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Optimization of Suspension System Parameters for a SUV

Murat Otkur, Narjes Alshammari, Noura Abdullah, Danah Alkandari, Hanan Thyab, Latifah Alduwaisan
College of Engineering and Technology,
American University of the Middle East, Kuwait
Murat.Otkur@aum.edu.kw, 40026@aum.edu.kw, 45123@aum.edu.kw, 37970@aum.edu.kw, 38931@aum.edu.kw, 38790@aum.edu.kw

Abstract **–** Ride comfort has been an important development parameter for transport vehicles starting from early horse carriages with simple leaf spring suspension systems, up to modern vehicles with the state-of-the-art suspension systems. A vehicle without a suspension system will transfer all the disturbances caused by bumps or holes on the road resulting to high acceleration and jerk values at the passenger compartment. Suspension system acts as the cushion of the vehicle when it undergoes road irregularities improving passenger comfort. Softer suspension systems provide better ride comfort via reducing the magnitude of the chassis oscillations however have negative effect on vehicle dynamics considering the fact that they result with loss of traction due to excessive roll motion of the vehicle causing weight transfer from the inner wheels to the outer wheels during cornering manoeuvres. Hence, optimization of suspension system parameters is essential considering both vehicle comfort and dynamics. Similar to all mechanical components, optimization using real hardware is considerably expensive and time consuming. Therefore, model-based optimization is essential to obtain the best performance parameters considering objectives as follows: minimize acceleration magnitude and pitch angle. Within this study a Half Car Model (HCM) for vehicle suspension system is developed in MATLAB / Simulink software and parameters used in the model are tuned for a Sport Utility Vehicle (SUV) using measurements captured via MATLAB Mobile software employed in a mobile phone. A full factorial Design of Experiment (DOE) is developed spanning ± %20 of original values. A regression model is built in Minitab software and it has been showed that optimized parameters result with %3.4 and 9.4% reduction in pitch angle and maximum acceleration values respectively.

Keywords**:** Ride Comfort, Suspension System, Half Car Model (HCM), DOE, Optimization

1. Introduction

In view of the evolution of passenger cars, driving comfort has significantly improved since beginning of 19th century, mainly due to the utilization of suspension systems and enhancements in shock absorber components. The main function of the suspension system of a vehicle is to dampen the irregularities and vibrations resulting from the road disturbances while maintaining contact between tires and road surface so that vehicle longitudinal and lateral dynamics is not degraded. For most applications there is a trade of between vehicle comfort and driving dynamics that can be adjusted via optimizing suspension system parameters such as shock absorber stiffness and damping coefficients.

Modern suspension systems of passenger cars have various types and geometries, consist of several components such as shock absorbers (consists of springs and dampers) and linkages that is responsible for connection between a vehicle body and its wheels. Shock absorber is the main component of a suspension system that allows to soften the vibrations generated by the road irregularities and obstacles [1]. Softer springs reduces the magnitude of the oscillations improving comfort, on the other hand degrade vehicle dynamic stability. Stiffer springs improves vehicle dynamics, reducing vehicle comfort. Considering this conflicting behaviour tuning suspension system parameters can be considered as an optimization problem that needs to be solved using simulation tools.

There are numerous studies for defining the optimum suspension system parameters. Similar to all mechanical components and systems, optimization of suspension systems without use of CAE is very time consuming and expensive considering the drawbacks of prototyping in terms of cost and duration. There are numerous studies performed in literature for modelling and optimization of suspension systems considering both components and kinematics design. Ranganathan et al. designed a helical and wave spring and performed a dynamic analysis using ANSYS software showed that for the investigated conditions use of wave spring reduces the stress developed and deformation by 80% and 89% respectively [2].

Bianco proposed a new enhanced spring model using Multi Body Simulation (MBS) of a rear suspension in Simpack Software and based on this advance model a novel target setting approach is proposed [3].

Considering 4-wheel vehicles, depending on number of wheels modelled there are 3 types of suspension system: Quarter Car Model (QCM), Half Car Model (HCM) and Full Car Model (FCM). QCMs are extensively used in automotive engineering due to their simplicity and capability to provide the qualitatively correct information, especially in the initial design stages of vehicle dynamics. It considers one-fourth of the vehicle weight, one suspension system and one tyre suspension system components assembly. Tiwari et al. developed a mathematical QCM in MATLAB software in order to perform basic analysis of 2 Degree of Freedom (DOF) motion for the passive QCM considering ride comfort metrics [4]. Luczko and Ferdek used a QCM in MATLAB for analyzing the effects of twin-tube hydraulic shock absorber that has an extra double-chamber cylinder in comparison to traditional dampers [5]. Compared to QCMs, HCMs have the capability to capture the pitch motion of the vehicle body which is an important parameter for driving comfort. In a HCM, vehicle is considered like a bicycle where front left and right suspension and wheel assembly is considered as single suspension system. Similarly rear left and right suspension and wheel assembly is considered as a single unit. HCM is capable of simulating the positions of the front and rear suspension system, vehicle mass and pitch angle: total 4 DOF. Goga and Klucik developed a HCM in MATLAB / Simulink and optimized the suspension system parameters using genetic algorithm with the 4 optimization criteria as: minimum of vertical acceleration of vehicle body, minimum of angular acceleration of vehicle body and minimum of vertical displacements of wheels and suspension system assemblies [6]. Similarly, Al-Ghanim and Nassar developed a HCM in MATLAB / Simulink for both passive and active suspension system employed vehicles, investigated their performance and performed different control strategies such as PID, fuzzy and their combination [7]. Khan et al. built a HCM in MATLAB / Simulink and afterwards converted the nonlinear HCM model into an equivalent linear system and employed a Linear Quadratic Regulator (LQR) controller in order to minimize the effects of road disturbances [8]. Compared to HCMs, FCM have the capability of capturing the roll motion of the vehicle body, that is a critical parameter for vehicle dynamic characteristics. FCMs have 7 DOF: 4 DOF resulting from the vertical positions of the suspension systems, 3 DOFs for the vertical position, pitch and roll angles of the vehicle body. Shirahatt et al. developed a mathematical FCM with 8 DOF with the addition of driver seat vertical motion and investigated comfort parameters such as maximum vertical acceleration of the passenger seat and tire displacement considering ISO2631 standard and optimized suspension system parameters using genetic algorithm [9]. Kaldas et al. developed a 9 DOF HCM with the addition of driver seat and engine vertical motion to the classical 7 DOF FCM for a passive suspension system vehicle and 13 DOF HCM with the addition of the 4 actuator positions for an active suspension vehicle [10]. They employed a Model Reference Controller (MRC) that contains 8 Proportional Integral Derivative (PID) controllers for both body and wheel control. Authors optimized control parameters employing a cost function containing road holding and ride comfort performance parameters.

The paper is organized as follows: Section 1 embodies the introduction and the literature review, section 2 contains the methodology whereas sections 3 and 4 presents the modelling and optimization content respectively. The paper is concluded in section 5 with the addition of future work as a recommendation.

2. Methodology

Figure 1 contains the flow chart of the conducted study. Within this study an analytical HCM with 4 DOF and capability of simulating vehicle centre of gravity position and pitch angle is developed in MATLAB / Simulink and validated using literature data [6]. MATLAB mobile software is used in an IOS mobile phone to get the vehicle measurements for the bump passing manoeuvre with vehicle vertical acceleration and pitch angles measurements. Built model is tuned for a SUV considering the pitch angle response of the vehicle and vehicle suspension parameters are determined. A full factorial Design of Experiment (DOE) is generated considering the shock absorber stiffness and damping coefficients for the front and rear axles spanning ± 20% variation of the original values. A regression model is built in Minitab software in order to minimize the vehicle response for the bump passing manoeuvre and optimized values are tested in the HCM developed in MATLAB / Simulink.

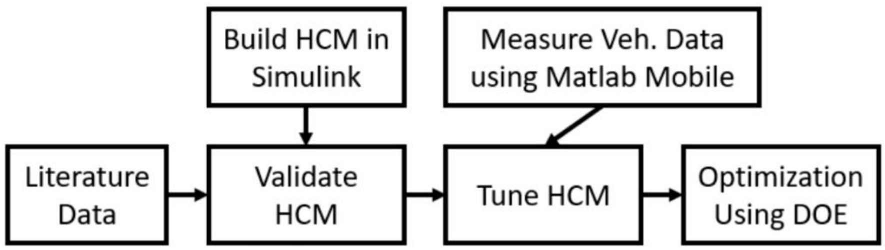


Figure 1: Flowchart of Research.

3. Modelling

Mathematical modelling is essential in order to optimize suspension system parameters as component-based testing is extremely expensive and time dependent. Therefore, a MATLAB / Simulink based time dependent analytical vehicle suspension system is developed.

3.1 Half Car Model

In the HCM all vehicle mass is considered in the vehicle longitudinal axis, therefore it is also referred as bicycle model. Vehicle is represented by a mass that is connected to the ground with two suspension systems: one at the front axle and at the rear axle. The geometry of the suspension systems is neglected, and it is assumed to have shock absorbers in vertical positions. Shock absorbers are modelled as mechanical systems consisting of a spring and a damper. Vehicle tire is modelled as a spring with high stiffness coefficient and damping effect of the tire is neglected in this study. Schematic of the HCM and definitions of the parameters used in the HCM are presented in figure 2 and table 1 respectively.

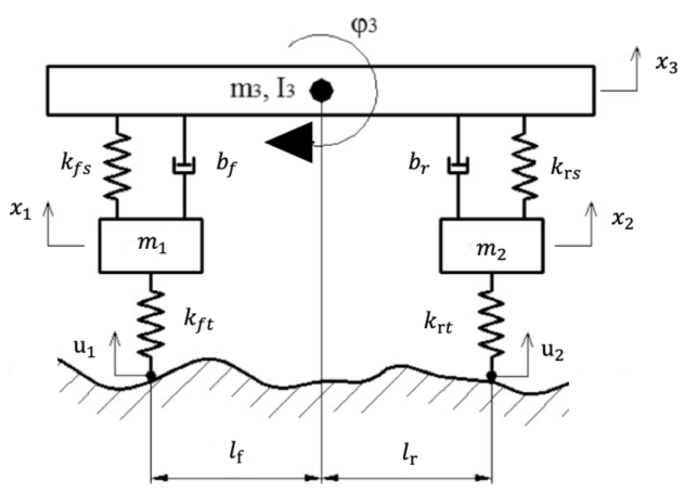


Figure 2: HCM schematic.

Translational Free Body Diagrams (FBDs) for the vehicle mass and suspension systems masses and rotational FBD for the vehicle mass are depicted in figure 3 a-d. Translational Equations of Motion (EQM) developed using the FBDs for each mass is represented in equations 1-3 and rotational EQM is represented in equation 4.

Table 1: Parameters used in HCM.

Parameters	Definition
m_1	Unsprung mass at front axel
m_2	Unsprung mass at rear axel
m_3	Vehicle mass / 2
I_3	Moment of Inertia about COG
k_{ft}	Front tire stiffness coefficient
k_{rt}	Rear tire stiffness coefficient
k_{fs}	Front shock absorber stiffness coefficient
k_{rs}	Rear shock absorber stiffness coefficient
b_f	Front shock absorber damping coefficient
b_r	Rear shock absorber damping coefficient
l_f	Distance of front axis from the vehicle center or gravity
l_r	Distance of rear axis from the vehicle center or gravity

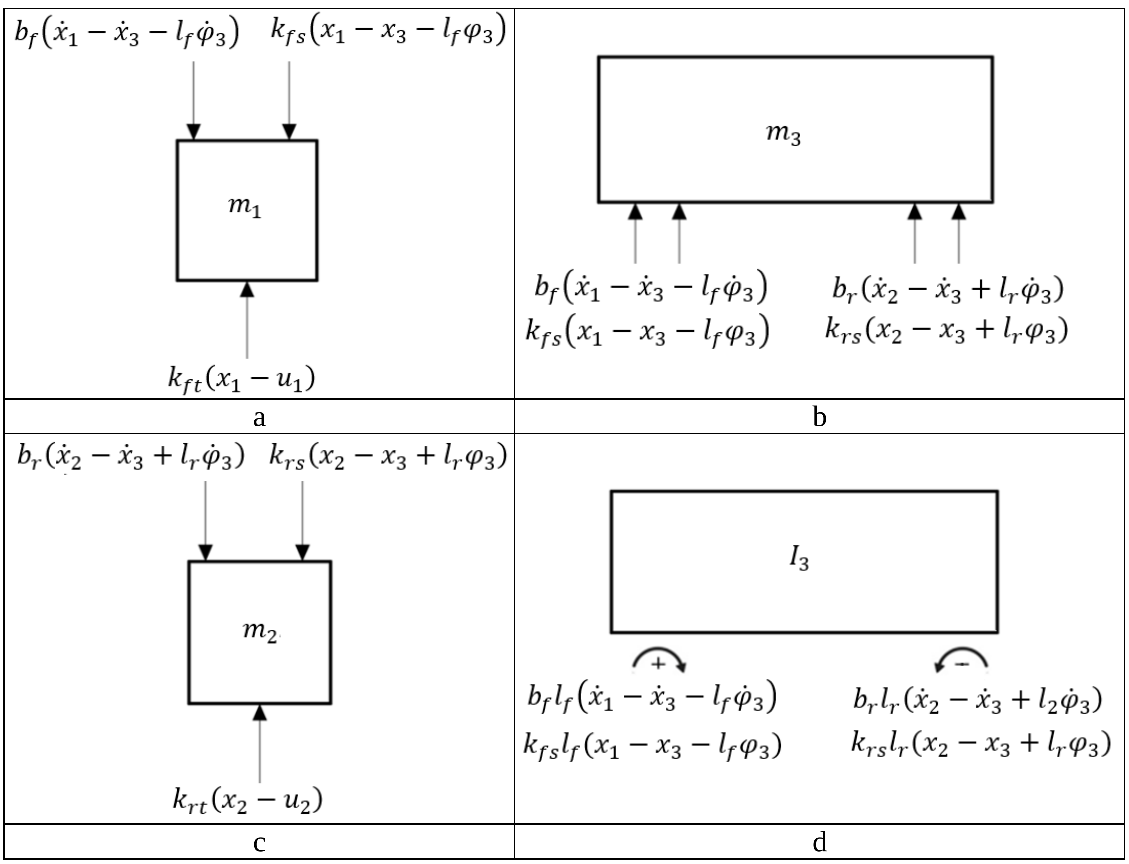


Figure 3: FBDs, a: Front suspension translational, b: Rear suspension translational, c: Vehicle translational, d: Vehicle rotational.

$$m_1\ddot{x}_1 + b_f(\dot{x}_1 - \dot{x}_3 - l_f\dot{\varphi}_3) + k_{fs}(x_1 - x_3 - l_f\varphi_3) - k_{ft}(x_1 - u_1) = 0 \tag{1}$$

$$m_2\ddot{x}_2 + b_r(\dot{x}_2 - \dot{x}_3 + l_r\dot{\varphi}_3) + k_{rs}(x_2 - x_3 + l_r\varphi_3) - k_{rt}(x_2 - u_2) = 0 \tag{2}$$

$$m_3\ddot{x}_3 - b_f(\dot{x}_1 - \dot{x}_3 - l_f\dot{\varphi}_3) - b_r(\dot{x}_2 - \dot{x}_3 + l_r\dot{\varphi}_3) - k_{fs}(x_1 - x_3 - l_f\varphi_3) - k_{rs}(x_2 - x_3 + l_r\varphi_3) = 0 \tag{3}$$

$$I_3\ddot{\varphi}_3 - b_f l_f\left(\dot{x}_1 - \dot{x}_3 - l_f\dot{\varphi}_3\right) + b_r l_r(\dot{x}_2 - \dot{x}_3 + l_2\dot{\varphi}_3) - k_{fs}l_f(x_1 - x_3 - l_f\varphi_3) + k_{rs}l_r(x_2 - x_3 + l_r\varphi_3) = 0 \qquad (4)$$

where $\dot{x}$,$\ddot{x}$, $\dot{\varphi}$ and $\ddot{\varphi}$ is vertical velocity, vertical acceleration, angular velocity and angular acceleration, respectively.

An HCM is developed in MATLAB / Simulink that is able to solve the differential EOMs simultaneously. Developed model is validated using the result of study from the literature [6]. Vehicle is subjected to a step input of 0.1 m and displacement of the front axle suspension versus time is plotted. As shown on the figure 4 literature data and MATLAB / Simulink simulation match each other well.

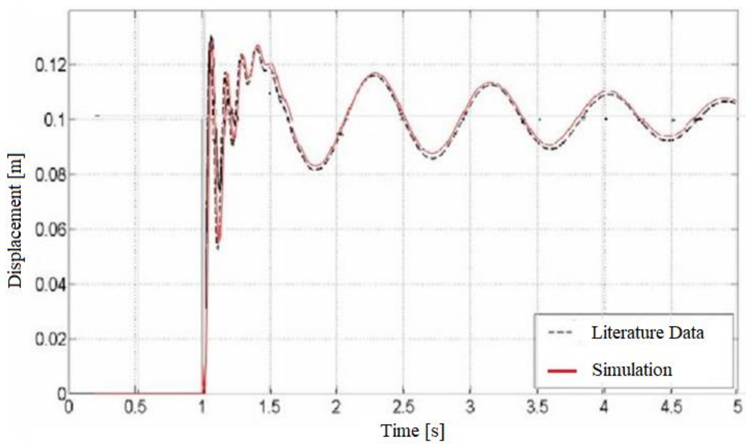


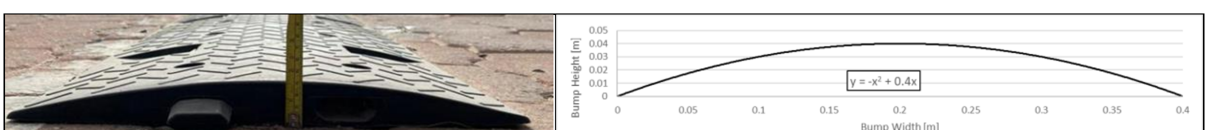
Figure 4: Displacement of front axle suspension for the 0.1 m amplitude step input.

3.2. Vehicle Measurement

Vehicle mass vertical acceleration and pitch angle is measured using MATLAB Mobile software operated in a IOS mobile phone. Sampling rate is set at 10 Hz. Specifications of the used SUV is summarized in table 2. Measurements are taken for a bump passing maneuver as shown in figure 5. Bump height is 4 cm and bump width is 40 cm.

Table 2: Vehicle specifications.

Parameter	Value	Unit
Length	4797	mm
Width	1940	mm
Height	1665	mm
Wheelbase	2874	mm
Curb Weight	1874	kg



a	b

Figure 5: Parabolic bump profile, a: Physical profile, b: Mathematical representation of the bump profile.

3.3. Tuning of the HCM:

Driver mass of 75 kg is added to the curb of the vehicle summing the gross weight as 1949 kg. Suspension system assembly with the tire is assumed as 50 kg, making the total mass for the left and right wheels as 100 kg for the front and rear axles. Therefore, m_3 is calculated as 1749 kg as the vehicle mass. Schock absorber stiffness and damping, tire stiffness coefficients and vehicle rotational moment of inertial are tunned manually in order to match the pitch angle simulation with the vehicle pitch angle measurements. Tunned parameters are listed in table 3. As shown in figure 6, pitch angle simulation matches well with the vehicle measurements.

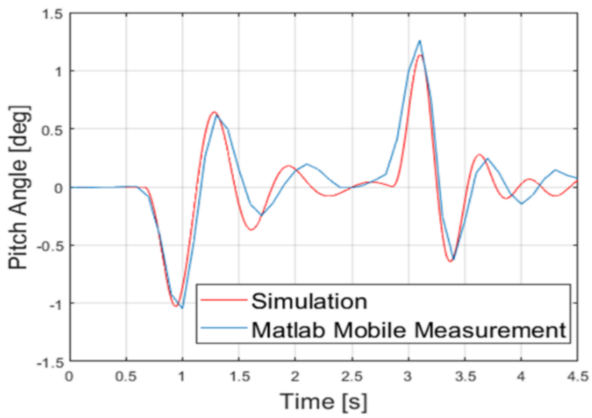


Figure 6: Validation of the HCM using Matlab mobile measurements.

Table 3: Vehicle specification.

Parameter	Value	Units
m_1	100	kg
m_2	100	kg
m_3	1749	kg
k_{fs}	100000	N/m
k_{rs}	200000	N/m
k_{ft} & k_{rt}	1000000	N/m
b_f & b_r	5000	Ns/m
I_3	5000	kg m^2
l_1 & l_2	1.4235	m

4. Optimization

A full factorial DOE with 4 input parameters spanning ±20% of the original values: kfs, krs, bf & br (Front and rear suspensions shock absorber stiffness and damping coefficients) is conducted and vehicle pitch angle and body vertical acceleration responses are captured. Maximum pitch angle and vertical acceleration values are used as the output parameters for the regression model using Minitab software. Response plots for the 4 inputs are depicted in figure 7. Optimizer in the regression module is used to minimize the pitch angle and maximum acceleration outputs. Original values and optimized values are listed in table 4. Comparison of pitch angle and vehicle vertical acceleration response during the simulation of the bump passing manoeuvre in MATLAB / Simulink for the original and optimized values are shown in figure 7. Results indicated that optimized values reduce the maximum pitch angle and vertical acceleration by 3.4% and 9.4% respectively. This trend can be clearly seen from the vehicle response plots in figure 7.

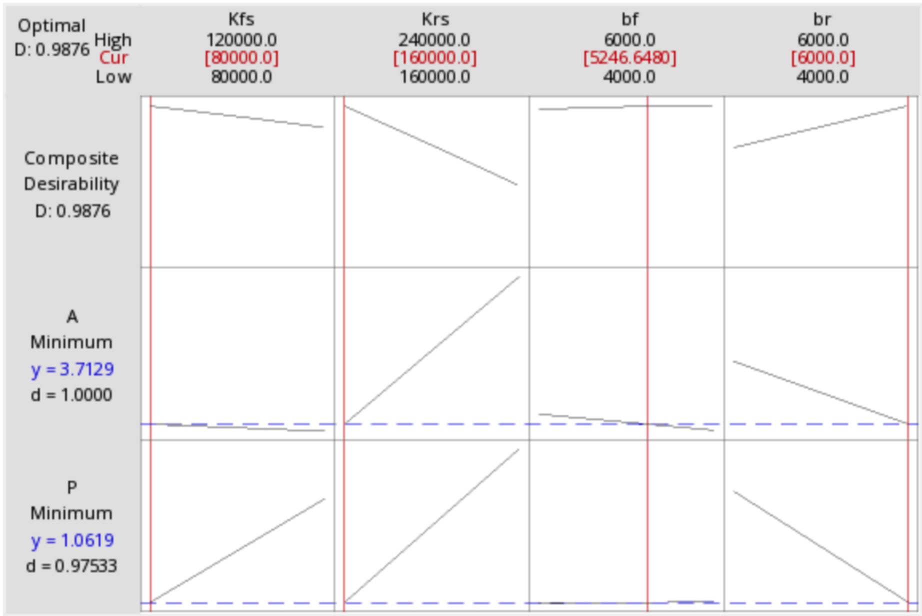
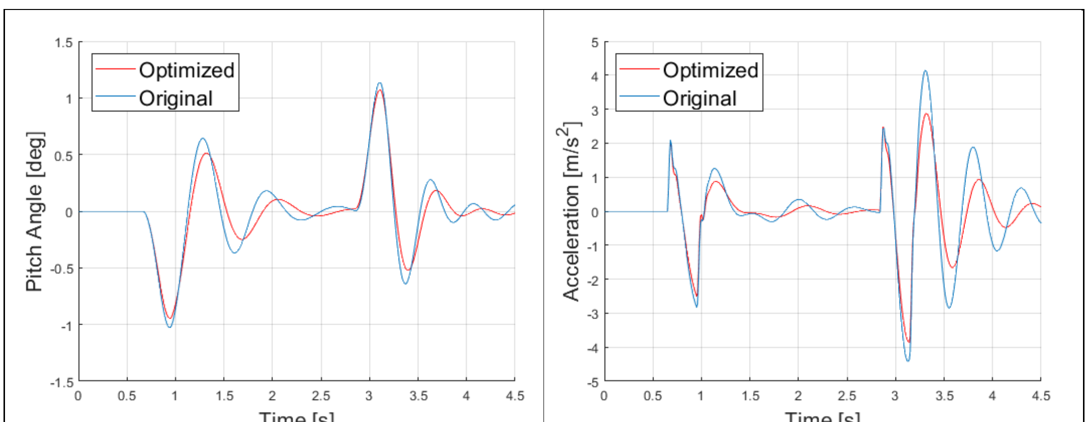


Figure 7: Response plots for the 4 inputs generated in Minitab software.

Table 4: Vehicle response summary for the original and optimized parameters.

	k_{fs}	k_{rs}	b_f	b_r	**Acceleration**	**Pitch**
Original	100000	200000	5000	5000	4.2628	1.108
Optimized	80000	160000	5246.65	6000	3.8602	1.070
% Change for the optimized values					-9.4%	-3.4%



a b

Figure 7: Comparison of pitch angle (a) and vehicle vertical acceleration (b) response for the original and optimized parameters.

5. Conclusion

Within the content of a capstone project for American University of Middle East – Kuwait for spring 2022 semester, analysis and optimization of a suspension system for a SUV is performed. A Matlab / Simulink HCM model with pitch angle and vertical acceleration simulation capability is developed and validated using literature data. Vehicle measurement for a bump passing maneuver is taken and used in order to tune the HCM. A full factorial DOE based optimization is performed spanning ± 20% of the original values for shock absorber stiffness and damping coefficients. Results indicates that considering only acceleration magnitude and pitch angle, using the values from the DOE optimization process improves driving comfort with 9.4% and 3.4% reductions respectively. For future work, the effect of optimized coefficients should be evaluated with respect to vehicle driving dynamics considering the roll over and weight transfer effect cornering maneuver as well.

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