Sathishkumar et al. 2015. *Int. J. Vehicle Structures & Systems*, 7(3), 123-127
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.7.3.07

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

Simulation of Electrical Actuator and Air Spring Actuator Controlled Suspension Systems for Automotive Vehicles

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ABSTRACT:

This article discusses methods to reduce the acceleration of a vehicle and increase its road holding ability. In the simulation using quarter car model, electric actuator and air spring based actuator are used as the main control elements. A three degrees of freedom system model is used in which the parameters for the tire, vehicle body and seat are considered. The required actuator force is calculated by a standard fuzzy controller. For analysing the performance of active suspension system, body acceleration and velocity are given as inputs to the controller according to ISO specified standards. Accelerations of the seat and vehicle body are used to judge the performance of the system.

KEYWORDS:

Quarter car model; Active suspension system; Electrical actuator; Air spring actuator; Fuzzy logic controller

CITATION:

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1. Introduction

The main purpose of any vehicle suspension system is to isolate the body from the disturbances caused by road unevenness and to maintain the continuous contact between road and the wheel. The suspension system is therefore responsible for the ride quality and vehicle stability [1]. A conventional suspension system has coil or leaf springs in combination with hydraulic or pneumatic dampers (shock absorbers) [2-4]. The design of a passive suspension system can only be a compromise between these conflicting demands. However, the improvement in vertical vehicle dynamics is possible by using a semi active damper [5, 6], an air spring actuator [7, 8] and electro-hydraulic actuator controlled suspension system [9, 10]. The active suspension system performance depends on the response time and force developed by the actuator.

Air spring actuators are well known for their low transmissibility coefficients and their ability to vary load capacities easily by changing only the gas pressure within the springs. The characteristics of air spring can be used for a mechatronic approach in the vehicle suspension design, which is the ability to provide a controlled variable force in terms of stiffness [11-13]. However, the response time and accurate control of an air spring is difficult. This may be overcome by using a more linear and quick responding actuators. Electric actuator characteristics are linear with respect to current, velocity and force [14]. Response time is also less due to the electrical power supply and natural characteristics of an electric actuator. Therefore both the actuators are able to develop a desired force between the sprung and unsprung masses.

In this paper, an active suspension system based on air spring is simulated using three degrees of freedom (DoF) quarter car model. The system model is controlled by an electric actuator using fuzzy logic. Body acceleration and velocity are given as inputs to the controller according to ISO specified standards whilst acceleration of the seat and vehicle body outputs are analysed to evaluate the performance of the proposed active suspension system.

2. Quarter car model and actuator forces

A quarter car model consists of a passenger seat, sprung mass, unsprung mass, air spring and a passive damper as shown in Fig. 1. The air spring is replaced later by the proposed electric actuator. Assumptions of the quarter car model used in this study are as follows:

- The tire is modelled as a linear spring without damping;
- There is no rotational motion in wheel and body;
- The behaviour of spring and damper are linear;
- The tire is always in contact with the road surface;
- The effect of friction is neglected so that the residual structural damping is not considered into vehicle modelling [15].

Equations of motion for active quarter car model are given by the following:

$$\mathrm{M_{ps}}\ddot{\mathrm{Z}}_{\mathrm{ps}} + \mathrm{C_{ps}}(\dot{\mathrm{Z}}_{\mathrm{ps}} - \dot{\mathrm{Z}}_{\mathrm{s}}) + \mathrm{K_{ps}}(\mathrm{Z_{ps}} - \mathrm{Z_s}) = 0 \quad (1)$$

$$\mathrm{M_s}\ddot{\mathrm{Z}}_{\mathrm{s}} + \mathrm{C_{ps}}(\dot{\mathrm{Z}}_{\mathrm{s}} - \dot{\mathrm{Z}}_{\mathrm{ps}}) + \mathrm{K_{ps}}(\mathrm{Z_s} - \mathrm{Z_{ps}}) + \mathrm{C_s}(\dot{\mathrm{Z}}_{\mathrm{s}} - \dot{\mathrm{Z}}_{\mathrm{us}}) + \mathrm{K_s}(\mathrm{Z_s} - \mathrm{Z_{us}}) + \mathrm{F_a} = 0 \quad (2)$$

$$\mathrm{M_{us}}\ddot{\mathrm{Z}}_{\mathrm{us}} + \mathrm{C_s}(\dot{\mathrm{Z}}_{\mathrm{us}} - \dot{\mathrm{Z}}_{\mathrm{s}}) + \mathrm{K_s}(\mathrm{Z_{us}} - \mathrm{Z_s}) + \mathrm{K_t}(\mathrm{Z_{us}} - \mathrm{Z_r}) - \mathrm{F_a} = 0 \quad (3)$$

Where M_{ps} is the passenger seat mass, M_s and M_{us} are sprung and unsprung mass, C_{ps} is passenger seat damping, C_s is suspension damping, K_{ps} is seat spring stiffness, K_{us} is stiffness of unsprung spring, Z_s and Z_{us} are the displacement of sprung and unsprung mass, Z_r is the road profile and F_a is the air spring force.

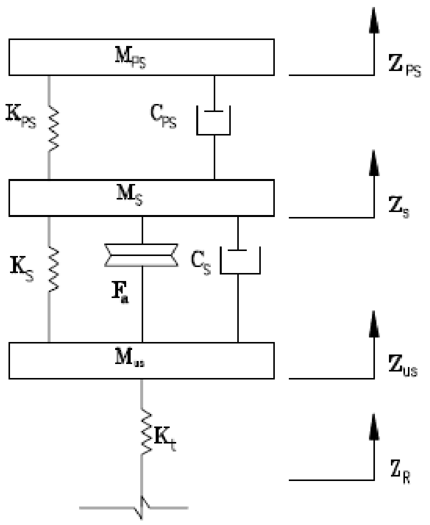

Fig. 1: Three DoF quarter car model

The air springs for passenger cars are commercially available but there is not enough research devoted to their dynamic characteristics [16]. Here we discuss the vehicle air suspensions from design aspects, but not from control viewpoint [17]. The gas in the air spring can be compressed to the pressure required to carry the load and therefore air spring does not require large static deflection. The compressibility of the gas thus provides the desired elasticity for the spring. The use of linear mathematical models allow us to obtain simple equations from which, if properly handled, more significant information can be extracted [18]. Fig. 2 shows the air spring actuator force diagram used in the system model. The electrical actuator consists of motor and gear. The motor gives a rotary motion which will be converted in to linear motion by the gear arrangement. The actuator characteristics of current vs. load [14] as shown in Fig. 3 is fed as an input for the system model.

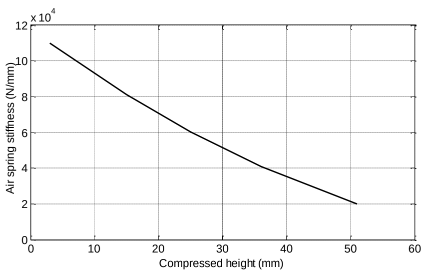

Fig. 2: Air spring actuator force diagram

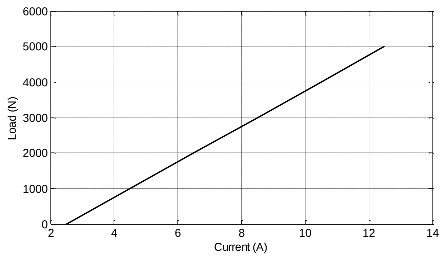

Fig. 3: Electric actuator force diagram

3. Controller design

Fuzzy logic control (FLC) is a practical alternative for a variety of challenging control applications. The FLC provides a human experience based representation to achieve high-performance control [19]. The FLC system consists of three stages namely, fuzzification, fuzzy inference machine and defuzzification. The fuzzification stage converts real-number or crisp input values into fuzzy values while the fuzzy inference machine processes the input data and computes the controller outputs in line with the rule-base and data base. These fuzzy outputs are again converted into real numbers by the defuzzification stage [20-23]. A possible choice of the membership functions for the three mentioned variables of the active suspension system represented by a fuzzy set are NL, NS, ZE, PS and PL. Output variable is the control force that must be tracked by the actuators. Figs. 4 and 5 show the inputs for the fuzzy controller. Fig. 6 shows the ouput of the fuzzy controller. The Mamdani-type fuzzy inference system is used. ‘If-Then’ rule-base is then applied to describe the experts' knowledge. The 2-in-1-out fuzzy rule-base cloud can drive the FLC inference mechanism. The 2-in-1-out FLC rule-base for the ride comfort of the two DoF active suspension system is given in Fig. 7 and Table 1.

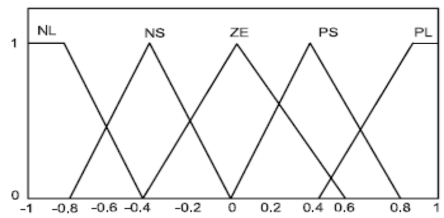

Fig. 4: Sprung mass velocity

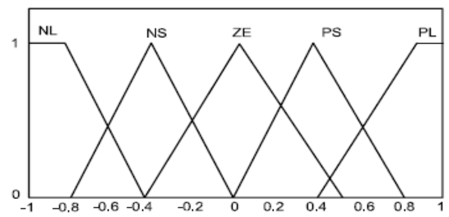

Fig. 5: Suspension deflection

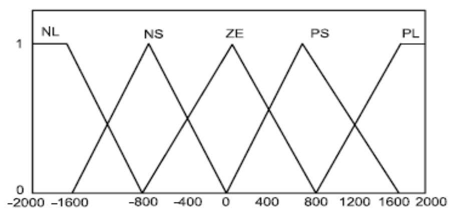

Fig. 6: Actuator force

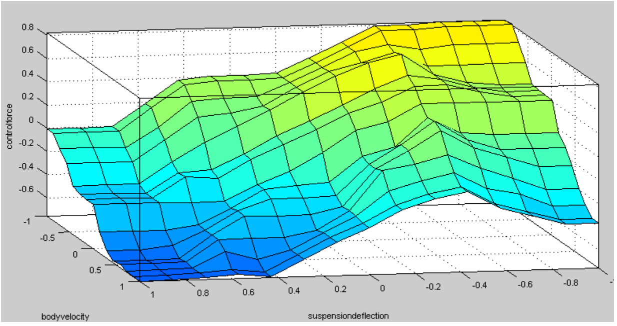

Fig. 7: FLC controller surface plot

Table 1: FLC rule-base

$\dot{z}_S$ / Z_S-Z_{us}	NL	NS	ZE	PS	PL
NL	PL	PS	PS	ZE	NS
NS	PL	PL	PS	PS	ZE
ZE	PS	PS	ZE	NS	NL
PS	PS	ZE	NS	NL	NL
PL	ZE	NS	NS	NL	NL

4. Road disturbances

To generate the road profile of a random base excitation for the 3-DoF active suspension simulation model, a spectrum of the geometrical road profile with road class roughness-D is considered as given in Table 2. The vehicle is travelling with a constant speed v0, the time history data of road irregularity are described by power spectrum density (PSD) method [24]. According to International Standard Organization (ISO) 2631 [25], the ride comfort is specified in terms of root mean square (RMS) acceleration as shown in Table 3.

Table 2: Road roughness values classified by ISO

Classification S (Ω)	Road Roughness K [m^2/(cycles/m)] (*10^{-6}) 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

Table 3: RMS acceleration vs. Ride comfort [25]

Acceleration level	Degree of comfort
Less than 0.315m/s^2	Not uncomfortable
0.315-0.63 m/s^2	A little uncomfortable
0.5-1 m/s^2	Fairly uncomfortable
0.8-1.6 m/s^2	Uncomfortable
1.25-2.5 m/s^2	Very uncomfortable
Greater than 2 m/s^2	Extremely uncomfortable

5. Results and discussion

The air spring actuator and electrical actuator controlled active suspension systems are compared. Matlab/Simulink is used as a computer aided-control system tool for modelling the non-physical three DoF quarter car with actuators in one analysis loop as shown in Fig. 8. To verify the system, the seat acceleration, body acceleration and suspension deflection parameters are considered. For the given input parameters, the response of the system is observed on a 10 second scale. For analysing passenger comfort, the evaluated seat and body or sprung mass acceleration parameters are shown in Fig. 9 and Fig. 10 respectively.

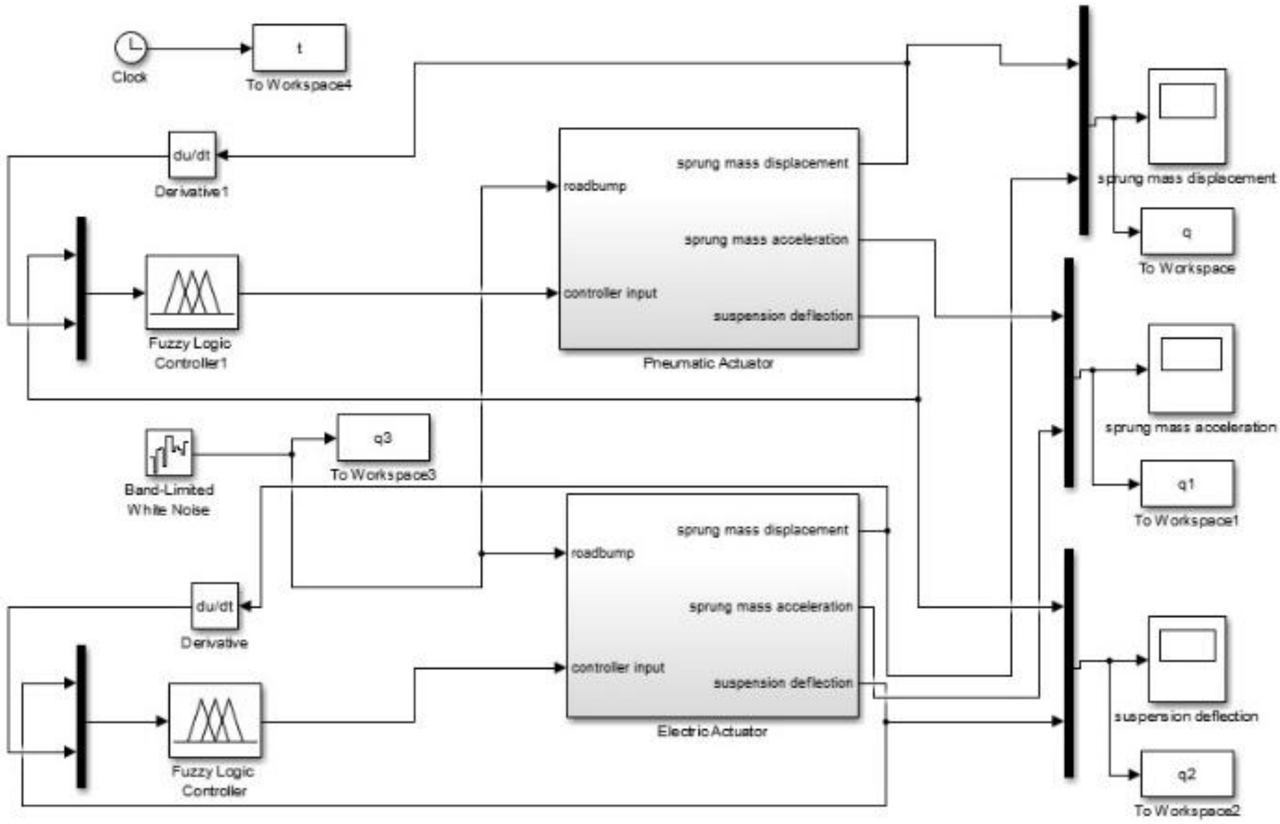

Fig. 8: Simulink block diagram for active suspension system model

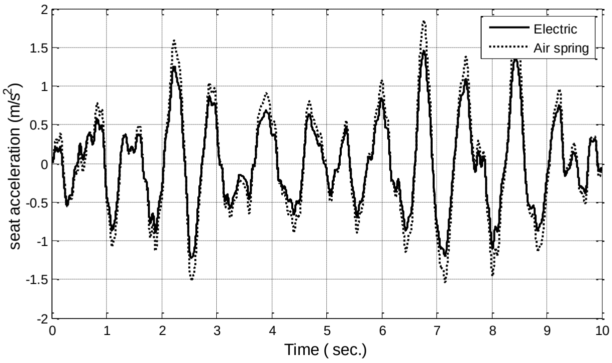

Fig. 9: Seat acceleration for D-class road roughness

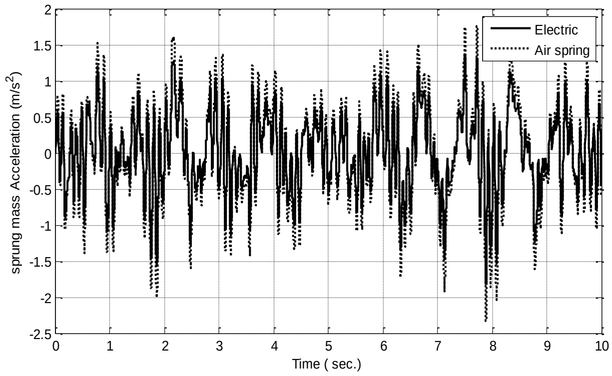

Fig. 10: Body acceleration for D-class road roughness

Due their response and characteristics of actuators and fuzzy rules, both the systems worked to nullify the ISO D-class road roughness. The RMS values of electrical actuator and air spring controlled suspension system seat acceleration are 0.588 and 0.743 m/s^2 respectively as given in Table 4. Both acceleration values are fairly uncomfortable; electrical actuator is less uncomfortable. The electrical actuator performs better than the air spring actuator [26]. Fig. 11 shows the relative displacement between vehicle sprung mass and unsprung mass, i.e., suspension deformation, for electric and air spring actuator based system. The response is better and ride comfort is stable when the electric actuator is used. In Fig. 11, the FLC controlled electric actuator has lower suspension deformation than the air spring suspension and passive suspension system, which also provides better ride comfort. Figs. 12 and 13 show the PSD of seat acceleration and sprung mass acceleration. From this, it is obvious that performance over the entire frequency range for both actuators are very similar but the electrical actuator is significantly more responsive in the mid frequency range.

Table 4: Comparison of RMS values of quarter car parameters

Parameter	Electrical actuator controlled	Air spring actuator controlled	Reduction in (%)
Seat acceleration (m/s^2)	0.5884	0.7439	20.90
Body acceleration (m/s^2)	0.6910	0.8903	22.38
Suspension deflection (m)	0.0068	0.0077	11.68

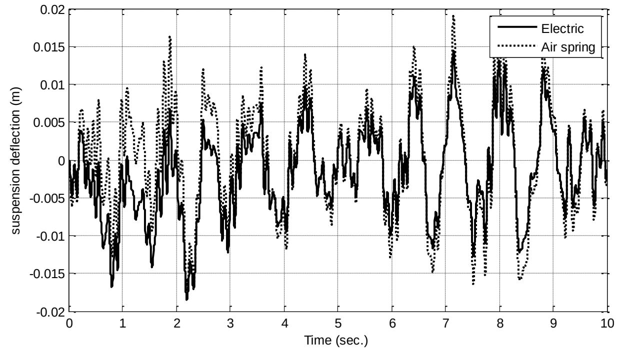

Fig. 11: Suspension deflection for D-class road roughness

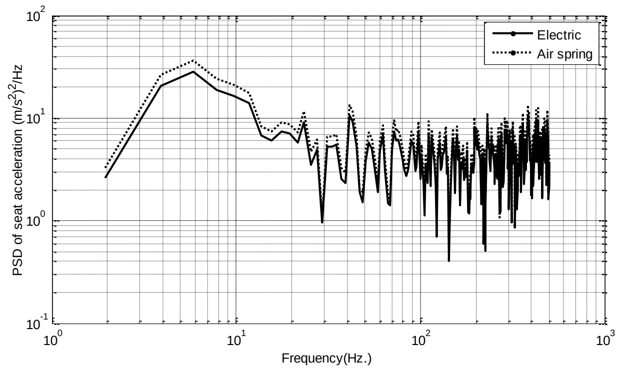

Fig. 12: PSD of seat acceleration for D-class road roughness

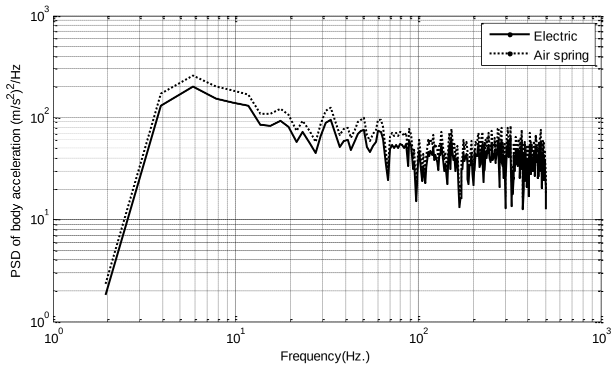

Fig. 13: PSD of body acceleration for D-class road roughness

6. Conclusion

In this paper, we discussed the characteristics of an air spring actuator and a proposed electrical actuator in a quarter car simulation model and their performances were analysed in frequency and time domains. The ISO D-class road roughness profile was given as an input for active suspension system model. Comparison of simulation results demonstrated that both the systems worked well and proposed electric actuator controlled active suspension system has significant improvements in performance over an air spring actuator based system.

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