

Front Rail Crashworthiness Design for Front Oblique Impact Using a Magic Cube Approach

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ABSTRACT

The front rail, as one main energy absorption component of vehicle front structures, should present steady progressive collapse along its axis and avoid bending collapse during the front oblique impact, but when the angle of loading direction is larger than some critical angle, it will appear bending collapse causing reduced capability of crash energy absorption. This paper is concerned with crashworthiness design of the front rail on a vehicle chassis frame structure considering uncertain crash directions. The objective is to improve the crash direction adaptability of the front rail, without deteriorating the vehicle's crashworthiness performance. Magic Cube (MQ) approach, a systematic design approach, is conducted to analyze the design problem. By applying Space Decomposition of MQ, an equivalent model of the vehicle chassis frame is generated, which simplifies the design problem. Based on this model, a two-layered front rail is proposed using a multi-step multi-domain topology optimization method and a response surface method. Numerical simulations are carried out with Altair/Hypermesh and LS-DYNA to compare the crashworthiness performances of the original front rail and the proposed two-layered design. The result shows that the two-layered front rail, with a reduced weight by 17%, can absorb 46% more energy in the 30 degree front impact than the original design, meanwhile, the proposed design eliminates the bending collapse on its rear end when the angle of loading direction reaches the critical angle. The energy absorption capability and the direction uncertainty adaptability of the front rail are significantly improved.

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INTRODUCTION

Crashworthiness, the ability of a structure to absorb energy for protecting occupants during an impact, is one of the most important design criteria for passenger cars. With the increase of government regulations of standards and consumer requirements, it has become the first design criterion considered during the vehicle development. Due to the complex loading and boundary conditions, the large nonlinear deformation, and many uncertainties of the modeling and simulation process, it is difficult to obtain a proper vehicle design which can optimize the

crashworthiness. For the vehicle front impact, including the front oblique impact and offset impact, the front energy-absorbing structure should present steady progressive collapse along its axis from the front end to the rear and avoid bending collapse. The end of front structure should be stiff enough to maintain passenger survival space. The front rail, as one main component to absorb crash energy in front collision, should follow this design principal. Analytical, numerical and experimental studies on geometrical shape design of the front rail in the axial collapse, including cross sectional area, gauges, triggers and holes, have been carried out [1,2,3,4,5]. In [6], Kitagawa, et al., proposed a buckling

analysis method to calculate the efficient position of beading for controlling collapse mode (axial and bending collapse). Yamazaki and Han [7] crashed tubes into a rigid wall and the thickness of tubes are optimized using a response surface method.

In above studies, only axial loadings are considered in the crashworthiness design of vehicle components. In a real crash event, the loading directions are rarely purely axial. Uncertainties in loading conditions will bring great difficulty in a vehicle structure design since loadings with a small angle to the travel direction may change the collapse mode. Han and Park [8] studied the thin-walled structure tube, provided formulations for the crush behavior of a square column subjected to oblique loads. They showed us that there are three collapse parts under oblique loading directions: axial collapse region, bending collapse region, and transition region from axial collapse to bending collapse. Nagel and Thambiratnam [9] compared the energy absorption response of straight and tapered thin-walled rectangular tubes under oblique impact loading. Ahmada et al. [10] studied the energy absorption ability of empty and foam-filled conical tubes. Qi et al. [11] applied a step-by-step failure mode modification approach considering the uncertainties in crashworthiness design. Recently, Qi et al. [12] compared four type axisymmetric thin-walled square tubes and conducted multiobjective crashworthiness optimization of the multi-cell cross section square tube. In all these aforementioned studies on thin-walled structure crashworthiness design with the oblique loading, researchers mainly focused on two traditional methods that are commonly adopted in the auto-industry. One is optimizing the component thickness and the other is adding reinforce structure or composite material such as formed aluminum honeycomb to enhance the front rail bending stiffness. However, these methods are only conducted after structure topology design, which will increase costs and the overall weight. The effect of the loading direction uncertainties has not been systematically considered during the crashworthiness design process of thin-walled structures.

In the present study, the oblique loading effect is considered in the front rail topology design process. The front rail is one typical thin-walled structure in vehicle body frame, so it must be sensitive to the crash angle. In other words, an optimal design under axial loadings may no longer perform high energy absorption capability under non-axial loadings. The goal of this research is to find a proper front rail design to improve its adaptability of the loading direction uncertainty and energy absorption ability in the front oblique crash. Based on a systematic Magic Cube (MQ) approach [13] and a multi-step multi-domain topology optimization method (MMTO) [14, 15, and 16], a two-layered front rail is proposed. Meanwhile, the topology design procedure of the front rail is worthy of conducting to consider the loading direction effect in the vehicle thin-walled structure topology design.

This paper is organized as follows: the MQ approach and MMTO method are briefly introduced first, followed by the problem description of the vehicle frame structure design for the vehicle oblique impact. The MQ approach is then employed to cascade the design task into subsystem and component design problems by applying four aspects of MQ: space decomposition, failure modes, optimization method and loading (crash). For the MQ element "Optimization Method", the multi-domain and multi-step topology optimization approach and the response surface method will be employed to chase a new front rail design. Finally, numerical simulations are carried out with Altair/Hypermesh and LS-DYNA to verify the crashworthiness performance of the proposed two-layered front rail in front oblique impact.

MAGIC CUBE APPROACH

The Magic Cube approach (MQ) is an advanced and systematic design approach, which has been developed in Chang Qi's research for vehicle crashworthiness design [13]. The Magic Cube is composited of 27 elements which are identified based on the general procedures and methodologies used in complex crashworthiness design problems as shown in Fig.1. The MQ approach includes three major dimensions: Decomposition, Design Methodology and General Consideration. The Decomposition dimension is related to the major decomposition approaches developed, and it contains three layers: Time (Process) Decomposition, Space Decomposition, and Scale Decomposition. The Design Methodology dimension is related to the techniques applied, which has three layers: Target Cascading, Failure Modes and Optimization Method. The General Consideration dimension also has three layers: Multidisciplinary Objectives, Loading Conditions, and Uncertainty Effects.

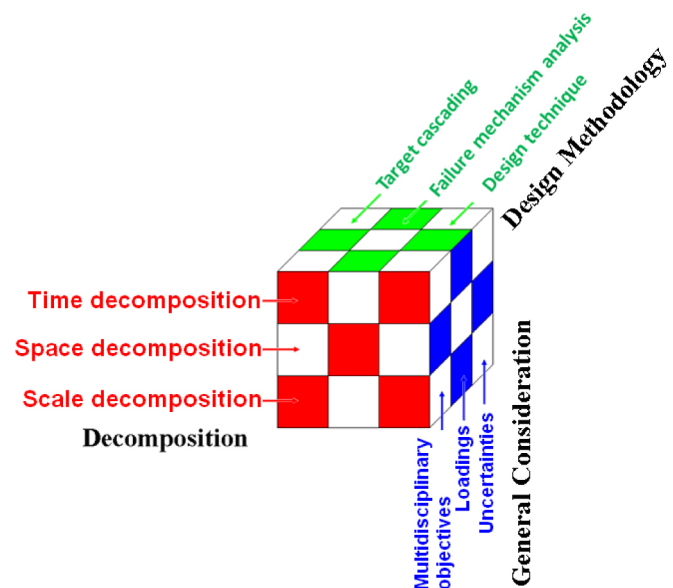


Figure 1. Magic Cube for crashworthiness design

MULTI-STEP MULTI-DOMAIN TOPOLOGY OPTIMIZATION METHOD

In the traditional topology optimization method, such as Homogenization Based Topology Optimization (HBTO), the structure is optimized within a single structure domain, subject to a given amount of the material for the entire structure. During the optimization process, the material in the design domain will be automatically distributed without designers' interaction. This process leaves little flexibility to the designer to control the material distribution. The optimization result is often too complex to be manufactured.

In order to solve these problems, a multi-domain multi-step topology optimization method is proposed for crashworthiness design by Ma, et al. [14, 15, and 16]. This method is composed of the multi-domain topology optimization (MDTO) method and the multi-step topology optimization (MSTO) method. In contrast to the single domain topology optimization, in which a given amount of the material is assigned to the entire design domain, MDTO allows the designer to assign different amounts of the material, or even different materials, to the different sub-domains of the structure. For example, Fig. 2 shows that a structural domain is divided into several sub-domains, where a certain amount of the material A is distributed into sub-domain 1, and a different amount of the material B is distributed into sub-domain 2. Furthermore, sub-domain 3 is considered as a non-design domain, where the material distribution stays unchanged at the current design stage.

In the general case, the optimization problem of the MDTO can be written as

$$\begin{aligned} & \text{Maximize: } f(\mathbf{X}) \\ & \text{Subject to: } h_j(\mathbf{X}) \leq 0 \quad (j = 1, 2, 3, \dots, m) \\ & \quad \underline{x}_j \leq x_j \leq \bar{x}_j \quad (j = 1, 2, 3, \dots, m) \end{aligned} \quad (1)$$

where $f = f(\mathbf{X})$ denotes the objective function; $h_j(\mathbf{X})$ denotes the j -th constraint function for the volume (or weight) of the j -th substructure in the j -th sub-domain (where $j = 1, 2, \dots, m$); $\mathbf{X} = \{x_1, x_2, \dots, x_n\}^T$ denotes the vector of the design variables; and $\underline{x}_j$ and $\bar{x}_j$ are the lower and upper bounds of design variable, respectively. Note that $f(\mathbf{X})$ in Eq. (1) must satisfy the state equations for the structural analysis problem at hand. These state equations may include, for example, the static equilibrium equation, the equation that defines the free vibration eigenvalue problem, or the equation for the dynamic response.

In order to improve the manufacturability of the design obtained from topology optimization, a multi-step topology optimization (MSTO) method was developed for crash by Wang, et al [16]. The main idea of MSTO is to decompose the complex crashworthiness design into a series of topology optimization design steps. In the current optimization step,

the optimized design results in the previous step will be imported as input conditions, such as structure, boundary condition or control parameters. Compared with the single step topology optimization, the process of MSTO will result in a simpler structure, which can significantly improve the manufacturability.

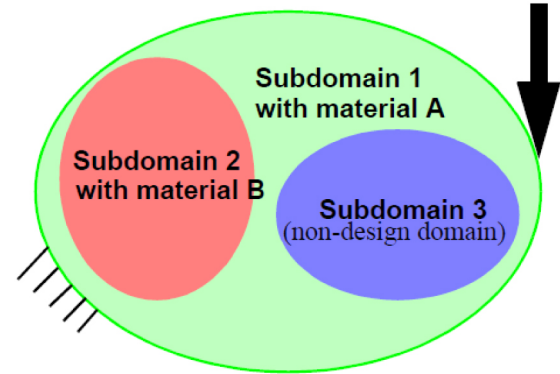


Figure 2. A multi-domain topology optimization problem

PROBLEM DESCRIPTION

Front Oblique Impact

In a real vehicle crash event, collision forms, contact components and loading directions are uncertain and random. Based on vehicle crash directions, collision types are commonly classified into front impact, side impact, rear impact, and roll over. Statistic indicates that the probability of all-type injury in the United State caused by front impact is nearly 40%, and up to 23% of total accidents are caused by front oblique collisions [17]. In the United States, vehicles should meet FMVSS 208 named occupant crash protection in the front impact, which regulates that the vehicle should impact longitudinally to a fixed rigid barrier at 48.3 km/h (30 mph). The barrier is perpendicular to the line of travel of the vehicle, and at any angle up to 30 degrees in either direction from the perpendicular to the line of travel of the vehicle as shown in Fig. 3.

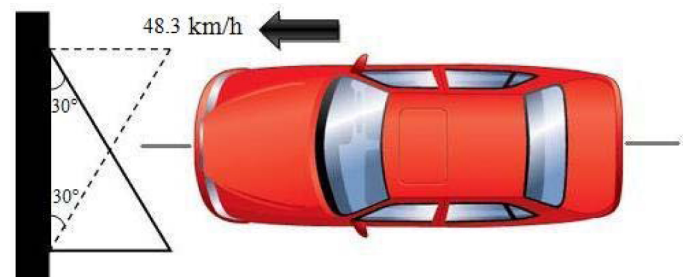


Figure 3. Front angle impact of FMVSS 208

Crash Analysis of Vehicle Frame Structure

The pre-processing of the finite element model in this investigation is conducted by Altair/Hypermesh.

Crashworthiness performance of the baseline design is firstly evaluated in the whole vehicle frame structure using LS-DYNA as depicted in Fig. 4. The vehicle frame structure is composed of 14 thin-walled parts: main members of the chassis frame and the front rail. The total weight is 2003.43 kg. According to FMVSS 208, the vehicle frame structure is crashed onto a rigid wall with a speed of 48.3 km/h. The material of the vehicle frame structure is steel, and the parameters are set as follows: material density $7.83 \times 10^3 \text{ kg/m}^3$, Young's modulus 205GPa, yield stress 167MPa~322MPa. The Mat-024, an elasto-plastic material type, is applied to the vehicle frame structure, and the strain rate is accounted for using the Cowper and Symonds model

which scales the yield stress with the factor $1 + (\dot{\epsilon} / C)^{1/p}$. The static coefficient and the dynamic coefficient of friction between the structure members is respectively 0.15 and 0.12 in the following crash simulations. The friction coefficient between the rigid wall and the structures is 0.3 except numerical models for searching the critical load angle of the front rail.

By adjusting the fixed rigid wall, the front oblique impact simulations of the vehicle frame structure are carried out to find the loading direction influence on the energy absorption capability. Compared with the pure axial front impact, the energy absorption of the front rail is reduced in the front oblique impact. The angles in this front oblique impact are $\alpha = 17^\circ$, $\beta = 6^\circ$ which will be introduced in the following. In Fig. 5, the value of energy absorption increases slowly between 16 ms and 30 ms, since there is a global bending at the end of the front rail. The energy absorbed by the front rail in pure axial impact is higher than that inclined impact ($\alpha = 17^\circ$, $\beta = 6^\circ$). Simulation results demonstrate that the energy absorption ability of the vehicle frame will be affected by crash directions. In [8], with the crash direction changing, the thin-walled structure will perform different collapse modes and crash energy absorptive capacity. Progressive collapse is always a high energy absorption deformation whereas bending collapse is a low energy absorption deformation. The structure design objective is to eliminate the bending collapse during the front angle collisions.

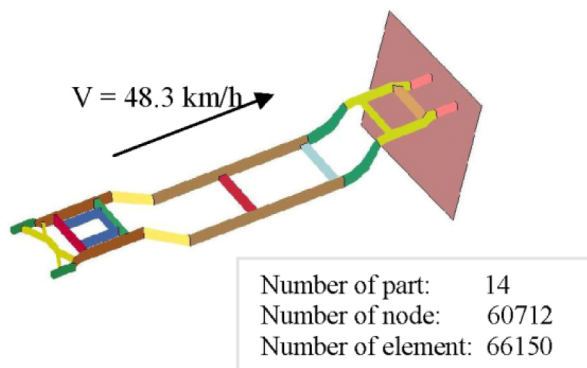


Figure 4. Vehicle chassis frame model

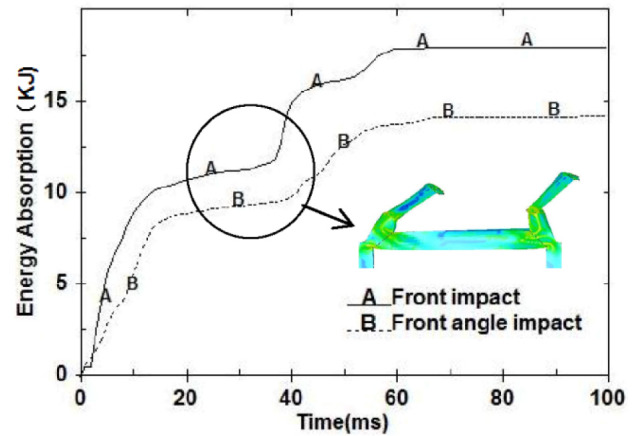


Figure 5. Comparison of the front rail energy absorption in the front impact and the front angle impact

Critical Load Angle of Front Rail

Han and Park's research [8] demonstrated that there is a critical load angle at which a transition takes place from an axial collapse mode to a bending collapse mode. The critical angle equation of a square column tube by non-linear exponential curve-fit method is:

$$\Phi_c = 4.81 + 1564.00e^{-1.12(l/b)} \quad (2)$$

Where l is the length and b is the width of the square column.

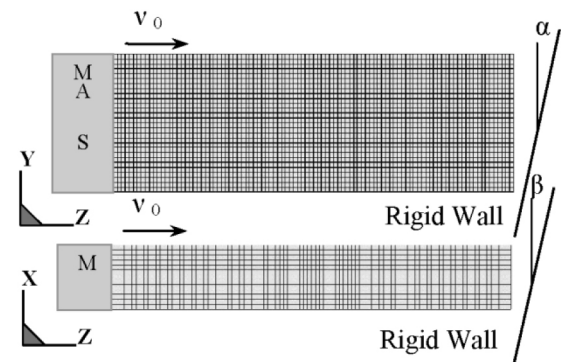


Figure 6. A front rail model for front oblique impact

Unfortunately, there is no equation of the critical angle for rectangle cross-section tubes. Therefore, a numerical model of the front rail for crash simulations is defined to find its critical load angle with no friction as shown in Fig. 6. The front rail has the baseline geometry of length $l = 400 \text{ mm}$, width $b = 64 \text{ mm}$ and height $h = 138 \text{ mm}$, thickness $t = 1.962 \text{ mm}$. Total mass of chassis frame structure (1999 Kg) is attached to the rear end of the front rail. Two critical angles are found from several front rail models built by increasing the angle of the rigid wall's normal to the line of travel of the front rail. As depicted in Fig. 7, when the rigid wall is rotated

$\alpha = 17^\circ$ along X axis, and $\beta = 6^\circ$ along Y axis, the global bending appears at the rear of the front rail. Therefore, the critical angles $\alpha = 17^\circ$, $\beta = 6^\circ$ for this front rail will be used in the crashworthiness design to simulate the load uncertainty.

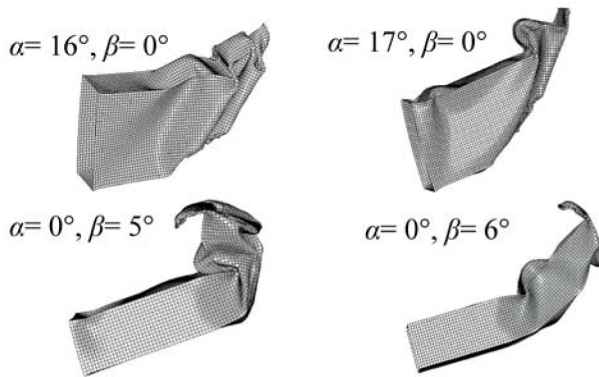


Figure 7. Front rail deformations in front oblique impacts

MAGIC CUBE APPROACH APPLICATION

In this study, the front rail is impacted onto a declined the rigid wall. The failure mode of the front rail is related to its energy absorption capability in the whole vehicle structure. Naturally, to analyse failure mode is helpful for this front rail design. As shown in Fig. 8, the MQ approach is applied to the crashworthiness design of the chassis frame structure: Space Decomposition, Failure Modes and Optimization Method in order to achieve the design objective.

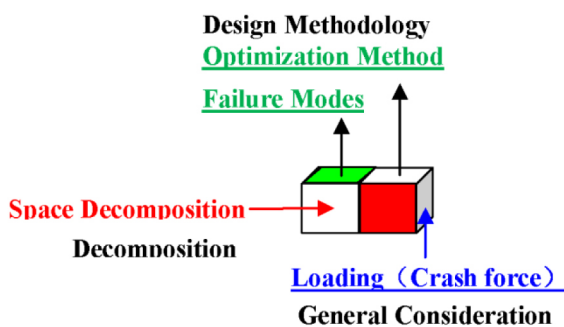


Figure 8. Element of magic cube (MQ) for the crashworthiness design considering front oblique loads

Space Decomposition

Space Decomposition is applied to obtain a simplified design model as depicted in Fig. 9. The vehicle chassis frame is decomposed into a simplified subsystem, according to the stress distribution and the permanent deformation of the whole chassis frame in the front oblique impact. The subsystem is composed of main deformed parts and equivalent mass elements on the side rail connection section. To validate this subsystem model, the crash force history of

the whole chassis frame model and the subsystem model is compared in Fig. 10. It can be seen that two crash force history curves are approximate during the first 70 ms, which indicates that subsystem can represent the chassis frame structure in the first 70ms.

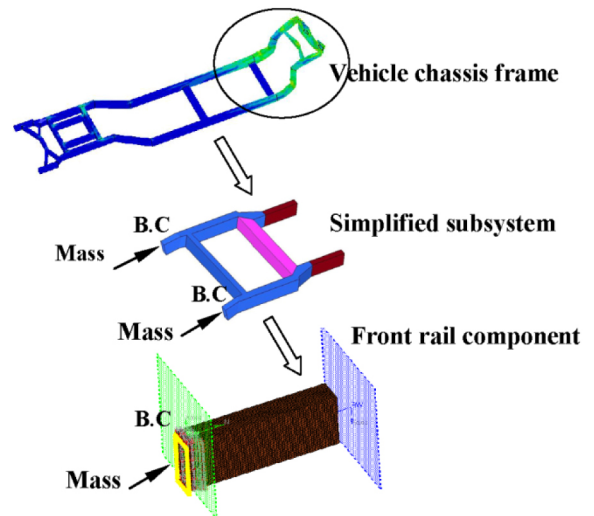


Figure 9. Space Decomposition of the vehicle chassis frame structure

The crash force-time curve and the chassis frame deformation are given in Fig. 11. The crash force onto the vehicle chassis frame is related to failure modes, such as bending, dynamic buckling and plastic hinge, so an important criterion of evaluating structure crashworthiness is to lower the peak crash force. There are two peak forces during the whole crash process: the first peak force is caused by a global bending of the front rail, and the second peak force occurs at 46.0 ms when the lower beam is totally crashed onto the rigid wall. At 30 ms, the major deformed part is the front rail and the front crossbeam. Therefore, the subsystem can be further decomposed into a front rail component as shown in Fig. 9.

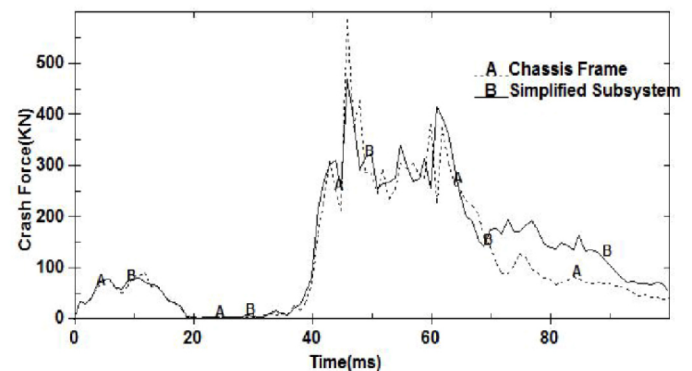


Figure 10. Crash force history comparison of the chassis frame model and the subsystem model

With the space decomposition, the complex problem of chassis frame is simplified: the design domain is the front