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PhD THESIS

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DAVID WHITE

An Investigation of Bus Design Parameters Affecting Rollover

Supervisor:

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November 1985

"...the LORD looked down from the pillar of fire and cloud at the Egyptian army and threw it into confusion. He made the wheels of their chariots swerve so that they had difficulty driving. The water flowed back and covered the chariots and horsemen - the entire army of Pharaoh that had followed the Israelites into the (Red) sea. Not one of them survived."

Exodus 14:24,25,28
The Holy Bible,
New International Version,
Hodder & Stoughton, 1978

[very early example of vehicle dynamics problems
and large-scale transport catastrophe]

ABSTRACT

Reduction in the number and severity of bus and coach rollover accidents is the broad aim of this thesis.

Chapter 1 outlines, from accident data, research needs and priorities regarding bus safety. Rollover accidents, particularly those due to sliding sideways into a "tripping" obstacle, are recognised as a major contributor to bus occupant casualties. Occupant injuries and fatalities are significantly higher if a coach rolls onto its roof or beyond, than if it rolls onto its side. A typical sequence of events leading to overturning accidents and consequently to occupant casualties is identified, and research needs into appropriate countermeasures for each step are suggested. Work towards two of these countermeasures is outlined.

Chapter 2 describes the first area. A full-scale coach rollover test for measuring superstructure impact strength against a standard, required a mathematical model to explain and predict the motion of unsymmetric vehicles. A computer model was developed, and reasonable agreement between simulation and actual tests is demonstrated, despite the large unmeasurable energy losses during the "kerb-impact" phase of the tests.

Research into "tripping" rollover stability, the second area, occupies Chapters 3 to 8. Experimental measurements of centre of gravity height, mass, moment of inertia, roll centre heights, suspension and tyre stiffnesses, and maximum stable tilt angle were undertaken on nine representative buses and coaches. A twelve degree-of-freedom computer simulation of a kerb-impact-initiated rollover for a vehicle with front and rear suspensions has been developed, checked against other methods, and used to predict rollover threshold velocities (which ranged from 2.8 to 4.2 m/s for the tested vehicles).

The effect of varying bus parameters on the rollover threshold was assessed and compared with their influence on tilt test performance as evaluated using a tilt test simulation. Centre of gravity height and track width had the greatest effect on results from both simulations. Correlation between tilt test results and computed rollover thresholds is reasonable. Both methods are recommended above a number of others as alternative ways of demonstrating adequate roll stability. Rollover threshold standards and new tilt platform limits are proposed, based on the current double-decker standard, but without self-levelling valves operating. Prohibition of tilt valves and disabling of self-levelling valves during a tilt test are recommended. These are some of the recommendations made to the U.K. Department of Transport concerning methods and standards of overturning stability testing.

The major conclusions are presented in Chapter 9, which gives an overview of the thesis contents and findings.

ACKNOWLEDGEMENTS

The achievements and findings described in this thesis were only possible with the assistance and cooperation of many people.

Mr Guy Tidbury, my supervisor and past Director of Cranfield Impact Centre, deserves special mention for his encouragement and direction. He and Mr A. Chatfield, of the Department of Transport, were the main instigators and co-ordinators of this research. When Mr Chatfield left the Department, his responsibilities were taken over by Mr A. Reynolds.

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The following operators and manufacturers cooperated by arranging test vehicles and/or discussing current designs and proposed changes:

DAF Bus,

Bedford Commercial Vehicles,

Jonckheere Carrosserie,

London Regional Transport,

National Bus Company,

Plaxtons,

Leyland Bus,
Metro-Cammell Weymann,
Ramoudt Tours,
Scania, and
Yeates of Scarborough.

The first version of the computer simulation of the bus rollover test was written by M. Openshaw as a major project in the course for his Master's Degree.

The assistance of the Computer Centre, Library and Photographic staff is acknowledged.

Because 'the first shall be last', the greatest thanks are for the one at the end. The unflagging support, interest, encouragement, devotion, tolerance and perseverance of my wife, Helen, has kept me sane, healthy and well-fed. She also proved to be most helpful in proof-reading and helping in the final assembly of this document. (Her reward is a trip for two to New Zealand!)

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NOTATION

Note that SI units are used throughout. Where references are not in SI units, the SI equivalent is also included.

a	longitudinal position of axle relative to sprung mass (a_f positive, a_r negative)
b	distance from sprung mass CG to roll axis
C	linear suspension damping
D	rotational suspension damping
F	force
FR	suspension roll moment
FS	suspension force
g	acceleration due to gravity (taken as 9.81 m/s ²)
h	distance from roll axis to axle
hrc	roll centre height
H	angular momentum
i	unit vector in X-axis direction
I	moment of inertia
j	unit vector in Y-axis direction
k	unit vector in Z-axis direction
K	linear suspension stiffness
m	mass
p	roll rate
q	pitch rate
Q	suspension roll stiffness
r	yaw rate
R	position vector
s	lateral displacement of axle relative to sprung mass
T	torque
u	forward velocity
v	lateral velocity
V	linear velocity vector
w	vertical velocity
x	forward displacement
y	lateral displacement
z	vertical displacement
ϕ	roll angle
θ	pitch angle
ψ	yaw angle
θ_0	roll axis inclination (with respect to bus)
ω	angular velocity vector

SUBSCRIPTS

B	bump stop
f	front axle
G	gravity
I	global
r	rear axle
R	roll
s	sprung mass
S	suspension
X	longitudinal
Y	lateral
Z	vertical

Note that the notation system adopted for vectors is, in full,

$$A_{cd}^b$$

where A is a vector in three dimensions, relating a property (such as position, velocity or acceleration) of point b relative to point c, expressed in axis system d. If d is not present, the vector is expressed in coordinates for the axis set c.

A^b is an abbreviated form which signifies the property A of axis origin b relative to the inertial axis set and expressed in axis set b.

CHAPTER 1

INTRODUCTION

1.1 EXAMINATION OF BUS ACCIDENT STATISTICS

Severe coach accidents receive significant media attention. Consequently, Government authorities are constrained by public pressure to act to reduce both the number and severity of coach accidents. Accident statistics are examined to determine the priority areas in which research activity should be focussed in order to effect the greatest potential improvement to bus safety.

1.1.1 Different Types of Bus Accidents

Table 1.1 uses data provided by the Vehicle Inspectorate branch of the Department of Transport from 358 coach accidents during the period 1982-1985. It is apparent that while frontal impacts account for the majority of recorded accidents, rollovers, which make up less than 9% of accidents, cause over 40% of the fatalities and serious injuries. This implies that overturning accidents should be the highest priority for bus safety research.

Table 1.1
Categories of Coach Accident Reports

Major Impact	Proportion of Accidents	Proportion of Injuries		
		Fatal	Serious	Slight
Front	60.9	23.8	27.1	53.4
Side	8.7	22.6	19.6	4.3
Rear	6.4	8.3	1.9	1.3
Not Known *	15.1	3.6	7.5	2.7
Rollover	8.9	41.6	43.9	38.3
Total Numbers	358	84	321	1093

* "Not Known" comprise non-impact accidents

1.1.2 Comparison with Accident Statistics from Some Other Countries

Published data about bus and coach accidents tends to be very general. Only a few recent studies targeted specifically at public service vehicle (abbreviated to PSV) accidents provide suitably detailed information.

1.1.2.1 West Germany

A random sample of 239 bus and coach injury accidents since 1978 was examined by Langwieder, et al (1985) for German motor insurers. In their study, 85% of fatalities and 50% of serious injuries occurred in rollover accidents, which comprised only 7% of the total. This reinforces the U.K. figures above.

Safety belts (preferably three-point belts) to prevent ejection, in conjunction with adequate roof strength to prevent excessive intrusion, are countermeasures recommended by the investigators. Anti-lock brakes are predicted to reduce accident severity in about 40% of the collisions.

1.1.2.2 France

A study of 48 fatal coach accidents in France (reported by Thomas, et al, 1985) over the period 1978-1984 shows an essentially similar trend. The main recommendations resulting from the investigations were to:

- (1) improve braking performance,
- (2) improve protection in head-on and offset collisions,
- (3) improve seat back design to provide passive restraint,
- (4) investigate means to minimise ejection without reducing emergency exits,
- (5) reduce smoke and fumes generated from the combustion of interior fabrics, and
- (6) work together with other countries to compare data, and assess feasibility of safety measures.

Safety belts were rejected as a candidate restraint method.

1.1.2.3 Eastern Europe

The proportions of accident types are different in eastern Europe. Buses are used predominantly for public transport in urban areas.

Steiner (1984) of Czechoslovakia reported 23 fatalities in 2028 accidents involving buses (32000 registered) in 1982. However, his breakdown into accident categories only indicates collision directions, with no mention of rollovers occurring.

Brief details of nineteen overturning accidents in Hungary between 1970 and 1977 are given by Matolcsy (1977), but analysis of bus accidents between 1978 and 1982 in the same country (Batiz, 1984) indicates that rollover accidents are less frequent and less significant for traffic casualties than collisions with cars and small trucks.

1.1.3 Effect of Rollover Angle on Injury

Table 1.2 indicates the effect on occupant casualties of increasing roll angle in overturning accidents, and, remembering the different bases for choosing accidents to investigate, compares PSV rollover accident data for three countries.

Table 1.2
Effect on Injuries of Rollover Angle

Rollover Angle	Number of Accidents			Number of Injuries					
				Fatal			Serious		
	F	WG	UK	F	WG	UK	F	WG	UK
90	6	3	15	12	7	8	38	34	50
>90	10	4	17	18	27	27	66	80	91

F = France, WG = West Germany, UK = United Kingdom

It is clear from this table that accident severity generally rises with total bus rollover angle.

1.1.4 Coach Safety Compared with other Transport Modes

Table 1.3 indicates the casualty rates for various modes of transport and countries. U.K. statistics are derived from Table P of 'Road Accidents Great Britain 1982' (Department of Transport, 1983). West German data is from their Federal Ministry of Transport (Der Bundesminister für Verkehr, 1985) as referred to by Langwieder, et al (1985). French sources are as previously mentioned.

Table 1.3
Comparison of Casualty Rates for Various Means of Transport

Transport Mode	User Fatalities/10 ⁹ passenger km		
	West Germany	France	United Kingdom
Ship	1.19	..	..
Aeroplane	1.35	..	..
Bus/Coach	1.9	0.5	0.57
Train	5.6	..	0.54
Truck	..	6	..
Car	20	16	5.5
Moped/ Motorcycle	141	100	130

Clearly, in all three countries, the relative risk of travelling in a PSV is significantly lower than for other forms of road transport. Bus travel is approximately ten times safer than travelling by car (30 times safer in France), and from 70 to over 200 times safer than riding on a two-wheeled motor vehicle.

It is an interesting reflection on modern society that a single severe coach accident is regarded as tragic news, yet the vastly greater number of deaths in other road accidents are virtually accepted as normal. There are a few differences, which only partly explain this observation: PSV passengers are often elderly or below school-leaving age, dependent entirely on the skills of the scheduled driver, and thus in an accident are regarded as innocent victims, whilst other road users have greater control over their own safety (being more likely to be a driver).

1.2 POSSIBLE AREAS FOR ACTION TO IMPROVE BUS ROLLOVER SAFETY

An extensive investigation conducted into 29 coach rollover accidents in the UK for the Department of Transport between 1975 and 1979 (reported briefly by Tidbury, 1977, Keman, et al, 1979 and Keman, 1979) gives a clear insight into the sequence of events leading up to a rollover. Although the specific details vary between different incidents, key common features have been identified and expressed in Figure 1.1.

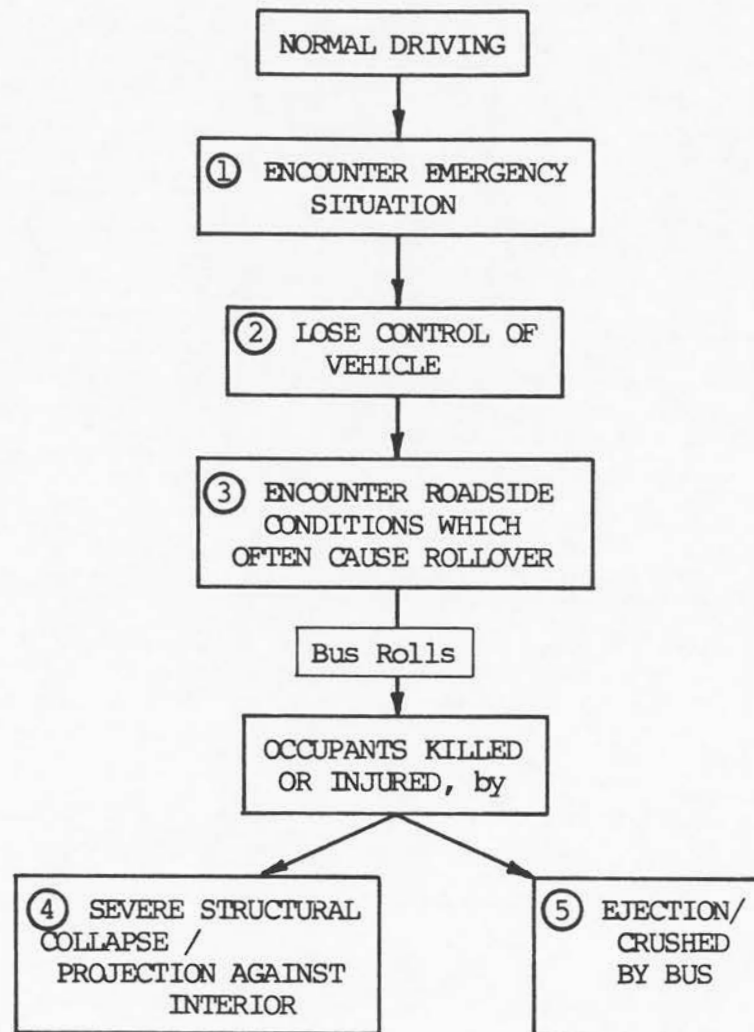


Figure 1.1 Typical Sequence of Events Leading to Bus Rollover Accident and Injuries

Kecman, et al (1981) summarised the typical rollover motion by stating that,

'in more than 75% of accidents the vehicle rolled purely sideways on level ground or into a depression up to one metre deep'.

Frequency of bus rollover accidents can be reduced by halting the sequence in Figure 1.1 at steps 1 to 3. Some legislative measures aimed at this area of active safety already exist. Severity of injuries can be lessened using passive safety measures at steps 4 and 5. Each of these aspects are discussed in turn below.

1.2.1 Emergency Situations

Since humans, and animals, are unpredictable, it is unlikely that emergency situations can be eliminated.

Careful design of road layout, especially at road works, can assist. Driver skill, judgement and attentiveness (including that of other road users) is probably the major factor determining the frequency and consequences of emergency situations.

1.2.2 Loss of Control

Loss of control sometimes occurs as the first step of a rollover sequence, if, for example, the driver falls asleep or suffers a sudden, disabling medical condition, or due to a mechanical defect such as brake failure.

Alternatively, control may be lost in an emergency situation due either to collision with another vehicle or object, or due to braking or steering forces in an accident avoidance manoeuvre which exceed available tyre/road friction, resulting in a skid.

Driver training, limitations on working hours, speed restrictions and construction laws, especially regarding tyre quality and braking and steering performance, are positive measures which already exist and aid in maintaining control.

London Regional Transport includes a session on a skid pad in a double-decker bus as part of its training course for new drivers. In view of the British winter climate and the importance of skid control in rollover accidents (a vehicle must be skidding if it is moving sideways), perhaps consideration could be given to making this aspect of training a mandatory pre-requisite before a PSV licence test is taken, or the skid-recovery skill be examined during the test.

Although many handling tests have been conducted for cars, no published work is known to the author comparing the performance limits (that is, controllability in extreme manoeuvres) for different buses and coaches. Weir, et al (1974) tested and simulated the handling of a truck and intercity bus. Preliminary full-scale tests on a range of PSVs would determine the variation in limit performance in several manoeuvres (such as, braking in a turn, and high-speed obstacle avoidance), and thus indicate whether this area is worth researching further, to ultimately improve the controllability of vehicles in emergency manoeuvres.

1.2.3 Roadside Rollover Conditions

Once a vehicle has lost control and is sliding sideways towards the edge of the road, the probability is high that the tyres will either impact a kerb or low wall, or plough into a soft verge. In fact, a survey of 18 British overturning accidents (Structural Design Group, 1979) revealed that the actual means by which the rollover torque was applied was one of these three "tripping" modes in over 60% of the cases, as shown in Figure 1.2.

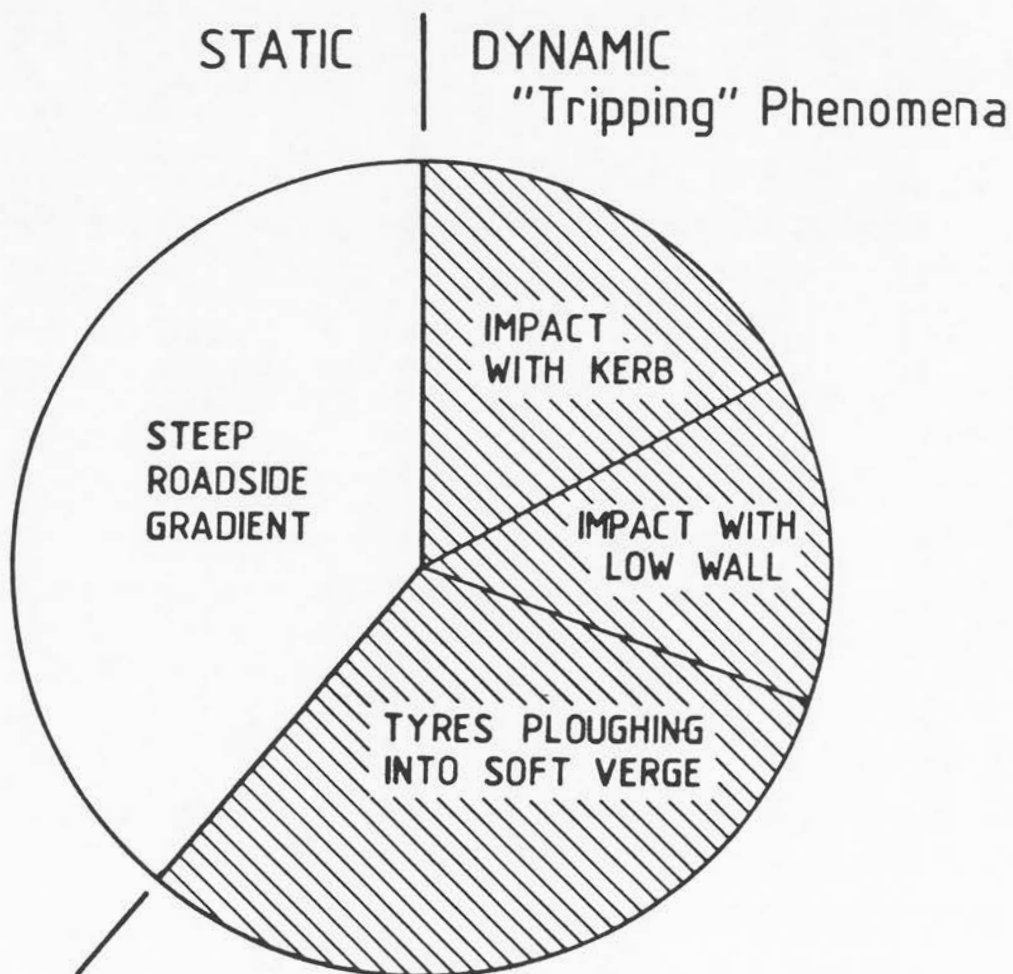


Figure 1.2 Phenomena Applying Bus Rollover Moments

Whether or not a vehicle rolls over in this situation is not easy to predict. Britain has one piece of legislation intended to set minimum standards for rollover stability which dates from 1931. The tilt test has, however, been criticised as bearing little resemblance to reality and being an unproven indicator of rollover stability. Danskin (1973) describes the tilt test as,

'a severe test, stretching at times the ingenuity of the designer, but having once passed the test the vehicle is presumed to be superior to one that fails. There are no statistics or evidence to clearly indicate that a vehicle not subjected to this test is inferior.'

Research is needed to determine whether the tilt test serves a useful purpose, or, widening the terms of reference further, whether there exists a need for an overturning stability test at all, and if so, what the best methods are for testing rollover stability.

1.2.4 Causes of Occupant Injuries

Occupant casualties occur during and after the actual rollover phase, due mainly to distinct causes, identified in Figure 1.3.

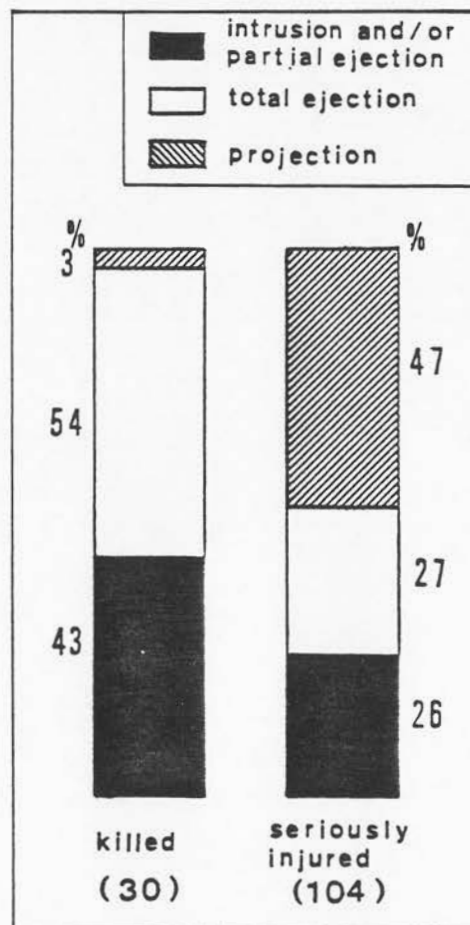


Figure 1.3 Injury Mechanisms in Overturning Accidents
(from Thomas, et al, 1985)

1.2.4.1 Structural Collapse/ Projection Against Interior

Structural collapse (labelled "intrusion" in Figure 1.3) is a serious problem identified in the last fifteen years. Figures 1.4 and 1.5 highlight two accidents in which the roof collapse is significant.

In 1978 the coach in Figure 1.4 suffered brake failure on a winding downhill road near Buxton, Derbyshire, and ran off the road until it halted on a 45° bank from which it overturned back onto the road. The steel/wood composite superstructure obviously collapsed onto the seat backs. Six of the 48 pensioners died (3 on the scene, 3 others within two weeks) and all occupants suffered injuries to some degree.

Fortunately, the casualties among the twenty or so band members were limited to a few slight injuries when the coach of steel construction in Figure 1.5 (courtesy of the Dundee Courier newspaper) rolled near Perth, Tayside, in 1976. The accident happened because the driver was unable to stop behind a queue of cars waiting at temporary traffic lights situated just beyond a downhill right hand curve. To avoid the cars, he steered towards the nearside verge, demolishing a stone wall and overturning onto level ground approximately two metres lower than the road.

To improve the strength and energy absorbing properties of coach superstructures, a standard test which specifies a minimum survival space is urgently required. Research into the suitability of proposed test methods has been continuing. Meanwhile, manufacturers have already been improving their structural designs for added rollover protection.

1.2.4.2 Passenger Ejection/ Crushed by Bus

The final major factor contributing to passenger casualties (Figure 1.1) identified in each of the accident studies in France, West Germany and Great Britain referred to earlier, is the phenomenon of ejection. Fifteen of the 35 coach rollover fatalities (43%) between 1982 and 1985 in Britain were occupants who had been ejected. The corresponding proportions for France and West Germany are 54% and 47% respectively. Ejection (and the almost inevitable crushing by the vehicle as it continues rolling) has been recognised as a major problem in all three countries.

Proposed methods of preventing passengers from being thrown out through skylights or windows have been suggested by different parties, but, lacking experimental evidence, these suggestions are open to question. In fact, recommendations concerning the implementation of seat belts in the French and West German reports are contradictory. Further research is needed to determine practical, acceptable and economic solutions.



Figure 1.4 Coach Rollover Accident near Buxton, Derbyshire (1978)



Figure 1.5 Coach Rollover Accident near Perth, Tayside (1976)

1.3 SELECTED AREAS FOR STUDY

Contributions to two of the outlined areas still requiring research (summarised in Section 9.1) are made with this work:

Firstly, a small advance is made in the fundamental knowledge concerning a test method proposed to reduce the occurrence of structural collapse in overturning accidents (Chapter 2).

Secondly, a major study of the rollover propensity of buses and coaches has been undertaken and is described in Chapters 3 to 8.

CHAPTER 2

SIMULATION OF DYNAMICS OF BUS
ROLLOVER TEST

2.1 INTRODUCTION

Two of the findings of the accident data analysis in Chapter 1 are:

- (1) the importance of adequate structural protection in a rollover accident, to preserve a specified survival space for the occupants, and
- (2) the need for a type-approval test to ensure that new PSVs meet a certain roof-strength standard.

Consultations, beginning in the early 1970s, were held under the auspices of the United Nations' Economic Commission for Europe (ECE) in a section known as GRSA (a French abbreviation for Group of Rapporteurs on Safety Provisions on Motor Coaches and Buses) to establish agreed provisions on superstructure strength regulations. Agreement was reached in 1985.

One method for testing roof strength was proposed by the British representative to GRSA. The suggested method was devised by the Structural Design Group at Cranfield, based on the typical overturning accident found in the accident investigations performed by the same group (see Section 1.2). Construction of the rig and execution of the full-scale tests was undertaken at the Motor Industry Research Association at Nuneaton.

2.2 TEST PROCEDURE AND PROBLEMS

2.2.1 Original Test Procedure

A test vehicle is slid sideways on a trolley down a ramp to strike a simulated "kerb" and roll over into the ditch at the bottom of the ramp (Figures 2.1 to 2.3). The intention is that the bus should land on its cantrail with certain linear and angular velocities, and therefore with a specific level of kinetic energy, in order to test the rollover protection of the vehicle's superstructure.

2.2.2 Problems Encountered

In practice, however, the first test vehicle had its suspension members grossly damaged during impact with the kerb. The impact force had to be reduced, and this was done in two ways: firstly, by lowering the kerb stiffness, using springs mounted on the impact face of the kerb, and secondly, by including an array of



Figure 2.1 Before Superstructure Strength Test



Figure 2.2 After Superstructure Strength Test

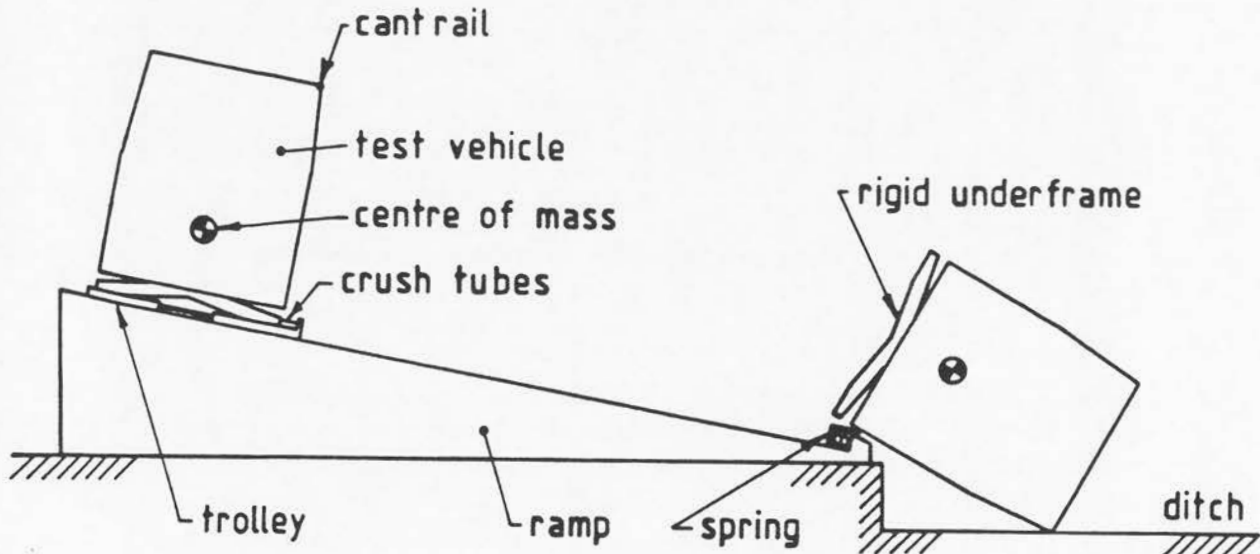


Figure 2.3 Schematic Diagram of Superstructure Strength Test Rig

steel "crush tubes" between each spring and the bus, which were designed to collapse at a known force limit. The wheels, axles and suspensions were removed from the test vehicle and replaced with rigid steel underframes of the appropriate mass distribution which were welded directly to the chassis to minimise the possibility of structural failure during the test (while at the same time, eliminating one source of unrepeatability - that of uncontrolled axle motion). A sled supported by air bearings superseded the wheeled trolley, giving better control of impact speed. With two leaf-spring overturning mechanisms of equal stiffness, satisfactory rollover tests were performed on mid-engined coaches (Figure 2.2).

If springs of equal stiffness are used to roll a vehicle with an asymmetric axle weight distribution (such as most rear-engined buses) the resultant yaw will normally cause the vehicle's cant-rail to contact at one end first (which is inappropriate for testing roof strength), rather than the desired uniform contact along the entire cant-rail. Air springs were thus introduced to provide readily adjustable spring stiffness (by changing the air pressure) compared with the leaf springs. However, the air spring pressures affect not only the motion of the test vehicle, but also its velocities and kinetic energy at the point of impact with the ground.

2.3 COMPUTER SIMULATION

A computer program, SIDBURT (SIMulation of the Dynamics of a BUS Rollover Test), was therefore developed to provide further insight into the vehicle motion, and particularly, to predict the air spring pressures which would satisfactorily roll a given vehicle.

SIDBURT is a three-dimensional digital simulation of the rigid body motion of a bus during a U.K.-proposed rollover test. Openshaw (1981) produced the initial computer model as part of an MSc thesis. The general layout of the simulation model is presented in Figure 2.4.

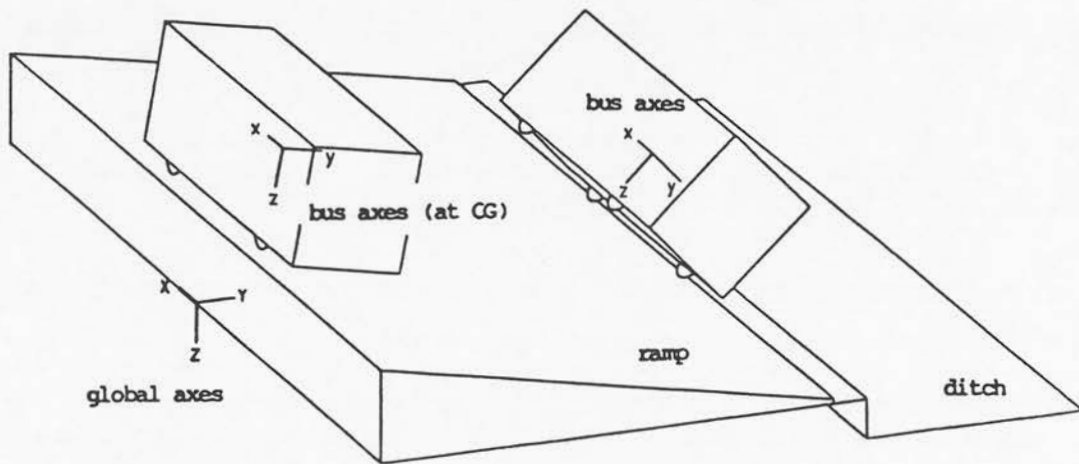


Figure 2.4 General Layout of Rollover Simulation Model

2.3.1 Features

While the aim was for two-dimensional rollover motion, yaw, pitch and longitudinal degrees of freedom are included to indicate whether significant displacement in these directions occurred. Figure 2.5 shows a summarised version of the flow diagram.

Apart from the gravity force, which is applied at the centre of mass, each of the other forces acting on the vehicle is only present if the vehicle is in contact with another object, such as the ramp surface, the overturning springs, the spring box or the edge of the ditch. Contact is determined by checking whether or not the point in question on the bus has intruded into a defined space, such as that occupied by the ramp or springs. If contact exists, the contact force is evaluated by firstly calculating the intrusion distance (compression) by geometry, and the relative velocity of the point concerned. The stiffness and damping

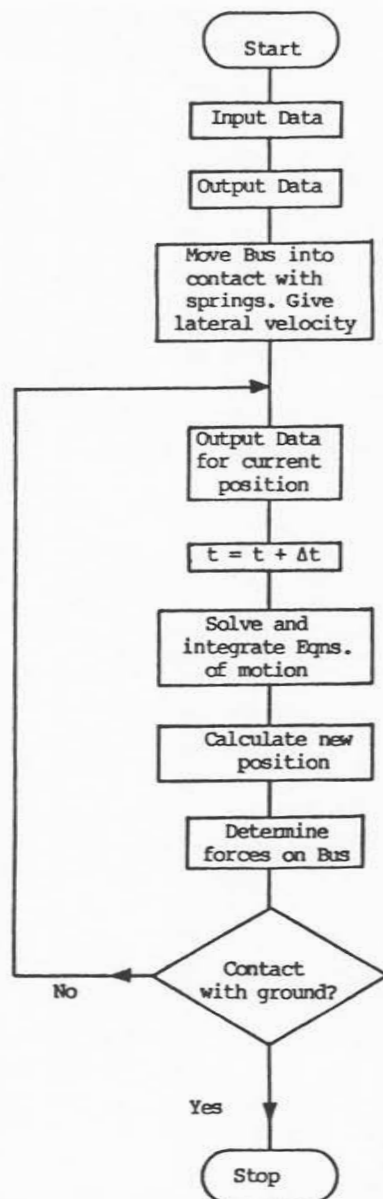


Figure 2.5 Simplified Flow Diagram of SIDBURT

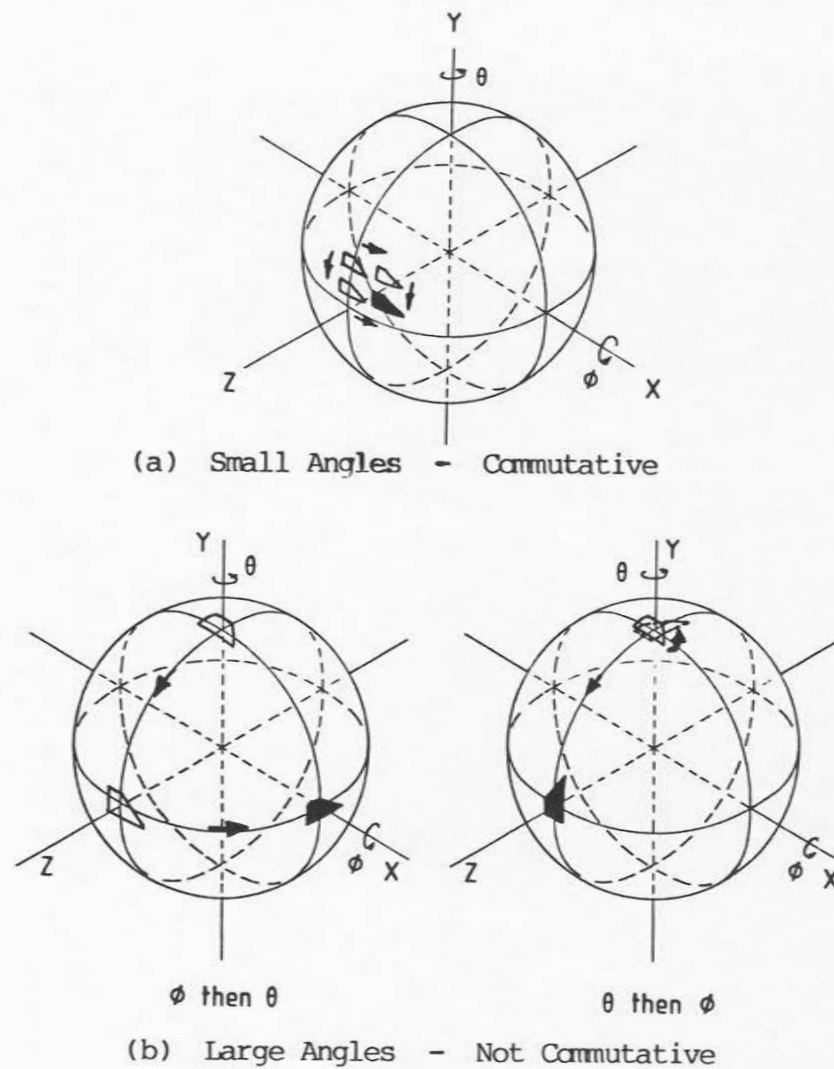


Figure 2.6 Comparison of Addition of Small and Large Angles

relationships for each respective surface are then used to compute the contact force, which is subsequently resolved into components of force in the bus axis system acting at the centre of mass, and torque components. Structural collapse is not simulated - the program stops when the vehicle hits the bottom of the ditch.

Equations of motion are the general moment equations for three-dimensional motion of a rigid body in space, with the reference axes attached to the body (Meriam, 1980).

Roll, pitch and yaw angles are used to transform position and velocity from the bus axis system to the global axis system (attached to the ground) and vice versa. The transformation matrix and method for updating vehicle rotations are obtained from aircraft stability and control theory (Babister, 1980, and derived in Section 4.5.2 for the Rollover Accident Simulation Program). Openshaw incorrectly assumed that since integration time steps are small, then rotation angles are small and can thus be regarded as commutative. Figure 2.6 demonstrates that combinations of small angles are commutative, whereas large angles are not. It is true that angle increments are small, but the axis transformation matrix requires the total angles to convert from global axes to bus axes.

The integration subroutine is based on the Runge-Kutta fourth order method.

2.3.2 Assumptions

- (1) The vehicle is represented as an infinitely rigid body and thus has no torsional or flexural flexibility.
- (2) Static and sliding friction between surfaces in contact are ignored.
- (3) The influence of the sled and spring boxes are neglected.
- (4) The Force v Deflection curve is represented by a straight line for leaf springs, and by a cubic equation for air springs, obtained from measurements taken during static tests. These curves are assumed to remain valid for dynamic response also.
- (5) The collapse behaviour of the crush tubes is approximated by a constant collapse force (Figure 2.7).

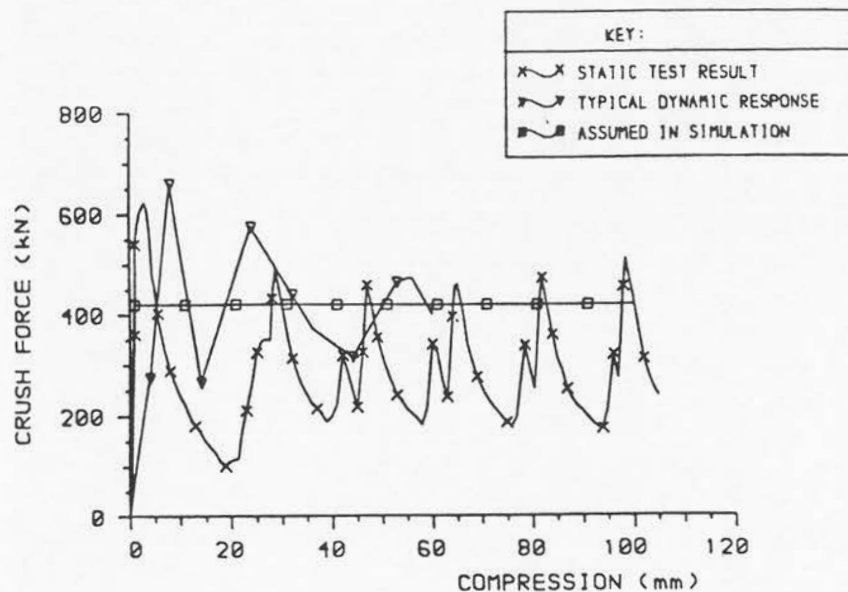


Figure 2.7 Collapse Behaviour of Crush Tubes

2.3.3 Input Data

The input data is divided into five groups, to allow quick alteration of most parameters:

- (1) Initial Data - The initial position and orientation of the vehicle, together with its lateral speed at the moment of impact with the overturning mechanisms, are specified, as well as the integration parameters and output data requirements.
- (2) Vehicle Data - The vehicle is modelled as a rigid body of given mass, inertial properties and exterior dimensions.
- (3) Ramp Data - Coordinates define the ramp position, the slope is given (the default value is 11 degrees) and the ramp surface is represented by high stiffness springs (and dampers, if required).
- (4) Spring Data - "Kerb" spring stiffness is expressed as an algebraic relationship between force and displacement. Position, orientation and travel of each spring are required, together with the crush tube length and the collapse force.

- (5) **Airspring Coefficients** - To simplify changes to spring stiffness for different air spring pressures a look-up table of cubic stiffness coefficients is provided for air pressures from 0 - 100 psi (0 - 690kPa).

2.3.4 Printed Output

Up to six different output files may be produced, and the time interval between successive data is also selectable (otherwise default values are used):

- (1) **Main Output** - The main output file reproduces the input data and records the elapsed time, position, orientation, and linear and angular velocities of the vehicle centre of mass throughout the rollover simulation. (Note that time is measured from the instant of contact with the overturning springs). Also the total linear and rotational kinetic energies and the overall energy checksum are given. An alphanumeric code indicates at each recorded step with which surfaces (ramp, springs or none) the bus is in contact.
- (2) **Spring Output** - The contact force, spring displacement and velocity, collapse distance of the crush tubes, and the total stored or absorbed energy of each overturning spring are printed. To save computer storage space, data is recorded in this file (and the next one) only if contact exists.
- (3) **Ramp Output** - Similarly, for the contact between the bus and the ramp, the force, displacement, velocity and stored energy are noted for each point in contact with the ramp.
- (4) **Contact Output** - As a precise check on the behaviour of the points in contact with the overturning mechanisms, their positions and velocities expressed relative to the ramp surface can be printed.
- (5) **Force Output** - Forces and moments acting in three co-ordinate directions at the vehicle centre of mass are given alongside a detailed breakdown of the components of the energy checksum.

- (6) Graphics Data - The graphics data file contains basic bus and ramp data and the bus position and direction matrix at each small time step, available for graphical presentation of the motion of the vehicle by a post-processor plotting program.

2.3.5 Graphical Output

Two interactive post-processing programs have been written to express the tabular files in graphical form, quickly summarising many pages of numbers into easily assimilated relationships and trends. These programs are separate from the simulation program so that the latter may be run in batch mode (though a typical run requires only 60-70 seconds of computer time).

One program produces perspective views of the three-dimensional vehicle motion with user-definable viewing position and scale and limited hidden-line removal capability. Figures 2.8 and 2.9 show typical perspective drawings.

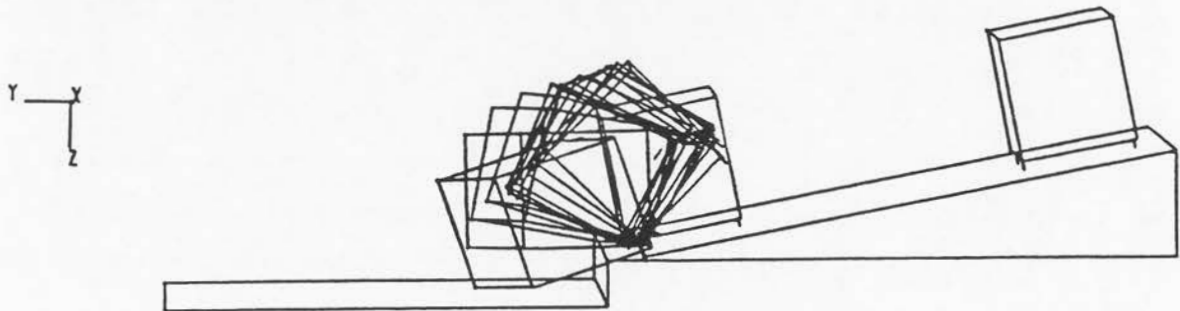


Figure 2.8 Perspective View of SIDBURT run for Leyland National

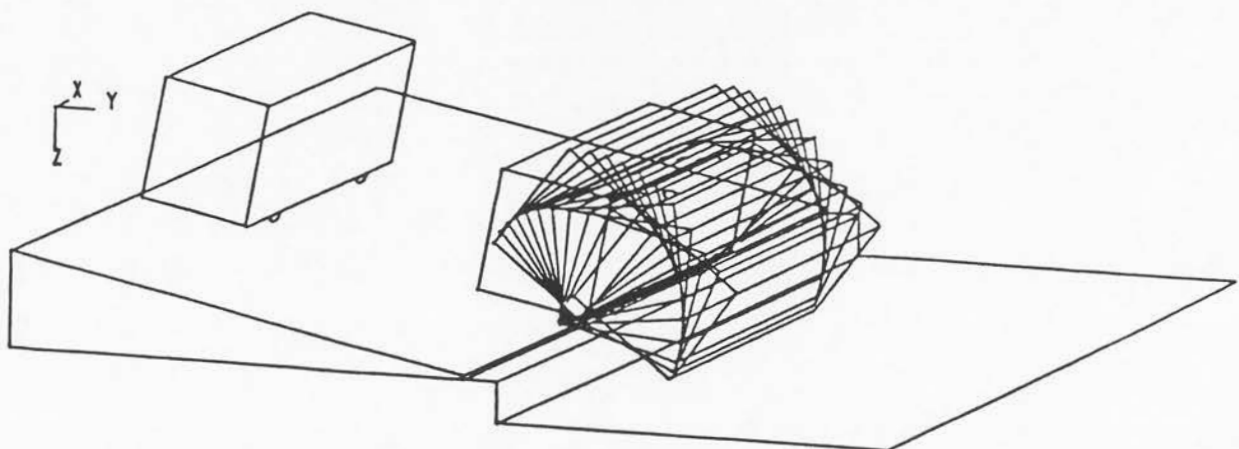


Figure 2.9 Perspective View of SIDBURT run for Plaxton Elite

The other program plots graphs of various variables, such as spring or ramp contact forces or time histories of the position of any point on the vehicle. Examples are shown in Figure 2.10.

2.4 COMPARISON WITH FULL-SCALE TESTS

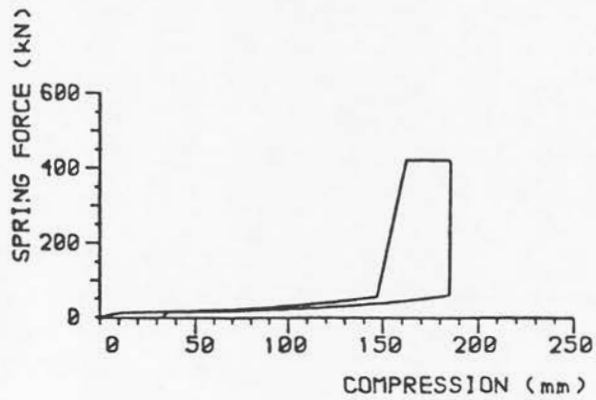
2.4.1 Initial Conditions

As part of a research programme undertaken for the Department of Transport (reported by the Structural Design Group), a number of vehicles were rolled using the rig described earlier. Two such tests are compared with computer simulations: one of a Leyland National integral bus with a rear engine and the other a Plaxton Elite coach on a Ford chassis with a symmetric axle weight distribution. Table 2.1 lists the relevant vehicle parameters and test conditions. This information is used as input data for the simulations.

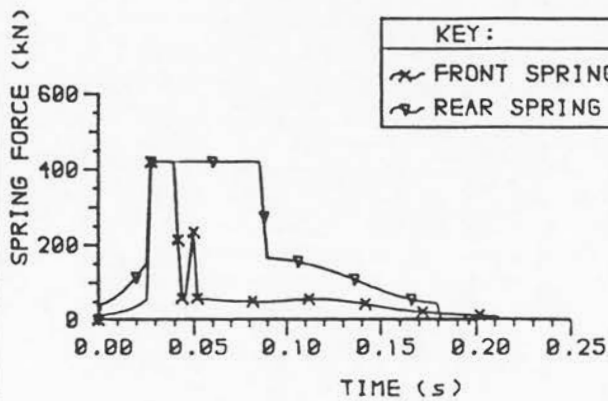
Table 2.1
Vehicle Data and Rollover Test Conditions

Parameter	Test Vehicle	
	Leyland National	Plaxton Elite.
Mass (kg):		
Front	2460	3305
Rear	6020	3338
Total	8480	6643
Ratio (Front:Rear)	29:71	50:50
Roll Moment of Inertia (kg.m^2)	9200	5850
CG Height (m)	0.88	1.14
Spring Stiffness:		
Type	air	leaf
Front	20 psi (140kPa)	280 (kN/m)
Rear	75 psi (520kPa)	280 (kN/m)
Spring Travel (m)	0.152	0.127
Speed at Impact with Springs (m/s)	5.60	5.38

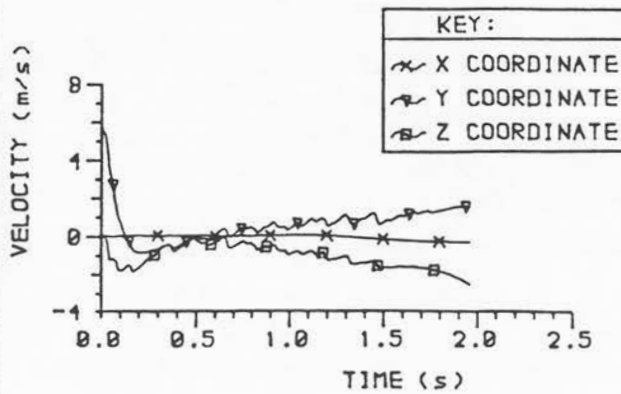
LEYLAND NATIONAL



(a) FORCE v COMPRESSION
at front spring

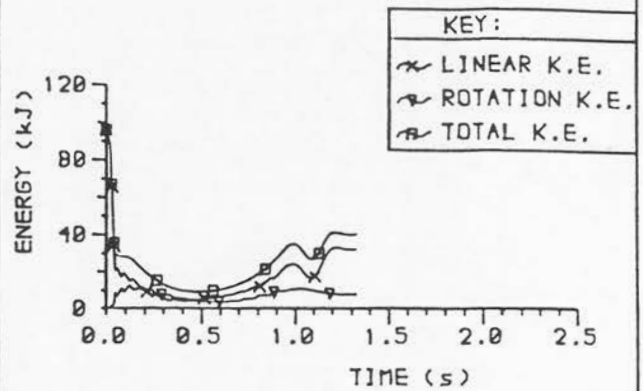


(b) FORCE v TIME
at both springs

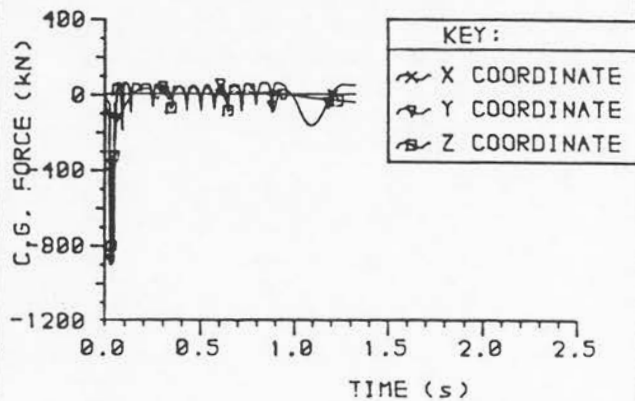


(c) VELOCITIES v TIME
(Bus Axis System)

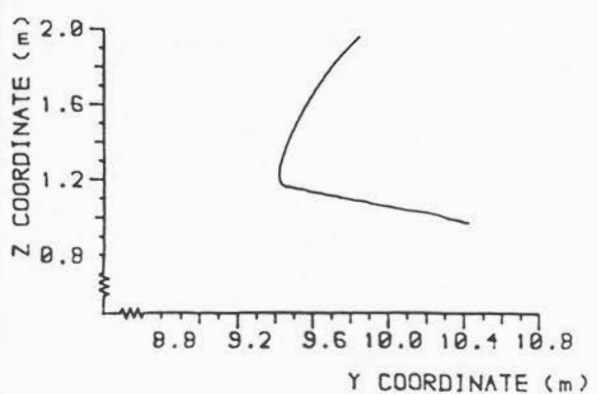
PLAXTON ELITE



(d) ENERGY COMPONENTS v TIME



(e) C.G. FORCES v TIME
(Bus Axis System)



(f) TRAJECTORY OF BUS NODE
(contacts front spring)

Figure 2.10 Typical Graphical Output from SIDBURT

2.4.2 Results

Figure 2.11 provides graphical expressions of the resultant motion of each vehicle as measured from high speed cine film in the tests, compared with the motion predicted by the computer simulations. (The Y coordinate is the lateral displacement towards the ditch, and the Z coordinate is the vertical displacement, expressed for clarity as positive upwards.)

Details of the global components of velocity at the moment of impact with the ground are given in Table 2.2, with energy components and performance of the crush tubes.

Table 2.2
Comparison of Bus Rollover Tests and Simulation

Parameters	Test Vehicle			
	Leyland National		Plaxton Elite	
	test	simulation	test	simulation
Crush Tubes				
Front crush (mm)	50.	33.1	5.	62.8
energy (kJ)	14.8	14.7	2.6	28.9
Rear crush (mm)	50.	186.0	2.	62.7
energy (kJ)	14.8	77.8	0.8	29.0
Total Crush Energy (kJ)	29.6	92.5	3.4	57.9
CG Velocities (global)				
X (m/s)	..	-0.04	..	0.01
Y (m/s)	1.68	1.95	2.55	3.09
Z (m/s)	1.56	2.25	0.39	0.29
Roll (rad/s)	1.71	2.08	1.22	1.64
Kinetic Energies (kJ)				
X	..	0.0	..	0.0
Y	12.2	16.1	22.0	31.7
Z	10.6	21.4	0.4	0.3
Roll	13.5	19.8	4.3	7.8
Pitch	0.0	0.1	0.0	0.0
Yaw	0.0	0.9	0.0	0.0
Total	36.3	58.1	26.7	39.8

LEYLAND
NATIONAL

KEY:

— SIMULATION
x x x x ROLL TEST

PLAXTON
ELITE

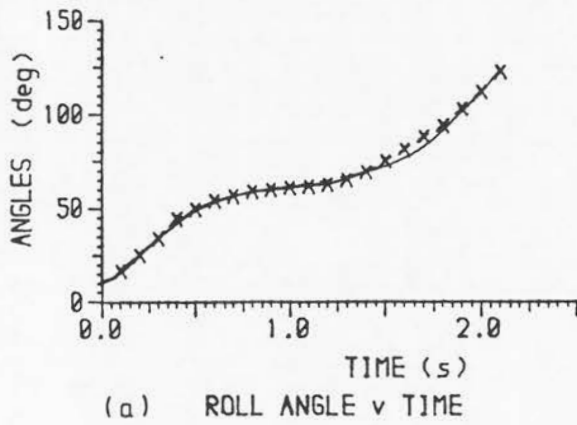
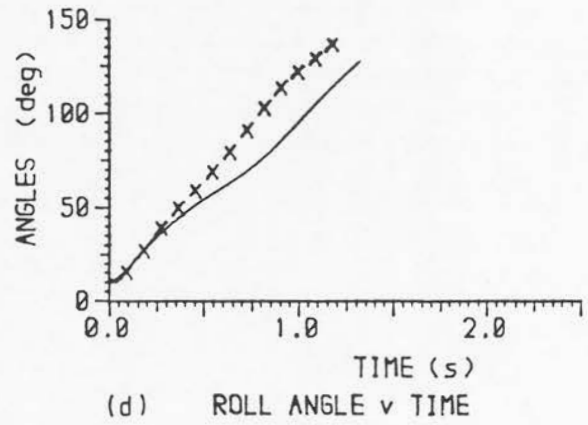
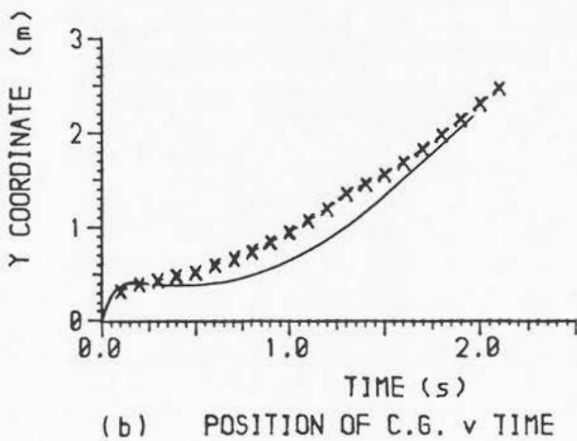
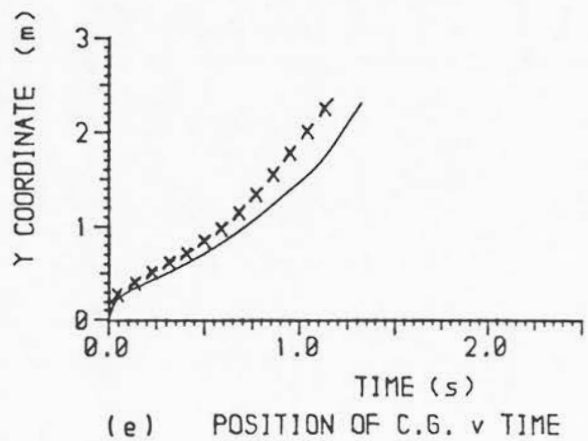
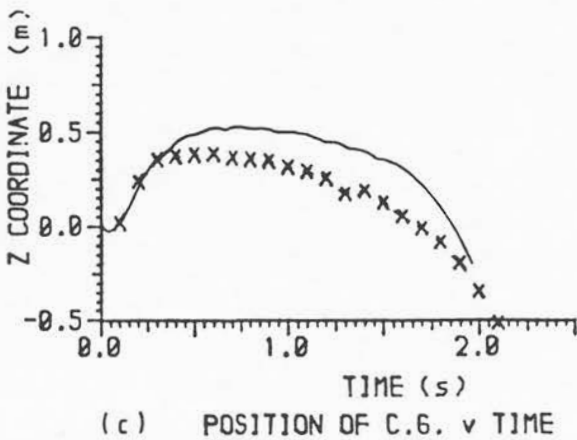
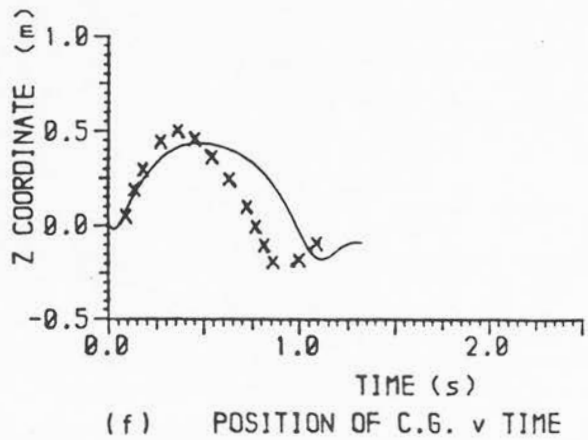


Figure 2.11 Comparison of Rollover Test and Simulation Results

2.4.2 Discussion of Results

2.4.2.1 Energy Comparison

From Table 2.2 it is apparent that the energies given by each simulation do not correspond very well with the test measurements. In particular, the energy absorbed by the crush tubes is far greater in the simulation than in reality. Table 2.3 shows the energy each vehicle possessed at the impact with the overturning springs and the energy at impact with the bottom of the ditch. It is obvious from the large discrepancy between the energy sums before and after the impact for the rollover tests, that a large proportion must have been dissipated, for example, into the ramp support structure, spring boxes, by deformation of the bus and its underframes (the latter visible as damage after repeated tests) and by movement of the bus-carrying sled. (In some rollover tests, as the bus rolled, the sled was observed to tip and scrape along the ramp surface). Most of this extraneous energy dissipation would occur during the impact with the overturning springs, and the remainder during contact with the spring boxes or the edge of the ditch.

Table 2.3
Energy Comparison Before and After Spring Impact

	Test Vehicle			
	Leyland National		Plaxton Elite	
	test (kJ)	simulation (kJ)	test (kJ)	simulation (kJ)
At impact with springs: Kinetic Energy	137.8	137.8	96.1	96.1
At impact with ground:				
Kinetic Energy	36.4	58.1	26.8	39.8
Crush Energy	29.6	92.5	3.4	57.9
* Potential Energy	-43.3	-17.4	-10.6	-6.5
Total Energy	22.7	133.2	19.6	91.2
Balance: "Other" Energy	115.1	4.6	76.5	4.9

* Potential energy has been defined as zero when the vehicle first contacts the overturning springs.

The simulation, however, has only one major mode of energy absorption: the crush tubes. (A small amount will be absorbed by the contact surfaces if a non-zero damping coefficient is specified). Thus the crush tubes in the simulation are equivalent to the crush tubes in the test plus most of the "other" modes of energy absorption listed above. So the large crush distances and energy absorption are to be expected in the simulation.

Another factor contributing to the differences between test and simulation is the accuracy of the measurements of the vehicle behaviour during the rollover tests. In particular, while the analysis of high-speed cine film produces adequate displacement data, errors are magnified when differentiating the results to obtain velocities, and then squared when calculating energies. In addition, any initial yaw of the vehicle (even as little as one degree) as it slides down the ramp, has a significant effect on the subsequent motion.

Considering these differences between rollover test and computer model, the simulation performs reasonably well to predict kinetic energies at impact with the ground which are not too different from the test results.

2.4.2.2 Trajectory Comparison

The graphs in Figure 2.11 show acceptable correlation between the trajectories as predicted by the computer model and the results obtained from the rollover test. In each case the general trend of the curve is correct, even in Figure 2.11f which has an interesting feature: the small hump towards the end of rollover of the Plaxton Elite is due to the vehicle rolling on its side over the spring box/ top of ditch, and it is also visible in the perspective view (Figure 2.9).

In the similar perspective drawing for the Leyland National (Figure 2.8), a more complex motion is evident by the small yaw and pitch oscillations. In reality this extra movement is partly absorbed by the torsional and flexural deflection of the vehicle structure and is not as noticeable.

2.5 OUTCOME OF TEST PROPOSAL

The proposed superstructure strength test method is realistic, producing deformations and collapse modes in test vehicles which closely resemble those observed in rollover accidents of vehicles of similar construction (Tidbury, 1984). By implication, the cantrail impact energy must correspond reasonably well with the typical level of energy a roof structure should be capable of absorbing. The energy requirement varies with the unladen mass, centre of gravity height, roll moment of inertia and cantrail height of the test vehicle. It was found in previous work

commissioned by the Department of Transport that it was possible to simplify this to a relationship between energy and mass only (Figure 2.12). Thus Figure 2.12 provides a reasonable indication of the energy at cantrail impact for a typical coach overturning accident in Great Britain.

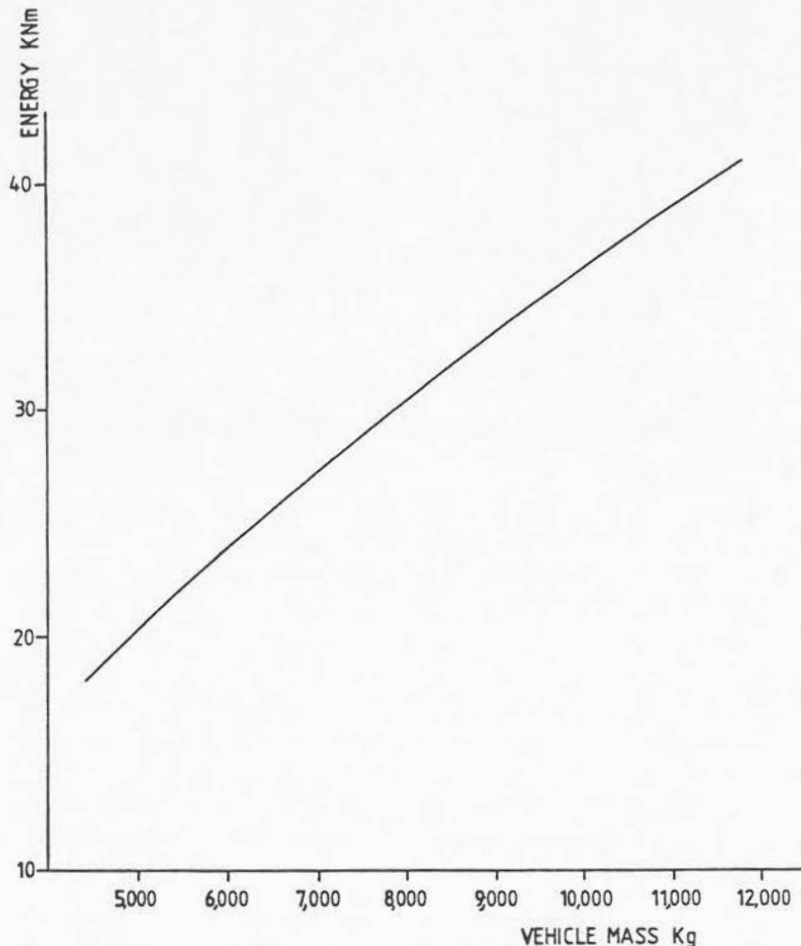


Figure 2.12 Minimum Energy Absorbing Potential of Coach Structures

However, despite the verification of the full-scale test by comparison with component testing (impacting one ring or a bay with a pendulum, Keenan, et al, 1981) and theoretical analysis, its disadvantages were regarded as too serious, and the method lost favour as a legislative test proposal. The criticism was addressed mainly at the unrepeatability introduced by the overturning mechanism, due to its largely unknown energy dissipation during the rollover tests.

The actual test method agreed by GRSA is to tilt a coach slowly on a platform positioned parallel with a ditch, until the vehicle reaches its point of balance and rolls over, striking its cantrail on the bottom of the ditch.

CHAPTER 3

VEHICLE DATA MEASUREMENT

3.1 INTRODUCTION

The approach adopted for the investigation of rollover propensity (given "tripping" roadside conditions discussed in Section 1.2.3) was to measure relevant properties of a range of buses and coaches, including their tilt performance, and develop a kerb-impact rollover simulation with which to compare the vehicles and conduct a parametric study.

3.2 DESCRIPTIONS OF TEST VEHICLES

Nine different Public Service Vehicles (PSVs) were tested. Two of these were especially imported from the Continent to examine models which are not designed to meet U.K. tilt test regulations. The following are brief descriptions of each model, in the order in which they were tested. (Note that all vehicles have air suspension and a rear-mounted engine except where stated otherwise.)

3.2.1 Leyland National 2

A 1980-81 integral single-decked service bus operated by London Regional Transport, the National has space for 24 seated and 46 standing passengers (Figure 3.1).

3.2.2 MCW Metrobus

Another 1980-81 red London bus, the version tested accommodates 91 passengers: 43 seated upstairs, and 28 seated and 20 standing downstairs (Figure 3.3).

3.2.3 Van Hool Astronomega

Imported from Ramoudt Tours in Belgium, this is a 1984 Continental version of a three-axle double-decker coach (Figure 3.2). Double-wishbone independent suspension with high-mounted air springs are employed for the front and trailing wheels (the latter are also steerable at low speed). Power is provided by a Mercedes Benz engine to the drive axle which is supported on a trailing A-frame. Its air springs are positioned outboard (eliminating the need for an anti-roll bar) and again high up (in fact 0.85 m above the ground at the base of the spring). The upper deck seats 55 passengers, and together with the 27 downstairs, the total capacity is 82 passengers.



Figure 3.1 Leyland National 2



Figure 3.2 Van Hool Astromega



Figure 3.3 MCW Metrobus

3.2.4 Jonckheere Jubilee P799

Klundert Tours from Belgium sent across this "one-and-a-half-decker" 11 metre coach (Figure 3.4). 54 passengers sit upstairs where the seats continue right to the front of the vehicle. The driver and co-driver (or hostess) are seated beneath, with stairs, bunk, toilet and luggage space behind them. The chassis in this case was by DAF.

3.2.5 Plaxton Supreme IV

This older (1978-79) coach from National Bus Company (NBC) has a Leyland Leopard chassis which incorporates parabolic multi-leaf springs with telescopic dampers and a mid-mounted engine (Figure 3.5). There is seating for 57 passengers in the relatively low body (3 metres high overall).

3.2.6 Plaxton Paramount 3200

An unused (1984) 3.2 metre high coach with a DAF SB2300 chassis was tested (Figure 3.6). Seating capacity was 55.

3.2.7 Duple Laser

Bedford Commercial Vehicles provided this new (1984) coach which seats 53 (Figure 3.7). The Bedford YNT chassis has a mid-engine and tapered leaf springs. Hydraulic dampers are fitted front and rear, as are anti-roll bars with diameters of 33 mm and 49 mm respectively. (Note that this coach was tested without the spare wheel fitted).

3.2.8 MCW Metroliner

Another three-axle double-decker coach - this one is 11.7 m long, and has the rearmost axle driven and a dead non-steering axle in front of it. Passenger capacity is 80: 57 upstairs and 23 downstairs, as used by NBC beginning in 1984 (Figure 3.8).

Front axle suspension design consists of inboard air springs mounted directly on the axle, which is located by leading links and a symmetric pair of laterally inclined rods. The dampers are also inboard, and roll stiffness is augmented by a 55 mm diameter anti-roll bar. The centre axle is attached to two trailing hinged leaves and a third link to further resist transverse forces. The leaves are kinked slightly outwards behind the axle in order to place the air springs a little further apart. A shock absorber and air spring at each extremity of a link-located H-frame provides the drive axle suspension.



Figure 3.4 Jonckheere Jubilee P799 /DAF



Figure 3.5 Plaxton Supreme IV /Leyland Leopard



Figure 3.6 Plaxton Paramount 3200 /DAF SB2300



Figure 3.7 Duplex Laser /Bedford YNT



Figure 3.8 MCW Metroliner



Figure 3.9 Leyland Royal Tiger Doyen

3.2.9 Leyland Royal Tiger Doyen

NBC also provided this (1982-83) 50-seater high-floor integral coach (Figure 3.9). It too has an H-frame rear suspension with four air springs and dampers but is located instead by two clamped tapered leaves. Hinged trailing leaves extend behind the front axle to a cross beam to which the panhard rod and, at the extremities, the dampers and air springs are mounted. (Two further shock absorbers are attached directly to the axle.)

3.3 HEIGHT OF CENTRE OF GRAVITY

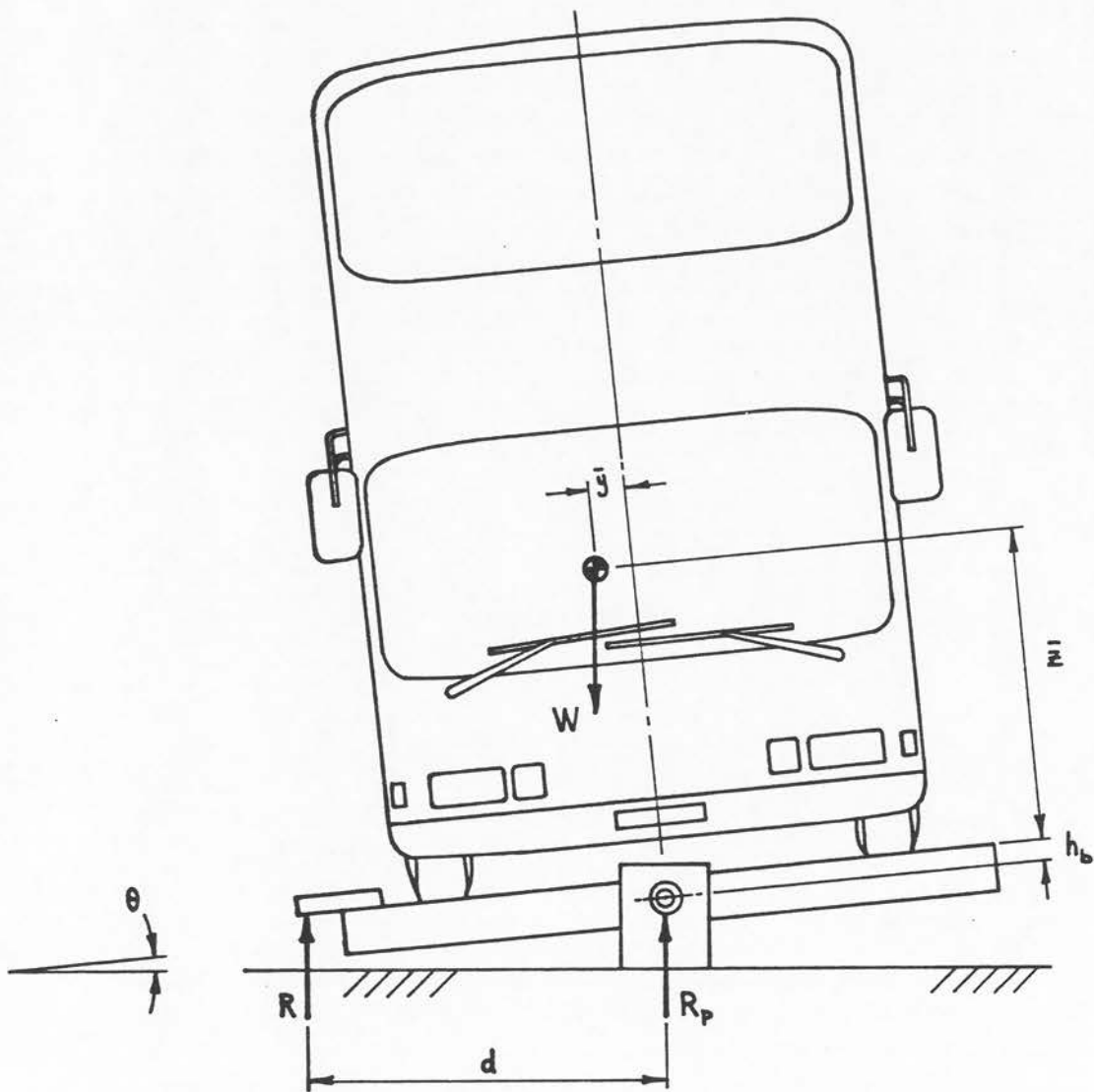
3.3.1 Method

To begin with, the term "centre of gravity" (which should more correctly be the "centre of mass", since it describes a point at which all the mass of a rigid body may be considered to be concentrated) will be abbreviated from now on to "CG". The same method and equipment used for determining the CG height of a vehicle as described by Knight and Tidbury (1981) was adopted. To summarise the technique, the vehicle is positioned with each axle supported from a transverse steel beam which has a central roller bearing, allowing the bus to roll if unrestrained. By measuring the reaction force ('R' in Figure 3.10) for a range of roll angles between 0 and 3 degrees from horizontal, and plotting a graph of these two variables, the CG height was calculated from the equation for static equilibrium of moments about the pivot.

The vertical distance from the top of the beam to the pivot is subtracted to give the unladen CG height above the ground. Note that the suspension and tyres are prevented from deflecting by suitable use of jacks or blocks, effectively converting the bus into a rigid body.

3.3.2 Calculation of Laden CG Height

The CG height of each vehicle was measured in the unladen state, to avoid overloading the test rig. The bus rollover simulations were conducted for vehicles in the laden condition defined by current tilt test regulations, as this is intended to be the "worst case" situation for rollover stability. This means that 63.5 kg per passenger must be added at the correct position for each seated or standing passenger, plus the driver. Double-deckers require the driver (and conductor) masses plus passengers on the upper deck only.



assume θ small,

$$R.d - W[\bar{y} + (\bar{z} + h_b)\theta] = 0$$

$$\Leftrightarrow R = \left[\frac{W(\bar{z} + h_b)}{d} \right] \theta + \left[\frac{W\bar{y}}{d} \right]$$

from graph of $R \vee \theta$,

$$\text{Gradient} = \frac{W(\bar{z} + h_b)}{d}$$

$$\text{R-intercept} = \frac{W\bar{y}}{d}$$

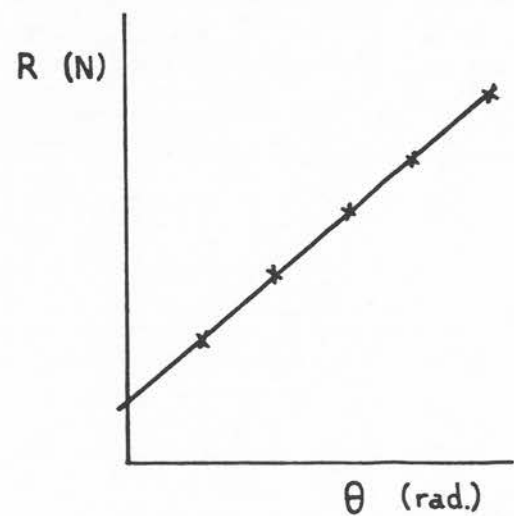


Figure 3.10 Method of CG Height Measurement

The combined CG height of two rigid bodies (Figure 3.11) is given by:

$$(m_1 + m_2)H = m_1h_1 + m_2h_2$$

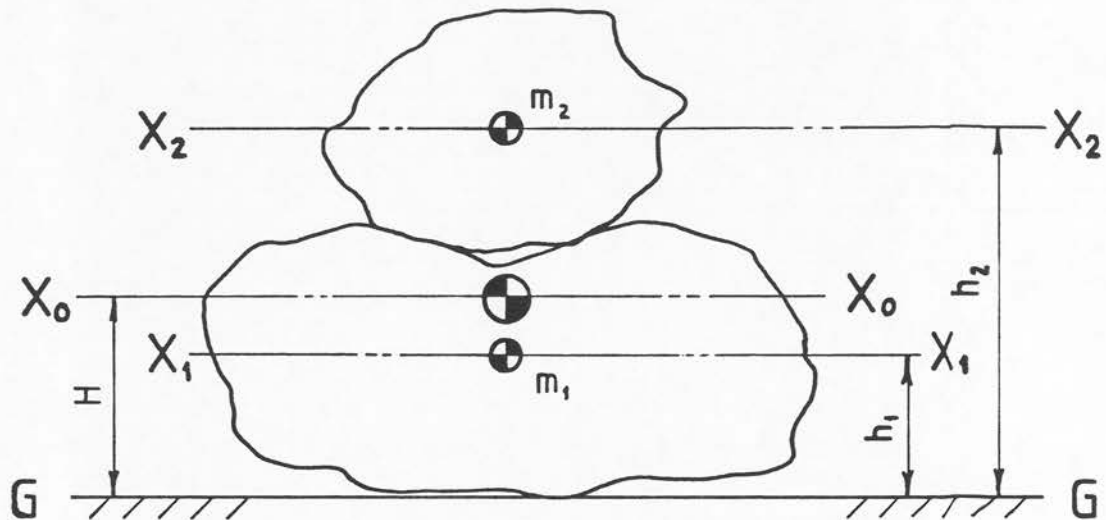


Figure 3.11 Calculation of Combined CG Height and Moment of Inertia of Two Bodies

Knowing the average seat height on each deck and given the regulation height for the CG of standing passengers is 2'8" (0.813m), the overall CG height for the laden vehicle can be determined. Allowance has been made for deflection of tyres due to the additional load. Vehicles with air springs have been assumed to maintain the same suspension displacement, laden or unladen, as intended by the self-levelling valves, whereas those with parabolic leaf springs have been assumed to displace according to the measured stiffness rates.

3.3.3 Results

Appendix A contains graphs of reaction force v roll angle for each vehicle. They all exhibit excellent linearity. Table 3.1 summarises the results of the CG height measurements and calculations.

Table 3.1
Measured and Calculated CG Heights

Vehicle	CG Height (± 0.02 m)		
	empty	laden	
		top deck	both decks
Leyland National 2	0.94	1.10	..
MCW Metrobus	1.08	1.48	1.43
Van Hool Astromega	1.43	1.67	1.63
Jonckheere Jubilee P799	1.56	1.75	..
Plaxton Supreme IV	1.06	1.21	..
Plaxton Paramount 3200	1.08	1.27	..
Duple Laser	1.16	1.30	..
MCW Metroliner	1.47	1.76	1.73
Leyland Royal Tiger Doyen	1.42	1.55	..

3.3.4 Discussion

The figures in the first column of Table 3.1 are the CG heights as measured for each unladen vehicle. The middle column contains the CG heights calculated using the method described in section 3.3.2. These values were used in the rollover simulations. For completeness, the CG heights of the double-deckers with a full complement of passengers and driver (typically 30 to 50 mm less than the "top deck laden" condition) are given.

It may be noted from Table 3.1 that the CG height of a high-floor coach (for example, Leyland Royal Tiger Doyen, or Jonckheere Jubilee P799) can exceed that of a double-decker bus. This is evidence that the traditional PSV categories are beginning to overlap, at least regarding stability.

3.4 ROLL MOMENT OF INERTIA

3.4.1 Method

With the vehicle on the same test rig as for the CG height measurement, and the suspensions and tyres blocked to effectively ensure rigid body motion, the method of determining the roll moment of inertia as outlined by Knight and Tidbury (1981) was used. (A very similar method was found to have been used earlier in Sweden by Nordström, et al, 1972).

Figure 3.12 and 3.14 illustrate the important feature: namely, the air springs (a) which provide a restoring moment when the bus is oscillated about an axis joining the two beam pivots (p). Figure 3.12 shows the details of the rig, and the rigid blocks (b) supporting the drive axle of the Astromega. The trailing wheels were immobilised in their ride positions with slings acting against the air springs.

The moment of inertia about the pivot axis is given by:

$$I_p = (2kl^2 + mgh) \left(\frac{T}{2\pi} \right)^2$$

where k = combined spring stiffness on one beam
 l = distance from spring to pivot
 m = total oscillating mass
 h = height of combined CG above pivot
 T = period of free oscillation

The roll moment of inertia of the vehicle about its CG is then obtained using:

$$I = I_p - I_B - mh^2$$

where I_B = moment of inertia of support beams about pivot

The time for several oscillations was recorded and the average calculated from several such measurements. This process was repeated for three different air spring pressures.

3.4.2 Calculation of Laden Moment of Inertia

As in the case of the CG height, the moment of inertia was measured in the unladen state. The laden moment of inertia was calculated by adding the effect of the passenger masses and taking into account the new CG position.

The combined moment of inertia of two rigid bodies about a given axis (Figure 3.11) is given by:

$$I_0 = I_1 + m_1(H-h_1)^2 + I_2 + m_2(H-h_2)^2$$

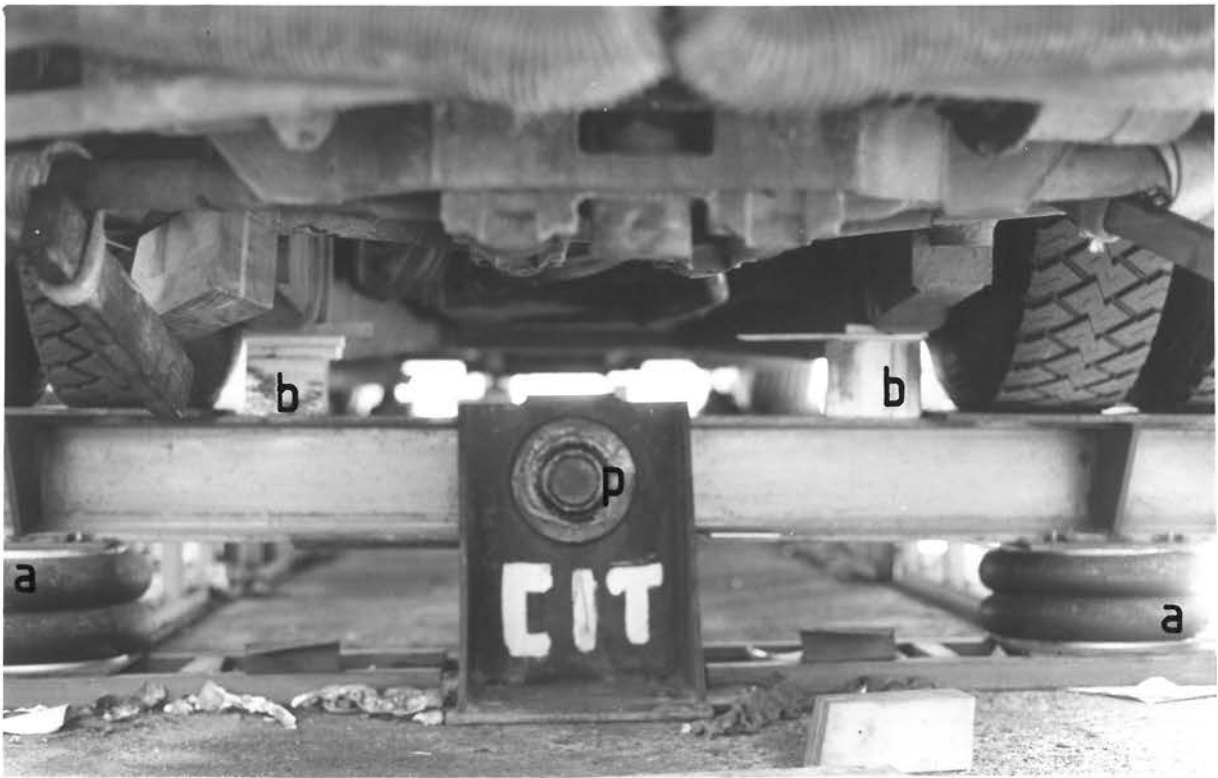


Figure 3.12 Moment of Inertia Measurement Rig Details



Figure 3.13 Force and Displacement Transducers for Tilt Test

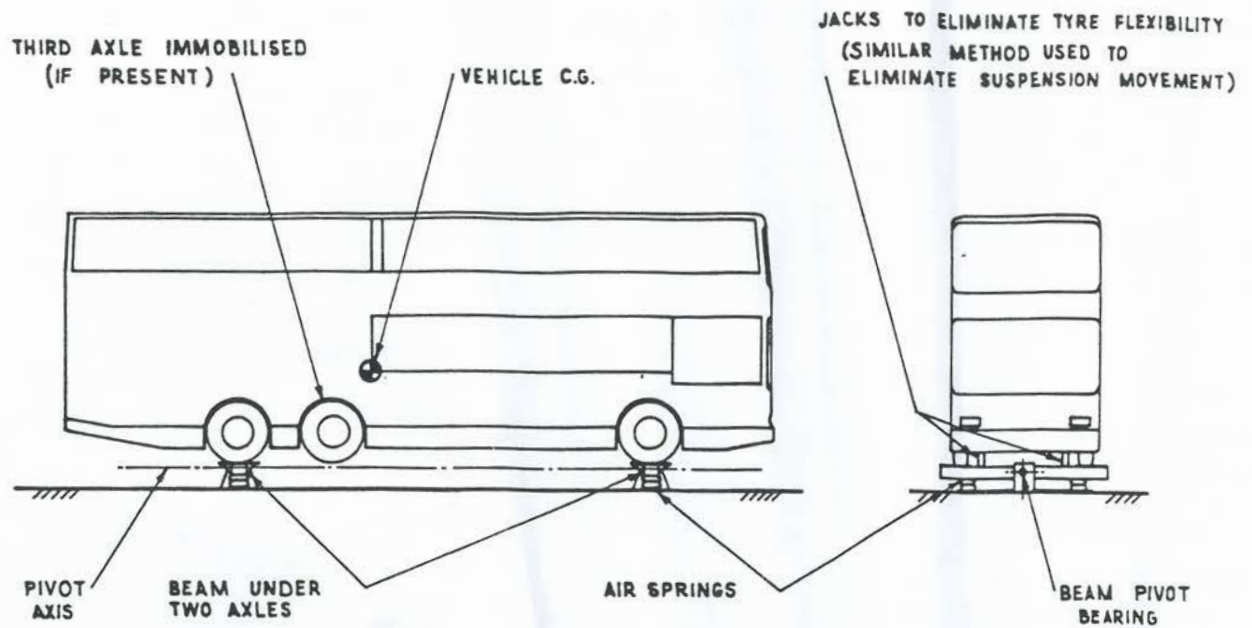


Figure 3.14 Method of Moment of Inertia Measurement

3.4.3 Results

Table 3.2 summarises the results of the moment of inertia measurements and calculations, together with mass data for comparison.

Table 3.2
Measured and Calculated Masses and Moments of Inertia

Vehicle	Mass (kg)		Mom. of Inertia (kgm^2)		
	empty	laden	empty		laden
	meas.	calc.	meas.	est.	calc.
Leyland National	9260	13770	12950	10000	13410
MCW Metrobus	10460	13250	14460		24810
Van Hool Astromega	16100	19660	10670	18000	25720
Jonckheere Jubilee	13700	17190	3330	16000	21710
Plaxton Supreme	9980	13660	8220		11440
Plaxton Paramount	10920	14480	10380		14150
Duple Laser	8750	12180	9310		12150
MCW Metroliner	15460	19140	11910	18000	27250
Leyland Royal Tiger	11940	15180	1880	13000	15690

3.4.4 Discussion

For reasons which are not obvious, the apparatus for moment of inertia measurement, which has given reasonably accurate results in the past, proved to be rather unreliable.

Measurements reported by Keenan, Sadeghi and Tidbury (1981) of six single-decked buses and coaches revealed a reasonably close relationship between mass and moment of inertia for this category of vehicle, as illustrated in Figure 3.15.

Using this relationship as the basis for assessing the accuracy of the measured moments of inertia, where the magnitude of the roll moment of inertia of an empty vehicle is significantly different from its unladen mass, this is considered to be in error and a more appropriate value has been estimated. The estimate was then used in the calculation of the laden moment of inertia.

The source of error was considered to be a combination of imperfect pivot bearings, inaccurate pressure measurement and equalisation, and the effect of a gusting side wind.

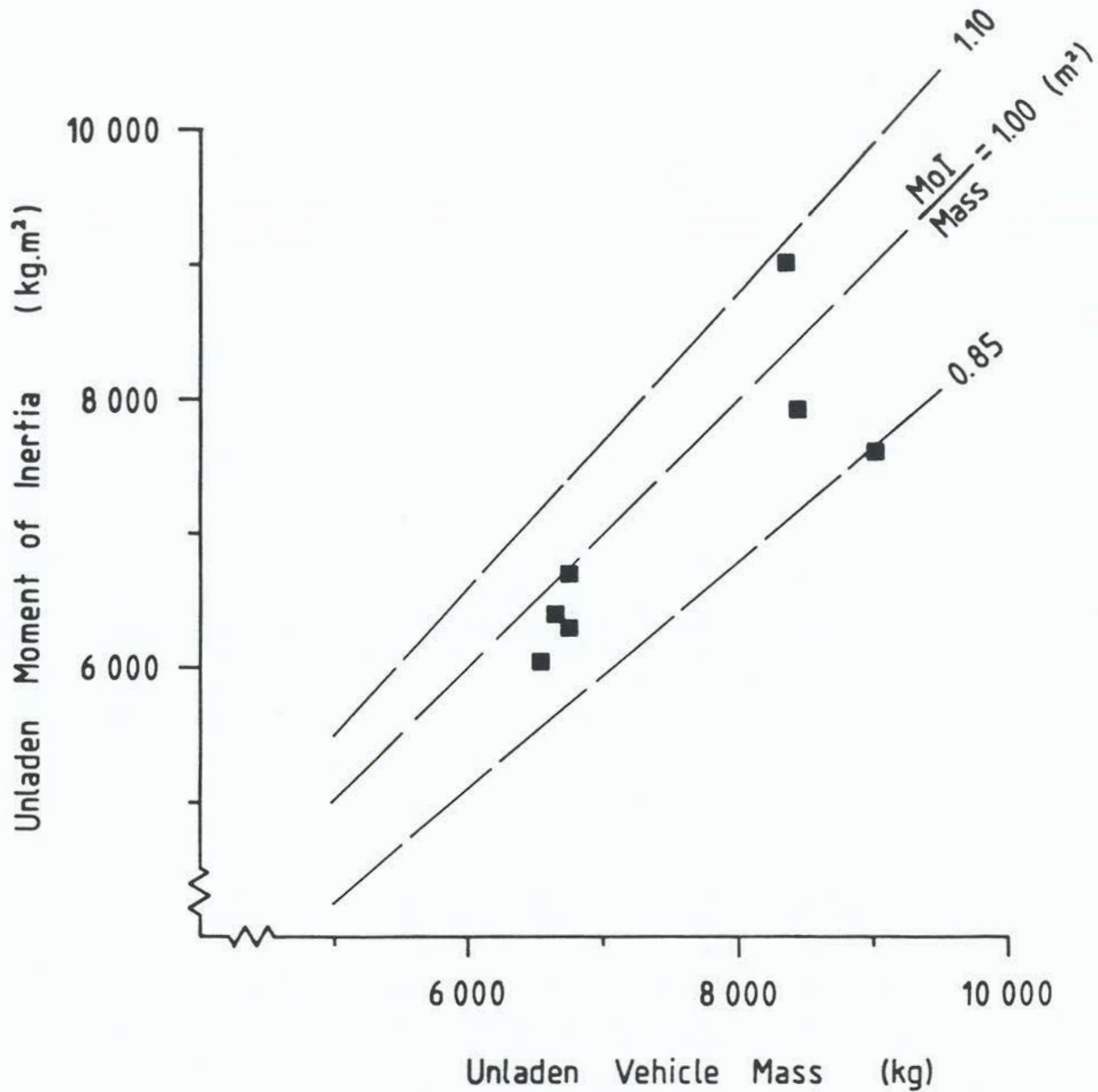


Figure 3.15 Relationship between Roll Inertia and Mass
(from Knight and Tidbury, 1981)

3.5 STIFFNESSES

3.5.1 Method

Vertical, lateral and roll stiffnesses were obtained for the suspension and tyres of each axle of a vehicle by applying a force in the particular direction, and measuring both the force and the associated deflection in the same direction. Vertical force was applied by loading sandbags, and was measured with

weighpads - one for each wheel. Roll moment was obtained by shifting the sandbags from one side of the bus to the other. By calculating the difference between weighpad readings and measuring the distance between wheels on one axle, the actual applied roll moment was determined. The lateral component of force during a tilt test is due to gravity (Figure 3.16. See also Figure 7.4), which was recorded using loadcells positioned between the tyre sidewall and the tilt table restraint (Figure 3.13). Linear bearings under the weighpads minimised friction losses from the lateral force.

By mounting potentiometers in the configuration shown in Figures 3.13 and 3.16 and measuring their initial positions in space, it was possible using geometric relationships to extract separately the vertical, lateral and roll displacements of one body relative to another. Graphs of Force v Displacement give an indication of linearity. Appendix B contains stiffness graphs for each vehicle. Further details of the behaviour of each vehicle are summarised in Appendix C. Instrumentation details are given in Appendix D. The analysis method for converting transducer displacements into coordinate displacements is outlined in Appendix H.

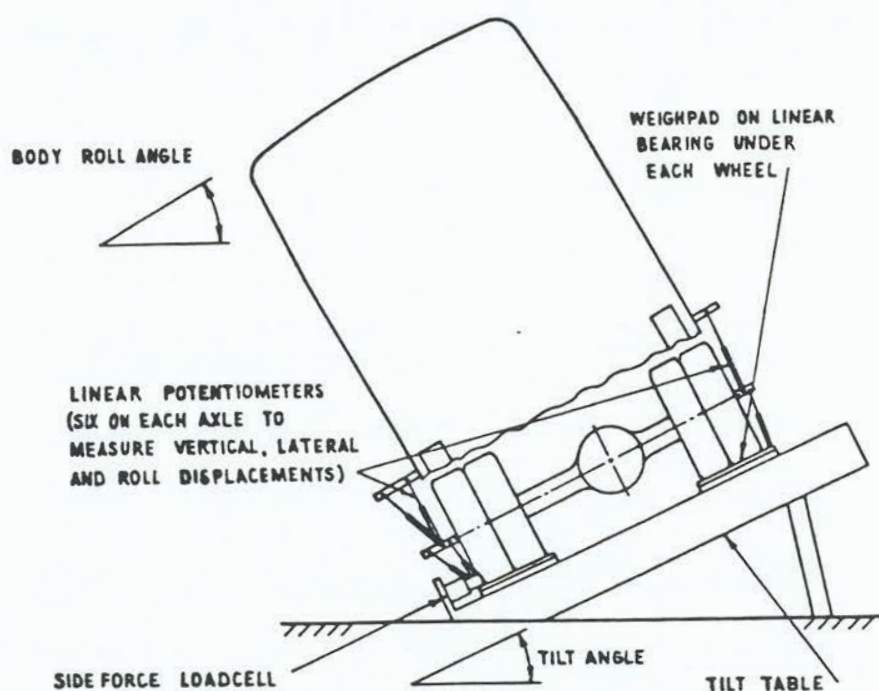


Figure 3.16 Tilt Test Instrumentation

3.5.2 Results

Tables 3.3 to 3.5 summarise the results of the stiffness measurements and calculations.

Table 3.3
Vertical Stiffness Measurements

Vehicle	Vertical Stiffnesses (kN/m)			
	Springs		Tyres	
	front	rear	front	rear
Leyland National 2	2810	1070	350	4220
MCW Metrobus	1050	2130	1900	3700
Van Hool Astramega	1120	36000	730	18700
Jonckheere Jubilee	480	1220	2610	2740
Plaxton Supreme IV	1180	2100	880	1270
Plaxton Paramount	9220	3840	1820	3200
Duple Laser	420	910	1810	3790
MCW Metroliner	890	5520	1930	5760
Leyland Royal Tiger	460	800	1890	3420

Table 3.4
Lateral Stiffness Measurements

Vehicle	Lateral Stiffnesses (kN/m)			
	Springs		Tyres	
	front	rear	front	rear
Leyland National 2	360	1290	530	6460
MCW Metrobus	660	810	260	1220
Van Hool Astromega	1430	1250	660	1210
Jonckheere Jubilee	910	1370	2670	4520
Plaxton Supreme IV	170	340	710	720
Plaxton Paramount	82	800	770	1160
Duple Laser	1010	1230	650	790
MCW Metroliner	960	1250	660	1220
Leyland Royal Tiger	40	9840	550	530

Table 3.5
Roll Stiffness Measurements

Vehicle	Suspension Roll Stiffnesses (kNm/rad)			
	Roll Moment Test		Tilt Test	
	front	rear	front	rear
Leyland National	440	470	210	2500
MCW Metrobus	180	1760	180	1090
Van Hool Astromega	470	5450	180	970
Jonckheere Jubilee	90	600	140	620
Plaxton Supreme	410	4140	1470	1060
Plaxton Paramount	110	1240	210	650
Duple Laser	160	490	190	510
MCW Metroliner	220	1220	180	1590
Leyland Royal Tiger	170	380	390	1160

3.5.3 Discussion

Vertical stiffnesses are expressed as axle rates, that is, as the stiffness of a single spring equivalent to the actual two or more springs.

Static tests on 8 different 11.00x22.5 radial ply tyres performed at TRRL resulted in vertical stiffness measurements averaging 1032 kN/m for an inflation pressure of 775 kN/m² (when loaded to 23.5 kN at 22°C on a flat surface). The average stiffness of 4 tyres at a pressure of 650 kN/m was 871 kN/m. These compare favourably with 950 kN/m typically obtained in these tests (ie, 1/2 of front axle rates and 1/4 of rear axle rates).

Measurement of vertical and roll stiffnesses of vehicles with air suspensions is not straightforward. This is because self-levelling valves are generally used to maintain the ride height within certain limits, by inflating the air bellows when the static axle load increases significantly, and exhausting the air if the load decreases. The stiffness of an air spring is essentially a function of its internal pressure and will therefore vary for a vehicle as a function of the static axle load. In addition, the addition of air to springs as the load

increases reduces the spring deflection, and thus the stiffness measured in a simple static test may easily be higher than the operating range of spring stiffnesses. This is evident in the cases of the National and Paramount front axles. More accurate measurements would be obtained if the self-levelling valves were made inoperative for the duration of the test.

The Duple Laser and Plaxton Supreme were the only vehicles tested which did not have air suspension: they each have leaf springs at both axles.

3.6 ROLL CENTRE HEIGHT

3.6.1 Description

"Roll Centre" in this report is used to describe the position, at the front or rear axle, of the axis about which the sprung mass (body and chassis) is assumed to rotate relative to the unsprung mass (axles and wheels).

It was possible to calculate the position of the instantaneous roll centre at each axle, using the successive positions of the displacement transducers during the roll moment test and the tilt test (Instrumentation layout was similar to that shown in Figure 3.16).

3.6.2 Results

Table 3.6 contains the average vertical component of the roll centre position at the front and rear axles for each vehicle.

Table 3.6
Mean Roll Centre Heights

Vehicle	Roll Centre Heights (m)			
	Roll Moment Test		Tilt Test	
	front	rear	front	rear
Leyland National	0.19±0.05	0.1 ±3.4	0.5±0.3	-0.1±3
MCW Metrobus	0.80±0.08	1.8 ±1.5	1.2±0.2	1.7±1.3
Van Hool Astromega	0.2 ±1.7	1.0 ±2.8	0.9±1.0	0.8±0.7
Jonckheere Jubilee	0.88±0.07	0.68±0.04	0.9±0.3	0.8±0.8
Plaxton Supreme	0.9 ±0.7	28. ±47	-8.6±2.7	-1.2±0.2
Plaxton Paramount	1.1 ±0.2	1.32±0.06	1.0±0.2	0.6±0.2
Duple Laser	0.9 ±0.5	1.46±0.09	1.0±0.3	0.9±0.2
MCW Metroliner	0.80±0.03	0.73±0.03	0.8±0.5	0.7±0.4
Leyland Royal Tiger	0.7 ±0.2	1.0 ±0.6	1.2±0.4	1.1±0.1

3.6.3 Discussion

Measurement of the position of the instantaneous roll centre proved to be not always as accurate as desirable (errors given are for 95% confidence intervals). In the extreme case (the Plaxton Supreme IV), it is thought that age (and therefore, rust) accentuates the interleaf friction, causing erratic suspension motion, due to the stick/slip behaviour. This phenomenon makes measurement of the roll centre position very difficult. It is known from previous work on commercial vehicle suspensions that the roll centre height for a leaf spring generally lies between the upper and lower spring mounting points (for example, see Figure 3.17 - from Mai and Sweatman 1984), and thus appropriate values have been estimated from measurements of these positions and are included in Table 3.8. Similarly, where other measured results appear unrealistic, the suspension geometry has been used

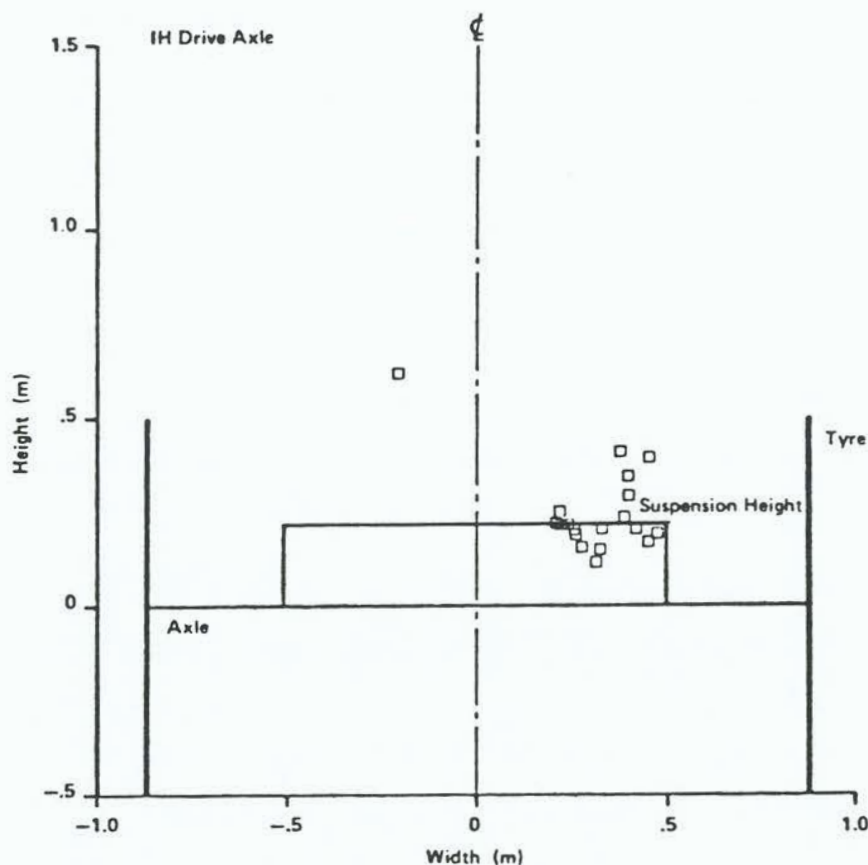


Figure 3.17 Roll Centre Position Measurements
for Typical Leaf-Spring Truck Axle
(from Mai & Sweatman, 1984)

Two of the coaches tested (the Jonckheere Jubilee P799 and the MCW Metroliner) show excellent repeatability between the roll moment test results and the tilt test results for the roll centre height. The tilt test is a quite complex loading situation for a vehicle - not only is the roll moment increasing as the platform is tilted, but the lateral force (relative to the platform) is increasing, and, to a lesser extent, the vertical load is decreasing. If the lateral or vertical suspension stiffnesses are relatively low, then the changing forces will result in appropriate lateral or vertical displacements. If these are significant with respect to the roll angle, then the accuracy of the roll centre height calculation (which assumes rotation only) deteriorates. It is therefore to be expected that those PSVs with higher front and rear lateral spring stiffnesses will generally give better results for roll centre height, which proves to be so.

3.7 TILT TEST PERFORMANCE

3.7.1 Method

The history of the tilt test is outlined in Section 5.1. The vehicle being tested was positioned on a tilting platform. Suitable chocks were strategically placed to prevent the tyres from slipping sideways, and some method of safety restraint was employed (such as chains over axles, or buffer bump stops) to stop the vehicle from tipping over when the tilt limit was reached. The tests were conducted by tilting the platform until the vehicle's point of instability was reached (Figures 3.18 and 3.19 show two tilt tests at such a stage).

To facilitate measurement of vertical and lateral forces, each vehicle was mounted on individual weighpads and linear roller bearings (to ensure all the lateral force was measured by the loadcell, rather than transmitted directly from the tyres to platform). The forces, displacements and roll angles were recorded at 5° intervals of increasing tilt angle.

3.7.2 Results

Table 3.7 gives the tilt angle (measured from the horizontal) for the platform and the roll angle for the body relative to the platform (averaged from readings at the front and rear of the body) for the maximum stable tilt angle of the vehicle. The rear axle was the first to lift in all cases except the Leyland National 2.



Figure 3.18 Van Hool Astronomega at Maximum Tilt Angle (28 deg.)



Figure 3.19 Leyland National 2 at Maximum Tilt Angle (38 deg.)

Table 3.7
Tilt Test Results

Vehicle	Roll Angles at Point of Instability (deg)		
	table angle (± 0.5)	body angle (± 0.2)	av. sandbag mass (± 1 kg)
Leyland National	38.0	3.6	65
MCW Metrobus	25.0	7.2	66
Van Hool Astramega	28.0	..	64
Jonckheere Jubilee	24.5	9.0 (1)	64
Plaxton Supreme	36.5	8.0	65
Plaxton Paramount	32.0	7.5	65
Duple Laser	33.6	7.8	65
MCW Metroliner	25.0	5.1 (2)	63
Leyland Royal Tiger	33.0	6.9	67

Notes: (1) Jonckheere only tested half-laden.
(Simulation predicts 21.5 degrees for rear wheel lift when fully laden.)
(2) Metroliner not quite tested to limit.

3.7.3 Discussion

It should be noted that these tilt tests were not identical to the legislative test. The average mass of a sandbag (nominally 63.5 kg) was found to vary from 63.3 to 66.9 kg. This was considered to be sufficiently important to include in Table 3.7. It was not uncommon for one or more sandbags to be displaced during a test, thus increasing the overturning moment. Also in the case of the MCW Metroliner, the test was not continued to its usual limit owing to uncertainty about the safety of the vehicle on the linear bearings at the point of balance. Its result will

therefore be conservative. The Jonckheere Jubilee was only half laden for the tilt test in order to satisfy the load limitations of the Aldenham tilt platform (in preference to the greater distance, time, expense and inconvenience of one of the higher-capacity outdoor sites). The result for the tilt test simulation of the fully laden Jonckheere is also given in Table 3.7, as a footnote.

3.8 INFLUENCE OF THE TILT TEST REGULATION ON COACH DESIGN

Rollover stability will be considered in detail in the following chapters. However, some comments on the vehicles tested, and in particular, distinguishing between the two imported coaches and the remainder which were built to pass the tilt test, are appropriate.

The Van Hool Astromega appears to match or exceed the standard of vehicles sold in the United Kingdom. The Jonckheere Jubilee P799 was quite a contrast. Subjectively, it felt unstable to ride in, with very large pitch and roll motions in normal traffic conditions. Objectively, this behaviour is predicted from the low roll stiffnesses and high laden and unladen CG heights. Actually, this vehicle had the highest unladen CG height of all nine PSVs tested, including doubledeckers. As will be shown in Section 4.9, the Jubilee P799 is also the easiest to overturn in the kerb-impact rollover accident simulation.

These observations indicate the need for an overturning stability test. It is not possible to argue this from accident statistics for this and similar coach designs, for two reasons:

- (1) Detailed accident statistics are not available - they would have to come from a foreign country, since coach models not tilt tested may not be sold in Great Britain, and
- (2) Rollover accidents occur so infrequently that when subdivided into smaller categories, statistical conclusions lose their significance.

3.9 SUMMARY

Table 3.8 summarises the vehicle data as used in the computer simulations.

Table 3.8
Summary of Experimental Data
(as used in computer simulations)

Parameter	V e h i c l e								
	Nat'l	M'bus	A'mga	Jonck	Supr.	Pa'mt	Laser	M'lnr	Royal
Mass (kg)	13770	13250	19660	17190	13660	14480	12180	19140	15180
CG Height (m)	1.10	1.48	1.67	1.75	1.21	1.27	1.30	1.76	1.55
Moment of Inertia (kg.m ²)	13410	24810	25720	21710	11440	14150	12150	27250	15690
Vertical Stiffness (kN/m)									
Front Spring	500	1050	1120	480	1180	500	420	890	460
Rear Spring	1070	2130	5520	1220	2100	900	910	5520	800
Front Tyres	1900	1900	1930	1900	1900	1820	1810	1930	1890
Rear Tyres	3600	3700	5760	3600	3600	3200	3790	5760	3420
Lateral Stiffness (kN/m)									
Front Spring	360	660	500	910	170	80	1010	960	500
Rear Spring	1290	810	1250	1370	340	800	1230	1250	1200
Front Tyres	530	260	660	650	710	770	650	660	550
Rear Tyres	900	1220	1210	900	720	1160	790	1220	530
Roll Stiffness (kNm/r)									
Front Spring	440	180	470	90	410	210	160	220	390
Rear Spring	470	1090	1590	600	1060	650	490	1590	1160
Roll Centre Height (m)									
Front	0.50	0.80	0.62	0.88	0.50	0.90	0.53	0.80	0.70
Rear	0.50	0.73	0.90	0.76	0.73	0.70	0.78	0.73	0.73
CG to Axle Distance (m)									
Front	3.32	3.17	4.28	3.70	3.50	3.80	3.15	4.58	3.98
Rear	1.76	1.80	2.23	2.17	2.58	2.06	2.53	1.72	2.07
Track Width (m)									
Front	2.05	2.09	2.10	1.91	2.05	2.05	2.02	2.09	2.05
Rear	2.00	2.04	1.94	2.01	2.01	2.01	1.99	2.06	1.98

CHAPTER 4

KERB-IMPACT ROLLOVER ACCIDENT
SIMULATION PROGRAM

4.1 PREVIOUS WORK

Chapter 1 established that a typical bus rollover accident involves a "tripping" phenomenon, in which the vehicle, out of control, laterally impacts a kerb, low wall or soft verge. This is therefore the mode of overturning accident investigated in detail.

It is a simple procedure to evaluate the "threshold overturning velocity", that is, the lateral velocity above which rollover will occur, when many assumptions are made. Appendix E contains the derivation of an equation for the threshold velocity, assuming two-dimensional rigid-body motion, zero initial roll velocity, short inelastic collision and pure rotation about the kerb after impact.

Bickerstaff (1976), using this method (formalised by Jones, 1976) showed that, neglecting suspension effects, all cars, light trucks and heavy trucks would roll over if they struck a kerb laterally at greater than 7 m/s (25 km/h). Bickerstaff's results have been converted to SI units to display them in Figure 4.1, and a hatched zone representing typical coach rollover characteristics added. It should be noted that, in practice, maximum lateral accelerations are limited to about 1.0 g for cars and 0.8 g for commercial vehicles, due to finite tyre/road friction. Those vehicles theoretically capable of higher lateral accelerations will understeer or oversteer rather than overturn.

Similar assumptions (rigid-body, two-dimensional motion, and inelastic collision) were made by Miles (1976), in developing a simple digital computer simulation of the kerb-impact rollover. His purpose was to study the energy absorption requirements of a coach superstructure when cant-rail impact with the ground occurs, and to investigate the first-order effects of varying parameters such as CG height, track width, and cant-rail height on the impact energy levels.

To aid investigations into roof structure strength requirements in Hungary, Voith (1981) developed a two-dimensional digital computer simulation. His model concentrates on the structural deformations due to roof-ground contact, using empirical equations to describe soil and framework deformation. The roll-initiating phase was not important and, if present, was not described.

A complete bus rollover accident simulation, from normal driving, through emergency steering and braking, loss-of-control, "kerb impact", and free flight, to structural collapse on roof impact,

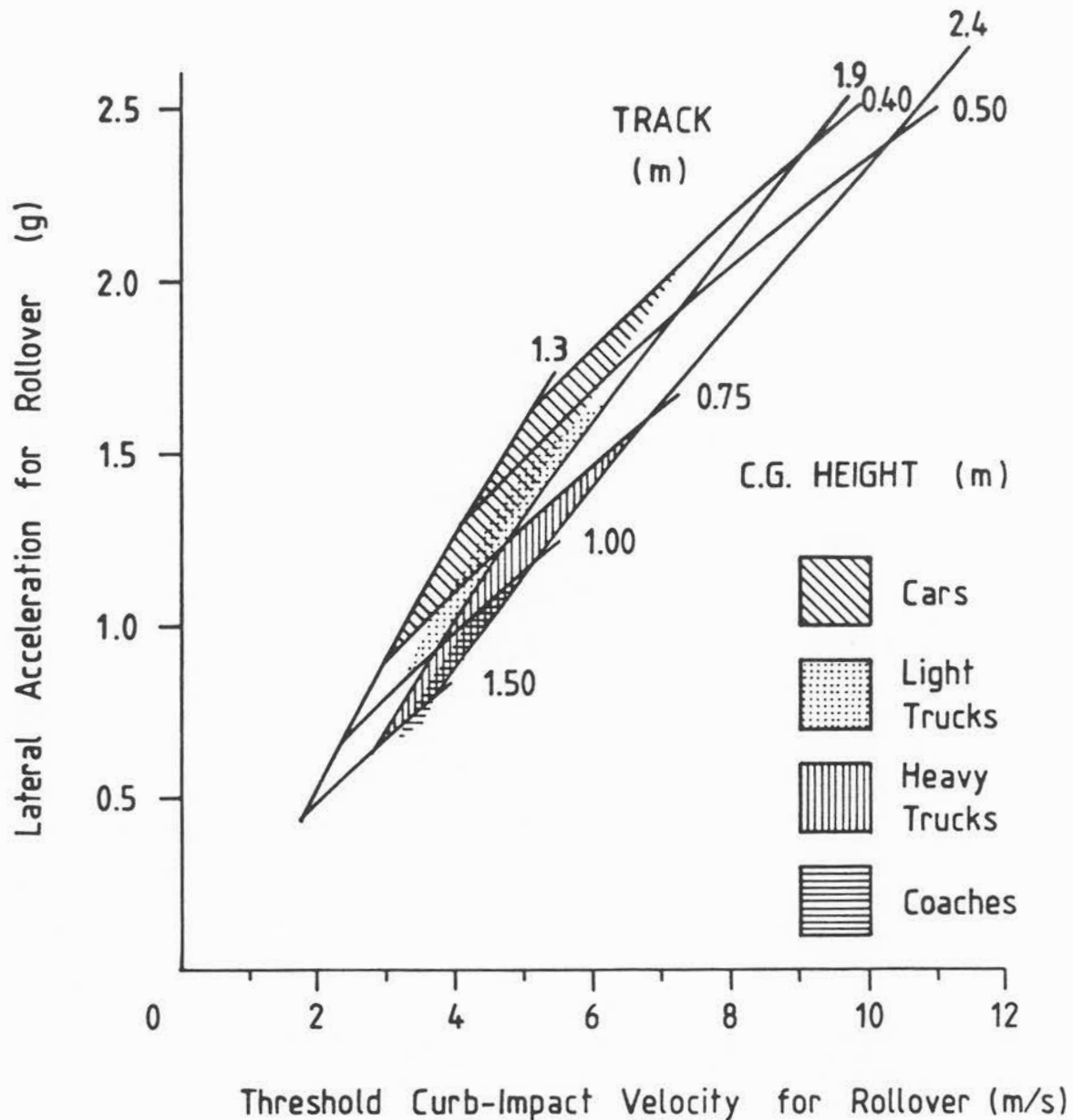


Figure 4.1 Effect of Varying Track and CG Height on Rollover Thresholds in Steady Cornering and Kerb Impact

was assembled by Sadeghi and Tidbury (1979). A vehicle dynamics program was used to model the behaviour until the bus was sliding sideways towards the kerb. Lateral suspension and tyre data were unavailable at the time, so a two-dimensional rigid-body impact was developed. The impact forces were calculated assuming certain stiffness and energy absorption characteristics for the tyre/kerb contact. CRASHD, the CRANfield Structural High-Deformation finite-element program [described by Miles (1976 and 1977), and

Tibury and Kerman (1978)] used the position, velocity and energy data at the point of roof/ground contact to determine the superstructure collapse behaviour. Multiple rollovers could be simulated. Among the suggested improvements to the simulation were:

- '● a more comprehensive simulation of the 'take-off' phase of the program allowing longitudinal as well as lateral motion;
- a true six degree of freedom free flight model...with pitching and yawing in addition to the rolling motion selected.'

Chapter 2 described a computer model (SIDBURT) which incorporates these two recommendations. However, the application was to predict trajectories and velocities for a specific roof impact strength test, rather than a generalized accident situation. SIDBURT was thus adapted and modified to simulate a typical kerb-impact overturning accident.

4.2 FEATURES OF SIMULATION

In order to investigate bus rollover accident propensity, a comprehensive, yet realistic simulation was deemed necessary. The following features were considered important:

1. General motion in three-dimensional space;
2. Consideration of axles as separate rigid bodies, free to bounce, roll and displace laterally with respect to the vehicle body;
3. Roll axis position independent of CG positions of either axles or sprung mass (Weir, et al, 1974, in the bus version of a vehicle dynamics program to investigate bus handling, constrained the sprung mass CG to lie on the roll axis, to simplify the equations. That was considered to be an undesirable restriction for this simulation);
4. Capability to incline the roll axis slightly from the horizontal; and
5. Energy absorption at the tyre/kerb impacts.

4.3 MATHEMATICAL MODEL

The vehicle is defined by its mass distribution, inertial properties, spring and damping constants, and dimensions. Figure 4.2 shows the configuration of the three rigid bodies (sprung mass, front axle and rear axle), their interconnection via the inclined roll axis, and the axis systems adopted.

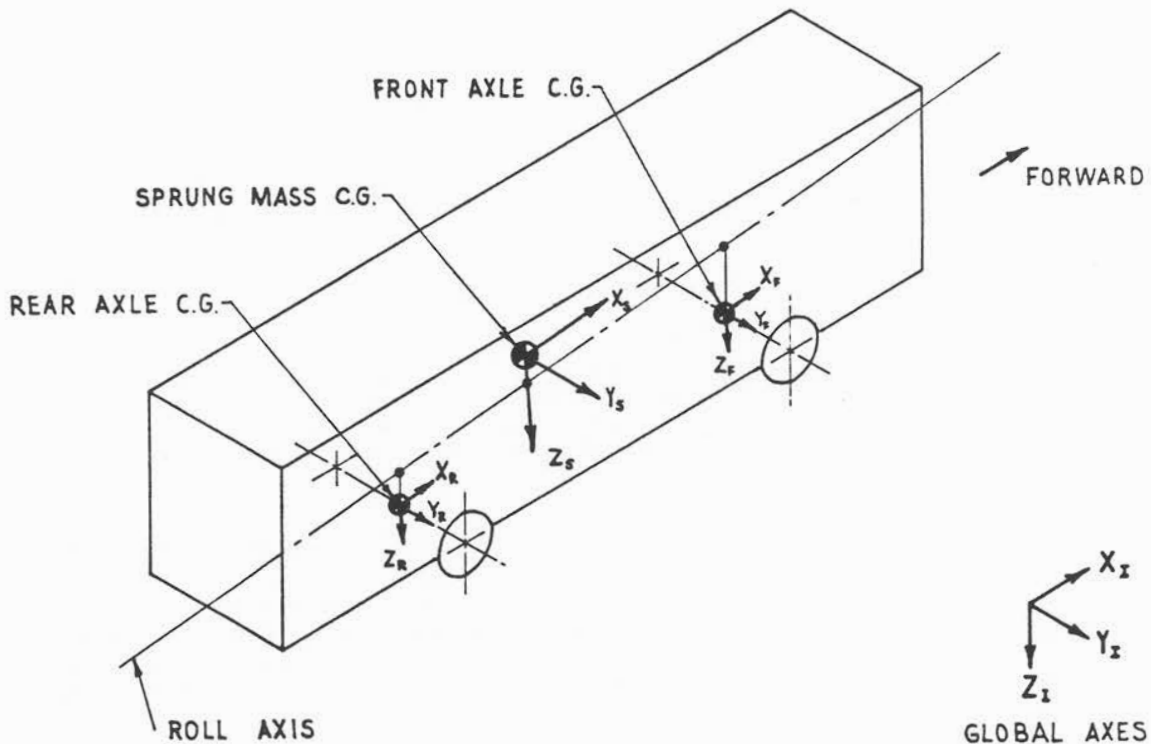
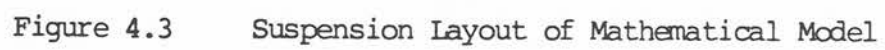


Figure 4.2 Mathematical Model for Rollover Accident Simulation

The sprung mass has six general degrees of freedom, while the axles are free to translate and rotate in the YZ plane relative to the sprung mass. Figure 4.3 shows more clearly the vertical, lateral and torsional spring/damper units between an axle and the roll axis in the model. Note that for simplicity the usual two or four suspension springs are represented by a single vertical spring at each axle, and the roll stiffness is considered to be provided via a separate spring.



The twelve degrees of freedom are summarised in Table 4.1.

Table 4.1
Degrees of Freedom in Rollover Accident Model

Rigid Body	Degree of Freedom	Assumptions
Sprung Mass	longitudinal displacement lateral displacement vertical displacement roll angle pitch angle yaw angle	
Front and Rear Axles	lateral displacement vertical displacement roll angle	all 3 measured relative to sprung mass roll angle small

4.4 ASSUMPTIONS

The more general a simulation is intended, the larger and more complex it becomes, requiring greater quantities of input data and ever increasing amounts of computer processing time. Some compromise between features and assumptions is necessary if a practically useful program is desired. The following assumptions were therefore made in deriving the equations of motion:

1. The road surface is assumed to be uniform and horizontal.
2. The kerb is positioned perpendicular to the vehicle's initial lateral velocity.
3. Axle roll angles relative to the sprung mass are also assumed to be small.
4. The relative roll motion of the sprung mass and axles is assumed to take place about a roll axis which is fixed with respect to the sprung mass. It is defined by specifying the initial values of the roll centre heights at the front and rear axles. The roll axis is taken as the line joining these two points.

5. Spring/ damper units act at the tyre contact points where the forces from the road and kerb are applied, and also between the sprung mass and each axle, simulating the suspension components.
6. Linear characteristics are used for every spring with two exceptions: the roll stiffnesses and tyre/kerb stiffnesses are as shown in Figures 4.4 and 4.5 respectively. The secondary roll stiffness models the effect of bump stops when the relative roll angle exceeds 15 degrees. (In practice, the only simulations in which the bump stops were contacted were those for the Duple Laser, Jonckheere Jubilee P799 and MCW Metrobus.) Similarly, the kerb stiffness takes into account the limited tyre lateral stiffness and attempts to model the increased stiffness when the kerb contacts a more solid part of the vehicle, such as a wheel rim.

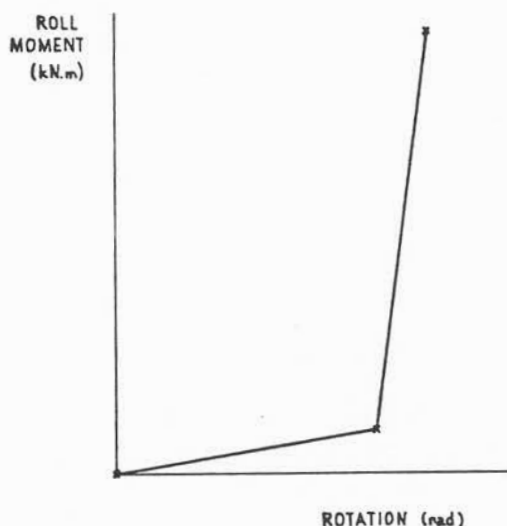


Figure 4.4
Representation of Roll
Stiffness Non-linearities

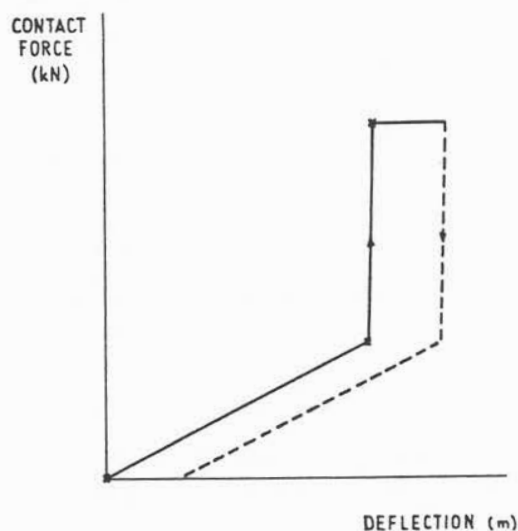


Figure 4.5
Representation of Tyre/Kerb
Stiffness Non-linearities

7. Beam axles are symmetric and hence products of inertia are neglected.
8. Beam axles can also represent the dynamic behaviour of independently suspended wheels.
9. Three-axle double-deckers are approximated to two-axle vehicles with mass and stiffnesses of the two rear axles combined.

Assumption 8 is actually a constraint and requires some explanation. To develop a computer model capable of realistically simulating an independent suspension requires considerably more work and more measured input data.

The major differences between the two generalised suspension types are a slightly lighter unsprung mass, and increasing tyre camber angles with body roll, for the vehicle with independent suspension (Figure 4.6a). The tyre camber angles of a vehicle with beam axles do not change significantly before wheel lift (Figure 4.6b). These effects are regarded as minor compared with the primary factors affecting rollover propensity outlined in chapter 6.

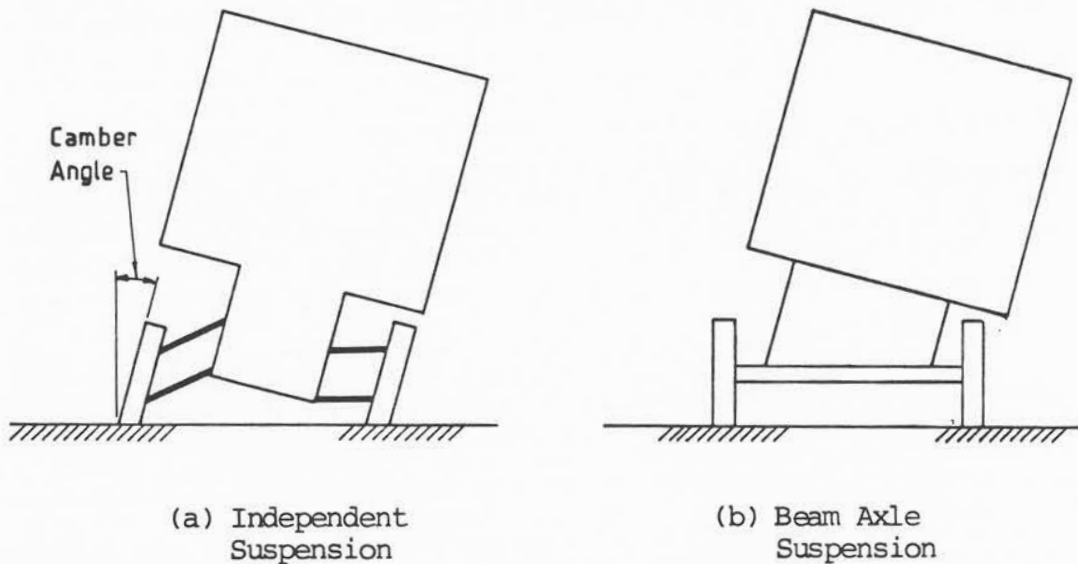


Figure 4.6 Comparison of Camber Changes with Body Roll Angle of Independent and Beam Axle Suspensions

The only tested vehicle with independent suspension was the Van Hool Astromega, whose front and trailing axles were this design. The drive axle was solid, however, and thus it is not thought to be a significant source of error to assume beam axles throughout.

4.5 DERIVATION OF EQUATIONS OF MOTION

4.5.1 Axis Systems

Four axis systems are used in formulating the equations of motion (Figure 4.2). The inertial axis set (X_i, Y_i, Z_i) is considered to be fixed with respect to the ground. The sprung mass axis set (X_s, Y_s, Z_s) is attached to the sprung mass CG with the convention X_s forward, Y_s to the right, and Z_s vertically downward. The front axle axis set (X_f, Y_f, Z_f) is attached to the front axle CG. Similarly, the rear axle axis set (X_r, Y_r, Z_r) is attached to the rear axle CG. Note that in the following formulation, subscript F (for front axle) and subscript R (for rear axle) can be exchanged. Thus, equations will be derived for the front axle only, but equivalent rear axle equations are produced by substituting subscripts.

Initially all four axis sets point virtually in the same directions. The exception is that the three rigid-body axis sets are rotated about the Y direction by θ_0 , the small angle of roll axis inclination. (See Figure 4.8 for more details.)

4.5.2 Transformation Matrices

4.5.2.1 Inertial Axis to Sprung Mass Axis Transformation

The order of angular rotations to transform from inertial coordinates to sprung mass coordinates that has been adopted for this analysis is Yaw, Pitch and Roll. This was chosen to allow roll angle to exceed 90 degrees without causing trigonometric difficulties.

Let w represent a general vector to be transformed. The angular orientation of the sprung mass relative to the ground is given by the rotation matrix product,

$$\underline{w}_s = [A_\theta][A_\phi][A_\psi]\underline{w}_i \quad (4.1)$$

$$\underline{w}_s = \begin{bmatrix} 1 & 0 & -\theta_0 \\ 0 & 1 & 0 \\ \theta_0 & 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\phi & \sin\phi \\ 0 & -\sin\phi & \cos\phi \end{bmatrix} \begin{bmatrix} \cos\theta & 0 & -\sin\theta \\ 0 & 1 & 0 \\ \sin\theta & 0 & \cos\theta \end{bmatrix} \begin{bmatrix} \cos\psi & \sin\psi & 0 \\ -\sin\psi & \cos\psi & 0 \\ 0 & 0 & 1 \end{bmatrix} \underline{w}_i \quad (4.2)$$

$$\underline{w}_s = \begin{bmatrix} \cos\theta\cos\psi - \theta_0\sin\phi\sin\psi & \cos\theta\sin\psi + \theta_0\sin\phi\cos\psi & -\sin\theta \\ -\theta_0\cos\phi\sin\theta\cos\psi & -\theta_0\cos\phi\sin\theta\sin\psi & -\theta_0\cos\phi\cos\theta \\ \sin\phi\sin\theta\cos\psi & \sin\phi\sin\theta\sin\psi & \sin\phi\cos\theta \\ -\cos\phi\sin\psi & +\cos\phi\cos\psi & 0 \\ \theta_0\cos\theta\cos\psi + \sin\phi\sin\psi & \theta_0\cos\theta\sin\psi - \sin\phi\cos\psi & \cos\phi\cos\theta \\ +\cos\phi\sin\theta\cos\psi & +\cos\phi\sin\theta\sin\psi & 0 \end{bmatrix} \underline{w}_i \quad (4.3)$$

To transform from sprung mass axis set to inertial axis set, the inverse transformation is used,

$$\underline{w}_I = [\text{DIR}] \underline{w}_S, \text{ where} \quad (4.4)$$

$$[\text{DIR}] = [A_\psi]^T [A_\theta]^T [A_\phi]^T [A_{\theta_0}]^T \quad (4.5)$$

$$[\text{DIR}] = \begin{bmatrix} \cos\theta\cos\psi - \theta\sin\phi\sin\psi & \sin\phi\sin\theta\cos\psi & \theta\cos\theta\cos\psi + \sin\phi\sin\psi \\ -\theta\cos\phi\sin\theta\cos\psi & -\cos\phi\sin\psi & +\cos\phi\sin\theta\cos\psi \\ \cos\theta\sin\psi + \theta\sin\phi\cos\psi & \sin\phi\sin\theta\sin\psi & \theta\cos\theta\sin\psi - \sin\phi\cos\psi \\ -\theta\cos\phi\sin\theta\sin\psi & +\cos\phi\cos\psi & +\cos\phi\sin\theta\sin\psi \\ -\sin\theta & \sin\phi\cos\theta & \cos\phi\cos\theta \\ -\theta\cos\phi\cos\theta & & \end{bmatrix} \quad (4.6)$$

4.5.2.2 Sprung Mass Axis to Front Axle Axis Transformation

To transform from sprung mass axis set to front axle axis set involves a rotation of ϕ_f about the roll axis,

$$\underline{w}_f = [A_{\phi_f}] \underline{w}_S \quad (4.7)$$

$$\underline{w}_f = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\phi_f & \sin\phi_f \\ 0 & -\sin\phi_f & \cos\phi_f \end{bmatrix} \underline{w}_S \quad (4.8)$$

Note that, although ϕ_f is assumed small, sine and cosine terms are included in the initial stages of the derivation of equations, so that when terms are differentiated with respect to time, no products are overlooked, such as,

$$\begin{aligned} \frac{d}{dt} \cos\phi_f &= -\dot{\phi}_f \sin\phi_f \\ &\doteq -\phi_f \dot{\phi}_f \end{aligned}$$

4.5.2.3 Angular Rate Transformation

As stated earlier, the transformation from the inertial axis set ($X_I Y_I Z_I$) to the sprung mass axis set ($X_S Y_S Z_S$) involves successive rotations:

YAW, ψ , about Z_S
 PITCH, θ , about Y_2
 ROLL, ϕ , about X_I (refer to Figure 4.7)

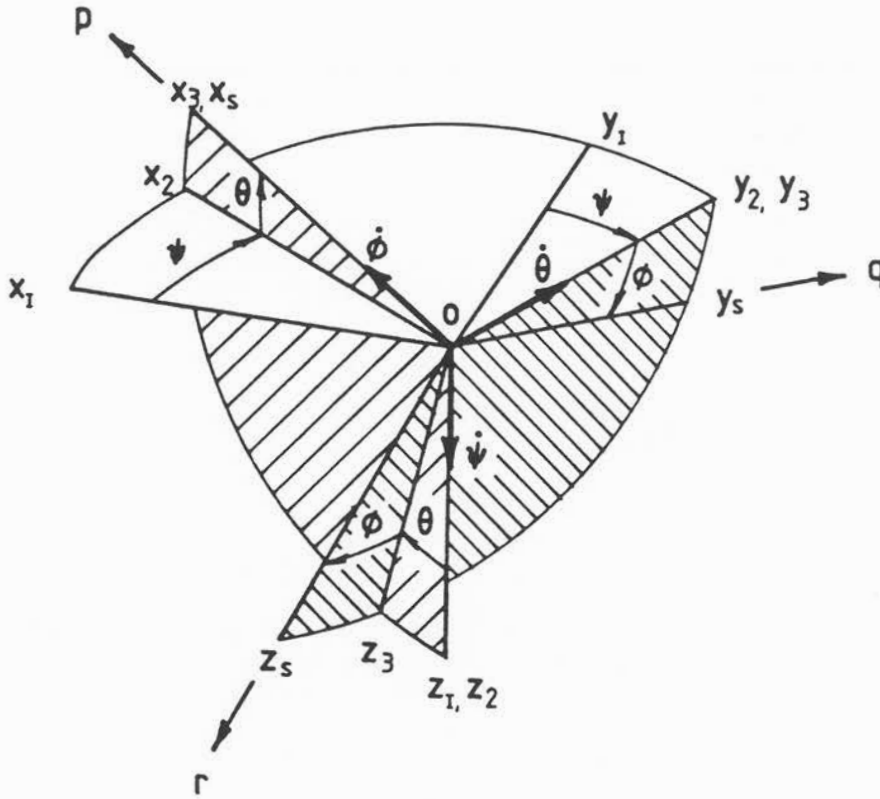


Figure 4.7 Angular Rate Transformations

Because the rotation angles are large, they are not commutative (as shown in Chapter 2) and therefore the defined order of rotations must be carefully observed. To be strictly correct, the final rotation, θ_0 , the roll axis inclination should be included. In practice, the maximum value for θ_0 is only 2.5° and can thus be considered as negligible in this regard.

The angular velocity of the sprung mass CG is defined as,

$$\underline{\omega} = p\underline{i}_s + q\underline{j}_s + r\underline{k}_s,$$

and angular rates given by,

$$\underline{\dot{\psi}} = \underline{k}_1 \dot{\psi} \quad \underline{\dot{\theta}} = \underline{j}_2 \dot{\theta} \quad \underline{\dot{\phi}} = \underline{i}_s \dot{\phi}$$

The roll rate, p , is the sum of the components of $\underline{\dot{\psi}}$, $\underline{\dot{\theta}}$, $\underline{\dot{\phi}}$ along OX_s :

$$p = \dot{\phi} - \dot{\psi} \sin \theta \quad (4.9)$$

$$\text{Similarly, } q = \dot{\theta} \cos \phi + \dot{\psi} \cos \theta \sin \phi \quad (4.10)$$

$$\text{and } r = \dot{\psi} \cos \theta \cos \phi - \dot{\theta} \sin \phi \quad (4.11)$$

In matrix form,

$$\begin{pmatrix} \dot{p} \\ \dot{q} \\ \dot{r} \end{pmatrix} = \begin{bmatrix} 1 & 0 & -\sin\theta \\ 0 & \cos\phi & \sin\phi\cos\theta \\ 0 & -\sin\phi & \cos\phi\cos\theta \end{bmatrix} \begin{pmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{pmatrix} \quad (4.12)$$

$$\text{Invert, } \begin{pmatrix} \dot{\phi} \\ \dot{\theta} \\ \dot{\psi} \end{pmatrix} = \begin{bmatrix} 1 & \sin\phi\tan\theta & \cos\phi\tan\theta \\ 0 & \cos\phi & -\sin\phi \\ 0 & \sin\phi/\cos\theta & \cos\phi/\cos\theta \end{bmatrix} \begin{pmatrix} \dot{p} \\ \dot{q} \\ \dot{r} \end{pmatrix} \quad (4.13)$$

$$\text{that is, } \dot{\phi} = \dot{p} + (q\sin\phi + r\cos\phi)\tan\theta \quad (4.14)$$

$$\dot{\theta} = q\cos\phi - r\sin\phi \quad (4.15)$$

$$\dot{\psi} = (r\cos\phi + q\sin\phi)/\cos\theta \quad (4.16)$$

4.5.3 Kinematic Relationships

The position of each CG and tyre contact point and the linear and angular velocities of each CG are required for the derivation of the equations of motion. The position and velocity of the point on the roll axis where the sprung mass and each axle are deemed to be joined (the front roll point, in the case of the front axle) is constrained to be equal for both rigid bodies.

Figure 4.8 indicates the various dimensions of the bus while stationary. This is used to establish certain relationships, before the dynamic case is considered.

The inclination of the roll axis, θ_0 , is assumed to be small,

$$\theta_0 = (hrc_f - hrc_r) / (a_f - a_r) \quad (4.17)$$

$$h_f = hrc_f - h_{hf} \quad (4.18)$$

$$h_r = hrc_r - h_{hr} \quad (4.19)$$

$$hrc_s = hrc_r - (hrc_f - hrc_r) \cdot a_r / (a_f - a_r) \quad (4.20)$$

$$b_s = h_{hs} - hrc_s \quad (4.21)$$

Transform this to the sprung mass axis system, using equation 4.8,

$$\begin{aligned} (\underline{R}_f^1)_s = & -z_f^1 \theta_o \underline{i}_s + (y_f^1 \cos \phi_f - z_f^1 \sin \phi_f) \underline{j}_s \\ & + (z_f^1 \cos \phi_f + y_f^1 \sin \phi_f) \underline{k}_s \end{aligned} \quad (4.26)$$

Add the sprung mass CG to front axle CG displacement, equation 4.23,

$$\begin{aligned} \underline{R}_s^1 = & [a_f - (b_s + h_f + z_f^1) \theta_o] \underline{i}_s \\ & + [(s_f + y_f^1) \cos \phi_f - (h_f + z_f^1) \sin \phi_f] \underline{j}_s \\ & + [b_s + (h_f + z_f^1) \cos \phi_f + (s_f + y_f^1) \sin \phi_f] \underline{k}_s \end{aligned} \quad (4.27)$$

4.5.3.3 Velocity of Front Right Tyre Contact Point

For simplicity, let the components in equation 4.27 be abbreviated to,

$$\underline{R}_s^1 = XBl \underline{i}_s + YBl \underline{j}_s + ZBl \underline{k}_s \quad (4.28)$$

The inertial velocity of a point in this rotating system is,

$$\underline{V}_{Is}^1 = \dot{\underline{R}}_s^1 + \omega^s \times \underline{R}_s^1 + \underline{V}^s \quad (4.29)$$

Differentiate equation 4.27 with respect to time,

$$\begin{aligned} \dot{\underline{R}}_s^1 = & -\dot{h}_f \theta_o \underline{i}_s + [\dot{s}_f - (s_f + y_f^1) \dot{\phi}_f - \dot{h}_f \phi_f - (h_f + z_f^1) \dot{\phi}_f] \underline{j}_s \\ & + [\dot{h}_f - (h_f + z_f^1) \dot{\phi}_f + \dot{s}_f \phi_f + (s_f + y_f^1) \dot{\phi}_f] \underline{k}_s \end{aligned} \quad (4.30)$$

Substitute into equation 4.29,

$$\begin{aligned} \underline{V}_s^1 = & [u - \dot{h}_f \theta_o + qZBl - rYBl] \underline{i}_s + [v + pb_s + \dot{s}_f \\ & - \dot{h}_f \phi_f - \{ \dot{h}_f + z_f^1 \dot{\phi}_f + \phi_f (s_f + y_f^1) \}] \underline{j}_s \\ & + [w + \dot{h}_f + \dot{s}_f \phi_f + \{ s_f + y_f^1 - \phi_f (h_f + z_f^1) \}] \underline{k}_s \\ & + pYBl - qXBl \underline{k}_s \end{aligned} \quad (4.31)$$

4.5.3.4 Linear Velocity of Sprung Mass

While the linear velocity components of the CG are u, v, w , the inertial velocity of the CG is given by (Figure 4.9):

$$\underline{V}^s = u \underline{i}_s + (v + pb_s) \underline{j}_s + w \underline{k}_s \quad (4.32)$$

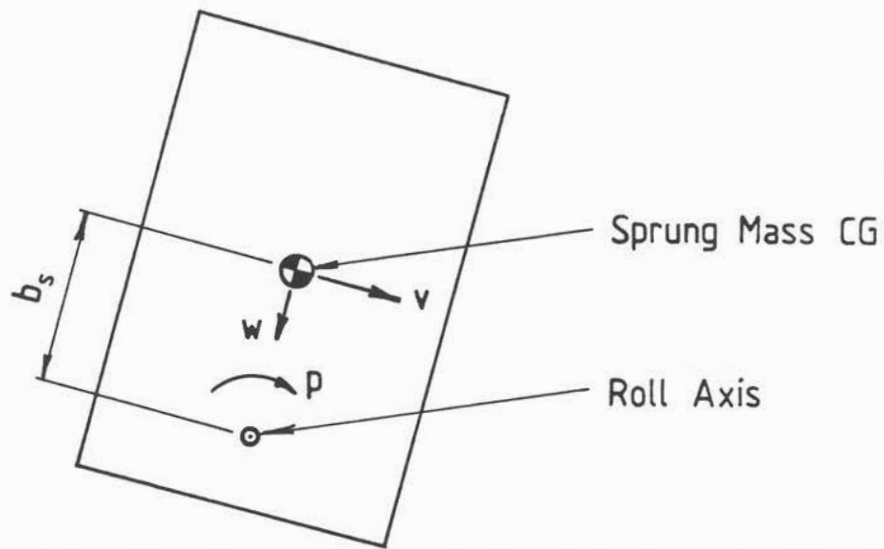


Figure 4.9 Calculation of Velocity of Sprung Mass CG

4.5.3.5 Angular Velocity of Sprung Mass

The inertial angular velocity of the sprung mass CG is given by,

$$\underline{\omega}^s = p\underline{i}_s + q\underline{j}_s + r\underline{k}_s \quad (4.33)$$

The linear velocity of the front roll point for the sprung mass is thus,

$$\underline{V}_{IS}^{rp} = (u + qb_s)\underline{i}_s + (v + ra_f)\underline{j}_s + (w - qa_f)\underline{k}_s \quad (4.34)$$

Express this in front axle axis set, using equation 4.8,

$$\begin{aligned} (\underline{V}_{IF}^{rp}) = & (u + qb_s)\underline{i}_f + [(v + ra_f)\cos\phi_f + (w - qa_f)\sin\phi_f]\underline{j}_f \\ & + [(w - qa_f)\cos\phi_f - (v + ra_f)\sin\phi_f]\underline{k}_f \end{aligned} \quad (4.35)$$

4.5.3.6 Linear Velocity of Front Axle

Define the inertial velocity of the front axle CG as,

$$\underline{V}^f = u_f\underline{i}_f + v_f\underline{j}_f + w_f\underline{k}_f \quad (4.36)$$

The linear velocity of the front roll point for the front axle is thus,

$$\underline{V}_f^{rp} = u_f\underline{i}_f + (v_f + p_f h_f)\underline{j}_f + (w_f - h_f)\underline{k}_f \quad (4.37)$$

Equate components with those in equation 4.35 to determine the inertial velocity of the front axle CG,

$$\begin{aligned}
\underline{\dot{V}}^f &= (u+qb_s)\underline{i}_f \\
&+ [(v+ra_f)\cos\phi_f + (w-qa_f)\sin\phi_f - p_f h_f]\underline{j}_f \\
&+ [(w-qa_f)\cos\phi_f - (v+ra_f)\sin\phi_f + h_f]k_f
\end{aligned} \quad (4.38)$$

4.5.3.7 Angular Velocity of Front Axle

$$\begin{aligned}
\underline{\omega}^f &= p_f \underline{i}_f + (q\cos\phi_f + r\sin\phi_f)\underline{j}_f \\
&+ (r\cos\phi_f - q\sin\phi_f)\underline{k}_f
\end{aligned} \quad (4.39)$$

where $p_f = p + \dot{\phi}_f$ (4.40)

4.5.4 Equations of Motion

4.5.4.1 Translational Equation for Sprung Mass

The linear form of Newton's second law of motion, expressed for a rigid body of constant mass, with rotating coordinates attached to the CG of the body is,

$$\underline{F} = m(\underline{\dot{V}} + \underline{\omega} \times \underline{V}) \quad (4.41)$$

Considering each component direction,

$$\underline{F}^S = F_x^S \underline{i} + F_y^S \underline{j} + F_z^S \underline{k} \quad (4.42)$$

Differentiate each component of sprung mass CG linear velocity (equation 4.32) with respect to time,

$$\underline{\dot{V}}^S = \dot{u}\underline{i}_s + (\dot{v} + \dot{p}b_s)\underline{j}_s + \dot{w}\underline{k}_s \quad (4.43)$$

Determine matrix product,

$$\underline{\omega}^S \times \underline{V}^S = (qw - rv - prb_s)\underline{i}_s + (ru - pw)\underline{j}_s + (pv + p_fb_s - qu)\underline{k}_s \quad (4.44)$$

Substitute 4.43 and 4.44 into equation 4.41,

$$\begin{aligned}
\underline{F}^S &= m_s(\dot{u} + qw - rv - prb_s)\underline{i}_s \\
&+ m_s(\dot{v} + \dot{p}b_s + ru - pw)\underline{j}_s \\
&+ m_s(\dot{w} + pv - qu + p_fb_s)\underline{k}_s
\end{aligned} \quad (4.45)$$

4.5.4.2 Rotational Equation for Sprung Mass

The equivalent form of Newton's second law of motion for the rotational case is,

$$\underline{\dot{T}}^S = \underline{\dot{H}}^S + \underline{\omega}^S \times \underline{H}^S \quad (4.46)$$

$$\text{where } \underline{H}^S = [I^S]_S \underline{\omega}^S \quad (4.47)$$

The inertia matrix is assumed to include no products of inertia, except the yaw-roll product. The angular momentum thus becomes,

$$\underline{H}^S = \begin{bmatrix} I_{xS}^S & 0 & I_{xzS}^S \\ 0 & I_{yS}^S & 0 \\ I_{xzS}^S & 0 & I_{zS}^S \end{bmatrix} \begin{pmatrix} p \\ q \\ r \end{pmatrix} \quad (4.48)$$

$$\begin{aligned} \text{thus, } \underline{H}^S &= (I_{xS}^S p + I_{xzS}^S r) \underline{i}_S + (I_{yS}^S q) \underline{j}_S \\ &\quad + (I_{zS}^S r + I_{xzS}^S p) \underline{k}_S \end{aligned} \quad (4.49)$$

Differentiating the angular momentum with respect to time gives,

$$\begin{aligned} \underline{\dot{H}}^S &= (I_{xS}^S \dot{p} + I_{xzS}^S \dot{r}) \underline{i}_S + (I_{yS}^S \dot{q}) \underline{j}_S \\ &\quad + (I_{zS}^S \dot{r} + I_{xzS}^S \dot{p}) \underline{k}_S \end{aligned} \quad (4.50)$$

The moments of inertia are constant with time, since the axis system rotates with the sprung mass. Determine the matrix product,

$$\begin{aligned} \underline{\omega}^S \times \underline{H}^S &= [(I_{zS}^S - I_{yS}^S)qr + I_{xzS}^S pq] \underline{i}_S \\ &\quad + [(I_{xS}^S - I_{zS}^S)pr + I_{xzS}^S (r^2 - p^2)] \underline{j}_S \\ &\quad + [(I_{yS}^S - I_{xS}^S)pq - I_{xzS}^S qr] \underline{k}_S \end{aligned} \quad (4.51)$$

Substitute 4.50 and 4.51 into equation 4.46,

$$\begin{aligned} \underline{\dot{T}}^S &= [I_{xS}^S \dot{p} + I_{xzS}^S (\dot{r} + pq) + (I_{zS}^S - I_{yS}^S)qr] \underline{i}_S \\ &\quad + [I_{yS}^S \dot{q} + (I_{xS}^S - I_{zS}^S)pr + I_{xzS}^S (r^2 - p^2)] \underline{j}_S \\ &\quad + [I_{zS}^S \dot{r} + I_{xzS}^S (\dot{p} - qr) + (I_{yS}^S - I_{xS}^S)pq] \underline{k}_S \end{aligned} \quad (4.52)$$

4.5.4.3 Translational Equation for Front Axle

Express equation 4.41 in the form for the front axle,

$$\underline{\ddot{F}}^f = m_f (\underline{\ddot{V}}^f + \underline{\omega}^f \times \underline{V}^f) \quad (4.53)$$

After the same process of differentiation and combination of matrices obtain,

$$\begin{aligned}
 \underline{F}^f = & m_f [\dot{u} + \dot{q}b_f + qw - rv - a_f(q^2 + r^2) \\
 & + (\dot{h}_f + p_f s_f)(q + r\phi_f) - (\dot{s}_f - p_f h_f)(r - q\phi_f)] \underline{i}_f \\
 & + m_f [\dot{v} + \dot{w}\phi_f + \ddot{s}_f - \dot{p}h_f - \dot{q}a_f\phi_f + \dot{r}a_f - \ddot{\phi}_f h_f \\
 & - 2p_f \dot{h}_f - p_f^2 s_f + ru - pw + qrb_s + pqa_f \\
 & + \phi_f(pv - qu + pra_f - q^2 b_f)] \underline{j}_f \\
 & + m_f [\dot{w} - \dot{v}\phi_f + \ddot{h}_f + \dot{p}s_f - \dot{q}a_f - \dot{r}a_f\phi_f + \ddot{\phi}_f s_f \\
 & + 2p_f \dot{s}_f - p_f^2 h_f + pv - qu + pra_f - q^2 b_s \\
 & - \phi_f(ru - pw + qrb_s + pqa_f)] \underline{k}_f
 \end{aligned} \quad (4.54)$$

4.5.4.4 Rotational Equation for Front Axle

Use a similar procedure to that already described in Section 4.5.4.2

$$\underline{T}^f = \underline{\dot{H}}^f + \underline{\omega}^f \times \underline{H}^f \quad (4.55)$$

$$\text{where } \underline{H}^f = [I^f]_f \underline{\omega}^f \quad (4.56)$$

The front axle inertia matrix is assumed to include no products of inertia, that is, the axis directions are assumed to be principal axes. The angular momentum is thus,

$$\underline{H}^f = \begin{bmatrix} I_{xf} & 0 & 0 \\ 0 & I_{yf} & 0 \\ 0 & 0 & I_{zf} \end{bmatrix} \begin{pmatrix} p + \dot{\phi}_f \\ q \cos \phi_f + r \sin \phi_f \\ r \cos \phi_f - q \sin \phi_f \end{pmatrix} \quad (4.57)$$

$$\begin{aligned}
 \text{Hence, } \underline{H}^f = & I_{xf}^f (p + \dot{\phi}_f) \underline{i}_f + I_{yf}^f (q \cos \phi_f + r \sin \phi_f) \underline{j}_f \\
 & + I_{zf}^f (r \cos \phi_f - q \sin \phi_f) \underline{k}_f
 \end{aligned} \quad (4.58)$$

Differentiating and combining leads similarly to,

$$\begin{aligned}
 \underline{T}^f = & [I_{xf}^f (\dot{p} + \ddot{\phi}_f) + (I_{zf}^f - I_{yf}^f) \{qr + \phi_f(r^2 - q^2)\}] \underline{i}_f \\
 & + [I_{yf}^f \{\dot{q} + r\dot{\phi}_f + \dot{\phi}_f(r - q\phi_f)\} + (I_{xf}^f - I_{zf}^f) p_f (r - q\phi_f)] \underline{j}_f \\
 & + [I_{zf}^f \{\dot{r} - q\dot{\phi}_f - \dot{\phi}_f(q + r\phi_f)\} + (I_{yf}^f - I_{xf}^f) p_f (q + r\phi_f)] \underline{k}_f
 \end{aligned} \quad (4.59)$$

4.5.4.5 Linear Equations of Motion

The longitudinal equation for the whole vehicle is obtained by adding the X components of the linear equations of motion for the sprung mass and axles,

$$\begin{aligned}\Sigma F_x &= F_x^s + F_x^f + F_x^r \\ \Sigma F_x &= m_s(\dot{u} + q\dot{w} - r\dot{v} - p\dot{r}b_s) + m_f[\dot{u} + \dot{q}b_s + q\dot{w} - r\dot{v} - a_f(q^2 + r^2) \\ &\quad + (\dot{h}_f + p_f s_f)(q + r\phi_f) - (\dot{s}_f - p_f h_f)(r - q\phi_f)] + m_r[\dots] \quad (4.60)\end{aligned}$$

$$\begin{aligned}\Sigma F_x &= m(\dot{u} + q\dot{w} - r\dot{v}) + \dot{q}b_s(m_f + m_r) - (q^2 + r^2)(m_f a_f + m_r a_r) \\ &\quad + m_f[(\dot{h}_f + p_f s_f)(q + r\phi_f) - (\dot{s}_f - p_f h_f)(r - q\phi_f)] \\ &\quad + m_r[(\dot{h}_r + p_r s_r)(q + r\phi_r) - (\dot{s}_r - p_r h_r)(r - q\phi_r)] \quad (4.61)\end{aligned}$$

The lateral equation of motion for the sprung mass is (from equation 4.45),

$$\Sigma F_y^s = m_s(\dot{v} + \dot{p}b_s + ru - pw) \quad (4.62)$$

The vertical equation of motion for the sprung mass is (from equation 4.45),

$$\Sigma F_z^s = m_s(\dot{w} + pv - qu + p^2 b_s) \quad (4.63)$$

The lateral equation of motion for the front axle is (from equation 4.54),

$$\begin{aligned}\Sigma F_y^f &= m_f[\dot{v} + \dot{w}\phi_f + \ddot{s}_f - \dot{p}h_f - \dot{q}a_f\phi_f + \dot{r}a_f - \ddot{\phi}_f h_f \\ &\quad - 2p_f \dot{h}_f - p_f^2 s_f + ru - pw + qrb_s + pqa_f \\ &\quad + \phi_f(pv - qu + pra_f - q^2 b_s)] \quad (4.64)\end{aligned}$$

The vertical equation of motion for the front axle is (from equation 4.54),

$$\begin{aligned}\Sigma F_z^f &= m_f[\dot{w} - \dot{v}\phi_f + \ddot{h}_f + \dot{p}s_f - \dot{q}a_f - \dot{r}a_f\phi_f + \ddot{\phi}_f s_f \\ &\quad + 2p_f \dot{s}_f - p_f^2 h_f + pv - qu + pra_f - q^2 b_s \\ &\quad - \phi_f(ru - pw + qrb_s + pqa_f)] \quad (4.65)\end{aligned}$$

4.5.4.6 Rotational Equations of Motion

The roll equation of motion for the sprung mass is obtained by adding the appropriate force equations x distance, to the roll component of equation 4.52,

$$\underline{T}_S = \underline{T}_S^S + \underline{R}_S^S \underline{x}_S^S \quad (4.66)$$

The sprung mass CG is separated from the roll axis by,

$$\underline{R}_S^S = -b_S k_S \quad (4.67)$$

The roll equation for the sprung mass is thus,

$$\begin{aligned} \Sigma T_{x_S} = & I_{x_S}^S \ddot{p} + I_{xz_S}^S (\dot{r} + pq) + (I_{z_S}^S - I_{y_S}^S) qr \\ & + m_S b_S (\dot{v} + p b_S + ru - pw) \end{aligned} \quad (4.68)$$

Pitch and yaw are described as taking place about the sprung mass CG, and therefore involve no significant force x distance components. The pitch equation of motion for the sprung mass is (from equation 4.52),

$$\Sigma T_{y_S} = I_{y_S}^S \ddot{q} + (I_{x_S}^S - I_{z_S}^S) pr + I_{xz_S}^S (r^2 - p^2) \quad (4.69)$$

The yaw equation of motion for the sprung mass is (from equation 4.52),

$$\Sigma T_{z_S} = I_{z_S}^S \dot{r} + I_{xz_S}^S (\dot{p} - qr) + (I_{y_S}^S - I_{x_S}^S) pq \quad (4.70)$$

To determine the roll equation of motion for the front axle,

$$\underline{R}_f^f = s_f j_f + h_f k_f \quad (4.71)$$

The roll equation of motion for the front axle is thus,

$$\begin{aligned} \Sigma T_{x_f} = & I_{x_f}^f (\ddot{p} + \ddot{\phi}_f) + (I_{z_f}^f - I_{y_f}^f) [qr + \phi_f (r^2 - q^2)] \\ & + m_f s_f [-\dot{v} \phi_f + \dot{w} + \ddot{h}_f + \dot{p} s_f - \dot{q} a_f - \dot{r} a_f \phi_f + \ddot{\phi}_f s_f + pv - qu + 2p_f \dot{s}_f \\ & - p_f^2 h_f + pra_f - q b_S - \phi_f (ru - pw + qrb_S + pqa_f)] \\ & - m_f h_f [\dot{v} + \dot{w} \phi_f + \ddot{s}_f - \dot{p} h_f - \dot{q} a_f \phi_f + \dot{r} a_f - h_f \ddot{\phi}_f + ru - pw - 2p_f \dot{h}_f \\ & - p_f^2 s_f + qrb_S + pqa_f + \phi_f (pv - qu + pra_f - q^2 b_S)] \end{aligned} \quad (4.72)$$

hence, $\Sigma T_{X_f} = -m_f(h_f + s_f \phi_f) \dot{v} + m_f(s_f - h_f \phi_f) \dot{w} - m_f h_f \ddot{s}_f + m_f s_f \ddot{h}_f$
 $+ (I_{X_f}^f + m_f s_f^2 + m_f h_f^2)(\ddot{p} + \ddot{\phi}_f) - m_f(s_f - h_f \phi_f) a_f \dot{q}$
 $- m_f a_f(h_f + s_f \phi_f) \dot{r} + 2m_f(s_f \dot{s}_f + h_f \dot{h}_f) p_f$
 $+ m_f(s_f - h_f \phi_f)(pv - qu + pra_f - q^2 b_s)$
 $- m_f(h_f + s_f \phi_f)(ru - pw + qrb_s + pqa_f)$
 $+ (I_{Z_f}^f - I_{Y_f}^f)[qr + \phi_f(r^2 - q^2)]$ (4.73)

4.5.4.7 Summary

These twelve equations of motion can be assembled in to a matrix by first writing each one in the form,

$$C_{i1}X_1 + C_{i2}X_2 + C_{i3}X_3 + \dots + C_{ij}X_j = F_i \quad (4.74)$$

Note also that the total mass is given by m.

That is, $m = m_s + m_f + m_r$ (4.75)

$$1. \quad m \dot{u} + (m_f + m_r) b_s \dot{q} = \Sigma F_x - m(qw - rv) + (q^2 + r^2)(m_f a_f + m_r a_r)$$

$$- m_f[(\dot{h}_f + p_f s_f)(q + r \phi_f) - (\dot{s}_f - p_f h_f)(r - q \phi_f)]$$

$$- m_r[(\dot{h}_r + p_r s_r)(q + r \phi_r) - (\dot{s}_r - p_r h_r)(r - q \phi_r)] \quad (4.76)$$

$$2. \quad m_s \dot{v} + m_s b_s \dot{p} = \Sigma F_y^s - m_s(ru - pw) \quad (4.77)$$

$$3. \quad m_s \dot{w} = \Sigma F_z^s - m_s(pv - qu + p^2 b_s) \quad (4.78)$$

$$4. \quad m_f \dot{v} + m_f \phi_f \dot{w} + m_f \ddot{s}_f - m_f h_f \dot{p} - m_f a_f \phi_f \dot{q} + m_f a_f \dot{r} - m_f h_f \ddot{\phi}_f$$

$$= \Sigma F_y^f + 2p_f \dot{h}_f + p_f^2 s_f - ru + pw - qrb_s - pqa_f$$

$$- \phi_f(pv - qu + pra_f - q^2 b_s)] \quad (4.79)$$

The fifth equation is identical to equation 4 except with the F subscripts changed to R. This also applies to equations 7 and 12.

$$6. \quad -m_f \phi_f \dot{v}_f + m_f \dot{w} + m_f \ddot{h}_f + m_f s_f \dot{p} - m_f a_f \dot{q} - m_f a_f \phi_f \dot{r} + m_f s_f \ddot{\phi}_f$$

$$= \Sigma F_z - 2p_f \dot{s}_f + p_f^2 h_f - pv + qu - pra_f + q^2 b_s$$

$$+ \phi_f(ru - pw + qrb_s + pqa_f)] \quad (4.80)$$

$$8. \quad m_s b_s \dot{v} + (I_{xs}^s + m_s b_s) \dot{p} + I_{xzs}^s \dot{r} = \Sigma T_{X_s} - I_{xzs}^s pq$$

$$- (I_{zs} - I_{ys})qr - m_s b_s(ru - pw) \quad (4.81)$$

$$9. \quad I_{ys}^s \dot{q} = \Sigma T_{ys} - (I_{xs}^s - I_{zs}^s)pr - I_{xzs}^s(r^2 - p^2) \quad (4.82)$$

$$10. \quad I_{xzs}^s \dot{p} + I_{zs}^s \dot{r} = \Sigma T_{zs} + I_{xzs}^s qr - (I_{ys}^s - I_{xs}^s)pq \quad (4.83)$$

$$\begin{aligned} 11. \quad & -m_f(h_f + s_f \phi_f) \dot{v} + m_f(s_f - h_f \phi_f) \dot{w} - m_f h_f \ddot{s}_f + m_f s_f \ddot{h}_f \\ & + (I_{xf}^f + m_f s_f^2 + m_f h_f^2) \dot{p} - m_f(s_f - h_f \phi_f) a_f \dot{q} - m_f a_f (h_f + s_f \phi_f) \dot{r} \\ & + (I_{xf}^f + m_f s_f^2 + m_f h_f^2) \ddot{\phi}_f = \Sigma T_{xf} - 2m_f(s_f \dot{s}_f + h_f \dot{h}_f) p_f \\ & - m_f(s_f - h_f \phi_f)(pv - qu + pra_f - q^2 b_s) \\ & + m_f(h_f + s_f \phi_f)(ru - pw + qrb_s + pqa_f) \\ & - (I_{zf}^f - I_{yf}^f)[qr + \phi_f(r^2 - q^2)] \end{aligned} \quad (4.84)$$

These equations are assembled into matrix form,

$$[C].(\ddot{x}) = (F) \quad (4.85)$$

where [C] is a 12x12 matrix of coefficients from the left hand sides of equations 4.76 to 4.84, $(\ddot{x})$ are the 12 degrees of freedom in their "acceleration" form, and (F) are the 12 net forces given by the right hand sides of the same equations.

4.6 APPLIED FORCES AND MOMENTS

4.6.1 Gravity

Gravity acts vertically downwards, which expressed as a unit vector is,

$$\underline{c} = \begin{pmatrix} 0 \\ 0 \\ 1 \end{pmatrix} \quad (4.86)$$

Transforming this into the sprung mass axis set using equation 4.4 gives,

$$\underline{c} = [DIR] \begin{pmatrix} 0 \\ 0 \\ 1 \end{pmatrix} \quad (4.87)$$

$$\text{or,} \quad \underline{c} = \begin{pmatrix} DIR(3,1) \\ DIR(3,2) \\ DIR(3,3) \end{pmatrix} \quad (4.88)$$

The gravity force for the first three equations of motion are respectively,

$$F_G(x) = mg.DIR(3,1) \quad (4.89)$$

$$F_G(y) = m_s g \cdot \text{DIR}(3,2) \quad (4.90)$$

$$F_G(z) = m_s g \cdot \text{DIR}(3,3) \quad (4.91)$$

(The letter in brackets indicates the degree of freedom to which the force applies.)

Gravity is considered to act through the CG (by definition) of a rigid body. It will therefore generate no moments about the CG. For this model, however, the roll equations have been written assuming roll occurs about the roll axis. The roll moment on the sprung mass due to gravity is thus,

$$F_G(\phi) = F_G(y) \cdot b_s \quad (4.92)$$

Equations for the axles describe their motion relative to that of the sprung mass. The forces involved are therefore also relative to the sprung mass. Gravity forces, then, are due only to the weight of the axles, resolved into the appropriate directions. The transformation is obtained by combining equations 4.8 and 4.88,

$$c = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & \phi_f \\ 0 & -\phi_f & 1 \end{bmatrix} \begin{pmatrix} \text{DIR}(3,1) \\ \text{DIR}(3,2) \\ \text{DIR}(3,3) \end{pmatrix} \quad (4.93)$$

$$\text{or,} \quad c = \begin{pmatrix} \text{DIR}(3,1) \\ \text{DIR}(3,2) + \phi_f \text{DIR}(3,3) \\ -\phi_f \text{DIR}(3,2) + \text{DIR}(3,3) \end{pmatrix} \quad (4.94)$$

The gravity components at the front axle CG in the lateral and vertical directions (respectively) are thus,

$$F_G(s_f) = m_f g [\text{DIR}(3,2) + \phi_f \text{DIR}(3,3)] \quad (4.95)$$

$$F_G(h_f) = m_f g [-\phi_f \text{DIR}(3,2) + \text{DIR}(3,3)] \quad (4.96)$$

and, similarly, the roll moment for the front axle is,

$$F_G(\phi_f) = -F_G(s_f) \cdot h_f + F_G(h_f) \cdot s_f \quad (4.97)$$

4.6.2 Tyre Reaction Forces

Tyre reaction forces are calculated, not on the basis of weight transfer or similar equations, but using the combined vertical tyre and road stiffness and the most recent value of the tyre compression. The latter is determined as shown in Figure 4.10. A tyre is modelled as a thin disk, which is permitted to penetrate the uniform horizontal ground plane. Obviously, if the tyre is entirely above the ground plane, the tyre compression and

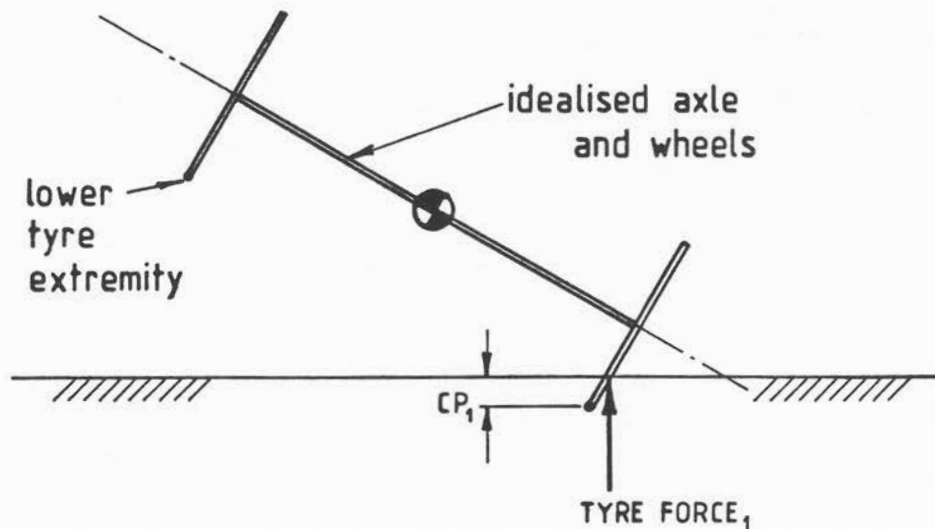


Figure 4.10 Method for Determining Vertical Tyre Compression

reaction force are zero. If, however, the lower tyre extremity lies below the road surface, the penetration distance (CP) gives the effective compression.

Except for the longitudinal equation of motion which is written for the whole vehicle (including axles), the tyre forces are considered to act only on the axles. The direction is opposite to that of gravity, and therefore if negative tyre reactions are considered, then similar relationships to those in Section 4.6.1 may be used.

4.6.3 Kerb Impact Forces

Adopting the same type of approach as used for calculating the tyre reactions, the kerb impact forces are determined from combined lateral tyre and kerb stiffness and calculation of the lateral tyre compression. Figure 4.11 indicates the application of the same principle - compression is again evaluated from the perpendicular penetration distance into the kerb of a point on the tyre.

The difference this time is that the point specified is on the kerb (the Kerb Extremity Point), rather than on the tyre. The kerb is assumed to be 100mm high, and the kerb impact force to act horizontally through this point.

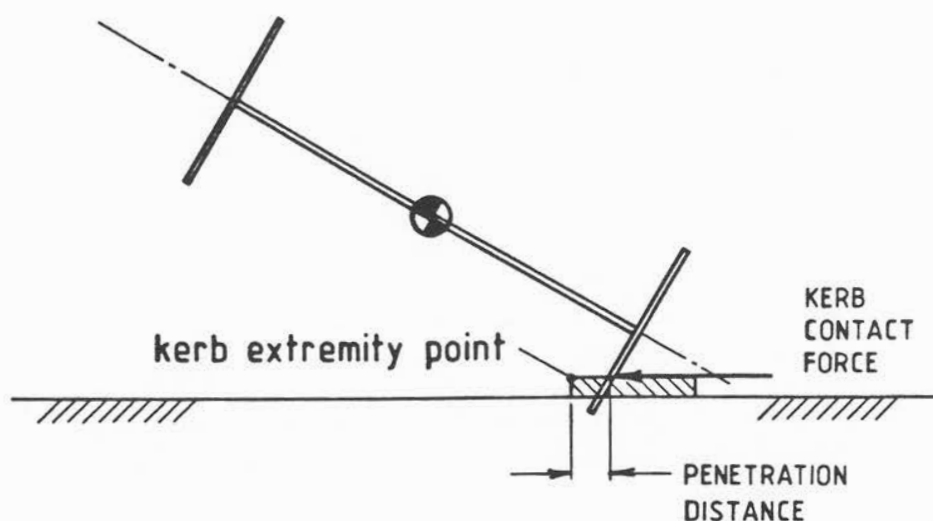


Figure 4.11 Method for Determining Lateral Tyre Compression

Work by Sadeghi and Tidbury (1979) and conclusions from Chapter 2 have highlighted the importance of allowing for energy absorption in the kerb impact. Figure 4.5 shows the force v deflection curve for a typical tyre/kerb impact as assumed in the simulation. The enclosed area in the graph represents energy absorbed, mainly as plastic deformation at a steady collapse force, in the manner of idealised crush tubes. The collapse load used is arbitrary. 500 kN, as used in SIDBURT (chapter 2) and corresponding approximately to measured dynamic collapse forces in superstructure strength rollover tests, was employed.

The global direction for the kerb impact forces is given by the unit vector,

$$\tilde{c} = \begin{pmatrix} 0 \\ -1 \\ 0 \end{pmatrix} \quad (4.98)$$

Like the tyre reaction forces, the kerb impact forces affect principally the axles. As the various suspension components displace, the suspension forces and moments are the means by which the large external forces are transmitted to the sprung mass.

4.6.4 Suspension Forces

4.6.4.1 Roll Moment

Figure 4.4 describes the variation in torque applied (equally and in opposite directions) to the sprung mass and an axle, as the relative axle angle is increased slowly. At higher speeds, the damping becomes significant, too. Expressed mathematically, if the roll angle is less than 15 degrees,

$$FS_{Rf} = \phi_f \cdot Q_f + \dot{\phi}_f \cdot D_f \quad (4.99)$$

Or, if the roll angle is at least 15 degrees,

$$FS_{Rf} = \phi_f \cdot Q_f + (\phi_f - 15/57.3) \cdot Q_B + \dot{\phi}_f \cdot D_f \quad (4.100)$$

where Q_B (Nm/rad) is the roll stiffness when a bump stop is contacted, and ϕ_f , the axle roll angle is expressed in radians.

The applied forces due the axle roll are,

$$F_R(\phi_f) = -FS_{Rf} \quad (4.101)$$

$$F_R(\phi_r) = -FS_{Rr} \quad (4.102)$$

$$F_R(\phi) = FS_{Rf} + FS_{Rr} \quad (4.103)$$

4.6.4.2 Lateral Force

The lateral suspension force is determined simply using Hooke's Law plus the viscous damping component. Thus,

$$FS_{Yf} = s_f \cdot K_{yf} + \dot{s}_f \cdot C_{yf} \quad (4.104)$$

Figure 4.12 sketches three views of a bus to indicate the correct direction of the applied lateral forces, to assist in evaluating their contribution to the force terms in the equations of motion.

The influence of the forces on the axles is limited to the lateral degree of freedom (since the lateral force is defined as acting through the roll axis, there is no contribution to the axle roll moment),

$$F_Y(s_f) = -FS_{Yf} \quad (4.105)$$

$$F_Y(s_r) = -FS_{Yr} \quad (4.106)$$

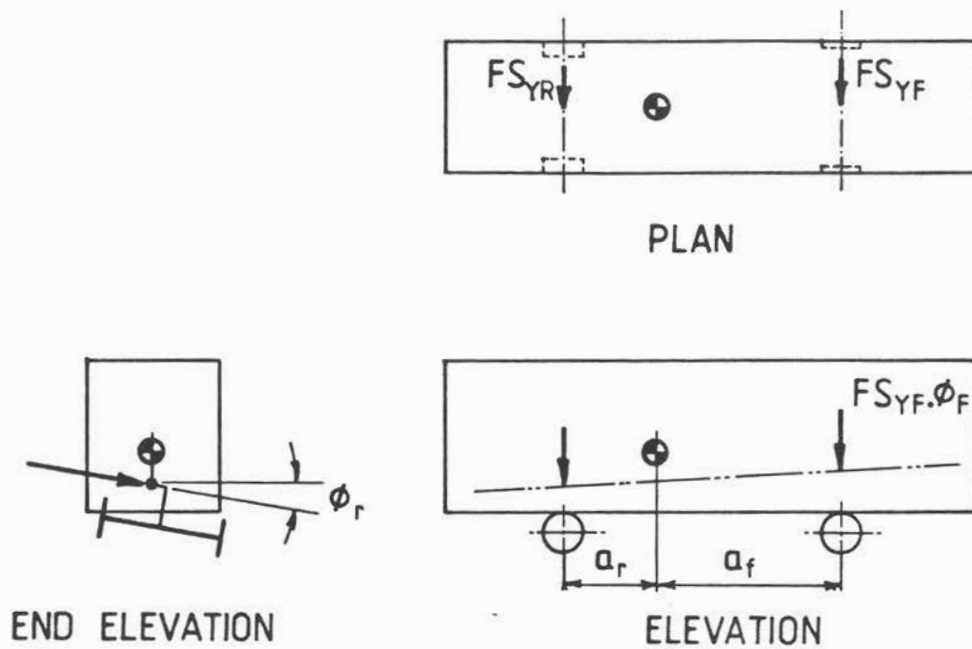


Figure 4.12 Direction of Lateral Suspension Forces Applied to Sprung Mass

The force on the sprung mass has components in the y and z directions, which also contribute to the pitch and yaw moments,

$$F_Y(y) = FS_{Yf} + FS_{Yr} \quad (4.107)$$

$$F_Y(z) = FS_{Yf} \cdot \phi_f + FS_{Yr} \cdot \phi_r \quad (4.108)$$

$$F_Y(\theta) = FS_{Yf} \cdot \phi_f \cdot a_f + FS_{Yr} \cdot \phi_r \cdot a_r \quad (4.109)$$

$$F_Y(\psi) = FS_{Yf} \cdot a_f + FS_{Yr} \cdot a_r \quad (4.110)$$

4.6.4.3 Vertical Force

The vertical suspension force at the front axle is given by,

$$FS_{Zf} = (h_f - h_{f0}) \cdot K_{Zf} - FS_{f0} + \dot{h}_f \cdot C_{Zf} \quad (4.111)$$

where $h_{f0} = h_f$ at time = 0
and, $FS_{f0} = FS_f$ at time = 0

Resolving both vertical suspension forces into the component directions gives,

$$F_Z(y) = -FS_{Zf} \cdot \phi_f - FS_{Zr} \cdot \phi_r \quad (4.112)$$

$$F_Z(z) = FS_{Zf} + FS_{Zr} \quad (4.113)$$

$$F_Z(\Theta) = FS_{Zf} \cdot a_f + FS_{Zr} \cdot a_r \quad (4.114)$$

$$F_Z(\psi) = FS_{Zf} \cdot \phi_f \cdot a_f + FS_{Zr} \cdot \phi_r \cdot a_r \quad (4.115)$$

The influence of the forces on the axles includes, in addition to the vertical force, a roll moment about the roll axis due to the lateral offset:

$$F_Z(h_f) = -FS_{Zf} \quad (4.116)$$

$$F_Z(h_r) = -FS_{Zr} \quad (4.117)$$

$$F_Z(\phi_f) = -FS_{Zf} \cdot s_f \quad (4.118)$$

$$F_Z(\phi_r) = -FS_{Zr} \cdot s_r \quad (4.119)$$

4.6.5 Total Forces and Moments

Separately for each equation of motion, these forces and moments are added together, to give the corresponding ΣF terms in equations 4.76 to 4.84. The remainder of the right hand side of each equation of motion, known as the inertia force, is added to the ΣF (or ΣT) term, and results in the net force, which appears in the column vector ($\underline{F}$) of equation 4.85.

4.7 METHOD OF SOLUTION

An overview of the program structure is given in Figure 4.13, a generalised flow diagram of the rollover accident simulation program.

The simulation is written in FORTRAN 77. Program development was done on a VAX 782, although some simulations were executed on VAX 750. Average CPU time varies with bus properties, but for typical vehicles is from 15 to 30 minutes.

The various stages of the rollover accident simulation program are relatively straightforward and outlined as follows:

4.7.1 Subroutine INTEGRATE

Numerical integration is employed to determine the velocity of each degree of freedom from its acceleration produced using the equations of motion.

The Runge-Kutta fourth-order method of integration is frequently used in dynamic simulations, such as Weir, et al (1974) and Openshaw (1981) mainly because of its simplicity and self-starting capability. Explanation of the method is given in many mathematics textbooks, for example, Hosking, et al (1978),

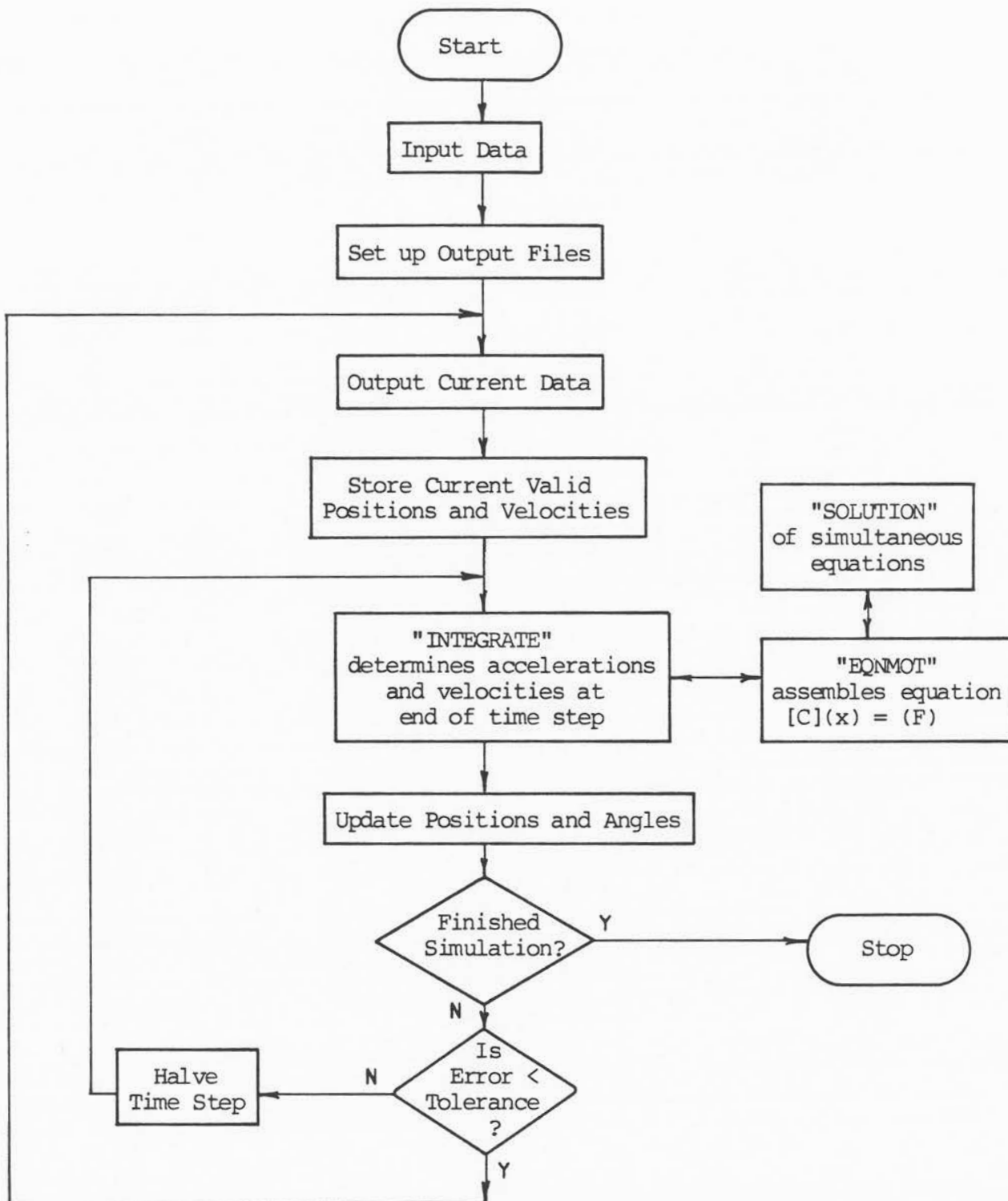


Figure 4.12 Flow Diagram for RASP

Ribbans (1969), Gault, et al (1973). Other numerical integration routines were considered, and some tested. One approach attempted, attributed to Richardson (Bender and Orszag, 1978) uses the fact that the accuracy of the Runge-Kutta increases as the integration step length (that is, the time increment for dynamics problems) is decreased. The true solution is assumed to be found as the step length tends to zero. If the solution at a particular point is evaluated using two or more step lengths, their solutions can be used to predict the solution for a zero step length with aid of a linear or higher order extrapolation (Figure 4.14). The theory is clever, but in practice the greater number of calculations and hence, computer time, required for the multiple solutions tended to outweigh the increased accuracy.

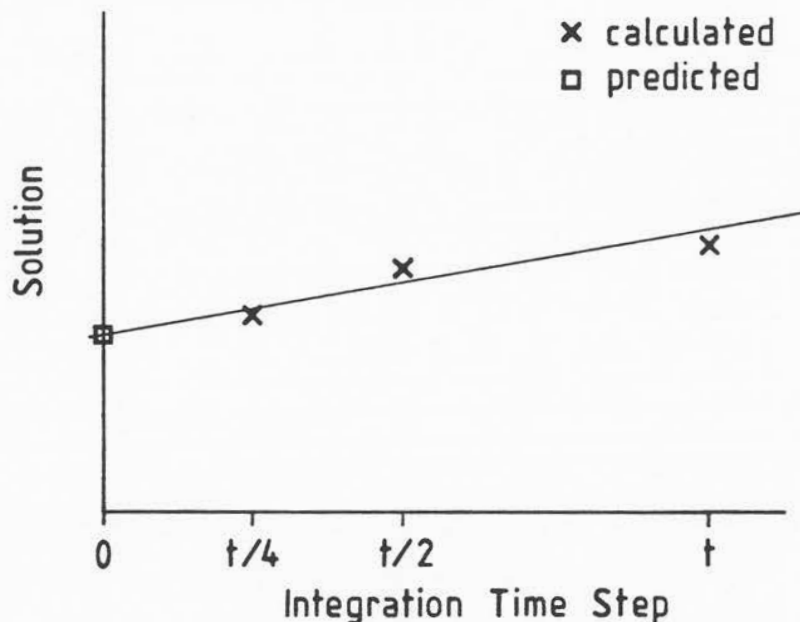


Figure 4.14 Richardson Approach Applied to Numerical Integration

The disadvantage of Runge-Kutta numerical integration is its lack of error checking. Openshaw (1981) overcame this by calculating current and predicted velocities, displacements and forces at each time step. If the difference between a current force magnitude and the value predicted for it from the last time interval exceeded a specified tolerance, the time step was halved. This became somewhat clumsy as the expressions for each force became more complex.

The compromise chosen for RASP between speed, accuracy, ease of error checking, and most of all, dependability, was the Runge-Kutta fourth-order method, determining the new solution after time step Δt and comparing the linear and angular velocities with those deduced from 2 steps of size $\Delta t/2$. Tolerance limits specified by the user on the size of these velocity differences indicate when the step length should be halved. If 50 time steps pass with no reduction in the time increment then it is adjudged to be too small, and is increased by 10%. Thus the time step regulates itself between acceptable limits set by the user. A very small time increment (approximately 50 μs) is necessary during the initial stages of the kerb impact, but this rises to typically about 2 ms as the impact forces decrease.

4.7.2 Subroutine EQNMOT

EQNMOT is called by INTEGRATE to assemble the matrix equation (4.85). Subroutine SOLUTION employs the Gauss elimination matrix method of solving the simultaneous equations to give the current accelerations for each degree of freedom. (Row and column swapping is not considered necessary due to the relative dominance of the terms on the diagonal).

4.7.3 Conditions for Halting Simulation

After each time step the following tests are performed, to determine whether the simulation should continue.

4.7.3.1 Rollover

The highest measured static roll angle of a PSV body relative to the ground was almost 45 degrees (Table 3.7). It is therefore not necessary to continue the simulation to 90 degrees of body roll angle, in order to conclude that rollover has taken place. If the adopted limit of 50 degrees is surpassed, that vehicle is assessed to have passed its stable limit and will overturn.

4.7.3.2 No Rollover

Alternatively, if the kerb-impact velocity is less than the threshold value, the bus will reach a maximum roll angle and fall back. If the current roll angle is more than 6 degrees less than the maximum roll angle, this indicates a "no rollover" case, and the simulation terminates.

4.7.3.3 Time Limit Expired

As a precaution against excessive computation resulting from erroneous input data, the user is able to specify a real time limit above which the program will stop and store sufficient data to enable the simulation to be restarted, if desired. (This restart facility is available after any one of these halt conditions. However if the execution is interrupted by the user or by crashing of the host computer, then this feature is lost.)

4.7.3.4 Computational Difficulties

If the time increment is continually halved to well below the typical values in order to bring the velocity errors within the specified tolerances, this usually indicates numerical problems, commonly caused by mistakes in the formulation of equations of motion or applied forces. In order to avoid excessive computation at very small time intervals in these cases during program development a minimum value for the time step was specified. If this limit is reached, the simulation is terminated with an appropriate error message.

One further precautionary test is performed after each integration step. That is to examine the total energy of the system, and stop the program if it exceeds an upper bound of 1000 kJ. Most simulations begin with about 250-420 kJ and decrease as energy is absorbed by hydraulic dampers and plastic deformation at the kerb/tyre interface. If the input data has not been correctly prepared, it is possible for numerical instability to occur, which usually results in the total system energy rising rapidly.

4.8 VALIDITY

A full-scale instrumented rollover test was not feasible due to resource limitations, so the simulation could not be checked directly. However, the work on the dynamic behaviour of buses during superstructure strength tests described in Chapter 2 shows reasonable correlation between measurements of actual tests and results predicted by the six degree-of-freedom model, SIDBURT (Figure 2.11). Since RASP contains many routines derived from the earlier model, including the numerical integration subroutine, the kerb-impact algorithm and the calculation of wheel forces from spring displacement considerations, rather than using weight transfer methods, the correlation with actual tests should be of a similar order.

Two theoretical methods were devised for checking the program further.

4.8.1 Comparison of Lateral Acceleration Simulations

The first was to use RASP to simulate a "steady-state" cornering test, in which the lateral acceleration was slowly increased (at a rate of $1g$ in 50 seconds) until each axle lifted. The results are compared in Table 4.2 with those obtained from a computer analysis which uses a static model. Figure 4.15 shows the similarity between quasi-static forces on a vehicle during a tilt test (a simulation of which is described in the following chapter) and those encountered during steady cornering. STATLAT, which models the latter, is derived directly from the tilt test simulation program, with only a few changes to the force equations.

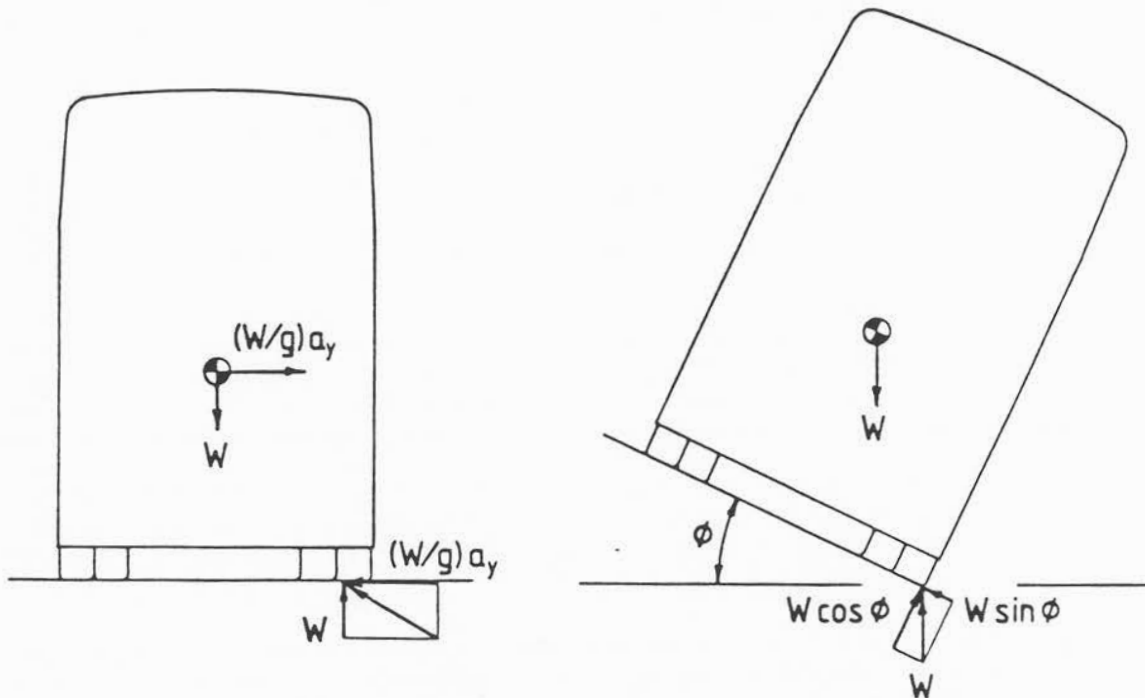


Figure 4.15 Loading Conditions During Steady Cornering Compared with Tilt Test Loading

Table 4.2
Validation of RASP by comparison of
lateral acceleration limit results

Vehicle	Limit Lateral Acceleration ($\pm 0.005g$)			
	using RASP		using STATLAT	
	1st Wheel Lift	Roll Limit	1st Wheel Lift	Roll Limit
Leyland National	0.665	0.760	0.665	0.730
MCW Metrobus	0.520	0.565	0.525	-
Van Hool Astronema	0.440	0.475	0.435	-
Jonckheere Jubilee	0.365	0.400	0.360	-
Plaxton Supreme	0.560	0.600	0.615	0.630
Plaxton Paramount	0.585	0.610	0.665	-
Duple Laser	0.540	0.580	0.530	-
MCW Metroliner	0.445	0.485	0.440	-
Leyland Royal Tiger	0.520	0.545	0.490	0.510

It was possible to obtain meaningful rollover limits from the STATLAT program for some of the PSVs. The simulations in the other cases (indicated with a dash) became unstable to varying degrees after the first wheel had lifted. This matter is discussed in Section 5.3.

Agreement for the lateral acceleration at which the first wheel leaves the ground is very good for most of these vehicles. In every case the simulations agree as to which axle this is (the rear axle for all buses except the Leyland National). The Plaxtons Supreme and Paramount have the lowest measured lateral spring stiffness in Table 3.8 and, since RASP includes this degree of freedom but STATLAT does not (that is, STATLAT assumes that these stiffnesses are infinite), the RASP program predictably gives lower results for them. The predicted First Wheel Lift Limits agree for the remainder within 0.03 g.

4.8.2 Comparison with Simple Rollover Threshold Analysis

The second method was comparison (in Figure 4.16) of the threshold lateral velocities obtained using RASP with results given by a simplified kerb impact analysis described by Jones (1976), and outlined in Appendix E.

The correlation coefficient for this relationship is 0.94 which is statistically significant at the 1% level (using the Pearson product moment correlation, from Cohen and Holliday, 1982). The higher threshold velocities given by the Simple Method are expected in view of the rigid body assumption, which neglects any suspension or tyre compliances.

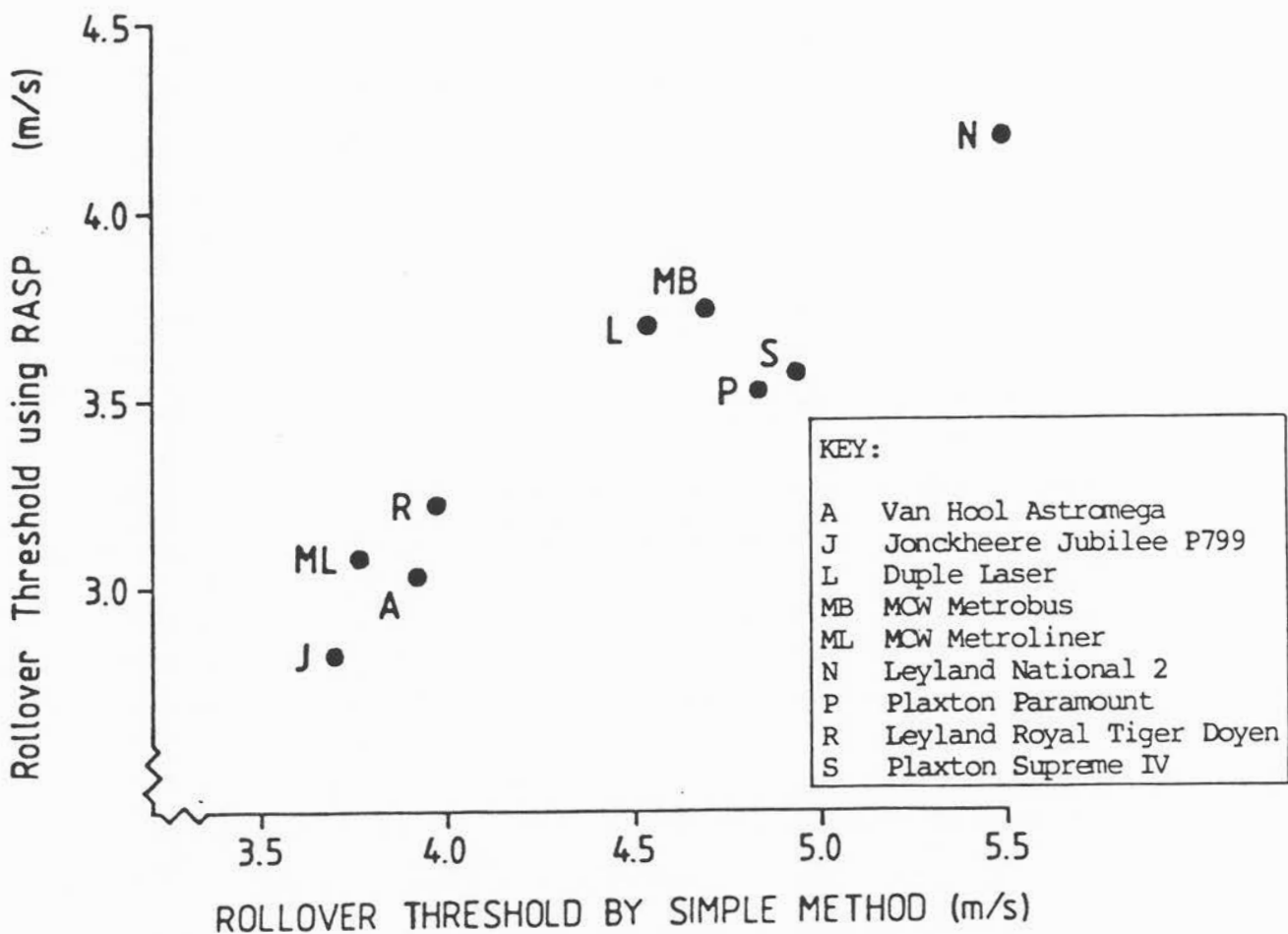


Figure 4.16 Comparison of Rollover Threshold Results Using RASP and Simple Method

4.9 RESULTS FOR TESTED VEHICLES

Table 4.3 indicates the simulated performance of each of the nine buses and coaches in a rollover accident, by describing the threshold lateral impact velocity above which overturning would occur.

Table 4.3
Results from Rollover Accident Simulation

Vehicle	Threshold Lateral Velocity for Rollover (m/s)
Leyland National 2	4.20
MCW Metrobus	3.74
Van Hool Astromega	3.03
Jonckheere Jubilee P799	2.82
Plaxton Supreme IV	3.57
Plaxton Paramount 3200	3.52
Duple Laser	3.70
MCW Metroliner	3.06
Leyland Royal Tiger Doyen	3.22

The threshold lateral velocities range from 2.8 to 4.2 m/s, a difference of almost 50% between the highest and lowest.

5.1 THE TILT TEST

The tilt test is a legislated type approval test required of all bus and coach models sold in Great Britain. When laden as described in Section 3.3.2, single-deckers must tilt to 35 degrees from the horizontal without falling over (a restraint system is used to arrest the overturning motion). The corresponding requirement for double-deckers is 28 degrees. (Details of tilt tests and results are given in Section 3.7.)

5.1.1 Historical Review

The tilt test originated in the 1920's when vehicle operators wished to fit roof structures to cover the upper deck of their double-deckers for all-weather use. Calculations were made at the time to determine the effect on the CG height and maximum lateral acceleration. Rather than disallow rigid covers to be fitted, it was decided to require all such vehicles to be tilt tested. Apparently tests were made at the time on various vehicles which indicated that the effect on the CG height of adding a rigid roof was less than the variation in CG height between different buses. It was reported (by the UK Rapporteur to GRSA, 1973) that consideration was given at the time to side wind forces and maximum cornering speed at full steering lock.

It is thought that the different tilt requirements for single- and double-deckers were probably obtained from actual measurements of contemporary vehicles, with regard also to different usage patterns (the double-deckers designed predominantly for urban transport with their higher passenger capacity, and the single-decked buses and coaches operating, not only within the city limits, but also in rural areas over longer routes at higher speeds).

5.1.2 Current Situation

Operating patterns have changed dramatically since the 1920's. Single-decked coaches are now approaching the same overall height as double-deckers, because operators desire increased luggage capacity. Both types are used for long-distance touring and on high-speed intercity passenger services. Speed limits are now up to 70 mph. Commercial vehicle tyre technology has advanced, with radial-ply and low-profile construction techniques contributing to better performance including higher available cornering forces. Suspension design has witnessed an almost total change from multi-leaf springs to air bellows with self-levelling valves. Separate chassis and body construction methods still exist,

although integral designs are becoming more common. Wood and steel structures have been superseded by steel and aluminium and even fibre-reinforced plastic for certain components. Regulations are being introduced which require suitable rollover protection for all single-decked PSV's. The engine is no longer positioned next to the driver, instead being located usually at the extreme rear, or sometimes under the floor between the axles. Engine power output now typically exceeds 180 kW (250 hp).

These developments, plus the criticisms of the tilt test by some in the bus industry, raise the following questions, which are addressed in the next chapters:

- (1) Should there be a distinction between the tilt angle requirements of single- and double-deckers, since in appearance and function they are now very similar?
- (2) Is the tilt test a valid test of overturning stability?
- (3) Is there a need for an overturning stability test at all?

These questions are addressed in this and the following chapters.

5.2 TILT TEST SIMULATION

The tilt test simulation enables the effect of variation of certain PSV design parameters on tilt test performance to be investigated.

5.2.1 Mathematical Model

Figure 5.1 illustrates the pertinent dimensions, forces and degrees of freedom involved. Equations of equilibrium are given in Appendix F and solved for the independent variables, as a static analysis. The approach used was to specify a tilt table angle and determine the values of the axle and sprung mass roll angles, axle lift and lateral displacements, and suspension movement, increment the platform angle by a small amount and repeat the procedure. Some details of the method of solution are included in Appendix F.

5.2.2 Assumptions

1. Pitch motions are negligible.
2. Axle roll angles are small.
3. The relative roll motion of the sprung mass with respect to the axles takes place about a fixed roll axis.

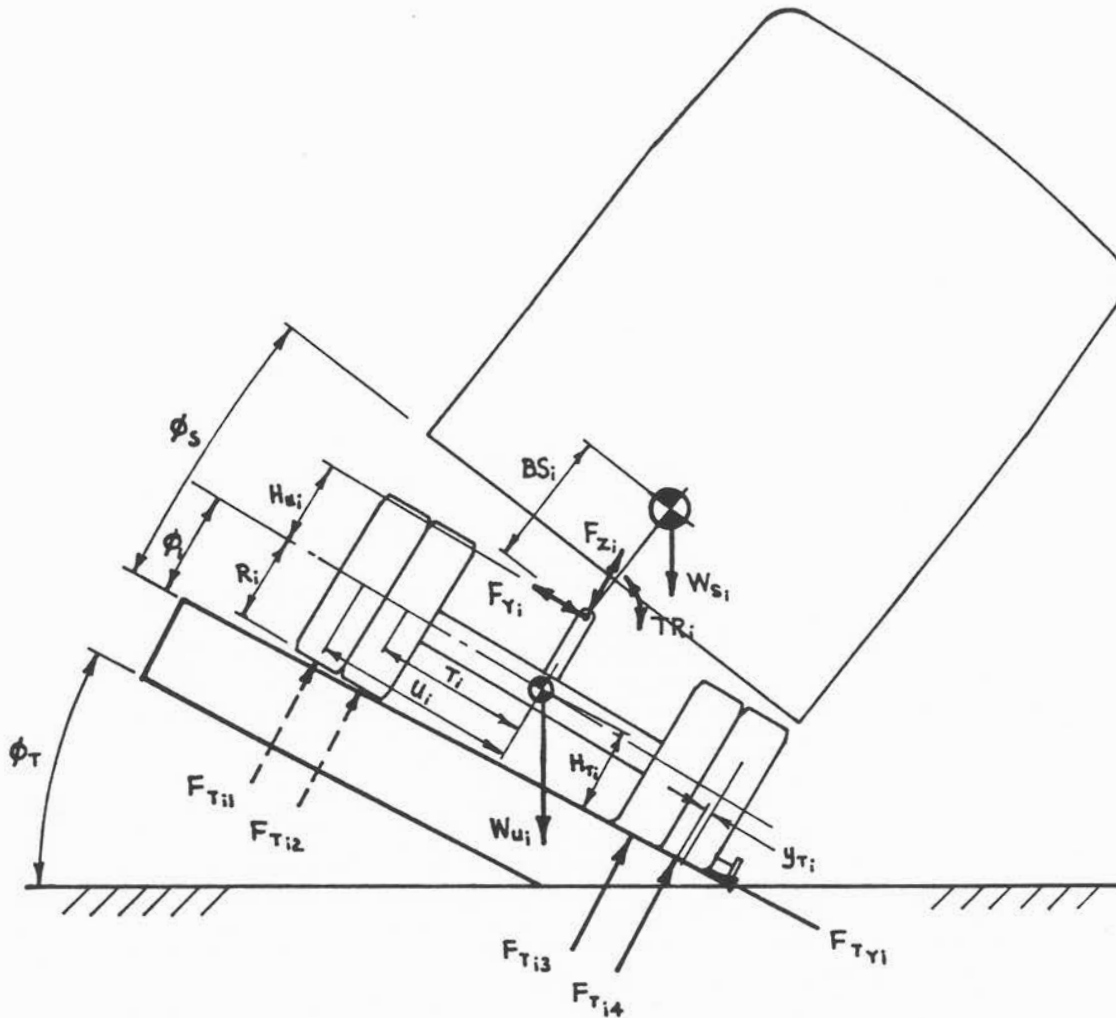


Figure 5.1 Mathematical Model for Tilt Test Simulation

4. The sprung mass and axles are modelled as rigid bodies. An important implication of this is that no allowance is made for torsional compliance of the vehicle structure. (This will be reasonable for integral and semi-integral PSVs, but may be important for some body-on-chassis vehicles, depending on the construction methods.)
5. The sprung mass is considered to be made up of two particles, one directly above each axle, at the correct CG height (for convenience in analysis). The ratio of masses is determined from weighpad measurements.
6. Springs are linear. No allowance is made for changes in suspension characteristics or CG height due to operation of levelling valves or tilt valve, or contact with bump stops.

7. The vehicle mass and CG height are calculated from the values for the empty vehicle, plus 63.5 kg per occupant, in the appropriate positions. (This does not coincide exactly with the vehicle as tilt tested, but does compare directly with the results from RASP.)
8. Lateral suspension compliance is assumed to be zero. That is, no allowance is made for the lateral displacement of the body with respect to the axles.

5.2.3 Results

Table 5.1 compares simulation results with actual tilt tests.

Table 5.1
Comparison of Tilt Test and Simulation Results

Vehicle	Maximum Stable Tilt Angle (deg)		
	Simulation		Test
	1st Axle	Limit	
Leyland National 2	35.4	39.0	38.0
MCW Metrobus	29.3	-	25.0
Van Hool Astromega	25.4	26.4	28.0
Jonckheere Jubilee P799	23.5	-	25.5
Plaxton Supreme IV	35.0	35.7	36.5
Plaxton Paramount 3200	36.0	36.0	32.0
Duple Laser	29.8	-	32.5
MCW Metroliner	25.8	-	25.0 *
Leyland Royal Tiger Doyen	28.9	29.3	33.0

* Test stopped after first wheel lift only

The simulation results given above are for the PSVs loaded as measured before the tilt test, rather than with the nominal 63.5 kg per occupant. The Jonckheere Jubilee P799 is simulated with only half of its quota of sandbags, as that was its test condition.

As a part of a general investigation into bus rollover in Hungary, two single-deck coaches were tilt tested, and the results reported by Hidvegi (1981). For comparison, the laden mass, CG height and tilt performance are given in Table 5.2.

Table 5.2
Results from Tilt Tests on Two Laden Hungarian Coaches
(from Hidvegi, 1981)

Description of Vehicle	Measured Data		
	Laden Mass (kg)	CG Height (m)	Maximum Tilt Angle (°)
10 metre airsprung rear-engine coach with roof rack	14380	1.62	25
11 metre leafsprung front-engine bus on truck chassis	15680	1.29	28

5.2.4 Discussion

The tilt angles obtained by the Hungarians are lower than those determined for the five U.K. specification single-deckers tested by the author (32 degrees minimum, 34.5 degrees average), indicating that, although the samples are very small, the British tilt regulation is probably having a positive effect on tilt performance.

Numerical instability after first wheel lift off occurred for some vehicles as with STATLAT (Section 4.8.1). This is indicated by a dash and is discussed further below.

Agreement is reasonable between the tilt simulation and measured results. The difference in tilt angle between first wheel lift and the rollover limit ranges up to 4 degrees tilt in the simulations. The tilt angle for first wheel lift was not measured in the tilt tests. Except for the MCW Metrobus and Plaxton Paramount 3200, in which cases the tilt test results are more doubtful than the simulation (due to reasons mentioned in Section 3.7.3), the simulation results are typically lower than the tilt limit. A review of the assumptions described in Section 5.2.2, in particular that of linear spring stiffnesses (and therefore neglecting effects due to bump stops, self-levelling valves and tilt valves) explains the other discrepancies.

5.3 NUMERICAL BEHAVIOUR OF STATIC SIMULATIONS

5.3.1 Problem Encountered

The simulation of the tilt test and the very similar program which models the steady cornering test (STATLAT, mentioned in Section 4.8.1) have a problem in common: numerical instability became evident after the first wheel lifted off the ground for most of the vehicles simulated.

5.3.2 Solution Methods Attempted

Many different solution techniques were tried in an attempt to successfully model the tilt test right up to the genuine tilt limit, including:

- a finite increment approach to integrating equations expressed in differential form, as used by Ervin, et al (1980) in a static simulation of steady cornering test for a three-axle representation of a tractor-semitrailer combination,
- corrected Gauss-Newton routines for least squares minimisation of a function, available as subroutines E04FCF and E04FDF in the NAG software library (Numerical Algorithms Group, 1983),
- Simplex method of minimising a general function (NAG routine E04CCF, and outlined in Nelder and Mead, 1965) chosen for its robustness and ability to cope with discontinuities, such as when a wheel lifts,
- Newton-Raphson technique (Hosking, et al, 1978) described in Appendix E with the tilt test equations.

While the tilt simulation worked perfectly for some vehicle parameters, such as those for the Plaxton Supreme, none of these methods made significant progress on simulating the other cases beyond first wheel lift. A more fundamental explanation is required.

5.3.3 Explanations

Ervin (1983) showed that if the roll stiffnesses at different axles are sufficiently disproportionate with respect to the ratio of the axle loads, then it is possible for a vehicle which still has three wheels on the ground to be unstable. However, preliminary calculations confirmed the tilt test observations that the particular vehicles tested were stable with only three wheels on the ground.

A more likely explanation is that there is more than one equilibrium position. To explain, consider the steady cornering simulation for simplicity since it is the same in principle as the tilt test without the complication of a changing ground plane angle. Figure 5.2 illustrates the two equilibrium positions that, in theory, are possible for the same lateral acceleration. One is stable and the other is not (Jones, 1976). The second position is employed by stunt drivers who drive four-wheeled vehicles on two wheels. When a simulation reaches the point where only three wheels are left on the ground, the solution may be flipping between these two solutions, consequently finding difficulty focussing on only one solution. This would explain the erratic, rather than divergent, nature of the attempted solution.

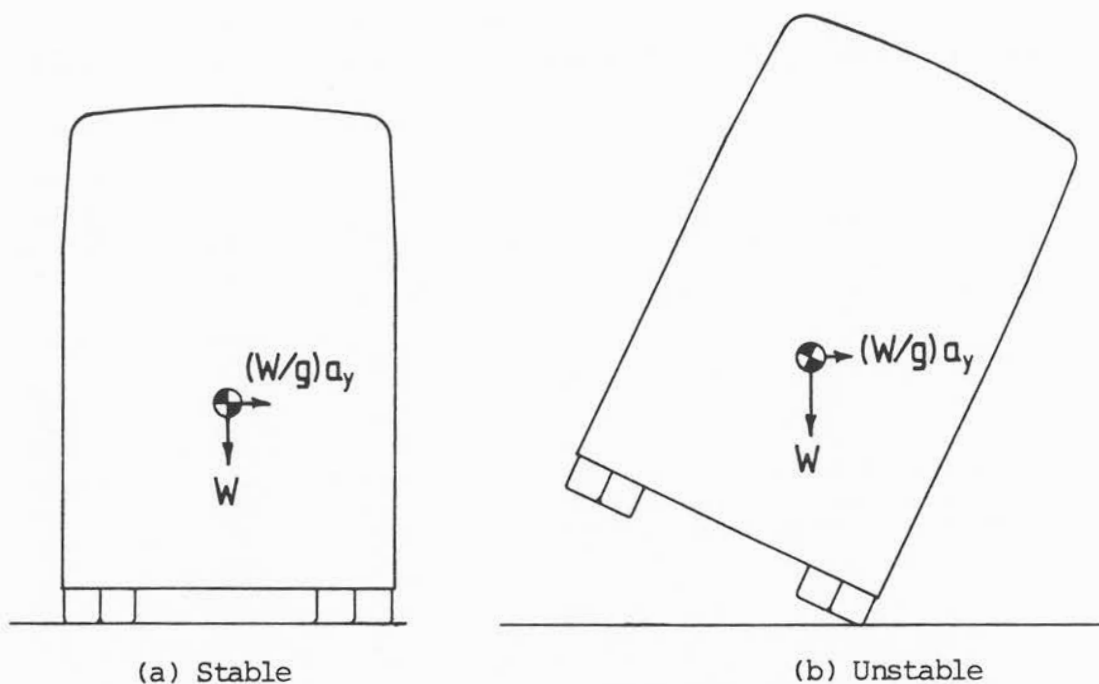


Figure 5.2 Two Equilibrium Positions at Steady Lateral Acceleration

It was discovered at a late stage that Mai and Sweatman (1984) using the Complex Method of function minimisation outlined by Box (1965) were able to continuously monitor the rollover stability during their simulation. They showed mathematically (in Appendix M) that the rollover limit may be reached at some point between one wheel lifting and another doing so. Thus, emphasis should be placed in a tilt test or similar simulation on determining mathematically, rather than from some preconceived supposition, when the point of instability is reached.

Fortunately, the difference between the First Wheel Lift Limit and the Rollover Limit are typically small compared to the possible improvements through redesign (as will be seen in the parametric study in Chapter 6). To quote from Mai and Sweatman (1984):

'the predicted First Wheel Lift Limit is considered to be a reasonable estimate of the stability threshold of the vehicle.'

Thus the First Wheel Lift Limit will be used in comparisons of tilt performance with other variables, since the Rollover Limits are not known for all the tested PSVs.

CHAPTER 6

COMPARISON OF ROLLOVER ACCIDENT AND
TILT TEST SIMULATIONS

6.1 COMPARISON OF RESULTS

Figure 6.1 indicates the relationship between the results from the Rollover Accident Simulation Program (RASP) and the maximum stable tilt angles as measured in the tilt tests (except for the Jonckheere Jubilee, in which case the simulated result is used instead of the test result, because it was only half-laden, rather than fully-laden). The correlation coefficient is 0.70 which is statistically significant at the 5% level (that is, there is a probability of at least 95% that these results are linearly related).

Figure 6.2 shows an even better correlation, this time between the tilt simulation results and the RASP results. The correlation of 0.82 is significant at the 1% level. The angle at which the first wheel lifts was used as the result from the tilt test simulation (for the reasons outlined in Section 5.3).

Lower, though still acceptable, correlation for Figure 6.1 is probably due to the tilt test method employed in the experimental phase (discussed in Section 3.7.3) and better correlation would be expected from normal tilt tests.

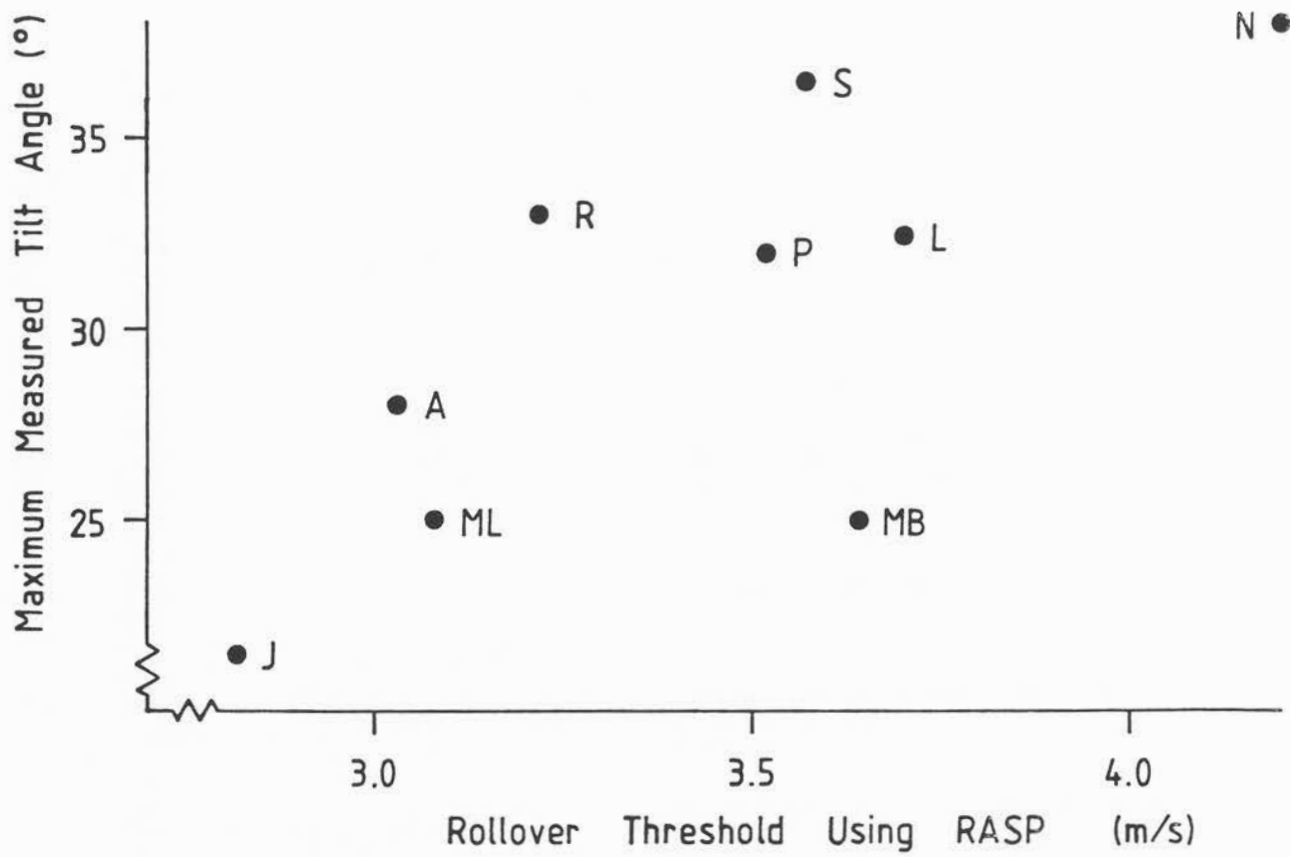


Figure 6.1 Correlation Between Tilt Test and Rollover Accident Simulation Results

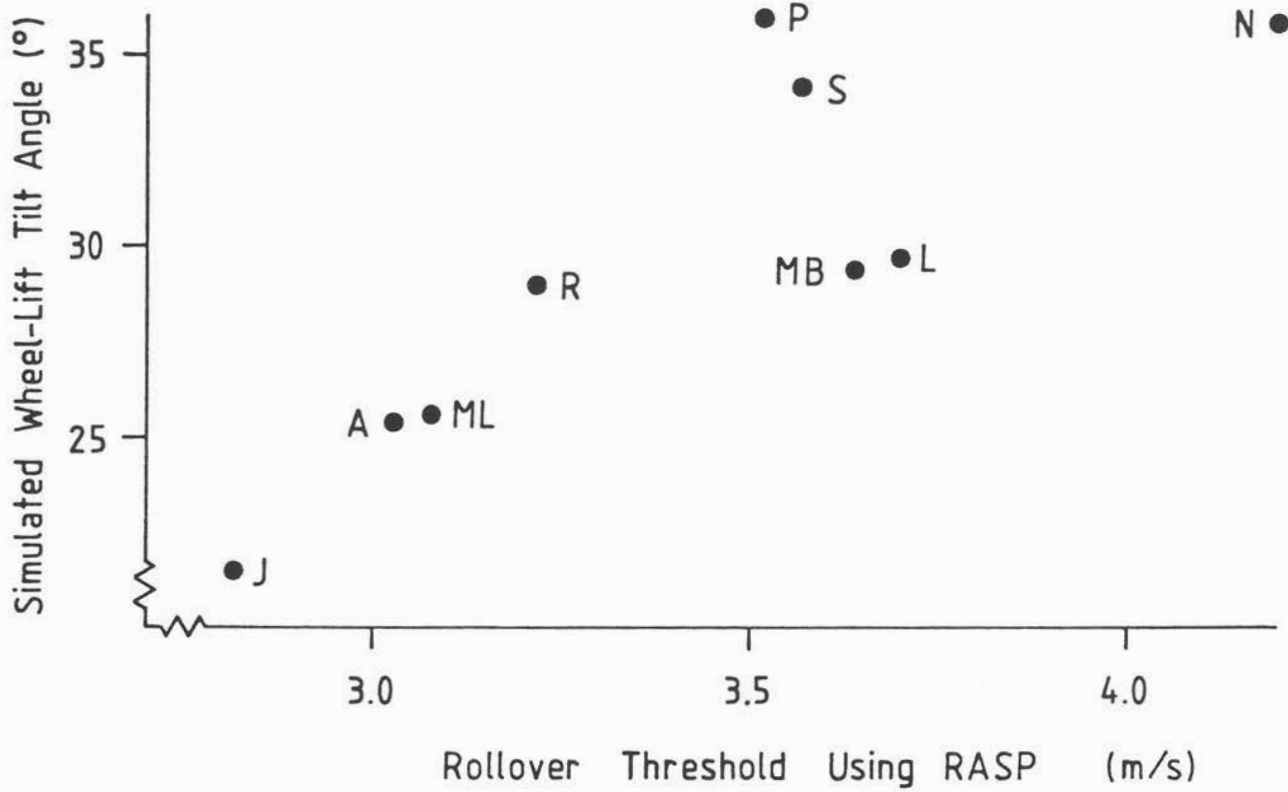


Figure 6.2 Correlation Between Tilt Simulation and Rollover Accident Simulation Results

6.2 COMPARISON OF PARAMETRIC INFLUENCES

The magnitudes of the various dimensional, inertial and stiffness parameters for two typical coaches are listed in Table 6.1. Unless stated otherwise, the parametric investigations are based on the "high-floor" coach, which represents the average of the entire range obtained. (The "low-floor" coach is typical of those measured vehicles with an overall height less than 3.3 metres.) Keeping the other variables at these base levels, each parameter in turn was varied to assess directly its effect on the threshold impact velocity and simulated tilt performance (that is, First Wheel Lift Angle).

Table 6.1
Data for Standard Coaches

Parameter	High-Floor	Low-Floor
Mass (kg)	15300	13000
CG Height (m)	1.50	1.25
Moment of Inertia (kg.m^2)	18500	12000
Vert. Stiffnesses: (kN/m)		
Front Spring	500	500
Rear Spring	1000	1000
Front Tyres	1900	1900
Rear Tyres	3600	3600
Lat. Stiffnesses: (kN/m)		
Front Spring	500	500
Rear Spring	1200	1000
Front Tyres	650	650
Rear Tyres	900	900
Roll Stiffnesses: (kNm/rad)		
Front Spring	350	250
Rear Spring	650	500
Roll Centre Height (m)		
Front	0.70	0.60
Rear	0.75	0.70
CG to Axle Distance (m)		
Front	3.80	3.32
Rear	2.00	1.76
Track Width (m)		
Front	2.05	2.05
Rear	2.00	2.00

6.2.1 Results

The results for the more significant factors are illustrated in Figures 6.3 to 6.6 which use normalised horizontal scales, obtained by dividing the parameter in question by its baseline magnitude. (Note that the corresponding axes in these figures do not always use the same scales.) Appendix F contains graphs which directly compare the effects of varying each dimensional and inertial parameter on both rollover threshold and tilt simulation performance. Table 6.2 summarises those graphs by expressing them as a sensitivity, that is, the expected change in rollover threshold (or tilt angle) for a given change in each parameter, assuming a linear range.

Table 6.2
Summary of Sensitivity Analysis

Parameter	Sensitivity	
	Rollover Threshold Velocity	Maximum Tilt Angle
CG Height	-2.2 (m/s)/m	-23 deg/m
Track Width	1.8 (m/s)/m	13 deg/m
Roll Centre Height	0.42 (m/s)/m	3.0 deg/m
Mass	-6.6×10^{-5} (m/s)/kg	-1.9×10^{-4} deg/kg
Moment of Inertia	2.3×10^{-5} (m/s)/(kgm ²)	0
Susp. Stiffnesses:		
vertical	6.7×10^{-5} (m/s)/%	2.1×10^{-3} deg/%
roll	1.4×10^{-4} (m/s)/%	2.2×10^{-2} deg/%
lateral	1.3×10^{-3} (m/s)/%	..
Tyre Stiffnesses:		
vertical	8.7×10^{-4} (m/s)/%	1.1×10^{-2}
lateral	6.3×10^{-4} (m/s)/%	..
Susp. Damping:		
vertical	0	0
roll	2.0×10^{-4} (m/s)/%	0
lateral	2.7×10^{-4} (m/s)/%	0
Tyre Damping:		
vertical	0	0
lateral	1.3×10^{-4} (m/s)/%	0

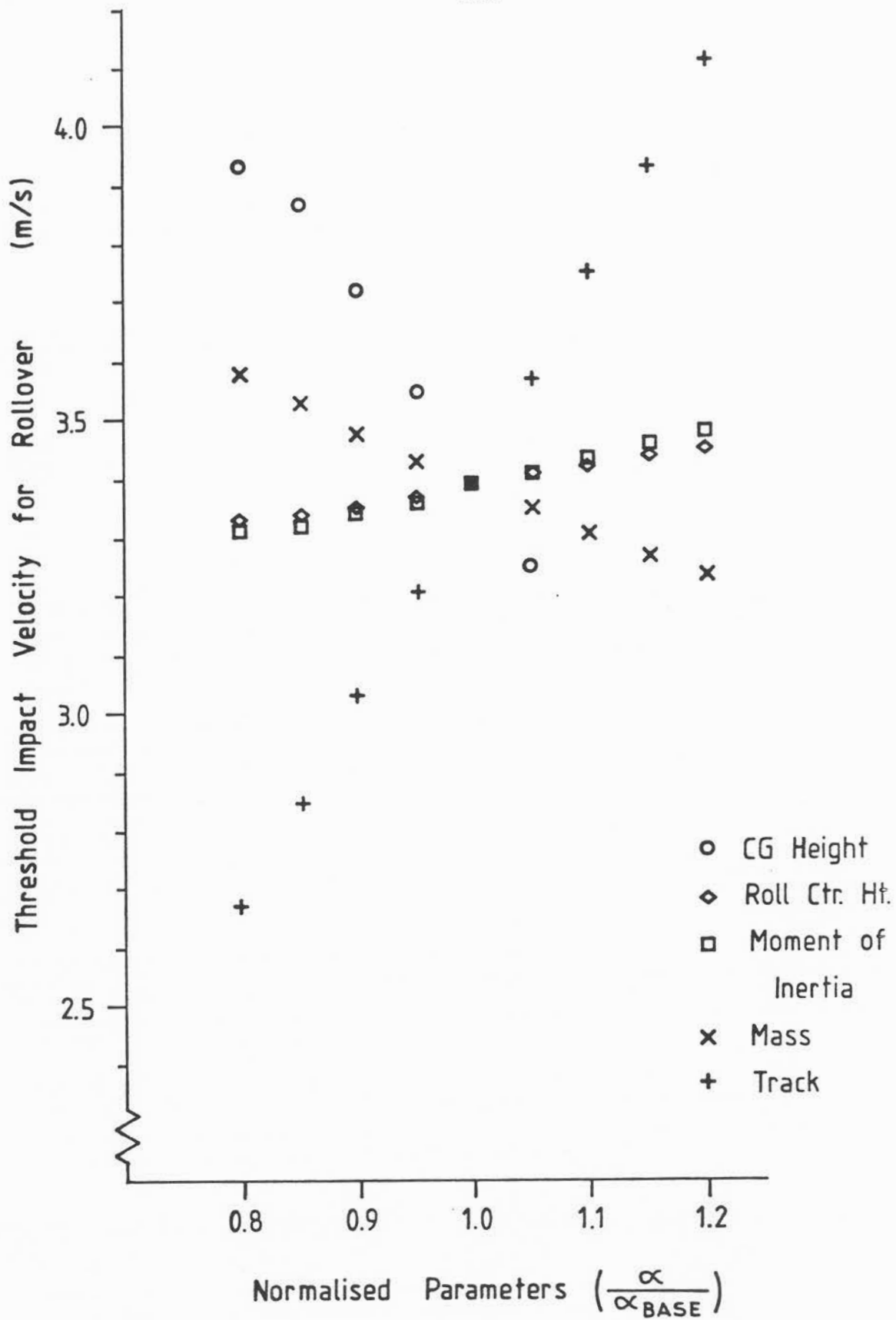


Figure 6.3 Effect of Varying Dimensional and Inertial Parameters on Rollover Threshold

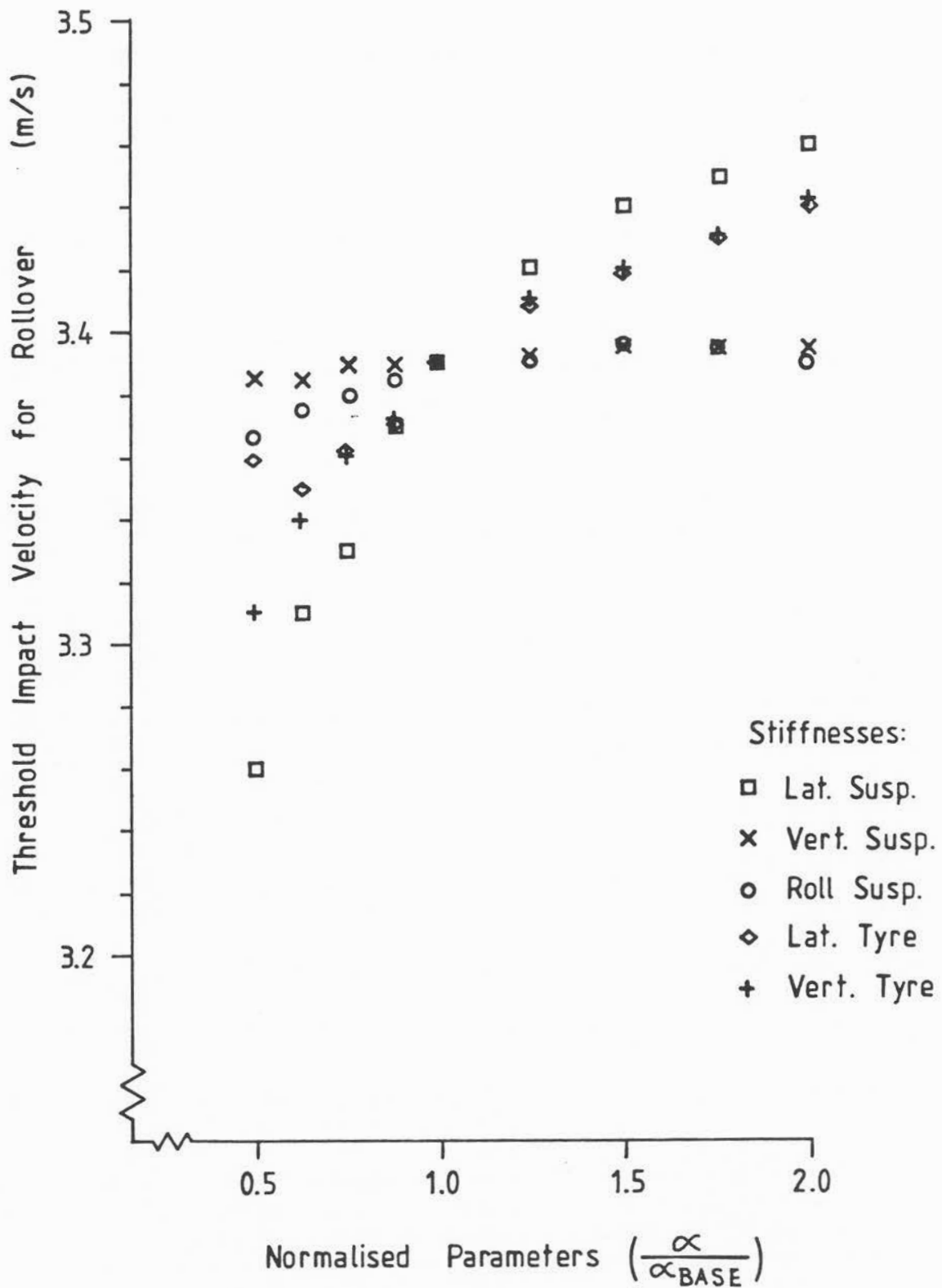


Figure 6.4 Effect of Varying Stiffnesses on Rollover Threshold

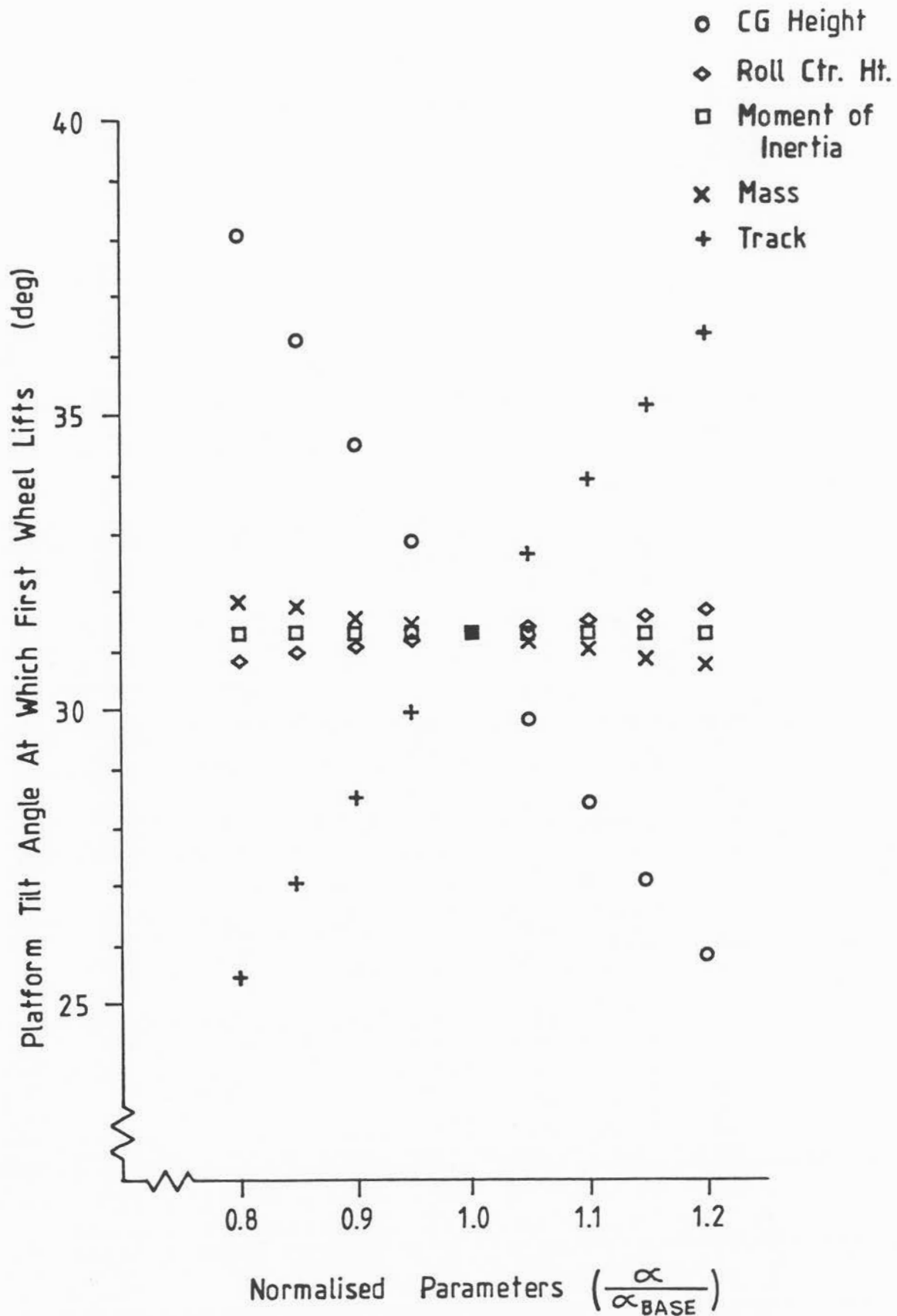


Figure 6.5 Effect of Varying Dimensional and Inertial Parameters on Tilt Simulation Results

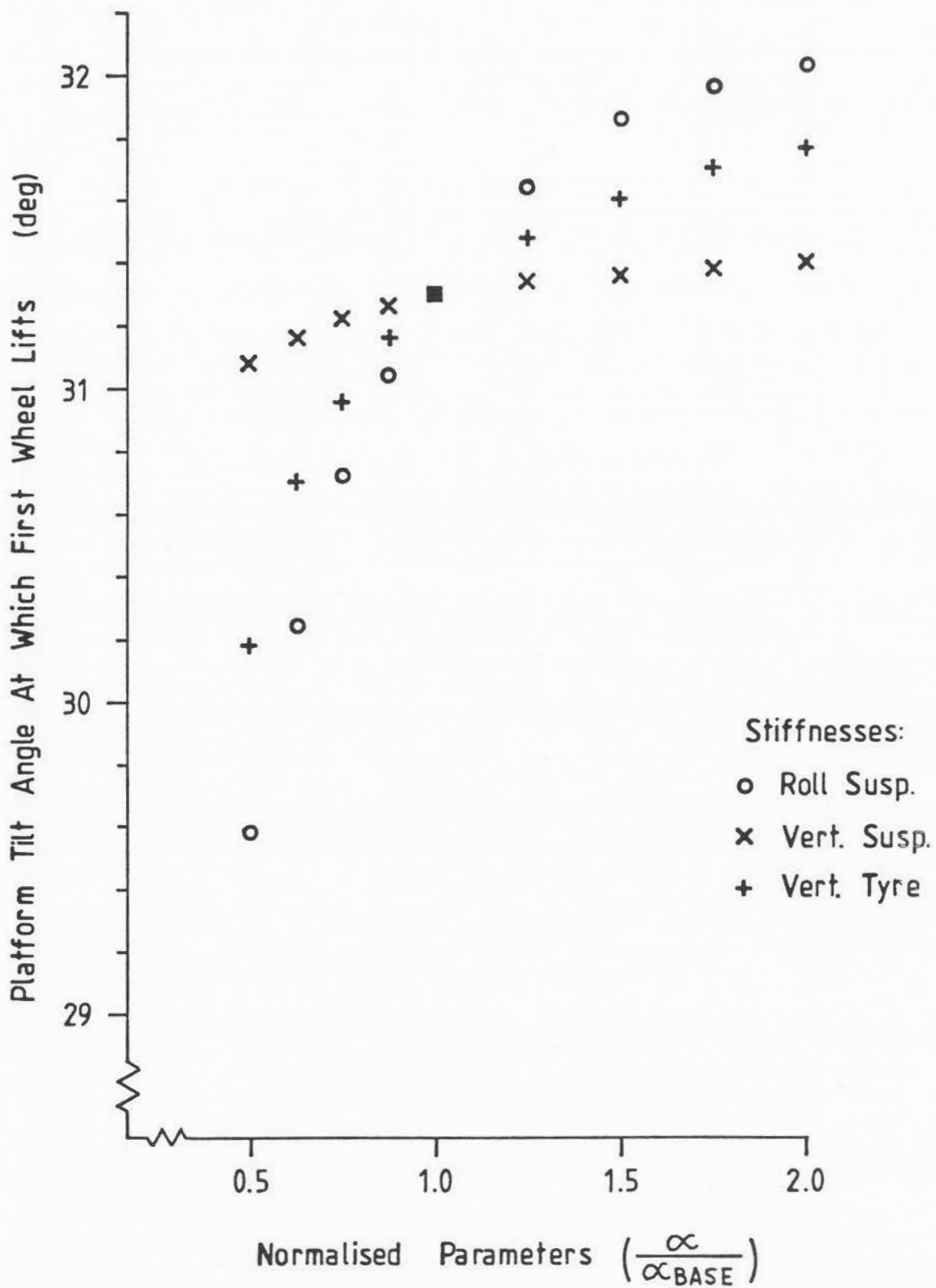


Figure 6.6 Effect of Varying Stiffnesses on Tilt Simulation Results

6.2.2 Discussion

It is apparent from Figure 6.3 to 6.6 that the most critical factors affecting rollover propensity and tilt performance are the CG height and track width.

The CG height is, not surprisingly, the single most important parameter available to a vehicle designer by which he can control the rollover threshold. CG heights in this sample of laden vehicles range from 1.10 to 1.76 metres. Figure 6.7 highlights the dominance of this parameter in affecting rollover threshold, evident from the good correlation (-0.89), despite the horizontal axis ignoring all effects apart from the CG height. (To a certain degree, however, the CG height will be related to vehicle mass and moment of inertia, because a higher CG generally implies a taller coach, and therefore greater mass. Also, variations in track width are relatively small.)

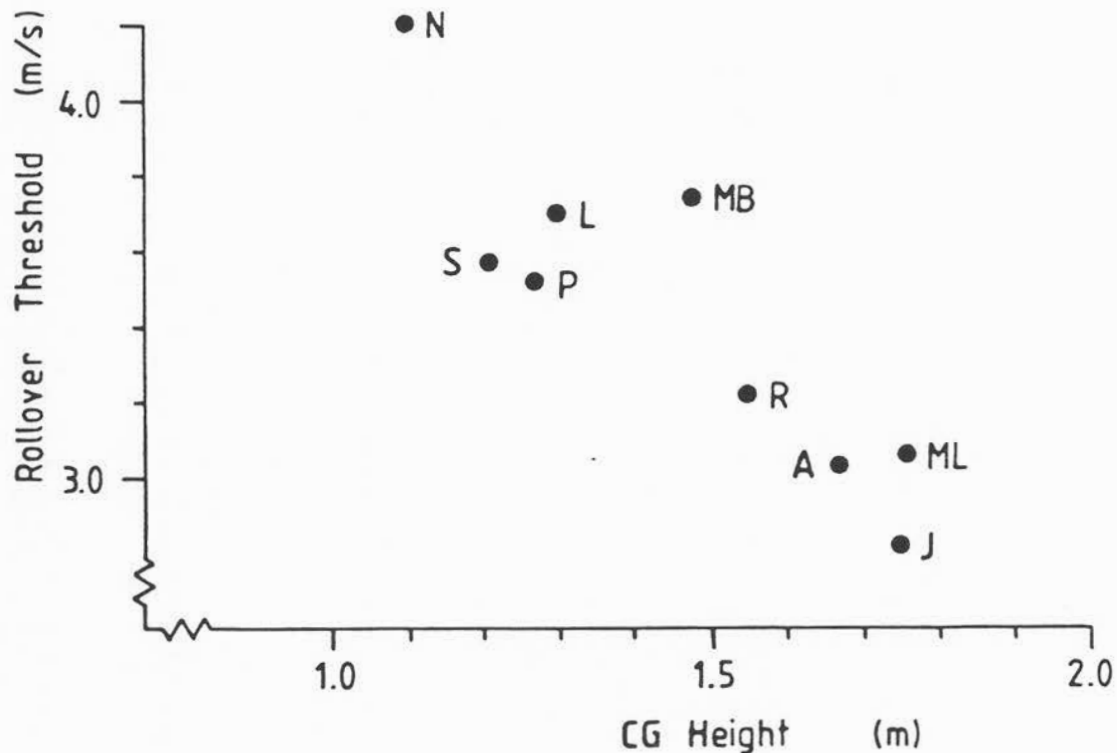


Figure 6.7 Relationship Between Rollover Threshold and CG Height

Owing to the tilt test requirements, British coach manufacturers should be well acquainted with the factors which influence CG height, such as overall height, seat mass and height above the ground, glass weight, roof material, position of air-conditioning unit, and so forth. Additional factors for consideration in this regard are the use of low-profile tyres, and the lowest sensible

ride height compatible with a suspension-lift option to allow the driver to raise the body for low speed negotiation of uneven terrain, including ferry ramps.

The track is limited by regulations on overall vehicle width. Most axle designs ensure the tyres are spaced as far apart as the law allows. In Canada, all provinces permit 102 inch (2.59 m) wide commercial vehicles to operate. Although the width limit for most commercial vehicles in the U.S.A. is 96 inches (2.44 m), buses operating on the interstate road system are allowed to be 102 inches wide. Ervin, et al (1980) recommended that the higher limit be permitted for tractors and semi-trailers transporting flammable liquids in Michigan, in order to significantly reduce the steady cornering rollover threshold. If width restrictions were eased for certain classes of commercial vehicle in Britain for which roll stability deemed the measure desirable, road width limitations would probably restrict such vehicles to particular classes of roads.

Increasing moment of inertia provides greater overturning stability, although the gains are small. No benefit is evident in the tilt test, however, because of its static nature.

In Appendix F, Figure F.5 demonstrates a peculiarity of the effect of varying roll centre height on tilt and rollover behaviour. The tilt test shows a steadily improving tilt limit, while the results from RASP reach a peak and drop off. For this typical vehicle the maximum corresponds to roll centre heights front and rear of 0.88 m and 0.94 m respectively. The same trends are evident in Figure F.6 which is for an average low-floor coach (whose parameters are also given in Table 6.1). In this case the peak in rollover threshold occurs for 0.69 m (front) and 0.81 m (rear) roll centre heights. These graphs highlight three discoveries related to roll centre height effects:

- (1) for any vehicle of this configuration there exist certain values of roll centre height at the front and rear axles which give optimum rollover stability;
- (2) the tilt test does not detect this optimum; and
- (3) predictions of kerb-impact rollover performance based on tilt test results become less accurate at roll centre heights above the optimum.

In practice, these findings are not greatly significant. Front and rear roll centre heights of the tested vehicles varied from 0.5 to 0.9 metres (as used in the simulations, Table 3.8). None are likely to seriously exceed their optimum values, and even if they did, their rollover performance would not be less than those with roll centre heights at the lower end of the range. Steady cornering limits for higher vehicles improve with roll centre height (as indicated by tilt test results). Underlying all this

is the fact that the overall effect of roll centre height on rollover threshold is minor compared with variations in CG height. This implies that it would be foolish to carefully optimise roll centre height without first minimising the CG height, and monitoring the effect on CG height of any subsequent design modifications.

Of the stiffness parameters, the most significant with respect to rollover threshold is the lateral suspension stiffness. (The tilt test simulation did not permit variation of this parameter.)

Roll stiffness, which was expected to be influential on the kerb-impact overturning behaviour, actually has negligible effect. For the tilt test, on the other hand, it is the most important stiffness (see Table 6.2) - another difference between tilt test and rollover accident simulations.

Table 6.2 also indicates the minor influence of damping on rollover threshold. Since damping is a function of dynamic behaviour, it has no influence on the static tilt test results.

6.3 DIFFERENCES BETWEEN ROLLOVER ACCIDENT AND TILT SIMULATIONS

Several parameters affect the kerb-impact rollover propensity differently from their influence on tilt simulation performance. Some (the roll moment of inertia, and the various damping constants) do so because their properties are only evident in a dynamic situation, and thus have not influence on the tilt test results. Others (the suspension roll and lateral stiffnesses and, to a lesser extent, the roll centre heights) affect the simulations differently due to their inherently different loading conditions.

Overall, though, the correlation is very good, implying that the tilt simulation, and hence the tilt test, are good indicators of kerb-impact overturning performance.

They are considered in Chapter 7 (along with several other possible methods) for their comparative suitability as a legislative test procedure for measuring rollover stability of PSVs.

CHAPTER 7

DISCUSSION OF BEST METHOD OF
EVALUATING ROLLOVER STABILITY

7.1 INTRODUCTION

Chapter 3 concluded that there is a case for having an overturning stability test for buses and coaches. This chapter addresses the question of how overturning stability can best be measured in a legislative test. It is essential that such a test be repeatable, and, in view of the legal requirement to pass the test, it should be kept as simple, repeatable and inexpensive as possible.

Chapter 1 described the deduction from accident investigations that a typical bus rollover accident involves lateral velocity into a "tripping" obstacle. An ideal test would therefore measure the performance of a vehicle in a scenario in which some form of calibrated, repeatable obstruction is struck at a prescribed lateral velocity. Realistically, any test method which correlates sufficiently well with the ideal test could be considered as a suitable legislative test procedure.

Finite time and funds limited this research to the development of a computer simulation of a rollover accident, and comparison of results from that simulation with actual and simulated tilt test results and simulated steady cornering results. Quantitative comments are given for the other candidate methods.

The following alternative methods of testing PSV rollover stability are considered:

- (1) Full-scale rollover accident test
- (2) Simulation of rollover accident test
- (3) Full-scale double lane-change test
- (4) Simulation of double lane-change test
- (5) Full-scale steady-state cornering test
- (6) Simulation of steady-state cornering test
- (7) Full-scale tilt test
- (8) Simulation of tilt test
- (9) CG height measurement.

A summary of the advantages and disadvantages of each method is given in Section 7.3.

7.2 REVIEW OF CANDIDATE TEST METHODS

7.2.1 Full-Scale Rollover Accident

A PSV is projected laterally (on a sled) at a prescribed velocity into a standard "kerb". The test is failed if the vehicle rolls past its point of balance.

Although this is precisely what is desired to be measured, Chapter 2 highlighted some complications found during research into a proposed method for testing PSV superstructure strength:

- the kerb-impact forces severely damage or even fracture the suspension components, even with force-limiting crush tubes interposed between wheel and kerb, and
- repeatability is low owing to the complexity of the impact forces and their significant effect on the outcome of the test.

In addition, even if a method of tethering were successfully employed to prevent rollover, the test vehicle would be seriously structurally damaged due to the high impact forces.

7.2.2 Simulation of a Rollover Accident

A mathematical model (such as RASP) which utilises experimentally determined vehicle data such as spring and inertial properties and appropriate dimensions, calculates the vehicle motion following impact with a standard kerb.

This method has the advantage of repeatability, because the kerb-impact is represented ideally and not subject to random effects.

However, it is the input data which determines whether or not a given coach passes the test. Therefore, measurement of the data would need to also be part of the legislative test. It was found during the experimental phase that approximately three days were required to determine the CG height, roll moment of inertia, stiffnesses, and roll centre heights for a typical coach. Double-deckers required about four days for the same tests. Calculations and simulation would take another one or two days. The time required (and the associated expense) is the main disadvantage for this method.

Further validation of the computer program by comparison with full-scale rollover accident tests may be desired by interested parties, to which the drawbacks listed in Section 7.2.1 also apply.

On the positive side, this method would give the best indication of kerb-impact rollover stability. It should therefore be available as an alternative method of demonstrating adequate rollover stability. As a safeguard, the simulation and data measurement could be restricted to Government or independent research units with the expertise to understand the procedures (and provide constructive criticism where necessary).

7.2.3 Full-Scale Double Lane-Change Test

A test course is marked out similar to that shown in Figure 7.1. The test consists of driving through the course at a prescribed forward speed, without exceeding certain constraints on such parameters as vehicle trajectory, roll and yaw angles, axle lift or rollover.

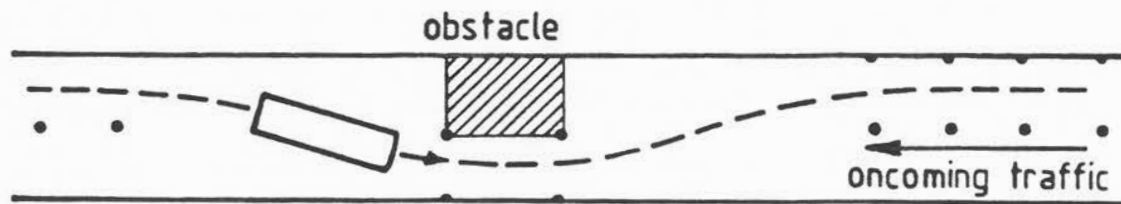


Figure 7.1 Typical Track Layout for Lane-Change Test

This is a commonly suggested method for investigating dynamic vehicle performance at the limits of tyre adhesion and yaw and roll stability. However, there are serious problems in the evaluation of performance. Fancher, et al (1976) identified these as:

- '(a) variations in driver skill,
- (b) differences in the amount of information imparted to the driver by the layout of the test course,
- (c) the asymmetry in response between right and left lane-changes, and

- (d) the lack of understanding of the connection between open- and closed-loop performance'.

As a test of PSV rollover stability, overturning is an additional hazard. It is logistically very difficult, if not impossible, to adequately secure some form of rollover restraint such as an outrigger (Figure 7.2 from Kemp, Chinn and Brock, 1978) to a bus, particularly one of chassisless construction. Even if it was possible, the device would need to be designed to withstand the anticipated loading which means that with the size of outrigger needed it would influence the mass, CG height and moments of inertia to the extent that the true vehicle was not being tested.



Figure 7.2 Outrigger to Prevent Accidental Rollover
(from Kemp, Chinn and Brock, 1978)

7.2.4 Simulation of Double Lane-Change Test

Computer simulation may overcome most of the difficulties of the full-scale test, though an acceptable driver model would be particularly important. The mathematical model, which requires extensive tyre data and measurement of the yaw moment of inertia in addition to the usual spring, dimensional and inertial parameters, simulates the manoeuvre described in Section 7.2.3.

Several vehicle dynamics computer programs exist which are capable of modelling this type of test, though for efficiency they may need to be simplified and tailored to suit this manoeuvre alone. Some examples of known validated simulation programs are:

- (1) HVOSM-VD2, (Highway-Vehicle-Object Simulation Model - Vehicle Dynamics, version 2) from Calspan Corporation.
- (2) Dynamic Roll Model, described by Gillespie and Verma (1978), from what used to be called the Highway Safety Research Institute (HSRI) at the University of Michigan. It is now known as the University of Michigan Transportation Research Institute (UMTRI).
- (3) Comprehensive Braking and Steering Model, otherwise known as the Phase 4 Model, also from UMTRI. It is reported in detail by MacAdam, Fancher, Hu and Gillispie (1980).

A purpose-built rig would need to be constructed to accurately measure the yaw moment of inertia of PSVs. Tests to determine input data would take a few weeks, although if the same suspension (such as a proprietary chassis) or tyres were used on more than one vehicle a database storing relevant test results may reduce the amount of experimental work required.

As for the equivalent full-scale test, the correlation with rollover accident propensity is unproven. Simulation is a useful means for investigating this question.

7.2.5 Full-Scale Steady-State Cornering Test

A vehicle is driven in a circular path of a specified radius. The forward speed is increased until a prescribed speed is reached, equivalent to a given lateral acceleration. This must be attained without exceeding tyre adhesion, or overturning.

The major difficulty with this method (apart from the fundamental question of whether it is testing the desired property) is the problem of attaching outriggers or some equivalent device which would prevent total rollover from occurring.

Most PSVs would reach tyre side force limits (and therefore understeer or oversteer) well before the rollover limit when unladen. However, when fully laden, the tested vehicles had Rollover Limits ranging from 0.40 to 0.76 g (Table 4.2).

Wilkins and Riley (1983) measured side force coefficients for commercial vehicle tyres, but only on wet surfaces (typically 0.30 to 0.34), perhaps because of the danger of rollover on dry surfaces. Braking force coefficients were measured on both wet and dry surfaces and give an indication of the effect of wetting a road on the tyre friction available. The peak wheel braking force coefficients are generally 0.3 to 0.6 at low speeds for a range of tyres on various wet roads, and 0.8 to 0.9 on dry road surfaces. Assuming a similar increase in side force coefficient on dry compared with wet surfaces yields values of approximately 0.5 to 0.8 g for the side force coefficient on a dry road. Thus, attaining a rollover condition in a steady cornering test could not be guaranteed for all vehicles.

7.2.6 Simulation of Steady-State Cornering Test

Because this test method is quasi-steady-state, involving forces and moments which can be held in equilibrium, simulation of the method is much simpler in principle than the previously described dynamic situations.

As is explained below in Section 7.2.7, the forces and moments in a steady-state cornering test are very similar to those in a tilt test. It was straightforward, then, to modify the tilt test simulation program to model the steady-state cornering test. Figure 7.3 compares the rollover accident simulation results with those from the cornering simulation. (The main difference between Figures 4.1 and 7.3 is that the latter does not assume rigid body motion, but allows for suspension, tyre and curb compliances.)

The correlation coefficient is 0.84, which is significant at the 1% level. Thus the steady cornering test is a good indicator of kerb-impact rollover performance.

7.2.7 Full-Scale Tilt Test

The tilt test is described in Section 5.1 and illustrated in Chapter 3.

At small tilt angles, the tilt test is fundamentally equivalent to the steady-state cornering test, with the in-plane component of the gravity force corresponding to the lateral inertial force (Figure 4.15). However at large tilt angles the normal load is reduced from W to $W \cos \theta$. For fully-laden heavy goods vehicles, which usually tilt before 20 degrees the effect is less than 6%. But the difference is 12% at the current 28 degree requirement

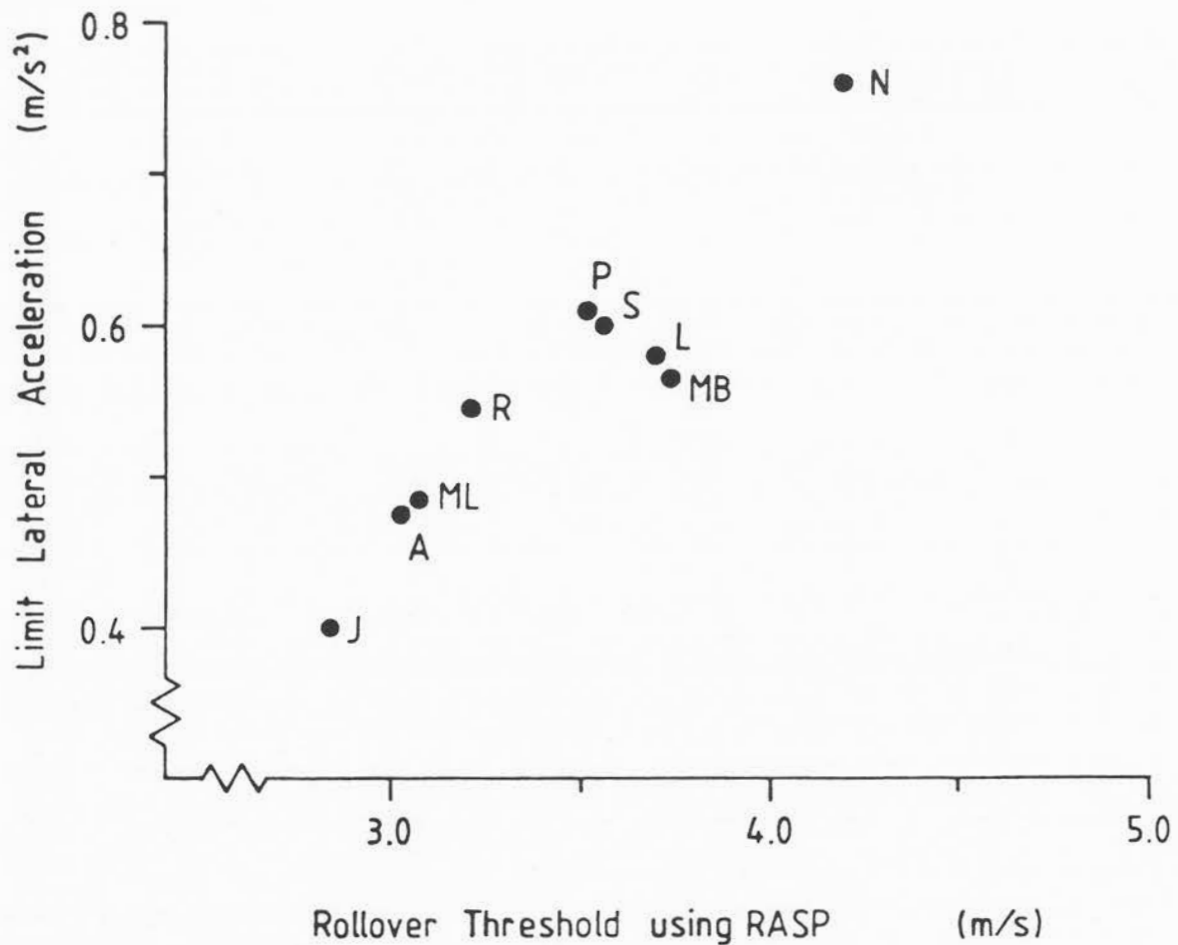


Figure 7.3 Comparison of Static Lateral Acceleration Results with RASP Rollover Threshold Results

for double-deckers, and 18% for single-decked vehicles at 35 degrees. This implies that at the higher tilt angles the vehicle body will lift with respect to the axles because of the lower normal force, and the CG position and suspension deflections will differ significantly from their level-ground values (Segel, Winkler and Fancher, 1981). However, the parametric study in Chapter 5 indicated that the effect of varying vertical suspension stiffness on tilt performance is actually small compared with the other parameters.

The advantages of the tilt test lie in its simplicity, repeatability and (in Britain) availability of test apparatus. This is reflected in the fact that tilt tests have been proposed for type-approval tests for articulated commercial vehicles in several countries: the United States (Ervin, et al, 1980), Sweden (Strandberg, et al, 1975) and Australia (Mai and Sweatman, 1984).

Incidentally, all the proposed tilt requirements fall in the range from 20.5 to 25.5 degrees.

7.2.8 Simulation of Tilt Test

For legislative purposes computer simulation of the tilt test offers no great advantages over the full-scale test. This is because the experimental work to determine the necessary input data is significantly more involved than the tilt test itself. Some PSV manufacturers use tilt test simulation during the design process to predict tilt performance before a vehicle is built.

7.2.9 CG Height Measurement Test

The method has been outlined in Section 3.3. While the parametric study (Section 3.5) indicated that the CG height is the most important parameter affecting propensity to rollover, to base a legislative test on measuring it alone would be to ignore completely the significant effect of suspension geometry and stiffnesses. For example, the Jonckheere Jubilee P799 and MCW Metroliner have almost identical laden CG heights, yet tilt test results differ by at least 14% and rollover threshold velocities by 8%. In addition, a CG height measurement test is more complex than a tilt test, requiring measurement of reaction forces and blocking of suspension and tyre movement.

7.3 RECOMMENDED TEST METHODS

Table 7.1 summarises the major advantages and disadvantages of each of these test methods.

The disadvantages for methods 1, 3 and 5 are sufficient to eliminate them from consideration. Method 4 is considered to be a "loss-of-control" test (Section 1.2.2) rather than a "kerb-impact-rollover" test. Methods 6 and 8 are very similar in principle, but require far more work than the full-scale tilt test (Method 7). Method 9 offers no advantages over the tilt test, which provides a more accurate indication of rollover performance. Thus, the tilt test appears to offer the best test procedure. The rollover accident simulation is considered to give the best indication of a vehicle's propensity to rollover in a curb-impact situation, at the cost of significantly more experimental testing to obtain input data. It is suggested that this method be permitted as an alternative means of meeting the rollover stability requirements. (This may even encourage further research into this area of rollover safety.)

Table 7.1
Candidate Overturning Stability Test Methods

Method	Advantages	Disadvantages	Further Work
1. Full-Scale Rollover Accident	<ul style="list-style-type: none"> ● tests directly for required property 	<ul style="list-style-type: none"> ● low repeatability ● serious damage to test vehicle 	
2. Rollover Accident Simulation	<ul style="list-style-type: none"> ● as above ● repeatability 	<ul style="list-style-type: none"> ● extensive testing for input values ● hard to truly validate 	
3. Full-Scale Lane-Change Test	<ul style="list-style-type: none"> ● good test of loss-of-control 	<ul style="list-style-type: none"> ● danger of rollover ● driver variation ● outrigger influence ● asymmetric response ● hard to evaluate ● tyre grip limited 	<ul style="list-style-type: none"> ● correlation with 1. ● auto-steer? (but other problems)
4. Lane-Change Simulation	<ul style="list-style-type: none"> ● repeatability 	<ul style="list-style-type: none"> ● extensive testing for input values 	<ul style="list-style-type: none"> ● correlation ● yaw inertia rig required
5. Full-Scale Steady Cornering Test	<ul style="list-style-type: none"> ● simple 	<ul style="list-style-type: none"> ● danger of rollover ● weather influence ● outrigger influence ● tyre grip limit 	
6. Steady Cornering Simulation	<ul style="list-style-type: none"> ● repeatable 	<ul style="list-style-type: none"> ● extensive testing for input values 	
7. Full-Scale Tilt Test	<ul style="list-style-type: none"> ● simple ● repeatable ● platforms exist 	<ul style="list-style-type: none"> ● suspension-raising effects at high tilt angles 	<ul style="list-style-type: none"> ● tilt test without levelling valves
8. Tilt Test Simulation	<ul style="list-style-type: none"> ● repeatable 	<ul style="list-style-type: none"> ● extensive testing for input values 	
9. CG Height Test	<ul style="list-style-type: none"> ● repeatable 	<ul style="list-style-type: none"> ● ignores suspension effects 	

CHAPTER 8

RECOMMENDED ROLLOVER STABILITY TEST
REQUIREMENTS

8.1 PROPOSED TEST METHODS

Chapter 7 highlighted the suitability of the tilt test as a legislative method for evaluating rollover performance. Simulation of a kerb-impact overturning accident is suggested as an optional alternative method.

8.2 PROPOSED TEST STANDARDS

The first of the three questions posed in Section 5.1.2 is now considered (the other two have already been answered). If a double-decker which has passed a tilt test regulation to 28 degrees is deemed to be a "safe" vehicle, does this not imply that a single-decked PSV which also tilts satisfactorily to 28 degrees and operates in the same circumstances is also a "safe" vehicle?

One notable difference in overturning behaviour between single- and double-deckers is the difficulty of rolling the taller vehicle beyond 90 degrees (that is, onto its roof or beyond). Overturning beyond 90 degrees introduces high cant rail loading and danger of roof collapse, although as mentioned earlier this is being addressed by new legislation. The severity of an accident in terms of occupant fatalities and injuries increases with roll angle, as was established in Section 1.1.3.

To arrive at the proposed overturning stability requirements the following line of reasoning was followed:

- Current double-deckers are considered "safe". (This assumption is discussed further in Section 8.4.)
- The current tilt requirement is therefore acceptable for double-deckers.
- Therefore, a single-decked vehicle which also meets the double-decker tilt test requirement is also considered "safe".
- The distinction between these two types of vehicle should be based, for the purposes of overturning stability, on the overall laden height of the vehicle, rather than the internal seating arrangement.
- A taller PSV is less likely to roll onto its roof or beyond, due to geometric and energy considerations.

- The new superstructure roof strength regulation should provide adequate survival space for the shorter coaches. (Note that it applies only to single-decked PSVs.)
- Death or injuries of occupants due to ejection or unrestrained motion within the vehicle is, however, more likely to occur in the rollover of a single-decked (or shorter) vehicle than in a similar accident for a double-decked (or, taller) vehicle, because of the generally higher roll angles.
- The greater potential accident severity for the lower vehicles is reflected in a tilt requirement higher than that of the taller vehicles by some reasonable margin (suggest 3 degrees).

To summarise the proposed overturning stability requirements:

- (1) PSVs at least 4.0 metres high to meet current double-decker standards, or 3.0 m/s lateral kerb impact velocity.
- (2) PSVs under 3.0 metres high to exceed current double-decker standards by a margin of 3 degrees, or 3.2 m/s kerb impact velocity.
- (3) A linear relationship be used to determine the standards for PSVs between 3 and 4 metres high.

Figure 8.1 expresses these proposals graphically.

8.3 DISCUSSION OF PROPOSALS

8.3.1 Choice of Rollover Standards

The basic assumption that current double-deckers are regarded as "safe" in respect of overturning stability is open to criticism. This is recognised, and the following paragraphs are intended to reveal the considerations behind the initial assumption.

Many safety standards involve some, perhaps reasoned, but nevertheless arbitrary decisions (for example, that frontal impact tests for cars be conducted at 30 mph rather than any other speed, or in this field, that a coach superstructure should absorb a certain amount of energy rather than a value greater or smaller).

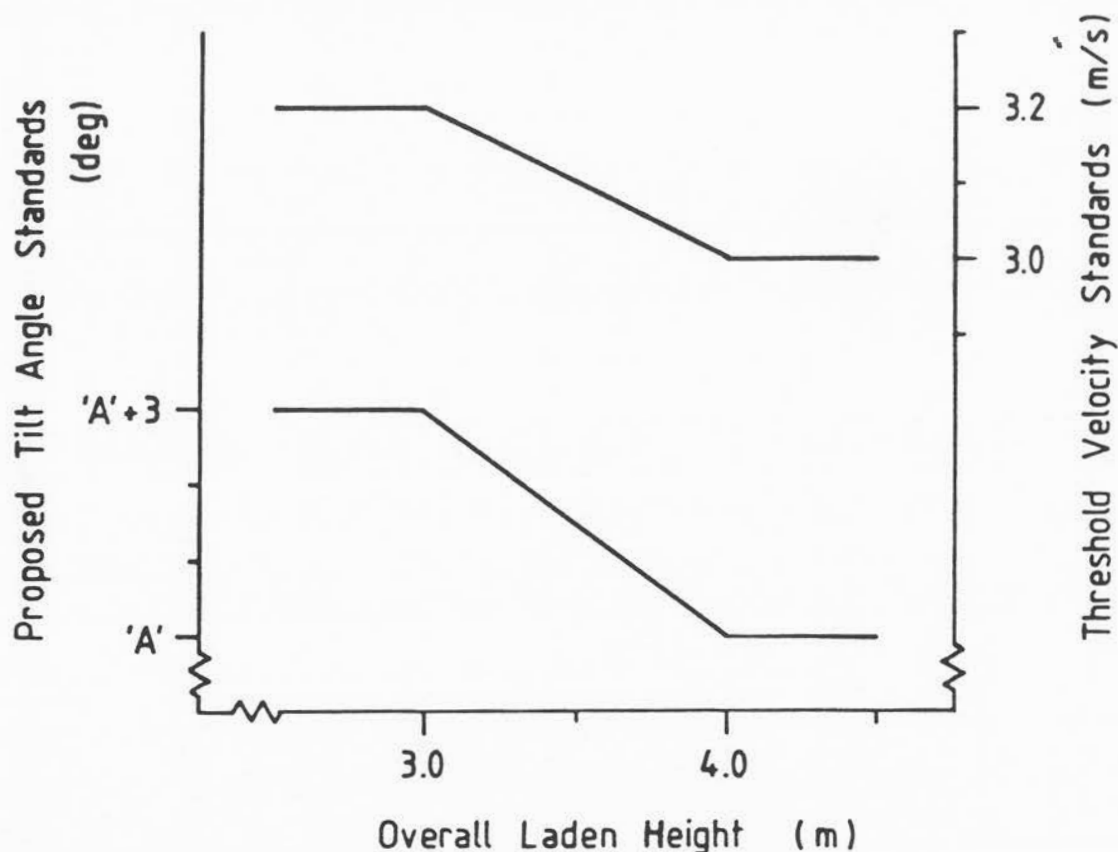


Figure 8.1 Summary of Proposed Rollover Stability Standards

It is usually possible for all concerned parties to agree on upper and lower bounds to a particular standard. In this case, 22 and 32 would be regarded as too low and too high respectively as a tilt limit for double-deckers. The lower constraint is the prevention of large numbers of rollover accidents. Apart from technical considerations, the upper constraint is perhaps less obvious. If the legal requirement is set too high with the intention of dramatic reductions in rollover accidents, manufacturers will probably react in the second of two possible ways (which appears to be the current situation for single-deckers):

1. Ideally, the manufacturers will recognise and concur with the intention of the regulation and will endeavour to improve overturning stability by keeping the CG height sufficiently low, even if it means voluntarily discontinuing very high-floor models for the sake of safety.
2. Conversely, manufacturers may adopt the attitude that the tilt test is archaic, bears no relationship to on-the-road rollover stability, and therefore dismiss it as of no value. Any means are therefore thought justified to fulfil the legal requirement, in order to sell their vehicles in Britain.

(It is hoped that this report will restore manufacturers' confidence in the usefulness of the tilt test as an indicator of rollover stability, Chapter 5.)

Unfortunately, it is feared that some manufacturers, past and present, may have compromised in other areas (such as, superstructure strength or seat and seat mounting impact properties) in order to satisfy tilt regulations. Clearly, this is undesirable, and highlights the need for a coordinated approach. The new ECE superstructure strength requirements are a step along that path.

Thus, if a stability requirement is established too high, it risks losing manufacturers' respect due to technical considerations and conflict with market economics, such as customer expectations of higher and more luxurious coaches.

Between the upper and lower bounds, the final choice is somewhat arbitrary and even political. The figures suggested are considered to be acceptable, from both "safety" and "technically-feasible" points-of-view.

8.3.2 Effects of Proposed Changes on Rollover Safety

Some may feel that these proposals will inevitably lead to a significant increase in occupant injuries and fatalities due to PSV rollover accidents. The standards for double-deckers would be virtually unchanged. The parametric study in Chapter 5 suggests that the actual effect of varying vertical suspension stiffness is small, implying that the effect of temporarily disabling self-levelling valve operation is likely to be less than one degree of tilt. The only double-deckers whose standard would be slightly lowered are the small proportion without air suspension, who previously were at a disadvantage in the tilt test. Those with air springs (now becoming universal in Britain) would face the same rollover stability standard as before.

The reduction in tilt angle required of current single-deckers is not as large as it may initially appear, because many existing coaches have a "tilt-aiding device" (further details in Section 8.4) to artificially improve their tilt performance from 33 or 34 degrees to meet the legal 35 degrees. Nevertheless, it is important that passenger safety does not suffer, and, to avoid this occurring, action taken must consider the situation as a whole.

The valid concerns expressed in the 1920s about overturning stability of the original double-deckers led to the tilt test regulations which are unchanged since 1930. The tilt angle limits were arrived at rather arbitrarily with no accident data on which to make informed decisions. Figure 8.2 (from Ervin, et al, 1980), which relates rollover threshold to overturning accident

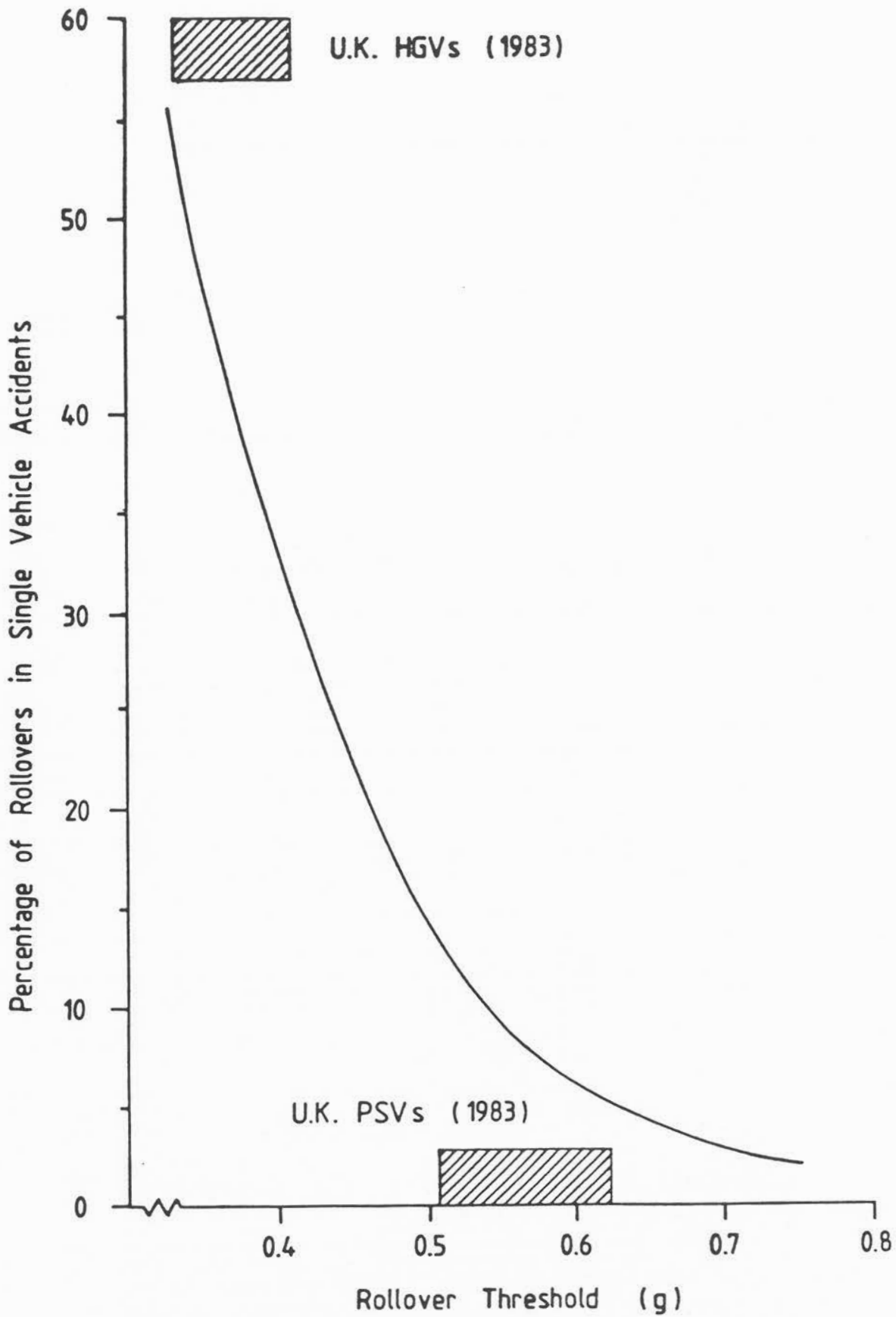


Figure 8.2 Relationship between Steady Cornering Rollover Threshold and Rollover Accident Statistics

frequency, was obtained by examining extensive computerized accident data files for heavy goods vehicles in the United States. Additional zones have been marked on Figure 8.2 representing the performance of PSVs and HGVs in the UK, using accident data from the Department of Transport (1983) and with knowledge of typical tilt test results for both types of vehicle, taking into consideration the range of possible loading conditions. It is apparent from this curve that the effect of altering the overturning stability standard (either increasing or decreasing) on rollover accident rates is very small for PSVs compared with HGVs (indeed, research attention throughout the world in the area of vehicle rollover stability appears to be focussed predominantly on articulated goods vehicles, particularly tankers).

No other country is known to have a roll stability test for buses or coaches. It is even possible that the existence in Britain of the tilt test has, in the past, diminished PSV rollover safety, by tempting some designers to decrease too far certain factors also affecting safety (mentioned above), in order to satisfy the tilt regulation. Provided adequate measures to address all the interrelated safety aspects are implemented together, it is the author's view that passenger safety would be increased.

8.4 REVIEW OF "TILT-AIDING DEVICES"

A "tilt-aiding device" is defined here as any device which improves the tilt test performance of a vehicle without improving its performance in a typical rollover accident.

8.4.1 Tilt Valves

8.4.1.1 Description

A "tilt valve" is an electropneumatic switch designed to operate when the vehicle body rolls beyond a specified angle. As a result, air pressure is reduced from one or more air bellows in order to improve the apparent performance of the vehicle in a tilt test.

There are three main types of switch mechanism:

- pendulum switch,
- mercury switch, and
- ball in vee (Van Hool patent).

It was suspected that tilt valves release air from only the uphill air springs. However, the facts obtained are that one manufacturer (Bedford) deflates all air springs quickly if 20

degrees body roll is exceeded for at least 30 seconds. Another (Scania) releases air slowly from the air springs at the front axle only, whenever the body roll angle exceeds 30 degrees.

The tilt performance is improved partly due to the lowering of the CG of the body, and also to the increase in roll stiffness when the bump stops are contacted.

8.4.1.2 Effects on Safety

Scania kindly demonstrated the effect of actuation of their tilt valve while driving on the perimeter track at Cranfield Airport. The subjective result was a decrease in ride comfort (due to running on front bump stops), and no indication that accidental release of air would cause an uncontrollable situation.

Because Britain is the only country with such a tilt test requirement, PSVs sold in Britain are the only ones to have tilt valves fitted. Otherwise identical vehicles are sold abroad without the tilt valves.

8.4.1.3 Implications for Tilt Test

The tilt test was instituted for the purpose of setting minimum standards for overturning resistance. Fitting a tilt valve enables a vehicle which would otherwise fail the legislative test, to pass. The valves have operating characteristics which are designed to, if possible, never function at all during the life of the vehicle, and they are not intended to operate in an accident situation.

Therefore, it is apparent that tilt valves contravene the original intentions of the tilt test regulations (if not the actual letter of the law). Because of this, these devices should be prohibited.

8.4.2 Self-Levelling Valves

8.4.1.2 Description

Suspension displacement sensors activate control valves which maintain a vehicle at its preset ride height, irrespective of its static loading. This is normal practice with air suspension and allows lower spring stiffnesses to be used (improving ride performance) without excessive spring travel or variation in step height.

Conventional design employs one levelling valve on the front axle (located close to the centreline) and two at the rear, near the axle extremities. This configuration responds to and compensates

for both vertical displacement and roll. For example, the ride height is essentially unchanged despite fluctuations in the number of occupants in the vehicle. Also, if all the passengers preferred to sit on one side of a coach, the self-levelling valves are able to eliminate the offset roll angle which would otherwise result. Similarly, if the air springs on one axle have different stiffnesses (possibly due to dimensional tolerances), or one side was leaking, the vehicle would appear lop-sided without self-levelling valves to maintain the trim condition. With appropriate design methods it is possible to control the response time to a given input, and together with a "dead-band" zone (that is, a limited range of control lever positions near the equilibrium position within which the valve does not respond) the influence of most road bumps on levelling valve operation is minimised. These valve characteristics can significantly influence the dynamic performance of a vehicle, particularly its ride and handling.

Many modern coaches have self-levelling valves with shorter response times than previous models. These are claimed to actually reduce the body roll during cornering (a simple form of active suspension!).

8.4.2.2 Implications for Tilt Test

Any bus or coach with self-levelling valves will reach a greater tilt platform angle while remaining stable, than without them. This is due to the roll-compensation feature described above. Some manufacturers use two self-levelling valves on the front axle, making a total of four valves, primarily to increase the roll compensation effects and hence improve the tilt performance.

8.4.3 Recommendations

Tilt valves are designed purely to enable a vehicle to pass a tilt test. They serve no other function. It is recommended that they be prohibited.

Self-levelling valves serve a legitimate and necessary function. They do, however, also improve the tilt performance to some degree (while having virtually no effect on kerb-impact rollover behaviour) and some manufacturers exploit this property. Their primary function concerns the ride and handling of the vehicle, and it is considered to be ill-advised to compromise this in order to meet tilt legislation. Expressed plainly: self-levelling valves should be used for their original purpose and not to improve tilt test results. To ensure that this occurs, it is recommended that self-levelling valves be made inoperative for the duration of a tilt test.

8.4.4 Discussion

The issue with tilt valves is straight-forward, and requires no further explanation.

The technical matter of disabling self-levelling valves for a tilt test may require some thought and consultation with manufacturers and certifying officers to establish the most feasible methods. Mai and Sweatman (1984) successfully tilt tested a tractor and semi-trailer combination with the self-levelling valves on the trailer made inoperative.

A separate matter concerns the tilt test performance of current double-deckers, which constitute the foundation of the proposed new overturning stability standards (outlined in Section 7.3). It is not known at present what tilt angle would be reached by current double-deckers without their self-levelling valves operating. Additional tilt tests are therefore required to determine the new tilt standard for tall coaches. It is suggested that at least two double-deckers without tilt valves and with only three self-levelling valves each (to avoid vehicles with enhanced tilt performance) be tilt tested both with and without the self-levelling valves functioning. The average reduction in tilt angle, subtracted from the current 28 degree tilt platform requirement would give the proposed new tilt limit for PSVs over 4 metres high.

Technical developments in the future could quite conceivably result in new or revised devices, concepts or systems which will need to be evaluated to determine whether they would enhance tilt test performance without improving curb-impact rollover performance. Active and semi-active suspension designs (in which conventional passive springs are wholly or partly replaced with hydraulic cylinders, whose lengths are controlled to compensate for road irregularities, cornering forces and such like, usually by feedback from wheel loads) should be mentioned as a potential problem case regarding overturning stability testing. Some investigations may be required to determine tilt testing policy for such vehicles. One suggestion is that, unless it can be clearly demonstrated that an active suspension system responds fast enough and would improve curb-impact rollover stability, these vehicles should be tilt tested with the suspension locked in the "normal ride position".

CHAPTER 9

CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

(Note that those conclusions and recommendations considered by the author to be of greater significance are indicated with an asterisk.)

9.1.1 Chapter 1 INTRODUCTION

Although coach travel is relatively safe compared with other forms of road travel, occasional severe accidents continue to occur. A review of detailed accident data from Great Britain, France and West Germany indicates that rollover accidents, although relatively few in number, cause a disproportionately large number of passenger casualties. Improving bus rollover safety ought to be a high priority among bus studies.

If a coach rolls further than just onto its side, the injuries to occupants are found to rise significantly. Passenger ejection and structural collapse of the vehicle are the major causes of fatalities.

Characteristics of a typical British bus rollover accident include lateral motion, impact with a "tripping" obstacle and rollover down a drop of up to 1 metre. A chain of events leading to occupant casualties in such an accident has been identified and measures aimed at breaking the chain at each of its links are considered. The main countermeasures suggested are to:

- Include skid-recovery as an essential skill required of PSV drivers, and therefore included in the licence test.
- Investigate the controllability in emergency manoeuvres of a range of buses and coaches to assess the potential for improvements in limit performance, particularly to minimise the loss of control.
- Study "tripping" rollover accident behaviour to evaluate whether the U.K. tilt test is a good indicator of the relative ease (or difficulty) of overturning a vehicle in this type of accident.
- Undertake an investigation into suitable means of retaining passengers within the bus, to prevent ejection, while maintaining full emergency exit standards.

In addition, in view of manufacturers' criticism of the tilt test as severe and having no established relevance to reality, an extra avenue of study was identified:

- Determine whether an overturning stability test is needed at all and, if so, study the suitability as a legislative test of other possible methods.

9.1.2 Chapter 2 SIMULATION OF DYNAMICS OF BUS ROLLOVER TEST

A full-scale bus rollover test, based on the typical "tripping" accident was devised and refined, primarily as a superstructure strength test to ensure adequate survival space during rollover impact loading. Problems of uneven cantrail contact at impact with the ground due to pitch and yaw were encountered when attempting to test vehicles with uneven axle weight distributions. A computer simulation known as SIDBURT (for SIMulation of the Dynamics of a BUS Rollover Test) was developed to examine the dynamic behaviour in detail. It features:

- six degrees-of-freedom rigid body motion,
- detailed non-linear "tripping"-mechanism characteristics, and
- comprehensive graphical output (including three-dimensional perspective viewing capability).

After correction of an error in the axis transformation routine the simulation compares favourably with full-scale rollover tests.

The trajectory of the vehicle shows particularly good agreement between experiment and theory, but the prediction of velocities and energies after the initial impact is not as accurate because of the largely unknown and unmeasurable energy dissipation during the rollover tests. A similar magnitude of energy was absorbed totally as plastic deformation of force-limiting crush tubes (at the kerb impact) in the computer model, but the energy losses in the full-scale test were more diverse and unpredictable and consequently unrepeatable. This factor caused this particular method of testing superstructure strength to be withdrawn as the U.K. proposal to a United Nations group of European bus experts. A simpler, though tougher, alternative test method was adopted instead.

The computer model is not made redundant by this decision, though. It has been developed further by the addition of front and rear suspensions and modified to be able to simulate the impact phase of actual rollover accidents.

9.1.3 Chapter 3 VEHICLE DATA MEASUREMENT

Tests were performed on nine vehicles ranging from city buses to luxury touring double-decker coaches to determine properties required for the computer simulations.

Centre of gravity heights ranged from 0.94 to 1.56 metres above the ground when unladen, and 1.10 to 1.76 metres when the worst case loading of passengers is included.

The equipment for roll moment of inertia measurement gave erratic results. For the purposes of kerb-impact rollover, however, the value is not critical, and an estimate based on a known approximate relationship between roll inertia and mass is adequate.

Evaluation of roll centre heights from analysis of carefully arranged displacement transducers proved to be feasible, although a dozen or more measurements are preferable and great care is necessary. If this is not done then the results have a great deal of scatter. Old multi-leaf springs give poorer accuracy due to the stick-slip nature of their inter-leaf friction. A pure roll moment test (in which masses are shifted from one side to the other) is considered to be preferable to a tilt test for measurement of roll centre heights, due to the additional lateral and vertical displacements influencing the tilt results.

The same displacement analysis method was used to resolve linear potentiometer output into three component displacements for the calculation of vertical, lateral and roll stiffnesses for the suspension and tyres. In retrospect, the lateral forces in the tilt test could be more simply calculated from the axle load and tilt table angle, with the resultant lateral tyre stiffness representing the combined effect of both wheels on the ground.

Maximum stable tilt angle was noted in the tilt tests, although test conditions were not exactly as for a type approval tilt test (due to such differences as the linear bearings and individual weighpads under each wheel, and the one or two unrestrained sandbags).

Consideration of the poor objective and subjective stability of one of the two coaches imported from the Continent highlights the benefit of an overturning stability test: such vehicles are prevented from being sold in Britain.

9.1.4 Chapter 4 KERB-IMPACT ROLLOVER ACCIDENT SIMULATION PROGRAM

A review of previous methods of "tripping" rollover analysis reveals that all available examples consider only the motion of a rigid block, neglecting both the suspension effects and the

three-dimensional behaviour which results from the different suspension properties at each axle.

RASP (Rollover Accident Simulation Program) includes both of these features. Vertical, lateral and roll displacement of each axle relative to the sprung mass are permitted. The relative roll motion is assumed to take place about a roll axis fixed to the sprung mass. The roll axis is defined by specifying its height above the ground at the front and rear axles. These roll centre heights need not be equal, implying that the roll axis can be inclined to the horizontal. In addition, the axle and sprung mass CGs are positioned independently of the roll axis, necessitating additional terms in the derived equations of motion due to the assumption that roll occurs about the roll axis, rather than about the CG.

Direct validation with an equivalent full-scale test would be difficult to achieve because of the variability of this type of impact, but the comparison in Chapter 2 between a full-scale bus rollover test and SIDBURT, an earlier single-rigid-body version of the simulation, shows reasonable correlation. Agreement is also good with a steady-state cornering simulation and a simple kerb-impact analysis.

Results from RASP for the tested vehicles gives threshold lateral velocities for rollover ranging from 2.8 to 4.2 m/s.

9.1.5 Chapter 5 THE TILT TEST AND TILT SIMULATION

The origins of the tilt test have been probed, and the developments in bus design and operation since that time are reviewed in order to assess the suitability of the tilt test in its present form to current practices.

A tilt test simulation which compares adequately with experimental results has been developed. For better agreement, non-linear spring characteristics and additional degrees of freedom could be included. The main purpose of this tilt test simulation, however, is to compare with the rollover accident simulation program (RASP), particularly to indicate the effect on tilt performance of varying certain vehicle parameters.

9.1.6 Chapter 6 COMPARISON OF ROLLOVER ACCIDENT SIMULATION AND TILT TEST SIMULATION

- * The CG height and track width are the parameters which affect the rollover and tilt performance the most.

Some differences have been noted between the two simulations:

- * ● Moment of inertia, and suspension and tyre damping do not affect tilt performance.
- * ● Roll stiffness has little effect on rollover threshold, but significantly influences tilt results.
- An optimum value of roll centre height exists for rollover performance. The threshold velocity cannot be increased by raising the roll centre heights further. Maximum tilt angle, however, continues to increase with roll centre height. In reality, this is worth noting but may not affect suspension design because the optimum values are towards the upper end of typical design practice.

In addition, the lateral stiffness is important for the kerb-impact behaviour, but its effect on the tilt limit could not be assessed without the addition of further degrees of freedom to the tilt simulation. While acknowledging that these differences exist, the direct comparison of results demonstrates that their combined effect is minor, implying that, in general, the tilt test provides a reasonably accurate indicator of kerb-impact rollover threshold.

9.1.7 Chapter 7 DISCUSSION OF BEST METHODS OF EVALUATING ROLLOVER STABILITY

- * Nine possible test methods are examined to ascertain their suitability as a legislative overturning stability test. Although the vertical load in the tilt test decreases with tilt angle (contrary to on-the-road conditions in which it is virtually constant), this effect was not found to be significant. It therefore offers the simplest, reasonably accurate assessment of the rollover stability of a bus or coach.
- * The rollover accident simulation is considered to give the best indication of a vehicle's propensity to overturn in a kerb-impact situation, at the cost of significantly more experimental testing to obtain the necessary input data. It is suggested that this method be permitted as an alternative means of meeting the rollover stability requirements (perhaps encouraging further research into this important area of rollover safety.)

9.1.8 Chapter 8 RECOMMENDED ROLLOVER STABILITY TEST REQUIREMENTS

Standards for the two overturning stability tests are proposed, considering the conflicting requirements of minimising rollover accidents and setting a technically attainable standard. If the standard is set too low then safety deteriorates. If the standard is set too high then safety will probably also deteriorate, as designers are forced to compromise in such other important areas as seat and seat mounting impact properties, to keep the CG as low as possible. Alternatively, some manufacturers may consider the regulation as an unnecessary obstacle, merely requiring some ingenuity to circumvent.

- * This is the present case with some coach builders who utilise a "tilt-aiding device" to satisfy the legal requirements for a vehicle which, without that device, would not pass. One such device, known as a tilt valve, is recommended to be prohibited since it serves no other function. Self-levelling valves also assist the tilt performance, but perform an essential function in controlling the suspension ride height of the vehicle. During a tilt test the self-levelling valves are recommended to be disabled to prevent their levelling function from operating.
- * Proposed stability standards are summarised in Table 9.1, with a sliding scale between the two standards. The lower category arises because of the greater probability of roll exceeding 90 degrees for the lower buses. Larger roll angles generally result in increased occupant fatalities and injuries. The higher stability standards are intended to reduce the occurrence of overturning accidents.

Table 9.1
Summary of Proposed Rollover Stability Standards

Method	Overall Laden Height (m)	
	> 4.0	< 3.0
Vehicle to be stable on tilt platform at: (deg)	'A'	'A'+3
Vehicle to laterally impact kerb without rollover at velocity of: (m/s)	3.0	3.2

(where 'A' is the tilt performance of typical current double-deckers without self-levelling valves operating)

- * The distinction between the two categories is based not on the traditional seating configurations (single- and double-deckers), but on overall height, since is the factor which affects the total rollover angle and hence accident severity, when a rollover occurs.
- * Finally, the current double-decker tilt standard is recommended to be remeasured both with and without self-levelling valve operation, and the new tilt standard for PSVs over 4 metres high be based on the results for the latter condition.

9.2 RECOMMENDATIONS FOR FURTHER WORK

9.2.1 Bus Safety

To complete this study requires the tilt testing of one or more double-deckers, with and without their self-levelling valves functioning. This forms the basis for the new recommended tilt angle.

Several more fundamental aspects of bus safety require further research. The two most important areas to study are:

- *
 - The problem of passenger retention, particularly in rollover, but also in frontal and side impacts.
 - The "loss-of-control" phase, to determine whether there is potential for improving the controllability of buses in extreme manoeuvres, such as braking in a turn.

9.2.2 Instrumentation

Owing to the uncertainty associated with the moment of inertia measurements, some calibration and even redesign may be necessary to restore confidence. The present apparatus was constructed largely from scrap components available at the time. As a minimum, the self-aligning bearings need to be cleaned, inspected and lubricated. The following steps are also recommended:

- Air to all air springs to be supplied from one filling point, with a single calibrated pressure gauge with a large dial.
- Metal tubing is preferred in connecting airsprings on the same beam.
- Flexible hose with connectors would be required to link springs on one beam to the others.
- Valves to isolate airsprings from each other are necessary for the moment of inertia test.

- Tests should be conducted with the beams bolted to the floor, preferably indoors (to minimise wind effects).

9.2.3 Development of Simulations

SIDBURT or RASP may be used for other applications - the choice depending on the chosen compromise between accuracy and speed. (If suspension motions may be neglected, then SIDBURT is appropriate.) For example, with suitable development either program could be used to study the motion of an occupant in a kerb-impact overturning accident, perhaps as one means of assessing various passenger restraint possibilities.

Either could be included into a comprehensive rollover accident simulation, incorporating vehicle dynamics and structural collapse phases.

If RASP was considered as a possible method for Type Approval evaluation of rollover stability, then some effort in improving the efficiency of the various routines may be justified. The plotting routines for SIDBURT could also be adapted to use with RASP output.

REFERENCES

- BATIZ, Z. Analyse von Tatsächlichen Autobusunfällen.
XV Meeting of Bus & Coach Experts,
Scientific Society of Mechanical Engineers,
Hungary, 1984.
- BENDER, C.M. Advanced Mathematical Methods for Scientists
ORSZAG, S.A. and Engineers.
International Student Edition, McGraw-Hill,
Tokyo, 1978.
- BICKERSTAFF, D.J. The Handling Properties of Light Trucks.
SAE 760710, Automobile Engineering Meeting,
Dearborn, Michigan, October 18-22, 1976.
- BOX, M.J. A New Method for Constrained Optimisation
and a Comparison with Other Methods.
Computer J., Vol. 6, p.42, 1965.
- DER BUNDESMINISTER Verkehrsnachrichten.
FÜR VERKEHR Heft 1, Bonn, January 1985
(referred to by Langwieder, et al, 1985).
- COHEN, L. Statistics for Social Scientists.
HOLLIDAY, M. Harper and Row, London, 1982.
- DANSKIN, N. Safety Regulations for Public Service
Vehicles.
Conf. on Vehicle Safety Legislation,
Cranfield, IMechE Paper No. C199/73, 1973.
- DEPARTMENT OF Road Accidents Great Britain 1982.
TRANSPORT HMSO, 1983.
- DEPARTMENT OF Road Accidents Great Britain 1983.
TRANSPORT HMSO, 1984.
- ERVIN, R.D. Future Configurations of Tank Vehicles
MALLIKARJUNARAO, C. Hauling Flammable Liquids in Michigan.
GILLESPIE, T.D. University of Michigan, Highway Safety
Research Inst., Report No. UM-HSRI-80-73-1,
1980.
- ERVIN, R.D. The Influence of Size and Weight Variables
on the Roll Stability of Heavy Duty Trucks.
University of Michigan, Transportation
Research Inst., SAE paper no. 831163, 1983.

- FANCHER, P.
SEGEL, L.
BERNARD, J.
ERVIN, R. Test Procedures for Studying Vehicle Dynamics
in Lane-Change Maneuvers.
SAE 760351, 1976.
- GILLESPIE, T.D.
VERMA, M.K. Analysis of the Rollover Dynamics of
Double-Bottom Tankers.
SAE 781065, 1978.
- GOULT, R.J.
HOSKINS, R.F.
MILNER, J.A.
PRATT, M.J. Applicable Mathematics - A Source for
Scientists & Engineers.
pp 366-369, MacMillan, London, 1973.
- HIDVEGI, L. Some Test Results of Lateral Stability
Tests of Buses.
Conference of Bus Experts, Hungarian
Scientific Society of Mechanical Engineers,
Budapest, 1981
- HOSKING, R.J.
JOYCE, D.C.
TURNER, J.C. First Steps in Numerical Analysis.
Hodder & Stoughton, Auckland, 1978.
- ISERMANN, H. Overturning Limits of Articulated Vehicles
with Solid and Liquid Loads (Die Kippgrenze
von Sattelkraftfahrzeugen mit fester und
flüssiger Ladung).
MIRA Translation No. 58/70, 1970.
- JONES, I. The Mechanics of Rollover as the Result of
Curb Impact.
SAE 750461, 1975.
- JONES, I. The Effect of Vehicle Characteristics on Road
Accidents.
pp130-132, Pergamon, 1976.
- KECMAN, D.
MILES, J.C.
SADEGHI, M.M.
TIDBURY, G.H. Investigation of PSV Rollover Safety.
Conf. on The Design, Construction and
Operation of Public Service Vehicles,
IMechE Paper C139/77, 1977.
- KECMAN, D. Bending Collapse of Rectangular and Square
Section Tubes in Relation to the Bus Roll
Over Problem.
PhD Thesis, Cranfield Institute of
Technology, U.K., 1979.
- KECMAN, D.
HARDY, R.N.
SADEGHI, M.M.
TIDBURY, G.H. Experimental Investigation into the Collapse
Behaviour of PSV Structures in Roll Over.
Conf. on Engines & Vehicles, J.U.M.V.
Kraguevac, 1979.

- KECMAN, D.
SADEGHI, M.M.
TIDBURY, G.H. Research Background to the UK Proposals for the New Bus Rollover Safety Regulations. Conf. on Engines & Vehicles, J.U.M.V. Kraguevac, 1981.
- KEMP, R.N.
CHINN, B.P.
BROCK, G. Articulated Vehicle Roll Stability: Methods of Assessment and Effects of Vehicle Characteristics. TRRL Report 788, 1978.
- KNIGHT, C.
TIDBURY, G. The SDG (Cranfield) Roll Moment of Inertia Test Rig. Conference of Bus Experts, Hungarian Scientific Society of Mechanical Engineers, Budapest, 1981.
- LANGWIEDER, K.
DANNER, M.
HUMMEL, T. Collision Types and Characteristics of Bus Accidents - Their Consequences for the Bus Passengers and the Accident Opponent. Tenth Int. Conf. on Experimental Safety Vehicles, Oxford, 1985.
- MacADAM, C.C.
FANCHER, P.S.
HU, G.T.
GILLESPIE, T.D. A Computerized Model for Simulating the Braking & Steering Dynamics of Trucks, Tractor-Semitrailers, Doubles and Triples Combinations - Users' Manual, Phase 4. HSRI, Report No. UM-HSRI-80-58, 1980.
- MAI, L.
SWEATMAN, P. Articulated Vehicle Stability - Phase II: Tilt Tests and Computer Model. Report No. AIR 323-2, Australian Road Research Board, 1984.
- MATOLCSY, M. Bus Research and Development at Autokut Research Institute, Hungary. Conf. on The Design, Construction and Operation of Public Service Vehicles, IMechE Paper C145/77, 1977.
- MILES, J.C. The Application of Finite Element Methods to Vehicle Collapse Analysis. PhD Thesis, Cranfield Institute of Technology, U.K., 1976.
- NELDER, J.A.
MEAD, R. A Simplex Method for Function Minimization. Computer J., Vol. 7, p.308, 1965.
- NORDSTRÖM, O.
MAGNUSSON, G.
STRANDBERG, L. Tunga Fordonskombinationers Dynamiska Stabilitet. National Swedish Road and Traffic Research Institute, Report No. 9, Stockholm, 1972.

- NUMERICAL
ALGORITHMS
GROUP User Manual.
Mk. 10, Vol. 3, 1983.
- OPENSHAW, M.J. The Dynamic Simulation of the Bus Rollover
Rig.
MSc Thesis, Cranfield Institute of
Technology, U.K., 1981.
- OPENSHAW, M.J.
TIDBURY, G.
WHITE, D. Simulation of the Dynamics of a Bus Rollover
Test.
XV Meeting of Bus & Coach Experts,
Scientific Society of Mechanical Engineers,
Hungary, 1984.
- RIBBANS, J. Basic Numerical Analysis - Book 1.
pp 106-108, Intertext, London, 1969.
- SADEGHI, M.M. Bus Roll Over Accident Simulation.
3rd Int. Conf. on Veh. Structural Mechanics,
Troy, Michigan, SAE 790994 or P-79/83, 1979.
- SEGAL, D.J. Highway Vehicle Object Simulation Model -
Version 2, Users' Manual.
Calspan Corp. Report ZR-5461-V-6 on Contract
No. DOT-FH-11-8265, 1976.
- SEGEL, L.
WINKLER, C.B.
FANCHER, P.S. The Mechanics of Heavy-Duty Trucks and Truck
Combinations.
International Assoc. for Vehicle Design,
Open University course, Sept 14-17, 1981.
- STRANDBERG, L.
NORDSTRÖM, O.
NORDMARK, S. Safety Problems in Commercial Vehicle
Handling.
National Swedish Road & Traffic Res. Inst.,
Report No. 82A, presented at Symposium on
Commercial Vehicle Braking & Handling, 1975.
- STEINER, M. Verkehrunfallanalyse von Omnibussen.
XV Meeting of Bus & Coach Experts,
Scientific Society of Mechanical Engineers,
Hungary, 1984.
- STRUCTURAL
DESIGN
GROUP Reports to the Department of Transport
on the PSV Rollover Research.
S.A.S., Cranfield Institute of
Technology, 1975-1982.
- THOMAS, C.
et al Severe Coach Accident Survey.
Tenth Int. Conf. on Experimental Safety
Vehicles, Oxford, 1985.

- TIDBURY, G.H. The Investigation of 8 Bus Roll Over
Accidents in Great Britain 1976-1977.
Invited Paper to the Hungarian Society
of Scientific Engineers, 1977.
- TIDBURY, G.H. The Design of Bus and Truck Structures
for Passenger and Crew Safety.
Int. J. of Vehicle Design, vol. 5, nos. 1/2,
pp. 67-85, 1984.
- TIMOSHENKO, S.
YOUNG, D.H. Advanced Dynamics.
pp. 119-123, 1st Edition, McGraw-Hill,
New York, 1948.
- U.K. RAPPORTEUR
to G.R.S.A. A Static Lateral Stability Test for Motor
Coaches and Buses.
Dept. of Environment, W/TRANS/WP29/495,
Para. 2, 1973.
- WHITE, D. P.S.V. Rollover Stability.
Tenth Int. Conf. on Experimental Safety
Vehicles, Oxford, 1985.
- WEIR, D.H.
et al Analysis of Truck and Bus Handling, Vol.2.
Systems Technology Inc., Report No.
DOT-HS-801-153, 1974.
- WILKINS, H.A.
RILEY, B.S. The Road Grip of Commercial Vehicle Tyres.
Conference on Road Vehicle Handling, MIRA,
IMechE Paper C135/83, 1983.

APPENDIX A
GRAPHS FOR CG HEIGHT MEASUREMENT

C.G. HEIGHT = 0.944 (m)

C.G. OFFSET = 0.079 (m)

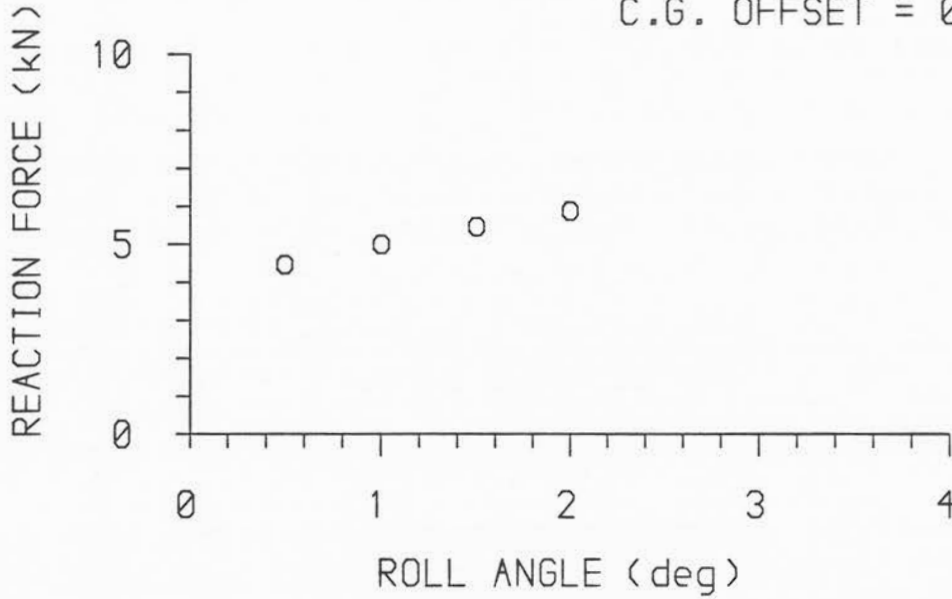


Figure A.1 Leyland National 2

C.G. HEIGHT = 1.076 (m)

C.G. OFFSET = 0.078 (m)

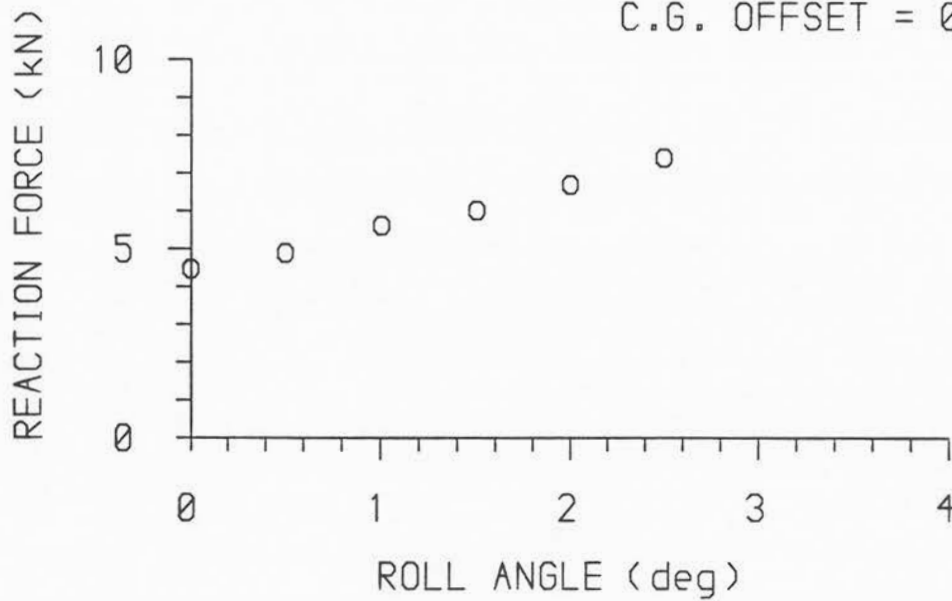


Figure A.2 MCW Metrobus

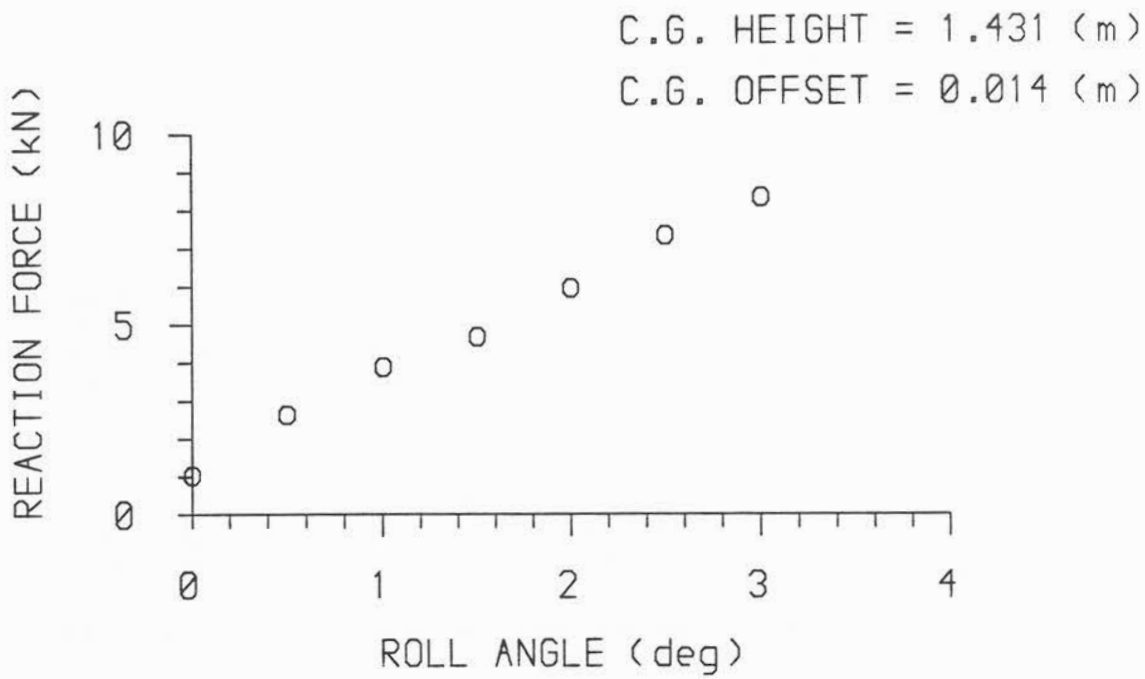


Figure A.3 Van Hool Astronema

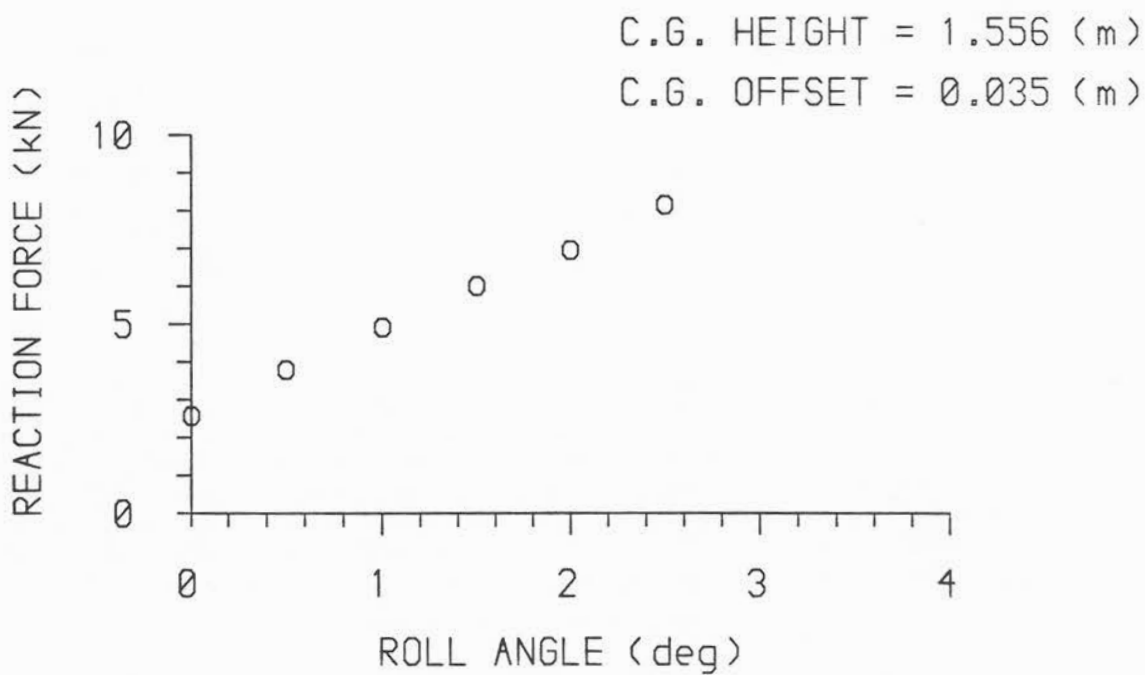


Figure A.4 Jonckheere Jubilee P799

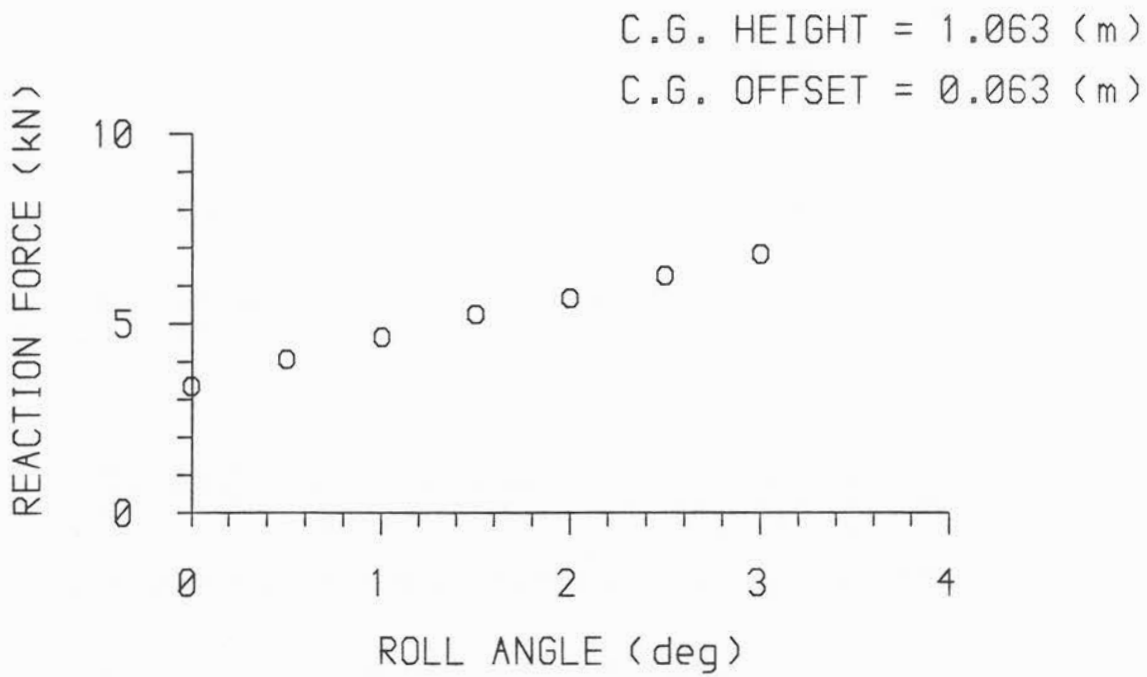


Figure A.5 Plaxton Supreme IV

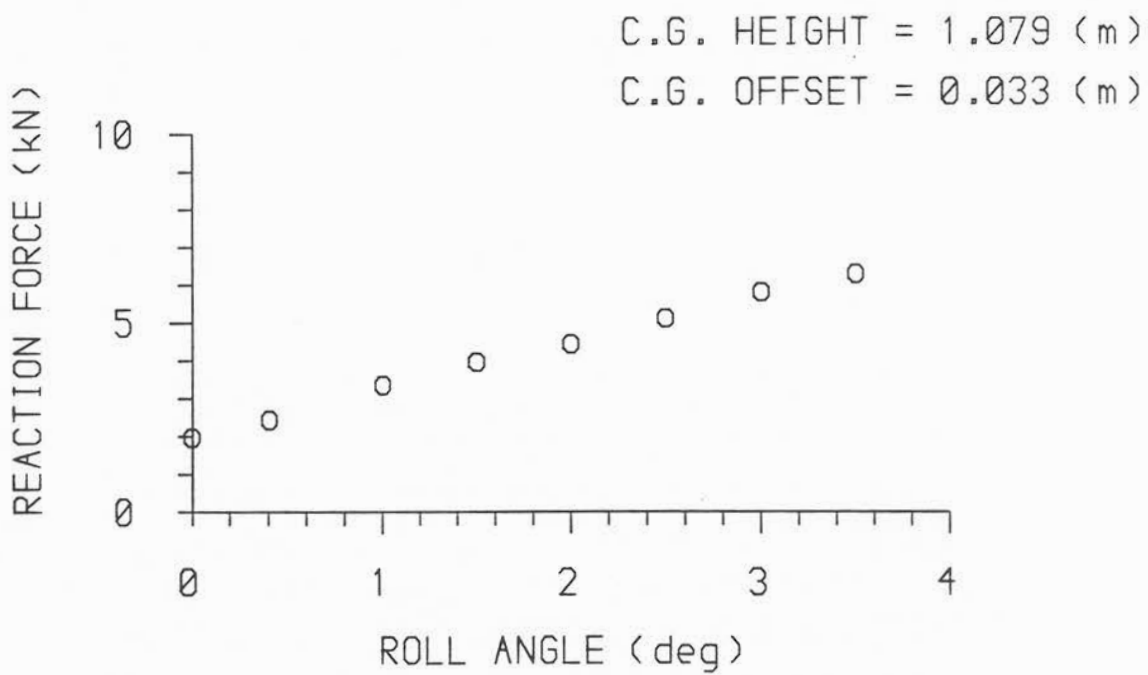


Figure A.6 Plaxton Paramount 3200

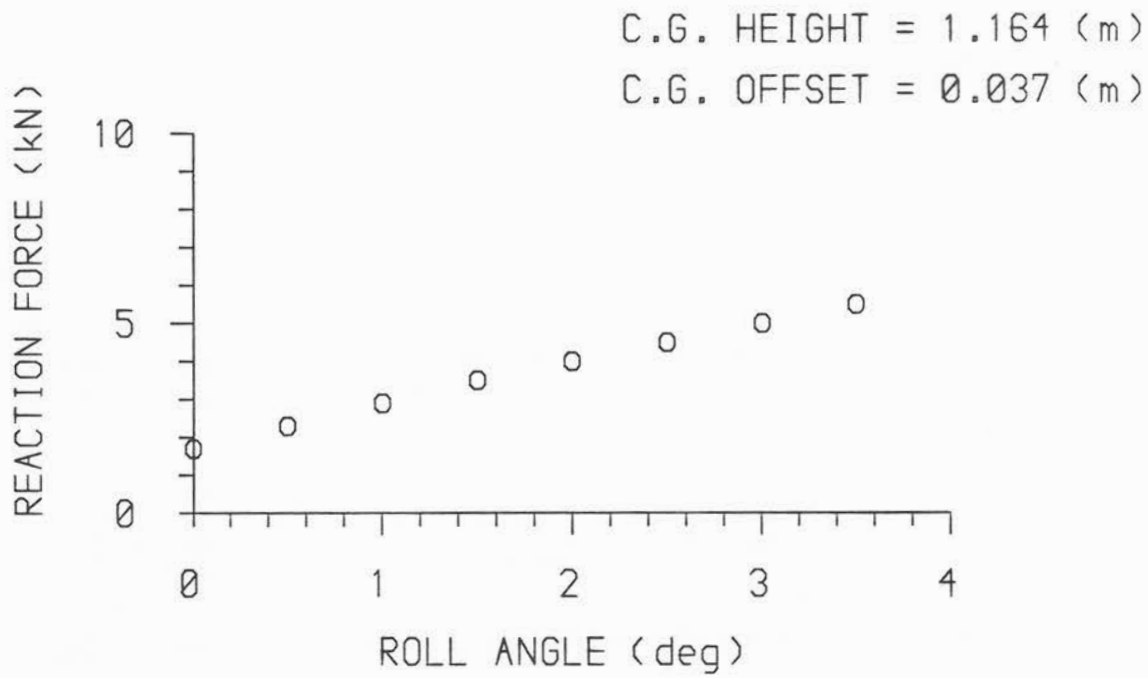


Figure A.7 Duple Laser

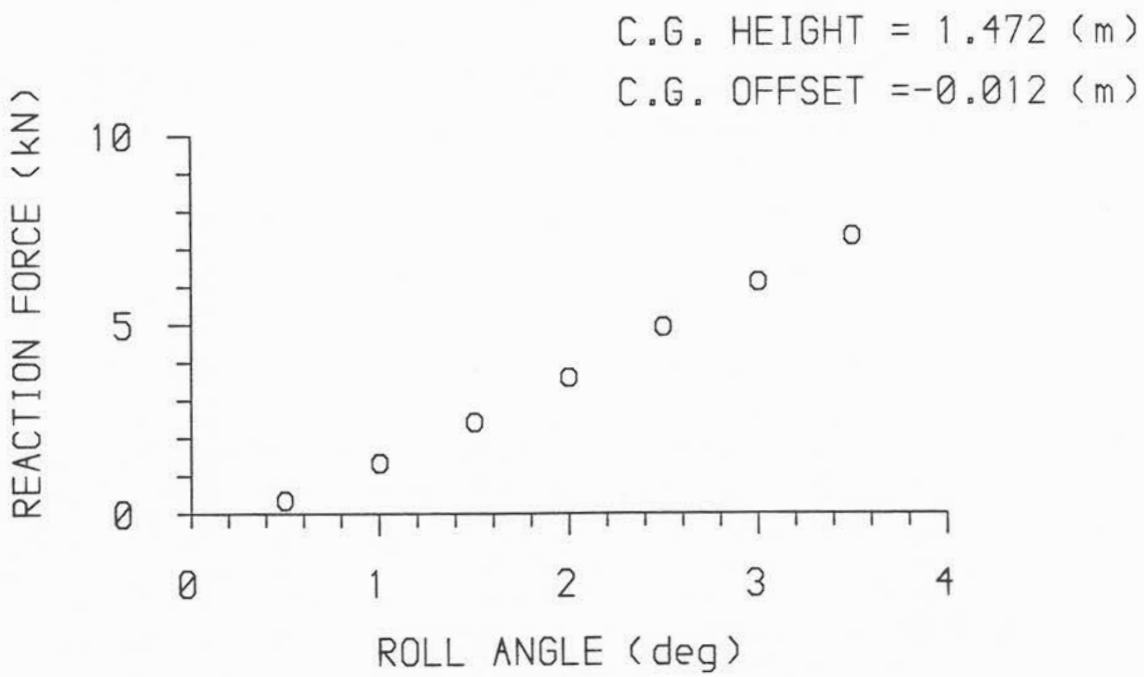


Figure A.8 MCW Metroliner

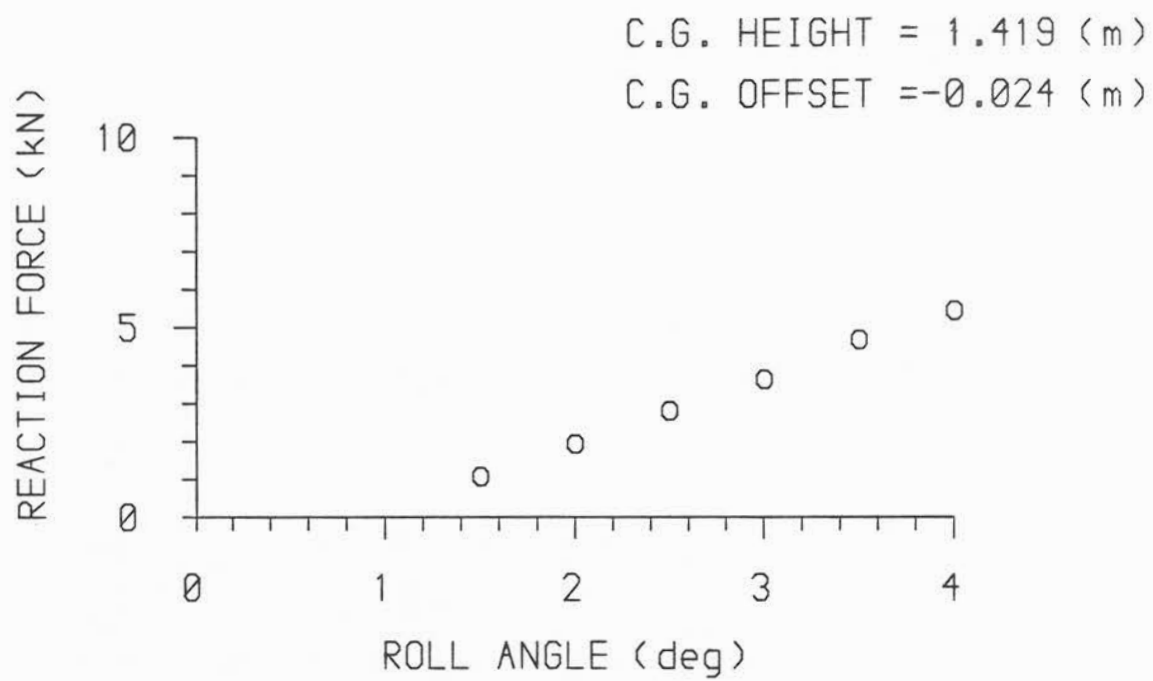


Figure A.9 Leyland Royal Tiger Doyen

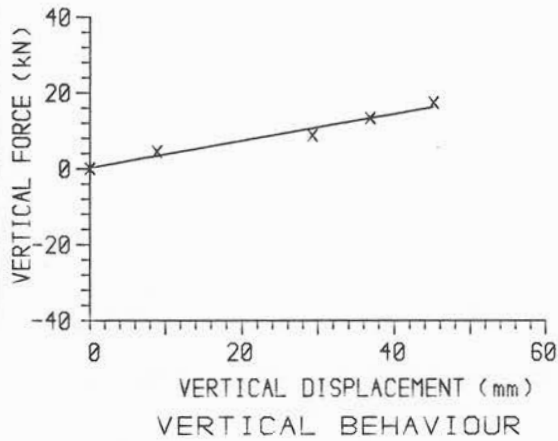
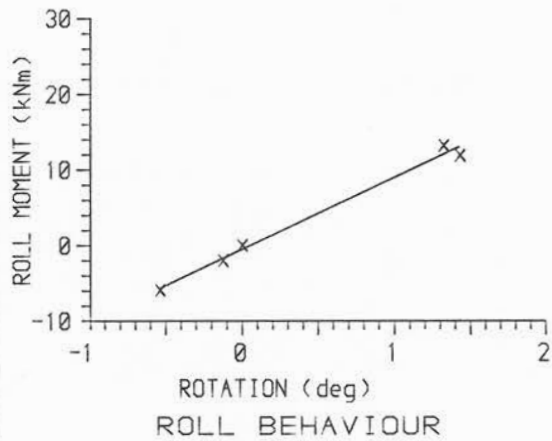
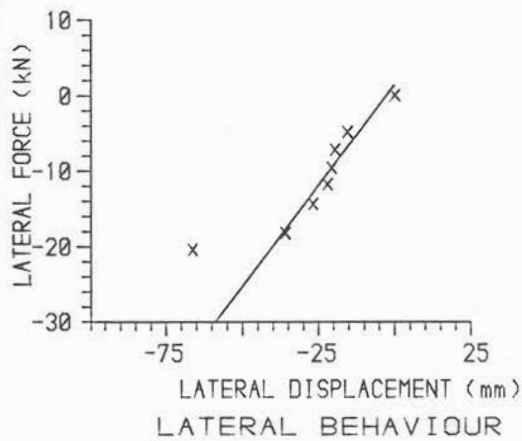
APPENDIX B
GRAPHS OF SUSPENSION AND TYRE STIFFNESS

VEHICLE: LEYLAND NATIONAL 2

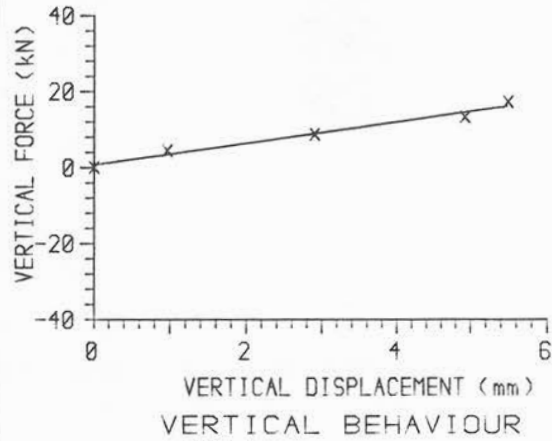
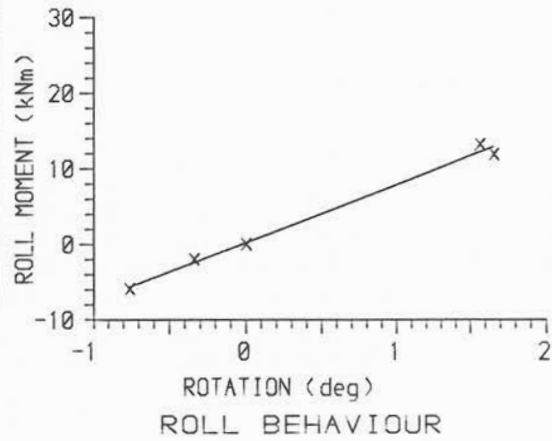
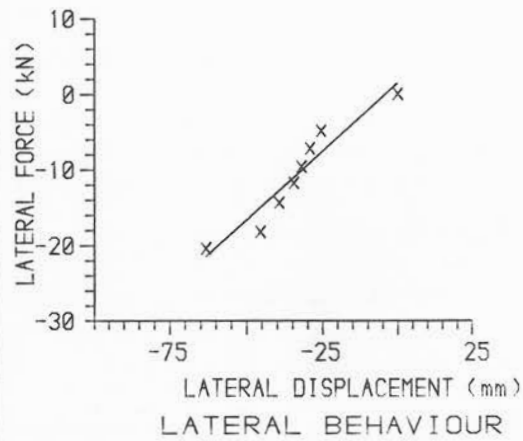
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES

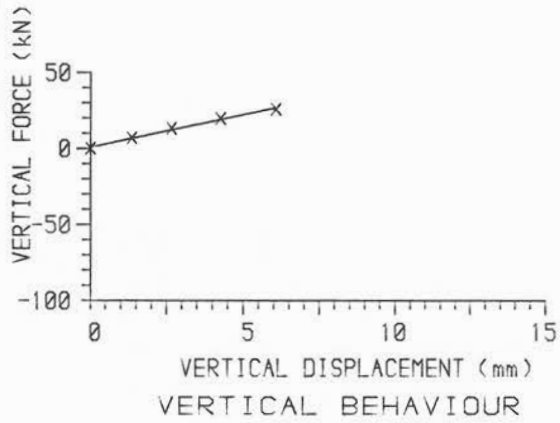
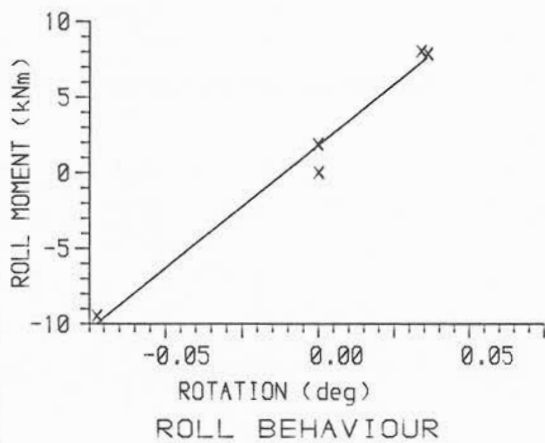
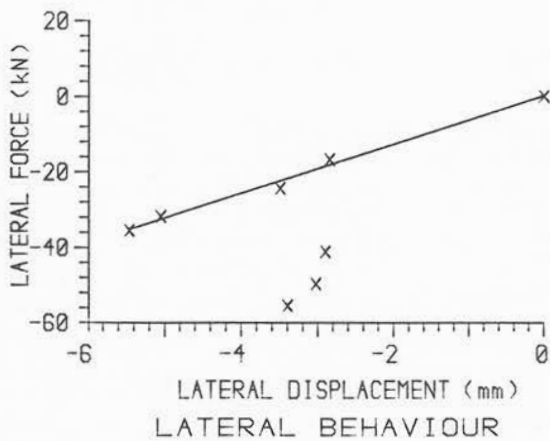


VEHICLE: LEYLAND NATIONAL 2

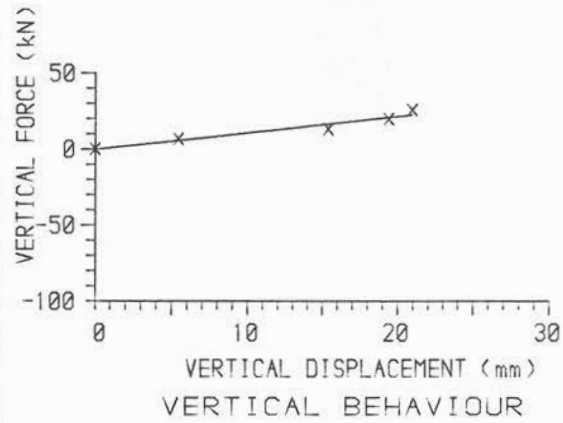
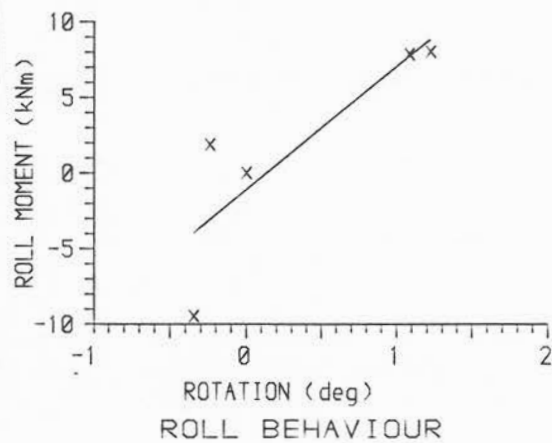
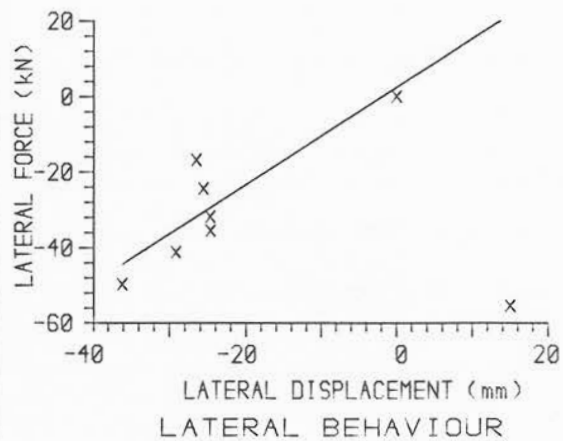
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

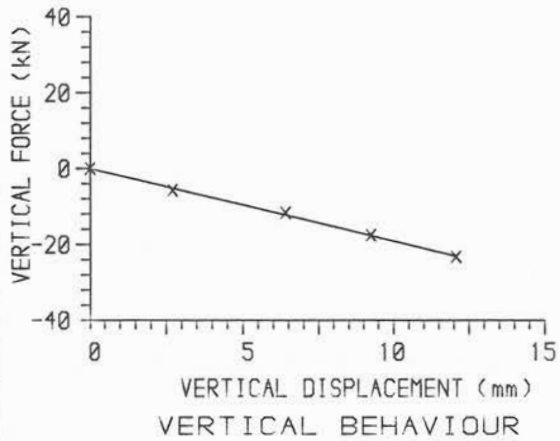
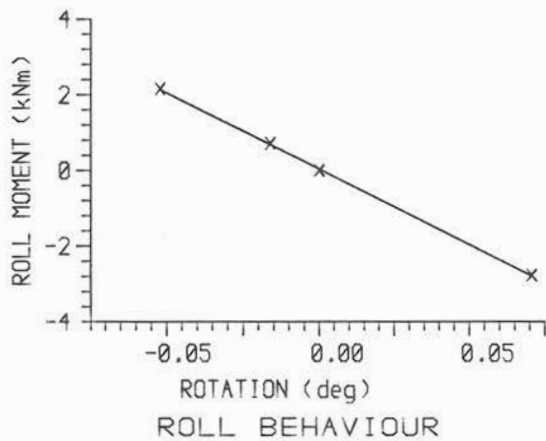
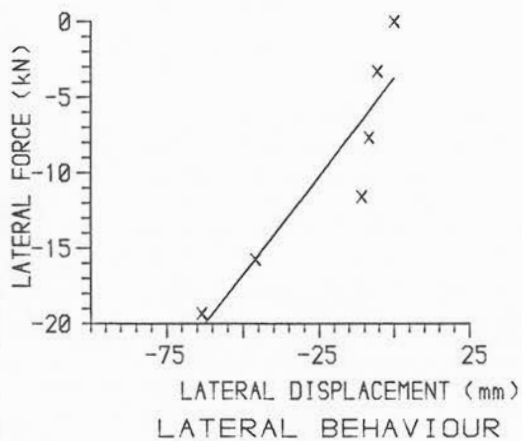
Fig. B.2

VEHICLE : MCW METROBUS

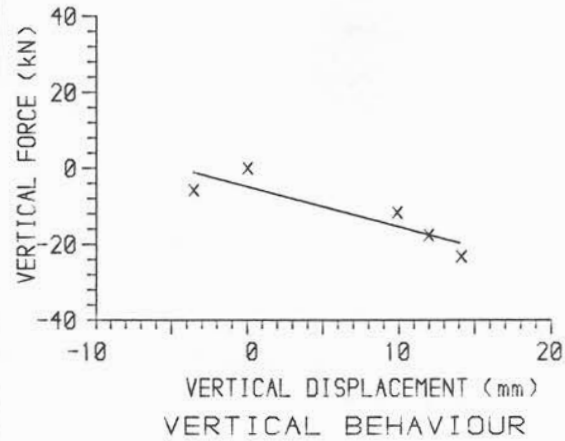
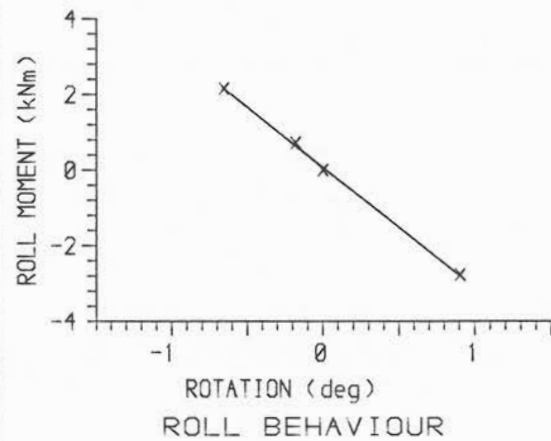
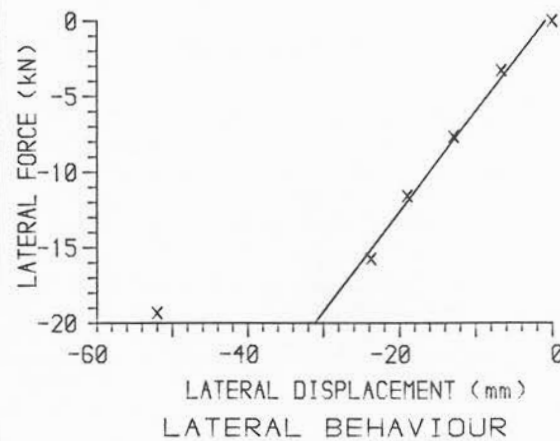
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES

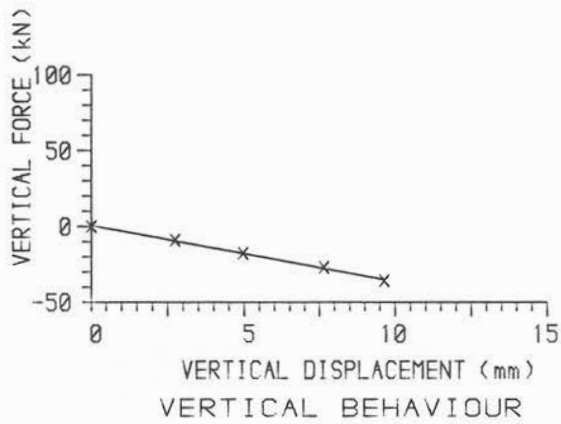
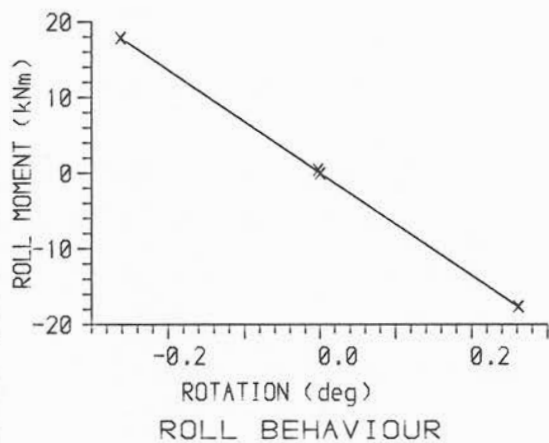
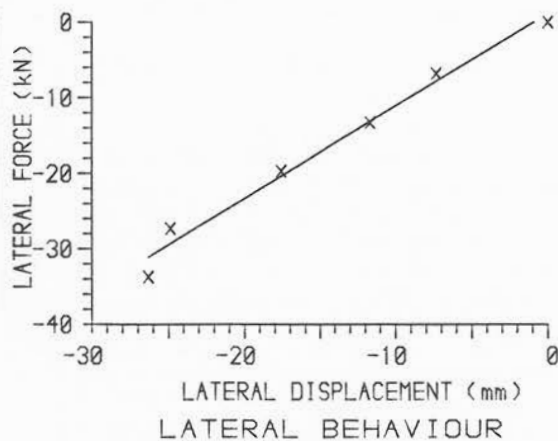


VEHICLE: MCW METROBUS

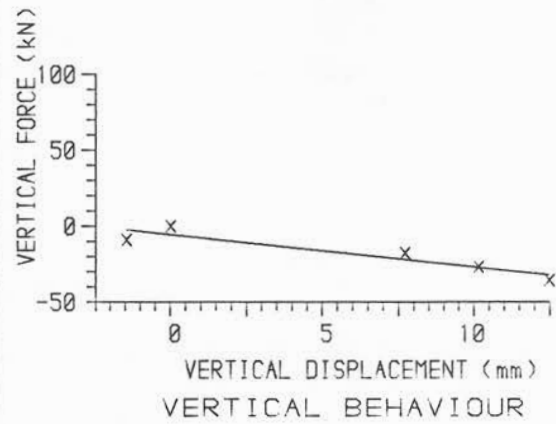
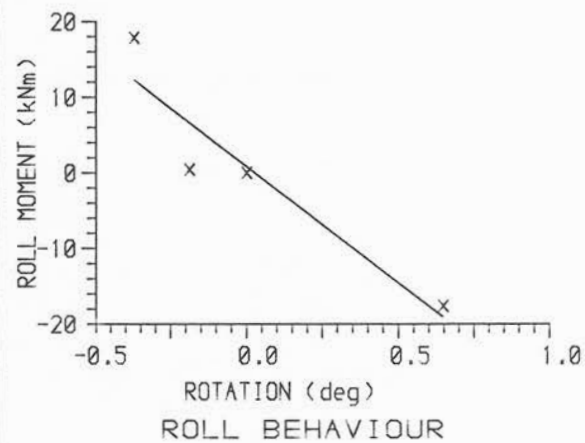
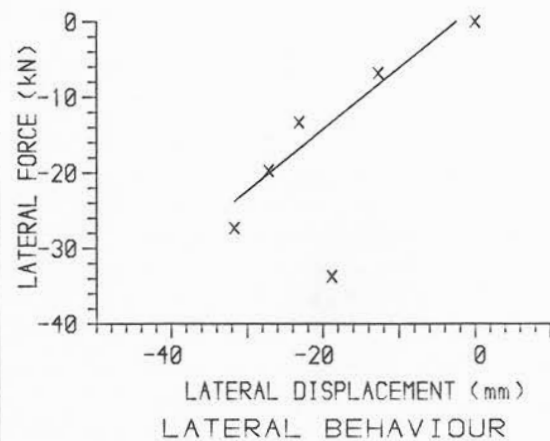
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

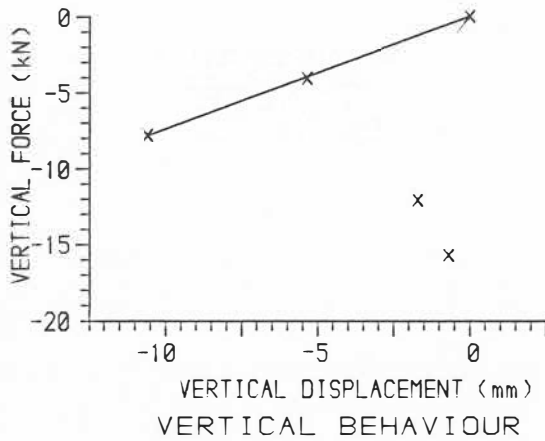
Fig. B.4

VEHICLE: VANHOOL ASTROMEGA (2-axle analysis)

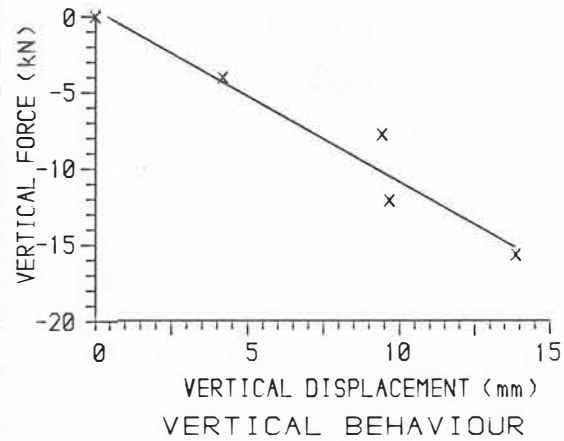
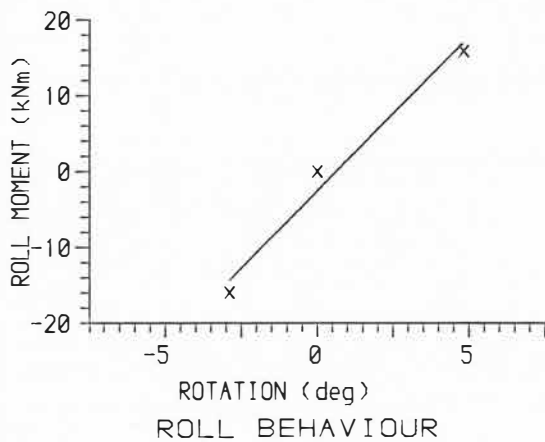
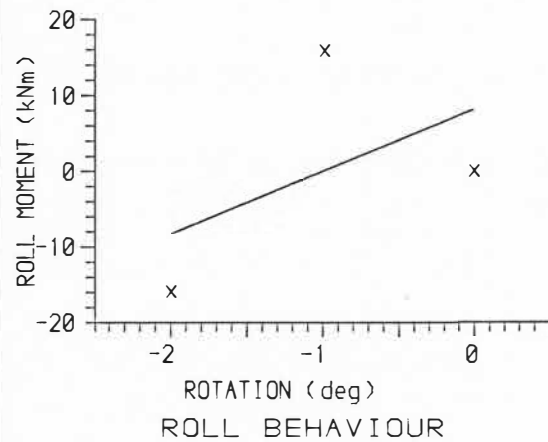
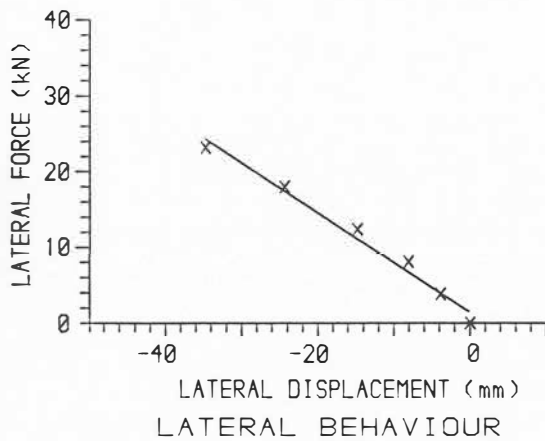
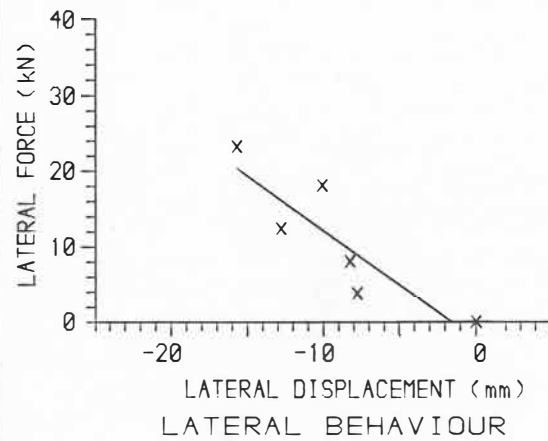
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

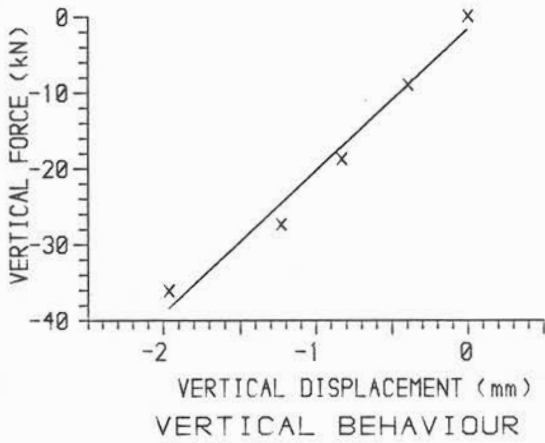
Fig. B.5

VEHICLE : VANHOO ASTROMEGA (2-axle analysis)

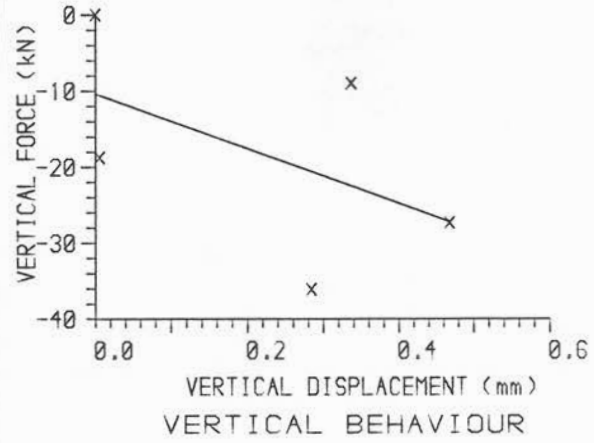
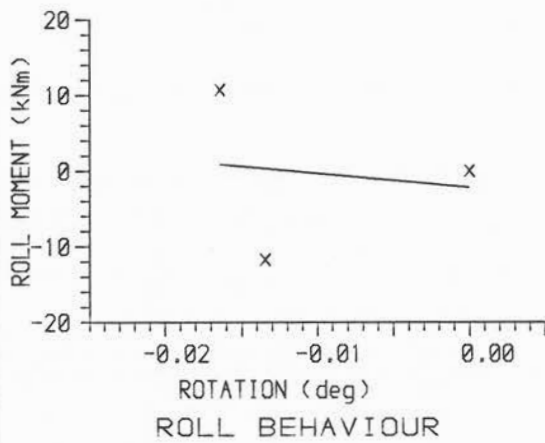
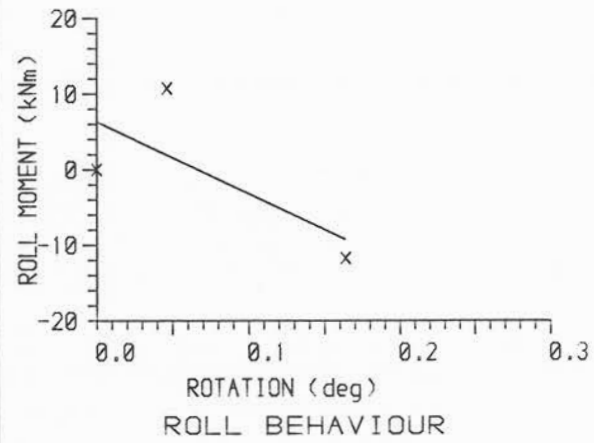
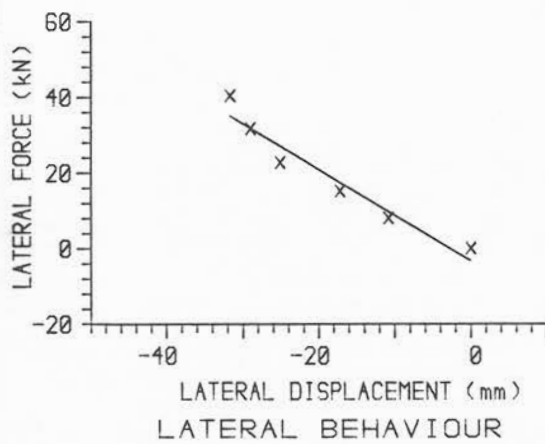
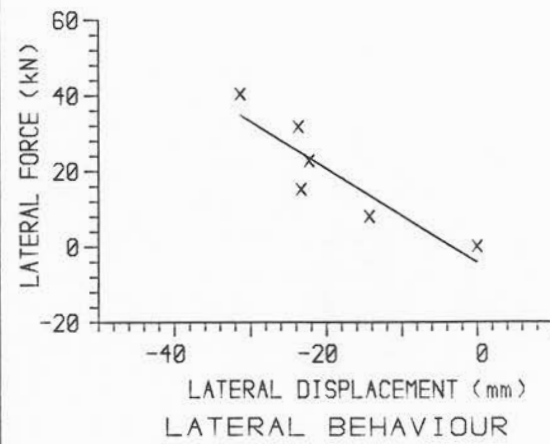
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

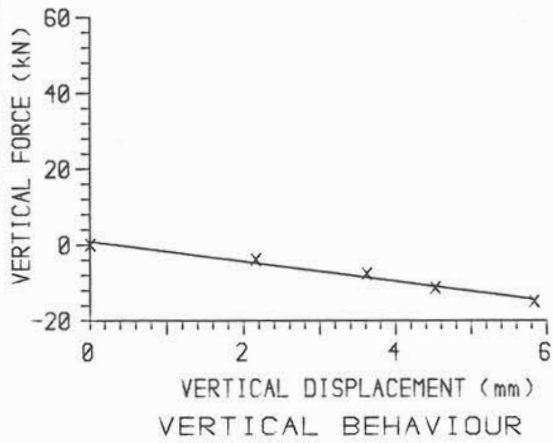
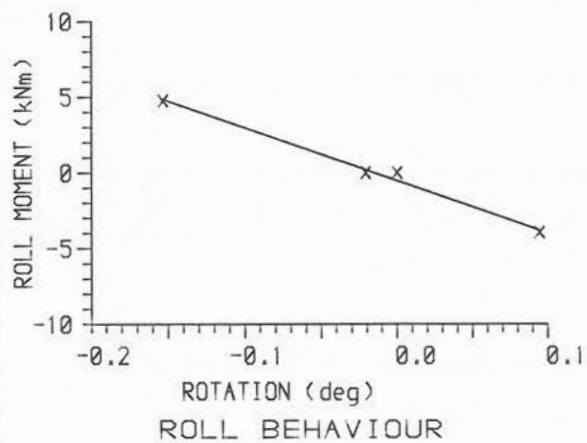
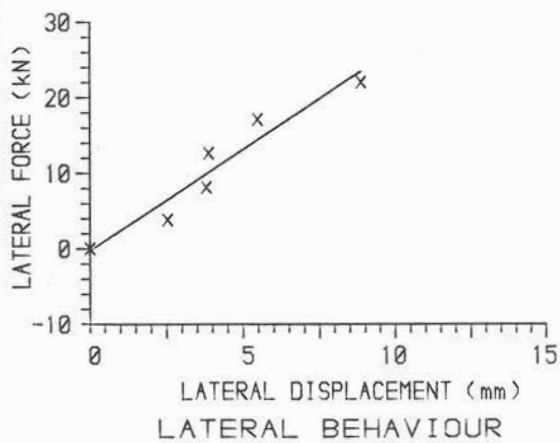
Fig. B.6

VEHICLE : JONCKHEERE JUBILEE P799

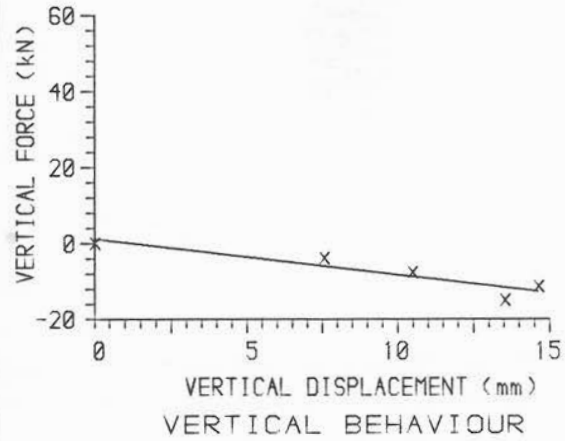
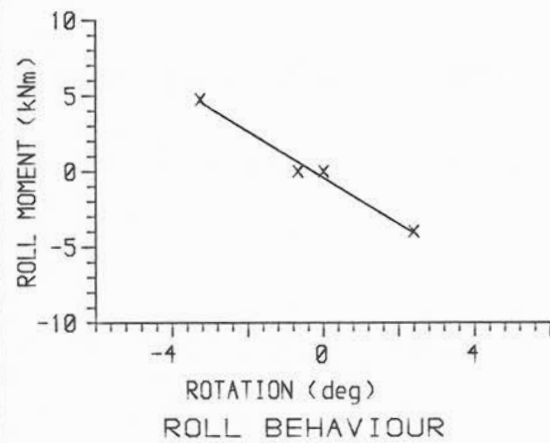
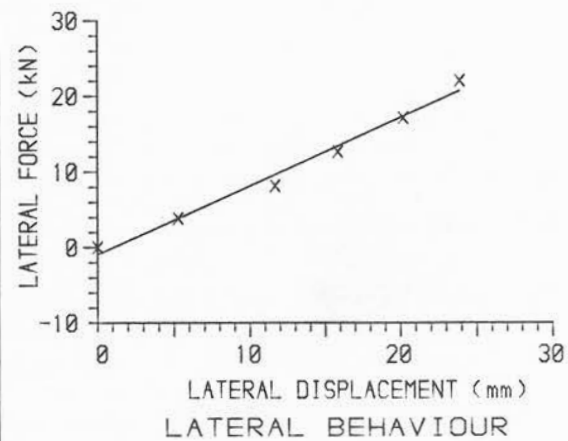
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

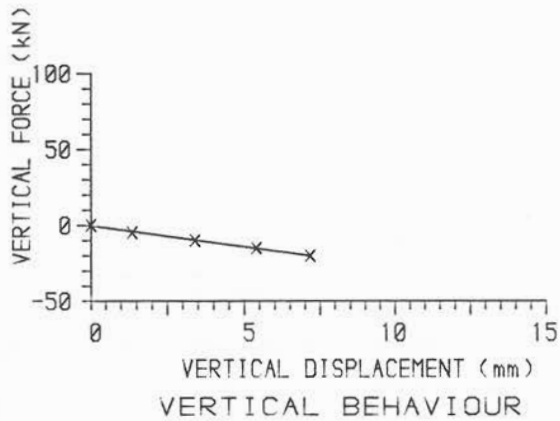
Fig. B.7

VEHICLE: JONCKHEERE JUBILEE P799

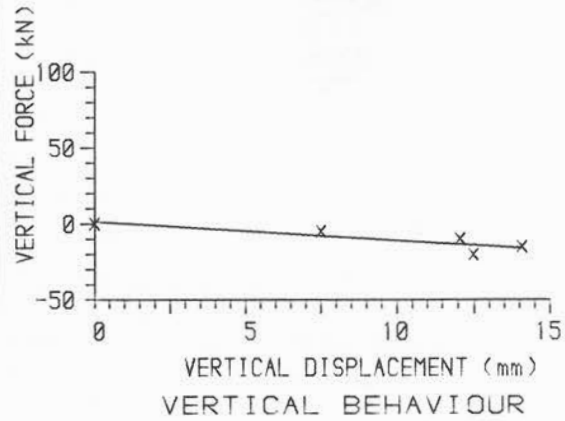
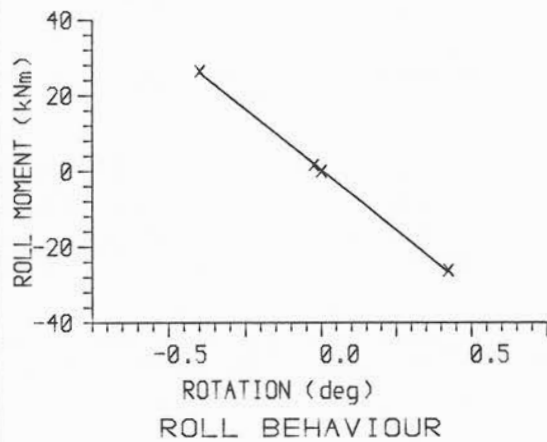
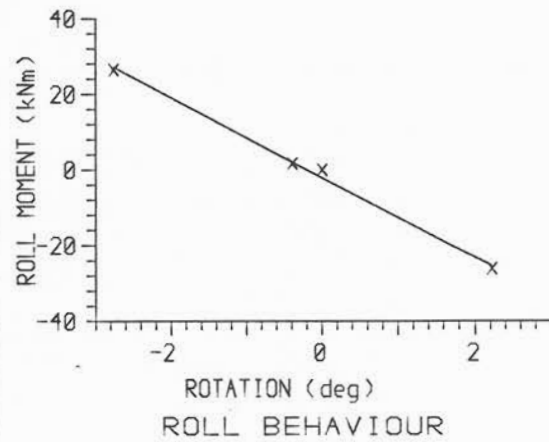
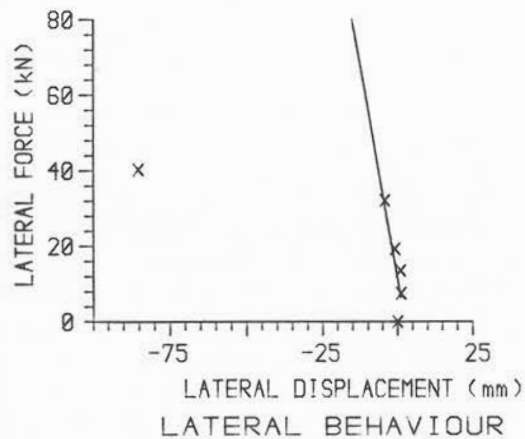
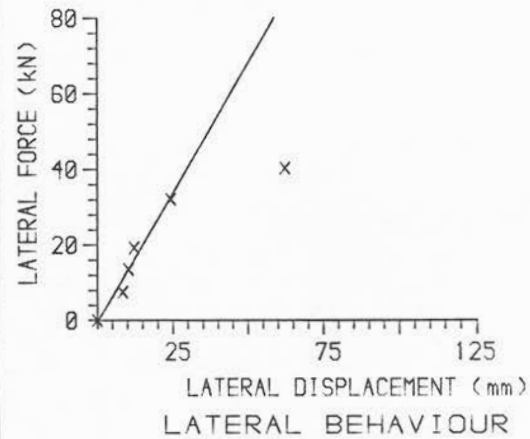
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES

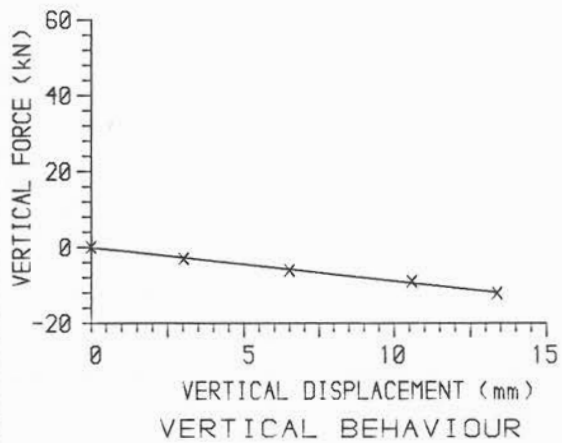
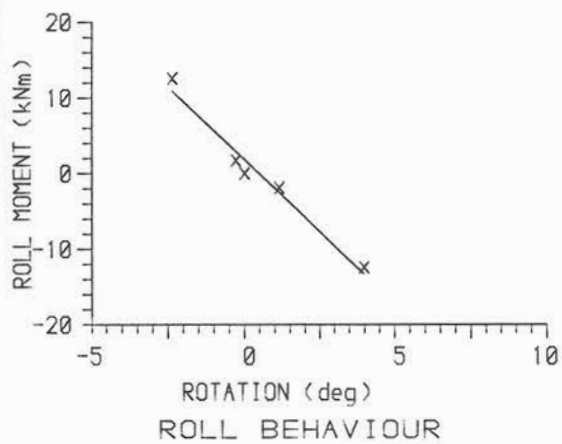
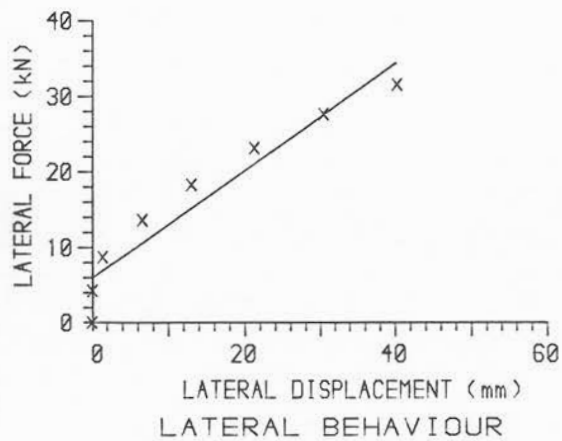


VEHICLE: LEYLAND LEOPARD/PLAXTON SUPREME IV

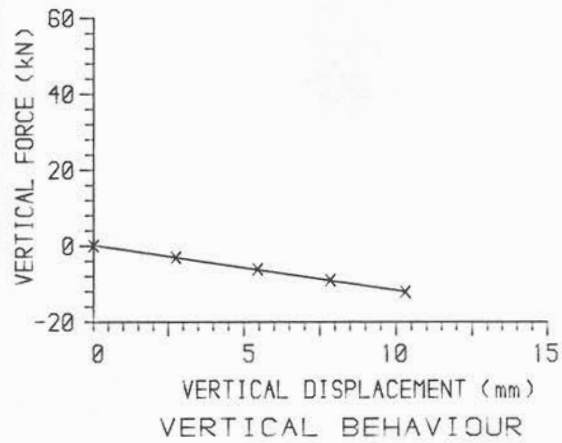
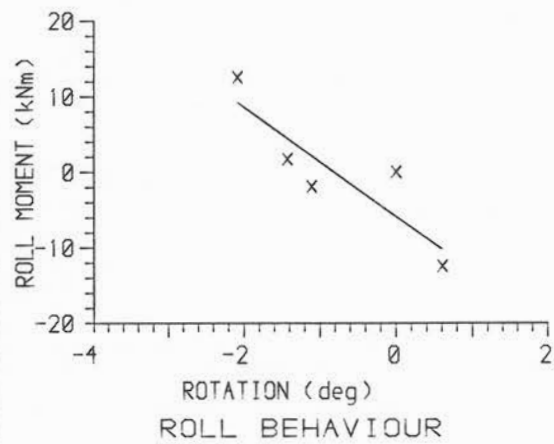
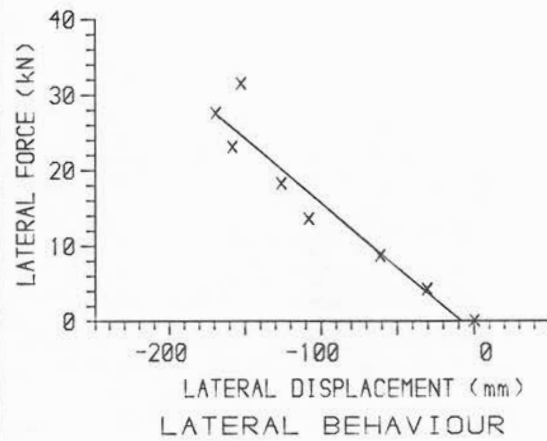
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

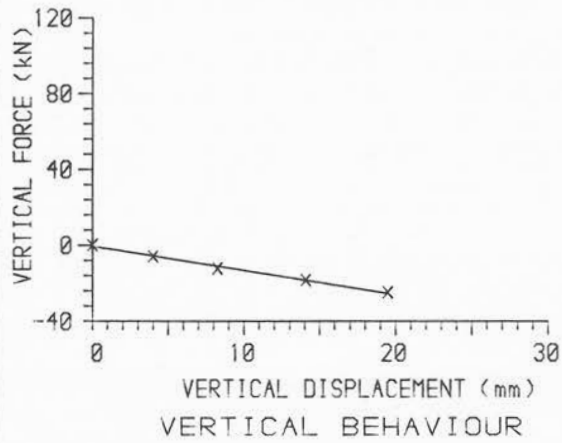
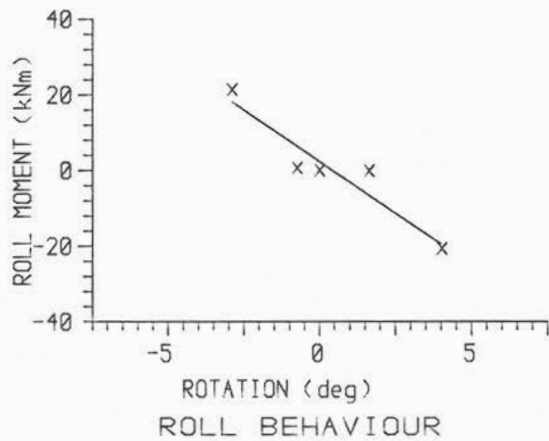
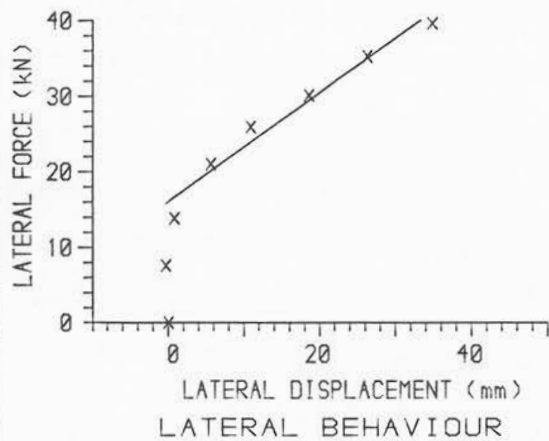
Fig. B.9

VEHICLE: LEYLAND LEOPARD/PLAXTON SUPREME IV

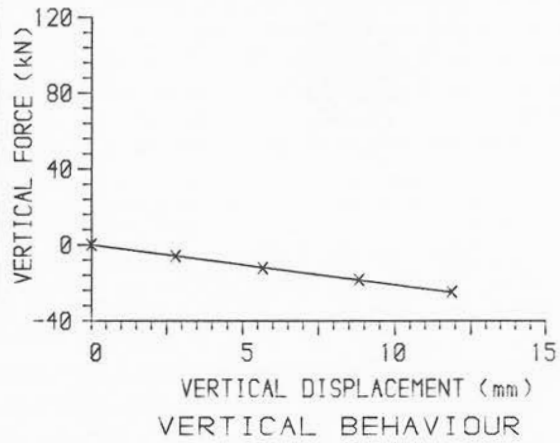
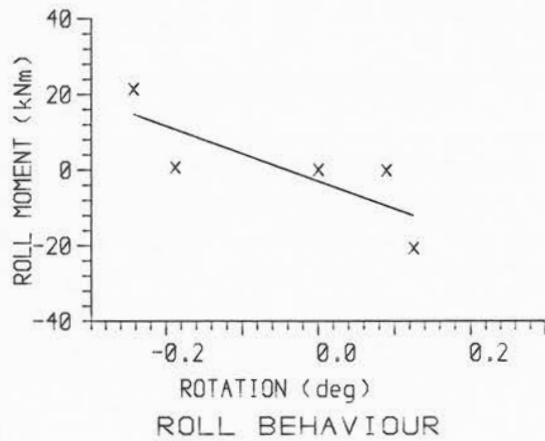
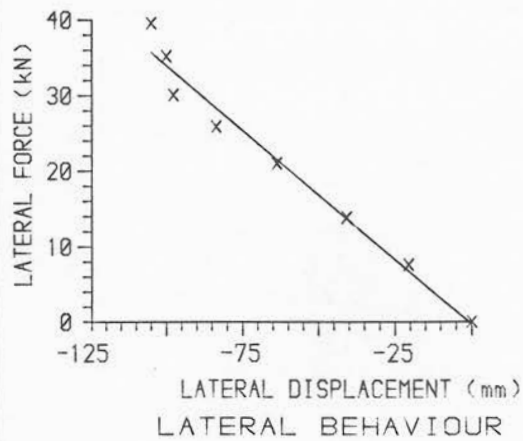
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

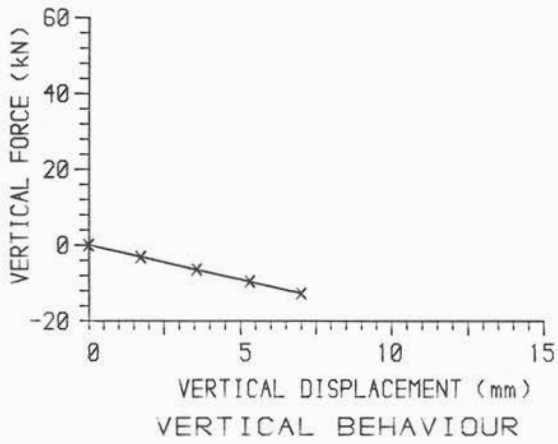
Fig. B.10

VEHICLE: DAF / PLAXTON PARAMOUNT 3200

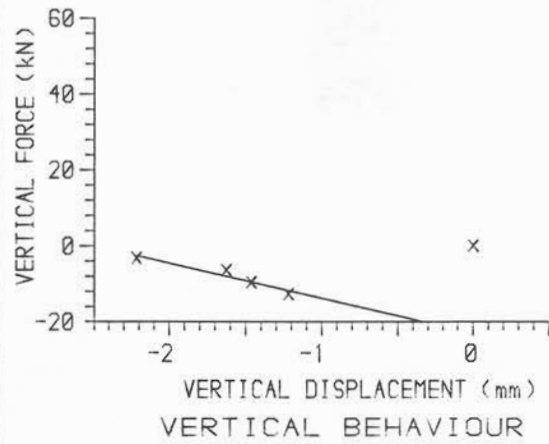
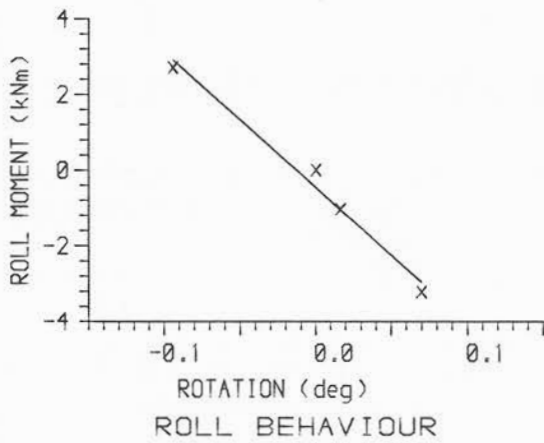
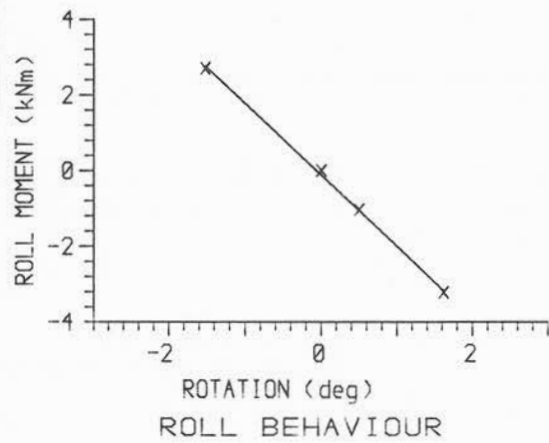
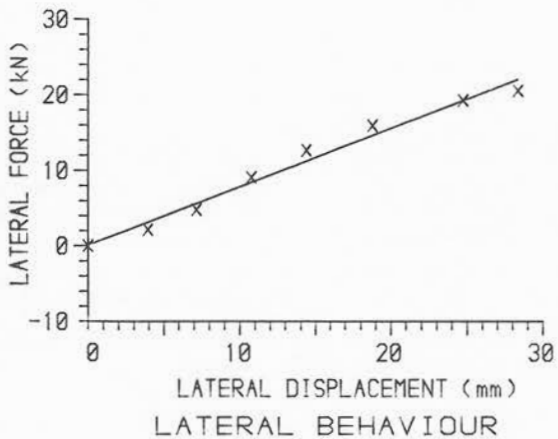
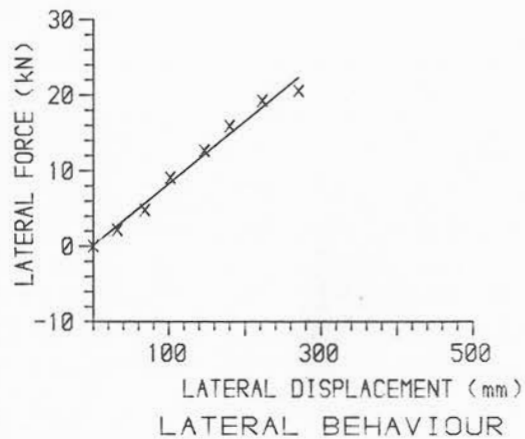
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES

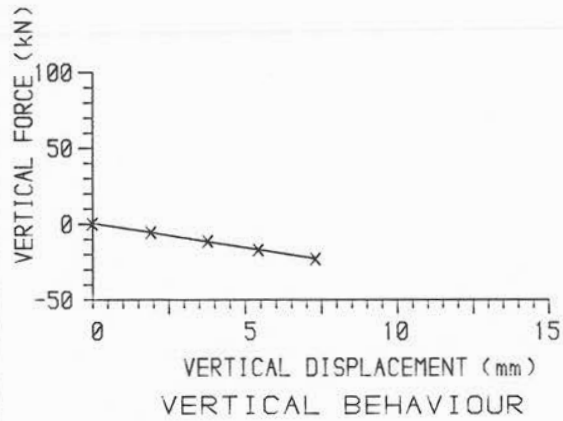
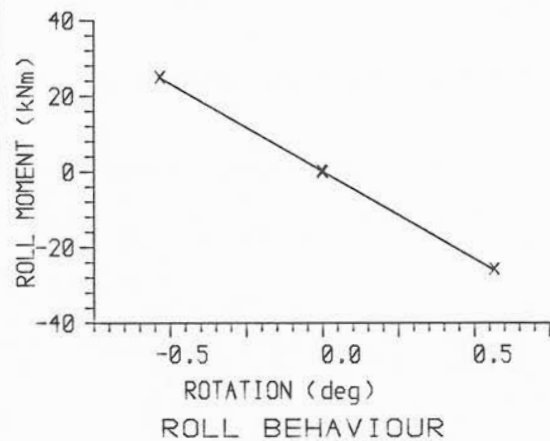
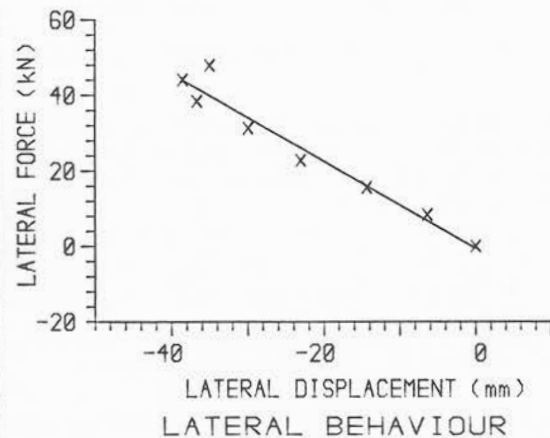


VEHICLE : DAF / PLAXTON PARAMOUNT 3200

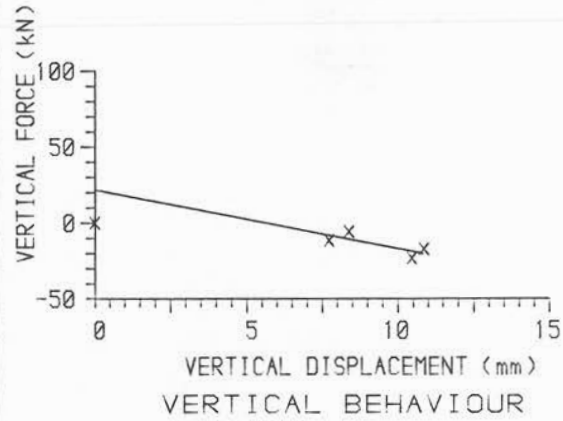
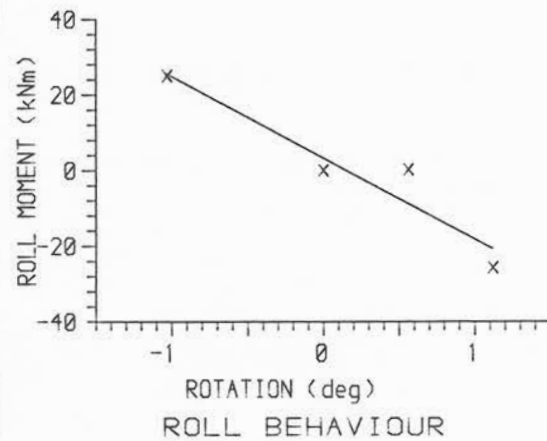
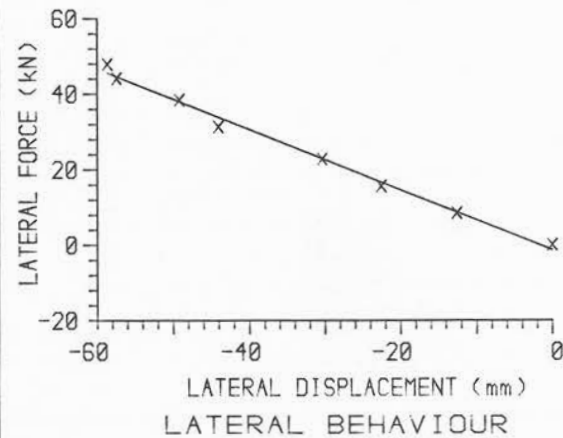
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

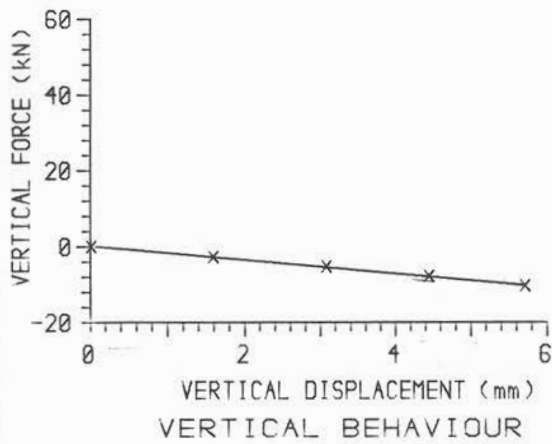
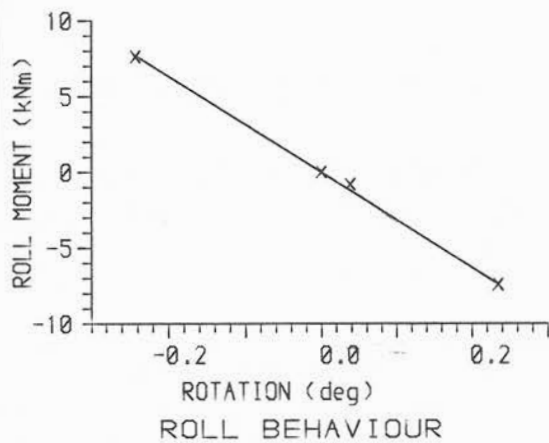
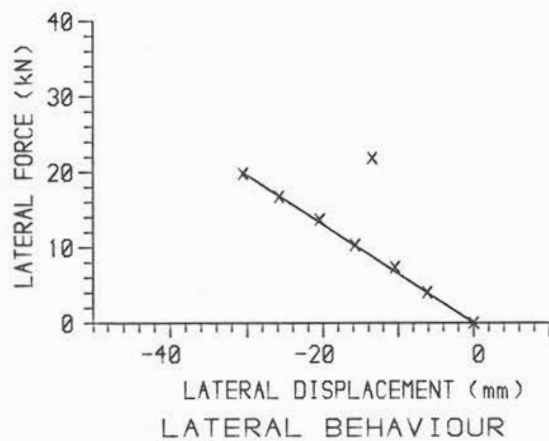
Fig. B.12

VEHICLE: BEDFORD YNT / DUPLÉ LASER

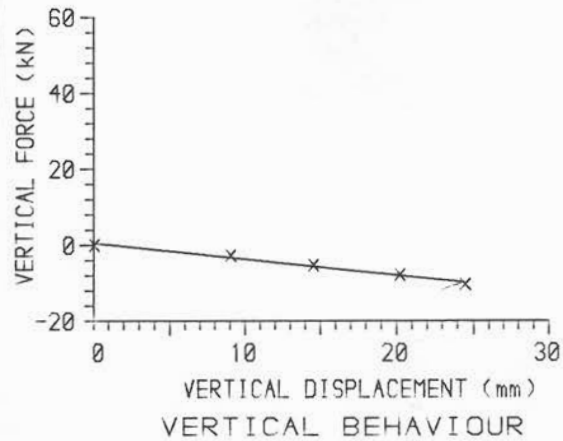
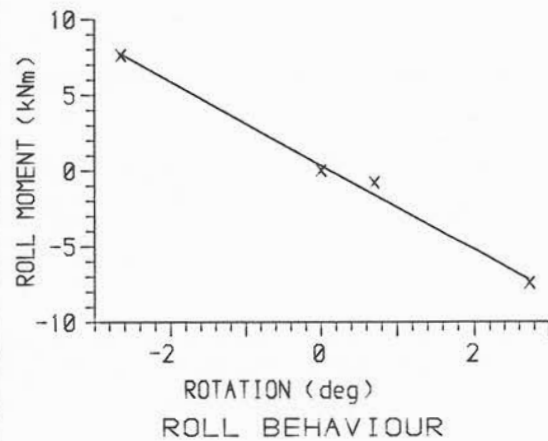
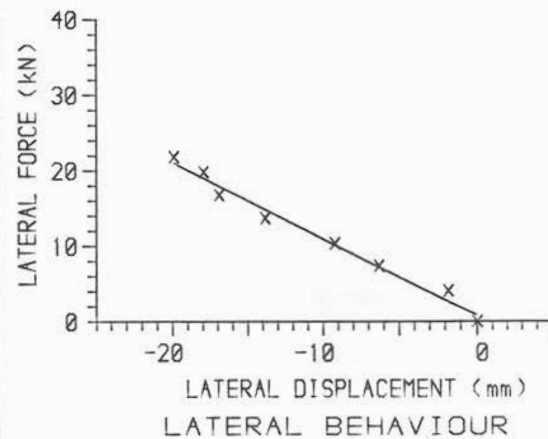
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

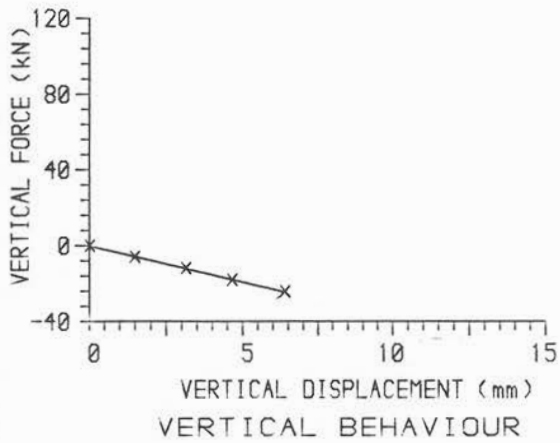
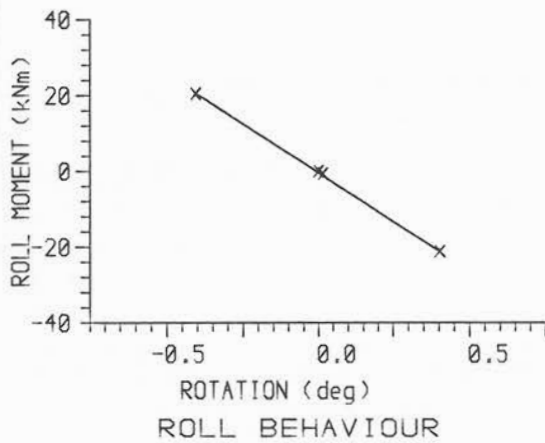
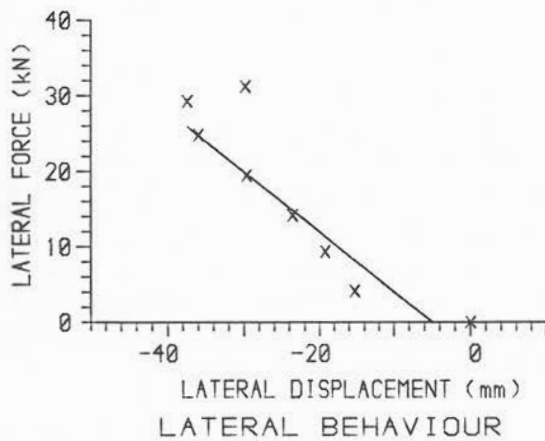
Fig. B.13

VEHICLE: BEDFORD YNT / DUPLÉ LASER

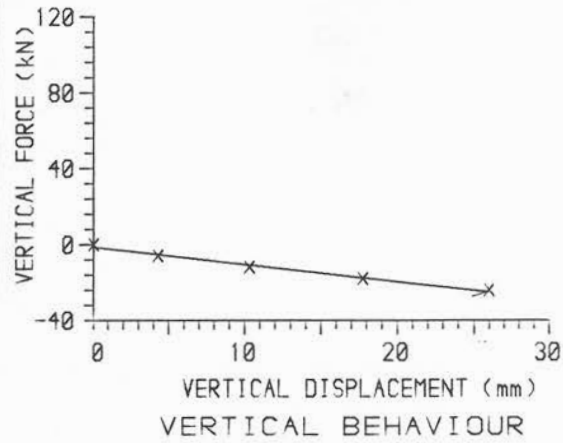
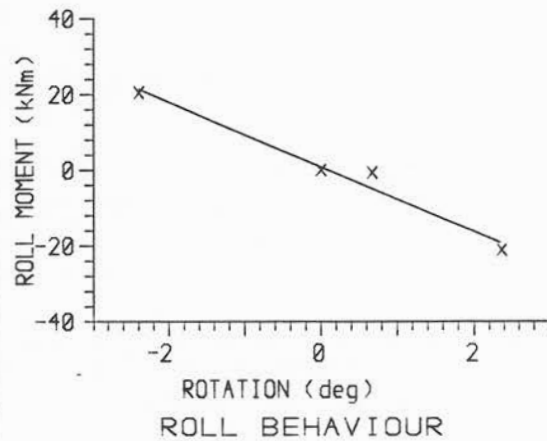
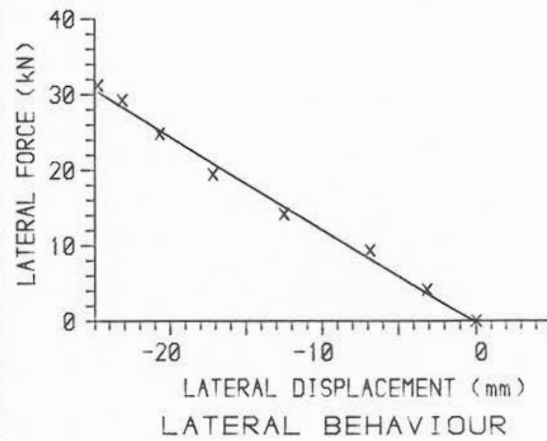
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



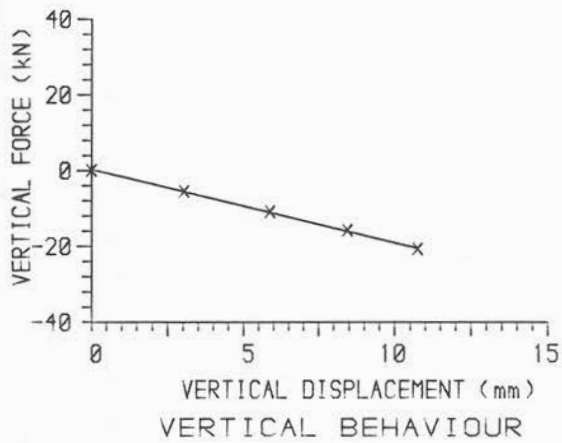
VEHICLE: MCW METROLINER

2 AXLE VERSION

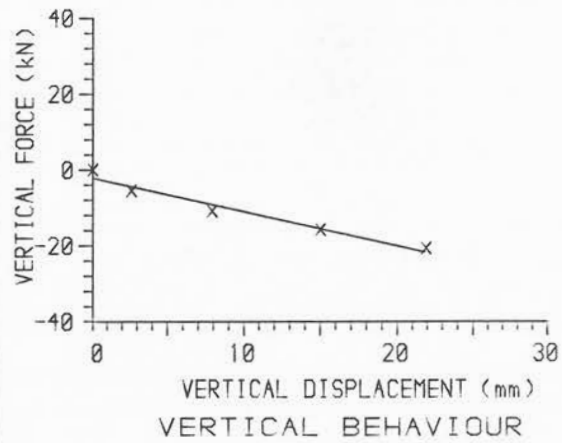
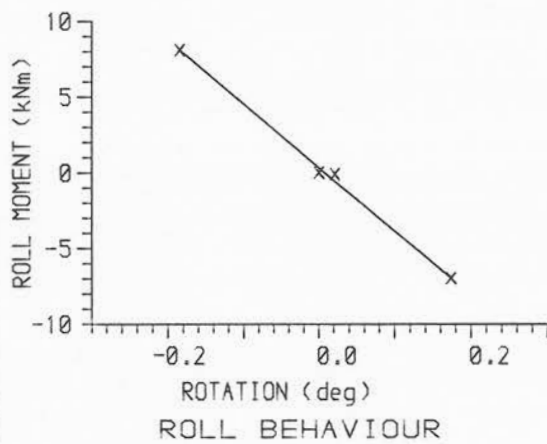
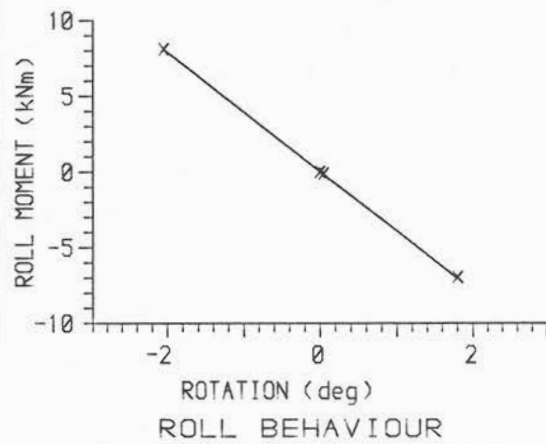
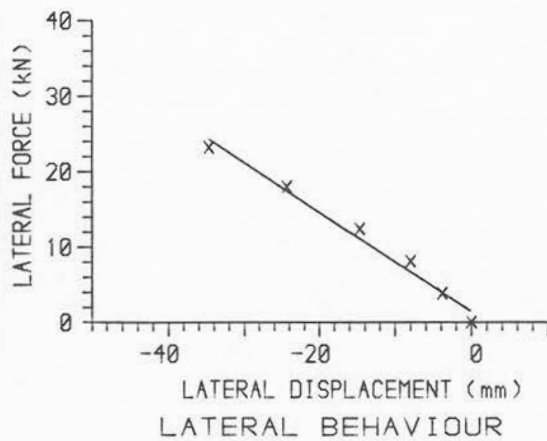
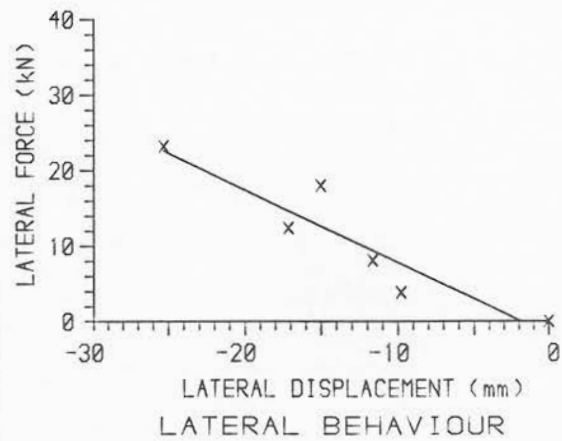
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

Fig. B.15

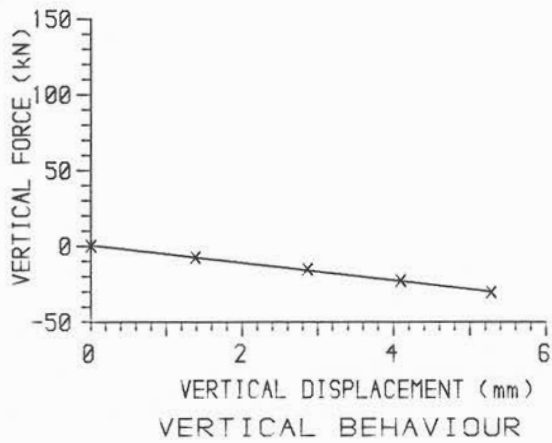
VEHICLE: MCW METROLINER

2 AXLE VERSION

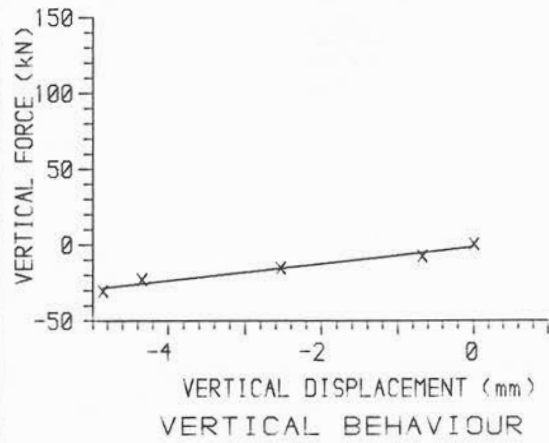
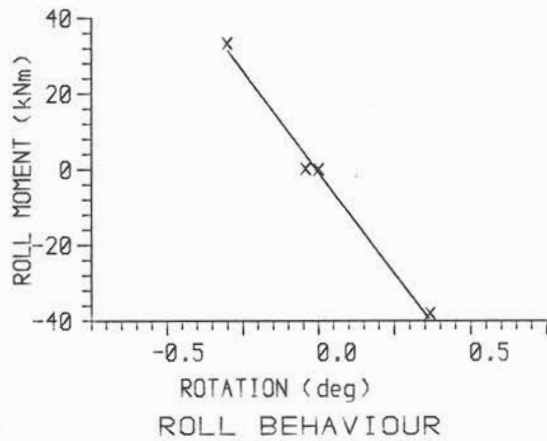
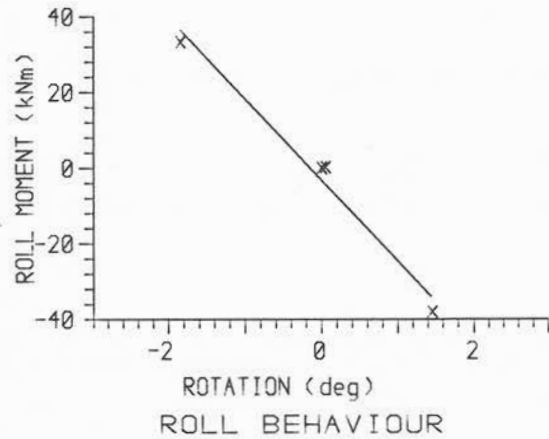
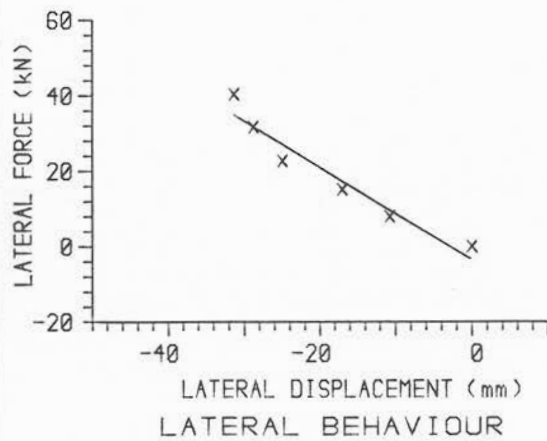
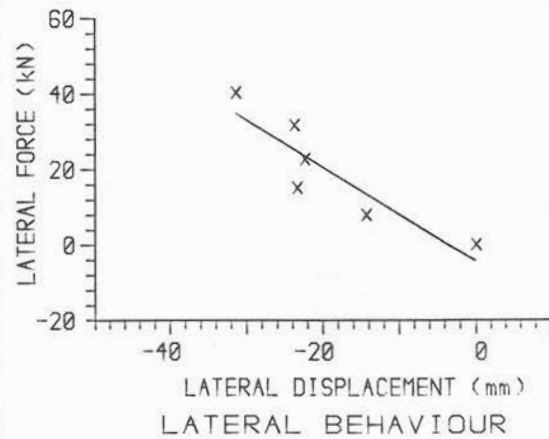
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

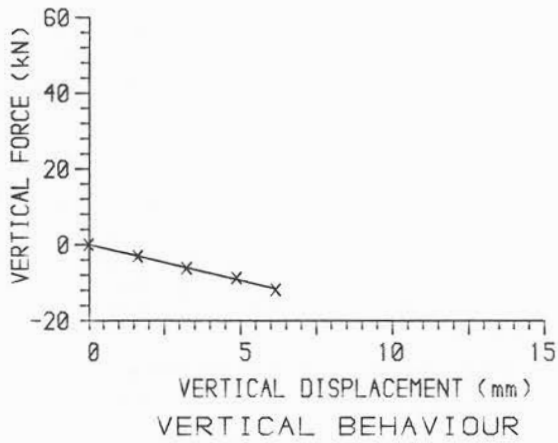
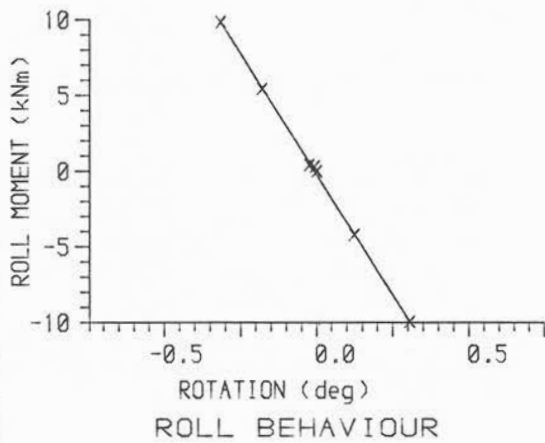
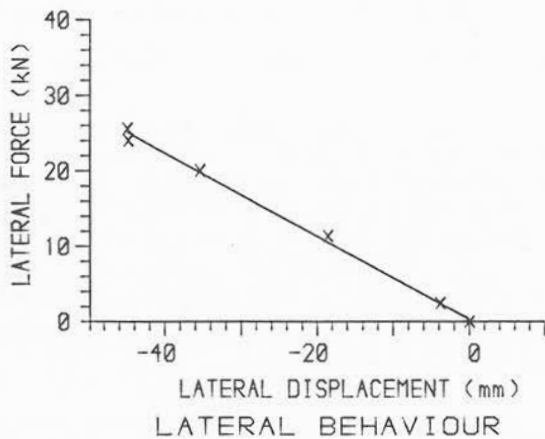
Fig. B.16

VEHICLE : LEYLAND ROYAL TIGER DOYEN

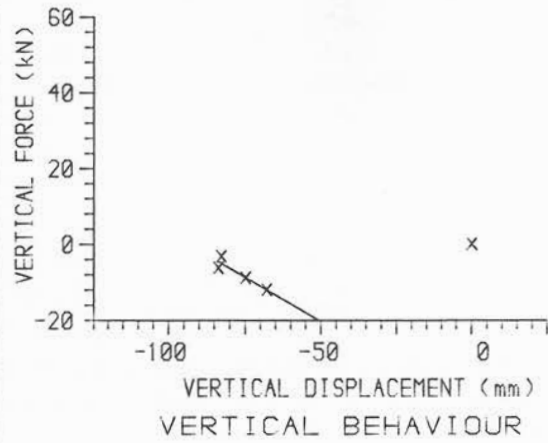
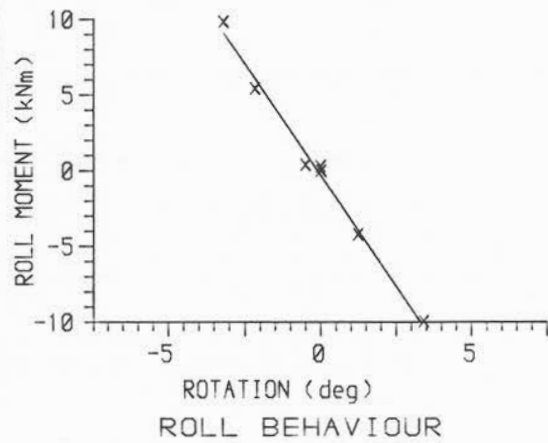
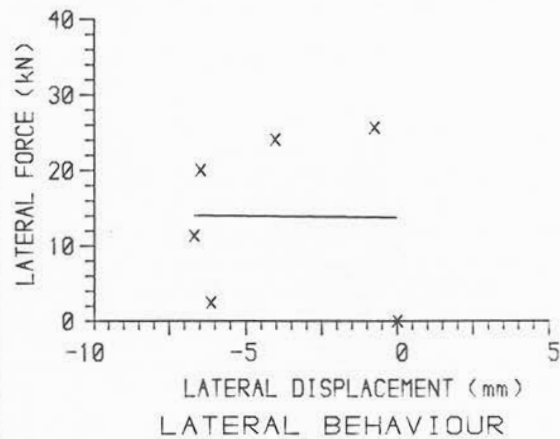
STIFFNESS SUMMARIES

AXLE 1

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



GRAPHS OF TYRE AND BODY STIFFNESS

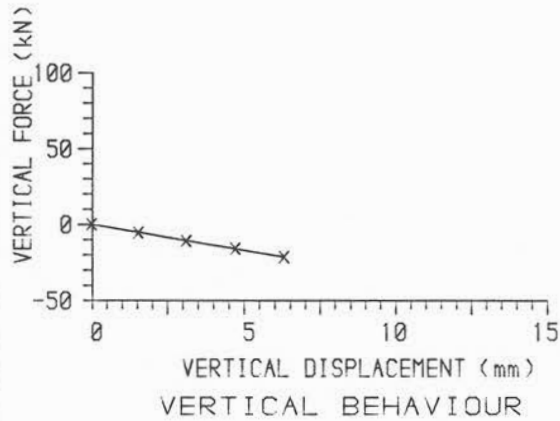
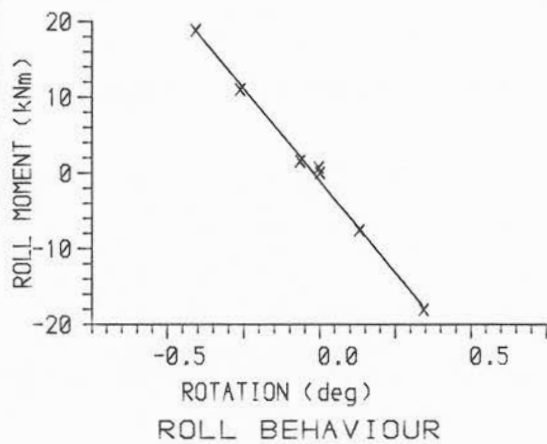
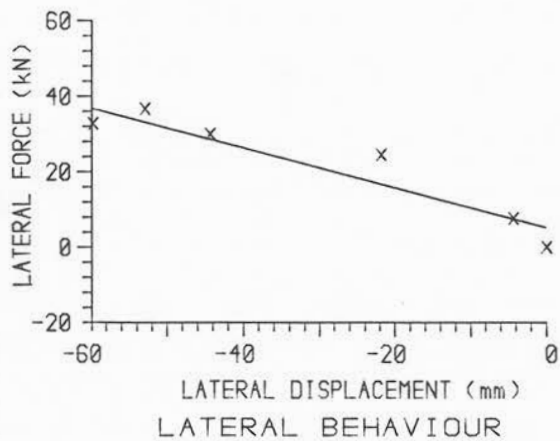
Fig. B.17

VEHICLE: LEYLAND ROYAL TIGER DOYEN

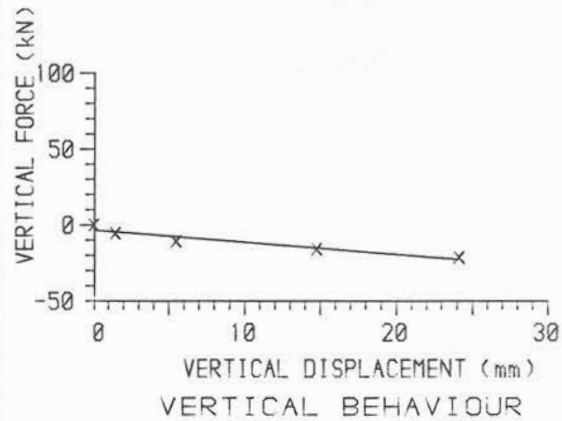
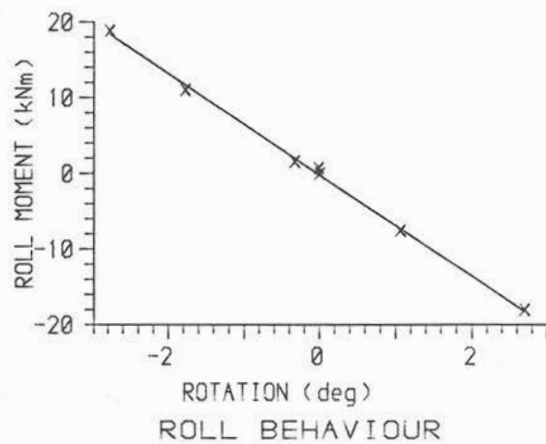
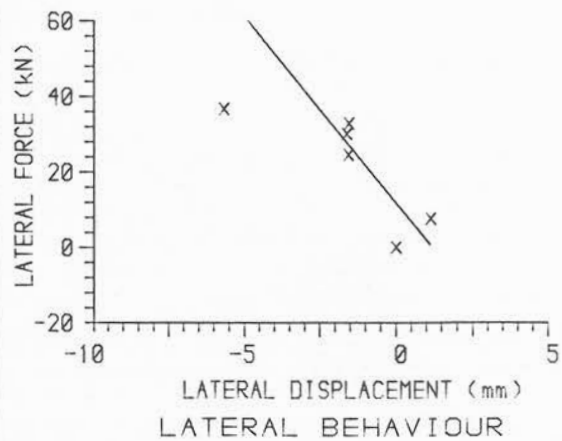
STIFFNESS SUMMARIES

AXLE 2

TYRE STIFFNESSES



SUSPENSION STIFFNESSES



APPENDIX C

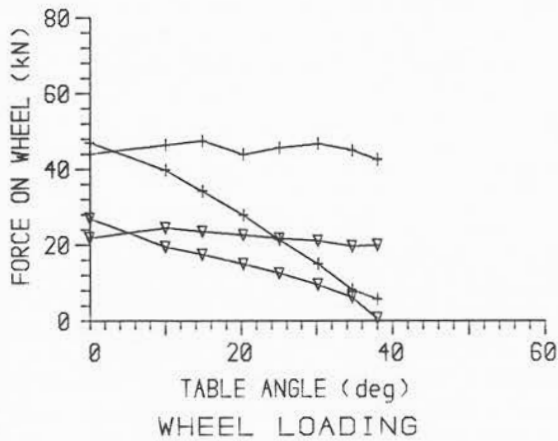
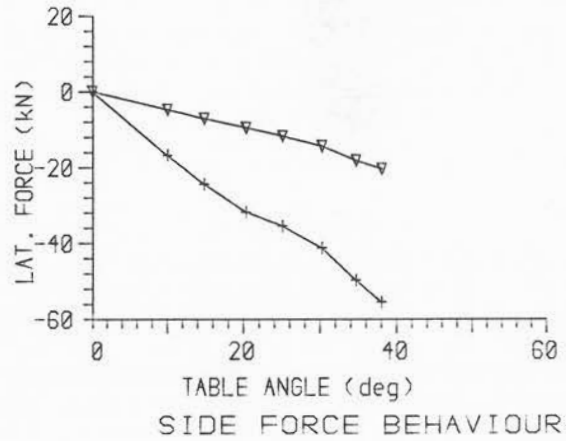
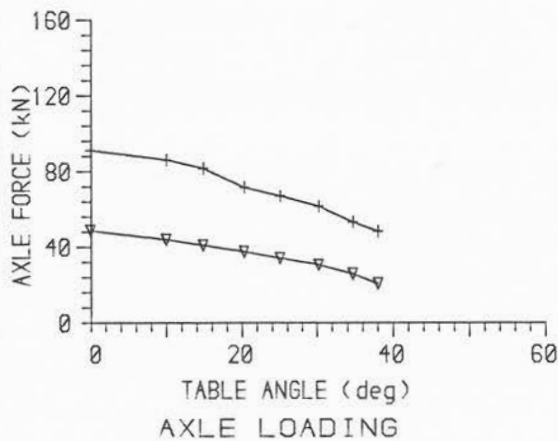
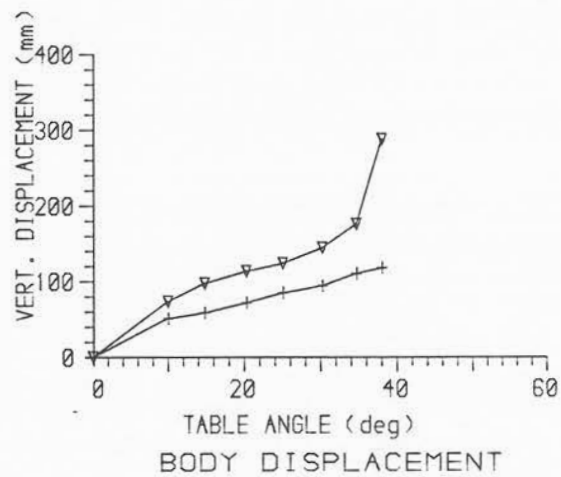
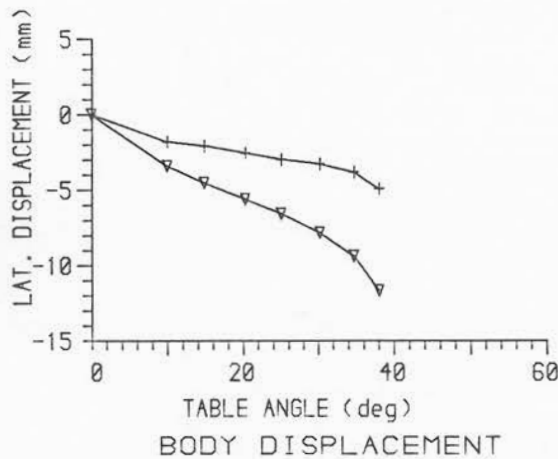
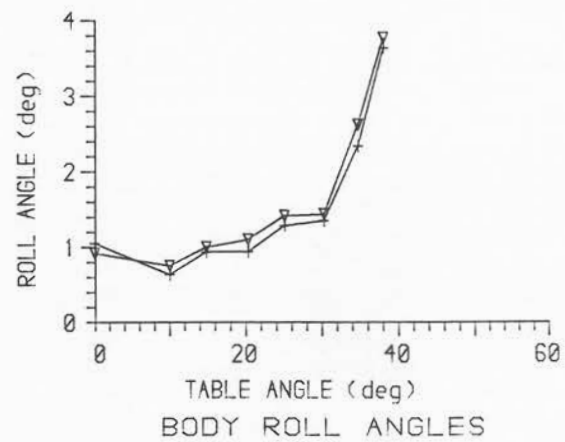
GRAPHICAL RESULTS FROM TILT TESTS

VEHICLE: LEYLAND NATIONAL 2

KEY:

▽▽AXLE 1

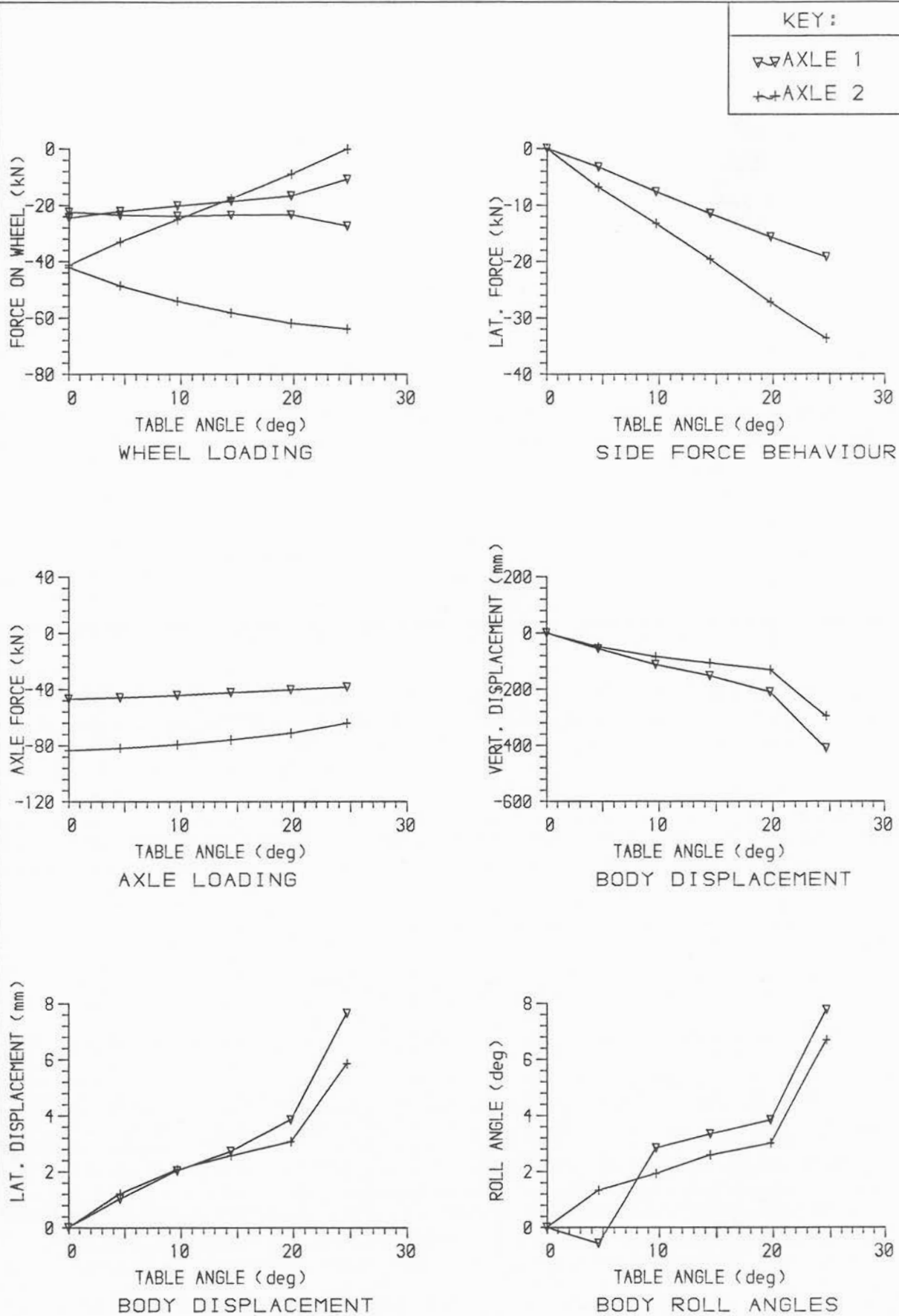
++AXLE 2



GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.1

VEHICLE: MCW METROBUS



GRAPHICAL RESULTS FROM A TILT TEST

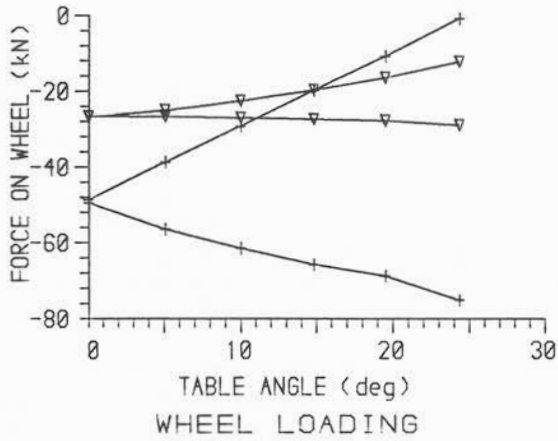
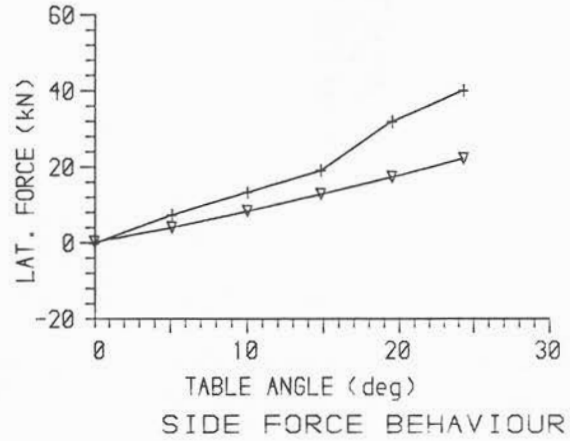
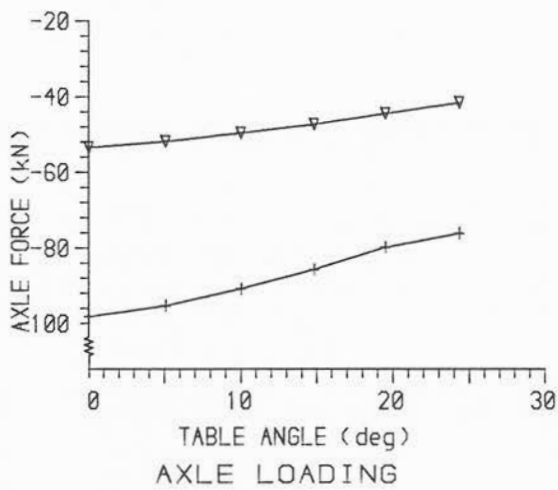
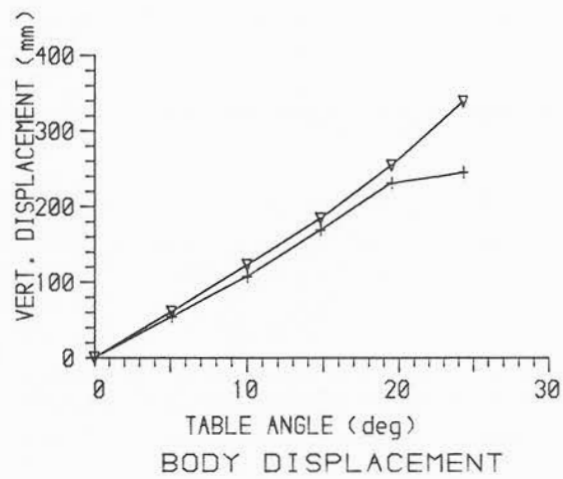
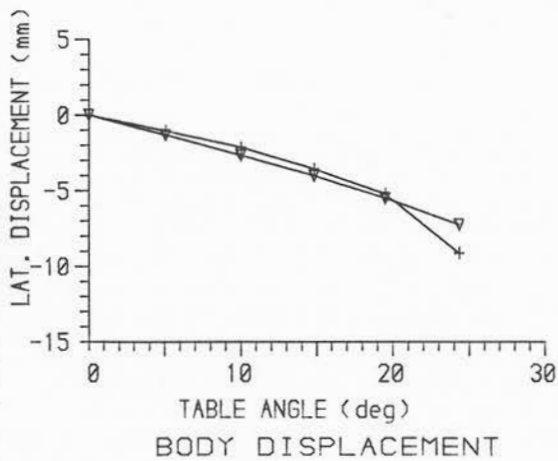
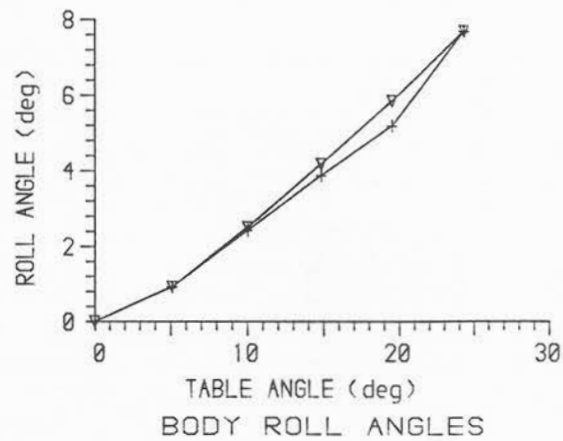
Fig. C.2

VEHICLE : JONCKHEERE JUBILEE P799

KEY :

▽ AXLE 1

+ AXLE 2

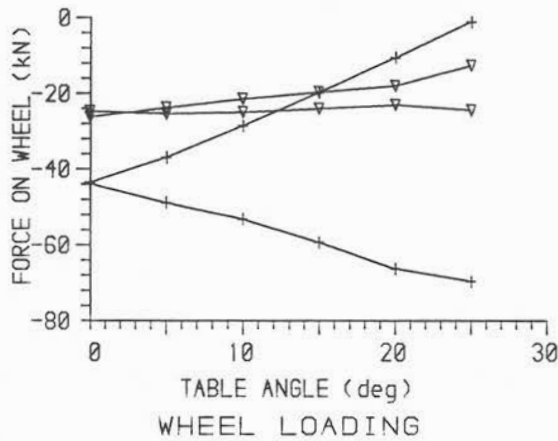
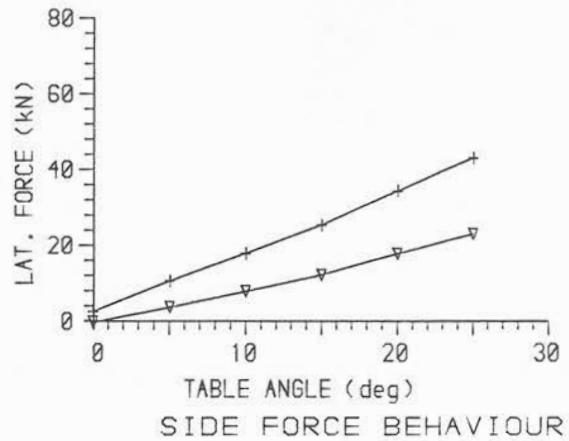
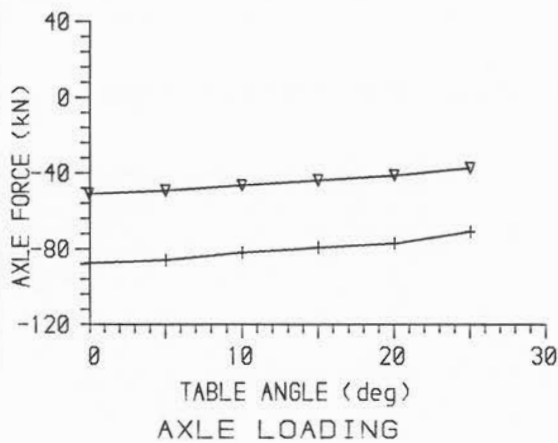
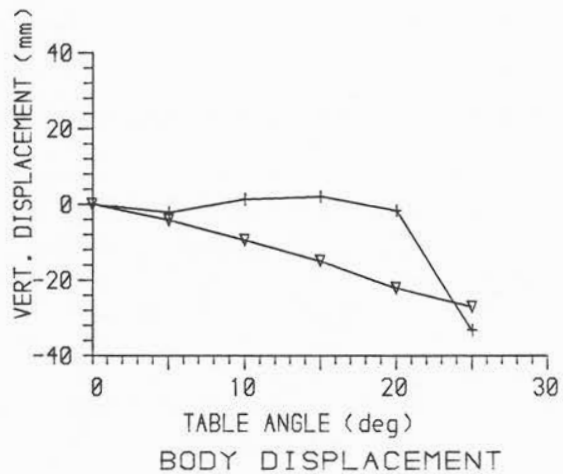
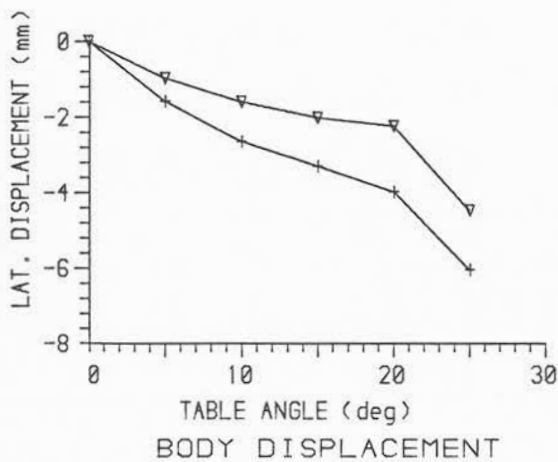
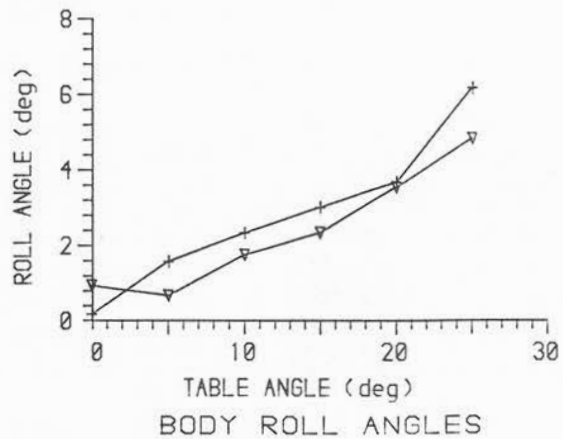


GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.3

VEHICLE: VANHOOL ASTROMEGA (2-axle analysis)

KEY:
 ▽ AXLE 1
 + AXLE 2



GRAPHICAL RESULTS FROM A TILT TEST

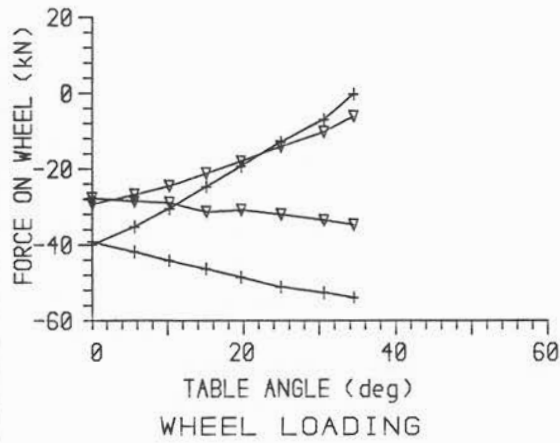
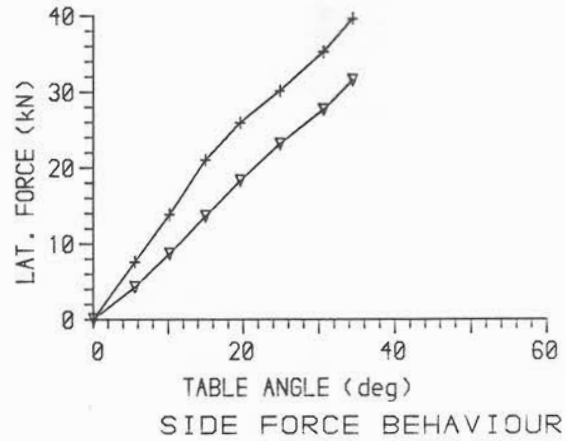
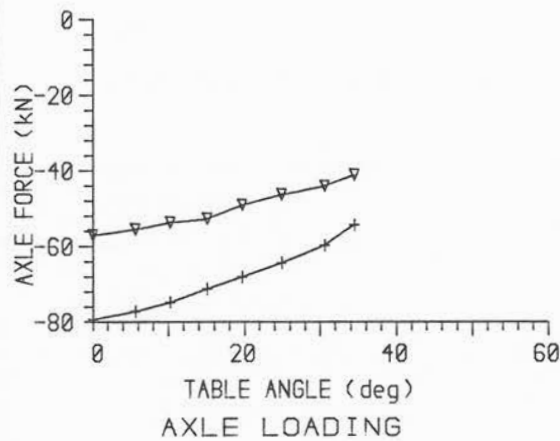
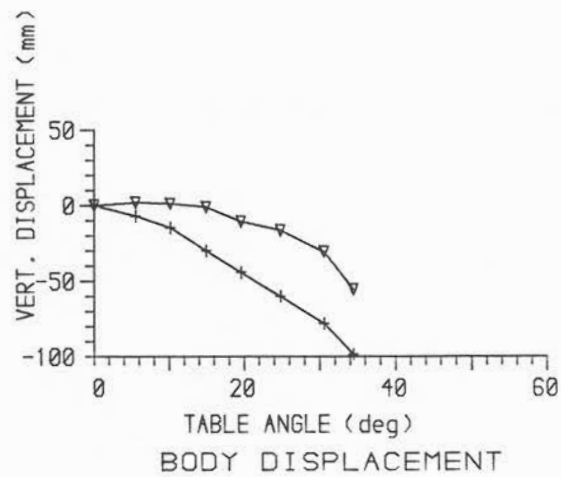
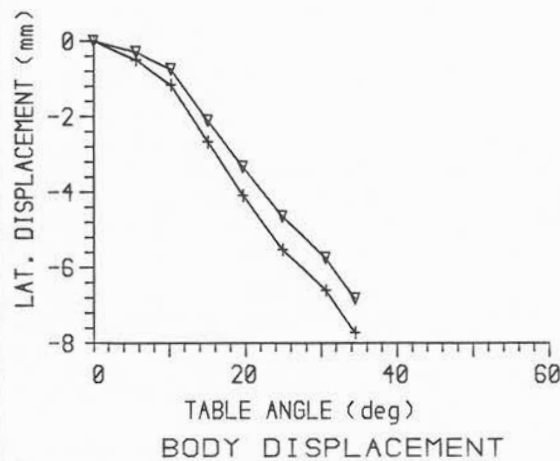
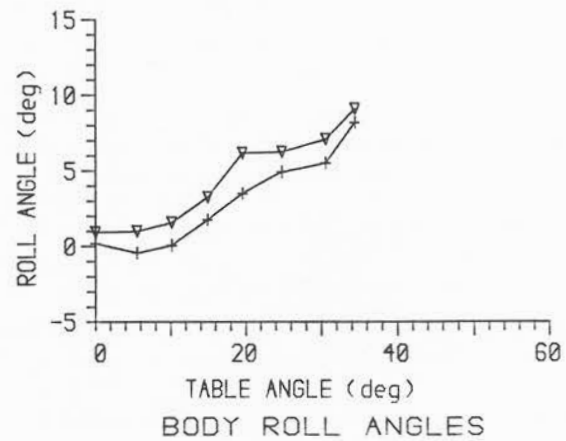
Fig. C.4

VEHICLE : LEYLAND LEOPARD/PLAXTON SUPREME IV

KEY :

▽ AXLE 1

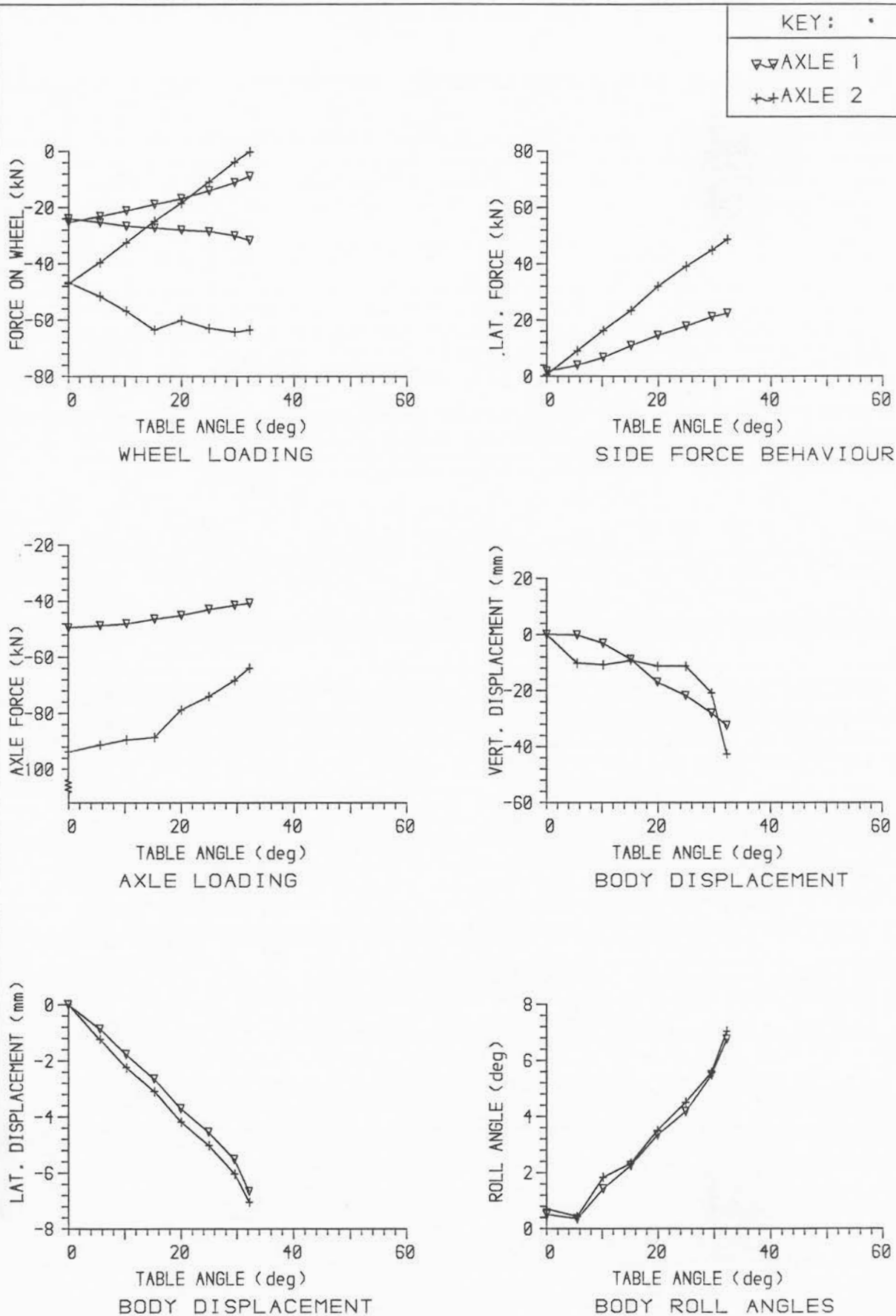
+ AXLE 2



GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.5

VEHICLE: DAF / PLAXTON PARAMOUNT 3200



GRAPHICAL RESULTS FROM A TILT TEST

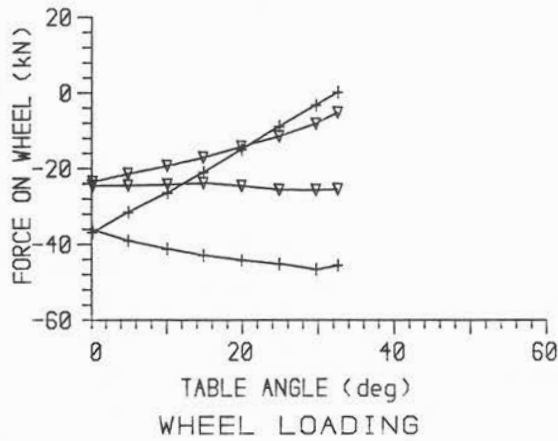
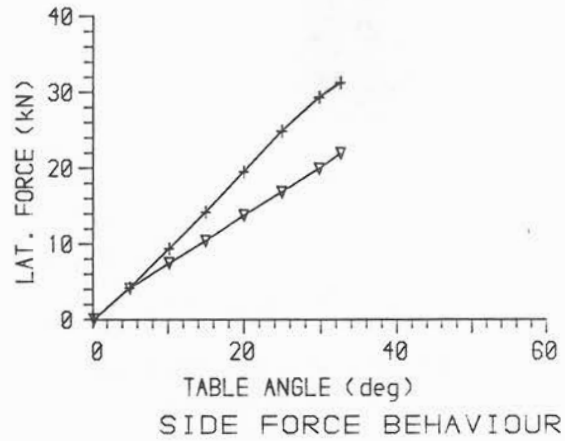
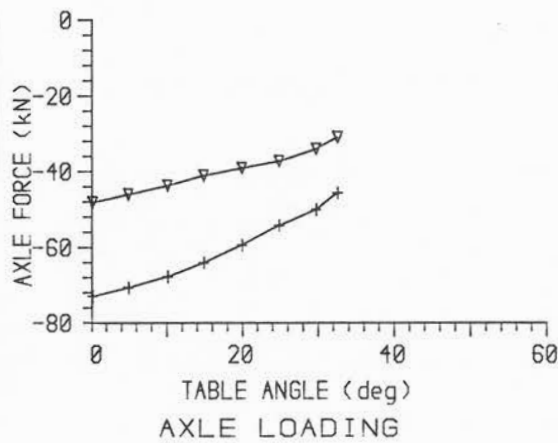
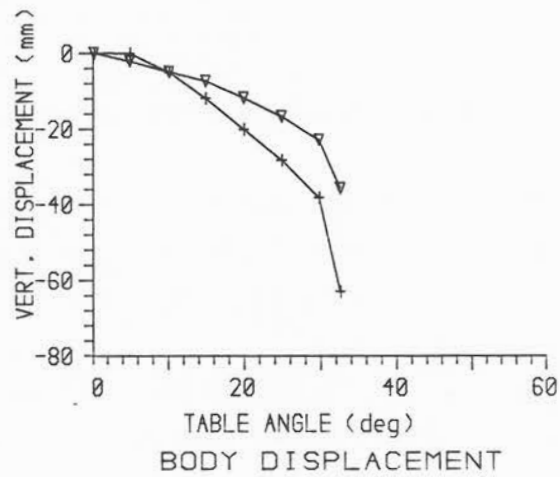
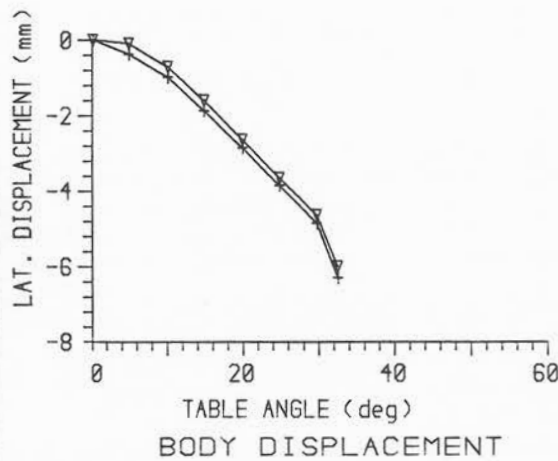
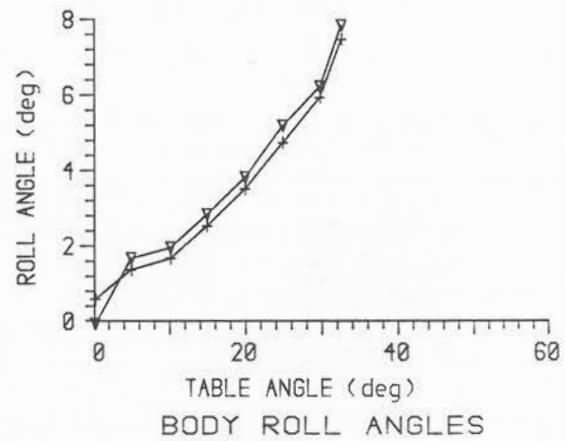
Fig. C.6

VEHICLE: BEDFORD YNT / DUPLÉ LASER

KEY:

▽ AXLE 1

+ AXLE 2



GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.7

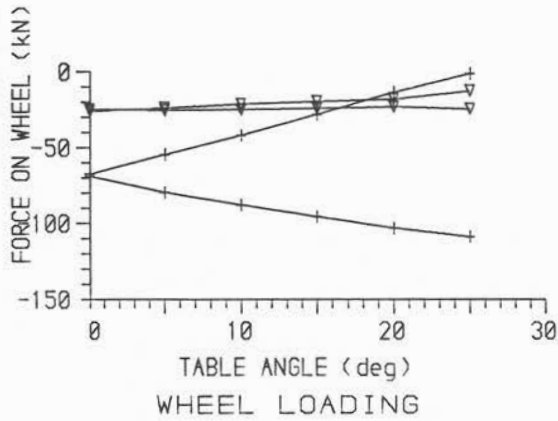
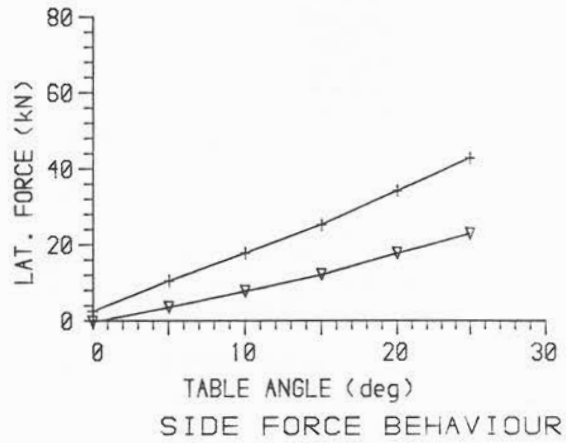
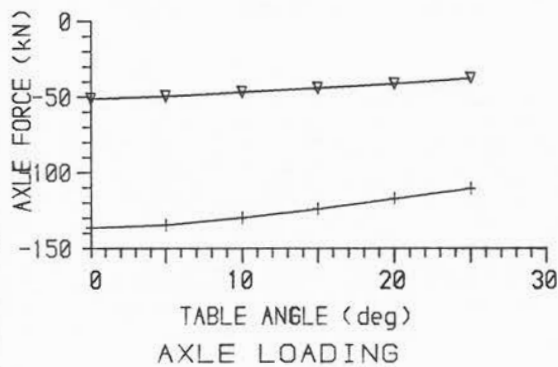
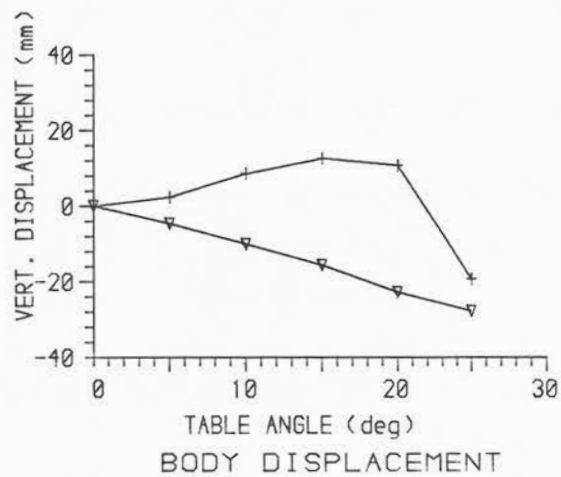
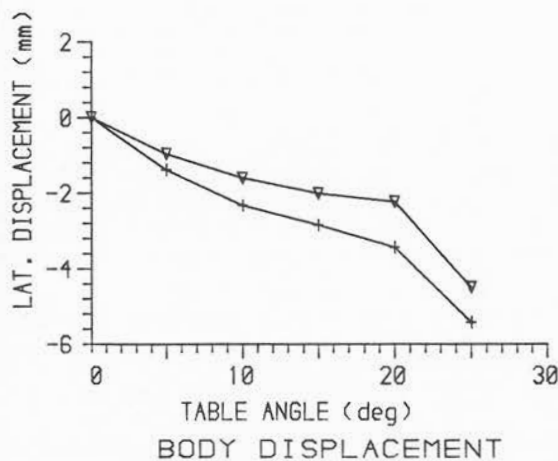
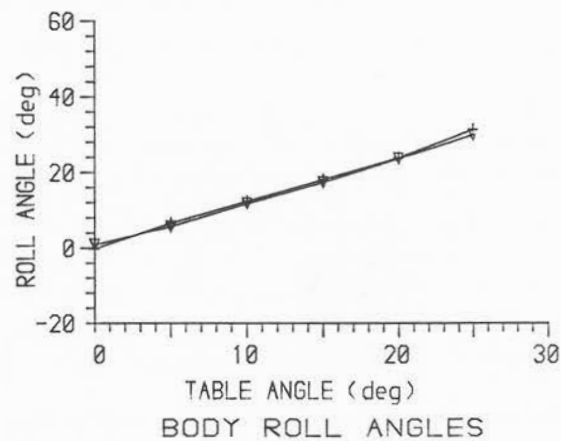
VEHICLE : MCW METROLINER

2 AXLE VERSION

KEY :

▽ AXLE 1

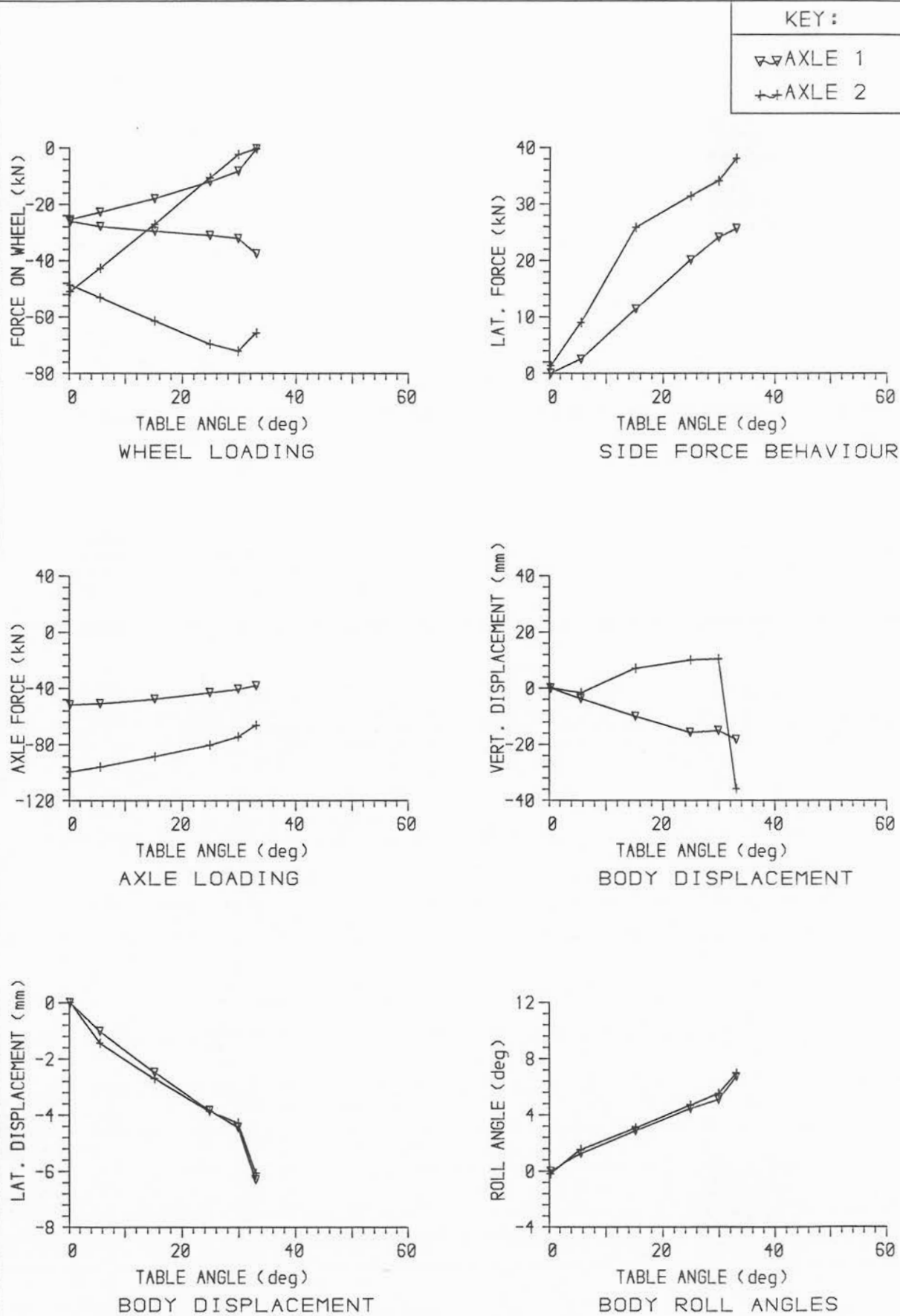
+ AXLE 2



GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.8

VEHICLE: LEYLAND ROYAL TIGER DOYEN



GRAPHICAL RESULTS FROM A TILT TEST

Fig. C.9

APPENDIX D DETAILS OF INSTRUMENTATION

D.1 FORCE MEASUREMENT

Individual wheel loads (normal to the local ground plane) were measured using strain-gauged Portable Vehicle Weighpads connected to a 6-channel digital display and printer.

Loadcells of 10 tonne capacity were positioned between tyre sidewalls and a reaction plate on the tilt platform to monitor the side force (refer to Figure 3.16). Up to 3 loadcells were connected to a suitable strain-gauge amplifier with its own power supply. Display was on a digital voltmeter via a multi-channel switching box. Calibration curves are given in Figure D.1.

D.2 DISPLACEMENT MEASUREMENT

Hybrid linear potentiometers in stroke lengths from 100 to 250 mm were utilised to determine vertical, lateral and roll displacements of the body and axles. Further details of the method of resolving the measured displacements into these 3 components are given in Appendix H. Power supply (stabilised 10V D.C.) and digital display were performed via the switching box, which enabled the displacements and lateral forces to be displayed rapidly in turn.

Roll angle of the tilt platform and front and rear of the bus body were measured with a clinometer.

D.3 OPERATIONAL DIFFICULTIES AND SOLUTIONS

The displacement transducers were positioned as shown typically in Figure 3.16 (after discovering that they were far easier to measure relative to the vehicle, than when attached directly between the body, axle and deck - that is, underneath the bus!). The first attachment method ($\phi 25 \times 16$ mm button magnets) was found to be inadequate to support the potentiometers' weight. Magnetic bases (50mm cube) were thus substituted and proved to be sufficiently strong.

Note that the calibration tests were performed using a Denison 150 ton compression test machine (No. 21051).

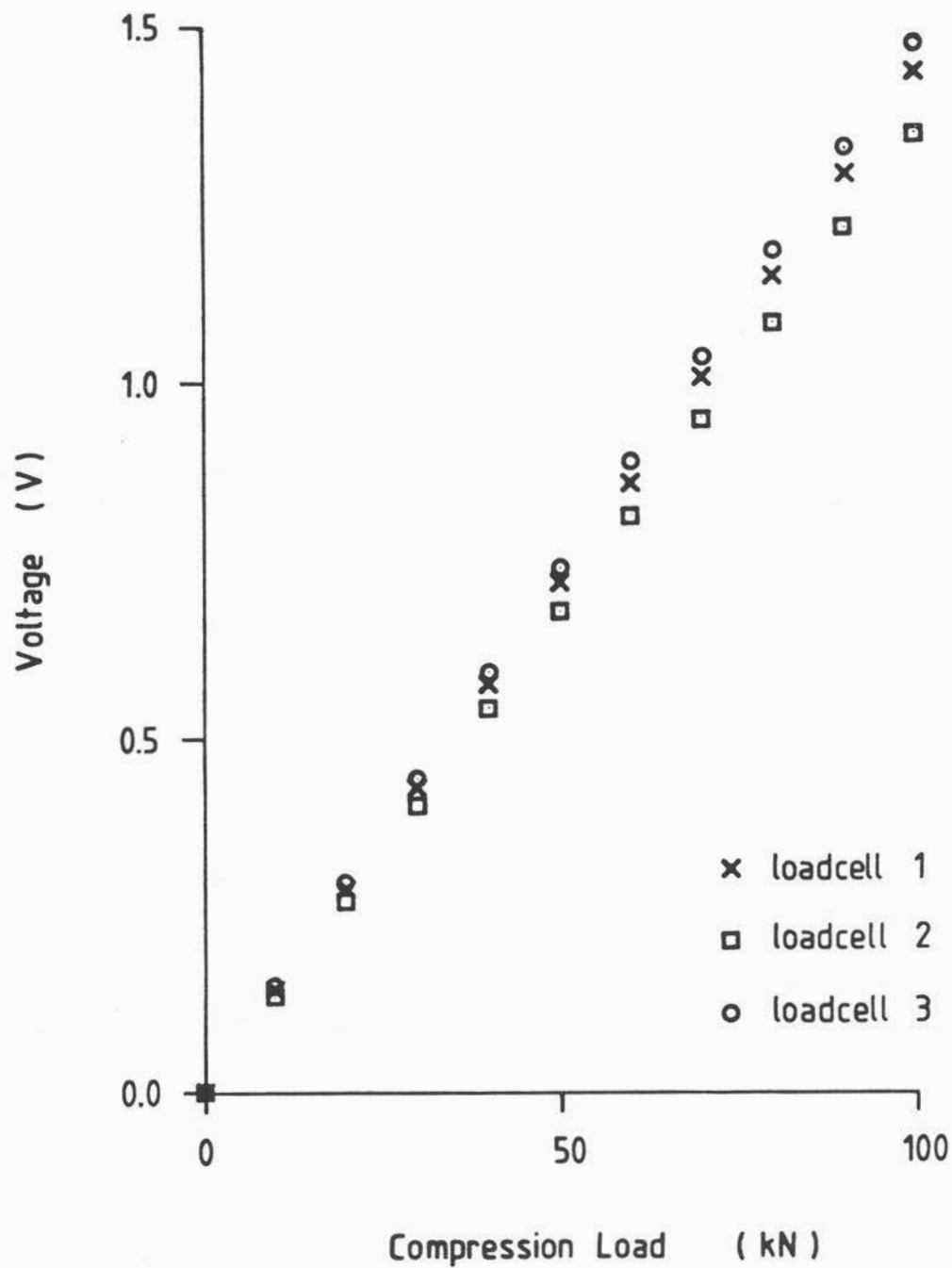


Figure D.1 Calibration Curves for Loadcells

The first (shakedown) tilt test discovered several other areas for improvement. The original ramp design was unsuitable for three reasons:

- too steep,
- difficult to climb both ramps simultaneously, while
- failing to stop in position on the weighpads.

Figure D.2 indicates the original and the subsequent ramp designs. The unequal lengths allow one axle to be driven up before the other, reducing the engine torque requirements. The horizontal sections allow speed control and limited manoeuvring, before driving into position on the weighpads and linear bearings.

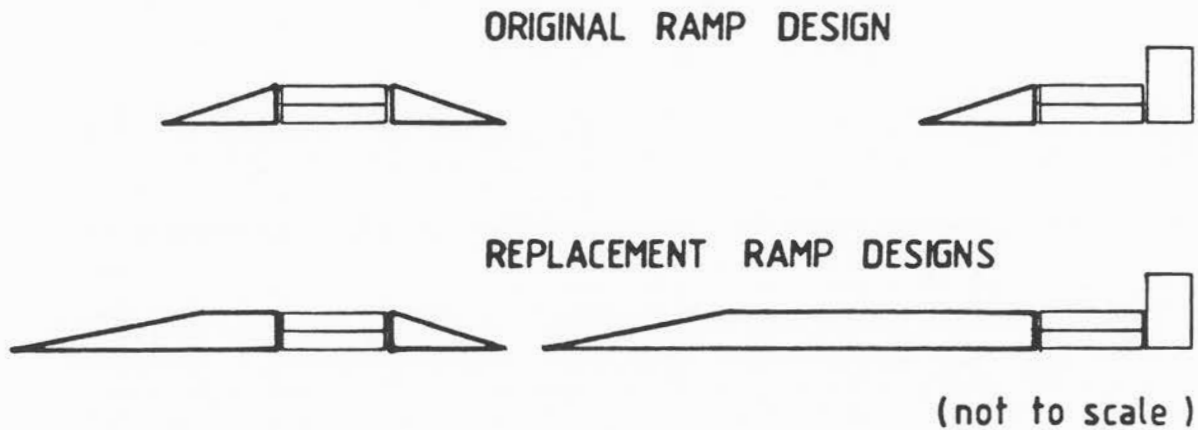


Figure D.2 Original and Redesigned Ramps

The upper plates on the linear bearings were found to permit the weighpads to deflect excessively. Reinforcing strips were added to double the thickness (to 25 mm) and increase the width and length. Provision was made, using bolts in tapped holes, to lock the upper plates in position on the tilt platform, temporarily overriding the low-friction bearing properties while the test vehicle was driven into position. Retaining strips were tacked onto the bearing cages to prevent disassembly.

Long spacers between nearside and offside ramp and bearing assemblies were found necessary to prevent the upper set from sliding down when a wheel lifted.

D.4 DESIGN OF LOW FRICTION PADS

D.4.1 Design Requirements

Six bearings pads were required to support a laden three-axle coach on a tilt test (to enable accurate measurement of the lateral forces). The maximum perpendicular load is therefore 100 kN, that is, the maximum capacity axle (nominally 10 ton) balancing on one wheel only.

Deflection was considered critical. If the bearing deformed noticeably under load it would seriously influence the tilt test results.

Pad height was to be minimised to lessen the ramp height required, and hence the difficulties and dangers in driving up the ramps with a £100000+ coach.

The maximum available area for each bearing was 400x500 mm - the dimensions of the weighpad which was to be positioned on top of the low friction pad.

D.4.2 Design Calculations

The deflection constraint eliminated all nonmetallic possibilities, such as air bearings. The two main options were either balls or rollers between solid plates.

Timoshenko (1930) gives a relationship between the maximum vertical load, P , which can be supported by a ball of diameter d , yield stress p_{max} , and modulus of elasticity E , (all expressed in Imperial units) as

$$p_{max} = 0.62 \sqrt[3]{\frac{PE^2}{d^2}} \quad (D.1)$$

Using high tensile steel, with a yield stress of 100000 psi (690 MPa) it was calculated that the load capacity per area is independent of ball diameter (assuming a square rather than hexagonal packing arrangement), and came to 35 kN/m². To support a design load of 100kN thus requires an area of 2.9 m², which vastly exceeds the available area of 0.2 m²!

Rollers between steel plates were therefore considered. The maximum load per unit length, P' , for a roller on a plate, is given by Timoshenko (1930) as

$$p_{\max} = 0.59 \sqrt{\frac{P'E}{d}} \quad (D.2)$$

For the same steel as the previous calculation, this equation gives the area (again independent of roller diameter) to support 100 kN is 0.015 m².

To reduce material costs, mild steel was specified and the calculations repeated with its lower yield stress of approximately 50000 psi (350 MPa). This resulted in a realistic area requirement, which translates into 10 rollers of dimensions $\phi 25 \times 300$ mm. Four linear bearings were constructed to this design, and two with only six rollers each were employed at the front axle.

APPENDIX E

SIMPLIFIED ANALYSIS OF ROLLOVER THRESHOLD FOR CURB IMPACT

This simple treatment is outlined in context in Jones (1976). a more thorough examination of the problem, together with investigation of a car curb-impact-rollover test is given in Jones (1975).

The suspension and tyre compliances are ignored and the vehicle is modelled as a rigid block (Figure E.1). Forward motion is neglected.

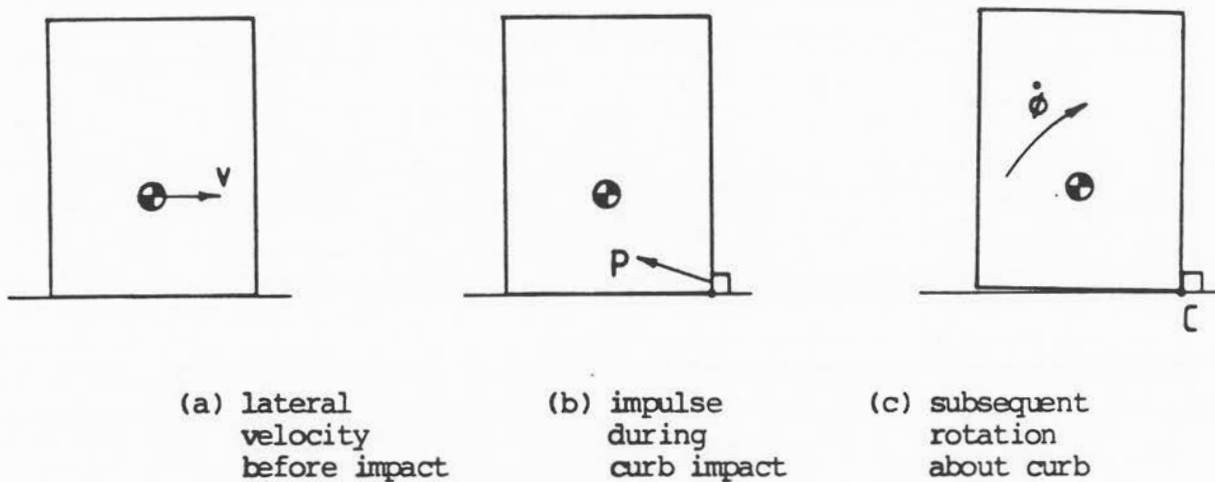


Figure E.1 Assumed Stages of Simple Curb-Impact Rollover

The duration of the impact is assumed sufficiently short to consider the impact force at the curb as an impulse (the car test reported in Jones, 1975, was in contact with the curb for 0.2 seconds). After the initial impact, the body is assumed to rotate about point C, ie. the curb. Using conservation of angular momentum about C gives,

$$I_C \dot{\phi} = mvz \quad (E.1)$$

where I = roll moment of inertia of vehicle about C
 $\dot{\phi}$ = roll velocity
 m = total vehicle mass
 v = lateral velocity prior to curb impact
 z = CG height above the ground

also y = half-track width
 g = acceleration due to gravity

The criterion for rollover is that the energy after impact must be sufficient to raise the CG to reach the point of balance. The increase in height is illustrated in Figure E.2, thus:

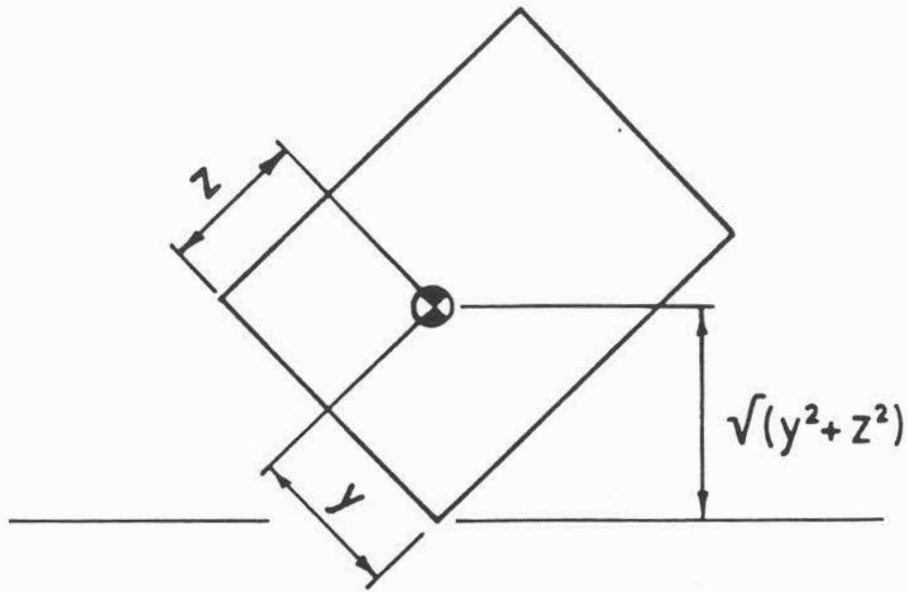


Figure E.2 Height of CG at Point of Balance

$$h = \sqrt{(y^2 + z^2)} - z \quad (\text{E.2})$$

Conservation of energy gives the condition for rollover to occur

$$\frac{1}{2} I_c \dot{\phi}^2 > mg[\sqrt{(y^2 + z^2)} - z] \quad (\text{E.3})$$

Squaring equation (E.1) and rearranging terms gives

$$\frac{1}{2} I_c \dot{\phi}^2 = \frac{m^2 v^2 z^2}{2 I_c} \quad (\text{E.4})$$

$$\text{thus } v^2 > \frac{2g I_c [\sqrt{(y^2 + z^2)} - z]}{m z^2} \quad (\text{E.5})$$

The rollover threshold velocity is that which makes equation (E.5) an equality. Note also that

$$I_c = I_x + m(y^2 + z^2) \quad (\text{E.6})$$

where I_x = roll moment of inertia of vehicle about its CG.

APPENDIX F

TILT TEST SIMULATION EQUATIONS

The essential features and assumptions of the tilt test simulation are outlined in Section 5.2. Referring to Figure 5.1, the following equilibrium equations may be written.

F.1 SPRUNG MASS EQUATIONS

Taking moments about the roll centre gives,

$$(Ws_1.Bs_1 + Ws_2.Bs_2)\sin(\phi_T + \phi_S) - TR_1 - TR_2 = 0 \quad (F.1)$$

Equilibrium of forces perpendicular to axle_i gives,

$$Fz_i = Ws_i \cos(\phi_T + \phi_i) \quad (F.2)$$

Equilibrium of forces parallel to axle_i gives,

$$Fy_i = Ws_i \sin(\phi_T + \phi_i) \quad (F.3)$$

F.2 AXLE EQUATIONS

Taking moments about the roll centre gives,

$$\begin{aligned} TR_i - (Ft_{i4} - Ft_{i1}).U_i - (Ft_{i3} - Ft_{i2}).T_i + Ftz_i(Yt_i + Hu_i\phi_i) \\ + Fty_i(Ht_i + Hu_i) - Wu_iHu_i\sin(\phi_T + \phi_i) = 0 \end{aligned} \quad (F.4)$$

Equilibrium of forces perpendicular to the platform gives,

$$Ftz_i = Wu_i\cos\phi_T + Fz_i + Fy_i\phi_i = 0$$

Substituting for Fz_i and Fy_i from (F.2) and (F.3),

$$Ftz_i = Wa_i\cos\phi \quad (F.5)$$

where $Wa_i = Wu_i + Ws_i$ (F.6)

Equilibrium of forces parallel to the platform gives,

$$\begin{aligned} Fty_i &= Wu_i\sin\phi_T + Fy_i - Fz_i\phi_i \\ \Rightarrow Fty_i &= Wa_i\sin\phi_T \end{aligned} \quad (F.7)$$

Substitute for Ftz_i and Fty_i in (F.4),

$$\begin{aligned} TR_i - (Ft_{i4} - Ft_{i1}).U_i - (Ft_{i3} - Ft_{i2}).T_i \\ + Wa_i(Yt_i\cos\phi_T + Ht_i\sin\phi_T) + Ws_iHu_i\sin(\phi_T + \phi_i) = 0 \end{aligned} \quad (F.8)$$

F.3 SPRING EQUATIONS

$$TR_i = Q_i (\phi_S - \phi_i) \quad (F.9)$$

$$Fz_i = Kz_i Z_i \quad (F.10)$$

$$Fty_i = Kty_i Yt_i \quad (F.11)$$

$$Ft_{i1} = Kt_{i1} (Zt_i - U_i \phi_i) \quad (F.12)$$

$$Ft_{i2} = Kt_{i2} (Zt_i - T_i \phi_i) \quad (F.13)$$

$$Ft_{i3} = Kt_{i3} (Zt_i + T_i \phi_i) \quad (F.14)$$

$$Ft_{i4} = Kt_{i4} (Zt_i + U_i \phi_i) \quad (F.15)$$

F.4 EQUILIBRIUM EQUATIONS

$$(Ws_1.Bs_1 + Ws_2.Bs_2)\sin(\phi_T + \phi_S) - Q_1(\phi_S - \phi_1) - Q_2(\phi_S - \phi_2) = 0 \quad (F.16)$$

$$Kz_i Z_i - Ws_i \cos(\phi_T + \phi_i) = 0 \quad (F.17)$$

$$\begin{aligned} Q_i(\phi_S - \phi_i) - (Kt_{i4} - Kt_{i1})U_i Zt_i - (Kt_{i1} + Kt_{i4})U_i^2 \phi_i \\ - (Kt_{i3} - Kt_{i2})T_i Zt_i - (Kt_{i2} + Kt_{i3})T_i^2 \phi_i \\ + Wa_i [Yt_i \cos \phi_T + (R_i - Zt_i) \sin \phi_T] \\ + Ws_i (H_i - Z_i) \sin(\phi_T + \phi_i) = 0 \end{aligned} \quad (F.18)$$

$$\begin{aligned} (Kt_{i1} + Kt_{i2} + Kt_{i3} + Kt_{i4})Zt_i + (Kt_{i4} - Kt_{i1})U_i \phi_i \\ + (Kt_{i3} - Kt_{i2})T_i \phi_i - Wa_i \cos \phi_T = 0 \end{aligned} \quad (F.19)$$

$$Kty_i Yt_i - Wa_i \sin \phi_T = 0 \quad (F.20)$$

F.5 METHOD OF SOLUTION

The procedure adopted was to fix ϕ_T and solve for the remaining variables ($\phi_S, \phi_1, \phi_2, Z_1, Z_2, Zt_1, Zt_2, Yt_1, Yt_2$). The lateral spring compliance, although modelled in the rollover simulation, was neglected to simplify solution.

A Newton-Raphson approach was used to iterate to the true solution. For example, for equation (F.16),

$$f_1(\phi_S, \phi_1, \phi_2) = 0$$

Initial values of ϕ_S , ϕ_1 , and ϕ_2 are estimated. The true solution is given by,

$$f_1(\phi_S + \delta\phi_S, \phi_1 + \delta\phi_1, \phi_2 + \delta\phi_2) = 0 \quad (F.21)$$

Newton-Raphson gives,

$$\frac{\partial f_1}{\partial \phi_S} \delta\phi_S + \frac{\partial f_1}{\partial \phi_1} \delta\phi_1 + \frac{\partial f_1}{\partial \phi_2} \delta\phi_2 = -f_1(\phi_S, \phi_1, \phi_2) \quad (F.22)$$

Express equations in this form:

$$\begin{aligned} & [(W_{S1} \cdot B_{S1} + W_{S2} \cdot B_{S2}) \cos(\phi_S + \phi_T) - Q_1 - Q_2] \delta\phi_S + Q_1 \delta\phi_1 \\ & + Q_2 \delta\phi_2 = -f_1(\phi_S, \phi_1, \phi_2) \end{aligned} \quad (F.23)$$

$$W_{Si} \sin(\phi_T + \phi_i) \delta\phi_i + Kz_i \delta Z_i = -f_2(\phi_i, Z_i) \quad (F.24)$$

$$\begin{aligned} & Q_i \delta\phi_i + [-Q_i - (Kt_{i1} + Kt_{i4}) U_i^2 - (Kt_{i2} + Kt_{i3}) T_i^2 \\ & + W_{Si} H_{ui} \cos(\phi_T + \phi_i)] \delta\phi_i - W_{Si} \sin(\phi_T + \phi_i) \delta Z_i \\ & + [(Kt_{i4} - Kt_{i1}) U_i + (Kt_{i3} - Kt_{i2}) T_i - W_{ai} \sin\phi_T] \delta Z_i \\ & + W_{ai} \cos\phi_T \delta Y_i = -f_4(\phi_S, \phi_i, Z_i, Z_i, Y_i) \end{aligned} \quad (F.25)$$

$$\begin{aligned} & [(Kt_{i4} - Kt_{i1}) U_i + (Kt_{i3} - Kt_{i2}) T_i] \delta\phi_i \\ & + (Kt_{i1} + Kt_{i2} + Kt_{i3} + Kt_{i4}) \delta Z_i = -f_6(\phi_i, Z_i) \end{aligned} \quad (F.26)$$

$$Kty_i \delta Y_i = -f_8(Y_i) \quad (F.27)$$

Equations (F.23) to (F.27) are solved simultaneously to determine ϕ_S , ϕ_1 , etc. Figure F.1 is a simple flow diagram indicating the program method and structure. The main features to note are that the simulation begins with the tilt platform horizontal (ie. $\phi_T = 0$) as the solution is readily calculated. ϕ_T is incremented by steps of 1 degree initially. This stepsize is reduced as the limit conditions are approached. The interval between previous solutions is used to determine the estimated solution at the end of the next increment.

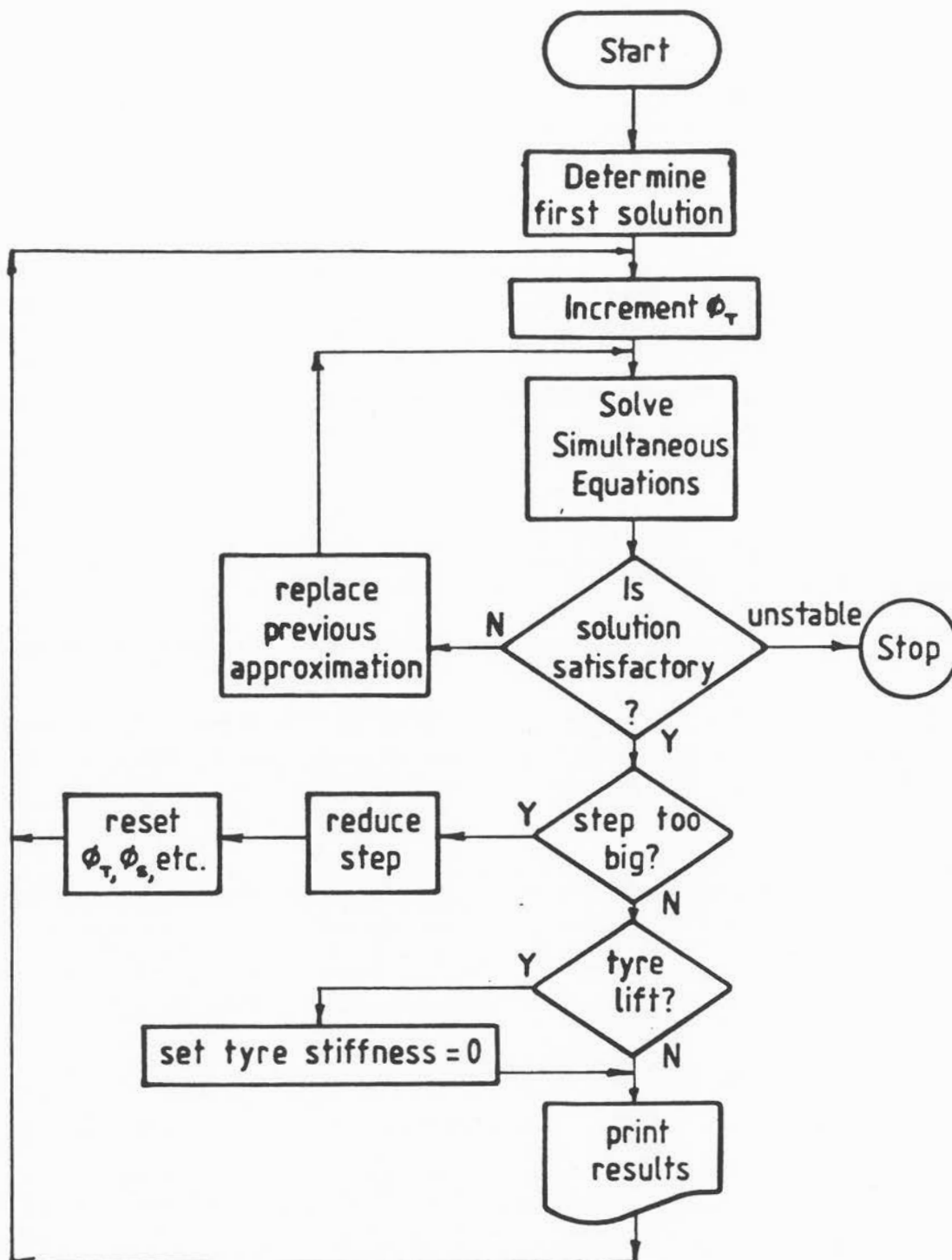
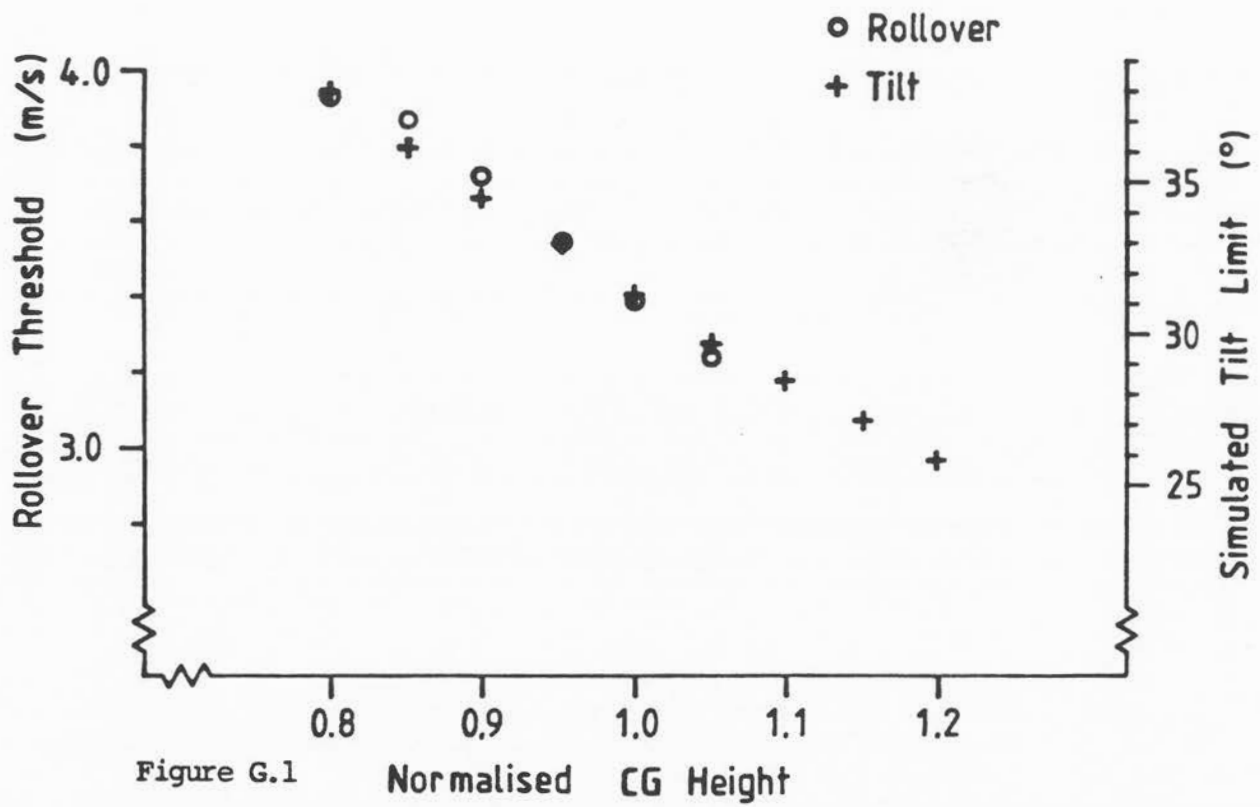
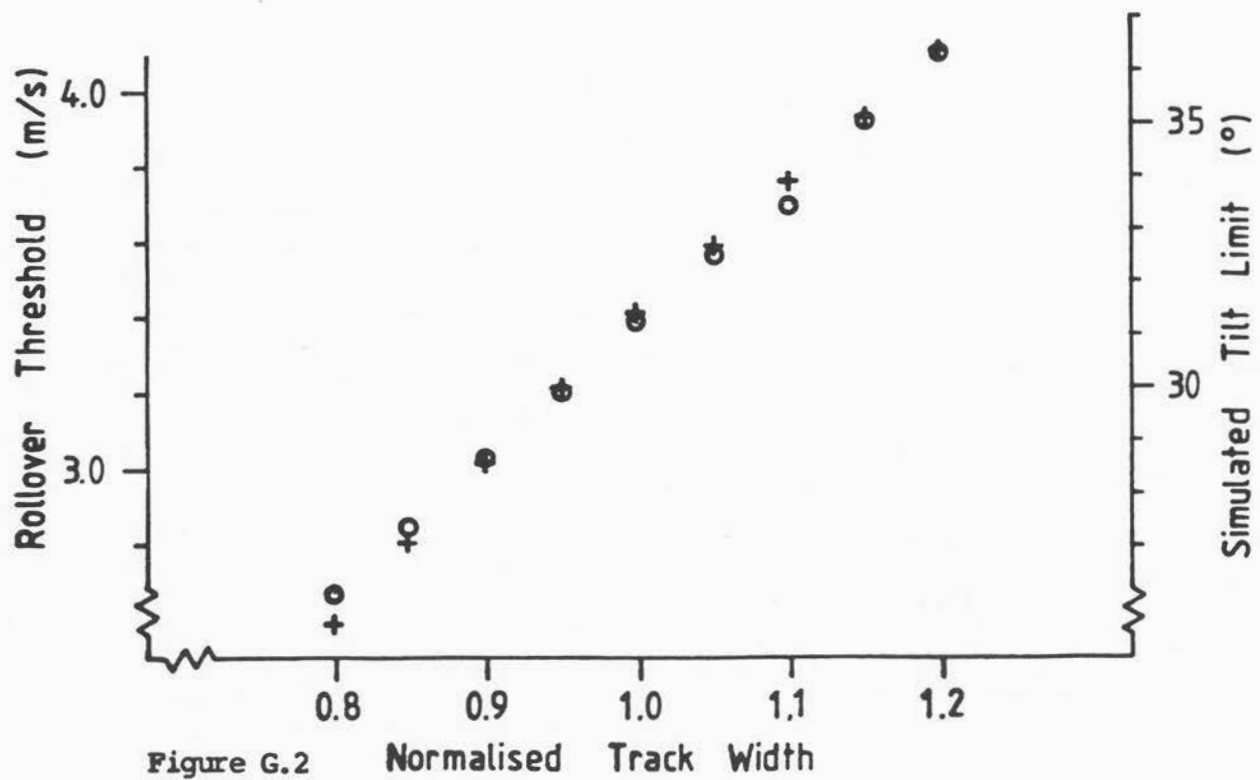


Figure F.1 Flow Diagram for Tilt Test Simulation

APPENDIX G

GRAPHS OF PARAMETRIC EFFECTS ON ROLLOVER THRESHOLD
AND TILT PERFORMANCE



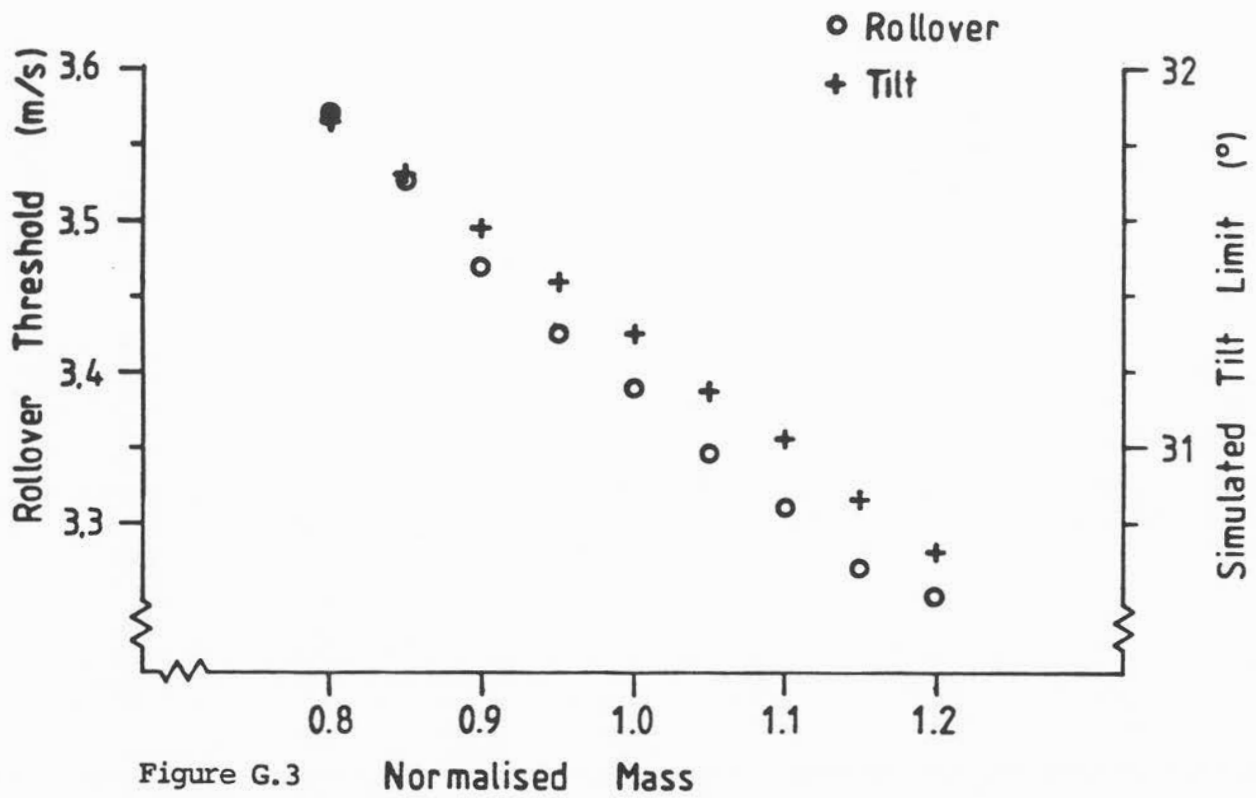


Figure G.3

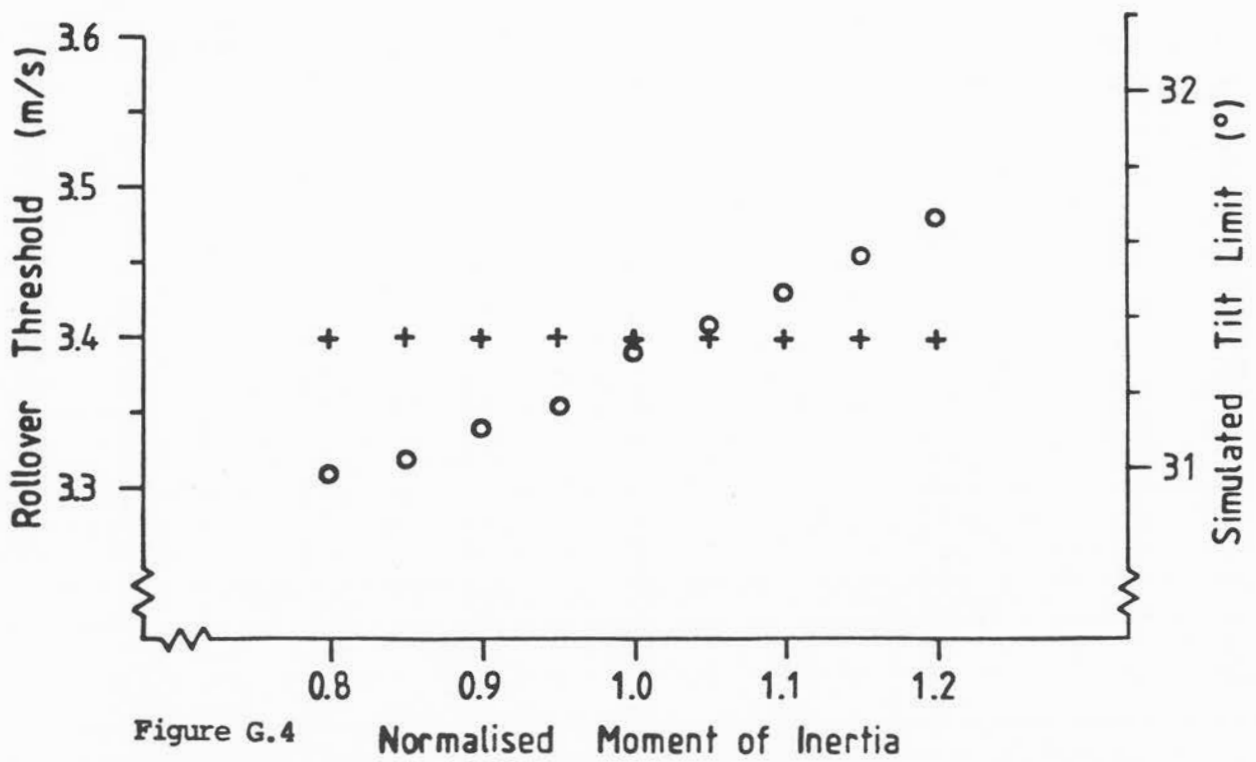


Figure G.4

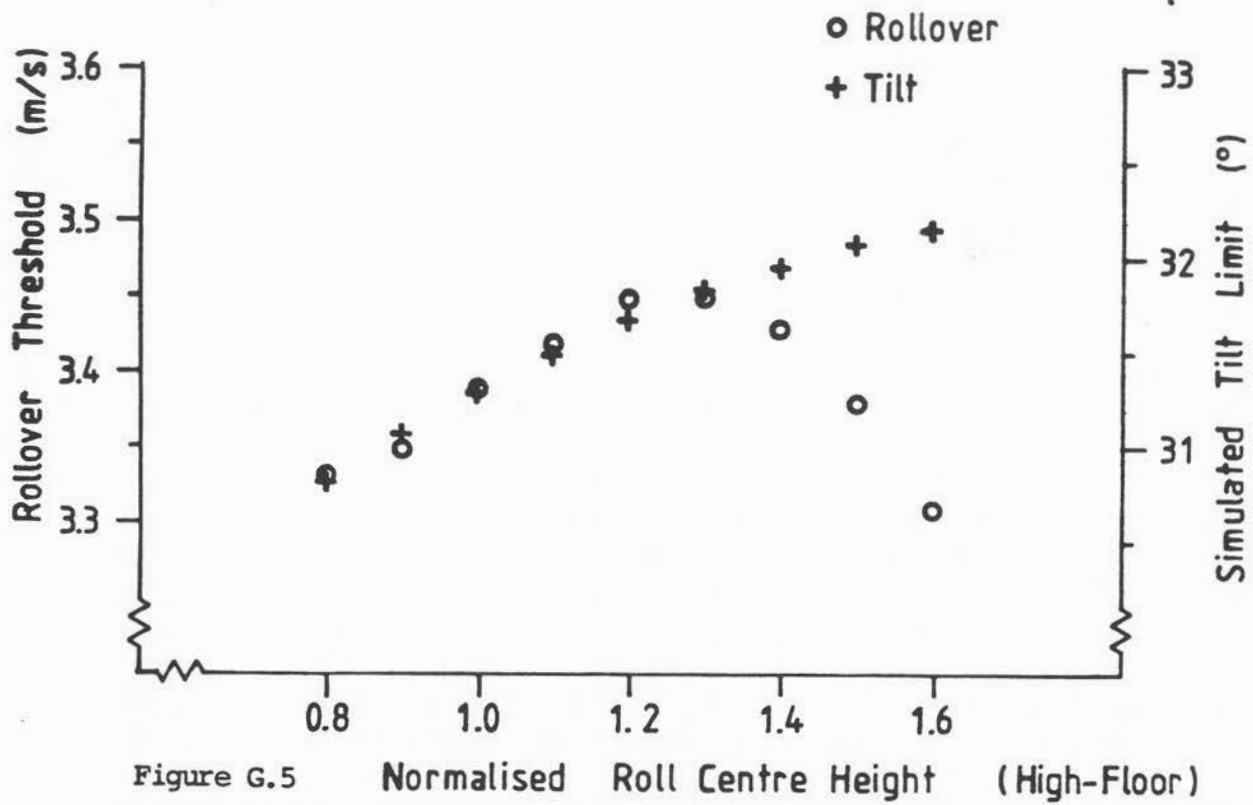


Figure G.5 Normalised Roll Centre Height (High-Floor)

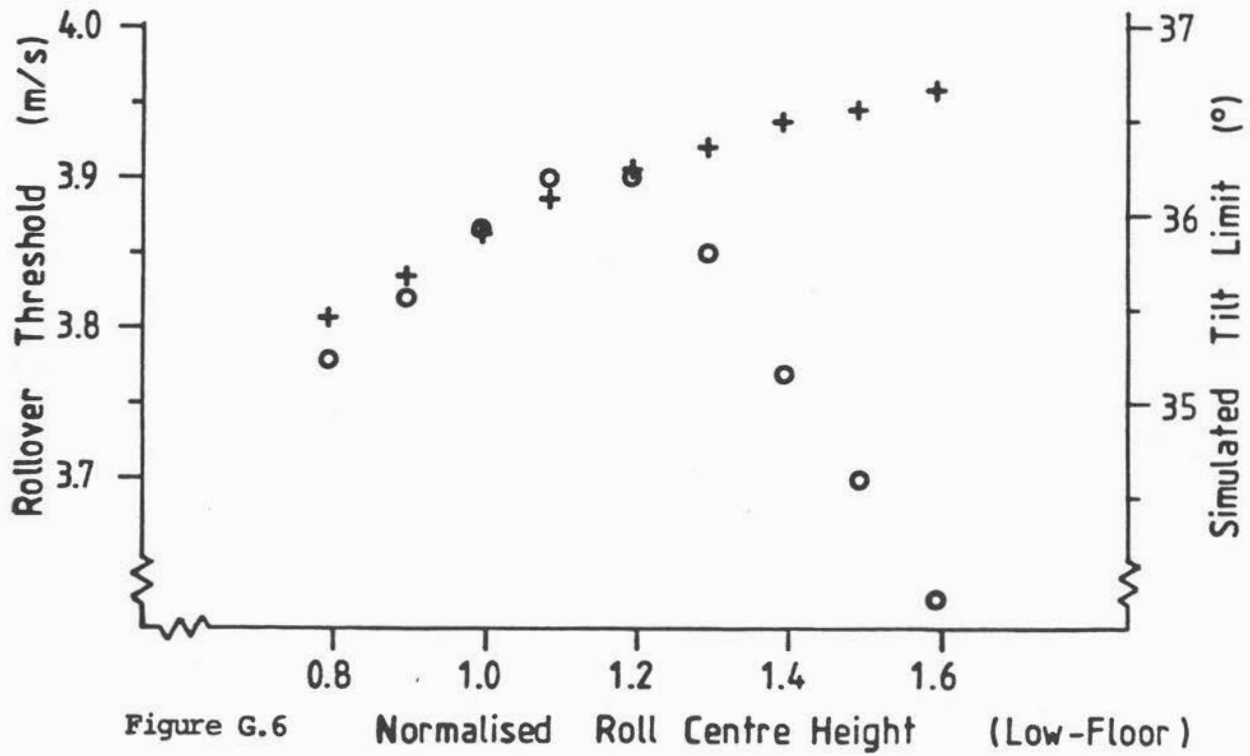


Figure G.6 Normalised Roll Centre Height (Low-Floor)

APPENDIX H ANALYSIS OF DISPLACEMENT DATA

H.1 MEASURED DATA

Before any test commenced, the coordinates of both ends of each linear potentiometer were measured relative to the point on the ground directly below the midpoint of the axle in question. As a check, the overall length of each linear potentiometer was also recorded.

At intervals in the loading sequence, the output voltage of each displacement transducer was noted.

All this raw data together with the force transducer outputs is analysed by the program CALCK (for CALCulation of Stiffness, K).

H.2 PROGRAM METHOD

CALCK converts all the raw data (in Volts or tonnes) into their physical quantities (millimetres or kiloNewtons) using the appropriate calibration constants.

H.2.1 Calculation of Linear and Rotational Displacements

The procedures for transforming changes in displacement transducer length into coordinate displacements and determining roll centre position are best explained with an example.

Figure H.1 illustrates a typical layout of displacement transducers for measuring axle movements relative to the ground.

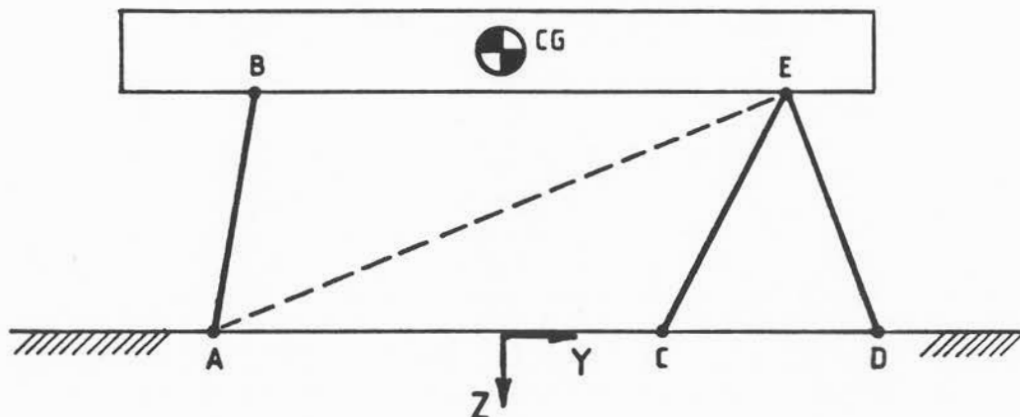


Figure H.1 Typical Transducer Positions

Let $\overline{AB}$ represent the distance from A to B, and so forth. $\overline{AC}$, $\overline{CD}$, and $\overline{BE}$ are lengths on rigid bodies, which are therefore constant. They are either measured directly or calculated from the coordinates of the end points. $\overline{AB}$, $\overline{CE}$ and $\overline{DE}$ are variable lengths representing displacement transducers and are obtained from the measured displacement transducer data. Knowing these six dimensions, it is possible to determine the position and orientation of body 2 relative to the reference axes on body 1.

The position of node E is first calculated, using the cosine rule for angle $\hat{CDE}$,

$$\hat{CDE} = \cos^{-1} \left(\frac{\overline{CD}^2 + \overline{DE}^2 - \overline{CE}^2}{2\overline{CD}\overline{DE}} \right) \quad (\text{H.1})$$

The coordinates of point E are evaluated from those of point D,

$$Y(E) = Y(D) - \overline{DE} \cos(\hat{CDE}) \quad (\text{H.2})$$

$$Z(E) = Z(D) - \overline{DE} \sin(\hat{CDE}) \quad (\text{H.3})$$

To determine coordinates of B, a line from A to E is first constructed, and its length and orientation readily calculated from the coordinates of both points,

$$\overline{AE} = \sqrt{[Y(E) - Y(A)]^2 + [Z(E) - Z(A)]^2} \quad (\text{H.4})$$

$$\hat{EAC} = \tan^{-1} \left(\frac{Z(A) - Z(E)}{Y(E) - Y(A)} \right) \quad (\text{H.5})$$

The cosine rule gives angle $\hat{BAE}$,

$$\hat{BAE} = \cos^{-1} \left(\frac{\overline{AB}^2 + \overline{AE}^2 - \overline{BE}^2}{2\overline{AB}\overline{AE}} \right) \quad (\text{H.6})$$

$$\text{Hence, } \hat{BAC} = \hat{BAE} + \hat{EAC} \quad (\text{H.7})$$

Coordinates of point B are thus,

$$Y(B) = Y(A) + \overline{AB} \cos(\hat{BAC}) \quad (\text{H.8})$$

$$Z(B) = Z(A) - \overline{AB} \sin(\hat{BAC}) \quad (\text{H.9})$$

A similar procedure is used to determine the position of the point on body 2 at which displacements are to be evaluated: usually its centre of gravity (CG).

The angle of body 2 relative to body 1 (positive clockwise) is given by,

$$\phi = \tan^{-1} \left(\frac{Z(E) - Z(B)}{Y(E) - Y(B)} \right) \quad (\text{H.10})$$

Subtraction from the initial values of $Y(OG)$, $Z(OG)$, and ϕ , give the lateral, vertical and roll displacements, respectively.

H.2.2 Calculation of Roll Centre Height

Once the positions of nodes B and E have been determined for two (usually successive) data measurements, it is possible to determine the instantaneous centre of rotation, Figure H.2.

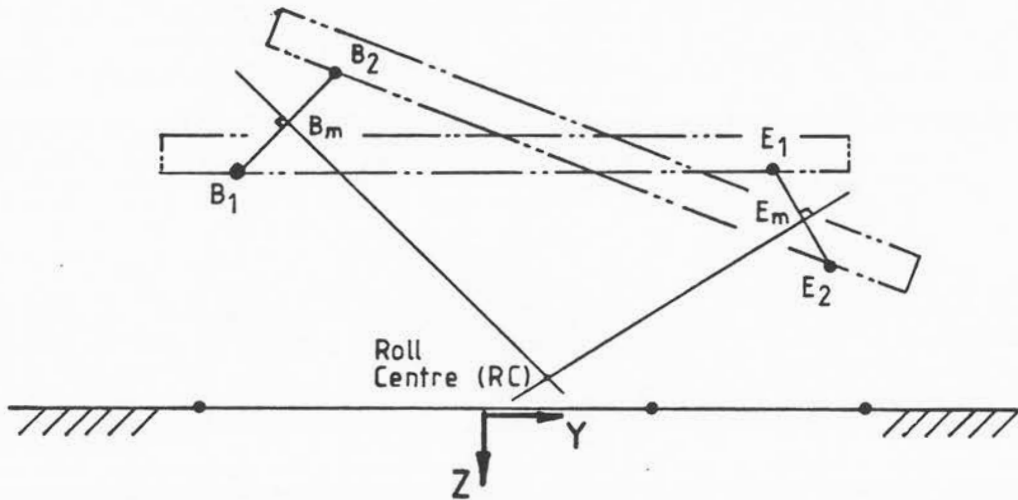


Figure H.2 Method of Evaluating Roll Centre Position

The midpoint of line $\overline{B_1B_2}$ is evaluated,

$$Y(B_m) = \frac{Y(B_1) + Y(B_2)}{2} \quad (H.11)$$

$$Z(B_m) = \frac{Z(B_1) + Z(B_2)}{2} \quad (H.12)$$

The line passing through the midpoint perpendicular to $\overline{B_1B_2}$ has a gradient given by,

$$G_B = \frac{Y(B_2) - Y(B_1)}{Z(B_1) - Z(B_2)} \quad (H.13)$$

The same procedure is used to determine the coordinates of point E_m , the midpoint of E_1E_2 , and the gradient, G_E , of the perpendicular line. The equations of the two lines are solved simultaneously to determine the intersection point, $Y(RC)$, $Z(RC)$, provided that the lines are not parallel.

$$G_B[Z(RC) - Z(B_m)] = Y(RC) - Y(B_m) \quad (H.14)$$

$$G_E[Z(RC) - Z(E_m)] = Y(RC) - Y(E_m) \quad (H.15)$$

Solve for $Z(RC)$,

$$Z(RC) = \frac{Y(E_m) - Y(B_m) + G_B Z(B_m) - G_E Z(E_m)}{G_B - G_E} \quad (H.16)$$

Substitute back to find $Y(RC)$,

$$Y(RC) = G_B [Z(RC) - Z(B_m)] + Y(B_m) \quad (H.17)$$