PERFORMANCE CRITERIA, DESIGN AND CRASH TESTS OF EFFECTIVE REAR UNDERRIDE BARRIERS FOR HEAVY VEHICLES

George Rechnitzer
Accident Research Centre, Monash University, Australia
Chris Powell
Department of Civil Engineering, Monash University, Australia
Keith Seyer
Department of Transport and Regional Services Australia
Paper No 218

ABSTRACT

During the 1990s a series of projects carried out at Monash University, together with reference to studies in Europe, USA and Brazil, have led to the development of recommendations for performance criteria for rear underride (or underride) barriers for heavy vehicles over 3.5 tonne. The work focussed on the development and crash testing of high capacity, yet practical, rigid and energy absorbing systems catering for both centred as well as offset impacts. The limitations of current international regulations are discussed, and recommendations for improved performance criteria to achieve effective rear underride barriers are presented.

INTRODUCTION

The usual occupant protection features built into cars such as seatbelts, airbags, crush zones are bypassed and ineffective in underride crashes involving heavy vehicles. This crash type presents severe injury risk to vehicle occupants, and represents extreme geometric and stiffness incompatibility [1, 2, 3,4, 5] between heavy vehicles and passenger cars.

Since about 1992, a program of work for the development and crash testing of rigid and energy absorbing underride barriers has been undertaken at Monash University by for VicRoads and the Department of Transport and Regional Services. From the work carried out at Monash University, and Internationally, performance and design criteria have been developed for rear underride barriers for heavy vehicles over 3.5 tonne. This work has focussed on the development and crash testing of high capacity, yet practical, rigid and energy absorbing systems catering for both centred as well as offset impacts.

Such tests have demonstrated that practical high capacity barriers can be fitted to trucks ranging from light through to heavy.

This paper includes a review of design principles for rear underride barriers, and current international regulations and their limitations. It is the aim of this paper to help promote the adoption of upgraded and realistic standards for truck underride barriers in Australia and internationally.

THEORETICAL ESTIMATE OF IMPACT FORCES ON UNDERRIDE BARRIERS

An essential starting point for determining the likely magnitude of forces that underride barriers need to withstand comes from the equation of impact mechanics. Murray (1988) develops the following basic equations relating impacts between cars and heavy vehicles [6].

Calculation of forces for impact with rigid underride barriers on heavy vehicles

Consider two vehicles approaching each other in opposite directions, the impacts are considered to be essentially plastic, such that the vehicles do not bounce apart, and after impact continue together with a combined mass of (m_1+ m_2) at speed V_3, in the same straight line.

Definition of variables:
a_1 = average impact deceleration of truck (m/s^2)
a_2 = average impact deceleration of the car (m/s^2)
F = average impact force acting between the vehicles (N)
m_1 = mass of truck (kg)
m_2 = mass of car (kg)
s = car crush (m), or overall deformation
t = impact duration (seconds)
V_1 = velocity of truck (m/s)
V_2 = initial velocity of car (m/s)
V_3 = post impact speed of vehicles (m/s)
V_c = closing velocity = V_1 + V_2
V_B = closing velocity = V_1 + V_2
G = energy absorption capacity of barrier

Average force on barrier. The average force F, on the barrier can be estimated using the following Equation 1.

$$F = \frac{m_1 m_2 V_c^{\,2}}{2(m_1 + m_2)s} \qquad (1).$$

Average deceleration of the car (m_2). This is given by Equation 2.

$$\boxed{a_2 = \frac{m_1 V_c^{\,2}}{2(m_1 + m_2)s}} \qquad (2).$$

Average deceleration of the heavy vehicle (m_1). This is given in Equation 3.

$$\boxed{a_1 = \frac{m_2 V_c^{\,2}}{2(m_1 + m_2)s}} \qquad (3).$$

Impact duration. The duration of the impact is given by Equation 4.

$$\boxed{t = \frac{2s}{V_c}} \qquad (4).$$

Using equations set out above, the average impact force and other crash parameters can be determined and are summarised in Table 1 below:

Table 1.
Example theoretical values for a car impacting the rear of a 40t truck with rigid underride barrier (V_1=0)

Car mass kg	V_2 car speed km/h	s car crush m (estim.)	F average impact force kN *(Eq.1)*	a_2 car average decel. G *(Eq.2)*
1000kg	50	0.6	157	16
	75	0.8	263	27
1400kg	50	0.6	218	16
	75	0.8	367	27
1800kg	50	0.6	277	16
	75	0.8	465	26

Of particularly interest is the magnitude of the impact forces as these form the basis, together with the crash test results (discussed in the following sections) for determining the appropriate strength requirements of an underride barrier. The average force ranges from **157**kN for the light car impacting at 50km/h to **465**kN for the heavy car (1800kg) impacting at 75km/h.

The calculations given in Table 1 are approximate and are based on impact with a 40t truck, which can be regarded as providing an upper bound or worst case scenario for such impacts. For truck masses in excess of 10t the forces only increase by around 10% compared with an impact with a 40t truck.

The forces and accelerations given here are *average* values, and can not be used directly for design without also considering the peak values measured from the crash tests.

Calculation of forces for impact with energy-absorbing underride barriers on heavy vehicles

Although well-designed rigid underride barriers provide the necessary interface between the car's occupant protection systems and the truck, the provision of energy absorption capacity on the truck can further reduce the crash severity at a given speed. Provision of energy absorption can also be seen as a measure to increase the survivable impact speed.

The following section summarises the key parameters and issues relating to design of an energy absorbing guard [6, 7] and follows on from the calculations for impacts between cars and trucks set out previously.

Consider the effect of adding to the truck an energy absorbing underride guard with a capacity of ***G.*** The effect of the guard is to share in the energy absorption, and reduce the car's share by some factor, for a given impact speed.

The energy absorbed during the impact can be rewritten in terms of the *energy absorption of the underride barrier*, ***G*** and the *crush energy of the car.* The crush energy of the car can be defined in terms of the *equivalent barrier speed* $\boldsymbol{V_B}$ (the impact speed that would result in the same deformation of the car) and is thus a measure of impact severity); or in terms of *closing velocity* ***Vc***.

Equivalent barrier impact speed. Equation 5 can be used to calculate the equivalent barrier impact speed $\mathbf{V_B}$, for a given closing velocity $\mathbf{V_c}$ of car and truck energy absorbing capacity **G**.

$$V_B = \sqrt{\left(\frac{m_1 m_2 V_c^{\,2}}{2(m_1+m_2)} - G\right)\frac{2}{m_2}} \quad (5).$$

To gauge the benefits of the guard for various levels of energy absorption G, the ratio V_c / V_B can be calculated.

$$\frac{V_c}{V_B} = \sqrt{(1+\frac{G}{(0.5 m_2 V_B^2)})(\frac{m_1+m_2}{m_1})} \qquad (6).$$

For $m_1 >> m_2$, Equation 6 simplifies to:

$$\frac{V_c}{V_B} = \sqrt{(1+\frac{G}{(0.5 m_2 V_B^2)})} \qquad (7).$$

The average force F, on the barrier can be estimated using the Equation 7, in terms of V_B.and ***G.***

$$F = \frac{1}{S}\left(\frac{1}{2} m_2 V_B^{\,2} - G\right) \qquad (8).$$

In Equation 8, **S** is the crush of the car after the energy absorption G of the truck has been expended, that is, after the barrier has 'bottomed' out.

Benefits of energy absorption. Table 2 sets out example results for the reduction in crash severity (V_B) for a range of car masses, impact speeds, and guard energy absorption capacities.

Table 2.
Examples of variation of V_B with different impact speeds and truck energy absorption level G

	G kJ	10kJ	40kJ	60kJ
$\mathbf{m_2}$ Car mass kgs	$\mathbf{V_C}$ km/h	$\mathbf{V_B}$ Equivalent Barrier Speed km/h		
1000	50	47	37	30
	75	72	67	63
1400	50	47	41	36
	75	73	68	66
1800	50	48	43	39
	75	72	69	67

The analysis in Table 2 shows that even for a lighter weight car (1000kg), a 10kJ barrier results in a small reduction in crash severity (about 3km/h). Therefore such low levels of energy absorption do not provide significant reductions in crash severity. This is not surprising, as this level of energy absorption is very small compared with the kinetic energy (KE) of the impacting vehicle. For a 1000kg car, the KE at 50km/h is 97kJ; at 75km/h the KE is 217kJ.

To be effective in terms of reduction in crash severity, energy absorption levels of the underride barriers need to be a significant fraction of the car's KE. This can be seen from Table 2 for the 60kJ capacity barrier: e.g. for the 1000kg car at 50km/h, V_B reduces to around 30km/h; at 75km/h V_B reduces to 63km/h.

This type of analysis can of course be extended to include assessment of performance requirements for energy absorbing front underride barriers (see [6, 7]).

RIGID REAR UNDERRIDE BARRIER TESTS

The design and testing of effective rigid rear underride barriers involved the use of theoretical analysis (as presented earlier), static load testing and crash tests. Three dynamic impact tests [8] were conducted as follows:

- Test No. 1. – Offset impact (50%) with truck without an underride barrier.
- Test No. 2. **-** Centred impact with truck with underride barrier.
- Test No. 3. **-** Offset impact (50%) with truck fitted with an underride barrier.

The underride barrier (shown in Figure 2) was designed for two off centre loads of P_2 = 150 kN, offset loads of P_1 = 100 kN, and centre load of P_3 = 100kN. The actual load capacity exceeded these values. The height above ground of the underside of the barrier on the unladened truck was 500 mm. In the design careful consideration was given to the load path into the truck chassis, to avoid lateral buckling and distortion of the chassis members.

Two almost identical barriers were fabricated and fitted to two almost identical trucks. The barriers were first tested with static forces (Figure 2, Photo 2) using the procedure in E.C.E.-R58. The main strut members of the barrier members were strain gauged to measure the impact forces on the barrier.

The tests were conducted with three same model cars with masses of 1420 kg. They were driven by remote control by their own engines to impact with a speed of 50 km/h. For Tests 2 and 3 a non-instrumented dummy was in the driver's seat; for Test 3 two shop-window dummies were in the front seats. In each case conventional lap and sash seat belts were used. In each test the truck was ballasted to 10 tonnes. The brakes of the truck were not engaged.

Figure 2 shows the static load tests and the crash tests at the moment of impact.

Three forms of high-speed instrumentation were used to capture data relating to decelerations, forces and displacements. Firstly the forces acting between the cars and trucks were measured by 16 strain gauges attached to the barriers (4 per strut). Data was captured at 10,000Hz. Secondly, high-speed video cameras operating at 500 frames per second and decals attached to the cars were used to evaluate the deceleration of the cars at any time before, during and after impact. Finally, accelerometers were attached to the bodywork of the cars.

Results for crash Tests 1, 2 and 3

Test No. 1. Severe underride occurred, with a peak deceleration of 130 m/sec^2, which is, not unexpectedly, considerably lower than for Tests 2 and 3. The mass of the car was 1420 kg and hence force can be calculated and plotted giving a peak load of 185 kN. The truck's tray penetrated as far as the B-pillar

Test No. 2 – The barrier performed very well with minor deformation only and no truck damage.

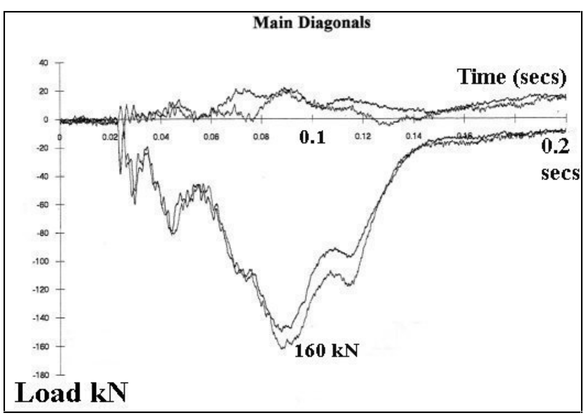


Figure 1. Load - time graph for the diagonals on the underride barrier in Crash Test 2.

The measured peak forces (Figure 1) in the two main diagonal struts were very similar and averaged 150 (total 300kN). The peak deceleration was seen to be 200 m/sec^2. With the car mass of 1420 kg, this represents a maximum crash force of 285 kN, in good agreement with that measured with the strain gauges. The crush length of the car was measured as 0.5 m when Equation (2) is used the average force is found to be 232 kN. Allowing for peaks this result again suggests a maximum force of about 300 kN.

Test	Test photos
Test 1. 50km/h Offset, No under-run barrier 10 t truck	
Proof load testing (offset) of prototype rigid underride barrier before crash test.	
Test 2. 50km/h Centred; Car mass =1,420; truck 10t	
Test 3. 50km/h Offset (50%), Car 1420kg, Truck 10t	

Figure 2. Crash tests with rigid truck, with and without the prototype underride barrier systems.

Test No. 3 **-** In this offset impact the barrier performed very well, and sustained minor damage only. Through a fault in the data triggering system, strain gauge readings were not obtained, however, analysis of the high speed video record for velocity and deceleration, gave a peak deceleration of 220 m/sec^2 , equivalent to a peak force of 310 kN, similar to Test 2 results.

ENERGY-ABSORBING REAR UNDERRIDE BARRIER SYSTEM DEVELOPMENT

Following from this work on rigid rear underride barriers the Federal Office of Road Safety funded an extension to this work to design, develop, and test an energy absorbing rear underride barrier (see Fig. 3).

The project involved the design and testing of a suitable energy absorbing module; the construction and static testing of a prototype energy absorbing underride barrier; and crash testing of the prototype energy absorbing rear underride barrier, including use of Hybrid 3 dummies.

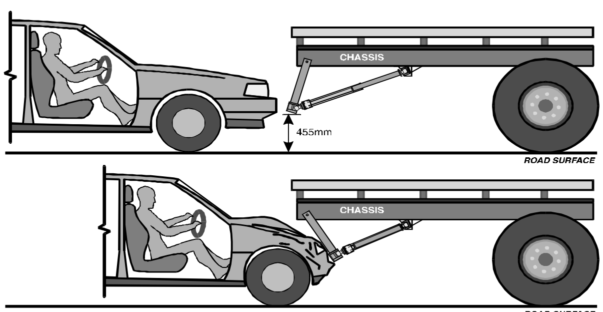


Figure 3. Illustration of energy absorbing rear underride barrier system on rear of truck, before and after impact.

Energy absorbing module design

From the preceding theoretical analysis, the energy absorbing system, to be effective, has to have a capacity of around 60kJ. It also needs to cater for lightweight and heavier vehicles. Due to the difference in force level required for this range of vehicle masses, a low force threshold is needed to activate the energy absorption mechanism for low mass vehicles. From a consideration of all these factors it was concluded that the system be designed for a 40kJ to 60kJ energy absorption capacity (shared amongst 4 units), at 400mm deformation.

The system comprised a tube in tube system, using square FRP (fibreglass reinforced epoxy) tubes as the crushable energy absorbing medium. The final selected tube is a 38mm square by 3.2mm wall thickness, 500mm long (Figure 5). The unit is contained within a 65mm square thin walled (1.6mm) steel tube, with a smaller 50mm square tube (1.6mm) reacting against the FRP unit, and acting as the 'piston'.

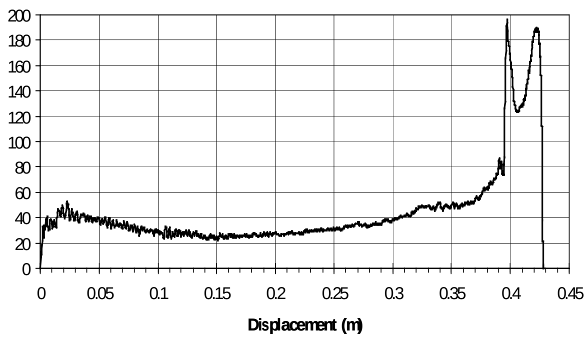


Figure 4. Load-deflection curve from the drop test on the tube-in-tube FRP system.

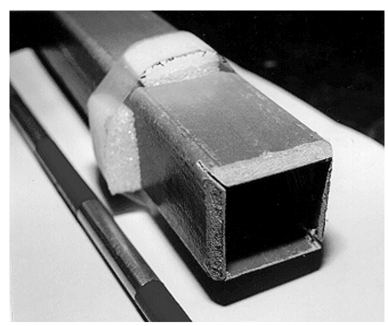


Figure 5. End preparation of FRP unit within the 65mm steel box showing 60deg. chamfer and corner cuts.

Overall crush length for the FRP tube is 400mm, at an average force of between 30-40kN (Figure 4). The steel tube system had an axial load capacity in excess of 180kN, at which stage local buckling of the tube wall occurs. Each of the four tubes absorbs around 16kJ of impact energy.

Static Testing of the first energy absorbing Prototype System

To test the full system a frame was constructed to simulate the rear structure of a truck (see Figure 7, Test 4). The underride barrier system consisted of the 4 energy-absorbing units, with a crossbeam and hangers. Static testing was carried out using two jacks loading simultaneously to simulate both centred and offset impacts.

Following the initial testing, design modifications were made which included the use of ball joints at each end of the strut so as to shield the struts from bending moments and to ensure good rotation in all directions. The system deformed satisfactorily with energy absorption of around 50kJ for the offset test and 60kJ for the centre static test.

Crash testing of the first prototype energy absorbing underride barrier system

Three crash tests were carried out on the prototype system at 48km/h. The prototype system was transported to the test barrier and fixed to the face of the concrete barrier (simulating a very high mass truck).

<u>Test 4 – centred impact, 48km/h</u>. The first test involved a large family sedan (1800kg), donated by Ford Australia, in a centred impact (see Figure 7). In this test the system performed very well with underride prevented and the two centre energy absorbing modules fully compressed (400mm) and the two outer modules by 270mm.

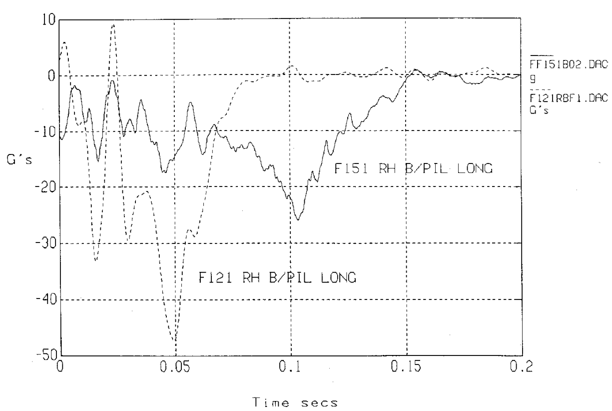


Figure 6. Crash pulse for 48km/h impact (1800kg sedan). Solid line is for impact with the energy absorbing rear underride system; the dashed line is a standard full frontal concrete barrier test.

A comparison of the acceleration pulse (Fig. 6), measured at the base of the B pillar of the car, for the normal rigid concrete barrier and the energy absorbing system shows a marked decrease in severity. The peak acceleration reduced from 50G to 25G with the pulse duration increased from 70ms to 150ms, with both results indicating a significant reduction in crash severity and important benefits in

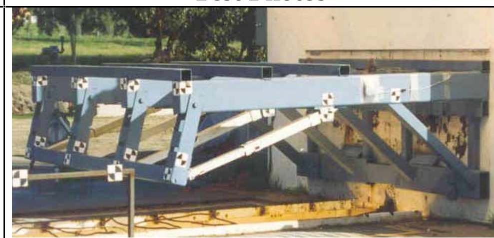
Test	Test Photos
Test 4. 48 km/h Centred, Car mass = 1800kg.	 
Test 5. 48 km/h Centred, Car mass 1700kg, & HIIIs	
Test 6. 48 km/h Offset, Car - 1700 kg, & HIIIs	
Test 7. 75 km/h Centred, car - 1350kg; truck - 9,100 kg With HIIIs	 

Figure 7. Crash tests 4-7 with the prototype energy absorbing underride barrier systems.

terms of occupant protection. It is noted that as the barrier height of 450mm was above the car's front longitudinals, the impact forces were concentrated on the car's engine via the underride barrier's crossbeam.

Tests 5 (Centred) and Test 6 (offset), 48km/h. The next two tests were carried out at Sydney's Crashlab facility and included both driver and front seat passenger Hybrid 3 ATDs (see Figure 7). Figures 8 and 9 show the crash pulse at the base of the vehicle's left and right B pillars. Once again the results show relatively low peak accelerations (20G for the centred test, 16G for offset) and an extended duration of up to 220ms for the offset test.

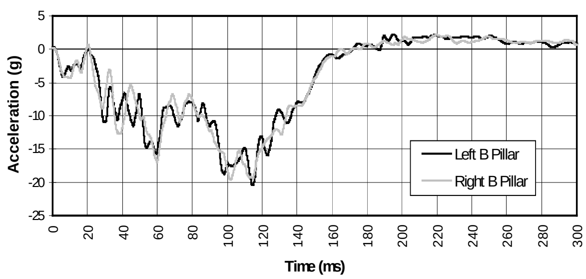


Figure 8. Test 5 -acceleration pulse at base of B pillar, 48km/h centred test with prototype energy absorbing underride barrier system (m=1700kg)

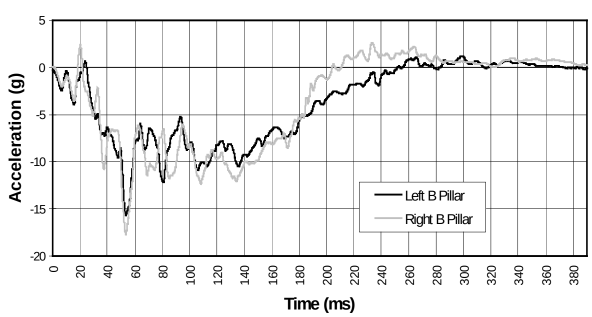


Figure 9. Test 6 -acceleration pulse for base of B pillar, 48km/h offset test with prototype, energy absorbing underride barrier system (m=1700kg).

In terms of the performance of the energy absorbing modules, some problems were identified from these crash tests. These included the need to increase the load capacity of the steel tubes, as the peak force measured was 200kN in the offset tests (particularly onerous loading due to the engine striking the corner of the barrier directly) and compression failure and buckling of the unit (Figure 10). In addition the ball joints require longer 'necks' to ensure adequate rotation prior to locking up. Significantly the measured injury criteria for the Hybrid 3 dummies showed low values and lower than normally measured for this vehicle type, as shown in Table 3, below.

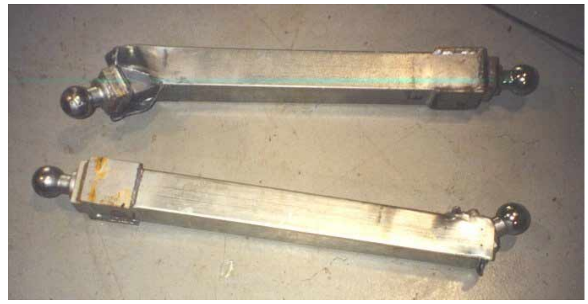


Figure 10. Compression failure of tubes due to overload (wall thickness to low).

Table 3.
Hybrid 3 results for Tests 5 and 6, comparison with ADR 69 full frontal rigid barrier test

	ADR 69 Test result		**Centred**		**50% Offset**	
Injury Criterion	Driver Dummy	Passeng. Dummy	Driver Dummy	Passeng. Dummy	Driver Dummy	Passeng. Dummy
Head Injury (HIC)	848	699	566	271	229	89
Max. femur compressive load (kN)	3.3	2.1	0.6	1.2	1.2	1.1
Chest compression (mm)	36.6	39.1	43	38	32	30

Test 7, centred, 75km/h. Following on from the lessons learnt from the initial prototype development work (Tests 4, 5 & 6), a redesigned energy absorbing barrier was fitted to a truck loaded to 9.1t (GVM) and tested in a full scale test in 1997, at the Autoliv crash test facility in Victoria.

The four energy absorbing struts were redesigned, using higher load capacity circular steel tubes, with ball joint ends. The same type FRP tubes were retained.

The underride barrier was fully redesigned incorporating a robust rear hinged frame mounted from the trucks floor structure. A cross beam bolted to the chassis is used to pick up the loads from the struts. Strain gauges were fixed to the four struts to measure the strut forces during the impact. The redesigned barrier system is shown in Figure 11, and on the truck in Figure 7.

Figure 11. 3D CAD image of the underride barrier system for Test 7.

The crash test used a 1350 kg car, (no airbag) with passenger and driver Hybrid III dummies, impacting the rear of the truck at 75 km/h. In this test the barrier performed very well, with underride prevented, the truck was undamaged with some minor deformation of the barrier itself. The car maintained moderately good cabin integrity, although the crash was clearly severe.

The energy absorbing struts absorbed around 50kJ, at a stroke of 300mm. The energy absorbing tubes were fully reusable, just requiring the insertion of a new fibreglass energy-absorbing module.

Table 4. Comparison of Hybrid 3 results for Test 7 at 75km/h with NCAP full frontal barrier test at 56km/h.

	NCAP centred impact m=1320kg 56km/h Test result		Test 7, centred impact; m=1350kg 75km/h impact (9.1 t truck with energy absorbing barrier)	
Injury Criterion	Driver HIII	Passeng. HIII	Driver HIII	Passeng. HIII
Head (HIC)	1499	1223	1842	1205
Femur (kN)	L 9.41 R 1.93	3.16 1.05	14 4.1	2.56 6.57
Chest 3ms clip	59.6g	49.1	56.2g	48.2g

The comparison of the Hybrid III results from Test 7 at 75 km/h with the equivalent 56km/h NCAP full frontal barrier test (Table 4), indicates that the energy absorbing system reduced the crash severity for the front passenger (HIC of 1205 vs 1223) to that of a 56km/h impact. For the driver the impact was a little more severe than the NCAP test (HIC 1842 vs 1499; high femur load of 14kN). Had the car been fitted with airbags HIC values below 1000 would be expected [12], although femur loads may still have been excessive.

__Barrier and strut forces.__ The force measured in the centre strut in the 75km/h crash test is given in Figure 12. The peak force measured for each of the two centre struts is 350kN, while for each of the two outer struts it is around 50kN. This latter value is the peak load sustained by the fibreglass energy-absorbing module in its collapse mode. The two inner struts fully locked up and provided 300mm of deformation. The outer struts did not lock up, and had additional travel remaining, as expected.

It is important to note that the measured peak strut force of 350 kN only lasts for a few milliseconds (strain readings are at 10,000Hz- ie. 10 readings per ms). Thus for the purpose of designing the struts a more appropriate load capacity per strut (all struts) would be in the order of 200 kN (see Figure 12).

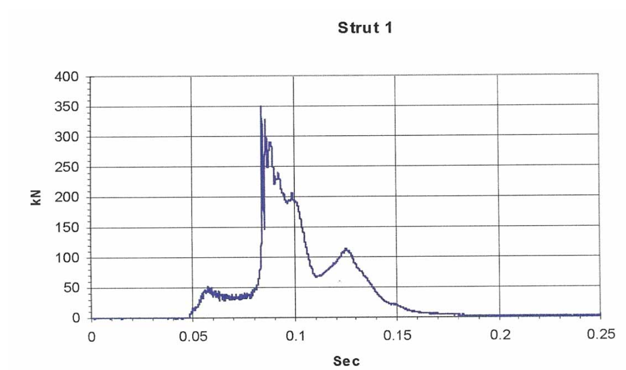


Figure 12. Test 6. Inner Strut - force vs time. 75km/h crash test. The peak load of over 300kN lasts for a few milliseconds only. Note the initial part of the curve (from t = 0.05s) shows the crush response of the FRP tube.

DISCUSSION AND CONCLUSIONS REGARDING REAR UNDERRIDE BARRIER DESIGN

A comparison of ‘theoretical’ calculations (using Equations 1-7 presented in this paper, with the crash test results shows these to be very useful (Table 5) and help underpin the values for barrier load capacity. However, the peak loads measured in the crash tests can exceed these average values by 50 %.

These test results suggest that individual struts (4-strut system) should have a load capacity of at least 200kN under dynamic loading. The underride barrier as a whole should have a load capacity in excess of 350kN. This applies for both centred and offset conditions.

Table 5.
Summary of crash test results from Tests 2 to 7

Test	Speed km/h	car mass kg	truck mass	Peak Strut. Force kN	Total force peak kN	Theory - total force (average) kN
Rigid barrier						
2. Centred	48	1,420	10,000	160*	285	220
3. Offset	48	1,420	10,000	-	310**	220
Energy absorbing						
4. Centred	48	1800	fixed	-	450**	230
5. Centred	48	1700	fixed	-	340**	200
6. Offset	48	1700	fixed	200*	275	200
7. Centred	75	1350	9,100	250+*	500+	300+

Note to Table 5: *Forces in struts measured; **Force determined from F=ma, where 'a' is Vehicle deceleration determined from high-speed video.

These values can be compared with the lightweight rear underride barrier designs developed for the IIHS by Moffat and Wong [2]. They calculated a horizontal load capacity ranging from 356kN (IIHS Guard 1), to 445kN (IIHS Guard 2).

The report by Deleys and Ryder [1] recommends static test loads of (i) 360kN total applied at two points about the centreline (i.e. two loads each of 180kN,; and (ii) 270kN towards the outer edge of the guard. This translates to two point loads (one near the outer edge) of 136kN each.

Deleys and Ryder also note that the actual impact forces measured for full size vehicles (1590kg - 1800kg) at 64km/h can exceed 680kN. They note, consequently, that the actual dynamic capacity of the tested barriers significantly exceeded the calculated static load capacity; and therefore design loads can be less than the measured short duration peak loads. The structures of trucks can readily sustain these realistic impact loads for both centred and offset impacts, using proper engineering attention to the barrier design, and of course the use of well designed connections.

The crash test results clearly highlight the benefits of provision of energy absorption in the order of 60kJ (or more) for rear underride barriers, and barrier deformation capacity of around 400mm.

IMPLICATIONS FOR INTERNATIONAL REGULATIONS and TEST LOAD REQUIREMENTS

The findings from this crash test program (and the theoretical calculations presented) highlight some unnecessary inadequacies in the performance requirement set out in various international standards. These inadequacies will lead to predictable failure of the barriers to prevent underride even at moderate speeds.

Table 6. Comparison of international force performance requirement for underride barriers, and proposed values

Load position	E.C.E R 58 maximum	USA (FMVSS 223/224)	Brazil	Recommended (this study)
	Test Load kN			
Outer P_1	25	50	100	200
Off centre P_2	100	100	150	200
Centre P_3	25	50	100	100

Table 6 compares the various international standards with those proposed from this study. Both the ECE and USA test load values for offset impacts (outer edge) are too low, and will not prevent underride in even low speed impacts. As the ECE load requirements are further reduced for lower mass trucks, these values are even less valid and quite ineffective in terms of underride protection. The recommended force requirements should be applicable for all heavy vehicles with little reduction for lower mass ranges.

The load requirements set out for Brazil in Table 6 are far more realistic. These are based on the extensive work done at Unicamp University in Brazil on improved underride design under the Impact Project [5]. The reader is referred to the excellent Unicamp website at www.fem.unicamp.br/~impact.)

It is noted that the 'Recommended' forces in Table 6 are increased in some cases from our earlier studies

[4, 7, 8, 10]. This is based on the need to allow for higher impact speeds (around 75km/h), and a more realistic recognition of the load capacity needed for offset impacts. It also recognizes that such performance criteria are achievable and practical for the trucking industry.

On a practical note (to emphasis this latter point), small section, lightweight steel struts, such as rolled rectangular steel sections of 75mm*50mm*3mm, are sufficient to resist compression loads of 200kN. These sections only weigh 5.43kg/m.

RECOMMENDATIONS

1. **Barrier test Forces:**

P1 (outer edge)	P2 (off centre)	P3 (centre)
200 kN	200 kN	100 kN

2. **Barrier height:** 400mm

3. **Barrier width:** Within 100mm of the outer frame of the rear of the truck

4. **Energy absorption:** 50kJ minimum

REFERENCES

1. Deleys NJ. & Ryder MO., (1971), Underride/Override Of Automobile Front Structures In Intervehicular Collisions, V1- Rear Underride, Cornell Aeronautical Laboratory, NY, for US DOT, NHTSA

2. Moffat CA and Wong J., (1980). Improved underride protection in moderate speed crashes; Proceedings AAAM Oct. 7-9, 1980.

3. Langwieder, K. And Danner, M. (1987) Priorities In The Active And Passive Safety Of Trucks. Eleventh ESV, Washington, May 1987.

4. Rechnitzer, G. and Foong Chee Wai. (1991), 'Truck Involved Crash Study: Fatal and Injury Crashes of Cars into the Rear of Trucks'. Monash Univ. Accident Research Centre, Report 26.

5. Bloch B & Schmutzler L. O, Improved Crashworthy Design For Truck Underride Guards; 16th ESV, May 31- June 4, 1998, Windsor.

6. Murray, N.W., 1988 A Study of Car-Truck Impacts and the Feasibility of Fitting Energy-Absorbing Guards to Heavy Trucks. Dept. of Civil Engineering, Monash University, 1988.

7. Rechnitzer, G. (1993), 'Truck Involved Crash Study: Fatal and Injury Crashes of Cars into the Front and Sides of Heavy Vehicles'. Monash University Accident Research Centre, Report 35.

8. Rechnitzer, G., Scott G. & Murray, N.W., (1993). 'The Reduction of Injuries to Car Occupants in Rear End Impacts with Heavy Vehicles'; 37th STAPP Conference, 8-10 Nov. 1993, San Antonio, Texas. paper 933123, SAE.

9. Rechnitzer G., Powell C., Seyer K., Development And Testing Of Energy Absorbing Rear Underride Barriers For Heavy Vehicles. 15th ESV, May 13- 16; Paper No. 96-S4-O-10, Melbourne.

10. Rechnitzer G., Design Principles for Underride Guards and Crash Test Results; SAE Heavy Vehicle Underride Protection TOPTEC; April 15-16 1997 Palm Springs, USA.

11. Rechnitzer G, Zou R, Grzebieta R., Madymo Computer Modeling Of Energy Absorbing Rear Underride Barriers For Heavy Vehicles - A Pilot Study; March 1997, Monash University Accident Research Centre. Report 112.

12. Zou R., Rechnitzer G. & Grzebieta, Simulation Of Truck –Rear Underride Barrier Impact. 17th ESV, June 4 – 7, 2001, Amsterdam.

ACKNOWLEDGMENTS

The considerable support and contribution of the Department of Transport and Regional Services Australia (sponsor); Vicroads (sponsor); Monash University Department of Civil Engineering Structural Laboratories; Ford Australia; RTA Crashlab; Autoliv and the Accident Research Centre, is gratefully acknowledged.