Articulated dumper truck rollover

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This report details a programme of practical testing and computer simulation of the roll characteristics of articulated site dumper trucks.

Trials were undertaken at MIRA using a Thwaites 6-Tonne, Power Swivel dumper truck to conduct a range of manoeuvres, at a range of speeds and carrying a range of loads. Each experimental case was modelled in a computer simulation. The modelled and measured results showed good correlation, in particular for manoeuvres incorporating larger bumps (100mm in height) and for circling manoeuvres.

From the measuring and modelling undertaken, it appears that the key contributions to rollover occur when working on uneven ground. Dumper trucks feature very low levels of damping since their only ‘suspension’ is by deformation of the tyres and drive train. On uneven ground, this can produce significant roll angles in response to impact with a bump and this effect can be exaggerated if the natural frequency of the response coincides with another wheel of the truck hitting the same bump.

An initial investigation of the response of dumper trucks impacting a bump when operating on a sloping plane was conducted in the computer model. This indicated that impacting a bump can cause the vehicle to roll over on slopes significantly shallower than the static roll angle of the truck.

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EXECUTIVE SUMMARY

This report details a programme of practical testing and computer simulation of the roll characteristics of articulated site dumper trucks.

Trials were undertaken at MIRA using a Thwaites 6-Tonne, Power Swivel dumper truck to conduct a range of manoeuvres, at a range of speeds and carrying a range of loads. Each experimental case was modelled in a computer simulation developed by Frazer-Nash in Phase 2 of this work.

The modelled and measured results showed good correlation, in particular for manoeuvres incorporating larger bumps (100mm in height) and for circling manoeuvres. Other tests conducted included J-turns and lane changes, as well as tests with smaller bumps, however these did not produce a significant response in the vehicle, at speeds which could be safely undertaken by the human test driver required in the MIRA trials.

Several practical considerations emerged from the programme of testing, most noticeably the difficulty in driving the trucks, and the unpredictability of steering arising from ‘slop’ in the vehicle bucket. For this reason, it is suggested that driver training could play a significant part in increasing safety in working with articulated dumper trucks.

From the measuring and modelling undertaken, it appears that the key contributions to rollover occur when working on uneven ground. Dumper trucks feature very low levels of damping since their only ‘suspension’ is by deformation of the tyres and drive train. On uneven ground, this can produce significant roll angles in response to impact with a bump and this effect can be exaggerated if the natural frequency of the response coincides with another wheel of the truck hitting the same bump.

An initial investigation of the response of dumper trucks impacting a bump when operating on a sloping plane was conducted in the computer model. This indicated that impacting a bump can cause the vehicle to roll over on slopes significantly shallower than the static roll angle of the truck.

It is recommended that a parametric study be undertaken to investigate how key parameters determined by the design of articulated dumper trucks can be varied to increase dumper stability, across a range of manoeuvres and on a range of slopes.
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1. INTRODUCTION

1.1 MOTIVATION

The HSE is concerned about the number of construction accidents involving compact dumper trucks (meaning dumper trucks where the load is in front of the driver). In a recent study, of accidents investigated by HSE inspectors, 39% involved the vehicle overturning:

- Some of these incidents involved high reach dumpers where the load (and hence the CoG) can be lifted and the cause of the accidents seems clear.
- Others involved the vehicle tipping about its front or rear axles when negotiating a steep slope, and again the cause is clear.
- However, well over half of the accidents were lateral overturns and in many cases these incidents occurred on relatively modest slopes (say 20%).

There are no requirements in the design standards (currently BS EN 474-6) covering dumpers for the truck to meet any particular level of stability, but most manufacturers are believed to test their vehicles to a code of practice involving tipping the vehicle with full articulation on a tilt table and measuring the angle at which one wheel lifts clear of the platform. No data is available at present for these tests but it is understood that a dumper will typically overturn at around 65% (33 degree) tilt angle. Manufacturers tend to recommend operation at up to half the tilt angle. This does not fit easily with the accident statistics where vehicles have turned over on 20% (11 degree) slopes.

As a result, the HSE wish to look further into the stability of dumpers. It seems possible that there are two instability modes for articulated dumpers:

- Mode 1 occurs when the vehicle tips about a line joining the contact points of the two wheels lowest down the slope.
- Mode 2, it is suggested, could occur when the front part of the truck tips about a line joining the contact point of the lower front wheel and the articulation joint. The articulation joint permits the two halves of the truck to not only swivel about a vertical axis in order to steer, but also to pivot to a limited extent about the horizontal fore and aft axis to enable all four wheels to remain in contact with the ground. It is suggested that if the uphill front wheel hits a bump, this could initiate instability of the front section of the truck. When the articulation joint reaches its limit, the question is whether the front section of the truck has sufficient inertia to overcome the stabilising moment of the rear section and cause the entire truck to fall on its side.

1.2 WORK TO DATE

Frazer-Nash studied a number of site dumper trucks, with assistance from Terex Compact Equipment and Thwaites Ltd, who, between them, supply the majority of site dumpers in use in the UK. Under Phase 1 of the research contract, a parametric model of an articulated site dumper was developed, and verified by simple hand calculations.
Phase 2 of the work developed a Graphical User Interface (GUI) to enable parametric studies of dumper designs to be quickly and easily undertaken, and the results displayed in a user-friendly format.

Phase 3 of this work, reported here, comprised a programme of testing of a 6-tonne site dumper truck, and validation of the computer model by comparison of the simulated and actual results arising from several manoeuvres.
2. SUMMARY OF TRIALS

Practical trials were undertaken under subcontract to MIRA, and took place from 25th June to 3rd July 2007, with Frazer-Nash in attendance. MIRA supplied all data files from the testing to Frazer-Nash, for comparison to the modelled cases. Additionally, all trials were digitally photographed and video-recorded and MIRA produced a report on the testing, which appears at Appendix A of this report.

2.1 MANOEUVRES TESTED

Four types of manoeuvre were investigated in the practical trials. The manoeuvres are described in detail in Appendix A, and are briefly summarised in sections 2.1.1 to 2.1.4 below. Each manoeuvre was repeated with the truck unloaded, half-filled with wet clay, and loaded to the maximum permissible front axle weight.

2.1.1 Steady state circular test

A constant steering input was applied, and the vehicle driven in a circle of nominally constant radius at gradually increasing speed. For the cases with a small turning radius the vehicle was unable to reach its maximum speed. This test was designed to validate the steering response of the vehicle and in particular to ensure the tyre model gave an appropriate response.

The test was repeated at three steering angles, giving circles of approximately 40m, 25m and 10m diameter.

2.1.2 J-Turn

The vehicle was driven in a straight line at constant speed and a steering input applied as rapidly as possible, then held constant, causing the vehicle path to describe a characteristic “J-Turn”. This test was performed with equal steering inputs at a range of up to four speeds, as permitted by the power characteristics of the vehicle.

2.1.3 Driving over a bump

The vehicle was driven at constant speed over a bump, both whilst travelling in a straight line and whilst on a constant radius circular path. The test was conducted such that the bump passed first under the front offside wheel, then immediately under the rear offside wheel on each occasion. The test was repeated at each of four speeds (10, 15, 20 and 25km/h where possible), and with both 50mm and 100mm bump heights.

This test was designed to validate the dynamic response of the vehicle to a step input.

2.1.4 Single lane change

The vehicle was driven through a set of coned boxes simulating a sudden change of lane to the lane on the immediate left. The test was based upon that described in the British Standard ‘Passenger cars. Test track for a severe lane-change manoeuvre. Obstacle Avoidance’ (ISO 3888-2:2002), suitably modified to reflect the capabilities of the vehicle under test. The modifications encompassed a reduction in the speed of the
vehicle and alterations to the size of the coned area in which the lane change has to take place.

2.2 SAFETY

MIRA undertook responsibility for site safety during the trials. The dumper was modified with roll stability bars and polystyrene "snubbers" to limit the maximum roll of any portion of the vehicle, as shown below.

The roll protection system added significant extra weight and inertia to the vehicle, and hence the computer model was adapted to account for this before simulation was undertaken.

The trials took place on a closed, single-vehicle test area and all driving was undertaken by a qualified test driver supplied by MIRA. A safety observer attended all tests, and held the authority to stop any test he deemed unsafe.

2.3 INSTRUMENTATION

The instrumentation of the vehicle was undertaken by MIRA, and details of the equipment used appear in Appendix A. Calibration and QA certification of the equipment was undertaken by MIRA. The following measurements were recorded:

- Vehicle Speed (by GPS)
- Lateral acceleration of rear of vehicle
- Longitudinal acceleration of rear of vehicle
- Vertical acceleration of rear of vehicle.
- Lateral acceleration of front of vehicle
- Longitudinal acceleration of front of vehicle
- Rear yaw angle
• Rear pitch angle
• Rear roll angle
• Rear yaw rate
• Rear roll rate
• Vertical acceleration of front of vehicle.
• Relative Roll between front and rear portions of vehicle
• Relative articulation between front and rear portions of vehicle.

2.4 MIRA

Frazer-Nash would like to acknowledge the contribution made by MIRA to this work. The professionalism shown by their staff, and their dedication to completing the trials and producing high quality data despite extremely inclement weather conditions was instrumental in permitting the validation of the computer model.
3. DATA CAPTURED

During the trials, all measurements were captured at 100Hz sampling frequency. The data was reviewed by the Frazer-Nash observer and MIRA test engineer at the completion of each test. Where obvious anomalies or unexpected results were observed (such as sensor malfunctions or driver error) the test was repeated and the initial data discarded. One data file for each manoeuvre was supplied, regardless of how many repetitions of that manoeuvre were necessary.

The data was supplied to Frazer-Nash in standard comma-separated-variable format, amenable to further analysis in proprietary packages.

The data supplied was of good quality, however significant noise was present in the accelerometer readings, likely attributable to vibration of the vehicle caused by the engine and transmission. Estimates of noise levels in the measurement of key vehicle parameters are presented below:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Maximum experienced noise component</th>
<th>Typical peak value for parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll (all)</td>
<td>1 degree</td>
<td>8 degrees</td>
</tr>
<tr>
<td>Pitch</td>
<td>0.5 degrees</td>
<td>2 degrees</td>
</tr>
<tr>
<td>Z acceleration (all)</td>
<td>2 m/s^2</td>
<td>21 m/s^2</td>
</tr>
</tbody>
</table>

A typical example of this is given in the results for the 100mm cornering bump test number three, for load case 1, presented in Appendix D.

Nonetheless, in all cases tested, the signal to noise ratio appears to be significantly less than 1.0, and hence comparisons between measured and modelled data could be made straightforwardly when the data was represented graphically.

Early in the programme of testing, it was noticed that the testing site was neither completely level nor flat, and indeed sloped up to 1.5 degrees in some positions in an unpredictable way which could not be compensated for in the model. Such undulations appear as steady-state measurements of roll and pitch for the vehicle, relative to the position of the vehicle when the instrumentation was calibrated.

This gives rise to two effects:

- Measurement Offset
  The measuring equipment was frequently reset during the test programme, since it was necessary to shut down the equipment and move the vehicle under cover each time inclement weather halted testing. All measurements of roll, pitch, and yaw are relative to the orientation of the vehicle when the instrumentation was reset. Hence, if the vehicle is pitched by 1 degree, and showing 1 degree of roll due to an un-level surface when the instrumentation was reset, future measurements of pitch and roll will be “offset” by this amount. Similarly, where there was relative roll or steer (however slight) between the
front and rear portions of the vehicle when the equipment was calibrated, this offset persists throughout the results.

Measurement offset also arose in the measurement of pitch in load cases 2 and 3. As the vehicle bucket was loaded, the vehicle pitches forwards, since the front axle load is higher than the rear, causing increased compression of the front tyres. When the measurement equipment was calibrated before the test, this steady state pitch was recorded as the zero level, introducing a further source of measurement offset on the measurement of pitch.

**Datum Uncertainty**

All manoeuvres involved traversing the test area in some way. Hence the “steady state” roll and pitch of the truck was seen to vary unpredictably in response to undulations in the ground, negating any dynamic effects. This was confirmed by driving the truck as slowly as possible in circles of various diameters and inspecting the recorded roll and pitch data.

Datum uncertainty is most apparent in the results of the steady-state circular tests. Since the vehicle described approximately the same path several times during the course of each steady-state circular test, the undulations can be seen to appear repeatedly (with ever shorter time period as the truck accelerates) in the measurement of both roll and pitch. The roll and pitch of the vehicle caused by the centripetal acceleration is of the order of 25 – 50% of this value, but nonetheless can be seen as a “trend” of the fundamental undulation. An example of the measured data from the medium radius, unloaded case can be seen below in Figure 1

![Figure 1 Measurements of front and rear roll angle during steady state circular test](image-url)
4. MODEL DESCRIPTION

4.1 GENERAL
A computer model of a site dumper truck has been developed in the programming language IDL, treating the front and rear portions of the truck as independent rigid bodies, constrained by the pivot and reacted by a spring damper system to represent each tyre. This is shown in Figure 2 below.

![Rigid-body Model](image)

**Figure 2** Rigid-body Model

The model has been developed to facilitate the investigation of the following parameters on vehicle stability:

- Bump profiles
- Bump heights
- Vehicle speeds
- Slope angles

Each of these parameters was investigated in the vehicle trials, with the exception of varying slope angles, since no suitably large, evenly sloped test area was available.

4.2 AXIS SYSTEMS
X, Y and Z axes were defined at the centre of gravity of each body, aligned with the centreline of that body, and roll, pitch and yaw defined as positive rotations about these Cartesian axes in the usual manner, i.e:

- Roll is a right-handed rotation about the x-axis
- Pitch is a right-handed rotation about the y-axis
4. Yaw is a right-handed rotation about the z-axis.

Separate axis systems were defined in each of the front (body 1) and rear (body 2) portions of the truck. A global axis system was also defined, centred at the origin from which movement of the truck in the global frame could be referenced. Other intermediate frames of reference were defined as required in order to facilitate transformations between the principal axis systems.

This is shown below.

![Sign convention]

4.3 INPUT PARAMETERS

The model takes the following input parameters:

- Mass of front section (kg)
- Mass of rear section (kg)
- Spring Stiffness (N/m) for tyre deflection, both front and rear
- Coefficient of Friction between tyre and ground
- Tyre Performance Coefficient
- Track width of front axle
- Track width of rear axle
- Diameter of front wheels
- Diameter of rear wheels
- Wheelbase
• Roll inertia of front portion of vehicle
• Roll inertia of rear portion of vehicle
• Compound pitch inertia for the vehicle
• Compound yaw inertia for the vehicle
• Type of pivot (articulate-oscillate or KingLink™)
• Position of centre of gravity of each body, relative to axle centre
• Position of pivot, relative to rear axle centre
• Slope angle
• Mass of Load
• Centre of Gravity position of load, relative to front axle centre.
• Profile of contact point of wheel as it passes over bump
• Time at which bump is first touched by front wheel
• Required steering angle and vehicle speed for each time step.

These parameters can be entered into the GUI and saved as XML files to represent different trucks or manoeuvres, allowing cases to be stored for later repetition.

4.4 MODELLING APPROACH

The model employs a vector method, since this simplifies the calculation of complex, changing 3D geometry. Furthermore, movement and rotation within frames is easily captured. In particular, a vector solution overcomes difficulties associated with:

• Roll causing pitch and yaw.
  For example, when the front of the vehicle rolls relative to the rear, the centre of gravity of the front portion moves upwards (i.e. the vehicle pivots slightly backwards about the rear axle) causing the vehicle as a whole to pitch backwards. This coupled motion is difficult to predict by a simple geometric method, whereas a suitable vector method automatically accounts for this.

• The roll axis may be moving in the global frame.
  The significant complexity of calculation involved in deducing this 3D geometry was likely to lead to errors.

The Shen tyre model (Ref. 1) was employed to calculate the side forces on the tyres, as a function of the reaction force, tyre slip coefficient and slip angle.

Where a bump was incorporated, it is represented by a vertical displacement of the ground as shown below. This provides a displacement to the unconstrained mass-
spring-damper system which represents the truck. An example of a bump of triangular profile is shown below in Figure 3.

Figure 3 bump input

It was noted, however, that where the radius of the wheel is large compared to the height of the bump, this model does not provide a truly accurate representation of the displacement imparted by the bump. Thus, a tyre contact model was developed to determine the “effective” profile of the bump for input into the model. An example output is shown below, where the dashed path represents the “pseudo-bump” to be inputted into the model.

Figure 4 Tyre contact model

This method assumes a perfectly stiff tyre, and applies this path as the displacement at the base of the spring-damper system which represents the tyres of the truck. The effect of the tyre contact model is to recognise that since the radius of the wheel is large compared to the height of the bump, the bump is effectively elongated by the tyre rolling over it.

The incorporation of this tyre contact model into the simulation significantly improved the simulation of all manoeuvres which incorporated a bump.

4.5 CALCULATION METHOD

4.5.1 Forces on the truck

The forces of the truck consist of the following:

- Gravity;
• Vertical forces at the wheels;

• Side forces at the wheels due to the slip angle between the direction of travel and the orientation of the wheel, and the vertical force on the wheel;

• Driving force on the front and rear wheels.

4.5.2 Gravity

When the truck is modelled on level ground, the gravitational force acts vertically upon each body of the truck and remains constant.

When the truck is modelled on a slope, the global reference frame is inclined, such that the X and Y axis of the global frame lie on the constant slope. Gravity is adjusted to act in the relevant direction and remains constant in this frame of reference.

4.5.3 Vertical Forces

Vertical forces are controlled dynamically by a spring-mass-damper system.

For each wheel, the length of the vector normal to the ground, from the wheel centre to the intersection with this plane is determined (zlength), as is the rate of change of this vector (VR).

Spring stiffness at the front of the vehicle is determined:

\[ k_{\text{Front}} = k \times K_{\text{Fac}} \]

Where the coefficient \( K_{\text{Fac}} \) allows tuning of the suspension system.

Similarly the rear spring stiffness is determined:

\[ k_{\text{Rear}} = k \times K_{\text{Fac}} \times 2.5/3.9 \]

where front tyre pressure is 3.9Bar and rear tyre pressure 2.5Bar.

Damping Factors are determined to give critical damping, then tuned using the coefficient \( CF_{\text{Fac}} \):

\[ CF_{\text{Front}} = CF_{\text{Fac}} \times (2.0 \times \sqrt{\text{k}_{\text{Front}} \times (m1)}) \]
\[ CF_{\text{Rear}} = CF_{\text{Fac}} \times (2.0 \times \sqrt{\text{k}_{\text{Rear}} \times (m2)}) \]

Reaction force at the wheel is determined:

\[ R11 = k_{\text{Front}} \times (\text{zlength11} \times \text{znatural}) - CF_{\text{Front}} \times (\text{DOT}([VR_{11}, 0.0, 0.0, 1.0]) - \delta_{\text{fbh}})) \]

Where \( \delta_{\text{fbh}} \) accounts for any undulation in the ground (i.e. bump input) which has occurred during the time-step.
4.5.4 Side Forces
The side forces are a function of vertical force and slip angle.

The slip angles are calculated for each wheel from the difference between the direction of travel of the relevant body and the orientation of the wheels, both projected onto the plane of the ground.

\[
\text{SLIP} = \frac{\text{ydot} + (\text{DOT}\left((\text{GR}), [1.0, 0.0, 0.0]\right) * \text{alphadot})}{(\text{abs}(\text{xdot}) + (\text{DOT}\left((\text{GR}), [0.0, 1.0, 0.0]\right) * \text{alphadot}))}
\]

Where:
- \text{xdot} and \text{ydot} are the x and y velocities of the body respectively
- \text{GR} is the vector from the common centre of gravity to the contact point of the wheel
- \text{alphadot} is yaw velocity
- \text{abs} represents the modulus of the number.
- \text{DOT} represents the scalar product.

The tyre side forces are given by:

\[
\text{F} = (\mu**\tanh((\text{CSLIP} * \text{SLIP}) / (\mu * \text{R})))
\]

Where:
- \(\mu\) is the coefficient of friction
- \(\text{R}\) is the reaction force
- \(\text{CSLIP}\) is the tyre slip coefficient.
- \(\text{SLIP}\) is the slip angle.

4.5.5 Driving Force
The required speed of the model at each time step is read from a data file, and the simulation determines an appropriate total driving force to be applied to generate this speed.

This is achieved by iteratively guessing a value of drive force and calculating the speed increase or decrease which would result in that time step from the application of that force. This is iterated until the required speed is converged upon.

The required total drive force is applied equally, at each wheel, in the local x-direction.

4.5.6 Total Forces and Moments
The applied forces are resolved in the x, y and z directions of body 2 by matrix transformation and matrix summation of the applied forces.

\[
\text{FORCE} = \text{G2} + \text{R212} + \text{R222} + \text{F212} + \text{F222} + \text{R112} + \text{R122} + \text{F112} + \text{F122} + \text{D112} + \text{D122} + \text{D212} + \text{D222}
\]

Where:
- \text{RXX2} is the reaction force at wheel XX, measured in frame 2.
- \text{DXX2} is the drive force at wheel XX, measured in frame 2
- \text{FXX2} is the tyre side force at wheel XX, measured in frame 2.
- \text{G2} is the gravity force, measured in frame 2.
The total moment about the centre of gravity, which is a key determinant of roll, pitch and yaw, is calculated through a series of vector products, thus:

\[
\text{MOMENT} = \text{CROSSP}((-\text{PG} + \text{C12} \# \text{PR11}), (\text{R112} + \text{D112} + \text{F112})) + \\
\text{CROSSP}((-\text{PG} + \text{C12} \# \text{PR12}), (\text{R122} + \text{D122} + \text{F122})) + \\
\text{CROSSP}((-\text{PG} + \text{PR21}), (\text{R212} + \text{D212} + \text{F212})) + \\
\text{CROSSP}((-\text{PG} + \text{PR22}), (\text{R222} + \text{D222} + \text{F222}))
\]

Where:
- \text{CROSSP} represents the vector product operation
- \text{PG} is the vector from the pivot to the common centre of gravity
- \text{PRXX} is the vector from the pivot, to the point of contact of wheel XX
- \text{C12} is the conversion matrix from frame 1 to frame 2
- \# represents matrix multiplication.

### 4.5.7 Accelerations

Linear accelerations are determined in frame 2, thus:

\[
\begin{align*}
\text{xdotdot} &= \frac{(\text{DOT}(\text{FORCE}, [1.0, 0.0, 0.0]))}{(m1+m2)} \\
\text{ydotdot} &= \frac{(\text{DOT}(\text{FORCE}, [0.0, 1.0, 0.0]))}{(m1+m2)} \\
\text{zdotdot} &= \frac{(\text{DOT}(\text{FORCE}, [0.0, 0.0, 1.0]))}{(m1+m2)}
\end{align*}
\]

i.e. the relevant linear acceleration is the appropriate resolution of the total force vector, divided by the total mass of the vehicle.

Global accelerations are determined by converting the local accelerations to the global frame, accounting for the coriolis affect.

### 4.5.8 Simulation

In summary, once inputs are applied, the calculation proceeds as follows:

(a) Calculate the geometry for this time step
   - Current position, slope direction, direction of forces, position of roll axes etc.

(b) Calculate the forces in this time step
   - Normal reactions, tyre side forces, necessary drive forces to achieve required vehicle speed, direction of gravitational forces

(c) Apply the forces, calculate the net force and moment about the common Centre of Gravity
   - Hence net accelerations for 5 degrees of freedom, \( \ddot{x}, \ddot{y}, \ddot{z}, \dot{\alpha}, \dot{\beta} \)

(d) Apply the forces, calculate the roll moment about the separate centres of gravity
   - Hence the net accelerations for 6th and 7th degrees of freedom, \( \ddot{\theta}_1, \ddot{\theta}_2 \)
(e) Integrate the accelerations, to give the displacements and velocities;
   • Using a Runge-Kutta Method
(f) Translate and rotate the bodies to their new positions at end of time step
(g) Repeat from step 1.

4.6 OUTPUT PARAMETERS
A major advantage of the modelling approach is that almost any parameter describing the behaviour of the truck may be reported from the model at every time-step. The following parameters are available to plot from within the GUI:

   • Steering Angle
   • Input Velocity
   • Bump profile
   • Vehicle Path
   • Velocity
   • Acceleration
   • Roll, Pitch and Yaw displacement
   • Roll, Pitch and Yaw velocity
   • Wheel Load
   • Drive Force
   • Vehicle Position
   • Path Curvature.

Furthermore, the graphical user interface presents a 3-D animation of the dumper truck performing the relevant manoeuvre.

In order to validate the model against the data from the trials, several other parameters were written out in a data file for each case. The model was also temporarily adjusted to directly provide linear accelerations at the positions of the instrumentation on the truck used by MIRA, rather than at the centre of gravity of the truck, as is ordinarily the case. This allowed a direct comparison of results when validating the model. And all of the results in this report are presented in this way.

4.7 VERIFICATION APPROACH
In order to verify the computer model, simulations were run using data for a Thwaites 6T Power-Swivel site dumper. An input file was created with the model parameters that encapsulated all the static data for the truck. Frazer-Nash would like to acknowledge the assistance of Thwaites Ltd. in providing the relevant information for this truck, despite it covering aspects of their proprietary knowledge.

The data files received from MIRA were then interrogated and used to produce XML input files to represent the manoeuvres undertaken, and the bump profiles encountered. Each case was simulated, and the results output as a data file. Both the
simulated and measured results were post-processed to provide plots of meaningful parameters.
5. MODELLING RESULTS

5.1 APPROACH

Model input files were created to simulate each of the 75 cases tested by MIRA. Every test undertaken at MIRA was simulated, and the results of the following key parameters plotted, and overlaid on the relevant measured parameters:

- Front Roll Angle
- Rear Roll Angle
- Roll difference (between front and rear portions of the vehicle)
- Vehicle Yaw Displacement
- Front Z Acceleration
- Rear Z Acceleration

5.2 STEADY-STATE CORNERING TESTS

The steady-state cornering tests were used to verify the tyre model, and the equilibrium roll adopted by the vehicle in response to varying levels of lateral acceleration. The lateral acceleration was well predicted by the model, with a slight over-prediction at larger radii. It is also noticeable that the measured signal is exceptionally noisy, particularly at the lesser steering angles (larger radii). These results correlate with observations made during the trial that the vehicle vibrated heavily at higher speeds (achievable only at larger radii) and struggled to hold its path without drifting out in response to small undulations in the test surface. This drifting was not predicted by the model (which assumed a perfectly level, flat surface with even tyre slip coefficient) and hence the modelled prediction of lateral acceleration may be taken as an upper-bound estimate of the behaviour of the real truck. This could have been minimised by using road tyres on the dumper rather than the “knobbed” off-road tyres used throughout the testing, however such a modification may have made the tyre stiffness model unrepresentative of dumper trucks in common usage in the UK.

The maximum roll angle experienced by any portion of the truck during testing was of the order of 3 degrees from the undisturbed position. Full results of the steady-state cornering tests for all load cases appear at Appendix B.
During the steady-state circular tests, the vehicle was predicted to roll slightly to the outside of the turn, and further that this roll would increase with vehicle speed as might be expected. The testing seemed to indicate that the vehicle would also pitch slightly, in response to relative roll between the front and rear portions of the vehicle, indicative of pitch-roll coupling. This does not appear to be predicted by the modelling, however the results are not conclusive from these manoeuvres due to the small amounts of relative roll predicted and experienced during these tests, and the large "noise" component of the uneven surface in the measured data.

Otherwise, the measured results, suitably filtered for the undulations in slope of the test area, appear to correlate well with the modelled data. However, since the undulations
of the test area are of several times the magnitude of the roll and pitch under consideration it is not possible to consider the detail of this correlation with any rigour.

An example of the predicted pitch and roll during the no-load, medium radius steady state circular test is shown below. In this example, the roll of the front and rear portions of the vehicle in response to the increasing lateral acceleration can be clearly seen.

![Figure 7](image1.png)

**Figure 7** Vehicle pitch in medium radius, steady-state circular test, unloaded case

![Figure 8](image2.png)

**Figure 8** Rear roll angle in medium radius, steady-state circular test, unloaded case
The modelling approach used predicts a slight steady state offset in rear roll of up to one degree due to the acceleration of the vehicle (Figure 8). This was not observed in the vehicle tests.

5.3 J-TURN AND S TESTS

Both sets of tests showed that these manoeuvres are unlikely to generate a rollover condition. The truck is stable when driving under these conditions and the trials showed that the gearing on the steering column is such that the driver cannot reasonably change the steering angle quickly enough for a rollover to occur.

The magnitude of roll and pitch, and the accelerations thereof were small compared to the noise in the data and undulations of the ground. For these reasons, J-Turn and lane-change tests contributed little to the understanding of the rollover dynamics of the truck, but does indicate that these manoeuvres are not responsible for vehicle rollovers.

5.4 DRIVING OVER A BUMP ON A STRAIGHT COURSE

The straight-line bump tests were repeated for all speeds and load-cases with both a 50mm and 100mm tall bump, of trapezoidal profile as described in Appendix A. This approach was adopted since there existed some prior uncertainty regarding whether a 50mm bump would cause sufficient disturbance to the motion of the truck to be apparent above the noise in the signal. There was also uncertainty as to whether the tests with a 100mm bump would be possible without exposing the driver to unnecessary shock loadings through the vehicle seat.

During the trials, it was possible to complete all manoeuvres at all speeds within the capabilities of the dumper truck for each load case, with both bump heights and hence the latter concern proved unfounded. However, upon examination of the data, the clearest results were apparent in the test utilising the 100mm bump, and hence these results have been focussed upon in this report since the accelerations and
displacements caused by the 100mm bump are more representative of the rollover condition.

At no time during testing did the vehicle reach the rollover condition, however in several of the tests with the 100mm bump the snubbers contacted the ground and limited the movement of the truck. In each of these instances, it was judged by MIRA to be safe to raise the snubbers to a suitable height and to repeat the test and recapture the data.

Wheel lift was observed in the majority of the bump tests, this included all speeds tested and also all of the bucket loads tested. Given that wheel lift was not observed in the other vehicle manoeuvres, it was concluded that this type of test would most closely simulate rollover.

The 100mm bump tests, both unloaded and at half-load, correlate well with the modelled data. In particular, predicted peak roll and pitch correlate exceptionally well, and the natural frequency of the truck’s z-acceleration after impacting the bump is well matched. This can be seen in the example results below:

**Figure 10** Results of straight-line 100mm bump test. Unloaded case, ~3 m/s.

**Figure 11** Results of straight-line 100mm bump test. Unloaded case, ~3 m/s.
Figure 12 Results of straight-line 100mm bump test. Unloaded case, ~3 m/s

Figure 13 Results of straight-line 100mm bump test. Unloaded case, ~3 m/s

Figure 14 Results of straight-line 100mm bump test. Unloaded case, ~3 m/s
The full results for all 100mm bump straight-line cases appear at Appendix C.

5.5  DRIVING OVER A BUMP ON A CURVED COURSE

As in the straight-line case, the 100mm bump cases provide the clearest insight into the correlation between the measured and modelled data.

The truck was driven towards the bump at constant speed, then a steering input applied by the driver, and held constant whilst the vehicle traversed the bump. Though it was not possible to determine a-priori what path the vehicle would follow, it was possible to replicate this manoeuvre straightforwardly in the model, by using the relative articulation of the vehicle halves as the input steering angle.

Note that this approach implies that any noise or inaccuracy in the measurement of the relative articulation of the front and rear portions of the vehicle will be applied as if that steering input had been applied, however, this is not considered to be a significant problem with the data modelled.

For each of the unloaded and half-load cases tested, the correlation of roll angle, roll difference, z acceleration, vehicle pitch and natural frequency of z acceleration was well matched. This can be seen in the example below:
Figure 16 Results of cornering over a 100mm bump.
Unloaded case, ~3.2m/s

Figure 17 Results of cornering over a 100mm bump.
Unloaded case, ~3.2m/s

Figure 18 Results of cornering over a 100mm bump.
Unloaded case, ~3.2m/s
Figure 19 Results of cornering over a 100mm bump. Unloaded case, ~3.2m/s

Figure 20 Results of cornering over a 100mm bump. Unloaded case, ~3.2m/s

Figure 21 Results of cornering over a 100mm bump. Unloaded case, ~3.2m/s
The full results for all 100mm bump cornering cases appear at Appendix D.

5.6 OVERALL CORRELATION OF RESULTS

Key metrics of correlation between the measured and modelled data for all 44 tests incorporating a bump were extracted from the full results. These are presented in the table below, which facilitates an ‘at a glance’ comparison of the fit between measured and modelled results, by type of manoeuvre and load case.
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<td>0.45</td>
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<td>0.48</td>
<td>0.33</td>
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<td>0.27</td>
<td>-0.54</td>
<td></td>
<td></td>
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<tr>
<td>50mm_bump_straight6</td>
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<td>1.02</td>
<td>0.24</td>
<td>0.65</td>
<td>2.23</td>
<td>1.58</td>
<td>0.09</td>
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<td></td>
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<tr>
<td>50mm_bump_straight7</td>
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<td>1.38</td>
<td>0.96</td>
<td>0.18</td>
<td>0.27</td>
<td>0.09</td>
<td>-0.55</td>
<td></td>
<td></td>
</tr>
<tr>
<td>50mm_bump_straight8</td>
<td>2.36</td>
<td>1.47</td>
<td>0.89</td>
<td>0.83</td>
<td>2.63</td>
<td>1.79</td>
<td>0.03</td>
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<tr>
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<td>1.34</td>
<td>-0.17</td>
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<tr>
<td>50mm_bump_straight10</td>
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<td>1.38</td>
<td>0.78</td>
<td>0.86</td>
<td>0.50</td>
<td>0.36</td>
<td>-0.27</td>
<td></td>
<td></td>
</tr>
<tr>
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<td>0.91</td>
<td>0.35</td>
<td>0.72</td>
<td>0.60</td>
<td>0.12</td>
<td>-0.13</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
In the table above, the maxima of front and rear roll angle, and vehicle pitch are identified for both the measured and the modelled case. For each parameter, the difference between the measured and modelled values is denoted as delta.

The following key has been used in the table to enable at-a-glance interpretation of the delta:

<table>
<thead>
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<th>Table 3 Key code to table 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>delta &lt; 1</td>
</tr>
<tr>
<td>1 &lt; delta &lt; 2</td>
</tr>
<tr>
<td>delta &gt; 2.0</td>
</tr>
</tbody>
</table>

It can be seen that only 7 of the 44 bump-cases modelled show deviations > 2 degrees. Of these, the majority (6/7) occur in the prediction of the roll of the rear portion of the truck. This is the hardest part to predict as it relies on an accurate prediction of the motion of the front of the vehicle as this passes over the bump in order for the simulation to be at the correct starting position for the simulation of the rear passing over the bump.

The results of greatest interest are those incorporating the 100mm bump. These represented 22 cases in total, and of these only 2 cases showed a delta in roll prediction of over 2 degrees.

This implies that the correlation between measured and modelled data is generally good across a wide range of test cases.
6. DISCUSSION

6.1 EFFECTS APPARENT FROM MODELLING

It was apparent from both the modelling and the testing that in none of the manoeuvres tested did the vehicle reach the rollover condition, however wheel lift was frequently observed in the tests with a bump.

In manoeuvres on flat ground, no more than 2 degrees of roll was experienced, save where such roll could be accounted for by a measurement offset. As expected, relative roll was observed between the front and rear portions of the vehicle, with the Kinglink™ joint allowing the portions of the vehicle to respond with a degree of independence to the applied manoeuvres. The maximum relative roll observed was of the order of 7 degrees, and hence the limiting condition for the Kinglink™ to lock, and the postulated “Mode 2” rollover to occur was not experimentally investigated.

Significantly more roll was generated during the bump tests, though the maximum roll experienced (of the front section of the vehicle) did not exceed 8 degrees. Significantly more roll was also experienced in the unloaded cases, and at the lower speeds. Indeed, the test driver reported that the 15 km/h test case was particularly uncomfortable, and this corresponds to the greatest predicted vehicle roll.

Unsurprisingly, the cornering bump tests generated higher vehicle roll angles than the equivalent straight-line bump tests. Indeed it appears that at the relatively small roll angles investigated here, it is possible to discern the total effect by superposition of the effects of the bump, and of the corner, implying that the system behaves linearly for small disturbances.

Where relative roll was generated in the vehicle, pitching was observed as a response. This is logical, since if the front of the vehicle rolls without the rear, then for one tyre to remain in contact with the ground, the other must lift and hence the height of the pivot point above the ground increases. With just three wheels in contact with flat ground there must be a pitching motion of the vehicle. For this reason, coupling between roll rate and pitch inertia was demonstrated in the model.

The pitch and roll inertias of the vehicle were not measured. They were estimated from the masses, inertias and locations of the constituent parts of the vehicle. Furthermore, the effect of varying the inertias of the vehicle was investigated, and it was found that the dynamic response of the vehicle is sensitive to the ratio of these parameters, affecting both the maximum pitch and roll experienced, and the frequency of the vehicle response to a bump input. However, it was not practicable to carry out a complete investigation within the scope of this project.

It was also observed that there is very little damping in the system, particularly in the loaded cases where a damped harmonic response in roll, pitch and z acceleration may be observed after impacting a bump. This can be seen most clearly in the lcl2_100mm_bump_straight2 case presented in Appendix C. Adding dampers to the wheels would theoretically improve this, however since the “spring” arises from the compliance of the tyres, wheels, and axles it is not immediately apparent how damping could be incorporated into the system, save by significantly altering the nature of the truck to include damping in axles or drive-train joints, or moving away from pneumatic tyres. These options could significantly impact on the cost, design and ease of
manufacture of the vehicles so a significant performance improvement would have to be demonstrated prior to this being recommended.

The simulation was found to be sensitive to the ratio of spring stiffness to damping coefficient. Indeed, this sensitivity was such that the model was adjusted to permit different spring stiffness’s to be applied to the front and rear of the vehicle. This mirrored the truck used in the MIRA trials, where the recommended tyre pressure differed significantly between the front and rear of the truck. Within the modelling, the spring stiffnesses were kept in the ratio of the recorded tyre pressures. Furthermore, a factor was applied to the measured static stiffness of the tyres to represent the dynamic stiffness.

6.2 EFFECTS EMERGING DURING THE TRIALS

During the trials, two practical effects were noticed, of relevance to the stability of the vehicles being assessed.

The trials benefited from a very experienced driver, with over 20 years experience as a test driver of heavy vehicles. The driver reported that the truck is both not particularly intuitive to drive, and moreover that the handling is somewhat unpredictable. This raises the issue of driver training to educate operators in the safe use of articulated site dumpers.

This was traced in part to backlash in the bucket-lock on the power-swivel dumper. The dumper supplied by Thwaites for testing was in good order, however even with the bucket locked, it could be swivelled a few degrees each way by hand. As the truck enters a corner, the inertia of the bucket causes it to swivel in its mounting until it hits the lock, whereupon the truck yaws away from its original path. When the tongue of the locking mechanism impacts the stop (see figure below), the swivelling of the bucket is arrested, transferring momentum into the front chassis of the truck. This gives rise to a momentary increase in acceleration in the y-direction for the front portion of the truck, and the vehicle responds by lurching sideways. This is not seen in the modelling, as discussed below.

Although the absolute amount of play in the bucket locking mechanism is small, its position close to the pivot point allows significant free rotation of the bucket on its pivot.
The effect is most pronounced with the vehicle fully loaded, when the bucket contains approximately 6 tonnes of load.

If such a relatively small movement of the bucket has a noticeable effect on vehicle handling then it can be implied that a loose load in the bucket would have a similar effect, if it slid in the bucket before suddenly impacting the side. However, tests were not conducted to validate this hypothesis, and the computer model would require modification to represent a shifting load.

As a result of the difficulty of driving the dumper, and the unexpected consequences of even slight movements of the bucket and load within it, it follows that the driver requires an understanding of the fundamental behaviour of the dumper, and its safe manner of operation, in order to operate the machine safely. The difficulty in driving the truck was highlighted during the steady state circling trials, where the truck would deviate from its path by up to 1m in response to undulations in the ground, even on a ‘flat’ tarmac surface.

The testing did not identify any fundamental instability of the vehicle, save that its handling was judged to be exceptionally poor compared to other road vehicles, and the inertias involved, even at relatively low speeds, are sufficient to affect the handling characteristics of the truck.

When conducting the trials the maximum roll difference between the front and rear portions of the truck was approximately 8°. In order to lock the King-link joint a relative roll of approximately 10.8° would have been required. Given the uncontrollability encountered during the trials it would have been unsafe to achieve a higher relative roll angle.

For these reasons, it would appear that the greatest contribution to prevention of rollover of articulated site dumper trucks would arise from increased driver training.

6.3 BEHAVIOUR ON A SLOPE

During the trials, the cornering, 100mm bump test produced the greatest vertical acceleration of the front of the vehicle, when conducted with the vehicle unloaded and at approximately 20 km/h.

Using this case, a preliminary investigation was undertaken regarding the effect of sloping ground. The manoeuvre was modelled on a range of slope angles, with all other parameters held constant. This suggested that the vehicle will roll over on a 30 degree slope, but will not do so on a 25 degree slope. This result is plausible, since the static roll angle of the truck, given the parameters entered into the model, is predicted to be 49.1 degrees. The impact with the bump causes the front portion of the truck to roll by over 13 degrees, hence the dynamic model predicts that 43 degrees of roll would be required to cause rollover in the case of the impact with a 100mm bump. The results of this case appear at Appendix E of this report.

It must be stressed that this result is merely indicative. The model has not been validated against test data on slopes of this magnitude, nor at roll angles of this magnitude. Indeed, during the course of this investigation it was noted that the animator in the Graphical User Interface does not correctly display the results at high slope angles.
6.4 SUGGESTIONS FOR FURTHER RESEARCH

In order to investigate the behaviour of articulated site dumper trucks at the rollover condition, it is suggested that a parametric study be conducted utilising the model to consider the effect of varying the following:

- Slope angles
- Bump sizes (height and profile)
- The effect of driving over a rut (i.e. a bump with negative height)
- Turn radii
- Speeds

And also investigate the effect of alterations to truck parameters such as:

- Track width
- Wheelbase
- Roll and pitch inertia
- Mass
- CoG Positions

This will allow the conclusions of this modelling work to be understood over a wide range of conditions, and it to be established whether the results are a peculiarity of the truck under test, or if they are more generally applicable across the population of vehicles, and a wider range of manoeuvres.

It was also noted in the model that at certain speeds, the impact of the rear wheel with the bump coincided with the front wheel touching down after passing the bump. This exaggerated the effect, and it is suggested that if the vehicle impacted multiple bumps, this could further contribute to the rollover condition. This could form the subject of further investigation.
7. CONCLUSIONS

Practical trials were conducted at MIRA using a Thwaites 6T Powerswivel dumper truck, typical of those in common use on UK construction sites. Trials were conducted over a range of manoeuvres, incorporating various bumps and vehicle speeds, and over a range of loads in the bucket of the vehicle. Key parameters were measured for each test case for comparison with the results of the same cases, modelled using the computer simulation developed by Frazer-Nash in phase 2 of this work.

The testing and modelling indicates that the main contributions to dumper truck rollover are likely to arise from:

- Uneven ground, including slopes and bumps.
- Low levels of damping present in the dumper trucks, since there is no explicit spring-damper arrangement.
- Dynamic response of the truck to bumps. In particular the frequency of response to bumps, which is controlled by the tyre stiffnesses, roll and pitch inertias of the vehicle, the track width and vehicle speed.

The practical trials undertaken at MIRA did not reach the rollover condition for the truck, since it was not practical to conduct such tests with a heavy vehicle operated by a human test driver. Nonetheless, when the test cases undertaken at MIRA were simulated in the computer model, a good correlation was observed between the measured and modelled results, particularly for those cases which are deemed to be most representative of aspects of the rollover condition, these being tests incorporating larger bumps, and steady-state circular tests.

From the modelling and testing completed, it is unlikely that the vehicle tested could be rolled over by impact with a bump of up to 100mm, on flat ground, at any radius of turn or speed within the operational envelope of the vehicle. However, it was verified that significant relative roll of up to 7 degrees can exist between the front and rear portions of the vehicle in response to even relatively slow impacts (~10 km/h) with relatively small (100mm) bumps. Were the vehicle on a significant slope, it is likely that this effect would increase to the point where the maximum relative roll allowed by the Kinglink™ is reached and momentum transfer from the front to the rear of the vehicle occurs. This would be characteristic of “mode 2” rollover as initially postulated.

The modelling predicts that the maximum roll and pitch experienced, and the roll and pitch rates of the vehicle are strongly dependant upon the pitch and roll inertias of the vehicle. It may be possible to determine values of pitch and roll inertia such that at speeds within the operational envelope of the vehicle, roll effects are damped by pitch inertia and vice versa. It is recommended that this is subject to further investigation, for a wide range of conditions for a range of slopes and bumps.

The modelling also predicts a strong correlation between the roll characteristics of the vehicle, and the ratio of spring stiffness to damping coefficient employed in the tyre model. If a practical method could be devised to manipulate the effective “spring stiffness” or damping coefficient of the system, this could allow roll stability of the vehicle to be controlled in the vehicle design.
Further work is recommended to conduct a parametric investigation of the key design parameters of articulated dumper trucks, over a range of manoeuvres likely to cause rollover. This work would be targeted to identify possible technical modifications to the design of articulated dumper trucks to make them less susceptible to rollover.

Furthermore it is recommended that efforts are made to improve the quality of driver training.
8. REFERENCES

1. Shen Tyre Model
   “The Dynamics of Vehicles on Roads and On Tracks.” ISSN 0042-3114
   Zhiyun Shen
APPENDICES

APPENDIX A
MIRA TEST SUMMARY REPORT

APPENDIX B
RESULTS OF STEADY-STATE CIRCULAR MANOEUVRES

APPENDIX C
RESULTS OF STRAIGHT LINE, 100MM BUMP MANOEUVRES

APPENDIX D
RESULTS OF CORNERING, 100MM BUMP MANOEUVRES

APPENDIX E
RESULTS OF CORNERING, 100MM BUMP MANOEUVRE, 30 DEGREE SLOPE, 20 KM/H
## Test Objective/Method/Specification No:

The objective of the test was to collect handling data on a 6 tonne capacity dumper truck whilst carrying out the four procedures described below. The data acquired will enable Frazer-Nash to validate a simulation model. Digital photographs and videos were taken of the vehicle during all of the tests.

### 1. Steady state circular test

The vehicle was driven in a circle in an anti-clockwise direction with the hand wheel held at a constant angle. As there were some undulations on the test surface which caused variation in the roll and pitch angles of the vehicle, a control lap was conducted at a constant speed of less than 10 km/h. This control lap was used to determine the effect of the surface on the vehicle. Once the first lap was completed the vehicle’s speed was gradually increased up to the maximum permissible speed for the steering angle. This test method was repeated at three different steering angles, which were full lock and approximately 8 and 12 degrees.

### 2. J Turn

The vehicle was driven in a straight line at a constant speed and then a negative steering input was applied (anti-clockwise turn). The test was performed at four different speeds with the same steering angle applied to provide steady state lateral accelerations from approximately 0.075 'g' up to the maximum achievable lateral acceleration.

### 3. Driving over bump test

The vehicle was driven at a constant speed over a bump, such that the bump only passed under the left hand side wheels. Two different bumps were used for the manoeuvres, which were 50mm and 100mm high and both had a trapezoidal profile, see Appendix A for dimensions. The vehicle was driven over the bumps in a straight line and also on a circular path. For the circular path the vehicle was driven in a straight line until it was within a close proximity of the bump and then the steering angle was applied, similar to a j-turn manoeuvre, and the bump passed under the inside wheels. The procedure was performed at up to 4 different speeds which were approximately 10, 15, 20 and 25km/h (when possible).

### 4. Single Lane Change

The vehicle was driven through a set of coned boxes arranged so that the vehicle moved from its original lane across to a lane on its left hand side. The dimensions of the boxes were based on those described in ISO 3888-2:2002, however, the distance between the boxes was modified to accommodate the speed and dimensions of the vehicle. The dimensions of the lanes are shown in Appendix A. The procedure was performed at up 4 different speeds which were approximately 10, 15, 20 and 25 km/h (when possible).
Load Cases
All of the test procedures were performed at three different load cases, which are detailed in Table 1.

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Front Axle Load (kg)</th>
<th>Rear Axle Load (kg)</th>
<th>Total (kg)</th>
</tr>
</thead>
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<tr>
<td></td>
<td>Left Wheel Right Wheel</td>
<td>Left Wheel Right Wheel</td>
<td></td>
</tr>
<tr>
<td>1*</td>
<td>1130 1190</td>
<td>1450 1450</td>
<td>5100</td>
</tr>
<tr>
<td>2</td>
<td>2200 2200</td>
<td>1500 1500</td>
<td>7400</td>
</tr>
<tr>
<td>3</td>
<td>3700 3700</td>
<td>1500 1500</td>
<td>10400</td>
</tr>
</tbody>
</table>

* For load case 1 there was no ballast located in the bucket but the stability bars and equipment increased the vehicle's weight above its kerb weight.

Table 1 - Wheel loads for each load case

Ballast was placed in the vehicle's bucket, which for the second load case comprised of clay and stone. For the third load case steel blocks were also used in addition to the clay and stone so that the required mass could be achieved without the level of the ballast being too high and unstable. The fill depth of the bucket for each load case and the location of the steel blocks are described in Appendix B.

Tyre Deflection Measurements
At each of the load case and kerb weight conditions, the wheel centre heights and loads were measured to allow Frazer-Nash to make an estimation of the static tyre stiffnesses.

Specimen Description/Part No(s):

Vehicle
Model: Thwaites Alldrive 6 tonne capacity power swivel dumper truck
VIN Number: SLCM466ZZ507A7566
Registration Number: BV55 UJW

Tyres
Model: Mich MPT-01 405/70-20
Front pressure: 3.9 Bar
Rear pressure: 2.5 Bar

Test Equipment:
To prevent the vehicle from rolling over during the tests, two stability bars were fitted, one at the front and one at the rear of the vehicle. Photographs and descriptions of the bars and fixtures are provided in Appendix C.

The vehicle was instrumented to measure the parameters listed in Table 2 throughout all of the tests at a sampling frequency of 100Hz. The instrumentation details and locations are described in Appendix D.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Channel</th>
<th>Units</th>
<th>Equipment</th>
<th>QA Number</th>
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</thead>
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<td>'g'</td>
<td>'g' accelerometer</td>
<td>28509</td>
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<tr>
<td>Front lateral acceleration</td>
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<td>'g'</td>
<td>'g' accelerometer</td>
<td>28508</td>
</tr>
<tr>
<td>Front vertical acceleration</td>
<td>4</td>
<td>'g'</td>
<td>'g' accelerometer</td>
<td>28510</td>
</tr>
<tr>
<td>Rear longitudinal acceleration</td>
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<td>'g'</td>
<td>'g' accelerometer</td>
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</tr>
<tr>
<td>Rear lateral acceleration</td>
<td>6</td>
<td>'g'</td>
<td>'g' accelerometer</td>
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<td>'g'</td>
<td>'g' accelerometer</td>
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</tr>
<tr>
<td>Rear yaw angle</td>
<td>8</td>
<td>Degrees</td>
<td>Inertial and GPS Navigation System (RT3000)</td>
<td>29193</td>
</tr>
<tr>
<td>Rear pitch angle</td>
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<td>Degrees</td>
<td></td>
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<td>Rear yaw rate</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Rear roll rate</td>
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<td>Degrees/sec</td>
<td></td>
<td></td>
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<td>Articulation angle</td>
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<td>380mm wire potentiometers</td>
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<tr>
<td>Roll angle difference between</td>
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<td>Degrees</td>
<td>Spring loaded linear potentiometer</td>
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<td>front and rear</td>
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<tr>
<td>Speed</td>
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<td>Inertial and GPS Navigation System (RT3000)</td>
<td>28193</td>
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</table>

Table 2 - Instrumentation details
Results:

The individual data sets for all of the tests were converted to text files (.csv format) and supplied to Frazer-Nash. A list of all of the files and their nomenclature are listed in Appendix E. The first channel in the data was 'Time' in seconds and the remaining channels and associated units correspond to those shown in Table 2.

Photographs of the vehicle undertaking some of the tests are shown in Figures 1 to 3. All photographs and video footage of the tests have been supplied to Frazer-Nash.

![Vehicle undertaking steady state cornering test](image1)

*Figure 1 - Vehicle undertaking steady state cornering test*

![Vehicle undertaking 100mm bump test](image2)

*Figure 2 - Vehicle undertaking 100mm bump test*
Figure 3 - Vehicle undertaking single lane change test

Tyre deflection measurements
The individual wheel loads and wheel centre (WC) heights measured are shown in Table 3.

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Front</th>
<th>Rear</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Left</td>
<td>Right</td>
</tr>
<tr>
<td></td>
<td>Load (kg)</td>
<td>WC height (mm)</td>
</tr>
<tr>
<td>Kerb</td>
<td>900</td>
<td>519</td>
</tr>
<tr>
<td>1</td>
<td>1100</td>
<td>517</td>
</tr>
<tr>
<td>2</td>
<td>2200</td>
<td>500</td>
</tr>
<tr>
<td>3</td>
<td>3700</td>
<td>477</td>
</tr>
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</table>

Table 3 - Tyre deflection measurements

Attachments/Notes:
- Appendix A: Test Details
- Appendix B: Details of Load Cases 2 and 3
- Appendix C: Stability Bars
- Appendix D: Instrumentation Location
- Appendix E: Data File Nomenclature

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<tr>
<td>Prepared By</td>
<td>Michael Naylor</td>
<td>Engineer, Vehicle Dynamics and NVH</td>
<td>27\textsuperscript{th} July 2007</td>
</tr>
<tr>
<td>Concurred By</td>
<td>Jon Parsons</td>
<td>Manager, Vehicle Dynamics and NVH</td>
<td>31\textsuperscript{st} July 2007</td>
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</table>
Appendix A

Test Details
Bump Profiles

![Bump Profile Diagram]

<table>
<thead>
<tr>
<th>Height, h (mm)</th>
<th>Top width, a (mm)</th>
<th>Base width, b (mm)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>44</td>
<td>84</td>
</tr>
<tr>
<td>100</td>
<td>60</td>
<td>132</td>
</tr>
</tbody>
</table>

Figure 4 - Bump profiles

Single Lane Change Dimensions

![Single Lane Change Diagram]

<table>
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<tr>
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</tr>
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<td>2.84</td>
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<td>2</td>
<td>18</td>
<td>6.19</td>
</tr>
<tr>
<td>3</td>
<td>6</td>
<td>3.36</td>
</tr>
</tbody>
</table>

Figure 5 - Single lane change dimensions
Appendix B

Details of Load Cases 2 and 3
All dimensions in this appendix are in mm, unless stated otherwise, and are quoted in the order x (longitudinal), y (lateral), z (vertical).

**Load Case 2**

The average distance below the horizontal side of the bucket to the level of the load was 215mm, as shown in Figure 6.

![Figure 6 - Location of average fill height for load case 2](image)

**Load Case 3**

The average distance above the horizontal side of the bucket to the level of the load was 170mm, as shown in Figure 7.

![Figure 7 - Location of average fill height for load case 3](image)
Steel blocks were also used to enable the required load to be placed in the bucket without the level of the clay and stone being too high and unstable, see Figures 8 and 9.

![Image of steel blocks used for load case 3]

**Figure 8 - Photograph of steel blocks used for load case 3**

<table>
<thead>
<tr>
<th>Block</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Distance below horizontal side of bucket to top of block (mm)</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>610</td>
<td>670</td>
<td>205</td>
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<td>500</td>
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<tr>
<td>B</td>
<td>670</td>
<td>610</td>
<td>205</td>
<td>110</td>
<td>500</td>
</tr>
<tr>
<td>C</td>
<td>670</td>
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<td>110</td>
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</tr>
<tr>
<td>D</td>
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**Figure 9 - Location of steel blocks used for load case 3**
Appendix C

Stability Bars
All dimensions in this appendix are in mm, unless stated otherwise, and are quoted in the order x (longitudinal), y (lateral), z (vertical).

Front Stability Bar and Attachments

Figure 10 - Photograph of front stability bar

Front Stability Bar Attachment Plate

Mass: 115kg (including all additional webbings and plates)

Dimensions: 455 x 1360 x 25mm

Location: The plate was attached underneath the body and protruded towards the forwards. The distance between the front of the plate and the front of the body was 336mm, see Figure 11.

Figure 11 - Location of front stability bar plate
**Front Stability Bar**

**Mass:** 180kg

**Dimensions:** 100 x 4000 x 200mm (10mm wall thickness)

**Location:**
- Distance from ground to underside of bar: 400mm
- Distance from rear edge of bar to front wheel centre: 620mm

![Diagram of Front Stability Bar](image)

**Figure 12 - Location of front stability bar**

**Feet and Sleeves**

**Mass:** 17kg (1 foot and 1 sleeve)

**Location:** One foot and sleeve located at each end of the stability bar.

**Rear Stability Bar**

![Photograph of Rear Stability Bar](image)

**Figure 13 - Photograph of rear stability bar**
Mass: 103kg

Cross sectional dimensions: 100 x 3560 x 150mm (8mm wall thickness)

Location:
Distance from ground to underside of bar 400mm
Distance from rear wheel centre to front edge of bar 790mm

Feet and Sleeves

Mass: 17kg (1 foot and 1 sleeve)

Location: One foot and sleeve located at each end of the stability bar.
Appendix D

Instrumentation Location
All dimensions in this appendix are in mm, unless stated otherwise, and are quoted in the order x (longitudinal), y (lateral), z (vertical).

Front Accelerometers

![Plate and Accelerometers](image)

**Figure 15 - Photograph of front accelerometers attached to vehicle**

The front accelerometers were attached to a vertical plate which was located forwards and above the transfer box. The top edge of this plate was 1015mm above the ground and the rear face was 469mm forwards of the articulation axis, as shown in Figure 16.

![Diagram](image)

**Figure 16 - Location of plate which front accelerometers were attached**
The locations of the front accelerometers in relation to the plate which they were attached are shown in Figure 17. The centres of the x and z axes accelerometers were aligned with the longitudinal centre line of the vehicle.

**Figure 17 - Location of front accelerometers**

**Rear Instrumentation**

**Figure 18 - Photograph showing instrumentation attached to rear of vehicle**
All of the rear instrumentation was located on top of the engine cover behind the driver's seat as shown in Figure 18. The rear accelerations, angles and rates were recorded by the RT3000 inertial and GPS navigation system at its geometric centre. The location of the RT3000, and the battery and Astech data acquisition system (DAS), are shown in Figure 19. All of this equipment was orientated so that the longitudinal centre lines corresponded with the longitudinal centre line of the vehicle. The masses of the other pieces of equipment were deemed negligible and have, therefore, not been included.
Appendix E

Data File Nomenclature
## Load Case 1

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APPENDIX B
RESULTS OF STEADY-STATE CIRCULAR MANOEUVRES
APPENDIX C
RESULTS OF STRAIGHT LINE, 100MM BUMP MANOEUVRES
APPENDIX D
RESULTS OF CORNERING, 100MM BUMP MANOEUVRES
APPENDIX E
RESULTS OF CORNERING, 100MM BUMP MANOEUVRE
30 DEGREE SLOPE, 20 KM/H
Articulated dumper truck rollover

This report details a programme of practical testing and computer simulation of the roll characteristics of articulated site dumper trucks.

Trials were undertaken at MIRA using a Thwaites 6-Tonne, Power Swivel dumper truck to conduct a range of manoeuvres, at a range of speeds and carrying a range of loads. Each experimental case was modelled in a computer simulation. The modelled and measured results showed good correlation, in particular for manoeuvres incorporating larger bumps (100mm in height) and for circling manoeuvres.

From the measuring and modelling undertaken, it appears that the key contributions to rollover occur when working on uneven ground. Dumper trucks feature very low levels of damping since their only ‘suspension’ is by deformation of the tyres and drive train. On uneven ground, this can produce significant roll angles in response to impact with a bump and this effect can be exaggerated if the natural frequency of the response coincides with another wheel of the truck hitting the same bump.

An initial investigation of the response of dumper trucks impacting a bump when operating on a sloping plane was conducted in the computer model. This indicated that impacting a bump can cause the vehicle to roll over on slopes significantly shallower than the static roll angle of the truck.

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