Design and experimental study of the rolling-enhanced acoustic system for gear tooth surface

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Abstract
Parameters of gear flanks, such as roughness, hardness, and residual stress, impact their working performance. Therefore, it is essential to optimize these parameters for increasing gears’ life. Thus, this study combined the meshing theory and the ultrasonic surface rolling technology and presented an innovative design to realize ultrasonic burnishing. First, the dimensions of the ultrasonic burnishing device with two nodes integrated with a gear at the bottom were calculated using frequency equations by a numerical method. Second, the correctness of the calculating results was verified by simulations and experiments. Third, the influence of the ratio between the width and diameter of the gear on the acoustic system was studied. Finally, experimental platform of the rolling-enhanced acoustic system for gear tooth surface is built successfully. A series of experiments were conducted, and the results showed that the roughness can be reduced by 76% at most, and the hardness and the residual stress can be increased by 53% and 415%, respectively. Therefore, this design provided a new theory and technology to machine complicated surfaces such as gear flanks.

Keywords Double Nodes · Gear flank · Horn · Ultrasonic Surface Rolling · Residual Stress

1 Introduction

Driving characteristics require that the gear should bear wear resistance, high transmission efficiency, large bearing capacity, and so forth. The surface integrity of gears, including roughness, residual stress, and surface microstructure, determines the gears’ performance. Tonshoff and Marzenel [1] produced three groups of gears with different roughness and residual stress by changing the parameters of gear honing and carried out pitting corrosion experiments using these three groups of gears. The results proved that the roughness and stress had a great influence on the fatigue life. Besides the quality of the initial gear surface, the running conditions in the initial cycle, that is, the running-in, also affected the fatigue life. Therefore, changing conditions of the initial surface affected the contact characteristics of gears.

During the initial gear meshing, the small bumps on the bear torque may lead to stress concentration at small bumps, thus causing micropits on the surface. Ariura et al. [2] proved that surface pits caused surface deterioration. The tooth surface needs to be smooth to avoid micropits so as to obtain better meshing characteristics. Soberg et al. [3] and Andersson [4] examined the contact area of hobbing and shaving gears after the initial running-in; they showed that the roughness could be reduced after the initial running-in. The hobbing surface roughness was first larger, and then reduced significantly.

Thus, proper running-in can change the surface morphology of the gear and improved the meshing characteristics. Furthermore, plastic deformation and wear caused surface micro-modification, especially in rolling/sliding contact [5]. Using a partial factorial design method, Oila and Bull [6] and Li and Kahraman [7] found pitting resistance was improved by reducing the surface roughness. Another study [2] confirmed that the coarse gear surface was more prone to micro-pitting.

Bergseth et al. [8] found that the gear contact area ratio during honing and pure hobbling was higher than that during grinding gears. Some studies [9, 10] also showed that the ultra-finishing gear had lower surface roughness and higher gear contact efficiency compared with the grinding gear. In addition, other published studies demonstrated that the initial surface morphology had a significant influence...
on the fatigue contact characteristics of the gear surface [11]. Therefore, if the surface properties of the gear can be improved during the final machining, the fatigue life and wear resistance of the gear may be positively affected.

Ultrasonic and traditional machining are combined to form a variety of machining methods, such as ultrasonic-assisted lapping [12], ultrasonic-assisted drilling [13–15], ultrasonic forming grinding and so on. However, they mainly focused on the processing method of material removal, but rarely on ultrasonic strengthening on the tooth surface. Simultaneously, ultrasonic rolling technology can improve the surface residual stress [16] and hardness [17], reduce the surface roughness, refine the grain [18, 19]. Despite its many advantages, studies on ultrasonic rolling [20, 21] could not be directly applied to the strengthening on the gear surface. At the same time, previous studies showed [22–27] that the initial running-in of the gear, and appropriate load and speed improved the surface integrity and affected the contact performance of the gear. Considering the excellent characteristics of ultrasonic rolling and the positive influence of initial running-in on gear surface performance, a set of rolling-enhanced acoustic system (REAS) for the gear tooth surface was designed in this study.

2 Strengthening mechanism of the gear tooth surface by ultrasonic rolling

The schematic diagram of the rolling-enhanced acoustic system for the gear tooth surface is shown in Fig. 1. The working principle is as follows. First, the hardness between the tool gear (TG) and the workpiece gear (WG) is different, and the hardness of TG is about 30HRC higher than that of WG. Second, ultrasonic vibration is applied on the TG. Specifically, the TG is installed on the longitudinal vibration horn, whose back end is connected with a high-power transducer. The design frequency is set as 20 kHz and a large-diameter chip design is chosen to manufacture the transducer. The TG can vibrate repeatedly along the axial direction under the drive of the transformer. The WG is mounted on dampers, which provides damping torques to the TG so that the pressure can be generated on tooth surfaces. The ultrasonic horn system (UHS) is installed on the spindle, and the spindle has a cavity to accommodate UHS. The servo synchronous motor is installed on the back end of the spindle to provide power for UHS. The TG and WG are rolled and meshed with each other, so that the ultrasonic rolling enhancement can be completed.

3 Theoretical calculation of UHS

3.1 The theoretical calculation

The key components of UHS are a transducer and a double-node horn. The transducers sold by merchants cannot meet the requirements of this system because of space, size, power, and other reasons. Therefore, the system in this study was a self-developed transducer and horn. The conventional horn \( h/4, h/2, 3h/4 \) cannot meet the requirements because the normal force on the tooth surface of the ultrasonic meshing roller is applied to the horn. Scholars [28, 29] have also designed various horns for specific application requirements. In this paper, the horn with two nodes was selected to increase the stiffness of the system, and the horn was designed and calculated with one wavelength. The theory in the transducer field is developed; there the calculation was ignored in this study.

\[ 1 - \lambda \text{ Multi-segment double-node horn (DNMS) is complex, as shown in Fig. 2a. According to the processing conditions, the horn must have two nodes, and one node should be located in the cylindrical section. To achieve this goal, first, the segment estimation method (SEM) was adopted to simplify the horn. The simplified horn is shown in Fig. 2b.} \]

Therefore, the following method was used for the SEM. Part I was the cylinder, and part II was the large cone with a small cylinder, which required the largest displacement (i.e., the force was 0) at the joint. The two parts were designed according to half wavelength. According to the propagation formula in the cylindrical rod, it was seen that: \( c = \sqrt{E/\rho} \); \( \lambda = c/f \); The horn was designed with 40Cr, and the fre-
quency was 20 kHz. $E = 2.11 \times 10^{11} \text{Pa}; \rho = 7850 \text{kg/m}^3$; $C = 5170 \text{m/s}$; $\frac{1}{2} \lambda$ is $L_1 = \frac{1}{2} \times \frac{5170 \times 10^3}{20 \times 10^3} = 130 \text{mm}$.

The resonance frequency equation of $1/2 \lambda$ cylinder with the conical horn was as follows:

$$\tan(kl_2) = \frac{Na}{k} - \tan(kl_2 + a_2)$$  \hspace{1cm} (1)

where $a_2 = \arctan(\frac{k}{N}) = \frac{D_1}{D_2}$; The input diameter of the transducer was 50 mm. Considering the wave transmission characteristics, the diameter of the horn $D_1 = 52, D_2 = 36$. If $1/2 \lambda$ design calculation was used, then $L_1 = 60$ was calculated using Eq. (1).

After the simplified size of the horn was estimated, the analytical method was used for accurate calculation. Neglecting the effects of shear deformation and moment of inertia, the displacement and strain equations of I, III, IV, and V sections in Fig. 2 were as follows:

$$\begin{align*}
\varepsilon_i &= A_i \cos(kx) + B_i \sin(kx); i = 1;3;4;5 \\
\varepsilon'_i &= -A_i k \sin(kx) + B_i k \cos(kx)
\end{align*}$$  \hspace{1cm} (2)

According to the theory of longitudinal vibration, the displacement and strain equations of the conical rod (Sect. 2) were obtained as follows:

$$\begin{align*}
\varepsilon_2 &= \frac{1}{x^2} \left( A_2 \cos(kx) + B_2 \sin(kx) \right) \\
\varepsilon'_2 &= \frac{\partial}{\partial x} \left( \frac{1}{x^2} \left( -A_2 \sin(kx) + B_2 k \cos(kx) \right) \right) \\
&= -\frac{1}{(x^2)^2} \times \left( A_2 \cos(kx) + B_2 \sin(kx) \right)
\end{align*}$$  \hspace{1cm} (3)

where $a = D_1 - D_2/D_1 l_2$, $k = \omega/\omega_1$ is the number of longitudinal wave, and $D_1$ and $D_2$ are the major and minor diameters of the cone, respectively. $A_1, A_2, A_3, A_4, A_5, A_6, A_7, A_8$ is the undetermined coefficient related to boundary and connection conditions. To satisfy the free boundary condition, the two ends of the multi-section horn were free:

$$\begin{align*}
E \cdot (\varepsilon'_i)_{x=l_1} &= 0 \\
E_5 \cdot (\varepsilon'_5)_{x=l_1+l_3+l_4+l_5} &= 0
\end{align*}$$  \hspace{1cm} (4)

If the displacement and force were continuous at the contact surface, the following conditions should be satisfied:

$$\begin{align*}
(\varepsilon_1)_{x=0} &= (\varepsilon_2)_{x=0} \\
E_1 \cdot (\varepsilon'_1)_{x=0} &= E_2 \cdot (\varepsilon'_2)_{x=0} \\
(\varepsilon_2)_{x=l_1} &= (\varepsilon_3)_{x=l_2} \\
E_2 \cdot (\varepsilon'_2)_{x=l_1} &= E_3 \cdot (\varepsilon'_3)_{x=l_2} \\
(\varepsilon_3)_{x=l_1+l_3} &= (\varepsilon_4)_{x=l_1+l_4} \\
E_3 \cdot (\varepsilon'_3)_{x=l_1+l_3} &= E_4 \cdot (\varepsilon'_4)_{x=l_1+l_4} \\
(\varepsilon_4)_{x=l_1+l_3+l_4} &= (\varepsilon_5)_{x=l_1+l_3+l_4} \\
E_4 \cdot (\varepsilon'_4)_{x=l_1+l_3+l_4} &= E_5 \cdot (\varepsilon'_5)_{x=l_1+l_3+l_4}
\end{align*}$$  \hspace{1cm} (5)

By introducing displacement $\varepsilon_i$ and strain $\varepsilon'_i$ into the aforementioned boundary conditions, Eq. (6) was obtained.

$$\begin{align*}
\begin{bmatrix}
11 & c_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{21} & c_{22} & c_{23} & c_{24} & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{31} & c_{32} & c_{33} & c_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & c_{43} & c_{44} & c_{45} & c_{46} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & c_{53} & c_{54} & c_{55} & c_{56} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & c_{65} & c_{66} & c_{67} & c_{68} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & c_{75} & c_{76} & c_{77} & c_{78} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & c_{87} & c_{88} & c_{89} & c_{90} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & c_{1009} & c_{1010} & 0
\end{bmatrix}
\begin{bmatrix}
A_1 \\
B_1 \\
A_2 \\
B_2 \\
A_3 \\
B_3 \\
A_4 \\
B_4 \\
A_5 \\
B_5
\end{bmatrix} &= 0
\end{align*}$$  \hspace{1cm} (6)

Equation (6) is a system of homogeneous equations of order 10 with $l_1, l_2, l_3, l_4, l_5$ and frequency as parameters and $A_1, B_1, A_2, B_2, A_3, B_3, A_4, B_4, A_5, B_5$ as unknowns.

where $c_{ij}(k = 1, 2, \ldots, 10)$ is the coefficient in Eqs. (4)–(5).

In order for homogeneous linear equation group has nonzero solutions, the determinant of the matrix of coefficients should be zero. This leads to the following frequency equation:

$$\Delta = \begin{bmatrix}
c_{11} & c_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{21} & c_{22} & c_{23} & c_{24} & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{31} & c_{32} & c_{33} & c_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & c_{43} & c_{44} & c_{45} & c_{46} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & c_{53} & c_{54} & c_{55} & c_{56} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & c_{65} & c_{66} & c_{67} & c_{68} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & c_{75} & c_{76} & c_{77} & c_{78} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & c_{87} & c_{88} & c_{89} & c_{90} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & c_{1009} & c_{1010} & 0
\end{bmatrix} &= 0$$  \hspace{1cm} (7)

For the given resonant frequency and sizes of the horn, the geometrical sizes of the horn can be calculated.

Excitation of a simple harmonic oscillation displacement at the input end surface O is

$$\varepsilon_{\text{inc}} = \varepsilon_{\text{inc}}^0 = (A_1 \cos k_1 l_1 + B_1 \sin k_1 l_1)e^{jwt}$$  \hspace{1cm} (8)

Combining Eq. (8) with Eqs. (4) to (5) yields:

$$\begin{align*}
\begin{bmatrix}
c_{11} & c_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{21} & c_{22} & c_{23} & c_{24} & 0 & 0 & 0 & 0 & 0 & 0 \\
c_{31} & c_{32} & c_{33} & c_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & c_{43} & c_{44} & c_{45} & c_{46} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & c_{53} & c_{54} & c_{55} & c_{56} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & c_{65} & c_{66} & c_{67} & c_{68} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & c_{75} & c_{76} & c_{77} & c_{78} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & c_{87} & c_{88} & c_{89} & c_{90} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & c_{1009} & c_{1010} & 0
\end{bmatrix}
\begin{bmatrix}
A_1 \\
B_1 \\
A_2 \\
B_2 \\
A_3 \\
B_3 \\
A_4 \\
B_4 \\
A_5 \\
B_5
\end{bmatrix} &= 0
\end{align*}$$  \hspace{1cm} (9)
Input the given geometrical sizes and the resonant frequency from Eq. (7) to solve for \( A_1, B_1, A_3, B_3, A_5, B_5 \) by expression of \( \varepsilon \) at \( x = l_1 \); according to Eq. (9), the displacement amplification factor can be calculated using:

\[
M = \left| \frac{\varepsilon|_{x=l_1}}{\varepsilon|_{x=l_1}} \right| = \left| \frac{A_2 \cos k_3 l_5 + B_3 \sin k_3 l_5}{A_1 \cos k_1 l_1 + B_1 \sin k_1 l_1} \right|
\]

(10)

The coefficient \( A_1, B_1, A_3, B_3 \) is known; The displacement equations of segment 1 and segment 3 can be known, when \( x_1 = 0; A_1 \cos k_1 x_1 + B_1 \sin k_1 x_1 = 0, x_2 = 0 \), \( A_2 \cos k_2 x_2 + B_2 \sin k_2 x_2 = 0 \); then \( x_1^1, x_2^1 \) are the double nodes of the horn.

### 3.2 Theoretical design

The length value of segmented estimation is \( L_1 = 130 \), \( L_2 = 60 \), \( L_3 = 60 \). If \( L_3 \) was evenly divided into three segments of \( l_1, l_2 \), and \( l_3 \), and each segment was 20, \( l_1 = l_2 = l_3 = 20 \). When the frequency was 20 kHz, the values of \( l_1 \) and \( l_3 \) were adjusted so that Eq. (7) was applicable. A computer program was written. In put the given sizes and set a loop to solve the resonant length according to Eq. (7), then \( l_1 = 120, l_3 = 31 \) were obtained (as shown in Fig. 3), similarly, the values of \( l_3 = 31 \) and \( l_5 = 9 \) can be obtained. The final dimensions of the horn were \( l_1 = 120, l_2 = 50, l_3 = 31, l_4 = 20, l_5 = 9 \).

If the length and frequency were included in Eq. (7), \( A_1, B_1, A_2, B_2, A_3, B_3, A_4, B_4, A_5, B_5 \) were obtained. Another program was written to get the amplification factor: \( M = 1.5 \). The third program was written to find double nodes, and the specific results are shown in Fig. 4. If the origin of coordinates is the big end of the whole horn, then coordinates of two nodes are: \( (56,0), (184,0) \).

![Fig. 4 Node near the big end and node near the small end](image)

### 4 Simulation result and analysis of UHS

In real applications, the question is how the replacement of the TG affects the performance of the UHS. The theoretical calculation is too complicated and not intuitive. In this section, the modal analysis and harmonic response module were used to study the characteristics of the UHS. The influence of the TG (diameter and thickness) on frequency and amplitude for UHS was studied. The theoretical calculations in Sect. 2 are also verified.

#### 4.1 Influence of gear diameter on the frequency/node span

The length dimension \( l_1 = 120, l_2 = 50, l_3 = 31, l_4 = 20 \), \( l_4 = 9 \) remained unchanged, while the gear diameter was changed from 40 to 70 mm (modulus was 2, \( z = 20/25/30/35/40 \)). The mode of longitudinal vibration (near 20 kHz) was found in ANSYS. The values in Table 1 were obtained under the longitudinal mode. \( f_a, N_a, S_a \) and \( f_e, N_e, S_e \) are the frequencies, nodes, and span (the distance between two nodes) arrived at using the analytical method described above and using the finite element method, respectively. The errors between \( f_a \) and \( f_e \), \( N_a \) (node coordinate) and \( N_e, S_a \) and \( S_e \) were \( \Delta_1 = |(f_a - f_e)/f_a| \), \( \Delta_2 = |(N_a - N_e)/N_e| \), and \( \Delta_3 = |(S_a - S_e)/S_a| \). The following conclusions can be drawn: As the diameter (mass) increased, the frequency of UHS decreased. The node position of the horn moved to the end of the gear with the continuous increase in the gear mass, and the span between the two node positions also increased with the increase in the gear mass.
TG was replaced, the mass of the whole gear increased as the number of teeth increased. To maintain the balance, the system needed to increase the span, and the double nodes moved toward the direction of mass increase.

The error between the two-node simulation value and the theoretical value is shown in Table 1. As the diameter of node 1 increases, the position of node 1 gradually gets closer to the small end, so the error becomes larger and larger, while the error of node 2 gradually decreases with the increase in diameter. The span error is stable and the span of simulation calculation is smaller than that of theoretical calculation. This is because only the influence of length is considered in the theoretical calculation, and the influence of gear diameter is not considered.

### 4.2 Influence of gear thickness on the frequency/node span

The size of the horn, \( l_1 = 120, l_2 = 50, l_3 = 31, l_4 = 20 \), \( l_5 = 9 \), remained unchanged, and the diameter \( D_1 \) was 60 mm \((m=2, z=30)\). The thickness of the gear ranged from 5 to 25 mm. The longitudinal vibration mode (near 20 kHz) was found in ANSYS, and the values in Table 2 could be obtained under the longitudinal vibration mode.

As shown in Table 2, the diameter was unchanged, but its mass continued to increase with the increase in thickness; also, the frequency tended to decrease, but the change was small. When the thickness was 5 mm, the frequency changed the most because the ratio of thickness to diameter was 0.083; also, the vibration of the gear was thin-plate vibration, which was a little bit different from the longitudinal mode. The vibration frequencies of other thicknesses varied very little. When the gear thickness was changed, the frequency and node position of the UHS changed little. Therefore, the change in thickness in a small range had little effect on the frequency and node position of the UHS.

**Table 1** Influence on node position and frequency

| Diameter (mm) | Mass (kg) | Frequency (kHz) | \( \Delta_1\% \) | Node 1 (mm) | Node 2 (mm) | \( \Delta_2\% \) For Node 1 | \( \Delta_2\% \) For Node 2 | Span (mm) | \( \Delta_3\% \) |
|--------------|-----------|-----------------|----------------|-------------|-------------|-----------------------------|-----------------------------|-----------|----------------|
| 40           | 0.101     | 22.336          | 11             | 56          | 170         | 1.7                        | 7.6                         | 114       | 10             |
| 50           | 0.22      | 21.512          | 7              | 59          | 173         | 3.5                        | 5.9                         | 114       | 10             |
| 60           | 0.35      | 20.268          | 1              | 64          | 176         | 11.2                       | 4.3                         | 112       | 12.5           |
| 70           | 0.52      | 19.025          | 4              | 68          | 178         | 19.2                       | 3.2                         | 110       | 14             |
| 80           | 0.65      | 18.012          | 9              | 70          | 184         | 22.8                       | 0                           | 114       | 10             |

**Table 2** Influence of gear thickness on node and frequency

| Thickness (mm) | Mass (kg) | Frequency (kHz) | Node 1 (mm) | Node 2 (mm) | \( \Delta_1\% \) For Node 1 | \( \Delta_2\% \) For Node 2 | Span (mm) | \( \Delta_3\% \) |
|----------------|-----------|-----------------|-------------|-------------|-----------------------------|-----------------------------|-----------|----------------|
| 5              | 0.088     | 23.301          | 64          | 179         | 12                         | 2.7                         | 100       | 21%            |
| 10             | 0.17      | 20.842          | 64          | 174         | 12                         | 5.4                         | 109       | 14%            |
| 15             | 0.26      | 20.676          | 64          | 179         | 12                         | 2.7                         | 110       | 14%            |
| 20             | 0.35      | 19.804          | 64          | 178         | 12                         | 3.2                         | 114       | 10%            |
| 25             | 0.44      | 19              | 64          | 181         | 12                         | 1.6                         | 117       | 8%             |

The horn was modeled according to the dimension calculated in Sect. 1. The thickness of the gear was 20 mm, and the diameter was 40/50/60 mm (the number of teeth was 20/25/30) for modeling. After the modeling was in Creo, the model was imported into the harmonic response module in ANSYS. Then, the effect of diameter variation on amplitude was studied.

In ANSYS, the displacement excitation applied was the same, and the magnitude was 4 \( \mu \text{m} \). Two flanges were set at the nodes. The constraint position was the same, with two flanges. The ultrasonic horn with different gears was used in the harmonic response. The amplitudes of the same system at different frequencies are shown in Fig. 6a–d. In a harmonic response, under the longitudinal vibration condition, the relationship between frequency and amplitude was recorded for the same system (Fig. 6a–c). When the frequency of longitudinal vibration was found at first, the amplitude was relatively large. The amplitude gradually decreased as the frequency increased. For the same system, the span of the system decreased with the decrease in vibration, but not much. Under longitudinal...
vibration, the positions of the two nodes had something in common: the position of the node close to the gear basically remained unchanged with the increase in vibration frequency, while the position of the node close to the transducer changed the span dimension with the strength of the amplitude.

When the vibration was strong, the span was large, and vice versa. As shown in Fig. 6, the amplitude of the horn without gear (HWG) was the largest; it was 0.019 mm. The HWG was compared with the system with gear (SWG). The node position of the HWG was closer to the end of the transducer than that of the SWG. Meanwhile, a high amplitude existed between the two nodes; this amplitude of the HWG was smaller than that of the SWG. This was because, with the addition of the gear, the mass of the whole system became larger, reducing the amplification effect. Consequently, the system wasted some vibration energy, thus weakening the amplification effect of the horn.

As shown in Fig. 6a–c, the three systems followed the same law: As the frequency increased, the amplitude decreased. The comparison among the systems of different gears showed that the same law in Sects. 3.1 and 3.2 was used. That is: When different gears were installed in the UHS, the frequency and amplitude decreased with the increase in the diameter of the gear.

Therefore, the node position and resonant frequency of the ultrasonic system were different with different gears installed. The resonant amplitudes of different gears at the same frequency were studied (4 cases: diameter 40/50/60/none). If the excitation displacement was the same, the UHS with different gears (diameter 40/50/60/none) was used for comparison at 21 kHz. Figure 7 shows that the amplitude of the system with no gear was the largest. It was also clear that after the installation of gears, the node position of the UHS moved toward the gear, and the span remained basically the same.

At the same frequency, the amplitude of the UHS (diameter 40/50/60) decreased as the diameter increased. The 40-mm-diameter gear had the largest amplitude because 21 kHz was close to the resonance point of the UHS with the 40-mm-diameter gear and away from the resonant point of the UHS with the 50/60-mm-diameter gear. It was meaningless to compare the amplitudes of different systems at the same frequency. However, the positions of double nodes of different systems were approximately the same at the same frequency, which provided a design basis for later manufacturing.

As shown in Sects. 4.1, 4.2 and 4.3, the increase in gear mass did not necessarily lead to the change in system frequency, which depended on the direction of mass increase. The increase in mass caused by diameter had a great influence on system frequency. Meanwhile, it was concluded that the increase in diameter had a major influence on nodes and frequency. The analysis in the aforementioned section indicated that the UHS could still resonate and the node position basically remained within a small range when changing different tool gears (with a small range of changes in the number of teeth), which provided a favorable basis for manufacturing.

5 Experiment on the vibration characteristics of UHS

The vibration effect of the horn was different with different materials; 40Cr and SKD11 were used to make the horn. The vibration performance of the two materials was tested on self-made 20-kHz transducers.

The parameters, such as natural frequency, dynamic resistance, and quality factor, were measured using a PV70A...
damping analyzer (Bandera Electronics co., Ltd). After the assembly of the horn (SKD11) and the self-made transducer, the resonant frequency was 20,910 Hz, the anti-resonant frequency was 22,419 Hz, the impedance was 11 Ω, and the quality factor was 2119. After the assembly of the horn (40Cr) and the transducer, the resonant frequency was 20,198 Hz, the anti-resonant frequency was 20,215 Hz, the impedance was 24 Ω, and the quality factor was 1039.

As shown in Fig. 8, the admittance circle curves of the two materials were normal, and the dynamic resistance value was small. First, compared with the design frequency of 20 kHz, the error (SKD11 and 40Cr) was 4.5% and 0.93%, respectively, which verified the correctness of the theoretical calculation. Second, the vibration transmission efficiency of the SKD11 horn was better. The impedance was only 11 Ω, and the quality factor was 103% higher than that of the 40Cr material. This was mainly caused due to SKD11's excellent characteristics, such as high toughness, high hardness, and compact structure [30, 31]. Therefore, the subsequent tests in this study adopted the horn made by SKD11.

As shown in Tables 1 and 2, two points had the smallest displacement, namely two nodes. However, flange thickness frequency was 22,419 Hz, the impedance was 11 Ω, and the quality factor was 2119. After the assembly of the horn (40Cr) and the transducer, the resonant frequency was 20,198 Hz, the anti-resonant frequency was 20,215 Hz, the impedance was 24 Ω, and the quality factor was 1039.

As shown in Fig. 8, the admittance circle curves of the two materials were normal, and the dynamic resistance value was small. First, compared with the design frequency of 20 kHz, the error (SKD11 and 40Cr) was 4.5% and 0.93%, respectively, which verified the correctness of the theoretical calculation. Second, the vibration transmission efficiency of the SKD11 horn was better. The impedance was only 11 Ω, and the quality factor was 103% higher than that of the 40Cr material. This was mainly caused due to SKD11’s excellent characteristics, such as high toughness, high hardness, and compact structure [30, 31]. Therefore, the subsequent tests in this study adopted the horn made by SKD11.

As shown in Tables 1 and 2, two points had the smallest displacement, namely two nodes. However, flange thickness
was required for design and manufacturing. The flange thickness was 8 mm. The thickness centerline of Flange I was 67 mm from the large end (flange span 63–70 mm). The thickness centerline of Flange II was 180 mm from the large end (flange span 176–184 mm). According to the FEA, gears with the diameter of 60/70/80 mm could be installed on the horn. \(f_m\) is the measured frequency. The errors between \(f_a\) and \(f_m\) was \(\Delta_4 = \frac{|f_a - f_m|}{f_m}\). The ultrasonic horn was assembled using a transducer, a horn, and a gear, as shown in Fig. 9. The influence of different gears on system frequency was examined with the impedance analyzer. The frequency of the system was reduced when the horn was added to the gear (Table 3), which was the same as the simulation results in Sect. 3. Although the frequency decreased, it still met the requirements. At the same current, the amplitude of the single horn was obviously larger than that of the horn with the gear. After installing the gear, energy was lost; therefore, the amplitude of the gear was generally smaller. When the diameter increased, the quality factor/frequency decreased, and the resistance increased. This suggested that the larger the gear, the weaker the vibration transmission effect. The experimental test frequency had a similar trend to simulation. With the increase in gear diameter, the frequency of the system gradually reduced, showing the effectiveness of finite element simulation.

One-way excitation was applied to the system, and the longitudinal vibration was generated on the system. The vibration characteristics were measured using the LK-G10 (Keyence, Japan) laser displacement sensor. Figure 10 shows that the largest amplitude of SKD11 horn (without gear) was 5.3 \(\mu m\), the minimum amplitude was -4.8 \(\mu m\), and the current was 0.8 A. A TG with a diameter of 60 mm was installed on the horn, and then the frequency of 20909 Hz was applied. The maximum amplitude of the system was 4.3 \(\mu m\), the minimum was -4.1 \(\mu m\), and the current was 0.8 A. This indicated that the amplitude of the former was large, and the amplitude decreased with the installation of the gear. The amplitude was adjusted by adjusting the current. Little
difference was observed in the system amplitude of different gears, indicating that the four gears could achieve good longitudinal vibration effects under the same horn.

6 Experimental and discussion on REAS

The ultrasonic rolling tooth surface was used on the self-developed experimental device (Fig. 11). The permanent magnet motor was used to provide a torque for TG (parameters: rated power 4.8 kW, rated speed 1000 R/min, rated current 9.2 A), and the brake (rated current 3.5 A, power 84 W, rated torque 400 Nm) provided a damping torque for the WG. The TG was subjected to ultrasonic vibration along the axial direction, and a damping torque was observed on the WG. The TG made the WG rotate, and then the tooth surface was strengthened.

The TG material was 20CrMnTi, the surface was carburized and quenched, and the hardness was 58–62HRC. The WG material was 12Cr2Ni4A; it was quenched and tempered, and the hardness was 26–30HRC (No special instructions are in this state). TG parameters were as follows: \( m = 2, z = 30 \). WG parameters were as follows: \( m = 2 \), and the number of teeth 30. The experimental parameters are shown in Table 4.

The following were described from two aspects: system characteristics and tooth surface properties after ultrasonic rolling.

6.1 System excellent characteristics

Tonshoff and Marzenell pointed out [32] that honing achieved a larger axial and contour direction of compressive residual stress because honing applied a larger mechanical load at a lower temperature, and grinding produced larger grinding heat. The ultrasonic rolling process also had excellent characteristics of honing. The temperature of the ultrasonic rolling process was measured with an infrared thermometer FLIR-T540 (FLIR, USA). The maximum temperature in the rolling process was not more than 40 °C, as shown in Fig. 12, which was very beneficial to the final processing of the tooth surface.

6.2 The result after ultrasonic rolling

After ultrasonic rolling, the gear teeth were divided into blocks using wire-electrode cutting to observe the state of the surface residual stress and surface topography, in the following tests, all measured positions are near the gear pitch circle. A digital Rockwell hardness tester (HRS-150) was used to measure the microhardness. The surface residual stress was measured by a PROTO-LXRD residual stress analyzer, with a Cr radiation target, a tube voltage of 30 kV, and current of 25 mA. A German-made scanning electron microscope (Merlin Compact) was used for morphological observation.

1. Residual stress

When the damping torque is 80Nm, the rolling time is 2–8 h and the residual stress on the tooth surface is shown in Fig. 13. It can be seen that the residual stress increases at first and then decreases with the accumulation of time. The maximum residual stress is 515.5 MPa after 6 h of rolling, which is 4.1 times higher than the initial residual stress.
-100 MPa, then, with the increase in time accumulation, the rolling pressure decreased to -366 MPa in 8 h. The results show that when rolling time is 6 h, the grain refinement degree is the highest, and the amount of deformed martensite is the largest, the initial residual stress is -100 MPa, the maximum residual stress can be increased by 415%. After rolling, the amount of deformed martensite decreases, grain defects and martensite damage result in the reduction of residual stress. This is similar to the conclusion in literature [33].

2. Hardness

In order to analyze the influence of time on the hardness of the tooth surface in the ultrasonic rolling method, the damping torque is selected as 30Nm, and the gears of 12Cr2Ni4A with and without tempered are tested separately. The hardness of the tooth surface is tested every 10 min. As can be seen from Fig. 14, the hardness increases monotonously during the initial rolling process and periodically with the increase in time. The material has a cycle softening and hardening. As the quenched and tempered steel (QTS) is relatively soft, its hardness range is larger than that of the unquenched and tempered steel (UTS), the initial hardness of UTS is higher, and the hardness change amplitude is smaller, however, it can be seen that the material is cyclic softening and hardening, and the hardness increases with time, which indicates that the increase in surface hardness takes time to accumulate. During the ultrasonic rolling process, some dislocations will cancel each other with the dipolar dislocations of the different symbols formed in the double sliding, or disappear by other ways, thus leading to softening, the lattice distortion in the softening zone restores and the sub-grains grow up, which leads to the decrease in hardness. The ultrasonic rolling method will lead to the increase in dislocation value again, and the grain refinement hardness will increase again, this results in cyclic softening and hardening, the hardness increases about 53%(QTS), 15% (UTS), respectively, compared with the initial value.

3. Surface quality

In order to study the influence of the damping torque on the surface topography, the area near the gear pitch circle was selected as the research object, and the surface topography under different torques was compared and studied. As shown in Fig. 15, the untreated gear surface has obvious peaks and valleys along the axial direction (gear width direction), which are characterized by grinding marks. This kind of peak and valley contributes a lot to the local stress concentration, so it is not conducive to the fatigue performance of the gear. After processing, for the 30Nm sample, the mechanical grinding marks along the axial direction are reduced. This smooth surface is caused by the effect of ultrasonic rolling. Under ultrasonic rolling, the material with high peaks on the tooth surface flows to the valleys. For the

Table 4 Processing experimental parameters

| Speed (r/min) | Frequency (kHz) | Amplitude μm | Torque (Nm) |
|--------------|----------------|--------------|-------------|
| 30           | 19.98          | 4.1 – 4.3    | 30 – 90     |

Fig. 11 Self-developed experimental device

Fig. 12 Temperature in process

Fig. 13 Effect of time on residual stress

Table 4

| Speed (r/min) | Frequency (kHz) | Amplitude μm | Torque (Nm) |
|--------------|----------------|--------------|-------------|
| 30           | 19.98          | 4.1 – 4.3    | 30 – 90     |
60Nm specimen, the distance between peaks and valleys is enlarged, and the peaks and valleys of the 80Nm specimens almost disappear. Larger plastic deformation occurs on the surface of the sample, the surface after treatment is more uniform, and the surface height difference is gradually reduced. It can be seen from the figure that the increasing damping torque leads to the increase in the contact pressure. With the increase in the contact pressure, the texture of the marks on the initial grinding surface is continuously changed, and the distance between the marks is gradually widened. When the traces of the processed ravines are processed, the surface morphology distribution is more smooth, and there are no obvious crack defects, and the texture is fine (Fig. 16).

The surface roughness (Ra and Rq) images of the untreated and treated samples are shown in. It can be noticed that there are obvious wear scars on the surface of the untreated sample, and the outline shows large peaks and valleys, resulting in relatively high Ra and Rq, which are 0.377 and 0.467, respectively. For the 30Nm sample, the grinding marks are basically removed, but the peak-to-valley surface is still larger than the distance between the grinding marks, which causes Ra and Rq to decrease by 0.114 and 0.132, respectively. When the torque is 60Nm, the surface profile is all small peaks and valleys, with large peaks and valleys interspersed in the middle, and the roughness is reduced to Ra of 0.111 and Rq of 0.134. Compared with 30Nm and 60Nm, the grinding marks of 80Nm specimens are almost removed, and the peaks and
valleys are no longer visible on the surface contour lines, and the roughness is reduced to Ra0.09 and Rq0.1; after treatment, the surface roughness Ra is decreased, respectively, 30%, 70%, 76%. After the treatment, the surface appears smooth as a whole, the roughness profile peaks fill the valleys, and the peaks and valleys are changed. The defect of stress concentration after grinding is removed, and the fatigue strength of the material can be improved.

To sum up, ultrasonic rolling can improve the roughness and surface appearance of the gear tooth, the main reasons are as follows: the displacement of the contact point strengthened by this mode is the longitudinal space sine curve, and the displacement of different nodes interweave with each other, which makes the peak-valley width of the surface become wider and the residual material height of the surface reduced after rolling and sliding repeatedly, this process reduces surface roughness and improves surface quality.

7 Conclusions

This paper presents a method of ultrasonic strengthening tooth surface. Firstly, DNMS is calculated theoretically. Then, the vibration characteristics of DNMS are demonstrated by finite element analysis. Finally, the corresponding experimental system is built and a series of experiments are carried out. By analyzing the theory, vibration characteristics, and experimental results, the following conclusions are obtained:

1. Combined with ultrasonic rolling strengthening technology and gear meshing principle, the rolling-enhanced acoustic system for the gear is proposed to strengthen gear surface, through theoretical calculation and experimental analysis, it is obtained that the gears with diameter 60/70/80 mm can be installed on the designed horn, which can ensure the minimum vibration of the flange position.

2. A multi-segment double-node compound horn system is designed to meet the requirements of REAS. The transducer is designed by half wavelength, and the horn is designed by one wavelength. The frequency equation of the double-node horn is established by the one-dimensional wave vibration theory, the correctness of which is proved by experiment and simulation.

3. By modal analysis and harmonic response analysis, it is concluded that influences of mass increase on the ultrasonic rolling system depended on the way of mass increase. The change of the thickness of gears with constant diameter has little effect on the system frequency. When the diameter increases with same thickness, the frequency of the URS become smaller, and the position of the double nodes is close to the gear end. The span difference in the double nodes to be varied in a small range. The amplitudes of different gear resonant systems (GRS) are different. The frequency of GRS decreases with the increase in diameter, and the amplitude of the horn (with no gear) is the largest. The amplitude was reduced after the gear was installed.

4. The self-built URES is used to strengthen the gears with the diameter of 60 mm. Some plastic deformation occurs in the tooth surface with ultrasonic rolling, and the surface texture changes. At the same time, the surface roughness reduces, and the surface brightness, tooth surface hardness, and residual stress increase. These findings indicate that the gear tooth surface strengthening treatment provided a new way of thinking.

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**Declarations**

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