

A New Proposed Variable Stiffness of the Vehicle Suspension System Passive Case: I

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Abstract

A theoretical analysis of a new proposed variable stiffness model of the suspension system is studied to improve the traditional suspension system. The fundamental idea of system centered on variable stiffness mechanism by added subsystem to suspension system depended on control rotating arm balancing the force between sides, it consists of a vertical control strut. The variation of the load transfer by rotate arm has spring and damper at another side of it where the point of rotation is supplement the body of car by sup system as vertical support. The investigations of the new variable by addition stiffness to the suspension system for reaches improved performance better than the variable stiffness systems for equivalent or traditional. The expending principles to described the performance of the characteristic behavior of system are fewer car body acceleration to ride comfort and lower suspension and tire deflection for road holding considered.

Introduction

The isolate vehicle and the passengers from the road disturbance is the primary aim of the suspension system, while keeping good contact with the road. An idea suspension should be received the vehicle to minimum the car body acceleration, dynamic tire force and satisfy the constraint imposed on the rattle space(small suspension deflection between the axis of body car and tire axis).

The Suspension system performance directly effects on handling stability and ride comfort of vehicles. However, The traditional passive suspension has significant limitation in coordination of these two performance, unable to meet requirement of specifications design of vehicle. Therefore, researchers carried out researches on non-passive suspension system. **Chen (2006)**

The types of the suspension systems can be divide as passive system, semi-active system and active suspension system. The suspension system in passive case is consider simplest, not required external energy and low cost but in case of semi-active suspension can be advance of compromise between the both, the simplicity and cheap of passive suspension systems, but the complication and high cost for higher-performance of suspension system in case of fully active. In contrast the suspension system in active case ,the requirement of the semi-active suspension power and complex much less than, which can provide considerable advance in car ride quality and more reliable. Consequently, The suspension systems in case semi-active are classified into two types skyhook and ground hook control strategy introduced by **Karnopp et al. (1983)**

getting more care in the advancement of suspension system. The skyhook control methodology is without a doubt the best generally utilized control arrangement for semi-active suspension system. The skyhook control can lessen the beak of the resonant for the body mass and in this way accomplishes a decent ride quality. In any case, keeping in mind the end goal to progress both the ride quality and the wellbeing of vehicle, both beak of the resonant of the body and the wheel need to be diminished.

The passive suspension design are change by a range of exploration, as recorded by the works of **Anubi et al. (2013)** offerings the design, examination, and test validation of the uninvolved instance of a variable stiffness of the suspension system for idea depends on an as of late composed variable solidness component. It contains of a level control strut and a vertical strut. **Anubi and Crane. (2014)** offerings the semi-active instance of a variable stiffness suspension system for the previous idea which depends on designed variable stiffness instrument that comprises of an even strut and a vertical strut. **T. Rao et al.(2010)** depicts the illustrating, and testing of passive skyhook semi-active suspension control approaches. The control execution of a three-level of-opportunity quarter car semi-active suspension system is considered utilizing Matlab/Simulink, model. **Y. Liu et al. (2008)** presents theoretical and experimental investigations of the another proposed structure utilizing two controllable dampers and two steady springs is proposed semi-active suspension with variable stiffness and damping have exhibited magnificent execution. A Voigt component and a spring in arrangement are utilized to control the system stiffness.

The papers **(13-22)** utilize one of three procedures adaptive, semi-active or fully active suspension. A adaptive suspension utilizes a detached spring and a movable damper with moderate reaction to advance the control of ride luxury and road holding. A semi-active suspension is practically identical, aside from that the versatile damper has a quicker reaction and the damping power is controlled progressively. A fully active suspension substitutes the damper with a hydraulic driven actuator, or different sorts of actuators, for example, electromagnetic actuators, which can accomplish ideal vehicle control, however at the expense of configuration difficulty, cost, and so forth. The e fully active suspension is additionally not reliable as in execution debasement results at

whatever point the control unsuccessful, which might be because of either mechanical, electrical, or programming disappointments.

In the suspension system of vehicles most semi-active suspension system are considered to keep the stiffness consistent while the shock absorber can varying the damping coefficient temporarily, in suspension optimization, both the damping coefficient and the spring rate of the suspension components are normally utilized as trustworthy impacts. In this manner, a semi-dynamic suspension framework that fluctuates both component the stiffness and damping of the suspension could offer more adaptability in adjusting rival plan targets, **Anubi(2013)**. The Suspension stiffness extends that presentation variable stiffness suspension system wonder are few in writing considering the enormous measure of looks into that has been done on semi-active suspension plans.

This paper proposed and investigation of a variable stiffness of the new modal for suspension system for the quarter car passive case are present , The stiffness variation conception used in this investigation utilize the “reciprocal actuation to effectually of energy transfer between the traditional vertical strut and the horizontal oscillating control arm in order to refining the energy dissipation for the suspension system generally. Relatively, due to the number of moving parts in this model, which it can easily be combined into existing traditional suspension for front and/or rear designs as application with a double wishbone suspension system.

Description and Mechanism of suspension Variable Stiffness

The model of variable stiffness system is appeared in Fig.(1). The Lever arm OB, of length L , is stuck at an altered point O and allowed to spin about O. The spring AB is stuck to the lever arm at B and is allowed to turn about B. The flip side A of the spring is allowed to pivoting about E by the lever arm AD is joint with O at E as appeared by the double headed arrow. The spring AB is likewise allowed to pivot about point A. The F is the outer force is relied upon to activity vertically upwards at point B without a doubt don't loss of sweeping statement. Arm AD comprise from two component $L1$ and $L2$ and turning about E by ψ . The sign is to changing the general solidness of the framework by development $L1$ and $L2$ fluctuating latently under the effect of a vertical spring-damper framework (not appeared in the figure) allude as U force.

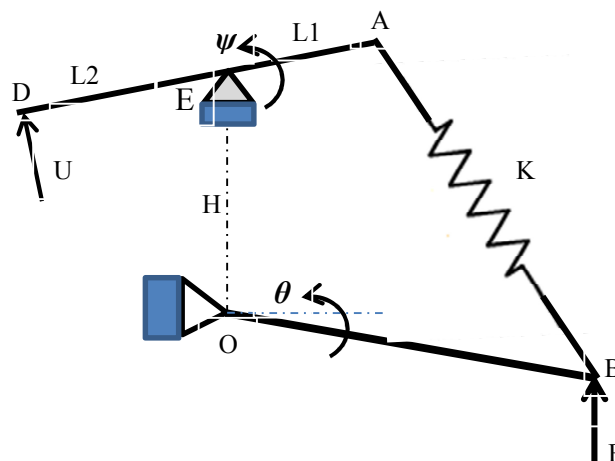


Fig.(1): Variable Stiffness Mechanism

Consideration the system of suspension as shown in Fig.(2). The schematic of model is consist from a quarter of body car as wheel assembly, two dampers, two springs, lower, upper wishbones and road disturbance.

The points O, A, B and D are identical as presented in Fig.(1) for the mechanism of the variable stiffness of Fig.(2). The Vertical regulator force U used to control by rotation of arm AD with angle ψ which in opportunity to controls on the mechanism for overall stiffness. The modulation of tire is consider as a linear spring for both spring constant and damping coefficient

The assumptions accepted in Fig.(2) are brief as tails:

- 1- The side translation or horizontal movement of the sprung mass is ignored, which is only the vertical displacement Z_s is measured.
- 2- The angle camber of wheel is zero at the position of equilibrium and its disparity is insignificant during the system route.
- 3- the unsprung mass is joint to the car body by two ways: first by damper and second by the arm OC (first control arm) where θ denotes the angular displacement of the first control arm.
- 4- The deflection in spring, damping and tire forces are in the linear regions of their operating ranges.
- 5- The sprung and the unsprung masses are expected to be particles.
6. Including both the mass and the stiffness of the control arm.

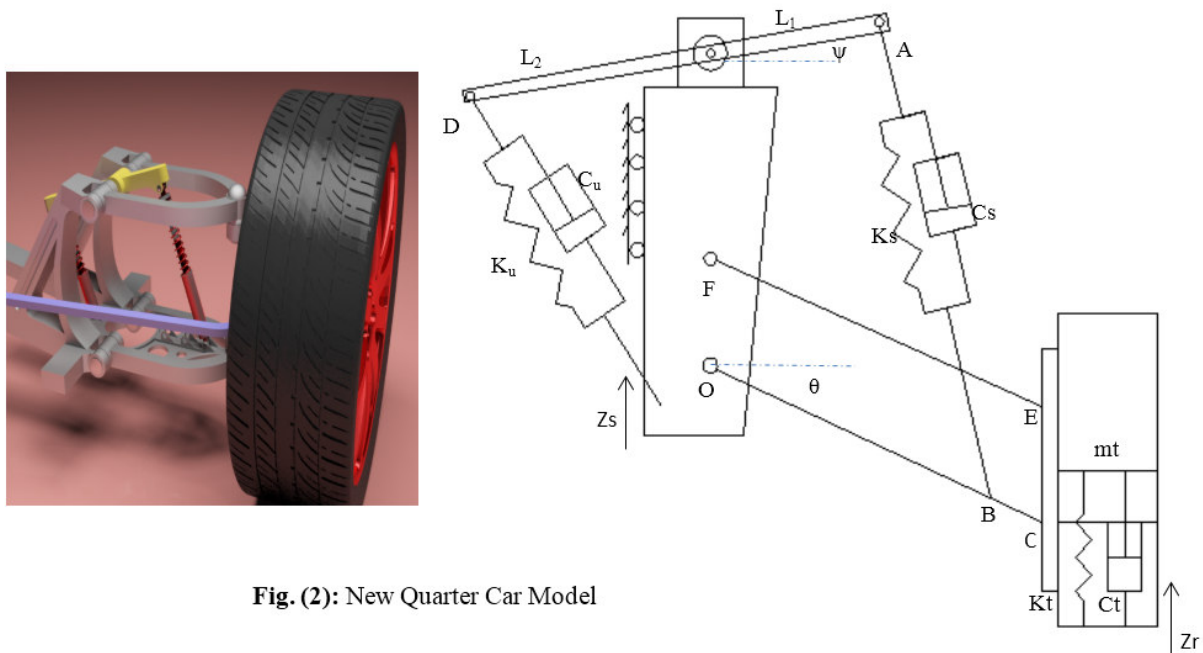


Fig. (2): New Quarter Car Model

Let $(y_A, z_A), (y_B, z_B), (y_C, z_C)$ and (y_D, z_D) denote the coordinate of point A, B, C and D, respectively, when the suspension system is at an equilibrium point, then following equation hold:

$$y_A = L_1 [\cos(\varphi - \varphi_0)] \quad (1a)$$

$$z_A = z_s + L_1 [\sin(\varphi - \varphi_0)] \quad (2a)$$

$$y_B = L_B [\cos(\theta - \theta_0)] \quad (3a)$$

$$z_A = z_s + L_B [\sin(\theta - \theta_0)] \quad (4a)$$

$$y_C = L_B [\cos(\theta - \theta_0)] \quad (5a)$$

$$z_C = z_s + L_C [\sin(\theta - \theta_0)] \quad (6a)$$

$$y_D = L_D [\cos(\varphi - \varphi_0)] \quad (7a)$$

$$z_D = z_s + L_D [\sin(\varphi - \varphi_0)] \quad (8a)$$

Where (θ_0) and (φ_0) are the original angular displacement of the arm OC and control arm AD at an evenness point.

For the small change of angles let

$\cos(\varphi) = 1, \cos(\theta) = 1, \sin(\theta) = \theta, \sin(\varphi) = \varphi$ and $(\varphi_0 = 0, \theta_0 = 0)$ then the following relation for the kinetic, potential and damping energies are obtained from the whole system.

$$K.E = \frac{1}{2} m_s \dot{z}_s^2 + \frac{1}{2} m_s (\dot{z}_s + L_c \dot{\theta})^2 + \frac{1}{2} I_1 \dot{\theta}^2 + \frac{1}{2} I_2 \dot{\varphi}^2 \quad (1b)$$

$$P.E = \frac{1}{2} k_s (z_s + L_c \theta - r)^2 + \frac{1}{2} k_t (L_c \theta - L_1 \varphi)^2 + \frac{1}{2} k_u L_2^2 \varphi^2 \quad (2b)$$

$$D.E = \frac{1}{2} C_s (L_c \dot{\theta} - L_1 \dot{\varphi})^2 + \frac{1}{2} C_t (\dot{z}_s + L_c \dot{\theta} - \dot{r})^2 + \frac{1}{2} I_1 \dot{\theta}^2 + \frac{1}{2} C_u L_2^2 \dot{\varphi}^2 \quad (3b)$$

Where I_1 and I_2 are the second moment of inertia of the arm OC and arm AD.

Equations of Motion

The quarter vehicle model with three degree of freedom Fig.(2) is employed for the suspension system. This model can capture bounce angle of wishbone (θ), angle of control arm (ψ) and the vertical displacement of vehicle z_s , therefore the generalized coordinates of system are:

$$q = \begin{bmatrix} z_s \\ \theta \\ \varphi \end{bmatrix} \quad (1c)$$

The equations of motion for the new model are derived utilize Lagrange's method are then given by:

$$(m_s + m_t) \ddot{z}_s + m_t L_c \ddot{\theta} + k_t z_s + k_t L_c \theta + C_t \dot{z}_s + C_t L_c \dot{\theta} = k_t z_r + C_t \dot{z}_r \quad (2c)$$

$$m_t L_c \ddot{z}_s + (m_t L_c^2 + I_1) \ddot{\theta} + C_t L_c \dot{z}_s + C_t L_c^2 \dot{\theta} + C_s L_B^2 \dot{\theta} - C_s L_B L_1 \dot{\varphi} + k_t L_c z_s + k_t L_c^2 \theta + k_s L_B^2 \theta - k_s L_B L_1 \varphi = k_t L_c z_r + C_t L_c \dot{z}_r \quad (3c)$$

$$I_2 \ddot{\varphi} - k_s (L_B \theta - L_1 \varphi) L_1 - C_s (L_B \dot{\theta} - L_1 \dot{\varphi}) L_1 + C_u L_2^2 \dot{\varphi} + k_u L_2^2 \varphi = 0 \quad (4c)$$

State Space Analysis

The states variable are introduce as $[x_1 \ x_2 \ x_3 \ x_4 \ x_5 \ x_6]^T = [z_s \ z'_s \ \theta \ \theta' \ \psi \ \psi']^T$ the the equation of motion written

in the state equation as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= F_1(x_1, x_2, x_3, x_4, x_5, x_6, z_r) \\ \dot{x}_3 &= x_4 \\ \dot{x}_4 &= F_2(x_1, x_2, x_3, x_4, x_5, x_6, z_r) \\ \dot{x}_5 &= x_6 \\ \dot{x}_6 &= F_3(x_3, x_4, x_5, x_6) \end{aligned} \quad (1d)$$

$$F_1 = \frac{-1}{Dy} \{ (DK_t + k_t L_c) z_s + (k_t L_c D + k_t L_c^2 + k_s L_B^2) \theta + (C_t D + C_t L_c) \dot{z}_s + (C_t L_c D + C_t L_c^2 + C_s L_B^2) \dot{\theta} - c_s L_1 L_B \dot{\varphi} - k_s L_1 L_B \varphi - (k_t D + k_t L_c) z_r - (C_t D + C_t L_c) \dot{z}_r \} \quad (2d)$$

$$F_2 = \frac{-1}{D\theta} \{ (D' C_t + C_t L_c) \dot{z}_s + (c_t L_c D' + C_t L_c^2 + C_s L_B^2) \dot{\theta} + (k_t D' + k_t L_c) z_s + (k_t L_c D' + k_t L_c^2 + k_s L_B^2) \theta - c_s L_1 L_B \dot{\varphi} - k_s L_1 L_B \varphi - (k_t D' + k_t L_c) z_r - (C_t D' + C_t L_c) \dot{z}_r \} \quad (3d)$$

$$F_3 = \frac{-1}{I_2} \{ k_s L_1 L_B \theta + C_s L_1 L_B \dot{\theta} - (k_s L_1^2 + k_u L_2^2) \varphi - (C_s L_1^2 + C_u L_2^2) \dot{\varphi} \} \quad (4d)$$

Where

$$Dy = m_t L_c - (m_s + m_t)(m_t L_c^2 + I_1) / m_t L_c$$

$$D\theta = -\frac{m_t L_c^2 + I_1}{m_s + m_t} + (m_t L_c^2 + I_1)$$

$$D = \frac{m_t L_c^2 + I_1}{m_t L_c} \quad D' = -m_t L_c / (m_s + m_t)$$

Linearization

The new model of system are linearized because of the small angles approximation and the fact that its equilibrium are zeros. The linearization is needed model forms that do not contain the sine are cosine function and whose equilibrium is nonzero.

$$\dot{x} = Ax + Bz_r \quad \text{and} \quad y = Cx + Dz_r$$

Where

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{\partial F_1}{\partial x_1} & \frac{\partial F_1}{\partial x_2} & \frac{\partial F_1}{\partial x_3} & \frac{\partial F_1}{\partial x_4} & \frac{\partial F_1}{\partial x_5} & \frac{\partial F_1}{\partial x_6} \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{\partial F_2}{\partial x_1} & \frac{\partial F_2}{\partial x_2} & \frac{\partial F_2}{\partial x_3} & \frac{\partial F_2}{\partial x_4} & \frac{\partial F_2}{\partial x_5} & \frac{\partial F_2}{\partial x_6} \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & \frac{\partial F_3}{\partial x_3} & \frac{\partial F_3}{\partial x_4} & \frac{\partial F_3}{\partial x_5} & \frac{\partial F_3}{\partial x_6} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ a21 & a22 & a23 & a24 & a25 & a26 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ a41 & a42 & a43 & a44 & a45 & a46 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & a63 & a64 & a65 & a66 \end{bmatrix} \quad (1e)$$

$$a21 = \frac{-(DK_t + k_t L_c)}{Dy}, a22 = \frac{-(DC_t + C_t L_c)}{Dy}, a23 = \frac{-(DL_c k_t + k_t L_c^2 + k_s L_B^2)}{Dy},$$

$$a24 = \frac{-(DL_c C_t + C_t L_c^2 + C_s L_B^2)}{Dy}, a25 = \frac{k_s L_1 L_B}{Dy}, a26 = \frac{C_s L_1 L_B}{Dy}$$

$$a41 = \frac{-(D' k_t + k_t L_c)}{D\theta}, a42 = \frac{-(D' C_t + C_t L_c)}{D\theta}, a43 = \frac{-(D' L_c k_t + k_t L_c^2 + k_s L_B^2)}{D\theta},$$

$$a44 = \frac{-(D' L_c C_t + C_t L_c^2 + C_s L_B^2)}{D\theta}, a45 = \frac{k_s L_1 L_B}{D\theta}, a46 = \frac{C_s L_1 L_B}{D\theta}$$

$$a63 = \frac{k_s L_1 L_B}{D\theta}, a64 = \frac{C_s L_1 L_B}{D\theta}, a65 = \frac{-(k_s L_1^2 + k_u L_2^2)}{I_2}, a66 = \frac{-(C_s L_1^2 + C_u L_2^2)}{I_2}$$

$$B = \begin{bmatrix} 0 & \frac{\partial F_1}{\partial z_r} & 0 & \frac{\partial F_2}{\partial z_r} & 0 & \frac{\partial F_3}{\partial z_r} \end{bmatrix}^T = \begin{bmatrix} 0 & \frac{k_t D + k_t L_c}{Dy} & 0 & \frac{k_t D' + k_t L_c}{D\theta} & 0 & 0 \end{bmatrix}^T \quad (2e)$$

D and C matrices depend on the output values from the system, and the $Z_r(t)$ be present the road movement indication as a function of the road surface and the vehicle speed. The term L_c and L_B are the length from pint O to tire and to point B respectively.

Effect The Stiffness And Damping Variation On Suspension System

A simulation study of the effect variable stiffness and damping on the suspension performance. The performances of interest are the ride luxury, suspension defection, and road holding. The ride comfort and road holding performances are characterized by the acceleration of body car and defection of the tire respectively. Fig .(2) shows the quarter car model used for the simulation study. The spring constant K_s and the damping coefficient C_s were varied in the intervals [10000 30000] N/m and [500 1500] N.s/m respectively **without added the subsystem** (K_{arm} , C_{arm}) to study the effect variation damping and stiffness on classic system.

The characterizes suspension of system is presentation by competence the ride luxury, road property and traveling . These performance principles are specified in expressions of the acceleration body of car, the deflection of suspension and tire respectively, therefore the performance index $J(x,v)$, articulated by way of the weighted quantity of the parameters present as [1]

$$J(X, V) = \int_{t_0}^{t_f} (CBA^2 + SD^2 + TD^2) dt$$

Where CBA: car body acceleration
 SD: suspension deflection
 TD: tire deflection

For each pair (K_s ; C_s) in the interval, the gain of the system for the three performances were computed at two frequencies 3.162 Hz and 316.2 Hz, corresponding to low and high frequencies respectively. Table(1) shows the gain of the system for car body acceleration against the given spring constants and damping coefficients.

Table(1): The value of gain(dB) of the suspension system

Ks N/m	Cs N.s/m	CBA		SD		TD	
		LF Hz	HF Hz	LF Hz	HF H	LF Hz	HF Hz
10000	500	14	-25.5	-18.2	-77.3	-40	-49
20000		13.5	-17.4	-17.8	-77	-41.3	-49
30000		13.1	-25.3	-17.2	-76	-41.2	-49.6
10000	1000	12.3	-25.3	-22.4	-77.1	-41.5	-49.5
20000		11.9	-18.2	-22.9	-77	-41.3	-49.67
30000		11.2	-25	-23	-77	-41	-49.9
10000	1500	11.3	-25	-27.5	-77.23	-42	--49
20000		10.8	-25	-27.5	-77.25	-42	-49
30000		10.4	-25	-28.5	-77.3	-42	-49

From table(1) it is seen that, at low frequency(LF), the best ride comfort performance is achieved by low damping. Also, can see that, at high frequency(HF), the best ride comfort performance is achieved by low stiffness. The best suspension defection performance is achieved by high damping at low frequency, and by a combination of high stiffness and low damping at high frequency. the best road holding performance is also achieved by high damping at low frequency, and by a combination of high stiffness and low damping at high frequency.

In practice, suspension systems are usually required to optimize a weighted combination of the above performance measures. It follows therefore, from the above simulation results, that the best suspension system will be the one that can modulate both its stiffness and damping values.

Passively Variable Stiffness and Simulation

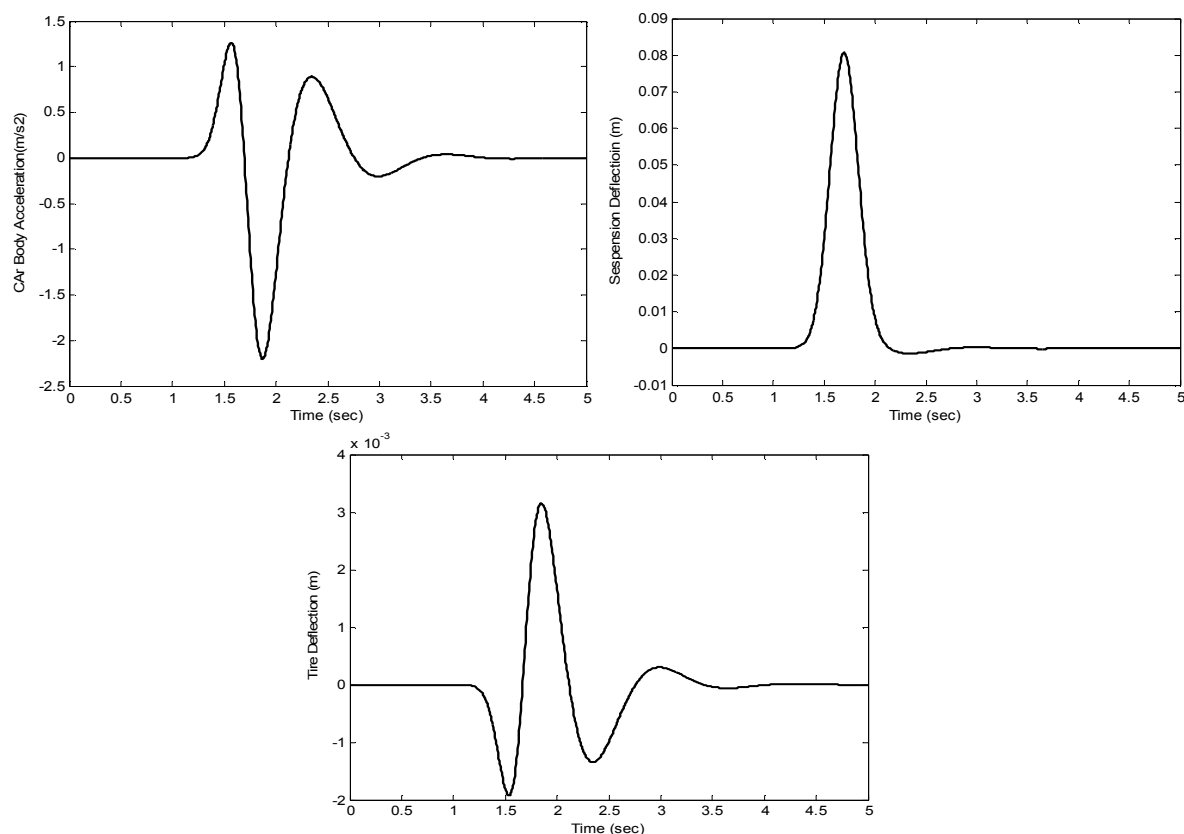
At this point, the control arm AD is allowable to rotate under the effect of forces by spring and damper. Where is no additional external force producer to system. Let K_{arm} is the constant of the spring and C_{arm} is the coefficient of damper, then the dynamics of the control arm is specified as:

$$I_{arm}\ddot{\phi} + C_{arm}L_2^2\dot{\phi} + K_{arm}L_2^2\phi + K_sL_1^2(\phi - \theta) + C_sL_1^2(\dot{\phi} - \dot{\theta}) = 0$$

In the simulation of the time domain, the car traveling on the road, which exposed to a bump of height 8cm for plane speed of 40 mph. The acceleration of body car, suspension defection, and tire defection responses are associated with the new variable stiffness of suspension system model by **Anubi** [1] and the present passive variable stiffness where the comparison between the two models as obtained results for acceleration of body car, suspension and tire deflection, and variance gain of system for the dynamic parameters structure (**Zin et al., 2004**). The data are fixed in Table 2.

Table (2): The value of dynamic and kinematic parameters

Parameters	Value	Parameters	Value	Parameters	Value
m_s	315 kg	I_a	0.1 kg m ²	C_s	1500N.s/m
m_t	37.5 kg	k_t	210 kN/m	K_u	5 kN/m
m_a	10 kg	C_t	600 N.s/m	C_u	2500N.s/m
I	0.015 kg m ²	k_s	29 kN/m	L_D	0.475m
H	0.4m	L_B	0.35m	L_1, L_2	0.2, 0.3m



Fig(3): Car Body acceleration ,Suspension Deflection and Tire deflection

From the above figures and comparison with Anubi [1] at passive case can see the following:

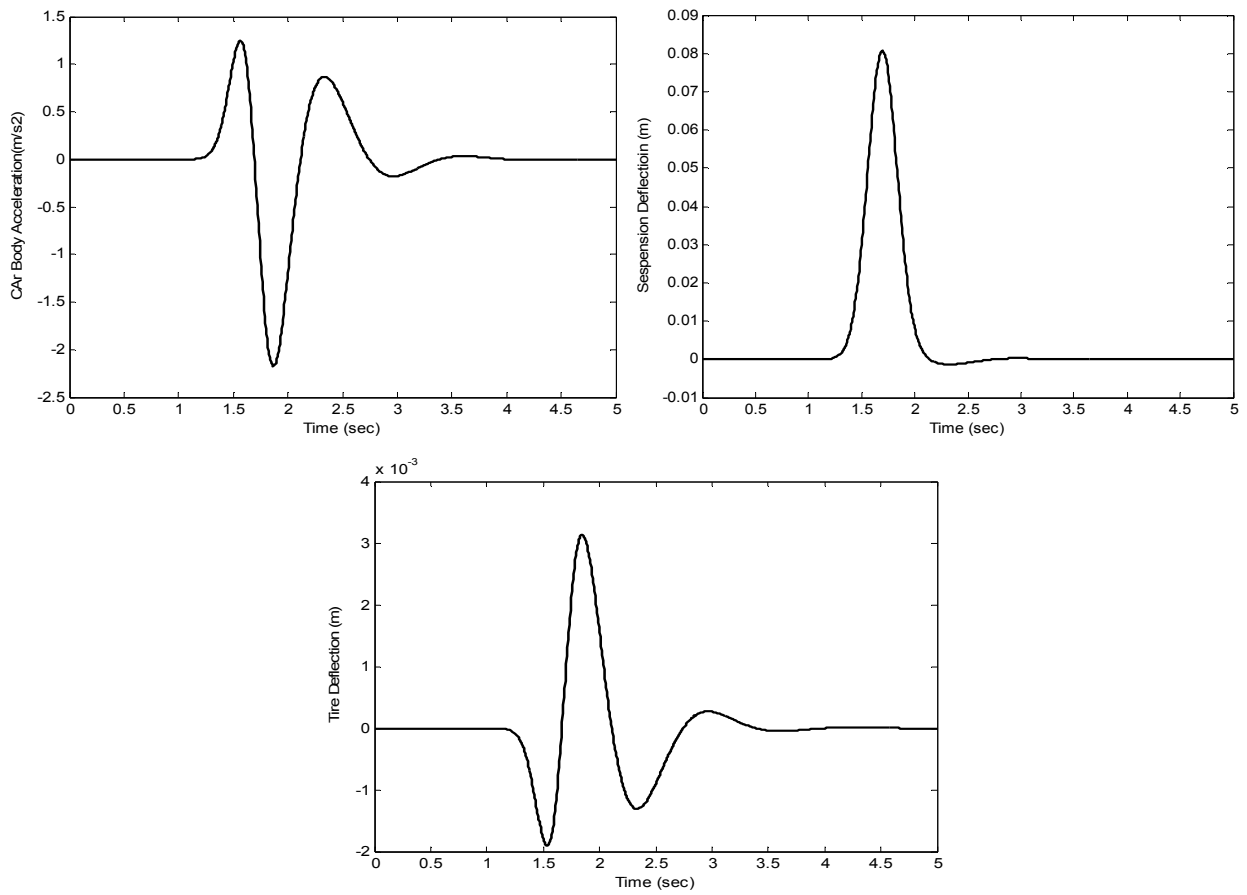
- 1- The value of acceleration of body car is little and suitable compared with results at [1] this attend to that “The model proposed has good results in this field”.
- 2- The suspension deflection between the body of car and road no longer than amplitude od road. For that advantageous to ride comfort and holding.
- 3- The tire deflection approximately equal between two models

On the other hand the results compared with respect to the root mean square value of the acceleration of body car , suspension and tire deflection shown smallest than values of paper [1]. As shown in table (3)

Table(3): Root Mean Square values of results

CBA: car body acceleration. SD: suspension deflection TD: tire deflection TDA: tire deflection acceleratio

	CBA	SD	TD	TDA
Anubi	0.5864	-	-	0.9685
Present work	0.5372	0.0055	0.0008	0.9239



Fig(4): Car Body acceleration ,Suspension Deflection and Tire deflection
 For $k_u=4000\text{N/m}$ $c_u=2500\text{N.s/m}$

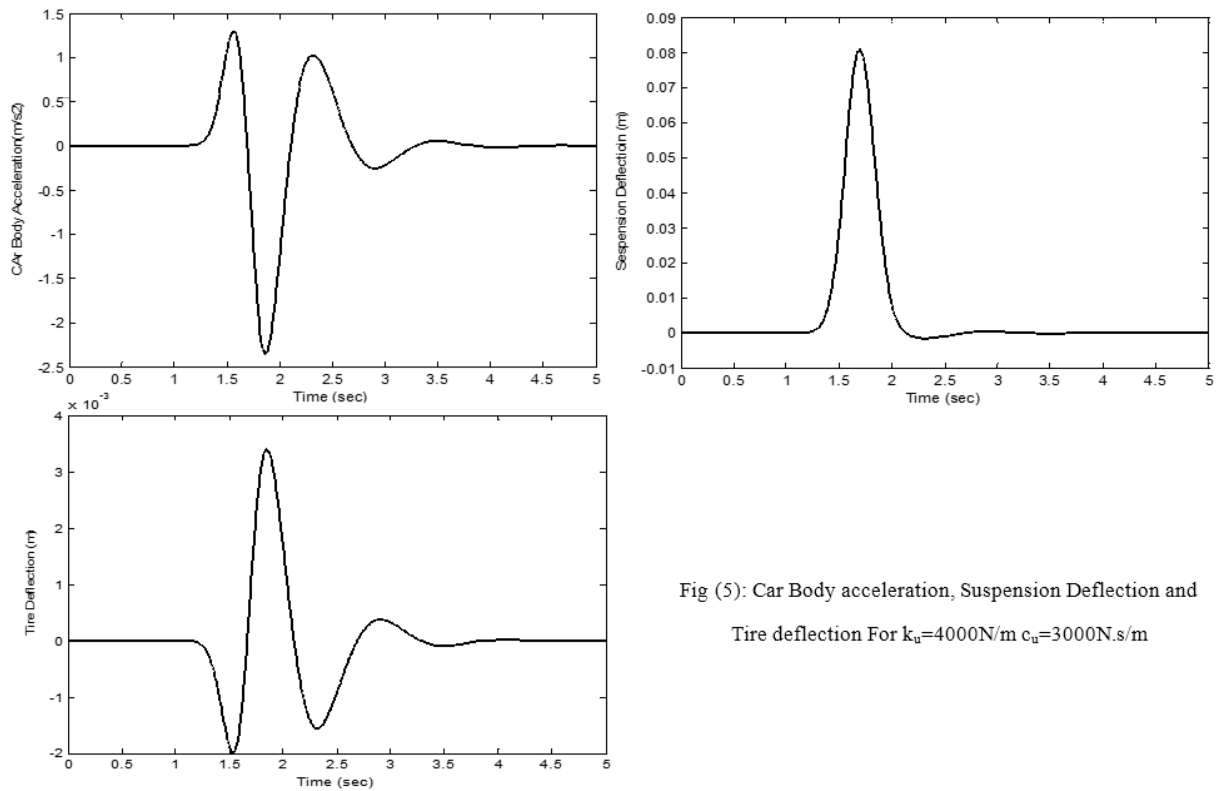


Fig (5): Car Body acceleration, Suspension Deflection and
 Tire deflection For $k_u=4000\text{N/m}$ $c_u=3000\text{N.s/m}$

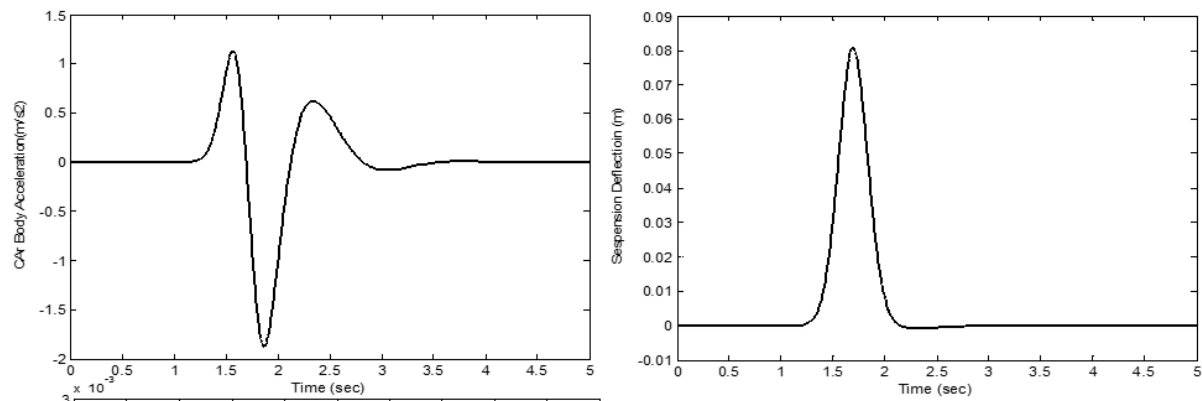


Fig (6): Car Body acceleration, Suspension Deflection and
 Tire deflection for $k_u=4000\text{N/m}$ $c_u=3000\text{N.s/m}$, $L_1=0.25\text{m}$, $L_2=0.3\text{m}$

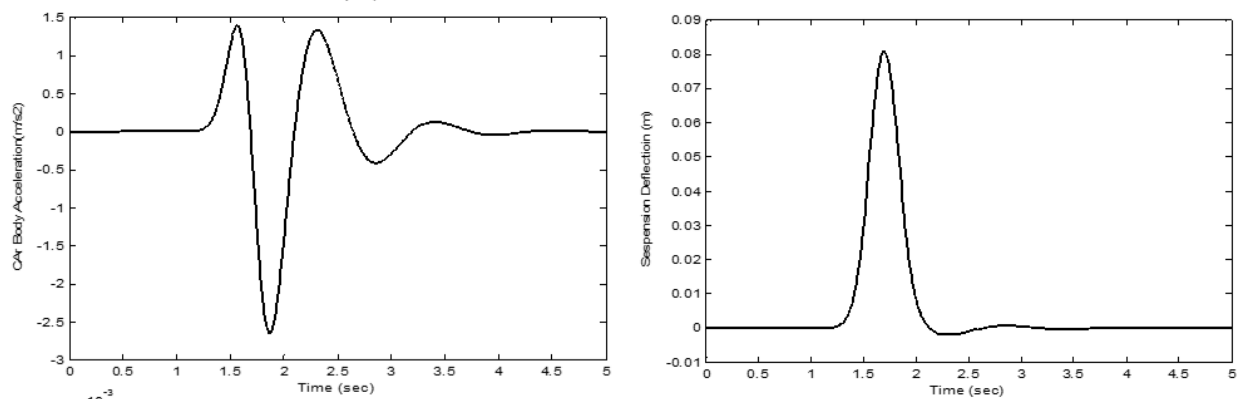
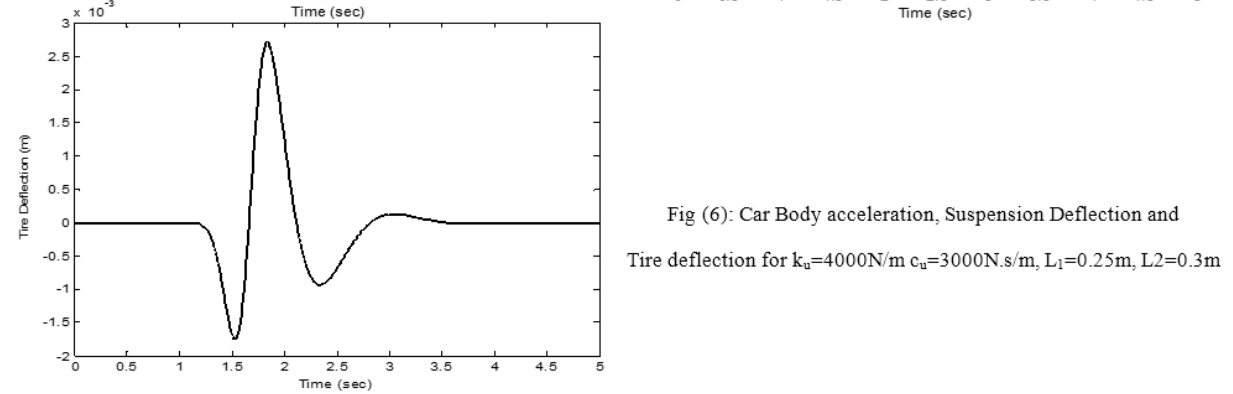
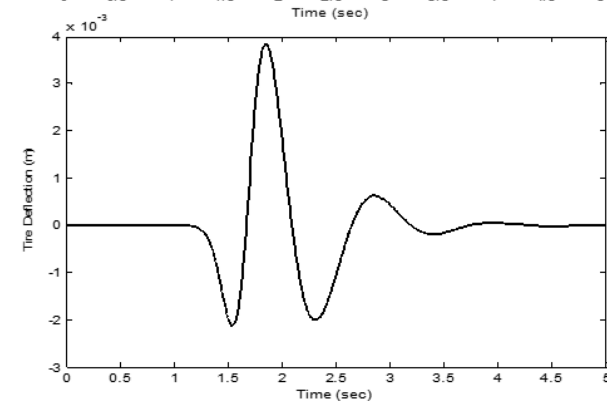
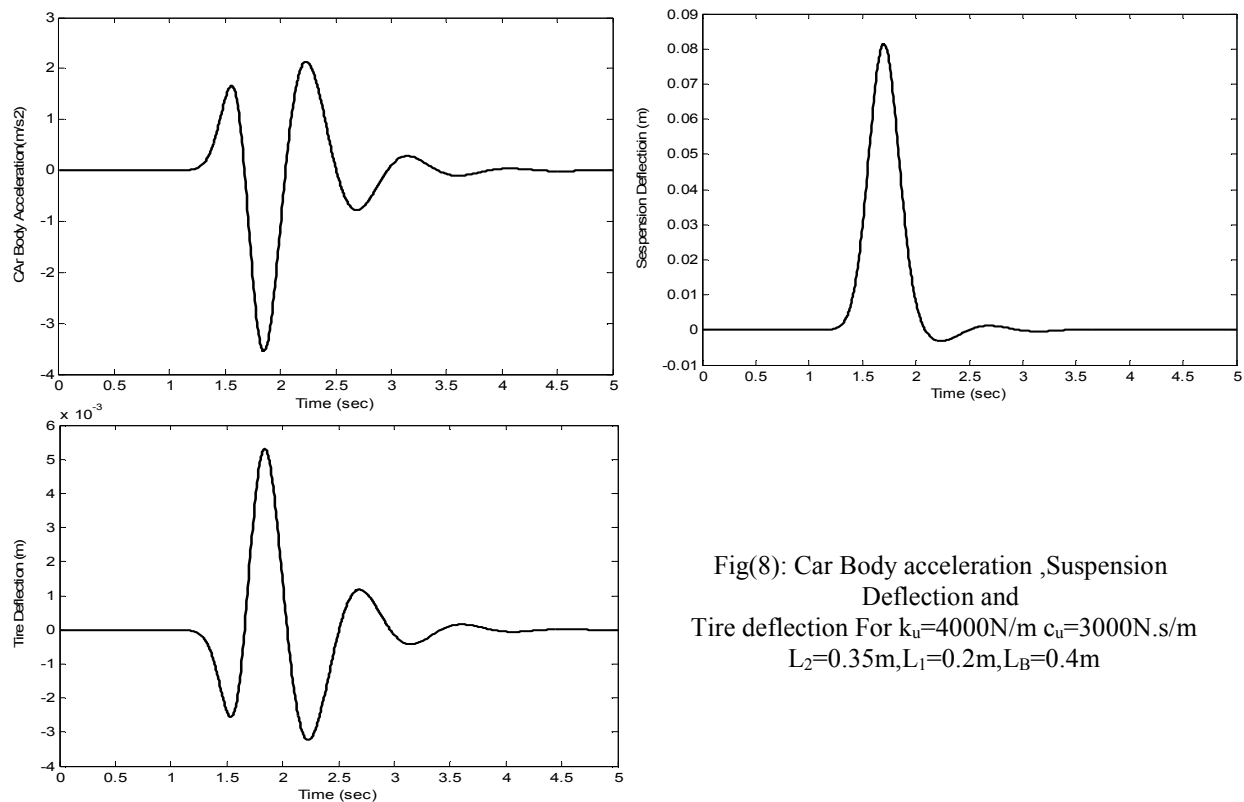


Fig (7): Car Body acceleration, Suspension Deflection and
 Tire deflection for $k_u=4000\text{N/m}$ $c_u=3000\text{N.s/m}$ $L_2=0.35\text{m}$, $L_1=0.2\text{m}$





Fig(8): Car Body acceleration ,Suspension Deflection and Tire deflection For $k_u=4000\text{N/m}$ $c_u=3000\text{N.s/m}$ $L_2=0.35\text{m}$, $L_1=0.2\text{m}$, $L_B=0.4\text{m}$

Results and Time domain

The results obtained in figures above discuss the effect of some parameters have effect on the behavior of the suspension system proposed such as increase the damping a of the subsystem added in suspension system in fig.(4) and fig.(5) increase the damping to (2500, 3000 N.s/m) indicated to decrease the acceleration and suspension deflection.

In fig.(6,7) study effect increase the length of arm control by expanded ($L_1=0.25$ or $L_2=0.35$), shown that the behavior of system is inverse but improved the acceleration and suspension deflection for L_1 the acceleration decrease to smallest value can be obtained and for L_2 the acceleration and suspension deflection begin tend to increase, for that the length of arm control has very sensitivity to change because it is do as stabiles arm between the force effect on sides.

In fig.(8) shown effect expanded the arm ($L_B=0.4$) on the system with constant another value of parameters (k_u, c_u, L_2, L_1) where the acceleration approximately steady but suspension and tire deflection tend to decrease.

Simulation of The Frequency Domain

An expected frequency response for the frequency area reproduction from the road disturbance influence info to the way of execution determined by the idea of variance gain [17,18]. The variance gain surmised is given, where z means the execution measure of consideration which is taken to be acceleration of body car, suspension deflection, and tire deflection.

The closed loop of the system is energize by the sinusoid $r = A \sin(\omega t)$; $t \in [0, 2\pi N/\omega]$, where N is a whole number big adequate to guarantee that the system achieves a steady state. The predictable yield signs were recorded and the evaluated difference additions were computed by.

$$G(\omega) = \sqrt{\frac{\int_0^{2\pi N/\omega} z^2 dt}{\int_0^{2\pi N/\omega} A^2 \sin^2 \omega t dt}}$$

Fig.(9) display the variance gain plots for the acceleration of body car and suspension deflection respectively. In this figures shown that the variable stiffness of the suspension gets healthy isolation of vibration according to range of the sensitive frequency for human (4-8Hz) [20], then improve handling in severity region ($> 59\text{Hz}$) [19].

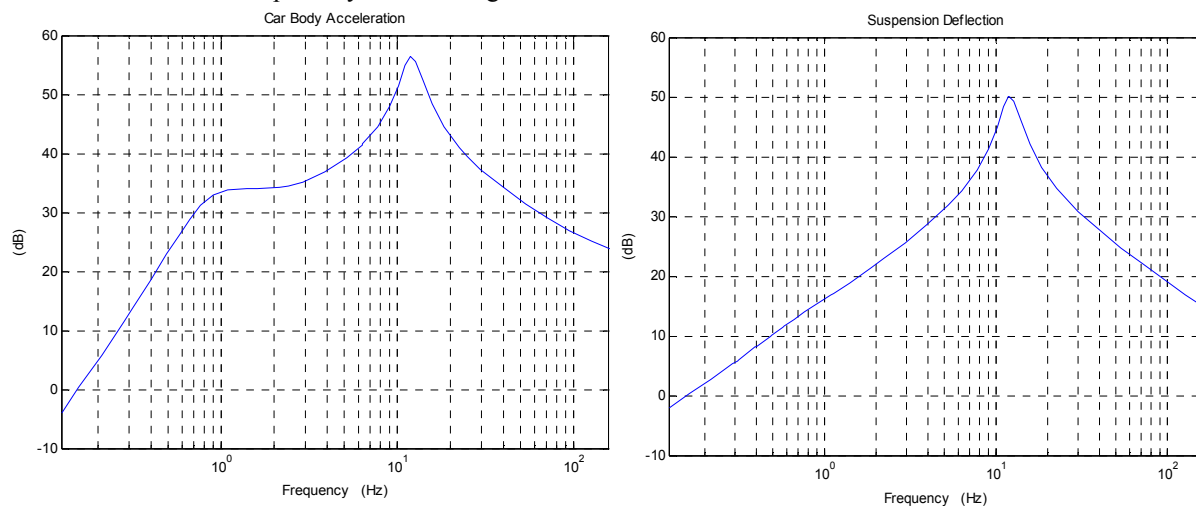
Conclusion

The analysis of a new variable stiffness suspension system of the passive case is presented. where is shown that

the system include of a new variable stiffness mechanism of the system proposal raises an development the old-style system as performance in terms suspension deflection, ride luxury, and road holding, for that, the following pinots can be draw:

- 1- The system appear good result with respect of acceleration more than traditional system.
- 2- The suspension system and tire deflection appear well result for road holding.
- 3- The little time of steady state response for all included parameters.

In the future, work will be done by experiment for two cases passive and active suspension where the passive cases consisted from spring and damper of subsystem can be substituted in the active suspension actuator such as hydraulic or pneumatic actuators to produced force generator. The new suspension system will also be examined the influence on roll and pitch dynamics using a half-car model.



Fig(9): Frequency domain Car Body acceleration and Tire deflection

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