# Modeling, Simulation and Validation of 14 DOF Full Vehicle Model


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**Abstract- An accurate full vehicle model is required in representing the behavior of the vehicle in order to design vehicle control system such as yaw control, anti roll control, automated highway system etc. There are many vehicle models built for the study of the vehicle dynamics specifically for the ride and handling behavior. This paper describes the vehicle model development of the vehicle model to study the behavior of the vehicle. The derivation of a 14 DOF vehicle model consisting of ride, handling and tire model is presented. The Magic tire formula was used as tire model. All the assumptions made for the 14 DOF vehicle model are stated. This 14 DOF vehicle model will be then validated using instrumented experimental vehicle for two steering inputs namely step steer and double lane change. The deviation of the outputs specifically the yaw rate, lateral acceleration and roll angle of the vehicle body and also the slip angle at each of the tire from the 14 DOF model simulation from the experimental results is discussed.**



**Keywords: ride, handling, tire model, instrumented experimental vehicle, step steer**


## I. INTRODUCTION

An accurate vehicle dynamics model should be built to represent the actual vehicle behavior and to be validated with vehicle dynamics simulation software and an instrumented real experimental car. If the behavior of the vehicle is not predicted when designing a vehicle, it can lead to improper handling behavior and threatening maneuver such as rollover. With mathematical models, the dynamic behavior and the safety of the vehicle can be investigated. With computer simulation tool, the vehicle dynamic behavior and safety can be investigated without the need to built or test a vehicle which is very costly.

The objective of this project is to develop, simulate and validate a 14 degree of freedom vehicle dynamics model with experimental data for the study of performance, ride, and handling of four wheel vehicles.

## II. VEHICLE MODELING

A comprehensive 14 DOF vehicle model that includes dynamic of roll center and nonlinear effect due to vehicle geometry change was developed for the study of roll dynamics by Shim [7]. The tire model used was the Pacejka tire model. This vehicle model was validated for three types of vehicle dynamics test namely step steer and, ramp steer and J turn test using CarSim and ADAMS/Car. The limitation, simplified equation validity and assumption of various modeling was discussed by analyzing their effect on the model roll response for step steer, ramp steer and J turn test. This paper presents the development of 14 DOF vehicle model by first comparing three tire models available and validation with an instrumented experimental vehicle for step steer and double lane change steering inputs.

### *Vehicle Model*

The 14 DOF vehicle model shown in Fig. 1 is used to study the vehicle behavior in longitudinal, lateral and vertical directions consists of a sprung mass of vehicle body and four unsprung masses of the wheels. The sprung mass has 6 DOF includes the longitudinal, lateral, vertical, roll, pitch, and yaw motion. Each of the wheels is allowed to have 2 DOF which consist of the vertical motion of the wheel and the wheel spin.

### *Modeling Assumptions*

Lumped mass is used to represent the sprung and unsprung masses. The vehicle body is being modeled as rigid. The outer and inner steer angle is assumed to be the same. The tires are assumed to having contact with the ground all the time.

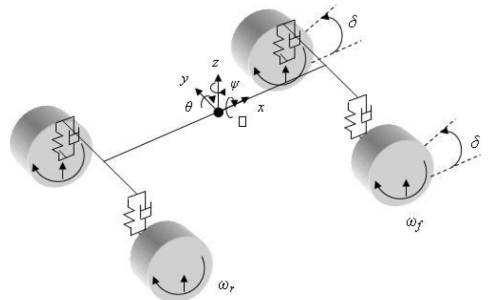


Fig. 1. 14 DOF Vehicle Model

### *Ride Model*

The ride performance parameters are the body acceleration and displacement, damper displacement and wheel acceleration. The vehicle ride model [6] shown in Fig. 2 has 7 DOF, is made up of the vehicle body which is connected to four wheels by the spring and damper at each corner. The vehicle body is allowed to have 3 DOF consisting of vertical, roll and pitch motion.

Summation of the vertical forces of the sprung mass:

$$\sum F_v = m_s \ddot{Z}_s$$

$$F_{fl} + F_{fr} + F_{rl} + F_{rr} = m_s \ddot{Z}_s$$

$$F_{sfl} + F_{dfl} + F_{sfr} + F_{dfr} + F_{srl} + F_{drl} + F_{srr} + F_{drr} = m_s \ddot{Z}_s \quad (1)$$

Summation of moment for pitching:

$$(F_{srl} + F_{drl} + F_{srr} + F_{drr})b - (F_{sfl} + F_{dfl} + F_{sfr} + F_{dfr})a = I_p \ddot{\theta} \quad (2)$$

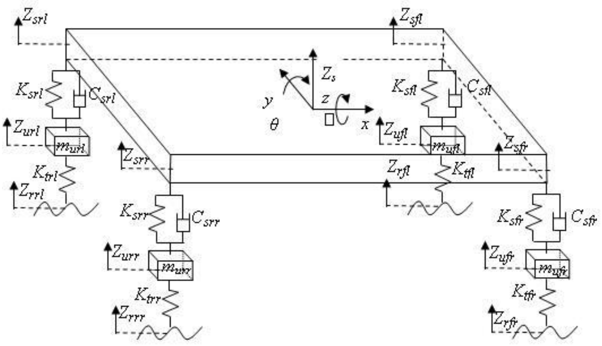


Fig. 2. Vehicle Ride Model

Summation of moment for rolling:

$$(F_{sfl} + F_{dfl} + F_{srl} + F_{drl})\frac{w}{2} - (F_{sfr} + F_{dfr} + F_{srr} + F_{drr})\frac{w}{2} = I_r \ddot{\phi} \quad (3)$$

Summation of vertical forces at front left:

$$F_{tfl} - F_{sfl} - F_{dfl} = m_{ufl} \ddot{Z}_{ufl} \quad (4)$$

Summation of vertical forces at front right:

$$F_{tfr} - F_{sfr} - F_{dfr} = m_{ufr} \ddot{Z}_{ufr} \quad (5)$$

Summation of vertical forces at rear left:

$$F_{trl} - F_{srl} - F_{drl} = m_{url} \ddot{Z}_{url} \quad (6)$$

Summation of vertical forces at rear right:

$$F_{trr} - F_{srr} - F_{drr} = m_{urr} \ddot{Z}_{urr} \quad (7)$$

The normal force acting on each tire is given below.

$$F_{zfl} = \frac{m_s g b}{2(a+b)} + m_{ufl} g + F_{fl}$$

$$F_{zfr} = \frac{m_s g b}{2(a+b)} + m_{ufr} g + F_{fr}$$

$$F_{zrl} = \frac{m_s g a}{2(a+b)} + m_{url} g + F_{rl}$$

$$F_{zrr} = \frac{m_s g a}{2(a+b)} + m_{urr} g + F_{rr} \quad (8)$$

*Handling Model*

The handling model as shown in Fig. 3 consists of 7 DOF. The vehicle body has 3 DOF: longitudinal, lateral and yaw motions. The equation of motion for longitudinal, lateral and yaw is similar to equations used in [1].The remaining 4 DOF correspond to the spin of each wheel.

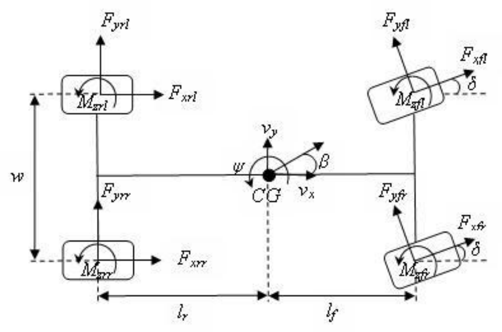


Fig. 3. Vehicle Handling Model

$a_x$ ,which is the inertial acceleration at the center of gravity of the vehicle in the direction of x axis is made up of two terms. The two terms are the acceleration which is due to the motion along x axis, $\dot{v}_x$ and the centripetal acceleration, $v_y \dot{\psi}$.

$$a_x = \dot{v}_x - v_y \dot{\psi} \quad (9)$$

Applying Newton's Second Law of motion, the following equation shows the summation of forces in longitudinal direction.

$$m_t a_x = F_{xfl}\cos\delta - F_{yfl}\sin\delta + F_{xfr}\cos\delta - F_{yfr}\sin\delta + F_{xrl} + F_{xrr} \quad (10)$$

$a_y$, which is the inertial acceleration at the center of gravity of the vehicle in the direction of y axis is made up of two terms. The two terms are the acceleration which is due to the motion along y axis, $\dot{v}_y$ and the centripetal acceleration, $v_x \dot{\psi}$.

$$a_y = \dot{v}_y + v_x \dot{\psi} \quad (11)$$

The following equation shows the summation of forces in lateral direction.

$$m_t a_y = F_{yfl}\cos\delta + F_{xfl}\sin\delta + F_{yfr}\cos\delta + F_{xfr}\sin\delta + F_{yrl} + F_{yrr} \quad (12)$$

The slip angles at the front and rear tire are obtained from the handling free body diagram.

$$\alpha_f = \tan^{-1}\left(\frac{v_y + l_f \dot{\psi}}{v_x}\right) - \delta \quad (13)$$

$$\alpha_r = \tan^{-1}\left(\frac{v_y - l_r \dot{\psi}}{v_x}\right) \quad (14)$$

Front tire longitudinal velocity is required to obtain the longitudinal slip

$$v_{wxf} = V_{tf}\cos\alpha_f \quad (15)$$

where the speed of the front tire is given by the equation below

$$V_{tf} = \sqrt{(v_y + l_f \dot{\psi})^2 + v_x^2} \quad (16)$$

Rear tire longitudinal velocity is required to obtain the longitudinal slip.

$$v_{wxr} = V_{tr}\cos\alpha_r \quad (17)$$

where the speed of the rear tire is given by the equation below

$$V_{tr} = \sqrt{(v_y - l_r \dot{\psi})^2 + v_x^2} \quad (18)$$

The longitudinal slip used in this mathematical model is under braking condition because the pitch motion is assumed to be positive when braking.

$$S_{af} = \frac{v_{wxf} - \omega_f R_w}{v_{wxf}} \quad (19)$$

$$S_{ar} = \frac{v_{wxr} - \omega_r R_w}{v_{wxr}} \quad (20)$$

The yaw equation of motion is given by the following equation. The aligning moment, $M_z$ is assumed to have the same direction with the yaw motion.

$$I_z\ddot{\psi} = [-\frac{w}{2}F_{xfl}\cos\delta + \frac{w}{2}F_{xfr}\cos\delta - \frac{w}{2}F_{xrl} + \frac{w}{2}F_{xrr} + \frac{w}{2}F_{yfl}\sin\delta - \frac{w}{2}F_{yfr}\sin\delta + l_f F_{xfl}\sin\delta + l_f F_{yfl}\cos\delta + l_f F_{xfr}\sin\delta + l_f F_{yfr}\cos\delta - l_r F_{yrl} - l_r F_{yrr} + M_{zfl} + M_{zfr} + M_{zrl} + M_{zrr}] \quad (21)$$

The longitudinal acceleration of the vehicle shown in Fig. 4 contributes to the pitch motion whereas the lateral acceleration causes the roll motion shown in Fig. 5. The pitch acceleration, $\ddot{\theta}$ can be determined from the Fig. 5. Summation of moment about the y-axis passing through pitch center is given as follow.

$$I_{sy}\ddot{\theta} = m_s a_x h + m_s g h\theta - k_\theta\theta - \beta_\theta\dot{\theta} \quad (22)$$

The roll acceleration, $\ddot{\phi}$ can be determined from the above free body diagram. Summation of moment about the x-axis passing through roll center is given as follow.

$$(I_r + m_s(h - h_{rc})^2)\ddot{\phi} = m_s a_y(h - h_{rc})\cos\phi + m_s g(h - h_{rc})\sin\phi - k_\phi\phi - \beta_\phi\dot{\phi} \quad (23)$$

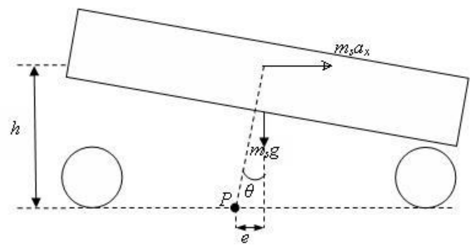

Fig. 4. Pitch Motion Due to Longitudinal Acceleration

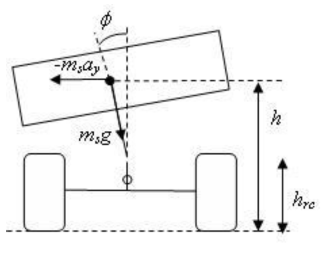

Fig. 5. Roll Motion Due to Lateral Acceleration

The degree of freedom of the spin of the tire is represented by the wheel angular velocity, $\omega$ as in Fig. 6. The summation of torque about wheel axle for each wheel is given by the following equation.

$$I_w\dot{\omega}_{fl} = T_{dfl} - T_{bfl} - F_{xfl}R_w$$

$$I_w\dot{\omega}_{fr} = T_{dfr} - T_{bfr} - F_{xfr}R_w$$

$$I_w\dot{\omega}_{rl} = T_{drl} - T_{brl} - F_{xrl}R_w$$

$$I_w\dot{\omega}_{rr} = T_{drr} - T_{brr} - F_{xrr}R_w \quad (24)$$

*Effect of Slip and Camber Angle*

When the tire rolls at a camber angle and zero slip angle, the lateral force generated is known as the camber thrust, $F_y$ as shown in the Fig. 7. The total lateral force generated is the sum of the lateral force due to the slip angle and the camber thrust and both acts in the same direction.

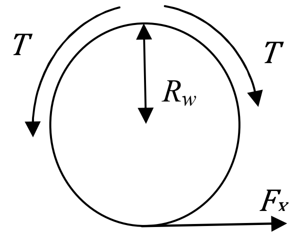

Fig. 6. Roll Motion Due to Lateral Acceleration

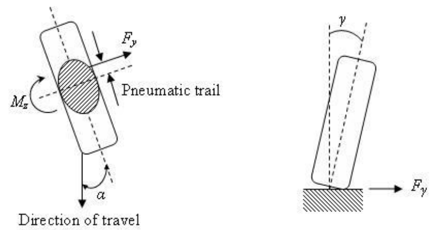

Fig. 7. Lateral Force and Aligning Moment Due to Slip Angle and Camber Thrust due to Camber Angle

## III. SIMULATION AND VALIDATION OF MODEL

Two types of the vehicle dynamics test were carried out for the purpose of validation using an instrumented experimental vehicle. The two tests are the step steer and double lane change. The steep steer test will be carried out at 180 degrees step steer at 35 kph. The double lane change test will be carried out at a constant speed of 80 kph. The steering input angle for the double lane change for the Simulink model will be taken from the steering wheel sensor as shown in Fig. 8.

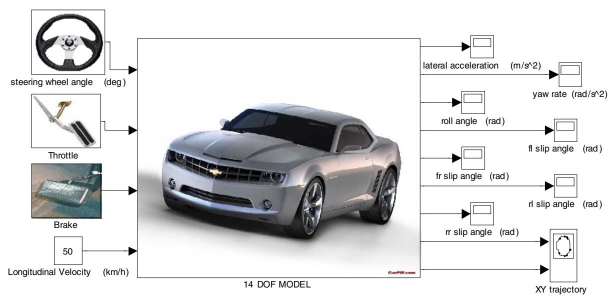

Fig. 8. 14 DOF vehicle Simulink model

The output response to be analyzed for step steer and double lane change will be yaw rate, lateral acceleration and roll angle of the vehicle body and also the slip angle at each of the tire. The difference between the simulation and the experimental results will be discussed.

The 14 DOF Vehicle Model was validated using practical experimental data which was conducted by the Smart Material and Automotive Control Lab of Universiti Teknikal Malaysia Melaka.

## III. RESULT AND DISCUSSION

*Step Steer Test*

The step steer was done using an instrumented vehicle for 180 degrees step steer angle at a speed of 35 kph. The steering wheel angle input for this test which is obtained from the steering wheel sensor is shown in Fig. 9. The steering wheel angle is negative because the steering angle input was given clockwise but the model assume counterclockwise steering angle as positive. It can be seen that the step steer is not exactly constant 180 degrees due to the difficulty for the driver to maintain the steering angle.

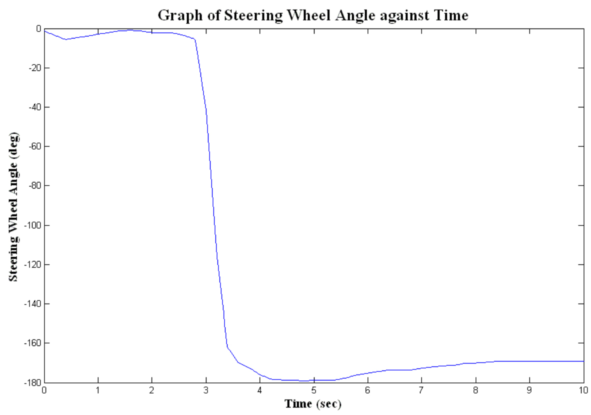


Fig. 9. Steering Angle Input for 180° Step Steer Test

The Figures 10 to 16 show the results of step steer simulation and the experimental data. The trend between the simulation and experiment results is almost the same with acceptable error. This error is due to the simplification in the vehicle dynamics model compared to real vehicle.

In terms of lateral acceleration and yaw rate as shown in Figures 10 and 11 respectively, the simulation results follow the experiment results quite closely.

In terms of the roll angle response as shown in Fig. 12, the simulation results has similar trend with the experimental results but difference in magnitude. The difference is magnitude is because of the simplification done in the vehicle modeling. One of the source of error is the effect of anti-roll bar is ignored in the simulation. Anti-roll bar reduces the roll angle of the vehicle, which is representation of the vertical motion.

The tire slip angle responses of the front and rear right tire is presented in Figures 13 and 14 consecutively. From all the slip angle responses, the experimental slip angle response is higher than the simulation slip angle response for both transient and steady state. This difference is because it was difficult for the driver to maintain the speed throughout the test. In the simulation, the vehicle was assumed to be travelling with a constant speed of 35 kph during the maneuver.

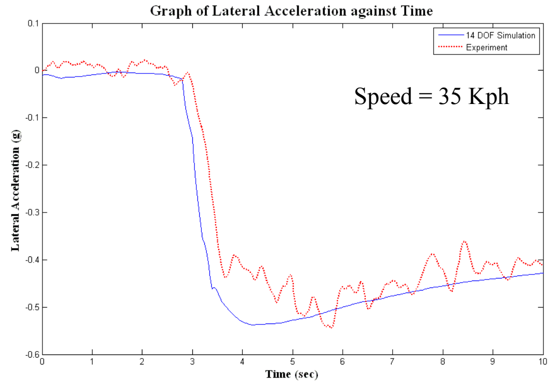


Fig. 10. Lateral Acceleration Response for 180° Step Steer

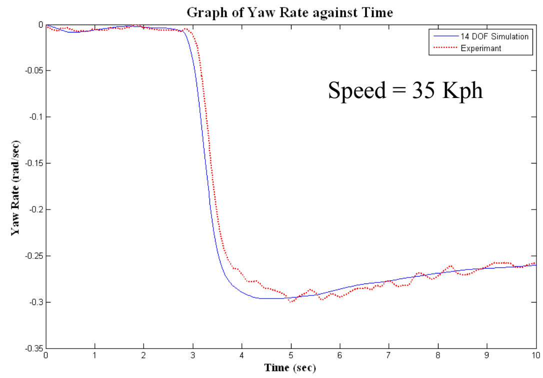


Fig. 11. Yaw Rate Response for 180° Step Steer Test

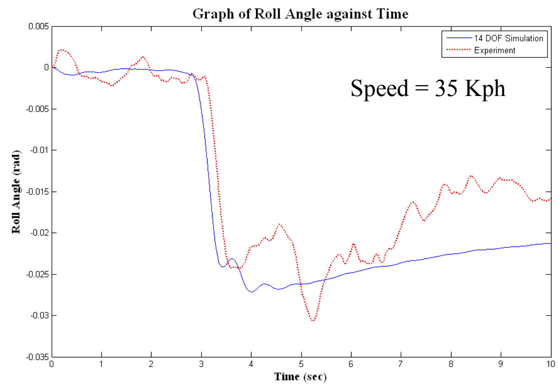


Fig. 12. Roll Angle Response for 180° Step Steer Test

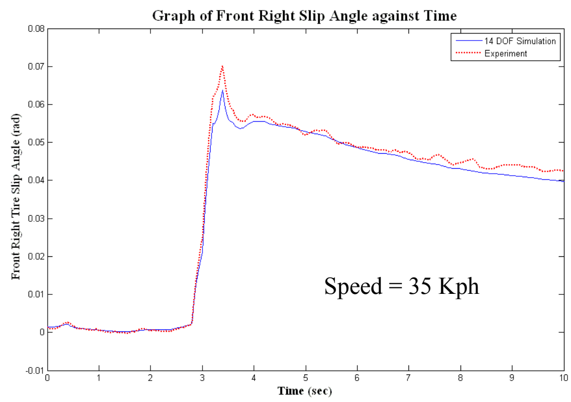


Fig. 13. Front Right Tire Slip Angle for 180° Step Steer Test

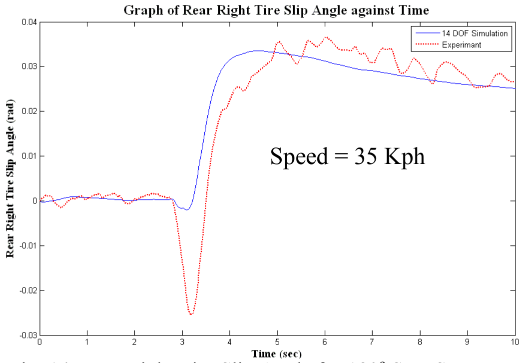

Fig. 14. Rear Right Tire Slip Angle for 180° Step Steer Test

*Double Lane Change Test*

The double lane change test was done using the instrumented vehicle at 80 kph. For simulation, the steering angle input as shown in Fig. 15 was taken from the steering wheel sensor.

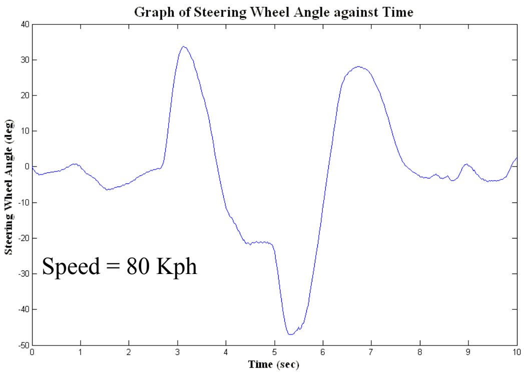

Fig. 15. Steering Angle Input for Double Lane Change Test

The simulation and experimental results for double lane change is shown in Figures 16 to 20. Fig. 16 shows the lateral acceleration response. It can be clearly seen that the simulation lateral acceleration results follows the trend and magnitude of experimental results very closely. The double lane change test yaw rate results for both simulation and experiment is presented in Fig. 17. The simulation yaw rate results closely follow the experimental yaw results with acceptable error in the magnitude.

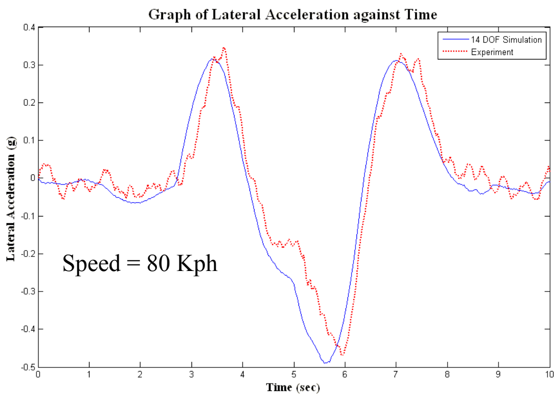

Fig. 16. Lateral Acceleration Response for Double Lane Change Test

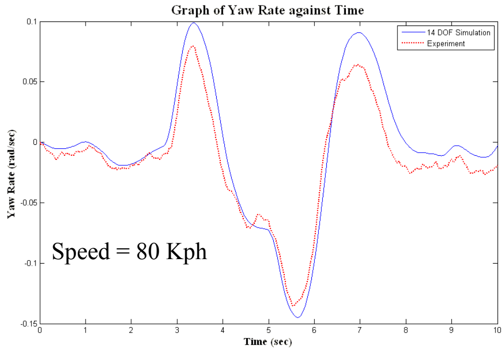

Fig. 17. Yaw Rate Response for Double Lane Change Test

Fig. 18 shows the roll angle results for both simulation and experiment for the double lane change test. It can be seen that the trend of simulation and experiment roll angle results is similar with error in the magnitude. Simplification in vehicle model causes the difference in magnitude between the simulation and experiment roll angle results. Ignoring the anti-roll bar is one of the sources of error of the roll angle results.

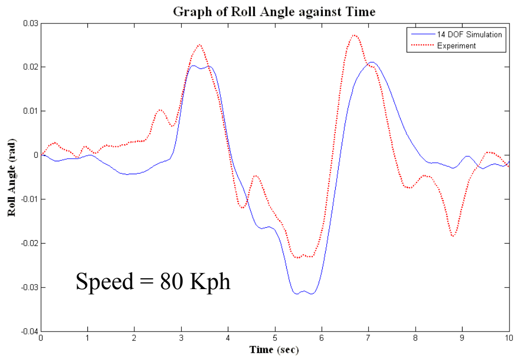

Fig. 18. Roll Angle Response for Double Lane Change Test

Figures 19 and 20 present the tire slip angle response for the front left, front right, rear left and rear right tire. The trends of the simulation tire slip angles results correlates very well with the experiment tire slip angle results. The difference arises because the vehicle is assumed to have a constant longitudinal velocity whereas during experiment, it was hard for the driver to maintain the vehicle throughout the double lane change test.

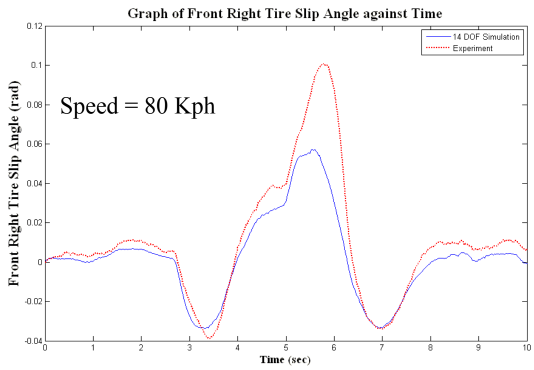

Fig. 19. Front Right Tire Slip Angle for Double Lane Change Test

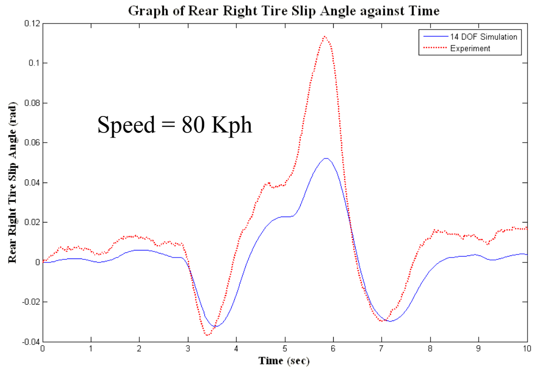


Fig. 20. Rear Right Tire Slip Angle for Double Lane Change Test

*Effect of the Roll Center*

The 14 DOF model without roll center considers that the roll center is fixed at the ground and the 14 DOF model with roll center assumes the roll center has a vertical distance of 100mm from ground. As seen from the Fig. 21, the magnitude of the roll angle for the 14 DOF model with roll center is lower than that of 14 DOF model with roll center.

Fig. 22 presents the normal force on front left, front right, rear left and rear right tires for 14 DOF model without and with roll center. It can be seen that normal tire forces for the 14 DOF model with roll center is lower than that of 14 DOF model without roll center. This means the roll center effects weight transfer from the inner wheel to outer wheel during cornering.

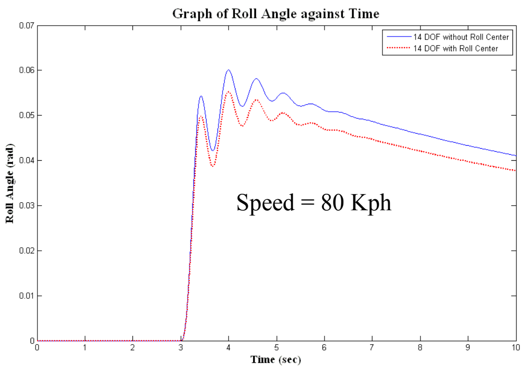


Fig. 21. Comparative Roll Angle Response for 180 Degrees Step Steer Test at 50 kph for Roll Center Effect

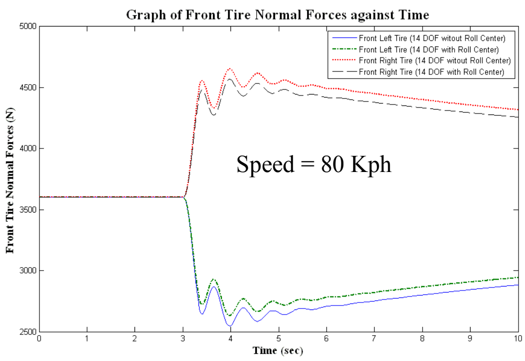


Fig. 22. Comparative Tire Normal Forces for 180 Degrees Step Steer Test at 50 kph for Roll Center Effect

## IV. CONCLUSION

A 14 DOF vehicle model which includes ride model, handling, and tire model was developed for the study of vehicle dynamics. The ride and handling model was validated separately using CarSimEd.

The 14 DOF vehicle model was validated with instrumented vehicle for 180 degrees step steer test at 35 kph and double lane change at 80 kph. The 14 DOF vehicle model was validated for lateral acceleration, yaw rate, roll angle and tire slip angle responses. From the 14 DOF model validation results, the trend between the simulation and experiment was similar with small difference in the magnitude. The difference arises due to the model simplification such as ignoring anti-roll bar, body flexibility, movement of roll center and also difficulty for the driver to maintain constant vehicle speed during the test maneuver. The validation result has proven the 14 DOF vehicle model can be used represent actual vehicle dynamic behavior.

The effect of roll center height to roll angle and tire normal forces were studied by varying the roll center height from the ground. As the distance of the roll center from the ground increases, the roll angle and the tire normal forces reduces.

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