



## A study on controlling air chamber pressure of hydro-pneumatic suspension system towards improving mining dump truck ride comfort

Le Xuan Long <sup>1</sup>, Vu Van Hai <sup>2</sup>, Vu Thi Hien <sup>3\*</sup>

<sup>1-3</sup> Faculty of Vehicle and Energy Engineering, Thai Nguyen University of Technology, Thai Nguyen 250000, Viet Nam

\* Corresponding Author: **Vu Thi Hien**

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### Abstract

This study presents the development of a mathematical model and control strategy for a hydro-pneumatic suspension system (HPS) to enhance ride comfort in a mining dump truck. A nonlinear dynamic model is formulated, incorporating the vertical force components of the HPS to accurately represent the system's behavior under real operating conditions. To improve ride comfort, two control methods are proposed: a Proportional-Integral-Derivative (PID) controller and a Fuzzy Logic controller. These controllers are designed to regulate the pressure in the air chamber of the suspension system, which plays a critical role in absorbing shocks and maintaining vehicle stability. Simulations are performed using a quarter-vehicle model in MATLAB Simulink to assess the effectiveness of the proposed control strategies. The results indicate that both PID and Fuzzy Logic controllers significantly reduce the vertical displacement and acceleration of the vehicle body when compared to a conventional passive suspension system. Specifically, the root mean square (RMS) value of body acceleration ( $a_{wb}$ ) is reduced by 18.39% with the PID controller and by 27.55% with the Fuzzy Logic controller, indicating a significant improvement in ride comfort.

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**Keywords:** Mining dump truck, hydro-pneumatic suspension system, semi active suspension system, air pressure chamber, ride comfort

### 1. Introduction

HPS have become a cornerstone in the advancement of vehicle suspension technology, offering significant improvements in ride comfort, handling, and adaptability compared to traditional systems. These systems utilize a combination of compressed gas and hydraulic fluid to absorb road disturbances effectively, offering the advantage of adjustable stiffness and damping. This flexibility allows HPSs to dynamically respond to various driving conditions, contributing to a smoother ride and better control. Recent studies <sup>[1]</sup> have highlighted the superior vibration isolation and load adaptability of hydro-pneumatic struts utilizing gas-oil emulsions, emphasizing their pivotal role in enhancing modern vehicle performance. Traditional passive suspension systems, although reliable and cost-effective, have intrinsic limitations due to their fixed mechanical properties. They are unable to adapt to varying road profiles or dynamic driving conditions. In contrast, active and semi-active suspension systems—particularly those based on hydro-pneumatic mechanisms—have emerged as powerful alternatives. These systems can adapt in real-time through advanced control strategies, enabling improved ride comfort and vehicle stability. Among the control approaches developed for hydro-pneumatic suspension systems, fuzzy logic control <sup>[2]</sup>, adaptive control <sup>[3]</sup>, and sliding mode control <sup>[4]</sup> have shown significant promise. These techniques are capable of dynamically adjusting suspension parameters in response to changing road and load conditions. Fuzzy logic controllers, for instance, have been successfully applied to optimize suspension behavior under diverse road scenarios <sup>[2]</sup>, while adaptive control strategies have been specifically tailored to meet the needs of heavy-duty vehicles <sup>[3]</sup>, ensuring consistent performance across varying operational contexts. Despite the benefits of active and semi-active suspension systems, energy efficiency remains a pressing concern. Active systems require substantial power input, raising the need for sustainable operation. Research has turned to energy regeneration technologies, which aim to recover and

reuse energy from suspension movements. Various studies have reviewed energy harvesting mechanisms <sup>[5]</sup>, demonstrating the feasibility of integrating regenerative components without compromising suspension performance. Additionally, researchers have explored multi-objective optimization methods to achieve a balance between ride comfort and handling stability <sup>[6]</sup>. These methods enable system designers to fine-tune control strategies to meet multiple performance criteria. Machine learning has also been introduced as a powerful tool for predicting road profiles and proactively adjusting suspension parameters <sup>[7]</sup>, further enhancing vehicle adaptability and passenger comfort. Recent advancements have addressed the nonlinear and complex nature of hydro-pneumatic suspension systems. Nonlinear modeling techniques <sup>[8]</sup> have been developed to accurately describe system dynamics, while the integration of energy harvesting systems has been validated through experimental and simulation studies <sup>[9]</sup>. Moreover, the use of neural networks <sup>[10]</sup> and advanced sensor technologies <sup>[11]</sup> has improved control accuracy and system responsiveness. Applications of these innovations have expanded into specialized contexts, particularly for heavy-duty and off-road vehicles <sup>[13]</sup>, where robust performance is critical. Further research has tackled issues of control robustness <sup>[14]</sup> and evaluated the broader implications on ride quality and driver health <sup>[15]</sup>. The authors' studies <sup>[17- 22]</sup> have proposed nonlinear dynamic models to influence the characteristics of HPSs on their efficiency. In this study, we focus on building a Fuzzy logic and PID controller to control the air chamber pressure towards improving vehicle ride comfort.

## 2. Vehicle dynamic model

Fig. 1 presents the structural configuration of the HPS. A nonlinear dynamic model has been formulated based on this structural representation, incorporating the stiffness coefficient  $k_h$  and damping coefficient  $c_h$ , as illustrated in Fig. 2.

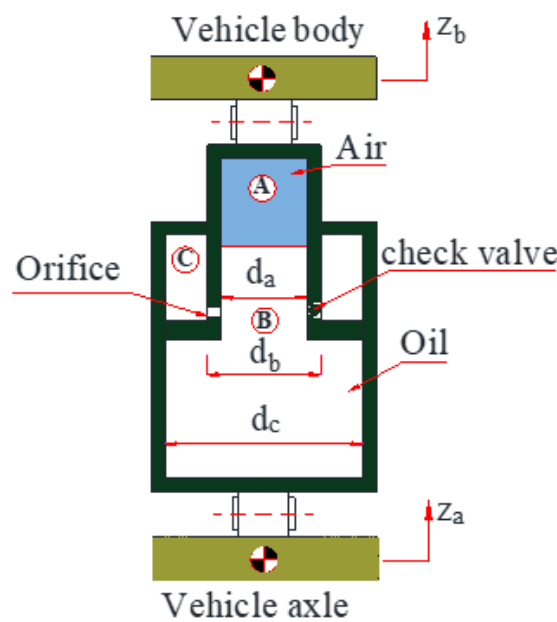


**Fig 1:** Structural schematic of HPS

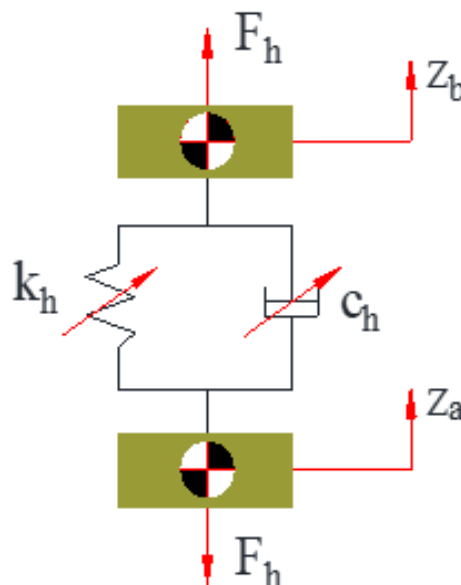


**Fig 2:** Dynamic model of HPS

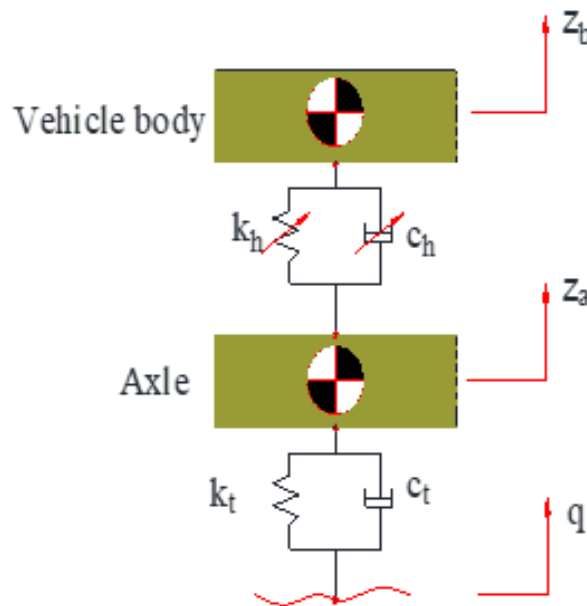


**Fig 3:** Dynamic model of 1/4 mining dump truck

The system consists of three main chambers: a gas chamber (A), a primary oil chamber (B), and an additional oil chamber (C), with corresponding pressures denoted as  $p_A$ ,  $p_B$ , and  $p_C$ , respectively. The geometric parameters of the HPS include the rod's inner diameter ( $d_a$ ), the rod's outer diameter ( $d_b$ ), and the cylinder's inner diameter ( $d_c$ ). Additionally, the HPS components incorporate an orifice and a check valve, characterized by areas  $A_1$  and  $A_2$ , respectively. The system's dynamic behavior is primarily influenced by the vertical displacements of the vehicle axle ( $z_a$ ) and the vehicle body ( $z_b$ ).

### 2.1. The vertical force of HPS

The vertical force ( $F_h$ ) is defined as the sum of the elastic force ( $F_k$ ), the damping force ( $F_c$ ) and the frictional force ( $F_f$ ).

$$F_h = F_k + F_c + F_f \quad (1)$$

The elastic force of the HPS is calculated based on the change in air pressure:

$$F_k = p_0 \left( \left( \frac{V_0}{V_0 + A_b(z_b - z_a)} \right)^n - 1 \right) A_b \quad (2)$$

Where,  $p_0$  is initial pressure of air chamber of HPS;  $V_0$  is initial volume of air chamber of HPS of front axle;  $V_a$  is the volume of gas at any position;  $n$  is the polytrophic rate.

The damping force is determined based on the pressure loss through the throttle hole and the friction between the rod and the cylinder:

$$F_c = - \left( \frac{\rho(A_a - A_b)^3(\dot{z}_b - \dot{z}_a)^2 \text{sign}(\dot{z}_b - \dot{z}_a)}{2C_d^2 \left( A_1 + \frac{(1 + \text{sign}(\dot{z}_b - \dot{z}_a))}{2} A_2 \right)^2} \right) \quad (3)$$

Where,  $\rho$  is the density of oil,  $C_d$  is coefficient of discharge through the hole.

The friction force between the rod and cylinder of the HPS is computed according to the wet lubrication theory <sup>[16]</sup>:

$$F_f = -\mu \frac{\pi d_c L}{h} (\dot{z}_b - \dot{z}_a) \quad (4)$$

In which:  $\mu$  is dynamic viscosity coefficient of the oil;  $L$  and  $h$  are diameter of rod and thickness of oil layer between rod and cylinder of HPS.

From Eq. (2), Eq. (3) and Eq. (4) the vertical force of HPS can be determined:

$$F_h = p_0 \left( \left( \frac{V_0}{V_0 + A_b(z_b - z_a)} \right)^n - 1 \right) A_b - \left( \frac{\rho(A_a - A_b)^3(\dot{z}_b - \dot{z}_a)^2 \text{sign}(\dot{z}_b - \dot{z}_a)}{2C_d^2 \left( A_1 + \frac{(1 + \text{sign}(\dot{z}_b - \dot{z}_a))}{2} A_2 \right)^2} \right) - \mu \frac{\pi d_c L}{h} (\dot{z}_b - \dot{z}_a) \quad (5)$$

## 2.2. Dynamic model of a 1/4 mining dump truck

A vibration model of a 1/4 mining truck with the HPS, consisting of two degrees of freedom, is developed to assess the impact of initial intake air pressure on the spring force and damping force, as illustrated in Fig. 3. In Fig. 3, the tire stiffness and damping coefficients are denoted as  $k_t$  and  $c_t$ , respectively; the stiffness and damping coefficients of the HPS are represented by  $k_h$  and  $c_h$ ; the vertical displacements of the axle and vehicle body are  $z_a$  and  $z_b$ ; and the road surface excitations are denoted as  $q$ . D'Alembert's principle is applied to formulate a system of differential equations that describe the vibrations of the mining dump truck. The differential equation governing the vibration of the vehicle body is:

$$m_b \ddot{z}_b = F_h \quad (6)$$

Vibration equation of the axle:

$$m_a \ddot{z}_a = F_t - F_h \quad (7)$$

Where,  $F_t$  is the vertical force of tires:

$$F_t = k_t(z_a - q) + c_t(\dot{z}_a - \dot{q}) \quad (8)$$

## 2.3. Road surface roughness

In this study, the random road surface function is considered as the input parameter for analyzing vehicle dynamics. According to the ISO 8068 standard [23], the roughness characteristics of the road surface are quantified by the spectral density, denoted as  $S_q(n_0)$  [24], which is defined as follows:

$$S_q(n) = S_q(n_0) \left( \frac{n}{n_0} \right)^{-\omega} \quad (9)$$

This roughness is modeled as a Gaussian random process and is generated using an inverse Fourier random variable:

$$q(t) = \sum_{i=1}^N \sqrt{\frac{2vn_0^2 S_q(n_0) \Delta f}{f_{mid,i}^2}} \cos(2\pi f_{mid,i} t + \phi_i) \quad (10)$$

In which:  $f_{mid,i} = f_i + \frac{2i-1}{2} \Delta f$ ; ( $i=1 \div N$ );  $\phi_i$  is the random phase of the magnetic distribution ( $0 \div \pi$ )

## 3. Results and discussion

In this study, a 1/4 vehicle oscillation model with a passive suspension system is chosen to simulate and analyze using a PID controller and a Fuzzy-logic controller to control the air pressure in the air chamber of the HPS to improve the ride comfort. The simulation results of displacement, vertical acceleration and pressure of the air chamber when the vehicle moves on the ISO E road surface, speed 30 km/h, full load state is shown in Fig. 4. From the results in Fig. 4, it shows that the amplitude values of  $z_b$  and  $a_b$  in the case with control are significantly reduced compared to the case without control and that of the Fuzzy controller is reduced more than that of the PID controller. The results also show that the maximum amplitude of the pressure value in the air chamber of the controlled suspension system is significantly reduced compared to the uncontrolled suspension system. From the simulation results, it also shows that the ride comfort is significantly improved with the controlled suspension system and the suspension system with the Fuzzy controller is better than the PID controller. From the results in Fig. 4, the RMS of acceleration of sprung mass ( $a_{wb}$ ) can be calculated according to ISO 2631(1997) standard through Eq. (11).

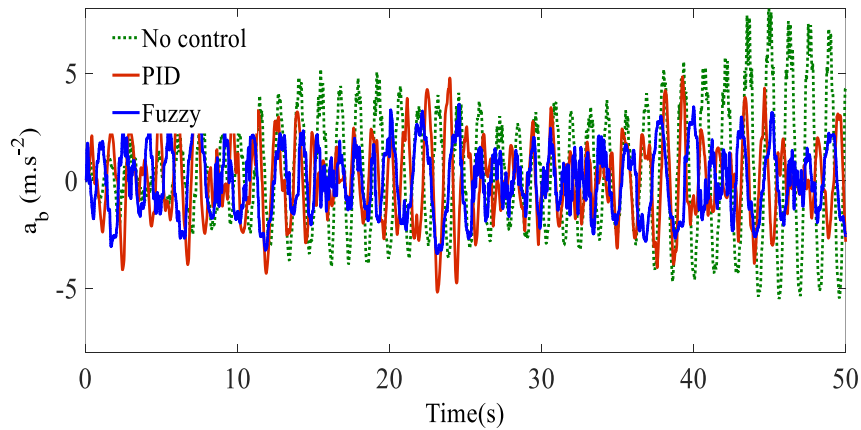
The vertical acceleration response of the vehicle body, expressed as the root-mean-square (RMS) value, is calculated in accordance with International Standard ISO 2631-1:1997 [25] as follows:

$$a_{wb} = \left[ \frac{1}{T} \int_0^T a_b^2(t) dt \right]^{0.5} \quad (11)$$

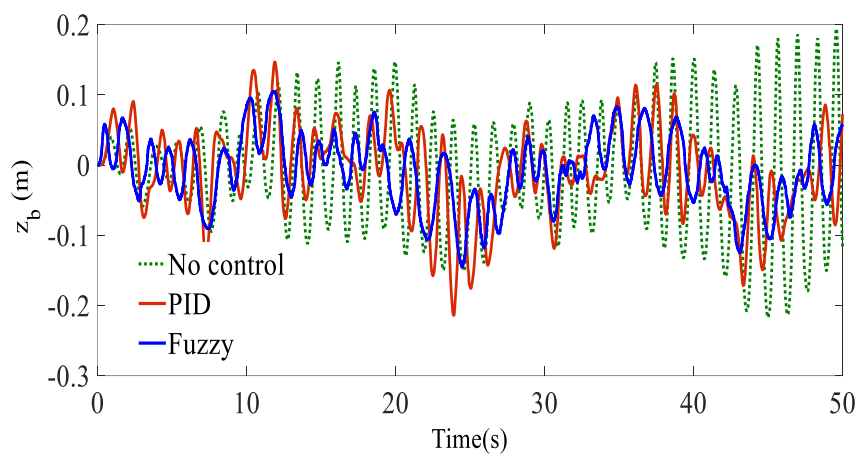
In which:  $a_b(t)$  represents the weighted acceleration as a function of time, measured in  $m/s^2$ , and  $T$  denotes the duration of the measurements.

Compared with the passive suspension system, the  $a_{wb}$  value of the suspension system with PID and Fuzzy controllers decreased by 18.39% and 27.55% respectively when the vehicle was moving on the ISO E-class road surface at a speed of 30 km/h in a

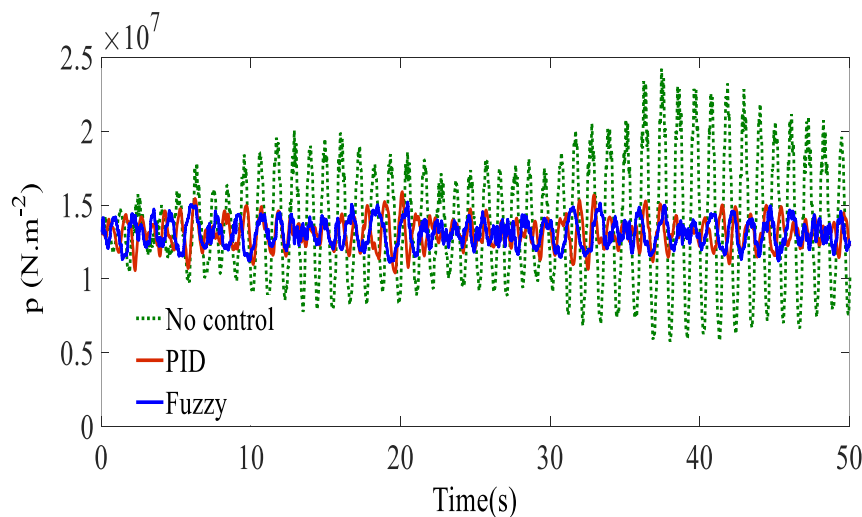
fully loaded state.



a) Acceleration of sprung mass



b) Displacement of sprung mass



c) Pressure of air chamber

**Fig 4:** Acceleration of sprung mass, displacement of sprung mass and pressure of air chamber

#### 4. Conclusions

In this study, the research team has established a mathematical model to determine the vertical force of the HPS, set up a PID controller and a Fuzzy controller to control the pressure in the air chamber to improve the ride comfort. The simulation results with a 1/4 mining dump truck model show the effectiveness of improving the ride comfort of the PID and Fuzzy controllers,

especially the Fuzzy controller. Compared to the passive suspension system, the  $a_{wb}$  value of the suspension system with PID and Fuzzy controllers decreased by 18.39% and 27.55%, respectively, when the vehicle traveled on an ISO E-class road surface at a speed of 30 km/h in a fully loaded condition.

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