Automation of Hydraulic Active Suspension System using Harmony Search Algorithm Tuned FOPID

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***Abstract* - Comfortable rides in vehicles require active suspension systems to give road holding ability. Suspension systems have a long history from horse drawn carriages with leaf springs to today's automobiles with automation solutions. They need to absorb dips and bumps in the road and involve complex control strategies with many control parameters, multi objective and uncertain disturbances and can be passive or active systems. In the passive type dampers absorb shocks and springs act as energy storing elements. Performance of the passive suspension system is decided by the selection of springs and dampers and a compromise is usually required to give smooth movement. Active suspension systems use actuators to provide comfortable rides, where sprung mass and un-sprung mass are measured and sent to a controller. In this paper we propose a Fractional order PID controller which involves five parameters, three gains together with integrator and differentiator. As mathematical approaches to determine the parameters are complicated, we propose a novel use of Harmony search as a stochastic optimizer to auto-tune the Fractional order PID controller.**

Keywords – Auto-tuning, Harmony Search Algorithm, Active Suspension System, FOPID controller.

I. INTRODUCTION

Suspension system finds application right from bullock carts to the heavy vehicles. In carts, flexible leaf springs are placed at the four corners and in the modern vehicles active suspension systems are provided with complex modern automation solutions. Suspension system includes springs, linkages and shock absorbers between vehicle and the wheel allowing relative motion between them. The road has random profiles and when the vehicles moves it is subject to vibrations which causes discomfort to the passengers and affects vehicle's durability. Suspension system is the one which guarantees comfort to passengers, vehicles durability and road holding ability. Vehicle suspension system distributes the vehicle's weight evenly which in turn absorbs the dips and bumps even on rough road conditions. The suspension system with complex objectives and stochastic disturbance involves control of multiple parameters to achieve the desired control performance. The springs provided in the suspension system forms the first line of defence against turbulent road disturbances and transfers it to the shock absorbers.

Different types of suspension systems are passive suspension system, semi-active and active suspension system. Passive suspension system comprises of damper and energy storage elements, the spring. Spring absorbs turbulent bumps and transfers it to shock absorber and it is passive since energy is not transferred. Working conditions plays an important role in the choice of springs and dampers. As the road conditions are stochastic in nature, the selection of the passive components are a mind boggling problem which motivates the need for the research on active suspension system. Active suspension system involves the measurement of acceleration of sprung mass and unsprung mass and control of the actuators which in turn exerts an independent force to guarantee smooth ride.

There are various types of vehicle models; quarter car model, half car model and full car model. Normally for research purposes, quarter car model are used. In the quarter car model, mass of the spring is the sprung mass (M) and tyre & axles are the unsprung mass (m).

Vehicle suspension system is required to enable comfort to the passengers. It is present even in bullock cart to the trucks. Many researchers have carried out solutions for driver's and passengers ride comfort. The vibrations caused by the bumps and dips causes discomfort to the person's comfort and health. Minimization of the seat suspension is the need of the day. There are three types of suspension system: Passive suspension system, Active Suspension System and Hydraulic Active Suspension System. Passive suspension system is the oldest technique with uncontrolled springs and dampers. In passive system, fixed springs and shock observing dampers are used.

II. LITERATURE REVIEW

Passive suspension system works under a limited range [1 & 2]. Passive systems can reduce or increase the vibrations. If the spring stiffness is low, it leads to rider comfort but leads to large suspension deflection caused by low frequency vibrations. Many times deflection reach the movement limit [3 & 4]. Studies on passive stiffness are carried out where the contribution is around the accomplishment of comfort and reduced suspension deflection under vibrations in low frequency. Active suspension system works under varied operating conditions which has springs and force actuators. The main disadvantage in the active suspension system is that it requires large actuating power, which limits its usage. Later, Hydraulic active suspension system has gained popularity since it consists of springs, dampers and force actuators. It consumes comparatively less power compared to active suspension systems [5,6]. Controllable dampers such as Electro Rhelogical (ER) dampers and Magneto-Rhelogical (MR) dampers are investigated [7,8]. Recently various swarm intelligence techniques like particle swarm optimization, bee colony optimization, bacterial foraging, particle swarm optimization, bat algorithm [9, 10], differential evolution (DE)[11], Genetic algorithm and some hybrid optimization [12,13,14], etc has gained popularity in the field of automation. Many meta-heuristic algorithms are sensitive to initial values and it requires large amount of record memory and uses complex derivatives which makes computational complex. Harmony Search (HS) algorithm is derivative free and it uses gradient search. This algorithm can be easily implemented for various engineering optimization. The main reason of easy adaptation is it does not require initial values of parameter to be optimized[15][16].

Section III describes about the different types of quarter car model and section III details the general concept of Harmony search Algorithm. Online tuning of the proposed fractional order PID control for the active suspension system is dealth in section IV followed by the simulation results and Conclusion.

III. QUARTER CAR MODEL

Quarter car model captures the entire behavior of a full car. Figure 1 shows the suspension system for a quarter car model which have springs and damper. Comfort for the passenger and driver is accomplished by controlling the vibrations transmitted due to the dips and bumps present in the road surface. The primary goal of the suspension system is to maintain contact between wheels & road and isolate passenger & vibration due to road disturbances. In addition, the tyres should always be kept on the ground. Without suspension, the tyre will float off the ground when subjected to bumps and on touching the ground will cause discomfort to the passengers. With suspension system, wheels should follow the road ripples resulting in vertical wheel movement.

A. Passive Suspension System

Passive suspension system consists of springs and dampers with fixed characteristics. The dynamics of vertical motion is controlled and suspension system doesn't supply energy. Spring constant and damping constant are constant and control is impossible in passive suspension system but is used widely in almost majority of the vehicles. Degree of freedom is two. Figure 1 shows the block diagram of passive suspension system.

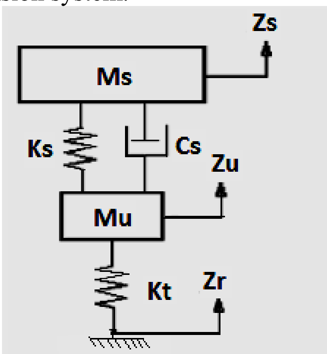

Figure 1. Passive Suspension System

Weight of car body, sprung mass is Ms, weight of wheel assembly is unsprung mass is Mu, spring constant is Ks (absorbs shock due to bump without creating oscillations in the vehicle) which is converted into potential energy, Cs is the coefficient of damper. Stiffness of tyre is Kt (tyre damping is neglected). Displacement in vertical direction of sprung mass, unsprung mass & terrain input disturbance are represented by variable Zs, Zu and Zr. Assumptions made are suspension spring and tyre stiffness are linear with tyre is contact with the road.

Mathematical first principle model is obtained using Newton's Second Law of Motion.

Equation of Motion for Sprung Mass for passive suspension system:

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) = 0 \qquad (1)$$

Equation of Motion for Unsprung Mass for passive suspension system:

$$M_u\ddot{Z}_u - K_s(Z_s - Z_u) - C_s(\dot{Z}_s - \dot{Z}_u) + K_t(Z_u - Z_r) = 0 \qquad (2)$$

State space representation of passive suspension system: State space modelling is obtained from the mathematical model of the suspension system. In passive suspension system, there is no actuation signal, so no input. The states are $x_1 = Z_s$, $x_2 = \dot{Z}_s$, $x_3=Z_u$, $x_4 = \dot{Z}_u$, and Z_r is the terrain road disturbance and the state space representation is:

$$\dot{x}(t) = Ax(t) + Bu(t) + EZ_r(t);$$
$$x(t) \in \Re^4, A \in \Re^{4x4}, B \in \Re^{4x1}, E \in \Re^{4x1}$$

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{-K_s}{M_s} & \frac{-C_s}{M_s} & \frac{K_s}{M_s} & \frac{C_s}{M_s} \\ 0 & 0 & 0 & 1 \\ \frac{K_s}{M_u} & \frac{C_s}{M_u} & -\frac{K_t + K_s}{M_u} & \frac{-C_s}{M_u} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{K_t}{M_u} \end{bmatrix} Z_r \qquad (3)$$

B. Active Suspension System

Passive suspension with hard suspension even though guarantees good vehicle handling, transfer road bumps to the vehicle and has fixed characteristics. If suspension is lightly damped will result in unstable condition during directional change even though it provides comfortable ride to the passengers. It provides the advantage of working under wide frequency range. Here, force actuators are used for automating the system under varied disturbance conditions and improving the quality of the ride. The schematic diagram of the active suspension system is given is figure 2. Damper is replaced with the force actuator.

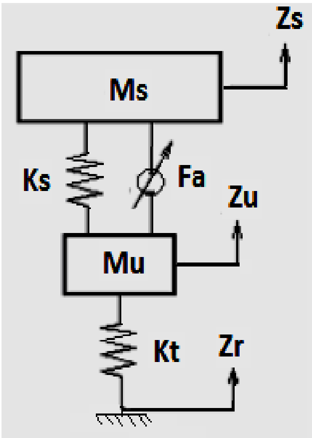

Figure 2. Active Suspension System

The mathematical model of the active suspension system is Equation of Motion for Sprung Mass in Active Suspension System:

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) = F_A \tag{4}$$

Equation of Motion for Unsprung Mass in Active Suspension System:

$$M_u\ddot{Z}_u - K_s(Z_s - Z_u) + K_t(Z_u - Z_r) = -F_A \tag{5}$$

All the parameters are the same as that of passive suspension system except F_A, electromagnetic actuator force which is the control input.

B1. State Space of Active Suspension System

In active suspension system, the control input is the force provided by the electromagnetic actuator i.e. $u = F_A$.

The state space representation of the active suspension system is:

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0\\ \frac{-K_s}{M_s} & 0 & \frac{K_s}{M_s} & 0\\ 0 & 0 & 0 & 1\\ \frac{K_s}{M_u} & 0 & -\frac{K_t+K_s}{M_u} & 0\end{bmatrix}\begin{bmatrix}x_1\\ x_2\\ x_3\\ x_4\end{bmatrix} + \begin{bmatrix}0\\ \frac{1}{M_s}\\ 0\\ -\frac{1}{M_u}\end{bmatrix}u + \begin{bmatrix}0\\ 0\\ 0\\ \frac{K_t}{M_u}\end{bmatrix}Z_r \tag{6}$$

TABLE 1. PARAMETERS OF QUARTER CAR SUSPENSION SYSTEM

Parameter	Value
Sprung Mass, Ms	282 (kg)
Unsprung Mass, Mu	45 (kg)
Spring Stiffness, Ks	17900(N/m)
Damping Constant, Cs	1000(N.s/m)
Tire Stiffness, Kt	165790 (N/m)

Differential equation (1) & (2) are simulated using MATLAB/SIMULINK environment. A two bump road disturbance is simulated which is shown in figure 3. Figure 4 shows the SIMULINK diagram of the passive suspension system and the hydraulic active suspension system.

The profile of the road disturbance is set by:

$$Zr = a\,\frac{1-\cos(8\pi t)}{2} \tag{7}$$

Where: $a = \begin{cases} 0.11 \text{ [m] for } 0.5 \le t \le 0.75 \\ 0.55 \text{ [m] for } 3.0 \le t \le 3.25 \\ 0, \text{ otherwise} \end{cases}$

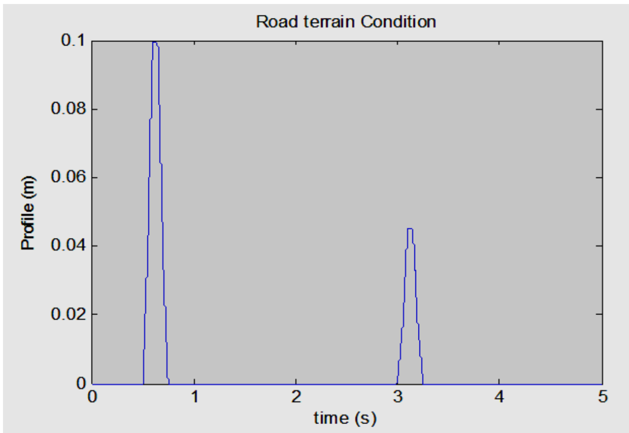

Figure 3. Road terrain Condition with two bumps.

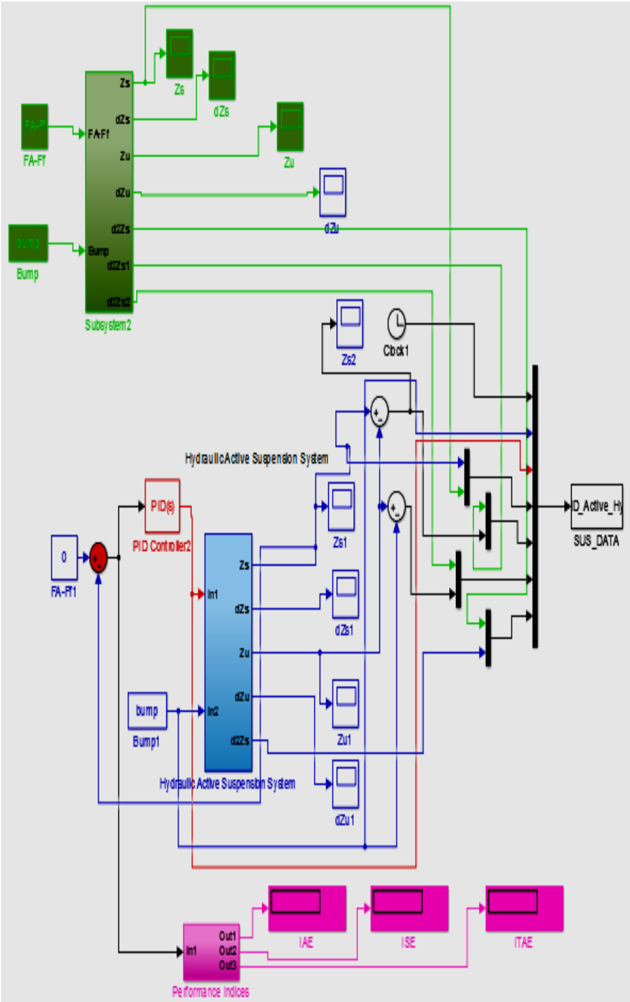

Figure 4. SIMULINK block diagram of passive and Active Suspension system.

IV. HARMONY SEARCH ALGORITHM

Harmony search algorithm (HS) is a music- based metaheuristic algorithm is developed by zong woo Geem et al. in 2001. Improvisation by Jazz musician is presented as an algorithm named as Harmony Search Algorithm. Jazz musician first plays any famous song and improvises by adjustment of the pitch ranges in the musical instrument aimed to deliver a pleasant harmony. This process is converted into an optimization algorithm with musician as decision variables and the improvised musical harmonies as objective function so as to provide upto the audience's standard. The repetition of the musician with the harmony memorized with improvisation in each turn is considered as the iteration of the optimization. The best composed pleasing harmony for optimal pitch ranges are considered as best pleasing harmony (final global objective value). The harmony memory is related to best fit individuals in GA, the best harmonies (solution vector) are stored in harmony memory and usage of harmonies in harmony memory is defined by the harmony memory accepting rate r_{accept} $\in [0,1]$. The pitch adjusted can be made by pitch bandwidth b_{range} and a pitch adjusting rate r_{pa}. The pitch solution x_{new} updated and stored in harmony memory after the adjusting action.

$$x_{new} = x_{old} + b_{range} * \delta \qquad (8)$$

where δ is a random number generator in the range of [-1,1] and usually pitch adjusting rate r_{pa} is used in the range 0.1 ~0.5.

Randomization is also like pitch adjusting but it used for the global search of system to bring new global solution.

The probability of randomization is,

$$P_{random} = 1 - r_{accept} \qquad (9)$$

The actual probability of pitch adjustment is

$$P_{pitch} = r_{accept} - r_{pa} \qquad (10)$$

V. ONLINE TUNING OF FRACTIONAL ORDER PID CONTROLLER FOR ACTIVE SUSPENSION SYSTEM

Fractional order PID controller parameters are tuned using the Harmony search algorithm. FOPID controller has five parameters to be optimized and the structure is given in equation (11):

$$C(s) = K_p + \frac{K_I}{s^{\lambda}} + K_D s^{\mu} \qquad (11)$$

where λ is the fractional order of integrator

μ are the fractional order of differentiator

K_P is the proportional gain

K_I is the integral gain

K_D is the derivative gain.

The controller parameters are real positive integer. If λ=1 and μ=1, a conventional PID controller is obtained. If λ=0, a fractional order PD controller is got and incase μ = 0, a fractional order PI controller is obtained.

Conventional techniques used for the controller parameter tuning involves complex mathematical derivations followed by optimization and approximations. As heuristic algorithms have gained popularity in near past, harmony search algorithm is attempted in this article. The controller parameters are optimized to satisfy the performance index. There are many performance indices such as minimum rise time, minimum settling time, minimum overshoot, minimization of Integral Absolute Error (IAE), Integral Square Error (ISE), Integral Time Absolute Error (ITAE), etc. ITAE involves combined optimization of overshoot, rise time and settling time. It is computed by taking the integral of the error and multiplying with clock signal and integrating it for the duration of the travel. Harmony search algorithm is used to obtain the FOPID controller parameters with ITAE as the objective function.

The objective function is represented by:

$$ITAE = \int_0^T t\,|e(t)|\,dt \tag{12}$$

The flow chart of harmony search is shown in figure 5. Figure 6 shows the block diagram of the harmony search algorithm optimised fraction order PID controller for active suspension system. From the block diagram, it is evident that the input to the active suspension system is the force actuation and the bump in the road are the disturbance which will excite acceleration to the vehicle and causes discomfort to the passengers and driver. The states of the active suspension system are the displacement of the sprung mass and unsprung mass, velocity of sprung mass and unsprung mass and acceleration. The desired performance is to minimize the displacement of the seat. When a bump is present in the road, it results in acceleration and displacement both in the wheel and the seat. If there exists a displacement in the seat, it will result in passenger's discomfort. The displacement of the seat should be reduced as much as possible. It cannot be nullified because of the physical constraints of the mechanical components present in the suspension system.

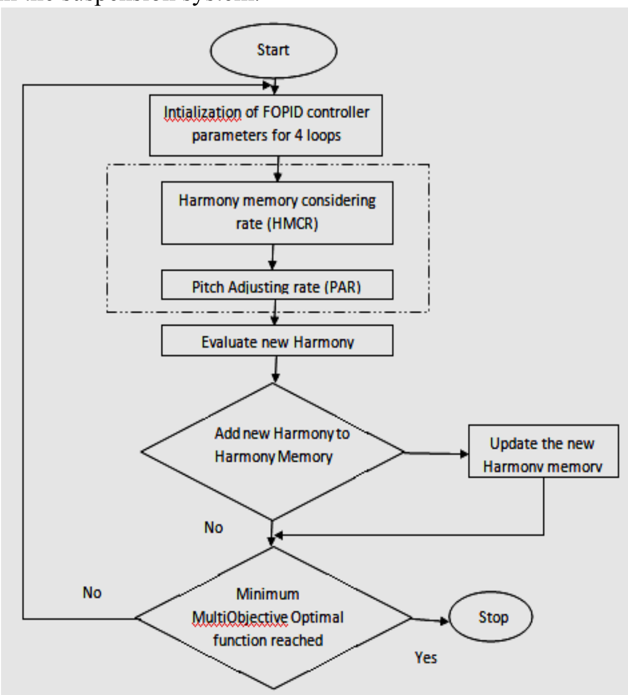

Figure 5. Flow chart of HS algorithm for FOPID controller parameter optimization

The displacement between the sprung mass and the unsprung mass needs to be reduced to zero which generates error. The absolute value of this deviation is multiplied with the time data and integrated to get the ITAE which is passed to the global optimizer, the harmony algorithm. The harmony algorithm determines the FOPID parameters and passes on to the closed loop control of active suspension system. The process is repeated for the specified number of iterations or till the global optimal solution is reached.

Parameters used for the Harmony search algorithm are:

Number of variable = 10

Maximum Iteration = 100

Harmony memory size = 16

Harmony consideration rate = 0.9

Minimum pitch adjusting rate = 0.4

Maximum pitch adjusting rate = 0.9

Minimum bandwidth = $1e^{-4}$

Maximum bandwidth = 1

Range of controller parameters is selected as

- $2.5e^3 < K_P < 3e^3$
- $0.5 < K_I < 0.7$
- $3e^4 < K_D < 5e^4$
- $0 < \lambda < 1$
- $0 < \mu < 1$

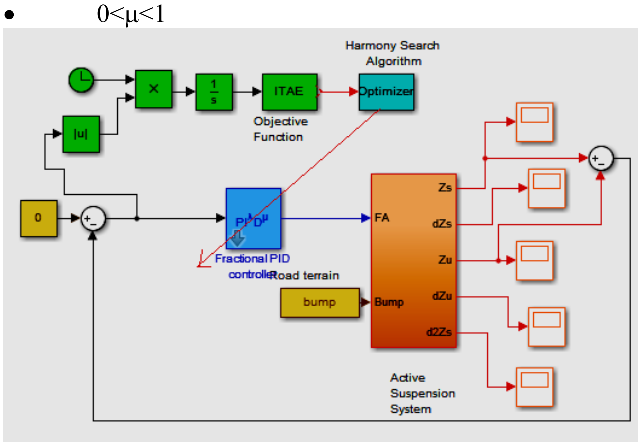

Figure 6. Harmony search algorithm optimized FOPID controller for active suspension system.

VI. SIMULATION RESULTS

Harmony Search Algorithm is used to tune the optimal FOPID controller parameters. Performance is analyzed for the two bump disturbance. Table 2 shows the controller parameters tuned using Harmony search algorithm.

TABLE 2. CONTROLLER PARAMETERS TUNED USING HARMONY SEARCH ALGORITHM

PID controller parameters for Active Suspension System	FOPID controller parameters for Active Suspension System
P = 7417255.5 I = 81998628.5 D = 41524.15	K_P = 2966.2 K_I = 0.5982 K_D = 39223 λ = 0.1625 μ = 0.7772

The tuned PID and FOPID controller generates the control output which is the actuator control force for the suspension system shown in figure 7. On analysis of the actuator force, it is evident that whenever there is a bump the actuator force changes drastically. The FOPID controller output is smoothed out compared to the conventional PID controller output and the amplitude variation is also reduced.

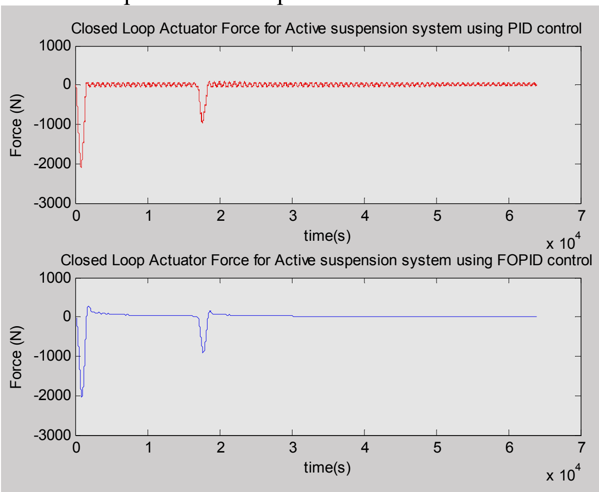

Figure 7. Actuator force for Active Suspension system.

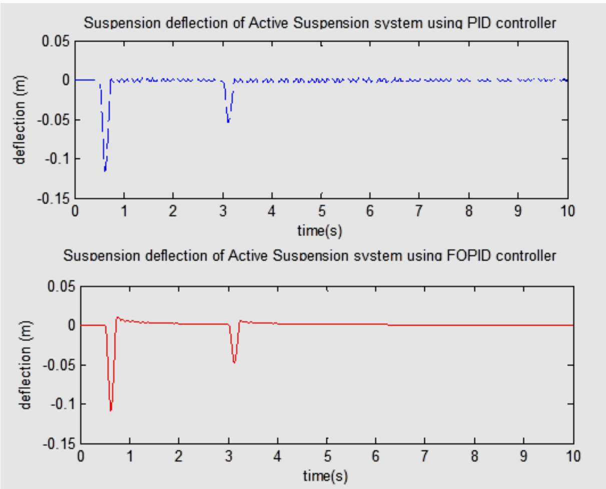

Figure 8. Suspension deflection of Active Suspension system.

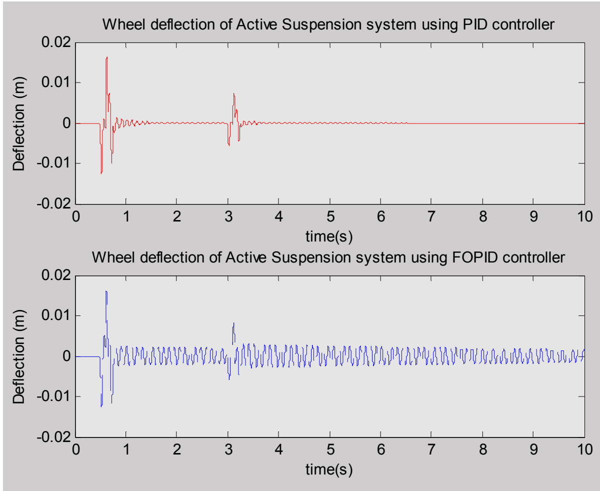

Figure 9. Wheel deflection of Active Suspension System

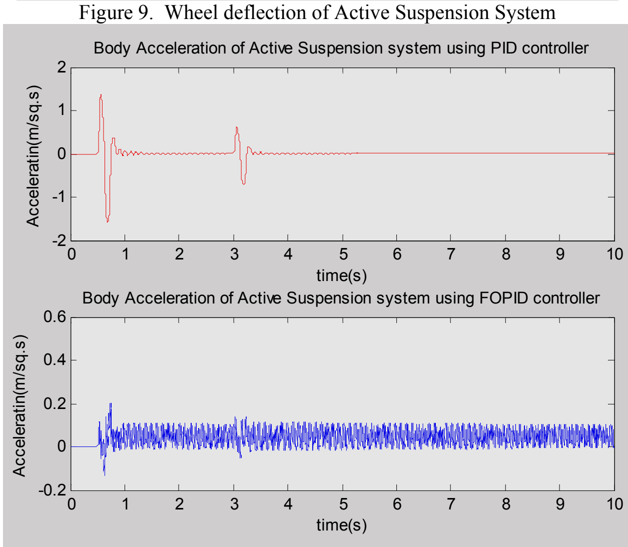

Figure 10. Body Acceleration of Active Suspension System

On investigation of the suspension deflection shown in figure 8, the deflection is very smooth for FOPID controller and oscillatory for the PID controller eventhough the wheel deflection is agitated in the FOPID scheme shown in figure 9. The body acceleration shown in figure 10 is vibrating but with a reduced amplitude of -0.01m to 0.2m for FOPID controller compared to the amplitude of -1.5 m/s^2 to 1.5m/s^2 for PID controller. The displacement felt in the seat is very minimal thereby providing comfort to the passenger/driver. Body acceleration is comparatively reduced as shown in figure 10.

VII. CONCLUSION

In the proposed work, Harmony Search algorithm is used for PID and FOPID tuning for an Active Suspension system. Harmony Search algorithm uses ITAE as the objective function. The tuned controller gives the optimal tuning parameter which considerably reduces the seat displacement thereby guaranteeing comfort to the passenger compensating on the wheel deflection.

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