VEHICLE DYNAMICS MODELING AND VERIFICATION WITH TEST OF ARTICULATED BUS

Erol ÇİFCİ[1], Semih SEZER[2]

ABSTRACT

Vehicle Dynamics model of articulated passenger bus were created. In the modeling stage firstly the vehicle's mass, moment of inertia and center of gravity were determined. Secondly front, middle and rear axle properties of damper, air spring and anti rollbar were defined and atteched the model. Finally, the articulated system was modeled and linked with the front and rear body. Vehicle model was created and simulations were made by using Adams / Car software. Additionally, the reduction ratio of steering system was added and verified with tests. After that model verification made with field measurement test. After the displacement and acceleration, the data was collected and processed on the prototype.

Keywords: *Vehicle Dynamics, Multiple Body Dynamics, Articulated Bus, Road Data Processing, Lumped Mass, Full Vehicle Model, MSC Adams, nCode*

INTRODUCTION

Vehicle dynamic is concerned with the movement of cars, trucks and buses on the road. Comfort, handling, cornering rotational, acceleration, braking are the main subjects of vehicle dynamicsThe dynamic behavior depends on the forces transmitted to the vehicle from the tires, resulting from gravity and aerodynamic factors. In general vehicle dynamic examines, it is sufficient to think of the vehicle as a single mass. Wheel and body masses should be considered separately for driving and suspension examine.[1]

In this context, the multi-body dynamics model of the bellows articulated passenger bus used in urban transportation was created in the MSC Adams Car workspace, as shown in figure 1. the general vehicle dynamics analysis of the prototip vehicle, the turn diameter, bump, lane change and acceleration analyzes were evaluated in the virtual simulation. Then, the acceleration and displacement data were collected under the test conditions determined on the prototype vehicle and multiple body dynamics model verification approaches were implemented. After obtaining the desired result from the validation studies, studies were carried out to improve the control of the vehicle dynamics model comfort, fatigue and bellows system. This study only covers the build of multi-body model of the articulated passenger bus, data collection via prototype vehicle, processing the collected data and model validation process.



Figure1. İsuzu 18M articulated bus and full vehicle dynamics model

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CREATING THE CAR DYNAMICS MODEL

A complete vehicle mathematical model will be used for vehicle dynamics analysis and model verification examine. The complete vehicle model consists of two bodies and a bellows system that connects them together and various sub-systems.

The model consists of steering wheel, front suspension, middle suspension, bellows system, rear suspension, center and rear anti-roll bars, wheels and front and rear body.

Front Suspension and Steering System

The front suspension multi-body dynamics model is shown in figure 2.

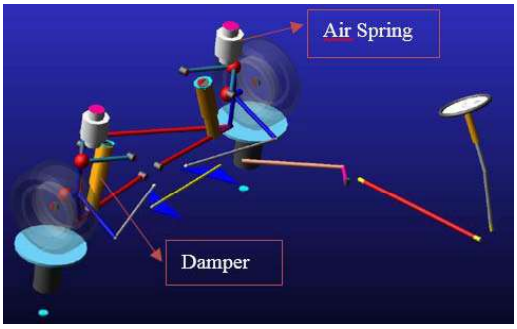


Figure 2. Front suspension and steering system multi-body dynamics model

While the pre-layout geometry shown in Figure 2 is started to be created, firstly the hardpoint coordinates that define the connection points of the parts with each other and the points in which they are located in space are entered. After the system kinematics is formed, the features of the elastic parts in the model as damper, air spring nonlinear are added to the model. The air spring is modeled with full open length, full closed length, design length, nominal working pressure, adiabatic force-displacement information.Weight information and moment of inertia information of the system were determined and added to the model.

Anti Roll Bar

The multi-body dynamics model of the middle and rear suspension anti-roll bar is shown in figure 3.

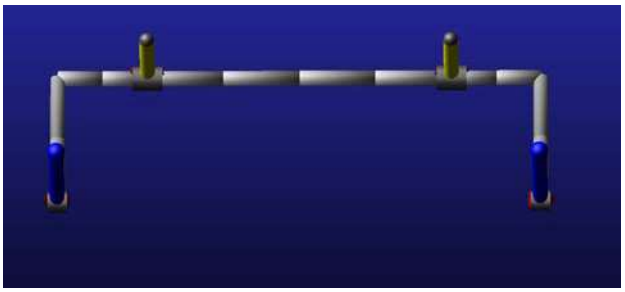


Figure 3. The multi-body dynamics model of the middle and rear suspension anti-roll bar

The anti-roll bar used in the middle and rear suspension improves the rollover characteristics of the vehicle. Increased vehicle stability during instantaneous lane change and cornering. The torsional stiffness value of the anti roll bar modeled with beam elements was calculated by the theoretical and finite element method and added to the model.

The torsional stiffness is calculated with the calculation presented by the Society of Automotive Engineers (SAE). The accepted geometry dimensions are shown in figure 4 and finite element models are shown in figüre 5.[2,3]

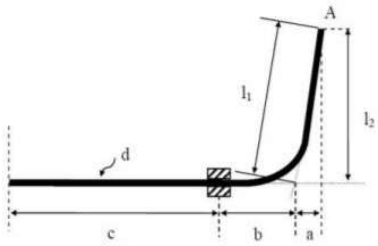


Figure 4. Anti-roll bar geometry used in SAE Spring Design Manual

The loading is applied at point A,inward to or outward from plane of the page.The roll stiffness of such a bar can be calculated as:

L=a+b+c (L:Half Track Lenght)

$$f_a = \frac{P}{3.E.I}\left(l_1{}^3 - a^3 + \frac{L}{2}.(a+b)^2 + 4.l_2^2.(b+c)\right) \quad (1)$$

$$f_a = \frac{1000}{3.(210000).(201288{,}959)}.\left(310^3 + \frac{970}{2}.165^2 + 4.(310^2).(485)\right) \quad (2)$$

$$f_a = 7{,}8856.10^{-9}(29791000 - 13204125 + 186434000) \quad (3)$$

$$f_a = 1{,}80mm \quad (4)$$

f_A:-Deflection of point A

$$KR = \frac{P * L^2}{2f_A} \quad (5)$$

$$KR = \frac{P * L^2}{2f_a} = \frac{1000(970)^2}{2(1.80)} = 260031.18\ Nm/rad \quad (6)$$

KR:Roll Stiffness of the bar

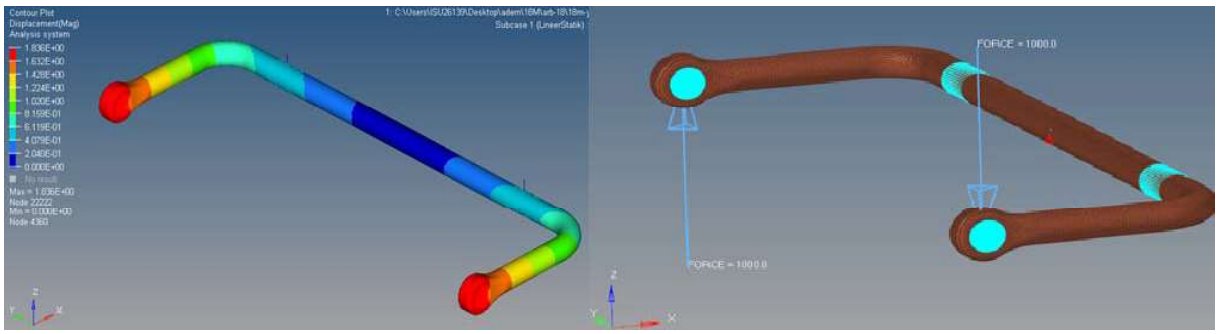


Figure 5. Anti-roll bar finite element A point displacement

$$KR = \frac{mgL^2}{2.f_a} = \frac{1000.(970)^2}{2.(1{,}83)} = 257076.5\ Nm/rad \quad (7)$$

$Roll\ stiffness_{teorik} = 260031.18\ Nm/rad$

$Roll\ stiffness_{sonlu\ eleman} = 257076.5\ Nm/rad$

When the deflection ratio was calculated between the theoretical calculations and the finite element model results, the deflection ratio was found to be 1.13% and the roll stiffness value was added to the multiple bodies model.

The Articulated System of Vehicle

The multi body dynamics model of the articulated system is shown in figure 6.

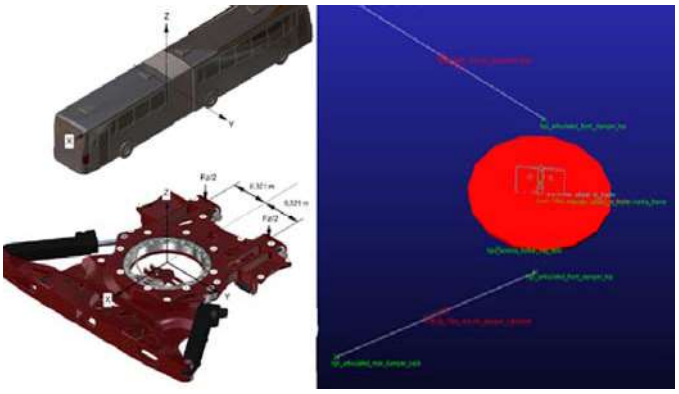


Figure 6 The multi body dynamics model of the articulated system

The articulated system connects the front and rear bodies. There are two dampers on the left and right side of the articulated model. The system also has its own controller to determine the damping rate of these dampers. In order to prevent the vehicle from skidding during instantaneous lane changes, the controllers step in and ensure the stability of the vehicle. [4]

Bodys of Vehicle

All sub-systems and articulated system are combined with the bodies. The multi-body dynamics model for the loading conditions of the front and rear body is shown in figure 7.

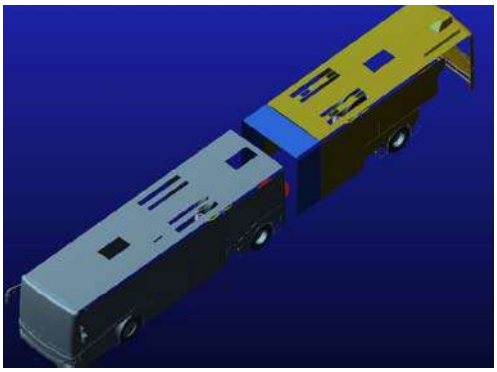


Figure 7. The multi-body dynamics model for the loading conditions of the front and rear body

All weight and mass inertial information of the front and rear body are defined by the center of gravity by the lumped mass approach. Body Weights information is quickly optimized using the full-vehicle analysis and adjustment mass method in the adams car.[5]

Tires

Belt Dynamics wheel model was chosen because it has higher frequency range than other wheel models and 3d enveloping contact algorithm was used from Adams / Tire Contact Methods. Belt dynamic tire frequency spectrum is shown in figüre 8.[6]

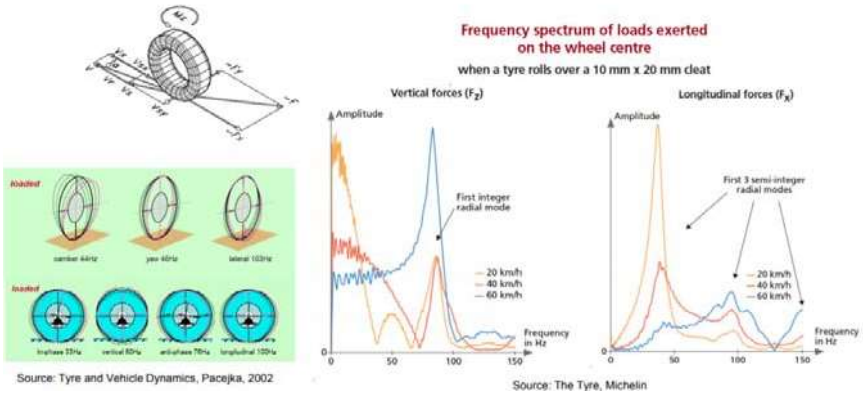


Figure 8. Belt dynamic tire frequency

Tire belt dynamics frequencies> 15Hz and wheel type 275 / 70R22.5 determined and multi-body model was created.

Wheel Slip Angle

$$\alpha = \operatorname{atan}\left(\frac{V_{sy}}{V_x}\right) \tag{8}$$

It is calculated from the equation. Vsy is the lateral speed of the wheel, Vx is the linear speed of wheel

RESULTS AND DISCUSSION

Lane Change Condition

Analysis and tests were performed as vehicle speed is 40 Km / h and road lane widths as shown in figure 9.

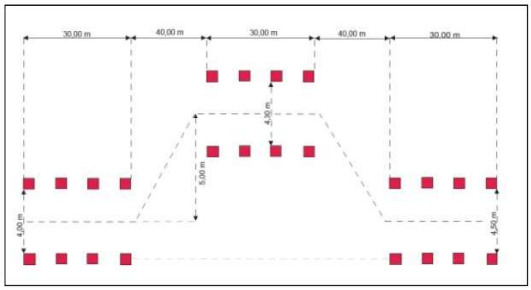


Figure 9. Lane change dimensions

Angle change of the articulated system at the time of lane change was measured by tests. The collected data and the multi-body dynamics model are shown in Figure 10 comparatively.

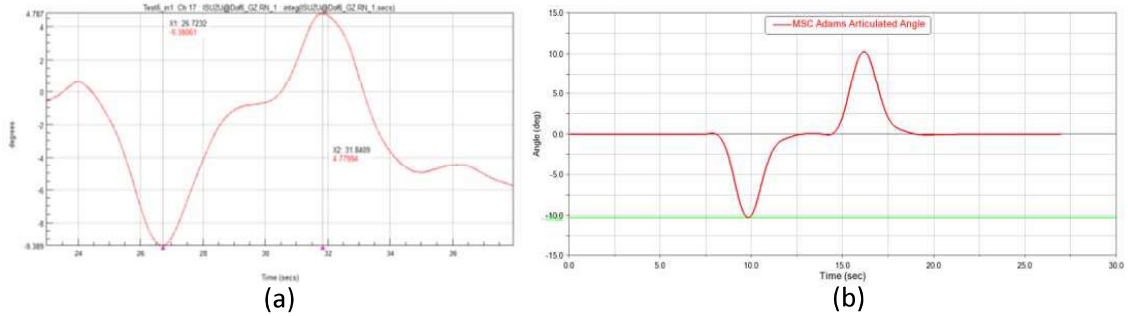


Figure 10. Lane change conditions. (a) Test result, (b)MBD result

Turn Diameter Conditions

Maximum steering angle is given to the steering wheel to determine the turning radius of the articulated passenger bus. In order to prevent lateral slippage of the vehicle, attention has been paid to keep the vehicle speed at a certain value. Analysis and tests are compared as shown in Figure 11.

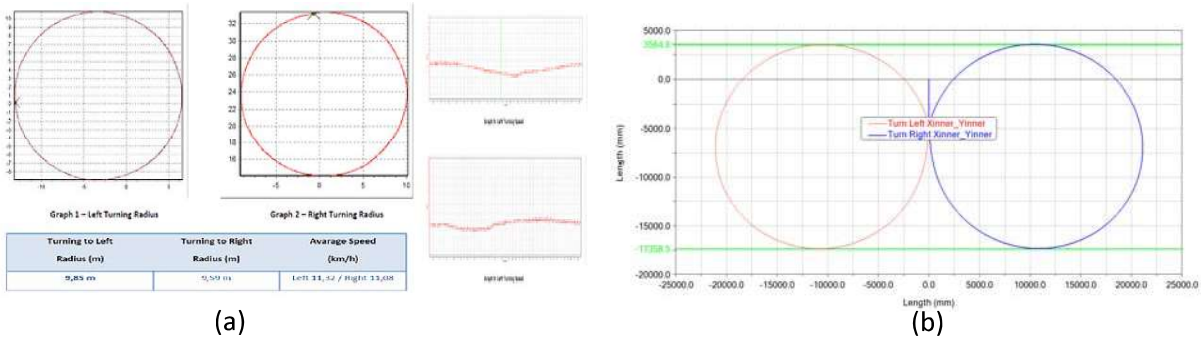


Figure 11. Vehicle turn diameter,(a)test result,(b) MBD result

The prototype vehicle internal radius obtained in the tests was 9.85m for left turn, whereas this value was determined as 10.46 in the multi-body dynamics model. A difference of about 5.7% was observed. This difference is due to the controller of articulated system of the vehicle being activated and causing the vehicle to slip slightly laterally.

Obstacle Passing Conditions

Multiple body dynamics analyses and physical tests were first performed with vehicle speed of 15km / h and 30km / h. The mathematical expression of the obstacle is y = 0.12 * sin (13.09 * x). In order to collect data on the prototype vehicle, physical tests have been performed at physically same bump dimensions and speeds. The displacement and accelerometer sensors positions on the prototype vehicle are shown in figure 12.



Figure 12. The displacement and accelerometer sensors positions.

(a) damper encoder, (b) front COG accelerometer

60-channel HBM Somat eDAQ data collection system was used as data collection system.
Accelerometers: ± 50g three-axis accelerometer (x10): Dytran 7523A5
Displacement meter:
LVDT Displacement Meter (x2): SIKO SG5
LVDT Displacement Meter (x4): Micro Epsilon WPS-750-MK30-P10
All channels are sampled at 1000 Hz. It was aimed to compare the test results with the results of the multi-body dynamics model. For this purpose, acceleration data collected from test vehicle wheel centers were converted to displacement data by signaling process. With this study, both the operating state of the suspension system was observed and the validation of the multi-body dynamics tire model was observed.

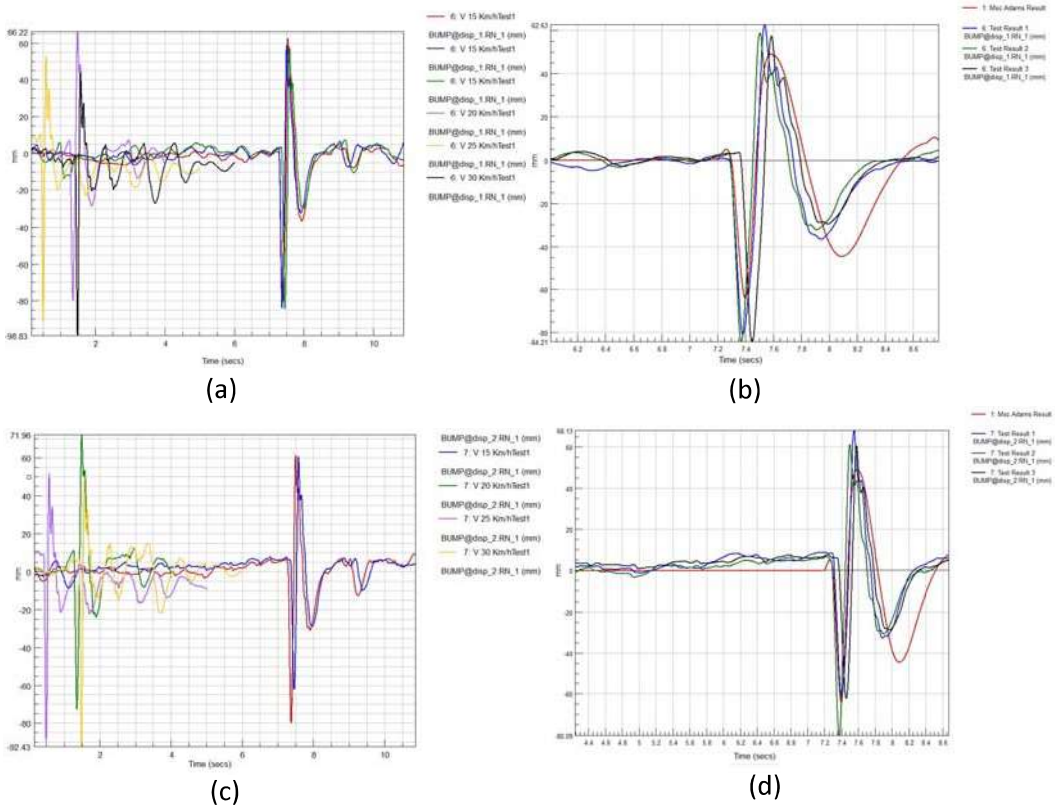


Figure 13. Front damper displacement data. (a) test data front axle left V = 15,20,25,30 km / h, (b) test-mbd front left V = 15km / h, (c) test data front axle right V = 15,20,25,30 km / h, (d) test-mbd front right V = 15km / h

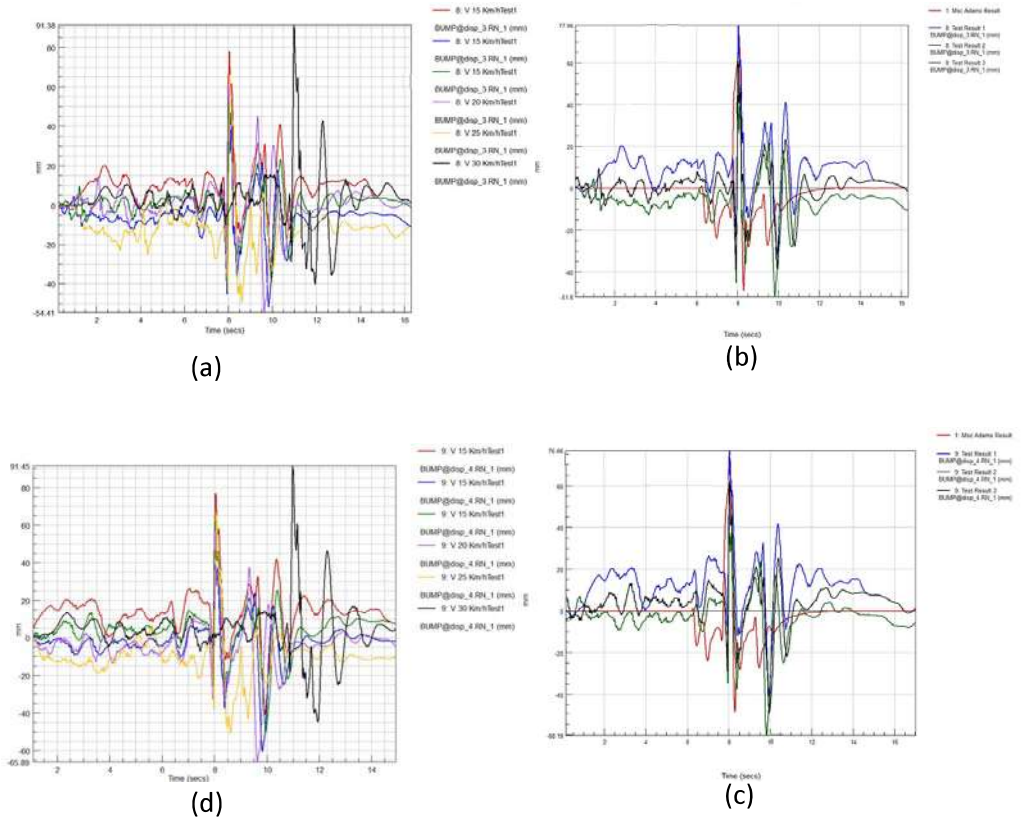


Figure 14. Middle daöper displacement data. (a) test data middle axle left V = 15,20,25,30 km / h, (b) test-mbd middle axle left V = 15km / h, (c) test data middle axle right V = 15,20,25,30 km / h, (d) test-mbd middle axle right V = 15km / h

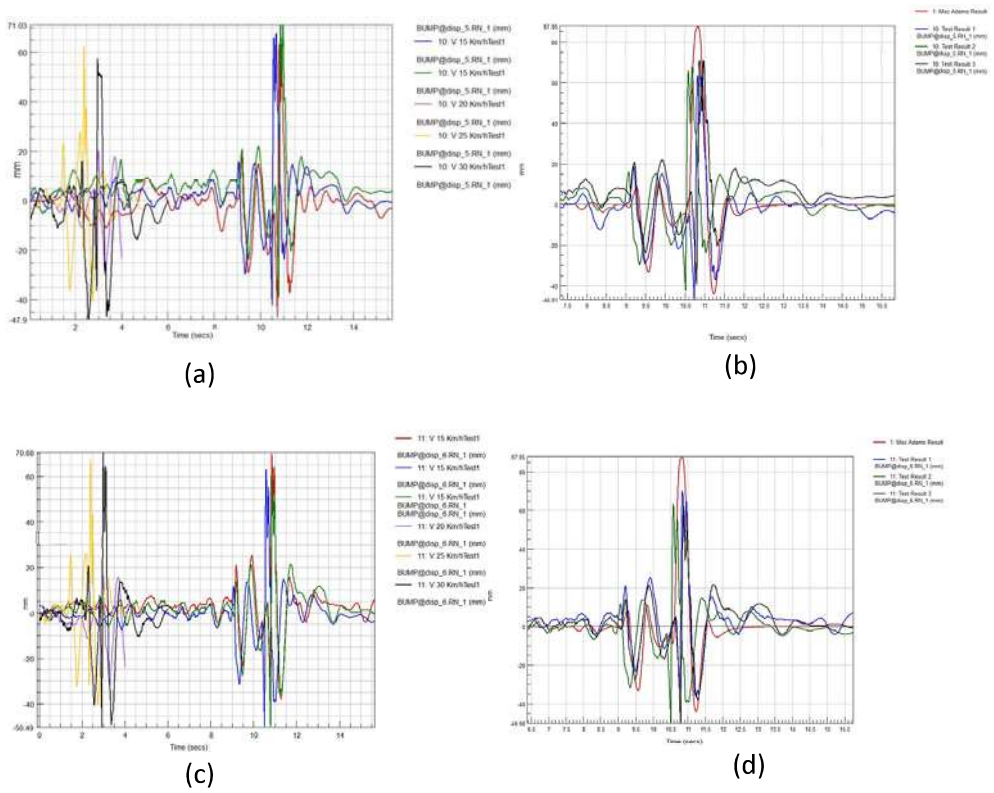


Figure 15. Rear damper displacement data. (a) test data rear axle left V = 15,20,25,30 km / h, (b) test-mbd rear axle left V = 15km / h, (c) test data rear axle right V = 15.20,25,30 km / h, (d) test-mbd rear axle right V = 15km / h

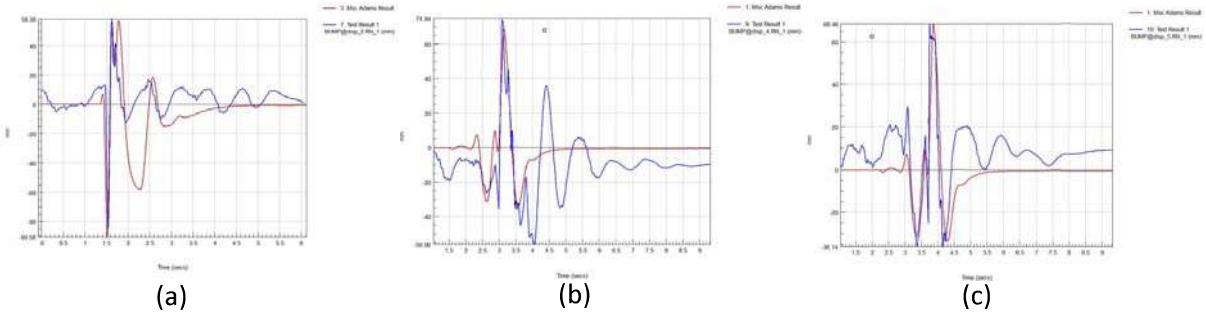
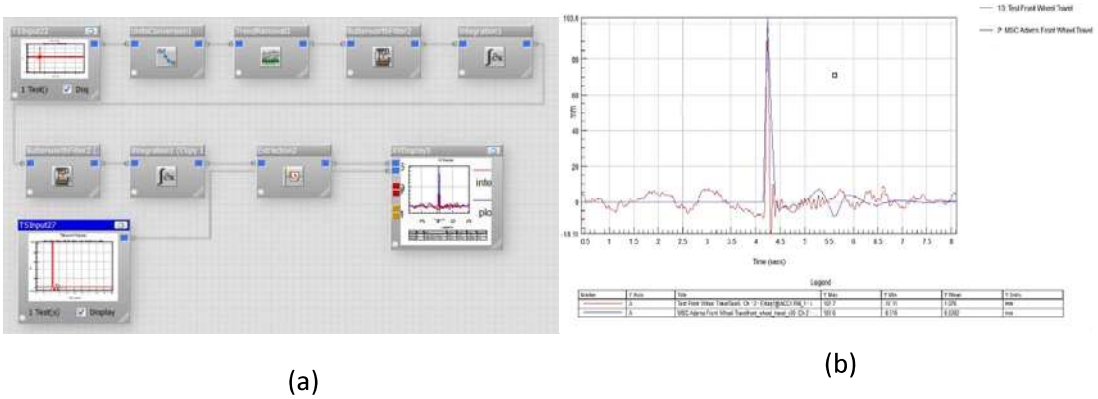
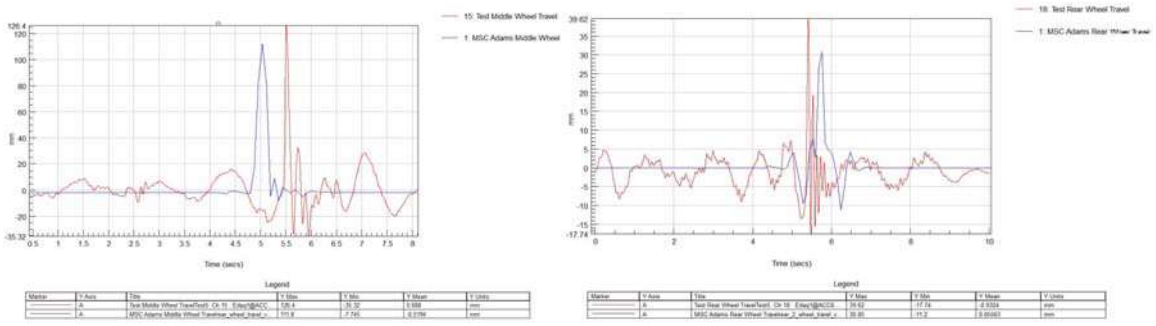


Figure 16.(a) test-mbd front axle left damper displacement. V=30km/h, (b) test-mbd middle axle left daöper displacement V=30km/h, (c) test-mbd rear axle left displacement V=30km/h

In the acceleration data collected from the wheel centers, the existing dirft was first corrected and then cleaned with a hight pass filter for 0.75 Hz. Afterwards, it was transformed into displacement data by integrating 2 times. Figure 17 compares the signal processing with the results of the test and multiple-body dynamics.





(c) (d)

Figure 17. Conversion of wheel center acceleration data to displacement. (a) Signal processing steps (b) test-mbd front axle left V = 30km / h, (c) test-mbd middle axle left V = 30km / h, (d) test -mbd rear axle left V = 30km / h

CONCLUSION

In this study, the multi-body dynamics model of the commercial articulated passenger bus was created in MSC Adams workspace. Displacement and acceleration data were collected from the prototype vehicle produced for the model validation examine. Multi body dynamics model and test results are tried to be verified. The multi-body dynamics model of the articulated passenger bus was mechanically modeled. Therefore, the analysis and test results at a maximum speed of 40 km / h were compared. Above 40km / the mbd model loses its stability. Analysis above this speed showed that the controller for the articulated system was needed. In turning radius determination analyses, a difference of 5.7% was observed between test and mbd model. This can be attributed to the activation of the controller in the test vehicle and the occurrence of minor lateral slip during the turns in the test vehicle. In the bump pass analysis, while the vehicle speed was 15 km / h, a small difference between the test vehicle damper displacements and mbd model displacements of 20 mm was observed. The test results were similar with the mbd model when the test was repeated at speeds of 30 km / h, because of the fact that the damper started to run more actively. When comparing bump pass analysis and test results, only damper displacements at bump pass were evaluated. When the acceleration data collected from the wheel hup are processed and converted to displacement data and compared with the results of the mbd model, the wheel vertical stiffness value in the mbd model is found to be a little smaller than the physical model. When the stiffness value of the wheel was reduced, mbd results and test results were similar. It is decided that vehicle dynamics analysis can be continued on the verified model.

NOMENCLATURE

MBD	Multi body dynamics
fA	Displacement,mm
KR	Roll stiffness,Nm/rad
α	Slip angle,deg
V	Longitudinal speed, km / h
Hz	Hertz

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