System Design and Characteristic Analysis of Follow-up Hydraulic Muffler Based on Gear-Screw Transmission

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Abstract. In order to improve the attenuation frequency range of traditional static two-stage H-type hydraulic muffler, a two-stage follow-up hydraulic muffler driven by gear-screw is designed based on PLC. The Muffler can change its structure parameters according to the rotational speed of the hydraulic pump, and attenuate the two pulsation frequencies of the hydraulic pump. Considering the structural parameters and gear transmission of the muffler, the static and dynamic characteristics of the muffler are analyzed systematically, and the stability of the muffler is verified by the case simulation test. The results show that the two-stage follow-up hydraulic muffler based on PLC can attenuate the two pressure pulsation frequencies at the same time, and the operation is stable and the response speed is fast. The two-stage follow-up hydraulic muffler has practical significance and value to provides a way for engineering hydraulic system noise reduction.

1. Introduction
Hydraulic muffler is a device used to attenuate pressure pulsation in hydraulic systems. Installing a hydraulic muffler at the outlet of a hydraulic pump is considered an effective and practical way. Traditional static hydraulic muffler has a narrow frequency range of attenuation pressure pulsation. In order to solve this technical problem, people have designed and developed a hydraulic muffler that can change its structural parameters according to the speed of the hydraulic pump. Because the two-stage H-type hydraulic muffler has two mass chambers and two volume chambers, as long as the structural parameters are matched, the inherent pulsation frequency and the backlash pulsation frequency generated by the hydraulic pump can be simultaneously reduced. Therefore, a follow-up device [1] is added on the basis of the secondary hydraulic muffler to widen the attenuation range of the secondary hydraulic muffler. The second-stage follow-up Hydraulic Muffler can achieve the best vibration reduction effect at all times to meet the attenuation requirements of the pressure pulsation of the pump source under different working conditions.

2. Structure Design and Working Principle Based on Gear-screw Transmission Type Two-stage Follow-up Hydraulic Muffler
The schematic diagram is shown in Figure 1. The flow area of the mass chamber has a great influence on the impedance, so the follow-up device is installed in the mass chamber. When the speed of the hydraulic pump changes, the speed sensor transmits the corresponding speed signal to the speed to the PLC. After automatic calculation by PLC, the pulse number and pulse frequency corresponding to the speed signal are given to the servo controller. After receiving the signal, the servo controller transfers
the voltage signal to the servo motor, so as to control the rotation number and speed of the servo motor. After receiving the command, the servo motor drives the gear to rotate through the coupling, and then converts the rotation of the nut to the translation of the ball screw, so as to realize the precise translation of the sliding blocks 1 and 2 to control the flow area of the pipelines in the quality room 1 and 2.

Through the displacement analysis of the system sliding blocks 1 and 2, it is found that the relationship between them can be approximately regarded as a positive proportion (the nut is fixed on the gear and rotates with the gear). In order to simplify the design and facilitate the engineering manufacture, it can be designed that a servo motor drives the driving wheel to drive the driven wheel 1 and the driven wheel 2 to move at the same time.

![Figure 1 Schematic diagram of a gear-screw drive-type secondary follow-up hydraulic muffler](image)

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Figure 1 Schematic diagram of a gear-screw drive-type secondary follow-up hydraulic muffler

The volume chamber of the hydraulic muffler is generally designed as a cylinder, and the length diameter ratio of the volume chamber can be selected between 1:1 and 1.5:1. The mass chamber is approximately regarded as a long orifice. Considering the convenience of actual processing, the quality room is designed to a square shape.

3. Characteristic Analysis of Two-stage Follow-up Hydraulic Muffler Based on Gear-screw Transmission Type

3.1. Static characteristic analysis of two-stage follow-up hydraulic muffler Based on gear-screw transmission type

The two natural frequencies of the two-stage hydraulic muffler are made equal to the natural pulsation frequency of the hydraulic pump and the backstroke pulsation frequency. Based on the pump speed and PLC control technology, the structural parameters of the two-stage follow-up hydraulic muffler with the change of the pump speed are reasonably designed as follows:

\[
\frac{L_v (V_v + V_s)}{d_1 - \frac{c D}{z_1 \pi n_1}} + \frac{L_v V_s}{d_2 - \frac{c D}{z_2 \pi n_1}} \left( \frac{2 \sqrt{2 \pi n_1 z_2}}{30a} \right)^2 = \left( \frac{L_v (V_v + V_s)}{d_1 - \frac{c D}{z_1 \pi n_1}} \right)^2 + \left( \frac{L_v V_s}{d_2 - \frac{c D}{z_2 \pi n_1}} \right)^2 \right] \left( \frac{4 L_v L_v V_s^2}{2} \right)
\]

(1)

In the formula, \( z \) is the number of plungers; \( z_i \) is the number of teeth in the driving gear; \( z_i \) is the number of teeth of the driven wheel 2; \( z_i \) is the number of teeth of the driven wheel 3; \( n_i \) is the speed of the hydraulic pump, \( r/min; L_i \) is the length of the pipeline of the mass chamber 1, m; \( L_i \) is the length of the pipeline of the mass chamber 3, m; \( V_s \) is the volume of the cavity I, \( m^3; V_s \) is the volume of volume II, \( m^3; D \) is the pitch of the lead screw, m; \( c \) is the thread number; \( n_i \) is the
rotational speed of the servo motor, \( r/\text{min} \); \( d_i \) is the initial diameter of the pipe of the mass chamber 1, m; \( d_3 \) is the initial diameter of the pipeline of the mass chamber 3, m.

The maximum speed of a 9-head plunger pump is 2200r/min. Taking the slider position corresponding to 50% \((f_1 = 330 \text{ Hz}, f_2 = 165 \text{ Hz})\) of the maximum speed as the balance position, the research speed range is 50% - 100%. In this study, water is selected as the fluid medium. Set the sliding block to complete accurate positioning within 0.1s. The rated speed of the servo motor is 6000r/min. The resolution is \( j = 2000 \text{ P/R} \). The pitch of the ball screw is 0.8mm. The number of thread is 1. Taking Table 1 as the research object, FIG.2~FIG.5 are obtained.

| Pump speed \( n_p / (r/\text{min}) \) | Natural pulsation frequency \( f_1 / \text{Hz} \) | Kickback pulsation frequency \( f_2 / \text{Hz} \) | Servo motor speed \( n_s / (r/\text{min}) \) | Displacement of sliding stop 1 \( \Delta x_1 / \text{mm} \) | Displacement of sliding stop 2 \( \Delta x_2 / \text{mm} \) | Pulse number \( J_2 / P \) | Pulse frequency \( f_2' / \text{Hz} \) |
|---------------------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| 2200                            | 660             | 330             | 4481.82         | 6.207           | 5.976           | 14939           | 149394          |
| 1980                            | 594             | 297             | 3371.28         | 4.614           | 4.495           | 11238           | 112376          |
| 1870                            | 561             | 281             | 2864.34         | 3.901           | 3.819           | 9548            | 95478           |
| 1760                            | 528             | 264             | 2338.74         | 3.234           | 3.118           | 7796            | 77958           |
| 1650                            | 495             | 248             | 1876.68         | 2.586           | 2.502           | 6256            | 62556           |
| 1540                            | 462             | 231             | 1434.78         | 2.002           | 1.913           | 4783            | 47826           |
| 1430                            | 429             | 215             | 1052.7          | 1.427           | 1.404           | 3509            | 35090           |
| 1320                            | 396             | 198             | 646.2           | 0.916           | 0.862           | 2154            | 21540           |
| 1210                            | 363             | 182             | 318.6           | 0.427           | 0.425           | 1062            | 10620           |
| 1100                            | 330             | 165             | 0              | 0              | 0              | 0              | 0              |

**Table 1** Change of displacement and speed when water is a fluid medium

![Figure 2](image1.png)  
**Figure 2:** Displacement relationship of sliding stop  

![Figure 3](image2.png)  
**Figure 3:** Displacement \( \Delta x_1, \Delta x_2 \) and pulse number \( J_2 \)

![Figure 4](image3.png)  
**Figure 4:** Natural pulse frequency and displacement diagram
Figure 5: Back pulse frequency and displacement diagram

It can be seen from Figure 2 that the displacement of the two sliding blocks (Δx₁ and Δx₂) is linear and the slope of the line is 1.03. It can be obtained that the tooth ratio of two driven wheels is 1.03, combined with the hydraulic muffler size, then the modulus and tooth number of the three gears are obtained. From the comprehensive consideration of processing technology and transmission efficiency [2], the modulus of gear can be 1 mm. The number of teeth of driving wheel and driven wheel 1 is 32, and the number of teeth of driven wheel 2 is 33. As can be seen from FIG. 3, the displacement is directly proportional to the number of impulses. And the slopes of the two lines are 1 and 1.03 respectively. As can be seen from Figure 4 and Figure 5, the relationship between the natural frequency and the backstroke pulsation frequency of the hydraulic system and the displacement of the sliding block is a power function. The attenuation of the two frequencies is the result of the combined action of the displacements of two sliding blocks. As the frequency increases, the displacement difference becomes more and more obvious.

4. The Dynamic Characteristic Analysis of Two-stage Follow-up Hydraulic Muffler Based on Gear-screw Transmission Type

According to the static characteristics analysis of the secondary follow-up muffler, it is necessary to change the structural parameters of the secondary follow-up hydraulic muffler when the speed of the hydraulic pump changes. Therefore, the study of the influence of the dynamic characteristics of the gear [4-6] on the follow-up device is the premise to ensure the reliable operation of the secondary follow-up hydraulic muffler.

4.1. Dynamic equation of gear system [3]

Since the forces on the two driven wheels are the same, they can be approximately regarded as two independent pairs of gears. The dynamic model of the gear system is shown in Figure 6.

Figure 6 Dynamic model of gear pair

According to literature [3], the dimensionless analysis model of gear pair system is as follows:

\[
\begin{bmatrix}
1 & 0 & 0 \\
0 & 1 & 0 \\
-1 & 0 & 1 \\
\end{bmatrix}
\begin{bmatrix}
\ddot{x}_p(t) \\
\ddot{x}_g(t) \\
\ddot{x}(t) \\
\end{bmatrix}
+ \begin{bmatrix}
\zeta_{11} & 0 & 0 \\
0 & \zeta_{22} & 0 \\
0 & 0 & \zeta_{33} \\
\end{bmatrix}
\begin{bmatrix}
\dot{x}_p(t) \\
\dot{x}_g(t) \\
\dot{x}(t) \\
\end{bmatrix}
+ \begin{bmatrix}
\kappa_{11} & 0 & \kappa_{1} \\
0 & \kappa_{22} & -\kappa_{23} \\
0 & 0 & 1 \\
\end{bmatrix}
\begin{bmatrix}
f_p(x_p) \\
f_g(x_g) \\
f_m(x_m) \\
\end{bmatrix}
= [F(t)]
\] (2)

\[
[F(t)] = \begin{bmatrix}
-F_p & 0 & 0 \\
F_g & 0 & \sin(\omega_{th} t + \phi_h) + 0 \\
F_m & F_{sh} \omega_{th}^2 & \sin(\omega_{th} t + \phi_t) \\
\end{bmatrix}
\] (3)

The dimensionless clearance nonlinear function of gear [7] is:

\[
f(x) = \begin{cases}
x + b, & x < -b \\
0, & -b \leq x \leq b \\
x - b, & x > b
\end{cases}
\] (4)
The parameters in the formula are described below:  
\[ \xi_{11} = \frac{c_p}{2m_p\omega_h}, \quad \xi_{22} = \frac{c_h}{2m_h\omega_h}, \quad \xi_{13} = \frac{c_m}{2m_c\omega_h}, \]
\[ \xi_{23} = \frac{c_m}{2m_h\omega_h}, \quad \xi_{33} = \frac{c_m}{2m_c\omega_h}, \quad k_{11} = \frac{\omega_h^2}{\omega_n^2}, \quad k_{13} = \frac{m_c}{m_h}, \quad k_{22} = \frac{\omega_h^2}{\omega_n^2}, \quad k_{23} = \frac{m_c}{m_h}, \quad F_p = \frac{F_{1p}}{m_p\omega_h^2}, \quad F_g = \frac{F_{1g}}{m_g\omega_h^2}, \]
\[ F_n = \frac{F_{1n}}{m_n\omega_h^2}, \quad F_{st} = \frac{F_{1st}}{m_{bst}\omega_h^2}, \quad F_{ab} = \frac{\bar{e}(t)}{b}, \quad m_c = \frac{l_p l_g R_p^2 f + R_g^2 l_p^2 f}{b}, \quad \omega_h = \sqrt{\frac{k_m}{m_c}}, \quad \omega_p = \sqrt{\frac{k_p}{m_p}}, \quad \omega_g = \sqrt{\frac{k_g}{m_g}}. \]

\( \omega_{eh} \) is excitation frequency; \( \omega_{at} \) is variable component frequency of meshing excitation; \( \bar{e} \) is gear error function; \( b \) is length characteristic number; \( m_c \) is equivalent mass; \( c_m \) is meshing damping coefficient; \( k_m \) is meshing stiffness coefficient.

4.2. Visual modeling and simulation of gear system [8-9]

Simulink[10] and the fourth-order Runge-Kutta calculation method are used to model and solve the nonlinear dynamic equation of gear system. The parameters of the system composed of driven wheel 2 and driving wheel are as follows:  
\[ \xi_{11} = 0.28, \quad \xi_{22} = 0.26, \quad \xi_{13} = 0.086, \quad \xi_{23} = 0.084, \quad \xi_{33} = 0.17, \quad k_{11} = 1.01, \quad k_{13} = 0.51, \quad k_{22} = 0.99, \quad k_{23} = 0.49, \quad F_p = F_g = 0, \quad F_m = 0.09, \quad b = 0.07. \]

The initial displacement is \( \Delta x = 0 \). The dimensionless excitation frequencies at the maximum and minimum speed of the servo motor are taken as the research objects. The values are respectively \( \omega_{eh} = 2.147, \quad \omega_{eh} = 0.365 \). The dynamic response of the gear pair system is shown in Figure 7-Figure 12. The parameters of the driven wheel 1 and the driving wheel system are:  
\[ \xi_{11} = \xi_{22} = 0.28, \quad \xi_{13} = \xi_{23} = 0.086, \quad \xi_{33} = 0.17, \quad k_{11} = k_{22} = 1.01, \quad k_{13} = k_{23} = 0.51, \quad F_p = F_g = 0, \quad F_m = 0.09, \quad b = 0.07. \]

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FIG.7 to FIG.18 show the phase diagram, velocity time diagram and vibration displacement time diagram of two gears at different frequencies. As can be seen from the above figure:

(1) The speed of gear system is different under different excitation frequencies. In dimensionless time, the change of velocity after 10 is approximately uniform. $v$ varies with the frequency of excitation. The larger the excitation frequency is, the faster the dimensionless velocity changes.

(2) The vibration displacement of gear system is controllable in a certain range. In dimensionless time, changes very quickly, and after 10, the amplitude of change is very small. This indicates that the system can run stably.
(3) The phase plan of all gear systems is not a closed curve. Motion is retaliatory. The phase trajectory is limited to a certain range (x between 1.02 and 1.25). The size of v is uniformly distributed between -0.25 and 0.25. It doesn't diverge to infinity. It shows that the gear-lead screw servo system is stable.

5. Conclusion
(1) In this paper, a two-stage hydraulic muffler based on gear-screw transmission type is designed. The structural parameters of the muffler vary with the speed of the pump and attenuate the two frequencies at different times. The working frequency range of two-stage follow-up hydraulic muffler is wide, and its frequency adaptability is better than that of static hydraulic muffler.

(2) Simulink module is used to simulate the follow-up system, and its dynamic characteristics are analyzed. The results show that the vibration of the gear-screw drive system has little influence, and the system can operate stably and respond quickly.

(3) The two-stage follow-up hydraulic muffler has a broad prospect in engineering practice. It provides a theoretical reference for the structural design of noise reduction devices in engineering practice.

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