AN ACTIVE SELF-TUNING SUSPENION SYSTEM TO IMPROVE DRIVER COMFORT

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ABSTRACT

This paper presents the simulation of two dimension a half-vehicle self –tuning active suspension system to simultaneously improve vehicle ride comfort. A validated 4-DOF of vehicle linear model was used to study the performance of passive suspension system and compared with the developed active suspension system. The governing equations of motion for the self- tuning active suspension was derived and used to reduce the effect of disturbances to the dynamics performance of the vehicle, which appear when the vehicle excited by a semi-circular sinusoidal bump road of a (0.1 m) height. The performance of passive suspension and the self-tuning active suspension are demonstrated by simulations and specially the vertical acceleration and the vertical root mean square (RMS) acceleration to observe the effect of the proposed system to the ride comfort. The active suspension system introduced in this work show good results for improving the ride comfort.

Keywords: ***Self-tuning active suspension, ride comfort.***

نظام تعليق ذو تضبيط ذاتي فعال لتحسين راحة السائق

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الخلاصة:

في هذا البحث نموذج محاكاةَ لموديل ثنائي البعد خطي ذو اربع درجات من الحرية لنِصْف مركبةٍ تحتوي على نظامِ تعليقِ فعال ذو تَضْبيط ذاتيٍ لتَحسين راحةِ الراكب بشكل آني. استند البحث على نموذج خطي كَانَ يُستَعملُ لدِراسَة أداءِ نظامِ التعليقِ السلبيِ للمقارنة بنظامِ التعليقِ النشيطِ المطورِ. تم اشتقاق المعادلات الحَاكِمة للحركةِ لنظام التعليقَ النشيطَ واستخدمت بتقليل تأثير الاضطرابات على أداءِ ديناميكا المركبة، الذي يَظْهرُ عندما تثار العربةَ بصدمة طريق على شكل نصف دائري لموجية جيبيه بارتفاع 0.1 متر. إنّ أداءَ التعليقِ السلبيِ وتعليقِ التَضْبيط الذاتيِ النشيطِ تم عرضه بالمحاكاةِ وخصوصاً التعجيلِ العموديِ

ومعدل مربع الجذر للتعجيل لمُلاحَظَة تأثيرِ النظامِ المُقتَرَحِ على راحةِ الراكب . وقد وجد انه هناك نتائج تظهران نظام التعليقِ النشيطِ المقدم في هذا البحثِ قد اعطى نتائج جيدة لتَحسين راحةِ الراكب.

NOMENCLATURE

M_s Sprung mass

M_u Un sprung mass.

$\ddot{Z}_s$ Vertical acceleration of sprung mass.

$\dot{Z}_s$ Vertical velocity of sprung mass.

Z_s Vertical displacement of sprung mass

C_s Main suspension coefficient of damping.

K_s Main suspension stiffness.

Z_t The centre of tyer (wheel) vertical displacement.

$\dot{Z}_t$ The centre of tyer (wheel) vertical velocity.

$\ddot{Z}_s$ The sprung mass CG Vertical acceleration.

$\ddot{Z}_t$ The centre of tyer (wheel) vertical acceleration.

Y_r Road profile vertical amplitude.

C_a Self-tuning Damper coefficient of damping.

ω The force frequency.

C.G The sprung mass center of gravity.

INTRODUCTION:

Ride quality and handling performance of road vehicles are very much affected by the design of the vehicle suspension systems. Good ride quality requires high damping setting at low frequencies to stifle bounce, roll and pitch, and lower damping settings at higher frequencies to avoid ride harshness. Unlike passive systems, which can only store or dissipate energy, active suspensions can continuously change the energy flow to or from the system when required, recently, the subject of active suspension design has been intensively reviewed by Jean-Gabriel Roumy [1], M .Frechin [2], and Mario Milanese and Carlo Novara [3], a simulation procedures by Jean-Gabriel Roumy [4], and Pakharuddin Mohd Samin, Hishamuddin Jamaluddin [5] . An optimization of active suspensions by applying methods of modern control theory has been reported by Bassam A. [6].

The vehicle suspension provides a means of isolating the vehicle's body from the road inputs. Several aspects of vehicle dynamics put different demands on the various suspension components. Occupant comfort requires the minimization of sprung mass accelerations, while lateral dynamic performance requires good road holding giving rise to a need for consistent normal forces at the tire interface. This all has to work within suspension rattle space and tire deflection limitations. In a passive suspension each improvement comes at the expense of performance in another area [10].In this work, it is proposed to develop an analytical self-tuning active suspension to effectively reduce the vehicle body acceleration for ride comfort, dynamic tyers deflections and body attitude for ride handling and suspension deflections for the purpose of packaging by means of active suspensions governing equations of motion. The objectives include: (1) Development of a half car model, which is adequate for understanding the effect of road disturbances on the ride and handling characteristics of the vehicle; (2) Deriving the optimal governing equations of motions of a self -tuning active system (3) Investigating the possibilities of performance improvements using the passive and active developed system for a simulation to see the vibration response of wheel center behavior and the C.G acceleration to study its effects on ride comfort.

ANALYTICAL FORMULATION:

1. Model of Vehicle Ride Dynamics:

The two dimensional 4-DOF half vehicle model as shown in figure 1 is based on a passive FEM vehicle model being validated by simulation results by Salim Y. Kasim [7], So in this work the passive suspension is replaced with an self-tuning active suspension which is represented parallel damper to the passive one, vertical motion in the z- direction, pitch motions about the pitch pole. It also consist two unsprung masses which are free to bounce vertically with respect to the sprung mass. The vehicle model parameters data used for the simulation were taken from [9] as shown in table (1).

2. Vehicle Model Governing Equations:

The equations governing the dynamic motion of the vehicle model equipped with self-tuning active and passive suspension systems can be expressed in the following state form:

$$M_s\ddot{Z}_s = -C_s(\dot{Z}_s - \dot{Z}_t) - K_s(Z_s - Z_t) - C_a(\dot{Z}_s - \dot{Z}_t) \quad (1)$$

$$M_u\ddot{Z}_t = C_s(\dot{Z}_s - \dot{Z}_t) - K_s(Z_s - Z_t) - K_t(Z_t - Y_r) + C_a(\dot{Z}_s - \dot{Z}_t) \quad (2)$$

Where:

$C_a(\dot{Z}_s - \dot{Z}_t)$: Self-tuning Damper force.

And the amplitude of vibration for both sprung and un sprung masses can be written as follows:

The vertical amplitude of vibration of the vehicle sprung mass is:

$$Z_s = \frac{M_S\omega^2 Z_t}{\sqrt{(K_s - M_s\omega^2)^2 + (C_s\omega)^2}} \quad (3)$$

And the vertical amplitude of vibration of the un sprung mass is:

$$Z_t = \frac{M_u\omega^2 Y_r}{\sqrt{(K_t - M_u\omega^2)^2}} \quad (4)$$

The displacements, velocity, and acceleration with respect to time can be obtained.

3. Vehicle and Bump Road Input:

The input excitation to the vehicle model is assumed to be the apparent vertical roadway motion, caused by the vehicle's forward speed along a road having a semi-circular sinusoidal profile of 0.1 meter amplitude and 0.6 meter length as shown in figure 8. The vehicle excitation model can be obtained in the shape of vertical elevation and horizontal distance as tabulated input.

RESULTS AND DISCUSSIONS:

A new approach of an active suspension self- tuning damper has been achieved, according to the results it clear that from the observation of the front and rear road wheels vertical displacement shown in figure (2), that the amplitude have been reduced to about the half . also in figure (4) it is clear that the in addition to a code contact between the road wheel and the terrain the vertical spring mass C.G's amplitude is reduced from which one can deduced the results shown in figure (3) with high response and low settling time. The root mean square (RMS) acceleration is one of a ride comfort criteria which is shown in figure (5) for a smooth low values in active compared to that in the passive one. In the low speed of the vehicle in both case active and passive there are deference in the response can be observed the reason is that a good contact were exist between the road profile and the tyre, but the deferent is appear small when the speed is increased, except at some speeds mainly at 40 km/h which is seem to be the critical speed, these facts can be observed in figures (5) and (7). And in this case a sudden shock can be avoided when the speed of the vehicle has increased. The proposed system show a good time response if we checked the simulation time and the length of the road bump which is 0.6 meter and the critical time need from the system to reply is about two second in this case we need a high response system and it is clear from the results shown in figures (2 through 7) that the system has managed to do. And the reduction of the vertical acceleration is about 2 times. It means that the proposed system has achieved the objects

CONCLUSION:

A self- tuning suspension simulation system based on an additional damper added to the passive one in a 2-D half vehicle 4 DOF model the new damper has the ability to modified itself according to the irregular road profile after it received a signal from the detector where it can be achieved on reading the input tabulated data before the first wheel reach it, and adjust itself to maintained a suitable damping help for overcome the non-uniformly road to make the ride comfort be better. The results have shown that the proposed system has achieved the aim.

REFRANCES:

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Parameter	Value	Description
kt1	200 kN/m	front tire stiffness
Ct1	125 N-s/m	font tire damping
mu1	100 kg	front suspension unsprung mass
ks1	30 kN/m	front suspension stiffness
Cs1	750 N-s/m	font suspension damping
Ixx	2704 kg-m^2	pitch rotational inertia
Ms	1700 kg	vehicle sprung mass
ks2	20 kN/m	rear suspension stiffness
Cs2	750 N-s/m	rear suspension damping
mu2	80 kg	rear suspension unsprung mass
kt2	200 kN/m	rear tire stiffness
Ct2	125 N-s/m	rear tire damping
L2	1.1 m	length from center of mass to rear axle
L1	1.6 m	length from center of mass to front axle

Table 1: Generic sedan vehicle parameters.

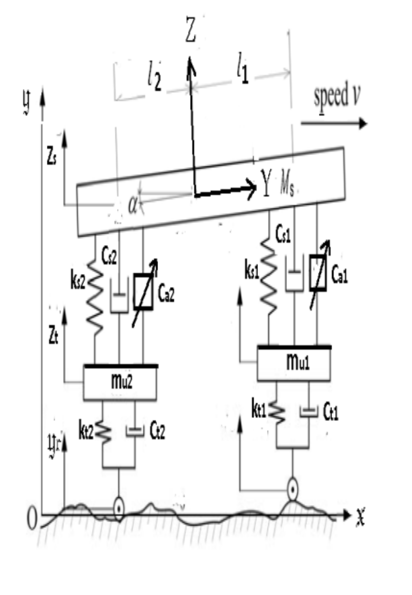
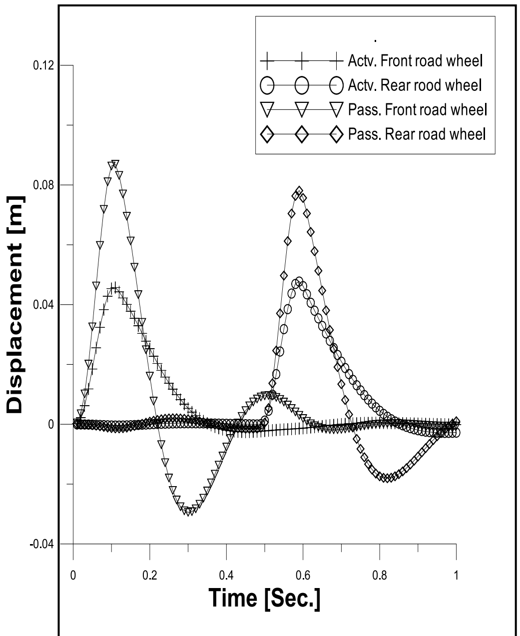


Figure (1) 2-D Half car model with self -tuning suspension.



Figure(2) comparison of unsprung Mass Displacement

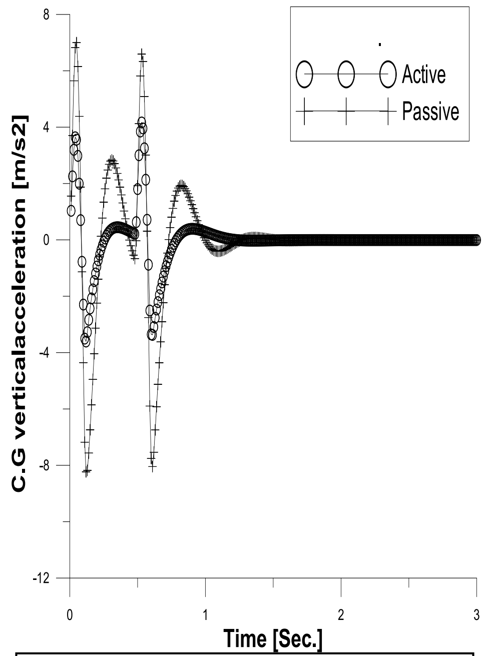
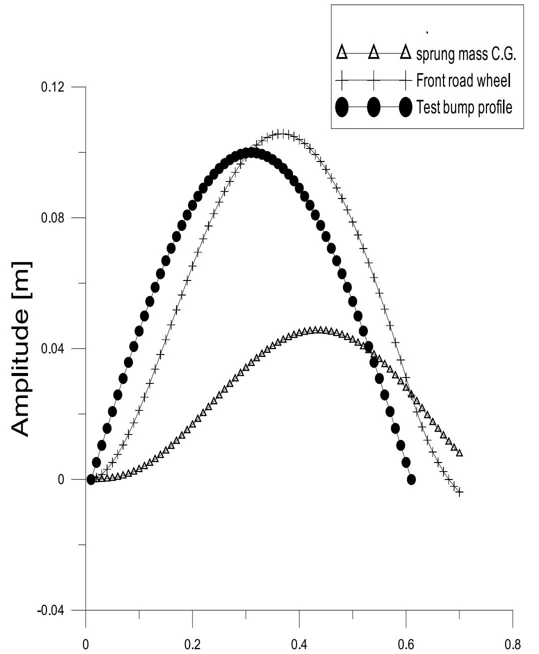


Fig (3) Sprung Mass C.G Vertical Acceleration Vs Time At 20 Km/h



Fig(4) Front wheel , sprung mass C.G displacements , and road test bump profile Vs Time at low speed

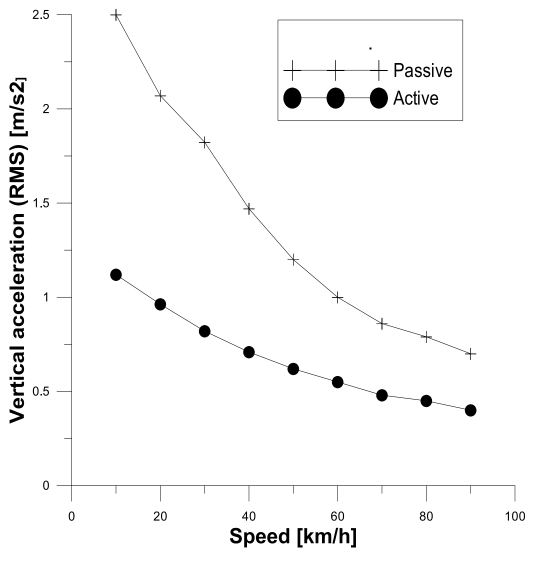


Fig (5) Sprung Mass C.G root mean square Vs vehicle speeds

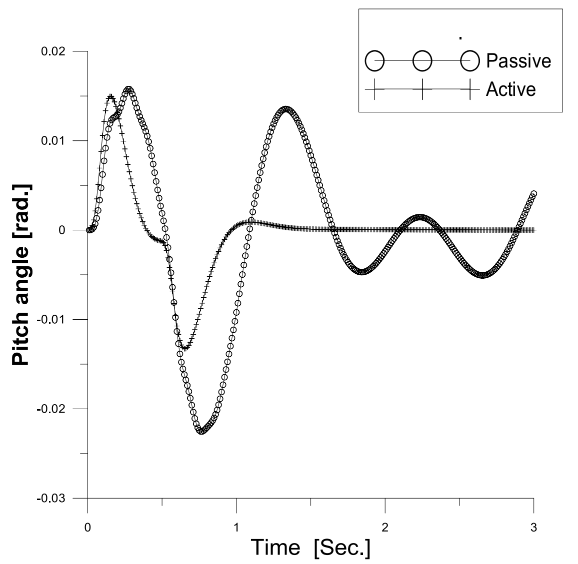


Fig (6) Sprung Mass C.G Pitch angle Vs Time At 20 Km/h

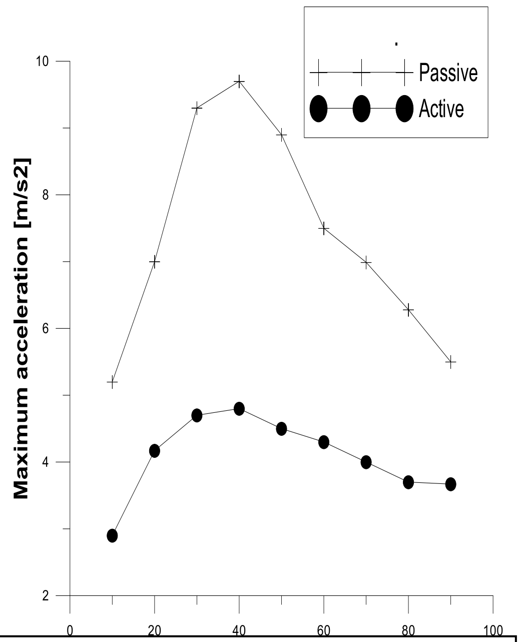
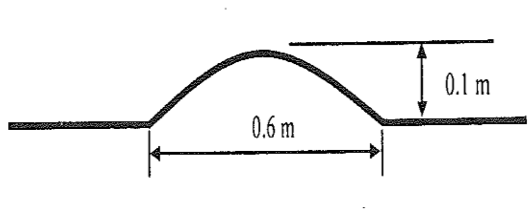


Fig (7) Sprung Mass C.G maximum acceleration Vs vehicle speeds.



Figure(8) The test bump road used in this study.