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SEMI-ACTIVE SUSPENSION FOR RIDE IMPROVEMENT USING STABILITY AUGMENTATION SYSTEM CONTROL ALGORITHM

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ABSTRACT

This paper presents the simulation of a full-vehicle semi-active suspension system to simultaneously improve vehicle ride comfort. A validated 7-DOF of vehicle model was used to study the performance of passive suspension system and theoretical desired force for the designed semi-active suspension system. The control algorithm developed for the semi-active suspension known as stability augmentation system (SAS) was used to reduce the effect of disturbances to the dynamics performance of the vehicle. The SAS with inner ride control loops and outer attitude loops are combined together by the input decoupling transformation scheme (DTS) to improve ride. The performance of passive suspension and the semi-active suspension control are demonstrated by simulations.

Keywords: *Semi-active suspension, ride improvement, stability augmentation system*

1.0 INTRODUCTION

In recent years, semi-active suspension systems are getting more attention than active suspension system because of their low cost, low power requirement, fail safe usage and competitive performance to the active suspension system [1-3]. The concept of semi-active suspension system was proposed by Karnopp's in the early of 1970's [4]. It was reported that the semi-active suspension system could be nearly as effective as fully active suspensions in improving ride quality [5].

The intent of this paper is to investigate the effectiveness of semi-active suspension in reducing unwanted vertical motion of passenger vehicle due to passing a bump or pot holes. A 7-DOF full vehicle model [6] is used with semi-active suspension system designed for the vehicle model to improve ride comfort. The study on ride and handling using active suspension control with 9 DOF [7]

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and 6 DOF [8], linearized 7 DOF [9,10] for ride only with active suspension has been studied. The control algorithm considered in this study is stability augmentation system adopted from stability augmentation system [9].

The paper is organized as follows. Section 2 describes mathematical of 7-DOF full vehicle model. Control structure of semi-active suspension system is in section 3. Furthermore, results of simulation of controller performance by comparing passive and semi-active suspension in the fourth section and finally conclusion is made in Section 5.

2.0 MATHEMATICAL 7-DOF VEHICLE MODEL

The 7-DOF vehicle model as shown in Figure 1 is based on a passive vehicle model being validated by experimental and simulation results with passive suspension being replace with an semi-active suspension which is represented as vehicle body three degrees of freedom, vertical motion in the z direction, roll and pitch motions about the roll center and pitch pole respectively. It also consist four unsprung masses which are free to bounce vertically with respect to the sprung mass.

The suspensions between the sprung mass and unsprung masses are modeled as passive viscous dampers and spring elements. The vehicle model parameter data used for the simulation were taken from a Malaysia National Car.

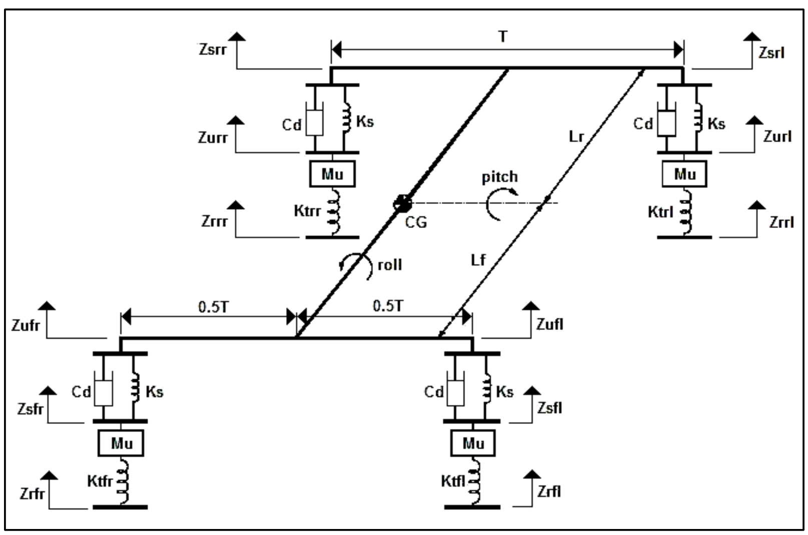

Figure 1: 7-DOF of ride model

Semi-Active Suspension Model (7-DOF)

1. Vehicle body (sprung mass) motions

$$M_s \ddot{Z}_b = \{(F_{sfl}+F_{afl})+(F_{sfr}+F_{afr})+(F_{srl}+F_{arl})+(F_{srr}+F_{arr})\} \qquad (1)$$

M_s is the vehicle sprung mass, $\ddot{Z}_b$ is the body vertical acceleration, F_{sij} is spring force at each corner and F_{aij} is the semi-active suspension force with the index (*i*) indicating front, (*f*) or rear (*r*) tyres and (*j*) indicating left (*l*) or right (*r*) tyres.

$$I_{pc}\ddot{\theta}=\{(F_{srl}+F_{arl}+F_{srr}+F_{arr})\cdot L_r\}-\{(F_{sfl}+F_{afl}+F_{sfr}+F_{afr})\cdot L_f\} \quad (2)$$

I_{pc} is the moments of inertia of the vehicle about the pitch center, $\ddot{\theta}$ is pitch acceleration, L_f and L_r denote the distance between the centre of gravity to the front and the rear axles, respectively.

$$I_{rc}\ddot{\phi}=\{(F_{sfl}+F_{afl}+F_{srl}+F_{arl})T/2\}-\{(F_{sfr}+F_{afr}+F_{srr}+F_{arr})T/2\} \quad (3)$$

I_{rc} is the moments of inertia of the vehicle about the roll center, $\ddot{\phi}$ is roll acceleration and T is the vehicle track width.

2. Wheel (unsprung mass) motions for all 4 corners.

$$F_{zij} - F_{sij} - F_{aij} = M_{uij}\ddot{Z}_{uij} \quad (4)$$

F_{zij} is tire force at each corner, M_{uij} is the unsprung mass at each corner and $\ddot{Z}_{uij}$ unsprung mass acceleration.

3.0 CONTROL STRUCTURE OF SEMI-ACTIVE SUSPENSION SYSTEM

The control structure of the semi-active suspension system for vehicle model is shown in Figure 2 as a closed-loop system. The control structure adopted from a stability augmentation system (SAS) [9, 11,12].

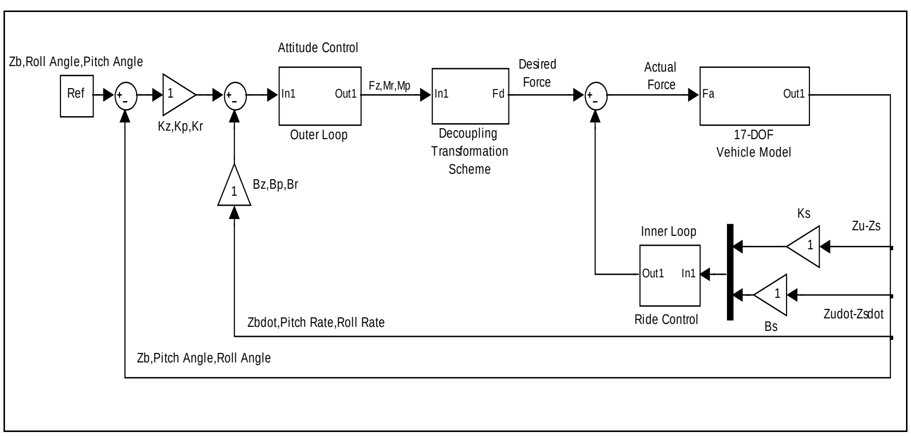

Figure 2: Block diagram for the control structure

The SAS consist of inner loop controller to reject the effect of road disturbances, outer loop controller to stabilize heave, pitch and roll response due to road disturbances and an input decoupling transformation that blend the inner and outer loops in the SAS. Then, inner loop controller provides the ride control that isolates the car body from wheel vibrations induced by road irregularities and outer loop controller provide the attitude control that maintains load-leveling and load distribution during vehicle maneuvers.

The outputs of the outer loop controller as shown in Figure 3 are vertical force to stabilize body bounce (F_z), moment to stabilize body pitch (M_θ) and moment to stabilize body roll(M_φ)

Outer loop controller distributed those force and moments into semi-active suspension using decoupling transformation scheme(DTS). The outputs of the DTS mainly the desired forces of the semi-active suspension are then subtracted with the relevant outputs of the inner loop controller to produce ideal forces of the suspension.

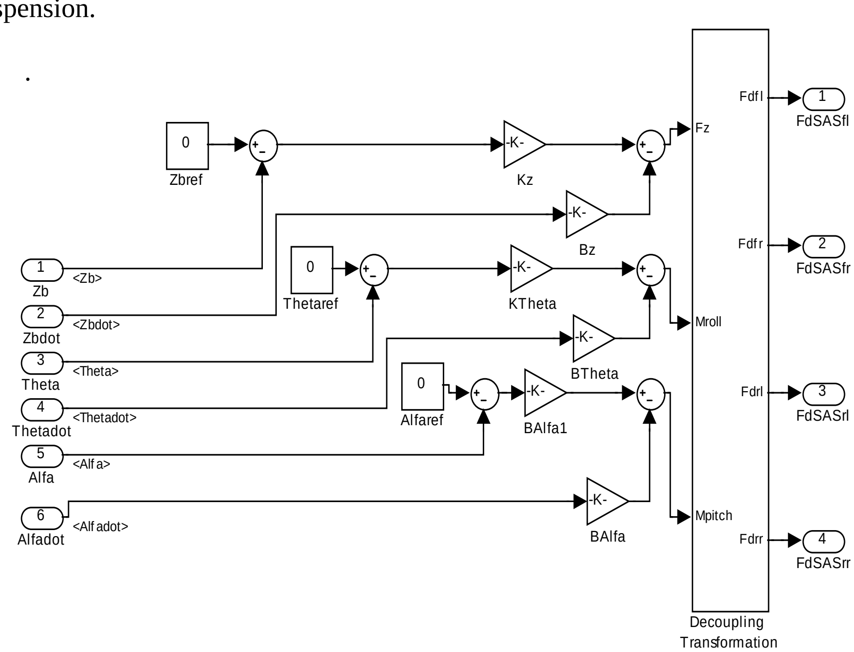

Figure 3: Outer loop and decoupling transformation for SAS

The understanding of the system dynamics in the previous section helps the development of DTS. From Equations (1), (2) and (3), equivalent forces for heave, pitch and roll can be defined by

$$F_z = (F_{afl}) + (F_{afr}) + (F_{arl}) + (F_{arr}) \tag{5}$$

$$M_\vartheta = -(F_{afl})L_f - (F_{afr})L_f + (F_{arl})L_r + (F_{arr})L_r \tag{6}$$

$$M_{\phi} = (F_{afl})0.5T - (F_{afr})0.5T + (F_{arl})0.5T - (F_{arr})0.5T \tag{7}$$

Equations (5), (6) and (7) can be rearranged in matrix format as follows

$$\begin{bmatrix} F \\ M_{\theta} \\ M_{\phi} \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ -L_f & -L_f & L_r & L_r \\ 0.5T & -0.5T & 0.5T & -0.5T \end{bmatrix} \begin{bmatrix} F_{afl} \\ F_{afr} \\ F_{arl} \\ F_{arr} \end{bmatrix} \tag{8}$$

A linear system of equations ***y=Dx,*** if $D \in \Re^{m \times n}$ has full low rank, then there exist a right inverse D^{-1} such that $D^{-1}D = I^{m \times m}$. The right inverse can be computed using $D^{-1} = D^{T}\left(DD^{T}\right)^{-1}$. Thus, the inverse relationship of equation (8) can be expressed as

$$\begin{bmatrix} F_{afl} \\ F_{afr} \\ F_{arl} \\ F_{arr} \end{bmatrix} = \begin{bmatrix} \frac{L_r}{2(L_f + L_r)} & -\frac{1}{2(L + L_r)} & \frac{1}{2T} \\ \frac{L_r}{2(L_f + L_r)} & -\frac{1}{2(L + L_r)} & -\frac{1}{2T} \\ \frac{L_f}{2(L_f + L_r)} & \frac{1}{2(L + L_r)} & \frac{1}{2T} \\ \frac{L_f}{2(L_f + L_r)} & \frac{1}{2(L_f + L_r)} & -\frac{1}{2T} \end{bmatrix} \begin{bmatrix} F_z \\ M_{\theta} \\ M_{\phi} \end{bmatrix} \tag{9}$$

This yields the input decoupling transformation scheme (DTS) shown in Figure 3 that blends the inner and outer control loops.

4.0 CONTROLLER PERFORMANCE IN COMBINATION OF ROLL AND PITCH MODE BUMP TESTS

In these combination test of the roll and pitch modes, the bumps were arranged in a way that it will firstly create the rolling effect when only one side of the vehicle’s wheels start to hop on and hop down the bump followed by the pitching effect (both wheels at the front axle hop on and hop down the bump followed by the wheels at the rear axle). The simulation results show a good improvement by the semi-active suspension system. It can be seen that the performance of the stability augmentation system with ride control is significantly better than the passive system and improve slightly better than without ride control.

It improves most of the body motion parameters that is jerk, the sudden body vertical movement, body acceleration, body heave, body roll rate, body roll angle, body pitch rate and body pitch angle. Table 1 shows the improvement of the semi-active suspension system by root-mean-square (RMS) method.

Table 1: RMS value of passive and semi-active suspension with and without ride control

	Passive	Semi-active w/o ride control	Improvement %	Semi-active with ride control	Improvement %
Jerk	806.63	141.11	82.51	59.71	92.60
Body Accel.	13.63	3.160	76.82	1.424	89.55
Body Vert. Heave	0.03	0.027	21.95	0.019	43.98
Body Roll Rate	29.89	12.764	57.30	9.340	68.75
Body Roll Angle	1.97	1.286	34.80	1.156	41.39
Body Pitch Rate	12.99	5.916	54.49	2.641	79.69
Body Pitch Angle	3.28	1.409	57.07	0.774	76.42

Figures 6 to 12 show significant improvement to all the body responses due to combination of roll and pitch mode bump test with the vehicle moving at constant speed of 50 km/h.

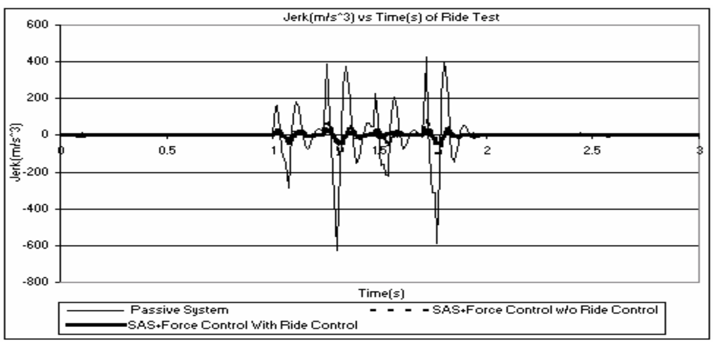

Figure 6: Jerk response performances

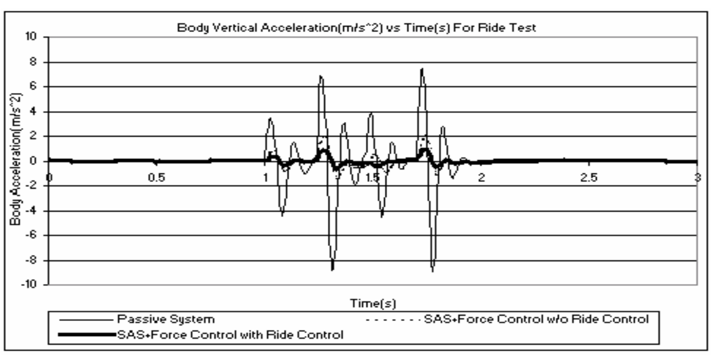

Figure 7: Body vertical acceleration performances

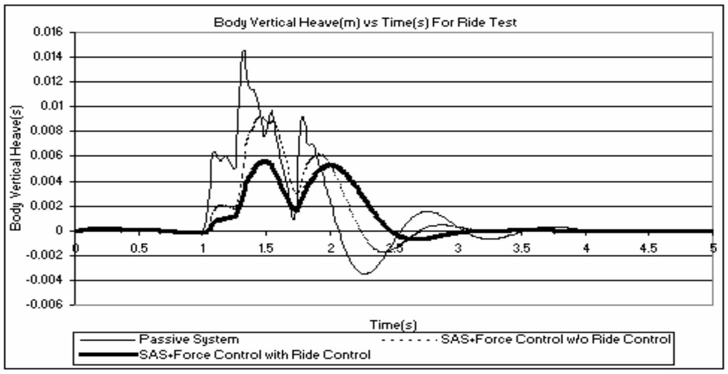

Figure 8: Body vertical heave performances

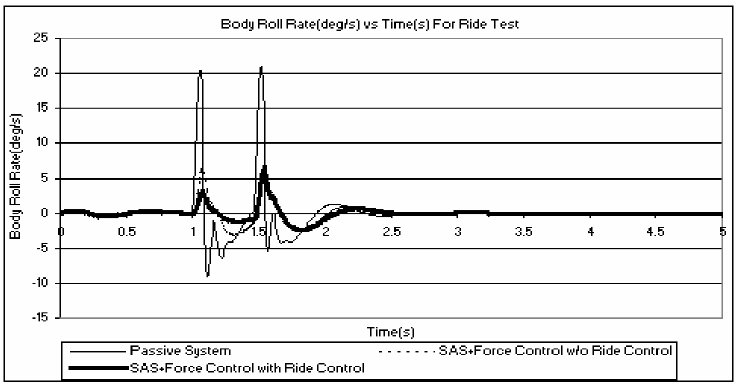

Figure 9: Body roll rate performances

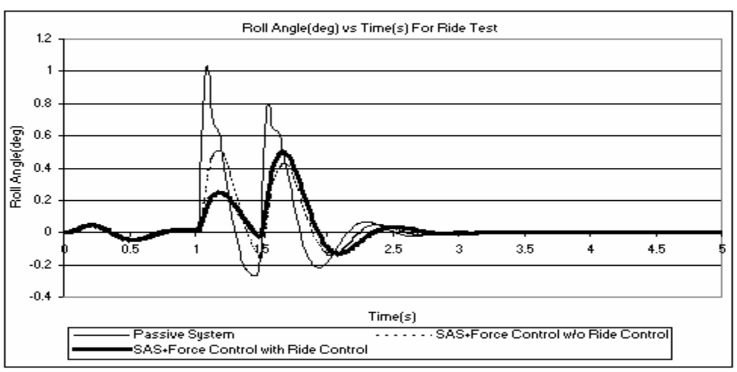

Figure 10: Roll angle performances

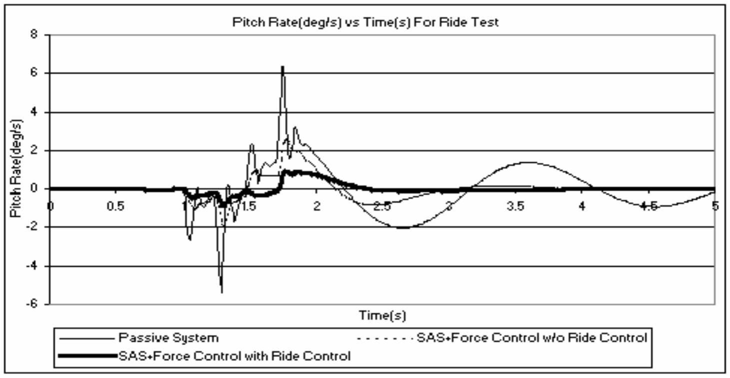

Figure 11: Pitch rate performances

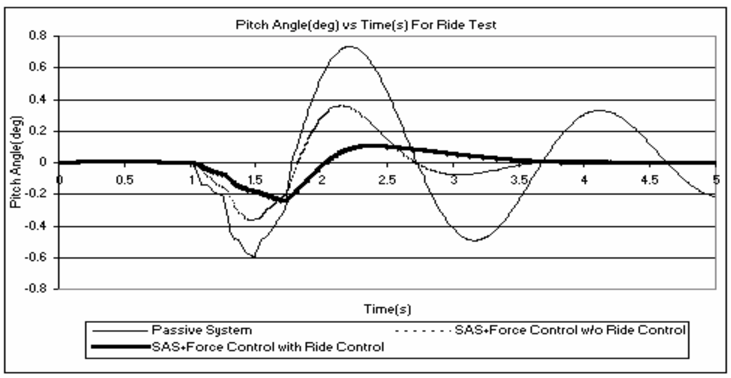

Figure 12: Pitch angle performances

5.0 CONCLUSIONS

A 7-DOF vehicle model has been developed and derived. A control structure namely stability augmentation system has been implemented in semi-active suspension system controller for the vehicle. The stability augmentation system has two controller loops of outer loop for attitude control and inner loop for ride control. Simulation studies have been performed for combination of roll and pitch mode bump test at 50 km/h. Performance of the proposed stability augmentation system with ride controller loop has been compared with passive system along with stability augmentation system without ride controller loop.

From the simulation results, it can be seen that semi-active suspension control using stability augmentation system with and without ride controller are able to reduce both the amplitude and the settling time of unwanted body motions in the forms of jerk, body displacement, body acceleration, roll angle, roll rate, pitch angle and pitch rate as compared with the passive system. The additional ride controller loop improved further the performance of stability augmentation system.

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