RESEARCH REVIEW JOURNALS
Volume-02
Issue-08
August-2017

ISSN: 2455-3085 (Online)
RESEARCH REVIEW International Journal of Multidisciplinary
www.rrjournals.com

Evaluation and Simulation of Semi-Active Suspension System by Modified Skyhook Control Theory Using Half Car Model using MATLAB/SIMULINK

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ARTICLE DETAILS

Article History
Received: 22 July 2017
Accepted: 09 August 2017
Published Online: 28 August 2017

Keywords
Modified skyhook control, mathematical modelling, half car model, simulation-evolution, semi-active suspension system

ABSTRACT

In present work modified control theory is developed for semi-active control of suspension system for MU (Multi Utility) cars. A mathematical model for half car model is developed for passive suspension system. Modified skyhook control theory for control of damping coefficient for variable damper is included. Half car model is developed on MATLAB/SIMULINK for results comparison for passive and semi-active controlled suspensions system. Responses of passive suspension system and semi-active suspension system for defined road test profiles are compared from which it is concluded that semi-active suspension system give best result. It can be test for different type of road profile also. Maximum acceleration goes in passive suspension system is higher than in semi-active suspension system which may comes into range of shock tolerance of human body for comfort.

INTRODUCTION

In automobiles main functions of suspension system are to carry the vehicle and its weight, maintain correct wheel alignment, control the vehicle's direction of travel, keep the tyres in contact with the road for safety and reduce the effect of shock forces to isolate the body from long and short wave vibrations for improve riding safety and comfort. The suspension system has to maintain the vehicle body at a uniform height above the axles over the long-wave unevenness in the road surface in the longitudinal direction without causing bouncing or pitching. The body should not rollout even though unevenness in the transverse direction to the road surface. Likewise, forces acting on the body as a result of driving maneuvers such as cornering, starting-off and braking, should not influence in the longitudinal direction, should be followed by maintaining the body in an exactly parallel position, acceleration forces in unevenness in the transverse direction should not exert any influences.

Hard suspension improves ride safety however it loses ride comfort and quality, while to improve ride quality and comfort it required soft suspension which causes losing in ride safety. Suspension system with fixed characteristics (damping and stiffness) can be optimally designed only for a particular road excitation, which is the result of the condition of the road surface and the speed driven. If the road surface or the speed driven differs, either comfort is diminished to an unnecessary extent or possible safety is not achieved or both. Semi-active suspension system is optimal solution for this (Gillespie, T.D., 2002). This paper thrashes out about semi-active suspension system. It is necessary to categorize suspension system according to the existence of control input. Suspension system can be categorized according to the existence of control input as follow (Chen, H., Liu, Z.Y., and Sun, P.Y., 2005),

- Conventional or passive suspension system
- Active suspension system
    - Fully active suspension system
    - Partially active suspension system with respect to load
    - Partially active suspension system with respect to frequency
    - Partially active pneumatic or hydro pneumatic suspension system
    - Semi-active suspension system

The semi-active suspension system uses a varying damping force as a control force. For example, in a hydraulic semi-active damper, varies the size of an orifice in the hydraulic flow valve to generate desired damping forces. Likewise in electro-rheological (ER) damper or a magneto-rheological (MR) damper can vary damping force by varies electric field or magnetic field respectively to cause various viscosities of the ER or MR fluids at various field intensities.

On the other hand, the fully active suspension system produces the control force with a separate hydraulic or pneumatic unit. Therefore, the cost and the weight of a fully active suspension system are much higher than those of a semi-active one. Semi-active suspension systems are getting more attention because of their low cost and competitive performance to the fully active ones (Keum-Shik Hong, Hyun-Chul Sohn, J. Karl Hedric., 2002)

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In present work semi-active control system is investigated using half car model with four degree of free of freedom. Considerations made in present work are as follow.

Four degree of freedom is considered which include bouncing motion of both unsprung masses and bouncing and pitching motion of chassis. Rolling motion and yawing motion of chassis and any lateral motion of vehicle is not considered. Both front and rear suspensions have only vertical linear motion. Spring using in front and rear suspensions have linear characteristic. Tyre is not loosing contact with road. Only road inputs are considered. Unbalance forces of rotating parts and aerodynamic effects are not considered. Vehicle velocity is taken as constant and test roads have no any gradient.

HALF CAR MODELLING OF MULTI UTILITY CAR.

A four degree of freedom (DOF) mathematical model of a half car is developed. The coordinate system was defined as follows:

- Positive x-axis pointing to the front of the vehicle.
- Positive y-axis pointing to the driver's left.
- Positive z-axis pointing up.
- Positive angles are defined by the right-hand rule.

Dynamic model (see figure 1) of a half car represents the whole vehicle. The model has spring mass and damper system. The model has

- Pitch and bounce motion for the chassis mass.
- Both front and rear axles have bouncing motion.
- Both tyres and suspensions are represents with spring and damping system.
- Chassis, front axle and rear axle represents with each single lump masses.
- Pitching motion is considered at the centre of gravity (C.G.) of the chassis mass.

Free body diagram shows the different kind of forces acting on the body mass, front axle mass and rear axle mass. Bouncing motion of all masses are considered as in 'Z_B' direction. Pitching motion is considered about 'Y_B' direction. In this model upward forces are considered as positive and clockwise angle is positive.

Z_{r1} and Z_{r2} are the road inputs at front and rear wheel respectively. Z_{r2} is delay by a_1+a_2 distance from Z_{r1}. Z_s, Z_{u1} and Z_{u2} are the vehicle sprung, front unsprung and rear unsprung masses displacements in 'Z' direction. Here θ is pitching angle. Pitching happens because of the unevenness of the Z_{r1} and Z_{r2}. In above free body diagram, all masses, stiffness, damping and dimensional values are known (Dabhi S.K., 2008) which are shown as below. Values of masses are taken half of actual values because of half car model is considered.

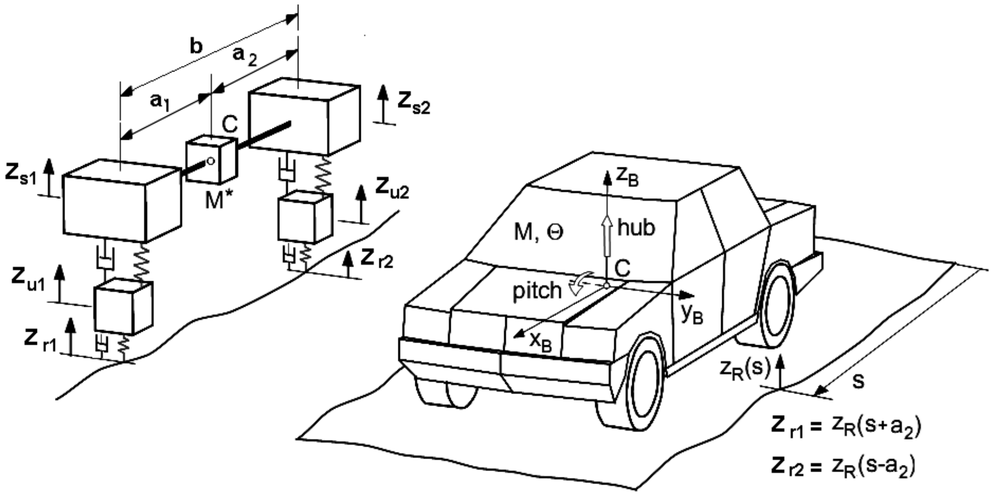


Figure 1: A dynamic half car model of multi utility car (Dabhi S.K., 2008)

Parameters (data):

m_{u1}	Front Unsprung mass, kg	40
m_{u2}	Rear Unsprung mass, kg	60
I	Moment of inertia of chassis,kg.m^2	3375
k_{s1}	Front suspension stiffness, N/m	22510
C_{s1}	Front suspension damping coefficient, Ns/m	2225
k_{t1}	Front wheel stiffness, N/m	134000
C_{t1}	Front wheel damping coefficient, Ns/m	700
M_s	sprung mass, kg	1100
a_1	Centre line distance between front wheel and C.G., m	1.375
a_2	Centre line distance between rear wheel and C.G., m	1.3
k_{s2}	Rear suspension stiffness, N/m	22740
C_{s2}	Rear suspension damping coefficient, Ns/m	2290
K_{t2}	Rear wheel stiffness, N/m	134000
C_{t2}	Rear wheel damping coefficient, Ns/m	700

Mathematical model has been developed from the figure 2 as follows.
For front unsprung mass,

$$-m_{u1}\ddot{Z}_{u1}-k_{t1}(Z_{u1}-Z_{r1})-C_{t1}(\dot{Z}_{u1}-\dot{Z}_{r1})+k_{s1}(Z_s-Z_{u1}+a_1\theta)+C_{s1}(\dot{Z}_s-\dot{Z}_{u1}+a_1\dot{\theta})=0 \quad (1)$$

For rear unsprung mass,

$$-m_{u2}\ddot{Z}_{u2}-k_{t2}(Z_{u2}-Z_{r2})-C_{t2}(\dot{Z}_{u2}-\dot{Z}_{r2})+k_{s2}(Z_s-Z_{u2}-a_2\theta)+C_{s2}(\dot{Z}_s-\dot{Z}_{u2}-a_2\dot{\theta})=0 \quad (2)$$

For sprung mass considering bouncing effect,

$$-\left[M_s\ddot{Z}_s+k_{s1}(Z_s-Z_{u1}+a_1\theta)+C_{s1}(\dot{Z}_s-\dot{Z}_{u1}+a_1\dot{\theta})+k_{s2}(Z_s-Z_{u2}-a_2\theta)+C_{s2}(\dot{Z}_s-\dot{Z}_{u2}-a_2\dot{\theta})\right]=0 \quad (3)$$

For sprung mass considering pitching effect,

$$I\ddot{\theta}+a_1\left[k_{s1}(Z_s-Z_{u1}+a_1\theta)+C_{s1}(\dot{Z}_s-\dot{Z}_{u1}+a_1\dot{\theta})\right]-a_2\left[k_{s2}(Z_s-Z_{u2}-a_2\theta)+C_{s2}(\dot{Z}_s-\dot{Z}_{u2}-a_2\dot{\theta})\right]=0 \quad (4)$$

By converting above equations into matrices form,

$$\begin{bmatrix} m_{u1} & 0 & 0 & 0 \\ 0 & m_{u2} & 0 & 0 \\ 0 & 0 & M_s & 0 \\ 0 & 0 & 0 & I \end{bmatrix}\begin{Bmatrix} \ddot{Z}_{u1} \\ \ddot{Z}_{u2} \\ \ddot{Z}_s \\ \ddot{\theta} \end{Bmatrix}+\begin{bmatrix} k_{t1}+k_{s1} & 0 & -k_{s1} & -a_1k_{s1} \\ 0 & k_{t2}+k_{s2} & -k_{s2} & a_2k_{s2} \\ -k_{s1} & -k_{s2} & k_{s2}+k_{s1} & a_1k_{s1}-a_2k_{s2} \\ -k_{s1} & k_{s2} & k_{s1}-k_{s2} & a_1k_{s1}+a_2k_{s2} \end{bmatrix}\begin{Bmatrix} Z_{u1} \\ Z_{u2} \\ Z_s \\ \theta \end{Bmatrix} \quad (5)$$

$$+\begin{bmatrix} C_{t1}+C_{s1} & 0 & -C_{s1} & -a_1C_{s1} \\ 0 & C_{t2}+C_{s2} & -C_{s2} & a_2C_{s2} \\ -C_{s1} & -C_{s2} & C_{s2}+C_{s1} & a_1C_{s1}-a_2C_{s2} \\ -C_{s1} & C_{s2} & C_{s1}-C_{s2} & a_1C_{s1}+a_2C_{s2} \end{bmatrix}\begin{Bmatrix} \dot{Z}_{u1} \\ \dot{Z}_{u2} \\ \dot{Z}_s \\ \theta \end{Bmatrix}=\begin{Bmatrix} k_{t1}Z_{r1}+C_{t1}\dot{Z}_{r1} \\ k_{t2}Z_{r2}+C_{t2}\dot{Z}_{r2} \\ 0 \\ 0 \end{Bmatrix}$$

Equation (5) can be written as follows,

$$[M]\{\ddot{U}\}+[C]\{\dot{U}\}+[k]\{U\}=\{F\} \quad (6)$$

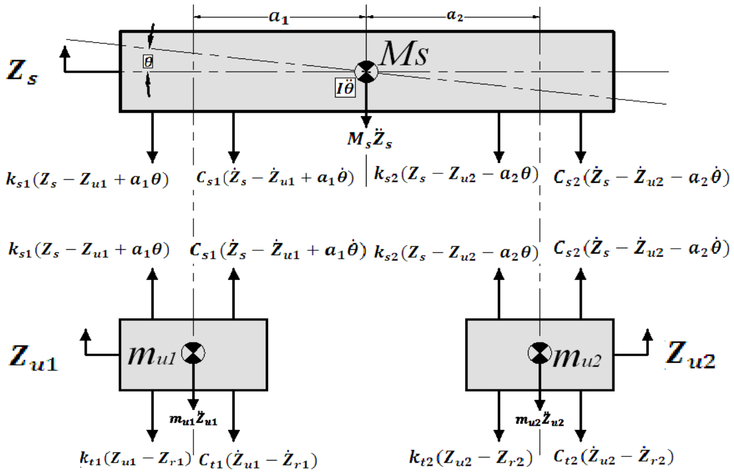


Figure 2: Free body diagrams of chassis, front suspension and rear suspension for half car model

Equation (6) (Patel C.B., 2009) is representing the half car mathematical model as shown in figure 2 where, M is mass matrix, C is damping matrix, k is stiffness matrix, $\ddot{U}$ is acceleration matrix, $\dot{U}$ is velocity matrix, U is displacement matrix and F is force matrix. Above state space equations of half car model are validated (Patel C.B., 2009).

MODIFIED SKYHOOK THEORY FOR SEMI-ACTIVE SUSPENSION SYSTEM.

Control system for semi-active suspension system is based on modified skyhook control theory. As per this theory, demanded suspension damping coefficient C_v* can be mention by equation (7) (Dabhi S. K., 2008).

$$C_v{*} = \frac{C_{hard}\left[\alpha\left(\dot{Z}_s - \dot{Z}_u\right) + (1-\alpha)\dot{Z}_s\right]}{\left(\dot{Z}_s - \dot{Z}_u\right)} \qquad (7)$$

Where, C_{hard} is upper limit of damping coefficient and α is coefficient of modified skyhook damper and its value can take from 0 to 1. If α is 0 then suspension system will work as ideal skyhook suspension system and if α is 1 then suspension system will work as purely passive suspension system. The optimum value of α is 0.3 (Kitching, K.J., Cebon, D. and Cole, D.J., 2000).

Semi-active controlled suspension system damping coefficient C_v is controlled by equation (8) (Dabhi S. K., 2008).

$$C_v = \begin{cases} C_{soft} & ; \quad if, (\dot{Z}_s - \dot{Z}_u).\dot{Z}_s \le 0, or \quad C_v{*} < C_{soft} \\ \left[\dfrac{C_{hard}\left[\alpha\left(\dot{Z}_s - \dot{Z}_u\right) + (1-\alpha)\dot{Z}_s\right]}{\left(\dot{Z}_s - \dot{Z}_u\right)}\right] & ; \quad if, (\dot{Z}_s - \dot{Z}_u).\dot{Z}_s > 0, \quad C_{soft} \le C_v{*} < C_{hard} \\ C_{hard} & ; \quad if, (\dot{Z}_s - \dot{Z}_u).\dot{Z}_s > 0, \quad C_v{*} \ge C_{hard} \end{cases} \qquad (8)$$

Where, C_{soft} is lower limit of damping coefficient.

In present work lower *(C_{soft})* and upper *(C_{hard})* limit of front and rear damper's damping coefficients are set at 0.2 and 0.5 damping ration ξ respectively.

SIMULATION RESULTS COMPARISON OF PASSIVE AND SEMI-ACTIVE SUSPENSION BY USING MATLAB/SIMULINK FOR EVALUATION OF COMFORT.

Half car model is developed in MATLAB/SIMULINK based on mathematical model given in this work is shown in figure 3. Sub systems of front and rear suspension systems are shown in figure 4. Figure 4(a) shows passive suspension system and figure 4(b) shows semi-active suspension system.

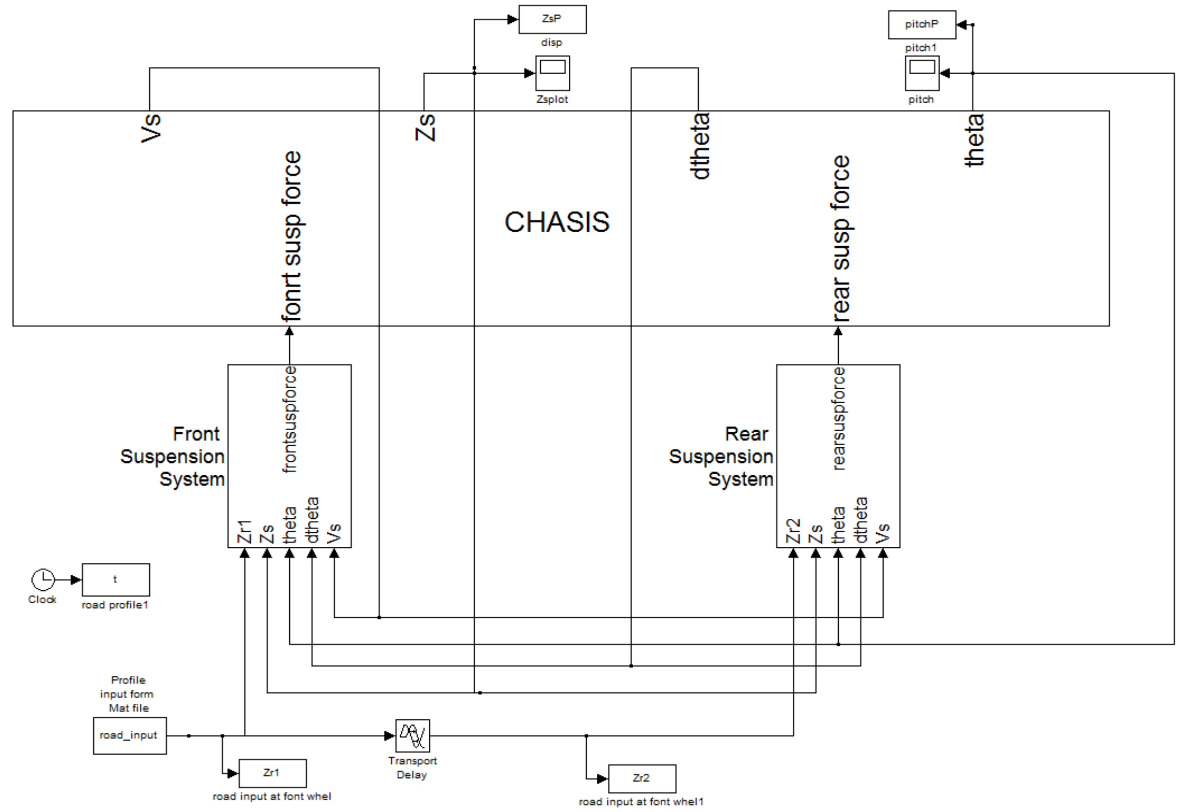
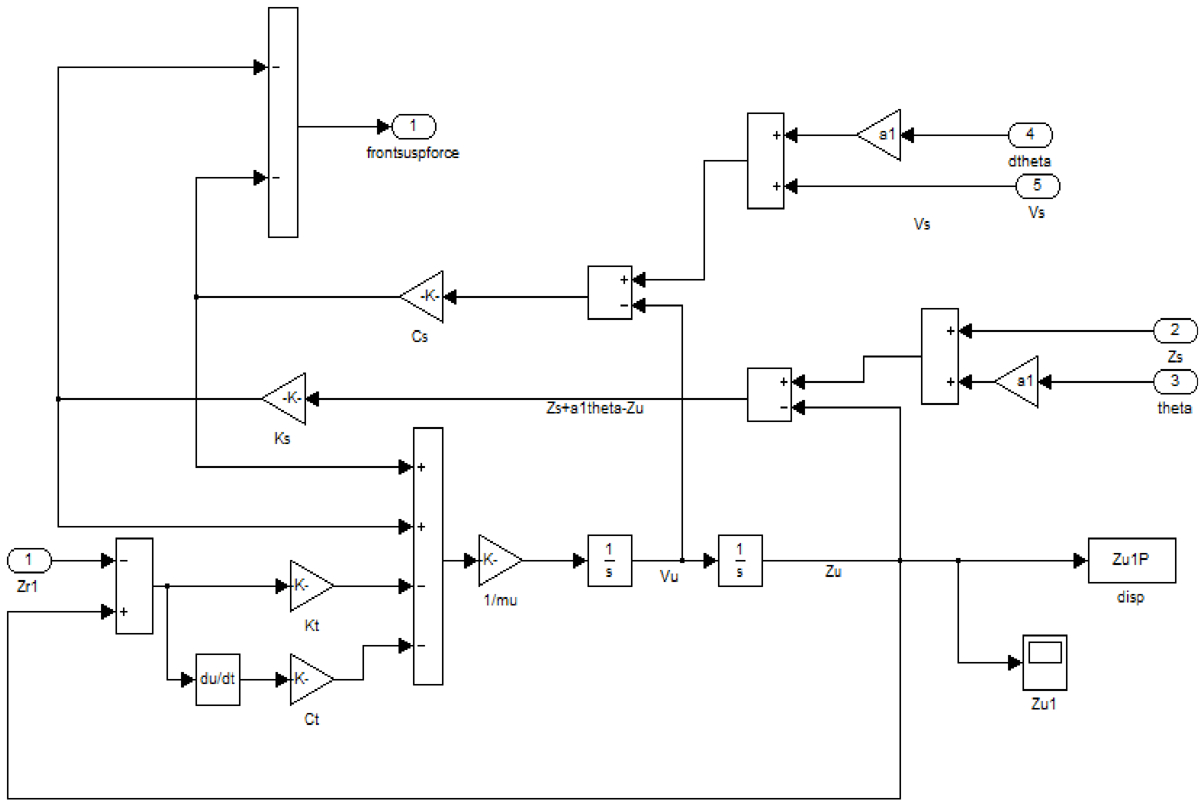
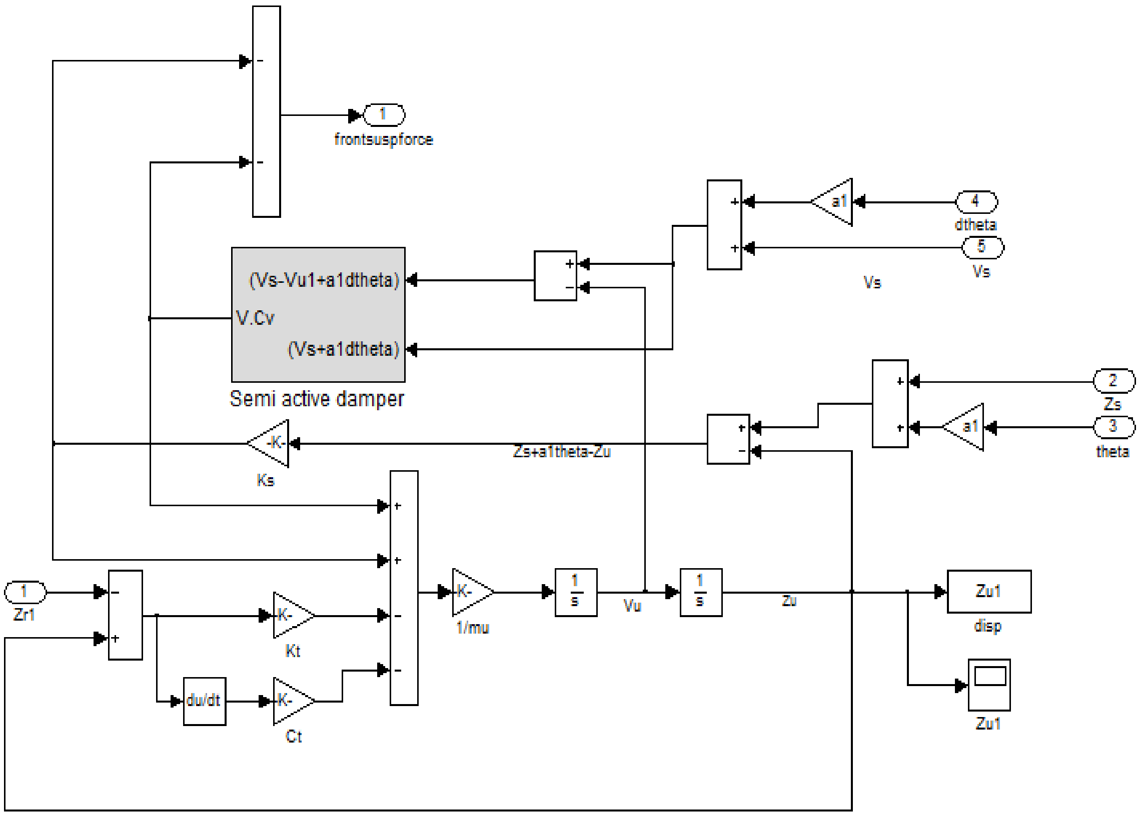


Figure 3: Half car model of suspension system in MATLAB/SIMULINK



(a) Passive suspension system.



(b) Semi-active suspension system.

Figure 4: Subsystem of front and rear suspension system,

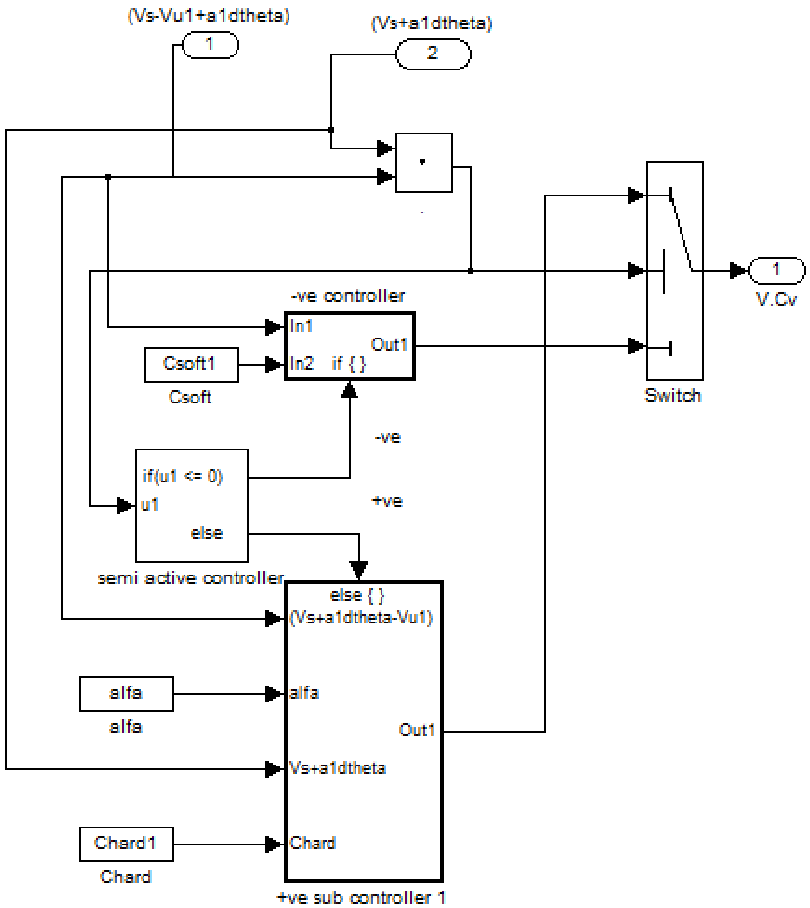
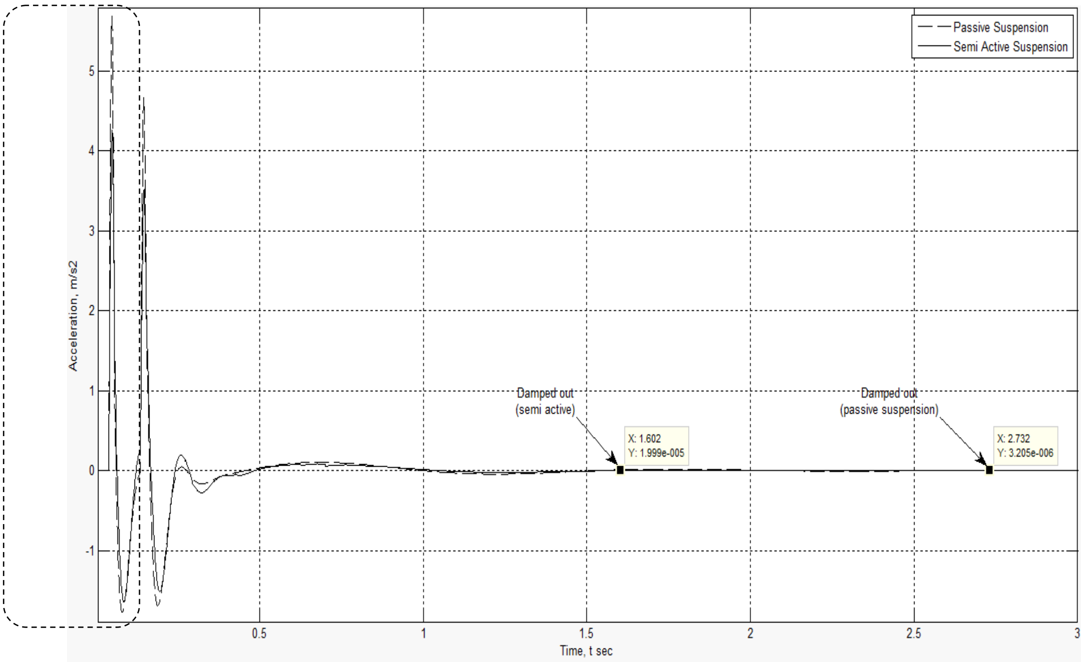


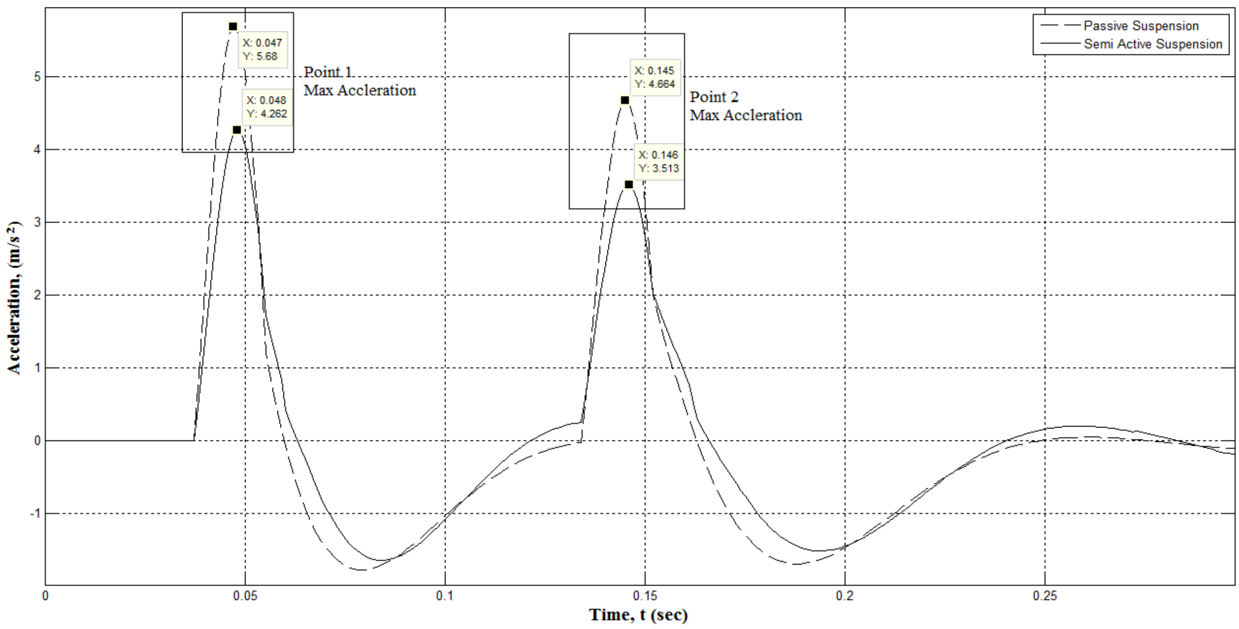
Figure 5: Subsystem of semi-active damper.

Controller for variable damper is developed which work on modified skyhook control theory discussed in section 3 which is included in semi-active damper in figure 4(b). MATLAB/SIMULINK model of semi-active suspension system is shown in figure 5.

Test road profiles are half sine bump, cosine bump, cycloid bump (Gawade, T.R., Dr. Mukherjee, S. and Prof. Mohan, D., 2004) and step of 0.1 m height. Test speeds are 30 km/h, 60 km/h and 100 km/h are used for comparison of passive and semi-active suspension system comfort levels. All taken road profiles are developed in MATLAB for testing purpose, has a bump profile start from 1 m distance, having height 0.1 m and length 0.5 m and rectangular cleat start from 1 m distance, having height 0.1 m and length 0.5 m. Sampling time is set at 0.001 s. Responses of designed systems are taken from centre of gravity (C.G.) of chassis in form of acceleration.



(a) Vertical acceleration between passive and semi-active suspension system for given model with 100 km/h vehicle speed on half sine bump profile.



(b) Enlarged View

Figure 6 Comparison of Vertical acceleration between passive and semi-active suspension system

Comparison of simulated result of passive and semi-active suspension system for 100 km/h vehicle speed with half sine bump profile is shown in figure 6.

Table 1 Result compression of passive and semi-active suspension system

Test Condition	Type of Suspension System	Point 1		Point 2		Time for damped out (sec)
		Acceleration (m/s^2)	Time (sec)	Acceleration (m/s^2)	Time (sec)	
100 km/h						
sine bump	Passive	5.680	0.047	4.664	0.145	2.732
	Semi-active	4.262	0.048	3.513	0.146	1.602
cosine bump	Passive	5.416	0.047	4.272	0.144	2.612
	Semi-active	3.916	0.047	3.098	0.144	1.062
cycloid bump	Passive	5.524	0.047	4.415	0.145	2.616
	Semi-active	4.044	0.048	3.229	0.145	1.602
Rectangle Cleat	Passive	6.270	0.194	5.958	0.296	4.965
	Semi-active	5.245	0.198	4.92	0.296	2.890
60 km/h						
sine bump	Passive	6.026	0.077	5.305	0.402	5.518
	Semi-active	4.817	0.079	3.965	0.404	3.516
cosine bump	Passive	5.856	0.077	4.938	0.401	4.982
	Semi-active	4.526	0.078	3.581	0.401	3.516
cycloid bump	Passive	5.975	0.078	5.106	0.402	4.443
	Semi-active	4.671	0.079	3.715	0.403	2.940
Rectangle Cleat	Passive	6.270	0.314	5.913	0.643	5.758
	Semi-active	5.245	0.318	4.775	0.648	4.312
30 km/h						
sine bump	Passive	5.710	0.148	5.552	0.475	4.519
	Semi-active	4.845	0.151	4.456	0.478	3.019
cosine bump	Passive	5.876	0.151	5.524	0.477	5.595
	Semi-active	4.819	0.153	4.328	0.478	3.593
cycloid bump	Passive	5.994	0.150	5.671	0.477	5.057
	Semi-active	5.016	0.153	4.476	0.479	3.593
Rectangle cleat	Passive	6.270	0.614	5.834	0.943	6.740
	Semi-active	5.245	0.618	4.701	0.948	4.637

Simulation results comparisons of passive and semi-active suspension system are shown in table 1. In table 1, point 1 gives maximum acceleration of spring mass when front wheel passing over defined obstruction and point 2 gives maximum acceleration of spring mass when rear wheel passing over same defined obstruction.

CONCLUSION

Passive suspension has limitation that it has fix damping coefficient, so design of passive suspension can be optimized for particular road condition. To improve ride quality and vehicle handling performance over wide range of road condition, semi-active suspension systems were introduced based on different control theories. In present work semi-active suspension system is evaluate for half car model for multi utility car, based on modified skyhook theory. From table 1, it is concluded that semi-active suspension system gives better isolation from road unevenness then passive suspension, which shows that semi-active suspension system gives more comfort. From table 1, it is also concluded semi-active suspension system damped bump or cleat disturbance is earlier then passive suspension system, which shows that semi-active suspension system is more stable then passive suspension system.

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