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LPV DESIGN OF FAULT–TOLERANT CONTROL FOR ROAD VEHICLES

PÉTER GÁSPÁR, ZOLTÁN SZABÓ, JÓZSEF BOKOR

Systems and Control Laboratory, Computer and Automation Research Institute
Hungarian Academy of Sciences, Kende u. 13–17, Budapest, H-1111, Hungary
e-mail: gaspar@sztaki.hu

The aim of the paper is to present a supervisory decentralized architecture for the design and development of reconfigurable and fault-tolerant control systems in road vehicles. The performance specifications are guaranteed by local controllers, while the coordination of these components is provided by a supervisor. Since the monitoring components and FDI filters provide the supervisor with information about the various vehicle maneuvers and the different fault operations, it is able to make decisions about necessary interventions into the vehicle motions and guarantee reconfigurable and fault-tolerant operation of the vehicle. The design of the proposed reconfigurable and fault-tolerant control is based on an LPV method that uses monitored scheduling variables during the operation of the vehicle.

Keywords: robust control, fault detection, LPV systems, faut-tolerant control, vehicle dynamics, vehicle control.

1. Introduction and motivation

Recently, there has been a growing demand for vehicles with ever better driving characteristics in which efficiency, safety and performance are ensured. In line with the requirements of the vehicle industry, several performance specifications are in the focus of research, e.g., improving road holding, passenger comfort, roll and pitch stability, guaranteeing the reliability of vehicle components, reducing fuel consumption and proposing fault-tolerant solutions (Gillespie, 1992). Integrated vehicle control methodologies are in the focus in research centers and automotive suppliers. The purpose of integrated vehicle control is to combine and supervise all controllable subsystems affecting vehicle dynamic responses. An integrated control system is designed in such a way that the effects of a control system on other vehicle functions are taken into consideration in the design process by selecting various performance specifications. Recently, several important papers have been presented on this topic (Yu *et al.*, 2008; Gordon *et al.*, 2003; Palkovics and Fries, 2001; Trachtler, 2004; Zin *et al.*, 2006).

This paper proposes a multi-layer supervisory architecture for integrated control systems in road vehicles. The supervisor has information about the various vehicle maneuvers and the different fault operations by monitoring components as well as fault-detection and identification (FDI) filters. For some examples of fault detection methods in industrial mechatronic products please refer to the works of, e.g., Muenchhof *et al.* (2009), Kanev and Verhaegen (2000), Fischer and Isermann (2004) or Theilliol *et al.* (2008).

The supervisor is able to make decisions about necessary interventions into the vehicle components and guarantee reconfigurable and fault-tolerant operation of the vehicle. In the proposed solution, local controllers are designed by taking into consideration the monitoring and fault signals received from the supervisor. The local control components are designed by Linear Parameter Varying (LPV) methods. These are well elaborated and successfully applied to various industrial problems. It is particularly appealing that nonlinear plants are treated as linear systems with *a priori* not necessarily known but online measurable, time-varying parameters. Hence, it allows linear-like control techniques to be applied to nonlinear systems.

The structure of the paper is as follows. In Section 2, the architecture of the integrated control is presented. In Section 3, the control-oriented modeling is performed and the control problem is set. In Section 4, the FDI design and the reconfigurable and fault-tolerant control design based on an LPV method are presented. Both the FDI design and the operation of the fault-tolerant control system are illustrated through simulation examples in Section 5. Finally, Section 6 contains concluding remarks.

2. Architecture of integrated control

A large number of theoretical problems have occurred in the design of the architecture of integrated vehicle control. The difficulty in the classical approaches is that the control design leads to hybrid and switching methods with a large number of theoretical problems (e.g., Hencey and Alleyne, 2010; Lu and Filev, 2009). Moreover, global stability and performance are difficult to guarantee. In order to implement a safety feature, the operation of a local controller must be modified by a supervisory command. However, in this case, the sharing of the responsibility of the designers is often unclear.

One remedy for these difficulties is to design a centralized controller which is able to integrate various controllable subsystems affecting the vehicle. For example, in the design of the global chassis control system, the brake, the steering, the anti-roll bars and the suspension systems are integrated. In this method, a sophisticated high-complexity and control-oriented model is constructed, which is augmented with a large number of performance specifications. The designed controller calculates the actuator commands based on measured signals and monitoring components.

This centralized control structure has several advantages: the designed controller guarantees performance specifications and robustness against uncertainties; the solution reduces the number of necessary sensors; it improves the flexibility of the actuators and avoids unnecessary duplications. This high-complexity control problem, however, is often difficult to handle, i.e., the more complex the vehicle model, the more numerical problems might occur. Moreover, this centralized approach is not suitable for the partial design tasks carried out by vehicle component suppliers. Furthermore, if a new component is added to the system, the entire system must be redesigned.

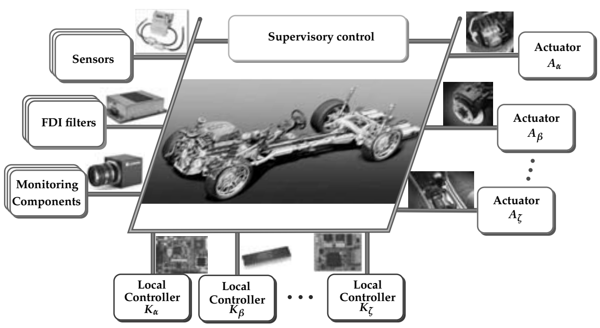

Fig. 1. Multi-layer supervisory architecture of integrated vehicle control.

The solution to integrated control proposed by this paper is to design a high-level controller which is able to supervise the effects of individual control components on vehicle dynamics, see the illustration in Fig. 1. In this supervisory decentralized control structure, there are logical relationships between the supervisor and the individually designed controllers. The communication between control components is performed by using the CAN bus. The advantage of this solution is that the components with their sensors and actuators can be designed by the suppliers independently. The role of the supervisor is to meet performance specifications and avoid interference and conflict between components. The supervisor performs the coordination of local controllers based on signals of the monitoring components and the FDI filters. It determines the various vehicle maneuvers and the different fault operations. The communication between the supervisor and the local components is realized through a well-defined interface, i.e., a set of signals with specific semantics. Once the interface is fixed after a conceptual design of the supervisor, the controllers of the individual local components can be designed separately.

The advantage of the proposed architecture for integrated vehicle control is that the complexity of the vehicle model is divided into several parts. In the formalism of the control-oriented model, the messages of the supervisor must be taken into consideration. Consequently, the signals of monitoring components and FDI filters are built in the performance specifications of the controller by using parameter-dependent weighting. In this way, the operation of a local controller can be extended to reconfigurable and fault-tolerant functions.

3. Actuator controller design

In this section, a motivation example for an integrated vehicle control is presented. The objective of the control design is to track a predefined path, guarantee road holding and increase yaw, roll and pitch stability. Four control components are applied in the system: the active brake, steering, anti-roll bars and the suspension system. In general, chassis control integrates the active steering and the active brake. A possible solution to the tracking problem uses active steering. When a rollover is imminent and this emergency persists, the brake system is activated to reduce the rollover risk. When the brake is used, however, the real path significantly deviates from the desired path due to the brake moment, which affects the yaw motion. This deviation must be compensated for by the active steering system. It is a difficult problem to perform tracking and rollover prevention at the same time since these tasks are in conflict with each other. In the paper, chassis control also includes two additional components. Active anti-roll bars are used to improve roll stability. Road holding and passenger comfort are improved by applying an active suspension system.

3.1. LPV models for local controllers. The local controllers are designed based on vehicle models with different complexity. In what follows, the dynamic models that correspond to the four control components are presented.

The lateral (yaw and roll) dynamics of the vehicle, which is modelled by a three-body system with a sprung mass m_s and two unsprung masses at the front m_{uf} and the rear m_{ur} including the wheels and axles, are illustrated in Fig. 2. I_{xx}, I_{xz}, I_{zz} are the roll moment of the inertia of the sprung mass, the yaw-roll product, and the yaw moment of inertia, respectively. The signals are 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, and v is the forward velocity. The lateral tire forces in the direction of the wheel ground contact are denoted by F_{yf} and F_{yr}. The roll 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 $k_{t,f}$, $k_{t,r}$. Here h is the height of CG of the sprung mass and h_{uf}, h_{ur} are the heights of CG of the unsprung masses, ℓ_w is the half of the vehicle width and r is the height of the roll axis from the ground.

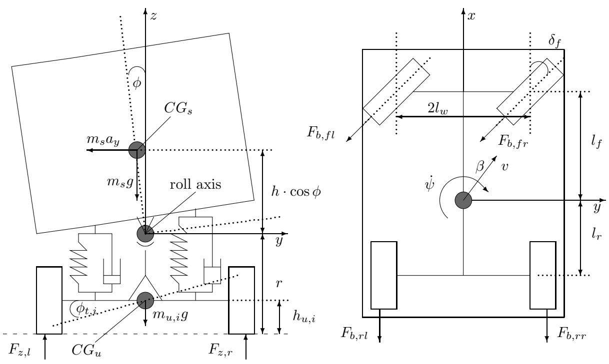

Fig. 2. Vehicle model with lateral dynamics.

In vehicle modelling, the motion differential equations are formalized. They are the lateral dynamics, the yaw moment, the roll moment of the sprung mass and the roll moment of the front and the rear unsprung masses. The equations are the following:

$$mv(\dot{\beta} + \dot{\psi}) - m_s h \ddot{\phi} = F_{yf} + F_{yr}, \tag{1}$$

$$-I_{xz}\ddot{\phi} + I_{zz}\ddot{\psi} = F_{yf} l_f - F_{yr} l_r + l_w \Delta F_b, \tag{2}$$

$$\begin{aligned}(I_{xx} + m_s h^2)\ddot{\phi} - I_{xz}\ddot{\psi} &= m_s g h \phi + m_s v h(\dot{\beta} + \dot{\psi}) \\ &\quad - k_f(\phi - \phi_{tf}) - b_f(\dot{\phi} - \dot{\phi}_{tf}) + u_{af} \\ &\quad - k_r(\phi - \phi_{tr}) - b_r(\dot{\phi} - \dot{\phi}_{tr}) + u_{ar}\end{aligned} \tag{3}$$

$$\begin{aligned}-rF_{yf} &= m_{uf} v(r - h_{uf})(\dot{\beta} + \dot{\psi}) + m_{uf} g h_{uf} \phi_{tf} \\ &\quad - k_{tf}\phi_{tf} + k_f(\phi - \phi_{tf}) \\ &\quad + b_f(\dot{\phi} - \dot{\phi}_{tf}) + u_{af},\end{aligned} \tag{4}$$

$$\begin{aligned}-rF_{yr} &= m_{ur} v(r - h_{ur})(\dot{\beta} + \dot{\psi}) - m_{ur} g h_{ur} \phi_{tr} \\ &\quad - k_{tr}\phi_{tr} + k_r(\phi - \phi_{tr}) \\ &\quad + b_r(\dot{\phi} - \dot{\phi}_{tr}) + u_{ar}.\end{aligned} \tag{5}$$

The lateral tire forces F_{yf} and F_{yr} are approximated linearly to the tire slide slip angles α_f and α_r, respectively: $F_{yf} = \mu C_f \alpha_f$ and $F_{yr} = \mu C_r \alpha_r$, where μ is the side force coefficient and C_f and C_r are tire side slip constants. In stable driving conditions, the tire side slip angle α_i can be simplified by substituting $\sin x \approx x$ and $\cos x \approx 1$. The classic equations for the tire side slip angles are then given as $\alpha_f = -\beta + \delta_f - l_f \cdot \dot{\psi}/v$ and $\alpha_r = -\beta + l_r \cdot \dot{\psi}/v$.

This structure includes several control mechanisms, which generate control inputs. They are the roll moments between the sprung and unsprung masses generated by the active anti-roll bars, u_{af} and u_{ar}, the difference in brake forces between the left and right-hand sides of the vehicle ΔF_b, and the steering angle δ_f.

The state space representation of the lateral dynamics, which is used in the control design of anti-roll bars, is formalized as follows:

$$\dot{x}_r = A_r(\rho) x_r + B_r(\rho) u_r, \tag{6}$$

where the state vector is $x_r = [\beta, \dot{\psi}, \phi, \dot{\phi}, \phi_{tf}, \phi_{tr}]^T$. The control inputs are the roll moments at the front and the rear between the sprung and unsprung masses: $u_r = [u_{af}, u_{ar}]^T$. The scheduling vector is $\rho = [v, \mu]^T$. Further details on the model structure can be found in the work of Gáspár *et al.* (2003b).

In the state equation, the matrix A_r depends on the forward velocity of the vehicle v nonlinearly. As a modelling assumption, the forward velocity is approximately equivalent to the velocity in the longitudinal direction while the slide slip angle is small. It is also assumed that the forward velocity is available, i.e., it is estimated online by using the on-board sensors (Song *et al.*, 2002). The adhesion coefficient of the vehicle μ depends on the type of road surface. There are several factors that can affect the value of the adhesion coefficient, which is a nonlinear and time varying function. Several methods have been proposed for the estimation of the adhesion coefficient and the side slip angle (de Wit *et al.*, 2003). Since the model contains a time-varying adhesion coefficient, an adaptive observer-based grey-box identification method has been proposed for its estimation (Gáspár *et al.*, 2010). Thus, the nonlinear model is transformed into an LPV one in which nonlinear terms are hidden with a suitably defined scheduling vector ρ. In practice, the components of the scheduling vector are measured or estimated.

In the design of the brake system, the same state space vector is used. The control input is the brake moment, which is able to generate unilateral brake forces at the front and the rear wheels at either of the two sides $u_b = \Delta F_b$,

$$\dot{x}_r = A_b(\rho)x_r + B_b(\rho)u_b. \tag{7}$$

In the design of the steering system The control input is the steering angle: $u_d = \delta_f$,

$$\dot{x}_r = A_r(\rho)x_r + B_d(\rho)u_d. \tag{8}$$

The vertical dynamics of the vehicle are modeled by a five-body system with a sprung mass m_s, and four unsprung masses at the front and the rear on the left and right hand sides. The state space representation of vertical dynamics, which is used for the control design of the suspension system, is formalized as follows:

$$\dot{x}_s = A_s(\rho_s)x_s + B_s(\rho_s)u_s, \tag{9}$$

where $x_s = [x_1, \theta, \phi, x_{2ij}, \dot{x}_1, \dot{\theta}, \dot{\phi}, \dot{x}_{2ij}]^T$ is the state vector which includes the vertical displacement of the sprung mass, the pitch angle and the roll angle of the sprung mass, the front and rear displacements of the unsprung masses on the left and right hand sides and their derivatives. The control inputs are generated by the suspension actuators: $u_s = \left[f_{fl}, f_{fr}, f_{rl}, f_{rr}\right]^T$. The state space representation of an active suspension system is found in the work of Gáspár *et al.* (2003a).

Throughout modelling, the nonlinear characteristics in the suspension spring and damper components are taken into consideration. The relative displacement δ_x and the relative velocity $\dot{\delta}_x$ between the sprung mass and the unsprung mass are assumed to be available. Thus the scheduling vector is chosen in the following form: $\rho_s = [\delta_x, \dot{\delta}_x]^T$.

3.2. Control design based on the LPV method. The closed-loop system applied in the design of integrated control includes the feedback structure of the model $G(\rho)$, the compensator, and elements associated with the uncertainty models and performance objectives. These elements define the parameter dependent augmented plant $P(\varrho)$. Using the controller K the closed-loop system $M(\varrho)$ is given by an LFT structure (see Fig. 3). The quadratic LPV performance problem is to choose the parameter-varying controller $K(\varrho)$ in such a way that the resultant closed-loop system $M(\varrho)$ is quadratically stable and the induced $\mathcal{L}_2$ norm from w to z is less than γ, i.e.,

$$\|M(\rho)\|_\infty = \inf_K \sup_{\|\Delta\|_2 \le 1} \sup_{\varrho} \sup_{\|w\|_2 \neq 0} \frac{\|z\|_2}{\|w\|_2} < \gamma. \tag{10}$$

By assuming an unstructured uncertainty and by applying a weighted small gain approach, the existence of a controller that solves the quadratic LPV γ-performance problem can be expressed as the feasibility of a set of LMIs, which can be solved numerically (see Packard and Balas, 1997; Wu *et al.*, 1996; Bokor and Balas, 2005). For the general case, see the works of Scherer (2001) and Wu (2001).

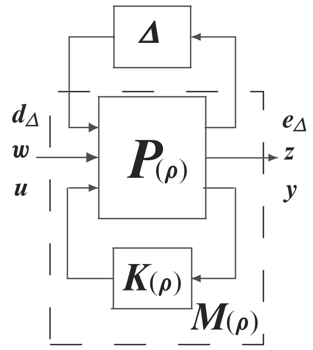

Fig. 3. General closed-loop interconnection structure.

In this framework, performance requirements are imposed by a suitable choice of the weighting functions W_p. The proposed approach realizes the reconfiguration of the performance objectives by suitable scheduling of these weighting functions. In what follows, this strategy is detailed.

The local components also include units for monitoring vehicle operations and FDI filters, and they are able to detect emergency vehicle operations, various fault operations or performance degradations in controllers. They also send messages to the supervisor. In the reconfigurable and fault-tolerant control of the local controller, several signals must be monitored and scheduling variables are added to the scheduling vector in order to improve the safety of the vehicle.

Below several examples of monitored components are presented.

Yaw stability is achieved by limiting the effects of the lateral load transfers. The purpose of the control design is to minimize the lateral acceleration, which is monitored by a performance signal: $z_a = a_y$. Unilateral braking is one of the solutions in which brake forces are generated in order to achieve a stabilizing yaw moment. In the second solution, an additional steering angle is generated in order to reduce the effect of the lateral loads. These solutions, however, require active driver intervention into the motion of the vehicle to keep the vehicle on the road.

Yaw tracking. Another control task is to follow a road by using a predefined yaw rate (angle). In this case, the current yaw rate must be monitored and the difference between the reference and the current yaw rate is calculated. The purpose of the control is to minimize the tracking error: $z_{\dot{\psi}} = \dot{\psi}_{cmd} - \dot{\psi}_{\mathrm{ref}}$.

Roll stability is achieved by limiting the lateral load transfers on both axles to below the levels for wheel lift-off during various vehicle maneuvers. The lateral load transfer is $\Delta F_{zy} = k_t \phi_t$, where ϕ_t is the monitored roll angle of the unsprung mass. The normalized lateral load transfer is introduced: $\rho_R = \Delta F_{zy}/mg$. The aim of the control design is to reduce the maximum value of the normalized lateral load transfer if it exceeds a predefined critical value.

Pitch stability is achieved by limiting the longitudinal load transfers to below a predefined level during sudden and hard braking. The normalized longitudinal load transfer is the normalized value of the pitch angle: $\rho_P = \theta/\theta_{\max}$, where θ is the monitored pitch angle and $\theta_{\max}$ is the maximal value of the pitch angle. The aim of the control design during braking is to reduce the pitching dynamics if the normalized longitudinal load transfer exceeds a critical value.

Road holding is achieved by reducing the normalized suspension deflections ρ_k between the sprung and unsprung masses at the four corner points of the vehicle. Since increasing road holding reduces the passenger comfort in the design of the suspension system its desired level is subject of a design decision.

4. Design of fault-tolerant LPV control

The fault-tolerant local controllers also require components for monitoring fault information. Faults in the operation of an actuator can be usually detected by using a built-in self-diagnostic method. In this case, fault information is sent by the actuator itself to the supervisor. When fault information is not achievable, the operation of an actuator must be monitored. In this case, an FDI component must be constructed which is able to detect fault or even performance degradation in the actuator. Fast and reliable operation of the actuator is critical. Since fault-tolerant control requires fault information in order to guarantee performances and modify its operation, an FDI filter is also designed.

4.1. Design of an FDI component. Any reconfiguration scheme relies on a suitable FDI component. There are a lot of approaches to design a detection filter (Chen and Patton, 1999). The LPV setting, however, narrows the available tools.

In contrast to the LTI case, in the LPV framework stability cannot be guaranteed in algebraic terms, e.g., by requiring that the "frozen" LTI systems be stable. Besides the technical difficulties of the potential design process, this fact implies that algebraic methods of the classical LTI FDI filter design (Gertler, 1998; Varga, 2008) are not suitable for the LPV setting. In the LPV framework, the only practical solution is to require quadratic stability, which can be cast as a set of Linear Matrix Inequality (LMI) feasibility problems (Rodrigues *et al.*, 2007). The so-called geometric approach of the FDI meets these requirements and often leads to successful detection filter design (for details, see Balas *et al.*, 2003; Bokor and Balas, 2004; Edelmayer *et al.*, 2004; Shumsky and Zhirabok, 2006).

As a high level approach, the FDI filter design problem can often be cast in the model matching framework depicted in Fig. 4 (Rank and Niemann, 1999).

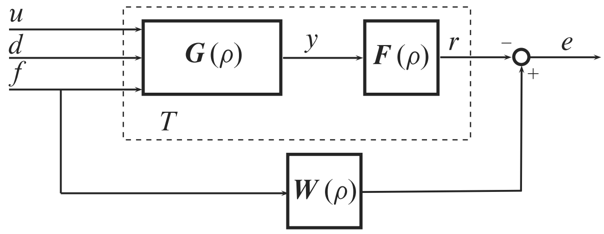

Fig. 4. Norm based detector design problem.

The LPV paradigm permits to cast a nonlinear system as a Linear Time-Varying (LTV) one, i.e., the residual can be expressed as

$$r = T_{ru}u + T_{rd}d + T_{rf}f. \tag{11}$$

Hence, to achieve robustness in the presence of disturbances and uncertainty, multiobjective optimization-based FDI schemes can be proposed where an appropriately selected performance index has to be chosen to enhance sensitivity to the faults and simultaneously attenuate disturbances: the robust disturbance rejection condition formulated as

$$||T_{rd}||_\infty = \sup_{||f||_2=1, \rho\in\mathcal{P}} ||r||_2 \tag{12}$$

is to be minimized.

This is a usual worst-case filtering problem and the corresponding design criteria can be formulated as a convex optimization problem by using LMIs. The main problem here is that the sensitivity and robustness conditions are conflicting. In the LTI framework, this means that both sensitivity to faults and insensitivity to unknown inputs cannot be achieved at the same frequencies. Faults having similar frequency characteristics as those of disturbances might go undetected. While the design problem is nonconvex, in general, Henry and Zolghadri (2004) propose a scheme that can handle it by using LMI techniques.

For illustrative purposes, the LPV detection filter design of the active anti-roll bars is sketched. The specific structure that fits the norm based approach, containing the weighted open-loop system, which includes the yaw-roll model $G(\rho)$ and the parameter-dependent FDI filter $F(\rho)$, and elements associated with performance objectives, is depicted in Fig. 5. In the diagram, u is the control torque, which is generated by anti-roll bars, y is the measured output, which contains the lateral acceleration a_y and the roll rate $\dot{\phi}$. The FDI filter takes the measured outputs and the control torques. The control torque is provided to the FDI filter so that the effect of the control input is attenuated on residual outputs. In the figure, f_a is the actuator fault and f_s is the sensor fault. Here e_a and e_s represent the weighted fault estimation errors associated with failures.

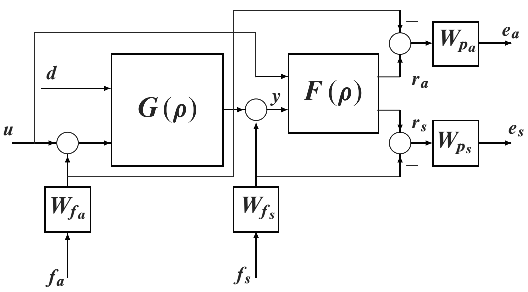

Fig. 5. Open loop interconnection structure for FDI filter design.

W_{p_a} and W_{p_s} are the fault detection performance weights, which reflects the relative importance of the different frequency domains. These weighting functions can be considered penalty functions, i.e., the weight should be large in the frequency range in which small errors are desired (low frequency) to achieve the integral action for fault estimation and small in the range where larger errors can be tolerated. The weight W_{f_a} represents the size of a possible fault in the actuator channel. The weight W_{f_s} takes sensor failure into consideration in the FDI filter design. The augmented plant of the filtering problem has $w = \begin{bmatrix}\delta_f & F_b & f_a & f_s & u\end{bmatrix}^T$ as the disturbance input and $e = \begin{bmatrix}e_a & e_s\end{bmatrix}^T$ as the performance output, which are used to evaluate the estimation quality. The design requirement for $\mathcal{H}_\infty$ residual generation is to maximize the effect of the fault on the residual and simultaneously minimize the effect of exogenous signals (δ_f, F_b, u) on the residual. Further details on the design can be found in the work of Grenaille *et al.* (2008).

The fault information provided by a fault detection filter triggers the control reconfiguration. At the level of local control design, the reconfiguration is achieved by scheduling the performance weights by a signal ρ_D related to the fault information and provided by a fault decision block. As a simple example, one might consider $\rho_D = f_{\text{act}}/f_{\max}$, where f_{act} is an estimation of the failure (output of the FDI filter) and $f_{\max}$ is an estimation of the maximum value of the potential failure (fatal error). The value of a possible fault is normalized into the interval $\rho_D = [0, 1]$. The estimated value f_{act} represents the rate of the performance degradation of active components.

4.2. Weighting strategy for reconfiguration. In what follows, the choice of the performance weights for the problems listed in Section 3.2 are detailed.

In order to solve the yaw rate tracking problem in the design of the steering system, the command signal must be fed forward to the controller ($\dot{\psi}_{\text{cmd}}$). The command signal is a pre-defined yaw rate signal and the performance signal is the tracking error $z_{\dot{\psi}} = e_{\dot{\psi}}$, which is the difference between the actual yaw rate and the yaw rate command. The weighting function for the tracking error is selected as

$$W_{pe} = \frac{1}{\epsilon}\frac{(T_{d1}s+1)}{(T_{d2}s+1)}, \tag{13}$$

where T_{di} are time constants. Here, it is required that the steady state value of the tracking error should be below an acceptable limit ϵ. In the design of the brake system, the command signal is the difference in brake forces while the performance signal is the lateral acceleration: $z_b = \begin{bmatrix}a_y, u_r\end{bmatrix}^T$. The weighting function of the lateral acceleration is selected as

$$W_{pa} = \Gamma_a(\rho_R)\frac{(T_{b1}s+1)}{(T_{b2}s+1)}. \tag{14}$$

Here Γ_a is a gain, which reflects the relative importance of the lateral acceleration and it is chosen to be parameter-dependent, i.e., the function of ρ_R. When ρ_R is small, i.e., when the vehicle is not in an emergency, Γ_a is small, indicating that the LPV control should not focus on minimizing the acceleration. On the other hand, when ρ_R approaches the critical value, Γ_a is large, indicating that the control should focus on preventing the rollover. As the gain Γ_a increases, the lateral acceleration decreases, since the active brake affects the lateral acceleration directly. In the control design, the parameter dependence of the gain is selected as follows:

$$\Gamma_a(\rho_R) = \frac{1}{R_a - R_b}(|\rho_R| - R_b). \tag{15}$$

Here R_b defines the critical status when the vehicle is close to the rollover situation, and R_a shows how fast the control should focus on minimizing the lateral acceleration. If a fault is detected in the suspension system or in the anti-roll bars, concerning roll stability their role is substituted by the brake system. In this case, the brake system is activated at a smaller critical value than in a fault-free case. Consequently, the brake is activated in a modified way and the brake moment is able to assume the role of the anti-roll bars or the suspension actuator in which the fault has occurred.

The modified critical values are

$$R_{a,\text{new}} = R_a - \alpha \cdot \rho_D, \tag{16}$$

$$R_{b,\text{new}} = R_b - \alpha \cdot \rho_D, \tag{17}$$

where α is a predefined constant factor. When there is performance degradation in the operation of the brake system, it is not able to create a sufficient yaw moment to improve roll stability. In this sense, the brake system is substituted by the steering system. The steering system receives the fault message from the supervisor and it modifies its operation in such a way that the effects of the lateral loads are also reduces. The difficulty in this solution is that performance degradation also occurs concerning the tracking task, since the steering system must create a balance between the tracking and the roll stability. The performance signals used in the suspension design are $z_s = \begin{bmatrix} a_z & s_d & t_d & u_s \end{bmatrix}^T$.

The goals of the suspension system are to keep the heave accelerations $a_z = \ddot{q}$, suspension deflections $s_d = x_{1ij} - x_{2ij}$, wheel travels $t_d = x_{2ij} - w_{ij}$, and control inputs small over the desired operation range. The performance weighting functions for heave acceleration, suspension deflections and pitch angles are selected as

$$W_{p,az} = \Gamma_{az}(\rho_k) \frac{T_{s1}s + 1}{T_{s2}s + 1}, \tag{18}$$

$$W_{p,sd} = \Gamma_{sd}(\rho_k) \frac{T_{s3}s + 1}{T_{s4}s + 1}, \tag{19}$$

$$W_{p,\theta} = \Gamma_{\theta}(\rho_P) \frac{T_{s5}s + 1}{T_{s6}s + 1}, \tag{20}$$

where T_{si} are time constants. The trade-off between passenger comfort and road holding is due to the fact that it is not possible to guarantee them together simultaneously. A large gain Γ_{az} and a small gain Γ_{sd} correspond to a design that emphasizes passenger comfort, while choosing Γ_{az} small and Γ_{sd} large corresponds to a design that focuses on road holding. The design procedure for an active suspension system is found in the work of Gáspár *et al.* (2003a).

The idea of the reconfigurable suspension system is based on the fact that active suspension systems are used not only to eliminate the effects of road irregularities but also to generate roll moments to improve roll stability or generate a pitch moment to improve pitch stability. For a reconfigurable suspension system, the parameter-dependent gains are selected as functions of the suspension deflection ρ_k, the normalized lateral load transfer ρ_R and the normalized value of the pitch angle ρ_P. In normal cruising, i.e., when $|\rho_R| < R_s$, the suspension system focuses on the conventional performances. If ρ_k and ρ_P do not exceed their critical values, the controller must create a balance between passenger comfort and road holding. If ρ_P exceeds a predefined critical value, the controller must focus on pitch moments. If ρ_k exceeds a critical value, the controller must focus on suspension deflection. In an emergency, however, i.e., when $|\rho_R| \geq R_s$, the suspension system must reduce the rollover risk, and guaranteeing passenger comfort (and pitch dynamics) is no longer a priority.

The weights (13),(14),(18)–(20) used in the paper are Proportional-Derivative (PD) components. Their time constants and gains reflect the required steady state and transient behavior of the different signals that describe the performance specifications.

5. Simulation examples

In the first simulation example, the FDI filter is tested during a double lane change maneuver. This maneuver is used to avoid an obstacle in an emergency. The maneuver has 2.5 m path deviation over 100 m. The size of the path deviation is chosen to test a real obstacle avoidance in an emergency on a road. The vehicle velocity is 70 km/h. The steering angle input is generated according to a human driver. The braking inputs are used to decelerate the vehicle during the maneuver. The velocity of the vehicle is changing from 70 km/h to approximately 64 km/h when applying braking forces. The fault scenarios used in the closed loop simulations are a 10 kNm anti-roll bar fault starting from 2 sec and a 0.1 rad/sec sensor fault occurring at the 5-th second. The step sensor fault means that the $\dot{\phi}$ sensor measures a signal with a constant additive fault. In our case, the step failure demonstrates a loss in effectiveness in the actuator. The time responses during the maneuver are illustrated in Fig. 6.

The weighting strategy of the FDI filter is formalized in the following way. The fault detection performance weight is selected in such a way that in the low frequency domain the fault estimation error should be rejected by a factor of 2 for both types of failure. Moreover, the weights for fault channels are selected in such a way that the size of a possible fault is 10 kNm in the actuator channel and there is a 0.1 rad/sec sensor failure.

As the lateral acceleration increases during the maneuver, the normalized load transfer also increases. The control torque between -40 kNm and 80 kNm at the front and rear axles. The fault at the actuator occurs at the 2 nd sec at the first actuator. The additive fault occurs at the measured yaw rate at the 5-th second. The first residual shows the actuator fault and the second one the sensor fault. The effects of the two failures are decoupled and the residuals give an acceptable estimation of faults in the anti-roll bars and the sensor fault, the time of their occurrence and their values, see Figs. 6(d) and (h).

In the second simulation example, the supervisory controller (solid) is compared with the conventional distributed controller without reconfiguration (dashed). The heavy vehicle performs a sharp double lane change manoeuver, which is defined by the signal yaw-rate. The

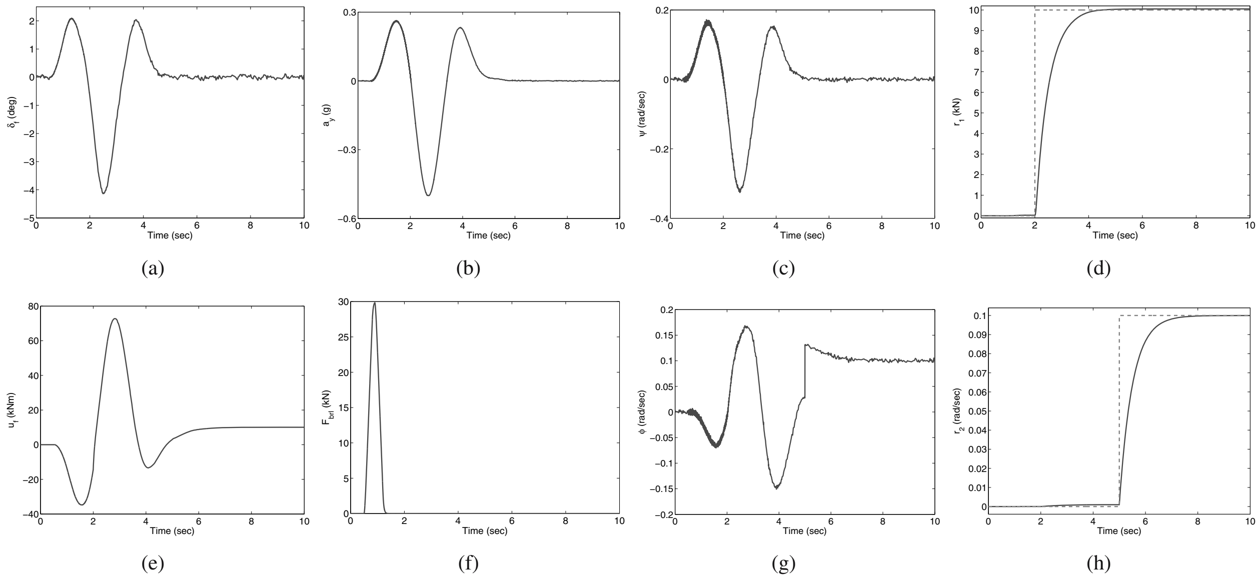

Fig. 6. Time responses to double lane change maneuver in the operation of the FDI filter: steering angle (a), lateral acceleration (b), yaw rate (c), residual with an actuator fault (d), roll moment at the front (e), brake force at the left (f), roll rate (g), residual with a sensor fault (h).

maneuver has a 4 m path deviation over 100 m. The velocity of the vehicle is 120 km/h. The example presents the integration of several components in order to follow a predefined path and guarantee roll and yaw stability simultaneously. Figure 7 shows the time responses of the controlled system.

In the example, the fault-tolerant control system in which there is a float failure in the active anti-roll bar at the front is examined, (see Fig. 7(c)). During the faulty operation, this component is not able to generate a stabilizing moment to balance the overturning moment. The supervisory control system uses fault information from an FDI filter, which monitors the operation of the active anti-roll bars.

The control design is based on the LPV method since it is able to handle the parameter dependence in the weighting strategy and guarantee that the designed controller meets the performance specifications. The integration is carried out through the parameter-dependent weighting function used in the design of the brake. The brake activates and generates a yaw moment in order to reduce the influence of the lateral loads, (see Figs. 7(d) and (h). When there is a failure in the front anti-roll bar, the braking lasts longer and the forces are greater than in the fault free case. Consequently, a greater steering angle is generated by the supervisory scheme to follow the predefined path and guarantee yaw/roll stability, see Fig. 7(a). The brake forces required by the control system are of impulsive nature for the conventional case while for the supervisory controller braking lasts longer. The deviation of the current yaw rate from the reference yaw rate is greater in the conventional case than in the supervisory case, (see Fig. 7(f)).

6. Conclusions

In the paper, a multi-layer supervisory architecture for the design and development of integrated vehicle control systems has been proposed. The local controllers are designed independently by taking into consideration the monitoring and fault signals by using LPV methods. In this architecture, the supervisor is able to make decisions about necessary interventions and guarantee reconfigurable and fault-tolerant operation of the vehicle.

As an illustration, integrated control is proposed for tracking the path of the vehicle, guaranteeing road holding, and improving pitch and roll stability. In a cruising mode the centralized solution when all the control components are designed together usually results in more balanced control actions. However, examples show that the distributed control system is less sensitive to fault operations than the centralized control.

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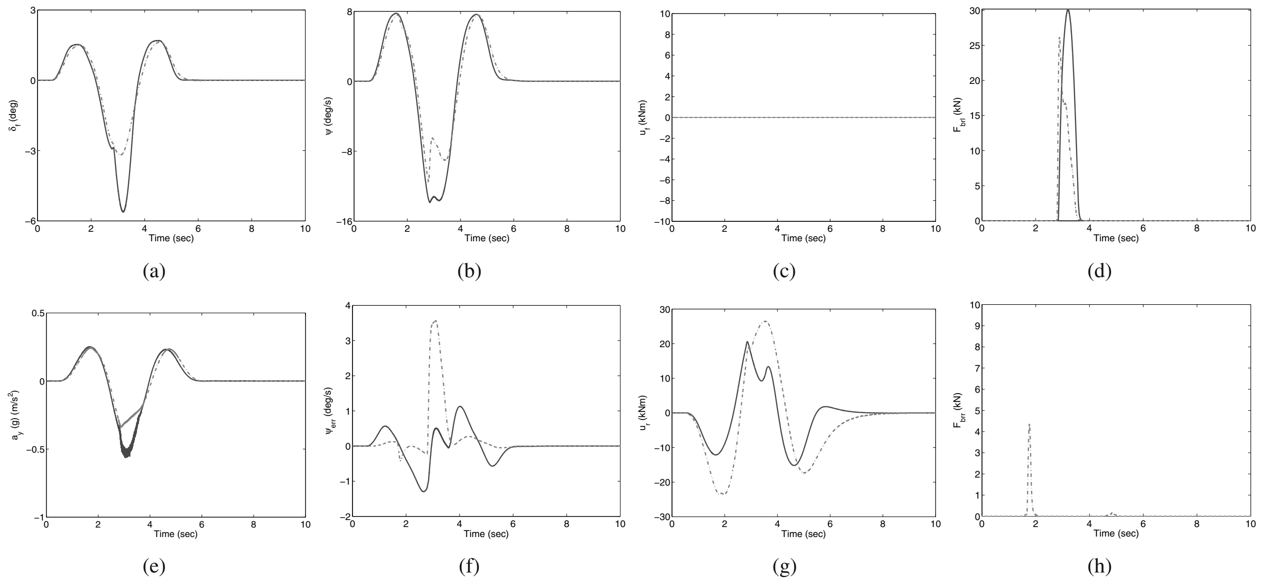

Fig. 7. Time responses of the fault-tolerant control in a fault in the operation of active anti-roll bars (solid—supervisory, dashed—conventional): steering angle (a), yaw rate (b), roll moment at the front (c), brake force at the left (d), lateral acceleration (e), tracking error (f), roll moment at the rear (g), brake force at the right (h).

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Péter Gáspár received both the M.Sc. and Ph.D. degrees from the Faculty of Mechanical Engineering, Budapest University of Technology and Economics, in 1985 and 1997, respectively, and the D.Sc. degree in control from the Hungarian Academy of Sciences (HAS) in 2007. Since 1990 he has been a research advisor in the Systems and Control Laboratory of the Computer and Automation Research Institute, HAS. He is the head of the Vehicle Dynamics and Control Research Group. He is also a professor at the Control and Transport Automation Department, Budapest University of Technology and Economics. His research interests include linear and nonlinear systems, robust control, multi-objective control, system identification and identification for control. His industrial work has involved mechanical systems, vehicle structures and vehicle control.

Zoltán Szabó obtained his Ph.D. degree in applied mathematics from Eotvos Lorand University, Budapest, in 2003, and the D.Sc. degree in control from the Hungarian Academy of Sciences (HAS) in 2011. Since 1996 he has been with the Systems and Control Laboratory (SCL) of the Computer and Automation Institute, HAS. Currently he is a research advisor of the SCL. His research interests include multivariable systems and robust control, system identification, fault detection with applications in power systems safety operations and also in the control of mechanical and vehicle structures. He is presently involved in activities related to reconfigurable control and control of linear hybrid systems.

József Bokor received the Dr.Tech. and later the Ph.D. degree in automation from the Technical University of Budapest, and the D.Sc. degree from the Hungarian Academy of Sciences in 1990. He was elected a corresponding member and later a full member of the Hungarian Academy of Sciences in 1999 and 2002, respectively. Currently he is a research director of the Computer and Automation Research Institute, Hungarian Academy of Sciences, where he is also the head of the Systems and Control Laboratory. He is a professor and the head of the Control and Transportation Automation Department, Faculty of Transportation Engineering, Budapest University of Technology and Economics. His research interest includes multivariable systems and robust control, system identification, fault detection with applications in power systems safety operations, and also the control of road and aerial vehicles and mechanical structures.

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