# Crashworthiness Design of Vehicle Structures via Equivalent Mechanism Approximations

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## Abstract

*A new method for crashworthiness optimization of vehicle structures is presented. In the new method, early design exploration is done by the optimization of an equivalent mechanism approximating a vehicle structure. An equivalent mechanism (EM) is a network of rigid bodies connected by prismatic and revolute joints with special nonlinear springs. These springs are tuned to mimic the force-displacement characteristics of thin-walled beams often found in the vehicle body structures. The EM models can be regarded as a super-set of lumped models and thus they are capable of providing better insight to the design issues. Proper selection of the nonlinear spring parameters is essential to successful implementation of the EM models. Identification of the spring parameters involves pre-compilation of databases of the crash characteristics of frequently used structures via LS-DYNA simulations. The pre-compiled databases and EM models are then used in the initial design phase in order to explore the crash deformation patterns and identify the good crash mode (CM). Once the good crash mode is identified, it becomes the target for detailed design stage which uses higher accuracy LS-DYNA models of the vehicle structures. A case study involving design optimization of the mid and lower rails of a vehicle subjected to frontal crash test conditions is presented. The case study demonstrates the effectiveness of the proposed method.*

## Introduction

Passenger vehicle crashworthiness is one of the essential attributes that vehicle designers strive to improve in order to satisfy government imposed regulations as well as making the vehicles attractive to potential customers in nowadays' highly competitive markets. However, crashworthiness is a difficult attribute to satisfy because of inherent trade-off between the need to cushion the impact and absorb the crash energy, which calls for a compliant structure and the need to prevent excessive deformation, which calls for a stiff structure. Achieving crashworthiness performance targets through a lightweight and cost efficient structure is also a requirement. Thus crashworthiness design is an area where optimization can be very beneficial.

However, vehicle crash simulations using nonlinear finite element (FE) models is often so computationally demanding, that many optimization techniques are unsuited for the task. The only practical techniques are those that emphasize obtaining a good design while utilizing only a few iterations [1-7]. Such techniques generally try to exploit any available information about the design problem in order to decrease the need for performing much iteration. Another approach is the use of reduced order models for approximate design estimations [8-13]. However, there are persistent difficulties in reduced order models concerning fidelity and ease of realization into a final design.

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The need for efficient, accurate and physically understandable approximation models motivated the idea of equivalent mechanism (EM) models [14]. The EM models are advanced reduced order models, in which the vehicle structure is lumped into equivalent springs and only the main masses (such as engine, passenger compartment …etc.) are considered. In the EM models, every main structural member is represented by a set of rigid masses connected through prismatic and revolute joints (Fig. 1). The joints have special nonlinear springs that are tuned to mimic the collapse behavior of the structural members.

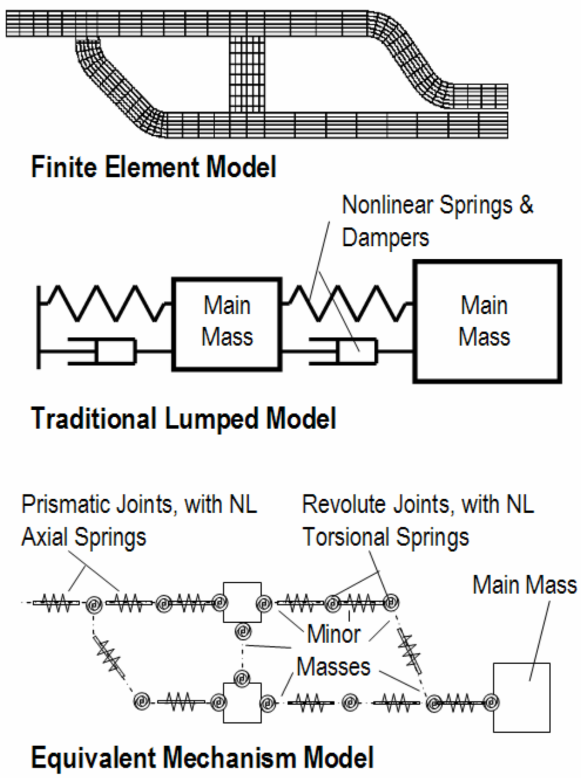


**Fig. 1. A finite element Model of a vehicle sub-structure and its corresponding lumped parameter and equivalent mechanism models**

The main idea in [14] was to do most of the optimization on the EM model rather than the FE model, thereby providing quick design insight and considerable savings on computational time. However, the algorithm presented in [14] encountered difficulty in the realization of the optimum mechanism into a real structure, which motivates the current work presented in this paper.

This paper starts with a brief literature review that motivates the current work, followed by explanation of the equivalent mechanism models and their implementation within a design context. Then, a case study involving frontal crash of a vehicle substructure is presented to demonstrate the effectiveness of the proposed approach at cutting down the computational cost for attaining good crashworthy structures.

## Equivalent Mechanism Approximations

The main idea in the EM approximations is that the main structural members of the vehicle frame, which are typically modeled using plate or shell elements in FE models, can be approximated as sets of rigid masses connected by prismatic and revolute joints that have special nonlinear springs (Fig. 1). The deformation resistance behavior of the springs is chosen to capture the behavior of the structural members. The EM models are then solved using conventional numerical simulation algorithms for mechanisms, thereby providing an efficient estimation of the vehicle structure behavior.

To characterize the nonlinear springs, a study of the deformation resistance forces and moments of thin-walled structural members is conducted [15]. The study involved many nonlinear FE simulations of axial crushing, bending and twisting of thin-walled box and hat sections. Typical deformation resistance curves for box and hat sections are provided in Fig. 2. It is observed that the overall deformation resistance behavior of thin walled structural members is similar in pattern and is characterized by:

- Deformation resistance rises quickly while still in the elastic stage (small deformation).
- Deformation resistance reaches a peak (usually near the onset of plate buckling) then collapses.
- Deformation resistance approaches a steady value as deformation keeps progressing.

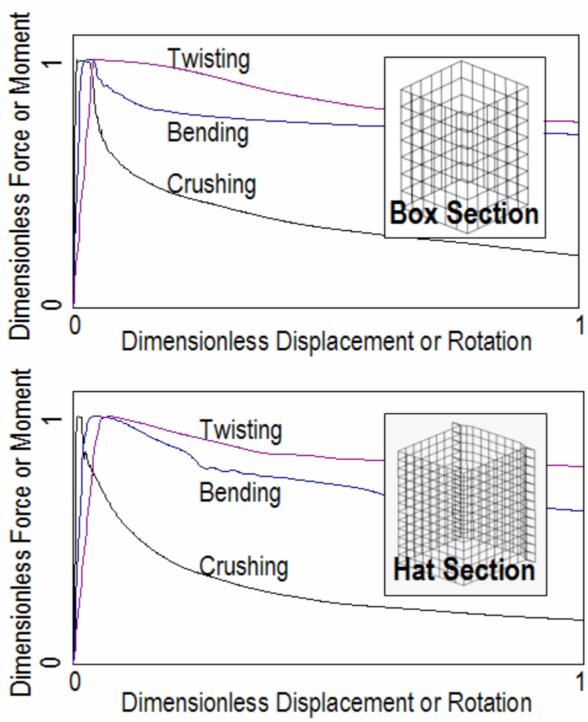


**Fig. 2. Typical deformation resistance curves for box and hat sections**

These characteristics that are observed in nonlinear FE simulations using LS-DYNA are also in agreement with reported experimental observations [3, 16] as long as the considered members are short enough so that no multiple folds of the sheet metal are formed.

Based on these observations, the nonlinear spring equation is made up of four regions (Fig. 4) that are blended together using sigmoid function [17], which is a continuous version of the hard limit function and provides a smooth blend of the deformation resistance during different stages of structural collapse. The spring force (or moment) is thus expressed as:

$$F_k = sig_1\, F_1 + sig_2\, F_2 + sig_3\, F_3 + sig_4\, F_4 \quad \textbf{(1)}$$

where:

$F_k$ is the total spring force

$F_1, F_2, F_3, F_4$ are the deformation resistance values for zones 1, 2, 3, 4 respectively (Fig. 3).

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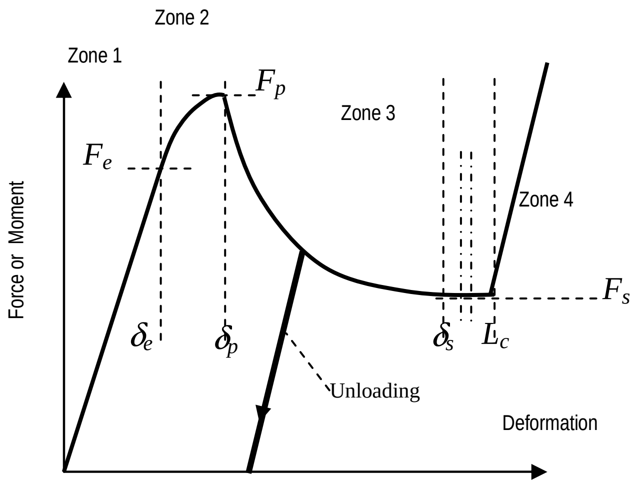


**Fig. 3. EM nonlinear spring behavior and main curve parameters**

Zone 1 represents the linear elastic behavior observed for small deformations. Zone 2 represents a quadratic approximation of the peak deformation resistance, then zone 3 represents an exponential collapse in the deformation resistance that approaches a steady state value. Zone 4 represents the high stiffness after crushing the full deformable length. The maximum deformable length $L_c$ is estimated from the length, geometry and of connectivity of the represented structural member. Estimation of the other tuning parameters ($F_e$, $\delta_e$, $F_p$, $\delta_p$, $F_s$ and $\delta_s$) is performed by building up databases of FE simulations for the design pool of structural members. In [15], databases of FE simulations are built using LS-DYNA for different sizes of box and hat sections. Values for the tuning parameters ($F_e$, $\delta_e$, $F_p$, $\delta_p$, $F_s$ and $\delta_s$) are identified such that they minimize the error in behavior between FE and the nonlinear spring. A surrogate model is then used to interpolate values for the tuning parameters for any dimensions of the structural member. It is thus possible to quickly estimate the values of the nonlinear parameters once the physical dimensions of the structural member are known.

## Optimization Algorithm

The optimization algorithm is shown in the flow chart in Fig. 4. The mechanism optimization is performed as a series of guess-and-evaluate. That is, the design variables are those representing the physical dimensions of the cross-sections. After a guess of the cross-section dimensions is made, those dimensions are looked up in the database of cross-sections (pre-compiled via LS-DYNA) to determine the equivalent nonlinear spring parameters. Then the EM models are used to estimate the crashworthiness performance (instead of the more expensive nonlinear FE). This form of simulation based optimization is well suited for derivative free global optimizers such as genetic algorithm (GA) [18, 19].

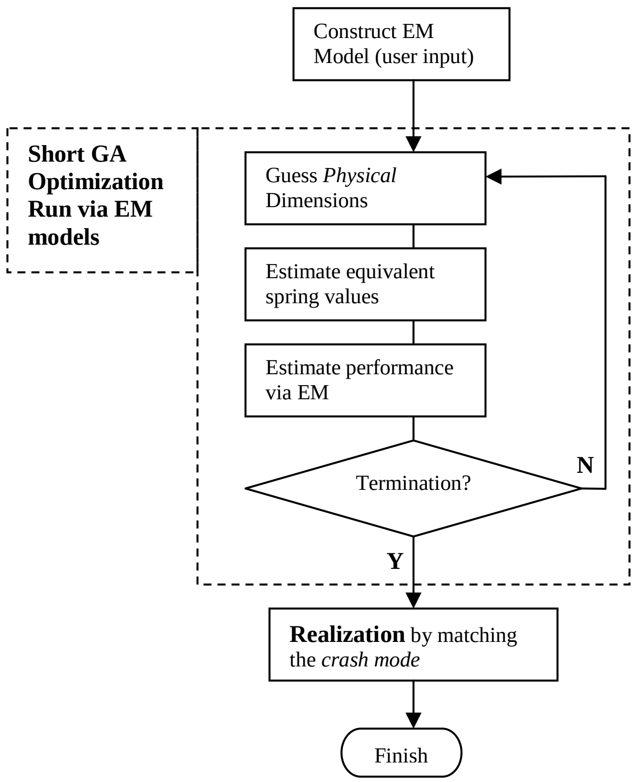


**Fig. 4. Flow chart of optimization algorithm**

Keeping in mind that the ultimate objective is to design the structure not the mechanism and that EM models are not very accurate, there is no point in making a long GA run for fine optimization of the mechanism. Instead, the GA run only provides a suggestion for the dimensions as well as the crash mode. A more rigorous study of CM is performed in [20], however, in this paper the crash mode (CM) is perceived as the sequence and extent of deformation in the different structural members during the crash event. As such, observing the CM allows the designer to manually tune the suggested dimensions by GA to match the deformation pattern observed in the EM model. The realization is thus performed using a few number of FE simulations (typically less than 10) and the realized design is usually of high quality.

## Case Study: Vehicle Substructure

A design case study is presented. The object of the study is a vehicle sub-structure comprised of the mid and lower rails, whilst the vehicle is subjected to full-lap frontal crash. This is the same design example of the second case study in [14]. The nonlinear FE and the corresponding EM models are shown in Fig. 5. There are 11 design variables in the problem, which are: $h_1$, $b_1$, $h_2$, $b_2$,that correspond to the height and breadth of the mid and lower rails respectively, and $t_1$, $t_2$, $t_3$, $t_4$, $t_5$, $t_6$ and $t_7$ that correspond to the box section thicknesses in zones 1 through 7 respectively. The design objective is to minimize the structural weight while preventing the crash deformation from exceeding allowed limits in both the frontal and rear zones of the sub-structure.

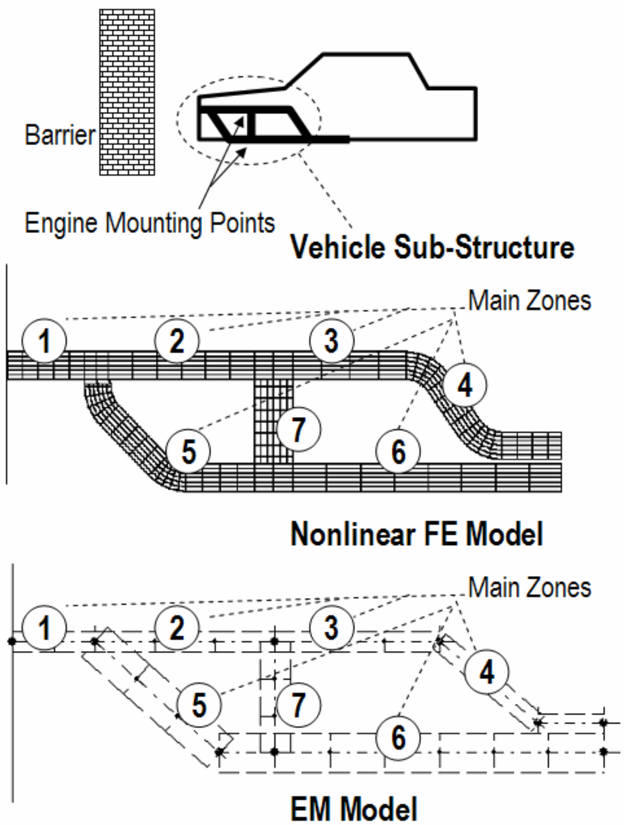


**Fig. 5. Nonlinear finite element and EM models of the vehicle sub-structure of the case study**

Design constraints are defined in the negative null form (i.e. less than zero denotes a feasible design). A constraint ($g_1$) represents the crush allowance in the frontal portion of the structure, while a constraint ($g_2$) represents admissible intrusion into the passenger compartment. Allowance on $g_1$ is typically larger than $g_2$. Therefore, a good design is expected to have sufficient compliance in its frontal portions to absorb the crash energy, but have enough stiffness in the rear portion to prevent excessive intrusion into the passenger compartment, all while having a light structural weight ($f$).

The best obtained design in [14] is used as a benchmark for comparison. By employing the optimization algorithm (Fig. 4), the results are shown in Table 1. It can be seen that the design suggested by GA is ultra-light, but infeasible. However after proper realization, through crash mode matching the resulting design is feasible and 10.5% better (lighter weight) than the previous best result. The crash mode matching was accomplished through 6 iterations that only required 6 FE simulations.

**Table 1 Case Study Results**

| | New Optimization Algorithm | | | Bench Mark* |
|---|---|---|---|---|
| | Estimated by EM | Actual as Realized | Matched to good CM | |
| $h_1$ (mm) | 50.00 | 50.00 | 70.00 | 60.40 |
| $b_1$ (mm) | 110.70 | 110.70 | 90.00 | 99.06 |
| $h_2$ (mm) | 50.00 | 50.00 | 90.00 | 119.74 |
| $b_2$ (mm) | 43.70 | 43.70 | 60.00 | 41.34 |
| $t_1$ (mm) | 1.78 | 1.78 | 2.20 | 1.00 |
| $t_2$ (mm) | 0.60 | 0.60 | 1.40 | 0.60 |
| $t_3$ (mm) | 1.01 | 1.01 | 1.10 | 0.63 |
| $t_4$ (mm) | 1.73 | 1.73 | 3.60 | 4.53 |
| $t_5$ (mm) | 4.60 | 4.60 | 3.60 | 4.52 |
| $t_6$ (mm) | 4.60 | 4.60 | 4.20 | 4.46 |
| $t_7$ (mm) | 1.20 | 1.20 | 1.40 | 4.60 |
| $f$ (kg) | 17.65 | 15.36 | 22.81 | 25.50 |
| $g_1$ (mm) | -44.70 | -94.10 | -11.00 | -55.20 |
| $g_2$ (mm) | -4.50 | +360.80 | -2.40 | -2.90 |

* Results are referenced to the second case study in [14]

## Conclusion

A new approach for crashworthiness optimization of vehicle structures is presented. The approach employs special equivalent mechanism (EM) models to approximate the behavior of the vehicle structure during the crash event. The EM models have the advantage of good approximation whilst preserving the overall sense of the vehicle motion thus allowing for simulation based optimization. A case study of a vehicle sub-structure is used to demonstrate the proposed design algorithm. The results of the case study showed the capability of the new algorithm to get good designs while utilizing very few number of nonlinear finite element simulations thereby significantly reducing the computational cost to complete the optimization task.

Future extensions of this research would include models to incorporate the effect of panels, which are non-negligible structural elements. Future extensions would also include studies of bigger and more complex vehicle structures.

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