SAFETY OF COUPLINGS ON HIGH PRODUCTIVITY VEHICLES

This project is funded by the Australian National Heavy Vehicle Regulator's (NHVR) Heavy Vehicle Safety Initiative (HVSI), supported by the Australian Government.

A RESEARCH PARTNERSHIP BETWEEN

ARTSA INSTITUTE

AND

AUSTRALIAN TRUCKING ASSOCIATION

HEAVY VEHICLE INDUSTRY AUSTRALIA

TRUCK INDUSTRY COUNCIL

EXECUTIVE SUMMARY

This field testing and simulation project seeks to understand road train coupling forces for gross combination masses (GCM) up to 160 tonne. Findings are critical for the safe selection and use of heavy vehicle fifth wheel and automatic pin couplings on increasingly heavy road train combinations.

Purpose

The D-Value formulae that prescribe the minimum dynamic strength of couplings in the existing Standards are based on a real world force measurements on triple road train combinations up to 125 tonne.

This project seeks to advance the evidence base by measuring the magnitude and character of the forces on quad road train couplings, with GCM up to 160 tonne.

A key hypothesis that motivated this project was to measure whether the forces through the couplings tend to plateau as the GCM increases, instead of increasing linearly with mass as predicted by the D-Value formulae in the Australian Standards. If so, this finding would have significant benefits in cost and convenience, allowing commonplace coupling types and sizes to be used throughout an increasingly heavy road train fleet.

Methodology

Quad road train combinations up to 160 tonne were tested on Northern Territory roads, with coupling forces measured via an instrumented dolly trailer. This novel approach allowed the test dolly to be moved to the three dolly positions to collect measurements on separate journeys. Force vectors, acceleration, GPS and other data was logged while underway in order to determine the nature and magnitude of forces on mechanical couplings. The test dolly incorporated onboard data acquisition equipment and local power supply in order to measure the coupling forces on the heavy vehicle combination in routine use.

Results from this testing were used to calibrate and adjust the parameters of a proprietary software model that was used to simulate road train forces at higher combination masses.

ARTSA institute coordinated and led this research project, while all field testing, post processing and simulation was conducted by Smedley's Engineers Pty Ltd.

Key findings

- There was no evidence of a 'plateau' in the force magnitudes at 160 tonne - a null hypothesis. Forces increase linearly in proportion to mass, consistent with the existing D-Value formulae.
- Peak Force magnitudes were similarly high for couplings situated different dolly locations.
    a. Similar peak force magnitudes were observed within couplings located centrally and forward within the combination.
    b. The D-Value formulae in the Standards, in line with classical equations of motion, predicts that the centremost dolly's couplings will experience the highest forces. Higher observed forces at forward and rear dolly's couplings are due to dynamic effects.
    c. Higher than expected overturn moment forces were observed at the couplings that connect the rearmost trailer. These appear to be due to a "whiplash" effect.

- Vehicle speed vs peak force plots show complex peak force distribution patterns:
    a. High coupling forces were experienced when at higher speeds when the combination met with undulating terrain. Peak forces correspond with the pitch of the hinged dolly;
    b. The majority of high-range forces were observed at low speed, typically less than 10km/h.
- Software simulation provided some insight into stability and coupling forces at higher masses.
    a. Simulation of road train behaviour predicts that forces through couplings could plateau when the overall mass increases to more than 215 tonne GCM;
    b. Force phenomena in multi combination vehicles were found to be complex in nature, therefore proprietary software simulation has poor correlation with real world peak forces;
    c. Simulated results may inform hypothesis for future research programs but cannot be interpreted as conclusive without being validated by real world testing.
- Several engineering design factors have a significant influence on peak coupling forces:
    a. Brake application timing - the signal delay with pneumatically signalled brake systems creates relative movement and therefore high forces on couplings;
    b. The size of the gap between mating coupling components contributes to coupling 'lash';
    c. Hinged drawbar dolly 'pitch' over road undulations corresponded with peak forces as the pitch releases its stored energy, contributing to force peaks through adjacent couplings.

Key recommendations

1. Worst case D-Value specification should apply to all couplings within the combination.
    a. As the centremost coupling will always return the highest value when the masses either side are closest to equal. Therefore, the prescribed D-Value formula can be simplified to return a required coupling D-Value, to apply to all couplings, as a function of just GCM.
    b. A corollary of this simplification is that road train couplings can be prescribed as a simple ratio of the GCM. Calculations result in the following simple rule: GCM_{MAX} = 2/3 x [D-Value].
    c. Combinations over 160 tonne remain subject to real world coupling force validation.
2. Consideration of engineering controls to limit forces through couplings, including:
    a. Mandatory electric-over-air controlled braking for heavy combinations;
    b. Limit coupling operational maximum fit tolerance to control coupling fit, or 'slack';
    c. Investigate the potential for increased use of rigid dollies to avoid the 'stored energy' effect that appeared to greatly increase pin-type coupling loads as a result of the hinged drawbar dolly pitch.
3. Further research is required to better understand and quantify the forces that couplings are subject to at low speed and high speed.
    a. The majority of high force events were at low speed, and collectively these may be the dominant operational environment that limits coupling life;
    b. The highest force events occurred at high speed over undulating terrain. Measured forces were a function of the speed and terrain, and therefore cannot be considered 'worst case';
    c. Higher force is proportionate to wear rates, therefore any initiative that can reduce force will also extend coupling life.

Future opportunities

This research project benefits from generous heavy vehicle industry support, both within Australia and overseas. Novel methodology to measure the coupling forces within the combination have been developed, providing a practicable research platform for the future study of heavy combination forces.

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INTRODUCTION

The "tyranny of distance" in Australia is an ongoing challenge and opportunity for Australia's road transport sector. Since historian Geoffrey Bainey first coined the phrase in 1967, continuous innovation and improvements to heavy vehicle design and road infrastructure has brought Australians closer together and underpinned a lifestyle that can enjoy both the open spaces and the economic opportunity that a vast land brings. Safe and efficient heavy vehicle fleets are now integral to the Australian economy and way of life.

At the vanguard of Australia's high productivity heavy vehicle fleet are the increasingly large road trains in routine use throughout Australia. These combinations push the boundaries of what is possible, with the dollar and environmental cost per payload tonne markedly lower than historical costs.

This report describes a unique research program that seeks to address one of the key engineering challenges that is presented by these large heavy vehicle combinations – understanding the forces and system variables that influence the safe selection and use of the mechanical couplings that connect the vehicles in the road train.

Road trains of increasing combination mass are operating on Australian roads, connected using conventional heavy vehicle couplings with minimum strength requirements that are prescribed by Australian Standards. The D-Value formulae provided in the Standards reflects the dynamic forces that a coupling is subject to. The current D-Value formulae incorporate correction factors that were developed during the 1980's based on real world measurement up to GCM 125 tonne. These correction factors have the effect of greatly reducing the D-Value requirement for couplings in use on multi-combination vehicles. The current research program seeks to learn about and validate the forces on couplings up to GCM 160 tonne.

The ARTSA Institute, along with the Australian Trucking Association (ATA), Truck Industry Council (TIC) and the Heavy Vehicle Industry Association (HVIA) have coordinated this project, bringing together industry expertise and experience.

This research project is required to fill a gap in knowledge regarding the forces in very large combinations. The results will provide the evidence base to inform the update of relevant Standards and Rules.

The project was funded by the National Heavy Vehicle Regulator's (NHVR) Heavy Vehicle Safety Initiative (HVSI), supported by the Australian Government.

BACKGROUND

The current Australian and international Standards for coupling components that are used between heavy vehicles are informed by an understanding of force as derived from classical physics, with theoretical levels of worst-case scenario force being calculated using equations of motion and deriving a figure that represents the dynamic relationship between adjacent vehicles - what has become known as a D-Value.

The D-Value figure and its method of attainment is now defined in European and Australian standards as a means of both certifying coupling strength, and for the end user, selecting appropriate couplings for their application.

Classical equations of motion that represent the forces between vehicles become increasingly complex and less reliable when considering multiple vehicle combinations such as road trains. Previous research has shown that as the size of the road train increases the forces on the coupling decrease as a proportion to the aggregate mass. A body of research that was completed in the 1980s by the Australian Road Research Board resulted in empirical evidence to support the generation of correction factors to the D-Value calculation that have the effect of reducing the approximated force that passes through the road train couplings. These D-Value formulae, incorporating the various correction factors, are now embodied in both Australian and European standards for multi combination vehicles.

A significant benefit of these correction factors is that the resultant maximum predicted dynamic force on the couplings is reduced from what would theoretically be predicted as a 'worst case' force magnitude. This has, in turn, resulted in the ability to select adequately rated common off-the-shelf couplings for road train use. This has significant benefits in terms of fleet interchangeability and efficient route logistics.

The largest combinations that were tested in the 1980s were around 125 tonne gross combination mass, comprising of three semi-trailers within the road train. This means that there is a limit to the empirical basis for the prescribed formulae in the Standards. As combination sizes increase in both mass and length and the total number of vehicles, it has become necessary to conduct testing to measure the forces that the coupling components are subject to within heavy multi-combination vehicles.

Based on simplified worse case assumptions for adjacent mass relationships using classical equations of motion, these D-Value equations attempt to predict a strength requirement that exceeds all anticipated forces while underway on normal roads. The empirical testing that was conducted in the 1980s added an evidence basis and enabled the application of evidence-based correction factors to the inherently conservative equations.

This approach of prescribing that a component is to exceed its strength requirements in all foreseeable scenarios is known as a *Safe Life* design philosophy. This engineering design approach is important on structurally critical components for which there exists no design redundancy. Put simply – the component must be designed for, manufacture and installation such that it never fails in service. This approach must not rely on inspection maintenance or routine replacement to ensure structural integrity unless there are prescribed systems to reliably support them. An example is in aerospace applications, where inspections fit within a strict compliance and enforcement regime including quality control procedures and record keeping protocols.

Given the many tens of thousands of relevant components that fall into this category, supported by tens of thousands of industry participants, sound design and supporting regulatory framework to guide best practice supply and maintenance is paramount.

Heavy vehicle operation in Australia operates under a fragile social licence, and there are many historical examples of where public fear or perceived risk has had a significant impact on heavy vehicle development. A good example is the extensive work that was put into the original introduction of B-Double combinations on Australian roads. B-Doubles were controversial at the time, but with a good evidence base and a sound track-record they are now commonplace. It is hoped that this project and complementary research that follows similarly informs and supports a sound evidence base to maintain community confidence in an increasingly safe and productive national heavy vehicle fleet.

ACKNOWLEDGEMENTS

The ARTSA Institute, along with the Australian Trucking Association (ATA), Truck Industry Council (TIC) and the Heavy Vehicle Industry Association (HVIA) have coordinated this project, with each industry organisation contributing expertise and enthusiasm, as well as engaging with their respective membership.

The practicalities of instrumenting an on-road application meant that this project could not have happened without industry support and in-kind contributions. Many partners contributed and the warm thanks are extended to a number of people and organisations for helping to ensure a successful outcome.

Direct Haul, operating out of Berrimah in NT, were key partners on the ground. Their team was very accommodating to the project's technicians, with their support making this very challenging project possible. The Direct Haul Berrimah team provided invaluable local knowledge of roads and seasonal conditions, allowed Smedley's Engineers to embed into their workshop and gave assistance, tools and space, facilitated equipment shakedowns, patiently took last minute schedule changes in their stride, worked the testing into their commercial operations transporting fuels to Katherine including special detours to find road features of interest, submitted their equipment to those 'road features', and provided stand-in drivers from the management team all to ensure the project was successful.

Warm thanks are also extended to Howard Porter Pty Ltd in WA, who provided the project with a tri-axle dolly at short notice during a stressful period involving Covid delays and disruptions. Delivery to a Melbourne workshop was made possible despite flood damage to the East-West rail line and it seemed the entire national fleet of road train equipment was in service. Howard Porter were also supportive when project delays meant the dolly was required for testing for a longer period of time than planned.

CIMC in Dandenong found space within their busy production schedule to prioritise the build of a custom drawbar to our specification, at a time when all trailer manufacturers were grappling with huge demand.

Coupling suppliers to the Australian market were without exception universally supportive of this project, with considerable actual and in-kind support contributed throughout the project. Technical expertise was generously provided from a number of suppliers.

The project particularly benefited from the sharing of relevant international expertise and physical testing experience by Swedish based engineer Tobias Johansson, Vice President of R&D at coupling supplier VBG (parent company of the Ringfeder brand).

This coupling safety project is funded by the National Heavy Vehicle Regulator's (NHVR) Heavy Vehicle Safety Initiative (HVSI), supported by the Australian Government.

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- BPW Australia
- VBG Truck Equipment [Europe]
- Jost Australia
- Fuwa K-Hitch Australia
- SAF-Holland / V.Orlandi - Australia

RESOURCES

Rules and Standards

- Automatic pin couplings:
  - AS 2213 – 2001 Commercial road vehicles – Mechanical connections between towing vehicles. Parts 1 & 2.
- Fifth wheels and related kingpins:
  - AS/NZS 4968.1:2003 Heavy road vehicles – Mechanical coupling between articulated vehicle combinations – Design criteria and selection requirements for fifth wheel, kingpin and associated equipment.
  - AS/NZS 4968.2:2003 Heavy road vehicles – Mechanical coupling between articulated vehicle combinations – Testing and installation of fifth wheel and associated equipment.
- Australian Design Rule ADR62/02 - Mechanical Connections between Vehicles
- Heavy Vehicle Standards Bulletin VSB6
- UNECE R55 – Addendum 54 - Uniform provisions concerning the approval of mechanical coupling components of combinations of vehicles
  - https://unece.org/fileadmin/DAM/trans/main/wp29/wp29regs/2015/r055r2e.pdf

Research reports and relevant academic publications

- Australian Road Research Board - ARRB; AIR 1083-1. (1980) Strength requirements for tow couplings in road trains; Peter Sweatman
- Australian Road Research Board - ARRB; AIR 1083-2. (1980) Instrumentation for measurement of coupling forces in road trains; Peter Sweatman
- Australian Road Research Board - Report ARR 149. (1987) Strength requirements for fifth wheel couplings in road trains and general articulated vehicles;
- HVTT14: (2016) Clevis couplings in multi vehicle combinations. In the 14th International Symposium on Heavy Vehicle Transportation Technology. Svensson B., Nilsson, J. Fröjd.
- HVTT16: (2021) Requirements on coupling strength of B-Triple and truck B-Double combinations; B. Augusto et al.

Additional unpublished proprietary resources were accessed from VBG Europe courtesy of senior engineer Tobias Johansson, Vice President of R&D at major coupling supplier VBG (parent company of Ringfeder).

PRINCIPAL FINDINGS

Higher mass, lower proportionate force hypothesis

It had previously been observed that as the size and mass of combinations increased, the force through coupling components decreased as a proportion of the overall mass. This phenomenon was not observed at the tested mass of around 160 tonne – a null hypothesis.

This effect was not observed within the simulation environment at overall mass less than around 215 tonne.

D-Value Equations correspond to measured force spectrum

The magnitude of large forces observed was consistent with the theoretical D-Value equations that are published in the relevant Australian Standards (AS2213 and AS4968). These equations are based on empirical evidence gathered during historical testing conducted on roads while underway up to 125 tonnes gross combination mass. The D-Value equations in the relevant Standards reasonably estimate the range of peak forces that were measured during the physical on-road testing.

Forces higher than expected at forward and rearward coupling locations

Similar peak force magnitudes were observed within couplings located centrally and forward within the combination.

The D-Value formulae in the Standards, in line with classical equations of motion, predicts that the centremost dolly's couplings will experience the highest forces. Higher observed forces at forward and rear dolly's couplings are due to dynamic effects.

Higher than expected overturn moment forces were observed at the couplings that connect the rearmost trailer. These appear to be due to a "whiplash" effect. It may be that similar dynamic effects increase the forces on the rearmost dolly's couplings in the same way that the forward dolly's couplings are increased, however these data were not collected due to a setup error for the journey in question.

Highest coupling peak forces at low speed and high speed

The highest forces were observed when at high speed when the combination met with undulating terrain. The magnitude of these forces are consistent with the published D-Value equations in the existing Standards.

Previous testing in the 1980's on road trains did not publish results regarding the nature of forces at low speed, and therefore these had not been considered significant. Observations in the testing conducted for this project indicate that the vast majority of high-range forces are observed at speeds below 10 km/h, associated with:

- Differential timing in the application and release of braking torque, owing possibly to the time response profile of pneumatic brake signalling.
- Momentum transfer between vehicles, exacerbated by the tolerance 'gap' between coupling bodies.
- The storage then release of potential energy by the pitching of convertor dolly chassis and the compression / extension of the suspension of the extreme axles on that dolly.

These real world phenomena have implications for the ability of researchers to computer-simulate coupling forces, as low speed events are subject to complex structural variables such as body pitch, drawbar stiffness, vibration modes and other dynamic factors. These inputs variables are difficult to repeatably model in a computer simulation environment, such as differences in driver brake pressure application.

Complex force relationships and dolly pitch

One would assume that the forces experienced on the dolly's Automatic Pin coupling, and the nearby fifth wheel coupling, would be similar. In reality, these force spectra were observed to be 'decoupled', and complex dynamic forces resulted in unexpected force peaks and dynamic behaviour.

Sources of dynamic complexity include:

- The most important dynamic behaviour to understand is the dolly pitch, which appears to have significant bearing on effective braking, tyre wear, combination stability, and coupling forces;
- Resonant frequencies of adjacent structures, leading to observable peaks in coupling force as a function of this behaviour;
- Forces created as a result of differential brake timing between adjacent trailers;
- Forces exacerbated by the gap between mating coupling components.

SOFTWARE SIMULATION

The relationship between adjacent couplings on convertor dollies was observed to be less linear or correlated than hypothesised. It is presumed that non-linear structural responses in the primary load path between the pin-type coupling and fifth wheel coupling cause there to be mis-match between the forces instantaneously measured at each location that is not explained by the mass inertia of the dolly itself.

Accordingly, multi-body computer simulation using proprietary modelling software cannot be comprehensively correlated to the physical force phenomena. For the purposes of this research program, a best-approximation correlation was determined. The simulation findings indicate areas of interest to research with further physical testing, given the limitations of the simulation environment.

Far more complex definitions of component structural responses, validated by in-field or laboratory measurement, would be required in order to develop a sufficiently high-fidelity model that can predict the distribution of force magnitudes between adjacent couplings, which are a result of complex non-linear dynamic relationships.

Simulation results indicated the following broad observations:

- Forces are highly influenced by road roughness, therefore vertical translation of the trailer chassis is the primary source of in-motion forces.
- Accordingly, forces are linearly proportionate to road speed within practical speed ranges.
- Forces increase in proportion to the size of the gap between mating coupling components, or coupling "lash", until an inflection point where the presence of lash has an isolating effect and forces are reduced. However at low speed, force increase due to increased lash is likely to have no upper bound.
- Rigid drawbar convertor dollies prevent the storage of potential energy in dolly chassis pitch, and so could mitigate high coupling forces and enable operation of very heavy combinations. This requires physical validation.
- Coupling lash has a direct contribution to force magnitudes.
- Very heavy combinations show that forces within couplings start to plateau, meaning that couplings may not need to increase in strength in direct proportion to overall mass. The simulated forces reach an inflection point at around 215 tonnes, and a bi-modal distribution shape was observed in place of the typical parabola described by the theoretical D-Value equations.

ENGINEERING FACTORS

Several engineering design factors have a significant influence on peak coupling forces.

Brake application timing

Some high forces, particularly at low speeds, are potentially attributable to the brake application timing.

As the measured combinations relied on the travel of a pneumatic brake signal between the prime mover and the trailers, the trailers can impact with each other before receiving a braking signal. This delay with pneumatically signalled brake systems therefore contributes to the quantity and magnitude of high forces on couplings, potentially limiting their life.

The use of trailer electronic braking system (EBS), would help to protect the coupling integrity.

Coupling maximum fit tolerance

The couplings in use to connect road trains have a small gap between the pin and the mating round surface, which allows them to connect and move without binding. The size of the gap increases over time as the coupling components wear. The larger this gap, the larger the force that is imposed on the couplings, as the trailers effectively collide with each other as the combination changes speed. This is typically referred to as coupling 'lash'.

Based on first principles, the magnitude of the force increases as a function of the size of the gap. A simplified model suggests that as the impact energy increases as a function of the square of the relative velocity – in other words, it would follow that if the gap doubles, the force quadruples.

Subject to further measurement to validate force measurements for different coupling gaps, and comparing results to combinations in which EBS is in use, it may be possible to limit these local impact forces by introducing a regulatory limit to the maximum wear on respective coupling components.

Hinged drawbar dolly pitch

Hinged drawbar dolly 'pitch' over road undulations, which was observed to correspond with peak forces as the pitch releases its stored energy, contributing to force peaks through adjacent couplings.

Rigid drawbar dollies could remove this source of peak force. Other problems exist with hinged drawbar dollies under heavy braking – namely excessive tyre wear, poor braking performance, and axle overload.

Any consideration of the use of rigid drawbar dollies is subject to field trials and further research to address other known limitations with their use.

VEHICLE IMPACT AT LOW SPEED

Engineers at the ARTSA Institute have raised the consideration of coupling strength in relation to vehicle impact, and in particular, low speed vehicle impact.

In short, the risk is that if the combination crashes into another truck or terrain, the prime mover stops but the trailer behind keeps going – crushing the prime mover cabin even at very low speed, as low as 10km/h.

Should the combination impact with another truck or terrain:

- The momentum of the aggregate mass of the combination behind the prime mover is easily enough to shear the fifth wheel coupling. The unimpeded momentum of the trailer/s can then crush the driver cabin
- Observations from real world fatal incidents, supported by calculated estimates suggests that this mode of impact can lead to fatal consequences for occupants of the prime mover cabin at impact speeds as low as 10km/h.
- This very low impact speed implies the need for a systemic solution given how common these events are - the risk exposure is much higher than previously thought.
- High risk of fatality exists during routine low speed activities, such as during shunting, manoeuvring, and at traffic intersections.

Potential solutions to manage this risk include:

- Requirements for static testing of couplings at a level of force that is a reasonable analog for impact forces that could be anticipated up to a suitable low velocity impact.
- Fitment of secondary hardware to restrain trailers such that the attached trailer combination cannot shunt forward under impact.
- Research into the available injury and fatality burden for relevant crash scenarios to better understand the risk profile.
- Consideration of cabin strength and geometry requirements to preserve a survivable space.

These observations regarding coupling strength are relevant to future consideration of Standards and performance criteria, and forms part of an ongoing effort to ensure that coupling hardware and associates design features reduce the risk to drivers as far as practicable.

DISCUSSION

THE EXISTING D-VALUE EQUATIONS

AS2213 and AS4968 are the Australian Standards that govern the selection, marking, design, testing and performance requirements of mechanical connections between vehicles intended for road use. Off road use is beyond the scope of these Standards and therefore beyond the scope of this research project.

Equation 1 and Equation 2, below, are the equations that dictate the minimum D-rating required for a coupling within a combination for pin type and fifth wheel respectively.

Pin Type Couplings

$$D = 0.6\frac{M_1 \times M_2}{M_1 + M_2} \times gravity$$

M_1 $-$ $Mass\ before\ the\ coupling\ (tonnes)$
M_2 $-$ $Mass\ after\ the\ coupling\ (tonnes)$
D $-$ $D\ rating\ value\ (kN)$

Equation 1 – Minimum D Rating for Pin Type Couplings

Fifth Wheel Couplings

$$D = \frac{4.9T_4(R_4 + 0.08T_4)}{M - U}$$

R_4 $-$ $Mass\ after\ the\ coupling\ (tonnes)$
T_4 $-$ $Mass\ before\ the\ coupling\ (tonnes)$
M $-$ $GCM\ of\ Combination\ (tonnes)$
U $-$ $Static\ vertical\ force\ onto\ 5th\ wheel\ (tonnes)$
D $-$ $D\ rating\ value\ (kN)$

Equation 2 - Minimum D Rating for Fifth Wheel Couplings

The physical concept underlying the theoretical D-Value equations is based on derivation from classical equations of motion, with correction factors informed by both assumptions in relation to road friction limitations, and following the collation of historical empirical evidence acquired during on-road testing of combinations up to 125 tonne gross combination mass (GCM) by the Australian Road Research Board (ARRB). ARRB is now the National Transport Research Organisation (NTRO).

This project seeks to determine if the existing D-value equations are suitable for selecting couplings at higher GCM, up to 160 tonne.

When combination variables are applied, Equations 1 and 2 return similar results for road train applications. (See Figure 30) This means that there is an opportunity to simplify these requirements in line with the results of this research program. A proposal to simplify these equations when applied to road trains is put forward in the *Recommendations* chapter below.

COUPLING MARKET SUMMARY

The following companies supply heavy vehicle couplings suitable for road trains to the Australian market – current at the time of publishing.

Table 1: Coupling market summary

Brand	Region of Origin	Automatic Pin-Type Coupling Product	Fifth Wheel Coupling Product
BPW	Europe	VBG Ringfeder	N/A
Jost	Europe	Rockinger	Jost
Fuwa K-Hitch	Europe	N/A	K-Hitch
SAF Holland / V.Orlandi	Europe	V.Orlandi	SAF
HYVA	Europe	HYVA	N/A
D'Angelo	Australia	N/A	D'Angelo

LIMITATIONS

Limitations of the field testing

- The majority of the test data that was collected was while underway on good quality roads. The most interesting results occurred while either on poor roads or when shunting and manoeuvring at slow speed. Therefore, any future research should focus on both high speed with undulating terrain and low speed shunting, in order to collate more data at these critical events.
    - The forces through the coupling while underway were observed to be very low compared with coupling rated capacity. This meant that most of the data collected could be ignored with respect to assessing coupling integrity.
    - The highest forces were observed during the relatively few elements of undulating terrain and appeared to increase in proportion with vehicle speed.
    - The majority of high forces were observed at low speed while start/stop/shunting.
- The results were measured in real world application, and therefore do not represent the worst case. In order to determine worst case, a controlled testing environment on prescribed terrain scenarios would be required, subjecting the combination to known extreme inputs, such as on a proving ground.
- The peak forces that were observed were a function of route, speed and driver behaviour. Therefore, these results cannot be used to quantify absolute values for safety factors or levels of redundancy.
- Different drivers tended to produce different levels forces on the same route. This variable may be a dominant factor in some cases, particularly at low speed. This is relevant to experiment repeatability, and follows experience with previous heavy vehicle testing, wherein some drivers are extra careful, and others are not – consistent with the Hawthorne Effect that can be a factor with all research involving humans inputs.

Limitations to the software simulation

- Empirical evidence when compared with the simulation outputs indicated that some model results are overly linear due to necessary simplifications in input parameters.
- As input parameter complexity is added (spring/dampener/flexible behaviour at connections such as couplings, drawbar hinges, drawbar mechanical structure, chassis mechanical structure etc), non-linear effects can be observed. In other words, as the model is tweaked to become more accurate for a given comparison scenario, it becomes less accurate for scenarios sufficiently different from the comparison. This reduces confidence in extrapolated data generated from different combination scenarios, such as much higher mass simulations.
- More understanding of the component structural responses, validated by in-field or laboratory measurement, is required in order to develop a sufficiently high-fidelity model that can predict the distribution of force magnitudes between adjacent couplings, which are a result of complex non-linear dynamic relationships. This includes, for example, being able to accurately model the hinged drawbar pitching, which is important to braking, stability, and coupling forces.
- Whilst the software in use represents current best practice, some complex force interactions observed in practice could not be accurately reverse engineered in the software environment, such as the complex dynamic interactions and force balance between adjacent Pin and fifth Wheel couplings.

RECOMMENDATIONS

Worst case D-Value should apply to all coupling types within the combination.

The magnitude of forces observed in the front dolly of the combination were similar to forces at the centre dolly.

In practice, combination interchangeability typically requires that couplings are capable of operating in all positions within the combination.

While some rear-coupling events were not measured at the fifth wheel due to a setup error, there is evidence that the rearmost coupling experience some higher forces as a result of a 'whiplash' effect along the combination.

These on-road measurements of peak forces indicates that industry guidance should recommend that all couplings of the same type within the combination are specified to the D-Value as calculated in the worst-case scenario.

Given that the centremost coupling will always correspond with the maximum required D-Value, and this in turn is maximum when the mass either side is equal – there is an opportunity to simplify and generalise the calculation for all couplings required within the combination.

Assuming a worst case in which the aggregate mass forward of the coupling = aggregate mass rearward of the coupling = GCM/2, and making other simplifying assumptions, the following simple relationship follows from both the Automatic Pin and fifth Wheel couplings (derived from equations 1 and 2 above):

Maximum Road Train GCM (tonnes) = 2/3 x Coupling D-Value (kN)

Or conversely:

Required Coupling D-Value (kN) = 1.5 x Maximum Road Train GCM (tonne)

Equation 3 – Minimum D Rating for Road Train Couplings

Engineering controls to limit peak forces on increasingly heavy combinations

Larger mass combinations are expected to experience proportionately higher forces, in line with the D-Value formula. As the overall mass increases, there are significant cost and logistical advantages with retaining the standard coupling size and type, if the resultant forces to which the coupling is exposed can be engineered to be sufficiently low. The following strategies are considered important engineering controls with a view to retaining coupling type status quo.

The following engineering controls can limit the incidence of the most damaging peak forces:

- Mandatory electronically controlled brake systems for combinations exceeding a minimum operational mass.
- Consideration should be given to placing an upper limit on the acceptable coupling fit tolerance for heavy road trains. This can reduce the high peak forces associated with adjacent trailers colliding, with the momentum transfer impact greatly increasing in proportion to the gap between mating coupling components, or coupling "slack".
    - This should be subject to further testing in order to quantify the real world magnitudes, especially during low-speed manoeuvring, shunting and braking.
    - Ensuring a limited wear window would involve regular coupling condition inspection and wear measurement within a suitable operational window. This has the additional advantage of increasing the likelihood of detecting high wear within structurally critical components, by virtue of reducing the inspection window.
- Further research to evaluate the role of rigid drawbar dollies.

Inspection and maintenance

The size of the gap between mating coupling components, or coupling 'lash' has a direct contribution to a significant increase in force magnitudes. Formal specific guidance to industry should be considered for provision to industry to support traceable measurement, inspection and maintenance of coupling components.
Specific guidance to industry should be provided to support traceable inspection and maintenance of coupling components. This is currently left to industry and 'workshop' level inspections take place, with formal Quality Assurance systems and traceability potentially inconsistent, absent or ad-hoc.

Further research into complex hinged drawbar related force phenomena

A number of observed force peaks corresponded with physical behaviour of the pitch of the hinged drawbar dolly. There is potential to significantly reduce the forces through couplings if rigid drawbars are used in place of hinged drawbars on heavy road trains, with other benefits including more effective braking, improved tyre wear, and potentially improved dynamic behaviour.

These potential benefits need to be balanced with some observed disadvantages of rigid drawbar dollies, and further research and validation is required in order to make sufficiently informed recommendations.

Low speed coupling forces

Further research is required to better understand the forces that couplings are subject to during braking, shunting, and manoeuvring at low speeds, typically under 10km/h. It was at this speed that the majority of high coupling forces were observed during testing, despite the time spent at these speeds being relatively low. It is noteworthy that simulation software does not provide useful simulation information in relation to low speed force due to the inability to accurately anticipate brake input pressure modulation.

Validation required for combinations over 160 tonne

A key hypothesis of this research was that the coupling forces would likely plateau with increased combination mass, however based on the simulation data it is now hypothesised that the inflection point where this relationship may change is greater than 160 tonne.

Simulations indicate that forces may plateau in the region of 215-230 tonnes. Incidentally, the greatest combination mass supported by a 320 kN rated automatic pin type coupling under current Standards is a combination mass of 217 tonnes.

Future testing should seek to validate the coupling forces experienced by these very heavy road train combinations with quad axle-grouped quad-trailer combinations in jurisdictions which allow them to operate at 30 t axle group mass (currently in use in Western Australia).

Road design

Road design implications may be relevant, including criteria around:

- the surface geometry of undulations as a proportion of speed, such as at crossings or causeways;
- avoiding design scenarios that could necessitate sudden stops.

CONCLUSIONS

This project has shown the value of detailed and coordinated research conducted with widespread industry cooperation within a collegiate and cooperative international engineering context.

The increased understanding regarding the nature and magnitude of forces that couplings are subject to within heavy combinations provides confidence in the safe use of high productivity heavy vehicle combinations. As combination masses continue to increase this work is critical to provide a sound evidence base to inform the ongoing development of Standards and the supporting regulatory framework.

The results of this research point to the safe use of conventional couplings into the future on the heavy quad road train combinations.

Detailed observations were made regarding the complex dynamic interactions that contribute to peak forces. These findings have pointed to a number of opportunities for regulatory and engineering improvements that could enable the safe use of conventional couplings on increasingly higher mass combinations.

TECHNICAL REPORT

The following technical report outlines the results from both the physical measurement and simulation analysis.

These technical report chapters have been principally authored by Smedley's Engineers Pty Ltd, who were contracted to undertake the field measurement, postprocessing analysis and software simulation for this project.

The research program that is summarised in the following chapters consisted of two parts:

- PHYSICAL TESTING - Real world measurement of instrumented road trains while underway
- SOFTWARE SIMULATION - Dynamic stability simulation of road train behaviour within a third party software environment.

A number of parameters can be adjusted and calibrated within the simulation environment to better tailor the complex dynamic stability simulation environment. The simulation undertaking benefited greatly from being able to calibrate to match the real world results that were recorded as part of this project. Critically, the simulation was also able to take advantage of technical advice and collaboration with coupling manufacturers.

Plain English:
The technical section of this paper includes technical terminology. Throughout, blue boxes like this one have been provided to summarise the important points while avoiding engineering terminology.

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CONVENTIONS AND DEFINITIONS

Vehicle axes used are the ISO8855:2011 automotive axes:

- X is the vehicle longitudinal axis;
- Y is the vehicle lateral axis; and,
- Z is the vehicle vertical axis;
- Rotation about X is roll;
- Rotation about Y is pitch; and,
- Rotation about Z is yaw.

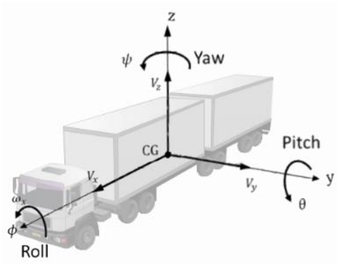


Figure 1 - ISO8855:2011 Vehicle Axes

Units used are suffixed but generally are SI in mixed magnitudes:
Metres, seconds, kilonewtons, MPa and degrees (°).

REPORT – PHYSICAL TESTING

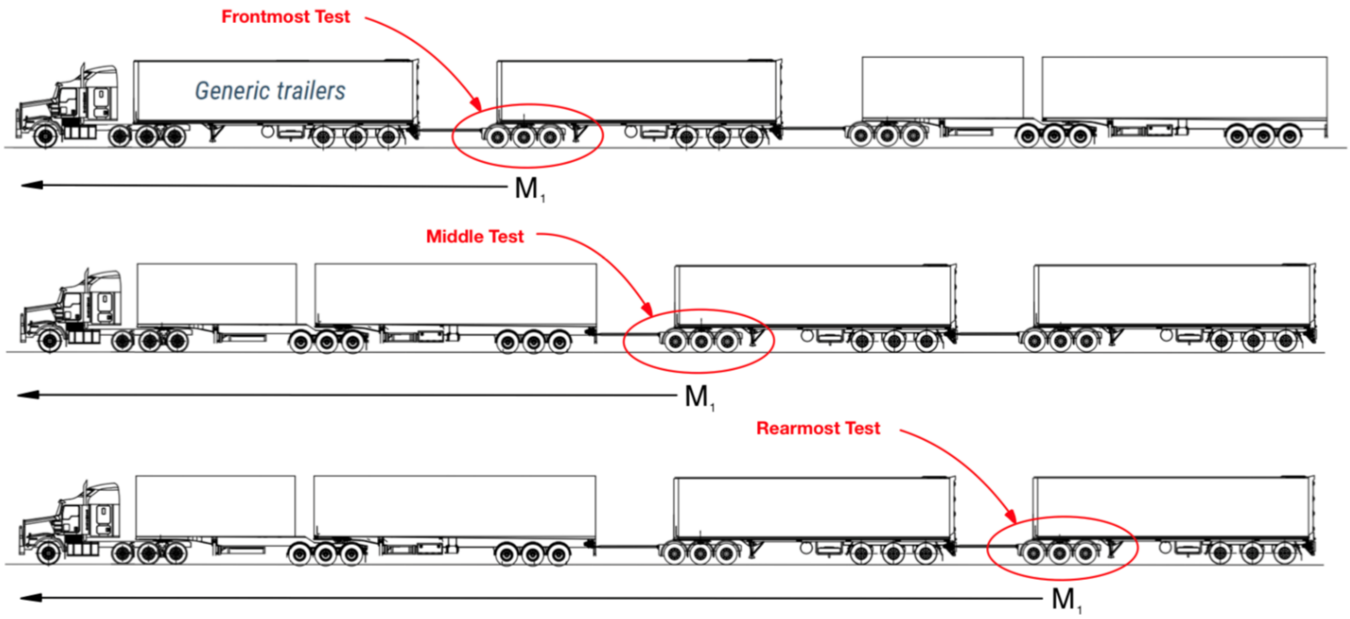
To investigate the actual forces experienced by the couplings at various locations within the road train combination compared to those predicted by the D-Value equations, a range of factors were considered.

The range of factors were:

- Testing at higher combination mass than previously tested, i.e. 125 tonne increased to 160 tonne GCM, to ensure real road loads are recorded for a moderately high GCM vehicle to inform conclusions.
- Testing at different coupling positions along the combination, to understand the relationship between total combination mass and mass before/after the coupling.
- Testing at the shortest possible drawbar length to minimise the drawbar travel arc radius, and therefore investigate the relationship between speed, trailer relative vertical movement before/after the coupling, and resultant forces.
- Test on both rough roads and on roads where high speeds can be attained with a 160t GCM vehicle, to determine if the increased energy of road inputs with speed has a significant relationship to coupling forces.

Testing in the Northern Territory provided ready access to 160 tonne GCM vehicles operating on public roads, allowing for embedment with an existing commercial operation to gather large datasets economically. Local advice was sought on routes containing a mixture of highway speeds and rough road features.

The published D-Value equations within Australian Standards AS2213 and AS4968 will be referred to as the "theoretical equations" in this technical report section. References throughout this section to pin couplings means the automatic pin couplings.



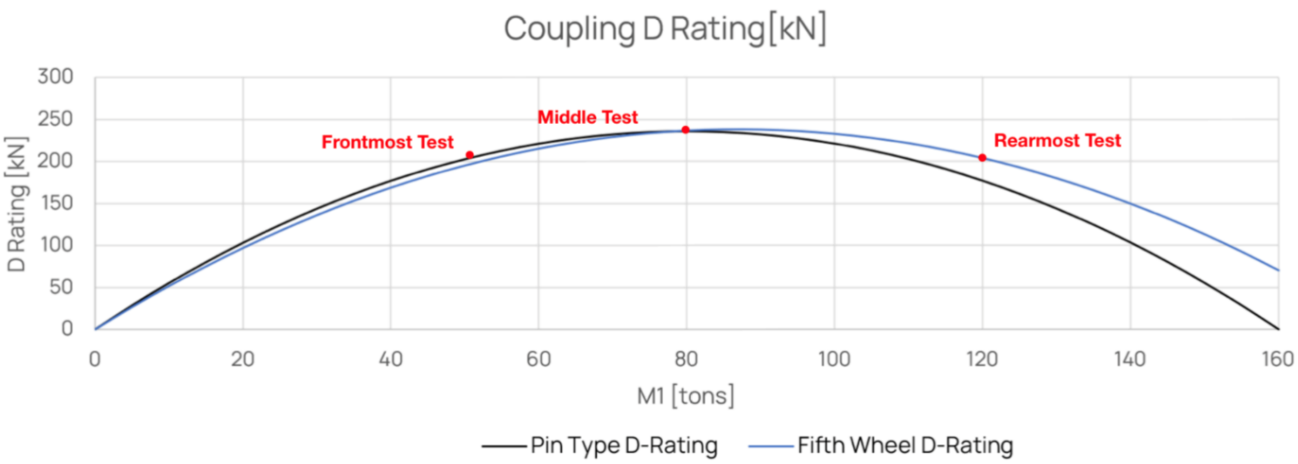


Figure 2 - D-Rating as a function of mass before coupling, AS2213.1 and AS4968.1

Each of these simplified force relationships is problematic when compared with what is known in real world application. The following hypothesis and assumptions inform this research project.

- Coupling forces are unlikely to increase linearly with GCM. Due to mass inertia within a combination, it is reasonable to hypothesise that they must tend towards some limit. Railroad trains with combined masses in the tens of thousands of tonnes are testament to this, notwithstanding that railroad trains are very different and do not have road input loads.

- Forces cannot reduce as sharply as predicted by these equations in closer proximity to the truck, due to unique local longitudinal force inputs such as:
    - brake application timing delay in pneumatic brake systems
    - traction loads
- Forces do not tend towards zero with increased proximity to the end of the combination, due to dynamic handling effects such as steady state and transient off-tracking. High forces can be associated with keeping high mass bodies together under increased body accelerations associated with increased off-tracking at units more distant from the lead unit. This phenomena could be described as a 'whiplash' effect.

A practical ramification of the theoretical D-Value equations is that for very high GCM combinations in operation today, the required coupling D-rating prescribed in the Australian Standards is beyond that which is currently available on the market.

While these equations are intuitive and easy to work with, they are limited by the simplifying assumptions that inform their derivation.

In summary:

- Multiple vehicle combinations may not be well represented by the D-Value equations due to dynamic and complex interactions between the many vehicles in the road train.
- Physical testing is required to provide the empirical evidence base to find out what the actual coupling forces are at the various locations along the road train.

Plain English:
According to the current standards, the couplings towards the middle of a combination see higher forces, and coupling D-ratings keep increasing as GCM increases.

A greater understanding of the real world coupling forces is required as combinations increase in length and mass. Under current guidance very heavy combinations (>200t GCM) cannot use conventional couplings. There are significant cost, convenience, and interchangeability benefits if conventional couplings may continue to be used for higher GCM vehicle combinations.

It also ignores things like force spikes during braking, and the increased movement of the last trailer on long road trains ("whiplash" effect).

TEST METHODOLOGY – PHYSICAL TESTING

To investigate the actual forces experienced by the couplings at various locations within the road train combination compared to those predicted by the D-Value equations, a range of factors were considered. These were:

- Testing at higher combination mass than previously tested, i.e. 125t increased to 160t GCM.
- Testing at different coupling positions along the combination, to understand the relationship between total combination mass and mass before/after the coupling.
- Testing at the shortest possible drawbar length to minimise the drawbar travel arc radius, and therefore investigate the relationship between speed, the trailer's relative vertical movement before/after the coupling, and resultant forces.
- Test on both rough roads and on roads where high speeds can be attained with a 160t GCM vehicle, to determine if the increased energy of road inputs with speed has a significant relationship to coupling forces.

Testing in the Northern Territory provided ready access to 160t+ GCM vehicles operating on public roads, allowing for embedment with an existing commercial operation to gather large datasets economically. Local advice was sought on routes containing a mixture of highway speeds and rough road features.

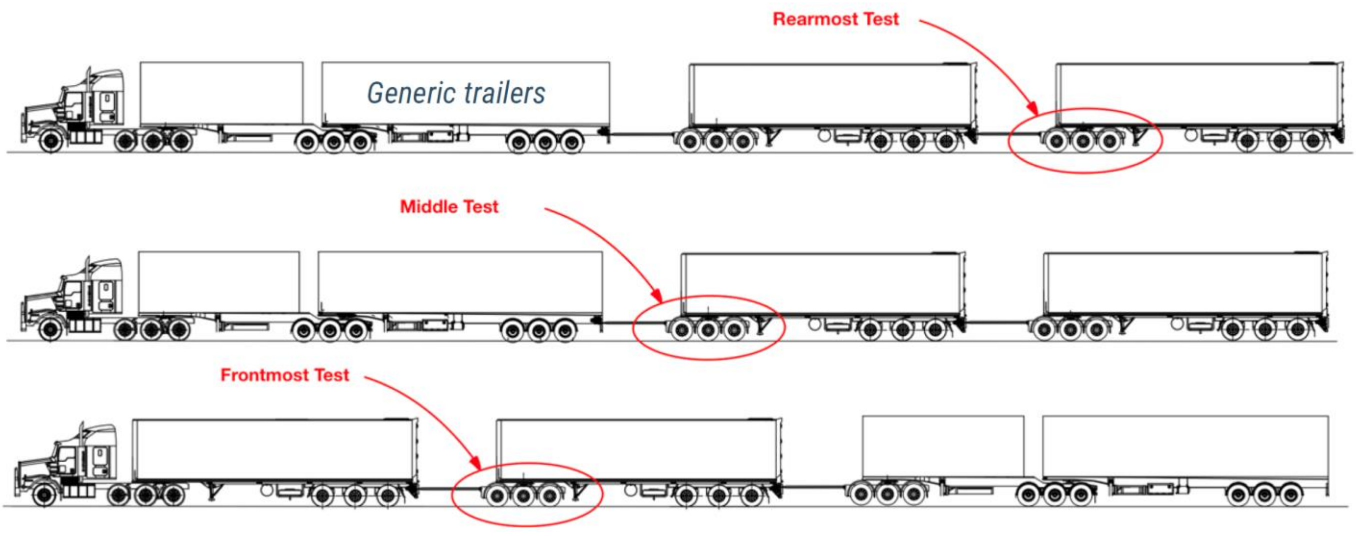


Figure 3 - Diagram showing the different configurations in the testing scheme

TEST EQUIPMENT

To investigate the factors highlighted, a test dolly was assembled which could be relocated along the combination whilst retaining the same remaining units, albeit reconfigured. Refer to Figure 3. Accordingly, a B-A-A Quad and A-A-B Quad vehicle combination were tested to facilitate test dolly relocation.

TEST EQUIPMENT DEVELOPMENT

To investigate the factors highlighted, a test dolly was assembled which could be relocated along the combination whilst retaining the same remaining units, albeit reconfigured. Refer to Figure 3. Accordingly, a BAA Quad and AAB Quad vehicle combination were tested to facilitate test dolly relocation.

Functional requirements:

- A minimum of IP56 rating for the exposed test equipment to deal with inclement weather and road dust.
- As an on-road test and operating over an extended duration within a commercial operation, complete ADR62 compliance was to be achieved by the test equipment.
- The load cells were to be capable of standalone laboratory calibration, independent of the dolly location due to project timeframe and dolly logistics.
- Prior testing in a similar smaller scale trial highlighted that direct strain gauging on the tow eye and fifth wheel feet could introduce unknown variables, owing to cast surface roughness, cross-sectional inconsistency and strain dead zones.
- Load cell design was commenced and completed before a test dolly was secured so the load cells were to be of a flexible interface to be adaptable to any dolly specifics.
- Measurements were to be taken in series with couplings rather than in parallel to eliminate unknown load path redundancies and provide measurement confidence.
- Load cells in series with the fifth wheel were not to contribute to an increase in skid plate height that could not be compensated in ride height adjustment.
- Minimal or no analogue signal cabling was to be exposed on the test vehicle and subject to strain events at mechanical couplings, so all remote signals were to be digitised at the load cell and transported via CANBUS.

Additional factors or learnings that influenced test requirements:

- The partner operator involved in the project was operating a dangerous goods vehicle on the most appropriate test for the route, so the data acquisition system had to comply with the relevant Dangerous Goods standards around electrical isolation, mechanical protection and double insulation.
- The data acquisition system was to be centralised in the truck cabin with only minimal cabling required to reconfigure the dolly location, so a single combined CANBUS and power trunk was deployed along the vehicle combination.
- Uncertainty of coupling load magnitudes lead to an over-design of load cell mechanical strength.

In the final design a modular M16 Grade 12.9 compression/tension stud with rolled threads was furnished with a full Wheatstone bridge strain gauging arrangement, providing both temperature compensation and bending moment elimination to measure purely axial strains. A 24-bit analogue-to-digital converter, power supply, microcontroller and CANBUS interface were integrated into an IP67 case on the stud, ensuring that all analogue signals were shielded and mechanically protected. A resolution of 1N was achieved, up to a maximum of ±16kN per load cell, with a linear calibration curve.

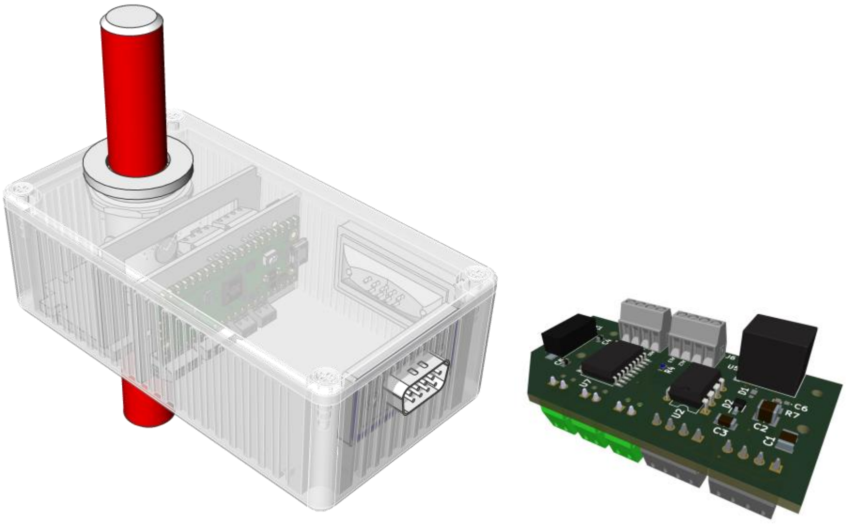


Figure 4 - Custom digital load cell developed for the project

Twenty identical digital load cells were manufactured in Australia, with six deployed to provide a 3D force vector measurement from the drawbar near to the pin-type tow eye, and 14 deployed to replace each fastener on the ball-race under the fifth wheel and provide a 3D force vector measurement. In order to accommodate the necessary CANBUS trunk length for a road train configuration, the baud rate was limited to 250 kb/s which then imposed an 80 Hz rate limit on each load cell. This was estimated to be more than double the frequency of any events of interest, satisfying the Nyquist theorem, which is a sampling rule which ensures sufficient measurement fidelity.

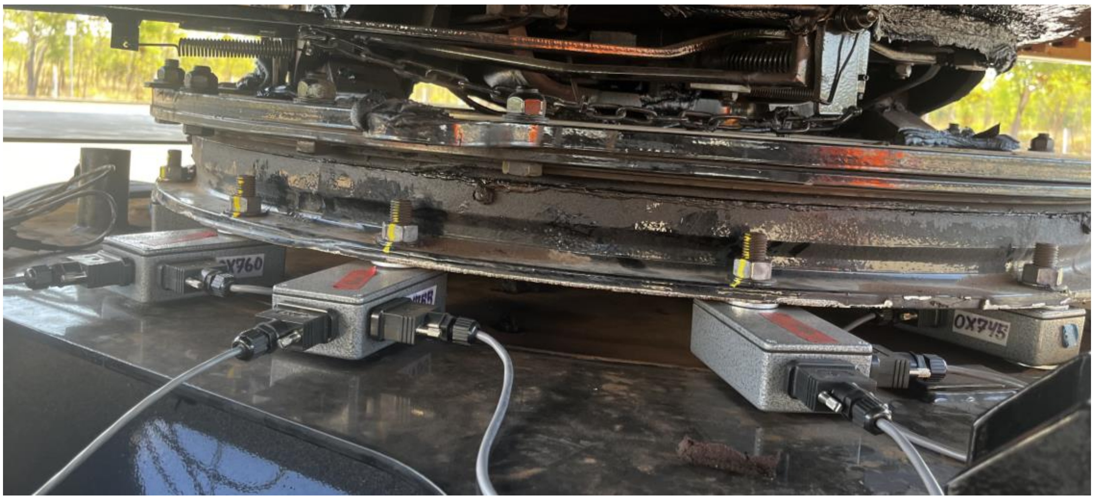


Figure 5 - Low profile load sensing studs/spacers mounted under the ball-race

Additional sensors were mounted to the test dolly to measure data to assist in correlating multi-body simulations. A three-channel ultrasonic road measurement apparatus was developed to record road features on each wheel path at the dolly, plus an additional measurement on the vehicle centreline. These sensors asynchronously measured at 30 Hz each. In the same processing unit an Inertial Measurement Unit measured three translational and three rotational axes of body movement at the front of the dolly chassis at 50 Hz.

Figure 6 - Ultrasonic distance measurement sensors recording wheel path vertical profile

An additional Inertial Measurement Unit was mounted to the dolly chassis as close to the centre point as possible, measuring dolly chassis movements at 50 Hz in three translational and three rotational axes.

A GPS unit recorded position and velocity at 10 Hz from the roof of the prime mover cab.

DRAWBAR LOAD CELLS

Load cells were arranged in a grouping of six to provide for a symmetrical measurement of lateral forces with a D-rating adequate to meet or exceed the D-rating of the pin type coupling.
The drawbar forces were measured by 6 load cells in-line with the longitudinal load path. Longitudinal force was found by summing the measurements from all the load cells. Lateral and vertical forces were found from the summations of the moments about the medial planes of the load cell group.

$$F_{Longitudinal} = \sum^{i} F_{Loadcell_i}$$

Equation 4 – Longitudinal pin type coupling forces

$$F_{Lateral} = \frac{1}{L}\sum^{i} F_{Loadcell_i} \times y_i$$

Equation 5 - Lateral pin type coupling forces

$$F_{Vertical} = \frac{1}{L}\sum^{i} F_{Loadcell_i} \times z_i$$

Equation 6 - Vertical pin type coupling forces

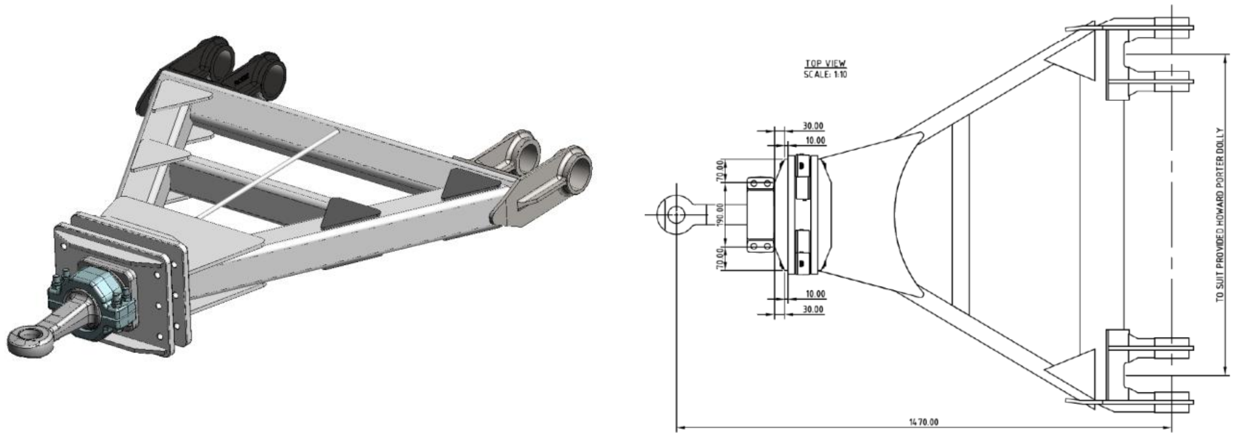


Figure 7 - CAD images from development of the custom test drawbar, with the measurement flange adjacent to the tow eye

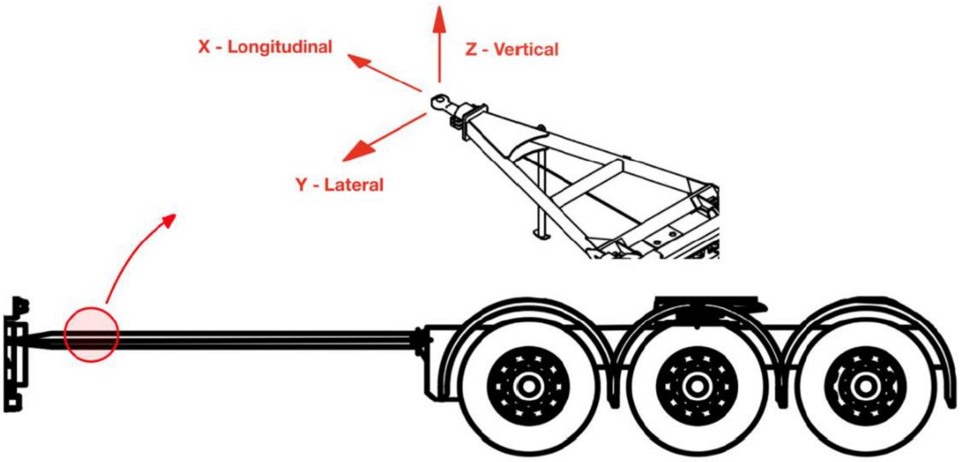


Figure 8 - Orientation of forces

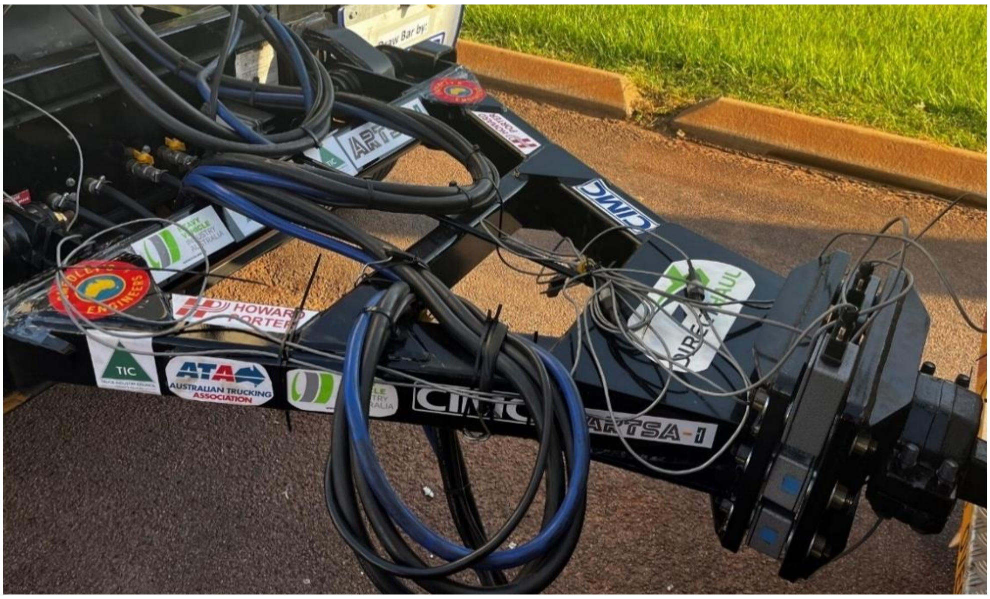


Figure 9 - The complete test drawbar

FIFTH WHEEL LOAD CELLS

Fifth wheel forces were measured using fourteen load cells introduced between the ball-race and the dolly chassis, replacing the bolts typically in this location with load sensing spacers/studs. Whilst the flange arrangement was not evenly spaced, based on the cartesian dimensions of these load cells from the centre of the ball-race, the longitudinal force and overturning moment was found from summation of moments, and vertical forces were found from the summation of forces. These outputs were calculated using the equations listed below.

$$F_{Longitudinal} = \frac{1}{L}\sum^{i} F_{Loadcell_i} \times x_i$$

Equation 7 - Fifth wheel longitudinal forces

$$M_{Overturning} = \frac{1}{L}\sum^{i} F_{Loadcell_i} \times y_i$$

Equation 8 - Fifth wheel overturning moments

$$F_{Vertical} = \sum^{i} F_{Loadcell_i}$$

Equation 9 - Fifth wheel vertical forces

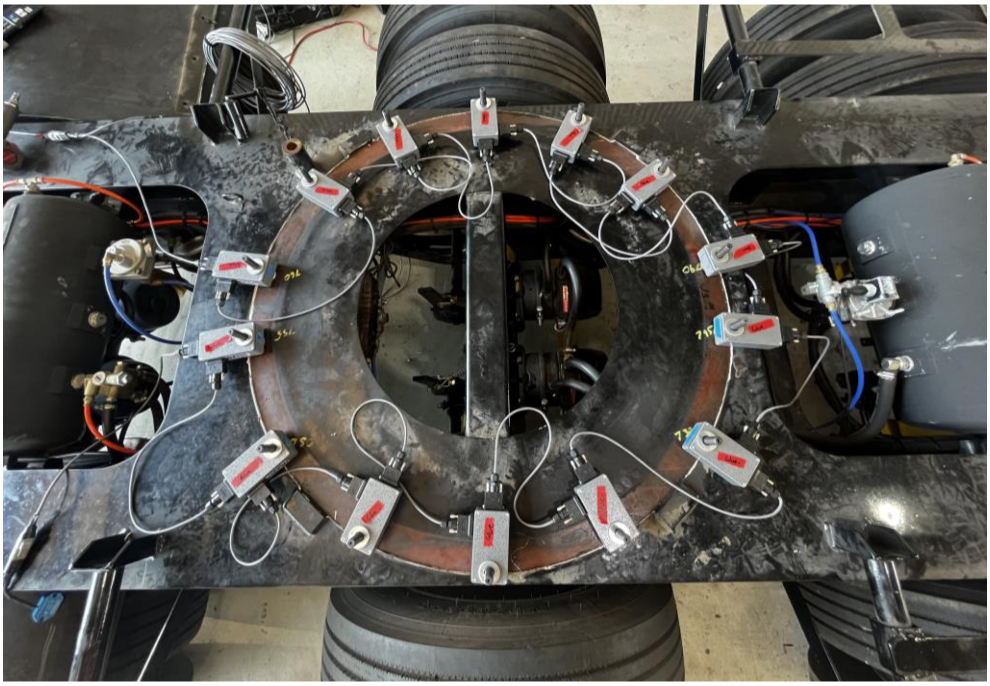


Figure 10 - Layout of the fifth wheel load cells

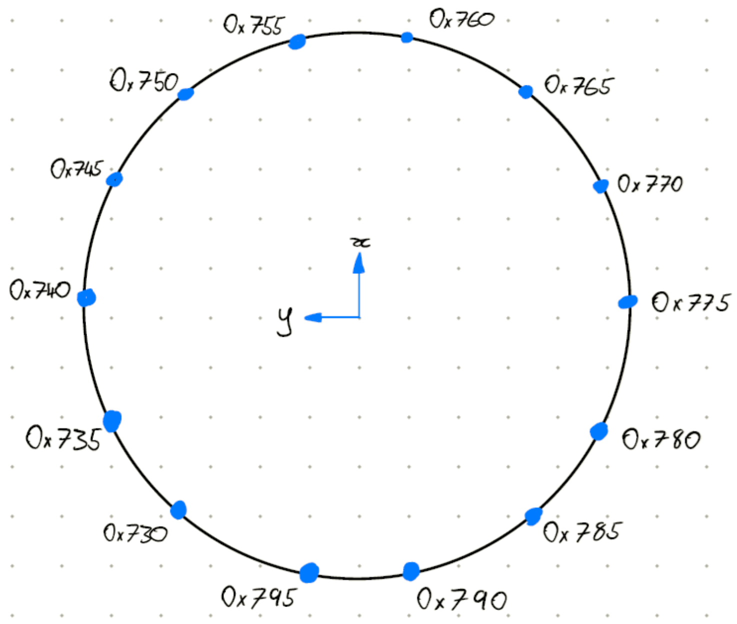


Figure 11 - Load cell identifier numbering, positive X direction is forwards on the combination, positive Y direction is to the left hand side of the vehicle

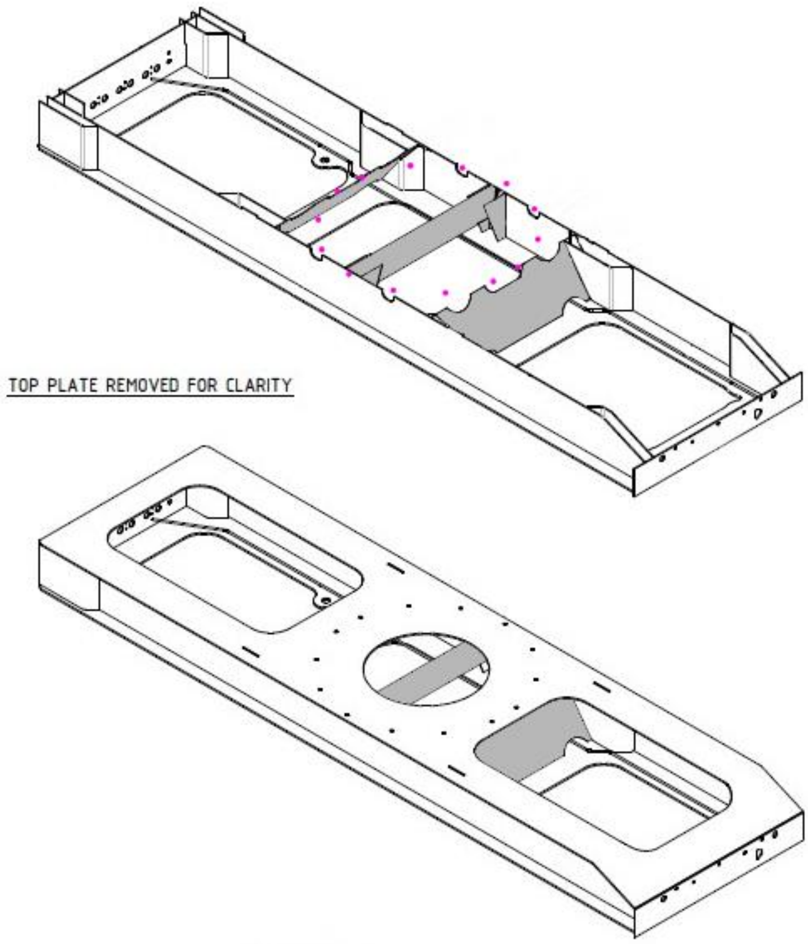


Figure 12 - Test dolly chassis, with load cell locations marked in magenta, and load bearing crossmembers highlighted in grey

Due to the composition of the dolly chassis, some load cells varied in reading more than others as they were in a more direct, stiffer, load path. In Figure 12, four locations with minimal bracing support can be seen. These correspond in Figure 11 to 0x730, 0x750, 0x765 and 0x785. These cells read in unison, but at lower magnitudes, than those around them.

Plain English:

To make sure the force measurements were easy to interpret, and accurate, custom studs were made to measure forces in the connections between the drawbar and the pin coupling, and between the dolly chassis and the fifth wheel. The only parts connecting the couplings all had simple force measuring functions so no forces could be missed.

FIFTH WHEEL ORIENTATION IN REAR DOLLY POSITION TEST

Figure 13 - Test Dolly fifth wheel oriented at 54° away from proper position during first (rear dolly) test

Due to the test dolly fifth wheel coupling assembly consisting of a combined ball race and fifth wheel, it was intended for use with vehicles with a Skid Plate Locking Key. The remainder of the test vehicle did not use fifth wheels of this type and so did not have Skid Plate Locking Keys. During the first test run, with the test dolly in the rear position, the fifth wheel was not oriented straight during coupling and the lateral axis of pivot of the fifth wheel was approximately 54° counter-clockwise from the proper position – see Figure 13. This led to very high fifth wheel M_y (moment about the lateral or Y-axis) measurements during that test, due to the partial restraint of the crucial pitch degree of freedom between the dolly and trailer chassis. As fifth wheel longitudinal force measurements were derived from the M_y measured about the ball-race, longitudinal fifth wheel force measurements were invalidated for the rear dolly position test.

Overturning moment (M_x) and pin type coupling measurements were unaffected. The issue was resolved for subsequent tests.

Plain English:
Mis-coupling of the test dolly during the first test, where the dolly was in the rear position, meant that fifth wheel force measurements were not correct for that test.

MEASUREMENT LIMITATIONS

While the best available measurement equipment and methods were utilised in this novel approach to instrumentation, it is important to estimate and quantify potential sources of error.

- Due to scheduling limitations, the calibration of the apparatus took place after removing the sub-assemblies from the test dolly, and at a laboratory in Melbourne a strain survey took place. Good hysteresis was observed, and this survey indicated a worst case 0.6% error was observed as compared to a Class A reference load cell.
- The measurement devices (in isolation) were subject to known error with an approximate maximum magnitude of 0.9% and uncertainty of ±0.25%.
- Unbolting the measurement apparatus from the dolly and situating in the calibration facility can result in measurable residual strains at the sensor locations. This was addressed via live tare of the sensor array in a known loading condition.
- Comparing different journeys over the same or substantially similar terrain when the dolly was in different positions was complicated by different laden masses of the respective tanker trailers. Differences were minimal however, with mass variation of 157.6t to 162.4t, a variation of 3%, meaning that results could be compared with reasonable confidence.
- A commercial weighbridge (Berrimah East Arm Weighbridge) was used to determine axle loads, the error for which was ±0.1% with a resolution of 50 kg.
- Some driver behaviour characteristics were observed by the technicians accompanying the vehicle on the journey. Due to operational constraints, not all tests were undertaken using the same driver. Anecdotally, due to relative familiarity with the routes and vehicle, one driver had a different approach to similar features than the other. This was reflected in the magnitudes of comparable measured forces. This may have been partly a 'Hawthorne effect', where the test subject changes behaviour as they know they are being tested. However, it may simply be differences in typical driving patterns and approach to the road.

TESTING ROUTE

As part of the commercial operations of the testing partner Direct Haul, the tankers were loaded with fuels in Darwin for delivery to Katherine. Weighbridge dockets and bills of lading were retained as record of combination mass data. Varying orders for fuel quantities of differing densities lead to some variance in GCM between 157.6 t and 162.4 t across the tests, a ±1.5% spread compared to the nominal GCM of 160 t.

The test route shown in Figure 14 below commenced in Berrimah on the outskirts of Darwin, in the north-west, travelling along the route denoted in **light blue,** deviating to and from Robin Falls along the **dark blue** route, then further detouring south on the **magenta** route to a turnaround at a stock station. Due to a tyre failure on the second test caused by large railroad ballast road material at this turnaround location, the third and fourth test routes added an additional leg shown in **purple** to access an alternative turnaround road. The two detours were included to include washouts and other more extreme features due to the condition of these roads. The return unladen journey was recorded once, with the test dolly in the middle position, traversing the same roads but not necessarily the same features in the other traffic lane.

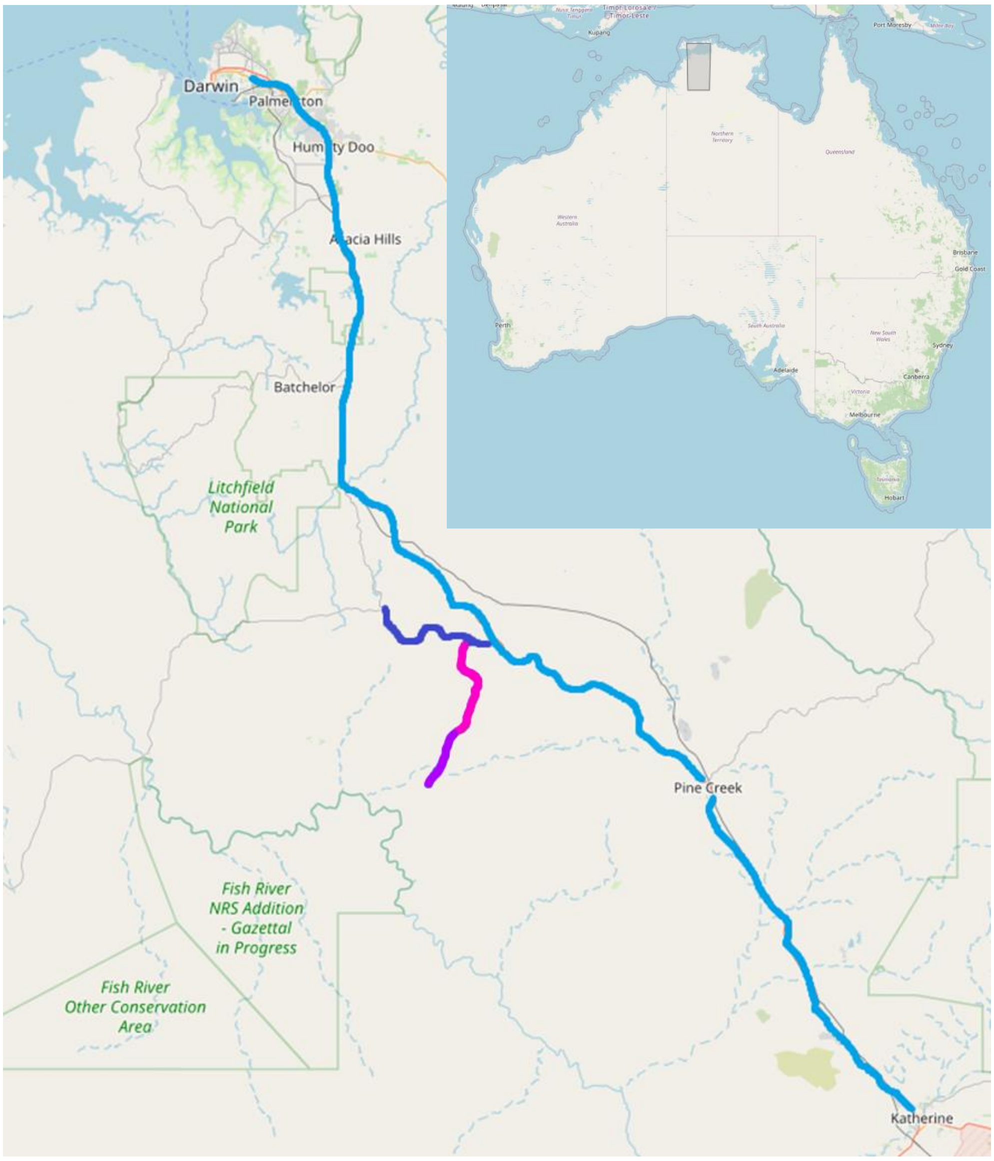


Figure 14 - The route from Berrimah, NT to Katherine NT, including detours to Robin Falls and Douglas-Daly

For the purposes of comparison and analysis, fifteen Stages were sub-selected from the more than 300 km long test route, due to containing features of interest and similarity of velocity between tests as compared with the broader dataset. Between Stages, the other segments have been named Transports. Stages and Transports were delineated/geofenced using GPS to ensure like-for-like road segments were compared.

FINDINGS: LONGITUDINAL FORCES ON COUPLINGS

The following section outlines the results and findings collected from the physical testing. The findings are laid out as a comparison between the theoretical forces based on the published D-value equations, and the loads physically seen by the couplings at each location along the vehicle. Several different methods have been used to analyse the results in order to understand the long-term significance of the forces.

Two methods common to vibrational analysis were used to order data and understand their frequency and magnitude. A further method of counting the threshold exceeding forces was also conducted to cover the distribution of high forces.

All methods were to undertaken to gain a better knowledge of the types of energy events seen at the couplings.

PIN TYPE COUPLING

Table 2 below shows the test GCMs for each test and calculates the required pin type D-Ratings based upon Equation 1, the theoretical D-Value equation.

Table 2 – Pin type D-Ratings for tested dolly positions

Test Number	2	3	4	1
Test Dolly Position	Front	Middle	Middle (Unladen)	Rear
Mass In Front of Coupling (t)	50.8	73.7	29.55	118.0
Mass Behind Coupling (t)	106.8	88.7	23.65	42.5
Gross Combination Mass (t)	157.6	162.4	53.2	160.5
AS2213 Pin Coupling D-Rating (kN)	203	237	77	184

The theoretical D-Value formula for a pin type coupling has a maximum at the mass centre of the combination and reduces at the front and rear of the combination.

The longitudinal force testing requirements for pin type couplings stipulate applying 2 million oscillations of magnitude ±0.6 x D-Rating. Based upon these assumptions, it would be reasonable to expect observed maximum forces to correspond with worst case theoretical forces for the pin type couplings at Front, Middle, Middle Unladen and Rear test dolly positions of ±121.5 kN, ±142.11 kN, ±46.2 kN and ±110.4 kN respectively.

FIFTH WHEEL COUPLING

Table 3 below shows the same tests as Table 2, but for the fifth wheel position which has a different mass before/after due to the mass of the dolly, and the D-Rating is based on Equation 2.

Table 3 - D Rating Requirements for fifth Wheel Coupling

Test Number	2	3	4	1
Test Dolly Position	**Front**	**Middle**	**Middle (Unladen)**	**Rear**
Mass In Front of Coupling (t)	54.7	77.6	33.5	121.9
Mass Behind Coupling (t)	102.9	84.8	19.8	38.6
Gross Combination Mass (t)	157.6	162.4	53.2	160.5
AS2213 fifth Wheel D-Rating (kN)	205	239	71	202

The theoretical D-Value formula for a fifth wheel coupling results in a similar curve to pin type couplings within the functional window of operation. Based on the longitudinal force testing requirements of AS4968.2, the theoretical worst case forces are ±0.6 x D-Rating, corresponding to ±123.0 kN, ±143.4 kN, ±42.6 kN and ±121.2 kN for the front, middle, middle unladen and rear dolly positions respectively.

GENERAL

Broadly, the data contained two types of loading:

- Moderate energy, high occurrence force oscillation, alternating about an approximate zero force average (excluding during elevation gain or descent)
- High energy, low occurrence 'peak' force events

Based on fracture mechanics fatigue theory, most coupling fatigue damage, contributing to coupling life reduction and risk of fatigue failure, occurs over threshold levels. These thresholds are only exceeded during occasional high energy events.

Figure 15 shows a sample of a typical high energy event to the left of the time window alongside more common moderate energy oscillations to the right. Notably, the 5th wheel overturning moments are largely unaffected by the by the high magnitude accelerations, as depicted on the graph. This observation is consistent with most of the testing data.

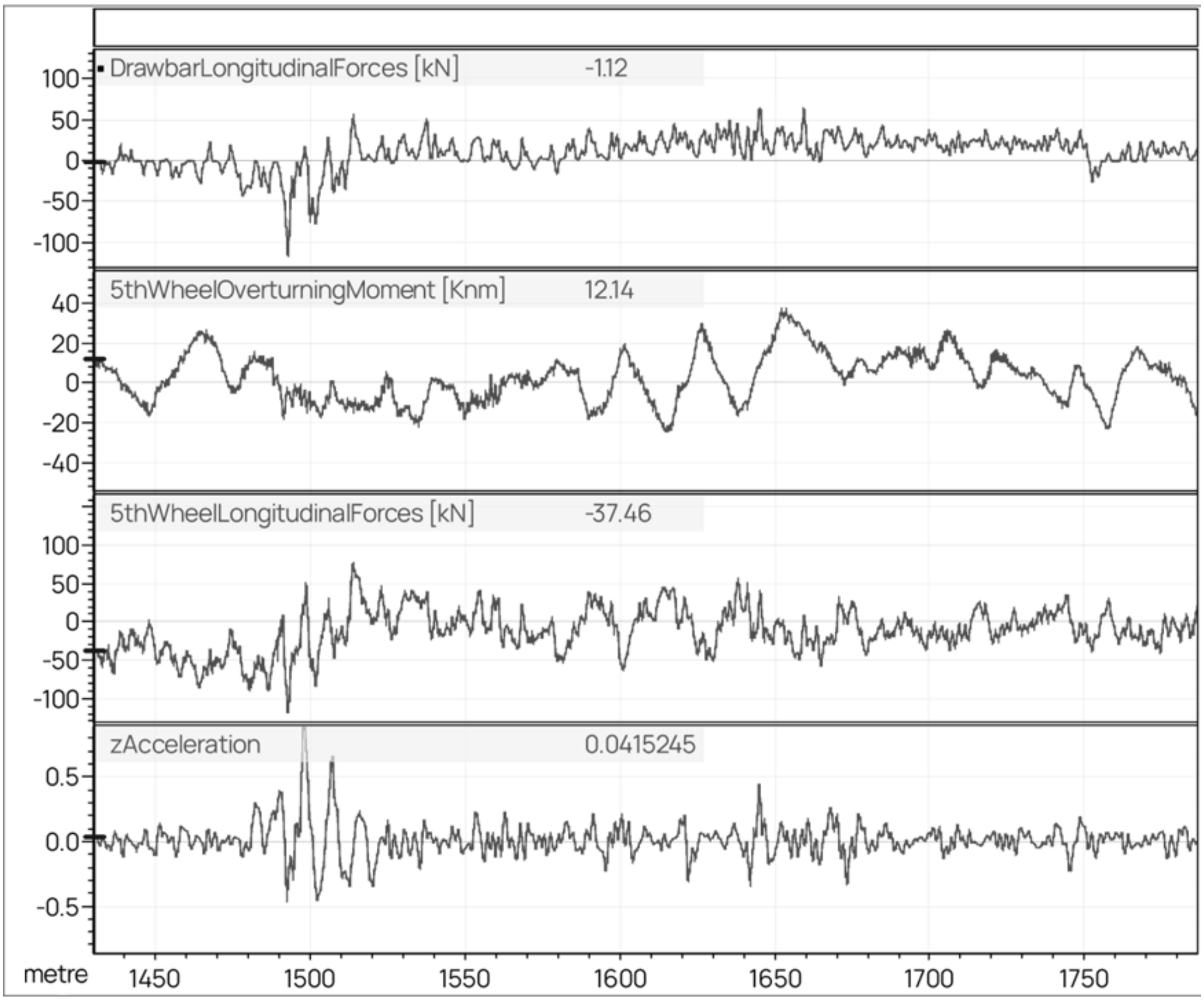


Figure 15 - Sample drawbar longitudinal forces, fifth wheel longitudinal forces and fifth wheel overturning moments plus dolly chassis vertical accelerations for a High Energy event alongside more common Moderate Energy oscillations

POWER SPECTRAL DENSITY

A Power Spectral Density (PSD) analysis over the complete range of the consistent test route is shown in the below results sections. A PSD is superior to a basic Fast Fourier Transform (FFT) as the relative quantity of samples in comparative channels does not influence the result.

PIN TYPE COUPLING

The PSD presented in Figure 16 implies a number of findings:

- A force oscillation peak ("mode") exists at around 1.55-1.65 Hz in all positions. This is of interest as heavy vehicle suspension vertical natural frequency is typically in the range of 1.5 Hz to 2.0 Hz, implying that relative vertical chassis movement between coupled units may be a contributor to coupling energy, however this does not necessarily imply it is a significant contributor to high energy events. An alternative possibility is that the dolly pitch mode is closely related to the suspension natural frequency and dolly chassis pitch stores and returns energy to the coupling system. This is supported by the dolly pitch axis gyroscopic mode, refer to Figure 17.
- The front position exhibited the strongest mode at 1.65 Hz followed by the middle position at a similar but lower-energy mode closer to 1.55 Hz, whilst the rear position did not exhibit a strong mode, implying that mass after the coupling is the primary variable influencing this behaviour. In accordance with the point made above regarding dolly chassis pitch storing energy, these forces may be most related to traction forces on the combination with increased traction forces leading to greater energy storage in dolly chassis pitch, resulting in a greater oscillation at or near the dolly chassis pitch mode.

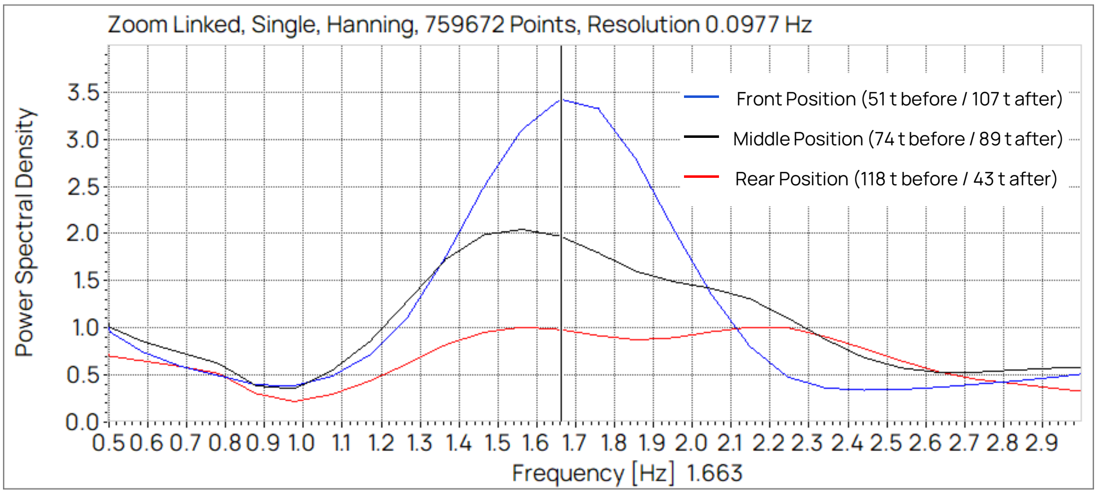


Figure 16 - Fast Fourier Transfominrm - drawbar forces (kN) for dolly position.

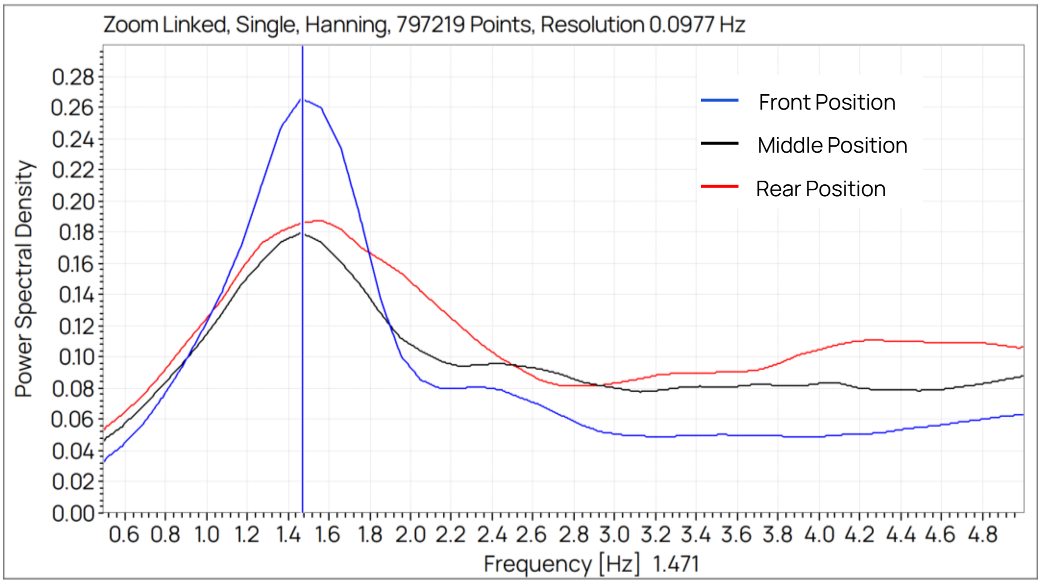


Figure 17 - Fast Fourier Transform of the test dolly pitch axis gyroscopic velocity (°/sec) for each dolly position

FIFTH WHEEL COUPLING

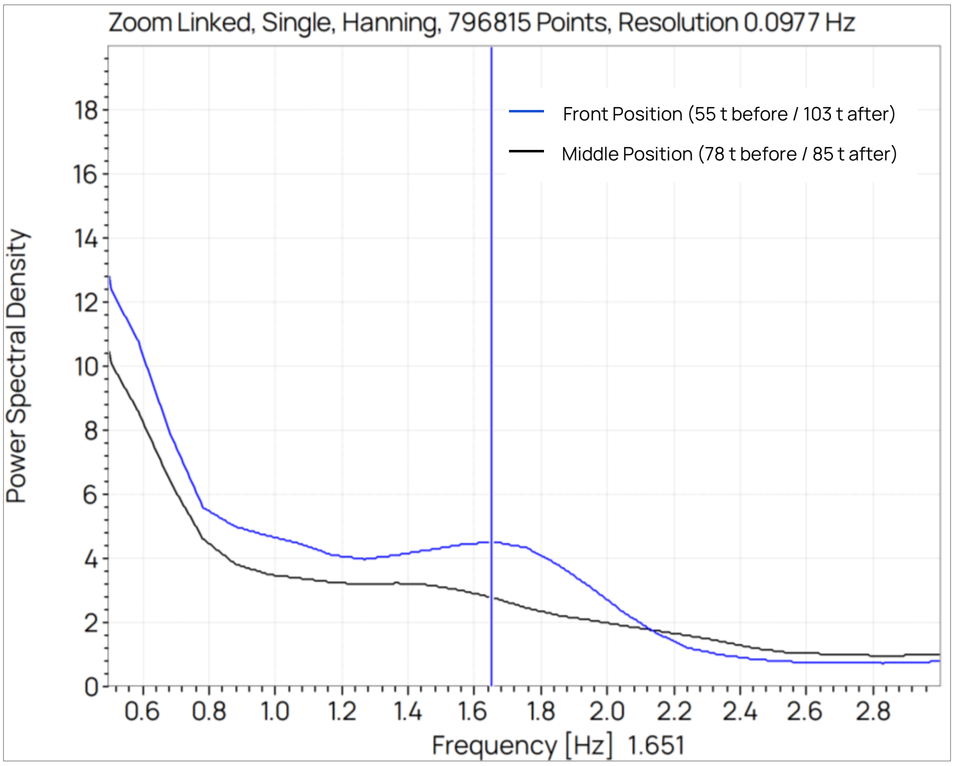


Figure 18 - Fast Fourier Transform of the fifth wheel forces for each dolly position

The PSD presented in Figure 18 implies that limited natural frequency mode is measured at the fifth wheel, although there is a weak result at 1.65 Hz in the front dolly position.

INTEGRAL OF ROOT MEAN SQUARE

To quantify the magnitude of long-term Moderate Energy force oscillation, absolute values and local minima and maxima are less relevant than the total amount of force and force oscillation seen by the coupling over time. Since the force measurement often oscillates about a zero-force or near-zero-force centre, a force time-average is not appropriate. In order to measure oscillating force, an integral of the RMS (Root Mean Squared, a concept borrowed from vibration analysis and signal processing and further explained in Equation 10 below) is taken of the drawbar longitudinal force, and the average of this integral of RMS is calculated across each stage from Stage 6 to 15. These stages were selected due to the consistency in road speed and conditions between the three tests. To provide context of road excitation input energy, average vehicle speed during each stage is also plotted.

$$RMS_{Longitudinal\ Drawbar\ Force} = \sqrt{\frac{1}{n}\sum_{i} Drawbar\ Longitudinal\ Force_i{}^2}$$

Equation 10 - Drawbar Longitudinal Force RMS

Results for the pin type and fifth wheel coupling are shown in Figure 19, below.

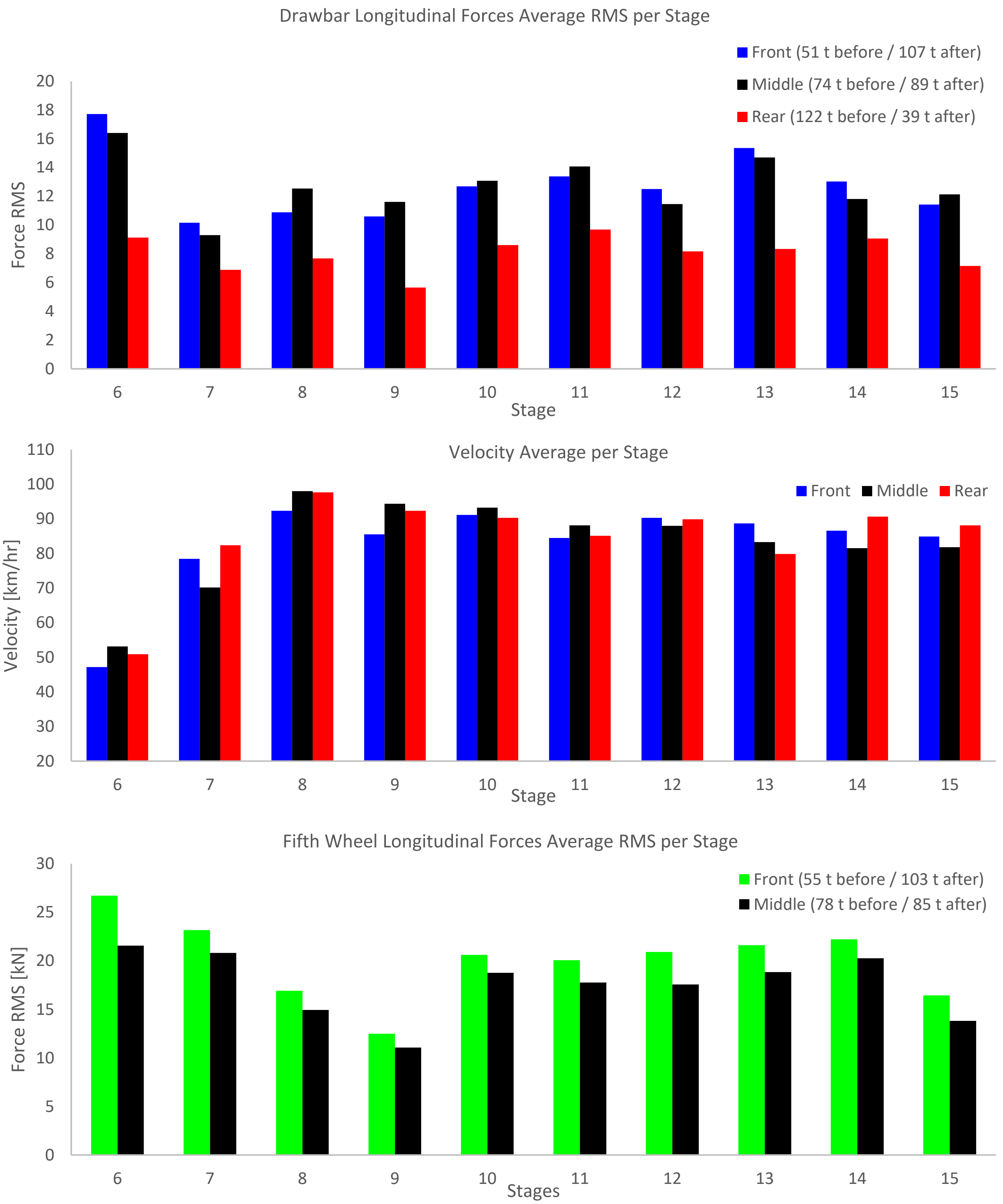


Figure 19 - Average drawbar and fifth wheel longitudinal RMS and vehicle velocity for Stages 6 to 15

Both the front and middle dolly position for pin type show an RMS result in the range of 10-18 kN RMS. Their relative strength compared with one another is directly correlated to average vehicle velocity, implying that they produce a near identical response to road excitation. The rear dolly position is the outlier in this case, at 6-10 kN RMS despite a similar average vehicle velocity.

The front fifth wheel position produces a higher RMS across all stages than the middle position. The rear position is not available due to the issues outlined in the section on *Fifth Wheel Orientation in Rear Dolly Position Test*.

The force in the fifth wheel is 40-50% higher than the pin type coupling for the same position, despite the theoretical equations' D-Ratings being very close between the coupling types.

Plain English:
According to the current standards, the couplings towards the middle should see higher forces. This was not reflected in an analysis of force– the test dolly in the front position often measured similar or higher force than in the middle.

Force isn't necessarily what causes coupling lifetime damage, so maximum/minimum forces also need to be considered in the sections coming up.

INTEGRAL OF ROOT MEAN SQUARE AND DISPERSED INDIVIDUAL EVENTS

The previous analysis is further aggregated in Table 4, below. By averaging longitudinal forces RMS for all stages at a specific coupling position, then dividing that by the average of longitudinal forces RMS for all dolly positions and stages (see Equation 11 below), then a single factor can compare the relative magnitude of a range of results.

$$Average\ Factor = \frac{\frac{1}{n}\sum_i a_i}{\frac{1}{n}\sum_i b_i}$$

$a = Longitudinal\ Forces: All\ stages, specific\ position$
$b = Longitudinal\ Forces: All\ stages, all\ positions$

Equation 11 - Average factor

Since individual events can exhibit very high singular force spikes without significantly impacting an RMS measurement, but likely contribute to coupling lifetime damage, a further comparison is included and is referred to as "Dispersed Individual Events". From the same stages in the same datasets, a range of minimum and maximum forces are gathered for specific events (chosen by author selection, seeking comparisons where vehicle velocity was closely matched in all tests) from each dolly position. This is then also aggregated through Equation 11.

PIN TYPE COUPLING

The above-described analysis is provided in Table 4 below, for the pin type coupling.

Table 4 - Summary of the RMS and Dispersed Individual Events for drawbar longitudinal forces

	Integral of RMS	**Dispersed Individual Events**
Dolly Position	Average Factor	Average Factor
Front (51 t before / 107 t after)	1.14	1.08
Middle (74 t before / 89 t after)	1.14	0.99
Rear (118 t before / 43 t after)	0.72	0.93

The results of Table 4 indicate that whilst RMS analyses suggest the 'energy' transmitted through the drawbar at the front and middle positions is near-equal, during notable events the front position experiences a higher force than the middle, which in turn experiences a higher force than the rear. This implies that the force magnitude for identical events increases as the mass after the pin type coupling increases, for the positions tested.

FIFTH WHEEL COUPLING

The same analysis is also provided for the fifth wheel below in Table 5. The absence of the rear measurement is detailed in the section on *Fifth Wheel Orientation in Rear Dolly Position Test*.

Table 5 - Summary of the RMS and Dispersed Individual Events for fifth wheel longitudinal forces

	Integral of RMS	**Dispersed Individual Events**
Dolly Position	Average Factor	Average Factor
Front (55 t before / 107 t after)	1.07	1.05
Middle (78 t before / 85 t after)	0.93	0.96
Rear (122 t before / 39 t after)	N/A	N/A

The results indicate that there is very good correlation between the RMS of the forces and the Dispersed Individual Events, and the Front position experiences moderately higher forces.

Plain English:

Over a longer term, for a pin type coupling the front and middle dolly positions have similar force.

In direct comparisons on the same road bumps, the front position sees a higher force on average.

The rear position is lower over the longer term and in direct comparisons.

On the fifth wheel, the front dolly position sees higher forces than the middle both over the long term and in direct comparisons.

Results for the rear coupling are not available due to a setup error.

COUNT OF FORCES EXCEEDING THRESHOLDS

Force magnitudes exceeding a threshold contribute to fatigue damage, which in turn can limit the life of the coupling and increase the risk of coupling failure while in service. Low force magnitudes for the high grade steel components in use in road train couplings do not influence coupling fatigue life, but may contribute to wear, which in turn can limit coupling life by virtue of physically removing material until eventually the coupling manufacturer's wear limits are reached.

An algorithm was applied to the datasets to count how many instances of forces crossing a series of thresholds were incurred, whilst the combination was travelling at more than 10 km/h. This was then divided by the distance of the dataset, to normalise to events/1000 km. It should be noted that:

- Whilst the identical routes were compared, speeds across features differ in some instances as a result of testing in real world conditions with traffic influence and unscheduled stops. The authors contend that the size of the input dataset minimises the influence of these factors.
- Long duration high force events were not witnessed, only transient peaks, so counting when a longitudinal force channel moves from below a threshold to above is offered as the most appropriate count of events rather than "samples above threshold" which can be influenced by double peaks.
- The bins are not exclusive, i.e. force events exceeding higher thresholds are included in the counts of lower thresholds, except where the measurement has oscillated above and below a higher threshold without dropping below a lower one. This is how some higher thresholds can record larger counts than lower thresholds.

Maximum thresholds exceeded are critical (horizontal axis), as is the frequency with which this occurs (vertical axis).

For each coupling type, two comparisons are presented. The first is a comparison across the entire dataset available but some unique features are present in each test. The second is a comparison solely across Stages and Transports of the test which presented consistent velocities and conditions between tests, namely Stage 6 onwards.

Very high magnitude forces were observed associated with combination "shunting" activities, which include braking, low speed dithering between acceleration and braking (i.e. crawling) and acceleration from a standstill. To quantify these, a count was also undertaken of how many instances were incurred of crossing force thresholds whilst travelling at less than 10 km/h. In this measurement however, as the datasets contained differing durations spent under 10 km/h, the normalisation is per 10 minute period at less than 10 km/h velocity, but greater than 0 km/h.

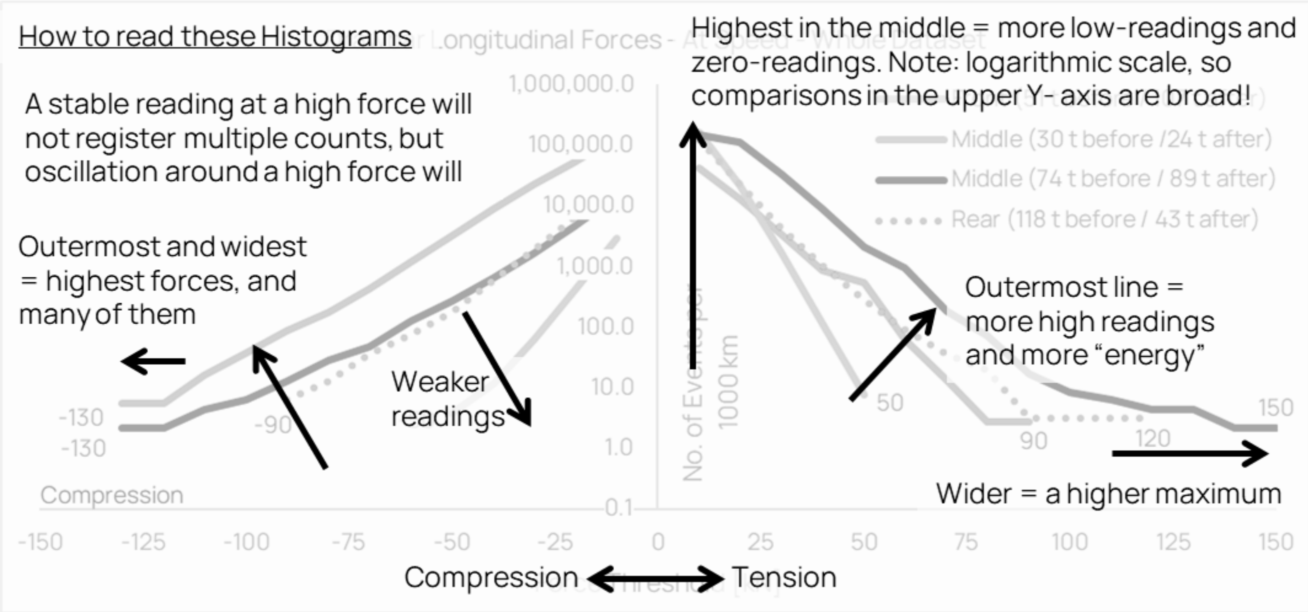


Figure 20 - Force Threshold Histogram interpretation guide

PIN TYPE COUPLING

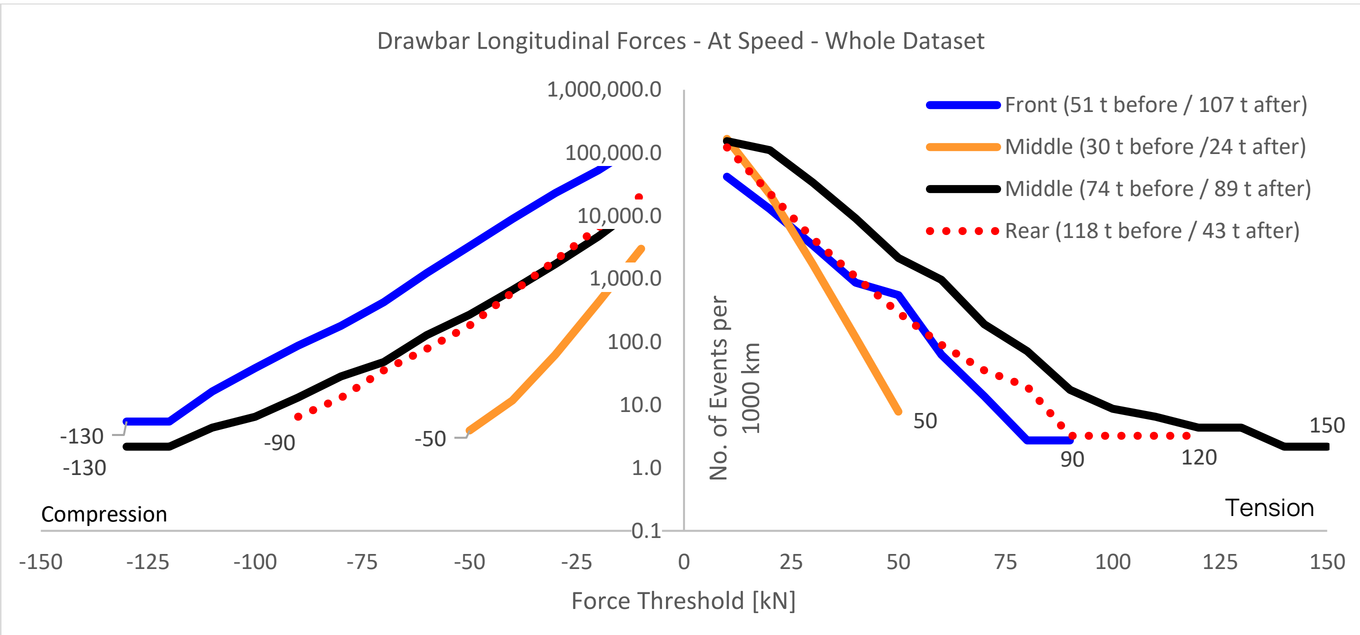


Figure 21 – Number of at speed (over 10 km/h) drawbar longitudinal force events exceeding force thresholds, per 1000 kms - whole dataset

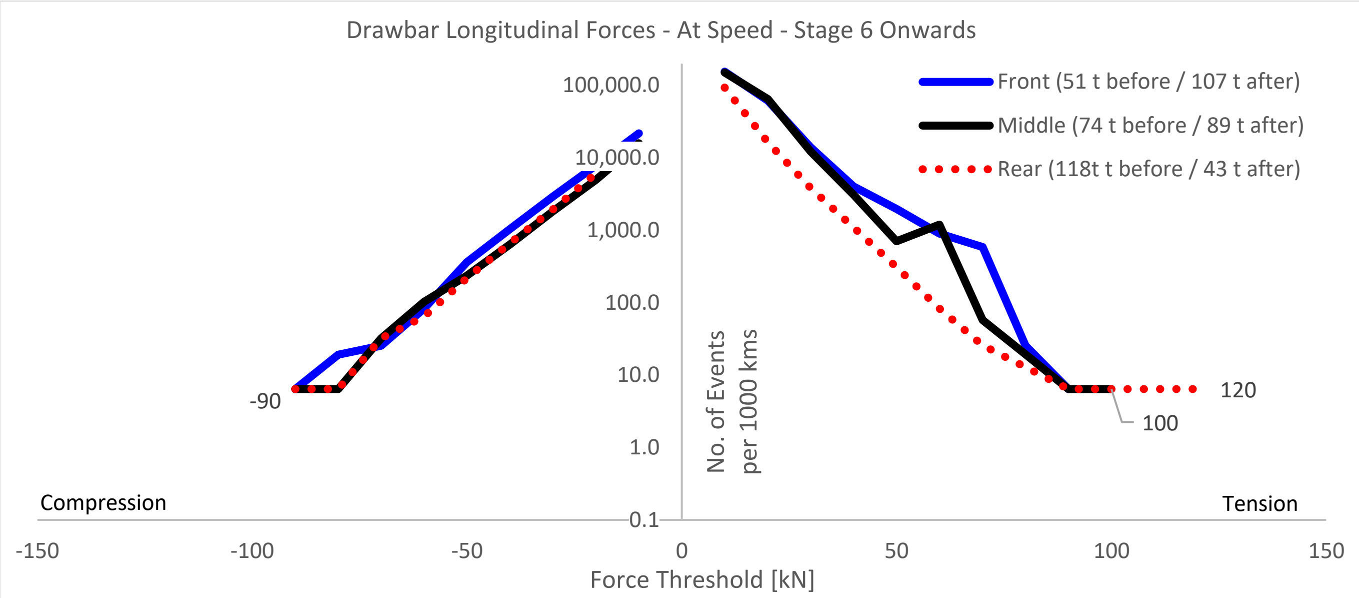


Figure 22 - Number of at speed (over 10 km/h) drawbar longitudinal force events exceeding force thresholds, per 1000 kms – Stage 6 onwards (high inter-test consistency).

Both the whole dataset (Figure 21) and restricted dataset analyses (Figure 22) present a strong compressive result for the front position, and a strong tensile result for the middle and rear positions.

Figure 23 indicates that in low speed conditions, the front dolly position experienced higher compressive forces than those in the middle or rear dolly positions, but these were not greater than the at-speed high range forces.

The mass of the combination in the unladen test was 33% of the mass of the laden test. The relationship of the peak forces in both the tensile and compressive directions is approximately linearly proportionate to the mass. Only the whole dataset comparison is made (Figure 21) as the unladen vehicle was not run across the restricted dataset in the same direction.

At low speed, the relationship of peak compressive forces was also approximately proportionate to combination mass, however tensile forces were comparatively low in both laden and unladen scenarios (Figure 23).

Tensile forces were in a similar range to those seen at speed; for the middle and rear dolly positions, this was not significantly different in maximum magnitude between findings at speed and at low speed. For the front dolly position, a significantly stronger result of tensile forces was observed at low speed compared with at speed, but this did not exceed the compressive at speed result. Accordingly, the envelope of forces is not extended significantly by low speed events across the whole dataset, as compared with events at speed also across the whole dataset.

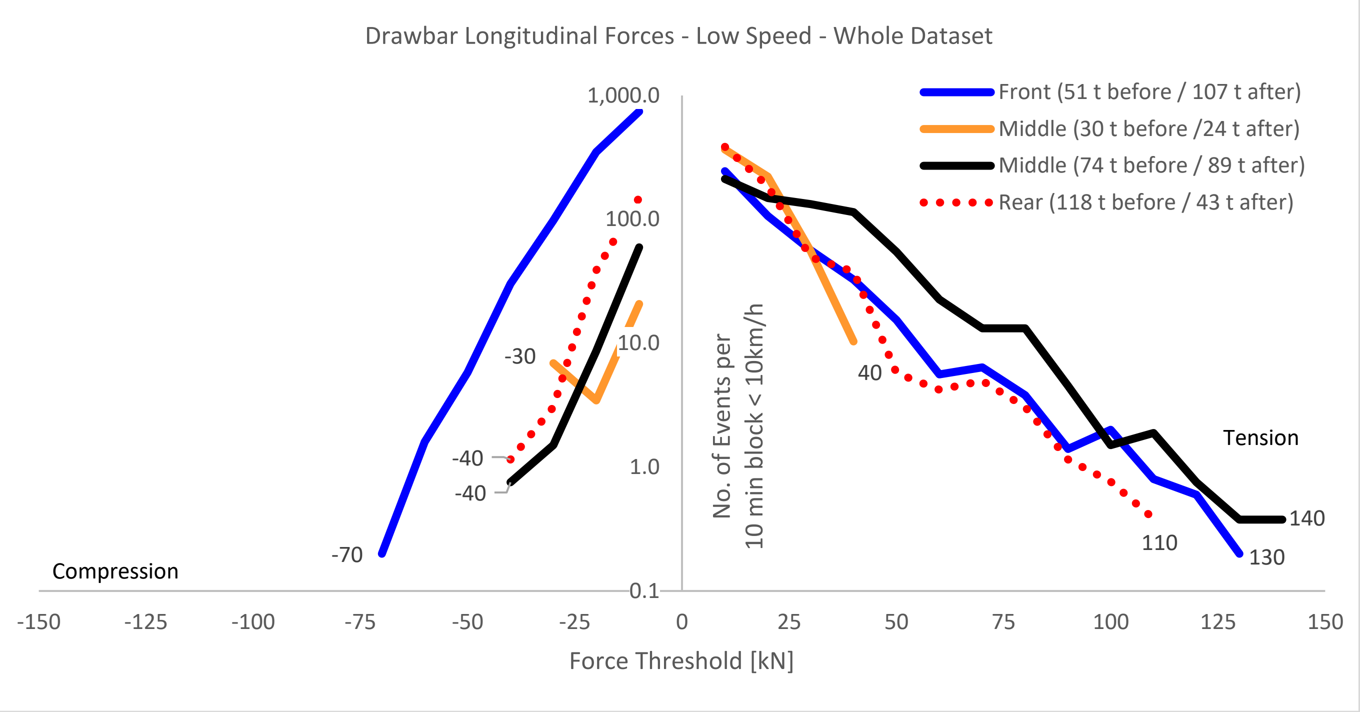


Figure 23 - Number of events exceeding force threshold per 10 minute block at a velocity of less than 10 km/h – whole dataset

FIFTH WHEEL COUPLING

Under free conditions the middle dolly position symmetrically exceeds higher maximum force thresholds than the front dolly position (see Figure 24), whilst under controlled comparative conditions the front position symmetrically exceeds slightly higher maximum force thresholds than the middle dolly position (see Figure 25). Refer to the section on *Fifth Wheel Orientation in Rear Dolly Position Test* regarding the absence of the rear dolly position measurements.

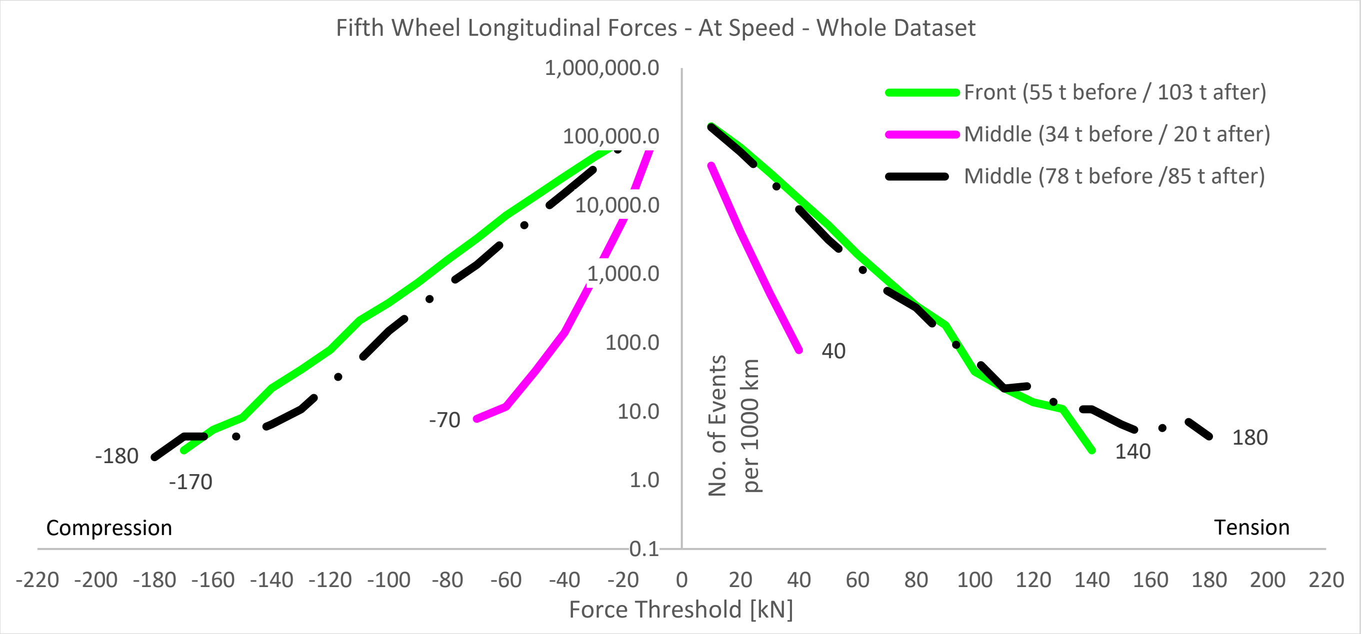


Figure 24 - Number of at speed (over 10 km/h) fifth wheel longitudinal force events exceeding force thresholds, per 1000 kms - whole dataset

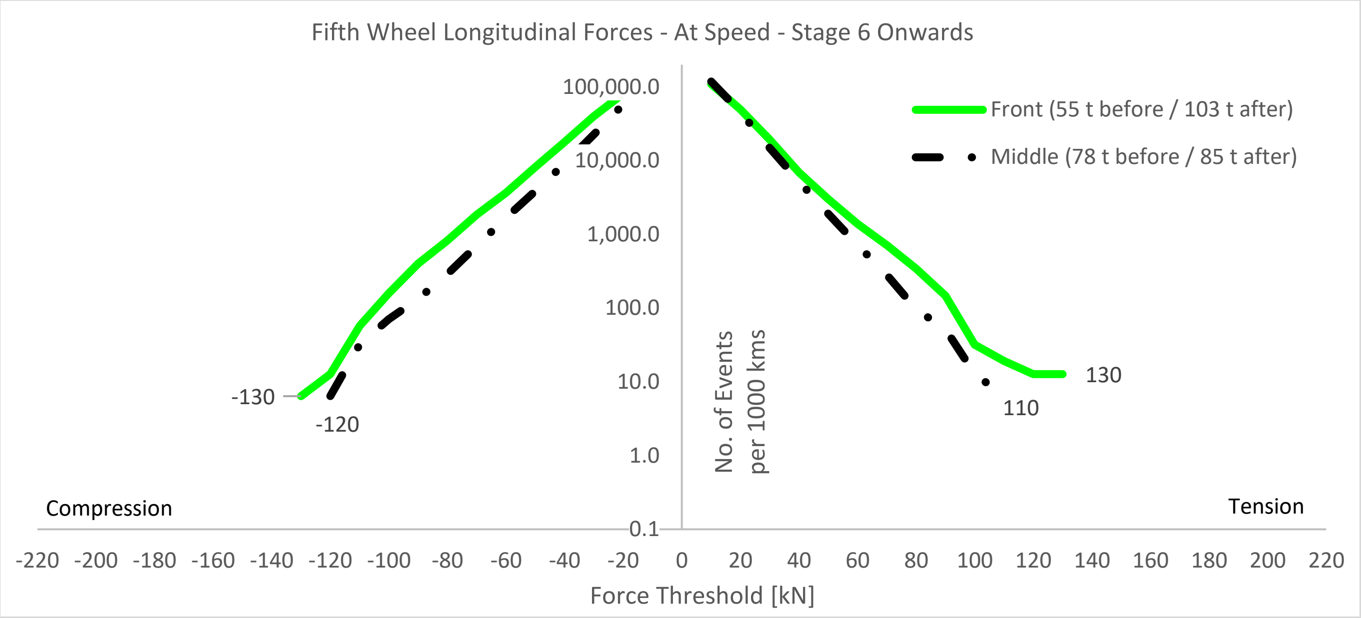


Figure 25 - Number of at speed (over 10 km/h) fifth wheel longitudinal force events exceeding force thresholds, per 1000 kms – Stage 6 onwards (high inter-test consistency)

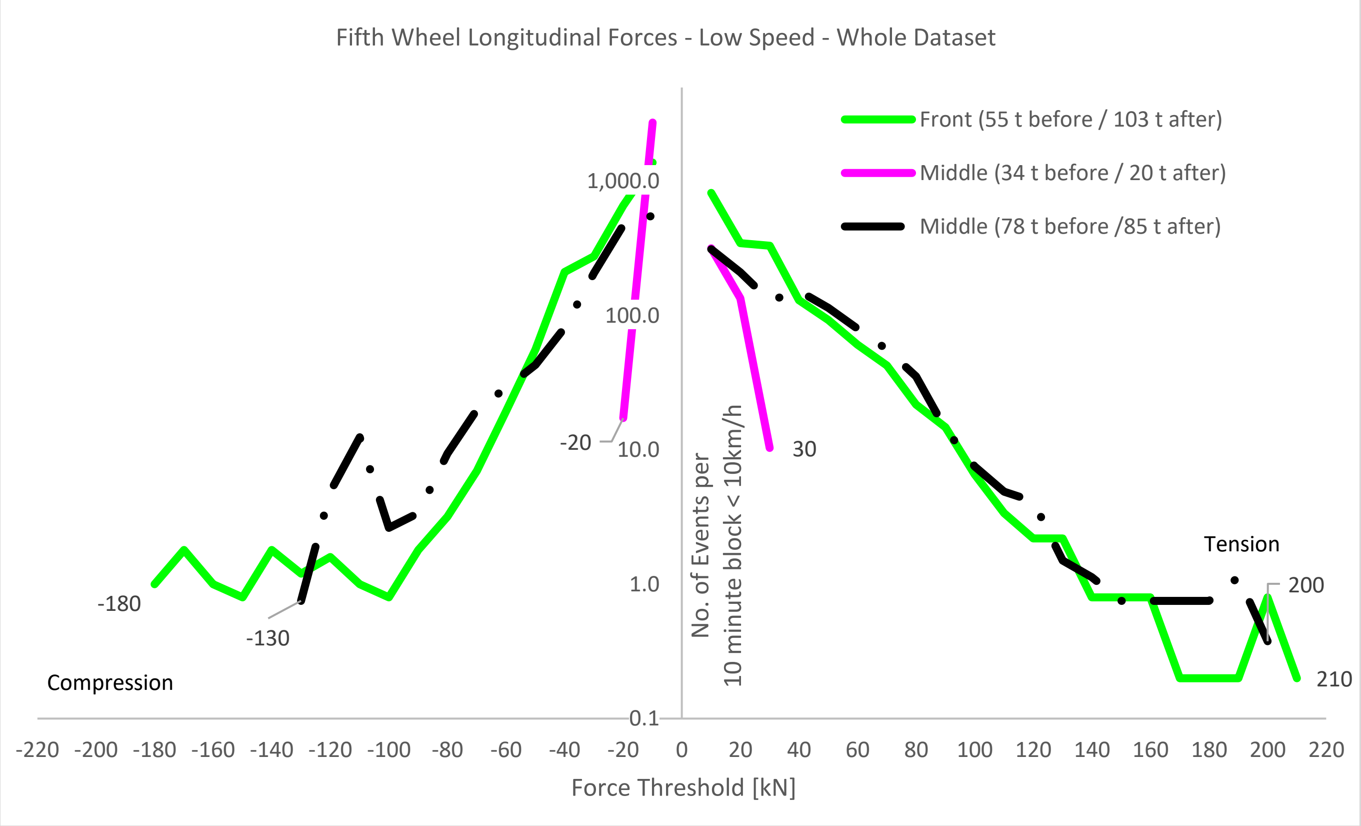


Figure 26 - Number of low speed (under 10 km/h) fifth wheel longitudinal force events exceeding force thresholds, per 10 minute block under 10 km/h – whole dataset

At the middle position, no forces were observed in excess of the D-rating.

In low speed conditions (refer Figure 26) very high tensile forces are encountered. Forces in excess of the D-rating of the front dolly position (207 kN) are observed.

Based on the probability found during the on-road testing, this force could be expected to be exceeded once every 50 minutes of low speed operation.

Plain English:

This testing shows that the highest forces that are routinely experienced by the coupling are when the vehicle is at low speed, typically less than 10km/h. Some of these measured forces are at a level that are in excess of the published D-Value guidance – possibly around once per hour of low speed operation.

This finding points to the importance of further research into low speed force magnitude and occurrence frequency.

In addition, means of controlling maximum levels of force within this operational window should be explored. These are discussed in more detail elsewhere in this report, and include consideration of controlling the maximum coupling fit tolerance, avoiding relative movement between trailers with more effective brake timing, and introducing mechanisms that serve to dampen the force applied to couplings.

OVERTURNING MOMENTS

Overturning moments present varying results between free and controlled test conditions. Under controlled conditions (Figure 28), the overturning moment did not vary substantially based on test dolly position. Under free conditions (Figure 27) similar results to the controlled conditions were observed in one direction, while higher results nearing 80 kNm were observed in the other.

For reference, AS4968.2 requires static testing of fifth wheels to an overturning moment of 112 kNm. Compared to observed forces, this results in a modest safety factor (approx. 1.35) incorporated in the Standards, particularly given that it is a dynamic load, and the measured forces during testing cannot be considered worst case.

Examples of raw overturning of data can be found in Figure 39 shows an example of a high overturning moment events.

The fifth wheel overturning moment measurements observed at low speed (Figure 29) do not exceed those observed at speed. Unladen measurements appear to be approximately proportionate to the ratio of laden to unladen trailer mass.

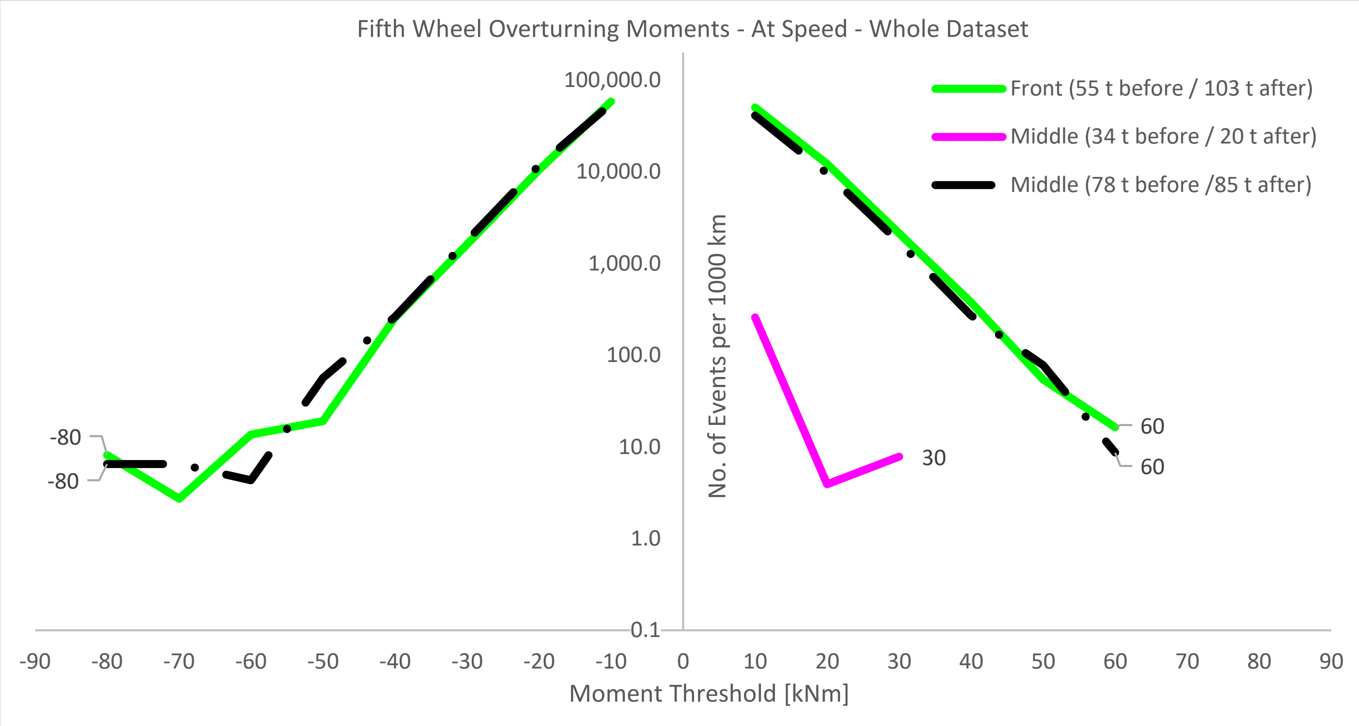


Figure 27 - Number of at speed (over 10 km/h) fifth wheel overturning moment events exceeding moment thresholds, per 1000 km – whole dataset

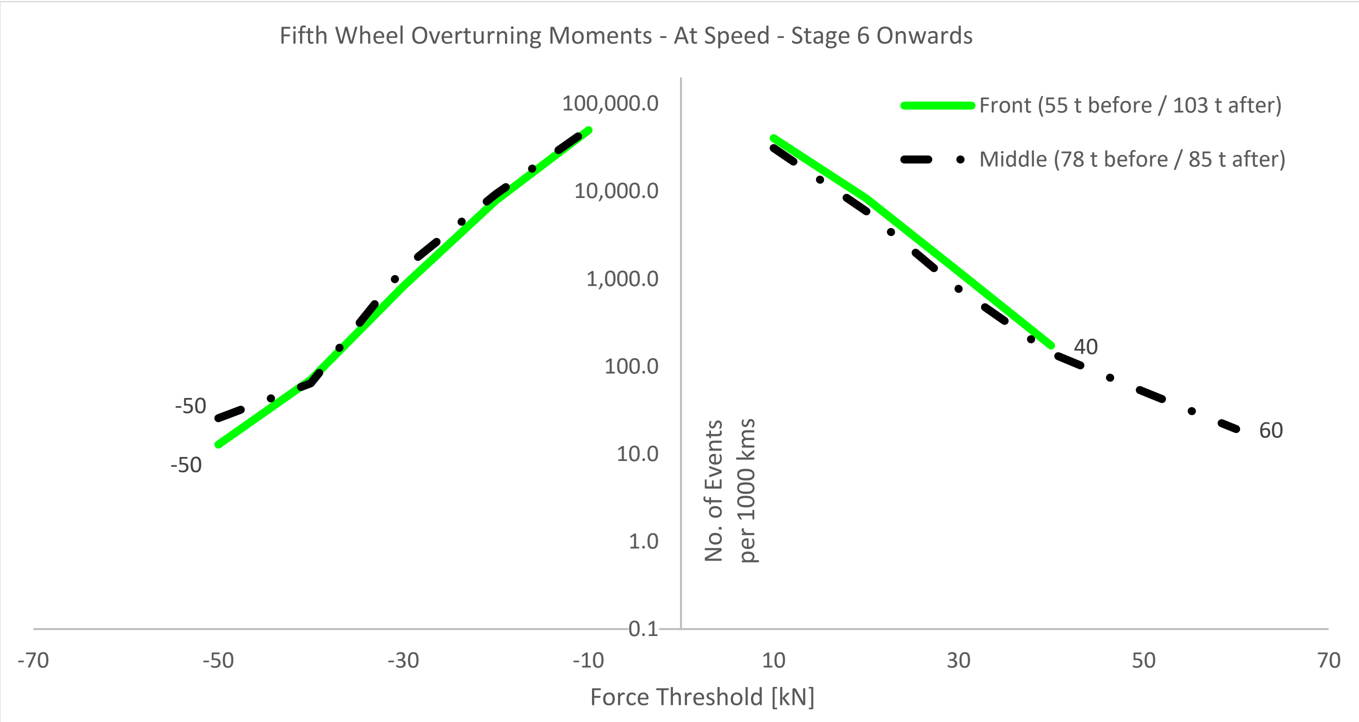


Figure 28 - Number of (over 10 km/h) fifth wheel overturning moment events exceeding moment thresholds, per 1000 kms– Stage 6 onwards (high inter-test consistency)

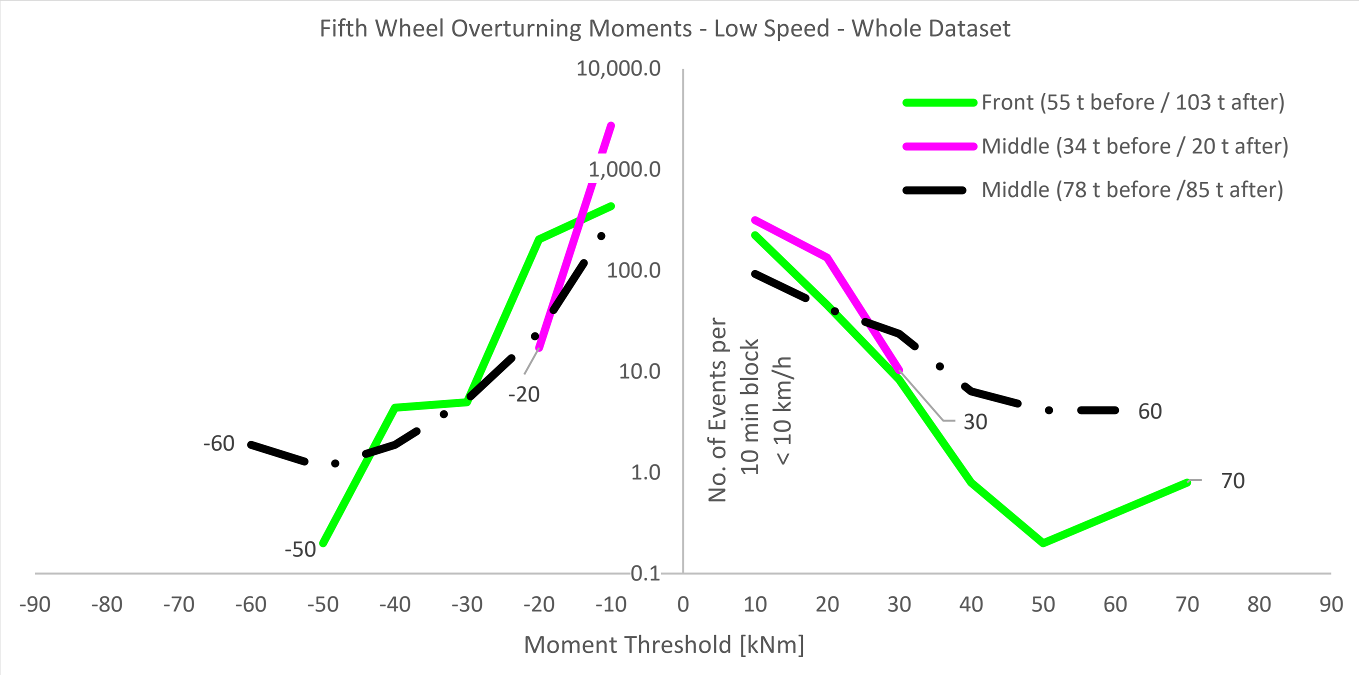


Figure 29 - Number of low speed (under 10 km/h) fifth wheel overturning moment events exceeding moment thresholds, per 10 minute block under 10 km/h – whole dataset

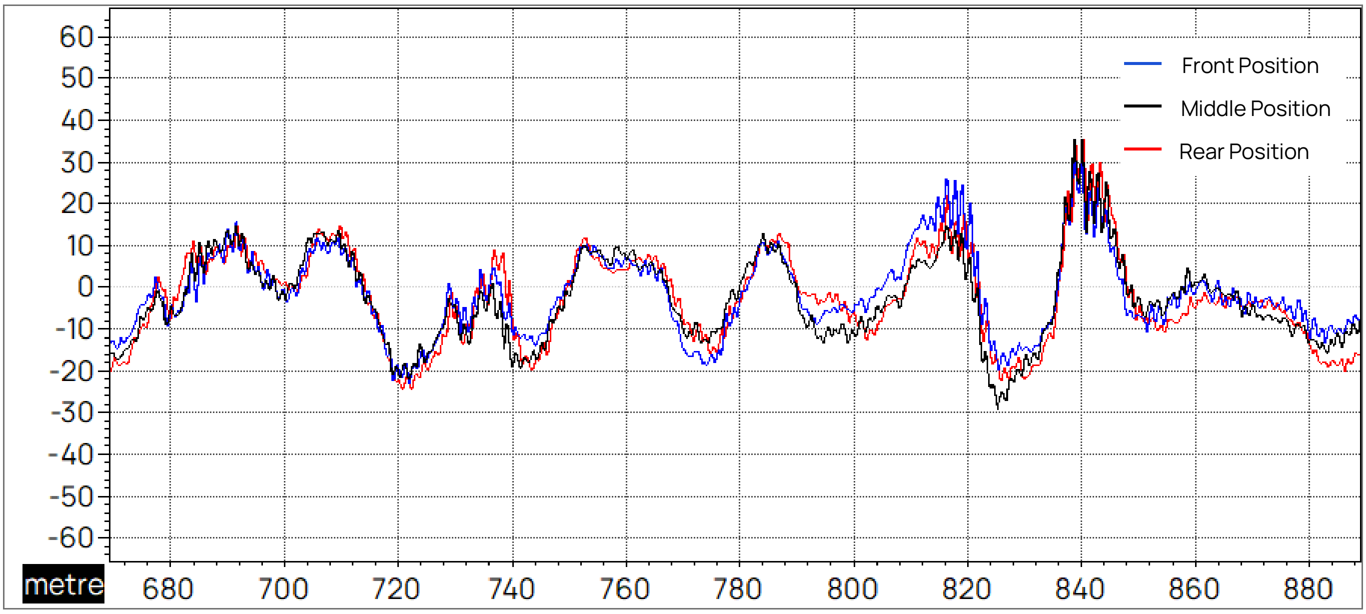


Figure 30 - Excerpt of fifth wheel overturning moment (kNm) data showing close correlation between dolly positions

ARTSA-I INSTITUTE

LONGITUDINAL DIRECTION FORCE - SUMMARY

Figure 31 below compares the forces exhibited to the required D-rating across the mass of the combination. As mentioned previously in the report, there are limitations in the data due to only being able to test two to three sampled points along the M1 curve. The trend shown on the coupling D-rating graph, indicating the largest forces towards the middle of the combination is not consistently reflected in the test data.

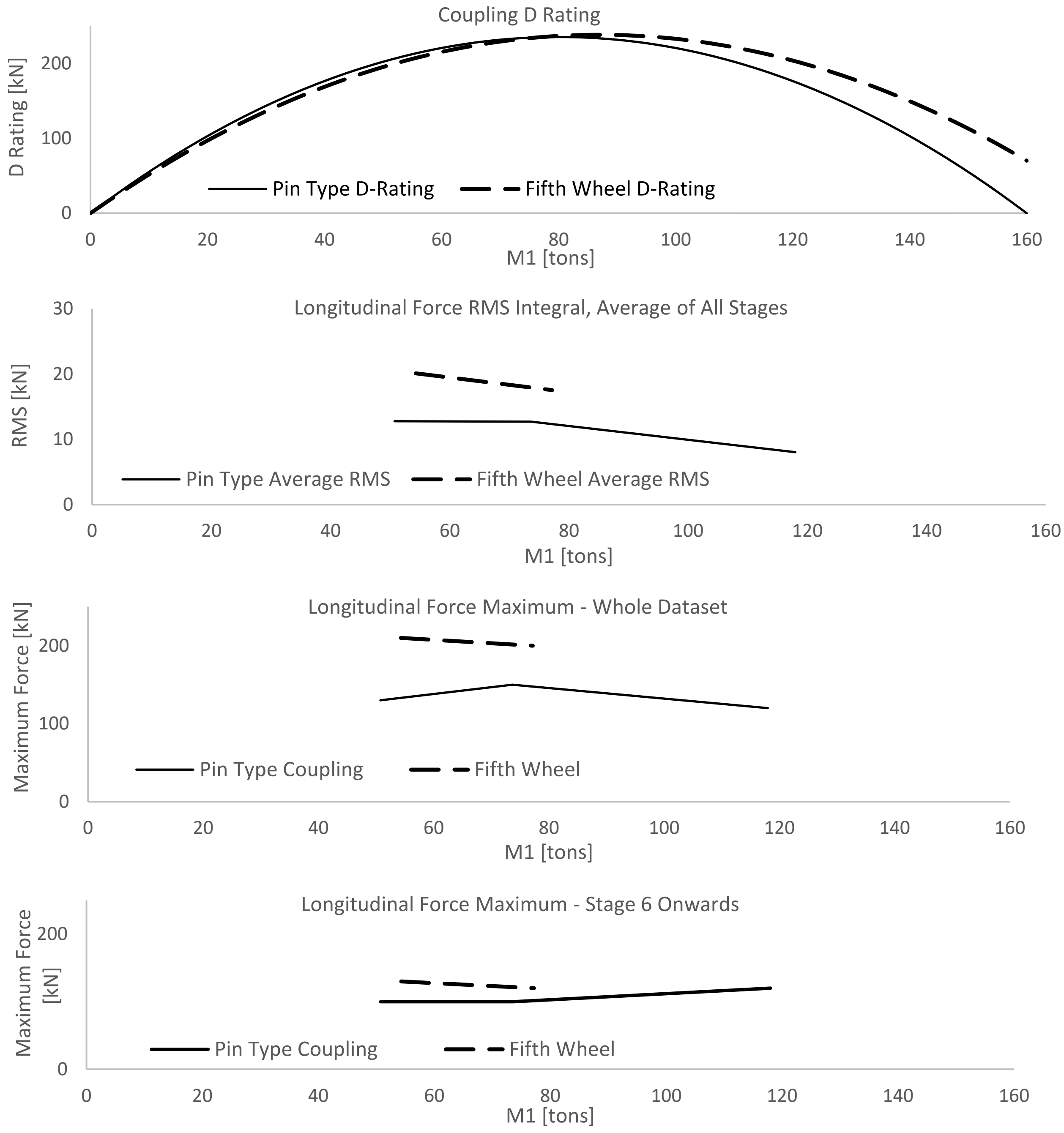


Figure 31 - Plot of theoretical D-Value against plots of laden Longitudinal Force measures

Plain English:
Over a longer term, for a pin type coupling the front and middle dolly positions have similar force but in direct comparisons on the same road bumps, the front position sees a higher force on average. The rear position is lower over the longer term and in direct comparisons.

On the fifth wheel, the front dolly position sees higher forces than the middle both over the long term and in direct comparisons. Results for the rear are not available.

FORCE VERSUS VELOCITY

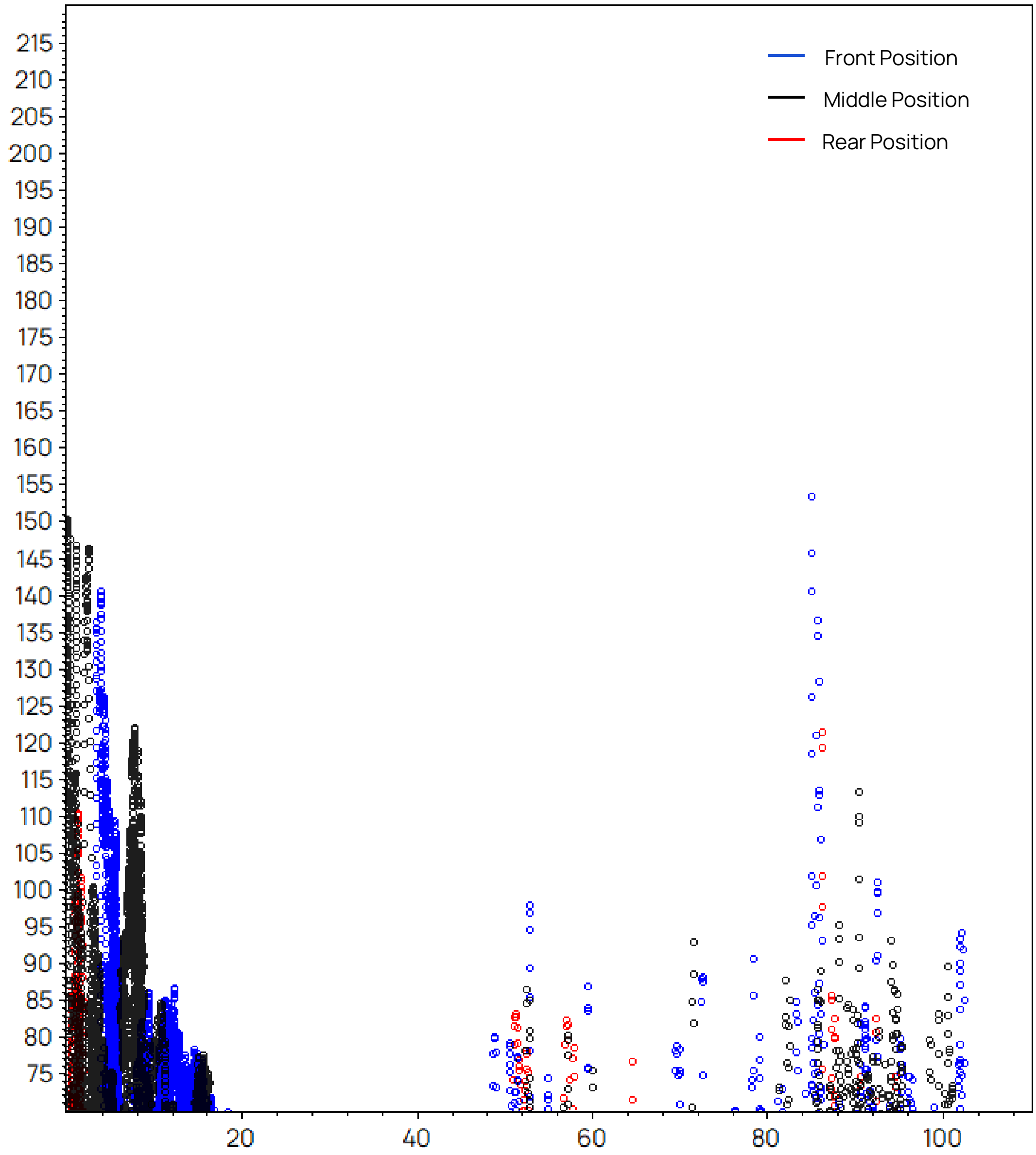


Figure 32 – Force (kN) – Velocity (km/h) plot for pin type coupling

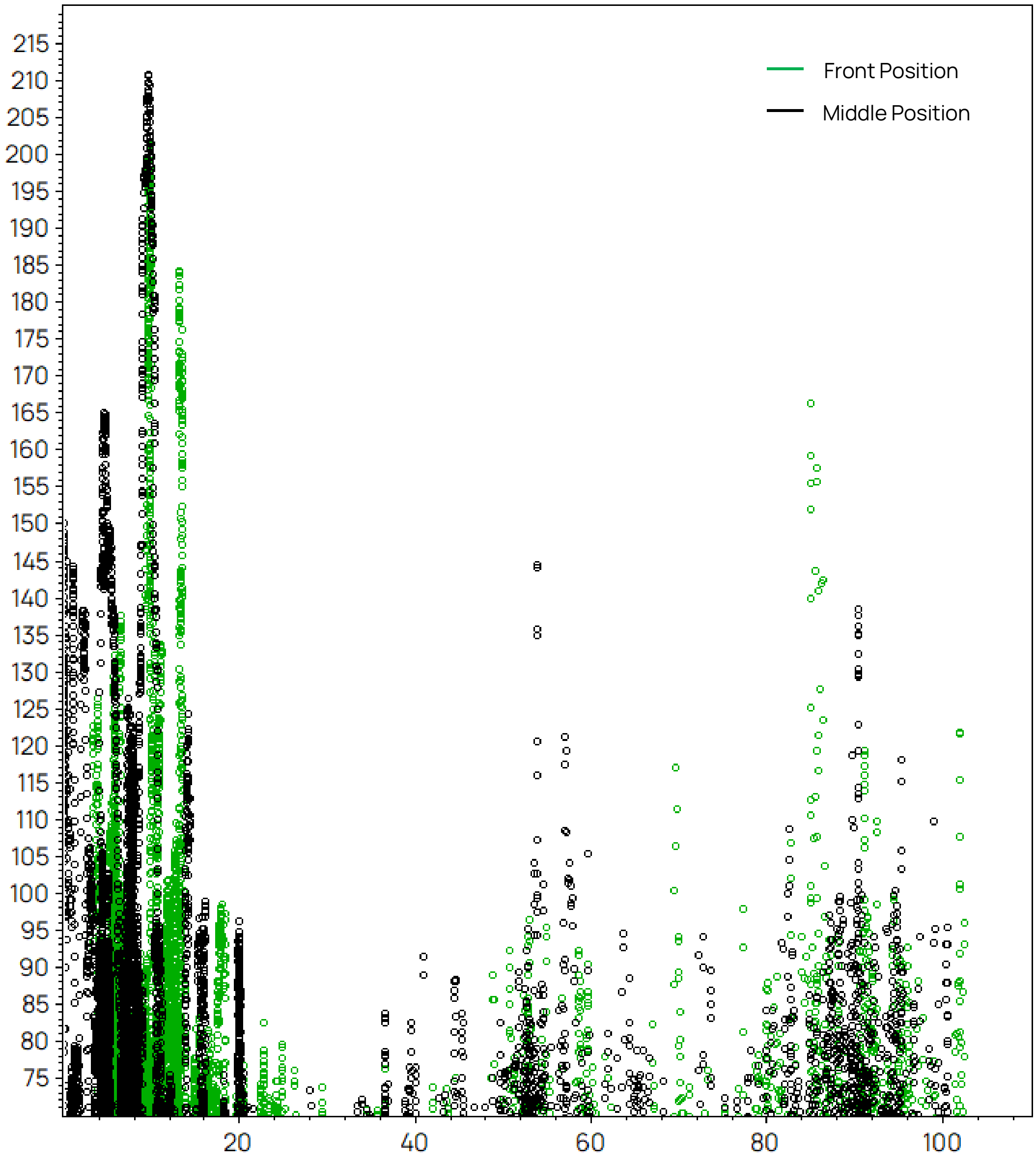


Figure 33 - Force (kN) – Velocity (km/h) plot for fifth wheel coupling

Based on the findings of prior international testing, and intuitively following kinetic energy calculation, it was hypothesised that there would be a squared relationship between vehicle velocity and longitudinal force. This was not observed; extreme forces frequently occurred at very low speed, and many peak at-speed forces occurred at lower speeds than the maximum. This observation applies to both the pin type (Figure 32) and fifth wheel (Figure 33) measurements.

The results also showed that the forces experienced by the fifth wheel coupling and the pin type coupling over the same excitation were often vastly different, often double the force in one vs the other. This is now understood to be a result of dynamic factors and complex excitation, including high frequency vibration adjacent to observed force peaks, as discussed elsewhere in this paper.

It is worth noting that the same road features were not crossed at a range of speeds in this testing, velocity was not an intentionally varied parameter in the gathering of 'operational' forces. This is the likely reason for not observing the force-velocity relationship observed elsewhere.

FINDINGS: LATERAL FORCES ON COUPLINGS

The lateral forces measured at the pin type coupling were minimal in comparison to the longitudinal forces. The highest observed lateral force scenario was during a low speed U-turn, reaching a peak force of 45 kN.

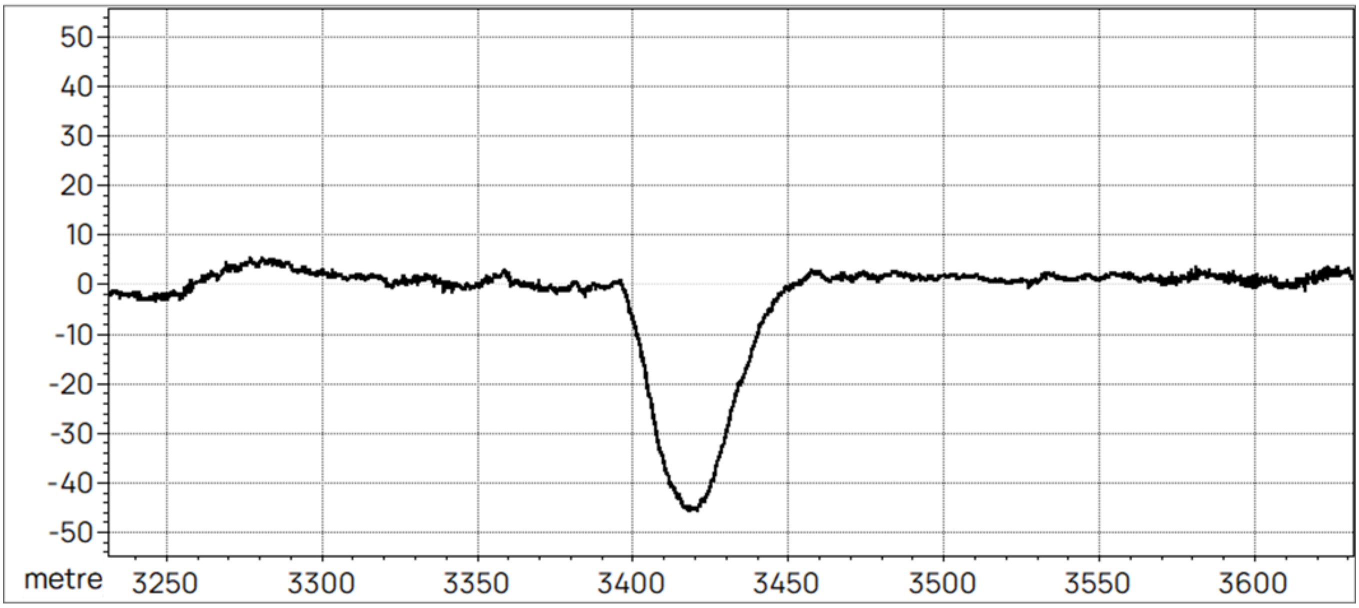


Figure 34 - Excerpt from test data showing the largest magnitude lateral force (kN) vs travel distance (m) on the drawbar

DOLLY POSITION:	FRONT
VEHICLE SPEED:	0 KM/HR
FIFTH WHEEL LONGITUDINAL FORCE:	146.21 KN
DRAWBAR LONGITUDINAL FORCE:	148.45 KN
OVERTURNING MOMENT:	-13.61 KNM
SCENARIO:	ACCELERATION FROM STANDSTILL
OCCURRENCE:	COMMON DURING STARTING ACCELERATIONS

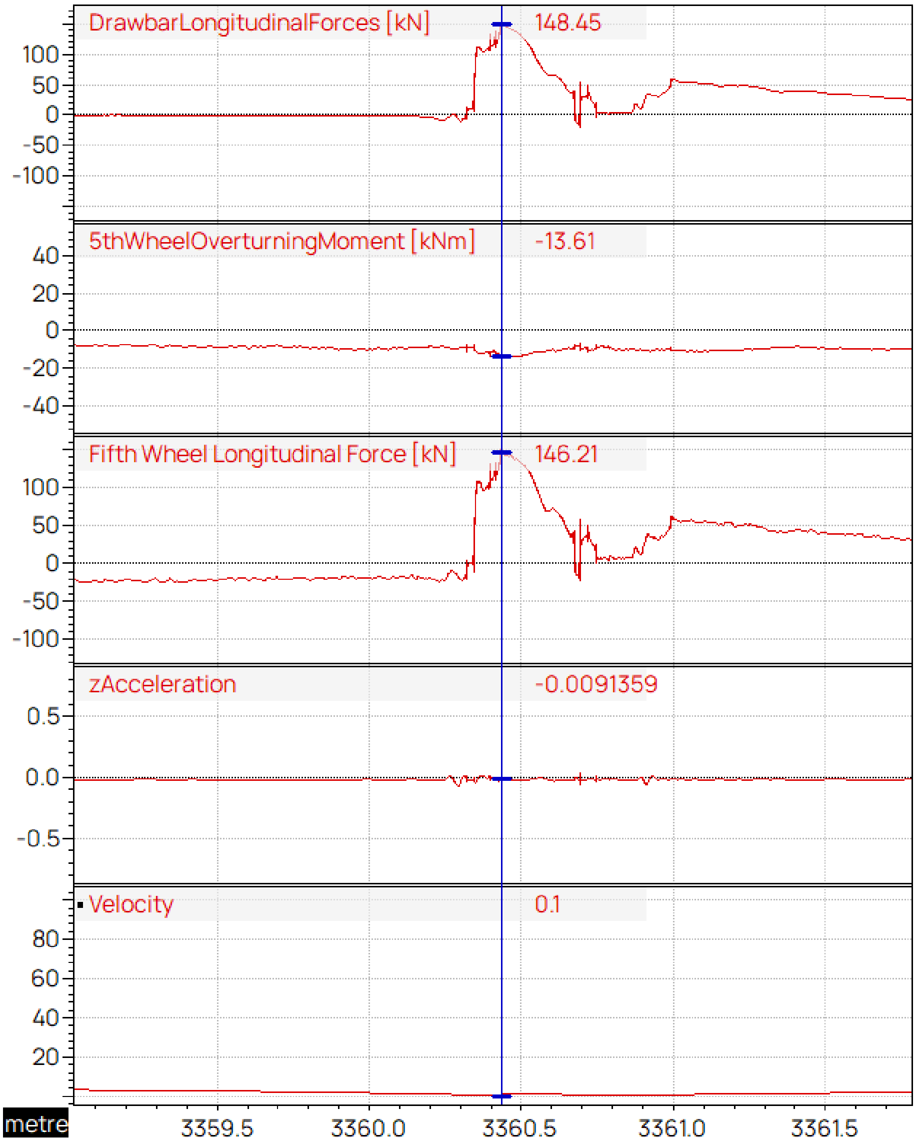


Figure 35 - An extract of measurements from a specific physical test event

DOLLY POSITION:	FRONT
VEHICLE SPEED:	95.3 KM/HR
FIFTH WHEEL LONGITUDINAL FORCE:	117.86 KN
DRAWBAR LONGITUDINAL FORCE:	75.53 KN
OVERTURNING MOMENT:	-24.45 KNM
SCENARIO:	TYPICAL ROAD EXCITATION (0.3G)
OCCURRENCE:	FREQUENT

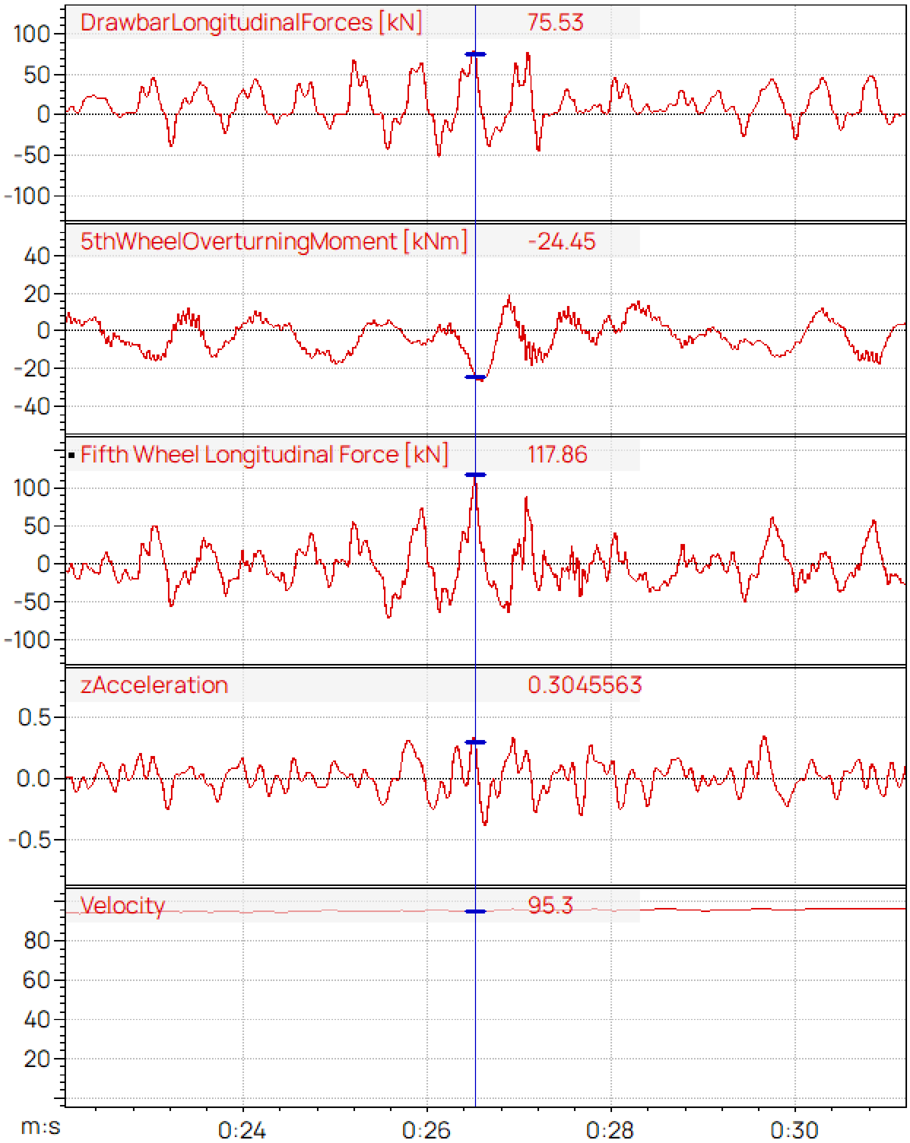


Figure 36 - An extract of measurements from a specific physical test event

DOLLY POSITION:	FRONT
VEHICLE SPEED:	59.5 KM/HR
FIFTH WHEEL LONGITUDINAL FORCE:	-169.65 KN
DRAWBAR LONGITUDINAL FORCE:	-67.77 KN (LOCAL MINIMUM: -119.80 KN)
OVERTURNING MOMENT:	49.24 KNM
SCENARIO:	TYPICAL LARGE ROAD EXCITATION (1G)
OCCURRENCE:	FREQUENT

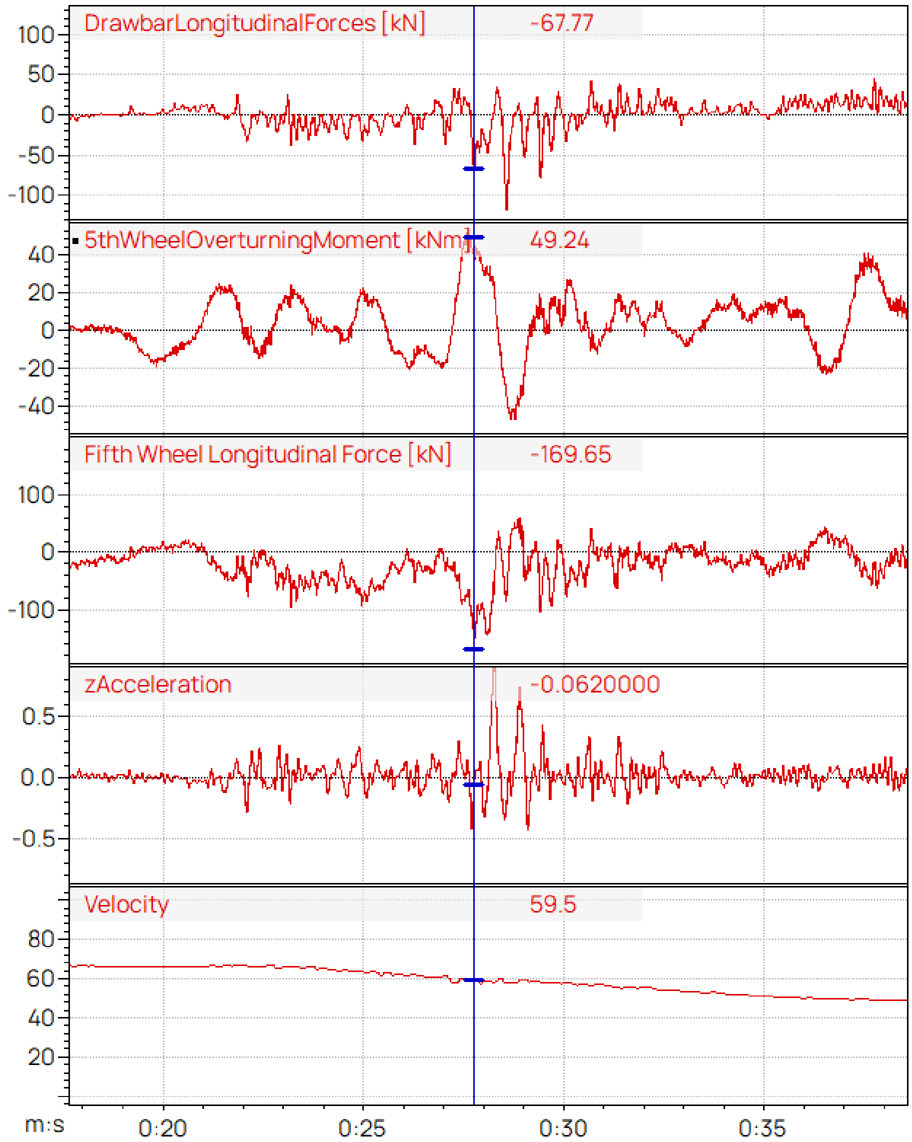


Figure 37 - An extract of measurements from a specific physical test event

DOLLY POSITION:	MIDDLE
VEHICLE SPEED:	84.9
FIFTH WHEEL LONGITUDINAL FORCE:	159.30 KN
DRAWBAR LONGITUDINAL FORCE:	153.39 KN
OVERTURNING MOMENT:	31 KNM (LOCAL MAXIMUM)
SCENARIO:	TYPICAL LARGE ROAD EXCITATION (1.2G)
OCCURRENCE:	FREQUENT

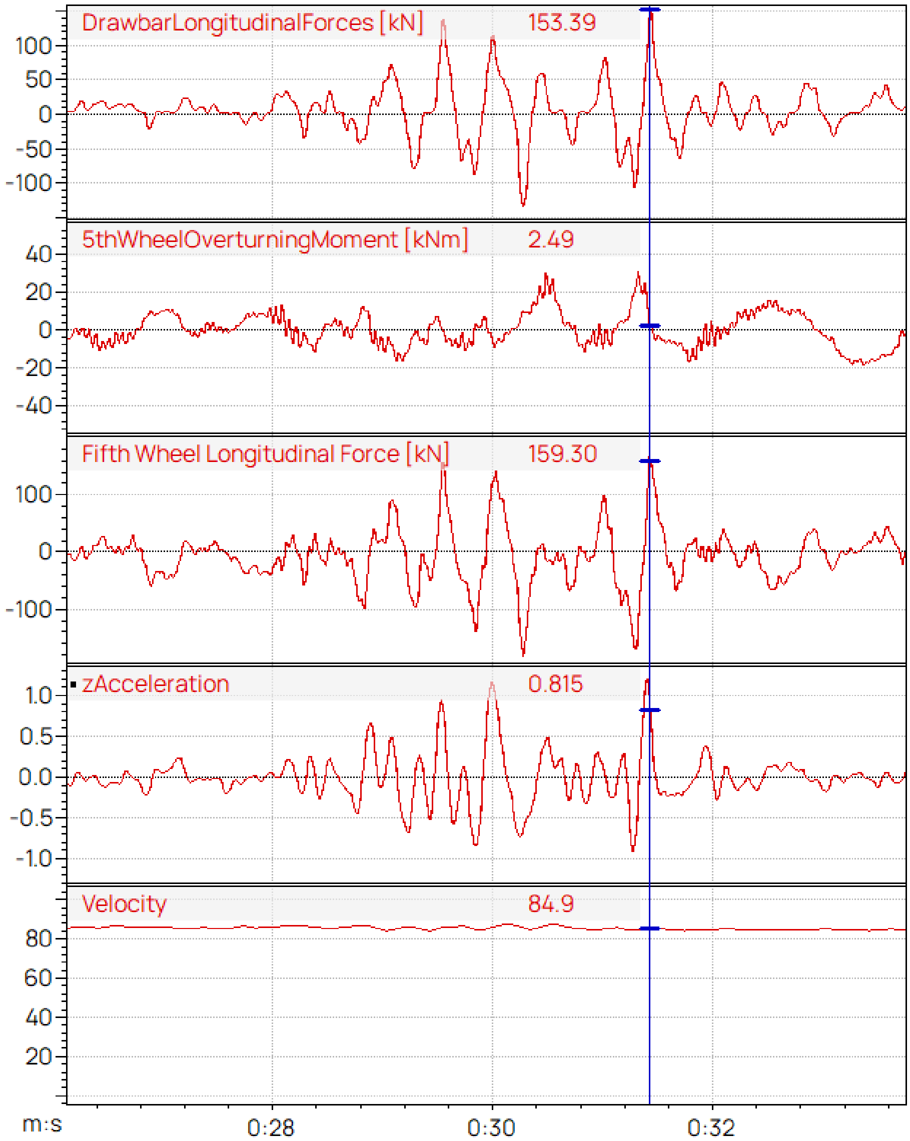


Figure 38 - An extract of measurements from a specific physical test event

DOLLY POSITION:	MIDDLE UNLADEN
VEHICLE SPEED:	72.7 KM/HR
FIFTH WHEEL LONGITUDINAL FORCE:	-49.3 KN
DRAWBAR LONGITUDINAL FORCE:	-55.11 KN
OVERTURNING MOMENT:	-14.70 KNM
SCENARIO:	LARGE LONGITUDINAL FORCE EXCITATION
OCCURRENCE:	MODERATELY FREQUENT

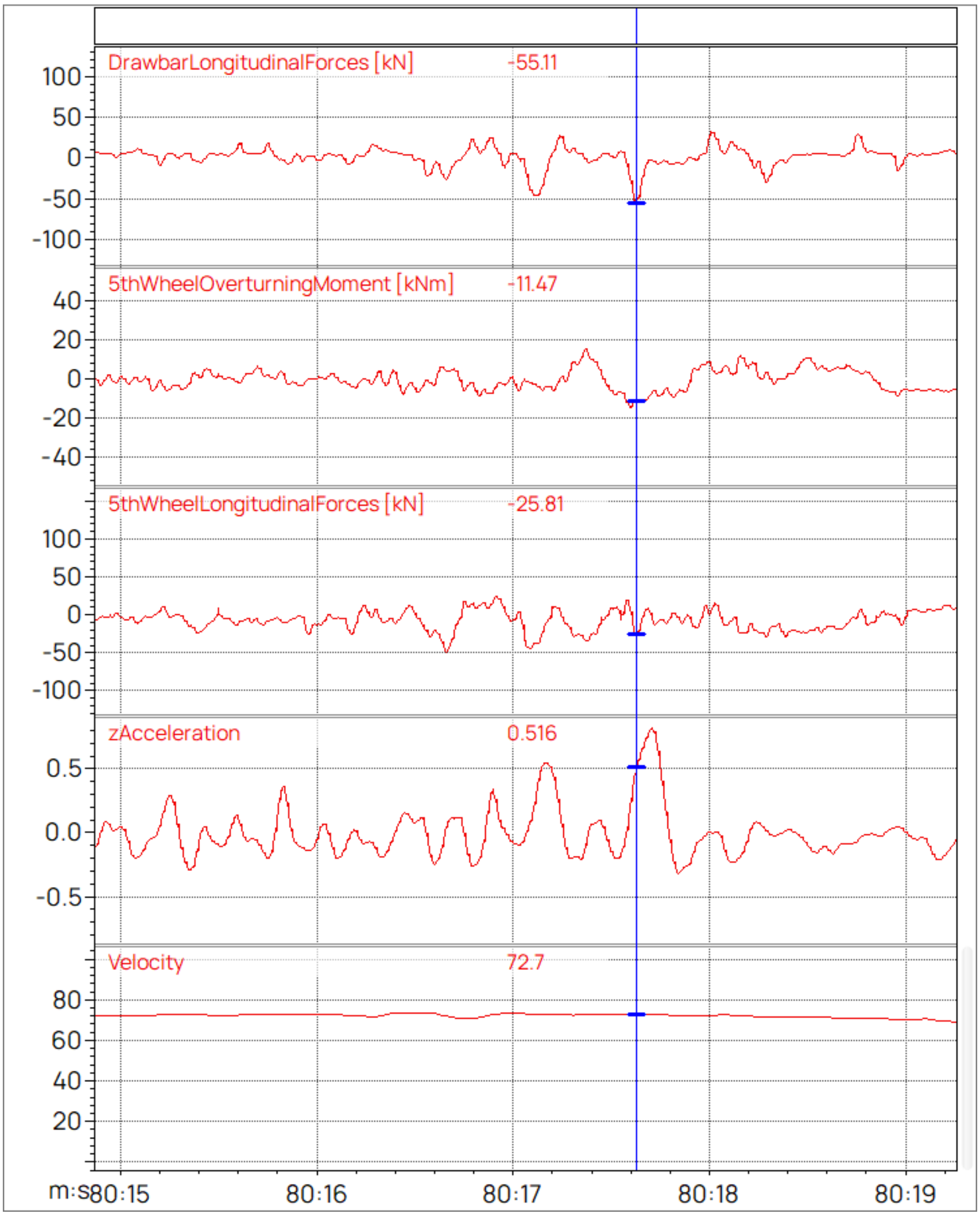


Figure 39 - An extract of measurements from a specific physical test event

DOLLY POSITION:	MIDDLE UNLADEN
VEHICLE SPEED:	14.4 KM/HR
FIFTH WHEEL LONGITUDINAL FORCE:	-70.39 KN
DRAWBAR LONGITUDINAL FORCE:	12.79 KN
OVERTURNING MOMENT:	30.31 KNM
SCENARIO:	LARGE FIFTHWHEEL OVERTURNING MOMENT EXCITATION
OCCURRENCE:	RARE

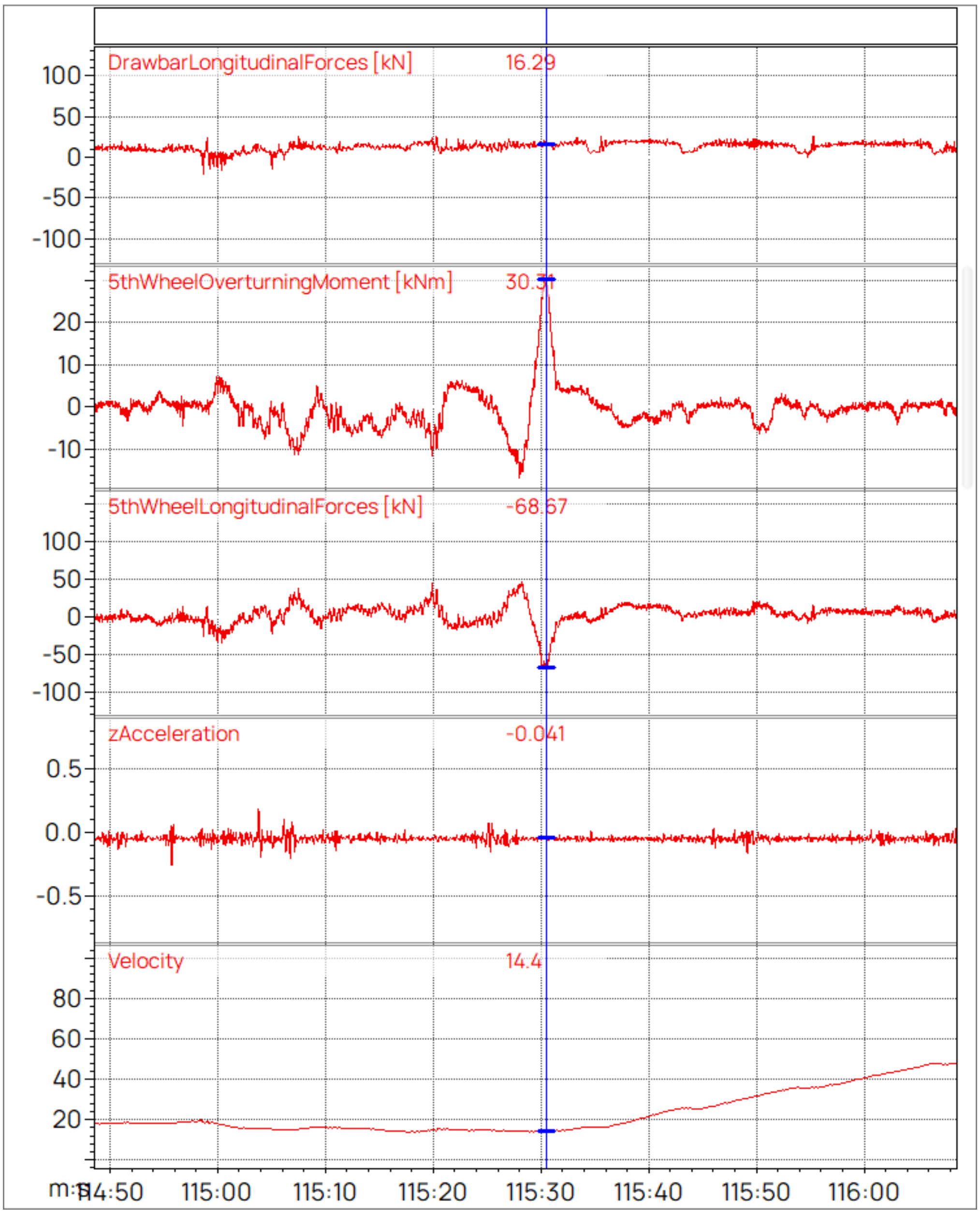


Figure 40 - An extract of measurements from a specific physical test event

REPORT – SOFTWARE SIMULATION

Some aspects of on-road testing were desirable but could not be accommodated within a practical test regime. In order to facilitate investigation of further factors, the on-road testing conducted was used to calibrate a multi-body computer simulation using proprietary software. These factors include:

- Simulation and plotting the relationship of GCM with coupling forces, namely at lower GCM than tested, and significantly higher GCM than tested, to research the mass-inertia relationship at extremes for D-ratings.
- Simulation on extremely harsh road features which are not safe or viable to repeatedly test an operator's vehicles on but which may be occasionally encountered in real world operations.
- Simulation over a range of vehicle combination types, ie mixed- and minimally-roll coupled.
- Simulation to research if the mass of a unit immediately adjacent the coupling is influential as compared with the mass of all units combined adjacent to a coupling.
- Simulation to research the function of road speed and coupling forces.
- Simulation to research the function of coupling lash and coupling forces.

SIMULATION CORRELATION

The multi-body simulation of a vehicle combination requires extensive inputs. These are:

- Vehicle assembly information:
  - The mass, three-dimensional location of the centre of gravity and moments of inertia of each discrete body;
  - The geometric arrangement of links between bodies;
  - The stiffness of joints connecting bodies; and
  - The rates of springs and damping of dampers.
- Driving manoeuvre information:
  - Road vertical profile and camber;
  - Vehicle velocity; and,
  - Steering controller behaviour.

These inputs must be very close approximations of the real physical quantities, otherwise the simulation will not correlate. For this simulation the source of the information was:

- Bill of lading for the vehicle combination based on weighbridge measurements to obtain axle group masses;
- Centre of gravity and moment of inertia estimations validated by hundreds of Performance Based Standards assessments;
- Suspension geometry provided by the suspension manufacturer;
- Trailer dimensions measured on the test vehicles;
- Suspension spring rates and damping provided by the manufacturer and "calibrated" by a simulated VSB11 Road Friendly Suspension test;
- Coupling bushing behaviour estimated from typical values and then tuned to obtain correlation;
- Vehicle velocity measured from GPS during testing; and,

- Road vertical profile and camber measured by a bank of ultrasonic sensors on the test dolly and then "calibrated" between the simulation and measured data by comparing the dolly chassis vertical acceleration between the simulation and the physical measurement.

Comprehensive road profile scanning at highway speed requires more sophisticated equipment than the scope of this project enabled. The measured road profile did not have the longitudinal resolution to fully represent the road profile at both wheel tracks, or road camber. After post processing the measured road profile was a close enough representation to facilitate manual tuning of the road profile to produce a correlated outcome in the dolly chassis vertical acceleration between the physical test data and the simulation.

Algorithms were written to iteratively modify the road profile, especially in proximity to road excitation features. One segment of road with a length of 1800 m, deemed Stage 10S, was selected as the test road and is described in Figure 41 below. Crossfall is omitted as the test road did not contain large crossfall events.

Automotive multi-body modelling software MSC Adams Car 2018 was the simulation tool used. Tyre modelling was by Pacejka Magic Formula 2002.

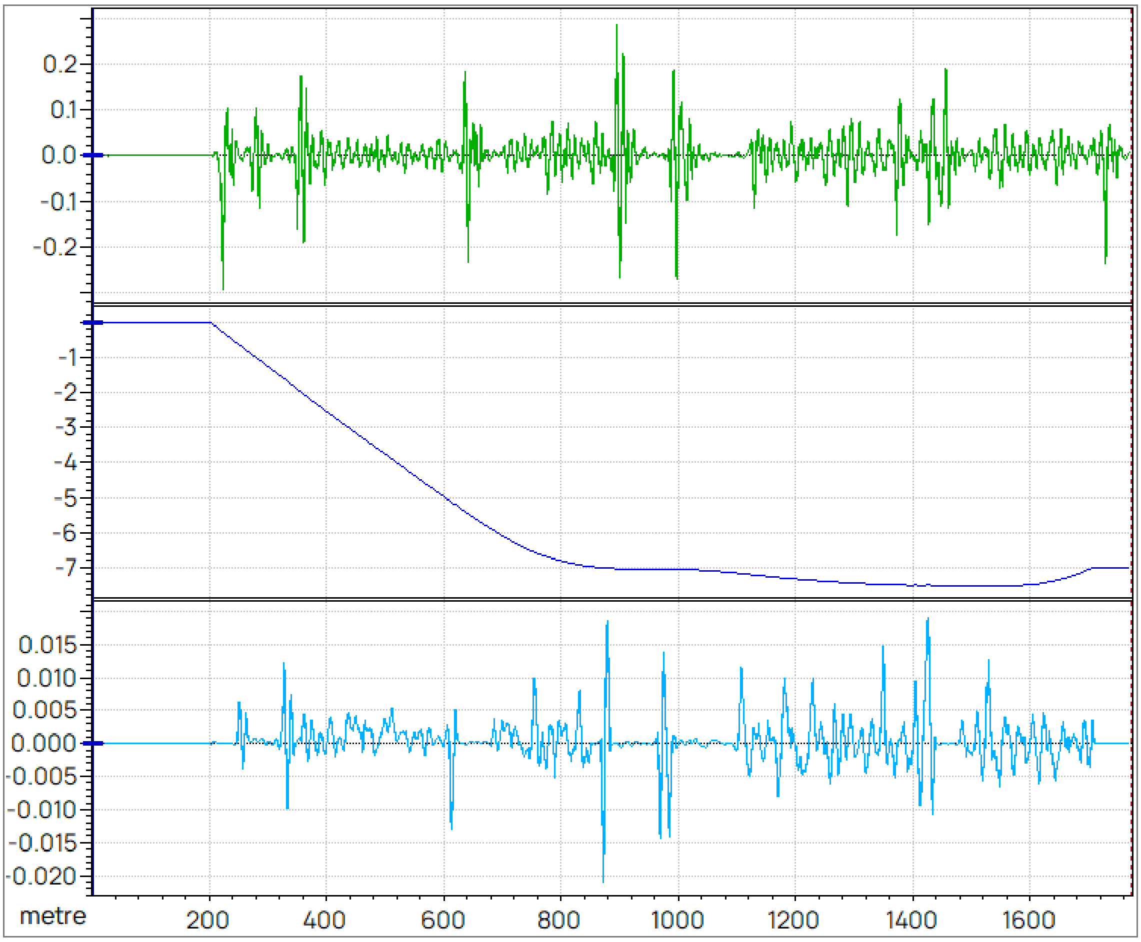


Figure 41 - Correlated Stage 10 vertical chassis acceleration (units of G) at the test dolly, road macro vertical profile and road micro

vertical profile (units of metres)

LIMITATIONS TO SIMULATION CAPABILITY

As the mechanical joints and body structures in multi-body modelling are "perfect" and "rigid", the relationship between the pin type coupling forces and the fifth wheel forces on the same dolly are related linearly by the inertias of the dolly chassis. This linearity is not observed in real world testing.

Creating increasingly realistic models would require isolated component testing of many joints and structures. The forces measured in the physical testing did not exhibit a consistently linear relationship, as non-linear damping (losses) and structural response effects are present in the structural path through a convertor dolly between a pin-type coupling and the associated fifth wheel coupling.

The result is that "perfect" correlation is not possible. Optimising the response of two couplings to match the physical test on one road feature leads to divergence on another. The configuration that has been selected is a compromise between the five major road features present on Stage 10, and the two adjacent couplings on the instrumented convertor dolly.

COUPLING JOINT PARAMETERS

After correlation of vertical acceleration by modifying the road profile input, the pin type coupling joint behaviour was modified to improve coupling force correlation. As the modelling undertaken incorporated rigid bodies, the simulation is only a system of masses, springs and dampers. The couplings of both types were represented as different "bushings", with a non-linear spring rate and longitudinal damping to represent friction and losses in real mechanical systems.

Pin type couplings require a minimum gap between mating components to facilitate their automatic operation. This is typically a minimum of 2 mm, however it is allowable to reach 7 mm. Actual operational gap size (or 'lash'), based upon time spent at zero force readings while passing from tension to compression readings, appeared to be less than stationary lash. Parameters shown below are interpreted into an Akima spline for simulation calculations so lash figures shown do not reflect effective lash as-modelled.

Table 6 - Pin-type coupling simulation parameters

Longitudinal Coupling Displacement	Longitudinal Coupling Force
-40 mm	-3,000 kN
-20 mm	-1,500 kN
-0.1 mm	-1 N
0 mm	0
0.1 mm	1 N
20 mm	1,500 kN
40 mm	3,000 kN
Longitudinal Damping:	100 kN/m/s

Table 7 - Fifth wheel coupling simulation parameters

Longitudinal Coupling Displacement	Longitudinal Coupling Force
-40 mm	-6,000 kN
-20 mm	-3,000 kN
0 mm	0
20 mm	3,000 kN
40 mm	6,000 kN
Longitudinal Damping:	100 kN/m/s

For both coupling types, lateral and vertical bushing parameters were effectively rigid (1,000 kN/mm) with no damping. Fifth wheel roll-mode lash was modelled in separate constructs, based on the Performance Based Standards requirements which are derived from fifth wheel lash testing undertaken by ARRB.

MULTI BODY VEHICLE SIMULATIONS

A range of experiments were conducted on the multi-body simulation which were impractical or unsafe to conduct on the physical combination. Note that in the following simulations, pin-type coupling forces are sometimes omitted from plots for clarity. This is owing to the absolute maximum forces between a pin type coupling for a convertor dolly and its paired fifth wheel being very similar (separated only by the inertia of a 4 t dolly), and the presence of more measurement points along the vehicle when using fifth wheel data.

MASS BEFORE AND MASS AFTER THE COUPLING

As the primary variable in the theoretical equations, mass was varied before and after the test coupling by testing with the dolly in the centre position and artificially inflating and deflating the unit masses. The results are then plotted below in Figure 42.

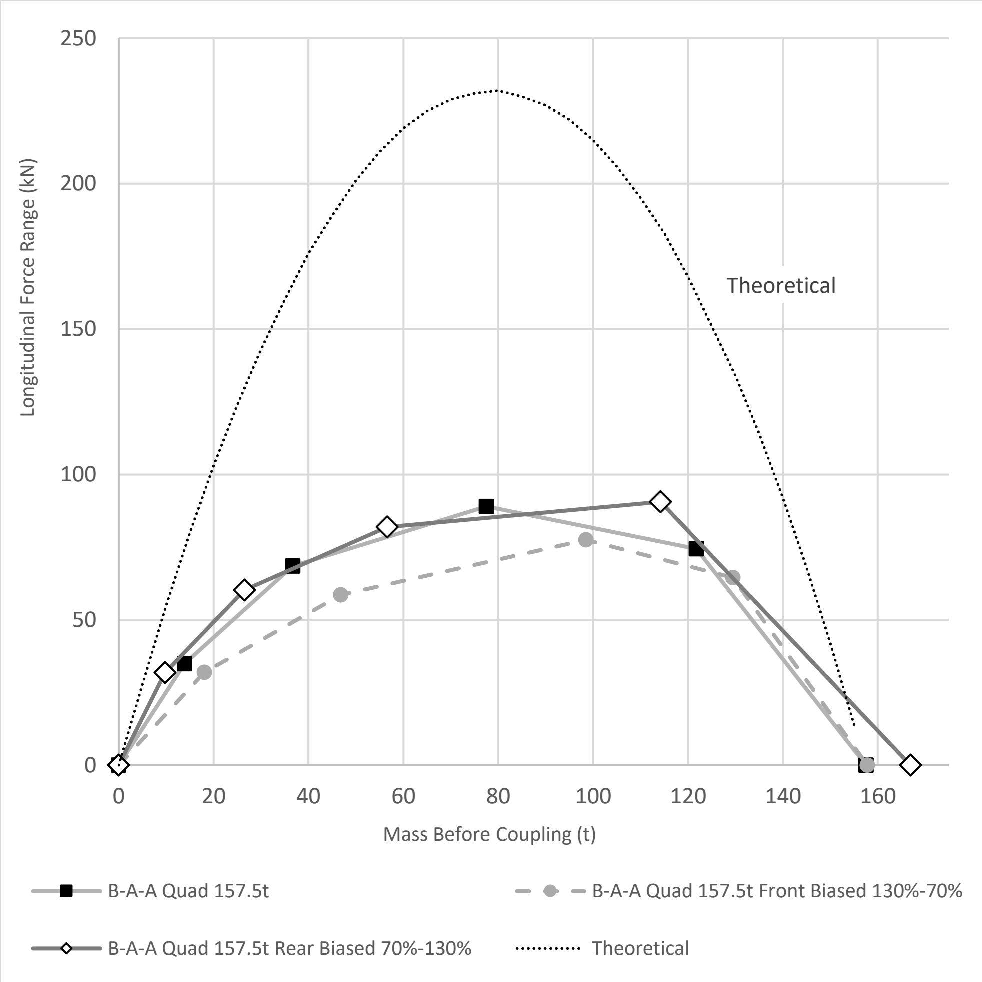


Figure 42 - Simulation results of longitudinal fifth wheel coupling forces in varying combination biases

At gross combination masses around 160 t, the force curve shape is consistent with the theoretical equations and is scaled down in these results due to lower bound road excitation. The subsequent section *Road Vertical Profile Roughness Sweep* demonstrates that road roughness can explain the gap to the theoretical force. Both forward *and* rearward mass biases were demonstrated to shift the force peak away from the mass centre. It is critical to note that the horizontal axis is mass, not length of vehicle, so the geometric centre is not shown. This investigation does not suggest a need to modify the theoretical shape.

COMBINATION CONFIGURATIONS

The combination was varied in structure, from an A-A-B Quad to a B-A-A Quad, and the test dolly was relocated within the combination. This is a representation of the physical tests. As it is representative of the physical tests, the gross mass is also varied in response to differing payloads. See Figure 43.

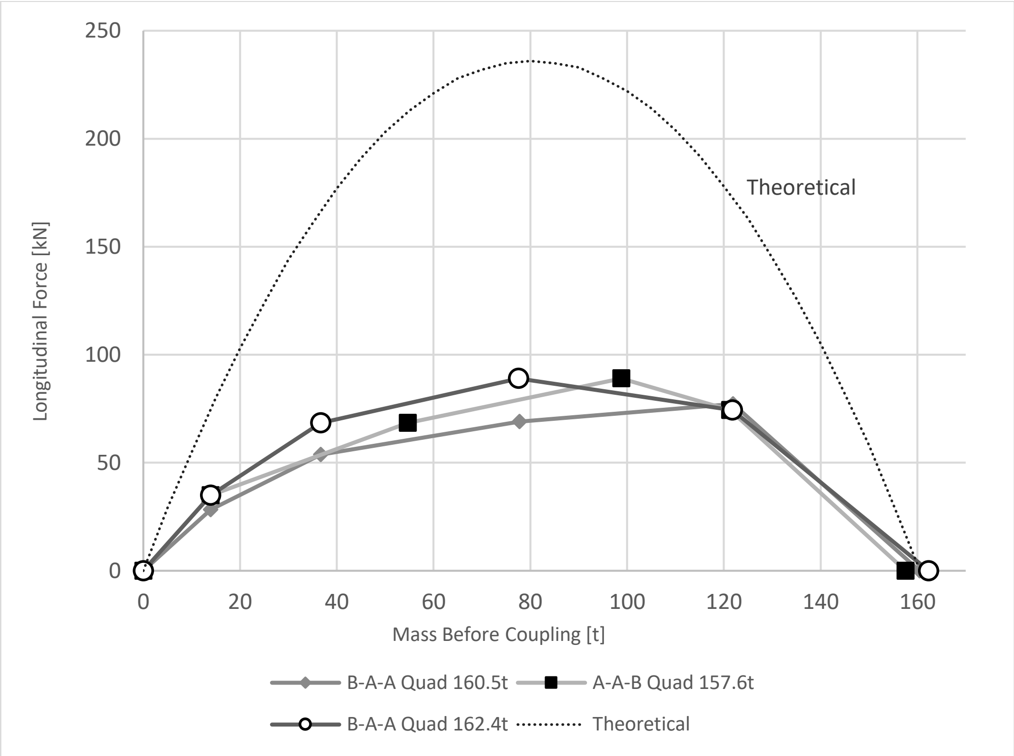


Figure 43 - Simulation results of longitudinal fifth wheel coupling forces in varying combination configurations

Similarly to Figure 42, re-structuring the combination shifts the peak forces slightly toward the rear of the combination. The B-A-A Quad at 162.4 t exhibits near perfect subscription to the shape of the theoretical curve. Vertical scaling is owing to lower bound road excitation. This investigation does not suggest a need to modify the theoretical shape.

COMBINATION MASS

Overall combination mass was varied by modifying axle group masses, and by the linking of longer combinations. See Figure 44. Please note that the character of the theoretical curve is of interest, as opposed to its magnitude in this case, given that road excitations simulated were not the worst case that the theoretical equations seek to predict.

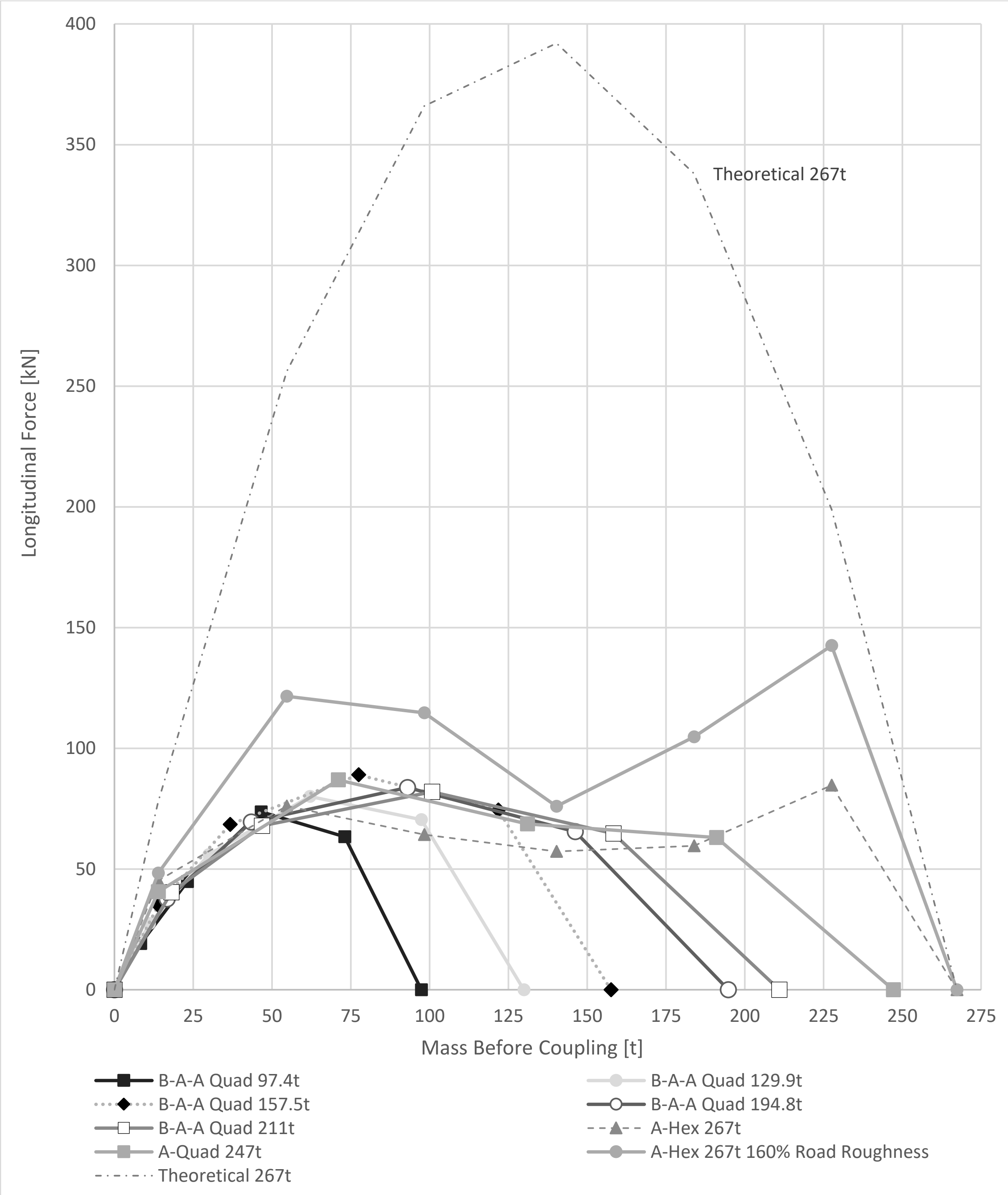


Figure 44 - Simulation results of longitudinal fifth wheel coupling forces at varying masses

This study indicates that the theoretical shape for coupling loads with relation to mass is supported by simulation at gross masses up to 210 t. This is evidenced by the curve shape observed by all combinations up to that mass exhibiting a parabola shape scaled by their gross combination mass.

However, beyond 210 t a bi-modal distribution is observed. Simulations of varying combinations at masses of 247 t and 267 t exhibit this phenomena. An additional simulation of the 267t vehicle was tested at 160% road roughness as confirmation. This scenario vertically scaled the forces observed but did not change the bimodal distribution shape. This warrants further investigation of vehicles in the range of 250 t plus gross combination mass to confirm if this behaviour is also observed in physical testing.

Given the limitations of the simulation environment, this points to the importance of future physical testing of road train combinations with masses over 160t. The novel measurement techniques developed as part of this project can be applied to these combinations in the future, given they have been shown to be relatively low cost and practicable to introduce within a real world working environment.

VEHICLE SPEED SWEEP

Vehicle travel speed was swept from 30 km/h to 140 km/h. See Figure 45 (pin-type couplings) and Figure 46 (fifth wheels).

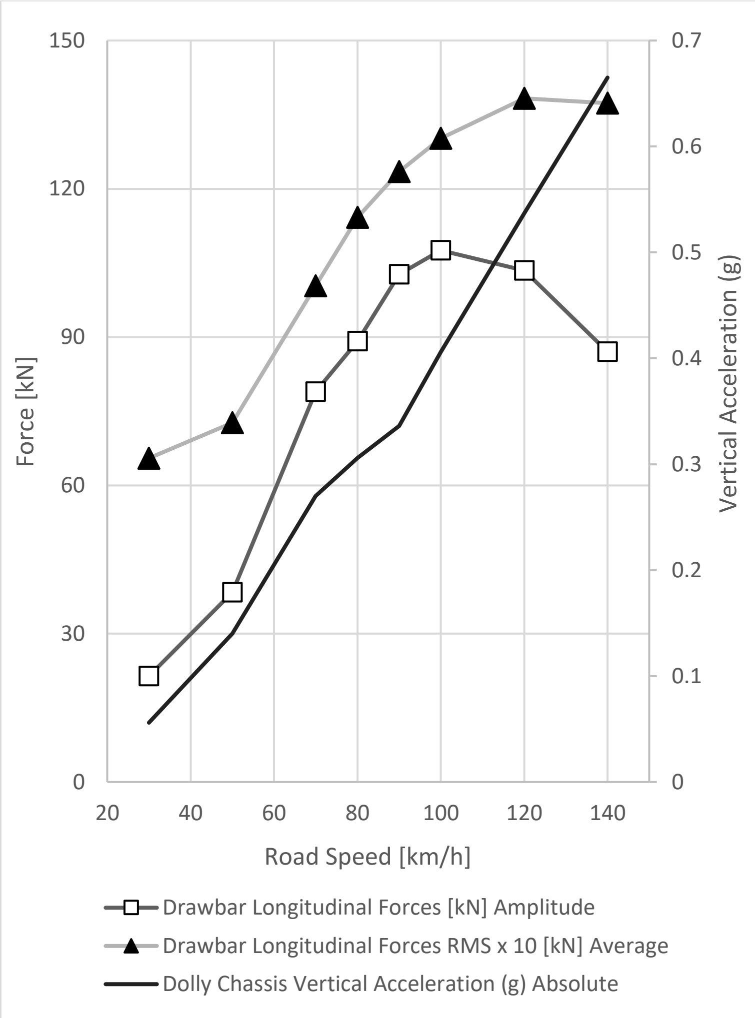


Figure 45 - Simulation results of longitudinal pin type coupling forces at first dolly position on an A-A-B Quad at varying road speed

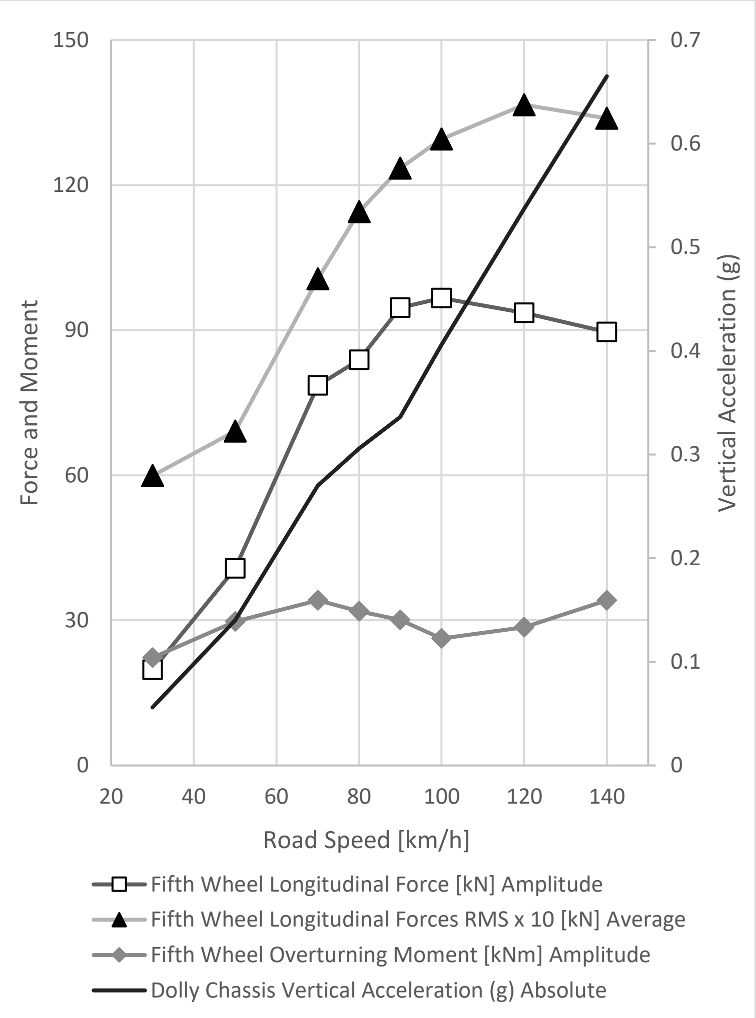


Figure 46 - Simulation results of longitudinal fifth wheel coupling forces at first dolly position on an A-A-B Quad at varying road speed

Both plots demonstrate that absolute maximum vertical acceleration bears a near-linear relationship to road speed, whilst overturning moment is nearly unaffected and longitudinal force increases linearly until the road speed is such that the frequency of road excitations exceeds the natural mode of the suspension, and there is a self-cancelling effect which reduces longitudinal forces. This does not occur until implausibly high road speed for a heavy combination, i.e. 120 km/h plus.

COUPLING LASH SWEEP

Pin type coupling lash (the force generated by adjacent vehicles colliding due to the gap between mating coupling components) was swept from 0 mm (total) to 7 mm in 1 mm increments, then additional tests were conducted at 10 mm and 20 mm for extreme cases. See Figure 47.

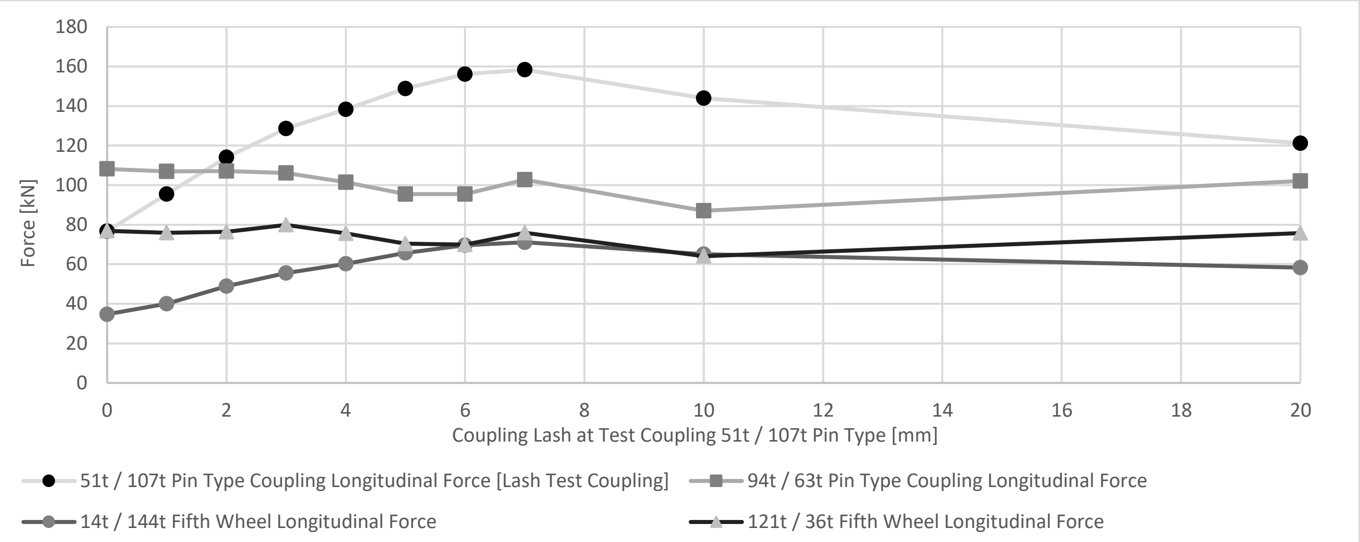


Figure 47 - Simulation results of longitudinal coupling forces at the test coupling and other couplings on an A-A-B Quad with varying lash in the test coupling

Coupling "lash", or local impact between adjacent vehicles, increases absolute maximum force at the local coupling near-linearly until a coupling gap of 7 mm. Interestingly, this trend is only observed for non-centred dolly positions.

For a gap of over 7mm, the software simulation predicts that greater lash can be seen to reduce peak forces, owing to oscillations between adjacent units being absorbed in the lash zone.

This result contrasts with both first principles calculation, and user feedback, which suggests that small increases in the connection gap can result in considerably higher 'lash' forces experienced by the driver.

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CONTROLLING MAXIMUM FORCE IN RELATION TO COUPLING FIT TOLERANCE.

This simulation suggests that there is a need to control the maximum peak forces to a level below the coupling capacity by limiting the maximum tolerance of the gap between adjacent coupling components. The corresponding maximum gap would be around 4 mm in the worst case.

Importantly, this simulation involves the combination underway on a proprietary 'test track' and not during start / stop manoeuvres. Previous experience within the research team, as well as calculations based on momentum transfer, suggest when braking, the relative movement causes very high spikes in force, with increasing exponentially as the 'gap' increases.

Given these results in the simulation, real-world testing of the correlation between the coupling fit tolerance, or 'lash' and force is prudent ahead of recommending updates to Standards or regulations.

The exploration of complimentary design features that serve to dampen the impact between adjacent trailers is also worthy of further investigation.

Plain English:

The gap between the mating coupling components increases as the coupling wears. Currently the maximum gap is as prescribed by coupling component suppliers, and is not independently regulated.

This simulation suggests that an upper limit to this gap is worthy of further investigation as a potential means to prevent very high forces from being applied to coupling components – potentially limiting their life and increasing the risk of fatigue failure.

Other means of controlling these peak forces are also worthy of consideration, such as introducing connection dampening.

RIGID DOLLY DRAWBAR

The drawbar hinge was stiffened to effectively infinite stiffness to emulate a rigid drawbar dolly. See Figure 48.

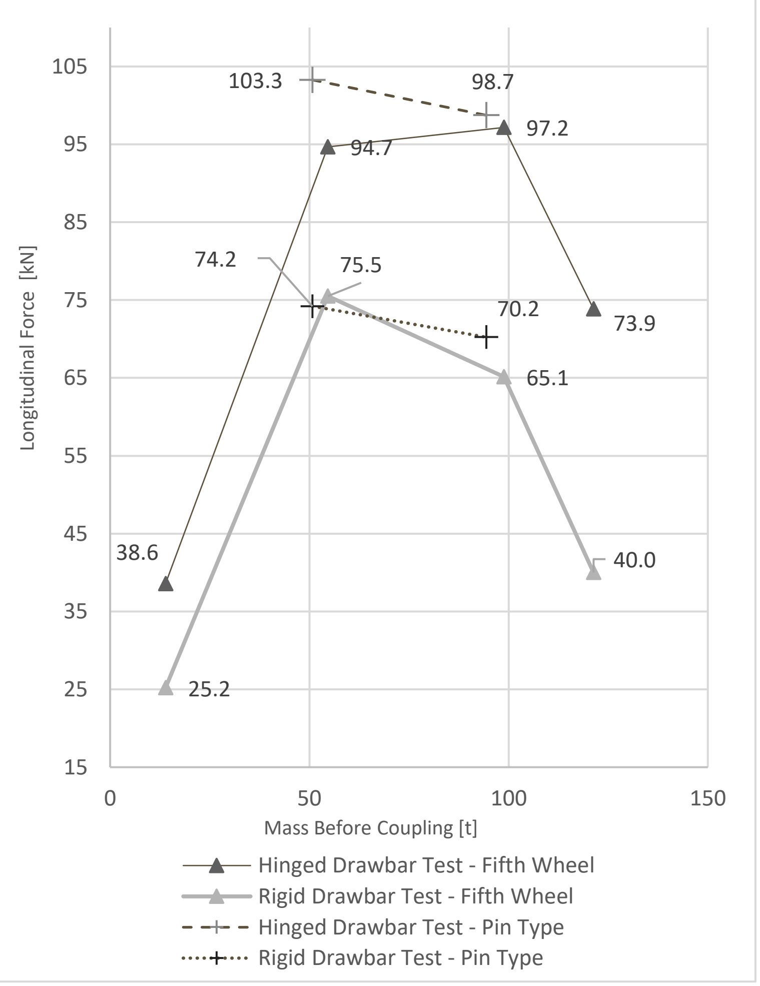

Figure 48 - Simulation results of longitudinal coupling forces at the test coupling and other couplings on an A-A-B Quad with the drawbar of both convertor dollies hinged versus fixed

The provision of a rigid drawbar, which prevents energy storage in the chassis of the converter dolly through chassis pitch, can be seen to reduce absolute maximum coupling loads in this simulation from 103.3 kN to 74.2 kN. The associated fifth wheel also sees a reduction in peak force. This justifies further exploration of rigid drawbar converter dollies for the potential to reduce coupling forces. Testing not shown here of a combination of rigid and hinged drawbar converter dollies showed forces can increase in the hinged drawbar where a rigid dolly is used elsewhere in the combination.

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ROAD EXPERIMENTS

The manner in which roads are defined in MSC Adams is by a series of rectangular faces. These are formed by a combination of longitudinal step length, vertical displacement, lateral displacement, road width and camber angle (defined at the road centreline). Friction is additionally considered but was set to 1.0 for this project.

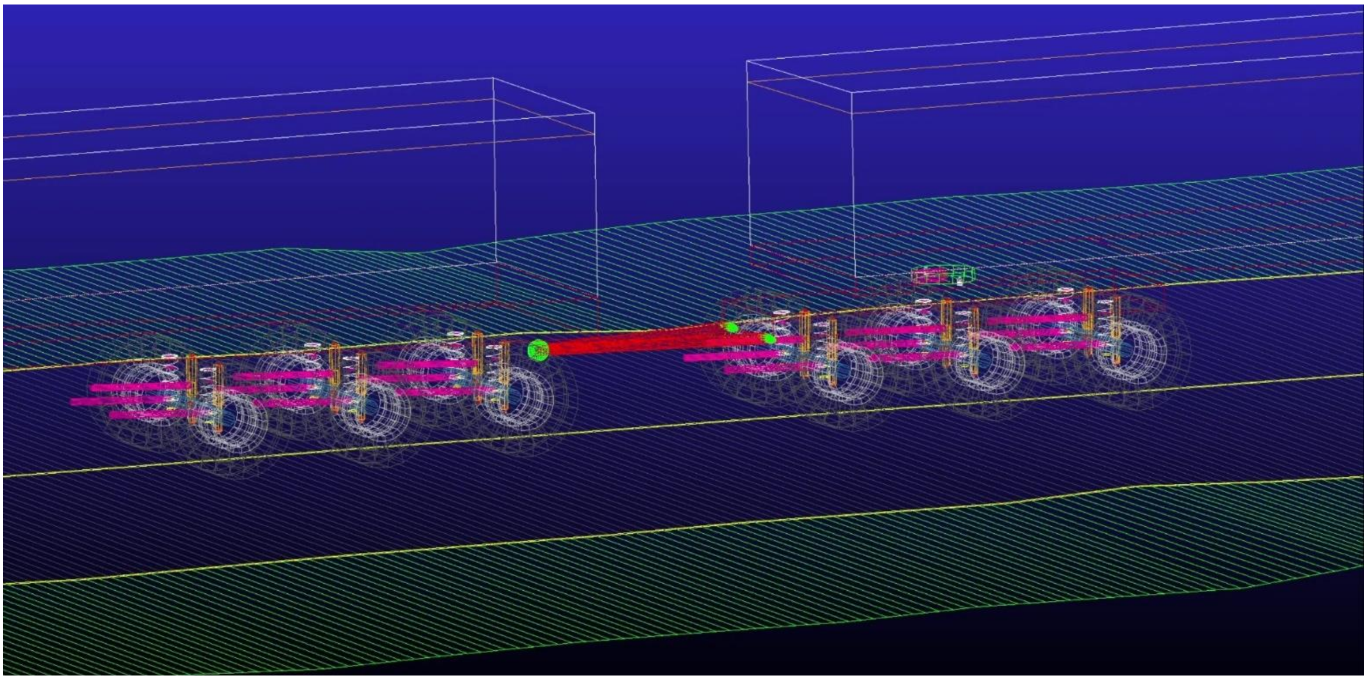

Figure 49 - A still from an animation of the vehicle simulations showing a road excitation feature

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ROAD VERTICAL PROFILE ROUGHNESS SWEEP

Road roughness was varied by a blanket application of a gain factor over the vertical displacement. The roughness was swept through 40%, 60%, 80%, 100% (representing the correlated road), 120%, 140%, and 160%. 180% and 200% roughness roads were attempted but the simulated vehicle suffered a rollover.

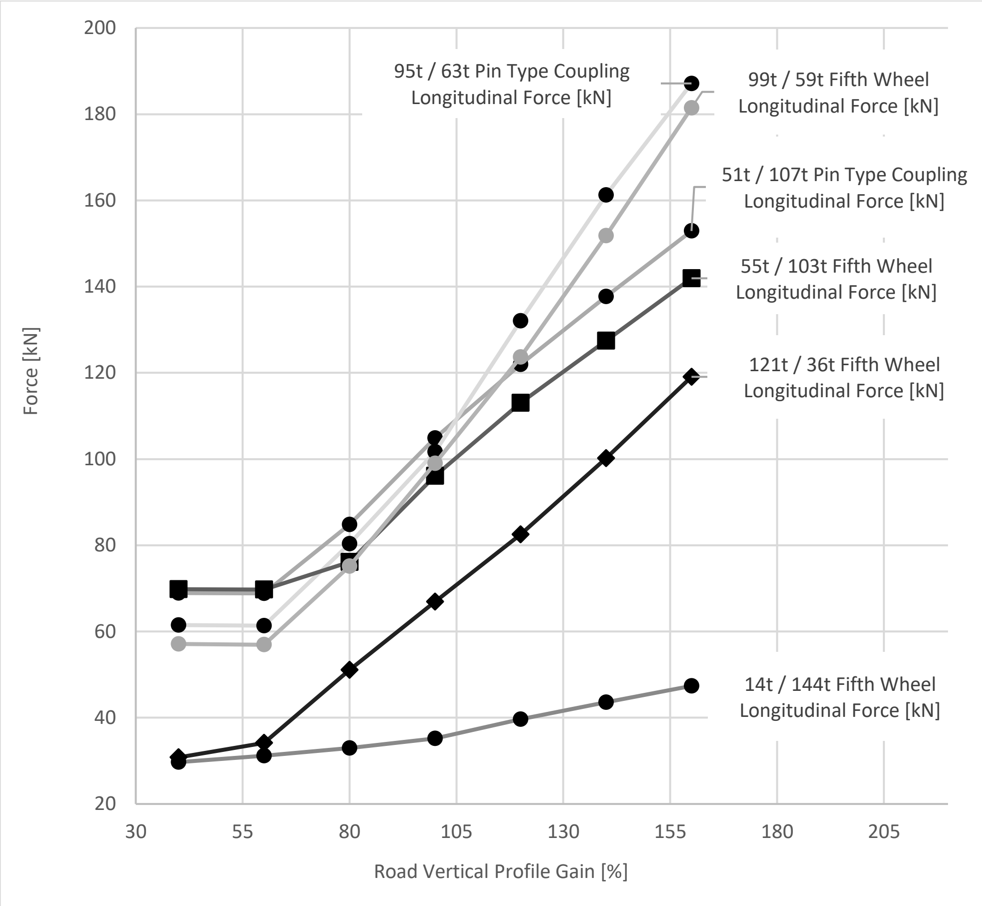

Figure 50 - Simulation results of longitudinal coupling force ranges at all coupling positions on an A-A-B Quad on varying road vertical profile gains

The results of Figure 50 indicate that for a 60% increase in road roughness, forces at the most central coupling increase by 85.4 kN, a slope of 1.42 kN per % road roughness. A linear relationship becomes established after an initial exponential relationship at lower roughness. For other couplings away from the centre, the gradient of the relationship is lower.