A piezoelectric direct-drive servo valve with a novel multi-body contacting spool-driving mechanism: Design, modelling and experiment

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Abstract
Stack-type piezoelectric actuators, which usually consist of several ceramic layers connected in series, are widely used in piezoelectric direct-drive servo valves (PDDSV). However, poor pulling force capacity of this kind of actuators affects the performances of the direct-drive servo valves. This article presents a new type of PDDSV, whose spool-driving mechanism is composed of a set of independent parts that are not fixed together but are in contact with each other. This multi-body contacting spool-driving mechanism provides bidirectional movement of the spool by a preloaded stack-type piezoelectric actuator and a driving disc spring. This prevents the stack-type piezoelectric actuator from bearing the pulling force due to the inertia and friction of the spool. Design of the proposed servo valve is illustrated in detail and its characteristics are also predicted. Based on a nonlinear dynamic model of the multi-body contacting spool-driving mechanism, a comprehensive dynamic simulation model of the proposed PDDSV is established. Static and dynamic characteristics of the proposed PDDSV have been studied experimentally and good agreements between experimental and simulation results are observed. The dynamic performances of the proposed PDDSV are compared with the existing piezoelectric servo valves, which demonstrate that the proposed PDDSV has satisfactory dynamic characteristics for high-frequency applications.

Keywords
Piezoelectric direct-drive servo valve, stack-type piezoelectric actuator, spool-driving mechanism, hysteresis nonlinearity, mathematical modelling, experimental characteristics

Introduction
Hydraulic actuation systems are widely used in aeronautical and industrial applications where high power density, high dynamic performances, robustness and over-load capability are required. The most important element in a hydraulic actuation system is the servo valve, whose dynamic characteristics greatly affect the performances of the whole hydraulic actuation system. High bandwidth servo valves are required in high-frequency actuation system applications, such as active vibration control, high cycle fatigue test systems for aerospace structures, high-speed motion simulators, and high-speed aerospace actuation systems and high bandwidth flight motion simulators.

The conventional two-stage electro hydraulic servo valves have bandwidth of the order of 50 Hz and they are not suitable for high-speed applications. The limitation of dynamic performances of the torque motor in the first stage limits the performances of the servo valve. With recent advancement in smart materials, bimorph and magnetostriction can be used as the electromechanical transducer for the servo valve. Some researchers adopted piezoelectric actuators and magnetostrictive actuators to drive the first stage by replacing the torque motor. Although the performances of these two-stage servo valves were improved (from 150 Hz to 280 Hz at −3 dB), their frequency response cannot be further improved due to compliance and fluid inertia of the hydraulic amplifier.

The need for faster actuating systems has led to the development of single-stage servo valves, where a single directly controlled system generates the force...
which is necessary to shuttle the valve spool and they are called 'direct-drive servo valves' (DDSVs). Piezoelectric materials are widely used in the DDSVs because of their high power density, high frequency and large output force. Existing piezoelectric DDSVs (PDDSVs) can be classified into two types. The first is the PDDSV driven by the piezoelectric actuator with a displacement-amplifying mechanism, which is used to increase the flow rate of servo valves.\textsuperscript{15–17} Despite this type of PDDSV breaking the bandwidth limitation of the torque motor and hydraulic amplifier, the flexibility of displacement-amplifying mechanisms still limits the frequency characteristics of these servo valves. Developments in actuator technologies have resulted in a new generation of stack-type piezoelectric actuators that have larger strokes to meet the design requirements of servo valves. Therefore, the second type of PDDSV was developed whose spool is driven by a stack-type piezoelectric actuator without the displacement-amplifying mechanism.\textsuperscript{15–17} Compared with the first type of PDDSV, this one has better dynamic characteristics because it has no displacement-amplifying mechanism. However, the pulling force capacity of the stack-type piezoelectric actuator is very poor, so Yokota and Hiramoto\textsuperscript{13,15} and Li\textsuperscript{16} adopted two stack-type piezoelectric actuators to push the spool alternately, which produces slide valve bidirectional movement. This has two disadvantages: (a) high cost of using two stack-type piezoelectric actuators and (b) difficulty in controlling the servo valve. Shen et al.\textsuperscript{17} utilised a spring plate mechanism and a single non-preloaded multi-layer piezoelectric actuator to realise the bidirectional spool control. Although it economises one piezoelectric actuator, the piezoelectric actuator is fixed together with the spool and extension rod, which results in another two disadvantages: (a) the piezoelectric actuator bears the pulling force generated by the inertia and friction of the spool-driving mechanism, while the pulling force is especially harmful to the non-preloaded stack-type piezoelectric actuator and (b) the piezoelectric actuator bears the lateral force generated by the poor concentricity between the piezoelectric actuator and the spool, which is also harmful to the stack-type piezoelectric actuator. These disadvantages reduced the performances of the PDDSV and even caused damage to the stack-type piezoelectric actuator.

In this article, a PDDSV with a novel multi-body contacting spool-driving mechanism is proposed. The multi-body contacting spool-driving mechanism is composed of several independent parts that are not fixed together, but are in contact with each other. The core component of the multi-body contacting spool-driving mechanism is a preloaded stack-type piezoelectric actuator that is first used as a spool-driving actuator for PDDSV. This preloaded stack-type piezoelectric actuator and a driving disc spring provide the pushing force to shuttle the spool. Therefore, the stack-type piezoelectric actuator, which is not fixed with the spool, does not bear the pulling force due to the inertia of the spool-driving mechanism or the lateral force generated by the poor concentricity between the piezoelectric actuator and the spool. Besides, the preloaded force of the piezoelectric actuator is enough to endure the pulling force due to its own friction. The proposed multi-body contacting spool-driving mechanism can provide better protection for the stack-type piezoelectric actuator and improve the performances of the PDDSV.

The purpose of this article is to illustrate the development research on the proposed PDDSV. The article is organised as follows. In the following section, the detailed design and theoretical characteristic estimation of the proposed PDDSV are described. The following section presents a comprehensive dynamic model of the proposed PDDSV, based on the non-linear dynamic model of the multi-body contacting spool-driving mechanism and considering the piezoelectric power amplifier’s dynamic characteristics. The subsequent section presents an experimental study of the static and dynamic characteristics of the proposed PDDSV and compares experimental and simulation results. The study concludes with final remarks in the last section.

**Design of the PDDSV**

**Operating principle of the PDDSV**

As shown in Figure 1, the spool-driving mechanism of the proposed PDDSV consists of four components: a piezoelectric actuator fixed with actuator rod, a spool, a disc spring rod and a driving disc spring. The four components are not fixed together but are in contact with each other to drive the spool which is called 'multi-body contacting spool-driving mechanism'. Before using the proposed servo valve, the driving disc spring should be preloaded in order to make the components of the spool-driving mechanism contact each other, then the null of the proposed servo valve is adjusted when half of the full-scale drive voltage is applied to the piezoelectric actuator. In addition, the piezoelectric actuator can generate a displacement proportional to the drive voltage (proportional to the amount of electric charge, to be exact) so that the spool position is approximately proportional to the drive voltage. Thus, when the drive voltage given to the piezoelectric actuator increases from half of the full scale, the piezoelectric actuator pushes the spool movement to the right from the null position and generates deformation of the disc spring. In this case, the hydraulic supply oil flows from port P (oil supply port) to port A (control port) and the oil of port B (the other control port) flows into port T (oil return port). In contrast, when the drive voltage given to the piezoelectric actuator decreases from half of the full scale, the spool...
moves to the left from the null position by the force of the disc spring. In this case, the hydraulic supply oil flows from port P to port B and the oil of port A flows into port T. With a constant supply of pressure, the outflow rate of the PDDSV is obtained, which is approximately proportional to drive voltage applied to the piezoelectric actuator.

The preloaded stack-type piezoelectric actuator

Configuration of the preloaded stack-type piezoelectric actuator. The stack-type piezoelectric actuator comprises several ceramic layers that are electrically connected in parallel and mechanically connected in series. The main advantage of using a stack-type piezoelectric actuator is its large output displacement, high generative force and very fast response coupled with the low driving voltage. However, the piezoelectric stacks cannot withstand the pulling force. In order to obtain better reliability in practical applications, the piezoelectric stacks are preloaded by a preloaded spring in series, which can make the piezoelectric actuator bear some pulling force. The displacement of the piezoelectric actuator is exported by a displacement-out slider. Therefore, the piezoelectric stacks, preloaded spring and displacement-out slider compose the preloaded stack-type piezoelectric actuator as shown in Figure 2(a).

The preloaded stack-type piezoelectric actuator used in the proposed PDDSV is a commercially available piezoelectric actuator with an integrated strain gauge sensor (SGS), produced by Physik

Figure 1. Schematic representation of the present PDDSV.
PDDSV: piezoelectric direct-drive servo valves.

Figure 2. (a) Schematic of the preloaded stack-type piezoelectric actuator and (b) photograph of the preloaded stack-type piezoelectric actuator.
Instrumente (model number P-843.40) as shown in Figure 2(b). The high-precision SGS is used to measure the displacement of the piezoelectric actuator. The drive voltage is applied to the piezoelectric actuator by the drive voltage cable. Parameters of the piezoelectric actuator are listed in Table 1.

Theoretical estimation of resonance frequency and displacement.

(a) Resonance frequency

Resonance frequency is an important parameter to be measured for predicting performances in high-frequency applications. With reference to Figure 2(a), the preloaded piezoelectric actuator can be equivalent to a mass–spring system, as shown in Figure 3, because the piezoelectric stacks can be seen as a spring whose stiffness can be expressed as

\[ k_p = E_p \frac{A_p}{l_p} \quad (1) \]

Considering that the left spring \( k_p \) is tensile and the right spring \( k_{ps} \) is compressed in Figure 3, the two springs are in parallel. The effective stiffness of the simplified mechanical model can be described as

\[ k_a = k_p + k_{ps} \quad (2) \]

The effective mass of the piezoelectric actuator is given by

\[ m_a = \frac{m_p}{3} + m_d \quad (3) \]

where \( m_p = \rho_p A_p l_p \) is the mass of the piezoelectric stacks.

Thus, the resonance frequency of the preloaded piezoelectric actuator can be calculated using the following equation:

\[ f_{ra} = \frac{1}{2\pi} \sqrt{\frac{k_a}{m_a}} \quad (4) \]

Substituting the parameters in Table 1 into equations (1) to (4) yields \( f_{ra} = 9400 \text{ Hz} \).

(b) Displacement

For simple estimates of the displacement, the nonlinearity of the piezoelectric actuator’s displacement is not considered here. According to the piezoelectric constitutive fundamental equations, the displacement of the no-load piezoelectric stacks can be expressed as

\[ x_p = n_p d_{33} u_d \quad (5) \]

where \( u_d \) is the drive voltage applied to the piezoelectric stacks.

The existence of the preloaded spring affects the displacement of the piezoelectric stacks. Applying Hooke’s law to the simplified mechanical model as shown in Figure 3, the displacement of the preloaded piezoelectric actuator is given by

\[ x_a = x_p \left( \frac{k_p}{k_p + k_{ps}} \right) \quad (6) \]

Substituting equation (1), equation (5) and the parameters in Table 1 into equation (6) gives \( x_a = 59.7 \text{ µm} \).

Measurement of resonance frequency and displacement.

(a) Resonance frequency

One method of determining the effect of bias condition and drive level on transducer performances is to analyse input electrical impedance versus frequency. The resonance frequency of the transducer is measured at the point of minimum impedance and the anti-resonant frequency at the point of minimum impedance. For the piezoelectric actuators, the resonance frequency can be measured at the point of maximum capacitance, because piezoelectric stacks can be seen as capacitances in parallel. This is accomplished...
with a swept sine excitation over the bandwidth of operation (500 mV, 100–1000 Hz) using a Wayne Kerr precision impedance analyzer (type number 6500B) as shown in Figure 4. The resonance plot of capacitance versus frequency for the piezoelectric actuator is shown in Figure 5.

From the plot, the resonance frequency of the piezoelectric actuator is observed from the peak of capacitance as 9533 Hz against the estimated value of 9400 Hz.

(b) Displacement

The quasi-static characteristic of the piezoelectric actuator has been tested, as shown in Figure 6. The signal generator generates a triangular control voltage whose amplitude is full scale and frequency is 0.05 Hz (shown in Figure 7). The signal is amplified to the drive voltage by the power amplifier, which is composed of the chassis (Physik Instrumente E501.00) and the piezomultiplexer module (Physik Instrumente E504.00F) and the drive voltage is supplied to the piezoelectric actuator. The displacement signal from the SGS and the control signal are both sent into the signal conditioning box, which provides signal conditioning functions and a signal interface to an industrial computer; and then the two signals are recorded by an analogue/digital (A/D) card (Advantech PCI1716) installed in the industrial computer. After simple data processing, the relationship between the displacement and the drive voltage is plotted in Figure 8.

Figure 8 shows that the relationship between the displacement and drive voltage is nonlinear, which is known as piezoelectric hysteresis. When drive voltages are 0 V and 100 V the displacements are observed at 0 µm and 59.3 µm, respectively, which matches the estimated values of 0 µm and 59.7 µm.

Design of the multi-body contacting spool-driving mechanism

As shown in Figure 1, the multi-body contacting spool-driving mechanism is the core part of the proposed PDDSV, because its dynamic characteristics directly determine the performances of the servo valves. Therefore, the design and analysis of the multi-body contacting spool-driving mechanism are
important. Based on the dimensions of the piezoelectric actuator, the multi-body contacting spool-driving mechanism is constructed as shown in Figure 9; it consists of a preloaded piezoelectric actuator, an actuator rod, a spool, a disc spring rod, a driving disc spring and a spring base. All the components are not fixed but are in contact with each other, except that the piezoelectric actuator is fixed to the actuator rod by threaded connection.

**Selection and analysis of the driving disc spring.**

(a) Theoretical estimation of stiffness

The multi-body contacting spool-driving mechanism shown in Figure 9 can be simplified as a mass–spring system (shown in Figure 10) without considering the friction and flow force acting on the spool-driving mechanism.

The effective mass of the simplified mass–spring system in Figure 10 is given by

\[ m_{sdm} = m_a + m_r \]  

where \( m_r \) is the total mass of rods, including the actuator rod, spool and disc spring rod.

In order to satisfy the design requirement of no-load flow rate (5 L/min when the hydraulic supply pressure is 21 MPa and the spool diameter \( d_s \) is 9.4 mm), \( x_e \) should be >50 \( \mu \)m. According to equation (8), \( k_{ds} \) should be <3.22 \( \times 10^6 \) N/m. Here, we chose a disc spring produced by Mubea Company (part number 170025) whose stiffness and stroke meet the design requirements. The dimensioned figure of the driving disc spring is shown in Figure 11, and its dimensions and material properties are listed in Table 2.

(b) Modal analysis

According to the simplified mass–spring system of the spool-driving mechanism shown in Figure 10, the natural frequency of the driving disc spring affects the bandwidth of the servo valve. The natural frequency should be much higher than the bandwidth of the servo valve. In order to analyse the natural frequency of the driving disc spring, modal analysis is carried out by applying Hooke’s law to the simplified mechanical model, the effective displacement of the piezoelectric actuator that suffered the driving disc spring force can be calculated by

\[ x_e = \frac{k_p}{k_p + k_{ds} + k_ps} \]  

\[ (8) \]
out using ANSYS V12.0. The material properties of the driving disc spring are given in Table 2. According to the working condition of the disc spring, its external edge (diameter \( D_e \) as shown in Figure 11) is constrained in all degrees of freedom. The first mode of the driving disc spring is shown in Figure 12.

From the analysis result, the natural frequency of the first modal is found to be 52,264 Hz (far greater than the resonance frequency of the piezoelectric actuator 9533 Hz) and the first mode shape meets the deformation law when it drives the spool. Therefore, this type of disc spring does not limit the frequency response of the spool-driving mechanism and it can satisfy the design requirement of the PDDSV.

**Resonance frequency of the multi-body contacting spool-driving mechanism**

Resonance frequency is one important parameter to predict the performances in high-frequency applications and to estimate time response. The multi-body contacting spool-driving mechanism, as a second-order system which is shown in Figure 10, is characterised by the first resonance frequency at

\[
f_{rs} = \frac{1}{2\pi} \sqrt{\frac{k_{sdm}}{m_{sdm}}} \tag{9}\n\]

where \( k_{sdm} \) is the effective stiffness of the spool-driving mechanism

\[
k_{sdm} = k_p + k_{ds} + k_{ps} \tag{10}\n\]

Using the parameter \( m_r = 0.03 \text{ kg} \), and substituting \( m_r, m_p, k_p, k_{ds}, k_{ps} \) and equations (7) and (10) into equation (9), yields \( f_{rs} = 4045 \text{ Hz} \). The bandwidth of this PDDSV is around 1300 Hz (one-third of the resonance frequency) so that it can be used in the high-frequency domain.

**Design of the spool-preloading mechanism and null-adjusting mechanism**

Based on the design of the multi-body contacting spool-driving mechanism in the section ‘Design of the multi-body contacting spool-driving mechanism’, the PDDSV has been constructed as shown in Figure 13(a). Because of the utilisation of the multi-body contacting spool-driving mechanism in the proposed PDDSV, two special mechanisms are designed to realise the servo valve’s function: the spool-preloading mechanism and the null-adjusting mechanism. In this section, the two mechanisms are illustrated with reference to the cutaway view of the

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**Table 2. Material properties and dimensions of disc spring.**

<table>
<thead>
<tr>
<th>Properties and dimensions</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>50 Cr</td>
</tr>
<tr>
<td>Density ( \rho_{ds} ) (kg/m(^3))</td>
<td>( 7.82 \times 10^3 )</td>
</tr>
<tr>
<td>Young’s modulus ( E_{ds} ) (N/m(^2))</td>
<td>( 2.06 \times 10^{11} )</td>
</tr>
<tr>
<td>Poisson’s ratio ( \mu )</td>
<td>0.29</td>
</tr>
<tr>
<td>Stiffness ( k_{ds} ) (N/m)</td>
<td>( 3.0453 \times 10^6 )</td>
</tr>
<tr>
<td>Diameter of inner edge ( D_i ) (m)</td>
<td>( 12.5 \times 10^{-3} )</td>
</tr>
<tr>
<td>Diameter of external edge ( D_e ) (m)</td>
<td>( 6.2 \times 10^{-3} )</td>
</tr>
<tr>
<td>Thickness of the disc ( \delta ) (m)</td>
<td>( 0.7 \times 10^{-3} )</td>
</tr>
<tr>
<td>Maximum stroke ( h ) (m)</td>
<td>( 0.3 \times 10^{-3} )</td>
</tr>
</tbody>
</table>

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*Figure 12. First mode of the driving disc spring.*
servo valve, shown in Figure 13(b) and its partial enlarged view shown in Figure 13(c).

For this PDDSV, the driving disc spring should be preloaded in order to make the components of the spool-driving mechanism contact each other. As shown in Figure 13(c), preloading of the driving disc spring is realised by the preloading screw. Then the preloading screw is locked by the locknut.

After the multi-body contacting spool-driving mechanism is preloaded, the null of the PDDSV is adjusted by the null-adjusting mechanism composed of null-adjusting screws, valve sleeve pushing ring and null-adjusting disc spring as shown in Figure 13(c). Null adjusting consists of: (a) setting the input hydraulic supply to 21 MPa and supplying half of the full drive voltage (50 V) to the piezoelectric actuator; (b) adjusting the null-adjusting screws to push the valve sleeve movement to the left through the valve sleeve pushing ring until the flow rate to the load is zero and (c) locking the null-adjusting screws with the locknuts. The two null-adjusting disc springs in series determine the adjustment range. The processing and assembly precision should ensure that the null position of the servo valve can be found in this adjustment range.

**Modelling of the PDDSV**

**Nonlinear dynamic model of the multi-body contacting spool-driving mechanism**

In the section ‘Design of the multi-body contacting spool-driving mechanism’, the characteristics of the multi-body contacting spool-driving mechanism were estimated. However, some details, such as the hysteresis nonlinearity of the piezoelectric actuator, the friction and the flow force of spool, were not considered. In order to carry out more in-depth analysis of the spool-driving mechanism, the nonlinear dynamic model is established in this section.

In addition to hysteresis nonlinearity, the piezoelectric effect also displays an electrical and mechanical coupling behaviour. Goldfarb and Celanovic\(^\text{19}\) provided a schematic representation for the electromechanical coupling behaviour of the piezoelectric stack-type actuator. Referring to Goldfarb's theory,
the electromechanical coupling model of the spool-driving mechanism is established, as shown in Figure 14.

In Figure 14, \( u_d \) is the drive voltage applied to the piezoelectric actuator. \( H \) represents the hysteresis effect and \( u_b \) is the voltage due to this effect. The piezoeffect is represented by \( T_{em} \), which is an electromechanical transducer with electromechanical coupling coefficient \( T_{em} = d_{33} \mu_0 k_p \).\(^{18}\) The capacitance \( C_p \) is the sum of the individual piezowafers, which are electrically connected in parallel. The total current flow through the circuit is \( q_p \) and \( q \) may be seen as the total charge in the piezoelectric actuator. The charge \( q_p \) is the transduced charge from the mechanical side. The voltage \( u_p \) is due to the piezoeffect. The force \( F_p \) is the transduced force from the electrical side. \( b_p \) and \( b_s \) are the viscous damping coefficients of the piezoelectric actuator and spool-driving rods (piezoelectric rod, spool and disc spring rod), respectively. \( x_s \) is the spool displacement. \( k_s \) is the equivalent stiffness of the flow force when the servo valve is at no-load status.

For a servo valve with valve port pressure drop \( P_v \), its flow force can be expressed as\(^{20}\)

\[
F_f = 2C_dC_vWx_{ds}P_v \cos \theta_f
\]  

where \( C_d \) is the flow coefficient of the valve port, \( C_v \) is the velocity coefficient of the valve port, \( W = \pi d_s \) is the flow area gradient of the valve port and \( \theta_f \) is the flow jet angle.

When the servo valve is at no-load status, \( P_v \) is a constant \((P_v = P_s/2)\) and \( F_f \) is directly proportional to \( x_s \). In this case, \( F_f \) can be seen as a spring force whose stiffness \( k_f \) is given by

\[
k_f = C_dC_vWP_s \cos \theta_f
\]  

where \( P_s \) is the hydraulic supply pressure.

According to the differential equation similarity of linear systems, the electrical circuit of the linear system can be decided, which is called ‘the analogical method of linear system’.\(^{21}\) Based on the analogical method of the linear system, the mechanical behaviour of the spool-driving mechanism can be analogised by its analogical electrical circuit. Therefore, the electromechanical coupling model of the spool-driving mechanism, as shown in Figure 14, can be described by the analogical electrical circuit shown in Figure 15. This can more easily explain the power flow through the spool-driving mechanism and establish the state equations more conveniently.

In the analogical electrical circuit of the spool-driving mechanism, the piezoeffect is represented by an ideal transformer with turns ratio \( 1/T_{em} \). \( F_p \) is seen as a voltage. The masses of \( m_a \) and \( m_t \) are analogised by inductances. The analogical electric elements of the spring \( k_p \), \( k_s \), \( k_{ps} \) and \( k_{ds} \) are capacitances whose values are \( 1/k_p \), \( 1/k_s \), \( 1/k_{ps} \) and \( 1/k_{ds} \). The damping \( b_p \) and \( b_s \) are seen as resistances and the velocity \( \dot{x}_s \) is seen as the current of the analogical electrical circuit of mechanical part. \( F_{p0} \) and \( F_{d0} \), which are the preload forces of the preload spring and the driving disc spring, are represented by constant voltage sources.

With reference to Figure 15, the complete set of the nonlinear dynamic equations of the multi-body contacting spool-driving mechanism is

\[
u_d = u_b + u_p
\]  

\[
g = H(u_b)
\]  

\[
u_p = F_p/T_{em}
\]  

\[
g = q_p + C_p u_p
\]  

\[
q_p = T_{em} \dot{x}_s
\]  

\[
(m_a + m_f) \frac{d^2 x_s}{dt^2} + (b_p + b_s) \frac{dx_s}{dt} + (k_p + k_f + k_{ps} + k_{ds}) x_s + F_{p0} + F_{d0} = F_p
\]  

where \( H \) represents the hysteresis operator.

Figure 14. Electromechanical coupling model of the spool-driving mechanism.
Identification of the hysteresis operator of the piezoelectric actuator

The hysteresis effect is an inherent characteristic of the piezoelectric material, whose existence limits the piezoelectric actuator’s performances. Researchers have proposed a mathematical model that can describe piezoelectric hysteresis, such as the Preisach model,22 the Duhem model,23 the Maxwell slip model19 and the Prandtle–Ishlinskii model.24 The number of parameters in these models is large. Therefore, to conveniently describe the hysteresis effect of piezoelectric actuator, Adriaens25 adopted a first-order differential equation with only three parameters and provided the identification method for the three parameters. The first-order differential equation is given by

\[ \dot{q} = a[u_h](au_h - q) + b u_h \]  
(19)

With reference to Adriaens,25 the hysteresis curve of \( H \), which is the relation between \( u_h \) and \( q \) as shown in equation (19), should be obtained to identify the three parameters. For the proposed piezoelectric actuator which is integrated with SGS, the hysteresis curve between displacement \( x_a \) and drive voltage \( u_d \) can be easily obtained as shown in Figure 8.

Combining equations (15) to (17) yields

\[ q = T_{em} x_a + \frac{C_p}{T_{em}} F_p \]  
(20)

\[ u_h = u_d - \frac{F_p}{T_{em}} \]  
(21)

Considering the piezoelectric actuator is at no-load status, \( F_p \) can be expressed as

\[ F_p = k_p x_a \]  
(22)

Substituting equation (22) into equations (20) and (21) yields

\[ q = \left( T_{em} + \frac{k_p C_p}{T_{em}} \right) x_a \]  
(23)

\[ u_h = u_d - \frac{k_p}{T_{em}} x_a \]  
(24)

Combining equations (23) and (24) and the relationship of \((u_d, x_a)\) as shown in Figure 8, the nonlinear hysteresis curve between \( u_h \) and \( q \) is obtained, as shown in Figure 16.

With reference to the centre point, average slope and hysteresis area, the parameters in equation (19) can be given by25

\[ a = \frac{q_c}{u_{hc}} \]  
(25)
Table 3. Parameters of the hysteresis model of the piezoelectric actuator.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$ (C/V)</td>
<td>$5.5 \times 10^{-4}$</td>
</tr>
<tr>
<td>$b$ (C/V)</td>
<td>$4.88 \times 10^{-5}$</td>
</tr>
<tr>
<td>$\alpha$ (V$^{-1}$)</td>
<td>$8.5 \times 10^{-3}$</td>
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Based on the analysis of the amplitude characteristics of E504.00F shown in Figure 17, the undamped natural angular frequency $\omega_n$ and the damping ratio $\zeta$ can be determined as: $\omega_n = 4.7124 \times 10^3$ rad/s and $\zeta = 0.707$. The amplitude characteristics of equation (28) are also plotted in Figure 17.

From Figure 17, the amplitude characteristic of the approximate second-order system agrees well with the power amplifier, which proves that the second-order system can be used to describe the dynamic characteristic of the power amplifier.

**No-load flow rate**

The load flow rate of the PDDSV can be expressed by the classical equation of traditional servo valve’s load flow, which is described as

$$Q_l = C_d W x_s \sqrt{\frac{1}{\rho_f} \left( \frac{P_s}{X_s} - P_L \right)}$$  \hspace{1cm} (29)

where $\rho_f$ is the density of the fluid and $P_L$ is the load pressure difference.

When the PDDSV is at no-load status ($P_L = 0$), the no-load flow rate can be written as

$$Q_{nl} = k_d x_s$$  \hspace{1cm} (30)

where $k_d = C_d W \sqrt{P_s/\rho}$ is the flow gain of the PDDSV.

With reference to the nonlinear dynamic model of the multi-body contacting spool-driving mechanism and the piezoelectric power amplifier’s dynamic characteristic, a dynamic block diagram of the PDDSV is obtained as shown in Figure 18.

In Figure 18, $G(s)$ is the transfer function representing the spool dynamics that is derived by equation (18) and is expressed as

$$G(s) = \frac{1}{(m_i + m_s)s^2 + (b_i + b_l)s + (k_p + k_l + k_m + k_{dl})}$$  \hspace{1cm} (31)

**Experimental characterisation**

**No-load flow rate characteristic**

The no-load flow rate is the most basic characteristic of a servo valve. For the proposed PDDSV, the main parameter measured in static characteristic test is the no-load flow rate versus the drive voltage. The standard servo valve test bench shown in Figure 19 was used to measure the no-load flow rate of the proposed PDDSV.

As shown in Figure 19, the hydraulic supply with instrumentation is the hydraulic test bench (1) for performance testing of servo valves. The PDDSV (2) under test is mounted on the manifold on the
The industrial computer (3) generates a triangular signal by a digital/analogue (D/A) card (Advantech PCI1723). The triangular signal conditioned by signal conditioning box (4) is amplified to the drive voltage of piezoelectric actuator by the power amplifier (Physik Instrumente E504.00F) (5). The no-load flow rate is evaluated on the flow test bench by using a gear-type flow meter, which is not visible in Figure 19(b). The signal from the flow meter sensor is sent to a signal conditioning box (4) and acquired by the A/D card (Advantech PCI1716) in an industrial computer (3). Then, the triangular signal and the flow rate signal are both recorded by the industrial computer (3) and plotted to give the no-load flow curve.

The PDDSV was mounted on the standard valve test bench. The input hydraulic supply was set to 21 MPa. According to section ‘Design of the spool-preloading mechanism and null-adjusting mechanism’, the null was adjusted when the drive voltage to the piezoelectric actuator is 50 V. A triangular voltage signal of amplitude 5 V and frequency 0.05 Hz, biased at 5 V, is input to the power amplifier. The data obtained from the flow meter were plotted against Figure 19.
the input voltage of the power amplifier, as shown in Figure 20. It is observed that a no-load flow rate of 4.45 L/min was obtained at the maximum input voltage of 5 V from null.

The no-load flow rate was simulated by the block diagram shown in Figure 18 and the parameters in simulation are listed in Table 4. From Figure 20, the simulated no-load flow rate curve agrees well with the experimental value. This validated the hysteresis operator of the piezoelectric actuator obtained in the section ‘Identification of the hysteresis operator of the piezoelectric actuator’.

Dynamic characteristics

The dynamic characteristic test of conventional servo valves is realised by mounting the valve on a dynamic cylinder that has low friction and inertia. The valve’s flow rate is measured indirectly by recording the output velocity of the piston in the dynamic cylinder.

The dynamic characteristics (frequency characteristic and time response) of the servo valve can be carried out by giving swept sinusoidal and step input signals to the servo valve. However, this dynamic test method is not applicable for the PDDSV because: (1) the bandwidth of the oil flow through the servo valve has a range around 1000 Hz which exceeds the bandwidth of the dynamic cylinder and (2) the fluid volume in the dynamic cylinder is too large, which will impact the dynamic characteristics of the servo valve.

In order to accurately test the dynamic characteristics of the proposed PDDSV, a test setup designed specifically for PDDSV by Lindler and Anderson12 was rebuilt in this study as shown in Figure 21(a). In this test setup, a hydraulic source supplies pressure to the proposed PDDSV (7) installed on a manifold block (2) by oil supply pipe (8) and return pipe (6). A variable orifice (4) is used to present a load. The pressure transducers (1) and (3), which are connected to the manifold, are used to measure the control pressures. This test setup offers an advantage for characterising the PDDSV, which results from the fact that the control ports are connected to a steel pipe (5) with extremely small volume. The small volume connected to the control ports prevents the flow from limiting the response of the PDDSV.

The schematic of the dynamic characteristics test system is shown in Figure 21(b), which consists of the dynamic characteristic test setup and the measuring system (signal generator, industrial computer, signal conditioning box and power amplifier). The test signal \( u_i \) is generated by a signal generator and amplified to the drive voltage by the power amplifier. The test signal \( u_i \), spool displacement \( x_s \) and pressures of control ports \( P_A, P_B \) are conditioned by the signal conditioning box and then acquired by an A/D card (Advantech PCI 1716) in an industrial computer.

Frequency characteristics. For servo valves, frequency characteristics are measured as an amplitude attenuation of the load flow rate to swept sinusoidal input signal of amplitude (one-quarter of the maximum value). The no-load flow rate is expressed by equation (30), which reveals that the no-load flow rate is proportional to the spool displacement. Therefore, the frequency characteristic can be tested by measuring the displacement of the piezoelectric actuator when the swept sinusoidal drive voltage is applied to the piezoelectric actuator.

The dynamic characteristics test system shown in Figure 21(b) was used to test the frequency characteristic of the proposed PDDSV. The variable orifice was set to open. The input hydraulic supply was set to 21 MPa. The sinusoidal signal of the amplitude (one-quarter of the maximum = 1.25 V), which is biased at 5 V, is generated and amplified to the drive voltage with amplitude (12.5 V) biased at 50 V by the power amplifier. The frequencies of the sinusoidal signal are chosen as 1 Hz, 2 Hz, 5 Hz, 10 Hz, 20 Hz,
50 Hz, 100 Hz, 200 Hz, 300 Hz, 400 Hz, 500 Hz, 600 Hz, 650 Hz, 700 Hz, 705 Hz, 710 Hz, 720 Hz, 730 Hz, 740 Hz and 750 Hz. The sinusoidal signal $u_i$ and the spool displacement $x_s$ are both recorded by the industrial computer. By applying a fast Fourier transform (FFT) to the measured data, the frequency characteristics of the proposed PDDSV are plotted as shown in Figure 22. From Figure 22, the measured frequency at $-3\,\text{dB}$ is found at 710 Hz.

Using the parameters in Table 4, the frequency characteristics were simulated by the block diagram shown in Figure 18. A swept sinusoidal signal (1–800 Hz) of amplitude 1.25 V, biased at 5 V, is chosen as the simulation input. The simulation result is also plotted in Figure 22. The theoretical frequency at $-3\,\text{dB}$ is observed at 780 Hz, which broadly agrees with the experimental result. The reason for this difference may be due to an oil momentum effect, which is not considered in the simulation.

However, the measured (710 Hz) and simulated (780 Hz) bandwidths of this servo valve are both lower than the estimated bandwidth (1300 Hz) in the section ‘Resonance frequency of the multi-body contacting spool-driving mechanism’. This is because the measured and simulated bandwidths are frequency characteristics of the system, which consists of the servo valve and power amplifier. Referring to the section ‘Piezoelectric power amplifier’, the dynamic characteristic of the power amplifier is lower than that of the PDDSV, which limits the bandwidth of the system.
**Time response characteristics.** Time response of the servo valve is the parameter to evaluate the transient characteristic, which is an important indicator of dynamic characteristics. The step response characteristic of the servo valve, which is expressed by the flow rate response under step input signal, is the most commonly used parameter to describe the transient characteristic. As the no-load flow rate is proportional to the spool displacement, the step response time of the proposed PDDSV can be described by the spool displacement response under a step drive voltage of the piezoelectric actuator. The pressure of the control port is usually measured to indirectly evaluate the transient characteristic for servo valves with piezoelectric actuators that have no displacement sensors.\(^1,8,12\) In this study, both the step responses of the spool displacement and the control port pressure have been measured.

The step response characteristic of the proposed PDDSV was tested by the dynamic characteristics test system, shown in Figure 21(b). The variable orifice was set to open. The input hydraulic supply was set to 21 MPa. A step signal with amplitude (from 5 V to 10 V, 10 V to 0 V and 5 V) and frequency 0.2 Hz is generated by the signal generator and amplified to the drive voltage by the power amplifier. The drive voltage is applied to the piezoelectric actuator. The signal of the SGS and the control port pressure \(P_A\) are both recorded by the A/D card in the industrial computer.

The measured step response of \(x_s\) is plotted in Figure 23(a), where the measured response time (from 10% to 90% of the steady value) of \(x_s\) is observed as 0.52 ms. Using the same input step signal as the experiment and the parameters listed in Table 4, the simulated step response of \(x_s\) is obtained and plotted in Figure 23(b) and the simulated response time is found to be 0.48 ms. The reason for the difference between the measured and simulated results is also supposed to be an oil momentum effect.

When measuring the step response of the spool displacement, the control port pressure \(P_A\) is also recorded as shown in Figure 24. From Figure 24, the time required to reach 0 MPa from 21 MPa is observed as 12 ms, which is much larger than the response time of spool displacement. This is because the no-load flow rate of the proposed PDDSV is small that it needs time to release or build up the pressure in the steel pipe (5) as shown in Figure 21(a).

The time required to reach 0 MPa from 21 MPa can be estimated by the definition of the hydraulic fluid bulk modulus, which is given by

\[
\frac{\Delta P}{\Delta t} = \frac{E_f}{V} Q_{nl}
\]

where \(\Delta P\) is the pressure variable quantity in the steel pipe, \(\Delta t\) is the time taken to reach this pressure variable quantity, \(E_f\) is the hydraulic fluid bulk modulus, \(V = \pi d^2 l/4\) is the volume of the steel pipe (5), and \(d\) and \(l\) are the inner diameter and length of the steel pipe.

Substituting \(\Delta P = 21\) MPa, \(E_f = 1000\) MPa, \(d = 10\) mm, \(l = 500\) mm, \(Q_{nl} = 4.5\) L/min into equation (32) gives \(\Delta t = 11\) ms. Considering that the spool movement time from 0 \(\mu\)m to 50 \(\mu\)m is about 1 ms as shown in Figure 23(a), the total time of \(P_A\) required to reach 0 MPa from 21 MPa is estimated to be about 12 ms, which agrees well with the measured value in Figure 24.

In order to evaluate the dynamic performances of the proposed PDDSV, a comparison with the existing piezoelectric servo valves has been carried out, as listed in Table 5. From Table 5, it is observed that the dynamic performances of the proposed PDDSV are significantly better than those of the two-stage piezoelectric servo valves\(^8,11\) and the PDDSVs with a displacement-amplifying mechanism.\(^12,14\) In the PDDSVs without the displacement-amplifying mechanism, the proposed PDDSV has better dynamic performances than Shen’s servo valve,\(^17\) while it is worse...
than Yokota and Hiramoto’s\textsuperscript{15} servo valve. As mentioned in the section ‘Frequency characteristics’, the proposed PDDSV can potentially get a bandwidth of 1300 Hz which will exceed that of Yokota’s servo valve if the power amplifier does not limit the dynamic performances of the proposed PDDSV.

Conclusions

A PDDSV with a novel multi-body contacting spool-driving mechanism, which has high dynamic performances and prevents the stack-type piezoelectric actuator from bearing the pulling force generated by the inertia and friction of spool, has been proposed. The development research, including design, modelling and experiment was done. The conclusions are as follows:

1. Theoretical estimation and experimental measurement of resonance frequency and displacement of the stack-type piezoelectric actuator were carried out. Good matching was observed, and hence the parameters used for the theoretical estimation of piezoelectric actuator were validated. Based on characteristic analysis of the piezoelectric actuator and selection of driving disc spring, the resonance frequency of the spool-driving mechanism and the bandwidth of the proposed servo valve were estimated as 4045 Hz and 1300 Hz, respectively.

2. The nonlinear dynamic model of the multi-body contacting spool-driving mechanism was established based on its analogical electrical circuit and the hysteresis operator was identified as a first-order differential equation. A second-order system was adopted to approximate the piezoelectric power amplifier. Combining the nonlinear model of the spool-driving mechanism and dynamic characteristics of power amplifier, a comprehensive dynamic model of the PDDSV was produced, which was validated by experimental results. This model is useful for controller design of the proposed servo valve.

3. Under a supply pressure of 21 MPa, the static and dynamic characteristics of the proposed PDDSV were studied experimentally. The no-load flow rate of the proposed servo valve was measured as 4.45 L/min at a maximum input voltage of 5 V from null. The frequency at $-3$ dB was observed at 710 Hz and the step response time of spool displacement was observed as 0.52 ms. It was found that the measured bandwidth of the proposed servo valve is lower than the estimated value (1300 Hz), which resulted from the limitation of the dynamic characteristics of the power amplifier. The dynamic performances of the proposed servo valve were compared with the existing piezoelectric servo valves, which demonstrated that it has satisfactory dynamic characteristics for high-frequency applications.

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Conflict of interest

None declared.

References


