track. The DARTS author was correcting this problem at the time of writing.

The DARTS model also required the definition of the wheel/rail contact conditions through a parameter known as the Hertzian Flexibility Constant (G). Jenkins (1974) provided an explanation of this term and how it should be used. The method used to calculate the Hertzian Flexibility Constant is provided in Section 6.3.3.

**Wheel and Rail Irregularity Simulation, Position and Data**

The DARTS program allowed the selection of a range of wheel and rail surface profile irregularities as shown in Table 6.2. The program also allowed the definition of arbitrary wheel and rail surface profiles, so that any combination of irregularities could be modelled.

The surface profile of the wheel flat is represented in DARTS as a track irregularity. Literature has shown that wheel flats may be adequately represented as a dip in the track as discussed in Chapter 2.

The location of the mid point of the particular irregularity chosen must also be defined as shown earlier in Figure 6.4. For the ‘No Irregularity’ and ‘Sinusoidal Corrugation’ the surface profile is included over the entire length of the simulated track, thus no mid point description is required.

The wheel radius and speed of the vehicle are also defined by the user in this section. Finally, as DARTS is a time domain model, the time step required for analysis must be defined. DARTS is capable of undertaking any time step, however the smaller the
step the more accurate the calculations would be. Unfortunately this will also increase the total time for analysis. The DARTS author recommends a time step of 0.02 milliseconds with the model reporting data for every fifth time step. Thus the simulation results would be presented in 0.1 millisecond time steps. This analysis typically takes around five to ten minutes on a Pentium 4, 1.6 GHz computer.

Table 6.2 Types of Irregularity that can be simulated by DARTS

<table>
<thead>
<tr>
<th>Type</th>
<th>Equation</th>
<th>Shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 No Irregularity</td>
<td>Nil</td>
<td></td>
</tr>
<tr>
<td>2 Wheel Flat</td>
<td>$f(x) = \frac{d}{2} \left(1 - \cos\left(\frac{2\pi x}{L}\right)\right)$</td>
<td></td>
</tr>
<tr>
<td>3 Sinusoidal Corrugation</td>
<td>$f(x) = \frac{d}{2} \sin\left(\frac{2\pi x}{L}\right)$</td>
<td></td>
</tr>
<tr>
<td>4 Dipped Joint</td>
<td>$f(x) = d \times \left(1 \pm \cos\left(\frac{\pi x}{L}\right)\right)$</td>
<td></td>
</tr>
<tr>
<td>5 Hollow Weld</td>
<td>$f(x) = \frac{d}{2} \left(1 - \cos\left(\frac{2\pi x}{L}\right)\right)$</td>
<td></td>
</tr>
<tr>
<td>6 Humped Weld</td>
<td>$f(x) = -\frac{d}{2} \left(1 - \cos\left(\frac{2\pi x}{L}\right)\right)$</td>
<td></td>
</tr>
<tr>
<td>7 Arbitrary Wheel Surface Profile</td>
<td>$x$ and $y$ coordinates must be defined</td>
<td></td>
</tr>
<tr>
<td>8 Arbitrary Rail Surface Profile</td>
<td>$x$ and $y$ coordinates must be defined</td>
<td></td>
</tr>
</tbody>
</table>

where:

$f(x)$ = shape function of the irregularity

$x$ = current coordinate on the rail

$d$ = depth of irregularity

$L$ = total length of the irregularity
6.2.4 Simulation Results

Table 6.3 presents the many types of output parameters that the DARTS model automatically calculates during a simulation. These results are stored in text files which may be imported into a program such as Microsoft Excel for interpretation in a graphical format as shown for the Benchmark Test in Chapter 4.

Table 6.3 DARTS Output Parameters and their Units

<table>
<thead>
<tr>
<th>File Name</th>
<th>Output Parameter</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-RT.OUT</td>
<td>Acceleration histories for the track:</td>
<td>m/s²</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Rail pad</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Sleeper</td>
<td></td>
</tr>
<tr>
<td>A-TRUCK.OUT</td>
<td>Acceleration histories for the vehicle:</td>
<td>m/s²</td>
</tr>
<tr>
<td></td>
<td>Both wheels</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Two points on the side frames</td>
<td></td>
</tr>
<tr>
<td>Y-RT.OUT</td>
<td>Displacement histories for the track:</td>
<td>m</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Rail pad</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Sleeper</td>
<td></td>
</tr>
<tr>
<td>Y-TRUCK.OUT</td>
<td>Displacement histories for the vehicle</td>
<td>m</td>
</tr>
<tr>
<td></td>
<td>Both wheels</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Two points on the side frames</td>
<td></td>
</tr>
<tr>
<td>FORCE.OUT</td>
<td>Wheel/rail contact force for both wheels</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>Rail/Sleeper contact force</td>
<td>Pa</td>
</tr>
<tr>
<td></td>
<td>Ballast pressure (Sleeper rail seat &amp; centre)</td>
<td></td>
</tr>
<tr>
<td>R-MS2.OUT</td>
<td>Rail moment and shear force histories:</td>
<td>kNm &amp; N</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td>R-SS2.OUT</td>
<td>Rail bending and shear stress/strain histories:</td>
<td>Pa</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td>T-MS2.OUT</td>
<td>Sleeper moment and shear force histories:</td>
<td>kNm &amp; N</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td>T-SS2.OUT</td>
<td>Rail bending and shear stress/strain histories:</td>
<td>Pa</td>
</tr>
<tr>
<td></td>
<td>Rail at both wheel/rail contacts</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fixed rail point</td>
<td></td>
</tr>
<tr>
<td>NATVIB.OUT</td>
<td>Natural Frequency of the Vehicle and Track</td>
<td>-</td>
</tr>
<tr>
<td>DAMP1.INF</td>
<td>Track coupled damping values</td>
<td>-</td>
</tr>
<tr>
<td>FREVIB.INF</td>
<td>Sleeper Model Information</td>
<td>-</td>
</tr>
</tbody>
</table>
6.3 A User-Friendly Interface for DARTS

The DARTS program was originally designed by its author to allow easy manipulation of the required simulation parameters. The DARTS.INP input file was the only program instruction that needed to be changed to allow different simulations. As the DARTS.INP file was difficult to create without detailed instructions and time, a user-friendly interface was developed to assist in the creation of this file.

Visual Basic .NET (2002) was chosen as the programming environment to allow development of the user-friendly interface. Figure 6.6 shows an example of the initial user input screen; every interface screen has been designed by the writer to be intuitive in operation for railway track design engineers. The general Microsoft Windows interface structure was used so that the environment was as familiar as possible to the user.

![Figure 6.6 Example of DARTS User Interface](image-url)
6.3.1 Interface Design and Structure

The writer interviewed a number of track design engineers from different organisations (Boyce & Herman, 2003; Ikaunieks, 2003) before creating the DARTS interface. At the time of writing, the TRACK Software (Grassie, 1994) was the only railway track dynamic analysis package that was readily available to railway organisations. As noted previously in this thesis, users of this software commented that use of the program was not intuitive and that the results produced by the program were unclear. It was also found that the parameters required for input in TRACK were not readily available. Users were also unsure of the range of parameters that were appropriate for input.

Therefore the interface created for DARTS needed to address these issues. Another Rail CRC Project associated with this research was undertaking investigations of appropriate parameters for input into models such as DARTS. These parameters would form a library of information available for the DARTS Interface, thus users would not be required to undertake time consuming searches for appropriate parameters.

When designing the DARTS Interface the writer examined the TRACK software and also a number of Microsoft products. The general look and feel of these products was used in the DARTS interface to provide a level of familiarity for the user. QR standards for interface design (Brady, 2001) were also examined so that the Interface would meet the railway organisations’ requirements. The Interface went through a series of iterations and was tested by track design engineers at QR and also by the Rail CRC Project 5/23 steering committee.

DARTS Interface Layout

In general there are two actions a user undertakes when first loading a program; creation of a new file or opening an existing file. For the DARTS interface the term ‘Investigation’ was used to describe the types of file created by the interface. The term ‘Investigation’ refers to a ‘Simulation’ of a vehicle travelling over a section of railway track. The DARTS interface allows the user to create and save
investigations; when the user is ready to undertake the simulation, the ‘Run DARTS’ button should be clicked on the Investigation Window (see Figure 6.6).

Figure 6.6 shows the Interface menus available to the user, these include:

- Investigation Menu: for creating, opening and saving investigations;
- Results Menu: for examining results from simulations;
- Library Menu: for definition of sets of parameters for various components;
- Window Menu: for management the open windows; and
- Help Menu: for further instruction on the interface.

The writer completed the Investigation Menu as part of the present research. This required the collection of simulation parameters needed for the creation of the DARTS.INP file. Flow charts describing the process of creation of a ‘New’ investigation and how to ‘Open’ an existing investigation are shown in Figure 6.7 and 6.8.

The parameters required for an investigation are managed through a window with a number of tabs as shown in Figure 6.6. The tabs allow the collection of data relating to Track, Vehicle, Irregularity, Analysis and Comments. The often confusing method of data collection for the DARTS.INP file shown in Figure 6.1 was restructured by the writer to allow an intuitive approach to data input.

Section 6.3.2 discusses the required user inputs for the interface when undertaking investigations. As mentioned previously various input parameters required calculation using other input parameters, whilst some parameters were not required to be changed for every simulation. The DARTS interface was designed to simplify the parameter input process, whilst still allowing more detailed input if required.
Figure 6.7 Flow Chart for the Operation of the DARTS Interface (New...)

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Figure 6.8 Flow Chart for the Operation of the DARTS Interface (Open...)
6.3.2 Undertaking an Investigation

The following sections present the main user interfaces required to undertake a DARTS Investigation. A CD is attached to this thesis that allows the installation of the DARTS interface onto a computer with Microsoft XP. Appendix I presents instructions on installing the DARTS Interface on a personal computer. A manual for the DARTS interface was being completed at the time of writing providing a discussion on all windows in the program interface including a number of examples.

The DARTS program interface may be executed from the Microsoft XP Windows desktop or START menu. Upon execution a blank program screen appears similar to Figure 6.6. The user may create a new or open an existing investigation by selecting the ‘Investigation’ menu and then clicking the ‘New…’ or ‘Open…’ menu items. Figure 6.7 and 6.8 detail the main operation of the New and Open menu items.

Upon creating a New Investigation the program screen shows an ‘Investigation Window’. This window has five tabs that allow the input of ‘Track’, ‘Vehicle’, ‘Irregularity’, ‘Analysis’ and ‘Comments’ data as mentioned previously. The user may choose to select pre-defined component parameters from the drop down boxes provided or may click the ‘Properties’ buttons to further define parameters.

The interface has a ‘Library’ of parameters for the various components of the vehicle and track. Parameter sets have been established for various components in the system including the vehicle, bogies, wheels, rails, rail pads, sleepers and track bed materials.

There are two types of parameter sets:

- ‘Standard Type’ components in the vehicle or track, such as Australian Standard rail sections or specific manufacturer components such as sleepers or rail pads;
- ‘User-Defined’ components that are created by the user.

Standard Type components cannot be edited or deleted by the user, whereas User-Defined components may be saved to the Library and edited or deleted at any time.
**Track Tab**

The Track Tab allows the collection of all ‘Below-Rail’ data. Figure 6.9 shows drop-down boxes detailing components types for the rail, rail pad or sleeper plate, sleeper and track bed. The user may click the ‘Properties’ buttons to view, edit, add or remove components types to or from the master library.

![Figure 6.9 Track Tab Interface for a DARTS Investigation](image)

The form was designed to seek input from the user in a sequence which flows from the top of the track structure to the bottom. Common parameters for the various sections of the railway track structure are grouped together. For example the parameters associated with the rail include the rail type, the rail gauge and any longitudinal or axial force experience by the rail. The spacing of the sleepers is also grouped together with the selection of sleeper type. The user may click the ‘View Example Diagram’ for a visual description of the track components (see Figure 6.3).
Rail Properties Window

The Rail Properties window shown in Figure 6.10 may be accessed from the Track Tab by clicking the ‘Properties’ button next to the Rail Type drop-down box. The layout of this window is common for all Properties Windows in the DARTS interface. By default all properties are presented as read-only so that parameters are not changed accidentally. If the user wishes to make changes to a component type, the ‘Edit’ button may be clicked, however this option is only available for user-defined components. The user may add a new component type by clicking the ‘Add’ button, and then enter the relevant parameters. The ‘Apply’ button may be clicked to save changes to the library and still remain in the properties window. The user may choose to remove a component type from the master library by clicking the ‘Remove’ button; however this option is also only available for user-defined components. When the user has finished viewing, editing, adding or removing parameters the ‘OK’ button may be clicked to save and exit, or the ‘Cancel’ button may be clicked to exit without saving.

Figure 6.10 Rail Properties Window
Vehicle Tab

The Vehicle Tab allows the collection of all ‘Above Rail’ data. Figure 6.11 shows drop-down boxes detailing component types for the vehicle, bogie and wheels. When a user selects a type of vehicle the software automatically chooses the appropriate bogie and wheel types. The combination of vehicle, bogie and wheel may be defined by clicking the ‘Vehicle Properties’ button. The user may also change this pre-defined vehicle type by choosing other component types from the bogie and wheel drop-down boxes.

![Vehicle Tab Interface for a DARTS Investigation](image)

*Figure 6.11 Vehicle Tab Interface for a DARTS Investigation*

The speed of the vehicle during the simulation may also be entered in the text box provided. Only one speed may be analysed per simulation. Finally the Hertzian Contact Coefficient is calculated automatically based on the vehicle and rail selections. This coefficient represents the contact stiffness between the wheel and the rail and its calculation is discussed later in Section 6.3.3.
**Vehicle Properties Window**

The Vehicle Properties window allows the definition of the vehicle’s car body gross mass. Bogie and Wheel types may also be defined for specific vehicle types by using the drop-down boxes provided. As with the Rail Properties window only user-defined component types may be edited or removed. New vehicle types may be added by clicking the ‘Add’ button.

![Vehicle Properties Window](image)

Figure 6.12 Vehicle Properties Window

A diagrammatic representation of the vehicle that is used in the DARTS simulation may be viewed by clicking the ‘View Example Diagram’ button on the vehicle tab shown on Figure 6.11. This window remains open for the user whilst the Vehicle, Bogie or Wheel Properties windows are viewed.

The Bogie Properties window is shown in Appendix I and allows the definition of the axle spacing, sideframe mass and mass moment of inertia, bolster mass and primary suspension stiffness’s. A check box is provided to allow for the simulation of a three-piece bogie. Checking this box sets the primary suspension to 2.0 MN/m. This was found by the writer to be the upper limit of this parameter.
**Irregularity Tab**

The irregularity table enables the definition of the contact conditions at the wheel/rail interface. A rail or wheel surface profile irregularity may be chosen. Table 6.2 detailed the various choices available for the wheel or rail surface profile; these may be selected via the drop-down boxes provided. If an arbitrary profile is chosen the ‘Load File’ button is enabled. If the user nominates this irregularity type a DARTS profile file must be opened and loaded for simulation (The DARTS manual in Appendix I details this further).

![Irregularity Tab Interface for a DARTS Investigation](image)

*Figure 6.13 Irregularity Tab Interface for a DARTS Investigation*

Depending on the irregularity type chosen the input parameters ‘Length’ and ‘Depth’ may change in accordance with the requirements of the irregularity. For example for a rail corrugation the terms ‘Wavelength’ and ‘Amplitude’ are used. The various diagrams shown in Table 6.2 may be viewed by clicking the ‘View Example Diagram’ button.
**Analysis Tab**

The Analysis Tab allows the definition of the various analysis positions that record data during a simulation. Figure 6.4 shown previously presents a diagrammatic representation of the various analysis positions. To simplify the calculation of the analysis positions the DARTS interface automatically calculates values based on their location in relation to a single sleeper. The choice of which particular sleeper should be analysed in the simulation may be defined by clicking the ‘Advanced Setup…’ button.

![Figure 6.14 Analysis Tab Interface for a DARTS Investigation](image)

The drop-down boxes provided allow the selection of predefined positions (such as the ‘Midspan Before Sleeper’) or user-defined positions. Positive or negative values must be used based on the relative to the direction of travel. A visual example of the analysis position is available by clicking the ‘View Example Diagram’ button.
Advanced Setup Window

The Program Analysis Setup window allows the definition of advanced features in DARTS program. Most of the parameters are predefined and read-only, however future versions of DARTS may allow informed adjustment of these values. For this version of DARTS the writer chose the most appropriate parameters based on the recommendations of the DARTS’ author.

![Program Analysis Setup Window](image)

*Figure 6.15 Program Analysis Setup Window*

The Time Step Analysis Setup group may be adjusted by the user. The time step for analysis may be defined, along with the time step write number that defines which analysis result will be reported in the DARTS output files. The actual time step reported is automatically calculated based on these inputs.
Keep Track Data Checkbox

Once a DARTS investigation has been completed and the user re-opens an investigation for further use, the ‘Keep Track Data’ checkbox appears. This checkbox allows the user to undertake a quicker analysis by fixing the details of the track structure. As mentioned previously an initial DARTS run may take five to ten minutes to complete. By keeping the track structure data the same the DARTS run time may be reduced to less than three minutes. This is especially useful for repetitive analysis of differing speeds or analysis positions.

Figure 6.16 below shows the checked Keep Track Data checkbox. The Track tab becomes read-only, however by un-checking the Keep Track Data checkbox track structure may still be changed. The DARTS interface automatically checks whether the Keep Track Data checkbox can be checked to ensure the shortest analysis time is always undertaken (refer Figure 6.8 for more details).

![Figure 6.16 Track Tab when Keep Track Data Checkbox Ticked](image_url)
6.3.3 Calculations undertaken by the Interface

As mentioned previously various input parameters are required to be calculated before completion of the DARTS.INP file. The DARTS Interface undertakes these calculations for the user, making input as simple as possible. The following sections outline the many calculations required.

Hertzian Flexibility Constant and Hertzian Contact Coefficient

The commonly used parameter to define the wheel/rail contact conditions is the Hertzian Contact Coefficient. DARTS however requires the input of a Hertzian Flexibility Constant; fortunately there is a simple relationship between the two parameters (Jenkins et al, 1974).

\[
G = \frac{1}{C_H^{2/3}}
\]  
\[(5.1)\]

where:

- \(G\) = Hertzian Flexibility Constant (m/N^{2/3})
- \(C_H\) = Hertzian Contact Coefficient (N/m^{3/2})

The Hertzian Contact Coefficient may be calculated using the following formula (Sun, 2002):

\[
C_H = \frac{4G_{wr} \sqrt{R_c}}{3(1 - \nu_{wr})^2}
\]  
\[(5.3)\]

where:

- \(E_{wr}\) = Shear Modulus of the wheel and the rail (Pa)
- \(\nu_{wr}\) = Poisson’s ratio of the wheel and the rail
- \(R_c = \sqrt{rR}\)
- \(r\) = rolling radius of wheel (m)
- \(R = \frac{\rho_w R_t}{\rho_w - r}\)
- \(\rho_w\) = wheel profile radius (m)
- \(R_t\) = rail profile radius (m)
**Timoshenko Coefficient**

In the equations for Timoshenko beams (rail and sleepers) the effective transverse shear strain is taken as equal to the average shear stress on a cross section divided by the product if the shear modulus and the shear coefficient $K$. The coefficient $K$ is a dimensionless quantity dependent on the shape of the cross-section. It is introduced to account for the fact that the shear stress and shear strain are not uniformly distributed over the cross section (Cowper, 1966).

According to the commonly accepted definition, $K$ is the ratio of the average shear strain on a section to the shear strain at the centroid. Literature points out that the distribution of the shear strain over a cross section depends on the mode of vibration of the beam and therefore varies with frequency.

Cowper (1966) presents various equations for the calculation of the Timoshenko Coefficient for different shapes. Cai (1992) recommends use of the following values for a rectangular sleeper and standard rail shape:

- Rectangular Sleeper: 0.833
- Standard Rail: 0.34

The coefficient for other shapes such as a steel or hybrid sleepers may be calculated using the methods discussed in Cowper’s paper.

**Sleeper Properties**

Various sleeper properties are required to be calculated by the user for the DARTS.INP file. These include the 1st moment of inertia at various sections on the sleeper and also the flexural rigidity of the sleeper.

The DARTS program assumes that sleepers used for simulation always have a rectangular cross-section. Thus other sleeper shapes such as steel and hybrid sleepers must be converted to rectangular sections. At the time of writing the DARTS
interface allowed the selection and input of parameters associated with steel sleepers. The conversion of the hollow shaped section into a rectangular section was undertaken by the DARTS interface using the following method:

\[
TD = \frac{TA}{TB} \quad (5.3)
\]

where:

- \(TD\) = Average depth of sleeper (m)
- \(TA\) = Sleeper cross-sectional area (m\(^2\))
- \(TB\) = Average with of sleeper (m)

These values were then used in the following formulas for the calculation of moment of inertia and flexural rigidity (It should be noted that due to the transformation of shape of the sleeper cross-section the DARTS results for stress and strains in the sleeper would be incorrect).

For a rectangular section the following formulas are used by the DARTS interface to calculate the moment of inertia of a cross-section:

\[
TI = \frac{TB \cdot TD^3}{12} \quad \text{or} \quad TIS = \frac{TB \cdot TDS^3}{12} \quad (5.4)
\]

where:

- \(TI\) or \(TIS\) = Moment of Inertia of centre or shoulder (m\(^4\))
- \(TB\) = Average width of sleeper (m)
- \(TD\) or \(TDS\) = Depth of sleeper at centre or shoulder (m)

The flexural rigidity of a rectangular section sleeper is calculated by the DARTS interface using the following formula:

\[
EIT = TI \cdot TE \quad \text{or} \quad EIS = TIS \cdot TE \quad (5.5)
\]

where:

- \(EIT\) or \(EIS\) = Flexural rigidity of sleeper (N.m\(^2\))
- \(TI\) or \(TIS\) = Moment of Inertia of sleeper cross-section (m\(^4\))
- \(TE\) = Elastic Modulus of sleeper (Pa)
Sleeper Dimensions and Track Bed Properties

The writer found when undertaking simulations with the DARTS program that a number of rules had to be followed with regard to the definition of sleeper dimensions and track bed stiffness properties. Figure 6.17 shows the various sleeper dimensions require for input into the DARTS program. For the non-uniform sleeper option in the DARTS program it was found that the rail gauge, sleeper length and shoulder length had a direct relationship.

\[ TL > DG \]
\[ TLS > TL - DG \]
\[ 0 < TLS - (TL - DG) < DG \]
\[ (1/3) \times TLS < TL - DG < (2/3) \times TLS \]
\[ TL - TLS < DG \]

However, the author of DARTS is in disagreement with any such limitations placed on the sleeper dimensions. Cai (2004) comments that the following rules may be followed for a successful simulation:

\[ TLS = \frac{2}{3} \cdot DG \times 2 \]
\[ TL = DG + \frac{2}{3} \cdot DG \]

where:

- DG = Half Rail Centre to Centre (m)
- TL = Half sleeper Length (m)
- TLS = Length of sleeper shoulder (m)
- TD = Sleeper depth at centre (m)
- TDS = Sleeper depth at shoulder (m)
The DARTS program also requires input of track bed stiffness parameters which directly relate to other sleeper dimensions. The track bed stiffness under the centre and shoulder of the sleeper directly relate to the ratio of centre depth to the shoulder sleeper depth.

\[
SK = \frac{TD}{TDS} \times SKS \tag{5.7}
\]

where:
- \(SK\) = Track bed stiffness at sleeper centre (N/m²)
- \(SKS\) = Track bed stiffness at sleeper shoulder (N/m²)

**Timber Sleepers and Steel Sleeper Plates**

In Section 4.5 in Chapter 4 the limitations of simulation timber sleepered railway track were discussed. The DARTS model was originally designed for concrete sleepered track simulation. However timber sleepered track may be represented by changing the definition of the various required parameters for DARTS.

For concrete sleepered track the prestressed concrete sleeper is very stiff and the rail pad has lower visco-elastic properties. However for timber sleepered track the hardwood timber sleeper has lower vertical elastic properties and the steel sleeper plate (if used) has very high stiffness. Thus when simulating timber sleepered track the combined vertical stiffness of the pad and sleeper should be used in place of the elastic rail pad property used for concrete sleepered track.
The following formula may be used to calculate the combined vertical stiffness for a timber sleepered track simulation:

\[
E_{\text{Tot}} = \frac{E_{\text{Pad}}E_{\text{Slp}}(H_{\text{Pad}} + H_{\text{Slp}})}{E_{\text{Pad}}H_{\text{Pad}} + E_{\text{Slp}}H_{\text{Slp}}}
\]

where:

- \(E_{\text{Tot}}\) = Combined elastic modulus of timber sleeper and steel plate
- \(E_{\text{Pad}}\) = Elastic modulus of steel sleeper plate
- \(E_{\text{Slp}}\) = Vertical elastic modulus of timber sleeper
- \(H_{\text{Pad}}\) = Depth of steel sleeper plate
- \(H_{\text{Slp}}\) = Depth of timber sleeper

**Track Bed Stiffness and Damping versus Track Modulus**

The DARTS interface allows the user to choose between two methods of representing the track structure stiffness and damping. The DARTS program requires the input of combined track bed stiffness and damping representing the stiffness and damping of the ballast, sub-ballast and subgrade layers. These stiffness and damping values are not readily available to track design engineers throughout the world, thus these parameters are usually based on research undertaken of specific railway track sections.

The more widely accepted parameter for railway track stiffness is known as the Track Modulus. This value represents the stiffness of the whole track structure including the rail, pad, sleeper, ballast, sub-ballast and subgrade. Recent research into the track modulus in Australia (Zhang, 1998; Murray & Murray, 2001) has identified methods to quickly evaluate values for track modulus.

Zhang (2001) also present a method to convert track modulus to track bed stiffness when the properties of the rail, pad and sleeper are known. This method has been implemented into the DARTS interface. If a user chooses to use the ‘Track Modulus’ option in the Track Bed Properties window the value is converted into an appropriate track bed stiffness and damping parameter.
6.4 Summary

DARTS was provided by its author (Cai, 2004) as a set of un-compiled Fortran 77 files. No formal manual was available, however some limited notes were scattered throughout the program code.

The READAT.F file was the most useful in deciphering the program structure. Figure 5.1 shows a flow chart created by the writer to describe the selection of input parameters for the DARTS program. The figure focuses on selection of the moving wheel/rail interaction simulation.

The DARTS program was originally designed by its author to allow manipulation of the required simulation parameters. The DARTS.INP input file was the only program instruction that needed to be changed to allow different simulations. As the DARTS.INP file was difficult to interpret and create without detailed instructions and time, a user-friendly interface was developed to assist in the creation of this file.

Visual Basic .NET (2002) was chosen as the programming environment to allow development of the user-friendly interface. Figure 5.6 shows an example of a typical user screen, which has been designed by the writer to be intuitive for railway track design engineers. The general Microsoft Windows interface structure was used so that the environment was as familiar as possible to the user.

The main user interfaces required to undertake a DARTS Investigation were presented and discussed. A CD is attached to this thesis that allows the installation of the DARTS interface onto a computer with Microsoft XP. Appendix I presents instructions on installing the DARTS Interface on a personal computer. A manual for the DARTS interface was being completed at the time of writing providing a discussion on all windows in the program interface including a number of examples.

Further development of DARTS was initiated at the time of writing for correction of the various problems that had been identified by the writer in this thesis, and also for completion of the DARTS interface for distribution to the Australian railway industry and research community.

200
7.1 Introduction

The research presented in this thesis has identified and developed a sophisticated computer model for the analysis of railway track dynamic behaviour for use by the Cooperative Research Centre for Railway Engineering and Technologies (Rail CRC) in Australia. Due the complex nature of railway track structures, track designer engineers have developed only limited knowledge of the static and dynamic loadings that track may be subjected to in its lifetime. To improve knowledge more investigation of the railway track dynamic behaviour is required and a comprehensive set of measurements of track forces under the varied mix of traffic and track must be accumulated. This would be best undertaken using sophisticated computer models capable of quantifying the effects of train speed, traffic mix, wheel impact loading and distribution of vehicle loads into the track. The thesis was separated into four main parts:

1. An examination of the Australian railway system and present design procedures;
2. Identification of models of railway track dynamic behaviour through an international state-of-the-art literature review;
3. Selection of an appropriate model of railway track dynamic behaviour for use by the Rail CRC; and
7.2 Findings and Conclusions

Traditional railway track design uses an allowable stress design approach, with economic considerations frequently dictating component selection and size. The railway track in reality is a complex non-linear system, which is a characteristic that current design procedures typically ignore. The Australian Standard AS1085.14 (2003) for prestressed concrete sleepers commented that when designing individual track components it would be desirable to consider the track components together as a dynamic system.

There are various tools available for track design engineers in Australia to assist in the design of railway track. These tools are typically based on quasi-static analysis of track. There are very few railway track dynamic analysis tools available for the designer to examine the track in depth component by component. Thus a need was found for a new approach.

The principal function of a railway track dynamic analysis model is to couple the components of the vehicle and track structure to each other so that their complex interaction is properly represented when determining the effect of traffic load on stresses, strains and deformations in the components of the railway system. Such a model would provide a foundation for predicting the track performance and serves as a technical and economical device for track design and maintenance.

A review of the various analytical models that represent the railway track structure under the transient loading of a passing train was undertaken. The following findings were made regarding the capabilities of models that have been developed:

- Models are usually two-dimensional allowing symmetry along the centreline of the track and vehicle;
- Vertical forces are of particular interest for railway dynamic behaviour;
- A single bogie is generally agreed adequate for modelling of the vehicle;
- The primary suspension of the bogie are often included in vehicle models, however secondary suspension is typically ignored when examining track dynamics;
- The wheel/rail interface may adequately be represented by the Hertzian contact theory;
- Elements with long extensions like rail and sleepers are best modelled using the Timoshenko beam theory as it incorporates the shear characteristics of the beam;
- The rail pad component is usually modelled as a spring and dashpot. The rail pad also usually takes on the characteristics of the fastening system; and
- Ballast, subballast and subgrade are often modelled as spring and dashpot systems; however some models incorporate a ballast mass and shear connecting springs, or treat the whole substructure as a half space.

Various benchmark tests were reported in the literature that compared the merits of vehicle and track simulation models available at the time; the latest test was in 1996. A new benchmark was undertaken as part of this research as a number of new models had been created internationally, a number of which had incorporated the capacity for modelling complex track component behaviour. The new benchmark provided an opportunity to compare various model complexities and capabilities with a view to selecting one for possible adoption by the rail organisations comprising the Rail Cooperative Research Centre in Australia.

The participant models had a range of complexity, from the TRACK model with one wheel on a symmetric railway track to the NUCARS™ model which allowed three-dimensional analysis of the whole vehicle and various layers of the track. Therefore the benchmark provided a good opportunity to compare the merit of having complex or simple models.

The benchmark results showed that significantly different results may be obtained by models of track dynamic behaviour, depending on the assumptions taken by the user for a particular track scenario. Even though a complete set of parameters was provided for each particular vehicle and track simulation, the participants were unable to produce the same results. This is an important finding regarding how parameters are interpreted in models of track dynamic behaviour.
Some reasons why it would be difficult to obtain the same results from different models with different users could include the:

- level of experience of the user and therefore the detail sought by the user when undertaking a simulation;
- differing modelling methods and complexities;
- number of different input parameters required;
- interpretation of the irregularities in the wheel and rail;
- differences in analysis time step and the frequency range;
- flexible and rigid representation of the sleepers; and
- varying methods (Euler and Timoshenko) used for modelling rails and the sleepers which were found to have significant effects on results.

It is apparent that a single set of simulations representing only one vehicle and track scenario is insufficient to draw firm conclusions regarding the behaviour of railway track components. Thus a more rigorous analysis of differing vehicles, varying speeds, varying track structures and differing discrete irregularities would allow a more thorough understanding. This type of analysis is planned as part of future research in CRC Project 5/23.

The comparison of benchmark results to empirical design methods currently used, showed that a detailed examination of the track structure should be undertaken when calculating forces arising from discrete irregularities, such as a dipped rail joint or weld. The results comparison showed inconsistency in agreement among the benchmark models and empirical results.

A more detailed examination of a series of discrete irregularities and track scenarios would have to be undertaken before either method could be ruled out as inappropriate. It does appear however that the calculation of dynamic forces experienced by the sleeper may be more realistically represented using analytical models rather than the simplistic empirical calculation methods which are based on the simple beam theory.
It also appears that the empirical methods are more suited to the calculation of dynamic forces experienced by timber sleepers. The concrete sleepered simulation results for the benchmark models were on the whole higher in magnitude than the empirical results, whilst the timber sleepered simulation results calculated by the benchmark models compared more favourably.

7.3 Recommendations

The DARTS model was shown to satisfy the greatest number of criteria required for selection and is recommended for use by the Australian railway industry and rail research community, subject to certain aspects of the model being addressed as described later. The research presented in this thesis aimed to make the calculation of dynamic forces more accessible to Australian railway track engineers by providing a comprehensive analysis tool. The DARTS model was found to be suitable for use for the following reasons:

- The program was supported by its author and looked as if it could be made reliable and accurate at the time of writing.
- DARTS was validated against other models of track dynamic behaviour in this thesis and was to be tested against real track data at the time of writing;
- A user-friendly interface that was readily interpretable was able to be created for DARTS by the writer as part of this research;
- At the time of writing DARTS was provided for use to the Rail CRC through an Intellectual Property agreement and therefore was cost effective in access and operation;
- DARTS showed the promise to provide the output quantities required for future research.

It is recommended that the DARTS author should be contracted to correct various problems in the DARTS program that have been identified by the writer in Chapters 4, 5 and 6. Once these problems have been rectified, the results for the benchmark
test should be recalculated for better assessment of the programs’ ability to simulate track. The problems included:

1. The quasi-static force shift problem;
2. The unusual rail pad reactions to a passing wheel; and
3. The sleeper dimension limitations for producing sleeper moment results.

It is also recommended that the user-friendly interface should be completed so that graphical results publishing features be readily used.

A second benchmark should also be undertaken to further validate the participant models against real track data. A comparison of the results may be made more readily and particular attention should be given to the following, with due consideration of and allowance for the participant models’ differing capabilities.

- The analysis time step and the frequency range to be considered in modelling should be specified;
- The sleepers should be modelled as flexible beams if possible; and
- Timoshenko beams should be used where possible for the rails and the sleepers as this modelling method incorporates shear characteristics which have been found to have significant effects on results.

Standards Australia is committed to transforming all of its standards to the more rigorous and defensible philosophy of ‘limit states’, which is a probability based approach. The AS1170 (2002) limit state loading code does not specify loading criteria for railways. Thus a statistical/probabilistic analysis of loads should be undertaken determining magnitudes of loads and the load combinations so the structure has an acceptably low risk of failure or of unserviceability. This would be best undertaken using the DARTS model detailed in this thesis.
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Appendix A

Vehicle & Track Parameters included in Dynamic Impact Factor Formulae
(Tew et al, 1991)
Table A.1 Vehicle & Track Parameters included in Dynamic Impact Factor Formulae (Tew et al, 1991)

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<td>Train Speed ($V$) ($\alpha'$)</td>
<td>or Speed Factor ($\eta$)</td>
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<td>$1 + 5.21 \frac{V}{D}$</td>
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<td>✓</td>
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<tr>
<td>Eisenmann (1972)</td>
<td>$1 + \delta \cdot \eta \cdot t$ ($t =$ Upper Confidence Limit)</td>
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<td></td>
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<tr>
<td>ORE (1965)</td>
<td>$1 + \alpha' + \beta' + \gamma'$</td>
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<td>✓</td>
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<tr>
<td>Indian Rail (1974)</td>
<td>$1 + \frac{V}{58.14 \sqrt{k}}$</td>
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<tr>
<td>German Railways (1961)</td>
<td>$1 + \frac{V^2}{3 \times 10^4}$</td>
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<td>South Africa Rail (1974)</td>
<td>$1 + 4.92 \frac{V}{D}$</td>
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<td>Clarke (1957)</td>
<td>$1 + \frac{19.65 \cdot V}{D \cdot \sqrt{k}}$</td>
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<td>WMATA (1974)</td>
<td>$(1 + 3.86 \times 10^{-5} \cdot V^2)^{2/3}$</td>
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Appendix B

Comparison Table for Railway Track Dynamic Analysis Models
Table B.1 Comparison Table for Railway Track Dynamic Analysis Models

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<th>DARTS (Section 3.3.2)</th>
<th>DIFF (Section 3.3.8)</th>
<th>NUCARS (Section 3.3.10)</th>
<th>SUBTTI (Section 3.3.4)</th>
<th>TRACK (Section 3.3.9)</th>
<th>VICT (Section 3.3.3)</th>
<th>3DWTSD (Section 3.3.1)</th>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Frequency / Time Domain</td>
<td>T</td>
<td>T</td>
<td>T</td>
<td>T</td>
<td>F</td>
<td>T</td>
<td>T</td>
<td>T</td>
<td>T</td>
<td>T</td>
</tr>
<tr>
<td>Moving Mass / Irregularity</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
<td>M</td>
</tr>
<tr>
<td>Infinite / Finite Track Length</td>
<td>F</td>
<td>F</td>
<td>F</td>
<td>F*</td>
<td>I</td>
<td>F</td>
<td>I</td>
<td>F</td>
<td>F</td>
<td>F</td>
</tr>
<tr>
<td>Maximum Frequency (Hz)</td>
<td>3000</td>
<td>2200</td>
<td>500</td>
<td>3000</td>
<td>2048</td>
<td>1000</td>
<td>?</td>
<td>?</td>
<td>?</td>
<td>?</td>
</tr>
<tr>
<td>Commercial Software available</td>
<td>✓*</td>
<td>✓</td>
<td>✓*</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
</tbody>
</table>

Terminology: ‘✓’ yes, ‘✓’ no, ‘?’ unknown, ‘/’ or
Rail or Sleeper: Timoshenko beam, Euler beam, Timoshenko elements, Rigid mass
Suspension, Contact, Railpad, Ballast, Subballast or Subgrade: Linear, Non-linear, Half-space

*Notes: DARTS – Track bed stiffness / damping; Software interface under development
NUCARS – Real Time Wheel Rail contact model; Track model not available commercially
SUBTTI – Sleeper lift-off is allowed; Ballast is a FE formulation for a prismatic volume of load distributed beneath each sleeper; FEs are placed onto the half-space subgrade; a ‘Ring’ model is used to represent the track as an infinite length;
TRACK – a Bogie mass is accounted for over one wheel
VICT – 3D model is also available; Ballast mass is also included
RAIL – Up to 5 car bodies can be modelled
RTRI Japan – The ballast layer is still into 3 sub-layers
Appendix C

Benchmark Tests for Models of Railway Track Dynamic Behaviour (Steffens, 2003)
Benchmark Tests for Models of Railway Track Dynamic Behaviour

Version 2, Amendment A

Initiated:
December 2003

Compiled by:
David Steffens

Queensland University of Technology
Brisbane, Australia
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1. INTRODUCTION

Many models of track dynamic behaviour exist throughout the world. A benchmark test was undertaken in 1995/96 by Dr Stuart Grassie to compare eight models of railway track and vehicle/track interaction. This test identified that substantially identical results could be obtained from both frequency and time domain models in the majority of conditions tested.

Over the past eight years much research has been undertaken to improve and extend models of railway track dynamic behaviour. A literature review (Steffens, 2003) has identified the key research bodies that have been exploring dynamic behaviour of railway track and vehicle/track interaction. Leaders in the field of railway track dynamics have been invited to take part in a benchmark test for the comparison and validation of analytical models of track dynamic behaviour. The invitees who expressed interest in the benchmark exercise are listed in Table 1.1.

Table 1.1. Participants of the Benchmark Test

<table>
<thead>
<tr>
<th>Participant</th>
<th>Model Name</th>
<th>Model Developed at</th>
<th>Email Contact</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dr Jens Nielsen</td>
<td>DIFF</td>
<td>CHARMEC, Sweden</td>
<td><a href="mailto:jens.nielsen@ingemansson.se">jens.nielsen@ingemansson.se</a></td>
</tr>
<tr>
<td>Dr Nick Wilson</td>
<td>NUCARS</td>
<td>AAR, TTCI, USA</td>
<td><a href="mailto:nicholas_wilson@aar.com">nicholas_wilson@aar.com</a></td>
</tr>
<tr>
<td>Dr Yan Quan Sun</td>
<td>3D WTSD</td>
<td>Central Queensland University, Australia</td>
<td><a href="mailto:y.q.sun@cqu.edu.au">y.q.sun@cqu.edu.au</a></td>
</tr>
<tr>
<td>Dr Stuart Grassie</td>
<td>TRACK</td>
<td>Cambridge University, UK</td>
<td><a href="mailto:sgrassie@aol.com">sgrassie@aol.com</a></td>
</tr>
<tr>
<td>Dr Zhenqi Cai</td>
<td>DARTS</td>
<td>Queen's University, Canada</td>
<td><a href="mailto:zjc_1412@yahoo.com">zjc_1412@yahoo.com</a></td>
</tr>
<tr>
<td>Dr Ulf Gerstberger</td>
<td>SUBTTI</td>
<td>Technical University Berlin, Germany</td>
<td><a href="mailto:ulf.gerstberger@bam.de">ulf.gerstberger@bam.de</a></td>
</tr>
<tr>
<td>Prof Wanming Zhai</td>
<td>VICT</td>
<td>Southwest Jiaotong University, China</td>
<td><a href="mailto:wmszhai@home.swjtu.edu.cn">wmszhai@home.swjtu.edu.cn</a></td>
</tr>
<tr>
<td>Dr Valéri Markine</td>
<td>RAIL</td>
<td>Delft University, Netherlands</td>
<td><a href="mailto:v.l.markine@citg.tudelft.nl">v.l.markine@citg.tudelft.nl</a></td>
</tr>
</tbody>
</table>

1 Model will be run and results analysed by David Steffens

1.1. Objectives

The objectives of this benchmark tests are as follows:

- To provide a forum for discussion and information sharing among researchers who are developing models of railway track dynamic behaviour.
- To compare model capabilities and simulation results amongst participants, using a standard set of parameters.
- To validate simulation results with data collected from the field.

It should be noted that sharing of information will only be undertaken with the full consent of participants. All information provided by the participants will remain confidential until agreements have been formed.
1.2. Benchmark Test Structure

*Part 1: Initial Simulation* will focus on comparison of the model capabilities and simulation results only. This document provides theoretical parameters for a vehicle travelling on two track structures.

*Part 2: Field Data Simulation* will use data currently being collected from a test site in Australia. Accurate parameters for the vehicle and track will be collated and provided in a supplementary document in early 2004. This part of the benchmark test will assist in validation of the models.

The benchmark test will focus on vertical vehicle/track interaction on straight standard gauge track. The test may be extended to include longitudinal and lateral vehicle/track interaction depending on the participant interest.

The requested simulations are presented in Section 2 and the testing parameters are presented in Section 3. Section 4 details the requested outputs and format of results for analysis.

If there are any clarifications requested during the benchmark testing, answers will be distributed to all participants so that testing be as consistent as possible.

It is requested that all participants keep information regarding the benchmark testing confidential. This information is owned by the Rail CRC Australia and is only provided to participants for use during the benchmark tests.

1.3. Assumptions and Limitations

From the literature available it is understood that some models may not be able to calculate some of the requested outputs (Section 4) or be able to make use of some the parameters suggested. It is requested that participants undertake as many calculations as possible to ensure that comparisons can be undertaken among all models.

To confirm descriptions presented in literature, participants are requested to provide a short description of their models including information such as:

- Frequency or time domain modelling?
- Moving mass or moving irregularity modelling?
- Infinite or finite length of track for modelling, and boundary conditions?
- Frequency range?
- Submodel inclusions?
- Wheel / rail interface conditions?
- Vertical, lateral and longitudinal modelling?
- Symmetrical modelling?
- Other important details?

It is also requested that any **assumptions or special requirements** be described so that a full understanding of the model's capabilities can be assessed.
2. ANALYSIS

Parameters to be used in Part 1: Initial Simulation are presented in Section 3. A passenger vehicle at particular speeds has been chosen to run on two track types with various irregularities. Table 2.1 describes the six runs requested to be undertaken.

Table 2.1. Benchmark tests for Part 1: Initial Simulation

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Vehicle (Section 2.1)</th>
<th>Speed (Section 2.2)</th>
<th>Track Structure (Section 2.3)</th>
<th>Condition (Section 2.4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Passenger</td>
<td>160 km/h</td>
<td>Concrete</td>
<td>Perfect Rail¹</td>
</tr>
<tr>
<td>2</td>
<td>Passenger</td>
<td>160 km/h</td>
<td>Concrete</td>
<td>Wheel Flat</td>
</tr>
<tr>
<td>3</td>
<td>Passenger</td>
<td>160 km/h</td>
<td>Concrete</td>
<td>Dipped Flat</td>
</tr>
<tr>
<td>4</td>
<td>Passenger</td>
<td>100 km/h</td>
<td>Timber</td>
<td>Perfect Rail¹</td>
</tr>
<tr>
<td>5</td>
<td>Passenger</td>
<td>100 km/h</td>
<td>Timber</td>
<td>Wheel Flat</td>
</tr>
<tr>
<td>6</td>
<td>Passenger</td>
<td>100 km/h</td>
<td>Timber</td>
<td>Dipped Joint</td>
</tr>
</tbody>
</table>

¹ “Perfect Rail” refers to continuously welded rail (CWR) with no surface irregularities.

The following sections explain in more detail the elements of each benchmark test.

The benchmark tests runs for Part 2: Field Data Simulation will be provided in a supplementary document in early 2004 when accurate parameters for the field testing have been collated.

2.1. Vehicle

The passenger vehicle used for the Manchester Benchmarks (Iwnicki, 1998) will be used for Part 1: Initial Simulation. The vehicle is a general passenger coach with two bogies and a simple primary suspension.

The vehicle model is symmetric and all bodies should be assumed rigid. Details of the bogies and wheel profile contact conditions are presented in Appendix A & B, detailed suspension characteristics and dimensions of the car body and bogies are presented in Section 3 (Some models may not be able to make use of all of the vehicle parameters provided, please detail any assumptions made).

An Australian locomotive and/or freight container wagon will be used for vehicles in Part 2: Field Data Simulation of the benchmark test.

2.2. Speed

Two train speeds have been chosen based on which track structure the train would be running on. For concrete sleepered ballasted railway track a speed of 160 km/h or 44.44 m/s shall be used. For timber sleepered ballasted railway track a speed of 100 km/h or 27.78 m/s shall be used.
2.3. Track Structure

Two typical track structures have been chosen with different rail and sleeper types. Standard gauge (1435 mm) ballasted railway track has been chosen that includes a subballast (capping layer) over the formation (subgrade). UIC Rail has been used in Part 1: Initial Simulation for simplicity. Part 2: Field Data Simulation will make use of 53 and 60 kg/m Rail as defined by the Australian Standard AS 1085.1.

![Figure 2.1. Track Structure Description](image)

Table 2.3. Concrete Sleepered Ballasted Track

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail</td>
<td>UIC 60 kg/m</td>
<td>Table 3.4.</td>
</tr>
<tr>
<td>Fastener</td>
<td>Pandrol ‘e’ 2003 clip</td>
<td>Table 3.4.</td>
</tr>
<tr>
<td>Rail Pad</td>
<td>7.5 mm HDPE</td>
<td>Table 3.4.</td>
</tr>
<tr>
<td>Sleeper</td>
<td>Concrete (30 tal rated)</td>
<td>Table 3.4.</td>
</tr>
<tr>
<td>Ballast</td>
<td>250 mm (below the sleeper base)</td>
<td>Table 3.6.</td>
</tr>
<tr>
<td>Subballast</td>
<td>150 mm</td>
<td>Table 3.6.</td>
</tr>
<tr>
<td>Formation</td>
<td>Medium Stiffness</td>
<td>Table 3.6.</td>
</tr>
</tbody>
</table>

Table 2.4. Timber Sleepered Ballasted Track

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail</td>
<td>UIC 54 kg/m</td>
<td>Table 3.5.</td>
</tr>
<tr>
<td>Fastener</td>
<td>Pandrol ‘e’ 2003 clip</td>
<td>Table 3.5.</td>
</tr>
<tr>
<td>Rail Pad</td>
<td>19 mm Steel sleeper plate</td>
<td>Table 3.5.</td>
</tr>
<tr>
<td>Sleeper</td>
<td>Timber (Hard wood)</td>
<td>Table 3.5.</td>
</tr>
<tr>
<td>Ballast</td>
<td>250 mm (below the sleeper base)</td>
<td>Table 3.6.</td>
</tr>
<tr>
<td>Subballast</td>
<td>150 mm</td>
<td>Table 3.6.</td>
</tr>
<tr>
<td>Formation</td>
<td>Medium Stiffness</td>
<td>Table 3.6.</td>
</tr>
</tbody>
</table>
2.4. Track Condition

Three condition types have been chosen for modelling including rail that has no irregularities (for base-line assessment), a wheel with a flat spot, and a dipped joint depending on the track structure.

Table 2.5. Irregularity Type for Modelling

<table>
<thead>
<tr>
<th>Irregularity Type</th>
<th>Description</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Perfect Rail</td>
<td>Continuously welded rail (CWR) with no surface irregularities</td>
<td>a = 50 mm (length of flat area on wheel tread) d = 0.3 mm* (depth of flat area on wheel tread) NB The width of the flat area on the wheel tread is assumed to be the width of the wheel/rail contact area</td>
</tr>
<tr>
<td>Wheel Flat</td>
<td></td>
<td>NB The wheel flat should strike directly above the sleeper</td>
</tr>
<tr>
<td>Dipped Rail Joint</td>
<td>Dip located in centre of sleeper bay</td>
<td>L = 1000 mm (total length of Irregularity, 500 mm either side of dip) d = 3.5 mm (depth of dip) NB The width of the dipped weld is assumed to be the width of the rail head</td>
</tr>
</tbody>
</table>
3. PARAMETERS

3.1. Unit and Sign Conventions

Figure 3.1 below presents the axis names and sign conventions to be used during testing (the arrow defines the positive direction). The x-axis in the positive direction shall be used as the direction of travel. Standard International (SI) units have been provided and results would be appreciated in these units.

![Figure 3.1. X Y Z Axis](image)

The parameters presented in Tables 3.1 to 3.3 have been supplied by the Manchester Benchmarks (Iwnicki, 1998) and every effort has been made to keep the parameters relevant for track behaviour modelling. Parameters presented in Tables 3.4 to 3.6 have been provided by Australian railway researchers, railway asset managers and railway manufacturing companies. Every effort has been made to keep the parameters as representative of real track behaviour as possible, however due to the high variability of railway track some parameters are very difficult to define.

3.2. Vehicle Parameters

Table 3.1. Vehicle Characteristic (Iwnicki, 1998)

<table>
<thead>
<tr>
<th>CAR BODY</th>
<th>Symbol</th>
<th>Input Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Axle Load*</td>
<td>$F_V$</td>
<td>$11.12 \times 10^3$</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of wagon car body</td>
<td>$m_w$</td>
<td>$32 \times 10^3$</td>
<td>kg</td>
</tr>
<tr>
<td>Roll Mass moment of inertia of wagon car body</td>
<td>$I_{wax}$</td>
<td>$56.8 \times 10^3$</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>Yaw Mass moment of inertia of wagon car body</td>
<td>$I_{way}$</td>
<td>$1.97 \times 10^6$</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>Pitch Mass moment of inertia of wagon car body</td>
<td>$I_{waz}$</td>
<td>$1.97 \times 10^6$</td>
<td>kg.m$^2$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>BOGIE</th>
<th>Symbol</th>
<th>Input Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of bogie</td>
<td>$m_{bo}$</td>
<td>$2.615 \times 10^3$</td>
<td>kg</td>
</tr>
<tr>
<td>Roll Mass moment of inertia of bogie</td>
<td>$I_{box}$</td>
<td>$1.772 \times 10^3$</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>Yaw Mass moment of inertia of bogie</td>
<td>$I_{boy}$</td>
<td>$3.067 \times 10^3$</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>Pitch Mass moment of inertia of bogie</td>
<td>$I_{boz}$</td>
<td>$1.476 \times 10^3$</td>
<td>kg.m$^2$</td>
</tr>
</tbody>
</table>
### WHEELSET
- Mass of wheelset: \( m_{wh} = 1.813 \times 10^3 \) kg
- Roll Mass moment of inertia of wheelset: \( I_{whx} = 1.120 \times 10^3 \) kg.m²
- Yaw Mass moment of inertia of wheelset: \( I_{why} = 1.120 \times 10^3 \) kg.m²
- Pitch Mass moment of inertia of wheelset: \( I_{whz} = 0.112 \times 10^3 \) kg.m²

### VEHICLE DIMENSIONS
- Bogie semi pivot spacing: \( L_{Bsp} = 9.500 \) m
- Bogie semi wheelbase: \( L_{Bsw} = 1.280 \) m
- Wheel radius: \( r_W = 0.460 \) m
- Height above rail level of bogie centre of gravity: \( H_{Bcg} = 0.600 \) m
- Height above rail level of body centre of gravity: \( H_{Wcg} = 1.800 \) m

<table>
<thead>
<tr>
<th>Table 3.2. Suspension Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Primary Suspension</strong></td>
</tr>
<tr>
<td>Longitudinal stiffness</td>
</tr>
<tr>
<td>Vertical stiffness</td>
</tr>
<tr>
<td>Lateral stiffness</td>
</tr>
<tr>
<td>Longitudinal damping</td>
</tr>
<tr>
<td>Vertical damping</td>
</tr>
<tr>
<td>Lateral damping</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Secondary Suspension</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal stiffness</td>
</tr>
<tr>
<td>Vertical stiffness</td>
</tr>
<tr>
<td>Lateral shear stiffness</td>
</tr>
<tr>
<td>Longitudinal damping</td>
</tr>
<tr>
<td>Vertical damping</td>
</tr>
<tr>
<td>Lateral damping</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Suspension Geometry (Appendix A)</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Primary springs</strong></td>
</tr>
<tr>
<td>Longitudinal semi spacing</td>
</tr>
<tr>
<td>Lateral semi spacing</td>
</tr>
<tr>
<td>Height above rail level of wheelset end</td>
</tr>
<tr>
<td>Height above rail level of bogie frame end</td>
</tr>
</tbody>
</table>

| **Secondary springs** |
| Longitudinal semi spacing | \( x_3 = 9.500 \) m |
| Lateral semi spacing | \( y_3 = 1.000 \) m |
| Height above rail level to top | \( h_3 = 1.130 \) m |
| Height above rail level of bottom | \( h_4 = 0.525 \) m |

NB Please see (Iwnicki, 1998) Manchester Benchmarks for more information

<table>
<thead>
<tr>
<th>Table 3.3. Wheel/Rail Interface</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Wheel/Rail Interface</strong></td>
</tr>
<tr>
<td>Radius of wheel (passenger vehicle)</td>
</tr>
<tr>
<td>Hertzian spring constant</td>
</tr>
</tbody>
</table>

NB Wheel and rail profiles have been used from the Manchester Benchmarks and are available as computer files for participants
### 3.3. Track Parameters

#### Table 3.4. Concrete Sleepered Track Superstructure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail mass per meter</td>
<td>60.34 kg/m</td>
</tr>
<tr>
<td>Rail profile radius on top</td>
<td>0.30 m</td>
</tr>
<tr>
<td>Area of rail cross-section</td>
<td>7.69 x 10^{-3} m²</td>
</tr>
<tr>
<td>Elastic modulus of rail</td>
<td>2.07 x 10^{11} N/m²</td>
</tr>
<tr>
<td>Shear modulus of rail</td>
<td>8.1 x 10^{10} N/m²</td>
</tr>
<tr>
<td>Rail second moment of area – x axis</td>
<td>30.55 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Rail second moment of area – y axis</td>
<td>5.13 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Timoshenko shear coefficient</td>
<td>0.34</td>
</tr>
<tr>
<td>Poisson's ratio of rail</td>
<td>0.27</td>
</tr>
</tbody>
</table>

**RAIL PAD (Appendix D, Figure D.2)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical stiffness of pad</td>
<td>200 x 10^5 N/m</td>
</tr>
<tr>
<td>Vertical damping of pad</td>
<td>50 x 10^3 N.s/m</td>
</tr>
<tr>
<td>Rail Pad Depth (Average)</td>
<td>0.0075 m</td>
</tr>
<tr>
<td>Rail Seat Length</td>
<td>0.180 m</td>
</tr>
<tr>
<td>Rail Seat Width (Rail foot width)</td>
<td>0.150 m</td>
</tr>
</tbody>
</table>

**RAIL (Appendix C, Figure C.2)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail mass per meter</td>
<td>54.43 kg/m</td>
</tr>
<tr>
<td>Rail profile radius on top</td>
<td>0.305 m</td>
</tr>
<tr>
<td>Area of rail cross-section</td>
<td>6.93 x 10^{-3} m²</td>
</tr>
<tr>
<td>Elastic modulus of rail</td>
<td>2.07 x 10^{11} N/m²</td>
</tr>
<tr>
<td>Shear modulus of rail</td>
<td>8.1 x 10^{10} N/m²</td>
</tr>
<tr>
<td>Rail second moment of area – x axis</td>
<td>23.46 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Rail second moment of area – y axis</td>
<td>4.89 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Timoshenko shear coefficient</td>
<td>0.34</td>
</tr>
<tr>
<td>Poisson's ratio of rail</td>
<td>0.27</td>
</tr>
</tbody>
</table>

**FASTENER (Appendix D, Figure D.1 & D.2)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fastener stiffness</td>
<td>1.0 x 10^5 N/m</td>
</tr>
<tr>
<td>Fastener damping</td>
<td>0.4 x 10^3 N.s/m</td>
</tr>
</tbody>
</table>

**CONCRETE SLEEPER (Appendix E)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sleeper mass</td>
<td>314 kg</td>
</tr>
<tr>
<td>Sleeper spacing</td>
<td>0.610 m</td>
</tr>
<tr>
<td>Elastic Modulus of sleeper</td>
<td>38 x 10^9 N/m²</td>
</tr>
<tr>
<td>Density</td>
<td>2.5 x 10^3 kg/m^3</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.15</td>
</tr>
<tr>
<td>Timoshenko shear coefficient</td>
<td>0.833</td>
</tr>
</tbody>
</table>

#### Table 3.5. Timber Sleepered Track Superstructure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail mass per meter</td>
<td>54.43 kg/m</td>
</tr>
<tr>
<td>Rail profile radius on top</td>
<td>0.305 m</td>
</tr>
<tr>
<td>Area of rail cross-section</td>
<td>6.93 x 10^{-3} m²</td>
</tr>
<tr>
<td>Elastic modulus of rail</td>
<td>2.07 x 10^{11} N/m²</td>
</tr>
<tr>
<td>Shear modulus of rail</td>
<td>8.1 x 10^{10} N/m²</td>
</tr>
<tr>
<td>Rail second moment of area – x axis</td>
<td>23.46 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Rail second moment of area – y axis</td>
<td>4.89 x 10^{-6} m^4</td>
</tr>
<tr>
<td>Timoshenko shear coefficient</td>
<td>0.34</td>
</tr>
<tr>
<td>Poisson’s ratio of rail</td>
<td>0.27</td>
</tr>
</tbody>
</table>

**STEEL SLEEPER PLATE (Appendix D, Figure D.4)**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic modulus of steel sleeper plate</td>
<td>2.07 x 10^{11} N/m²</td>
</tr>
<tr>
<td>Sleeper Plate Stiffness</td>
<td>309.6 x 10^9 N/m</td>
</tr>
<tr>
<td>Sleeper Plate Damping</td>
<td>0 N.s/m</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sleeper Plate Depth (Average)</td>
<td>0.019 m</td>
</tr>
<tr>
<td>Property</td>
<td>Value</td>
</tr>
<tr>
<td>----------------------------------------------</td>
<td>----------------</td>
</tr>
<tr>
<td>Rail Seat Length</td>
<td>L_Spl</td>
</tr>
<tr>
<td>Rail Seat Width</td>
<td>W_Spw</td>
</tr>
<tr>
<td><strong>FASTENER</strong> (Appendix D, Figure D.3)</td>
<td></td>
</tr>
<tr>
<td>Fastener stiffness</td>
<td>K_Fx</td>
</tr>
<tr>
<td>Fastener damping</td>
<td>C_Fx</td>
</tr>
<tr>
<td><strong>TIMBER SLEEPER</strong></td>
<td></td>
</tr>
<tr>
<td>Sleeper mass</td>
<td>M_TS</td>
</tr>
<tr>
<td>Sleeper length</td>
<td>L_TSz</td>
</tr>
<tr>
<td>Sleeper width</td>
<td>L_TSx</td>
</tr>
<tr>
<td>Sleeper depth</td>
<td>L_TSy</td>
</tr>
<tr>
<td>Sleeper spacing</td>
<td>S_S</td>
</tr>
<tr>
<td>Vertical (Cross Grain) Elastic Modulus</td>
<td>E_Tsv</td>
</tr>
<tr>
<td>Longitudinal (Along the Grain) Elastic Modulus</td>
<td>E_Tsl</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho_{TS}$</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>$\nu_{TS}$</td>
</tr>
</tbody>
</table>

Table 3.6. Track Substructure

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>BALLAST</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Height of ballast</td>
<td>H_B</td>
<td>0.25</td>
</tr>
<tr>
<td>Elastic modulus of ballast</td>
<td>E_B</td>
<td>$130 \times 10^6$</td>
</tr>
<tr>
<td>Ballast Stiffness (Concrete Sleepered Superstructure)</td>
<td>K_B</td>
<td>$125 \times 10^6$</td>
</tr>
<tr>
<td>Ballast Damping (Concrete Sleepered Superstructure)</td>
<td>C_B</td>
<td>$310 \times 10^6$</td>
</tr>
<tr>
<td>Ballast Stiffness (Timber Sleepered Superstructure)</td>
<td>K_B</td>
<td>$85 \times 10^6$</td>
</tr>
<tr>
<td>Ballast Damping (Timber Sleepered Superstructure)</td>
<td>C_B</td>
<td>$220 \times 10^3$</td>
</tr>
<tr>
<td>Density of ballast</td>
<td>$\rho_B$</td>
<td>$1.40 \times 10^3$</td>
</tr>
<tr>
<td><strong>SUBBALLAST</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Height of subballast</td>
<td>H_SB</td>
<td>0.15</td>
</tr>
<tr>
<td>Elastic modulus of subballast</td>
<td>E_SB</td>
<td>$200 \times 10^6$</td>
</tr>
<tr>
<td>Subballast Stiffness (Concrete Sleepered Superstructure)</td>
<td>K_SB</td>
<td>$660 \times 10^6$</td>
</tr>
<tr>
<td>Subballast Damping (Concrete Sleepered Superstructure)</td>
<td>C_SB</td>
<td>$1.15 \times 10^6$</td>
</tr>
<tr>
<td>Subballast Stiffness (Timber Sleepered Superstructure)</td>
<td>K_SB</td>
<td>$500 \times 10^6$</td>
</tr>
<tr>
<td>Subballast Damping (Timber Sleepered Superstructure)</td>
<td>C_SB</td>
<td>$940 \times 10^3$</td>
</tr>
<tr>
<td>Density of subballast</td>
<td>$\rho_{SB}$</td>
<td>$2.00 \times 10^3$</td>
</tr>
<tr>
<td><strong>SUBGRADE</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Subgrade Modulus (Subgrade Stiffness per square metre)</td>
<td>E_Sub</td>
<td>$50 \times 10^6$</td>
</tr>
<tr>
<td>Subgrade Stiffness (Concrete Sleepered Superstructure)</td>
<td>K_Sub</td>
<td>$155 \times 10^6$</td>
</tr>
<tr>
<td>Subgrade Damping (Concrete Sleepered Superstructure)</td>
<td>C_Sub</td>
<td>$1.05 \times 10^6$</td>
</tr>
<tr>
<td>Subgrade Stiffness (Timber Sleepered Superstructure)</td>
<td>K_Sub</td>
<td>$135 \times 10^6$</td>
</tr>
<tr>
<td>Subgrade Damping (Timber Sleepered Superstructure)</td>
<td>C_Sub</td>
<td>$895 \times 10^3$</td>
</tr>
<tr>
<td>Density of Subgrade</td>
<td>$\rho_{Sub}$</td>
<td>1700</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>$\nu_{Sub}$</td>
<td>0.25</td>
</tr>
<tr>
<td>Shear Wave Velocity (models using halfspace subgrade)</td>
<td>$v_{shw}$</td>
<td>200</td>
</tr>
<tr>
<td>Elastic Modulus (models using halfspace subgrade only)</td>
<td>E_Subh</td>
<td>$170 \times 10^6$</td>
</tr>
</tbody>
</table>

NB Stiffness and damping values are calculated based on the theoretical assumptions presented in Sun & Dhanasekar (2002). Only models using a halfspace formulation for the subgrade should use the Shear Wave Velocity and Subgrade Elastic Modulus values.
4. RESULTS

It is requested that participants calculate the following and present a graph of results for each output parameter.

Table 4.1. Quantities to be calculated

<table>
<thead>
<tr>
<th>Code</th>
<th>Output Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Normal contact force between the wheels and rail</td>
<td>kN</td>
</tr>
<tr>
<td>B</td>
<td>Bending moment in the rail immediately above the sleeper</td>
<td>kNm</td>
</tr>
<tr>
<td>C</td>
<td>Vertical acceleration of the rail immediately above the sleeper</td>
<td>m/s²</td>
</tr>
<tr>
<td>D</td>
<td>Force in railpad at railseat on sleeper</td>
<td>kN</td>
</tr>
<tr>
<td>E</td>
<td>Bending moment in sleeper at railseat</td>
<td>kNm</td>
</tr>
<tr>
<td>F</td>
<td>Bending moment in sleeper at sleeper centre</td>
<td>kNm</td>
</tr>
<tr>
<td>G</td>
<td>Summation of the force transmitted from the ballast to the sleeper OR The peak pressure at the ballast / sleeper interface</td>
<td>kN</td>
</tr>
<tr>
<td></td>
<td></td>
<td>kN/m²</td>
</tr>
</tbody>
</table>

NOTE: Analysis should be undertaken at the point directly above “Sleeper A” as described in Table 2.5.

4.1. Time Domain Models

Participants that have time domain models are requested to undertake a total minimum run distance of 20m using a minimum time step of 0.0001 seconds (0.1 milliseconds).

We suggest that participants should provide an ‘Output Parameter vs Time’ graph and a data file detailing the results for each run combination (Table 2.1 & 4.1).

4.2. Frequency Domain Models

Participants that have frequency domain models participants should undertake the calculations which they think most relevant, and include (if possible) a description of what calculations have been undertaken in order to represent the situation described in above.

We suggest that participants should provide an ‘Output Parameter Amplitude vs Frequency’ graph and a data file detailing the results of each run combination (Table 2.1 & 4.1).

4.3. Format of Results

The data files provided should be in the form of a comma or tab delimited text file (or similar) and be named according to the model name, run number and output parameter code previously discussed (eg. Jens Nielsen’s DIFF, run number 2 and output B would be named ‘DIFF2B.txt’).
5. BENCHMARK TIMELINE

It would be appreciated if participants could have Part 1: Initial Simulation benchmark test results returned by January 9th 2004. During January 2004 supplementary benchmark instructions will be forwarded for Part 2: Field Data Simulations. Part 2 testing would most likely be completed by March 2004.

6. CONTACT DETAILS

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7. REFERENCES


8. ACKNOWLEDGEMENTS

The author of this Benchmark Instruction would like to thank the various contributors that assisted in the selection and format of information presented.

- Queensland University of Technology
- Central Queensland University
- Rail Infrastructure Corporation
- Australian Rail Track Corporation
- Pandrol
- Austrak
- BHP Billiton
- Manchester Metropolitan University – Rail Technology Unit
Appendix C1 – Passenger Vehicle Details

Figure A.1. Passenger Vehicle Bogie (side view)

Figure A.2. Passenger Vehicle Bogie (end view and section)
Appendix C2 – Wheel & Rail Profile Details (Iwnicki, 1998)

The measured wheel and rail profiles used in the Manchester Benchmarks should be used in this benchmark exercise, and are available as a file of x,y coordinates. The profiles are the common S1002 wheel profile and UIC60 rail section.

Wheel profile data

Left and right wheels are defined when facing the direction of travel. The profiles were measured close to the rail level with the wheelset in the loaded direction.

Lateral coordinates for both wheels are given relative to the mid point between the flangebacks. The positive direction is to the right when facing the direction of travel. Hence the lateral coordinate of the outside edge of the right wheel tread will have a large positive value and the lateral coordinate of the side edge of the left wheel tread will have a large negative value.

Vertical coordinates for both wheels are given relative to a fixed horizontal datum line through the trad of the wheels. The positive direction is radially outwards away from the axle. Hence the vertical coordinate of the wheel flange for both wheels will have a positive value and the outside edge of the wheel tread a small negative value.

Rail profile data

Left and right rails are defined when facing the direction of travel. The profiles were measured with the track in the unloaded condition.

Lateral coordinates for both rails are given to the mid point between the gauge faces. The positive direction is to the right when facing the direction of travel. Hence the lateral coordinate of the field side of the right rail will have a large positive value and the lateral coordinate of the field side of the left rail will have a large negative value.

Vertical coordinates for both wheels are given relative to a line parallel to the plane of the top of the rail crowns passing through approximately the bottom of the gauge corner of each rail. The positive direction is upwards. Hence the vertical coordinate of the crown of each rail will have a positive value and the vertical coordinate of the bottom of the gauge corner of each rail will have a small negative value.
Appendix C3 – Rail Section Details

Figure C.1. UIC 60 kg/m Rail
Figure C.2. UIC 54 kg/m Rail
Appendix C4 – Pandrol Fastening System Details

Figure D.1. Concrete Sleeper Fastening System

Figure D.2. Rail Pad and Insulator Details
Figure D.3. Timber Sleeper Fastening System

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.
Appendix C5 – Concrete Sleeper Details

Figure D.1. Standard Gauge Concrete Sleeper (30 Tonne – 60 kg/m Rail)
Appendix D

Equations used for the Worn Wheel Flat and Dipped Rail Joint or Weld Profiles
<table>
<thead>
<tr>
<th>Model</th>
<th>Worn Wheel Flat</th>
<th>Dipped Rail Joint or Weld</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>DARTS</strong></td>
<td>( z(x) = \frac{d}{2} \left( 1 - \cos \left( \frac{2\pi x}{a} \right) \right) )</td>
<td>( z(x) = d \left( 1 - \cos \left( \frac{\pi x}{L} \right) \right) ) for ( 0 &lt; x &lt; L / 2 )</td>
</tr>
<tr>
<td></td>
<td>( 0 &lt; x &lt; a )</td>
<td>( z(x) = d \left( 1 + \cos \left( \frac{\pi x}{L} \right) \right) ) for ( L / 2 &lt; x &lt; L )</td>
</tr>
<tr>
<td><strong>DIFF</strong></td>
<td>( z(x) = \frac{d}{2} \left( 1 + \cos \left( \frac{2\pi x}{a} \right) \right) )</td>
<td>( z(x) = \frac{2d}{L} \left( x \right) + d ) for ( -L / 2 &lt; x &lt; 0 )</td>
</tr>
<tr>
<td></td>
<td>( -a / 2 &lt; x &lt; a / 2 )</td>
<td>( z(x) = -\frac{2d}{L} \left( x \right) + d ) for ( 0 &lt; x &lt; L / 2 )</td>
</tr>
<tr>
<td><strong>NUCARS</strong></td>
<td>( z(x) = d - d \left[ e^{-a_2} + e^{-a_1} - 2e^{-2a_1} \right] ) for ( d = 0.38 \text{mm}, a_2 = 50 \text{mm}, a_1 = 5 \text{mm} )</td>
<td>( z(x) = d - d \left[ e^{-x_{L_2}/2} + e^{-x_{L_1}/2} - 2e^{-2x_{L_1}} \right] ) for ( L_2 = 1000 \text{mm}, L_1 = 216.22 \text{mm} )</td>
</tr>
<tr>
<td></td>
<td>( -a / 2 &lt; x &lt; a / 2 ) ( r = \text{radius of wheel} )</td>
<td></td>
</tr>
<tr>
<td><strong>SUBTTI</strong></td>
<td>( z(x) = \frac{d}{1 - \cos \left( \frac{a}{2r} \right)} \left( \cos \left( \frac{x}{r} \right) - \cos \left( \frac{a}{2r} \right) \right) ) for ( -a / 2 &lt; x &lt; a / 2 ) ( r = \text{radius of wheel} )</td>
<td>( z(x) = 0.0 ) for ( x &lt; x_m - \frac{L}{2} )</td>
</tr>
<tr>
<td></td>
<td>( x &lt; x_m - \frac{L}{2} ) ( x &gt; x_m + \frac{L}{2} )</td>
<td>( x_m = \frac{L}{2} ) for ( x_m &lt; x &lt; x_m + \frac{L}{2} )</td>
</tr>
<tr>
<td><strong>TRACK</strong></td>
<td>( z(x) = \frac{d}{2} \left( 1 - \cos \left( \frac{2\pi x}{a} \right) \right) )</td>
<td>( z(x) = \frac{ad}{2\pi} \left( 1 - \cos \left( \frac{\pi x}{L} \right) \right) ) for ( 0 &lt; x &lt; L / 2 )</td>
</tr>
<tr>
<td></td>
<td>( 0 &lt; x &lt; a )</td>
<td>( z(x) = \frac{ad}{2\pi} \left( 1 + \cos \left( \frac{\pi x}{L} \right) \right) ) for ( L / 2 &lt; x &lt; L )</td>
</tr>
<tr>
<td></td>
<td>( \alpha = \text{angle of ramp} )</td>
<td></td>
</tr>
<tr>
<td><strong>VICT</strong></td>
<td>( z(x) = \frac{d}{2} \left( 1 - \cos \left( \frac{2\pi x}{a} \right) \right) )</td>
<td>( z(x) = \frac{d}{2} \left( 1 - \cos \left( \frac{2\pi x}{L} \right) \right) ) for ( 0 &lt; x &lt; L )</td>
</tr>
<tr>
<td></td>
<td>( 0 &lt; x &lt; a )</td>
<td></td>
</tr>
</tbody>
</table>

* The NUCARS worn wheel flat is a variation of the wheel radius. All other model represent the worn wheel flat as a track irregularity.

Definitions except where noted in the table:

- \( d \) = Depth of worn wheel flat or dipped rail joint or weld
- \( a \) = Length of worn wheel flat
- \( L \) = Length of dipped rail joint or weld
Appendix E

Benchmark Test Results for Simulations 4 to 6 (Timber Sleepered Ballasted Track)
Appendix E1  Quasi-Static Forces when there are No Irregularities

4A - Wheel/Rail Contact Force for Bogie Leading Wheel

Figure E1.1 (a); 4A - Wheel/Rail Contact Force from Bogie Leading Wheel (Type 1)

* Distance marks = Position of sleepers

4A - Wheel/Rail Contact Force for the Bogie Leading Wheel (Type 1)

Figure E1.1 (b); 4A - Wheel/Rail Contact Force for the Bogie Leading Wheel (Type 1) (Vertical scale zoomed)
Figure E1.2; 4B - Rail Bending Moment at Sleeper A (Type 2)
Figure E1.3 (a); 4C - Rail Acceleration at Sleeper A (Type 2)

Figure E1.3 (b); 4C - Rail Acceleration at Sleeper A (Type 2) (Horizontal scale zoomed as shown in Figure E.3 (a))
Figure E1.4; 4D - Rail Pad Force on Sleeper A (Type 2)
Figure E1.5; 4E - Bending Moment in Sleeper A at the Rail Seat (Type 2)

Figure E1.6; 4F - Bending Moment in Sleeper A at the Centre (Type 2)
Figure E1.7 (a); 4G - Total Ballast Force on Sleeper A from two Rails (Type 2)

Figure E1.7 (b); 4G - Total Ballast Pressure on Sleeper A from two Rails (Type 2)
Appendix E2  Dynamic Wheel/Rail Forces from Worn Wheel Flats

Figure E2.1 (a); 5A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1)

Figure E2.1 (b); 5A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1) (Horizontal scale zoomed as shown in Figure E2.1 (a))
Figure E2.2 (a); 5B - Rail Bending Moment in Sleeper A (Type 2)
Figure E2.2 (b); 5B - Rail Bending Moment in Sleeper A (Type 3)

Figure E2.2 (c); 5B - Rail Bending Moment in Sleeper A (Type 3)
(Horizontal scale zoomed as shown in Figure E2.2 (a))
Figure E2.3 (a); 5C - Rail Acceleration at Sleeper A (Type 2)

* Time marks = Time when wheel passes over a sleeper

Figure E2.3 (b); 5C - Rail Acceleration at Sleeper A (Type 2)
(Horizontal scale zoomed as shown in Figure E2.3 (a))
**Figure E2.4 (a); 5D - Rail Pad Force at Sleeper A (Type 2)**

(Vertical and horizontal scale zoomed as shown in Figure E2.4 (a))
Figure E2.4 (c); 5D - Rail Pad Force at Sleeper A (Type 3)
(Vertical and horizontal scale zoomed as shown in Figure E2.4 (a))
Figure E2.5 (a); 5E - Bending Moment in Sleeper A at the Rail Seat (Type 2)
Figure E2.5 (b); 5E - Bending Moment in Sleeper A at the Rail Seat (Type 3)

Figure E2.5 (c); 5E - Bending Moment in Sleeper A at the Rail Seat (Type 3)

(Horizontal scale zoomed as shown in Figure E2.5 (b))
5F - Bending Moment in Sleeper A at the Centre

* Time marks = Time when wheel passes over a sleeper

Figure E2.6 (a); 5F - Bending Moment in Sleeper A at the Centre (Type 2)
Figure E2.6 (b); 5F - Bending Moment in Sleeper A at the Centre (Type 3)

Figure E2.6 (c); 5F - Bending Moment in Sleeper A at the Centre (Type 3)
(Horizontal scale zoomed as shown in Figure E2.6 (b))
Figure E2.7 (a); 5G - Total Ballast Force on Sleeper A from two Rails (Type 2)

Figure E2.7 (b); 5G - Total Ballast Force on Sleeper A from two Rails (Type 2)
(Vertical and horizontal scale zoomed as shown in Figure E2.7 (a))
Figure E2.7 (c): 5G - Total Ballast Pressure on Sleeper A (Type 2) (Vertical and horizontal scale zoomed as shown in Figure E2.7 (a))
Appendix E3  Dynamic Wheel/Rail Forces from Dipped Rail Joints

6A - Wheel/Rail Contact Force for Bogie Leading Wheel

Figure E3.1 (a); 6A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1)

Figure E3.1 (b); 6A - Wheel/Rail Contact Force for Bogie Leading Wheel (Type 1)  
(Horizontal scale zoomed as shown in Figure E3.1 (a))
Figure E3.2 (a); 6B - Rail Bending Moment in Sleeper A (Type 2)

Figure E3.2 (b); 6B - Rail Bending Moment in Sleeper A (Type 3)
Figure E3.3; 6C - Rail Acceleration at Sleeper A (Type 2)
Figure E3.4 (a); 6D - Rail Pad Force at Sleeper A (Type 2)

Figure E3.4 (b); 6D - Rail Pad Force at Sleeper A (Type 3)
(Vertical scale zoomed)
Figure E3.5 (a): 6E - Bending Moment in Sleeper A at the Rail Seat (Type 2)

Figure E3.5 (b): 6E - Bending Moment in Sleeper A at the Rail Seat (Type 3)
6F - Bending Moment in Sleeper A at the Centre

Figure E3.6 (a); 6F - Bending Moment in Sleeper A at the Centre (Type 2)

6F - Bending Moment in Sleeper at Centre (Dynamic Increment Only)

Figure E3.6 (b); 6F - Bending Moment in Sleeper A at the Centre (Type 3)
Figure E3.7 (a); 6G - Total Ballast Force on Sleeper A from two Rails (Type 2)

Figure E3.7 (b); 6G - Total Ballast Pressure on Sleeper A from two Rails (Type 2)
Appendix F

Summary of Benchmark Output Parameters Results for Simulations 1 to 6
### Table F1.1 Summary of Max/Min Output Results from Simulation Run 1

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL based on AS 1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS</th>
<th>SUBITI</th>
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*(Dynamic Increment only)*

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(Dynamic Increment Only)

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</tr>
<tr>
<td>G (kPa)</td>
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Table F5.2 Summary of Max/Min Output Results from Simulation Run 5
(Dynamic Increment)

<table>
<thead>
<tr>
<th>OUTPUT PARAMETER</th>
<th>EMPIRICAL based on AS1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS</th>
<th>SUBTI</th>
<th>TRACK*</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>B (kNm)</td>
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<td>3.9</td>
<td>7.3</td>
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<td>-7.3</td>
<td>-97.1</td>
<td>-</td>
<td>-6.5</td>
<td>-</td>
</tr>
<tr>
<td>D (kN)</td>
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<td>46.4</td>
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<td>1.8</td>
<td>11.7</td>
<td>-</td>
<td>1.6</td>
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<tr>
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<td>-3.3</td>
<td>-</td>
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<td>F (kNm)</td>
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<td>DIFF</td>
<td>NUCARS</td>
<td>SUBTI</td>
<td>TRACK*</td>
<td>VICT</td>
</tr>
<tr>
<td>------------------</td>
<td>--------------------------------</td>
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<td>------</td>
<td>--------</td>
<td>-------</td>
<td>--------</td>
<td>------</td>
</tr>
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<td>269.3</td>
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<td>B (kNm)</td>
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<td>-11.6</td>
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<td>C (m/s²)</td>
<td>-</td>
<td>1524.3</td>
<td>1035.9</td>
<td>1202.4</td>
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<td>7.0</td>
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<td>F (kNm)</td>
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<td>G (kN)</td>
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<td>144.0</td>
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<td>-0.1</td>
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<td>G (kPa)</td>
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<td>218.2</td>
<td>246.1</td>
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<td>-36.8</td>
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Table F6.2 Summary of Output Results from Simulation Run 6 (Dynamic Increment)

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<tr>
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<th>EMPIRICAL based on AS 1048.14</th>
<th>DARTS</th>
<th>DIFF</th>
<th>NUCARS</th>
<th>SUBITI</th>
<th>TRACK*</th>
<th>VICT</th>
</tr>
</thead>
<tbody>
<tr>
<td>B (kNm)</td>
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<td>-</td>
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<tr>
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<td>5.4</td>
<td>118.4</td>
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<td>2.1</td>
<td>-</td>
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<tr>
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<td>D (kN)</td>
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<td>57.4</td>
<td>27.3</td>
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</tr>
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<td>E (kNm)</td>
<td>3.7</td>
<td>6.6</td>
<td>4.7</td>
<td>11.4</td>
<td>-</td>
<td>5.0</td>
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<td>-2.4</td>
<td>-3.7</td>
<td>-</td>
<td>-2.0</td>
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<tr>
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<td>F (kNm)</td>
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<td>1.2</td>
<td>4.4</td>
<td>-</td>
<td>1.0</td>
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<td>-13.8</td>
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<td>-2.4</td>
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Appendix G

List of Input Parameters used by the DARTS program
<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Run &amp; write parameters</th>
<th>Units</th>
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</thead>
<tbody>
<tr>
<td>IRUN</td>
<td>0</td>
<td>DYNamic COMpliance analysis (DYNCOM)</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>DYNamaic analysis of Stationary Loads (DYNSL)</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>DYNamaic analysis of Moving Loads (DYNML) with vehicle</td>
<td>-</td>
</tr>
<tr>
<td>JRUN</td>
<td>0</td>
<td>Run whole program</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>Run program with input free vibration data generated from previous runs, can change parameters other than track parameters and vehicle damping</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Run program with input free vibration data generated from previous runs, can change parameters other than track parameters</td>
<td>-</td>
</tr>
<tr>
<td>IWR</td>
<td>0/1</td>
<td>Write rail mode curves (off/on)</td>
<td>-</td>
</tr>
<tr>
<td>IWT</td>
<td>0/1</td>
<td>Write tie mode curves (off/on)</td>
<td>-</td>
</tr>
<tr>
<td>IWFFT</td>
<td>0/1</td>
<td>Write an individual free-free tie mode curves (off/on)</td>
<td>-</td>
</tr>
<tr>
<td>NMODE0</td>
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<td>?</td>
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</tr>
<tr>
<td>NMODE1</td>
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<td>?</td>
<td>-</td>
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<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Rail &amp; sleeper type</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>IRAIL</td>
<td>1</td>
<td>Euler beam</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Timoshenko beam</td>
<td>-</td>
</tr>
<tr>
<td>ITIE</td>
<td>0</td>
<td>Solid mass</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>Euler beam</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Timoshenko beam</td>
<td>-</td>
</tr>
<tr>
<td>LTIE</td>
<td>1</td>
<td>Uniform sleeper cross-section</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Non-uniform sleeper cross-section</td>
<td>-</td>
</tr>
<tr>
<td>NTIE</td>
<td>&lt;37</td>
<td>Sleeper number for which calculation is undertaken (mode only)</td>
<td>-</td>
</tr>
<tr>
<td>IPAX</td>
<td>0/1</td>
<td>Rail axial force indicator - (off/on)</td>
<td>-</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Rail data</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>RE</td>
<td></td>
<td>Rail elastic Modulus</td>
<td>N/m²</td>
</tr>
<tr>
<td>RMU</td>
<td></td>
<td>Rail Poisson’s ratio</td>
<td>-</td>
</tr>
<tr>
<td>SHK</td>
<td></td>
<td>Timoshenko shear coefficient</td>
<td>-</td>
</tr>
<tr>
<td>RDEN</td>
<td></td>
<td>Rail density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>RI</td>
<td></td>
<td>Rail moment of inertia</td>
<td>m⁴</td>
</tr>
<tr>
<td>RA</td>
<td></td>
<td>Rail cross-sectional area</td>
<td>m²</td>
</tr>
<tr>
<td>SL</td>
<td></td>
<td>Sleeper spacing</td>
<td>m</td>
</tr>
<tr>
<td>PAX</td>
<td></td>
<td>Axial force in rail</td>
<td>N</td>
</tr>
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<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Rail section data for stress &amp; strain cal.</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>RY</td>
<td></td>
<td>Neutral axis to any point in rail cross-section</td>
<td>m</td>
</tr>
<tr>
<td>RQ</td>
<td></td>
<td>Section modulus of head</td>
<td>m³</td>
</tr>
<tr>
<td>RB</td>
<td></td>
<td>Web width at neutral axis</td>
<td>m</td>
</tr>
</tbody>
</table>

<table>
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<th>Units</th>
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</thead>
<tbody>
<tr>
<td>NOT</td>
<td>&lt;37</td>
<td>Total number of sleepers in model - must be &lt; 37</td>
<td></td>
</tr>
<tr>
<td>TL</td>
<td></td>
<td>Half length of sleeper</td>
<td>m</td>
</tr>
<tr>
<td>TLS</td>
<td></td>
<td>Length of sleeper shoulder</td>
<td>m</td>
</tr>
<tr>
<td>Code</td>
<td>Input</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>------</td>
<td>-------</td>
<td>------------------------------------------------------------------------------</td>
<td>--------</td>
</tr>
<tr>
<td>TB</td>
<td></td>
<td>Breadth of sleeper</td>
<td>m</td>
</tr>
<tr>
<td>TD</td>
<td></td>
<td>Depth of sleeper at centre</td>
<td>m</td>
</tr>
<tr>
<td>TDS</td>
<td></td>
<td>Depth of sleeper at shoulder</td>
<td>m</td>
</tr>
<tr>
<td>DG</td>
<td></td>
<td>Half gauge</td>
<td>m</td>
</tr>
<tr>
<td>TI</td>
<td></td>
<td>Moment of inertia at centre</td>
<td>m^4</td>
</tr>
<tr>
<td>TIS</td>
<td></td>
<td>Moment of inertia at shoulder</td>
<td>m^4</td>
</tr>
<tr>
<td>TDEN</td>
<td></td>
<td>Sleeper density</td>
<td>kg/m^3</td>
</tr>
<tr>
<td>TE</td>
<td></td>
<td>Sleeper elastic modulus</td>
<td>MN/m</td>
</tr>
<tr>
<td>TMU</td>
<td></td>
<td>Sleeper Poisson’s ratio</td>
<td>N/m^2</td>
</tr>
<tr>
<td>TSHK</td>
<td></td>
<td>Timoshenko shear coefficient for sleeper cross-section</td>
<td>N/m/m</td>
</tr>
<tr>
<td>EIT</td>
<td></td>
<td>Flexural rigidity at centre (TI*TE)</td>
<td>MN/m^2</td>
</tr>
<tr>
<td>EIS</td>
<td></td>
<td>Flexural rigidity at shoulder (TIS*TE)</td>
<td>MN/m^2</td>
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<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Rail pad and ballast stiffness</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>PK</td>
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<td>Pad stiffness</td>
<td>N/m</td>
</tr>
<tr>
<td>SKS</td>
<td></td>
<td>Track bed stiffness at shoulder</td>
<td>N/m/m</td>
</tr>
<tr>
<td>SK</td>
<td></td>
<td>Track bed stiffness at centre</td>
<td>N/m/m</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Frequency range &amp; interval for determining free vibration frequencies</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>RFREQM</td>
<td>Max frequency for rail free vibration</td>
<td>Hz</td>
<td></td>
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<tr>
<td>RDELF</td>
<td>Frequency interval for rail free vibration</td>
<td>Hz</td>
<td></td>
</tr>
<tr>
<td>TFREQM</td>
<td>Max frequency for sleeper free vibration</td>
<td>Hz</td>
<td></td>
</tr>
<tr>
<td>TDELF</td>
<td>Frequency interval for sleeper free vibration</td>
<td>Hz</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Rail pad and ballast damping values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>PC</td>
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<td>Pad damping</td>
<td>N.s/m</td>
</tr>
<tr>
<td>SC</td>
<td></td>
<td>Track bed damping (centre &amp; shoulder)</td>
<td>N.s/m</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Modes coupling parameter and method of integration</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>ICOUP</td>
<td>0</td>
<td>Uncoupled modes, proportional damping</td>
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<tr>
<td></td>
<td>1</td>
<td>Coupled modes, un-proportional damping</td>
<td>-</td>
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<tr>
<td>METHOD</td>
<td>GAU</td>
<td>Gauss Quadrature</td>
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<td></td>
<td>RK0</td>
<td>Runge-Kutta fixed step method</td>
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<td>RK1</td>
<td>Runge-Kutta varying step method</td>
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<tr>
<td>XP</td>
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<td>Position on rail where responses are calculated</td>
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<tr>
<td>NNTIE</td>
<td></td>
<td>Sleeper number where responses are calculated</td>
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</tr>
<tr>
<td>TY</td>
<td></td>
<td>Position on Sleeper where responses are calculated</td>
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</tr>
</tbody>
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<table>
<thead>
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<th>Units</th>
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<tr>
<td>IV</td>
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<td>Vehicle type (1-DOF)</td>
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<tr>
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<td>2</td>
<td>Vehicle type (2-DOF)</td>
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<table>
<thead>
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<th>Code</th>
<th>Input</th>
<th>Description</th>
<th>Units</th>
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<tr>
<td>LD</td>
<td>0</td>
<td>Unit impulsive load</td>
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<tr>
<td></td>
<td>1</td>
<td>Stationary arbitrary load</td>
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</tr>
<tr>
<td></td>
<td>2</td>
<td>Moving arbitrary load</td>
<td>-</td>
</tr>
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<td>DT</td>
<td></td>
<td>Integration time step</td>
<td>-</td>
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<td>TDUR</td>
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<td>Total duration of force for LD=0,1</td>
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</tr>
<tr>
<td>Code</td>
<td>Input</td>
<td>Unit</td>
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</tr>
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<td>-------</td>
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</tr>
<tr>
<td>VEL</td>
<td>Velocity</td>
<td>m/s</td>
<td></td>
</tr>
<tr>
<td>DELT</td>
<td>Time Step</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>NW</td>
<td>Time Step write number (every NW steps)</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Code</th>
<th>Input</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>IR</td>
<td>Wheel/rail irregularity data</td>
<td>-</td>
</tr>
<tr>
<td>XIRR</td>
<td>distance to midpoint of irregularity (IRR = 0,2 XIRR = 0.0)</td>
<td>m</td>
</tr>
<tr>
<td>WRAD</td>
<td>Wheel radius</td>
<td>m</td>
</tr>
<tr>
<td>FL</td>
<td>Irregularity length (wavelength)</td>
<td>m</td>
</tr>
<tr>
<td>FD</td>
<td>Irregularity depth (amplitude)</td>
<td>m</td>
</tr>
<tr>
<td>NPOINT</td>
<td>Number of points in arbitrary profile</td>
<td>-</td>
</tr>
</tbody>
</table>
Appendix H

Flowchart for the Creation of the DARTS.INP file
Run & Write Selection

Rail & Sleeper Selection

Rail Data

Data Points on Rail Selection

Sleeper Data

Pad & Track Damping Data

Coupling Parameters & Method of Integration

Rail & Sleeper Analysis Positions

Vehicle Selection & Data

Wheel/Rail Irregularity Selection, Position & Data

IRR
Type of Wheel or Rail Irregularity

IV
Type of Vehicle

METHOD
Type of Integration to be used

ICOUPL
Type of Coupling and Damping

IRAIL
Type of Rail Simulation

ITIE
Type of Sleeper Simulation

LTIE
Type of Sleeper Cross-Section

DYNML

RA
RA1 or RA2

SL
Sl2 or Sl4

B

0 to I

I to 17
DATA POINTS ON RAIL SECTION for stress & strain calculations

**SLEEPER DATA**

**Solid Mass**

- **ITIE = 0**
  - Total Number of Ties
  - Half length of Sleeper
  - Sleeper Breadth
  - Sleeper Depth at Centre
  - Sleeper Density
  - Pad Stiffness
  - Track Bed Stiffness

- **ITIE = 1**
  - Total Number of Ties
  - Half length of Sleeper
  - Sleeper Breadth
  - Sleeper Depth at Centre
  - Sleeper Density
  - Pad Stiffness
  - Track Bed Stiffness

**Uniform Euler Beam**

- **ITIE = 1**
  - Total Number of Ties
  - Half length of Sleeper
  - Sleeper Breadth
  - Sleeper Depth at Centre
  - Tie moment of inertia
  - Sleeper Density
  - Sleeper Elastic Modulus
  - Sleeper Poisson's Ratio
  - EIT = TI*TE
  - Flexural Rigidity of Sleeper

**Non-Uniform Euler Beam**

- **ITIE = 1**
  - Total Number of Ties
  - Length of Shoulder
  - Sleeper Depth at Shoulder
  - Tie moment of inertia at Centre
  - Sleeper Density
  - Sleeper Elastic Modulus
  - Sleeper Poisson's Ratio
  - EIS = TIS*TE
  - Flexural Rigidity of Sleeper at Centre
### WHEEL OR RAIL IRREGULARITY SELECTION & POSITION

**IRUN** = 2
**IV** = 4

**IR** (Irregularity)
- 0 – Smooth Rail
- 1 – Wheel Flat
- 2 – Corrugation
- 3 – Dipped Joint
- 4 – Hollow Weld
- 5 – Humped Weld
- 6 – Arbitrary Wheel Surface Profile
- 7 – Arbitrary Rail Surface Profile

**Type of Wheel or Rail Irregularity**

**XIRR** (Location of Irregularity)
- **IRR** = 0, 1, 2, 3, 4, 5
- Midpoint of Irregularity
- **IRR** = 6 or 7
- Start Point of Irregularity

**Irregularity**
- 0 to 7

**VEL** (Velocity of moving load)

**DELT** (Time step of moving load)

**NW** (Time step report number)

**WRAD** (Wheel Radius)

**FL** (Corrugation Wave Length)

**FD** (Amplitude of Corrugation)

**NPOINT** (Number of Points in Irregularity)

**LINE, XPOINT, YPOINT**
- Line Number
- X and Y Coordinates of Arbitrary Profile

**IRR** (Irregularity)
- 0 – Smooth Rail
- 1 – Wheel Flat
- 2 – Corrugation
- 3 – Dipped Joint
- 4 – Hollow Weld
- 5 – Humped Weld
- 6 – Arbitrary Wheel Surface Profile
- 7 – Arbitrary Rail Surface Profile
Appendix I

Instructions for the Installation of the DARTS Interface
Instructions for the Installation of the DARTS Interface

The DARTS program is run through a windows interface that may be installed on any computer that has Microsoft Windows XP installed. To install the DARTS Interface Software, please follow the instructions below:

1. Ensure you have the right to install programs on your computer (administration rights). You can not install this program unless you have these rights.

2. Insert the DARTS Interface disk attached to this thesis into your CD drive and an installation screen will appear after a few seconds.

3. If you do not have the ‘Dot Net Framework’ installed on your computer you will be warned to install it before continuing. This program may be found on the DARTS Interface disk in the folder titled ‘Dot Net Framework v1.1’. Run the DOTNETFX.EXE file in this folder to install this program and follow the prompts.

4. Once the DOT NET Framework has been installed on your computer continue to follow the DARTS interface installation prompts as required and complete the installation.

5. You should now have a ‘DARTS v1.0’ shortcut on your desktop, run this to load the program.

PLEASE NOTE:

The program provided is ONLY THE DARTS INTERFACE, Dr Zhenqi Cai's actual DARTS program is not included, and has been replaced by a simulation executable to act like his program. Therefore no results are available for examination.

Questions on installation may be directed to DAVID STEFFENS on (07) 3235 2718 or 0404 064 574.