Design and Experimental Study of Ultrasonic Vibration Feeding Device With Double Symmetrical Structure

JIAN ZHANG AND XIAOZHU WANG
Yingkou Institute of Technology, Yingkou, Liaoning 115014, China
Corresponding author: Xiaozhu Wang (330608566@qq.com)

This work was supported in part by the Liaoning Natural Science Foundation Joint Fund Project under Grant 2021-YKLH-07 and Grant 2020-YKLH-28, in part by the Liaoning Provincial Department of Education Funding for Scientific Research under Grant L2020007, and in part by the Program for Excellent Talents of Science and Technology in Yingkou Institute of Technology under Grant RC202003.

ABSTRACT In view of the limitations of traditional feeder driving mode in bulk material conveying, a new scheme of ultrasonic vibration feeding device with double symmetrical axis structure is proposed based on the existing traveling wave ultrasonic material conveying device. The outer sides of the upper and lower straight beam sections of the symmetrical conveying vibrator contain convex tooth structures, and piezoelectric ceramic sheets are pasted on the inner side; Based on the piezoelectric driving principle and the quarter wavelength principle, the conveying vibrator is dynamically designed, the influence of geometric size on the vibrator frequency is analyzed, the bending vibration mode of the vibrator is selected as 18 order, and the principle prototype of ultrasonic vibration feeding device is made. The resonant frequency difference measured in the experiment is close to that obtained by theoretical analysis, Moreover, the amplitude distribution of the conveying vibrator tooth surface is basically consistent with the finite element analysis results, and the influence law of material types on the conveying rate is compared and analyzed. The test results show that the ultrasonic vibration feeding device has a certain conveying capacity, the conveying rate is adjustable in a certain range, has high control accuracy, and meets the engineering application.

INDEX TERMS Vibratory feeding device, symmetric structure, ring type piezoelectric vibrator, traveling wave, ultrasonic vibration.

I. INTRODUCTION Ultrasonic material conveying device is a new type of ultrasonic driver derived from ultrasonic motor, which is completely different from the traditional material conveying device in principle[1]. Compare with the traditional conveying device, it has no shaft, rod and other mechanical transmission parts, simple structure and easy to manufacture [2]; The conveying rate can be controlled by changing any parameter of the driving signal (such as voltage, frequency, amplitude, etc.) [3], so there are many control parameters and good controllability [4]. And it don’t produce interference electromagnetic field and is not affected by electromagnetic interference signal [5]; it has the characteristics of quiet, no noise and fast response [6].

When the traveling wave ultrasonic vibration feeding device working, the contact pressure and contact deformation are generated between the conveying vibrator and the material through the self-weight of the material [7]. Because of the longitudinal displacement at the contact, the material is out of contact with the particle at the peak position on the conveying vibrator, The transverse displacement at the contact pushes the material forward in a single direction by friction [8]. It can be seen that the conveying vibrator is the major component of the ultrasonic vibration feeding device [9]. In order to design a high-performance ultrasonic vibration feeding device, only by clarifying the dynamic characteristics of the conveying vibrator [10].

The structure of the traditional long ring vibrator is shown in Figure 1. Because the vibrator of this structure is designed with convex tooth structure on the outside of the upper straight beam section, and the ceramic sheet is pasted on the outside of the lower straight beam section, which causes the
upper and lower parts of the vibrator structure are strictly asymmetric, it is difficult to find the optimal mode in the finite element analysis, and with the further widening of the width dimension, it is easy to have obvious bending deformation in the width direction [11]. Therefore, the industrialization of ultrasonic material conveying technology depends on the in-depth discussion of ultrasonic driving principle [12].

In view of the problems existing in the traditional long ring ultrasonic material conveying device, in this paper, an ultrasonic vibration feeding device with symmetrical structure is designed. The conveying vibrator with double symmetrical axis structure can well solve the above problems and has better conveying performance for different materials.

II. THEORETICAL BASIS OF ULTRASONIC VIBRATION FEEDING DEVICE DESIGN

In this paper, a symmetrical structure is introduced base on the traditional long ring vibrator structure. The service life of the symmetrical structure vibrator is twice that of the traditional long ring vibrator. Since any one of the upper and lower conveying surfaces is worn for a long time, and then the conveying performance of the conveying device is reduced, The other conveying surface is used to make the conveying vibrator continue to work.

At the same time, in order to improve the conveying rate of the vibrator (amplifying the surface amplitude of the vibrator), a convex tooth structure is designed outside the upper and lower straight beam sections of the vibrator elastomer. Since the condition for synthesizing in-plane bending traveling waves is to excite two-phase approximately orthogonal in-plane bending vibration modes of the same order in the vibrator [13], it is required to find two-phase approximately orthogonal in-plane bending modes of the same order in the theoretical analysis.

Considering that in order to reduce the hollow structure, the pasting difficulty of piezoelectric ceramic sheet is increased [14], so that the pasting position of piezoelectric ceramic sheet only is located on the inner side of the upper and lower straight beam sections. Therefore, the prototype of the designed symmetrical structure vibrator is shown in Figure 2.

The section line is piezoelectric ceramic sheet in Figure 2. Compare with Figure 1 and Figure 2, it is found that the number of convex teeth in the upper and lower straight beam sections of the symmetrical structure vibrator is reduced and the tooth width is increased. Because the wire cutting technology is used in the later processing of the vibrator [15], so as to achieve higher cost performance [16].

A. IN-PLANE VIBRATION ANALYSIS OF CIRCULAR RING STRUCTURE

The long ring structure evolves from the ring [17], Referring to the bending vibration of the ring in its torus, the ring structure is shown in Figure 3 [18].

In Figure 3, $h$ is the thickness of the ring and $b$ is the radial width of the ring.

It is assumed that the plane of the ring (x-y plane) is the symmetrical plane of each cross-section [19], and the cross-sectional dimension of the ring is very small compares with its centerline radius $r$ [20]. From radial displacement (positive outward) $u$ and tangential displacement (increasing direction of $\theta$ is positive) $v$, and the unit elongation at any point on the ring center line is

$$ e = \frac{u}{r} + \frac{\partial v}{r \partial \theta} \quad (1) $$

In the Equation (1): The angle at one point on the ring is $\theta$.

The expression for the change of curvature is

$$ \frac{1}{r + \Delta r} - \frac{1}{r} = -\frac{\partial^2 u}{r^2 \partial \theta^2} - \frac{u}{r^2} \quad (2) $$

In the case of bending vibration in the ring plane, the radial displacement $u$ is expanded into trigonometric series

$$ u = \sum_{i=1}^{\infty} [a_i \cos(i\theta) + b_i \sin(i\theta)] \quad (3) $$

In the Equation (3): the coefficients $a_i$ and $b_i$ are functions of time.

And it is defined that the number $i$ for circumferential wavelengths of ring bending deformation is the order of vibration mode.

Considering the bending vibration without expansion, Set $e = 0$, which is obtained from equation (1)

$$ u = -\frac{\partial v}{\partial \theta} \quad (4) $$
Substitute Equation (3) into Equation (4) and integrated to obtain

\[ v = \sum_{i=1}^{\infty} \left[ -\frac{a_i}{l} \sin(i\theta) + \frac{b_i}{l} \cos(i\theta) \right] \]  

(5)

The bending moment of any cross section on the ring is

\[ M = -\frac{EI}{r^2} \left( \frac{\partial^2 u}{\partial \theta^2} + u \right) \]  

(6)

In the Equation (6): \( M \) is the bending moment of any cross section on the ring; \( E \) is the elastic modulus of the material; \( I \) is the inertia moment of the ring cross-sectional plane.

Thus, the bending potential energy is obtained

\[ U = \frac{EI}{2r^4} \int_0^{2\pi} \left( \frac{\partial^2 u}{\partial \theta^2} + u \right)^2 r d\theta \]  

(7)

In the Equation (7): \( U \) is the bending potential energy. Substitute Equation (3) into Equation (7) and apply the following Equation (8)

\[ \int_0^{2\pi} \cos m\theta \cos n\theta d\theta = 0 \]

\[ \int_0^{2\pi} \sin m\theta \sin n\theta d\theta = 0 \quad (m \neq n) \]

\[ \int_0^{2\pi} \cos m\theta \sin n\theta d\theta = 0 \]

\[ \int_0^{2\pi} \cos^2 m\theta d\theta = \int_0^{2\pi} \sin^2 m\theta d\theta = \pi \]  

(8)

Obtain

\[ U = \frac{EI\pi}{2r^3} \sum_{i=1}^{\infty} \left( 1 - \frac{l^2}{R^2} \right)^2 \left( \frac{a_i^2}{l} + \frac{b_i^2}{l} \right) \]  

(9)

The kinetic energy of the vibrating ring is

\[ T = \frac{A\rho}{2} \int_0^{2\pi} (\dot{a}_i^2 + \dot{b}_i^2) r d\theta \]  

(10)

In the Equation (10): \( T \) is the kinetic energy of the vibrating ring, \( A \) is the cross-sectional area of the ring, \( \rho \) is the density of the material.

Substitute Equation (3) and (5) into Equation (10) to derive

\[ T = \frac{Ar\pi\rho}{2} \sum_{i=1}^{\infty} \left( 1 + \frac{l^2}{R^2} \right) (\dot{a}_i^2 + \dot{b}_i^2) \]  

(11)

Applying Lagrange equation to conservative system and generalized coordinate \( a_i \), the differential equation of motion is obtained

\[ Ar\rho(1 + \frac{l^2}{R^2})\ddot{a}_i + \frac{EI\pi}{r^3}(1 - \frac{l^2}{R^2})a_i = 0 \]  

(12)

Or

\[ \ddot{a}_i + \frac{EI\pi^2(1 - \frac{l^2}{R^2})}{\rho Ar^4(1 + \frac{l^2}{R^2})}a_i = 0 \]  

(13)

The equation of the same form is obtained for the generalized coordinate \( b_i \).

Therefore, the frequency expression for the \( i \)-th mode of bending vibration in torus is

\[ f_i = \frac{1}{2\pi} \sqrt{\frac{EI^2(1 - \frac{l^2}{R^2})}{\rho Ar^4(1 + \frac{l^2}{R^2})}} \]  

(14)

The bending vibration mode belongs to the \( i \)-th order and recorded as \( B_i \). When \( i = 1 \), \( f_1 = 0 \), the ring moves as a rigid body; when \( i \geq 2 \), the ring vibrate according to the basic bending mode.

The straight and curved beams are composed of two straight beam segments and two semicircular arc segments. As shown in Figure 4, \( b \) is the beam width and \( h \) is the thickness.

**FIGURE 4.** Structural diagram of long ring beam.

It is assumed that the beam is uniform and equal section, and the cross-sectional dimension of the beam is very small compared with the centerline radius \( r \). When the antinode number of long ring beam bending vibration mode is \( 2n \), the vibration mode of the bending mode is defined as the \( n \)-th order, which is recorded as \( B_n \).

According to the equation \( r = (2\pi R_0 - 2l)/2\pi \), keep the length of the ring axis as a fixed value, where \( R_0 \) is the radius of the ring centerline, set \( R_0 = 80 \text{mm} \), the thickness of the long ring beam \( h = 4 \text{ mm} \), the width \( b = 6 \text{ mm} \), \( l/r = 10 \), where \( l \) is the length of the straight beam section, \( r \) is the radius of the ring centerline, the material is brass, the elastic modulus \( E = 110 \text{GN/m}^2 \), the density \( \rho = 8400 \text{kg/m}^3 \), and the unit type is BEAM3. Using the finite element software, the in-plane bending vibration modes of the ring and long ring beam are analyzed, and the \( B_2 \sim B_{10} \) vibration modes and natural frequencies are obtained, the analysis results of ring bending vibration mode are compared with that of long ring beam, as shown in Figure 5.

**B. SYMMETRY ANALYSIS OF MODE SHAPES FOR LONG RING BEAMS**

Any size combination of long ring structure is selected. When \( h = 5 \text{mm} \), \( b = 6 \text{mm} \), \( l = 155 \text{mm} \), \( r = 30 \text{mm} \), the material is brass, the element type is BEAM3, considering the accuracy requirements, the element scale is 1mm, and the even and odd orders are optional.

Taking \( B_{12} \) and \( B_{13} \) as examples, the coordinate of any node and its displacement in x and y directions are extracted by finite element method to judge the symmetry.
The modal analysis results of $B_{12}$ in-plane bending vibration are shown in Figure 6; the natural frequencies of the two modes are 17945Hz and 18109Hz respectively, with a frequency difference of 164Hz.

Take three nodes 1, 2 and 3 from the upper straight beam section locate in the left half plane, and correspondingly take three nodes 4, 5 and 6 symmetrically on the right half plane. So there are six independent nodes in the upper half plane, and the six nodes $1'$, $2'$, $3'$, $4'$, $5'$ and $6'$ in the lower half plane symmetrical to the upper half plane are also generated in turn.

Table 1 lists the coordinates of the selected node and the deformation of the selected node under two modes.

### Table 1. Coordinate and deflection of selected nodes for $B_{12}$.

| Point number | Coordinate (mm) | Mode 1 | Mode 2 |
|--------------|-----------------|--------|--------|
|              | $\Delta x$      | $\Delta y$ | $\Delta x$ | $\Delta y$ |
| 1            | 0.917881        | 3.78628 | -0.32908 | 0.32908 |
|              | 252514          | 5024491  | 8188785  | 8188902 |
| 2            | 0.969555        | -3.75525 | 0.18518 | -0.18518 |
|              | 799576          | 3830337  | 2025411  | 6852752 |
| 3            | 0.201908        | -2.13763 | 0.78027  | -0.78027 |
|              | 867802          | 3844367  | 4282031  | 4282034 |
| 4            | -0.20109        | -2.13763 | 0.78027  | -0.78027 |
|              | 8627608         | 3844350  | 4282032  | 4282034 |
| 5            | -0.969555       | -3.75525 | 0.18518  | -0.18518 |
|              | 5795399         | 3830341  | 6852556  | 6852572 |
| 6            | -0.917881       | 3.78628  | -0.32908 | 0.32908 |
|              | 1252469         | 5024513  | 8188814  | 8188806 |
| 1'           | 0.917881        | -3.78628 | -0.32908 | -0.32908 |
|              | 252503          | 5024485  | 9525650  | 9525664 |
| 2'           | 0.969555        | 3.75525  | 0.23141  | 0.23141 |
|              | 799576          | 3830326  | 3554663  | 3554677 |
| 3'           | 0.201908        | 2.13763  | 3.12332  | 3.12332 |
|              | 867822          | 3844351  | 2024324  | 2024332 |
| 4'           | -0.20109        | 2.13763  | -3.12332 | -3.12332 |
|              | 8627608         | 3844350  | 2024411  | 2024402 |
| 5'           | -0.969555       | 3.75525  | -0.23141 | -0.23141 |
|              | 5795396         | 3830334  | 3534864  | 3534848 |
| 6'           | -0.917881       | -3.78628 | 0.20343  | 0.20343 |

Note: $\Delta x$ and $\Delta y$ are the deformation along the x and y directions respectively, as shown in Table 1, and they are relative values.

The observation reveals that the vibration mode of mode 1 is symmetrical about both x-axis and y-axis, and the vibration mode of mode 2 is symmetrical about the origin. The results are consistent with the judgment of theoretical analysis.
The modal analysis results of \( B_{13} \) in-plane bending vibration are shown in Figure 7. The natural frequencies of the two modes are 21226Hz and 21270Hz respectively, and the frequency difference is 44Hz.

The node selection method is the same as \( B_{12} \). Coordinates of selected nodes and deformation of selected nodes in two modes are shown in Table 2.

The observation also reveals that the vibration mode of mode 1 is symmetrical about y-axis, and the vibration mode of mode 2 is symmetrical about x-axis. The results are consistent with the judgment of theoretical analysis.

The natural frequencies of two same order vibration modes are generally different, moreover, when the \( l/r \) is kept within a certain numerical range, the vibration mode with almost zero frequency difference is found by optimizing the geometric size of the beam. However, when the \( l/r \) continued to increase, it is difficult to find the two bending modes with similar frequency.

Keep \( n \) constant and double the values of \( l \) and \( r \) in Table 3 at the same time, the thickness of the long ring beam \( h \) is also doubled, and the width \( b \) remains unchanged. Starting from the ring with \( R_0 = 160mm \), the vibration modes and natural frequencies of modes \( B_{15}, B_{16}, B_{17} \) and \( B_{18} \) are still analyzed. The results are shown in Figure 9.

It is obvious that the curve in Figure 9 is basically consistent with the corresponding curve in Figure 8, only under the same \( n \) value; the two natural frequencies of the same order vibration mode are almost reduced to half of the original size. Therefore, it is inferred that the long ring beam structure maintains both geometric similarity and dynamic similarity.

### III. MODAL CONTROL OF SYMMETRICAL CONVEYING VIBRATOR

The key of traveling wave ultrasonic vibration feeding device is to generate unidirectional traveling wave in the vibrator, the condition for synthesizing in-plane bending traveling waves is that two-phase approximately orthogonal in-plane bending vibration modes at the same order of the vibrator are excited, and the natural frequency difference of the two standing waves should be as small as possible. In this paper, parametric modeling is carried out for the prototype of the proposed conveying vibrator by using the finite element software, and the size parameters of some structures are strictly controlled.

#### A. STRUCTURAL SLOTTING AND VIBRATOR MODE CONTROL

Modal analysis is to find two ideal in-plane bending vibration modes of symmetrical conveying vibrator, and then determine the structural dimensions of each part in the vibrator. Therefore, it is necessary to set the structural parameters before analysis. The specific settings of some parameters during their frequencies are, the conclusion of vibrator symmetry is demonstrated accurately by mathematical method.
TABLE 3. Numerical value of \( N, R, L \).

| \( n \)   | \( r \)   | \( l \)   |
|-----------|-----------|-----------|
| 0.2000    | 75.2119   | 15.0424   |
| 0.4000    | 70.9645   | 28.3858   |
| 0.6000    | 67.1712   | 40.3027   |
| 0.8000    | 63.7629   | 51.0103   |
| 1.0000    | 60.6838   | 60.6838   |
| 1.2000    | 57.8883   | 69.4660   |
| 1.4000    | 55.3390   | 77.4747   |
| 1.6000    | 53.0049   | 84.8078   |
| 1.8000    | 50.8596   | 91.5473   |
| 2.0000    | 48.8812   | 97.7625   |
| 2.2000    | 47.0510   | 103.5123  |
| 2.4000    | 45.3529   | 108.8470  |
| 2.6000    | 43.7731   | 113.8101  |
| 2.8000    | 42.2997   | 118.4391  |
| 3.0000    | 40.9222   | 122.7666  |
| 3.2000    | 39.6316   | 126.8211  |
| 3.4000    | 38.4199   | 130.6277  |
| 3.6000    | 37.2801   | 134.2084  |
| 3.8000    | 36.2060   | 137.5829  |
| 4.0000    | 35.1921   | 140.7683  |
| 4.2000    | 34.2334   | 143.7801  |
| 4.4000    | 33.3255   | 146.6322  |
| 4.6000    | 32.4646   | 149.3370  |
| 4.8000    | 31.6470   | 151.9055  |
| 5.0000    | 30.8896   | 154.3478  |
| 5.2000    | 30.1294   | 156.6730  |
| 5.4000    | 29.4240   | 158.8893  |
| 5.6000    | 28.7508   | 161.0042  |
| 5.8000    | 28.1077   | 163.0245  |
| 6.0000    | 27.4927   | 164.9564  |
| 6.2000    | 26.9041   | 166.8056  |
| 6.4000    | 26.3402   | 168.5772  |
| 6.6000    | 25.7994   | 170.2762  |
| 6.8000    | 25.2804   | 171.9067  |
| 7.0000    | 24.7818   | 173.4729  |
| 7.2000    | 24.3026   | 174.9786  |
| 7.4000    | 23.8415   | 176.4271  |
| 7.6000    | 23.3976   | 177.8217  |
| 7.8000    | 22.9699   | 179.1653  |
| 8.0000    | 22.5576   | 180.4607  |
| 8.2000    | 22.1598   | 181.7104  |
| 8.4000    | 21.7758   | 182.9167  |
| 8.6000    | 21.4049   | 184.0820  |
| 8.8000    | 21.0464   | 185.2082  |
| 9.0000    | 20.6997   | 186.2974  |
| 9.2000    | 20.3643   | 187.3512  |
| 9.4000    | 20.0395   | 188.3714  |
| 9.6000    | 19.7250   | 189.3596  |
| 9.8000    | 19.4201   | 190.3173  |
| 10.0000   | 19.1246   | 191.2458  |

FIGURE 8. Change of natural frequencies for the same two order modes with \( l \) and \( r \).
finite element analysis are shown in Table 4. In finite element analysis (without considering the influence of bonding layer), SOLID45 eight node hexahedron element is adopted for elastomer, SOLID5 electromechanical coupling element is adopted for piezoelectric ceramics, mapping mode is adopted for meshing, and the element size is specified as 1mm.

**TABLE 4. Setting of partial parameter in finite element analysis.**

| Parts                        | Elastic modulus GPA | Density (kg/m³) | Poisson's ratio |
|------------------------------|---------------------|-----------------|----------------|
| Elastomer (brass 59)         | 1.1                 | 8400            | 0.3            |
| Piezoelectric ceramics (pzt-181) | anisotropy      | 7850            |                |

The partial enlarged view of the tooth part in the symmetrical conveying vibrator is shown in Figure 10. In the Figure 10, \( l_1 \) is the tooth length of the vibrating elastomer, \( l_2 \) is the tooth gap, \( h_1 \) is the substrate thickness, \( h_2 \) is the tooth height, and \( c \) is the tooth gap spacing. Since the condition for synthesizing in-plane bending traveling waves is to find two bending vibration standing waves with equal frequency and amplitude and 90° difference between space and time [21], [22], the length sum of the tooth and the tooth gap is strictly controlled to meet the quarter wavelength in the parametric design [23], and there is \( c = l_1 + l_2 = \lambda/4 \) in the corresponding figure. At the same time, the perimeter of the vibrator is an integral multiple of the bending vibration wavelength, so as to reduce the difficulty of searching [24].

It is found that no matter how to adjust and combine the size parameters; it is difficult to find two ideal bending vibration modes. When the frequency difference decreases, the corresponding vibration mode is not ideal, but when the vibration mode is more reasonable, the frequency difference is very different, and some vibration modes also have obvious wavelength non-uniformity, and even other order vibration modes are mixed between the two same order modes.

At the same time, it is required that the symmetrical conveying vibrator is modular combined in the width direction [25], so it is necessary to take into account the traverse vibration of the vibrator in the width direction. In fact, there is obvious bending vibration in the width direction. The key factor causing the bending vibration mode of the vibrator is the structure of the ceramic sheet and its bonding.

**FIGURE 9. Change of natural frequencies for the same two order modes with 2l and 2r.**

**FIGURE 10. Partial enlarged drawing of vibrator tooth.**
position [26]. Therefore, the whole ceramic sheet is replaced with independent small ceramic sheets. However, it is found that the position accuracy of ceramic sheet paste to vibrator is difficult to ensure [27].

Therefore, it is envisaged that proper slotting on the matrix surface of the vibrator elastomer can improve the positioning accuracy of the ceramic sheet on the one hand [28] [29], it is expected that the modal correction is achieved while shortening the analysis process on the other hand [30], such as the structural diagram of slotted vibrator with independent piezoceramics is shown in Figure 11.

Through the strict control of size parameters, the B_{18} bending mode of the vibrator is finally optimized. The vibrator width $b$ is 16mm, the length of the upper and lower straight beam sections $l$ is 208.5mm, the matrix thickness $h_1$ and tooth height $h_2$ are 2mm, the tooth length $l_1$ and the ceramic sheet length are 6mm, the tooth gap $l_2$ is 0.5mm, the groove depth $h_3$ and groove length are 0.5mm, the ceramic sheet thickness $h_4$ is 1mm, and the number of teeth contained in the upper and lower straight beam sections is 32.

Because the large frequency difference is not conducive to the synthesis of traveling waves, it is necessary to adjust some geometric dimensions of the vibrator to reduce the frequency difference without affecting the vibration mode.

Through parameter analysis, it is found that the inner radius of the circular arc at both ends $r$ is the best optimization object. Therefore, the inner radius of the circular arc at both ends $r$ is adjusted slightly, and finally set to 5.3mm; the final result of finite element analysis is shown in Figure 12.

The shape of the vibrator before and after deformation is shown in Figure 12. The natural frequencies of the two orthogonal modes are 21921Hz and 21923Hz respectively, and the frequency difference is 2Hz.

It is observed that the tooth gap at the antinode and the slotting are basically maintained in the same vertical direction before and after deformation. Therefore, the slotting treatment of the vibrator elastomer can effectively control the bending vibration mode of the vibrator.

**B. OPTIMAL DESIGN OF SYMMETRICAL CONVEYING VIBRATOR**

A vibrator structure with 32 teeth in both the upper and lower straight beam sections is determined by the finite element analysis. However, by observing the $B_{18}$ vibration mode of the vibrator, it is found that the waveform of each standing wave is very similar to that of the Saw tooth wave, but in fact, the two standing wave waveforms constituting traveling wave are similar to sinusoidal waveforms.

During the analysis, the reason for this phenomenon is that the length ratio between the tooth and its gap is too large. In addition, increasing the number of teeth on the vibrator surface is conducive to increase the transverse amplitude of the vibrator.

Therefore, the teeth number of this structure vibrator is adjusted, that is, a tooth gap is added at the center of the original tooth, and the corresponding number of teeth is doubled. The structure diagram of the modified vibrator is shown in Figure 13.

The vibration modes of the two vibrator modes are shown in Figure 14. By observing the finite element analysis results, it is found that the increases of tooth gap effectively reduce the bending stiffness of the vibrator, but it doesn’t affect the vibration order of the conveying vibrator, which is still $B_{18}$. Slightly adjust the inner radiuses of the arc at both ends $r$ to 5.9mm, the natural frequencies are 20326Hz and 20327Hz respectively, the frequency difference is 1Hz. obviously, the vibration mode in Figure 14 is significantly improved compared with Figure 12.

**C. SENSITIVITY ANALYSIS OF VIBRATOR GEOMETRY ON VIBRATION MODE**

In the production process, when there are strict requirements on the conveying width, the vibrator width is adjusted; however, with the increasing of conveying width, transverse bending vibration occurs in the width direction of vibrator. Therefore, it is required that the conveying vibrator is modular combined in the width direction, the conveying width
is designed within the size range without large transverse bending vibration.

In order to select the best design width of the vibrator and keep the other dimensions of the vibrator structure in Figure 13 unchanged, only change the dimensions in the width direction. Observe the bending mode of the vibrator from the ANSYS figure. It is found that the corresponding vibration modes under different widths are very similar, but the deformation in the width direction is slightly different.

Therefore, the width dimensions are designed in series, the vibrator natural frequencies and frequency differences of the two bending modes under different widths are shown in Table 5.

**TABLE 5. Natural frequency and frequency difference of two modes for vibrator with different width.**

| Width (mm) | Mode 1 frequency (Hz) | Mode 2 frequency (Hz) | Frequency difference (Hz) |
|------------|-----------------------|-----------------------|---------------------------|
| 14         | 20301                 | 20299                 | 2                         |
| 15         | 20314                 | 20313                 | 1                         |
| 16         | 20327                 | 20326                 | 1                         |
| 17         | 20338                 | 20339                 | 1                         |
| 18         | 20349                 | 20350                 | 1                         |
| 19         | 20359                 | 20360                 | 1                         |
| 20         | 20368                 | 20370                 | 2                         |
| 21         | 20376                 | 20379                 | 3                         |
| 22         | 20384                 | 20388                 | 4                         |

Compare the modes of different models, and finally, the vibrator structure with a width of 16mm is selected as the object of solid processing.

The vibration mode is closely related to the design of the conveying vibrator; the parameter sensitivity of geometric size on the B18 mode of vibrator is analyzed. Corresponding to Figure 13, the main objects of analysis are: vibrator substrate thickness is $h_1$, tooth height is $h_2$, boss height is $h_3$, inner radius of arc at both ends is $r$, tooth gap is $l_2$ and piezoelectric ceramic thickness is $h_t$. During the analysis, ensure that other dimensions of the vibrator remain unchanged, change any one size of the analysis object and observe the influence of the size on the mode, and adjust the step size is 0.1mm. The analysis results are shown in Figure 15.

It is found out from Figure 15 that the coincidence degree of every two curves under the substrate thickness $h_1$, tooth height $h_2$, tooth gap $l_2$ and piezoelectric ceramic thickness $h_t$ are better, indicating that the change of these parameters have little effect on natural frequency difference between two modes, which is relaxes when considering the machining accuracy. But from Figure 15 (c) and (d), it is seen that the variation of $h_3$ and $r$ have a great influence on the natural frequency difference between the two modes, so the dimensional accuracy should be strictly controlled during machining. Through parameter sensitivity analysis, it is found that the inner radius of the circular arc at both ends $r$ is the best optimization object, and finally set to 5.9mm; The above analysis results not only shows the sensitivity of geometric dimensions to vibrator modes, but also provides a theoretical basis for the design of this type conveying vibrator in the future.

**FIGURE 15. Parameter sensitivity analysis of vibrator modes.**

**IV. WHOLE MACHINE EXPERIMENT OF ULTRASONIC VIBRATION FEEDING DEVICE**

The ultrasonic vibration feeding device is mainly composed of conveying vibrator, support structure and support. The physical drawing of the prototype is shown in Figure 16, the conveying vibrator is overlapped in the through hole of the support through the support rod, and the support rod and the support are fastened with screws.

The prototype design adopt 64 independent ceramic sheets, and two fixtures with the same structure are designed and manufactured for the pasting of ceramic sheets, as shown
in Figure 16 (a); The polarity (+ - sign represent the polarization direction of the ceramic sheet) and position are shown in Figure 16 (b), the piezoelectric ceramics are accurately arranged in the groove of the fixture with plastic film and double-sided adhesive tape; the wiring of the prototype is welded, as shown in Figure 16 (c). Each piece of piezoelectric ceramic is welded with a wire, and the ground wire is led out from the substrate.

A. EXPERIMENTAL INSTRUMENTS AND SCHEME

The experimental device and test scheme are shown in Figure 17. The sinusoidal signal is generated by the signal generator, and the appropriate voltage is output by the power amplifier, and load onto the piezoelectric ceramic chip of the conveying vibrator, the vibration information is transmitted to the oscilloscope by the laser vibrometer, and the amplitude of vibration is displayed in the oscilloscope.

B. RESONANCE FREQUENCY TEST OF PROTOTYPE

Through finite element analysis and calculation, the resonant frequencies of the B18 order two modes are 20326Hz and 20327Hz respectively. One group of ceramic sheets is excited, a fixed point is selected on a conveying surface of the vibrator, the frequency of the signal generator is adjusted near the resonant frequency of the theoretical analysis, and several resonance frequency points are found. It is tested that the B18 order two-phase working frequencies of the vibrator are 18330Hz and 18420Hz respectively, and the frequency difference is 90Hz. There is a certain deviation compared with the frequency value of theoretical analysis. There are many possible reasons for this phenomenon, such as the adhesive layer is not considered in the finite element modeling, the selection of grid size in finite element analysis, the setting of material parameters, and subsequent processing errors, the position error when the piezoelectric ceramic is pasted with the vibrator elastomer, etc.

C. TESTING OF MODAL CONTROL

In order to quantitatively verify whether the actual working mode of the conveying vibrator was consistent with the theoretical analysis; that was, whether the real modal control was achieved, it was necessary to analyze the amplitude distribution of the vibrator tooth surface and the orthogonality of the vibrator’s two-phase modes.

1) ORTHOGONALITY OF TWO-PHASE MODES

In order to verify the existence of orthogonality in the theoretical analysis, the data in the length direction amplitude test are extracted, as shown in Figure 19(a) and (b). It is found that the actual working mode has good orthogonality.

2) AMPLITUDE DISTRIBUTION ALONG THE LENGTH

Set the input voltage peak value as 40V, excite the ceramic sheets of groups A and B respectively, and the excitation frequencies are 18350Hz and 18440Hz respectively. On the symmetrical plane of the vibrator, on the symmetrical plane of the vibrator, starting from the position of the first tooth at the left end of the working face, took three points on each tooth as the test point, a total of 64 teeth, i.e. 192 test points. Record the voltage peak value of each test point, and then convert it into amplitude. According to this method, the amplitude distribution of the tooth surface along the length of A and B side is tested respectively, as shown in Figure 20 and Figure 21.

In the figures, the thick line represents the actually measured amplitude curve, and the dotted line represents the amplitude curve of finite element analysis. Since the actually measure amplitude values are positive, it isn’t convenient to compare with the finite element analysis results, the actual
measure amplitude curve is simulated and represented by dotted lines, and the thin line represents the tooth profile curve.

As illustrated in Figure 20 and Figure 21, the actual working mode is basically the same as that of the finite element analysis, and the side A and B achieve good modal control.

3) AMPLITUDE DISTRIBUTION IN WIDTH DIRECTION

In theoretical analysis, not only observe the vibration mode in the length direction, but also pay attention to the vibration mode in the width direction. Set the input voltage peak value as 40V, excite the ceramic sheets of groups A and B respectively, and the excitation frequencies are 18340Hz and 18420Hz respectively. Select a wavelength range (take the middle part of the vibrator structure) on the working face, a total of 8 teeth, select the center position of each tooth along the width direction, and select 7 points at each center position (take from the side close to the lead), a total of 56 test points, record the voltage peaks of each test point, and then convert them into amplitude. Test the amplitude distribution along the width direction of side A and B according to this method, as shown in Figure 22 and Figure 23.

The amplitude distribution along the length direction of the tooth widths is shown in Figure 22 (a) and (c). However, the curves shown in Figure 22 (a) are not uniform, the amplitude drop of each tooth along the tooth width direction is large, and the curves in Figure 22 (c) are relatively better. In order to more intuitively describe the amplitude distribution in the width direction, the amplitude distribution of each tooth along the center of the tooth width is shown in Figure 22 (b) and (d). Only the amplitude curve of individual teeth along the width direction has no obvious warping in Figure 22 (b), while the curve is relatively stable in Figure 22 (d).

Although there are similar problems in Figure 23, it is better than in Figure 22. Since the test is carried out only within one wavelength range of the vibrator working face, and there are individual warps in the width direction, there is no large bending deformation in the width direction for the whole conveying vibrator.

D. PERFORMANCE TEST AND ANALYSIS OF PROTOTYPE

The conveying performance of the whole device depends on the conveying rate, and it is an important standard to judge whether the conveying vibrator structure is feasible and achieve optimization.

1) FREQUENCY-AMPLITUDE CHARACTERISTICS

In order to analyze the Frequency-amplitude characteristics of the conveyor, select a fixed point on side A of the vibrator and set the peak value of the two-phase input voltage at 40V, the ceramic sheets of group A are excited, the working
frequency of phase A is 18350Hz as the reference value, adjust the frequency of the signal generator in steps of 100Hz, and record the corresponding amplitude value at each frequency. Similarly, the ceramic sheets of group B are excited, record the corresponding amplitude value at each frequency, draw the Frequency-amplitude characteristic curve, and fit the curve. The results are shown in Figure 24.

As shown in Figure 24, with the increase of working frequency, the amplitude of the conveying vibrator tooth surface also increases; when the working frequency reaches the resonant frequency of the system, the amplitude of the conveying vibrator tooth surface reaches the maximum; when the working frequency continues to increase, the amplitude of the conveying vibrator tooth surface is shown a decreasing trend.

2) RELATIONSHIP BETWEEN AMPLITUDE AND CONVEYING RATE
The conveying rate is an important standard to measure the conveying capacity of ultrasonic vibration feeding device, and the conveying rate is closely related to the amplitude of the conveying vibrator surface.

Select side A of the conveying vibrator for test. The conveying material is tapered roller, and the excitation frequency of the vibrator is set as the average value of two-phase
The corresponding voltage peaks under different amplitudes are obtained, and the conveying vibrator is moved to find the peak positions of two standing waves when the two-phase of the vibrator are excited respectively; By adjusting the power amplifier, the voltage peak of the particle at the standing wave peak position is consistent with the voltage peak corresponding to the calibrated amplitude, and measure the conveying rate under different amplitudes, the relationship curve between amplitude and conveying rate is obtained, as shown in Figure 25. The conveying rate of ultrasonic vibration feeding device increases with the amplitude increasing of vibrator surface.

3) EFFECT OF MATERIALS ON CONVEYING RATE
Using the above test method of conveying rate, the excitation frequency is set at 18390Hz, and the amplitude of the particle on the vibrator surface is adjusted to 1µm, the conveying
Taking the finite element software ANSYS as the main tool, the in-plane bending vibration modes of ring and long ring beam are compared and analyzed. The vibration mode symmetry of long ring beam is demonstrated by mathematical method, the inherent law of in-plane bending vibration of long ring beam is explored, and finally the prototype of symmetrical conveying vibrator is proposed.

In the dynamic design process of the conveying vibrator, the principle of quarter wavelength is strictly followed, and a feasible scheme that can control the vibration mode of the vibrator is proposed, that is, the elastomer surface of the vibrator is properly slotted, so as to force the antinode position of the vibration mode to make the conveying vibrator work in the ideal mode. The experimental verification proves the feasibility of the mode control method; it provides a theoretical basis for the design of this type conveying vibrator in the future.

Through the modal analysis of finite element software, Two B₁₈ order in-plane bending vibration modes of the vibrator are found, the natural frequencies are 20326Hz and 20327Hz respectively, and the frequency difference is only 1Hz, which meets the engineering requirements. By using the design method of pasting independent ceramic sheets, the transverse vibration in the width direction of the vibrator is minimized, which provides a basis for the serialization design of width dimensions. The influence of the main geometric size of the vibrator on the B₁₈ order bending mode is analyzed. The analysis shows that the substrate thickness, tooth height, tooth gap and piezoelectric ceramic thickness of the vibrator has little influence on the natural frequency difference between the two modes, while the boss height of the vibrator and the inner radius of the arc at both ends has a great influence on the natural frequency difference between the two modes.

According to the results of theoretical analysis, the principle prototype is developed, and the resonant frequency and vibration mode of the prototype are experimentally studied. The experimental results show that the B₁₈ order two-phase working frequencies of the vibrator are 18330Hz and 18420Hz respectively, and the frequency difference is 90Hz. There is a certain deviation compare with the frequency difference of theoretical analysis, but this deviation is within the allowable range. The amplitude distribution of the conveying vibrator tooth surface is studied. The results show that the actual working mode is basically consistent with the finite element analysis results. From the perspective of traveling wave synthesis, the existence of orthogonality in theoretical analysis is verified.

The conveying performance of ultrasonic vibration feeding device is tested, and the effects of amplitude and material type on the conveying rate are analyzed. The experimental results show that there is an approximate linear relationship between the conveying rate and the amplitude of vibrator surface; under the same working frequency and amplitude, different kinds of materials are selected for the conveying capacity test, and the results show that different conveying

| Material type       | Material characteristics                                                                 | Conveying rate (mm/s) |
|---------------------|------------------------------------------------------------------------------------------|-----------------------|
| Cap nut (M8)        | The material was stainless steel, the surface was very smooth, the quality was heavier, and it was easy to transport | 79.28                 |
| Hexagon nut (M5)    | The material was steel, with smooth surface, light weight and easy transportation         | 67.48                 |
| Bearing (604)       | The material was stainless steel with smooth surface and easy transportation               | 74.20                 |
| Tapered roller (2.5g)| The material was stainless steel, the surface was smooth and easy to transport            | 77.51                 |
| Shim (0.2g)         | The material was stainless steel, the surface was very smooth and easy to transport        | 75.82                 |
| Huangyao tablet (Ø10,0.5g)| The surface was smooth, the shape was large, and it was easy to transport | 65.57                 |
| Baiyao tablet (Ø7,0.1g)| Rough surface, small shape, difficult to transport                                       | 53.32                 |
objects has different conveying rates, and the conveying effect is better for materials with smooth surface and heavy quality.

REFERENCES
[1] H. Li, W. Chen, X. Tian, and J. Liu, “An experiment study on temperature characteristics of a linear ultrasonic motor using longitudinal transducers,” Ultrasonics, vol. 95, pp. 6–12, May 2019.
[2] L.-K. Chang and M.-C. Tsai, “Design of single-phase driven screw-thread-type ultrasonic motor,” Rev. Sci. Instrum., vol. 87, no. 5, May 2016, Art. no. 055002.
[3] X. C. Chu, J. Wang, S. Yuan, L. Li, and H. Cui, “A screw-thread-type ultrasonic actuator based on a Langevin piezoelectric vibrator,” Rev. Sci. Instrum., vol. 85, no. 6, Jun. 2014, Art. no. 065002.
[4] Y. He, Z. Yao, S. Dai, and B. Zhang, “Hybrid simulation for dynamic responses and performance evaluation of linear ultrasonic motors,” Int. J. Mech. Sci., vol. 153–154, pp. 219–229, Apr. 2019.
[5] C. Jiang, X. Wu, D. Lu, Z. Xu, and L. Jin, “Contact modeling and performance evaluation of ring type traveling wave ultrasonic motors considering stator teeth,” Ultrasonics, vol. 117, Dec. 2021, Art. no. 106518.
[6] Y. Deng, G. Zhao, X. Yi, and W. Xiao, “Contact modeling and input-voltage-region based parametric identification for speed control of a standing wave linear ultrasonic motor,” Sens. Actuators A, Phys., vol. 295, pp. 456–468, Aug. 2019.
[7] Y. Tanoue and T. Morita, “Opposing preloads type ultrasonic linear motor with quadrupled stator,” Sens. Actuators A, Phys., vol. 301, Jan. 2020, Art. no. 111764.
[8] J. L. Fan, Y. H. Zhang, and Y. H. Zhou, “Study on a novel in-plane longitudinal-bending compound linear ultrasonic motor,” Piezoelectrics Acustooptics, vol. 41, no. 6, pp. 819–823, 829, Dec. 2019.
[9] R. Niu, H. Zhu, and C. Zhao, “A four-legged linear ultrasonic motor: Design and experiments,” Rev. Sci. Instrum., vol. 91, no. 7, Jul. 2020, Art. no. 076107.
[10] B. Zhang, Z. Yao, Z. Liu, and X. Li, “A novel L-shaped linear ultrasonic motor operating in a single resonance mode,” Rev. Sci. Instrum., vol. 89, no. 1, Jan. 2018, Art. no. 015006.
[11] S. Izuhara and T. Mashimo, “Characterization of bulk piezoelectric element-based ultrasonic motors,” ROBOMECH J., vol. 7, no. 1, p. 31, Aug. 2020.
[12] S. Miyake, Tomohiro Harada, Hiroyuki Shimizu, Sumiaki Kishimoto, and Takeshi Morita, “High-power characteristics of (Bi, Na) TiO$_3$-BaTiO$_3$ ceramics and application in miniature ultrasonic motor,” Sensors Mater., vol. 32, no. 7, pp. 2443–2452, Jul. 2020.
[13] R. Niu, J. Liu, H. Zhu, and C. Zhao, “Design and evaluation of a novel light arc-shaped ultrasonic motor,” AIP Adv., vol. 9, no. 6, Jun. 2019, Art. no. 065009.
[14] F. Wang, U. Nishizawa, H. Tanaka, and S. Toyama, “Development of miniature spherical ultrasonic motor using wire stators,” J. Vibroeng., vol. 20, no. 8, pp. 2939–2950, Dec. 2018.
[15] K. Miyoshi and T. Mashimo, “Miniature direct-drive two-link using a micro-flat ultrasonic motor,” Adv. Robot., vol. 32, no. 20, pp. 1102–1110, Sep. 2018.
[16] Y. Liu, S. Shi, C. Li, W. Chen, and J. Liu, “A novel standing wave linear piezoelectric actuator using the longitudinal-bending coupling mode,” Sens. Actuators A, Phys., vol. 251, pp. 119–125, Nov. 2016.
[17] M. Kucera, E. Wistrela, G. Pfüsterschmied, V. Ruiz-Díez, T. Manzaneque, J. Hernando-García, J. L. Sánchez-Rojas, A. Jachimowicz, J. Schalke, A. Bittner, and U. Schmid, “Design-dependent performance of self-actuated and self-sensing piezoelectric-AlN cantilevers in liquid media oscillating in the fundamental in-plane bending mode,” Sens. Actuators B, Chem., vol. 200, pp. 235–244, Sep. 2014.
[18] S. L. Sharp, J. S. N. Paine, and J. D. Bloyter, “Design of a linear ultrasonic piezoelectric motor,” J. Intell. Mater. Syst. Struct., vol. 21, no. 10, pp. 961–973, Jul. 2010.
[19] W.-H. Lee, C.-Y. Kang, D.-S. Paik, B.-K. Ju, and S.-J. Yoon, “Butterfly-shaped ultra slim piezoelectric ultrasonic linear motor,” Sens. Actuators A, Phys., vol. 168, no. 1, pp. 127–130, Jul. 2011.
[20] Z. Li, Z. Wang, P. Guo, L. Zhao, and Q. Wang, “A ball-type multi-DOF ultrasonic motor with three embedded traveling wave stators,” Sens. Actuators A, Phys., vol. 313, Oct. 2020, Art. no. 112161.
[21] S. Borodinas, P. Vasiljev, D. Mazeika, R. Bareikis, and Y. Yang, “Design optimization of double ring rotary type ultrasonic motor,” Sens. Actuators A, Phys., vol. 293, pp. 160–166, Jul. 2019.
[22] S. L. Zhou and Z. Y. Yao, “Design and optimization of a modal-independent linear ultrasonic motor,” IEEE Trans. Ultrason., Ferroelectr., Freq. Control, vol. 61, no. 3, pp. 46–53, Mar. 2014.
[23] Z. Huang, S. Shi, W. Chen, L. Wang, L. Wu, and Y. Liu, “Development of a novel spherical stator multi-DOF ultrasonic motor using in-plane non-axisymmetric mode,” Mech. Syst. Signal Process., vol. 140, Jun. 2020, Art. no. 106658.
[24] X. Yang, Y. Liu, W. Chen, and J. Liu, “A cylindrical traveling wave ultrasonic motor using bonded-type composite beam,” Ultrasonics, vol. 65, pp. 277–281, Feb. 2016.
[25] X. Xin, Y. Yu, J. Wu, X. Gao, Z. Li, X. Yi, W. Chen, and S. Dong, “A ring-shaped linear ultrasonic motor based on PSN-PMS-PZT ceramic,” Sens. Actuators A, Phys., vol. 309, Jul. 2020, Art. no. 112036.
[26] D. Lu, Q. Lin, B. Chen, C. Jiang, and X. Hu, “A single-modal linear ultrasonic motor based on multi vibration modes of PZT ceramics,” Ultrasonics, vol. 107, Sep. 2020, Art. no. 106158.
[27] L. Zhou, Z. Yao, S. Dai, Y. He, and H. Xu, “Modeling and verification of life prediction of a V-shaped linear ultrasonic motor,” Rev. Sci. Instrum., vol. 92, no. 4, Apr. 2021, Art. no. 045003.
[28] J. Yan, Y. Liu, J. Liu, D. Xu, and W. Chen, “The design and experiment of a novel ultrasonic motor based on the combination of bending modes,” Ultrasonics, vol. 71, pp. 205–210, Sep. 2016.
[29] H. Li, J. Deng, S. Zhang, H. Yu, and Y. Liu, “Design and experiment of a three-feet linear ultrasonic motor using third bending hybrid modes,” Sens. Actuators A, Phys., vol. 331, Nov. 2021, Art. no. 112990.
[30] S. Izuhara and T. Mashimo, “Design and characterization of a thin linear ultrasonic motor for miniature focus systems,” Sens. Actuators A, Phys., vol. 329, Oct. 2021, Art. no. 112797.