Control of active car suspension system based on 2PID and PSO Optimization

Hajer Emad[1] University of Baghdad Al-Khwarizmi College of Engineering Mechatronics Engineering Department. E-mail: enghajer9796@gmail.com

Ali Mary[1] University of Baghdad Al-Khwarizmi College of Engineering Mechatronics Engineering Department. E-mail: Alimary76@kecbu.uobaghdad.edu.iq

Malik Mohammed Ali[1] University of Baghdad Al-Khwarizmi College of Engineering Mechatronics Engineering Department.

ABSTRACT

Active suspension systems are essential to improve vehicle performance and ride comfort under various road vibrations. Active suspension systems proved a better option compared to a passive suspension system. It has an actuator that produces a force that reduces the turbulence caused by road interference. The suspension in this research is controlled by a 2Proportional-Integral-Derivative (PID) and optimize the gains using the Particle Swarm Optimization (PSO) method. Three forms of road disturbances are studied one is sine and step input and a single and two bumps. Simulation results demonstrated the advantage of 2PID controllers-based PSO algorithm in terms of improving the dynamic performance of the active suspension system regarding to body acceleration position and deflection and provide good tire-road contact.

Keywords: Active suspension, PSO, PID controller, quarter car

1.Introduction

Modern trends in technology are drastically modified the car industry. Suspension plays an important role in rising ride comfort and road-handling ability. THE most perform of A suspension system is to reduce the vertical acceleration of the vehicle body that's transmitted to the occupants so as to realize ride comfort and conjointly keep the tires to bear with road throughout disturbance conditions of the vehicle. The vehicle suspension system will be controlled by exploitation of three basic approaches that are classified as passive, semi-active and active suspensions. Active suspension system employs actuators to supply desired management action between the vehicle sprung body and wheel shaft to reduce the impact of road disturbances on the passengers and therefore they're a lot of economical compared to passive suspension system and semi-active suspension systems [1-2]. Many researchers, withinside the final decade, have distinct strategies including implemented optimal control [10], H∞ based controllers [11-12],nonlinear adaptive control [13], adaptive sliding control with self-tuning fuzzy compensation [14], fuzzy logic [15-16], synthetic neural network [17-18], backstepping control [19], intelligent controllers [20], model reference adaptive controllers [21], and proportional-integral-derivative (PID) controllers [22] to the vibration control problem of vehicle suspension system.

The author [3] presents a PID controller to change the damper of the vehicle suspension system. The PID controller keeps the car body stable by adjust coefficient. The improvement PI controller for the quarter-car model was present by [4]. The gain of the active suspension system is tunned by using PSO to improve body acceleration and tire deviation. A mathematical model for active and passive suspension system for the quarter car model is presented by [5] PID controller is used to compare between passive and active suspension system. The authors [6] is present a PID controller and PSO is used to design active quarter car suspension model. The spring mass displacement was observed to be reduced which indicates a good improvement in the ride comfort and sprung mass. The authors [7] planned a strategy utilizing the PSO calculation to tackle PID issues and the difficulties of deciding the three boundaries of the PID module were settled well. The choosing capability and control execution were flawless. further developing to the three exhibition pointers of the vehicle suspension. [8] They fostered the quarter vehicle and street knock models allocated to MATLAB Simulink for the powerful investigation of the vehicle's suspension framework, and through the consequences of Simulink, they saw that the presentation of the PSO based P-I-D regulator was greatly improved contrasted with the uninvolved framework and the Ziegler-Nichols tuning technique depends on the dynamic framework as far as position, speed and speed increasing and to furnish great contact with the street tires during tempestuous rough terrain conditions. [9] they concluded that a functioning suspension with further developing of the spring firmness, damping coefficients, and PID gains gave the best tenant solace and street taking care of equilibrium and the hereditary calculation can be utilized to make viable streamlining.

In this research particle swarm optimization (PSO) based on optimum 2PID controllers are suggestion to achieve the optimal damping force as resultant with enhancement of passenger comfort, riding stability and road handling for the vehicle's active suspension system. The modeling, simulation and optimization were done with the help of MATLAB SIMULINK.

2.Modeling of Active Suspension System of Quarter Vehicle Model

The active suspension system is implemented using the quarter car model presented in Fig. 1 with the highlighted elements. [6]

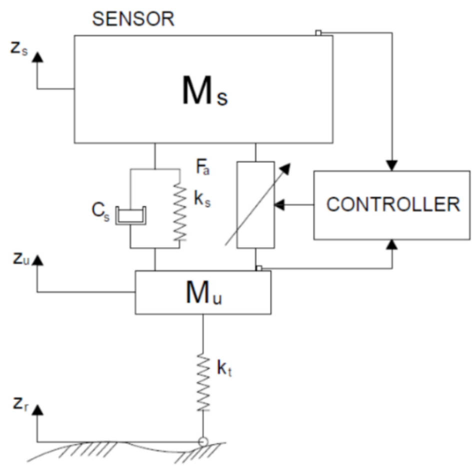


Figure 1: Graphic of a quarter car model

$M_{S:}$ car chassis mass.

M_u: wheel mass.
K_s: Spring Stiffness.
K_t: Tire Stiffness corresponded to spring
C_s: Dumper force.
Fa: Actuator force.
Z_r : Road profile
Z_s : Sprung Mass Displacement
Z_u : Unsprung Mass Displacement
$\dot{Z}_s$: Sprung Mass Velocity
$\dot{Z}_u$: Unsprung Mass Velocity

The parameter of Quarter vehicle model is shown in table 1.

Table 1. Values of Parameters in Quarter Car Model [2]

Sprung mass	MS = 300Kg
Un sprung mass	Mu = 60Kg
Suspension damping	Cd=1000Ns/m
Suspension stiffness	Ks 16000N/m
Tire stiffness	Kt=190000N/m

Quarter car model active suspension system can be written based on Newton's laws. [24], [25], [26] equations:

$$M_s\ddot{Z}_s = -C_s(\dot{Z}_s - \dot{Z}_u) - k_s(Z_s - Z_u) + F_a$$
$$M_u\ddot{Z}_u = -C_s(\dot{Z}_u - \dot{Z}_s) - k_s(Z_u - Z_s) - k_t(Z_u - Z_r) - F_a \quad (1)$$

Where, A = State space matrix, B = Input matrix, C = Output matrix, D = Direct transmission matrix, X = State variables, U = Input of system, Y = Output of system.

It is critical to observe that State Space Model (SSM) refers to a category of model that describes the dependence among the latent state variable and the found dimension. The state or the dimension may be both non-stop or discrete. In control engineering, a state-space illustration is a mathematical model of a physical machine as a fixed of input, output and state variables associated through first-order differential equations or distinction equations [27]. The state variables are selected ($X(t)$) primarily based totally at the preferred outputs in Eq. (2).

$$x_1 = Z_s$$
$$x_2 = \dot{x_1} = \dot{Z}_s$$
$$x_3 = Z_u \quad (2)$$
$$x_4 = \dot{x_3} = Z_1$$
$$x_5 = Z_r$$

The left-hand aspect of the primary state-space equation is acquired through taking the primary by-product of the state variable vector to be

$$X(t)= ([x_1\ x_2\ x_3\ x_4\ x_5]^{T}) \Rightarrow X(t)= [\dot{x_1}\dot{x_2}\dot{x_3}\dot{x_4}\dot{x_5}]^{T} \quad (3)$$

To get the desired values of first derivatives of state variables, substitute the state variables in the equations of motion:

$$M_s\dot{x_2} = -C_s(x_2 - x_4) - k_s(x_1 - x_3) + F_a$$
$$\Rightarrow \dot{x_2} = \frac{1}{M_s}[-C_s(x_2 - x_4) - k_s(x_1 - x_3) + F_a]$$
$$M_u\dot{x_4} = C_s(x_2 - x_4) + k_s(x_1 - x_3) - k_t(x_3 - x_5) \quad (4)$$
$$\Rightarrow \dot{x_4} = \frac{1}{M_u}[C_S(x_2 - x_4) + k_S(x_1 - x_3) - k_t(x_3 - x_5) - F_a]$$

The first derivatives of the state variables can be represented by rearranging the previous equations.

$$\dot{x_1} = x_2$$
$$\dot{x_2} = -\frac{k_s}{M_s}x_1 - \frac{C_s}{M_s}x_2 + \frac{k_s}{M_s}x_3 + \frac{C_s}{M_s}x_4 + \frac{1}{M_s}F_a$$
$$\dot{x_3} = x_4 \quad (5)$$
$$\dot{x_4} = \frac{k_s}{M_u}x_1 + \frac{C_s}{M_u}x_2 - \frac{(k_s+k_t)}{M_u}x_3 - \frac{C_s}{M_u}x_4 + \frac{k_t}{M_u}x_5 - \frac{1}{M_u}F_a$$
$$\dot{x_5} = \dot{Z}_r$$

The following state-space formulation is produced by expressing the previous equations in matrix form.

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\\ \dot{x}_5\end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0\\ -\frac{k_s}{M_s} & -\frac{C_s}{M_s} & \frac{k_s}{M_s} & \frac{C_s}{M_s} & 0\\ 0 & 0 & 0 & 1 & 0\\ \frac{k_s}{M_u} & \frac{C_s}{M_u} & -\frac{(k_s+k_t)}{M_u} & -\frac{C_s}{M_u} & \frac{k_t}{M_u}\\ 0 & 0 & 0 & 0 & 0\end{bmatrix}\begin{bmatrix}x_1\\ \frac{x_1}{x_2}\\ x_3\\ x_4\\ x_5\end{bmatrix} + \begin{bmatrix}0 & 0\\ 0 & \frac{1}{M_s}\\ 0 & 0\\ 0 & -\frac{1}{M_u}\\ 1 & 0\end{bmatrix}\begin{bmatrix}\dot{Z}_r\\ F_a\end{bmatrix} \quad (6)$$

It's important to note that the input is $\dot{Z}_r$, while the road profile is Z_r, as can be shown by recognizing the input.

3. Modeling of Road Profiles

Different types of road profiles are taken into consideration in order to explore the suspension dynamics in terms of ride comfort and handling capability of the vehicle. In general, road disturbances that last for a short time and have a strong impact have a negative impact on passenger ride comfort and may cause the road surface to lose its hold.

Many of the road disturbances are caused by bumpy roads. The road disturbances are represented with basic sinusoidal functions for the dynamic response study of the vehicle suspension system on rough roads. In this study, road disturbances with one and two bumps are evaluated for the car suspension system's performance examination [8].

3.1 Road disturbance with one bump (*Rd1*)

Following equation can be used to simulate a single bump in the road.

$$R_{d1}(t) = \begin{cases} \frac{A}{2}\left(1 - \cos\, 2\pi\left(\frac{t}{T_{b1}}\right)\right) & \text{for } T_{b1} \le t \le 2T_{b1} \\ 0 & Otherwise \end{cases} \quad (7)$$

A is the height of the bump, and Tb1 is the duration of the bump in Eq. (7), The ratio of bump length (L) to vehicle velocity (V) determines the size of the bump's duration. The road bump disturbance is thought to be formed between Tb1 and 2Tb1 seconds, with a plain road profile assumed for the remainder of the time [8].

3.2 Road disturbance with two bumps (*Rd2*)

The road disturbance with two bumps of different magnitudes is modeled using the following equation.

$$R_{d2}(t) = \begin{cases} \frac{A}{2}\left(1-\cos\left(2\pi\left(\frac{t}{T_{b1}}\right)\right)\right), \text{for } T_{b1} \le t \le 2T_{b1} \\ \frac{B}{2}\left(1-\cos\left(2\pi\left(\frac{t}{T_{b2}}\right)\right)\right), \text{for } 8T_{b2} \le t \le 9T_{b2} \\ 0 \text{ Otherwise} \end{cases} \quad (8)$$

In the Eq. (8) it is considered that the height of the initial road bump disturbance is A during the first
disturbance time interval between Tb1 and 2Tb1 seconds and the second bump disturbance of height
B is considered during the time interval between 8Tb2 and 9Tb2 seconds [8].

4. Simulation

The current study considers the minimizing of vehicle body acceleration using a PSO-based 2PID controllers and inserts it into the Simulink model. For the purpose of analyzing the performance of active suspension systems, MATLAB Simulink models for road disturbances have been built.
Simulink models for two types of road disturbances are constructing in this study one with a single bump and the other with two bumps According to Eq. (7) and Eq. (8) in the previous section [8].

4.1 *Simulink model of road disturbance with one Bump*

Figure 2 shows a Simulink model of a road disturbance with a single bump built from Eq. (7).

model

with

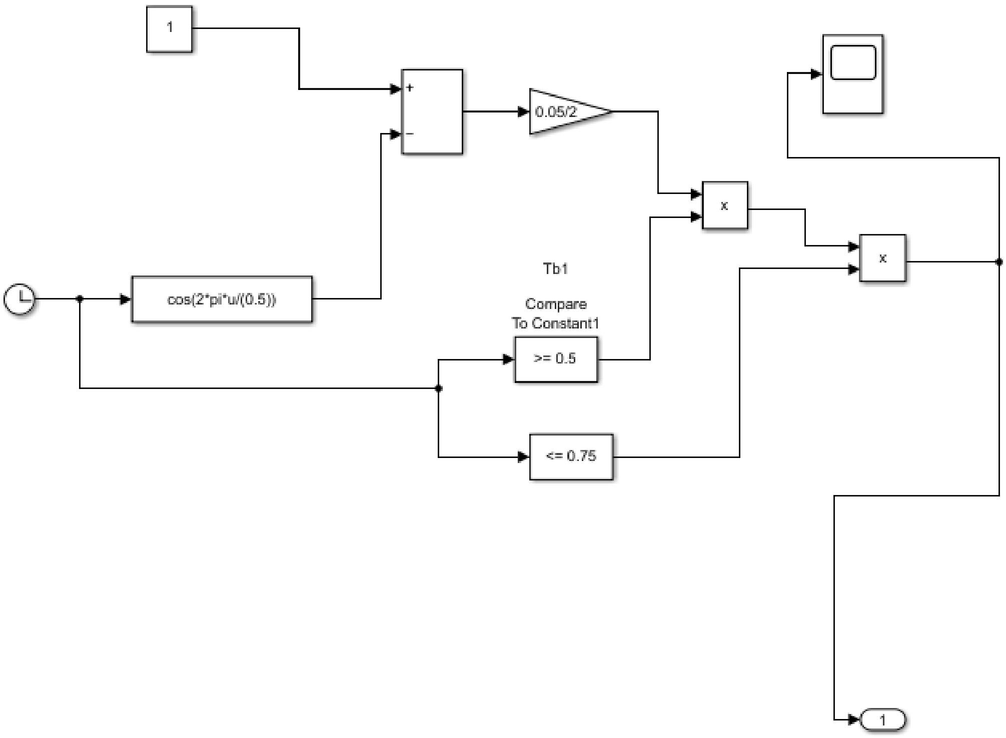


Figure 2: Simulink road disturbance one bump

Figure 2 : ***Simulink model road disturbance with one Bump***

The values of T1 and T2 in Fig.2 are assumed to be equal to bump starting time Tb1 and bump ending time 2Tb1 as described in Eq. (7).

4.2 *Simulink model of road disturbance with two bumps*

Equation (8) has been translated into a MATLAB Simulink model of a road disturbance with two bumps, as seen in Fig.3.

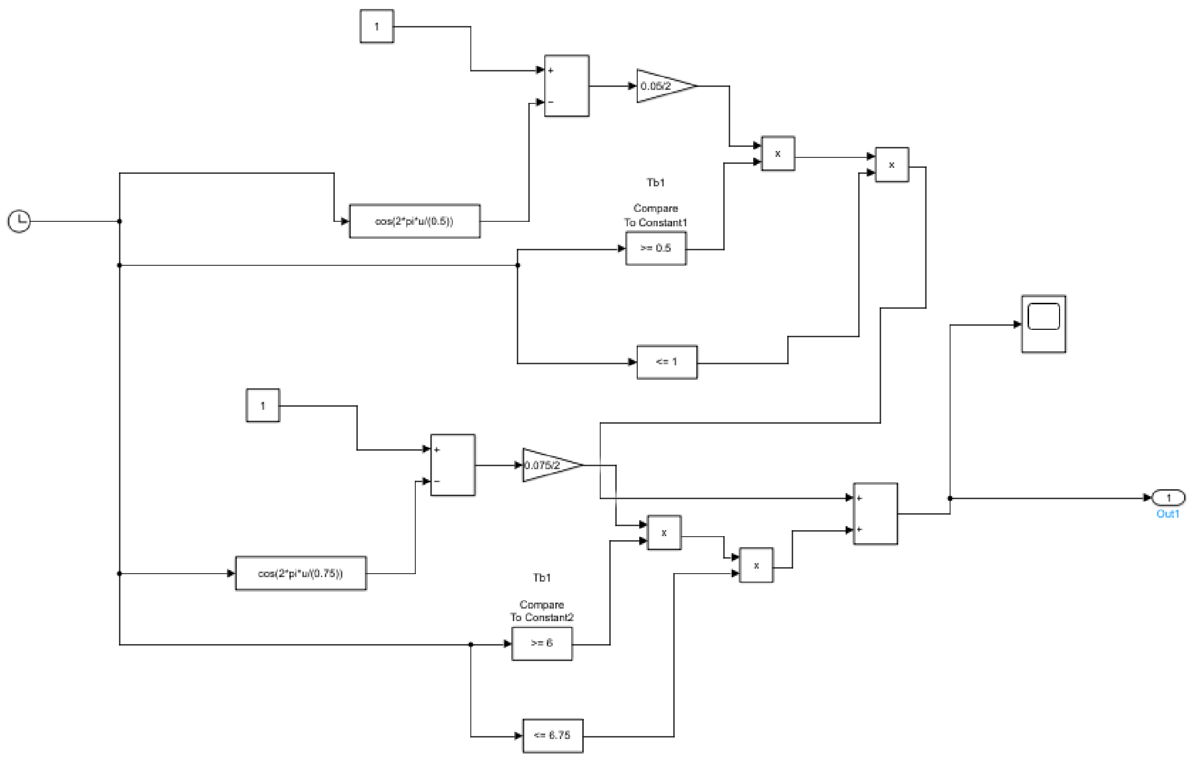


Figure 3: Simulink model road disturbance with two bumps.

The values of T1 and T2 in Fig.3 are taken to be equal to the bump-starting time Tb1 and the bump ending time 2Tb1 during the first road disturbance interval, as given in Eq. (8).
During the second road disturbance, the values of T3 and T4 are taken to be 8Tb2 and 9Tb2, respectively with two bumps in the second road disruption interval. Figure 4 shows a Simulink mode of third road disturbance with sine product step input.

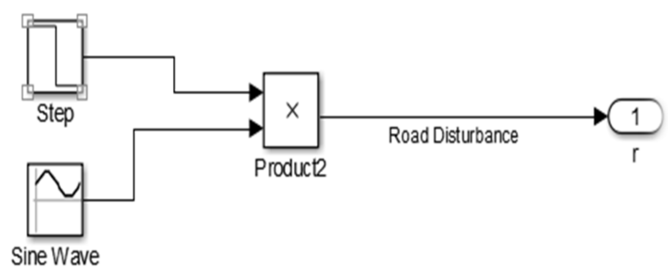


Figure 4: Road profile

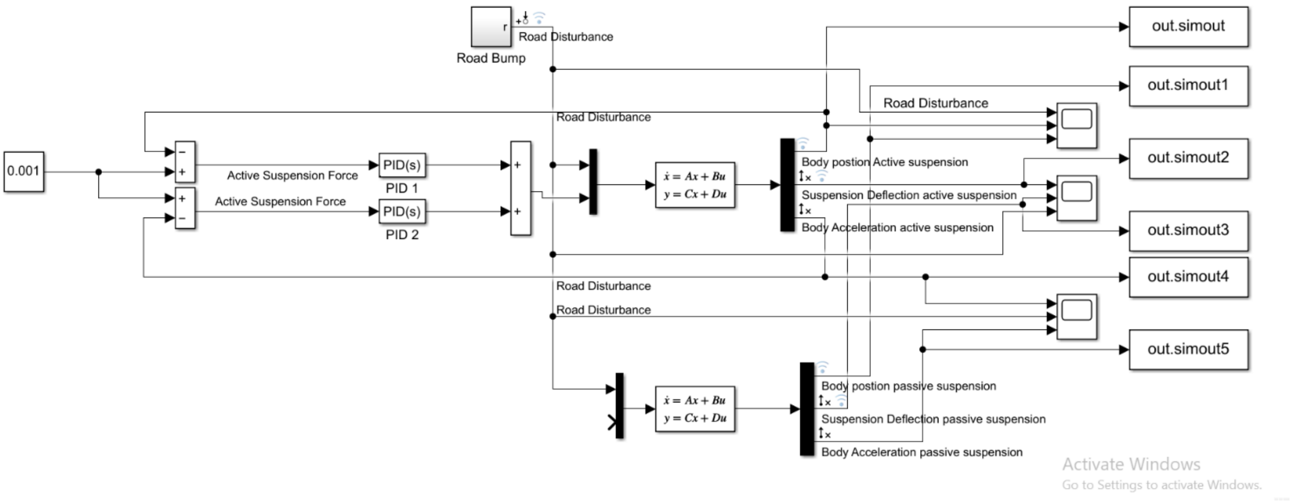


Figure 5: Simulink model of quarter vehicle active suspension system

5. Optimal Pid Using Pso

Particle Swarm Optimization is a new evolutionary computational technique invented by James Kennedy, an American social psychologist, and Russell C. Eberhart [28].
PSO was created mostly by simulating bird flocking. The following is a list of PSO definitions: The attributes of each individual particle i are as follows: x_{id} is the current position in search space; $p_{id}$is the current velocity; and the personal best position in search space, PID.

• In the case of a minimization work, the personal best position, p_{id}, corresponds to the position in search space where particle i has the smallest error as specified by the objective function f.

• The global best position, represented by, is the location with the lowest error over all p_{gd}.

Every particle in the swarm is updated using equations at each repetition. Eq. (9) is Particles Velocity and Eq. (10) is Particles Position.

$$v_{id} = w^* v_{id} + cl^* \text{ rand } ()^* (p_{id} - x_{id}) + c2^* \text{ rand } ()^* \left(p_{gd} - x_{id}\right) \quad (9)$$

$$X^{i+1} = X^i + V^{i+1} \quad (10)$$

where c1 and c2 are two positive constants, c1 and c2 are two random numbers within the range [0, l], and w is the inertia weight, the inertia weight linearly decreases from 0.9 to 0.4 as the algorithm proceeds from one iteration to another iteration. The velocity is limited to predetermined minimum and maximum values. The *Pbest, Gbest* are populatic best and global best of i particle of PSO algorit
Figure 6 shows the flow diagr
optimal P-I-D gains using
suspension systems.
The steps of the PSO algorith

a) Begin this project by assigning each particle in the swarm a random position.
b) For each particle, calculating the fitness function (in this case, mean square error).
c) By comparing each particle's fitness to its Pbest, we may determine which particle has the superior current value, which we can use to designate as the Pbest.
d) The best fitness value's outcome is referred to as guest, and its position is referred to as p_{gd}.

e) Using (a) and (b), upgrade the values of all the particles' velocities and positions.

f) Repeat steps b–e until the maximum number of iterations stops for a sufficiently correct fitness value [30].

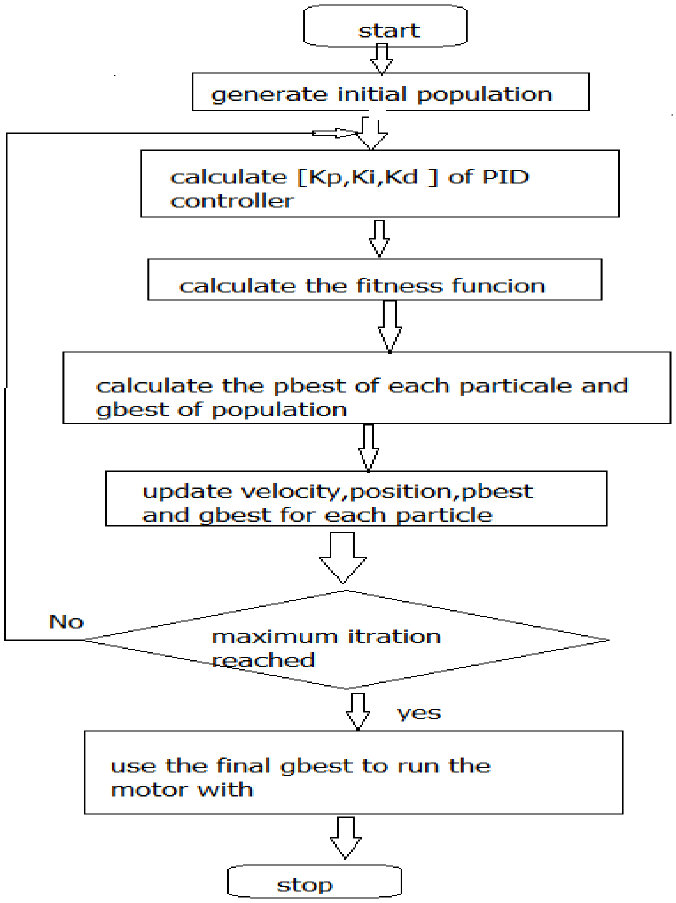


Figure 6: System flow chart [31]

6-*Simulation* results

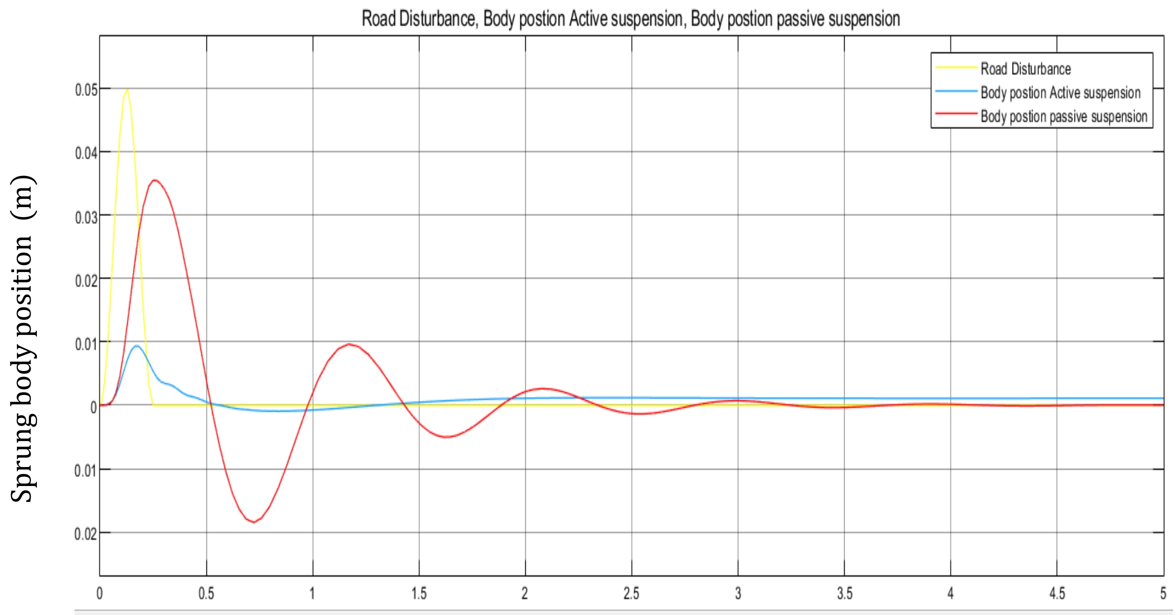
In this section, MATLAB program has been used to simulate the proposed 2PID controllers and passive suspension quarter car system as shown in fig. 6. The values of parameters in quarter car model are shown in table 1. A road profile, which is shown in Fig .4, used to determine the time response. During the movement of the car on the road profile, the body vibration is compensated by the proposed 2PID controllers. PSO optimization algorithm has been used to tune the parameters of the proposed controller with the following objective function by this objective function, the parameters of the proposed controller have been tuned to minimize body acceleration and deflection. PSO algorithm is used with the following parameters:

(Swarm size=500, Cognitive parameter=1, Social parameter=1)

The following parameters are use the ride comfort can be determined based on the time response of position and acceleration of sprung mass. The settling time and overshoot are the important factors by which it can evaluate the performance of the controller. Figures 7 ,8 and 9 show the position, acceleration, and deflection for proposed controller and passive suspension car systems**.** PID controllers' gains calculated as shown in table (2)

Table 2. Optimal Values of the 2P-I-D

	Kp	Ki	Kd
PID1	27.9596636282343	84.8411742135087	2.0058943211592 6
PID2	0	8.35567393631263	0



Time(seconds)

Figure 7: Sprung body position with 2PID

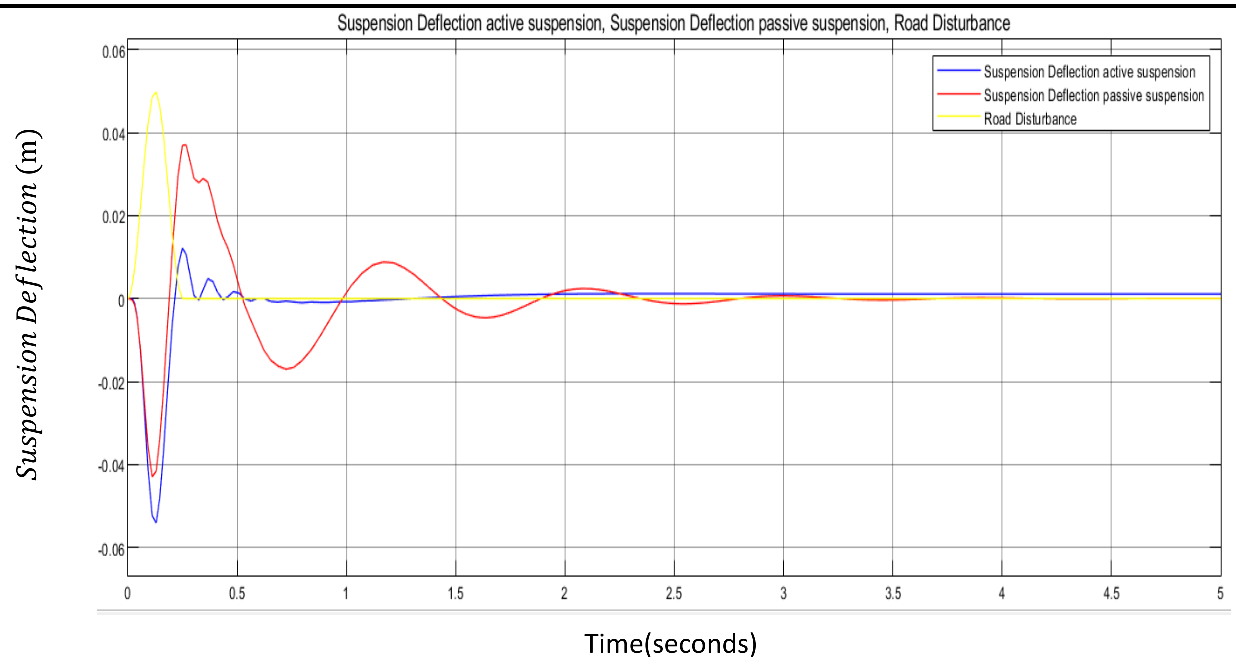


Figure 8: Sprung body Deflection with 2PID

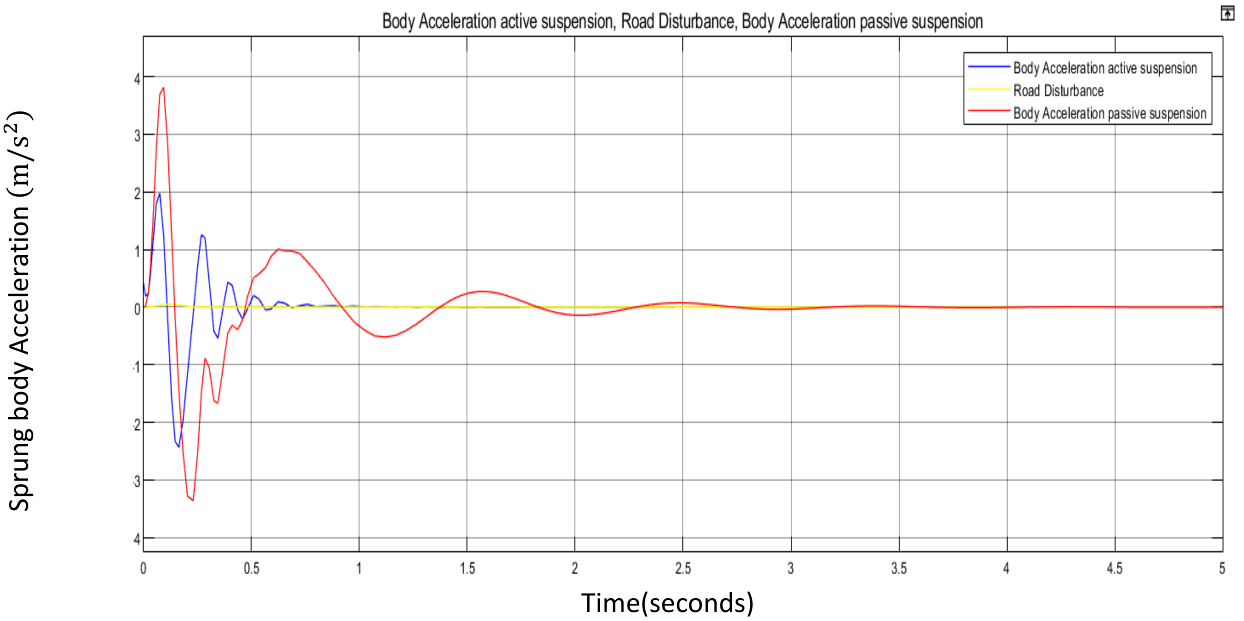


Figure 9: Sprung body acceleration with 2PID

Two forms of road bump disturbances are studied in this study: one with a single bump and the other with two bumps of varying height and length. In the first scenario, the road disruption is caused by a single bump. It is considered a height (A) bump of 0.05m. The length of the bump (L) is supposed to be 0.5m, and the vehicle's velocity (V) is considered to be 3.6km/h.

The bump duration Tb1 is equal to 0.5 seconds and is computed as (L/V).

and the vehicle is assumed to be passing the bump within the time intervals of 0.5 and 1.0 seconds. The position, deflection and acceleration of the vehicle sprung body subjected to single bump disturbance are significantly improved in the case of active suspension system with PSO based suggested 2PID controllers compared to passive suspension system, as shown in Fig.10, Fig.11 and Fig.12.

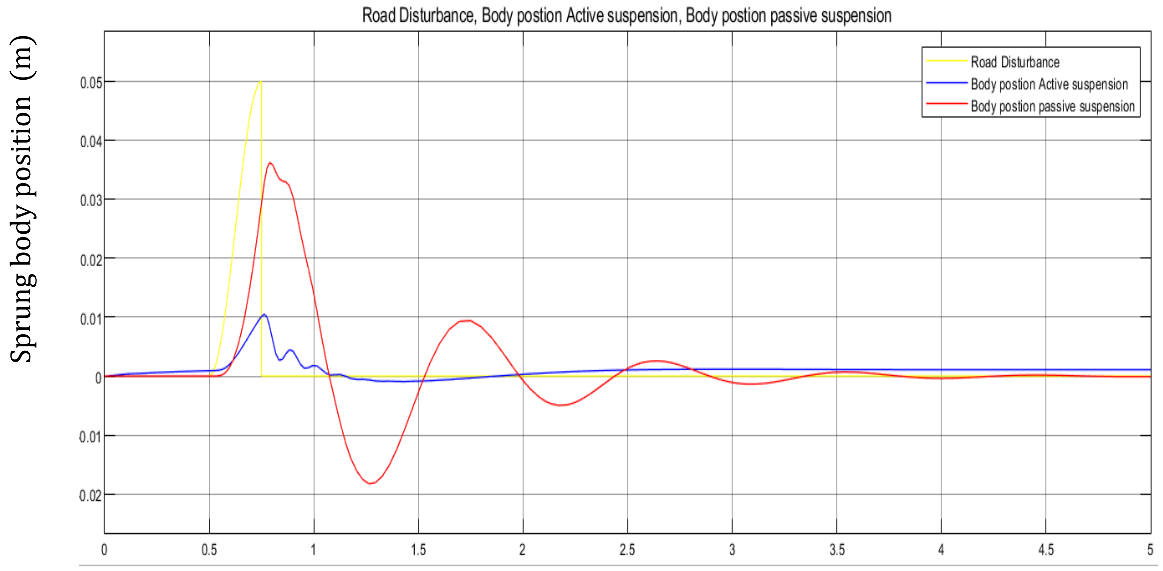
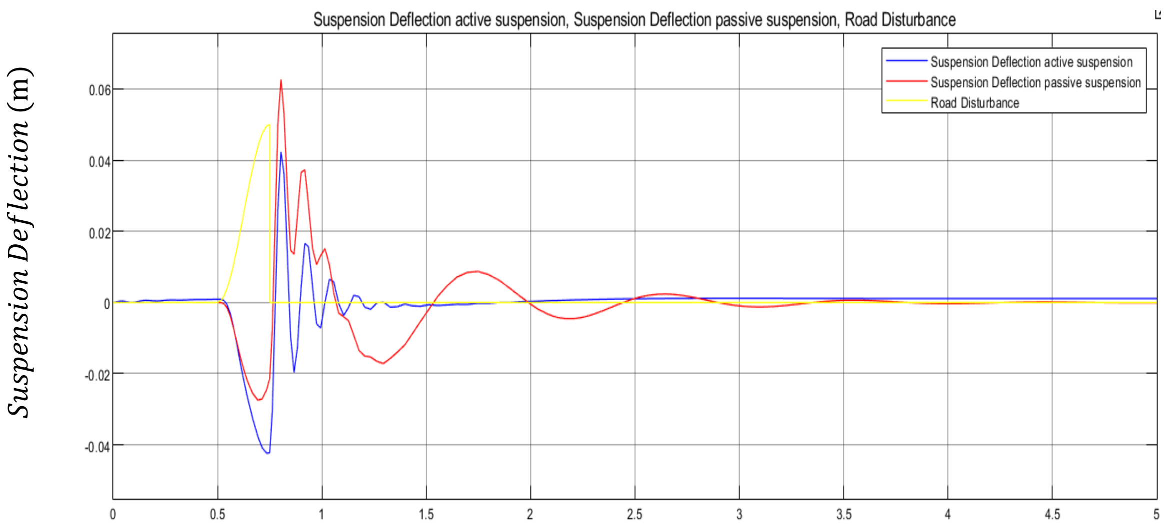
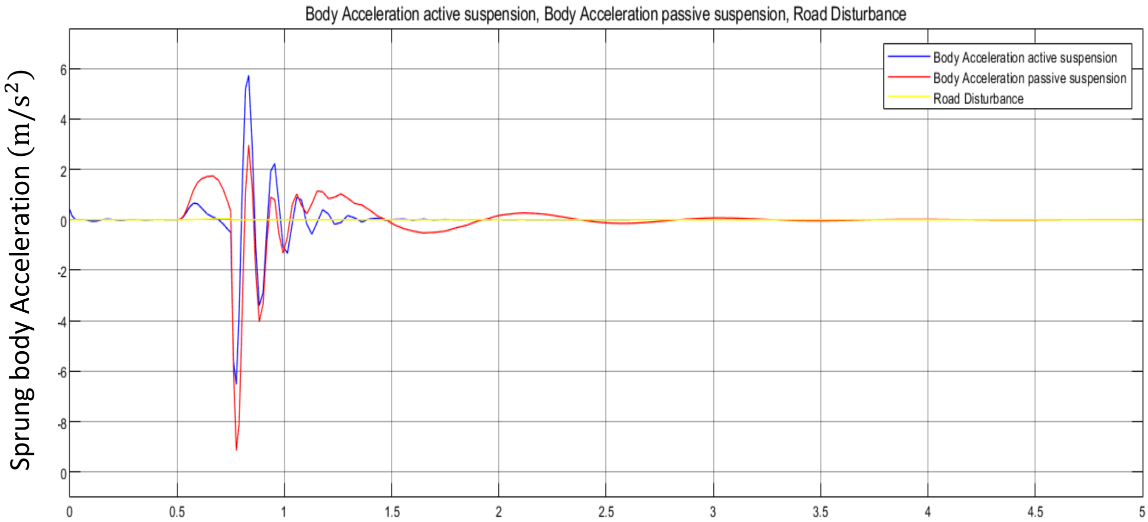


Figure 10: Sprung body position with single bump disturbance.



Time(seconds)

Figure 11: Sprung body Deflection with single bump disturbance.

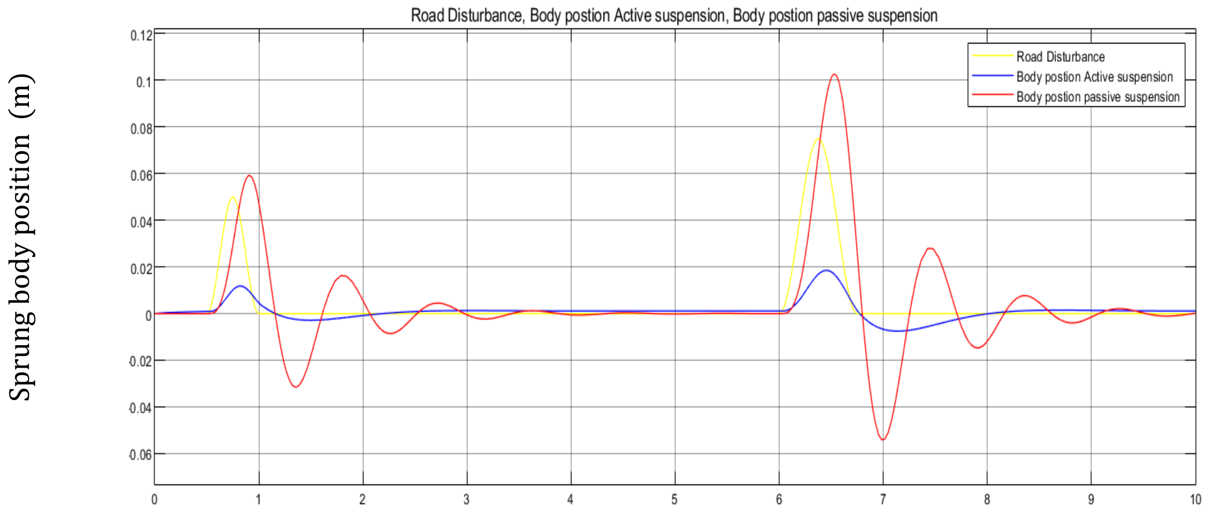


Time(seconds)

Figure 12: Sprung body acceleration with single bump disturbance.

The road disturbance with two bumps of height (A) 0.05m and height (B) 0.075m is studied in the second case. The first bump is 0.5m long, the second bump is 0.75m long, and the car is driving at a velocity of 3.6km/h. The duration of the first bump Tb1 is 0.5 seconds and in the case of the second bump 0.75 seconds. Between the time intervals of 0.5 to 1.0 seconds and 6.0 to 6.75 seconds, the car crosses the bump.

The dynamic performance of the active suspension system is substantially superior in the case of sprung mass position, deflection and acceleration when compared to the passive suspension system, as shown in Fig.13, Fig.14 and Fig.15.



Time(seconds)

Figure 13: Sprung body position with two bump disturbances

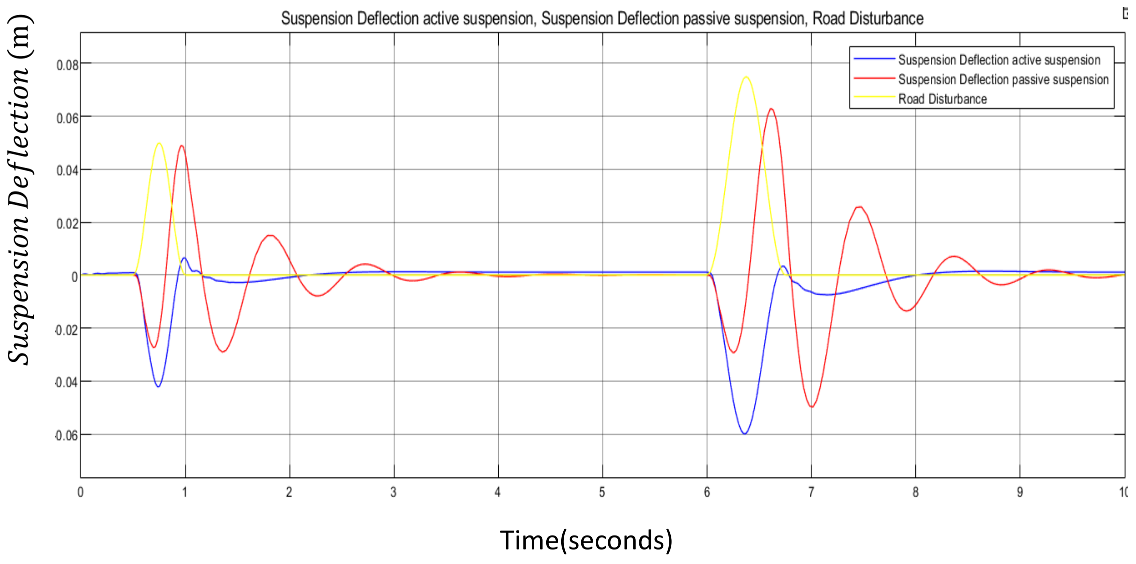


Figure 14: Sprung body Deflection with two bump disturbances

The negative deflection is greater in the case of active suspension system with suggested controller during bump disturbance periods, as shown in Fig.11 and Fig.14 During the disturbance conditions, the larger negative deflection indicates better road holding capacity and good tire-road contact.

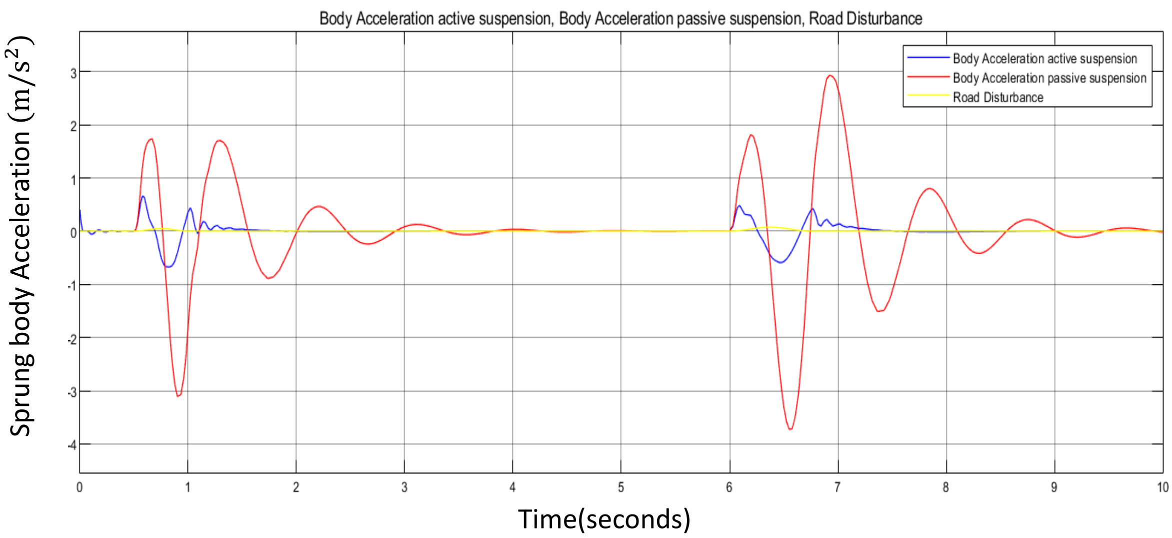


Figure 15: Sprung body acceleration with two bump disturbances.

These figures indicate clearly superior of propped controller. It can notice form these figures that the active system reduced overshoot and settling time when compared to the passive suspension car system. Tables lists the overshoot, rise time and settling time for the two models. It is noticed that the response of the active model is better, and the reduction is good compared with the passive model.

Table 3. Overshoot, settling time and risetime of body position with one bump

Parameter	active	passive	Reduction (%)
Rise time	0.554	0.585	5.299
Settling time	1.004	3.5552	71.760
Overshoot	0.00959	0.03589	73.280

Table 4. Overshoot, settling time and risetime of body Deflection with one bump

Parameter	active	passive	Reduction (%)
Rise time	0.489	0.768	36.328
Settling time	1.169	2.506	53.352
Overshoot	0.04122	0.06214	33.666

Table 5. Overshoot, settling time and risetime of body acceleration with one bump

Parameter	active	passive	Reduction (%)
Rise time	0.507	0.521	2.687
Settling time	1.184	2.127	44.335
Overshoot	3.008	5.688	47.117

Table 6. Overshoot, settling time and risetime of body position with two bumps

Parameter	active	passive	Reduction (%)
Rise time	0.532	0.568	6.338
Settling time	2.245	2.491	9.876
Overshoot	0.01149	0.06535	82.418

Table 7. Overshoot, settling time and risetime of body deflection with two bumps

Parameter	active	passive	Reduction (%)
Rise time	0.100	0.812	87.685
Settling time	1.301	2.497	47.898
Overshoot	0.005931	0.04923	87.953

Table 8. Overshoot, settling time and risetime of body acceleration with two bumps

Parameter	active	passive	Reduction (%)
Rise time	0.501	0.506	0.988
Settling time	1.206	2.010	40
Overshoot	0.6828	1.7273	60.470

8- Conclusion

In this paper, a suspension quarter car model problem is used as optimization problem to determine the optimum parameters of the proposed 2PID controllers to get the best performance of the suspension car system. MATLAB program has been used to simulate the proposed method. PSO is used as optimization algorithm. controller achieve good performance with small settling time and small overshoot. Moreover, the proposed method includes high suppression of on vehicle vibration and good enhancement in ride comfort. When compared to passive suspension, simulation results showed that active suspension had improved overshooting, settling time, and rising time.

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