

Article

Design and Optimization of Hydropneumatic Suspension Simulation Test Bench with Electro-Hydraulic Proportional Control

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Abstract: Available hydropneumatic suspension simulation test benches have insufficient loading accuracy and limited functionality rendering them unsuitable for performance testing of heavy vehicles with this type of suspension. Therefore, a multi-functional compound simulation test bench was designed that used an electro-hydraulic proportional control technique. A mathematical model was established to describe the hydraulic loading system, and the transfer function of the system was derived. The gain and phase margins confirmed the stability of the system. A simulation model was established in the Simulink environment and step and sine signals of different frequencies were applied separately to analyze the dynamic characteristics of the system. The results showed that the system responded slowly and exhibited phase lag and signal distortion. The dynamic characteristics of the system were improved by incorporating an adaptive fuzzy PID controller. Simulation results showed that the response of the system to the step signal stabilized at the preset value within 0.3 s with no oscillation or overshoot. The improved system performed well in replicating the random vibrations of heavy vehicles operating on Class B and C roads. This confirmed that the system can satisfy the loading requirements of heavy vehicle hydropneumatic suspensions and can be used as a simulation test bench for such suspensions.

Keywords: hydropneumatic suspension; simulation test bench; electro-hydraulic proportional control; adaptive fuzzy PID controller; road vibration simulation



Citation: Wu, Z.; Jiao, B.; Sun, C.; Zhang, Y.; Zhao, H. Design and Optimization of Hydropneumatic Suspension Simulation Test Bench with Electro-Hydraulic Proportional Control. *Machines* **2023**, *11*, 907. <https://doi.org/10.3390/machines11090907>

Academic Editors: Min Cheng, Qun Chao and Ruqi Ding

Received: 17 July 2023

Revised: 1 September 2023

Accepted: 7 September 2023

Published: 13 September 2023



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1. Introduction

A hydropneumatic suspension is an integration of elastic elements and dampers, using an inert gas stored in an accumulator as the elastic medium and hydraulic oil as the load transfer medium of the dampers [1,2]. It has the functions of balancing load, generating damping vibration, and improving vehicle attitude [3–5]. Hydropneumatic suspension can meet a variety of driving conditions and improve vehicle ride comfort and handling stability; hence, has been widely used in heavy vehicles that operate under varied road conditions [6–8]. The performances of hydropneumatic suspension are generally tested by simulation using a test bench. The test bench loading method can significantly affect its ability to replicate the expected load or road conditions [9].

Since the end of the last century, some countries have carried out research on suspension test benches. Mechanical Testing & Simulation Company (MTS, in Eden, Minnesota, United States) and General Motors Company (GM, in Detroit, Michigan, United States) collaborated to invent the first vehicle road simulator [10]. Staiger GmbH & Co (in Stuttgart, Germany) developed a spindle-coupled suspension test bench [11]. By applying force

and motion in six degrees of freedom through vehicle spindles, the MTS 329 test bench (MTS, in Eden, Minnesota, United States) can conduct several different tests to replicate real-world road loads on vehicles and their subsystems [12,13]. The research on suspension test benches in China started relatively late and has developed rapidly in recent years. Its loading methods have mainly experienced mechanical, electro-hydraulic servo, etc. [14]. Chen et al. [15] designed a vertical loading test bench with adjustable hard points that output several vertical vibration signals such as step and sine signals and integral white noise. Yang [16] developed a bench for testing the kinematic and compliance characteristics of vehicle suspension in a single-axis direction that applied load using a servo motor and a ball screw. Si [17] designed a vehicle suspension test bench that applied mechanical loads whose amplitude and frequency were controllable within the respective ranges and preliminarily discussed the application of the test bench. Zhao et al. [18] and Qin [19] designed a combined control suspension test bench using a servomechanism with a feedforward interference compensator and a system-optimized feedback controller. Liu et al. [20] designed a high-precision electro-hydraulic servo suspension test bench using a fuzzy, self-tuning proportional integrative derivative (PID) controller. Du et al. [21] realized road simulation iteration and durability tests using RPC simulation iteration and other technologies and verified relevant requirements for suspension road simulation test benches. Zhang [22] developed a hydropneumatic suspension simulation system using a three-state control strategy and designed the controller parameters using the pole placement method, thereby increasing the bandwidth and improving the dynamic characteristics of the system while maintaining its stability.

The above literature review shows that the loading methods used for existing test benches mainly include mechanical and electro-hydraulic servo loading methods. Mechanical loading test bench output waveform is not accurate, with large errors and other shortcomings, and thus is unsuitable for performance testing of hydropneumatic suspensions. The electro-hydraulic servo loading test bench is accurate; however, it is expensive and requires stringent environmental conditions, limiting its application [23,24]. Therefore, a simulation test bench needs to be designed that is cost-effective, reliable, functional, and suitable for analyzing the performance of hydropneumatic suspensions for heavy vehicles. Because the performance of electro-hydraulic proportional control technology meets the test requirements, the operation of its control system is relatively simple, and the cost of electro-hydraulic proportional control system is lower than servo control, and the requirement for oil cleanliness is low, which greatly reduces the system failure and makes the loading system more stable and reliable [25]. Therefore, the electro-hydraulic proportional control method is selected to complete the loading of the test bench.

In this paper, a multi-functional compound hydropneumatic suspension test bench was designed based on the electro-hydraulic proportional control technology, and the schematic diagram of the hydraulic loading system was completed. A mathematical model was established to describe the hydraulic loading system, and the transfer function of the system was derived. A simulation model was established in the Simulink environment and step and sine signals of different frequencies were applied separately to analyze the dynamic characteristics of the system. The results showed that the system responded slowly and exhibited phase lag and signal distortion. The fuzzy control algorithm was added on the basis of the conventional PID controller, and the fuzzy PID controller of the electro-hydraulic proportional system was established. The dynamic characteristics of the system were improved by incorporating the fuzzy PID controller. Simulation results showed that the response of the system to the step signal stabilized at the preset value within 0.3 s with no oscillation or overshoot. The improved system performed well in replicating the random vibrations of heavy vehicles operating on Class B and C roads. This confirmed that the system can meet the requirements of the hydropneumatic suspension test and verified the correctness of the mathematical model and the feasibility of the test bench design.

2. Simulation Test Bench

2.1. Functional and Performance Parameters

The simulation test bench was used to determine several performance characteristics of hydropneumatic suspensions such as displacement, velocity, stiffness, damping, and durability. In addition, it could be used for a 1/4 vehicle road simulation test. The key performance parameters of the test bench were 5 t maximum load, ± 120 mm maximum stroke, 0–30 Hz excitation frequency, and 0.5 m/s maximum velocity of the piston rod [26–29].

2.2. Mechanical Platform

The mechanical platform comprises a base, vertical circular columns, upper and lower cross beams, upper and lower sliders, a sliding lock, weight plates, and transducers. Hydropneumatic suspensions, wheels, and other heavy vehicle components necessary for performance analysis and 1/4 vehicle road simulation test can be mounted at different positions, making the test bench capable of several different types of simulation, as shown in Figure 1.

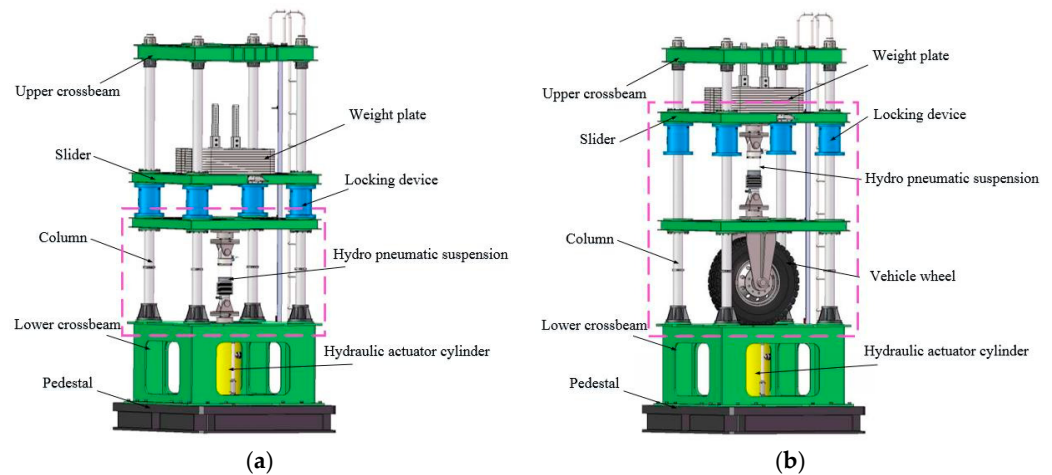


Figure 1. Mechanical platform of simulation test bench. (a) Hydropneumatic suspension performance analysis; (b) 1/4 vehicle road simulations.

Figure 1a illustrates the setup for hydropneumatic suspension performance analysis. The working principle is as follows. The sliding lock is locked. The hydropneumatic suspension is mounted between the lower slider and the lower cross beam and in direct contact with the hydraulic cylinder. The upper slider is loaded with weights on the top and presses downward. The hydraulic loading system drives the hydraulic cylinder, which outputs a load simulating the road vibration on the hydropneumatic suspension. A force transducer measures the total output force from the hydropneumatic suspension. A displacement transducer measures the relative displacement between the barrel and piston rod of the hydraulic cylinder, with the measurements fed back to the control loop. The displacement and output force measurements enable the displacement characteristics of the hydropneumatic suspension to be determined. The displacement measurements are recorded against elapsed time to determine the velocity characteristics of the hydropneumatic suspension.

Figure 1b illustrates the setup for the 1/4 vehicle road simulation test. The working principle is as follows. The sliding lock is unlocked. The wheel is mounted between the lower cross beam and lower slider. The hydropneumatic suspension is mounted between the two sliders. The upper slider is loaded with weights and pressed downward to simulate the sprung mass. During the simulation, the hydraulic loading system provides the hydraulic cylinder with pressure and flow to drive the tire, hydropneumatic suspension, and weights to vibrate simultaneously. A computer provides electrical signals to simulate different road vibrations on the tire, thereby simulating the real operating conditions of the vehicle. The displacement and tensile/compressive force transducers

record and output the measurements to complete the 1/4 vehicle road simulation of the hydropneumatic suspension.

2.3. Hydraulic Loading System

The hydraulic loading system comprises a hydraulic power supply, an electro-hydraulic proportional valve, a hydraulic cylinder, and an accumulator. Its main function is to shake the test bench to simulate road vibrations. Figure 2 shows a schematic diagram of the system. The electro-hydraulic proportional directional valve is the key component. It controls the direction of flow of the hydraulic oil and proportionally controls its flow depending on the input electrical signals received that control both the direction and rate of flow.

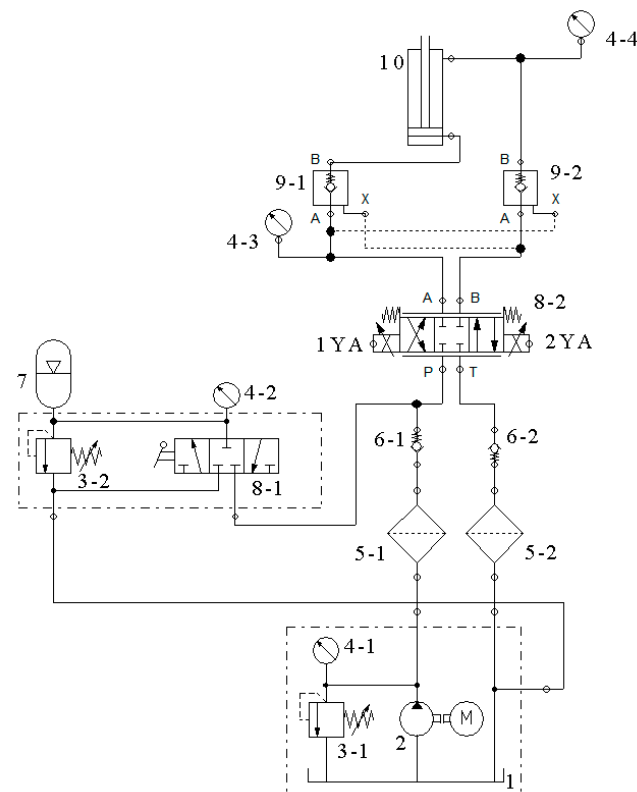


Figure 2. Schematic diagram of the hydraulic loading system with electro-hydraulic proportional control. Key: 1: oil tank, 2: hydraulic pump and electric motor, 3: relief valves, 4: pressure gauges, 5: filters, 6: hydraulic check valves, 7: accumulator, 8: directional control valves (8-1 manual, 8-2 electro-hydraulic), 9: pilot hydraulic control check valves, 10: hydraulic cylinder.

The working process of the hydraulic loading system is the motor drives the hydraulic pump 2 to work, and the hydraulic oil flows through the filter 5-1 and the hydraulic check valve 6-1 to enter the hydraulic system. At the same time, manually control the directional valve 8-1 to the left position, and some of the oil enters the accumulator for storage. When the input electrical signal is positive, the left coil 1YA of the electro-hydraulic proportional directional valve 8-2 is energized, and the oil flows into the rod cavity of the hydraulic cylinder 10 through the electro-hydraulic proportional directional valve 8-2 and the pilot hydraulic check valve 9-1. The velocity of flow is proportional to the electrical signal. At this time, pilot hydraulic check valve 9-2 is open, and the oil from the piston cavity of hydraulic cylinder 10 flows through hydraulic check valve 6-2 and oil filter 5-1 to return to oil tank 1. When the input electrical signal is negative, the right coil 2YA of the electro-hydraulic proportional directional valve 8-2 is energized, and the oil flows into the piston cavity of the hydraulic cylinder 10. The velocity of flow is proportional to the electrical signal, and the oil from the piston cavity of the hydraulic cylinder 10 returns to the oil tank 1. When a shutdown command is issued, the electro-hydraulic proportional directional

valve 8-2 stops the input of external electrical signals, and the right coil 2YA continues to be energized. The oil flows into the piston cavity of hydraulic cylinder 10, and its velocity is proportional to the electrical signal. The oil in the piston cavity of hydraulic cylinder 10 returns to the oil tank 1. Manually control the directional valve 8-1 in the right position to relieve pressure from accumulator 7. Subsequently, the electrohydraulic proportional directional valve 8-2 was powered off, the valve core returned to the neutral position, the motor stopped running, and the system completed the shutdown.

2.4. Parameter Calculation and Selection of Key Components

2.4.1. Hydraulic Cylinder

The load of the hydraulic cylinder is mainly caused by the weight of the vehicle body, suspension, and tires. The maximum load of this simulation test bench is 5 t. In order to leave a certain margin, a safety factor of 1.2 is taken, and the maximum load of the hydraulic cylinder is calculated to be 60,000 N. Considering the size specifications of the simulation test bench, it is possible to install a larger hydraulic cylinder. The hydraulic cylinders selected in this paper have the dimensions and performance parameters shown in Table 1. The thrust of the hydraulic cylinder is provided by the piston cavity, and the calculated thrust is 73,594 N, which is greater than 60,000 N, thus meeting the design requirements.

Table 1. Hydraulic cylinder dimensions and performance parameters.

Maximum Stroke	Piston Diameter (d_1)	Piston Rod Diameter (d_2)	Rated Pressure	Maximum Pressure
± 120 mm	125 mm	60 mm	6 MPa	15 MPa

2.4.2. Hydraulic Pump

When hydraulic oil is injected into the piston cavity of the hydraulic cylinder, it is the push stroke, and when hydraulic oil is injected into the rod cavity, it is the backward stroke. The speed of the piston rod during the push stroke is always lower than the speed of the backward stroke. The maximum speed of the piston rod occurs when oil is injected into the rod cavity. The maximum speed of piston rod movement v_m is 0.5 m/s. Therefore, the maximum flow of the hydraulic cylinder q_m is written as:

$$q_m = \frac{\pi}{4} (d_1^2 - d_2^2) \cdot v_m \quad (1)$$

Taking $d_1 = 125$ mm, $d_2 = 60$ mm, and $v_m = 0.5$ m/s into Equation (1), the maximum flow of the hydraulic cylinder is calculated to be 4.68×10^{-3} m³/s, which is 280 L/min.

Based on the above calculation results, the hydraulic pump parameters selected in this paper are shown in Table 2. The maximum flow of the hydraulic pump is 326 L/min, which is greater than 280 L/min, which meets the design requirements.

Table 2. Hydraulic pump parameters.

Displacement	Maximum Speed	Rated Power	Rated Pressure	Maximum Pressure
250 mL/r	1450 rpm	45 kW	35 MPa	40 MPa

2.4.3. Electro-Hydraulic Proportional Valve

According to the design and calculation results of the hydraulic cylinder, the maximum flow q_m of the hydraulic cylinder is 280 L/min. Therefore, the flow of the electro-hydraulic proportional valve is [30]:

$$q_v = q_m \sqrt{\frac{\Delta p_1}{\Delta p_v}} \quad (2)$$

where q_v is the working flow of the electro-hydraulic proportional valve; q_m is the maximum flow of the hydraulic cylinder; Δp_1 is the pressure drop at the proportional valve port when the hydraulic cylinder is working, taken as 1 MPa; Δp_v is the nominal pressure drop of the proportional valve, taken as 0.9 MPa.

According to Equation (2), the working flow of the electro-hydraulic proportional valve is 295 L/min. The parameters of the electro-hydraulic proportional valve selected in this paper are shown in Table 3.

Table 3. Parameters of electro-hydraulic proportional valve.

Valve Weight	Rated Flow	Maximum Flow	Drift Diameter	Current Control Signal	Maximum Pressure
11.5 kg	300 L/min	450 L/min	16 mm	4–20 mA	31.5 MPa

3. Electro-Hydraulic Proportional Control System

3.1. System Composition and Working Principle

The electro-hydraulic proportional control system comprises a proportional amplifier, an electro-hydraulic proportional directional control valve, a hydraulic cylinder, and a displacement transducer. It controls the waveform of the output from the hydraulic loading system. Figure 3 shows a block diagram of the control system. The working principle is as follows. The external electrical signal input to the controller is proportionally amplified and input to the electro-hydraulic proportional directional control valve to adjust the rate of flow, thereby controlling the displacement of the hydraulic cylinder piston rod. To improve the control sensitivity, the measurements of the displacement of the piston rod recorded by the displacement transducer are converted to electrical signals and fed back to the input of the controller. The control system determines the difference between the displacement measurements and the input signal and adjusts the input signal to reduce that difference, achieving closed-loop control of the displacement.

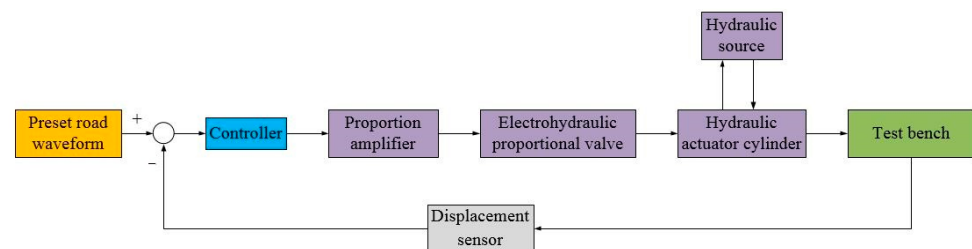


Figure 3. Block diagram of electro-hydraulic proportional control system.

3.2. Proportional Amplifier

The proportional amplifier converts the input signal from voltage to current. This proportional control element can be modeled using the following transfer function [31]:

$$G_a(s) = K_a = \frac{I(s)}{U(s)} \quad (3)$$

where I and U are the output current (A) from and input voltage (V) to the proportional amplifier, respectively; K_a is the gain (A/V) of the proportional amplifier.

3.3. Electro-Hydraulic Proportional Directional Control Valve

The electro-hydraulic proportional directional control valve positions the proportional electromagnet depending on the input current signal, thereby adjusting the opening sizes of the valve ports and controlling the rates of flow into and out of the hydraulic cylinder. This control element converts electrical signals to hydraulic signals [32]. Usually, there is a

median dead zone (5–10%) in the electro-hydraulic proportional directional valve, which is the main cause of system nonlinearity. In order to simplify and facilitate modeling, this article ignores the dead zone characteristics of the electro-hydraulic proportional directional valve. So, it can be modeled using the following second-order transfer function [33]:

$$G_{sv}(s) = \frac{Q_L(s)}{I(s)} = \frac{K_{sv}}{\frac{s^2}{\omega_{sv}^2} + \frac{2\zeta_{sv}s}{\omega_{sv}} + 1} \quad (4)$$

where Q_L is the output flow (m^3/s) from the electro-hydraulic proportional directional control valve, K_{sv} is the flow current gain [$\text{m}^3/(\text{s}\cdot\text{A})$], ω_{sv} is the natural frequency of the control element (rad/s), and ζ_{sv} is the damping ratio of the control element, which is assigned a value of $\zeta_{sv} = 0.6$.

3.4. Hydraulic Cylinder

Figure 4 shows a schematic diagram of the hydraulic cylinder. Here, p_s is the oil supply pressure; p_0 is the return pressure, which is assigned a value of zero ($p_0 = 0$); x_v is the displacement of the spool of the electro-hydraulic proportional valve; p_1 , q_1 , and A_1 are the pressure, flow, and effective area of the piston cavity, respectively; p_2 , q_2 , and A_2 are the pressure, flow, and effective area of the rod cavity, respectively; y is the displacement of the hydraulic cylinder piston rod, with the extensional displacement indicated by a positive sign.

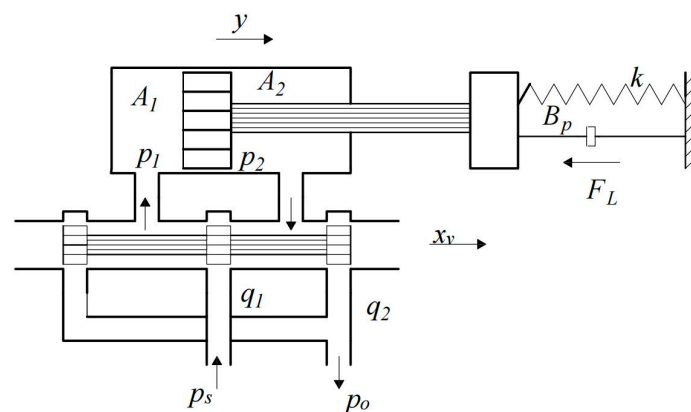


Figure 4. Schematic diagram of valve-controlled hydraulic cylinder.

3.4.1. Linear Flow Equation

When modeling a valve-controlled hydraulic cylinder system, two very important parameters need to be involved: load pressure p_L and load flow q_L [34]. When the hydraulic cylinder piston rod extends, the output force from the piston rod F_L is:

$$F_L = p_1 A_1 - p_2 A_2 \quad (5)$$

Let $F_L = p_L A_1$, where p_L is the load pressure; Equation (5) can then be rewritten as:

$$p_L A_1 = p_1 A_1 - p_2 A_2 \quad (6)$$

Which can be transformed as:

$$p_L = \frac{p_1 A_1 - p_2 A_2}{A_1} \quad (7)$$

Let $n = A_2/A_1$; then, Equation (7) can be rewritten as:

$$p_L = p_1 - n p_2 \quad (8)$$

When the piston rod extends, the flow in the piston and rod cavities are [35]:

$$q_1 = C_d \omega x_v \sqrt{\frac{2}{\rho} (p_s - p_1)} \quad (9)$$

$$q_2 = C_d \omega x_v \sqrt{\frac{2}{\rho} p_2} \quad (10)$$

According to the working principle of the hydraulic cylinder, when the piston rod moves, the flow velocities in the two cavities are equal, and the flow in the two cavities is proportional to the respective effective as:

$$\frac{q_2}{q_1} = \sqrt{\frac{p_2}{p_s - p_1}} = \frac{A_2}{A_1} = n \quad (11)$$

By combining Equations (8) and (11) and replacing p_1 and p_2 with p_s and p_L , respectively, the following equations can be derived:

$$p_1 = \frac{n^3 p_s + p_L}{1 + n^3} \quad (12)$$

$$p_2 = \frac{n^2 (p_s + p_L)}{1 + n^3} \quad (13)$$

Because of the difference in the area between the rod and piston cavities of a single-rod hydraulic cylinder, the incoming and outgoing flow of the hydraulic cylinder are not equal. The load flow q_L is defined as the average of the flow in the piston and rod cavities:

$$q_L = \frac{q_1 + q_2}{2} \quad (14)$$

By substituting Equations (9) and (10) into Equation (14), the following equation can be derived:

$$q_L = \frac{C_d \omega x_v}{2} \left(\sqrt{\frac{2}{\rho} (p_s - p_1)} + \sqrt{\frac{2}{\rho} p_2} \right) \quad (15)$$

By substituting Equations (12) and (13) into Equation (15) and replacing p_1 and p_2 with p_s and p_L , respectively, the following equations can be derived:

$$q_L = C_d \omega x_v \frac{1 + n}{2\sqrt{1 + n^3}} \left(\sqrt{\frac{2}{\rho} (p_s - p_L)} \right) \quad (16)$$

Thus, the linear flow equation of the proportional valve can be obtained:

$$q_L = K_q x_v - K_c p_L \quad (17)$$

where K_q is the flow gain (m^2/s):

$$K_q = \frac{\partial q_L}{\partial x_v} = C_d \omega \frac{1 + n}{2\sqrt{1 + n^3}} \sqrt{\frac{2}{\rho} (p_s - p_L)} \quad (18)$$

and K_c is the flow-pressure coefficient [$\text{m}^5/(\text{Pa} \cdot \text{s})$]:

$$K_c = \frac{\partial q_L}{\partial p_L} = -C_d \omega x_v \frac{1 + n}{2\sqrt{1 + n^3}} \sqrt{\frac{2}{\rho} \frac{1}{2\sqrt{p_s - p_L}}} \quad (19)$$

3.4.2. Flow Continuity Equation

The flow in the piston and rod cavities can be described using the following continuity equation [36]:

$$\begin{cases} q_1 = C_{ip}(p_1 - p_2) + C_{ep}p_1 + \frac{V_1}{\beta_e} \frac{dp_1}{dt} + A_1 \frac{dy}{dt} \\ q_2 = C_{ip}(p_1 - p_2) - C_{ep}p_2 + \frac{V_2}{\beta_e} \frac{dp_2}{dt} + A_2 \frac{dy}{dt} \end{cases} \quad (20)$$

where C_{ip} and C_{ep} are the internal and external leakage factors [$\text{m}^5/(\text{N}\cdot\text{s})$], respectively, of the hydraulic cylinder; V_1 and V_2 are the volumes (m^3) of the piston and rod cavities, respectively; β_e is the elastic modulus of the hydraulic oil (Pa); and y is the displacement (m) of the piston.

The working volumes of the piston and rod cavities are:

$$V_1 = V_{10} + A_1 y \quad V_2 = V_{20} - A_2 y \quad (21)$$

where V_{10} and V_{20} are the initial volumes (m^3) of the piston and rod cavities, respectively.

When the piston is at the midpoint of the hydraulic cylinder, the compressibility of the hydraulic oil is most affected, and the stability of the system is the lowest. In other words, the system is relatively safe when the piston is at any position other than the mid-point. Therefore, the mid-point of the hydraulic cylinder is defined as the initial position of the piston.

$$V_{10} = V_{20} = \frac{V}{2} \quad (22)$$

where V is the total volume (m^3) of the hydraulic cylinder.

According to the characteristics of the hydraulic cylinder, the following equation can be derived:

$$V_2 = \frac{V - V_1}{n} \quad (23)$$

By combining Equations (12)–(15) and (18)–(21), the load flow in the hydraulic cylinder satisfies the following continuity equation:

$$q_L = \left(C_{ip} + \frac{1}{2} C_{ep} \right) \left[\frac{n^2(n-1)}{1+n^3} p_s + \frac{1+n^2}{1+n^3} p_L \right] + \frac{1}{2\beta_e} \frac{(1-n^3)V_1 + n^3V}{1+n^3} \frac{dp_L}{dt} + \frac{1}{2} (A_1 + A_2) \frac{dy}{dt} \quad (24)$$

where

$$C_{ie} = \left(C_{ip} + \frac{1}{2} C_{ep} \right) \frac{1+n^2}{1+n^3} \quad (25)$$

$$C_f = \left(C_{ip} + \frac{1}{2} C_{ep} \right) \frac{n^2(n-1)}{1+n^3} \quad (26)$$

and V_1 is the equivalent volume:

$$V_t = 2 \frac{(1-n^3)V_1 + n^3V}{1+n^3} \quad (27)$$

By designating $A_m = (A_1 + A_2)/2$, Equation (24) can be rewritten as:

$$q_L = C_{ie}p_L + C_f p_s + \frac{1}{4\beta_e} V_t \frac{dp_L}{dt} + A_m \frac{dy}{dt} \quad (28)$$

3.4.3. Load Equilibrium Equation

When the piston rod extends, the output force from and load on the hydraulic cylinder satisfy the following equilibrium Equation:

$$p_L A_1 = m_t y + B_p y + K_p y + F_{ex} \quad (29)$$

where m_t is the total mass (kg) of the piston and the converted load on the piston, B_p is the damping coefficient [N/(m·s)] of the piston and load, K_p is the stiffness of the load spring (m/s), and F_{ex} is the external load on the piston.

3.4.4. Output Displacement Equation

By performing the Laplace transform to Equations (17), (28), and (29), the following Equation can be obtained:

$$\begin{cases} Q_L(s) = K_q X_v(s) - K_c P_L(s) \\ Q_L(s) = C_{ie} p_L(s) + C_f p_s(s) + \frac{s}{4\beta_e} V_t p_L(s) + s A_m Y(s) \\ P_L(s) A_1 = s^2 m_t Y(s) + s B_p Y(s) + K_p Y(s) + F_{ex}(s) \end{cases} \quad (30)$$

By defining $K_{ce} = K_c + C_{ie}$ as the system flow coefficient and canceling the intermediate variables $P_L(s)$ and $Q_L(s)$, the output displacement of the hydraulic cylinder piston rod $[Y(s)]$ can be expressed as:

$$Y(s) = \frac{A_1 K_q X_v(s) - C_f p_s(s) A_1 - \left(K_{ce} + \frac{V_t}{4\beta_e} s \right) F_{ex}(s)}{\frac{V_t}{4\beta_e} m_t s^3 + \left(K_{ce} m_t + \frac{V_t}{4\beta_e} B_p \right) s^2 + \left(K_{ce} B_p + \frac{V_t}{4\beta_e} K_p + A_1 A_m \right) s + K_{ce} K_p} \quad (31)$$

By dividing both the numerator and denominator by $A_1 A_m$, Equation (31) can be transformed as:

$$Y(s) = \frac{\frac{K_q X_v(s)}{A_m} - \left(\frac{1}{K_{cem}} + \frac{1}{K_y} s \right) F_{ex}(s)}{\frac{m_t}{K_y} s^3 + \left(\frac{m_t}{K_{cem}} + \frac{B_p}{K_y} \right) s^2 + \left(\frac{B_p}{K_{cem}} + \frac{K_p}{K_y} + 1 \right) s + \frac{K_p}{K_{cem}}} \quad (32)$$

where $K_y = (4\beta_e A_1 A_m) / V_t$ is defined as the stiffness of the hydraulic spring, i.e., the stiffness of the compressed liquid in the piston and rod cavities; $K_{cem} = (A_1 A_m) / K_{ce}$.

The load on the hydraulic cylinder of the loading system of the simulation test bench is dominated by inertial load, and the spring load and resistance load are negligible. Thus, Equation (32) can be simplified as:

$$Y(s) = \frac{\frac{K_q X_v(s)}{A_m} - \left(\frac{1}{K_{cem}} + \frac{1}{K_y} s \right) F_{ex}(s)}{s \left(\frac{s^2}{\omega_p^2} + \frac{2\zeta_p s}{\omega_p} + 1 \right)} \quad (33)$$

where ω_p is the hydraulic natural frequency (rad/s), which satisfies the following equation:

$$\omega_p = \sqrt{\frac{K_y}{m_t}} \quad (34)$$

and ζ_p is the hydraulic damping ratio, which satisfies the following equation:

$$\zeta_p = \frac{1}{2} \left(\frac{\sqrt{K_y m_t}}{K_{cem}} + \frac{B_p}{\sqrt{K_y m_t}} \right) \quad (35)$$

3.4.5. Transfer Function

The first term of the numerator in Equation (33) is the output displacement from the hydraulic cylinder controlled by the electro-hydraulic proportional directional control valve. Using the displacement of the valve spool as the input, the transfer function can be expressed as:

$$G_{LSD}(s) = \frac{Y_S(s)}{X_v(s)} = \frac{K_q}{s \left(\frac{s^2}{\omega_p^2} + \frac{2\zeta_p s}{\omega_p} + 1 \right) A_m} \quad (36)$$

Using the output flow from the valve spool as the input, the transfer function can be expressed as:

$$G_{LSQ}(s) = \frac{Y_S(s)}{Q_L(s)} = \frac{Y_S(s)}{K_q X_v(s)} = \frac{1}{s \left(\frac{s^2}{\omega_p^2} + \frac{2\zeta_p s}{\omega_p} + 1 \right) A_m} \quad (37)$$

The second term of the numerator in Equation (33) is the variation in the output displacement from the piston that depends on the external load, and the transfer function can be expressed as:

$$G_{LE}(s) = \frac{Y_{ex}(s)}{F_{ex}(s)} = - \frac{\left(\frac{1}{K_{cem}} + \frac{1}{K_y} s \right)}{s \left(\frac{s^2}{\omega_p^2} + \frac{2\zeta_p s}{\omega_p} + 1 \right)} \quad (38)$$

3.5. Displacement Transducer

The displacement transducer is a metal linear sensing element and can be considered a proportional control element [37]. The transfer function of this control element can be expressed as:

$$H_d(s) = \frac{U_d(s)}{Y_d(s)} \quad (39)$$

where H_d is the gain coefficient (V/m) of the displacement transducer, U_d is the output voltage (V) from the displacement transducer, and Y is the displacement (m) of the hydraulic cylinder piston rod.

3.6. Overall Transfer Function and Major Parameters of the System

The open-loop transfer function of the electro-hydraulic proportional control system can be obtained by computing the transfer functions of the system's components—Equations (3), (4), and (37)–(39):

$$G(s)H(s) = K_a \frac{K_{xv}}{\frac{s^2}{\omega_{sv}^2} + \frac{2\zeta_{sv}s}{\omega_{sv}} + 1} \frac{1}{s \left(\frac{s^2}{\omega_p^2} + \frac{2\zeta_p s}{\omega_p} + 1 \right) A_m} H_d \quad (40)$$

Table 4 shows the major parameters of the system. Figure 5 shows a block diagram of the transfer functions of the hydraulic loading system with electro-hydraulic proportional control.

Table 4. Major parameters of the hydraulic loading system with electro-hydraulic proportional control.

Summary	Parameter	Summary	Parameter
Proportional amplifier gain (K_a)	0.2 V/A	Hydraulic oil elastic modulus (β_e)	7×10^8 Pa
Proportional valve gain (K_{sv})	$0.09 \text{ m}^3/(\text{s}\cdot\text{A})$	Proportional valve flow gain (K_q)	$0.05 \text{ m}^2/\text{s}$
Proportional valve damping ratio (ζ_{sv})	0.6	Displacement sensor gain (K_d)	1 V/m
Natural frequency of proportional valve (ω_{sv})	314 rad/s	Equivalent total mass of piston (m_i)	5100 kg
Effective area of piston cavity (A_1)	$1.23 \times 10^{-2} \text{ m}^2$	Natural frequency of hydraulic system (ω_p)	154.42 rad/s
Effective area of rod cavity (A_2)	$9.44 \times 10^{-3} \text{ m}^2$	Damping ratio of hydraulic system (ζ_p)	0.111
Total volume of hydraulic cylinder (V)	$3.07 \times 10^{-3} \text{ m}^3$	Internal leakage coefficient (C_{ip})	$3.3 \times 10^{-3} \text{ m}^5/(\text{N}\cdot\text{s})$
Initial volume of two chambers (V_{10}/V_{20})	$3.74 \times 10^{-4} \text{ m}^3$	External leakage coefficient (C_{ep})	0
Hydraulic spring stiffness (K_y)	$1.22 \times 10^8 \text{ N/m}$	Equivalent leakage coefficient (C_{ie})	$3.61 \times 10^{-11} \text{ m}^5/(\text{N}\cdot\text{s})$
System flow coefficient (K_{ce})	3.76×10^{-11}	Additional leakage coefficient (C_f)	$-2.81 \times 10^{-12} \text{ m}^5/(\text{N}\cdot\text{s})$

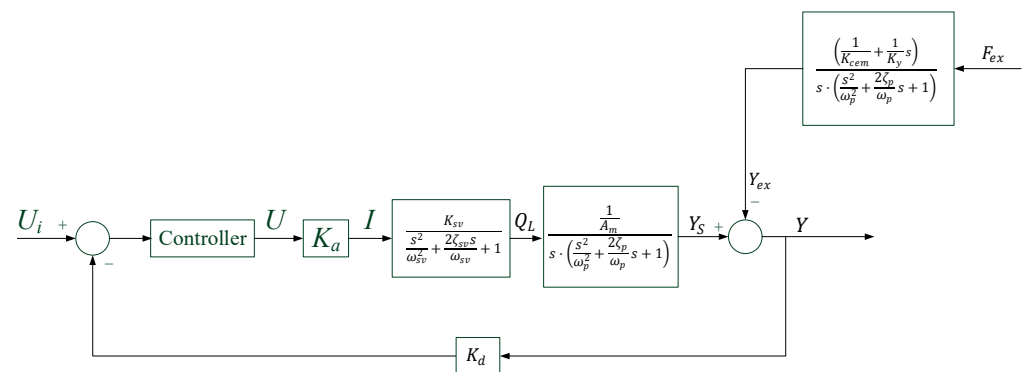


Figure 5. Block diagram of the transfer functions of the control system.

4. Results and Discussion

4.1. Analysis and Optimization

4.1.1. Stability Analysis

An open-loop Bode diagram of the hydraulic loading system with electro-hydraulic proportional control was plotted using the MATLAB 2018b software, as shown in Figure 6. The gain margin of the hydraulic loading system, gain crossover frequency, phase gain, and phase crossover frequency were 26.3 dB, 143.1 rad/s, 89.5° , and 1.66 rad/s, respectively. The criterion for performance assessment of electro-hydraulic proportional control systems is that the gain margin should be >6 dB and phase margin $>45^\circ$. Thus, the dynamic process of the electro-hydraulic proportional control system converges and performs well.

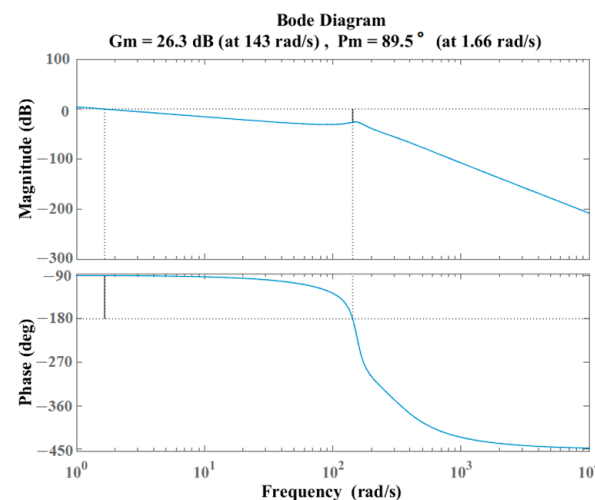


Figure 6. Open-loop Bode diagram of hydraulic loading system with electro-hydraulic proportional control.

4.1.2. Analysis of Dynamic Characteristics

As mentioned above, the dynamic process of the system converges. A model of the electro-hydraulic proportional control system was established in the Simulink environment. Figure 7 shows the response of the system to a 0.1 m step input signal. The response curve of the electro-hydraulic proportional closed-loop control system to the step signals is smooth, exhibiting no oscillation or overshoot. However, the response was slow (steady-state error was mostly eliminated after 2.7 s and completely eliminated after 3.1 s) and did not satisfy the requirements for the simulation test of hydropneumatic suspensions.

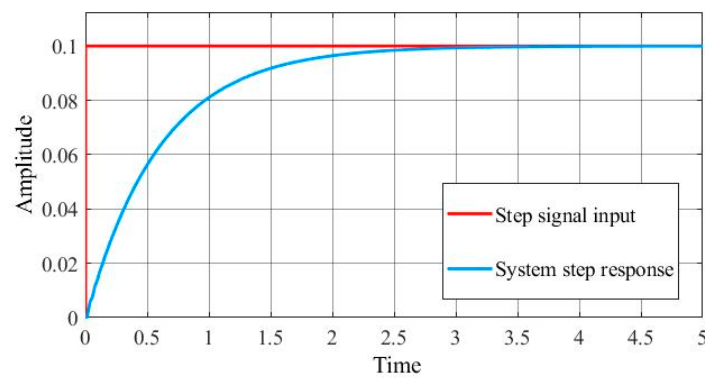


Figure 7. Response curve of system to step input signal.

The frequency of the vehicle vibration caused by road roughness falls into the range of 0.5–25 Hz [38]. Sine wave signals of 1, 10, and 25 Hz were used separately as input to the system. Figure 8 shows the responses of the system. The output signal had the same frequency as the input signal. However, as the frequency of the input signal increases, the magnitude of the output signal decreases, the phase lag increases, and the signal distortion increases. Thus, it is necessary to improve the accuracy of the hydraulic loading system by incorporating an appropriate control algorithm.

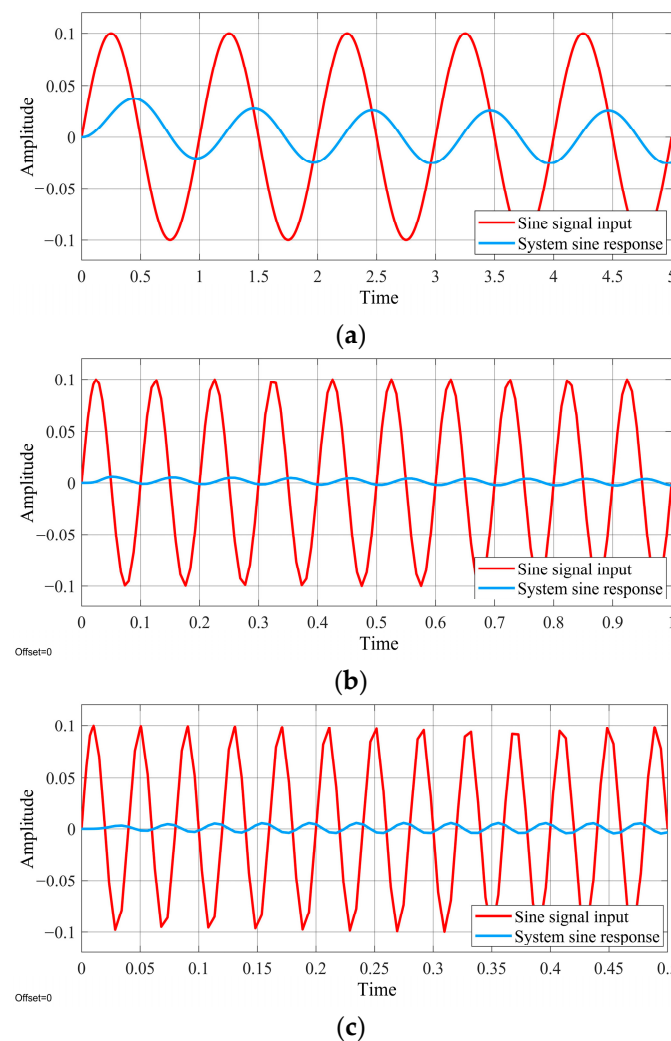


Figure 8. Response curves of the system to sine signals: (a) 1 Hz, (b) 10 Hz, and (c) 25 Hz.

4.1.3. Optimization with Adaptive Fuzzy PID Control Algorithm

The electro-hydraulic proportional control system was combined with an adaptive fuzzy PID control algorithm. The error of the closed-loop control system and its variability, e and de/dt , were fuzzified, and the corresponding fuzzified variables, E and E_c , were used as input to the fuzzy controller. The fuzzy domains of discourse were defined as $E \in (-0.12, 0.12)$; $E_c \in (-6, 6)$.

Three fuzzy variables were output from the PID controller. The fuzzy domains of discourse were defined as: $\Delta K_p \in (-9, 9)$; $\Delta K_i \in (-0.3, 0.3)$; $\Delta K_d \in (-0.006, 0.006)$.

According to the characteristics of the simulation test bench, each of the input and output variables was divided into seven subsets: (NB, NM, NS, ZO, PS, PM, PB). Table 5 shows the fuzzy inference rules.

Table 5. Fuzzy inference rules.

$\Delta K_p/\Delta K_i/\Delta K_d$		NB	NM	NS	E_c ZO	PS	PM	PB
E	NB	PB/NB/PS	PB/NB/NS	PM/NM/NB	PM/NM/NB	PS/NS/NB	ZO/ZO/NM	ZO/ZP/PS
	NM	PB/NB/PS	PB/NB/NS	PM/NM/NB	PS/NS/NM	PS/NS/NM	ZO/ZO/NS	NS/ZO/ZO
	NS	PM/NB/ZO	PM/NM/NS	PM/NS/NM	PS/NS/NM	ZO/ZO/NS	NS/PS/NS	NS/PS/ZO
	ZO	PM/NM/ZO	PM/NM/NS	ZS/NS/NS	ZO/ZO/NS	NS/PS/NS	NM/PM/NS	NM/PM/ZO
	PS	PS/NM/ZO	PS/NS/ZO	ZO/ZO/ZO	NS/PS/ZO	NS/PS/ZO	NM/PM/ZO	NM/PB/ZO
	PM	PS/ZO/PB	ZO/ZO/NS	NS/PS/PS	NM/PS/PS	NM/PM/PS	NM/PB/PS	NB/PB/PB
	PB	ZO/ZO/PB	ZO/ZO/PM	NM/PS/PM	NM/PM/PM	NM/PM/PS	NB/PB/PS	NB/PB/PB

A simulation model was established in the Simulink environment. Figures 9 and 10 show the responses of the system to step and sine signals, respectively. With the addition of the adaptive fuzzy PID control algorithm, the response curve of the hydraulic loading system with electro-hydraulic proportional control to a step signal reached and stabilized at the preset value within 0.3 s and had no oscillation or overshoot. The optimized system also performed well in replicating sine signals of different frequencies.

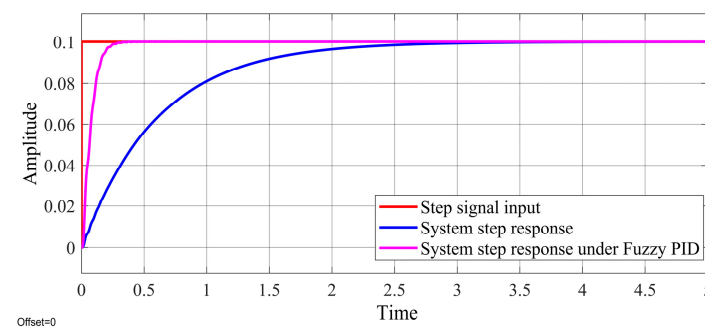


Figure 9. Response curve to a step signal of the hydraulic loading system with an electro-hydraulic proportional controller combined with an adaptive fuzzy PID control algorithm.

4.2. Response of the System to Random Road Vibration

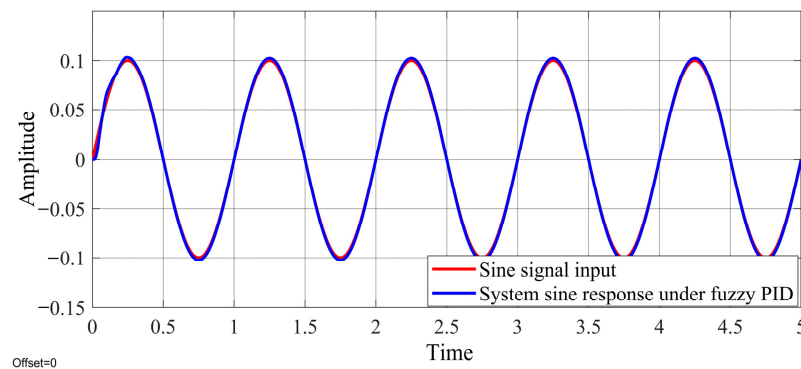
4.2.1. Model of Road Vibration

For the hydropneumatic suspension test bench to closely simulate the real operating conditions of vehicles, its output signal must accurately simulate road roughness-induced random road vibrations. Road roughness is usually obtained through two methods. The first method is direct detection, which uses sensor technology to directly measure the longitudinal profile of the road surface. Then, using road roughness evaluation technology and theory, the measured longitudinal profile curve of the road surface is converted into various roughness indicators. The second method is responsive detection, which uses sensors to detect the dynamic response of the vehicle body to road roughness, and then converts the dynamic response of the vehicle body into various roughness indicators.

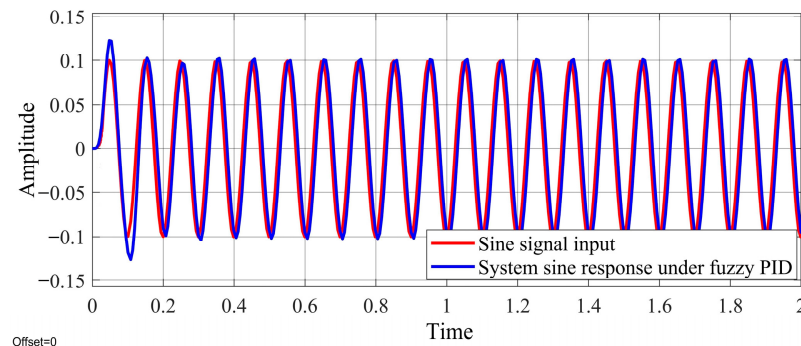
through a roughness evaluation model [19]. The first method is costly and not suitable for long-distance roads, so in this study, the second method was adopted. A time domain model of random road vibration was established by combining first-order filter and band-limited white noise:

$$\dot{q}(t) = -2\pi f_0 q(t) + 2\pi n_0 \sqrt{G_0 v} \omega(t) \quad (41)$$

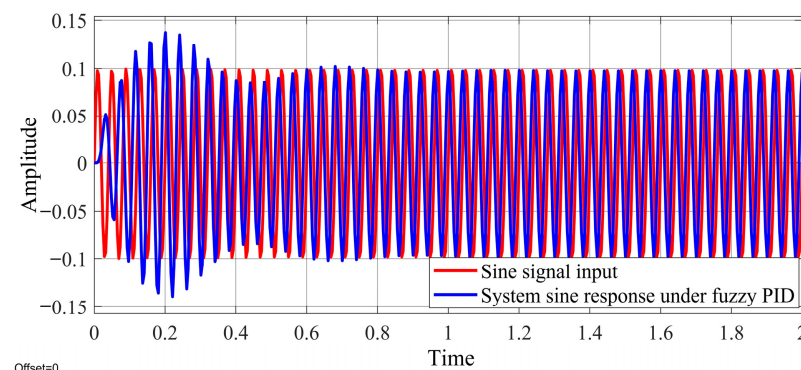
where $q(t)$ is the road excitation function, $\omega(t)$ is the Gaussian white noise with a mean = 0, f_0 is the lower cutoff frequency (=0.1 Hz), n_0 is the reference spatial frequency (=0.1 Hz), v is the vehicle velocity (=15 m/s), and G_0 is the road roughness coefficient obtained based on the road power spectral density that is determined by the road classification into one of eight (A–H) classes, as shown in Table 6.



(a)



(b)



(c)

Figure 10. Response curves to sine signals of different frequencies of the hydraulic loading system with an electro-hydraulic proportional controller incorporated with an adaptive fuzzy PID control algorithm: (a) 1 Hz, (b) 10 Hz, and (c) 25 Hz.

Considering their higher proportions, roads of Class B and C were simulated. The road roughness coefficients of Class B and C roads are $G_{0B} = 64 \times 10^{-6} \text{ m}^3$ and $G_{0C} = 256 \times 10^{-6} \text{ m}^3$, respectively. A simulation model of the excitations from Class B and C roads was established in the MATLAB/Simulink environment. Figure 11 shows the simulated random vibrations from Class B and C roads.

Table 6. Road roughness-based road classification.

Road Classification	Road Roughness Coefficient (10^{-6} m^3)		
	Lower Limit	Geometric Average	Upper Limit
A	8	16	32
B	32	64	128
C	128	256	512
D	512	1024	2048
E	2048	4096	8192
F	8192	16,384	32,768
G	32,768	65,536	131,072
H	131,072	262,144	524,288

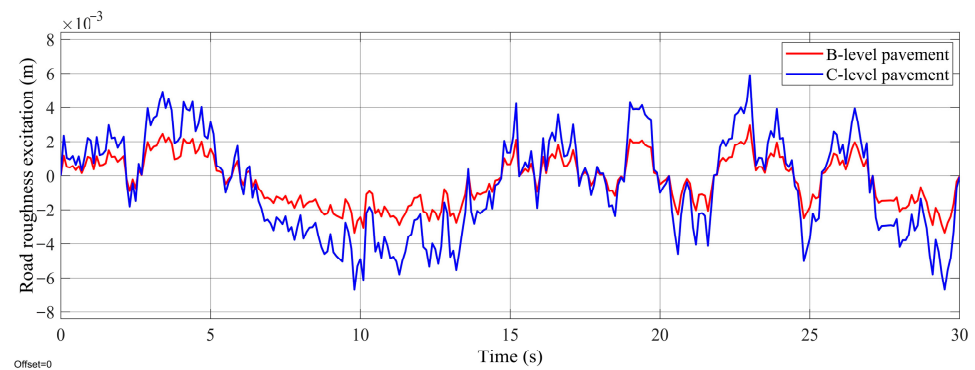


Figure 11. Waveforms of road vibrations from Class B and C roads.

4.2.2. Response to Random Road Vibration

Figure 12 shows the responses of the hydraulic loading system with an electro-hydraulic proportional controller combined with a fuzzy PID control algorithm to input signals of random vibrations from Class B and C roads. The fuzzy PID control system exhibited a good response, with the output waveform being consistent with that of the input road vibrations. The relative errors in the simulation of the vibrations from Class B and C roads were 12.2% and 13.1%, respectively, which were both less than the 15% error required for engineering systems, confirming that the proposed fuzzy PID control system satisfied the requirements for practical engineering applications.

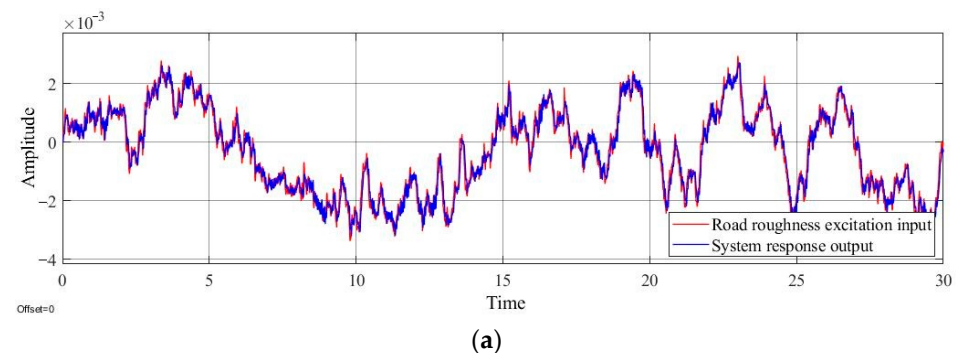


Figure 12. Cont.

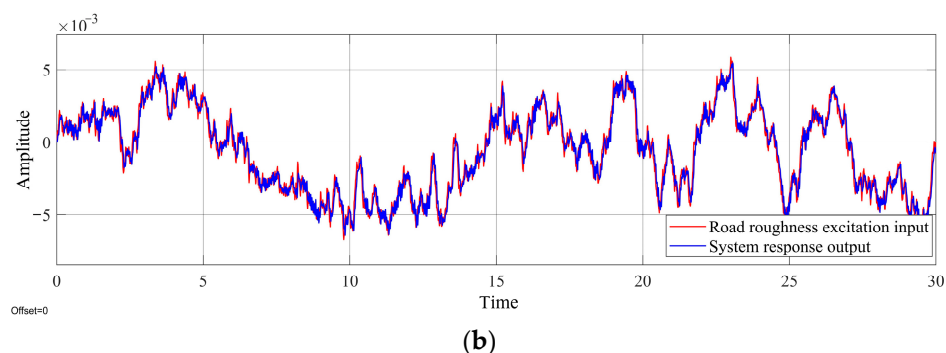


Figure 12. Response curves to random road vibrations: (a) Class B roads, and (b) Class C roads.

5. Conclusions

(1) A multi-functional compound simulation test bench was designed for the performance assessment of heavy vehicle hydropneumatic suspensions using the electro-hydraulic proportional control technique.

(2) A mathematical model was developed for describing the hydraulic loading system with electro-hydraulic proportional control, and the transfer function of the system was derived.

(3) Based on the established mathematical model of the electro-hydraulic proportional control hydraulic loading system, a Bode diagram was drawn using MATLAB 2018b software to analyze the stability of the system. A simulation model of the system was established in the Simulink environment to analyze its dynamic characteristics. The results showed that the hydraulic loading system with electro-hydraulic proportional control had good stability, smooth response, and no overshoot. However, the response was slow.

(4) The fuzzy control algorithm is added on the basis of the PID controller, and the fuzzy PID controller of the electro-hydraulic proportional system is established. The dynamic response characteristics of the hydraulic loading system with electro-hydraulic proportional control were improved. The improved system performed well in replicating the road vibrations from Class B and C roads.

(5) Due to time and funding constraints, this article was unable to complete the development and experimental work of an electro-hydraulic proportional control hydropneumatic suspension simulation test bench. Instead, it only analyzed the stability and dynamic characteristics of the system based on the established mathematical model. In the future, more experimental verification work will be carried out on this basis.

Author Contributions: Conceptualization, Z.W. and Y.Z.; methodology, Z.W. and Y.Z.; software, C.S.; validation, B.J., C.S. and H.Z.; formal analysis, B.J.; investigation, Z.W.; resources, B.J.; data curation, B.J. and C.S.; writing—original draft preparation, Z.W.; writing—review and editing, Z.W.; visualization, Y.Z.; supervision, Z.W.; project administration, Z.W.; funding acquisition, Z.W. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the Fund for Guiding the Transformation of Scientific and Technological Achievements of Shanxi Province, grant number 2021104021301061, and by the Fund for Fundamental Research Program of Shanxi Province, grant number 202203021221106.

Data Availability Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

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