World of Computer Science and Information Technology Journal (WCSIT)
ISSN: 2221-0741
Vol. 3, No. 2, 26-31, 2013

An Optimized PID Control Strategy For Active Suspensions Applied To A Half Car Model

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Abstract— In this paper, we propose a new control strategy for the active control of a hydraulically controlled half-car active suspension system. Our study proposes a special construction of the suspension system where the hydraulic actuator is to be placed in series with the conventional passive one to form a special case of low-frequency active suspension. The full dynamics of the electro-hydraulic servo-valve and hydraulic actuator were employed. Response of proposed control strategy has been tested for different road profiles and riding conditions including car chassis rolling effect when cornering and pitch movements when braking/ acceleration. Results have shown superior performance of our modified controller over passive suspensions and many other controllers of previous studies. Simulations show that our proposed controller provide better passenger comfort as it lowers maximum body acceleration by 94.3% and with reduction in body travel by 98.8% of that of passive one. Also, our proposed control strategy has shown better road handling and car stability over a whole range of road and inertial disturbances.

Keywords- Half-car active suspension; Electro-hydraulic servo actuator; PID control ; Optimization.

I. INTRODUCTION

Active suspensions have attracted interest for research in the few recent decades especially after the large integration in electronic controllers for industrial applications. Also, control engineers have a wide space to contribute in this field by applying modern automatic control schemes with the new developments in active suspension components, i.e., actuators, sensors, precise low-cost electronics and fast microprocessors, [1].

Developments of vehicle suspensions are made to achieve some performance characteristics in order to have a good suspension system. Main characteristics are shown in [2] and summarized in [3] to be; a) regulation of body movement by isolating body from road bumps in addition to minimize inertial disturbances resulting from vehicle cornering and control of body pitch and bounce and, b) regulation of wheel hop (vehicle handling) and, c) force distribution of car weight between the four wheels to get good handling characteristics needs all-time wheel-tire contact. Usually, suspension design should compromise between these conflicting trade-offs and also is related to cost, [3].

Active suspensions have additional hydraulic actuators to the passive elements of conventional ones. These actuators give the suspension the ability to generate needed forces to improve performance characteristics of vehicle ride for all driving situations, [4].

There are two types of commonly recognized active suspension structures [5], low bandwidth and high bandwidth, as shown in Fig. 1. High bandwidth refers to regulation of both the sprung mass of the suspension for vibrations below 3Hz and the unsprung mass, wheel assembly, for vibrations around 12 Hz [3], [5]. While the low bandwidth suspension regulates only the sprung mass for its low frequency vibrations and the high frequency vibrations are controlled by the passive damper of the system, [5]. The structure of this paper will be as follows; In the following section we introduce model of proposed suspension structure for a half-car with employing full dynamics of the hydraulic actuator for simulation purposes. Next we show proposed suspension control methodology followed by controlled system simulation and discussion for results obtained, then we come to our conclusion.

II. PROPOSED SUSPENSION MODEL

Our proposed system is a special structure of the low bandwidth active suspension system. We propose that a hydraulic actuator to be placed above the sprung mass, which will be assumed a small value relative to car chassis mass.

As shown in Fig. 2, car body, m_c, is to be supported only by the hydraulic actuators while the passive suspension components will remain the same. In Fig. 2 the sprung masses, m_f and m_r, represent small disc-like masses with appropriate mechanical hardness to support the hydraulic actuator. These masses will be helpful in slowing road disturbances transfer to car chassis and this would give our controllers the time to be able to track road profile. Unsprung masses, m_{uf} and m_{ur} represent front and rear wheel assemblies, respectively. The springs, c_r and c_s, the dampers, b_f and b_r, and springs, c_{tf} and c_{tr}, represent stiffness of passive springs, stiffness of passive dampers, and stiffness of

Publication fees of this paper are funded through *Creatives* project held at IUG through a grant from Welfare Association, Arabic Monetary Fund, and Islamic Development Bank.

pneumatic compressed tires, for front and rear assemblies, respectively. The variables u_f, x_{uf}, x_{sf}, x_f represent road disturbance, wheel travel, disc travel, and body travel of the front side, respectively and variables u_r, x_{ur}, x_{sr}, x_r represent road disturbance, wheel travel, disc travel, and body travel of the rear side.

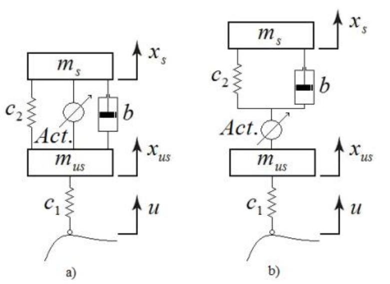


Figure 1. Active suspensions: a) high bandwidth (parallel), b) low bandwidth (series).

The forces F_f and F_r generated between sprung masses and car body is built up by pressure difference across a hydraulic actuators. Hydraulic actuators are taken here to be a four-way critically lapped spool valve controlled by a flapper valve with force feedback, the same as in [4], [6], and [7]. The force generated by each actuator can be written as

$$F_i = A.P_{Li}, \quad i = f, r \tag{1}$$

where A is the cross sectional area of the actuator cylinder and P_{Li} is pressure drop across cylinders pistons. As shown in [8], the change in pressure drop can be written as

$$\frac{V_t}{4\beta_e}\dot{P}_{Li} = Q_{Li} - C_{tp}P_{Li} - A(\dot{x}_i - \dot{x}_{si}), \quad i = f, r \tag{2}$$

where V_t is total actuator volume, β_e is the effective bulk modulus, Q_{Li} is hydraulic load flow for the two sides, and C_{tp} is the total leakage coefficient of the piston. The load flow is given by

$$Q_{Li} = \text{sgn}[P_s - \text{sgn}(x_{vi})P_{Li}]C_d w x_{vi}\sqrt{\frac{1}{\rho}\left|P_s - \text{sgn}(x_{vi})P_{Li}\right|}, \quad i = f, r \tag{3}$$

where C_d is discharge coefficient, w is spool valve area gradient, x_{vi} is displacement of spool valves for each actuator, ρ is hydraulic fluid density, and P_s is hydraulic supply pressure.

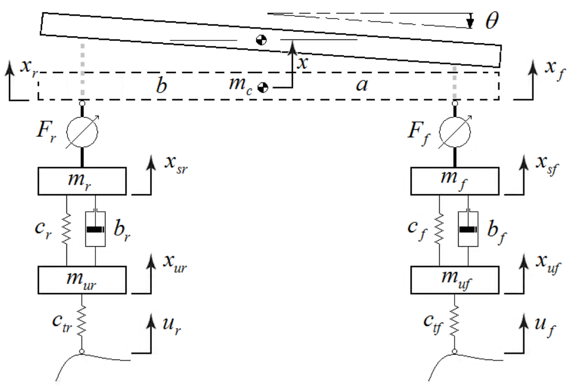


Figure 2. Proposed half-car active suspension system.

The pressure drop across actuators can be described by

$$\dot{P}_{Li} = -\beta P_{Li} - \alpha A(\dot{x}_i - \dot{x}_{si}) + \gamma x_{vi} w_s, \quad i = f, r \tag{4}$$

where,

$$\alpha = \frac{4\beta_e}{V_t}, \beta = \alpha C_{tp}, \gamma = \alpha C_d w\sqrt{\frac{1}{\rho}},$$
$$and\, w_s = \text{sgn}[P_s - \text{sgn}(x_{vi})P_{Li}]\sqrt{\left|P_s - \text{sgn}(x_{vi})P_{Li}\right|} \tag{5}$$

The spool valve displacement is controlled by a voltage or current input r_i to the servo valve. The dynamics of the servo valve can be approximated by a linear filter as

$$\dot{x}_{vi} = \frac{1}{\tau}(-x_{vi} + r_i), \quad i = f, r \tag{6}$$

Now, following [9] the mathematical model of the proposed half-car suspension using Newton's second law, along with the full dynamics of the hydraulic actuator shown in (1) to (4) will be:

for unsprung masses (wheels)

$$\ddot{x}_{ui} = \frac{1}{m_{ui}}[c_i(x_{si} - x_{ui}) + b_i(\dot{x}_{si} - \dot{x}_{ui}) - c_{ti}(x_{ui} - u_i)], \quad i = f, r \tag{7}$$

for sprung masses (discs)

$$\ddot{x}_{si} = \frac{1}{m_i}[-c_i(x_{si} - x_{ui}) - b_i(\dot{x}_{si} - \dot{x}_{ui}) + AP_{Li}] \tag{8}$$

while the dynamics of motion for the front side of car body can be described by

$$\ddot{x}_f = \frac{1}{m_c}[F_f + F_r - am_c\ddot{\theta}] \tag{9}$$

and the dynamics of motion for the rear side of car body is

$$\ddot{x}_r = \frac{1}{m_c}[F_f + F_r + bm_c\ddot{\theta}] \tag{10}$$

where F_f and F_r are shown in (1) and (4), a is the distance from the front axle to center of gravity of car body, b is the distance from the rear axle to center of gravity of car body, and θ is body pitch angle and can be described as

$$\ddot{\theta} = \frac{1}{J_y}[-aF_f + bF_r] \tag{11}$$

where J_y is moment of inertia of car body. Note that positive direction of pitch angle is assumed to be clock wise. Finally, travel of center of gravity of car body, bounce, can be calculated by

$$x = x_f + a\sin\theta = x_r - b\sin\theta \tag{12}$$

Now the full dynamics of the half-car model is described in the equations (1) to (12).

III. Control Methodology And Design

The proposed suspension structure gives us the advantage of that, at any fault in active suspension, conventional suspension will remain operating. In this section we show design procedure and justify our proposed structure. Our analysis and control design is applied to quarter-car model, then the control scheme is applied in the same manner to every wheel. The proposed structure for each quarter-car suspension forms two series components; passive suspension and hydraulic actuator. The main idea of this proposed suspension is to drive the hydraulic actuator as a displacement compensator for variation of the passive suspension deflection due to road or inertial disturbances.

IV. PID control

This control scheme is applied to regulate car body displacement by closing our control loop with the value of car body travel only. Our reference input for car body travel is *zero* and road variations or load variations are considered as disturbances in the control loop. Schematic diagram of this control is shown in Fig. 3 below

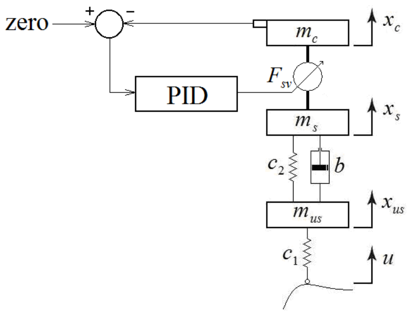


Figure 3. PID control scheme for car body.

Our control law for this case is

$$r=K_{sv}[-K_p x_c(t)-K_i \int x_c(t)dt-K_d \frac{dx_c(t)}{dt}] \qquad (13)$$

Assuming that our reference value for car body displacement is *zero*. Setting of control law values shown in equation (13) is performed by two ways throughout this paper; first by tuning conventional PID control parameters by trial and error, and the second is by the use of *SIMULINK Design Optimization* tool.

This tuning is performed using the *SIMULINK Design Optimization* tool supported by MATLAB®. Procedure of tuning PID parameters tuning is done by setting our desired car body level to some value, say 0.01 m, and performing iterative optimization problem for achieving desired step response characteristics, as shown in Fig. 4. This is done using *Signal Constraint* block with specifying our tuned parameters to be the PID controller gains K_p, K_i, and K_d. Note that we have chosen to have a rapid response with very small rise and settling times in order to cut vibration disturbances as fast as possible. But this forced us to widen the accepted overshoot range up to 40%.

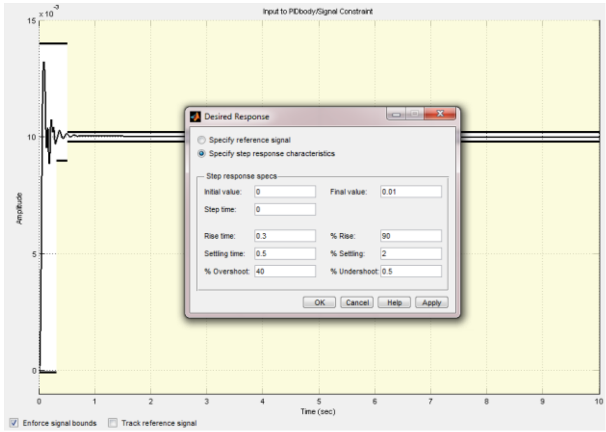


Figure 4. Desired step response for optimized PID control.

Desired response is set to have the following bounds shown in Table I below

TABLE I. Desired step response characteristics for optimized PID control.

Characteristics			
Rise time (sec)	0.3	% Rise	90
Settling time (sec)	0.5	% Settling	2
% Overshoot	40	% Undershoot	0.5

V. Simulation Results

Our proposed suspension with control strategy discussed is compared with passive suspension response for the same input. Using standard parameters and values taken from [7] and used in [4] and [6] with the half-car model parameters shown in [9]:

$$\begin{aligned}
&m_c = 575kg,\quad m_{sf} = m_{sr} = 20kg,\quad m_{uf} = m_{ur} 60kg,\\
&c_{tf} = c_{tr} = 190000 N/m, c_{sf} = c_{sr} = 16812 N/m,\\
&b_f = b_r = 1000 N/m/s,\quad J_y = 769 kg/m^2, a = 1.38m,\\
&b = 1.36m,\quad \tau = 1/30\sec,\quad P_s = 10342500 Pa,\quad A = 3.35\times 10^{-4} m^2,\\
&\alpha = 4.515\times 10^{13} N/m^5,\quad \beta = 1\sec^{-1},\quad \gamma = 1.545\times 10^{9} N/m^{5/2} kg^{1/2}.
\end{aligned} \qquad (14)$$

(3.12)

Note that value of sprung mass is set by trial and error and cannot be eliminated as it is important for our proposed suspension stability. Also, it is not justified to increase it bigger than this value due to extra weight of the car. Optimal selection of sprung mass value may be a separate research topic.

First we test system performance for a single road bump represented as

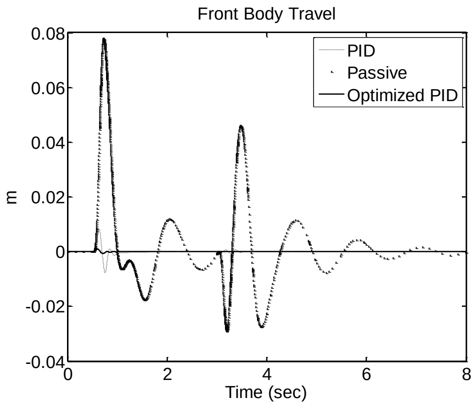
$$u_f(t) = \begin{cases} h(1-\cos 8\pi t), & 0.5 \le t \le 0.75 \\ 0, & otherwise \end{cases} \quad m \qquad (15)$$

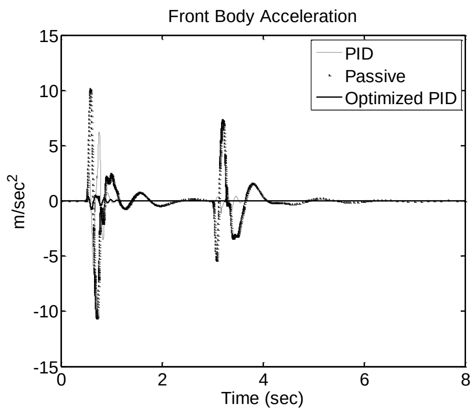
and

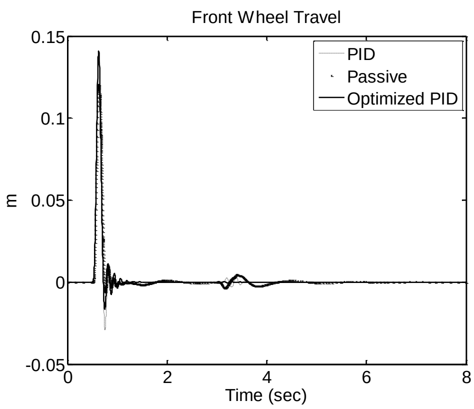
$$u_r(t) = \begin{cases} h(1-\cos 8\pi t), & 3 \le t \le 3.25 \\ 0, & otherwise \end{cases} \quad m \qquad (16)$$

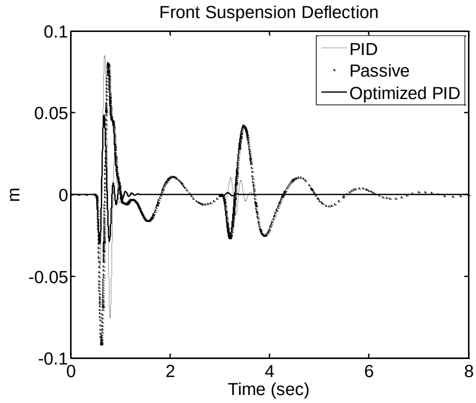
Note that the bump to the rear wheel is a delayed version of that of the front one. Bump height is said to be 11 cm, i.e. *h=0.055*. Since Many previous studies showed that most important performance index for a good suspension system are; minimization of body vertical acceleration which directly affects passenger comfort and, minimization of suspension deflection, i.e. relative displacement between sprung and unsprung masses which affects handling, [4], [6], [7], for each case we examine body travel, body acceleration, wheel travel, and suspension deflection (for our proposed active suspension, deflection is denoted by passive component deflection. This performance index is important because reaching suspension limits may damage vehicle components in addition to its discomfort effect to passengers, [4].

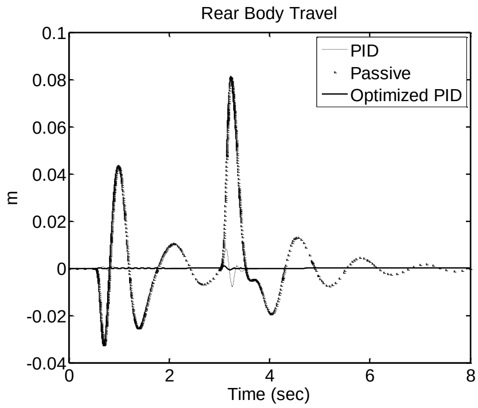
Note that throughout this paper, displacement of servo valve spool is limited to ±1cm. Figure 5 shows comparison between passive suspension and our proposed active suspension responses and we note that acceleration for both front and rear sides has been reduced by almost 94.3% and body travel by almost 98.8% compared to passive suspension. Also, handling performance still accepted.

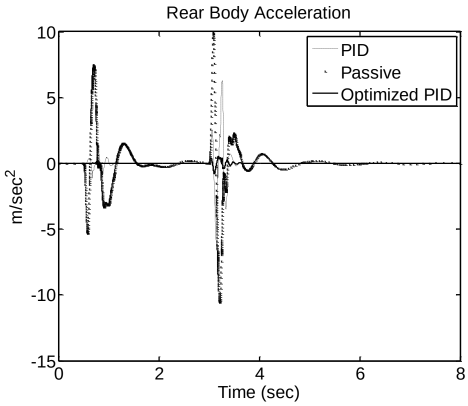


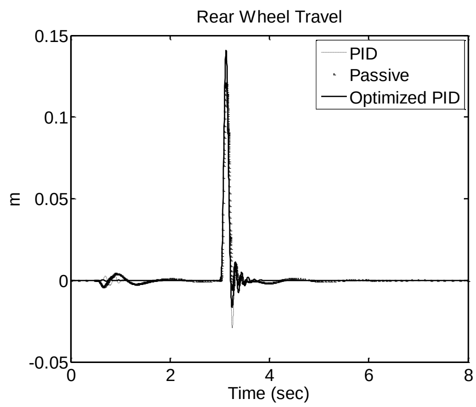












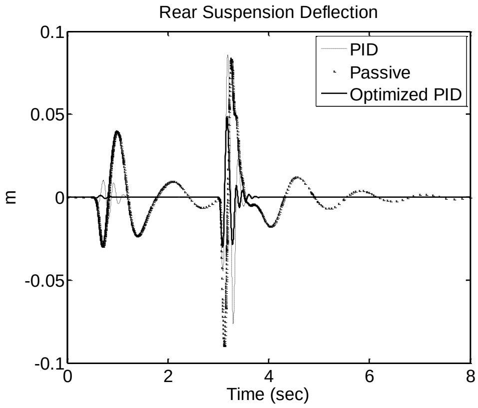


Figure 5. Half-car response of PID control schemes for an 11 cm bump.

Our proposed suspension performance is tested for inertial disturbances by applying a force to the body mass acting downwards for some time period, [10], and this force can be generated in real in the cases of cornering, braking, and accelerating. We first simulate car body cornering when inertial forces are acting on both front and rear sides of the body. Thus, inertial forces are set as follows

$$IF_f(t)=IF_r(t)=\begin{cases} 300(1-\cos 2\pi t), & 1\le t\le 1.5 \\ 600, & 1.5\le t\le 5.5 \\ 300(1-\cos 2\pi t), & 5.5\le t\le 6 \\ 0, & otherwise \end{cases} \qquad (17)$$

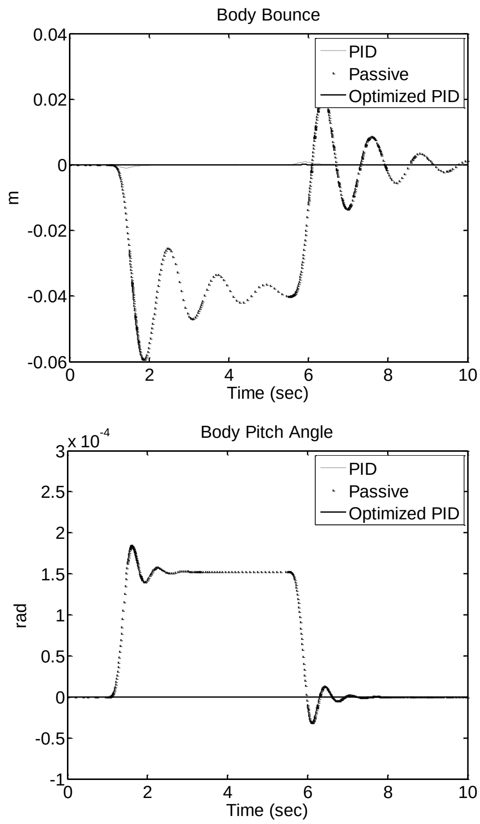


Figure 6. Half-car response due to inertial forces generated with car cornering situation- PID control schemes.

Note that both our PID and optimized PID controllers gave excellent response in preventing car body roll in a smooth manner. Also, our proposed system with control strategy are tested for *braking* condition by applying 600N to the front side of car body, say in the period [1,6] sec, and then we simulate car *acceleration* condition where inertia force is applied to the rear side, say in the period [9,14] sec. The input forces are applied to model dynamics and are given values according to the following functions

$$IF_f(t)=\begin{cases} 300(1-\cos 2\pi t), & 1\le t\le 1.5 \\ 600, & 1.5\le t\le 5.5 \\ 300(1-\cos 2\pi t), & 5.5\le t\le 6 \\ 0, & otherwise \end{cases} \qquad (18)$$

$$IF_r(t)=\begin{cases} 300(1-\cos 2\pi t), & 9\le t\le 9.5 \\ 600, & 9.5\le t\le 13.5 \\ 300(1-\cos 2\pi t), & 13.5\le t\le 14 \\ 0, & otherwise \end{cases} \qquad (19)$$

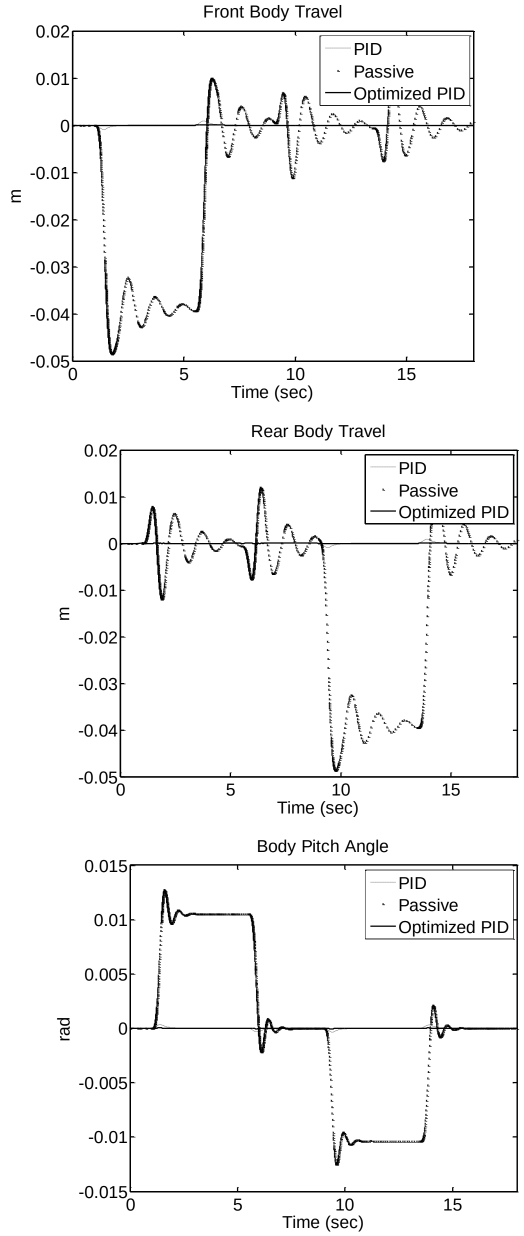


Figure 7. Half-car response due to inertial forces generated in car body braking(t=[1,6]sec) and acceleration(t=[9,14]sec) situations.

VI. Conclusion

In this paper we have introduced a new proposed active suspension structure, a special prototype of low bandwidth suspension, in order to make use of conventional techniques in active suspension control problem. Performance of our proposed structure is enhanced by a simple control strategy, based on optimized PID scheme.

Extra weights, sprung masses, were added to the vehicle because the limitation of performance for conventional PID control of electro-hydraulic actuator. The proposed control strategy has shown superior performance for the active suspension for different ride situations of bumps and inertial loads. Results have shown numerous reduction in body acceleration and good handling characteristics. Also, suspension deflection is reduced and this would give the chance to decrease passive suspension length.

Acknowledgment

The authors would like to thank *Creatives* project at IUG for publishing this paper through a grant from Welfare Association, AMF, and IDB.

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