Research Paper

STEERING CONTROL FOR ROLLOVER AVOIDANCE OF HEAVY VEHICLE USING FUZZY TUNED PI CONTROLLER

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Abstract

The aim of this article is to develop an active steering assistance system in order to avoid the rollover of heavy vehicle. This approach is applied on single body model of heavy vehicle. The main objectives are to prevent rollover and minimize the deviation from the desired trajectory. The mathematical model of the vehicle is implemented using MATLAB as an initial work. Then found a measure for potential rollover, considering the term Load transfer ratio (LTRd) for that. The simulation for varying parameters like speed and center height of gravity is also covered in this research. Hence analyze the system response by tuning the PI parameter using fuzzy logic controller. Finally compare both controller responses for the given vehicle and choose a best controller among them.

Key Terms: Steering assistance system, Roll over, Load transfer ratio (LTRd), PI controller, fuzzy logic controller

1. Introduction

In recent years, rollover has been the subject of intensive research, especially by the major automobile manufacturers. That research is geared towards the development of rollover prediction schemes and occupant protection devices. It is however, possible to prevent such a rollover incident by monitoring the car dynamics and applying proper control effort ahead of time. Therefore there is a need to develop driver assistance technologies which would be transparent to the driver during normal driving conditions, while acting in emergency situations to recover handling of the vehicle until the driver recovers control of the vehicle.

In this a new approach was presented focusing on rollover avoidance by active steering. There, an actuator sets a small auxiliary front wheel steering angle in addition to the steering angle commanded by the driver. The aim was to robustly decrease the rollover risk due to transient roll overshoot of the vehicle's body when performing lane change or obstacle avoidance maneuvers. The control law consists of proportional feedback of both the roll rate and the roll angle. The gains were fixed according to robustness and performance considerations in parameter space and time domain. The resulting controller was shown to robustly reduce the maximum roll angle overshoot after steering input steps for large variations of the CG height in particular at high velocity. Moreover, the roll damping was robustly improved. In this controller was modified by gain scheduling against velocity and CG height to achieve comparably good results also at low speeds and different heights of CG. With this linear control concept, however, the vehicle may still roll over in case of too large steering wheel inputs.
This paper proposes a rollover control design using PI and fuzzy tuned PI and a final comparison among controller for ensuring robustness is also included in this work. Section 2 gives an idea about the research works going on this area. Section 3 describes the mathematical model of the system. Section 4 deals with rollover control design of PI controller and fuzzy tuned PI controller and simulation results. Section 5 discussed about the simulation analysis. Conclusion and outlook are given in section 6.

2. Literature Review

It is well known that vehicles with a high center of gravity such as vans, trucks and the highly popular SUVS (Sport Utility Vehicles) are more prone to rollover accidents. According to the 2004 data (NTHSA, 2006), light trucks (pickups, vans and SUVS) were involved in nearly 70% of all the rollover accidents in the USA, with SUVs alone responsible for almost 35% of this total. The fact that the composition of the current automotive fleet in the U.S. consists of nearly 36% pickups, vans and SUVs (Carlson et al., 2003), along with the recent increase in the popularity of SUVs worldwide, makes rollover an important safety problem.

Rollover is the second most dangerous car crash on American Highways. The year 2000, 9,882 people were killed in light vehicle rollover crashes, including 8,146 killed in single vehicle rollovers. Vehicles with high centers of gravity (center of gravity is hereafter denoted CG), e.g., Sport Utility Vehicles, SUVS, are becoming more and more popular. These vehicles are more likely to rollover during extreme maneuvers compared to ordinary cars (Forkenbrock et al. 2003). The tripped rollover is well understood, and it is responsible for the majority of the fatalities. The untripped rollover is not well understood, and it is only responsible for a small portion of the fatalities (Howe et al. 2001). Still, people are getting killed by un tripped rollovers, and research on this subject is motivated. Tripped rollovers can be avoided if a control system prevents the vehicle from uncontrolled skidding, e.g., Electronic Stability Program, ESP (van Zanten et al. 1996). Untripped rollovers can be avoided if a control system lets the vehicle deviate from the nominal trajectory.

Rollover prevention is a topical area of research in the automotive industry (see, for example, http://www.safercar.gov/Rollover for a good introduction to the problem) and several studies have recently been published. “Damping of vehicle roll dynamics by gain scheduled active steering in this paper active steering with PD-feedback of the roll rate was considered and the control was significantly reduce the rollover risk of a vehicle with a high center of gravity (CG) in transient steering maneuvers. Hence the controller gain scheduling with speed achieves good performance as a result of a combined approach in parameter space and by constrained optimization.” “untripped SUV rollover detection and prevention”In this paper they have covered a lot of objectives as to provide rollover detection by giving warning signal, rollover prevention by using linear control allocation technique. “A methodology for the design of robust rollover prevention controllers for automotive vehicles: Part 1-Differential braking” here also they are using LTRd term to prevent rollover but they are using a differential braking system. “Nonlinear steering and braking control for vehicle rollover avoidance” here a they considered that the vehicle dynamics control concept composed of steering and braking control which significantly reduces the rollover hazard caused by steering inputs. Gain scheduled continuous steering control forms an inner control loop which achieves improved roll dynamics. For rollover emergency case, an outer nonlinear steering control loop avoids rollover at the cost of some course deviation. This lane tracking error is, however, reduced by simultaneous deceleration which also supports rollover counteraction.

“A Methodology for the Design of Robust Rollover Prevention Controllers for Automotive Vehicles with Active Steering”, this paper they first develop a linearized vehicle model and then considered the term LTRd in order to avoid rollover. Then they first uses a PI controller to avoid rollover finally they proposed a robust control design to avoid rollover. “Second Order Sliding Modes: Theory and Applications” and “Sliding Mode Controllers for Active Suspensions” these papers give a detail idea about sliding mode technique.
3. Research Design and Methodology

In this gives a brief idea about the research design approach and methodology. In order to capture the salient features of vehicle rollover and for controller design purposes, we utilize the well known linearized vehicle model commonly referred as the single-track model (or bicycle model) with a roll degree of freedom; this is illustrated in Figure 3.1. This specific model or its variations are widely used in vehicle dynamics control applications (see for example Carlson et al. (2003), Takano et al. (2001), Ackermann et al. (1998), Odenthal et al. (1999), Chen et al. (2001), Hac et al. (2004), Kiencke et al. (2000)). In this linear model the steering angle $\delta$, the roll angle $\phi$, and the vehicle sideslip angle $\beta$ are all assumed to be small. We further assume that all the vehicle mass is sprung, which implies insignificant wheel and suspension weights.

![Figure 3.1. Single track model with roll degree of freedom.](image)

Also the lateral forces on the front and rear tires, denoted by $S_v$ and $S_h$, respectively, are represented as linear functions of the tire slip angles $\alpha_v$ and $\alpha_h$, that is, $S_v = C_v \alpha_v$ and $S_h = C_h \alpha_h$, where $C_v$ and $C_h$ are the front and rear tire stiffness parameters respectively. The assumptions of small angles and linear tire forces are probably an oversimplification of the nonlinear vehicle behavior at the rollover limit, yet these provide a good balance between capturing the salient features of vehicle behavior while keeping the complexity at a manageable level. In order to simplify the model description, we further define the following auxiliary variables

$$\sigma = C_v + C_h$$

$$\rho = C_h l_h - C_v l_v$$

$$\kappa = C_v l_v^2 + C_h l_h^2$$

where the lengths $l_v$ and $l_h$ are defined in Figure 3.1. For simplicity, it is assumed that the sprung mass rolls about a horizontal roll axis which is along the centerline of the track and at ground level. Using the parallel axis theorem of mechanics, $J_{seq}$, the moment of inertia of the vehicle about the assumed roll axis, is given by

$$J_{seq} = J_{xx} + mh^2$$

where $h$ is the distance between the center of gravity (CG) and the assumed roll axis and $J_{xx}$ is the moment of inertia of the vehicle about the roll axis through the CG. We introduce the state vector $\xi = [v_y \ \dot{\Psi} \ \dot{\phi} \ \dot{\Phi}]^T$

$v_y$: Lateral velocity of the vehicle,
$\dot{\Psi}$: Yaw rate of the undercarriage,
$\dot{\phi}$: Roll rate of the sprung mass about the roll axis,
$\dot{\Phi}$: Roll angle of the sprung mass about the roll axis.
The linearized equations of motion corresponding to this model are as follows

\[ \dot{\xi} = A\dot{\xi} + B\delta \]  \hspace{1cm} (3)

\[ A = \begin{bmatrix}
\frac{\sigma J_{xx}}{mv J_{xx}} & \frac{\rho J_{xx}}{mv J_{xx}} - \nu & -\frac{hc}{J_{xx}} & \frac{h(mgh-k)}{J_{xx}} \\
\frac{\rho}{J_{zzv}} & -\frac{\kappa}{J_{zz}} & 0 & 0 \\
-\frac{h\sigma}{\nu J_{xx}} & \frac{h\rho}{\nu J_{xx}} & -\frac{c}{J_{xx}} & \frac{(mgh-k)}{J_{xx}} \\
0 & 0 & 1 & 0
\end{bmatrix} 
\]

\[ B = \begin{bmatrix}
\frac{C_v J_{xx}}{mv J_{xx}} \\
\frac{C_{vl}}{J_{zz}} \\
hC_v & J_{xx} \\
0
\end{bmatrix} \]

\[ \ddot{\zeta} = \begin{bmatrix} 0 & 0 & -\frac{2c}{mgT} & -\frac{2k}{mgT} \end{bmatrix} \]

Model parameters appearing in (3) are given in Table 1.

### Table 1

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>1224</td>
<td>( [kg] )</td>
</tr>
<tr>
<td>( J_{xx} )</td>
<td>362.6</td>
<td>( [kg\cdot m^2] )</td>
</tr>
<tr>
<td>( J_{zz} )</td>
<td>1280</td>
<td>( [kg\cdot m^2] )</td>
</tr>
<tr>
<td>( l_v )</td>
<td>1.102</td>
<td>( [m] )</td>
</tr>
<tr>
<td>( l_l )</td>
<td>1.25</td>
<td>( [m] )</td>
</tr>
<tr>
<td>( T )</td>
<td>1.51</td>
<td>( [m] )</td>
</tr>
<tr>
<td>( h )</td>
<td>0.375</td>
<td>( [m] )</td>
</tr>
<tr>
<td>( c )</td>
<td>4000</td>
<td>( [N\cdot m\cdot s/rad] )</td>
</tr>
<tr>
<td>( k )</td>
<td>36075</td>
<td>( [N\cdot m/rad] )</td>
</tr>
<tr>
<td>( C_v )</td>
<td>90240</td>
<td>( [N/rad] )</td>
</tr>
<tr>
<td>( C_h )</td>
<td>180000</td>
<td>( [N/rad] )</td>
</tr>
</tbody>
</table>

3.1 The dynamic load transfer ratio, \( LTR_d \):

The load transfer ratio (Odenthal et al., 1999; Kamnik et al., 2003) can be simply defined as the load (i.e., vertical force) difference between the right and left wheels of the vehicle, normalized by the total load (i.e., the weight of the car). In other words,

\[ \text{Load transfer Ratio} = \frac{\text{Load on right tires} - \text{Load on left tires}}{\text{Total weight}} \]  \hspace{1cm} (4)

Clearly, this quantity varies between \(-1\) and \(1\), and for a perfectly symmetric vehicle and driving in a straight line, it is zero. The extrema are reached in the case of a wheel lift-off on one side of the vehicle, in which case the load transfer ratio is 1 or \(-1\) depending on the side that lifts off. If roll dynamics are ignored, it is easily shown (Odenthal et al., 1999) that the corresponding load transfer ratio (which we denote by \( LTR_s \)) is approximated by

\[ LTR_s = \frac{2a_y h}{gT} \]  \hspace{1cm} (5)
where $a_y$ is the lateral acceleration of the CG and $T$ is the vehicle track width. Note that rollover estimation based upon (5) is not sufficient to detect the transient phase of rollover (due to the fact that it is derived ignoring roll dynamics). In (Solmaz et al., 2006) we obtain an exact expression for the vehicle load transfer ratio which does not ignore roll dynamics; we denote this by $LTR_d$. To aid exposition we repeat the derivation here. Recall that we assumed the unsprung mass weight to be insignificant and the main body of the vehicle rolls about an axis along the centerline of the track at the ground level. We can write a torque balance for the unsprung mass about the assumed roll axis in terms of the suspension torques and the vertical wheel forces as follows:

$$-F_R \frac{T}{2} + F_L \frac{T}{2} + k\dot{\phi} + c\ddot{\phi} = 0$$

(6)

Now substituting the definition of load transfer from (4) and rearranging yields the following expression for $LTR_d$:

$$LTR_d = \frac{2}{mgT} (c\dot{\phi} + k\dot{\phi})$$

(7)

In terms of the state, $LTR_d$ can be represented by the following relationship

$$LTR_d = \tilde{C}\xi$$

$$\tilde{C} = [0 0 \frac{-2v}{mgT} \frac{-2k}{mgT}]$$

We also assume in this paper that all the model parameters $m, J_{xx}, J_{zz}, l_v, l_h, C_v, C_h, k, c$ are known. This is an unrealistic assumption: yet our control design is easily extended to account for uncertainty in these parameters which we demonstrate by designing our controllers to be robust with respect to uncertainties in vehicle speed $v$ and center of gravity height $h$. As a side note, although we assumed all the vehicle model parameters to be known, it is possible to estimate some of these that are fixed (but unknown) using the sensor information available for the control design suggested here.

### 3.2 Mathematical model of system:

**Case 1:** Vehicle speed=140km/h, Centre height of gravity=0.375m & Peak steering magnitude=$100^\circ$. Further definitions of the parameters appearing in (3) are given in Table 1.

$$\dot{A} = \begin{bmatrix}
-8.376 & -43.97 & -4.14 & -32.7 \\
2.54 & -7.89 & 0 & 0 \\
-7.198 & 3.36 & -11 & -87.2 \\
0 & 0 & 1 & 0 
\end{bmatrix} \quad \dot{B} = \begin{bmatrix}108.77 \\
77.75 \\
93.48 \\
0 \end{bmatrix}$$

$$\tilde{C} = \begin{bmatrix}0 & 0 & 0.4419 & 3.98 \end{bmatrix}$$

**Case 2:** Vehicle speed=70km/h, Centre height of gravity=0.45m & Peak steering magnitude=$100^\circ$

$$\dot{A} = \begin{bmatrix}
-19.172 & -10.44 & -4.97 & -38.13 \\
5.09 & -15.82 & 0 & 0 \\
-17.32 & 8.09 & -11.05 & -84.7 \\
0 & 0 & 1 & 0 
\end{bmatrix} \quad \dot{B} = \begin{bmatrix}124.2 \\
77.75 \\
112.18 \\
0 \end{bmatrix}$$

$$\tilde{C} = \begin{bmatrix}0 & 0 & 0.4419 & 3.98 \end{bmatrix}$$

### 3.3 Rollover Prevention Controller design (PI controller):

Mathematical model used for the analysis is of the form

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In this paper we designate the driver commanded input $\delta_d$ to be the disturbance input ($\omega$) and active steering input $\delta_c$ as the control input.

\[ \omega = \delta_d \]  
\[ u = \delta_c \]  

Hence the total steering angle $\delta = \delta_d + \delta_c$. In this problem we considered a proportional integral (PI) type state feedback controller of the form.

\[ u = K_p e + K_i \int e \, dt \]  
where \( e = e(t) = \phi_{ref} - \phi \)

Reference yaw rate $\phi_{ref}$ is the steady yaw rate which result from a constant driver input and zero control input.

\[ \phi_{ref} = \alpha \delta_d \]

$\alpha$ is the constant gain. The control structure is schematically depicted on figure 3.2 and the parameter $h = [0 \ 1 \ 0 \ 0]$.

Figure 3.2  Active steering system using PI controller

3.4 Fuzzy tuned PI controller Design:

Manually tuning PI parameters based on the vehicle parameter variations is a difficult task in order to overcome these difficulties a fuzzy tuned PI controller is introduced. FLC has four main components: the fuzzifier, knowledge base, inference mechanism and defuzzifier. Based on membership functions and fuzzy logic, the fuzzifier converts a crisp input signal to fuzzified signals. The knowledge base houses rule base and the data base. The inference mechanism fires relevant control rules and then decides the plant input. Finally the defuzzification process converts the fuzzy output into crisp control signal. Fuzzy supervisory controller attempts to provide nonlinear action for the controller output using fuzzy reasoning. The PI gains are tuned based on a fuzzy inference system rather than the conventional approaches. The inputs considered are error and change in error. The input ranges are obtained by simulations using the minimum and maximum range of steering angle. Error ranges from [-1 2000] and change in error ranges from [-3000 3000]. The outputs are $K_p$ ranges from [0 100], $K_i$ ranges from [1 15]. Fuzzy control input error has five fuzzy sets Zero (Z), Positive Small (PS), Positive Medium (PM), Positive Big (PB) and Positive Big (PVB) and control input change in error has seven fuzzy sets Negative Big (NB), Negative medium (NM), Negative Small (NS), Zero (Z), Positive Small (PS), Positive Medium (PM) and Positive Big (PB). The input membership functions are shown in figure 3.3. The output of fuzzy set, $K_p$ has two fuzzy sets small and big whereas $K_i$ have three
fuzzy sets small, medium and big. The output membership functions are shown in figure 3.4. Mamdani method is used for the fuzzy inference. There are general rules of thumb for tuning PI parameters and the rule bases are given in table 2.

1. If the input is positive large, then the proportional gain $K_p$ must be large, integral term $K_i$ small; hence, system output speed increases.

2. If the input is very small, then the PI parameters $K_p$ should be smaller, $K_i$ larger; thus, the output will have reduced overshoot and faster response.

3. These types of rules are not easy to implement using traditional tuning methods; however, this is very helpful for intelligent tuning.

![Figure 3.3 Input membership functions](image1)

![Figure 3.4 Output membership functions](image2)

**Table 2**

<table>
<thead>
<tr>
<th>E</th>
<th>CE</th>
<th>Z</th>
<th>PS</th>
<th>PB</th>
<th>PVB</th>
</tr>
</thead>
<tbody>
<tr>
<td>NB</td>
<td>PS</td>
<td>PS</td>
<td>PB</td>
<td>PVB</td>
<td></td>
</tr>
<tr>
<td>NM</td>
<td>PS</td>
<td>PB</td>
<td>PB</td>
<td>PVB</td>
<td></td>
</tr>
</tbody>
</table>
4. Simulation Results and Discussion:

By using a PI controller, analyze the vehicle dynamics and control the rollover of vehicle. The simulations are done for different parameters such as vehicle velocity, centre height of gravity and peak steering magnitude also different types of inputs like step and square wave are also included in this research. First applied a step input of magnitude 1.75 (rad) to the system i.e. without control then considered the system with PI control and a fuzzy tuned PI controller for tuning PI gains. Finally did the same analysis for square wave with different parameters.

**Step input:** Vehicle speed=140km/h, Centre height of gravity=0.375m & Peak steering magnitude=100°

**Case 1:** Without control

**Case 2:** With control $K_p=1; K_i=1$ and fuzzy PI controller for a step input of magnitude 100°
4.2 Square wave input: Vehicle speed=140km/h, Centre height of gravity=0.375m & Peak steering magnitude=100°

Case 1: Without control

Case 2: With PI control $K_p=1; K_i=1$, and fuzzy PI controller for a square wave input of magnitude 100°
Figure 4.7 PI controller output for square wave, Figure 4.8 Fuzzy PI controller output for square wave

\( v=140\text{km/h}, h=0.375, \delta=100^\circ \)  
\( v=140\text{km/h}, h=0.375, \delta=100^\circ \)

4.3 Step input: Vehicle speed=70km/h, Centre height of gravity=0.45m & Peak steering magnitude=150°

Case 1: Without control

Figure 4.9 Vehicle steering angle of step magnitude 150°(2.62 radians)

Figure 4.10 LTRd curve without controller  
\( h=0.45m, v=70\text{km/h}, \delta = 150^\circ \text{(step input)} \)

Case 2: With control \( K_p=1; K_r=1 \) and applied a step input of magnitude 150°
4.4 Square wave input: Vehicle speed=70km/h, Centre height of gravity=0.45m & Peak steering magnitude=150°

Case 1: Without control

Case 2: With control $K_p=1; K_f=1$ and applied a square wave input of magnitude 150°(2.62 radians)
**Figure 4.15** PI controller output for square wave, **Figure 4.16** Fuzzy PI controller output for square wave

\[ v=70 \text{km/h}, h=0.45, \delta=150^0 \]

**4.5 Simulation Analysis:**

<table>
<thead>
<tr>
<th>Type of input</th>
<th>Vehicle speed (km/h)</th>
<th>Steering angle (degree)</th>
<th>Centre height of gravity (m)</th>
<th>Control strategy</th>
<th>LTRd</th>
<th>Settling Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step</td>
<td>140</td>
<td>100</td>
<td>0.375</td>
<td>Without control</td>
<td>18 - 22</td>
<td>1.5</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>PI controller</td>
<td>1.44 - 3.5</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fuzzy PI</td>
<td>0.8 - 1.1</td>
<td>1.1</td>
</tr>
<tr>
<td>Square</td>
<td>140</td>
<td>100</td>
<td>0.375</td>
<td>Without control</td>
<td>-26 - 22</td>
<td>0.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>PI controller</td>
<td>-5 - 3.5</td>
<td>0.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Fuzzy PI</td>
<td>1.1 - 1.4</td>
<td>0.8</td>
</tr>
<tr>
<td>Step</td>
<td>70</td>
<td>150</td>
<td>0.45</td>
<td>Without control</td>
<td>18 - 20</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>PI controller</td>
<td>1.2 - 3.6</td>
<td>1.9</td>
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<td></td>
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<td>Fuzzy PI</td>
<td>0.7 - 1.1</td>
<td>1.3</td>
</tr>
<tr>
<td>Square</td>
<td>70</td>
<td>150</td>
<td>0.45</td>
<td>Without control</td>
<td>-24 - 21</td>
<td>0.8</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>PI controller</td>
<td>-5.6 - 3.6</td>
<td>0.9</td>
</tr>
</tbody>
</table>
Table 4.1 Simulation analysis of PI controller

| Controller  | Fuzzy PI | -1.6 – 1.1 | 0.7 |

From the analysis it's clear that for fuzzy PI the output value lies below the 1 except at the starting instants. But still as compared with other case its peak value and settling time is also low. Hence Fuzzy PI controller is the best controller among them. Because in this system some parameters are continuously varying with situation in that point of view the better controlling effect is provided by fuzzy PI controller.

5. Conclusion

This research is mainly focusing on the methodology for the design of vehicle rollover prevention system using active steering actuators by introducing the term LTR. Firstly, simulation using a PI controller and fuzzy PI controller by applying different types of input and parameter variations are included in this research work. A comparative analysis of output response and settling time with variation in parameters are also covered in this research. From the obtained results it's clear that in case of vehicle system to avoid rollover fuzzy PI controller is best as compared with PI controller.

Reference


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