Research Article

Research on Damping Characteristics of Interconnected Hydropneumatic Suspension considering the Effects of Hose and Check Valve

Hongmin Zhang and Xin Fang

1School of Intelligent Engineering, Shandong Management University, Jinan 250357, China
2Department of Intelligent Manufacturing, Shandong Labor Vocational and Technical College, Jinan 250300, China

Correspondence should be addressed to Hongmin Zhang; zanghm@sdmu.edu.cn

Received 8 February 2021; Accepted 9 July 2021; Published 17 July 2021

Copyright © 2021 Hongmin Zhang and Xin Fang. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

The interconnected hydropneumatic suspension (ICHPS) has not only the nonlinear stiffness and damping of the independent hydropneumatic suspension (IDHPS) but also antiroll and antipitch functions. The existing analysis of hydropneumatic suspension damping mainly focuses on the orifice and check valve in the suspension cylinder. In this study, the calculation formula of the damping force of ICHPS is established, and the numerical simulation results show that the damping characteristics of the hydraulic hose cannot be ignored. The influence of check valve and hose on the damping characteristics is analyzed. Through the equivalent energy method, the equivalent compression damping ratio and the equivalent recovery damping ratio of the ICHPS are established. It is pointed out that when designing the damping characteristics of the ICHPS, it is necessary to select the orifices, check valves, and hose damping reasonably to make the damping characteristics get the best match.

1. Introduction

The suspension system is the general name of all force transmission connecting devices between the frame and the axle or wheel of the vehicles, and it has good shock absorption effect and is the key device to ensure the good driving of vehicles [1]. To further improve driving stability and handling comfort, a hydropneumatic suspension (HPS) was introduced by Paul Magès of Citroën during the 1940s [2–4]. HPS is generally composed of a cylinder and an accumulator. Inert gas nitrogen is used as the elastic medium, and the force is transmitted by the incompressibility of hydraulic oil. Orifices and check valves are set inside the cylinder or in the connecting oil circuit between the cylinder and the accumulator. The throttling loss of oil through the orifices is used to attenuate energy and suppress vibration. The HPS has the characteristics of nonlinear vertical stiffness and stiffness increasing (decreasing), which makes the suspension dynamic stroke smaller and the suspension stiffness smaller in good road conditions. Therefore, it can reduce the vertical force and acceleration on the body and improve the smooth driving of the vehicle. Meanwhile, when the road is rough, the increase of suspension dynamic travel leads to the increase of suspension stiffness, which increases the shock resistance of the suspension, reduces the probability of suspension breakdown, and improves the off-road driving performance of the vehicle [5]. HPS has higher static stiffness and nonlinear damping. Compared with the traditional rubber bearing, it has an obvious effect on reducing the low frequency vibration of vehicles and controlling the vibration of a cab [6] and has various merits: space saving, light weight, and flexible design [7]. Now various HPS structures have been used in different vehicle suspensions such as automobiles [8, 9], city-bus [10, 11], and construction vehicles [12–16].

At present, there are mainly two kinds of HPS used in engineering vehicles: independent and interconnected. The IDHPS is that each suspension cylinder is not
interconnected with each others, which can independently realize the free adjustment of the body height. However, for ICHPS, the left and right (front and back or cross) hydraulic cylinders are interconnected with each other, not only has the advantages of IDHPS but also can restrain each other when the road is not smooth, to keep the whole vehicle in a balanced state. A large body of research work mainly focuses on the following four aspects: (i) optimisation of the stiffness and damping characteristic of the IDHPS [9–11, 17, 18], (ii) research on antiroll and antipitch performance of IDHPS [15, 19–25], (iii) control of active or semiactive HPS system [26–30], and (iv) energy recovery of HPS [31–34].

The ICHPS not only has the characteristics of nonlinear stiffness and damping but also can improve the antipitch and antiroll performance through the interconnection of hoses. The main research object of this study is the damping characteristics of the ICHPS. For the damping characteristics, the previous literature focused on the analysis of the influence of the change of the orifice and check valve on the damping characteristics but did not analyze the damping effect of the cross-interconnected hose. At the same time, for the orifice and check valve, only their influence on the damping coefficient is analyzed, but their influence on the suspension cylinder itself is not analyzed.

Therefore, this study establishes a mathematical model with hose damping, analyzes the influence of the length and diameter of the hose, analyzes the influence of orifice and check valve on vibration attenuation, and analyzes the influence of check valve on oil replenishment and orifice on pressure holding. At the same time, the equivalent energy method is used to linearize the nonlinear damping to design the ICHPS.

2. Schematic Diagram of ICHPS

Figure 1 shows the model of ICHPS with two interconnected chambers, which is a scheme to improve the roll stiffness by using the characteristics of hydropneumatic. A cylinder and an accumulator are arranged in the suspension of a bridge. The rod-side chamber of one cylinder is connected to the piston-side chamber of the other. The two accumulators have the same function, and each one is shared by the cylinders on both sides. If one side provides the pressure of the piston-side chamber (chamber I), the other side provides the pressure of the rod-side chamber (chamber II and III). The oil in chamber III (inner ring chamber) enters chamber II (outer ring chamber) through check valve 4 and orifice 5, but the oil in chamber II (outer ring chamber) can only enter chamber III (inner ring chamber) through orifice 5. The structure of this scheme is completely symmetrical on the left and right, which is consistent with the symmetry of the mass distribution of the vehicle. It can be regarded as composed of two identical cylinders with the reverse pressure chamber sharing the accumulator.

When the car body rolls, the cylinder on one side of the car body bears additional compression force, and the car body on this side sinks; meanwhile, the cylinder on the other side bears additional tension force, and the car body rises. On the compressed side, the pressure in the piston-side chamber of the oil cylinder increases. Due to the communication of the oil circuit, the pressure in the small chamber of the oil cylinder on the other side will also increase, forcing the body on the other side to have a downward trend. Similarly, on the rising side, the pressure in the piston-side chamber of the oil cylinder decreases, and the body on the other side will move upward when the oil is inhaled. It can be seen that the communication between the left and right oil lines has an antiroll effect.

3. Calculation of Damping Force

For the ICHPS shown in Figure 1, suppose that the body displacement $x$ (m) and the body velocity $v$ (m/s) are both compressed to positive and restored to negative. Assuming that the car body vibrates up and down without roll, it is easy to list the following force balance relations:

$$ F = 2(p_1 A_1 - p_2 A_2), $$

where $F$ is the output force of ICHPS; $p_1$ and $p_2$ are the pressure of chamber I and chamber II of the cylinder; $A_1 = (\pi D^2/4)$ and $A_2 = (\pi (D^2 - d^2)/4)$ are the effective areas of chamber I and chamber II of the cylinder, $D$ is the piston diameter of the cylinder, $d$ is the rod diameter of the cylinder.

Considering the pressure loss along the interconnected hose:

$$ p_1 = p + \lambda g_1 \frac{l_{g1}}{d_g} \frac{\rho v^2}{2} \text{sign } v, $$

$$ p_2 = p - \lambda g_2 \frac{l_{g2}}{d_g} \frac{\rho v^2}{2} \text{sign } v, $$

where $p$ is the actual gas pressure of accumulator; $p_1$ is the chamber pressure of cylinder; $\rho$ is the hydraulic oil density; $l_{g1}$ and $l_{g2}$ are the lengths of hose from the left (right)
accumulator to chamber I and chamber III of the left (right) cylinder; \( d_g \) is the diameter of hose connecting accumulator and cylinder; \( v_{g1} = (vD^2/d_g^2) \) and \( v_{g2} = (v(D^2 - d^2)/d_g^2) \) are the oil flow velocities in hose from left (right) accumulator to chamber I of the left (right) cylinder and right (left) suspension cylinder chamber III; sign \( v \) is sign function, defined as follows:

\[
\text{sign } v = \begin{cases} 
1, & v > 0, \\
0, & v = 0, \\
-1, & v < 0,
\end{cases}
\]

\( \lambda_g \) is the resistance coefficient along the hose, determined by the following formula:

\[
\lambda_g = \begin{cases} 
\frac{80}{\text{Re}} & \text{Re} \leq 2000, \\
\frac{0.3164}{\text{Re}^{0.25}} & 2000 < \text{Re} \leq 10^5, \\
\frac{0.032 + 0.221}{\text{Re}^{0.25}} & 10^5 < \text{Re} \leq 3 \times 10^6, \\
\left(2\lg \frac{d}{\Delta} + 1.74\right)^{-2} & \text{Re} > 900 \frac{d_g}{\Delta},
\end{cases}
\]

where \( \text{Re} = (v, d_g/\mu) \) is the Reynolds number; Reynolds number of fluid in the hydraulic hose of ICHPS is generally less than \( 10^5 \); \( \mu \) is the kinematic viscosity of hydraulic oil; \( \Delta \) is the surface roughness of pipe wall, 0.03 mm for rubber hose, and \( (\Delta/d) \) is relative roughness of pipe wall.

By analyzing chambers II and III of the cylinder, the oil in chamber III can enter chamber II through the orifice and check valve, and the oil in chamber II can only enter chamber III through the orifice. The oil flow between the orifice and the check valve is simplified as short orifice throttling. According to the short orifice throttling theory, the pressure relationship between chamber II and chamber III can be obtained:

\[
p_2 = p_3 - \frac{\rho}{2C_zA_z + 0.5C_dA_d} (1 + \text{sign } v) \quad q, \quad (6)
\]

where \( q = (\pi \nu (D^2 - d^2)/4) \) is the oil flow rate of chamber II and chamber III of the cylinder; \( C_z = 0.7 \) and \( C_d = 0.6 \) are the flow coefficients of orifice and check valve [13]; \( A_z \) and \( A_d \) are the effective flow areas of orifice and check valve.

The analysis of the accumulator shows that the gas in the HPS is nitrogen, and its performance is close to that of the ideal gas. The state equation of the ideal gas is used to describe the state change process of the gas:

\[
pV^r = p_jV_j^r,
\]

where \( r \) is the gas polytropic index; \( p_j \) is the gas pressure of accumulator at static equilibrium position; \( V_j \) is the gas volume of the accumulator at static equilibrium position; \( V \) is the actual gas volume of the accumulator.

For the value of the gas polytropic index, the thermodynamic process of gas state change is regarded as an adiabatic process when the load is fast, \( r = 1.4 \); when the load is slow, the thermodynamic process of gas state change is regarded as an isothermal process, \( r = 1 \). Feng [35] showed that for 2.5 L and 3.5 MPa accumulator, the variable index is 1 when the excitation frequency is below 1 Hz, 1.4 when the excitation frequency is 1–3 Hz, and 1.7 when the excitation frequency is above 3 Hz. Wang et al. [36] considered that the gas compression rate has a great influence on the gas polytropic index and constructed a real gas polytropic index model based on the volume compression rate. Lin et al. [3] pointed out that the exponent value increases with an increase in excitation frequency. However, it tends to saturate near 1.4 at frequencies above 1 Hz. Considering the working frequency range of HPS is generally higher than 1 Hz, the exponential value of 1.4 is considered appropriate. According to the research of vehicle dynamics and ergonomics, the vertical vibration frequency that the human body is used to is the frequency of the body moving up and down when walking, which is about 1–1.6 Hz. To meet the habit of the human body, the natural vibration frequency of the body should be in or close to this frequency range as far as possible. To sum up, in the actual work process, take the gas polytropic index \( r = 1.4 \).

By synthesizing equations (1)–(3) and (6)–(8), the output force \( F \) of the ICHPS is obtained:

\[
F = \frac{\pi d^2 p_jV_j^r}{4(V_j - (\pi d^2/4)x)^2} \left[ \pi \rho(D^2 - d^2)^3 \left( \frac{\nu}{128} \left( C_zA_z + 0.5C_dA_d(1 + \text{sign } v) \right) \right)^2 \right. \\
+ \left. \frac{\pi \rho v^2 \text{sign } v}{8d_g^2} \left( \lambda_{g1}l_{g1}D^6 + \lambda_{g2}l_{g2}(D^2 - d^2)^3 \right) \right].
\]

Equation (9) shows that the output force of the ICHPS consists of three parts: the gas elastic force \( F_1 \), the damping force \( F_2 \) of the orifice and check valve, and the damping force \( F_3 \) of the hydraulic hose. Namely,

\[
F_1 = \frac{\pi d^2 p_jV_j^r}{4(V_j - (\pi d^2/4)x)^2}, \\
F_2 = \frac{\pi \rho(D^2 - d^2)^3}{128} \left( \frac{\nu}{C_zA_z + 0.5C_dA_d(1 + \text{sign } v)} \right)^2 \text{sign } v, \\
F_3 = \frac{\pi \rho v^2 \text{sign } v}{8d_g^2} \left( \lambda_{g1}l_{g1}D^6 + \lambda_{g2}l_{g2}(D^2 - d^2)^3 \right).
\]
4. System Analysis and Simulation Research

4.1. Force-Displacement and Force-Velocity Characteristics. Equation (9) is easy to draw its force-displacement and force-velocity characteristics through mathematics software. In addition, the simulation model of ICHPS is established in AMESim, and the damping effect of the hydraulic hose is observed by comparing the results of equation (9) and the simulation model. Furthermore, it is acceptable to use the simulation model to find the optimal design solution of HPS because the characteristics of HPS are determined by various design parameters and complex nonlinear interaction [37].

Figure 2 shows the simulation model of the ICHPS. The simulation data adopts the actual parameters of a certain type of all-terrain crane, as shown in Table 1.

Apply the sinusoidal excitation signal $A = 0.1 \text{ m}$, $f = 1.67 \text{ Hz}$ [38] to the cylinder, and the excitation signal is shown in Figure 3. Figure 4 shows a comparative analysis of the force-displacement characteristics and force-speed characteristics of the simulation model, the mathematical model with hose damping, and mathematical model without hose damping. It can be seen that the mathematical model considering hose damping is similar to the simulation result, while the mathematical model not considering hose damping has an obvious deviation from the simulation result.

4.2. Response to Step Excitation. Figure 5 shows the model of single bridge ICHPS, and the suspension mass of the single bridge is 12000 kg. Figure 6 shows an equivalent mechanical suspension model, in which the damping element 3 is set with two kinds of damping forces. One is to consider hose damping, i.e., $F = F_2 (v) + F_3 (v)$; and the other is not to consider hose damping, i.e., $F = F_2 (v)$. When step displacement excitation is applied to the system, there is no excitation displacement in $0 \sim 0.1 \text{ s}$, and the displacement jumps to $0.01 \text{ m}$ in $0.1 \text{ s}$, as shown in Figure 7.

Figure 8 shows the comparison curves of body displacement, velocity, and acceleration of the simulation model, the equivalent mechanical suspension model with hose damping, and the equivalent mechanical suspension model without hose damping. Figure 8 shows that for step excitation, the simulation model and the equivalent mechanical suspension model considering hose damping tend to be stable after 3 or 4 oscillations, and their displacement, velocity, and acceleration responses are the same. The equivalent mechanical suspension model without considering the hose damping will continue to oscillate.

5. Analysis of the Influence of Check Valve and Hose on Damping Characteristics

In fact, the damping force of the ICHPS is related to the piston diameter $D$, the rod diameter $d$, the density $\rho$ of the hydraulic oil, the flow area $A_s$ of the orifice, the flow area $A_d$ of the check valve, the diameter of the hose $d_h$, the length of the hose $l_{h1}$ and $l_{h2}$, and the kinematic viscosity $\mu$ of the hydraulic oil. A large number of researchers have analyzed the influence of cylinder size and orifice size on the damping force [6, 12, 13, 19–21]. In order to shorten the calculation time, Chen et al. [39] proposed a technique to make the model reduction method and applied it to a practical model for dynamic analysis of wheel-rail interactions in railway engineering. However, this study focuses on the influence of the check valve and hose.

5.1. Function of Check Valve. Due to the existence of a check valve, the damping of the compression stroke and recovery stroke of the HPS system is different. For the traditional shock absorber, the relationship between the damping of the compression stroke $f_f$ and recovery stroke $f_j$ is as follows: $f_j = (0.25 \sim 0.5) f_f$ [40]. Wu [41] and Zhang [42] considered that for general vehicles, the ratio should be $0.2 \sim 0.3$. For off-road and military vehicles, the ratio should be $0.5 \sim 0.7$. However, all the above works of literature considered that the ratio could be achieved by orifice and check valve. However, this study argues that the main function of the check valve is to supply oil to the rod-side chamber of the cylinder rather than to obtain the damping force during compression.

Apply step excitation to the model shown in Figure 7; the step displacement is $0.1 \text{ m}$, the size of the orifice is kept at $2.7 \text{ mm}$, and the diameter of the check valve seat hole is set as $1 \text{ m}$, $3 \text{ mm}$, and $5 \text{ mm}$, respectively.

Figure 9 shows the displacement curve of the car body. It can be seen that although the diameter of the check valve seat hole has changed from $1 \text{ mm}$ to $5 \text{ mm}$, the change of the displacement attenuation curve of the car body is very small. On the contrary, when the diameter of the check valve decreases, the displacement of the highest point of the car body increases. For example, when the diameter is $1 \text{ mm}$, the displacement of the highest point is $17.7 \text{ mm}$, and when the diameter is $5 \text{ mm}$, the displacement of the highest point decreases to $15 \text{ mm}$, but this change is very small. It can be said that the change of check valve diameter does not affect the attenuation of body vibration displacement.
Figure 10 shows the pressure curve of the rod-side chamber. With the decrease of the diameter of the check valve, the pressure of the rod-side chamber of the cylinder becomes smaller and smaller. For example, the pressure of the rod-side chamber with the check valve seat hole of 1 mm, 3 mm, and 5 mm is −0.2 bar, 29 bar, and 54 bar, respectively. The rod-side chamber of the cylinder with the check valve seat hole of 1 mm has been sucked into negative pressure. To

Table 1: Simulation model parameters.

| Description                                    | Notation | Value | Units |
|------------------------------------------------|----------|-------|-------|
| Piston diameter                                | $D$      | 0.11  | M     |
| Rod diameter                                   | $d$      | 0.09  | M     |
| Gas pressure of accumulator in static balance  | $p_j$    | $9.25 \times 10^6$ | Pa    |
| Gas volume of accumulator in static balance    | $V_j$    | $1.2 \times 10^{-3}$ | m$^3$ |
| Gas polytropic index                           | $r$      | 1.4   | —     |
| Hydraulic oil density                          | $\rho$   | 870   | kg/m$^3$ |
| Flow area of orifice                           | $A_z$    | $1.41 \times 10^{-5}$ | m$^2$ |
| Flow area of check valve                       | $A_d$    | $1.41 \times 10^{-5}$ | m$^2$ |
| Flow coefficient of orifice                    | $C_z$    | 0.7   | —     |
| Flow coefficient of check valve                | $C_d$    | 0.6   | —     |
| Diameter of the hose                           | $d_g$    | 0.013 | m     |
| Length of hose from left (right) accumulator to chamber I of left (right) suspension cylinder | $l_{g1}$ | 0.5 | m |
| Length of hose from left (right) accumulator to chamber III of right (left) suspension cylinder | $l_{g2}$ | 2.5 | m |

Figure 3: Displacement and velocity excitation. (a) Displacement signal and (b) speed signal.

Figure 4: Force-displacement and force-velocity characteristics. (a) Force-displacement characteristic and (b) force-velocity characteristic.
prevent this situation, the flow area of the check valve should be increased in the design to improve the oil filling ability of rod-side chamber. At the same time, it has been pointed out from Figure 9 that increasing the diameter of the check valve does not affect the vibration curve of the body displacement.

Figure 11 shows the acceleration curve of the car body. With the decrease of the flow area of the check valve, the acceleration value of the car body becomes larger. The highest acceleration values of the car body with the check valve seat holes of 1mm, 3mm, and 5mm are 5.6 m/s², 4.7 m/s², and 3.9 m/s² respectively. This is because the pressure of the rod-side chamber decreases while the pressure of the piston-side chamber increases during instantaneous compression. The smaller the pressure of the rod-side chamber, the greater the upward force of the cylinder and the greater the acceleration. Therefore, to reduce the body acceleration, the flow capacity of the check valve should also be increased.

According to the above analysis, when designing the check valve, it is necessary to meet the need for oil replenishment and consider the limit condition that the pressure in the rod-side chamber is 0. The diameter $D_s$ of the check valve seat hole should be calculated according to the following formula:

$$D_s > \sqrt{\frac{(D^2 - d^2)v_{y\text{max}}}{nC_d \sqrt{2p_j/\rho}}} \quad (11)$$

where $v_{y\text{max}}$ is the maximum compression speed, the maximum recovery speed of 0.3 m/s when an all-terrain crane passes through a high bump on a class D road at 70 km/h; $D$ is the piston diameter; $d$ is the rod diameter; $C_d$ is the flow coefficient of the check valve; $p_j$ is the static equilibrium pressure; $\rho$ is the hydraulic oil density; $n$ is the number of check valves, generally 2. The diameter of the check valve seat hole can be calculated by determining the maximum compression speed.

5.2. Influence of Hose

5.2.1. Influence of Diameter of the Hose. Figure 12 shows the effect of the diameter of the hose on the damping characteristics. As can be seen from Figure 12, with the increase of the diameter of the hose, the damping coefficient decreases, and the ratio of the compression and recovery damping coefficient also decreases. However, when the diameter increases to a certain value, the change of the hose damping is very small. For example, when the diameter is 16 mm and 20 mm, the difference between the damping coefficient and the ratio of the compression and recovery is small. This can also be seen from Table 2.

Because the hose is long damping, its damping value is related to the viscosity of the hydraulic oil, and the viscosity is greatly affected by the oil temperature, while the orifice and check valve are short dampings, which are not sensitive to the change of oil temperature. Therefore, the too thin hose
should be avoided, but there is no need to choose too thick hose because when the diameter of the hose increases to a certain extent, the difference in damping effect is very small.

5.2.2. Influence of the Length of the Hose. Figure 13 shows that the effect of the length of the hose on damping is almost linear. With the increase of the length of the hose, the damping coefficient increases, and the ratio of compression damping coefficient to recovery damping coefficient increases, but this effect is very small. The values are shown in Table 3. Therefore, if the diameter is set reasonably, the change of the length in a certain range has little effect on the change of system damping. Therefore, the length of the hose is directly determined by the structural arrangement.

5.2.3. Influence of Hose on the Damping Ratio of Compression and Recovery. Figure 14 shows the comparison curve between the damping coefficient of the orifice plus the check valve and that of the hydraulic hose. It can be seen that in the recovery stroke, the damping of the hose is very small, mainly because the orifice is working, while in the compression stroke, the damping of the hose is more obvious than that of the check valve. Considering the hose damping, the damping ratio of compression and

Figure 8: Comparison of the response of the simulation model and equivalent mechanical suspension model to step excitation. (a) Comparison of body displacement, (b) comparison of body speed, and (c) comparison of body acceleration.

Figure 9: Displacement curve of suspension mass.
Figure 10: Pressure curve of rod-side chamber of cylinder. (a) Pressure curve of rod-side chamber of cylinder and (b) enlarged view of the pressure curve.

Figure 11: Acceleration curve of suspension mass. (a) Acceleration of the car body and (b) enlarged view of acceleration.
recovery is 0.27, but without considering the hose damping, the damping ratio of compression and recovery is only 0.058. Therefore, it is considered that the damping ratio of the compression and recovery stroke is a combination of the damping orifice, the check valve, and the hose.
6. Damping Characteristic Design

Since the damping characteristics of the hydropneumatic suspension system are affected by the orifice, check valve, and hoses, the matching of them should be considered in the design.

The equivalent damping ratio $\psi_y$ in compression and $\psi_f$ in recovery is calculated by using the equivalent energy method [43].

$$
\psi_y = \frac{\left( \int_0^1 (F_2 + F_1) dv \left( \frac{v}{2} \right) \right)}{2 \sqrt{Km}}
= \frac{1}{2 \sqrt{Km}} \left[ \frac{\nu \pi \rho (D^2 - d^2)}{192 (C_z A_z + C_d A_d)^2} \Delta t \right] \left[ \frac{80 \mu d_g}{D^2} (Re \leq 2000) + \frac{0.2301 v^{0.75}}{(D^2 / \mu d_g)^{0.25}} (Re > 2000) \right] + \frac{\pi \rho (D^2 - d^2)^3 l_g}{8d_g} \left( \frac{80 \mu d_g}{D^2 - d^2} (Re \leq 2000) + \frac{0.2301 v^{0.75}}{((D^2 - d^2)/\mu d_g)^{0.25}} (Re > 2000) \right),
$$

The equivalent damping ratio $\psi_f$ is calculated similarly.

**Table 3: Influence of the length of the hose on damping characteristics.**

| Length of the hose (mm) | Restoring damping coefficient at 0.3 m/s (N/m/s) | Compression damping coefficient at 0.3 m/s (N/m/s) | Ratio of damping coefficient of compression and recovery |
|------------------------|-----------------------------------------------|-----------------------------------------------|--------------------------------------------------|
| 0.5                    | 95281.5                                       | 33294.4                                       | 0.35                                             |
| 1                      | 101819                                        | 39830.6                                       | 0.39                                             |
| 1.5                    | 108357                                        | 46366.9                                       | 0.43                                             |

**Figure 14: Comparison of damping coefficient.**
In the formula, $\nu$ is generally in the range of 0.15–0.3 m/s, usually 0.3 m/s. The damping value at this speed reflects the damping capacity of the vehicle driving on a better road surface under low frequency excitation conditions to ensure comfort. When the damping ratio $\psi$ ($\psi = (\psi_y + \psi_f)/2$) is large, the vibration can be quickly attenuated, but at the same time, it will transfer the greater road impact force to the car body, and even make the wheels not able to rebound quickly to the ground and lose the adhesion; when the value of $\psi$ is small, the vibration duration becomes longer, which is not conducive to improving the comfort. The damping ratio suggested in this study is $\psi = 0.2$–0.4. For heavy vehicles and vehicles with poor road conditions, the value of $\psi$ should be larger. At the same time, the damping ratio $\psi_y$ of the general compression stroke is smaller, and the damping ratio $\psi_f$ of the recovery stroke is larger. The relationship between them is maintained as $\psi_y = (0.25–0.5) \psi_f$, and a larger ratio is taken for heavy vehicles and vehicles with poor road conditions.

7. Conclusion

In this study, the mathematical model of the damping force of the ICHPS is established. Through the analysis of its force-displacement and force-velocity characteristics and its response to step excitation, it is proved that the hose damping cannot be ignored in the ICHPS. For the design of damping characteristics, it is pointed out that the dimension of the check valve does not affect the damping characteristics of the system but affects the oil supplement to the rod-side chamber, so the check valve should be larger as far as possible. For the hose, it should not be too thin to prevent increasing the hose damping, but not too thick, because when the diameter of the hose is increased to a certain value, the influence on the system damping is small. After a reasonable selection of the diameter of the hose, the length of the hose has little influence on the system and can be selected according to the needs of the structural layout. The equivalent compression damping ratio and the equivalent restoring damping ratio of the ICHPS are established by the equivalent energy method. It is pointed out that when designing the damping characteristics of the ICHPS, not only the damping effect of the orifice and the check valve but also the damping effect of the hydraulic hose should be considered. By reasonably matching these three relations, the ICHPS can obtain the best damping characteristics.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

This work was financially supported by the Research Project of Education and Teaching Reform in Shandong Province (grant no. 205JG089) and Shandong Vocational Education Teaching Reform Research Project (grant no. 2019678), and these supports are greatly appreciated.

References

[1] Y. Zhao, H. Xu, Y. Deng, and Q. Wang, “Multi-objective optimization for ride comfort of hydropneumatic suspension vehicles with mechanical elastic wheel,” *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 233, no. 11, pp. 2714–2728, 2018.

[2] L. Konieczny, R. Burdzik, and T. Wegryn, “Analysis of structural and material aspects of selected elements of a hydropneumatic suspension system in a passenger car,” *Archives of Metallurgy and Materials*, vol. 61, no. 1, pp. 79–84, 2016.

[3] D. Lin, F. Yang, D. Gong, and S. Rakheja, “Design and experimental modeling of a compact hydropneumatic suspension strut,” *Nonlinear Dynamics*, vol. 100, no. 4, pp. 3307–3320, 2020.

[4] H. Hua, L. Wang, H. Qi, J. Zhang, and N. Zhang, “Implementation and experimental study of a novel air spring combined with hydraulically interconnected suspension to enhance roll stiffness on buses,” *SAE Technical Papers*, vol. 2015, 2015.

[5] X. Wang, X. Fang, S. Gao, and W. Zhao, “Analysis on stiffness characteristic of a connected hydropneumatic suspension system,” *Machine Tool & Hydraulics*, vol. 40, no. 9, pp. 55–57, 2012.

[6] R. Jiao, V. Nguyen, and V. Le, “Ride comfort performance of hydro pneumatic isolation for soil compactors cab in low frequency region,” *Journal of Vibeengineering*, vol. 22, no. 5, pp. 1174–1186, 2020.

[7] K. Kwon, M. Seo, H. Kim, T. H. Lee, J. Lee, and S. Min, “Multi-objective optimisation of hydropneumatic suspension with gas-oil emulsion for heavy-duty vehicles,” *Vehicle System Dynamics*, vol. 58, no. 7, pp. 1146–1165, 2020.

[8] A. Pazooki, S. Rakheja, and D. Cao, “Modeling and validation of off-road vehicle ride dynamics,” *Mechanical Systems and Signal Processing*, vol. 28, pp. 679–695, 2012.

[9] J. A. Tamboli and S. G. Joshi, “Optimum design of a passive suspension system of a vehicle subjected to actual random road excitations,” *Journal of Sound & Vibration*, vol. 219, no. 2, pp. 193–205, 1999.

[10] K. Deprez, D. Moshou, and H. Ramon, “Comfort improvement of a nonlinear suspension using global optimization and in situ measurements,” *Journal of Sound & Vibration*, vol. 284, no. 3–5, pp. 1003–1014, 2005.

[11] A. F. Naudé and J. A. Snyman, “Optimisation of road vehicle passive suspension systems. Part 1. Optimisation algorithm and vehicle model,” *Applied Mathematical Modelling*, vol. 27, no. 4, pp. 249–261, 2003.

[12] E. Zheng, X. Zhong, R. Zhu et al., “Investigation into the vibration characteristics of agricultural wheeled tractor-implement system with hydropneumatic suspension on the front axle,” *Biosystems Engineering*, vol. 186, pp. 14–33, 2019.

[13] Y. Yin, S. Rakheja, and P.-E. Boileau, “Multi-performance analyses and design optimisation of hydropneumatic suspension system for an articulated frame-steered vehicle,” *Vehicle System Dynamics*, vol. 57, no. 1, pp. 108–133, 2019.

[14] J. Féliz and C. Vera, “Bond graph assisted models for hydropneumatic suspensions in crane vehicles,” *Vehicle System Dynamics*, vol. 16, no. 5–6, pp. 313–332, 1987.
[15] K. Arikan, K. Küçük, H. Yurt, and H. Imrek, "Modelling and optimisation of an 8 x 8 heavy duty vehicle’s hydropneumatic suspension system," *International Journal of Vehicle Design*, vol. 71, pp. 122–138, 2016.

[16] W. Tian, X. Yi, L. Guo, and Y. Guo, "Simulation and experiments for seven spindled hybrid coupled hydro pneumatic suspension vehicles," *China Mechanical Engineering*, vol. 29, no. 9, pp. 1084–1089, 2018.

[17] K. Suh and H. Yoon, "Design optimization of a rear independent suspension for the korean light tactical vehicle," *International Journal of Automotive Technology*, vol. 19, no. 2, pp. 245–252, 2018.

[18] A. Kuznetsov, M. Mammadov, I. Sultan, and E. Hajilarov, "Optimization of a quarter-car suspension model coupled with the driver biomechanical effects," *Journal of Sound and Vibration*, vol. 330, no. 12, pp. 2937–2946, 2011.

[19] H. Qi, N. Zhang, Y. Chen, and B. Tan, "A comprehensive tune of coupled roll and lateral dynamics and parameter sensitivity study for a vehicle fitted with hydraulically interconnected suspension system," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 235, Article ID 2061157284, 2020.

[20] B. Tan, Y. Wu, N. Zhang, B. Zhang, and Y. Chen, "Improvement of ride quality for patient lying in ambulance with a new hydropneumatic suspension," *Advances in Mechanical Engineering*, vol. 11, no. 4, Article ID 2072152868, 2019.

[21] H. Qi, B. Zhang, N. Zhang, M. Zheng, and Y. Chen, "Enhanced lateral and roll stability study for a two-axle bus via hydraulically interconnected suspension tuning," *SAE International Journal of Vehicle Dynamics, Stability, and NVH*, vol. 3, 2018.

[22] W. Tian, Y. Yang, and W. Wang, "The influence of the interconnection forms of hydropneumatic suspension on vehicle stability," *Automotive Engineering*, vol. 39, no. 12, pp. 1362–1367, 2017.

[23] M. Wang, B. Zhang, Y. Chen, N. Zhang, S. Chen, and J. Zhang, "Frequency-based modeling of a vehicle fitted with roll-plane hydraulically interconnected suspension for ride comfort and experimental validation," *IEEE Access*, vol. 8, pp. 1091–1104, 2020.

[24] H. Qi, Y. Chen, N. Zhang, B. Zhang, D. Wang, and B. Tan, "Improvement of both handling stability and ride comfort of a vehicle via coupled hydraulically interconnected suspension and electronic controlled air spring," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 234, 2020.

[25] M. Zheng, P. Peng, B. Zhang et al., "A new physical parameter identification method for two-axis on-road vehicles: simulation and experiment," *Shock and Vibration*, vol. 2015, Article ID 191050, 9 pages, 2015.

[26] C. Zhou, X. Liu, F. Xu, and W. Chen, "Sliding mode switch control of adjustable hydropneumatic suspension based on parallel adaptive clonal selection algorithm," *Applied Sciences*, vol. 10, no. 5, p. 1852, 2020.

[27] C. Zhou and G. Wen, "Hydraulic-electrical energy regenerative semi-active hydropneumatic suspension system based on a modified skyhook damping control algorithm," *Journal of Vibration and Shock*, vol. 37, no. 14, pp. 168–174, 2018.

[28] W. Yue, Y. Shi, A. Peng, and S. Li, "Study on ride comfort of a heavy vehicle based on active hydropneumatic suspension," *Journal of Vibration and Shock*, vol. 35, no. 24, pp. 183–188, 2016.

[29] F. Zhao, J. Guan, L. Gu, Y. Li, and R. Liu, "Control algorithm for vehicle height adjustable system of hydropneumatic suspension based on Lugre model," *Transactions of Beijing Institute of Technology*, vol. 25, no. 1, pp. 49–55, 2016.

[30] Y. Tsuruga, "Study on hydraulic active suspension for wheeled hydraulic excavator," *Proceedings of the IFPS International Symposium on Fluid Power*, vol. 1996, no. 3, pp. 367–372, 1996.

[31] Y. Zhang, H. Chen, K. Guo, X. Zhang, and S. Y. Li, "Electro-hydraulic damper for energy harvesting suspension: modeling, prototyping and experimental validation," *Applied Energy*, vol. 199, pp. 1–12, 2017.

[32] T. Yi, F. Ma, C. Jin, and Y. Huang, "A novel coupled hydropneumatic energy storage system for hybrid mining trucks," *Energy*, vol. 143, pp. 704–718, 2018.

[33] W. Wu, S. Zhang, Z. Zhang, and P. Mućka, "Mathematical simulations and on-road experimental study of the vibration energy harvesting from mining dump truck hydropneumatic suspension," *Shock and Vibration*, vol. 2019, Article ID 4810472, 16 pages, 2019.

[34] J. Petric, "Modeling of hydropneumatic energy storage system," in *Proceedings of the 8th EUROSIM Congress on Modelling and Simulation*, Cardiff, UK, September 2013.

[35] S. Feng, "Determining the air polytropic exponent value related theoretically and practical for bladder accumulator," *Chinese Hydraulics & Pneumatics*, vol. 5, pp. 3–5, 2002.

[36] Y. Wang, B. Wei, and Y. Yang, "Establishment and verification of real gas multivariate index model for hydropneumatic suspension system," *Transactions of the Chinese Society of Agricultural Engineering*, vol. 35, no. 20, pp. 10–16, 2019.

[37] J. Feng, C. Matthews, S. Zheng, F. Yu, and D. Gao, "Hierarchical control strategy for active hydropneumatic suspension vehicles based on genetic algorithms," *Advances in Mechanical Engineering*, vol. 7, 2015.

[38] Automobile Industry Standard of China, *Bench Test Method of Automobile Shock Absorber*, Automobile Industry Standard of China, Beijing, China, 1999.

[39] Y. Chen, B. Zhang, S. Chen, and L. Dong, "Model reduction technique tailored to the dynamic analysis of a beam structure under a moving load," *Shock and Vibration*, vol. 2014, Article ID 406093, 13 pages, 2014.

[40] W. Wang, *Automobile Design*, China Machine Press, Beijing, China, 4 edition, 2004.

[41] R. Wu, *Dynamic modeling, simulation and experimental study of hydro pneumatic suspension system*, Ph.D. dissertation, School of Mechanical Engineering, Zhejiang University, Hangzhou, China, 2000.

[42] Y. Zhang, "Research of hydropneumatic spring suspension by modeling, simulation and test," Master dissertation, School of Mechatronic Engineering, China Mining University, Xuzhou, China, 2003.

[43] Y. Wei, J. Yang, and Y. Rong, "A design to make the damping coefficient of a vehicles absorber suit the damping ratio of a suspension system," *Journal of Wuhan Automotive Polytechnic University*, vol. 06, pp. 24–27, 2000.