Heavy Vehicle Crash Forces

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Abstract

This paper extends the previous work by presenting actual crash simulations that calculate the forces that occur on two types of heavy vehicles involved in barrier crashes. The results can be compared to the simplistic estimates based upon the energy momentum approach. The result show that the average forces that exist are broadly in agreement with the simplified estimates.

These considerations are directly relevant to road safety because it is widely accepted that a combination vehicle that stays together during a crash is likely to suffer less damage and cause less damage that one that separates. Therefore, the question of whether the coupling is likely to break is a central road-safety consideration. The paper will inform considerations about the necessary longitudinal strength of mechanical couplings. Simulations of a rigid-truck, a semi-trailer and a B-double truck striking an immovable barrier at various speeds and weights are presented to inform the considerations.

The strength requirements of mechanical couplings are specified in technical standards such as UN ECE Regulation 55 and SAE standards J133 and J874. Surprisingly, none of these standards explicitly define a *Factor-of-Safety*. In the Australian experience, mechanical couplings are being used on longer and heavy multi-combination vehicles that were originally designed for single-trailer applications. It is argued that changes to the technical requirements in couplings rules are needed to explicitly introduce a *Factor-of-Safety* in addition to fatigue withstand considerations.

Keywords: Heavy vehicle crashes, chassis-rail ladder, mechanical couplings, strength of couplings, crash forces.

1. The Nature of Vehicle Crashes

Collisions involving one vehicle and a barrier or two vehicles can be analysed using energy and momentum conservation considerations, as was described by one of the authors previously (Ref[1]). These methods identify two extreme crash behaviours which are elastic (energy returning) or plastic (energy absorbing). The real-world collision is intermediate between the plastic and elastic collisions.

The purpose of the current investigation is to inform discussion about what Factor-of-Safety should exist for the mechanical coupling(s) between parts of a combination vehicle. It is commonly accepted by road-safety experts that it is desirable for the parts of a combination vehicle to stay connected when a crash occurs. Separation of the vehicle into parts in the event of a crash, results in multiple objects that need to be avoided by other road users and it puts the heavy-vehicle driver at risk of being struck by a following part of his vehicle.

Figure 1 shows fifth-wheel pedestals that broke during a low-speed collision (< 15 km/h) between two fully-laden B-double trucks. The heavy-vehicle driver was killed. Metallurgical tests shows that the pedestals were made from low-grade cast iron with a nominal yield tensile strength of 400 MPa and nominal yield shear strength of 330 MPa. The manufacturer's D-value for the fifth wheel is 200 kN. A debate occurred between experts involved in investigating the crash about how the *Factor-of-Safety* of the coupling should be defined and how it could be calculated. This crash investigation motivated the authors to write this paper.

Figure 1 Failed fifth wheel (cast) pedestal legs on a fifth wheel fitted to a truck-tractor.

Figure 2 illustrates three stages in the impact between a B-double truck that is initially travelling at speed V_i and an immovable barrier. Plastic and elastic deformation occurs at the front of the vehicle during the impact. At the instant that the vehicle centre of mass is zero, some of the initial energy is absorbed by plastic distortion at the front and the rest is stored as elastic energy in all sections of the vehicle.

Soon after separation, the vehicle has a recoil speed of V_f. At this time the elastic energy has been returned as vehicle kinetic energy. The difference between the initial kinetic energy $\frac{1}{2} MV_i^2$ and the final kinetic energy $\frac{1}{2} MV_f^2$ is the energy absorbed in plastic deformation and heat absorbed due to shock waves and mechanical rubs that occur during the crash.

The crash illustrated in Figure 2 is highly idealized as off-line motion after the crash is not shown. It is sufficient to assume that the final kinetic energy is ½ M V_f^2 which may have sideways velocity components and rotations. The crush zone has a final Crush Length CL that is shorter than the original length. Mechanical work is done by the forces experienced in the crash zone and this is equal to the lost kinetic energy.

The parts of the vehicle that are behind the crush zone experience the same average deceleration during the crash because they are rigidly connected. Shock wave forces will also occur so the peak deceleration forces can be higher than those to be expected based upon the average deceleration.

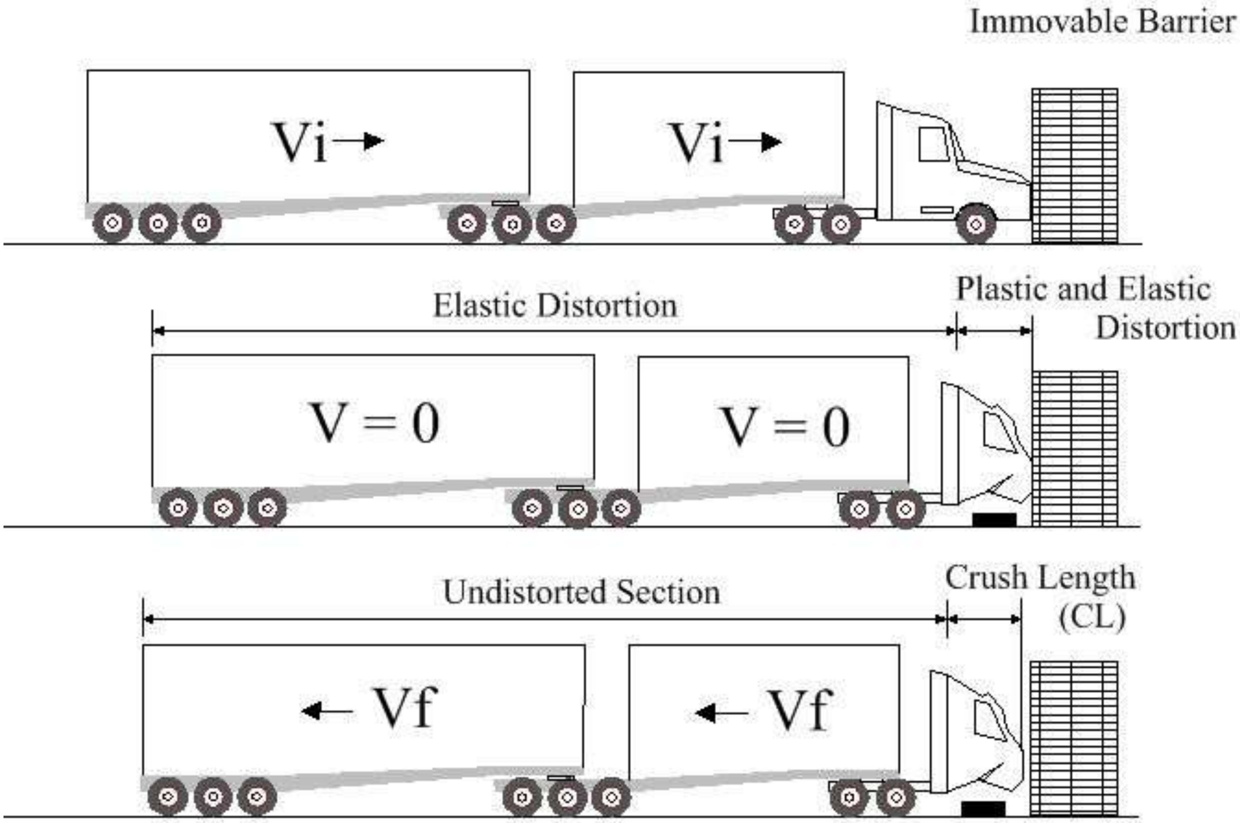


Figure 3 Simplistic representation of plastic and elastic sections of the crash vehicle.

In reality the interaction is very non-linear because materials yield, the chassis ladder buckles and unyielding items such as the engine strike the boundary. In a previous paper [1], one of the authors (Hart) described how estimates of crash forces might be obtained based on energy-momentum methods. This paper provides a contrasting approach, which is to simulate the impact.

In Part 2 we present the results of detailed simulations of vehicles hitting a perfectly rigid barrier. The simulations are based upon detailed finite-element models of firstly; a rigid truck and secondly; a B-double truck. The semi-trailer simulation uses the B-dosubel truck model with the second trailer deleted. The simulations provide estimates of the 'abnormal' forces that occur at the front of the crush zone and in the 'elastic' deformation zone (which is where the mechanical couplings are installed).

The FEA simulations were run as explicit nonlinear analyses using Altair Radioss software. The models are based upon public domain LS-DYNA models that have been converted to Radioss format. The modelling of the truck-tractor chassis and trailer structures have been improved to provide better resolution and accuracy. They better reflect the chassis ladders used on heavier combinations such as B-double vehicles. The connections are rigid with no slackness allowed for. The timestep used in the simulations is 1E-6 seconds.

Part 3 of this paper provides a detailed consideration of the ratings of; and the 'normal' forces that mechanical couplings could experience. This leads to detailed consideration about the Factors of Safety that exist. Part 4 presents a discussion about the strength requirements in coupling technical standards and argues for revision of the key standards; which are UN ECE Regulation 55 and SAE standards J133 and J847.

2. Simulation of Truck Barrier Crashes

2.1 Rigid Truck

The first set of FEA analyses a collision between a rigid truck weighing 8t and travelling initially at 60 km/h and an immovable barrier. The truck has a familiar chassis-ladder construction made from mid-strength steel (390 MPa material). The forces were computed at nine sections (Figure 5). Section 1 is 0.25 m from the bumper and sections behind the first are at 0.5 m spacings. Results are shown in Figure 3.

The rigid truck model consists of 41,000 nodes and 38,000 elements. The barrier is represented using a RWALL rigid wall which is effectively a fixed, non-deformable infinite plane and all elements of the truck are able to make contact with the barrier. It is acknowledged that the unyielding barrier represents a worst-case and somewhat unrealistic practical case. All barriers exhibit some deformation and this reduces the crash forces experienced by the vehicle. Therefore, the simulations will give higher than realistic vehicle crash forces.

The rigid-truck simulation showed that:

- The chassis-rail damage is mainly confined to Sections 1, 2 and 3.
- The barrier force is considerably higher than the vehicle section forces because multiple regions of the vehicle absorb crash forces.
- The barrier contact is over within 0.15s.
- The barrier force displays an oscillation at about 3.6 Hz.
- The forces experienced in Sections 1, 2 and 3 and considerably higher than for Sections 4 – 9.
- There is little deformation beyond Section 2, which is 0.45 m from the front of the chassis rails.
- There is very little deformation in the chassis rails beyond Section 2, which is 0.75 m from the front of the chassis rails.
- Sections 1 & 2 come to rest earlier than following sections.
- The Sections 3 – 9 decelerate at a peak value of about 16g.

If the chassis ladder alone of the rigid truck is impacted into the barrier then a shock wave with a frequency of 176 Hz is set-up (Figure 4). This identifies the natural frequency of the chassis ladder. The shock wave travels at the speed of sound in the steel (which is about 5 km/s). The shock wave dissipates in less than ~ 0.3 second. At this time the rear sections of the truck are still moving forward.

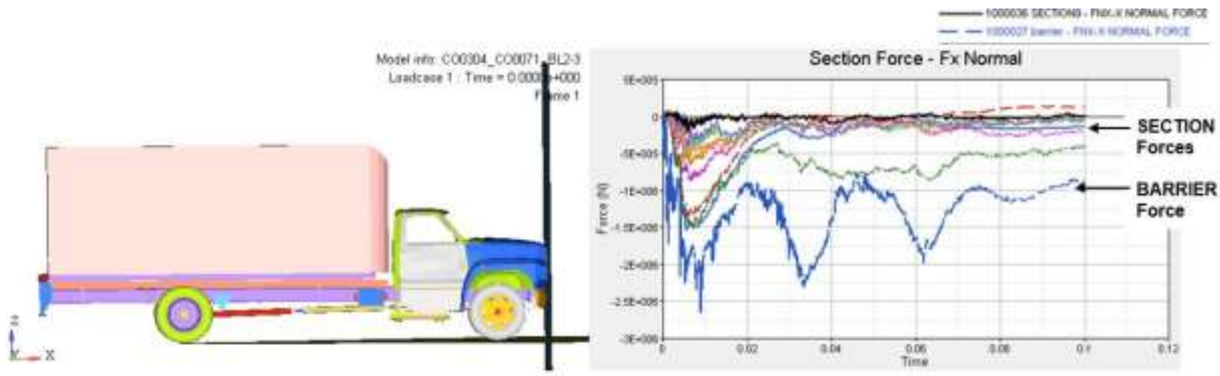


Figure 3a Force results for the barrier crash

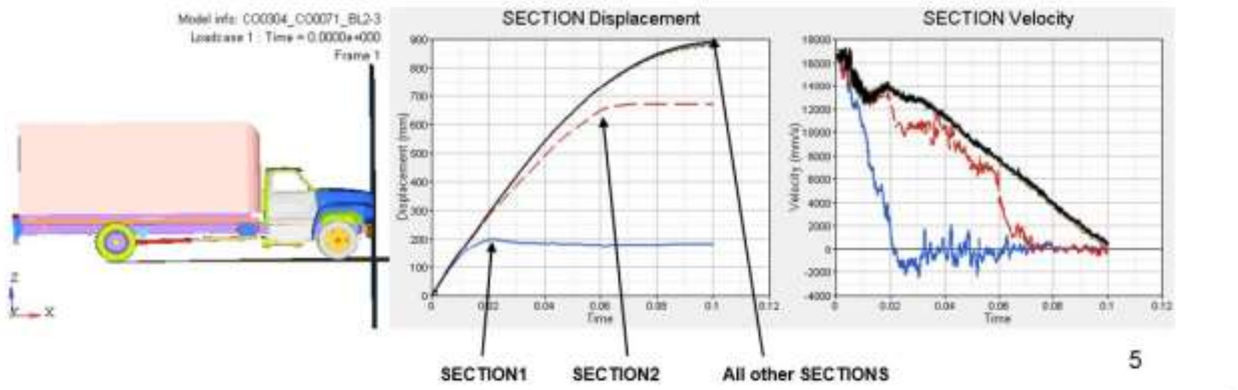


Figure 3b Force results for the barrier crash.

The deceleration at the first truck section is severe. The deceleration falls off very rapidly with distance from the front.

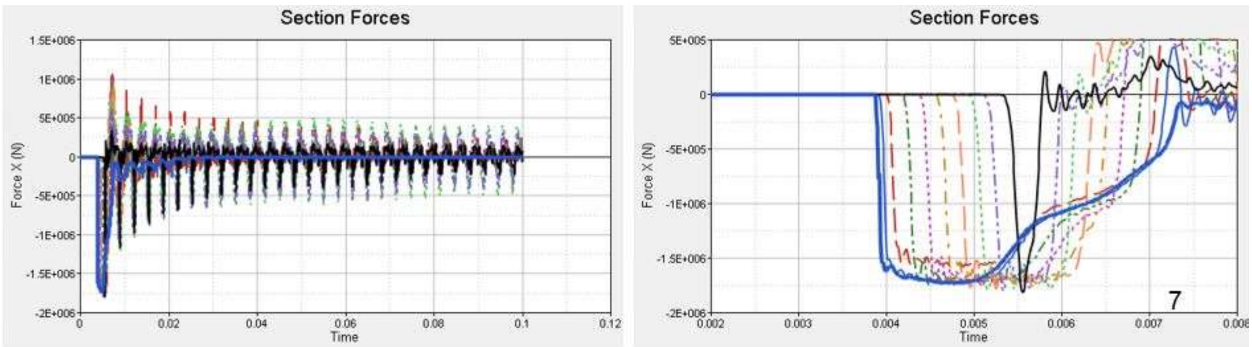
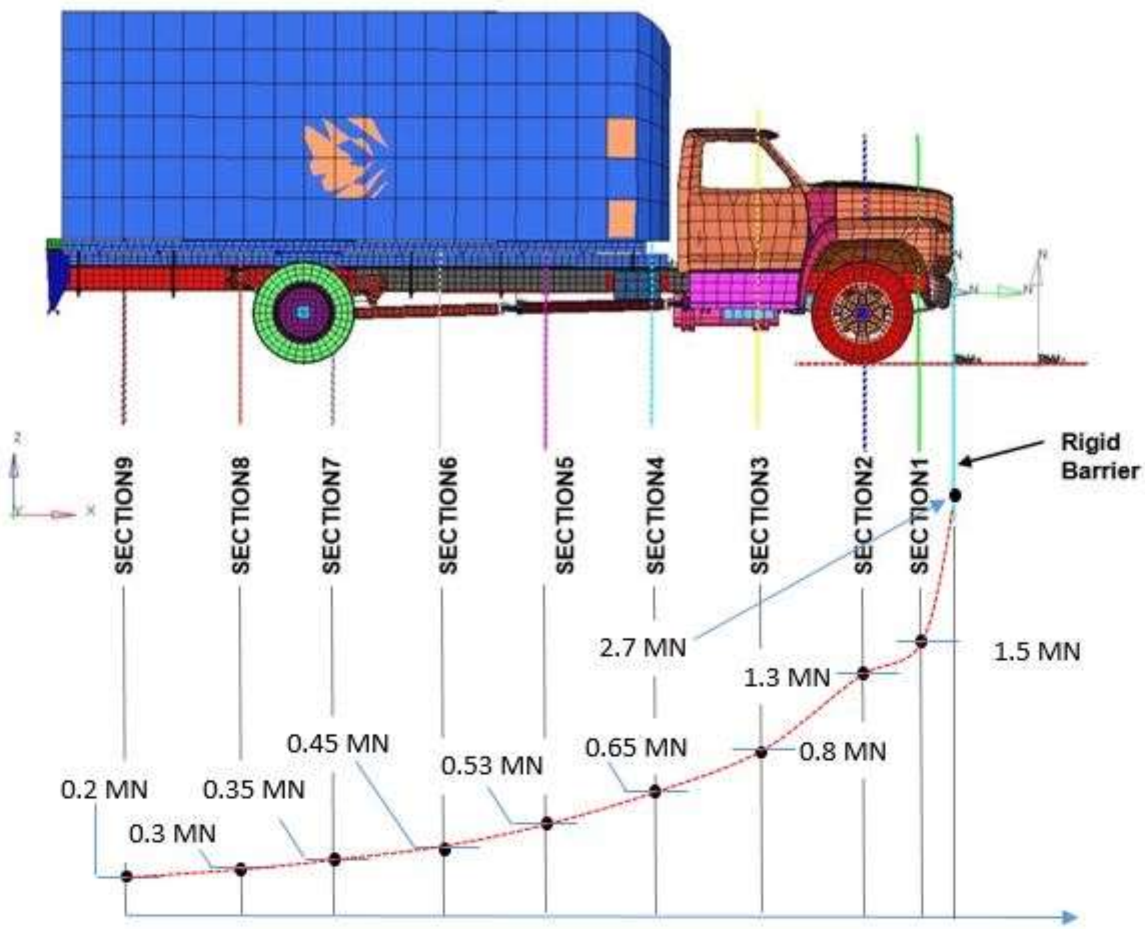


Figure 4 Shock waves in the chassis-rail ladder of the rigid truck

Ringing in the chassis rail ladder persists for ~ 0.3s.



- The peak deceleration at Section 1 is about 80g.
- The peak deceleration at Section 2 is about 23g.
- The peak deceleration of the following 7 sections is about 16g.

Figure 5 Peak section forces.

2.2 B-Double Truck

A second series of simulations was conducted involving a B-double truck colliding with an immovable barrier. The full B-double vehicle model has 181,000 nodes and 217,000 elements. Numerous contact conditions are used to represent contacts between the various components of the vehicles. The fifth-wheel joints are modelled using a cylindrical joint between the kingpin and the fifth-wheel plate, with the joint axis fixed to be perpendicular to the plate. This allows the kingpin to rotate and move up or down relative to the fifth-wheel plate whilst locking all other degrees of freedom. Vertical downward displacement is limited by contact between the fifth-wheel plate and the trailer skidplate, whilst upwards displacement is limited by a travel stop in the cylindrical joint which represents the locking jaws engaging with the kingpin. Forces acting on the joint were output as the fifth wheel forces given in the results.

As for the rigid-truck simulation, the vehicle is unconstrained and can ricochet off line. In the first case the vehicle is unloaded (tare weight =15t) and in the second case it is loaded with a total weight of 62t. Both cases were run at three speeds, which are 15 km/h, 30 km/h and 60 km/h.

Figure 6 illustrates the predicted crash damage. The crush zone extends about 3m back from the front of the unladen truck and about 5m back from the front of the laden truck. The chassis rails buckle on the laden truck due to the over-ride moment that comes from the trailer momentum forces acting on the front fifth wheel. The simulation treats each of the two couplings as rigid connections of infinite strength. The chassis rails on the truck-tractor was set to have a yield strength of 650 MPa steel and an ultimate tensile strength of 900 MPa.

Selected forces and velocities during the crash for each of the three speeds and two masses are shown in Figures 8 a, b & c. Table 1 shows the peak forces that occur at the barrier and at each of the couplings. The interaction between the barrier and the truck are over in about 0.5s after which time recoil of the vehicles occurs. The peak barrier force is greatly affected by speed but not by mass. The peak barrier force is about 6.3 MN for an input speed of 60 km/h and about 2.6 MN for a 30 km/h input speed.

The barrier force appears to have a squared relationship with input speed. The barrier force persists for longer for the laden truck than the unladen truck, which indicates that it is the impulsive force (Ns) that is mainly affected by mass.

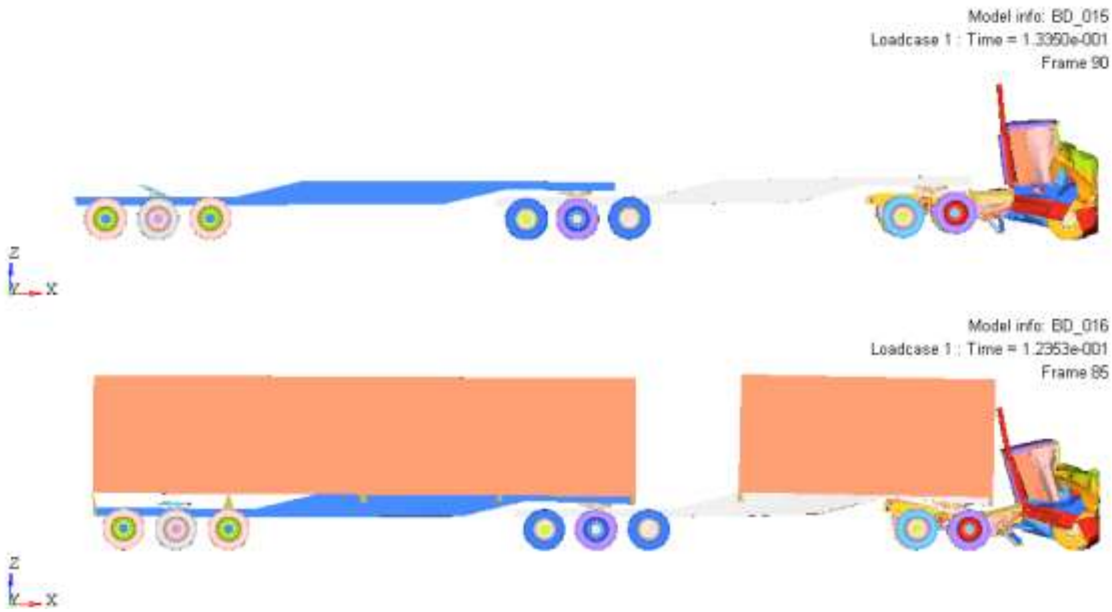


Figure 6 A pictorial representation of the crash damage to the B-double (60 km/h).

Table 1 shows that forces occur at the fifth wheels that are significant proportion of the peak barrier force. Higher values occur for the laden case (62t) than for the unladen case (15t). The fifth wheel forces increase with total vehicle mass but not proportionately so. The fifth-wheels forces increase with speed but not proportionately with speed. Indeed, fifth wheel forces only increase marginally with impact speed. This indicates that most of the crash energy goes into the distortion of the front of the truck-tractor. The crumple behaviour of the truck-tractor is a key determinant of the forces that occur at the fifth wheels.

Figure 7 shows the results from Table 1 plotted against crash speed. The relationship between barrier peak forces and crash speed is non-linear whereas the relationship between the couplings' peak forces and speed is approximately linear. Doubling the speed results in an approximate 20% increase in the peak forces at the couplings. Most of the additional collision force is absorbed in the crush zone. The forces at the couplings are not proportional to the initial energy.

For the 15 km/h impacts, the peak decelerations at the first coupling are 12.8g unladen and 2.9g laden. The greatest vulnerability to coupling failure is the laden case which has the highest forces.

Condition **B-double Vehicle**	**Peak Barrier Force (MN)**	**Peak Force at front fifth wheel (MN)**	**Peak force at the rear fifth wheel (MN)**	**Peak Deceleration at front fifth wheel (Peak force/trailing mass)**
Unladen (15t) at initial speed = 15 km/h	2.0 (200 %)	1.00 (10 %)	0.50 (50%)	12.8 g
Laden (62t) at initial speed = 15 km/h	2.5 (156 %)	1.60 (100 %)	1.10 (69%)	2.9 g
Unladen (15t) at initial speed = 30 km/h	2.5 (211 %)	1.18 (100 %)	0.62 (53%)	15.1 g
Laden (62t) at initial speed = 30 km/h	2.8 (165 %)	1.70 (100 %)	1.16 (68%)	3.1 g
Unladen (15t) at initial speed = 60 km/h	5.9 (437 %)	1.35 (100 %)	0.77 (57 %)	17.3 g
Laden (62t) at initial speed = 60 km/h	6.6 (347 %)	1.90 (100 %)	1.38 (73 %)	3.4 g

Table 1 Comparison of the peak forces experienced at the barriers and at the two fifth wheel kingpins of the B-double vehicle. The peak forces that occur at the barrier and at the second coupling are also given as a percentage of the applicable peak force at the first coupling.

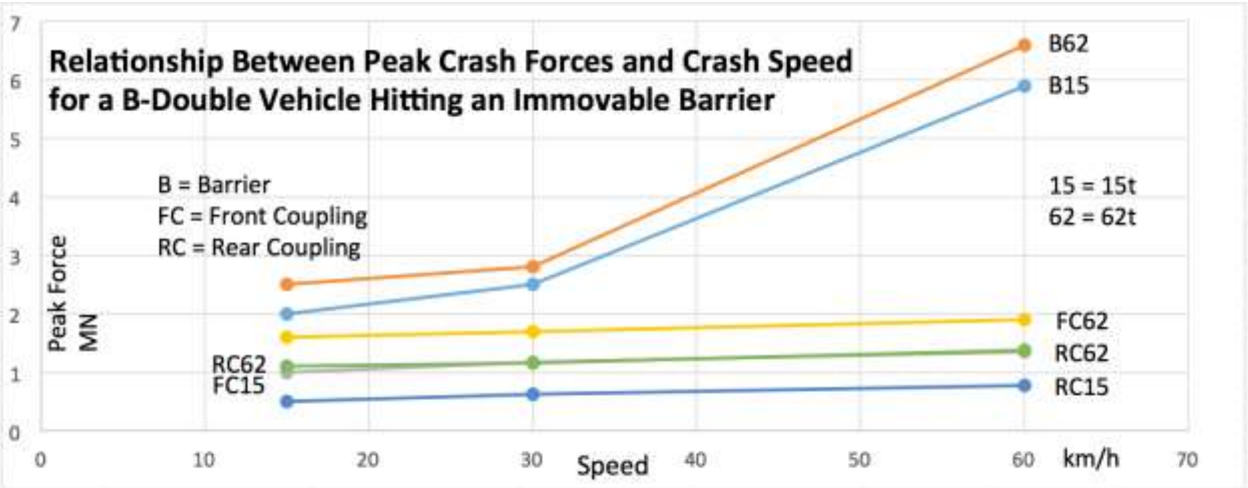


Figure 7 Peak force for both masses plotted against initial speed; for the B-double truck.

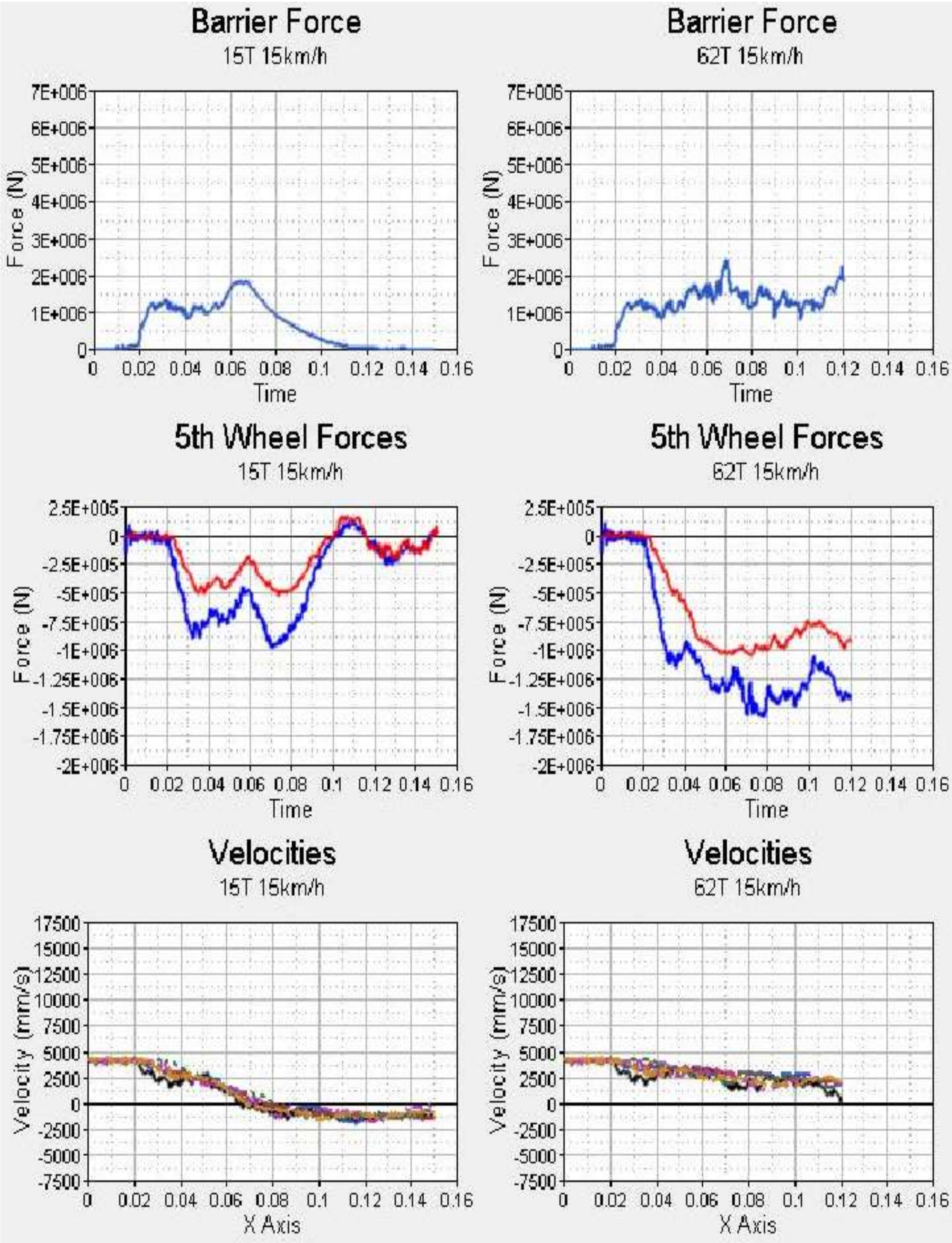


Figure 8a Force and velocity results for 15 km/h impacts.

Fifth Wheel Forces - Blue: front fifth wheel. Red: rear fifth wheel

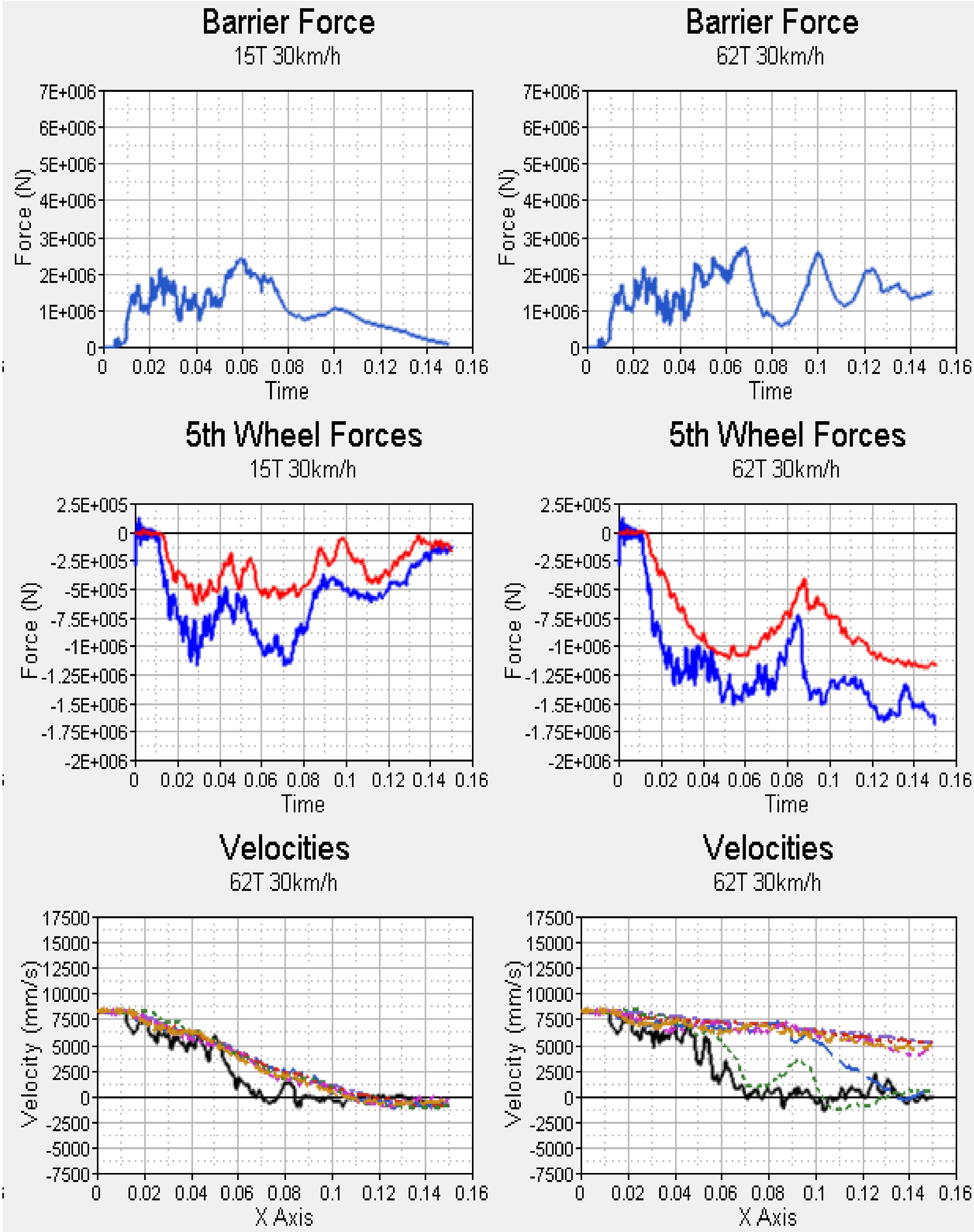


Figure 8b Force and velocity results for 30 km/h impacts.

Fifth Wheel Forces - Blue: front fifth wheel. Red: rear fifth wheel

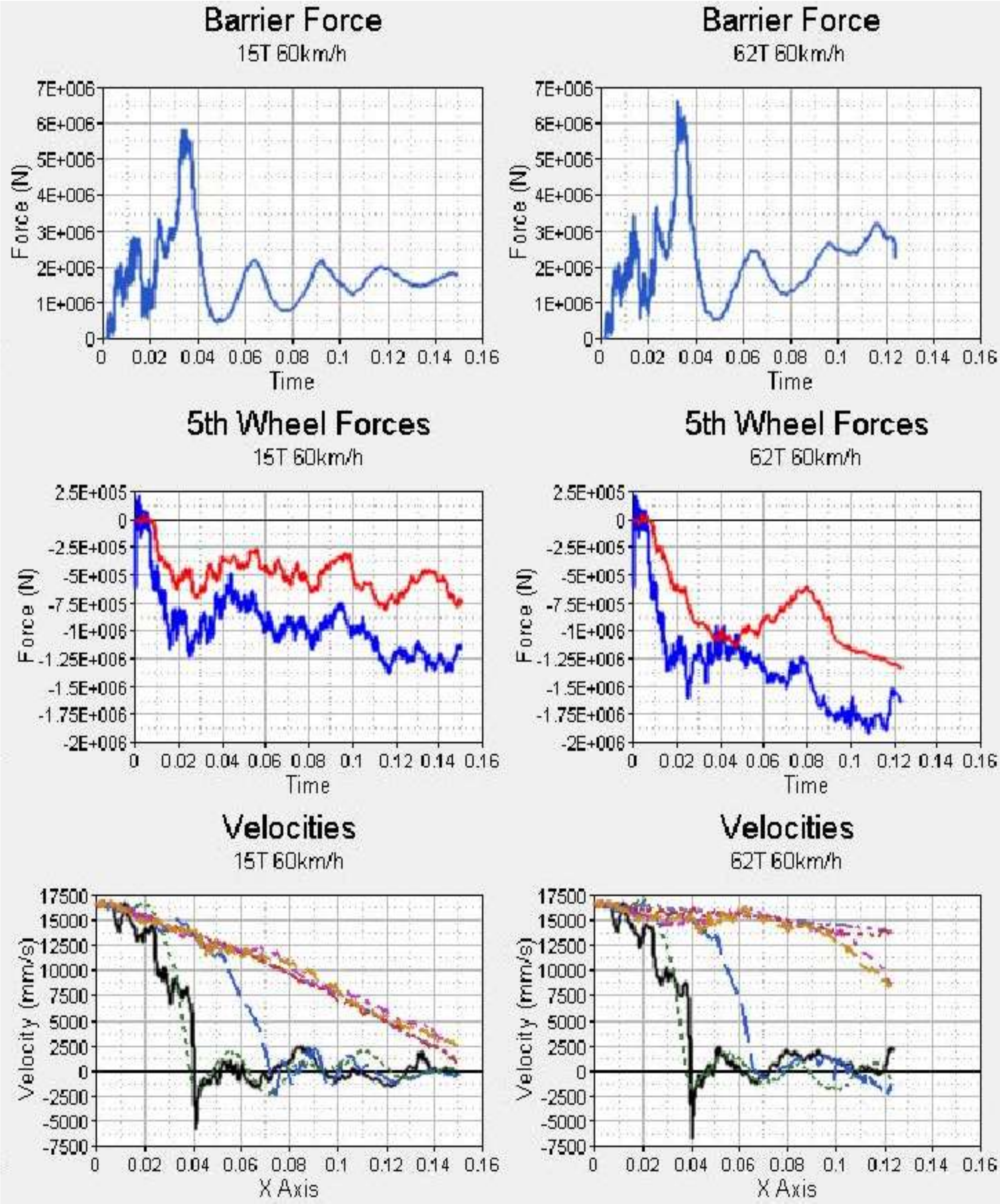


Figure 8c Force and velocity results for 60 km/h impacts.

Fifth Wheel Forces - Blue: front fifth wheel. Red: rear fifth wheel

2.3 Semi-Trailer Truck

The crash force simulations were also done for a semi-trailer vehicle. The truck-tractor and first trailer models were unchanged from the B-double simulation. The unladen mass for the semi-trailer vehicle was 11T and laden mass was 26T. The second B-double trailer was deleted. Table 2 and Figure 9 show the results for the semi-trailer. As for the B-double vehicle, the worst case forces at the coupling occur in the laden condition.

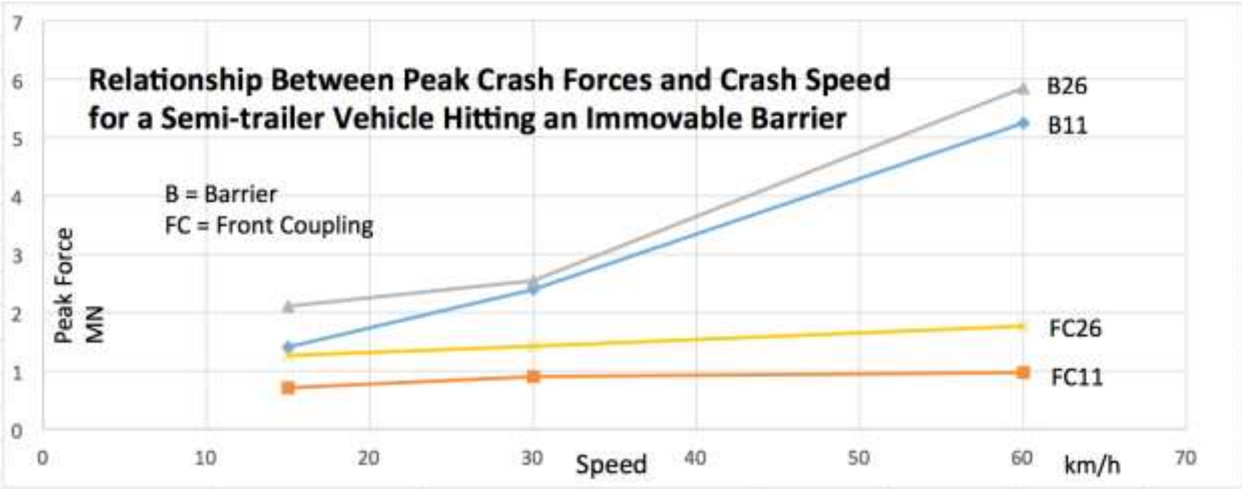


Figure 9 Peak force for both masses plotted against initial speed; for the B-double truck.

Condition **Semi-trailer Vehicle**	**Peak Barrier Force (MN)**	**Peak Force at Fifth Wheel (MN)**	**Peak Deceleration at fifth wheel (Peak force/trailing mass)**
Unladen (11.3t) at initial speed = 15 km/h	1.4 (210 %)	0.7 (100 %)	17.1 g
Laden (26.3t) at initial speed = 15 km/h	2.10 (167 %)	1.26 (100 %)	6.6 g
Unladen (11.3t) at initial speed = 30 km/h	2.40 (266 %)	0.90 (100 %)	22.0 g
Laden (26.3t) at initial speed = 30 km/h	2.55 (180 %)	1.42 (100 %)	7.4 g
Unladen (11.3t) at initial speed = 60 km/h	5.25 (541 %)	0.97 (100 %)	23.7 g
Laden (26.3t) at initial speed = 60 km/h	5.85 (332 %)	1.76 (100 %)	9.2 g

Table 2 Comparison of the peak forces experienced at the barriers and at the fifth wheel kingpin of the semi-trailer vehicle. The peak forces that occur at the barrier and at the second coupling are also given as a percentage of the applicable peak force at the first coupling.

3. Factors of Safety for Mechanical Couplings

3.1 Factors of Safety Guidance

A guide to good safety-factor engineering practice as proposed by Juvinall and Marshek in *Fundamentals of Machine Design*, Ref [5] is:

Appropriate Factor of Safety	Application
1.25 – 1.5	Material properties known in detail. Operating conditions known in detail. Loads and resultant stresses and strains known with a high degree of certainty. Material test certificates, proof loading , regular inspections and maintenance. Low weight is important to design.
1.5 - 2	Known materials with certification under reasonably constant environmental conditions, subjected to loads and stresses that can be determined using qualified design procedures. Proof tests, regular inspections and maintenance required.
2 – 2.5	Materials obtained for reputable suppliers to relevant standards operated in normal environments and subjected to loads and stresses that can be determined using checked calculations.
2.5 - 3	For less tried materials or for brittle materials under average conditions of environment, load and stresses.
3 - 4	For untried materials used under average conditions of environment, load and stress. Also for better-known materials that are to be used in uncertain environments or subject to uncertain stresses.
Notes	The *Factor-of-Safety* = Maximum tolerable load / Actual normal load. In this context the maximum tolerable load is the load that just produces yielding of the material.
Notes	For repeated cyclical loads, the factors of safety established above must be based on the endurance limit (fatigue limit) rather than on the yield strength of the material. The strength calculations should also take account of stress concentration factors and hence a low material strength may need to be assumed.
Notes	Impact shock forces. The higher factors of safety given above (2.5 – 4) may be used but estimation of the maximum tolerable load must take account of dynamic magnification factor.

Table 3 Guidance on appropriate factors of safety for machines.

Mechanical couplings are subjected to repeated cyclical loads and they may experience impact loads if there is slack in the connections. The materials may be well known (such as grades of steel for kingpins) or they might be potentially less well controlled (such as cast pedestals for fifth wheels). The above guidance suggests that a *Factor-of-Safety* (FoS) of 3 is applicable based upon endurance limit considerations.

Factors-of-Safety are needed to account for abnormal conditions and for degradations under couple plausibly occur under normal conditions. For mechanical couplings, crash forces can be regarded as abnormal even though they are foreseeable in a very small proportion of cases. The *Factor-of-Safety* framework presented in Table 2 is based upon normal foreseeable loads. Because couplings are routinely subject to time-varying loads, the material limits should be based upon the endurance limits.

The question of whether a coupling is likely to break during abnormal conditions must take into account the inherent *Factor-of-Safety*, which implies a strength limit, and knowledge of the abnormal force levels that could plausibly occur.

3.2 Endurance Limit Considerations

Vehicle standards rules for mechanical couplings, such as UNECE Regulation 55, identify three principle strength ratings which are:

D-value: The manufacturer's declared horizontal strength rating.

V-value: The manufacturer's declared minimum vertical strength rating.

S-value: The manufacturer's declared static vertical load rating.

According to UN ECE Regulation 55, the D- and V-value ratings must be proven by fatigue testing that is to be conducted by application of oscillating forces of (at least) ±60% of the rating. The fatigue stress limit of a material ("endurance limit") is less than the yield strength. Kingpins have cylindrical symmetry and well defined material properties so that they can be conveniently modelled. A typical kingpin steel is 42CrMo4 which has a yield strength of 990 MPa and an ultimate strength of 1100 MPa.

Australian Standard AS4968 provides guidance about the assessment of the endurance limit for high-tensile strength kingpins. The standard specifies that when an 'FEA steady-state test force' equal to a pushing or pulling load of 0.6x D-Rating is applied, the maximum von mises stress everywhere should be less 40% of UTS. FEA Modelling of kingpins by one of the authors (Longhurst) has shown that the stress levels that occur in a common 50 mm kingpin 'bolt-in' design that has a D-value rating of 190 kN, is about 440 MPa when a pushing or pulling force of 114 kN (0.6 D-value) is applied.

Assuming the D-value rating of 190kN is the maximum peak load, then the FEA predicts a maximum stress of 440/0.6 = 733MPa. Comparing this to the yield strength of 990MPa gives a FoS of 990/733 = 1.35 for low cycle stress compared to material yield strength. This estimate is based upon two important assumptions; which are:

- The abnormal load is applied to the same part of the coupling as does the normal load.
- The D-value load is the peak operation force that could occur.

Considering the first assumption, the crash forces applied to a kingpin could be applied by the fifth-wheel skid-plate pushing into the flange section of the kingpin immediately above the 50 mm diameter working section of the kingpin (for 50 mm hardware, which is commonly used in Australia). The top of the kingpin shaft is substantially thicker than the working section that is clasped by the fifth-wheel jaws. The bending moment and stresses arising from an abnormal load applied above the 50 mm diameter section that is clasped by the jaws, are consequently lower than might be expected. Therefore, the kingpin is stronger than might be expected based upon the D-value level, which is tested with forces applied to the 50 mm diameter section. This assessment is not true for the fifth wheel. The legs and attachment hardware of a fifth wheel experience the abnormal forces and moments directly.

The discussion so far has not considered what normal load levels occur in practice. It may be that the D- and V-value ratings are conservative and that the operational loads implied by these ratings are never reached. Therefore, the maximum operational forces that might occur need to be determined.

3.3 Maximum Operational Forces for a B-Double Configuration Truck

The discussion will focus on a B-double truck because this is a common type of vehicle on sub-urban roads in Australia and because B-doubles are likely to become increasingly common in Argentina and eventually in Europe.

The minimum D-value required for the mechanical coupling(s) between vehicle parts in a combination vehicle is specified in ISO standard 18868. The horizontal coupling strength requirements for a B-double combination truck, which has two fifth-wheel couplings is calculated using the following formula:

D-value requirement at truck-tractor = $4.9R_2 (T_2 + 0.08 \times R_2) / GCM$ (kN)

R_1 = total laden weight on the first trailer. Assume that the laden weight of the second trailer is the same.

R_2 = total mass (tonnes) carried by the two trailers, including the load imposed onto the truck-tractor.

T_2 = the sum of the truck-tractor axle loads (tonnes). That is, the truck-tractor GVM.

GCM = the total combination vehicle weight (tonnes).

Tare of the truck-tractor = uncoupled weight of the tractor-trailer.

Example 1: Assume GCM = 62 t, T_2 = 22 t, R_2 = 54 t. Truck-tractor tare weight = 8 t.

The D-value requirement for the truck-tractor coupling = 112.3 kN. (“$D_{minimum}$”)

The maximum normal operational forces that occur can be calculated if the worst-case operating conditions can be identified. The three obvious candidate conditions are pulling up a steep hill or severe deceleration under braking.

3.3.1 Maximum Hill Climbing Force in Example 1

On an 18 % grade, which is the slope specified in UN Regulation 13 for parking performance, the low-speed pulling force between truck-tractor and the trailers in the example is:

$R_2 \times 9.8061 \times \sin \Theta = 94$ kN $\qquad \Theta = 10.2^o$

18% grades are never experienced on the sealed public road network in Australia, at least as experienced by loaded B-double vehicles which require good quality roads. Therefore, this choice for maximum operational force is rejected.

A more realistic maximum grade that a B-double vehicle could experience is 10%. The maximum low-speed pulling force is then 53 kN.

3.3.2 Maximum Deceleration Force in Example 1

If somehow a deceleration of 0.5g was achieved by the truck-tractor brakes acting alone, the force acting on the first coupling would be R_2 x 0.5 x 9.8061 = 265 kN. This situation is unrealistic because such a high deceleration could not be achieved by the truck-tractor brakes acting alone. Therefore, this choice for maximum operational force is rejected.

If the laden vehicle stops at a deceleration of 0.5g due to the braking action of the two trailers only, the horizontal deceleration force acting via the first coupling onto the truck-tractor = Tare x 0.5 x 9.80161 = 39kN. This could plausibly occur because the trailers have brakes on six axles. If they are disc brakes the retardation force capability might be high.

2.3.3 Maximum Operational Force in Example 1

The maximum low speed pulling force is larger than the maximum practical braking force. Hence the maximum operational force = 53 / 112.3 = 47 % of the D-value requirement; and

53 / 62 x 9.8061 = 9 % of the total vehicle weight.

3.3 Maximum Operational Forces for a Truck Pulling a Centre-Axle Trailer

For a truck pulling one centre-axle trailer the minimum D- and V-values are:

D_c-value requirement = 9.8062 x T_2 x R_3 / {T_2 + R_3} (kN)

V-value requirement = α (X^2 / L^2) C

α = 2.4 m/s^2 for a spring suspension and on the towing vehicle;
X = length of the loading area of the trailer, in (m);
L = drawbar length in (m);
C = sum of the axle loads on the centre-axle trailer in tonnes.
T_2 = total weight of the truck.
R_3 = weight on trailer axle group.

Example 2: Assume GCM = 42 t, T_2 = 22 t, R_3 = 20 t. Weight on coupling = 0.1 x R_3 = 2 t.

Truck tare weight = 8 t. Rear spring suspension on the truck.
Trailer drawbar length L = 4m. Trailer loading length X = 6m. C = 0.9 x 20t = 18 t.

Minimum coupling D_c-value = 9.80621 x 22 x 20 / 44 = 98 kN.

Minimum coupling V-value = 2.4 x 1.5^2 x 18 = 97 kN.

Centre-axle trailers are usually pulled by either automatic pin couplings or pintle hook couplings. Both these types of couplings are used in association with a towing eye that is installed onto the drawbar.

3.3.1 Maximum Hill Climbing Force in Example 2

On an 18 % grade, which is the slope specified in UN Regulation 13 for parking performance, the low-speed pulling force between truck-tractor and the trailers in the example is:

R_3 x 9.8061 x Sin Θ = 35 kN Θ = 10.2^o

18% grades are very rarely experienced on the sealed public road network in Australia but could be traversed by a truck and single-trailer combination.

3.3.2 Maximum Deceleration Force in Example 2

Because the weights of the truck and the trailer are similar in this example, a maximum deceleration due to the brake action of one vehicle acting alone is likely to be much less than 0.5g. A level of 0.3g might be achievable.

If the trailer-only was braking the vehicle and if a deceleration of 0.3g was achieved, the horizontal force on the coupling would be 0.3 x 9.8061 x 22 t = 65 kN.

3.3.3 Maximum Operational Force in Example 2

The maximum operational force is assessed to be braking effort due to one vehicle only that achieves a deceleration of 0.3g.

The maximum operational force = 65 / 98 = 66 % of the D_c-value; and

65 / 42 x 9.8061 = 16 % of the total vehicle weight.

3.4 *Factors-of-Safety* Specified in Couplings Rules

Neither ISO standard 18868 nor UN Regulation 55 explicitly declare a *Factor-of-Safety* requirement. Consequently UN R55 does not require a proof test and there is no requirement for the manufacturer to declare or prove a *Factor-of-Safety*.

The situation in the USA is more complicated. There is no mandatory technical standard for fifth wheel couplings in the USA that we have identified. There is a technical standard for kingpins - SAE J133. Hook couplings should comply with SAE J847.

The J133 specifies that a 2” (~ 50 mm) kingpin must not break when a static proof test of 115 % of the total pulled load weight. Additionally the kingpin must withstand a cyclical force of 0.4 x total pulled weight. Therefore, SAE J133 has both a proof test and a fatigue test.

Considering the B-double in Example 1, the proof test requirement from SAE J133 for the first kingpin is 1.15 x 54 t = 609 kN, which is 5.42 x $D_{minimum}$. The maximum operational force of the B-double is 53 kN. The implied minimum *Factor-of-Safety* for the first kingpin based on the SAE J133 proof test requirement is 1.15 x 54 x 9.8061 / 53 = 11.5. If the *Factor-of-Safety* were based upon endurance limit rather than the yield limit, then the minimum *Factor-of-Safety* would be 0.4 x 11.5 = 4.6.

Considering hook couplings that are intended to support between 5% and 20% of the trailer weight (application type II), SAE standard J847 specifies that the hook coupling should be proof-tested by application of a single test force R, as is illustrated below – Figure 10.

The proof force R in Example 2 is then $\sqrt{(1.15 \text{ x } R_3)^2 + (0.5 \text{ x } R_3)^2} = 246$ kN

This force R is: 3.78 x maximum operational force (65 kN). That is, a minimum *Factor-of-Safety* exceeding 3 is implied, based upon static (yield) force considerations. If the factor of safety considerations were based upon endurance then the fatigue limit might again be set at 40 % of yield strength, then the implied minimum *Factor-of-Safety* = 0.4 x 3.78 = 1.5.

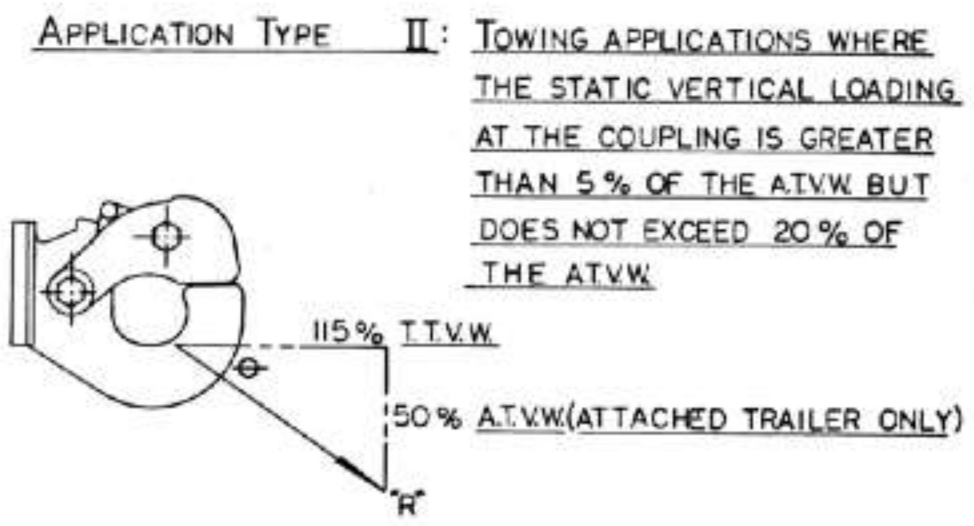


Figure 10 The proof-test specification that is in SAE J847.

3.5 Operational and Crash Forces for the 62t B-Double

The operational and crash forces for the B-double of Example 1 are summarized in Table 4 below.

Condition	**Maximium operational force (kN)**	**Minimum fifth wheel D-value required (kN)**	**Peak crash force at front fifth wheel (kN)**	**Peak crash force at the rear fifth wheel (kN)**
Laden (62t) at initial speed = 15 km/h	53 (47%)	112.3 (100%)	1600 (1425%)	1100 (980 %)
Laden (62t) at initial speed = 30 km/h	53 (47%)	112.3 (100%)	1700 (1514%)	1160 (1033 %)
Laden (62t) at initial speed = 60 km/h	53 (47%)	112.3 (100%)	1900 (1692%)	1380 (1229 %)

Table 4 Comparison of the normal and abnormal forces that the couplings on the B-double truck in Example 1 might experience. Forces are expressed as a percentage of the minimum D-value requirement.

4. Discussion

Three distinct impact forces arise when a truck hits a immoveable barrier. These are:

1. Shockwave forces, which excite the resonances in the impacting vehicle.
2. Crush forces (plastic deformation) at the front of the impacting vehicle.
3. Compression forces in the elastic zone (which is behind the crush zone) in all parts of the vehicle.

The shock wave forces travel at the speed of sound in the chassis ladder and is dissipated very quickly. It is mainly confined to the truck-tractor chassis ladder. This wave excites the natural frequencies of the vehicle and multiple reflections occur. Most of the collision energy is absorbed in the crush zone. This is where very high collision forces occur. Whilst there is some elastic energy stored and returned in the crush zone, mostly the force and energy is absorbed. The force that is transferred to from the crush zone to the elastic zone in the examples might be 20% of the barrier force. Further information about absorption of force in crush zones of cellular metal can be found in a paper by Tan and Qu, Ref [6].

The mechanical couplings are likely to be located well behind the crush-zone (plastic-zone). They experience the initial shock wave force, which dies away extremely quickly and then the elastic force which persists for longer and redirects the vehicle. The force that reaches the mechanical couplings varies approximately linearly with speed and not with speed squared (see Figures 8 & 9.

If the first mechanical coupling on a B-double or semi-trailer combination vehicle breaks, there is a significant risk that the driver of the vehicle will be seriously or fatally injured as a result of a disconnected trailer hitting the cabin. Therefore, the *Factor-of-Safety* that is inherent in the mechanical coupling is an important consideration. It is acknowledged that an unattached trailer might catch on the rear of the truck-tractor which could provide additional protection. However, this protection is problematical because the trailer might also rise-up and come clear.

Based upon the practical considerations that led to Table 4, there is a *Factor-of-Safety* in the D-value specification of about 2. The D-value certification of couplings according to UN ECE Regulation 55 involves endurance testing. If as is often assumed, the design forces should be less than 40% of yield, then a *Factor-of-Safety* of 2 / 0.4 = 5 is implied. This level is less than values needed to ensure that the front coupling does not break during a relatively low-speed impact at 15 km/h with an immovable barrier. However, it is acknowledged that an immovable barrier test is unrealistic.

The UN ECE Regulation does not require that a proof test be conducted. This is a failing because there is no *Factor-of-Safety* specification. Because the mechanical coupling represents a single point of failure, it is basic and good engineering practice to specify a minimum *Factor-of-Safety*. In out opinion an appropriate minimum 'regulatory' *Factor-of-Safety* for a mechanical coupling based upon the D-value is 5. That is, the coupling should be able to withstand a proof force of 5D without breaking. This proposal is based upon tensile and compressive failure considerations rather than endurance considerations.

The USA technical standards for mechanical couplings, which are based upon SAE standards, do have a proof test specification. The kingpin standard J133 also contains an endurance test specification. In the authors' opinions both endurance tests and proof tests are necessary.

The authors believe that new couplings regulations (which could be Global Technical Regulations under the UN ECE umbrella) should be developed. These should contain both an endurance-test specification and a proof-test specification.

5. References

1 P.M. Hart, *Occupant protection considerations in heavy-vehicle crashes*, HVTT11, Melbourne, March 2010.

2 United Nations Regulation 55, *Uniform provisions concerning the approval of mechanical coupling components of combinations of vehicles.* Revision 1.

3 SAE Standard J133-2003, Kingpin Performance

4 *SAE Standard J847-1987*, Trailer Tow Eye and Pintle Hook / Coupler Performance.

5 R C Juvinall and K M Marshek, *Fundamentals of Machine Component Design*, 2003, Wiley & Sons.

6 H Tan, S Qu, Impact of Cellular Materials, Cellular and Porpus Metrials in Structures and Processes, CISM International Centre for Mechanical Sciences Volume 521, 2010, pp 309 – 334, Springer Link.