Thiyagarajan et al. 2018. *Int. J. Vehicle Structures & Systems*, 10(3), 195-198
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.10.3.08

International Journal of
Vehicle Structures & Systems
Available online at www.maftree.org/eja

Development and Control of Active Suspension System with Energy Regeneration Implementation Scheme

J. Thiyagarajan[a,b], P. Sathishkumar[c], J. Arivarasan[a,d], S. Rajeshkumar[a,e] and T.S. Rajalakshmi[a,f]

[a]*Dept. of Mechatronics Engg., SRM Institute of Science and Technology, Kattankulathur, India*
[b]*Corresponding Author, Email: thiyagarajan.j@ktr.srmuniv.ac.in*
[c]*Automotive Engineering Research Institute, Jiangsu University, Zhenjiang, China*
Email: sathishkumar8989@gmail.com
[d]*Email: arivarasan.j@ktr.srmuniv.ac.in*
[e]*Email: rajeshkumar.s@ktr.srmuniv.ac.in*
[f]*Email: rajalakshmi.ts@ktr.srmuniv.ac.in*

ABSTRACT:

Active suspension systems have been used in the recent years as they provide better ride comfort, road handling and safety. The effect of vehicle vibration caused by road roughness is effectively reduced by active suspension system which plays an important role in improving the vehicle performance indices. The application of active suspension system is limited because it consumes high amount of energy. From the point of energy saving, a regenerative active suspension system is designed and its working principle with two modes switched in different conditions was implemented. In this implementation scheme, operating electric circuits are designed based on different working status of the actuator and power source. In the first stage an electromotor mode in which an active suspension system uses a linear electric actuator controlled by constrained PID controller. In the generator mode, under certain circumstances using linear motor as actuators enables to transform mechanical energy of the car vibrations to electrical energy and accumulated to charge the energy-storage capacitor and fed back into the power source when needed.

KEYWORDS:

Active suspension; Quarter car; Linear actuator; PID controller; Electromotor mode; Generator mode

CITATION:

J. Thiyagarajan, P. Sathishkumar, J. Arivarasan, S. Rajeshkumar and T.S. Rajalakshmi. 2018. Development and Control of Active Suspension System with Energy Regeneration Implementation Scheme, *Int. J. Vehicle Structures & Systems*, 10(3), 195-198. doi: 10.4273/ijvss.10.3.08.

1. Introduction

The main parts of the conventional (passive) suspension systems are the wheels with the tyres, the wheel carrier systems, damper and spring elements, the steering and the brakes [1]. The dynamic behaviour of passive automotive suspension systems is primarily determined by the choice of the spring (stiffness) and the damper (damping coefficient) besides different aspects being taken into account. On the one hand, the suspension should provide excellent ride comfort by a soft spring and damper setup which isolates the chassis from the road induced vibrations. On the other hand, the vehicle should be controllable by the driver to ensure ride safety, which requires a stiff, well damped combination between the vehicle and the road, specifically for non-stationary driving manoeuvres, e.g. driving at a rough road or cornering. Consequently, the requirements regarding comfort and safety are conflicting [2-5]. In the last few decades, many researches have been carried out to improve vehicle suspensions. Among the suggested solutions, active suspension is a potential way to improve suspension performance although the passive suspension system can effectively handle some control of suspension system [6-7]. The types of actuators used in AVSS include hydraulic, pneumatic, electric and electromagnetic actuators. The use of electromagnetic actuators has good prospect for FASS applications but the technology is relatively new and still evolving. Rotational electromagnetic actuators require gears or ball screw designs to convert the rotary motion and this adds to the complication of the system. The biggest set-backs for electromagnetic actuators include its cost, the increase in moving parts as well as, additional masses. The system also requires between 12-24V to provide continuous excitation force, whereas electrohydraulic actuators require about 10V.

The benefits of the electromagnetic actuator include increased efficiency, mechanically stiffer, improved stability and dynamic behaviour, and more accurate force control [8-9]. AlešKruczek et al [10] designed the electric linear motor as suspension system actuator for active suspension. They used the H-Infinity control scheme for full car, half as well as quarter models. Focus was on comparison of different controllers designed for quarter, half and full car models. Hyniova [11] with co-researchers [12-13] designed a unique advanced suspension system that used a linear electric motor as a suspension system actuator and provided desired forces between unsprung mass and car body. Hyniova [11] conducted several experiments on energy management in the system. In order to validate various control strategies and to test different ways of energy consumption optimization, he designed and constructed a unique one quarter car test stand. His work also dealt with the way and results of experimental validation of vehicle active suspension system behaviour when robust control was applied and with energy management approach that was used in the system. The main drawback of active suspension system is power demand is more than the semi-active and other controllable suspension system. So this paper describes energy recuperation from suspension system and controlling of the same by proposing an electric motor cum generator.

2. Linear motor

Fig. 1 shows the basic design of linear motor. The mean feature of linear motors is that it directly translates electrical energy into linear mechanical motion and force, and vice versa. Linear electric motor can transform electrical energy to linear motion of the rotor. In fact, linear motor works on the same basis as electric motor, the only difference is that linear motor has stretched out windings to longitudinal direction. It can be imagined as common rotary motor with diameter equal to infinity [12]. The key benefit of the suggested solution using a linear actuator is the capability to generate desired forces acting between the sprung & unspring masses of the vehicle, providing better insulation of the car sprung mass from the uneven road surface disturbances. In addition, under certain situations using linear motors as actuators will lead to recover energy by converting heave motions of the vehicle, accumulate it, and use it when needed.

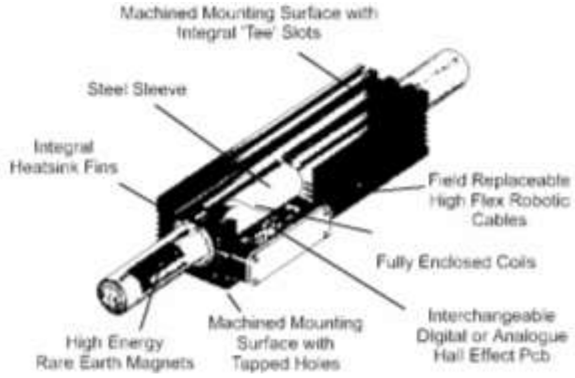

Fig. 1: Basic design of linear motor [12]

The digital signal from the deflection sensor and impulse signal from motor is imported into the microprocessor. Then they are handled with control algorithm and active suspension law. Then it is exported to drive and storage unit to control the motor actuator's mode. In this work electrical motor and generator modes are possible. When the car is travelling on the rough surface the enough amount of energy stored in the drive and storage unit from the movements of suspension. Then the digital signal from the deflection sensor will transmit signal to the controller and from the control algorithm actuator switching takes place. In this mode it reduces the movement of suspension and provides comfort to the passenger's. For this mode power supply for the motor is given by drive and storage unit.

When the car is travelling in the smooth/even surface there will be some vibrations in the suspension that can be used for recovery of energy and stored in the storage unit. Fig. 2 shows that block diagram of the linear actuator/motor controlled active suspension. In the case of active suspension the passivity limitation is fully overcome and energy may be introduced into the system. The control input is the suspension force F delivered by the force actuator in addition to the passive devices of the suspension. The difference between slow and fully-active suspensions is in terms of bandwidth. The fully active suspension actuator is capable of reacting in milliseconds (30-40Hz).

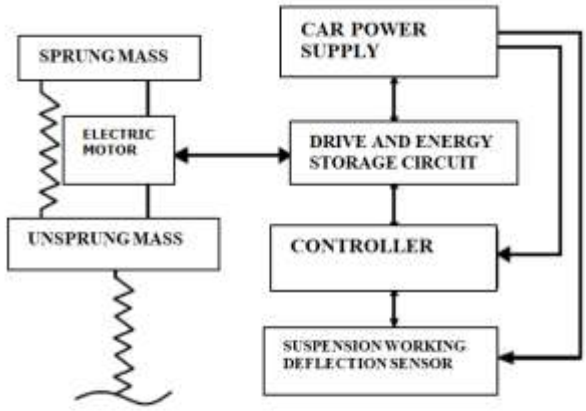

Fig. 2: Block diagram of linear electric actuator in active suspension system

3. Mathematical modelling of suspension system

The equations of motion for the sprung and unspring masses of the passive quarter car model are given by,

$$M_1\ddot{Z}_1 + C_1(\dot{Z}_1 - \dot{Z}_2) + K_1(Z_1 - Z_2) = 0 \quad (1)$$

$$M\_2\, \ddot{Z}\_2 + C\_1\, (\dot{Z}\_2 - \dot{Z}\_1\,) + K\_1\, (Z\_2 - Z\_1\,) + K\_2\, (Z\_2 - Z\_r\,) = 0 \quad (2)$$

$$\ddot{Z}_S = \frac{1}{M_S}\left[C_S\left(\dot{Z}_{uS} - \dot{Z}_S\right) + K_S(Z_{uS} - Z_S)\right] \quad (3)$$

$$\ddot{Z}_{uS} = \frac{1}{M_{uS}}\left[C_S\left(\dot{Z}_S - \dot{Z}_{uS}\right) + K_S(Z_S - Z_{uS}) + K_{uS}(Z_r - Z_{uS})\right] \quad (4)$$

Let us assume the state variables are as follows,

$$Z_1 = Z_S - Z_{uS}, Z_2 = \dot{Z}_S, Z_3 = Z_{uS} - Z_r, Z_4 = \dot{Z}_{uS} \quad (5)$$

$$\dot{Z}_1 = \dot{Z}_S - \dot{Z}_{uS} \approx Z_S - Z_{uS}, \dot{Z}_2 = \ddot{Z}_S\,, \dot{Z}_3 = \dot{Z}_{uS} - \dot{Z}_r$$
$$\dot{Z}_4 = \ddot{Z}_{uS} \quad (6)$$

General form of state space equation is given by,

$$\dot{Z} = AZ + BF_a + \dot{Z}_r \quad (7)$$

Quarter car suspension system state space equation is

$$\begin{bmatrix} \dot{Z}_1 \\ \dot{Z}_2 \\ \dot{Z}_3 \\ \dot{Z}_4 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & -1 \\ \frac{-K_S}{M_{uS}} & \frac{-C_S}{M_{uS}} & 0 & \frac{C_S}{M_{uS}} \\ 0 & 0 & 0 & 1 \\ \frac{-K_S}{M_S} & \frac{C_S}{M_S} & \frac{-K_{uS}}{M_S} & \frac{-C_S}{M_S} \end{bmatrix} \begin{bmatrix} Z_1 \\ Z_2 \\ Z_3 \\ Z_4 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ -1 \\ 0 \end{bmatrix} \dot{Z}_r \quad (8)$$

Fig. 3 shows conventional suspension system and it consists of coil or leaf spring with combination of hydraulic damper. The parameters spring constant and damping constant are fixed from the design stage itself, so cannot be controlled. Advantages of conventional suspension, is enough simple design, high reliability and no need of power supply [14-15 & 19]. The drawbacks of conventional suspension, is if it is high damping or high stiffness suspension it will transfer the vibration of uneven road surface [16-17].

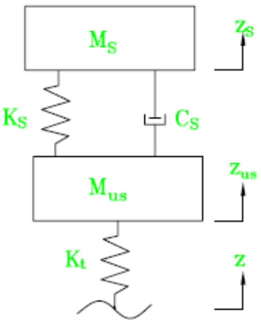

Fig. 3: Quarter vehicle passive suspension 2DOF model

4. Controller design

The error signal e(t) is used to make the proportional, integral and derivative (PID) control actions. The resulting signals weighted and added to form the control signal u(t) applied to plant model using,

$$u(t) = K_p\left[e(t) + \frac{1}{T_i}\int_0^t e(t)dt + T_d\,\frac{de(t)}{dt}\right] \qquad (9)$$

Where u(t) is input signal of the plant model. e(t) - error signal = r(t)-y(t), and r(t) is the reference input signal. Overshoot is when a signal or function exceeds its target. Settling time is the time elapsed from the application of an ideal instantaneous step input to the time at which the output has entered and remained within a specified error band. And steady-state error is the difference between the desired final output and the actual one [18, 20].

5. Simulation

To verify the linear electric actuator controlled active suspension system, they are compared with passive suspension system. Passive system and active suspension system are all included in one analysis loop. The parameters of the quarter car model are listed in Table 1. According to ISO 2631 [20], the ride comfort is specified in terms of Root Mean Square (RMS) acceleration. The ISO road roughness input is presented in Table 2. The response of the system is observed on 10 seconds scale. To generate the road profile of a random base excitation for the 3-DOF Active suspension simulation disturbance, a spectrum of the geometrical road profile with road class roughness-D is considered. The vehicle is travelling with a constant speed v0, the time histories data of road irregularity are described by PSD method [17-19].

Table 1: Quarter car parameters used in the simulation

Parameter	Value
Sprung Mass (Ms)	250 kg;
Unsprung Mass (Mus)	50 kg
Suspension spring constant (Ks)	18,600 N/m
Suspension damping coefficient(Cs)	1000 Ns/m
Tire spring constant(Kt)	196,000 N/m

Table 2: Road roughness value classified by ISO

Classification S	Road roughness (m^2/(cycles/m)) (*10^{-6})	
(Ω)	Range	Average
A (very good)	2 to 8	4
B (good)	8 to 32	16
C (average)	32 to 128	64
D (poor)	128 to 512	256
E (very poor)	512 to 2048	1024

6. Results and discussion

Figs. 4-6 show that the simulation of sprung mass acceleration, suspension travel and unsprung mass displacement of passive and active suspension system respectively for ISO E-class road. Fig. 5 shows that the suspension travel of passive and active suspension, significant reduction in active suspension compared to passive suspension system. Sprung mass acceleration of active is 35.3% reduced than the conventional system as presented in Table 3. This is evident that the proposed system provides better comfort than passive system. There is no change unsprung mass displacement, therefore the proposed is not affecting the road holding capability.

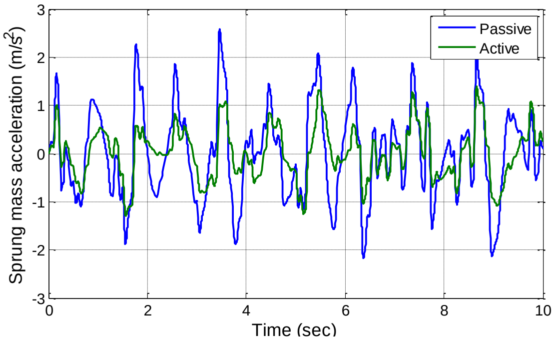

Fig. 4: Sprung mass acceleration

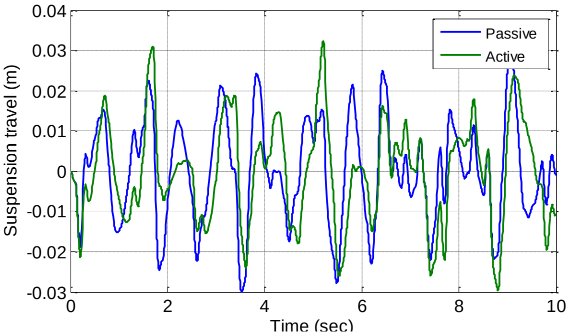

Fig. 5: Suspension travel

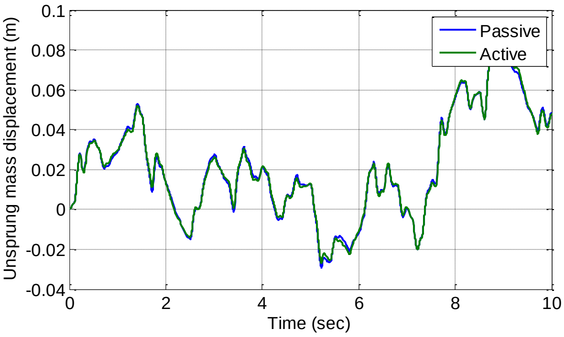

Fig. 6: Unsprung mass displacement

Table 3: RMS values of sprung mass acceleration, suspension travel and unsprung mass displacement for ISO E-class road

Parameters	Passive suspension	Active suspension	% of Improvement
Unsprung mass displacement (m)	0.0353	0.0352	-
Suspension travel (m)	0.0136	0.0098	27.9
Sprung mass acceleration (m/s^2)	1.0838	0.7012	35.3

7. Conclusion

The energy-regenerative active suspension with two modes is studied by simulation and test in this paper. In the electrical motor mode, the optimal ride comfort or the expected vehicle body attitude can be obtained by active control. While in the generator mode, the system can not only recycle the vibration energy excited by uneven road surfaces to some extent, but can also improve ride comfort performance meanwhile. The combination of these modes can be potentially valuable and switching strategy is particularly important.

REFERENCES:

[1] B. Gao, J. Darling, D.G. Tilley, R.A. Williams, A. Bean and J. Donahue. 2006. Control of a hydro-pneumatic active suspension based on a non-linear quarter-car model, *IMechE Int. J. Systems and Control Engg.*, 220(1), 15-31.

[2] J. Lin, R.J. Lian, C.N. Huang and W.T. Sie. 2009. Enhanced fuzzy sliding mode controller for active suspension systems, *Mechatronics*, 19, 1178-1190 https://doi.org/10.1016/j.mechatronics.2009.03.009.

[3] G. Koch, S. Spirk, E. Pellegrini, N. Pletschen and B. Lohmann. 2011. Experimental validation of a new adaptive control approach for a hybrid suspension system, *American Control Conf.*, 4580-4585. https://doi.org/10.1109/ACC.2011.5991450.

[4] D. Fischer and R. Isermann. 2004. Mechatronic semi-active and active vehicle suspensions, *Control Engg. Practice*, 12, 1353-1367.

[5] R. Rajamani. 2012. *Vehicle Dynamics and Control*, 2nd Edn. https://doi.org/10.1007/978-1-4614-1433-9.

[6] I. Maciejewski. 2012. Control system design of active seat suspensions, *J Sound Vib.*, 331, 1291-1309. https://doi.org/10.1016/j.jsv.2011.11.010.

[7] S.K. Sharma and A. Kumar. 2017. Ride performance of a high speed rail vehicle using controlled semi active suspension system, *Smart Materials and Structures*, 26(5), 55026. https://doi.org/10.1088/1361-665X/aa68f7.

[8] B.T. Fijalkowski. 2011. Automotive mechatronics: operational and practical in intelligent systems, control and automation: *Sci. and Engg.*, 52(2).

[9] S. Lee and W.J. Kim. 2010. Active suspension control with direct-drive tubular linear brushless permanent magnet motor, *IEEE Transactions on Control Systems Tech.*, 18(4), 859-870. https://doi.org/10.1109/TCST.2009.2030413.

[10] A. Kruczek, A. Stříbrský, J. Honců and M. Hlinovský. 2009. Controller choice for car active suspension, *Int. J. Mechanics*, 3(4), 61-68.

[11] Hyniova. K 2014. On experimental verification of vehicle active suspension robust control, *Latest Trends on Systems*, 1, 353-358.

[12] A. Stribrsky and K. Hyniova. 2007. Energy recuperation in automotive active suspension systems with linear electric motor, *Proc. Mediterranean Conf. on Control and Automation*, 1-5.

[13] K. Hyniova, A. Stribrsky, J. Honcu and A. Kruczek. 2009. Active suspension system - energy control, *IFAC Proc.*, 42(19), 146-152.

[14] P. Sathishkumar, J. Jancirani and J.D. John. 2014. Reduction of axis acceleration of quarter car suspension using pneumatic actuator and active force control technique, *J. Vib., Engg.*, 16 (3).

[15] P. Sathishkumar, J. Jancirani and D. John. 2014. Reducing the seat vibration of vehicle by semi active force control technique, *J. Mech. Sci. and Tech*, 28 (2), 473-479.

[16] R.C. Sharma. 2016. Evaluation of passenger ride comfort of Indian rail and road vehicles with ISO 2631-1 standards: Part 1 - Mathematical modelling, *Int. J. Vehicle Structures and Systems*, 8(1), 1-6. https://doi.org/10.4273/ijvss.8.1.01.

[17] R.C. Sharma. 2016. Evaluation of passenger ride comfort of Indian rail and road vehicles with ISO 2631-1 standards: Part 2 - Simulation, *Int. J. Vehicle Structures and Systems*, 8(1), 7-10. https://doi.org/10.4273/ijvss.8.1.02.

[18] J. Jancirani, P. Senthilkumar, M. Eltantawie and D. John. 2015. Comparison of air spring actuator and electro-hydraulic actuator in suspension system, *Int. J. Vehicle Structures and Systems*, 7(1), 36-39. http://dx.doi.org/10.4273/ijvss.7.1.07.

[19] R.C. Sharma. 2017. Ride, eigenvalue and stability analysis of three-wheel vehicle using Lagrangian dynamics, *Int. J. Vehicle Noise and Vibration*, 13(1), 13-25. https://doi.org/10.1504/IJVNV.2017.086021.

[20] S.K. Mouleeswaran 2012. Design and development of PID controller-based active suspension system for automobiles, *PID Controller Design Approaches-Theory, Tuning and Application to Frontier Areas*, 71-98. https://doi.org/10.5772/32611.