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Original Research Article

Suspension damping optimization using genetic algorithms

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Ali Hemati[1], Alireza Shooshtari[2]

[1 *]Bu Ali Sina University

[2]Department of Mechanical Engineering, Bu Ali Sina University

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* Corresponding author
A_hemati65@yahoo.com

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ABSTRACT

Vehicle ride comfort is a function of the frequency of transmitting vibrations to passengers from road irregularities. The objective of this research is to provide a systematic method into the requirement of the passive suspension based on genetic algorithm (GA). Objective function of GA method is driver’s seat acceleration to provide driver ride comfort. The passive suspension damping coefficient and driver’s seat suspension damping coefficient have been optimized. In this paper, a vehicle with eight degrees of freedom has been simulated to achieve driver’s seat comfort. Analytical results explained by mathematical equations presented in the graphical form have been included.

Keywords: Genetic Algorithm, Passive suspension, Full vehicle model, Ride comfort, Objective function

1. Introduction

Vibration intensity is major importance which has a direct relation with vehicle speed; so often an increase of speed is unbearable. That is why the operators feel discomfort in driving the larger tractors at high speed and full nominal power [1]. The characteristics of air springs, such as the effects of bellows and those of heat transfer on spring constant and damping factor, have been studied [2]. The application of an additional air reservoir makes the stiffness characteristics of the air spring softer and in consequence decreases the natural frequency of the system. [3] Present the design and performance of a vibration isolation and suppression system using a novel hybrid actuator concept to provide both passive isolation and active damping. According to [4] the particular on-road usage of SUV’s makes these vehicles more prone to accidents. [5] Proposed a vibration isolation system using a conventional spring connected in parallel with a fully passive, negative-stiffness-mechanism to produce a reduced value of the supporting stiffness. Possibility of using slow active suspension control to reduce the body roll and thus reduce the rollover propensity is presented [6]. Guclu has been presented a vibration control performance of a seat suspension system of non-linear full vehicle model using fuzzy logic controller [7]. Vibration control performance of a semi-active electro rheological seat suspension system using a robust sliding mode controller has been searched by Huang and Chen [8]. The control system automatically switches between “ride comfort” and “handling” modes of evaluating several vehicle parameters. Ride height control is also built into the system to level the vehicle when the suspension struts are loaded with gas [9]. In [10] a general procedure is presented for the analytical synthesis of an optimal vibration isolation system of a human body including its sensitivity to vibration. A genetic algorithm method is applied to the optimization problem of a linear 1-DOF vibration isolator mount and the method is extended to the optimization of a linear quarter car suspension model [11]. For efficiency, genetic algorithm is employed to search for the parameters like damping ratio and spring constant to achieve an optimum trade off among ride comfort, handling quality, and suspension stroke simultaneously for random input. A dynamic model of an on-highway truck seat is proposed by Hassanin et al. [12]. In this paper, a full vehicle model with passive suspension has been optimized. The main objective of this study is to prepare a ride comfort improvement. The GA objective function is driver's seat acceleration. In addition, using GA method the suspension parameters have been optimized for driver's seat stability.

2. Mathematical Modeling

Dynamic simulation is created based on the dynamic mathematical model. This model consists of the K_{tf} and K_{tr} represent the front tires stiffness and the rear tires stiffness, m_f and m_r are the front and the rear suspension mass, respectively. C_f and C_r are damping coefficients of the front and the rear suspension, respectively. Figure 1 shows the full vehicle model.

The full vehicle model equations of motion consist of body, driver seat and suspension motion; Vertical motion (2), pitching motion (3), roll motion (4), front left suspension motion (5), front right motion (6), rear left suspension motion (7), rear right motion (8) and driver's seat motion (9) is shown [14];

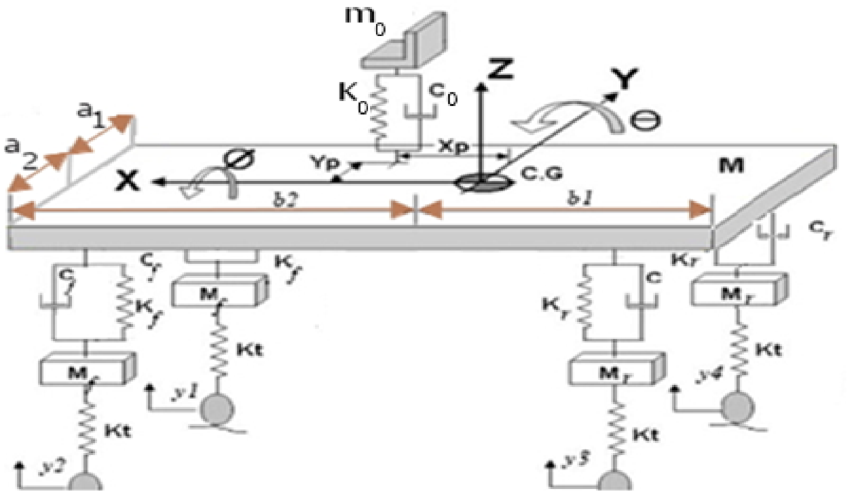

Figure 1. Full model vehicle

$$m[\ddot{x}] \quad C[\dot{x}] \quad K[x]=f \tag{1}$$

$$m\ddot{x} \quad c_f \, \dot{x} \quad \dot{x}_1 \quad a_1 \quad b_1 , \quad c_f \, \dot{x} \quad \dot{x}_2 \quad a_1 \quad b_2 , \quad c_r \, \dot{x} \quad \dot{x}_3 \quad a_2 \quad b_1 , \tag{2}$$

$$c_r \big(\dot{x} \quad \dot{x}_4 \quad a_2 \quad b_2 , \quad c_0 \, \dot{x} \quad \dot{x}_0 \quad a_2 \quad b_2 , \quad k_f \, x \quad x_1 - a_1\theta + b_1\varphi\big) + k_f\left(x - x_2 - a_1\theta - b_2\varphi\right) +$$

$$k_r\left(x - x_3 + a_2\theta + b_1\varphi\right) + k_r\left(x - x_4 + a_2\theta - b_2\varphi\right) + k_0\left(x - x_0 + a_2\theta - b_2\varphi\right) = f_f + f_f + f_r + f_r$$

$$I_Y \ddot{} \quad c_f.a_1 \, \dot{x} \quad \dot{x}_1 \quad a_1 \quad b_1 , \quad c_f.a_1 \, \dot{x} \quad \dot{x}_2 \quad a_1 \quad b_2 , \quad c_r.a_2 \, \dot{x} \quad \dot{x}_3 \quad a_2 \quad b_1 , \tag{3}$$

$$c_r.a_2\big(\dot{x} \quad \dot{x}_4 \quad a_2 \quad b_2 , \quad c_0.a_2 \, \dot{x} \quad \dot{x}_0 \quad a_2 \quad b_2 , \quad k_f.a_1 \, x \quad x_2 - a_1\theta - b_2\varphi\big) +$$

$$k_r.a_2\left(x - x_3 + a_2\theta + b_1\varphi\right) + k_0.a_2\left(x - x_0 + a_2\theta - b_2\varphi\right) = -f_f.a_1 - f_f.a_1 + f_r.a_2 + f_r.a_2$$

$$I_x \ddot{} \quad c_f.b_1 \, \dot{x} \quad \dot{x}_1 \quad a_1 \quad b_1 , \quad c_f.b_1 \, \dot{x} \quad \dot{x}_1 \quad a_1 \quad b_1 , \quad c_f.b_2 \, \dot{x} \quad \dot{x}_2 \quad a_1 \quad b_2 , \tag{4}$$

$$c_r.b_1\left(\dot{x}-\dot{x}_3+a_2\dot{\theta}+b_1\dot{\varphi}\right)-c_f.b_2\left(\dot{x}-\dot{x}_4+a_2\dot{\theta}-b_2\dot{\varphi}\right)-c_0.b_2\left(\dot{x}-\dot{x}_0+a_2\dot{\theta}-b_2\dot{\varphi}\right)+$$

$$k_f.b_1\left(x-x_1-a_1\theta+b_1\varphi\right)-k_f.b_2\left(x-x_2-a_1\theta-b_2\varphi\right)+k_r.b_1\left(x-x_3+a_2\theta+b_1\varphi\right)-$$

$$k_r.b_2\left(x-x_4+a_2\theta-b_2\varphi\right)-k_0.b_2\left(x-x_0+a_2\theta-b_2\varphi\right)=f_f.b_1-f_f.b_2+f_r.b_1-f_r.b_2$$

$$m_1\ddot{x}_1-c_f\left(\dot{x}-\dot{x}_1-a_1\dot{\theta}+b_1\dot{\varphi}\right)-k_f\left(x-x_1-a_1\theta+b_1\varphi\right)+k_{tf}\left(y_1-x_1\right)=f_1 \tag{5}$$

$$m_2\ddot{x}_2-c_f\left(\dot{x}-\dot{x}_2-a_1\dot{\theta}-b_2\dot{\varphi}\right)-k_f\left(x-x_2-a_1\theta-b_2\varphi\right)+k_{tf}\left(y_2-x_2\right)=f_2 \tag{6}$$

$$m_3\ddot{x}_3-c_r\left(\dot{x}-\dot{x}_3+a_2\dot{\theta}+b_1\dot{\varphi}\right)-k_r\left(x-x_3+a_2\theta+b_1\varphi\right)+k_{tr}\left(y_3-x_3\right)=f_3 \tag{7}$$

$$m_4\ddot{x}_4-c_r\left(\dot{x}-\dot{x}_4+a_2\dot{\theta}-b_2\dot{\varphi}\right)-k_r\left(x-x_4+a_2\theta-b_2\varphi\right)+k_{tr}\left(y_4-x_4\right)=f_4 \tag{8}$$

$$m_0\ddot{x}_0-c_0\left(\dot{x}-\dot{x}_0+a_2\dot{\theta}-b_2\dot{\varphi}\right)-k_0\left(x-x_0+a_2\theta-b_2\varphi\right)=0 \tag{9}$$

The vehicle parameters are given in Table 2.

Table 1. Full model vehicle parameter

Parameter	Symbol	Value
Mass of the vehicle body	m	840 kg
Un sprung mass in front left/right side	m_f	53 kg
Un sprung mass at rear side	m_r	76 kg
Distance between front wheel and full-vehicle model at its mass center	a_1	1.4 m
Distance between rear wheel and full-vehicle model at its mass center	a_2	1.47 m
Spring constant of front suspension	k_f	10000 N/m
Spring constant of rear suspension	k_r	10000 N/m
Spring constant of front tire	k_{tf}	200000 N/m
Spring constant of rear tire	k_{tr}	200000 N/m
Fixed damping coefficient of the front suspension damper	c_f	2000 N.s/m
Fixed damping coefficient of the rear suspension damper	c_r	2000 N.s/m
Distance between front and rear right side wheel and full-vehicle model at its mass center	b_1	0.7 m
The distance between front and rear left side wheel and full-vehicle model at its mass center	b_2	0.75 m
Roll moment of inertia of the vehicle body	I_X	820 kgm2
Pitch moment of inertia of the vehicle body	I_Y	1100 kgm2
Spring constant of Driver's seat	K_0	1200 N/m
Fixed damping coefficient of the constancy of the Driver's seat	C_0	400N. s/m
Mass of the Driver's seat	m_0	80 kg

3. Genetic Algorithm

The MATLAB/SIMULINK toolbox has been used to optimize the passive suspension coefficients. The genetic algorithms for determining the optimal suspension coefficients is used. When the GA method is used, it is necessary to determine an objective function. The ride comfort is improved using driver's seat acceleration as an objective function. Using the GA method, the optimal parameters for passive suspension are obtained. The best values for vehicle and driver's seat damping coefficients determined using the GA method by search and iteration. Driver's seat acceleration is objective function of the genetic algorithm equation. Objective function of the GA method shown in equation (10);

$$J=\sum_{i=1}^{1000}\left|\ddot{x}_0=\frac{c_0}{m_0}\left(\dot{x}-\dot{x}_0+a_2\dot{\theta}-b_2\dot{\varphi}\right)+\frac{k_0}{m_0}\left(x-x_0+a_2\theta-b_2\varphi\right)\right| \tag{10}$$

The GA research method to determining passive damping coefficients shown on Figure 2;

Minimizing the (10) applied the ride comfort; using GA, passive damping parameters have been optimized. When suspension damping coefficients have different values, consist of suspension's damping with lower bound and upper bound, the vehicle vibration has a different bandwidth, according to the time domain simulation results. The lower, upper and optimal bound of damping coefficients and damping shown in Table 2.

Figure 3 shows the GA method's flowchart to optimization suspension coefficients to achieve best driver's seat behavior and to improve ride comfort.

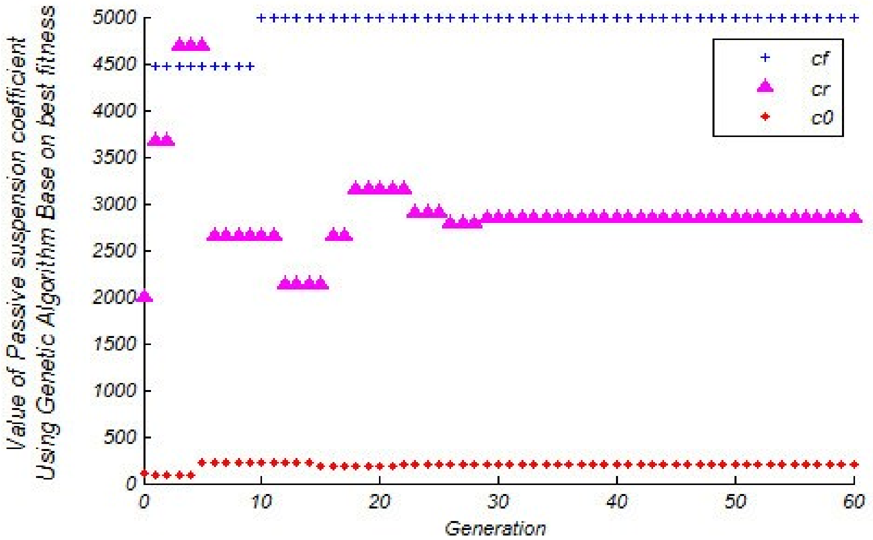

Figure 2. GA optimization based on objective function

Table 2. Variable suspension coefficient

Parameters	Lower bound	Genetic algorithms	Upper bound
C_f, N.s/m	800	4850	5000
C_r, N.s/m	800	2850	5000
C_0, N.s/m	50	205	1000

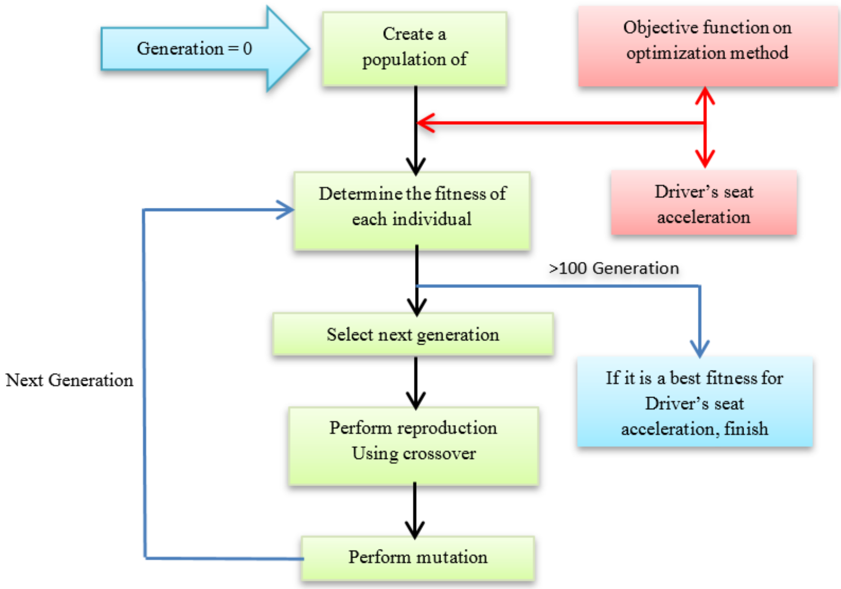

Figure 3. The GA method flowchart

4. Simulation Results

The optimization of damping coefficients helps to have a ride comfort. Figure 4. Shows the vehicle input for simulation. The time of the simulation is ten seconds. Vehicle's running speed is 80 km/h in this simulation so its simulation can compare to [13] for passive Suspension simulation output for a same speed. All of the unfavorable vibrations of the full vehicle model have been simulation based on driver's seat acceleration optimization, here. Figure 5, Figure 6, Figure 7, Figure 8, Figure 9 and Figure 10 are roll angle deflection, body's vertical acceleration, body's pitching acceleration, body's roll acceleration, driver's seat acceleration and flowchart of suspension that designing in Matlab/Simulink toolbox, respectively.

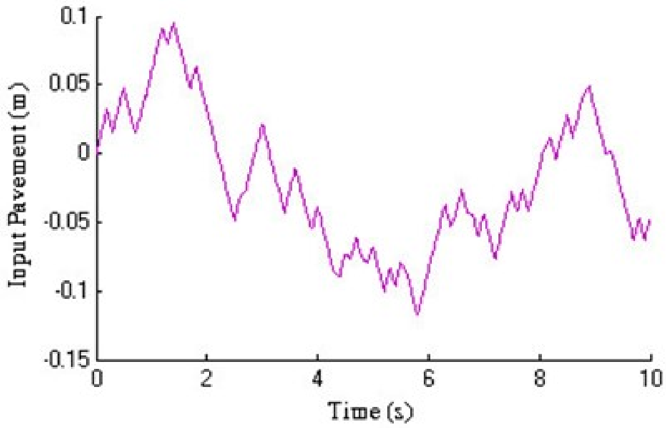

Figure 4. Input Pavement simulation curve

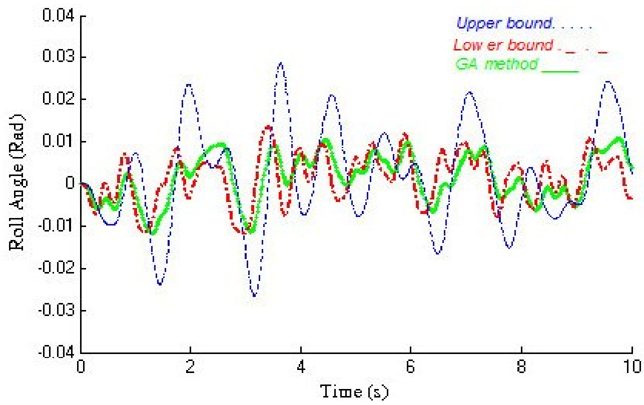

Figure 5. Roll angle's simulation curve

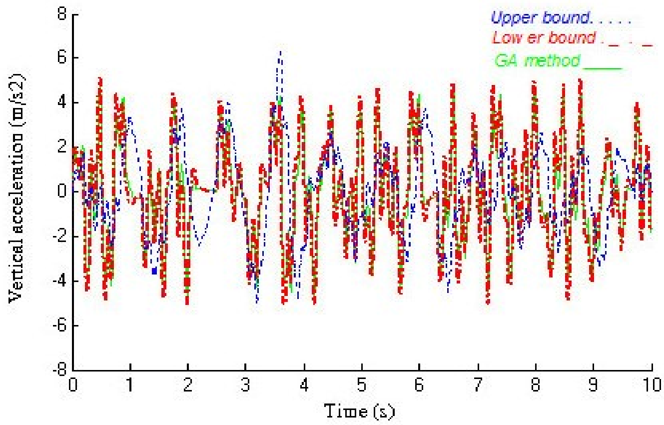

Figure 6. Body's Vertical acceleration's simulation curve

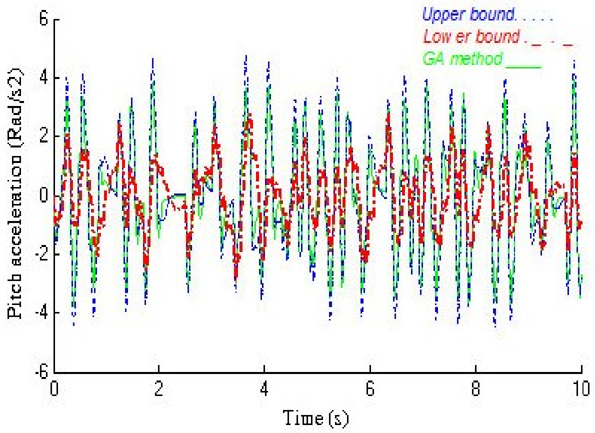

Figure 7. Body's pitching acceleration's simulation curve

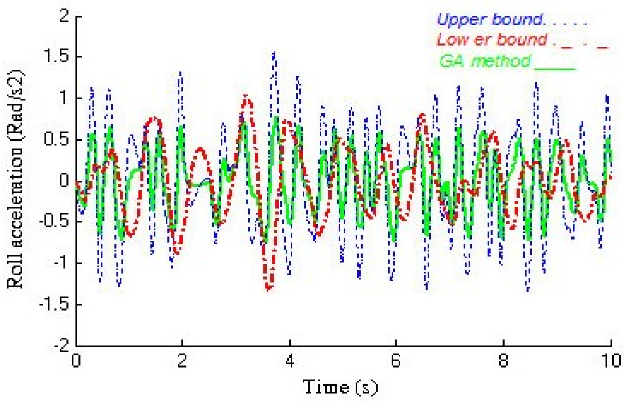

Figure 8. Body’s Roll acceleration simulation curve

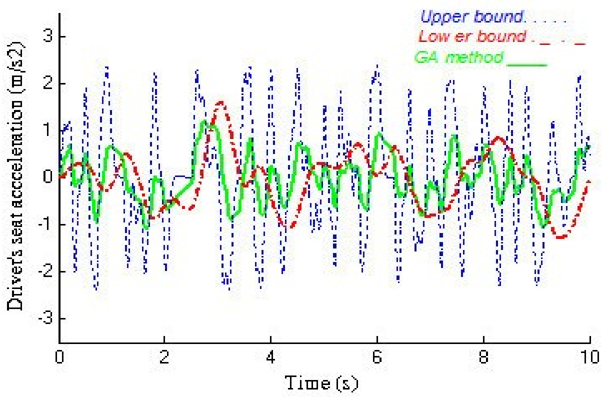

Figure 9. Driver’s Seat acceleration’s simulation curve

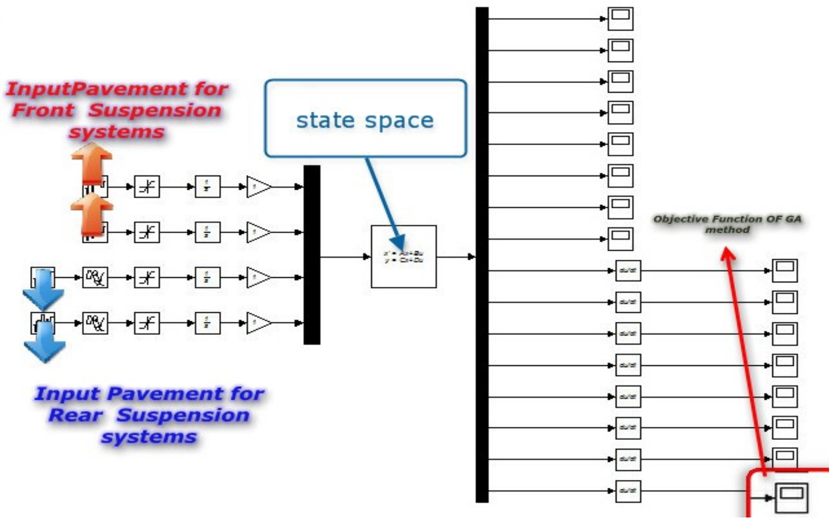

Figure 10. Flowchart of full model vehicle suspension

According to the time domain simulation results, when suspension damping coefficients have a different value, the body and driver’s seat performance have a different vibration bandwidth. When the damping coefficients changing between 800 N.s/m-5000 N.s/m, the vehicle behavior and ride comfort have a different bandwidth. When the front and rear suspension damping coefficients become higher than 800 N.s/m, the driver’s seat acceleration value become better, and the ride comfort become better, when the front suspension damping coefficients go beyond 4850 N.s/m, its value becomes higher, the smooth gets worse. When the rear suspension damping coefficients goes beyond 2850 N.s/m, its value becomes higher, the smooth gets worse. When the driver’s seat suspension damping coefficients become higher than 50 N.s/m, the driver’s seat acceleration value become better. When the driver’s seat suspension damping coefficients go beyond 205N.s/m, its value becomes higher, the ride comfort gets worse. When the objective function of GA method is driver's seat acceleration, the main objective of optimization is determining of best damping coefficients to achieve the good ride comfort. The simulation and optimization result shown in Table 3.

Table 3. Value of Simulation result

Performances		Lower Bound	GA method	Upper Bound
Body Vertical Acceleration	m/s^2	5	4	6
Body Pitch Acceleration	Rad/s^2	2.5	3	4
Body Roll Acceleration	Rad/s^2	1.5	0.6	1.5
Driver's Seat Vertical Acceleration	m/s^2	1.8	1	2.5
Roll Angle	Rad	0.015	0.01	0.03

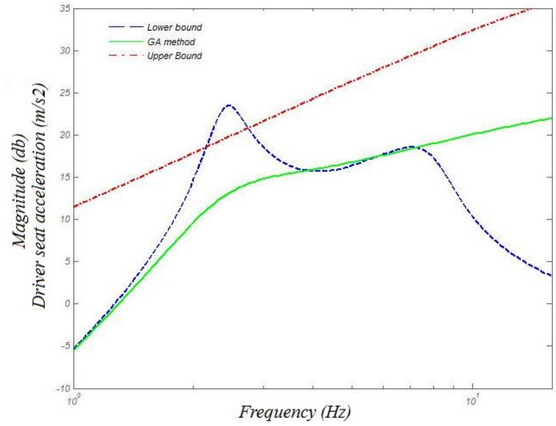

Figure 11. Driver's seat acceleration frequency response

Figure 11 shows compare between the driver's seat acceleration using GA, upper bound and lower bound of passive suspension damping coefficients in frequency response.

5. Conclusion

In this paper, the genetic algorithm method is applied to optimization problem of suspension with linear eight-degree-of-freedom (8-DOF). Damping coefficients are optimized using genetic algorithms for best result of the driver's seat vibrations. Although the suspension systems are linear, it is difficult to find such optimal relation analytically. Damping coefficients have been determined for minimum bandwidth of driver's seat acceleration, here. Optimization of objective function can reduce another unfavorable vibration to achieve an acceptable bandwidth, in the general case.

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