A model free control design approach for a semi-active suspension of a passenger car

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Abstract— **Comfort in passenger cars can be improved by including a semi-active suspension system, controlled by online varying the damping based on measurements of the car motions. The selection of an appropriate control structure is crucial, since it determines the complexity of the control design and parameter tuning process. This paper presents a flexible model-free control structure based on physical insights in the car and semi-active suspension dynamics which are used to linearize and decouple the system. A static decoupling is used to decouple the system into its modal motions heave, roll and pitch, which are then controlled by modal (diagonal) controllers, consisting of several feedback and feedforward modules, each taking care of a specific comfort or handling issue. The feedback controller is based on the Skyhook principle. Steering angle, throttle, and break pedal feedforward control roll during cornering, and pitch during acceleration and deceleration, respectively. All control parameters are physically interpretable and therefore can be easily tuned online according to guidelines given by test pilots.**

I. INTRODUCTION

Comfort and road handling performance of a passenger car are mainly determined by the damping characteristic of the shock absorbers. Passive shock absorbers have a fixed damping characteristic determined by their design. Depending on the road excitation, however, it is desirable to adjust this characteristic to increase performance. Semi-active and active suspension systems offer the possibility to vary the damper characteristics along with the road profile e.g. by changing the restriction of one or two current controlled valves or by changing the viscosity of a magneto rheological fluid. An active shock absorber has the additional advantages that negative damping can be provided and that a larger range of forces can be generated at low velocities, thereby potentially allowing an increase in system performance. Semi-active suspensions on the other hand are less complex, more reliable and commercially available. They do not require an external power source (e.g. hydraulic pump) and are more safe because they can only dissipate energy and therefore cannot render the system unstable.

Literature describes several linear and nonlinear techniques to control a car using an active or semi-active suspension. [1],[2] and [3] apply **linear** control strategies based on linear physical car models consisting of lumped masses, linear springs and dampers, and a shock absorber modelled as an ideal force source. Real car dynamics are much more complex and active shock absorbers are not ideal force sources but have a complex nonlinear dynamic behavior. These unrealistic assumptions make these linear control approaches less appropriate for practical applications.

Nonlinear control strategies such as linear parameter varying gain scheduling [4],[5], backstepping [6] and adaptive control [7],[8], have been applied to active suspension systems. These controllers are based on a nonlinear physical car and damper model which have a large number of parameters. The experimental identification of these model parameters is a complex problem. In addition, the design and tuning of the above mentioned nonlinear controller is not straightforward. Basically the use of nonlinear models and controllers lead to very time-consuming designs, since no standard techniques or software tools are available.

[9] present a practical, experimental approach using linear identification and robust control techniques on an active suspension of a quarter car test rig. A linear robustly performant controller was obtained using μ-synthesis based on an experimentally identified linear model of both the active suspension and the quarter car dynamics. The relatively simple construction of the test-rig and the linearity of the active suspension made it possible to apply linear identification and control design techniques. The dynamics of a real car are much more complex and a semi-active suspension behaves quite differently then an active suspension: it becomes uncontrollable when the rattle velocity [1] is zero. Because of these two reasons, the techniques developed in [9] are not extendable to this application.

While the above mentioned model based approaches may, in theory, yield optimal controllers for a certain shock-absorber and car model, their application to a full car and highly nonlinear semi-active shock-absorber is complex or even infeasible.

This paper presents a model free control structure that does not directly aim at optimality, but incorporates many physically interpretable parameters that can be easily tuned

[1] the rattle displacement/velocity is the relative displacement/velocity of the rod with respect to the cylinder

online according to guidelines given by test-pilots and based on test-results. This approach is based on physical principles of semi-active shock-absorbers and cars in general, but does not require a model of its dynamics. Therefore it is applicable to any semi-active or active suspension system and any type of car.

The semi-active suspension hardware, the available sensor signals, the car and the hydraulic shaker are described in Section II. Sections III and IV discuss the systems linearization and decoupling. The structure of the controller, consisting of several feedback and feedforward modules, is explained in Section V. Each of these modules tackles a particular comfort or handling issue. Section VI discusses the tuning of the parameters of the resulting control structure. Final conclusions are given in Section VII.

II. Hardware description

A. Semi-active shock-absorber

A classic passive shock absorber consists of a cylinder filled with oil and a rod connected to a piston, which contains a calibrated restriction called the piston valve (see figure 1 left). The change in volume caused by the rod moving in or out the cylinder is compensated for by oil flowing in or out the accumulator (accu) through the base valve. The pressure drop over both the base valve and the piston valve results in a damping force acting on the piston.

The semi-active shock absorber hardware (see figure 1 right) corresponds to that of a passive shock absorber in which the piston and base valve are each replaced by a check valve. A current controlled CVSA [2] valve connects the two damper chambers. The current to this valve is limited between $i^- = 0.3A$ and $i^+ = 1.6A$, which corresponds to the least and most restrictive positions of the valve (i.e. open and closed), respectively.

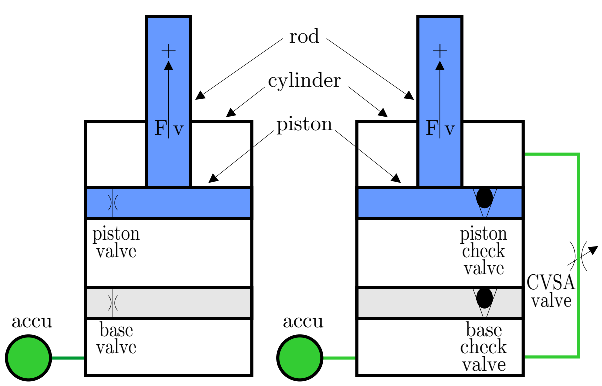

Fig. 1. Working principle of a passive and semi-active shock-absorber

When the rod moves up (positive rattle velocity), the piston check-valve closes and oil flows through the CVSA valve. Because the volume of the rod inside the cylinder reduces, oil is forced from the accumulator into the cylinder through the base check-valve.

[2]CVSA: continually variable semi-active

When the rod moves down (negative rattle velocity), the piston check-valve opens. Because the volume of the rod inside the cylinder increases, the base check-valve closes and oil flows from the cylinder into the accumulator through the CVSA valve.

Figure 2 shows a measured velocity-force damping characteristic for different currents of the considered semi-active shock-absorber. A low/high current to the CVSA valve corresponds to a small/large restriction yielding a low/high damping ratio. This characteristic is obtained experimentally by applying a sinusoidal rattle displacement signal for different settings of the control current to the CVSA valve.

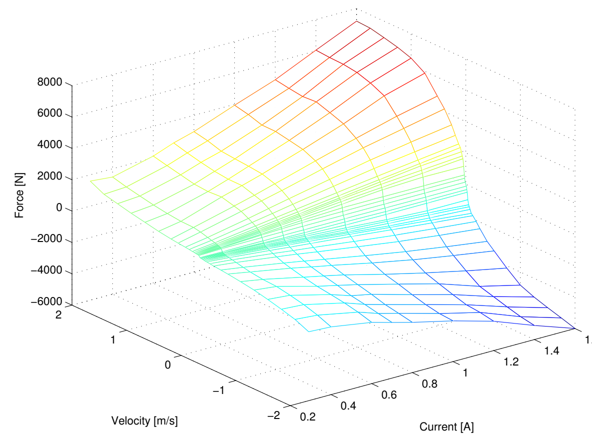

Fig. 2. Semi-active shock absorber damping characteristic relating the damper force to the control current and the rattle velocity

B. Passenger car

A passenger car equipped with 4 semi-active shock-absorbers is available as a test-vehicle. Figure 3 shows the car on an 4 hydraulic shakers which are capable of independently exciting the 4 wheels of the car with a desired road profile.

Fig. 3. Picture of the passenger test-car on the hydraulic shaker

The body acceleration of the car is measured using 4 accelerometers mounted on the 4 corners of the chassis. Also the rattle displacement of all 4 shock-absorbers is measured using linear displacement sensors. For this test-setup, the disturbance inputs of the system are the displacement of the shakers under the wheels of the car. The control inputs are the currents to the 4 semi-active shock-absorber CVSA valves. The developed controllers are implemented on a dSpace 1003 controller board located in the trunk of the car.

III. Feedback linearization

The goal of feedback linearization is to transform the original control inputs of the system (the currents to the semi-active shock-absorber CVSA valves) into virtual control inputs, in order to linearize the dynamic relation between these new control inputs and the outputs of the system to be controlled (the measured body accelerations). If the relation between the system inputs and outputs is linear (or sufficiently linear), control design and tuning simplifies since well known and CACSD-supported linear control design techniques can be applied successfully.

When this transformation includes a physical damper model [4], [5], [6], [7],[8], the new control input corresponds to the damper force. This paper presents an alternative transformation, using a bilinear damper model, which results in another new control input that is *not* the damper force, but which, however, results in a better linearization of the system.

A. Combining feedback linearization and linear control

In order to explain how feedback linearization and linear control are combined, only one corner of the car is considered. The shock absorber generates a force f_d depending on the rattle velocity v_r and the damper valve current i_v. The disturbance input is the road displacement x_a. The measured output is the body acceleration a_b. To control the body of the car, i.e. to reduce the body acceleration, a nonlinear controller is required that feeds back the body acceleration a_b, to the damper control current i_v (see Figure 4).

The design and tuning of such a nonlinear controller is not straightforward. Since the semi-active shock absorber is the most nonlinear element of the system, a linearizing controller is introduced, which calculates an appropriate damper current i_v, such that a desired damper force f_c is realized for the given rattle velocity v_r (see Figure 5). The desired damper force f_c is generated using a linear controller based on the measured body acceleration a_b.

The linearizing controller is based on an inverse damper characteristic. Two different characteristics are considered: the measured velocity-force characteristic shown in Figure 2, which represents a simplified physical damper model (see Section III-B), and an analytically derived bilinear characteristic (see Section III-C).

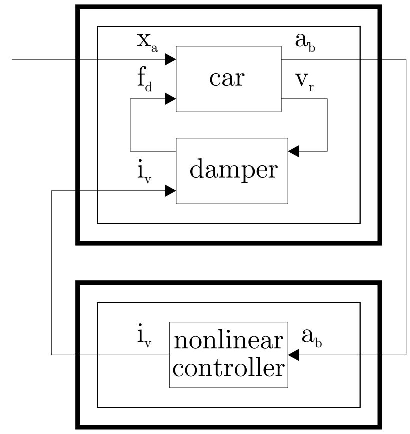

Fig. 4. Original car and damper system controlled by a nonlinear controller which calculates a damper current i_v from a measured body acceleration a_b

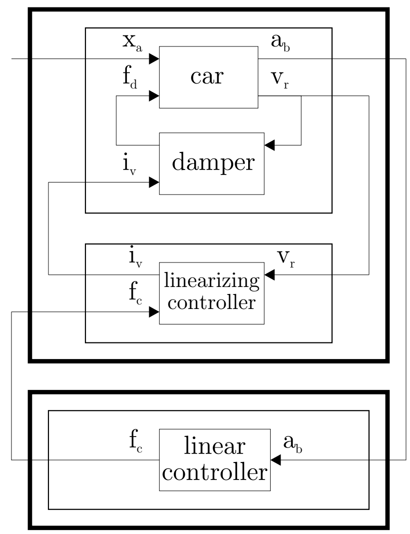

Fig. 5. Feedback linearized system controlled by a linear controller which calculates a desired damper force from a measured body acceleration

B. Linearizing controller based on physical damper model

The classic approach to feedback linearization is to use a physical model. In this application the physical model is a simplified 2D-lookup table (see Figure 2) which relates the damper force f_d to the control current i_v and the rattle velocity v_r. The inverse model is obtained by using 2D-interpolation techniques such that the damper current i_v can be calculated from the rattle velocity v_r and the desired damper force f_c.

C. Linearizing controller based on bilinear damper characteristic

A semi-active shock absorber is a device that delivers a force f_c related to the rattle velocity v_r and the control signal i_v (equation 1). A bilinear approximation of this relation (equation 2) can be simplified (equation 3) by setting coefficients F_0 and F_{10} to 0 since a semi-active shock absorber cannot deliver any force when the rod is not moving ($v_r = 0$). Equations 4 and 5 show the forward and inverse damper model similarity relations. Based on this bilinear approximation, the damper force is linearly related to the product of the rattle velocity and the biased control signal.

$$f_c = F(v_r, i_v) \quad (1)$$
$$f_c = F_0 + v_r F_{01} + i_v F_{10} + v_r i_v F_{11} \quad (2)$$
$$f_c = v_r F_{11}(\frac{F_{01}}{F_{11}} + i_v) \quad (3)$$
$$f_c \sim v_r(i_0 + i_v) \quad (4)$$
$$i_v \sim \frac{f_c}{v_r} - i_0 \quad (5)$$

Since a scaling is a linear operation that is compensated for by the linear controller, a new input f_c can be created which is equal to the product of the rattle velocity v_r and the biased control current $i_v + i_0$. This input no longer has the physical dimension of a damper force. Therefore it is called a *virtual* damper force. Note that this linearizing controller contains only one parameter i_0, the control current bias, around which the controller will operate.

D. Linearizing controller comparison

To check and compare the performance of the linearizing controllers, the following experiment was setup. The test car was placed on the dynamic shakers and excited with the following signals:

- Uncorrelated pink noise road profile displacement signals to the 4 shakers under the wheels of the car.
- Uncorrelated white noise currents to the 4 CVSA valves of the semi-active shock absorbers.

The following signals where measured:

- The 4 accelerometer signals on the 4 corners of the car.
- The 4 rattle displacements of the 4 shock absorbers.

The 4 (virtual) damper forces were calculated offline based on the physical and the bilinear damper models.

Note that the performance of the damper models is *not* validated by comparing the real (measured) and the calculated (virtual) damper forces, since reproduction of this force is not the issue here and moreover since the bilinear model produces a virtual damper force which no longer has this physical meaning since it has been scaled and offset. The intention was to calculate a signal that is more linearly related to the body acceleration then the original control signal. Figure 6 shows the multiple coherence [10] between the calculated damper forces, based on the physical and the bilinear model, and the body acceleration measured on one of the corners of the car. The coherence obtained with the bilinear model is higher, therefore indicating a more linear dynamic behavior, then the one obtained with the physical damper model.

The problem with the physical damper model is that it tries to compensate for the nonlinear current-velocity-force characteristics of the shock-absorber, which is much

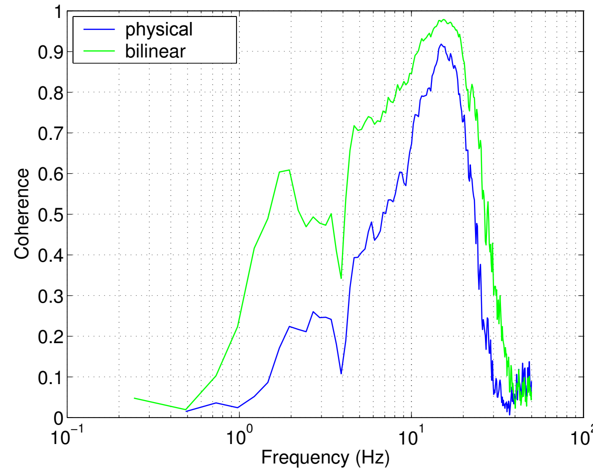

Fig. 6. Linearity comparison based on the multiple coherence of the feedback linearized systems using a physical and bilinear damper characteristic

more complex then the lookup table in Figure 2). This lookup table was obtained with specific harmonic excitation signals for a range of fixed current settings. However, the response of a nonlinear system can be amplitude dependant. Therefore, the model is only valid for excitation signals with similar amplitude levels to the ones that were used for its identification. The bilinear model on the other hand does not include these nonlinear characteristics but implements a bilinear approximation which turns out to be more linearizing for broadband excitation signals, such as stochastic road excitations. Note that this bilinear damper model includes only one parameter i_0 while the lookup table damper model consists of a large amount of data points to describe the nonlinear characteristics of the damper.

IV. MODAL DECOUPLING

The system as described in section II is a 4-by-4 multiple-input multiple-output (MIMO) system which requires a 4-by-4 MIMO controller (see figure 7). The control inputs of the system (outputs of the controller) are the virtual forces of the semi-active shock-absorbers $\mathbf{f_d^c}$. The transformation of these virtual forces into control currents is not considered in this section, and also not shown in figures 7 and 8. The measured outputs of the system (inputs of the controller) are the body accelerations at the 4 corners of the car $\mathbf{a_b^c}$.

A static decoupling matrix D (see equation 6) and its pseudo-inverse $D^\dagger$ are used to decouple the system into its modal motions heave, roll and pitch, which are then controlled by a *modal* (diagonal) controller (see figure 8). The transformed control inputs of the system (outputs of the controller) are 3 virtual modal forces acting on the car through the 4 semi-active shock-absorbers $\mathbf{f_d^d}$. The transformed outputs of the system (inputs of the controller)

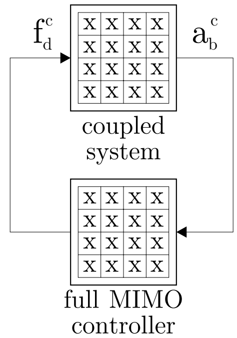

Fig. 7. Controller scheme for the 4-by-4 coupled system

are the 3 modal motions of the car $\mathbf{a_b^d}$.

$$D = \begin{pmatrix} +1 & +1 & +1+\delta & +1+\delta \\ +1 & -1 & +1+\delta & -1-\delta \\ +1 & +1 & -1-\delta & -1-\delta \end{pmatrix} \tag{6}$$

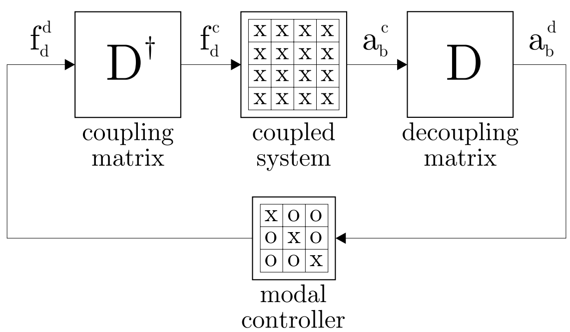

Fig. 8. Controller scheme where the coupled 4-by-4 system is modally decoupled by the static transformation matrices D and $D^\dagger$ and controlled by a diagonal 3-by-3 modal controller

A perfectly symmetric car would be decoupled by the transformation matrices D and $D^\dagger$ with $\delta = 0$. The columns of this matrix correspond to a location of the sensors on the car: front-left, front-right, rear-left, rear-right. The rows represent the modal motions: heave (all in phase), roll (left in anti-phase with right), pitch (front in anti-phase with rear). δ is introduced as a control parameter that can be tuned online in order to achieve symmetric car dynamics as a result of the longitudinally eccentricity of the center of gravity.

V. Control strategies

The linear controller consists of several feedback and feedforward modules, each tackling a specific comfort or handling issue. The output of all modules are summed to a desired modal virtual damper force.

A. Integral feedback control for comfort improvement

The goal is to suppress the modal motions of the car to increase the passengers comfort. The feedback linearization controller and modal decoupling transformations allow to directly specify desired modal forces, to be delivered by the shock-absorbers, from measured modal motions. Based on the skyhook principle ([1]) the diagonal modal controllers consist of 3 first order low-pass filters of which the bandwidth f_b and gain can is tuned online to meet on optimal trade-off between desired comfort specifications and input saturation.

B. Damping adjustment for optimal road handling

Wheelhop is a resonance mode where the wheels of the car move with large amplitude with respect to the road while the car body remains relatively still. This phenomenon deteriorates the handling performance of the car because of the large tire contact force variations. The wheelhop mode can be damped by increasing the control current bias i_0 (see Equation 5) around which the control currents are varied.

C. Steering angle feedforward control

When driving the car in a turn, it will roll because of the centrifugal force, which is proportional to the driving velocity squared and the curvature of the turn. This roll motion is compensated for using a feedforward controller, which adds a modal roll force to the desired damper forces, opposite to the roll motion caused by the turn and proportional to the measured driving velocity squared and the steering angle.

D. Throttle and break pedal feedforward control

When accelerating or braking, the car will pitch respectively backward and forward, proportional to the imposed lateral acceleration. This pitch motion is compensated for using a feedforward controller, which adds a modal pitch force to the desired damper forces. This additional desired pitch force is proportional to the measured breaking force and the desired wheel torque (both available on the CAN bus of the car).

VI. Controller parameter tuning

No model is available to tune the controller e.g. in simulation. The translation of subjective issues like comfort and road handling, into classical control specifications, e.g. bandwidth and settling time, is very difficult and ambiguous. Therefore, the developed controller is equipped with a number of parameters that can be tuned separately on-line, based on comments provided by an experienced test-pilot driving the car over calibrated test tracks. All tunable parameters have a physical interpretation such that their effect on the total behavior of the suspension is clear. The following discusses the different control parameters, their physical interpretation and their effect on the behavior of the car.

- The modal decoupling matrix D contains one parameter δ representing the longitudinal offset of the center of gravity. This parameter is tuned in order to get a balanced car response where the front and rear dynamics behave similarly.

- The integral feedback controller, which consists of 3 first-order low-pass filters, contains 6 parameters: 3 gains and 3 bandwidths. Increasing these gains and bandwidths improves the low-frequency attenuation of the modal motions of the car up to a certain point where they also start to deteriorate the high-frequency harshness.
- The bilinear damper model used to linearize the system dynamics, includes the rattle velocity to calculate the control currents based on the desired virtual damper force. Therefore the measured rattle displacement is differentiated and filtered with a low-pass filter in order to prevent high-frequency noise amplification. The bandwidth of this filter is an important parameter which is tuned to optimize the trade-off between controller bandwidth and noise sensitivity.
- The control current bias i_0 (see Equation 5) determines the average amount of damping in the system and is mainly tuned to optimize the handling performance of the car. Increasing this value provides the car with a better tire force contact but deteriorates the passengers comfort. Experimental tuning showed that the optimal value depends on the type of road: a smooth road allows for a soft setting while a rough road requires a harder setting.

  An appropriate measure to monitor the handling performance is the tire contact force variations. Since these tire contact forces cannot be measured online, it is assumed that the average amount of kinetic energy of the wheels is related to the average amount of tire contact force variation. The absolute wheel velocity can be approximated by the rattle velocity, since at wheelhop resonance, the body of the car remains relatively still with respect to the wheels. This leads to the following adaptive control law to maintain constant handling performance: adapt the control current bias i_0 such that the mean amount of kinetic energy of the wheels remains constant.

  A measure E proportional to the (moving) average amount of kinetic energy of the wheels is calculated online by filtering the sum (of all 4 wheels) of the rattle velocities v_r squared with a first order low-pass filter. The time-constant τ of this filter determines the time over which the average is calculated.

  $$E(t) = \frac{1}{\tau s + 1} (\sum v_r^2) \qquad (7)$$

  The control current bias i_0 adaptation mechanism is implemented as a simple proportional feedback controller that uses i_0 to regulate E towards a desired value E_0. E_0 and the proportional gain are tuning parameters for this adaptation mechanism.

VII. Conclusions

The generic controller structure presented in this paper is derived based on physical insight in car and semi-active suspension dynamics without explicitly using a model. It does not require a model of the cars dynamics, and consequently, is applicable to any semi-active or active suspension system and any type of car. The control structure consists of three basic parts. First the system is linearized by transforming the original current control inputs to virtual damper force input signals. Then the system dynamics are decoupled into their modal components using static decoupling matrices. Finally this linearized and decoupled system is controlled by a linear decentralized controller, which consists of several modules that all tackle a specific comfort or handling issue.

This algorithm has been tested extensively by an experienced test-pilot on a number of standard handling and comfort test-tracks. Since an objective measure to quantify the overall car performance is not available, we rely on the findings of the test-pilot, which can be summarized as follows: *The parameters of the developed controller could be tuned on-line very easily and intuitively based on my feedback during all manoeuvres and critical handling situations, yielding a significant improvement of the car behavior w.r.t. comfort and handling.*

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