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Improvement of vehicle ride comfort using genetic algorithm optimization and PI controller

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Abstract In this paper a MATLAB SIMULINK model of seven Degrees Of Freedom (DOF) full vehicle model is developed. Mathematical equations are obtained using Newton's second law and free body diagram concept. Validation of the SIMULINK model is obtained to ensure that the model is suitable for studying the ride comfort. A Genetic algorithm optimization technique is used to find the optimum values of spring stiffness and damping coefficient for front and rear passive suspension system of the seven DOF vehicle model at variable velocities which improve the performance of the suspension system of the vehicle. Also Proportional Integral (PI) controller is implemented to the model to study its effect on ride comfort. Comparison of the results for body acceleration and sprung mass displacement of the optimized data of suspension parameters and model with PI controller are illustrated. The results show that the optimized parameters and PI controller give significant improvements of the vehicle ride performance over the passive suspension system.

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1. Introduction

The major target of the vehicle suspension system is to enhance the ride comfort and to provide good ride handling capability. A vehicle suspension may be classified as passive suspension, semi active suspension and active suspension system [1]. A quality suspension must achieve a good behavior of the vehicle and a degree of comfort depending on the interaction with the uneven road surface [2]. Modeling of suspension systems is

very important as it can be used for studying the performance of the ride comfort and road handling of the vehicle. There are different methods of representation the model. State space modeling, transfer function and MATLAB Simulink are used to solve these models. There are different models used in this field, quarter car model [2–4], half car model [5–7] and full car model [8–11]. The performance of active suspension system using Linear Quadratic Regulator (LQR) and Proportional Integral Derivative (PID) techniques was compared with the passive suspension system. Simulation is based on the mathematical model by using MATLAB/SIMULINK software in [3]. Mathematical half-car model was created in Matlab/Simulink. Results from simulations of model with original and optimized suspension parameters were compared in [5]. Simulation of active half car suspension model was done using MATLAB,

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Nomenclature

a	distance from CG to front wheel, m	M_S	mass of vehicle, kg
b	distance from CG to rear wheel, m	M_{wl1}	mass of front left wheel, kg
C_{sl1}	damping coefficient of front left damper, N-s/m	M_{wl2}	mass of rear left wheel, kg
C_{sl2}	damping coefficient of rear left damper, N-s/m	M_{wr1}	mass of front right wheel, kg
C_{sr1}	damping coefficient of front right damper, N-s/m	M_{wr2}	mass of rear right wheel, kg
C_{sr2}	damping coefficient of rear right damper, N-s/m	Z_{cg}	displacement of CG of vehicle body, m
c	distance from CG to right wheel, m	Z_{rl1}	road input to front left wheel, m
d	distance from CG to left wheel, m	Z_{rl2}	road input to rear left wheel, m
I_{xx}	moment of Inertia about X - X axis, kg-m ²	Z_{rr1}	road input to front right wheel, m
I_{yy}	moment of Inertia about Y - Y axis, kg-m ²	Z_{rr2}	road input to rear right wheel, m
K_{sl1}	spring stiffness of front left suspension, N/m	Z_{wl1}	displacement of front left wheel, m
K_{sl2}	spring stiffness of rear left suspension, N/m	Z_{wl2}	displacement of rear left wheel, m
K_{sr1}	spring stiffness of front right suspension, N/m	Z_{wr1}	displacement of front right wheel, m
K_{sr2}	spring stiffness of rear right suspension, N/m	Z_{wr2}	displacement of rear right wheel, m
K_{wl1}	spring stiffness of front left tire, N/m	θ	pitch angle of the body at CG, degree
K_{wl2}	spring stiffness of rear left tire, N/m	Φ	roll angle of the body at CG, degree
K_{wr1}	spring stiffness of front right tire, N/m		
K_{wr2}	spring stiffness of rear right tire, N/m		

Simulink and a comparison between controllers such as classical PID controller; self-tunable fuzzy inference system (STFIS) controller and the passive system controller were founded in [6]. The full car suspensions for passive and active one with PID or Fuzzy controls were simulated using MATLAB/SIMULINK at [12]. The results indicated that the active with PID and fuzzy controls suspension systems were better than the passive system. An active suspension for the Quarter/Half/Full car model of car was developed by [13] to improve its performance by using a PID controller. Optimization was also applied to one of the these types as in [14] who made an investigation to determine the spring and damper settings that will ensure optimal ride comfort of vehicle for different speeds using Design of Experiment Method (DOE) for seven degrees of freedom full car model.

This paper is organized as follows; Full car model is obtained in Section 2. Mathematical equations are presented in Section 3. In Section 4 the MATLAB, Simulink model is built. The road profiles used in this paper are given on Section 5. Section 6 shows the optimization and controller. Results and discussion of validation are obtained in Section 7. Section 8 presents the conclusions of the paper. An Appendix A is given at the end of the paper.

2. Full car model

Fig. 1 shows a full car model with seven degrees of freedom system considered for analysis [11]. It is consisting of sprung mass, M_s referring to the part of the car that is supported on springs and un-sprung mass which refers to the mass of the wheel assembly. The suspensions between the sprung mass and un-sprung masses are modeled as linear viscous dampers and spring components. The tire has been replaced with its equivalent stiffness and tire damping is neglected as it's influence on tire behavior is negligible [9,11 and 12].

3. Mathematical modeling

3.1. Mathematical equations

Using the Newton's second law of motion and free body diagram concept; the following seven equations of motion are derived.

For vehicle body bounce motion (Sprung Mass):

$$\begin{aligned}
 M_s \ddot{Z}_{cg} = & (-K_{sr1} - K_{sl1} - K_{sr2} - K_{sl2})Z_{cg} \\
 & + (-C_{sr1} - C_{sl1} - C_{sr2} - C_{sl2})\dot{Z}_{cg} \\
 & + (K_{sr1}a + K_{sl1}a - K_{sr2}b - K_{sl2}b)\dot{\theta} \\
 & + (C_{sr1}a + C_{sl1}a - C_{sr2}b - C_{sl2}b)\dot{\theta} \\
 & + (K_{sr1}c - K_{sl1}d + K_{sr2}c - K_{sl2}d)\dot{\phi} \\
 & + (C_{sr1}c - C_{sl1}d + C_{sr2}c - C_{sl2}d)\dot{\phi} \\
 & + K_{sr1}Z_{wr1} + K_{sl1}Z_{wl1} + K_{sr2}Z_{wr2} + K_{sl2}Z_{wl2} \\
 & + C_{sr1}\dot{Z}_{wr1} + C_{sl1}\dot{Z}_{wl1} + C_{sr2}\dot{Z}_{wr2} + C_{sl2}\dot{Z}_{wl2}
 \end{aligned} \quad (1)$$

For vehicle body pitching motion (Sprung Mass):

$$\begin{aligned}
 I_{yy} \ddot{\theta} = & (K_{sr1}a + K_{sl1}a - K_{sr2}b - K_{sl2}b)Z_{cg} \\
 & + (C_{sr1}a + C_{sl1}a - C_{sr2}b - C_{sl2}b)\dot{Z}_{cg} \\
 & + (-K_{sr1}a^2 - K_{sl1}a^2 - K_{sr2}b^2 - K_{sl2}b^2)\dot{\theta} \\
 & + (C_{sr1}a^2 - C_{sl1}a^2 - C_{sr2}b^2 - C_{sl2}b^2)\dot{\theta} \\
 & + (-K_{sr1}ac + K_{sl1}ad + K_{sr2}bc - K_{sl2}bd)\dot{\phi} \\
 & + (-C_{sr1}ac + C_{sl1}ad + C_{sr2}bc - C_{sl2}bd)\dot{\phi} - K_{sr1}aZ_{wr1} \\
 & - K_{sl1}aZ_{wl1} + K_{sr2}bZ_{wr2} + K_{sl2}bZ_{wl2} - C_{sr1}a\dot{Z}_{wr1} \\
 & - C_{sl1}a\dot{Z}_{wl1} + C_{sr2}b\dot{Z}_{wr2} + C_{sl2}b\dot{Z}_{wl2}
 \end{aligned} \quad (2)$$

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