Design and Rig Test of Hydro-Pneumatic Spring for Heavy Off-Road Vehicles

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Abstract: In order to meet the requirements of heavy off-road vehicles for long-stroke elastic elements, a new type of double-chamber hydro-pneumatic spring was innovatively designed. According to the structure and working characteristics, the mathematical models of the spring under two working conditions are established, and its stiffness characteristics are analyzed. The force and displacement data of the spring at different frequencies were obtained by rig test. The static friction, damping and elastic force of the spring are separated. Based on the data of elastic force and displacement, the polytropic exponent of hydro-pneumatic spring under different working conditions is identified by curve fitting. The structure and theoretical model of double-chamber hydro-pneumatic spring designed in this paper provide support for the application of hydro-pneumatic spring in long-stroke off-road vehicle.

1. Introduction

The oil-gas spring is a suspension element that uses oil as the force transmission medium and high-pressure gas as the elastic medium, and integrates the functions of elastic elements and shock absorbers \cite{1}. According to the different air chambers, the oil and gas springs can be divided into single air chamber oil and gas springs, double air chamber oil and gas springs and oil and gas springs with back pressure air chambers. Compared with the single-chamber oil-gas spring, the double-chamber gas spring has a variable stiffness characteristic\cite{2,3}.

For heavy off-road vehicles, this paper designs a small damping gas spring, which is used in parallel with the shock absorber to solve the problems of elastic characteristics change and seal life reduction caused by the temperature rise of the gas spring \cite{4,5}. In this paper, a double-chamber oil-gas spring model is established, and the structural parameters are designed. A bench test is carried out on the sample to verify the correctness of the design parameters and the validity of the oil-gas spring model.

According to the target model, the oil-gas spring designed in this paper should meet the theoretical stroke between 620mm and 900mm, and the limit load is 250kN.

The traditional oil-gas spring integrates the elastic element and the shock absorber. The advantage is that it has nonlinear stiffness damping and good vibration damping performance. The disadvantage is that the mechanical energy is converted into heat energy due to the damping, and the oil temperature increases. At the same time, a rubber diaphragm is used to separate the air chamber and the oil. Poor heat dissipation and rubber life problems will lead to a decrease in suspension performance and durability.

The double-chamber oil-gas spring in this paper cancels the damping valve design, which avoids the decrease of spring performance due to temperature rise. And a metal piston is used to separate the air
chamber and the oil chamber to enhance the spring load capacity. At the same time, the use of rubber parts is reduced, so that the spring has good stability and longer service life.

The double-chamber oil-gas spring is characterized by having two air chambers with different working pressures, as shown in Fig.1. The working front part of the spring is similar to that of the single-chamber oil-gas spring. When the load reaches a certain value, the overall stiffness of the spring decreases, which reduces the stiffness in the second half of the stroke. The upper and lower lifting rings are respectively connected with the body and the wheels. There are three chambers in the oil and gas spring, which are the oil chamber, the low-pressure air chamber and the high-pressure air chamber, and the air chamber is filled with high-pressure nitrogen.

![Fig.1 Double air chamber gas spring](image)

2. Mathematical Modeling and Analysis

2.1. Mathematical model establishment

The mathematical model of the double-chamber oil-gas spring in this paper is based on the following assumptions [6]:

- There is no leakage in all parts; the pressure in each gas-liquid chamber is uniform; the elastic deformation of structural parts and the compressibility of oil are not considered; the influence of the quality of moving parts is not considered.

At low load and small stroke, only the low pressure chamber is compressed. According to the laws of thermodynamics, the following relationship exists between pressure and volume when a gas is compressed:

\[ p_{L0}V_{L0}^m = p_LV_L^m = C \]  

\[ p_{L1}A_L = p_{H0}A_H \]  

\[ p_{H0} \] is the initial inflation pressure of the high-pressure air chamber. At this time, the oil and gas spring is in a critical working state. The volume of the low-pressure air chamber in the critical working state is calculated as follows:
\[ V_{L1} = V_{L0} \left( \frac{p_{L0}}{p_{H0}} \right)^{\frac{1}{m}} = V_{L0} (B_p \frac{A_H}{A_L})^{\frac{1}{m}} \]  

(3)

The relationship between the working load \( F \) and the displacement \( s \) of the oil and gas spring of the main and auxiliary air chambers is as follows:

\[
F = \begin{cases} 
A_L p_{L0} \left(1 - \frac{A}{A_L h_{L0}}\right)^m, & s \leq s_1 \\
\frac{p_{L0} A_L}{h_{L0}} \left(1 - \frac{A}{A_L h_{L0}}\right)^{(m+1)}, & s > s_1 
\end{cases}
\]

(4)

The stiffness of the oil-gas spring can be derived from formula (12) with respect to \( s \), and the relationship between stiffness \( K \) and displacement \( s \) is as follows:

\[
K = \begin{cases} 
\frac{p_{L0} A_L}{h_{L0}} \left(1 - \frac{A}{A_L h_{L0}}\right)^{(m+1)}, & s \leq s_1 \\
\frac{p_{L0} A_L}{h_{L0}} \left(1 + B_p A_L h_{L0} \right)^{m} \left(1 + B_H h_{L0} \right), & s > s_1 
\end{cases}
\]

(5)

2.2. Simulation analysis

According to the load and stiffness characteristics of the double-chamber oil and gas spring, a simulation mathematical model is established in MATLAB. The setting parameters are shown in Table 1. Set different initial inflation pressures of high and low pressure air chambers, and set the excitation conditions as sinusoidal excitation with an amplitude of 120 mm and a frequency of 0.1 Hz.

Table 1 Model variable settings

| variable physical meaning | variable name | Initial air column height of high pressure air chamber | Initial air column height of low pressure air chamber | High pressure chamber initial inflation pressure | Low pressure chamber initial inflation pressure |
|---------------------------|--------------|------------------------------------------------------|-----------------------------------------------------|-----------------------------------------------|-----------------------------------------------|
| /mm                       | \( h_{H0} \) | /mm                                                  | /mm                                                 | /MPa                                          | /MPa                                          |
| /mm                       | \( h_{L0} \) | /mm                                                  | /mm                                                 | /MPa                                          | /MPa                                          |

Select an appropriate specific pressure coefficient according to the requirements of the working state of the spring, where the specific pressure coefficient is the ratio of the initial charging pressure of the high-pressure air chamber to the initial charging pressure of the low-pressure air chamber. Change the pressure of the high and low pressure air chambers, draw the stroke-load characteristic curve as shown in Fig.2, and draw the stroke-stiffness characteristic curve as shown in Fig.3.

![Fig.2 Load characteristic diagram](image1)

![Fig.3 Stiffness characteristic map](image2)

According to the mathematical model, the load capacity of the oil-gas spring is positively correlated with the initial pressure of the low-pressure air chamber, the area of the piston, and the initial air column height of the low-pressure air chamber. Within a reasonable specific volume coefficient range, the initial air column heights of different high and low pressure air chambers are simulated, and the stroke-load characteristic curve is drawn as shown in Fig.4, and the stroke-stiffness characteristic curve is drawn as shown in Fig.5.
Under the condition of keeping the initial charging pressure of the high and low pressure air chambers unchanged, as the initial air column height of the high and low pressure air chambers becomes smaller and smaller, the stiffness of the oil and gas springs increases gradually. In the case of a certain specific pressure coefficient, reducing the initial air column height of the low-pressure air chamber will reduce the critical working displacement $s_1$ of the oil-gas spring, and the high-pressure air chamber will enter the working state in advance. Considering the simulation analysis results and actual processing capabilities, the final parameters of the oil and gas spring samples are selected as shown in Table.2.

**Table.2 Selection of sample parameters of oil and gas spring**

| Parameter Name                        | Parameter Value | Parameter Name                        | Parameter Value |
|---------------------------------------|-----------------|---------------------------------------|-----------------|
| Initial air column height of high pressure air chamber /mm | 235             | Low pressure chamber initial inflation pressure /MPa | 5.6             |
| Initial air column height of low pressure air chamber /mm | 140             | High pressure air chamber piston diameter /mm | 70              |
| High pressure chamber initial inflation pressure /MPa | 9.1             | Low pressure air chamber piston diameter /mm | 100             |

3. Test and result analysis

According to the designed parameters, the gas spring test sample is manufactured and installed on the test bench. The ultimate load test, friction test and temperature rise test were carried out to verify whether the designed double-chamber oil-gas spring can meet the needs of heavy-duty off-road vehicles.

3.1. Ultimate load test

Adjust the pressure of the oil-gas spring to the normal equilibrium position, take 715 mm (static equilibrium position) as the center and the amplitude of ±120 mm, and apply sinusoidal excitation at 0.1 Hz to complete the ultimate bearing test of the oil-gas spring.

According to the main piston size, specific volume coefficient and specific pressure coefficient of the oil and gas spring sample, curve fitting can be performed on the ultimate load test data, and the
damping coefficient, friction force and the real gas variability exponent \( m \) can be identified. The oil and gas spring load \( F \) includes elastic force \( F_K \), damping force \( F_C \) and friction force \( F_f \) namely:

\[
F = F_K + F_C + F_f
\]

The first half of the stroke (single-chamber working state) and the second half (two-chamber working state) of the hydraulic spring sample's ultimate load test data were exponentially fitted by MATLAB software. Fig. 8 shows the comparison between the fitting curve of the front and rear strokes and the test curve.

![Fig.8 Fitted curve and test data curve comparison chart](image)

According to the fitting curve of the ultimate bearing test data of the oil-gas spring sample, the damping coefficient, friction force, and gas variability index values of the oil-gas spring sample are identified and obtained as shown in Table 3.

| Parameter identification of oil and gas spring samples | damping coefficient | friction force | gas variability index |
|--------------------------------------------------------|---------------------|----------------|------------------------|
| 120000 N/(m·s\(^{-1}\)) | 3.5 kN | 1.165 |

By correcting the gas variability index, the simulation accuracy can be improved. The modified gas variability index is substituted into the model, and the comparison between the simulation data before and after the modification and the experimental data is shown in Fig. 9.

![Fig.9 Comparison of load curves](image)  ![Fig.10 Comparison of stiffness curves](image)

It can be seen that the modified hydraulic spring load and stiffness simulation curve fits the test data curve better than the modified one. Compared with the traditional oil and gas spring, the damping force is significantly reduced, so the oil and gas spring will also have better temperature rise performance.
The test curve is processed, and the ultimate load test results of the oil-gas spring sample are obtained as shown in Table 5. It can be seen that the maximum load of the test sample is 178 kN at the maximum stroke of 120 mm, which meets the actual use requirements.

### Table 4: The ultimate load test results of the oil-gas spring

| Journey | Stroke | Load |
|---------|--------|------|
| -120mm  | 55.9mm | 29kN |
| 0       |        | 59kN |
| 120mm   |        | 178kN|
| High Pressure Chamber Intervention Point Stroke | 55.9mm |

#### 3.2. Friction Resistance Test

Adjust the pressure of the oil-gas spring to the normal equilibrium position, take the static equilibrium position as the center and the amplitude of 50mm, apply sinusoidal excitation at 0.8Hz and 1.6Hz respectively to complete the friction resistance characteristics of the oil-gas spring, complete the three-time sinusoidal excitation, and obtain the elastic curve of the oil-gas spring as shown in Figure 11.

![Figure 11: Oil-gas spring sample friction resistance test](image)

The test curve is processed, and the friction resistance test results are shown in Table 5.

### Table 5: Oil-gas spring sample friction resistance test

| Frequency (Hz) | First | Second | Third | Mean |
|---------------|-------|--------|-------|------|
| 0.80          | 3.47  | 3.38   | 3.71  | 3.52 |
| 1.67          | 3.73  | 3.27   | 3.87  | 3.62 |

It can be seen that the average values of the three tests of the test sample at 0.8 Hz and 1.6 Hz are 3.52 kN and 3.62 kN, respectively, which are similar to the friction force data 3.5 kN identified in the ultimate load test, and the friction resistance is relatively small, which meets the design requirements.

#### 3.3 Temperature rise test

Adjust the pressure of the oil-gas spring to the normal equilibrium position, take the static equilibrium position as the center, 50mm amplitude, 1.67Hz excitation frequency, and perform sinusoidal excitation for 10 minutes. The test curve is processed, and the temperature rise test is arranged as shown in Table 6.

### Table 6: Oil and gas spring sample temperature rise test

| Ambient Temperature | Temperature After Test | Temperature Difference |
|---------------------|------------------------|------------------------|
| 29.4°C               | 58.4°C                 | 29.0°C                 |

### 4. Conclusion

In this paper, a double-chamber oil-gas spring is designed with a small damping coefficient. It can be
used in parallel with the shock absorber to avoid the change of the elastic characteristics and the reduction of the sealing life caused by the temperature rise of the oil-gas spring. This study can draw the following conclusions:

1. According to the structure and working characteristics, the mathematical model of the oil-gas spring in two working states is established.

2. Carry out a bench test to obtain the force and displacement data of the oil and gas spring at different frequencies. Using elastic force and displacement data, through curve fitting, the gas variability index under different working conditions can be effectively identified.

3. After experiments, the correctness of the design parameters and the validity of the oil-gas spring model are verified. This double-chamber oil-gas spring model can provide design support for oil-gas springs.

References
[1] Sang Z.G., Dong M.M., Qin Y.C., et al. (2017) Modelling and Analysis on the Characteristics of Dual-Chamber Hydro-Pneumatic Suspension. Journal of Beijing Institute of Technology, 37: 401-405.
[2] Dong M.M., Meng H.Q. (2013) Simulation of Stiffness and Damping Characteristics of Hydro-pneumatic Spring and Analysis of Influencing Factors. Vehicle & Power Technology. 02:47-50.
[3] Zhao K., Dong M.M., Wang Z.G., et al. (2017) Vibration-Based Terrain Classification for Autonomous Vehicles. Journal of Beijing Institute of Technology, 26: 440-448.
[4] Liu P., Wang X.Z., Zheng S.H., et al. (2019) Bench Design for Static and Dynamic Characteristics Test of Large Hydro-pneumatic Suspension Cylinders. CHINESE HYDRAULICS & PNEUMATICS, 03: 74-79.
[5] Li M.J., Wu Z.F., Xu G.Z. (2019) The Simulation and Experiment Research on the Influence of Main Parameters of Accumulator on Hydraulic Excitation Table System. CHINESE HYDRAULICS & PNEUMATICS, 09: 70-77.
[6] Liu B.G. (2016) Parameters’ Optimization Design of Nonlinear Hydro-Pneumatic Suspension. Beijing Institute of Technology, MA thesis.