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# Robust control of an electromagnetic active suspension system: Simulations and measurements

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ABSTRACT

This paper considers the control of a novel high bandwidth electromagnetic active suspension system for a quarter car model in both simulations and experiments. The nature of the control problem with multiple objectives that have to be optimized as well as the uncertain parameters of the plant call for an $H_\infty$-controller. By changing weighting filters different controllers can be designed, emphasizing either comfort or handling. Using the high bandwidth of the actuator comfort can be improved by 40% over the passive BMW whilst keeping suspension travel within the same limits. Using a different controller, handling can be improved up to 30%, limited by RMS actuator force.



## 1. Introduction

A conventional car suspension is always a trade-off between comfort and handling. Over the last decade, top of the line manufacturers have therefore developed active suspension systems to enhance comfort when driving straight whilst improving handling while cornering. Current day examples are the ABC system [3] employed by Mercedes which contains a hydraulic actuator in series with a passive suspension. Whilst this system can provide energy to the suspension, a bandwidth of only 5 Hz is obtained and continuous pressurization is required making the energy demands very high. Another example is the Delphi magneto-rheological damper [2] which, under the influence of a magnetic field, can change its damping value within a specified range. Benefits of this system are its high bandwidth and low power requirement. Energy can, however, not be supplied to the system making this a semi-active system.

Various other solutions have been developed to provide active suspension, however they all suffer from either low bandwidth [19], high power demands or limited usability. The only suspension that is claimed to solve all of these problems is developed by Bose Corp. [1], details of this system are, however, not provided.

Given these drawbacks, a novel tubular permanent magnet electromagnetic actuator [9] was designed as shown in Fig. 1. It is capable of delivering direct drive in a small volume. Furthermore, the bandwidth it can achieve is much higher than that of other fully active systems. Power consumption is lower than that of a hydraulic system since no continuous pressurization is necessary and energy can even be recuperated depending on the damping value and controller settings [10].

To prove the increased efficiency and bandwidth of this electromagnetic suspension a controller has to be developed. Numerous publications exist on the control of active suspension systems, for example Lee and Kim [17] considers lead-lag, LQ and fuzzy control for a brushless tubular permanent magnet actuator. Due to the limited peak force (29.6 N) of the actuator, a scaled down (sprung mass 2.3 kg, unsprung mass 2.27 kg) test setup is considered making the setup not representing a typical vehicle (sprung-unsprung ratio 10:1). Furthermore, the parameters of the setup are considered to be fixed and fully known. Performance gains up to 77% were achieved, however, no notion was made on the deterioration of suspension travel or handling. On the other hand, Lauwerys et al. [16] does use a full size quarter car setup and also includes uncertainties in the design of the controller. However, with the actuator being hydraulic, only a reductions in the sprung resonance is considered (1.5 Hz) opposed to the region where humans are most sensitive (4–10 Hz). Furthermore, RMS power requirements of the hydraulic suspension system are 500 W per corner, making the system inefficient.

This paper considers control of the active suspension using an $H_\infty$-controller. This control topology was chosen such that variations in the plant that occur in practice can be accounted for. Furthermore, weighting filters can be added to emphasize comfort (vertical acceleration) or handling (tire compression), whilst keeping both actuator force and suspension travel within its limits. By making use of frequency dependent weighting both objectives can be optimized in their most sensitive frequency band.

Design of the controller was done using a quarter car model based on a BMW 530i. Furthermore, a full size quarter car test

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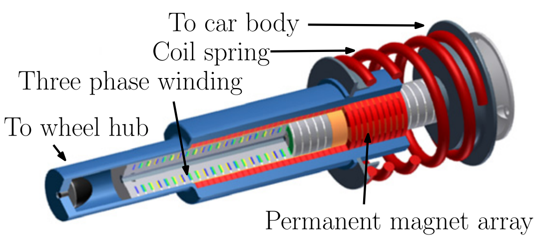


**Fig. 1.** Tubular permanent magnet actuator in parallel with a passive spring.

setup has been built that describes the vertical dynamics of a vehicle. Multiple suspension struts can be fitted, thereby making it possible to compare the suspension Strut of the BMW with the controlled electromagnetic suspension. It will be shown that a 41% improvement in comfort can be achieved in the frequency range of highest human sensitivity (4–10 Hz), limited by the available suspension travel. Handling can be improved by 31%, limited by the available RMS actuator force.

The outline of the paper is as follows. In Section 2 the system model will be discussed which consists of an introduction to the active suspension, the quarter car model that will be used for simulations and the road description. After this the main control problem will be presented in detail. In Section 4 the full size test setup will be shown. Design of the controller will be explained in Section 5. Results of the controllers designed can be seen in Section 6. Finally, conclusions are drawn in Section 7.

## 2. System model

A quarter car model will be used. For this all parameters are based on a BMW 530i. This German built saloon car is well known for its sportiness, agility and comfort. With its aluminum bonnet and front quarter panels a near to perfect 50.9/49.1% front to rear weight distribution is achieved. The front suspension system, which will be replaced by the active suspension system, is a MacPherson strut. This suspension strut has a spring stiffness of 30.01e3 N/m as is shown in Table 1, together with the other vehicle parameters. The damping of the passive suspension strut is a nonlinear function of the vertical velocity and can be seen in Fig. 3. Performance of the active suspension will be compared with this passive front suspension.

**Remark 1.** A half or full car model is not chosen as the test setup is designed as a quarter car setup. The model is therefore also a quarter car. Furthermore, as we are most interested in comfort, which is defined by vertical acceleration of the vehicle body, a model that only describes this is chosen.

**Table 1**
Technical data of the BMW 530i.

| Parameter | Value |
|---|---|
| Unloaded mass | 1546 kg |
| Maximum loaded mass | 2065 kg |
| Unsprung mass front (2 wheels) | 96.6 kg |
| Unsprung mass rear (2 wheels) | 89.8 kg |
| Spring stiffness | 30.01e3 N/m |
| Tire stiffness min–max (single tire) | 3.1e5–3.7e5 N/m |
| Weight distribution front-rear | 50.9–49.1% |
| Maximum compression (bump) | 0.06 m |
| Maximum extension (rebound) | 0.08 m |

### 2.1. Active suspension system

The retrofit suspension system, as shown in Fig. 1 consists of an electro-magnetic actuator in parallel with a mechanical spring to maintain the height of the vehicle. The actuator is a tubular slotted three-phase permanent magnet actuator with a maximum RMS force of 1000 N and similar suspension travel as the passive BMW suspension. Fail safe passive damping is provided by means of eddy-currents, see Fig. 3. Further specifications and its mode of operation are given in [7,8].

Three sensors are fitted on the actuator, being the sprung mass acceleration sensor, suspension travel sensor and unsprung mass acceleration sensor. The sprung acceleration sensor is fitted because it gives a direct measure of comfort of the vehicle. Second, the laser sensor is installed such that the suspension travel can be directly measured using this sensor, thereby commutation of the actuator is executed precisely. Finally, the unsprung acceleration sensor is installed to estimate the state of the tire since it is impossible to measure tire compression directly. This set of sensors provides all information necessary and is most commonly used in literature [21].

### 2.2. Quarter car model

A quarter car model represents one corner of a vehicle for which only the vertical dynamics are considered. Fig. 2 shows a graphical representation of the quarter car including an actuator. The body of the car is represented by the sprung mass, $m_s$. The wheels, brakes and part of the suspension is represented by the unsprung mass $m_u$. The suspension stiffness and damping are denoted by $k_s$ and $d_s$ respectively with the tire stiffness denoted by $k_t$. The degrees of freedom are the displacement of the sprung ($z_s$) and unsprung mass ($z_u$). The road displacement, $z_r$, is prescribed by the road profile as will be discussed below. Finally, the actuator force is denoted by $F_{act}$. The equations of motion are given by

$$m_s\ddot{z}_s = -k_s(z_s - z_u) - d_s(\dot{z}_s - \dot{z}_u) + F_{act} \tag{1}$$

$$m_u\ddot{z}_u = k_s(z_s - z_u) + d_s(\dot{z}_s - \dot{z}_u) - k_t(z_u - z_r) - F_{act}, \tag{2}$$

where it is assumed that tire-road contact is maintained at all time, i.e. $k_t(z_u - z_r) > 0$.

Variations that occur in the quarter car are based on physical changes such as the number of people in the car, the damper veloc-

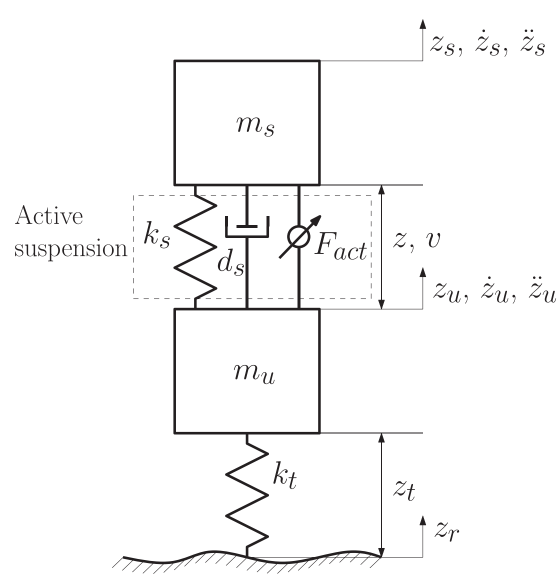


**Fig. 2.** Quarter car model.