Research on a hydraulic displacement amplifier for a piezoactuator

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Abstract. A piezoactuator-based hydraulic displacement amplifier with two flexible diaphragm spools and one rigid chamber is designed. The elastic characteristics and effective area of the diaphragm spool are theoretically and numerically analyzed. The influencing factors on the effective area of the diaphragm spool are studied by the finite element analysis method. On this basis, a prototype is produced, and its measurement and control system is built. Experimental results show that the hydraulic amplifier prototype can achieve a stroke range of 540 μm and its amplification ratio is approximately 6.

1. Introduction

As an electro-mechanical device, piezoelectric actuators are widely used in different engineering applications, such as precision micro-displacement positioning platforms [1] and servo valves [2], due to their advantages of high positioning accuracy, dynamic response, and wide frequency range. However, the output displacement of the piezoelectric actuator is very small, generally a few micrometres to tens of micrometres [3]. Therefore, its output displacement is often amplified by means of amplification mechanisms to meet applications requiring greater displacement.

According to different principles, amplification mechanisms are divided into two groups: the flexure hinge-displacement amplifier [4] and the hydraulic-displacement amplifier [5].

The common flexure hinge-displacement amplifiers mainly include the lever-type flexure hinge mechanism, Scott-Russell mechanism and bridge-type flexure hinge mechanism, as well as the rhombic mechanism. However, the flexure hinge-displacement amplifier has shortcomings, such as a small load and easy fatigue damage, which limits its application.

The hydraulic amplification mechanism based on the hydraulic transmission principle, which enlarges the micro-displacement by multiplying the area ratio of different pistons, fits in overload conditions and is insensitive to impulses. According to the structure type of the piston, the hydraulic amplification mechanism can be divided into a rigid piston and flexible piston [6].

Wu [7] developed a PZT micro-valve with a rigid piston hydraulic amplifier, and the enlarged stroke reached 37 μm at 150 V. Muralidhara [8] developed a piezoactuator-based prototype micro-actuator with a hydraulic displacement amplification system. A piezoactuator was used to deflect a diaphragm that displaced a certain volume of hydraulic fluid into a smaller-diameter piston chamber, thereby amplifying the displacement at the other end of the rigid piston.

In microelectromechanical systems, hydraulic amplifiers using membranes as flexible pistons are applied extensively to enlarge the stroke of micro-actuation systems. These works primarily concern the...
structural design; thus, the corresponding dynamic analysis is lacking in the literature. In this paper, a piezoactuator-based hydraulic amplifier with two flexible diaphragm spools and one rigid chamber is studied.

2. Structure of the hydraulic amplifier
The structural principle of the diaphragm-type hydraulic amplifier is shown in Figure 1. It consists of a large and small flexible diaphragm spool, a rigid chamber and hydraulic oil. Its working principle is as follows: when the piezoactuator acts on the large diaphragm spool, it makes the diaphragm bend horizontally and squeezes the hydraulic oil in the chamber; the hydraulic oil then pushes the small diaphragm spool to generate a larger displacement, as shown in Figure 1(b).

![Figure 1. Structural principle of the diaphragm-type hydraulic amplifier](image)

3. Theoretical analysis
3.1. Definition of the effective area of the diaphragm spool
The diaphragm spool structure is shown in Fig. 2, which is composed of a concentric hard disk, a rubber diaphragm and a chamber shell. The force transferred by the diaphragm spool under the action of pressure difference is partly balanced by the chamber shell, which fixes the diaphragm; thus, the rest of the force is the output working force of the diaphragm spool. Therefore, the effective area of the diaphragm spool can be understood as the imaginary equivalent area of the piston, rather than the total area of the diaphragm spool.

The effective area of the diaphragm spool can be defined as the ratio of the force acting on the diaphragm spool to the pressure difference acting on the diaphragm spool when the diaphragm spool is in equilibrium. As shown in Figure2, the effective area of the diaphragm spool $A_e$ can be expressed as

$$ A_e = \frac{F}{\rho} $$

![Figure 2. Diaphragm spool structure](image)
3.2. Effective area of the diaphragm spool in the initial position

When the displacement of the diaphragm spool is very small (for example, less than 5% of the diaphragm diameter), the effective area can be regarded as a constant and can be derived by the following two methods:

(1) Method one: deflection formula calculation method

According to the small-displacement elastic theory of a circular thin plate with equal thickness, the maximum deflection \( \omega_{p_{\text{max}}} \) of the diaphragm with a hard disk under the action of fluid pressure \( p \) is [9]

\[
\omega_{p_{\text{max}}} = \frac{p}{64D} \left[ R^4 - r_0^4 + 4R^2r_0^2 \ln \frac{r_0}{R} \right] = \frac{3(1-\mu^2)}{16} \frac{pR^4}{Eh} \left( 1 - \frac{r_0^4}{R^4} + 4 \frac{r_0^2}{R^2} \ln \frac{r_0}{R} \right)
\]

Where \( D = Eh^3/12(1-\mu^2) \) denotes the bending stiffness.

Similarly, the maximum deflection \( \omega_{F_{\text{max}}} \) of the hard disk under the action of the concentrated force \( F \) is

\[
\omega_{F_{\text{max}}} = \frac{3(1-\mu^2)}{4\pi} \frac{FR^2}{Eh} \left[ 1 - \left( \frac{r_0}{R} \right)^2 + 4 \ln^2 \left( \frac{r_0}{R} \right) \right] \]

When \( \omega_{p_{\text{max}}} \) is equal to \( \omega_{F_{\text{max}}} \), according to formula (1), the effective area \( A_e \) is

\[
A_e = \frac{\pi}{4} \frac{R^4 - r_0^4 + 4R^2r_0^2 \ln \frac{r_0}{R}}{R^2 - r_0^2 - 4R^2 \ln^2 \left( \frac{r_0}{R} \right)}
\]

(2) Method two: integral calculation method

Force \( F \) on the diaphragm spool consists of two parts: force \( F_1 \) on the hard disk and force \( F_2 \) on the diaphragm in the annular space between the hard disk and shell.

\[
F = F_1 + F_2
\]

Thus, \( F_1 = p\pi r_0^2 \).

It is assumed that the diaphragm is an ideal elastic element. The force of the annular area formed by taking the increment \( dr \) at radius \( r \) is \( 2\pi r \cdot p \cdot dr \). The force is transmitted to the hard disk and the chamber shell in proportion. Therefore, the force transmitted to the hard disk is

\[
dF_2 = \frac{a_1}{a_1 + a_2} 2\pi r \cdot p \cdot dr
\]

Where \( a_1 = r - r_0 \) is the distance from the hard disk to the unit area and \( a_2 = R - r \) is the distance from the edge of the shell to the unit area.

Therefore,

\[
F_2 = \int_{r_0}^{R} \frac{R - r}{R - r_0} 2\pi r \cdot p \cdot dr = p \cdot \frac{\pi}{3} \left( R^2 + Rr_0 - 2r_0^2 \right)
\]

The resultant force \( F \) on the hard disk is

\[
F = F_1 + F_2 = p \cdot \frac{\pi}{3} \left( R^2 + Rr_0 + r_0^2 \right)
\]

Thus, the effective area \( A_e \) is

\[
A_e = \frac{F}{p} = \frac{\pi}{3} \left( R^2 + Rr_0 + r_0^2 \right)
\]
3.3. Amplification ratio calculation
The large diaphragm and the small diaphragm are made of the same material; however, the large hard disk film is subjected to the concentrated force $F_b$ and the small hard disk film is subjected to the load $F_s$, as shown in Fig. 3.

![Figure 3. Force analysis of the hydraulic amplifier](image)

As shown in Fig. 3, according to equations (2) and (3), the displacement $x_b$ of the large diaphragm spool under the action of $F_b$ and $p$ can be obtained as follows

$$x_b = F_b \cdot \frac{3(1 - \mu^2)}{4\pi E h_b^3} \left[ R_b^3 - r_b^2 \cdot \frac{R_b^3 - r_b^2 + 4R_b^2 \ln \left( \frac{r_b}{R_b} \right)}{R_b^3 - r_b^2} \right] - p \cdot \frac{3(1 - \mu^2)}{16E h_b^3} \left( R_b^4 - r_b^4 + 4R_b^2 r_b^2 \ln \frac{r_b}{R_b} \right)$$

Similarly, the displacement $x_s$ of the small diaphragm spool is

$$x_s = p \cdot \frac{3(1 - \mu^2)}{16E h_s^3} \left( R_s^4 - r_s^4 + 4R_s^2 r_s^2 \ln \frac{r_s}{R_s} \right) - F_s \cdot \frac{3(1 - \mu^2)}{4\pi E h_s^3} \left[ R_s^3 - r_s^2 \cdot \frac{R_s^3 - r_s^2 + 4R_s^2 r_s^2 \ln \left( \frac{r_s}{R_s} \right)}{R_s^3 - r_s^2} \right]$$

Then, the amplification ratio $k$ formula is

$$k = \frac{x_s}{x_b}$$

4. Simulation of the effective area based on FEM
As the hard disk moves, the effective area of the diaphragm is constantly decreasing, but equations (4) and (9) consider that the affected area is constant. Equations (4) and (9) only show a small error when the hard disk displacement is small. In the case of a large displacement, the effective area of the diaphragm can be determined by FEM.

When the small diagram is displaced by the pressure, its effective area also changes, as shown in Fig. 4. The diagram moves from the initial position through A and B to point C, as shown in Fig. 4(a). Therefore, the corresponding output force and effective area are changed, as shown in Fig. 4(b). Additionally, as the hard disk moves from A to C, the output force and effective area of the hard-disk diaphragm decrease accordingly. Similarly, the FEM simulation result of the large diagram is shown in Fig. 5.

As shown in Figs. 4 and 5, when the hard-disk displacement is zero, the error of the effective area of the small diagram analysed by FEM is 0.5 and 4.5%, respectively, compared with the effective areas calculated by equations (4) and (9). Similarly, the error of the effective area of the small diagram is 0.09 and 0.84%, respectively. This result shows the accuracy of the FEM simulation. At the same time, it can also be seen that the calculation result of equation (4) is closer to the FEM result than that of equation (9), so the effective area can be calculated by using equation (4) when the diaphragm is in the initial state. However, as the hard-disk displacement increases, the error of the theoretical equation calculation...
The result will also increase. In this case, the FEM results will be relatively more representative of the true value of effective area of the diaphragm.

\[
\text{Figure 4. FEM simulation result of the small diagram}
\]

The effects of oil pressure and diaphragm thickness on the effective area are shown in Figs. 6. It can be seen that for the same diaphragm, the higher the oil pressure is, the larger the output force and the smaller the change rate in the effective area in relation to the displacement. At the same oil pressure, the thicker the diaphragm is, the larger the change rate in the effective area in relation to the displacement.

\[
\text{Figure 5. FEM simulation result of the large diagram}
\]
5. Experimental Study

The hardware part of the measurement and control system mainly includes the data transmission module, PI power amplifier, amplifier prototype, data acquisition module, control module, and computer installed with LabVIEW. The hardware schematic diagram of the experimental platform is shown in Fig. 7, and an experimental photograph is shown in Fig. 8. Using the LabVIEW software, the prepared control program is run and calculated in the NI CRI09024 control module, and the control signal is generated and fed to the PI power amplifier through the NI CRI09263 data transmission module. Then, the amplified voltage signal is supplied to the prototype amplifier. The grating displacement sensor measures the output displacement of the prototype amplifier, which is recorded in the computer through the NI CRI0 9411 digital acquisition module.

Fig. 9 shows the displacement of the prototype amplifier and the piezoactuator when the control voltage is slowly increased from 0 to 90 V and then slowly reduced to 0 V.

Figure 6. Effect of the chamber pressure and the diaphragm thickness on effective area

(a) Different chamber pressure          (b) Different diaphragm thickness

Figure 7. Hardware schematic

Figure 8. Experimental system

Figure 9. Deformation-voltage diagram
It can be seen from Fig. 9 that the maximum output displacement of the prototype actuator is approximately 540 $\mu$m and the amplification ratio is approximately 6, which is close to the amplification ratio of 6.6 calculated by the above theoretical analysis; this result verifies the correctness of the derived amplification ratio formula. However, due to the presence of micro-leakage, cavitation, compression and other factors in the closed chamber and the displacement loss caused by the non-axial deformation of the rubber diaphragm, the actual amplification ratio is slightly lower than the theoretical analysis.

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