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Enhanced the Damping Efficiency of Hydraulic Regenerative Suspension System Comparing with the Active and Passive Suspension Systems

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Abstract
With the growing number of automobiles on the road and the introduction of electric and hybrid vehicles, it has become critical to lower each vehicle's energy usage., as several renewable methods are used on a large scale for harvesting energy from the various systems of the vehicle, including energy harvesting using suspension systems. In this paper, the power generation system was derived and modeled using the non-linear suspension system for a full model of the vehicle using the MATLAB-SIMULATION, where the system consisting of a hydraulic cylinder, piston, a spool valve, and hydraulic motor. The system's performance characteristics were compared to those of passive and active suspension systems, and the system's performance was found to be comparable to that of the active system. Three different pressure levels (10, 30, and 50 bar) were used, along with three different input signals (random, sinusoidal, and square) and a fractional order proportional integral derivative controller (FOPID). The results showed the performance characteristics of the regenerative suspension systems can be close to the performance characteristics of active suspension systems.
Keywords: Hydraulic Suspension, Active Suspension, FOPID Controller, Full body Modelling.

Introduction

There are many innovative methods of energy harvesting, such as hybrid vehicles, electric vehicles, fuel cell hybrid vehicles, regenerative braking, exhaust heat energy harvesting, variable compression ratio engines, and regenerative suspension system [1,2]. In this paper, a regenerative suspension system has been studied to harvest energy and provide comfortable rides for passengers compared to the passive and active suspension system. The electro-hydraulic system has been used and controlled by Fractional Order proportional integral derivative controller (FOPID) by using Matlab/Simulink software.

Power generation systems of suspensions can be divided briefly into two parts, the first part is a direct system which includes an electromagnetic system, Piezoelectric material, and a combination of MR damper and electromagnetic system. The indirect system, which comprises of a rack and pinion mechanism, a ball screw mechanism, a hydraulic mechanism, and a pneumatic mechanism, is the second component. This system converts reciprocating (up and down) motion to rotational motion. The harvesting of energy from vibrations is a frequent use of piezoelectric crystals or smart materials. [3,4,5]. The electrohydraulic and electromagnetic are considered the major and main fields in the suspension energy harvesting systems [6]. Numerous researchers have investigated energy harvesting from suspension systems, such as Demetgul and Guney (2017) [7], who propose for a hydraulic regenerative shock absorber that collects energy from both hydraulic fluid and shock body motion. In comparison to traditional suspension systems, Jia Mi et al. (2017) [8] suggested incorporating HESA into a bogie system for railway cars. The simulation findings indicate that the system is capable of successfully reducing the carriage's vibration while retaining a high potential for vibratory energy recycling. Ahmad and Monis (2017) [9] have replaced the shock absorber in a common suspension system with a hydraulic cylinder, hydraulic motor, and dynamo to harvesting the energy that generates from the hydraulic pressure and made a comparison with shelf and pinion regenerative suspension system. To study the damping performance and energy regeneration for 4DOF, Zou et al. (2018) [10] offered a comparison between a passive suspension system and a hydraulic interconnected suspension system (HIS-HESA) based on a hydraulic energy regenerative shock absorber. B. Lafarge et al. (2018) [11] proposed a suspension model based on a bond graph with an integrated piezoelectric beam because it provides the optimal configuration for the energy harvester. Bai and Guo (2018) [12] demonstrated an improved active suspension system model. This study completely evaluated the dynamics of a hydraulic actuator in an active suspension system in comparison to the current active suspension system model. Magdy Naeem et al. (2018) [13] have developed a system for Hydro-Pneumatic Energy Harvesting Suspension (HPEHS). The concept incorporates a hydraulic rectifier to maintain a one-way flow direction to maximize power production from the suspension system's vertical oscillation and to accomplish vehicle control and comfort. Zeyu Sun et al. (2019) [14] investigated the energy feedback potential of the hydraulic linked energy-regenerative suspension, a technique for collecting and reusing energy. When experimental and simulation results are compared, it is obvious that the hydraulic linked power feedback suspension has a significant energy feedback component, which provides ideas for novel model suspension architectures as well as energy recovery and use in vibration systems.

Shiying Li et al. (2019) [15] made the following proposal: A revolutionary energy-harvesting variable/constant damping suspension system using an electromagnetic damper powered by a motor. The proposed approach aims to accomplish the following: vibration energy from the vehicle's suspension system is not only gathered but also regulated to be stored in the battery for future use. Junyi Zou et al. (2019) [16] investigated the modeling and ride characteristics of a seven-degree-of-freedom whole vehicle suspension model coupled with a hydraulically linked suspension based on energy regeneration shock absorbers. Zhanwen Wang et al. (2019) [17] proposed that range-extended electric cars use twin-screw gearboxes to provide high-efficiency regenerative shock absorbers. Shuang Liu et al. (2020) [18] proposed a novel adaptive dynamic surface control technique that incorporates both known and unknown suspension parameters for the electrohydraulic actuator, as well as a dynamic model of the vehicle's active suspension, in order to increase the vehicle's control accuracy.

Modeling of the Entire Vehicle Suspension System Using a Hydraulic Actuator

The equations of regenerative hydraulic suspension systems have been derived for the modeling and ride analysis of a 7-DOF full vehicle suspension model.

The system consists of a hydraulic cylinder with a piston rod, servo valve, and hydraulic motor as shown in Figure (1).

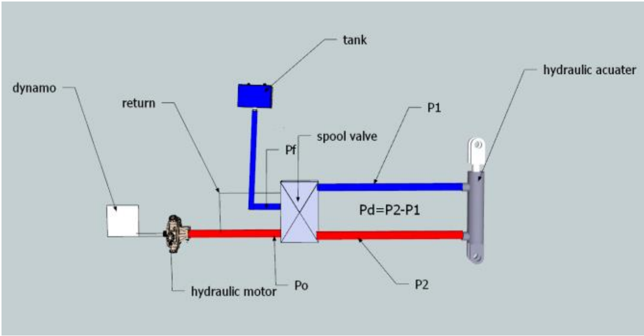


Figure 1. The hydraulic regenerative suspension system

The spool valves have been used for controlling the hydraulic actuators. The general flow rate equations through orifices are given by

$$Q_{Ldi} = C_{dc}\omega x_{svi}\sqrt{\frac{1}{\rho}(P_{fi} - \mathrm{sgn}(x_{svi})P_{di} - P_{oi})} \quad \text{Eq.(1)}$$

Change in pressures concerning time is given by:

$$\dot{P}_{di} = -\delta P_{di} - \Psi A_p \dot{x}_{pi} + \Psi Q_{Ldi} \qquad \text{Eq. (2)}$$

The displacements of the spool valves (x_{svi}) that controlled by input signals (u_{mi}) at a constant time (τ) are

$$\dot{x}_{svi} = \frac{1}{\tau}(u_{mi} - x_{svi}) \qquad \text{Eq. (3)}$$

The hydraulic actuator forces are:

$$F_{hyi} = A_p P_{di} \qquad \text{Eq. (4)}$$

The full vehicle motion has seven degrees of freedom as shown in Figure (2).

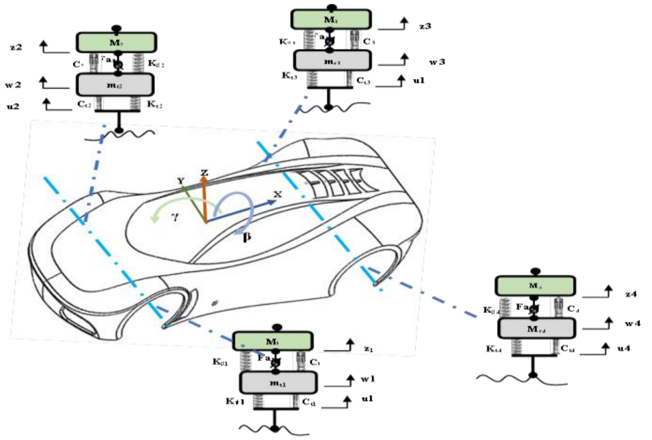


Figure 2. Full Vehicle suspension system [19].

The equations of this modeling [20,21] as:

The motions of the sprung mass

Vertical motion

$$f_{Ksi} = K_{si}(z_i - w_i) + \mu K_{si}(z_i - w_i)^3 \qquad Eq.(5)$$

$$f_{Ci} = C_i(\dot{z}_i - \dot{w}_i) + \mu C_i(\dot{z}_i - \dot{w}_i)^2 \operatorname{sgn}(\dot{z}_i - \dot{w}_i) \qquad Eq.(6)$$

$$F_{ai} = F_{hyi} - F_{fri} \qquad Eq.(7)$$

Rolling motion

$$J_x\ddot{\gamma} = (f_{Ks1} - f_{Ks2} - f_{Ks3} + f_{Ks4})\frac{b}{2} + (f_{C1} - f_{C2} - f_{C3} + f_{C4})\frac{b}{2} + (f_{a3} - f_{a1} + f_{a2} - f_{a4})\frac{b}{2} + T_x$$

Pitch motion

$$J_y\ddot{\beta} = (f_{ks3} + f_{ks4})l_2 - (f_{Ks1} + f_{Ks2})l_1 + (f_{C3} + f_{C4})l_2 - (f_{C1} + f_{C2})l_1 + (f_{a1} + f_{a2})l_1 - (f_{a3} + f_{a4})l_2 + T_y$$

The vertical motions equation of unsprung masses can be rewritten as

$$m_i\ddot{w}_i = C_i\dot{z}_c - 0.5bC_i\dot{\gamma} - C_il_i\dot{\beta} + \mu C_i(\dot{z}_i - \dot{w}_i)^2 \operatorname{sgn}(\dot{z}_i - \dot{w}_i) - (C_i + c_{ti})\dot{w}_i + c_{ti}\dot{u}_i + K_{si}z_c - 0.5bK_{si}\gamma - K_{si}l_e\beta + \mu K_{ti}(z_i - w_i)^3 - (K_{si} + k_{ti})w_i + k_{ti}u_i - A_pP_{Li} + f_{fri}$$

Eq. (10)

Where : e = 1 for i=(1,2) and e = 2 for i= (3,4)

Results and Discussion

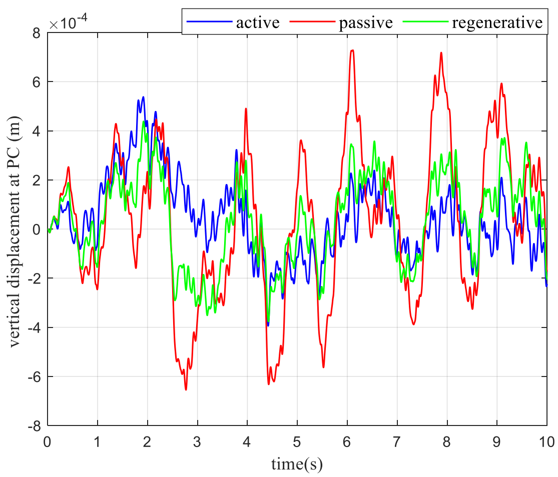
The PIλDμ was employed and adjusted using Evolutionary Algorithm (EA) in this study to manage the entire car regenerative suspension systems using hydraulic actuators. Using the MATLAB SIMULATION program, the simulation results indicated the road profiles, vertical displacement at the center of gravity, pitch angles, and roll angles with reaction time. The simulation was carried out for three signal inputs (random, sine, and square) waves with amplitudes of 10 mm over a period of 10 seconds, taking into account three levels of hydraulic pressure (10, 30, and 50) bars in the system.

Figure (3) compares the vertical displacements of the regeneration (with a regulated system using a FOPID controller system when exposed to 10, 30, and 50 bars) and active and passive systems with random signal input. The result showed the vertical displacement of the center of gravity of the vehicle with using the regenerative system be better than the passive suspension system and close to that in the active system when increasing output pressure of the regenerative suspension system.

The comparison of the pitch angle of the passive, regenerative, and active with random road profiles is shown in figure (4). The comparison of the rolling angle of the passive, regenerative, and active with random road profile as shown in figure (5).

The figure (6) shows the comparison of the vertical displacement of the passive, regenerative and active with a single road profile. Figure (7) depicts a comparison of the pitch angles of the passive, regenerative, and active road profiles with a single road profile.The figure (8) shows the comparison of the roll angle of the passive, regenerative and active with a single road profile.

The figure (9) illustrates the vertical displacement of the passive, regenerative, and active road profiles in relation to the square road profile. The pitch angles of the passive, regenerative, and active road designs are compared to the pitch angles of the square road profile in Figure 10. The figure (11) shows the comparison of the roll angle of the passive, regenerative and active with a square road profile.



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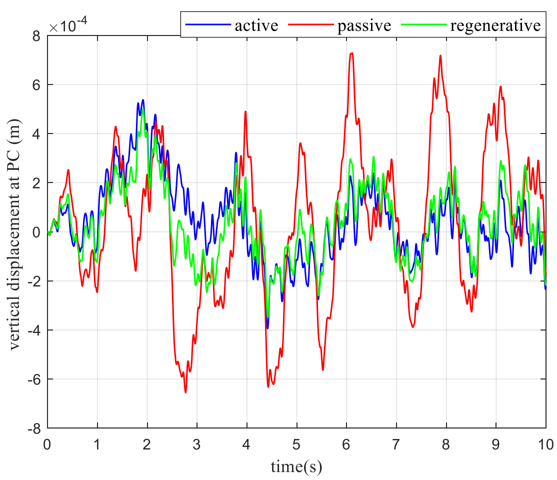

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Advances in Mechanics
Volume 9, Issue 3, 2021
Page 108-125

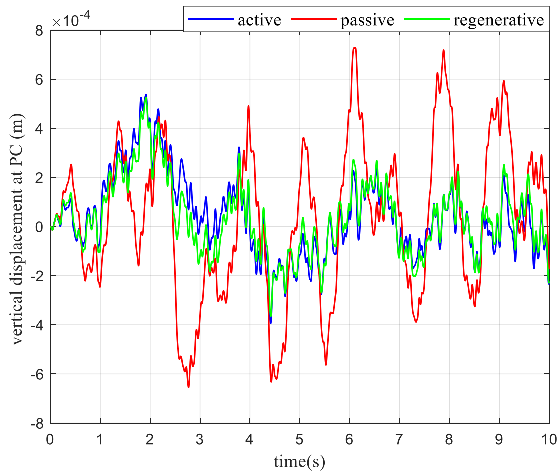

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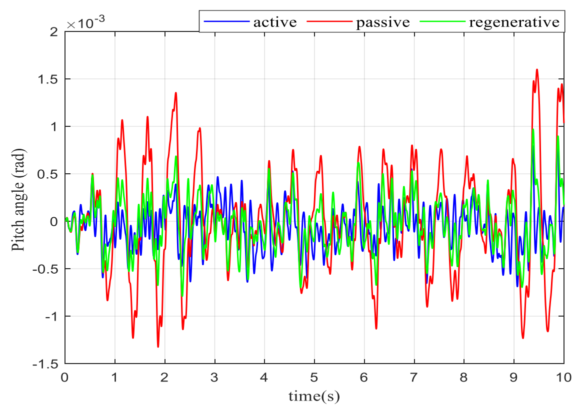
Figure 3. The vertical displacement of Pc (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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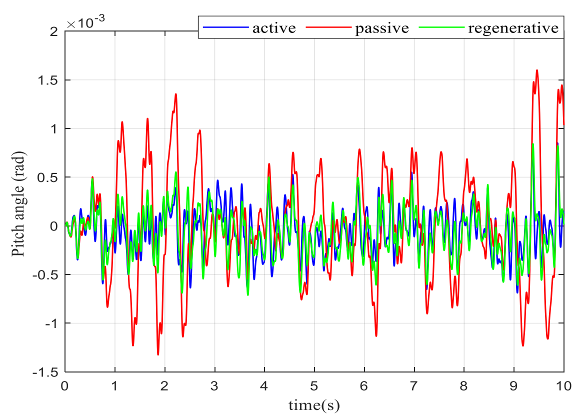

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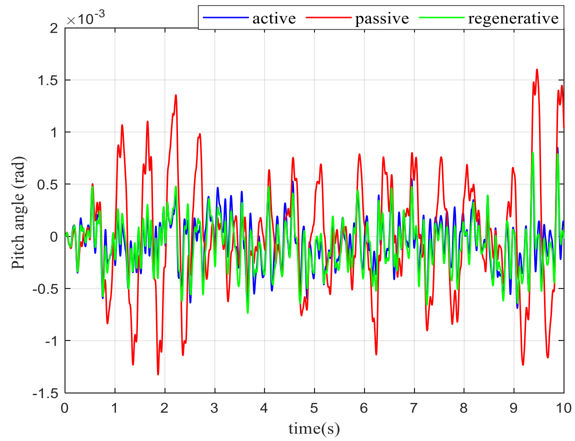

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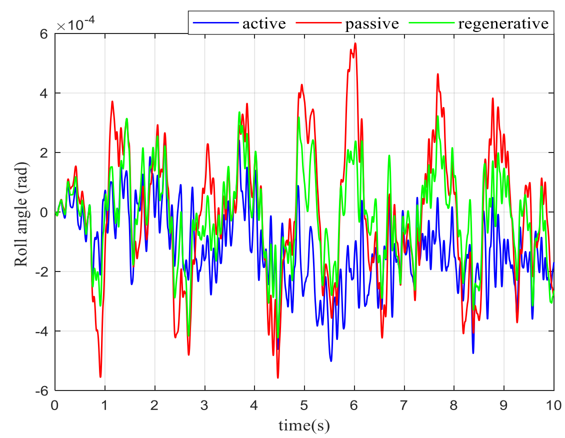
Figure 4. The pitch angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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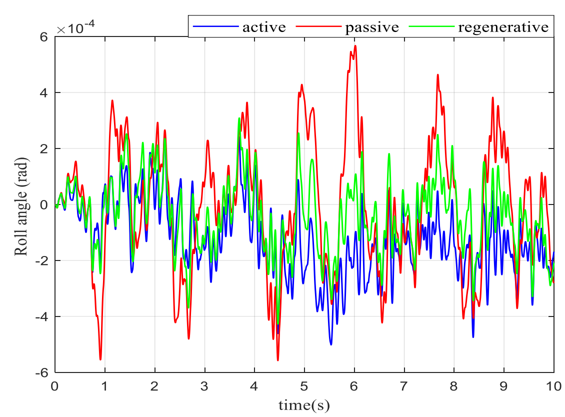

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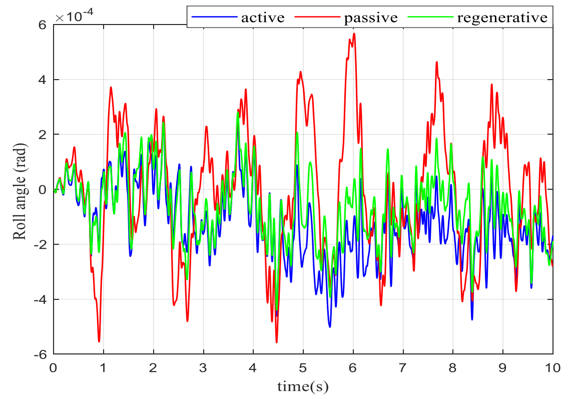

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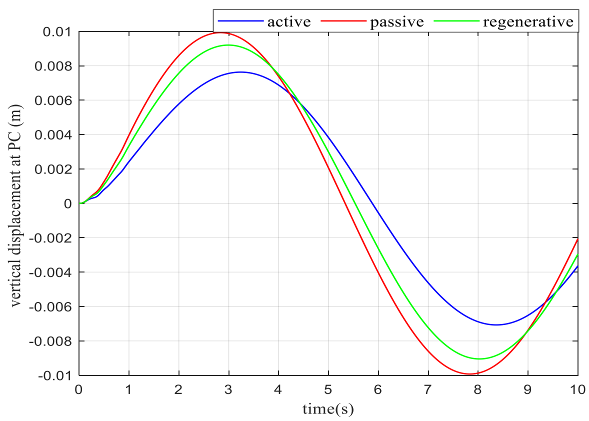
Figure 5. The roll angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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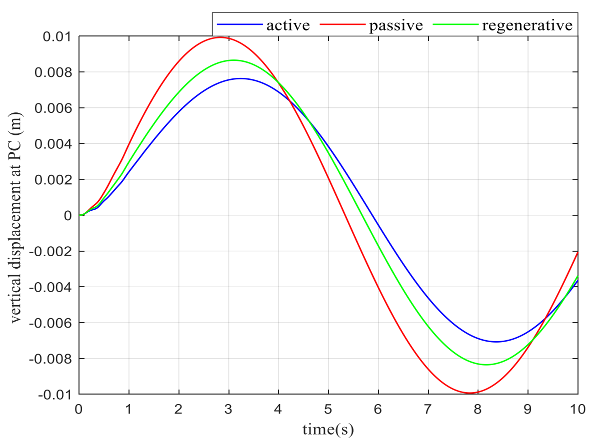

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Advances in Mechanics
Volume 9, Issue 3, 2021
Page 108-125

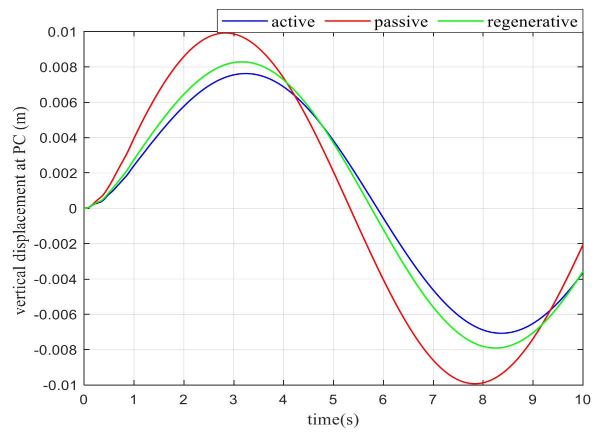

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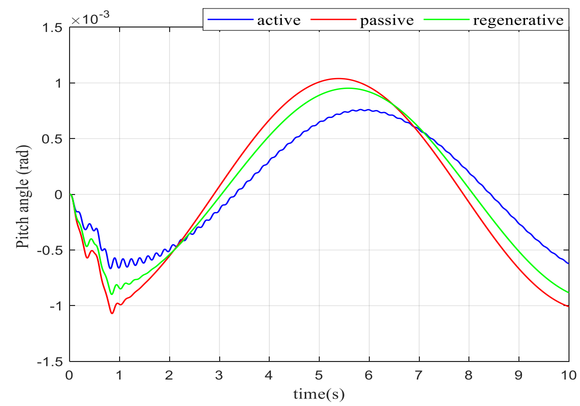
Figure 6. The vertical displacement (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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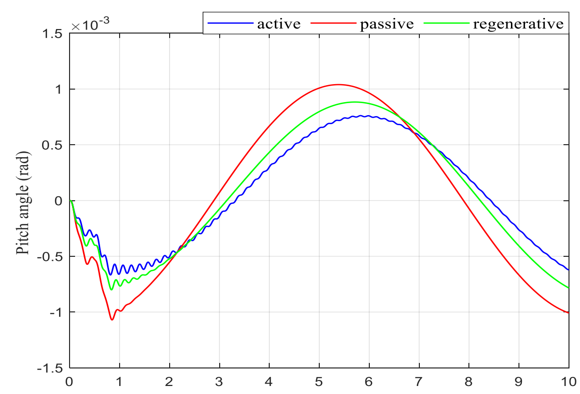

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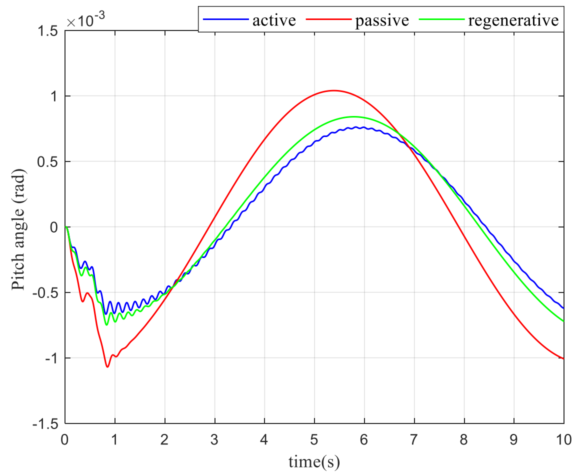

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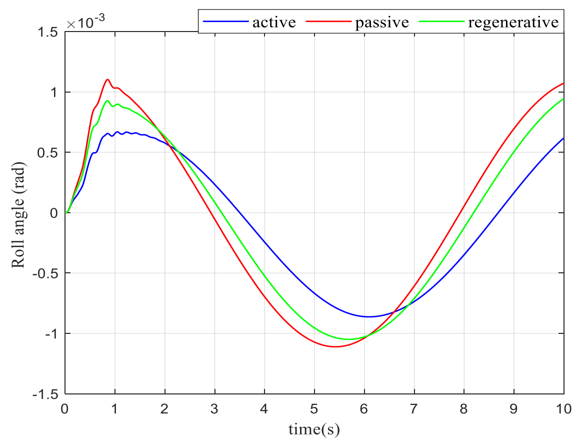
Figure 7. The pitch angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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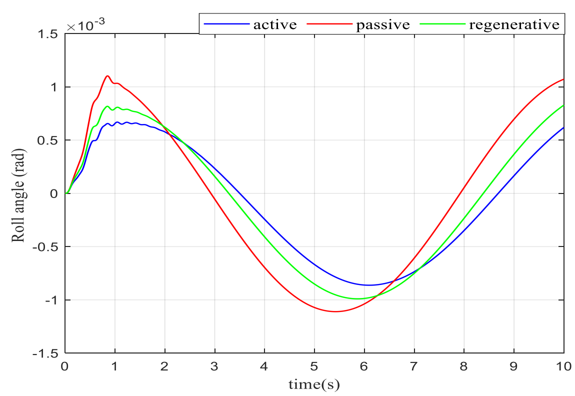

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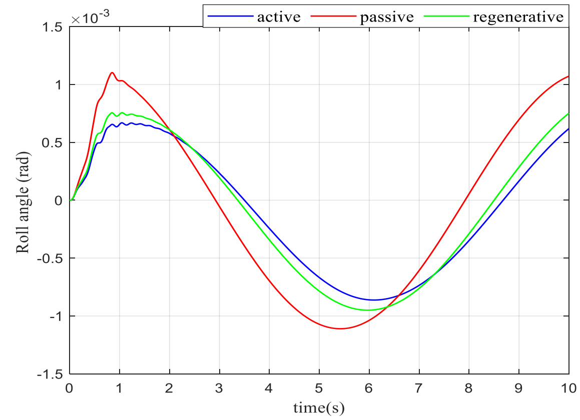

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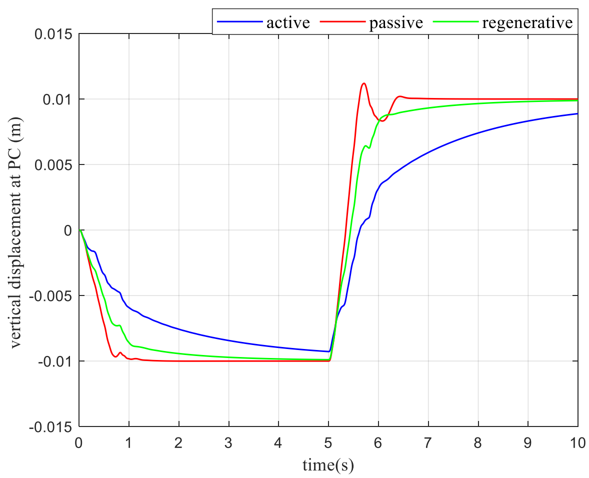
Figure 8. The roll angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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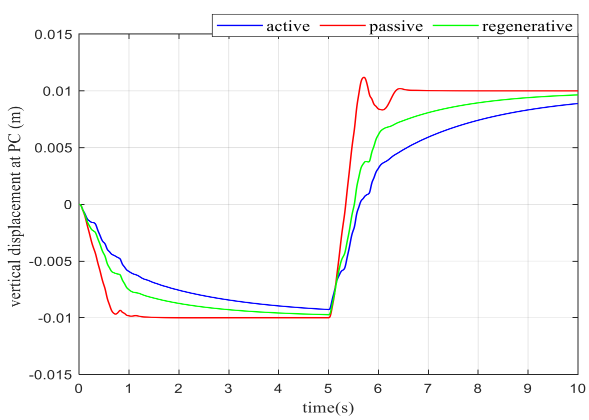

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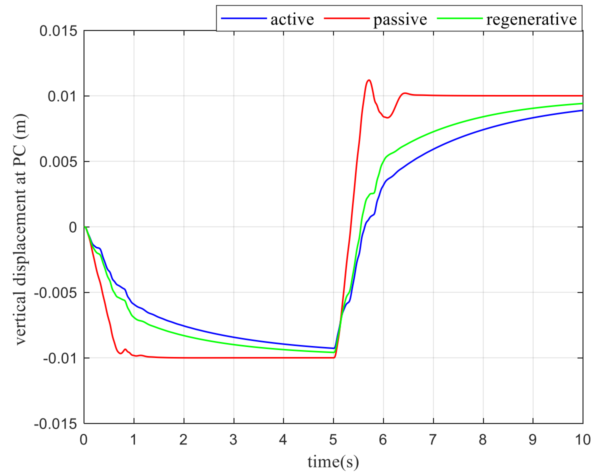

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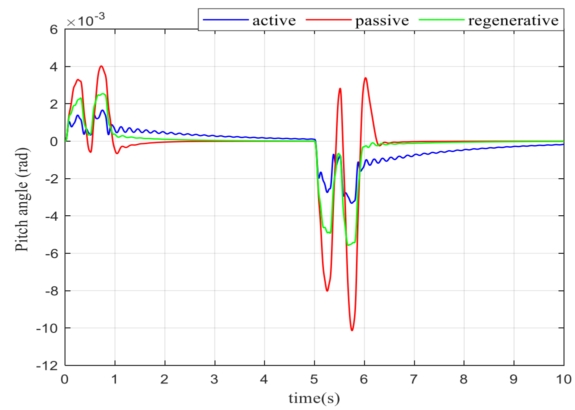
Figure 9. The vertical displacement(a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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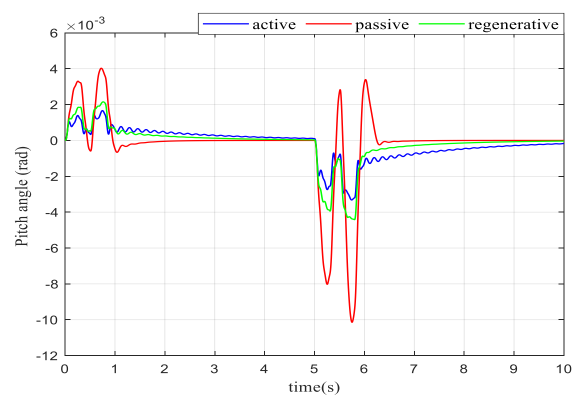

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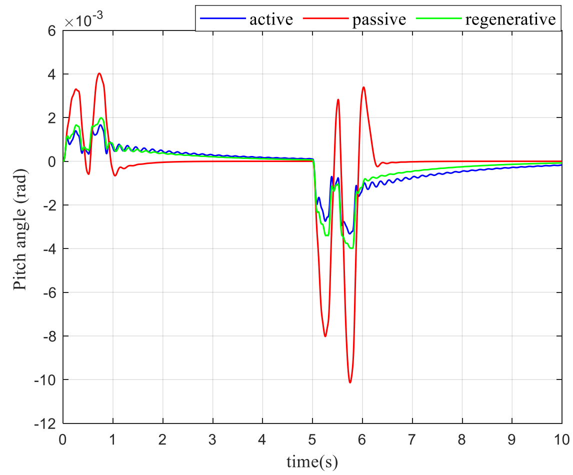
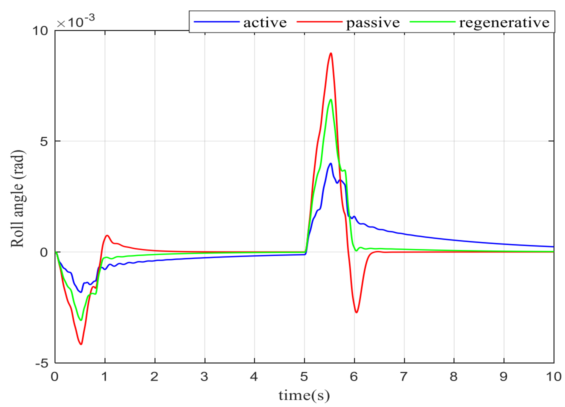


Figure 10. The pitch angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.



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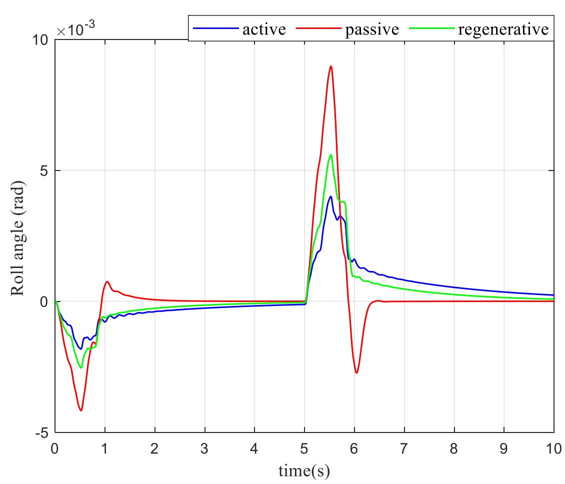

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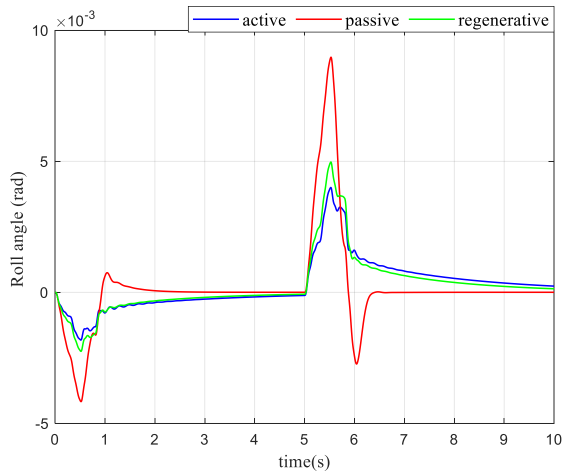

(c)

Figure 11. The roll angle (a) at pressure 10 bars, (b) at pressure 30 bars, and (c) at pressure 50 bars.

Conclusion

The purpose of this work is to develop unique equations for the Full Vehicle Regenerative Suspension system with hydraulic actuator and to simulate them using the Matlab Simulink tool. The system generally consists of (piston, hydraulic cylinder, spool valve and hydraulic motor) in addition to the control system. The comparison between active, passive and regenerative suspension was performed by using three signals (random, sinusoidal and square waves) and three pressure levels (10, 30 and 50 bars) with a wave amplitude of 0.1 m and a frequency of 0.1 Hz for 10 s. The conclusion shows that the hydraulic regenerative suspension system is better than the simple suspension system in terms of damping efficiency, in addition to harvesting wasted energy. The conclusion shows that the hydraulic regenerative suspension is better than the simple suspension in terms of damping efficiency as well as harvesting wasted energy. It was also found that it is slightly less than the damping efficiency of the active system, but it is better than it in terms of energy harvesting because it harvests energy instead of consuming it.

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Nomenclature

Symbol	**Description**	**Units**
Q_{Ldi}	Flow rate through piston	m^3/s
c_{dc}	Discharge coefficient	-
□	Area gradient	-
ρ	Hydraulic density	kg/m^3
P_{fi}	Pressure feed	N/m^2
P_{di}	Deferent pressure	N/m^2
P_{oi}	outlet pressure	N/m^2
δ , Ψ	Actuator parameters	-
x_{pi}	displacement of the piston inside the cylinder	m
F_{hyi}	Hydraulic force for actuator	N
	force of spring	N
K_{si}	stuffiness constant	N/m
z_i	Vertical displacements at suspensions	m
w_i	Vertical displacements for unsprung masses	m
$\boldsymbol{\mu}$	Empirical parameter	-
F_{ci}	forces of damping	N
c_i	damping factors	N.s/m
F_{ai}	Actual force of hydraulic actuator	N
F_{fr}	Frictional force inside the actuator	N
j_x	Moment of inertia in x-direction	kg.m2
j_y	Moment of inertia in y-direction	kg.m2

Γ	Roll angle	red
Β	Pitch angle	red
Tx	Cornering torque	N.m
Ty	Braking torque	N.m
L1	Distance between the center of gravity of sprung mass and front axle	m
L2	Distance between the center of gravity of sprung mass and rear axle	m
B	Width of vehicle	m
m_i	Unsprung masses	kg
c_{ti}	tire damping factors	N.s/m
u_i	The road profile inputs	m
z_c	Vertical displacements at center of gravity	m
k_{ti}	Tire stuffiness factors	N/m
A_p	Cross section piston area	m2