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Comparison of Air Spring Actuator and Electro-Hydraulic Actuator in Automotive Suspension System

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ABSTRACT:

The present article introduces an approach that combines modelling and simulation of air spring actuator and electro-hydraulic actuator for comparison in automotive suspension system. Both hydraulic and air spring actuators are controlling the vehicle body by developing a desired force between sprung mass and unsprung mass using fuzzy logic controller. The vehicle body along with the wheel system is modelled as a two degree of freedom quarter car model. The actuator performance is investigated using the quarter car suspension model under single road bump with severe peak amplitude excitations and random road input. From the results of simulation, it can be concluded that air spring actuator gave better performance than electro-hydraulic actuator in all conditions under vertical body deflection.

KEYWORDS:

Air spring actuator; Electro-hydraulic actuator; Quarter car model; Automotive suspension; Fuzzy logic controller

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1. Introduction

The major purpose of any vehicle suspension system is to isolate the body from road unevenness disturbances and to maintain the contact between road and the wheel. Therefore, the vehicle suspension system is responsible for the ride quality and driving stability [1]. The conventional suspension system has coil or leaf springs in combination with hydraulic or pneumatic shock absorbers [2-4]. The design of a classical passive suspension system is a compromise between these conflicting demands. However, the improvement in vehicle dynamics in vertical direction is possible by developing an air spring actuator [5, 6] and electro-hydraulic actuator controlled suspension system [7, 8]. 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. Another important characteristic of air springs, which can be used for a mechatronic approach in suspension design, is the ability to provide a controlled variable force in terms of spring rate [9-11]. Moreover, they offer simple and inexpensive automatic levelling [12].

Nonlinear electro-hydraulic actuator can develop a desired force between the vehicle body and wheel axle [13]. This desired force is to achieve certain performance objective under external disturbances, such as passenger's comfort under road imperfections [14-16]. Developing a desired force between masses according to the incoming signal is difficult. Due to this both the actuators are tested for automotive suspension application and their performance is evaluated. The following sections detail the development of a quarter car model, control design using air spring actuator and electro-hydraulic actuator followed by simulations using MATLAB Simulink to compare their performances.

2. Quarter car model

The quarter car suspension system model consists of one-fourth of the body mass, suspension components and one wheel [15] as shown in Fig. 1. The assumptions of a quarter car model are as follows:

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

Both of the actuators will provide a desire force. The equations of motion for the sprung and unsprung masses of the quarter car model are given by,

$$M_s \dot{Z}_s + C_s\left(\dot{Z}_s - \dot{Z}_{us}\right) + K_s\left(Z_s - Z_{us}\right) - F_a = 0 \qquad (1)$$

$$M_{us}\ddot{Z}_{us} + C_s\left(\dot{Z}_{us} - \dot{Z}_s\right) + K_s\left(Z_{us} - Z_s\right) + K_{us}\left(Z_{us} - Z_r\right) + F_a = 0 \tag{2}$$

$$\ddot{Z}_s = \frac{1}{M_s}\left[C_s\left(\dot{Z}_{us} - \dot{Z}_s\right) + K_s\left(Z_{us} - Z_s\right) + F_a\right] \tag{3}$$

$$\ddot{Z}_{us} = \frac{1}{M_{us}}[C_s\left(\dot{Z}_s - \dot{Z}_{us}\right) + K_s\left(Z_s - Z_{us}\right) + K_{us}\left(Z_r - Z_{us}\right) - F_a] \tag{4}$$

Where F_a is control force. Z_s, Z_{us} and Z_s-Z_{us} are the sprung mass, unsprung mass displacement and suspension deflection respectively. The state space equation for active & semi active suspension is given by,

$$\begin{bmatrix}\dot{Z}_1\\ \dot{Z}_2\\ \dot{Z}_3\\ \dot{Z}_4\end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & -1\\ \frac{-K_s}{M_{us}} & \frac{-C_s}{M_{us}} & 0 & \frac{C_s}{M_{us}}\\ 0 & 0 & 0 & 1\\ \frac{-K_s}{M_s} & \frac{C_s}{M_{us}} & \frac{-K_{us}}{M_s} & \frac{-C_s}{M_s}\end{bmatrix}\begin{bmatrix}Z_1\\ Z_2\\ Z_3\\ Z_4\end{bmatrix} = \begin{bmatrix}0\\ \frac{1}{M_{us}}\\ -1\\ \frac{1}{M_s}\end{bmatrix}F_a + \begin{bmatrix}0\\ 0\\ -1\\ 0\end{bmatrix}\dot{Z}_r \tag{5}$$

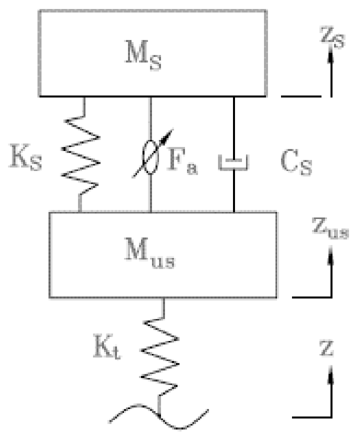

Fig. 1: Two degree of freedom quarter car model

3. Air spring actuator

The mathematical model in [11] is simplified with the following assumptions were made.

- The geometry of piston was cylindrical.
- Operating conditions were constant temperature and low-vibration frequency.
- The gas in the air spring actuator was ideal gas.
- The spring coefficient of the air spring is nonlinear.

The air spring force can be expressed as [6]:

$$F = \left(P_a - P_A\right)A \tag{6}$$

Where A is valve active area, P_A is the compressed air spring pressure, P_a is the indoor air spring pressure gas. The product of gas volume and gas pressure in the air spring was considered as constant. From the ideal gas equation, the following applies,

$$PV^n = const \tag{7}$$

$$P_1V_1^n = P_0V_0^n \tag{8}$$

Where subscripts 0 and 1 are for the initial and compressed air values. P and V are pressure and volume in the air spring. When the air spring was compressed at the length of h, the following function can be derived,

$$F(h) = \left(P_1 - P_A\right)A = \left(P_0V_0^n/V_1(h)^n - P_A\right)A(h) = \left(\frac{P_0}{\left(V_1(h)/V_0\right)^n} - P_A\right)A(h) \tag{9}$$

Fig. 2 shows a schematic of the air spring actuator. When the air spring was compressed to the length of h, the air volume in the chamber of air spring and the effective piston area can be expressed as the following nonlinear function of h:

$$\begin{aligned} V_1 &= V_0 + \alpha_1 h + \alpha_2 h^2 + \alpha_3 h^3\\ A_1 &= V_0 + \beta_1 h + \beta_2 h^2 + \beta_3 h^3 \end{aligned} \tag{10}$$

Combining Eqns. (6) and (10),

$$F(h) = \left(P_0\Big/\left(1 + \frac{\alpha_1}{V_0}h + \frac{\alpha_2}{V_0}h^2 + \frac{\alpha_3}{V_0}h^3\right)^n - P_A\right)\left(A_0 + \beta_1 h + \beta_2 h^2 + \beta_3 h^3\right) \tag{11}$$

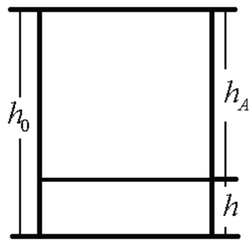

Fig. 2: Schematic diagram of the air spring

Since the operating condition for the spring was assumed as isothermal low-frequency environment, the value of n was equal to 1 and becomes,

$$\left(1 + \frac{\alpha_1}{V_0}h + \frac{\alpha_2}{V_0}h^2 + \frac{\alpha_3}{V_0}h^3\right)^n = 1 + \frac{1}{2}\left[\frac{\alpha_1}{V_0} + \left(\frac{\alpha_1}{V_0}\right)^m\right]h + \frac{1}{2}\left[\frac{\alpha_2}{V_0} + \left(\frac{\alpha_2}{V_0}\right)^m\right]h^2 + \frac{1}{2}\left[\frac{\alpha_3}{V_0} + \left(\frac{\alpha_3}{V_0}\right)^m\right]h^3 = 1 + b_1 h + b_2 h^2 + b_3 h^3 \tag{12}$$

Substituting Eqn. (12) into the Eqn. (11),

$$\frac{A(h)}{\left(V_1(h)/V_0(h)\right)^n} = A_0 + q_1 h + q_2 h^2 + q_3 h^3 \tag{13}$$

Combining Eqns. (13) and (11), Eqn. (9) can be written in the nonlinear function of h, as,

$$F(h) = \left(P_0 - P_A\right)A_0 + \left(P_0q_1 - P_A\beta_1\right)h + \left(P_0q_2 - P_A\beta_2\right)h^2 + \left(P_0q_3 - P_A\beta_3\right)h^3 \tag{14}$$

$$F(h) = \left(P_0 - P_A\right)A_0 - k_1 h - k_2 h^2 - k_3 h^3 \tag{15}$$

According to the assumption of cylindrical shape of piston, it was clear that $A_0 = A_A$. With the piston compression height h and Eqn. (7),

$$P_0V_0 = P_AV_A \tag{16}$$

Then Eqn. (6) can be rewritten as,

$$P_0 h_0 A_0 = P_A h_A A_A \tag{17}$$

According to Eqn. (17) and Fig. 2,

$$P_0 - P_A = P_0\left(\frac{-h}{h_0 - h}\right) \tag{18}$$

Finally, based on the Eqn. (16), the function of h was rewritten as:

$$F(h) = P_0 A_0\left(\frac{-h}{h_0 - h}\right) - k_1 h - k_2 h^2 - k_3 h^3 \tag{19}$$

4. Electro-hydraulic actuator

The hydraulic actuator is placed between body and wheel axles. This force is governed by the following equation [7, 17],

$$F_{Ai} = A_P P_{Li} \tag{20}$$

Where A_p is the cross section area of the piston inside i^{th} actuator. P_{Li} is the hydraulic pressure inside i^{th} actuator. The nonlinear pressure is given by,

$$\dot{P}_{Li} = -\beta P_{Li} - \sigma\left(A_P \dot{x}_{Pi} - Q_i\right) \tag{21}$$

$$\sigma = \frac{4\beta_e}{V_t}, \beta = \sigma C_{tP} \text{ and } x_{pi} = z_i - w_i \tag{22}$$

The spool valve displacement is controlled by an input voltage u_m. The corresponding dynamic relation can be simplified as a first order differential equation as,

$$\dot{Z}_{vi} = \frac{1}{\tau}\left(u_{mi} - Z_{vi}\right) \tag{23}$$

The parameters of the quarter car model and hydraulic actuator are given in Table 1.

Table 1: Quarter car suspension system model parameters [18, 19]

Notation	Parameter	Value
M_s	Sprung mass	290 kg
M_{us}	Unsprung mass	60 kg
K_s	Suspension stiffness	16200 Nm^{-1}
K_{us}	Tire stiffness	191000 Nm^{-1}
C_s	Suspension damping	1000 Nsm^{-1}
α, β, γ	Actuator parameters	4.52×10^{13}, 1, 1.55×10^{9}
A_P	Piston cross section area	$3.35*10^{-4}$ m^2
	Supply pressure	103.354 bar
C_d	Discharge coeff.	0.7
ρ	Fluid density	970 kgm^{-3}
ω	Area gradient	1.436×10^{-2} m^2

5. Controller design

The fuzzy logic controller used in the active suspension has body velocity and suspension deflection as inputs and desired actuator force F_a as output. The control system consists of fuzzification, fuzzy inference and defuzzification. The fuzzification stage converts real number input values into fuzzy value, while the fuzzy inference machine processes the input data and computes the controller outputs coping with the rule base and data base [20]. These fuzzy value outputs are converted into real numbers by the defuzzification stage. The membership functions for the considered variables of the active suspension system represented by a fuzzy set is as - Positive Small (PS), Positive Large (PL), Zero (ZE), Negative Small (NS) and Negative Large (NL). The universe of discourse for both the input and output variables were classified into five sections [21] as - Positive Small (PS), Positive Large (PL), Zero (ZE), Negative Small (NS) and Negative Large (NL). Each rule is derived from the characteristic of the passive suspension system. Mamdani's [20] minimum operation rule is used as a fuzzy implication function. As the process usually requires a non-fuzzy value of control, a centroid method of defuzzification is used.

6. Results and discussion

The random road inputs are used to verify the developed control system and study the characteristics of air spring actuator (ASA) controlled and electro-hydraulic actuator (EHA) controlled suspension systems. Matlab/Simulink is used as a computer aided control system tool for modelling the non-physical two degrees of freedom quarter car model with actuators and their modelling are included in one analysis loop. To verify the performance of actuators, the body displacement, acceleration and suspension deflection responses of the system are observed on 5 seconds scale. Figs. 3 & 4 show the acceleration responses of sprung and unsprung masses respectively. Fig. 5 shows the suspension travel for random road condition. Table 2 lists a summary of comparison results between ASA and EHA controlled suspension system models. The response of ASA controlled suspension system is lower at all points due to air cushioning and stiffness changing. From the graph ASA controlled suspension maximum peak point is 0.0075m and RMS value is 0.0037m. The ASA controlled suspension is better than EHA controlled suspension for good road holding and lesser rattle space.

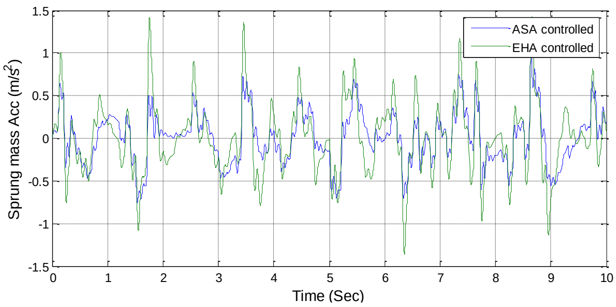

Fig. 3: Sprung mass acceleration for random road input

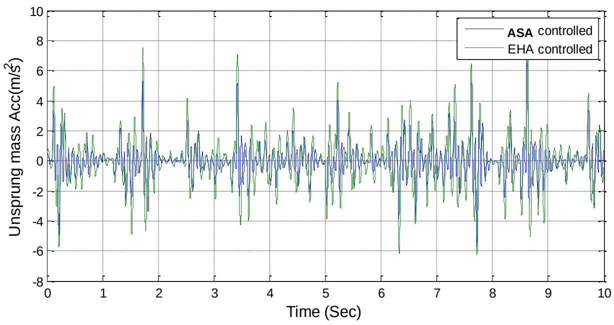

Fig. 4: Unsprung mass acceleration for random road input

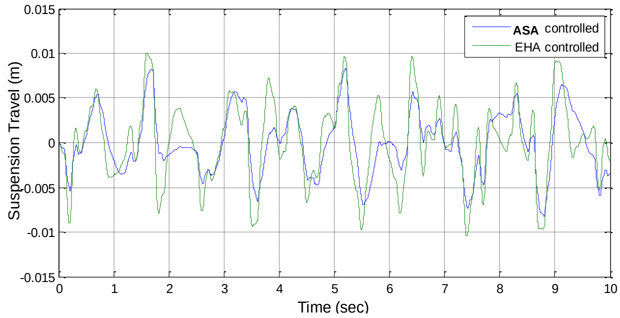

Fig. 5: Suspension travel for random road input

Table 2: Comparison of ASA & EHA controlled suspension

Comparison	Sprung mass acceleration (m/s^2)	Unsprung mass acceleration (m/s^2)	Suspension deflection (m)
RMS – ASA controlled	0.3900	1.1162	0.0037
RMS – EHA controlled	0.5571	1.9322	0.0049
ASA – EHA (%)	29.99	39.85	24.48

7. Conclusion

The air spring actuator and electro hydraulic actuator utilized for light passenger vehicle suspension system were tested under same condition. Peak to peak and root mean square values of acceleration and displacement responses from a quarter car model were used to compare the performance of two system models. From the results of simulation, it can be concluded that air spring actuator gave better performance than electro-hydraulic actuator in all conditions under considered vertical body deflection.

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