Development of a low-frequency vibration-assisted drilling device for difficult-to-cut materials

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Abstract
In the drilling process of difficult-to-cut materials, conventional drilling has resulted in various problems such as high drilling temperature and poor machining quality. Low-frequency vibration-assisted drilling has great potential in overcoming these problems since broken chips are generated. In this paper, a low-frequency vibration-assisted drilling device is developed by using a novel ring flexure hinge as the elastic recovery mechanism. Firstly, based on the theory of elastic mechanics and mechanical vibration, the deflection of ring flexure hinge is designed theoretically, and the influence of its structural parameters on its deflection is analyzed. Then, the correctness of theoretical design is further verified by static and dynamic simulation and stiffness test. Finally, the vibration performance of the device is tested under no-load condition, and actual drilling test is conducted to verify the drilling performance. The results show that the device could realize the axial low-frequency vibration with constant frequency-to-rotation ratio and amplitude stepless adjustment and present good working stability under no-load and load conditions. In the actual drilling test of titanium alloy and carbon fiber–reinforced plastic (CFRP)/titanium alloy laminated structure, the device under appropriate processing parameters breaks titanium alloy chip into small pieces and reduces drilling temperature by 44% and inhibits secondary damage of CFRP. It provides the reference and guidance for the development of LFVAD device in presented work.

Keywords Low-frequency vibration-assisted drilling · Ring flexure hinge · Titanium alloy · Carbon fiber reinforced plastic

1 Introduction
Difficult-to-cut materials such as titanium alloy, high-temperature alloy, and composite materials have a great prospect in aerospace and aircraft manufacturing. Due to tens of thousands of holes to be drilled to meet the mechanical bolting or riveting demand for aircraft assembly, it is still a challenge to make holes on the difficult-to-cut materials with high quality [1, 2].

The machining difficulties of the difficult-to-cut materials have been reported, such as delamination damage during drilling of CFRP and CFRP/titanium alloy stacked materials [3], chip evacuation during drilling, and low thermal conductivity combined with a high chemical affinity of titanium alloys [4, 5]. These machinability problems lead to high machining temperature, short tool life, and poor machining quality.

Compared with conventional drilling (CD), vibration-assisted drilling has great potential in reducing cutting temperature, prolonging tool life, and improving machining quality [6, 7]. According to the frequency of vibration, there are low-frequency vibration-assisted drilling (LFVAD) and ultrasonic vibration-assisted drilling (UVAD). UVAD has significant technological effects in the processing of difficult-to-cut materials [8–10]. However, in ultrasonic vibration drilling, the vibration amplitude can reach several microns, which restricts the machining efficiency [11, 12]. LFVAD process with appropriate processing and vibration parameters becomes an
intermittent cutting, which has more advantages in chip breaking and chip removal, reducing drilling temperature and improving processing efficiency [13–17].

Some research on the LFVAD of aluminum alloy, titanium alloy, and carbon fiber–reinforced plastic (CFRP)/titanium (aluminum) stacked materials was performed. Pecat et al. [18, 19] developed a kinematic model of the vibration-assisted drilling process for a two-fluted cutter and studied the chip shape under different processing parameters and vibration parameters through LFVAD experiment. It was found that the tool wear and the cutting temperatures were significantly decreased by the application of LFVAD, whereas the thrust force was raised compared to CD. Okamura et al. [20] carried out the experiment of LFVAD of titanium alloy. The results showed that the drilling temperature of titanium alloy decreased greatly, and the drill wear rate was suppressed largely by applying low-frequency vibration. Besides, the chip was divided into short length, and the hole exit burr height and thickness became small under LFVAD. As the amplitude becomes large and the vibration frequency becomes high, these effects appear more remarkably. Hussein et al. [21] carried out the experiment of LFVAD of CFRP/Ti6Al4V stacked material and found that the temperature of LFVAD was significantly lower than that of CD. The delamination defects at the entry and exit of the CFRP were significantly improved due to an enhancement in the chip evacuation mechanism and a reduction in the cutting temperature. In addition, LFVAD could significantly reduce tool wear, but the mechanical load was larger than that of CD.

Numerous significant studies have shown that LFVAD has great potential in machining difficult-to-cut materials. Low-frequency vibration-assisted drilling is the basis of experimental research on LFVAD. At present, low-frequency vibration-assisted drilling device mainly adopts mechanical, electromagnetic, and hydraulic modes [22]. The electromagnetic vibration drilling device has simple structure and high vibration frequency. However, it is only suitable for small hole machining because of its small power. The hydraulic vibration drilling device has advantages of long life, low noise, high output power, and good buffering effect, while its structure is complex and its manufacturing cost is high. The mechanical vibration drilling device has advantages of simple structure, good rigidity, and long service life. However, cam mechanism, eccentric wheel mechanism, and crank slider mechanism are often used in the vibration device [23, 24], which makes the amplitude adjustment inconvenient and requires additional power supply. Meanwhile, spring is generally used as the elastic recovery mechanism in these devices, resulting in weak rigidity of the process system and poor amplitude stability in vibration drilling, which limits the application of vibration drilling technology.

In order to promote the application of LFVAD, Laporte et al. [25] has developed a mechanical low-frequency vibration drilling device. The spring pile is used as elastic recovery mechanism of the device, which reduces rigidity of the system. What is more, additional pins are needed to realize torque transmission, and linear bearing is used to ensure the axial transmission of vibration, which makes structure more complex.

In this paper, a novel ring flexure hinge is used as elastic recovery mechanism to develop a mechanical low-frequency vibration-assisted drilling device through theoretical design, static and dynamic simulation, and stiffness test. The ring flexure hinge can be machined with different deflection according to working requirements by changing structural parameters of the ring flexure hinge. In addition, the ring flexure hinge also transfers torque to rotate the vibration output part and provides guidance for axial vibration, which makes the whole structure more compact. The LFVAD device could be directly installed on the spindle of machine tool, and with the rotation of the spindle, it could realize the axial low-frequency vibration with constant frequency-to-rotation ratio and amplitude stepless adjustment. The vibration performance of the device is tested under no-load condition, and actual drilling test of titanium alloy and CFRP/titanium alloy laminated structure is conducted to verify the drilling performance.

2 Working principle of the LFVAD device

The low-frequency vibration system composition and schematic diagram of the LFVAD device are shown in Figs. 1 and 2.

The low-frequency vibration system mainly includes four parts: sinusoidal end cam, slider mechanism, ring flexure hinge, and fixed block, which are integrated into BT40 tool holder.

Sinusoidal end cam, as excitation of the low-frequency vibration system, determines amplitude range and frequency-to-rotation ratio of the device. As shown in Fig. 2, a sinusoidal curved surface is processed on one side of the ring plate. The axial low-frequency vibration of tool is the result of superposition of two cams with same sinusoidal surface, and the axial vibration displacement equation can be expressed as follows:

\[ A = \sqrt{2(1-\cos \varphi)}A_0 \cos (\omega t + \psi) \]  

(1)

Fig. 1 The low-frequency vibration system composition
where $A$ is the axial amplitude of the device, $A_0$ is the amplitude of a single sinusoidal surface, $\omega$ is the circular frequency of vibration, $\varphi$ is the phase difference between the sinusoidal surface of the upper and lower end cams, and $\psi$ is the phase angle of tool vibration.

The stepless adjustment of amplitude in the range of $0 - 2A_0$ is realized by changing phase difference between the two cams. The low-frequency vibration system mainly plays the role of chip breaking in drilling process. For common double-edged twist drill, the chip breaking condition [26] is:

$$\frac{4A}{f_r} \geq \frac{1}{\sin\left(\frac{\omega_f n}{2}\right)}$$

(2)

where $f_r$ is the feed rate, and $\omega_f$ is the frequency-to-rotation ratio. When the feed rate $f_r$ satisfies $4A/f_r > 1$, the frequency-to-rotation ratio needs to meet certain conditions to realize theoretical chip breaking. When $4A/f_r < 1$, no matter how the frequency-to-rotation ratio is changed, the chip will not break theoretically. Therefore, low-frequency amplitude $A$ of the designed device must satisfy $A > f_r/4$. In addition, the number of cycles of sinusoidal end cams and the number of cylindrical sliders are both designed as odd numbers, that is, the frequency-to-rotation ratio $\omega_f$ is odd number. Considering the overall structure and size of the device, the number of cycles of the sinusoidal surface is set to 5, and the amplitude of sinusoidal surface is set to 0.05 mm.

As the transmission mechanism, slider mechanism includes a fixed plate with several through holes and cylindrical sliders. In order to ensure that the tool vibrates along the axis without radial vibration, the number of cylindrical sliders in the fixed plate should be equal to the number of periods of sinusoidal surface. The fixed plate and cylindrical slider are as shown in Fig. 2. Five identical cylindrical sliders are placed in five through holes of fixed plate. Two hemispherical grooves are machined on the top and bottom of the cylindrical slider respectively. A steel ball is placed in each groove. The upper steel ball of cylindrical slider is in contact with the upper sinusoidal end cam, and the lower steel ball is in contact with lower sinusoidal end cam. The cylindrical slider can move axially along the through-hole wall of the fixed plate, which is fixed with the outer shell by screw.

As elastic recovery mechanism, the ring flexure hinge provides elastic restoring force for axial vibration of the system. The detailed design and analysis of ring flexure hinge will be carried out in the next section.

The LFVAD device can be directly installed on the spindle of the machine tool. With the rotation of the spindle, the upper sinusoidal end cam rotates with the BT40 tool holder, providing harmonic excitation. The shell of device is connected with fixed part of the machine tool by fixed block and does not rotate during operation. The fixed plate of slider mechanism is connected with shell of the device by screws, and cylindrical slider of slider mechanism slides up and down along the through-hole of fixed plate under harmonic excitation. The amplitude of harmonic excitation can be changed by adjusting the lock nut. The frequency of harmonic excitation is related to the number of sinusoidal period (frequency-to-rotation ratio) and spindle speed of machine tool. The ring flexure hinge provides restoring force of harmonic excitation and makes the same harmonic vibration at the same time. Through the cooperation of slider mechanism, double sinusoidal end cam and
ring flexure hinge, the tool can generate axial low-frequency harmonic vibration while rotating at high speed.

According to the above, the LFVAD device could realize the low-frequency vibration of cutting tool with constant frequency-to-rotation ratio and amplitude stepless adjustment.

3 Design of ring flexure hinge

3.1 Design principle of the ring flexure hinge

The flexure hinges are formed by the middle elastic deformation unit connecting two rigid bodies, and the relative movement between the two rigid bodies is achieved through the deformation of the intermediate unit [27]. Flexure hinges have the advantages of no mechanical friction, no clearance, compactness, and high motion sensitivity. Therefore, in some mechanisms, it often replaces the spring to realize the relative motion of the internal structure of the device [28, 29]. The LFVAD device adopts a novel ring flexure hinge as elastic recovery mechanism, and the material of ring flexure hinge is 65Mn spring steel. From Fig. 3, it can be seen that the ring flexure hinge is located at the bottom of connecting rod. The inner ring of thin ring plate is connected with connecting rod, and the outer ring of thin ring plate is connected with a large connecting rod, and the inner ring of thin ring plate is fixed with the outer ring of thick ring surface. The ring flexure hinge can be regarded as a thin ring plate. The elastic recovery mechanism, and the material of ring flexure hinge is LFVAD device adopts a novel ring flexure hinge as elastic recovery mechanism, and the material of ring flexure hinge is 65Mn spring steel. From Fig. 3, it can be seen that the ring flexure hinge is located at the bottom of connecting rod.

According to the above, the LFVAD device could realize the low-frequency vibration of cutting tool with constant frequency-to-rotation ratio and amplitude stepless adjustment.

As shown in Fig. 3, the ring flexure hinge is simplified to a ring thin plate model for axial stiffness analysis. The inner ring radius is \( b \), the outer ring radius is \( a \), and the thickness is \( h \).

The differential equation of circular plate bending in polar coordinate system based on the research [30] is expressed as

\[
\left( \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} \right) \left( \frac{\partial^2 w}{\partial r^2} + \frac{1}{r} \frac{\partial w}{\partial r} + \frac{1}{r^2} \frac{\partial^2 w}{\partial \theta^2} \right) = \frac{q(r, \theta)}{D} \tag{3}
\]

where \( D = \frac{Eh^3}{12(1+\nu)} \) is the bending stiffness of the plate, \( E \) is the elasticity modulus of the plate, \( h \) is the thickness of the plate, \( \nu \) is Poisson’s ratio, and \( q \) is the load.

For axisymmetric loads \( q = q(r) \), because circular (ring) plates are axisymmetric in geometry, their deformation will be axisymmetric. The derivatives of deflection \( w \) to \( \theta \) are all 0. The basic Eq. (3) of the circular plate bending is simplified to the Euler type ordinary differential equation as:

\[
\left( \frac{d^2}{dr^2} + \frac{1}{r} \frac{d}{dr} \right) \left( \frac{d^2 w}{dr^2} + \frac{1}{r} \frac{dw}{dr} \right) = \frac{q(r)}{D} \tag{4}
\]

The general solution of Eq. (4) can be obtained as:

\[
w = Ar^2 + Br^2 \ln \frac{r}{a} + C \ln \frac{r}{a} + K + w(r) \tag{5}
\]

where \( a \) is the external radius of circular (ring) plate, \( w(r) \) is the special solution of the equation, and the constants \( A, B, C \), and \( K \) are determined by the boundary conditions of the plate.

The general solution of the ring plate and the circular plate is the same. Since there is no uniformly distributed load on the ring plate, that is, \( q(r) = 0 \), the deflection \( w \) can be expressed as:

\[
w = Ar^2 + Br^2 \ln \frac{r}{a} + C \ln \frac{r}{a} + K \tag{6}
\]

The inner ring of the thin ring plate is fixed with the connecting rod, and the outer ring of the thin ring plate is subjected to uniform transverse shear force \( Q_0 \) without rotation. According to working conditions of the ring plate, two boundary conditions can be obtained as:

when \( r = b \), there are

\[
w_{r=b} = 0, \quad \frac{\partial w}{\partial r}_{r=b} = 0 \tag{7}
\]
When \( r = a \), there are:

\[
\left( \frac{\partial w}{\partial r} \right)_{r=a} = 0, 
(Q)_{r=a} = -Q_0
\]  

(8)

The internal force element formulas (bending moment, torsion, and transverse shear force) of ring and circular plates are expressed as

\[
\begin{align*}
M_r &= -D \left[ \frac{\partial^2 w}{\partial r^2} + \nu \left( \frac{\partial w}{\partial r} + \frac{1}{r} \frac{\partial^2 w}{\partial \theta^2} \right) \right] \\
M_\theta &= -D \left[ \frac{1}{r} \frac{\partial w}{\partial r} + \frac{1}{r^2} \frac{\partial^2 w}{\partial \theta^2} \right] + \nu \frac{\partial^2 w}{\partial r^2} \\
Q_r &= -D \frac{\partial}{\partial r} (\nabla^2 w) 
\end{align*}
\]  

(9)

Substituting Eq. (7) into Eq. (6), we get the following equations:

\[
\begin{align*}
0 &= A b^2 + B b^2 \ln \frac{b}{a} + C \ln \frac{b}{a} + K \\
0 &= 2Ab + 2Bb \ln \frac{b}{a} + Bb + C \frac{1}{b}
\end{align*}
\]  

(10)

Substituting Eq. (8) into Eqs. (8) and (9), we get the following equations:

\[
\begin{align*}
0 &= 2Aa + Ba + C \frac{1}{a} \\
Q_0 &= \frac{4DB}{a}
\end{align*}
\]  

(11)

The constants A, B, C, and K can be obtained by simultaneous Eqs. (10) and (11), and the deflection of the ring plate can be calculated by introducing them into Eq. (6):

\[
w = Q_0 a \left[ \left( \frac{r^2 \ln \left( \frac{r}{a} \right)}{4D} \right) - \frac{b^2 (b^2 - a^2 - 6a^2 \ln \left( \frac{b}{a} \right))}{8D (a^2 - b^2)} - \frac{r^2 \left( 2b^2 \ln \left( \frac{b}{a} \right) - a^2 + b^2 \right)}{8D (a^2 - b^2)} - \frac{a^2 b^2 \ln \left( \frac{b}{a} \right) \ln \left( \frac{r}{a} \right)}{2D (a^2 - b^2)} \right]
\]  

(12)

In the design process of ring flexure hinge, it is necessary to know the influence of structure parameter thickness \( h \) and inner and outer diameter on its deflection. Figure 4 shows the influence of the thickness of the ring flexure hinge on its deflection, and Figure 5 shows the relationship between the outer diameter \( a \) and deflection \( w \) when the inner diameter is determined.

It can be seen from Figs. 4 and 5 that the thickness has a great influence on the deflection of the ring flexure hinge, and the deflection decreases with the increase of the thickness. In addition, with the increase of the thickness, the influence degree becomes smaller and smaller. When the inner diameter is determined, the deflection increases with the increase of the outer diameter.

Because the stiffness of the thick ring connected by the outer ring is very large, the transverse shear force acting on the outer ring of the thin ring plate can be approximately transformed into the concentrated force on the thick ring. Thus, the axial stiffness \( K \) of the ring flexure hinge can be written as:

\[
K = 2\pi r Q_0 / \omega
\]  

(13)

According to the theory of mechanical vibration [31], the differential equation of undamped motion of the plate for free vibration can be expressed as:

**Fig. 4** Influence of thickness \( h \)
\[ D\nabla^4 w + \rho_A \frac{\partial^2 w}{\partial t^2} = 0 \]  

where \( \rho_A \) is the surface density of the plate.

Assuming that the plate performs a simple harmonic motion, the displacement of the plate can be expressed as:

\[ w = W \cos(\omega t) \]  

where \( W \) is the vibration mode function of the plate which is related to the coordinates and must meet the corresponding boundary conditions, and \( \omega \) is the circular frequency of the vibration.

Substituting Eq. (15) into Eq. (14), we get the following equations:

\[ (\nabla^4 - k^4) W = 0 \]  

where \( k^4 = \frac{\rho_A \omega^2}{D} \).

In a polar coordinate system, the Laplace operator in the Eq. (16) is expressed as

\[ \nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} \]  

The total solution of Eq. (16) in polar coordinate system can be obtained as follows:

\[ W(r, \theta) = \sum_{n=0}^{\infty} \left[ A_n J_n(kr) + B_n Y_n(kr) + C_n I_n(kr) + D_n K_n(kr) \right] \cos n\theta + \sum_{n=1}^{\infty} \left[ A_n^* J_n(kr) + B_n^* Y_n(kr) + C_n^* I_n(kr) + D_n^* K_n(kr) \right] \cos n\theta \]  

where \( J_n \) and \( Y_n \) are Bessel functions of the first type and second type respectively; \( I_n \) and \( K_n \) are deformed Bessel functions of the first type and the second type respectively; \( A_n, A_n^*, \ldots, D_n, D_n^* \) represent unknown constants related to the mode shape of circular thin plates, and their specific values are determined by boundary conditions; \( n \) is the number of pitch diameter when the circular plate vibrates.

When the origin of polar coordinates coincides with the center of the circular plate, the solution of the circular plate vibration differential equation can be simplified as follows:

\[ W_n = [A_n J_n(kr) + C_n I_n(kr)] \cos n\theta \]  

When the ring flexure hinge works, it can be regarded as the vibration of the ring plate with the inner hole boundary fixed and the outer ring connected with the rigid mass block. Thus, the boundary conditions of the ring plate with outer diameter \( a \) and inner diameter \( b \) can be expressed as follows:

\[ \begin{align*}
W(b) &= 0 \\
\frac{\partial W(b)}{\partial r} &= 0
\end{align*} \]

Therefore, the circular frequency of the transverse vibration of the ring plate can be calculated as follows:

\[ \omega = \frac{a_{ns}}{b^2} \sqrt{\frac{D}{\rho_A}} \]  

where \( a_{ns} \) is the frequency constant of the transverse free vibration of the plate, which is determined by the boundary conditions, the outer diameter \( a \), the inner diameter \( b \), and the surface density \( \rho_A \) of the rigid mass [31].
3.2 Simulation of the ring flexure hinge

The static and dynamic simulation analysis of ring flexure hinge is carried out by ANSYS workbench, and the axial stiffness is compared with the theoretical design. The strength of ring flexure hinge is checked, and the limit speed of the device is determined.

Considering the overall size of the device and the processing factors of the annular flexure hinge, the outer diameter \( a = 33.5 \) mm, the inner diameter \( b = 17.5 \) mm, and the thickness \( h = 0.6 \) mm of the ring flexure hinge are determined. The material is 65 Mn spring steel with elastic modulus \( E = 1.986 \times 10^{11} \) Pa, Poisson’s ratio \( \nu = 0.288 \), and density \( \rho = 7.81 \) g/cm\(^3\).

Firstly, the finite element model is established according to the size of ring flexure hinge. According to Eq. (12), when the transverse shear force \( Q_0 = 1762.6 \) N/m is applied to the outer ring of the thin ring plate, the axial displacement \( w \) can be calculated as 200 \( \mu \)m, which meets the design requirements.

The boundary conditions of simulation are consistent with theoretical boundary conditions that the inner ring of ring plate is fixed, and the outer ring of thin ring plate is subject to transverse shear force of 1762.6 N/m. In addition, the outer ring is restrained from rotation. The results of static simulation of ring flexure hinge are shown in Fig. 6.

From Fig. 6, it is easy to know that the maximum axial displacement \( w \) was 191.64 \( \mu \)m. According to Eq. (13), it can be calculated that the axial stiffness \( K \) is 1.935 N/\( \mu \)m, which differs 4.37% from the theoretical value of 1.854 N/\( \mu \)m.

If there are no chamfers in the inner ring and outer ring joints of ring flexure hinge, singular stress points will appear in the strength analysis (as the mesh becomes more and more dense, the stress increases infinitely). Therefore, chamfers should be added to the inner and outer ring joints of ring flexure hinge in the strength analysis. Adding chamfering to inner and outer ring joints increases the average thickness of ring thin plate, which will affect its stiffness. In actual processing, chamfering is often used to reduce stress concentration, so it is necessary to establish a finite element model with chamfering for stiffness and strength simulation analysis.

A finite element model of the connecting rod with a chamfer radius of 1 mm at the inner and outer ring joints was established, and the stiffness and strength were analyzed. The upper part of connecting rod is fixed, and the downward uniform transverse shear force is 1.7626 N/mm acting on the outer ring connection of thin ring plate. The results of axial displacement of thick ring are shown in Fig. 7.

It is evident that the chamfers have a great influence on the axial stiffness of ring flexure hinge. The simulation results show that the axial stiffness \( K \) of ring flexure hinge is 2.151 N/\( \mu \)m, which differs 11.2% from that of the non-chamfering flexure hinge.

Next, the strength of ring flexure hinge with chamfers is analyzed. The top of connecting rod is fixed, and thick ring is loaded with the downward displacement load of 200 \( \mu \)m. The results are shown in Fig. 8. It is found that the maximum stress occurs at the joint between the ring thin plate and the interior, which is 355.16 MPa. The yield strength of 65Mn spring steel after heat treatment is 785 MPa and the safety factor \( s = \frac{785}{355.16} = 2.21 > 1 \). Therefore, the strength condition is satisfied when the amplitude is 100 \( \mu \)m.

Then, the dynamic simulation analysis is carried out. First, the modal analysis of ring flexure hinge with chamfers is carried

Fig. 6 Static simulation of ring thin plate
out. The previous four natural frequencies are obtained as shown in Table 1, and the first four modes are shown in Fig. 9.

Next, the harmonic response is analyzed. The frequency range is set to 0–500 Hz. The top of the connecting rod is fixed, and the outer ring of thin ring plate is subject to the downward transverse shear force of 1762.6 N/m. The results are shown in Fig. 10.

The resonance frequency is about 390 Hz, so the vibration frequency of the low-frequency vibration system cannot be higher than 390 Hz. When the frequency-to-rotation ratio of low frequency is 5, the spindle speed of the machine tool cannot exceed 4680 r/min. Therefore, in the actual processing process, the spindle speed of machine tool is limited.

3.3 Test of the ring flexure hinge

The test of low-frequency structure is mainly to verify the axial stiffness of ring flexure hinge. The connecting rod is...
machined according to the designed size, and the axial static stiffness of connecting rod is tested by using the SANS-CMT5205-200 microcomputer-controlled electronic universal testing machine, as shown in Fig. 11.

The axial force-z displacement curve of the ring flexure hinge is obtained as shown in Fig. 12. It can be found that the axial displacement of connecting rod is approximately proportional to the applied axial force, and its axial stiffness is calculated to be 1.82 N/μm.

By comparing the theoretical and experimental stiffness, the difference between the experimental and theoretical values is 1.84%, which verifies the correctness of the theoretical calculation. The difference between the experimental value and the simulation value of the ring flexure hinge with chamfers is large, which is 15.39%. The reason may be that the thickness

\[ h \]

of ring flexure hinge has a great influence on its axial stiffness. It is unavoidable that there are machining errors in processing. Also in compression tests, there may be some test errors. Meanwhile, the uniformity of heat treatment and creep characteristics of materials may affect their axial stiffness.

| Table 1 | Previous four natural frequencies |
|---------|----------------------------------|
| Mode    | 1      | 2      | 3      | 4      |
| Frequency (Hz) | 224.96 | 225.21 | 390.47 | 1655.9 |

Fig. 9  Previous four modes. (a) First mode; (b) second order mode; (c) third order mode; (d) fourth order mode

Fig. 10  Harmonic response analysis curve
4 Performance test of the LFVAD device

4.1 Vibration performance test

The LFVAD device and its vibration performance test site under no-load condition are shown in Fig. 13. The device can be directly installed on VMC-850E of vertical machining center (speed range 50–6500 r/min, positioning accuracy ±0.0075 mm, tool handle specification BT40). The KEYENCE LK-G10 non-contact laser measuring system (sampling frequency 6.5 kHz) is used to measure the axial low-frequency vibration of the output terminal of the device without load. Because there is no reference plane perpendicular to the axis of twist drill in the cutting part of twist drill, the carbide twist drill is replaced with cylindrical rod for measurement.

When the phase difference between upper and lower sinusoidal end cam is adjusted to 30° by rotating the locknut, the maximum low-frequency amplitude can be calculated as 17.36 μm from Eq. (1) in the third section.
When the spindle speed of the machine tool is set to 600 r/min, the low-frequency axial vibration results of the tool are shown in Fig. 14. The overall average low-frequency amplitude is about 15 μm, which is lower than the theoretical calculation amplitude, and the deviation is within the acceptable range. It can be seen from Fig. 14b that the frequency of low-frequency vibration is 50 Hz, which is consistent with the frequency-to-rotation ratio at this spindle speed. It is demonstrated that the low-frequency vibration of the device under no-load condition is stable, which meets the design requirements.

4.2 Drilling performance test

The drilling performance test of the LFVAD device was carried out with two cases of typical drilling operation. The first one is titanium alloy drilling, and the second one is CFRP/
titanium alloy laminated structure drilling, as shown in Fig. 15 (a and b).

### 4.2.1 Titanium alloy drilling

In this experiment, the uncoated tungsten carbide twist drill was used to drill titanium alloy (Ti-6Al-4V, 300 mm × 150 mm × 10 mm). The main geometrical features of the used drills include a 5-mm diameter, a 135° point angle, a 30° helix angle, and two cutting edges. The experimental setup is illustrated in Fig. 15 (a).

The material properties of Ti-6Al-4V are shown in Table 2, and the drilling conditions are presented in Table 3. In the drilling process, three-way piezoelectric ceramic dynamometer (KISTLER 9257B) was used to collect thrust force, and the sampling frequency was 20 kHz.

Figure 16 shows the time and frequency domain signals of thrust force during the drilling process when amplitude is 20 μm. LFVAD presents intermittent cutting process, and increases the instantaneous impact force in the cutting process. It is noticed from Fig. 16b that the frequency of low-frequency vibration is still stable under load conditions, which further proves that the ring flexure hinge is feasible as elastic recovery mechanism of the device.

The chip morphology reflects the movement path of tool to some extent. The chip morphology of titanium alloy under different amplitudes was analyzed and compared. Figure 17 shows the chip morphology of titanium alloy with different amplitudes under the same processing parameters.

As shown in Fig. 17a, when the amplitude is 0 μm, it is conventional drilling (CD). The CD produces continuous conical helix type chip with long length. It is difficult to break the titanium alloy chips because of the constant cutting thickness in the CD process.

Figure 17b shows the chip morphology of titanium alloy with amplitude of 10 μm. According to Eq. (2), when the amplitude A is 10 μm and the feed rate f is 0.05 mm/r, the theoretical chip breaking condition is not satisfied (4A/f < 1). Therefore, what is generated is still conical helix type chip with short length.

Broken titanium alloy chips are generated when the amplitude is 20 μm and 30 μm, shown in Fig. 17c and d. Because the amplitude and feed rate meet the

| Table 2 Material properties of Ti-6Al-4V |
|----------------------------------------|
| Properties              | Value |
|-------------------------|-------|
| Density (g/cm³)         | 4.4   |
| Elastic modulus (GPa)   | 109   |
| Tensile strength (MPa)  | 950   |
| Poisson’s ratio          | 0.34  |
| Elongation (%)           | 8     |
| Hardness (HV)            | 360   |
| Thermal conductivity(W/(m·°C)) | 7.5   |

| Table 3 Drilling conditions of titanium alloy drilling |
|-------------------------------------------------------|
| Variable                   | level |
|----------------------------|-------|
| Spindle speed/(r/min)      | 400   |
| Feed rate/(mm/r)           | 0.05  |
| Amplitude/(μm)             | 0, 10, 20, 30 |
| Frequency/(Hz)             | 33.33 |
theoretical chip breaking conditions ($4A/f_1 > 1$), the tool and workpiece can be separated and contacted periodically in the drilling process, resulting in broken chips. Therefore, the chip breaking effect can be achieved by selecting appropriate processing parameters and vibration parameters. The drilling performance of the designed LFVAD device meets the actual processing requirements.

The titanium alloy hole edge burr affects the surface quality and the performance. Burr height is often used to characterize the size of burr. As shown in Fig. 18, the exit burr height of titanium alloy is much higher than the entrance, so the exit burr of titanium alloy is mainly discussed. Ten points are selected evenly along the circumference of the hole exit to measure the burr height, and their average value is taken as the average exit burr height, as shown in Fig. 19a. Figure 19b shows the effect of amplitude on the titanium alloy exit burr height.

In the process of drilling, the thrust force at exit of titanium alloy is an important factor affecting the burr height. It can be seen from Fig. 19b that the exit burr height of titanium alloy increases with the increase of amplitude. The maximum thrust force in LFVAD increases with the increase of vibration amplitude. The larger the maximum thrust force is, the earlier the time of plastic deformation occurs when the material at the hole bottom reaches the yield limit, which leads to the increase of the bottom uncut thickness, and the more uncut material flows with plastic deformation after cutting, resulting in the larger the exit burr height. Therefore, in order to reduce the exit burr height of titanium alloy, the small vibration amplitude is more desirable when the chip breaking effect is achieved.
4.2.2 CFRP/titanium alloy laminated structure drilling

In this experiment, the uncoated tungsten carbide twist drill with the same structural parameters as section 4.2.1 was used to drill CFRP (T700-12K/AG80, 32 lays of unidirectional laying, 300 mm × 120 mm × 5 mm)/titanium alloy (Ti-6Al-4V, 300 mm × 210 mm × 5 mm) laminated structure, as shown in Fig. 15b. K-type thermocouple was used to detect the interface temperature at 0.5 mm away from the hole wall in the drilling process, and the data was read by the fast response thermocouple acquisition instrument (HR-USB-T008). Finally, the machining quality of CFRP was observed by using the super depth of field microscope and scanning electron microscope.

The material properties of CFRP are shown in Table 4, and the drilling conditions of CFRP/titanium alloy drilling are presented in Table 5.

Figure 20 presents the measured temperature curves during drilling at feed rates of 0.03 mm/r and 0.06 mm/r. Compared with CD (A=0 μm), the maximum temperature of LFVAD with amplitude of 20 μm is reduced by about 44%. LFVAD changes the cutting process from continuous cutting to intermittent cutting, which increases the duty cycle of the tool. In addition, the discontinuous titanium alloy chips are efficiently evacuated from the hole along spiral groove, which takes away a large amount of cutting heat.

In the process of drilling CFRP/titanium alloy laminated structure, the drilling temperature of titanium alloy and the chip morphology of titanium alloy have great influence on the machining quality of CFRP. The delamination factor “Φd” is evaluated using Eq. (22):

\[
Φ_d = \frac{S_{\text{actual}} - S_{\text{nominal}}}{S_{\text{nominal}}}
\]

where \( S_{\text{actual}} \) represents the area of a circle that is concentric to the hole and circumscribing the delamination extents, while \( S_{\text{nominal}} \) is the nominal hole area [32].

Figure 21 shows the observation results of the entrance and exit of the CFRP hole and the SEM morphologies of the drilled CFRP hole walls with the fiber orientation angle at 90°.

In the drilling process of CFRP/titanium alloy laminated structure, the titanium alloy under the CFRP plate plays a
good supporting role, which reduces the delamination defects at CFRP hole exit, resulting in the delamination factor of hole entrance larger than hole exit. In addition, compared to CD, the delamination factor of CFRP hole in LFVAD is smaller and increases slower with the increase of feed rate, as shown in Fig. 22.

When the feed rate is 0.03 mm/r, the delamination at CFRP hole exit in CD is slightly better than LFVAD. This is because when the feed rate is small, the drilling temperature of CD and LFVAD is very low. Moreover, the chip thickness is thin and chip bending section coefficient is small, so it is easy to occur bending deformation in the removal process, resulting in less thermal-mechanical damage to the hole wall. In this case, the drilling phase of titanium alloy has little influence on exit delamination of CFRP, and the exit delamination caused by CFRP drilling phase will be dominant. Compared with CD, LFVAD has larger maximum thrust force, which is easier to cause delamination at CFRP hole exit.

However, with the increase of feed rate, the drilling temperature and chip thickness increase, which leads to the increasing influence of titanium alloy drilling phase on the exit delamination of CFRP. The drilling temperature of LFVAD is lower than CD. Besides, discontinuous titanium alloy chips can be efficiently evacuated out of hole along the spiral groove, which reduces thermal-mechanical damage of chips on CFRP hole wall, thus significantly reducing the delamination factor and improving the quality of CFRP hole.

### 5 Conclusions

In this paper, a novel ring flexure hinge is used as elastic recovery mechanism to develop the low-frequency vibration-assisted drilling device. Through performance test, the LFVAD device presents good working stability under no-load and load conditions. The results are summarized as follows:

1. The simulation and test of ring flexure hinge verified the correctness of theoretical design. The ring flexure hinge has good linear elasticity, and the structural parameters determine its stiffness, which affects limit speed of the device.
2. The ring flexure hinge can be machined with different deflection according to working requirements by changing structural parameters of the ring flexure hinge.

### Table 4 Material properties of T700-12K/AG80 CFRP

| Property                          | Value |
|----------------------------------|-------|
| Density (g/cm³)                  | 1.55  |
| Fiber volume fraction (%)        | 60%   |
| Transverse modulus (GPa)         | 40    |
| Longitudinal modulus (GPa)       | 230   |
| Tensile strength (MPa)           | 4900  |
| Glass transition temperature (°C)| ≥230  |

### Table 5 Drilling conditions of CFRP / titanium alloy laminated structure drilling

| Variable                        | Level          |
|---------------------------------|----------------|
| Spindle speed (r/min)           | 400            |
| Feed rate (mm/r)                | 0.03, 0.04, 0.05, 0.06 |
| Amplitude (μm)                  | 0, 20          |
| Frequency (Hz)                  | 33.33          |
thickness $h$ of ring flexure hinge has a great influence on its deflection. The deflection increases with the increase of thickness, and the influence degree of thickness on the deflection gradually decreases with the increase of thickness.

3. In LFVAD of titanium alloy, exit burr height of titanium alloy increases with the increase of amplitude. Therefore, in order to reduce exit burr height of titanium alloy, the small vibration amplitude is more desirable when the chip breaking effect is achieved.

4. The LFVAD under appropriate processing parameters improves chip removal effect and significantly reduces the drilling temperature, which has a promising application in CFRP/titanium (aluminum) alloy laminated structure and other difficult-to-cut materials.

Fig. 20 Temperature of drilling process

Fig. 21 The entrance and exit morphology of CFRP: a morphology of hole wall with $A=0 \, \mu m$, b morphology of hole wall with $A=20 \, \mu m$
Fig. 22 Delamination factor

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