Design of a Flextensional Transducer with Adjustable Prestress

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Abstract: Prestress has a great bearing on the performance of flextensional transducer. In order to enhance the adjustability of prestress, a design method for the flextensional transducer with adjustable prestress was proposed in this study. Through the conversion design of rotational and rectilinear motions of the transition block, the clamping length of major axis of class IV flextensional transducer shell was changed, enabling the adjustment of prestress. Next, the related experiment was further carried out. The results show that the return displacement of adjustable transition block is slightly greater than the travel displacement, and both displacements show high overlap ratios in the two tests. Because of the prestress adjustment, the frequency response of flextensional transducer is reduced to some extent, within one frequency band (40-63 Hz), but its sound pressure level (SPL) substantially increases nearby the resonant frequency (about 75 Hz).

1. Introduction
Flextensional transducer drives a larger area of shell to generate greater flexural vibration through the stretching vibration of driving element, thus radiating the sound wave with greater energy inside the water. Class IV flextensional transducer has been studied and analyzed most adequately so far, and now it is still a research hotspot. Zhang et al. [1] used the finite element method to study the stress distribution in the driving vibrator of the piezoelectric monocrystal PMNT transducer, and compared it with the piezoelectric ceramic PZT transducer. Zhou et al. [2] put forward a transducer structure driven conformably with the flextensional shell, which could effectively avoid the change of vibration mode induced by the traditional longitudinal driving crystal stack and obtain a lower operating frequency. Guo et al. [3] reduced the working frequency of transducer by 22.2% and expanded the working band by 21.5% through opening double grooves in the shell. On the contrary, Ma Z et al. [4] enhanced the resonant frequency and working bandwidth of the transducer by adding reinforcements outside the shell. Based on the fish-lip flextensional transducer invented by Mo [6], Liu et al. [5] further explored a double-shell nested fish-lip flextensional transducer and optimized its structural parameters. In addition, Chen L G et al. [7] simulated the half-wavelength array performance of class IV flextensional transducer, and laid the particular emphasis on studying the influence of the size of water area. Comparatively speaking, the close-packed array studied by Gu et al. [8] could more effectively reduce the resonant frequency, expand the bandwidth and achieve all-round directivity. Driving elements like piezoelectric ceramic crystal stacks or giant magnetostictive rods are installed along the major axis of elliptic shell of the flextensional transducer. In order to ensure the normal operation of transducer and give full play to the performance of driving elements, the length of each driving element is generally required to be slightly greater than the clamping distance of major axis of transducer shell, and then
the transducer will generate a prestress. As found in [9,10], the performance of flexextensional transducer is greatly impacted by the prestress. He et al. [11] studied the assembling prestress of flexextensional transducer as well as the relation between the diving depth and the prestress.

For most class IV flexextensional transducers, the prestress is unchangeable once the fabrication is completed. Therefore, a design method of flexextensional transducer with adjustable prestress was proposed in this study. Through the conversion design of rotational and rectilinear motions of transition block, the clamping distance of major axis of class IV flexextensional transducer shell was changed, so as to realize the adjustment of prestress. Subsequently, the displacement adjustment of transition block and radiation performance of transducer were tested.

2. Design of Prestress-Adjustable Structure

2.1. Schematic design of adjustable prestress

The traditional class IV flexextensional transducer applies the prestress to the driving elements by reasonably designing the clamping distance of driving elements along the major axis of shell and the length of driving element (generally speaking, the former is slightly smaller than the latter). The prestress of the fabricated transducer cannot be changed, so high precision is required for the design and fabrication in the clamping distance of shell and the length of driving elements, which undoubtedly increases the difficulty in the transducer design and fabrication. As for the scheme proposed in this study, the prestress of the flexextensional transducer is adjusted by changing the clamping distance of driving elements along the major axis, and the clamping distance is adjusted by changing the thickness of the transition block, and the specific scheme design is as shown in Figure 1.

The transition block was divided into fixed plate and moving plate, where the latter contacts and fits with the driving element and the former contacts and fits with the shell. The prestress is loaded by changing the distance between the fixed plate and moving plate of transition block. When assembled into the transducer structure, the transition block only has two exposed end faces, so the adjusting and operating component (hand wheel) of transition block could only be designed on an end face. From the aspect of kinematics analysis, the rotational motion of transition block might be transformed into rectilinear motion, the screw drive could be considered, as it could not only guarantee the conversion of motion but also realize self-locking. But the drive direction disaccords with the direction of rotation of the current hand wheel, so the direction of rotation should be changed, which could be realized through the conical gear set. Meanwhile, the problem of force transfer is taken into account, and the conical gear ratio could be adjusted so that not too large force is needed to drive the hand wheel. The specific structural diagram is as shown in Figure 1 (b), and the implementation process is as follows: As the hand wheel is rotated, the drive shaft drives the small conical gear to rotate, the small conical gear is geared to the large conical gear, thus the drive is transmitted to the large conical gear, the gear engagement between the small and large conical gears conforms to the principle of effort-saving drive, and the large conical gear is rotated to drive the rotation of threaded portion, and the threaded hole on the fixed plate fits with the fixed plate well. Driven by the screw thread, a rectilinear motion is generated between large conical gear and fixed plate, and the shaft on the large conical gear fits with the moving plate, the large conical gear drives the rectilinear motion between moving plate and fixed plate, their distance then changed, so did the size of transition block along the major axis of flexextensional transducer shell, thus realizing the prestress adjustment of driving element.

2.2. Design of structure with adjustable prestress

The design of structure with adjustable prestress is mainly composed of screw drive design and drive displacement design.

2.2.1. Screw drive design

Screw drive generally converts the rotational motion into rectilinear motion, or on the contrary, converts the rectilinear motion into rotational motion, during which the energy and force are
transferred. As the dimensional space of transition block is limited, only the screw drive with simple structure can be used, which can realize the self-locking when certain conditions are met. The thread of sliding screw is generally divided into three types: trapezoid, sawtooth and rectangle, among which the trapezoid thread has been mostly widely applied, and the sawtooth-shape screw is mainly applied to one-way force condition, while the rectangular thread has high drive efficiency, but its machining is difficult with low strength, so it has been less applied. Given this, the trapezoid screw was used for the analysis and design in this study.

\[ d_z \geq 0.8 \left( \frac{F}{\psi p_p} \right)^{1/2} \]  

where \( F \) is the axial load (N); \( p_p \) is the permissible pressure intensity of thread pair, which is 7.5-13 MPa by checking the manual; \( \psi \) is the nut form, and the integral-type nut takes the value interval: 1.2-2.5. In order to meet the related requirements, the related values are taken as the minimum, and thus the following is obtained:

\[ d_z \geq 0.8 \left( \frac{4000}{1.2 \times 7.5 \times 10^5} \right)^{1/2} = 16.87 \text{mm} \]

The above parameter \( d_z \) is taken as 18 mm by checking the manual, so the nominal diameter: \( d=20 \text{ mm} \) and \( P=2 \text{ mm} \). The nut height is \( H=\psi d_z=22.8 \text{ mm} \), and here it is set at 20 mm. The number of screwing turns is \( n=H/P=10-12 \), which is calculated as 10. The height of basic tooth form is \( H_1=0.5P=1 \text{ mm} \).

The working pressure intensity is as below:

\[ p = \frac{F}{\pi d_z H_1 n} = \frac{4000}{\pi \times 19 \times 1 \times 10} = 6.7013 \text{MPa} \leq p_p = (7.5 - 13) \text{MPa} \]

2) Self-locking condition. The lead angle needs to satisfy the following equation:

\[ \lambda = \arctan \frac{S}{\pi d_z^2} = \arctan \frac{2}{\pi \times 19} = 1.92^\circ \]

where \( S \) is the screw lead (mm).

The equivalent frictional angle is:

\[ \rho' = \arctan \frac{f}{\cos \frac{\alpha}{2}} = \arctan \frac{0.15}{\cos 15^\circ} = 8.83^\circ \]

where \( f \) is the friction factor, which ranges from 0.15 to 0.17. When \( \lambda \leq \rho' \), the self-locking is enabled.

3) Screw strength design. The equivalent stress is as below:
\[ \sigma_{ca} = \sqrt{\frac{4F}{\pi d_3^2}} + 3 \left( \frac{M_i}{0.2d_3} \right)^2 \]  

(2)

where \( M_i \) is the torsional moment, and \( M_t = M_{i1} + M_{i2} \); and the friction moment of screw thread is as below:

\[ M_{i1} = \frac{1}{2} d_2 F \tan(\lambda + \rho') = \frac{1}{2} \times 0.019 \times 4000 \times \tan(1.92^\circ + 8.83^\circ) = 7.215 \text{ N} \cdot \text{m} \]

The friction moment on the axial bearing surface of drive shaft is expressed in the following equation:

\[ M_{i2} = \frac{1}{3} f_s F D_0 = \frac{1}{3} \times 0.17 \times 4000 \times 0.02 = 4.533 \text{ N} \cdot \text{m} \]

where \( f_s \) is the friction factor between axial bearing surfaces, and it is taken as 0.17 if both bearing surfaces are made of steel; \( D_0 \) is the outer diameter of bearing surface, which is taken as 20 mm. Hence, \( M_t = 11.748 \text{ N} \cdot \text{m} \). \( d_3 \) is the minor diameter of screw thread, which is taken as 17.5 mm. The equivalent stress is:

\[ \sigma_{ca} = \sqrt{\frac{4F}{\pi d_3^2}} + 3 \left( \frac{M_i}{0.2d_3} \right)^2 = 10^6 \sqrt{\frac{4 \times 4000}{\pi \times 17.5^2}} + 3 \times \left( \frac{11.748}{0.2 \times 17.5^2 \times 10^{-3}} \right)^2 = 25.24 \text{ MPa} \]

For 45 steel, \( \sigma_s = 225 \text{ MPa} \), and \( \sigma_p = \frac{\sigma_s}{3 - 5} = (45 - 75) \text{ MPa} \), which meets the requirement for screw strength.

4) Thread strength design. The width at the thread root is \( b = 0.5P = 1 \text{ mm} \). The screw and nut are made of the same material, so it is only necessary to check the screw and thread strength:

(1) Shear strength

\[ \tau = \frac{F}{\pi d_3 b n} = \frac{4000}{\pi \times 17.5 \times 10} = 7.28 \text{ MPa} \leq \tau_p \]

The permissible shear stress \( \tau_p = 0.6 \sigma_p = (27 - 45) \text{ MPa} \), which meets the strength requirement.

(2) Bending strength

\[ \sigma_b = \frac{3FH}{\pi d_3 b^2 n} = \frac{3 \times 4000 \times 1}{\pi \times 17.5^2 \times 10} = 21.83 \text{ MPa} \leq \sigma_{bp} \]

The permissible bending stress \( \sigma_{bp} = (1 - 1.2) \sigma_p = (45 - 90) \text{ MPa} \), which also meets the strength requirement.

To sum up, the screw strength meets the requirement.

5) Screw stiffness design. The axial deformation of screw lead generated due to the axial load is calculated as below:

\[ \Delta S_p = \frac{FS}{EA} = \frac{4FS}{\pi Ed_3^4} = \frac{4 \times 4000 \times 2 \times 10^{-3}}{\pi \times 17.5^2 \times 10^6 \times 210 \times 10^6} = 1.583816 \times 10^{-7} \text{ m} \]

The axial deformation of screw lead generated due to the torsional moment is as below:

\[ \Delta S_M = \frac{16M_s L}{\pi GD_3^2} = \frac{16 \times 11.748 \times 2 \times 10^4}{\pi^2 \times 8.5 \times 10^6 \times 17.5^4 \times 10^{-12}} = 9.556 \times 10^{-9} \text{ m} \]

The total axial deformation: \( \Delta S = \Delta S_M \pm \Delta S_f \), and the maximum deformation: \( 1.6794 \times 10^{-7} \text{ m} = 0.16794 \mu \text{m} \). The following is further solved:

\[ \left( \frac{\Delta S}{S} \times 10^9 \right) = \frac{0.16794}{2} \times 10^3 = 84 \mu \text{m/m} \]

The Class 9 precision requirement can be satisfied.
2.2.2. Drive displacement design

The axial displacement of drive screw is expressed as below:

\[ d = \frac{\phi}{2\pi} S = \frac{\phi}{2\pi} P x \]  

(3)

where \( \phi \) is the screw corner (rad); \( S \) is the screw lead (mm); \( P \) is the screw pitch (mm); \( x \) is the number of leads.

The drive ratio of conical gear is set as \( i = 2 \), \( n_1 = 16 \), \( n_2 = 32 \), rotational angle of gear as \( \phi_1 \), and rotational angle of conical gear connecting to the screw as \( \phi_2 \), and the following can be obtained according to the definition of drive ratio:

\[ i = \frac{\phi_1}{\phi_2} \]  

(4)

By combining the formula of axial displacement of screw, the following can be obtained:

\[ d = \frac{\phi_1}{4\pi} P x \]  

(5)

If the angular precision is taken as \( 1^\circ = \frac{\pi}{180} \), screw pitch as \( P = 1.5 \text{ mm} \) and \( x = 1 \), the drive accuracy of axial displacement is obtained as follows:

\[ \Delta d = \frac{180}{4\pi} \times 1.5 = 2.083\mu\text{m} \]

The rectilinear motion displacement generated by the rotational angle (\( \theta \) (°)) is \( d = \theta \Delta d \), so the rectilinear motion displacement when the hand wheel rotates by one circle is \( d = 0.75 \text{ mm} \).

According to the above analysis and design, the structural design of adjustable transition block and material object picture are as shown in Figure 2.

Figure 2. Design drawing and material object picture of adjustable transition block
3. Experiment and Results

The experiment mainly aimed to test the drive displacement of adjustable transition block and the influence of prestress adjustment of transition block on the transducer radiance.

The drive displacement of adjustable transition block was tested twice at three location points (left end point, midpoint and right end point) each time, each test is divided into travel and return, and the relation between displacement and number of rotational turns of hand wheel can be obtained. As shown in Figure 3, the return displacement is larger than the travel displacement, which mainly ascribes to the error of screw drive clearance. The results of two tests show that the overlap ratios of both travel displacement and return displacement are high, where the former is slightly higher than the latter (Figure 4). The actual displacement values of travel and return are deviated, to some extent, from the theoretical design.

![Figure 3. Comparison of travel and return displacements in two tests](image1)

![Figure 4. Overlap ratio analysis of travel and return displacements in two tests](image2)

The designed and fabricated adjustable transition block was assembled into the flextensional transducer, and the two sides of shell are sealed. The flextensional transducer is then hoisted for the sound production test in the outfield air. The frequency-dependent changes of converted sound pressure levels (SPL) under two circumstances—unregulated transition block and regulated transition block—in the same test direction of the same distance, are specifically shown in Figure 5. After the prestress was further added to the regulated transition block, the frequency response of flextensional transducer was reduced within one frequency band (40-63 Hz) to a certain degree, but the SPL might increase by a large margin nearby the resonant frequency (about 75 Hz). Therefore, the SPL frequency response of transducer could be regulated by changing its prestress, and the needed SPL in the target frequency band has an obvious gain.
Figure 5. Frequency-dependent change curves of converted SPL under two working conditions

4. Conclusions
In this study, a design method of flextensional transducer with adjustable prestress was proposed. The clamping distance of major axis of class IV flextensional transducer shell changes through the conversion design of rotational and rectilinear motions of transition block, so as to regulate the prestress. Next, further experiments are carried out on the displacement adjustment of transition block and radiation performance of the transducer. The results reveal that the return displacement of adjustable transition block is slightly greater than the travel displacement, and both travel and return displacements show high overlap ratios in the two tests. Due to the prestress adjustment, the frequency response of adjustable transition block decreases to some extent in one frequency band, but the SPL increases considerably nearby the resonant frequency (about 75 Hz).

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