Study on the Design of Cutting Disc in Ultrasonic-assisted Machining of Honeycomb Composites

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Abstract. To study the influences of geometrical parameters on the design of ultrasonic-assisted cutting disc tool for honeycomb composites, a finite element model integrating the cutting disk and ultrasonic horn was established. The performance of the output amplitude of the cutting disc was analyzed under different geometrical parameters including cutter radius, cutter thickness and anterior angle. Then the influences of the geometrical parameters on the energy density and bending rigidity of the cutting disc were discussed. The simulation results shown that with the increase of cutter radius, both the energy density and rigidity decreased, and the amplitude at the cutting edge increased firstly and then decreased under the interaction of the energy density and rigidity. There was an ideal interval of radius within which the amplitude reached the maximum. With the increase of cutter thickness and anterior angle, the energy density decreased while the rigidity increased, the amplitude at the cutting edge presented a declining trend.

1. Introduction

Honeycomb composite has been more and more widely applied in such fields as the aerospace, auto, ship, etc. for its excellent physical properties and chemical properties[1][2]. It is mainly processed in two methods, traditional high-speed milling and ultrasonic-aided processing. Specifically, the latter applies ultrasonic vibration in traditional tool motion to effectively reduce the cutting force and improve the processing quality and processing efficiency[3][4]. Special cutters applied in the ultrasonic-assisted processing of honeycomb composites mainly include triangular blade knife and cutting disc. Cutting disc is mainly used for plane and surface machining, as a tool for finish machining. As an actuating element in the cutting process, cutting disc’s geometrical shape and dimension parameters would impact the vibration performance, machining accuracy and processing efficiency of the ultrasonic system greatly. Therefore, the selection of optimal parameters in the design of cutter is of great importance.

Regarding the technical research on the ultrasonic-assisted processing of honeycomb composites in foreign countries, Austrian GFM Company took the lead in producing a special ultrasonic machine by aiming at honeycomb composites; French CRENO also made a planer-type ultrasonic machine and developed a mechanical-arm machine, which could finish the processing of honeycomb materials of complicated shape. But since related applications involve the military industry, there were few research papers. In terms of the design of ultrasonic vibration system, Lucas[5] et al. established a finite element model of a single-blade ultrasonic-assisted cutting device by simulating the cutting of cheese and two other kinds of materials that could be used for simulating food and biological tissue; Nick[6] et al. built a mathematical model of the design process and came up with the optimization method. With the maximum output amplitude in resonant mode as the objective, an objective function was defined to
optimize the ultrasonic vibration system; BJORNSSON \[7\] et al. studied the ultrasonic system of triangular blade applied on the mechanical arm, achieved the maximum amplitude 29um, and carried out the machining experiment of carbon fiber reinforced materials.

As for the shape design of special cutters used for ultrasonic-assisted cutting of honeycomb composites in China, finite element simulation is mainly applied in the design of cutter and ultrasonic horn. For instance, Shen Y B \[8\] applied the finite element analysis method to study the dynamic characteristics of sharp cutter, designed the sharp cutter for cutting honeycomb composites, and decided the geometric dimensioning of cutter in the best mode of vibration. Wu X \[9\] put forward the concept of substitution method and finite element analysis method, and designed ultrasonic straight-blade cutter based on the ultrasonic-assisted processing principle. Huang S \[10\] designed a disc cutter with sawtooth, built a cutting force model, and tested that sawtooth disc cutter could reduce the cutting force. Fang L \[11\] optimized the cutter by combining the statics and modal analysis results of disc cutter, made the structure of cutter compact, and reduced the motor power. Zhang Y D \[12\] et al. designed a broad-blade ultrasonic cutter, and concluded that slotting at a specific position of the cutter would realize the consistent lateral oscillation. Some scholars also optimized the design of cutter through the research on the processing technology. For instance, Hu X P \[13\] et al. built a theoretical model arguing that the cutting force of straight-blade cutter was related to such parameters as the cutting angle, depth, anterior angle of cutter, etc. which was of instructive to the design and optimization of acoustic system in ultrasonic-assisted cutting. Zhang S F \[14,15\] analyzed the impact of geometrical parameters of the disc cutter on the cutting temperature based on ABAQUS software, and built the ultrasonic vibration frequency model with the numerical fitting method. Zhuang Y \[16\] et al. argued that the key to the effective cutting of AFRP lay in the sharpness of cutting blade, designed the ultrasonic cutter and explored cutting parameters through the cutting experiment. Chen J Q \[17\] analyzed the wear pattern of the cutter during the machining process, conducted the software development based on the computer digital image, and implemented the real-time detection of the wear pattern of disc cutter.

In conclusion, domestic and foreign studies on the special disc cutter for the ultrasonic-assisted cutting of honeycomb composites mainly focus on the modal analysis of cutter and matching with the ultrasonic vibration system. Besides, technological tests mainly focus on the impact of machining parameters on the machining quality. But there are few studies on the influence law of geometrical parameters on the output amplitude and cutting effect. This paper carries out the modelling and simulation analysis of the harmonic response of ultrasonic disc cutter based on ANSYS Workbench software, concludes the laws of influence of geometrical parameters on the amplitude output of the ultrasonic vibration system, and analyzes the results from the energy density and structural rigidity.

2. Simulation Model of the Cutting Disc and Ultrasonic Horn

Ultrasonic horn is the core of the ultrasonic vibration system, which mainly magnifies the supersonic microvibration caused by the transducer till the amplitude meets the requirement of machining. Generally, when the shape of ultrasonic cutter is similar to the end of ultrasonic horn or the cutter is light in weight, the impact of cutter on the resonant frequency of ultrasonic vibration system can be ignored, and the ultrasonic horn can be optimized directly. But when the structure of the disc cutter used for the ultrasonic cutting of honeycomb composites differs greatly from the ultrasonic horn and it is heavy in weight, with a great impact on the structure of the ultrasonic vibration system, integrated design of the disc cutter and ultrasonic horn is necessary.

The disc cutter is mainly applied for facing or surface machining, a type of finish machining in the machining process of paper honeycomb. Specifically, it is in threaded connection with the ultrasonic horn, as shown in Figure 1.
Traditional ultrasonic vibration system is mainly designed by setting up a dynamic differential equation and solving the equation according to the boundary conditions to obtain all parameters. But once the disc cutter is integrated with the ultrasonic horn, it is complicated in structure, and it is also difficult to solve the equation. This paper mainly conducts the finite element analysis of the integrated structure of disc cutter and ultrasonic horn with ANSYS Workbench to study its dynamic characteristics, so that it shows better dynamic acoustic features at the resonant frequency. Besides, the geometric dimensioning of the cutter at the optimal inherent frequency and vibration mode is selected.

2.1. Simulation Model

According to the geometrical shape of existing ultrasonic horn and cutter, a finite element simulation model is set up, as shown in Figure 2. The key to the ultrasonic-assisted cutting of honeycomb composites lies in the smooth cutting of honeycomb fiber, and the rigidity of material in the cutting direction is small, so the blade of cutter shall be sharp, and the material of cutter shall show good machined characteristics. Moreover, the cutter is sheet-like, it is required to have good abrasive resistance. But since honeycomb material, in hallow cell-size structure, displays low harness, and the area of cut is small, it sets a low requirement on the hardness and heat-conducting property of the cutter. The principle of ultrasonic-assisted machining is to exert ultrasonic vibration on the cutter, so the material is required to display strong toughness, and endure long-term high-frequency vibration. At present, common cutter materials include high-speed steel, alloyed tool steel, cemented carbide, tool carbon steel, ceramics, etc. Since alloyed tool steel is more suitable for making low-speed cutting tool, while ceramic material is poor in toughness, and tool carbon steel is poor in abrasive resistance, and cemented carbide is fragile, featuring poor shock resistance and anti-vibration performance, high-speed steel is relatively suitable, and its material performance is shown in table 1.

| Material  | Elastic modulus | Density  | Poisson's ratio |
|-----------|----------------|----------|-----------------|
| W18Cr4V   | 2e11Pa         | 7800kg/m³| 0.3             |

2.2. Analysis of Vibration Characteristics

Related studies have shown that during the cutting of honeycomb composites with special ultrasonic disc cutter, the greater the amplitude of ultrasonic vibration at the blade is, the greater the role of ultrasonic-assisted cutting will be, namely the smaller the cutting force will be, and the better the cutting effect will be. For this reason, the mode of vibration of the greatest vibration amplitude at the blade should be selected as the target mode.

Integrated modal analysis of the cutter and ultrasonic horn is carried out with the modal modular in ANSYS Workbench. Different modes of vibration of the cutter at different resonant frequencies can be obtained. In general, the mode of vibration of the cutter consists of three types, namely the wave type, longitudinal vibration type and twist vibration, as shown in Figure 3.
Harmonic response simulation analysis is carried out for the cutter within 18 kHz~22 kHz by making use of the harmonic response module of ANSYS Workbench, and the curve of the mean value of longitudinal vibration amplitude and excitation frequency at the blade is obtained, as shown in Figure 4. It can be seen from the frequency-amplitude curve of the harmonic response analysis that the system reaches the maximum amplitude at point A near 20 kHz, and the mode of vibration corresponding to the modal analysis is shown in Figure 5.

According to the simulation result, in case of the same input, the vibration amplitude of resonance at the blade of longitudinal vibration is the greatest, and it is a mode of vibration featuring the highest energy conversion efficiency, as the target mode of vibration for subsequent designs.

As stated previously, under the ultrasonic assistance, the greater the vibration amplitude of the disc cutter at the blade is, the greater the role of ultrasonic-assisted cutting will be. This paper takes different geometrical parameters for the harmonic response analysis to get the mean vibration amplitude of the cutter at the blade in longitudinal resonance, and it also analyzes the laws of influence of different geometrical parameters on the vibration amplitude of the cutter.

This paper studies the laws of influence of the cutter thickness, anterior angle and blade radius on the vibration amplitude in longitudinal vibration mode with the single factor method. At first, the resonant frequency H of longitudinal vibration mode is obtained through the modal analysis. According to the shell size and material characteristics of the matched transducer of ultrasonic vibration system, 2 MPa of pressure is input, and the scope of harmonic response frequency is set to be (H-100, H+100) Hz, for solving the output amplitude of the ultrasonic vibration system at the blade in case of longitudinal resonance, and analyzing the change rules of output vibration amplitude in case of the same input.

3.1. Effect of Tool Radius R on the Amplitude
According to the requirement on the inner arc surface machining size and machining frequency of work piece, the radius of small disc cutter is decided to be about 15mm. When the cutter radius R ranges between 12 mm and 17 mm, it can be learnt from the modal analysis results that when the thickness b ranges between 0.6 mm and 1.3 mm, and the anterior angle γ0 between 11°and 17°, the resonant frequency of the vibration system is about 20 kHz, and it is matched with the transducer of the system.
This paper conducts the harmonic response analysis of the output vibration amplitude of the system within the choice range of parameters.

In Figure 6, it is the curve of influence of value on the vibration amplitude when the blade thickness $b$ is 0.9 mm, the anterior angle $\gamma_0$ is 13°, 15° and 17°, and radius $R$ ranges between 12 mm and 17mm.

![Figure 6. Curve chart of amplitude change with radius](image)

It can be seen from the three curves in the figure that when the radius ranges between 12 mm and 13.5 mm, with the increase of the radius, the output vibration amplitude increases from 25 $\mu$m to 35 $\mu$m. When the radius is over 13.5, the output vibration amplitude decreases gradually from 35 $\mu$m to 15 $\mu$m at a slower speed with the increase of the radius, and the curve tends to be flat. Generally, the maximum vibration amplitude shows when the radius ranges between 13 mm and 14 mm, within which the optimal radius of the vibration system occurs.

### 3.2. Effect of tool Thickness $b$ on the Amplitude

When the tool radius $R$ is 13 mm and the thickness $b$ of the tool is in the range of 0.6 mm to 1.2 mm, the overall longitudinal vibration resonance frequency of the vibration system is around 20 kHz. However, it is difficult to machine a tool with a thin thickness, and the rigidity of the tool will be too weak to guarantee stability when cutting honeycomb materials, the value of the tool thickness $b$ is set from 0.8 mm to 1.2 mm.

In Figure 7, it is the curve of influence of tool thickness $b$ on the vibration amplitude when the cutter radius $R$ is 13 mm, and anterior angle $\gamma_0$ is between 11° and 17°.

![Figure 7. Curve chart of amplitude change with thickness](image)

![Figure 8. Curve chart of amplitude change with anterior angle](image)

It can be seen from the curves in the figure that when the tool thickness ranges between 0.8 mm and 1.2 mm, with the increase of the thickness, the output vibration amplitude significant decreases from 40 $\mu$m to 25 $\mu$m. Therefore, the thickness of the tool should be taken as small as possible during design. Considering the tool machining process, tool stiffness and vibration effect, the tool thickness should be around 0.9 mm.
3.3. Effect of Anterior Angle $\gamma_0$ on the Amplitude

According to the section 2.2, the value of the tool thickness should not be too small. Therefore, in this section, the tool thickness is in the range of 0.8mm to 1.2mm and the tool radius $R$ is 13 mm. The influence of the anterior angle on the amplitude is simulated. The result is shown in Figure 8.

It can be seen from the curves that when the anterior angle ranges between 11° to 17°, with the increase of the anterior angle, the output vibration amplitude significant decreases. When the tool is thin, the change of the anterior angle will reduce the output amplitude value from 43 $\mu$m to 35 $\mu$m, which is a big change. When the tool is thick, the total variation of the output amplitude is about 2 $\mu$m as the angle of the rake angle changes. In this case, the change in the rake angle has little effect on the output amplitude.

Since the smaller the tool's anterior angle is, the sharper the tool is, which means more conducive to cutting the fiber. Therefore, when the tool thickness is taken to a larger value, the tool rake angle should be taken as small as possible while preserving the tool life.

4. Analysis of Simulation Results

The influences of the geometrical parameters on the vibration characteristics were analyzed from the energy density and bending rigidity.

4.1. Analysis of Energy Density

The energy accumulation principle of ultrasonic horn is: on the premise of ignoring the energy loss during the transmission, the energy remains the same when going through any cross section of the ultrasonic horn. Therefore, the expression of the energy density $\rho_0$ on the cross section of the sectional area $S_0$ is:

$$\rho_0 = E / S_0$$  \hspace{1cm} (1)

In which: $E$ is the sectional energy, which is a constant value.

It is clear in equation (1) that the smaller the sectional area is, the greater the energy density will be. The relational expression between the vibration amplitude $A$ and the energy density is:\hspace{1cm} (18):

$$\rho_0 = \frac{1}{2} K_e A^2$$  \hspace{1cm} (2)

Namely

$$A = \frac{2\rho_0}{K_e}$$  \hspace{1cm} (3)

In which: $K_e$ is a coefficient, related to the system structure.

According to equation (3), the greater the energy density is, the greater the vibration amplitude will be. To make the vibration amplitude transfer of the ultrasonic vibration system steady, the diameter at the joint of disc cutter is set as 10 mm according to the end size of the ultrasonic horn, while the total height of the cutter is set to be 2.7 mm according to the experience. A rectangular plane coordinate system is set for the disc cutter, as shown in Figure 9, in which, $b$ is the thickness of blade, $\gamma_0$ is the anterior angle, $R$ is the radius of cutter. The expression of two straight lines in the figure is:

Line 1:

$$y_1 = \tan \alpha (R - x_1)$$  \hspace{1cm} (4)

Line 2:

$$y_2 = 7.7 - \sqrt{2}b - x_2$$  \hspace{1cm} (5)

By ignoring the stepped surface, and taking the sectional area of the disc cutter within $y (0, b)$ into account, the sectional area equation of the cutter when $y = y_0$ is:

$$S_0 = \pi (x_1^2 - x_2^2) = \pi [(R - y_0 / \tan \alpha)^2 - (7.7 - \sqrt{2}b - y_0)^2]$$  \hspace{1cm} (6)
The change relation between geometrical parameters $R$, $b$, $γ_0$ and sectional area can be obtained according to equation (6), and the relation between geometrical parameters and the energy density and vibration amplitude of the cross section can be obtained according to equation (1) and (3), as shown in table 2. It is clear in table 2 that when only the energy density is taken into account and other factors are ignored, the energy density of the cross section would decrease with the increase of the radius $R$, thickness $b$ and anterior angle $γ_0$.

Table 2. Change relation between geometric parameters and energy density

| Geometric parameters | Increase or decrease | Sectional area | Energy density | Vibration amplitude |
|----------------------|----------------------|----------------|----------------|---------------------|
| Radius $R$           | ↑                    | ↑              | ↓              | ↓                  |
| Thickness $b$        | ↑                    | ↑              | ↓              | ↓                  |
| Anterior angle $γ_0$ | ↑                    | ↑              | ↓              | ↓                  |

4.2. Analysis of Bending Rigidity

When the energy density is the same, the greater the bending rigidity is, the smaller the vibration amplitude will be. Therefore, bending rigidity of the cutter is also one of the influence factors. This paper mainly measures the structural rigidity by calculating the deflection at the cross section of the disc cutter.

The cross section of the disc cutter is abstracted to be the cantilever beam in Fig 10. $q(x)$ is related to the energy density. During the calculation, $q(x)$ is regarded as a constant value. The elasticity modulus of the beam $E$ is the elasticity modulus of the cutter material; the inertia moment $I$ of the beam is not a constant value in practical conditions, but it is regarded as a constant value for simplifying the calculation.

The equation of bending moment is:

$$M(x) = \int q(x)dx = \int q \cdot xdx = \frac{1}{2}qx^2 \quad (0 \leq x \leq R)$$

(7)

The differential equation obtained is:

$$\frac{d^2\omega}{dx^2} = \frac{M(x)}{EI} \Rightarrow EI\omega'' = \frac{1}{2}qx^2$$

(8)

According to the first integration:

$$EI\omega' = EI\theta = qx^3/6 + C$$

(9)

According to the double integration:

$$EI\omega = qx^4/24 + Cx + D$$

(10)

According to the boundary conditions, $x=0$, $θ=0$ are replaced in equation (8), and it can be worked out that $C=0$; $x=0$, $ω=0$ are replaced in equation (9), and $D=0$. $C$ and $D$ are replaced in equation (10), the deflection curve equation can be obtained.
The maximum deflection at the blade is:

$$\omega = \frac{q x^4}{24EI}$$

(11)

It can be seen from equation (12) that the greater the radius $R$ is, the greater the deflection at the blade will be, namely the greater the vibration amplitude will be. When the thickness $b$ increases, the equivalent inertia moment of the structure $I$ increases, and the deflection at the blade and the vibration amplitude decrease. When the anterior angle increases, the equivalent inertia moment increases, while the deflection decreases. Therefore, the relation between geometrical parameters of the cutter and the inertia moment, and deflection at the blade can be obtained, as shown in table 3.

| Geometric parameters | Increase or decrease | Inertia moment | Deflection | Vibration amplitude |
|----------------------|----------------------|----------------|------------|--------------------|
| Radius $R$           | ↑                    | →              | ↑          | ↑                  |
| Thickness $b$        | ↑                    | ↑              | ↓          | ↓                  |
| Anterior angle $\gamma_0$ | ↑                  | ↑              | ↓          | ↓                  |

According to the equations in table 2 and table 3, it can be seen that with the increase of radius, the energy density decreases, but the rigidity becomes weaker. Therefore, there would be the choice range of optimal radius. With the increase of cutter thickness, the energy density decreases, and the rigidity becomes stronger, which would decrease the vibration amplitude at the blade. With the increase of the anterior angle, the energy density decreases, and the rigidity becomes stronger, so it would also decrease the vibration amplitude at the blade.

4.3. Analysis of Simulation Results

By analyzing the simulation results in Section 3 from two aspects of energy density and bending rigidity, it can be seen that when the radius is less than 13.5 mm, although the energy density at is high, the overall rigidity of the tool is strong and the vibration effect is suppressed, resulting in a small amplitude value. Conversely, when the tool radius is larger than 13.5 mm, the energy density reduced too much, also resulting in a smaller amplitude value. This means the vibration system has an ideal radius interval to maximize the amplitude value, and the parameter value of this interval should be preferentially selected in the design parameter selection. For thickness and anterior angle, as the value of the parameter increases, the energy density of the tool decreases while the bending rigidity increases, which causes the amplitude to decrease. These two parameters should take as small a value as possible within the allowable range. To design the geometric parameters of the cutting disc, the influences of the geometric parameters of the tool on the energy density and the tool stiffness should be considered comprehensively to maximize the conversion efficiency of the tool.

5. Conclusion

1) A finite element model of the disc cutter and amplitude transformer was established, and modal analysis and harmonic response analysis were carried out. Characteristics of vibration amplitude at the blade of the integrated structure with the frequency were obtained. The simulation results shown that the maximum mean longitudinal vibration amplitude could be achieved for the integrated structure, and longitudinal vibration mode can be used in the ultrasonic-assisted cutting.

2) Harmonic analysis results shown the vibration amplitude increased firstly and decreased later with the increase of radius $R$, and the ideal values of the vibration amplitude was within 13mm and 14mm, the vibration amplitude decreased with the increase of cutter thickness, and within 0.8mm and 1mm, the cutter shown better stability in case of great vibration amplitude. The vibration amplitude decreased with the increase of anterior angle, the smaller the anterior angle the better within enough bending rigidity.
3) From the aspects of energy density and bending rigidity, the increase of the tool radius would reduce the energy density and decrease the bending rigidity, so there existed an optimized value range for the tool radius. The increase of the tool thickness and the anterior angle would reduce the energy density and increase the bending rigidity, which could increase the output amplitude.

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