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# Using Modelling and Simulation to Predict Dynamics of Converted Ground Vehicle

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**ABSTRACT**

In order to redesign and convert the passenger ground vehicle Land Rover defender 110 into military vehicle for different surveillance and reconnaissance missions it is necessary, prior to equipment integration, to assess its future dynamic response. For this purpose, the 19-degree of freedom multibody simulation model of the defender was developed using the software package MSC.ADAMS/Car. The simulation model was validated using the instrumented experimental vehicle for two scenarios namely bump test and double lane change manoeuver. Comparison of numerical predictions suggests reasonably good agreement with the actual vehicle responses. The validated model was then used to assess the effect of longitudinal and vertical position of the added equipment on the responses of the upgraded vehicle. Lateral stability degradation due to the added equipment was also investigated defining the rollover threshold as an objective assessment criterion. The obtained results show considerable lateral stability degradation for both marching and operating heights of the added equipment.

**Keywords:** Computer modelling, model validation, field testing, vehicle dynamic, vehicle conversion

## 1. INTRODUCTION

Simulation and computer modelling have transformed the unidirectional process of vehicle design and development into a next level strategy which allows engineers to reproduce manoeuvers and tests on virtual models to assess the dynamic behaviour of vehicles at different design levels (complete redesign, derivative design, variant design, model update, etc.)[1]. Nevertheless, vehicle simulations are intended to reduce the cost and the duration of the vehicle development process and help identifying errors and deficiencies at early stages of the design process. Ride comfort and handling properties are one of the major key features to be investigated using vehicle simulation models. Ride comfort is expressed as the level of discomfort experienced by the passenger in terms of frequency and amplitudes of mostly vertical oscillations induced by road geometry and engine vibration. Meanwhile, handling properties are related to the response and the stability of the vehicle to driver and environmental inputs such as gust, wind and road disturbances[2,3]. Literature suggests that the requirements for vehicle simulation models should be in accordance with the considered dynamic characteristics. Thus, the simulation model should be kept as simple as possible, but good enough to accurately represent the dynamic behaviour to be investigated[4]. Depending on the application, many scientific papers investigate one aspect of vehicle dynamics. In[5], the author has suggested a method to identify lateral tire forces using a simple vehicle model and can be applied to the analysis of vehicle handling performance. Pazooki[6] has developed a comprehensive off-road vehicle model for ride analysis using a 3D tire-terrain interaction model. The author has investigated both suspended and unsuspended vehicle model responses arising from road roughness profile. A high resolution computer based simulation model was developed by Leatherwood[7] which aims to emulate the ride and handling performance of a ground military vehicle.

Even though most vehicle dynamics can be simulated using computer based models, the conceptual simulation model should be validated against experimental data vectors obtained by performing tests on the real vehicle. The documented literature in the field of vehicle model validation highlights different applications of this process depending on the general purpose and the author's individual understanding of the subject. According to Carson[8], the process of verification and validation aims to build a virtual model which is sufficiently accurate to reproduce the behaviour and the performance of the physical system that it represents. Kleijnen[9] emphasizes the use of statistical techniques and how this tool can improve the quality of the simulation using objective and quantitative assessment criteria. In another source concerning FEM application for suspension analysis, verification and validation are used to check out whether assumptions and simplifications within the model are as accurate as they should be[10]. Leatherwood[7] intensively worked on the development of a high-resolution simulation model by including single component characteristics measured directly on the vehicle under study. In a literature survey presented by Kutluay and Winner[11], the authors have stated that universal methodology for vehicle model validation does not exist. It depends on the

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field of application and the dynamic behaviour to be assessed. In most cases, the validation procedure is highly related to the developer's visual judgement and to the individual appreciation of the validation data.

### 1.1 Background and Motivation

Modern threats and challenges require prompt reaction to face the changing character of modern warfare and the use of new technologies. Thus a respectful army should quickly and efficiently equip its military units with appropriate equipment. Here comes the advantage of developing military mobile solutions by converting existing vehicles. Furthermore, army can achieve significant savings in terms of maintenance when using one platform for different purposes. Through the last 20 years it has been noticed an increasing need to outfit land forces with lightweight ground vehicle for surveillance and reconnaissance of battlefield, borders and military facilities. While leading military industries develop new concepts of these vehicles, others upgrade an existing vehicle by integrating the required equipment. The most common equipment for the above mentioned vehicles consists of combination of different sensors, transducers and telecommunication devices such as TV and thermal cameras, ground surveillance radars, laser devices, acoustic sensors and radio stations, all integrated in one block mounted on a telescopic mast. The analysis of the integration possibility of this equipment on wheeled vehicle should answer questions about arrangement, positioning, additional mass, mechanical stress of vehicle structure, dynamic stability, ride and manoeuvrability.

## 2. FULL VEHICLE SIMULATION MODEL

The aim of this work is to build and validate a multibody model of the wheeled road vehicle Land Rover Defender 110 through series of carefully selected ride and handling tests. The principal technical characteristics of the vehicle which are relevant to the model, as well as moments of inertia about the CoG and the location of the CoG[12], are presented in Table 1.

**Table 1. Principal technical characteristics of the vehicle to be modelled.**

| Model | Land Rover Defender 110 |
|---|---|
| Wheel base | 2794 mm |
| Track width | 1486 mm |
| Unloaded weight | 2125 kg |
| Front suspension | Live beam axle |
| Rear suspension | Live beam axle |
| Axle ground clearance | 250 mm |
| Tire size | 235/85R16 |
| Roll moment of inertia | 744 kg.m$^2$ |
| Pitch moment of inertia | 2440 kg.m$^2$ |
| Yaw moment of inertia | 2478 kg.m$^2$ |
| CoG vertical position (from ground) | 1000 mm |
| CoG longitudinal position (from front axle) | 1400 mm |

### 2.1 Component Parameters Identification

To quantify all the required parameters for the multibody simulation model, it was necessary to carry out a number of measurements on the target vehicle. The suspension topology was determined by estimating the coordinates of the most representative bushing centres of the front and rear suspensions with reference to a coordinate system located at the mid contact point between the wheel and the ground. The dimensions of the linkages are measured and drawn into ADAMS/Car to estimate their mass as well as their inertia properties. Individual spring stiffness characteristics were measured using a displacement sensor attached to an axial force transducer to obtain the corresponding force-displacement curves. Compression speed was kept small enough to consider quasi-static loadings and the hysteresis effect was neglected. Front and rear anti-roll bars were tested in the same way to obtain the correspondent force-angle curves. Shock absorber force-velocity curves were determined using a hydraulic press. Transmission ratio was calculated using a displacement sensor to measure wheel deviation along with an angle transducer to record steering wheel angle.

### 2.2 Suspension System Configuration

The Defender's front suspension system consists of a live rigid axle connected to vehicle body by means of two radius arms to provide longitudinal guidance of the axle and to react to longitudinal forces. A Panhard rod attaches the axle to the vehicle body to prevent lateral axle displacement. This configuration allows the axle to move only up and down along z-axis, as well as rotation about the x-axis. A pair of coaxial coil spring and shock absorber are mounted vertically on each end of the axle. The rear suspension has the same configuration. However, the Panhard rod is replaced with a triangular linkage. This combination forms Robert's mechanism which provides lateral axle positioning and makes the connection point between the axle and the linkage (coupler point) to move on a straight line vertically. Springs are mounted vertically at each end of the axle, while the shock absorbers are tilted slightly at an angle about y-axis to introduce some longitudinal damping. Both front and rear suspension systems are fitted with anti-roll bars to help reduce body roll when the vehicle corners.

### 2.3 Model Implementation

In order to adapt the complexity of the model to the behaviour being assessed, a number of assumptions and simplifications were adapted and used within the model, those are:

(i) The vehicle chassis is modelled as a rigid body;
(ii) Default values are used for the aerodynamic properties of the vehicle body;
(iii) Default properties are used for all bushings within the model;
(iv) Suspension components are considered to be rigid bodies;
(v) Because of lack of technical documentation, the steering subsystem was modelled using a predefined MSC. ADAMS template.

The Land Rover Defender is modelled as a multibody dynamic system using MSC.ADAMS/Car, comprising beam axle front and rear suspension, steering subsystem, front and rear wheels and vehicle chassis. The primary purpose of this model is to simulate the ride and handling behaviour of the vehicle. Therefore, having accurate representation of the brake system and the power train is not meaningful, apart from considering the effect of their mass properties on the vehicle behaviour. A non-linear MSC.ADAMS Pacejka 2002 tire model[13] was used and fitted with experimental data to consider vertical dynamic of the tire under normal loads. The front suspension is modelled as a rigid axle attached to the vehicle body by means of two longitudinal rods connected to the chassis by hook joints at the rear end and to the axle by spherical joint at the front end. The Panhard rod is modelled as a lateral rod attached to the chassis by hook joint and to the axle by spherical joint. The steering system is modelled as a rack pinion system using a predefined MSC.ADAMS/CAR template and fitted with measured steering ratio. Each end of the rack is connected to the tie rod by constant velocity joint (convel joint). The tie rod is connected to the wheel hub through the steering arm by a spherical joint. The anti-roll bar is modelled as two separate bars attached to the axle by spherical joints at the lower ends and supported by bushings fixed to the vehicle chassis. The other ends are connected to each other by a torsional spring which applies moment around the axis joining the two bars. This moment is function of the angular displacement of the two bars around the aforementioned axis. The rear suspension is modelled in a similar way. However, the Panhard rod was replaced by a flexible square section beam to be representative of the triangular linkage on the real vehicle. The beam is attached at the rear end to the middle of the axle by a spherical joint while the front end is fixed to the chassis. Coil springs and dampers are modelled with nonlinear curves using measured experimental data. The suspension bushings are modelled as three-dimensional spring damper allowing transmission of forces and moments. These forces and moments are calculated using the specified translational and rotational stiffness and damping found in the bushing property files. The 3D models of front and rear suspension and the steering system as well as their kinematic schemes are shown in Figs. 1 and 2. The full model has 19 degrees of freedom, 42 moving parts, 18 spherical joints, 12 revolute joints, 9 Hooke joints, 3 translational joints, 2 convel joints, 1 cylindrical joint, 8 fixed joints and one motion defined by the rotation of the steering wheel.

## 3. FIELD MEASUREMENTS

The ADAMS model was validated against measured test data of the instrumented Land Rover Defender 110 negotiating a discrete bump and performing a double lane change manoeuver according to STANAG-4357[14]. The tests were carried out on a dry airport pavement. In order to record the required signals, different sensors and transducers were installed on the vehicle (Fig. 3). The triaxial navigational sensor (TANS), which gives

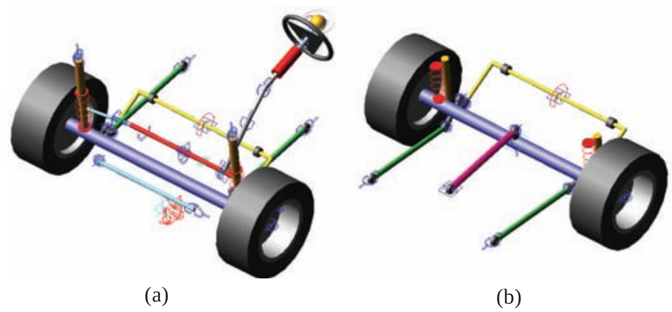


**Figure 1. MSC.ADAMS/Car full vehicle model. (a) front suspension and steering and (b) rear suspension.**

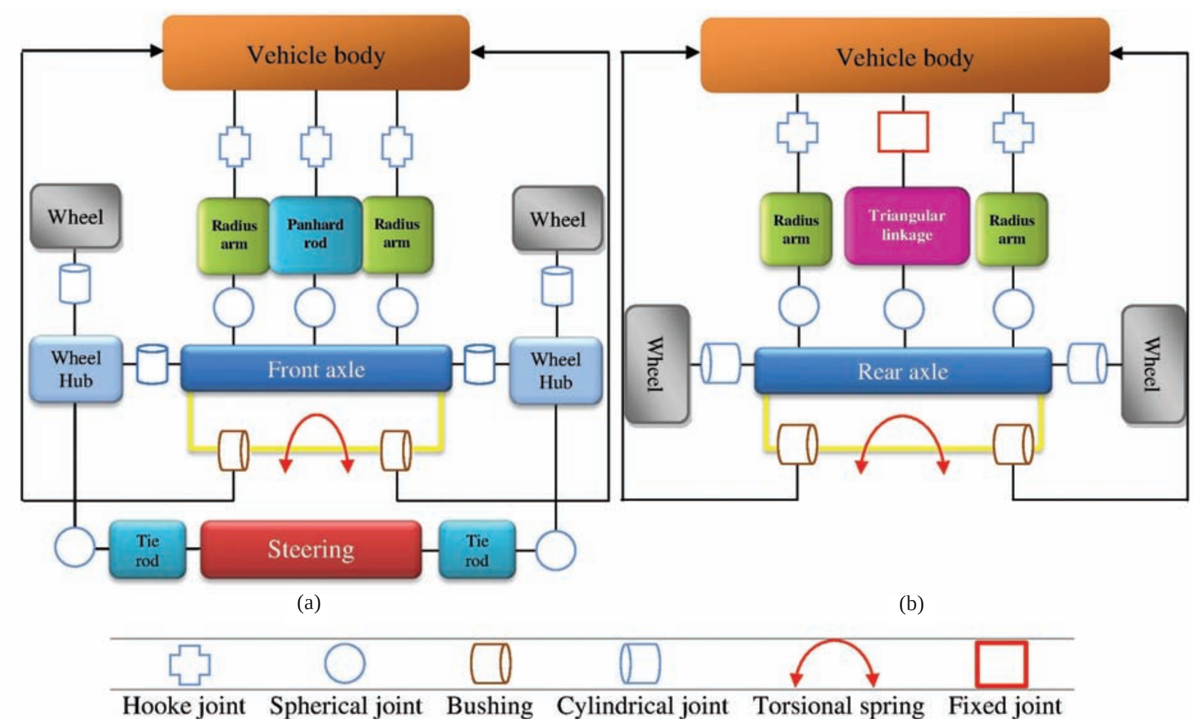


**Figure 2. Kinematic schemes of the suspension systems: (a) front suspension and (b) rear suspension.**

simultaneous measurement in up to 6 axes (3-axes gyro and 3-axes accelerometer) was installed near the vehicle's CoG. Three HF laser height sensors were installed on the vehicle body at three different locations to measure pitch and roll angles by simple trigonometric calculation. The optical sensor S-350 was mounted at the front bumper and used to measure vehicle longitudinal and lateral velocity. Inductive displacement transducers WA100 and WA200 were mounted at the left front and rear springs to measure relative displacement chassis/axle. A displacement sensor was fixed to the steering rod to record steering wheel displacement. The data were recorded at 50 Hz sampling rate. Sensor positions and measurement parameters are defined in Table 2.

## 4. MODEL VALIDATION

The MSC.ADAMS model was validated by comparing the relevant parameters of the real vehicle negotiating a bump obstacle at 20 km/h, 30 km/h, and 40 km/h and performing a double lane change manoeuever at 50 km/h, 60 km/h, and 70 km/h with those obtained by simulation. Figure 4 represents the correlation obtained between the simulation and the experimental results for the vehicle crossing a bump obstacle at 30 km/h. It is evident that a satisfactory correlation was achieved in terms of shape and amplitude values, especially for vertical acceleration which is a direct measure of ride comfort. Calculated average mean root square deviations of front and rear spring displacements, pitch angle, pitch velocity

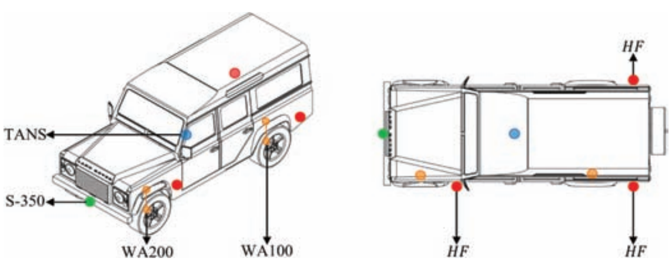


**Figure 3. Vehicle measurement positions.**

**Table 2. Measurement parameters and sensor positions**

| Sensor | Position | Channel | Measure | Symbol |
|---|---|---|---|---|
| $HF_1$ | left front chassis | 1 | roll angle<br>pitch angle | $\theta_x$<br>$\theta_y$ |
| $HF_2$ | right rear chassis | 1 | | |
| $HF_3$ | left rear chassis | 1 | | |
| TANS | CoG | 2 | longitudinal acceleration | $a_x$ |
| | | 3 | lateral acceleration | $a_y$ |
| | | 4 | vertical acceleration | $a_z$ |
| | | 5 | roll velocity | $w_{xz}$ |
| | | 6 | pitch velocity | $w_y$ |
| | | 7 | yaw velocity | $w_z$ |
| S-350 | mid front chassis | 8 | longitudinal velocity | $v_L$ |
| | | 9 | lateral velocity | $V_Q$ |
| WA200 | left front spring | 10 | displacement | $d_1$ |
| WA100 | left rear spring | 11 | displacement | $d_1$ |
| SA | steering rod | 12 | steering angle | α |

and vertical acceleration suggest good agreement between the model responses and the measured vehicle parameters for ride comfort (Table 3).

**Table 3. Comparison of calculated RMS values between simulation and measured parameters for bump test at 30 km/h**

| Parameter | $d_1$ (mm) | $d_2$ (mm) | $\theta_y$ (deg) | $\omega_y$ (deg/s) | $a_z$ (g) |
|---|---|---|---|---|---|
| Model | 5.25 | 6.29 | 0.17 | 1.86 | 0.094 |
| Measured | 6.13 | 5.88 | 0.18 | 1.72 | 0.084 |
| Deviation (%) | 14.3 | 6.97 | 5.55 | 8.14 | 11.9 |

Figure 5 indicates the correlation for the vehicle performing a double lane change manoeuvre at 70 km/h. The simulation model truly predicts trends and amplitudes of the observed parameters. For double lane change, simulated roll velocity and lateral acceleration, which are significant for handling assessment, correlate good enough with those obtained from field measurements (Table 4). However, a significant time history deviation was observed which is explained by the difference in driver steer input. Because there is no steering control during the field test, the obtained results are highly dependent on the driver skills while the MSC.ADAMS/Car driver model steers the vehicle so that it follows very closely the mid line of the test track. The simulation model is then considered validated and can be used for analysis of the upgraded vehicle.

**Table 4. Comparison of calculated RMS values between simulation and measured parameters for double lane change manoeuver at 70 km/h**

| Parameter | $\theta_x$ (deg) | $\omega_x$ (deg/s) | $\omega_z$ (deg/s) | $a_y$ (g) | $d_1$ (mm) | $d_2$ (mm) |
|---|---|---|---|---|---|---|
| Model | 1.11 | 2.54 | 6.33 | 0.22 | 9.07 | 9.48 |
| Measured | 0.96 | 2.85 | 5.68 | 0.2 | 8.54 | 8.89 |
| Deviation (%) | 15.6 | 10.88 | 11.44 | 10 | 6.21 | 6.63 |

## 5. DYNAMICS OF THE UPGRADED VEHICLE

After the considered model had been validated, series of simulation were undertaken to investigate the effect of the added equipment on the dynamic behaviour of the vehicle. The simulation model was modified so it includes mass and inertia parameters of the equipment chosen by the sponsoring institution. The mass of the equipment is estimated to be 350 kg, so the total mass of the vehicle becomes 2475 kg. The equipment was modelled as rigid body connected to the vehicle chassis by a fixed joint. Three different layouts of the vehicle will be considered. The baseline configuration is related to the original vehicle. The folded configurationis related to the upgraded vehicle with the equipment mounted at the CoG of the baseline vehicle and its vertical coordinate set to the marching height (595 mm from the baseline vehicle CoG). The extended configuration is the same as the second but the equipment is mounted at the operating height (1900 mm from the baseline vehicle CoG).

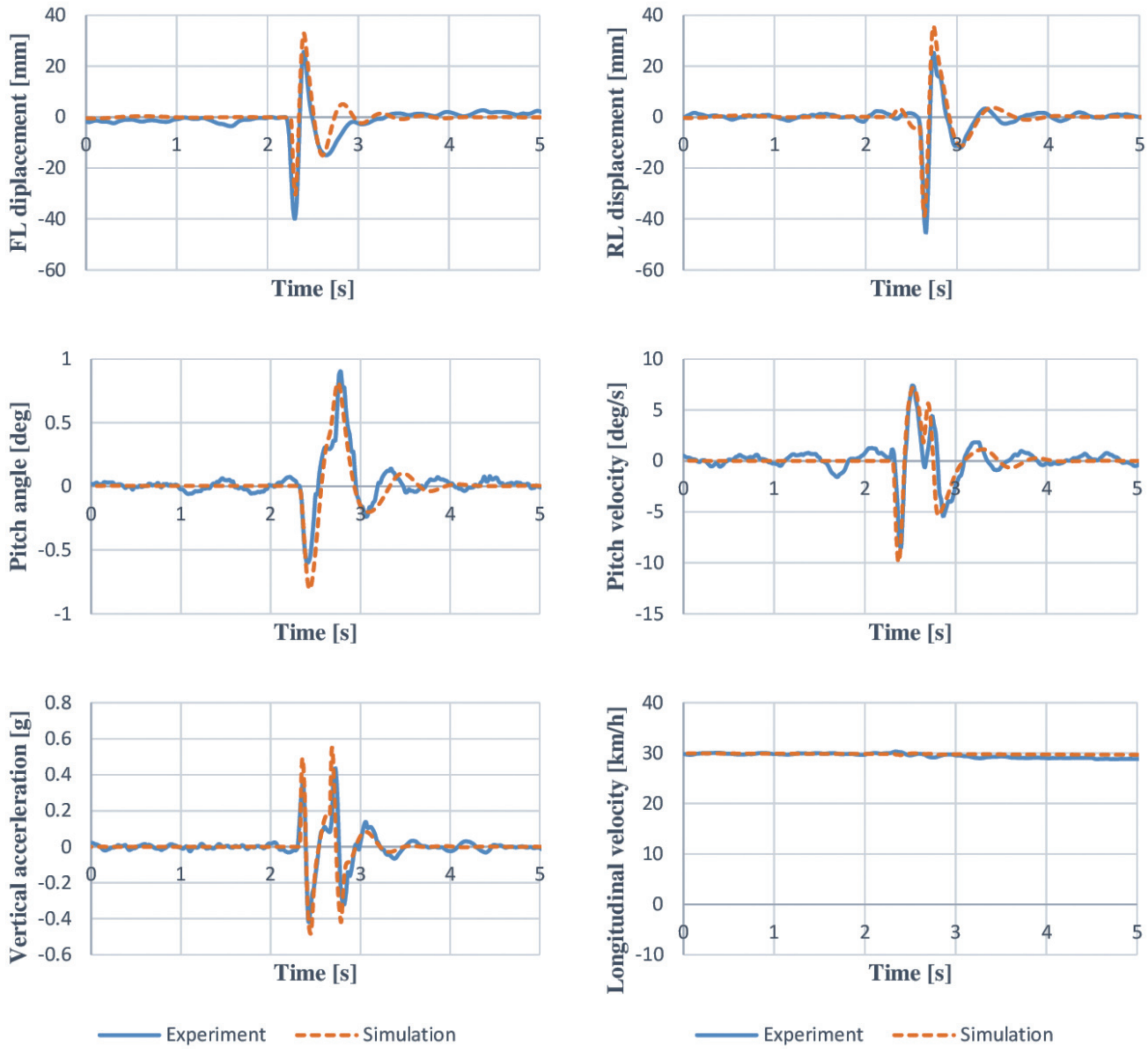


**Figure 4. Simulation and experimental results for a bump test at 30 km/h.**

Firstly, the longitudinal position of the integrated equipment CoG was varied from -600 mm to +400 mm about the vehicle CoG and the vertical position was set to the marching position. The analyses are performed for the vehicle negotiating a discrete obstacle and performing a double lane change. The obtained responses are expressed in term of vertical RMS acceleration and maximum roll angle as functions of the distance of the CoG, for bump test and double lane change test respectively (Fig. 6). The results show minimal effect of the CoG longitudinal coordinate on the vertical RMS acceleration with about 3.5 per cent RMS variation through the considered distance range. However, vertical acceleration decreases as the added mass is shifted forward, which is explained by the fact the front suspension hasslightly higher damping coefficient. Similarly, the effect longitudinal positioning of the added equipment on maximum roll angle is almost negligible with less than 1 per cent variation through the considered distance range.

Secondly, series of simulation were performed to assess to effect of the added equipment on the ride and handling behaviour of the vehicle. The results are expressed in term of RMS vertical acceleration and maximum roll angle for the vehicle model configurations performing bump test at 30 km/h and double lane change manoeuver at 50 km/h, respectively (Table 5). As it was expected, the results show that the added equipment yields considerably lower acceleration responses (22 per cent) compared to the baseline vehicle when negotiating the bump obstacle. However, the model exhibits almost insignificant sensitivity to variation of the equipment vertical position. On the other hand, the simulated roll angle of the folded and the extended configuration are nearly 50 per cent and 110 per cent greater than that of the baseline vehicle.

Lastly, a ramp steer event was simulated to determine rollover threshold of the vehicle when the front wheels are subject to ramp steer function of 2 deg/s. Figure 7(a) represents the evolution of the chassis lateral acceleration as function of

**Table 5. Influence of the added equipment on the RMS vertical acceleration and maximum roll angle.**

| Parameter | RMS vertical acceleration [g] | Maximum roll angle [deg] |
|---|---|---|
| Baseline vehicle | 0.094 | 1.59 |
| Folded configuration | 0.073 | 2.38 |
| Extended configuration | 0.074 | 3.37 |

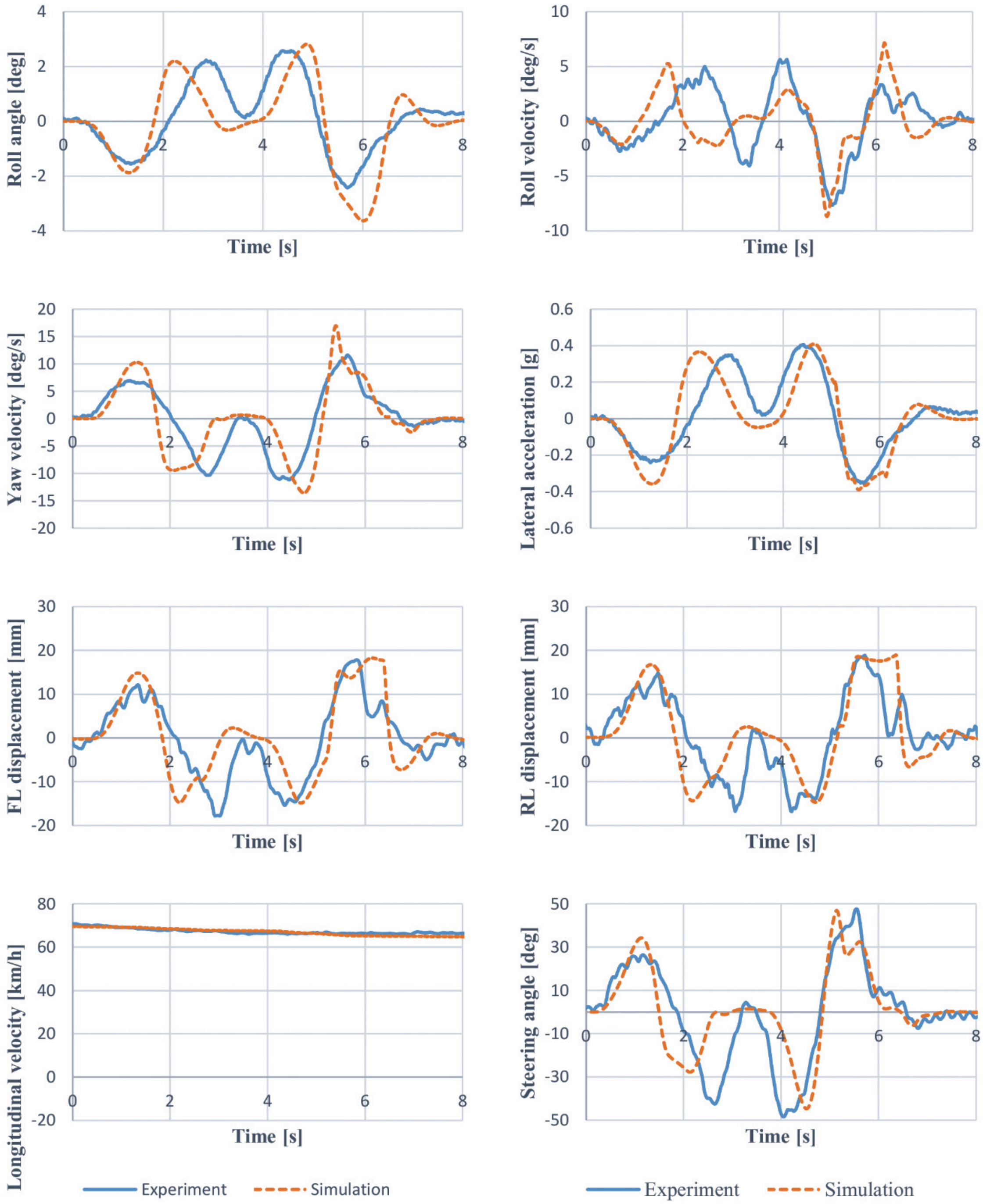


**Figure 5. Simulation and experiment results for a double lane change manoeuver at 70 km/h.**

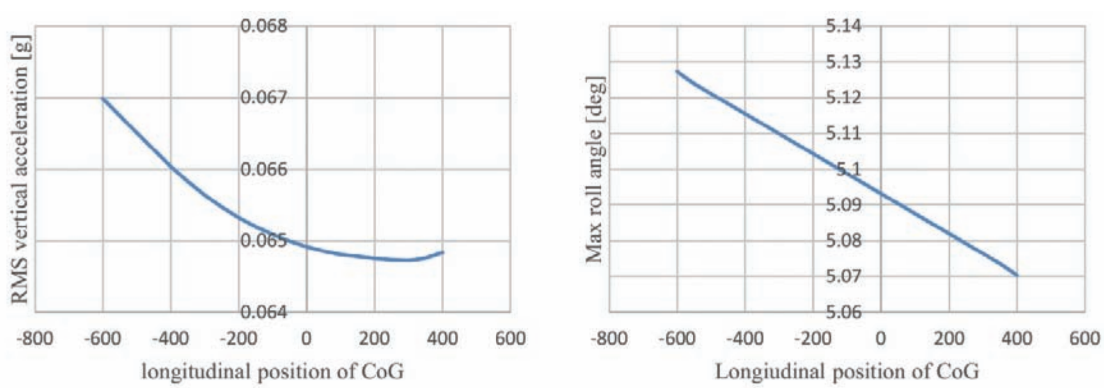


**Figure 6. Effect of equipment longitudinal position.**

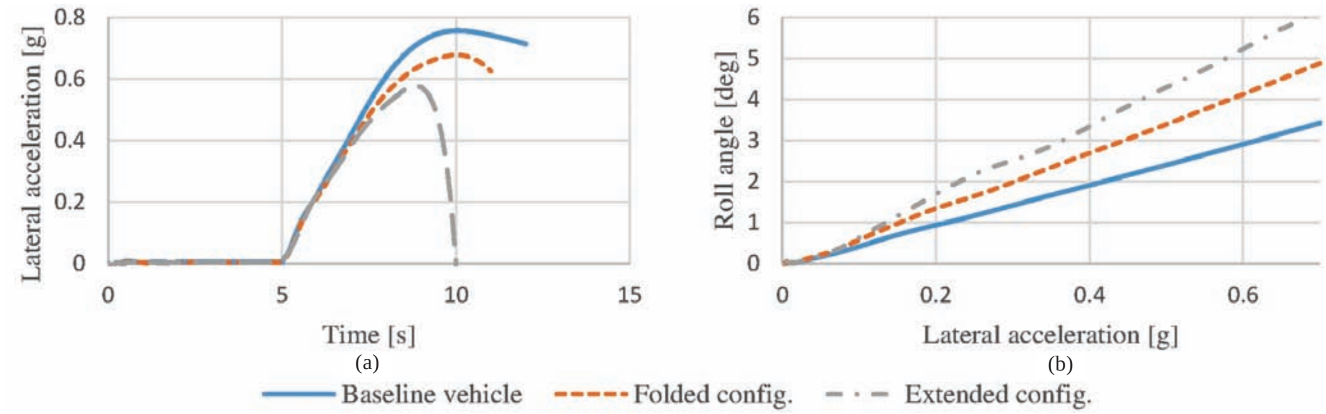


**Figure 7. Simulation results of the ramp steer event.**

time for the considered configurations. The results show that the added equipment significantly affects the lateral stability of the vehicle. The effect of the added equipment can be also represented by the variation of the chassis roll angle as function of lateral acceleration (Fig. 7(b)). The rollover threshold decreases from 0.75 g to 0.68 g when the added mass of 350 kg is mounted at the marching height, which represents about 10 per cent lateral stability degradation. When the added mass is mounted at the operating height, rollover threshold decreases from 0.75 g to 0.59 g which represents about 23 per cent lateral stability degradation.

## 6. CONCLUSIONS

This paper deals with the development of a multibody vehicle model that can be used to predict vehicle dynamic performance after conversion or modification with particular attention to the effect of the added equipment.

A multibody simulation model of the Land Rover Defender was developed in MSC.ADAMS/Car using vehicle parameter measurement. The 19-degree of freedom simulation model was validated against experimental data measured on the instrumented experimental vehicle performing bump test and double lane change manoeuver at various speeds. The validation results suggest good agreement in trend and magnitude between the simulation and field measurement. The differences can be attributed to both experimental procedures and model simplifications such as instrument installation, driver skills, rigid bodies assumption, simplified tire and steering models.

The validated model was used to investigate the effect of adding surveillance equipment to the baseline vehicle. Simulation results showed minimal effect of the longitudinal position of the equipment on the vehicle ride and handling behaviour. However, the equipment vertical position considerably affects behaviour of the vehicle model especially handling dynamics. Simulation of the ramp steer event of the folded and the extended configuration of the converted vehicle provided quantitative evidence of vehicle instability and higher potential for rollover.

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His main contribution was in the definition, implementation and the supervising of the experimental tests as well as the analysis of the experimental data.