6th IFAC Symposium Advances in Automotive Control
Munich, Germany, July 12-14, 2010

Reference model based adaptive control of a hybrid suspension system

Guido Koch *, **Sebastian Spirk** * and **Boris Lohmann** *

* *Institute of Automatic Control, Technische Universität München 85748 Garching, Germany (e-mail: {guido.koch, sebastian.spirk, lohmann}@tum.de).*

Abstract:
The paper presents a new adaptive controller structure for a hybrid quarter-vehicle suspension system containing a low bandwidth actuator and a semi-active damper. The optimal control force to ease the conflict between ride comfort, ride safety and limited suspension deflection is obtained from a reference model based on a passive suspension system with timevarying stiffness and damping. By allocating the resulting control force to the two actuators, the power demand of the hybrid suspension system is significantly lower compared to a high bandwidth active system although their performance is similar. Stability of the adaptive controller structure is guaranteed using a Lyapunov function approach. The hybrid suspension is compared with benchmark systems in simulations under realistic assumptions regarding nonlinearities of the suspension elements, the actuator and sensor architecture as well as the road profile.

Keywords: Active vehicle suspensions; Semi-active dampers; Vehicle dynamics; Vehicle suspension; Adaptive control.

1. INTRODUCTION

A vehicle suspension system should provide a maximum of ride comfort (characterized by low vertical chassis acceleration $\ddot{z}_c(t)$) but has to guarantee ride safety by keeping constraints on the dynamic wheel load $F_{dyn}(t)$ in order to ensure the transmission of longitudinal and lateral forces between tire and road. Passive suspensions can only moderately ease the conflict between these aims due to their fixed spring (c_c) and damper (d_c) configuration (see Figure 1, "rms" abbreviates root mean square). Due to rising customer demand for increased ride comfort, semi-active or low bandwidth active suspension systems are integrated in modern production vehicles (see e.g. [Pyper et al. 2003]). Fully active systems (see e.g. [Jones 2005]) include high bandwidth actuators (cutoff frequency > 20 Hz), their costs and high energy demand have however obviated their integration in production vehicles so far.

In [Koch et al. 2008b] it has been shown that the combination of a semi-active damper and a low bandwidth actuator integrated in series to the primary spring (the configuration is called *hybrid suspension* in this paper) can almost achieve the same performance as a high bandwidth active suspension system, especially if an adaptive controller is used. Adaptive controllers for suspension systems change the controller parametrization according to the driving state such that ride comfort is optimized while the safety limits on the dynamic wheel load and suspension travel are not violated [Lin and Kanellakopoulos 1997, Fialho and Balas 2002, Koch et al. 2008a].

The stability proof for adaptive or switching suspension control approaches [Fialho and Balas 2002, Koch et al. 2008a] is frequently based on the existence of a common

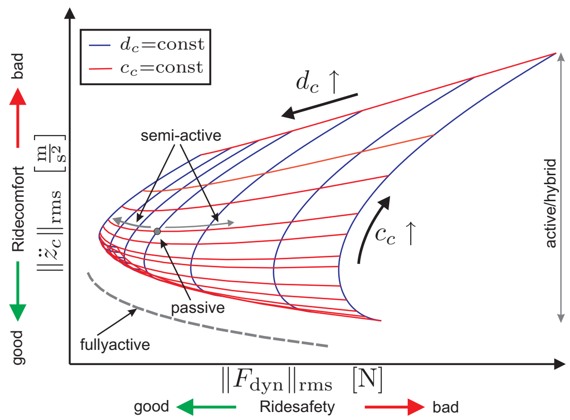

Fig. 1. Vehicle suspension conflict diagram (carpet plot)

Lyapunov function [Lin and Antsaklis 2009] for the resulting closed loop systems, which is calculated numerically. However, the existence of a solution for the involved linear matrix inequalities (LMIs) depends on the controller parametrizations and it is not guaranteed that a solution still exists if the controller tunings are changed or plant parameters vary. Therefore, a transparently parametrizable controller structure with analytically guaranteed stability of the adaptive system is desired in order to support the implementability and tuneability of the mechatronic suspension system. Moreover, economical aspects regarding the actuator and sensor architecture, computational effort and energy efficiency of the system have to be considered for suspension controller design.

This paper presents a new adaptive controller structure for the recently proposed hybrid suspension system. The considered hardware is already available in todays production vehicles (although not in the proposed combination). The controller structure uses a reference model in combination with an adaptation logic to calculate the optimal resulting natural frequency and damping ratio of the chassis mass according to the optimal passive suspension configuration for the current driving state. The resulting control force to simulate the corresponding suspension setting is allocated to the continuously variable semi-active damper and a hydraulic actuator. In order to evaluate the performance of the proposed concept, the hybrid suspension system is compared with benchmark systems including a semi-active and a high bandwidth active suspension system under realistic conditions (excitation signals, sensor configuration, nonlinearities of the suspension) in simulations.

The remainder of the paper is organized as follows: In Section 2 models of the suspension system and the actuators as well as the system requirements are presented. The control strategy is introduced in Section 3. In Section 4 simulation results are given and the paper is concluded in Section 5.

2. MODELING OF THE SUSPENSION SYSTEM

2.1 Vehicle model

Quarter-car models are used as vehicle models since the frequency range of interest for suspension control is below 25 Hz and only the vertical translatory movements of chassis and wheel mass are considered [Mitschke and Wallentowitz 2004]. Figure 2 shows the quarter-car models of the fully active, the semi-active and the hybrid suspension system.

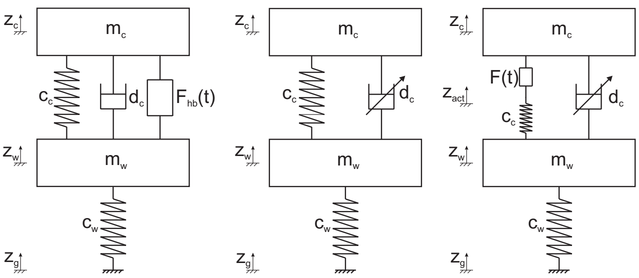

Fig. 2. Fully active, semi-active and hybrid suspensions

The model of the passive suspension results from the fully active suspension model if the control force vanishes ($F_{hb}(t) = 0$). The state-vector $\mathbf{x}$ and the output vector $\mathbf{y}$ are introduced as

$$\mathbf{x} = [\, z_c - z_w, \;\; \dot{z}_c, \;\; z_w - z_g, \;\; \dot{z}_w \,]^T , \tag{1}$$

$$\mathbf{y} = [\, \ddot{z}_c, \;\; F_{dyn}, \;\; z_c - z_w \,]^T , \tag{2}$$

where $F_{dyn} = c_w(z_g - z_w)$ denotes the dynamic wheel load. With the control input of the actuator $u(t) = F(t)$ and the semi-active damper $u_{cvd}(t) = F_d(t)$ as well as the disturbance input $u_d(t) = \dot{z}_g(t)$, the quarter-car model of the hybrid vehicle suspension system can be expressed as a fourth order state space model in the form [1]

$$\dot{\mathbf{x}}(t) = \mathbf{f}(\mathbf{x}(t), u(t), u_{cvd}(t), u_d(t)) , \tag{3}$$

$$\mathbf{y}(t) = \mathbf{g}(\mathbf{x}(t), u(t), u_{cvd}(t)) . \tag{4}$$

The nonlinearity of the model is introduced by the spring and damper characteristics depicted in Figure 3. The tire stiffness is considered to be linear in the frequency and amplitude range of interest and a constant friction force of $F_{df} = F_f \mathrm{sgn}(\dot{z}_c - \dot{z}_w)$ with $F_f = 50\,\mathrm{N}$ in the suspension strut is included in the model. A detailed description of the nonlinear quarter-car model with explicit equations is given e.g. in [Mitschke and Wallentowitz 2004]. It is assumed that the measured variables are $\mathbf{y}_m = [\, \ddot{z}_c, \;\; \ddot{z}_w \;\; z_c - z_w \,]^T$, which reflects a realistic sensor configuration of modern automobiles equipped with mechatronic suspension systems.

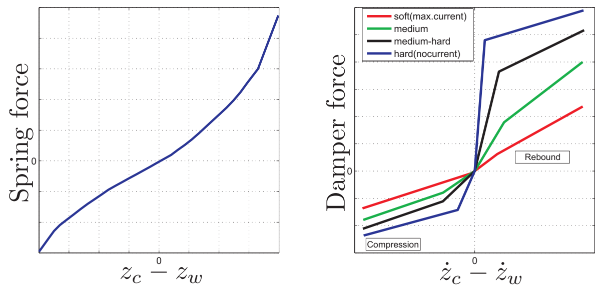

Fig. 3. Nonlinear spring and damper characteristic

2.2 Actuator models

Semi-active damper The measured force vs. velocity characteristics of the variable damper (*CVD*) are given in Figure 3. The damper's power electronic unit generates a driving current $i_d(t)$ resulting in a change of the velocity dependent damper force $F_d = d_c(t)(\dot{z}_c - \dot{z}_w)$ according to a nonlinear characteristic. The dynamic behavior of the hydraulic damper (the electromagnetical valve dynamics and the mechanical force generation) is approximated by

$$G_{el}(s) = \frac{1}{5 \cdot 10^{-3}s + 1} \quad \text{and} \quad G_m(s) = \frac{1}{9 \cdot 10^{-3}s + 1} . \tag{5}$$

The actual damper current $i_d(t)$ is a measurement signal provided by the power electronic unit. The damper model and its feedforward control (described in Section 3.4) for a desired damper force F_d^* is shown in Figure 4 with i_d^* denoting the desired damper current.

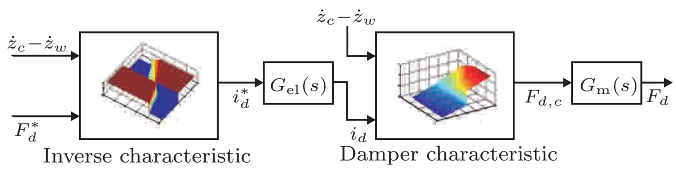

Fig. 4. Damper model and feedforward control

Actuators For the slow hydraulic actuator (*SAC*) a bandwidth of 5 Hz is assumed (see Pyper et al. [2003]), which is sufficient to exploit the potential of the hybrid suspension system [Koch et al. 2008b]. Assuming a time-delay of the actuator of 3 ms, which is approximated by a

[1] Due to the fact that the presented results are part of a research project in cooperation with an industry partner, the vehicle parameters, details of the semi-active damper and the scaling of the axes in Figure 3 have been omitted.