



Improvement of ride performance of hydro-pneumatic suspension system of mining dump using PID and Fuzzy logic controllers

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Abstract

This paper proposes a mathematical model and control strategy for a hydro-pneumatic suspension system (HPS) aimed at enhancing ride comfort in a mining dump truck. This study investigates the effectiveness of Proportional-Integral-Derivative (PID) and Fuzzy Logic controllers in improving the performance of a semi-active hydro-pneumatic suspension (HPS) system. A dynamic model for a mining truck equipped with an HPS system is developed, and the vertical forces, including elastic, damping, and frictional forces, are calculated and is simulated by using MATLAB Simulink software. The system's performance is evaluated under different operating conditions. Simulation results demonstrate that the HPS, controlled by PID and Fuzzy logic controllers, significantly improves ride comfort, with reductions in the root mean square of acceleration (a_{wb}) values. The Fuzzy controller outperforms the PID controller, with improvements of up to 36.03% in poor road conditions and 31.77% at higher speeds.

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

1. Introduction

Vehicle suspension system plays a vital role in determining a vehicle's ride comfort and handling stability. A well-designed suspension ensures that the vehicle maintains effective contact with the road while minimizing body vibrations caused by road surfaces. Conventional passive suspension systems, while simple and cost-effective, operate with fixed damping characteristics. This limitation prevents them from adapting to different road conditions or dynamic changes in vehicle behavior, making it difficult to achieve an optimal compromise between comfort and handling performance. To overcome these shortcomings, semi-active suspension systems have been introduced. These systems offer a greater degree of adaptability by continuously adjusting the damping force in response to road surface variations and vehicle dynamics ^[1]. The ability to modulate damping characteristics in real time allows semi-active suspensions to significantly improve both ride comfort and road handling compared to passive systems. Recent research has focused on the application of advanced control strategies to enhance the performance of semi-active suspension systems. Among the most used techniques are Proportional-Integral-Derivative (PID) controllers and Fuzzy Logic controllers ^[2, 3]. PID controllers are particularly favored due to their simplicity, ease of implementation, and ability to maintain system stability under many operating conditions ^[3]. However, their performance is often compromised in the presence of nonlinearities and rapidly changing road conditions. PID controllers rely on a precise system model and fixed parameters, making them less effective when faced with time-varying disturbances. To address the limitations of PID control, fuzzy logic controllers have been introduced into the design of semi-active suspension systems. These controllers are capable of handling system nonlinearities and uncertainties more effectively than traditional methods ^[4, 5]. Fuzzy logic control does not require an exact mathematical model and instead utilizes a rule-based system that mimics human reasoning, making it more flexible and robust under a wide range of driving conditions. Hybrid control approaches that combine PID and fuzzy logic techniques have emerged as promising solutions that leverage the advantages of both strategies.

These hybrid controllers can deliver improved vibration suppression and enhanced ride comfort compared to using either control technique independently [6, 7]. The studies have demonstrated to complex dynamic behaviors and maintaining performance. Further advancements have been made through the development of adaptive and neural network-based fuzzy controllers. These approaches allow for the real-time adjustment of control parameters, enabling the suspension system to respond dynamically to changing conditions [8, 9]. Such intelligent control methods have shown great potential in further enhancing the performance and reliability of semi-active suspension systems. Building upon these recent developments, this research aims to explore the integration of Fuzzy Logic and PID control strategies for regulating air chamber pressure in a vehicle's suspension system. The primary objective is to improve ride comfort under diverse road conditions by optimizing suspension performance through intelligent, adaptive control methods [10-12]. In addition, the research team proposed the nonlinear models of passive HPSs to investigate their effects on efficiency [13-18]. This study focuses on the development of a Fuzzy Logic and PID controller to regulate air chamber pressure, thereby improving ride comfort.

2. Vehicle dynamic model

Fig. 1 illustrates the structural configuration of the hydro-pneumatic suspension (HPS). Based on this structural representation, a nonlinear dynamic model has been developed, incorporating the stiffness coefficient k_h and the damping coefficient c_h , as depicted in Fig. 2. The system comprises three primary chambers: the gas chamber (A), the main oil chamber (B), and an auxiliary oil chamber (C), with their respective pressures represented as p_A , p_B , and p_C .

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 system integrates an orifice and a check valve, characterized by areas A_1 and A_2 , respectively. The dynamic response of the system is predominantly influenced by the vertical displacements of the vehicle axle (z_a) and the vehicle body (z_b).

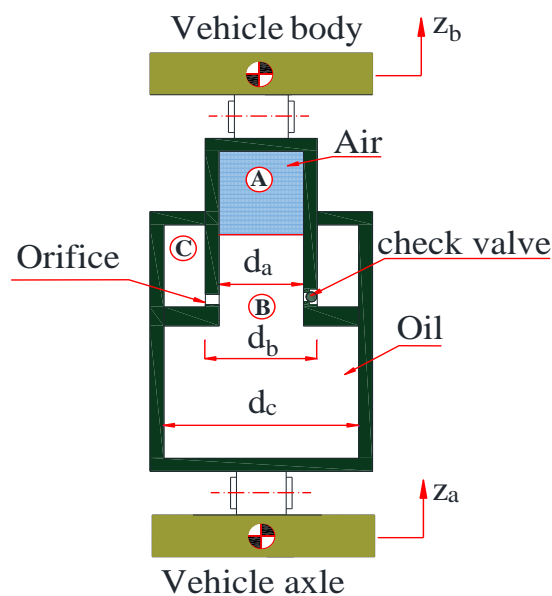


Fig 1: Structural schematic of HPS

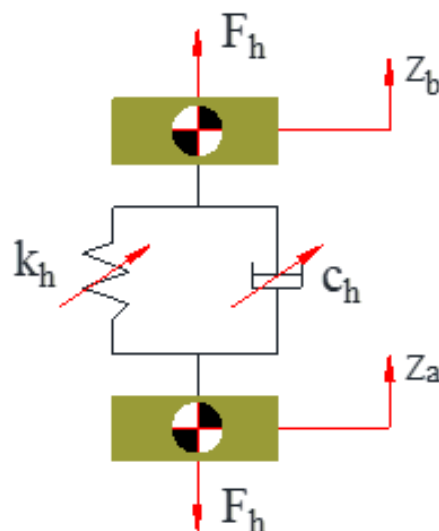


Fig 2: Dynamic model of HPS

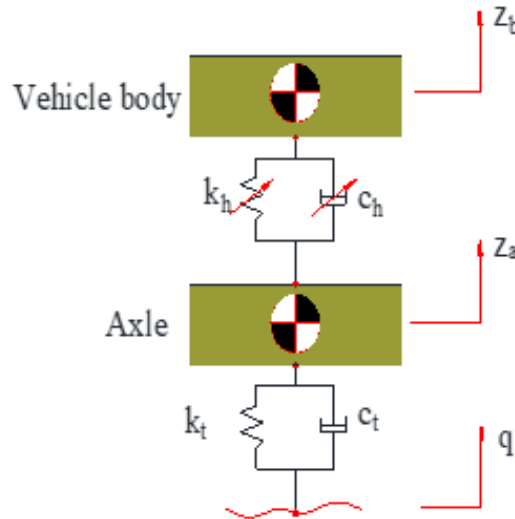


Fig 3: Dynamic model of 1/4 mining dump truck

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\$).

$$F_h = F_k + F_c + F_f \tag{1}$$

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

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

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:

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

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:

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

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:

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

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

A quarter-vehicle vibration model of a mining truck equipped with an HPS system is developed to evaluate the effects of initial intake air pressure on the elastic and damping forces. This model, which consists of two degrees of freedom, is illustrated in Fig. 3. In this figure, the tire stiffness and damping coefficients are denoted as \$k_t\$ and \$c_t\$, respectively.

To describe the vibration behavior of the mining truck, D'Alembert's principle is applied, leading to the formulation of a system

of differential equations. The equation governing the vibration of the vehicle body is:

$$m_b \ddot{z}_b = F_h \tag{6}$$

Vibration equation of the axle:

$$m_a \ddot{z}_a = F_t - F_h \tag{7}$$

Where, F_t is the vertical force of tires:

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

In this study, the random road surface function according to the ISO 8068 standard [19] is considered as the input parameter for analyzing vehicle dynamics.

2.3. PID and Fuzzy logic controller

Design a PID controller with the input signal being the displacement of the suspended mass (z_b), the output signal being the control pressure of the air chamber (p) with the objective function being the vertical acceleration of the suspended mass. Applying the Ziegler-Nichols method, the coefficients: K_p , K_i , K_d can be found.

Design a Fuzzy logic controller with two input signals, the displacement and vertical velocity of the suspended mass (z_b , v_b) and one output signal, the control pressure of the air chamber (p).

3. Results and discussion

To evaluate the improvement of the ride comfort of the two PID and Fuzzy logic controllers, different road surface conditions and speeds were simulated and calculated. The simulation results, comparing the root mean square of acceleration of sprung mass (a_{wb}) values through Eq. (9) when comparing the passive suspension system with the PID and Fuzzy logic controllers are shown in Fig. 4.

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 [20] as follows:

$$a_{wb} = \left[\frac{1}{T} \int_0^T a_b^2(t) dt \right]^{0.5} \tag{9}$$

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.

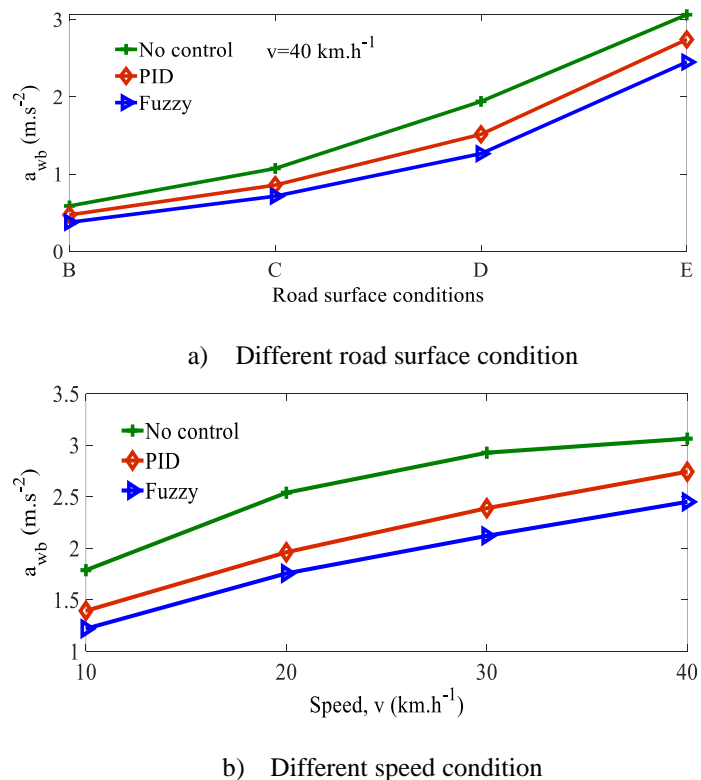


Fig 4: Values of a_{wb} under different operating conditions

Road surfaces from ISO class B to ISO class E (ISO 8068 standard) are selected for simulation analysis when the vehicle moves at 40 km/h, full load condition. The a_{wb} values when the vehicle moves on different road surfaces are shown in Fig. 4a. The results in Fig. 4a indicate that the a_{wb} values increase rapidly when the vehicle moves on poor road surfaces. When the vehicle moves from the road surface from ISO level B to ISO level E, the speed of 40 km/h compared to the passive suspension system, the a_{wb} value with the HPS with PID controller decreases by 19.64%, 20.03%, 21.9% and 10.47% respectively, with the suspension system with Fuzzy controller decreases by 36.03%, 33.36%, 34.8% and 20.01% respectively. The analysis results show that the road surface condition greatly affects the ride comfort. The simulation results when the vehicle moves on different road surfaces show that the effectiveness of the HPS with air chamber pressure control greatly improves the ride comfort when the vehicle moves on poor road surface conditions.

The speed values from 10 km/h to 40 km/h were selected to analyze the effect of the moving speed on the ride comfort when moving on the ISO class E road surface, fully loaded state. The a_{wb} values are shown in Fig. 4b. The results in Fig. 4b show that the a_{wb} value increases rapidly when the vehicle moves at a higher speed. The results also show that when the vehicle moves at different speeds, the ride comfort with the HPS with air chamber pressure control is improved very well compared to the passive suspension system and the suspension system with the Fuzzy controller is improved better than the PID controller but the difference is not much. When the vehicle moves from 10 km/h to 40 km/h, on the ISO class E road surface compared to the passive suspension system, the a_{wb} value with the suspension system with PID controller decreases by 22.07%, 22.74%, 18.39% and 10.47% respectively, with the suspension system with Fuzzy controller decreases by 31.77%, 30.82%, 27.55% and 20.01% respectively. The simulation results when the vehicle moves at different speeds show that the effectiveness of the HPS with air chamber pressure control also greatly improves the ride comfort.

4. Conclusions

In this research, a mathematical model was established to determine the vertical force of the HPS, and a PID and Fuzzy logic controller was set up to control the air chamber pressure to improve the ride comfort. Simulation results show that the PID and Fuzzy logic controllers significantly improve the ride comfort compared to the passive HPS under different operating conditions. The simulation results demonstrate that the road surface condition significantly impacts ride comfort, with the HPS showing substantial improvements over the passive suspension system. On poor road surfaces (ISO class E), the a_{wb} values decreased by 19.64%, 20.03%, 21.9%, and 10.47% for the PID-controlled HPS and 36.03%, 33.36%, 34.8%, and 20.01% for the Fuzzy-controlled HPS, compared to the passive system. Similarly, for varying speeds from 10 km/h to 40 km/h, the HPS with air chamber pressure control showed enhanced ride comfort, with the PID controller decreasing the a_{wb} values by 22.07%, 22.74%, 18.39%, and 10.47%, and the Fuzzy controller decreasing them by 31.77%, 30.82%, 27.55%, and 20.01%. These results highlight the effectiveness of the HPS with air chamber pressure control in improving ride comfort under different operating conditions.

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