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Multiobjective optimization for foam-filled multi-cell thin-walled structures under lateral impact

Hanfeng Yin [a,b], Youye Xiao [a,b], Guilin Wen [a,b,*], Qixiang Qing [b], Yufeng Deng [b]

[a] State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University, Changsha 410082, Hunan, PR China
[b] Key Laboratory of Advanced Design and Simulation Techniques for Special Equipment, Ministry of Education, Hunan University, Changsha 410082, Hunan, PR China

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ABSTRACT

Nowadays, foam-filled multi-cell thin-walled structure (FMTS) has been widely used in the field of automotive due to their extraordinary energy absorption capacity and light weight. In this study, nine kinds of FMTSs with different cross-sectional configurations under lateral crushing load conditions were investigated using nonlinear finite element method through LS-DYNA. The complex proportional assessment (COPRAS) method was used to make clear which kind of FMTSs has the most excellent crashworthiness. According to this method, it can be found that FMTSs with 2, 3 and 9 cells are the top-3 excellent structures in our considered cases. In order to improve the crashworthiness of the three FMTSs, they were optimized by metamodel-based multiobjective optimization method which was developed by employing polynomial regression (PR) metamodel and multiobjective particle swarm optimization (MOPSO) algorithm. In the optimization process, we aimed to achieve maximum value of specific energy absorption (SEA) and minimum value of maximum impact force (MIF). Based on the comparison of the Pareto fronts obtained by multiobjective optimization, we can find that FMTS with 9 cells (FTMT9) performs better than FMTSs with 2 and 3 cells. Thus, the optimal design of FMTS9 is exactly an excellent energy absorption candidate under lateral impact and can be used in the future vehicle body.

1. Introduction

As we all know, the bumper, side-door beam and B-pillar can absorb impact energy in a car accident to protect the safety of the driver and passengers, they have become very important energy absorbers in vehicle body. Nowadays, foam-filled thin-walled structures have drawn increasing attention due to their excellent energy absorption capacity and extraordinary light weight. Thus, a lot of work on studying the energy absorption characteristics of foam-filled thin-wall structures under lateral impact condition has been done by employing experimental, analytical and numerical methods. Santosa et al. studied the effect of ultralight metal filler on the bending collapse behavior of thin-walled prismatic columns [1]. Through analyzing, he drawn conclusions that the columns filled with low-density metallic such as foam or honeycomb can effectively increase the bending strength of the column. Chen [2] studied bending collapse of foam-filled hat profiles though experiment and numerical simulation. In the experimental study, he found that foam-filled hat profiles of sectional geometry and foam density can achieve 30–40% increase in the specific energy absorption (SEA), compared to traditional non-fill members, which proved that foam-filled structures were potential weight-efficient energy absorbers. Dynamic bending responses of foam-filled double tube structures were studied by Guo and Yu [3] experimentally and numerically. They found that the new structure had steadier load carrying capacity and much higher energy absorption efficiency than those of the traditional foam-filled single tube under bending condition. Shojaeifard et al. [4] investigated the bending behavior of empty and foam-filled aluminum tubes with different cross-sections numerically. They found that the SEA of elliptic foam-filled tube with 1.5 mm thickness is 22% higher than untreated elliptic tube with the same thickness.

Since the energy absorption was found to be significantly dependent on the foam density and tube geometry [5], optimization techniques have been widely employed as useful methods to further improve the energy absorption capability of foam-filled thin-walled structures. Zarei and Kröger [6] applied structural optimization technique to study bending behavior of empty and foam-filled beams and they found that foam-filled beam could absorb the same energy as the empty beam with lower weight. To achieve the optimum design of the uniform foam (UF) and functionally graded foam (FGF) filled beams, Fang et al. carried out the multiobjective crashworthiness optimizations for these

* Corresponding author at: State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University, Changsha 410082, Hunan, PR China. Tel.: +86 731 88821482; fax: +86 731 88822051.
E-mail address: glwen@hnu.edu.cn (G. Wen).

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structures [7]. From their results, they pointed out that the FGF filled beams performed better than the UF counterparts. In addition, from the previous studies on multi-cell thin-walled structures [8–11], we can find that the crashworthiness of the multi-cell thin-walled structure is better than that of the single-cell thin-walled structure under axial impact. Yin et al. investigated six kinds of foam-filled multi-cell thin-walled structures (FMTSs) under axial impact using the nonlinear finite element code LS-DYNA [12]. They concluded that the FMTS with nine cells had the most excellent crashworthiness in their considered cases. Therefore, the energy absorption capacity of the FMTS is very likely to be better than that of the foam-filled single-cell thin-walled structure. However, to our knowledge, there are few research papers that study the crashworthiness of FMTSs under lateral loading condition.

In this paper, nine kinds of FMTSs with different cross-sectional configurations under lateral loading conditions were investigated using the nonlinear finite element code LS-DYNA. The complex proportion assessment (COPRAS) [13–16] method was used to find the most excellent one among those FMTSs. Based on COPRAS method, three kinds of FMTSs were found to have top-3 excellent crashworthiness. Then, these three FMTSs were optimized by a metamodel-based multiobjective optimization method, which was developed by employing polynomial regression (PR) metamodels as well as multiobjective particle swarm optimization (MOPSO) algorithm.

2. Finite element modeling

2.1. Geometrical description

The structure investigated in this study is foam-filled multi-cell thin-walled structure (FMTS) subjected to lateral impact loading. The schematic and finite element model of FMTS with cell number $n=1$ is shown in Fig. 1. FMTS lays on two cylindrical supports [6]. The span and diameter of the two cylindrical supports are 430 mm and 50 mm, respectively. The length of the tube is 550 mm, the side length of the cross section is 55 mm and the thickness of the tube is 2 mm. A cylindrical punch with a diameter of 50 mm and a mass of 998 kg impacts onto the column at an initial velocity of $v=4.4$ m/s in the mid-span. In this study, nine kinds of FMTSs were investigated using nonlinear finite element code LS-DYNA. These nine kinds of FMTSs are shown in Fig. 2.

2.2. Material properties

The material of thin-walled structure is aluminum alloy AlMg-0.5F22 with mechanical properties of density $\rho=2.7\times 103$ kg/m^3, Young's modulus $E=68.566$ GPa, Poisson's ratio $\mu=0.29$, initial yield stress $\sigma_y=227$ MPa, and tangential modulus of elasticity $E_t=321$ MPa [7,17]. As the aluminum is insensitive to the strain rate, the rate-dependent effect was neglected in the finite element (FE) modeling [18]. Thus, the tube was modeled with a bilinear elastic–plastic material with strain hardening (material model 24 in LS-DYNA).

The material model chosen to represent the material behavior of aluminum foam filler was Deshpande–Fleck foam (material model 154 in LS-DYNA). The model was proposed by Deshpande and Fleck [19]. The yield criterion of the foam material is defined by

$$\Phi=\hat{\sigma}-\sigma_y\le 0 \tag{1}$$

where σ_y denotes the yield stress and $\hat{\sigma}$ is the equivalent stress. The equivalent stress $\hat{\sigma}$ is given as follow

$$\hat{\sigma}^2=\frac{1}{[1+(\alpha/3)^2]}(\sigma_e^3+\alpha^2\sigma_m^2) \tag{2}$$

where σ_e is the effective Von Mises stress and σ_m represents the mean stress. Parameter α defines the shape of the yield surface, which can be written as

$$\alpha^2=\frac{9(1-2\upsilon_p)}{2(1+\upsilon_p)} \tag{3}$$

where υ_p is the plastic Passion's ratio. In most cases, the plastic Passion's ratio of aluminum foam is zero, thus $\alpha=2.12$.

The strain hardening rule is implemented in this material model as

$$\sigma_y=\sigma_p+\gamma\frac{\hat{\varepsilon}}{\varepsilon_D}+\alpha_2\ln\left[\frac{1}{1-(\hat{\varepsilon}/\varepsilon_D)^{\beta}}\right] \tag{4}$$

where $\hat{\varepsilon}$ represents equivalent strain, σ_p, α_2, γ, β and ε_d are the material parameters and can be related to the foam density as

$$\begin{cases}(\sigma_p,\alpha_2,\gamma,\frac{1}{\beta},E_P)=C_0+C_1(\frac{\rho_f}{\rho_{f0}})^{\kappa}\\ \varepsilon_d=-\ln(\frac{\rho_f}{\rho_{f0}})\end{cases} \tag{5}$$

where ρ_f is the foam density and ρ_{f0} is the density of base material. C_0, C_1 and κ are the constants, which are listed in Table 1. From Eq. (5) [20], we find that the Young's modulus of foam material E_p is also a function of ρ_f.

2.3. Finite element modeling

The finite element models were established using explicit nonlinear finite element code LS-DYNA. The tube wall was modeled with the Belytschko–Lin–Tsay thin shell elements. To model foam materials, the eight-node solid elements with one-point reduced integration were adopted. The punch and supports were set as rigid bodies.

The interface between the foam and wall was modeled with "automatic surface to surface" contact. While an "Automatic single

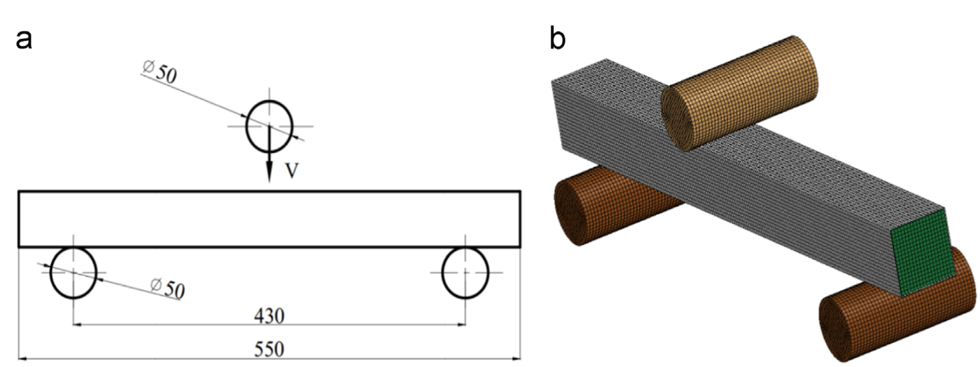

Fig. 1. The FMTS with cell number $n=1$: (a) schematic [6] and (b) finite element model.