

MULTIBODY HANDLING ANALYSIS OF A LIGHT COMMERCIAL TRUCK CONSIDERING FRAME FLEXIBILITY

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Abstract. *This paper presents a complete multibody model of a light commercial truck considering the frame flexibility for handling characteristics studies. The flexible multibody model was developed using MSC.ADAMS, and it includes a complete primary suspension, the steering system, the powertrain, a tire model (Delft) and the flexible frame. The directional response behaviour, the drivability and the vehicle/road interaction are the factors that define vehicle handling performance. The most important parameters concerning lateral behaviour are: longitudinal velocity, tire cornering stiffness, suspension system elasticity, steering systems elasticity, lateral load transfer, mass centre location and aerodynamics properties. Consequently, regarding lateral load transfer, the use of a flexible frame is very important. The complete model was validated with experimental results and for the purpose of vehicle analyses, typical standard handling manoeuvres were undertaken including constant radius turn, constant velocity turn and double lane change. The results obtained were lateral acceleration, yaw rate, side slip angle, roll angle and the understeer gradient. Analysing the results, the vehicle showed a changing behaviour concerning steer. At low velocities and lateral accelerations, the vehicle is slightly understeer. However, at higher velocities and lateral accelerations, it can become oversteer.*

Keywords: *lateral vehicle dynamics, multibody systems, finite element method, and virtual prototyping*

1. Introduction

Multibody modelling has been used in order to make vehicle dynamics parametric studies easier. Virtual prototyping enables simulation engineers to infer about a vehicle's behaviour in different operating stages, resulting in a decrease in test time and number of physical prototypes, which means significant cost reduction. In order hand, the development of new light weight materials and the demand for higher operating speeds required the development of more accurate methodologies to evaluate the effect of bodies undergo large deformation. So the flexible multibody models have been used to accomplish these necessities.

The principal concern of this paper is to describe the methodology applied to infer about the handling capability of a light commercial vehicle. The directional response behaviour, the drivability and the vehicle/road interaction are the factors that define handling performance of a vehicle. The most important parameters concerning lateral behaviour are: longitudinal velocity, tire cornering stiffness, suspension system elasticity, steering systems elasticity, lateral load transfer, mass centre location and aerodynamics properties.

For this purpose, a complete flexible multibody vehicle model was built. The frame of the vehicle under study presents a notable flexibility due to the save of material and the geometry. This fact influences the lateral load transfer during a manoeuvre. In order to capture this behaviour, the frame was included as a flexible body in a multibody environment. A flexible body is introduced in a multibody model through generalized mass and stiffness matrices based on the Craig-Bampton method. A finite element model of the frame was created and a modal analysis was performed generating the necessary information for MSC.ADAMS to create the flexible representation of the frame.

Besides this, the non-linear tire characteristics play an important role in vehicle lateral dynamics. The model includes Delft Tyre model, which is based on Pacejka's Magic Formula.

During the development of a new vehicle, modelling can decrease time and cost, however experimental tests will be always necessary. Thus, experimental tests were undertaken in order to validate the virtual vehicle model proposed in this paper.

For evaluating directional behaviour, typical standard handling manoeuvres were performed including constant velocity turn, constant radius turn and double lane change.

The vehicle studied is a two-axle light commercial truck with a loaded weight of 9000 kg. Both suspensions presents leaf springs, dampers and anti-roll bars.

2. Methodology

For the handling analyses, a complete multibody model of the vehicle was developed using the software MSC.ADAMS/Car 12.0. The methodology of this software is based on a template builder and an analysis interface. Although ADAMS/Car presents a complete set of subsystems for passenger's vehicles, the kind of subsystems usually used in commercial vehicles are not available, such as leafsprings, multiple wheels, flexible frame, etc. Therefore, the majority of the templates were built for these studies.

The two most important template built were the flexible frame and the leafsprings. The procedures applied are described at the next sections.

Another important role during modelling for handling studies is the accuracy of the tire model. The most used tire model is the Deft Tyre Model that is based on Pacejka's Magic Formula (Pacejka, 1991). The Magic Formula tyre aims at an accurate description of the steady-state behaviour of a tyre by providing a set of mathematical formulae, which are partly based on a physical background. It calculates the lateral and longitudinal forces and the aligning torque acting on a tyre under pure and combined slip conditions, using longitudinal and lateral slip, wheel camber and vertical force as input quantities (Deft-Tyre, 1996). The set-up of the tire parameters model was made by Pirelli using experimental measurements.

2.1. Flexible Bodies

A flexible multibody system consists of elastic and rigid bodies connected by joints and/or force elements such as springs, dampers and actuators. Using flexible multibody dynamics theory, it is possible to create and analyse constrained bodies that suffer large displacements, including large rotations. This field emerged in the early seventies as the result of the need to simulate many industrial and technological systems. The development of new light weight materials and the demand for higher operating speeds required the development of more accurate methodologies to evaluate the effect of bodies undergo large deformation. Some related theory was reviewed and some research directions were identified by Shabana (1997).

Nowadays a large amount of codes are able to include flexible bodies. In these studies, the software MSC.ADAMS was used and a short review of the theory that the software uses are presented below (Mechanical Dynamics, 2002).

A finite element model represents an infinite number of degrees of freedom (DOF) using a finite number. The linear deformations of the nodes of this finite element mode, \mathbf{u} , can be approximated as a linear combination of a smaller number of mode shapes, Φ .

$$\mathbf{u} = \sum_{i=1}^M \Phi_i \mathbf{q}_i \quad (1)$$

where M is the number of mode shape and \mathbf{q} the modal coordinates. The deformation behaviour can be captured with a smaller number of modal DOF. So the Eq. 1 can be presented in a matrix form:

$$\mathbf{u} = \Phi \mathbf{q} \quad (2)$$

where Φ is the modal matrix built with the modes of interest Φ_i . At an early stage, the software used the eigenvectors of an unconstrained system, but to achieve model fidelity, an excessive number of modes were required. The solution was to use Component Mode Synthesis (CMS), based on the Craig-Bampton method (Craig & Bampton, 1968).

The Craig-Bampton method partitioned the DOF of the system into boundary DOF, \mathbf{u}_b , and interior DOF, \mathbf{u}_i . In this way, two sets of modes are defined. The constrained modes are obtained by giving a unit displacement to each boundary DOF and the displacement \mathbf{q}_c in the corresponding boundary DOF is the same as the deformation at the boundary \mathbf{u}_b . And the fixed-boundary normal modes are obtained by fixing the boundary DOF and computing an eigensolution, \mathbf{q}_n .

The relationship between the physical DOF, the Craig-Bampton modes and their modal coordinates is illustrated by the following equation.

$$\mathbf{u} = \begin{Bmatrix} \mathbf{u}_b \\ \mathbf{u}_i \end{Bmatrix} = \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \Phi_{ic} & \Phi_{in} \end{bmatrix} \begin{Bmatrix} \mathbf{q}_c \\ \mathbf{q}_n \end{Bmatrix} \quad (3)$$

where \mathbf{u}_b are the boundary DOF, \mathbf{u}_i are the interior DOF, \mathbf{I} is the identity matrix and the $\mathbf{0}$ is the zero matrix, Φ_{ic} are the physical displacements of the interior DOF in the constraint modes, \mathbf{q}_c are the modal coordinates of the

constraint modes, Φ_{in} are the physical displacement of the interior DOF in the normal modes and the q_n are the modal coordinates of the fixed-boundary normal modes.

The generalized stiffness and mass matrices corresponding to the Craig-Bampton modal basis are obtained by a modal transformation, Eq.4 and Eq. 5.

$$\hat{\mathbf{M}} = \Phi^T \mathbf{M} \Phi = \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \Phi_{ic} & \Phi_{in} \end{bmatrix}^T \begin{bmatrix} \mathbf{M}_{bb} & \mathbf{M}_{bi} \\ \mathbf{M}_{ib} & \mathbf{M}_{ii} \end{bmatrix} \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \Phi_{ic} & \Phi_{in} \end{bmatrix} = \begin{bmatrix} \hat{\mathbf{M}}_{cc} & \hat{\mathbf{M}}_{nc} \\ \hat{\mathbf{M}}_{cn} & \hat{\mathbf{M}}_{nn} \end{bmatrix} \quad (4)$$

$$\hat{\mathbf{K}} = \Phi^T \mathbf{K} \Phi = \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \Phi_{ic} & \Phi_{in} \end{bmatrix}^T \begin{bmatrix} \mathbf{K}_{bb} & \mathbf{K}_{bi} \\ \mathbf{K}_{ib} & \mathbf{K}_{ii} \end{bmatrix} \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ \Phi_{ic} & \Phi_{in} \end{bmatrix} = \begin{bmatrix} \hat{\mathbf{K}}_{cc} & \mathbf{0} \\ \mathbf{0} & \hat{\mathbf{K}}_{nn} \end{bmatrix} \quad (5)$$

where the subscripts i and b denote internal and boundary DOF and the subscripts, n and c the normal mode and the constraint mode.

The generalized stiffness and mass matrices of the Craig-Bampton method represents the system since the natural frequencies are associated with them. These matrices are used by ADAMS.Car to build the flexible body.

The finite element model of the frame were elaborated using the software MSC.PATRAN and are shown at Fig. 1. The natural frequencies of the frame were calculated by the software MSC.NASTRAN and are described at the Tab. 1.

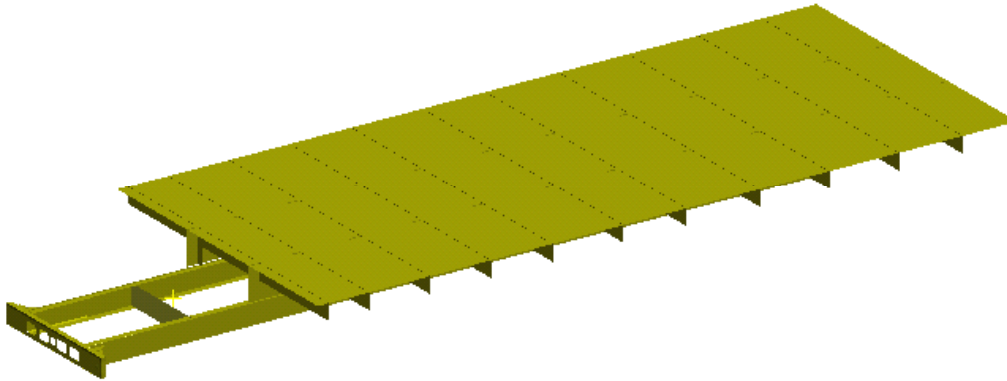


Figure 1. Frame finite element model

Table 1. Main Chassis Mode Shapes.

| Mode Number | Mode Description | Frequency (Hz) |
|-------------|---|----------------|
| 1° | 1 st torsion | 2.9 |
| 2° | 1 st torsion + 1 st lateral bending | 3.2 |
| 3° | 2 nd torsion | 7.0 |
| 4° | 1 st vertical bending | 8.6 |
| 5° | Front lateral bending | 19.1 |

2.2. Leaf spring generation procedure

The ADAMS.Car does not present a tool for the leafspring generation, so, a different methodology was required. The leafspring model was built in three steps.

Firstly, the software ADAMS.Pre was used to create the unloaded leafspring geometry. The modelling required the unloaded leafspring geometry, the axle location, the shackle dimensions and location, the leaf eyehook description and location, the bushing damping and stiffness rates and the material characteristics. The ADAMS.Pre uses beam elements to model the leafspring and the user can define the number of the beam elements used. This procedure resulted in an unloaded leafspring model (Fig. 2a).

Secondly, the normal load was applied to the leafspring in order to generate a loaded leafspring model (Fig. 2b). The unloaded and the loaded leafspring models are shown in Fig. 2.

Finally, this loaded leafspring model was imported into the ADAMS.Car environment and the interface communicators were built.

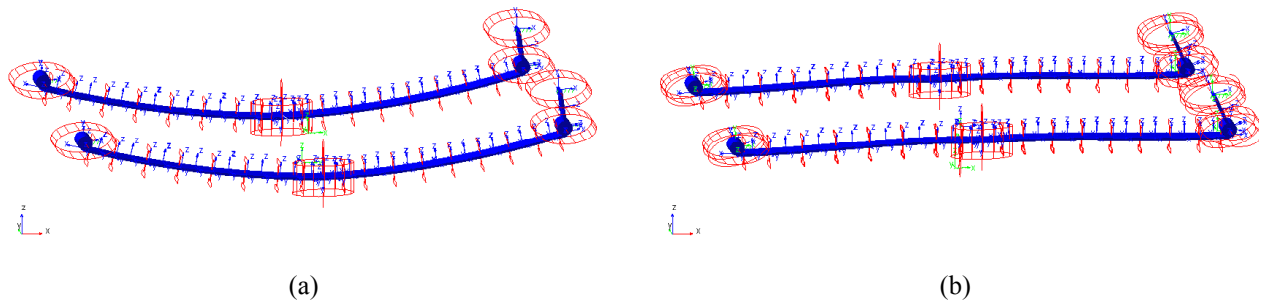


Figure 2. (a) Unloaded front leafspring model and (b) Loaded front leafspring model

2.3. Complete flexible multibody vehicle

The methodology of the ADAMS.Car is based on a template builder and an analysis interface. After all the templates were built, they were added to the analysis interface. So; the axles, the wheels, the leafsprings, the steering system, the cabin, the suspension cabin, the occupants, the powertrain and the flexible frame were connected by communicators into the analysis interface. The complete model used to simulate the manoeuvres is shown in Fig. 3.

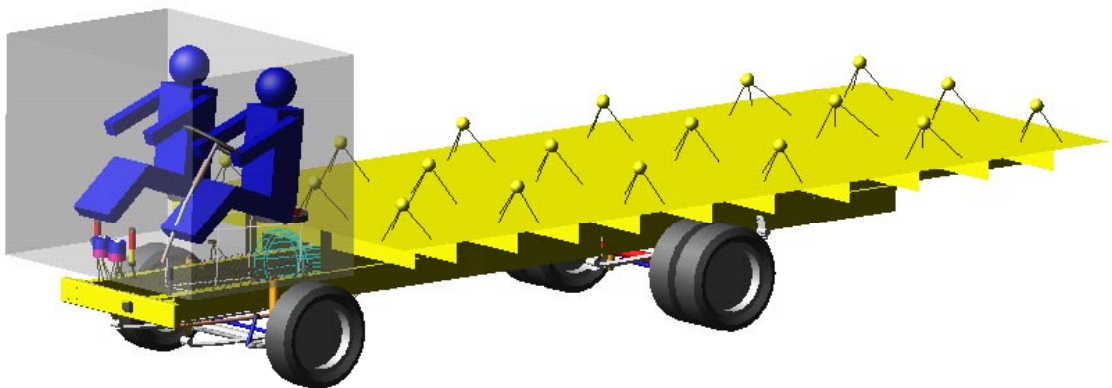


Figure 3. The complete flexible multibody vehicle model

3. Experimental Validation

The experimental validation was made carrying out two types of analysis: transient and steady state. Yaw velocity, lateral acceleration, side slip angle and roll angle were the parameters analysed and they were chosen using the criteria to evaluate lateral vehicle dynamics. The validation was realized imposing the experimental steering wheel angle motion and the longitudinal velocity to the complete model.

An important measure of vehicle performance is its transient behaviour immediately following a steering command. The transient manoeuvre analysis used was the standard double lane change specified by ISO 3888 (Road vehicles, 1975). The specified manoeuvre is a combination of cornering action and the tyres play an important role on this performance, developing the appropriate slip angles and lateral forces. Under this procedure, the vehicle is tested under severe conditions. Most modern vehicles can perform a transient cornering manoeuvres with a lateral acceleration up to 0.8 g (Hegazy *et. al.*, 2000).

A double lane change at the constant vehicle speed 60 km/h was undertaken. Figure 4 shows the vehicle yaw velocity (a), lateral acceleration (b), the side slip angle (c) and the roll angle (d) during the manoeuvre. The maximum lateral acceleration was 0.45 g and the maximum roll angle of 2.5°, which are acceptable limits.

According to the results, the model achieves a good correlation with the experimental results.

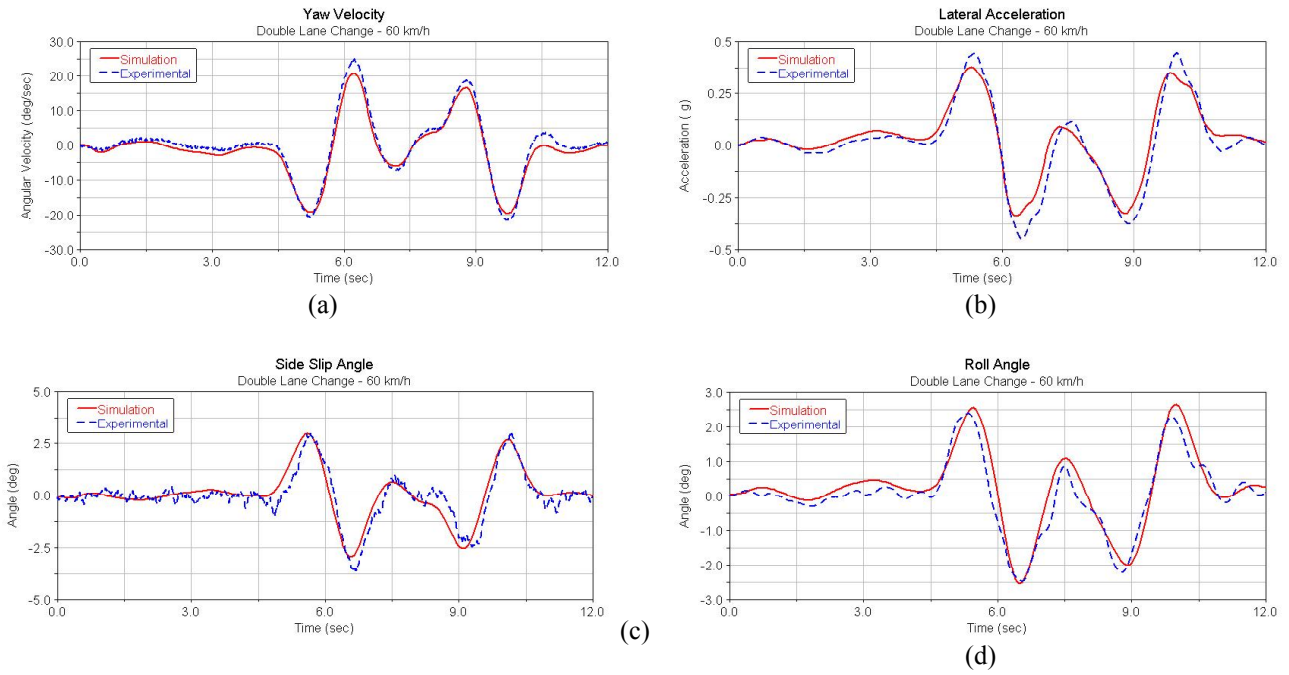


Figure 4. Experimental and simulated results during a double lane change manoeuvre at 60 km/h: (a) Yaw Velocity, (b) Lateral Acceleration, (c) Side Slip Angle and (d) Roll Angle

The steady state was investigated by performing a steering pad testing manoeuvre. In this test the driver should maintain the vehicle at a constant velocity and a constant radius. The manoeuvre was undertaken at 40 km/h at a constant radius of 47.5m. Figure 5 shows the vehicle yaw velocity (a), lateral acceleration (b), the side slip angle (c) and the roll angle (d) during the manoeuvre. The experimental results from this manoeuvre presented a low signal to noise ratio. In order to improve the visibility of the data, a low pass frequency filter was used. Probably, this fact implied that the correlation found on this steady state procedure was not so good as that found at the transient procedure. However, the model achieves a satisfactory correlation with the experimental results, once results such as yaw velocity and lateral acceleration were well captured by the model.

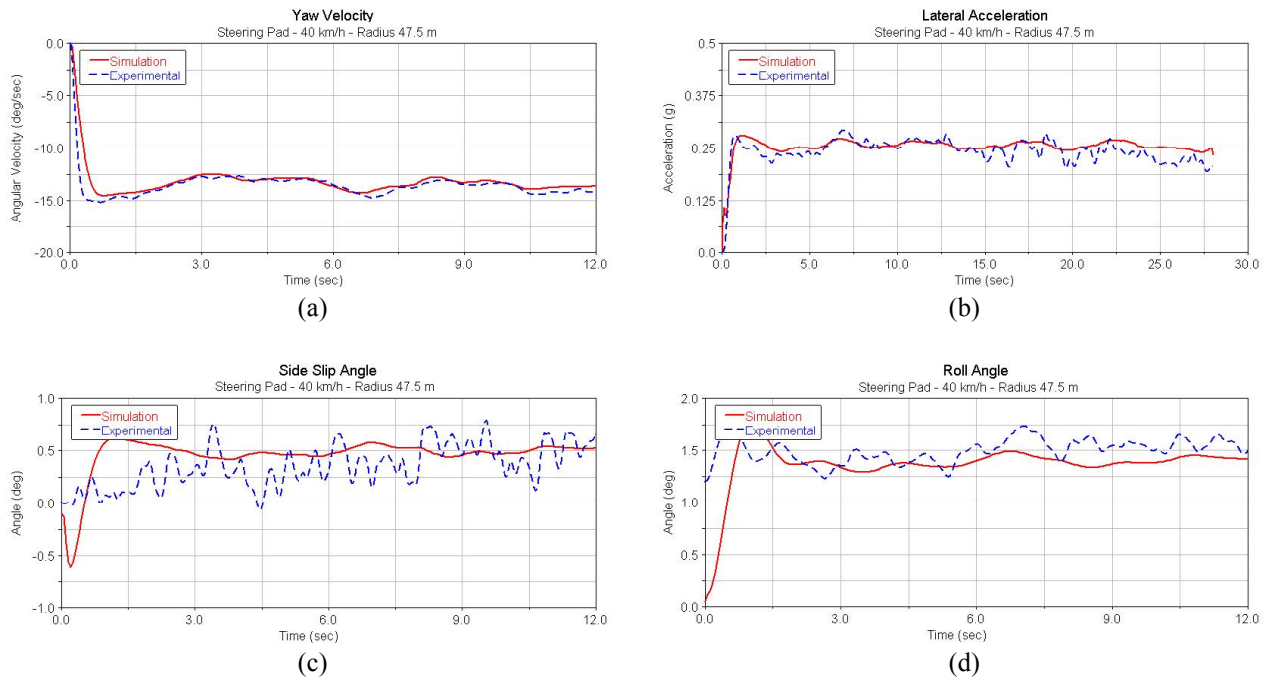


Figure 5. Experimental and simulated results during a steering pad manoeuvre at 40 km/h and constant radius of 47.5 m: (a) Yaw Velocity, (b) Lateral Acceleration, (c) Side Slip Angle and (d) Roll Angle

The same analyses were carried out with a complete multibody model with rigid frame. The results of these analyses showed practically the same result as the flexible multibody model concerning the yaw velocities and the

lateral accelerations. However, better results for side slip angle and roll angle were obtained considering the frame flexibility.

4. Handling Analyses

A vehicle under increasing lateral acceleration on a constant radius turn can be classified in three categories according to the directional behaviour. In this situation, a vehicle in which no change in the steer angle is required to keep a constant radius while the velocity is increased is classified as neutral steer. However, a vehicle in which the steer angle will have to increase with the velocity is classified as understeer. In another case, a vehicle that the steer angle will have to decrease when the velocity is increased is classified as oversteer (Gillespie, 1992).

Experimental steering pad manoeuvres were undertaken at 20, 30 and 40 km/h at a constant radius of 47.5 m. The steering wheel angle necessary to maintain the vehicle at these velocities and the lateral acceleration developed are shown in Fig. 6(a) and Fig 6(b), respectively. The averages of these values are shown in Tab. 2.

According to the experimental results, the vehicle under study has understeer behaviour until 30 km/h, once it increased the steering wheel angle with the increasing velocity and a neutral behaviour from 30 until 40 km/h, once it kept the steering wheel angle with the increasing velocity. It is expected that the vehicle would present oversteer behaviour if the velocity were greater.

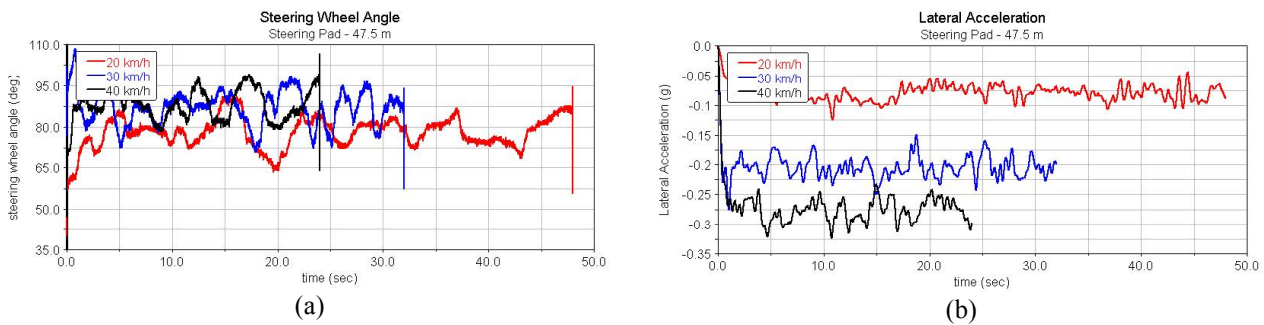


Figure 5. (a) Steering Wheel Angles and (b) Lateral Accelerations for steering pad manoeuvres at velocities 20, 30 and 40 km/h

Table 2. Experimental steering wheel angles and lateral accelerations for steering pad manoeuvres at velocities 20, 30 and 40 km/h

| Velocity (km/h) | Steering Wheel Angle (degrees) | Lateral Acceleration (g) |
|-----------------|--------------------------------|--------------------------|
| 20 | 78.5 | -0.079 |
| 30 | 87.3 | -0.200 |
| 40 | 87.6 | -0.249 |

As an aid to investigate this expectation, constant steering wheel angle manoeuvres were simulated using the validated multibody model. An imposed steering wheel angle of 100° was applied at the velocities 20, 40, 50, 60 and 70 km/h. The yaw velocity resulted of these simulations are shown in Fig. 6(a). A slight increase and a notable increase can be noted at 60 and 70 km/h, respectively. This fact means that the vehicle has oversteer behaviour and the steering wheel angle is greater than the necessary to perform a constant radius corner. So at these velocities the vehicle is always turning at smaller radius and consequently the yaw velocity is increasing.

Based on these manoeuvres, a yaw gain, that is the yaw velocity divided by the steering wheel angle, was obtained. Figure 6(b) shows the yaw gain value with the increase of the velocity. The neutral behaviour is also shown (black line). The results below this line represent understeer behaviour and above this line oversteer behaviour.

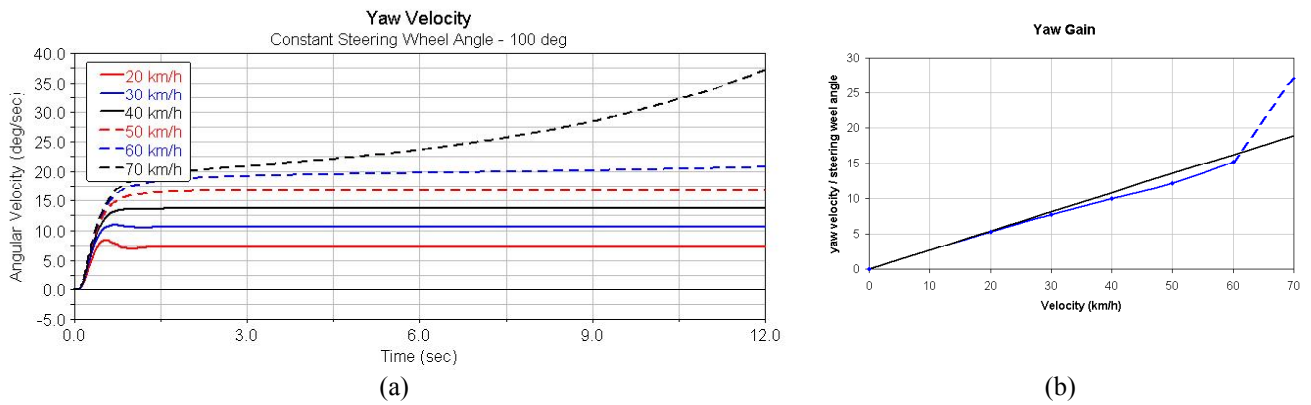


Figure 6. Yaw Velocity for constant steering wheel angle manoeuvres at velocities 20, 30, 40, 50, 60 and 70 km/h

Another usual form to evaluate the lateral behaviour of a vehicle was created by Pacejka and is called handling diagram. The handling diagram is a two-dimension plot with acceleration on the ordinate and radians on the abscissa. The handling equation and the lines of constant velocities (V-lines) and lines of constant path curvature are plotted on this plane (Winkler, 1998). Equation 6 shows the handling equation and δ is the steering angle, l is the vehicle wheelbase, a_y is the lateral acceleration and K is the understeer gradient defined by Society of Automotive Engineers.

$$\delta - \frac{l}{R} = a_y K \quad (6)$$

The handling qualities of the vehicle are readily apparent from the diagram. Since the handling curve is a plot of a_y versus $\delta - \frac{l}{R}$, the slope of the handling curve provides a direct indication of the understeer/oversteer quality of the vehicle. A positive slope indicates understeer, a vertical slope neutral steer and a negative slope indicates oversteer.

Instability can occur when K is a negative value and the vehicle is operating above the critical velocity, $V_{cr} = \sqrt{-\frac{gl}{K}}$. The inverse of the slope of the V-line is $\frac{gl}{V^2}$, g is the gravitational acceleration and V is the longitudinal velocity. Thus, critical velocity at an operating point corresponds to the velocity of the V-line that is parallel to the handling curve (Winkler, 1998).

The V-lines (dash lines) and the handling equation curves (solid lines) for manoeuvres at constant velocity of 20 and 60 km/h are shown in Fig. 7. During the manoeuvre at 20 km/h, the vehicle has understeer behaviour until 0.28 g of lateral acceleration, after that the behaviour is oversteer, but only near 0.50 g of lateral acceleration, some instability can occur. During the manoeuvre at 60 km/h, the vehicle has a neutral steer behaviour until 0.23 g of lateral acceleration, above this, the vehicle is oversteer and some instability can occur above 0.54 g of acceleration.

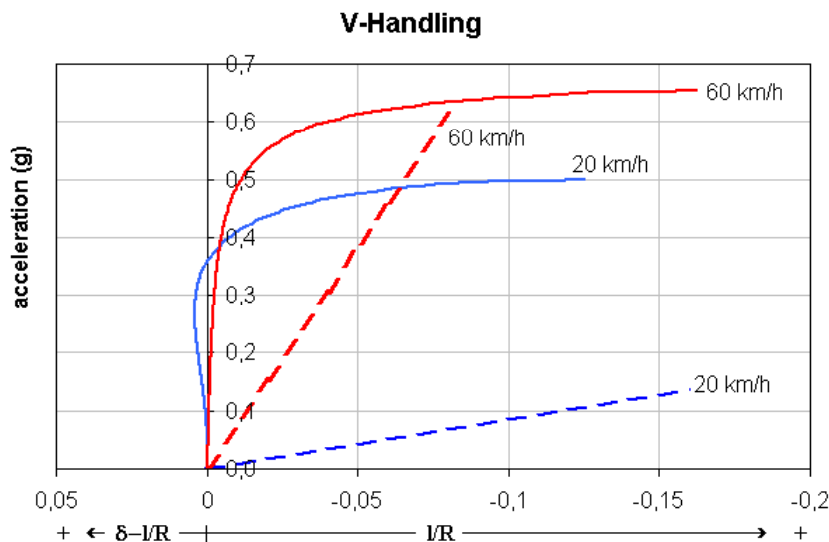


Figure 7. Handling diagram for constant velocity manoeuvres at 20 and 60 km/h

5. Summary and Conclusions

This paper showed the development and validation of a flexible multibody vehicle model. Since the vehicle frame suffers large displacements, the frame was included in the model as a flexible body.

The methodologies, such as leafspring generation and flexible body inclusion on a multibody environment, were discussed. In spite of ADAMS/Car does not present a direct procedure to create leafsprings, the methodology described here is an efficient way of accomplishing this task.

The experimental validation of the vehicle model was realized using the transient and the steady state analyses. However, both sets of analyses presents good correlation with the experimental results, transient results were better captured by the model than the steady state results were. The flexible model had better results concerning side slip angle and roll angle than the model with rigid frame.

Finally, a handling study was performed concerning the steer behaviour. At low velocities and lateral accelerations, the vehicle is slightly understeer. However, at higher velocities and lateral accelerations, it can become oversteer. A handling diagram was shown to evaluate the handling and stability behaviour, once it readily shows the handling qualities.

The methodology proposed in this paper included flexible multibody modelling and standard manoeuvres to evaluate lateral vehicle dynamics. Theses procedures enable design engineers to infer about lateral vehicle dynamics in different operating stages. The result is a decrease in test time and number of physical prototypes so desired for the automotive industry nowadays.

6. Acknowledgements

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7. References

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