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CRASH MODE ANALYSIS OF VEHICLE STRUCTURES BASED ON EQUIVALENT MECHANISM APPROXIMATIONS

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ABSTRACT

This paper presents a method for crashworthiness design of vehicle structures based on analyses of crash mode (CM), a sequence of axial crushing, twisting, and transversal bending during a crash event. The method emulates a process commonly called "mode matching" by vehicle designers, where the crash performance of a structure is improved by manually modifying the design until its CM matches the one the designers deem as optimal. Instead of relying on the insight of experienced designers, an optimal CM of a structure is identified via the numerical sampling of the design space of an "equivalent" mechanism, which approximates the structure as a network of rigid beams joined by prismatic and revolute joints with special nonlinear springs. The sampled designs of the mechanism are first classified according to their CM, and a finite element (FE) model of a baseline design is then manually modified to match the best CM. A case study on a 2D vehicle front substructure subjected to full-lap crash demonstrated that the method yield a better design than numerical optimization with a far less number of nonlinear FE simulations.

KEYWORDS

Vehicle crashworthiness, crash modes, design optimization, physical surrogate models, robust design.

1. INTRODUCTION

Crashworthiness, an ability of structures to absorb impact energy to protect occupants in the event of crash, naturally is one of the most important design criteria for passenger vehicle bodies. Since vehicle manufacturers must meet the government regulations of standard crashworthiness performances to sell their products, it is usually the first design criteria considered during the design process before other issues, such as noise and vibration. Despite its priority, crashworthiness a difficult attribute to satisfy since it essentially requires the structure to be both stiff at some areas (to prevent intrusions to the passenger cabin and fuel system), and soft at other areas (to absorb the impact energy), all while being lightweight and cost efficient.

Nonlinear finite element (FE) simulations are predominant computational tool employed by vehicle designers for crashworthiness design. The downside of nonlinear FE simulations is that they require enormous computational resources when full vehicles or large sub-structures are considered. This practically inhibits the use of numerical optimization techniques except for special cases, where small substructures are considered or fast approximation models, such as response surfaces, are employed.

Experienced vehicle designers, on the other hand, improve the crash performance of a structure without a large number of nonlinear FE simulations, by observing the *crash modes* of the structure. A crash mode (CM) is a sequence of axial crushing, twisting, and transversal bending in a structure during a crash event. Viewing a CM as a strategy for energy absorption, the experienced designers utilize the process commonly called "mode matching," where a structure is modified until its CM matches the one the designers deem as optimal based on their experiences.

For instance, consider the mid-rail of a vehicle subject to full-lap frontal crash shown in Fig. 1, where crush modes of two different designs are illustrated as sequences of figures. In design A, zone 1 fully collapses first (during the first 40 milliseconds) then zone 3 partially deforms all while zone 2 does not deform much. In Design B, zone 1 only partially collapses, followed by a sever bending in zones 2 and 3. These two designs have totally different energy absorption characteristics, as indicated by the differences in their crush modes. Further, one can guess design A is better, *i.e.*, absorb more energy with less intrusion at zone 3, than design B due to the occurrence of an axial crushing (which tends to absorb more energy than twisting or transversal bending) immediately after the impact. To match the crash mode of design B to the one of design A, an experienced designer would reduce the stiffness of zone 1 so it would crush easily, and then increase the stiffness of zones 2 and 3 so they would not bend as much.

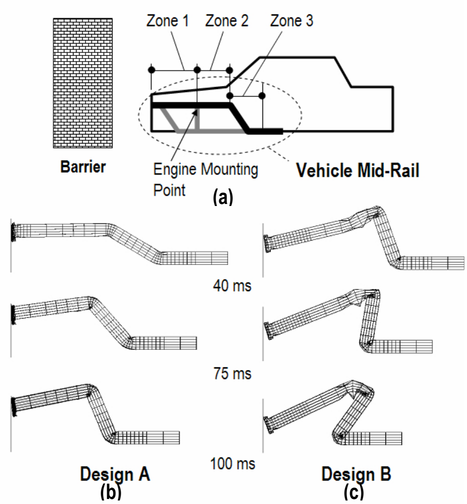


Figure 1 Vehicle mid-rail structure subject to full-lap frontal crush: (a) schematic layout, (b) Design A that represents one crash mode and (c) Design B that represents a different crash mode

The method presented in this paper attempts to emulate this process of mode matching, with an over arching goal of improving the efficiency of crashworthiness optimization. Instead of relying on the insight of experienced designers, an optimal CM of a structure is identified via the numerical sampling of the design space of an "equivalent" mechanism (EM), which approximates the structure as a network of rigid beams joined by prismatic and revolute joints with special nonlinear springs (Hamza and Saitou 2003). The sampled designs of the mechanism are first classified according to their CM, and a finite element (FE) model of a baseline design is then manually modified to match the best CM. A case study on a 2D vehicle front substructure subjected to full-lap crash demonstrated that the method yield a better design than numerical optimization with a far less number of nonlinear FE simulations. It is also demonstrated that a robust design insensitive to the variations in design variables can be identified through the observations of the classified CMs.

The rest of the paper is organized as follows: a review of relevant literature on crashworthiness design, a brief description of the equivalent mechanism (EM) model of a vehicle structure developed in our previous work, a representation of crash modes using the EM model, a case study of a 2D vehicle front frame substructure. The paper concludes with a summary and future work.

2. RELATED WORK

While a significant amount of research is directed to crashworthiness, they can be classified into two categories: topology optimization and parametric optimization.

Topology optimization poses the design problem as the optimal allocation of a structural material within a fixed design domain, without assuming a predefined connectivity (*i.e.*, topology) among structural members. Examples of structural topology optimization for crashworthiness include the work by Mayer, Kikuchi and Scott (1996), Gea and Luo (2001) and Soto (2001). The results of topology optimization, however, lack the critical details to be a manufacturable design, limiting their use only for the generation of alternative design concepts.

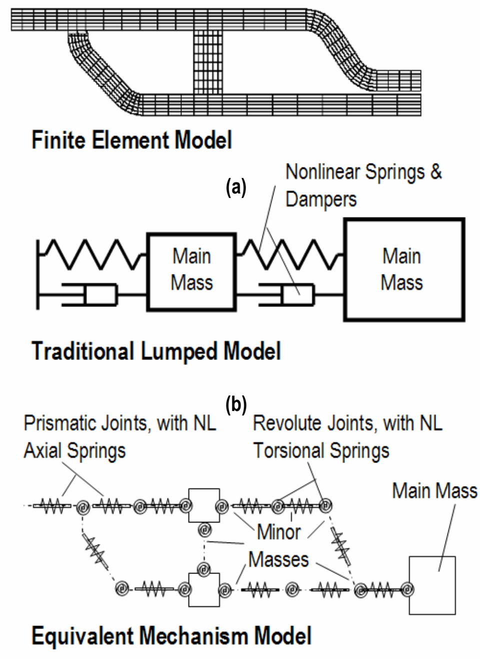
Parametric optimization on the other hand poses the design problem as the optimization of the parameters of a predefined shape, *e.g,.* dimensions. Since the result of parametric optimization can be detailed enough to be a manufacturable design, it had been applied to small, but realistic vehicle substructures (Han and Yamada, 2000; Kurtaran *et. al.*, 2001; Chen, 2001). When full vehicles or large sub-structures are considered, however, the use of nonlinear FE simulation becomes impractical due to the excessive computational time. For example, Yang *et al* (2001a) reported an example where two iterations of optimization required 512 computers running in parallel for 72 hours.

To improve the computational time of crash simulation, several approximation models are proposed, including phenomenological models such as response surface methods (RSM) (Yang *et. al.* 2001b) and neural networks (NN) (Omar *et. al.* 2000), and reduced-order models such as reduced lattice and lumped parameter models (White *et. al.* 1985; Bennett *et. al.* 1991; Soto and Diaz, 1999; Kim and Arora 2003).

Since phenomenological models represent the input-output relationship of a crash simulation by "observing" the sampled input-output pairs, it requires many runs of nonlinear FE simulation to achieve adequate accuracy over ranges of parameter values. Also, its phenomenological nature prohibits the physical interpretation of the resulting parameter values, providing little insights to the designers. The reduced order models, on the other hand, does not require nonlinear FE simulation to build and also allows some physical interpretations. Its drawback, however, is the difficulty in realizing the optimal reduced order model as the detailed FE model, which itself is an optimization problem involving nonlinear FE simulations.

To overcome this drawback of reduced order models, the authors have proposed an equivalent mechanism (EM) model (Hamza and Saitou, 2003), where a vehicle structure is approximated as a network of imaginary rigid beams connected by prismatic and revolute joint with nonlinear springs, which mimics force-displacement relationship of thin-walled structures subject to axial crush and transversal bending. Dissimilar to the conventional lumped parameter model (Fig. 2 (b)) whose springs, dampers and masses do not have direct correspondence with the structural members, the EM model (Fig. 2 (c)) approximates each structural member with a rigid link with several linear and revolute joints, representing the overall geometry of the structure.

By choosing the cross sections and wall thicknesses of each beam in a structure, the crash performance of an EM model can be optimized through the multi-body dynamic simulations with the corresponding nonlinear spring properties chosen form a database of pre-analyzed FE models of thin-walled beams. Since the spring properties are chosen among the ones available in the database, the realization of an EM model as a FE model is just a matter of replacing each EM member with the corresponding FE member in the database (Hamza and Saitou, 2003).



(c)

Figure 2 (a) finite element model of a vehicle sub-structure and its (b) lumped parameter and (c) equivalent mechanism models

In addition to the ease of realization, the high geometric fidelity of the EM model makes it an ideal choice for the analyses of crush modes, which is not possible with lumped parameter models. The next section provides a brief overview of equivalent mechanism model.

3. EQUIVALENT MECHANISM MODEL

The main idea of approximating a vehicle structure as a mechanism is based the observation that 1) majority of members in a vehicle structure during a crash event undergo either axial crush or transversal bending, and 2) beams subject to axial crush and transversal bending can be seen as rigid links (with masses) connected by a prismatic joint and a revolute joint, respectively. By attaching these joints to special nonlinear springs that mimic the behavior of thin-walled structures subject to crush and bending, the resulting mechanism can approximate the overall deformation behavior of a vehicle structure by the conventional multi-body dynamic simulation.

To characterize the nonlinear springs at the joints, a study of the deformation resistance forces and moments of thin-walled structural members is conducted (Hamza and Saitou, 2004a). The study involved many nonlinear FE simulations of axial crushing, twisting and transversal bending of thin-walled box and hat sections. Typical deformation resistance curves for box and hat sections are provided in Fig. 3. 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.

These characteristics are also in agreement with reported experimental observations (Han and Yamada, 2000) and (Koanti, and Caliskan, 2001), provided that the considered members are short enough so that no multiple folds of the sheet metal are formed.

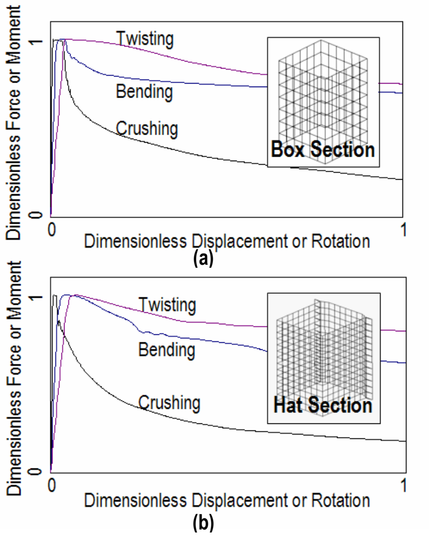


Figure 3 Typical deformation resistance curves for (a) box and (b) hat sections (Hamza and Saitou, 2004a)

The details of the force-displacement curve of the nonlinear springs are provided in (Hamza and Saitou, 2003) and updated in (Hamza and Saitou, 2004b). While this paper presents the results by two-dimensional EM models, the idea naturally extends to 3D, which is currently under development.

4. REPRESENTATION OF CRUSH MODES

The *mode matching* is a process adopted by experienced vehicle designers to improve the crash performance of a structure without a large number of nonlinear FE simulations. In essence, it consists of the following steps:

1. Observe the crush behavior of the current design, typically via an animation of the FE simulation results. If the current design exhibits what the designer considers as an ideal crush mode, stop the process move to more detailed design. Otherwise go to step 2.
2. Modify the design by increasing/reducing plate thickness or adding/removing reinforcements, to bring the current crush mode closer to the ideal crush mode. Go to step 1.

While practically very effective, the process heavily relies on the insights of experienced designers as to what crush mode is ideal for a given structure. In this paper, instead, an EM model of a structure is utilized to efficiently seek for an ideal crush mode via the numerical sampling of the design space of the EM model, followed by a classification of the sampled designs according to their crush modes.

To facilitate the objective classification, a mathematical representation of a crush mode is developed, which specifies the following:

- Amount and type of each major deformation (*eg*., axial, twisting, or bending) during an crush event.
- Locations of each deformation in the structure.
- Time of occurrences of each deformation.

Namely, a crush mode is defined as a hyper-matrix:

$$C = (c_{ijk});\quad c_{ijk} \in [0,1], i \in T, j \in Z, k \in D \tag{1}$$

where c_{ijk} is a normalized amount of deformation of type k in zone j at time i, T is a set of discrete event time, j is a set of zones in the structure, and D is a set of deformation types.

Given the definitions of sets T, Z, and D, the crush mode C of a structure can be computed from the result of the dynamic simulation of a crush event. For example, let us consider the crush modes illustrated in designs A and B in Fig 1. Tables 1 and 2 show the crush modes as defined in Equation (1) obtained from the results of nonlinear FE simulations, where $D = \{\text{axial, bend}\}$, $Z = \{1, 2, 3\}$ as defined in Fig. 1, and $T = \{1, 2, 3, 4\}$ with 1 being the first 25 milliseconds after the impact *etc.* The amount of deformation shown in Tables 1 and 2 are normalized by the total length of the zone (if axial), or by 180^o (if bending). It can be seen from Tables 1 and 2 that design A undergoes a significant axial collapse in zone 1 (90% of the length of zone 1) and almost no deformation in zone 2, while design B experiences much less axial collapse in zone #1 (approximately 48%) but an appreciable bending in zone 2. One could also observe that the bending in zone 3 was much more in design B than in design A.

Table 1 Crush mode C of Design A in Fig. 1

Time	Zone 1		Zone 2		Zone 3	
Event	Axial	Bend	Axial	Bend	Axial	Bend
1	0.55	0.00	0.00	0.00	0.00	0.00
2	0.40	0.00	0.00	0.00	0.01	0.08
3	0.00	0.00	0.00	0.01	0.02	0.23
4	0.00	0.00	0.00	0.00	0.01	0.19

Table 2 Crush mode C of Design B in Fig. 1

Time	Zone 1		Zone 2		Zone 3	
Event	Axial	Bend	Axial	Bend	Axial	Bend
1	0.45	0.00	0.03	0.01	0.00	0.00
2	0.03	0.00	0.16	0.06	0.01	0.29
3	0.00	0.00	0.00	0.08	0.01	0.32
4	0.00	0.00	0.01	0.05	0.00	0.34

While it is possible to calculate the crush modes C directly from the results of a nonlinear FE simulation as illustrated in Tables 1 and 2, the determination of deformation amount becomes somewhat subjective due to the difficulty in determining the exact location of reference to measure the deformation amount. With an EM model, on the other hand, the calculation of the crush mode is a straightforward task since the deformations occur only at the joints at pre-specified locations. The next sections present a case study that demonstrates the use of EM model to quickly identify an ideal crush mode for mode matching.

5. CASE STUDY

This section discusses a case study with a vehicle sub-structure comprised of the mid and lower rails subjected to full-lap frontal crash (Hamza and Saitou, 2003) shown in Fig. 4. Design variables are: h_1, b_1, h_2, b_2, which 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, t_7, which 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. The problem is summarized as follows (all units are in mm):

minimize: f = structural weight (2)

subject to:

$$g_1 = \delta_{13} - 950 \le 0 \qquad (3)$$

$$g_2 = \delta_4 - 100 \le 0 \qquad (4)$$

$$40.0 \le h_1, b_1, h_2, b_2 \le 150.0 \qquad (5)$$

$$0.6 \le t_1, t_2, t_3, t_4, t_5, t_6, t_7 \le 4.6 \qquad (6)$$

where δ_{13} is the total displacement in zones 1 – 3 and δ_4 is the displacement in zone 4 in Fig. 4.

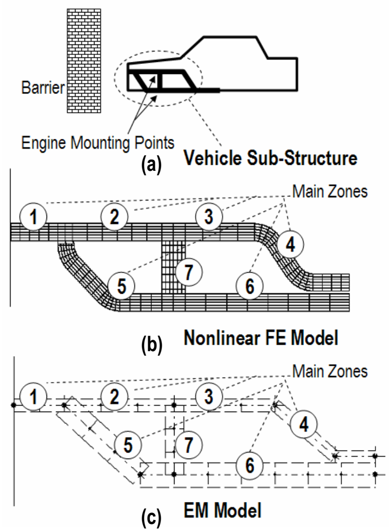


Figure 4 Vehicle sub-structure of the case study: (a) schematic layout, (b) nonlinear finite element model and (c) EM model

5.1. Mode matching for optimal design

Instead of numerical optimization of the EM model as done in (Hamza and Saitou, 2003), the design variables are randomly sampled within the ranges to obtain 50 sets of design variables. For each set of design variables, the crush mode *C* is calculated using an EM model. The resulting crush modes of all 50 designs are shown in Appendix, where the total crash event time of 100 ms is discretized into 5 event times. By visual inspection, the crush modes of the 50 samples are classified into 5 major types, as shown in Table 3.

To verify that these five mode types can be realized by FE models and they exhibit similar crush performances (so one can know a good mode in EM is also good in FE), nonlinear FE simulations are performed for the designs in the neighborhood of the EM models of each mode type. The results are shown in Fig.5, where it is qualitatively confirmed that similar types of crush modes exist in FE models with similar crush performances.

Table 3. Classification of major identified crush modes and representative sample designs

Crash Mode Type	Distinctive Qualities in CM Quantitative Measure	Performance Characteristics	Sample ID
1	Complete axial crushing of zone 1, early and significant contribution by zone 2, little or no deformation in zones 3 and 4	Tightly feasible and light weight.	8, 17
2	Complete axial crushing of zone 1, delayed and insignificant contribution in zone 2, partial collapsing in zones 3 and 4	Both infeasible and heavy weight	9, 15, 31, 33, 40, 45
3	Complete axial crushing of zone 1, partial contributions by zone 2 and/or zone 3, little or no deformation in zone 4	Abundantly feasible and heavy weight	4, 26, 35
4	No axial crushing in zone 1, sever collapsing in zone 2, partial collapsing in zones 3 and 4	Either badly infeasible or just feasible but very heavy weight	16, 19
5	Partial axial crushing in zone 1, , partial contributions by zone 2 and/or zone 3, partial collapsing in zone 4	Either infeasible or just feasible but heavy weight	14, 21

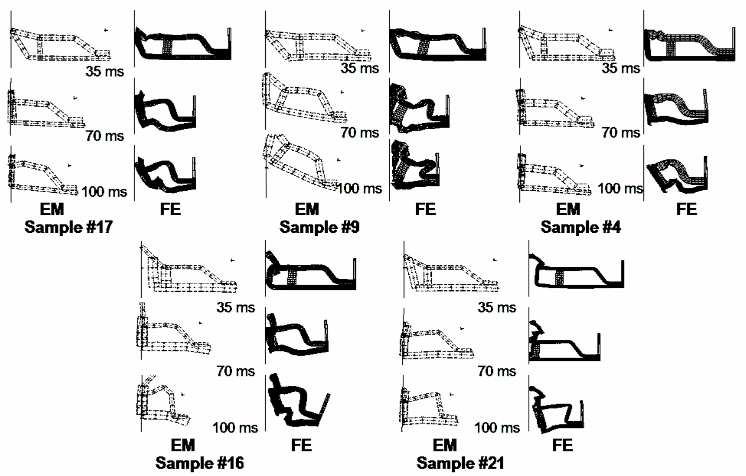


Figure 5 Five types of crush modes in Table 3 by FE and EM models

Table 4. Realization of sample 17 of an ideal crush mode (type 1)

	sample 17			Bench mark
	estimated by EM	ealized	mode matched	
h_1 (mm)	83.04	83.04	85.00	60.40
b_1 (mm)	124.37	124.37	125.00	99.06
h_2 (mm)	64.23	64.23	85.00	119.74
b_2 (mm)	65.15	65.15	70.00	41.34
t_1 (mm)	1.63	1.63	2.00	1.00
t_2 (mm)	1.62	1.62	1.40	0.60
t_3 (mm)	0.62	0.62	1.20	0.63
t_4 (mm)	2.73	2.73	4.00	4.53
t_5 (mm)	3.41	3.41	2.60	4.52
t_6 (mm)	2.64	2.64	4.60	4.46
t_7 (mm)	3.40	3.40	2.00	4.60
f (kg)	22.76	18.76	25.45	25.50
g_1 (mm)	-150.32	+341.90	-67.56	-55.20
g_2 (mm)	-30.50	+67.00	-1.80	-2.90

Crush mode type 1 in Table 3 is identified as an ideal crash mode, and an EM model (sample 17) of the type is realized to a FE model. As shown in Table 4, the realization produces an infeasible design, having a completely different crush mode. The realized design is then manually modified to match the crush mode to type 1. With an ideal crush mode fully visualized via an EM, mode matching is very easily done by making specific zones stronger or weaker. After *only* 6 nonlinear FE simulations, the resulting design is better than the best design reported in (Hamza and Saitou 2003), indicated as benchmark in Table 4, which required approximately 130 nonlinear FE simulations.

5.2. Mode matching for robust design

Due to the inherent variations manufacturing and crash conditions, it is highly desirable that the crush performance of a structure is robust, *i.e.,* the crush performance is insensitive to the uncontrollable variations in parameter values. However, the extreme computational cost of crash simulations renders the conventional method for robust design, (*e.g.*, Taguchi, 1986; Phadke, 1989) virtually impractical.

As a practical alternative, experienced designers utilize the mode matching process to achieve a robust design, by seemingly searching for a design that does not "jump" to another mode with small variations in the parameters. While simple, the method makes sense since 1) the changes in crush mode, when they occur, are caused by only minute changes in parameter values (*i.e.*, crush mode "jumps") and 2) the "jump" in crush mode is always associated with a dramatic change in the crush performances. By searching for a design that is "in the middle of" a high quality crush mode, therefore, a robust design can be identified with the sampled EM designs. For this, the following simple steps can be used:

1. **Identify a robust CM type:** Select a high-quality CM type with some allowances to the constraint values, so that minor performance variations do not cause constraint violations.
2. **Identify a robust design of the robust CM type:** Select a design of the CM type whose design variables conform the relationship to keep the relative strengths of the different zones that realize the selected CM type, so that small variations do not cause a jump of CM type.

By applying these steps, EM sample 4 in Table 3 is identified as a robust design. The simulation results of the realized FE model with the nominal values of the design variables are given in Table 5. The design has abundance on the constraints and the relative strengths of the zones agree with other observed designs in the same CM. As a comparison, Table 5 also shows a perturbation of the realization of EM sample 17, identified as a non-robust design by the reverse application of the above steps. The design, however, is tight on the second constraint and has the thickness t_1 larger (relative to t_2 and t_3) than the most designs of the CM type. Interestingly, the nominal performance of sample 17 shows it is a good design if robustness was not considered.

Table 5. Two designs identified as robust and as un-robust

	Identified as robust	Identified as un-robust
h_1 (mm)	124.00	85.00
b_1 (mm)	124.00	125.00
h_2 (mm)	75.00	85.00
b_2 (mm)	63.00	70.00
t_1 (mm)	0.70	2.15
t_2 (mm)	1.20	1.40
t_3 (mm)	1.80	1.20
t_4 (mm)	4.60	4.00
t_5 (mm)	2.30	2.60
t_6 (mm)	4.60	4.60
t_7 (mm)	3.90	2.15
f (kg)	27.01	25.58
g_1 (mm)	-61.38	-60.59
g_2 (mm)	-25.30	-6.01

To examine the robustness of these designs, the averages and standard deviations of objective functions f and constraints g_1 and g_2 are estimated via

Monte-Carlo simulation with sample size of 100, subject to ± 5% normally distributed variations on all the design variables. Although both designs perform well with their nominal values, Table 6 shows that their robustness of constraint values vary significantly, confirming that the design selected according to the above steps is in fact robust against the variations in the design variables. It should be noted that the selection of sample 4 (robust design) is merely based on the observation of the crush modes of the sampled EM designs, without using *any* FE simulations.

Table 6. Robustness of two designs identified as robust and as un-robust

		Identified as robust	Identified as un-robust
Nominal Values	f (kg)	27.01	25.58
	g_1 (mm)	-61.38	-60.59
	g_2 (mm)	-25.30	-6.01
Average of 100 Samples	f (kg)	26.96	25.56
	g_1 (mm)	-62.98	-58.34
	g_2 (mm)	-24.56	-1.41
Standard Deviation of 100 Samples	f (kg)	0.31	0.28
	g_1 (mm)	2.45	14.39
	g_2 (mm)	5.48	10.08
Number of Infeasible Designs in 100 Samples		Zero	40

6. CONCLUSIONS AND FUTURE WORK

This paper presented a computer emulation of mode matching, a design process employed by experienced vehicle designer for the improvement of structural crashworthiness. Rather than relying on the experiences of the human designers, a method was presented on the use of an equivalent mechanism model and random sampling to objectively identify an ideal crush mode unbiased by the human intuition.

The concept of a crush mode is mathematically defined as a hyper-matrix of a normalized amount of deformation of a type in a zone in a structure at a time during a crush event. The definition was utilized to classify the crush modes of the sampled EM. A case study with a vehicle front substructure demonstrated that mode matching can obtain better designs with significantly less computational efforts than conventional techniques. Further, robust designs can be identified by observing the crash modes of the sampled EM designs, without any nonlinear FE simulations.

While a number of open questions exist (*e.g.,* how to decide T, Z, and D most suitable for a given structure), the case study confirmed a strong potential of the mode matching process with EM models as an effective tool for simulation-based crashworthiness design. Future work includes the automation of crush mode classification, mode matching, and three-dimensional extensions. These efforts are currently in progress and will be reported in future opportunities.

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APPENDIX

Listing of the CM Quantitative Measure for Selected Designs

Type 1: Good Crash Performance and Light Weight

Simulation ID	F	G1	G2
8	24.21	-0.14	-0.03

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.49	0.00	0.26	0.12	0.00	0.00	0.00	0.00	0.00	0.07	0.00	0.05
2	0.44	0.00	0.08	0.05	0.00	0.00	0.00	0.00	0.00	0.03	0.00	0.02
3	0.00	0.06	0.23	0.25	0.00	0.00	0.00	0.00	0.09	0.10	0.00	0.07
4	0.00	0.04	0.11	0.18	0.00	0.00	0.00	0.00	0.14	0.08	0.00	0.00
5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.04	0.00	0.05	0.04	0.04

Simulation ID	F	G1	G2
17	22.76	-0.15	-0.03

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.93	0.00	0.30	0.13	0.00	0.00	0.00	0.00	0.00	0.13	0.00	0.00
2	0.00	0.25	0.31	0.47	0.00	0.00	0.00	0.00	0.06	0.47	0.00	0.00
3	0.00	0.12	0.05	0.52	0.00	0.08	0.00	0.00	0.13	0.13	0.00	0.00
4	0.00	0.09	0.00	0.13	0.00	0.15	0.00	0.06	0.05	0.05	0.06	0.00
5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00

Type 2: Bad Crash Performance yet Heavier Weight

Simulation ID	F	G1	G2
9	36.68	-0.39	0.138

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.89	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2	0.04	0.00	0.26	0.09	0.00	0.00	0.00	0.03	0.00	0.20	0.00	0.04
3	0.00	0.00	0.04	0.13	0.00	0.04	0.00	0.15	0.00	0.04	0.00	0.13
4	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.05	0.00	0.06	0.00	0.04
5	0.00	0.00	0.00	0.02	0.00	0.03	0.00	0.11	0.03	0.05	0.00	0.04

Simulation ID	F	G1	G2
31	31.93	-0.27	0.06

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.93	0.00	0.16	0.08	0.00	0.00	0.00	0.00	0.00	0.06	0.00	0.00
2	0.00	0.00	0.21	0.10	0.00	0.00	0.00	0.05	0.00	0.10	0.00	0.00
3	0.00	0.00	0.13	0.07	0.00	0.00	0.00	0.05	0.00	0.06	0.00	0.00
4	0.00	0.04	0.09	0.00	0.00	0.00	0.00	0.04	0.13	0.05	0.00	0.05
5	0.00	0.00	0.02	0.00	0.00	0.00	0.00	0.08	0.04	0.02	0.00	0.10

Type 3: Very Good Crash Performance yet Heavier Weight

Simulation ID	F	G1	G2
4	29.85	-0.15	-0.07

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.93	0.00	0.03	0.00	0.00	0.00	0.00	0.00	0.00	0.03	0.00	0.00
2	0.00	0.00	0.29	0.12	0.00	0.00	0.00	0.00	0.00	0.14	0.00	0.00
3	0.00	0.00	0.20	0.09	0.00	0.00	0.00	0.00	0.00	0.10	0.00	0.00
4	0.00	0.00	0.10	0.31	0.00	0.00	0.00	0.00	0.18	0.53	0.00	0.33
5	0.00	0.00	0.00	0.00	0.00	0.03	0.00	0.00	0.00	0.07	0.00	0.05

Simulation ID	F	G1	G2
35	31.85	-0.11	-0.03

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.69	0.00	0.00	0.00	0.04	0.00	0.00	0.00	0.00	0.00	0.00	0.00
2	0.23	0.00	0.00	0.05	0.20	0.02	0.00	0.00	0.00	0.00	0.00	0.09
3	0.00	0.00	0.00	0.15	0.37	0.05	0.00	0.00	0.00	0.04	0.00	0.24
4	0.00	0.00	0.00	0.00	0.00	0.05	0.00	0.04	0.00	0.22	0.00	0.18
5	0.00	0.00	0.00	0.02	0.00	0.00	0.00	0.03	0.00	0.00	0.00	0.02

Type 4: Not Crushing Zone1 - Very Bad Performance

Simulation ID	F	G1	G2
16	22.13	-0.08	0.175

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.00	0.00	0.37	0.17	0.00	0.00	0.00	0.00	0.02	0.17	0.00	0.00
2	0.00	0.00	0.22	0.13	0.00	0.00	0.00	0.00	0.00	0.11	0.00	0.04
3	0.00	0.07	0.10	0.25	0.00	0.00	0.00	0.00	0.17	0.32	0.00	0.07
4	0.00	0.08	0.03	0.14	0.00	0.22	0.00	0.09	0.07	0.10	0.12	0.04
5	0.00	0.17	0.00	0.09	0.00	0.28	0.00	0.14	0.00	0.05	0.28	0.06

Type 5: Partial Crushing of Zone1 - Bad Performance

Simulation ID	F	G1	G2
21	26.81	-0.14	0.137

Event	Zone 1		Zone 2		Zone 3		Zone 4		Zone 5		Zone 6	
	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend	Crush	Bend
1	0.64	0.00	0.13	0.06	0.00	0.00	0.00	0.00	0.00	0.07	0.00	0.00
2	0.06	0.02	0.25	0.10	0.00	0.00	0.00	0.00	0.00	0.14	0.00	0.00
3	0.06	0.30	0.25	1.29	0.00	0.00	0.00	0.11	0.24	0.39	0.00	0.05
4	0.03	0.00	0.00	0.08	0.00	0.00	0.00	1.96	0.00	0.02	0.28	0.05
5	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00