Research Article

On Design of Through-Hole Structure Inspired by Vascular Bundle of Bamboo for Multiple Compression Load Cases

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In this paper, the multiangle’s crashworthiness of the bionic tubes inspired by the cylindrical fibers of bamboo vascular bundles as well as the through-hole structure are studied. A finite element model validated by the theoretical analysis and drop hammer impact experiments is used to assess the optimal core distribution ratio of the bionic tube at 0°, 10°, 20°, and 30° impacts. Finally the effects of the number of circular core and wall thickness on crashworthiness are investigated. The results show that the bionic tube has better crashworthiness at small angle impacts compared to the square tubes, and l14r6 is obtained as the optimum bionic tube: the SEA at small angles improves by 12.72% (0°), 14.91% (10°), and 18.12% (20°), respectively; however, the SEA at 30° impact decreases by 2.53%. As the number of circular core increases, the crashworthiness becomes more significant, but the deformation mode becomes worse. The SEA of the tube with 3 cores increases by an average of 1.85 times (0°), 1.89 times (10°), 1.34 times (20°), and 1.41 times (30°) for each impact condition, respectively, compared to the tube with single core.

1. Introduction

Thin-walled tubes as energy-absorbing structures [1] usually need to be lightweight [2] while satisfying high-energy absorption, so aluminum alloy materials [3] are widely used for the processing of tubes. The crashworthiness of thin-walled tubes is strongly influenced by the shape [4–6]. Guillow et al. [7] investigated the effect of the ratio of diameter to thickness (D/t) and the ratio of length to thickness (L/t) on the loading-bearing characteristics of circular tubes, respectively, and plotted the distribution of deformation modes. Meanwhile, comprehensive crashworthiness under multiole impacts is necessary for the tubes designed, as real-world collisions do not usually occur entirely in the axial direction and are often accompanied by the oblique impacts. Therefore, the crashworthiness and deformation mode of thin-walled tubes under oblique impact conditions are receiving increasing attention. Han and Park [8] conducted a numerical study of the oblique impact of square tube and showed that the energy absorption performance of square tube under the oblique impact was significantly lower than that of an axial impact. Reyes et al. [9] investigated the effect of wall thickness and material on the oblique impact of tubes. Ahmad et al. [10] analyzed the crashworthiness between empty tubes and foam-filled conical tubes under the oblique impact by experiment. In addition, the crashworthiness of multiole tubes obtained by combining the same or different shapes of the same mass has been shown to have superior energy absorption [11–13]. The deformation modes and crashworthiness of the multiole tubes are influenced by the number of cells and wall thickness [14–16].

Structures inspired by nature offer a significant improvement over conventional structures in terms of crashworthiness [17]. Bamboo is widely used [18, 19] because of its low density and good mechanical properties [20], with a higher stiffness-to-mass ratio than steel, in addition to excellent tensile [21–23], compressive [24, 25], and bending strength [26]. Song et al. [27] proposed a biomimetic step-thickness thin-walled structure based on the structural characteristics of bamboo with gradient wall thickness, and verified the
effectiveness of the biomimetic design by quasi-static compression experiments under the oblique loads. Zou et al. [28] did an experimental investigation on the impact resistance of bamboo structures and proposed a thin-walled tube based on the spatial distribution pattern of the number of vascular bundles.

In this paper, with reference to the microstructure of the vascular bundles, a thin-walled tube is designed based on the cylindrical fiber bundles as well as through-hole structure inspiration. First, the crashworthiness of the bionic structure is analyzed experimentally and theoretically, and the accuracy of the finite element model (FEM) is verified. Then, the optimum structural ratio of bionic unit on crashworthiness is found by the finite element method based on the evaluation index under multiangle impact conditions. Finally, the effects of the number of bionic unit and wall thickness on crashworthiness are investigated based on the optimum ratio.

2. Bionic Design

2.1. Structural Characteristics of Bamboo Central Through-Hole. As shown in Figures 1(a)–1(e), the presence of a through-hole in the center of each cylindrical fiber of the bamboo vascular bundle is observed by Micro-CT. The through-hole structure enhances the crashworthiness of the bamboo while making the structure lightweight. Therefore, in order to improve the crashworthiness of thin-walled tubes, a bionic structure imitating the through-hole feature of bamboo fibers is proposed.

Inspired by the structural features of the vascular fiber through-hole, and after the evolution of Figure 2 to obtain the bionic unit. In order to investigate the effect of the above simplified structure on the structural crashworthiness performance, as well as to explore the spatially optimal allocation ratio of the circular radius \( r_i \) and the length of reinforcements \( l_i \), this paper carried out comparative analysis of 14 allocation ratio designed tubes through the structural crashworthiness evaluation index.

2.2. Model Structural Parameters. The radius range of the circular is taken as 4–16 mm in this study, considering that too small radius of the circular is not conducive to practical machining. The design ensures that the equal material consumption of the 14 tube core units for comparison. Taking \( l_{10}r_{10} \), which has approximately the same area of each cells, as the benchmark, and setting its thickness \( t_{10} \) as 1.0 mm, then the \( t_i \) can be expressed as follows:

\[
t_i = \frac{m_{10}}{(4l_i + 2\pi r_i)H\rho},
\]

where \( m_{10} \) is the core unit mass of \( l_{10}r_{10} \). \( H \) is the height of the tube and \( \rho \) is the density. The calculated results of each tube wall thickness are shown in Figure 3. The \( r_i \) decreases
linearly with $l_i$, while the core wall thickness $t_i$ increases nonlinearly with $l_i$.

In this paper, cases both axial impact and oblique impact are considered, and the proposed bionic tubes are studied comprehensively. It is hoped that the optimal size combination of $l_i r_i$ can be obtained after comparative analysis. The analysis condition and comparison structure are shown in Figure 4. The thin-walled tube with outer wall side length of 40.0 mm and height of 80.0 mm is fixed on the rigid bottom plate and compressed by a moving rigid plate with an initial velocity of 10.0 m/s. The crushing angle between the rigid plate and the sample varies from 0° to 30°.

2.3. Evaluation Index. In order to evaluate the crashworthiness of the bionic tubes under various impact conditions, some evaluation indexes are introduced.

The energy absorption (EA) denotes the total energy absorption during crushing and can be formulated as follows:

$$EA = \int_0^d F_x \, dx,$$  \hspace{1cm} (2)

where $d$ is given crushing displacement and $F_x$ is the instantaneous crushing force in Equation (2). PCF represents the peak crushing force. SEA represents the energy absorbed per unit mass of the structure. The mean crush force (MCF) is the ratio of the total EA to the corresponding crushing displacement $d$, as follow:

$$SEA = \frac{EA}{m_i},$$  \hspace{1cm} (3)
where \( m_i \) is the mass of tube.

\[
MCF = \frac{EA_i}{d},
\]

(CFE) represents the crushing force efficiency:

\[
CFE = \frac{MCF}{PCF},
\]

\( \omega \) represents the platform volatility coefficient:

\[
\omega = \frac{F_{\text{max}} - F_{\text{min}}}{MCF}.
\]

We put forward a reasonable calculation method based on the probability of collision accidents at various angles of the vehicle and considering the above crashworthiness indexes, so as to determine the optimal structure ratio. Among the collision accidents occurring in all directions, the proportion of frontal collision is 39\%, among which the proportion of frontal nonangle collision is 16\%, and the proportion of frontal tilted collision is 23\% [29]. The tilted angle is set as 10\°, 20\°, and 30\° in this study, respectively.

Among the crashworthiness indexes, SEA and MCF represent the energy absorption performance and load-bearing crashworthiness of the structure most intuitively from the definition, and their evaluation characteristics are that the larger the value, the better the mechanical performance of the structure. Therefore, the calculation formula of the optimal structure is defined as follows:

\[
l_{10r_{10}}^{\text{opt}} = \frac{16\%}{39\%} (\text{SEA}_{\psi} + \text{MCF}_{\psi}) + \frac{23\%}{39\%} \times \frac{1}{3} \left( \text{SEA}_{i_{\psi}} + \text{MCF}_{i_{\psi}} \right).
\]

2.4. Analysis of Mesh Convergence. The material used here is the AA6061T4 and the material failure of aluminum alloy tube is not considered [30, 31]. Mechanical properties: mass density \( \rho \) is 2,700 kg/m\(^3\), Young’s modulus \( E \) is 70.0 GPa, initial yield stress \( \sigma_y \) is 114.34 MPa, the ultimate stress \( \sigma_u \) is 257.83 MPa, and the Poisson’s ratio \( \mu \) is 0.31 [32]. The stress–strain curve tested is shown in Figure 5.

In order to determine the appropriate element size, convergence analysis is performed in this section, with the friction coefficient of each contact surface set to 0.2. Figure 6 shows the convergence behavior of \( l_{10r_{10}} \) with different element sizes. It can be seen that the energy absorption gradually converges when the element size is reduced to 1.0 mm. Therefore, the element size used in this paper is 1.0 mm.

3. Verification of Finite Element Model

3.1. Test Verification. The tubes codenamed \( l_{15r_{15}}, l_{10r_{10}}, l_{8r_{8}} \), and \( l_{20r_{0}} \) are manufactured by CNC wire cutting technology as shown in Figure 7. Their masses are 51.6, 51.5, 51.3, and 51.4 g, respectively, which are approximately equal and showed that the machining accuracy of 0.05 meets the requirements and the mass control of the FEM is 56.76 g. The large mass of the reason for FEM is due to the overlapping of materials at the panel joints. The maximum drop height of the impact tester (Figure 8) is 6.6 m. In the experiment, the weight of the hammer is set at 120 kg and the height at 2 m. The calculated impact velocity of the drop hammer is 6.26 m/s.

Figure 9 shows the deformation of \( l_{15r_{15}}, l_{10r_{10}}, l_{8r_{8}}, \) and \( l_{20r_{0}} \) during the impact experiment. The deformation of the specimens is progressive folding, which is close to the FEM results (Figure 10 (0°)). As can be seen from Figure 11, \( l_{15r_{15}} \) (Figure 11(a)), \( l_{10r_{10}} \) (Figure 11(b)), \( l_{8r_{8}} \) (Figure 11(c)), and \( l_{20r_{0}} \) (Figure 11(d)) show good agreement between the load and displacement curves under experiment and simulation, while the simulation results in a higher peak load, which is
3.2. Theoretical Analysis of Energy-Absorbing Structures. The EA during compression process is the sum of the bending energy \( E_b \) and membrane energy \( E_m \) \[33\]. That is:
\[
EA = 2\mu_e H_w P_m = E_b + E_m, \tag{8}
\]
where \( \mu_e \) is the effective crushing distance coefficient, \( \mu_e \) is taken as 0.75 in this study. \( H_w \) is the wavelength of the fold obtained after compression. \( P_m \) is MCF during the compression process.

\( E_b \) is expressed as follows:
\[
E_b = 2\pi L_c M_0, \tag{9}
\]
where \( M_0 \) is the fully plastic bending moment, \( M_0 = 0.25 \sigma_0 t^2 \), \( \sigma_0 \) is the flow stress of material, \( \sigma_0 = (\sigma_y + \sigma_u)/2 \), where \( \sigma_y \) and \( \sigma_u \) represents the yield strength and ultimate strength, respectively. \( L_c \) is the edge length, and the \( E_b \) of three corner (Figure 12(a)) elements can be expressed as follows:
\[
E_{bi} = \begin{cases} 
2\pi M_0 \left( \frac{1}{4} + \frac{1}{2} \right) & (I) \\
2\pi \left( \frac{L}{2} + \frac{L - 2r}{4} \right) M_0 & (II), \\
2\pi M_0 \left( \frac{L - 2r}{4} + \frac{\sigma_y}{2} \right) & (III)
\end{cases} \tag{10}
\]
where \( r_i \) denotes the value of the radius of the circle corresponding to the circular arc. \( L \) is the length of the outer wall edge. Therefore, the total energy absorbed by the material bending deformation is 

\[
E_b = N_l E_{bI} + N_{II} E_{bII} + N_{III} E_{bIII},
\]

where \( N_l \) is the number of the corner element.

The membrane energy \( E_m \) is closely related to the shape of the different corner elements. Zhang et al. [13] analyzed \( E_m \) by a single corner element (I) during crushing at one wavelength as follows:

\[
E_{mbi} = 2 \sigma_0 t_i H_w^2 \left( \frac{2L - 2r_i}{3L - 2r_i} \frac{M_{s0}}{t_i} + \frac{L - 2r_i}{3L - 2r_i} \frac{M_{w0}}{t_i} \right) \left( 1 + 2 \tan \left( \frac{\theta_i}{2} \right) \right).
\]

For the single-corner element (III) with an arc unit, we take the arc AB equivalent into a line segment AB (Figure 12(b)). Then the corner element (III) can then be regarded as a symmetric three-panel corner unit [35], and considering the effect of the central corner on the compressive strength of the corner unit, the membrane energy absorbed by the single corner element (III) during one wavelength crushing is:

\[
E_{mci} = 4 M_{s0} H_w^2 \left( \frac{\tan \Phi}{\tan \Phi + 0.05/\tan \Phi} \right) \left( 1 + 2 \tan \left( \frac{\Phi}{2} \right) \right).
\]

The membrane energy absorbed by the single-corner element \( d \) in one wavelength crushing is:

\[
E_{md} = 4 \sigma_i t_i H_w^2 \left( \frac{16 M_{s0} H_w^2}{t_i} \right).
\]
FIGURE 11: Comparison of force–displacement between experiment and simulation: (a) $l_r15$, (b) $l_r10$, (c) $l_r6$, and (d) $l_r0$.

TABLE 1: Comparison of PCF(kN), MCF(kN) and SEA (J/g) between experiment and simulation.

<table>
<thead>
<tr>
<th></th>
<th>PCF_Exp</th>
<th>PCF_Sim</th>
<th>Error</th>
<th>MCF_Exp</th>
<th>MCF_Sim</th>
<th>Error</th>
<th>SEA_ Exp</th>
<th>SEA_Sim</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>$l_r15$</td>
<td>42.80</td>
<td>43.35</td>
<td>1.27%</td>
<td>35.83</td>
<td>32.51</td>
<td>9.27%</td>
<td>33.71</td>
<td>34.37</td>
<td>1.92%</td>
</tr>
<tr>
<td>$l_r10$</td>
<td>47.74</td>
<td>49.92</td>
<td>4.37%</td>
<td>37.20</td>
<td>34.62</td>
<td>6.94%</td>
<td>34.28</td>
<td>36.60</td>
<td>6.34%</td>
</tr>
<tr>
<td>$l_r6$</td>
<td>49.65</td>
<td>51.06</td>
<td>2.76%</td>
<td>38.60</td>
<td>36.51</td>
<td>5.41%</td>
<td>36.40</td>
<td>38.60</td>
<td>5.70%</td>
</tr>
<tr>
<td>$l_r0$</td>
<td>48.72</td>
<td>51.82</td>
<td>5.98%</td>
<td>35.74</td>
<td>32.39</td>
<td>9.37%</td>
<td>33.53</td>
<td>34.24</td>
<td>2.07%</td>
</tr>
</tbody>
</table>
The mean crushing force can be expressed as follows:

\[
\text{EA}_i = \begin{cases} 
2\pi \left[ 4L \mu_i + (2L + (2\pi - 4)\tau_i)\mu_i \right] + \\
16 \left[ \frac{6L}{3L - 2\tau_i} + 1 \right] \frac{M_{\mu_i}}{t_i} + \left( \frac{3(L - 2\tau_i)}{3L - 2\tau_i} + 4.09 \right) \frac{M_{\mu_i}}{t_i} \times H_w \left( 4 \leq \tau_i \leq 16 \right) \\
2\pi (4L \mu_i + 2L \mu_i) + 16 \left( \frac{M_{\mu_i}}{t_i} + 2 \frac{M_{\mu_i}}{t_i} \right) \times H_w^2 \left( r = 0 \right)
\end{cases}
\]

For \( t_i = 1.00 \text{ mm}, \ M_{\mu_i} = M_{\mu_i} t_i^2 \), therefore, the average crushing force can be expressed as follows:

\[
\text{P}_m = \begin{cases} 
\frac{4}{3} \pi \mu_i \left[ 4L + (2L + (2\pi - 4)\tau_i)\mu_i \right] \times \frac{1}{H_w} + \\
\frac{32}{3} \mu_i \left[ \frac{6L}{3L - 2\tau_i} + 1 \right] + \left( \frac{3(L - 2\tau_i)}{3L - 2\tau_i} + 4.09 \right) \times t_i \times H_w \left( 4 \leq \tau_i \leq 16 \right) \\
\frac{4}{3} \pi (4L \mu_i + 2L \mu_i) \times \frac{1}{H_w} + \frac{32}{3} \left( \frac{M_{\mu_i}}{t_i} + 2 \frac{M_{\mu_i}}{t_i} \right) \times H_w \left( r = 0 \right)
\end{cases}
\]

Then \( H_w \) can be obtained from \( \frac{\partial \text{P}_m}{\partial H} = 0 \):

\[
H_w = \sqrt{\frac{8 \times \frac{4L + (2L + (2\pi - 4)\tau_i)\mu_i}{6L \left( 3L - 2\tau_i \right) + 1} + \left( \frac{3(L - 2\tau_i)}{3L - 2\tau_i} + 4.09 \right) \times t_i}{3}} \times H_w \left( 4 \leq \tau_i \leq 16 \right)
\]

Substituting into the Equation (17), and introduce a dynamic enhancement factor of 1.12 to balance the inertia effect [13], the mean crushing force \( \text{P}_m \) of the design element in the compression process is obtained as follows:

\[
1.12 \times \frac{16\sqrt{2\pi} \mu_i}{3} \sqrt{\frac{4L + (2L + (2\pi - 4)\tau_i)\mu_i}{6L \left( 3L - 2\tau_i \right) + 1} + \left( \frac{3(L - 2\tau_i)}{3L - 2\tau_i} + 4.09 \right) \times t_i} \left( 4 \leq \tau_i \leq 16 \right)
\]

\[
1.12 \times \frac{16\sqrt{2\pi} \mu_i}{3} \sqrt{\frac{4L + (2\tau_i^2) + 2\tau_i + 3}{\left( 3 + 2\tau_i \right)} \times H_w} \left( r = 0 \right)
\]
The response surface method is applied to analyze the effect of the radius of the circular core $r$ and the thickness of the core $t$ on the MCF (or $P_m$) of the bionic tube in Equation (19), as shown in Figure 13. From the response surface, the MCF decreases as the radius $r$ decreases when the thickness $t$ is equal. In this study, the thickness of the inner core unit is related to the radius of the circular core $r$. Bringing Equation (1) into Equation (19), it is found that the only factor affecting the MCF is the radius of the circular core $r$. Therefore, a one-factor two-dimensional curve of the theoretical calculated value of MCF is obtained, as shown in Figure 14. The figure also compares the results of the axial impact simulation in the next section, with an average error of 2.09%.

Therefore, this section is able to verify the accuracy of the FEM through impact experiments and theoretical analysis, which can be used for subsequent finite element parametric analysis under multiangle impact conditions.

4. Results of Simulation

4.1. Multiangle Compression Simulation. Figure 10 compares the deformation modes of different ratio tubes under axial (0°) and oblique (10°, 20°, and 30°) impact, respectively, the top row of tubes is $r_{16} - r_{10}$ from left to right and $r_9 - r_0$ for the bottom row of tubes. There are differences in the deformation modes of the thin-walled tubes at different impact angles due to the differences in the dimensions of the inner units. Specifically, under axial impact, when $r_c$ is large, the deformation mode of the inner core unit is irregular because more folds and concavities in the circular core cylinder (red box), while as $r_c$ decreases, the deformation mode improved. At oblique angles of 10° and 20°, the deformations show a progressively more compact folded deformation as the $t/r_c$ ratio increases, but at angle of 30°, the tubes show a progressively more bent deformation mode as a whole. At 10° oblique impact, the $l_4 r_{16} - l_4 r_{12}$ tubes (red frame) only forms twofold and the lower part of the tube is not strong enough to support the upper deformation and bends. This bending phenomenon due to poor support is also seen in the $l_4 r_{16} - l_4 r_{13}$ tubes (red box) at 20° oblique impact.

The difference is that the larger circular core units form folding folds in the upper part of the tube together with the fascia, showing the potential for greater energy absorption, whereas the circular core units of the $l_1 r_9 - l_1 r_5$ tubes (red frame) do not appear to fold and absorb energy during impact.
Figure 15 shows the comparison of the load displacement relationship for 14 tubes under multiangle impact. As shown in Figure 15(a), the initial peak load gradually increases with decreasing radius of the circular element during axial impact. The trend of load variation before the compression displacement of 12.5 mm (the first fold formation) is the same, and when the radius of the circular element ≥10 mm, the load bearing curve of the tube shows a clear cascade of increasing...
cases as the radius decreases. When the radius ≤10 mm, the load curves are crossed and overlapped. The common tube \( l_{20\theta_0} \) exhibits a lower loading capacity throughout the compression process, \( \text{MCF}_{l_{r_0}} \) is 32.39 kN, which is close to \( \text{MCF}_{l_{r_13}} \), while \( \text{MCF}_{l_{r_14}} \) has a maximum of 36.51 kN, which is 12.72% higher than the common tube, which indicates that the presence of the central circular element can change the loading capacity of the tube, and it can be well improved when the radius ≤12 mm. The trend of the load curves of 14 kinds of tubes shows a radius of 9 mm as the limit, when \( r \leq 9 \text{ mm} \), the load fluctuation is more violent with the increase of compression displacement, indicating that the loading capacity of this part of the tubes are not stable. The initial peak load shows a trend of gradually increasing and then decreasing as the radius of the circular element decreases when the impact angle reaches 10° (Figure 15(b)). And the initial peak load of \( l_{10\theta_0} \) reaches a maximum of 38.85 kN. \( l_{20\theta_0} \) shows a lower loading capacity, \( \text{MCF}_{l_{r_5}} \) is only 27.88 kN, which is 82.44% of \( \text{MCF}_{l_{r_0}} \) (maximum 33.82 kN). The load curves show a significantly larger second peak with increasing compression displacement when the radius is below 12 mm at impact angle of 20° (Figure 15(c)). The analysis concludes that the peak of load fluctuation is generally accompanied by the generation of deformation folds, the folds will extend to the inside, near the circular element as the tilt compression proceeds, which indicates that the fold deformation of the circular element will absorb more energy, once again proving that the presence of the circular element enhances the loading capacity of the tube. The load curve of each tube has an obvious peak when the compression displacement is about 23 mm at impact angle of 30° (Figure 15(d)), where rigid compression plate just touch the last upper edge of the outer wall of the tube by the compression angle of the tangent function, so the force of the tube will become a large angle of oblique push when the outer edge of the tube is completely compressed, the loading capacity will gradually reduce. Along this line of thought, it means that the overall loading capacity of the tube is insufficient if the peak load occurs before the compression displacement of 23 mm, the rigid plate in the case of not yet completely compressed the edge of the tube, there is a fold underneath causing the tube to bend; the opposite means that the tube has a strong loading capacity.

4.2. Determination of the Optimal Ratio. Figures 16(a), 16(c), 16(e), and 16(g) show the comparison of crashworthiness of each tube under multiangle impact, whose indicators are related to energy absorption and passive safety, respectively. Figures 16(b), 16(d), 16(f), and 16(h) show the comparison of MCF which represents the compressive capacity, and CFE and \( \omega \) represent the compressive stability.

The PCF and SEA of the bionic tube increase with decreasing \( r \) under axial impact. The SEA of the \( l_{1.4_{r_0}} \) tube is the largest (Figure 10 in green box), with an increase of 12.72% and a decrease of 1.48% in PCF compared to the square tube. At 10° impact angle, the SEA and PCF of the tubes show increase and then decrease, the maximum SEA of \( l_{1.4_{r_0}} \) (green box) is 35.75 J/g, which is 21.27% higher than that of the square tube \( l_{20_{r_0}} \). A stable change at 20° impact angle of SEA and PCF, with the maximum SEA of \( l_{1.4_{r_0}} \) (green box) being 32.04 J/g, an increase of 18.12% compared to \( l_{20_{r_0}} \), and its PCF being 36.81 kN, 1.23 times that of \( l_{20_{r_0}} \). At an impact angle of 30°, the SEA and PCF of the tubes show a wave trend, with the maximum SEA for \( l_{1.0_{r_0}} \) (green box) being 21.12 J/g, an improvement of 5.16% compared to \( l_{20_{r_0}} \), the PCF being 32.52 kN, 1.18 times that of \( l_{20_{r_0}} \). It can be seen that at 30° or large angular impacts, the circular core unit does not contribute much to the improvement in energy absorption as it does not undergo large compact fold deformations. It is worth noting that the thin-walled tubes \( l_{4_{r_0}}-l_{12_{r_0}}(10^\circ) \), \( l_{6_{r_0}}-l_{12_{r_0}}(20^\circ) \), and \( l_{1.0_{r_0}} \) have smaller CFE and larger \( \omega \) at the corresponding impact angles, indicating poorer loading-bearing stability as well. The rest of the tubes are able to control the crushing force efficiency CFE in the range of 0.85–0.89 (10°), 0.80–0.86 (20°), and 0.57–0.66 (30°) for each oblique impact, and the platform fluctuation factor \( \omega \) in the range of 0.16–0.26 (10°), 0.18–0.37 (20°), and 0.67–0.98 (30°).

Summing up the above analysis of the data results and the calculation method of Equation (7), we obtained \( l_{1.4_{r_0}} \) as the optimal bionic tube.

5. Crashworthiness Parameters Study

In summary, it can be proved that the combination of circular element and reinforcement improves the crashworthiness of the bionic tube through the simulation analysis. And the optimal allocation ratio of dimensions \( l: r = 7: 3 \) is obtained. Therefore, this section further investigates the effect of the number of circular core and thickness factors on the crashworthiness of multiangle impacts, based on the optimum ratio.

5.1. Effect of the Number of Circular Element. In order to explore the effect of the number of circular element on the crashworthiness of tubes, tubes with the number of circular element of 1, \( 2^\circ \), \( 3^\circ \), \( 4^\circ \), \( 5^\circ \), and \( 6^\circ \) are designed in Figure 17 (a). The calculated parameters for each dimension of the six tubes using the controlled mass method are shown in Figure 17(b).

It is clear from Figure 18 that the loading capacity of the tube still increases gradually with the increase of the number of circular element under the constant total mass. The load curves of \( 1^2 \) and \( 2^2 \) specimens are nearly flat, while the load curves show significantly large peaks when the number of circular element \( \geq 3^2 \). We find that \( 1^2, 2^2 \), and \( 3^2 \) specimens show progressive folding deformation during the compression combined with Figure 19, while when the number of circular element \( \geq 4^2 \), the internal stiffness increases due to the increase of hinging within the unit area, resulting in bending deformation of the tube from the middle position, where \( 4^2 \) and \( 5^2 \) specimens show progressive folding deformation again at the upper until the end of compression. The \( 6^2 \) specimen continues to bend at the point where the bending occurs and shows signs of collapse overall. In general, this
FIGURE 16: Continued.
uncontrolled deformation from the middle of the bend is not allowed in the engineering applications.

Figure 20 shows the comparison of the axial crashworthiness of the tubes with increasing number of circular cores under the same mass. From Figure 20(a), we can see that the SEA and PCF tend to increase as the number of circular core unit increases, but the only decrease occurs when the number is 42, probably due to the deformation mode of bending from the middle and the lack of internal compression, which leads to a lower energy absorption capacity. 62 specimens have the highest SEA, which is 1.55 times higher than 12 specimens but the PCF also increases by 1.32 times. The SEA of the 32 specimen is 1.37 times higher than that of the 12 specimen and the PCF is increased by a factor of 1.20. From Figure 20(b) we can see that the 12 specimen has the smallest CFE and the largest ω, indicating the lowest pressure stability, while the 22 specimen has the largest CFE and the smallest ω, indicating the most stable loading-bearing capacity.

The axial crashworthiness tends to increase as the number of circular core increases, but in terms of deformation, it does not show a positive correlation to the ideal deformation. In general, the uncontrolled deformation of energy-absorbing tubes with circular core’s number $\geq 42$ bending from the middle is not allowed in engineering applications, and their thickness of $<0.5$ mm is already very small, and if the thickness is increased, the deformation will only get worse, because it means that there will be more compact crushing, and the worse deformation mode under axial impact means worse overall collapse of the oblique impact. In contrast, the deformation of tubes with the number of circular unit number $\leq 32$ show the ideal progressive collapse from one end. Therefore, we only investigate the crashworthiness of tubes with circular core number 12, 22, and 32 for multiangle impact conditions with differences in thickness in the next section.

5.2. Effect of Thickness. Table 2 shows the mass information for each tube for differences in thickness and number of cores. The tubes are identical for the multiangle impact conditions and the impact angles are also set to 0°, 10°, 20°, and 30°, respectively.

<table>
<thead>
<tr>
<th>Numbers</th>
<th>$l (\text{mm})$</th>
<th>$r (\text{mm})$</th>
<th>$n (\text{mm})$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1²</td>
<td>14.00</td>
<td>6.00</td>
<td>1.03</td>
</tr>
<tr>
<td>2²</td>
<td>8.48</td>
<td>3.64</td>
<td>0.74</td>
</tr>
<tr>
<td>3²</td>
<td>6.09</td>
<td>2.61</td>
<td>0.58</td>
</tr>
<tr>
<td>4²</td>
<td>4.75</td>
<td>2.03</td>
<td>0.47</td>
</tr>
<tr>
<td>5²</td>
<td>3.89</td>
<td>1.67</td>
<td>0.40</td>
</tr>
<tr>
<td>6²</td>
<td>3.29</td>
<td>1.41</td>
<td>0.35</td>
</tr>
</tbody>
</table>

Figure 16: Comparison of SEA, PCF, MCF, CFE, and $\omega$ under multiangle impact: (a and b) axial impact; (c and d) 10° impact; (e and f) 20° impact; (g and h) 30° impact.

Figure 17: (a) Distribution of circular elements and (b) dimensional parameters.
FIGURE 18: Comparison of load–displacement curves under the different number of circular element.

FIGURE 19: Deformation modes under the different number of circular element (a–f).
Figure 21(a)–21(d) shows the variation of the force–displacement curves of each tube with the number of circular cores $N$ and the compression displacement $D$, respectively. It can be seen that the load curves of the tubes increase gradually, no matter the increase in the number of circular cores at equal thickness or the increase in thickness at the same number of cores. The load curves for each impact angle follow the same trend and are analyzed in the following text in conjunction with the compressional deformation mode (Figures 22–23):

The load curves under axial impact (Figure 21(a)) and $10^\circ$ impact (Figure 21(b)) show a wave change. However, some of the tubes show more dramatic wave changes, and for the purpose of the quoted analysis, a tube with a thickness of 1.0 mm and a number of circular cores of $2^2$ is named $2^2.1.0$. We find that the load curves for $0^\circ.2^2.1.0$, $0^\circ.3^2.0.7–1.0$, and $10^\circ.2^2.1.0$, $10^\circ.3^2.0.8–1.0$ fluctuate are even more dramatic. It is noteworthy that these are the very same tubes with poor deformation modes marked in red in Figures 22 and 24, some exhibiting deformation where the ends are folded and squeezed and then bent in the middle, and others where the deformation starts in the middle, with similar deformation greatly abating the potential of the circular core unit to carry loading and absorb energy. In contrast, in the progressive folding and deformation tubes, the circular core unit is fully compressed to maximize EA.

The change in the load curves for $20^\circ$ impact (Figure 21(c)) and $30^\circ$ impact (Figure 21(d)) are not the same as for a small angle impact, where the load curve shows an increase followed by a decrease, except that $20^\circ.1^2$ still shows a wave change. Of course, we can also explain the load variation in terms of deformation, as shown in Figure 23. We find that $20^\circ.1^2$ forms a good progressive folding condition between the stiffness at the top and the support force at the bottom when subjected to impact, regardless of the thickness, whereas with an increased number of circular cores, the stiffness at the top is so large that its resistance to deformation far exceeds the support force at the bottom, resulting in a buckling deformation at the bottom. The resulting flexural deformation of the bottom under this large angular impact causes a sharp drop in the load curve once it has formed its peak, as the axial loading-bearing capacity is basically lost.

We have described the force–displacement curves and deformation modes above. Figure 25 develops a comparative analysis of crashworthiness indexes. The increase, both in the number of circular core and in thickness increase the PCF and MCF. However, the SEA does not increase progressively, at $20^\circ$ impact, the SEA of $2^2$ and $3^2$ are similar, and the SEA of $2^2$ tube with thicknesses of 0.6 and 0.7 mm exceeds that of $3^2$ tube. The SEA for different thicknesses are compared and the maximum values of SEA for each number of circular core are obtained at: $0^\circ.1^2.1.0$ (37.39 J/g), $0^\circ.3^2.1.0$ (36.46 J/g), $0^\circ.2^2.0.9$ (36.46 J/g), $0^\circ.3^2.1.0$ (64.94 J/g), $10^\circ.1^2.1.0$ (32.42 J/g), $20^\circ.1^2.1.0$ (50.74 J/g), $10^\circ.3^2.1.0$ (54.71 J/g), $20^\circ.3^2.1.0$ (54.71 J/g), $30^\circ.3^2.1.0$ (64.94 J/g), $30^\circ.3^2.1.0$ (35.80 J/g), $20^\circ.2^2.1.0$ (35.80 J/g), and $30^\circ.3^2.1.0$. It can also be seen that the energy absorption capacity of the tube gradually decreases as the
angle of oblique impact increases. The CFE and $\omega$ show a steady wave variation as the thickness increases, the mean values of CFE at different thicknesses are: 0.68 ($0^\circ$), 0.82 ($0^\circ$), 0.84 ($0^\circ$), 0.87 ($10^\circ$), 0.85 ($10^\circ$), 0.80 ($10^\circ$), 0.82 ($20^\circ$), 0.72 ($20^\circ$), 0.63 ($20^\circ$), 0.60 ($20^\circ$), 0.67 ($30^\circ$), 0.67 ($30^\circ$), and 0.68 ($30^\circ$). The mean values of $\omega$ at different thicknesses are: 0.67 ($0^\circ$), 0.37 ($0^\circ$), 0.38 ($0^\circ$), 0.22 ($10^\circ$), 0.20 ($10^\circ$), 0.35 ($10^\circ$), 0.32 ($20^\circ$), 0.60 ($20^\circ$), 0.88 ($20^\circ$), 0.93 ($30^\circ$), 0.73 ($30^\circ$), and 0.73 ($30^\circ$). It can be seen that the tube will

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**FIGURE 21:** Force–displacement curve with increasing displacement $D$ and number of circular core $N$: (a) $0^\circ$, (b) $10^\circ$, (c) $20^\circ$, and (d) $30^\circ$. 

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FIGURE 22: Deformation modes with increasing thickness and number of circular core $0^\circ$.

FIGURE 23: Deformation modes with increasing thickness and number of circular core $10^\circ$.

FIGURE 24: Deformation modes with increasing thickness and number of circular core $20^\circ/30^\circ$. 
**FIGURE 25:** Continued.
exhibit stable loading-bearing capabilities in the axial and 10° impact, but less stable at 20° and 30° impact.

6. Conclusion

In this study, inspired by the cylindrical fibers through-hole structure in bamboo, bionic tube is designed and analyzed. The accuracy and feasibility of the FEM are verified by the drop hammer impact experiments and theoretical analysis, and the optimum ratio of the bionic unit is found by multi-angle impact simulation. Finally, the effects of the number of bionic unit and wall thickness on the crashworthiness of multangle impact are investigated based on the optimal structure, and the main conclusions are as follows:

(1) The results of the impact experiments and the theoretical analysis show that the accuracy of the established FEM is high.

(2) The energy absorption capacity of the bionic tube is significantly increased due to the presence of the circular element structure, the optimal ratio of the bionic structure (length of reinforcement: radius of the circular element) is obtained from the multiangle impact calculation and analysis as 7:3. The crashworthiness
at small angles is significantly improved compared to normal square tubes: SEA is increased by 12.72% (0°), 14.91% (10°), and 18.12% (20°), respectively; but SEA at 30° is reduced by 2.53%.

(3) The axial crashworthiness tends to increase as the number of circular core increases under constant mass, but the tube exhibits poor intermediate twist deformation when the number of circular core ≥4. The deformation shows progressive folding for the number of circular cores ≤3, where the 3° tube shows a 1.37 times higher SEA in axial compression compared to single circular core tube under equal mass.

(4) The load-bearing and energy absorption of tubes with different numbers of circular core tends to increase with increasing thickness, but the deformation mode becomes progressively worse. If the deformation mode is not taken into account, compared to the single core tube, the SEA of a tube with 3° cores increases by an average of 1.85 times (0°), 1.89 times (10°), 1.34 times (20°), and 1.41 times (30°) for each impact condition, respectively.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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