Bus Handling Validation and Analysis Using ADAMS/Car

Marcelo Prado, Rodivaldo H. Cunha, Álvaro C. Neto
debis humaitá ITServices Ltda.

Argemiro Costa
Pirelli Pneus S.A.

José E. D'Elboux
DaimlerChrysler do Brasil S.A.

ABSTRACT

This paper describes the modeling, experimental procedures and validation process used for a project together with DaimlerChrysler, Pirelli and debis humaitá to validate the bus model in ADAMS/Car. It is described the bus subsystems which have been generated by ADAMS/Car templates. A comparison of experimental results and model simulation for an ISO lane change at 80 km/h and a sweep steer at 40 km/h with steering wheel imposed motion are made. Some of the quantities ilustrated are: steering wheel angle, lateral acceleration and yaw rate. Frequency domain analysis was made and the yaw rate and lateral acceleration gains and phases due to steering wheel angle were plotted. Finally, a comparison is made between ISO lane change with imposed motion versus a machine control (driver) with varying driver parameters.

INTRODUCTION

Vehicle dynamic simulation using multibody vehicle model has been very useful in reducing time and cost in the development stages. Using computer simulation the engineer may understand the dynamic behavior of the vehicle in different operating conditions.

In this paper, the main objective is the validation of the multibody vehicle model in handling maneuvers. In this kind of analysis tire dynamics dominate the handling behavior. So, it is very important to have a good characterization of tire characteristics.

In order to validate the multibody model, experimental measurements were carried out with steering wheel imposed motion and the results were compared to simulation results, with the same motion.

MULTIBODY MODEL OF THE BUS

Figure 1 shows the bus multibody model that was built in ADAMS/Car version 10.1 The model was divided into eight subsystems: front suspension, rear suspension, front wheel and tire, rear wheel and tire, steering system, brake system, powertrain and car body (comprised of seats, chassis and all the agregate elements).

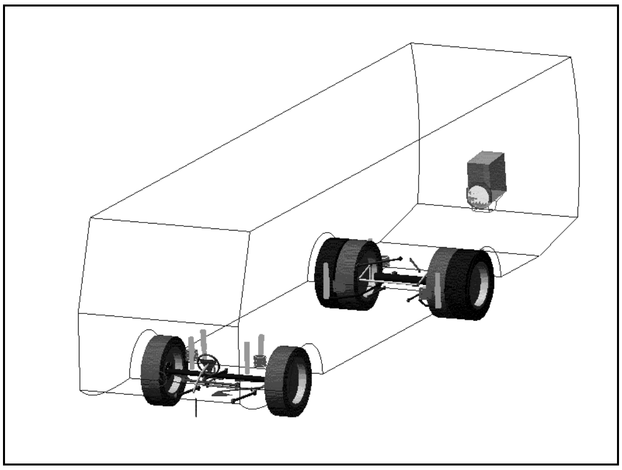

Figure 1: Full bus model in ADAMS/Car.

Figure 2 shows the front suspension subsystem. This subsystem is comprised of a rigid axle, two air springs , four telescopic shock absorbers, a stabilizer bar, three longitudinal tension bars and a Panhard rod.

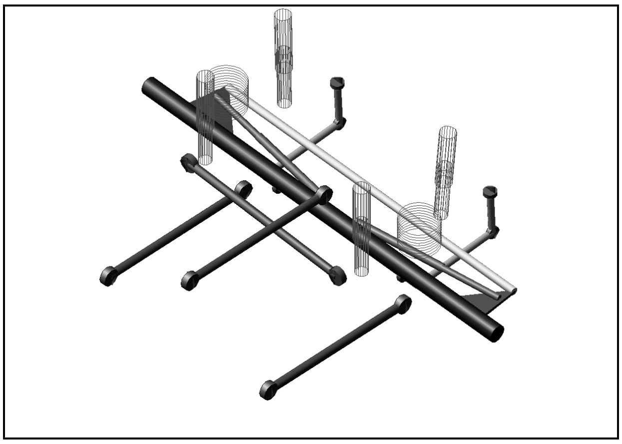

Figure 2: Front suspension subsystem.

Figure 3 shows the rear suspension subsystem. This subsystem is made of a rigid axle, four air springs, a stabilizer bar, four shock absorbers, two longitudinal tension bars and two inclined bars.

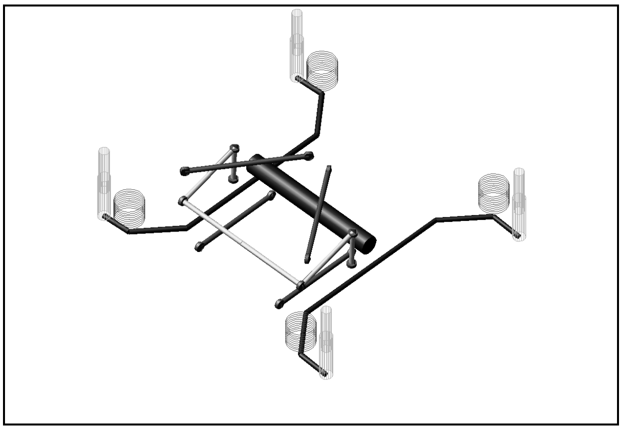

Figure 3: Rear suspension subsystem..

The air spring was modeled using an adiabatic curve since only transient analysis has been performed (in this type of analysis the process is adiabatic, without heat transfer). The front and rear axle were modeled as rigid bodies . The shock absorbers were modeled using non linear curves. The non linearities of the bushings stiffnesses were also included. The car body was modeled as rigid body because the type of chassis used in this bus is very stiff .

The steering system model is shown in Figure 4. The steering compliance and damping were included in this model. The steering compliance was obtained from experimental results.

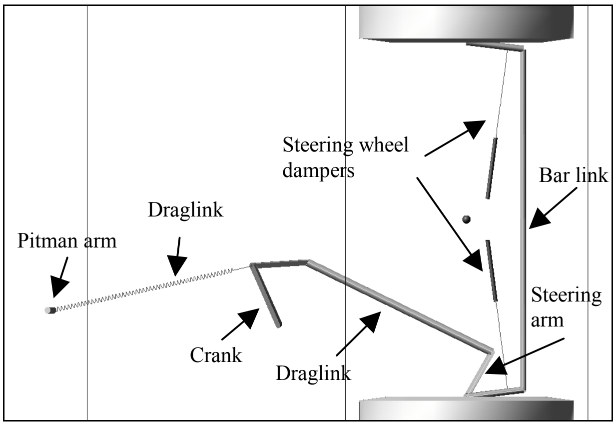

Figure 4: Steering system model

To setup tire parameters experimental measurementes were made by Pirelli. With these parameters, a wheel and tire susbsystem was built using the Delft Tyre Model. It is based on Magic Formula. Magic Formula calculates the longitudinal and lateral forces and aligning moment acting on the tire under pure and combined slip conditions, using longitudinal and lateral slips, wheel camber and the vertical force as input quantities. In a handling analysis, it is very important a good tire characterization, because tire dynamics has a strong influence in vehicle dynamics.

In order to perform closed-loop simulations (imposing steering wheel angle and maintaining the logitudinal velocity), the brake and powertrain systems were modeled. The ADAMS/Car Driving Machine needs these subsystems to perform closed-loop maneuvers. Figure 5 shows the engine curve of the bus.

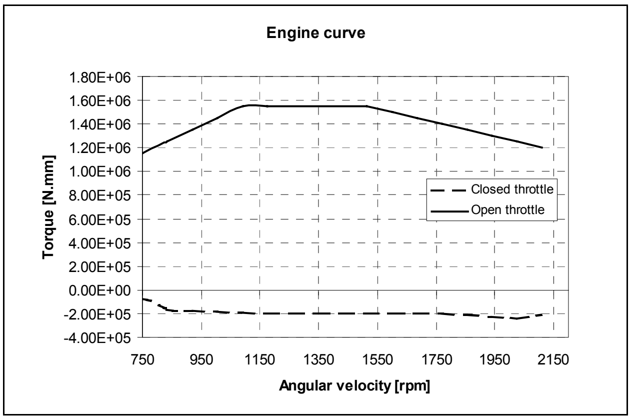

Figure 5: Engine curve of the bus model.

HANDLING ANALYSIS

Open-loop and closed-loop maneuvers were carried out. The open-loop maneuvers have been performed using experimental steering wheel angle imposed at steer wheel at the bus model. Closed-loop maneuvers were performed using ADAMS/Car Driving Machine. In these maneuvers, the Driving Machine attempts to perform a ISO lane change maneuver at 60 km/h at prescribed path and velocity.

EXPERIMENTAL VALIDATION

The experimental validation was carried out with experimental steering wheel angle imposed at the ADAMS/Car model and comparing the model results to experimental measurements (Figure 6).

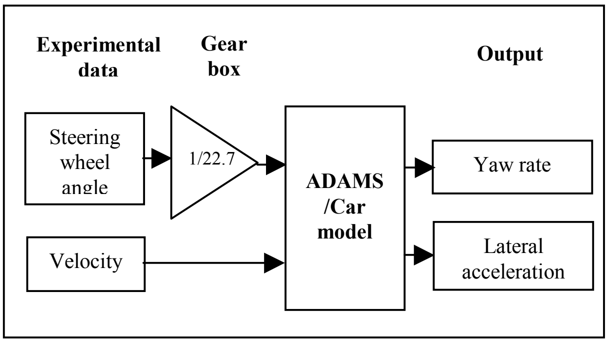

Figure 6: The experimental steering wheel angle was imposed and the results were compared to experimental measurements.

DOUBLE LANE CHANGE AT 80 km/h

The double lane change at 80 km/h was performed according to ISO Technical Report 3888 [8] with experimental steering wheel imposed motion (Figure 7).

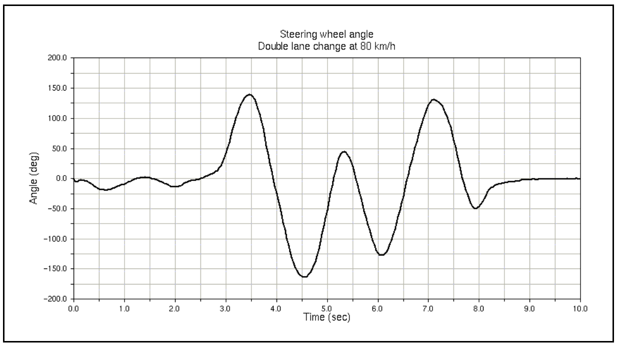

Figure 7: Steering wheel angle.

Figure 8 shows the power spectral density (PSD) of the steering wheel angle. It can be seen that the band-width is limited and the frequency-dependent behavior of the vehicle coul not be determined.

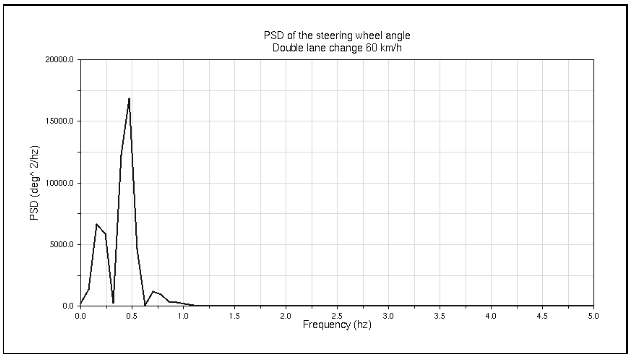

Figure 8: PSD of the steering wheel angle.

Figure 9 shows the experimental and simulated results of the lateral acceleration. It can be seen that the simulation results are very close to experimental results.

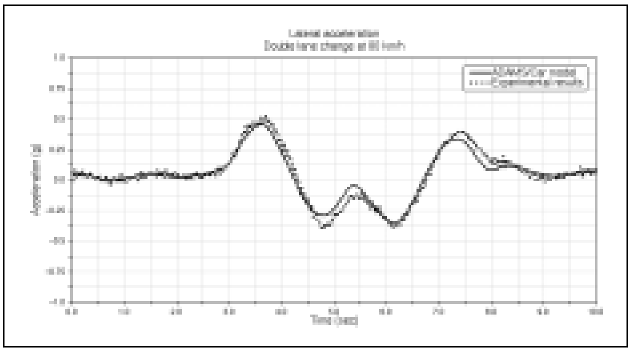

Figure 9: Lateral acceleration.

Yaw rate response is shown in Figure 10. The peaks of experimental results were higher than simulation. Figure 10 shows that the bus model has more damping in yaw mode than the real vehicle.

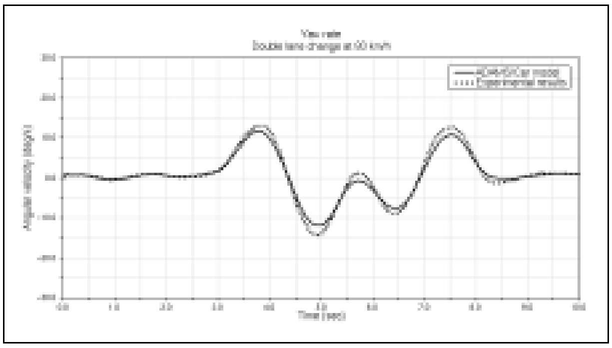

Figure 10: Yaw rate response.

SWEEP STEER AT 40 km/h.

The sweep steer maneuver was performed. Figure 11 shows the steering wheel imposed motion.

Figure 12 shows the PSD of the steering wheel angle It can be seen from Figure 12 that the band-width of this maneuver is limited and it is not possible to obtain much information about the frequency-dependent behavior of the vehicle from this test.

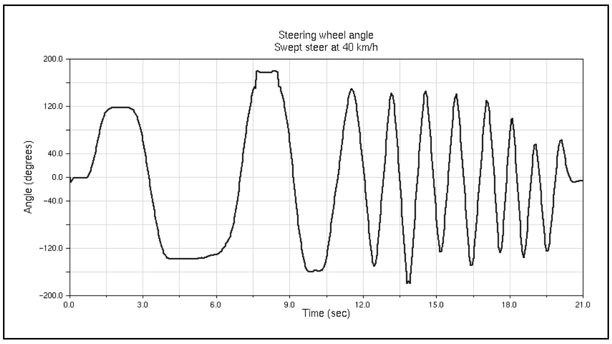

Figure 11: Steering wheel angle.

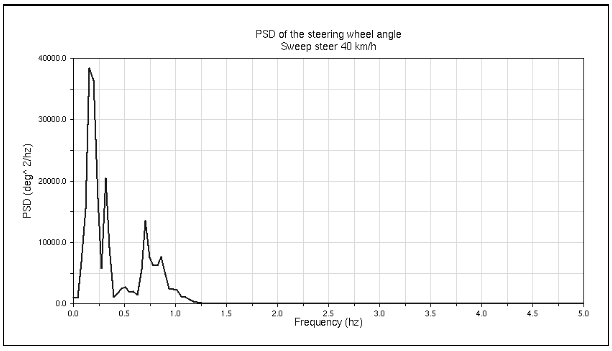

Figure 12: PSD of the steering wheel angle.

Figure 13 shows the lateral acceleration. The correlation of the simulation was very close.

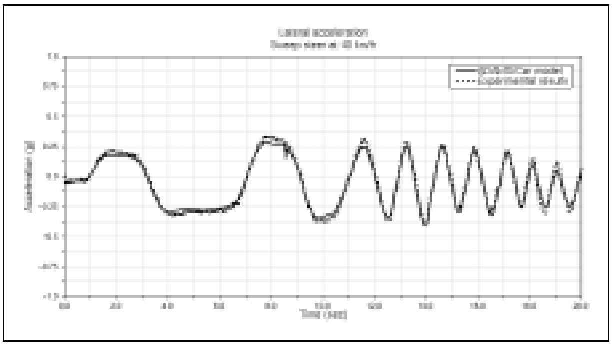

Figure 13: Lateral acceleration.

The yaw rate response is shown in Figure 14. The correlation of the simulation was very close for low frequencies. However, with increasing frequency, the simulation began to deviate from experimental results. This behavior was due to yaw damping of the bus model.

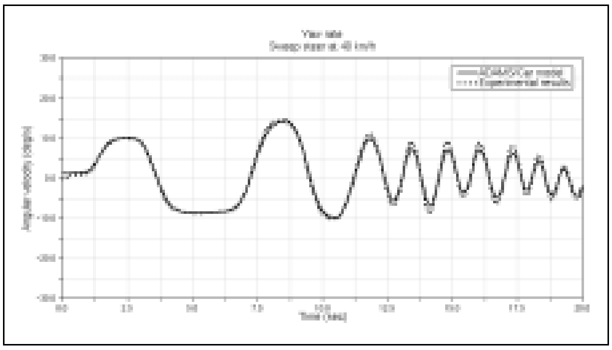

Figure 14: Yaw rate.

FREQUENCY DOMAIN ANALYSIS

In order to investigate the vehicle frequency-dependent behavior, the sweep steer maneuver at 70 km/h was carried out in ADAMS/Car model. The steering wheel angle is shown in Figure 15.

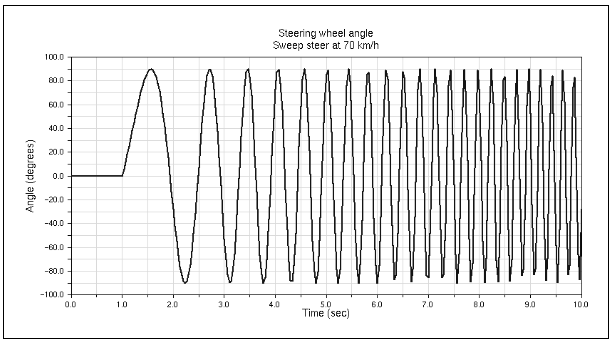

Figure 15: Steering wheel angle.

The PSD of the steering wheel angle is shown in Figure 16. The band-width of this maneuver was large enough to analize the frequency-dependent behavior of the vehicle.

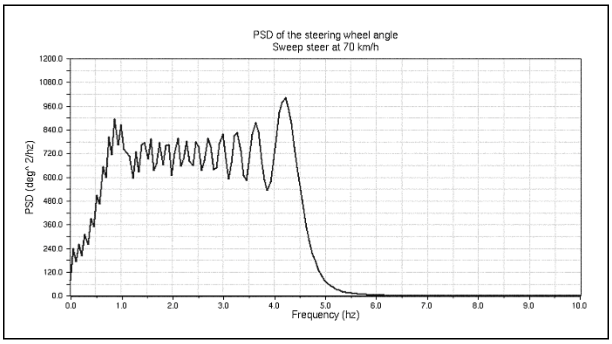

Figure 16: PSD of the steering wheel angle.

Figure 17 shows the lateral acceleration gain due to steering wheel angle. The shape of the lateral acceleration response is explained by the fact that the lateral acceleration is composed of the centrifugal acceleration caused by yawing velocity and a linear acceleration along the lateral axis [1].

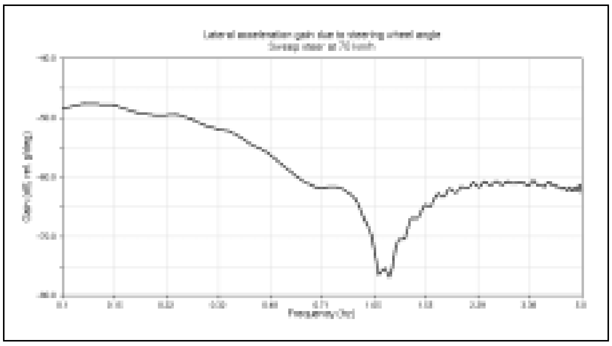

Figure 17: Lateral acceleration gain.

The phase angle between lateral acceleration and steering wheel angle is shown in Figure 18.

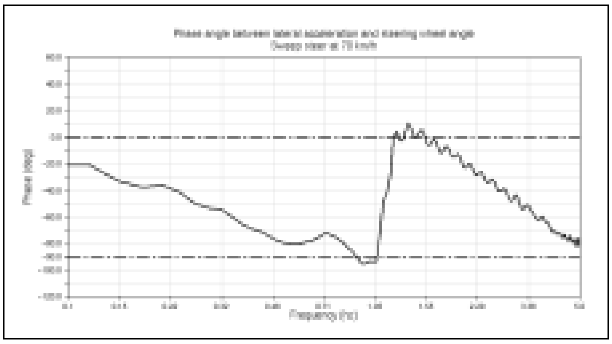

Figure 18: Phase angle between lateral acceleration and steering wheel angle.

Figure 19 shows the yaw rate gain due to steering wheel angle. At low frequencies, the yaw rate remains nearly constant. As frequency increases beyond 0.5 Hz, the inertia of the vehicle begins to predominate and the yawing velocity response tends to decrease. A high frequency steering input produces very little response [1].

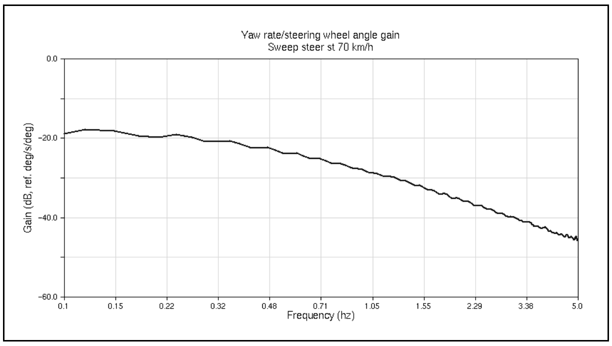

Figure 19: Yaw rate gain

Figure 20 shows the phase angle between yaw rate and steering wheel angle.

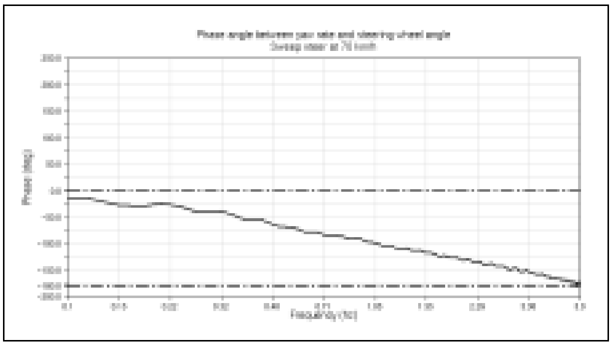

Figure 20: Phase angle between yaw rate and steering wheel angle.

ISO DOUBLE LANE CHANGE MANEUVERS USING ADAMS/CAR DRIVING MACHINE

One of the maneuvers that is avaliable in ADAMS/Car Driving Machine is the ISO Lane Change Maneuver. In this maneuver, the Driving Machine attempts to perform a double lane change maneuver at prescribed path and velocity.

Driving Machine defines the driving style with one parameter called Lateral Preview Time (LPT), representing the time the drivers knows in advance the desired trajectory [3]. Low values of LPT represents nervous driver reactions and high values of LPT represents very smoothing driving styles.

ISO double lane change maneuver using machine control has been performed using different values of LPT.

Figure 21 shows the steering wheel angle of the ISO double lane change maneuver at 60 km/h with different values of LPT compared to experimental steering wheel angle.

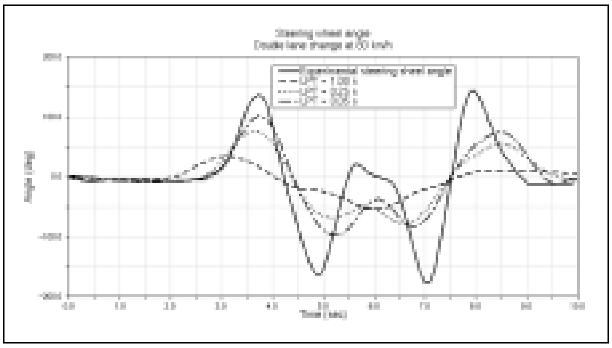

Figure 21: Steering wheel angle in an ISO double lane change at 60 km/h with varying LPT and with experimental steering wheel imposed motion.

Figure 22 shows the bus trajectory with different values of LPT and with experimental steering wheel imposed motion.

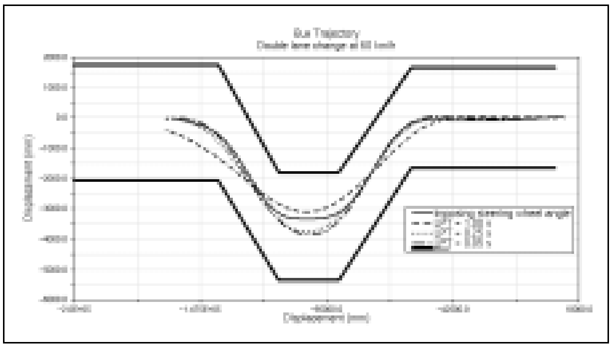

Figure 22: Bus trajectory in an ISO double lane change at 60 km/h with varying LPT**.**

Figure 23 shows the lateral acceleration with different values of LPT and with experimental steering wheel imposed motion.

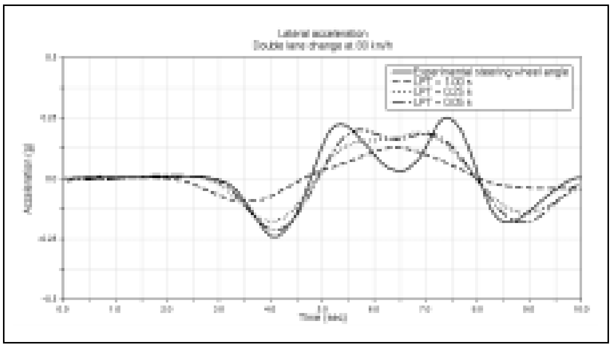

Figure 23: Lateral acceleration in an ISO double lane change at 60 km/h with varying LPT.

Figure 24 shows the yaw rate at different values of LPT and with experimental steering wheel imposed motion.

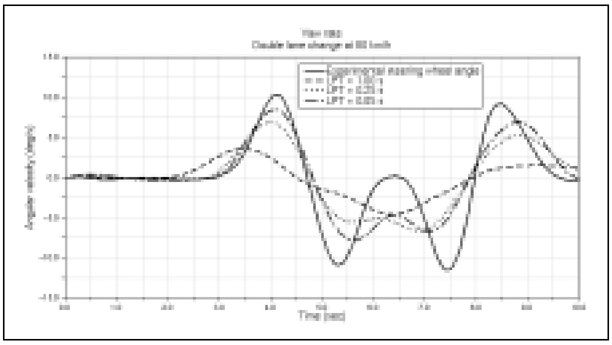

Figure 24: Yaw rate in an ISO double lane change at 60 km/h with varying LPT.

It can be seen from Figure 21 to 24 that the LPT strongly influences the behavior of the vehicle during the maneuvers. The experimental result compared to simulation results were shown that the real driver has low value of LPT.

CONCLUSION

The bus model has been validated for transient maneuvers (the experimental results were very close to simulation results). The bus model was not validated in steady-state maneuvers because the air spring model was not modeled with height and roll control valves

The influence of driver styles in bus handling behavior was analyzed and the results were shown that driver styles has a strong influence in bus handling behavior.

For future investigations, the air spring will be modeled with height and roll control valves in order to perform steady-state analysis.

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