The ride comfort optimization of mining dump truck based on eight degree of freedom

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Abstract. The working environment of mining dumping truck is poor and in order to improve the vibration reduction effect of mining dump trucks, a vehicle damping adjustable semi-active suspension based on connected hydro-pneumatic suspension is proposed. The 8 DOF model of hydro-pneumatic suspension and complete vehicle was established for mining dump truck. The fuzzy PID control method is used to optimize the smoothness of the vehicle by controlling the vertical acceleration of the truck, pitch angle acceleration and seat vertical acceleration. The damping performance of semi-active suspension is analyzed by Matlab/Simulink software. The simulation results show that compared with the passive suspension under the condition of random road power spectrum input, the root mean square values of the vertical acceleration of the truck and the vertical acceleration of the seat of the semi-active suspension are reduced by 25.76\% and 48.7\%. At the same time, the effect of vehicle vibration reduction is also obvious under the different speeds.

1. Introduction

As the vehicle for transporting mineral materials, the mining dump truck has advantages like high transport efficiency and large-scale load. The mining dump truck vehicle suspension is mainly used to attenuate the vibration caused by the impact of mine road surface, reduce the damage to mining dump truck itself, to meet the requirements of riding comfort of workers in different working conditions.

There have been many papers research on the vehicle multi DOF semi-active suspension, such as the study of energy-fed suspension and hybrid electromagnetic suspension [1, 2]. Zhang Jie established a two DOF model of vehicle active hydro-pneumatic suspension, LQG and fuzzy PID control strategy were combined to control the active hydro-pneumatic suspension and improve its performance [3]. Meng Xiaojie realized the canopy damping control for the magnetorheological damper semi-active seat suspension and the vibration dampening effect at the seat [4].

In this paper, Matlab was used to establish the eight DOF vehicle model and use fuzzy PID control to design connected hydro-pneumatic suspension system, and then the front and rear parts of the hydro-pneumatic springs are coupled separately. The vehicle signal is collected by the sensor and input it into the controller, and then operation current of the electro-hydraulic servo valve in different states is obtained by the control strategy of the controller. By controlling the opening and closing of the electro hydraulic servo, the external oil tank and oil pump can reasonably press in or release the hydraulic oil in the hydraulic cylinder, so that the semi-active oil and gas suspension can keep the best performance under different working conditions.
There FUZZY PID controllers are used in this paper. The first controller takes the vertical velocity and acceleration at the center of mass of the suspension as the control object. The second controller takes the vehicle body pitch angle velocity and acceleration as the control object. The third controller takes the speech and acceleration of the seat as the control object. The sum of the current output of the three controllers is used as the input of each hydro-pneumatic spring. In this way, the semi-active suspension can be controlled and the vehicle performance can be improved.

2. Vehicle model of connected hydro-pneumatic suspension

2.1. Eight DOF vehicle model
The eight DOF models were composed of four wheel vertical vibration, body pitch angle, vehicle roll angle, body moves vertically and the vertical movement of the seat. The following mathematical model has been established according to the vehicle model shown in Figure 1.

Figure 1. 8 degrees-of-freedom vehicle model.

\[
\begin{align*}
& m_1 \ddot{z}_1 + k_1 (z_1 - q_1) - k_3 (z_{t1} - z_1) - c_1 (\dot{z}_{t1} - \dot{z}_1) = 0 \\
& m_2 \ddot{z}_2 + k_2 (z_2 - q_2) - k_4 (z_{t2} - z_2) - c_2 (\dot{z}_{t2} - \dot{z}_2) = 0 \\
& m_3 \ddot{z}_3 + k_3 (z_3 - q_3) - k_5 (z_{t3} - z_3) - c_3 (\dot{z}_{t3} - \dot{z}_3) = 0 \\
& m_4 \ddot{z}_4 + k_4 (z_4 - q_4) - k_6 (z_{t4} - z_4) - c_4 (\dot{z}_{t4} - \dot{z}_4) = 0 \\
& m_5 \ddot{z}_5 + k_5 (z_{t5} + b\dot{\theta}_2) + c_5 (\dot{z}_{t5} + b\dot{\theta}_2) = 0 \\
\end{align*}
\]

\[
\begin{align*}
& m_b \ddot{\dot{z}}_b + k_1 (z_{t1} - z_1) + c_1 (\dot{z}_{t1} - \dot{z}_1) + k_2 (z_{t2} - z_2) + c_2 (\dot{z}_{t2} - \dot{z}_2) + k_3 (z_{t3} - z_3) + c_3 (\dot{z}_{t3} - \dot{z}_3) + k_4 (z_{t4} - z_4) + c_4 (\dot{z}_{t4} - \dot{z}_4) - k_5 (z_{t5} + b\dot{\theta}_2) - c_5 (\dot{z}_{t5} + b\dot{\theta}_2) = 0 \\
& L_1 \ddot{\dot{\theta}}_1 - [k_1 (z_{t1} - z_1) + c_1 (\dot{z}_{t1} - \dot{z}_1)] L_1 + [k_2 (z_{t2} - z_2) + c_2 (\dot{z}_{t2} - \dot{z}_2)] L_1 + [k_3 (z_{t3} - z_3) + c_3 (\dot{z}_{t3} - \dot{z}_3)] L_1 + [k_4 (z_{t4} - z_4) + c_4 (\dot{z}_{t4} - \dot{z}_4)] L_1 + [k_5 (z_{t5} + b\dot{\theta}_2) - c_5 (\dot{z}_{t5} + b\dot{\theta}_2)] a = 0 \\
& L_2 \ddot{\dot{\theta}}_2 - [k_1 (z_{t1} - z_1) + c_1 (\dot{z}_{t1} - \dot{z}_1)] L_2 + [k_2 (z_{t2} - z_2) + c_2 (\dot{z}_{t2} - \dot{z}_2)] L_2 + [k_3 (z_{t3} - z_3) + c_3 (\dot{z}_{t3} - \dot{z}_3)] L_2 + [k_4 (z_{t4} - z_4) + c_4 (\dot{z}_{t4} - \dot{z}_4)] L_2 + [k_5 (z_{t5} + b\dot{\theta}_2) - c_5 (\dot{z}_{t5} + b\dot{\theta}_2)] b = 0
\end{align*}
\]
Where: \( z_{t1} = z_b - L_2 \theta_1 - L_4 \theta_2 \), \( z_{t2} = z_b + L_4 \theta_1 - L_4 \theta_2 \), \( z_{t3} = z_b + L_1 \theta_1 - L_3 \theta_2 \), \( z_{t4} = z_b - L_2 \theta_1 + L_3 \theta_2 \), \( z_{t5} = z_5 - z_b - a \theta_1 \)

Where: \( m_1, m_2, m_3, m_4 \) are the mass of left front, right front, left rear and right rear suspension; \( z_1, z_2, z_3, z_4 \) are the vertical displacement at the left front, left rear, right front and right suspension; \( m_5, m_6 \) are the total mass of cab and body mass respectively; \( k_1, k_2, k_3, k_4 \) are the stiffness of front and rear suspension leaf springs; \( c_1, c_2, c_3, c_4 \) are the damping of front and rear suspension leaf springs respectively; \( I_1, I_2 \) are the roll angle and pitch moment of inertia respectively; \( \theta_1, \theta_2 \) are the angular displacement of body roll and pitch motion respectively; \( k_t_1, k_t_2, k_t_3, k_t_4 \) are the stiffness of each tire; \( q_1, q_2, q_3, q_4 \) are the corresponding road surface excitation of each tire; \( z_1, z_2, z_3, z_4 \) are the displacement at each supporting point and the center of mass of the truck body; \( a, b \) are the horizontal and longitudinal distances from the center of mass of the cab to the center of mass of the body.

2.2. **Hydro-pneumatic spring model**

According to the regulation, the mining dump truck is at static equilibrium state when unload. The downward movement of plunger is set as the positive movement. To the contrary, the upward one is negative. The stiffness force in the hydro-pneumatic spring is mainly produced by the compression of the gas in the hydro-pneumatic and the damping force is mainly produced by the viscosity between the oil and is realized by the pressure loss caused by the damping hole. Starting from the balanced position of the hydro-pneumatic spring, in the compression stroke, the gas is compressed, the check valve opens and operates. The hydraulic fluid in the rod-less cavity flows into the annular cavity from the check valve and the damping hole. When in the drawing stroke, the gas state is restored, the check valve is closed. The hydraulic fluid from the annular cavity flows into the rod less cavity for the damping hole.

**Stiffness characteristics of Hydro-pneumatic spring:** [5]

\[
F_K = P_m A_2 \left( \frac{L_m}{L_m - \frac{A_2}{A_1}} \right)^x
\]  

**Damping characteristics of hydro-pneumatic spring:**

\[
F_c = \frac{(A_1 - A_2)^2 \rho^2 \rho}{2 \left[ c_d A_2 + c_d A_d \left( \frac{1 + \frac{1}{2} \text{sgn}(v)}{2} \right) \right] \text{sgn}(v)}^2
\]

Where: \( F_k \) is the hydro-pneumatic spring stiffness force; \( P_m \) is gas pressure at full load; \( A_2 \) is the cross-sectional area of the piston rod; \( A_1 \) is the cross-sectional area of the air cavity; \( L_m \) is the height of the gas at full load equilibrium position; \( x \) is the relative displacement of piston rod rod cylinder; \( r \) is the variable coefficient of gas; \( F_c \) is the damping force of the hydro-pneumatic spring; \( \rho \) is the oil density; \( v \) is the relative speed of piston rod and cylinder; \( C_d \) is the flow coefficient; \( A_z \) is flow area of the damping hole; \( A_d \) is flow area of check valve.

In this paper, the road random excitation input model in time domain is establish based on the idea of filtering white noise method. Taking a white noise conforming to Gaussian distributing as the road random excitation input model, the road surface time domain model with single wheel input is obtained as:

\[
\dot{q}(t) = -2 \pi f_0 q(t) + 2 \pi \sqrt{G_q(n_0)} \dot{u}w(t)
\]

3. **Design of fuzzy PID controller**

3.1. **PID control**

PID control is the most commonly used controller. PID controller as a linear controller, the control deviation is formed by comparing the difference between the actual output value and the given value. The control deviation is processed by proportion, integration and differentiation, and an output value is given to the original control system, so as to play the role of optimization. The control law can be expressed as:
The transfer function is written as:

\[ G(S) = \frac{u(t)}{e(t)} = k_p (1 + \frac{1}{T_1 s} + T_D s) \]  

(13)

Where: \( k_p \) is the proportionality coefficient; \( T_1 \) is the integral time constant; \( T_D \) is the different time constant.

### 3.2. Fuzzy controller

Fuzzy PID control is composed of traditional PID controller and fuzzy controller in parallel, combining the advantages of the two controllers. The fuzzy control selects the actual output as the fuzzy control quantity, carries on the inference and the judgment of fuzzy rule, carrying on the defuzzification then output the final value.

The output of the three variables with the change of the input to achieve self-adjustment, output three values and the original PID parameters to add, constitute a fuzzy PID control.

\[ K_p = K_{po} + \Delta K_p \]
\[ K_i = K_{io} + \Delta K_i \]
\[ K_d = K_{do} + \Delta K_d \]  

(14)

Where: \( K_p, K_i, K_d \) are the final value of the ratio, integral and differential; \( K_{po}, K_{ip}, K_{do} \) are the initial value; \( \Delta K_p, \Delta K_i, \Delta K_d \) are the increment value.

In this model, the PID controller’s proportional, integral and differential values are adjusted adaptively in the form of two inputs and three outputs. After fuzzification, fuzzy approximate reasoning and anti-fuzzification, the modified value is input into the PID controller, and the PID controller parameter value is further adjusted, so as to optimize the smoothness.

Based on the established vehicle model simulation, the basic theory domains of the body vertical velocity and acceleration of the vertical controller obtained by simulation values are \([-2,2]\) and \([-22,30]\), the basic theoretical domains of the speed and acceleration of body pitch angle are \([-1,0.8]\) and \([-15,16]\), the basic theory domains of the vertical velocity and acceleration of car seats are \([-4,3]\) and \([-45,45]\). The fuzzy field of the selected fuzzy input and the output of \( K_p, K_i \) and \( K_d \) is \([-4,4]\).

### 4. Simulation analysis

According to ‘Road irregularities indication method protocol’ from International Standardization Organization document ISO/TC108/S C2N67 and ‘Mechanical vibration - Road surface profiles - Reporting of measured data’ from national standardized document GB/T7031-2005, the road irregularities is divided into eight levels with difference of pavement power spectral density in space frequency \([6, 7]\). In this paper, the road random excitation input model in time domain is establish based on the idea of filtering white noise method. Based on the established model, the road surface grade is selected as E-class road surface, and the vehicle speed is selected as 40Km/h for simulation and the simulation time is 10s. The fuzzy control rules are shown in Table 1. The comparison of suspension performance indexes is shown in Table 2.

| \( K_p \), \( K_i \), \( K_d \) | \( ke \) |
|-----------------|-----------------|
| NB   | NM   | NS   | ZO   | PS   | PM   | PB   |
| NB   | PB/NB/PS | PB/NB/NS | PM/NM/NB | PM/NM/NB | PS/NS/NB | PS/ZO/NM | ZO/ZO/PS |
| NM   | PB/NB/PS | PB/NB/NS | PM/NM/NB | PM/NM/NB | PS/NS/NM | PS/ZO/NS | ZO/NS/ZO |
| NS   | PM/NM/ZO | PN/NM/NS | PM/NS/NS | PM/NS/NS | PS/NS/NS | ZO/NS/NS | ZO/NS/NS |
| kec | ZO   | PS   | NS   | ZO   | NS   | PS   | PM   |
| PS   | PS/NS/NS | PS/NS/NS | ZO/NS/NS | PS/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS |
| PM   | ZO/NS/PB | ZO/NS/PB | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS |
| PB   | ZO/NS/PB | ZO/NS/PB | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS | ZO/NS/NS |

Table 1. Fuzzy control rules table.
Table 2. Comparison of suspension performance indexes.

|                         | Passive Suspension | Semi-active Suspension | Period |
|-------------------------|--------------------|------------------------|--------|
| Vertical Acceleration(m/s²) | 5.6123             | 4.1666                 | 25.76% |
| Elevation Acceleration(rad/m²) | 0.4260             | 0.2205                 | 48.24% |
| Acceleration At Roll Angle(rad/s²) | 2.9359             | 1.9903                 | 32.1%  |
| Seat Acceleration(m/s²)     | 11.3694            | 5.8253                 | 48.76% |

Maintain the E-class road surface and select the speed simulation of 30, 35, 40, 45, and 50Km/h. The root mean square values of each indicator were compared, as shown in Figure 2-Figure 5.

5. Conclusions
The simulation results show that the optimization effects of vehicle vertical acceleration, pitch angle acceleration, and seat vertical acceleration are 25.76%, 48.24%, and 48.76%, when the vehicle driving speed is 40Km/h under the condition of E-class road surfaces. The connected hydro-pneumatic suspension system based on fuzzy PID control can optimize the ride comfort of the vehicle at different speeds.

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