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Fuzzy control of a hydro-pneumatic suspension system for dump trucks

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ABSTRACT: *A dump truck is a heavy-duty vehicle that operates frequently in uneven terrains. This study aims to enhance the vehicle's ride comfort by controlling the hydro-pneumatic suspension system parameters. Initially, a dynamic model is developed for the vehicle. Subsequently, a fuzzy controller is designed to regulate the suspension system parameters. Finally, Matlab/Simulink software is utilized for simulation, and the average vertical acceleration and pitch acceleration of the vehicle body are selected as objective functions. The simulation outcomes reveal that the active hydro-pneumatic suspension system (A-HPS) can substantially reduce the vibration of the vehicle body and enhance the suspension system's performance compared to the passive hydro-pneumatic suspension system(P-HPS).*
KEYWORDS: *Dump truck, hydro-pneumatic suspension, fuzzy control*

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I. INTRODUCTION

Heavy truck vibration is a significant issue that affects the driver's ride comfort and can cause structural damage to the vehicle. Researchers have studied various methods to reduce truck vibration and improve driver comfort. One study was used a dynamic simulation model to analyze the vibration levels of a heavy truck cabin and suggested improving the suspension system to reduce vibration levels and improve ride comfort [1]. Another study was compared the dynamic wheel loads and ride comfort of a semi-trailer truck with air-spring and leaf-spring suspension systems, and found that the air-spring suspension system resulted in better ride comfort [2]. Le Van Quynh et al. have investigated the effects of suspension design parameters on ride comfort and road surface friendliness, indicating the importance of finding a balance between the two [3]. Vanliem Nguyen et al. was proposed a three-dimensional model of the suspension system and analyzed its dynamic characteristics using the finite element method [3]. Yongzhu Hu et al. were focused on the sensitivity analysis of dynamic parameters on ride comfort, and found that suspension damping coefficient and tire stiffness significantly impact ride comfort [5]. Peng Guo et al. were proposed an optimization method for suspension parameters to improve ride vibration, and found that the optimized suspension system effectively reduced ride vibration [6]. Van Liem Nguyen et al. were explored the ride comfort performance of heavy trucks with semi-active isolation systems [7]. Basaran Sinan et al. were proposed an adaptive vectorial backstepping control strategy for an electromagnetic active suspension system to reduce vibration in truck cabins [8]. Wangqiang Xiao et al. were investigated the influence of particle damping on ride comfort in mining dump trucks, and found that particle damping significantly improves ride comfort [9]. Gangfeng Wang et al. were analyzed the loading impact damping characteristics of a two-stage pressure hydro-pneumatic suspension system for a mining dump truck, and found that the system effectively reduces impact force and improves damping[10]. The study was discussed the impact of truck vibration on road surface damage and suggests that the use of air suspension systems can effectively reduce road surface damage caused by heavy-duty trucks [1].

In this study, a half-model vertical dynamic model is established to develop a new mathematical model for comparing the active hydro-pneumatic suspension system with the passive hydro-pneumatic suspension system to determine vertical force. The average vertical acceleration and pitch acceleration of the vehicle body are chosen as objective functions. The simulation and calculation of objective functions are carried out using Matlab/Simulink software. Finally, the performance of the active hydro-pneumatic suspension system is evaluated and compared with that of the passive hydro-pneumatic suspension system while the vehicle was traveling on bumpy roads

II. HALF-VEHICLE DYNAMIC MODEL OF A DOUBLE –DRUM VIBRATORY ROLLER

In order to analyze the performance of the hydro-pneumatic suspension system, a half model vehicle dynamic model with both the active hydro-pneumatic suspension system and the passive hydro-pneumatic suspension system is established under the random road surface excitation for analysis, as shown in Fig.1(a).

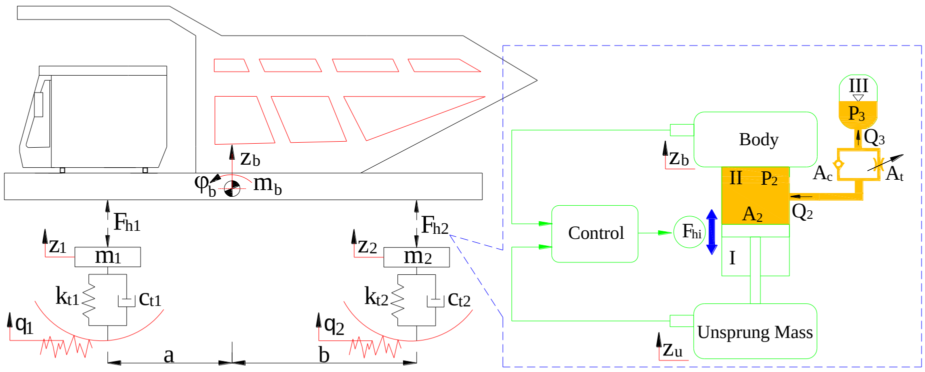

Fig. 1. Half model vehicle suspension models with hydro-pneumatic suspension systems.

From Fig.1(a), the motion equations of unspung and sprung masses using Newton's second law are written as follows

$$\begin{cases} m_b\ddot{z}_b = -(F_{h1}+F_{h2}) \\ I_b\ddot{\theta}_b = F_{h1}.a - F_{h2}.b \\ m_1\ddot{z}_1 = F_{h1} - \left[k_{t1}(z_1-q_1)+c_{t1}(\dot{z}_1-\dot{q}_1)\right] \\ m_2\ddot{z}_2 = F_{h2} - \left[k_{t2}(z_2-q_2)+c_{t2}(\dot{z}_2-\dot{q}_2)\right] \end{cases} \tag{1}$$

where, m_b and m_1, m_2 are the sprung of body and unsprung masses, respectively; k_t and c_t are the tire stiffness and damping coefficients, respectively; z_b and z_1, z_2 are the vertical displacements of the sprung and unsprung masses, respectively; q is the random road surface roughness, F_{hi} is the vertical dynamic forces of the hydro-pneumatic suspension systems (i=1-2)

Determining the vertical dynamic forces of hydro-pneumatic suspension system. Based on the mathematical model of the hydro-pneumatic cylinder shown in Fig. 1, the vertical force of the hydro-pneumatic suspension system is defined.

The volume flow rates through the orifices are related to pressure differentials across the orifices in the following manner:

$$Q_3 = C_d A_s \sqrt{2g\frac{(p_2-p_3)}{\rho}} \tag{2}$$

where, C_d is the discharge coefficient; A_s is the orifice area and consists of the area of the check valve and throttle valve, $A_s = A_t + A_c$; ρ is the mass density of the fluid; p_2 and p_3 are the real-time pressures in chamber II and chamber III, respectively.

According to the continuity equations, we can obtain that:

$$Q_2 = -Q_3 \tag{3}$$

The rate of change of the fluid volume in hydraulic cylinder is related to the relative velocity across the respective strut:

$$Q_2 = A_2(\dot{z}_b - \dot{z}_u)\,\mathrm{sign}(\dot{z}_b - \dot{z}_u) \tag{4}$$

From (2) and (4), the pressure in the ring oil chamber can be determined by

$$p_2 = p_3 + \frac{\rho A_2^{\,2}(\dot{z}_b - \dot{z}_u)^2}{2C_d^2 A_s^{\,2}}\,\mathrm{sign}(\dot{z}_s - \dot{z}_u) \tag{5}$$

When the damper is compressed, the adjusted gas chamber pressure is given by

$$p_0V_0^k = pV^k \tag{6}$$

where p_0 and V_0 are the initial pressure and volume in air chamber, p and V are the pressure and volume in air chamber when the damper is compressed, k is the polytrophic rate (1< k <1.4).

$$p_3 = p_0\left(\frac{V_0}{V_0 + A_2(z_b - z_u)}\right)^k \tag{7}$$

In which $V = V_0 + A_2(z_b - z_u)$

The vertical dynamic force of hydro-pneumatic suspension system can be computed by

$$F_{hi} = -\frac{\rho A_2^{\ 3}(\dot{z}_b - \dot{z}_u)^2}{2C_d^2 A_s^{\ 2}}\mathrm{sign}(\dot{z}_s - \dot{z}_u) + p_0\left(\left(\frac{V_0}{V_0 + A_p(z_b - z_u)}\right)^k - 1\right)A_2 \tag{8}$$

Bump road profile. A bump road profile refers to a type of road surface that has a series of small, sharp, and abrupt changes in elevation, resulting in a rough and uneven ride for vehicles traveling over it. The formula for determining the vertical displacement of a single bump on a road profile is:

$$q_1 = \begin{cases} \frac{a}{2}(1-\cos\left(\frac{2\pi v t_f}{\lambda}\right) & 1 \le t_f \le 1+ \\ 0 & \text{otherwise} \end{cases} \tag{9}$$

$$q_2 = \begin{cases} \frac{a}{2}(1-\cos\left(\frac{2\pi v t_r}{\lambda}\right) & t_d \le t_r \le \frac{\lambda}{v}+ \\ 0 & \text{otherwise} \end{cases}$$

where, $t_r = t_f + t_d = t_f + 1 + \frac{l_f + l_r}{v}$; a is the amplitude of the bump, is the amplitude of the bump, λ is Input disturbance wavelength, v is vehicle forward velocity, length between sprung mass centroid and front l_f and rear l_r ends

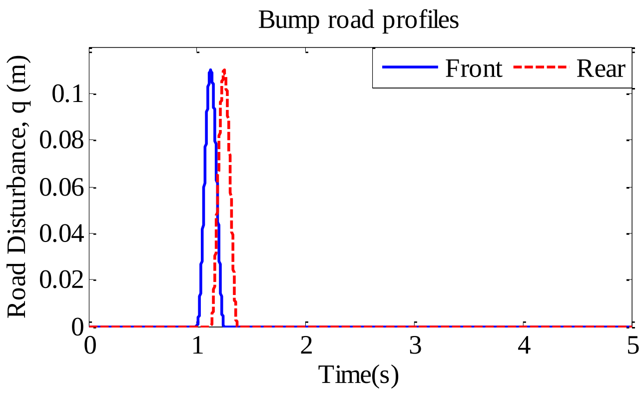

Fig.2. Bump road profile

Vehicle ride comfort evaluation method. Vehicle ride comfort indicator is based on ISO 2631-1 (1997)[15], vibration evaluation based on the basic evaluation method including measurements of the weighted root-mean-square (rms) acceleration is defined by:

$$a_w = \left[\frac{1}{T}\int_0^T a^2(\mathrm{t})\,\mathrm{dt}\right]^{\frac{1}{2}} \tag{10}$$

where a(t) is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurements.

III. FUZZY LOGIC CONTROL DESIGN

In this study, the fuzzy logic controller comprises two inputs, namely, the body acceleration and suspension relative displacement, while the spool displacement of electric hydraulic proportional valve is the resulting outputs that eventually control the vehicle's response, fuzzy control is shown in Figure 2.

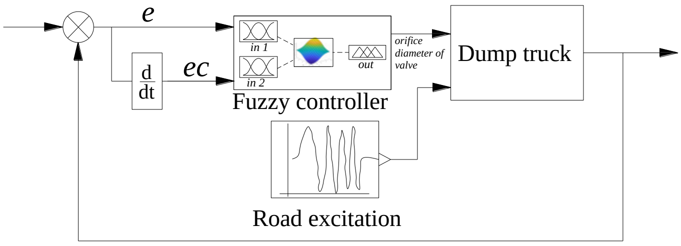

Fig.3. Layout of fuzzy logic control

The active suspension's fuzzy controller employs e and ec, both ranging from [-6, 6], and the valve spool displacement of the domain is within [0, 10]. Specifically, two inputs, e and ec, and one output are defined as seven fuzzy subsets of PB PM PS Z NS NM NB, with the output variable's membership function identical to the input error and error rate of change, both featuring a triangular membership function. By using MAX-MIN fuzzy inference after fuzzification based on the number of fuzzy sets and membership functions, the fuzzy rule table of fuzzy output variables can be derived. The fuzzy control rules are established through the empirical induction method, with the self-tuning fuzzy control rules of the output displacement presented in Table 1[13].

The fuzzy output can be obtained via the fuzzy rules outlined in Table 1, and the parameters can be acquired through the inverse model. By doing so, the hydro-pneumatic suspension system can be controlled effectively.

Table 1. Fuzzy control of output displacement of valve

ec \ e	NB	NM	NS	Z	PS	PM	PB
NB	PB	PB	PM	PM	PS	PS	Z
NM	PB	PB	PM	PM	PS	Z	Z
NS	PM	PM	PM	PS	Z	NS	NM
Z	PM	PS	PS	Z	NS	NM	NM
PS	PS	PS	Z	NS	NS	NM	NM
PM	Z	Z	NS	NM	NM	NM	NB
PB	Z	NS	NS	NM	NM	NB	NB

IV. RESULTS AND DISCUSSION

In order to analyze the efficiency of the fuzzy controller for the active hydro-pneumatic suspension system in improving vehicle ride comfort, Matlab/Simulink software is used to simulate the differential equations of motion in Section 2 and calculate the set of parameters using Eq. (10) based on the reference Table 2. The simulation is conducted on an Bump road with the vehicle speed set at 60 km/h.

Table 2. The constraint of parameters of hydro-pneumatic suspension system

Parameters	Value	Parameters	Value	Parameters	Value
$\mu/(N.s/m^2)$	0.6	$A_t(m^2)$	2x10^-4	$p_0(Mpa)$	2
$\rho/(kg/m^3)$	850	$A_c(m^2)$	5x10^-4	$v_0(m^3)$	0.0026
k	1.3	$A_2(m^2)$	0.01	C_d	0.62

The acceleration responses of the vertical body (a_b) and pitch angle of the body (a_φ) shown in Fig.3 and Fig.4

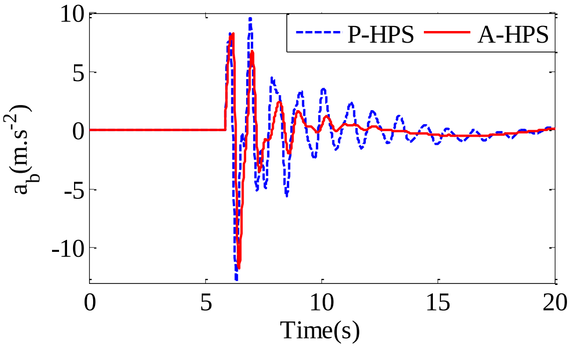

Fig.3. Acceleration responses of the vertical body

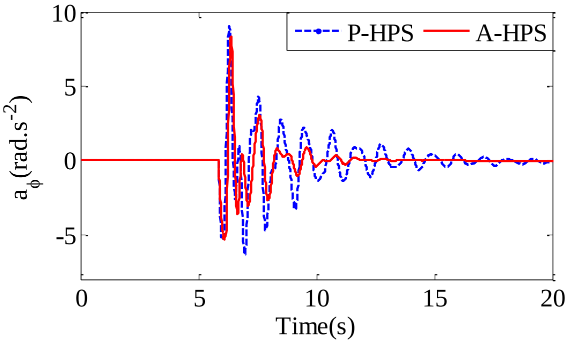

Fig.4. Acceleration responses of body pitch

From the results of Fig. 3 and Fig.4 , we could be determined the values of the weighted root-mean-square (r.m.s.) accelerations of the vertical body (a_{wb}), body pitch ($a_{w\varphi}$) are 2.07 m.s-2 and 1.43 rad.s-2, respectively with the passive hydro-pneumatic suspension system(P-HPS). The values of the weighted root-mean-square (r.m.s.) accelerations of the vertical body (a_{wb}), body pitch ($a_{w\varphi}$) are 1.7 m.s-2 and 1.08 rad.s-2, respectively with the active hydro-pneumatic suspension system (A-HPS)

The results presented in Fig. 3 and Fig. 4 demonstrate that the values of AWB and AWPhi are significantly reduced when using the active hydro-pneumatic suspension system (A-HPS) compared to the passive hydro-pneumatic suspension system (P-HPS) reduction 22.24% and 32.01%, respectively. The analysis results indicate that the active hydro-pneumatic suspension system has a significant positive effect on vehicle ride comfort.

V. CONCLUSION

In this paper, a half-vehicle suspension model with a hydro-pneumatic suspension system has been established based on modeling and verification. The damping force of the hydro-pneumatic suspension was adjusted by controlling the electro-hydraulic proportional valve using fuzzy method. Simulation results show that the active hydro-pneumatic suspension system has reduced the values of the weighted root-mean-square (r.m.s.) accelerations of the vertical body (a_{wb}) and body pitch ($a_{w\varphi}$) by 22.24% and 32.01%, respectively with bump road excitation compared to the passive hydro-pneumatic suspension. The results demonstrate that the active hydro-pneumatic suspension system can effectively improve the vehicle ride comfort.

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