ANALYSIS AND PERFORMANCE OF A PROPOSED VARIABLE STIFFNESS SUSPENSION SYSTEM

Mohammed Jawad Aubad* **Hatem RahemWasmi****
*alrobaiymoh@yahoo.com
***Mechanical Engineering Department /Babylon University**
****Mechanical Engineering Department /BaghdadUniversity**

ABSTRACT

A theoretical analysis of a proposed variable stiffness model of the suspension system is studied to improve the traditional suspension system. The fundamental idea of system focused 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 variable by addition stiffness to the suspension system for improvement of 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.

KEY WORDS : performance, proposed variable stiffness, suspension system, ride comfort, suspension deflection.

التحليل والاداء لمقترح نظام تعليق متغير الجساءة

محمد جواد كاظم عبيد **حاتم رحيم وسمي**

الخلاصة :

يتضمن هذا البحث التحليل النظري لمقترح موديل لنظام التعليق متغير الجساءة لغرض تحسين النظام التقليدي في التعليق. الفكرة الاساسية للنظام تتبلور حول الية متغير الجساءة بإضافة نظام فرعي لنظام التعليق يعتمد على التحكم بذراع دوار موازن بين قوتين جانبيتين يكون متكون من دعامة تحكم عمودية متغير الحمل المنتقل بواسطة الذراع الدوار الحاوي للنابض والمخمد على الجانب الاخر يكون متصل في نقطة دوارة و مثبتة على بدن السيارة . الدراسة لنظام الجساءة المتغيرة بإضافة النابض لنظام التعليق لتحسين الاداء كان افضل من الانظمة المماثلة او التقليدية. المبدأ الاساس كان بوصف اداء خواص النظام السلوكية باقل تعجيل لبدن السيارة للدلالة على الركوب المريح و اقل تعليق و تشوه في الاطار بالنسبة الى قيادة الطريق.

كلمات دالة: اداء، مقترح متغير الجساءة،نظام التعليق، الركوب المريح، تشوه التعليق.

Received:2016-6-19
Accepted:2016 -10-18

INTRODUCTION :-

The isolation of the 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 minimize 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 the requirement of the specifications design of vehicle. Therefore, researchers carried out researches on non-passive suspension system **. Chen (2006)** .The types of the suspension systems can be divided as passive system, semi-active system and active suspension system. The suspension system in passive case is considered simplest, not required external energy and cheap but in case of semi-active suspension can be advanced 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 semi-active case 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 changed by a range of investigation, 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)** presented 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)** depicted 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)** presented 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 other papers utilized 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 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 vary the damping coefficient temporarily, in suspension optimization, both the damping coefficient and the spring rate of the suspension components are normally utilized as trust worthy 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 presents proposed and theoretical investigate a variable stiffness of the new modal for suspension system for the quarter car in passive case, 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 the variable stiffness system is shown in Figure (1). The Lever arm OB, of length L, is stuck at an altered point O and allowed to rotate 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 which 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 comprises from two component L1 and L2 and turning about E by ψ . The sign is to change 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. Consideration the system of suspension as shown in Figure(2). The schematic model is consisting 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 presented in Figure(1) for the mechanism of the variable stiffness of Figure(2). The Vertical regulator force U used to control by rotation of arm AD with angle ψ which in opportunity to control on the mechanism for overall stiffness. The modulation of tire is considered as a linear spring for both spring constant and damping coefficient
The assumptions assumed in Figure(2) can be summarized as:

1- The side translation or horizontal movement of the sprung mass is ignored, which is only the vertical displacement *Zs* is measured.

2- The angle camber of the 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.

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 the following equations hold:

$$y_A = L_1[\cos(\varphi - \varphi_0)] \quad (1a)$$
$$z_A = z_s + L_1[\sin(\varphi - \varphi_0)] \quad (2a)$$
$$y_B = L_B[\cos(\theta - \theta_0)] \quad (3a)$$

$z_A = z_s + L_B[\sin(\theta - \theta_0)]$ (4a)
$y_C = L_B[\cos(\theta - \theta_0)]$ (5a)
$z_C = z_s + L_C[\sin(\theta - \theta_0)]$ (6a)
$y_D = L_D[\cos(\varphi - \varphi_0)]$ (7a)
$z_D = z_s + L_D[\sin(\varphi - \varphi_0)]$ (8a)

Where (θ_0) and $(\varphi_0)$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.

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

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

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

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. Therefor the generalized coordinates of the system are:

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

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

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

$$m_t L_c \ddot{z}_s + (m_t L_c^2 + I_1)\ddot{\theta} + C_t L_c \dot{z}_s + C_t L_c^2 \dot{\theta} + C_s L_B^2 \dot{\theta} - C_s L_B L_1 \dot{\varphi} + k_t L_c z_s +$$

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

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

State Space Analysis

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

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 = F_1(x_1, x_2, x_3, x_4, x_5, x_6, z_r)$$

$$\dot{x}_3 = x_4$$

$$\dot{x}_4 = F_2(x_1, x_2, x_3, x_4, x_5, x_6, z_r)$$

$$\dot{x}_5 = x_6$$

$$\dot{x}_6 = F_3(x_3, x_4, x_5, x_6) \quad (1d)$$

$$F_1 = \frac{-1}{Dy}\{(Dk_t + k_tL_c)z_s + (k_tL_cD + k_tL_c^2 + k_sL_B^2)\theta + (C_tD + C_tL_c)\dot{z}_s + (C_tL_cD + C_tL_c^2 + C_sL_B^2)\dot{\theta} - c_sL_1L_B\dot{\varphi} - k_sL_1L_B\varphi - (k_tD + k_tL_c)z_r - (C_tD + C_tL_c)\dot{z}_r\} \quad (2d)$$

$$F_2 = \frac{-1}{D\theta}\{(D'^{C_t} + C_tL_c)\dot{z}_s + (c_tL_cD' + C_tL_c^2 + C_sL_B^2)\dot{\theta} + (k_tD' + k_tL_c)z_s + (k_tL_cD' + k_tL_c^2 + k_sL_B^2)\theta - c_sL_1L_B\dot{\varphi} - k_sL_1L_B\varphi - (k_tD' + k_tL_c)z_r - (C_tD' + C_tL_c)\dot{z}_r\} \quad (3d)$$

$$F_3 = \frac{-1}{I_2}\{k_sL_1L_B\theta + C_sL_1L_B\dot{\theta} - (k_sL_1^2 + k_uL_2^2)\varphi - (C_sL_1^2 + C_uL_2^2)\dot{\varphi}\} \quad (4d)$$

Where

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

$$D\theta = -\frac{m_tL_c^2 + I_1}{m_s + m_t} + (m_tL_c^2 + I_1), \qquad D = \frac{m_tL_c^2 + I_1}{m_tL_c} \qquad D' = -m_tL_c/(m_s + m_t)$$

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 to disregard the sine or cosine function and whose equilibrium is nonzero.

$\dot{x} = Ax + Bz_r$ And $y = Cx + Dz_r$

Where

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

$$a21 = \frac{-(Dk_t + k_tL_c)}{Dy}, a22 = \frac{-(DC_t + C_tL_c)}{Dy}, a23 = \frac{-(DL_ck_t + k_tL_c^2 + k_sL_B^2)}{Dy},$$
$$a24 = \frac{-(DL_cC_t + C_tL_c^2 + C_sL_B^2)}{Dy}, a25 = \frac{k_sL_1L_B}{Dy}, a26 = \frac{C_sL_1L_B}{Dy}, \quad a41 = \frac{-(D'k_t + k_tL_c)}{D\theta}$$

$$a42 = \frac{-(D'C_t + C_tL_c)}{D\theta}, a43 = \frac{-(D'L_ck_t + k_tL_c^2 + k_sL_B^2)}{D\theta}, a44 = \frac{-(D'L_cC_t + C_tL_c^2 + C_sL_B^2)}{D\theta}$$
$$a45 = \frac{k_sL_1L_B}{D\theta}, a46 = \frac{C_sL_1L_B}{D\theta}, a63 = \frac{k_sL_1L_B}{D\theta}, a64 = \frac{C_sL_1L_B}{D\theta}, a65 = \frac{-(k_sL_1^2 + k_uL_2^2)}{I_2}$$

$$, a66 = \frac{-(C_sL_1^2 + C_uL_2^2)}{I_2}$$

$$B = \begin{bmatrix} 0 & \frac{\partial F_1}{\partial z_r} & 0 & \frac{\partial F_2}{\partial z_r} & 0 & \frac{\partial F_3}{\partial z_r} \end{bmatrix}^T = \begin{bmatrix} 0 & \frac{k_t D + k_t L_c}{Dy} & 0 & \frac{k_t D\prime + k_t L_c}{D\theta} & 0 & 0 \end{bmatrix}^T \quad (2e)$$

D and C matrices depend on the output values from the system, and the $Z_r(t)$ presents 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 of The Stiffness And Damping Variation On Suspension System

A simulation study of the effect variable stiffness and damping on the suspension performance is presented. The performances of interest are the ride comfort, 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 Ks and the damping coefficient Cs 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 systems.

The characterizes suspension of the system is presented by competence the ride luxury, road property and traveling. These performance principles are specified in expressions of the acceleration, body of the 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 presented as **Anubi (2013)**

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

Where CBA: car body acceleration

SD: suspension deflection, and TD: tire deflection

For each pair (Ks; Cs) 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. 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:

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

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 (2013)** 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 of suspension system are fixed in Table [2] by take the same value of the first model (Model for Anubi) with respect of the masses, springs and dampers, but for the length L_1 and L_2 by taken the values are chosen to give the smallest data for acceleration of body car, suspension and tire deflection. From the above figures and comparison with **Anubi(2013)** at passive case, one can see the following:

1- The value of acceleration of body car is little and suitable compared with the results at **Anubi** (**2013)** 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 of 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 show smallest than values of paper **Anubi** (**2013).** As shown in table (3)

RESULTS AND TIME DOMAIN :

The results obtained in the figures above discuss the effect of the same parameters effect on the behavior of the proposed suspension system such as increasing the damping a of the subsystem added to the suspension system in fig. (4) And fig. (5) Increase the damping to (2500, 3000 N.s/m) indicate decreasing the acceleration and suspension deflection.

In fig.(6,7)study the effect of increasing the length of arm control by expanding (L_1 =0.25 or L_2=0.35),shown that the behavior of the 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 tend to increase, for that the length of arm control has very sensitivity to change because it is as stabiles arm between the force effect on sides.

In fig.(8) shown an 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 in to the way of execution determined by the idea of variance gain **Stack(1995).** 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 energized by the sinusoid r = A sin(wt); t ϵ [0, 2πN/w], 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. **Schoukens(2001).**

$$G(wj) = \sqrt{\frac{\int_0^{2\pi N/w} z^2 dt}{\int_0^{2\pi N/w} A^2 sin^2 wt\, dt}}$$

Fig.(9) display the variance gain plots for the acceleration of body car and suspension defection respectively. In this figure shown that the variable stiffness of the suspension gets healthy isolation of vibration according to range of the sensitive frequency for human (4-8Hz) **Liu(2008),** then improve handling in severity region (> 59H) **Sung(2007).**

CONCLUSIONS :-

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 a development the old-style system as performance in terms of suspension deflection, ride comfort, and road holding, for that, the following pinots can be drawn :

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

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

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

Table(3): Root Mean Square values of results

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

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

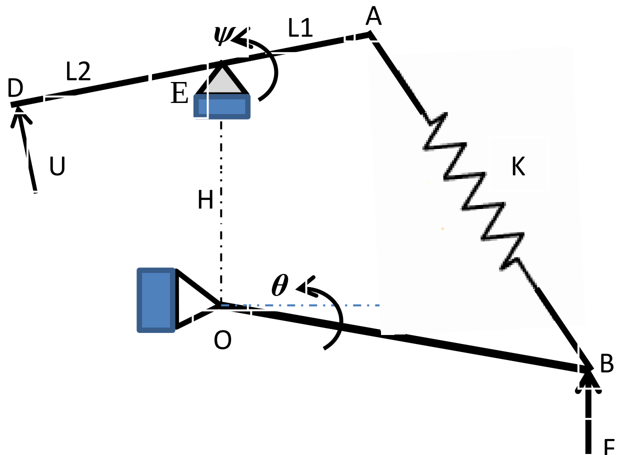


Fig.(1): Variable Stiffness Mechanism

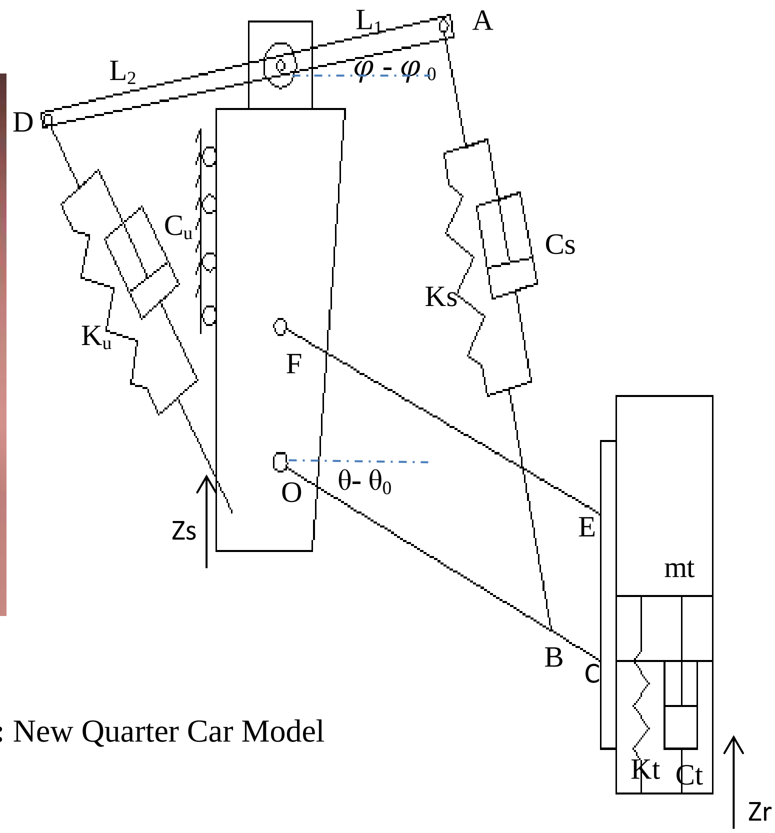
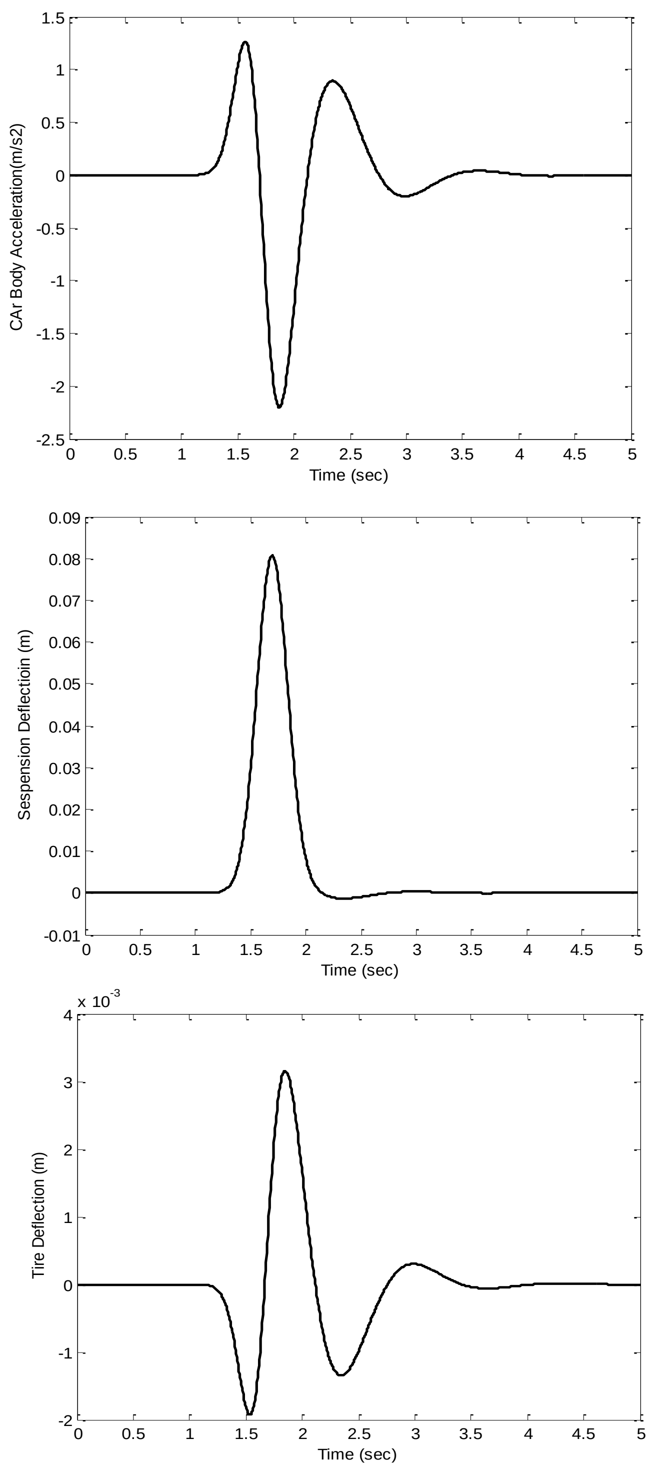
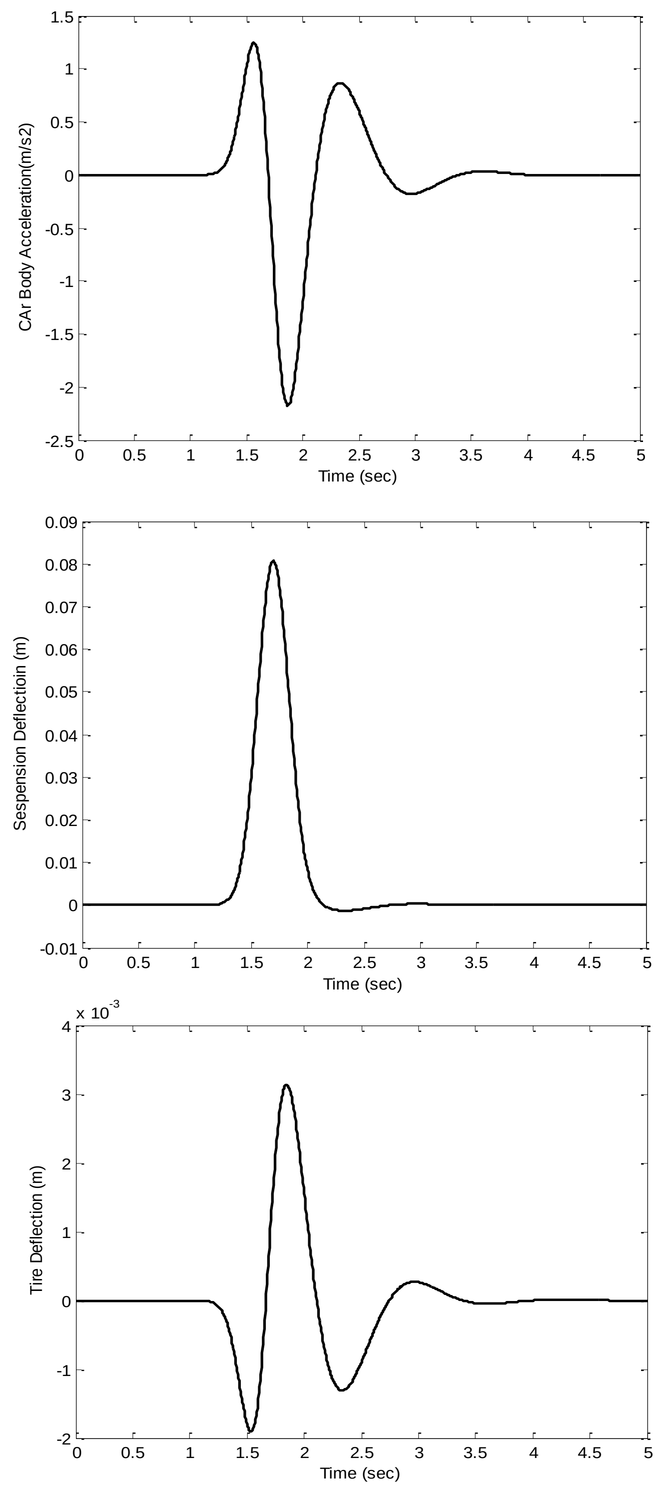


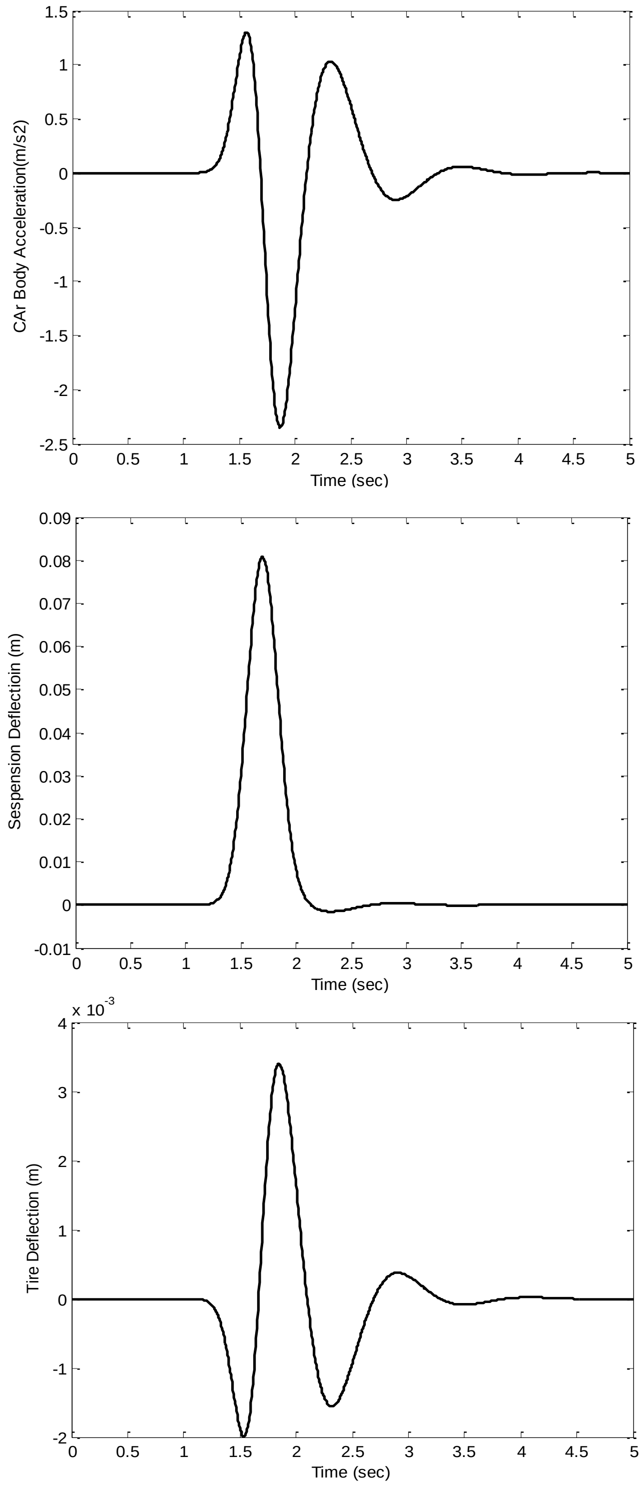
Fig.(2): New Quarter Car Model



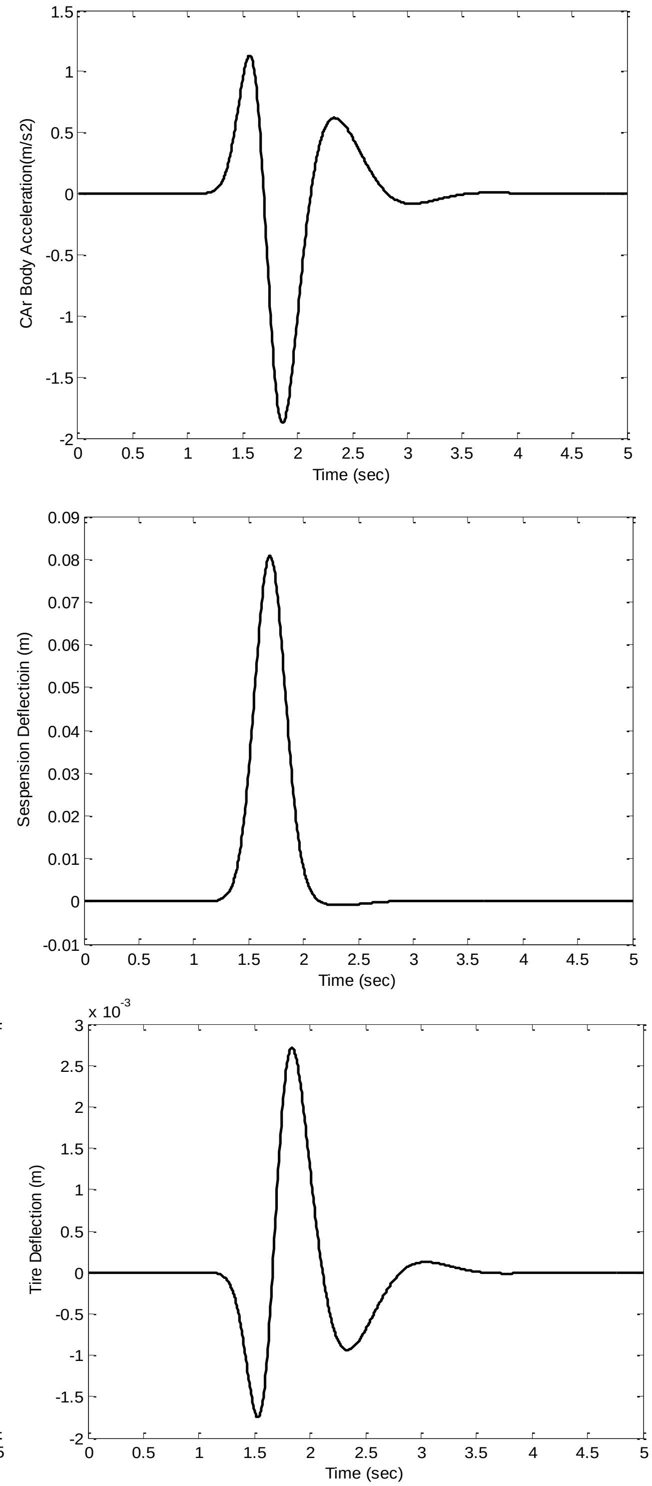
Fig(3): Car Body acceleration , Suspension Deflection and Tire deflection



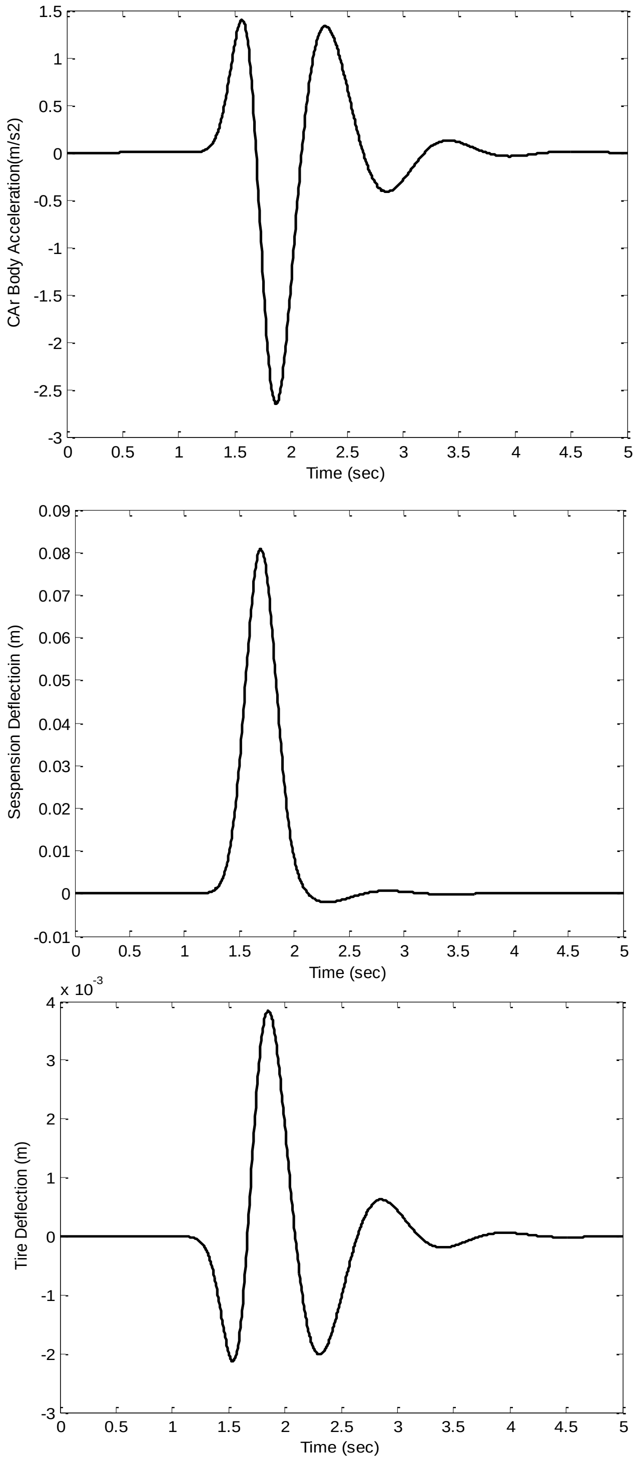
Fig(4): Car Body acceleration ,Suspension Deflection and Tire Deflection For k_u=4000N/m c_u=2500N.s/m



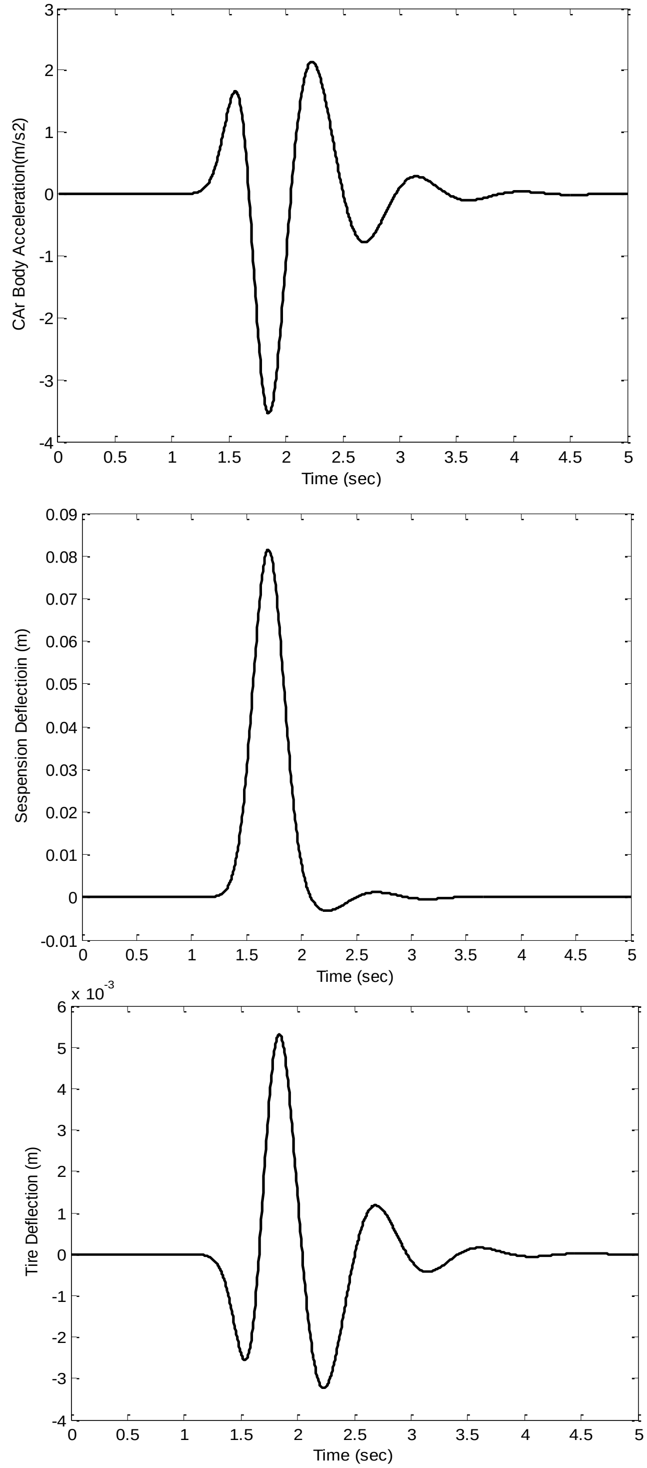
Fig(5): Car Body acceleration ,Suspension Deflection and Tire deflection For k_u=4000N/m c_u=3000N.s/m



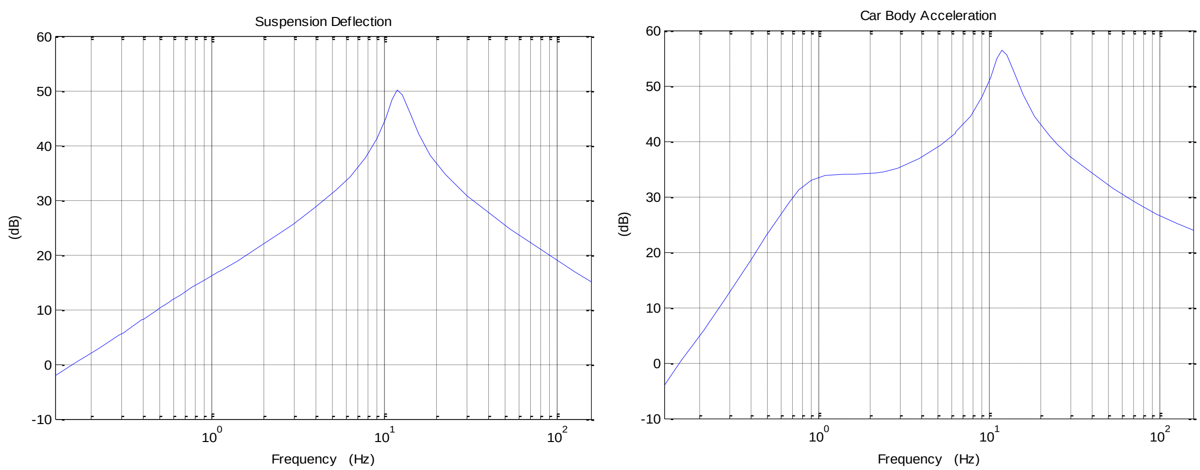
Fig(6): Car Body acceleration ,Suspension Deflection And Tire deflection For k_u=4000N/m c_u=3000N.s/m, L_1=0.25m,L_2=0.3m



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



Fig(8): Car Body acceleration ,Suspension Deflectionand Tire deflection For k_u=4000N/m c_u=3000N.s/m, L_2=0.35m,L_1=0.2m,L_B=0.4m



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

REFERENCES :-

Anubi O. M., D. R. Patel, and C. D. Crane,2013" A new variable stiffness suspension system: passive case" Mechanical Sciences. Vol. 4, pp.139–151.

Anubi, O. M., Crane, C., and Ridgeway, S. 2010" Design and Analysis of a Variable Stiffness Mechanism" Proceedings IDETC/CIE2010. ASME 2010 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference.

Anubi O. M, Carl D. Crane, 2014" A new active variable stiffness suspension system using a nonlinear energy sink-based controller" Vehicle System Dynamics.

Ashfak, A., Saheed, A., Abdul-Rasheed, K. K., and Jaleel, J. A. 2011: Design, Fabrication and Evaluation of MR Damper, International Journal of Aerospace and Mechanical Engineering. Vol. 1,pp.27–33.

Alkhatib, R., Jazar, G. N., and Golnaraghi, M. F. 2004 "Optimal design of Passive Linear Suspension Using Genetic Algorithm, Journal of Sound and Vibration, vol.275,pp. 665–691.

Chen Jiarui, 2006 "Structure of Automobile," China Communications Press, Beijing

Do, A.-L., Sename, O., and Dugard, L. 2010" An LPV Control Approach for Semi-active Suspension Control with Actuator Constraints" in American Control Conference.

I. Fialho and G.J. Balas, 2002 "Road Adaptive Active Suspension Design Using Linear Parameter-Varying Gain-Scheduling, IEE Transactions on control systems technology vol.10, pp. 43–54.

K.S. Hong, D.S. Joen, and H.C. Sohn,1999 “A New Modeling of the Macpherson Suspension System and its Optimal Pole-Placement Control” Vehicle System Dynamics.

.

Karnopp D, Crosby M, Harwood R. 1974 ”Vibration control using semi-active force generators”. J. Eng Ind.vol. 96, pp.619–626.

T. R. Rao, G. V. Rao, K.S. Rao & A. Purushottam,2010” Analysis Of Passive And Semi Active Controlled Suspension Systems For Ride Comfort In An Omnibus Passing Over A Speed Bump” IJRRAS,Vol.5,pp.7-16.

Karnopp, D. 1983: Active damping in road vehicle suspension systems, Vehicle System Dynamics, vol. 12, pp.291–316.

Kobori, T., Takahashi, M., Nasu, T., Niwa, N., and Ogasawara, K. 1993.”Seismic response controlled structure with active variable stiffness system” Earth quake Engineering Structure Dynamic. vol. 22, pp.925–941,

Gongyu Pan and F. Fan, 2011 ” Fuzzy Control and Simulation of a New Semi-Active Suspension” International Conference on Transportation, Mechanical, and Electrical Engineering (TMEE), p.p. 2290-2293.

Schoukens, J., Pintelon, R., Rolain, Y., and Dobrowiecki, T. 2001 “Frequency Response Function Measurements in the Presence of Nonlinear Distortions” Automatica, 37, 939–946.

Stack, A. J. and Doyle, F. J. 1995 ”A Measure for Control Relevant Nonlinearity, in: American Control Conference, Seattle, 2200–2204.

Sung, K., Han, Y., Lim, K., and Choi, S. 2007 ” Discrete-time fuzzy sliding mode control for a vehicle suspension system featuring an electro rheological fluid damper” VTT Symp., 16, 798–808.

Y. Liu_, H. Matsuhisa, H. Utsuno, 2008 ” Semi-active vibration isolation system with variable stiffness and damping control” Journal of Sound and Vibration 313 pp.16–28.

Yang Lin, Zhao Yuzhuang, Chen Sizhong and Wu Zhicheng,Jan. 2011,”Neural Network Model Reference Control for Semi-Active Hydro-Pneumatic Suspension” Transactions of Beijing Institute of Technology, vol. 31, pp:24-28.

Zin, A., Sename, O., Basset, M., Dugard, L., and Gissinger, G. 2004 ” A nonlinear vehicle bicycle model for suspension and handling control studies” in: Proceedings of the IFAC Conference on Advances in Vehicle Control and Safety (AVCS), Genova, Italy, 638 – 643.