Analytic and Simulation Based Modeling of Vehicle Lateral Stability and Handling for Double Lane Change (DLC) Maneuvering

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Abstract

In this paper, some stability aspects of lateral motion of a vehicle are presented. In this manner, analytic and simulation based modeling of vehicle lateral stability and handling for double lane change (DLC) maneuvering are covered. It is shown that vehicle stability depends on several parameters. Therefore, a certain initial state condition and a constant steering angle are assumed. A mathematical model of the vehicle stability based on a 3 DOF model is also presented considering yaw and roll is presented. CarSim software is applied to further verify the DLC model of the objective vehicle (i.e. SUV). The results showed that for the DLC maneuvering condition, the cornering direction affects the lateral slip product too. The maximum and minimum value of lateral slip variations vary within the range of ± 30 degree. Also it was concluded that the steering angle limit variation lie within the range of ±400 degree.

Keywords: CarSim; Double Lane Change; Handling; Stability

1. Introduction

Vehicle handling and stability have gained a lot of attention and importance within the automotive engineering industry. The problem occurs when a contradictory issue is desired, driving at high speed and keeping the vehicle stability and handling at the greatest level during various driving conditions such as maneuvering and travelling over road irregularities.

Currently, vehicle stability improvement techniques have tendency towards the application of kinetics such as braking and traction procedures to assist the stability of vehicle during motion. The active safety systems, that control the vehicle lateral and longitudinal dynamics, aids the driver to control the vehicle in different operational conditions such as slippery roads or sudden steering and maneuvering.

There are different chassis control methods, such as ABS (anti-lock braking system), four-wheel steering, and active front steering, the purpose of all of which is to control the vehicle and react properly to the driver and keep the vehicle steady under emergency situations [1]. It should be also noted that the kinetics of motion that affect the stability of the vehicle are function of tire-ground interaction process and parameters such as traction, braking, rolling resistance, etc. The tire-ground contact patch characteristics and surface-tire friction along with normal load are determinant factors on stability and handling. In order to keep the appropriate vehicle performance at limited handling, it is essential to make the forces applied by all tires to operate efficiently and keep them inside the tire friction ellipse [1].

Yaw and roll motions are two substantial lateral dynamic features of the ground vehicles. Yaw and rollover stability are very influential on driving safety given that yaw stability is the possibility to keep vehicle turning with minimal skidding, and roll stability is to avoid vehicle turnover during sudden lateral acceleration. There are studies documented in literature dealing with different strategies concerned with planar motion control of the vehicle.

The optimum tire force to maximize acceleration/deceleration of a four-wheel vehicle during cornering was carried out with the aim of analyzing the potential improvement within the scope of vehicle steering and driving mechanisms [2].

For 4-wheel-distributed steering and traction/braking systems, a vehicle dynamics integrated control algorithm was proposed using an on-line non-linear optimization method with optimization of work load of tires while keeping them at a constant value. Moreover, the influence of the objective vehicle dynamics control for the 4-wheel-distributed steering and 4-wheel-distributed traction/braking systems was also presented using simulation to compare with the combination of the various actuators [3].

The design of the integrated active front steering and active differential control for handling improvement of road vehicles is a dynamic field of studying interest too. In this regard, a controller design algorithm was developed based on the solution of a set of linear matrix inequalities that guarantee robustness against a number of vehicle parameters such as speed, cornering and braking stiffness while it was confirmed that static-state feedback controllers designed by the proposed
method could reach an outstanding road handling performance versus uncontrolled ones [4].

In another investigation, the impact of tire vertical stiffness on ride, handling, accelerating/braking performance, and fuel consumption of a vehicle were covered along with a method for determining the optimum vertical stiffness of tires. To this end, a performance index was proposed considering that the tire vertical stiffness is a design variable and its optimization provides a compromise among ride, handling, accelerating/braking performance, and fuel consumption of the vehicle. The analytical optimization results were confirmed by performing precise numerical simulations [5]. In a relevant paper, a gain-scheduled active steering control and active differential design method to preserve vehicle stability in extreme handling conditions were proposed with a new formulation of the bicycle model in which tire slip angles, longitudinal slips and vehicle forward speed seem to be varying vehicle parameters [6].

Of the other vehicle controller types, a vehicle roll motions using active antiroll bar [7], continuous damping control (CDC) [8], or active suspension control [9] to enhance the rollover stability through influencing the lateral load transfer and longitudinal speed control [10, 11] can be pointed out.

A study was performed to design an integrated controller for the processes of the yaw and rollover stability controls using a nonlinear model with a piecewise linearization tire model. Further confirmation were made based on some experimental data obtained from the experiments and simulations were carried out in CarSim software environment including slalom condition and double-lane change. The obtained results showed that the coordinated control algorithm improves vehicle yaw and roll stability satisfactorily [12].

As documented in literature, rollover accidents are responsible for about 33% of all deaths of SUVs while it is simpler to control rollover when compared to any other vehicular accidents due to a low center of gravity position [13-15].

This highlights the need to assess the vehicle stability and lateral forces of vehicles particularly those of off-road vehicles owing to various emergency conditions that vehicles may encounter.

To the best knowledge of authors, little is known about vehicle lateral stability and handling for double lane change (DLC) maneuvering for a typical sport utility vehicle (SUV) vehicle. The aim of the present work is to investigate the vehicle lateral stability and handling for double lane change (DLC) maneuvering using CarSim software. A mathematical model of the vehicle stability based on a 3 DOF model is also presented considering yaw and roll.

2. Mathematical Model of Vehicle Dynamics

A three-degree-of-freedom rollover and yaw prediction model is shown in Fig. 1. The kinematics and moments applied to the vehicle roll bar and chassis is demonstrated as well as 3 DOF model for rolling and yaw process for the vehicle.

The equations presented in this paper are appropriate to determine the longitudinal and lateral motion the vehicle and also the yaw motion. Regarding the vehicle roll motion, it relies on the suspension parameters. When a vehicle is driven through a corner, a lateral acceleration increases and the vehicle chassis rolls about the longitudinal axis. The roll centers are different for different cars and depend on the suspension geometry, however, for a passenger car the roll centers and the roll axis are always below the center of gravity of vehicle. As documented in the literature, the most common approach to investigate the roll motion of a vehicle is by computing the vehicle rotational stiffness. This can be performed considering the suspension as two torsion springs located in the front and rear roll centers and evaluating the motion as a function of the lateral acceleration. Based on the above assumptions, the balance equations of the force along the y-axis and torques around the y-axis and x-axis can be expressed as:

\[ m (v_y + v_{r,r}) = (F_{y_{r1}} + F_{y_{r1}}) + F_{y_{r2}} + F_{y_{r2}} \]  
\[ I_{r} \dot{\phi} = I_{1} (F_{y_{r1}} + F_{y_{r1}}) - I_{z} (F_{y_{r2}} + F_{y_{r2}}) + M_{z} \]  
\[ M_{z} = mgh_{\phi} - k_{\phi} \phi - C_{\phi} \phi \]  

As a roll prediction model, vehicle model with three degrees-of-freedom should represent the real vehicle. When rollover occurs, the lateral acceleration of vehicle will gradually increase and subsequently, the lateral forces of tires treat non-linearly. Accordingly, the limit of tire’s lateral force due to road adhesion limit should be included. In addition to normal force, the tire slip angle is another main quantity for the calculation of tire lateral force. The following equations define the slip angles of the front and rear tires:

\[ \alpha_{1} = \delta_{j} - \arctan \left( \frac{V_{y} + l_{r}\tau}{V_{r} + 0.5W_{r}\tau} \right) \]  
\[ \alpha_{2} = \delta_{j} - \arctan \left( \frac{V_{y} + l_{r}\tau}{V_{r} - 0.5W_{r}\tau} \right) \]  
\[ \alpha_{3} = \arctan \left( \frac{l_{r}\tau - V_{y}}{V_{r} + 0.5W_{r}\tau} \right) \]  
\[ \alpha_{4} = \arctan \left( \frac{l_{r}\tau - V_{y}}{V_{r} - 0.5W_{r}\tau} \right) \]
It should also be pointed out that longitudinal forces are function of the tire slip coefficient while the lateral forces depend on the tire slip angles \( \alpha \). In this part, the longitudinal slip is ignored and the longitudinal forces are considered for the condition in which a quasi-constant forward velocity is provided.

The transfer function of yaw rate and sideslip angle can be formulated as following:

\[
\frac{\gamma(s)}{\delta(s)} = G_\gamma \frac{\tau_s + 1}{T_s^2 + T_z s + 1},
\]

\[
\frac{\beta(s)}{\delta(s)} = G_\beta \frac{\tau_s + 1}{T_s^2 + T_z s + 1},
\]

where,

\[
G_\gamma = \frac{V_s}{(l_1 + l_2)(1 + PV_s^2)},
\]

\[
P = \frac{m}{2(l_1 + l_2)^2} \frac{l_z C_m - l_z C_{sf}}{C_m C_{sf}}.
\]

The linear tire model used to calculate the lateral tire force is calculated as following:

\[
F_y = C_s \alpha
\]

This model can define the vehicle lateral dynamics under normal driving conditions with low lateral acceleration satisfactorily. In emergency situations when lateral acceleration is great, the vehicle slip angle increases and tire forces increase at the road friction limit. Hence, the dynamic behavior of vehicle is nonlinear and using the linear tire model will be insufficient. The Dugoff’s tire model based on the friction ellipse idea has been widely used for nonlinear simulations [16]. In this model, the relation for lateral force of each tire is as follows:

\[
F_y = C_s \tan \alpha \frac{f(S)}{1 - i}
\]

where,

\[
f(S) = \begin{cases} 
S(2 - S) & \text{if } S < 1 \\
1 & \text{if } S > 1 
\end{cases}
\]

3. Double Lane Change (DLC) Model

The DLC maneuver is a test procedure for a severe lane-change maneuver and object avoidance simulations, and is described in the standard ISO/TR 3888–1. In the DLC maneuver, the vehicle enters into the first lane-section with a constant initial longitudinal velocity, and continues with a released throttle valve. Originally known as the ”moose test” the lane change maneuver was transferred to the International Standard ISO 3888–2 after a revision by the Association of the German Automotive Industry (VDA). The ISO and VDA lane-change test is used to evaluate the handling performance of a vehicle and is an integral part of the vehicle design procedures and vehicle assessment. Based on 3 cone lanes with a total length of 61 m a double lane change is defined, which must be completed with maximum speed.

The ISO double lane change test consists of an entry and an exit lane and with a length of 12m and a side lane with a length of 11m. The width of the entry and side lane are de-
pendent on the width of the vehicle, the width of the exit lane is constantly 3 m wide. The lateral offset between entry and side lane is 1 m and the longitudinal offset is 13.5 m. For the same lateral offset the side and exit lane has a slightly shorter longitudinal displacement of 12.5 m. 2 m after the start of the entry lane the throttle is released so that the entire maneuver is completed in the overrun mode with the top gear and an engine speed of at least 2000 rev/min. At the end of the entry and exit lane the velocity is measured. The entry velocity is increased gradually. If no cones are overturned, the test is passed.

Figure 2. Schematic representation of DLC maneuvering [17]

There are some certain advantages of the automatically driven ISO lane change tests as following:
1) Objective test results without driver influence
2) Fast installation and configuration of the steering robot - ready to use for testing in about 1 hour
3) No elaborate positioning of cone lanes needed
4) All data is recorded and displayed under real-time conditions by the steering robot
5) Simple graphic analysis by driving robot software

Figure 3. Automatic DLC in x-y planar motion based on ISO 3888 [17]

4. CarSim Model

CarSim simulates a complete 3D vehicle-suspension multi-body model, concerning the car vertical dynamics, as well as the lateral and longitudinal motions. It delivers is of efficient approaches for simulating the performance of passenger vehicles and light-duty trucks. CarSim is a preferred tool for analyzing vehicle dynamics, developing active controllers, calculating a car’s performance characteristics, and active safety systems. In addition to VS Commands (the built-in scripting language), it can run with custom programs (MATLAB, Visual Basic, C/C++) using VS API (application program interface).

Fig. 5 shows the vehicle sprung mass, roll, pitch and yaw inertia, $I_x$, $I_y$, and $I_z$ respectively. Also the radii of gyration are presented. It should also be pointed out that inertia and radius of gyration are related by the equation:

$$ I = M \cdot R \cdot R $$  (15)

Due to the symmetry of the problem concerning the road profile that is the same for both left and right wheels, only the simplified problem of car vertical dynamics is studied in this paper, considering the lateral and longitudinal motions.

5. Results and Discussion

The longitudinal velocity, lateral velocity, yaw rate, roll rate and rotational speeds of four wheels constitute the degrees of freedom for this model. Therefore, the normal load transfers of tires, the roll steer, the roll camber and other model complexity effects are considered on evaluation of the model performance. The aims of yaw stability and roll stability are to follow the driver intention and limit the maximum of the vehicle lateral acceleration, respectively. The DLC maneuvering road profile for the target path and controlled vehicle path are presented in Fig. 6 which presents the lateral and longitudinal distance to path. As it can be deduced, after the second lane change, there is a peak in difference between the target and vehicle path difference in lateral direction. However, it is clear-cut that the difference starts from the first lane change. However, the controlled motion of the SUV is a main cause to return the vehicle at the desired path.

Steering wheel angle versus time for the objective vehicle following the DLC path is presented in Fig. 7. The variations in steering wheel angle is well in accordance with the vehicle driving path. The vehicle runs at a constant speed for 8 s and then the steering and braking is conducted. The driver’s inputs is provided in Fig. 7. The controller subtracts or adds a steering angle to the driver command. When the road adhesion is at its nominal value even when the speed varies, the control effort vanishes within driver reaction time which is assumed to be about 1 s. When the road adhesion is decreased, there is a remaining steering angle. The steering angle limit variation lie within the range of ±400 degree.

In vehicle dynamics, slip angle or sideslip angle is the angle between a rolling wheel's actual direction of travel and the direction towards which it is pointing (i.e., the angle of the vector sum of wheel forward velocity $V_f$ and lateral velocity $V_l$). For a free-rolling wheel this slip angle results in a force parallel to the axle and the component of the force perpendicular to the wheel's direction of travel is the cornering force. This cornering force increases approximately linearly for the first few degrees of slip angle, then increases non-linearly to a maximum before beginning to decrease. A non-zero slip angle arises because of deformation in the tire carcass and tread. As
the tire rotates, the friction between the contact patch and the road results in individual tread 'elements' (finite sections of tread) remaining stationary with respect to the road. If a side-slip velocity $u$ is introduced, the contact patch will be deformed. When a tread element enters the contact patch, the friction between the road and the tire causes the tread element to remain stationary, yet the tire continues to move laterally. Thus the tread element will be 'deflected' sideways. While it is equally valid to frame this as the tire/wheel being deflected away from the stationary tread element, convention is for the coordinate system to be fixed around the wheel mid-plane.
Figure 7. Steering wheel angle with respect to driving time

Figure 8. The lateral slip (side slip) angle with respect to time

Figure 9. Roll variations with respect to time
While the tread element moves through the contact patch it is deflected further from the wheel mid-plane. This deflection gives rise to the slip angle, and to the cornering force. The rate at which the cornering force builds up is described by the relaxation length. The ratios between the slip angles of the front and rear axles (a function of the slip angles of the front and rear tires respectively) will determine the vehicle's behavior in a given turn. If the ratio of front to rear slip angles is greater than 1:1, the vehicle will tend to understeer, while a ratio of less than 1:1 will produce oversteer. Actual instantaneous slip angles depend on many factors, including the condition of the road surface, but a vehicle's suspension can be designed to promote specific dynamic characteristics. A principal means of adjusting developed slip angles is to alter the relative roll couple (the rate at which weight transfers from the inside to the outside wheel in a turn) front to rear by varying the relative amount of front and rear lateral load transfer. This can be achieved by modifying the height of the roll centers, or by adjusting roll stiffness, either through suspension changes or the addition of an anti-roll bar. Because of asymmetries in the side-slip along the length of the contact patch, the resultant force of this side-slip occurs away from the geometric center of the contact patch, a distance described as the pneumatic trail, and so creates a torque on the tire. For the vehicle lateral side-slip angle variations with respect to time, Fig. 8 is presented. As it can be seen, the lateral slip for rear tires are harmoniously changing within a small limit when compared to the steering tires (front tires). Also for the DLC maneuvering condition, the cornering direction affects the lateral slip product too. The maximum and minimum value of lateral slip var-
The safety of driving vehicle is mainly dependent on yaw and rollover stability. Instability of yaw takes place on all roads, while rollover mainly happens on the high adhesion road. Roll and yaw products are very substantial parameters for vehicle stability and handling conditions. Figs. 9 and 10 show the vehicle roll and yaw rates, respectively, that also explicitly exhibit a uniformity in variations. The greatest roll value corresponds to 2.5 degree that occurs during cornering from each of lanes and then stabilizes after the double lane change gradually. The same trend is also attributable to the yaw rate variations during the DLC process (Fig. 10).

It should be noted that in order to decrease the control input, the value of weighting ratio can be increased proportionally, otherwise the yaw rate can’t follow the behavior of reference model and the vehicle remains unstable. Due to the weight transfer effect during cornering and the lateral force variations, the vertical forces applied to the tires change. Roll and yaw parameters due to the moments applied to the tires affect the vertical force variations. As shown in Fig. 11, at the start of process the vertical load on front tires and rear tires are equal. With the start of cornering in the first lane, the tires on the right and left side of the vehicle behave similarly. It is obvious that, as expected, the right and left tires are subject to transferred weight oppositely. It should be noted that in order to decrease the control input, the value of weighting ratio can be increased proportionally, otherwise the yaw rate can’t follow the behavior of reference model and the vehicle remains unstable.

6. Conclusions

In this paper, some stability aspects of lateral motion of a vehicle are presented. In this manner, analytic and simulation based modeling of vehicle lateral stability and handling for double lane change (DLC) maneuvering are covered. It is shown that vehicle stability depends on several parameters. Therefore, a certain initial state condition and a constant steering angle are assumed. A mathematical model of the vehicle stability based on a 3 DOF model is also presented considering yaw and roll is presented. CarSim software is applied to further verify the DLC model of the objective vehicle (i.e. SUV). The results showed that for the DLC maneuvering condition, the cornering direction affects the lateral slip product too. The maximum and minimum value of lateral slip variations vary within the range of ± 30 degree. Also it was concluded that the steering angle limit variation lie within the range of ±400 degree.

References