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ABSTRACT
By analyzing the contact mechanism between the stator and rotor, this study demonstrates that the prepressure on the rotor is not a constant value because of the structural stiffness of the preload structure. In addition, this study explains the driving mechanism of the traveling wave ultrasonic motor under unsteady prepressure and deduces a dynamic model considering the structural stiffness of the preload structure. Furthermore, the relationship between the speed characteristics of the motor and the structural stiffness of the preload structure is obtained through model simulation. The simulation results are found to be in good agreement with the experimental results, which demonstrates the correctness of the dynamic model. Moreover, the output characteristics of the motor can be improved by designing an appropriate structural stiffness of the preload structure.

I. INTRODUCTION
The rotary traveling wave ultrasonic motor (TWUSM) exhibits a good potential for application in many precision servo control platforms owing to its advantages, such as fast response, good position, speed control, and being free of external magnetic field interference. Extensive studies have been conducted on its theoretical model to improve the design efficiency and achieve accurate control of the ultrasonic motor. Furthermore, the finite element method (FEM) is used to design ultrasonic motors. Numerical investigation has been used in many fields, and many scholars have also applied it to the simulation of an ultrasonic motor. A few studies have presented an equivalent circuit model of ultrasonic motors and discussed its applicability to evaluate the characteristics of the motor. Hagood and McFarland proposed a general framework for modeling to predict the motor performance as a function of design parameters. Moreover, the contact model of the TWUSM considering the preload and the load torque has been established. Giraud et al. established the original modeling of a TWUSM to provide a new torque control law for this category of actuators. Kühne et al. presented a novel second-order model for TWUSMs to control them at low speeds. This study analyzes the performance evolution of a traveling-wave type rotary ultrasonic motor in a high-rotating environment and establishes a dynamic model of the stator of the TWUSM in a centrifugal field.

Under a specific prepressure, the rotor and stator of ultrasonic motors are in contact and the normal pressure is needed for the friction drive. Whether the disk spring of the rigid rotor or the flexible rotor is used, the objective is to make the rotor flexible (structural stiffness) to realize an appropriate adjustment space to apply the required prepressure. The stator amplitude of the TWUSM changes significantly when starting or adjusting the speed, which leads to an axial displacement of the rotor. Considering the structural stiffness of the preload structure, the axial movement of the rotor affects the preloading between the stator and the rotor. The mass-spring system comprising the mass of the rotor and structural stiffness of the preload structure directly affects the output characteristics of the TWUSM, when the contact interface force changes because of the change in the stator amplitude. When the parameters...
of the mass-spring system are specific values, the motor speed may fluctuate for a long period. Although this phenomenon has been experimentally observed, no intensive study and detailed analysis are available. Most of the present studies assume that the rotor and the stator of the ultrasonic motor are in contact under a constant pre-pressure, establish a dynamic model, and analyze the output characteristics of the motor. However, the experimental phenomenon of large speed fluctuation of the motor has not been reasonably explained.

Based on the analysis of the contact mechanism between the stator and rotor, this study demonstrates that the prepressure in the rotor is generally not a constant value because of the structural stiffness of the preload structure. Considering the structural stiffness of the preload structure, the rotor additionally acts as a mass-spring system in the axial direction, whose boundary excitation originates mainly from the contact interface force between the stator and the rotor. When the motor starts or adjusts its speed, the stator amplitude changes significantly, which changes the interface excitation and the driving effect of the rotor, which affect the output characteristics of the motor. Specifically, when the structural stiffness, rotor mass, and moment of inertia of the preload structure are under specific conditions, the speed of the motor fluctuates significantly for long periods. When the motor is required to adjust its speed frequently in a servo system, these characteristics inevitably affect the overall efficiency of the entire system.

Considering the structural stiffness of the preload structure in the above-mentioned scenario, this study establishes a dynamic model of the rotor, explains the driving mechanism of the TWUSM under unsteady pre-pressure, and deduces a dynamic model. The relationship between the speed characteristics of the motor and the structural stiffness of the preload structure is established through simulation. Rotors with different structural stiffness are trial-manufactured based on the simulation results. The experimental results were found to be in good agreement with the simulation results. Thus, it is proved that the appropriate structural stiffness of the preload structure can improve the output characteristics of the motor.

II. MOTOR PERFORMANCE MODEL

The TWUSM has two main components, namely, stator and rotor, as depicted in Fig. 1. The rotor remains pressed on the stator under the action of the preload, which provides the necessary pressure for friction transmission. During the functioning of the motor, a specific alternating voltage is applied to the piezoelectric ceramics, a travelling wave can be derived in the stator based on the inverse piezoelectric effect of the piezoelectric ceramics, and electrical energy can be transformed into ultrasonic vibration of the stator. Then, the high-frequency and small-amplitude vibrations of the stator are transformed into the rotating motion of the rotor driven by contact friction, thereby realizing the energy conversion from electric energy (input) to mechanical energy (output).

The friction transmission between the stator and the rotor has a considerable effect on the output performance of ultrasonic motors and is mainly affected by the preload. In the early stages of establishing the friction drive model of the motor, irrespective of the preload originating from the deformation of either the flexible rotor itself or the flexible mechanisms, such as disc springs (for the combination of the rigid rotor with the disc spring and rigid rotor), the preload was assumed to be constant, as illustrated in Fig. 2. During the functioning of the motor, the stator is excited to generate traveling waves, the profile of the stator surface is sinusoidal, regardless of the vibration amplitude of the stator, and the preload applied to the rotor is set as a constant value ($F_N$).

Moreover, both the flexible rotor and the combination of the rigid rotor with the disc spring and rigid rotor have a deformation stiffness, as depicted in Fig. 3. In this figure, $K_R$ and $C_R$ represent the stiffness and damping of the preload structure, respectively.

The values of $K_R$ and $C_R$ for the flexible rotor and the combination of the rigid rotor with the disc spring and rigid rotor are different. For the combination of the rigid rotor with the disc spring and rigid rotor, the preload arises from the elastic deformation of the disc spring and $K_R$ and $C_R$ represent the stiffness and damping of the disc spring, respectively. Furthermore, the preload of the flexible rotor arises from its own deformation, and $K_R$ and $C_R$ represent the stiffness and damping of the flexible rotor under preload deformation.

A. Rotor dynamic model

Considering the structural stiffness of the above-mentioned preload structure, the dynamic model of the rotor can be expressed as

![FIG. 1. Schematic of the stator and rotor of the TWUSM.](image1)

![FIG. 2. Contact state between the stator and rotor with a constant prepressure.](image2)
FIG. 3. Contact state between the stator and rotor with a preload structure.

\[
M_R \ddot{Z}_R + C_R \dot{Z}_R + K_R Z_R = F_{RN} - F_N, \quad (1a)
\]

\[
J_R \ddot{\beta} + C_J \dot{\beta} = T_C - T_I, \quad (1b)
\]

where \(M_R, J_R, C_J, Z_R, \beta, F_{RN}, F_N, T_C,\) and \(T_I\) represent the rotor mass, rotor rotary inertia, rotary damping, axial freedom of the rotor, rotation freedom of the rotor, normal support force from the contact interface to the rotor, preload on the rotor, drive torque from the contact interface to the rotor, and motor load torque, respectively.

It can be seen that, when the structural stiffness \(K_R\) is considered, the axial motion of the rotor is additionally represented as a spring oscillator system: \(M_R - K_R\); the external excitation of the system originates from the interface force \(F_{RN}\). Apparently, Eqs. (1a) and (1b) are coupled, and changing \(F_{RN}\) causes the axial vibration of the rotor. The contact area and stress distribution between the stator and the rotor are affected, and the driving effect of the contact interface and the output characteristics of the motor are thus affected. Specifically, if the structural stiffness of the preload applying structure, rotor mass, and moment of inertia \((M_R - K_R - J_R)\) reach a certain condition, when the interface excitation changes, the output speed of the motor continues to significantly fluctuate. For example, when the motor starts, turns off, or controls its speed by frequency modulation, the vibration amplitude of the stator changes significantly and the oscillation duration of the rotor is longer subjected to varying interfacial axial excitation. Furthermore, the interface driving torque \(T_C\) experiences an oscillatory change, and the output speed of the motor fluctuates for long periods.

B. Contact interface model

To improve the contact characteristics of the stator and rotor, a soft layer of the friction material is attached on the contact face of the rotor, as illustrated in Fig. 4, where \(\vec{r}, \vec{\theta},\) and \(\vec{z}\) represent the radial, circumferential, and axial unit vectors in the cylindrical coordinate system. During the functioning of the ultrasonic motor, the traveling wave is excited in the stator and the contour of the stator surface is wavy. Under the action of the preload, because the material of the stator and rotor is hard, it can be assumed that the stator and rotor undergo no contact deformation and only the friction layer has a corresponding deformation, indicating that the deformation of the friction layer is consistent with the contour of the stator surface on the contact interface between the stator and the rotor. Neither the combined rigid rotor nor the flexible rotor produces additional contact deformation because of contact, and the flexible rotor undergoes only structural deformation under preloading.

Figure 4 depicts that when the stator and the rotor are in contact, only the friction layer undergoes deformation because of contact, the contact area is an arc with a very small radial width \(\varepsilon\), and the radius of its middle position is \(r_C\). The contact area between the stator and rotor is a ring area, as depicted in Fig. 5.

The force acting on the contact interface between the stator and the rotor can be decomposed into normal contact force \(f_n\) and tangential friction force \(f_t\). Assuming that the normal component of the contact force is proportional to the compressive deformation of the friction layer, the tangential friction meets the Coulomb friction theory; thus,

\[
f_n = k_s \delta, \quad \text{(2a)}
\]

\[
f_t = \mu_d f_n, \quad \text{(2b)}
\]

where \(k_s\) represents the equivalent distributed spring stiffness coefficient for the friction layer and \(\mu_d\) represents the dynamic friction coefficient of the contact interface between the stator and the rotor.
Within a specific wavelength, the contact between the stator and rotor is illustrated in Fig. 6, where \( v_r \) and \( v_t \) represent the moving direction of the travelling wave in the stator and the rotating direction of the rotor, respectively. Furthermore, \( \theta_0 \) and \( \theta_3 \) represent an equal speed point where the contact points on the stator share the same circumferential speed and the contact point where the stator and rotor start contact, respectively. In a specific wavelength, the equal speed points \( \theta_0 \) divide the contact area into two different regions of intervals: the first interval is \((-\theta_0, -\theta_3)\) and \((\theta_0, \theta_3)\), where the circumferential velocity of the stator surface particles is less than the linear velocity of the rotor, resulting in the stator preventing the rotor (the intervals are marked by "-" in Fig. 6). The second interval is \((-\theta_3, \theta_0)\), where the circumferential velocity of the stator surface particles is larger than the linear velocity of the rotor, and the stator causes the rotor to rotate (the interval is marked by "+" in Fig. 6). The friction layer is deformed because of the contact between the stator and rotor, and the axial deformation of the teeth is extremely small in comparison with that of the stator substrate. Thus, the compressive deformation of the friction layer can be expressed as

\[
g(r, \theta, z, t) = w + h_z + h_t - z(t),
\]

where \( w, h_z, h_t, \) and \( z \) represent the axial displacement of the stator substrate, height from the top of the stator substrate to the midplane of the stator, tooth height, and distance from the bottom surface of the friction layer to the midplane of the stator, respectively.

The axial support force from the interface to rotor \( F_{RN} \) can be obtained by integrating Eq. (2a) on the contact area depicted in Fig. 6 as

\[
F_{RN} = \int s f_s ds.
\]

Furthermore, the interface driving torque of the rotor \( T_C \) can be obtained by integrating Eq. (2b) on the contact area,

\[
T_C = \int s f_t r ds,
\]

where \( r \) represents the contact radius and the sign function \( \gamma \) is determined by the relative speed of the stator and rotor at the contact point,

\[
\gamma = \text{sign}(V_R, V_S) = \begin{cases} +1 & V_S > V_R \\ -1 & V_S < V_R \end{cases}.
\]

Apparently, when the motor starts, turns off, or controls its speed by frequency modulation, the traveling wave amplitude of the stator changes, causing the contact area and the amplitude between the stator and rotor to change. Moreover, the interfacial excitation to the rotor \( F_{RN} \) changes accordingly, a new disturbance is generated in the spring vibration system \((M_R - K_R)\) of the rotor in the axial direction, and the output speed of the motor continues to significantly fluctuate under a specific condition. Generally, such characteristics of the motor are disadvantageous to the control system in servo control.

C. Dynamic model of stator

The stator of the ultrasonic motor is a bending vibration circular plate excited by a piezoelectric, and the displacement field of the stator can be described based on the thin plate theory and the hypothetical modal method.

The displacement of the neutral plane of the stator in bending vibration can be discretized by the Ritz method,

\[
u_{s0} = \begin{bmatrix} u_0 \\ v_0 \\ w_0 \end{bmatrix} = \begin{bmatrix} \varphi_{s01} & \varphi_{s02} & \cdots & \varphi_{s0n} \\ \varphi_{s11} & \varphi_{s12} & \cdots & \varphi_{s1n} \\ \varphi_{s21} & \varphi_{s22} & \cdots & \varphi_{s2n} \end{bmatrix} \varphi_{smech} q_s = \varphi_{smech} q_s.
\]

where \( u_0, v_0, \) and \( w_0 \) represent the displacement of the neutral surface along the radial, circumferential, and z directions, respectively, in cylindrical coordinates, \( \varphi_{s0i}, \varphi_{s1i}, \varphi_{s2i} \) (\( i = 1, 2, 3, \ldots, n \)) is the shape function that satisfies the boundary conditions, \( \varphi_{smech} \) is the vibration mode function of the neutral surface, and \( q_s \) represents the modal coordinate. When the motor functions, only two orthogonal modes \( B_{0m} \) (0 for pitch circle and \( m \) for pitch diameter) of the same frequency, are excited in the stator, the modal coordinates in the above-mentioned formula are reduced to two and the modal function of the neutral surface is simplified,

\[
q = \begin{bmatrix} q_1 \\ q_2 \end{bmatrix}^T,
\]

\[
\varphi_{smech} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix},
\]

where \( \varphi_1 \) and \( \varphi_2 \) are the two hypothetical shape functions corresponding to the modes and are written as

\[
\varphi_1 = F(r) \cos(m \theta),
\]

\[
\varphi_2 = F(r) \sin(m \theta),
\]
TABLE I. Physical parameters of the presented motor.

| Parameter                        | Values/properties |
|----------------------------------|-------------------|
| Outer diameter (mm)              | 60                |
| Working frequency (kHz)          | 40–45             |
| Driving voltage (peak-to-peak) (V) | 400            |
| Preload (N)                      | 200               |
| Stator substrate material        | Phosphor bronze   |
| Piezoelectric ceramics           | Pb(ZrxTi1-x)O3-8 (PZT-8) |
| Rotor material                   | Duralumin         |
| Rotor mass (g)                   | 28                |
| $K_R$ (N m$^{-1}$)               | 10 000            |
| Contact layer material           | Polytetrafluoroethylene |
| Thickness of the contact layer (mm) | 0.3            |
| Friction coefficient             | 0.18              |
| Contact stiffness (N m$^{-1}$)   | $6.4 \times 10^8$ |

where $F(r)$ is the radial displacement distribution function and $\cos(m\theta)$ and $\sin(m\theta)$ represent the displacement distribution functions of two orthogonal modes along the circumferential direction. Furthermore, the expression of the displacement field in the stator can be obtained by the Kirchhoff thin plate theory,

$$
\begin{bmatrix}
    u \\
    v \\
    w
\end{bmatrix} = L_{\text{mid}} \Phi_{\text{mech}} q = 
\begin{bmatrix}
    \Phi_{u} \\
    \Phi_{v} \\
    \Phi_{w}
\end{bmatrix} q = \Phi_{\text{mech}} q, \quad (11)
$$

where $u$, $v$, and $w$ indicate the radial, circumferential, and axial displacement vectors of the stator, $L_{\text{mid}}$ is a differential operator matrix, and $\Phi_{\text{mech}} = L_{\text{mid}} \Phi_{\text{mech}}$ is the mode function matrix of the whole stator. The energy expressions of the stator can be obtained using the displacement field function, and the dynamic equation of the stator can be derived by the integral operation based on Hamilton's variational principle,

$$
M_s \ddot{q} + C_s \dot{q} + K_s q = \Delta v + (F_{sn} + F_{st}), \quad (12)
$$

where $M_s$ and $K_s$ represent the mass matrix and the stiffness matrix of the stator, respectively, and $\Delta$ is a constant coefficient matrix that converts the excitation voltage $v$ into modal force (known as the electromechanical coupling coefficient) and reflects the ability to convert the electrical input into the mechanical response. $F_{sn}$ and $F_{st}$ represent the modal force corresponding to the interface contact force in normal and tangential directions. The explicit expressions of the

FIG. 7. Time history of the axial force and vibration displacement of the rotor. (a) Force acting on the rotor along the $z$ direction. (b) Driving torque acting on the rotor along the $z$ direction. (c) Rotor displacement along the $z$ direction. (d) Rotor speed along the $z$ direction.
mass and stiffness matrix of the stator are as follows:

\[
M_s = \rho_s \int_V \Phi_{smech}^T \Phi_{smech} dV + \rho_p \int_V \Phi_{smech}^T \Phi_{smech} dV, \quad (13a)
\]

\[
K_s = \int_V N_{smech}^T \Phi_{smech}^T \Phi_{smech} dV + \int_V N_{smech}^T \Phi_{smech} N_{smech} dV, \quad (13b)
\]

where \(\rho_s\) and \(\rho_p\) represent the densities of the stator substrate and piezoelectric ceramics, respectively, \(c_s\) and \(c_p\) are the stiffness matrices of the stator substrate and piezoelectric ceramic, \(V_s\) and \(V_p\) denote the volume integral of the stator substrate and the piezoelectric ceramics, and \(N_{smech}\) is the strain matrix given by

\[
N_{smech} = L_{smech} \Phi_{smech}
\]

where \(L_{smech}\) is an operator matrix of a pure differential sign.

Combining Eqs. (1), (2), (4), (5), and (12) and considering the damping of the structure, the dynamic equation of the entire motor can be obtained as

\[
\begin{align*}
M_s \ddot{q} + C_s \dot{q} + K_s q &= \Delta v + (F_{st} + F_{fr}) \\
M_R \ddot{Z}_R + C_R \dot{Z}_R + K_R Z_R &= \int f_s ds - F_N \\
J_R \ddot{\beta} + C_R \dot{\beta} &= \int y f s ds = T_l
\end{align*}
\]

(15)

III. SIMULATION AND ANALYSIS

Considering TWUSM-60 as an example, a performance simulation is performed based on the dynamic equation (15) of the whole motor. The parameters of TWUSM-60 used are listed in Table I; the rotors are selected with nonlinear stiffness, the stiffness of which is small (<20 000 N m \(^{-1}\)) when the rotor is under 200 N, and the stiffness of the preloading structure is obtained by the FEM or a pressure experiment.

The time history of the axial force and the vibration displacement of the rotor can be obtained by simulation, as depicted in Fig. 7. The axial contact force and the driving moment of the rotor tend to stabilize after oscillation and eventually stabilize at a certain displacement. In addition, the speed of the rotor along the axis produces an oscillation, which eventually tends to stabilize at a slight oscillation near zero during the starting process of the motor. Apparently, during the starting of the motor, the amplitude response of the stator excites the vibration of the rotor along the axis, which makes the rotor-preloaded structural stiffness system \((M_R - K_R - J_R)\) generate a vibration response along the \(z\) direction.

IV. EXPERIMENT

Considering the effects of \(K_R\) on the response of the ultrasonic motor, this study focuses on designing an experimental platform consisting of devices, such as high precision speed sensor, agency of linear displacement, pressure sensor, and linear encoder, as shown in Fig. 9. The agency linear displacement works forward to make the preload of the motor, the linear encoder is used for getting the deformation of the rotor, the pressure sensor is used for getting the preload of the motor, and the speed torque sensor is used for getting the speed and torque of the motor. Furthermore, the starting and the velocity jump characteristics of the TWUSM with different \(K_R\) are tested and are compared with the simulation results.
A. Analyzing the starting characteristics of the motor with different $K_R$

Figure 10 depicts the experimental and simulation results of the starting characteristic curves of motors with different $K_R$ at different frequencies. Because the parameters of simulation differ from those of the actual motor and the systematic damping of the actual motor varies at different frequencies, the simulation can observe the effect of only one parameter. Therefore, some discrepancies exist between the simulation and the experimental results; however, they are in good agreement. The motor speed reaches an average value with obvious pulsation during the starting process when $K_R$ is $10000 \text{ N m}^{-1}$, and the motor speed attains a constant value in a very short time when $K_R$ is $500 \text{ N m}^{-1}$.

B. Analyzing the velocity jump characteristics of the motor with different $K_R$

Adjusting the driving voltage and the frequency can change the speed of the motor by adjusting the amplitude of the stator. Because adjusting the frequency can be easily realized technically, it is performed in actuality to control the speed of the motor. Apparently, in the process of adjusting frequency, the change in the stator amplitude causes a change in the contact interface force on the rotor, which results in a change in the motion of the rotor both along and around the axis. However, in previous studies, the dynamic response in the process of adjusting the frequency often neglected is that change of the frequency changes the amplitude of the stator and the axial ($Z_R$) excitation of the rotor, which affects boundary conditions of the rotor axial ($M_R$-$K_R$) system and finally changes rotation response of the rotor.

The velocity jump curves obtained by experiment and simulation are illustrated in Fig. 11. The curves obtained by simulation and experiment are consistent, and $K_R$ affects the speed response of the motor in the process of adjusting the frequency. When the value of $K_R$ is $10000 \text{ N m}^{-1}$, the frequency is varied from one value to another specific value; the speed of the motor is unstable for a long period. Furthermore, if the external inertial load is considered, the speed of the motor becomes more unstable, which is obviously unfavorable to the servo control system in terms of accuracy and speed tracking. However, when the value of $K_R$ is $500 \text{ N m}^{-1}$, frequency is varied from one value to another specific value; the speed of the motor is unstable for a very short period. Thus, designing an appropriate $K_R$ can reduce the speed fluctuation time when the speed is varied.
V. CONCLUSION

In this study, the structural deformation characteristics of the flexible rotor and the combination of the rigid rotor with the disc spring and rigid rotor under preloading were analyzed. The results indicated that the structural stiffness constitutes a mass-spring system along the axis of the rotor and that the vibration system of the stator, vibration system of the rotor along the $(M_R-K_R)$ axis, and the rotating motion equation are coupled together, which considerably influences the speed characteristics of the TWUSM. Furthermore, the rotor (axial) vibration equation, contact model, and stator model considering the stiffness of the preload structure were established. Furthermore, the simulation results were in good agreement with the experimental results, indicating that the driving moment of the stator to rotor is similar to the pumping action because of the stiffness of the preload structure, which causes the rotor to vibrate. This effect can be attenuated by changing the inertia of the rotor or reducing the stiffness of the preload structure to improve the response characteristics of the ultrasonic motor.

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