

# Case Study and Simulation of Vehicle Dynamics Using Motion-View/Motion-Solve

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## Abstract

This paper presents case study on the vehicle dynamics of car in two scenarios, case 1 is single lane change analysis and case 2 is double lane change analysis, and the result are then formulated later according to the corresponding case study. A millidegrees-of-freedom non-linear multibody dynamic model of a vehicle, comprising front and rear suspensions, steering system, road wheels, tyres and vehicle inertia. The model incorporates all sources of compliance, stiffness and damping, all with non-linear characteristics. The vehicle model is modified using in-built library of Hyper works – Motion-view/Motion-solve (automatic dynamic analysis of mechanical systems) formulation. The model is used for the purpose of vehicle handling analysis. Simulation runs, in-line with vehicle manoeuvres specified under ISO and British Standards, have been undertaken and reported in the paper.

**Keywords:** Multi-body dynamics, Suspensions, Steering, Non-linear characteristics, Simulations.

## 1. INTRODUCTION

Motion View is a general pre-processor for Multi-body Dynamics. It is unique in that one can utilize the Motion View modelling language to build models that will simulate both within Altair's solver suite as well as with third-party solvers.

Motion View's native modelling language is called MDL (Model Definition Language).

MDL has several unique characteristics; such as the ability to set any data as an algebraic parametric expression of anyone (or more) other model data. In addition, models or assemblies in MDL can have unlimited hierarchy and can pass topology (or data) from the parent systems to the child systems for use locally. MDL follows an object-oriented approach so that definitions can be reused as many times as needed in order to construct a model.

Motion View MDL models can be exported to a variety of solvers including Altair Motion Solve, MSC ADAMS, Dassault ABAQUS, as well as others. Users can create their own solver writers or data readers using TCL which is the command language used throughout Motion View. TCL can be used to drive the entire software in batch operation or can be used to create a customized GUI (Graphical User Interface).

The topics following this overview will provide much more detail on these and other aspects of Motion View.

Whether we are riding in an expensive sports car, an inexpensive sub-compact, or a rugged SUV or off-road vehicle, the ride and handling characteristics determine much of the perception of comfort and performance of your vehicle. Vehicle dynamics engineers need sophisticated methods for analysing and improving the "feel" of an automobile, and simulation methods are essential in finding the optimal solution.

Altair multibody solutions help to simulate and optimize all the important parameters that govern vehicle dynamics:

**Suspension Design:** Study the suspensions to find optimum suspension design factors.

**Ride and Handling:** Tune the full vehicle handling and ride characteristics using appropriate handling or durability tires.

**Loads Prediction for Fatigue Analysis:** Perform a four-post durability analysis or a full vehicle analysis with durability tires and generate loads for the fatigue analysis with the virtual model.

An important measure of vehicle performance is its handling characteristics when subjected to a given steering input. The stability of the vehicle under various specified manoeuvres can be investigated. Typical vehicle handling simulations include transient cornering, lane changing and slalom motions, including double-lane changes, with or without braking. The current analysis is concerned with a transient cornering manoeuvre with a constant forward velocity. Several important parameters are investigated.

These include the tyre forces, spring forces, damper forces and bump and rebound stop forces, particularly on the inside wheels as the vertical force diminishes with increasing lateral acceleration. The vertical excursion of the front and rear roll centre heights is also of interest as a large displacement of these can affect the vehicle stability.

Most modern vehicles can undergo cornering manoeuvres with lateral accelerations of up to 0.8 g, during which body roll in the region  $2\pm 88$  can occur. The specified manoeuvre should represent a realistic test of vehicle behaviour under severe conditions. Test procedures have been specified by international standards in ISO 7401± 1988 or in the British Standard BS AU 230:1989. Figure 6 shows an animated output for the transient manoeuvre during a simulation time of 5 s. Five hundred-time steps of simulation were undertaken, after an initial static equilibrium analysis is carried out to ensure vehicle placement at the kerb height.

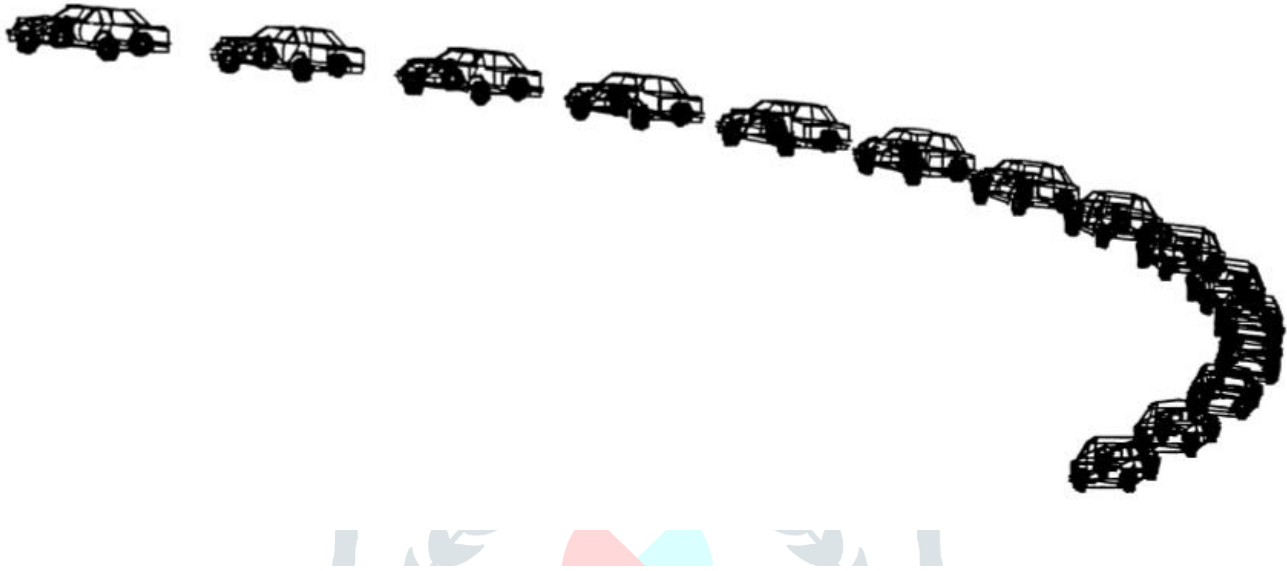


Figure 1

## 2. DESCRIPTION OF THE VEHICLE MODEL

The full vehicle model comprises vehicle mass and inertia, front and rear double-wishbone suspensions, a rack and pinion steering system, road wheels and tyres. Vehicle motions are described in terms of the fixed global frame of reference X, Y, Z shown in Fig. 1. Local part frames of reference  $x_i, y_i, z_i$  are attached to all the moving parts. A generic formulation method, based upon Lagrange's equation for constrained systems, is employed for the derivation of equations of motion for all parts in the model in a body  $3\pm 1\pm 3$ Euler frame of reference.

### 2.1. SUSPENSION AND STEERING SYSTEM MODEL

The front and rear suspensions are of double-wishbone configuration. Each quarter-suspension comprises two control arms, referred to as the lower and upper control arms respectively. The control arms are connected to the vehicle body by elastic rubber bushings with non-linear characteristics. Typical characteristic curves for these are shown in Fig. 2. The bushings provide appropriate longitudinal and torsional compliance which influences the dive (during braking) and squat (during acceleration) characteristics of the vehicle. The shock absorber is attached to the vehicle underbody and the lower control arm. A bump stop is situated on the lower control arm, while a rebound stop is located on the upper control arm.

In order to study the handling behaviour of a vehicle, the tyre cornering forces must be carefully determined. The simplest form of tyre modelling involves the computer storage of a large amount of measured tyre data which are used in conjunction with an interpolation method to represent the tyre forces characterized by the measured data. This method is currently used in general purpose dynamics software. Most tyre models currently used in vehicle dynamic simulations involve empirical representations of the measured tyre data.

### 2.2. TYRE SYSTEM MODEL

There are six components of force and moment generated as a result of tyre interaction with the road. These are the vertical tyre force, longitudinal traction force and lateral force, as well as the self-aligning moment, the overturning moment and the rolling resistance moment (see Fig. 4a). The tyre model reported here does not include the overturning moment.

### Tyre vertical force

The radial tyre contact force, acting in the radial plane of the tyre, has a component that acts in the direction of the contact normal at the tyre road contact patch. This component is used to calculate the tyre vertical force. The radial force is dependent on tyre deflection and its rate of change, both measured along the tyre vertical directional vector. The deflection is obtained by an instantaneous evaluation of the distance between the position of the wheel centre and the road surface plane in the contact patch. The time rate of change in deflection is obtained by the vector scalar product of the instantaneous tyre radius vector and the wheel centre global velocity. The quantities thus obtained are employed to obtain the stiffness and damping contributions to the tyre vertical force, as indicated by the first and second terms in equation.

$$F_{\text{vert}} = k\delta - c \frac{\partial \delta}{\partial t} \quad \text{Eq (1)}$$

### Tyre lateral force

There are two alternative methods for determination of the tyre lateral force. These are through the use of an equation method or by an interpolation process. The equation method, employed in this analysis uses the following simplified equation to generate the lateral force-

$$F_{\text{lat}} = \mu F_{\text{vert}} (1 - e^{-k_{\text{lat}}|\alpha|}) \quad \text{Eq (2)}$$

Where,

$$\alpha = \tan^{-1} \left( \frac{v + l\psi_{10}}{u} \right) - \beta \quad \text{Eq (3)}$$

### Tyre Longitudinal Force

The rolling resistance force and the traction (or braking) force together constitute the Tyre longitudinal reaction force [17]. The rolling resistance force is calculated by multiplying the coefficient of rolling resistance,  $f$ , with the vertical force. The traction force is obtained by multiplying the instantaneous value of the coefficient of friction with the vertical force. These forces oppose the motion of the vehicle at the road surface contact patch. The combined force is referred to as the longitudinal force.

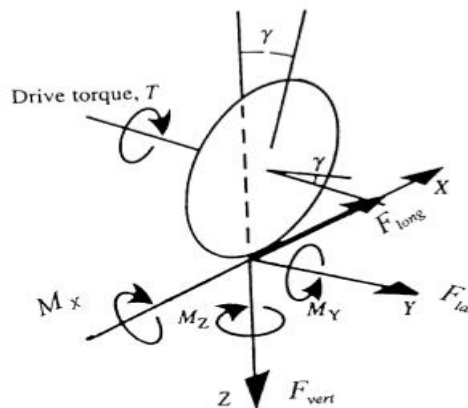


Figure 2

### 3. ANALYSIS

#### CASE 1 – FULL VEHICLE BODY WITH SLA SUSPENSION SYSTEM ON A ROAD USING SINGLE LANE CHANGE ANALYSIS TYPE

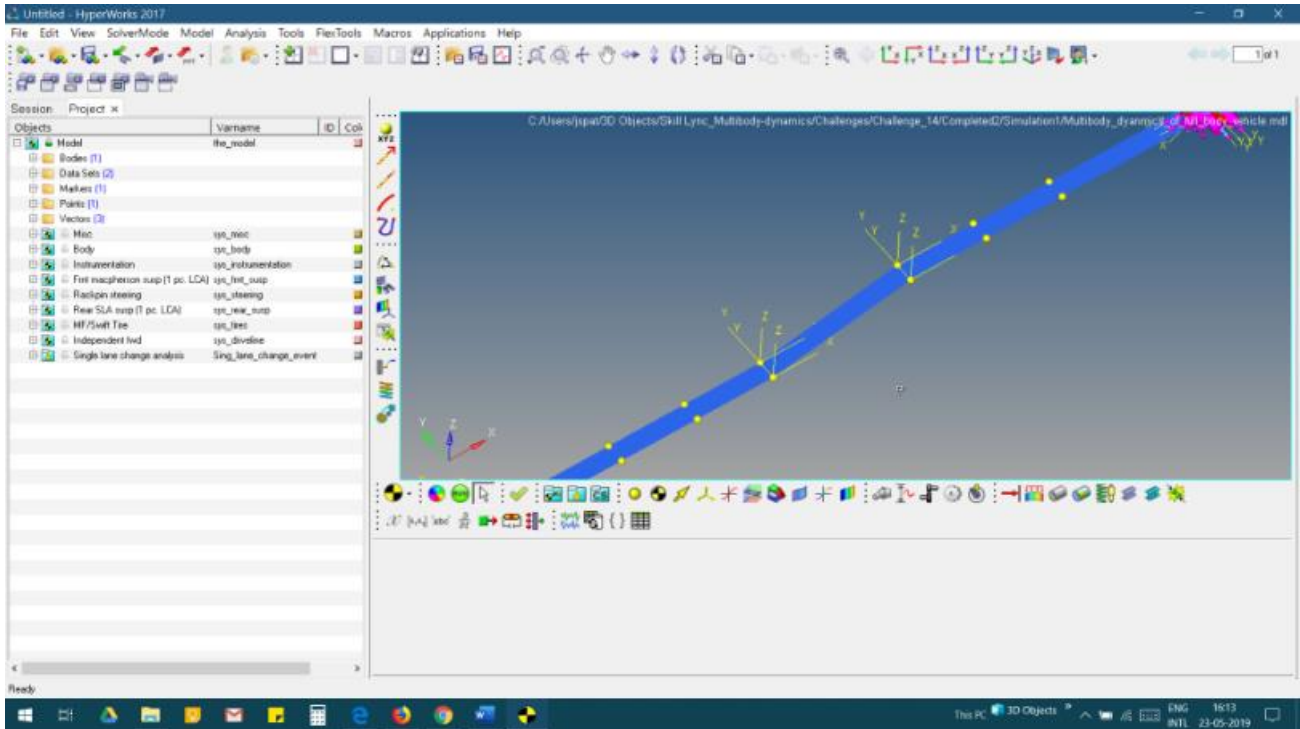


Figure 3

#### CASE 2 – FULL VEHICLE BODY WITH SLA SUSPENSION SYSTEM ON A ROAD USING DOUBLE LANE CHANGE ANALYSIS TYPE

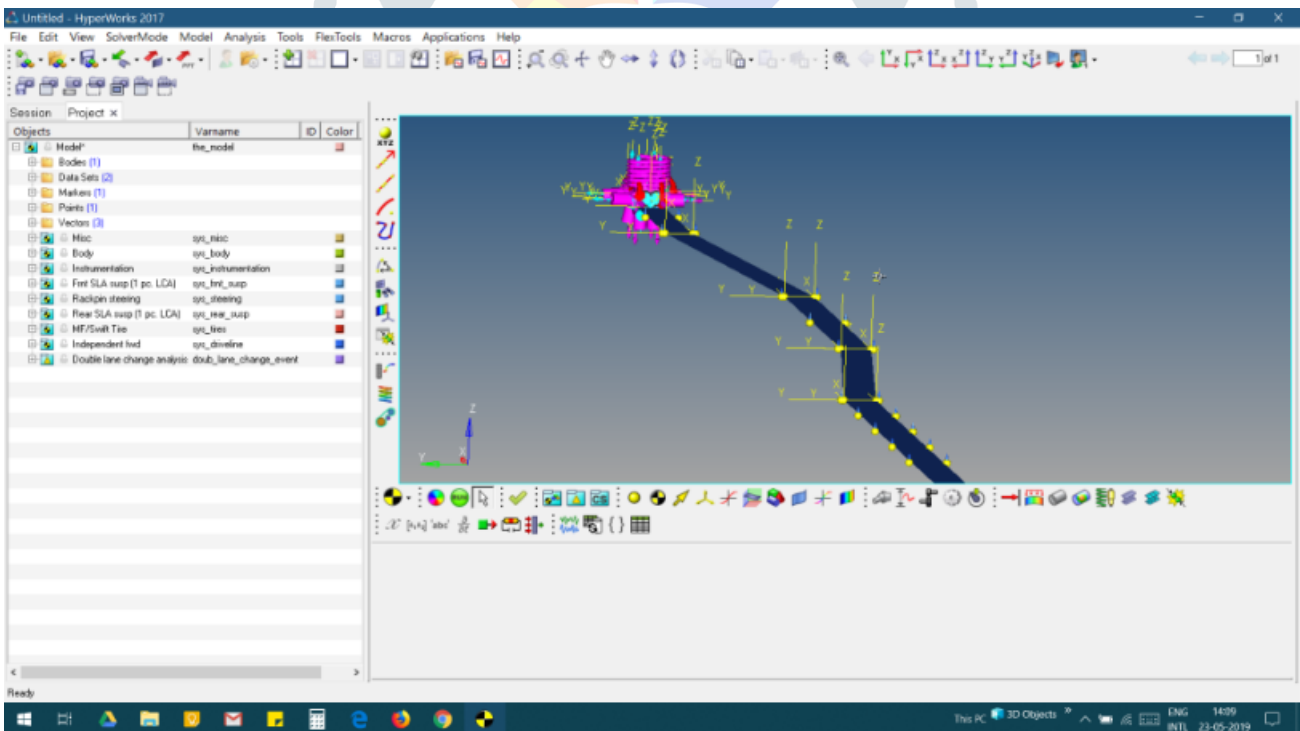


Figure 4

### 4. CASES SIMULATION

#### 4.1. CASE 1 simulation —

Analysis of the Full vehicle was done with single lane change road type as shown in the image above.

To begin with, the model type for the simulation was chosen. Here, it was Full vehicle body along with SLA suspension system built in.

By going to Model—Model Type—Full Vehicle

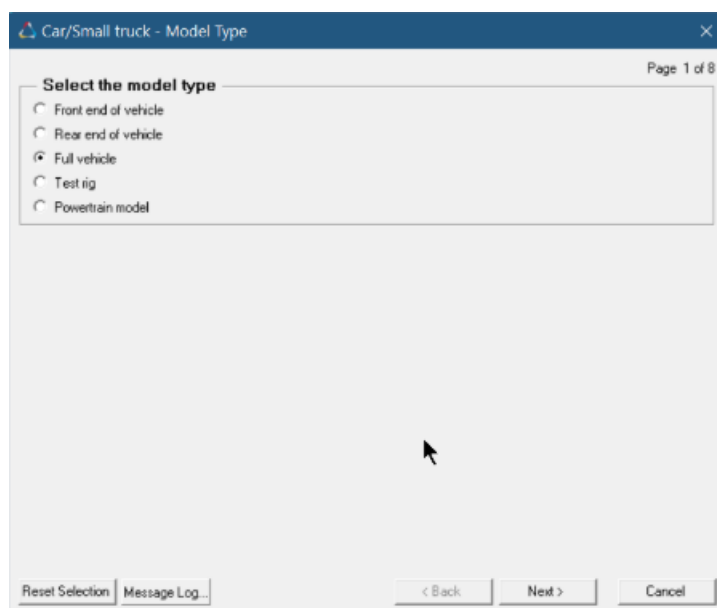


Figure 5

After that by going next and choose the FRONT WHEEL DRIVELINE.

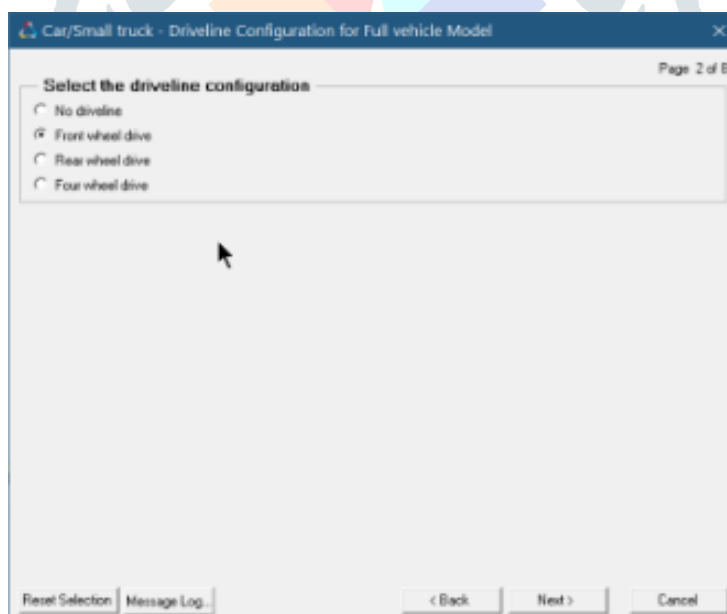


Figure 6

Going further, we choose both front and rear wheel suspension type as-

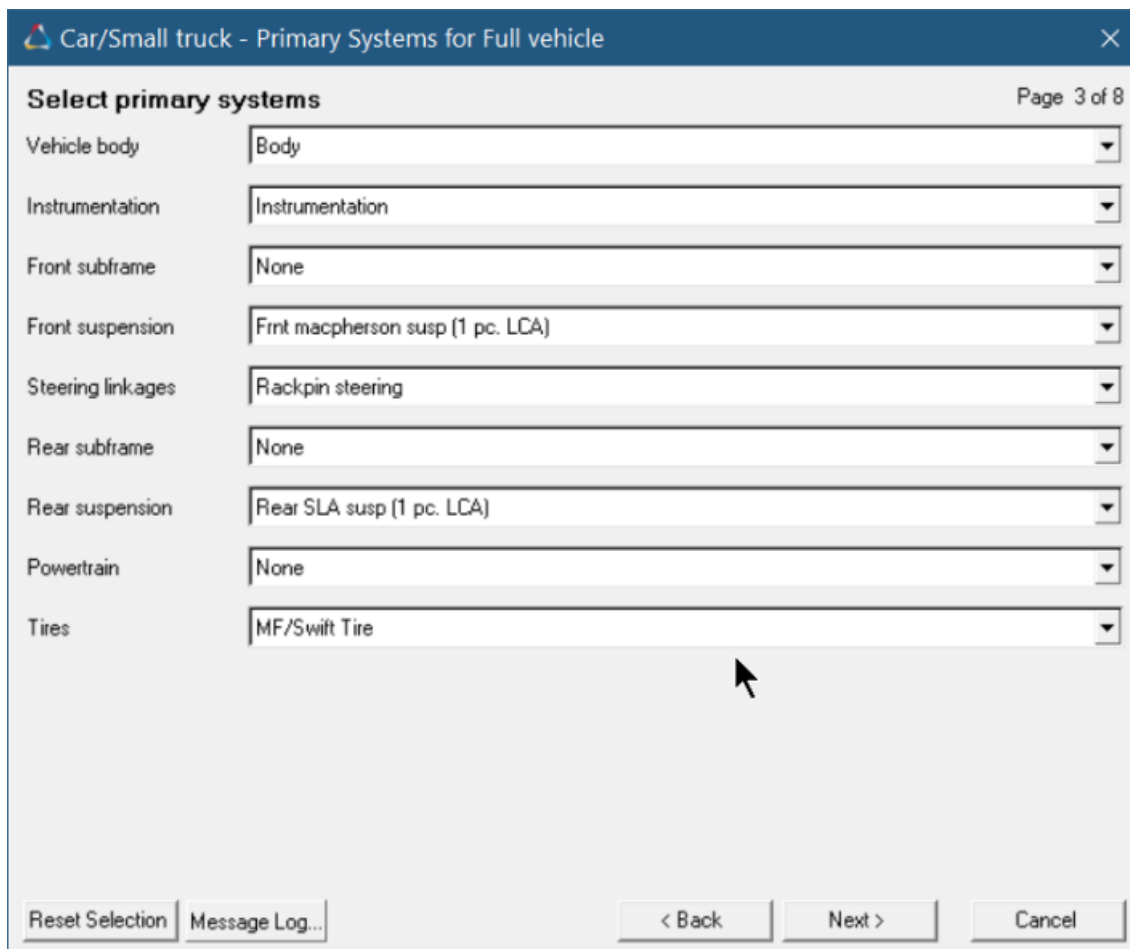


Figure 7

Then further keeping rest of the option at the default stage, we got a Full vehicle Graphic model along with the whole suspension system inbuilt with the type of suspension we chose.

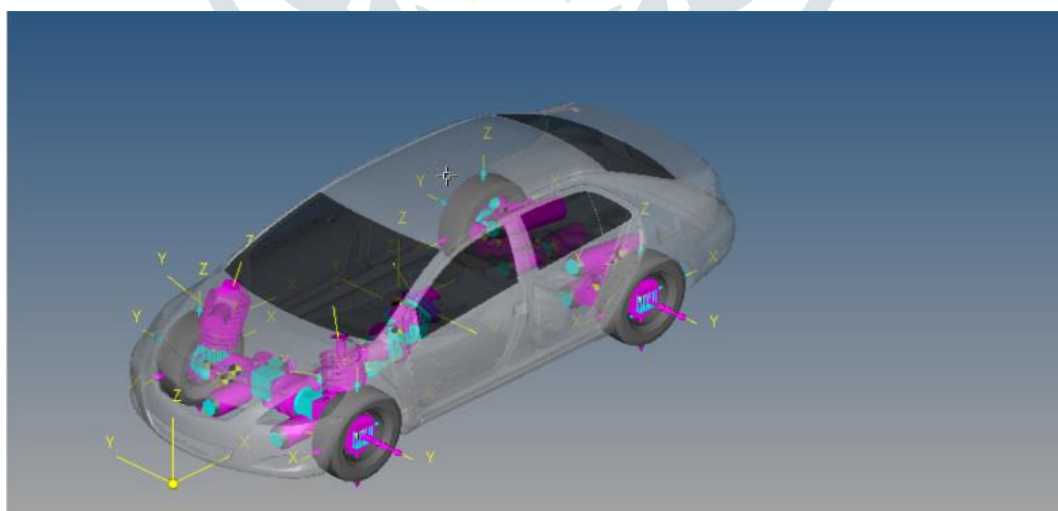


Figure 8

Further, the following Parameter in the options was modified by going into the model and DATA Summary option-

a. Vehicle Body CG – 235

	LEFT			Sym	RIGHT		
	X	Y	Z		X	Y	Z
Model							
Global Origin	0.000	0.000	0.000				
Misc							
Body							
Vehicle Body CG	235	0.000	1250.000				
Instrumentation							
Fmt macpherson susp (1 pc. LCA)							
Wheel CG	1000.000	-750.000	1000.000	<input checked="" type="checkbox"/>	Symmetric	Symmetric	Symmetric
Wheel center	1000.000	-750.000	1000.000	<input checked="" type="checkbox"/>	Symmetric	Symmetric	Symmetric
Spindle align	1000.000	-650.000	1000.000	<input checked="" type="checkbox"/>	Symmetric	Symmetric	Symmetric
Knuckle CG	1000.000	-650.000	1000.000	<input checked="" type="checkbox"/>	Symmetric	Symmetric	Symmetric
LBJ	991.000	-668.000	916.000	<input checked="" type="checkbox"/>	Symmetric	Symmetric	Symmetric

Figure 9

b. Couplers –

Rack coupler ratio – 9.500

Pinion Coupler ratio -- - 3.0

	LEFT	Sym	RIGHT
	LRatio		RRatio
Model			
Misc			
Body			
Instrumentation			
Fmt macpherson susp (1 pc. LCA)			
Fmt strut (with inline its)			
Rackpin steering			
Rack/Pinion Coupler	9.500		
Steering column 1 (not for abaqus)			
Rear SLA susp (1 pc. LCA)			
Rear shock absorber ( with inline its )			
Rear coil spring			
MF/Swift Tire			
Independent fwd			
Differential Coupler	-0.500		
Pinion Coupler	3.0		
Single lane change analysis			
Steer controller			
Drive torque controller			

Figure 10

c. Spring Dampers-

Front Struct---

Preload - 3700 for Coil Spring

and Stiffness to 30.00

Default values

Data Summary						
System Model						
	LEFT				Sym	Stif
	Stiffness	Damping	Preload	Length/Angle		
Model						
Misc						
Body						
Instrumentation						
Front macpherson susp (1 pc. LCA)						
Front strut (with inline its)						
Coil spring	30.000	0.000	3700	156.429	<input checked="" type="checkbox"/>	Symmet
Strut damper	0.000	1.000	0.000	0.000	<input checked="" type="checkbox"/>	Symmet
Rackpin steering						
Steering column 1 (not for abaqus)						
Rear SLA susp (1 pc. LCA)						
Rear shock absorber (with inline its)						
Shock damper	0.000	1.000	0.000	0.000	<input checked="" type="checkbox"/>	Symmet
Rear coil spring						
Coil spring	90.000	0.000	6150.000	254.951	<input checked="" type="checkbox"/>	Symmet
MF/Soft Tire						
Independent fwd						

Figure 11

Further, we choose the Analysis type which is -- **Single Lane Change type analysis**

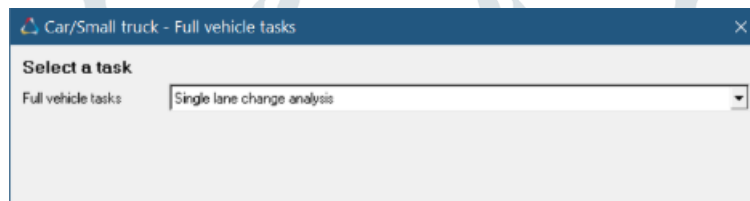


Figure 12

Thus, finally after changing the default values and choosing the simulation type, we get the final setup as shown below,

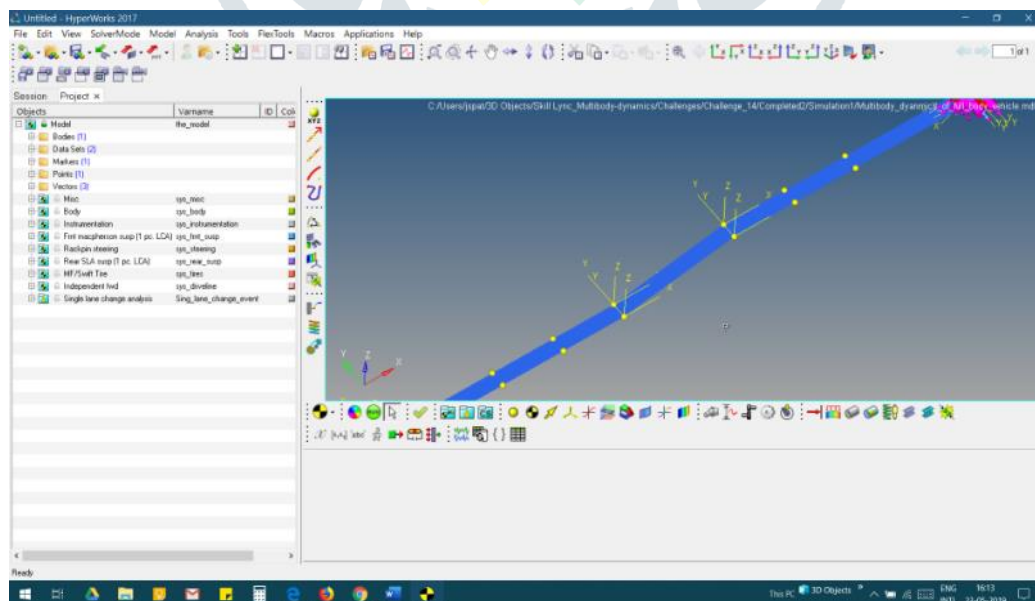


Figure 13

Now, the parameters which were changed from the default parameters in case 1 are,

The parameters that were changed for this analysis were-

a. Vehicle Mass – 1500 Kg i.e. 15000 N

b. Vehicle Velocity- 70 Mph

Front Cornering Stiffness – 48000

Rear Cornering Stiffness – 48000

Then, the analysis was performed, at default Simulation time and time step which was successful.

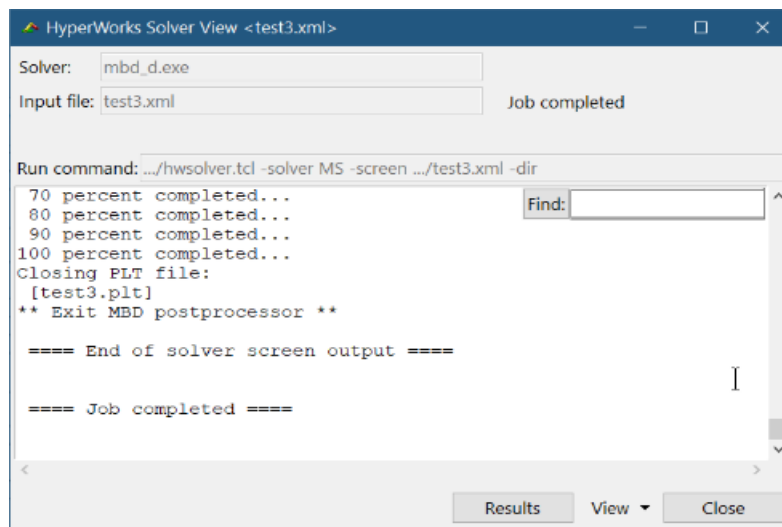


Figure 14

A systematic report was generated after the simulation. And the changes were observed.

#### 4.2. CASE 2 simulation-

Analysis of the Full vehicle was done with single lane change road type as shown in the image above.

The procedure followed is the same, but, instead of Analysis type- Single Lane Change Type, we choose, Double lane change analysis. The use of multi-body dynamics in vehicle handling analysis approach is particularly useful under transient conditions arising from the application of a steering function. A non-linear dynamic analysis for a realistic multi-degrees-of-freedom vehicle model subjected to a step steering function has been presented. The results of such an analysis can be used to measure the vehicle 'responsiveness' in terms of given parameters specified in ISO and BS standards for non-steady conditions. The extent of lateral acceleration overshoot roll angle variation and generated vertical tyre forces, particularly on the inside wheels, in turn can provide a good measure of vehicle stability. The simulation highlighted in the paper indicates that enough tyre forces are generated, ensuring vehicle adherence to its path. The change in the roll angle is within the specified limit for the type of vehicle under investigation. The vehicle lateral acceleration overshoot is 0.07 g and its response time to the steering input is adequately short. The procedure highlighted in this paper can be employed for simulation of vehicle models under transient manoeuvres. A large amount of proprietary physical and geometrical data has been included in the paper which can be used by others who intend to carry out similar vehicle handling studies.

To begin with, the model type for the simulation was chosen. Here, it was Full vehicle body along with SLA suspension system-built in. Keeping the initial properties of the double lane change simulation like that of the first case, the parameters that are changed are –

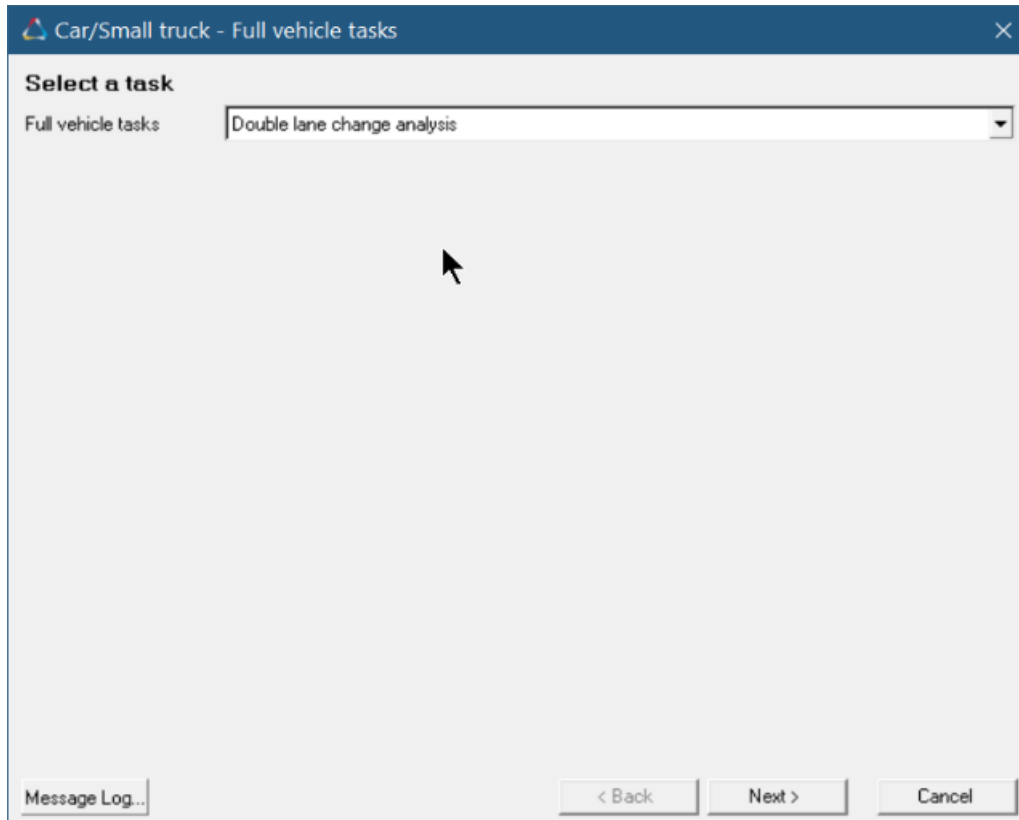


Figure 15

- a. Weight- 2500 kg i.e. 25000 N
- b. Vehicle Velocity on the track, Velocity- 70 Mph

Thus, after changing the default values and choosing the simulation type, we get the final setup as shown below,

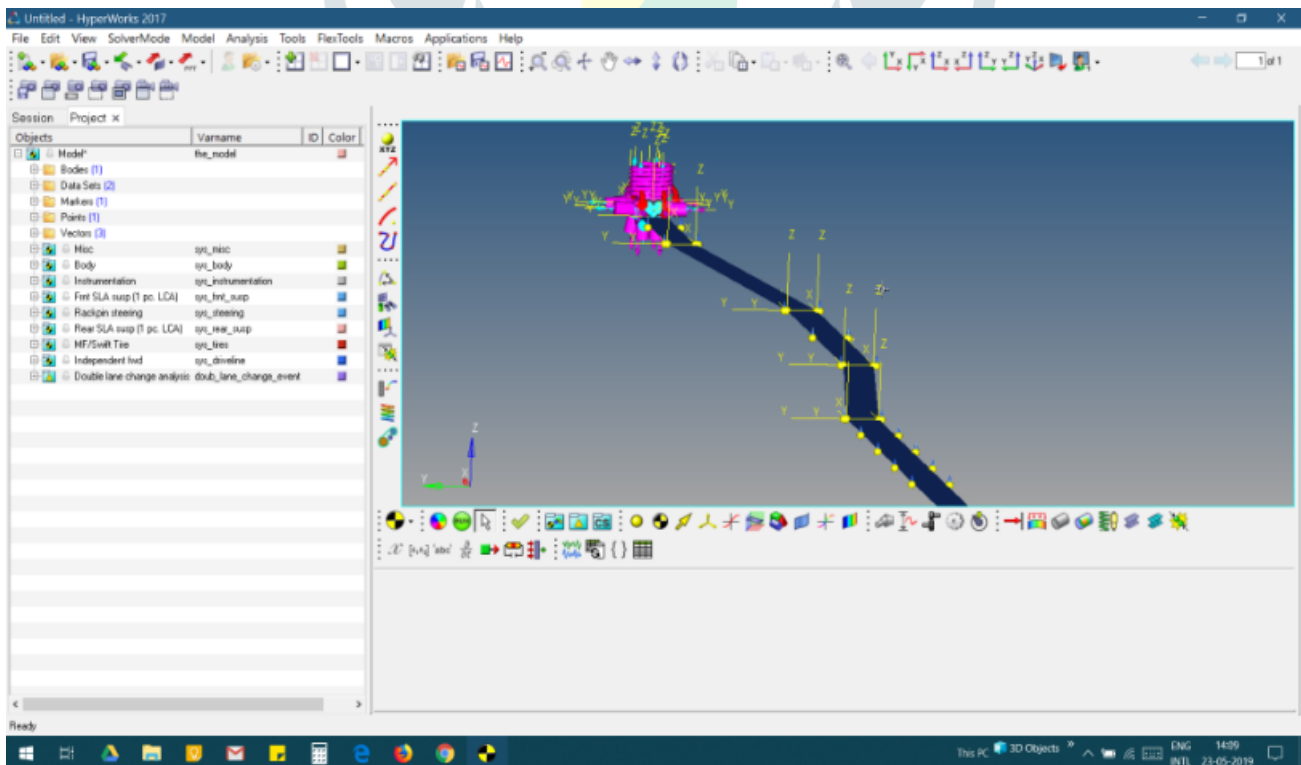


Figure 16

5. RESULTS

5.1. Results for Case 1

Therefore we can observe here that, because of the change in certain parameter and most important parameters such as CG and the Velocity of the Vehicle during the analysis, there is the condition of slippage at the end of the lane change, but later on the Vehicle tend to become stable further as seen in the Hypergraph which is plotted in the Report, which is the CG displacement. Thus, in this way, by simulation the problems can be rectified and thereby changing the parameters, can be resolved easily.

a. Vehicle CG Displacement

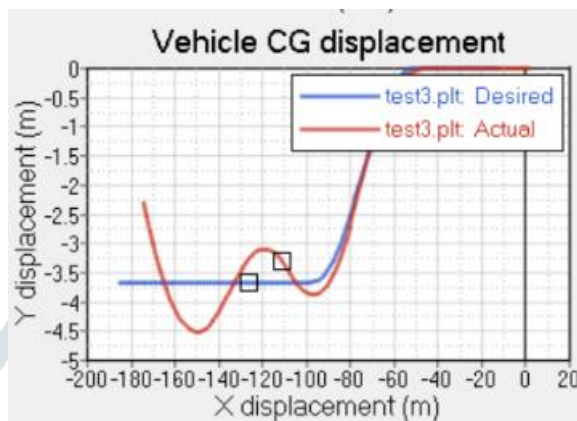


Figure 17

Here, in Case 1, we can observe that the desired line graph is initially a straight line, but this is the ideal condition when all the parameters of the simulation are ideal, while the actual line graph is aberrated by 2.5 meter on Y axis initially, whereas, it's steady along the x axis of the graph. The initial unsteadiness is caused mainly due to change in the vehicle velocity along the lane change from default 40 Mph to 70 Mph as well as in the Tyre type as well as the suspension system that was used in the modified system which can be easily corrected. Point (-100, -3.5) is the point of the single lane change whereas, the graph of desired output is for the ideal condition with slow speed.

b. Lateral Acceleration Vs Time

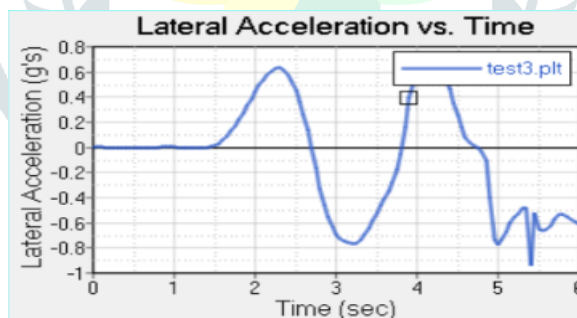


Figure 18

c. Yaw rate vs. Time

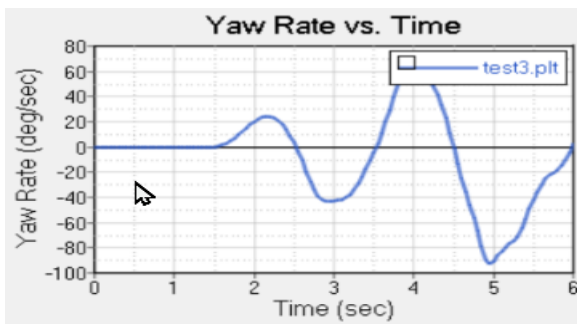


Figure 19

d. Steering wheel torque Vs Time

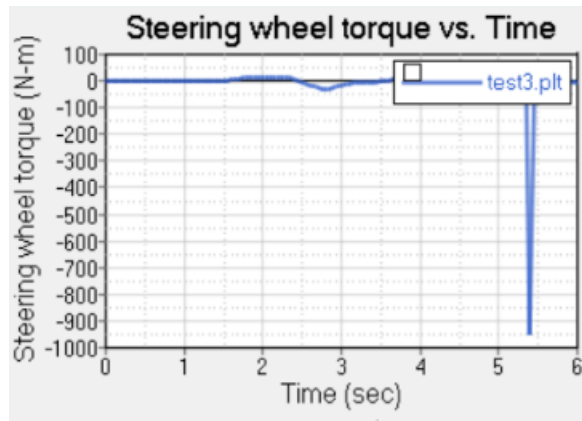


Figure 20

Here, the line graph of steering wheel torque is steady along 0 value with a slight disturbance due to the change in the single lane of the vehicle.

Overall result obtained is shown below –

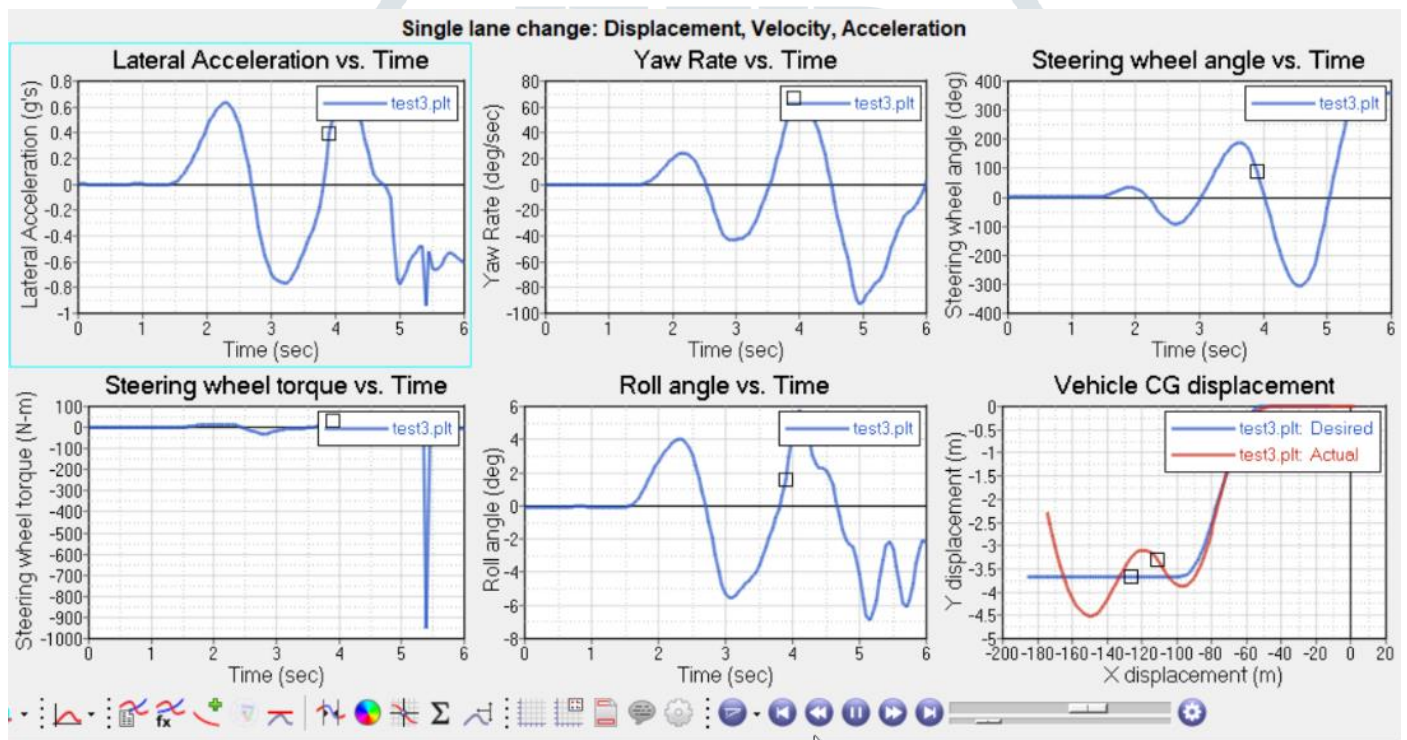


Figure 21

5.2. Results for Case 2

Therefore we can observe here that, because of the change in certain parameter and most important parameters such as Weight and the Velocity of the Vehicle during the analysis, there is the no condition of slippage as that was observed during the first of the Simulation, and the Vehicle seems to be stable throughout the simulation and the Hypergraph of the vehicle CG also predicts the same thing.

Thus, in this way, by simulation the problems can be rectified and thereby changing the parameters, can be resolved easily.

A. Vehicle CG Displacement

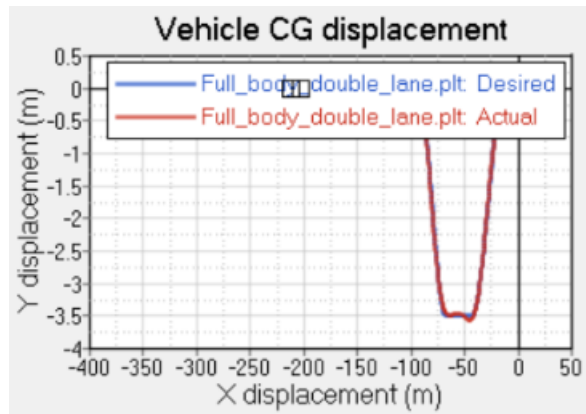


Figure 22

Here, unlike the result obtained in the case 1 simulation of single lane change analysis, the graph obtained in double lane change analysis is exact like the desired graph with ideal parameters. Though the velocity of the vehicle was increased by 30 Mph, which is to 70 Mph, the var managed to stay on track and maintain its stability. This is because of the change in suspension system from the previous model, in this case, the suspension system in the front as well as in the rear was same, which is Front SLA suspension System and Rear SLA suspension System, which helped the vehicle to gain the stability at higher speed during lane change.

B. Lateral Acceleration Vs Time

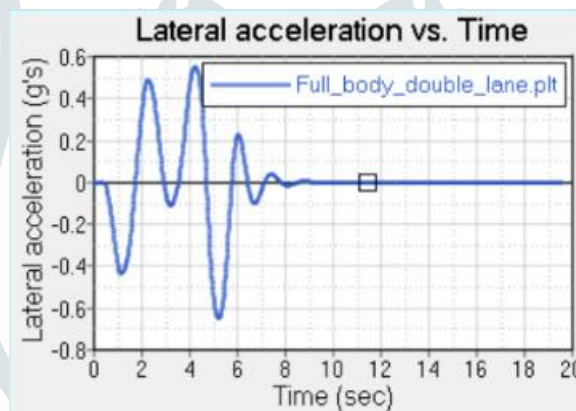


Figure 23

C. Yaw rate Vs Time

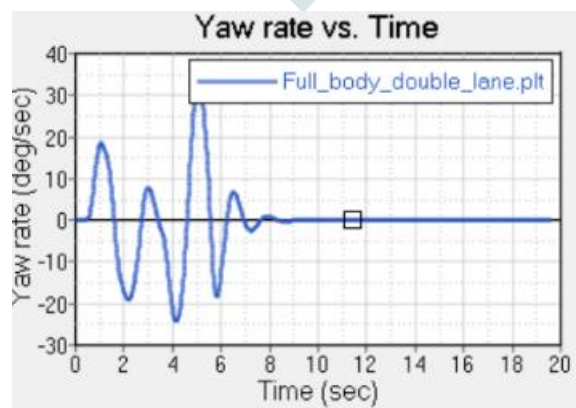


Figure 24

## D. Steering Wheel Torque Vs Time

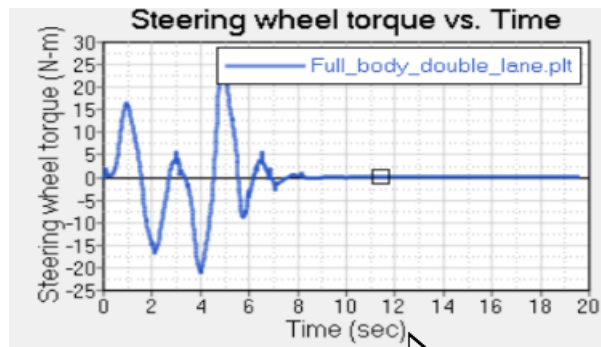


Figure 25

Overall result obtained is shown below –

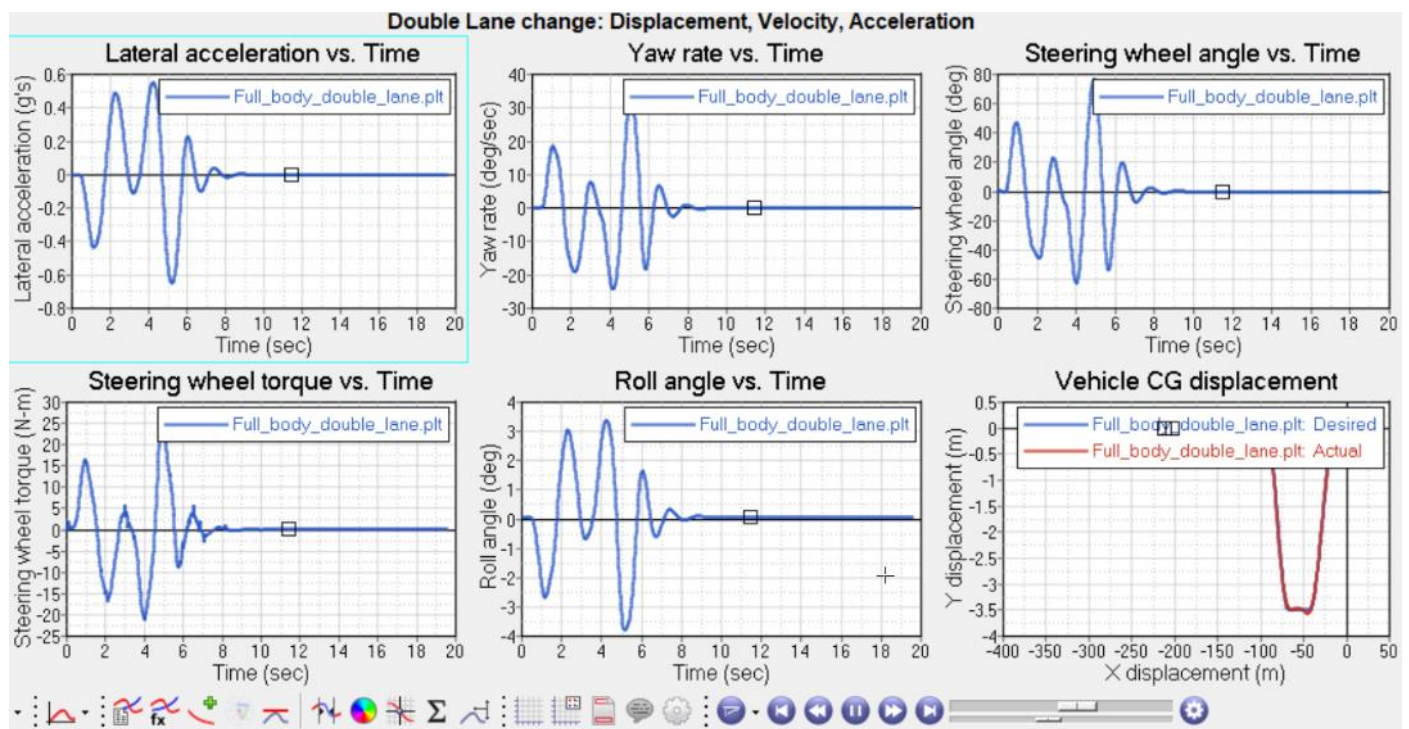


Figure 26

## 6. CONCLUSION

From the simulation contrast analysis above, we can see that Motion-view as analysis module used specially for vehicle dynamics, because each process from the mechanics modeling to dynamic simulation analysis, are parametric process, it can quickly and easily establish the physical prototype development scheme. Therefore, Motion-view and Motion-solve has promising application prospect. Furthermore, according to the above analysis, we can obtain following conclusions.

1. The application of simulation software Hyper works – Motion-view/Motion-Solve, including the establishment of the vehicle steering system, front and rear suspension assembly, vehicle simulation model of body and tires, car, can simulate more real in the steering wheel to the steering characteristic of sinusoidal input conditions, provides the simulation as an effective method for handling and stability analysis of automobile.
2. The car in low velocity turning lane changing, along with the increase of the steering angle and vehicle side-slip angle and yaw rate and lateral acceleration change amplitude increasing, still did not exceed the range of stability; the car at high velocity turning or changing lanes too large steering angle to make the car lost control stability. Reduce velocity and reduce the steering angle is the most secure scheme.
3. Velocity and steering angle will make the vehicle lateral motion increases, performance depends not only on the automobile tire and suspension components to ensure the operation stability, you need to add the electronic stability program electronic control system to regulate and enhance vehicle handling and stability.

**7. REFERENCES**

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