Vehicle restraint systems —

Part 2: Fundamentals of highway restraint systems

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Committees responsible for this Published Document

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Contents

Foreword

This part of PD 6634 has been prepared by Subcommittee B/509/1. The other parts in the series are:

— *Part 1: Fundamentals — Database;*

— *Part 3: Development of vehicle highway barriers in the United Kingdom;*

— *Part 4: Development of bridge parapets in the United Kingdom;*

— *Part 5: Development of barrier transitions and terminals;*

— *Part 6: Crashworthy roadside features — Impact attenuators.*

BSI Subcomittee B/509/1, whose constitution is shown in this Published Document, takes collective responsibility for its preparation under the authority of the Standards Committee. The Subcommittee wishes to acknowledge the personal contribution of Mr I. B. Laker.

Over the last 30 years the Department of the Environment, Transport and the Regions (DETR), the Transport Research Laboratory (TRL), the British Standards Institution (BSI) and other organizations have been involved in research, testing, design and the preparation of specifications and standards for vehicle restraint systems such as safety fences, barriers and bridge parapets. Much of this work has been published in the form of Transport Research Laboratory reports, drawings, specifications and standards.

In recent years, particularly since the introduction of quality assurance schemes for both the manufacture of components and the erection of safety fences and parapets, the need for additional advice, guidance and background information has been highlighted. In 1988 the then Department of Transport (DTp) and BSI agreed to the preparation of a comprehensive British Standard or reference manual on vehicle restraint systems.

A steering group of representatives from BSI, DTp and TRL was formed to supervise the project and the following terms of reference were formulated:

"To prepare the draft of a comprehensive document on safety fences, barriers and bridge parapets covering research and development, design, specification, manufacture, installation, repair and maintenance."

It was decided to split the reference manual into several parts and the following groups were formed:

a) Working Group 1 — Part 1 dealing with the fundamentals of safety fences, barriers, parapets and transitions in the UK;

b) Working Group 2 — Part 2 dealing with the specification and layout of safety fences and barriers;

c) Working Group 3 — Part 3 dealing with the installation, inspection and repair of safety fences;

d) Working Group 4 — Part 4 dealing with the installation, inspection and repair of safety barriers;

e) Working Group 5 — Part 5 dealing with all aspects of bridge parapets.

Of these proposed parts PD 6634 forms part 1 and BS 7669-3 forms part 3. Work on the other parts has been suspended.

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

This Published Document is not to be regarded as a British Standard.

Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 47 and a back cover.

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1 Scope

This part of PD 6634 describes the fundamentals of highway restraint systems in terms of their economic justification, impact test techniques, measurement of test data, measurement of vehicle characteristics, impact severity indices, computer modelling and the impact test reporting format with reference to British and European Standards.

2 The economic justification for safety barriers

2.1 General

Early constructions of safety barriers were elementary in design and materials (Freeman, 1934 [1]; Coburn, 1949 [2]), although bridge parapets were more robustly constructed in masonry and steel. Their vehicular impact characteristics were unsuitable for high speed vehicles. On impact the vehicle was likely to be brought to a severe stop with the possibility of overturn (Hewes, 1942 [3]).

Before the 1960s, the use of safety barriers in the UK as an established method of restraining vehicles leaving the carriageway was almost negligible. Exceptions were where there were particularly hazardous areas, such as on bridges and their approaches, or where the roadway ran alongside steep embankments. The rapid growth in traffic after World War II prompted the need to examine the effectiveness of safety barriers both from the consideration of their cost efficiency and from their vehicle impact performance.

2.2 The case for no median safety barriers on motorways

The separation of opposing traffic lanes reduces the risk of severe head-on collisions if the central median is sufficiently wide for the encroaching vehicle to be brought under control. In a study on cross median accidents on 22 roads (Hurd, 1956 [4]), 10 % of the accidents occurred where there was a 5 ft wide median, 5 % where there was an 18 ft median, 2 % where there was a 30 ft median and 1 % where medians were wider than 50 ft. On two roads with a 40 ft wide unpaved median sloped 3 ft to the centre (Hutchinson, 1962 [5]), observations revealed that vehicles travelled considerably long distances on the median. About 50 % travelled more than 300 ft and 10 % travelled more than 500 ft. There was some indication that a depressed median hinders a driver in regaining control of the vehicle.

A broad summary suggests that to reduce cross median accidents to an acceptable minimum, a median needs to be 50 ft to 100 ft wide, free of obstacles, fairly level and with a running surface that permits a driver to retain control of the vehicle. The saving in accidents needs to offset the cost of the land to achieve a positive cost advantage.

2.3 The case for median safety barriers on motorways

In the UK it is unusual for roads to have wide central reserves or medians. On most motorways, median width is about 4 m, although there is a growing need to encroach on the median area for road widening purposes, so reducing the width even further.

In an early analysis of non-median, median crossing and median-noncrossing accidents (Newby, 1962 [6] and Moore, 1964 [7]), it was shown that in median crossing accidents, about twice as many people were injured as in the other types of accident and about one third of the crossing accidents resulted in fatal or serious injuries.

The installation of a safety barrier is a cheaper option than a wide median; unfortunately there is a penalty to pay. The installation of barriers close to the carriageway increases the total number of accidents. The purpose of the safety barrier is primarily to restrain an errant vehicle from leaving its carriageway or traffic lane and entering an area more dangerous to the occupants. It follows that impact into the safety barrier needs to offer the possibility of a less severe injury to the occupants than that which might occur had the barrier not been installed.

Where traffic flow is light the risk of head-on collisions in median crossing accidents is less. To have an effect on the cross median accident rate it was considered that the average daily traffic flow should be in excess of 60 000 per day (Moskowitz and Schaeffer, 1960 [8]).

Consequently the potential benefits from the installation of safety barriers needs to derive from the consideration that the injuries arising from the inevitable increase in the number of accidents are of a low level compared to the severe or fatal accidents saved by the presence of the barrier.

One of the earliest studies in the UK to determine whether safety barriers are beneficial was carried out on the M1 motorway (Moore and Jehu, 1968 [9]). Two trial lengths of double sided blocked out w-section beam barrier [untensioned corrugated barrier (UCB)] of length 29 km were erected on the three lane, double carriageway. The rails were blocked out 20 cm on both sides of strong wooden posts, the top edge of the rail was 67.5 cm above road level.

Accident data were recorded during 1962-1963 before the barrier was installed. After installation, records were kept for the period 1965-1967. In the interval, traffic flow increased 12 %, to allow for this, accident data were recorded for an adjacent 61 km length of M1 motorway where no barrier was installed. Accident classes were divided into "on-median", "cross median no collision", "cross median with collision" "and median not entered".

The effect of the barrier practically eliminated all cross median accidents, but substantially increased other non injury accidents. Allowing for accident rate adjustments based on the 61 km control, and assigning monetary values (1966 prices) to the accident savings, the net effect of the cost/benefit calculation showed a saving of £460 per kilometre per year. However, there was a very large statistical standard error arising from the fact that there were few fatal accidents and their high tariff had a considerable influence on the analysis (Dawson, 1965 [10]). The barrier and maintenance costs were in excess of the savings and consequently, at that time, there was a reluctance to generally install safety barriers.

Based on the double sided tensioned corrugated beam (DSTCB) barrier, a later re-assessment of the economic benefits of safety barriers on motorway central reserves (Moore and Newby, 1969 [11]) produced a refined analysis by introducing selective costing according to the larger number of casualties per vehicle in cross median accidents, and to the greater frequency of fatalities. The revised calculations indicated that the annual cost of safety barrier per kilometre was about £457 (1969) and the annual net saving in accidents was £1 165 per kilometre. Statistically, there was a 70 % chance of the true value lying between minus £393 and plus £2 723; with a 2 to 1 chance that the overall benefit was positive. It was concluded it was probably beneficial to install safety barriers on motorways where the traffic flow was in excess of 35 000 vehicles per day. There is also an underlying feeling that a driver who is killed by a vehicle that crosses a central median is the “innocent” party. Accordingly, if there is an engineering possibility that such accidents can be prevented then, although it might not be a sound basis for making policy decisions, it is incumbent upon highway authorities to make such provisions to implement that safety engineering.

2.4 The case for verge safety barriers on motorways

A safety fence working party (Benson, 1975 [12]) examined the cost/benefit of installing safety barriers on the verges of safety fences. Any conclusions drawn were very sensitive to the assumptions made on the severity of the accidents. It was possible to estimate the cost of accidents per mile where vehicles had left the motorway on the nearside. However, there were no comparable data available showing the cost of accidents involving verge safety fences, making a meaningful comparison impossible. Assumptions needed to be made that the cost of accidents involving verge barriers would be less severe and consequently less costly than accidents where the vehicles left the road to the nearside where no verge barrier was installed. There was little evidence to suggest a reduction in severity was to be expected.

To be cost effective verge safety fences would have to bring a marked reduction in total accident costs. For example, at an installation cost of £15 000 per mile (1973), and a repair and maintenance cost of £500 per mile per year, accident savings of £1 800 per mile per year would be required to offset the costs. It was assumed 41 % of injury accidents involved only single vehicles, and that half of these would leave the road to the left.

Without knowing the efficacy of verge safety fences, there seemed to be no case for their use over the whole motorway network. Where there were high flows, reduction of accident costs of only 20 % could be sufficient to justify the installation of verge safety fences.

Taking into account the above restrictions, there is a case for the provision of verge safety fences only where there are special risks such as near bridge piers, lighting columns and sign poles.

2.5 The case for median safety barriers on all purpose roads

All purpose roads are those 60 mile/h to 70 mile/h dual carriageway roads not classed as motorways. The study (Sowerby, 1987 [13]) examined data from 350 km out of an existing total of almost 3 000 km of dual carriageway road. The sample included 142 km of roadway that had a safety barrier already installed on the central reserve.

The objective was to investigate accident data, road features and other parameters and, from the subsequent analysis, make prescribed recommendations for the installation of safety barriers at sites where they would be most cost effective. Accident data were categorized into fatal and serious, and then correlated by multiple regression with roadway variables which ranged over traffic flow, the presence of safety fence, median width, kerbs, cats-eyes, road lighting, metre strips and minimum radii of curvature.

For the unfenced sample of roadway, the accident frequency was 0.44 injury accidents per kilometre per year. Almost half of the total involved cross median accidents where vehicles entered the opposing carriageway.

The number of median accidents (accident frequency) varied strongly with traffic flow. The unfenced sample of 5.32 accidents per 100 million vehicle kilometres fell by 29 % to 3.76 accidents per 100 million vehicle kilometres. The reduction in median casualty rates was even more marked; the fenced sample at 6.72 injuries per 100 million vehicle kilometres was 35 % lower than the unfenced sample at 10.3 injuries per 100 million vehicle kilometres.

The regression analysis was found to be more successful in explaining the correlation of the chosen variables with accident frequency rather than casualty rate. Accident frequency positively correlated with average annual daily total (AADT).

The safety barrier term was statistically significant in both the "median" and "all" accident models. Geometric variables had little discrimination between accident frequency on the fenced and unfenced sites. A somewhat unexpected result showed that roads with kerbs had higher accident probabilities. Roads with offside metre strips were found to be safer.

On average a safety fence installation on a new road needs to recover a cost of £31 000/km (1985) including maintenance. On roads where kerbs are in place this threshold is reached at traffic flow above 10 000 vehicles per day.

Dual carriageways without kerbs formed about 70 % of the unfenced sample; on these roads the accident frequencies were lower.

On new roads, safety barriers were cost effective where traffic flows were above a base level of 10 000 vehicles per day, given an average barrier cost.

On existing roads, where traffic delay costs are included, safety barriers are justified at all rates of traffic flow in the range from about 10 000 to 35 000 vehicles per day.

For the majority of dual carriageway roads without kerbs, safety barriers are always cost effective for traffic flow of 10 500 to 33 500 vehicles per day. The cost model was based on the "all" accident model, from which casualties by severity are desegregated. The benefit of installing safety fences is highly dependent on reductions in fatalities per accident in the fenced sample; care was taken by further regression analyses to verify the assumptions. They demonstrated that the model was robust.

A further analysis in the form of a "before and after" study, revealed that the overall accident rate fell by 35 % after the installation of a safety barrier; the casualty rate of combined fatal and serious accidents fell by 52 %.

2.6 The case for higher containment barriers on motorways

2.6.1 *Higher containment barriers on the central median*

Higher containment safety barriers (HCSBs) are those that, on impact, are able to safely redirect vehicles of mass greater than private cars. Starting in the early 1980s HCSBs made of steel and of concrete were developed and tested by TRL.

The availability of these systems was stated publicly in the Motorway Safety Package announced in April 1988, (Thorndike, 1988 [14]). Whereas the car safety barrier was designed to safely contain a 70 mile/h, 1.5 t vehicle impacting at 20°, HCSBs were effective against heavy commercial vehicles (HCVs) up to 38 t, travelling at 50 mile/h and impacting at 20°.

Crossover accidents are considered to be random events; otherwise, if accidents frequently occur at one particular location then it is conceivable that the cause is due to a local defect in the road design. Over the years 1984 to 1987, on average there were 77 central median crossover accidents per year reported on motorways in England. Of this total, 42 accidents involved penetration of the barrier, of which about one half involved vehicles of over 1.5 t. The average number of casualties per year was 17 fatalities, 42 serious injuries and 79 slight injuries.

The HCSB was not expected to be successful against every heavy vehicle impact as there is a possibility of impact at speeds higher than 50 mile/h by vehicles of higher mass; also secondary collisions with the barrier, in which rebound occurs after primary impact with another vehicle, can induce barrier impacts at high angles. However, it was estimated that of the 77 recorded accidents it was expected that in 70 of these, the vehicle could be contained.

A discounted cost/benefit analysis at 1988 prices showed that, over a 20 year period the increased costs of installing HCSBs were three times greater than the increased benefits. This analysis did not take account of the potential increase in casualties to car occupants arising from the added stiffness of the HCSBs.

On these figures the installation of HCSBs on motorway medians in England was not economically justifiable.

Subsequent to the analysis carried out in 1988, a need arose for widening the M25 motorway. To achieve room for a fourth traffic lane some encroachment was necessary on the existing 4 m wide central median. The resulting narrower median, as well as the inclusion of central street lighting columns, restricted the full performance of the previously installed car based safety barrier. In this particular case, it was considered advisable to install an intermediate high containment concrete barrier. It had been proved, by controlled tests, that such a barrier can give successful impact response for cars and middle range commercial vehicles of 16 t.

2.6.2 *Higher containment barriers on motorway verges*

The case for installation of HCSBs on motorway verges is even less than that for the installation of HCSBs on central medians. However, there may well be sensitive sites where their use in small lengths is a justifiable precaution. One such case occurred adjacent to the Corley service station on the M6 motorway where it is possible that an HCV, which left the carriageway on the nearside, could crash through into a public restaurant. An HCSB has been installed in this area. Regional Authorities are advised to deal with similar roadside situations and install HCSBs according to the prevailing merits.

3 Controlled vehicle impact testing for safety barrier development

3.1 General

Vehicle accidents frequently bring with them the trauma of human distress. In the understanding that comes from accident reconstruction, there is the underlying possibility of avoiding, or at least reducing, the severity of impact. The mechanisms of vehicle impact into fixed objects is complex. The variables that need to be taken into account include vehicle speed, mass, structural stiffness of the vehicle and the object struck, angle of contact, road surface condition and the influence that these variables have on the severity of injury to the vehicle occupants.

Reconstruction of vehicle accidents by mathematical models, computer simulation, and in a few cases, the use of practical scale model cars, can play a part in the development of safety barriers. Nevertheless, there needs to be confirmation of proposed barrier designs by full scale vehicle impact testing.

The cost of executing and analysing such tests is high and so the essential information on barrier performance has to be gleaned from a restricted number of tests. The difficulty of the task is compounded by the range of possible barrier accidents and the concomitant number of impact variables that need to be considered. For example, the characteristics of the population of vehicles in the UK, in terms of their mass, maximum speed and dimensions, cover a very wide spectrum. The mass of a small private car can be as little 0.65 t, whereas a six axle heavy commercial vehicle has a maximum gross mass of 38 t and may increase to 44 t. The maximum speed permitted on UK designated roads is 70 mile/h although many vehicles, including the smaller private cars, can easily exceed this limit. The angle at which a vehicle, at speed, strikes and is constrained by a barrier is closely linked to the kinetic energy that the barrier and vehicle need to absorb in the impact; high impact angles generate high impact energy. It follows that the higher the value of energy dissipated, the greater the elastic and inelastic deformation of vehicle and barrier. It is evident that barriers that can absorb high levels of impact energy, and then successfully return vehicles on to a safe path, are likely to attract both high first costs and high repair costs.

However, cost/benefit analysis of accidents (see clause **2**) has shown that the greatest saving derives from those barrier designs which were likely to be effective against private cars. Accordingly, the early consideration in the development of UK safety barriers was to set up impact test criteria that would meet the requirements for evaluation of car based safety barriers.

Subsequently, as the needs warranted, criteria for the testing of heavy commercial vehicles (HCVs) had to be taken into account.

3.2 Test criteria for safety barriers

3.2.1 *Impact angle and critical speed*

In a primary impact into a safety barrier, that is, where the vehicle has not rebounded from another object, the maximum angle at which the vehicle can turn to approach the line of the fence is limited by the adhesion, or coefficient of friction, μ_{tr}, acting between the tyres and the road surface (Jehu and Pearson, 1977 [15]). The critical speed, v_c, is the forward speed of the vehicle at the point at which adhesion is about to be lost and the vehicle will start to slide out of driver steering control. If the barrier is involved in a secondary impact, perhaps following contact with another vehicle, then this situation does not apply and the impact angle cannot easily be predicted.

The minimum radius of turn at the critical speed occurs at equilibrium of the inertial force due to the radius of turn of the vehicle, and the resistive side forces acting on the tyres, (see Figures 1 and 2) that is:

$$\frac{mv_c^2}{r} = mg\mu_{tr} \qquad (1)$$

where

m is the mass of the vehicle;

r is the radius of turn;

g is the gravitational constant.

From Figure 1 it can be shown that:

$$\text{Cos}\ \theta = 1 - \frac{r - w}{r} = 1 - \frac{w}{r} \qquad (2)$$

where

w is the lateral distance of the car from the barrier at the moment it starts turn.

Combining equations (1) and (2) gives:

$$\text{Cos}\ \theta = 1 - \frac{gw\mu_{tr}}{v_c^2} \qquad (3)$$

It can be seen that equation (3) is independent of the vehicle's mass and so it applies equally to private and heavy commercial vehicles.

The theoretical curves for a 2-lane and 3-lane carriageway, derived from equation (3), are shown on Figure 3; the value of the coefficient of friction between tyre and road is taken to be 0.7. Also included in Figure 3 are data gathered from a road side investigation of accidents on motorways within a 130 km radius of the Transport Research Laboratory (TRL) at Crowthorne, Berkshire (Macdonald, 1986 [16]). The 3-lane curve shows that for approach speeds of 113 km/h (70 mile/h) and 80 km/h (50 mile/h), the limiting angles are 20° and 30° respectively. Out of a total of 25 accidents investigated, only three exceeded the predicted maximum impact angle; burst tyres or secondary impacts could account for these.

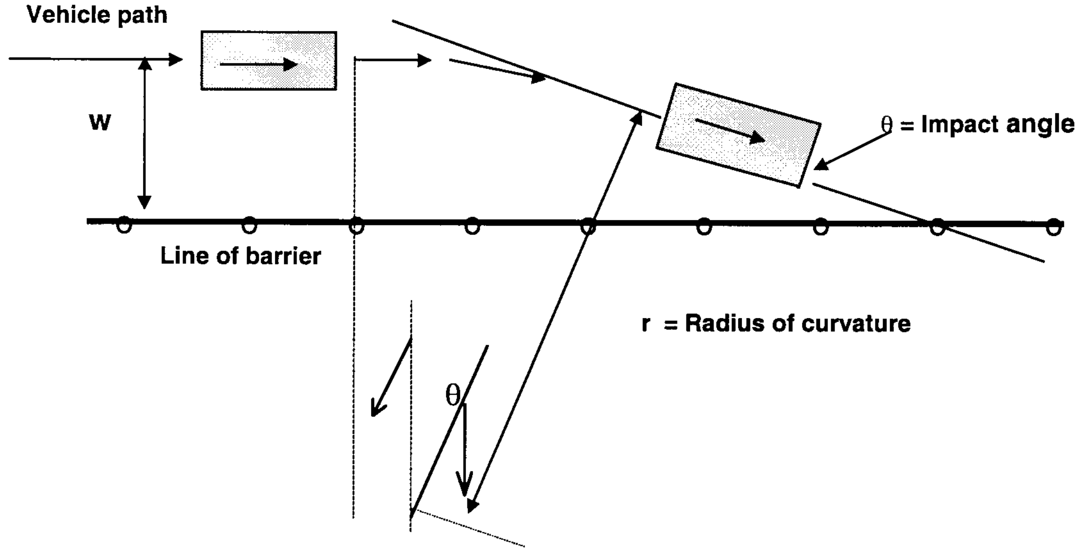


Figure 1 — Critical speed and barrier impact angle

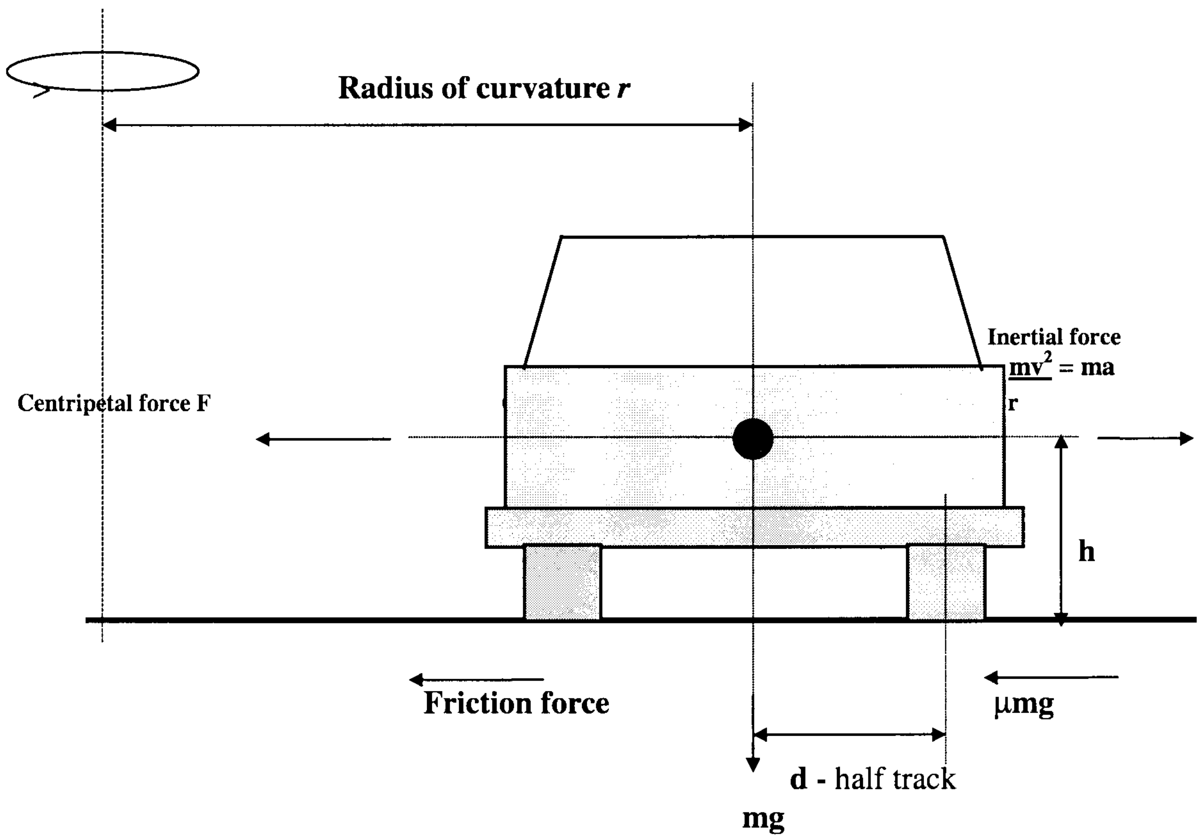


Figure 2 — Vehicle overturn at critical speed

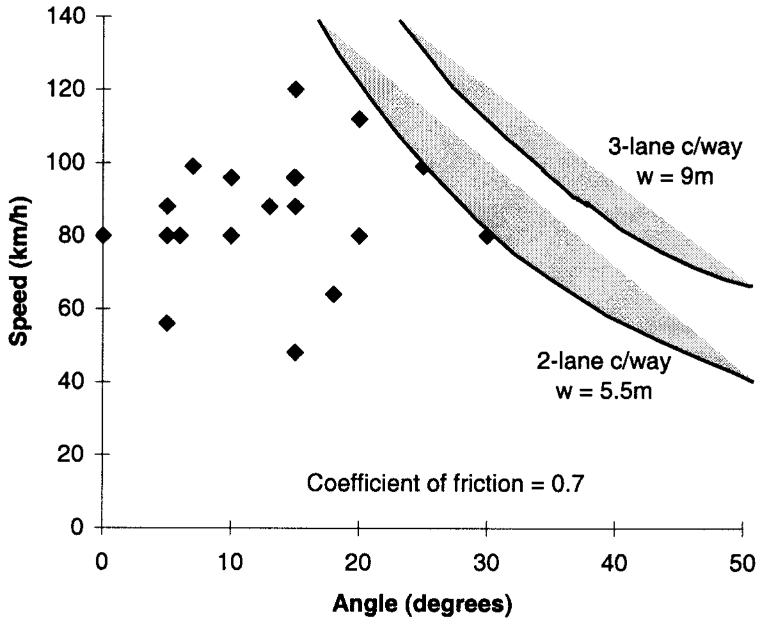


Figure 3 — Actual and theoretical impact speed versus angle

3.2.2 ***Vehicle stability — Sliding and overturning***

A vehicle travelling at critical speed on a curve is at the point of losing adhesion between tyre and road. If adhesion is lost, simple theory of circular motion predicts that the vehicle will slide in a straight line on a tangent to the circular path; it will not slide radially outwards. The angle that the straight line makes with a longitudinally installed safety barrier is the angle of impact (see Figure 1).

It has been shown in **3.2.1** that a vehicle turning across a 3-lane road, 9 m wide, at the critical speed of 70 mile/h can hit the barrier at a limiting angle of 20° and when adhesion between tyres and road surface is lost, it initially tends to slide in a straight line. However, if the coefficient of friction is high, and the vehicle has a high centre of gravity (CG) compared with the width of the wheel track, there is a limiting condition wherein the vehicle will not slide, but can overturn.

It is of interest to examine whether a 70 mile/h vehicle at critical turning speed and at the limiting impact angle is in the pseudo stable condition of sliding, or is on the point of overturning.

Restating equation (1):

$$\frac{mv_c^2}{r} = ma_{rad} = mg\mu_{tr} \tag{4}$$

where

$a_{rad} = v^2/r$ = radial acceleration.

From Figure 2, the vehicle is on the threshold of overturning when:

$$ma_{rad}h = mgd \tag{5}$$

where

h is the height of CG;

a_{rad} is the radial acceleration;

d is the half width of wheel track.

Also from Figure 2:

$$a_{rad} = \mu_{tr}g \tag{6}$$

Combining equations (5) and (6) gives the threshold condition for overturn as:

$$\mu_{tr} \geq \frac{d}{h} \tag{7}$$

That is, when the coefficient of friction between tyre and road surface is greater than the ratio of half the width of the wheel track to the height of the CG, then the vehicle will start to overturn.

A typical value for half the wheel track width of a 1 500 kg car is 0.75 m. If the vehicle is negotiating a curve at the critical speed of 70 mile/h on a bend which has a tyre to road surface coefficient of friction of 0.7 then, from equation (7), the vehicle is on the point of overturning if the height of the CG exceeds 1.07 m. The CG height of such a vehicle when unladen is typically 0.53 m, consequently it is unlikely to overturn unless the contents of the vehicle doubles the height of the CG.

The speed at which it would overturn can be derived from equations (1), (4) and (5). For the conditions stated, equation (1) gives a radius of curvature of 143 m. equations (4) and (5) indicate the vehicle is on the point of overturning at a speed of just under 100 mile/h.

Elementary analysis has shown that a typical 1 500 kg vehicle, that has a half wheel track width of 0.75 m and a CG height of 0.53 m, when turning across a 9 m wide motorway at the critical turning speed of 70 mile/h, will enter a pseudo stable sliding condition and is not likely to overturn; the critical overturning speed of 100 mile/h is very much higher than the sliding speed of 70 mile/h; clearly, for this vehicle, the overturning speed cannot be achieved.

3.3 Standard controlled barrier impact test conditions

3.3.1 *Private cars*

The maximum speed for private cars on UK motorways is 70 mile/h. Examination of the national fleet revealed that the overwhelming majority weighed less than 1 500 kg. The exceptions were a small percentage of large luxury cars.

For a known coefficient of friction between tyre and road surface, a simple analysis of the maximum angle that a vehicle, at steady speed, can turn across a carriageway and impact a longitudinal safety barrier has been presented in **3.2**. On a 9 m wide motorway carriageway a 70 mile/h vehicle is likely to impact the barrier at a maximum angle of about 20°, if the coefficient of friction is 0.7.

For private cars, this standard impact test criterion has been adopted in BS 6579 and BS 6779. That is, private car based barriers have to be successfully tested by collision with a 70 mile/h, 1 500 kg car impact, at an angle of 20°.

Although not included in BS 6579 and BS 6779, an impact test at the same speed and angle by a lightweight car (700 kg) was considered as an advisable additional test. Its purpose was to investigate the post-impact stability of small cars; they were considered likely to be more vulnerable to instability than the heavier car.

Cost/benefit analysis had shown that car based safety barriers were likely to be the most beneficial and so, from the mid 1960s, development of roadside barriers was concentrated in this area using the standard test impact conditions, supplemented by an impact with a small car.

Several successful designs were produced by DTp (TRL) and industry using these test criteria. The installation of safety barriers on UK roads began to take place in quantity in the early 1970s.

3.3.2 *Heavy commercial vehicles*

The economic case for heavy commercial vehicle (HCV) safety barriers could not be so well made as that for private cars. However, there had always been the need to protect sensitive areas from incursion by vehicles of all kinds; the most prominent case being the need for parapets on bridges. Early bridge parapet designs provided a starting point for the development of higher containment safety fences. Clearly, the potentially higher impact energy of HCVs compared with that of private cars, made the containment problem significantly more difficult. However, impact tests on concrete bridge parapets, which later became known as normal containment parapets, had shown they easily contained the standard private car impact. In addition, the theoretical critical speed and impact angle into a safety barrier applies equally to the heaviest commercial vehicle as it does to lightweight cars (Figure 3).

In setting out an impact test matrix for a range of bridge parapet designs, for those parapets designated as having low containment characteristics, the impact angle was retained at 20° and the vehicle mass remained at 1.5 t, but the impact speed was lowered to 50 mile/h (BS 6779). For normal containment, the impact test speed, angle and vehicle mass were retained to be the same as for safety barriers, (the standard impact). But for high containment bridge parapets, although the 20° angle was retained, the vehicle mass was increased to a value of 30 t and the impact speed reduced to 40 mile/h. The balance of vehicle mass and speed was an engineering compromise to accommodate one of the most impact aggressive HCVs on UK roads, the rigid 4-axle 30 t tanker, and yet retain a realistic impact test speed and energy level that would not result in damage to the bridge deck on which the parapets were mounted.

The stock of UK bridges covers a range of designs; it was important to develop parapets of such strength, that impact into them would not cause more damage to the bridge than to the parapet. The repair of the parapet was likely to be much less difficult and costly than repairing the bridge deck.

Consequently, the impact test criteria for bridge parapets had to be a balance between the parapet stiffness and the bridge deck strength. With initial guidance from bridge parapet design, it was clear at the early stages of development of roadside higher containment safety barriers, that a compromise had to be reached between the energy level of impact against which the barriers would be effective and the complexity of the barrier design and cost. The compromise was reached by taking a unity value for the energy level of the standard impact of 1.5 t, 20° at 70 mile/h (1.5: 20: 70), due to the velocity normal to the barrier, and comparing this to a similarly derived energy level of the HCV test for high containment parapets, (30: 20: 40). The HCV parapet test has an energy level of about 6.5 times the standard impact.

For comparative purposes, an HCV on a motorway at its maximum permitted speed of 60 mile/h with gross mass of 38 t, has an energy level of about 18.5 times the standard impact; although, on impact, the articulated tractor of a 38 t HCV is more easily redirected, with the result that the impact tends to be less severe than impact by a 4-axle rigid, yet lighter, 30 t vehicle.

Taking account of environmental considerations, cost based engineering solutions and the possibility of high energy impacts, it seemed reasonable to place the impact conditions for higher containment roadside safety barriers at about midway between the energy levels of the standard impact and the energy level adopted for the 30 t HCV impact into high containment parapets. Accordingly, the impact energy for higher containment safety barriers was set at about three times that of the standard level.

The use of energy level, as a means of defining impact conditions, is useful in the sense that it provides a single value description of the collision, but it should be borne in mind that energy is a scalar rather than a vector quantity. A practical interpretation of this is to consider two impact conditions at the same energy level, derived from velocities normal to the barrier; one at an impact angle of for example 10° and the other at 90°. In the first case, because of the shallow angle of impact, the energy dissipated will be spread over a much longer length of barrier compared to the 90° impact. Consequently the disparate concentration of energy over the different lengths of barrier in the two examples means, for successful vehicle containment, the two barriers would need to be of very different design.

Based on this heuristic approach, the test conditions for the development of higher containment safety barriers were set at 16 t, 15°, at a vehicle impact speed of 50 mile/h. The selection of the 16 t flat bed HCV represented the most common heavy commercial vehicle in the UK fleet (although it can take many designs of chassis and body dimensions). After an exploratory period of development, it became evident that the impact angle could be increased to 20° to conform to the impact angle in other safety barrier and bridge parapet test regimes.

The test matrix of vehicle speeds, masses and angles for safety barriers and bridge parapets does not claim to represent every possibility of impact that can occur on the road; such a test matrix would make a test program prohibitively expensive. However, the matrix is intended to be sufficiently stringent to separate the performance of good barrier designs from those which could create unnecessary hazards.

3.3.3 *Impact performance acceptance criteria*

The requirements in the UK, for acceptable vehicle and barrier performance under controlled impact test conditions, are given in the British Standards BS 6579 (for safety barriers) and BS 6779 (for bridge parapets).

The standards deal in detail with the integrity of vehicle and parapet, the pre-impact and post-impact trajectories of the vehicle, and the impact severity in terms of theoretical head impact velocity (THIV) and post-impact head velocity (PHD). In brief, the standards require that during the impact period the test vehicle shall be safely contained and redirected on a path close to the line of the barrier. There are exceptions to this general statement and the appropriate standard for either safety barriers (BS 6579) or bridge parapets (BS 6779) should be referenced for specific vehicle inertial masses, barrier types and their impact test requirements.

3.4 The European Standard for road side restraint systems

In 1990, under a Directive from the European Union (EU), the European Committee for Standardization (CEN) was instructed to begin work on the harmonization of the impact performance of roadside restraint systems, including vehicle safety barriers, that were prevalent in Member Countries. Under the Directive, CEN Technical Committee TC 226/WG1, Road restraint systems, formulated, as part of its mandate, a matrix of vehicle characteristics and impact conditions for the performance testing of vehicle safety barriers (EN 1317 series: EN 1317-1, EN 1317-2, prEN 1317-3).

Tables 1 and 2 are extracts from the European Standards EN 1317-1 and EN 1317-2. The tables give the range of vehicle characteristics, speed values and impact angles that are currently used in the performance testing of road restraint systems; both safety barriers and bridge parapet impact requirements are included in the tables.

4 Test vehicle propulsion and steering control

4.1 General

It was clear at the beginning of work on the development of road restraint systems that the use of manned test vehicles was prohibitive except for the most predictable of barrier tests. Impacts into rigid utility poles had shown that at low speed, about 20 mile/h, vehicle decelerations were sufficiently high to generate forces likely to cause occupant injury, (Moore and Christie, 1960 [17]). Even so, a few manned vehicle tests were made on shaped kerbs and barriers, but this technique was soon abandoned in favour of remote control testing.

4.2 Vehicle propulsion

Development of safety barriers in the UK began in about 1962, that is before the standard impact test condition had been formulated. The choice of vehicle propulsion system, at that time, was not therefore restricted to methods that could reach a speed of 70 mile/h. The intention was to achieve a “high speed impact”, where the term “high speed” inferred speed values considerably in excess of the current experience, such as 30 mile/h in vehicle crash-block testing, or 20 mile/h to 25 mile/h achieved at that time, in lighting column testing.

Table 1 — Test vehicle characteristics

Inertial mass								
kg								
Vehicle mass[a]	825 ± 40	1 300 ± 65	1 500 ± 75	10 000 ± 300	13 000 ± 400	16 000 ± 500	30 000 ± 900	38 000 ± 1 100
Including maximum ballast[b]	100	160	180	—	—	—	—	—
Dummy	75	—	—	—	—	—	—	—
Total vehicle static mass	900 ± 40	1 300 ± 65	1 500 ± 75	10 000 ± 300	13 000 ± 400	16 000 ± 500	30 000 ± 900	38 000 ± 1 100
Dimensions								
m								
(tolerance ±15 %)								
Wheel track (front and rear)	1.35	1.40	1.50	2.00	2.00	2.00	2.00	2.00
Wheel radius (unloaded)	—	—	—	0.46	0.52	0.52	0.55	0.55
Wheel base (between extreme axles)	—	—	—	4.60	6.50	5.90	6.70	11.25
Number of axles[c]	1S + 1	1S + 1	1S + 1	1S + 1	1S + 1	1S + 1/2	2S + 2	1S + 3/4
Ground clearance of front bumper measured at corner	—	—	—	0.58	—	0.58	0.58	0.58
Centre of gravity[d]								
m								
Distance from axle ±10 %	1.00	1.10	1.24	2.70	3.80	3.10	4.14	6.00
Lateral distance from centre line.	±0.07	±0.07	±0.08	±0.10	±0.10	±0.10	±0.10	±0.10
Height above ground. (inertial test mass ±10 %)	0.50	0.53	0.53	—	—	—	—	—
Load (+15 % to −5 %)	—	—	—	1.50	1.40	1.60	1.90	1.90
Type of vehicle	Car	Car	Car	Rigid	Rigid	Bus	Rigid	Articulated HGV

[a] Includes load for heavy goods vehicle (HGV).
[b] Includes test equipment.
[c] S: steering (or steered) axle.
[d] Inertial test mass

Table 2 — Vehicle impact test criteria

Test level	**Impact speed** km/h	**Impact angle** degrees	**Total vehicle mass** kg	**Type of vehicle**
TB 11	100	20	900	Car
TB 21	80	8	1 300	Car
TB 22	80	15	1 300	Car
TB 31	80	20	1 500	Car
TB 32	110	20	1 500	Car
TB 41	70	8	10 000	Rigid
TB 42	70	15	10 000	Rigid
TB 51	70	20	13 000	Bus
TB 61	80	20	16 000	Rigid
TB 71	65	20	30 000	Rigid
TB 81	65	20	38 000	Articulated

Energy for vehicle propulsion is a choice between the motive power of the engine, or the use of external power derived from motorized winch or towing systems. Running vehicles down short gradients is of limited use. A ramp height of 10 m would produce a terminal speed of less than 30 mile/h. The use of new vehicles for the purposes of crash testing is expensive. The used car market is usually the source for barrier test vehicles. Vehicles should be in road worthy condition, although vehicles of that period, whether new or second hand, could not be relied upon to reach an impact speed of 70 mile/h under their own power within the restricted distances available on crash test sites.

For the earliest higher speed barrier tests (1962), it was considered that acceptable and informative results could be obtained on barrier performance if the impact speed was in excess of 45 mile/h, for a moderate sized four door saloon of about 1.36 t. In preliminary tests of a concrete beam barrier, the test vehicles were manned and driven into the barrier at angles up to 20° at speeds of about 30 mile/h (V.J. Jehu, 1967 [18]). Using remote steering methods, higher impact speeds were possible by pulling the test car, attached to a long steel cable, with a high powered tow vehicle; the engine of the test car was not engaged. The cable passed through a pulley mounted on the back of the tow car, to a ground anchor. This mechanism provided a 2 to 1 speed ratio; the tow car had only to reach about 23 mile/h for the test car to reach impact speed at about 45 mile/h. Speeds in excess of about 50 mile/h were found to be difficult to achieve because of limitations in the remote steering system. Steering control was dependent on a balanced yoke connection between the tow cable and the car's track rod. It was important to keep a steady tension in the tow cable, otherwise the car could run off course; the method was found to be not suitable for speeds in excess of 45 mile/h to 50 mile/h.

It had been shown in earlier trials that the method of accelerating an unpowered test vehicle was feasible using two tow vehicles with pulleys that give a 4 to 1 speed advantage, (Laker and Blamey, 1961 [19]). Using this technique, a lightweight, unpowered, 1936, Morris 8 car had been accelerated to 40 mile/h in a distance of 50 m; the acceleration is equivalent to more than 3 m/s^2; this is considerably in excess of the maximum acceleration of most present day high performance private cars. The vehicle was steered by a guidance shoe linked between the car's steering track rod and a ground rail; target impact precision was of the order of $\pm$50 mm.

Subsequently, on a 190 m long test site at TRL, using the two towing vehicles method and remote radio control of the car's steering mechanism, impact test speeds of 62 mile/h were achieved (Jehu and Laker, 1967 [20]). This tow method of vehicle propulsion and radio control of an unpowered vehicle was found to be successful, and used, over the period of time during which current designs of safety barrier, presently in use on UK highways, were initially developed. The benefit of not requiring engine power from the test car simplified the purchase of second hand test vehicles.

Later, improved crash site facilities became available at TRL. Barrier approach distances of over half a mile were possible. After being push-started by a high performance, 350 horse power car, impact speeds of more than 70 mile/h could be reached by using the engine power of the more modern test vehicle. The vehicles were steered by a radio link; the controller operated a remote steering wheel and braking system from a following car (LF78, 1967 [21]).

As the need grew for the use of heavy commercial vehicles in the development of higher containment barriers, it was clear that the installed engine power was not sufficient for regulatory testing of prototype barriers; in addition, impacts within small speed tolerances were needed. Furthermore, the precision of radio controlled steering could not provide the tolerances necessary in targeting critical impact points. To overcome the problem, TRL and the Motor Industry Research Association (MIRA) combined to build a high energy impact test facility on the MIRA proving ground. The motive power was provided by a winch system driven by two 500 horse power induction motors. The design specification was to propel a 38 t truck up to a speed of 50 mile/h in a distance of 366 m; this is equivalent to an acceleration of about 0.7 m/s^2. Experience with the winch system demonstrated that accuracies of better than 0 to +2 mile/h could be achieved for speeds up to 100 mile/h, (Bacon, 1986 [22]).

Table 3 gives some examples of the MIRA/TRL power winch characteristics.

Table 3 — Examples of winch performance

Vehicle type	Mass t	Distance m	Speed mile/h
Car	1.5	110	70
HCV	16	200	50
HCV	38	305	50
Sled	2.5	180	100

4.3 Remote steering control

Remote steering by radio control and vehicle guidance by a ground rail linked to the test car have been referred to in **4.1**.

Vehicle control by radio has the benefit of flexibility; that is, the desired impact angle can be easily changed from test to test, and the technique lends itself to use between test sites. The disadvantages include the need for installation of fairly complex steering control equipment in the test car where it might suffer damage during the impact. In addition, steering by a controller operating a remote steering wheel from a following car may not provide sufficiently accurate contact with the target point on the barrier. The impact accuracy problem can be overcome by the replacement of manual remote steering control, with an induction loop buried in the test track approach to the barrier; but the risk of damage to complex equipment remains and the flexibility of moving from site to site is lost.

Test vehicle guidance by a shoe connection between a ground rail and the car has the benefit of high accuracy in pin pointing the desired impact point on the barrier. The cross-section of the guidance rail needs to be substantial if the shoe connects directly to the chassis of the test vehicle, rather than through the steering linkage; particularly if the rail is intended to guide heavy commercial vehicles. It has been estimated that a lateral force of at least 100 kN can be exerted on the ground rail during remote guidance of a 40 t HCV.

An alternative compromise between radio control and a heavy ground rail is the use of a guide wire strung between two moveable ground anchors, and connected to the steered stub axle of the test vehicle, (Bacon and Roy, 1988 [23]). This method has the benefit of flexibility wherein it can be easily moved to approach the barrier at chosen impact angles, or transferred to another test site as required. The link between the guide wire and the car is disconnected just before impact; the vehicle steering is then free from any external constraints and the guidance mechanism, having been separated from the vehicle, is not subject to damage during the impact. Computer simulation of the system demonstrated typical lateral force on the cable peaking at about ±2.5 kN, transitional loads ranged between 0.25 kN; the average lateral load was of course zero.

Using guide wire steering and propulsion from a 1 000 horse power induction motor winch, practical trials demonstrate the propulsion and guidance of a 16 t vehicle to a speed of 50 mile/h in a distance of 135 m.

4.4 Test car speed control

In early barrier testing, precise speed control of the test vehicle was not the main factor under consideration; barrier design and elimination of hazardous features was the prime concern. Limitations in test-site space, or engine performance, or towing power, usually meant that the maximum speed achievable was the only one of interest. Eventually, drafting of technical standards that included parametric tolerances, highlighted the need for improved control of vehicle impact speeds.

Speed control of a powered winch is simpler than controlling the speed of used car engines. Propulsion of the test vehicle by a towing winch also avoids the need to purchase used test vehicles with engines that are in good working order. Clearly there could be a cost saving. Calibration trial runs provided settings for the winch that permitted impact speeds to be met within selected tolerances.

The main factor, with regard to stopping the test vehicle after impact with the barrier is the space available on the test site. The extent of vehicle damage sustained can be of interest in assessing the overall performance of the barrier. Damage caused by secondary impact into protective structures obscures such useful information. If site space permits, braking of the vehicle by aggregate arrester beds, or catch netting, is likely to be most effective.

Where space is limited, remote control of the car's braking system is an alternative. Unfortunately, damage to test vehicles during contact with a barrier can include the car's brakes. Remote braking is usually not progressive, and hard application of the brakes by perhaps stored pneumatic pressure, to damaged or unbalanced brakes, can result in vehicle instability and overturn. There is always the need to ensure that all personnel are protected should test vehicles go out of control.

5 Measurement of vehicle and restraint system test data

5.1 General

A central purpose for carrying out full scale impact testing of vehicle prototype road restraint systems is, at the simplest level, to separate devices of good performance from those that are unacceptable; the impact test can be said to act as a discriminating filter which separates good systems from bad. Some restraint systems, such as a flexible roadside safety barrier, might have impact characteristics that are easily observable; for example, a test vehicle might be seen to be contained and redirected in a safe and satisfactory manner. However, the task of discriminating between good and bad performance becomes more complex when a restraint system has necessarily to be more resistive, and so generates higher deceleration forces in the vehicle. This can arise where roadside conditions require a more rigid system for vehicle containment within a limited space, or where high containment systems, suitable for commercial vehicles, might produce undesirable effects on small car response.

Recording vehicle impact characteristics allows objective assessment of a restraint system's impact performance to be made, compared with an observer's momentary subjective appreciation of an event that may take place in only a fraction of a second. In addition, a permanent record of the results of the impact test is needed to assist approval procedures for the acceptance of candidate systems.

The specific vehicle criteria for impact test acceptance have mainly centred on speed, acceleration and angular position. Instrumentation for recording at this level allows confirmation to be made on whether the regulatory pre-impact speed has been met, it gives a measure of the forces generated during the impact and it permits the calculation of the severity indices theoretical head impact velocity (THIV) and post-impact head deceleration (PHD), discussed in clause **7**. Approach speed is derived by timing the vehicle over a short distance (6 m), just before impact point. Cameras placed overhead and normal to the path of the test car provide checks on entry and exit speeds.

Anthropomorphic instrumented dummies are used to observe the effect of their weight on a small car's stability after impact. The recorded accelerations monitored from the dummy give an insight into the severity of impact, although indices derived from vehicle decelerations are simpler measures of severity.

Before the test takes place, vehicle and restraint system parameters need to be recorded. Vehicle data should include its mass, wheel track, wheel base and position of the centre of mass. The range of information logged, relating to the restraint system, is dependant on the particular device under test; nevertheless, it should include full engineering details, with reference to techniques for roadside installation, instrumentation for strain gauge records and prescriptions for soil strength or foundation requirement (see **5.5**).

Means for recording the deflection of the barrier and trajectory of the vehicle as it progresses through the test are essential. Cine and still photographs of vehicle and restraint system, taken before and after impact, can be of fundamental value, particularly in cases where a design modification is required. Overhead cameras suitably positioned are a main source of information. These can be reinforced by end-on cameras and displacement transducers strategically placed on the restraint system. Simple devices such as sand strips can provide valuable information on the vehicle's approach angle and trajectory during contact, and exit path.

The details for setting up restraint system approval tests in the United Kingdom are given in the British Standards BS 6579 and BS 6779 and in the European Standard EN 1317.

The information in **5.2** and **5.3** outlines typical instrumentation requirements for testing a roadside safety barrier.

5.2 Photography

Cine cameras are strategically placed to record the movement of the test vehicle and the deflection of the restraint system. Figure 4 shows a typical layout for coverage of a private car test on a safety barrier.

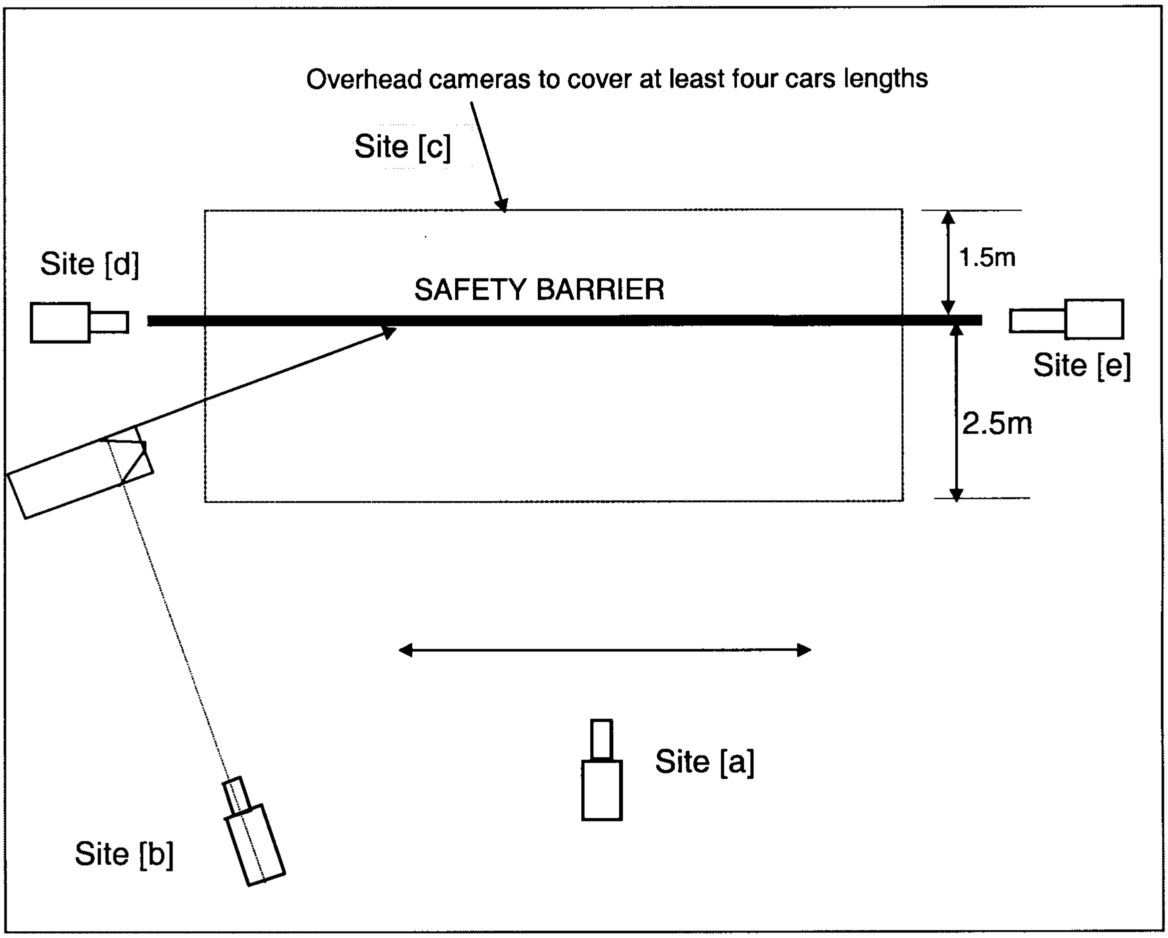


Figure 4 — Typical camera layout for a car test

Film speeds should run at not less than 200 frames per second (fps). A facility for recording strobe timing marks on the edge of the film at intervals of not greater than 0.01 s should be available. Comparable timing techniques should be employed if video tape recording is used.

The following camera schedule has been found to give satisfactory results in recording tests with private cars (see Figure 4). Commercial vehicles, because of their extra lengths, might require additional coverage to encompass the total movement of the vehicle.

Site [a]: One normal speed camera panned to follow the path of the vehicle.

Site [b] One fixed high speed camera placed at right angles to the vehicle's approach path.

Site [c]: Two or more overhead high speed cameras to record vehicle trajectory, including the exit path. These cameras, in conjunction with a grid of 1 m squares enables the lateral and longitudinal displacements to be recorded.

Site [d]: One high speed camera looking down the line of the barrier from a position behind impact point. Vehicle roll, barrier penetration and information on parallax corrections for the overhead cameras can be derived from this position.

Site [e]: One high speed camera looking along the barrier towards vehicle approach.

The cameras positions described for sites [a] to [e] are the minimum requirement for assessing barrier performance. Extra cameras should be deployed if there is a risk of not recording the essential features of the vehicle/barrier interaction.

Facilities to enable accurate analysis of the high speed film should be available.

Adequate still photographs should be taken of the barrier and car before and after the test.

The photographic information is required as part of an overall test report.

5.3 Vehicle instrumentation

The impact vehicle is considered to be a point mass at the origin of three orthogonal axes. As shown in Figure 5, the axes can rotate in yaw, roll and pitch.

Vehicle movement is recorded by accelerometers mounted at the centre of mass. Corrections to the accelerometer outputs can be made if there are difficulties in placing them sufficiently close to this point (NCHRP 350 [24]). It is important that spurious oscillations should not contaminate the deceleration signals; accelerometers should be mounted on a stiff block firmly attached to the structure.

Angular rotation of one or more planes in yaw, roll or pitch can be monitored by a rate gyroscope, angular accelerometers or two sets of linear accelerometers (EN 1317).

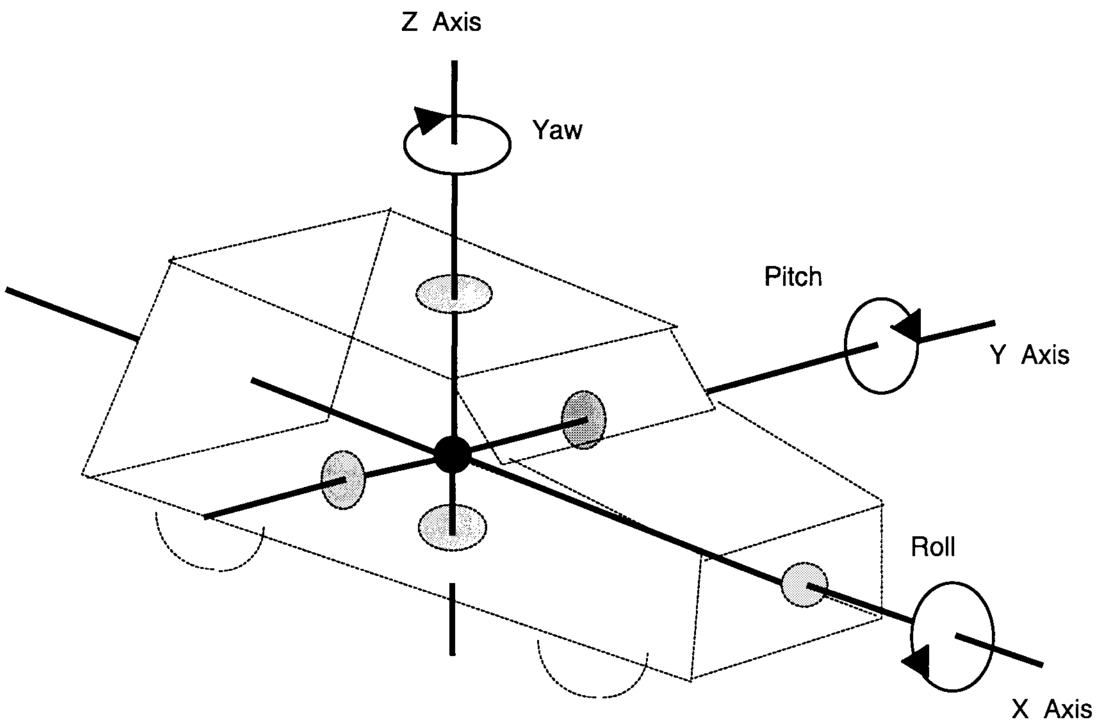


Figure 5 — Vehicle orthogonal axes

In the text, the term "deceleration" denotes a negative value of acceleration. Techniques for logging output signals from the transducers include on-board recorders or ground based recording stations linked to the vehicle by land lines, or by radio transmission. The present trend is to use analogue transducers with analogue to digital converters, (A/D), and digital loggers using magnetic tape or random access memory (RAM). Digital recorders cyclically log the data flow from the transducers, continuously overwriting the memory, until a predetermined signal level is detected; it then permanently records the incoming signals, and stops when the memory is full.

Sign conventions for installing vehicle instrumentation have been changed from time to time; Table 4 gives some of the more prominent sources.

The main purposes of installing vehicle instrumentation are to:

— monitor levels of impact deceleration for comparative purposes from test to test;

— evaluate dynamic loading on the restraint system;

— compute impact severity indices;

— track the path of the vehicle within terrestrial axes.

The minimum vehicle instrumentation for recording linear accelerations and angular velocities consists of three linear acceleration transducers, mutually orthogonal (tri-axial), and aligned with the vehicle axes in the longitudinal, transverse and vertical directions, plus one angular rate transducer. Alternatively, two sets of tri-axial accelerometers placed a known distance apart on or near the longitudinal axis are sufficient.

The acceleration and angular velocity transducers, and the relevant signal conditioning and recording equipment should conform to ISO 6487.

The selection of a data channel frequency class (CFC) is dependant upon the ultimate use of the data; in many cases the selected bandwidth can be a result of good engineering judgement. For example, a graphical presentation of vehicle crash data may contain such unwanted high frequencies from body panel vibrations that the underlying low frequency depiction, of the vehicle movement as a whole, may be concealed. For visual presentation of traces, a rule of thumb is to apply filtering, with decreasing low pass filters, to the point where the total area under the data trace is at the point of being affected. Practical examples, from high speed impacts with private cars, show underlying trends can be revealed by filtering down to a 10 Hz low pass filter.

The channel frequency class is designated by a number indicating that the channel frequency response lies within the limits specified in Figure 6.

It is important to note that valid comparisons of test results, using different frequency response classes, can be difficult to make. Specific frequency response classes should be selected when comparing results from different sources. The classes given in Table 5 are recommended examples; there may be other considerations that impose special requirements.

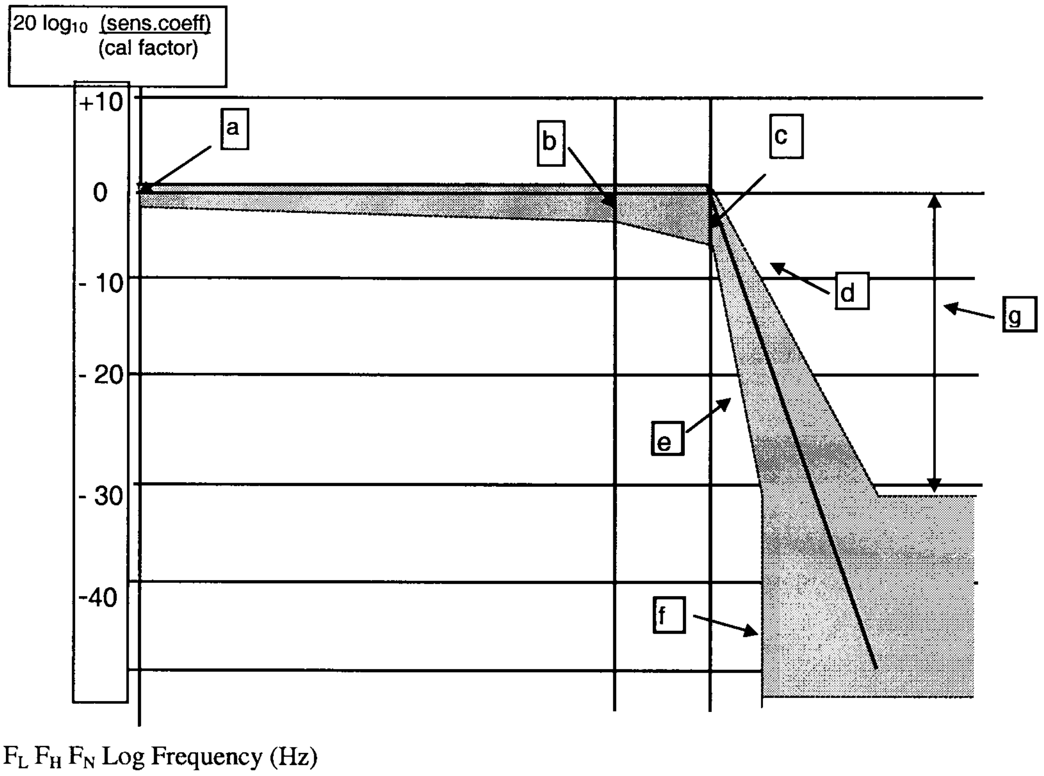
Table 5 — Typical CFC classes

Test measurement	CFC
Vehicle:	
Total vehicle	60
Components	600
Integration of velocity or displacement	180
Dummy occupant:	
Head acceleration	1 000
Chest acceleration	180
Chest deflection	180

The CFC number is numerically equal to the first break point on Figure 6, marked as F_H.

Table 4 — Sign conventions for vehicle instrumentation

Source	Long acceleration X Forward	Lateral acceleration Y to the left	Vertical acceleration Z upwards	Yaw To the right	Pitch Front down	Roll To the right
TRL	Positive	Negative	Negative	Positive	Negative	Positive
MIRA	Positive	Positive	Positive	Positive	—	Positive
ISO 6487	Positive	Negative	Negative	Positive	Negative	Positive
NCHRP 230 [25]	Positive	Positive	Positive	Negative	Positive	Positive
NCHRP 350 [24]	Positive	Negative	Negative	Positive	Negative	Positive
SAE J211 [26]	Positive	Negative	Negative	Positive	Negative	Positive



CFC	F_L	F_H	F_N
1 000	<0.1	1 000	1 650
600	<0.1	600	1 000
180	<0.1	180	300
60	<0.1	60	100

a ±0.5 db
b +0.5 to −1db
c +0.5 to −4 db
d −9 db/octave
e −24 db/octave
f ∞
g −30 db

Figure 6 — Frequency response limits

5.4 Measurement of the position of a vehicle's centre of mass

5.4.1 *Background and summary*

The trajectory of a vehicle involved in a collision, either with a fixed object or another vehicle, is influenced by the position of its centre of mass (CM). For example, the height of the vehicle's CM above the running surface affects its roll stability, and the longitudinal position affects the tracking or yawing characteristics of the vehicle. The CM position has an effect on pitch characteristics, but this is not usually prominent unless the suspension is damaged or the vehicle is overloaded. The method employed for measuring the position of the CM of test vehicles used in crash studies for the development of roadside safety features, needs to be accurate and repeatable if conformity of test results is to be maintained. To help achieve accuracy and conformity of measurement, a procedure is given in the following paragraphs for locating the position of the CM, in the vertical and longitudinal planes, of two-axle vehicles and multiaxle heavy commercial vehicles (HCVs).

In order to measure the CM height of two-axle vehicles by tilting and weighing the vehicle it is recommended that a linear plot is constructed of tan ϕ versus M_f' and M_r', where ϕ is the tilt angle and M_f', M_r', are the axle masses of the front and rear axles. To achieve uniform and accurate results the vehicle should be tilted through an angle range of ±12° centred on an operating point of 23° (see **5.4.4**). An example is given in **5.4.4.5**.

Full scale vehicle impact tests involving HCVs are usually made with the HCVs loaded or partially loaded. Although the tilt method for measurement of CM can be applied to some 2-axle HCVs, practical difficulties could result because of their heavy masses. To help overcome the difficulty, a method is given whereby the HCV masses and dimensions, which relate to chassis, transmission, engine, ballast and payload weights, are treated separately.

The longitudinal position of the CM of both two-axle and multi-axle vehicles requires measurement of axle weights and spacings. The CM of the whole vehicle is calculated by taking moments about a selected datum (see **5.4.3.1**).

The procedure for locating the CM height of multi-axle HCVs requires that the mass and CM position of the component parts of the vehicle are estimated. The CM of the whole vehicle is then calculated by taking moments about a selected datum; an example is given in **5.4.3.2**.

The vehicle tilting method has been documented in ISO 10392. The standard should be read in conjunction with **5.4.2**.

Other techniques, such as tilt tables and pendulum devices, are not discussed; the availability to test houses of compatible equipment is likely to be limited.

5.4.2 ***Determination of the CM longitudinal position (x) and height (z) of two-axle vehicles***

5.4.2.1 *General*

The category of two-axle vehicles includes private cars, car based vans, pickup trucks and HCV vehicles up to about 5 t unladen mass. Higher mass vehicles can be included in this category if weighing and tilting apparatus of appropriate accuracy is available. The required accuracies of measurement are given in ISO 10392.

5.4.2.2 *Measurement of CM longitudinal position (x) of two-axle vehicles*

Figure 7 shows a cross section of a two-axle vehicle of wheel base L, total mass M_v, front axle mass M_f and rear axle mass M_r. The CM position is at a longitudinal distance X from the front axle and a distance Z from the horizontal running surface.

Definitions:

$$L = 0.5 \times [L_{left} + L_{right}]$$

$$M_v = M_f + M_r; \quad M_f = M_1 + M_2; \quad M_r = M_3 + M_4.$$

where M_1 to M_4 are the measured wheel loads.

Taking moments about the front axle:

$$X = (M_r/M_v) \times L \qquad (8)$$

Taking moments about the rear axle:

$$X = L \times (1 - M_f/M_v) \qquad (9)$$

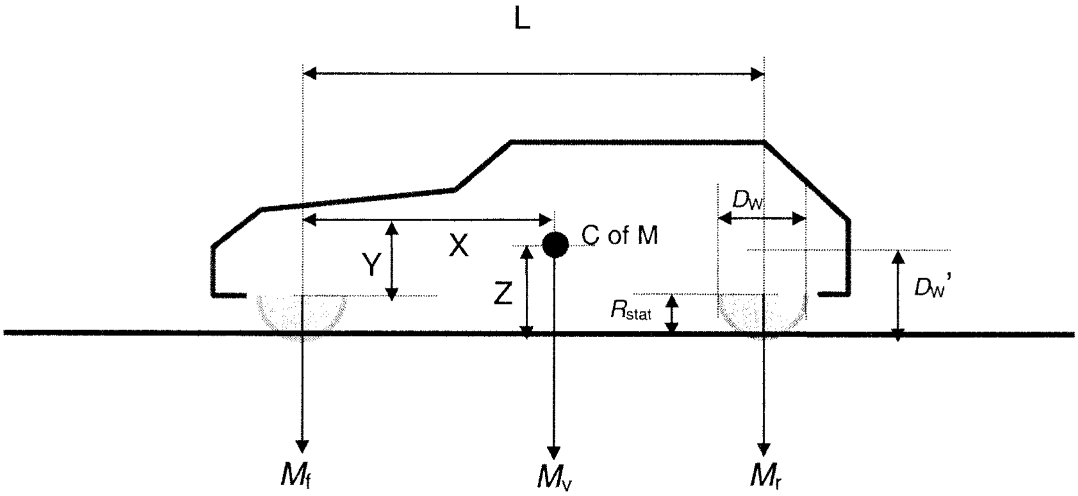
The mean value of equations (8) and (9) are taken as the measured distance (X) of the CM from the front axle.

5.4.2.3 *Lateral position of the CM*

Similar techniques to those discussed in **5.4.2.1** are used for locating the lateral position of the CM relative to a longitudinal vertical median plane through the vehicle. However, it is assumed that left and right hand wheel loads are equal, the CM therefore lies in the longitudinal centre line of the vehicle.

5.4.2.4 *Measurement of CM height (Z) of two-axle vehicles*

Figure 7 shows the height of the CM at a distance Z above the running surface. Figure 8 shows the vehicle tilted through an angle ϕ, about the rear axle. The measured loads, on the front and rear axles whilst tilted, are denoted by M_f' and M_r'.



$X = [M_r/M_v] \times L$
$X = L\,[1 - M_f/M_v]$

$Z = Y + R_{stat}$
$R_{stat} = D_w' - D_w/2$

$L = 0.5\,[L_{left} + L_{right}]$

$M_v = M_f + M_r$
$M_f = M_1 + M_2$
$M_r = M_3 + M_4$

Figure 7 — Distance of centre of mass from the front axle (X) and from the ground (Z)

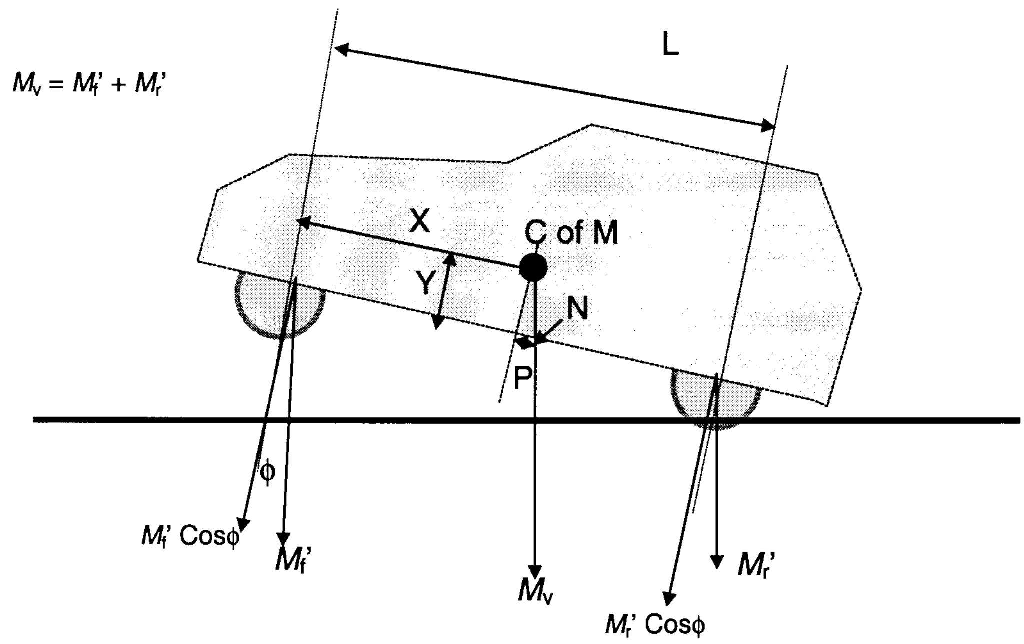


Figure 8 — Vehicle tilted to measure CM height (Y) above the axle height

From Figure 8:

$$P = Y \operatorname{Tan} \phi \qquad (10)$$

$$M_v = M_f' + M_r' \qquad (11)$$

$$Z = Y + R_{stat} \qquad (12)$$

$$R_{stat} = D_w' \; D_w/2 \qquad (13)$$

Taking moments about Point N in Figure 8 gives:

$$(X + P)\, M_f' \operatorname{Cos} \phi = (L - X - P)\, M_r' \operatorname{Cos} \phi \qquad (14)$$

Substituting equations (10) and (11) in equation (14) gives:

$$Y = L(M_r' - M_r)/M_v \operatorname{Tan} \phi \qquad (15)$$

Substituting equation (12) in equation (15) gives:

$$Z = \{L(M_r' - M_r)/M_v \operatorname{Tan} \phi\} + R_{stat,r} \qquad (16)$$

The equation for tilting about the front axle is given by:

$$Z = \{L(M_f' - M_r)/M_v \operatorname{Tan} \phi\} + R_{stat,f} \qquad (17)$$

where

Z is the height of the CM above the running surface.

The mean value of equations (16) and (17) should be taken as the height (Z) of the CM above the running surface.

5.4.2.5 *Linear equation for CM height (Y) above axle centres*

Measurements are plotted in the form of a linear equation.

Equation (15) is recast as follows:

M_r' (or M_f') = $YM_v \operatorname{Tan} \phi/L + M_r$; (or M_f about front wheel) (18)

Equation (18) is a linear equation of slope YM_v/L and constant M_r as shown in Figure 9.

Measurements of load M_r' for a range of tilt angle ϕ are taken and plotted in Figure 9. A best fit linear regression line is drawn through the plotted points and the value of Y determined. The measurement is repeated by tilting about the front axle, and the mean value of Y taken from the two results (see **5.4.4.2** and **5.4.4.5**).

Substitution in equation (12) gives the value of the height Z of the CM above the running surface.

The range of tilt angles ϕ, should be between $23° \pm 12°$ (see **5.4.4.4**).

5.4.2.6 *Preparation of a two-axle vehicle for centre of mass measurement*

During the measurement of wheel loads against tilt angles the following vehicle preparation and measurement procedures should be adopted.

a) The vehicle body and suspension, when on a level surface, are clamped to prevent relative movement during the tilting process.

b) Measurements are made whilst increasing and reducing the tilt angles; the mean values are calculated.

c) Front and rear wheel loads are measured in separate experiments; the mean of the derived values of Y is calculated.

d) The requirements of ISO 10392 should be met.

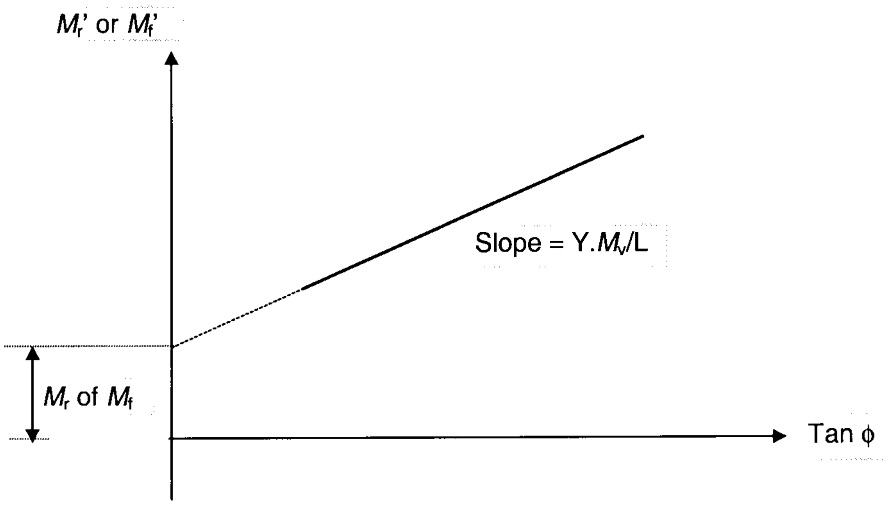


Figure 9 — Plot of M_r' against Tan ϕ

5.4.3 *Determination of the longitudinal position (X_{cm}) and height (Z_{cm}) of the centre of mass of a multi-axle HCV*

5.4.3.1 *Longitudinal CM position (X_{cm}) of multi-axle HCVs*

The longitudinal position of the CM of a multi-axle HCV is determined by measuring the axle weight and position of each axle from a datum line, and by taking moments about the datum (Figure 10). The CM position (X_{cm}) is then determined from the following equation.

where

X_{cm} = distance of cm from datum;
M_t = total mass of vehicle;
n = number of axles;
M = weight of each axle;
X = distance of each axle from the datum.

The measurements are repeated for datum points located at the front and the rear of the vehicle, and the mean value of X_{cm} is taken.

5.4.3.2 *Centre of mass height (Z_{cm}) of a multi-axle HCV*

The height of the CM above the running surface for a multi-axle HCV is determined by estimating the masses and CM heights of the major components of the vehicle, and by taking moments about the running surface (Figure 11). The height of the CM (Z_{cm}) shall be determined from the following equation.

$$Z_{cm} = (1/M_t) \sum_{n=n}^{n=1} M_n Z_n$$

where

Z_{cm} = height of CM above running surface;
M_t = total mass of vehicle;
n = number of individual vehicle masses;
M = weight of each mass;
Z = height of each mass above running surface.

$$Xcm = \sum_{n=n}^{n=1} Mn.\, Xn$$

n = number of axles

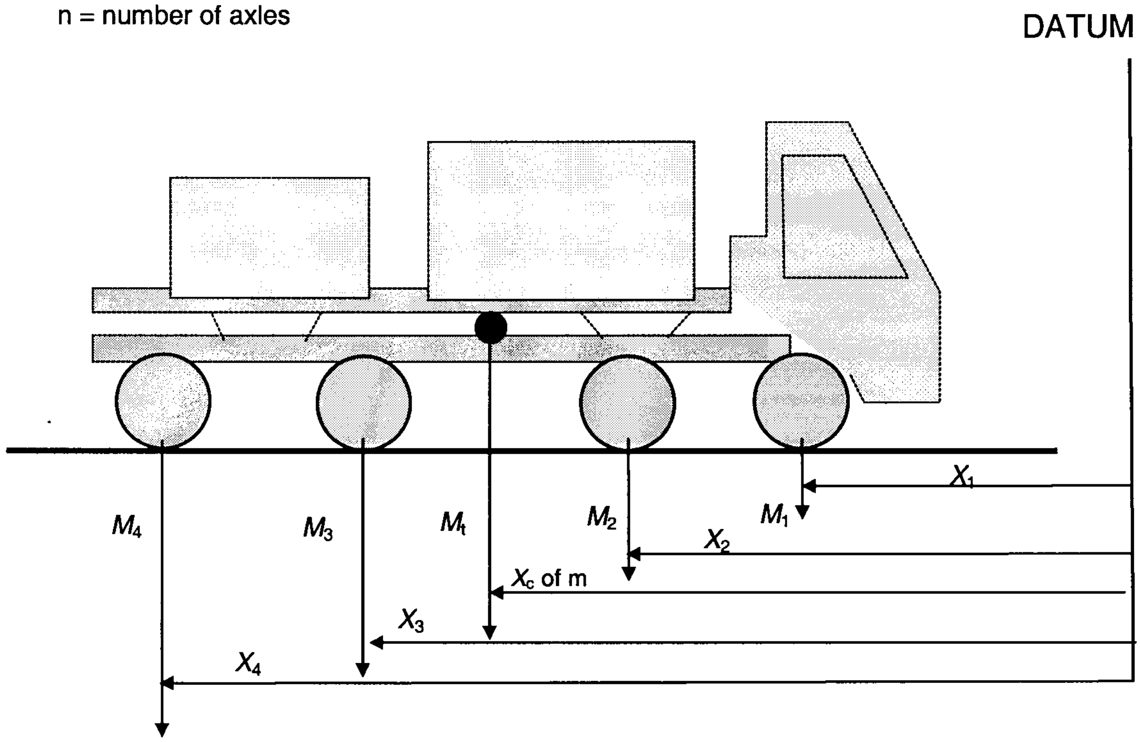


Figure 10 — Longitudinal position of centre of mass from datum

$$Zcm = (1/Mt)\sum_{n=n}^{n=1} Mn.Zn$$

n = numberof masses

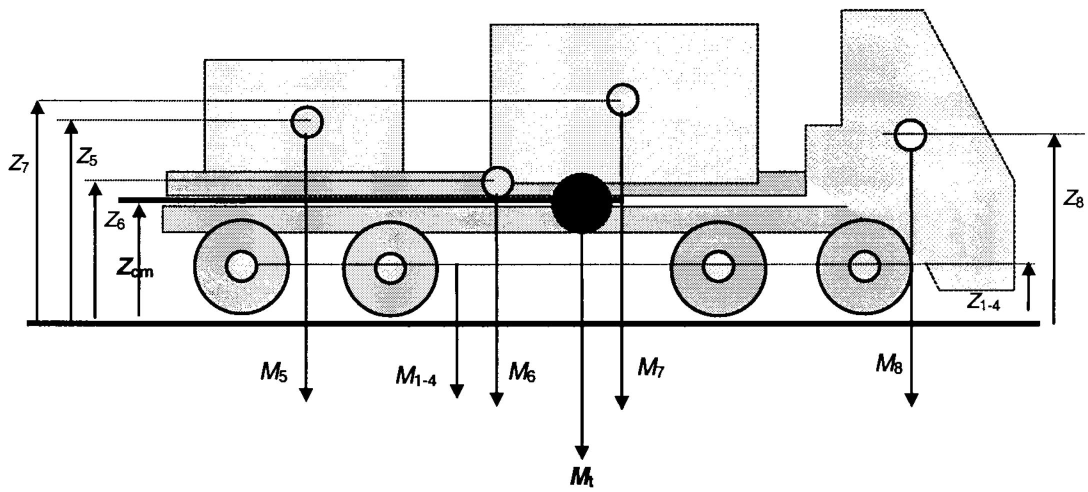


Figure 11 — Centre of mass height (Z_{cm}) of multi-axle HCV of mass M_{t}

5.4.3.3 *Example: To find Z_{cm} of a rigid HCV*

Table 6 shows the total mass of individual masses.

Table 6 — Total mass of individual masses

Component	Mass kg	Height m
Engine and gear box	1 500	0.95
Front axles (2)	680	0.46
Rear axles (2)	1 520	0.48
Cab	250	1.40
Chassis	4 180	0.79
Tank and load	23 961	2.01
Total mass	32 091	

Taking moments, with masses in kilograms and height in metres, gives the CM height (Z_{cm}) at 1.69 m.

5.4.4 ***Practical techniques for the measurement of tilt angle and wheel load***

5.4.4.1 *Accuracy*

The accuracy of measurement should follow the recommendations given in ISO 10392.

They are repeated in Table 7 for convenience.

Table 7 — Accuracy of determined parameters

Parameter		Accuracy
Absolute axle load		±0.2 %
Change in axle load due to tilting (this applies to weighing scales which do not measure absolute loads, but change in load.		±2.5 %
Dimensions:	< 2 000 mm	±1 mm
	> 2 000 mm	±0.05 %
Angles		±1.0 mm

5.4.4.2 *Measurement techniques for evaluating the height (Y) of the CM, above axle centres by tilting*

Equation (18) relates the measured rear wheel load M_r', to the tilt angle ϕ, as follows:

$$M_r' = Ym_v \times \text{Tan}\,\phi/L + M_r \tag{18}$$

where

Y = height of CM above axle centres;

M_v = static total weight of vehicle;

M_r = static load on real wheel.

A similar equation can be generated for the front wheel loads.

Equation (18) shows that the relationship between the wheel load M_r', and the tilt angle ϕ is proportional to the tangent of the angle. Consequently, the sensitivity of measurement changes according to the angles chosen at which to make measurements. This means a unit change in angle at a chosen tilt angle will produce a change in load which is proportionally different from the change in load due to a unit change in angle at another value of tilt angle.

The accuracy of measurement, as given above, does not take into account these non-linearities. The effect could be that one operator who chose to make measurements over a low range of tilt angles, could experience an overall resolution of measurement different from another who operated over a higher range of tilt angles. This could lead to non-conformity of measurement, they could be working to different accuracies of measurement.

5.4.4.3 *Sensitivity of measurement*

The sensitivity of measurement can be defined as:

the change in load (M_r') for a unit change in angle (ϕ)

which equates to:

$$\frac{d\,(M_r')}{d\varphi}$$

Differentiating equation (18) gives:

$$\frac{d\,(M_r')}{d\varphi} = \frac{YM_v\text{Sec}^2\,\varphi}{L} \tag{19}$$

Equation (19) is plotted in Figure 12. It shows that the sensitivity of measurement is reasonably constant over a range of angles from about 0° to about 35°.

The analysis indicates that to achieve conformity, tilt measurements should be restrained to fall within a range of angles from 0° to 35° from the horizontal.

The value of Y is determined from the slope of the graph of M_r' against Tan ϕ (see **5.4.2.4** and **5.4.3.3**).

5.4.4.4 *Sensitivity and its rate of change*

Unit rate of change of sensitivity can be defined as the condition under which a unit change in the angle ϕ produces unit change in the sensitivity of measurement. Zero or constant rate of change of sensitivity over the range of tilt angle would give the best opportunity for accuracy and conformity of measurement between operators. However Figure 12 is a non-linear curve; consequently constant sensitivity over the range of ϕ is not possible, and zero rate of change only occurs when the tilt angle $\phi = 0$; so a compromise needs to be reached and the following is proposed.

Differentiating equation (19) gives:

$$\frac{d^2\,(M_r')}{d\varphi^2} = \frac{\text{Sin}\,2\,\varphi}{\text{Cos}^4\,\varphi} \tag{20}$$

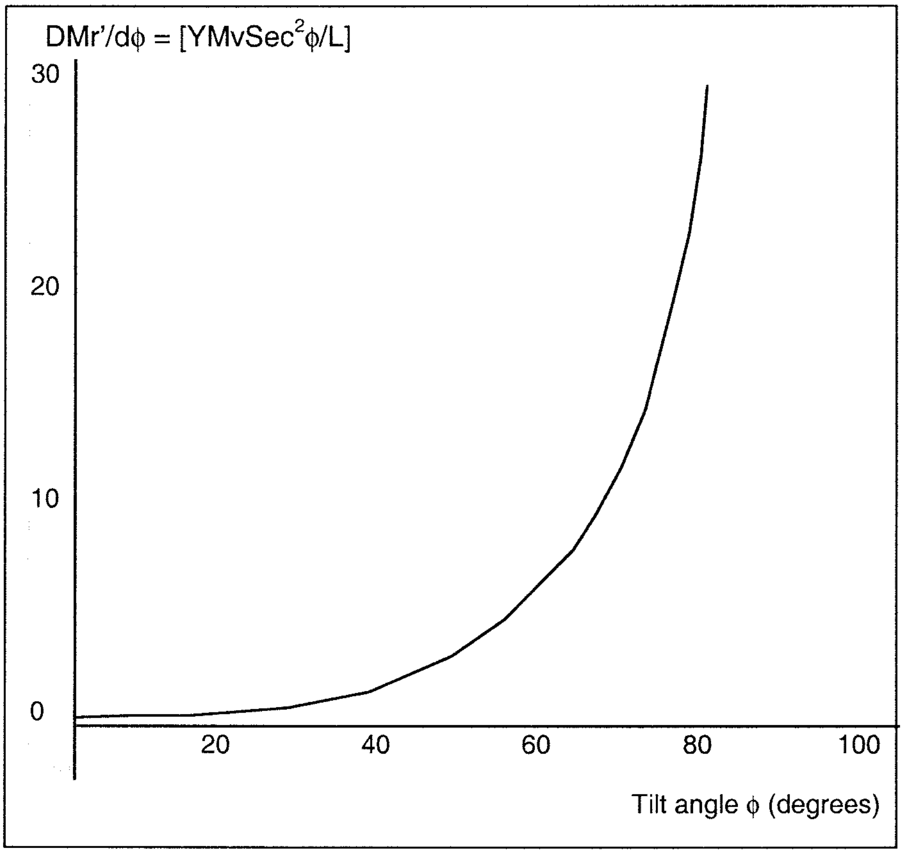


Figure 12 — Sensitivity analysis

Figure 9, which represents sensitivity of measurement, has unit slope when:

$$\frac{\text{Sin } 2\ \varphi}{\text{Cos}^4\ \varphi} = 1 \qquad (21)$$

The solution for ϕ in equation (21) gives:

$\phi = 23°$ (approximately)

This means, in the limit, that a unit change in angle ϕ, when $\phi = 23°$, produces a unit change in the sensitivity of measurement. It can be seen that the sensitivity of measurement (Figure 12) is reasonably linear over the range $23° \pm 12°$. Equation (20) shows that, due to non-linearities in Figure 12, unit rate of change in sensitivity cannot be maintained; it ranges from about 0.5 to 2.0 over this range of angle ϕ.

Consequently, for conformity of measurement, readings taken over a tilt angle range of about $\pm 12°$, centred at an angle of 23°, give a reasonable compromise. There can be difficulties with the body work of tilted vehicles making contact with the ground; suitable blocking under the wheels can often overcome this problem.

An example measurement for assessing the CM height of a two-axle vehicle is given in **5.4.4.5**.

5.4.4.5 *An example of the tilt/weighing method for the measurement of CM height*

Equation (18) relates in the form of a linear equation, the vehicle tilt angle ϕ to the increase in wheel load (M_r' or M_f'). The slope of the line is YM_v/L, where Y is the height of the CM above axle centres, M_v is the total vehicle mass and L is the wheel base. The complete equation is:

$$M_r' \text{ (or } M_f') = YM_v \text{ Tan } \phi/L + M_r \text{ (or } M_f) \qquad (18)$$

The height (Z) of the CM above ground is:

$$Z = Y + R_{stat} \qquad (12)$$

where R_{stat} is the load deflected radius of the tyre [see equation (13)].

Figure 13 shows a plot of increase in wheel loading due to tilting a vehicle through angle ϕ. The tilt angles cover a range of $\pm 12°$ centred about an operating point at 23° (see **5.4.4**).

Weighings are made whilst lifting and lowering the vehicle on the front and on the rear wheels. The mean values are taken of the lifting and lowering measurements. Figure 13 shows the data plotted for front and rear load measurements; a linear regression line has been drawn through each set of points.

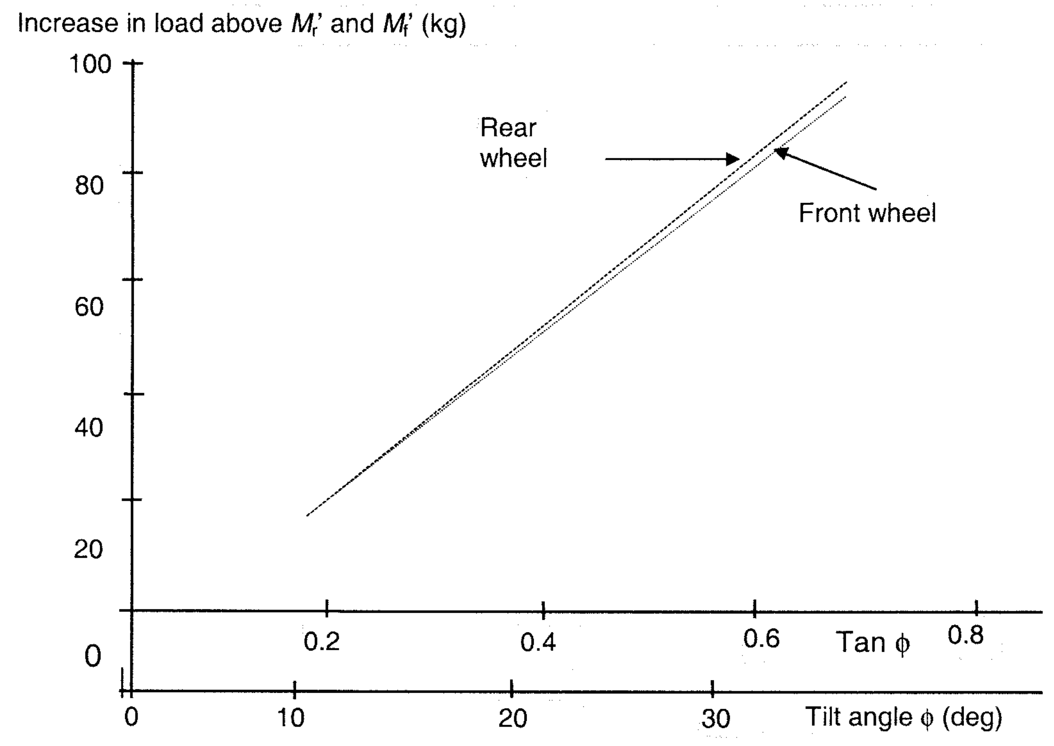


Figure 13 — Linear plot of M_r' and M_f'

The following calculations, derived from Figure 13, are based on an early Ford Cortina Estate car:

M_r = 519 kg; M_f = 522 kg; M_v = 1 041 kg; R_{stat} = 0.28 m; L = 2.59 m.

Regression slope for front wheels = $Y_f\ M_v/L$ = 134.9, therefore:

$Y_f = 2.59 \times 134.9/1\,041 = 0.34$ m.

A similar analysis for the rear wheels gives:

$Y_r = 2.59 \times 131.7/1\,041 = 0.33$ m.

Hence, the height of CM above ground is:

$Z = 0.5(0.34 + 0.33) + 0.28 = 0.61$ m.

5.5 Vehicle restraint system test report

5.5.1 *General*

The headings relating to the type of information for inclusion in a restraint system test report are given in EN 1317-1. Some are repeated here for convenience, with some amendments.

5.5.2 *Test laboratory*

Name
Address
Test site location
Telephone number

5.5.3 *Client*

Name
Address
Telephone number

5.5.4 *Test item*

Date received
Date tested
Name of test item
Design specification
Drawing details

5.5.5 *Test specification*

CEN test category
Target impact speed
Target impact angle
Target inertial vehicle test mass

5.5.6 *Installation*

Detailed description of item for test
Test site drawing including ground work
Material specification
Soil specification
Design specification for on-road installation
Photographs

5.5.7 *Vehicle*

Model
Model year
Identification number
Total vehicle mass
Test inertial mass
Ballast, position and mass
Dummy details (if fitted)
Dimensions and characteristics of vehicle (private car, bus, etc.)
Position of centre of mass
Photographs

5.5.8 ***Presentation of results***

5.5.8.1 *General presentation*

Test number
Test date
Weather conditions
Description of test sequence
Summary one page chart showing vehicle trajectory relative to barrier

NOTE The chart should also show essential details of the test (e.g. vehicle type, mass, speed, entry angle, exit path, impact severity index, deceleration values, restraint system damage, pass or fail class.)

Deceleration graphs
Strain/force graphs if recorded
Photographs

5.5.8.2 *Restraint system*

Impact point
Length of contact
Maximum dynamic deflection
Maximum permanent deflection
Working width
Major parts fractured or detached
Damage description of restraint system
Details of ground fixing design performance
Forces on restraint system if measured
Whether restraint system was breached
Photographs
Details of whether design criteria have been met.

5.5.8.3 *Vehicle related data.*

Impact speed and variation from target speed and whether within tolerance

Impact angle and variation from target angle and whether within tolerance
Exit speed
Exit angle and whether CEN box criterion was met
Rebound distance if applicable
Whether restraint system was breached
Whether vehicle overturned and where
Vehicle cockpit deformation index (VCDI)
Whether major part of vehicle became detached
Photographs

5.5.8.4 *Impact severity indices*

THIV and PHD impact severity indices
THIV
PHD
Flail distances
Time of flight
Contact point
Vehicle movement relative to ground plane
ASI (acceleration severity index)
OIV and RDA impact severity indices
Flail distances
Forward velocity
Lateral velocity
Ridedown deceleration

5.5.9 ***Test approval***

Test house
Name of approving officer and title
Signature
Date

6 Vehicle stability and safety barrier characteristics

6.1 General

The purpose of a safety barrier is to restrain a vehicle from straying from its carriageway into a more hazardous area where its occupants could be in greater danger than had it remained in its own lane. The severity of impact with the barrier should be restrained to permit an acceptable vehicle to barrier response, i.e. the vehicle should be safely contained and redirected on to a path, either close to the line of the barrier, or where it would be of least impediment to other vehicles. Ideally, the driver would be able to retain or recover control of the vehicle after impact such that he/she could steer the vehicle or bring it safely to rest. Many of the early designs of post and rail safety barriers demonstrated impact responses that did not meet the desired criteria. Phenomena such as pocketing, snagging, spinout and rollover occurred wherein the vehicle became unstable and driver steering control was lost.

Vehicle stability arising from collision with rigid and deformable barriers is discussed in **5.2** to **5.6**. Plain faced barriers mounted on a continuous foundation, of perhaps concrete construction, are not likely to cause spinout unless the coefficient of friction between vehicle and barrier surface is high, or there is some pronounced modelling to the profile. Plain barrier cross-sections that have sloped profiles can induce climbing by the leading impact wheel to the extent that the vehicle rolls out of the barrier and overturns.

During impact, a deformable barrier can absorb energy in addition to that dissipated by deformation of the vehicle's structure. If the barrier is not plastically deformed, energy may be returned to the vehicle and cause a high angle exit path.

The design task is to produce a barrier constraint system to match the available roadside space, that meets the requirements of vehicle redirection, without spinout or rollover, onto a path where it is likely to be of least hazard to following traffic. It is clear that more than one type of barrier material may be used to meet these constraints. Some of the basic factors to be considered in their development are outlined, but the ultimate performance of prototype designs should always be confirmed by full scale impact tests.

6.2 The concept of strong and weak post barriers

6.2.1 ***Weak post barriers***

If, during the impact period, the change in forward velocity per unit time is low, the longitudinal deceleration forces acting on the vehicle will also be low. Consequently, for those barriers where the vehicle is likely to make direct contact with the posts, their strengths, longitudinal to the line of the fence, need to be relatively weak. As the vehicle progresses throughout the barrier contact period, the posts need to be able to be run down without inducing vehicle spinout.

Similarly, lateral deceleration forces will be low if the loss in lateral velocity takes place over the largest transverse distance permitted by the imposed roadside restrictions such as narrow median width or proximity to utility poles. The lateral strength of the posts needs to be sufficient to meet these constraints and so retain the vehicle within the deflection space available behind the barrier. Barriers designed to meet these requirements are designated weak post barrier systems.

6.2.2 *Strong post barriers*

For strong post and rail barriers, care needs to be taken to match the beam stiffness with the post stiffness to avoid pocketing. Pocketing occurs if the beam stiffness is low compared with the rigidity of the posts. If the post spacing is large, the colliding vehicle can generate excessive beam deflection and form a deep bay, or pocket, between post centres. The outcome can either be collision with the post or the vehicle leaves the barrier at a high exit angle.

If the leading wheel snags on a post the vehicle tends to rotate horizontally about a vertical axis. If the post is rigid and does not yield, the phenomenon of spinout occurs with inevitable loss of vehicle control by the driver. To avoid such a hazardous response, the beam needs to be blocked out by a stiff spacer of sufficient depth to prevent contact between wheel and rigid post. Barriers of this type are designated strong post barrier systems and are often restricted to use on roads with low speed restrictions.

Although strong post barriers are relatively unyielding, the magnitude of the impact force normal to the barrier is reduced by movement of the post in its soil footing (wooden posts), or by transverse bending of the post (steel) if the footing is rigid. As the impact rotates the post, blocking out has the effect of raising the beam's effective height. This can be an advantage in restraining vehicles of high centre of mass.

6.3 Factors leading to vehicle spinout

A vehicle that makes contact with a barrier at a small angle is likely to swing towards the line of the barrier moving anticlockwise as presented in Figure 14. Under conditions of increasing approach angle, there is a threshold limit where the vehicle will pivot about the contact point and swing away from the barrier, leading to it spinning out in an unwanted uncontrollable condition.

In Figure 14, the average reaction normal to the barrier is shown as R and the force along the line of the barrier is shown as $F + \mu R$; where μ is the coefficient of sliding friction between the vehicle and barrier, and F is the average value of the forces generated by striking the posts or other obstructions.

It can be deduced from Figure 14 that spinout is prevented if the following conditional equation applies (Moore and Jehu, 1964 [27]):

$$\frac{F + \mu R}{R} < \frac{X}{Y} = \frac{c \cos\theta - b \sin\theta}{c \sin\theta + b \cos\theta} \qquad (22)$$

Equation (22) shows that when $\tan\theta$ is equal to the ratio of the distance of the centre of mass from the front of the vehicle, to its distance from the side, that is $\tan\theta = c/b$, the vehicle will spin out regardless of the value of $F + \mu R$. Examples of these ratios for a 2.3 litre and a 1.1 litre car are, 70° and 67° respectively.

It is of interest to examine equation (22) for values of friction that represent steel sliding on steel (0.1) and steel sliding on concrete (0.3) for the condition when the post bending strength along the line of the barrier is negligible, that is $F = 0$.

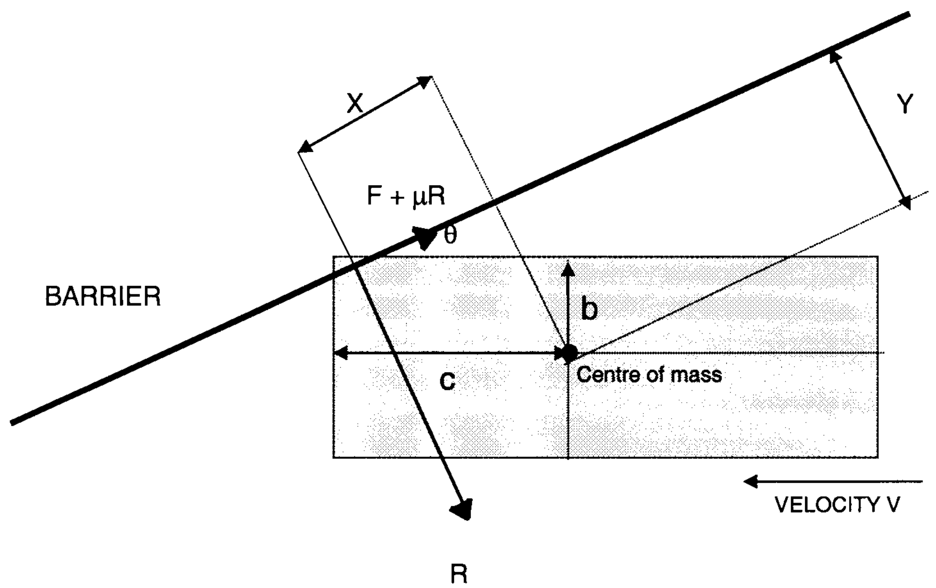


Figure 14 — Diagrammatic presentation of vehicle contact with a barrier

Table 8 gives the critical angle of spinout for two sizes of vehicle for coefficients of friction of 0.1 and 0.3. For this range of coefficients the critical angle for each car differs by only about 2° to 4°. This suggests that critical spinout is reasonably independent of the two sizes of car. However, the effect of spinout on barrier material is more pronounced. The analysis indicates that a car contacting a concrete barrier will spinout at an angle of about 10° to 12° lower than a similar impact on a steel barrier.

Table 8 — Critical spinout angles for selected coefficients of friction

Coefficient of friction	Critical angle (degrees)	
	2.3 litre car	1.1 litre car
0.1 (steel on steel)	64	62
0.3 (steel on concrete)	54	50

For the on-road barrier impact angle of about 20° predicted in **3.2.1**, vehicle spinout is only likely to occur if the coefficient of friction is very large, or if retardation is engendered by contact with posts or other barrier features. To ensure that this retardation is small the vehicle should not make contact with the posts, as prescribed in the design of the blocked out strong post barrier, or the post should be easily run down in the line of the barrier, as required for a weak post barrier.

The interaction of vehicles and barriers during impact is a complex mechanism. The simple analysis presented here can only give a basic insight to this mechanism. It may be of benefit to examine fully designed prototype barrier systems by computer simulation and by full scale impact tests.

6.4 Vehicle lateral deceleration

Figure 14 gives a diagrammatic representation of a vehicle striking a barrier with velocity V, at an angle θ between the car and the barrier. During the impact, the barrier may deflect and the car may crumple. As the vehicle progresses through the impact, its centre of mass moves through a total lateral distance of $Y - b + Z$, where Z is the combined deflection of the barrier and the crushing of the vehicle (Moore and Jehu, 1964 [27]).

The following equation may be derived from Figure 14:

Mean lateral deceleration,

$$\bar{a} = \frac{(V \sin \theta)^2}{2\{c \sin \theta + b(\cos \theta - 1) + Z\}} \qquad (23)$$

Equation (23), when combined with the mass of a vehicle (M), can give a first order of magnitude of the interactive force (R) between the vehicle and the barrier, as given in equation (24) and plotted in Figure 15.

$$R = M\bar{a} \qquad (24)$$

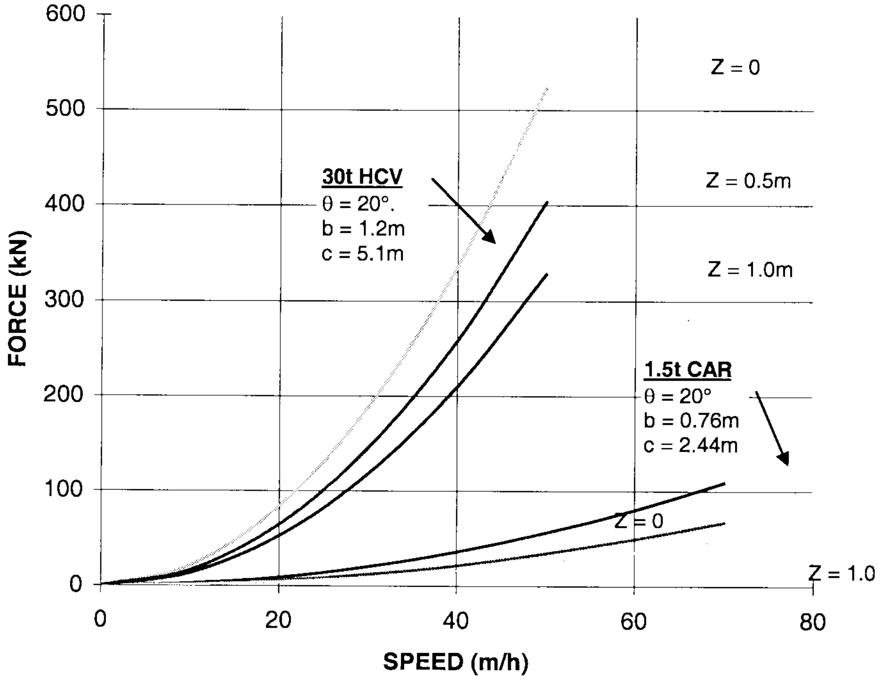


Figure 15 — Average impact force derived from equation (24)

The force R shown in Figure 15 for a 30 t HCV and for a 1.5 t car, is an average value taken from the moment of first contact, to the position where the vehicle has become parallel to the line of the fence; barrier deflection and vehicle crumpling may take place during this period. The moment the vehicle is parallel might not be coincident with the maximum deflection, but it is frequently taken to be so for first order calculations.

Figure 15 demonstrates the reduction in lateral force that can be achieved if the barrier deflects and the vehicle crushes a distance Z. For example a 30 t vehicle impact at 40 mile/h generates a lateral force of about 350 kN when $Z = 0$, however if barrier deflection and vehicle crush permit a value for Z of 0.5 m, the average lateral force reduces to 250 kN, a reduction in load on the barrier of about 30 %.

It should be noted that in full scale impact tests, instrumentation records have shown that peak values can be up to four times greater than average values.

6.5 Vehicle overturn stability against a barrier

Figure 16 illustrates the forces acting on a vehicle tending to rollover a barrier in the vertical plane. From a consideration of the balance between the opposing moments of the reaction forces on the barrier and the vehicle's own mass, it tilts either towards or away from the barrier (Moore and Jehu, 1964 [27]).

The following approximate relationship describes the effect of barrier and vehicle centre of mass heights on the stability of an impacting vehicle under imposed impact force R.

Taking moments about the centre of mass gives:

$$M_a (h_1 - h_2) = (W_1 - W_2)d$$

and:

$$W_1 + W_2 = M_g$$

Combining the two equations gives:

$$2W_1 = M_g - \frac{M_a}{d}(h_1 - h_2) \qquad (25)$$

The vehicle will become laterally unstable and start to roll *towards* the barrier when $W_1 = 0$, that is, when:

$$R = M_a > \frac{Mgd}{h_1 - h_2} \qquad (26)$$

and the vehicle will start to roll *away* from the barrier when $W_2 = 0$, that is:

$$R = M_a > \frac{Mgd}{h_2 - h_1} \qquad (27)$$

The equations illustrate the importance of differences in height between the centre of mass and the effective heights of the barrier. If the heights are the same the vehicle cannot roll over, if the centre of mass is higher then the vehicle will roll towards the barrier and in the limit may overturn, if the centre of mass is lower the vehicle will tend to roll away.

6.6 Impact force and energy absorption

6.6.1 *General*

An appreciation of the factors determining the redirective properties of a barrier may be found in the energy balance of the lateral motion of the vehicle (Jehu and Prisk, 1967 [28]). During impact, energy is exchanged by elastic deformation of the vehicle (E_v) and barrier (E_b), and energy is absorbed by plastic deformation of the vehicle (P_v) and barrier (P_b).

The energy exchanged by the elastic lateral deformation is proportional to the final kinetic energy of the lateral motion of the vehicle. The energy absorbed by plastic deformation is proportional to the difference between the initial and final kinetic energies of the lateral motion of the vehicle.

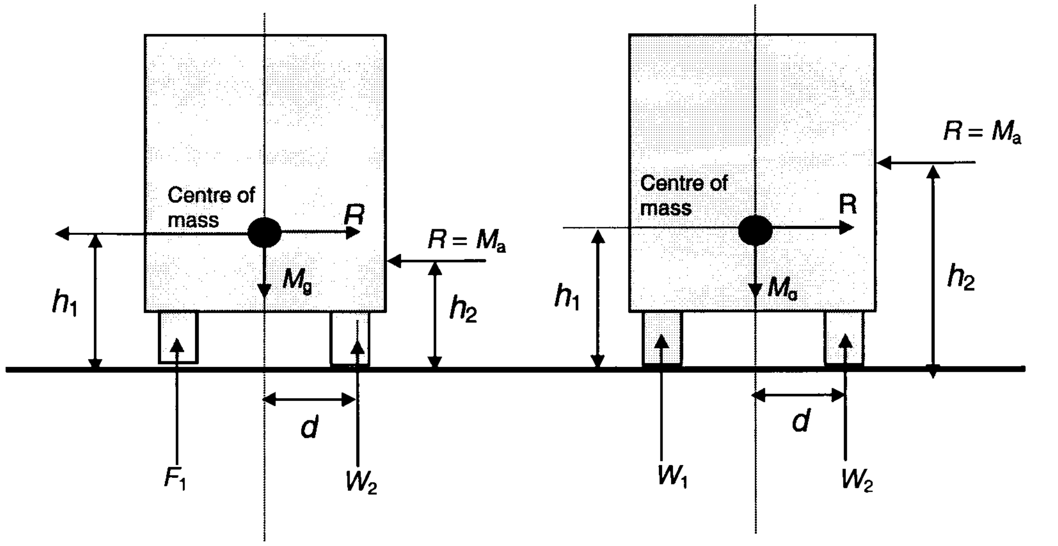


Figure 16 — Vehicles striking barriers of heights h_2; first below and then above centre of mass

If V_b and V_a are the velocities of the vehicle before and after impact, and θ_b and θ_a are the entry and exit angles then:

$$\frac{M}{2}\left(V_b^2 \times \sin^2\theta_b - V_a^2 \times \sin^2\theta_a\right) \approx (P_v + P_b) \qquad (28)$$

$$\frac{M}{2}\left(V_a^2 \times \sin^2\theta_a\right) \approx (E_v + E_b) \qquad (29)$$

or:

$$\sin^2\theta_a \approx \frac{2(E_v + E_b)}{MV_a^2} \qquad (30)$$

Expression (28) shows that damage to the vehicle is reduced if plastic deformation to the barrier is increased.

Expression (30) shows that the angle of departure is reduced if the elastic properties of the barrier are reduced.

6.6.2 ***Characteristics of energy absorbing brackets***

The severity of vehicle impact, into a rigid barrier manufactured in either concrete or steel, can be reduced by fitting energy absorbing brackets between the horizontal beam and the rigid barrier. During impact the hexagonal brackets, shown in Figure 17, plastically deform and cushion the blow to the vehicle (Jehu and Laker, 1972 [29]).

If the wall is assumed to be completely rigid and the crumpling of the vehicle is assumed to be zero, then the value Z in equation (23) represents the crushing of the energy absorber. As an example, by entering typical dimensions in equation (23) for a 113 km/h (70 mile/h) car impact at 20°, the deceleration of the vehicle, over a bracket crush distance of 0.3 m, reduces from 8.5 g to 5.9 g. Similar calculations for a coach show the proportional reduction in deceleration of the centre of mass is lower at 2.3 g down to 1.9 g. The lower value arises through the coach having a greater distance than a car, between the pivot point and its centre of mass. The example demonstrates that where it is necessary to install a high containment rigid barrier on the highway, the impact severity sustained by a large vehicle is likely to be acceptable without the need for cushioning the blow. However, the smaller vehicle will suffer high decelerations and these can be effectively ameliorated by the addition of an energy absorbing bracket.

6.6.3 ***Stiffness requirements of an energy absorber***

To determine the stiffness required for the energy absorber it is necessary to estimate the length that will be plastically compressed under vehicle impact. Assuming the centre of mass of the vehicle follows a curved path of radius D, whilst the vehicle is in contact with the energy absorber over a length L, then:

$$L = 2Z\sqrt{\{(2D/Z) - 1\}} \qquad (31)$$

Since D is much greater than Z, equation (31) approximates to:

$$L = 2\sqrt{2DZ} \qquad (32)$$

For motion in a circular path, the radial acceleration a_D is given by:

$$a_D = V^2/D$$

Hence:

$$L = 2V\sqrt{2Z/a_D} \qquad (33)$$

For a 113 km/h (70 mile/h) car impact at 20° with $Z = 0.3$ m, equation (33) gives a contact length L, of 6.3 m. The load/deflection characteristics of an open box beam requires brackets to be fixed at 1.5 m centres if the beam bending is to be limited to a small value. Hence four brackets are needed to support a 6.3 m length, of which the outer two would be only partly compressed, that is, the equivalent of three brackets are fully compressed.

The compressive force per bracket is $Ma_D/3$, which for a typical car is 29 kN.

In practice, hexagon shaped brackets fabricated from 6.4 mm mild steel plate, having a compression distance of 0.3 m and a vertical height of 0.2 m, were found to have a mean compression load of 30 kN.

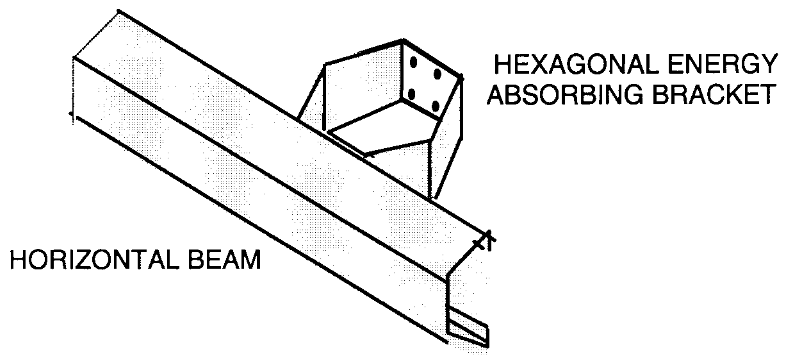


Figure 17 — Open box beam fitted to a hexagonal energy absorbing bracket

7 Vehicle impact severity indices

7.1 General

In the development of restraint systems during the mid 1960s, the measurement of impact severity arising from vehicle collision with vehicle restraint systems was, to a degree, initially of secondary importance to the primary and overriding necessity for practical solutions to the problems of vehicle containment and redirection. Later, progress in the development of novel restraint devices was such that additional methods were required for the assessment of their impact performance, particularly with regard to the generation of harmonized European Standards. The introduction of a numerical index, in addition to evaluating the quality of restraint systems, provided the possibility of ranking acceptable systems and rejecting unacceptable designs.

The measurement of vehicle impact severity involves examining one or more of the following parameters: vehicle crush, vehicle deceleration, vehicle loss in velocity during the impact or measurement of the forces acting on anthropomorphic dummy occupants. The background to the properties of these parameters as good impact severity indicators is considered in **7.2** to **7.10**. The choice taken by the UK to select a dual combined severity index based on the loss in vehicle velocity and on vehicle deceleration, is discussed in some detail.

Within the context of this presentation, the impact severity indices are concerned with the generation of force, or other parameters, which might give rise to injury in a vehicle collision. The specific injuries, or frequency of injury to occupants are not discussed, other than where reference is made to the bio-mechanical literature enabling parallels to be drawn between collision forces in vehicles and medical experiments. The discussion in **7.2** to **7.10** draws heavily from this literature and due recognition is given to those authors.

7.2 Impact velocity and injury

A starting point for the investigation in the UK of vehicle impact severity is the statistical analysis of road accidents where passengers were shown to be injured by being thrown forward into the windscreen or facia panel (Gurdjian, Webster and Lissner, 1949 [30]).

Gurdjian et al [30] considered that the total injurious effect was due to the magnitude and rate of absorption of energy by the head and by the body. They took as a measure for injury, the amount of energy necessary to produce fractures in the skull of a cadaver head. Energy in a moving object is related to the square of its velocity.

They noted that there was no correlation between the severity of cerebral damage and linear skull fracture; that is, fatal injury may result from concussion without skull fracture. It is also true that skull fractures may occur without any damage to the brain. These brief observations highlight the complexities that exist in injury related accidents.

In laboratory experiments, the energy of the blow to the head was controlled by dropping the skull (stresscoated), of known weight, from a measured height onto a polished steel slab of mass 160 lb (72.6 kg). The interior stress patterns were found to be remarkably consistent no matter where the skull was struck. In order to verify the results from the stresscoat tests and to determine their significance if any, of the hair, scalp and head contents, impact tests were made on 55 completely intact human heads. The presence of hair, scalp and contents did not alter the position of the resulting fracture. One surprising result was that after sufficient energy had been absorbed to produce a single line fracture, very little more was required for complete destruction of the head.

The weights of the heads varied from 7.3 lb (3.3 kg) to 14.6 lb (6.63 kg) and the average energy necessary to produce a single line fracture with blows in the frontal region was 571 in·lbf [64.5 J (1 J = 1 N·m)]. The heights from which the heads were dropped, and caused at least one fracture, varied from 34 in to 96 in (0.6 m to 2.5 m).

The corresponding velocities from these heights were 9.2 mile/h to 15.5 mile/h (4.1 m/s to 6.9 m/s). The velocities spanned a range to the extent that at least one fracture to the skull had occurred.

NOTE It is thought that bone tends to become more rigid and brittle after death, and in addition, scalp tissue becomes more flabby and therefore it might be expected to give less protection to the skull so that, after death, smaller impacts might cause fracture to the skull; or conversely, passengers in colliding vehicles might be able to withstand blows of greater intensity than those derived in experiments.

7.3 Deceleration and injury

Shoemaker (1961) [31] produced Table 9 giving the tentative limits of tolerable deceleration. The table originates from the work of Stapp (1955) [32] and Severy (1957) [33]; the studies were based on full scale tests without theoretical guidance.

Shoemaker emphasized that his criteria were tentative, nevertheless Table 9 was produced by many researchers as a standard, even though certain factors such as duration and onset rate of deceleration were controversial.

Graham and Burnett (1969) [34] adopted as a severe injury threshold, a limit of 10 g over a period of 50 m/s. The discrepancy between the time duration of 200 m/s in Shoemaker's table and the 50 m/s quoted by Graham was not addressed, although Olson and Post (1970) [35] considered that Graham's 10 g theory was supportable if the duration of the impact was less than 200 m/s. Nordlin et al (1971) [36] inserted under an abridged version of Shoemaker's table, the comment "highest 50 m/s average, vehicle passenger compartment". Michie and Bronstat (1971) [37] included the amendment on NCHRP 118 (1971) [37] and reintroduced the 500 g/s onset rate. NCHRP 153 (1974) [38] adopted Nordlin's version of the Shoemaker table with some exceptions. The most important were: the lap shoulder restraint figures were deleted and the impact angle was limited to 15°.

Table 9 — Possible limits of tolerable deceration

Restraint condition of occupant	Lateral			Longitudinal			Total		
	Maximum deceleration g	Duration s	Rate of onset g/s	Maximum deceleration g	Duration s	Rate of onset g/s	Maximum deceleration g	Duration s	Rate of onset g/s
Unrestrained	3	0.200	500	5	0.200	500	6	0.200	500
Restrained by lapbelt	5	0.200	500	10	0.200	500	12	0.200	500
Lapbelt and shoulder harness	15	0.200	500	25	0.200	500	25	0.200	500

Chi (1976) [39] in his assessment of injury criteria in roadside barrier tests conducted a wide review of the literature relating to impact severity criteria and summarized his findings as shown in Table 10. He concluded that terms such as peak acceleration, maximum acceleration and onset rate are controversial and are not pertinent to barrier crash studies; they have caused confusion and led experts to the wrong conclusions. They should either be abandoned or standardized with precise meanings.

Chi closed by saying that a too conservative approach to adjustments to the values in Table 10 could set tolerance levels lower than necessary. Even though many existing designs of barrier could meet tighter tolerances, their adoption might have a tendency to disqualify rigid safety barriers that are acceptable and are essential under certain roadside conditions.

7.4 Injury mechanisms

The literature relating to injuries to occupants of vehicles and aircraft is very wide and far ranging. Although it might be an oversimplification it gives some insight to say that the information on deceleration linked injuries originates mostly from the aircraft industry where concern is paramount about the effect on pilots of prolonged high levels of deceleration that arise from aeroplane manoeuvres, or propulsion from ejector seats, rather than the physical contact made by the pilot against solid surfaces in aeroplane crashes. To continue the oversimplification, parallels to injury to occupants of cars might be more closely associated with the events in an aeroplane crash where contact is made between passengers and the interior of the occupant compartment. However, the analogy probably differs after contact is made with the interior, although the car occupant can be exposed to relatively prolonged periods of deceleration whilst the vehicle slows down and remains in contact with the interior of the vehicle.

Michie (1981) [43] commented that dynamic impact force is characterized by the impulsive force generated when a head strikes a car's windscreen and the skull is fractured. The force/time history in this type of deceleration takes place over such a short interval that the impact has elapsed before the body organs, of comparatively high natural frequency, have responded. As far as the body organs are concerned, there is only a change in momentum. Neither the intensity of deceleration nor the duration of the pulse is independently important.

Table 10 — Criteria for recommended tolerance limits for fatal or irreversibly disabling injuries

Occupant restraint	Longitudinal *g*		Lateral *g*		Vertical *g*	
	ΔV	a	ΔV	a	Δv	a
Lap and shoulder belt	12[a]	25[b]	12[c]	25[c]	3[d]	12[d]
Lap belts only	12[a]	20[e]	6[f]	20[c]	3[d]	12[d]
Unrestrained system	6[a]	20[e]	6[c]	20[c]	3[d]	10[d]

ΔV = Occupant impact velocity.

g = average deceleration over duration of impact.

[a] Swearingen (1965) [40].

[b] Douglas Aircraft tests, Kornhauser and Hegenwald quoted by Payne (1961) [41] and HIAD (1960) [42] upper limits.

[c] Assumed to be the same as longitudinal.

[d] Based on Geertz (1914), HIAD (1960) [42] and Payne (1961) [41]

[e] Douglas Aircraft tests, Kornhauser and Hegenwald quoted by Payne (1961) [41] and HIAD (1960) [42] lower limits.

[f] Restraining effect to lateral motion neglected.

An injury mechanism suggested by Kornhauser and Gold (1961) [44] is:

$$\int_0^t a \cdot dt = \Delta V \leq (\Delta V) \text{ limit} \qquad (34)$$

where

a = deceleration of the head in metres per second squared (m/s^2);

t = duration of the pulse;

ΔV = change in velocity of the head in metres per second (m/s).

Equation (34) indicates the injury potential at the end of the first stage of the impact (Michie 1981 [43]).

After the first stage the occupant has moved forward and may well be crumpled against the bulkhead of the vehicle, as has been demonstrated many times in experimental block crashes with dummy occupants. The deceleration force now has had sufficient time for the body response to be fully developed; during this stage the injury mechanism depends essentially on the amount of force that acts on the body rather than the momentum.

A sustained deceleration force results in the deformation and crushing of the body organs and elements and is measured by:

$$a \leq a_{(\text{limit})} \qquad (35)$$

where

a = the deceleration of the head.

Payne (1961) [41] estimated that the minimum duration for body response varies from 7 ms to 40 ms depending on the direction of the force and the particular body element.

By setting a threshold value for $a_{(\text{limit})}$ for a duration of say 10 ms or more, the dynamic injury force is defined; and so the potential at the end of the second stage of the impact is defined (Michie, 1981 [43]).

There is a third, hydraulic mechanism in which dynamic forces act for long periods of time; several minutes or more. An example might be in aircraft manoeuvres where fluids drain from the brain and cause a “blackout” condition. In extreme cases blood vessels can rupture and vital organs haemorrhage. Time periods of this duration are unlikely to occur in barrier accidents.

7.5 Injury thresholds

Results were gathered by Michie (1981) [43] from acceleration sled tests with human and animal subjects and from vehicle and other types of accident data.

Based principally on head impacts into windscreens at velocities ranging from 13 m/s to 16 m/s and an FMVSS 208 [45] head injury criterion (HIC value) less than 1 000, Michie considered that a value of 12 m/s for $\Delta v_{(\text{limit})}$ is a reasonable upper limit for an occupant that moves forward and strikes the windscreen panel. He proposed a forward travel, or flail, distance of 0.6 m.

For lateral impacts where the compartment space had not been intruded, Michie proposed an upper limit of 9 m/s threshold was achievable for the impact velocity. This proposal was consistent with USA, FMVSS 214 [46] and from accident statistics from France derived from car to car accidents.

In the second stage, when the occupant remains in contact with the interior surface, Michie concluded that for both the lateral and longitudinal directions, an upper limiting value of 20 g for the post-impact deceleration was survivable even for pulses of long durations.

NOTE The USA impact severity flail-space model OIV and RDA, discussed in **7.7**, considers the motion of the head inside the vehicle as two uncoupled separate components, that is, the longitudinal direction and the lateral direction. Consequently values of the velocity of impact and the post-impact decelerations, proposed by Michie (1981) [43], are required for each direction. The UK THIV and PHD model mathematically couples these two components and so only single values are required for the impact velocity and the post-impact deceleration.

7.6 THIV and PHD

7.6.1 *General*

Occupants of a vehicle involved in a collision do not necessarily follow the same deceleration profile as the vehicle, even though they might be wearing safety belts. A simplified, but descriptive example would be to consider the motion of a ball tossed into the air inside a vehicle at the moment preceding a very severe impact in which no bodywork crushing takes place. The vehicle instantly decelerates to rest and the ball, whilst airborne, retains its speed. As the vehicle comes to a rapid instantaneous stop, the ball continues forward and contacts the interior of the passenger compartment at the pre-impact speed of the car.

If it is inelastic, and so does not rebound, it takes up the same speed as the car and adopts the same deceleration pattern. If on another occasion there is some crushing of the vehicle’s structure, the car’s loss in speed per unit time is lessened. It follows that the rate of change in the speed difference between ball and car is reduced and so the ball impacts at a lesser speed.

In a similar manner, an occupant in a decelerating vehicle will move forward in a decelerating car, and his head, if unrestrained, will strike the inside of the passenger compartment at a velocity equal to the pre-impact speed of the car. If the vehicle is designed to crush on impact then the occupant will strike the interior at a lower velocity. Clearly, the same effect of lessening the velocity will occur whether it is the car, or the object stuck by the car, that crushes.

In oblique collisions with roadside safety barriers the vehicle experiences longitudinal, vertical and lateral forces as well as rotational motions of pitch, roll and yaw. Consequently a freely moving occupant (head) could impact at any point on the interior surface of the vehicle. In the THIV and PHD analysis, pitch, roll and vertical forces are omitted; this simplification constrains the head to movement in only a horizontal plane. The simplification is supported by the knowledge that high levels of pitch, roll or vertical motion are not usually excessive unless the vehicle overturns; the acceptance or rejection of such an event can be visually assessed.

The description of the vehicle impact given above takes place in a very short time. Depending on the characteristic of the vehicle and the object struck, the duration of the impact period for a restraint system is typically from about 0.1 s to as much as 2.0 s. During this time period the occupant is considered to have experienced two injury producing stimuli, the impact of his head with the interior of the passenger compartment at the relative speed difference between the slowing car and his freely moving head and the deceleration of the vehicle for the remaining duration of the impact whilst his head remains in contact with the vehicle's interior.

The vehicle impact severity criteria adopted in the UK encapsulate these two injury producing phenomena and are known as the theoretical head impact velocity (THIV) and the post-impact head deceleration (PHD).

In a range of impact tests into stiff and flexible barriers, Laker and Payne (1991) [47] found good correlation between THIV and the anthropomorphic dummy impact criteria HIC and CSI used in the testing of new vehicles.

7.6.2 *The background to THIV and PHD*

An early unpublished paper (Moore, 1961 [48]) considered the dynamics of a freely moving head within a vehicle undergoing impact deceleration. Taking the head as the point of reference, during the collision the vehicle is being decelerated towards the head and the speed of impact, V, is:

$$V = \sqrt{2ad} \tag{36}$$

where

a = time averaged deceleration $\left(\frac{1}{T}\right)\int a{\cdot}dt$;

d = distance of head from interior surface;

T = time of flight.

The head may deform on impact so that the movement of the centre of mass of the head might be greater than the skull, but for practical purposes the above relationship is sufficiently accurate.

If the time-average of the deceleration and the distance the head is situated from the interior impact surface are known, the THIV value can be found for any vehicle speed. If the collision is inelastic the head remains in contact with the interior surface for the remaining duration of the impact until the vehicle comes to rest.

During this phase the head experiences the same deceleration as the vehicle. It is most probable that the magnitude of the equivalent deceleration force is much lower during this ridedown phase, than the impulsive force of the initial collision. The deceleration during this period was designated the post-impact head deceleration (PHD); literature in the USA (NCHRP 230 [25]) refers to this as the ridedown acceleration (RDA).

Early application of the single degree of freedom THIV concept (Laker and Walker, 1964 [49]) ranked the impact severity of vehicles colliding with lighting columns according to peak decelerations, mean decelerations during the first 0.1 s after impact and by the THIV values. The three indices placed the impact severity of collision with the columns in different ranking orders of severity. Appraisal of cine film and impact damage to vehicles in previous tests (Walker, 1962 [50]) gave better correlation with THIV than peak accelerations did, suggesting that THIV was a better indicator of impact severity than deceleration.

Head impact velocity with a single degree of freedom on the longitudinal axis of the vehicle continued to be used in the UK for the assessment of impact severity into lighting columns, telegraph poles and similar roadside furniture (Blamey, 1964 [51]). Moore (1961) [48] observed that if the head collision is inelastic the head remains in contact with the impacted interior surface of the vehicle.

In the UK, the concept of THIV and PHD was not used for evaluating oblique impacts into roadside safety barriers at the beginning of developments in the 1960s, partly because at that time the fundamental problems and urgency of producing safety barriers that gave satisfactory vehicle collision response, took priority. Up to 1983 safety fences in the UK were predominately designed for the containment of private cars; subsequently, work started on the development of barriers for heavy commercial vehicles (HCVs). The higher containment barriers were necessarily stiffer, and their effect on the severity of impact by private cars was of significant importance. One of the important benefits of the THIV and PHD concept enabled impact severity to be compared for rigid and flexible barriers under impact by vehicles of all sizes, weights, speeds and angles of impact. The single degree (X-axis) of freedom model was expanded to a three dimensional model to include the lateral axis (Y) and the yaw angle (ϕ) shown in Figure 18.

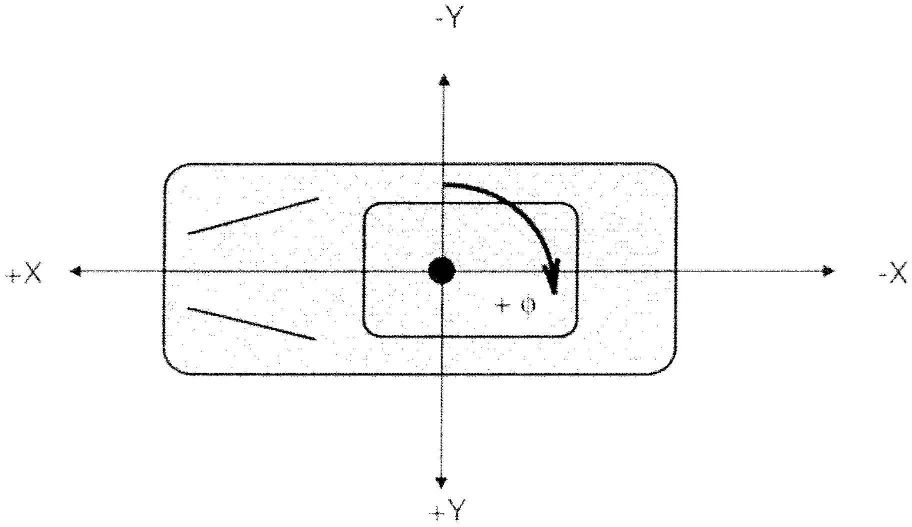


Figure 18 — Three degrees of freedom impact severity model

Consequently, the expanded model permitted the analysis of the impact severity in angled barrier impacts. The longitudinal, lateral and angular components of vehicle dynamics were combined to produce the theoretical head impact velocity (THIV) and the post-impact head deceleration (PHD), (Laker, 1986 [52]).

7.6.3 *The algorithm of THIV and PHD*

7.6.3.1 *General*

THIV and PHD concepts were developed for assessing occupant impact severity for vehicles involved in collisions with roadside safety features. The occupant is considered to be a freely moving object (head) that continues moving as the vehicle loses speed during contact with a restraint system until the head strikes a surface within the interior of the vehicle at a speed designated as the theoretical head impact velocity (THIV) (see Figure 19).

The head is presumed to remain in contact with the surface during the remainder of the impact period. In so doing it experiences the same levels of deceleration as the vehicle during the remaining collision period. The deceleration during this period is described as the post-impact head deceleration (PHD).

The magnitudes of THIV and PHD are considered to be measures of the vehicle's impact severity when in collision with a roadside obstacle. Impact criteria levels are set such that the magnitudes of THIV and PHD, measured in a test, shall not exceed predetermined limiting values.

Vehicle motions may be recorded by accelerometers and rate gyros, but other adequate techniques are available (Laker, Nov 1994 [53]; Dec 1994 [54]).

7.6.3.2 *The theoretical head impact velocity (THIV)*

Figure 20 shows trajectories, relative to the ground, of the centre of mass of a vehicle and a freely moving object within the vehicle. Ground based co-ordinates are denoted by upper case letters X and Y; vehicle based co-ordinates by lower case x and y.

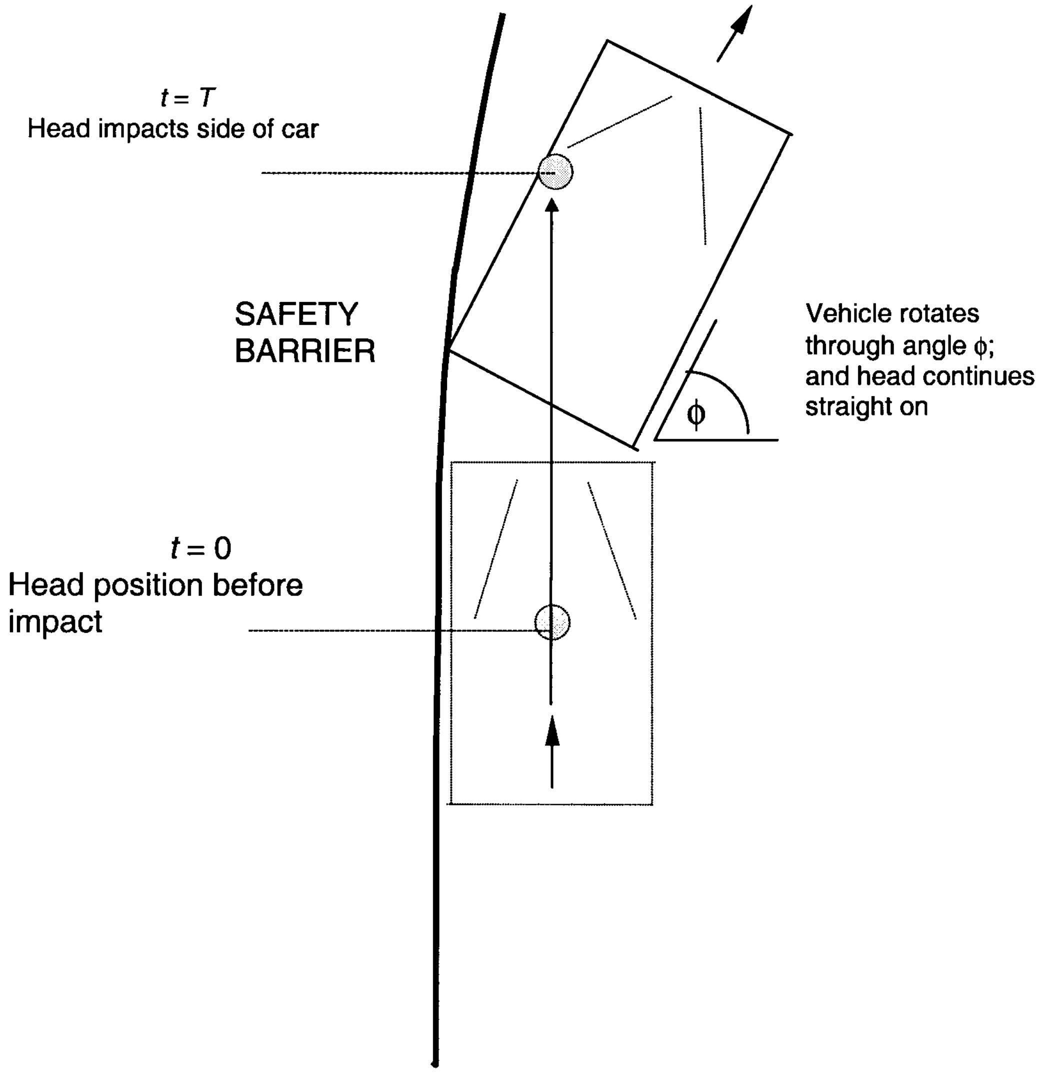


Figure 19 — Head impact in 3-degree of freedom model (oblique collision with barrier)

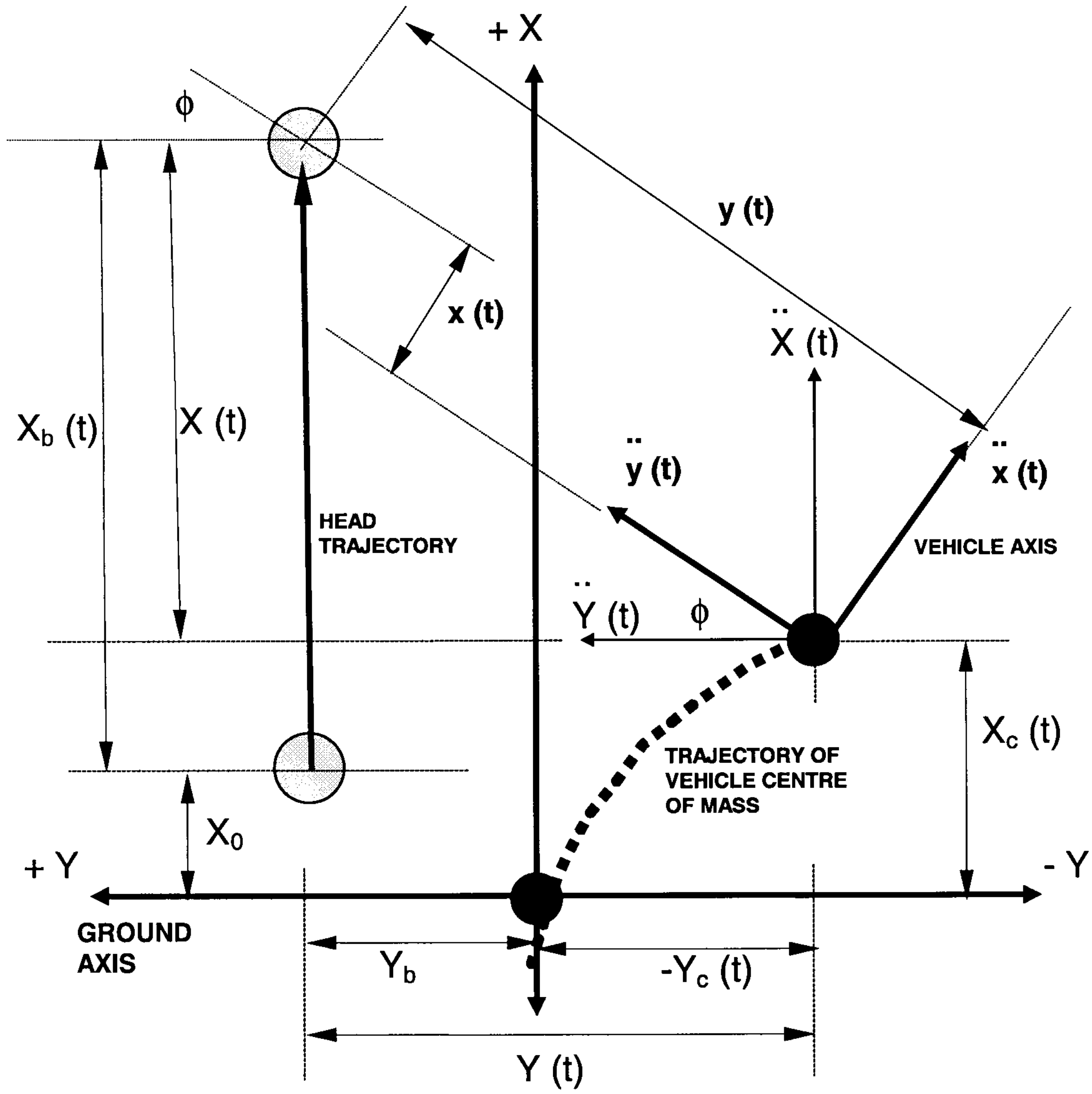


Figure 20 — Head and vehicle trajectories

With reference to Figures 18, 19 and 20, the THIV and PHD values are derived as follows.

Vehicle: movement relative to ground based co-ordinates

Deceleration of the vehicle relative to the ground:

Forward (positive in the forward direction):

$$\dot{X}_c\ (t) = \dot{x}(t) \times \cos \varphi + \dot{y}(t) \times \sin \varphi \qquad (37)$$

Lateral [positive towards the left hand side (LHS)]:

$$\dot{Y}_c(t) = \dot{y}(t) \times \cos \varphi - \dot{x}(t) \times \sin \varphi \qquad (38)$$

where

$\dot{x}$ and $\dot{y}$ are the forward and lateral accelerations of the vehicle as measured by accelerometers (x positive forward, y positive to vehicle's LHS); and

ϕ is the angle of yaw (positive clockwise looking from above).

Velocity of the vehicle relative to the ground (impact point):

$$\dot{X}_c\ (t + dt) = \frac{\dot{X}_c\ (t) + \dot{X}_c\ (t + dt)}{2} \times dt + \dot{X}_c\ (t) \qquad (39)$$

$$\dot{Y}_c\ (t + dt) = \frac{\dot{Y}_c\ (t) + \dot{Y}_c\ (t + dt)}{2} \times dt + \dot{Y}_c\ (t) \qquad (40)$$

where

dt is the time interval of the numerical calculation.

The quantity, t, in parenthesis, references the variable as a function of time.

Displacement of the vehicle relative to the ground (impact point):

$$X_c\ (t + dt) = \frac{\dot{X}_c\ (t) + \dot{X}_c\ (t + dt)}{2} \times dt + X_c\ (t) \qquad (41)$$

$$Y_c\ (t + dt) = \frac{\dot{Y}_c\ (t) + \dot{Y}_c\ (t + dt)}{2} \times dt + Y_c\ (t) \qquad (42)$$

Head: movement relative to ground based co-ordinates

Velocity of head relative to the ground:

$$\dot{X}_b = V_0 \tag{43}$$

$$\dot{Y}_b = 0 \tag{44}$$

where

V_0 is the initial vehicle impact velocity with the barrier.

Displacement of the head relative to the ground (impact point):

$$X_b(t) = \dot{X}_b t \tag{45}$$

$$Y_b = Y_0 \tag{46}$$

Head: movement relative to vehicle based co-ordinates

Transposing from ground based to vehicle based co-ordinates gives:

$$x_b = X \cos\varphi - Y \sin\varphi \tag{47}$$

$$y_b = X \sin\varphi - Y \cos\varphi \tag{48}$$

where

$$X = X_b(t) + X_0 - X_c(t) \tag{49}$$

$$Y = Y_b - Y_c(t) \tag{50}$$

NOTE If X_0 and Y_b are zero the head is positioned at the vehicle's centre of mass.

Velocity of head relative to the vehicle co-ordinates

Differentiating equations (49) and (50), noting X_0 and Y_b are constants and substituting in equations (47) and (48) gives:

$$\dot{x}_b(t) = (\dot{X}_b - \dot{X}_c)\cos\varphi + \dot{Y}_c \sin\varphi \tag{51}$$

$$\dot{y}_b(t) = (\dot{X}_b - \dot{X}_c)\sin\varphi + Y_c \cos\varphi \tag{52}$$

The resultant head velocity (RV) is:

$$RV(t) = \{\dot{x}_b^2\,(t) + \dot{y}_b^2\,(t)\} \tag{53}$$

THIV is calculated at a standard distance, D, after a flight time, T seconds, measured from the moment of vehicle impact. The lateral and forward standard distances are selected as appropriate to the location of the head within the interior of the vehicle. The values adopted in Europe harmonize with the USA, NCHRP 350 [24] document, that is the lateral and forward distances are 0.3 m and 0.6 m respectively.

Then the theoretical head impact velocity (THIV) is:

$$\text{THIV}(T) = \{\dot{x}_b^2(T) + \dot{y}_b^2(T)\}^{\frac{1}{2}} \tag{54}$$

where

T is the time of flight at which the head has struck either the lateral surface or the forward surface.

For the acceptance of a prototype restraint system, the THIV value should not exceed the appropriate limiting value.

7.6.3.3 *The post-impact head deceleration*

The post-impact head deceleration (PHD) is the resultant of the forward and lateral component decelerations of the vehicle's centre of mass.

The resultant deceleration (RD) is:

$$\text{RD}(t) = \{\ddot{x}_c^2(t) + \ddot{y}_c^2(t)\}^{\frac{1}{2}} + \omega^2\, r \tag{55}$$

where ω is the angular velocity in radians per second (rad/s) and r in metres (m) is the radial displacement. The term, ω^2r, is relatively small if the accelerometers are fixed near the centre of mass of the vehicle; the term may then be omitted.

Then the post-impact head deceleration (PHD) is:

$$\text{PHD (after T)} = \{\ddot{x}_c^2(>T) + \ddot{y}_c^2(>T)\} \tag{56}$$

The PHD should not exceed a limiting value, whilst the vehicle is in contact with the barrier.

7.7 The flail-space model, occupant impact velocity (OIV), and ride down acceleration (RDA)

A paper by Michie (1981) [43] provided the corner stone for the inclusion of the flailspace model into the USA, FHWA, NCHRP 230 [25] report. The paper reviewed the injury criteria for three injury models then in use. In particular his review references the work of Shoemaker (1961) [31], Edwards et al (1970) [55] and Tamani and Viner (1970) [56].

Michie found these models either to be inconsistent or to be inadequate measures of occupant risk. He acknowledged that all three criteria indicated occupant risk at least in an overall manner, however the criteria of a maximum average deceleration of 50 ms was probably the least adequate since this period of deceleration could occur and end prior to impact between the occupant and the interior of the vehicle.

As in the case of the THIV and PHD severity criteria, the flail-space hypothesis divides the collision into two stages. In the first stage, designated occupant impact velocity (OIV), an unrestrained occupant is propelled forward and collides with the interior of the passenger compartment. Thus the occupant experiences no injurious forces prior to contact.

In the second stage, referred to as the ride down acceleration (RDA), the occupant remains in contact with the interior surface and experiences the same deceleration force as the vehicle.

Clearly THIV has similar features to OIV. The difference between the two criteria is that THIV takes into account the yaw angle of the vehicle and OIV does not. The main effect of this difference is that the movement of an object (head) inside the vehicle cannot be predicted by OIV. With regard to deceleration, the lateral and longitudinal components of vehicle motions derived from orthogonal accelerometers remain mathematically uncoupled in RDA. PHD combines these two components to give a single value resultant for the head deceleration. Consequently, two sets of values for the impact velocities and decelerations have to be formulated for severity threshold criteria for OIV and RDA, while THIV and PHD require only one set.

To summarize, the THIV model tracks the trajectory of the freely moving head through the passenger compartment and discloses the value of the head impact velocity. THIV also tracks the displacement trajectory of the vehicle relative to a ground plane as the vehicle progresses through the impact. Whereas, the OIV and RDA model simply summates the displacement of the vehicle in the longitudinal and transverse directions, and places pass/fail boundaries on the velocities recorded at the prescribed distance of 0.3 m lateral and 0.6 m derived from the respective accelerometer traces; irrespective of the attitude of the vehicle relative, or to the movement of the occupants in the passenger compartment.

7.8 The ASI ellipsoidal envelope

A USA Military Specification (1967) [57] for defining multiaxial acceleration limits was developed at the Wright-Paterson Airforce base. The Texas Transportation Institute (TTI) (Ross, 1972 [58]) adopted this concept in its investigation of the traffic safety conditions wherein vehicles could become airborne over the terrain in the vicinity of sloping grate culverts.

The term "acceleration severity index" (ASI) was coined by TTI. It required the combined longitudinal, lateral and vertical accelerations of the vehicle at its centre of mass to take a value equal to less than unity as defined by the following equations of an ellipsoid:

$$^{k}\overline{a}_{x} = \frac{i}{m}\sum_{j=k}^{k+m} {}^{j}a_{x} \quad 57)$$

$$^{k}\overline{a}_{y} = \frac{i}{m}\sum_{j=k}^{k+m} {}^{j}a_{y} \quad 58)$$

$$^{k}\overline{a}_{z} = \frac{i}{m}\sum_{j=k}^{k+m} {}^{j}a_{z} \quad 59)$$

$$^{k}\mathrm{ASI} = \left\{\left(\frac{^{k}\overline{a}_{x}}{G_{XL}}\right)^2 + \left(\frac{^{k}\overline{a}_{y}}{G_{XL}}\right)^2 + \left(\frac{^{k}\overline{a}_{z}}{G_{ZL}}\right)\right\} \quad 60)$$

where

G_{long} = acceleration on X-axis;
G_{lat} = acceleration on Y-axis;
G_{vert} = acceleration on Z-axis;
G_{XL} = limit acceleration on X-axis;
G_{YL} = limit acceleration on Y-axis;
G_{ZL} = limit acceleration on Z-axis.

The limit accelerations in the denominator of the above equations are defined as the highest automobile accelerations that an occupant could sustain without serious injury.

The limit accelerations shown in Table 11 have been used by the various authors listed.

The numerator terms are the measured vehicle decelerations along the respective axis averaged over a 50 m/s interval.

7.9 Vehicle interior deformation index (VIDI)

The vehicle interior deformation index (VIDI) was developed in 1970/1971 and is referenced in the European Standard EN 1317-1. The index describes the location and the extent of the deformation of the passenger compartment. It consists of two alphabetic and seven numeric quantities in the format:

X X a b c d e f g

The index provides a standard description of the impact deformation of a vehicle's passenger compartment. The recommended accuracy of measurement is ±0.02 m.

The location of the compartment deformation is indicated by the first two characters "XX", as shown in Figure 21.

The seven characters "a" to "g" indicate the percentage reduction of seven designated interior dimensions:

— a: horizontal distance between the dashboard and top of rear seat;
— b: vertical distance between roof and floor panel;
— c: horizontal distance between rear seat and the engine facia panel;
— d: vertical distance between the lower facia panel and floorpan;
— e: horizontal interior width;
— f: angular distance between lower edge of right window and upper edge of left window;
— g: angular distance between the lower edge of left window and the upper edge of right window;

The values of each of the seven "a" to "g" quantities are determined by the following scale:

— 0: if the reduction is less than 3 %;
— 1: if the reduction is less than 3 % and less or equal to 10 %;
— 2: if the reduction is more than 10 %.

If there are reductions of more than 10 % then a photograph of the damage should be provided.

Table 11 — Limit accelerations for ASI

Longitudinal G_{XL}	Lateral G_{YL}	Vertical G_{ZL}	Author
7	5	6	H. Ross 1972 [58]
20	15	17	G.D. Weaver [59]
5	3	—	Preferred NCHRP 153 [38]
10	5	—	Acceptable NCHRP 153 [38]
25	15	—	Occupant NCHRP 115 [60] restrained
12	9	10	France: with seatbelt [43]

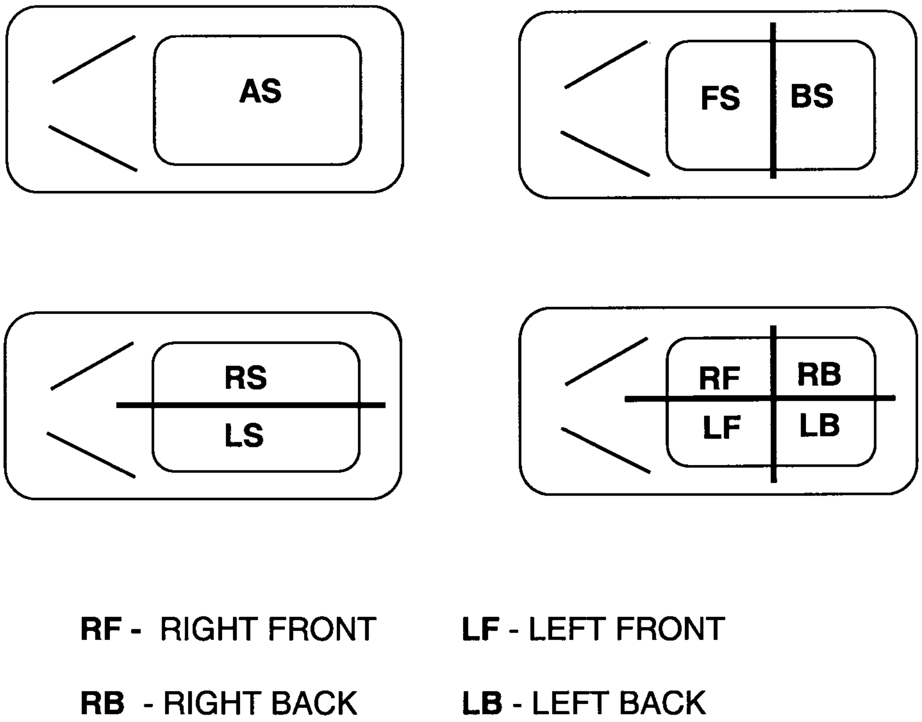


Figure 21 — Location of passenger compartment deformation

7.10 Summary of clause 7

Three concepts for assessing the severity of vehicle impact, based on predictions of the forces acting on a notional vehicle occupant, have been reviewed: the THIV and PHD model, the OIV and RDA model, and the acceleration severity index (ASI).

The ASI deals with the injuries caused by deceleration forces acting on elements of the human frame and organs. The index is calculated from measurements of deceleration, taken in three orthogonal planes, using an ellipsoidal equation; an index value greater than unity implies a serious, if not fatal, level of deceleration.

Most probably the ASI concept originates from the USA, (Wright-Paterson Air Force Base) where it was used to quantify the effects of relatively prolonged durations of acceleration forces on pilots involved in seat ejection experiments. Periods of injurious deceleration mentioned in the relevant medical literature vary from 50 ms to 200 ms; this span of time, when related to vehicle impact, might conceal injurious spikes of shorter duration.

During the early stages of vehicle impact, the occupant, whether restrained or not, does not experience the same deceleration forces as the vehicle's centre of mass; the position in most impact tests where the recording instruments are placed. Consequently levels of deceleration could be measured that, in reality, are not experienced by the occupant.

To overcome this problem, researchers in vehicle accident injury in the USA and the UK have developed a technique that considers injurious forces in two stages. The first stage begins at the moment of vehicle impact into an object, and ends when the car's occupants collide with the interior of the vehicle. Medical workers have studied injuries arising from head impacts into solid surfaces. Injury levels observed in head impact velocity experiments can be compared to the injury occurring from passenger impact within the compartment interior. The free flight trajectory of the passenger is derived from measurements, in the lateral and longitudinal planes, of the deceleration of the vehicle's centre of mass; from these records, the passenger's impact with the interior of the vehicle can be evaluated. Vertical decelerations were considered to be of little value in assessing vehicle impact response unless they were so violent that the vehicle overturned. In which case, visual assessment is sufficient to accept or reject the test, as appropriate.

The observation that passengers wearing seat belts do not experience equivalent velocity changes is perfectly correct, however this does not detract from the merits of the technique in evaluating vehicle impact severity.

The second stage considers the period after the passenger has made contact with the interior of the vehicle and, from then onwards, experiences the same decelerations as the vehicle; as observed by the movements of anthropomorphic dummies in controlled vehicle crash block tests. During this second stage it is the comparatively prolonged deceleration forces that are likely to cause injury; an upper limit of 20 g has been established from biomechanical experiment. It is of some note, that in frontal block test collisions, the engine begins to be pushed backwards into the passenger compartment at levels of deceleration of the centre of mass above 20 g to 25 g; thus a deceleration severity limit which avoids this condition is beneficial.

The essential difference between the USA and UK measures of vehicle impact criteria is that in the UK model the longitudinal and lateral decelerations are mathematically coupled with the aid of yaw rate measurement while in the USA model they are not. This means the UK method requires only single threshold values for impact velocity and deceleration, whereas the USA index requires two values for each axis. In addition, although the USA index attempts to quantify the flail distance, it does not take account of the rotation of the car, so the prescribed flail distance necessarily has to be only an approximation of the true distance. The UK index does track the movement of the theoretical occupant through the passenger compartment from any pre-impact position. It also tracks the vehicle, against a terrestrial axis, as the car progresses through the contact with the roadside restraint system. Consequently in the THIV model there is the benefit of a realistic model of the passenger's movement and an assessment of his impact into the interior of the car, plus a record of the path of the vehicle during the test.

8 Model simulation of vehicle impact into highway restraint systems

8.1 General

The cost of full scale impact testing of prototype vehicle restraint systems during the design development stage, forms a very large part of the development costs. For example, the present cost of one full scale impact test with a 1 500 kg car, is equivalent to the installed price of about 500 m length of a basic roadside safety barrier. Assuming profit on the sale of safety barrier is 10 %, it would be necessary to sell 5 000 m (3.1 miles) of barrier to recover the cost of one test. In the UK, prototype designs of safety barrier require two proving tests; the total number of development tests can be well in excess of the number of statutory tests. Clearly, the cost of full scale testing, as a proportion of the development costs of a novel restraint system, needs to be closely considered at the outset of a new initiative.

The use of scale models for restraint system development, either in the form of mathematical, physical or computer simulation, seems an attractive route both financially and practically if the models are reliable. However, as a general statement, it is likely that because of the complexity of the mechanical response characteristics of vehicle impact into restraint systems, modelling of such events are only likely to provide interpolation between full scale testing rather than extrapolation. In other words, full scale testing can provide the data for validation of models, and useful results can be obtained from the validated model, particularly where the effect of simple parametric changes are investigated, such as vehicle speed, vehicle mass, post spacings or impact angle. Eventually, though, full scale testing is necessary to pass statutory regulations.

8.2 Mathematical models

The interaction of an impacting vehicle and even the simplest restraint systems is sufficiently complex almost to exclude general analytic mathematical modelling other than the basic concepts discussed in clause **6**. Certainly, refinement and optimization of the detailed mechanical designs are virtually ruled out. The difficulties arise from the large number of variables that need to be considered which include vehicle crush characteristics, steering dynamics, road surface and tyre adhesion, vehicle speeds, position of centre of mass, moments of inertia, vehicle dimensions, suspension characteristics etc., and for commercial vehicles the number of axles and tractor trailer articulation are added complexities. This abbreviated list of variables excludes the mathematical description required of the restraint system under test. For the simplest elastic or energy absorbing post and rail restraint system, the bolted fixings, beam strengths, post strengths, material characteristics, friction, plastic deformation and failure characteristics of the constituent components must be mathematically modelled.

An early paper (Giavotto, 1967 [61]) produced analytic expressions for the deformation of a continuous beam of infinite length (barrier horizontal rail) on various types of support, including elastic-plastic and rigid plastic (barrier posts), when subjected to both a concentrated and a distributed force.

In addition to the difficulties described, the interaction of the vehicle's variables with the restraint system's variables need to be modelled to predict the performance of the total system under analysis. In general it is reasonable to say that the complexity of such interaction has ruled out precise predictions of restraint system deformation and the vehicle's pre- and post-impact trajectories, by analytic mathematical models.

8.3 Physical scale models

The possibility of using scaled down physical models of vehicles and restraint systems seems attractive. However, it should be recalled that where a reduced scale model is used it needs to be able to predict the full scale behaviour of the prototype. To achieve this, the dimensions of the physical model and the characteristics of the material used in its construction must bear certain definite relationships to the dimensions and materials of the prototype.

The principles which govern these relationships are referred to as principles of similitude (Hetényi, 1966 [62]). The principles govern both the design of the model and the means of extrapolating the results from the model tests to predict the performance of the prototype. For models of metal construction it can be advantageous to employ a material which has a lower modulus of elasticity than that of the prototype, so that distortions which arise under test are sufficiently large to be accurately measured, without resorting to excessive and unrealistic loading forces. As the scale of the model is reduced it becomes increasingly difficult to maintain exact geometric similarity, and so the duplication of all the details of the prototype becomes physically more difficult to the point of impossibility. It should be borne in mind that there is the possibility that the practical scale model route for restraint system development might be equally, or more expensive than full scale impact testing.

8.4 Computer simulation

8.4.1 *General*

Attempts were made to construct analogue models (Giavotto and Caprille, 1967 [63]) of vehicle/barrier impact response. Analogue computers were restrictive in the number of elements that could be modelled; scaling and instability of computer amplifiers also added to the difficulty in achieving reliable results.

With their growth, availability and reliability, analogue computers were quickly replaced by digital computers for vehicle impact simulation. Digital simulations of vehicle to barrier impacts have been developed world wide. Some of the computer software developments in Holland, USA, and the UK are mentioned in **8.4.2**, **8.4.3** and **8.4.4**, either because the program listings are available, or because digital simulation has been used in a specific area of vehicle restraint system design.

8.4.2 *Computer models developed in Holland*

The VEDYAC model was developed by the SWOV laboratory in Holland (Heyer, 1988 [64]). It was developed from an earlier model, MAMIAC, written specifically for simulation of passenger car impacts against steel guard rails. MAMIAC could only be run on a main frame computer. The decision was taken to rewrite the software according to the following requirements under the program name of VEDYAC:

— the model should be able to be run on desktop computers;

— the range of vehicles should cover small passenger cars to heavy commercial vehicles (HCV's);

— the model output should predict collision deformation and vehicle manoeuvring behaviour;

— the model should be easy to use (user friendly).

VEDYAC represents vehicle and barrier components moving in three dimensional space, by springs and masses. The shapes of components, such as vehicles and restraint systems, are described in terms of:

— geometrical planes: one model component might contain an unlimited number of planes;

— cylinders: of varying length and diameter, attached together in unlimited numbers;

— pseudo finite element structures: a network of point masses connected by rods and beams.

NOTE A detailed summary is given by Heyer (1988) [64], [65].

The model has been used to examine vehicle roll inducing characteristics of two concrete safety barriers of different cross-section profile; the General Motors and the New Jersey shaped profiles. The results of the analysis showed the maximum height climbed by the leading wheel of the impacting vehicle tended to be lower for the New Jersey shaped barrier. The difference in height climbed was more pronounced for the smaller cars. The severity of impact, as defined by ASI (see **7.8**), always exceeded the accepted safety limit of 1.0. Under severe impact conditions the VEDYAC model was able to predict vehicle rollover, (Heyer, 1988 [65]).

8.4.3 *Computer models developed or used in the UK*

TRL commissioned Cranfield Institute of Technology, under contract, to investigate the use of computer models for the simulation of vehicle to barrier impacts. The programs considered were CRASH, SMAC, CRUNCH, BARRIER VII, CRASH-D, KRASH, HVOSM and CVS. Reference to some of these early USA models, and their characteristics, are reproduced in NCHRP 230 (Michie, 1981 [25]). The program CALSPAN CVS simulates the movement of an occupant within the passenger compartment. The project covered a variety of vehicle and barrier types with only a minimum of vehicle data available and so a combination of CRASH-D, KRASH and the CVS program were selected as the most effective tools for the analysis at hand, (Sadeghi and Blake, 1986 [66]).

Program CRASH-D (Kecman, 1981 [67]) is essentially a finite element program. A steel post and rail barrier was modelled as a single beam supported by posts at selected regular intervals. The collapse analysis under the CRASH-D program enabled the load bearing curve of the barrier to be generated.

The load curve acts as an input to the program KRASH, that is, CRASH-D produced the impact force that the vehicle simulation program KRASH would “see” during contact with the barrier. Concrete barriers were modelled in the form of an infinitely stiff spring which could eventually collapse at a prescribed yield point.

Program KRASH (La Barge, 1978 [68]; Gamon, 1978 [69]) simulates an impacting vehicle by a number of masses interconnected by a number of beam elements. Systems of point masses and springs represent tyres and suspension, and they in turn, were connected to springs and masses which represented the bodywork. Vehicle to barrier impact loading was simulated by the load and deflection properties derived from program CRASH-D.

Calspan CVS (crash victim simulation), (Sadehgi and Blake, 1986 [66]), simulates movement of a vehicle occupant. The program presents the occupant’s head, torso, arms and legs by three dimensional ellipses which have the properties of mass and inertia and are jointed together by hinges that have both viscous and damping characteristics. The model’s components are constrained to have a range of movement similar to that of a human body or anthropoid. The interior of the vehicle is composed of planes that represent the seats and interior surfaces of the passenger compartment. Contact of the human model with the vehicle interior is generated by defining the acceleration characteristics of the vehicle; the acceleration values are an output parameter from the program KRASH.

The accuracy of simulation is affected by the extent to which the vehicle structure collapses under impact. Whilst parameters such as vehicle mass and inertia can be measured fairly accurately, vehicle crush characteristics are difficult to estimate without the benefit of a costly exercise to achieve whole vehicle structural collapse analysis.

After calibration of the suite of programs by crash tests with a passenger car, a 16 t rigid commercial vehicle and a 38 t articulated heavy commercial vehicle were used to investigate the effect of halving the post spacing of a three rail steel parapet, (Sadeghi and Walton, 1988 [70]).

The capacity of a parapet at the normal containment level is intended to withstand impact by a 70 mile/h, 1 500 kg vehicle at 20°. The maximum energy dissipation under such an impact is 86 kJ; at this energy level maximum parapet deflection was predicted by the model to be 0.64 m. Halving the post spacing showed that, at the same deflection, the impact speed for containment increased to about 108 mile/h. The proportional increase in energy dissipated at this level of impact increased about 2.4 times.

Alternatively, the simulation predicted that by halving the post spacing containment was possible of a 16 t commercial vehicle impact on an approach line of 15°, whose speed was near 45 mile/h.

The simulation model using programs CRASH-D and KRASH indicated that the capability of a steel bridge parapet designated as normal containment in BS 6779 could be substantially improved by halving the post spacings. The model results have not been subjected to checking by full scale impact testing.

Program DYNA3D, a finite element program, developed in the USA, was released in 1990 under the name of LS-DYNA. After some enhancement, including the addition of program MADYMO, it became available in the UK in 1991 as DYNA3D 5.1. It was used by TRL to examine the interaction between an impacting vehicle and a crash cushion. DYNA3D was developed for use by the motor car industry whose interest centred on the detail of vehicle impact zones; the buckling and deformation of body panelling and structures in impacts with single objects were of prime importance.

Such refined detail is not required in the modelling of vehicle collision into the relatively longer structures associated with safety barriers. The program was adapted to permit reasonable computer run time by splitting the number of degrees of freedom permissible, between the vehicle and the roadside hardware. Using this technique, impact at 107 km/h of a 1991 Saturn passenger car and lighting column was simulated and adequately confirmed by a full scale impact test; the simulation ran for 0.15 s. The comparison between model and crash test indicated very close comparison of engine block deceleration magnitudes up to the maximum generated value of about 60 g, (Oskard et al, c1993 [71]).

Finite element models of barrier posts and w-section beam rails have been simulated, and work is continuing to develop DYNA3D into an effective tool for vehicle impact simulation into highway restraint systems (TRL Ltd., 1992 [72]).

Program ROPES was developed by the University of Sheffield under contract to Bridon Plc. (Howard and Johnson, 1990 [73]). The model consists of a set of mathematical equations which represent vehicle impact into a highway restraint system, which is composed of collapsible posts and horizontal wire ropes loosely connected to the posts. The governing equations describe the motion of the vehicle’s centre of gravity, the geometric shape of the vehicle, the deformation of the posts, the transfer of momentum between vehicle and posts and the deformation of the rope. The model does not include the rotational retardation due to friction between the tyres and the road, and its interaction with the dynamic response of the ropes.

Simple approximations permit analytic solution of the set of equations without the need for computer simulation. The vehicle is represented as a point mass. Simple assumptions are made concerning the release of the ropes from the posts and the effective stiffness of the barrier.

Although fairly elementary, the mathematical model provides valuable interpretative results in advance of the more precise, complex numerical analysis that is available within the computer program ROPES.

Program ROPES is written in the programming language FORTRAN and it is suitable for installation on a personal computer with a 40 Mb hard disk storage capacity. The simulation has been used to compare the impact performance of a wire rope fence mounted on traditional steel posts, with the barrier's performance when then steel posts are replaced by glass fibre reinforced plastic posts (GFRP) (Bateman, 1995 [74]). Preliminary results indicate that GFRP box section posts compare favourably with the steel Z-section posts. The fence becomes less stiff with increase in post spacing although surprisingly less than might be expected, pre-loading of the ropes has some effect on barrier penetration and post strength is a larger contributor to fence stiffness than post spacing.

8.4.4 *Computer models developed in the USA*

The USA has been most prolific in the generation of the number of computer programs that simulate vehicle dynamics during collision with highway restraint systems. Additionally, several models have been developed to simulate the movements of occupants in vehicle passenger compartments. Some of these programs are referenced in the NCHRP Report 350 (Ross et al, 1993 [24]) produced under the auspices of the Federal Highway Administration (FHWA); they are reproduced in Table 12.

The publication NCHRP 350 [24] confirms that simulated results are sensitive to vehicle parameters that are sometimes difficult to obtain or estimate. Vehicle moments of inertia, crush properties and suspension characteristics are not generally published and need to be found by experimentation. Barrier movements can be difficult to predict because of the non-linearities in the system such as slack in bolted joints, soil conditions, friction and the stability or collapse of structures. The publication also affirms that where a program has been validated for a range of impact conditions, it can be used with confidence to predict events within that range; although investigation of conditions beyond the validated range can provide some insight, caution should be used in conditions where results have been derived through extrapolation of the model parameters.

Table 12 — Summary of some USA highway computer programs

Name	Producer and date of last modification	Principal use	Model	Validation	Source	Comments
HVOSM	Calspan, 1989	Simple vehicle Rigid barrier	3D Lump mass vehicle	Extensive	FHWA R&D	Excellent wheel suspension system analog. Simplified vehicle crush analog
BARRIER VII	University of California, 1973	Simple vehicle Flexible barrier	2D Finite element	Extensive	FHWA R&D	For use where vehicle roll and pitch are negligible
GUARD	HTRI, 1989	Simple vehicle with bumper model Flexible or rigid barrier	3D Finite element Vehicle/barrier model	Limited	FHWA R&D	Developed from CRUNCH; simple wheel/suspension model
NARD	HTRI, 1989	Articulated vehicle Flexible or rigid barrier	3D Finite element	Limited	FHWA R&D	Similar to GUARD but with an articulated vehicle
SMAC	Calspan, 1975	Rigid 2D vehicle to vehicle motion No barrier	Vehicle to vehicle or other fixed object	Limited	NHTSA	For use in vehicle to vehicle accident reconstruction Amended for barrier impacts

9 British and European Standards on road restraint systems

9.1 General

BS 6579 *Safety fences and barriers for highways*: the drafting of British Standards on safety fences and barriers for highways began in the early 1980s. A safety barrier was defined as an installation which is continuously in contact with its supporting foundation (e.g. concrete barrier). A safety fence was defined as a construction of beams mounted on posts. The first publication was under the British Standard series BS 6579, in 1985. No requirement for vehicle impact testing was listed in the preliminary documents, although an appendix listed dynamic characteristics in terms of vehicle impact speeds, fence and barrier deflections; also a desirable trajectory for the vehicle during and after impact was described. Later, an impact performance requirement was added to the British Standard which included prescribed classes for dynamic deflection of fences and barriers, and a measure for the determination of the impact severity sustained by a colliding vehicle.

BS 6779 *Highway parapets for bridges and other structures*: the drafting of British Standards for bridge parapets began in the early 1980s under the British Standard series BS 6779. A parapet was defined as a barrier at the edge of a bridge, or on top of a wall, or similar structure. Whereas the British Standard for safety fences and barriers dealt mainly with standardization of components, the British Standard for parapets concentrated on design parameters for the construction of parapets. Requirements were included for the impact testing of prototype parapets.

BS 7669 and PD 6634 *Vehicle restraint systems*: the information contained within BS 6579 and BS 6779 is being compiled and expanded to form a comprehensive publication in the form of a reference manual or handbook to permit easy reference to the fundamentals of design of safety barriers, barrier specifications, layout, installation, inspection and repair.

European Standard EN 1317 on road restraint systems: the European Standards authority CEN (Comité Européen de Normalisation) was instructed, in 1990, to prepare performance standards for road restraint systems. The work was allocated to TC 226, working group 1 (WG1) with the assistance of task group 1 (TG1) to advise on technical matters. The standards are in preparation under the title of EN 1317 Road restraint systems. The standards encompass the performance of vehicle restraint systems which include safety fences, safety barriers, parapets, terminals, transitions, crash cushions and pedestrian restraint systems. Durability, product identification and evaluation of conformity of production are covered by prEN 1317-5.

9.2 BS 6579 Safety fences and barriers for highways

BS 6579 is divided into the following parts:

— *Part 1: Specification for components for tensioned corrugated beam safety fences on Z-posts*;

— *Part 3: Specification for components for tensioned rectangular hollow section beam (100 × 100 mm) safety fence*;

— *Part 4: Specification for components for tensioned rectangular hollow section beam (200 × 100 mm) safety fence*;

— *Part 5: Specification for open box beam safety fence (single height)*;

— *Part 6: Specification for components for open box beam safety fence (double height)*;

— *Part 7: Specification for untensioned corrugated beam safety fences*;

— *Part 8: Specification for concrete safety barriers*.

Work on the following parts is under consideration:

— *Part 9: Specification for permanent and temporary vertical concrete safety barriers*;

— *Part 10: Specification for high containment transitions links between safety fences and bridge parapets*;

— *Part 11: Specification for wire rope safety fences*.

9.3 BS 6779 Highway parapets for bridges and other structures

BS 6779 is divided into the following parts:

— *Part 1: Specification for vehicle containment parapets of metal construction*;

— *Part 2: Specification for vehicle containment parapets of concrete construction*;

— *Part 3: Specification for vehicle containment parapets of combined metal and concrete construction*;

— *Part 4: Specification for parapets of reinforced and unreinforced masonry construction*.

9.4 BS 7669 Vehicle restraint systems

The following part of BS 7669 is currently available:

— *Part 3: Guide to the installation, inspection and repair of safety fences*.

The following parts of BS 7669 are under consideration:

— *Part 2: Specification and layout of safety fences and barriers*;

— *Part 4: Guide to the installation, inspection and repair of safety barriers*;

— *Part 5: Aspects of bridge parapets*.

9.5 EN 1317 Road restraint systems

EN 1317 is divided into the following parts:

— *Part 1: Terminology and general criteria for test methods*;

— *Part 2: Safety barriers — Performance classes, impact test acceptance criteria and test methods.*

The following parts of EN 1317 currently exist as draft European Standards (prEN 1317):

— *Part 3: Crash cushions — Performance classes, impact test acceptance criteria and test methods*;

— *Part 6: Pedestrian restraint systems, pedestrian parapet.*

The following parts are under consideration:

— *Part 4: Barrier systems: Terminals and transitions — Performance classes, impact test acceptance criteria and test methods*;

— *Part 5: Durability, product identification and evaluation of conformity.*

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