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Modeling and control of an off-road truck using electrorheological dampers

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Abstract. This work deals with the mathematical modeling and control of the semi-active suspension of an MAN off-road truck with a payload of 5 tons which comprises electrorheological dampers. Thereby, a cascaded control structure with four controllers for the control of a quarter-car in the inner control loop and a superimposed control strategy for the overall vehicle is used. The main goal of the control strategy is to reduce the motion of the chassis (especially roll, pitch and vertical movement) while increasing driving stability. The capability of the overall control strategy is demonstrated by means of simulation studies and measurement results.

1. Introduction

Increasing demands on driving comfort and driving stability over the last years require the development of new active and semi-active suspension systems for automotive applications. In general, in semi-active suspension systems as considered in this paper, the damping characteristics of the shock absorber has to be varied over a large range. Nowadays, this change of the damping characteristics is in most cases achieved by using electro-hydraulic adjustable dampers, see, e. g., [1].

In order to circumvent the drawback of an electro-hydraulic suspension system regarding e. g. the dynamics, magnetorheological and electrorheological semi-active suspension systems were introduced in the recent years, see, e. g., [2], [3] and [4]. Such dampers usually exhibit a symmetrical damping characteristics whereas passive shock absorbers used in this field of application possess an asymmetrical damping characteristics, i. e. the damping is considerably higher for rebound than for compression. This is mainly due to the fact that such a characteristics yields a good compromise between driving stability and driving comfort. Therefore, one basic idea in the design of the ER dampers used in this work was to adopt the asymmetrical behavior of a passive automotive shock absorber and to combine it with an active adjustment of the damping characteristics by means of an ER valve.

Based on the mathematical modeling of the ER damper and the vehicle in the next section, a nonlinear control strategy for the damper and the vehicle is developed, which is designed to meet the requirements on driving stability and comfort. The feasibility of the proposed control strategy is shown by means of simulation studies and measurement results.

2. Mathematical Modeling

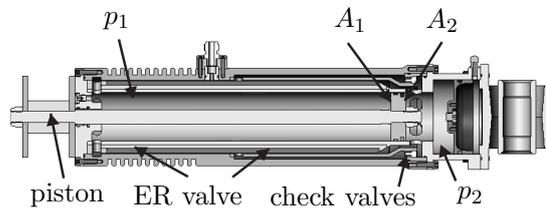


Figure 1. Longitudinal section of the ER damper.

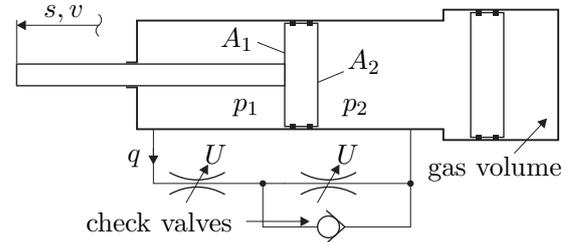


Figure 2. Simplified schematics of the ER damper in figure 1.

A longitudinal section of the ER damper is depicted in figure 1. Therein, A_1 and A_2 are the effective piston areas and p_1 and p_2 denote the pressure in chamber 1 and 2 of the cylinder. Furthermore, s and $v = \frac{d}{dt}s$ denote the position and the velocity of the piston. In figure 1, it can be seen that the ER valve forms an annular gap and is together with a number of check valves integrated within the ER damper. The corresponding schematics in figure 2 shows the fluid path between the two chambers composed of the ER valve and the check valves. During compression ($v < 0$), the check valves are open and the left hand part of the ER valve dominates the pressure difference while during rebound ($v > 0$) the check valves are closed and the fluid has to flow through the full length of the ER valve leading to higher pressure differences and thus to higher damping forces. The mathematical modeling of a quite similar ER damper prototype under the assumption of a Bingham-like material model for the ER fluid is show in [5]. Accordingly, the volume flow q through the ER valve and the check valves can be derived in the form $q = q(p_2 - p_1, U)$ with U the voltage applied to the ER valve. For the subsequent mathematical modeling of the truck, one is mainly interested in the total force $F_{ER} = F_d + F_c$ generated by the ER damper acting on the truck, where the damping part $F_d = A_1(p_2 - p_1)$ is responsible for the dissipation of energy and the part $F_c = (A_2 - A_1)p_2$ refers to a spring force resulting from the different areas A_1 and A_2 of the damper.

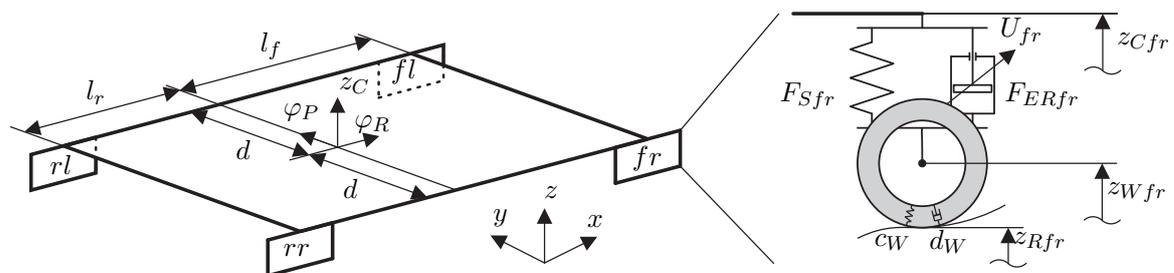


Figure 3. Schematics of the vehicle with the relevant degrees-of-freedom.

The mathematical model of the truck incorporates the vertical degree-of-freedom and the roll and pitch movement of the chassis while effects like braking, cornering or acceleration are not considered. This leads to the generalized positions of the chassis $\mathbf{p}_C = [\varphi_R, \varphi_P, z_C]^T$ with the roll angle φ_R , the pitch angle φ_P and the vertical position of the center of gravity of the chassis z_C . Furthermore, each of the four wheels has an independent vertical degree-of-freedom, whereas the tires are modeled in form of a linear spring/damper system with the tire stiffness c_W and the tire damping d_W , cf. figure 3. In order to indicate each of the four wheels individually, the index pair $\alpha\gamma$ with $\alpha \in \{f = \text{front}, r = \text{rear}\}$ and $\gamma \in \{l = \text{left}, r = \text{right}\}$ is used. Doing so, the vertical excitation of the wheels due to the road surface is denoted by $\mathbf{z}_R = [z_{Rfl}, z_{Rfr}, z_{Rrl}, z_{Rrr}]^T$ and the vertical position of the wheels and the points of the chassis where the dampers and

springs are mounted read as $\mathbf{z}_W = [z_{Wfl}, z_{Wfr}, z_{Wrl}, z_{Wrr}]^T$ and $\mathbf{z}_C = [z_{Cfl}, z_{Cfr}, z_{Crl}, z_{Crr}]^T$, respectively. With the four spring forces $\mathbf{F}_S = [F_{Sfl}, F_{Sfr}, F_{Srl}, F_{Srr}]^T$, which are nonlinear functions of $\mathbf{z}_W - \mathbf{z}_C$, and the mass matrix of the chassis $\mathbf{M}_C = \text{diag}(J_R, J_P, m_C)$ (J_R and J_P are the mass moments of inertia about the φ_R and φ_P axis, respectively and m_C is the total mass of the chassis) the equations of motion of the chassis read as

$$\frac{d}{dt}\mathbf{p}_C = \mathbf{v}_C \quad \frac{d}{dt}\mathbf{v}_C = \mathbf{M}_C^{-1}\mathbf{T}_C(\mathbf{F}_S + \mathbf{F}_{ER}) - [0, 0, g]^T \quad (1)$$

with

$$\mathbf{T}_C = \begin{bmatrix} d & -d & d & -d \\ -l_f & -l_f & l_r & l_r \\ 1 & 1 & 1 & 1 \end{bmatrix} \quad \text{and} \quad \mathbf{F}_{ER} = [F_{ERfl}, F_{ERfr}, F_{ERrl}, F_{ERrr}]^T. \quad (2)$$

Thereby, a small angle approximation is used and g denotes the gravitational constant. Furthermore, the four equations of motion of the wheels take the form

$$\frac{d}{dt}\mathbf{z}_W = \mathbf{v}_W \quad \frac{d}{dt}\mathbf{v}_W = \mathbf{M}_W^{-1}(\mathbf{F}_W - \mathbf{F}_S - \mathbf{F}_{ER}) - g[1, 1, 1, 1]^T \quad (3)$$

with the mass matrix of the wheels $\mathbf{M}_W = \text{diag}(m_{Wf}, m_{Wf}, m_{Wr}, m_{Wr})$. Therein, the forces due to the tire deflection $\mathbf{F}_W = [F_{Wfl}, F_{Wfr}, F_{Wrl}, F_{Wrr}]^T$ are of the form $\mathbf{F}_W = \mathbf{C}_W(\mathbf{z}_R - \mathbf{z}_W) + \mathbf{D}_W(\mathbf{v}_R - \mathbf{v}_W)$ with $\mathbf{v}_R = \frac{d}{dt}\mathbf{z}_R$, the matrix of the tire stiffnesses $\mathbf{C}_W = \text{diag}(c_W, c_W, c_W, c_W)$ and the matrix of the tire damping coefficients $\mathbf{D}_W = \text{diag}(d_W, d_W, d_W, d_W)$.

3. Controller Design

The fundamental task is to track a desired damping force $\tilde{F}_{d\alpha\gamma}$ on each ER damper as accurate as possible within the physical limits. Using the velocities $v_{\alpha\gamma} = v_{C\alpha\gamma} - v_{W\alpha\gamma}$, the required voltage $U_{\alpha\gamma}$ to be applied to the ER valves can be calculated in the form [5]

$$U_{\alpha\gamma} = U_{\alpha\gamma}(v_{\alpha\gamma}, \tilde{F}_{d\alpha\gamma}), \quad \alpha \in \{f, r\}, \quad \gamma \in \{l, r\} \quad (4)$$

where the desired damping forces $\tilde{F}_{d\alpha\gamma}$ serve as virtual control inputs for the superimposed controller. The superimposed control comprises of two parts. The first one is a combination of sky-hook and ground-hook control, the so-called hybrid-control strategy [6], for each wheel individually to assure good road handling. The second part is concerned with a strategy to reduce the roll, pitch and vertical movement of the chassis. Thereby, the dampers are controlled in such a way that linear damping forces are realized for these three degrees-of-freedom

$$\mathbf{F}_{Cd} = -\mathbf{K}\mathbf{v}_C \quad (5)$$

with $\mathbf{K} = \text{diag}(k_R, k_P, k_Z)$. The resulting desired force $\tilde{F}_{d\alpha\gamma}$ of each damper in (4) is determined by a suitable force allocation algorithm from the three generalized forces (5) and the forces calculated by the hybrid-control strategy.

4. Simulation Studies

The results of the simulation studies are demonstrated by means of a trapezoidal obstacle [7] with a height of 200 mm, a length of 500 mm and a slope of 30°. Since both (pitch and roll) degrees-of-freedom are of interest, two virtual obstacles are used in simulation, one on the left hand side

and another on the right hand side, where the contact of the right wheels with the trapezoid is 0.5 s later than on the left hand side. With a velocity of the truck of 3.75 km/h, the resulting roll angle is depicted in figure 4 which shows a significant improvement with the ER damper (ERD) compared to the currently used commercial original damper (OD). The corresponding accelerations of the chassis show similar results. By contrast, the driving stability is amongst others determined by the tire deflection as depicted in figure 5. One can see a reduced tendency for take-off of the wheel also leading to a better driving stability.

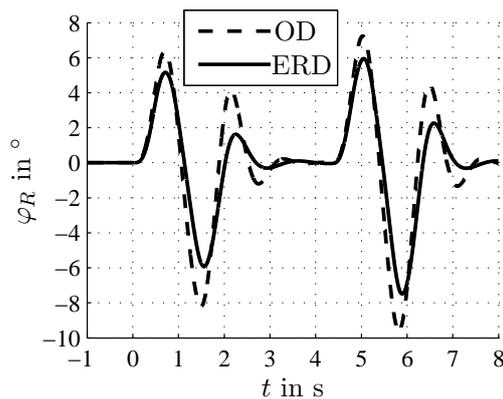


Figure 4. Roll angle.

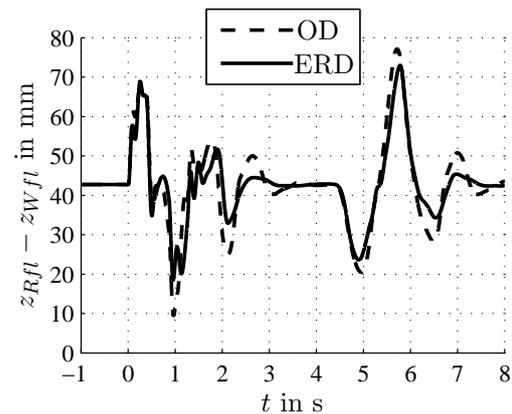


Figure 5. Tire deflection.

5. Measurement Results of the Truck

Measurements of the truck equipped with ER-dampers and of the original truck were performed on different test tracks with various velocities. Since it is almost impossible to perform identical measurements twice for the same track, it is not meaningful to compare and depict the time evolutions of the measurement results in a plot. Therefore, the measurement data is analyzed by means of the rms-value [7]

$$\varepsilon_m = \sqrt{\frac{1}{T} \int_0^T \varepsilon^2(t) dt} \quad (6)$$

for a given signal ε . The improvement for example w. r. t. the acceleration of the chassis on the so-called Schweizer Bahn with 12 km/h and 20 km/h was approximately 11 – 18% in both cases.

6. Conclusion

The proposed control strategy for a truck equipped with ER dampers results in improved driving stability and comfort. This was shown by means of simulation studies and measurement results.

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