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A FAULT-TOLERANT VEHICLE CONTROL DESIGN ⋆

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Abstract: To improve the performance properties of heavy vehicles, i.e. to reduce the risk of rollovers, improve passenger comfort and road holding, a reconfigurable fault-tolerant control design of the active suspensions and the active brake is performed. However when a fault (loss in effectiveness) occurs at one of the suspension actuators a reconfiguration is needed in order to maintain the same performance level. The proposed reconfiguration scheme is based on an $\mathcal{H}_\infty$ Linear Parameter Varying (LPV) method that uses the fault information as one of the scheduling variables. The LPV based control design and the operation of the control mechanism are demonstrated in a vehicle maneuver.

Keywords: fault-tolerant control; reconfiguration; fault detection; linear parameter varying control; nonlinear modelling; robust control; road vehicle.

1. INTRODUCTION

These days there is a growing demand for vehicles with ever better driving characteristics, in which efficiency, safety, and performances, such as passenger comfort, road holding, rollover stability, yaw stability, suspension working space and energy consumption, are ensured. Several individual active control mechanisms are applied in road vehicles to solve different control tasks. In this paper a combined control mechanism is proposed which creates a balance between different components in order to enhance the performances and safety of the vehicle. The control mechanism includes active suspensions and an active brake. The role of the active suspension system is to improve passenger comfort, i.e. to reduce the effect of harmful vibrations on the vehicle and passengers, Gillespie [1992], Hrovat [1997], Gáspár et al. [2003b]. The role of the active brake is to apply unilateral braking since it reduces the lateral tire forces directly and decelerates the vehicle, Chen and Peng [2001], Palkovics et al. [1999], Gáspár et al. [2003a].

In this paper the control components mentioned above are integrated in order to improve the efficient and reliable operations. The goal is to design a combined controller that uses an active suspension system all the time to improve passenger comfort, road holding and guarantee the suspension working space. It activates the controlled braking system only when the vehicle comes close to rolling over. When a rollover is imminent the active suspension system generates a stabilizing moment to balance an overturning moment. When this dangerous situation persists, the active brake system must generate unilateral brake forces in order to reduce the risk of rollovers. This is an integrated control system, since several actuators operate in co-operation with each other and meet different performance requirements, i.e. passenger comfort, road holding, rollover prevention and fault-tolerant operation.

In order to design a fault-tolerant control, fault information must be used as an input signal. Thus, in case of a failure an adequate tuning of the control mechanism guarantees roll stability. It means that if a hydraulic actuator fault occurs in the active suspension system, the control system assume the role of the fault suspension component to enhance rollover prevention. In the case of a detected failure the operation of the control mechanism must be modified. For this purpose, the fault parameter is also applied in the control design. The solution of the fault-tolerant operation is based on the reconfigurability of the active brake. The basis of this solution is that the active brake is able to modify the yaw dynamics to reduce the rollover risk.

In this paper the model for control design is constructed in a Linear Parameter Varying (LPV) structure, in which changes in forward velocity and other varying variables are selected as scheduling parameters. The LPV modeling techniques allow us to take into consideration the nonlinear effects in the state space description, thus the model structure is nonlinear in the parameters, but linear in the states. In the control design the performance specifications both for rollover and suspension problems, and the model uncertainties are also taken into consideration. The solution is based on the reconfigurability of the active brake: if a fault occurs in the active suspension system and it is detected by a Fault Detection and Identification (FDI) filter, the brake is activated and the rollover is prevented. A method for the design of the FDI filter has also been proposed.

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The structure of the paper is as follows. In Section 2 the combined yaw-roll model of heavy vehicles is presented. In Section 3 the design of the FDI filter is presented. In Section 4 the performance objectives, the uncertainty components and the weighting functions in the reconfigurable control structure are defined and the control design of the suspension system is presented. In Section 5 the operation of the rollover prevention system is demonstrated through simulation examples. Section 6 contains some concluding remarks.

2. THE LPV MODELING OF YAW, ROLL AND VERTICAL DYNAMICS

Figure 1 illustrates the combined yaw-roll dynamics of the vehicle, which is modelled by a three-body system, in which m_s is the sprung mass, $m_{u,f}$ is the unsprung mass at the front including the front wheels and axle, $m_{u,r}$ is the unsprung mass at the rear with the rear wheels and axle, and m is the total vehicle mass. The signals are the lateral acceleration a_y, the side slip angle of the sprung mass β, the heading angle ψ, the yaw rate $\dot{\psi}$, the roll angle ϕ, the roll rate $\dot{\phi}$, the roll angle of the unsprung mass at the front axle $\phi_{t,f}$ and at the rear axle $\phi_{t,r}$. δ_f is the front wheel steering angle. v is the forward velocity.

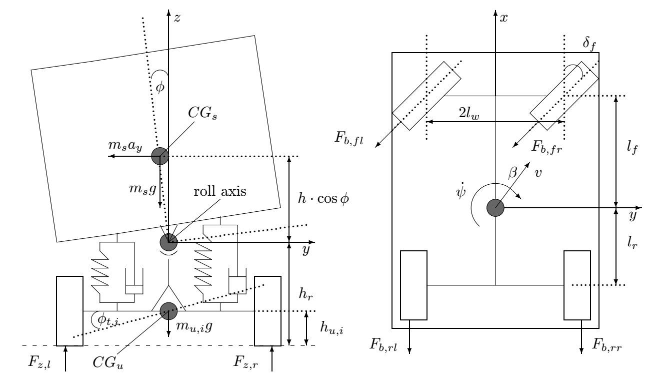

Fig. 1. The yaw-roll model for control design

The motion of the sprung mass is damped by suspensions with damping coefficients b_f, b_r and stiffness coefficients k_f, k_r. The tire stiffnesses are denoted by $k_{t,f}$, $k_{t,r}$. This structure includes three control mechanism, which generate control inputs. They are the difference in brake forces between the left and right-hand side of the vehicle ΔF_b, and roll moments between the sprung and unsprung masses generated by control forces u_{s1} and u_{s2} with the half of the vehicle width ℓ_w. The relationship between the actual forces and the fictitious forces is the following: $u_{s1} = u_{fl} - u_{fr}$ and $u_{s2} = u_{rl} - u_{rr}$.

In the design of suspension systems a simplified half-car model is substituted for the full-car model, since the roll motion of the vehicle is handled in the modelling of the rollover system. The half-car vehicle model, which is shown in Figure 2, comprises three parts: the sprung mass m_s and two unsprung masses m_{uf}, m_{ur}. The sprung mass is assumed to be a rigid body and has freedoms of motion in the vertical and pitch directions. x_1 denotes the vertical displacement at the center of gravity and θ is the pitch angle of the sprung mass. The front and rear displacements of the sprung and the unsprung masses are denoted by x_{1f}, x_{1r} and x_{2f}, x_{2r}, respectively. The disturbances in the suspension system are caused by road irregularities w_f, w_r. The input forces of the suspension system are generated by the actuators u_{s3} and u_{s4}.

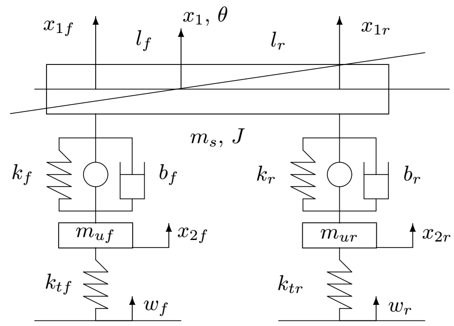

Fig. 2. The half-car model for suspension design

It is important to note that although the control design for the suspension system is based on the half-car model and the performance signals are the heave and the pitch accelerations, the control system designed is operated on the whole full-car model. The control of the roll dynamics is to be solved in the design of the rollover prevention system. In the half-car model the fictitious forces are moved to the center plane of the vehicle. The relationship between the actual forces and the fictitious forces are the following: $u_{s3} = u_{fl} + u_{fr}$ and $u_{s4} = u_{rl} + u_{rr}$.

The motion differential equations can be formalized both for the rollover and the suspension dynamics, and by selecting state variables the state space representation can be formalized.

$$\dot{x}_r = A_r(v)x_r + B_{1r}(v)\delta_f + B_{2r}(v)u_r \tag{1}$$
$$\dot{x}_s = A_s x_s + B_{1s} w + B_{2s} u_s \tag{2}$$

with $x_r = \begin{bmatrix}\beta \ \dot{\psi} \ \phi \ \dot{\phi} \ \phi_{t,f} \ \phi_{t,r}\end{bmatrix}^T$ and $x_s = \begin{bmatrix}x_1 \ \theta \ x_{2f} \ x_{2r} \ \dot{x_1} \ \dot{\theta} \ \dot{x_{2f}} \ \dot{x_{2r}}\end{bmatrix}^T$.

The disturbance signals are the front wheel steering angle δ_f, and the road disturbances w_f, w_r. The control inputs are the roll moments, the suspension forces and the braking force: $u_r = [u_{s1} \ u_{s2} \ \Delta F_b]^T$ and $u_s = [u_{s3} \ u_{s4}]^T$.

In the equation (1) the system matrices depend on the forward velocity of the vehicle nonlinearly. In the linear yaw-roll model the velocity is considered a constant parameter. However, forward velocity is an important parameter, so that it is considered to be a variable of the motion. One characteristics of the LPV system is that it must be linear in the pair formed by the state vector, x, and the control input vector, u. The matrices A and B are generally nonlinear functions of the scheduling vector ρ. If v is chosen as a scheduling parameter, the differential equations of the yaw-roll motion are linear in the state variables: $\rho = v$.

In this paper the detection of an imminent rollover is based on the monitoring of the lateral load transfers for both axles. The lateral load transfer can be given: $\Delta F_{z,i} = \frac{k_{t,i}\phi_{t,i}}{l_w}$, where i denotes the front and rear axles. The lateral load transfer can be normalized in such a way that the load transfer is divided by the total axle load:

$R_i = \frac{\Delta F_{z,i}}{F_{z,i}}$. If the normalized load transfer R_i takes on the value ± 1 then the inner wheels in the bend lift off. The normalized load transfer increases more quickly at the rear axle R_r than at the front axle R_f since the ratio of the effective roll stiffness to the axle load is larger at the driven axle. Thus, R_r is considered as a critical normalized load transfer.

3. FAULT DETECTION IN THE SUSPENSION SYSTEM

In the design of a reconfigurable control systems, besides the performances and uncertainties, fault information must also be taken into consideration. The fault signal is obtained by an FDI filter independently from the control system.

3.1 Inversion based FDI filter

For the linear system with m inputs and p outputs:

$$\dot{x}(t) = Ax(t) + Bu(t) \tag{3}$$
$$y(t) = Cx(t). \tag{4}$$

Denote by $\mathcal{V}^*$ the maximal parameter varying (A, B)-invariant subspaces contained in $KerC$. If $dim\, ImB = m$ and $\mathcal{V}^* \cap ImB = 0$ are fulfilled, the system is invertible and one can always choose a coordinate transform of the form

$$z = Tx, \text{ where } \quad T^{-1} = [\,\Lambda\ ImB\ \mathcal{V}^*\,], \quad \Lambda \subset \mathcal{V}^{*\perp},$$

such that the system will be decomposed to:

$$\dot{x}_1 = A_{11}x_1 + A_{12}x_2 + B_1 v \tag{5}$$
$$\dot{x}_2 = A_{21}x_1 + A_{22}x_2 \tag{6}$$
$$y = C_1 x_1. \tag{7}$$

Applying the feedback

$$u = F_1 x_1 + F_2 x_2 + v, \tag{8}$$

that makes $\mathcal{V}^*$ $(\mathcal{A} + BF, ImB)$ invariant, one can obtain the system:

$$\dot{x}_1 = A_{11}x_1 + B_1 v \tag{9}$$
$$y = C_1 x_1. \tag{10}$$

The linearly independent system $\tilde{y}$, where

$$\tilde{y} = \left[y_1, \cdots, y_1^{(\gamma_1)}, \cdots, y_p, \cdots, y_p^{(\gamma_p)}\right]^T. \tag{11}$$

spans the dual space of X_1, hence the inverse system can be expressed as:

$$\dot{\eta} = A_{22}\eta + A_{21}S^{-1}\tilde{y}, \tag{12}$$
$$u = F_2\eta + B_1^{-r}S^{-1}(\dot{\tilde{y}} - SA_{11}S^{-1}),$$

where B_1^{-r} is the right inverse of B_1 and S is the map from x_1 to $\tilde{y}$.

If the fault affected system is invertible for the fault, as input, one can apply the steps described above obtaining an inversion based fault detection filter, see Balas et al. [2004], Szabó et al. [2003].

This construction will be applied for a setting based on the quarter–car model.

3.2 FDI design for the suspension system

The state space representation of the quarter-car model is:

$$\dot{x}_1 = x_3, \tag{13}$$
$$\dot{x}_2 = x_4, \tag{14}$$
$$\begin{aligned}\dot{x}_3 = &-\left(\frac{k_s^l}{m_s} + \frac{k_s^{nl}}{m_s}\rho_k\right)x_1 + \left(\frac{k_s^l}{m_s} + \frac{k_s^{nl}}{m_s}\rho_k\right)x_2 - \\ &-\left(\frac{b_s^l}{m_s} - \frac{b_s^{sym}}{m_s}\rho_b\right)x_3 + \left(\frac{b_s^l}{m_s} - \frac{b_s^{sym}}{m_s}\rho_b\right)x_4 + \\ &+\frac{1}{m_s}b_s^{nl}\rho_b\sqrt{\rho_b(x_4 - x_3)} - \frac{1}{m_s}F,\end{aligned} \tag{15}$$
$$\begin{aligned}\dot{x}_4 = &\left(\frac{k_s^l}{m_u} + \frac{k_s^{nl}}{m_u}\rho_k\right)x_1 - \left(\frac{k_s^l}{m_u} + \frac{k_s^{nl}}{m_u}\rho_k + \frac{k_t}{m_u}\right)x_2 + \\ &+\left(\frac{b_s^l}{m_u} - \frac{b_s^{sym}}{m_u}\rho_b\right)x_3 - \left(\frac{b_s^l}{m_u} - \frac{b_s^{sym}}{m_u}\rho_b\right)x_4 + \\ &+\frac{k_t}{m_u}d - \frac{1}{m_u}b_s^{nl}\rho_b\sqrt{\rho_b(x_4 - x_3)} + \frac{1}{m_u}F,\end{aligned} \tag{16}$$

where $\rho_b = sgn(x_4 - x_3)$ and $\rho_k = (x_2 - x_1)^2$ are selected as scheduling variables. The control signal F is generated by the actuator. x_1 and x_2 denote the vertical displacement of the sprung mass and the unsprung mass, respectively. The disturbance d is caused by road irregularities.

Possible faults of the actuators (loss of effectiveness) can be detected by reconstructing the actual suspension forces F. Since the real actuators might present a saturation effect it is necessary to check, in addition, if the actual forces are lower then those corresponding to the saturation level of the actuators.

Having measured the signals $y_1 = \dot{x}_3, y_2 = \dot{x}_4$ and $y_3 = x_2 - x_1$ an inversion based detection filter is proposed. In the constructions of the filter the first step is to express F from the equations (15), (16) and in these expression we plug in the known values y_i:

$$\begin{aligned}F = &-\left(b_s^l - b_s^{sym}\rho_b\right)x_3 + \left(b_s^l - b_s^{sym}\rho_b\right)x_4 + \\ &+ b_s^{nl}\rho_b\sqrt{\rho_b(x_4 - x_3)} + \left(k_s^l + k_s^{nl}\rho_k\right)y_3 - m_s y_1,\end{aligned} \tag{17}$$
$$\begin{aligned}F = &\,k_t x_2 - \left(b_s^l - b_s^{sym}\rho_b\right)x_3 + \left(b_s^l - b_s^{sym}\rho_b\right)x_4 + \\ &+ b_s^{nl}\rho_b\sqrt{\rho_b(x_4 - x_3)} - k_t d - m_u y_2 + \\ &+ \left(k_s^l + k_s^{nl}\rho_k\right)y_3.\end{aligned} \tag{18}$$

By plugging back the obtained expressions in the original equations the resulting LPV system will have the same states as the original one and it will be observable with the output y_3:

$$\begin{aligned}\dot{x}_3 = &-\left(\frac{k_s^l}{m_s} + \frac{k_s^{nl}}{m_s}\rho_k\right)x_1 + \left(\frac{k_s^l}{m_s} + \frac{k_s^{nl}}{m_s}\rho_k\right)x_2 + \\ &+\left(\frac{k_s^l}{m_s} + \frac{k_s^{nl}}{m_s}\rho_k\right)y_3 - y_1,\end{aligned} \tag{19}$$
$$\begin{aligned}\dot{x}_4 = &\left(\frac{k_s^l}{m_u} + \frac{k_s^{nl}}{m_u}\rho_k\right)x_1 - \left(\frac{k_s^l}{m_u} + \frac{k_s^{nl}}{m_u}\rho_k\right)x_2 - \\ &-\left(\frac{k_s^l}{m_u} + \frac{k_s^{nl}}{m_u}\rho_k\right)y_3 + y_2,\end{aligned} \tag{20}$$

moreover, this equation has as a scheduling variable the measured signal $\rho_k = y_3^2$.

For a LPV system that depends affinely on the scheduling variables an LPV observer can be designed using LMI techniques: let us recall that an LPV system is said to be quadratically stable if there exist a matrix $P = P^T > 0$ such that $A(\rho)^T P + PA(\rho) < 0$ for all the parameters ρ. A necessary and sufficient condition for a system to be quadratically stable is that this condition holds for all the corner points of the parameter space, i.e., one can obtain a finite system of linear matrix inequalities (LMIs) that have to be fulfilled for $A(\rho)$ with a suitable positive definite matrix P, see Gahinet et al. [1996].

In order to obtain a quadratically stable observer the LMI $A_o^T(\rho)P + PA_o(\rho) < 0$ must hold for suitable $K(\rho)$ and $P = P^T > 0$, with $A_o = A + KC$. By introducing the auxiliary variable $L(\rho) = PK(\rho)$, one has to solve the following set of $LMIs$ on the corner points of the parameter space: $A(\rho)^T P + PA(\rho) - C^T L(\rho)^T - L(\rho)C < 0$.

By using the estimated state signals the values of the actual actual suspension forces F are computed using expression (17).

The actuator which generates the necessary force for the suspension system is a four-way valve-piston system. Denoting by z the relative displacement one has $F = A_P P_L$, where A_P is the area of the piston and P_L is the pressure drop across the piston with respect to the front and rear suspensions. The derivative of P_L is given by

$$\dot{P}_L = -\beta P_L + \alpha A_P z + \gamma Q, \tag{21}$$

in which $Q = Q_0 x_v$ is the hydraulic load flow (with the notation $Q_0 = sgn(r)\sqrt{|r|}$ and $r = P_S - sgn(x_v)P_L$, moreover, α, β, γ are constants, $z(= \dot{x}_2 - \dot{x}_1)$ is the the relative velocity, P_S is the supply pressure and x_v is the displacement of the spool valve. The cylinder velocity acts as a coupling from the position output of the cylinder to the pressure differential across the piston. It is considered a feedback term, which has been analyzed by Alleyne and Hedrick [1995], Alleyne and Liu [2000].

The displacement of the spool valve is controlled by the input to the servo-valve u:

$$\dot{x}_v = \frac{1}{\tau}\left(-x_v + u\right). \tag{22}$$

where τ is a time constant.

Let x_5 and x_6 denote P_L and x_v, respectively. Then, the actuator model can be written separately as

$$\dot{x}_5 = -\beta x_5 + \gamma Q_0 x_6 + \alpha A_P z, \tag{23}$$

$$\dot{x}_6 = -\frac{1}{\tau}x_6 + \frac{1}{\tau}u_a. \tag{24}$$

By using a backstepping technique one can obtain the values of u_a that corresponds to the given suspension force under given operational conditions, Gáspár and Szederkényi [2007].

4. THE DESIGN OF A RECONFIGURABLE CONTROL

In the control design both the rollover and the suspension problems are taken into consideration. In this combined structure a new weighting strategy is proposed in order to meet several performance demands, such as enhancing passenger comfort, increasing rollover stability and road holding, guaranteing suspension working space and reducing energy consumption. In the rollover problem the performance outputs for control design are the lateral acceleration, the lateral load transfers at the front and the rear, and the control inputs: $z_r = [a_y \ \Delta F_{z,f} \ \Delta F_{z,r} \ u_r]^T$, where the lateral acceleration can be calculated in the following way:

$$z_r = C_{1r}(v)x_r + D_{11r}(v)\delta_f + D_{12r}(v)u_r \tag{25}$$

In the suspension problem the performance outputs for control design are passenger comfort (i.e. heave acceleration), the suspension deflections and the control inputs: $z_s = [a_z \ z_{sf} \ z_{sr} \ u_s]^T$.

$$z_s = C_{1s}x_s + D_{11s}w + D_{12s}u_s \tag{26}$$

The measured outputs are the lateral acceleration of the sprung mass, the derivative of the roll angle and the suspension deflections at the suspension components.

Although the performance signals are formalized to the full-car model, the design is performed in two subsystems instead of the full-car model. One of them is the yaw-roll model for rollover prevention, which is controlled by u_r and ΔF_b. The second is the half-car model for suspension design, which is controlled by u_s. The two subsystems with their control inputs are independent of each other, i.e. the two subsystems are controlled independently in normal cruising situation. When a fault occurs in the active suspension system, its role is assumed by the active brake. The reconfiguration of the control structure is solved by a weighting strategy, which is presented in this section.

The purpose of the weighting functions is to keep the lateral acceleration, the lateral load transfers, the have acceleration, the suspension deflection and the control inputs small over the desired frequency range. The weighting functions chosen for performance outputs can be considered as penalty functions: they are selected large in a frequency range where small signals are desired, and small where larger performance outputs can be tolerated.

$$W_{p,ay} = \phi_{ay}\frac{A_1(\frac{s}{T_a}+1)}{(\frac{s}{T_b}+1)} \tag{27}$$

$$W_{p,az} = \frac{A_2(\frac{s}{T_c}+1)}{(\frac{s}{T_d}+1)} \tag{28}$$

$$W_{p,zs} == \frac{A_3(\frac{s}{T_e}+1)}{(\frac{s}{T_f}+1)} \tag{29}$$

with time constants T_i and proportional coefficients A_i. The weighting functions for the control inputs guarantee the limitation of the control forces $W_{p,us}$ and $W_{p,ur}$.

The gain ϕ_{ay} in the weighting functions is selected as a function of parameter R_r in the following way:

$$\phi_{ay} = \begin{cases} 0 & \text{if} \quad |R_r| < R_1 \\ \dfrac{(|R_r| - R_1)}{(R_2 - R_1)} & \text{if} \quad R_1 \le |R_r| \le R_2 \\ 1 & \text{if} \quad |R_r| > R_2 \end{cases} \tag{30}$$

where R_1, R_2 are constants. The gain ϕ_{ay} is increased in order to minimize the lateral acceleration and prevent the rollover of the vehicle. As the gain ϕ_{ay} increases the lateral acceleration decreases, since the active brake influences the lateral acceleration directly. R_1 defines the critical status when the vehicle is in an emergency. Parameter R_2

shows how fast the control should focus on minimizing the lateral acceleration.In the lower range of R_r the gain ϕ_{ay} must be small, and in the upper range of R_r the gains must be large. Consequently, the weighting functions must be selected in such a way that they minimize the lateral load transfers in emergency situations. However in normal cruising situations the control do not focus on the lateral load transfers since the weight is small.

In the event of a fault the range of the operation of the brake system must be extended and the wheels are decelerated gradually rather than rapidly if the normalized load transfer has reached its critical value. A small value of R_1 corresponds to activating the brake system early and gradually, whereas a large value of R_1 corresponds to activating the brake system rapidly. Thus, the design parameter R_1 is chosen to be scheduled on fault information ρ_f.

$$R_1 = R_1 - \frac{\rho_f}{\alpha} \tag{31}$$

where ρ_f is the normalized value of the fault information and α is a constant factor (in our case it is chosen 10).

The uncertainties of the nominal model are represented by W_r and Δ_m. The uncertainties of the nominal model are represented by the weighting function W_r in such a way that in the low frequency domain the uncertainties are about 10% and in the upper frequency domain they are up to 100%. The input scaling weights W_δ and W_w normalize the disturbances to the maximum expected command. $W_{n,ay}$, $W_{n\dot{\Phi}}$, $W_{n,sf}$ and $W_{n,sr}$ take into account the sensor noises in the control design.

The solution of an LPV problem is based on the set of infinite dimensional LMIs being satisfied for all $\rho \in \mathcal{F}_\mathcal{P}$, thus it is a convex problem, Rough and Shamma [2000], Wu [2001]. In practice, this problem is set up by gridding the parameter space and solving the set of LMIs that hold on the subset of $\mathcal{F}_\mathcal{P}$, see Balas et al. [1997]. The number of grid points depends on the nonlinearity and the operation range of the system. The LPV control is constructed by the Parameter Dependent Lyapunov Functions (PDLF) in which the conservatism of the control design is reduced.

For the interconnection structure, $\mathcal{H}_\infty$ compensators are synthesized for several values of velocity in a range $v =$ [20kph, ..., 120kph]. The spacing of the grid points is based upon how well the $\mathcal{H}_\infty$ point designs perform for plants around the design point. The rear load transfer parameter space is grided as $R_r = [0, R_1, R_2, 1]$. The scheduling parameter ρ_f, which is the fault information provided by the FDI filter, can be taken from interval $\rho_f = [0, 1]$. The zero value of ρ_f corresponds to the non-faulty operation and the value 1 to the full hydraulic actuator failure. Hence the parameter ρ_f is grided by selecting 3 grid points.

5. DEMONSTRATION OF THE RECONFIGURABLE CONTROL

In the first example the operation of the FDI filter is illustrated. The measured signals are the accelerations of the sprung mass and the unsprung mass and the relative displacement between the two masses at the front and the right-hand side of the vehicle. The FDI filter calculates the current force by using an inversion method and the measured signals. The reconstructed force is illustrated by the solid line. The force is compared with the force required by the suspension system (dashed line) within a time interval, which is practically selected between 0.2..0.4 *sec* depending on the forward velocity. If there is a large deviation between the required and the actual forces, it must be decided whether this deviation is a consequence of a performance degradation or an actuator saturation. Using the backstepping method the control input is also reconstructed as it is shown by the solid line in Figure 3. Since the values of the actuator do not exceed its upper limit, an actuator fault resulting in performance degradation is detected.

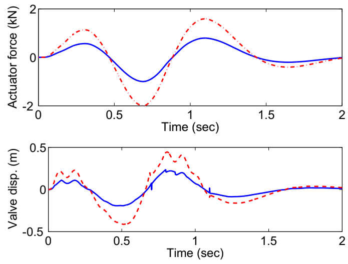

Fig. 3. The result of the FDI procedure

In the next example the operation of a conventional suspension system is compared with a reconfigurable suspension system. In the example a cornering maneuver with 70 *kph* velocity is presented. The cornering maneuver starts at the 1^{st} second and at the 4^{th} second a huge bump with 10 cm maximal value disturbs the motion of the vehicle. Figure 4 shows the suspension forces u_{fl} and u_{fr} (signals u_{rl} and u_{rr}). The suspension system operating in conventional manner generates suspension forces in order to reduce the effects of harmful vertical vibrations. Thus, it focuses on the huge bump which disturbs the motion at the 4^{th} second and so, the minimization of the heave acceleration as it is illustrated by the dashed line. In the reconfigurable case when the vehicle maneuver causes a critical value regarding rolling over, the suspension system generates moments to balance the overturning moments, thus the control force focuses only on reducing the normalized lateral load transfer and guaranteeing passenger comfort is no longer a priority (solid line). The purpose of reconfigurable active suspensions is to meet conventional performances in normal cruising and guarantee rollover prevention and improve safety in emergencies.

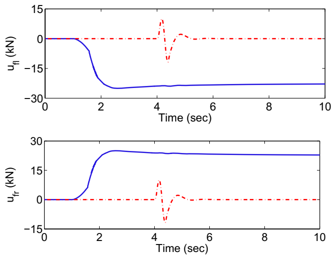

Fig. 4. The operation of the suspension system

In the third example the operation of the fault-tolerant integrated control is illustrated. The vehicle performs the same maneuver as in the previous example, however, it is assumed that an actuator failure has already been detected at the front and rear. The time responses of the steering angle, the normalized lateral load transfer at the rear, the braking force at the rear and the suspension force are presented in Figure 5. The solid line illustrates the fault operation and the dashed line illustrates the fault-free case. It is observed that the normalized load transfer increases due to the reduced power of the actuators. According to the detected actuator fault the brake is activated at a smaller value of the critical normalized load transfer. Moreover, the duration of the required brake force is longer in the case of a suspension fault. The figure also presents the suspension force at the front left-hand side.

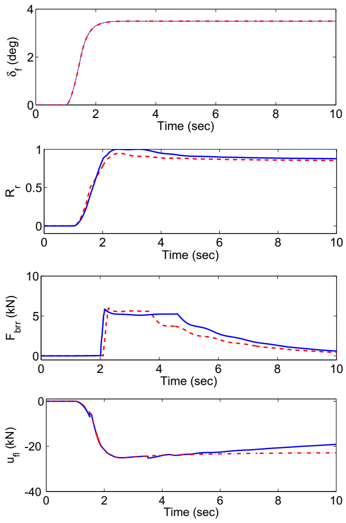

Fig. 5. The operation of the reconfigurable control system

6. CONCLUSIONS

In this paper a fault-tolerant reconfigurable controller which includes an active suspension system and an active brake has been proposed. With this structure both rollover prevention and passenger comfort can be guaranteed. Moreover, if a fault occurs in the active suspension system and it is detected by the FDI filter, the active brake assumes the role of the active suspension to enhance rollover prevention. A weighting strategy is applied in the closed-loop interconnection structure, in which the normalized lateral load transfers and the residual output of the FDI filter play an important role. This control mechanism guarantees the balance between rollover prevention and passenger comfort.

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