Int. J. Advance. Soft Comput. Appl., Vol. 6, No.3, November 2014
ISSN 2074-8523

Intelligent optimal Controller of Active Suspension System Based on Particle Swarm Optimization

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Abstract

This paper proposes a cascade control algorithm of an active suspension system (ASS) with hydraulic actuator dynamic for a quarter car model. The objective of designing a controller for the car suspension system is to improve the ride comfort while maintaining the constraints on to the suspension travel and tire deformation, which is subjected to different road profiles. The control algorithm is based on the fusion of robust control and computational intelligence techniques which consists of the inner loop controller for force tracking control of the hydraulic actuator model and the outer loop controller for disturbance rejection control. Particle swarm optimization (PSO) algorithm is employed to optimize the Proportional-integral (PI) controller parameters for force tracking control of the hydraulic actuator model. Similarly, the PSO algorithm is utilized in the outer loop controller to search for the optimal values of the weighting matrices for the linear quadratic optimal control (LQR) such that the desired performance of the ASS is guaranteed. In comparison with the passive suspension system, the simulation results demonstrate the superiority of proposed PSO-based controller, where it significantly improved the ride comfort by maintaining the other constraints (the suspension travel, tire deflection, and control force) in their limits.

Keywords: *Active Suspension System, linear quadratic regulator, Proportional Integral Control, Particle Swarm Optimization.*

1 Introduction

A car suspension system is the mechanism that physically isolates the vehicle body from the wheels. It governs many crucial functions of the vehicle, i.e., supports the vehicle weight, isolates effectively the vehicle body from the road excitations, maintains tire-ground contact, and assigns the vehicle wheels at appropriate positions of road terrain [1]. In addition, the main functions of a vehicle suspension system are to efficiently improve the control performance and the ride comfort for passengers in a vehicle while maintaining adequate vehicle stability.

The role of a conventional passive car suspension model is to achieve a trade-off between the ride comfort and road handling performance. Nevertheless, an active suspension system (ASS) varies from the conventional car suspensions in its ability to store, dissipate and to introduce energy to the system. Fig. 1 shows the schematic view of the ASS, where the hydraulic actuator is installed in parallel with the passive components. The objective of designing the ASS is to optimize the compromise between ride comfort and road handling performance. Typically, a high-quality ASS can isolate the car body from the vibration arising from road surface. Moreover, it ensures the contact between the wheels and road surface to provide a desirable ride comfort and safety.

The improvement of a vehicle active suspension control system currently receives a lot of attention from both academics and automobile industrial researches. Many active suspension control methods have been recently proposed to optimize the compromise between ride comfort and road handling properties, i.e., optimal state feedback control [2-3], propositional derivative (PD) control [4], robust control [5], fuzzy logic control [6-8], and sliding mode control [9-10]. An adaptive nonlinear controller is designed in [11] to reduce the model error of uncertain ASS. In addition, Sharkawy [12] presented an adaptive fuzzy control technique to improve the riding quality of ASS. Similarly, a fuzzy logic controller is developed by Sakman et. al [13] for four degrees of freedom non-linear ASS.

Most of the existing ASS studies focused on the main-loop controller design without considering the dynamics of the hydraulic actuators. Meanwhile, the desired force of the main-loop controller is calculated as a function of vehicle suspension states and road disturbance input. Although it is widely assumed that the hydraulic actuators can carry out the desired force of the main loop controller accurately. In reality, however, actuator dynamics are complicated and exhibit highly nonlinear behavior. Moreover, strong interaction exists between the force that the actuators can generate and the vehicle body motions. Therefore, some researchers include the force tracking controller of a hydraulic actuator to ensure that the desired force commanded by the main loop vehicle controller is achieved precisely [14-16]. For instance, a modular adaptive robust control is developed in [8] to design the force loop controller for an ASS with considering the hydraulic actuator dynamics. Similarly, a sliding mode control is proposed by Sam and Osman [14] to improve the riding quality of an electro-hydraulic active suspension system.

The purpose of this paper is to develop a robust controller that uses the Particle Swarm Optimization (PSO) algorithm to solve the multi-objective optimization problem for an electro-hydraulic ASS. Although the control algorithm described here is, in principle, similar to that of Chantranuwathana [15] and Sam et. al [16] which consists of the inner loop controller for force tracking control of the hydraulic actuator model and the outer loop controller for disturbance rejection control, the proposed controller, however, differs from the latter mainly by the inclusion of PSO algorithm in the inner loop controller as well as outer loop controller.

Proportional-integral (PI) control based on PSO algorithm is developed for force tracking control of the hydraulic actuator model, where PSO algorithm is employed to optimize the selection of PI controller gains to improve the overall system performance. Similarly, PSO algorithm is utilized in the outer loop controller to obtain the optimum weighting matrices for the linear quadratic optimal control (LQR), where the car body acceleration is used as a fitness function. The objective of LQR based on PSO algorithm is to improve the riding quality of ASS by minimizing the vertical body acceleration, in which the constraints of the ASS, i.e., suspension travel, tire deflection, and control force are satisfied. This multi-objective controller is realized by using the PSO algorithms to search for the optimal values of the weighting matrices for the LQR such that the desired performance of the ASS is guaranteed.

The paper is outlined as follows: In section 2, the methods are presented, which include the model of the quarter car ASS, the PSO algorithm and the controller design. The simulation setup and measurement method is given in section 3. The results of the designed controller as compared to passive suspension system are discussed in section 4. Finally, the conclusion is presented in section 5.

2 Materials and Methods

2.1 Quarter Car Model

Based on the study [16], a quarter car suspension system is used and the model is shown in Fig. 1. The parameters of a quarter car model are presented in Table 1. The dynamic equations of the two-degree-of-freedom quarter car suspension system are:

$$m_s \ddot{z_s} = k_s(\tag{1}$$

$$m_u \ddot{z_u} = -k_s(z_u - z_s) - b_s(\dot{z_u} - \dot{z_s}) + k_t(z_r - z_u) - f_a \tag{2}$$

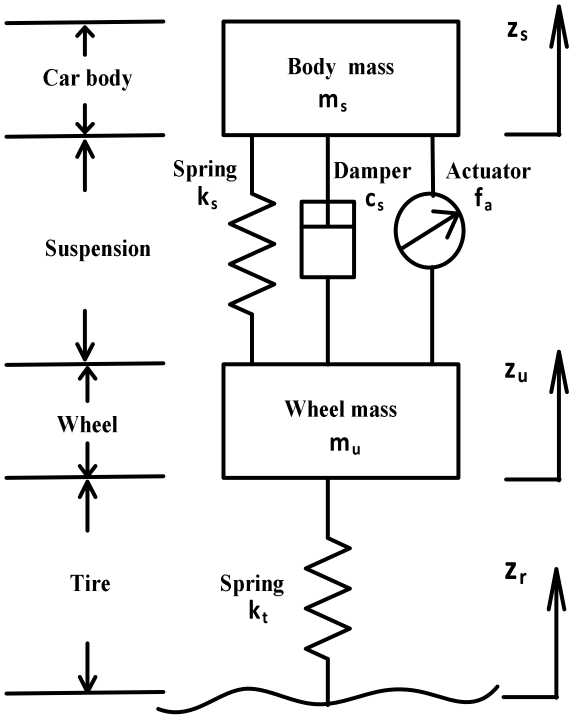

Fig. 1 Quarter-car active suspension model

Table 1: The parameters of quarter car model

Parameter	Description
m_s	sprung mass = 282 kg
m_u	unsprung mass = 45 kg
k_s	spring constant = 17900 N/m
k_t	spring constant of tire =165790 N/m
b_s	damper coefficient = 1500 N/(m/s)
Z_s	displacement of vehicle chassis relative to plain ground
Z_u	displacement of wheel relative to plain ground
Z_r	uneven road surface relative to plain ground
	output force provided by the servo-hydraulic cylinder.

The complete mathematical model of the system that describes the dynamical behavior of the electro hydraulic actuator consists of the hydraulic dynamics and the servo-valve dynamics. The dynamics equation of the hydraulic actuators is given as follows [17]:

$$F_a^{\cdot} = A_p \alpha \left[C_{d1} w u_1 \sqrt{\frac{P_s - sgn(u_1)P_L}{\rho}} - C_{d2} u_2 sgn(P_L) \sqrt{\frac{2P_L}{\rho}} - C_{tm} P_L - A_p (z_u^{\cdot} \right. \quad (3)$$

The servo-valve dynamics is computed using the following form [18]:

$$\tau u^{\cdot} + u = kv \quad (4)$$

The parameters of hydraulic actuator model are taken from [17]. The parameters are presented in Table 2.

Table 2: The parameters of hydraulic actuator model [17].

Parameter	Description
C_{d1}	Discharge coefficient = 0.7
A_p	Piston area =
ρ	Specific gravity of hydraulic fluid = 3500
α	Hydraulic coefficient =2.273e9 N/
w	Spool valve width = 0.008 m
P_L	Pressure induced by load
P_s	Supply pressure = 20684 kN/
u_1	Spool valve position
u_2	Bypass valve area
C_{tm}	Leakage coefficient = 15e-12
F_a	Actuator force
V	Input voltage command

The mathematical model of a quarter-car active suspension system equipped with a hydraulic actuator dynamics model are given by the following state space representation

$$\begin{bmatrix} x_1^{\cdot} \\ x_2^{\cdot} \\ x_3^{\cdot} \\ x_4^{\cdot} \\ x_5^{\cdot} \\ x_6^{\cdot} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & -1 & 0 & 0 \\ \frac{-k_s}{m_s} & \frac{-b_s}{m_s} & 0 & \frac{b_s}{m_s} & \frac{1}{m_s} & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_s}{m_u} & \frac{b_s}{m_u} & \frac{-k_t}{m_u} & \frac{-b_s}{m_u} & \frac{-1}{m_u} & 0 \\ 0 & -A_p^2\alpha & 0 & A_p^2\alpha & -\alpha C_{tm} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{-1}{\tau} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ \Phi \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{K}{\tau} \end{bmatrix} V + \begin{bmatrix} 0 \\ 0 \\ -1 \\ 0 \\ 0 \\ 0 \end{bmatrix} z_r^{\cdot} \quad (5)$$

where, f_a is the control force from the hydraulic actuator and assumed as the control input. In general Eq. (5) can be written in a compact form as:

$$x^{\cdot}(t) = Ax(t) + Bu(t) + f(x,t) \tag{6}$$

where $x(t) \in R^{n*m}$ is the state vector, $u(t) \in R^{m*n}$ is the control input, and the continuous function $f(x,t)$ represents the uncertainties with the mismatched condition. Thus, in order to simplify the analysis, the following assumptions are made:

Assumption 1. There exists an $\beta > 0$ such that $\|f(x,t)\| \leq \beta$, where $\|*\|$ represented the standard Euclidian norm.

Assumption 2. The pair (A, B) is controllable and the input matrix B has a full rank.

2.2 Particle Swarm Optimization (PSO) Algorithm

PSO algorithm is a population based optimization method that was originally developed by Kennedy and Eberhart in 1995 [19] . PSO algorithm is initialized with a population of randomly generated solutions, which are called particles. Then, PSO searches in these particles for optimal solution. The position and the velocity of each particle are updated in each iteration according to its previous best position. Each individual particle has a current position x_i, velocity v_i, and personal best position $(x_{idpbest})$. The position amongst all the particles' personal best positions that yielded the smallest error is called the global best position $(x_{idgbest})$. The particle's velocity is updated during each iteration and the new velocity is added to the particle's current position to determine it is new position.

The velocity and the position of each particle are updated according to the following equations [20]:

$$w(t) = w_{min} + (w_{\text{I}} \tag{7}$$

$$v_{id}(t) = w(t)v_{id}(t-1) + 2\alpha(x_{idpbest}(t-1) - x_{id}(t-1) + 2\alpha\left(x_{idgbest}(t-1) - x_{id}(t-1)\right) \tag{8}$$

$$x_{id}(t) = v_{id}(t) + x_{id}(t-1) \tag{9}$$

where, wmin and wmax are the maximum and minimum values of the inertia weight w, vid(t) is the velocity of the particle i at iteration t, xid(t) is the current position of particle i at iteration t, m is the maximum number of iterations, i is the number of the particles that goes from 1 to n, d is the dimension of the variables, and α is a uniformly distributed random number in (0,1).

The described PSO algorithm is utilized in the inner loop controller to optimize the selection of PI parameters, and the force tracking error is used as a fitness function. Similarly, the PSO algorithm is adopted in the outer loop controller to calculate the optimal values of the weighting matrices for the LQR controller such that the riding quality of ASS can be improved.

2.3 Controller Design

Fig. 2 shows the controller structure of a suspension system implemented in this study. The controller is composed of two controller loops namely inner loop controller and outer loop controller. Proportional-integral (PI) control based on PSO algorithm is utilized in the inner loop controller for force tracking control of the hydraulic actuator model, where PSO algorithm is employed to optimize the selection of PI controller gains to improve the overall system performance. LQR based on PSO algorithm is adopted in the outer loop controller to improve the riding quality of ASS by minimizing the vertical body acceleration, in which the constraints of the ASS, i.e., suspension travel, tire deflection, and control force are satisfied.

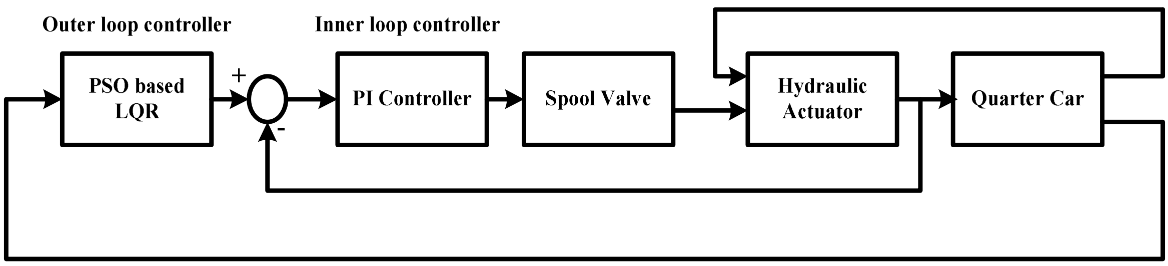

Fig. 2 Controller structure of the ASS

2.3.1 Inner Loop Controller Design (Actuator Controller)

Fig. 3 shows the inner-loop control system that used for force tracking control of the hydraulic actuator. PI control is implemented which takes force tracking error as the input and delivers control voltage to drive the spool valve. The forcing functions depend on the type of road disturbance, which can be represented by sinusoidal, saw-tooth, square, and random functions.

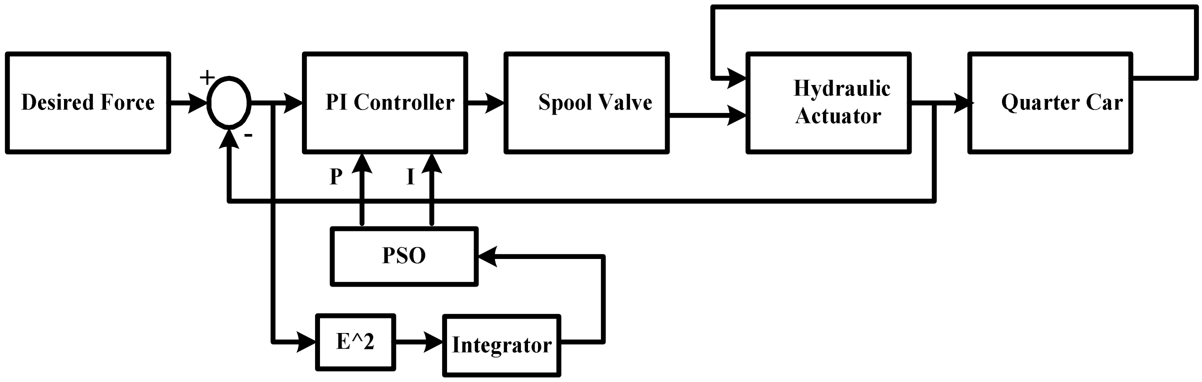

Fig. 3 Force traking controller (Inner loop controller)

The PI gain parameters of the inner loop controller are optimized by algorithm. The integral square of the force trucking error is used as a f function. The objective of the optimization is to minimize the fitness function performance index as: (10)

2.3.2 Outer Loop Controller Design Using LQR

PSO algorithm is utilized in the outer loop controller to obtain the optimum weighting matrices for the LQR, where the car body acceleration is used as a fitness function. The inputs of the outer loop controller are wheel velocity and the vehicle body velocity, while the output of the outer loop controller is the desired force that must be exerted by the hydraulic actuator. Typically, the inner loop controller is used to control the spool-valve displacement, in which the actuator force can track the desired force requested by the outer loop controller accurately.

Consider a controllable quarter car suspension model given as follows:

$$x^{\cdot}(t) = Ax(t) + Bu(t) + f(x,t) \tag{11}$$

where $\mathrm{x(t)} \in R^{n*m}$ is the state vector, $u(t) \in R^{m*n}$ is the control input, and the continuous function $f(x,t)$ represents the uncertainties with the mismatched condition. The aim is to find a linear state-feedback law *u*=-*kx*, which minimize the following fitness function performance:

(12)

where, $\bar{y}$ (t) is the acceleration of the car body and T is the integral period time.

3 Simulation Setup

The proposed PSO-based controller and the mathematical model of the system as defined in Eq. (5) are simulated in MATLAB-SIMULINK. The parameters of the quarter car suspension model selected for this study are listed in Table 3.

Table 3: The parameters values used in the car suspension model.

Parameter	m_s	k_s	b_s	k_t	m_u
Value	$282\,kg$	$17900\,N/m$	$1500\,N/(m/s)$	$165790\,N/m$	$45\,kg$

The road disturbance z_r used in this simulation is represented by a bump as shown in Fig. 6.

$$z_r = \begin{cases} 0.025(1-\cos 8\pi t), & 0.5\,sec \le t \le 0.75\,sec \\ 0.036(1-\cos 8\pi t), & 3\,sec \le t \le 3.25\,sec \\ 0, & otherwise \end{cases} \quad (13)$$

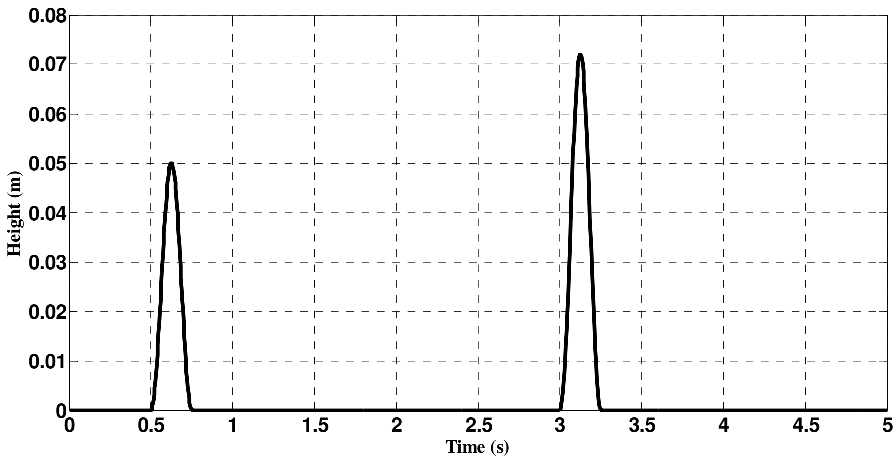

Fig. 4 The road disturbances.

Initially, the parameters of the inner loop controller should be optimized using the proposed PSO algorithm until the hydraulic actuator is able to track the desired force accurately. Then; the inner loop controller is combined with the outer loop controller to improve the riding quality of the ASS. The proportional and integral gains of the PI controller are set to 0.94 and 0.22, respectively. These values are obtained by the proposed PSO algorithm, in which the integral tracking error is used as a fitness function. The number of particle in each swarm is set to 50 and the maximum number of iteration is set to 100.

Figs. 4-5 show, respectively, the force tracking control performance of the hydraulic actuator model using PI controller for sinusoidal and square functions. The results show that the hydraulic actuator can accurately track the desired force.

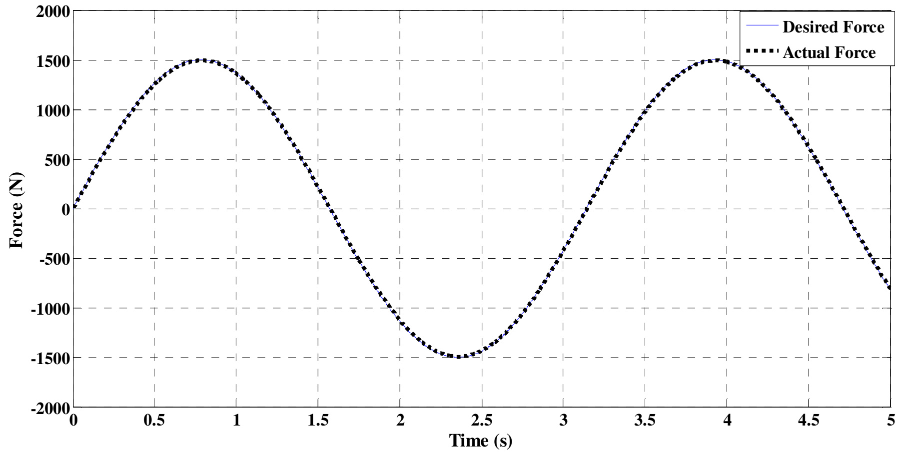

Fig. 5 Force tracking of a sinusoidal function

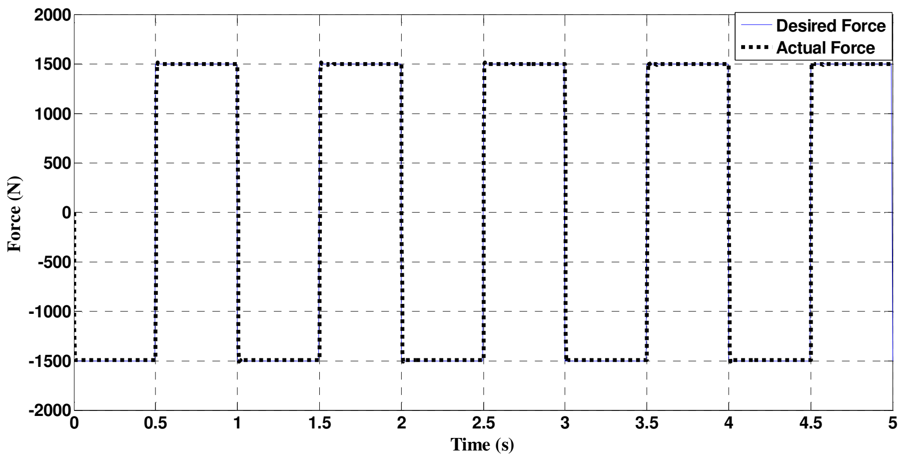

Fig. 6 Force tracking of a square function

The values of Q and R for the LQR controller are obtained by the proposed PSO algorithm. The number of particle in each swarm is set to 20 and the maximum number of iteration is set to 70. The PSO search process should be terminated when there is no improvement in the value of the fitness function for a particular number of iterations or the maximum number of iterations is reached.

$$Q = \begin{bmatrix} 0.1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 0.01 & 1 \\ 0 & 0 & 0 & 0.01 \end{bmatrix} * 10^5$$

Using Matlab the poles and state feedback gains for the LQR controller are as follows:

$$\text{Poles} = \begin{bmatrix} -17.3220 + 59.8905i \\ -17.3220 - 59.8905i \\ -7.7588 + 0.3860i \\ -7.7588 - 0.3860i \end{bmatrix}$$

$$K_{lqr} = [55 \quad 3277 \quad -15209 \quad 5]$$

The simulation was performed for a period of 5 second with a variable step size using ode45 (Dormand-Prince) solver. There are two parameters to be observed in this study namely, the car body acceleration and the wheel deflection. The main objective is to minimize the car body acceleration for ride comfort by maintaining the following constrains:

Suspension travel limit is ± 8 cm [21].

Maximum tire deflection,

$$(x_u - x_r) \leq \frac{9.8*(m_s+m_u)}{k_t} = 1.9 \text{ cm}$$ [22].

Spool valve displacement limits ± 1cm.

Force limits (1000N) [21].

4 Result and Discussion

This section discusses the simulation results of the proposed cascade controller for the mathematical model of the system as defined in Eq. (5).

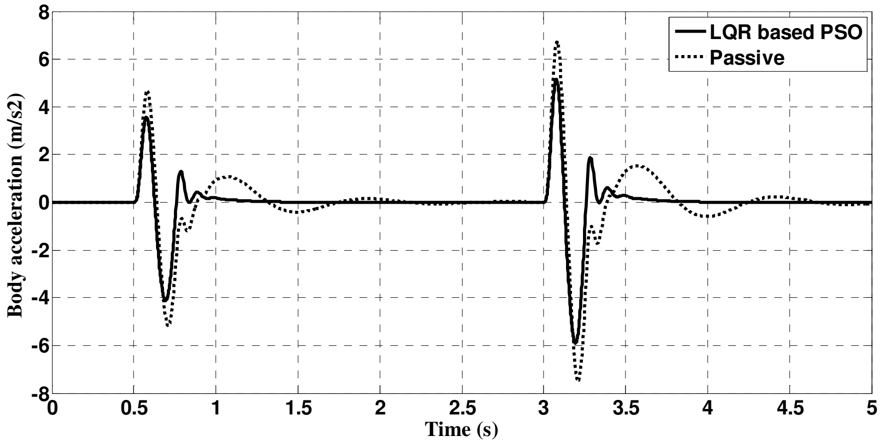

Fig. 7 Unsprung mass acceleration

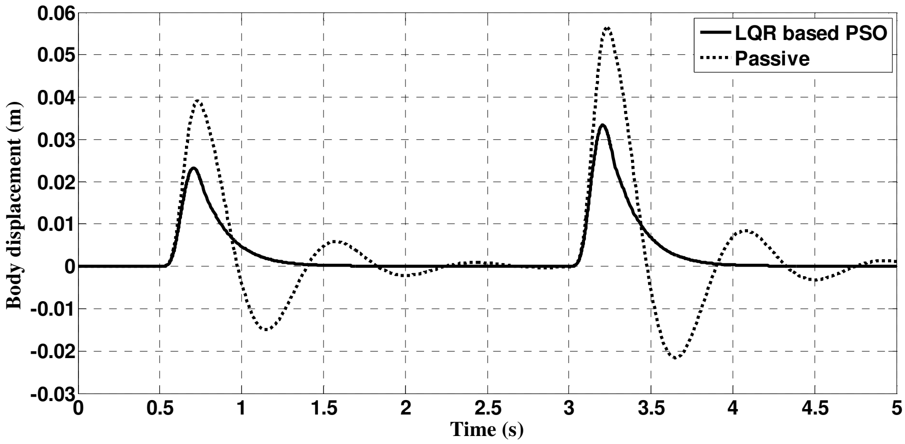

Fig. 8 Unsprung mass displacements

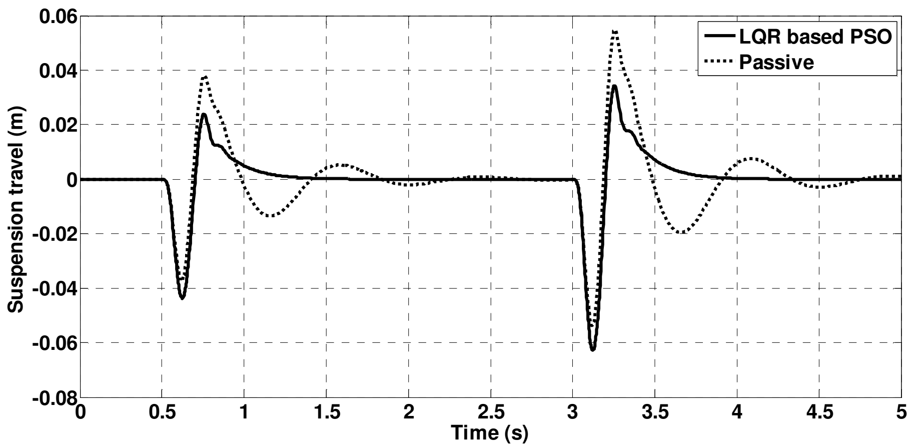

Fig. 9 Suspension travel

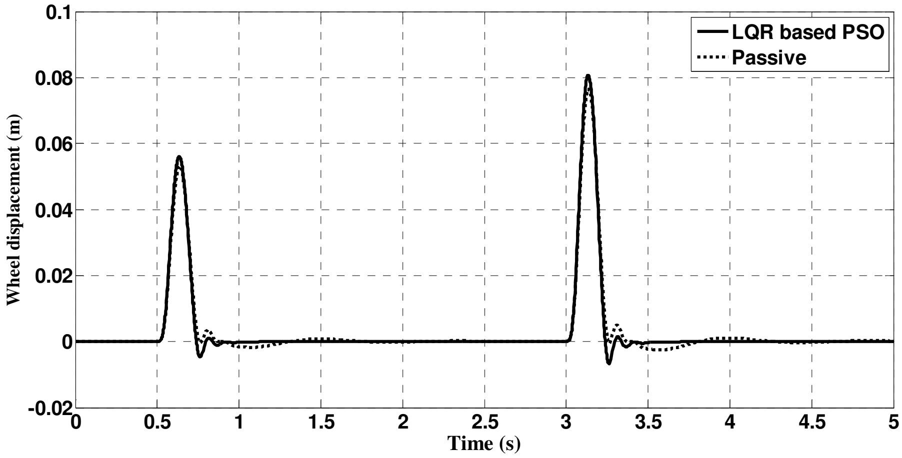

Fig. 10 Wheel displacement

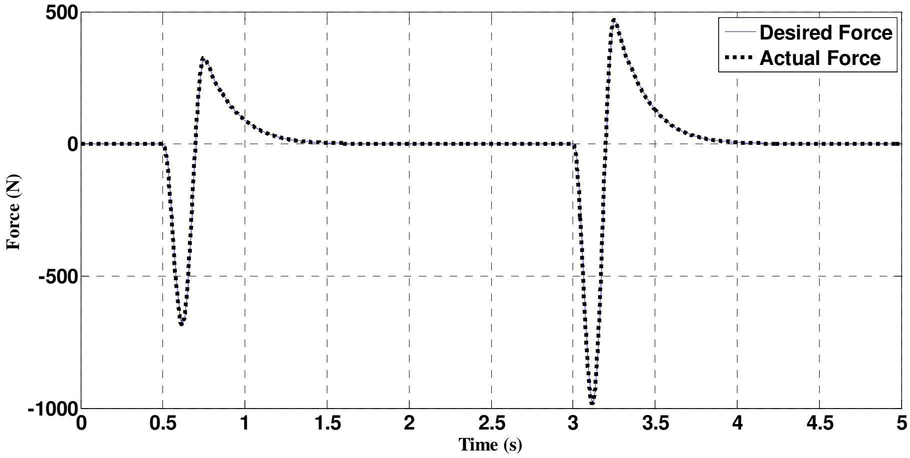

Fig. 11 Control force

Figs. 7-11 show, respectively, the responses of the unsprung mass acceleration, unsprung mass displacements, suspension travel, wheel displacement, and control force. As shown in Fig. 7, the unsprung mass acceleration and displacement are cleary reduced using the proposed PSO-based controller compared to the passive suspension system. In addition, the proposed PSO-based controller provides a significant improvement in ride comfort compared to the passive suspension system. The peak value of body acceleration which is a measure of ride quality is reduced from 6.73 m/s^2 in the case of passive suspension system into 5.15 m/s^2 using the proposed PSO-based controller. . The car body displacement is shown in Fig. 8. It can be observed that using the proposed PSO-based controller the car displacement is considerably reduced compared to to the passive suspension system. In addition, the car displacement using the proposed LQR based PSO is smoothely changed which provids a desirable ride comfort and safety.

The performances of suspension travel and wheel displacement are shown in Figs. 9 and 10, respectively. It can be observed that there is a little increment in the suspension travel and wheel displacement in the case of the proposed PSO-based controller compared to the passive suspension system, but they are still in their limits (suspension travel limit and tire displacement limit). Therefore, the constraints of suspension travel limit as well as maximum tire deflection are guaranteed. Similarly, as shown in Fig. 11 the control force does not exceed the limit of the hydraulic actuator (1000N) using the proposed PSO-based controller. The proposed controller can greatly reduce the vibration as well as the settling time of unsprung mass, suspension travel, and wheel displacement compared to the passive suspension system.

5 Conclusion

This paper proposed a robust control algorithm for a quarter car suspension system with hydraulic actuator dynamic. A proportional integral control (PI) is used in the inner loop controller for force tracking control of the hydraulic actuator model, where the PI parameters are optimized by PSO algorithm. Similarly, PSO algorithm is utilized in the outer loop controller to search for the optimal values of the weighting matrices for the LQR such that the desired performance of the ASS is guaranteed. The proposed controller is compared with the existing passive suspension system and it has shown a better performance, such that it improves the ride comfort by maintaining the suspension travel, tire deflection, and control force constrains in their limits. In comparison with the other control methods, the results demonstrate that the proposed PSO-based controller clearly decreases both car body displacement as well as vehicle body acceleration. Moreover, the result shows that the hydraulic actuator is able to track the desired force accurately.

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