

Development of Front End Crash Structure for Lightweight Hybrid Electric Vehicle

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Abstract

Rooted in the £29 million Low Carbon Vehicle Technology Project (LCVTP), Coventry University has continued to conduct research into lightweight Body In White (BIW) design and lightweight crash structure development utilising structural optimisation for alternatively fuelled vehicles such as Hybrid Electric Vehicles (HEV). This paper explains how a lightweight HEV front end crash structure has been developed, refined and validated using numerical analysis. This is based on structural optimisation results, benchmarking of similar sized vehicles and previous experience of crash structure development.

Index terms— Body In White (BIW); Topology optimisation; Crashworthiness; Lightweight Hybrid Electric Vehicle (HEV); NCAP; NHTSA.

Introduction his paper will be concerned with presenting and discussing the development of a front end crash structure for a lightweight Hybrid Electrical Vehicle (HEV). This will be based on topology optimisation results, which has been published and discussed in Bastien (2010), ??astien The structural loadpaths to be used for the Body In White (BIW) and the crash structures were extracted from an initial design volume, i.e. Computer Aided Design (CAD) model, by employing Finite Element (FE) based linear static topology optimisation and New Car Assessment Program (NCAP) representative loading. Figure 1 illustrates the design volume used for the topology optimisation study which will be utilised as a reference point to summarise the topology optimisation throughout this section. The results of the topology optimisation revealed that the floor area, i.e. "5" in Figure 1, was subject to distinguishable changes, primarily as a function of the structural integrity of other components such as a battery pack, Christensen et. al (2011).

In addition, the generalised topology of the roof area ("1" in Figure 1) remained consistent throughout the entire study. The simple conclusion was that the topology of this area had converged. The converged roof topology was unconventional when compared to the roof bow structures of many modern day passenger vehicles, Christensen et. al. ??2011). There were however some concerns with respect to the structures ability to withstand the loads associated with a vehicle rollover, Christensen et. al. (2012) and Christensen et. al. (2012c).

Finally, the side area topology did, in line with the roof topology, also remain consistent, yet, a significant number of the models displayed a rather vague definition of the side area topology.

The results relating to the roof, floor and partially the side area topologies, which in essence make up the "safety cage" of the vehicle generally display relatively well defined loadpaths.

Thereby, the individual model topologies (of the safety cage) were found to be viable solutions which can be implemented in the BIW design in order to successfully withstand the dynamic crash loading scenarios. Nevertheless, this is solely based upon mechanical engineering judgements and is not at this point backed up by any calculations.

The above thus suggests that even though the "correct" method of representing the crash scenarios includes explicit (dynamic) modelling, useful results (load path extraction) can be obtained by utilising relatively simplistic linear static topology optimisation.

The key benefit of this approach was the low CPU cost, a typical calculation time of one topology optimisation model, was approximately 45 minutes, using 2 cores.

When the focus of attention was shifted to the front and rear area topologies ("3" and "4" in Figure 2) significant changes due to variations of force application angles and stiffness values were found, Christensen et. al. (2011).

The response of the topology optimisation seemed to be "triangulation", i.e. the widespread use of triangles within the geometry, i.e. design space. This made perfect sense from a linear static point of view, as the stiffest geometry in solid mechanics is a triangle. However, this raises serious concerns when the subsequent step is taken into dynamic loading, primarily because of the triangles resistance to buckling, which undoubtedly will have a negative influence on the crushability, and therefore the dynamic crash performance of these very vital areas, more specifically design of the crumple zone. This is evidently one of the major limitations of the linear static (implicit) solver and highlights the necessity for further steps in the development of topology optimisation algorithms, particularly with an emphasis on non-linear material behaviour.

The extend of this limitation will be further highlighted and analysed during the remaining sections of this paper.

With the brief summation of the topology optimisation complete, the focus of attention will now be aimed at developing the front crash structure of the vehicle using shape and size optimisation, with the basic loadpath definitions defined by the topology optimisation.

II.

2 Front Crash Structure Benchmarking

The development of the front end crash structure commenced with a benchmarking vehicles of similar size (external dimensions) and mass in order to define the performance requirements for the front crash structure.

3 a) NCAP HEV Target setting

The aim of this task was to define a target setting for the HEV front crash structure, in order to meet a 35mph rigid barrier impact (56.65km/h nominal) NHTSA (2012).

In order to do so, the first step of this study was to investigate the current state of art in vehicle's structural performance and understand how an "ideal" crash pulse could be obtained for a lightweight HEV.

Five vehicles were initially chosen for this study, primarily due to their structural layout and associated crash performance, courtesy of NHTSA testing, NHTSA (2011). The relevant data for the five chosen vehicles is listed in Table 1. In addition to the above justification of the selection of vehicles for comparison, a further justification can be made based on the above vehicle layouts. The first 3 (Fiesta, Cooper and FourTwo) are similarly sized to the proposed structure of this paper (external dimensions and mass values), whereas the Jaguar and the Honda were chosen in order to better understand the effects of a long front overhang.

Due to publishing restrictions only the Ford Fiesta will be presented in greater detail below.

The data available from the above NHTSA test reports, NHTSA (2011) were mainly focused on occupant injuries, with considerably less data available on the actual structural performance of the vehicles in question. In general, the Vehicle Acceleration Pulse (VAP) may be considered as an 'enabler' for reducing the severity of the occupant injuries, i.e. reducing VAP leads to a reduction in severity of occupant injuries. Other factors such as the restraint system does however also significantly influence the severity of occupant injuries. Due to the nature of the overall study, the VAP was nevertheless considered in isolation.

4 b) 2011 Ford Fiesta

The Ford Fiesta was the newest model year vehicle under investigation, and was one of the highest rated small vehicles tested by the Insurance Institute of Highway Safety (IIHS), IIHS (2012), offering a good performance benchmark target for the new vehicle design.

Newton's second law of motion was used to extract the VAP, equation (5), assuming that all the vehicles' mass remain coupled during the crash scenario.

5 F VAP m

(1)

In equation (??), 'F' is the force exerted on the vehicle (from the barrier), this was extracted from the NHTSA data NHTSA (2011), and 'm' is the vehicle test mass available from Table 1. Thereby the pulse can be obtained, Figure 3 represents the resulting VAP for the FORD Fiesta test. The first (local) VAP peak of approximately 15g ($g = 9.82 \text{ m/s}^2$) occurs at 18ms, this was caused by the initiation of the crush can. The highest VAP peak occurs at approximately 28ms, and was caused by the engine contacting the rigid wall. Between 30ms and 60ms, the main longitudinals (longits) collapsed, as well as the engine ancillary bay, giving rise to a relatively "horizontal" profile of the VAP. Around 60ms the wheel made contact with the sill, leading to a local increase in VAP, ultimately followed by the vehicle ride down.

From the test videos, NHTSA (2011), it was noticeable that the plastic deformation, i.e. structural damage was very much localised at the extreme front of the vehicle, with no visible deformation of doors or door apertures. This was collaborated by the test report, as no change in door aperture pre to post test was measured, and only

2mm difference in seat mounting positions were measured. This fact was consistent with the approach of using linear static topology optimisation, for the development of the passenger cell, as originally assumed.

The approach of the above analysis was also adopted for the remaining four vehicles listed above. As previously mentioned, these will however not be further addressed in this paper. The average vehicle acceleration was calculated by taking the total impact energy, defined as the integral of the contact force of test vehicle against the rigid wall and the vehicle motion, divided by the maximum intrusion.

6 d) Global Acceleration pulse target setting

It can be seen from the vehicles investigated, they are developed for several load cases. For a realistic front concept structure to be created from this investigation, both low and high-speed frontal impacts will be considered. No stiffness or NVH load cases will be assessed, nevertheless the structure will be developed with these load cases also in mind. The front end was developed to create a global vehicle pulse which will work for both impact load cases, with the targets outlined below.

Front Low Speed (FLS) damageability a.k.a. "Thatcham insurance rating" tests have recently been adapted to better represent real world crash scenarios related to insurance claims. The FLS load case therefore consists of a frontal impact at 15kph, with a 40% offset barrier, applied at an angle of 10° relative to the x-axis in Figure 1 and Figure 2.

This assess the cost of repair of the full vehicle, in which major structural damage is a significant concern, as repair costs (and thus vehicle insurance category) will be high. Consequently the parameter for the FLS scenario is no visible longit deformations. This can be quantified by setting a limit of all plastic strain to a maximum value of 2%, suggesting all damage is localised to the bumper beam and crush cans.

The high-speed frontal 35mph (FHS) for this concept structure is in essence the NCAP test, as previously discussed. This is of course based on occupant injury, however as seen in the tests the average accelerations are similar between all vehicles. the target, idealised global pulse shape metrics can therefore be visualised for the concept front end, as shown in Figure 5. The idealised pulse profile illustrated in Figure 5 is based on the following ideologies:

1. L Low speed damageability control. Using replaceable crush cans to absorb a specific amount of energy, equal to the FLS low speed Thatcham insurance rating test. The crush cans should be as stiff as possible, in order to ramp up acceleration as rapidly fast as possible without damaging the longitudinals.
2. T The acceleration should rapidly ramp up, in order to engage the occupant(s) in the restraint system early on. Thereby the occupants(s) acceleration will be coupled to the vehicle acceleration thereby minimising any lag between the two offering increased control.
3. A Acceleration peak duration. This should be kept at short as possible whilst maintaining the pulse shape, i.e. not bottoming out the crush space prior to all impact energy being absorbed.
4. T The peak acceleration should not to exceed 42g. This is the maximum value found during the benchmark study. In addition, this value is well below the 80g legal requirement.
5. T The crash duration should be as long as possible, in order to reduce the average accelerations as much as possible. This can be obtained by using at least 400mm of the available crush space in the front end of the vehicle, based on the target setting.

Lack of front end ancillaries simplifies the development of this pulse shape, as the interaction of the engine to the crash structure has less effect at the front end of the vehicle. The front-end stiffness will be dominated by the controlled crush of the main longitudinal members, and their interaction with the adjacent structure. The lack of front end ancillaries will however also affect the stiffness requirements of the occupant safety cell.

The crash investigations also showed that bulkhead intrusions are very small for most vehicles, again ensuring the deceleration distances for the occupants are maintained. This is a key target for the design and prediction restraint systems performance, and reduction of occupant injury. It can be assumed that this concept vehicle will be designed with a very stiff bulkhead with this in mind, so only the structure forward of the front bulkhead was be considered in the subsequent analyses.

7 III. Development of HEV Crash Structure

To create a front-end structure suitable for crash events, the data and information gathered from previous section has been be used to create the targets for the structural performance, as previously discussed.

Spring mass damper modelling was envisaged as a possible concept-modelling tool, however, further 110 39.8 14.0 545 investigation showed this modelling technique is mainly based on empirical test data of known sections / stiffness. With none of this data available to initially set up a 1D spring damper model, AISI (2012), 3D Finite element non-linear analysis will be used throughout to develop a front end design, using the industry standard solver code LS-Dyna.

To develop a suitable front end structure further research into structural deformation modes for crash energy management for very short front end vehicles was required, in addition to material investigations. This aimed to improve the structural efficiency of the design, whilst ensuring the viability in terms of manufacturing volume and methodology.

8 a) Additional benchmarking

The 5 vehicles investigated previously gave a lot of insight in to the mechanics of a crash event, however only the Smart is of real relevance in terms of BIW architecture. To further progress this project, it was deemed necessary to further investigate the forward structures of more modern vehicles with very short front ends.

To do this, the Peugeot 107 / Citroen C1 platform, Toyota IQ and Audi A2 were analysed. The Peugeot and Toyota utilises a "conventional" steel construction whilst the Audi A2 utilises an aluminium space frame.

9 b) Initial Concept

Based on the topology optimisation results, the interpretation thereof, previous crash design knowledge, ideas generated from the above benchmarking and crash analysis investigations, the primary loadpaths for the front end crash structure was defined as illustrated by Figure 6. Using the outcome of the initial shape and size optimisation the topology optimisation results were used to guide structural hard point locations and attachment points. This ensured the BIW structure created would be "compatible" with the "safety cage", and the primary load paths were maintained throughout the length of the vehicle. This led to the generation of the front crash structure illustrated in Figure 7. MacPherson strut setup was utilised using hard points identified from the topology optimisation, Christensen et. al. (2011). 2. The shock absorber turret was placed as far back as possible in order to maximise the crush distance. This was conducted with consideration of the vehicle dynamics. 3. Based on the benchmark and associated analyses the wheel to sill interaction combined with the subframe deformation were found to be key parameters of the smart car energy management. Consequently, these were incorporated into the crash structure, as they were likely to have a significant effect on the global crash pulse. 4. The length of the 1 longitudinals were maximised in order to increase the available crush distance. 5. Manufacturability was considered throughout the development of the crash structure. Therefore, the structure was designed using pressed steel parts.

The subframe was intended to be bolted on the BIW from underneath the vehicle, eliminating the need for the fixings to pass through the main longitudinal sections.

As structural efficiency was a key part of the design spot welds were used to create a stiff but lightweight structure. This meant that the structure was not designed to promote failure for energy management as a function of the assembly. Instead this was to be attained through geometry and a better use of material.

Based on the design illustrated by Figure 7 an FE model was created, as illustrated by Figure 8. The FE model was subsequently used to run a series of crash model analyses in order to correlate the model in addition to incorporating a series of adjustments in order to meet the criteria identified in section II of this paper.

On completion of the adjustments the performance of the Last Concept Iteration Model (LCIM) the global acceleration pulse was overlaid with the Smart Four Two and the Ford Fiesta pulses. This was done in order to compare the LCIM performance to vehicles with a similar structural configuration, including the class leader for occupant injury reduction. The pulses of the three vehicles in question is illustrated in Figure 9. Figure 9 demonstrates that the duration of the LCIM pulse is comparable to the Smart pulse, however the average accelerations are higher due to the increased mass of the concept vehicle that LCIM is based on. The peak accelerations of the LCIM is lower than that of both the Smart and the Ford. However, in this connection it should be mentioned that a standard SAE J211 CFC 180 filter, SAE (2012), was applied to the LCIM results in order to remove numerical noise from the curve.

The overalls shape of the LCIM pulse is similar to the Smart most likely as a result of the similar frontend configuration. The short front end of the two vehicles forces early engagement of the tyre to sill contact, which ramps up the accelerations from approximately 30ms, consequently a rear loaded pulse shape occurs.

It must be emphasised that the pulses in Figure 9 are based on simplified modelling of the concept structure in Computer Aided Engineering (CAE), relative to real world vehicles. This does for example result in an overly stiff front end structure (in CAE). This is because a significant amount of crush distance is used for pedestrian protection and low speed insurance rating impact tests in real world vehicles. Consequently the initial crush of the crash structure will have a considerably lower stiffness than the straight steel beam used for the LCIM concept model. This additional crush distance (low stiffness foam compression etc before structure beings to collapse) is likely to be the reason why the duration t_1 , Figure 9, is longer on the Smart and Ford vehicles when compared to the LCIM.

The duration of t_2 Figure 9, for the Ford Fiesta impact test is significantly longer than that of the LCIM. This reduces the average accelerations of the occupant(s), reducing the load transferred through the ??011), it is unlikely that a significantly better crash performance than that of the Smart FourTwo can be obtained.

IV.

10 Conclusion

This investigation has focused on the design of vehicle front-end structures for crashworthiness. This has been accomplished by initially benchmarking the crash performance of similar sized vehicles with excellent crashworthiness. This was done in order to fully understand the underlying mechanics of such structures. The investigation then focused on the first stage of the crash event, the structural behaviour of the vehicle itself, as an enabler for the reduction of occupant injuries during crash scenarios.

Five cars were benchmarked and compared, all of which were subject to the NCAP 35mph rigid barrier frontal crash. This demonstrated the fact that the pulse shape is highly dependant upon vehicle configuration. Vehicles with front mounted engines and front wheel drive were found to provide the best characteristic pulse. Structural targets were subsequently derived from analysing the NHTSA data. This was used to guide the concept design and development of the front structure of the LCIM. The peak allowable acceleration target was limited to 42g, as this was the peak of the benchmarked vehicles. The dynamic intrusion target for the vehicle was set to the interval of 454-482 mm. The target pulse shape to reduce occupant injury and improve restraint system loadings was defined in Figure 5.

Additional analysis of similar vehicle body structures including crush mode characteristics and materials was then completed before an initial crash structure (loadpath) was defined, taking the outcomes of the topology optimisation into account. This ensured that no discontinuation of loadpaths would occur throughout the length of the vehicle.

Next, shape and size optimisation was used to obtain initial information about the required crosssectional properties.

Initial model correlation and energy checks were carried out prior to refining the structure, thereby ensuring that the real world physics were represented in the FE model. The stiffness of the initial concept was found to be much too low. Therefore additional studies were conducted in order to develop the global structure stiffness and subsequently a suitable longitudinal crush mode for robustness.

The low speed performance of the structure was also investigated and the crush cans developed to meet the required energy absorbance. Further studies were conducted assessing the mesh convergence which proved the chosen mesh size of x 15 mm to be a good compromise between computational efficiency and result accuracy.

A full crash model incorporating the wheels, sub frame and suspension was created in order to capture the wheel to sill interactions. The incorporation of these assemblies allowed the gauge of the longitudinals to be reduced whilst maintaining the stability of the crush mode, thus improving the structural efficiency of the front-end structure, i.e. reducing mass whilst maintaining performance. This work showed the target pulse shape could not be attained using this vehicle configuration, due to the late interaction of the wheel and sill creating a load path, spiking the reaction force. Peak acceleration was found to be well under the 42g target at 36.6g. Comparing the results to those of similar configuration vehicles found that the shape of the LCIM pulse was comparable. The LCIM concept design could be further developed in order to meet all targets set.

V.

11 Next Steps

To further the engineering of the lightweight front end crash structure of this paper several aspects of the structure, concept development tools, modelling structure, boundary conditions and mass reduction should be revised, including:

A Additional structural research. Specific larger vehicles (external dimensions and mass) utilise tapered longitudinals, or swages, to obtain required crush characteristics, this was not found to be the case in the smaller vehicles analysed during the benchmarking exercise. Further studies could be conducted to better understand how the new front end structure could be utilised to control the pulse profile of the LCIM concept, which could lead to improved structural efficiency.

Topology optimisation. This step was conducted using linear static topology optimisation which clearly has severe limitations with respect to crashworthiness, as discussed in e.g. Christensen et al. (2012b) and Christensen et al. (2012c). A truly non-linear topology optimisation algorithm catering for large levels of non-linearity would drastically improve the starting point (primary loadpath definition) for the crash structure.

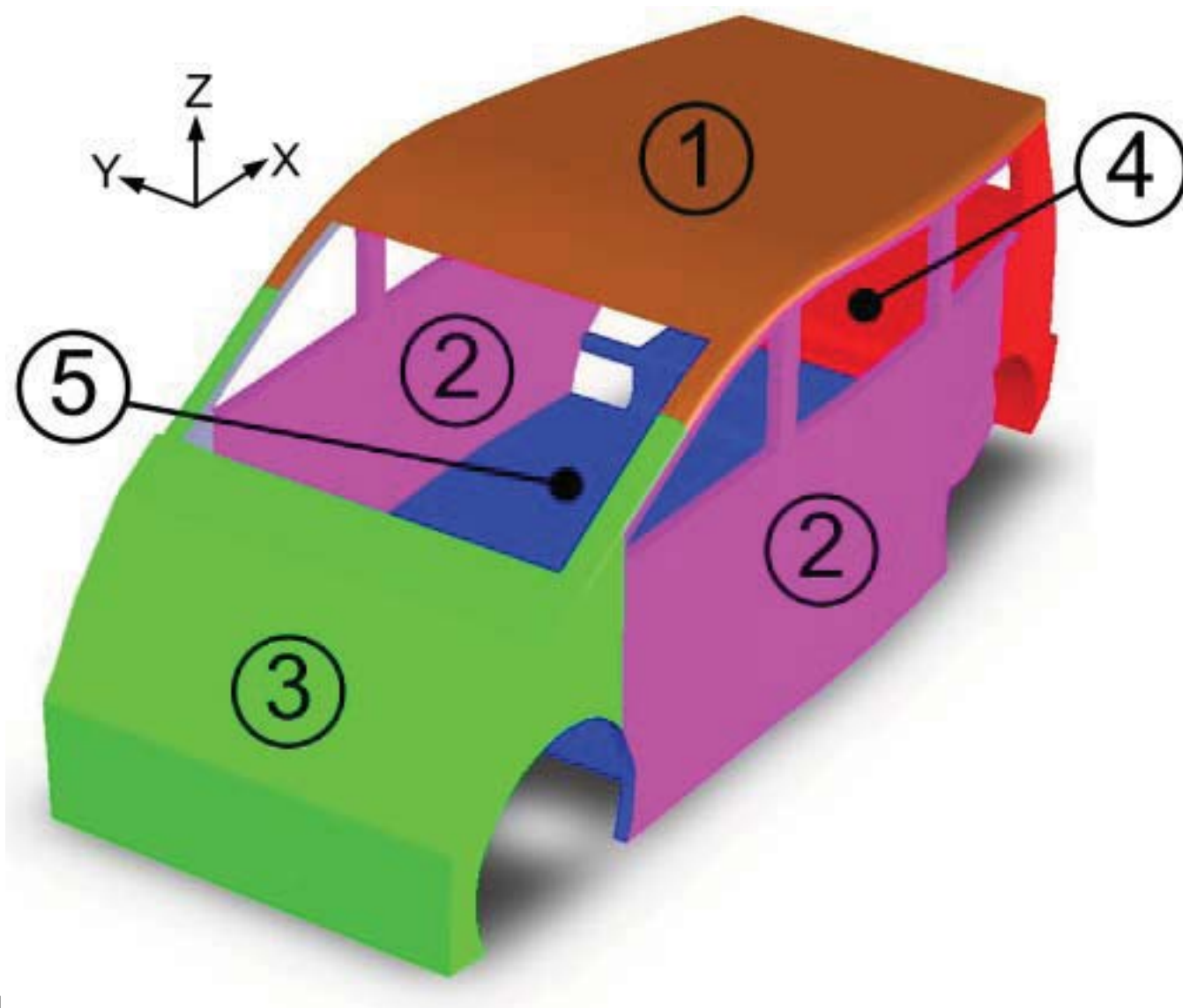
Boundary conditions. All crash model utilised in the development of the LCIM utilised a rigid bulkhead to constrain the model. This is not truly representative of the motion of the vehicle during impact, and could be improved with the use of a sled model. This would allow the pitching of the car during impact to be captured. Modelling the centre of mass as a point mass and shape for occupant injury reduction (front biased pulse). Data from NHTSA crash tests and modelling were investigated in order to quantify different pulse shapes including the interactions that caused them. utilising a simple rigid sled would not affect the computational time significantly.

M Manufacturing methods.

As "traditional" manufacturing methods were considered throughout the development of the LCIM only steel pressings were utilised. Further investigations on the feasibility of other steel manufacturing methods could be analysed in order to further improve the structural efficiency. This could for example include the use of seamless hydro formed parts for structural members, or even other materials. These improvements would also need to include the pressing manufacturing process, which would remap material strains and thinning due to the manufacturing in order to provide a more production-ready solution.

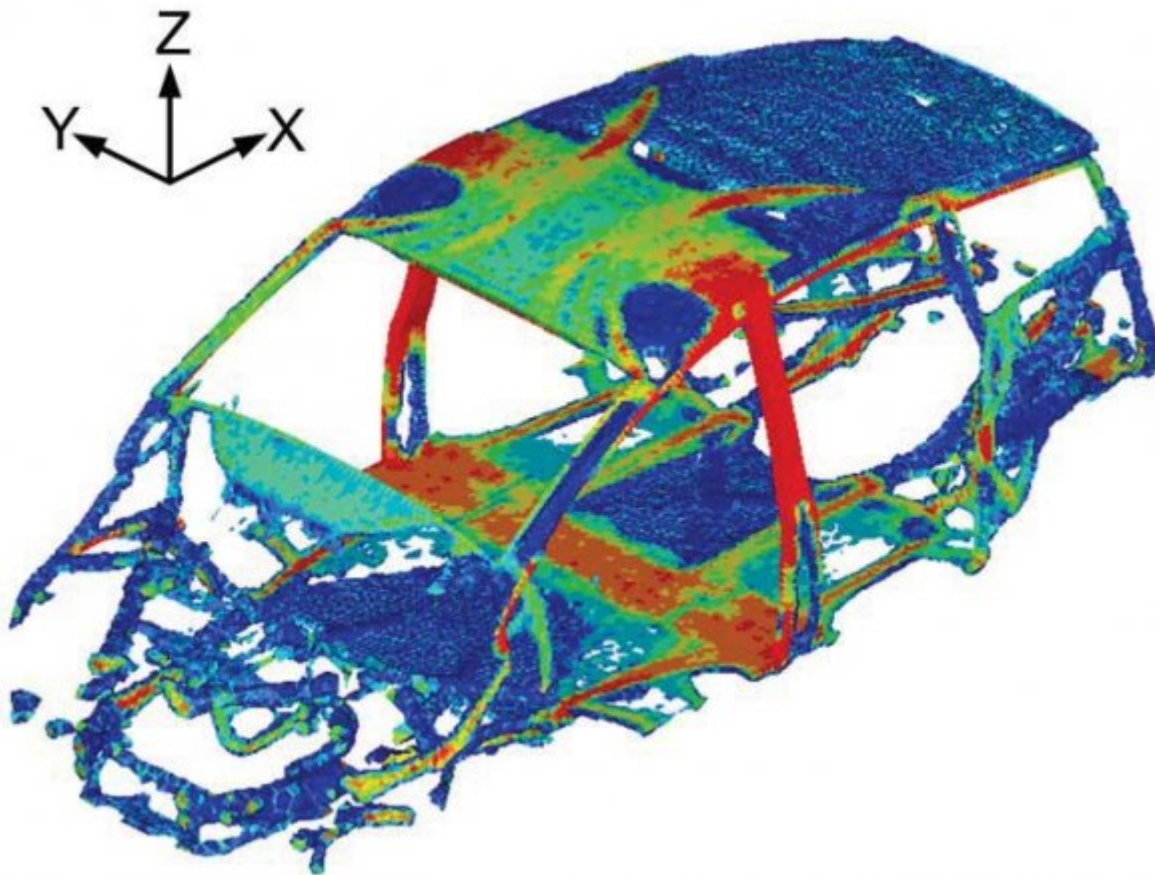


Figure 1:



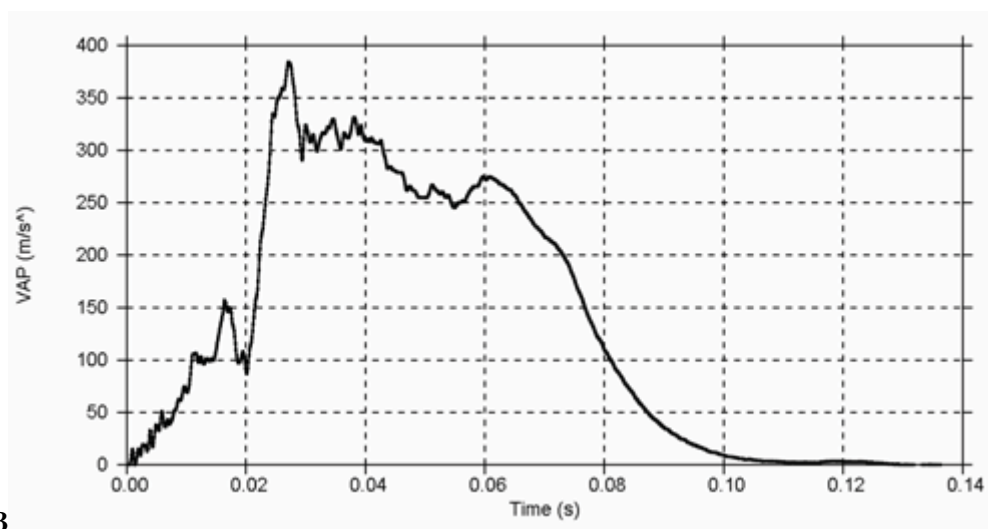
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Figure 2: Figure 1 :



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Figure 3: Figure 2 :



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Figure 4: Figure 3 :

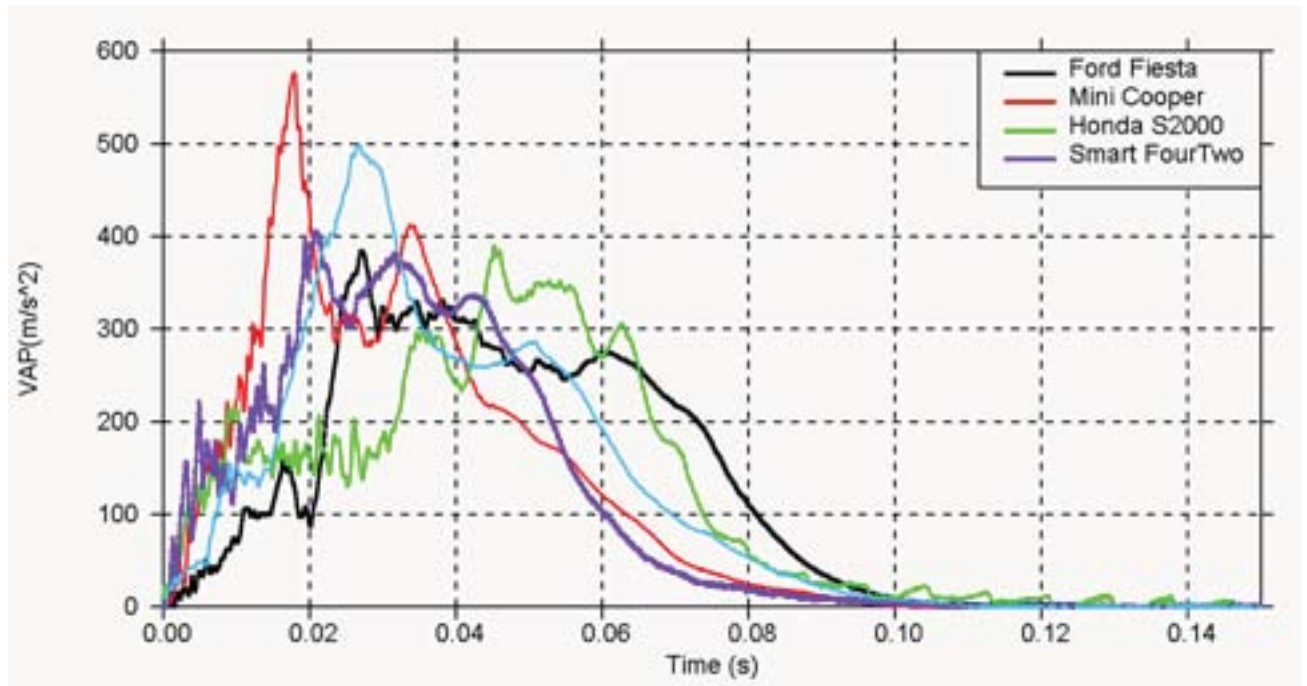


Figure 5:

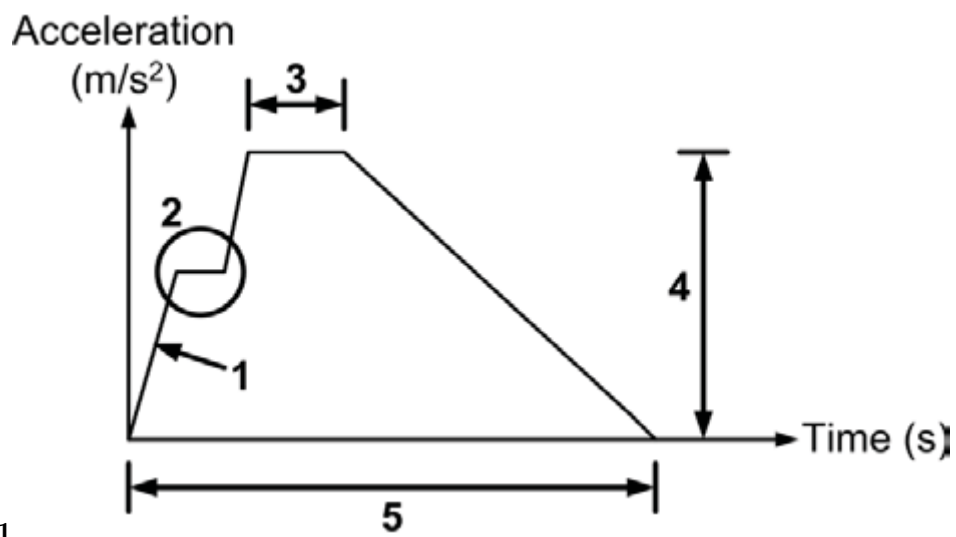
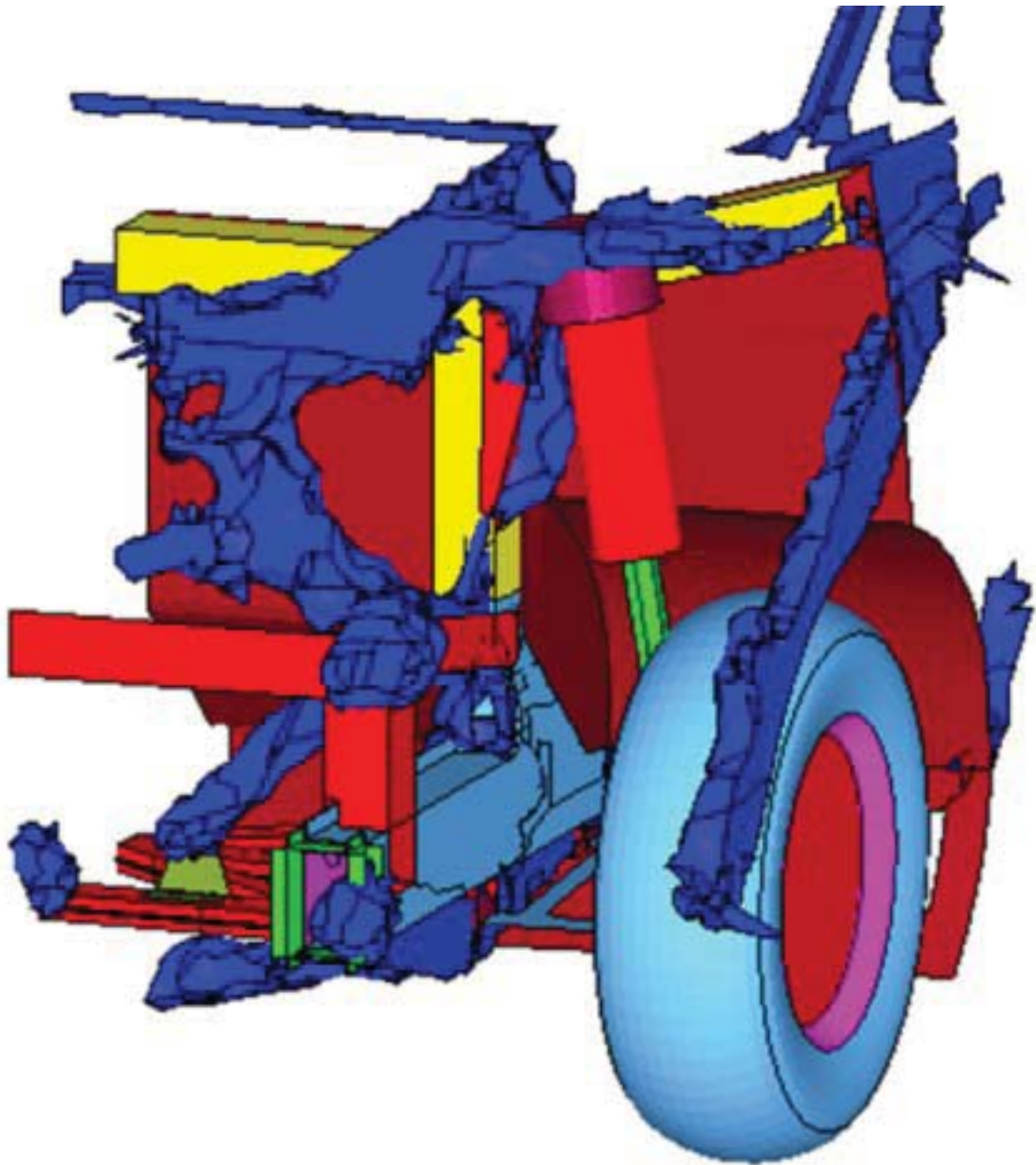


Figure 6: Figure 1 :



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Figure 7: Figure 5 :

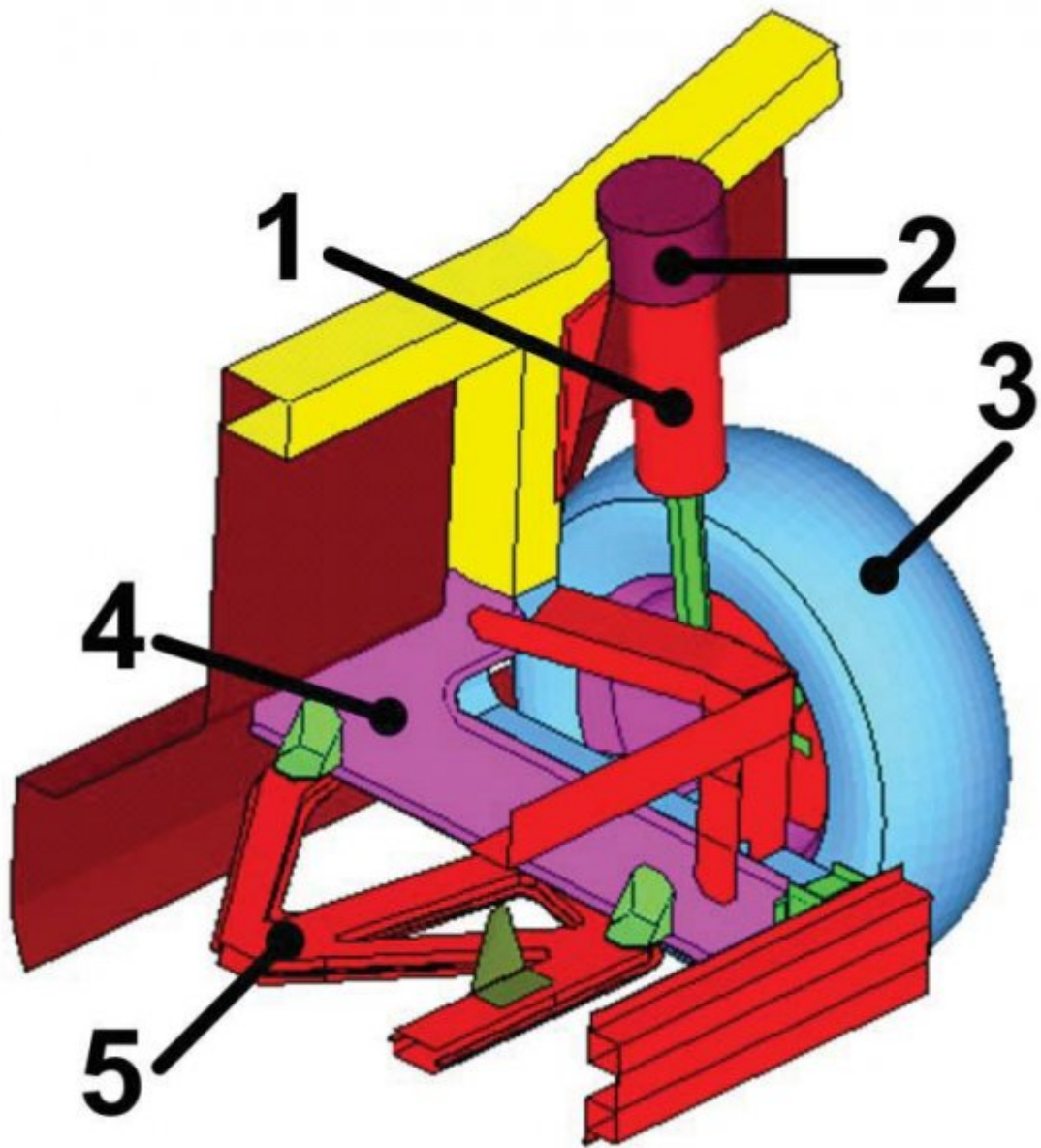
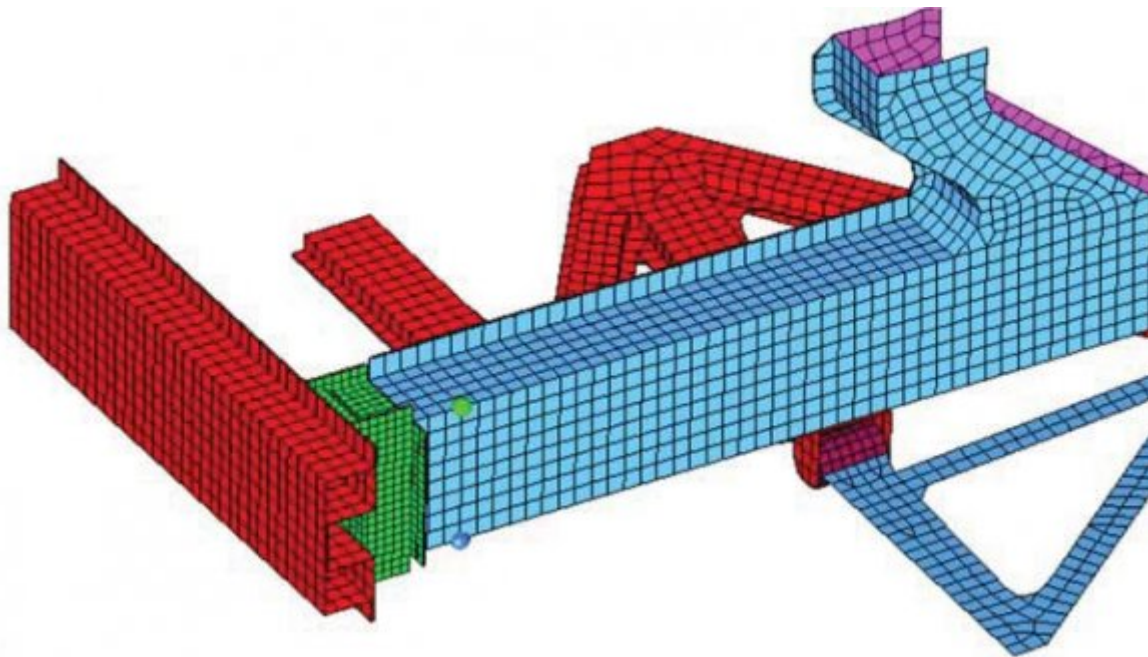
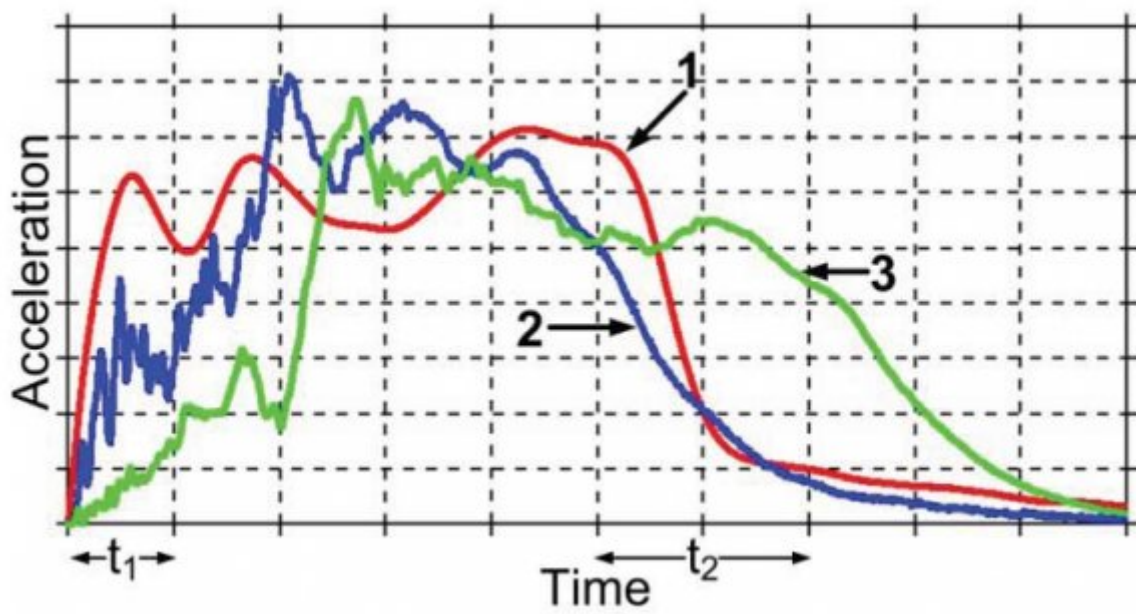


Figure 8:



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Figure 9: Figure 6 :



7

Figure 10: Figure 7 :

1

Vehicle	Model(year)	Mass(kg)	Test speed(km/h)	Impact number	NIH test (Gms)	Max. Post-crush impact
Ford Fiesta	2011	1359	56.5	6996	612	
Mini Cooper	2008	1371	56.3	6291	398	
Smart FourTwo	2008	1057	55.9	6332	320	
Jaguar Xtype	2003	1777	55.7	4484	413	
Honda S2000	2003	1465	57.0	4462		

The layouts of the five chosen vehicles are listed below:

1. 2011 Ford Fiesta: Front transversely mounted engine, front wheel drive, 5 seats.
2. 2008 Mini Cooper: Front transversely mounted engine, front wheel drive, 4 seats, short front overhang.
3. 2004 Smart FourTwo: Rear transversely mounted engine, rear wheel drive, 2 seats, very short front overhang.
4. 2003 Jaguar X type: Front transversely mounted engine, four wheel drive, 4 seats, long front overhang.

Figure 11: Table 1 :

2

Vehicle	performance Impact duration (ms)	Acceleration g (m/s ²)	Max. Ave.	Intrusion (mm)
Ford Fiesta	100	39.4	16.0	612
Mini Cooper	90	58.6	17.7	398

Figure 12: Table 2 :

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