Heavy Vehicle Suspensions – Testing and Analysis.

A thesis submitted for the degree of Doctor of Philosophy

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“All those that have raised themselves a name by their ingenuity, great poets, and celebrated historians, are most commonly, if not always, envied by a sort of men who delight in censuring the writings of others, though they never publish any of their own.” - Miguel de Cervantes
Keywords

Heavy vehicle; truck; lorry; suspension test; Vehicle Standards Bulletin 11 (VSB 11); suspension health; suspension model; dynamic force; wheel force; pavement force; tyre force; spatial repeatability; spatial repetition; road roughness; pavement roughness; surface roughness; suspension metrics; on-board mass; heavy vehicle telematics; tamper evidence; tamper metrics; load sharing; dynamic load sharing; heavy vehicle suspension frequency; heavy vehicle suspension wavelength; heavy vehicle suspension model; suspension software model.

Abstract

Transport regulators consider that, with respect to pavement damage, heavy vehicles (HVs) are the riskiest vehicles on the road network. That HV suspension design contributes to road and bridge damage has been recognised for some decades. This thesis deals with some aspects of HV suspension characteristics, particularly (but not exclusively) air suspensions. This is in the areas of developing low-cost in-service heavy vehicle (HV) suspension testing, the effects of larger-than-industry-standard longitudinal air lines and the characteristics of on-board mass (OBM) systems for HVs. All these areas, whilst seemingly disparate, seek to inform the management of HVs, reduce of their impact on the network asset and/or provide a measurement mechanism for worn HV suspensions. A number of project management groups at the State and National level in Australia have been, and will be, presented with the results of the project that resulted in this thesis. This should serve to inform their activities applicable to this research.

A number of HVs were tested for various characteristics. These tests were used to form a number of conclusions about HV suspension behaviours.

Wheel forces from road test data were analysed. A “novel roughness” measure was developed and applied to the road test data to determine dynamic load sharing, amongst other research outcomes. Further, it was proposed that this approach could inform future development of pavement models incorporating roughness and peak wheel forces. Left/right variations in wheel forces and wheel force variations for
different speeds were also presented. This led on to some conclusions regarding suspension and wheel force frequencies, their transmission to the pavement and repetitive wheel loads in the spatial domain.

An improved method of determining dynamic load sharing was developed and presented. It used the correlation coefficient between two elements of a HV to determine dynamic load sharing. This was validated against a mature dynamic load-sharing metric, the dynamic load sharing coefficient (de Pont, 1997). This was the first time that the technique of measuring correlation between elements on a HV has been used for a test case vs. a control case for two different sized air lines.

That dynamic load sharing was improved at the air springs was shown for the test case of the large longitudinal air lines. The statistically significant improvement in dynamic load sharing at the air springs from larger longitudinal air lines varied from approximately 30 percent to 80 percent. Dynamic load sharing at the wheels was improved only for low air line flow events for the test case of larger longitudinal air lines. Statistically significant improvements to some suspension metrics across the range of test speeds and “novel roughness” values were evident from the use of larger longitudinal air lines, but these were not uniform. Of note were improvements to suspension metrics involving peak dynamic forces ranging from below the error margin to approximately 24 percent.

Abstract models of HV suspensions were developed from the results of some of the tests. Those models were used to propose further development of, and future directions of research into, further gains in HV dynamic load sharing. This was from alterations to currently available damping characteristics combined with implementation of large longitudinal air lines.

In-service testing of HV suspensions was found to be possible within a documented range from below the error margin to an error of approximately 16 percent. These results were in comparison with either the manufacturer’s certified data or test results replicating the Australian standard for “road-friendly” HV suspensions, Vehicle Standards Bulletin 11.

OBM accuracy testing and development of tamper evidence from OBM data were detailed for over 2000 individual data points across twelve test and control OBM
systems from eight suppliers installed on eleven HVs. The results indicated that 95 percent of contemporary OBM systems available in Australia are accurate to +/- 500 kg. The total variation in OBM linearity, after three outliers in the data were removed, was 0.5 percent. A tamper indicator and other OBM metrics that could be used by jurisdictions to determine tamper events were developed and documented. That OBM systems could be used as one vector for in-service testing of HV suspensions was one of a number of synergies between the seemingly disparate streams of this project.
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Role of publications that contributed to this project

A number of activities and data gathering processes contributed to the project *Heavy vehicle suspensions – testing and analysis*. That project started as a joint QUT/Main Roads project. Main Roads (MR) was merged with Queensland Transport on 1 March 2009 and became the Department of Transport and Main Roads (DTMR).

Some of these activities and the data therefrom have been documented previously as shown in the publications in Appendix 5. To ensure that this thesis is a stand-alone document, some repetition of the concepts from those documents (significant references follow) was necessary in this thesis.

The work of the project was in the areas of developing low-cost in-service heavy vehicle (HV) suspension testing (Davis & Bunker, 2008a), the effects of larger-than-industry-standard longitudinal air lines (Davis & Bunker, 2008d), and the characteristics of on-board mass (OBM) systems for HVs (Davis, Bunker, & Karl, 2009; Karl, Cai, Koniditsiotis *et al.*, 2009a; Karl, Davis, Cai *et al.*, 2009b). The documents produced in the lead up to, and during, this project sought to inform the management of HVs, reduce their impact on the network asset and/or provide measurement mechanisms for worn HV suspensions. State and National project management groups seeking to better manage HVs have been, and will be, informed by the result from this project, including the Australian Treasury (Clarke & Prentice, 2009) in its efforts to explore mass-distance charging for heavy vehicles.

Suspension data for three heavy vehicles (HV) were gathered before and during the inception stage of the joint QUT/Main Roads (now DTMR) project with a confluence of events leading to the start of the project proper on 1 August 2007. Some of this inception work resulted in papers written and presented very shortly after the project started but prepared prior to enrolment (Davis, Kel, & Sack, 2007) as well as subsequently (Davis & Bunker, 2008d).

The on-board mass data were gathered during the project and, due to administrative and organisational arrangements involving necessary separation and/or essential overlaps in roles and data sharing between QUT, Transport Certification Australia (TCA) and MR/DTMR, not all of that data were relevant to this thesis. Accordingly, the publications in Appendix 5 were co-authored by different participants; any given publication was, properly, not necessarily co-authored by all concerned.
Statement of original authorship

The work contained in this thesis has not been previously submitted to meet requirements for an award at this or 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.

Lloyd Davis

Signature

31st May 2010
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The following people and their organisations deserve a vote of thanks:

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- Queensland Transport staff who provided in-kind support;

- the RTA of NSW for additional funding when the test programme vehicle numbers increased by two more than in the original budget;

- Mylon Motorways staff for sourcing drivers and buses and for technical assistance;

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Transport Certification Australia staff for their continued faith in our combined test programmes.

I would like to convey my sincere and deep appreciation to my family and friends for continuing to provide to me unlimited patience, tolerance and encouragement.

To my wife especially, thank you for having the faith that I could discover something worthwhile on this journey and the inner strength that you used to carry us both to this point.

The definition of faith is belief without proof. Paradoxically, unaccompanied faith in the existence of the unknown fails the rational warrant to investigate. New territories therefore lie unexplored where voyages to them cannot be justified by reason. Hence, journeys of discovery proceed on faith alone.
1 Introduction and problem definition

Road transport in Australia is essentially an economic argument. Governments at all levels provide and maintain road network assets. Provision of these assets incurs costs. The assets are provided within an engineering framework that includes economic considerations. The consumer of the transport service is not necessarily levied the actual cost of the transport task (Productivity Commission, 2006). Responsible asset consumption, cost recovery mechanisms and equity thereof, user charges and cross-subsidisation all form the basis for on-going debate that properly includes the economics that has always been part of good engineering practice.

1.1 About this chapter

The purpose of this chapter is to introduce the research framework for this thesis. It presents the broad setting and history that has lead to the problem statements for the research. How these problem statements have been taken up and have driven the study is then presented as the set of research aims for the project Heavy vehicle suspensions – testing and analysis. That project originated as a joint QUT/Main Roads project. Main Roads was merged with Queensland Transport on 1 March 2009 and became the Department of Transport and Main Roads.

1.2 Background

Regulators make decisions on heavy vehicle (HV) access. These decisions are partially informed by examining the parameters and metrics of those HVs. Some of these are obvious such as static axle mass or rollover threshold. Some are not as obvious such as the “road-friendliness” of suspensions or whether suspension designs reduce dynamic pavement loading. Static parameters are more easily specified and measured than dynamic measures. This thesis concentrates on some dynamic properties of HV suspensions with a view to informing the transport industry and jurisdictions on engineering and related issues.
1.3 Load sharing

1.3.1 Evolution from static to dynamic load sharing

Load sharing can be defined as the equalisation of the axle group load across all wheels or axles. A variation on that definition is that a HV with a “load equalising system” needs to have, p. 26 (Stevenson & Fry, 1976):

- an axle group [that] utilises a suspension with the same spring types on each axle; and
- a design that delivers “substantially equal sharing by all the ground contact surfaces of the total load carried by that axle group”.

Soon after this study, early efforts to define “load-sharing” in Australia were made (Australia Department of Transport, 1979). The suspension on the right in Figure 1.1 and Figure 1.2 is an example of a centrally pivoted suspension although the one shown is not the only expression of this design. It is apparent from Figure 1.1 and Figure 1.2 that load sharing was seen at the time to be a static or quasi-static phenomenon. The suspension on the left was defined as non-load sharing because of effect shown in Figure 1.2. This was recognised by Sweatman (1983) as only part of the issue. Sweatman (1983) as well as others (Cole & Cebon, 1991) contended that centrally-pivoted suspensions with inadequate damping by design would be less “road-friendly”. The report concluded that centrally pivoted suspensions caused underdamped transmission of front-axle perturbations to the rear axle via the rocker-arm mechanism, leading to high wheel forces.

![Figure 1.1. Early attempts to define load sharing (Australia Department of Transport, 1979).](image-url)
1.3.2 Regulatory framework

Road authorities are concerned about the damage HVs do to the road network asset. One of these concerns arises from any imbalance in loads between wheels in a multi-axle HV suspension group. It is generally regarded as beneficial for a multi-axle HV suspension group to share the load evenly across all the wheels. Any imbalance results in one wheel creating more pavement force than do the others. To this end, the Australian specification for “road-friendly” HV suspensions, Vehicle Standards Bulletin 11 (Australia Department of Transport and Regional Services, 2004a) and the Australian vehicle standards (Australia, 1999) specify limits of imbalance between wheels in a HV axle group.

The methodology within the regulations for determining compliance with load sharing in HV suspensions uses static processes. Curiously, Vehicle Standards Bulletin 11 (VSB 11) does not specify a methodology for determining static load sharing (Prem, Mai, & Brusza, 2006), Further, this specification does not address dynamic load sharing at all.

Regulators specify HV mass limits in terms of static wheel loads. In contrast to the use of static performance measures, vehicles bounce as they travel. This creates dynamic forces within HV frames and imparts dynamic forces to the pavement. Regulations or standards in Australia do not specify dynamic parameters for HV suspensions in general and dynamic load sharing in particular.

HVs with air suspensions and carrying increased loads were introduced to Australia at the end of the 1990s as part of the micro-economic reform fashionable at the time (see below). At that time, it was known that concomitant increases in dynamic wheel loads would result from the inability of these suspensions to share loads in the
dynamic sense. This inability was recognised as having the potential to cause greater road damage than might otherwise be the case should air-sprung HVs incorporate more dynamic load equalisation into their design (OECD, 1992, 1998). With the clarity of hindsight, the disbenefits resulting from heavier air suspended HVs, due to higher road network asset damage, were probably not recognised as discounting somewhat the societal and economic benefits of higher payloads.

Three-axle HV semi-trailer groups are commonplace and quad-axle semi-trailers are being introduced on the Australian eastern seaboard. That air-sprung HV multi-axle groups do not load share in the dynamic sense has been demonstrated (Davis & Sack, 2004). There is now a growing recognition of the phenomenon of insufficient dynamic load sharing within air-sprung HV suspension groups. This phenomenon, particularly for quad-axle semi trailers, is of increasing interest to transport regulators (Blanksby, George, Peters et al., 2008a; Blanksby, George, Peters et al., 2008b; Blanksby, Gemanchev, Ritzinger et al., 2009).

1.3.3 Dynamic load sharing metrics for HV suspensions

Blanksby (2007) stated that there was no dynamic load sharing measure for HVs. Others (Patrick, Gemanchev, Elischer et al., 2009) have stated that expert opinion has it that load sharing in the dynamic sense cannot be achieved by current HV suspension designs (Patrick et al., 2009). Despite this, de Pont (1997) and other researchers (Cebon, 1999; Potter, Cebon, Collop et al., 1996; Sweatman, 1983) have been investigating dynamic load sharing of HVs for almost three decades.

Further to the previous work on load sharing, de Pont (1997) developed two dynamic load sharing metrics but only one, the dynamic load sharing coefficient (DLSC), was applicable to HV axle groups with more than two axles. It was able to be applied per wheel or per axle (de Pont, 1997) but not per group without aggregating the results across wheels or axles.

The most widely used load-sharing measure is the load sharing coefficient (LSC). This metric is not the most useful and has been criticised since it is really the average load sharing behaviour of the group (de Pont, 1997). LSC averages the forces on a
wheel over a test run and is derived \textit{per} wheel or \textit{per} axle. These factors render difficult a judgement regarding the quality of a HV axle group’s load sharing ability in totality since six individual and different LSC measures arise from a tri-axle group test run, for instance. However, little work has been done on replacement for this metric (de Pont, 1997). Even the DLSC provides a number of different values for each test run requiring further aggregation to present the data in plain format.

Greater emphasis on, and specification of, the dynamic load sharing ability and other dynamic parameters of air suspensions is required.

1.3.4 Dynamic load sharing systems

Larger longitudinal air lines on air sprung HVs have been developed (Davis, 2006a, 2007; Davis & Kel, 2007; Willox, 2005). Anecdotal evidence suggests that use of these systems can improve dynamic load sharing and reduce dynamic forces both within the HV and transmitted to the pavement \textit{via} the HV’s wheels. Alterations to these forces may lead to potential savings in HV suspension designs as well as savings on structures, surfacings and pavement maintenance. Alterations in these forces (and the quantum of such alterations) from the use of such systems had not been determined rigorously until the data for this project were gathered.

Noting that the system tested for this project altered the size of longitudinal air lines on a HV, side-to-side load sharing is counterproductive to HV handling, resulting in promotion of roll.

1.3.5 Problem statement # 1

\textit{A dynamic load sharing metric without the drawback of aggregating data across wheels or axles does not exist for axle groups of more than two axles.}
1.3.6 **Problem statement # 2**

Whether fitting larger longitudinal air lines to air-sprung HVs alters dynamic forces at the axle-to-body interface has not been determined adequately.

1.4 **In-service HV suspension testing**

1.4.1 **Higher Mass Limits, history and imperatives**

The transport industry exerts continuous pressure on road authorities and transport regulators to allow “freight efficient” vehicles (for which read fewer drivers, more trailers, more axles and higher axle masses) onto the network.

As part of the micro-economic reform of the 1980s and 1990s the Australian Government commissioned the mass limits review (MLR) project undertaken by the National Road Transport Commission (Pearson & Mass Limits Review Steering Committee, 1996a). The MLR project concluded that HVs operating at higher mass limits (HML) and equipped with “road friendly” suspensions (RFS) would be no more damaging to the road network asset than conventional HVs operating at statutory mass with conventional steel springs (Pearson & Mass Limits Review Steering Committee, 1996a). Further, the project report stated that HV air suspensions in poor condition would damage the infrastructure more than HV steel suspensions and recommended, amongst other things, that eligibility to operate at HML was dependent on maintenance of dampers. This last point was due to the heavy reliance air suspensions place on suspension dampers (shock absorbers) for correct damping. This was in contrast to HV steel suspensions that possess intrinsic residual damping via the Coulomb friction between the leaves of the steel springs (Prem, George, & McLean, 1998).

Supported by the findings of the final report of the DIVINE project (OECD, 1998), the findings of the MLR report led to a reform termed “higher mass limits” implemented under the “second heavy vehicle reform package” (National Transport Commission, 2003). Details on the original HML access and conditions when implemented varied between Australian States and still do. However, in terms of additional mass, HML generally allowed increases above statutory mass of Δ2.5t on
a HV tri-axle group and Δ0.5t on a HV tandem axle group.

In return for HML payloads, HVs carrying them needed to be fitted with RFS that had been certified as meeting the requirements of VSB 11. This compliance with VSB 11 was for new suspensions only. VSB 11 was released in 1999 and revised subsequently (Australia Department of Transport and Regional Services, 2004a, 2004c).

That HVs were permitted to carry greater mass in return for, amongst other requirements, being equipped with RFS was the first indication that specific axle-mass increases were to be tied to vehicle design improvements. This approach signalled to the transport industry there would be minimal scope for any further blanket increases in gross vehicle mass (GVM). Further, any economic benefits from increases in GVM were not necessarily going to be balanced against the increasing costs of maintenance and capital for infrastructure capable of carrying heavier HVs. It was an acknowledgement that the road network asset had reached a point where any further gains in HV productivity would need to be traded off against more efficient boutique vehicles with improved design. That the network would reach this point had been foreseen (Sweatman, 1994) as well as the prediction of incentives to encourage HV characteristics which did not consume the asset as quickly as in the past (Woodrooffe, LeBlanc, & Papagiannakis, 1988).

Interestingly, a decade on from the release of VSB 11, the Austroads guideline for HV access to local roads (Geoff Anson Consulting & InfraPlan [Aust], 2009) states, on p. 16: “It is now generally recognised by road authorities that large parts of the road network infrastructure have reached its capacity in being able to handle heavy vehicle operation on a general access basis.”

1.4.2 Higher Mass Limits and suspension health

The Mass Limits Review Report and Recommendations by the National Road Transport Commission (Pearson & Mass Limits Review Steering Committee, 1996a) found that HV air suspensions in poor condition would damage the infrastructure more than HV steel suspensions and recommended, amongst other things, that
eligibility to operate at HML was dependent on maintenance of dampers. Regulators have become increasingly concerned about this point in recent years.

It is for noting that the only HVs meeting the requirements for RFS in 1999 were air-suspended although steel suspensions meeting the RFS standard have been released to the market in the intervening years (Australia Department of Transport and Regional Services, 2004b).

The DIVINE report showed the results of ineffective shock absorbers on HV wheel loadings. Figures IV.33 and IV.34 on p. 99 of the DIVINE report, taken from Woodroffe (1997), are shown in Figure 1.3. These plots show that, over and above any static load, when the wheels of a loaded HV were subjected to a 1mm sinusoidal input, dynamic wheel loads increased by $\Delta 50kN$ for the case where an air suspension had ineffective shock absorbers.
The Marulan survey – snapshot of HV suspension health in Australia

The Roads and Traffic Authority (RTA) of NSW commissioned a survey of HVs to determine in-service compliance to RFS requirements at its Marulan checking station in 2006. 121 air-sprung HVs were tested to the requirements of VSB 11 in the survey.

Since Marulan was not on a HML route at the time of the survey, it could be argued that some of the HVs surveyed at Marulan were not RFS compliant by choice of non-HML load. This idea needs to be balanced against the concept that the transport industry does not allocate HVs with RFS for HML duty only. HVs are purchased with RFS on the basis that some of their activity will be at HML, the rest of the time.
they operate at statutory mass. Unofficial estimates put the number of RFS suspension sold in Australia per year at 90 percent to 95 percent of the total (Patrick et al., 2009). Actual figures are difficult to verify since members of the HV suspension supply industry are reluctant to release official numbers due to the highly competitive nature of the industry in Australia.

A sample of only 68 HVs with RFS would have provided a 10 percent tolerance level for the Marulan data (Blanksby, George, Germanchev et al., 2006). Taking the lowest estimate of RFS-equipped HVs from above, a sample of 121 HVs would have yielded 108 RFS-equipped HVs; well above the 68 that would have provided a statistical tolerance level of 10 percent (Blanksby et al., 2006) for the survey result. From the statistically valid sample of 121 air-sprung HVs, approximately 50 percent did not meet the damping values specified in VSB 11. Further, 16 percent did not meet the requirements for frequency values specified in VSB 11. The results from the Marulan survey indicated strongly that air-sprung HVs were not having their “road friendliness” maintained during normal work. Dr. Cebon, one of the authors of the Sweatman et al., (2000) report investigating in-service testing of RFS suspensions on HVs had already recommended, at the international level, type-testing of RFS using parametric or other means combined with annual in-service testing (Cebon, 1999). Others (Potter, Cebon, & Cole, 1997; Woodrooffe, 1995) had already proposed similar concepts.

In parallel with, but retrospectively encouraged by, the results at Marulan, the States of Queensland and New South Wales included the development and application of an in-service HV suspension test in their respective Bilateral Infrastructure Funding Agreements with the Australian Government (Australia Department of Transport and Regional Services, 2005a, 2005b).

All Australian State Governments have a Bilateral Infrastructure Funding Agreement (BIFA) with the Australian Government. These are also known as the “AusLink agreements”. Each BIFA is an agreement between individual States of Australia and the Commonwealth and covers arrangements applying to Australian Government funding to all Australian States under the first five-year AusLink investment programme (2004-05 to 2008-09) and any agreed modifications thereto. They also cover actions for infrastructure planning, prioritisation of infrastructure investments.
opportunities, development and assessment of project proposals and evaluation of completed projects (Australia Department of Transport and Regional Services, 2005b).

1.4.4 Higher Mass Limits and a “road friendly” suspension test

In return for extending available HML routes, the States of Queensland and NSW both agreed with the Commonwealth that an in-service test for “road-friendly” HV suspensions would be developed and implemented (Australia Department of Transport and Regional Services, 2005a, 2005b). A project by the National Transport Commission continues. This thesis and its originating project *Heavy vehicle suspensions – testing and analysis* continue to support the National Transport Commission (NTC) in its endeavours. The road freight industry works to tight financial margins. Any test for shock absorber health will drive up transport costs. Accordingly, low cost is beneficial.

1.4.5 Problem statement - in-service HV suspension testing

*In Australia at present, there is no:*

- requirement to have air-sprung HV air suspensions comply with the Australian specification for “road-friendly” suspensions, VSB 11 (Australia Department of Transport and Regional Services, 2004a) once the HV is in service;

- recognised low-cost HV suspension test.

1.5 On-board mass monitoring of HVs

1.5.1 The Intelligent Access Program

Transport Certification Australia Limited (TCA) was created to certify service-providers of HV telematics and administer the Intelligent Access Program (IAP). Stage 1 of the IAP monitors HV location, time, speed, tamper-evidence, and
proprietary trailer identification (Davis, Bunker, & Karl, 2008a).

1.5.2 On-board mass management - program

An alteration to the focus of TCA occurred with respect to on-board mass (OBM) monitoring. As mentioned in Section 1.4.4, the funding arrangements between the Australian Government and the various State Governments are covered, in part, by each State’s BIFA. The BIFAs for NSW and Queensland from 2004-05 to 2008-09 (Australia Department of Transport and Regional Services, 2005a, 2005b) included obligations requiring HML vehicles to be monitored by the IAP for position and, importantly for this thesis, on-board mass monitoring.

1.5.3 On-board mass monitoring

TCA, jointly with the NTC, has undertaken a project to investigate the feasibility of on-board vehicle mass-monitoring devices to be incorporated into Stage 2 of the IAP (Davis et al., 2008a). On-board mass monitoring (OBM) increases jurisdictional confidence in operational HV compliance. This research project has identified technical issues regarding on-board mass monitoring systems including:

- determination of HV mass using OBM systems at an evidentiary level (i.e. accurate enough to be used as evidence in a prosecution);
- accuracy, robustness and tamper issues of OBM components (mass sensors, connections, power supply, display unit etc.); and
- potential use of dynamic data to cross check measurement results from OBM systems.
1.5.4 Problem statement – on-board mass monitoring of HVs

Tamper-evidence and accuracy of current OBM systems needed to be determined before any regulatory schemes were put in place for its use. There are no regulatory on-board mass schemes for HVs operating anywhere in the world. Australia, via the Intelligent Access Programme (Davis et al., 2008a; Karl, 2007; Karl, Davis, Cai et al., 2009; Transport Certification Australia, 2005) is examining the feasibility of such implementation.

1.6 Research aims

The following Research Aims have been developed from the problem identification in Sections 1.3 to 1.5.

1.6.1 Aim 1: Dynamic load sharing 1

Hypothesis:

“An improved dynamic load sharing measure can be developed for heavy vehicle suspensions.”

The research aimed to test this hypothesis by:

- examining the existing load sharing measures available for heavy vehicles;
- determining whether these are suitable and; if not
- developing a dynamic load sharing parametric measure (or measures) that can be applied to consecutive wheels or axles.
1.6.2 Aim 2: Dynamic load sharing 2

Hypothesis:

“Alterations to dynamic parameters, particularly dynamic load sharing, occur from the use of larger longitudinal air lines in air-suspended HVs.”

The research project aimed to:

- determine if larger air lines on air sprung HVs make a difference to wheel force parameters and axle-to-body force parameters and, if so;
- determine the quantum of such alterations to wheel forces and axle-to-body forces from the use of larger longitudinal air lines.

1.6.3 Aim 3: In-service HV suspension testing

Hypothesis:

“A low-cost in-service suspension test for HVs is possible.”

The research project aimed to explore low-cost methods to evaluate

- if equivalent outcomes to VSB 11 testing for body bounce and damping ratio may be achieved. This for validity, via “proof-of-concept” of the “pipe test” (Davis & Bunker, 2008a) consisting of driving the HV over a 50mm steel pipe and analysing the body bounce and damping; and
- the development of a modified brake tester to impart resonant forces into HV suspensions.

The latter would be used to:

- compare previous work on wheel forces from suspensions in good condition vs. those equipped with poor/worn shock absorbers, such as in the lower window of Figure 1.3, after Woodrooffe (1997); and
determine a “proof-of-concept” for a second, low-cost HV suspension tester to inform the NTC project referenced above.

1.6.4 Aim 4: On-board mass monitoring of HVs – search for accuracy and tamper-evidence

Hypothesis:

“On-board mass measurement systems for HVs are accurate and tamper-evident for Australian regulatory purposes.”

The research project aimed to determine:

- the accuracy of currently-available on-board mass (OBM) systems by analysis of weighbridge vs. OBM reading;
- the use of dynamic data from OBM systems to detect tamper events; and
- whether dynamic data from OBM systems could be used to derive static HV mass.

1.7 Objectives

A programme was established to address the Research Aims in Section 1.6 to meet the following objectives.

1.7.1 Objective 1 – dynamic load sharing metric

Develop at least one dynamic load sharing parametric measure that can be applied to the axle group, consecutive axles or to consecutive wheels and for more than two axles.
1.7.2 **Objective 2 – differences for larger longitudinal air lines**

Determine if larger air lines on air sprung HVs make a difference to wheel forces and axle-to-body forces and, if so, determine the quantum of such alterations from the use of larger longitudinal air lines.

1.7.3 **Objective 3 – development of in-service suspension test(s)**

Explore low-cost HV suspension test methods. Evaluate if low-cost HV suspension test methods can be made equivalent to VSB 11 outcomes for body bounce and damping ratio. Evaluate the validity, *via* “proof-of-concept” of the “pipe test” (Davis & Bunker, 2008e). Develop a modified roller-brake tester to impart resonant forces into a HV suspension and, in part, validate previous work on wheel forces from suspensions in good condition *vs.* those equipped with poor/worn shock absorbers (Woodroofe, 1997). Determine, *via* “proof-of-concept”, that a modified roller-brake tester may be used to detect a worn HV suspension compared with one within specification. This latter to inform the NTC project referenced above.

1.7.4 **Objective 4 – on-board mass measurement feasibility**

Determine the accuracy of currently-available OBM systems. Examine the accuracy of OBM readings *vs.* weighbridge readings. Validate the use of dynamic data from OBM systems to detect tamper events.
1.8 **Scope, definitions, conventions and limitations of the study**

1.8.1 **Glossary, terms, acronyms and abbreviations**

<table>
<thead>
<tr>
<th>Terms, abbreviations and acronyms</th>
<th>Meaning</th>
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<tr>
<td>AASHO</td>
<td>American Association of State Highway Officials. Became AASHTO.</td>
</tr>
<tr>
<td>AASHTO</td>
<td>American Association of State Highway and Transportation Officials.</td>
</tr>
<tr>
<td>ALF</td>
<td>Accelerated loading facility.</td>
</tr>
<tr>
<td>APT</td>
<td>Air pressure transducer. A device that converts an air pressure signal to a proportional electrical signal.</td>
</tr>
<tr>
<td>ARRB</td>
<td>Australian Road Research Board – now privatised, has changed its name to ARRB Group Limited.</td>
</tr>
<tr>
<td>ARRB</td>
<td>Australasian Road Research Board.</td>
</tr>
<tr>
<td>ARTSA</td>
<td>Australian Road Transport Suppliers Association.</td>
</tr>
<tr>
<td>ATRF</td>
<td>Australasian Transport Research Forum. A conference for presentation of papers and colloquia on matters of transport planning, policy and research.</td>
</tr>
<tr>
<td>Axle-hop</td>
<td>Vertical displacement of the wheels (and axle), indicating dynamic behaviour of the axle and resulting in more or less tyre force onto the road. Usually manifests in the frequency range 10 – 15Hz.</td>
</tr>
<tr>
<td>BIFA</td>
<td>Bilateral Infrastructure Funding Agreement. Also known as “AusLink agreement”. An agreement between individual States of Australia and the Commonwealth which “covers arrangements applying to funding made available by the Australian Government to Queensland under the first five-year AusLink investment programme (2004-05 to 2008-09) and any agreed subsequent changes to, and extensions of, the programme. It also covers agreed arrangements for infrastructure planning, identification of investment priorities, development and assessment of project proposals and evaluation of completed projects.” (Australia Department of Transport and Regional Services, 2005b). Queensland’s BIFA may be viewed at: <a href="http://www.auslink.gov.au/publications/policies/pdf/Queensland_bilateral.pdf">http://www.auslink.gov.au/publications/policies/pdf/Queensland_bilateral.pdf</a>.</td>
</tr>
<tr>
<td>Term</td>
<td>Definition</td>
</tr>
<tr>
<td>----------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Body bounce</td>
<td>Movement of the sprung mass of a truck as measured between the axles and the chassis. Results in truck body dynamic forces being transmitted to the road via the axles &amp; wheels. Usually manifests in the frequency range 1 – 4Hz.</td>
</tr>
<tr>
<td>CAPTIF</td>
<td>Canterbury Accelerated Pavement Testing Indoor Facility.</td>
</tr>
<tr>
<td>CoG</td>
<td>Centre of gravity. The point at which a body’s mass may be said be concentrated for purposes of determining forces on that body.</td>
</tr>
<tr>
<td>∆</td>
<td>Greek letter “delta” – denoting increment.</td>
</tr>
<tr>
<td>Damping ratio</td>
<td>How rapidly a dynamic reduces impulsive response. It is found from the ratio of the peak of one dynamic cycle to the peak in the next cycle. The damping ratio, zeta (ζ) is given as a value under 1 (e.g. 0.3) or a percentage (e.g. 30 percent).</td>
</tr>
<tr>
<td>DIF</td>
<td>Dynamic impact factor (Woodrooffe &amp; LeBlanc, 1987). See also PDWF &amp; PDLR.</td>
</tr>
<tr>
<td>DIVINE</td>
<td>The Dynamic Interaction between Vehicles and Infrastructure Experiment. The Dynamic Interaction between Vehicles and Infrastructure Experiment (DIVINE) Project was formulated to report on dynamic effects of heavy vehicles on infrastructure to inform transport policy decisions regarding that infrastructure and road transport costs.</td>
</tr>
<tr>
<td>DLC</td>
<td>Dynamic load coefficient (Sweatman, 1983).</td>
</tr>
<tr>
<td>DLSC</td>
<td>Dynamic load sharing coefficient (de Pont, 1997).</td>
</tr>
<tr>
<td>Dot operator</td>
<td>Denotes derivative with respect to time. Where (say) x is a time-dependent variable, $\frac{dx}{dt}$ denotes the derivative or the rate of change with respect to time. This concept is sometimes designated $\dot{x}$ as a shortened form. The derivative with respect to time of $\dot{x}$, that is $\frac{d^2x}{dt^2}$ or acceleration, is designated $\ddot{x}$.</td>
</tr>
<tr>
<td>DoTaRS</td>
<td>Department of Transport and Regional Services. An Australian Government department.</td>
</tr>
<tr>
<td>DTMR</td>
<td>Department of Transport and Main Roads. Queensland Government Department formed from the amalgamation of the Departments of Main Roads and Queensland Transport in March 2009.</td>
</tr>
<tr>
<td>Eigenfrequency</td>
<td>Frequency of a body at one of its vibrational resonance modes.</td>
</tr>
<tr>
<td>ESA</td>
<td>Equivalent standard axle.</td>
</tr>
<tr>
<td>EU</td>
<td>European Union.</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier transform. A method whereby the Fourier transform is found using discretisation and conversion into a frequency spectrum (see Fourier transform).</td>
</tr>
<tr>
<td>Term</td>
<td>Definition</td>
</tr>
<tr>
<td>------</td>
<td>------------</td>
</tr>
<tr>
<td>Fourier transform</td>
<td>A method whereby the relative magnitudes of the frequency components of a time-series signal are converted to, and displayed as, a frequency series (Jacob &amp; Dolcemascolo, 1998).</td>
</tr>
<tr>
<td>GVM</td>
<td>Gross vehicle mass.</td>
</tr>
<tr>
<td>HML</td>
<td>Higher Mass Limits. Under the HML schemes in Australia, heavy vehicles are allowed to carry more mass (payload) in return for their suspension configuration being “road friendly”. See VSB 11.</td>
</tr>
<tr>
<td>HV</td>
<td>Heavy vehicle.</td>
</tr>
<tr>
<td>Hz</td>
<td>Hertz. Unit of vibration denoting cycles per second. Units are s⁻¹.</td>
</tr>
<tr>
<td>IRI</td>
<td>International roughness index.</td>
</tr>
<tr>
<td>LSC</td>
<td>Load sharing coefficient. A measure of how well a suspension group equalises the total axle group load, averaged during a test. This is a value which shows how well the average forces of a multi-axle group are distributed over each tyre &amp;/or wheel in that group (Potter et al., 1996).</td>
</tr>
<tr>
<td>MCV</td>
<td>Multi-combination vehicle. HVs with general arrangement or GVM greater than that of a semi-trailer.</td>
</tr>
<tr>
<td>MLR</td>
<td>Mass Limits Review. The national project that resulted in the implementation of HML in Australia.</td>
</tr>
<tr>
<td>NHVAS</td>
<td>National Heavy Vehicle Accreditation Scheme. A voluntary scheme that certifies transport operators against a set of industry-specific quality assurance requirements. Membership of this scheme is a pre-requisite for HML.</td>
</tr>
<tr>
<td>NRTC</td>
<td>National Road Transport Commission. A national body set up by the States of Australia to facilitate economic reform of the road transport industry. Became the NTC earlier this decade.</td>
</tr>
<tr>
<td>NSW</td>
<td>New South Wales.</td>
</tr>
<tr>
<td>NTC</td>
<td>See NRTC.</td>
</tr>
<tr>
<td>OECD</td>
<td>Organisation for Economic Co-operation and Development.</td>
</tr>
<tr>
<td>OBM</td>
<td>On-board mass. Systems for measuring the mass of a heavy vehicle using on-vehicle telematics.</td>
</tr>
<tr>
<td>PDF</td>
<td>Peak dynamic force.</td>
</tr>
<tr>
<td>PDLR</td>
<td>Peak Dynamic Load Ratio (Fletcher, Prem, &amp; Heywood, 2002).</td>
</tr>
<tr>
<td>PDWF</td>
<td>Peak dynamic wheel force. The maximum wheel force experienced by a wheel during dynamic loading as a result of a step input (Fletcher et al., 2002). If applied to axle forces, this measure is related to dynamic impact factor (DIF) as the numerator in the equation.</td>
</tr>
<tr>
<td>PSD</td>
<td>Power spectral density.</td>
</tr>
<tr>
<td>QUT</td>
<td>Queensland University of Technology.</td>
</tr>
<tr>
<td>-----</td>
<td>-------------------------------------</td>
</tr>
<tr>
<td>RFS</td>
<td>“Road-friendly” suspension. A HV suspension conforming to the performance requirements defined by VSB 11.</td>
</tr>
<tr>
<td>RSF</td>
<td>Road stress factor. An estimation of road damage due to the 4th power of instantaneous wheel force (Potter <em>et al.</em>, 1997).</td>
</tr>
<tr>
<td>RTA</td>
<td>Roads and Traffic Authority (NSW).</td>
</tr>
<tr>
<td>Shock absorber</td>
<td>See suspension damper.</td>
</tr>
<tr>
<td>Sigma ($\sigma$)</td>
<td>Greek symbol lower-case ‘sigma’, denoting standard deviation.</td>
</tr>
<tr>
<td>Spatial repeatability</td>
<td>The tendency for HV suspensions with similar characteristics to concentrate wheel forces at particular points on any given length of road.</td>
</tr>
<tr>
<td>Suspension damper</td>
<td>A device used to provide vibration damping to the suspension of a vehicle. This device reduces perturbations in suspension travel over time. It damps out axle-hop; contributes to tyre contact with the pavement during travel over undulations and during braking.</td>
</tr>
<tr>
<td>TCA</td>
<td>Transport Certification Australia.</td>
</tr>
<tr>
<td>TMR</td>
<td>See DTMR.</td>
</tr>
<tr>
<td>WiM</td>
<td>Weigh-in-motion. Technology that uses sensors in the road to measure the wheel force of vehicles.</td>
</tr>
</tbody>
</table>
1.8.2 Scope

The scope of the project was limited to measurement of data at the wheels and the springs of heavy vehicles. This was to:

- develop low-cost methods for in-service testing of HV suspensions as alternatives to the method defined in the Australian specification for “road friendly” suspensions Vehicle Standards Bulletin 11 (VSB 11);

- determine whether larger longitudinal air lines fitted to HV air spring suspensions altered forces at the springs or the wheels and, if so, by how much;

- derive frequency-series data from the testing to inform the activities of the project;

- develop HV suspension models to support the above activities and alter their parameters to determine areas for future HV suspension development;

- determine the accuracy and precision of on-board mass (OBM) telematics for HVs; and

- determine whether a tamper indicator could be developed for OBM systems for HVs.

The study considered legal (statutory mass and HML) heavy vehicle loads. Road network damage due to overloading was not part of the study parameters.

1.8.3 Numbering convention

In general, numbers have been shown to three significant figures. In places where this has not occurred, it was due to the derived values being sensitive to the least (usually fourth) significant figure or where mass could be measured to 0.1 kg.
1.9 Outline of the research methodology

1.9.1 The scientific method

The standard experimental methodology of performing a test with a known or standard system, changing one feature, running the experiment again and then comparing the two sets of data was performed. Thus the standard scientific method where the results from a control case were compared with those from a test case (Mill, 1872) was implemented for the experimental design and the analysis of the data.

1.9.2 Dynamic load sharing metric

Currently available load sharing metrics were reviewed. By deriving suspension metrics from the data collected during the testing (Davis & Bunker, 2008e), it was apparent that a dynamic load sharing parameter using instantaneous data was available. This was developed further and the results documented.

1.9.3 Differences for larger longitudinal air lines

Three air-suspended HVs were instrumented to measure air spring and wheel forces during typical travel. Two sets of tests were conducted. The control case had standard air lines and the test case was for larger longitudinal air lines between the air springs. Suspension parameters for load sharing and dynamic forces were derived. By comparing these for the test case vs. the control case, conclusions were drawn. Using the suspension parameters derived from a combination of manufacturer’s data and observed behaviour during other testing (Section 1.9.4), the fundamental parameters of the HV suspension were determined. This allowed a computer model of a theoretical HV suspension to be created. By varying the load-sharing ability of this model, further improvements in HV suspension design were apparent. These were documented but not developed as this line of research was tending out-of-scope.
1.9.4 Development of in-service suspension test(s)

Method 1. Using the testing of three HVs described above, a novel suspension test, the “pipe test” was performed. The control test was a replication of the step-down test in the Australian specification for “road-friendly” suspensions, VSB 11. The results from the novel test were compared with those from the VSB 11 test.

Method 2. A used FKI Crypton HV brake tester (FKI Crypton Ltd, 1990) was modified and instrumented. A test HV wheel was excited by two flat areas on the roller when it was spun up to speed. Instrumentation measured the wheel forces thus created. Dampers with differing levels of wear were fitted. The control case was for a new shock absorber, the two test cases were for a damper worn to the point where tyre degradation was occurring and the other test case was for no shock absorber. The wheel forces were analysed for magnitude and frequency for the cases tested. Threshold values for shock absorber wear vs. wheel forces were determined as a “proof-of-concept” in-service HV suspension test. This informed the NTC project and assisted in proposing low-cost HV air-suspension test methods.

1.9.5 On-board mass feasibility

The testing program was conducted in line with Davis et al. (Davis et al., 2008a; Davis, Bunker, & Karl, 2008b). Twelve test and control OBM systems from eight suppliers were installed on eleven HVs. The HVs were loaded to tare, 1/3, 2/3 and full load points. The HVs were then weighed six times per load point on weighbridges after a short road circuit. Dynamic data were recorded for the road circuits. The weighbridge readings were compared with the static OBM readings. A range of dynamic data was recorded as the test vehicles encountered speed bumps, braking and cornering.

Tests were performed to determine potential for, and ease of, tampering. Various tamper tests were performed; the one that concerns this thesis directly was that where the air lines to the OBM primary transducer were closed off. Dynamic data were recorded for open and closed air line states. The two sets of data were compared and an algorithm developed to detect the difference.
1.10 Structure of the thesis

This thesis is structured in chapters. The chapter numbers and their titles are as shown in the diagram, Figure 1.4. The diagram defines the interconnections and precedent/outcome succession within the methodology used to achieve the objectives, described above.
Figure 1.4. Diagram showing structure of thesis.
1. **Introduction and problem definition** (this chapter) sets the scene for the thesis. It describes the aims, objectives, background, limitations and scope of the project. Definitions, abbreviations and acronyms are set out.

2. **Review of heavy vehicle suspension metrics** contains a summary of measures and parameters used by regulators, researchers and the heavy vehicle industry in their attempts to quantify the relative values of HV suspensions.

3. **Data collection** documents the test programme methodology used to gather the data for the project.

4. **Development of heavy vehicle suspension models** covers the process where HV suspensions were conceptualised and then models assembled for use in the project.

5. **Heavy vehicle suspension model calibration and validation** applies the data gathered for the project to the concept models developed in the previous chapter. This to calibrate and validate the HV suspension models.

6. **Quasi-static testing** provides results from the quasi-static test programme. It also uses the models developed, calibrated and validated in the previous chapters. This to determine some metrics for the suspensions tested.

7. **On-road testing and roller bed test analysis** provides results from the roller-bed testing and the on road test programme, save for the OBM testing. It also details the development of an innovative metric relating to roughness. This is then used to illustrate wheel loadings and discuss pavement damage model parameters.

8. **On-board mass system characterisation** describes the accuracy, precision and other characteristics determined from testing of OBM systems.

9. **Development and validation of a tamper metric** provides the rationale for, and subsequent development and validation of, an indicator of tampering with on-board mass (OBM) telematics systems where the air lines to the air-pressure transducers have been restricted.

10. **Efficacy of larger longitudinal air lines** details derived measures from the testing of the control case of standard air lines vs. larger longitudinal air lines. Further, it details a novel dynamic load sharing test.
11. **HV suspension in-service testing** discusses how various testing methods, particularly low-cost ones, may be applied to in-service testing of HV suspensions and the implications of this approach within a regulatory framework.

12. **Contribution to knowledge – industrial practice** details the contribution to knowledge, from this project, that is readily applicable to industrial practice. It is in four Sections:
   - **Alterations to HV suspensions** explores the possibilities discovered during the analysis of the different HV suspension types and configurations. Further, by varying the internal parameters of the models developed in Chapters 4 and 5, theoretical improvements are mooted for novel suspension configurations.
   - **Application of in-service HV suspension testing** provides a framework for applying the in-service suspension tests developed earlier.
   - **Implications for network assets** discusses the issues related to poor HV suspension performance if allowed to deteriorate from the new state.
   - **Application of OBM to HV monitoring policy** explores the possibilities of OBM as a regulatory tool.

13. **Contribution to knowledge – theory and future work** details the contribution to theoretical knowledge from this project. It also provides suggestions to researchers who may wish to develop some of the concepts explored in this thesis in academic, theoretical or abstract variable-space in the future.

14. **Conclusions and recommendations** summarises the work and how the objectives were met.
2 Partial literature review of heavy vehicle suspension metrics

2.1 About this chapter
The purpose of this chapter is to summarise the suspension metrics used later in this thesis. It is not exhaustive and the interested reader is referred to the literature review (Davis & Bunker, 2007) for more information.

This chapter presents novel work on load sharing from the project and sets the scene for the development of models with respect to dynamic load sharing, suspension design and road damage.

2.2 Introduction to this review
This review is presented in three parts; one part relating to temporal measures, the second concerning spatial measures and the third presenting a study into the load sharing coefficient (LSC) and its relationship to dynamic load coefficient (DLC). New material developed since the literature review of this project (Davis & Bunker, 2007) is included, noting particularly the spatial correlation work outlined in Section 2.4.5 (Blanksby, Germanchev, Ritzinger et al., 2009) and the study of DLS vs. LSC (Davis & Bunker, 2009d) in Section 2.5.

The split between the first two sections reflects the two major research philosophies regarding heavy vehicle (HV) dynamics and their impact on pavements. The temporal approach (section 2.3) tends to see HV dynamics in terms of inter-axle or inter-wheel interactions. The spatial approach (section 2.4) views dynamic forces from HVs in terms of their wheel forces passing a point on the road, particularly where homogeneity of suspension types or specifications increases the probability of wheel force repetition at a particular point of the pavement. Such forces are generated by dynamic response due to a defect in the pavement at some previous point or from axle-hop due to defective dampers. Australian research in the spatial metric domain has had an addition since the literature review for this project was completed. This work is reviewed in Section 2.4.5.

Road asset owners are primarily concerned with pavement damage caused by HVs.
Subtleties of suspension design are not the intrinsic focus of this approach but are a means to minimise asset degradation. Accordingly, pavement damage models as they relate to HV suspension designs and metrics are covered in this chapter to the extent that they are related.

2.3 Temporal measures

The following measures for HV suspensions are designated ‘temporal’ (Cebon, 1987) since they are dependent on the forces on the chassis or wheels within a particular history.

2.3.1 Damping ratio

The damping ratio in the context of a HV suspension is a measure of how quickly a HV body returns to steady state motion after encountering a bump in the road. Damping ratio is designated by the Greek letter zeta (ζ), is dimensionless and usually shown as a number less than 1.0 (e.g. 0.3) or as a percentage (e.g. 30 percent) denoting the damping present in the system as a fraction of the critical damping value (Doebelin, 1980).

Apart from other specified parametric thresholds, HV “road friendly” suspensions are required to have:

- a damping ratio, zeta (ζ), of greater than 0.2 or 20 percent with dampers fitted; and

- a contribution of more than 50 percent to the overall viscous damping value of the measured damping ratio from the dampers (Australia Department of Transport and Regional Services, 2004a).
2.3.2 Damped natural frequency

The damped natural frequency of the sprung mass (body), in the HV suspension environment, is the measure of how many times per second the HV body bounces after some perturbation (e.g. bump in the pavement).

For a HV suspension to be defined as “road-friendly”, not only does the damping ratio have to exceed certain values (above), but also the damped free vibration frequency ($f$) of body bounce needs to be less than 2.0 Hz.

Davis and Sack (Davis & Sack, 2004, 2006) as well as Sweatman (Sweatman, 1983) have shown the magnitude of parameters of interest (including body bounce but also axle-hop, etc.) either as a frequency series using the output of a fast-Fourier transform (FFT) or as a power spectral density (PSD) against frequency (Gyenes & Simmons, 1994; Jacob & Dolcemascolo, 1998; LeBlanc & Woodrooffe, 1995; OECD, 1998). Essentially, the two methods (PSD vs. FFT) do not differ in their application to finding resonant frequencies. The vertical scales differ between the two methods in that the vertical axis of the PSD graph is proportional to the square of the magnitude of the FFT graph (Vernotte, 1999).

2.3.3 Digital sampling of dynamic data – Shannon’s theorem (Nyquist criterion)

Following from the previous section on frequency and to inform the rationale behind data sampling choices detailed later in this thesis (Chapter 3), an expansion on digital sampling theory follows (Davis & Bunker, 2007).

In order to re-create a signal with frequency components from 0 Hz to $f_i$ Hz, where $f_i$ Hz is the maximum frequency of interest, the Shannon-Nyquist sampling theorem states that the sampling rate, $f_s$, must be a minimum of $2 \times f_i$ Hz (Considine, 1985).

Noting that frequency, $\omega$, in radians is related to conventional frequency $f$ in Hz by:

$$\omega = 2\pi f$$

**Equation 2.1**
then the following proof may be considered.

Let $T$ be the system sampling time in seconds (s) of a continuous time-series signal where the highest frequency of interest in the sampled signal is $\omega_i \text{ rad.s}^{-1}$ and $\theta$ is the phase angle. The continuous time-series signal may be represented by:

$$e(t) = \sin(\omega t + \theta)$$

Equation 2.2

Let the sampling frequency in radians be $\omega_s$, where $0 < \omega_i < \omega_s/2$.

The sampling frequency, $\omega_s$, relates to conventional sampling frequency, $f_s$, in Hz by:

$$2\pi f_s = \omega_s = \frac{2\pi}{T}.$$  

$$=> \quad T = \frac{2\pi}{\omega_s}$$

Equation 2.3

Let $t = kT$ where $k$ is a constant.

Substituting $kT$ for $t$ in Equation 2.2, the equation for the sampled signal may be represented:

$$=> \quad e(kT) = \sin(k\omega T + \theta)$$

Equation 2.4

Now consider a different continuous time-series signal with higher frequencies where the higher frequencies contain a component $n\omega_s$ in addition to $\omega_s$ and represented by:

$$f(t) = \sin[(\omega_i + n\omega_s)t + \theta]$$

Equation 2.5

where $n = 1, 2, 3…$

Substituting $2\pi/T$ for $\omega_s$ from Equation 2.3, Equation 2.5 becomes:

$$f(t) = \sin\left[(\omega_i + 2\times\pi\times\frac{n}{T})t + \theta\right]$$

Equation 2.6
for which the sampling time \( T \) is as above.

Substituting \( kT \) for \( t \) as before, the equation for the reconstructed signal is then:

\[
f(kT) = \sin \left( \omega_i + \frac{2\pi \times n}{T}kT + \theta \right)
\]

\[
=> f(kT) = \sin(kT\omega_i + \theta)
\]

Equation 2.7

As the signal frequencies in Equation 2.6 increase proportional to \( 2\pi/T \), the reconstructed signal in Equation 2.7 becomes indistinguishable from Equation 2.4, even though the frequency of the signal has been increased.

Hence, \( \omega_i < \omega_s/2 \) is the criterion for the process of regular time-based sampling. At the limit, as \( \omega_i \) approaches \( n\omega_s/2 \), or:

\[
\omega_i \lim_{n\omega_s/2}
\]

the reconstruction process for sampled signals is unable to make the distinction between any two sampled signals for all cases of integer values of \( n \).

Beyond this limit, where \( \omega_i \geq \omega_s/2 \), the reconstructed frequencies are “folded” back into the frequency spectrum interval \( 0 < \omega_i < \omega_s/2 \) and it becomes impossible to reconstruct \( \omega_i \) frequencies greater than \( \omega_s/2 \) from the sampled signal.

The critical sampling frequency \( \pi/T \) is often referred to as the “Nyquist frequency”.

The selection of small enough values of \( T \) to achieve valid reconstruction of signals is termed Shannon’s theorem, which may be stated as either of the following two imperatives:

\[
\omega_s > 2 \omega_i
\]

or

\[
\pi/T > \omega_i
\]

(Houpis & Lamont, 1985).
2.3.4 Dynamic load coefficient

Sweatman (1983) developed a measure designated the dynamic load coefficient (DLC) in his work “A study of dynamic wheel forces in axle group suspensions of heavy vehicles. Special Report No. 27” (Sweatman, 1983). This was, in part, based on earlier work (Sweatman, 1980) and was to account for, and allow comparison between, the relative effects of dynamic wheel force behaviour of differing suspension types.

The DLC was defined as the coefficient of variation of dynamic wheel forces relative to the static wheel force; *i.e.* the ratio of variation in dynamic wheel forces to static wheel force. That approach utilised the concept that a measure of road damage could incorporate a damage component due to:

- dynamic forces present from wheel loads; plus
- a component due to the static forces present.

The static wheel force was represented in this measure by the “mean wheel load” \(F_{mean}\) (Figure 2.1). The dynamic forces were represented in this measure as the standard deviation (\(\sigma\)) or root-mean-square (RMS) of the dynamic wheel force (Figure 2.1).

The DLC may be defined mathematically:

\[
DLC = \frac{\sigma}{F_{mean}}
\]

*Equation 2.8*

where:

\(\sigma\) = the standard deviation of wheel force; and

\(F_{mean}\) = the mean wheel force (Sweatman, 1983).

It assumes that:

- dynamic loads are random and have a Gaussian distribution about \(F_{mean}\) as shown in Figure 2.1, after DIVINE (OECD, 1998); and
road damage is distributed evenly along a length of road (Collop & Cebon, 2002).

![Visualisation of Dynamic Load Coefficient](image)

**Figure 2.1. Summary of DIVINE report illustration for dynamic load coefficient.**

Sweatman used various independent variables against which DLCs were plotted for the suspensions tested. These approaches included, for example, plotting averaged DLCs against speed over all the runs made, regardless of the road surface (Sweatman, 1980, 1983), and DLCs for specific determinations of roughness, e.g. “smooth” and “rough” roads (Sweatman, 1983).

Differences in interpretation of the denominator in the DLC formula have been evident (de Pont, 1992). Both “static wheel force” and “mean wheel force during testing” have been defined as the denominator (Potter *et al.*, 1997; Sweatman, 1983). It is for noting that Sweatman (1983) defined DLC with $F_{\text{mean}}$ as the denominator.

Other work (Potter *et al.*, 1997) redefined the DLC denominator to be the static force, ($F_{\text{stat}}$) on the wheel. There is a subtle but distinct difference between the two approaches. If the static wheel force measurements are made on level ground, the measured value will differ from on-road measurements since the cross-fall of the road will place the centre of gravity (CoG) of the vehicle slightly to one side. $F_{\text{mean}}$ will therefore differ from $F_{\text{stat}}$, depending on the degree of cross-fall. It will also vary depending on the load-sharing ability of the suspension in question (de Pont, 1992) and the suspension characteristics, particularly damper non-linearity (Karl *et al.*, 2009b).

Under DLC evaluation, a perfect suspension would have a DLC of zero, *i.e.* the
wheel force would not vary above or below the static value. The range, in reality, is somewhere between 0 and 0.4 (Mitchell & Gyenes, 1989). Many researchers (Gyenes, Mitchell, & Phillips, 1992; Mitchell & Gyenes, 1989) have used DLC as one measure to differentiate suspension types from each another (e.g. steel vs. air). Despite this, the use of DLC has been criticised for purposes of attempting to distinguish between the damage potential of suspensions with different axle groups (Potter et al., 1996) and despite being adopted as the de-facto standard as a road-damage determinant (OECD, 1998).

DLC continues to be criticized, most recently by Dr. Cebon at the Fifth Brazilian Congress on Roads and Concessions; along the line of: “how this [DLC] method leads to false conclusions regarding where and how to use road maintenance funds, spatial repeatability of road surface stress being the key issue.” (Lundström, 2007). This criticism is not new (Cebon, 1987). DLC shares a drawback with other suspension parameters that derive a value per wheel or per axle; the need to aggregate or average the different DLC values per axle or wheel to get a manageable picture of it against independent variables or for comparison of differing suspension designs.

### 2.3.5 Load sharing coefficient

Early attempts to determine load sharing of HV suspensions (Sweatman, 1976) were by driving a test HV over a 40mm plank and measuring the load under the plank. Changes in axle loads, during the dynamic load conditions created thereby, were then compared with static loads.

Sweatman (1983) attempted to quantify the load sharing ability of a multi-axle group in a number of ways, amongst which was the load sharing coefficient (LSC). This was designed to be a measure of how a suspension group shared the total axle group load across the axles within the group. It is a value of the ability of a multi-axle group to distribute its load over each tyre and/or wheel in that group during travel.
The original definition of LSC was:

\[
LSC = \frac{2 \times n \times F_{\text{mean}}}{F_{\text{group (stat)}}}
\]

Equation 2.9

where:

\(n\) = number of axles in the group;

\(F_{\text{group (stat)}}\) = axle group static force and

\(F_{\text{mean}}\) = the mean wheel force in Figure 2.1 (Sweatman, 1983).

Note that this approach treated the load sharing as being between axles.

Sweatman (1980) stated that the net increase in road damage, \(\Delta_{\text{damage}}\), due to unequal loading of (say) 10 percent between axles in a tandem group assuming, again, the ‘fourth-power law’, may be calculated by:

\[
\Delta_{\text{damage}} = 0.5 \times [1.1^4 + 0.9^4 - (1+1)] \times 100\%
\]

Equation 2.10

This approach did not necessarily agree with other, early definitions such as that of Stevenson & Fry (1976) p. 24, that a HV with a “load equalising system” meant that an axle group utilised a suspension with the same spring types on each axle and that this resulted in “substantially equal sharing by all the ground contact surfaces of the total load carried by that axle group”. Note the emphasis on wheel forces in the context of “ground contact surfaces”, not axle forces.

LSC has been simplified and modified more recently to:

\[
LSC = \frac{F_{\text{mean (i)}}}{F_{\text{stat (nom)}}}
\]

Equation 2.11

where:

\(F_{\text{stat (nom)}}\) = Nominal static tyre force = \(\frac{F_{\text{group (total)}}}{n}\);

\(F_{\text{group (total)}}\) = Total axle group force;
\( F_{\text{mean}} (i) \) = the mean force on tyre/wheel \( i \); and

\( n \) = number of tyres in the group (Potter et al., 1996).

Equation 2.9 and Equation 2.11 differ in that the latter focuses on the equalisation of wheel forces and the former on equalisation of axle forces. This may be attributed to a difference in interpretation between schools of road damage: the vehicle modellers vs. the pavement modellers.

Potter et al., (Potter et al., 1996) examined variations in quantitative derivation of measures to describe the ability of an axle group to distribute the total axle group load. That work indicated a judgement that inter-axle relativities were the key to inter-wheel load sharing.

The worth of the LSC as a prime determinant of suspension behaviour has declined but it is still used when describing the ability of a multi-axle group to distribute its load across all the wheels in its group. One of the drawbacks of LSC is the need to aggregate or average the different LSC values per axle or wheel to get a tractable depiction of that metric against an independent variable or to compare differing suspension designs.

2.3.6 Peak dynamic wheel force

The peak dynamic wheel force (PDWF) is the maximum wheel force experienced by a wheel during dynamic loading in response to a step (up or down) input (Fletcher et al., 2002). This measure is important as a link between analysis of wheel force history and the work that promotes spatial repetition (Section 2.4) of HV wheel forces as a measure of damage (Cebon, 1987; Collop & Cebon, 2002; Potter, Cebon, & Cole, 1997; Potter, Cebon, Cole et al., 1995).

In an alternative view that includes non-Gaussian wheel force distributions in the spatial domain, PWDF may form part of a damage model applied to those points of maximum force on the pavement. When applied to historical data of wheel forces on a particular length of pavement, the peak dynamic wheel force may be used as an indicator of potential damage when raised to the appropriate power. This measure is
readily understandable and provides a direct result without requiring aggregation across axles or wheels.

### 2.3.7 Peak dynamic load ratio (dynamic impact factor)

One of the criticisms of DLC is that it assumes that a Gaussian distribution of wheel forces in the time domain will be Gaussian as a spatial variable. Where the wheel forces may not be Gaussian (which suggests an alternative to DLC) and when considering longitudinal position variable-space, peak dynamic load ratio (PDLR) may be considered. It is the ratio of the maximum wheel force dynamic load to the static wheel force:

\[
PDLR = \frac{PDWF}{F_{stat}}
\]

**Equation 2.12**

where:

- \(PDWF\) = peak dynamic wheel force measured instantaneously during the test (Section 2.3.6); and
- \(F_{stat}\) is the static wheel force.

It is not based on a particular distribution and is useful when comparing data with similar distribution sets (Fletcher *et al.*, 2002).

A similar measure for axle forces has also been designated the “dynamic impact factor” (DIF) and was used in earlier evaluations of different types for suspensions for road damage:

\[
DIF = \frac{PDF}{F_{stat(axle)}}
\]

**Equation 2.13**

where:

- \(PDF\) = peak dynamic force measured instantaneously during the test; and
$F_{stat\ (axle)}$ is the static axle force (Woodrooffe & LeBlanc, 1987).

Again the difference in philosophy is apparent between the allocation of road network asset damage to axle forces or wheel forces. The DIF is another measure that allows direct comparison of two test cases or suspension designs without requiring aggregation across axles or wheels.

### 2.3.8 Dynamic load sharing coefficient

The original Sweatman research which examined different LSCs per suspension type instrumented only one hub per vehicle due to the cost (Sweatman, 1983). That work derived wheel forces in multi-axle groups by taking the complement of measured wheel-load. Whilst understandable in terms of expense, inferring the other wheel loads as a complement of the measured load somewhat contradicted earlier work (Sweatman, 1980) that found instantaneous axle forces across the axle group have a tendency to be unequal due to the dynamic forces generated by the road profile. If the wheel forces were only out-of-phase, and there were no in-phase, common-mode or random wheel forces present, then deriving wheel forces by taking the complement of measured wheel-loads would have been valid.

Accordingly, the original research into LSC was questionable. de Pont (1997) also noted that dynamic load sharing had not been addressed adequately and proposed a modification to the concept of load sharing which took into account the dynamic nature of wheel forces and any load sharing which may occur during travel, designated the dynamic load sharing coefficient (DLSC):

$$DLSC = \sqrt{\frac{\sum (DLS_i - \bar{DLS})^2}{k}}$$

*Equation 2.14*

where:

Dynamic load sharing (DLS) at axle $i = DLS_i = \frac{nF_i}{\sum_{i=1}^{n} F_i}$

*Equation 2.15*
$n =$ number of axles;

$F_i =$ instantaneous wheel force at axle $i$; and

$k =$ number of instantaneous values of DLS, i.e. number of terms in the series (de Pont, 1997).

It is noted from Equation 2.14 that DLSC is the standard deviation of the dynamic load sharing function, DLS$i$. Whilst this approach is an evolution from assumptions regarding complementary wheel-loadings and more inclusive of random, in-phase or common-mode relative excitation between consecutive axles, it does not consider that an axle can have differing wheel-loads at either end. This since the instantaneous wheel forces at axle $i$ are summed to get $F_i$ for comparison with the other axle/s in a multi-axle group. Again, there is an emphasis on inter-axle comparison. However, the DLSC approach differs from other approaches in that it may be applied to consecutive wheels in groups. Accordingly, the DLSC provides a value of comparison per two or more wheels or axles and could be applied to the side of a HV or an entire group. It was the most versatile load sharing metric discovered in the literature review.

2.4 Spatial measures

2.4.1 History

From 1958 to 1960, the American Association of State Highway Officials (AASHO) conducted testing on purpose-built road pavements. Approximately $1.1 \times 10^6$ axle repetitions occurred, at varying loads, from US Army trucks being driven at 56km/h by drivers commissioned for the task. Arising from these tests, the “fourth power rule” for granular pavements was determined empirically, viz; pavement damage was proportional to the fourth power of the static load of an axle (Cebon, 1999). This approach, whilst used almost universally for flexible pavement design has been criticised (Cebon, 1987, 1999) in that it did not take into account the concentration of dynamic loads at certain points over a length of road, effectively averaging the HV “bounce” forces into the empirical data (de Pont, 1992).
2.4.2 Quasi-static wheel loadings and pavement damage

Pavement design life is determined by, and based on, repetitive loadings arising from repeated passes of a theoretical heavy vehicle axle. Conceptually, this pavement life design parameter is based on the number of passes of a standard axle over a pavement. This measure is, in turn, based on the AASHO work (Section 2.4.1) where axle repetition occurred until the pavement became unserviceable (Cebon, 1999). The number of HV passes (i.e. the number of axle repetitions) determined the design parameter for pavement life. This basic theory for determining pavement life as a value of vehicle passes has not altered significantly since the US military experiments last Century (Alabaster, Arnold, & Steven, 2004; Main Roads Western Australia, 2005; Romanoschi, Metcalf, Li et al., 1999). Pavement loadings including wheel force frequencies higher than just the one occurrence per wheel pass have not figured prominently in pavement models in use today. Were dynamic and steady state forces able to be separated from the original AASHO data, separation of rutting failure due to quasi-static loadings vs. fatigue failure due to dynamic wheel forces may have been evident. On this line of argument, the following exponents and associated failure modes have been determined to be probable (de Pont & Steven, 1999; Pidwerbesky, 1989):

- 1 to 2 (probably 1.4) for rutting of NZ roads;
- 2 for fatigue and between 3.3 – 6 for rutting on Australian roads;
- 3.3 for fatigue and 4 for rutting on Finnish roads;
- 2 for fatigue and 8 for rutting on French roads;
- 1.2 to 3 for fatigue on Italian roads; and
- 1.3 to 1.9 for fatigue and 4.3 for rutting on North American roads.

The damage exponents have been reported to vary due to road construction and HV configurations used in the country (de Pont & Steven, 1999; Pidwerbesky, 1989). The quasi-static load passes causing pavement distress are in contrast to the dynamic forces that produce point-failure in flexible pavements which gave the empirical “fourth power rule” historically where \( q = 4 \) for flexible pavements (Cebon, 1999;
Cole, Collop, Potter et al., 1996). Similarly, other damage exponents (such as 12 for concrete pavements) may be chosen, depending on the material (Vuong, 2009).

Australia’s accelerated loading facility (ALF) and New Zealand’s Canterbury accelerated pavement testing indoor facility (CAPTIF) have been used to determine pavement life in a similar manner to the original US testing; repeated passes of a test wheel at a particular load over a pavement (Main Roads Western Australia, 2005; Moffatt, 2008). In fairness, much work has been done using New Zealand’s CAPTIF system to correlate dynamic wheel forces with axle passes (de Pont, 1997; de Pont & Pidwerbesky, 1994; de Pont & Steven, 1999). Further, a considerable body of work in the UK (Cebon, 1987, 1993, 2007, 1999; Cole & Cebon, 1989, 1992, 1996; Cole, Collop, Potter et al., 1992; Cole et al., 1996; Collop, Cebon, & Cole, 1996; Collop, Potter, Cebon et al., 1994; Gyenes & Mitchell, 1992, 1996; Gyenes et al., 1992; Gyenes & Simmons, 1994) has reported dynamic wheel loadings from HVs. Results of that work have not yet been incorporated into general pavement design, particularly in Australia (Main Roads Western Australia, 2005; Moffatt, 2008).

### 2.4.3 Stochastic forces – probabilistic damage

Cebon (Cebon, 1987, 1993, 1999), amongst others, has championed the concept of spatial damage assessment for dynamic wheel loads. This is an approach where the damage due to HV dynamics is quantified over a particular length of pavement. It is based on the probabilistic nature of road damage. It contains the concept that road damage leading to loss of serviceability occurs at only a small proportion of the length of road (Cebon, 1987). Spatial repetition predicts pavement damage due to HV wheel forces, sometimes using the highest frequency of interest in the process. “Spatial” measures include weighted stress, aggregate force, strain fatigue damage and pavement deformation (Cebon, 1987; Collop & Cebon, 2002; Potter, Cebon, & Cole, 1997; Potter, Cebon, Cole et al., 1995). These, as well as other studies (Hahn, 1987a, 1987b; LeBlanc & Woodroofe, 1995), have indicated strongly that wheel-loads along a length of road are not distributed randomly but are concentrated at specific points on the length of road for a specific type of vehicle at a specific speed. This effect is known as “spatial repeatability”.
For specific classes of tested vehicles the spatial correlation of wheel-loads was reported as moderate-to-high and judged highly dependent on travel speed, HV suspension and chassis configurations (Cole, Collop, Potter et al., 1996; LeBlanc & Woodrooffe, 1995). Pavement models using this approach require an intimate knowledge of pavement behaviour resulting from wheel forces. Some studies have used instrumented pavements to correlate spatial and temporal measurements of wheel-loads (Cole, Collop, Potter et al., 1996; Gyenes & Mitchell, 1992). Those approaches broke the pavement down into a number of short segments to determine peak forces per segment.

The issue of allocating road damage at specific points on the pavement to the entire HV fleet becomes less clear, however, when examining attempts made to correlate wheel loads measured from test HVs against spatial wheel-loads measured from the pavement. Diversity of suspension types, such as steel walking-beam or air-sprung, and diversity of speed reduced the correlation to moderate-to-low (LeBlanc & Woodrooffe, 1995). Those lower correlations were noted even on a test semi-trailer tanker that was configured specifically to have its prime-mover and trailer suspensions replaced with steel or air for testing (LeBlanc & Woodrooffe, 1995). Increasing homogeneity of the parameters of the HV fleet equipped with “road friendly” suspensions (RFS) will nonetheless result in more highly correlated wheel forces. LeBlanc predicted that spatial repetition would therefore need to be addressed (LeBlanc, 1995); suspensions with common parameters will bounce their wheels onto the same places on the pavement after encountering a bump.

2.4.4 Spatial repetition and HML

When considering the introduction of higher mass limits (HML) into Australia, spatial repeatability measures contributing to pavement damage were acknowledged as an approach to determining road damage but not included in the methodology (National Road Transport Commission, 1993) for determining damage due to HVs at masses greater than statutory. Attempts have been made to harmonise the spatial damage models with temporal models that rely on measurements from HVs (de Pont & Pidwerbesky, 1994). This approach has not been adopted widely.
2.4.5 Cross-correlation of axle loads

Blanksby et al., (Blanksby et al., 2009) reported the use of laser deflectometers to measure the distance between the hubs of a 3-axle air-suspended semi-trailer and the road surface. The basic premise of the theory behind the testing was that dynamic load sharing could be detected from the correlation of instantaneous forces on a wheel. Where a particular wheel encountered (say) a bump, this action would result in forces being present on other wheels as the load was evened out. The cross-correlation of forces would be strong where dynamic load sharing was occurring and weak where it was not, therefore. This approach was applied to forces on successive axles and analysed in the spatial domain. This was a new definition of load sharing not used before but one that seemed to be valid. The remainder of the test programme could be questioned, however, as outlined below.

The testing procedure measured HV hub heights as surrogates or indicators of tyre deflection. The experimental design assumed that tyre deflection was proportional to, and an indicator of, tyre load. Calibration of the relationship between tyre deformation and hub height used static loads and a quasi-static “bounce test” where the HV was dropped from a height of 80 mm with dynamic wheel forces recorded. The relationship between dynamic load and tyre deflection was derived from this quasi-static process. The HV was driven on suburban roads of varying roughness and at different speeds with the laser signals recorded. Some testing was performed with the dampers removed from the middle axle. The signals from the lasers were analysed for DLC against roughness and suspension health; the removal of the shock absorber from the middle axle nominated as a surrogate of a worn suspension.

The methodology in the report did not address the differences between calibrating tyre forces dynamically vs. the static calibration performed. A number of researchers (Hartree, 1988; O'Keefe, 1983; Popov, Cole, Cebon et al., 1999; Segel, 1975; van Eldik Thieme & Pacejka, 1971) have documented the differences between quasi-static tyre deflection forces and those forces exerted by tyres in the dynamic rolling state. Other research undertook dynamic calibration of tyre carcass deflection vs. load (Tuononen, 2009) but it also acknowledged that inflation pressure and temperature greatly influenced the accuracy. How the difference between dynamic readings, with their attendant variables, and static calibration was overcome was not
explored in the Blanksby *et al.*, (2009) report other than to perform a drop test using the ARRB Group’s drop test rig. This quasi-static bounce test was then equated to a rolling dynamic test without referring to the work that documented the problems with equating quasi-static effects with dynamic rolling deflection (Hartree, 1988; O’Keefe, 1983; Popov, Cole, Cebon *et al.*, 1999; Segel, 1975; van Eldik Thieme & Pacejka, 1971).

In particular, the issue highlighted by previous research (van Eldik Thieme & Pacejka, 1971) but unexplored by the Blanksby *et al.* (2009) report was that tyre radius (axle height) does not remain linear with speed, particularly for low speed vs. medium and high speeds.

Suspending belief in the reality of non-uniformity of tyre deflection vs. load vs. speed for a moment, let us assume that:

- the methodology of using the lasers was valid;
- that tyre deformation was proportional to load at all speeds;
- a 10 percent experimental sensitivity compared with a full load was required; and
- HV tyres in dual configuration have a tyre spring rate \( (k_t) \) of 1.96 MN/m (Costanzi & Cebon, 2005, 2006; de Pont, 1994; Karagania, 1997) or approximately 2 kN/mm.

Accordingly, for a 10 percent detectable load variation at the tyres of a full tri-axle wheel; *i.e.* 375 kg in 3.75 t (22.5 t / 6), the tyres would deform 1.84 mm. For surfacing aggregates of (say) 10 mm or 15 mm, this meant that signal excursions for 10 percent sensitivity would have been less than 2 mm in magnitude superimposed onto a 10 mm or 15 mm noise signal from the peak vs. trough range in pavement aggregate. Hence, a data signal with an experimental sensitivity of 10 percent was recorded with a noise signal an order-of-magnitude greater than the data range. How this anomaly was dealt with was not explained adequately.

With the exception of removing the middle shock absorber to simulate a wear situation, only test data were analysed. No other control data were measured. That
is, the standard scientific method proposed by Mill (1872) and outlined in Section 1.9.1 was not used for the testing or analysis of the data.

Other work by most of the same authors (Germanchev, Blanksby, Ritzinger et al., 2008) describing flaws in the wheel-load analysis methodology. In particular, compensation for tyre damping force, compensation for outboard offset of the laser during axle roll and investigation of high frequency vibrations from the laser deflectometers were nominated as issues yet to be resolved (Germanchev et al., 2008). The report (Blanksby et al., 2009) did not quote from the previous work outlining these flaws and did not address them. Further, the mechanism for compensating for other mechanical factors that could introduce error, such as wheel-bearing wear, was not mentioned.

### 2.5 Load sharing coefficient vs. dynamic load coefficient

#### 2.5.1 Introduction

The aim of this section is to present a summary of the mathematical relationship between the dynamic load coefficient (DLC) and the load sharing coefficient (LSC) after they were reviewed above. The contents of the following section are a summary of other work for this project developed in detail by Davis and Bunker (2009d).

#### 2.5.2 Brief recap on DLC and LSC

Sweatman (1983) needed a numerical value to ascribe a relative road damage value to a HV suspension in comparison to other suspensions. The dynamic load coefficient (DLC) was one of the measures derived from earlier work (Eisenmann, 1975).

Equation 2.8 defines the dynamic load coefficient (DLC). Also developed in his 1983 study, Sweatman developed the load sharing coefficient (LSC) as a measure of how well any particular axle or wheel of a multi-axle HV suspension shared the load.
of the entire group (Sweatman, 1983). The LSC (Equation 2.11) had slight modifications made subsequently (Potter et al., 1996).

2.5.3 Relationship between LSC and DLC

Recapping Equation 2.8:

\[
DLC = \frac{\sigma}{F_{\text{mean}}(i)}
\]

and Equation 2.11:

\[
LSC = \frac{F_{\text{mean}}(i)}{F_{\text{stat}}(i)}.
\]

Reformatting Equation 2.11:

\[
F_{\text{mean}}(i) = LSC \times F_{\text{stat}}(i)
\]

Reformatting Equation 2.8:

\[
F_{\text{mean}}(i) = \frac{\sigma}{DLC}
\]

Now, equating \(F_{\text{mean}}(i)\) from Equation 2.16 and Equation 2.17; therefore:

\[
DLC = \frac{\sigma}{LSC \times F_{\text{stat}}(i)}
\]

Accordingly, we see that, for a given HV suspension, the LSC will have an inverse relationship with the DLC for that suspension. The slope of the line plotted on the graph of the relationship will be \(\frac{\sigma}{F_{\text{stat}}(i)}\).

The LSC of a perfect suspension would be 1.0 (Potter et al., 1996) with a DLC of 0 (Mitchell & Gyenes, 1989).
Assume that the static mass value remains constant, as does the standard deviation of the wheel/axle force signal over the recorded test run. Plotting an indicative relationship between DLC vs. LSC using Equation 2.18 allows a visual analysis of the next logical step. Figure 2.2 shows this relationship and indicates that increasing the LSC means a decreasing DLC. LSC variation away from 1.0 is undesirable with a LSC locus moving away from this value implies increasingly uneven distribution of load during travel (Potter et al., 1996). Figure 2.2 shows that there is a mutual exclusivity of optimisation between the two measures. Implementing design improvements that bring about reductions in DLC will increase the LSC value. The scale for DLC was not used in Figure 2.2 or Figure 2.3; these are a conceptual plots of the relationship.

![Figure 2.2. DLC vs. LSC relationship.](image)

Further developing this reasoning, Figure 2.3 shows an arbitrary optimum LSC band. Where this band intersects with the DLC corresponding to that optimum range of LSC, the range of DLC available (or resulting from) that design is shown as 'x' in Figure 2.3. Given any optimum (or at least, desirable) LSC range, the designer has no choice about the resultant DLC in the range 'x' in Figure 2.3.
Accordingly, LSC and DLC are, for any given suspension, not separate parameters but mutually dependent and inversely proportional variables arising from suspension design. Suspension forces and their transmission to the chassis (and therefore to the wheels) are influenced by other factors such as damper characteristics (Karl et al., 2009b). After choosing dampers and suspension components that result in compliance with Vehicle Standards Bulletin (VSB) 11, HV designers may not have a wide range of choices when attempting to minimise DLC or centre LSC around 1.0 since these are inversely proportional and related to the dynamic range of the HV suspension forces.

2.6 Summary of this chapter

2.6.1 Model conflict

There were discovered, broadly, two approaches to determination of HV wheel forces:

- vehicle models; and
- pavement models.

The school of vehicle modellers treated vehicle dynamics in terms of inter-axle force relationships. These were used for analysis on both internal vehicle forces and as an
indication of pavement forces via the wheels. Early work treated load sharing as a phenomenon involving sequential axles, for example, as opposed to individual wheels. In contrast, pavement modellers did not concern themselves particularly about the measurement of pavement forces via axles but applied theoretical and empirical wheel forces to pavements and modelled the forces where the tyre meets the road.

2.6.2 Pavement damage models

Pavement damage models may be further separated into differing approaches:

− the “Gaussian” - a random distribution of HV wheel forces on a road length with all pavement segments subject to a roughly evenly-distributed probability of road damage; or

− the “spatial” - the characteristics of HV suspensions and travel speeds incline wheel forces to occur at specific locations on any length of road and that road damage is therefore concentrated at those specific points.

As noted in Section 1.4.1, although spatial repetition was acknowledged in the MLR report, Gaussian distribution of wheel forces was the approach that the National Road Transport Commission (NRTC) took when considering the introduction of HML into Australia (National Road Transport Commission, 1993; Pearson & Mass Limits Review Steering Committee, 1996b).

2.6.3 Spatial repeatability

Adherents to the spatial philosophical school of pavement damage focus on the road asset, measure wheel forces directly and develop road-damage models that account for dynamic loads as HVs travel over segments of pavement. This approach generally requires in-road sensor-based technology to measure actual wheel forces at the road surface accurately (Cole & Cebon, 1989; Cole, Collop, Potter et al., 1992; Potter et al., 1996; Potter, Collop, Cole et al., 1994).
2.6.4 Spatial repeatability vs. Gaussian distribution

There are some disadvantages with spatial measurement. One is the need for long lengths of instrumented road, up to 250m (Potter et al., 1997). Another is the inability to measure conveniently a variety of roads with varying surfaces. Instrumented vehicles can record data from on-road excitations for longer distances than an instrumented pavement. On-vehicle measurement ability is more portable, allowing recording of different road surfaces more conveniently. Further, since road profiles alter over time, the wheel forces, and therefore road damage, can only be determined using instrumented pavement methods for the particular set of circumstances at the time of the testing. Repeatability would require the same length of pavement to be measured periodically. This is easier with an instrumented vehicle than an instrumented pavement.

Spatial measures attempt to deal with rutting damage in a different manner from fatigue damage. Rutting is the result of repeated passes of wheels and is related to speed and vehicle static mass. Fatigue is a result of many individual dynamic wheel forces impacting the pavement at the same or similar points along a length of road (Potter, Cebon, Cole et al., 1995).

Whilst there may be medium-strong spatial correlation of wheel forces for particular vehicles at particular speeds (Cole et al., 1996), that correlation reduces to moderate-to-low for the fleet. This is very likely due to different suspension types and different speeds of operation across the fleet. Accordingly, Eisenmann’s formula underestimates pavement wear (Gyenes, Mitchell, & Phillips, 1994) and the 95th percentile formula overestimates pavement wear from dynamic wheel loads (LeBlanc & Woodroofe, 1995). That the dominant vehicle types have approximately Gaussian axle load distributions and therefore may have their wheel forces characterised by mean and standard deviations has, however, been validated in more recent work (de Pont, 2004). This assurance was based on DIVINE project WiM data and incorporated tare as well as loaded data for vehicles representing a greater majority of HVs on Australian roads.

If spatial measures were valid in predicting pavement failures then peak dynamic forces would cause those failures (Cole et al., 1996; Collop, Potter, Cebon et al., 1994). Even if wheel forces were not spatially correlated, the “power law” model for
wheel force damage would still indicate pavement failure. The difference between these two models appears to be the number of repetitions before failure rather than whether to choose average dynamic forces over peak ones.

As pointed out by critics of methods that assume Gaussian distribution of wheel forces, the “fourth power rule” was developed with dynamic loadings already “built into” the AASHO experimental data. Different “power law” model exponents have been derived in the intervening half-Century since the original AASHO work with military vehicles. The existence of these differing exponents as applied to different pavement types shows that the “science” of damage models is still not exact.

To exploit spatial repetition as a pure approach, both steady state wheel force load and dynamic wheel force load need to be separated; pavement damage treated as two distinct phenomena: rutting and fatigue respectively. In a perfect world, the rutting and the fatigue damage predictors would be combined after this point in the process to form a composite damage model. As pointed out by the critics of Gaussian wheel force models, this is in contradistinction to the “one-size-fits-all” lumped empirical formula containing both concepts. Nonetheless, and with some awkward logic, spatial models of pavement damage still nominate the fourth power exponent in their damage predictions (Collop et al., 1994) for flexible pavements.

### 2.6.5 Defining road damage from a vehicle-based framework

Mitchell & Gyenes (1989) noted that one measure alone was not sufficient to determine the road-damaging potential of a particular suspension and concluded that a judgement based on:

- LSC;
- DLC;
- low and high frequency vibration forces (body bounce and wheel hop); and
- peak axle loads
would be necessary since the measures developed up to that point described different suspension parameters, depending on behaviour. Further, it may be valid to question which parameters should be desirable for “road friendly” suspensions (RFS) as measured by testing methods other than the European Union method (European Council, 1996).

de Pont’s work (1999) showed that the values measured for resonant frequencies, etc. at different loads and speeds do not vary significantly from those derived from the EU testing if the centre-of-gravity is placed over the particular suspension (component) under test.

Other research pointed out that determining suspension characteristics measured from a shaker-bed or reaction frame necessitated either test a single axle with the body fixed or test the whole vehicle. Testing selected axles did not reflect suspension nor vehicle performance (Stanzel & Preston-Thomas, 2000). This inferred that whole-of-vehicle testing is the only valid method but this:

- ignored the reality that prime-movers and trailers are rarely used continuously as a unit;
- was hard to reconcile against the supposed validity of individual axle tests; and
- did not seem to encompass de Pont’s work (de Pont, 1999).

2.7 Conclusions of this chapter

2.7.1 Relative views of pavement/wheel load

Pavement forces from a HV may be measured from the viewpoint of the wheels of a HV or at the pavement. That choice and the assumption of Gaussian force distribution in the temporal or spatial domains leads to various possibilities for viewing forces at the HV tyre/pavement interface. The approach of Gaussian modelling to pavement damage would be valid were HV wheel forces to be distributed randomly along a length of road. This relies on measurement of wheel force histories (statistical analysis) and development of road-damage models based
on (say) the “fourth power rule”. Pavement design life is then based on repetitive loadings arising from repeated passes of a theoretical heavy vehicle axle. That thinking may be applied to either vehicle-based or pavement-based measurements. The former approach, where pavement effects are assumed from measured wheel forces, is more common since HVs are more easily instrumented than pavements.

Alternative approaches in the pavement modelling domain measure wheel forces directly and develop road-damage models that account for dynamic loads as HVs travel over segments of pavement. This generally requires in-road sensors to measure wheel forces at the road surface (Cole & Cebon, 1989; Cole et al., 1992; Potter et al., 1996; Potter et al., 1994).

### 2.7.2 Spatial repeatability vs. Gaussian distribution

The spatial school, backed by not inconsiderable empirical data, rejects the thinking of the Gaussians and contends that spatial repetition will cause failure of the pavement from localised peak dynamic forces damaging the pavement at recurrent points along a length of road. It rejects the use of averaged dynamic forces as a predictor of pavement damage (Potter et al., 1994). This philosophy then proposes that network utility is reduced by denying service on the road where such localised pavement failures occur (Cebon, 1987, 1993, 1999). This may be balanced against a pragmatic view that such service denial is short-term; potholes are patched with subsequent resumption of access, whilst conceding that patches increase future failure point probability.

### 2.7.3 Vehicle-centric measurement of metrics

In striving to improve the LSC or DLC suspension metrics, the vehicle designer may not have much choice as these are mutually dependent variables and inversely proportional to each other. Improving one degrades the other.

There is a broad range of suspension metrics available to the vehicle researcher or designer. Some of these are more easily derived than others. The mechanisms for
creating the necessary vehicle dynamics to measure the metrics vary from the simple to the complex. Computational requirements for deriving the various metrics vary likewise.

Early work (Eisenmann, 1975; Sweatman, 1983) postulated certain approaches, particularly assuming Gaussian distribution of wheel forces. Later development, leveraging off spatial repetition, adopted a reductionist approach and concentrated more on peak wheel forces as an indicator of pavement damage (Fletcher et al., 2002; Woodroffe & LeBlanc, 1987). This latter metric much more easily measured, either from the vehicle or the pavement.

2.7.4 Pavement models

Whether wheel forces are measured at the pavement or the vehicle, whether they are aligned spatially or which metrics are valid; pavement models, particularly in Australia, ascribe pavement life design to quasi-static HV axle load repetitions with a “power law” damage exponent. These models are empirically derived from data that contains both static and dynamic wheel forces. Section 2.4.2 expanded on these damage exponents.

Pavement damage models have an entrained “power law” damage exponent to account for empirical data drawn from different countries, differing materials and different HV configurations (Pidwerbesky, 1989). Damage exponents varying from 1 to 8 for flexible pavements (de Pont & Steven, 1999; Pidwerbesky, 1989) and up to 12 for concrete pavements (Australroads, 1992; Vuong, 2009) are used. dePont & Steven (1999) showed quasi-static loadings caused rutting but not fatigue, with damage exponents tending toward 1.0.

The damage exponent order of magnitude varied in the literature from $x^1$ to $x^{12}$. This range is greater than the order of magnitude variation in the exponents allocated to human scale ($10^0$ m) vs. Earth diameter ($10^7$ m). From the original derivation of the repetitive axle loading and subsequent evolution of pavement damage models, it appears that the variation in order-of-magnitude of damage exponents has been a result of explaining subsequent pavement damage empirical data in various ways.
Notwithstanding, it is difficult to reconcile that variations in HVs, materials and country of construction would account for damage outcomes greater than the proportion of a human to the size of the Earth.

In summary, the Gaussian vs. spatial and HV instruments vs. in-road sensor debates will largely be academic until standard axle repetition (SAR) or equivalent standard axle (ESA) measures incorporate scientifically-determined dynamic wheel force measures. Whilst the fourth power rule and ESAs have been questioned, they remain in pavement engineering lore as the basis of measures to determine asset damage.

2.8 Chapter close

The research framework, problem statements and objectives for this thesis were outlined in Chapter 1. However, without tools to execute the work (or parameters to measure, analyse and report) the research programme would be for nothing. Accordingly, this chapter introduced the philosophy, derivation and range of available HV suspension metrics and their measurement. The background described in this chapter informed the methodology and rationale embedded in the testing procedures, particularly digital sampling theory (Section 2.3.3), used to gather the data for the project Heavy vehicle suspensions – testing and analysis. These procedures are detailed in Chapter 3 following.

Chapter 4 details the development heavy vehicle (HV) suspension models for the thesis. This process would be less intelligible without the introduction to damping ratio and damped natural frequency in Sections 2.3.1 and 2.3.2.

Analysis of the data testing from the testing outlined in Chapter 3 will be presented later in this thesis in Chapters 5 to 10. These chapters will make extensive use of HV suspension metrics detailed in Chapter 2.

Chapter 8, whilst mostly concerned with on-board mass (OBM) measurement systems for HVs, will require a working knowledge of the fundamental characteristics of a HV suspension as provided in Chapter 2, particularly digital sampling theory (Section 2.3.3).
Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“If you are not measuring it, you are not managing it.” - W. Ed Deming.
3 Test methodology arising from problem identification

3.1 About this chapter

This chapter describes the methodology used to gather the data for the project: *Heavy vehicle suspensions – testing and analysis*. This includes the rationale for choices made when designing the tests.

3.1.1 Rationale for sampling frequency – general statement regarding Sections 3.2 and 3.3

Data for use to meet Objectives 1 and 2 (the three heavy vehicles in Section 3.2) and Objective 3 (the roller bed testing in section 3.3) were recorded using an advanced version of a CHEK-WAY® HV telemetry system. The telemetry system sampling rate was 1 kHz giving a sample period of 1.0 ms. The natural frequency of a typical heavy vehicle axle (axle-hop) is 10 to 15 Hz; a natural periodicity in the range 100 to 66.7 ms (Cebon, 1999). The sprung mass damped natural frequency or body bounce is usually in the range 1 to 4 Hz; a natural periodicity in the range 1000 to 250 ms and therefore of a relatively lower frequency range (i.e. greater periodicity values) than axle-hop (de Pont, 1999).

The Nyquist sampling criterion (Shannon’s theorem) defines the minimum required sampling frequency for any dynamic data. For accurate recreation of dynamic data from sampled data without loss of information, Shannon’s theorem specifies that measured dynamic data be sampled at a frequency at least twice that of the highest data frequency of interest (Houpis & Lamont, 1985). The supporting theoretical background for this has been covered in Section 2.3.3. When using time-based recording, measuring and reconstruction of signals with higher frequencies than (say) body bounce, such as axle-hop, necessitates higher sampling rates than would be used for body bounce data. The maximum frequency in the data for the testing described in Section 3.2 and Section 3.3 was 15 Hz (axle-hop); the corresponding minimum necessary sampling frequency was therefore 30 Hz. The CHEK-WAY® HV telemetry system sampling frequency of 1 kHz was well above the minimum requirement of 30 Hz; the requirement for the Nyquist sampling criterion (Shannon’s


3.2 HV suspension testing - Objective 2 and part of Objective 3

A test programme used three instrumented heavy vehicles (HVs) to gather data used for determining whether larger longitudinal air lines alter wheel or chassis forces (Objective 2) in addition to informing part of the in-service suspension testing (Objective 3) portion of this project.

3.2.1 General description

To provide data for Objective 2 and partially for Objective 3, the wheel forces and chassis forces on three test HVs were measured. The test HVs and their axle group configurations were as follows:

- School bus: one front steer axle, one rear (drive) axle;
- Interstate coach: one front steer axle, one rear drive axle, one rear tag axle. The tag and drive axle comprised the rear axle group. Axle spacing between the tag axle and the drive axle was 1.4 m; and
- Articulated HV: prime mover (not tested) and trailer tri-axle group with axle spacing of 1.4 m.

These are shown from Figure 3.1 to Figure 3.4. The semi-trailer was towed by a prime mover that was not tested in this programme.
Figure 3.1. Prime mover and test semi-trailer with test load.

Figure 3.2. Three-axle coach used for testing.

Figure 3.3. Two-axle school bus used for testing.
The axle groups of interest for these vehicles were:

- **bus**: rear (drive) axle;
- **coach**: rear (tag and drive) axle group; and
- **semi-trailer**: tri-axle group.

The axle/axle group of interest on each HV was configured such that standard sized air lines or larger longitudinal air lines, the “Haire suspension system”, could be connected; the former being the control case, the latter being the test case. All axle/axle groups of interest had a single height control valve. The “Haire suspension system” is a proprietary suspension system that uses larger-than-standard air lines, one down each side of the vehicle. These connect successive air springs on their respective sides of the vehicle. Strictly there are no “standard” size for air lines but industry norms are approximately 4 mm to 10 mm inside diameter (Simmons, 2005).

The “Haire suspension system” comprises air lines with an inside diameter of approximately 50mm, an order-of magnitude greater than found in general air suspension applications. A schematic of this system is shown in Figure 3.5. The alteration to the size of the air lines between successive air springs on the same side of the HV, *per* the “Haire suspension system”, was the only alteration made to the HVs for the control (standard air line) vs. test (Haire suspension system) cases.
analysed. Auxiliary roll stiffness of, and Coulomb friction within, the HV suspensions tested were per the manufacturer’s specification and were incorporated into the methodology by testing the three HVs in the “as delivered” state; these variables were therefore unaltered from one test to the next.

The bus and the coach used a drive axle arrangement with four air springs supporting the chassis and connected with beams as shown in Figure 3.6. This figure also shows the larger air lines for the case of the “Haire suspension system”. The coach tag axle had two air springs in a conventional arrangement for HV axles; one air spring on each end, similar to that shown in Figure 3.5. The school bus drive axle did not have another axle with which to share load. It had been fitted with the “Haire suspension system” as shown in Figure 3.6. Similar to the other axle groups tested, it was possible to connect either the “Haire suspension system” or standard air lines. Accordingly, it was tested but load sharing was not in-scope for that vehicle.

![Figure 3.5. Schematic of the “Haire suspension system” (left) and standard air suspension system (right).](image)

The “Haire suspension system” uses 20 mm connectors to connect the air springs to the 50 mm air lines; Figure 3.7. This figure shows the arrangement for the semi-trailer but was typical for the HVs tested.

![Figure 3.6. Schematic layout of the bus and coach drive axles.](image)
The bus had one suspension levelling valve for the drive axle (the axle of interest). The semi-trailer had one suspension levelling valve for the tri-axle group, the axle group of interest. The coach had two suspension levelling valves on the drive/tag rear group, one on each side. The suspension levelling valves were initialised before each test by powering up the HV under test and allowing the air pressure in the air springs to stabilise, resulting in correct ride height for the HV under test. After the HVs were manoeuvred into position for the quasi-static tests (Section 3.2.3), the brakes were released. This allowed the suspensions to settle into a quiescent state with as little brake wind-up and bushing hysteresis as possible.

All test HVs were equipped with new suspension dampers to ensure that they were returned as closely as possible to the manufacturer’s damping specifications for the tests.

### 3.2.2 On-road tests

To provide data for Objective 2, the wheel forces and chassis forces on the three test HVs were measured. Highway and suburban roads were used. The HVs were driven at different speeds. The roads chosen were considered a representative mix of speed, roughness and surfaces that would be expected during typical low, medium and high-
speed HV operation. The road sections were in the Brisbane area; locations and speeds were as shown in Table 3.1.

The test HVs were tested both at tare and loaded, the loaded case being as close to the maximum general access mass for the group under test. Scrap steel in bins was used to load the semi-trailer (Figure 3.1 and Figure 3.11); sacks of horse feed were used to load the buses (Figure 3.4).

The test HVs were instrumented to measure axle-to-chassis forces and wheel forces. The dynamic signals from the on-board instrumentation were recorded for 10 s at 1.0 kHz. This resulted in test data in the form of 10,000 data points over a 10 s time-series signal from the transducers at each axle-end of interest on each test HV at the various test speeds.

The same section of road was not used for each speed during these tests. For reasons of logistics, safety and consideration of other road users, the general speed limit on each road section was used as the test speed.

Table 3.1. Test speeds, locations and details for the three HVs.

<table>
<thead>
<tr>
<th>Location</th>
<th>Description</th>
<th>Speed (km/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sherwood Rd, Rocklea</td>
<td>Westbound after the traffic signals at the Rocklea markets</td>
<td>40 and 60</td>
</tr>
<tr>
<td>Fairfield Rd, Rocklea and Fairfield</td>
<td>Northbound after the roundabout at Venner Rd</td>
<td>60</td>
</tr>
<tr>
<td>Fairfield Rd, Rocklea and Fairfield</td>
<td>Northbound after the Hi-Trans depot</td>
<td>60 and 70</td>
</tr>
<tr>
<td>Ipswich Mwy</td>
<td>Westbound under the Oxley Rd/Blunder Rd overpass</td>
<td>80 and 90</td>
</tr>
<tr>
<td>Ipswich Mwy</td>
<td>N/Eastbound after the Progress Rd on-ramp</td>
<td>80 and 90</td>
</tr>
</tbody>
</table>

Nonetheless, different roads with different roughness at different speeds have been used previously and was not an unusual approach for this type of testing (Woodrooffe, LeBlanc, & LePiane, 1986). Another consideration was that a variety of surface roughness was not available over one section of road for a variety of speeds within the 10 s recording window of the telemetry system. The testing
procedures have been documented for further reference (Davis, 2007; Davis & Bunker, 2008a; Davis & Bunker, 2008b, 2009b).

3.2.3 Quasi-static suspension testing

Air spring forces and the accelerations at the wheels on the three test HVs were measured to provide data partially for Objective 3. The test HVs were loaded to maximum legal mass and all wheels of interest driven off 80 mm steps simultaneously. This was to replicate the VSB 11 step test (Australia Department of Transport and Regional Services, 2004a). Figure 3.8 to Figure 3.10 shows this procedure for the coach left drive wheel, for instance, but all vehicles were tested in this manner. The HVs were then driven at 5km/h over a 50 mm nominal diameter heavy wall steel pipe (Figure 3.11 and Figure 3.12). The resulting impulse was thus applied to the axle of interest. The pipe had bars welded to it to prevent rotation as the tyres moved over it. The VSB 11-style tests were included to yield data for the control case against which to test the lower-cost “pipe test” results as the test case.

Figure 3.8. Before: showing preparation for the step test.
Figure 3.9. During: the rear axle ready for the step test.

Figure 3.10. After: the step test that was set up in Figure 3.9.
3.2.4 Rationale for “pipe test”

VSB 11 specifies a number of methods for determining the damping ratio and damped natural frequency of the body bounce. Amongst these is a procedure where the HV’s wheels are rolled off a step of 80 mm height and the body bounce signal
analysed. System theory has it that a resonant system may be characterised by analysis of the system output after an impulse input (Chesmond, 1982; Considine, 1985; Doebelin, 1980). Inputs into systems are also known as “forcing functions”. The background to this portion of systems theory is that a perfect impulse of infinitely short period will contain all frequencies. Infinitely short impulses have, theoretically, infinite power. The characteristic frequencies of the system will be the ones transmitted to the output after such an input and thus available for analysis. Pragmatically, no impulse is of infinitely short duration, nor of infinite power. However, the application of this theory works well in practice, provided the forcing function is as short as possible and of a period less than 0.35 the damped natural period (Doebelin, 1980). That work also showed that the shape of the forcing function was not relevant provided that it was short enough (i.e. of a shorter duration than 0.35 of the natural period of the system being characterised). The 80 mm step, in terms of system theory behind that particular VSB 11 method, was the impulse imparted to the HV suspensions to excite them, as systems, so that their outputs, being the axle-to-body displacements, could each be analysed for damped natural frequency and damping ratio. To inform Objective 3 (in-service suspension testing) partially, the “pipe test” was proposed as an alternative, lower cost, forcing function to the step test.

3.2.5 Rationale for instrumentation to measure dynamic wheel forces

To inform Objective 2 (alterations to dynamic load sharing from larger longitudinal air lines) and partially Objective 3 (in-service suspension testing) wheel forces and hub acceleration data were required. A combination of accelerometers and strain gauges was necessary to measure wheel forces.

Strain gauges were mounted on the sides of the axles of interest. This positioned them across the neutral axis of the axle to be measured. Mounting was as close as possible to the hub and used cyanoacrylate glue after surface polishing. Figure 3.13 shows the arrangement for the semi-trailer, for example, but this was typical for all the test vehicles. Waterproofing foil was installed over the strain gauges.
The strain gauges thus mounted allowed measurement and recording of static and
dynamic shear forces in the axle of interest at the point of their mounting. Previous
work showed that mounting strain gauges at the top and/or bottom of test axles
yielded greater sensitivity and improved signal-to-noise ratios for strain gauge output
vs. wheel force compared with side mounting (de Pont, 1997). However, mounting
on the top and/or the bottom of the axle also meant that bending moments induced in
the axle by side forces on the wheels formed part of the dynamic strain gauge
signals. This resulted in complex signals that were difficult to analyse since the
signals included bending moment in the axle combined with the shear force at that
point (de Pont, 1997).

Previous research (Woodroofe et al., 1986) used strain gauge mounting
arrangements that, simply due to the commonality of strain gauge design, resulted in
spatial separation of the gauges. Similarly, when designing the tests for this thesis,
physical separation of strain gauge elements was inevitable since they could not be
installed on top of each other. To minimise, to the greatest degree, transverse wheel
forces or other axle bending moments being measured by the strain gauges, the
chevrons in the strain gauge arrays were mounted as close as possible to, and evenly
spaced either side of, the neutral axis of the axle on which they were installed.
Accordingly, the $F_{\text{shear}}$ data (Equation 3.1) were more easily analysed since they
were uninfluenced to the greatest pragmatic extent by any bending moment present
(de Pont, 1997).

Figure 3.13. Strain gauge mounted on the semi-trailer axle.
The strain gauges were not able to measure the inertial component of wheel forces further outboard from the point at which they were mounted. Accordingly, accelerometers were mounted outboard of the strain gauges and as closely as possible to the hub of interest (arrow, Figure 3.14). These measured acceleration data outboard of the strain gauges. The arrangement in Figure 3.14 was for the bus but this example was typical for all hubs measured.

![Figure 3.14. Accelerometer mounted on bus drive axle.](image)

### 3.2.6 Derivation of dynamic wheel forces

By recording the accelerometer and strain gauge signals, the terms of Equation 3.1 were used to derive dynamic wheel forces for each test. This is known as the “balance of forces” technique and is illustrated graphically in Figure 3.15 (Davis & Bunker, 2007).

The formula used for this technique is:

\[
F_{\text{wheel}} = F_{\text{shear}} + ma
\]

Equation 3.1

where:

- \(a\) = the acceleration experienced by the mass outboard of the strain gauge;
- \(m\) = the mass outboard of the strain gauge and acting at the centre of gravity (CoG)
of that mass; and

\[ F_{\text{shear}} = \text{the shear force on the axle at the strain gauge} \]

(Cebon, 1999; de Pont, 1997; LeBlanc, Woodrooffe, & Papagiannakis, 1992; Whittemore, 1969; Woodrooffe et al., 1986).

Figure 3.15. Showing variables used to derive dynamic tyre forces from an instrumented HV axle.

Note that the sense of the forces in Figure 3.15 and Equation 3.1 are such that the downward direction is positive.

As mentioned above, the accelerometers were mounted as closely as possible to the hubs on the axle/s of interest. Noting that the distance from the CoG of the axle to the CoG of the wheel is denoted \( d \) and the distance from the CoG of the axle to the accelerometer \( r \) in Figure 3.15. Pragmatic considerations such as physical access and wheel mechanical components precluded mounting the accelerometers at exactly the centre of gravity (CoG) of the masses outboard of the strain gauges, i.e., \( d \neq r \) (ref. Figure 3.15). The difference in values between \( d \) and \( r \) was able to be neglected since the roll angles were small and the value of \( (d-r) \ll d \) (Cebon, 1999). Further, when comparing the data from the test case with that for the control case for the same axle, errors due to \( d \neq r \) were common to both and therefore cancelled (Woodrooffe & LeBlanc, 1987). Previous work stated that even large variations in
the forces outboard of the strain gauges do not contribute greatly to the overall variation in measured wheel forces (de Pont, 1997).

The values of $m$ for the three test HVs were found from manufacturer’s data (Giacomini, 2007; Mack-Volvo, 2007a), calculation of trailer axle tube mass using steel density and by direct measurement. The direct measurement technique was used for the bus and the coach and involved cutting through damaged axle housings at the corresponding strain gauge mounting point. The axle remnants outboard of the respective strain gauge mounting points were then weighed (Appendix 1).

Further details of the rationale for the instrumentation, calibration and the methodology used in the derivation of the wheel forces have been documented in Appendix 1 and previous publications (Davis, 2007; Davis & Bunker, 2008a; Davis & Bunker, 2008b, 2009b).

### 3.2.7 Rationale for instrumentation – indicative pavement roughness

To inform Objective 2 (alterations to dynamic load sharing from larger longitudinal air lines) roughness values of the pavements encountered were derived. The accelerometer signals from the on-road testing were used to derive indicative roughness values. This was designated the “novel roughness” value to distinguish it from the standard definition of roughness used by pavement engineers. The usual definition of road roughness is an international standard measure termed the international roughness index (IRI). It is found by calibrated vehicles performing test runs. The data from such a run is a roughness measure of the tested pavement in mm/m or m/km. This is indicative of the amount of vertical movement of the calibrated vehicle relative to the horizontal distance travelled along the road during the test run. This measure may also be visualised as an upward slope that the vehicle negotiates as it travels. The steepness of the slope is proportional to the roughness of the road. This measure is now standardised for world-wide use (Sayers, Gillespie, & Hagan, 1987; Sayers, Gillespie, & Paterson, 1986).
### 3.2.8 Rationale for instrumentation – computer model of suspension

Under Objective 3, data from observed HV behaviour were used to determine fundamental HV suspension parameters during the quasi-static testing (Section 3.2.3). These data were gathered by recording the accelerations at the hubs of interest and the concomitant air spring responses and used to develop and validate computer models of the HV suspensions.

An air pressure transducer (APT) was connected into each air line supplying each air spring of interest on each axle of interest. APTs convert air pressure into a proportional electrical signal. Accordingly, signals proportional to the static and dynamic air spring forces were recorded for the tests. An APT (arrowed) is shown in Figure 3.16. Accelerometer installation details have been covered above.

![Figure 3.16. APT used for measuring air pressure at the axle/chassis interface.](image)

Accelerometer and air spring data were recorded during the “pipe tests” and the VSB11-style tests. The acceleration values at the hubs during on-road testing were recorded. Chapters 4 and 5 expand on this but suffice to say that these data were recorded for use as input signals to an analogue of a HV suspension as a second-order system (section 4.3.3). The APT data were likewise recorded as the corresponding outputs of that second-order system model.
3.2.9 Rationale for instrumentation - spring forces

To inform Objective 2 (alterations to dynamic load sharing from larger longitudinal air lines) and partially Objective 3 (in-service suspension testing) axle-to-chassis (air spring) force data were required. The air pressure transducer signals recorded from APTs installed as described above during the tests described above provided the data for that analysis.

3.2.10 Data recording

The test HVs were instrumented at each axle-end of interest to gather data for use in Objective 2 and partially Objective 3 as described above.

Each test run was recorded (Section 3.1.1) which resulted in 10,000 data points over a 10 s time-series signal from each APT, accelerometer and strain gauge on each axle-end of interest on each test HV for the road tests. The recording did not always start at the same point on test road segment due to human triggering. Accordingly, the data were examined, time series by time series, and the data matched in time to the same position of the road segment as determined from observing the same set of impulses in each pair (i.e. control case and test case) of data recordings. Start and finish times for the test case and the control case were then aligned so that the same recording interval over the particular road segment was used for each matched pair of time-series data.

3.3 HV suspension tests- remainder of Objective 3

3.3.1 General description

A used roller-type HV brake tester was modified and instrumented. The testing was carried out in accordance with the test and safety plan for this portion of the project (Davis & Bunker, 2008b). This section of Chapter 3 summarises a portion of previous work documented elsewhere (Davis & Bunker, 2009c) and elaborates on test methodology not yet published.

A test HV wheel was excited by two flat areas on a roller when it was spun up to
speed. Instrumentation measured the wheel forces thus created.

Dampers with differing levels of wear were fitted. The control case was for a new shock absorber. The HV industry advises that flat spots on tyres (Australian Road Transport Suppliers Association, 2001) are indicative of worn shock absorbers. An example of a HV tyre exhibiting this type of wear is shown in Figure 3.17 with flat spots (arrowed). Accordingly, a shock absorber causing that symptom was chosen as one test case. The other test case was for no shock absorber, simulating a damper that had ceased to function.

The wheel forces were analysed for magnitude and frequency for the cases tested. Threshold values for shock absorber wear vs. wheel forces were determined as a “proof-of-concept” for an in-service HV suspension test. This informed the NTC in-service suspension testing project (Section 1.4.4) and assisted in proposing low-cost HV air suspension test methods.

![Figure 3.17. HV tyre exhibiting symptoms of damper wear.](image)

### 3.3.2 Detail

A used HV brake testing machine was acquired. The brake testing functions were not used for the test programme. The only features of the brake tester used were two rollers and the frame.

The test rig developed was modified from the original brake tester in that the wheel under test sat on top of one roller only. A general view of the test rig used is shown in Figure 3.18; the power unit is in the foreground with the roller mounted at the end.
of the frame furthermost from the viewer.

![End view of modified roller brake tester.](image)

Figure 3.18. End view of modified roller brake tester.

Two different rollers were used. They had two diametrically opposed flat areas (Figure 3.19 and Figure 3.20) machined into them. The flat areas on one roller were of radial depth 2 mm and the other roller had flat areas of radial depth 4 mm. The depth of the flat areas was defined as the radial distance from the original roller circumference to the centre of the final flat area (Figure 3.19).
The amount of material removed to create the flat areas was chosen to provide sufficient depth of eccentricity so that results could be compared with similar research (Woodrooffe, 1996). The eccentricity created provided repetitive input to the test vehicle’s wheel.
The brake tester’s original drive motors and ancillary devices such as the arm on the idler roller and the testing control console were discarded. A 22 kW capacity electric motor drove a hydraulic pump that, in turn, drove a hydraulic motor. The hydraulic motor was connected to the roller and used as the final power unit for the testing (Figure 3.21).

![22 kW motor (foreground), coupled to hydraulic pump in the hydraulic fluid reservoir.](image)

Effects of motor torque altering the readings on the load cells were minimised to the greatest extent by mounting the motor on a plate attached to a torque arm. This torque arm is shown in Figure 3.22. In this way, vertical loads presented to (and measured by) the load cells were from the HV wheel under test and did not contain a force component due to motor torque.
Figure 3.22. Hydraulic motor with roller and coupling removed. Note torque arm, slightly obscured (arrow A) was connected to hydraulic motor mounting plate, B. Locating frame (C) allowed motor mount to float.

Figure 3.23. The test rig in the pit. Arrow A shows hydraulic safety cut off switch.
The hydraulic motor was controlled *via* valves and a safety switch. The safety switch is shown (Arrow "A") in Figure 3.23. The speed of the roller was controlled *via* a coupling to the hydraulic bypass valve shown (arrow "C") in Figure 3.24.

### 3.3.3 Roller bed installation

The modified test rig retained the overall frame dimensions of the original tester from which it was made. Brake testers are designed to be installed in roadside pits during HV interceptions by regulatory authorities or in pits as part of a HV service facility. A pit was constructed as specified by the manufacturer of the brake tester (FKI Crypton Ltd, 1990).
3.3.4 Positioning the test wheel

The test HV (a semi-trailer) was positioned such that the right middle wheel of its tri-axle group was on the roller. The left wheel of the middle axle was constrained using extendable retaining frames (indicated by arrows "A", Figure 3.24) and chained down (arrow "B", Figure 3.24) as an added precaution (Davis & Bunker, 2008b). The chains were applied firmly but not so tightly that the axle was constrained from rolling and thereby unduly influencing the results. The brakes on the trailer were applied except for the brake on the wheel under test. These measures were a safety precaution due to the uncertain nature of any testing. It is expected that this level of restraint would not be necessary should this test procedure be implemented commercially.

The final arrangement of the wheel and its relationship with the drive roller is shown in Figure 3.25.

![Figure 3.25. The final position of the roller in relation to the HV wheel under test.](image)

3.3.5 Instrumentation

Commercial load cells (strain gauges measuring deflection in a steel block) were installed under the bearings at each end of the roller. Figure 3.26 and Figure 3.27 show this detail. The dynamic wheel force was found from the sum of the forces on the load cells. This was calibrated as wheel force in kilograms. The dynamic signals from the load cells were recorded for 10 s at 1.0 kHz during each test run using a modified CHEK-WAY® on-board HV telemetry system as used for testing the three HVs as detailed in Section 3.1.1. This resulted in test data in the form of 10,000 data...
points over a 10 s time-series signal from each strain gauge on each end of the roller for each test.

Figure 3.26. Load cell (indicated A) under roller LHS bearing.

Figure 3.27. Load cell (indicated B) under roller RHS bearing.

3.3.6 Operation

Figure 3.28 shows the test HV wheel run up to speed on the roller. This resulted in the HV wheel being presented with a cyclic loading from the two flat areas on the roller.
3.3.7 Tested conditions

The condition states of the suspension dampers were:

- no shock absorber;
- a shock absorber that had worn to the point where it had been removed from its donor vehicle because abnormal tyre wear (i.e. flat spots) had become apparent; and
- a fully functional shock absorber (i.e. within specification) for the suspension being measured.

Tests were conducted comprising combinations of three shock absorber condition states, two different values of flat areas on the rollers and loads of 1/3, 2/3 and full. The full load condition approximated a full load (3 t) on the wheel, the 2/3 load was 2049 kg and the 1/3 load was 1167 kg on the wheel. The tests are summarised in Table 3.2.

The hydraulic valves of the test rig were set so that the final speed of the roller reached approximately 360 revolutions per minute. This meant that, with two flats per revolution, the frequency of the forcing function from the flats of the roller was
approximately 12 Hz. This was at or above the highest axle-hop frequency measured for this trailer and suspension (Davis & Bunker, 2008e).

Table 3.2. Combination of load conditions, damper condition states and roller flats tested.

<table>
<thead>
<tr>
<th>Roller flat dimension</th>
<th>Load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1/3</td>
</tr>
<tr>
<td>2 mm</td>
<td>No damper</td>
</tr>
<tr>
<td>2 mm</td>
<td>Worn damper</td>
</tr>
<tr>
<td>2 mm</td>
<td>New damper</td>
</tr>
<tr>
<td>4 mm</td>
<td></td>
</tr>
<tr>
<td>4 mm</td>
<td></td>
</tr>
<tr>
<td>4 mm</td>
<td></td>
</tr>
</tbody>
</table>

3.4 On-board mass (OBM) accuracy and tamper-objective 4

3.4.1 Introduction and overview

The testing program was conducted in line with the test plans (Davis et al., 2008a, 2008b). A pilot test programme was followed by full-scale testing. This section summarises the data gathering procedures relevant to this thesis. Other tests were undertaken; their descriptions and results have been documented elsewhere (Davis et al., 2008a, 2008b; Davis, Bunker, & Karl, 2009; Karl et al., 2009).

Twelve test and control OBM systems from eight suppliers were installed on eleven HVs. The details and brands of the OBM systems have been made anonymous here and in other work (Karl et al., 2009b) due to the competitive nature of the OBM industry in Australia.

The HVs were loaded to tare, 1/3, 2/3 and full load points. The HVs were then weighed six times per load point on weighbridges after a short road circuit. Dynamic data were recorded during the road circuits. Examples of the tested HVs, some parked on the weighbridges used, are shown from Figure 3.29 to Figure 3.32.
Figure 3.29.  Test HV – small road train.

Figure 3.30.  Test HV – detail of small road train trailers.

Figure 3.31.  Test HV – truck and dog trailer on weighbridge with similar combination following.
The weighbridge readings were compared with the static OBM readings. A range of dynamic data was recorded as the test vehicles encountered speed bumps, braking and cornering. The intermediate loadings at the 1/3 and 2/3 load points were included to determine linearity in the relationship between OBM data and the weighbridge data. Loads at those load points were not exact and did not need to be; loading at approximately these load points sufficed for the purpose of determining linearity.

Tests were performed to determine potential for, and ease of, tampering. Various tamper tests were performed; the one that concerns this thesis directly was where the air lines to the OBM primary transducers were closed off. Dynamic data were recorded for open and closed APT air line states. The two sets of data were compared and an algorithm developed to detect the differences.
3.4.2 Sampling frequency

It was postulated that dynamic data could inform the development of a tamper indicator. Further to this, it was theorised that signal frequencies of the pressure in the air springs and influenced by axle-hop would be key to this development.

As noted in Chapter 3, Section 3.1.1 and subsequently, HV axle-hop occurs at frequencies of up to 15 Hz; periodicity 66.7 ms (Cebon, 1999). For the Nyquist sampling criterion (Shannon’s theorem, Chapter 2, Section 2.3.3) to be satisfied, data capture needed to be at least twice the frequency of interest (Houpis & Lamont, 1985) or at least 30 Hz; periodicity 33.3 ms.

Most commercial OBM systems do not record dynamic data at frequencies that would have allowed axle-hop to be recorded but most manufacturers have stated that their systems would be capable of such measurement, were it required (Davis, 2008). Accordingly, data for use in Objective 4 were recorded using a HV OBM system whose manufacturer had specified a sample interval of 24.0 ms, providing a sampling rate of 41.6 Hz (Davis et al., 2008a, 2008b). As outlined in the background theory for dynamic data sampling (Chapter 2, Section 2.3.3) this sampling frequency was more than adequate to capture the test signal data since 41.6 Hz was higher than the 30 Hz value derived from twice axle-hop. Accordingly, the Nyquist sampling criterion (Shannon’s theorem) was met (Houpis & Lamont, 1985) for the testing.

3.4.3 Procedural detail

Each OBM system tested had a data set recorded for that unit. These data sets were measured against a weighbridge or certified scales.

As a cross-validation of the data set from each test OBM, particularly for the dynamic data recorded, another set of data was recorded. These data were recorded by systems that were universal for all tests, regardless of vehicle. The data from these systems provided sets of static and dynamic control data. Two OBM systems common to all the tests and vehicles were used and were termed the reference OBM systems. One of them recorded static and dynamic data contemporaneously with the OBM system under test. The other recorded static data only. One outcome of the
use of the reference OBM systems was the ability to compare the measured mass (MM) reading of the reference OBM systems with the MM reading of the test system and with the reference mass (RM) reading from the weighbridge. Accordingly, at least three measured mass (MM) readings were taken per test, one from each of the reference OBM systems and one from the OBM system under test (Davis et al., 2008a, 2008b).

3.4.4 Tamper tests

HV regulators regard tampering as a major issue. Controlled tampering during the tests was carried out to determine whether the effects of that tampering could be detected from changes in the data. Accordingly, some basic tampering that involved changing the operation of the test vehicle or its systems occurred. Some tested OBM systems used air spring pressure to determine on-board mass readings. For those cases, a ball valve or turncock valve was interposed in the air line between the APT and its associated air springs (Figure 3.33). This allowed the pressure in the air springs of the axle group being measured to determine the mass on that group to be presented to the APT or not. Accordingly, this simulated the case where an operator may block the APT air line in an attempt to provide an OBM system with a false, and lower, group mass than the actual mass (Davis et al., 2008a, 2008b).

Figure 3.33. Ball value interposed between air spring and APT.
3.4.5 Sample size

For each load condition, viz: tare, 1/3, 2/3 and full load; data readings were taken a number of times to improve the reliability and accuracy of the results.

The sample size was determined from the process outlined in Appendix 3 and summarised here.

A 95 percent level of confidence was chosen. Reasonable assumptions about:

- an experimental error value; and
- the spread of measurements from the population of OBM systems,

led to six readings per load condition. This meant six readings of the reference mass (RM) and the measured mass (MM) per test load condition without changing any other variables.

3.4.6 Exercising the HV suspensions

The suspensions of the tested HVs were exercised between readings to ensure that influences due to bushing hysteresis, inter-leaf friction and air bag stretch, etc., were averaged out over the readings. That meant that each test HV was required to perform some travel activity before returning to be weighed again, typically a circuit around a suburban block or within a transport depot.

3.5 Summary and conclusions of this chapter

This chapter has set down the background philosophy, methodology and choices made in the experimental designs for the parts of the projects involved in preparation of this thesis.

Some of the areas explored by the objectives and aims of this project were not mainstream. A distinction needs to be made, however, between that which was being tested and the methods used to test it. Ultimately, the data gathered involved exercising the test HV in question and recording the data. Wherever a choice needed
to be made between unproven or conservative methodology, the latter was chosen. This methodology was then applied to the experimental design that provided the data used in this thesis.

Where stochastic influences were foreseen in the experimental design, such as those experienced with weighing devices (i.e. on-board mass measurement systems or weighbridges), appropriate sample population numbers were developed using mature methodology (Dubes, 1968; Snedecor & Cochran, 1967).

Likewise, conservative and proven methodology was chosen to determine wheel forces from the experimental HVs (de Pont, 1997; Woodroofe et al., 1986).

### 3.6 Chapter close

This chapter has described the procedures used for the testing and data gathering, including (and/or referenced in Appendices) the conditions and rationale surrounding each method.

Data thus gathered were analysed to:

- evaluate systems such as the “Haire suspension system” in Chapter 10 by making use of HV suspension metrics from Chapter 2 and the roughness measures derived in Chapter 7;
- validate computer models in Chapter 5 after development of those models in Chapter 4, noting that vertical acceleration data at the hubs of interest and the concomitant air spring responses form the input and output data for those models;
- evaluate variations in the load-sharing ability of computer models from Chapter 4 and Chapter 5 to propose further improvements in HV suspension designs in Chapter 12, Section 12.2.4 and Chapter 13, Section 13.3.1; and
- characterise on-board mass measurement systems in Chapter 8 and develop and validate algorithms for OBM tamper-evidence in
Chapter 9.

That data and the models developed will then be used, amongst other outcomes, to propose an in-service HV suspension health regime in Chapter 11 by a fusion of OBM implementation and low-cost methodology for HV suspension testing.

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“There are two possible outcomes: if the result confirms the hypothesis, then you've made a measurement. If the result is contrary to the hypothesis, then you've made a discovery.” - Enrico Fermi.
4 Development of heavy vehicle suspension models

4.1 About this chapter

This chapter details the development of heavy vehicle (HV) suspension models used later in this thesis. Further, this chapter contains linkages between those models and actual HV suspensions tested and described in Section 3.2. This work will be used to address Objectives 1 to 3 in the areas of dynamic load sharing and in-service suspension testing. This chapter then allows the scene to be set for validation of the theoretical models, analysis of their outputs and conclusions drawn therefrom in Chapter 6, particularly when correlated to actual HV suspension test data gathered as detailed in Chapter 3.

4.2 Dynamic load sharing

4.2.1 Suspension model

Figure 4.1 presents a simple theoretical model that may assist to explore the concept of dynamic load sharing between wheels or axles in a HV. The wheels, axles, etc. on one axle in Figure 4.1 are assumed to be of equal mass to their corresponding equivalent component on the other axle.

Let us consider Figure 4.1 in the context of a connection mechanism using conventionally sized or “industry standard” longitudinal air lines which have dimensions of approximately 4 to 12 mm inside diameter. Such a design does not allow transfer of air between the air springs quickly enough when either of the wheels encounters a bump at typical HV operating speeds (Blanksby et al., 2008a; Davis & Sack, 2004; Simmons, 2005). This configuration may be considered to exhibit no dynamic load sharing; therefore, its axles may be considered independent of each other. Should one wheel encounter a non-uniformity of large enough magnitude, were tyre elasticity not able to accommodate it; the other wheel could be lifted off the ground momentarily. This would transfer the group mass to the wheel still in contact with the road at that moment.
Now consider the model in Figure 4.1 with an air spring connection where air is transferred effortlessly (and pressure equalised instantaneously) between air springs, should a non-uniformity be encountered. Any inequality of load between the two wheels would be balanced out during travel of the HV. A continuum of possibilities arises between the two connection mechanism scenarios and is dependent on the magnitude of irregularities in the pavement and tyre elasticity. Whether increased load sharing designs can be implemented will also figure in any outcomes.

To develop the simple model in Figure 4.1 further, for validation against real-world situations, the axle spacing may be chosen to be 1.4 m.

The testing programme described in Section 3.2 used a semi-trailer and a coach with that axle spacing. It is for noting that a typical 11R22.5 HV tyre has a radius of 525mm with a rolling radius of 489mm (Continental tyres, 2005; Goodyear, 2005).

Let the air spring connection mechanism in Figure 4.1 be able to transfer air effortlessly between the air springs. The elapsed time between one wheel encountering a bump and its rear neighbour meeting the same bump will be dependent on the speed of travel. The time delay between the two wheels being disturbed will be inversely proportional to the speed of travel. This elapsed time may
be designated:

\[ t = \frac{d}{v} \]  

Equation 4.1

where:

\( v = \text{speed in ms}^{-1}; \)

\( d = \text{distance in m}; \) and

\( t = \text{time in s}. \)

The time constant of a system thus modelled may be represented as the difference in time between the events of the two wheels encountering the same bump. In any system, the time constant may also be defined as the inverse of the fundamental (or resonant) frequency of the system.

The modes of vibration and frequencies of axles and bodies of HVs have been documented extensively (Cebon, 1999; Davis & Bunker, 2008a, 2008e; de Pont, 1997). Providing the wheels and other rotating parts are balanced and suspension components such as wheel bearings and shock absorbers are in good order, axle-hop will occur when a wheel encounters a discontinuity in the road surface. Such an impulse will start axle-hop that is characterized by a series of reducing excursions of the wheel. Axle-hop frequencies occur in the range 10 to 15 Hz and, as speeds increase, tend to the same order of magnitude as those of body bounce frequencies (Davis & Bunker, 2008e). How quickly axle-hop excursions reduce will be a function of suspension damper health and design characteristic (Section 2.3.1). For the model in Figure 4.1, axle-hop forces would be transmitted along the connection mechanism by the air spring first encountering a bump. These would be passed, in a series of pneumatic pulses, to the rear air spring via the connection mechanism. Shock absorbers in good condition should damp out this vibration but not before the second air spring received a series of pressure impulses at the axle-hop frequency common to both axles.

The time between these events for 1.4 m axle spacing would depend on travel speed (Equation 4.1) and is shown in Table 4.1.
Assuming an idealised transfer of air from one air spring to its rear neighbour, Table 4.1 shows that speeds between 60 km/h and 70 km/h for axles spaced at 1.4 m would result in a resonant frequency in the system of between 12 and 14 Hz.

Table 4.1. Relationship between different speeds and the elapsed time between wheels at 1.4 m spacing.

<table>
<thead>
<tr>
<th>Speed (km/h)</th>
<th>Elapsed time between axles passing a point on the road (ms)</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>0.126</td>
<td>8</td>
</tr>
<tr>
<td>60</td>
<td>0.084</td>
<td>12</td>
</tr>
<tr>
<td>70</td>
<td>0.072</td>
<td>14</td>
</tr>
<tr>
<td>80</td>
<td>0.063</td>
<td>16</td>
</tr>
<tr>
<td>90</td>
<td>0.056</td>
<td>18</td>
</tr>
</tbody>
</table>

These frequencies are coincident with the axle-hop frequencies in the 10-15 Hz range (Cebon, 1999; Davis & Bunker, 2008e; de Pont, 1997). This detail will inform the discussion on dynamic load sharing in Chapter 12.

4.3 HV suspension computer model

4.3.1 Free-body diagram

In order to construct a computer model of a HV suspension air spring, system equations were developed. Considering the free-body diagram of a half-axle (i.e. the wheel in one corner of a HV, or “quarter-HV”) in Figure 4.2, then the pressure in the air springs may be considered proportional to the displacement between the body and the axle, or a variable derived from the result of subtracting displacement $x$ from displacement $y$. 

Section 3.2 detailed the methodology for recording suspension data from instrumented HVs. Since an instrumented pavement was not used, wheel forces were not measured directly. Wheel forces were derived from a combination of acceleration and strain gauge data. These results are detailed later and yielded force at \( z \) derived by using Equation 3.1. Acceleration data at the axles were measured directly, not derived. To develop the most accurate model from the data available, the acceleration data were chosen as the input since these data were available as low-error direct measurements (Section 3.2.8 and Appendix 2).

Consider axle acceleration, \( \ddot{y} \), influenced by axle mass, \( m_\text{a} \). The displacement variable, \( y \), may be derived from a double integral of the acceleration at the axle, or the integral of the axle velocity, \( \dot{y} \), where these are known data. For the model developed here, the acceleration at the axle was a known. It was used as an input signal to the model since:

\[
\begin{align*}
    y &= \text{displacement of the axle in m}; \\
    \dot{y} &= \text{velocity of the axle in ms}^{-1}; \text{ and} \\
    \ddot{y} &= \text{acceleration of the axle in ms}^{-2}.
\end{align*}
\]

Hence, the diagram in Figure 4.2 was redrawn by considering the forces and displacement at the axle; this is shown in Figure 4.3.
Nota bene: the tyre spring constant, $k_t$, does not appear in Figure 4.3. This was not to discount the influence of tyre spring rate within the model; the model in Figure 4.3 was developed to be viewed from the forces at the axle, not the wheels. This from the pragmatic construct of available low-error empirical acceleration data at the axle. The influence of the tyre spring constant, $k_t$, was included later and is detailed in Section 4.3.8.

Note that the sense of the displacements (and forces) in Figure 4.2 and Figure 4.3 were such that the downward direction was positive.

Assumptions:

a HV suspension is a classically underdamped second-order system governed by a second-order differential equation;

any second-order system has the differential equation, from first principles:

\[ \ddot{p} = f(t, \dot{p}, p) ; \]

Equation 4.2
and the generalised system equation for such a body with damping (Meriam & Kraige, 1993) is:

\[ m\ddot{p} + c\dot{p} + k_p = 0 \]

Equation 4.3

where:

- \( m \) is the mass of the body;
- \( c \) is the damping coefficient;
- \( k \) is the spring rate;
- \( p \) is the time-dependent system response.

Applying the generalised system equation, Equation 4.3, to the free-body diagram in Figure 4.3, some detailed assumptions may reasonably be made:

- the system equation for the free-body diagram in Figure 4.3 for the sprung mass is: \( m_s\ddot{x} + c_s\dot{x} + k_s x = 0 \);
- the spring rate, \( k_s \), is linear regardless of direction;
- the damping coefficient, \( c_s \), varies piecewise-linearly according to direction of movement (Costanzi & Cebon, 2005, 2006; Duym, Stiens, & Reybrouck, 1997; Prem et al., 1998; Uffelmann & Walter, 1994);
- bushings, locating rods or other suspension components add no differential spring action, dead-band or hysteresis; and
- the springs and the dampers do not reach their limits of travel (i.e. no spring or damper hysteresis; no “bang-bang” limiting).

These assumptions were proven valid as detailed later (Chapter 5); should they not have been, they would have been revisited.
4.3.2 Regarding spring rate linearity and the damping characteristic

Modelling of HV suspensions that included air springs with non-linear spring rates over the entire range of spring travel has been carried out (Costanzi & Cebon, 2005, 2006). That work involved complex modelling and allocated some hysteresis to the springs at the limits of air spring travel (Costanzi & Cebon, 2005, 2006). However, Costanzi and Cebon (2005, 2006) noted that air spring hysteresis occurred over a very small portion of the overall range. Other researchers have produced less complex models of HVs suspensions that were used for pavement damage research (Cole & Cebon, 2007; Prem et al., 1998). Those models ascribed a linear value to the spring rate, $k_s$. Accordingly, for the models used in this thesis, linear spring rates were chosen. Manufacturer’s data (Mack-Volvo, 2007b) and the previous work on simple-to-moderately complex HV suspension models justified this assumption.

The moderately complex models of HV suspensions used for pavement damage research and development of HV suspension designs used models with a piece-wise linear damping coefficient, $c_s$, which varied depending on direction (Cole & Cebon, 2007; Duym et al., 1997; Prem et al., 1998). Even the complex modelling of Costanzi and Cebon (2005, 2006) assumed linear, but unequal, damping coefficients in either direction. For the models used in this thesis, piecewise-linear damping coefficients were derived for damper bounce and rebound cases.

These assumptions were proven valid as detailed later (Chapter 5); should they not have been, they would have been revisited.

4.3.3 System equations

Newton’s Second Law and Newtonian mechanics allowed a system equation to be developed using the relationship that the axle has with the body of the HV in Figure 4.3 as follows:

Newton’s Second Law states:

$$\text{Force} = \text{mass} \times \text{acceleration}$$

Equation 4.4
The first force to consider on the body of the HV in Figure 4.3 is due to its mass and acceleration; using Equation 4.4:

\[ \text{Force (due to mass and acceleration of the sprung mass)} = m_s \ddot{x} \quad \text{Equation 4.5} \]

where:

\( m_s \) = sprung mass of the HV; and

\( \ddot{x} \) = acceleration at the body of the HV.

Now consider the spatial relationship that the body and the axle have with respect to each other. That relationship may be defined by the forces on the components in Figure 4.3. The force on a damper is proportional to the relative velocities between its ends as defined by its damping coefficient, \( c_s \). From Figure 4.3, this force may be described as:

\[ \text{Damper force} = c_s (\dot{x} - \dot{y}) \quad \text{Equation 4.6} \]

where \( c_s \) = damping coefficient of the shock absorber;

\( \dot{x} \) = velocity of the body of the HV; and

\( \dot{y} \) = velocity of the axle.

The spring force is defined by its spring rate, \( k_s \), and is proportional to the relative displacement between its two ends. From Figure 4.3, this force may be described as:

\[ \text{Spring force} = k_s (x - y) \quad \text{Equation 4.7} \]

where \( k_s \) = spring rate;

\( x \) = displacement of the body of the HV; and

\( y \) = displacement of the axle.

The sum of the total force (\( \text{Force}_{\text{total}} \)) on the HV body in Figure 4.3 may be found
by adding the forces found from Equation 4.5, Equation 4.6 and Equation 4.7:

\[ \text{Force}_{(\text{total})} = (m_y \ddot{x}) + c_s (\ddot{x} - \ddot{y}) + k_s (x - y) \]

Equation 4.8

from Newtonian mechanics, the forces on a body may be summed to zero (Meriam & Kraige, 1993):

\[ \text{Force}_{(\text{total})} = 0 \]

Equation 4.9

since the HV body exhibits second-order underdamped behaviour, Equation 4.8 and Equation 4.9 may be equated and re-written:

\[ \text{Force}_{(\text{total})} = (m_y \ddot{x}) + c_s (\ddot{x} - \ddot{y}) + k_s (x - y) = 0 \]

Equation 4.10

From Equation 4.10 therefore:

\[ (m_y \ddot{x}) + c_s (\ddot{x} - \ddot{y}) + k_s (x - y) = 0 \]

Equation 4.11

\[ \Rightarrow \quad m_y \ddot{x} + c_s (\ddot{x} - \ddot{y}) + k_s (x - y) = 0 \]

\[ \Rightarrow \quad (m_y \ddot{x}) = c_s (\ddot{y} - \ddot{x}) + k_s (y - x) \]

Equation 4.12

and restating the variables and their units:

\[ m_y = \text{the mass of the body in kg}; \]
\[ c_s = \text{the damping coefficient (\textit{nota bene: not} the damping ratio) of the shock absorber in kNs/m}; \]
\[ k_s = \text{the spring constant in kN/m}; \]
\[ y = \text{displacement of the axle in m}; \]
\[ \dot{y} = \text{velocity of the axle in m.s}^{-1}; \]
\[ x = \text{displacement of the body in m}; \]
\[ \dot{x} = \text{velocity of the body in m.s}^{-1}; \] and
\[ \ddot{x} = \text{acceleration of the body in m.s}^{-2}. \]

Equation 4.12 shows that the forces on the body are created by the forces from the shock absorber and the spring combined. This equation was used to develop the computer models used in the following sections and in Chapter 5.

### 4.3.4 Damped natural frequency

Assuming underdamped behaviour, with some justification from empirical evidence (Davis & Bunker, 2008a; Davis, Kel, & Sack, 2007; Davis & Sack, 2004, 2006), the equation of motion from an underdamped second-order system equation provided the relationship between the undamped natural frequency, \( \omega_n \), the damped natural frequency, \( \omega_d \), and the damping ratio, \( \zeta \) (defined in Section 4.3.5):

\[
\omega_n = \frac{\omega_d}{\sqrt{1 - \zeta^2}}
\]

Equation 4.13

where:

\( \omega_d \) = the damped natural frequency; or body bounce frequency, in rad.s\(^{-1}\);

\( \omega_n \) = undamped natural frequency; and

\( \zeta \) = the damping ratio (Meriam & Kraige, 1993; Thomson & Dahleh, 1998).

The damped natural frequency, \( f \), is the inverse of the time (period) between successive points on the output waveform (e.g. successive peaks or successive zero-crossings) of a second-order system response to an impulse input. Figure 4.4 shows this time as \( T_d \).

From first principles:

\[
f = \frac{1}{T_d} = \frac{\omega_d}{2\pi};
\]

Equation 4.14
hence inverting $T_d$ provided the damped natural frequency for the development of computer models in this chapter and the next, noting that the Système International d'Unités (SI) derived unit for frequency or vibration is Hertz (Hz) of which the derivation is $s^{-1}$ with the appropriate multiplier $2\pi$ to convert frequency to rad.$s^{-1}$.

These equations will be used later.

![Figure 4.4. Illustrating the values used to derive system equations of a second-order system.](image)

### 4.3.5 Damping ratio – full wave data

The damping ratio ($\zeta$) may be determined by comparing the values of any two consecutive peaks in the same phase (i.e. comparing the magnitudes of the first and third excursions or the second and fourth excursions) of the response output signal of an underdamped second-order system after an impulse function input has been applied (Meriam & Kraige, 1993).

Prem et al., (2001) used the following formula (Meriam & Kraige, 1993) to determine the damping ratio of a HV suspension:
\[
\delta = \frac{2\pi\zeta}{\sqrt{1 - \zeta^2}} = \zeta \omega_n \tau_d
\]

Equation 4.15

where:

\(\zeta\) = the damping ratio;

\(\tau_d\) = the damped natural period;

\(\omega_n\) = the undamped natural frequency = \(\frac{\omega_d}{\sqrt{1 - \zeta^2}}\);

Equation 4.16

\(\omega_d\) = the damped natural frequency; and

\(\delta\) = the standard logarithmic decrement (Meriam & Kraige, 1993) given by the following formula:

\[
\delta = \ln \left( \frac{A_1}{A_2} \right)
\]

Equation 4.17

where:

\(A_1\) = amplitude of the first peak of the response; and

\(A_2\) = amplitude of the third peak of the response or

\(A_1\) and \(A_2\), as the first two peaks of the response that are in the same direction, \(i.e.\) on the same side of the x-axis of the time-series signal of the response; as shown in Figure 4.4 (Meriam & Kraige, 1993).

These may be derived from first principles from the equations of motion for second-order systems (Meriam & Kraige, 1993; Thomson & Dahleh, 1998).

Note: Starting with Equation 4.15 and solving for \(\zeta\) (Meriam & Kraige, 1993) the following relationship between \(\zeta\) and \(\delta\) may be found:
\[ \zeta = \delta \sqrt{\left(\frac{2\pi}{\zeta}\right)^2 + \delta^2} \]

Equation 4.18

as shown in other work (Davis & Bunker, 2007).

### 4.3.6 Damping ratio – half wave data

Where a half-cycle of the response from a second-order system to an impulse is available, the half-cycle damping ratio may be found by using:

- the first two peaks: \( A_1, A_{1.5} \); and
- half the damped natural period \( \frac{\tau_d}{2} \);

from those variables, as shown in Figure 4.4.

Hence the period between \( A_1 \) and \( A_{1.5} \) is half the damped natural period or \( \frac{\tau_d}{2} \).

The damping ratio from a half-wave signal, \( \delta_{1/2} \), may be derived from the same equations of motion used to derive the full-wave damping ratio above by restating Equation 4.15 (Thomson & Dahleh, 1998):

\[ \delta = \xi \Theta \frac{\tau_d}{2} = \frac{2\pi \xi}{\sqrt{1 - \xi^2}} \]

Equation 4.19

then substituting \( \frac{\tau_d}{2} \) for the period and adjusting the other sides of the equation for equality:

\[ \delta_{1/2} = \xi \Theta \frac{\tau_d}{2} = \frac{\pi \xi}{\sqrt{1 - \xi^2}} \]

Equation 4.20

where:

\( \zeta \) = damping ratio;
\[ \delta_{1/2} = \ln \left( A_1 / A_{1.5} \right) ; \]

\( \omega_n \) = undamped natural frequency; and

\( \tau_d \) = damped natural period.

Equating only the first and last terms of Equation 4.20 yields:

\[ \delta_{1/2} = \frac{\pi \zeta}{\sqrt{1 - \zeta^2}} \]

Equation 4.15

\[ \Rightarrow \delta_{1/2}^2 = \frac{\pi^2 \zeta^2}{1 - \zeta^2} \]

\[ \Rightarrow \zeta = \frac{\delta_{1/2}}{\sqrt{\delta_{1/2}^2 + \pi^2}} \]

Equation 4.21

These equations (Davis & Bunker, 2008a, 2008c) will be used later.

4.3.7 Second-order system generic model

Since the acceleration at the axle, \( \ddot{y} \), was known from testing, it was used as an input to the model. This variable was not part of Equation 4.12, but the vertical velocity of the axle, \( \dot{y} \), was. This allowed Equation 4.12 to be developed into a simple Simulink Matlab\textsuperscript{®} control system block diagram shown in Figure 4.5, given that the integral of \( \ddot{y} \) is \( \dot{y} \) (\textit{nota bene}: this model did not yet have the bump and rebound damping coefficients, that subtlety was incorporated later and addressed below):
where:

- the output (Scope) was the APT pressure proportional to the displacement between the body and the axle \((y - x)\); and
- the input signal (Signal 2) was the vertical acceleration signal measured at the axle, \(\ddot{y}\).

As mentioned in Section 4.3.2, suspension dampers have non-linear characteristics related to directional velocity; *i.e.* the damping characteristic varies with speed and direction of movement. This is to provide different dynamic resistances (*i.e.* damping coefficients) when the wheels hit a bump and then undergo rebound. This differential damping characteristic allows suspensions to control and optimise tyre contact with the road during travel over undulations and non-uniformities. This design feature required the inclusion of two damping coefficients (and therefore bump and rebound damping ratios) in the models.

This will be expanded in Chapter 5.
4.3.8 Regarding the influence of the tyres

The input for the model was taken to be from the vertical acceleration, $\ddot{y}$, of the axle mass. Even so, tyre spring rate and tyre damping both influenced measurement of this parameter. This was because axle-hop and tyre bounce contribute to axle and air spring behaviours. Previous researchers have noted this effect (Fletcher et al., 2002). Table 4.2 shows some of the variables and their units contained in vehicle models incorporating tyre parameters from Fletcher et al. (2002).

Table 4.2. Parameters used in HV suspension models that include tyre characteristics.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body bounce frequency</td>
<td>$\omega_d$</td>
<td>rad.s$^{-1}$</td>
</tr>
<tr>
<td>Axle-hop frequency</td>
<td>$\omega_{axle}$</td>
<td>rad.s$^{-1}$</td>
</tr>
<tr>
<td>Damping ratio</td>
<td>$\zeta$</td>
<td>n/a</td>
</tr>
<tr>
<td>Sprung mass</td>
<td>$m_s$</td>
<td>kg</td>
</tr>
<tr>
<td>Unsprung mass $m_u$</td>
<td>$m_u$</td>
<td>kg</td>
</tr>
<tr>
<td>Suspension spring rate</td>
<td>$k_s$</td>
<td>N/m</td>
</tr>
<tr>
<td>Suspension damping coefficient</td>
<td>$c_s$</td>
<td>Nm/s</td>
</tr>
<tr>
<td>Tyre spring rate</td>
<td>$k_t$</td>
<td>N/m</td>
</tr>
<tr>
<td>Tyre damping coefficient</td>
<td>$c_t$</td>
<td>Ns/m</td>
</tr>
</tbody>
</table>

The relationship between the damping ratio, $\zeta$, sprung mass, $m_s$, and damping coefficient, $c_s$, may be derived from first principles as shown in the equality portions of Equation 4.22 and Equation 4.23 (Thomson & Dahleh, 1998). However, Fletcher et al., (2002) also documented the relationship between the variables in Table 4.2 to undertake HV modelling that included the influence of the tyre spring rate, $k_t$. For a quarter-truck model, its two predominant modes of oscillation being characterised by body-bounce and axle hop, estimates of undamped body bounce natural frequency and damping ratio may be made using approximations shown as expressions on the RHS of Equation 4.22 and Equation 4.23 (Fletcher et al., 2002).

It is for noting that the units in the simplifications (RHS of the expressions) do not match the units of the derived variables for frequency or damping ratio; the
expressions are provided here, and were used as indicative checks, when deriving the model parameters in Chapter 5. The differences in the derived variables (discounting the mis-match in units) between the approximations and the equalities were negligible.

\[
\omega_n = \frac{c_s}{2\zeta m_s} \approx \frac{k_s k_t}{\sqrt{(k_s + k_t)m_s}}
\]

Equation 4.22

\[
\zeta = \frac{c_s}{2m_s \omega_n} \approx \frac{c_s}{2\sqrt{k_s m_s}} \left[ \frac{k_t}{k_s + k_t} \right]^{1.5}
\]

Equation 4.23

The expression in Equation 4.24 likewise provided reassurance that the influence of tyres was accounted for in the models; acknowledging that the units in this simplification do not match the units of the derived variable (Fletcher et al., 2002).

\[
\omega_{asle} \approx \frac{k_s + k_t}{m_u}
\]

Equation 4.24

Typical parameters for tyre spring rates and tyre damping coefficients have been reported (Costanzi & Cebon, 2005, 2006; de Pont, 1994; Karagania, 1997). These are shown in Table 4.3.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tyre spring rate</td>
<td>(k_t)</td>
<td>1.96 MN/m</td>
</tr>
<tr>
<td>Tyre damping coefficient</td>
<td>(c_t)</td>
<td>1.76 kNs/m</td>
</tr>
</tbody>
</table>

Table 4.3. Typical values used for tyre and HV suspension parameters.

To incorporate dynamic tyre phenomena into the models, Equation 4.22 to Equation 4.24 were used.

The derivation of \(k_s\) for known values of \(k_t, m_s\) and \(\omega_n\) was as follows:
\[ \omega_n = \frac{c_s}{2m_s} \approx \sqrt{\frac{k_s k_i}{(k_s + k_i)m_s}} \]

Equation 4.22

\[ k_s \approx \frac{k_s m_s \omega_n^2}{k_i - m_s \omega_n^2} \]

Equation 4.25

Provided the damped natural frequency, damping ratio and mass of a HV suspension were known, the damping coefficient, \( c_s \), could be derived for the model in Figure 4.5 from a re-stated Equation 4.23 (Thomson & Dahleh, 1998):

\[ \zeta = \frac{c_s}{2m_s \omega_n} \approx \frac{c_s}{2\sqrt{k_s m_s}} \left[ \frac{k_i}{k_s + k_i} \right]^{1.5} \]

Equation 4.23

\[ c_s = 2\zeta m_s \omega_n = \frac{2\zeta \sqrt{k_s m_i}}{\left[ \frac{k_i}{k_s + k_i} \right]^{1.5}} \]

Equation 4.26

From the spring rate, \( k_s \), found from Equation 4.25 and a known value of damping ratio, \( \zeta \), from either Equation 4.18 or Equation 4.21, Equation 4.26 provided both the generalised damping coefficient values and the bump and rebound damping coefficients for the models developed in the next chapter. Accordingly, contributory components from tyre influence on the variables \( k_s \) and \( c_s \) (Figure 4.5) were incorporated into the models.

### 4.4 Summary and conclusions of this chapter

Computer simulations were necessary for completion of this project and thesis. This was because performing variations in parameters on live suspensions would have been risky to personnel and possibly destructive to tested HVs. Accordingly, simulations were performed using computer models as analogues of HV suspensions. To create those models, system equations governing second-order underdamped systems were theorised. Relationships between the various components of HV suspensions were gathered from known sources. By applying a combination of those system equations and known HV suspension characteristics, a generic computer
model was developed. This chapter has documented the fundamentals upon which the development of that generic computer model was based.

4.5 Chapter close

Two models have been developed in this chapter. The first, a simple model of two axles will allow exploration of dynamic load sharing. The second, a computer simulation model, was developed in generic form. This generic computer model will allow the data gathered from HV testing (Section 3.2) to be developed into three separate analogues of the HVs axles tested. Individual models for each HV axle will allow empirical data to be input with analysis of the associated outputs to be detailed in Chapters 5 and 6.

Accordingly, these models will inform the discussion regarding:

- the efficacy of systems such as the “Haire suspension system” (Objective 2) from analysis (Chapters 10, 12 and 13);

- in-service suspension testing (Objective 3) of air-sprung HVs in Chapter 6; and

- implications of increased dynamic load sharing (Chapters 10, 12 and 13).

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“All models are flawed – some are useful.” - W. Ed Deming.
5 Heavy vehicle suspension model calibration and validation

5.1 About this chapter

This chapter documents the development of the heavy vehicle (HV) suspension computer model built up in the previous chapter. It shows the application of this suspension model to the three test vehicles in Section 3.2. System equations detailed in Chapter 4 and empirical data gathered as described in Section 3.2 were used to determine parameters for three HV axle models. Validation of the models was then undertaken by showing the results achieved when the computer model outputs were compared with empirical outputs for the same input data. These models will be used to meet Objective 2, dealing with dynamic load sharing, and Objective 3, development of low-cost in-service HV suspension testing.

5.2 Introduction

The generic computer model developed in Chapter 4 was designed to have the acceleration at the axle as the input signal and air spring pressure, being a surrogate of the axle-to-body displacement, as the output signal. This matched a portion of the data from the experimental design outlined in Section 3.2. This chapter documents the analysis of input (accelerometer) and output (air spring) data signals from the VSB 11-style step tests to determine suspension parameters for the three HVs tested in Section 3.2. The various blocks in the computer model were then populated from the relationship between those signals to develop three “quarter-HV” models. These models were then calibrated against the VSB 11-style step test output data recorded per test vehicle as described in Section 3.2. Once the computer model parameters were determined, data from the accelerometers recorded during the VSB 11-style step tests were input to the computer models. The output data from the models as analogues for air spring responses were then compared with empirical results for air spring data with those results documented in this chapter. A diagrammatic summary of the process and how this chapter relates to, and uses data from, previous chapters is illustrated in Figure 5.1.
5.2.1 Regarding data smoothing

The outputs of the air pressure transducers (APTs) during the testing were recorded for each test vehicle. A 5 Hz low-pass filter was applied to the empirical APT signals to smooth the waveforms and eliminate noise, particularly axle-hop. This allowed more accurate reading of the excursions and periodicity. Similarly, a 5 Hz low-pass filter was included in the output signal chain in the models developed, for the same reasons.
5.2.2 Regarding displayed data, left/right variation in data and choice of axes

The lengths of signal periods shown in the plots below have been chosen to best illustrate the signal characteristics. This was since the various impulse events were not always evident in the traces at the same point for each time-series. Compensation has been made for this by choosing the most appropriate window period for the traces.

As mentioned in Appendix 1, the quiescent outputs of the instruments showed slight variations due to vehicle supply voltage fluctuations. This phenomenon resulted in differing values for APT and accelerometer readings when comparing the amplitudes of the signals from the left and right sides of the test vehicles. The suspension responses shown (and used to develop specific HV axle models) below also differed between left and right sides. This was almost certainly due to natural variation in any manufacturing process and uneven side-to-side wear in suspension components, even with replacement of the dampers. Previous work (Davis, 2007; Davis & Bunker, 2008c, 2008e; Davis & Kel, 2007) compensated for these variations by:

- averaging the derived parameters for left and right data values;
- noting the steady state quiescent values of the instrumentation outputs; and
- adjusting the relevant calculations accordingly.

For the development of the models in this chapter, the variations in the steady state signal amplitudes between left and right side data were not of great concern. This was because relative amplitudes between signal excursions were used to determine ratios and not their absolute values. Further, zero-crossing periods to determine frequency from time-domain series were unaffected by instrumentation drift.

To develop specific HV models from the generic model developed in Chapter 4, variations between the data (and therefore derived parameters) from the left and the right hand sides of the HVs tested needed to be addressed. This was done by averaging the derived left and right parameters for the specific models for each HV as shown below. This resulted in three HV models representing a standardised set of
behaviours for a wheel on an axle and with a blend of left/right parameters. The alternative was to generate six separate HV wheel models representing each wheel of each axle tested. As will be seen below, the former choice yielded a valid way forward, otherwise that decision would have been reviewed.

There were differences and variations between the model outputs and the empirically derived values for damped natural frequency and damping ratio. These are noted briefly in each section below and dealt with in detail in Section 5.8.2.

5.2.3 Regarding the choice of axles for analysis and modelling

The two rear axles of the semi-trailer produced similar APT waveforms to that illustrated in Figure 5.8 for the front semi-trailer axle. Accordingly, the front axle of the semi-trailer was chosen for model development; multiple arrangements of axles in the model used in Chapter 12, Section 12.2.4 were achieved by repetition of the axle model developed here.

The maximum dynamic drive wheel forces for the coach were approximately 50 percent greater than the tag axles forces (Davis & Bunker, 2008e). This phenomenon is shown in Figure 5.2 for coach dynamic wheel forces averaged per test speed. The wheel forces created at the drive axle were measured at levels potentially more damaging than those at the tag axle were. Accordingly, the drive axle of the coach, being the more critical of the two coach rear axles in terms of network asset damage, was chosen to be modelled.

![Figure 5.2. Coach tag and drive axle wheel forces during dynamic tests – average values vs. speed.](image-url)
5.3  Calibrating the models – accelerometer data as inputs

Considering the diagram of a half-HV axle (i.e. the wheel in one corner of a HV, or the “quarter-HV”) in Figure 4.3, then the acceleration at the hub, $\ddot{y}$, may be considered to be the double-derivative, with respect to time, of the vertical displacement of the hub, $y$.

Known inputs were recorded (Section 3.2) at the axles of interest from the outputs of accelerometers mounted at the respective hubs. Examples of these data are shown from Figure 5.3 to Figure 5.5 for accelerometer time series data recorded during the VSB 11-style step tests. The two rear axles of the semi-trailer produced accelerometer waveforms very similar to that of the front semi-trailer axle.

![Bus drive axle accelerometer signal - VSB 11-style step test](image)

Figure 5.3. Time series of bus drive axle hubs’ vertical acceleration during VSB 11-style step test.
5.4 Calibrating the models – air spring data as outputs

The air spring pressures were assumed proportional to the relative displacement between the axle and the chassis and/or the force on the air springs. This was not an unreasonable assumption since excursions with amplitudes in the order of 80 mm, as experienced by the tested HVs in Section 3.2, were well within the range of air spring linear response (Davis, 2006b; Davis, 2008; Germanchev & Eady, 2008; Karl et al., 2009). The air spring pressure was considered a variable derived from the result of subtracting displacement $x$ from displacement $y$ in Figure 4.3. These data had been recorded for the VSB 11-style step tests (Section 3.2); examples are shown.
from Figure 5.6 to Figure 5.8 for APT time series signals.

Figure 5.6. Time series of bus drive axle APT output during VSB 11-style step test.

Figure 5.7. Time series of coach drive axle APT output during VSB 11-style step test.

These data were chosen as the reference cases for the three HVs tested, VSB 11 being the standard for “road friendliness” of HV suspensions (Australia Department of Transport and Regional Services, 2004a, 2004c).
Figure 5.8. Time series of front semi-trailer axle APT output during VSB 11-style step test.

It is for noting that there were some differences in the quiescent values of the APT outputs when LHS was compared with the RHS, particularly for the bus and the coach. Section 5.2.2 and Appendix 1 refer.

That the systems being measured were classically underdamped second-order responses was indicated by the APT output waveforms. A computer model of the suspension conceptualised in Figure 4.2 and Figure 4.3 for the three HVs tested was then developed in line with that shown generically in Figure 4.5.

### 5.5 Developing the bus drive axle model

#### 5.5.1 Bus suspension damping ratio

The impulse response at the bus drive axle air springs was as shown in Figure 5.6. Using the variables shown in Figure 4.4, an averaged damping ratio for the single drive axle on the bus was derived from full cycle values of the variables $A_1$ and $A_2$ (as shown generically in Figure 4.4) in Figure 5.6. This was by substituting LHS and RHS values into Equation 4.18 (Meriam & Kraige, 1993; Thomson & Dahleh, 1998) and averaging. The input values and the results for damping ratio are shown in Table 5.1.
Table 5.1. Damping ratios for left and right air springs - VSB 11-style step test on the bus drive axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test results</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Quiescent signal value</td>
<td>1814</td>
<td>1777</td>
</tr>
<tr>
<td>$A_1$</td>
<td>169</td>
<td>163</td>
</tr>
<tr>
<td>$A_2$</td>
<td>27.4</td>
<td>27.0</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.270</td>
<td>0.280</td>
</tr>
</tbody>
</table>

The damping ratios per side derived from the responses measured from the APTs on the bus drive axle ranged from 0.27 to 0.28 or an error of 3.6 percent between sides. This variation was not of concern since it was expected that individual axle damping ratio values would show a discrepancy due to natural variation in the manufacturing process, uneven left/right wear and tear on components, etc; even given the renewal of the dampers. It was dealt with by averaging the values for the different sides and applying a general value of 0.275 for damping ratio in the bus model equations. Further expansion on these differences will be covered in Section 5.8.2.

The bus manufacturer was unable to supply type-tested damping ratio values for this axle (Mack-Volvo, 2007b).

5.5.2 Bus suspension damped natural frequency

Using the inversion of damped natural period, or $T_d^{-1}$ (Equation 4.14) where $T_d$ is the damped natural period (Figure 4.4), the damped natural frequency, $f$, was obtained from the plot in Figure 5.6. The resultant values for damped natural frequency are shown in Table 5.2.

Table 5.2. Damped natural frequencies, left and right air springs - VSB 11-style step test, bus drive axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Damped natural frequency, $f$</td>
<td>1.07</td>
<td>1.05</td>
</tr>
</tbody>
</table>

The damped natural frequency derived for the bus drive axle had a difference of 1.9 percent attributable to side-to-side variation. As for damping ratio, this variation was compensated for by averaging the LHS and RHS derived values. This resulted in a
damped natural frequency for the model of 1.06 Hz or 6.66 rad.s\(^{-1}\). The bus manufacturer was unable to supply type-tested body-bounce frequency data for this axle (Mack-Volvo, 2007b).

### 5.5.3 Bus suspension model variables

A computer model from the generalised diagram in Figure 4.5 was developed. To populate the variables in Equation 4.12 as they applied to the bus, variation in derived damping ratios between the two sides was compensated for by averaging the LHS and the RHS damping ratio. The VSB 11-style step test provided an averaged damping ratio value of 0.275 (Table 5.1). The damped natural body bounce frequency, \(\omega_d\), for the model (Table 5.2) was 6.66 rad.s\(^{-1}\) (1.06 Hz) after averaging the LHS and the RHS values. The undamped natural frequency for the model was found from Equation 4.16 using a damping ratio, \(\zeta\), of 0.275, yielding an undamped natural frequency, \(\omega_d\), of 6.93 rad.s\(^{-1}\) or 1.10 Hz. A sprung mass for the system model, \(m_s\), of 4.47 t was derived from a measured wheel mass of 5 t (Davis & Bunker, 2009e) less half the total unsprung mass of the bus axle being 530 kg (Prem, 2008).

Table 5.3 lists the totalised variables for the model after this process, those listed in Table 4.3 and those derived from Equation 4.25 and Equation 4.26.
Manufacturer’s data were provided for a static spring rate, \( k_s \), range varying between 47.6 and 286 kN/m (Mack-Volvo, 2007a); the derived spring rate, \( k_s \), in Table 5.3 was within this range. The lower value of 47.6 kN/m was for tests at very small excursions; a low incremental spring rate to provide a soft ride over small perturbations.

As a check for the axle-hop frequency value derived here, fast Fourier transforms (FFTs) of the accelerometer signal from the bus axle (in Figure 5.3) showed axle-hop frequencies between 8.5 Hz and 10.8 Hz (Davis & Bunker, 2008e); 10.2 Hz was within this range.

The bump and rebound damping ratios were determined from the excursions in the positive and negative directions of the signals from the VSB 11-style step tests (Figure 5.6). Figure 4.4 illustrates the starting points and conventions for derivation of differing damping ratios, depending on the relative direction of movement between the axle and the body. From Figure 4.4 and using Equation 4.21:

- the convention for the signal excursion from R to B was taken as the case of rebound damping where the axle was moving away from the chassis; and

- the signal excursion from B to Q was for the case of bump damping where the axle was moving toward the chassis.

The damping ratios were determined for the cases of:

- bump, where the body and axle move toward each other. This resulted in a positive sense for \( \hat{y} - \hat{x} \) which, in turn, required the model to recognise only positive values of \( \hat{y} - \hat{x} \) (i.e. a lower limit of zero for \( \hat{y} - \hat{x} \)). This limiting condition was applied to the feedback loop controlling the bump damping coefficient; and

- rebound, where the body and axle move away from each other. This resulted in negative values for \( \hat{y} - \hat{x} \) which, in turn, required the model to consider only the negative values of \( \hat{y} - \hat{x} \) (i.e. an upper limit of zero for \( \hat{y} - \hat{x} \)). This limiting condition was applied as the
rebound damping coefficient feedback loop.

Accordingly, the values for $A_1$ and $A_{1.5}$ (Figure 4.4) for the bus were used to derive the bump damping ratio, $\zeta_{\text{bump}}$, using those excursions in Figure 5.6 and Equation 4.21. $A_{1.5}$ and $A_2$ (Figure 4.4) were used to derive the rebound damping ratio, $\zeta_{\text{rebound}}$, using those excursions in Figure 5.6 and Equation 4.21. The two direction-specific damping ratios are shown in Table 5.4.

Table 5.4. Determining the bump and rebound damping ratios for the bus from the VSB 11-style step test.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Quiescent signal value</td>
<td>1814</td>
<td>1777</td>
</tr>
<tr>
<td>$A_1$</td>
<td>169</td>
<td>163</td>
</tr>
<tr>
<td>$A_{1.5}$</td>
<td>33.4</td>
<td>38.0</td>
</tr>
<tr>
<td>$A_2$</td>
<td>27.4</td>
<td>27.0</td>
</tr>
<tr>
<td>Bump damping ratio, $\zeta_{\text{bump}}$</td>
<td>0.060</td>
<td>0.110</td>
</tr>
<tr>
<td>Rebound damping ratio, $\zeta_{\text{rebound}}$</td>
<td>0.460</td>
<td>0.420</td>
</tr>
</tbody>
</table>

Having determined the spring rate, $k_s$, and knowing the tyre spring rate, $k_t$, and the sprung mass, $m_s$ (Table 5.3), the bump and the rebound damping coefficients, $c_{\text{bump}}$, and $c_{\text{rebound}}$, respectively, were found by substituting the left/right average of the derived bump and rebound damping ratio values, $\zeta_{\text{bump}}$ and $\zeta_{\text{rebound}}$, in Table 5.4, into Equation 4.26:

$$c_{\text{bump}} = 2\zeta_{\text{bump}}\omega_n m_s = 2\zeta_{\text{bump}} \sqrt{k_s m_s} \left[ \frac{k_t}{k_s + k_t} \right]^{1.5}$$

Equation 4.26

$\Rightarrow \quad c_{\text{bump}} = 6.69 \text{ kN} \cdot \text{m}$; and

$$c_{\text{rebound}} = 2\zeta_{\text{rebound}}\omega_n m_s \approx 2\zeta_{\text{rebound}} \sqrt{k_s m_s} \left[ \frac{k_t}{k_s + k_t} \right]^{1.5}$$

Equation 4.26

$\Rightarrow \quad c_{\text{rebound}} = 34.2 \text{ kN} \cdot \text{m}$. 
5.5.4 **Bus drive axle software model**

From the derivation of the necessary variables above, the block values from the generic model shown in Figure 4.5 were populated to create a Simulink Matlab® model of the bus quarter-HV for its drive axle as shown in Figure 5.9. Noise from axle-hop and other sources made derivation of data from plots difficult. Accordingly, to render excursions and other data more easily obtained from the output plots, a 5 Hz Butterworth filter was added to the output signal chain before the final output “Scope” element; top right corner, Figure 5.9 as discussed in Section 5.2.1. The constant for the gain block before this filter was determined from the relationship between the APT outputs and the air spring excursions.

The constant for the gain block after the input (Figure 5.9, bottom left) was the telemetry system’s input sensitivity determined from the mathematical combination of accelerometer sensitivity and the telemetry system’s count range and then dividing by the acceleration due to gravity in m/s². The accelerometer signal as an input was adjusted for gravity offset; the signal on the accelerometer had a constant equivalent to gravity subtracted from it before running any simulations. This eliminated the gravity component from the accelerometer empirical data input. Accordingly, the input signal represented only net acceleration values fluctuating around zero. Otherwise, the constant offset gravity component input to the integrator would have resulted in a ramp time-series signal output, rendering any analysis invalid.

![Figure 5.9. Matlab® block diagram showing individual blocks for bus half-axle suspension simulation.](image)

---

**239.63**

**spring k**

**34.2**

**bump damping coefficient**

**6.69**

**bump damping coefficient**

---
Manufacturer’s data did not vary per side (Mack-Volvo, 2007a) and did not always match the characteristics derived. This was particularly noticeable for the generalised damping coefficient, $c_s$, which featured in the generic model (Figure 4.5) and Table 5.3. This parameter was provided as an average value of 12.3 kNs/m for the H96 setting on this axle (Mack-Volvo, 2007a). Empirical extremes of the bump damping coefficient, $c_{bump}$, at 3.20 kNs/m and the rebound damping coefficient, $c_{rebound}$, of 30.3 kNs/m on this axle were also provided by the manufacturer (Mack-Volvo, 2007a). The range of these manufacturer’s values was similar but the values differed from those derived empirically. Even so, the derived values of damper coefficients used were justified on the grounds of derivation from empirical data. The manufacturer’s data were for type tests; those may be expected to vary from manufactured unit metrics as will be expanded in Section 5.8.2.

5.5.5 Validation of the bus suspension model

The bus axle model shown in Figure 5.9 had a representative sample (Figure 5.3) of data recorded from the accelerometers during the VSB 11-style step test applied to it as an input. A resultant simulation time-series output (from the “Scope” block in Figure 5.9) is shown in Figure 5.10. As discussed in Section 5.2.2, the model used averaged left/right parameters (Table 5.2 and Table 5.4) to compensate for the differences in empirical data. Since compensation for the differences between the empirical data from each side had been performed by averaging, the output from the model was that for a combined average of the LHS and RHS responses. The plots within Figure 5.10 have been aligned for better comparison.

Zeroing the input mean (Section 5.5.4), resulted in some non-alignment of the zeros on the y-axes in the output data. The absolute values of the excursion maxima and minima from these data compared with those from the empirical data were not of great concern. This was since the damping ratios for the model were derived from the ratios of relative dynamic excursions in the y-axes data, not the y-axes offsets or absolute excursions. Similarly, the damped natural frequency was derived from the period between zero-crossings or peak excursions, not the absolute values of those excursions.
The values for damping ratio, $\zeta$, and damped natural frequency, $f_d$, were then derived from the simulated output response to the empirical step test data as an input, noting the 5 Hz filtering in the output signal chain of the model. These are shown in Table 5.5 and Table 5.6 respectively.
Table 5.5. Comparison between simulation model damping ratio and result from empirical data - bus.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides, Table 5.1)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value $A_i$</td>
<td></td>
<td>6.70</td>
</tr>
<tr>
<td></td>
<td></td>
<td>163</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.275</td>
<td>0.265</td>
</tr>
<tr>
<td>Error compared with average of actual VSB</td>
<td></td>
<td>-3.60%</td>
</tr>
<tr>
<td>11-style step test damping ratio, $\zeta$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5.6. Comparison between simulation model damped natural frequency and result from empirical data - bus.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides, Table 5.2)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency, $f$ (Hz)</td>
<td>1.06</td>
<td>1.057</td>
</tr>
<tr>
<td>Error compared with average of actual VSB</td>
<td></td>
<td>-0.280%</td>
</tr>
<tr>
<td>11-style step test damped natural frequency, $f$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

There was a difference of -0.280 percent and -3.60 percent between the model output and the empirically derived values for damped natural frequency and damping ratio respectively. In brief, the error was small and therefore a good result was obtained from the simulation, especially since the bump damping ratio, $\zeta_{bump}$, derived per side (Equation 4.21, shown in Table 5.4) varied significantly between sides. Further expansion on the reasons for these differences is dealt with in Section 5.8.2.
5.6 Developing the coach drive axle model

5.6.1 Coach suspension damping ratio

The impulse response at the coach drive axle air springs was as shown in Figure 5.7. In a similar manner to the bus, the coach drive axle was analysed for damping ratio, $\zeta$. By applying Equation 4.18 to the $A_1$ and $A_2$ (Figure 4.4) values of the coach drive axle response to the step-test, damping ratio values for each side were derived from signal excursions. These results are shown in Table 5.7.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Quiescent signal value</td>
<td>1954</td>
<td>1805</td>
</tr>
<tr>
<td>$A_1$</td>
<td>139</td>
<td>170</td>
</tr>
<tr>
<td>$A_2$</td>
<td>11.4</td>
<td>16.8</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.370</td>
<td>0.350</td>
</tr>
</tbody>
</table>

The damping ratio results derived from the APT responses at the coach drive axle averaged 0.360 +/- 0.010 or an error of 5.50 percent between sides. These differences will be discussed in more detail in Section 5.8.2. The variation was not of particular concern since it was expected that individual axle damping ratio values would show a discrepancy due to natural variation in the manufacturing process and uneven left/right wear and tear on components. It was dealt with by averaging the values for the different sides and applying a general averaged value of 0.360 for damping ratio.

Similar to the absence of certified damping ratio data for the bus, the manufacturer was unable to supply type-tested damping ratio values for this axle (Mack-Volvo, 2007b).
5.6.2 Coach drive axle damped natural frequency

The damped natural frequency, \( f \), of the coach drive axle was found from inverting the period, \( T_d \), between successive peaks in Figure 5.7 (Equation 4.14). The resultant values are shown in Table 5.8.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency, ( f ) (Hz)</td>
<td>1.14</td>
<td>1.10</td>
</tr>
</tbody>
</table>

Note that the damped natural frequency derived for the coach drive axle had a difference of 3.5 percent attributable to side-to-side variation. This variation was compensated for by averaging the LHS and RHS derived values as discussed above. This resulted in a damped natural frequency for the model of 1.12 Hz or 7.037 radians.s\(^{-1}\). The coach manufacturer was unable to supply type-tested damped natural frequency data for this axle (Mack-Volvo, 2007b).

5.6.3 Coach suspension model variables

A computer model from the generalised diagram in Figure 4.5 was developed. To find the model remaining variables (Equation 4.12) as they applied to the coach, a general averaged value of 0.36 for damping ratio (Table 5.7) was used. Similarly, left/right variation was averaged to provide the model with a damped natural body bounce frequency of 1.12 Hz from Table 5.8. From these values and the application of Equation 4.16 a model undamped natural frequency, \( \omega_n \), of 7.54 rad.s\(^{-1}\) or 1.20 Hz was derived. A system sprung mass, \( m_s \), of 3.79 t was determined from a measured wheel mass of 4.3 t (Davis & Bunker, 2009e) less half the total unsprung mass of the coach axle being 510 kg from measured data (Table A1.1) and Prem (2008). That data indicated that the coach axle was lighter than the bus axle since it was equipped with alloy wheels (Table A1.1).
The remaining variables from Equation 4.25 and Equation 4.26 were found using known values derived above. These are shown in Table 5.9.

Table 5.9. Given and derived tyre and HV suspension parameters - coach.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped natural body bounce frequency</td>
<td>( \omega_n )</td>
<td>7.54</td>
<td>rad.s(^{-1})</td>
</tr>
<tr>
<td>Sprung mass</td>
<td>( m_s )</td>
<td>3.79</td>
<td>t</td>
</tr>
<tr>
<td>Unsprung mass</td>
<td>( m_u )</td>
<td>0.510</td>
<td>t</td>
</tr>
<tr>
<td>Suspension spring rate</td>
<td>( k_s )</td>
<td>242</td>
<td>kN/m</td>
</tr>
<tr>
<td>Suspension damping coefficient</td>
<td>( c_s )</td>
<td>24.5</td>
<td>kNs/m</td>
</tr>
<tr>
<td>Tyre spring rate</td>
<td>( k_t )</td>
<td>1.96</td>
<td>MN/m</td>
</tr>
<tr>
<td>Axle-hop frequency</td>
<td>( \omega_{ axle} )</td>
<td>10.6</td>
<td>Hz</td>
</tr>
</tbody>
</table>

Manufacturer’s data were provided for a static spring rate, \( k_s \), value ranging from 146 to 242 kN/m (Mack-Volvo, 2007a). Dynamic spring rates may vary by a multiple of up to 1.4 of static spring rates (Costanzi & Cebon, 2005; Duym et al., 1997; Prem et al., 1998). This is because static spring rate measurement does not always account for adiabatic conditions occurring during short, transient excursions of the air spring (Costanzi & Cebon, 2005; Duym et al., 1997; Prem et al., 1998). Certainly short, transient excursions were an accurate description of the VSB 11-style step tests performed as described in Section 3.2. Accordingly, the derived dynamic spring rate value, \( k_s \), fell at the upper limit of the manufacturer’s range. Nonetheless, it could have been up to 1.4 times greater and still have been valid since this parameter was derived dynamically.

Checking for axle-hop frequency validity, other work for this thesis (Davis & Bunker, 2008e) showed axle-hop frequencies between 8.5 Hz and approximately 12 Hz for the coach drive axle; the derived 10.6 Hz was well within this range.

The damping ratios for bump and rebound cases were determined from the signal excursions in the positive and negative directions from an indicative and representative sample of the air spring signals during a VSB 11-style step test, an example of which is shown in Figure 5.7.

As for the bus, the air spring excursions in Figure 5.7 for values R to B (\( A_{1} \) and \( A_{1.5} \) in Figure 4.4) and B to Q (\( A_{1.5} \) and \( A_{2} \) in Figure 4.4) were used to derive the bump
and rebound damping ratios, $\zeta_{\text{bump}}$ and $\zeta_{\text{rebound}}$ respectively, using Equation 4.21. They are shown in Table 5.10. The same processing and offset compensation used for the bus model was made for the acceleration and air spring quiescent state signals (Section 5.2.2).

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value</td>
<td>1954</td>
<td>1805</td>
</tr>
<tr>
<td>$A_1$</td>
<td>139</td>
<td>170</td>
</tr>
<tr>
<td>$A_{1.5}$</td>
<td>38.4</td>
<td>44.8</td>
</tr>
<tr>
<td>$A_2$</td>
<td>11.4</td>
<td>16.8</td>
</tr>
<tr>
<td>Bump damping ratio, $\zeta_{\text{bump}}$</td>
<td>0.360</td>
<td>0.300</td>
</tr>
<tr>
<td>Rebound damping ratio, $\zeta_{\text{rebound}}$</td>
<td>0.380</td>
<td>0.390</td>
</tr>
</tbody>
</table>

Having determined the spring rate, $k_s$, and knowing the tyre spring rate, $k_t$, and the sprung mass, $m_s$ (Table 5.9), the bump and the rebound damping coefficients, $c_{\text{bump}}$ and $c_{\text{rebound}}$ respectively, were found. This was done by substituting the left/right average of the derived bump and rebound damping ratio values, $\zeta_{\text{bump}}$ and $\zeta_{\text{rebound}}$, shown in Table 5.10, into Equation 4.26:

$$c_{\text{bump}} = 2\zeta_{\text{bump}}\omega_n m_s = 2\zeta_{\text{bump}}\sqrt{k_s m_s} \left[ \frac{k_t}{k_s + k_t} \right]^{1.5}$$

Equation 4.26

$$c_{\text{rebound}} = 2\zeta_{\text{rebound}}\omega_n m_s = 2\zeta_{\text{rebound}}\sqrt{k_s m_s} \left[ \frac{k_t}{k_s + k_t} \right]^{1.5}$$

Equation 4.26

As for the bus, data for the bump and rebound damping coefficients were provided by the manufacturer (Mack-Volvo, 2007a). For the coach drive axle at the H96 setting, this ranged between manufacturer’s extremes of 1.90 kNs/m in bump to 25.0
kNs/m in rebound with an average of 11.8 kNs/m. The derived values of the model bump and rebound damping coefficients, \( c_{\text{bump}} \) and \( c_{\text{rebound}} \), differed from manufacturer’s type test data that, by its nature, varied from manufactured unit values. Accordingly, the parameters derived (such as the damped natural frequency and the bump and rebound damping coefficients) were justified on the basis that they were derived from empirically derived data from the APT output signals. This allowed the computer model of the drive axle of the coach to be developed as shown in Figure 5.11. A 5 Hz filter was added to the final output signal processing chain for purposes of smoothing similar to that noted in Section 5.2.1.

![Matlab block diagram showing individual blocks for coach half-axle suspension simulation.](image)

Similar to the bus, the coach computer model gain constants after the input signal (Signal 2) and before the output were determined from the relationship between the accelerometer signals and the resultant APT outputs.

### 5.6.4 Validation of the coach suspension model

The model shown in Figure 5.11 had an input applied from a representative sample (Figure 5.3) of data recorded from the accelerometers during the VSB 11-style step test. The resultant time-series output (from the “Scope” block in Figure 5.11) is
shown in Figure 5.12. The gravity steady state offset on the accelerometer input signal was eliminated by an equal and opposite signal before processing. As seen previously for the bus, this resulted in some mismatching of the zeros on the y-axes in the graphs following. The resulting disparity was not important and did not affect the results since damping ratio was derived from relative dynamic excursions in the y-axes data, not the offsets or absolute excursions. The output provided a combined average model of the left and right responses since the differences between the sides had been averaged. Note that the axes in the plots within Figure 5.10 have been adjusted for better comparison.
Chapter 5

model suspension response to empirical step test data input

-150
-100
-50
0
50
100
150

0
50
100
150
200

2.0 2.2 2.4 2.6 2.8 3.0 3.2 3.4 3.6 3.8 4.0 4.2 4.4 4.6 4.8 5.0 5.2 5.4 5.6 5.8 6.0

time (s)
magnitude of simulated response (arbitrary linear scale)

Coach drive axle APT signal - VSB 11-style step test

APT output (arbitrary linear scale)

LEFT

RIGHT

Time (s)

Figure 5.12. (above) time series of Matlab® Simulink coach half-axle model output for a vertical acceleration input during VSB 11-style step test. Figure 5.7 (repeated for information) below.

The values for damping ratio, $\zeta$, and damped natural frequency, $f$, were then derived from the simulated output response to the empirical step test data as an input, noting the 5 Hz filtering in the output signal chain of the model. These are shown in Table 5.11 and Table 5.12 respectively.
Table 5.11. Comparison between simulation model damping ratio and result from empirical data - coach.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides Table 5.7)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value</td>
<td>-4.40</td>
<td>-4.40</td>
</tr>
<tr>
<td>$A_1$</td>
<td>143</td>
<td>143</td>
</tr>
<tr>
<td>$A_2$</td>
<td>10.7</td>
<td>10.7</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.360</td>
<td>0.340</td>
</tr>
<tr>
<td>Error compared with average of actual VSB 11-style step test damping ratio, $\zeta$</td>
<td>-</td>
<td>-5.50%</td>
</tr>
</tbody>
</table>

There was a difference of 0.890 percent and -5.50 percent between the model output and the empirically derived values for damped natural frequency and damping ratio respectively. This was considered a good result since the errors were small. Further expansion on the reasons for these differences is dealt with in Section 5.8.2.

Table 5.12. Comparison between simulation model damped natural frequency and result from empirical data - coach.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides, Table 5.8)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency (Hz)</td>
<td>1.12</td>
<td>1.13</td>
</tr>
<tr>
<td>Error compared with average of actual VSB 11-style step test damped natural frequency</td>
<td>-</td>
<td>0.890%</td>
</tr>
</tbody>
</table>

5.7 Developing the semi-trailer axle model

5.7.1 Semi-trailer suspension damping ratio

The impulse response at the suspension of one of the axles of the semi-trailer was as shown in Figure 5.8. In a similar manner to the bus and the coach, this response was analysed for damping ratio, $\zeta$. Equation 4.18 provided the theory to derive damping ratio from the relative values of $A_1$ and $A_2$ (Figure 4.4) in Figure 5.8. These results
are shown in Table 5.13.

Table 5.13. Damping ratios for left and right air springs - VSB 11-style step test on the semi-trailer axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Quiescent signal value</td>
<td>1479</td>
<td>1500</td>
</tr>
<tr>
<td>$A_1$</td>
<td>94.0</td>
<td>98.0</td>
</tr>
<tr>
<td>$A_2$</td>
<td>17.0</td>
<td>21.0</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.240</td>
<td>0.260</td>
</tr>
</tbody>
</table>

Comparing the APT damping ratios per side, it may be noted that the variation in damping ratio results between sides of the semi-trailer was 0.02 (or a left/right variation of approximately 8 percent) for an averaged value of 0.25. From a “pull-up-and-drop” method (Australia Department of Transport and Regional Services, 2004a, 2004c), the manufacturer quoted VSB 11 type-tested damping ratio values for these axles (Colrain, 2007) of 0.2501. The difference between the derived value and the manufacturer’s value will be addressed briefly in Section 5.7.3 in preparation for the detail in Section 5.8.2 which will also address the variation in damping ratios per side.

5.7.2 Semi-trailer axle damped natural frequency

The damped natural frequency, $f$, of the semi-trailer axle was found from the inverse of the time between successive peaks in Figure 5.8 by inverting the damped natural period ($T_d^{-1}$, Equation 4.14), where $T_d$ is the damped natural period. The resultant values for damped natural frequency are shown in Table 5.14.

Table 5.14. Damped natural frequencies for left and right air springs - VSB 11-style step test on the front axle of the semi-trailer.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
<td>RHS</td>
</tr>
<tr>
<td>Damped natural frequency $f$</td>
<td>1.68</td>
<td>1.72</td>
</tr>
</tbody>
</table>
Comparing the two sides in Table 5.14, the damped natural frequency derived for this axle differed by a maximum of 2.3 percent. To address this difference briefly, the manufacturer quoted the damped natural frequency for the semi-trailer axle at 1.89 Hz (Colrain, 2007) from a “pull-up-and-drop” VSB 11 type test method with masses as used during the testing for this thesis. These data are type-test values, as are all VSB 11 parameters (Davis & Bunker, 2007). Some potential reasons for differences between these and the tested values were evident. These will be dealt with briefly in Section 5.7.3 and in detail in Section 5.8.2.

5.7.3 Empirical data and metrics derived thereby vs. VSB 11 type test data

The VSB 11-style test used for the testing in Section 3.2.3 was a step down, without first pulling up the HV. Individual axle metrics will differ with natural variation in the manufacturing process as well as other factors mentioned previously such as mechanical wear and tear. VSB 11 is a type test and type tests do not always reflect production values. Further, differing HV test methods will produce different results (Uffelmann & Walter, 1994). That work, for instance, noted differences of up to 60 percent in damping ratio results depending on whether the test was a “lift and drop”, traverse over a bump or a step down. That dampers have differential rates depending on direction is a large contributory factor to this phenomenon. Wear and tear in the HV tested may also have been a factor, even with the precaution of installing new shock absorbers. More discussion on this issue is contained in Section 5.8.2.

5.7.4 Semi-trailer suspension model variables

A semi-trailer half-axle model was developed from the generalised diagram in Figure 4.5. To find the model remaining variables (Equation 4.12) as they applied to the axle tested, a general averaged damping ratio value of 0.25 was used from Table 5.13, compensating for the differences in left and right values of damping ratio on both sides of the semi-trailer axle as discussed briefly above. Similarly, by averaging the left and right side values for the damped natural frequency for the model, 1.70 Hz or 10.68 rad.s\(^{-1}\) (Table 5.14). The undamped natural frequency, \(\omega_n\), was then found
from Equation 4.16 to yield a value for this model parameter of 11.0 rad.s\(^{-1}\) or 1.75 Hz. The system sprung mass, \(m_s\), value of 2.92 t was derived from a measured wheel mass of 3.26 t less the unsprung mass of the semi-trailer axle being 336 kg (Davis & Bunker, 2009e; Giacomini, 2007).

Using known variables listed above and from Equation 4.25 and Equation 4.26, the remaining system parameters were derived, as summarised in Table 5.15.

Table 5.15. Given and derived tyre and HV suspension parameters – semi-trailer.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamped natural body bounce frequency</td>
<td>(\omega_n)</td>
<td>7.54</td>
<td>rad.s(^{-1})</td>
</tr>
<tr>
<td>Sprung mass</td>
<td>(m_s)</td>
<td>2.90</td>
<td>t</td>
</tr>
<tr>
<td>Unsprung mass</td>
<td>(m_u)</td>
<td>0.336</td>
<td>t</td>
</tr>
<tr>
<td>Suspension spring rate</td>
<td>(k_s)</td>
<td>427</td>
<td>kN/m</td>
</tr>
<tr>
<td>Suspension damping coefficient</td>
<td>(c_s)</td>
<td>22.5</td>
<td>kNs/m</td>
</tr>
<tr>
<td>Tyre spring rate</td>
<td>(k_t)</td>
<td>1.96</td>
<td>MN/m</td>
</tr>
<tr>
<td>Axle-hop frequency</td>
<td>(\omega_{axle})</td>
<td>12.4</td>
<td>Hz</td>
</tr>
</tbody>
</table>

Empirical axle-hop frequency data (Davis & Bunker, 2008e) indicated that the range for this parameter was from 10 to 12 Hz; the derived \(\omega_{axle}\) of 12.4 Hz was slightly outside this range by an acceptable margin of derivational error.

The damping ratios for bump and rebound cases were determined from the signal excursions in the positive and negative directions of the VSB 11-style step tests. Similar to the cases for the other two tested vehicles, Equation 4.21 and the signal excursions for values R to B (\(A_1\) and \(A_{1.5}\) in Figure 4.4) and B to Q (\(A_1.5\) and \(A_2\) in Figure 4.4) were used to derive bump and rebound damping ratios, \(\zeta_{\text{bump}}\) and \(\zeta_{\text{rebound}}\), respectively. These are shown in Table 5.16 as derived after the same processing and offset compensation for the other two models previously.
Table 5.16. Determining the bump and rebound damping ratios for the semi-trailer front axle from the VSB 11-style step test.

<table>
<thead>
<tr>
<th>Variable</th>
<th>LHS VSB 11-style step test</th>
<th>RHS VSB 11-style step test</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value</td>
<td>1479</td>
<td>1500</td>
<td></td>
</tr>
<tr>
<td>$A_1$</td>
<td>94.0</td>
<td>98.0</td>
<td></td>
</tr>
<tr>
<td>$A_{1.5}$</td>
<td>32.0</td>
<td>38.0</td>
<td></td>
</tr>
<tr>
<td>$A_2$</td>
<td>17.0</td>
<td>21.0</td>
<td></td>
</tr>
<tr>
<td>Bump damping ratio, $\zeta_{bump}$</td>
<td>0.200</td>
<td>0.190</td>
<td>0.195</td>
</tr>
<tr>
<td>Rebound damping ratio, $\zeta_{rebound}$</td>
<td>0.320</td>
<td>0.290</td>
<td>0.305</td>
</tr>
</tbody>
</table>

No manufacturer’s value for the general damping coefficient was available (Colrain, 2007).

Having determined the spring rate, $k_s$, and knowing the tyre spring rate, $k_t$, and the sprung mass, $m_s$ (Table 5.15), the bump and the rebound damping coefficients, $c_{bump}$ and $c_{rebound}$ respectively, were found. This was done by substituting the left/right average of the derived bump and rebound damping ratio values, $\zeta_{bump}$ and $\zeta_{rebound}$, in Table 5.16, into Equation 4.26:

$$c_{bump} = 2\zeta_{bump}\omega_n m_s \approx 2\zeta_{bump} \sqrt{k_s m_s \left[ \frac{k_s}{k_s+k_t} \right]^{1.5}}$$

**Equation 4.26**

$$\Rightarrow c_{bump} = 18.45 \text{ kNs/m}; \text{ and}$$

$$c_{rebound} = 2\zeta_{rebound}\omega_n m_s \approx 2\zeta_{rebound} \sqrt{k_s m_s \left[ \frac{k_s}{k_s+k_t} \right]^{1.5}}$$

**Equation 4.26**

$$\Rightarrow c_{rebound} = 28.86 \text{ kNs/m}.$$  

Having derived the required variables to populate Equation 4.12, Figure 5.13 was developed for the semi-trailer half-axle computer model.
Similar to the other two half-axle models, the constants for the gain after the input and before the output were determined from the relationship between the accelerometer signal values and the resultant APT output values with appropriate elimination of the steady state signal due to gravity.

Further parametric investigation was then undertaken to derive simulation outputs for derived damped natural frequency and damping ratio values from the Simulink Matlab® model for the semi-trailer axle using empirical data from the accelerometers during the VSB 11-style step test.

5.7.5 Validating the semi-trailer suspension model

As for the other two test vehicles, the output from the Simulink Matlab® model for the semi-trailer front half-axle suspension was analysed for an empirical data input (Figure 5.5) from the accelerometers during a VSB 11-style step test. This is shown in Figure 5.14. Comparing the output from the model with Figure 5.8, it may be seen by inspection that the period and excursions were very similar; a visual check that the model provided good correlation with the empirical data.
The model's values for damping ratio and damped natural frequency were then derived for this input. These are shown in Table 5.17 and Table 5.18 respectively, after 5 Hz filtering.
Table 5.17. Comparison between simulation model damping ratio and result from empirical data – semi-trailer.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides, Table 5.7)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value $A_1$</td>
<td></td>
<td>2.50</td>
</tr>
<tr>
<td>$A_2$</td>
<td></td>
<td>104</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.250</td>
<td>0.240</td>
</tr>
<tr>
<td>Error compared with</td>
<td></td>
<td>-4.00%</td>
</tr>
<tr>
<td>average of actual VSB 11-style step test damping ratio</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

There was a difference of -4.00 percent between the model output and the empirically derived values for damping ratio. Expansion on possible reasons for this result is contained in Section 5.8.2.

Table 5.18. Comparison between simulation model damped natural frequency and result from empirical data – semi-trailer.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test (average, both sides, Table 5.8)</th>
<th>Simulink model with empirical input from VSB 11-style step test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency (Hz)</td>
<td>1.70</td>
<td>1.65</td>
</tr>
<tr>
<td>Error compared with average of actual VSB 11-style step test damped natural frequency, $f$</td>
<td>-</td>
<td>-2.90%</td>
</tr>
</tbody>
</table>

There was a difference of -2.90 percent between the model output and the empirically derived values for damped natural frequency. The variation between sides of these parameters for the VSB 11-style step tests (Table 5.13 and Table 5.14) was 8.3 percent and 2.3 percent respectively.

Comparing the left/right variation of suspension parameters derived empirically using the “gold standard” test procedure, the errors between the model output and the actual data were of the same order-of-magnitude. This was considered a satisfactory result with respect to generally-accepted experimental error. Further expansion on the reasons for any differences is dealt with in Section 5.8.2.
5.8 **Summary of this chapter**

5.8.1 **Error analysis – totalised summary**

Table 5.19 to Table 5.21 provide a summary of the errors for:

- side-to-side empirical output data from the VSB 11-style step tests; and
- the model simulation outputs for empirical data inputs to the particular axle.

**Table 5.19. Summary of errors – bus drive axle.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Totalised errors across all testing and simulations – bus drive axle Method</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>VSB 11 (left/right variation)</td>
<td>VSB 11 result vs. model</td>
</tr>
<tr>
<td>Damping ratio</td>
<td>3.60%</td>
<td>-3.60%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>1.90%</td>
<td>-0.280%</td>
</tr>
</tbody>
</table>

**Table 5.20. Summary of errors – coach drive axle.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Totalised errors across all testing and simulations – coach drive axle Method</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>VSB 11 (left/right variation)</td>
<td>VSB 11 result vs. model</td>
</tr>
<tr>
<td>Damping ratio</td>
<td>5.50%</td>
<td>-5.50%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>3.50%</td>
<td>0.890%</td>
</tr>
</tbody>
</table>

**Table 5.21. Summary of errors – semi-trailer axle.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Totalised errors across all testing and simulations – semi-trailer axle Method</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>VSB 11 (left/right variation)</td>
<td>VSB 11 result vs. model</td>
</tr>
<tr>
<td>Damping ratio</td>
<td>8.30%</td>
<td>-4.00%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>2.30%</td>
<td>-2.90%</td>
</tr>
</tbody>
</table>
5.8.2 Regarding the left/right differences from empirical data, VSB 11 data and also the model outputs

The errors shown in Table 5.19 to Table 5.21 may be considered in light of the following discussion. Manufacturer’s data for the semi-trailer were supplied for damped natural frequency and damping ratio. These data were for VSB 11 type-tested values (Davis & Bunker, 2007). Differences between these and the tested values from this project were evident. As with all manufacturing process, there will be natural variation in any system that will cause data from one individual unit to vary from the type-tested metrics. This will also be the case for differences between type tested damping coefficients and those derived empirically herein for the bus and the coach.

Even though the dampers were renewed for all the HVs tested, other suspension components would undoubtedly have undergone mechanical wear and tear compared with their new state. Notably, where supplied, the coach and bus manufacturer’s parametric data did not differ per side. Accordingly, any variations in the empirically derived data per side were due to uneven wear and tear, manufacturing tolerances and measurement error. This last should have been evenly divided between sides and was minimal, as shown in other work (Davis, 2006b; Karl et al., 2009b) and in Appendix 2. Potholes predominate on the left hand side of the road in Australia and instantaneous axle forces as derived later in this thesis were higher on the left compared with the right hand side (Davis & Bunker, 2009a). It is not surprising to expect that such wear and tear would have been greater on the left therefore, causing an imbalance in both suspension component wear and also more frequent component replacement on that side. Further, differences of up to 60 percent in damping ratio values have been reported (Uffelmann & Walter, 1994) depending on direction of excitation. As noted in Table 5.4, Table 5.10 and Table 5.16 as well as from the manufacturer (Mack-Volvo, 2007a), dampers have differential rates depending on direction. The directionality of damper response is an important contributory factor to the phenomenon of different results, dependent on different test methods. That is, some test methods drop the HV axle, exercising one direction before the other. Other methods lift and drop, reversing the order of the halves on the excitation impulse (Uffelmann & Walter, 1994).
The models developed were not particularly complex compared with some others such as those of Costanzi and Cebon (2005, 2006). There were, however, of similar complexity to other work (Cole & Cebon, 2007; Duym et al., 1997; Prem et al., 1998). The model parameters were composites derived from averaged empirical left and right hand side data from the vehicles tested. This resulted in the model outputs being composites of the left and right averages of all the model parameters and their inputs. Nonetheless, the computer models produced damped natural frequency and damping ratio values very close to the empirical results. Further, should the differences in parameters between sides derived by the use of the VSB 11-style step test inform likely error margins, the errors between the model outputs compared with the empirical data were of the same order of magnitude.

VSB 11 (Australia Department of Transport and Regional Services, 2004c) is the Australian “gold standard” for parametric measurement of “road friendly” HV suspensions. There was only one set of available VSB 11-derived suspension data for the HVs tested as described in Section 3.2; that for the semi-trailer axles. Empirical data from the VSB 11-style step test in Section 3.2 were used as an input to the semi-trailer model. The difference between the damping ratio from that model’s output, the VSB 11-certified damping ratio and the VSB 11-style step test damping ratio result was 0.01 (Table 5.17). This was an error of -4.0 percent. As noted in Section 5.7.3, variations of up to 60 percent have been reported between results from different types of impulse testing (Prem et al., 1998; Uffelmann & Walter, 1994). Similarly, the difference between the VSB 11-certified damped natural frequency and the semi-trailer model damped natural frequency was -2.9%.

All the model output errors were within, or less than, the same order-of-magnitude of left/right variation apparent from the empirical data recorded during the step test defined in VSB 11. This result, combined with the potential for variation in data due to different excitation methods, was considered a satisfactory outcome with respect to generally-accepted experimental error.
5.9 Conclusions from this chapter

5.9.1 General

The maximum time of impulse duration recommended by Doebelin (1980) for characterising a system is \(0.35/f\) where \(f\) is the damped natural frequency (Equation 4.14). Therefore, to characterise the bus and coach suspension systems, the impulse duration would, taking their damped natural frequencies from Table 5.2 and Table 5.8, ideally have been:

\[
0.35 \times \frac{1}{f} = 0.31 \text{ to } 0.33 \text{ s}
\]

and the semi-trailer impulse duration with a damped natural frequency of 1.7 Hz (Table 5.14) would ideally have been:

\[
0.35 \times \frac{1}{f} = 0.18 \text{ s}
\]

The VSB 11-style step test input impulse durations were all approximately 0.4 s (Figure 5.3 to Figure 5.5); slightly longer than Doebelin’s recommendation. This slight increase in duration may have contributed to the small variations in the measured values for damped natural frequency as predicted by Doebelin, pp. 79 - 81 (1980). Nonetheless, the model outputs showed good correlation with empirical output data for the same input data.

5.10 Chapter close

Chapter 3, Section 3.2 described methodology for a novel, low cost HV suspension test, the “pipe test” where a HV wheel was rolled over a 50 mm steel pipe at low speed with the air spring response measured. The models developed in this chapter, now validated against empirical data for damped natural frequency and damping ratio, will be used to determine the validity of that low cost test method in Chapter 6. Further, Chapter 12, Section 12.2.4 will explore the results of using three of the semi-trailer axle models (Figure 5.13) developed in this chapter formed into a tri-axle group. This meta-model will be used to explore the ability of a simulated semi-
trailer group to distribute load at the air springs for varying levels of air line connectivity between axles. Accordingly, the models developed and validated in this Chapter will allow exploration of the theoretical limits of air spring suspension load sharing in Chapter 12.

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“Let not the perfect be the enemy of the good.” - Voltaire (attrib.)
6 Quasi-static suspension testing and parametric model outputs

6.1 About this chapter

This chapter presents analysis of the data gathered as described in Section 3.2.3. In it, “pipe test” data are compared with the VSB 11-style step test results. Simulation models were developed in Chapter 4 and validated in Chapter 5. These simulation models were used in an exercise where empirical “pipe test” data were used as inputs to those models. Correlation of the results of that exercise with the results from VSB 11-style testing in Section 3.2.3 is detailed.

6.2 Introduction

As detailed in Chapter 1, a low cost suspension test needed to be developed to meet Objective 3 of this project. The quasi-static testing detailed in Section 3.2.3 for the “pipe test” vs. the VSB 11-style test was analysed to compare the VSB 11-style step test results with the results from the low-cost “pipe test”.

Due to scheduling and logistical constraints of the overall test programme, only one “pipe test” was conducted per test heavy vehicle (HV). Accordingly, the results and analysis of the “pipe test” in this chapter are detailed as a “proof-of-concept” rather than a full implementation approach to that test within a statistically significant developmental framework.

The coach and the semi-trailer “pipe tests” did not yield signals analysable for damping ratio. Nonetheless, acceleration data from the bus “pipe test” were used as an input to the three HV models developed and validated in Chapter 5. This was to broaden the scope of the project to more than just the one successful “pipe test”. The output data from these simulations, as an analogue of air spring pressure, were then compared with the derived values for the suspensions tested using the VSB 11-style step tests. The results of those comparisons are detailed and analysed in this chapter.
6.3 Low-cost suspension testing – “pipe test” vs. VSB 11-style step test – empirical results

6.3.1 General

The test methodology detailed in Section 3.2.3 to meet Objective 3 of this project resulted in, amongst other data, accelerations at the hub of interest during the low-cost “pipe test” and VSB 11-style step tests. The corresponding air pressure transducer (APT) output data from the air springs during those tests were also recorded. The accelerometer data and the APT data for the VSB 11-style step tests have been documented in Chapter 5.

6.3.2 The “pipe test” as an input to the tested HV suspensions

The accelerometer data recorded at the hub of interest on the three HVs tested during their “pipe test” is shown Figure 6.1, Figure 6.2 and Figure 6.3.

![Bus drive axle accelerometer signal - pipe test](image)

Figure 6.1. Time series of bus drive axle hubs’ vertical acceleration during the “pipe test”.

The signals in Figure 6.2 and Figure 6.3 show an asymmetry with a superposition of low and high frequency signals compared with the signal in Figure 6.1.
This issue is better illustrated in the frequency domain. From Chapter 5, and later in this chapter, the axle-hop frequency was determined for the bus and the coach to be approximately 10 Hz. Consider the FFT of the bus accelerometer signal during the “pipe test” compared with the FFT of the bus accelerometer signal during the VSB 11-style test (Figure 6.4).

The FFT of the bus drive axle accelerometer shows maxima in amplitudes varying around the axle-hop frequency and altering slightly depending on excitation method. Nonetheless, the two frequency spectra for the pipe and the VSB 11-style step tests
on the bus drive axle are similar, indicating the reasons for the classical second-order response to the step test as shown in Figure 5.6 and the response to the “pipe test” shown in Figure 6.7.

Now consider the FFT of the coach accelerometer signal during the “pipe test” compared with the FFT of the coach accelerometer signal during the VSB 11-style test. These are compared in Figure 6.5. The spectra differ for the two impulse functions.

Figure 6.4. Indicative frequency spectrum of the bus drive axle vertical acceleration for VSB 11-style step test (left) compared with indicative frequency spectrum of the bus drive axle vertical acceleration for “pipe test” (right).

Figure 6.5. Indicative frequency spectrum of the coach drive axle vertical acceleration for VSB 11-style step test (left) compared with indicative frequency spectrum of the coach drive axle vertical acceleration for “pipe test” (right).
Contrasted with the pair of FFT spectra in Figure 6.4, the “pipe test” (right hand window of Figure 6.5) FFT for the coach drive axle shows a combination of axle-hop around the 10 Hz range in addition to predominant low-frequency signals centred around 3 Hz. The step test on the left hand window of Figure 6.5 does not indicate such a peak at this frequency and is more similar to the left hand window of Figure 6.4. FFTs to compare the two types of test impulse were performed for the semi-trailer axle in the same way as for the coach. These are shown in Figure 6.6.

Figure 6.6. Indicative frequency spectrum of the trailer front axle vertical acceleration for VSB 11-style step test (left) compared with indicative frequency spectrum of the trailer front axle vertical acceleration for “pipe test” (right).

Similar to the coach, the trailer axle had a mixture of frequencies induced by the “pipe test”. These were centred around 4 Hz, as shown in the right hand window of Figure 6.6. The VSB 11-style step test accelerometer signal (left hand window, Figure 6.6) for the semi-trailer axle did not contain the same proportion of low frequency signals as the “pipe test” on that axle.

These results, as impulse inputs to a proposed low-cost test, will be examined in the next section in terms of how they affected the output measured at the air springs.

6.3.3 HV suspension responses to the “pipe test”

The bus speedometer registered from 0 km/h whereas the prime mover and the coach speedometer scales both started at 5 km/h. Accordingly, it was difficult for the driver to moderate the test speeds of the coach and the semi-trailer to balance the
requirement to excite the suspensions with enough energy (\textit{viz;} sufficient speed) with the need to keep the speed below 5 km/h to prevent other, unwanted, stimuli. As a result, the “pipe tests” for the coach and the semi-trailer were conducted at the upper end of the speed scale as described in Section 3.2.3. The higher traverse speeds for the coach and the semi-trailer “pipe tests” excited frequency spectra shown at the right hand windows of Figure 6.5 and Figure 6.6.

The APT output signals during the “pipe tests” are shown as plots from Figure 6.7 to Figure 6.9.

![Figure 6.7. Time series of APT outputs during the “pipe test” on the bus.](image)

The data in Figure 6.7 from the bus were for a test speed lower than the semi-trailer or the coach. The bus “pipe test” yielded a response that could be classified as a second-order system response to an impulse. It approximated that described in Section 4.3.1, governed by system equations in Section 4.3.3 and exemplified in Figure 4.4.
Figure 6.8. Time series of APT outputs from the coach drive axle during the “pipe test”.

Figure 6.8 and Figure 6.9 show APT output data that were not analysable as second-order underdamped systems owing to their responses not aligning with those expected for classical second-order system responses as referenced previously. This point especially so for damping ratio analysis since the responses did not decay exponentially as would be expected for a second-order system (Section 4.3.1, 4.3.3 and Figure 4.4).

Figure 6.9. Time series of APT outputs from the front semi-trailer axle during the “pipe test”.

Consider the “pipe test” result (Figure 6.7) as a classical second-order system response, as modelled in Figure 4.4, compared with those for the non-classical second-order system responses shown in Figure 6.8 and Figure 6.9. For the cases of
the higher traverse speeds, the impulse periods for the “pipe test” in Figure 6.2 and Figure 6.3 were not markedly different when comparing the two cases of classical vs. non-classical responses. Hence, the impulse period did not make a contributory difference. HVs have a pitch mode of 3 to 4 Hz (Cole & Cebon, 1991; OECD, 1998). It was expected, therefore, that the larger amount of energy present at the higher speeds caused pitching of the HVs. Hence, the extraneous 3 Hz (right hand window, Figure 6.5) and 4 Hz (right hand window, Figure 6.6) signals dominated the input signals to the coach and semi-trailer respectively and transferred those forces to the air springs. This affected the impulses at the axles.

Another contributor to the anomalous results may have been the semi-trailer and coach chassis bottoming-out (Woodrooffe, 1995) and/or the air lines being choked and unable to pass high-velocity air between the air springs during vigorous excitation (Li & McLean, 2003a). This particularly so for the semi-trailer where the APT signal flattened after the first positive excursion (circle, Figure 6.10).

Figure 6.10. Expanded view of APT output showing “bottoming-out” of semi-trailer air spring after initial excitation during “pipe test”.

To analyse as much data as available and readily applicable to second-order system theory, the bus APT data were analysed (Section 6.3.5) to compare the “pipe test” with the VSB 11-style step test data for damping ratio and damped natural frequency. Further, the coach drive axle and the semi-trailer axle were analysed for damped natural frequency from the empirical data. These results are detailed in Sections 6.3.6 and 6.3.7.
6.3.4 Regarding the later use of bus “pipe test” empirical data

The “pipe test” on the bus resulted in a classical second-order response from its air springs; Figure 6.7. Due to the low traverse speed, this was the only “pipe test” to elicit a classical second-order response.

The testing described in Section 3.2.3 recorded accelerations at the axles and air pressures in the air springs. The models developed in Chapter 5 were surrogates for the HV suspensions tested. The acceleration data recorded during the testing described in Section 3.2.3 represented the input variable for those models. The output variable for the models was air spring pressure. To analyse the “pipe test” more widely than for just the bus, the simulation models developed in Chapter 5 for the other HVs had the bus “pipe test” accelerometer data applied as an input. This exploration is detailed in Section 6.4.

6.3.5 Bus suspension parameters: step vs. pipe from empirical data

Figure 6.7 was analysed using Equation 4.18 to derive the damping ratio, $\zeta$, for the bus drive axle. These data were good examples of a classical second-order underdamped system response to an impulse function. The results are shown in Table 6.1 after 5 Hz filtering to remove axle-hop noise (see Section 5.2.1 regarding data smoothing). The damping ratios from the VSB 11-style step tests on the bus ranged from 0.27 to 0.28; a variation of 3.6 percent between sides (Table 5.1). The bus “pipe test” produced a signal that indicated a damping ratio of 0.23. The figures for the derivation of this parameter are shown in Table 6.1.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Pipe test results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value</td>
<td></td>
</tr>
<tr>
<td>$A_1$</td>
<td>1823 1773</td>
</tr>
<tr>
<td>$A_2$</td>
<td>86.5 81.4</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>19.4 18.4</td>
</tr>
<tr>
<td></td>
<td>0.230 0.230</td>
</tr>
</tbody>
</table>
The VSB 11-style step tests carried out as described in Section 3.2 resulted in data (Chapter 5) that varied per side. To overcome this variation, the derived metrics were averaged per side to create models that were amalgams of both left and right side derived HV parameters as expanded previously in Section 5.2.2.

A comparison could then be made between the averaged left and right side values of damping ratio for:

- the two types of impulse forcing function; and
- damping ratio using full cycle values for the variables $A_1$ and $A_2$ (Figure 4.4).

Table 6.2 combines the results from Table 6.1 with the left/right averaged value for damping ratio in Table 5.1 derived from the bus data recorded during the VSB 11-style step test.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Pipe test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damping ratio, $\zeta$, averaged LHS and RHS</td>
<td>0.270</td>
<td>0.230</td>
</tr>
</tbody>
</table>

The “pipe test” produced a signal that indicated a damping ratio of 0.230. The damping ratios derived from the “pipe test” vs. the averaged result from the VSB 11-style step tests varied by -16.4 percent. The reasons for this variation between test results were likely the difference in impulse characteristic and experimental error. The latter has been determined previously to be less than 1 percent (Davis, 2006b) for this data recording system. Further, any experimental error was common to both test types. Accordingly, the differences in damping ratio may be attributed to the difference in excitation methods.

The inversion of period, or $T_d^{-1}$ method (where $T_d$ is the damped natural period, Equation 4.14) was used to find the damped natural frequency, $f$, measured from the
plot shown in Figure 6.7. The resultant values for damped natural frequency are shown in Table 6.3.

Table 6.3. Damped natural frequencies for LHS and RHS air springs – “pipe test” on the bus drive axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Pipe test results</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
</tr>
<tr>
<td>Damped natural frequency, $f$ (Hz)</td>
<td>1.17</td>
</tr>
</tbody>
</table>

A combination of the results from Table 6.3 with those found earlier (Table 5.2) for the damped natural frequency from the VSB 11-style step is shown in Table 6.4.

Table 6.4. Comparison between left/right averaged damped natural frequencies for the two types of impulse testing on the bus drive axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Pipe test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency averaged LHS and RHS, $f$ (Hz)</td>
<td>1.06</td>
<td>1.17</td>
</tr>
</tbody>
</table>

The derived damped natural frequency from the “pipe test” result did not vary between sides for the bus drive axle; noting that the results from the VSB 11-style step test for the bus damped natural frequency in Table 5.2 varied per side by ±0.010 in 1.06. The overall result for damped natural frequency differed between methods by 0.110 in 1.06 or 10.4 percent.

Fundamental differences in the derived damped natural frequency and damping ratio values were evident from the two different excitation methods. These differences may be attributed to dissimilar excitation methods, mechanical variation in the suspension and experimental error. The latter was minimal, as mentioned above and in Appendix 2. The mechanical parts of the suspension for the “pipe test” and the step test were common to both tests. Accordingly, the differences in damping ratio and damped natural frequency may be attributed to the difference in excitation methods.
6.3.6 Coach suspension parameters: step vs. pipe from empirical data

Using the inversion of damped natural period, or $T^{-1}_d$ method (where $T_d$ is the damped natural period, Equation 4.14), $T_d$ was found from the data in Figure 6.8. The resultant values for damped natural frequency are shown in Table 6.5.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Pipe test results</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
</tr>
<tr>
<td>Damped natural frequency, $f$ (Hz)</td>
<td>1.12</td>
</tr>
</tbody>
</table>

The results from Table 6.5 for the “pipe test” were combined with those found earlier (Table 5.8) for the damped natural frequency derived from the data from the VSB 11-style step test on the coach. This allowed comparison of the averaged results for the two methods, shown in Table 6.6, for the coach drive axle.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Pipe test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency averaged across LHS and RHS, $f$ (Hz)</td>
<td>1.12</td>
<td>1.15</td>
</tr>
</tbody>
</table>

As with the VSB 11-style step test, the derived damped natural frequency from the coach drive axle “pipe test” results varied between sides. Noting that the results from the VSB 11-style test for the coach damped natural frequency in Table 5.8 varied per side by +/- 0.020 in 1.12, the same order-of-magnitude variation per side was evident for the coach “pipe test” results shown in Table 6.6. Hence, side-to-side variation between the two excitation methods was similar for the coach. The overall result for damped natural frequency differed between methods by 2.70 percent.
Since experimental error was both small and common to both test methods and the mechanical parts of the suspension were also common to both tests, the differences in derived damped natural frequency were attributable to the difference in excitation methods.

### 6.3.7 Semi-trailer suspension damped natural frequency: step vs. pipe from empirical data

Inversion of the damped natural period (where \( T_d \) is the damped natural period, Equation 4.14) or \( T_d^{-1} \), yielded the semi-trailer damped natural frequency from \( T_d \) measured from the data plotted in Figure 6.9. The resultant values for damped natural frequency are shown in Table 6.7.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Pipe test results</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LHS</td>
</tr>
<tr>
<td>Damped natural frequency, ( f ) (Hz)</td>
<td>1.75</td>
</tr>
</tbody>
</table>

The results in Table 6.7 from the semi-trailer “pipe test” and the VSB 11-style step results for the semi-trailer in Table 5.14 were aggregated into Table 6.8.

<table>
<thead>
<tr>
<th>Variable</th>
<th>VSB 11-style step test</th>
<th>Pipe test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency averaged across LHS and RHS, ( f ) (Hz)</td>
<td>1.70</td>
<td>1.64</td>
</tr>
</tbody>
</table>

Comparing the results from the two different impulses as forcing functions, the damped natural frequency derived for the semi-trailer axle using the two test methods had a difference of -3.50 percent between the averaged values used to
eliminate side-to-side variation. The variation in averaged values of the damped natural frequency from the “pipe test” was of a similar order-of-magnitude to that from the averaged values from the VSB 11-style step test (Table 5.14). As with the other two tested HVs, the differences in the derived values for damped natural frequency were attributable to the difference in excitation methods since other error sources were common to both tests.

### 6.4 Computer modelling using the “slow” “pipe test” excitation

The “pipe test” data for the coach and the semi-trailer (Section 3.2) shown in Figure 6.8 and Figure 6.9 were not recognisable as second-order underdamped system outputs from an impulse input. As mentioned above, the “pipe tests” for the semi-trailer and the coach were run, unfortunately, at a speed too high to elicit classical second-order responses from the suspensions. It is for noting that the “pipe test”, using the same pipe and methodology as used for the impulse testing described in Section 3.2 had been used in previous work (Davis & Sack, 2004, 2006) to provide an impulse into a semi-trailer suspension. Those tests resulted in APT data which were well aligned with the suspension manufacturer’s parameters derived from VSB 11 certification (Davis & Sack, 2004, 2006). They were at a similar low traverse speed to that used for the “slow speed” bus “pipe test” as detailed in Section 3.2.

The output data from the APTs on the bus drive axle in Section 3.2 had classical second-order responses to the slow-speed run over the pipe. That run produced analysable signals yielding good correlation with VSB 11-style step test values. To develop the “pipe test” further than just one vehicle during the project for this thesis, the computer models for the coach drive axle and the semi-trailer developed in Chapter 5 had the “slow” bus accelerometer data (Figure 6.1) applied to them as input data.

Figure 6.11 shows a 5 Hz low-pass filtered time-series of the coach drive axle simulation. This was the result of using the accelerometer data from the slow bus run over the pipe as an input to the coach simulation model. Similarly, Figure 6.12 shows the response of the semi-trailer model when the bus drive axle accelerometer
data were used as the input.

Figure 6.11. Time series of Matlab Simulink model of coach drive axle APT output for empirical coach hub vertical acceleration input during “pipe test”.

Figure 6.12. Time series of Matlab Simulink model of semi-trailer axle APT output for empirical trailer front hub vertical acceleration input during “pipe test”.
Damping ratios and damped natural frequencies were derived from the data in Figure 6.11 and Figure 6.12; the coach drive axle and the semi-trailer axle model outputs using the slow bus traverse over the pipe as the input. These were derived using Equation 4.18 and Equation 4.14 respectively, as before. These values are shown in Table 6.9 and Table 6.10.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Coach model using empirical input data from the slow “pipe test”</th>
<th>Semi-trailer model using empirical input data from the slow “pipe test”</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quiescent signal value</td>
<td>5.96</td>
<td>0</td>
</tr>
<tr>
<td>$A_1$</td>
<td>117</td>
<td>103</td>
</tr>
<tr>
<td>$A_2$</td>
<td>21.8</td>
<td>19.1</td>
</tr>
<tr>
<td>Damping ratio, $\zeta$</td>
<td>0.297</td>
<td>0.259</td>
</tr>
<tr>
<td>Error compared with average of actual VSB 11-style step test damping ratio, $\zeta$</td>
<td>-17.5%</td>
<td>3.60%</td>
</tr>
</tbody>
</table>

The results summarised in Table 6.9 indicated that the semi-trailer model, when presented with the “slow” “pipe test” as an input, yielded a damping ratio that varied from the empirically derived VSB11-style step test result for damping ratio by 3.6 percent. This was the same order-of-magnitude error for the models when empirical VSB 11-style step test data were used as inputs (Table 5.19 to Table 5.21).

The derived damping ratio from the coach drive axle model with the slow “pipe test” had an error of -17.5 percent compared with the derived result for damping ratio from the VSB 11-style step test. This result indicated that the “pipe test” needed to be considered further, including the appreciable error in the derived coach damping ratio.
Table 6.10. Comparison of simulation models’ damped natural frequencies for the slow “pipe test” vs. VSB 11-style values.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Coach model using empirical input data from the slow “pipe test”</th>
<th>Semi-trailer model using empirical input data from the slow “pipe test”</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damped natural frequency (Hz)</td>
<td>1.06%</td>
<td>1.71%</td>
</tr>
<tr>
<td>Error compared with average of actual VSB 11-style step test damped natural frequency, f</td>
<td>-5.30%</td>
<td>0.580%</td>
</tr>
</tbody>
</table>

The data summarised in Table 6.10 indicated that the errors in the derived parameters from the models were of the same order-of-magnitude as those for the models when empirical step test data from the VSB 11-style step test were used as inputs (shown Table 5.19 to Table 5.21). This was considered a very good result and indicated that the “pipe test” needed to be considered further as a test. This issue will be expanded later in Section 6.5 and Chapter 11.

6.4.1 Regarding errors; the “pipe test” vs. the VSB 11-style step test

Table 6.11 to Table 6.13 provides a summary of the errors for:

- the model simulation outputs for empirical data inputs to the particular axle; and

- empirical output data from the VSB 11-style step tests vs. the “pipe test”.

Table 6.11. Summary of errors – bus drive axle.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Comparison of VSB 11-style step test vs. “pipe test” – bus drive axle Live drive method – empirical data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damping ratio</td>
<td>-16.4%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>10.4%</td>
</tr>
</tbody>
</table>
The “pipe test” input signals from Figure 5.3 to Figure 5.5 may be compared with the VSB 11-style step down test input signals shown from Figure 6.1 to Figure 6.3. The “pipe test”, when compared with the VSB 11 step down test:

- imparted a similarly shaped time-domain impulse into HV suspensions;
- imparted the required impulse for a comparable time; and
- had a similar amplitude at the axles.

The discontinuity provided by the pipe was sufficient to provide an impulse of the appropriate characteristics and equivalent to that of the step down test used in VSB 11 to excite a HV suspension. The results, from actual experimental data, of the response to the “pipe test” input were sufficient to yield dynamic suspension parameters with a worst-case error of approximately 16 percent (Table 6.11) for the damping ratio.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Comparison of VSB 11-style step test vs. “pipe test” – coach drive axle</th>
<th>Using bus “pipe test” data as input vs. VSB 11 empirical result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Live drive method – empirical data</td>
<td></td>
</tr>
<tr>
<td>Damping ratio</td>
<td>n/a</td>
<td>-17.5%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>2.70%</td>
<td>-5.30%</td>
</tr>
</tbody>
</table>

This was for the “pipe test” case when compared with the VSB 11-derived parameters, since manufacturer’s parameters were not available. Further, any errors were within this same margin as the results for VSB 11-style tests conducted contemporaneously with the “pipe tests”.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Comparison of VSB 11-style step test vs. “pipe test” – semi-trailer axle</th>
<th>Using bus “pipe test” data as input vs. VSB 11 empirical result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Pipe test result vs. VSB 11 result</td>
<td></td>
</tr>
<tr>
<td>Damping ratio</td>
<td>n/a</td>
<td>3.60%</td>
</tr>
<tr>
<td>Damped natural frequency</td>
<td>-3.50%</td>
<td>0.580%</td>
</tr>
</tbody>
</table>

As noted in the previous work (Davis & Kel, 2007; Davis et al., 2007), the tag and the drive axle damping ratios and damped natural frequencies were determined to be independent and distinct. This result was from the use of accelerometers fixed to the chassis of the coach. It could be proposed that close coupling due to the air spring connection mechanism between the dissimilar coach tag and drive axles produced APT signals that were synchronised and therefore indistinct from each other. This possibility may have contributed to the inability of the computer simulation model to accommodate the successful “slow” “pipe test” accelerometer data to produce a damping ratio result within the error envelope of the other simulation parameter results.

Another possibility may have been that the coach axle accelerometer data were somehow different from that of the bus axle data for the “pipe test”. Were this the case, it could explain why the substitution of bus axle accelerometer data did not provide a correlated result for the exercise where the bus axle data were used as an input into the coach axle model (Table 6.12). However, the semi-trailer axle model provided results well within the envelope of difference in empirical test results, Table 6.13. This axle’s sprung and unsprung masses were almost half of that of the coach axle. Accordingly, that supposition may be ruled out, leaving the close-coupling scenario the most likely.
6.5 Summary of this chapter

6.5.1 General

This section summarises the results and analysis detailed in this chapter in preparation for a discussion in Chapter 11.

6.5.2 The “pipe test” vs. VSB 11-style step test - duration

VSB 11 (Australia Department of Transport and Regional Services, 2004a) defines testing of heavy vehicle (HV) suspensions for “road friendliness”. The step test is one VSB 11 method for an impulse to be imparted to a HV suspension; the tyre is rolled off an 80mm step. The resultant axle-to-chassis transient signal is then measured and analysed for damping ratio and damped natural sprung mass frequency. This action is akin to characterising a control system by the application of an impulse as a forcing function (Chesmond, 1982; Doebelin, 1980).

The time taken by a HV tyre to execute the VSB 11 specified step is finite. Figure 5.3 to Figure 5.5 shows that this time was approximately 0.4 to 0.5 s. Similarly, the impulse duration from the “pipe test” shown from Figure 6.1 to Figure 6.3 is also approximately 0.4 to 0.5 s. Previous work provided a theoretical study of the duration of the impulse signal from the “pipe test” (Davis & Sack, 2006). That theoretical work predicted 0.43 s impulse duration at the axle for an 11R22.5 tyre and a 50 mm steel pipe and allowed for tyre enveloping using a point-follower model. The validation of this work was in the match between the impulse lengths from the theoretical study and the empirical data. Accordingly, a combination of:

- 5 km/h approach speed, and
- traverse over a 50 mm pipe

would provide an impulse of sufficient signal strength and duration to allow the resultant signal at a HV spring to be analysed for body bounce frequency and damping ratio.
6.5.3 The “pipe test” vs. VSB 11-style step test - errors

The error summary shown from Table 6.11 to Table 6.13 indicated that the maximum error between “live drive” “pipe test” results and empirical VSB 11-style step test results was approximately 16 percent and this for the damping ratio. Damped natural frequency error between the two test methods was approximately 10 percent. Damped natural frequency is dependent on air spring size and suspension geometry and does not change that much in service (Blanksby et al., 2006; Patrick et al., 2009). Frequency has a component that is proportional to \( \frac{1}{\sqrt{1-\xi^2}} \) as detailed in Equation 4.16. Accordingly, halving the damping ratio from (say) 0.2 to 0.1 alters the frequency by 1.5 percent. The damping ratio on the other hand, being dependent on damper health, changes with time (Blanksby et al., 2006); any alterations to shock absorber performance affect the damping directly and proportionally, as detailed in Chapter 5. Accordingly, damping ratio may be considered to be the most critical (and dependent on damper health) of the two parameters that define a RFS. The error values found for this test programme, when determining damping ratio values in particular, were surprisingly low since previous researchers noted difficulty in determining this parameter accurately (Woodrooffe, 1995) and others who noted a 60 percent difference in derived results, depending on method (Prem et al., 1998; Uffelmann & Walter, 1994). The work of Uffelmann and Walter (1994) examined the potential for measured parameter values to differ, even when derived from the same suspension. The conclusion from that work was that these differences derived directly from, and were dependent on, excitation method; the damping ratio derived from a “pull down and release” type impulse to a HV suspension differed 42 percent from the damping ratio derived from the same suspension subjected to a “lift and drop” impulse. It is noted that both of these methods are allowed under VSB 11. Further, VSB 11 testing in Australia does not generally involve multi-axle groups (Section 11.3.1). The pipe test, provided the traverse speeds are constrained within a manageable but narrow band, would allow multi-axle groups to be tested within an in-service environment.
6.5.4 The “pipe test” vs. VSB 11-style step test – need for development

As mentioned in the introduction to this chapter, the “pipe tests” undertaken for this project were singular occurrences. Different project funding and logistical arrangements may have facilitated multiple runs over the pipe and step tests resulting in analysis for statistical significance.

As the results stand, they are a “proof-of-concept” only. Nonetheless, previous work in an earlier Main Roads project performed multiple runs over the “pipe test” (Davis & Sack, 2006). The results for that programme indicated that errors between the “pipe test” and the manufacturer’s VSB 11 specifications varied by no more than 14 percent for frequency and 12 percent for damping ratio, similar to the results detailed in this chapter.

6.6 Chapter close

6.6.1 General

Chapter 3 described two methodologies for low cost testing of HV suspension. One was the “pipe test” where a heavy vehicle (HV) wheel was rolled over a 50 mm steel pipe at low speed with the air spring response measured. Those test procedures have been analysed in this chapter.

The models developed in Chapter 5, previously validated against empirical data for damped natural frequency and damping ratio, were used to determine the validity of the low cost “pipe test” method in this chapter.

Chapter 12 will use the semi-trailer axle model developed in this chapter. Three of the axle models in Figure 5.13 will be formed into a tri-axle group. This meta-model will be used to explore the ability of a simulated semi-trailer group to distribute load at the air springs for varying levels of air line connectivity between axles. Accordingly, the models developed and validated in this chapter will allow exploration of the theoretical limits of air spring suspension load sharing in Chapter 12.
Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“Public opinion in 1491 held that the world was flat - no progress without analysis.”
7 Data analysis - on road testing and roller bed

7.1 About this chapter

This chapter presents analysis of the data from the roller bed testing detailed in Section 3.3. This was to inform the remainder of Objective 3 of this project; a low cost suspension test. This chapter also presents the results and analysis of empirical dynamic wheel forces from the on-road testing (Section 3.2.2) in terms of test speeds and a “novel roughness” measure developed for this project. These are presented as an innovative contribution to the project’s body of knowledge and for use in Chapter 10 in evaluating the changes to air sprung heavy vehicle (HV) suspensions from larger longitudinal air lines.

7.2 Introduction

As detailed in Chapter 1, Sections 1.4 and 1.7.3, a low cost suspension test was developed to meet Objective 3 of this project; development of a low cost suspension tester. The results from the roller bed testing data gathered as described in Section 3.3 were analysed. These, as well as analysis of the data gathered from the on-road testing detailed in Section 3.2.2 are presented. A useful outcome was evident from this analysis; the ability to show dynamic wheel forces measured against speed and roughness values for the test roads used. These dynamic forces vs. a “novel roughness” value (also denoted a “novel roughness” measure), developed for this project, are documented in this chapter.

7.3 Wheel forces vs. roughness

7.3.1 “Novel roughness” metric - derivation

As mentioned in Section 3.2.7, road roughness is usually designated by a standard measure, the international roughness index (IRI), found using calibrated vehicles. The units of this roughness measure are mm/m or m/km. IRI indicates an amount of vertical movement relative to travelled horizontal distance. This roughness measure
is standardised (Sayers et al., 1987; Sayers et al., 1986). Each hub on each axle of interest had acceleration data recorded during the on-road testing for this thesis, as detailed in Section 3.2. Net vertical acceleration measured at the hub was used after the constant gravity component was removed. A double integration was performed on the vertical acceleration data at a representative axle of each test heavy vehicle (HV). This yielded a “novel roughness” value of positive vertical movement of the axle for a given horizontal distance travelled. The horizontal distance travelled for each 10 s of recorded data was different for each test speed. Accordingly, the velocity of each HV during each test needed to be included in the derivation of the roughness results.

Equation 7.1 provides a mathematical derivation of the “novel roughness” value used.

\[
\text{“novel roughness”} = \frac{\int \int_{a} a \, da}{v} \text{ m/m}
\]

where:

\[ a = \text{net upward hub acceleration during the recording period; } \]
\[ v = \text{velocity in metres per 10 s and } \]
\[ n = \text{the number of data points recorded over 10 s. } \]

\textit{nota bene:} only the positive values of } a \text{ were integrated, in line with the philosophy that the IRI measure is determined as a positive slope.}

The units in Equation 7.1 were resolved as follows:

\[ a: \text{ metre.s}^{-2} \]
\[ a \text{ integrated twice: } \int \int \text{ metre.s}^{-2} \Rightarrow \text{ metres } \]
\[ v: \text{ horizontal metres/10 s. } \]
Equation 7.2 provided the transformation of measured acceleration into the positive vertical displacement (in metres) that the hubs moved during the 10 s recording period (vertical metres/10 s) per test run. Returning to Equation 7.1, the units of “novel roughness” from Equation 7.1 may then be resolved:

\[
\text{“novel roughness” units} = \frac{f[a]}{v} = \frac{\text{vertical metres/10 s}}{\text{horizontal metres/10s}} = \frac{\text{vertical metres}}{\text{horizontal metres}}
\]

A factor of 1000 was applied to render this “novel roughness” value into mm/m.

This “novel roughness” value or “novel roughness” measure should not be equated to the IRI value of the roads used for the testing. It was derived to provide an indicative measure of roughness as experienced by each test HV axle at a representative hub accelerometer. The tyres, axle mass and wheel mass varied with each test vehicle. Accordingly, the “novel roughness” value derived was unique to each vehicle. It arose from the contributions of the unsprung mass dynamics combined with those from surface irregularities. In this way, it was similar to the methodology for determining IRI; that methodology does not distinguish between contributory forces from the axle-to-body dynamics of the test vehicle compared with those from the surface irregularities of the pavement (Sayers et al., 1987; Sayers et al., 1986). Even so, the “novel roughness” value provided an independent variable against which to plot wheel force as the dependent variable.

7.3.2 “Novel roughness” vs. wheel load

The data plotted from Figure 7.1 to Figure 7.9 show the peaks, standard deviations and means of the wheel forces vs. “novel roughness” values for each test vehicle axle of interest. The drive axle of the coach and the front axle of the semi-trailer were chosen as the axles of interest for those HVs for the plots. The coach drive axle was chosen as its forces were higher (and therefore potentially more damaging) than the tag axle of the coach. The semi-trailer’s front axle plots were very similar to its other axles.

A brief commentary is provided, where appropriate, on each figure in the sections on the bus, coach and semi-trailer below.
Discussion and conclusions regarding these plots is contained in Section 7.6.2 and Chapter 13 including left/right variations shown later in this section.

### 7.3.3 Wheel forces vs. “novel roughness” - bus

Figure 7.1 indicated that smoothest roads did not always have the lowest peak wheel forces for the bus drive axle peak wheel forces vs. the corresponding “novel roughness” values. Peak wheel forces generally increased with “novel roughness” values for this test HV but not to the extent that the correlation coefficient of the linear regression between the “novel roughness” range and peak wheel forces was above 0.707, i.e. not in a statistically significant manner. It is likely that this was due to, for instance, isolated patches of distress in otherwise relatively smooth sections. Further, peak values occurred on the right side when the test circuit incorporated the right-hand lane of a one-way section (i.e. where the cross-fall sloped down to the right).

![Figure 7.1. Bus drive axle peak wheel forces vs. “novel roughness”](image)

Figure 7.2 indicated no correlation between the mean wheel forces from the bus drive axle and increasing “novel roughness” values. Linear regression correlation coefficients were substantially below 0.707 for this relationship.
The maxima in standard deviations of the bus wheel forces did not always occur at peak “novel roughness” values, as seen in Figure 7.3. The standard deviations did not correlate to increasing roughness over the range, with regression coefficients below 0.707.

A few rough patches in otherwise comparatively smooth sections were likely to be the cause for this result. High standard deviation values on the right hand side of the bus occurred for one-way sections with cross-fall opposite to the usual construction.

The linear regression correlation coefficients for the relationship between the bus drive wheel forces parameters are shown from Figure 7.1 to Figure 7.3 and
increasing “novel roughness” values is summarised in Table 7.1. These parameters were not correlated with increasing “novel roughness” values for the bus.

Table 7.1. Correlation coefficients for bus wheel force parameters with increasing roughness.

<table>
<thead>
<tr>
<th>Correlation coefficient, R, of wheel force parameters over “novel roughness” range – bus drive axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Std. dev.</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>&lt;0.707</td>
</tr>
</tbody>
</table>

Figure 7.1 and Figure 7.2 indicated visually that the bus peak and mean drive wheel forces were dependent on the side from which the measurement was taken. Figure 7.3 data indicated that the standard deviations were not. As a check, a t-test for variations per side in standard deviation, mean and peak wheel forces against the range of increasing “novel roughness” values was performed. A value for $\alpha = 0.1$ was chosen since road-damage business cases generally use this $\alpha$ value as an upper bound. The choice of $\alpha = 0.1$ was conservative; 0.2 has been used for business cases in mechanical engineering applications with skewed distribution data (Kleyner, 2005). A heteroscedastic test option was chosen since the data had unequal variances (Kariya & Kurata, 2004) over the range of “novel roughness” values. A two-tailed test was used since there was no control case, only two test cases: left and right (Hamburg, 1983). Variation per side in peak, standard deviation and mean wheel forces for increasing “novel roughness” values was tested and, where the results of the t-test met this criterion, they are shown in shaded cells in Table 7.2.

Table 7.2. t-test results for bus wheel forces over “novel roughness” range.

<table>
<thead>
<tr>
<th>Left/right wheel force t-test table for range of “novel roughness” – bus drive axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Std. dev.</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>0.320</td>
</tr>
</tbody>
</table>
The result of this t-test confirmed that the mean and peak forces on the bus drive axle varied per side for the range of increasing “novel roughness” values but the standard deviations did not. This result and the lack of correlation between some bus wheel force parameters and “novel roughness” values will be discussed in Section 7.6.2.

7.3.4 Wheel forces vs. “novel roughness” - coach

The coach drive axle wheel forces were plotted against “novel roughness” values as for the bus. These data are shown from Figure 7.4 to Figure 7.6 with a brief commentary on each plot. Figure 7.4 indicated that, similar to the bus, peak forces did not always occur on the roughest roads.

![Figure 7.4. Coach drive axle peak wheel forces vs. novel roughness.](image)

Even so, the correlation coefficient for the linear regression relationship between increasing “novel roughness” values and peak wheel forces was above 0.707. Again, where the peak forces occurred on the right, the test sections had cross-fall opposite to the norm.
Figure 7.5. Coach drive axle mean wheel forces vs. “novel roughness”.

Figure 7.5 indicated that there did not appear to be a correlation between “novel roughness” values and mean wheel forces. The correlation coefficient for this relationship was checked; it fell substantially below 0.707.

The linear regression correlation coefficient for the coach drive wheel force standard deviations was substantially below 0.707 in Figure 7.6; there was no correlation between this variable and increasing “novel roughness” values.

Figure 7.6. Coach drive axle std. dev. of wheel forces vs. “novel roughness”.
The linear regression correlation coefficients for the relationship between the coach drive wheel force parameters, shown from Figure 7.4 to Figure 7.6, and increasing “novel roughness” values is summarised in Table 7.3.

**Table 7.3. Correlation coefficients for coach wheel force parameters with increasing roughness.**

| Correlation coefficient, R, of wheel force parameters over “novel roughness” range – coach drive axle |
|-------------------------------|----------------|----------------|
| Std. dev.                     | Mean           | Peak           |
| <0.707                        | <0.707         | >0.707         |

The data for the coach drive axle forces in Figure 7.5 indicated that mean forces varied per side with increasing “novel roughness” values. This contention, and that of variation per side in standard deviation and peak wheel forces for “novel roughness” values, was tested. As for the bus, a t-test was performed for variations per side in standard deviation, mean and peak wheel forces against increasing “novel roughness” values. The results of those tests, shown in Table 7.4, indicated that only mean coach drive axle forces varied per side.

**Table 7.4. t-test results for coach drive axle wheel forces over “novel roughness” range.**

| Left/right wheel force t-test table for range of “novel roughness” – coach drive axle |
|-------------------------------|----------------|----------------|
| Std. dev.                     | Mean           | Peak           |
| 0.922                         | 0.065          | 0.299          |

This result suggested that the coach, with its drive and tag axle arrangement, was able to reduce the effect of peak forces per side more effectively than could the bus, with its single rear axle.
7.3.5 Wheel forces vs. “novel roughness” – semi-trailer

The semi-trailer wheel forces were plotted against “novel roughness” values as for the bus and coach. Indicative data for the front semi-trailer axle are shown from Figure 7.7 to Figure 7.9 with a brief commentary on each plot included below.

![Figure 7.7. Semi-trailer axle peak wheel forces vs. “novel roughness”](image_url)

In general, the semi-trailer’s increasing peak wheel forces, exemplified in Figure 7.7, corresponded to increasing “novel roughness” values with linear regression correlation coefficients well above 0.707 for all axles and sides.
The plots in Figure 7.8 and Figure 7.9 suggest that neither the semi-trailer’s wheel force standard deviations nor its mean wheel forces correlated with increases in “novel roughness” values.

The linear regression correlation coefficients for the relationship between semi-trailer wheel force parameters shown from Figure 7.7 to Figure 7.9 and increasing “novel roughness” values is summarised in Table 7.5.
Table 7.5. t-test results for semi-trailer axle wheel forces over “novel roughness” range.

<table>
<thead>
<tr>
<th></th>
<th>Std. dev. per axle</th>
<th>Mean per axle</th>
<th>Peak per axle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Rear</td>
<td>Mid</td>
<td>Front</td>
</tr>
<tr>
<td>Correlation coefficient, R, of wheel force parameters over “novel roughness” range – semi trailer axle group</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>&lt;0.707</td>
<td>&lt;0.707</td>
<td>&lt;0.707</td>
<td></td>
</tr>
</tbody>
</table>

Whole-of-axle results are shown in Table 7.5 since the linear regression values for the three derived parameters on the left side did not vary from those on the right. This is not to be confused with any left/right t-test dependency of the actual derived metrics from the forces correlated to “novel roughness” values, below in Table 7.6.

As for the other two tested HVs, a t-test (Table 7.6) was performed for variations per side in standard deviation, mean and peak wheel forces against increasing “novel roughness” values.

Table 7.6. t-test results for semi-trailer axle wheel forces over “novel roughness” range.

<table>
<thead>
<tr>
<th>Left/right wheel force t-test table for range of “novel roughness” – semi trailer axle group</th>
</tr>
</thead>
<tbody>
<tr>
<td>Std. dev. per axle</td>
</tr>
<tr>
<td>Rear</td>
</tr>
<tr>
<td>0.923</td>
</tr>
</tbody>
</table>

The figures in Table 7.6 indicated that the only forces that varied per side were the mean wheel forces. This result is significant since it is similar to the result from the coach, noting that the semi-trailer axle group and the coach rear group are multi-axle groups. The plots in this section and the results for left/right variation shown in Table 7.6 will be further explored in Section 7.6.2 and Chapter 13.
7.4 Wheel forces left/right variation vs. speed

7.4.1 Introduction

Wheel force data during the road tests over a range of speeds were gathered in the test programme as detailed in Section 3.2.2. Statistical analysis was performed using a t-test to determine significance for left/right variation of wheel forces. This analysis, unlike that undertaken in the previous section, grouped wheel force data per test speed. The analysis below does not include the data from the 70 km/h road test. This was because the variances in wheel force data were unequal within each speed range. This detail ruled out a paired t-test and necessitated a heteroscedastic t-test (Kariya & Kurata, 2004) performed for speeds of 40, 60, 80 and 90 km/h with multiple sets of data recorded per speed. Some of this work has been published previously, including plots of mean, standard deviation and peak wheel forces vs. test speeds (Davis & Bunker, 2009e). Those plots have been omitted from the following section; the interested reader is referred to them for further information. The t-test summary table results for this data are shown from Table 7.7 to Table 7.9.

Similar to the t-test for left/right variation over the range of “novel roughness” values in the previous section, a value for $\alpha = 0.1$ with the heteroscedastic test option (Kariya & Kurata, 2004) and a two-tailed test was used (Hamburg, 1983). Where the results of the t-test met these criteria, they are shown from Table 7.7 to Table 7.9 in shaded cells. A brief commentary is provided, where appropriate, on each table below. Discussion and conclusions regarding these plots are contained in Section 7.6 and Chapter 13 including left/right variation shown from Table 7.7 to Table 7.9.

7.4.2 Left/right variation in wheel forces vs. speed

The bus wheel forces were tested for, and found dependent on, left/right position sensitivity for some speeds as indicated in Table 7.7.
### Table 7.7. t-test summary table for left/right variation in bus drive axle forces.

<table>
<thead>
<tr>
<th>Speed (km/h)</th>
<th>Left/right wheel force t-test table – bus drive axle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Std. dev.</td>
</tr>
<tr>
<td>40</td>
<td>0.105</td>
</tr>
<tr>
<td>60</td>
<td>0.481</td>
</tr>
<tr>
<td>80</td>
<td>0.095</td>
</tr>
<tr>
<td>90</td>
<td>0.546</td>
</tr>
</tbody>
</table>

This result was only for the standard deviation at 80 km/h and the mean at 40 km/h and 80 km/h, however. Differences in the result for left/right variation in wheel forces were drawn from the same data as the previous section but with a different result. This will be explored in Section 7.6.2, sufficing to observe that different results may be derived from the same data by use of different techniques.

The coach drive wheel forces were tested for left/right position sensitivity using the same t-test as above. The t-test analysis results (Table 7.8) showed that there were no significant differences in left/right variation for the mean, peak or standard deviations for the coach drive wheel forces for aggregated speed data over all the runs, however.

### Table 7.8. t-test summary table for left/right variation in coach drive axle forces.

<table>
<thead>
<tr>
<th>Speed (km/h)</th>
<th>Left/right wheel force t-test table – coach drive axle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Std. dev.</td>
</tr>
<tr>
<td>40</td>
<td>0.967</td>
</tr>
<tr>
<td>60</td>
<td>0.926</td>
</tr>
<tr>
<td>80</td>
<td>0.593</td>
</tr>
<tr>
<td>90</td>
<td>0.795</td>
</tr>
</tbody>
</table>

This was in contrast to the result for left/right variation in mean wheel forces for the roughness values in the previous section. Even given the proposition, mooted above, that the coach design provided greater left/right stability, the result of less-than-statistically significant variation per side in coach mean wheel forces indicated that different results arose from different analytical techniques, given the same sets of data.
Semi-trailer wheel forces were subjected to the t-test for left/right position correlation; the results are shown in Table 7.9. These indicated that the mean wheel forces on the front and middle axles of the semi-trailer varied per side for all speeds and with a 90 percent confidence value. This was predominantly on the left but was biased toward the right for one-way right lane test sections as mentioned in the previous section and in previous work (Davis & Bunker, 2009e).

<table>
<thead>
<tr>
<th>Speed (km/h)</th>
<th>Std. dev. per axle</th>
<th>Mean per axle</th>
<th>Peak per axle</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Rear</td>
<td>Mid</td>
<td>Front</td>
</tr>
<tr>
<td>40</td>
<td>0.576</td>
<td>0.883</td>
<td>0.768</td>
</tr>
<tr>
<td>60</td>
<td>0.978</td>
<td>0.867</td>
<td>0.887</td>
</tr>
<tr>
<td>80</td>
<td>0.851</td>
<td>0.809</td>
<td>0.767</td>
</tr>
<tr>
<td>90</td>
<td>0.881</td>
<td>0.909</td>
<td>0.885</td>
</tr>
</tbody>
</table>

It is likely that these variations resulted from the centre-of-gravity (CoG) of the semi-trailer shifting to the left or the right, depending on cross fall. This result was not too dissimilar from that for the mean forces being dependent on side as seen for “novel roughness” vs. mean wheel forces in Table 7.6. The semi-trailer’s front and middle axles were particularly affected by left/right variation but the rear axle was only affected at the highest test speed. This would seem to indicate that the front two axles on the semi-trailer had left/right imbalances where the CoG was thrown to one side or the other by the cross-fall of the road for suburban up to intermediate speeds. The rear axle was not so affected until highway speeds were reached.

The results in this section and the left/right variation shown in Table 7.9 will be further explored in Section 7.6.2 and Chapter 13.
7.4.3 Frequency of forces at the hubs and at the wheels.

A fast Fourier transform (FFT) programme was applied to the time series data from the accelerometers at the hubs and the time series data of the wheel forces. These FFT plots of wheel force data were derived by using Matlab® FFT code developed for the project Heavy vehicle suspensions – testing and analysis and used previously (Davis & Bunker, 2008a, 2008e). This allowed the dominant frequencies at the axle hub and at the wheels to be made more distinct as peaks in the spectra of both. Indicative plots showing the predominant frequency range at the axle hubs of the test HVs are shown from Figure 7.10 to Figure 7.12. Note that these are not necessarily typical since some have been chosen with forces predominant on the right side, some with left side forces predominant.

![Figure 7.10. Frequency spectrum of drive axle hub vertical acceleration – bus, 90 km/h.](image1)

![Figure 7.11. Frequency spectrum of drive axle hub vertical acceleration – coach, 90 km/h.](image2)
Figure 7.12. Frequency spectrum of front axle hub vertical acceleration – semi-trailer, 90 km/h.

Figure 7.12 indicates that axle-hop in the region of 10 Hz was the dominant frequency at the hubs of the semi-trailer at 90 km/h in the indicative plot. This was the lower bound for axle-hop of this HV. Other work (Davis & Bunker, 2008e) showed that the axle-hop for this HV ranged from 10 to 12 Hz. The bus and coach exhibited axle-hop frequency spectra peaking in a band centred around 10 Hz as shown in Figure 7.10 and Figure 7.11. All these results aligned with other research that found axle-hop for HVs in the range 10 to 15 Hz (Cebon, 1999; de Pont, 1997). Further details on the FFTs of accelerometer data from the testing is available from other work from this project (Davis & Bunker, 2008e).

Indicative plots of the frequency spectrum of the wheel forces of the three test HVs are shown from Figure 7.13 to Figure 7.15. These plots show that, in addition to axle-hop frequencies in the 10 to 15 Hz range, additional peaks in the range of 1 to 2 Hz were evident.
Damped natural body bounce frequencies for the tested HVs were shown in Table 5.2, Table 5.8 and Table 5.14. The coach and bus sprung mass damped frequencies were indicatively 1.0 Hz and the semi-trailer approximately 1.7 Hz for this parameter.

A substantial body of work has been documented regarding the ranges of HV sprung
mass (body bounce) and axle-hop (unsprung mass) frequencies. Cebon (1987), Cole and Cebon (1994) and Chatti and Lee (2002), among many others, have noted sprung mass damped natural frequencies of HVs in the 1 to 4 Hz range with unsprung mass damped natural frequencies in the 10 to 15 Hz range.

The twin peaks in the measured wheel force frequency spectra were indicative of a combination of body bounce at these frequencies and a component at approximately 10 Hz from axle-hop. This indicated that forces from body bounce were being added to axle-hop forces to produce the total wheel forces. Figure 7.16 shows an indicative method for visualising this concept. This example is illustrative of wheel forces derived from the testing being the sum of sprung and unsprung mass signals containing the peaks in damped natural frequencies of the sprung and unsprung masses.

The damped natural frequency of the sprung mass in Figure 7.16 is indicated by a peak in the frequency spectrum amplitude of the signals measured at the air pressure transducers during the testing detailed in Section 3.2. Similarly, the axle-hop frequency is indicated by a peak in the amplitude of the FFT spectrum from the indicative accelerometer signals shown.
Figure 7.16. Indicating that peak in the vertical wheel force spectrum may be seen as an addition of damped and undamped natural frequencies.
These results will be used in Section 7.4.4 to expand on spatial repetition of wheel forces.

### 7.4.4 Suspension wavelength and spatial repetition

The axle-hop and body bounce frequencies, shown from Figure 7.13 to Figure 7.15, were translated back into wavelengths. Since frequency may be found as the inverse of the signal period (Equation 4.14), wavelength was derived from the test speed and the fundamental relationship between speed and distance as follows:

\[

\text{d} = \frac{v}{f} \tag{Equation 7.4}

\]

where:

\(v\) = speed in metres.s\(^{-1}\);
\(d\) = distance in m; and
\(t\) = time in s.

Substituting \(1/f\) for \(t\), from first principles, the distance between successive peaks in the wheel forces may be derived:

\[

\text{d} = \frac{v}{f} \tag{Equation 7.4}

\]

Applying Equation 7.4 to the frequency data sets from this project (Davis & Bunker, 2008e), the tested HVs’ suspension wavelengths were derived after examining the predominant axle-hop and body bounce frequencies and corresponding test speeds. These are summarised in Table 7.10.

The "spatial repetition" approach has been discussed in Section 2.4 (Cebon, 1987; Collop & Cebon, 2002; Potter, Cebon, & Cole, 1997; Potter, Cebon, Cole et al., 1995). How this concept relates to the data derived in Table 7.10 will be expanded later in this chapter. For the purposes of presentation of derived data, however, consideration may be given to peak loadings from the HVs tested for this project.

Note that the bold figures in Table 7.10 are for the predominant frequencies and wavelength distances at the corresponding speed; \(i.e.\) body bounce predominated at 60 km/h but both body bounce and axle-hop were approximately equal magnitude at 80 km/h and 90 km/h (Davis & Bunker, 2008e).
Table 7.10. Predominant suspension frequencies at the test speeds and associated wavelength distances.

<table>
<thead>
<tr>
<th>Vehicle/axle group</th>
<th>Speed (km/h)</th>
<th>Body bounce frequency (Hz)</th>
<th>Axle-hop frequency (Hz)</th>
<th>Suspension wavelength distance corresponding to the predominant body bounce frequency (m)</th>
<th>Suspension wavelength distance corresponding to the predominant axle-hop frequency (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bus and coach drive axle</td>
<td>60</td>
<td>1.0</td>
<td>10.0</td>
<td>16.7</td>
<td>1.67</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>1.0</td>
<td>10.0</td>
<td>22.2</td>
<td>2.22</td>
</tr>
<tr>
<td></td>
<td>90</td>
<td>1.0</td>
<td>10.0</td>
<td>25.0</td>
<td>2.5</td>
</tr>
<tr>
<td>Semi-trailer tri-axle group</td>
<td>60</td>
<td>1.7</td>
<td>12.0</td>
<td>9.8</td>
<td>1.4</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>1.7</td>
<td>10.0</td>
<td>13.1</td>
<td>2.2</td>
</tr>
<tr>
<td></td>
<td>90</td>
<td>1.7</td>
<td>10.0</td>
<td>14.7</td>
<td>2.5</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>-</td>
<td>12.0</td>
<td>-</td>
<td>1.9</td>
</tr>
<tr>
<td></td>
<td>90</td>
<td>-</td>
<td>12.0</td>
<td>-</td>
<td>2.1</td>
</tr>
</tbody>
</table>

The distances between peak wheel forces, and their concentrations for a particular HV, may be predicated on the wavelengths shown in Table 7.10. From the indicative distances in this table, the potential pavement distress patches at highway speeds from the bus and coach tested would be likely to occur in a complex pattern of concentrated dynamic impact areas approximately 2m apart with a superposition of additional patches at approximately 22 m. Where one wavelength was an integer multiple of the other wavelength, superposition would be expected to occur at the longer wavelength resulting in a greater probability of pavement failure at longitudinal points where both body bounce and axle-hop were acting together. For (say) a speed of 90km/h, this distance would be approximately 22m. Using the same approach for the bus and coach at low speeds, the potential for concentrated pavement wheel loads, and therefore pavement distress, may be proposed for intervals of approximately 17 m.

The semi-trailer was more likely to induce pavement stress at patches approximately 10m apart for suburban travel. Highway running for the semi-trailer would be expected to produce peak loadings at distances of approximately 14 to 15 m apart with superimposed periodic force maxima approximately 2 m apart. For 100 km/h speeds, these spacings would be expected to increase to approximately 16 m for body

---

1 Lower bound for semi-trailer axle-hop.

2 Upper bound for semi-trailer axle-hop.
bounce and range from 2.3 m to 2.7 m for axle-hop (Equation 7.4 and Table 7.10). Superposition would suggest that, where the longer spacings coincided with the spatial repetition of axle-hop forces, greater pavement distress would occur as a patch.

7.5 In-service heavy vehicle suspension testing - roller bed

7.5.1 Introduction

Objective 3 of this project was to develop two low-cost HV suspension tests. The “pipe test” results have been described and analysed (Chapter 6). For the second low cost test, as detailed in Section 3.3, a wheel within a HV suspension group was vibrated on a roller bed modified from a HV brake testing unit. The forces at the roller for a control case of new dampers were compared with two test cases for worn dampers. The first test case was for a damper worn to the point where tyre wear was apparent due to flat spots on the tyre; the second test case was for no damper. The peak-to-peak values (i.e. the dynamic range) of wheel force signals were derived for these cases. This was in line with the description in Section 3.3 for combinations of the three cases of shock absorber health, two test speeds and rollers with machined areas of 2 mm and 4 mm radial depths.

7.5.2 Peak dynamic forces

As mentioned in Section 3.3, the two test speeds were designed to envelop, but just exceed, body bounce (1.7 Hz) and axle-hop (12 Hz) frequencies for this semi-trailer. The dynamic range concept is illustrated in Figure 7.17.
Figures 7.17 and 7.18 show the time series of the wheel force at the roller for the new shock absorber – 4 mm flat, full load. The range (peak-to-peak value) of the vertical wheel forces is depicted in Figures 7.17 and 7.18. Figures 7.18 to 7.20 illustrate portions (and representative samples) of the time series of the wheel force signals for the three states of shock absorber health: new shock absorber, worn damper, and no damper, for the case of 4 mm deep machined areas and at full load. As reported by other researchers such as Woodrooffe (1996), resonance occurred during the transients from starting the roller compared with the steady state running dynamic forces. This may be seen in the starting transients for the test cases for the worn damper in Figure 7.19 and no damper, Figure 7.20, during the 0.2 to 1.2 s or so (arrowed) before the signal settled down to fairly even excursions.
Figure 7.19. Time series of the vertical wheel forces for worn shock absorber – 4 mm flat, full load.

Figure 7.20. Time series of the vertical wheel forces for no shock absorber – 4 mm flat, full load.

The maximum values of dynamic wheel force range for the three conditions of shock absorber health are shown from Figure 7.21 to Figure 7.24.

Figure 7.21. Maximum vertical wheel forces above static for the different shock absorber health conditions, 1.9 Hz excitation, and 2 mm flat.
Max. dynamic wheel force range for 2 mm flat roller - fast speed

![Graph](image)

**Figure 7.22.** Maximum vertical wheel forces above static for the different shock absorber health conditions, 12 Hz excitation, and 2 mm flat.

For the 2 mm deep flat areas on the roller, the dynamic range of wheel forces did not provide data that were distinguishable to a particular damping condition at the slow test speed (Figure 7.21) or for the fast test speed at other than the full load condition (Figure 7.22). This result will be explored in Section 7.6.3.

Max. dynamic wheel force range for 4 mm flat roller - slow speed

![Graph](image)

**Figure 7.23.** Maximum vertical wheel forces above static for the different shock absorber health conditions, slow speed, and 4 mm flat.

The dynamic range of the wheel forces from the test with a 4 mm machined depth at the:

- slow test speed, approximating body bounce (Figure 7.23); and
fast test speed, approximating axle-hop (Figure 7.24), did, however, prove to be conditional on damper health.

![Graph of Max. dynamic wheel force range for 4 mm flat roller - fast speed](image)

Figure 7.24. Maximum vertical wheel forces above static for the different shock absorber health conditions, fast speed, and 4 mm flat.

The results from Figure 7.24 indicated that wheel forces from a suspension with shock absorbers worn to the point where tyre wear was apparent were clearly distinguishable from the forces present when new or no dampers were present. Section 7.6.3 will discuss the implications from the data shown in Figure 7.23 and Figure 7.24.

### 7.5.3 Maxima of wheel forces in the frequency spectra

The data for the 2 mm excitations shown in Figure 7.21 and Figure 7.22 were indeterminate with respect to damper condition vs. dynamic range with the exception of the 2 mm deep machined areas at the fast speed. The correlation between damper condition-state and wheel forces for the 4 mm deep machined areas excitations in Figure 7.23 and Figure 7.24 prompted more analysis. Accordingly, frequency-domain analysis using fast Fourier transforms (FFTs) was performed on the data for the 4 mm excitation tests at fast and slow speeds for the full load condition. The data from those tests is shown from Figure 7.25 to Figure 7.30.
Figure 7.25. Frequency spectrum of the vertical wheel forces for new damper – 4 mm flat, fast test speed, full load.

Figure 7.26. Frequency spectrum of the vertical wheel forces for worn damper – 4 mm flat, fast test speed, full load.

Figure 7.27. Frequency spectrum of the vertical wheel forces for no damper – 4 mm flat, fast test speed, full load.
The maxima of dynamic wheel forces at frequencies approximating axle-hop (Figure 7.25 to Figure 7.27), indicated that:

1. the dynamic range of axle-hop forces when no damper function was present was approximately twice that for the case where new dampers were fitted; and
2. the dynamic range of axle-hop forces with dampers worn to the point where tyre damage was apparent from visual inspection was approximately one-third greater than that for the new damper condition.

![FFT of signal for new damper](image1.png)

**Figure 7.28.** Frequency spectrum of the vertical wheel forces for new damper – 4 mm flat, slow test speed, full load.

![FFT of signal for worn damper](image2.png)

**Figure 7.29.** Frequency spectrum of the vertical wheel forces for worn damper – 4 mm flat, slow test speed, full load.
In contrast to the data shown from Figure 7.25 to Figure 7.27, Figure 7.28 to Figure 7.30 reflected that varying the damper condition and using the slow speed to excite the suspension did not yield definitive differences in peaks in the frequency series for full load using the roller with the 4 mm deep machined areas for excitation. This was indicated strongly by the existence of multiple peaks in the spectra for low-speed excitation of the HV wheel under test, particularly in comparison with the single, unmistakable peak shown in each of the plots shown from Figure 7.25 to Figure 7.27. The slow-speed test FFTs contained a combination of body bounce, pitching, tyre imbalance and axle-hop. The fast speed excitation reflected only axle-hop.

### 7.6 Summary and conclusions from this chapter

#### 7.6.1 General

This section summarises the results and analysis detailed in this chapter in preparation for an expanded discussion to indicate where they fit into the future of industry research in Chapter 12 and academic research in Chapter 13.
7.6.2 HV suspension metrics derived from wheel forces

Mean wheels forces varied per side for the bus and the semi-trailer. This appeared to be the result of the effect of the cross-fall of the road shifting the heavy vehicles’ centres-of-gravity to the left or the right and the ability of the heavy vehicle (HV) to compensate for this action. Peak wheel forces from the coach and the semi-trailer correlated to increasing values in “novel roughness” but those for the bus did not appear to do so. Conversely, the bus wheel force standard deviations, mean wheel forces and peak wheel forces did not correlate with increasing “novel roughness” values. There was no correlation between standard deviations and mean wheel forces for the coach and semi-trailer with increasing “novel roughness” values.

By using a different independent variable (test speed) instead of roughness, different results were obtained for the derived parameters from the same data sets. A preliminary conclusion may be made that care needs to be exercised, therefore, in ensuring that results from one method are cross checked against results from another method.

Chapter 2 detailed some pavement damage models using a “power law” exponent to attempt to account for the variation in empirical pavement life correlated to axle passes. To “build in” an allowance for dynamic wheel forces, damage exponents ranging from 1.0 to 12 have been used (Austroads, 1992; Pidwerbesky, 1989). These approaches to pavement damage models need to be considered in terms of the results in this chapter.

The adherence to HV suspension dynamic metrics containing only standard deviations (Eisenmann, 1975; Sweatman, 1983) to the exclusion of other metrics such as peak forces (Cebon, 1987; Collop et al., 1994) needs to be re-considered. Further, the concept that mean forces are generally higher on the lower part of the cross fall of the pavement due to shifting centres-of-gravity needs to be incorporated into pavement designs. This particularly so as indicated by the left/right forces dependency results from both roughness and speed for the “workhorse” of the Australian HV fleet: the semi-trailer. These concepts will be expanded in Chapter 13, Section 13.4.
7.6.3 Regarding the dynamic range for different damper conditions

The experimental design was framed to determine if the damper condition was apparent from measuring the dynamic range of wheel force signals. By exciting one side and leaving the other side free to vibrate, axle roll was generated. This was considered in terms of how the auxiliary roll stiffness of the axle would have affected the results. The only variable that was altered between tests was the damping characteristic of the shock absorber. The axle auxiliary roll stiffness leading to the phenomenon of axle roll was common to all tests. Accordingly, any influence that axle roll had on the measured net dynamic forces from the HV wheel for each case of damping characteristics was present in the results of all tests. Since only relative values of wheel force dynamic ranges were considered, the axle roll component for individual results therefore cancelled.

Damper conditions were varied from new to no damping. With 4 mm deep machined areas on the roller, the overall dynamic wheel forces above static were approximately doubled when comparing the control case of new shock absorber of vs. the test case of no shock absorbers. Further, with dampers worn to the point where they were causing tyre wear, peak dynamic forces were approximately 30 percent greater than for the case of the new damper.

The frequency and time series analysis of the data for 2 mm deep machined areas on the roller at either 1.9 Hz or 12 Hz did not yield definitive conclusions regarding damper health. Further, the frequency domain analysis of the data from the 4 mm deep machined areas at 1.9 Hz did not prove conclusive with respect to damper condition. However, as a “proof-of-concept”, both the frequency and time domain analysis showed that damper condition could be detected using the test rig at axle-hop frequencies with 4 mm deep machined areas on the roller. Accordingly, these results will need to be considered in any future designs for in-service testing for HV suspensions.

Development of in-service testing of HV suspensions, should the current National Transport Commission project consider a method similar to that detailed in this project, will need to ensure that sufficient amplitude is imparted to the HV wheel for meaningful differentiation of damper health to occur.
The results also allowed a judgement to be made regarding the additional damage that worn dampers contribute to the community cost of providing the transport network. This will be explored in Chapter 12, Section 12.4.2.

7.6.4 Regarding the use of tyre wear as an indicator of damper health

Using tyre wear as an indicator that suspension dampers are worn is not an indicator that results in timely shock absorber replacement. The results of in-service compliance testing by Blanksby et al., (2006) showed that HVs without noticeable tyre wear fell outside the “road friendly” criteria for damping ratio in over 50 percent of the tested population. This issue will be further explored in Chapter 12, Section 12.4.1.

7.7 Chapter close

7.7.1 General

Chapter 3 described methodologies that resulted in large data sets for wheel force data. Some of those data sets have been analysed in this chapter for wheel forces during the on-road tests. A “novel roughness metric” or “novel roughness” measure was developed. Application of this “novel roughness” parameter led to results and preliminary conclusions regarding how some wheel force metrics, such as peak values and standard deviations to name two, are related to left/right side bias and “novel roughness”. Analysis of left/right differences in wheel forces was also presented for speed ranges as recorded and described in Chapter 3.

The “novel roughness” measure will be used in Chapter 10 to explore whether larger longitudinal air lines alter air suspension behaviour. Chapter 11 will explore in-service HV suspension testing informed by the results of the roller bed tests analysed in this chapter. Further, the issue of tyre wear as an indicator that suspension dampers are worn will be further explored in Chapter 12, Section 12.4.1, supported by the analysis detailed herein. That differing levels of wheel force were evident for different levels of damper health was shown in this chapter. These data will be used
in an indicative exercise regarding the community cost of poor damper maintenance in Chapter 12, Section 12.4.2. Future research directions will be expanded in Chapter 13, Section 13.3 as informed by the approaches shown above.

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.
8 On-board mass system characterisation

8.1 About this chapter

This chapter summarises the applicable results of the non-tamper testing (methodology as documented in Section 3.4) conducted as part of the on-board mass (OBM) system accuracy and tamper-evidence activity to meet Objective 4.

The OBM testing was overseen by Transport Certification Australia (TCA) as lead agency. Various data from that test programme contributed to the project “Heavy vehicle suspensions – testing and analysis”.

8.2 Introduction

OBM systems were tested as part of this research programme. Those test procedures were covered in Section 3.4 with the rationale for the test programme covered in Section 1.5. The on-board mass (OBM) systems presented to the testing team were tested in their “as offered” condition (Section 3.4). Sometimes this condition was without particular prior preparation, save for calibration, or without the OBM systems installed on vehicles selected with a great deal of prior notice. The test team’s original view was that the supplier of the OBM system under test would borrow a heavy vehicle (HV) from their client (Davis et al., 2008b) and this was the case for majority of the HVs and OBM systems tested. This arrangement allowed testing to proceed on HVs and OBM systems in regular service. In some cases, however, the prevailing economic situation during the testing precluded the use of working HVs since they could not be freed from normal service to allow the testing to proceed. In these instances, the tested HVs were hire units with OBM systems installed and calibrated immediately prior to the testing.

The test plans under which the testing was carried out contained elements developed to determine the accuracy and tamper evidence of contemporary, commercially-available OBM units in Australia (Davis et al., 2008a, 2008b). From the data recorded during the testing, evidence was collected that allowed determination of accuracy and tamper-evidence of those systems. This chapter provides:
the detailed analysis of data concerned with accuracy as determined by measuring OBM outputs vs. certified weighbridges; and

data from normal OBM operation (as compared to that for OBM tampering detailed in Chapter 9), for comparison and development of tamper-evident metrics apparent from changes to dynamic signals from OBM systems.

8.2.1  Regarding statistical measures used in this chapter

This chapter refers to x-y (scatter) plots of data. The values of axle group mass from the weighbridges were plotted on the x-axis against the reported mass from the OBM system under test on the y-axis. An example of an x-y plot from the testing programme is shown in Figure 8.1.

Figure 8.1. An example of an x-y (scatter) plot for weighbridge readings vs. OBM system readings.
Figure 8.1 has sets of points plotted with lines of best fit (Dubes, 1968; Hamburg, 1983; Hoel & Jessen, 1971; Layfield, 2003; Werner, 2002). In terms of the OBM systems tested, the lines of best fit provided an indicator of the alignment between the test vehicles’ OBM readings and the actual group mass as measured by the weighbridge(s).

### 8.2.2 Regarding the amount of data to be presented in this chapter

For the OBM systems tested as detailed in Section 3.4, each static test generated a great deal of data. Indicatively, these data were generated from three OBM systems (two reference systems plus one OBM under test) with multiple sensors per axle group at four load points and six measurements at the weighbridges per load point. This was just for the static readings. At the primary transducer level, a total of 4,479 samples were collected across all tested conditions, including 2,175 samples from OBM reading measurement with brakes off, on level ground and no tampering (Karl et al., 2009b). In addition to this, 30 s of dynamic data were captured at 41.6 Hz from the primary transducers for each HV per circuit per load point.

Accordingly, the following section presents representative samples and aggregates data into more tractable data sets.

### 8.3 OBM testing programme results

#### 8.3.1 Results – static tests

During the consultation phase of the OBM project, the axle group mass was defined as the smallest “unit of measurement” for a HV component in which jurisdictions were interested (Davis et al., 2008a, 2008b). The values for the coefficient of determination ($R^2$), y-axis offset from zero ($c$) and the slope of the best-fit line ($m$) were plotted for all axle groups.

Extensive analysis of this data has been published in related work (Karl et al.,

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3 inaccuracies in the mean of the best fit lines for the range of OBM readings.
Figure 8.2 and Figure 8.3 provide visual guides to summarised test data. Figure 8.2 shows a scatter (x-y) plot of zero offsets (c) plotted against the slope (m) of the best-fit lines for all non-tamper tests. Figure 8.3 shows a scatter plot of $R^2$ values plotted against the slope (m) of the best-fit lines for all 111 non-tamper tests.

Figure 8.2. Scatter (x-y) plot of the OBM systems' offset (c) values against the value of the slope of the relationship between OBM reading and weighbridge (m).

Figure 8.2 shows that the offsets from zero on the y-axis of all the data set plots, as exemplified in Figure 8.1, for the OBM systems tested varied. This variation was for a range, between the maxima and minima, of just less than +/- 1500 kg. The relationship between the majority of the OBM system readings when plotted against the weighbridge readings (i.e. the slope, m, of the scatter plots) did not vary by more than 5 percent from the ideal slope designated by 1.0. Further, the values for the offset from zero (c) on the y-axis in Figure 8.2 indicated strongly that 19 in 20 contemporary OBM systems available in Australia are accurate to +/- 500 kg per axle group (Karl et al., 2009b). That this was the case is indicated visually in the spread of data between the lines at the +500 kg and -500 kg points on the y-axis in Figure 8.2.
As a visual guide, the cluster of \( R^2 \) values in Figure 8.3 indicates that all OBM individual system tests (save for three) had \( R^2 \) values in the range 0.995 to 1.0; other analysis (Karl et al., 2009b) provides more detail. Further, as a visual account, the coefficient of determination (\( R^2 \)) values for all OBM system tested were very closely clustered around 1.0. This spread of \( R^2 \) values represented a total variation in linearity of 0.5 percent when the three outliers were removed. This was judged a very good result for jurisdictions and the OBM industry as will be discussed later in this chapter. How these results will influence OBM, if implemented in the future, and implications for the OBM industry and jurisdictions will be expanded later in this chapter and in Chapter 12, Section 12.5.

8.3.2 Results – analysis of dynamic data for non-tamper events

Section 3.4 detailed the testing procedures. In brief, as they relate to dynamic data gathering, the test HVs were driven around a convenient test circuit of a typically...
uneven road or transport yard to exercise their suspensions. This was for every test OBM system reading vs. every weighbridge reading. This action relieved the suspensions of brake wind-up, bushing hysteresis, interleaf Coulomb friction in the springs, etc. Accordingly, the static tests were as free of built-in mechanical bias as possible. Because the reference OBM system was capable of recording dynamic data, exercising the HV suspensions provided an opportunity to record such data from the air springs of the test HVs. The sample rate of the reference system that recorded this data was 41.6 Hz, periodicity 24.0 ms. The testing recorded data sets, exemplified by two time series in Figure 8.4, from the air pressure transducers connected to the air springs (Section 3.4) for each test HV for each circuit for each load point.
Figure 8.4. Examples of dynamic on-board mass data from an APT.

Figure 8.4 shows typical time series data for non-tamper events recorded from the OBM system used to record dynamic data. These test data sets were for air spring data recorded during HV travel on uneven pavements. Such plots were in alignment with typical air spring time series data for other testing programmes (Davis &
Bunker, 2008e). The results from analysis of these dynamic data are below and will be used to derive and validate tamper-evident algorithms in Chapter 9.

For reasons described later in Section 12.5.5, dynamic data from the load cell OBM systems were not recorded dynamically.

Dynamic air pressure transducer (APT) data were analysed using the fast Fourier transform (FFT) program developed for this project and used on the data presented in Sections 7.4.3 and 7.5.3. The frequency spectra at the air springs for all axle groups for all circuits (Davis et al., 2009) were derived. Rectangular FFT windowing was used (Brigham, 1988) to analyse the 1248 data points recorded at 41.6 Hz for each dynamic recording as described in Chapter 3, Section 3.4. Examples of these FFT plots are shown from Figure 8.5 to Figure 8.7. The vertical scales used from Figure 8.5 to Figure 8.7 are arbitrary linear scales but are congruent within the three plots.

Figure 8.5. Example of frequency spectrum of dynamic on-board mass data from an APT.
Figure 8.6. Example of frequency spectrum of dynamic on-board mass data from an APT.

Figure 8.7. Example of frequency spectrum of dynamic on-board mass data from an APT.
The FFTs shown from Figure 8.5 to Figure 8.7 and others derived from the data recorded for this project indicated that, for healthy suspensions without tampering, the frequency spectra of the air springs had various peaks from 1.0 Hz to 15 Hz. Further, the frequency peaks shown were a combination of vehicle Eigenfrequencies, vehicle geometry, and road surface unevenness. Vehicle Eigenfrequencies are related to vehicle dynamics and include such phenomena as body bounce, body pitch and axle hop. Body bounce is usually in the approximate range of 1 to 4 Hz (de Pont, 1997). The pitch mode frequency is usually in the 3 to 4 Hz range (Cole & Cebon, 1991; OECD, 1998). The axle hop frequency is usually in the 10 to 15 Hz range (Cebon, 1999). Other influences that would be manifest in a FFT of air spring data would be where frequency matching between the vehicle Eigenfrequency and the road profile wavelength occurs for a given vehicle speed (OECD, 1998). Peaks in the FFT plots shown from Figure 8.4 to Figure 8.7 correspond to these types of influences. It is not necessary to identify the origin of each peak in the FFT plot in detail any further. This is because the development of the tamper indicators in the next chapter was based on the amplitude of the FFT being uneven, comprising peaks and troughs, across the frequency range. These findings will be used in Chapter 9 for validating a set of indicators that will indicate OBM systems tampering events.

8.3.3 Results – dynamic data vs. static data

The data shown in Figure 8.4 were typical of the data recorded during the test HV circuits. One avenue of investigation for the OBM project was to determine if the static mass aligned with the mean of the dynamic signals from the air springs. A brief visual examination of the data in Figure 8.4 may indicate this possibility. Were this to be the case, it was hypothesised that this equivalence would provide a further indicator for the data being measured by the OBM system (Davis et al., 2008a, 2008b; Karl et al., 2009; Karl & Han, 2007).

Averaging the dynamic data did not provide correlation with the static mass readings (Karl et al., 2009b). This result was likely due to a number of factors. Force on an air spring does not always indicate the total force on the wheel. This has been shown in other work (Davis & Bunker, 2008e) in both time and frequency domains. The non-linearity of dampers (see Sections 4.3.2, 5.5.3, 5.6.3 and 5.7.4) very likely
resulted in differential forces being transmitted to the chassis of the test HVs, depending on the direction of suspension movement (Karl et al., 2009; Prem et al., 1998; Uffelmann & Walter, 1994). This phenomenon would have suggested that the mean of the force at the air spring during travel would be shifted compared with the static force in the air spring. Further, OBM systems on air-sprung HVs generally measure the air spring pressure as a surrogate of wheel force, after suitable calibration. However, there are other load paths from the wheels to the chassis, shown (arrows) in Figure 8.8, after Karl et al., (2009b). It may be seen from that figure that the wheel forces, particularly dynamic wheel forces, may be passed to the chassis by other suspension components. Accordingly, the forces at the air springs are indicators of, but not necessarily equal to, wheel forces.

![Figure 8.8. Load paths from the wheel to the chassis of a HV (after Karl et al., 2009).](image)

**8.4 Summary and conclusions of this chapter**

The testing carried out under Transport Certification Australia’s on-board mass (OBM) accuracy and tamper-evidence programme provided a considerable amount of data. Static accuracy results for OBM systems and their analysis, including subtleties of weighbridge variation, have been detailed extensively elsewhere (Karl et al., 2009b). This issue will be left for readers to inform themselves further as it is verging out of scope for this project.
The accuracy results indicated strongly that, without particular preparation other than calibration, 95 percent of contemporary OBM systems available in Australia are accurate to +/- 500 kg; this being the offset values in the y-axis in Figure 8.2. The results also indicated a total variation of 0.5 percent in OBM linearity for all (save for three) OBM individual system tests (Figure 8.3). This was indicated from the coefficient of determination ($R^2$) values in the range 0.995 to 1.0, when the three outliers were removed. Noting that the OBM systems were tested in “as presented” condition, including being installed on hire HVs in some cases, this appeared to be a very good result for jurisdictions and the OBM industry. Jurisdictions may now be more confident about the performance of these systems, should they begin to specify OBM as part of their regulatory frameworks. The details of such implementation form part of Stage 2 of the Intelligent Access Program outlined in Section 1.5. With some more refinement, the OBM industry may be able to better the figures found from the testing. Implications for the OBM industry and jurisdictions will be expanded in Chapter 12, Section 12.5.

8.5 Chapter close

8.5.1 General

This chapter presented results from the on-board mass (OBM) testing programme conducted by Transport Certification Australia (TCA) where applicable to the project “Heavy vehicle suspensions – testing and analysis”. These results were for the straightforward portion of the test programme and informed the accuracy question posed in Section 1.5.

Dynamic data from the air springs were introduced with some preliminary analysis of the frequency spectra of non-tamper operating conditions. The tampering issue was raised in Section 1.5 with some descriptions of the deliberate tampering events during the tests detailed in Section 3.4.

The next chapter (Chapter 9) will expand on data analysis from the deliberate tampering events, i.e. atypical conditions of operation for OBM systems. This will lead to development of tamper indicators and include an algorithm to indicate a tamper event. These indicators and the algorithm will be validated also in Chapter 9.
Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“All government, indeed, every human benefit and enjoyment, every virtue and every prudent act, is founded on compromise and barter.” - Edmund Burke.
9 Development of tamper metrics

9.1 About this chapter

This chapter details the development of metrics that would inform jurisdictions and regulators of an attempt to tamper with an on-board mass (OBM) system on a heavy vehicle (HV). Tampering events were deliberately introduced into the testing (Section 3.4) for the OBM system accuracy and tamper-evidence work to meet Objective 4. Oversight of the OBM testing was by Transport Certification Australia as lead agency. Some data from that test programme contributed to the project “Heavy vehicle suspensions – testing and analysis”.

9.2 Introduction

HV regulators regard tampering with vehicle instrumentation or measurement systems, particularly those on heavy vehicles (HVs), as a major issue. Transport Certification Australia was the lead agency for a test programme to determine accuracy and tamper evidence for heavy vehicle on-board mass (OBM) systems (Davis et al., 2008a, 2008b). Some controlled tampering was included in that test programme to determine if the effects of that tampering could be detected from changes in the data. This involved changing the operation of the test vehicles or their systems. Those tampering events and the data derived contributed to the project “Heavy vehicle suspensions – testing and analysis”.

Section 3.4.4 detailed the mechanism of interposing ball valves between the air pressure transducers (APTs) and the air springs of the HVs tested. The dynamic data from the air springs were recorded at various load points, depending on logistical arrangements, for the open and closed states of the ball valves. The closed state represented a deliberate tamper event. This was to simulate the instance where an operator may block air lines to an OBM system in an attempt to under-report the mass of the HV in order to overload it or, in the case of mass-distance charging (Clarke & Prentice, 2009), reduce the charge for the trip, for instance.

Examples of data and analysis for the case where the ball valve was open (normal operation) have been detailed in Chapter 8. The tamper events where the ball valve
was closed and mechanisms to detect such events are the subject of the investigations in this chapter.

9.3 **Results – analysis of dynamic data from tamper events**

Testing procedures were detailed in Section 3.4. As mentioned in Chapter 8, test HVs were driven around a convenient circuit of a typically uneven road or transport yard to exercise their suspensions for purposes of relieving the suspensions of brake wind-up, bushing hysteresis, interleaf Coulomb friction in the springs, etc. During this activity, dynamic data were captured at 41.6 Hz, periodicity 24.0 ms, by the OBM reference system capable of this recording. These data were recorded off the air pressure transducers connected to the air springs (Section 3.4) for each test HV for each circuit for each load point. Data for non-tamper events have been exemplified in the time and frequency domains in Chapter 8.

The ball valves were closed to simulate tamper events deliberately designed to prevent detection of correct HV mass while the test HV performed a circuit on typically uneven pavements. Examples of dynamic data recorded for these cases are displayed in the time domain from Figure 9.1 to Figure 9.3.
The choice of left or right side, drive or trailer axle data in these figures was not particularly significant; they are presented as illustrative of typical dynamic data recorded during the tamper testing for all circuits and tested HVs. The plots of the air spring data in Figure 9.1 indicate that the signal is steady, with some small variation due to electronic noise present. It differs markedly from the dynamic data that would be expected from air springs in the time domain as illustrated in (say) Figure 8.4.

Figure 9.2 shows one case where one signal was similar to those in Figure 9.3 and the other signal declined over time. This latter effect was due to a leak in the air line between the APT and the ball valve, leading to a loss of pressure.
Figure 9.2. Example of dynamic on-board mass data from an APT when tampering occurred.

Figure 9.3. Example of dynamic on-board mass data from an APT when tampering occurred.

Figure 9.3 shows two signals from the drive axle of a test HV where both air lines were leaking. A decline in pressure is apparent approximately 5 seconds after the
recording started and before the HV started its test run. Note that the ball valves were turned off at different times for this plot, leading to different starting points for air pressure depletion. Not all air lines in HVs leak; some do and that is a normal condition that is compensated for by the supply of more air from the compressor on the HV. The three examples shown from Figure 9.1 to Figure 9.3 were representative and typical of the results from the tamper events where the air lines to the APTs were blocked. Accordingly, both the leaky and the air-tight states of the air lines for the controlled tamper events in the test programme were indicative with respect to the conditions surrounding potential tamper events in the field. Any tamper measure needed to address these.

Using the fast Fourier transform (FFT) program developed for this thesis and as used on the data presented in Sections 7.4.3 and 7.5.3, the dynamic air pressure transducer (APT) data shown from Figure 9.1 to Figure 9.3 were analysed. Examples of these are shown from Figure 9.4 to Figure 9.6. Rectangular FFT windowing was used (Brigham, 1988) to analyse the 1248 data points recorded at 41.6 Hz for each dynamic recording as described in Chapter 3, Section 3.4. The vertical scales used in Figure 9.4 to Figure 9.6 are arbitrary linear scales but are of the same units.

**Figure 9.4.** Example of frequency spectrum of dynamic on-board mass data from an APT during tampering event.
Figure 9.5. Example of frequency spectrum of dynamic on-board mass data from an APT during tampering event.

Figure 9.6. Example of frequency spectrum of dynamic on-board mass data from an APT during tampering event.
Compared with the FFTs shown from Figure 8.5 to Figure 8.7 and the others derived from the data recorded for this project, the frequency spectra of the air springs when tampering occurred had some distinct and common characteristics, namely:

- a smooth transition from maxima at the lower end of the frequency spectrum to minima at the upper end; and
- no distinct peaks in the frequency spectrum from physical or mechanical influences on the HV air springs save for a maximum amplitude at the lower end of the spectrum.

It is noted that the FFTs did not alter markedly in their overall shapes or characteristics for the leaky tamper events compared with the non-leaky ones. These FFT results were very likely due to the absence of dynamic signals at the APTs that would be expected at the air springs. These dynamic signals would typically comprise a combination of road surface signals, frequency matching of HV Eigenfrequencies (OECD, 1998) and the surface wavelength and HV suspension characteristics imposed by vehicle dynamics such as axle-hop (Cebon, 1999), body bounce (Davis & Bunker, 2008e; de Pont, 1997) or pitch (Cole & Cebon, 1991; OECD, 1998). This combination of signals was not evident in any of the FFT outputs from any of the tamper tests. Previous analysis arrived at a similar conclusion (Davis et al., 2009).

9.4 Tamper indicators

9.4.1 General

Transport industry regulators in Australia would welcome an operational environment where tampering with HV systems did not occur. However, should OBM systems be implemented in a regulatory framework in future, transport industry regulators take the pragmatic view that attempts at tampering will occur. That such a framework is being considered more widely is evident (Clarke & Prentice, 2009). Accordingly, reliable tamper-evident metrics indicating tamper events are required.
### 9.4.2 Tamper index

To detect tampering events, a number of quantitative tampering measures have been developed. This thesis contains results for one such measure, designated the ‘tampering index’ (TIX). The TIX algorithm is an innovative contribution to the project’s body of knowledge and that of this thesis. Other tampering metrics are being developed by TCA and the Department of Transport and Main Roads. As is proper, details of these are being kept confidential so that they are not undermined by unscrupulous activity such as reverse engineering being applied to them. The TIX algorithm is the subject of a joint intellectual property agreement between the Queensland University of Technology and the Department of Transport and Main Roads (DTMR). As such, access will be restricted to this project’s review team and a small number of staff from QUT and DTMR, with due regard to confidentiality requirements. A general description of the TIX algorithm and the results of its application are made public here, but specific details are restricted for the reasons given above.

The tampering index (TIX) is a non-dimensional number and is proposed as one of a range of indicators to be applied to OBM data. These indicators are anticipated as triggers to notify jurisdictions of potential tamper events.

The TIX indicator was derived from the range of dynamic data from the OBM system primary transducers. Were the load or the speed to increase, then the range of the dynamic data would be expected to increase as shown in other work in this project (Davis & Bunker, 2008e). Accordingly, the TIX algorithm normalises the data’s dynamic range so that the TIX values remain within a reasonably constant envelope, irrespective of speed or load. Applying the TIX algorithm to the dynamic data for the non-tamper circuits resulted in a range of values that were indicative of “healthy” operation, *i.e.* no tampering. Figure 9.7 shows a graphical representation of this concept by example. The “healthy” range for TIX values is arrowed and between the upper and lower TIX results. In this figure, the TIX tamper value is the lowest square. Where the derived TIX values resulted in loci, rather than points of maxima and minima, the healthy range for the TIX values was bounded by those loci.
The TIX algorithm was applied to the dynamic data recorded during the test circuits (described in Section 8.3.2) for both the tamper and non-tamper test states. This was to provide TIX values for control data when the HVs were moving without tampering vs. test TIX values derived from controlled tamper events. Thirty-eight instances of tamper vs. non-tamper events were analysed using the TIX algorithm. Three indicative examples for this analysis are plotted Figure 9.8 to Figure 9.10 below.
Figure 9.8. Annotated example of TIX algorithm applied to empirical on-board mass APT data for typical operation and during tamper event.

Figure 9.9. Annotated example of TIX algorithm applied to empirical on-board mass APT data for typical operation and during two tamper events.
In all thirty-eight cases, the TIX values for tamper tests were distinct from TIX ranges or areas between the loci for maxima and minima from the non-tamper data (Davis et al., 2009).

The TIX algorithm was also applied to data collected from air springs during other testing (Davis & Bunker, 2008e) that incorporated a range of speeds from urban to highway. The order-of-magnitude of these TIX results remained constant for this range of speeds. This validated the concept of the inclusion of a normalising factor in the TIX algorithm.

### 9.5 Summary and conclusions of this chapter

The following logical chain of scenarios derived from the testing in this project may inform Stage 2 of the IAP for implementation of on-board mass (OBM) systems on heavy vehicles (HVs). Given any normal operational scenario for a monitored HV, a number of metrics would be expected to be present, including:

1. movement of the vehicle (detected via GPS tracking capability);
2. frequency spectrum of the dynamic data being characterised as:
   
a. not smooth;

b. not declining in magnitude with increasing frequency;

c. with multiple peaks and all of them above (say) 1.0 Hz; and

3. tamper index (TIX) within healthy bounds.

Were scenarios 1 to 3 (and hence their associated metrics) to be present, then tampering would be highly improbable.

Some OBM systems have an accelerometer built into their housing. These elements are used by manufacturers to compensate the OBM reading when the HV is parked on a slope. Other applications of accelerometers in OBM modules are used to determine if the HV is on level ground. Both these applications of accelerometers to OBM readings allow the HV operator to make a judgement of the reliability of the OBM readout accuracy or, in more advanced applications, allow the OBM system to correct the raw transducer data before displaying a mass value (Davis, 2008).

Further to scenarios 1 to 3 above, then, another indicator may be added:

4. dynamic signals from the chassis of the vehicle as it is in motion and as measured by an accelerometer in the OBM unit.

Were a combination of scenario 1, above, to be present without scenarios 2 to 4, it is highly likely that tampering has occurred.

Other combinations, such as where all or part of scenario 2 or scenario 3 were not present individually but scenario 4 was detected, would indicate a high probability that a tamper event had occurred.

Even for basic implementation of tamper-evidence without the multiple verifications of scenarios 1 to 4 (and their associated metrics) above, the simple metrics of:

- movement of the vehicle, as detected by GPS;
- dynamic data being present at the chassis; and
absence of primary transducer signal (due to blocked air line or cables cut);

would be a basic set of conditions that would indicate a high probability of a tamper event.

Deliberate tampering with the OBM system of a HV is easier and more probable when the vehicle is stationary compared with when it is moving. Accordingly, a tampering event attempted when the HV were stationary should be able to be detected from analysis of dynamic data by the use of the rules set out in this chapter. This issue and specification of OBM systems, including sampling rates, will be discussed in detail in Chapter 12, Section 12.5.

9.6 Chapter close

9.6.1 General

This chapter has presented typical tamper data from the heavy vehicle (HV) on-board mass (OBM) testing programme conducted by Transport Certification Australia (TCA) as applicable to the project “Heavy vehicle suspensions – testing and analysis”. It has expanded on data analysis of the deliberate tampering events during the OBM system testing.

A set of tamper indicators was developed and included an algorithm to indicate a tamper event. This was to inform the tamper question posed in Section 1.5, with some descriptions of the deliberate tampering events during the tests detailed in Section 3.4. These indicators and the algorithm have been presented, with some validation, in this chapter.

How these developments will inform road transport regulators and jurisdictions in the implementation of OBM systems for HVs will be further expanded in Chapter 12 and concluded in Chapter 13. Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“Eternal vigilance is the price of liberty.” - Thomas Jefferson.
10 Dynamic load sharing and larger longitudinal air lines for air-sprung heavy vehicles

10.1 About this chapter

Chapter 2 covered a number of significant indicators for heavy vehicle (HV) suspensions. This chapter (Chapter 10) has two purposes. The first is to meet Objective 1 by documenting the development of a dynamic load sharing metric for HV suspensions. The second, to meet Objective 2, is to apply this metric and some of the suspension indicators, outlined in Chapter 2, to data from testing heavy vehicle (HV) suspensions with larger longitudinal air lines to indicate whether larger longitudinal air lines alter the suspension parameters of air-sprung HVs.

10.2 Introduction

10.2.1 Dynamic load sharing in heavy vehicles

Perfect load equalisation would result in a load sharing coefficient (LSC) of 1.0 (Potter et al., 1996). Early research into dynamic load sharing (Sweatman, 1983) documented steel suspension LSCs in the range 0.791 to 0.957. That work placed air suspensions somewhat in the top third of this range with LSCs of 0.904 to 0.925, indicating that they did not necessarily load share any better than the steel suspensions available at the time.

Previous work (Davis, 2006a; Davis & Sack, 2004) has shown that air-sprung “road friendly” suspensions (RFS) do not load share dynamically when in multi-axle groups. That testing, in February 2003 (Davis & Sack, 2004), was on a semi-trailer fitted with “industry-standard” longitudinal air lines (6.5 mm inside diameter, 9.5 mm outside diameter). Note that there is not, strictly speaking, an “industry-standard” air line size but the majority of air sprung HVs have air lines with inside diameters ranging from approximately 4 mm to approximately 12 mm (Davis & Sack, 2004; Simmons, 2005). Where “standard” is used to describe air line sizes in this chapter, it is to this size range that this description refers. The results (Davis & Sack, 2004) showed that the transfer of air between air springs on the test vehicle
was in the order of three seconds. Were HV axle spacings 1 m apart at their closest (worst case), then at 100 km/h (27.7 ms\(^{-1}\)) the reaction time for air to transfer between air springs would need to be in the order of 0.036 s for any reasonable dynamic load sharing to occur. This value could be relaxed to about 0.047 s for axle spacings of 1.3 m at 100 km/h. Hence, air transfer with time constants in the order of three seconds will not load share dynamically. This inability causes potentially more distress to the road network than the case where air-sprung HVs had an improved ability to load-share (OECD, 1992, 1998).

Recent work has measured dynamic load sharing on tri-axle and quad-axle semi-trailers (Blanksby, George, Peters et al., 2008; Blanksby et al., 2009). Even in light of the unaddressed concerns within that body of research and further to Section 2.4.5, that work showed load sharing in air-sprung HVs with industry-standard air lines does not occur dynamically at urban speeds.

10.2.2 Dynamic load sharing in heavy vehicles – larger longitudinal air lines

Two reports (Estill & Associates Pty Ltd, 2000; Roaduser Systems Pty Ltd, 2002) were commissioned by the former National Road Transport Commission (NRTC) after greater mass concessions were granted to HVs in Australia. These considered countermeasures that corrected HV handling problems in air-suspended HVs. Within those reports, HV air suspension modifications using larger-than-industry-standard air lines were prominent. The nominated modifications were a commercial system (the “Haire suspension system”) that used two 50 mm diameter air lines longitudinally, along each side of an air-sprung HV, to connect fore-and-aft air springs in lieu of the relatively smaller air lines normally used (Figure 3.5). These air lines connect to the air springs via 20mm diameter connections (Figure 3.7). The Roaduser report recommended tests to assess the effect of installing larger air lines between sequential axles (Roaduser Systems Pty Ltd, 2002). That report ventured the opinion that this measure should improve the load-sharing capability with two reported cases where installation of such a system rectified handling problems. Further, as far back as 2000, p. 32 of an Estill & Associates (2000) report recommended that the NRTC, now the National Transport Commission (NTC):
“…investigate and evaluate ‘after market improvements’ to air suspensions…” from installation of “…larger diameter pipes to supply and exhaust air flow to the bag quickly and hence improve the response time of the air bag. The modification also reduces the roll and has improved stability.”

Since those recommendations were made, the 2005 test programme funded by the former Queensland Department of Main Roads (Davis, 2006a) and the 2007 test programme (Davis, 2007; Davis & Kel, 2007) comprise the only known published testing, to date, of HVs with larger longitudinal air lines.

10.2.3 Dynamic load sharing in heavy vehicles – regulatory framework

As mentioned in Sections 1.3.1 and 1.3.2, air-sprung HVs were granted concessions to carry greater mass at the end of the 1990s and this trend continues today, conditional on their being fitted with “road friendly” (Australia Department of Transport and Regional Services, 2004a, 2004b) suspensions.

Within the decision making process to allow HVs carrying greater mass in return for being equipped with “road friendly suspensions”, Australian road authorities were advised (OECD, 1992, 1998) that air-sprung HVs with industry-standard (or conventionally sized) air lines between air springs did not load share in the dynamic sense. The OECD research work found that ineffective dynamic load sharing had the potential to cause greater road damage than might otherwise be the case were air-sprung HVs to have improved dynamic load equalisation (OECD, 1992, 1998) over and above that which was available at that time. Hence, the effects of poor dynamic load equalisation were published and known at the time of granting air-sprung HVs concessions to carry greater mass.

10.2.4 Objectives

One portion of the testing detailed in Section 3.2 was designed to produce data that would allow comparison of suspension indicators for the control case of standard longitudinal air lines vs. suspension indicators for the test case of larger longitudinal air lines, the latter being the proprietary “Haire suspension system”. These results
are shown in this chapter, allowing a commencement of meeting Objectives 1 and 2. Alterations to dynamic load sharing for the case of the bus fitted with the “Haire suspension system”, compared with its factory air lines, were not considered because the bus drive axle had no other instrumented axle with which to “share” its load (Section 3.2.1). However, results for alterations to the bus drive axle parameters, other than dynamic load sharing (such as peak dynamic wheel forces and other suspension metrics from Chapter 2), for the two cases of air line size have been documented in this chapter alongside those derived for the other two HVs tested as described in Section 3.2.

10.3 Larger longitudinal air lines

10.3.1 Dynamic load sharing – correlation metric

It should be possible to detect dynamic load sharing from cross-correlation of forces between any two elements in a HV. Were dynamic load sharing occurring, any force on one element, such as on an air spring or a wheel, should result in a force being transferred to a corresponding element. Detecting the degree of correlation between those pairs of forces should indicate the extent of dynamic load sharing; the higher the correlation, the greater the amount of dynamic load sharing occurring. A similar approach was attempted in Austroads project AT1212 (Blanksby et al., 2008; Blanksby et al., 2009), but without definitive results. A contributory factor to the indeterminate results from that test programme may have been that, with the exception of removal of a shock absorber from the middle axle of the semi-trailer, no control vs. test case data were recorded. Apparently, no other comparative testing was undertaken during that programme (Blanksby et al., 2008; Blanksby et al., 2009). Other issues, such as development of an algorithm to convert wheel force correlation to a spatial measure, may have also contributed to the lack of definitive conclusions from that test programme.

The testing detailed in Section 3.2 for the project Heavy vehicle suspensions – testing and analysis followed the scientific method (Section 1.9.1) proposed generically by Mill (1872). For the specific details of this project, standard air lines were the control case and larger longitudinal air lines the test case. An innovative
contribution to this project’s body of knowledge and that of this thesis was derived from analysis of those results. This analysis determined the correlation coefficient, \( R \), between pairs of wheels or air springs on the same sides of the semi-trailer and the coach in real time. This was undertaken for the data sets recorded from each air spring and derived for each wheel as detailed in Section 3.2. The closer the correlation coefficient, \( R \), was to 1.0, the greater the statistical significance between the elements compared (i.e. between wheels or air springs).

This concept will be used in the next section to determine whether changes to the correlation coefficients (and therefore the dynamic load sharing) between pairs of wheels and air springs on the coach and the semi-trailer were detected for the control case of standard air lines vs. the test case of larger longitudinal air lines.

### 10.3.2 Dynamic load sharing – correlation results

Data sets of wheel forces and air spring forces derived from the road tests described in Section 3.2 for the multi-axle groups of interest were analysed by comparing correlation coefficients for the control case of standard longitudinal air lines vs. the test case of larger longitudinal air lines. This comparison applied to pairs of wheels and pairs of air springs from each side of the coach and the semi-trailer in real time as illustrated in Figure 10.1 and Figure 10.2.

![Figure 10.1. Illustrating the pairs of wheels and air springs tested for load sharing using correlation - coach.](image-url)
The coach drive wheels had two data sets recorded from two air springs per test (Figure 3.6); that from the front air spring was chosen for analysis to reduce the number of pairs tested. Each wheel on the semi-trailer had one associated wheel force data set and one air spring data set per test.

![Diagram of wheel and air spring pairs](image.png)

**Figure 10.2. Illustrating the pairs of wheels and air springs tested for load sharing using correlation – semi-trailer.**

As discussed in the previous section, were dynamic load sharing altered by the use of larger longitudinal air lines along each side of a HV, such changes would be expected to show up as an alteration to correlated instantaneous forces between pairs of wheels or pairs of air springs on that side. Alterations consisted of changing the size of the longitudinal air lines along each side of the vehicles; the size of the transverse air lines from one side to the other on the tested HVs were not altered (Section 3.2.1 and Figure 3.5). Accordingly, only pairs of air springs and wheels on the same side of the coach and the semi-trailer were investigated for changes to dynamic load sharing.

To reduce the amount of data displayed to a tractable level, representative and indicative samples from each test speed’s correlation coefficients for each front-to-
back pair of air springs and wheels (Figure 10.1 and Figure 10.2) were plotted for the test case and the control case. The distribution of the correlation coefficients within the variable-space was then bounded to indicate their maxima and minima for the test case and the control case. The results are shown in Figure 10.3 to Figure 10.6.

Figure 10.3. Correlation coefficient distribution of air spring forces for larger (Haire) longitudinal air lines and standard air lines vs. test speed – coach.

Figure 10.3 suggested strongly that the larger longitudinal air lines altered the dynamic load sharing at the air springs by a statistically significant amount. The distribution bounded by the correlation coefficients for the case of larger longitudinal air lines occupied a different variable-space from that for the control case of standard air lines.
Figure 10.4 likewise showed a clear differentiation between the distributions of correlation coefficients for the two cases tested. It was indicative of a situation where larger longitudinal air lines made a statistically significant difference to dynamic load sharing at the air springs. These results for dynamic load sharing at the air springs of the coach and the semi-trailer will be discussed in detail in Section 10.4.2.

The coach’s wheel force correlation coefficient distributions for the two cases tested were not as clear-cut as they were for the air springs. The distributions in Figure 10.5 for the test case and the control case overlapped, with no clear segregation of the two, save for the lowest test speed. This indicated that dynamic load sharing at the coach wheels was not altered in a statistically significant manner by the use of larger longitudinal air lines across the speed range. This result will be discussed in detail in Section 10.4.2.
Similar to the results for the coach, those for the semi-trailer, illustrated in Figure 10.6, indicated that the distribution of wheel force correlation coefficients did not alter by a statistically significant amount for the two cases of longitudinal air line size.
These results will be discussed in detail in Section 10.4.4; suffice to acknowledge here that other work in this project (Davis & Bunker, 2009b) indicated a limit to increases in dynamic load sharing within HV suspensions where other components were kept standard. This was due to the possibility, covered briefly in Section 4.2 and Table 4.1, that increasing load sharing, without associated and considered alterations to the damping characteristics, could incline a HV suspension to become unstable (Davis & Bunker, 2009b).

10.3.3 Alterations to heavy vehicle suspension metrics from larger longitudinal air lines – metrics and methodology

Suspension metrics detailed in Chapter 2 were derived from the data gathered during the road tests described in Section 3.2 for the test case of larger longitudinal air lines vs. standard air lines. Some results from this approach, using test speed as the
independent variable against which to plot the suspension metrics for the test case and the control case, have been published elsewhere (Davis, 2006a, 2007; Davis & Bunker, 2008d). An alternative approach to presenting derived suspension metrics for the case of standard air lines as the control case and larger longitudinal air lines as the test case is shown in the following section.

Section 7.3.1 detailed the development of a “novel roughness” measure or “novel roughness” value. The values given by Sayers et al., (1986) were used as a guide to develop a set of independent variables based on three bands of “novel roughness” values. The bands, in “novel roughness” units of mm/m, were chosen as:

- below 3;
- from 3 to 4; and
- above 4.

These choices within the total range of the “novel roughness” values provided approximately equal numbers of tests, and therefore roughly equal numbers of suspension metrics, within each band. It is emphasised again here, as it was in Section 7.3.1, that the “novel roughness” value was derived for each tested HV and was not related to the IRI developed by Sayers et al., (1986).

The suspension metrics, detailed in Chapter 2, of dynamic load coefficient (DLC), dynamic impact factor (DIF) and peak dynamic suspension force (PDSF) were derived at the air springs of the bus for each of the “novel roughness” bands given above. This was for both the cases of standard air lines as the control case and for the test case of larger longitudinal air lines. Noting that the bus had no ability for its drive axle to share load, neither the dynamic load sharing coefficient (DLSC), nor the load sharing coefficient (LSC), were derived for that HV.

The same metrics were derived from wheel force data for the tests that used the two sizes of longitudinal air lines. For suspension metrics using peak values, the peak dynamic suspension force (PDSF) for air spring data mapped to peak dynamic wheel force (PDWF) when using wheel data. The DLC, LSC, DIF and PDSF/PDWF at the air springs and the wheels were derived for the coach and the trailer. Due to the nature of the DLC and LSC metrics being specific to particular wheels or air springs,
and to bring the data presented down to a tractable level, these values were averaged, after derivation, across all wheels or all air springs within the group of interest. They are shown in the tables below. Peak forces were not averaged; they were processed as maxima with the results shown below.

The DLSC developed by de Pont (1997) was applied to the air spring data and the wheel force data for the coach and the semi-trailer for the two cases of larger longitudinal air lines. Individual results for each wheel or air spring compared with the other wheels or air springs on that side of the HV were derived. This technique altered the application of Equation 2.14 to deriving the DLSC for each air spring or wheel using the forces measured at all of the air springs or wheels on the same side as the particular air spring or wheel:

\[
DLSC_i = \sqrt{\frac{\sum (DLS_i - \overline{DLS})^2}{k}}
\]

\textbf{Equation 10.1}

where:

Dynamic load sharing (DLS) at wheel or air spring \(i = DLS_i = \frac{nF_i}{\sum_{i=1}^{n} F_i}\)

\textbf{Equation 10.2}

\(n = \text{number of wheels or air springs on the side of wheel or air spring } i;\)

\(F_i = \text{instantaneous force at wheel or air spring } i; \text{ and}\)

\(k = \text{number of instantaneous values of DLS, } i.e. \text{ number of terms in the data set (de Pont, 1997).}\)

Similar to the work in Section 10.3.2, front-to-back dynamic load sharing alterations arising from the use of larger longitudinal air lines was the focus since side-to-side air line sizes had not been altered.

A t-test for variations in metrics for the two cases within each “novel roughness” band was performed for the wheel force data and the air spring data. A heteroscedastic test option was chosen since the data sets had unequal variances.
(Kariya & Kurata, 2004) within each band of “novel roughness”. A conservative value for $\alpha = 0.1$ was chosen since, as mentioned in Chapter 7, road-damage business cases generally use this $\alpha$ value as an upper bound. A one-tailed test (StatPac Inc, 2007) was chosen since the results in Section 10.3.2 indicated that the larger longitudinal air lines altered some dynamic measures in a particular direction. A two-tailed test was considered but discarded since its other tail would have informed the case for performance improvement beyond the confidence limit (Hamburg, 1983).

Statistically significant variation between the control case and the test case was tested. Where the results of the t-test indicated that there was a 90 percent or greater probability (i.e. a result less than or equal to 0.1) that the population means of the two cases varied due to the experimental difference and not error, these occurrences are shown in shaded cells below. Where the t-test indicated this statistical significance, the percentage change between the averaged derived parameter values within each “novel roughness” band population for the two cases is shown parenthetically.

10.3.4 Alterations to heavy vehicle suspension metrics at the air springs from larger longitudinal air lines

A t-test for alterations in bus suspension metrics at the air springs was performed for the different sized air lines. The results in Table 10.1 indicated that these alterations did occur but not in a uniform manner across all “novel roughness” bands. Any alterations to suspension metrics at this interface would likely not have been due to any improvements to load sharing since there was no ability to shift air from the air spring on one axle to another. Rather, where such alterations proved statistically significant, the larger air lines were probably acting, as would an accumulator, to reduce axle-to-chassis dynamic forces by compression of the relatively larger air volume for the test case.

The phenomenon of extra compressible fluid acting as an accumulator to soften peak dynamic forces has been hypothesised previously (Davis, 2007; Davis & Bunker, 2008d). Further, this possibility may have led to the result that alterations were not consistent across all “novel roughness” bands due to standing waves in the lines. Where improvements were evident, they were approximately 3 to 5 percent for the
case of larger longitudinal air lines and these for the suspension metrics involving peak dynamic forces.

Table 10.1. t-test table and percent alterations to suspension metrics for the bus air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to bus drive axle air spring force metrics, and, where significant, (percent change)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
<td>Dynamic impact factor</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.307</td>
<td>0.0001 (4.53)</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.350</td>
<td>0.163</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.385</td>
<td>0.138</td>
</tr>
</tbody>
</table>

The tag axle of the coach and the drive axle were treated separately as they were of different masses and had different spring arrangements (Section 3.2.1). Suspension metrics were derived from their data. There was a statistically significant indication of alterations to the coach metrics for the two cases of larger longitudinal air lines as shown in Table 10.2 and Table 10.3.

It was noted that the two metrics utilising peak forces (DIF and PDSF) differed in the significance of their outcomes at the highest band of “novel roughness” for the tag axle of the coach and at the lowest “novel roughness” band for the drive axle.
Table 10.2. t-test table and percent alterations to suspension metrics for the coach tag axle air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>“Novel roughness” (mm/m)</th>
<th>t-test results for alterations to coach tag axle air spring force metrics, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.179</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.246</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.219</td>
</tr>
</tbody>
</table>

These results will be discussed in detail in Section 10.4.3. It is sufficient to note that a number of metrics need to be considered in making a judgement on HV suspensions (Mitchell & Gyenes, 1989), even when they have common factors, such as peak forces, in their derivation.

Table 10.3. t-test table and percent alterations to suspension metrics for the coach drive axle air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>“Novel roughness” (mm/m)</th>
<th>t-test results for alterations to coach drive axle air spring force metrics, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.191</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.247</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.209</td>
</tr>
</tbody>
</table>

The DLSC for the coach air springs of interest was derived, using the air spring data for that side, for the two sizes of longitudinal air lines. The t-test results of these DLSC values for those air springs for the control case vs. the test case are shown in Table 10.4. The results of deriving the DLSC for the coach indicated a uniform
alteration to dynamic load sharing between the air springs along each side when they were connected with larger longitudinal air lines.

Table 10.4. t-test table and percent alterations to dynamic load sharing coefficient (DLSC) for the coach air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>“Novel roughness” (mm/m)</th>
<th>t-test results for alterations to coach rear group air spring force dynamic load sharing coefficient, and, where significant, (percent change)</th>
<th>Tag left</th>
<th>Tag right</th>
<th>Drive left</th>
<th>Drive right</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 3</td>
<td></td>
<td>0.00229 (62.9)</td>
<td>0.000771 (55.5)</td>
<td>0.00356 (52.6)</td>
<td>0.00121 (48.0)</td>
</tr>
<tr>
<td>3 to 4</td>
<td></td>
<td>0.000235 (64.1)</td>
<td>0.00664 (58.3)</td>
<td>0.000185 (58.0)</td>
<td>0.00733 (54.8)</td>
</tr>
<tr>
<td>&gt; 4</td>
<td></td>
<td>0.0393 (43.2)</td>
<td>0.0629 (40.5)</td>
<td>0.0982 (31.4)</td>
<td>0.0882 (34.8)</td>
</tr>
</tbody>
</table>

The values shown in Table 10.4 aligned with the results shown in Figure 10.3, albeit for a different independent variable. The results in Table 10.4 indicated that dynamic load sharing was occurring at the air springs to a greater degree with the larger longitudinal air lines than for the case of standard air lines. An improvement of approximately 30 to 60 percent was evident for the case of larger longitudinal air lines.

The semi-trailer DLC, LSC, DIF and PDSF were derived and the results are shown in Table 10.5. As a preliminary note, in preparation for the discussion in Section 10.4.3, these results indicated that peak forces and some other derived metrics from air spring data were altered for the test case of larger longitudinal air lines connecting air springs along each side of the trailer. The quantum of the alterations, where statistically significant, varied. This will be discussed in Section 10.4.3.
Table 10.5. t-test table and percent alterations to suspension metrics for the semi-trailer air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to semi-trailer group air spring force metrics, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.157</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.0134 (22.1)</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.239</td>
</tr>
</tbody>
</table>

Individual DLSC values per semi-trailer air spring, using the air spring data for that side, are shown in Table 10.6. These data indicated that, similar to the coach in Table 10.4 and as suggested in Figure 10.4, dynamic load sharing at the air springs was improved by approximately 45 to 80 percent with the larger longitudinal air lines connecting fore-and-aft air springs than with the standard longitudinal air lines on the semi-trailer.

Table 10.6. t-test table and percent alterations to dynamic load sharing coefficient for the semi-trailer air springs against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to semi-trailer air spring force dynamic load sharing coefficient, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Rear left</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.00337 (78.4)</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.0242 (76.1)</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.0325 (66.2)</td>
</tr>
</tbody>
</table>

Section 10.4.2 will expand on the results of this section. Briefly, the results from this section may be compared with those in Section 10.3.2; both sets of results indicated that the multi-axle HV groups tested increased their dynamic load sharing at the air springs with larger longitudinal air lines.
10.3.5 Alterations to heavy vehicle wheel force suspension metrics from larger longitudinal air lines

Table 10.7 indicated that the chosen bus suspension metrics derived from wheel force data were not altered for the two cases. Different sized air lines did not make a statistically significant difference to the bus wheel force suspension metrics when they were tabulated against the “novel roughness” value bands.

Table 10.7 t-test table and percent alterations to suspension metrics for the bus drive wheel forces against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to bus drive axle wheel force metrics</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.457</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.202</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.484</td>
</tr>
</tbody>
</table>

Similar to the air spring suspension metrics in the previous section, metrics were derived from coach wheel force data, with the tag axle and the drive axle being treated separately.

The statistical significance for the two cases of longitudinal air line size was apparent but not uniform over the range of “novel roughness” bands for the coach tag and drive axle wheel forces as shown in Table 10.8 and Table 10.9.
Table 10.8. t-test table and percent alterations to suspension metrics for the coach tag axle wheel forces against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to coach tag axle wheel force metrics and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.0597 (39.7)</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.424</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.436 (1.34)</td>
</tr>
</tbody>
</table>

As for the air springs, the two metrics utilising peak forces (DIF and PDWF) did not necessarily align in the statistical significance of their results for either axle. This indicated that suspension metrics using the same elements (i.e. peak forces) did not always provide the same statistical significance for the two cases tested.

Table 10.9. t-test table and percent alterations to suspension metrics for the coach drive axle wheel forces against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to coach drive axle wheel force metrics, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Dynamic load coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.0383 (38.4)</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.347</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.253</td>
</tr>
</tbody>
</table>

The DLSC values from the coach drive wheel forces and tag wheel forces were derived per wheel using the forces measured from the wheels on that side. The DLSCs for each wheel for the cases of the two sizes of longitudinal air lines were subjected to a t-test; the results of which are shown in Table 10.10. These indicated that the larger longitudinal air lines facilitated dynamic load sharing between wheels on the same side when pavements with low “novel roughness” values were being
encountered; medium-to-high “novel roughness” bands resulted in little or no dynamic load sharing between the wheels on any one side.

The results in Table 10.10 indicated that dynamic load sharing was not occurring with certainty for the coach across the range of “novel roughness” values for the cases of the two different sized air lines. This finding reinforced the results for dynamic load sharing using wheel force correlated to speed in Figure 10.6 and will be discussed in detail in Section 10.4.4.

Table 10.10. t-test table and percent alterations to dynamic load sharing coefficient for the coach wheels against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to coach wheel force dynamic load sharing coefficient, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tag left</td>
<td>Tag right</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.0544 (28.9)</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.491</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.483</td>
</tr>
</tbody>
</table>

The DLC, LSC, DIF and PDWF were derived from the semi-trailer wheel forces. The results are shown in Table 10.11 indicating that the larger longitudinal air lines did not make a statistically significant difference to these derived suspension metrics at the wheels, with the exception of the LSC within the high “novel roughness” band.

Table 10.11. t-test table and percent alterations to suspension metrics for the semi-trailer against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to semi-trailer wheel force metrics, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic load coefficient</td>
<td>Load sharing coefficient</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.301</td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.380</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.338</td>
</tr>
</tbody>
</table>
Table 10.12 shows the individual DLSC values per wheel, indicating that wheel force DLSC, and therefore dynamic load sharing at the wheels, did not occur more significantly across the range of “novel roughness” bands for the larger longitudinal air lines than for the case of standard air lines.

Table 10.12. t-test table and percent alterations to dynamic load sharing coefficient for the semi-trailer against “novel roughness” bands.

<table>
<thead>
<tr>
<th>Novel roughness (mm/m)</th>
<th>t-test results for alterations to semi-trailer group wheel force dynamic load sharing coefficient, and, where significant, (percent change)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Rear left</td>
</tr>
<tr>
<td>&lt; 3</td>
<td>0.247</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>3 to 4</td>
<td>0.144</td>
</tr>
<tr>
<td>&gt; 4</td>
<td>0.235</td>
</tr>
</tbody>
</table>

The results for suspension metrics derived from wheel force data sets derived for the three HVs tested will be discussed in detail in Section 10.4.4.

10.4 Discussion of the results from this chapter

10.4.1 General

Indicative suspension metrics for the three tested heavy vehicles (HVs) and the tests described in Section 3.2.2 were derived. Some suspension indicators for the control case of standard air lines vs. larger longitudinal air lines have been documented previously (Davis, 2006a, 2007; Davis & Bunker, 2008d). To contribute further to research, an innovative approach to dynamic load sharing was developed. This technique was added to an alternative determination of dynamic load sharing using the dynamic load sharing coefficient (DLSC) after de Pont (1997). Both approaches were then deployed using different independent variables. The results from using two sets of independent variables supported the hypothesis that improved dynamic load sharing was possible from the use of larger longitudinal air lines.

As noted in Section 4.2.1, increased dynamic load sharing between axles or air springs may not necessarily be conducive to reductions on other suspension parameters. Further, the frequency spectrum of wheel forces may be viewed as a
combination of body bounce and axle-hop (Section 7.4.3), the former predominantly in the 1 to 4 Hz range, the latter in the 10 to 15 Hz range (Cebon, 1999; Davis & Bunker, 2008e; de Pont, 1997). The results of the analysis of spring forces and wheel forces will be treated separately in the following sections.

10.4.2 Alterations to air spring dynamic load sharing from larger longitudinal air lines

Dynamic load sharing using correlated air spring force data was plotted against the independent variable of test speed. This process indicated strongly that there was an increase in dynamic load sharing for the test case of larger longitudinal air lines at the axle-to-chassis interface for the multi-axle HVs tested (Figure 10.3 and Figure 10.4). These results were for an indicative range of correlations and indicative test speeds and showed, in both of these figures, that the range of correlation coefficients was separate and distinct for the two cases of air line size. The strong indicators from the correlation coefficient approach required a different methodology to determine if it was valid. The dynamic load sharing coefficient (DLSC) was applied to the same air spring data but determined against a different independent variable in the form of a range of “novel roughness” measure bands (Section 7.3.1) to separate causality even further. This was to determine alterations to air spring DLSC along either side of the coach and the semi-trailer for the two cases of air line size. The t-test results for the DLSC alterations per air spring per side were uniform across all “novel roughness” bands. It was noticeable that dynamic load sharing at the axle-to-chassis interface, derived from the DLSC, improved across the board for the coach and the trailer in the order of 30 to 80 percent. This result reinforced the validity of, and results from, the correlation coefficient methodology.

Section 2.5 postulated that improvements to the dynamic load coefficient (DLC) and the load sharing coefficient (LSC) were mutually exclusive. The results in Section 10.3.4 indicated that this might be the case for the air spring data from the two multi-axle HVs tested; the LSC improved only when the DLC did not and vice versa. This preliminary result will be left for other researchers to pursue since this is verging out of scope.
10.4.3 Alterations to air spring suspension metrics from larger longitudinal air lines

The dynamic load coefficient (DLC), the load sharing coefficient (LSC), dynamic impact factor (DIF) and peak dynamic suspension force (PDSF) were derived from air spring test data for different bands of the “novel roughness” metric. The results were mixed but, overall, the larger longitudinal air lines made a statistically significant difference to the derived metrics compared with the control case. The metrics involving peak forces such as the PDSF and the DIF showed the most consistent improvement. Whilst some metrics altered in a statistically significant manner, the actual change, when averaged across the “novel roughness” band, was sometimes below the margins of error (Appendix 2). After accounting for this, improvements within the traditional suspension metrics such as the DLC, LSC, DIF and PDSF, where they were statistically significant, varied in the order of 3 to 8 percent.

The bus had no other axle with which to “share” its dynamic load but some alterations to the bus PDSF and the DIF were evident (Table 10.1) from the use of larger longitudinal air lines. Accordingly, such alterations did not necessarily arise from better dynamic load sharing (even though this phenomenon seemed to be occurring between axles and wheels in the multi-axle HVs tested). Any alterations to suspension metrics derived from the peak dynamic forces from any of the HVs tested may have arisen from a combination of factors. One of these may have been improved dynamic load sharing from the larger longitudinal air lines, as noted in the previous section. The other factor, common to both the bus and the multi-axle HVs tested, was an increased air volume contained in the larger air lines. This increased air volume may have, it is suggested, compressed more readily over undulations and acted as would gas in an accumulator, softening peak forces at the springs compared with the control case (Davis, 2007; Davis & Bunker, 2008d).

10.4.4 Alterations to wheel force dynamic load sharing from larger longitudinal air lines

In contrast to the results for the dynamic load sharing at the air springs, the results for wheel force dynamic load sharing indicated a different outcome for the test case vs. the control case of air line sizes. Figure 10.5 indicated that there was an
improvement in dynamic load sharing for the case of larger longitudinal air lines at the 40 km/h test speed. Otherwise, the two ranges of correlation coefficients for the two cases overlapped without a distinct variation between them. The velocity of the air in the air lines for low “novel roughness” and/or low speed would have been relatively low compared with higher test speeds and/or higher “novel roughness” (Li & McLean, 2003a, 2003b). Figure 10.5 indicated that, for the 40 km/h test speed (i.e. low air line velocity) there was an improvement in the ability of the coach to load share dynamically at the wheels in the case of the larger longitudinal air lines. As a cross check, alterations in the coach DLSC for the case of low air flow (Table 10.10) indicated that dynamic load sharing was occurring during low “novel roughness” events with an improvement of approximately 30 to 60 percent. This was signified by the alteration in DLSC per wheel for all wheels, at the lowest “novel roughness” band, compared with the other wheels on that side. This agreement between the two approaches served as a useful verification of the validity of the correlation coefficient as a determinant of dynamic load sharing against the more mature DLSC metric.

Figure 10.6 indicated strongly that there was no difference in dynamic load sharing at the semi-trailer wheels for the two cases of air line size since the correlation ranges overlapped. This result suggested that dynamic load sharing did not alter at the wheels of the semi-trailer for the increased air line size test case for any test speed. Testing this contention for validity, as for the coach wheel forces, the wheel force DLSC derived for the semi-trailer per wheel per side did not alter uniformly across all wheels for any “novel roughness” band (Table 10.12).

10.4.5 Alterations to wheel force suspension metrics from larger longitudinal air lines

Suspension metrics such as the dynamic load coefficient (DLC), the load sharing coefficient (LSC), dynamic impact factor (DIF) and peak dynamic wheel force (PDWF) were derived at the drive wheels of the bus for each of the “novel roughness” bands given above. The results indicated that the bus wheel force suspension metrics did not alter to a statistically significant degree for the two sizes of air line tested.
The coach and the trailer had mixed results for suspension metrics derived from wheel forces. Overall, the larger longitudinal air lines did not make a statistically significant difference to the derived metrics compared with the control case, with the exception of low flow events as noted above, although some metrics involving peak dynamic forces were reduced by up to 24 percent in isolated cases. Section 4.2.1 developed a simple model of load sharing and indicated that an increase in dynamic load sharing between wheels may increase the transmission of axle-hop from one axle to the next. The frequencies of axle-hop have been shown in this project (Section 7.4.3) and in other work (Cebon, 1999; Davis & Bunker, 2008e; de Pont, 1997) to range from 10 Hz to 15 Hz. These are the frequency ranges that coincide with speeds of 60 km/h and above for axle spacings of 1.4 m. Both the coach and the semi-trailer used for the testing described in Section 3.2 had axle spacings of 1.4 m. Accordingly, the transmission of axle-hop frequencies, via improved dynamic load sharing from one wheel or axle to the next, is not necessarily advantageous. The possibility that continuous suspension oscillation at axle-hop frequencies would occur with increased dynamic wheel load sharing has been explored in other work during this project (Davis & Bunker, 2009b). This issue will be expanded in Chapter 12 by undertaking a sensitivity analysis for differing load sharing abilities of a HV suspension model.

With regard to the DLC vs. LSC issue discussed in Section 2.5 and in the previous section, with the exception of the tag axle at the low “novel roughness” band, improvements in these two parameters derived from wheel forces for the two cases of air line size were mutually exclusive. Further, the result for the LSC from the tag axle at the lowest “novel roughness” band improved by less than the error value (Appendix 2). Accordingly, the contention that DLC and LSC are mutually exclusive may well still hold, given that detail. This issue will be left at that point as a research avenue for the future since further pursuit of this concept is verging out-of-scope.
Summary and conclusions from this chapter

10.5.1 Alterations at the air springs from larger longitudinal air lines

Two dynamic load sharing suspension metrics, the DLSC and the correlation coefficient between pairs of air springs, were derived for the multi-axle heavy vehicles (HVs) tested. These were shown to alter for the test case of larger longitudinal air lines by a statistically significant degree leading to the conclusion that dynamic load sharing was facilitated by larger air lines.

Other suspension metrics such as the PDSF and the DIF also altered for all three HVs for the two cases tested. However, changes in these metrics, particularly for those utilising peak force data, could not necessarily be separated in causality from the alterations to the DLSC and the correlation coefficient between pairs of air springs. This was due to increased volumes of compression media, in the form of more air in the larger longitudinal air lines, between the air springs. Nonetheless, the results indicated that, in the majority of cases, dynamic forces at the axle-to-chassis interface were reduced by up to 8 percent with the use of the larger air lines. This leads to the possibility that chassis and air spring forces could be reduced with the implementation of this system. Further, approximate improvements from 30 and 80 percent in dynamic load sharing at the axle-to-chassis interface occurred with the larger longitudinal air lines.

A preliminary observation may be that improving dynamic load sharing does not necessarily lead to improvements in other suspension metrics. These results and issues associated with altering the dynamics between air springs will be explored in Chapter 12.

10.5.2 Alterations at the wheels from larger longitudinal air lines

For the HVs with multi-axle suspension arrangements, the suspension metrics associated directly with dynamic load sharing at the wheels did not alter in a statistically significant manner across the “novel roughness” bands for the case of larger longitudinal air lines. Some alterations to dynamic load sharing at the coach
wheels did occur in a statistically significant manner during low air line flow events. Suspension metrics involving peak dynamic forces altered by up to approximately 24 percent for the test case. However, this range of improvement was not consistent across all wheels and it was not consistent across all roughness bands or speeds.

Alterations to other suspension metrics for all HVs tested, such as the PDWF and the DIF, did not alter in the majority of cases for the test air line size case vs. the control case. Increased dynamic load sharing at the wheels was explored briefly in terms of any potential instability arising from continued oscillations, at axle-hop frequencies, leading to undesirable consequences with respect to vehicle stability. Given the presence of moderate numbers of “Haire suspension systems” in the Australian HV fleet without obvious distress or HV instability (Estill & Associates Pty Ltd, 2000; Roaduser Systems Pty Ltd, 2002), the operational evidence suggests that forces due to axle-hop frequencies are not shared dynamically at the wheels for either size of longitudinal air line. This contention matches with the analysis of the results from this chapter.

10.6 Chapter close

This chapter presented analysis of data gathered as described in Section 3.2.2 using some of the suspension metrics from Chapter 2. An innovative method of determining the presence of dynamic load sharing was also developed using the correlation of forces between pairs of heavy vehicle (HV) components such as wheels or air springs.

Indicative air spring and wheel force data from this analysis were plotted against test speed as the independent variable. Some of the suspension metrics from Chapter 2 were derived for air spring and wheel force data and presented using a range of “novel roughness” bands (Section 7.3.1) as the independent variable. Both these approaches indicated that the issue of larger longitudinal air lines increasing dynamic load sharing at the air springs of the tested HVs has, in general, been resolved in the affirmative.

Any dynamic load sharing improvement at the wheels due to larger longitudinal air
lines was not evident save for low speed/low “novel roughness” events. This was not unexpected since increases in wheel force dynamic load sharing would incline potentially undesirable axle-hop oscillations to propagate continuously during travel. This issue will be explored in Chapter 12 by undertaking a sensitivity analysis for differing load sharing abilities of a HV suspension model.

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“What I dream of is an art of balance.”- Henri Matisse.
Chapter 11  Heavy vehicle in-service suspension testing

11.1 About this chapter

Chapter 1, Section 1.4 detailed the rationale behind in-service heavy vehicle (HV) suspension testing. To meet Objective 3, two low-cost HV suspension tests were developed. Chapter 6 detailed the results for a “proof-of-concept” low-cost HV suspension test that excited a HV suspension during the traverse of a 50 mm steel pipe and measured the air spring response. Chapter 7, Section 7.5 detailed the results of a second, low cost HV suspension test using a modified brake testing machine. This chapter explores the outcomes of those tests in terms of a discussion about the policy and strategic implication of an in-service HV suspension test regime. This will lead on to a discussion on the contribution to knowledge from this project in Chapter 12.

11.2 Introduction

11.2.1 General

Heavy vehicles (HVs) with “road friendly” suspensions (RFS) were granted concessions to carry greater masses under the various higher mass limits (HML) schemes (Section 1.4.1) arising from some of the micro-economic reforms in Australia over the last 15 years. The HV suspension requirements under HML schemes in Australia broadly followed the European assumption that air suspensions on HVs should be allowed a payload advantage over conventional steel-sprung axles based on “damage equivalence” (de Pont, 1997). The health of HV suspensions and the related effect on the road network asset when they move out-of-specification has been well documented (Gyenes & Mitchell, 1992; National Road Transport Commission, 1993; OECD, 1998). That air suspensions, deemed to be “road friendly” at the time and fitted to HVs carrying HML loads, remained healthy was recommended as one of the conditions of access (Pearson & Mass Limits Review Steering Committee, 1996a).

Australia incorporated the 92/7/EEC parameters and tests into its VSB 11 certification regime for “road friendly” suspensions as well as adding additional
requirements regarding static load sharing. “Road friendly” suspensions in Australia generally incorporate air springs, although some steel RFS have come onto the market (Australia Department of Transport and Regional Services, 2004b).

11.2.2 In-service suspension testing of HVs in Australia

Methods to determine the health of air-sprung HV suspensions were proposed before their introduction into Australia (Cebon, 1999; Potter et al., 1997; Woodrooffe, 1995; Woodrooffe et al., 1988). A specification was developed, Vehicle Standards Bulletin 11 (Australia Department of Transport and Regional Services, 2004a, 2004c). This Vehicle Standards Bulletin (VSB) was applied to new suspensions going into service on HVs carrying out HML duties. An in-service suspension test was not specified at the time.

The bi-lateral infrastructure funding agreements (BIFAs) in force between both the State Governments of New South Wales and Queensland with the Australian Government at the time of writing contain commitments to develop an in-service test for air-sprung HVs operating under HML schemes. A project by the National Transport Commission (NTC) to develop such a test is now underway.

The NTC’s brief for the consultancy to that project stated, in part:

..."The objective of this consultancy is to identify and report (with recommendations) on devices that will deliver:

- an in-service field test for road friendly suspension systems including consideration of dynamometer testing or other low cost test methods; and

- a roadside enforcement test..."

Note the emphasis on "low-cost test methods".

As inputs to that NTC project, the former Queensland Department of Main Roads (now part of the Department of Transport and Main Roads) has (amongst other efforts) contributed in-kind support, via a joint project with QUT entitled Heavy vehicle suspensions – testing and analysis (Davis, 2007, 2008; Davis & Bunker,
2007, 2008a, 2008b, 2008c, 2008d, 2008e, 2009; Davis & Kel, 2007; Davis, Kel, & Sack, 2007). The joint project used data from (former Main Roads’) low-cost and preliminary test programmes carried out on a limited scale (Davis & Sack, 2004, 2006) as well as major and subsequent testing work (Davis, 2007, 2008; Davis & Bunker, 2008c, 2008d, 2008e, 2009; Davis & Kel, 2007; Davis et al., 2007).

11.2.3 In-service HV testing in the transport environment

Any test on the air suspensions of the HV fleet operating at HML in Australia would need to be low-cost, given the tight financial margins on which road transport operates. VSB 11 type-testing is comparatively expensive with an order of magnitude of $AUD5,000 per test (Bisitecniks, 2007). This cost is for either single axles or axles in a group. A cost structure of this magnitude would not be acceptable for testing every air-sprung axle on every HV operating under the various HML schemes in Australia. This point especially since the tests would conceivably need to be done annually for typical operation and more frequently under arduous conditions. To achieve acceptance of in-service testing of air-sprung HV suspensions the costs per test would have to be one or two orders-of-magnitude lower than for VSB 11 testing.

11.3 In-service HV suspension testing – issues and discussion

11.3.1 Test standards for in-service HV testing

The standard defining RFS characteristics in Australia at the time of writing is VSB 11. Various regulations govern HVs operating at HML loads. One of these is the requirement for HVs operating at HML loads to be equipped with RFS as they go into service. Their suspensions are certified to VSB 11, by definition. VSB 11 is structured such that a “type test” of one axle at varying loadings is then applicable to that suspension model across the range of installations on HVs, including multi-axle installations. The issue of load sharing between axles tested singly is not addressed definitively in VSB 11 (Davis & Bunker, 2007). Further, the composition of VSB 11
is well suited to the laboratory situation but not as an application appropriate to testing a large volume of HVs.

Despite the issue noted above regarding a low-cost test, any proposal for an in-service test on HV suspensions faces a challenge as defined by the following logic:

- RFS are certified to VSB 11 at the time the HV so equipped commences HML service (usually when new);
- the HV operator is required to maintain the HV suspension such that it keeps its characteristics within the bounds of VSB 11;
- since there is no other definition of “road friendliness”, jurisdictions wishing to specify a test for in-service HV suspension characteristics have, at the time of writing, no other choice but to specify VSB 11;
- therefore, jurisdictions testing a HV for RFS compliance in an in-service environment have no choice, at the time of writing, but to use VSB 11 and suffer the higher costs accordingly.

This chain of logic leads to the conclusion that the only RFS test that is acceptable in Australia at present is VSB 11, in-service or otherwise. Given the nature of VSB 11 as a “type test” more suited to laboratory environments, and as noted above, VSB 11 testing is relatively expensive.

Low cost testing, such as the “proof-of-concept” testing documented in Chapter 6 for the “pipe test” excitation and in Chapter 7, Section 7.5 for the roller bed, are not recognised test methods. Further, there are no low-cost methods for determining VSB 11 compliance. This would seem to be a challenge for regulators at the time of writing. Similar situations have, however, been resolved in the past by jurisdictions. This was usually by specifying a different standard for the new state vs. any in-service requirements. A simple example would be tyre groove depth: the tyre is installed by the retailer and is covered by the applicable standards for new tyres. The vehicle owner replaces the tyre when the groove depth reaches a minimum value as specified by the appropriate regulator. The tyre, by definition, never meets its “as new” specifications during its lifetime but provides service until worn out. It is
suggested that a similar pragmatic relaxation of the “as new” requirements for RFS specifications should occur for in-service suspension testing of HVs.

These issues will need to be addressed by the NTC in developing in-service test standards for RFS suspensions.

11.3.2 In-service HV testing & on-board mass measurement systems

On-board mass (OBM) systems are becoming increasingly prevalent in Australia’s HV fleet. The national project to determine the feasibility of OBM systems in a regulatory framework has formed part of the project for this thesis (Davis et al., 2008a, 2008b; Karl, 2007; Karl et al., 2009). Chapter 8 showed that 95 percent of OBM systems available commercially were accurate to within +/- 500 kg. Further, the results from the reference system and one other used for the testing described in Chapter 8 (Karl et al., 2009b) indicated that reliable dynamic data were available from OBM systems. As noted in Section 3.4.2, most OBM manufacturers have supplied sufficient information to the OBM project team to indicate that dynamic data from air springs or accelerometers could be recorded by OBM systems (Davis, 2008).

A HV may be subjected to an impulse of appropriate magnitude and duration, such as would result from traversing a slightly protruding bar embedded conveniently in a transport operator’s marshalling yard. An OBM system of appropriate sampling speed may record the data from the air springs or chassis accelerometers of that HV. The resultant signals from such data could be analysed for body bounce and damping ratio, i.e. VSB 11 characteristics. This technique need not be confined to air-suspended HVs; analysis of steel-suspended HV chassis acceleration would be applicable.
11.3.3 In-service HV testing – impulse testing and a way forward

The dilemma outlined in Section 11.3.1 becomes slightly more difficult when the results of differences in test methods are examined. Uffelmann and Walter (1994) noted differences in damping ratio results of over 60 percent, depending on suspension test method and excitation impulse direction. Further, Sweatman et al., (2000) allowed a margin of 25 percent between an in-service value for damping ratio and the acceptable threshold for damping ratio under VSB 11. The parameters derived from exciting HV suspensions using the “pipe test” (Chapter 6) indicated an envelope of inaccuracy of approximately 16 percent from the manufacturer’s VSB data or data generated from VSB 11-style step tests. Given the difference in excitation functions between the “pipe test” and the VSB 11 results, this error was relatively small compared with that found by the previous research of Uffelmann and Walter (1994).

Some of these issues would be mitigated were HV operators to perform their own testing under (say) the National Heavy Vehicle Accreditation Scheme (NHVAS). The NHVAS is a national scheme for HVs to meet “alternative compliance procedures” to those enforced by road authorities. It specifies that HV operators abide by a set of standards, management procedures, audits and business rules. There are two modules under NHVAS, maintenance management and mass management. HV operators may choose accreditation in one or both modules. Accreditation is recognised across Australia. Jurisdictional representatives perform regular audits on HV operators accredited under NHVAS to ensure that the provisions are met (National Transport Commission, 2008).

An in-service test would not need to be particularly accurate if the suspension in question were monitored over some months, under arduous service; or years, in suburban service, under a scheme such as NHVAS. The indicative inaccuracy of approximately 16 percent between the “pipe test” results and VSB 11 test results may be acceptable for pass/fail field testing. By testing a HV when new, the errors between VSB 11 values and any in-service test would be clear and, even with inaccuracies due to test method, constant. Accordingly, the in-service test results for that HV suspension would become calibrated against the certified VSB 11
parameters of the HV axle when new. Further, the margins of error between the new values for the suspensions tested and those for (say) the “pipe test” results for this project could be accommodated by plotting the histories of test results. After the suspension reached some threshold, it could be referred to a more precise measurement technique or simply have its shock absorbers (and bushes, etc.) replaced as a matter of course when the HV next came in for servicing.

A pipe or other suitable discontinuity such as used for the testing in this project could be installed in the driveway of a transport facility. Other discontinuities such as increasingly prevalent speed bumps in transport yards may also suffice as forcing functions. With the increasing prevalence of on-board-mass (OBM) measurement systems installed on HVs, such a traverse would result in an OBM system record of air spring or chassis accelerometer data. Were the HV loaded appropriately, these data could be analysed to determine body bounce damping ratio and damped natural frequency (Davis et al., 2007; Davis & Sack, 2004, 2006) within known margins of error as shown in Chapter 6. Any suitable discontinuity such as a bar, pipe or channel would suffice were it of sufficient height to elicit an impulse of enough energy to impart a readable response from the HV’s OBM system. This would need some consideration so that discontinuity height or traverse speed were low enough that the energy thus imparted did not result in body pitch or other unanalysable behaviour in the HV as was detailed in Chapter 6.

For HVs without OBM, the simple expedient of attaching a data-recording accelerometer to the chassis (regardless of suspension type) and driving over the discontinuity should provide a suitable trace for analysis of body bounce damping ratio and sprung mass frequency. This approach could also apply to roadside testing where transport inspectorate staff could attach data-logging accelerometers to the chassis of a HV under test and have it traverse a standardised discontinuity. The trace from such an event could be analysed against known characteristics of that suspension when new. Analysis of such data either from OBM systems or accelerometer data loggers is estimated to cost approximately $AUD100 per axle group, based on the time spent on analysis of data for this project. It is for noting that commercial applications of data analysis from HV on-board mass systems are already emerging onto the market in allied areas (Sack & Doust, 2009).
Some research would be needed to determine the characteristics of available suspensions, including those steel-sprung that are VSB 11-compliant. Given that there are, admittedly, quite a few VSB 11-compliant suspension models available in Australia, their numbers are finite, nonetheless. Accordingly, the “as new condition” characteristics of each could be documented after some form of standardised in-service or field impulse test. The differences in results between the VSB 11-certified values and the measured field values could be noted with a margin applied as agreed by regulators. Such a programme would be the subject of further work in the regulatory field. Hence the developers of any in-service HV suspension test will need to consider these issues; they will need to be addressed by the NTC in developing an in-service testing regulatory environment for RFS suspensions.

11.3.4 In-service HV testing – procedural considerations for impulse testing

The correct procedure to impart a forcing function for an in-service test would need to consider speed and positioning of the HV wheels. Care with traverse speed would be required as has been noted from the results and analysis in Chapter 6. To test HV suspension in an in-service application, some pre-conditions would need to be met. The following list is not exhaustive but relates to the findings from this project:

- vehicle speed at or below approximately 1.2 $\text{ms}^{-1}$ (5 km/h) to keep tyre deformation and chassis pitching to a minimum;
- correct and parallel positioning of the HV wheels with respect to the pipe or other discontinuity used for excitation;
- discontinuity not too great in height (e.g. 40 to 60 mm) to keep tyre deformation and chassis pitching to a minimum; and
- sufficient sampling rate of the measurement system, theoretically above approximately 6 Hz but in the order of 50 Hz to allow for a margin of error.
11.3.5 The roller bed as a low-cost in-service suspension test

Damper condition was detectable using the roller bed test rig (Section 3.3) as designed. This “proof-of-concept” experimental design met the requirement for a low-cost suspension test in that, after approximately $AUD10,000 in set-up costs, the cost of analysis per axle-group would be approximately $AUD100. The roller with two shaved areas of 4 mm depth provided good detection of damper health at fast and slow speed running. It could be argued that the dynamic loads detected as excursions above static wheel forces will vary with suspension design, model and type. This challenge may be overcome in a similar manner to that proposed above for the impulse testing of HV suspensions. Further research will be needed to overcome this issue and will be discussed in Chapter 13, Section 13.2.2.

11.4 Summary and conclusions

11.4.1 The “pipe test” as an in-service suspension test

Pass/fail testing from the “pipe test” would be valid for inaccuracies in reading of no more than 16 percent, noting that the VSB 11-style step test had left/right variation in results of up to approximately 8 percent. Since the in-service testing would likely be conducted over a period on the same vehicles, a known and constant margin of error between the VSB 11 figures and those from tests using a device similar to the pipe used for this thesis would allow trends to be detected.

The damped natural frequency is most attributable to air spring size and suspension geometry. It does not change that much in service (Blanksby et al., 2006; Patrick et al., 2009). Halving the damping ratio from (say) 0.2 to 0.1 alters the frequency by 1.5 percent. The damping ratio on the other hand, being dependent on damper wear, changes with time (Blanksby et al., 2006); alterations to shock absorber performance affect the damping directly and proportionally and are the most critical of the two parameters that define a “road friendly” suspension (RFS). The errors in derived damping ratio values under the test programme for this project were surprisingly low given previous researchers noting difficulty in determining this parameter accurately (Woodroofe, 1995) and others who noted a 60 percent difference in values. This latter variation was noted as resulting from method dependence (Prem et al., 1998;
Uffelmann & Walter, 1994).

The “pipe test” has, in the past (Davis et al., 2007; Davis & Sack, 2004, 2006), provided valid and reliable outputs provided each axle of the suspension group was allowed to settle before the subsequent axle encountered the pipe. For application to a commercial low-cost in-service suspension test, the need for axles in multi-axle groups to settle before subsequent axles traverse the pipe (or other impulse) will need to be balanced against the requirement to have a short enough impulse imparted to the wheels to excite the suspension into a measurable response. Short impulses will require some speed to be effective; too slow will lengthen the impulse and render the results invalid, from the work of Doebelin (1980). Single axles have a greater margin as they do not have the two competing requirements. Tyre enveloping is influenced by a number of factors, amongst which are inflation pressure, size, construction materials and methods, depth of tread and whether in a dual combination or single tyre on a wheel. Determining the elasticity of damping ratio and body bounce frequency results for alterations in these variables will need to be determined.

11.4.2 The roller bed as an in-service suspension test

Damper condition was detectable using the roller bed test rig (Section 3.3) as designed. This “proof-of-concept” experimental design met the requirement for a low-cost suspension test in that, after approximately $AUD10,000 in set-up costs, the cost of analysis per axle-group would be approximately $AUD100. The roller with 4 mm shaved off in order to provide sufficient repetitive excitation provided good detection of damper health at fast and slow speed running.

11.5 Chapter close

This chapter has explored some of the technical and regulatory issues that will need to be addressed to implement in-service testing of HV suspensions. It has drawn from:
the two low-cost suspension testing methods detailed in Chapter 3;

the outputs of the HV suspension models developed in Chapter 5;

and

the HV suspension models analysed in Chapter 6

to propose a way forward, within the documented inaccuracies of the “pipe test” vs. Vehicle Standards Bulletin 11.

The on-board mass systems explored in Chapter 8 were also suggested as possible measurement system vectors for in-service suspension testing. The results of the roller bed tests detailed in Chapter 3 and analysed in Chapter 7 were used to propose a way forward for a second, low cost in-service suspension test for HVs. These discussions and the results of the testing and analysis presented in this thesis in Chapters 5, 6, 7 and 8 will be summarised in Chapter 12, Section 12.3 to show the contribution to knowledge, via Objective 3 from this project, that could lead to development of low-cost HV suspension testing methodologies.

Links where this chapter informs other chapters of this thesis are shown in Figure 1.4.

“We don’t have the money, so we have to think.” - Lord Rutherford.
12 Contribution to knowledge – industrial practice

12.1 About this chapter

This chapter details the contribution to knowledge, readily applicable to industry and regulators, from this project. It draws together the innovative methods and approaches developed under the project entitled *Heavy vehicle suspensions – testing and analysis* and shows how the objectives of that project were met.

12.2 Application to heavy vehicle suspension designs – Objectives 1 and 2

12.2.1 Dynamic load sharing

Objective 1 hypothesised that an improved dynamic load sharing measure for heavy vehicle suspensions could be developed. Objective 2 hypothesised that improvements to dynamic load sharing could be effected by the use of larger longitudinal air lines to air-suspended heavy vehicles (HVs). The following sections outline how those objectives were achieved for the project as well as other results that were developed during the process.

12.2.2 Introduction

Maximum transfer of air from one air spring to its associated rear air spring could be seen to be an ideal situation for load equalisation. However, the phenomenon of axle-hop requires that some imperfection be introduced into the transfer mechanism to reduce the possibility of standing waves in the air spring connector exciting sympathetic oscillations in neighbouring air springs. This phenomenon has been discussed briefly in Sections 4.2.1 and 10.4.5.

A computer model was derived from empirical data for the semi-trailer half-axle developed in Chapter 4. It was validated in Chapter 5. That model was used to create a theoretical model of one side of a multi-axle HV suspension with varying
values of dynamic load sharing. Three HV half-axle models were then linked together with the ability to vary the level of dynamic load sharing between them. The results from that exercise were documented for this project (Davis & Bunker, 2009b). A summary of the approach and the results from that exercise are presented in the following sections.

### 12.2.3 An improved dynamic load sharing metric

Objective 1 was met by deriving the correlation coefficients of forces between different elements, such as air springs or wheels, on a HV. This approach was validated by the use of control vs. test case data for different sized air lines and cross validated by the use of different, but mature, independent variables in Chapter 10, Sections 10.3.4 and 10.3.5 for the same control vs. test case methodology.

### 12.2.4 How much dynamic load sharing is beneficial?

The half-axle model for the semi trailer has been shown previously in Figure 5.13. For the purposes of the exercise, the input to the three half-axle models linked together was empirical accelerometer data from the accelerometer signals measured during the on-road testing as detailed in Chapter 3.2, Section 3.2.2. The accelerometer signals were derived empirically and the model validated in Chapter 5 (Davis & Bunker, 2008a, 2008e) by comparing the output of the model with the empirical signals measured from the air springs. The accelerometer data were entered per test speed via the "Signal builder" block, upper left, Figure 12.1. The air line manifold (upper right Figure 12.1) represented the pneumatic interconnection between air springs. The “transport delay” blocks simulated the delay between the front axle and subsequent axles encountering the empirical acceleration data. The outputs from the model, as analogues of air spring pressure at each air spring, were recorded for the simulations.
The value of gain in the “load sharing fraction” block in Figure 12.1 represented the amount of dynamic load sharing between the air springs as a dimensionless number varying between 0 and 1.0. The load sharing fraction was a construct of the model. Varying the gain in the load sharing fraction (LSF) block allowed differing values of load sharing to be programmed into the model. A sensitivity analysis was performed for differing values of LSFs. It found that increasing values of LSF encouraged axle-hop to dominate the air spring frequency spectra, particularly at high speeds (Davis & Bunker, 2009b). Were this to be reflected in reality, axle-hop frequency forces would have been transmitted to the chassis of the HV, cancelling out the conventional suspension design approach where axle inertia combined with suspension damping act to de-couple the pavement frequencies from the chassis. To overcome this undesirable situation, attenuation of high frequency transmission of air between air springs was added to the model. This modification resulted in instability of the model at LSF values above $1 \times 10^{-5}$.

The complexity involved in modifying HV suspension systems was indicated by this, seemingly beneficial, modification since the suspension became more unstable at high LSF values than for the cases where axle-hop was allowed to resonate through the air spring connections. Altering one variable to overcome perceived drawbacks
did not result in improvements to the model in other areas.

The damper characteristics were varied (Davis & Bunker, 2009b) to ones that were different, but known. The characteristics for the dampers on the coach were substituted for those in the semi-trailer axle model (Figure 5.13) and then configured as in Figure 12.1. With this substitution, the model proved stable for LSFs up to 1.0 (i.e. 100 percent transmission of air between air springs). Altering the shock absorber characteristics allowed the model to maintain stability and have increased air line sizing. These results indicated that air lines between sequential air springs could have diameters in the order of those of the air springs themselves, provided the shock absorber characteristics were more aligned with those found on passenger coaches. This proposal will be left at that point as it is venturing out-of-scope, except to refer to Chapter 13, Section 13.3 for potential future research.

12.2.5 Alterations to HV suspension metrics from the use of larger longitudinal air lines

Objective 2 was met by the use of the innovative approach to correlation coefficients between air spring forces and wheel forces as well as deriving metrics for different independent variables such as test speed and “novel roughness” values in Chapter 10. The results indicated strongly that dynamic load sharing was facilitated at the air springs of the multi-axle HVs tested with larger longitudinal air lines fitted. Further, suspension metrics involving peak dynamic forces measured at the air springs were, in general, reduced somewhat by the use of larger longitudinal air lines. This second phenomenon was likely due to the presence of a greater volume of compressible fluid (air) in the larger air lines absorbing peaks in dynamic forces.

Suspension metrics derived from wheel forces did not indicate consistent improvements from the use of larger longitudinal air lines for the HV tested. It is suggested that this is due to the limitation of the damper characteristics as explored briefly in Section 12.2.4 and extensively in other work developed for this project (Davis & Bunker, 2009b). This latter proposal is mentioned here since the developers of the larger longitudinal air line system tested for this project, properly, did not alter the damper characteristics on the HVs tested. To do so would have
invalidated the VSB 11 certification of the axles used. Were such modifications undertaken, in contravention of suspension certification, potentially more alterations to wheel and chassis forces may have been made. This will be expanded in Chapter 13 Section 13.3.

12.3 Development of heavy vehicle in-service suspension testing methods – Objective 3

12.3.1 In-service HV testing

Objective 3 hypothesised that low cost in-service HV suspension testing was possible. The following two sections outline how that aim was achieved for the project. This work will be provided to the National Transport Commission (NTC) to further the progress of the in-service HV suspension testing project.

12.3.2 In-service HV testing – impulse testing

Systems theory (Doebelin, 1980) informs that the perfect impulse of infinitely small duration contains all frequencies at the same amplitude. The reality falls somewhat short of this ideal since all impulses are finite. Nonetheless, system characterisation may be performed by measuring the output of a system after the application of an imperfect impulse as an input (Chesmond, 1982; Doebelin, 1980). Chapter 6 detailed the results of applying this methodology to heavy vehicle (HV) suspensions as a system.

The innovative “pipe test” results were compared to those of a VSB 11-style step test. These were discussed in detail in Chapter 11. The “proof-of-concept” for this innovative, low cost method was shown to have errors in the order of approximately 16 percent when compared with VSB 11-style test results or manufacturer’s data. These results were an order of magnitude lower than indicated by other research (Prem et al., 1998; Uffelmann & Walter, 1994) indicating that the methodology described herein has contributed an improvement to any proposals for in-service HV suspension testing.
12.3.3 In-service HV testing – roller bed

The roller bed testing results detailed in Chapter 7, Section 7.5 indicated that dynamic oscillatory excitation to a HV wheel could be used to detect damper health from wheel forces. The worse the shock absorber condition, the greater the dynamic range of the wheel force signal measured at the roller. The dynamic wheel force range for 4 mm excursions approximately doubled when the dampers were removed compared with the case for new shock absorbers. The trend found in the test results suggested that even greater dynamic wheel force ranges are being imposed for on-road excursions larger than 4 mm when HVs are operating with dampers in a condition where they are wearing tyres abnormally. These results will be extrapolated in a limited fashion in Section 12.4.2 regarding the additional damage that worn dampers cause vs. those in healthy HV suspensions.

The project showed that damper condition was detectable using the test unit (Section 3.3) as designed. This “proof-of-concept” experimental design met the requirement for a low-cost suspension test in that, after approximately $AUD10,000 in set-up costs, the cost of analysis per axle-group would be approximately $AUD100. The results indicated that a “proof-of-concept” for a moderate-cost testing machine and low cost-per-test has been achieved with the use of a modified roller-brake tester.

The brake-testing machine from which the test rig was made was a standard unit that was not altered with respect to its original overall dimensions. A commercial variant of it should therefore be able to fit into existing infrastructure at roadside interception sites or vehicle maintenance facilities where brake testers are designed to fit.

12.4 Implications for network assets

12.4.1 Using tyre wear as an indicator of damper health

The roller bed testing results detailed in Chapter 7, Section 7.5 showed that, for a damper causing tyre wear, the axle-hop forces were more than 10 percent greater than for those where the damper was in good condition for excitations using a local radial reduction of 2 mm in the roller diameter. The analogous forces with a 4 mm radial shave off the roller were more than 30 percent greater than for those where the
damper was in good condition.

54 percent of 121 air-sprung HVs surveyed at Marulan, NSW, in 2006 had dampers that did not meet VSB 11 but did not have apparent tyre wear (Blanksby et al., 2006). As mentioned in Section 1.4.3, Marulan was not on a HML route at the time; some anecdotal views have opined that the Marulan results had little relevance since VSB 11 compliance was not a requirement for operation on that route. That argument is specious and has been dealt with in Section 1.4.3. The main issue is that HV suspensions worn to the point where tyre wear is apparent do not meet VSB 11 and therefore impose greater forces than do healthy HV suspensions.

The Australasian Road Transport Suppliers Association (ARTSA) advises, amongst other activities, the air suspension industry via a code of practice (Australian Road Transport Suppliers Association, 2001). That document lists (p. 21), under the heading “shock absorber troubleshooting indicators” the advice that replacement of shock absorbers should be considered when an “increase in tyre wear and flat spots” is/are apparent. This advice to use tyre wear as a trigger for damper replacement needs to be reviewed.

12.4.2 Regarding the community cost of poor HV suspension health

Costanzi & Cebon (2007; 2005) modelled a fleet of HVs with 50 percent ineffective dampers. The conclusion from that work was that, at higher mass limits (HML) loadings, pavement and surfacing damage would be 20 to 30 percent greater than for a comparable freight task with a fleet equipped with dampers in good condition. That was for the Newell Highway, which has thicker pavements and surfacings than the majority of Queensland HML routes (Queensland Department of Main Roads, 2007). Nonetheless, to start an approximate exercise; the Main Roads maintenance budget in 2008 was AUD$253M (Queensland Department of Main Roads, 2008). The HML network in Queensland comprises approximately one-third of the road network. It is not unreasonable to propose that road asset damage is due to heavy vehicles compared with the damage from any other form of road traffic. On an, admittedly, broad-brush approximation, the Costanzi & Cebon figures of 20 to 30
percent translate into an approximate upper bound of AUD$26M per annum saved
on that portion of the network used at HML loads, were HML HVs to have dampers
maintained across the fleet.

Consider then, the roller bed testing described in Chapter 7, Section 7.5. For that
eexercise, 4 mm excitations were successful in deterministic detection of damper
health when axle-hop frequencies were induced into a HV wheel. Previous research
(Davis & Bunker, 2008e) has shown that HV wheel loads at highway speeds resonate
continuously at axle-hop frequencies in the range 10 Hz to 12 Hz. For the case of
worn shock absorbers on the roller bed, the 4 mm excitation at 12 Hz resulted in an
increased peak wheel-load of 70 kg more than for the case for new shock absorbers.
The axle-hop excitation for the case no shock absorbers resulted in a peak wheel load
of 205 kg more than for the case for new shock absorbers.

Let us estimate that, conservatively, road excitation has a 4 mm range (in reality, a
low estimate). Let us choose HVs with worn shock absorbers in condition-states
somewhere between tyre wear being apparent to totally ineffective. The wheel force
increases above static ranging from 70 to 205 kg for that range of condition-states
indicate that peak loadings on pavements and bridges would be between 70 kg (2.3
percent of statutory mass) and 205 kg (6.8 percent of statutory mass) higher than for
the case of well-maintained shock absorbers.

The fourth-power rule has been questioned earlier (section 2.4.2) by many
researchers. Nonetheless, applying this rule allows a comparison with generally
accepted civil engineering lore for the relationship between loadings on granular-
based pavements and pavement damage. Accordingly, the 2.3 percent and 6.8
percent range (above) indicates that instantaneous loading from worn, out-of
specification or defective shock absorbers on HVs would create instantaneous
loading damage increases in the range of 9.7 percent to 30.2 percent when compared
with well-maintained dampers. This range of increased damage due to worn
dampers is similar to the damage increase estimates from the approach developed by
Costanzi & Cebon (2005). It is, however, then able to be extrapolated across the
entire network as the testing in the roller bed testing described in Chapter 7, Section
7.5 was performed at maximum statutory mass wheel loads, not HML.
This exercise does not account for the twelfth-power rule for concrete pavements; that exponent would indicate a greater increase in damage, nor does it account for excitation due to road irregularities or axle-hop values of greater than 4 mm.

12.5 Application of OBM to heavy vehicle mass monitoring policy – Objective 4

12.5.1 On-board mass system tamper evidence and accuracy

Objective 4 was to determine the accuracy and potential for tamper-evidence of on-board mass (OBM) measurement systems for the purposes of an Australian regulatory regime. This Objective was met from the results of the testing of OBM systems detailed in Chapter 8 and Chapter 9. These chapters showed the results of characterising the available OBM systems in Australia and development of tamper indicators, including the tamper index (TIX). The following sections address how these developments may be used for Stage 2 of the Intelligent Access Project.

12.5.2 Tamper metrics

The Intelligent Access Programme (IAP), administered by Transport Certification Australia and mentioned in Section 1.5.1, has implemented satellite tracking of heavy vehicles (HVs) (Koniditsiotis, Taylor, Davis et al., 2004). One of the enabling platforms for this programme is the global navigation satellite system (GNSS), commonly referred to as the global positioning system4 (GPS). The success and robustness of the implementation of this tracking under Stage 1 of the IAP has been by careful specification of multiple quality indicators derived from the GPS output data. Stage 2 of the IAP will likewise have a strong emphasis on multiple quality indicators.

Jurisdictions are considering specifying OBM as part of their regulatory frameworks but require confidence regarding the performance of those systems (Australia Department of Transport and Regional Services, 2005a, 2005b; Karl et al., 2009).

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4Strictly speaking, GPS is a US system; GNSS is the generic term for all satellite navigation systems.
The details of such implementation are part of Stage 2 of the Intelligent Access Program outlined in Section 1.5. Accordingly, one tamper indicator will not be sufficient to prosecute, or even indicate with certainty, a tampering event. Where a combination of the scenarios developed in Chapter 9 were apparent, it would be highly likely that a tamper event had occurred. This approach will be presented to TCA. This will be in conjunction with arrangements for supply of the tamper algorithms either developed as part of this project or as part of the ongoing OBM Stage 2 project under IAP.

12.5.3 Tamper evident specifications

Jurisdictions and transport regulators require OBM solutions (Karl, 2007; Karl et al., 2009; Karl & Han, 2007) to an evidentiary level (i.e. accurate enough to be used as evidence in a prosecution). Previous experience with Stage 1 of the IAP was that more than one quality indicator was required for an evidentiary approach to GPS data (Koniditsiotis et al., 2004). That experience was also informed by the philosophy that, where technical monitoring rules were pragmatic and were not a great impost on HV operators, they were adopted. However, where technical solutions to monitoring were difficult, costly or complex to implement, “business rules” were adopted instead. Stage 1 of the IAP has a mix of business and technical rules for its governance framework.

The testing carried out under Transport Certification Australia’s on-board mass (OBM) accuracy and tamper-evidence programme provided a considerable amount of data. From the results shown in Chapter 9, dynamic data, amongst other indicators, will be required to derive robust tamper evidence for OBM systems for IAP Stage 2. Both frequency-domain and algorithmic approaches to dynamic data proved useful indicators of tamper events. It appears feasible that tamper detection may be implemented using logical rule sets and some simple algorithms, most of which are in the public domain. The example of the GPS data indicating movement without matching frequency data would be an example of a business rule; the tamper index (TIX) metric would be an example of a technical rule.

The issue of applying the TIX algorithm developed to detect tampering (Chapter 9)
to load cell data needs to be addressed. This is because HVs use load cells as well as APTs to determine OBM readings. The issue of tampering with load cells in the static environment has been shown to be of concern (Karl et al., 2009b). The TIX algorithm has not yet been validated against corrupted vs. typical load cell data. A modified OBM reference system will need to be used to measure dynamic signals from load cells, before and after they have been wedged, to explore further the validity of the tampering algorithms and any supporting technical and business rules developed.

12.5.4 Sampling frequency for OBM systems

The sampling frequency of the reference OBM unit used (Chapter 3, Section 3.4.2) was 41.6 Hz. The axle-hop frequencies of the HVs tested would very likely be in the range 10 to 15 Hz and not more than 15 Hz (Cebon, 1999; de Pont, 1997). The Nyquist sampling criterion (Shannon’s theorem, Section 2.3.3) indicates that the frequency of sampling needs to be at least twice that of the frequency of the signal of interest. Accordingly, the dynamic data recorded during the tamper vs. non-tamper events had a frequency range that was detectable up to approximately 20 Hz (i.e. half of 41.6 Hz). The derivation of the algorithms in Chapter 9 was predicated on data with that frequency range.

Assuming that OBM is implemented under Stage 2 of the IAP and that tamper-evident rules are applied similar to those developed in Chapter 9. Those rules were validated using data containing axle-hop frequencies. For those rules to be successful, the dynamic data from the OBM systems in the field will need to have a similar frequency range to that used for development of those rules in this chapter, i.e. more than twice any axle-hop frequency. Hence, the sampling frequency of the OBM units in the field under IAP Stage 2 will need to be at least 30 Hz, otherwise additional testing will need to be undertaken to validate other algorithms to those developed for this project. This conclusion has implications for OBM manufacturers in Australia.

Most OBM systems do not sample at frequencies capable of detecting axle-hop. Indeed, most do not sample at frequencies capable of detecting body bounce, which
would require a maximum sampling frequency of approximately 5 Hz (Davis, 2008). There are a number of reasons for this limitation within OBM units. One is to keep costs down, since the cost of an electronic chip is related to its sample rate; higher sample rate, higher cost. Another reason is that lower sample rates do not require much buffering of the final OBM readout to stop display jitter. Higher primary transducer data sample rates will require more buffering than used at present. Were the same buffering to be used as at present but with higher sample rates, OBM displays would jitter due to inherent noise and the digitisation indeterminacy within all electronic systems. Accordingly, as implemented at the time of writing, the majority of OBM manufacturers choose low sample rates for their equipment to keep buffer and sampling chip costs down (Davis, 2008). Both the use of dynamic signals from the OBM units for the tamper index (TIX) metric and business rules regarding movement of the vehicle compared with dynamic data will need to be considered as Transport Certification Australia moves into the specification phase for the OBM feasibility project.

12.5.5 Load cell tampering

Anecdotal data had it that wedges may be inserted under load cells resulting in the reading from the load cell indicating less than the actual applied load (Davis, 2008). The OBM test programme showed that view to be the case (Karl et al., 2009b).

The issue of using dynamic data to detect load cell tampering has yet to be determined. This was due to the inability of the load cells, where they were the primary OBM transducers, to be connected to both the dynamic reference system and the test OBM system simultaneously during the OBM test programme. Parallel verification of primary transducer signals via the reference OBM system vs. the test OBM system was the aim of the test programme as designed. This arrangement could not be implemented for the load cell cases, due to their inability to be connected to two measurement devices at once. This limitation, imposed by the physics of the load cell bridges, meant that load cell dynamic signals were not measured during the OBM testing. However, dynamic data have been recorded from load cells during other testing (Davis & Sack, 2005). An example of the frequency signature of load cell dynamic data, in the form of the FFT of data recorded from the
fifth wheel load cell of a semi-trailer during travel, is provided below in Figure 12.2. Rectangular FFT windowing was used (Brigham, 1988) to analyse 10,000 data points recorded at 41.6 Hz for this dynamic recording (Davis & Sack, 2005). Notably pitch at approximately 4 Hz and axle hop at approximately 9 Hz are prominent peaks in the spectrum. This preliminary result aligns with other work that indicted that the pitch mode frequency of a HV is usually in the 3 to 4 Hz range (Cole & Cebon, 1991; OECD, 1998) and the axle hop frequency is usually in the 10 to 15 Hz range (Cebon, 1999).

![FFT of signal](image)

**Figure 12.2. Frequency spectrum of vertical forces recorded by a load cell under a turntable.**

The load cell FFT signature in Figure 12.2 is similar to the FFT signatures shown from Figure 8.5 to Figure 8.7 in as much as it has irregular peaks and troughs in its spectrum. Accordingly, the issue of applying the tamper index (TIX) algorithm to load cell data appears to be worth pursuing. The TCA team has proposed a small test programme. It will use a modified OBM reference system to measure the dynamic signals from load cells, before and after they have been wedged, to explore further the validity of any tampering algorithms developed.
12.6 Summary of this chapter

12.6.1 Dynamic load sharing

That dynamic load sharing may be facilitated by larger longitudinal air lines on air-suspended heavy vehicles (HVs) is indicated strongly by the results in Chapter 10 and discussed further in this chapter. What is not so clear is the potential for improvements to HV suspension metrics from further increases in dynamic load sharing. The standard dampers fitted to the semi-trailer axle tested, for example, appeared to act against further improvements to HV suspension indicators for larger longitudinal air lines. Conversely, were HV suspension designs to have a holistic revision which incorporated the increased flow of air between sequential air springs with a concomitant variation in damper characteristics, even greater reductions in peak loadings may be possible than those found during this project. This area needs to be explored by HV designers and the HV manufacturing industry.

12.6.2 In-service HV testing

Chapter 11 outlined some issues and concepts that regulators will need to address to implement in-service HV suspension testing. This chapter, supported by findings from Chapter 7, built on those concepts to outline practical methodologies that need to be considered by the National Transport Commission to implement such testing as required under the current Queensland and NSW Government funding arrangements (Australia Department of Transport and Regional Services, 2005a, 2005b).

12.6.3 Community cost of poor HV suspension health

Two indicative exercises on increased instantaneous pavement loadings have been detailed in Section 12.4.2. The first, applicable to HML routes, indicated a potential saving of approximately AUD$26M/annum. The second applied the wheel force values from the roller bed testing described in Chapter 7, Section 7.5. This exercise indicated that up to approximately 30 percent of the maintenance portion of the Department of Transport and Main Roads' budget, or approximately $AUD75M/annum could be gained from the simple expedient of road transport
operators testing their HVs and replacing worn shock absorbers. This figure is over and above damage due to overloading; a comment made for future reference but verging out-of-scope for this project.

It is for noting that this conclusion does not address the community cost of reduction in HV safety arising from defective, out-of-specification or worn dampers, nor does it address any Local Government infrastructure costs.

12.6.4 On-board mass systems on heavy vehicles – sampling rates and synergy with other requirements

The testing of on-board mass (OBM) systems for HVs under this project indicated certain accuracy and evidentiary levels for OBM systems available in Australia. The results of these tests (Karl et al., 2009b), to be considered for use in Stage 2 of the Intelligent Access Project, will be presented to road transport regulators in Australia. The tamper algorithms developed as part of this project or alternatives created under other, on-going development arrangements with Transport and Main Roads, will be presented to Transport Certification Australia’s OBM Stage 2 project under IAP. The implications for the OBM industry in Australia will be that higher sampling frequencies than those currently deployed at the primary transducer level will be required to implement a robust and tamper-evident platform for implementing OBM monitoring using the algorithms developed during this project or other methods based on analysis of dynamic data from OBM systems.

Higher sample rates would also facilitate the recording of air spring transients should the HV traverse a discontinuity. This could facilitate the derivation of suspension damping characteristics within an in-service HV suspension testing regime. Some OBM manufacturers already use an accelerometer in an OBM unit to detect the angle of the HV stance where it is not on level ground. Recording the accelerometer data during a similar traverse over a discontinuity to that postulated above could be used to determine suspension health for a different approach, but similar outcome, to implementation of in-service HV suspension testing.
12.7 Conclusions from this chapter

The following list condenses the future industrial research directions from this project:

1. alter current damping characteristics of commercial shock absorbers, in conjunction with larger longitudinal air lines, to improve dynamic load sharing;

2. investigate improvements to suspension metrics from alterations in 1, above;

3. develop a cheap and simple standardised impulse test for air-suspended HVs with due consideration to the variability of tyre parameters influencing the results of the tests;

4. measure the error between test data from a standardised impulse test and VSB 11-certified data for the air suspensions available in Australia. Circulate to road authorities and implement in-service suspension testing using an impulse into air suspended HVs;

5. develop a standardised roller bed test for air-suspended HVs;

6. measure the error between test data from a standardised roller bed test and VSB 11-certified data for the air suspensions available in Australia. Circulate to road authorities and implement in-service suspension testing using a roller bed test on in-service air suspended HVs;

7. review the advice to transport operators that tyre wear is an indicator of worn dampers; this point is too late in the maintenance cycle to keep the dampers within specification;

8. specify a minimum of 30 Hz as the sample rate for any on-board mass system for HVs (suggest 40 Hz to provide a margin for future applications requiring analysis of HV frequencies above 15 Hz);

9. use OBM systems to measure suspension health after application of
a standardised impulse under 3, above; and

10. use a modified OBM system to measure dynamic signals from load cells, before and after tampering, to explore further the validity of tampering algorithms.

“Opportunity is missed by most people because it is dressed in overalls and looks like work.” - Thomas Edison.
13 Contribution to knowledge – theory and future work

13.1 About this chapter

Avenues of research and other work have been discovered as the joint project entitled *Heavy vehicle suspensions – testing and analysis* has progressed. Those central to the Objectives (and some others) have been included in this thesis. Other areas of investigation have led to the conclusion that, were they to be pursued, they would have ventured out-of-scope. Nonetheless, it is important to provide some directions to future researchers where this has happened. This chapter serves that purpose.

13.2 Application of heavy vehicle in-service suspension testing

13.2.1 Future research into the “pipe test” as a low-cost in-service suspension test

To calibrate the “pipe test” against Vehicle Standards Bulletin (VSB) 11, research will be needed to determine the characteristics of available suspensions. VSB 11-compliant suspension models available in Australia in “as new condition” would need to be matched and documented against some form of standardised in-service or field impulse test. The margins between the field test results and the VSB 11 characteristics could be noted and distributed, as agreed by regulators. Such a programme would be the subject of further work in the regulatory field. The results of some of the background work from this project (Davis, Kel, & Sack, 2007) as well as the body of work on the “pipe test” herein, as a “proof-of-concept” (and allowing for the finite resources of this project), should be applied to the issue of testing the health of suspensions on HVs with steel suspensions by the use of attaching an accelerometer to their chassis. Other areas of research would be into the alteration of measured suspension parameters by variations in tyre characteristics such as inflation pressure, tyre construction, tyre configuration (dual/single), size, tread depth and case design as these will influence the phenomenon of tyre enveloping and its effect on the forcing function provided by the pipe test. It is also noted that variations in
tyre characteristics and tyre enveloping influencing derived suspension parameters for VSB 11-approved forcing functions are not addressed in VSB 11; a potential additional avenue for future research.

Chapter 6, Section 6.3 detailed the results for the testing of the semi-trailer and the coach when subjected to the “pipe test” as a forcing function. The traverse speeds were too great to elicit a second-order (Section 4.3.1) response that was analysable for damping ratio. The coach axle responses were analysed in other work leading to this project (Davis et al., 2007). The tag axle and the drive axle were not able to be distinguished from examination of the air spring data. This was likely due, as mentioned in Section 6.4.1, to close coupling of the tag axle and the drive axle within the same group. Such groups with dissimilar axles make up a small minority of the heavy vehicle (HV) fleet. Any low-cost test programme developed under the National Transport Commission in-service HV suspension testing project (Section 1.4) need not be stalled based on the perfect being the enemy of the good. Nonetheless, further research into low-cost testing for dissimilar axles will need to be considered given the anomalous results for the coach rear axle group in the testing described in Chapter 6.

13.2.2 Future research into in-service HV testing – roller bed
The roller bed testing results detailed in Chapter 7, Section 7.5 indicated that dynamic oscillatory excitation to a HV wheel could detect damper health from wheel forces. Given that the numbers of HV suspension models available in Australia certified to VSB 11 are finite (Australia Department of Transport and Regional Services, 2004b), all available models could be subject to roller bed testing similar to that undertaken in Chapter 7, Section 7.5. The results for these could be correlated, documented and referenced against “as new” certified VSB 11 values and then known values of degradation of suspension dampers and corresponding decay in VSB 11 parameters, particularly in the critically important damping ratio (Section 11.4.1) plotted over time.
13.3 Future research into dynamic load sharing

13.3.1 Future research into improvements dynamic load sharing by use of larger longitudinal air lines

It is venturing outside the scope of the Heavy vehicle suspensions – testing and analysis project to explore the fluid mechanics of airflow between air springs. However, a few points may be central to future research in this area:

- the comparison of wheel and suspension forces for standard vs. larger longitudinal air lines between air springs has been documented (Davis, 2006a, 2007; Davis & Bunker, 2008d; Davis & Bunker, 2008e; Davis & Kel, 2007);

- the nature of improved air flow between air springs using larger longitudinal air lines has been addressed (Li & McLean, 2003a, 2003b; McLean, Lambert, & Haire, 2001) with research continuing; and

- the Li and McLean (2003b) study postulated an improvement of 22 times greater air flow between air springs for 20 mm diameter air pipes vs. 4.15 mm diameter pipes.

The frequency spectra of the larger longitudinal air lines used in the "Haire suspension system" did not move radically into axle-hop (Davis & Bunker, 2008e) during “live-drive” testing. That is, the modification did not result in instability or increase in axle-hop during actual testing. This was borne out by analysis of the data for this project (Davis & Bunker, 2008e). Mapping this result to the model used in Chapter 12, Section 12.2.4, leads to the conclusion that the longitudinal air line diameter modification did not result in a corresponding model load sharing factor (LSF) any greater than an order-of-magnitude of approximately 1 x 10^{-6} (i.e. an order-of-magnitude below 1 x 10^{-5}).

If the smaller air lines have flows 22 times less than 20 mm air lines (the size of the connectors used in the “Haire suspension system”), this concept indicates that their LSFs are in the order of magnitude of sub-1 x10^{-7}. Further, given that body bounce for the test case where the "Haire suspension system" was fitted did not result in
vehicle instability, it is reasonable to infer that this system extracts the benefits of speedier transfer of air between successive air springs without the potential for instability arising from low-frequency runaway. The issue of suspension dampers needs to be addressed in light of the findings of Chapter 12, Section 12.2.4 and other work within this project (Davis & Bunker, 2009b). Those approaches indicated that increased dynamic load sharing is being hampered by the characteristics of the standard dampers on the axles of (say) the semi-trailer tested. This since altering the damper characteristics in the suspension model to those from another, known, suspension resulted in reduced transmission of axle-hop, no instability at high speed and extremely low restriction in air flow between air springs.

The industry behind the supply of HVs should consider more research into providing better load sharing than found currently in most HV suspensions. Societal benefits accruing from reductions in peak loadings leading to reduced road damage, reduced payload damage (especially for fragile goods), less fatigued drivers and passengers, and greater life from heavy vehicle chassis, suspension and coachwork components may be the result.

13.3.2 Future research into load sharing metrics

Any improvement to a HV’s dynamic load coefficient (DLC) was shown mathematically to be mutually exclusive to an improvement to its load sharing coefficient (LSC) in Section 2.5. The results shown in Sections 10.3.4 and 10.3.5 indicated that this might be the case for the two multi-axle HVs tested. With the exception of the tag axle wheel forces, the LSC did not improve when the DLC did and vice versa for the two cases tested. Further research should be undertaken on this issue since both the LSC and the DLC have been used extensively over the past few decades as measures of HV suspension effectiveness (Davis & Bunker, 2007).
13.4 Wheel forces within pavement damage models

Three HVs were tested as described in Chapter 3, Section 3.2. The data from those tests were used in Chapter 7, Section 7.3, Section 7.4 to show derived wheel forces against a “novel roughness” measure and against test speed. When checked for statistical significance, mean wheels forces varied per side for the bus and the semi-trailer. This appeared to be the result of the effect of the cross-fall of the road shifting the HVs’ centres-of-gravity to the left or the right and the ability of the heavy vehicle (HV) to compensate for this action.

The bus wheel force standard deviations, mean wheel forces and peak wheel forces did not correlate to increasing “novel roughness” values. Neither did the coach and semi-trailer wheel force standard deviations and mean wheel forces correlate with increasing “novel roughness” values. However, peak wheel forces from the coach and the semi-trailer did correlate to increasing values in “novel roughness”. Left/right variations were evident in peak and mean wheel forces derived from the bus over the range of “novel roughness” values. Only mean wheel forces varied per side for the coach and the semi-trailer in the same variable-space.

Indicatively, the semi-trailer exhibited variation per side in mean wheel forces both per test speed and per “novel roughness” value. Peak forces for the bus and the coach did not. By using data per test speed instead of roughness, different results were obtained for the derived parameters. These results were from the same data sets. Care needs to be exercised, therefore, in ensuring that results from one method are cross-checked against results from another method.

As noted in Chapter 2, pavement damage models use a “power law” damage exponent to attempt to account for the variation in empirical pavement life correlated to axle passes. A single pass from an axle that does not bounce causes rutting and has a damage exponent tending to 1.0 (de Pont & Steven, 1999). To “build in” an allowance for dynamic wheel forces, damage exponents ranging from 1.0 to 12 are used (Austroads, 1992; Pidwerbesky, 1989). These approaches to pavement damage models need to be considered in terms of these results. In particular, the left/right variation apparent in mean wheel forces and the highly-dependent relationship between “novel roughness” values and peak wheel forces needs to be investigated.
further by pavement technologists, geotechnical engineers, and other domain experts. This particularly for the pavement peak loading issue and left/right variations. This latter particularly indicating that the pavement under the outer wheel path should have a different design standard from that of the inner wheel path pavement.

Augmentation of pavement models needs to account for dynamic wheel loading effects now that the dynamic data for HV wheel forces from this project have been documented. This augmentation needs to take into account a more complex set of considerations than simply the peak forces or the static loads. Chapter 2 detailed some pavement damage models that used static load values. The results in Chapter 7 indicated that the mean wheel forces might be indicated by static loads. However, the on-board mass portion of this project found that mean wheel forces are not equal to static wheel forces (Karl et al., 2009b). Further, neither roughness values nor peak wheel forces are included in any Australian pavement design model (Austroads, 1992; Main Roads Western Australia, 2005; Moffatt, 2008). The results in Chapter 7 indicated that the correlation of wheel forces to roughness needs to be explored further, with this concept being noted in other research (OECD, 1992, 1998). Further, the adherence to HV suspension dynamic metrics containing only standard deviations (Eisenmann, 1975; Sweatman, 1983) needs to be re-examined since peak wheel forces of the most common line-haul HV in Australia varied proportional to “novel roughness” in a statistically significant manner whereas neither the wheel force standard deviations nor the mean wheel forces so varied.

### 13.5 Suspension wavelength and the HV fleet

Chapter 7, Section 7.4 showed indicative fast Fourier transform (FFT) plots of peak wheel forces. In these, the contribution that body bounce forces made to pavement forces was shown to be approximately equal in magnitude to that of axle-hop forces. Accordingly, two sets of suspension wavelengths need to be examined as they both contribute approximately equally to instantaneous pavement forces from HV wheels. The results shown in Chapter 7, Section 7.4 indicated that wheel forces from body bounce at highway speeds would be repeated at approximately 15 to 28 m spacings\(^5\).

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\(^5\) 27.7 m for 10 Hz and 100 km/h
Axle-hop repetitive forces will occur at approximately 2 to 2.5 m intervals, depending on speed of travel. These effects are termed “spatial repetition” and have been well documented (Cole & Cebon, 1992; Cole et al., 1996; Collop et al., 1996; Collop et al., 1994; Jacob, 1996; LeBlanc & Woodrooffe, 1995).

Should a particular suspension have its axle-hop frequency (i.e. axle-hop force repetition) as a multiple of its body bounce frequency, an approximate doubling of the instantaneous pavement force will occur where the two coincide at a common wavelength node. Pavement distress will then be concentrated as a patch at that point with damage proportional to a suitable exponent.

The Australian road transport network has been opened up to increasing numbers of HML vehicles and lengths of road declared HML routes. With this increase, there has been a concomitant number of HVs fitted with RFS. Accordingly, the suspension characteristics of the HV fleet have become more homogenous than in the past. This then provides for a continuing homogenisation of HV suspension characteristics, particularly with respect to body bounce frequencies, namely:

- more highly correlated wheel forces from the heavy vehicle fleet;
- pavement distress likely concentrated in patches distributed longitudinally; and
- that concentration to predominate at intervals of approximately 15 to 30 m for highway segments, particularly those with laden semi-trailer traffic.

That spatial repetition would need to be addressed eventually had been foreseen (LeBlanc, 1995). The results from this project agree with that proposition. There is a need for research into the effects of mandated uniform HV suspension characteristics and that is now more urgent than when first mooted by researchers such as LeBlanc (1995) about 15 years ago. Where the body bounce wavelength of a HV suspension creates a maximum instantaneous force at a point on a pavement and the axle-hop forces are coincidental at that point, a potential doubling of the instantaneous wheel forces will likely occur, especially at highway speeds. Hence distress points at 25 m intervals would be expected, coincident with distress points at
2.5 m intervals caused by HVs with homogeneous suspensions all bouncing at or below 2 Hz.

The measures of standard deviation, mean and peak dynamic wheel forces combine to show a picture of pavement forces in the real world. Instantaneous values of these forces can be up to double those of the static force for which the pavement was designed. Further, pavement “novel roughness” provides a reasonable indicator of maximum wheel forces and their range. Wheel force maxima and dynamic range for three HVs have been determined and documented for this project.

Dynamic pavement forces created by heavy vehicles have been modelled for some decades. The current pavement models that use a number of quasi-static passes of a HV axle at a theoretical loading to determine pavement life do not always account for peak dynamic forces or the variation of those forces around the mean. It is the peak (within the upper tail) of wheel loads that do the damage, however, whether or not an empirical “power law” is applied. The quasi-static application of pavement loads from HV wheel forces in traditional models for pavement design needs to be reviewed to incorporate the dynamic data from the research presented here and by others. This especially since the regulatory system and the pavement design process both define static wheel forces as the determining factor for HV impacts. Clearly, this measure has little value when the dynamic pavement loads from HVs are understood.

13.6 Future directions of research into on-board mass monitoring of heavy vehicles

13.6.1 Future OBM research

Some significant areas of potential tampering have been addressed in the test programme; development of both technical and business options to detect these events will need to be addressed for implementation of regulatory OBM system application to Australian HVs. Agreed best practice guidelines and procedures for installation, calibration, operation and maintenance will be developed as TCA moves into the regulatory phase for OBM implementation. The test programme investigated the potential use of additional intelligence derived from analysis of
dynamic OBM data as validation against data from the static data recorded by those OBM systems.

Vulnerability of mechanical and pneumatic systems to degradation, drift and tampering will need to be further investigated in order to achieve OBM implementation in the IAP environment but these issues verge outside the scope of this thesis.

13.7 Conclusions from this chapter

The following points condense the theoretical research directions identified during this project:

1. further development of suspension models to better analyse the process of air transfer between air springs on HVs and improve load sharing and dynamic performance;

2. development of pavement damage models incorporating peak wheel forces and left/right variation in wheel forces;

3. development, refinement and adoption of pavement damage models incorporating spatial repetition adopting the work herein and that of others (Cebon, 1987; Cole & Cebon, 1992; Cole et al., 1996; Collop, Cebon, & Cole, 1996; Collop et al., 1994; LeBlanc & Woodrooffe, 1995); and

4. investigation of the susceptibility of HV on-board mass telematic components to degradation, drift and tampering, and the design parameters of OBM systems to give greater flexibility for future applications.

“A prophet is not without honour, save in his own country, and in his own house.” - Matthew xiii, 57.
14 Conclusions

14.1 Introduction

This research had a number of objectives. They concerned development of low-cost in-service heavy vehicle (HV) suspension testing, the effects of large longitudinal air lines and characteristics of on-board mass (OBM) systems for HVs. Whilst those objectives may appear to be disparate, they were all centred on measurement of HV suspension behaviours with the ultimate purpose of reducing the damage and repair costs to Australian road network assets. This chapter presents the main conclusions of the study in relation to the Objectives presented in Chapter 1 and as met in the subsequent chapters. The appropriate areas of research and the results therefrom will be presented to the various project management groups where their role is applicable to research carried out in undertaking this project and writing this thesis.

Chapter 2 presented a summary of the HV suspension metrics found in the literature review for this project (Davis & Bunker, 2007) as well as literature released since that search.

Chapter 3 presented the details of the experimental procedures used to gather data for the project as well as the rationale behind the choices of instrumentation and methodologies used.

Chapter 4 developed some abstract models of HV suspensions. These were used later in Chapters 5 and 6 with respect to in-service suspension testing. The model for the semi-trailer was also used later in Chapter 12 to indicate the potential for, and future directions of research into, implementation of large longitudinal air lines for HVs.

Chapter 7 analysed wheel forces from the “live-drive” data as described in Chapter 3 as well as the roller bed testing. The wheel force data were developed into “novel roughness” measures derived from the wheels of each test HV and used in Chapter 10 for determination of dynamic load sharing. The wheel force data were also analysed and presented in Chapter 7 for left/right variations and for different speeds. This led on to some conclusions regarding suspension and wheel force frequencies, their transmission to the pavement and repetitive wheel loads in the spatial domain.
The data from the roller-bed testing, gathered as described in Chapter 3, were analysed as presented in Chapter 7 to inform one part of the in-service suspension testing portion of the project that was taken up in Chapters 11, 12 and 13.

Chapters 8 and 9 were concerned with the OBM accuracy testing and tamper evidence portions of the project. These areas documented the first public results of a programme to test all available OBM systems in a country anywhere in the world. These results were not stand-alone, however. They also served to inform portions of the discussion on in-service suspension testing in Chapters 11 and 12.

Chapter 11 was concerned with the technical and policy issues regarding in-service suspension testing for HVs as informed by this project.

Chapters 12 and 13 together outlined some future directions for theoretical, regulatory and industrial research that were identified from the work over the course of the project but which, if pursued, would have taken those portions of the project out of scope.

14.2 Main conclusions

14.2.1 General

The project objectives were presented in Chapter 1, Section 1.7. These are repeated below with an accompanying discussion on the results from each. The applicable chapters of the thesis are cross referenced as sub-headings under each Objective and within that discussion. Reference to Chapter 1, Figure 1.4, shows the interconnection of the chapters and provides guidance on how the subject of each chapter is related to the other chapter subjects. The cross references to the chapters in the following sections are not sequential due to the fundamental interconnectedness of the subject matter and the parallel development of different models as the project proceeded.
14.3 Objective 1

14.3.1 Dynamic load sharing 1

“Develop at least one dynamic load sharing measure that can be applied to the axle group, consecutive axles or to consecutive wheels and for more than two axles or wheels.”

Chapter 2, Sections 2.3.5 and 2.3.8 documented load sharing suspension metrics applicable to this thesis, their merits and drawbacks discovered during the literature review. Chapter 2, Section 2.5 explored the relationship that the load sharing coefficient (LSC) has with the dynamic load coefficient (DLC) and determined that they were inversely proportional to each other. Chapter 3, Section 3.2.2 detailed the experimental design and implementation of the testing of three heavy vehicles. An interesting approach to determining wheel forces was developed in Chapter 7, Section 7.3.1 which led the development of an independent variable, “novel roughness” as measured at the wheels of each of the test vehicles.

The dynamic load sharing coefficient (DLSC), as developed by de Pont (de Pont, 1997), was derived from the data collected from the testing outlined in Chapter 3, Section 3.2.2. These results were used in Chapter 10 to show that the DLSC was improved at the air springs of the two multi-axle HVs, in a statistically significant manner, by the use of large longitudinal air lines (i.e. the test case). This improvement was consistent across all “novel roughness” bands. The wheel force DLSC was not altered consistently across all “novel roughness” bands for the test cases, however. That the DLSC was improved at the air springs was shown to be statistically significant for all “novel roughness” bands. The DLSC was improved for the test cases of the larger longitudinal air lines at the coach wheels only for low “novel roughness” (low air line flow events). These were the outcomes of application of the DLSC as a mature methodology.

An improved method of determining dynamic load sharing was then applied to the wheel forces and to the air spring forces. This method comprised cross correlation of forces between wheels and between air springs to determine whether dynamic load sharing was occurring and, if so, whether the larger longitudinal air lines had made a statistically significant improvement. The DLSC results indicated that dynamic load
sharing was occurring at the air springs for all test cases but at the wheels of the coach for low air line flow events only. To validate the improved method, the correlation coefficients from the air springs and the wheel forces were plotted against HV test speeds. The results indicated that dynamic load sharing was improved at the air springs, but only at low “novel roughnesses” (low air flow events) at the coach wheels, by their use. This was the same result as for the application of DLSC and therefore validated the improved methodology using correlation of forces.

Determining the extent of dynamic load sharing using the correlation coefficient between two elements of a HV was the first time that this method had been used on a HV for a test case vs. a control case for two different sized air lines. This was in accordance, however, with the scientific method of altering one variable at a time (Mill, 1872). It is submitted that Objective 1 was met.

14.4  Objective 2

14.4.1  Dynamic load sharing 2

“Determine if large air lines on air sprung HVs make a difference to wheel forces and axle-to-body forces and, if so, determine the quantum of such alterations from the use of larger longitudinal air lines.”

**Chapter 3, Section 3.2.2** detailed the gathering of data during on-road testing of three instrumented HVs. These HVs were configured such that small (4 mm to 12 mm diameter), “industry-standard” longitudinal air lines could be interchanged with larger (50 mm diameter proper with 20 mm diameter connectors) longitudinal air lines: the “Haire suspension system”. This allowed comparison of the forces at the air spring and the wheels for the control case of small air lines vs. the use of larger longitudinal air lines. **Chapter 10** detailed the results of the analysis of that data using some mature suspension metrics detailed in **Chapter 2**. This approach concluded that there were statistically significant improvements to some suspension metrics with the use of larger longitudinal air lines, but not in a uniform manner. Of note were improvements to suspension metrics involving peak dynamic forces ranging from below the error margin to approximately 24 percent. However, this
range of improvement was not consistent across all air springs, wheels, roughness bands or speeds.

The suspension metric that was improved consistently across all air springs for all roughness bands and all test speeds was the improvement in dynamic load sharing from the use of larger longitudinal air lines. The improvement in dynamic load sharing coefficient (DLSC) in all cases varied in a statistically significant manner from approximately 30 percent to 80 percent.

It is submitted that Objective 2 was therefore met.

Software models of the suspensions of the three HVs tested were developed in Chapter 4, Section 4.3.7 and Chapter 5, Sections 5.5.4, 5.6.4 and 5.7.5. The software model for the semi-trailer suspension was used in Chapter 11 to inform future industrial research into the deployment of larger longitudinal air lines on HVs. Some exploratory excursions into varying damper characteristics with increasing air transfer between air springs were undertaken in Chapter 11. These led to the preliminary conclusion that future research into dynamic load sharing would need to alter currently available shock absorber damping characteristics to accommodate potential further gains in dynamic load sharing.

14.5 Objective 3
14.5.1 In-service HV suspension testing

“Explore low-cost HV suspension test methods. Evaluate if low-cost HV suspension test methods can be made equivalent to VSB 11 outcomes for body bounce and damping ratio. Evaluate the validity, via “proof-of-concept” of the “pipe test”. Develop a modified roller-brake tester to impart resonant forces into a HV suspension and, in part, validate previous work on wheel forces from suspensions in good condition vs. those equipped with poor/worn shock absorbers. Determine, via “proof-of-concept”, that a modified roller-brake tester may be used to detect a worn HV suspension from one within specification. This latter to inform the NTC project.”
Two low-cost methods that may prove useful in developing in-service tests for HV suspensions were described in Chapter 3, Sections 3.2.3 and 3.3. The results of these tests were analysed in Chapter 6 and Chapter 7, Section 7.5. The results indicated that, as a “proof-of-concept”, low-cost HV suspension testing was possible. The results from the test programme were, at worst, approximately 16 percent in error from those for the same vehicles tested to Australian specification for “road friendly” HV suspensions, VSB 11 (Australia Department of Transport and Regional Services, 2004a). The roller bed testing showed that varying degrees of damper degradation were detectable from the wheel forces at the roller bearings. These results have been communicated to the in-service HV suspension testing project staff at the National Transport Commission. It is submitted that Objective 3 was therefore met.

The results in Chapter 6 and Chapter 7, Section 7.5 informed a discussion on the implications of in-service HV suspension testing in Chapter 11, Chapter 12, Section 12.3 and Chapter 13, Section 13.2.

The results of the quasi-static testing described in Chapter 3, Section 3.2.3 were used to develop software models of the suspensions of the three HVs tested in Chapter 4, Section 4.3.7 and Chapter 5, Sections 5.5.4, 5.6.4 and 5.7.5. The model for the semi-trailer was used in Chapter 11 to inform the future directions of dynamic load sharing in HVs under Objective 2.

### 14.6 Objective 4

#### 14.6.1 On-board mass monitoring of HVs – search for accuracy and tamper-evidence

“Determine the accuracy of currently-available OBM systems. Examine the accuracy of OBM readings vs. weighbridge readings. Validate the use of dynamic data from OBM systems to detect tamper events.”

Chapter 1, Section 1.5 outlined the rationale for on-board mass (OBM) systems on HVs to be deployed as part of Stage 2 of the Intelligent Access Program (IAP). Chapter 3, Section 3.4 described the data gathering process during the OBM system
test programme. **Chapter 8** presented the analysis of accuracy and linearity of the OBM systems tested. **Chapter 9** described the development of a tamper indicator and other OBM metrics that could be used by jurisdictions to determine tamper events.

The results in **Chapter 8** indicated that 95 percent of contemporary OBM systems available in Australia are accurate to +/- 500 kg. When three anomalous results out of 111 scatter plots of OBM linearity were removed, a total variation of 0.5 percent in OBM linearity from individual system tests was the result.

Typical deliberate tampering event data from the OBM system testing were analysed in **Chapter 9**. A set of tamper indicators was developed in that chapter, including testing and validation of an algorithm that would indicate a tamper event. This algorithm, in combination with other logical operations that could be applied to HVs in the field, was proposed as a way forward for road transport regulators and jurisdictions in the implementation of OBM systems for HVs.

The application of OBM systems to in-service suspension testing as a crossover result between objectives from this project (i.e. applicability of OBM to Objective 3) was broached in **Chapter 11, Section 11.3.3**. Future directions in OBM research and industrial applications were developed in **Chapter 12, Section 12.5** and **Chapter 13, Section 13.6**.

The results in **Chapter 8** and **Chapter 9** have been presented to jurisdictions for their judgement on the applicability of OBM to Stage 2 of the IAP (Karl et al., 2009b). Further, jurisdictions may also now consider other applications of OBM as part of their regulatory frameworks. It is submitted that Objective 4 was met.
“It has always been the fate of those who have made new discoveries to be disesteemed and slightly spoken of by such as either have had no true relish and value for the things themselves that are discovered, or have had some prejudice against the persons by whom the discoveries were made. It would be vain, therefore, and unreasonable in me to expect to escape the censure of all, or to hope for better treatment than far worthier persons have met with before me.

But this satisfaction I am sure of having, that the things themselves in the discovery of which I have been employed are most worthy of our diligent search and inquiry, being the various and wonderful works of God in different parts of the world; and, however unfit a person I may be in other respects to have undertaken this task, yet, at least, I have given a faithful account, and have found some things undiscovered by any before, and which may at least be some assistance and direction to better qualified persons who shall come after me.”

- William Dampier.
Appendix 1 – Instrumentation and calibration of three HVs to measure wheel force data

Introduction

The following Appendix details the methodology used to derive the values of the unsprung masses and calibration of axle-mounted strain gauges. The strain gauge calibration graphs include the correlation of strain gauge readings to wheel forces for the wheel and the hub jacked off the ground and/or tare and/or full load readings for the various test vehicles.

Masses outboard of the strain gauges

To find the total dynamic wheel force ($F_{\text{wheel}}$), the two terms $F_{\text{shear}}$ and $ma$ (Equation 3.1) were derived for the instantaneous data recorded during the tests of the three HVs in Section 3.2. The unsprung mass $m$ outboard of the strain gauges contributed the $m$ coefficient of $a$ in the $ma$ term of Equation 3.1. The value of $m$ in Equation 3.1, the unsprung mass outboard of the strain gauges, was found as described below.

In order to determine the masses of the coach and bus axles outboard of the strain gauges, a bent tag axle and a cracked drive axle housing were procured and cut through completely at the strain gauge mounting points. These portions of axle were then weighed on certified scales (Figure A1.1, Figure A1.3, Figure A1.4, and Figure A1.5). The tag axle was not identical to the one installed on the coach but it was similar enough to provide a valid mass for this portion of the unsprung mass value.

Unfortunately, a drive axle half-shaft was not available for destruction but a sound spare was made available on loan. It was weighed and measured. Its mass outside the strain gauge mounting points was able to be calculated owing to the uniformity of its shape and by using a standard value for the density of steel (Figure A1.2). The resultant measurement was added to the measured masses of the wheel/tyre/hub assembly on the HV tested, the measured mass of the requisite portion of the axle housing and to the manufacturer’s specified masses (Mack-Volvo, 2007a) for the other components for the relevant axle/s. This process yielded the value for $m$ (Table
A1.1) in Equation 3.1 that was applied to the derivation of wheel forces for each HV wheel under test, speed and test case. These values were also used for calibrating the strain gauge readings as discussed shortly. Signals representing a value of $a$ from the accelerometers allowed completion of the wheel force data for each axle-end of interest (de Pont, 1997).

Figure A1.1. Weighing the half-shaft.

Figure A1.2. Calculating the half-shaft mass outboard of the strain gauges.

Figure A1.3. Weighing the drive axle housing mass outboard of the strain gauges. This photo shows the bus axle portion.
Figure A1.4. Weighing the drive axle housing mass outboard of the strain gauges. This photo shows the coach axle portion.

Figure A1.5. Weighing the mass of the tag axle portion outboard of the strain gauges.
## Axle mass data

Table A1.1. Unsprung mass outboard of the strain gauges for the test vehicles.

<table>
<thead>
<tr>
<th>Component</th>
<th>Coach drive axle/wheels</th>
<th>Coach tag axle/wheels</th>
<th>School bus axle/wheels</th>
<th>Semi-trailer axle/wheels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub, brakes, bearings, nuts etc</td>
<td>192.3kg (Mack-Volvo, 2007a)</td>
<td>140.2 kg (Mack-Volvo, 2007a)</td>
<td>187.3 kg (Mack-Volvo, 2007a)</td>
<td>149.4 kg (Giacomini, 2007)</td>
</tr>
<tr>
<td>Wheels</td>
<td>160.0 kg (measured on TI scales)</td>
<td>83.0 kg (measured on TI scales)</td>
<td>180.0 kg (measured on TI scales)</td>
<td>180.0 kg (measured on TI scales)</td>
</tr>
<tr>
<td>Housing/axle portion</td>
<td>30.8 kg (Figure A1.4)</td>
<td>5.2 kg (Figure A1.5)</td>
<td>32.8 kg (Figure A1.3)</td>
<td>7.1 kg (Giacomini, 2007)</td>
</tr>
<tr>
<td>Half shaft</td>
<td>10.7 kg (Figure A1.1 and Figure A1.2)</td>
<td>n/a</td>
<td>11.4 kg (Figure A1.1 and Figure A1.2)</td>
<td>n/a</td>
</tr>
<tr>
<td>Total</td>
<td>393.8 kg</td>
<td>228.4kg</td>
<td>411.5kg</td>
<td>336.5kg</td>
</tr>
</tbody>
</table>
Calibrating wheel forces vs. axle shear

In order to determine $F_{\text{shear}}$ in Equation 3.1, the relationship between the strain gauge readings and the forces in the axle resulting from wheel forces needed to be determined. Accordingly, the $F_{\text{shear}}$ component, measured via strain gauge readings, was calibrated from its relationship with the wheel force (Woodrooffe et al., 1986) as follows, with commentary regarding the rationale included in each point:

- the static wheel force exerted by each wheel of the axle group under test on the test vehicle was measured as a mass from calibrated scales usually used by transport inspectors for enforcement purposes. This static mass value was recorded with the test vehicle on the flat, level floor of an industrial shed. This was done in conjunction with the calibration of the on-board telemetry system, for efficiency, after it was installed;

- when static weighing using the scales under each wheel of interest was undertaken, the static value of $F_{\text{wheel}}$ measured on the scales contained two force components. These were a static $F_{\text{shear}}$ component at the strain gauge mounting point inboard of the strain gauge and acting through the spring; and a static $ma$ force component. The former due to the mass of the chassis and suspension components, etc. transmitted via the axle to the wheel; and the latter a force component due to gravity acting on the unsprung mass outboard of the strain gauge;

- the measured static wheel force had a constant offset compared with the strain gauge readings. This margin was the difference between $F_{\text{shear}}$ and the total force ($F_{\text{shear}} + ma$); i.e. the latter of the two components of the static value of $F_{\text{wheel}}$ measured on the scales, remembering that the strain gauges measured shear proportional only to the forces inboard of their mounting point;

- to find the constant offset due to the static $ma$ force component, the chassis of the test vehicles were jacked so that the wheel force registered as close to zero as possible (+5/-0 kg) on the portable
scales. Figure A1.6 shows the method of jacking the chassis so that
the wheel forces could be read off the scales when balanced to zero.
With the static wheel force registering zero, the strain gauges on
their axle were being subjected to the negative shear force due to
gravity acting on \( m \) at that point; \textit{i.e.} as the unsprung mass of the
wheel/hub of interest was in equilibrium and registering zero wheel
force at the scales, the strain gauges were registering a shear force
across the axle equivalent to a negative value of \( m \) at that point;
in this condition, the strain gauge reading (corresponding to \( F_{\text{shear}} \) at
that point) was recorded as the static but negative value analogous
to \( m \) under gravity for that hub/axle stub. The value of \( m \) was \textit{per}
Table A1.1; the \( a \) for the static reading was acceleration due to
gravity: \( 9.81 \text{ ms}^{-2} \). The value of unsprung mass outboard of the
strain gauges became the first (lowest) point on the static wheel
force \( \text{vs.} \) strain gauge plots shown from Table A1.2 to Table A1.4.
Hence, for the purposes of the wheel force \( \text{vs.} \) strain gauge graphs,
the negative value of \( m \) became the lowest point on the graph for
each wheel of interest; \textit{i.e.} when plotting the relationship between
the strain gauge readings \( \text{vs.} \) known static mass values, the lowest
point was the static force from gravity acting on each wheel/hub
outboard of the strain gauge;

the reading of the strain gauges under the resultant zero wheel force
load was set to zero in the telemetry system using set
potentiometers;

the chassis was lowered to normal operating mode; and then

the static reading of the strain gauges at that point yielded a tare
signal that matched the calibrated wheel force \textit{via} transport
inspector scales less the axle/wheel static mass force outboard of the
strain gauges for each corresponding wheel.

Each test vehicle was driven to the loading site after this procedure was completed.
This allowed the suspensions to neutralise any lateral or other residual forces in the
springs, bushings or tyres. The static signals from the strain gauges were recorded as tare values before loading. The test HVs were loaded with test weights and the strain gauge readings recorded for the fully loaded condition. Where possible, logistical considerations allowing, the procedure was repeated after unloading to provide another tare point on the load/strain reading graph. These readings then provided the offset and slope on the strain vs. load graph (Woodroofe et al., 1986) for each axle-end of the axle/s of interest. Using the linear regression lines of these graphs, direct mapping (or correlation) of dynamic signals recorded from the strain gauges during the testing could then be correlated directly to a wheel force value extrapolated from the corresponding linear regression formula that defined the relationship between wheel force (calibrated scale readings) vs. strain gauge readings for the particular wheel of interest. This provided the dynamic values for $F_{\text{shear}}$ in Equation 3.1. Adding this term to the derived term $ma$ in Equation 3.1 produced dynamic $F_{\text{wheel}}$ data for each wheel of interest.

The logistical considerations for loading the semi-trailer were minimal: a forklift and standard loads in bins. However, the loading and unloading of the horse feed to provide the test loads in the buses was time and resource intensive.

![Figure A1.6. Jacking the test vehicle so that the static wheel force could be set to zero.](image)

Daily checks on the quiescent outputs of the strain gauges showed slight variations due to vehicle supply voltage fluctuations. The strain gauge digital count values
were noted and the calibration graph equations for that series of tests were adjusted accordingly.

Telemetry equipment failure after tests 197 and 238 necessitated recalibration of the replacement system. The graph for recordings after test 238 used a different calibration constant and slope since the bus could not be unloaded and re-loaded to determine the tare values for the replacement measurement system. This detail is as noted in the titles in Table A1.3.

The school bus had its strain gauges mounted slightly more inboard on its drive axle than for those positions on the coach drive axle. This resulted in a slightly greater drive axle unsprung mass outboard of the strain gauges on the school bus compared with the coach.

As noted above, one of the steps in calibrating the strain gauges was to jack up the chassis of the test vehicle so that the wheel force registered as close to zero as possible (+5/-0 kg) on the portable scales. Figure A1.7 shows the detail of setting the wheel force to equilibrium for the purposes of setting the recording equipment.
Figure A1.7. Gradually reducing the wheel force as the chassis was jacked up: top panel, very small wheel force; bottom panel, no wheel force.
## Static wheel force vs. strain readings: coach

### Table A1.2. Static wheel force vs. strain readings: coach

<table>
<thead>
<tr>
<th>Coach Rear Left Wheel, Tests with Haire Suspn</th>
<th>Coach Rear Right Wheel, Tests with Haire Suspn</th>
</tr>
</thead>
<tbody>
<tr>
<td>( y = 13.843x - 15682 ) ( R^2 = 1 )</td>
<td>( y = 14.672x - 16923 ) ( R^2 = 1 )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Coach Left Drive Wheel, Tests with Haire Suspn</th>
<th>Coach Right Drive Wheel, Tests with Haire Suspn</th>
</tr>
</thead>
<tbody>
<tr>
<td>( y = 56.467x - 63644 ) ( R^2 = 0.9981 )</td>
<td>( y = 62.156x - 70192 ) ( R^2 = 0.9982 )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Coach Rear Left Wheel, Tests with Std Suspn</th>
<th>Coach Rear Right Wheel, Tests with Std Suspn</th>
</tr>
</thead>
<tbody>
<tr>
<td>( y = 13.881x - 15721 ) ( R^2 = 0.9998 )</td>
<td>( y = 14.554x - 16795 ) ( R^2 = 0.9997 )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Coach Left Drive Wheel, Tests with Std Suspn</th>
<th>Coach Right Drive Wheel, Tests with Std Suspn</th>
</tr>
</thead>
<tbody>
<tr>
<td>( y = 56.485x - 63663 ) ( R^2 = 0.9976 )</td>
<td>( y = 61.609x - 69562 ) ( R^2 = 0.9992 )</td>
</tr>
</tbody>
</table>
### Static wheel force vs. strain readings: School bus

#### Table A1.3. Static wheel force vs. strain readings: bus

<table>
<thead>
<tr>
<th>Wheel Location</th>
<th>Test Range</th>
<th>Tare Suspension</th>
<th>Regression Equation</th>
<th>Mass (kg)</th>
<th>Digital Count Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear Left</td>
<td>197 to 215</td>
<td>Tare with hair suspension</td>
<td>$y = 19.196x - 23200$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450</td>
</tr>
<tr>
<td>Rear Right</td>
<td>197 to 215</td>
<td>Tare with hair suspension</td>
<td>$y = 13.538x - 15532$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450</td>
</tr>
<tr>
<td>Left</td>
<td>216 to 238</td>
<td>Tare with std suspension</td>
<td>$y = 19.361x - 23395$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450</td>
</tr>
<tr>
<td>Right</td>
<td>216 to 238</td>
<td>Tare with std suspension</td>
<td>$y = 13.368x - 15342$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450 1500</td>
</tr>
<tr>
<td>Rear Left</td>
<td>239 to 258</td>
<td>Loaded with std suspension</td>
<td>$y = 25.271x - 31269$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450 1500</td>
</tr>
<tr>
<td>Rear Right</td>
<td>239 to 258</td>
<td>Loaded with std suspension</td>
<td>$y = 16.205x - 18927$</td>
<td>0</td>
<td>1000 1050 1100 1150 1200 1250 1300 1350 1400 1450 1500</td>
</tr>
</tbody>
</table>
Table A1.3 cont’d. Static wheel force vs. strain readings: bus (cont’d).

<table>
<thead>
<tr>
<th>mass (kg)</th>
<th>digital count value</th>
</tr>
</thead>
<tbody>
<tr>
<td>-500</td>
<td>1050</td>
</tr>
<tr>
<td>500</td>
<td>1100</td>
</tr>
<tr>
<td>1500</td>
<td>1150</td>
</tr>
<tr>
<td>2500</td>
<td>1200</td>
</tr>
<tr>
<td>3500</td>
<td>1250</td>
</tr>
<tr>
<td>4500</td>
<td>1300</td>
</tr>
<tr>
<td>5500</td>
<td>1350</td>
</tr>
<tr>
<td>1050</td>
<td>1400</td>
</tr>
<tr>
<td>1100</td>
<td>1450</td>
</tr>
<tr>
<td>1150</td>
<td>1500</td>
</tr>
</tbody>
</table>

Bus rear left drive wheel, tests 259 to 277 (haire suspn, loaded)

$y = 23.828x - 29507$

Bus rear right drive wheel, tests 259 to 277 (haire suspn, loaded)

$y = 15.693x - 18343$
## Static wheel force vs. strain readings: semi-trailer

Table A1.4. Static wheel force vs. strain readings: semi-trailer.

<table>
<thead>
<tr>
<th>Location</th>
<th>Equation</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear left</td>
<td>$y = 18.118x - 19637$</td>
<td>0.9994</td>
</tr>
<tr>
<td>Rear right</td>
<td>$y = 17.389x - 16121$</td>
<td>0.9996</td>
</tr>
<tr>
<td>Mid left</td>
<td>$y = 17.721x - 18763$</td>
<td>0.9997</td>
</tr>
<tr>
<td>Mid right</td>
<td>$y = 18.004x - 19814$</td>
<td>0.9997</td>
</tr>
<tr>
<td>Front left</td>
<td>$y = 18.222x - 19135$</td>
<td>1</td>
</tr>
<tr>
<td>Front right</td>
<td>$y = 18.66x - 19375$</td>
<td>0.9997</td>
</tr>
</tbody>
</table>

### Graphs

- **Trailer rear left wheel**: $y = 18.118x - 19637$, $R^2 = 0.9994$
- **Trailer rear right wheel**: $y = 17.389x - 16121$, $R^2 = 0.9996$
- **Trailer mid left wheel**: $y = 17.721x - 18763$, $R^2 = 0.9997$
- **Trailer mid right wheel**: $y = 18.004x - 19814$, $R^2 = 0.9997$
- **Trailer front left wheel**: $y = 18.222x - 19135$, $R^2 = 1$
- **Trailer front right wheel**: $y = 18.66x - 19375$, $R^2 = 0.9997$
Table A1.4 cont’d. Static wheel force vs. strain readings: semi-trailer (cont’d).

<table>
<thead>
<tr>
<th>Trial position</th>
<th>Equation</th>
<th>( R^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear left std tests</td>
<td>( y = 18.591x - 20159 )</td>
<td>0.9999</td>
</tr>
<tr>
<td>Rear right std tests</td>
<td>( y = 17.413x - 16131 )</td>
<td>0.9991</td>
</tr>
<tr>
<td>Mid left std tests</td>
<td>( y = 17.953x - 18977 )</td>
<td>0.9986</td>
</tr>
<tr>
<td>Mid right std tests</td>
<td>( y = 18.061x - 19894 )</td>
<td>0.9989</td>
</tr>
<tr>
<td>Front left std tests</td>
<td>( y = 18.007x - 18890 )</td>
<td>0.9999</td>
</tr>
<tr>
<td>Front right std tests</td>
<td>( y = 18.573x - 19304 )</td>
<td>1.0</td>
</tr>
</tbody>
</table>
Appendix 2 – Error analysis for the three test HVs

The details following have been outlined elsewhere (Davis & Bunker, 2008d; Davis & Bunker, 2009e). The dynamic and inertial forces at the wheels of a HV are in a constant state of flux during travel. An overall conclusive error value which holds for all conditions is therefore virtually impossible to derive (Cole, 1990). The work of Cole (1990) and, slightly earlier, Mitchell and Gyenes (1989), examined dynamic suspension measures. That work was, in part, trying to allocate indicators to different types of HV suspensions.

The test programme for the three HVs tested for this thesis recorded the dynamic shear forces on the axles at the strain gauges and any dynamic forces from the axles outboard of strain gauges due to the inertial mass of the wheels and other outboard components. This was achieved with axle strain gauges and accelerometers. The readings from these were used to derive both static and dynamic wheel forces. This method used modern instrumentation to reduce error compared with previous methods. Cole (1990) proposed an error of approximately 7 percent due to that work not measuring, nor recording, the unsprung mass outboard of the axle strain gauges. This meant that forces from the dynamic inertia outboard of the strain gauges were not derived. Overall dynamic wheel forces were, at best, estimated as a contribution to the totality of dynamic wheel forces (Cole, 1990).

The overall errors in wheel force measurements from the programme for this thesis were due to the errors in the accelerometer, telemetry system, \( d \neq r \) (Figure 3.15), and strain gauge readings. Overall error in the telemetry system used for these tests has been documented previously at ±1.0 percent (Davis, 2006b). This has been verified by the OBM test programme (Karl et al., 2009b). The individual error of each strain gauge reading by the telemetry system used for these tests can be seen in Table A1.2 to Table A1. The regression values varied from 99.22 percent, or an error of less than one percent at worst for one strain gauge on the coach, to 100 percent (best) for the semi-trailer. These figures indicate that the errors in strain gauge readings were, even at the worst, very small.

Nonetheless, a cross check for overall error was undertaken, and an indicative error value ascribed from the results for the bus, as follows:
noting that the telemetry system sampled at 1 kHz for 10 s per run; 1 x 10⁴ instantaneous values from each strain gauge and accelerometer were recorded per test run;

the bus LSC was derived (Figure A2.1) by averaging those 1 x 10⁴ values across all transducers (by definition, since LSC requires an averaging process for its derivation);

LSC uses an average of instantaneous wheel force for each run and divides it by the mean (or static) wheel force of the group during the test; but

the bus had only 2 wheels under test, therefore:

the bus LSC for each test speed should therefore have been 1.0.

Any deviation from 1.0 for the bus LSC was therefore due to a combined measurement error at the strain gauges, the telemetry system and the accelerometers. The worst-case LSC (i.e. largest error) at each test speed for the bus was plotted in Figure A2.1. This indicated that the measurement error ranged from -2.2 percent to +3.2 percent. Hence an error range of +/- 3 percent seems a reasonable conclusion.

Further, for any instantaneous metrics such as the dynamic load sharing coefficient (DLSC), the averaging process was not used. Therefore, any error would be per instantaneous reading. Instantaneous reading error vs. the compounded error for averaged metrics would be reduced proportionally to the number of samples. With 1 x 10⁴ samples, this made any error in instantaneous metrics very small.
Figure A2.1. Showing the load-sharing coefficient for the bus wheel forces for a range of test speeds.
Appendix 3 – Sample size for OBM testing

The spread of the readings from the test cases for the OBM testing, viz: tare, 1/3, 2/3 and full load was at first assumed to have a normal distribution around a mean value. Accordingly, an appropriate sample size (number of readings per load condition) was found. This was for a desired accuracy value and level of confidence that the mean of the population of OBM systems, as a whole, would not differ from the measured mean of the OBM system under test.

The process of determining sample size required input values such as:

- the expected (or known from previous trials) standard deviation of the experimental measured values;
- desired accuracy of the data; and
- level of confidence regarding that accuracy.

The number \((n)\) of samples (readings) was then determined from the following formula (Snedecor & Cochran, 1967):

\[
 n = \left[ \frac{\left( \frac{Z_u}{2} \right) \sigma}{E} \right]^2
\]

Equation A3.1

Where:

\(n\) = sample size (number of readings) to achieve the specified error;

\(\frac{Z_u}{2}\) = the critical value of the standardised normal \((z)\) distribution used to determine the level of confidence;

\(\sigma\) = the standard deviation of the population data; and

\(E\) = the desired accuracy (specified error) of the test. This was the pre-defined acceptable difference between the mean of the experimental data and the mean of the
The value of $Z_{\alpha/2}$ was determined from the choice of the level of significance known as $\alpha$ which was, in turn, used to derive the level of confidence. The level of confidence is usually designated as a percentage that can be visualised as being bounded by the critical values of $+/- Z_{\alpha/2}$ under the normal population distribution curve (Davis et al., 2008a, 2008b) and related to half of the level of significance ($\alpha/2$) therein.

The level of confidence is a value (or percentage) of certainty that the mean of the sample data will be within the specified error of the mean of the entire population.

The level of confidence is designated:

1 - $\alpha$ as a value; or

$$(1 - \alpha) \times 100$$ as a percentage.

For instance, a value of $\alpha = 0.1$ provides a level of confidence of 0.90 or 90 percent that the mean value in the sample readings will be equal to or smaller than the desired error of the population mean (Snedecor & Cochran, 1967).

The OBM pilot test plan used an initial value for standard deviation ($\sigma$) of 350 kg from earlier work (Davis, 2006b). Early in the pilot testing a maximum value for $\sigma$ for OBM readings of 130 kg for loads approximating 22 t was observed. These results, and choosing a maximum value of $E$ as 140 kg for loads of 22 t (approximately), led to the following calculations to choose the number of readings:

to find the number of readings required to determine the mean accuracy of OBM systems for:

$\square$ a level of confidence of 95 percent; and

$\square$ a standard deviation $\sigma$ of 130 kg; with

---

*In this case, the total population would be the total population of OBM systems.*
a desired maximum error \( E \) of 140 kg between the test data mean accuracy and the population mean accuracy;

let:

\[ \alpha = 0.05 \quad (i.e. \text{ a 95 percent level of confidence}); \]

\[ \sigma = 130 \text{ kg}; \text{ and} \]

\[ E = 140 \text{ kg} \]

\[ \Rightarrow \text{ the area in the region to the left of } \frac{Z_u}{2} \text{ and to the right of } z = 0 \text{ in a normal distribution curve is:} \]

\[ 0.5 - (0.05/2) = 0.475; \]

the table of the standardised normal \((z)\) distribution (CTQ Media LLC, 2008) gave a \( Z_u/2 \) value of 1.96;

Substituting into Equation A3.1:

\[
\begin{align*}
    n &= \left[ \frac{1.96 \times 130}{140} \right]^2 \\
    &\Rightarrow n = 3.3
\end{align*}
\]

\( \therefore \) the number of readings for these experimental parameters for each load condition, \textit{viz:} tare, 1/3, 2/3 and full load; will be \( n \) (rounded up) = 4.

Allowing for a conservative approach for surety, given that 4 is a low number, by adding a margin of (say) 50 percent to four; this made the number of readings \( 4 + 2 = 6 \). This meant six repeated readings per test load condition without changing any

\[ i.e. \text{ assuming that 95 percent of the population of OBM systems return a mass reading within a maximum range of 260kg either side of the mean. This was a deliberately conservative assumption. } n.b. \text{ this 95 percent is not the same 95 percent that was chosen for the degree of confidence in this exercise but relates to the fact that 95 percent of a normally distributed population will lie between a range two standard deviations either side of the mean.} \]
other variables.

The choice of 6 readings was validated in the middle and latter stages of the pilot testing where the values of $\sigma$ (standard deviation of the test and reference system population data) were about or below 140 kg after anomalous data were removed (Germanchev & Eady, 2008).

The conservative approach of six readings increased the level of confidence and reduced the value of $E$ to approximately 100 kg when the maximum value for $\sigma$ dropped below 90 kg.
Appendix 4 – Copyright release

The document (Australia Department of Transport, 1979) from which the diagrams in Figure 1.1 and Figure 1.2 have been taken is not copyright.

The diagram in Figure 1.3 was originally developed by John Woodroofe (Woodroofe, 1997) and reproduced in the DIVINE report (OECD, 1998). Kind permission has been granted by the original author to reproduce the diagram. This permission is shown below (Figure A4.1).
Figure A4.1. Copyright permission from John Woodroffe.

“The first thing we do, let’s kill all the lawyers.” - William Shakespeare, Henry IV, Part II, Act IV, Scene II.
Appendix 5 – Publications

QUT & MR/DTMR publications


Davis, L. and Bunker, J. (2009). Heavy vehicle suspension testing and analysis: dynamic load sharing. Report. Department of Main Roads and Queensland University of Technology, Brisbane, Queensland, Australia.


**Davis**, L. and Bunker, J. M. (2008). In-service testing of heavy vehicle suspensions: background report for the NTC project. Department of Main Roads and Queensland University of Technology, Brisbane, Queensland, Australia.


**QUT, MR/DTMR and TCA publications**


Appendix 5

QUT, MR/DTMR, ARRB and TCA publications


TCA Publications


MR Publications


References


Davis, L. & Kel, S. (2007). Heavy vehicle suspension testing: parametric changes from larger longitudinal air lines; Innovative systems for heavy vehicle suspension testing. Main Roads Technology Forum, 13th, Brisbane, Queensland, Australia: Brisbane, Queensland, Australia: Department of Main Roads. Engineering and Technology Group.


Hahn, W. D. (1987b). Effects of commercial vehicle design on road stress; quantifying of the dynamic wheel loads for stage 3: single axles; stage 4: twin axles; stage 5: triple axles; as a function of the springing and shock absorption system of the vehicle. Working paper ED/87/40. Crowthorne, United Kingdom: Transport and Road Research Laboratory (TRRL).


Kleyner, A. V. (2005). *Determining optimal reliability targets through analysis of product validation cost and field warranty data*. University of Maryland, College Park, Maryland, USA.


