Rollover Propensity Control Via Active Suspension for Off-road Vehicle

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Abstract. A control strategy that improves rollover stability via active suspension is presented. The control structure is proposed considering the load transfer rate as the rollover propensity index. The controller is based on fuzzy logic and different stiffness on each active suspension limit the rollover propensity during cornering. Simulation results for off-road vehicle of fishhook test show that the proposed algorithm can effectively reduce rollover risk.

Introduction

It is well known that automobile rollover is an important traffic safety problem [1]. For off-road vehicle, there are two typical driving situations which can induce rollover. When cornering at high speed, lane change for obstacle avoidance or disturbance of side wind, vehicle rollover is caused directly, which is called ‘rollover on a plane surface’. The other rollover occurs when the vehicle hit an obstacle after skidding [2,3]. The elevated center of gravity (CG) is the most critical factor affecting rollover stability as a sacrifice for off-road vehicle's passing quality.

For common sense, rollover stability is also affected by the track width in addition to the CG height. But these parameters are nearly fixed especially for a particular car ignoring the impact of varying loadings [3]. As a result, the parameters on behalf of vehicle dynamic process during rollover, such as the load transfer, require more concern.

Physical characteristics of suspension act a critical role during the load transfer between wheels. In general when the vehicle is imminent to roll, stiffer suspension settings for active suspension are engaged to reduce the magnitude of the rollover detection metric, such as the roll angle, as well as the rollover propensity [4].

There are plenty of researches about ride comfort via active suspension and rollover via active steering or braking. A switched suspension setting was shown in [4] which used switched stable control and two settings of suspension to prevent rollover. [5] introduced a conceptual strategy that uses different actuating forces between the front and rear axle which causes a pitch motion along with the body roll to enhance handling of a vehicle during cornering maneuver while improving rollover stability.

As vehicle rollover is closely linked to the roll degree of freedom of vehicle dynamics, it is effective to directly control this degree of freedom by active suspension [6,7]. This paper focuses on the rollover avoidance by active suspension, while care more about the influence
of characteristics altering of active suspension, especially for stiffness biasing of left and right suspension.

The rest of the paper is organized as follows. The next section describes the system structure and the details about the rollover propensity index. Then the proposed control structure is presented, followed by simulation results. The summary of the work are given at last.

**System Setup**

Fig.1 shows the proposed control system architecture which contains a vehicle module, a rollover index compute and a controller of active suspension. The rollover index module uses the parameters from the vehicle as input and provides necessary signal to the active suspensions controller.

\[
LTR = \frac{2m}{mT} \left( h_s + h \cos \phi \right) \frac{a_s}{g} + h \sin \phi
\]

**Figure 1. System architecture.**

It’s uncomfortable for passengers to have stiffer suspension for common driving conditions. However soft roll stiffness implies roll propensity. As a result, there are two kinds of control strategy in the active suspension module, ride mode and rollover mode.

When the absolute value of rollover index over the set value, rollover model works to reduce rollover propensity. Otherwise ride model acts. The altering of suspension parameters can provide a trade-off between the comfort and safety requirements. But this paper only focuses on the rollover propensity controller.

The controller and the rollover index were built in MATLAB/Simulink [8]. The simulations were performed using the nonlinear dynamic vehicle model in CARSIM from Mechanical Simulations [9]. The important vehicle parameters are listed in following table.
Table 1. Numerical vehicle data.

| Parameter and numerical data | Descriptions                        |
|-------------------------------|--------------------------------------|
| $g = 9.81 \text{m/s}^2$       | Acceleration due to gravity          |
| $h = 1050 \text{mm}$          | Nominal height of CG over roll axis  |
| $h_r = 650 \text{mm}$         | Height of roll axis over ground      |
| $T = 1540 \text{mm}$          | Track width                          |
| $l_f = 1150 \text{mm}$        | Distance between front axle and CG   |
| $l_r = 1450 \text{mm}$        | Distance between rear axle and CG    |
| $m = 2030 \text{kg}$          | Vehicle mass                         |
| $m_s = 1592 \text{kg}$        | Sprung mass                          |
| 146kn/m (Front)               | Suspension spring rate               |
| 46kn/m (Rear)                 |                                      |

Rollover propensity index

Any rollover prevention controller requires a performance output or parameters of the vehicle to measuring the propensity of rollover. A lot of different index has been suggested in the literature. A rollover coefficient as defined in [3] was used in this paper. The load transfer ratio $L_{TR}$ is defined as the vertical load difference between the right and left wheels of the vehicle normalized by the total force.

Then

$$L_{TR} = \frac{F_{z,R} - F_{z,L}}{F_{z,R} + F_{z,L}}$$

$$= \frac{2m_s}{mT} \left[ (h_r + h \cos \phi) \frac{a_y}{g} + h \sin \phi \right]$$

(1)

Where $a_y$ means the lateral acceleration

$$a_y = u(\dot{\beta} + \dot{\gamma}) - h\ddot{\phi}$$

(2)

The system states are the velocity of the vehicle $u$, the side slip angle $\beta$, the yew rate $\gamma$, the roll angle $\phi$. $F_{z,R}$ and $F_{z,L}$ represent the tire vertical loads. $L_{TR}$ represents load transfer within an axle and ranges between -1 to 1. If the vertical load of right wheels is equal to zero $F_{z,R} = 0$, then the wheels lift off and the rollover propensity index, $L_{TR} = -1$. If a perfectly symmetric vehicle go straight driving on a horizontal road, then $L_{TR} = 0$. In general each axle will have different values for $L_{TR}$ at any time and the larger one is used as the rollover propensity index.
Rollover Propensity Control

As shown in Fig.1, the control structure was proposed considering the load transfer rate as the rollover propensity index. The suspension controller system used the signals from the vehicle and sent the altering of stiffness to the active suspension. The accuracy and time lag of the actuator were not taken into account in this article.

The rollover controller was based on fuzzy logic which took the velocity, the steering angle, the roll angle and the roll rate as inputs. It passed the altering value of stiffness to the active suspension. The rollover control region was selected to be above a rollover propensity index, \( LTR \), value of 0.5 (absolute).

Based on the fuzzy logic, there were two levels for the steering angle (left & right), the velocity as well as the roll rate (low & high) and five levels for the roll angle (clock-high, clock-middle, low, anti-clock-middle & anti-clock-high) as input. For the output, there were three levels for the altering: (low, middle & high).

Fig 2 shows the structure of the fuzzy logic system. The altering on the suspension depends on the velocity and the steering angle. If the cornering velocity is low, the controller only changes the outside of the suspension. On the other hand, at high velocity, the controller changes both left and right suspensions to enhance stability. The quantity of stiffness altering is affected by the roll angle and the roll rate.

![Figure 2. Fuzzy controller of rollover mode.](image)

Simulation Results

The simulations were performed using the commercial software SIMULINK/MATLAB and CARSIM. The vehicle parameters have listed in Tab.1. It is assumed that each simulation was performed at constant speed on a dry road condition (the road adhesion coefficient \( \mu = 0.85 \)).

In Fig.3, a steering wheel angle which applied to the vehicle is shown on the left. The velocity of the vehicle during the fishhook at an initial speed of 41 km/h is shown on the right.
Figure 3. Steering angle & velocity of the vehicle.

Fig. 4 shows the roll angle of the vehicle with active suspension controller and without. The comparison of roll rate during the test is shown in Fig. 4 on the right.

Figure 4. Roll angle & roll rate of the vehicle.

The proposed control strategy reduced the roll angle and the roll rate especially on peak value when the rollover mode activated. The rollover mode deploys stiffness altering of the active suspension to reduce rollover propensity at a cost of riding comfort during risk. That is the reason for rollover index compute firstly to decide which mode works.

Summary

In this paper a control strategy of active suspension that improves rollover stability during cornering maneuver was presented. Simulation results of fishhook test showed that both the roll angle and the roll rate were decreased. Hence, the proposed algorithm can effectively reduce rollover risk for off-road vehicle.

The performance of the system can be further improved with the application of advanced control methods or by independently control every active suspension actuation.
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