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# Evaluation of Human Exposure to Vibration Subjected to Active Suspension Actuators

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**ABSTRACT:**

*This paper details the assessment of human response to vibration through modelling of seated human body using seven degrees of freedom lumped mass model. Continued human exposure to chronic vibrations may subsequently leads to person's discomfort. To avoid this discomfort, an active suspension with combination of electro-hydraulic, pneumatic or air spring actuator is introduced between sprung mass and the unsprung mass which is controlled by a PID controller. For the simulation, ISO D-class road is given as input for the designed Matlab Simulink model and the results were compared. The simulation result shows that air spring actuators based active suspension can effectively attenuate the vertical vibration acceleration and increase the riding comfort.*

**KEYWORDS:**

*Human body; Lumped mass; Vibration control; Active suspension; Hydraulic and pneumatic actuator*

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## 1. Introduction

The main aim of the automotive suspension system is to isolate the vehicle body from road uncertainness. The vehicle suspension is categorized into passive, semi-active and active suspension. Passive suspension cannot control over the design range, due to fixed parameters of passive elements. Semi-active suspension control system is characterised into combination of variable stiffness and conventional damper, variable damper with conventional coil spring. The actuator is placed parallel with conventional spring and damper is known as active suspension system. The active suspension system performance depends on actuator characteristics and controller design [1-4]. Therefore the active suspension simulation block model was developed with the combination of Electro-hydraulic and Air spring actuator. Although human body is a unified and complex active dynamic system, lumped parameter models are often used to evaluate the human dynamic properties [5].

The human body is a very sophisticated dynamic system whose mechanical properties vary from one moment to another and from one individual to another. From the results of a large amount of experimental data, various biodynamic models have been developed to describe human motion. These models can be grouped as lumped-parameter (LP), finite element (FE), and multi-body (MB) models [6]. LP models consisting of multiple lumped masses inter connected by ideal springs and ideal dampers have proven to be effective in many applications, including those involving human exposure to whole body vibration [5]. In this work, the human body along with the seat, vehicle body and tire is modelled using seven degrees of freedom (DoF).

## 2. Mathematical modelling

Fig. 1 shows the biodynamic lumped human linear seat model coupled with quarter model of ground vehicles. It is assumed that the system does not vibrate in lateral or longitudinal directions, the subject is exposed to vertical vibration [7-12] only. The equations of motion for the developed 7 DoF model are given by,

$$m_1\ddot{z}_1 = -k_1(z_1 - z_2) - c_1(\dot{z}_1 - \dot{z}_2)$$

$$m_2\ddot{z}_2 = -k_2(z_2 - z_3) - c_2(\dot{z}_2 - \dot{z}_3) + k_1(z_1 - z_2) + c_1(\dot{z}_1 - \dot{z}_2)$$

$$m_3\ddot{z}_3 = -k_3(z_3 - z_4) - c_3(\dot{z}_3 - \dot{z}_4) + k_2(z_2 - z_3) + c_2(\dot{z}_2 - \dot{z}_3)$$

$$m_4\ddot{z}_4 = -k_4(z_4 - z_5) - c_4(\dot{z}_4 - \dot{z}_5) + k_3(z_3 - z_4) + c_3(\dot{z}_3 - \dot{z}_4)$$

$$m_5\ddot{z}_5 = -k_5(z_5 - z_5) - c_5(\dot{z}_5 - \dot{z}_6) + k_4(z_4 - z_5) + c_4(\dot{z}_4 - \dot{z}_5)$$

$$m_6\ddot{z}_6 = -k_6(z_6 - z_7) - c_5(\dot{z}_5 - \dot{z}_6) + k_5(z_5 - z_5) + c_5(\dot{z}_5 - \dot{z}_6) - F$$

$$m_7\ddot{z}_7 = -k_6(z_6 - z_7) + c_6(\dot{z}_6 - \dot{z}_7) + k_6(z_6 - z_7) + F$$

where $m_1$ to $m_7$ are the human head, upper torso, lower torso, pelvis, passenger seat, sprung mass, unsprung masses respectively. $k_1$ to $k_7$ are the stiffness and $c_1$ to $c_7$ are the damping constant respectively.

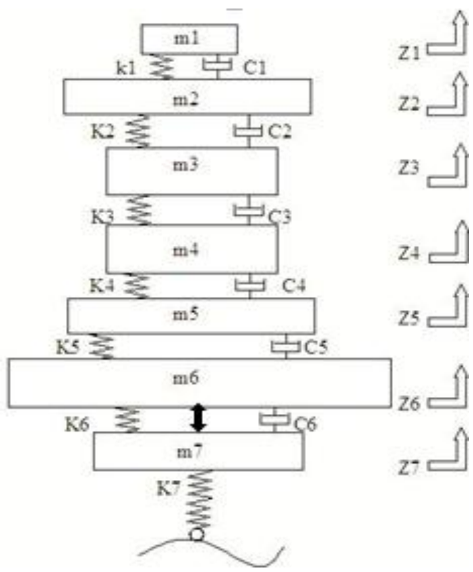


**Fig. 1: Active suspension with human body lumped mass model**

## 3. Dynamics modelling

To determine the force at any given point in the stroke of an air spring, the atmospheric pressure must be backed out of the absolute pressure values to yield gauge pressure. Typical value of the absolute pressure is p = 300 - 700kPa. The force on the spring at any given point of the load deflection curve is the gauge pressure multiplied with the effective area [13] using,

$$F_{pa} = P_{pa} * A_e \tag{1}$$

The force developed by the pneumatic actuator can be evaluated as follows,

$$K_{as,dynamic} = \gamma\left(P_\gamma + P_a + P_g\right)A_e^2 / V \tag{2}$$

$$K_{as,static} = \left(P_\gamma + P_a + P_g\right)A_e^2 / V \tag{3}$$

The nonlinear force produced by the active hydraulic actuator is placed between body and wheel axles. This force is governed by [14, 15],

$$F_{Ai} = A_P P_{Li} \tag{4}$$

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

$$\dot{P}_{Li} = -\beta P_{Li} - \sigma\left(A_P \dot{x}_{Pi} - Q_i\right) \tag{5}$$

$$\sigma = \frac{4\beta_e}{V_t},\ \beta = \sigma C_{tP}\ and\ x_{pi} = z_i - w_i \tag{6}$$

Where $\beta_e$ is the effective bulk modulus of hydraulic system. $V_t$ is the total volume of fluid under compression. $C_{tp}$ is leakage coefficient of piston.

$Q_i$ is the hydraulic flow through the piston inside the $i^{th}$ actuator as follows,

$$Q_i = C_d\,\omega x_{vi}\sqrt{P_{si} - \mathrm{sgn}\left(x_{vi}\right)P_{Li} / \rho} \tag{7}$$

where $C_d$ is the discharge coefficient, $\omega$ is the area gradient, $x_v$ is the spool valve displacement, $\rho$ is the fluid density, and $Ps$ is the supply piston pressure. The spool valve displacement is controlled by an input voltage um. The corresponding dynamic relation can be simplified as a first order differential equation as follows,

$$\dot{Z}_{vi} = \left(u_{mi} - Z_{vi}\right)/\,\tau \tag{8}$$

The PID controller design is defined by,

$$U_c = K_P e(t) + \frac{K_p}{T_i}\int_0^t e(t)dt + K_P T_d \frac{de(t)}{dt} \tag{9}$$

Where $U_c$ is the current input from the controller, $K_P$ is the proportional gain, $T_i$ and $T_d$ is the integral and derivative time constant of the PID controller respectively [16]. The values of the parameter $K_P$, $T_i$ and $T_d$ are set according Zeigler-Nichols tuning rules.

To generate the road profile of a random base excitation for the 2-DoF Semi-active suspension simulation disturbance, a spectrum of the geometrical road profile with road class roughness-D is considered as given in Table 1. The vehicle is travelling with a constant speed $v_0$, the time histories data of road irregularity are described by PSD method [17-19]. According to International Standard Organization (ISO)2631 [18], the ride comfort is specified in terms of root mean square (RMS) acceleration as per Table 2 [20]

**Table 1: Road roughness value 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 2: Acceleration level and degree of comfort defined in ISO2631-1 [21-23]**

| 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 |

## 4. Actuator robustness test

The electro-hydraulic and pneumatic actuator controlled suspension with human body block model developed separately, and both the systems are tested on the same input condition. Figs. 2 and 3 show the head and upper torso acceleration respectively. Due to lower transmissibility coefficients, the pneumatic actuator controlled suspension peak point and root mean square values are lower than hydraulic active suspension system. Head RMS acceleration value of hydraulic and pneumatic actuator 0.5283 and 0.4808m/s$^2$ respectively. Figs. 4 and 5 show the lower torso and pelvic acceleration respectively. Both the lower torso and pelvic acceleration of pneumatic controlled suspension RMS is 4.3% lesser than the hydraulic controlled suspension. Table 3 presents the comparison of RMS values for hydraulic and pneumatic controlled systems.

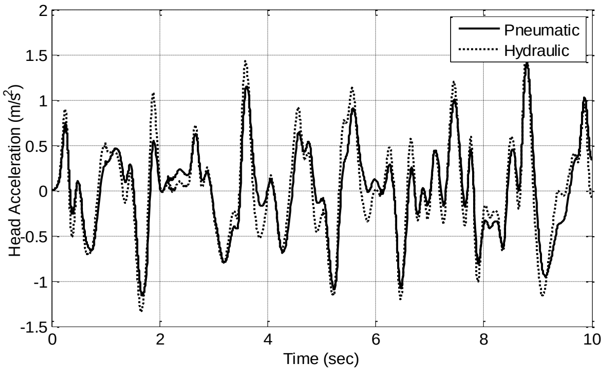


**Fig. 2: Head acceleration**

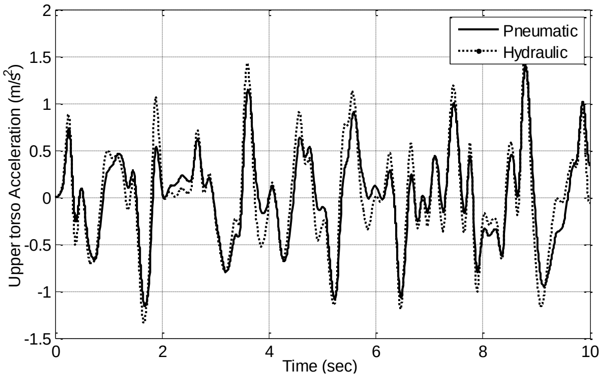


**Fig. 3: Upper torso acceleration**

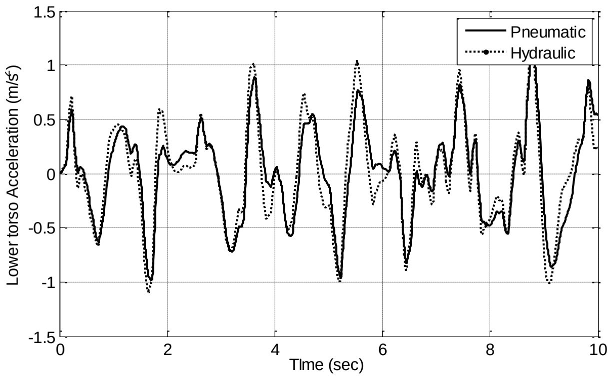


**Fig. 4: Lower torso acceleration**

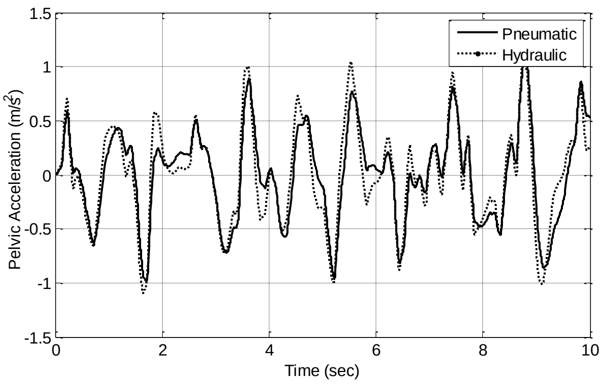


**Fig. 5: Pelvic acceleration**

**Table 3: RMS values comparison of human body**

| Acceleration (m/s$^2$) | Pneumatic | Hydraulic | % of reduction |
|---|---|---|---|
| Head | 0.4808 | 0.5283 | 8.99 |
| Upper Torso | 0.4794 | 0.5262 | 8.89 |
| Lower Torso | 0.4240 | 0.4432 | 4.32 |
| Pelvic | 0.4232 | 0.4421 | 4.27 |

## 5. Conclusions

The hydraulic and pneumatic actuator controlled active suspension with human body was analysed. The peak to peak method and root mean square method has been used to investigate the performance of different control methods. The ISO D-class road profile was given as an input for active suspension system. From the results of simulations, it can be stated that the pneumatic actuator control suspension can achieve substantial reduction of peak acceleration than that of hydraulic controlled suspension. Intelligent controller resulted in more comfort and substantial reduction of the vertical acceleration.

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