Parametric Analysis of Four Wheel Vehicle Using Adams/Car

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Abstract:

Inspiring from the Multibody dynamics this paper has been carried out for estimating the dynamics of vehicle in motion. As there is more and more importance is given to the handling performance to the vehicle for its comfort, certain parameters like roll, yaw, pitch and, side slip angle etc., are to be studied, analyzed, controlled and adversely changed for increasing the overall performance and vehicle behavior. Here the focus is intended on the roll, pitch and yaw phenomena of the vehicle while changing the lane tack and its effects are simulated at different speeds. Results show certain variation and its effect. Hence this tool for multibody dynamics proves more and more efficient for such conditions.

Keywords: ADAMS/CAR, Lateral force, Multibody Dynamics, Pitch, Roll, Tire slip angle and, Yaw.

1. Introduction:

The focus of the paper is to estimate vehicle response in terms of roll ,yaw and pitch effects in front-wheel steered, rear-wheel driven four wheeled vehicles in real time. It gives a measure of the lateral forces produced at the tire-road contact patches while cornering and lane change during vehicle motion which make the vehicle turn. Physically it represents the twist in the treads of the tires and body frame. It is very difficult if not possible to directly measure the yaw rate angles of the chassis in vehicle, hence indirect methods have to be applied to estimate them. Knowledge of side slip angle is a required for advanced vehicle control systems like braking control, stability control, security actuators and for validating vehicle simulators. These controllers increase the safety of the vehicle, and make the response more predictable. The knowledge of vehicle dynamics can also be applied to decrease road damage caused by vehicles. The wheel hub is coupled to the vehicle through the suspension and steering mechanisms. Thus the steering and suspension mechanisms affect each other's behaviour. While cornering, changes will be induced in variables associated with both the mechanisms. The project endeavours to analyse these changes to estimate yaw rate rolling and pitch angle based on chassis deflection information. This work has been performed to meet the objectives:

1. To develop a method of estimating roll, yaw and, pitch in front-wheel steered four wheel vehicles.
2. To instrument a vehicle and conduct on road tests to validate the estimation method.

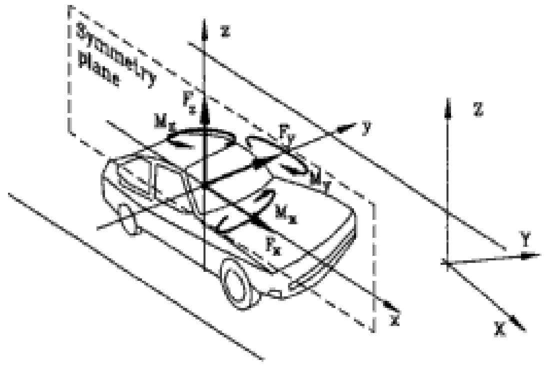
A model is constructed in ADAMS/CAR to simulate test conditions and predict the results for the tests to be conducted. The predictions of this model are verified with experimental results from literature. An open-loop estimator that uses a three degree of freedom vehicle model is used to estimate results using real time experimental data from tests conducted. We conclude that a three degree-of-freedom model is a good start for use in estimation techniques of vehicle dynamics and performance.

2. Vehicle dynamics:

The basic information required to understand vehicle architecture as well as dynamic behaviour of front wheel steered, four wheel vehicles is its mathematical formulation. It also explains how lateral forces are generated while a vehicle negotiates a turn, and their relation with the slip angle. Four wheel vehicle model is discussed in detail. This model is used to build observers (estimators) for estimating vehicle handling behaviour. A brief explanation of currently employed methods of estimation lateral forces and slip angles is given below. Also, the system consists of the dependent mechanism: steering and suspension system. The system shown here consists of rack and pinion steering mechanism and Double Wishbone Suspension.

The vehicle coordinate system shown in the Figure is explained below:

- Linear motion along x direction is known as longitudinal motion.
- Rotational motion about x axis is known as roll.
- Linear motion along y direction is known as lateral or transverse motion.
- Rotational motion about y axis is known as pitch.
- Linear motion along z direction is known as vertical motion.
- Rotational motion about z axis is known as yaw.



3. Vehicle Model Formulation For Parameters Identification:

For simulating the lateral dynamics of the vehicle a four wheel 3 Degree of freedom model is used containing lateral velocity (V), yaw rate (r) and, roll angle (ϕ).The input of the model is the steering angle on the front tires. Also the continuum mass of the vehicle is modelled by three lumped masses which are front and rear unsprung masses (M_{uf}, M_{ur}) and sprung mass (M_s) so the entire vehicle mass is:

$$M_t = M_s + M_{uf} + M_{ur}. \tag{1}$$

In order to derive the equation of motion a moving reference frame is attached to the vehicle with its origin at the centre of gravity as shown in Figure. Since the coordinate system is attached to the vehicle the inertia properties of the vehicle will remain constant Also as the result of symmetry assumption all the products of inertia are ignored .The state variables are assumed to be lateral velocity yaw rate and roll angle.

Using the above assumptions the equations describing the motion are:

$$M_t(\dot{V} + ru) + M_s h_s \ddot{\phi} = F_{yfr} + F_{yfl} + F_{yrr} + F_{yfl}, \tag{2}$$

$$I_{xx}\ddot{\phi} + M_s h_s(\dot{V} + ru) = L_s, \tag{3}$$

Where:

$$L_s = M_s g h_s \phi - K_\phi \phi - C_\phi \dot{\phi}, \tag{4}$$

Where, Ms is the sprung mass, which is the mass supported by the vehicle suspension, (I_{xx})is the sprung mass moment of inertia about longitudinal axis (x), (I_{zz})is the moment of inertia of the entire vehicle about vertical axis (z), K_ϕ and C_ϕ are roll stiffness and roll damping coefficient of suspensions respectively (h_s)is the vertical distance of CG from the roll axis, a and b are the distances of the front and rear axles from CG ,u is the longitudinal speed of the vehicle which is constant in vehicle maneuvers and F_{yfr}, F_{yfl}, F_{yrr}, F_{yrl} are the tire cornering forces of front right front left rear right and rear left respectively.

Figure 2 .Vehicle Model

The cornering force of a tire is mainly dependent on the slip angle vertical load longitudinal slip and camber angle of that tire. In this paper a tire model in which the cornering force of the tire is a function of the cornering stiffness vertical load slip angle and longitudinal slip has been used and the effects of camber angle and aligning moments are ignored. In order to compute tire forces the slip angles of the tire should be calculated as below:

$$\propto_{fr} = \delta - \tan^{-1}\left(\frac{V+ar}{u-t_f\frac{r}{2}}\right) \tag{5}$$

$$\propto_{fl} = \delta - \tan^{-1}\left(\frac{V+ar}{u+t_f\frac{r}{2}}\right) \tag{6}$$

$$\propto_{rr} = \tan^{-1}\left(\frac{br-V}{u-t_r\frac{r}{2}}\right) \tag{7}$$

$$\propto_{rl} = \tan^{-1}\left(\frac{br-V}{u+t_r\frac{r}{2}}\right) \tag{8}$$

Where δ is the steer angle as the input of the model and t_f , t_r are the front and rear tread widths of the vehicle respectively.

3.1 Lateral load for its effects:

Also, for obtaining the lateral load transfer some equations to describe vertical forces on each tire have been written. In this concept lateral load transfer is assumed to be the result of three phenomena which are body roll, roll centre height and unsprung mass.

Lateral load transfer, due to body roll, is an follows:

$$F_{f1} = \left(\frac{k_f h_s M_s}{K_\emptyset t_f}\right).\left(a_y \cos\emptyset + g\sin\emptyset\right) \tag{9}$$

$$F_{r1} = \left(\frac{k_r h_s M_s}{K_\emptyset t_r}\right).\left(a_y \cos\emptyset + g\sin\emptyset\right) \tag{10}$$

Where k_f and k_r are the front and rear roll stiffness and a_y is the lateral acceleration Which is given as:

$$a_y = \dot{V} + ru + \frac{M_s h_s \ddot{\emptyset}}{M_t} \tag{11}$$

Lateral load transfer, due to roll center height, is as follows:

$$F_{f2} = \frac{M_s b h_f a_y}{t_f(a+b)} \tag{12}$$

$$F_{r2} = \frac{M_s a h_r a_y}{t_r(a+b)} \tag{13}$$

Where h_f and h_r are the front and rear roll center heights respectively.

And lateral load transfer,due to unsprung masses, is :

$$F_{f3} = M_{uf} a_y \frac{n_f}{t_f} \quad (14)$$

$$F_{r3} = M_{ur} a_y \frac{n_r}{t_r} \quad (15)$$

The vertical load on each tire can be described by the following equations :

$$F_{zfr} = \frac{W_f}{2} - F_{f1} - F_{f2} - F_{f3} \quad (16)$$

$$F_{zfl} = \frac{W_f}{2} + F_{f1} + F_{f2} + F_{f3} \quad (17)$$

$$F_{zrr} = \frac{W_r}{2} - F_{r1} - F_{r2} - F_{r3} \quad (18)$$

$$F_{zrl} = \frac{W_r}{2} + F_{r1} + F_{r2} + F_{r3} \quad (19)$$

Where W_f and W_r are the static load distribution on the front and rear axles and can be computed by the following equations:

$$W_f = M_t g \frac{b}{a+b} \quad (20)$$

$$W_r = M_t g \frac{a}{a+b} \quad (21)$$

3.2 Lateral Force and Tire Slip Angle:

While cornering, a vehicle undergoes lateral acceleration. As the tires provide the only contact of the vehicle with the road, they must develop forces which result in this lateral acceleration. When a steering input is given, the successive treads of the tires that come in contact with the road are displaced laterally with respect to the treads already in contact with the road. Thus an angle is created between the angle of heading and the direction of travel of the tire. This angle is known as the tire slip angle which gives an estimate of twist of the treads of the tire. It can also be defined as the ratio of the lateral and forward velocities of the wheel. The twisted treads try to get back to their original positions, thus producing the force required for lateral acceleration. This force is known as the Lateral Force (F_y) or the Cornering Force. At a given load, the cornering force grows with slip angle. At low slip angles (5 degrees or less) the relationship is linear. In this region, cornering force is often described as $F_y = C_\alpha \alpha$. The proportionality constant C_α is known as cornering stiffness and is defined as the slope of the curve for F_y versus α at $\alpha = 0$.

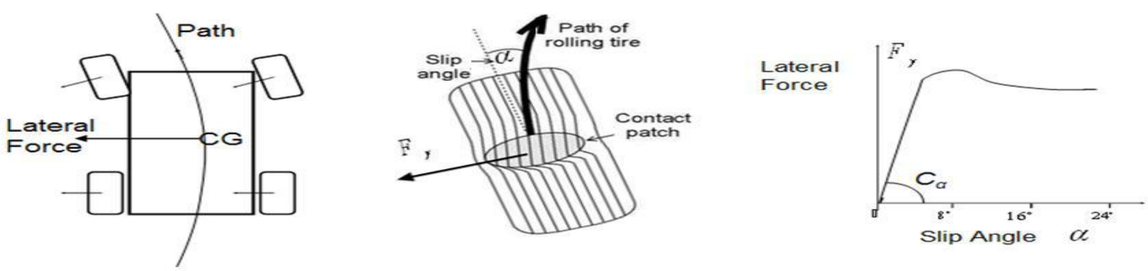
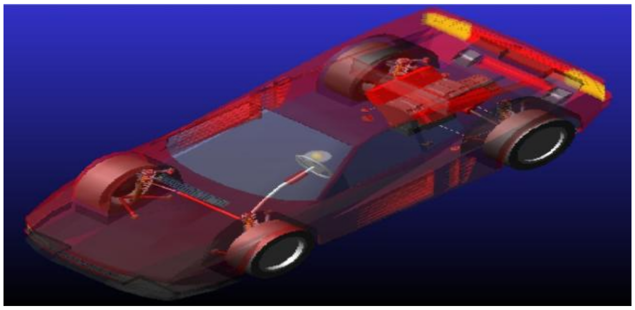


Figure 3. Three Degree of Freedom Automobile Model

Since the stability control system has the ability to affect the vehicle's attitude and motion, a function normally reserved for the driver, it needs to accurately interpret what the driver intends for the vehicle motion in order to provide added directional control (within physical limitations) as a driver's aid. Responsiveness, consistency, and smoothness are essential for a driver's confidence and comfort with the system. These are the guiding principles for our development. A driver typically expresses directional intent through the steering wheel. The angular position of the steering wheel is the first measure of driver intent.

4. Simulation RESULTS:

The below shown figure indicates the model which is to be simulated for the estimation of vehicle dynamics and its handling. The model shown below is ADAMS/CAR generated model:



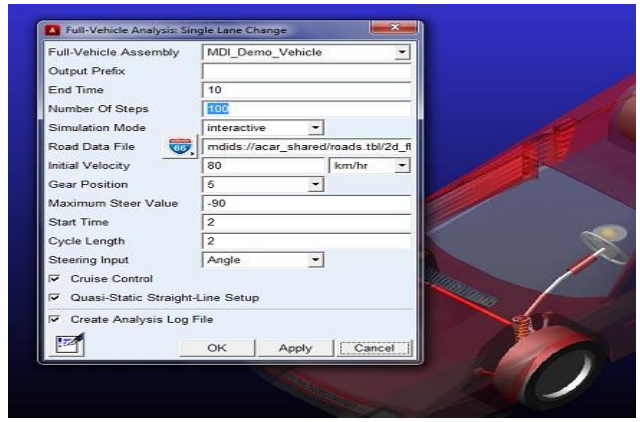


Figure 4. ADAMS/CAR and Full Vehicle Analysis

With reference to the car model here full vehicle analysis is to be carried out. Certain graphs are discussed here.The graph shown here is angular v/s time for the rolling motion of vehicle chassis at speed of 40, 80 and, 120 km/hr.

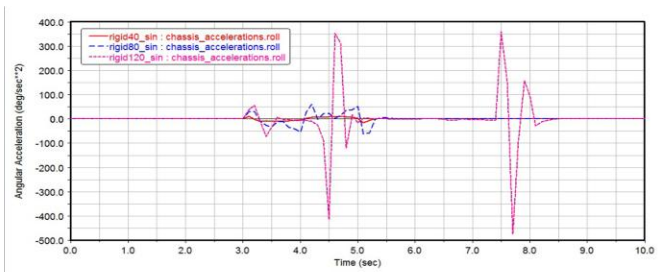


Figure 5. Angular acceleration v/s time for Roll

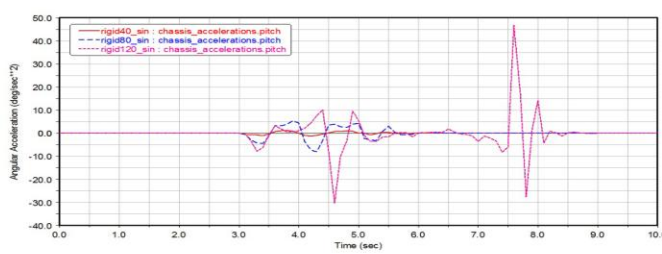


Figure 6. Angular acceleration v/s time for Pitch

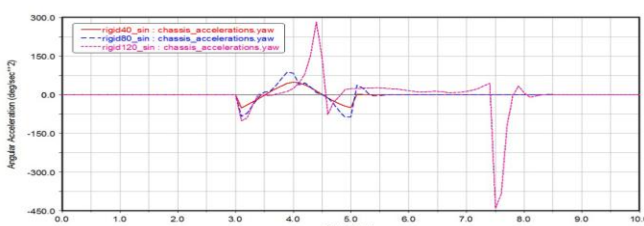


Figure 7.Angular acceleration v/s time for Yaw

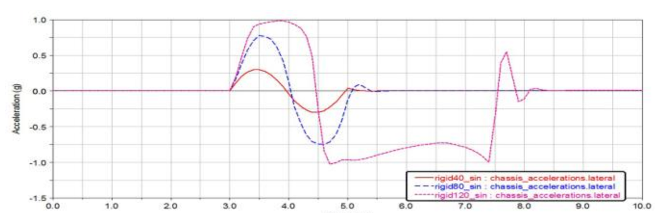


Figure 8.Angular acceleration v/s time for lateral acceleration

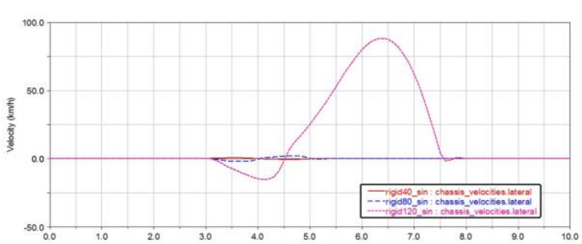


Figure 9.Chassis velocity v/s time for lateral

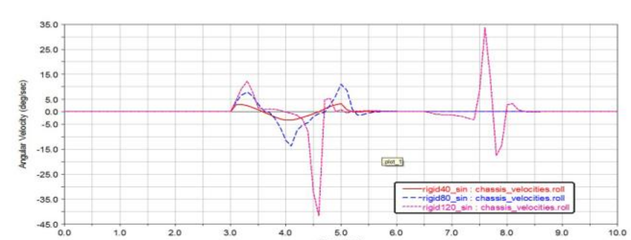
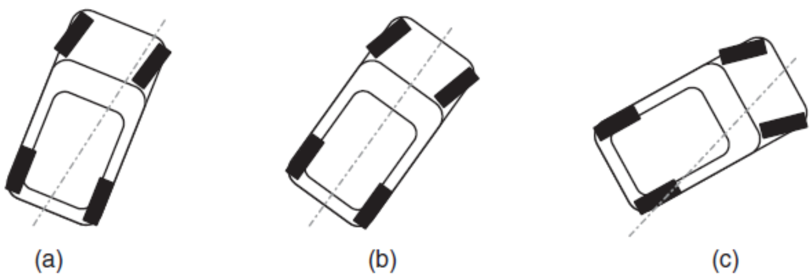


Figure 10.Angular Velocity v/s time for Roll

Here the lateral effect on vehicle handling is shown over here and its effects to yaw, roll and, pitch is mentioned:In scenario (a), yaw rate gain is reduced further than lateral acceleration gain. In order to accommodate the changes in both lateral acceleration and yaw rate, the radius of the path must increase and so the vehicle has a period of adjustment to a new, wider line in the curve. Most drivers notice this and instinctively reduce vehicle speed to restore the desired path over the ground. If uncompensated, it leads to a vehicle departing the course (road, track, etc.) in an attitude that is basically forwards. This is by far the most common behaviour for road vehicles.In scenario (b), lateral acceleration and yaw rate gain change in some connected manner and the vehicle will maintain course although it might need some modification to steering input. Excess speed for a curve will lead to the vehicle running wide but with no sense of 'turning out of the curve'. Such a vehicle generally feels benign although the progressive departure can mean it is unnoticed by inattentive drivers.In scenario (c), lateral acceleration gain reduces more than yaw rate gain. This leads to an 'over-rotation' of the vehicle when viewed in plan. Depending on the severity of the mismatch, the change may lead to a spin out of the curve. From inside the vehicle there is a pronounced sense of the rear end of the vehicle departing first but objectively the vehicle may not actually oversteer in the classical sense.it may simply move 'towards neutrality'. Vehicles that preserve yaw rate gain as they lose linearity are widely regarded as fun to drive and sporty.



4.1 Yaw, Rate and angular Acceleration

The vehicle is driven at known speeds, on a pre- marked course. Hence, by knowing the trajectory of the curve and speed of the vehicle, the yaw rate and lateral acceleration is calculated.

5. Conclusion

This paper aimed at estimating the dynamic parameters like roll, yaw, pitch and side slip angle in front-wheel steered, rear-wheel driven four wheeled vehicles. Side slip angle cannot be measured directly; hence estimation has been built to calculate side slip angle from measurable variables. A model was constructed in ADAMS/CAR to predict this variable for a candidate set of test conditions. Since the predictions of the ADAMS/CAR model were found to be in reasonable agreement with the experimental results reported in literature, it can be conclude that the three degree-of-freedom model provides good prediction capabilities for estimation of vehicle dynamics.

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