Analysis of aluminum front bumper beam for heavy quadricycles under impact loading

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Abstract
A heavy quadricycle is a small four-wheeled vehicle suitable for urban mobility owing to its compact, lightweight and energy-efficient advantages. However, such small vehicles have a comparatively short crumple zone leading to a higher risk of occupant injury in case of frontal collision. In this paper, the structural performance and energy absorption of aluminum alloy AA6063-T6 thin-walled front bumper beams for heavy quadricycles are studied. Mechanical characteristics of the beams with simple cross-sectional profiles, i.e., square, rectangle and double-cell rectangle as well as a commercially available bumper beam of a heavy quadricycle under cylindrical-rod impact loading are investigated through finite element analysis using LS-DYNA. A parametric study is also performed to examine the effects of open and closed sections, different cross-sectional shapes and various fillet radii on impact responses of the bumper beam including energy absorption, specific energy absorption, peak force, and crush force efficiency. The results show that although section profiles with sharp corners possess higher energy absorption, a larger filament radius could mitigate the unfavourable peak force from impact up to 40%. The partition plate of the double-cell rectangle section induces a much higher peak force compared to others. In addition, when comparing an open-section beam with closed-section beams, it is found that an open-cell hat-shape model with fillet radius of 8 mm is the most efficient design. The crush force efficiency and peak force of the section are 32% higher and 17% lower, respectively, when compared to the closed-cell section with fillet radius of 8 mm.

Keywords: Bumper beam, Frontal impact, Energy absorption, Heavy quadricycle

1. Introduction
At present, one of the greatest challenges faced by automotive industry is to provide safer vehicles with high energy efficiency at a competitive cost [1]. The use of high-performance lightweight materials for structural members offers great potential for increasing vehicle efficiency while crashworthiness requirements are also satisfied. A heavy quadricycle, categorized by the European Union as L7e, is a small four-wheeled vehicle that has an unladen mass equal to or less than 450 kg without battery, a maximum net motor power not exceeding 15 kW and a maximum speed not less than 90 km/h [2]. Quadricycles are suitable for urban mobility owing to their compact, lightweight and energy-efficient
advantages. However, such vehicles have a comparatively short crumple zone leading to a higher risk of occupant injury in case of frontal collision.

Front crumple zone is mostly designed by mounting thin-walled stiff bumper beam to crash boxes so as to absorb and dissipate kinetic energy in accidental impacts. Crash energy must be absorbed in a controlled manner with limiting forces exerted on occupants and unstable behaviors such as buckling or catastrophic failures be avoided [3]. The energy absorption capacity and crashworthiness performance of the bumper and crash boxes have received extensive attention in the engineering fields [4]. To achieve higher energy absorption efficiency, researchers proposed various methods to achieve the better design of the members [5-8]. Energy dissipation mechanisms for diverse design of crumple zone are also substantially explored using basic theory of deformation and observation of beam under indentation-mode impact loading [9-14]. Based on the Technique for Order of Preference by Similarity to Ideal Solution (TOPSIS) used in bumper beam profile selection process for small road vehicle, it is found that the double hat profile fits the bumper beam for a small car. The selected concept can be strengthened by adding ribs or increasing the thickness of the bumper beam to make the best geometry according to defined product design specifications [15-16]. Dynamic tests on a double hat profile and a simple hat profile under impact showed that the double hat profile exhibits better stress distribution and lower deflection after impact [17-18] which benefits both passenger and pedestrian protection [19].

This paper aims to apply finite element analysis to investigate the energy absorbing behavior and failure mechanism of the aluminum alloy AA6063-T6 thin-walled front bumper beam for heavy quadricycles under impact loading. A parametric study is also performed to examine the effects of open and closed sections, different cross-sectional shapes and various fillet radii on impact responses of the bumper beam.

2. Bumper Beam Model

2.1 Material Properties

The bumper beam in this study is made of aluminum AA6063-T6. The material mechanical properties as listed in table 1 are obtained from uniaxial tensile tests according to the ASTM E8M-04 standard [20] using three specimens. The tensile tests are conducted using Zwick Roell HB250 universal testing machine. The loading speed is set constant at 1 mm/min. The true stress-strain curve as illustrated in Figure 1 is assigned to the bumper material model in the finite element simulation.

Table 1. Material properties of AA6063-T6 obtained from the uniaxial tensile test.

| Material   | E (GPa)     | ν [14] | ρ (kg/m³) [14] | σy (MPa) | σu (MPa) | εmax     |
|------------|-------------|--------|----------------|-----------|-----------|-----------|
| AA6063-T6  | 64 ± 2.72   | 0.33   | 2700           | 197.72 ± 2.52 | 214 ± 2.88 | 0.0991 ± 1.25 |

Figure 1. True stress-strain curve of AA6063-T6.
2.2 Bumper Beam Profiles
In order to assess the most suitable profile for the bumper beam of an L7e quadricycle, four aluminum profiles, which are commercially available profile, rectangular section, double-cell rectangular section and square section, are proposed and their dimensions are shown in Table 2. The length of the bumper beam should not exceed the vehicle width of 1,500 mm. In this study, the bumper beam length is 720 mm taken from a commercial model of heavy quadricycle. To facilitate result comparison of energy absorbing capacity of different beam sections, the mass of all beam profiles is controlled at 0.6 kg by changing the beam thickness.

| Section profile | A (mm) | B (mm) | Thickness (mm) |
|-----------------|--------|--------|----------------|
| Commercial      | 50     | 50     | 1.89           |
| Rectangle       | 48     | 40     | 2.00           |
| Double-cell rectangle | 30     | 26     | 1.97           |
| Square          | 38     | 38     | 2.00           |

Table 2. The geometry of the bumper beam cross section for the impact loading test.

3. Cylindrical-rod Impact Simulation
The cylindrical-rod impact loading tests are simulated using non-linear explicit finite element code in LS-DYNA. The bumper beam of length 720 mm is supported by two cylindrical rods with a span of 620 mm at the locations of the crash boxes (Figure 2a). The beam is discretized using Belytschko-Tsay 4-node shell elements. The 20-mm diameter cylindrical rod and supports are modelled using 8-node solid elements. An automatic single surface contact algorithm is applied to model the contact of the beam to itself. To account for the contact between the impactor (master) and the beam (slave), and between the beam (master) and the supports (slave), automatic node-to-surface contacts are defined. The friction coefficient for all contact pairs is set to be 0.3 [21]. The MAT_024 Piecewise Linear Plasticity Material model is assigned to the beam material using the properties obtained from the test results and the stress-strain curve as shown in Figure 1 and MAT_020 Rigid is used to model the cylindrical-rod and supports. The cylindrical rod of mass 25 kg is assigned with an initial impact velocity of 4.71 m/s (17 km/h), which provides an initial impact energy of 279 J. Mesh sensitivity is performed by comparing the simulated average force of models with element sizes from 0.5 mm to 5 mm as shown in Figure 2(b). The resulting average impact force obtained from mesh sizes of 0.5, 1 and 2 mm are relatively similar. Therefore, 2-mm element size was chosen to reduce computation time without compromising result accuracy. Strain rate effect is not considered in the present numerical model since the aluminum alloy 6063-T6 is strain-rate insensitive during low strain-rate impact between 110 s\(^{-1}\) and 850 s\(^{-1}\) [22].
4. Simulation Results and Discussion

4.1. Effects of Bumper Beam Profiles

The deformed shapes and effective plastic strain of the bumper beams under cylindrical-rod impact loading are illustrated in Figure 3. After impact, all beam sections exhibit similar fracture failure starting from the top surfaces underneath the point of impact and propagating to the sides of the beams. Force-displacement curves of each cross-section profile under impact are compared in Figure 4 and similar trend can be clearly seen. Peak forces are observed at the beginning stage of the impact when failure at the corners of the top surface first occurs. Then, the forces drastically drop afterward and continuously decrease when the beam’s side walls damage causing large deformations to the beam. The double-cell rectangular profile shows the highest peak force \( F_{\text{max}} \) of 11.3 kN due to the stiffened vertical wall inside the cross-section. The commercially available hat-section gives a noticeably lower peak force compared to that of other profiles owing to the deformation mechanism of the open-section in which its vertical walls could undergo larger horizontal deformation and plastic strain to absorb the first impact force.

![Effective plastic strain of the bumper beam under cylindrical-rod impact](image)

Figure 3. Effective plastic strain of the bumper beam under cylindrical-rod impact.

Results on energy absorption (EA), specific energy absorption (SEA), crush force efficiency (CFE), average force \( F_{\text{avg}} \) and maximum deformation \( d_{\text{max}} \) are listed in Table 3. The average force is computed by averaging the force over the entire distance of the impact event. Crush force efficiency is defined as the ratio of the average force to the peak force. Thus, a preferable profile for a bumper is one with a low peak force to mitigate the injury risk of the occupant during crash, a high SEA for mass...
and cost efficiency, and CFE should be close to 1 where the impact force is well transferred to different parts of the structure. The commercial open-cell model is found to exhibit the highest SEA of 202.8 J/kg. However, the double-cell rectangular section gives the highest CFE among other sections. Figure 5 illustrates result comparison of the rectangular, double-cell rectangular, and square sections with the commercially available baseline model. It is found that the closed-section bumper beams have lower EA and SEA than those of baseline model by 30-40%. The double-cell rectangle although provides a higher CFE and $F_{\text{avg}}$, its $F_{\text{max}}$ is somewhat dangerously higher than other models. Therefore, considering the above discussion, the optimal shape for a heavy quadricycle is the commercial section as it exhibits the highest EA and SEA and less peak force compared to other sections.

![Figure 4. Force-displacement curves.](image1)

![Figure 5. Section performance compared with baseline.](image2)

**Table 3.** Energy absorption capability of different bumper beam cross sections.

| Cross section   | Commercial | Rectangle | Double-cell rectangle | Square |
|-----------------|------------|-----------|-----------------------|--------|
| EA (J)          | 123.3      | 70.6      | 78.2                  | 84.1   |
| SEA (J/kg)      | 202.8      | 115.9     | 128.3                 | 138.1  |
| CFE             | 0.205      | 0.161     | 0.228                 | 0.184  |
| $F_{\text{max}}$ (kN) | 7.7        | 7.8       | 11.3                  | 8.2    |
| $F_{\text{avg}}$ (kN) | 1.6        | 1.3       | 2.6                   | 1.5    |
| $d_{\text{max}}$ (mm) | 109.77     | 116.16    | 107.58                | 112.68 |

4.2 Effects of Fillet Radius

The failure mechanism simulation results of the bumper beams under rod impact testing have shown that the crack initiates from the top corners of the beams and propagating to the side walls. Therefore, adding rounded fillets at corners will help improve the load carrying and energy absorption capability of the bumper beam. Therefore, the effects of fillet radius for both open-cell and closed-cell commercially available sections with the fillet radii of 0, 2, 5, 8 mm are investigated.

The force-displacement results of various fillet radii for both open-cell and closed-cell cross sections are illustrated in Figure 6. The energy absorption performance comparison with the baseline model is also shown in Figure 7. It is noticed that, for open-cell sections, the size of fillet radius largely affects the section performance. The beam section with a larger fillet radius has a distinguishably less peak force and thus higher CFE and $F_{\text{avg}}$. The most preferable open-cell section is one with fillet radius of 8 mm in which the $F_{\text{max}}$ is reduced by 40% compared to the baseline model while CFE and $F_{\text{avg}}$ are
improved by 100.9% and 20.5%, respectively. However, for closed-cell sections, the advantages of fillet radius are not as evident. Force-displacement curves for closed-cell section with different fillet radii are similar. Although the closed-cell sections with fillet radius of 8 mm show a lower $F_{\text{max}}$ compared to the open-cell and the baseline sections, the SEA and $F_{\text{avg}}$ are also reduced due to the characteristics of the closed-cell bumper damage during impact. In addition, the performance of all closed-cell sections is found to be less favorable when compared with those of open-cell sections.

Figure 6. Force-displacement curves of (a) open-cell (b) closed-cell sections with different fillet radii.

Figure 7. Performance comparison of commercial section with various fillet radii to the baseline model.

5. Conclusion
The behavior and performance of the heavy quadriclec bumper beam under the cylindrical-rod impact loading with various section profiles and fillet radii are examined. Among the closed-cell sections, the double-cell rectangular bumper beam provides the highest CFE and $F_{\text{avg}}$. However, the open-cell commercial section gives a much higher SEA than other beam profiles with the lowest $F_{\text{max}}$. The analyses of fillet size for open-cell and closed-cell sections indicate that the use of larger fillet radii up to 8 mm can improve the performance of the open-cell sections while the effect of fillet radius to the closed-cell sections are not as prominent. The open-cell hat-shape model with fillet radius of 8 mm is found to be the most efficient design as it exhibits the highest CFE and the lowest $F_{\text{max}}$. 
This model should be examined further under other impact and crash loads in order to use efficiently for a heavy quadricycle.

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