System design and experimental research on the tangential ultrasonic vibration-assisted grinding gear

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Abstract
A tangential ultrasonic vibration-assisted gear grinding (TUVAGG) system was proposed in this paper to enhance the machining accuracy and performance of gears. Firstly, the longitudinal resonant vibration system of the TUVAGG was designed based on the non-resonant theory. The frequency equation and the displacement characteristics for the vibration system were obtained for the particular boundary conditions. Secondly, the vibration system composed of simplified disc and gear was simulated by the finite elements analysis method (FEM) and verified by the resonant measurement test. Finally, the vibration system effectiveness was verified through the ultrasonic vibration-assisted gear grinding test. The normal and tangential grinding forces in TUVAGG were lower by 7.4–28.2% and 8.9–18.9%, respectively, than conventional gear grinding (CGG). Besides, the grinding temperature and surface roughness in TUVAGG declined by 7.6–25.7% and 8.6–21.8% respectively compared with CGG, while the former tooth surface residual compressive stress and microhardness exceeded the latter ones by 13.2–29.3% and 8.9–12.7%. The non-resonant theory was suitable for the designation of longitudinal vibration system for TUVAGG, and also provided a novel process technology for gear machining.

Keywords Ultrasonic vibration-assisted gear grinding · Longitudinal vibration system · FEM · Residual compressive stress · Non-resonant theory

1 Introduction

Power- and motion-transferring gears are widely used in machinery, transport, robotics, aerospace, and many other fields [1–3]. Growing demand for high-precision gear dimensions and performance implies strict requirements for the accuracy of gear-manufacturing technologies. Compared with conventional machining, ultrasonic machining is known to reduce the cutting force [4], extend tool life [5], enhance machining accuracy [6], and improve workpiece surface morphology [7] and wear resistance [8] and corrosion resistance [9]. In particular, ultrasonic vibration has been successfully combined with conventional gear machining, giving rise to such innovative processing techniques as ultrasonic vibration-assisted gear hobbing, gear shaving [10], gear lapping [11], gear honing [12], and ultrasonic electrochemical gear machining [13]. According to the state-of-the-art relevant publications, ultrasonic vibration-assisted gear machining can improve the processing technology and manufacturing process and significantly enhance the surface integrity and fatigue performance [14]. However, to the best of the authors’ knowledge, no experimental and numerical studies on the effect of displacement amplitude and ultrasonic frequency on cutting forces and processing performance for the TUVAGG have been performed yet.

In the TUVAGG, the designation of longitudinal vibration system directly affects the machining effect. Wei et al. [15] designed the longitudinal vibration system for grinding the steel arc bevel gear with a 1.25-mm modulus and 17 teeth based on the four-terminal network method. They modified the size of the vibration system through the mass reciprocity method. However, the designation error increased with the gear diameter. When conducting the ultrasonic lapping test, Wu et al. [16] combined the tool and...
horn into the integrated design, in which the tool was simplified by a small round cone. With the help of the wave equation of variable cross-section along the longitudinal vibration, the continuity conditions of stress and velocity, and the boundary conditions, the frequency equation of the combination was established, and the relationship between the size variation of small bevel gear and the resonant frequency was derived. However, the designation error was higher due to ignoring section variation in the combination junction surface. Based on the longitudinal vibration wave equation of variable cross-section, Gong et al. [17] simplified the gear into a cylindrical rod. They deduced the frequency equation and the amplification coefficient calculation formula, which were complicated. Lü et al. [18] analyzed the horn and gear via the mass reciprocity method to deal with the fixed part of the gear and proposed the general theoretical model of the ultrasonic gear honing. It was found that the designed vibration system could achieve better performance.

In the case of the TUVAGG system, the processed gear should be treated as the active load. Besides, the gear structure is determined not by the resonant frequency but by its application requirements. In the practical application, because the ultrasonic generator restricts the frequency ranges of the ultrasonic vibration system, it is impossible to maintain the processed gear frequency within the required range accurately. However, it can be resonated in the frequency range by properly designing the vibration system composed of gear and horn. When designing the ultrasonic honing vibration system with a large load, Zhao [19] adopted the local resonance theory to design the system and made the system well-resonant by tuning the ultrasonic generator frequency. Subsequently, the non-resonant design theory was utilized to design the ultrasonic vibration system of gear honing. Lü et al. [20] found that it could simplify the design of resonant system, and avoid the complexity of the resonant design theory. In addition, the exciting frequency of the system was adjusted to eliminate the adverse effect of tool wear on the resonance system during the ultrasonic machining. However, the effect of gear dimensions on the vibrational characteristics (resonant frequency and ultrasonic amplitude) of the vibration system was not considered during the designation. Meanwhile, the ultrasonic vibration system for gear grinding has not been covered by available studies. Therefore, it is essential to investigate the vibration system for the TUVAGG.

To achieve the designation of the longitudinal vibration system for TUVAGG, the respective theoretical model is first developed on the basis of non-resonant theory. Then, the frequency equation and the displacement characteristics of the longitudinal vibration system are derived from the boundary conditions. The effect of gear dimensions on the vibrational characteristics is assessed by the FEM, and the system’s dynamic characteristics are numerically simulated to obtain the resonant frequency and ultrasonic amplitude. The resonant tests are also conducted to verify the theoretical and FEM-predicted results. Finally, the actual grinding test is performed to assess the developed vibration system’s effectiveness.

2 TUVAGG operation principle

The TUVAGG operation principle is illustrated by Fig. 1. The TUVAGG experimental setup mainly comprises a machine tool spindle, form-grinding wheel, and ultrasonic vibration system. The latter primarily consists of an ultrasonic generator, transducer, horn, and gear. The ultrasonic generator is used to convert the alternating current into high-frequency electric oscillation, while the transducer converts the current of ultrasonic frequency into mechanical vibration of the ultrasonic vibration system. However, the amplitude of mechanical vibration produced by the transducer is too low for mechanical machining. Therefore, the horn as a concentrator is added to amplify the ultrasonic vibrational amplitude to the applicable magnitude and drive the longitudinal gear vibration along the tangential direction of the grinding wheel. The form-grinding wheel is utilized for machining the gear. During the processing, the grinding wheel axis is normal to the gear axis, and the center of the grinding wheel truncation coincides with that of the ground tooth groove. The grinding wheel speed is \( v_w \), the radial grinding depth is \( a_r \), and the gear feed rate is \( v_{fu} \).

3 Model of the longitudinal resonance system

For the vibration system of TUVAGG, the gear dimensions strongly influence the vibrational characteristics, and the resonant design theory is not suitable for the designation of the vibration system, while the non-resonant theory is more...
appropriate. The vibration system composed of the gear and horn is firstly regarded as a whole, and the frequency equation is established through the continuous and boundary conditions. Then, the frequency is made resonant with the working frequency by adjusting the horn size. Finally, the resonant and natural (modal) frequencies are obtained to match the operation.

Generally, a transducer is regarded as the energy input end during the designation of the vibration system, and which design is not considered in this study. In order to acquire a higher amplification coefficient, a composite horn is designed. For facilitating the vibration system design, the following assumptions are made [21]:

1. The vibration system is simplified as a rigid structure, and the contact conditions between the horn, gear, and nut are ignored.
2. The gear is simplified as a cylinder of equal diameter to the reference circle, and the effect of gear shape on longitudinal vibration is neglected.

The vibration system contains four main components: transducer, horn, gear, and nut, as presented in Fig. 2.

The gear is installed on the small end of the horn and fixed with a nut, and the xoy coordinate system with the origin in the center of the horn’s large end is adopted. The vibration system is composed of five sections, namely cylindrical section I, conical section II, and cylindrical sections III, IV, and V, respectively, with corresponding lengths $l_1$, $l_2$, $l_3$, and $l_4$, and respective radii $R_1$, $R_2$, $R_3$, $R_4$, and $R_5$.

### 3.1 Model of the longitudinal vibration system

In the longitudinal resonance system, it is supposed that a homogeneous and isotropic material is utilized to fabricate the composite horn, and the mechanical loss and damping effects are neglected. According to the wave equation for the variable cross section [22], we get:

$$\frac{\partial^2 \xi_i}{\partial x_i^2} + \frac{1}{S_i} \frac{\partial}{\partial x_i} \frac{\partial \xi_i}{\partial x_i} + k_i^2 \xi_i = 0$$ (1)

where $i = 1, 2, 3, 4, 5$ correspond to segments I, II, III, IV, and V, respectively; $\xi_i = \xi_i(x_i)$ is the displacement function of each segment; $S_i = S_i(x_i)$ is the cross-sectional area function of the vibration system; $k_i$ is the circular wavenumber, $k_i = \omega/\xi_i$, $\omega$ is circular frequency, $c_i = \sqrt{E_i/\rho_i}$ is propagation velocity of longitudinal waves, $\rho_i$ is the material density, and $E_i$ is the Young modulus.

Consequently, for the cylindrical sections I, III, and V, the displacement and stress functions are derived as follows:

$$\begin{align*}
\xi_i(x_i) &= a_{i1} \cos k_i x_i + a_{i2} \sin k_i x_i \\
\frac{\partial \xi_i}{\partial x_i} &= -a_{i1} k_i \sin k_i x_i + a_{i2} k_i \cos k_i x_i
\end{align*}$$ (2)

where $i = 1, 3, 5$ denote segments I, III, and V, respectively.

For the conical section II, the displacement and stress functions can be expressed as:

$$\begin{align*}
\xi_2(x_2) &= \frac{1}{x_2-1/\alpha} (a_{21} \cos k_2 x_2 + a_{22} \sin k_2 x_2) \\
\frac{\partial \xi_2}{\partial x_2} &= \frac{1}{x_2-1/\alpha} (-a_{21} k_2 \sin k_2 x_2 + a_{22} k_2 \cos k_2 x_2) - \frac{1}{(x_2-1/\alpha)^2} (a_{21} \cos k_2 x_2 + a_{22} \sin k_2 x_2)
\end{align*}$$ (3)

where $\alpha = (R_1 - R_2)/R_1 l_2$.

For section IV, the displacement and stress functions can be written as:

$$\begin{align*}
\xi_4(x_4) &= \frac{1}{R_4 - R_3} (a_{41} \cos k_4 x_4 + a_{42} \sin k_4 x_4) \\
\frac{\partial \xi_4}{\partial x_4} &= \frac{1}{R_4 - R_3} (-a_{41} k_4 \sin k_4 x_4 + a_{42} k_4 \cos k_4 x_4)
\end{align*}$$ (4)

When the vibration system vibrates freely along the longitudinal direction, the boundary conditions and coupled conditions for the forces and displacements should be satisfied as follows:

$$\begin{align*}
\xi_1(x_1)|_{x_1 = l_1} &= \xi_2(x_2)|_{x_2 = l_1} \\
\xi_2(x_2)|_{x_2 = l_1 + l_2} &= \xi_3(x_3)|_{x_3 = l_1 + l_2} = \xi_4(x_4)|_{x_4 = l_1 + l_2} \\
\xi_3(x_3)|_{x_3 = l_1 + l_2 + l_3} &= \xi_4(x_4)|_{x_4 = l_1 + l_2 + l_3} = \xi_5(x_5)|_{x_5 = l_1 + l_2 + l_3}
\end{align*}$$ (5)
For the left and right ends of the vibration system, the following boundary conditions apply:

\[
\begin{align*}
\frac{\partial \xi_1(x_1)}{\partial x_1}|_{x_1=0} &= 0 \\
\frac{\partial \xi_5(x_5)}{\partial x_5}|_{x_5=l_1+l_2+l_3+l_4} &= 0
\end{align*}
\] (7)

The theoretical model of the vibration system can be obtained by combining Eqs. (5)–(7), which yields a complex transcendental equation.

\[
\Delta = \begin{vmatrix}
C_{11} & C_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
C_{21} & C_{22} & C_{23} & C_{24} & 0 & 0 & 0 & 0 & 0 & 0 \\
C_{31} & C_{32} & C_{33} & C_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & C_{43} & C_{44} & C_{45} & C_{46} & 0 & 0 & 0 & 0 \\
0 & 0 & C_{53} & C_{54} & 0 & 0 & C_{57} & C_{58} & 0 & 0 \\
0 & 0 & C_{63} & C_{64} & C_{65} & C_{66} & C_{67} & C_{68} & 0 & 0 \\
0 & 0 & 0 & 0 & C_{75} & C_{76} & C_{77} & C_{78} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & C_{89} & C_{90} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & C_{109} & C_{1010}
\end{vmatrix}
\] (8)

where \(C_{mn}\) are coefficients of \(a_{ij}\).

3.2 Frequency equation of the longitudinal vibration system

The homogeneous equations composed of coefficients of \(a_{ij}\) can be obtained by substituting Eqs. (2)–(4) into Eqs. (5)–(7), respectively. In order to ensure that the system will have non-zero solutions, the determinant of the system coefficients should be equal to 0. The frequency equation for the system has the following form:

\[
\xi_1(x_1)|_{x_1=0} = a_{11} \cos k_1 l_1 + a_{12} \sin k_1 l_1 = \xi_0
\] (9)

According to the functional structure, Eqs. (8) and (9) yield the statically indeterminate Eq. (10).

\[
\begin{pmatrix}
C_{11} & C_{12} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
C_{21} & C_{22} & C_{23} & C_{24} & 0 & 0 & 0 & 0 & 0 & 0 \\
C_{31} & C_{32} & C_{33} & C_{34} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & C_{43} & C_{44} & C_{45} & C_{46} & 0 & 0 & 0 & 0 \\
0 & 0 & C_{53} & C_{54} & 0 & 0 & C_{57} & C_{58} & 0 & 0 \\
0 & 0 & C_{63} & C_{64} & C_{65} & C_{66} & C_{67} & C_{68} & 0 & 0 \\
0 & 0 & 0 & 0 & C_{75} & C_{76} & C_{77} & C_{78} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & C_{89} & C_{90} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & C_{109} & C_{1010}
\end{pmatrix}^{-1}
\begin{pmatrix}
\xi_0 \\
a_{11} \\
a_{12} \\
a_{21} \\
a_{22} \\
a_{31} \\
a_{32} \\
a_{41} \\
a_{42} \\
a_{51} \\
a_{52}
\end{pmatrix}
= 0
\] (10)
A group of special solutions of $C_{mn}$ can be obtained from Eq. (10), and then substituted into the displacement function of each section, respectively. The longitudinal displacement curve along the $x$-direction can be acquired.

4 Design and dynamic analysis of the longitudinal resonance system

4.1 Gear size effect on the vibration system characteristics

It is well known that the length of the composite horn should be determined according to the load [23]. In order to assess the effect of gear dimensions on the resonant frequency and the horn length, the reference radius and its thickness are varied in the FEM analysis. As shown in Fig. 3, when the horn length and gear thickness are kept constant, the resonant frequency declines with the increase of the reference radius and gear thickness. As the ultrasonic frequency is kept at 20 kHz, the horn length decreases with the reference radius, and increases with the thickness.

It can be seen in Fig. 3 that the reference radius and gear thickness strongly affect the resonant frequency and the required horn length. Compared with the reference radius, the thickness has a slight effect on the frequency. If the thickness is increased and the reference radius is decreased, the resonant length $l$ of the horn increased and vice versa. Thus, $l$ is directly proportional to the gear thickness-to-diameter ratio. According to Lü et al. [20], if the reference circle diameter is lower than 100 mm, and the thickness-to-diameter ratio exceeds 0.3, the longitudinal resonant designation is suitable for the vibration system. As shown in Fig. 3, the resonant frequency deviation from the design frequency grows with increased reference radius and thickness. Simultaneously, taking the horn stiffness effect into consideration, the gear is selected with a modulus of 3 mm, a number of teeth of 21, and a thickness of 20 mm.

4.2 Design of the longitudinal vibration system

The longitudinal vibration system is composed of a disc, horn, and nut, which are made from different steel grades: 12Cr2Ni4A, 42CrMo, and 45# steel, respectively. According to Lin [22], the length of the cylindrical section of the composite horn $l_1$ should be smaller than $\lambda/4$. It is assumed that length $l_1$ is 60 mm, and the numerical method is used to solve the conical section length $l_2$. Taking the variation range of the length $l_2$ as 40–100 mm, the relationship between the value of the determinant of coefficients $\Delta$ and the conical section length can be obtained, as shown in Fig. 4. Therefore, the conical section length $l_2$ is 65.4 mm.

4.3 Dynamic characteristics of the longitudinal resonance system

It is assumed that the maximum displacement is gained at the output end of the transducer, while $\zeta_0$ 4 $\mu$m. A group of special solutions of $C_{mn}$ can be derived from Eq. (10), and substituted into the displacement function of each section, respectively. Then, the longitudinal displacement curve along the $x$-direction can be constructed, as shown in Fig. 5.

As seen in Fig. 5, the node position is located at a 58.5 mm distance from the horn input end, and the ultrasonic amplitude at the gear’s left and right ends are 7.25 and 7.36 $\mu$m, respectively.

4.4 Finite element analysis

To verify the accuracy of the above theoretical results, the vibration system composed of a simplified disc (named disc vibration system) and the vibration system composed of gear
named gear vibration system) were analyzed using the FEM method. The 3D models of the respective vibration system were elaborated using the software PROE and imported into the ANSYS17.0 commercial software. The displacement constraint of 4 μm was applied to the left end of the horn. The models of both vibration systems are shown in Fig. 6.

The resonant frequency values of the disc and gear vibration systems were 19.64 and 19.20 kHz, respectively, differing from the design frequency of 20 kHz by 1.8 and 4.0%.

The ultrasonic amplitude distribution curves of the vibration system and gear along the x-direction are plotted in Fig. 7.

As presented in Fig. 7, the ultrasonic amplitude at the disc’s left and right ends reached 6.50 and 6.76 μm, differing by 10.3 and 8.2%, respectively, from the theoretical results. The corresponding ultrasonic amplitudes at the gear’s left and right ends reached 6.62 and 6.93 μm, with the respective errors of 8.7 and 5.8%.

5 Performance test of the longitudinal vibration system

5.1 Impedance characteristic test

The components were processed according to the above parameters. In order to reduce energy losses, the bonding surfaces were ground and coated with Vaseline before assembly. The impedance analyzer PV70 (Beijing Band Era Co. Ltd., China) was utilized to analyze the impedance characteristics of both vibration systems. The measurement results are presented in Fig. 8.

The resonant frequencies of the disc and gear vibration systems were 20.75 and 20.46 kHz, respectively. The corresponding dynamic resistances of the above vibration systems were 11.14 Ω and 24.05 Ω. As shown in Fig. 8, the admittance circle was normal, which meant that the both vibration system achieved higher-quality factor.

5.2 Ultrasonic amplitude test

The laser displacement sensor (LK-G10, KEYENCE, Japan) was employed to measure the ultrasonic amplitudes in both systems. In order to improve measurement accuracy, seven measurement points were selected at the disc (gear) edges. During the disc vibration system measurement, the laser beam from the laser displacement sensor was focused on the disc output end. The ultrasonic amplitude test results on the disc vibration system are presented in Fig. 9.

When measuring the gear vibration system, the gear vibration system in Fig. 8b replaced the disc vibration system. In order to achieve reliable data, each test was conducted three times, and a final value was obtained by averaging the measured results. The measured results are presented in Fig. 10.
The results obtained for both vibration systems are presented in Table 1, where $f_M$ denotes the theoretical design frequency, $f_A$ and $f_E$ are FEM and resonant test frequencies, respectively; $f_{AM}$ is the deviation rate between the $f_A$ and $f_M$, namely $f_{AM} = |f_A - f_M|/f_M \times 100$; $f_{EM}$ is the deviation rate between the $f_E$ and $f_M$, namely $f_{EM} = |f_E - f_M|/f_M \times 100$; $A_M$ is the amplitude of the theoretical analysis, $A_A$ and $A_E$ are FEA and resonance test amplitudes, respectively; $A_{AM}$ represented the deviation rate between the $A_A$ and $A_M$, namely $A_{AM} = |A_A - A_M|/A_M \times 100$; $A_{EM}$ is the deviation rate between $A_E$ and $A_M$, namely $A_{EM} = |A_E - A_M|/A_M \times 100$.

It can be observed from Table 1 that there was a deviation between the measured and simulated results, which was mainly related to the material’s non-uniformity. In the FEM calculations, the material was assumed to be uniform; while in actual fabrication, the local material’s removal affected the experimental results. Since deviations $f_{AM}$, $f_{EM}$, $A_{AM}$, and $A_{EM}$ of both vibration systems were below 10%, the experimental results were close to the theoretical and FEM results.

### 6 Ultrasonic vibration-assisted gear grinding test

#### 6.1 Experimental setup and methodology

A modified CNC machine center (VMC850E, SMTCL, China) was used to perform the UVAFGG test. The basic experimental setup is presented in Fig. 11.

The above setup included the ultrasonic vibration system (which was earlier described and depicted in Fig. 1), the grinding force acquisition system, and grinding temperature acquisition system. The involute cylindrical gear was utilized to conduct the tests, and the gear vibration system was connected to the fixture through the flange...
with four bolts. The ultrasonic vibration system was also connected to a dynamometer (9257B, Kistler Instrument Corp., Switzerland) through a fixture. The dynamometer was fixed on the workbench with a clamp.

An amplifier (5070A) was used to amplify the electrical signal from the dynamometer and transmit it to a data collector (2825A). The Dyno Ware commercial software was employed to monitor and process the recorded data using a PC.

The thermocouple method was used to measure the grinding temperature. Two blind holes with a diameter of 1 mm were prefabricated in the reference circle at a distance of 0.2 mm from the gear surface. Two K-type thermocouples were placed in the two blind holes, respectively, and contacted at the end of the hole. The hole opening was filled with plasticine to fix the thermocouple. During the grinding test, the signals from the thermocouples were measured by a digital thermometer.

In the grinding test, the TUVAGG and CGG were achieved by switching on or off the ultrasonic generator. The respective parameters are listed in Table 2. Each group parameter was replicated three times, and the averaged result was taken as the final value. After the TUVAGG test, the machined gear teeth were cut using wire-electrode cutting to observe the state of the surface residual stress and surface topography. The surface residual stress was explored with the PROTO X-ray via the XRD method. A digital Rockwell hardness tester (HRS-150) was used to measure the microhardness. The white-light

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**Fig. 8** Impedance test of longitudinal resonance system: (a) disc vibration system, (b) gear vibration system

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**Fig. 9** The ultrasonic amplitude measurements of disc vibration system
interferometer (GT-K, BRUKER, USA) was utilized to measure the gear surface roughness. Three points near the tooth root, index circle, and tooth tip were selected to reduce the measurement error, respectively, and their average value was taken as the final result.

### 6.2 Results and discussion

#### 6.2.1 Grinding force

Grinding force is generally employed to evaluate the grinding performance. The relationship between the grinding force and grinding parameters is presented in Fig. 12.

It can be seen from Fig. 12 that the TUVAGG can significantly reduce the grinding force, as compared to the CGG: the normal and tangential grinding forces were reduced by 7.4–28.2% and 8.9–18.9%, respectively. Furthermore, in both cases, grinding forces grew with the radial grinding depth and dropped with an increase in spindle speed. This should be attributed to the fact that the grinding depth of single grain would increase with radial grinding depth. Moreover, the number of effective grains in the grinding zone would increase. However, the grinding arc length during the TUVAGG was increased due to a longer trajectory of single grain, which reduced its undeformed chip thickness. Moreover, the surface was very smooth, and was similar to attach a layer of oil film [24]. It could effectively reduce the friction as the grain cut into the workpiece. Therefore, the grinding force in TUVAGG was reduced by the compressive factors.

As shown in Fig. 12, the maximum reduction ratio of grinding force was obtained at the lower spindle speed. According to Ding et al. [25], when the spindle speed exceeded a critical speed, the overlapping and interaction of different grains’ trajectories were weakened. This would lead to the ultrasonic wavelength being closer to the single grain trajectory in the CGG. In other words, the benefit of ultrasonic vibration was suppressed in the high spindle speeds, so the grinding force reduction ratio was higher at low spindle speeds. This agreed well with the fact that the most effective CGG processing required high spindle speeds, while the TUVAGG processing should be conducted at low spindle speeds, which can also prolong the grinding wheel life.

#### 6.2.2 Grinding temperature

Grinding temperature is regarded as the key factor influencing the processed surface quality. Figure 13 presents the relationship between the grinding temperature and grinding parameters.

It can be observed in Fig. 13 that the grinding temperature in TUVAGG was reduced by 7.6–25.7%, as compared to CGG. If the radial grinding depth was kept constant, the grinding temperature enhanced with the spindle speed. This occurred because the total heat increased with the spindle speed, and the heat distribution ratio between the grinding wheel and gear also increased. Therefore, more heat flew to the gear and led to the temperature elevation. If the spindle speed was kept constant, an increase in radial grinding depth caused the grinding temperature to rise. This resulted from the elongation of the contact arc length between the grinding wheel and gear with an increase in radial grinding depth, which increased the number of grains involved in grinding. Meanwhile, an increase in undeformed chip thickness raised the energy consumption. Besides, the ultrasonic vibration could increase the grain-workpiece impact acceleration and produce grain’s staggering trajectories, which would achieve the material removal with less energy consumption. Besides, the intermittent grinding conducted between grains and workpiece was conducive to heat dissipation. Consequently, the grinding temperature in TUVAGG was relatively lower than that in CGG.

### Table 1 Test results of the vibration system

| Parameters          | $f_{AM}$ kHz | $f_{EM}$ kHz | $f_{AM}$ % | $f_{EM}$ % | $A_M$ μm | $A_A$ μm | $A_E$ μm | $A_{AM}$ % | $A_{EM}$ % |
|---------------------|--------------|--------------|------------|------------|----------|----------|----------|------------|------------|
| Disc vibration system | 20           | 19.64        | 20.75      | 1.8        | 3.75     | 7.36     | 6.76     | 6.67       | 8.20       | 9.38       |
| Gear vibration system | 20           | 19.20        | 20.64      | 4.0        | 2.30     | 7.36     | 6.93     | 6.84       | 5.80       | 7.07       |
Table 2  Experimental parameters

| Type                        | Parameters                      | Value                      |
|-----------------------------|---------------------------------|----------------------------|
| Grinding wheel              | CBN form-grinding wheel         |                            |
|                             | Diameter $D$ (mm)               | 80                         |
|                             | Grinding wheel granularity (#) | 400                        |
| Grinding parameters         | Radial grinding depth $a_r$ ($\mu$m) | 10, 20, 30, 40             |
|                             | Spindle speed ($r$/min)         | 1000, 2000, 3000, 4000    |
|                             | Workpiece feed rate (mm/min)    | 400                        |
|                             | Coolant                         | None                       |
| Gear                         | Material                        | 12Cr2Ni4A steel            |
| Ultrasonic parameters       | Ultrasonic frequency $f$ (kHz)  | 20.46                      |
|                             | Ultrasonic amplitude $A$ ($\mu$m) | 4.8                       |
| Methods                     |                                 | TUVAGG, CGG                |

Fig. 11  Experimental setup of TUVAGG

Fig. 12  Effect of grinding parameters on the grinding force: (a) normal grinding force, (b) tangential grinding force
6.2.3 Residual stress

It is well known that the residual compressive stress can be generated in the machined surfaces by adopting an appropriate processing technology and adjusting/controlling the input energy to enhance the fatigue strength and corrosion resistance [26]. Figure 14 presents the residual stress under different parameters in both cases, and at the below of the figure, the increasing ratio of residual compressive stress of TUVAGG to CGG is presented.

It was found that the tooth surface presented residual compressive stress during both processing. The TUVAGG induced residual stress in the gear teeth surface exceeded the CGG ones by 13.2–29.3%. In addition, the residual compressive stress decreased with the radial grinding depth and spindle speed. It was leadingly attributed to that the grinding temperature enhanced with the radial grinding depth and spindle speed, and caused the thermal stress to offset the mechanical squeezing effect, and ultimately the residual compressive stress declined. Moreover, when the ultrasonic vibration was superimposed on the gear, the arc length of the grain was effectively elongated. The undeformed chip thickness was reduced, and the material’s plastic removal rate was enhanced. It could as well introduce the reciprocating ironing and grinding effects. Meanwhile, the vibrational stress induced by ultrasonic vibration could offset the internal stress and reduce the material’s yield strength, increasing the residual compressive stress.

6.2.4 Surface microhardness

The relationship between the surface microhardness and grinding parameters was presented in Fig. 15. It was concluded that the surface microhardness increased with the radial grinding depth and decreased with the spindle speed. Compared with CGG, the surface microhardness could be enhanced by 8.9–12.7% in the TUVAGG. When the spindle speed was increased, the grinding force gradually dropped, and the grinding temperature increased, causing the microhardness to decline.
However, with an increase in the radial grinding depth, both grinding force and temperature increased. Besides, the separate contact between the abrasive grains and workpiece under the action of ultrasonic vibration could be reduced to a certain extent. In addition, the abrasive grains had a certain strengthening effect on the workpiece, enhancing the microhardness at the higher radial grinding depth.

### 6.2.5 Surface roughness

The relationship between the surface roughness and grinding parameters is presented in Fig. 16. It can be observed from Fig. 16 that the surface roughness $Ra$ during the TUVAGG is significantly lower than that of CGG. The surface roughness $Ra$ could be reduced by 8.6–21.8% with the variation of spindle speed and radial grinding depth. It was enhanced with the radial grinding depth, and deteriorated with the spindle speed. This was attributed to the fact that the undeformed chip thickness of a single abrasive grain increased with the radial grinding depth and caused the plastic-stack on the grain side. With an increase in spindle speed, the number of dynamic abrasive grains participating in grinding increased, reducing the undeformed chip thickness. Under the action of tangential ultrasonic vibration, the increased arc length of grain also reduced the undeformed chip thickness. Due to the abovementioned reciprocating ironing and grinding of the machined surface, the TUVAGG produced a better surface roughness than CGG.

The gear surface machined by the CGG and TUVAGG was explored by observing the surface texture and microstructure. Figure 17 illustrates the surface topographies of the gear surface in CGG and TUVAGG. The gear surface texture during the TUVAGG was more flat and smooth than that in CGG. As presented in Fig. 17a, under the CGG, some residual material remained on the surface of the workpiece and was parallel to the grinding direction due to the gap between the effective grain edges. It resulted in obvious grooves and ridges on the surface. However, with the help of ultrasonic vibration, the trajectory between grain and workpiece interfered with each other, producing the reciprocating impact on the workpiece. This promoted the processed material removal in small chips. Therefore, as shown in Fig. 17b, a better surface quality was obtained in TUVAGG than in CGG.

### 7 Conclusions

In this study, a longitudinal vibration system of TUVAGG was designed based on non-resonant theory, and the performed gear grinding tests verified its effectiveness. According to the findings, the following conclusions were drawn:

1. It was observed that the simplified method for the designation of longitudinal gear vibration system was reasonable. The resonant frequency decreased with the gear thickness and reference radius. The horn length increased with an increase in the thickness and the reference radius decreasing.
2. Compared with the CGG, the normal and tangential grinding forces during the TUVAGG were reduced by 7.4–28.2% and 8.9–18.9%, respectively, and the grinding temperature was reduced by 7.6–25.7%. The grinding force reduction was more pronounced at low spindle speeds, thus extending the grinding wheel life.

3. It was revealed that the ultrasonic vibration increased the surface residual compressive stress by 13.2–29.3%, especially at low spindle speed, and enhanced the surface microhardness by 8.9–12.7%. The surface roughness was reduced by 8.6–21.8%, and the surface texture appeared more flat and smooth compared with the CGG.

Nomenclature

TUVAGG, Tangential ultrasonic vibration-assisted gear grinding; CGG, Conventional gear grinding; \( v_i \), Grinding wheel speed; \( a_i \), Radial grinding depth; \( v_m \), Gear feed rate; \( l_i \), Length of horn’s cylindrical section I; \( R_i \), Radius of horn’s cylindrical section I; \( R_0 \), Radius of horn’s conical section II; \( R_s \), Radius of horn’s cylindrical section III; \( R_4 \), Radius of horn’s cylindrical section IV; \( \xi_0 \), Maximal input displacement of the horn’s left end; \( \xi_s \), Displacement function of horn’s section I; \( \xi_s \), Displacement function of horn’s section II; \( \xi_t \), Displacement function of horn’s section III; \( \xi_s \), Displacement function of horn’s section IV; \( k_s \), Circular wave number of each section; \( S_s \), Cross-sectional area function; \( a_s \), Coefficient of homogeneous equations; \( C_s \), Coefficients of \( a_s \); \( \rho_s \), Material density; \( \lambda \), Wavelength; \( A \), Ultrasonic amplitude; \( A_{EM} \), FEM-predicted amplitude; \( A_p \), Amplitude measured by the resonance test; \( A_{TM} \), Amplitude of the theoretical analysis; \( A_{EM} \), Deviation between \( A_p \) and \( A_{EM} \) values; \( f_{EM} \), Experimental resonance frequency; \( f_{EM} \), FEM-predicted frequency; \( f_{TM} \), Theoretical design frequency; \( f_{EM} \), Deviation between \( f_{EM} \) and \( f_{TM} \) values; \( f_{EM} \), Deviation between \( f_{EM} \) and \( f_{TM} \) values.

Author contribution

Wenbo Bie conceived the analysis and wrote the manuscript. Bo Zhao provided supervisions on experimentation and manuscript preparation. Chongyang Zhao collected the data and revised the manuscript. Long Yin and Xingchen Guo performed the experiment. The authors discussed each reference paper together and contributed useful ideas for this manuscript.

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Availability of data and material

All data generated or analyzed during this study are included in the present article.

Declarations

Ethical approval

The article follows the guidelines of the Committee on Publication Ethics (COPE) and involves no studies on human or animal subjects.

Consent to participate

Not applicable.

Consent to publish

Not applicable.

Competing interests

The authors declare no competing interests.

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