Design of a Novel Fuzzy Controller to Enhance Stability of Vehicles

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Abstract **-** This paper presents the design of a novel fuzzy control structure to improve stability of vehicles with semi-active suspension system. The proposed fuzzy controller adjusts the damping coefficient to stabilize the sprung mass and hence reduce the tendency of vehicle to rollover. A full car model with eight degrees of freedom is adopted that includes the vertical, roll, yaw, and pitch motions as well as the vertical motions of each wheel. Four decentralized fuzzy controllers are developed and applied to each individual damper in the vehicle suspension system. The controllers input(s) are lateral acceleration and vehicle states and the output is an adaptive damping coefficient. Mamdani's inference engine is used to obtain the required damping coefficient of each suspension system. To evaluate the performance of the proposed controller, experiments were performed for simple turn and lane change maneuvers. To show the effectiveness of the proposed controller, comparison is made with Cadillac controller. Results show that the fuzzy controller reduces roll angle, linear transfer ration (LTR) and hence decreases the propensity to rollover in vehicles.

I. Introduction

Due to the safety of passengers and environment, design of rollover controllers for vehicle has shown a great interest among researchers. There are different parameters contributing to rollover such as suspension, steering, and tires. To account for the complexity of the vehicle system, stability of vehicles is attributed to the suspension as well as steering systems. In this paper, we investigate the effects of suspension on rollover stability and hence design a controller to reduce the propensity to rollover.

Amongst different influencing parameters on rollover, two major factors have great influences:

- Road input
- Vehicle dynamics that is mostly related to the lateral acceleration and hence dictated by yaw and roll motion

The input from the road is very important. Depending on different road conditions, stability of the vehicle might be jeopardized significantly. For instance, driving over bumpy surfaces is more dangerous than on the even ones. Therefore, simulations must be carried out on based on the real or at least close to real road data.

Over the years, many different models for determining the dynamics of rollover have been captured. The simplest model which can be adopted is a half-car model. In this model, effects of tires may or may not be included.

Gillespie [1] presents an expression for the rollover threshold due to turning with constant velocity. This expression is simply called Static Safety Factor (SSF) and gives a prudent criterion for measuring the rollover. Since the dynamic of the rollover is not considered in these models, they are unable to present the effect of tire parameters and also roll motion.

Bernard [2] included the effect of lateral movement of vehicle center of gravity and lateral tire distortion. Later on, Dixon [3] provided a more refined formula, which he included the effect of body roll and tire lateral distortion. Hac [4] formulated the different parameters affecting rollover of a car for a non-dynamic case (steady-state cornering) such as tires, half-track width change, gyroscopic motion due to wheel rotation, and jacking forces (vertical forces tend to lift the vehicle center of mass). He concluded that amongst the considered parameters, lateral displacement of vehicle center of gravity has a great influence. All the secondary effects can reduce the rollover threshold up to 25%. Thereafter, he investigated the effect of suspension kinematics on stability of a vehicle. It was determined that an optimum value for the roll center height exists and an analytical expression was given for that. Possible future improvement in rollover may be achieved by considering nonlinearities in the suspension system. It was concluded that variations in suspension parameters can stabilize the system. As well, Whitehead et. al. [5] investigated the effect of various vehicle parameters on rollover propensity by simulation.

Also energy method was used to determine the roll stability of vehicles. Dahlberg [6] developed a dynamic rollover threshold (DRT) method to predict rollover in dynamic cases. However, the method is sufficient but not enough for a vehicle to rollover.

All the models adopted to determine the rollover threshold were manoeuvre dependent. Since the dynamics of the vehicle and hence the other influencing parameters on rollover threshold play critical rule in stability of vehicles,

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Chen and Peng [7], used NN to predict the time to rollover for various manoeuvres. They simulated 15 different manoeuvres resulting in rollover and developed a NN system to train the data. The NN takes TTR, vehicle roll angle, and change of roll angle to predict the instability. They showed that the proposed method can accurately estimate the time to rollover.
The developed controllers are mostly model based and hence the reliability of the system for different maneuvers might be at risk. This in fact is a motivation for developing a controller that is not model based and at the same time real data from sensors can be directly incorporated into it. Therefore, in this paper, a fuzzy controller is proposed due to its effectiveness in dealing with noisy date from the sensors. developed to enhance the stability of vehicles.

The organization of this paper is as follows: first in section II roll and yaw models are reviewed. Mathematical equations and state-space models are derived. In section III, the Fuzzy controller is presented. In section IV, analysis and results are shown and conclusions are drawn.

II. Modeling

To capture the dynamic behaviour of the system, two dynamic models shown in figure 1 and figure 2, are considered. Figure 1 shows the roll plane model of the vehicle vertical. Tire forces, lateral acceleration, and sprung and unsprung weights are the external forces applied to the system. Note that we have separated sprung mass and upsprung masses weights.

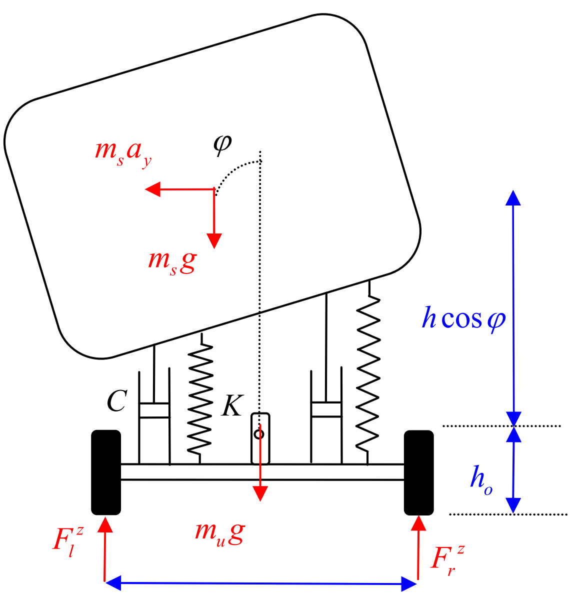

Fig. 1. Roll plane model

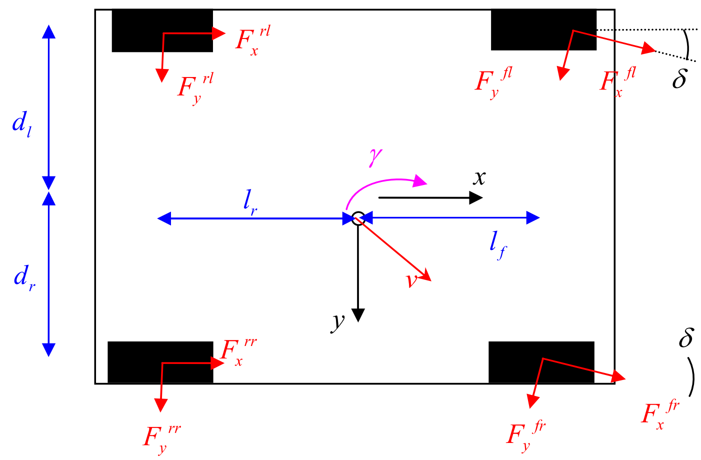

Fig. 2. Yaw model

Figure 2 depicts the yaw plane model of the vehicle. A Cartesian coordinate system has been located at the mass center of the vehicle. Longitudinal and lateral tire forces shown in this coordinate frame. The steering angle, the angle that tire makes with the longitudinal direction, is introduced as δ.

Consequently, components of acceleration along x and y directions are as follows:

$$\begin{aligned} a_x &= \dot{v}_x - \dot{\gamma} v_y \\ a_y &= \dot{v}_y + \dot{\gamma} v_x \end{aligned} \tag{1}$$

Consequently, equations of motion are derived as follows:

$$(F_x^{fl}+F_x^{fr})\cos\delta-(F_y^{fl}+F_y^{fr})\sin\delta+F_x^{rl}+F_x^{rr}-m(\dot{v}_x-\dot{\gamma}v_y)=0$$
$$(F_x^{fl}+F_x^{fr})\sin\delta-(F_y^{fl}+F_y^{fr})\cos\delta+F_y^{rl}+F_y^{rr}-m(\dot{v}_y+\dot{\gamma}v_x)=0$$

$$\begin{aligned} &(-F_x^{fl}d_l+F_x^{fr}d_r)\cos\delta+(F_y^{fl}d_l-F_y^{fr}d_r)\sin\delta+ \\ &(F_x^{rl}+F_x^{fr})l_f\sin\delta+(F_y^{rl}+F_y^{fr})l_f\cos\delta+F_x^{rl}d_l-F_x^{rr}d_r \\ &-F_y^{rl}l_r-F_y^{rr}l_r=I\ddot{\gamma} \end{aligned} \tag{2}$$

Assuming linear tire model:

$$\begin{aligned} F_y^{fl}+F_y^{fr} &= F_y^f;\, F_y^f = C_f\alpha_f \\ F_y^{rl}+F_y^{rr} &= F_y^r;\, F_y^r = C_r\alpha_r \end{aligned} \tag{3}$$

Where C_f and C_r are front and rear cornering coefficients, respectively, α_f and α_r are front and rear slip angles.

$$\alpha_f = \delta - \frac{\dot{v}_y + l_f \dot{\gamma}}{\dot{v}_x}$$

$$\alpha_r = -\frac{\dot{v}_y - l_r \dot{\gamma}}{\dot{v}_x} \qquad (4)$$

Using equations (3) and (4), we can rearrange equations (3) as follows:

$$\begin{aligned}
&m\dot{v}_x = (F_x^{fl} + F_x^{fr})\cos\delta - (F_y^{fl} + F_y^{fr})\sin\delta + F_x^{rl} + F_x^{rr} + m\dot{\gamma}v_y \\
&m\dot{v}_y = (F_x^{fl} + F_x^{fr})\sin\delta + (F_y^{fl} + F_y^{fr})\cos\delta + F_y^{rl} + F_y^{rr} + m\dot{\gamma}v_x \\
&(-F_x^{fl}d_l + F_x^{fr}d_r)\cos\delta + (F_y^{fl}d_l - F_y^{fr}d_r)\sin\delta \\
&+(F_x^{fl} + F_x^{fr})l_f\sin\delta + (F_y^{fl} + F_y^{fr})l_f\cos\delta \\
&+F_x^{rl}d_l - F_x^{rr}d_r - F_y^{rl}l_r - F_y^{rr}l_r = I\ddot{\gamma}
\end{aligned} \qquad (5)$$

In equation (5), $\dot{r}$ is the yaw rate.

The roll equation can be expresses as:

$$I_0\ddot{\varphi} + C_\varphi\dot{\varphi} + (K_\varphi - m_s gh)\varphi = m_s gha_{sy} \qquad (6)$$

In the above equation, C_φ and K_φ represent roll damping and roll stiffness of the suspension system accordingly, a_{sy} is the lateral acceleration of the C.G. of sprung mass system. This acceleration can be expressed as:

$$a_{sy} = a_{uy} - h\ddot{\varphi} \qquad (7)$$

Where a_{uy} is the lateral acceleration of the center of unsprung mass. Note that in equation (10), roll angle is assumed to be small.

Equation (5) to (7), are the key equations to solve for the rollover dynamics of the vehicle.

In order to assess and measure the rollover, we define a criterion that will indicate an estimation of tendency to rollover. The criterion is defined as Linear Transfer Ratio (LTR). LTR Expression based on the vertical tire forces is given by:

$$LTR = \frac{F_z^r - F_z^l}{F_z^r - F_z^l} \qquad (8)$$

Where F_z^r and F_z^l are left and right tire vertical forces.

Using the Newton second law in the vertical direction as well as equilibrium of roll moments, one can find LTR based on the parameter of the system:

$$LTR = \frac{F_z^r - F_z^l}{F_z^r - F_z^l} = \frac{2m_s}{mT}((h_o + h\cos\varphi)\frac{a_{sy}}{g} + h\sin\varphi) \qquad (9)$$

On a horizontal surface for a straight driving, LTR is equal to zero. If F_z^r or F_z^l approaches zero, this means that one of the wheels is lifting off the ground and vehicle is in the threshold of rollover. Therefore, the more LTR is deviated from zero the more vehicles' safety is jeopardized.

When the vehicle is driven with a constant speed, *v*, the equation 8-12 are simplified as:

$$\begin{aligned}
&mv\dot{\alpha} - m_s h\ddot{\varphi} = \dot{\gamma}\left[\frac{C_r l_r - C_f l_f}{v} - mv\right] + C_f\delta - (C_f + C_r)\alpha \\
&I\ddot{\gamma} = -(C_f l_f^2 + C_r l_r^2)\frac{\dot{\gamma}}{v} + (C_r l_r - C_f l_f)\alpha + C_f l_f \delta \\
&(I_0 + m_s h^2)\ddot{\varphi} + C_\varphi\dot{\varphi} + (K_\varphi - m_s gh)\varphi = m_s gv(\dot{\alpha} + \dot{\gamma})
\end{aligned} \qquad (10)$$

III. Fuzzy Stability Controller

The structure of the integrated vehicle and stability controller is shown in Fig.3. The stability controller inputs roll angle or lateral acceleration, and roll rate, and outputs the damping coefficient. Depending on increasing the roll angle, to each side, it adjusts the damping to reduce the roll angle and hence to increase the stability of the sprung mass. Using this stability control will help to reduce the propensity to roll over.

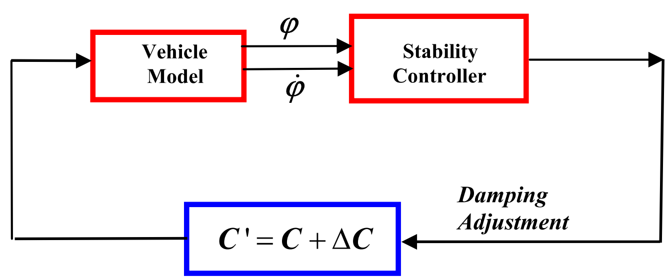

Fig. 3. Structure of the stability controller

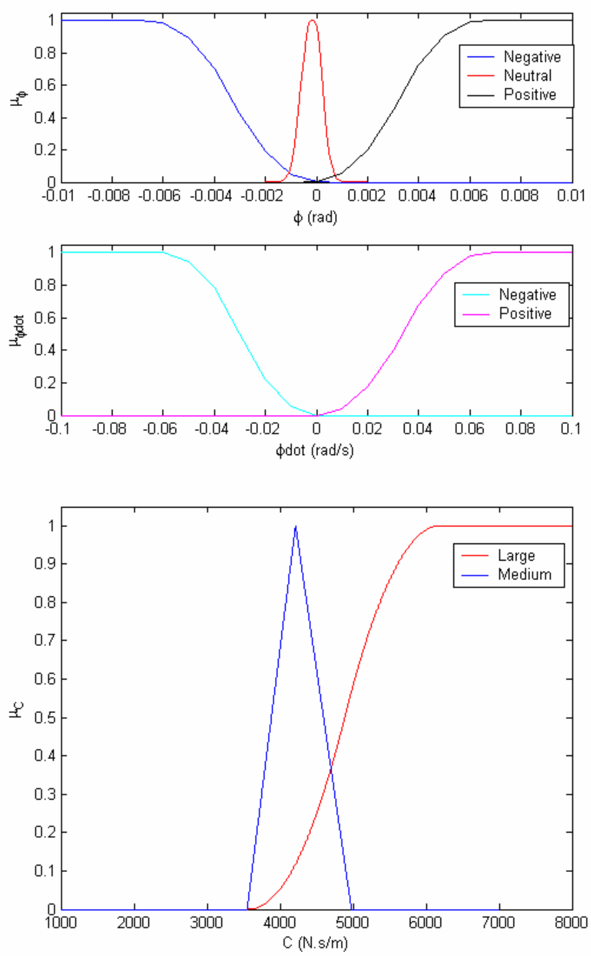

Fig. 4. Fuzzy membership functions

Table 1: Fuzzy logic rules

Roll Rate / Roll angle	Negative	Positive
Negative	Large	Medium
Neutral	Medium	Medium
Positive	Medium	Large

IV. Analysis and Results

To validate the stability controller, two different manoeuvres were conducted: double lane change and a simple turn. The results are reported accordingly.

Double lane change:

The vehicle is driven on a double lane change with the velocity of 30 km/h. The schematic of the double lane change maneuver is depicted below:

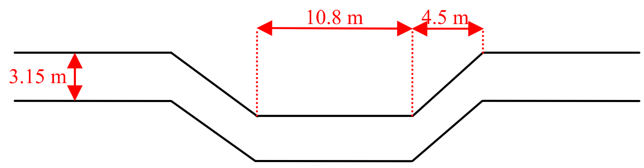

Fig. 5. Double lane change maneuver

2 controllers; Cadillac controller and fuzzy controller were implemented on the Cadillac.
The experimental results are shown in Fig 6. It should be mentioned that the peak values of the lateral acceleration is considered in the course of investigating the performance of the controllers. It is observed that the fuzzy controller outputs less lateral acceleration compared to the Cadillac controller. The fluctuations seen in before or after the turns are not important in studying the performance of the controllers.

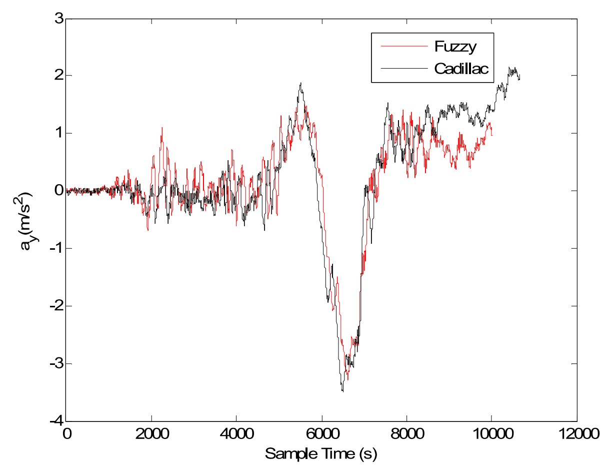

Fig. 6. Double lane change maneuver

Tables 2 and 3 tabulate the performance of the controllers in terms of percentage.

Table 2: RMS Controllers comparison of a_y in %

	Fuzzy	Cadillac
Fuzzy	-	8.31%

Table 3: Peak to peak comparison of a_y in %

	Fuzzy	Cadillac
Fuzzy	-	6.17%

Simple Turn:

Similar to the the double lane change maneuver, the vehicle is driven on a simple turn but with different velocity of 35 km/h. The schematic of the simple turn manoeuvre is depicted below:

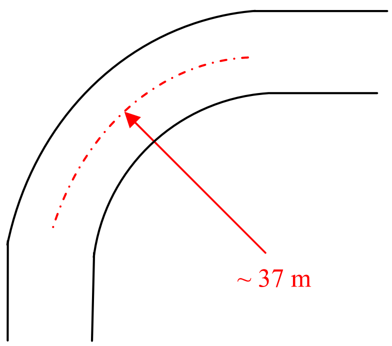

Fig. 6. Simple turn maneuver

Similarly, the results for the simple turn are shown in the Fig 8. It is shown that the fuzzy controller reduces lateral acceleration as well in this manuever.

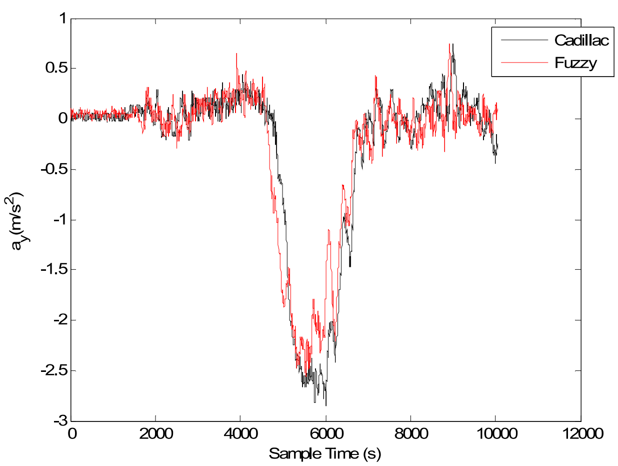

Fig. 8 Simple turn maneuver

Tables 4 and 5 tabulate the performance of the controllers in terms of percentage.

Table 4: RMS Controllers comparison of a_y in %

	Fuzzy	Cadillac
Fuzzy	-	8.23%

Table 5: Peak to peak comparison of a_y in %

	Fuzzy	Cadillac
Fuzzy	-	7.77%

V. Conclusions

In this paper, a novel fuzzy control strategy for a full car model with a semi-active suspension system was proposed to enhance rollover stability. To compare the effectiveness of the developed fuzzy controller, experimental analyses were performed on a Cadillac SRX. It was shown the proposed fuzzy controller reduced lateral acceleration and hence Linear Transfer Ratio (LTR) which is an indication of enhancement in rollover stability.

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