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H_∞ active anti-roll bar control to prevent rollover of heavy vehicles: a robustness analysis

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Abstract: Rollover of heavy vehicle is an important road safety problem world-wide. Although rollovers are relatively rare events, they are usually deadly accidents when they occur. In order to improve roll stability, most of modern heavy vehicles are equipped with passive anti-roll bars to reduce roll motion during cornering or riding on uneven roads. This paper proposes an H_∞ approach to design active anti-roll bars using the yaw-roll model of a single unit heavy vehicle. The control signals are the torques generated by the actuators at the front and rear axles. Simulation results in both frequency and time domains are provided to compare two different cases: passive anti-roll bars and H_∞ active anti-roll bars. It is shown that the use of two H_∞ active (front and rear) anti-roll bars drastically improves the roll stability of the single unit heavy vehicle to prevent rollover.

Keywords: Vehicle dynamics, Active anti-roll bar control, Rollover, Roll stability, H_∞ control, μ-analysis.

1. INTRODUCTION

1.1 Context

The rollover is a very serious problem for heavy vehicle safety, which can result in large financial and environmental consequences. In order to improve roll stability, most of modern heavy vehicles are equipped with **passive anti-roll bars** to reduce roll motion. The passive anti-roll bar has the advantages to reduce the body roll acceleration and roll angle during single wheel lifting and cornering maneuvers. However, the passive anti-roll bar also has drawbacks. During cornering maneuvers, it transfers the vertical forces of one side of suspension to the other one, creating therefore a moment against lateral force.
In order to overcome the drawbacks of the passive anti-roll bar systems, several schemes with possible active intervention into the vehicle dynamics have been proposed. One of them employs **active anti-roll bars**, that is, a pair of hydraulic actuators which generate a stabilizing moment to balance the overturning moment. Lateral acceleration makes vehicles with conventional passive suspension tilt out of corners. The center of the sprung mass shifts outboard of the vehicle centerline, which creates a destabilizing moment that reduces roll stability. The lateral load response is reduced by active anti-roll bars which generate a stabilizing moment to counterbalance the overturning moment in such a way that the control torque leans the vehicle into the corners (see Sampson and Cebon (2003), Gaspar et al. (2004)). Other methods can be used (active steering, electronic brake mechanism,...) but they are beyond the scope of this paper.
The disadvantage of the active anti-roll bars is that the maximum stabilizing moment is limited physically by the relative roll angle between the body and the axle (Sampson and Cebon (2002)).

1.2 Related works

Some of the control methods applied for active anti-roll bar control on heavy vehicle are briefly presented below:
a- Optimal control: Sampson *et al* (see Sampson and Cebon (1998), Sampson and Cebon (2002)) have proposed a state feedback controller which was designed by finding an optimal controller based on a linear quadratic regulator *(LQR)* for single unit and articulated heavy vehicles.
The *LQR* was also applied to the integrated model including an electronic servo-valve hydraulic damper model and a yaw-roll model of a single unit heavy vehicle. The input control signal is the input current of the electronic servo-valve (Vu et al., 2016).
b- Neural network control: Boada et al. (2007) proposed a reinforcement learning algorithm using neural networks to improve the roll stability for a single unit heavy vehicle.
c- Robust control *(LPV)***:** Gaspar *et al* (see Gaspar et al. (2005a), Gaspar et al. (2004) and Gaspar et al. (2005b)) have applied Linear Parameter Varying *(LPV)* technique for the active anti-roll bar combined with an active brake control on the single unit heavy vehicle. The forward velocity is considered as the varying parameter.

1.3 Paper contribution

Based on the model presented in (Gaspar et al. (2004)), this paper proposes an H_∞ control method for active anti-roll bar, that focuses on the uncertainties due to the vehicle forward velocity and the sprung mass variations. Hence the following contributions are brought:
- We design an H_∞ robust controller for active anti-roll bar system on the single unit heavy vehicle. The aim is to maximize the roll stability to prevent rollover of heavy vehicles. The normalized load transfer and the limitation of the torque generated by actuators in various maneuver situations are considered.
- The performance analysis, made in frequency domain, shows

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that the H_∞ active anti-roll bar control drastically reduces the normalized load transfer, compared to the passive anti-roll bar. It also shows that the H_∞ active anti-roll bar control is robust w.r.t. the forward velocity and the sprung mass variation. The robust stability analysis of the designed controller is performed using the μ- analysis method.

- In time domain, we use a double lane change as the heavy vehicle maneuver. The simulation results indicate that the *Root Mean Square (RMS)* of the H_∞ active anti-roll bar control have dropped from 15% to 50% compared to the passive anti-roll bar with all the forward velocities considered in interval from $50Km/h$ to $110Km/h$.

The paper is organised as follows: Section 2 gives the model of a single unit heavy vehicle. Section 3 gives the H_∞ robust control synthesis to prevent rollover of heavy vehicles. Section 4 illustrates the robustness analysis in the frequency domain using the μ- tool. Section 5 presents the simulations in time domain. Finally, some conclusions are drawn in section 6.

2. SINGLE UNIT HEAVY VEHICLE MODEL

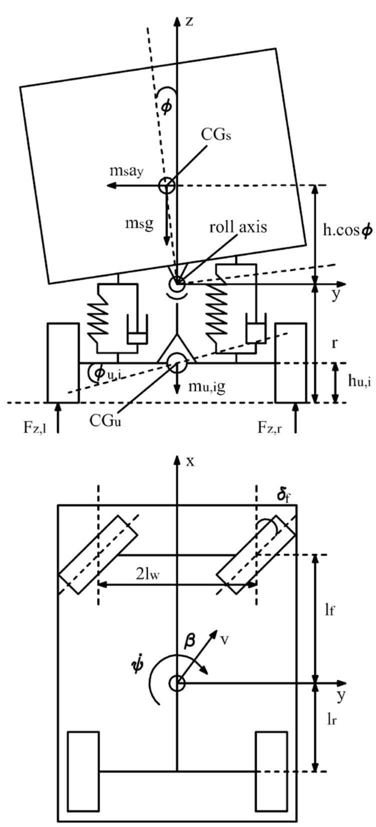

Fig. 1. Yaw-Roll model of single unit heavy vehicle (see Gaspar et al. (2004)).

Fig 1 illustrates the combined yaw-roll dynamics of the vehicle modelled by a three-body system, in which m_s is the sprung mass, m_{uf} the unsprung mass at the front including the front wheels and axle, and m_{ur} the unsprung mass at the rear with the rear wheels and axle. The parameters and variables of the yaw-roll model are shown in (Gaspar et al. (2004)).

In the vehicle modelling, the differential equations of motion of the yaw-roll dynamics of the single unit vehicle, i.e. the lateral dynamics, the yaw moment, the roll moment of the sprung mass, the roll moment of the front and the rear unsprung masses, are formalized in the equations (1):

$$\begin{cases} mv(\dot\beta+\dot\psi)-m_sh\ddot\phi = F_{yf}+F_{yr} \\ -I_{xz}\ddot\phi+I_{zz}\ddot\psi = F_{yf}l_f-F_{yr}l_r \\ (I_{xx}+m_sh^2)\ddot\phi-I_{xz}\ddot\psi = m_sgh\phi+m_svh(\dot\beta+\dot\psi) \\ \quad -k_f(\phi-\phi_{tf})-b_f(\dot\phi-\dot\phi_{tf})+M_{ARf}+U_f \\ \quad -k_r(\phi-\phi_{tr})-b_r(\dot\phi-\dot\phi_{tr})+M_{ARr}+U_r \\ -rF_{yf} = m_{uf}v(r-h_{uf})(\dot\beta+\dot\psi)+m_{uf}gh_{uf}.\phi_{tf}-k_{tf}\phi_{tf} \\ \quad +k_f(\phi-\phi_{tf})+b_f(\dot\phi-\dot\phi_{tf})+M_{ARf}+U_f \\ -rF_{yr} = m_{ur}v(r-h_{ur})(\dot\beta+\dot\psi)-m_{ur}gh_{ur}\phi_{tr}-k_{tr}\phi_{tr} \\ \quad +k_r(\phi-\phi_{tr})+b_r(\dot\phi-\dot\phi_{tr})+M_{ARr}+U_r \end{cases} \tag{1}$$

In (1) the lateral tire forces F_{yf} and F_{yr} in the direction of velocity at the wheel ground contact points are modelled by a linear stiffness as:

$$\begin{cases} F_{yf} = \mu C_f\alpha_f \\ F_{yr} = \mu C_r\alpha_r \end{cases} \tag{2}$$

with tire side slip angles:

$$\begin{cases} \alpha_f = -\beta+\delta_f-\dfrac{l_f\dot\psi}{v} \\ \alpha_r = -\beta+\dfrac{l_r\dot\psi}{v} \end{cases} \tag{3}$$

The moment of passive anti-roll bar impacts the unsprung and sprung masses at the front and rear axles as follows (Vu et al., 2016):

$$M_{ARf} = 4k_{AOf}\frac{t_At_B}{c^2}\phi-4k_{AOf}\frac{t_A^2}{c^2}\phi_{uf} \tag{4}$$

$$M_{ARr} = 4k_{AOr}\frac{t_At_B}{c^2}\phi-4k_{AOr}\frac{t_A^2}{c^2}\phi_{ur} \tag{5}$$

where k_{AOf}, k_{AOr} are respectively the torsional stiffness of the anti-roll bar at the front and rear axles, t_A half the distance of the two suspensions, t_B half the distance of the chassis and c the length of the anti-roll bars's arm.

Using the previous equation, the single unit heavy vehicle is represented by the linear system in the state space form (6):

$$\begin{cases} \dot x = Ax+Bu \\ y = Cx \end{cases} \tag{6}$$

with the state vector:

$$x = \begin{bmatrix} \beta & \dot\psi & \phi & \dot\phi & \phi_{uf} & \phi_{ur} \end{bmatrix}^T$$

the input vector:

$$u = \begin{bmatrix} \delta_f & U_f & U_r \end{bmatrix}^T$$

and the output vector:

$$y = \begin{bmatrix} \beta & \dot\psi & \phi & \dot\phi & \phi_{uf} & \phi_{ur} \end{bmatrix}^T$$

Remark: Note that matrix A mainly depends on the forward velocity (V) and the sprung mass (m_s). The design of H_∞ controller will be done considering the nominal matrix A and the effect of uncertainties will be analysed in section 4.

3. H_∞ ROBUST CONTROL SYNTHESIS TO PREVENT ROLLOVER OF HEAVY VEHICLES

3.1 Control objective, problem statement

The objective of the active anti-roll bar control system is to maximize the roll stability of the vehicle. An imminent rollover can be detected if the calculated normalized load