

Optimization of Damper Top Mount Characteristics for Semi-Active Suspension System

Mina M.S. Kaldas
Minia University

Kemal Çalışkan, Roman Henze, and Ferit Küçükay
IAE, TU Braunschweig

ABSTRACT

Semi-active suspension offers variety of damping force range which demands greater need to optimize the top mount to ensure multiple objectives of ride comfort, harshness and safety can be achieved. For this purpose, this paper proposes a numerical optimization procedure for improving the harshness performance of the vehicle through the adjustment of the damper top mount characteristics of the semi-active suspension system. The proposed optimization process employs a frequency dependent combined objective function based on ride comfort and harshness evaluation. A detailed and accurate damper top mount mathematical model is implemented inside a validated full vehicle model to provide a realistic simulation environment for the optimization study. The semi-active suspension system employs a Rule-Optimized Fuzzy-Logic controller. The ride comfort and harshness of the full vehicle are evaluated by analyzing the body acceleration in different frequency ranges. The dynamic stiffness of the damper top mount is used to describe the optimum damper top mount characteristics for different optimization case studies. The results show that, the proposed optimization routine enables - compensation of the harshness degradation in the semi-active suspension system by adjusting the characteristics of the damper top mount.

CITATION: Kaldas, M., Çalışkan, K., Henze, R., and Küçükay, F., "Optimization of Damper Top Mount Characteristics for Semi-Active Suspension System," *SAE Int. J. Commer. Veh.* 10(1):2017, doi:10.4271/2017-01-0412.

INTRODUCTION

The main task of the vehicle suspension system is to isolate the vehicle from the road irregularities while providing adequate road holding capability. The limitations of passive suspensions regarding the conflict between ride comfort and road holding have motivated the investigation of controlled suspension systems, both active and semi-active [1]. The foundation of controlled suspensions for the car mass-market can probably be dated back to the early 1960s, when Citroën introduced hydro-pneumatic active suspension in its luxury cars; at that time those suspensions were still untouched by electronics [2]. Given this tribute to Citroën, the 1980s was the real start of electronic suspension; analog electronics were already developed, and the magic of the fully active suspensions attracted both the Formula 1 competition and the car manufacturers [3]. During this decade the exceptional potential of replacing the conventional spring-damper system with a fully electronically controllable fast-reacting hydraulic actuator was demonstrated. However, the high costs, significant power absorption, bulky and unreliable hydraulic systems, and uncertain management of the safety issues prevented its impact on the mass-market of the vehicles [4].

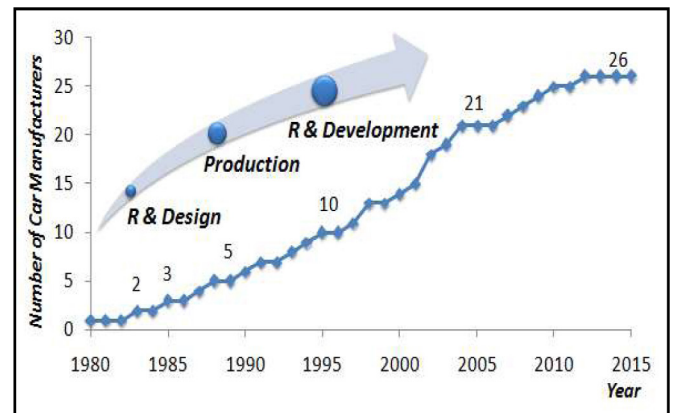


Figure 1. Number of the car manufactures for controlled suspension [4]

Since the 1990s, it became increasingly clear that the best compromise of cost (component cost, weight, electronics and sensors, also power consumption, etc.) and performance (comfort, handling and safety) was to be found in another technology of electronically controllable suspensions: the semi-active suspension or the Continuous Variable Damping suspension. For example, the car manufacturers like Audi, VW, Mercedes, Ford, Volvo and GM cooperated together with the controlled dampers suppliers such as Sachs, Tenneco, Breda and Monroe to design, manufacture and develop different types of the

semi-active suspension systems which are introduced in the mass-market of cars [4]. In last few years most of the luxury car manufactures have started offering semi-active suspension as standard equipment. In fact, now it is not uncommon to see semi-active suspension system as an option on non-luxury vehicles.

After a survey of the previous research studies on the semi-active suspension systems in the automobile market, it can be concluded that the semi-active dampers with solenoid valve are well developed and made commercially available, and therefore have already found their way to the vehicle market in comparison with the other types of the controlled dampers. The semi-active control algorithms have been developed by suppliers or OEM's to achieve the correct balance of comfort and road holding control [5], [6], [7], [8], [9], [10], [11]. On the other hand, the available research studies on the semi-active suspension systems have been applied only for the ride comfort frequency range up to 15 Hz. However, semi-active suspension systems might have a negative effect on the vehicle harshness performance, which was not considered yet in the available research studies [4].

The damper top mounts are used in the vehicles not only to provide ideal Noise-Vibration-Harshness (NVH) performance but also to improve ride comfort, driving safety, and handling. The ride comfort and harshness can be considered as the vibration response of the vehicle body at different frequency ranges. Vehicle ride comfort can be evaluated with vertical acceleration of the body up to 20 Hz while the harshness can be considered as the body vertical acceleration in the frequency range over 20 Hz until 100 Hz [1]. Another type of the vehicle harshness is the impact harshness (IH) which also affects the subjective impression of ride comfort. This type of harshness involves the vibration response of the vehicle which is referred to as IH events [12], [13], [14], [15].

It is relatively easy and cost effective to change the compliance of the damper top mounts compared with changing the characteristics of other suspension components to improve the vehicle ride comfort and harshness performance [16], [17], [13], [14]. For the adjustment of top mount characteristics to improve the ride comfort and harshness performance, the correlation between these has to be understood. For this purpose, in [18], a detailed mathematical damper top mount model based on restoring force mapping technique is presented which enables accurate simulation of the damper top mount force response. The presented damper top mount model consists of three elements, which are the nonlinear elastic element, the nonlinear friction element, and the nonlinear viscous element. The amplitude dependence of the top mount characteristics is modeled by using the friction element and the elastic element, while the frequency dependence of the top mount is modeled by using the restoring-force-mapping technique. Furthermore, a new procedure is proposed for the identification of the model parameters, which is based on a two-stage optimization routine where two sets of measurement data from the amplitude and frequency dependent tests are used. The mathematical model parameters of different commercial damper top mounts are identified using experimental data. The model is then validated by comparing the measured and simulated forces for three different top mounts. Good agreement between the measured and the simulated forces is obtained. Furthermore, the proposed model is found to be superior to the existing rubber isolator models. The damper top

mount model proposed in [18] is used first to study the influence of the damper top mount characteristics on the ride comfort and harshness performance of a vehicle [19], and to optimize the damper top mount characteristics for the passive suspension systems to improve the harshness performance of the vehicle [4]. All top mount parameters investigated in this work have been shown to have visible effects on either one or both of the ride comfort and harshness characteristics of the vehicle. Therefore, it was concluded that, significant improvements can be achieved in both ride comfort and harshness performance of a vehicle through proper adjustment of the damper top mount characteristics.

In this study, an optimization technique to optimize the damper top mount characteristics of the semi-active suspension system is developed. The proposed optimization technique employs a new combined objective function based on ride comfort and harshness evaluation. A detailed and accurate damper top mount mathematical model is implemented inside a validated full vehicle model to constitute the simulation environment required for the optimization study. The damper top mount characteristics are optimized for the semi-active suspension system based on the Rule-Optimized Fuzzy-Logic controller published by the authors in previous papers [20], [21], [22], [23]. The ride comfort and harshness of the full vehicle are evaluated by analyzing the body acceleration in different frequency ranges. While, the dynamic stiffness of the damper top mount is used to describe the optimum damper top mount characteristics for different optimization case studies.

VEHICLE MATHEMATICAL MODEL

In this study, a generic full vehicle suspension model with non-linear suspension component characteristics has been used. The vehicle model contains seven masses which represent the body, four wheels, engine, and driver. The model has eleven degrees of freedom which include the bounce, pitch and roll motions of the body and engine, and the bounce motion of the wheels and driver. The suspension between the vehicle body and wheel is modeled using a linear spring and a viscous damper in series with a rubber top mount at each corner. The damper and top mount masses are neglected. The nonlinear characteristics of the damper and the top mount are taken into account by using detailed damper top mount model presented in [18] and non-linear Force-Velocity damper characteristics. Point contact is assumed between the road and the tire at each corner and tire-road separation is taken into account. The non-linear force characteristics of the tires are considered within the model by using an experimentally identified non-parametric Force-Displacement-Velocity map for the tires. Furthermore, the anti roll bars at front and rear axles are also included in the vehicle model. The engine is assumed to have bounce, pitch and roll degrees of freedom with respect to the vehicle body. The engine body is supported by three equivalent vertical mounts, which are modeled using linear springs in parallel with viscous dampers. The driver is included inside the full vehicle model as a one degree of freedom system with the vertical motion only, and it is modeled as a mass supported by a linear spring and a linear damper. The axis convention and the notations used for the vehicle body, wheels and road profile are shown in Figure 2.

The equations of motion for the bounce, pitch and roll responses of the vehicle body are written as follows:

$$M_b \ddot{Z}_b = -F_{sb} - F_{sb2} - F_{sb3} - F_{sb4} + F_{e1} + F_{e2} + F_{e3} + F_{st} \quad (1)$$

$$I_b \ddot{\phi}_b = L_1 F_{sb1} - L_2 F_{sb2} + L_1 F_{sb3} - L_2 F_{sb4} + T_e^p + T_{st}^p \quad (2)$$

$$J_b \ddot{\chi}_b = B_{s1} F_{sb1} + B_{s2} F_{sb2} - B_{s1} F_{sb3} - B_{s2} F_{sb4} + T_{stf} + T_{str} + T_e^r + T_{st}^r \quad (3)$$

The vehicle is front-engine front-drive type and the engine-gearbox block is supported by three mounts. First mount is situated at the left side while the second one installed at the right edge of the engine. The third engine mount is the sub-frame mount and installed at the middle and near to the left side. The equations of motion for the engine vertical, pitch and roll responses are:

$$M_e \ddot{Z}_e = -F_{e1} - F_{e2} - F_{e3} \quad (4)$$

$$I_e \ddot{\phi}_e = L_{e1} F_{e1} - L_{e2} F_{e2} - L_{e3} F_{e3} \quad (5)$$

$$J_e \ddot{\chi}_e = B_{e1} F_{e1} - B_{e2} F_{e2} - B_{e3} F_{e3} \quad (6)$$

The equations of motion for vertical response of the driver can be written as:

$$M_{st} \ddot{Z}_{st} = -F_{st} \quad (7)$$

The equations of motion for the bounce motion of the wheels can be written as:

$$M_{wi} \ddot{Z}_{wi} = -F_{wi} + F_{swi} \quad (8)$$

The equation of motion for the damper top mount dynamic characteristics can be written as the following, and more details about it is shown in [18].

$$F_{mi} = F_{elastic_i} + F_{friction_i} + F_{viscous_i} \quad (9)$$

$$F_{elastic_i} = K_{p1i} K_{e1i} (Z_{bi} - Z_{ki}) + K_{p2i} K_{e2i} (Z_{bi} - Z_{ki})^2 + K_{p3i} K_{e3i} (Z_{bi} - Z_{ki})^3 \quad (10)$$

$$F_{friction_i}(z) = \begin{cases} F_{fr_s} & \text{if } z = z_s \\ F_{fr_s} + \frac{z - z_s}{z_{fr}(1-\delta) + (z - z_s)} (K_{p4i} F_{fr_{maxi}} - F_{fr_s}) & \text{if } z > z_s \\ F_{fr_s} + \frac{x - x_s}{z_{fr}(1+\delta) - (z - z_s)} (K_{p4i} F_{fr_{maxi}} + F_{fr_s}) & \text{if } z < z_s \end{cases} \quad (11)$$

$$F_{viscous_i} = K_{p5i} f_{damper}((\dot{Z}_{bi} - \dot{Z}_{ci}), (Z_{bi} - Z_{ci})) \quad (12)$$

$$F_{ej} = K_{ej} (Z_{ej} - Z_{bej}) + C_{ej} (\dot{Z}_{ej} - \dot{Z}_{bej}) \quad (13)$$

$$F_{st} = K_{st} (Z_{st} - Z_{bs}) + C_{st} (\dot{Z}_{st} - \dot{Z}_{bs}) \quad (14)$$

$$F_{wi} = K_{ti} (Z_{wi} - Z_{oi}) \quad (15)$$

$$F_{swi} = K_{si} (Z_{bi} - Z_{wi}) + F_{di} \quad (16)$$

$$T_e^p = -L_{be1} F_{e1} - L_{be2} F_{e2} - L_{be3} F_{e3} \quad (17)$$

$$T_e^r = B_{be1} F_{e1} - B_{be2} F_{e2} - B_{be3} F_{e3} \quad (18)$$

$$T_{st}^p = -F_{st} L_{bs} \quad (19)$$

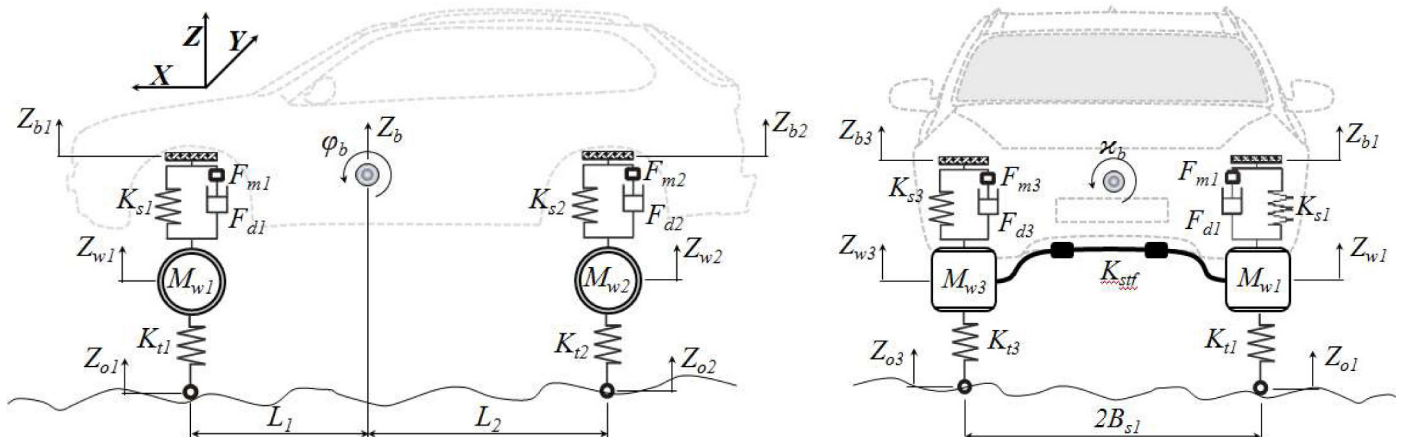


Figure 2. The 11 DoF vehicle model