DESIGN OPTIMISATION OF VEHICLE COMPONENTS FOR FULL FRONTAL CRASH

By

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ABSTRACT

Vehicular passive energy absorption plays an important part during frontal crash for passenger safety. Optimization of the frontal components is the key to increasing energy absorption due to large parameters. A full frontal crash of Volvo V40 model has been done in this project using ANSYS Explicit Dynamics module. Simplified geometry to represent major energy absorption mechanism of the individual components involved in frontal crash have been modelled on ANSYS modeler and analyzed under standard test conditions. The components considered for optimization are bumper cover stiffener, energy absorber, bumper cross-beam and chassis longitudinal member. MOGA and NLPQL implemented in ANSYS is utilized to optimize the geometries for individual components to give maximum energy absorption. The optimized geometry of each component was finally assembled together and analyzed again in under the same conditions.

good job on individual part A-
but I'm not sure
if I understand the
system-level simulation...

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1. PROBLEM STATEMENT

The automobile industry is trying to improve three themes which are energy conservation, safety and comfort. Further, due to the humongous increase of vehicles on the road, many research universities are focusing on improving the vehicle safety standards [1,2]. Given the current volume of vehicles on the road, a lot safety measures are being developed in order to safeguard the passengers, which in turn means to make the vehicles to absorb more energy during vehicle collision. In the current study, design optimization of full- frontal structure of a vehicle to increase the energy absorption is conducted. There are four sub-systems in the frontal structure, namely Bumper cover stiffener, Energy absorber, Bumper longitudinal beam and chassis member.

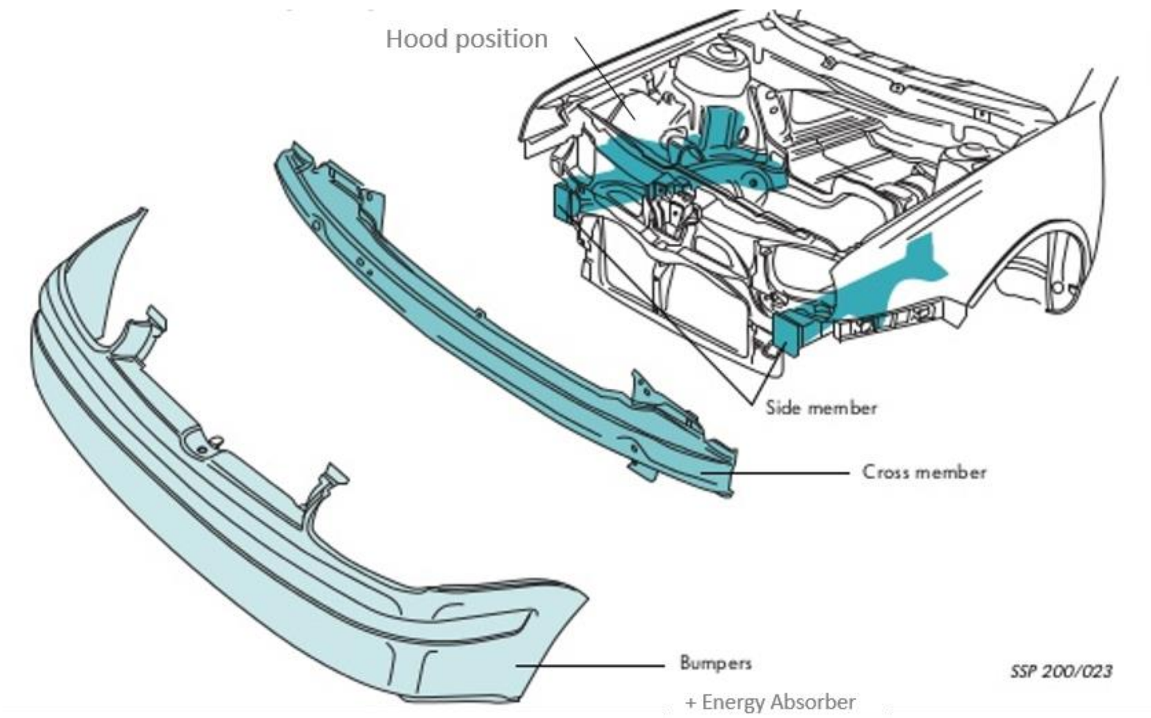


Fig.1: Extruded view of frontal crash components on vehicle

Numerous analysis was studied in order to carry out simulations of full frontal structure on ANSYS. Douglas Gabauer et al. [3] Studied about injury criteria during a road side vehicle collision. They have analyzed the Head Injury Criterion (HIC) and Acceleration Severity Index (ASI) for each simulation to find correlation between the both. Energy absorption and design optimization of the full – frontal structure wasn't discussed in this paper.

Zhida Shen et al. [4] presented the FEA modeling of the whole vehicle to increase the safety standards in the vehicle. They have analyzed for different parameters in order to find an optimal solution to meet their main objective. They have done for whole vehicle collision for high- velocity, low- velocity impact as well as the side wise collision. This model however doesn't optimize the design of vehicle.

Hao Chen et al. [5] studied the energy absorption on the whole vehicle and carried out the orthogonal design optimization, structural optimization using LS-DYNA. They have carried out the analysis on bumper cross beam (A), bumper cross thickness material (B), energy absorber groove distance (C) and front longitudinal beam groove member (D) along with 3 levels of each factor. Under these parameters, they have found their best condition to absorb 51% of energy during a collision.

The sub-system level division of work among the group members is as follows.

- Ram Mohan Telikicherla- **Bumper cover stiffener**
- Viswanathan Parathasarthy- **Energy Absorber**
- Pulkit Sharma- **Bumper Cross-Beam**
- Sai Nizampatnam- **Chassis longitudinal member**

2. NOMENCLATURE:

- T1= Thickness of the stiffener
- T2= Thickness of the support member
- H1= Height of the first support
- H2= Height of the second support
- L= Length of the absorber
- H= Height of the absorber
- SHAPE1= Vertical shape of the absorber
- SHAPE2= Horizontal shape of the absorber
- A21= Angle between bead and base plate of bumper beam
- V27= Vertical length is front bead surface
- H22= Length of bead from front surface inwards
- t =Thickness of longitudinal beam
- L_1 = Length of first sub member
- L_2 = Length of second sub member
- L_3 = Length of third sub member
- θ_1 = Angle between first and second sub member
- θ_2 = Angle between second and third sub member
- a = Height of the cross section
- b = Width of the cross section

3. BUMPER COVER STIFFENER

3.1 MATHEMATICAL MODEL:

3.1.1 OBJECTIVE FUNCTION:

The objective of the problem is to maximize the energy that can be absorbed by the stiffener and minimize the equivalent Von-misses stress in the member during low velocity crash. From literature it has been found that the component taken for optimization is useful for energy absorption during low velocity crash. A stiffener is a component that is attached behind the bumper cover and it is the main member in the assembly that absorbs maximum energy during low velocity crash.

3.1.2 STIFFENER PROPERTIES:

- Material- Aluminum alloy
- Weight of the stiffener- 3.3 kgs.
- Height of the stiffener- 197 mm
- Width of the stiffener- 1310 mm
- Depth of the stiffener- 202 mm

All dimensions are in millimeters and the weight is in kilograms. The dimensions and material properties has been taken from manufacturer's data sheet and has not been optimized.

3.1.3 DESIGN PARAMETERS:

The design parameters taken for the optimization study are thickness of the stiffener, thickness of the support, height of first support and height of the second support.

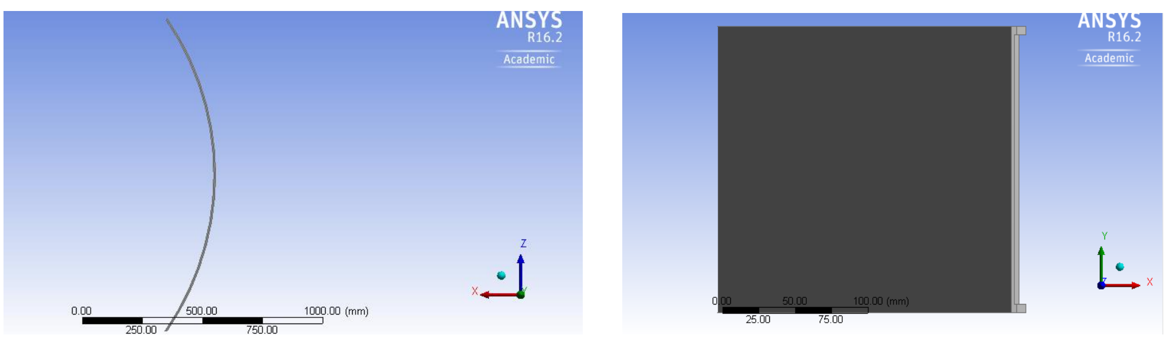


Fig.2: Top view and side view of the stiffener member

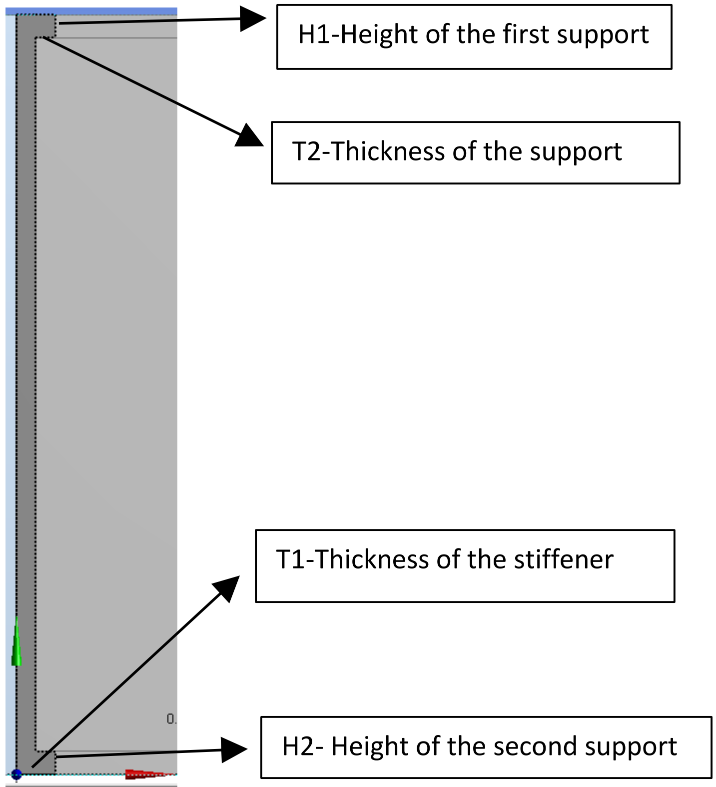


Fig.3: Cross section of the stiffener with design parameters

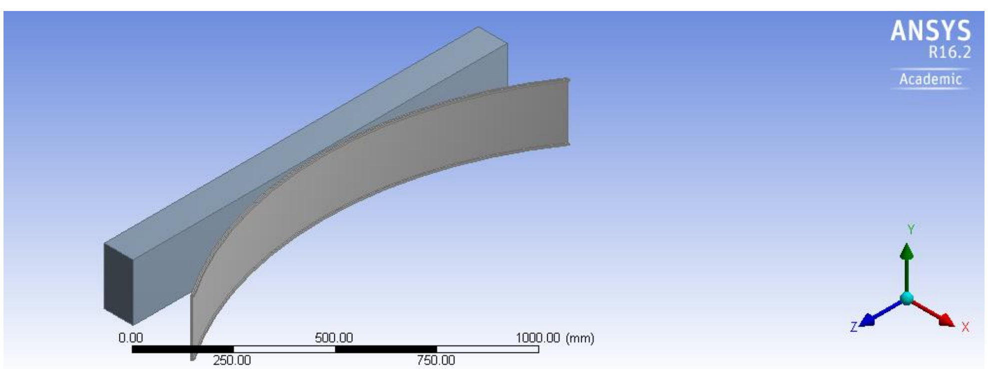


Fig.4: Isometric view of the stiffener with wall.

3.1.4 CONSTRAINTS:

The upper bounds and lower bounds for each design parameter that has been taken to optimize has been constrained within the range given below in Table 1.

Table 1: Design parameters with initial values and bounds.

DESIGN PARAMETER	INTIAL VALUE	UPPER BOUND	LOWER BOUND
T1 Thickness of stiffener	10	11	9
T2 Thickness of support	5	5.5	4.5
H1 Height of the first support	6	6.5	5.5

H2 Height of the second support	6	6.5	5.5

All dimensions are in (mm).

3.2 MODEL ANALYSIS:

3.2.1ANALYSIS METHOD AND BOUNDARY CONDITIONS:

Explicit dynamics analysis has been performed on the central stiffener by fixing the stiffener and giving velocity to the wall. The reason to choose explicit dynamics analysis in ANSYS is because in the absence of LS-DYNA if a crash simulation has to be performed then explicit dynamics module will provide the best possible results. The boundary conditions that have been taken for getting the results that can be compared with literature are as following.

The bumper cover stiffener has been fixed at the end faces. This has been done because in the assembly the stiffener component is snap fitted onto the main frame of the car body and during crash the end members are fixed and the stiffener fails at the center. A wall was modelled and given an initial velocity to show the deformation of the stiffener due to varying velocity.

3.3 OPTIMIZATION STUDY:

3.3.1 DESIGN OF EXPERIMENTS:

LATIN- HYPERCUBE SAMPLING ALGORITHM:

In order to perform the design of experiments on ANSYS Workbench, Latin- hypercube algorithm has been used. The reason to use this algorithm is because as there are more number of inputs variables that increase the number of simulation runs which also increases the number of permutations. Latin hypercube sampling algorithm developed by McKay et al., offers a proven mechanism to reduce the number of simulation runs needed to complete the sensitivity analysis.

6	⊟ Design of Experiments	
7	Design of Experiments Type	Latin Hypercube Sampling Design
8	Samples Type	User-Defined Samples
9	Random Generator Seed	50
10	Number of Samples	30

Fig.5: Design of experiments input

The number of samples in the above picture taken from ANSYS Workbench shows the number of data points taken to run the design of experiments.

3.3.2 SENSITIVITY ANALYSIS:

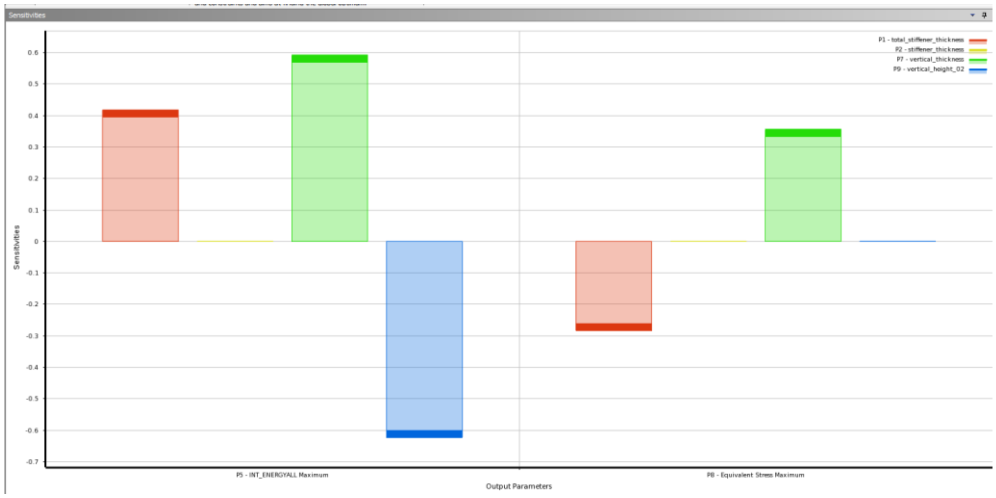


Fig.6: Sensitivity analysis of design parameters with output data

The above sensitivity analysis tells that the stiffener thickness, support thickness and height of second support are very sensitive to the energy absorption and equivalent von-misses stress.

3.3.3 OPTIMISATION METHOD:

3	⊟ Optimization Method			
4	MOGA	The MOGA method (Multi-Objective Genetic Algorithm) is a variant of the popular NSGA-II (Non-dominated Sorted Genetic Algorithm-II) based on controlled elitism concepts. It supports multiple objectives and constraints and aims at finding the global optimum.		
5	Configuration	Generate 100 samples initially, 100 samples per iteration and find 3 candidates in a maximum of 20 iterations.		
6	Status	Converged after 732 evaluations.		
7	⊟ Candidate Points			
8		Candidate Point 1	Candidate Point 2	Candidate Point 3
9	P1 - total_stiffener_thickness (mm)	10.686	10.686	10.518
10	P2 - stiffener_thickness (mm)	4.517	4.5267	4.5147
11	P7 - vertical_thickness (mm)	5.515	5.5103	5.5171
12	P9 - vertical_height_02 (mm)	5.7758	5.6972	5.9219
13	P8 - Equivalent Stress Maximum (MPa)	★★★ 767.75	★★★ 768.57	★★★ 769.64

Fig.7: Candidate points that are obtained after running MOGA for optimization

The optimization method used in this subsystem is **MOGA (Multiple objective genetic algorithm)**. The reason to choose this algorithm is because there are two conflicting objective functions. They are:

1. Maximizing the energy absorption.
2. Minimizing the equivalent Von-Misses stress.

These parameters are conflicting since the energy absorption calculation in ANSYS is done by taking the maximum strain energy and the stress is calculated by taking the maximum equivalent Von-misses stress. The two parameters taken are interdependent and there is system level trade-off in order to get the optimal design.

3.4 DISCUSSION OF RESULTS:

From the optimization study the point where there is maximum energy absorption and minimum equivalent stress is candidate point 2 from the below results.

Optimization , Candidate Points

B	C	D	E	F	G	H	I
Name	P1 - total_stiffener_thickness (mm)	P2 - stiffener_thickness (mm)	P7 - vertical_thickness (mm)	P9 - vertical_height_02 (mm)	P5 - INT_ENERGYALL Maximum (J kg^-1)	P8 - Equivalent Stress Maximum (MPa)	
						Parameter Value	Variation from Reference
Candidate Point 1	10.686	4.517	5.515	5.7758	3727	767.75	-0.25 %
Candidate Point 2	10.686	4.5267	5.5103	5.6972	3745.2	768.57	-0.14 %
Candidate Point 3	10.518	4.5147	5.5171	5.9219	3568.2	769.64	0.00 %
New Custom Candidate Point	10	5	6	6			

Fig.8: Candidate point showing the percentage variation from initial value

We can see that the variation in the three candidate points is very less. This clearly tells us that the results obtained are repeatable.

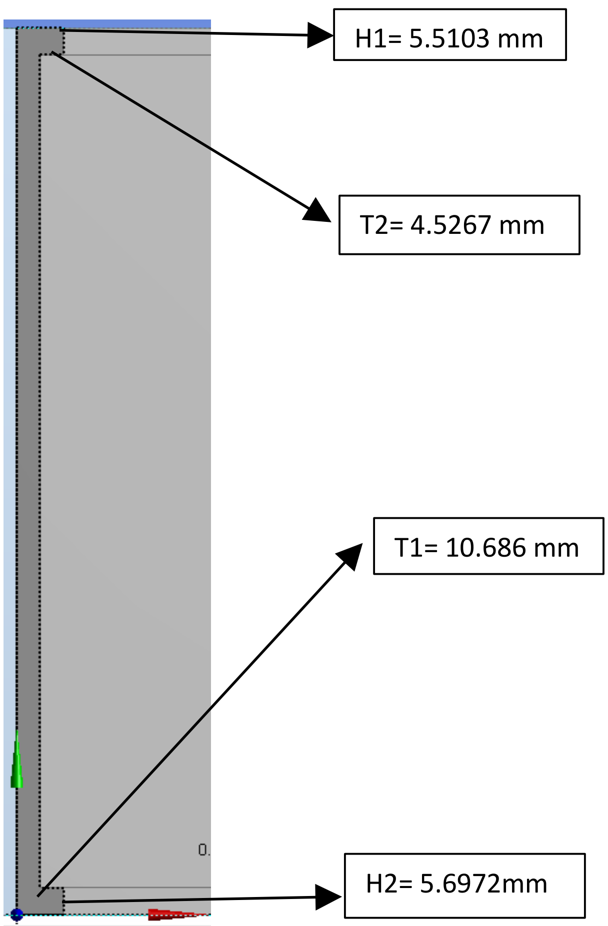


Fig.9: Cross section of the stiffener with optimized values.

Table 2: Design parameters with initial and optimized values

DESIGN PARAMETER	INTIAL VALUE	OPTIMIZED VALUE
T1 Thickness of stiffener	10	10.686
T2 Thickness of support	5	4.5267
H1 Height of first support	6	5.5103
H2 Height of second support	6	5.6972

All dimensions are in mm (millimeters).

The optimized results obtained has been validated by changing the mesh size from coarse to medium. The results converged and this convergence determines the validity of the simulation. The results obtained have also been validated by comparing the internal energy values obtained from literature. The value obtained from the simulation is 11,235 Joules which is comparable. The value will not be exact since the conditions taken for analysis in the literature is different from the one taken in the project.

ANSYS SIMULATION RESULTS:

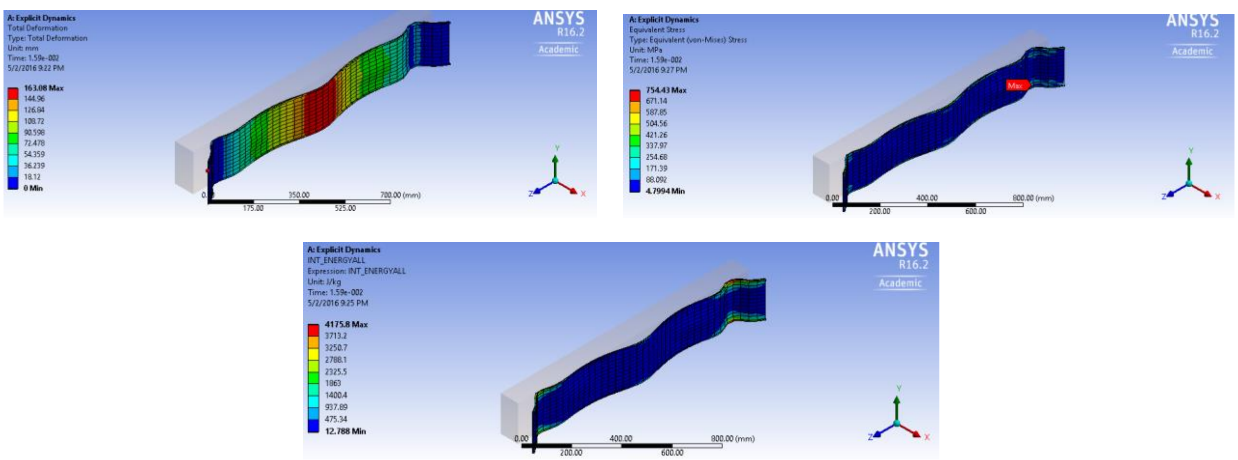


Fig.10: Deformation (Top left), Equivalent von-mises stress (Top right) and Internal energy

The above images show the simulation contour plots of deformation, stress and internal energy is for an initial run. The results obtained makes sense because the expected maximum deformation is at the center since during crash the stiffener breaks at the center. The stress is maximum at the ends due to stress concentration acting at the edge of the stiffener during crash.

4. ENERGY ABSORBER

Energy absorber is key member which is fixed in the stiffener. It's mainly used in low-velocity impact such that the other component's damage is restricted. In the current work, we have modelled energy absorber with the same dimensions as on Volvo V40.

4.1 MATHEMATICAL MODEL:

4.1.1 OBJECTIVE FUNCTION:

The main objective of the project is to maximize the energy that can be absorbed by the energy absorber. After studying numerous analysis done by different authors and it's found that the optimization of energy absorber for a low-velocity collision would be useful rather than a higher-velocity one. In order to increase the energy absorption, the shape of the absorber is parameterized within the original dimension of the car to avoid any conflict with other parts of systems.

4.1.2 ENERGY ABSORBER PROPERTIES AND DIMENSIONS:

- Material – Polyurethane
- Mass – 0.45474 kg
- Height of the absorber – 105 mm
- Length of the absorber – 76 mm
- Horizontal shape of the absorber – 44.828 mm
- Vertical Shape of the absorber – 22.146 mm

4.1.3 DESIGN PARAMETERS:

There are two energy absorbers attached to the stiffener which is symmetrical in shape and properties, so we have taken only one energy absorber to improve the energy absorber and multiply the result obtained by a factor of 2. The geometrical parameters of the absorber as shown in the figure 11 and the shape of the absorber is shown in figure 12 below would be varied to increase the energy absorption.

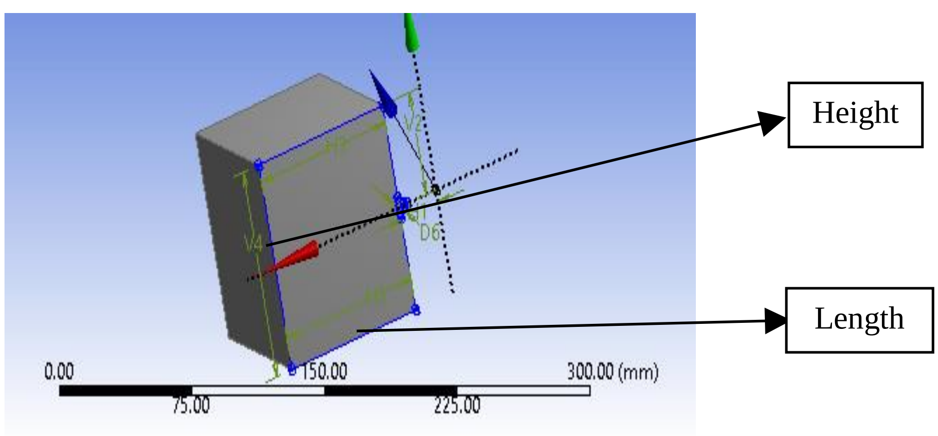


Fig.11: Model of the energy absorber

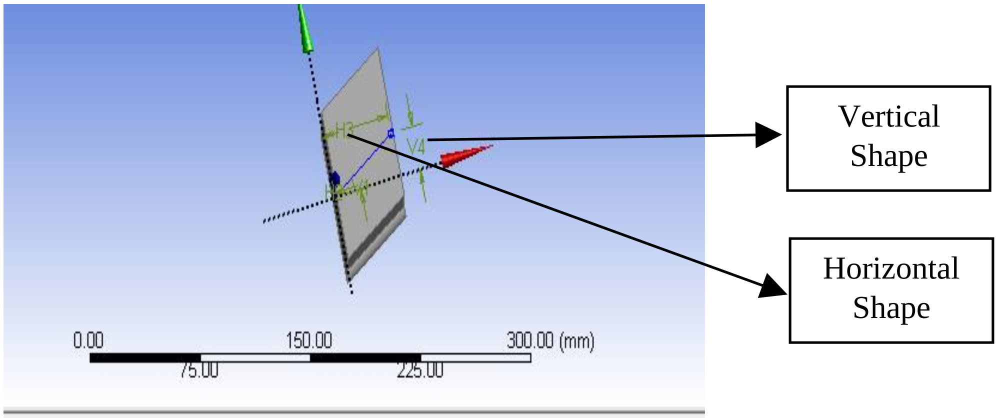


Fig.12: Side-View of the energy absorber

4.1.4 CONSTRAINTS:

The upper and lower bound of each parameter which is considered for optimization is tabulated below

Table 3: Upper Bound and Lower Bound considered during an optimization study

Design Parameter	Lower Bound (mm)	Upper Bound (mm)
Height	50	105
Length	40	76
Horizontal Shape (Shape 1)	30	44.828
Vertical Shape (Shape 2)	15	22.5

4.2 MODEL ANALYSIS:

4.2.1 ANALYSIS METHOD AND BOUNDARY CONDITIONS:

The model has been modeled in ANSYS and they have been analyzed in Explicit Dynamics, which gives an edge over LS-DYNA for the crash simulations and it also provides better results. The constraints mentioned above have been taken as parameters to conduct a Design of Experiment using Latin-Hypercube Algorithm for 25 design points. The initial dimension of the vehicle which was given on the website as the upper bound and values vary within that range. After conducting the Design of Experiment (DOE) in ANSYS and response surface was determined in ANSYS in which the sensitivities for a response point was found out by varying each parameters to find how each parameters does constitute in the change of the energy absorption. Further, in accordance with that we can decide the key parameters and optimize them to increase the energy absorber.

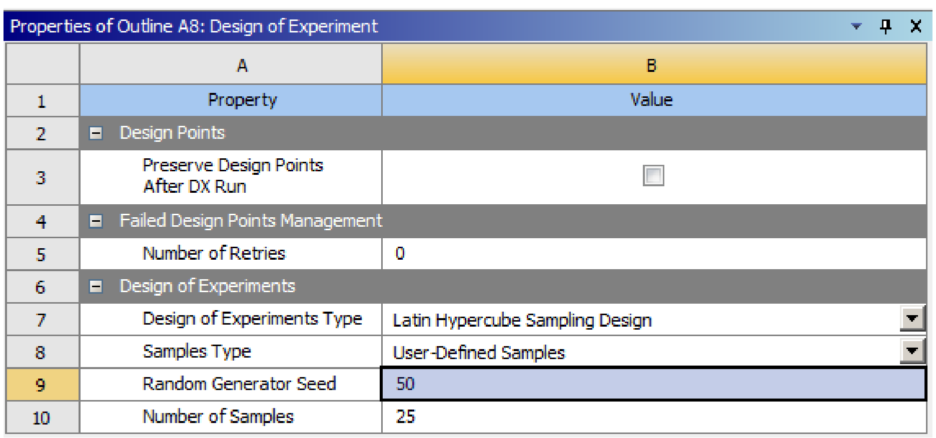
The boundary conditions that have been taken for getting the results that can be compared with literature are as following. Assuming the vehicle structure in which energy absorber is fixed at the end face. This is means that the energy absorber is fitted to the stiffener of the vehicle and during a collision (low-velocity) impact, energy absorber fails at the front. For the simulations, wall was modeled and initial velocity was given to see the deformation of the energy absorber and energy absorbed by the part.

4.3 OPTIMIZATION STUDY:

4.3.1 DESIGN OF EXPERIMENTS:

LATIN- HYPERCUBE SAMPLING ALGORITHM:

In order to perform the design of experiments on ANSYS Workbench Latin- hypercube algorithm has been used. It’s used here because the user can provide the number of samples for running the simulations. Since, each simulation takes more than 20 minutes to calculate, I have taken 25 design samples to work with. Latin hypercube sampling algorithm developed by McKay et al., offers a proven mechanism to reduce the number of simulation runs needed to complete the sensitivity analysis.


Properties of Outline A8: Design of Experiment

	A	B
1	Property	Value
2	Design Points	
3	Preserve Design Points After DX Run	
4	Failed Design Points Management	
5	Number of Retries	0
6	Design of Experiments	
7	Design of Experiments Type	Latin Hypercube Sampling Design
8	Samples Type	User-Defined Samples
9	Random Generator Seed	50
10	Number of Samples	25

Fig.13: Design of Experiment Properties

The above shown figure 13 presents the number of samples(design-points) used for the optimization problem.

4.3.2 SENSITIVITY ANALYSIS:

The sensitivity analysis was carried out with help of response points as shown in figure 14 below. The response point would be a reference point in which the parameters would be varied in order to find the sensitivity of each parameter.

Table of Schematic C3: Response Surface

	A	B	C	D	E
1	Name	P2 - Height (mm)	P3 - Length (mm)	P4 - Shape1 (mm)	P5 - Shape2 (mm)
2	⊟ Refinement Points				
*	New Refinement Point				
4	⊟ Response Points				
5	Response Point	77.5	58	37.414	18.75
*	New Response Point				

Fig.14: Response Point Data

The below figure 15 represents the sensitivity of each parameter with the response points. From the figure, we can observe that the length of the absorber is more sensitive to the energy absorbed by the sub-system and similarly vertical shape of the absorber is more sensitive to the total deformation. The response surface was determined by ANSYS and we observed how the output parameters react to the input parameters.

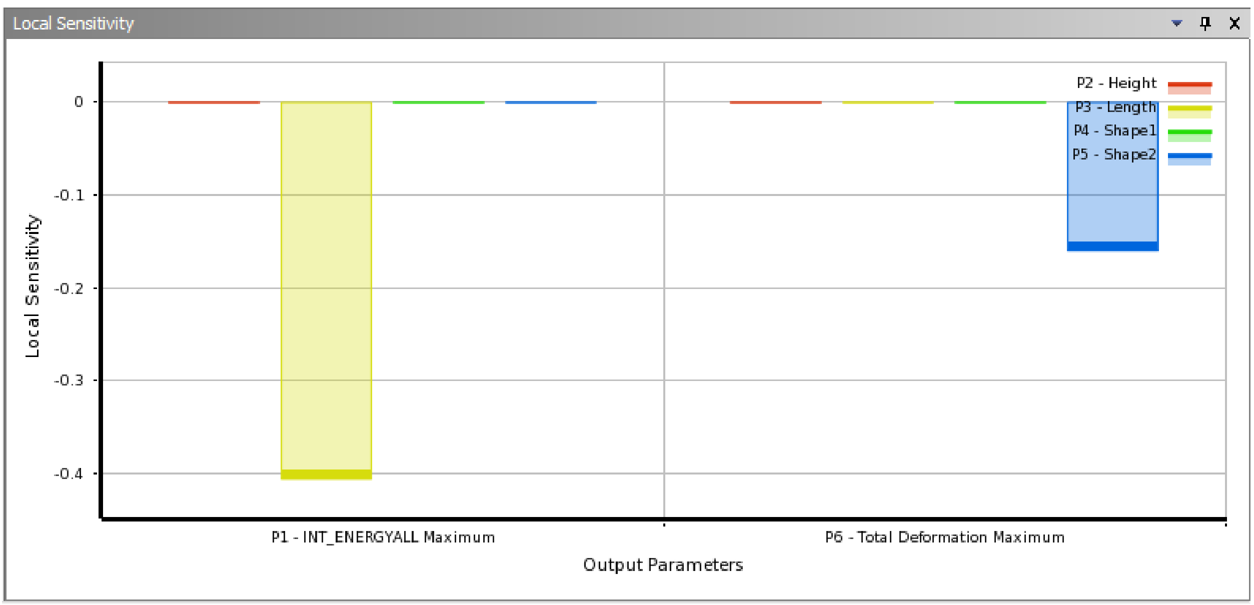


Fig.15: Sensitivity Analysis of the Sub-System

4.3.3 OPTIMISATION METHOD:

The optimization method used in this subsystem is NLPQL (Non-Linear Programming by Quadratic Lagrangian). The reason to choose this algorithm is because there was only one objective function and one constraint. It uses a quadratic approximation of a Lagrangian function and a linearization of the constraints. The parameters taken are interdependent and there is system level trade-off in order to get the optimal design.

4.4 DISCUSSION OF RESULTS:

From the optimization study the point where there is maximum energy absorption is the candidate point 1.

Table 3: Optimized Result of Sub-System

Name	Height (mm)	Length (mm)	Shape1 (mm)	Shape2 (mm)	Energy Absorption (J/kg)	Total Deformation (m)
Candidate Point 1	105	43.395	37.414	18.75	6590.8	0.085844
Candidate Point 2	94.353	43.393	37.414	18.75	5619.6	0.085844
Candidate Point 3	77.5	53.888	37.414	18.75	4936.5	0.085844

We can see that the variation in the three candidate points is less. This clearly tells us that the results obtained are repeatable. Energy absorbed by an energy absorber is 5994.20 J which is comparable to the value obtained by Hao Chen et al [5].

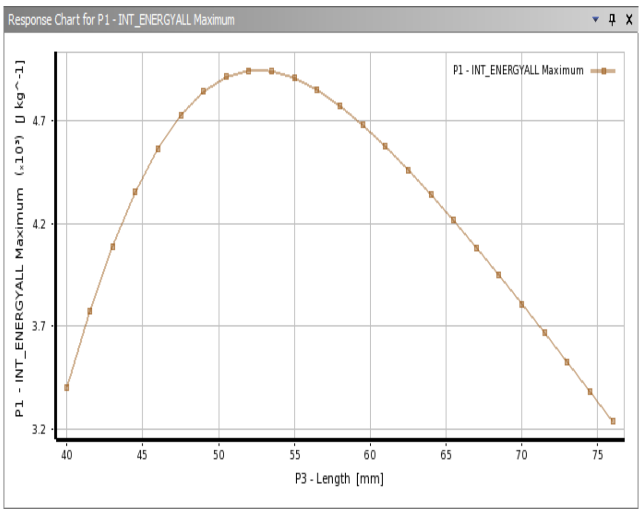


Fig.16(a): Variation of Energy absorption with Length

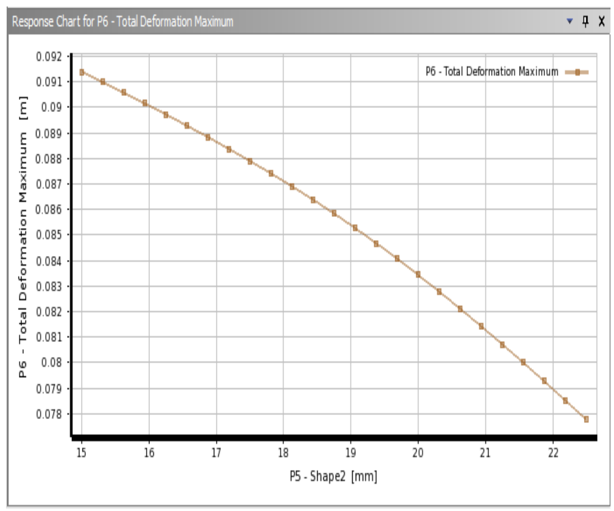


Figure 16(b): Variation of Total Deformation with Shape2

Figure 16(a) & 16(b) represents how variation of energy absorption changes with the length parameter and variation for total deformation happens with shape2 parameter.

Table 4: Comparison between optimized and initial values

Design Parameter	Initial Value (mm)	Optimized Value (mm)
Length of the absorber	76	43.393
Height of the absorber	105	105
Horizontal Shape of the absorber	44.828	37.414
Vertical Shape of the absorber	22.146	18.75

5.BUMPER CROSS BEAM:

Bumper Cross-beam (or simply referred as bumper beam) is third major component from front in longitudinal direction for frontal crash protection. It plays role in both low and high speed crash by absorbing impact energy by bending and crushing of beads. In the current work, we have modelled bumper beam with bead with same outer box dimensions as on Volvo V40.

5.1 MATHEMATICAL MODEL:

5.1.1 OBJECTIVE FUNCTION:

The objective of this sub-problem is to maximize *energy absorption (internal/strain energy)* and minimize *maximum Von-Mises stress* to increase the longevity of the beam under high velocity impact (worst case criteria). Internal energy and Von-Mises stress considered is *nodal averaged* values.

5.1.2 DESIGN PARAMETERS:

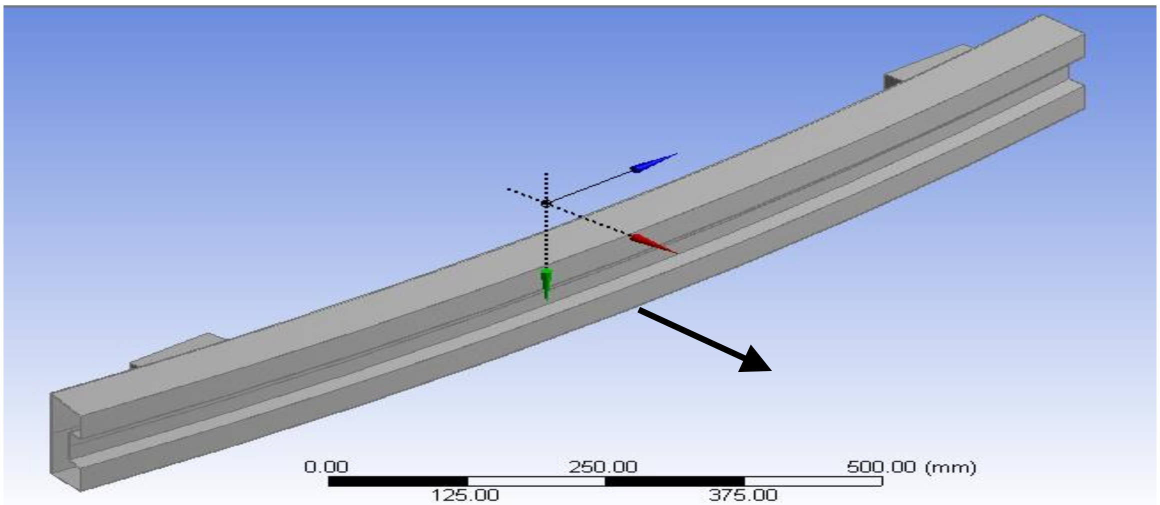


Fig. 17: Isometric view of bumper beam with bead

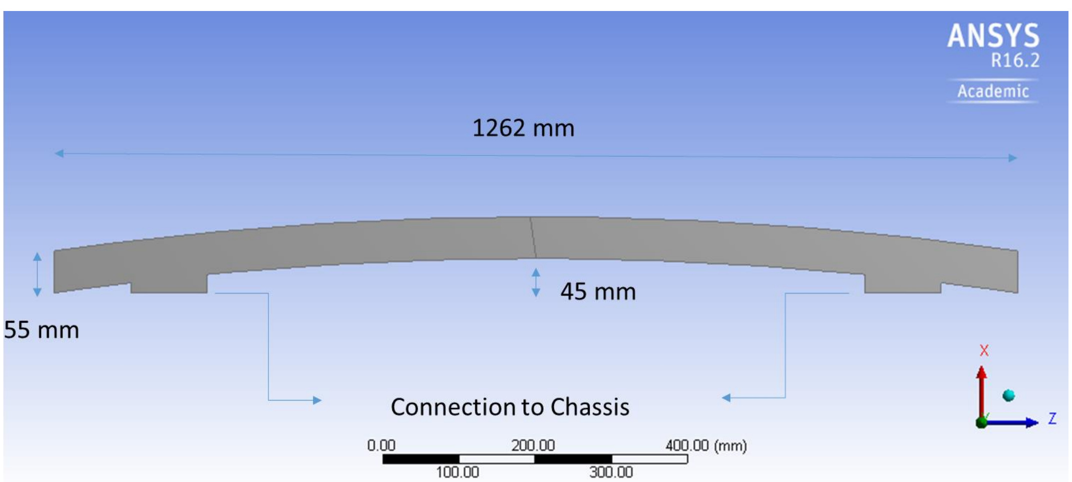
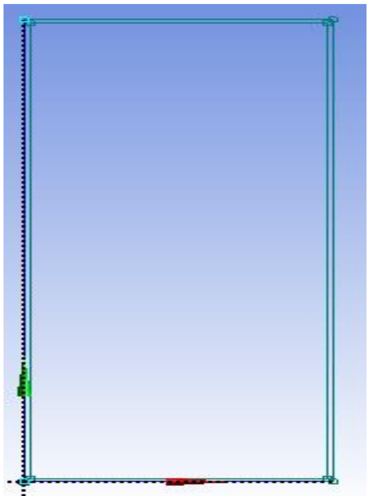


Fig. 18: Top view of bumper beam with outer box dimensions

(a)



(b)

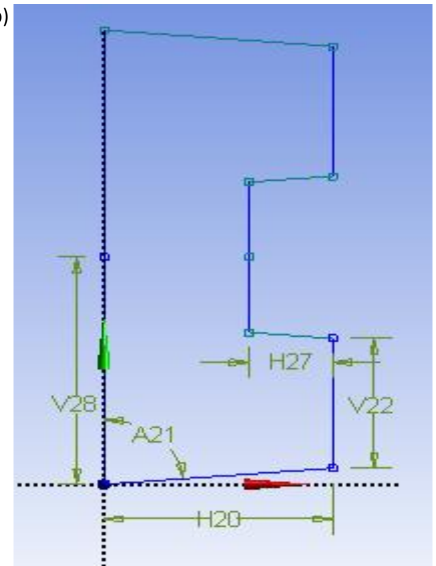


Fig. 19: Cross-sections of bumper beam (a) without bead structure (135mmX55mm, thickness=2mm) (b) with bead structure (V28=67.5mm, A21=85deg, H20=55mm, V22=38.59mm, H27=20mm)

Fig. 17, 18 and 19 shows the geometry of bumper beam with initial values of dimensions. The material considered is structural steel and the mass is 4.0186 kg compared to 4.124 kg on actual vehicle. Mass, stiffness and damping of structure and material are going to be vital properties to correlate the CAE results with the actual component.

For the current analysis, Table 5 describes the parameters considered for optimization with reason for selection.

Table 5: Design parameters* and initial values for optimization for bumper beam

S/N	Parameter	Initial value	Comment
1	A21	85 deg	Affects the crushing of bead structure
2	V22	38.59 mm	Affects the front surface area and hence transmission of load
3	H27	20 mm	Affects the integrity of base plate for intrusion of bead

5.1.3 CONSTRAINTS:

The upper bounds and lower bounds for each design parameter that has been taken to optimize has been constrained within the range given in Table 6.

Table 6: Upper and lower bounds considered during optimization study

S/N	Parameter	Lower Bound	Upper Bound
1	A21	85 deg	110 deg
2	V22	30 mm	45 mm
3	H27	16 mm	24 mm

*Due to computation capability constraint, three most important parameters are considered. In case of no such constraints all the mentioned geometric dimensions shown in fig. 19 and thickness have to be considered.

5.2 MODEL ANALYSIS:

5.2.1 ANALYSIS METHODOLOGY AND BOUNDARY CONDITIONS:

The analysis is conducted in *Explicit Dynamics* module of ANSYS Workbench 16.2. The impact was achieved with a rigid wall (mass = 760 kg) moving with a velocity of 15 m/s (run time =0.04 s). The boundary condition for bumper beam was finalized iteratively to avoid stress concentration, refer Appendix 1 for details of iterations and results achieved. Further convergence of the analysis was achieved by refining the mesh from *coarse* and *medium* in *sizing option.* No significant change was observed between the two mesh sizes considered. So all the further analysis will be conducted with coarse mesh to save on computations.

From the iterations conducted for boundary conditions (refer Appendix 1) it was concluded that chassis long member have to be used with the bumper beam to give realistic results. Fig. 4 shows the analysis setup and results obtained for initial values of parameters.

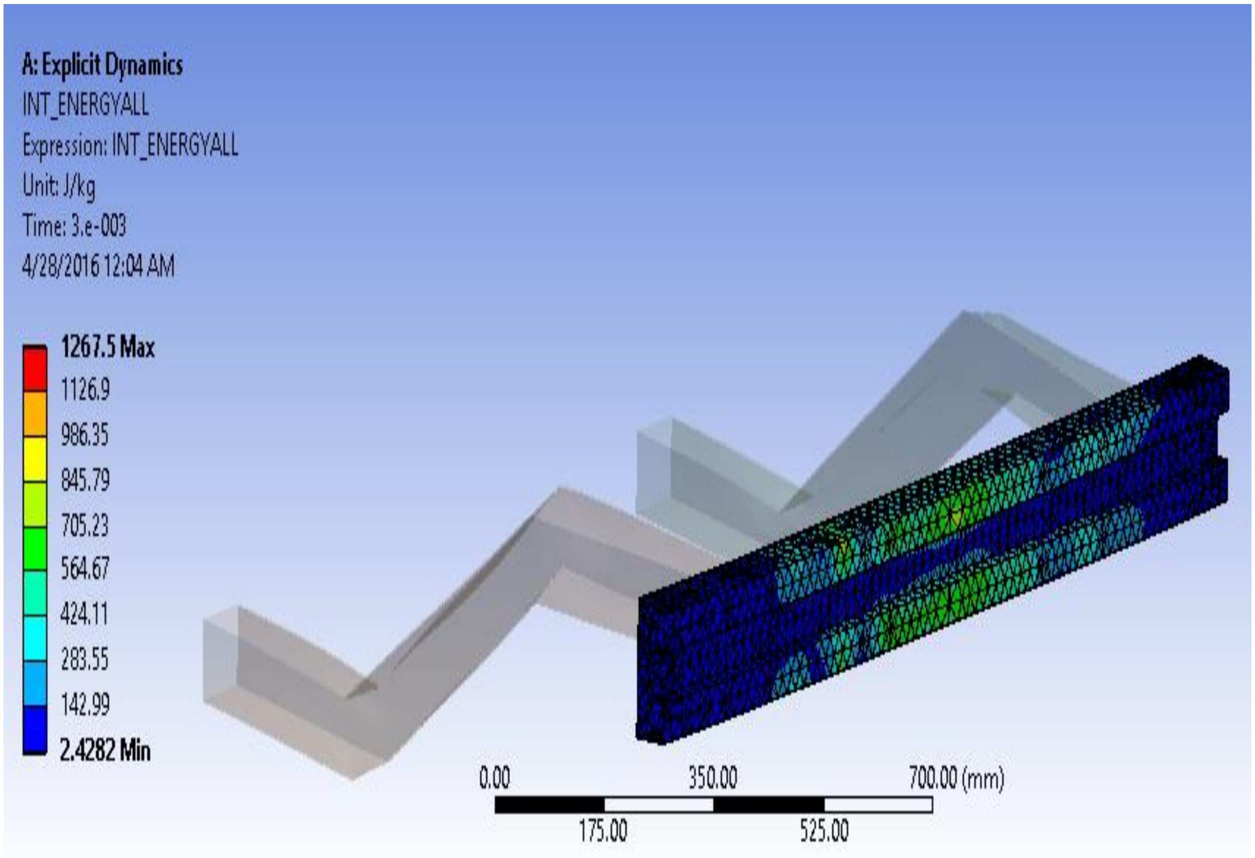


Fig. 20: Results of explicit dynamic analysis with chassis as high stiffness support for beam. Max Internal energy = 1267.5 J/kg and Max. Von-Mises Stress = 1674.5 MPa.

The simulation with chassis as support gives uniform spread of stress (no stress concentration) which was expected considering the symmetry of structure and loading. Also both the mechanisms of bending and crushing were simulated during impact. The obvious disadvantage with the setup being that the analysis time increased 4 folds (about 4 hours per simulation). Energy absorbed for bumper beam is 5179.6 J, which is comparable to value obtained by Hao Chen et al [4].

5.3 OPTIMIZATION STUDY:

5.3.1 DESIGN OF EXPERIMENTS:

In order to perform the design of experiments on ANSYS Workbench Latin- hypercube algorithm has been used. The reason to use this algorithm is because as there are more number of inputs the number of simulation runs increases due to the increase in number of permutations. Latin hypercube sampling algorithm developed by McKay et al., offers a proven mechanism to reduce the number of simulation runs needed to complete the sensitivity analysis.

The upper and lower bounds already mentioned are used to generate data points and corresponding results.

5.3.2 OPTIMIZATION METHOD:

The optimization method used in this subsystem is **MOGA (Multiple objective genetic algorithm)**. The reason to choose this algorithm is because there are two conflicting objective functions. They are:

5. Maximizing the energy absorption.
6. Minimizing the equivalent Von-Misses stress.

These parameters are conflicting since the energy absorption calculation in ANSYS is done by taking the maximum strain energy and the stress is calculated by taking the maximum equivalent Von-misses stress. The two parameters taken are interdependent and there is system level trade-off in order to get the optimal design.

5.4 DISCUSSION OF RESULTS:

There are two results concluded for the analysis. First optimization is conducted with objective function as given above and second optimization is conducted with objective function for minimizing Von-Mises stress only. Fig. 5-8 are for first optimization. Fig. 5 and Fig. 6 shows the variation of Energy absorption and Max. Von-Mises stress with A21. H27 and V22 doesn’t affect energy absorption but only the Von-Mises stress as given in Fig. 7 and Fig. 8.

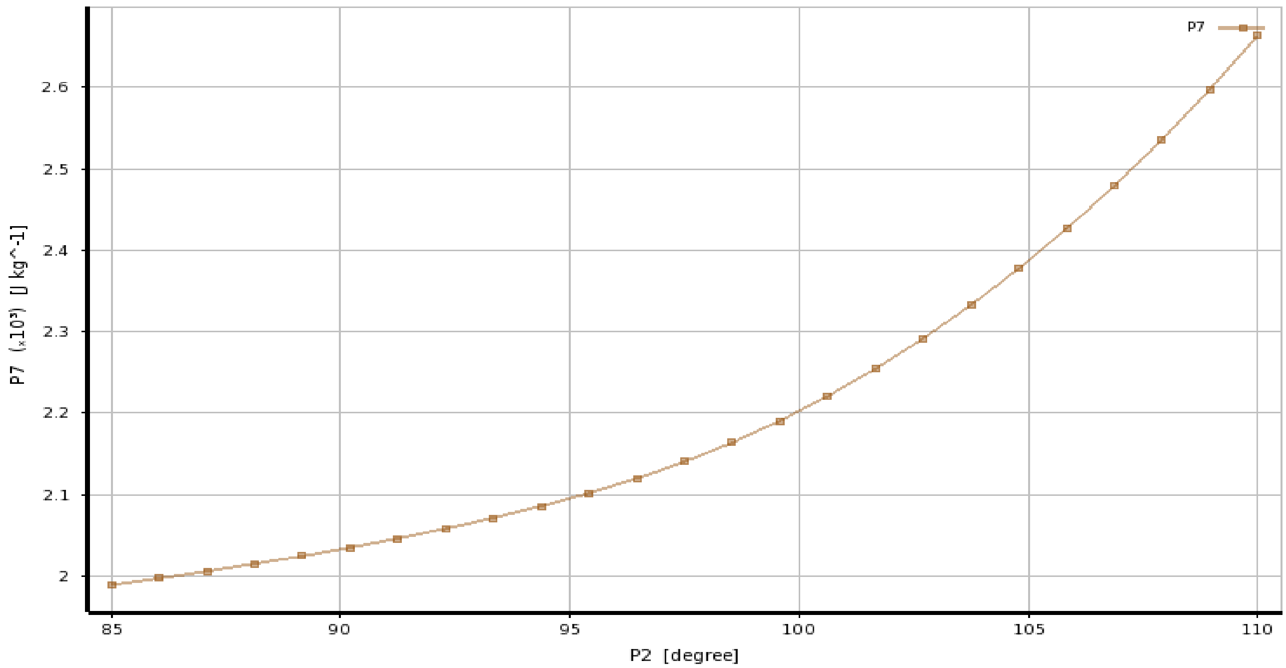


Fig. 21: Variation of Energy absorption with A21 (angle) in the design space

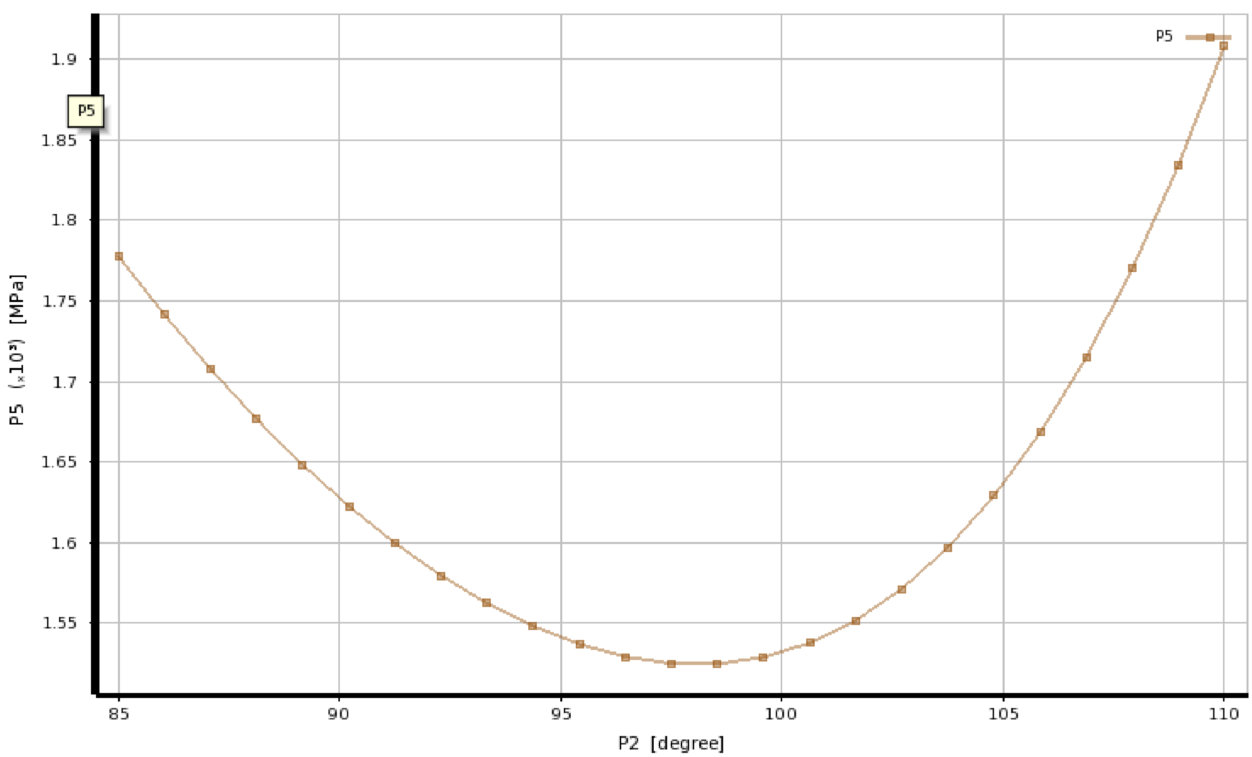


Fig. 22: Variation of Max. Von-Mises stress with A21 (angle) in the design space

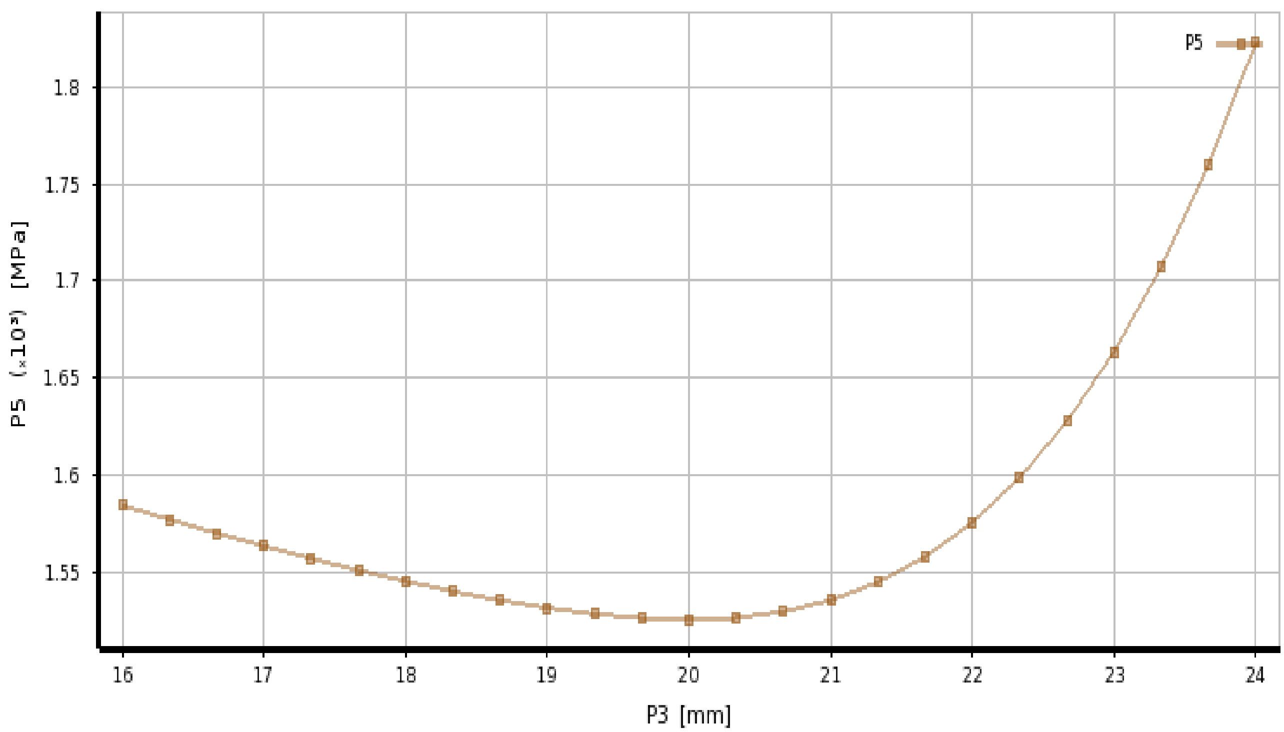


Fig. 23: Variation of Max. Von-Mises stress with H27 in design space

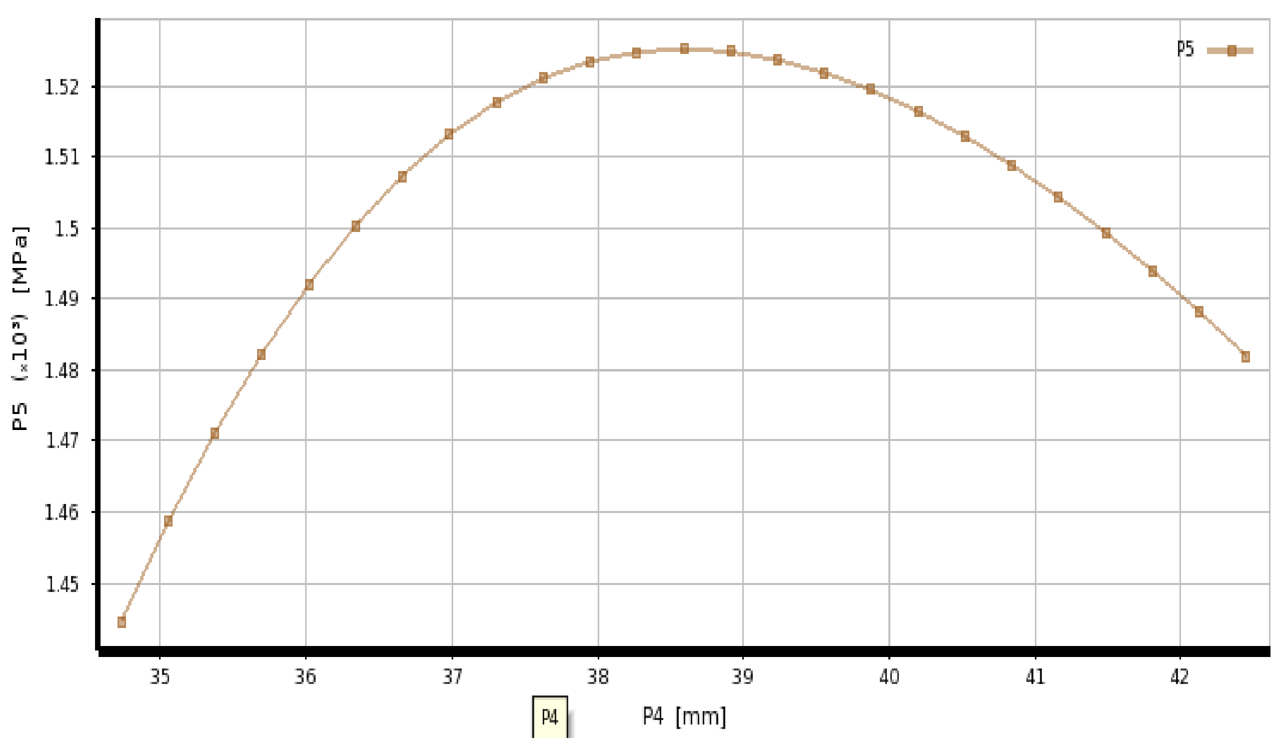
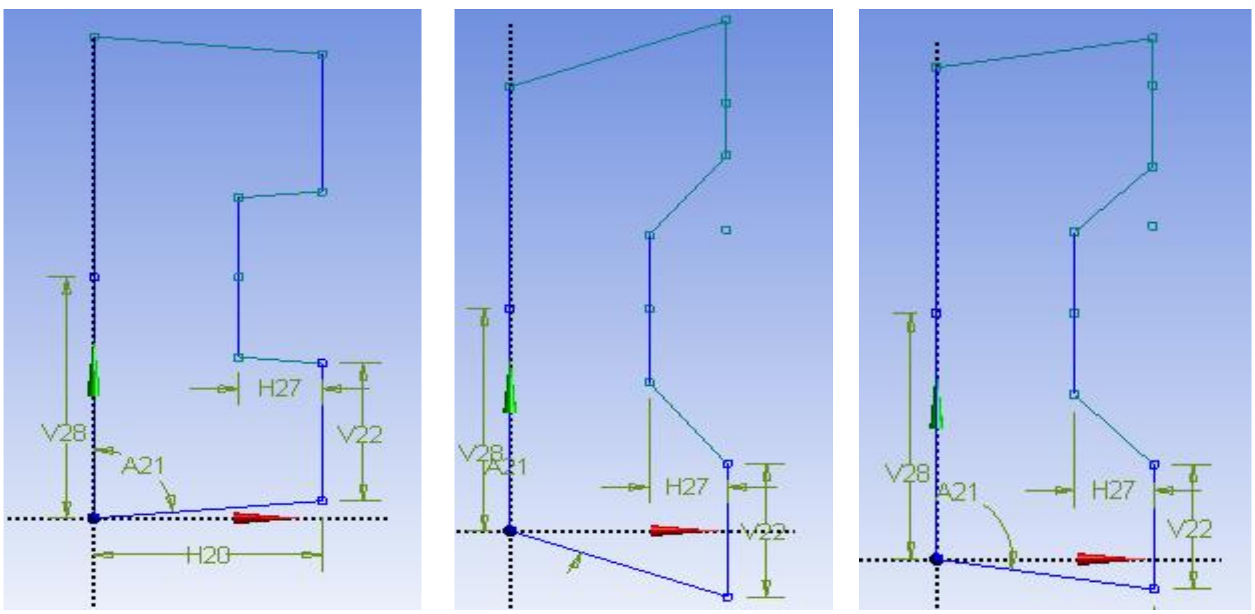


Fig. 24: Variation of Max. Von-Mises stress with V22 in design space

Table 7: Summary of results from both optimization and comparison with initial values

Objective	A21	H27	V22	Energy absorption (J/kg)	Stress (MPa)	Comment
Initial Values	85	20	38.59			
Energy + Stress	109.98	19.813	42.403	2662	1600.7	Design1 (D1)
Stress	98.54	20	34.73	2001	1542.9	Design2 (D2)

Considering fracture mechanical aspect to compare the two design obtained, we observe that ultimate tensile strength of the beam is reached at 0.00175s and 0.00205s in D1 and D2 respectively. Furthermore, Energy absorbed for two cases are comparable at critical point of fracture. Hence both the design gives equivalent performance till first fracture. A more detailed fracture mechanic study is required to choose the best design out of two.



(a) (b) (c)

Fig. 25: Comparison of beam cross-section (a) Start value (b) Design1 (c) Design2

6. CHASSIS LONGITUDINAL BEAM

6.1 MATHEMATICAL MODEL:

6.1.1 OBJECTIVE FUNCTION:

The objective of the problem is maximizing the energy absorbed during the crash resulting in less damage to engine and passenger compartment. It is the most critical part as it absorbs maximum energy during high velocity crash. This is attached just behind the bumper cross beam. Total deformation is taken as the constraint which is given in detail under constraint section.

Now much simplified model of chassis longitudinal member is incorporated as it provides rapid estimate of crash behavior. [6]

The optimization results are given in the later section.

6.1.2 DESIGN PARAMETERS:

The design parameters that are taken for the optimization study are as follows:

1. Thickness of longitudinal beam (t)
2. Length of first sub member (L_1)
3. Length of second sub member (L_2)
4. Length of third sub member (L_3)
5. Angle between first and second sub member (θ_1)
6. Angle between second and third sub member (θ_2)
7. Height of the cross section (a)
8. Width of the cross section (b)

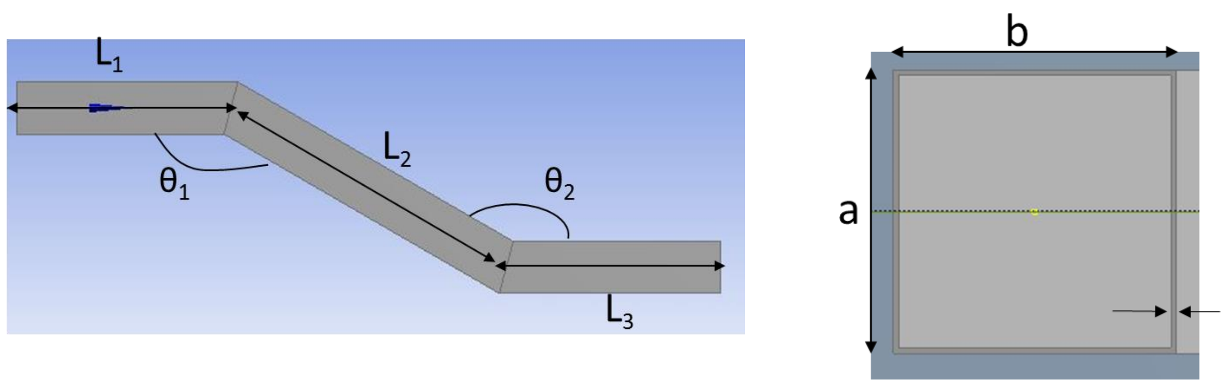


Fig.26: Side view and cross section of chassis longitudinal member

6.1.3 CONSTRAINTS:

- Total deformation ≤ 0.752 mm
- Upper bound and lower bound of each design parameters are as given in the following table

Table 8: Upper bound and lower bounds of corresponding parameter

DESIGN PARAMETER	UPPER BOUND	LOWER BOUND
L_1	360	300
L_2	540	460
L_3	360	300
θ_1	180	90
θ_2	180	90
T	2	1.2
A	90	72
B	90	72

Lengths are in (mm) and angles in degrees.

6.2 MODEL ANALYSIS:

The following assumptions are taken to make the problem easy to solve.

6.2.1 ASSUMPTIONS:

- Designed simplified model instead of detailed model of chassis longitudinal member as simplified model provides rapid estimate of crash behavior. [6]
- Assumed the car's body as block of mass.
- Assumed mass of car as 1500 Kgs.
- The upper limit and lower limit of some parameters are taken intuitively.
- Assigned structural steel as the material for longitudinal member and the block of mass which is acting as car body.

6.2.2 ANALYSIS METHOD AND BOUNDARY CONDITIONS:

Explicit dynamics analysis has been performed on the chassis longitudinal member by giving initial velocity to the chassis longitudinal member and crashing into the wall. The reason to choose explicit dynamics analysis in ANSYS is because in the absence of LS-DYNA if a crash simulation has to be performed then explicit dynamics module will provide the best possible results. Also a block a mass has been attached to the end of the member to replicate the more realistic conditions. The mass of the block is taken as 750 Kgs which is half the mass of the car as the explicit dynamics analysis is performed on single longitudinal member.

The boundary conditions that have been taken during explicit dynamics are as follows

The wall has been taken as fixed rigid body and the chassis longitudinal member along with the mass of the block have been given an initial velocity.

Also the degree of freedom is fixed for the front end of longitudinal member such that only the direction of crash (Z axis) is free.

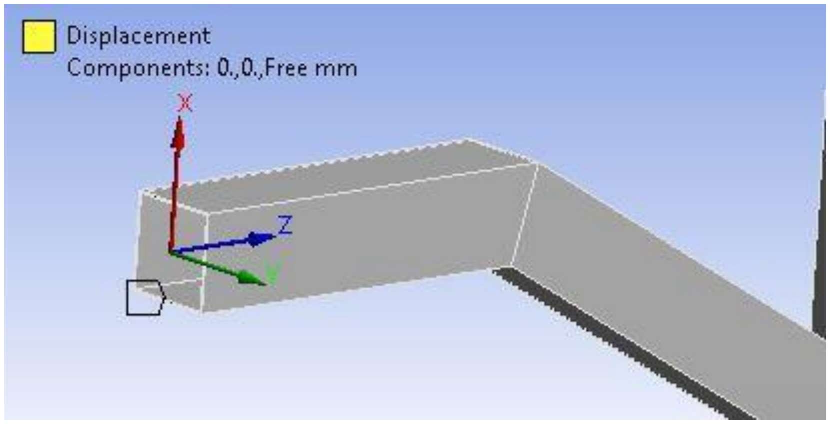


Fig.27: Fixing the degrees of freedom of front end

The initial velocity is taken as 15 m/s.

6.3 OPTIMIZATION STUDY:

6.3.1 DESIGN OF EXPERIMENTS:

LATIN- HYPERCUBE SAMPLING ALGORITHM:

Latin Hypercube sampling design algorithm is used for Design of experiments (DOE). The reason to use this algorithm is because as there are more number of inputs the number of simulation runs increases due to the increase in number of permutations and it is a proven mechanism to reduce the number of simulation runs needed to complete the sensitivity analysis.

It was infeasible to run DOE analysis for 100 points due to time constraint so analysis was done for **50 Design points**.

Name	P6 - L3 (mm)	P5 - Theta1 (degree)	P4 - L1 (mm)	P3 - L2 (mm)	P2 - b (mm)	P1 - a (mm)	P7 - Theta2 (degree)	P8 - t (mm)	P9 - INT_ENERGYALL Maximum (BTU lbm^-1)
1	333	148.5	324.6	473.6	72.9	84.42	107.1	1.672	13.047
2	301.8	166.5	305.4	502.4	79.02	77.22	139.5	1.288	14.118
3	327	143.1	357	534.4	76.14	74.7	157.5	1.224	7.9315
4	304.2	110.7	310.2	516.8	85.5	79.74	114.3	1.624	5.6812
5	355.8	117.9	336.6	464	87.66	78.3	148.5	1.544	6.7934
6	325.8	155.7	354.6	491.2	84.06	82.26	152.1	1.464	12.699
7	354.6	144.9	300.6	497.6	75.06	89.82	171.9	1.432	13.762
8	311.4	123.3	318.6	513.6	79.38	81.54	141.3	1.88	5.4071
9	307.8	179.1	306.6	505.6	85.86	87.66	179.1	1.512	5.0806
10	306.6	134.1	327	468.8	87.3	76.86	123.3	1.784	10.078
11	321	130.5	316.2	483.2	86.58	81.18	112.5	1.336	10.496
12	336.6	173.7	322.2	510.4	81.18	83.34	155.7	1.848	16.215

Fig.28: DOE Results

6.3.2 SENSITIVITY ANALYSIS:

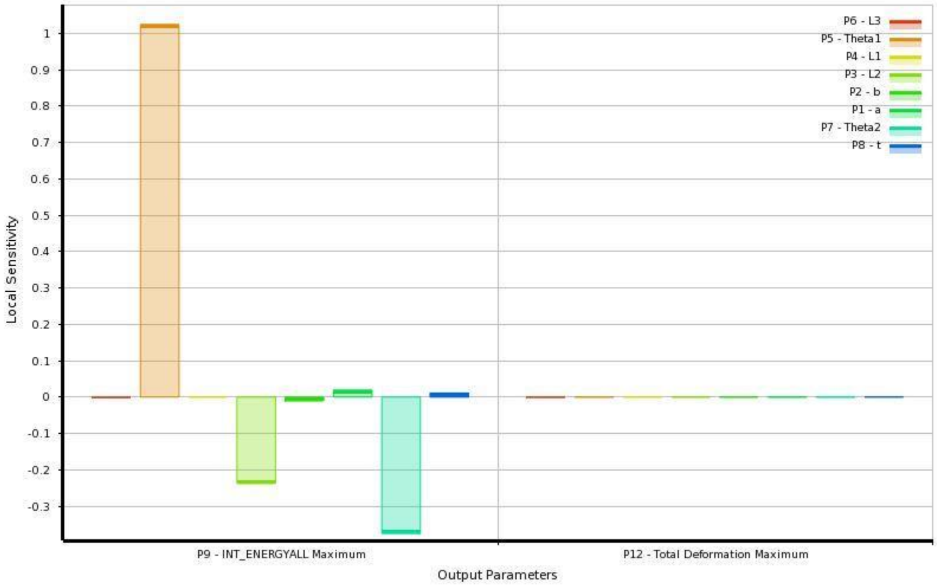


Fig.29: Sensitivity analysis of design parameters with output data

The above sensitivity analysis shows that angle between first and second sub member (θ_1), angle between second and third sub member (θ_2), length of second sub member (L_2) are very sensitive to the energy absorption and whereas no design parameter is sensitive to total deformation.

6.3.3 OPTIMISATION METHOD:

The optimization method used for this subsystem is NLPQL (Nonlinear programming by quadratic lagrangian). This algorithm is used because it has single output parameter objective i.e. maximizing the energy absorption and one constraint on total deformation.

	A	B	C	D	E
1	Optimization Study				
2	Maximize P9	Goal, Maximize P9 (Default importance)			
3	P12 <= 0.029588 in	Strict Constraint, P12 values less than or equals to 0.029588 in (Default importance)			
4	Optimization Method				
5	NLPQL	The NLPQL method (Nonlinear Programming by Quadratic Lagrangian) is a gradient-based algorithm to provide a refined, local, optimization result. It supports a single output parameter objective, multiple constraints and is limited to continuous parameters. The starting point must be specified to determine the region of the design space to explore.			
6	Configuration	Approximate derivatives by Central difference and find 3 candidates in a maximum of 20 iterations.			
7	Status	Converged after 95 evaluations.			
8	Candidate Points				
9		Starting Point	Candidate Point 1	Candidate Point 2	Candidate Point 3
10	P6 - L3 (mm)	330	360	341.3	330.83
11	P5 - Theta1 (degree)	135	180	180	180
12	P4 - L1 (mm)	330	330	330	330
13	P3 - L2 (mm)	500	460	460	460
14	P2 - b (mm)	81	72	75.789	80.15
15	P1 - a (mm)	81	90	90	84.025
16	P7 - Theta2 (degree)	135	133.42	131.91	130.76
17	P8 - t (mm)	1.6	2	1.7644	1.6322
18	P9 - INT_ENERGYALL Maximum (BTU lbm^-1)	9.3401	29.052	22.11	15.779
19	P12 - Total Deformation Maximum (in)	0.029539	0.029554	0.029543	0.02954

Fig.30: Candidate points that are obtained after running NLPQL for optimization

6.4 DISCUSSION OF RESULTS:

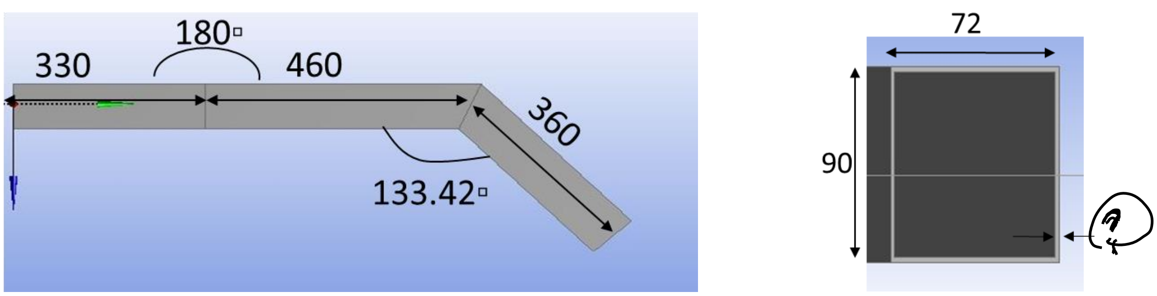
The optimization results gave three candidate points. The next step is to select the best candidate point out of these three.

Reference	Name	P6 - L3 (mm)	P5 - Theta1 (degree)	P4 - L1 (mm)	P3 - L2 (mm)	P2 - b (mm)	P1 - a (mm)	P7 - Theta2 (degree)	P8 - t (mm)	P9 - INT_ENERGYALL Maximum (BTU lbm^-1)	
										Parameter Value	Variation from Reference
◉	Starting Point	330	135	330	500	81	81	135	1.6	★ 9.3401	0.00 %
○	Candidate Point 1	360	180	330	460	72	90	133.42	2	★★★ 29.052	211.05 %
○	Candidate Point 2	341.3	180	330	460	75.789	90	131.91	1.7644	★★ 22.11	136.72 %
○	Candidate Point 3	330.83	180	330	460	80.15	84.025	130.76	1.6322	★★ 15.779	68.94 %

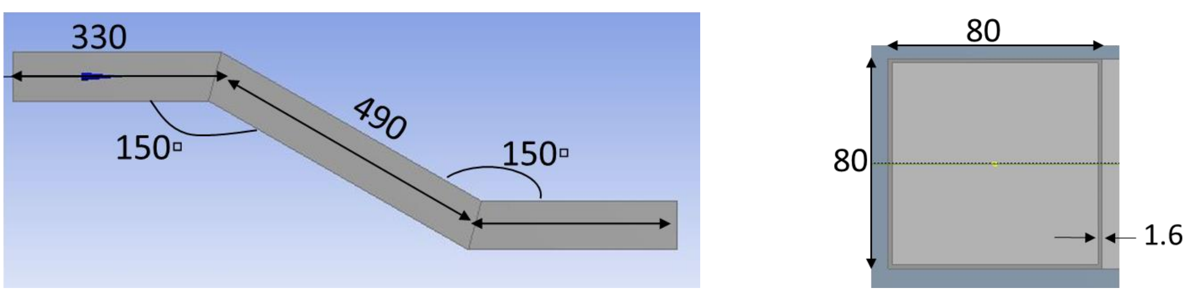
Fig.31: Candidate point showing the percentage variation from Starting point

Candidate point 1 gives highest internal energy so this point is selected as best design point. The internal energy at this point is coming out to be 29.052 BTU/lbm which is equivalent to 67574.952 J/Kg.

COMPARISON BETWEEN BASE AND OPTIMIZED DESIGN:



Optimized Design



Base Design

Fig.32: Side view and cross section of optimized design and base design with dimensions

Table 9: Optimal value and base value of corresponding parameter

DESIGN PARAMETER	OPTIMAL VALUE	BASE VALUE
L_1	330	330
L_2	460	490
L_3	360	330
θ_1	180	150
θ_2	133.42	150
T	2	1.6
A	90	80
B	72	80

Lengths are in (mm) and angles in degrees.

The internal energy for the optimized design is coming out to be 67574.952 J/Kg which is significantly higher than the base model value of 35560 J/Kg. The value of internal energy is significantly higher than any other subsystem which is validated from the study done by Hao Chen et al [4]. This also shows chassis longitudinal member is the critical component during the frontal crash. Although the internal energy in this case is coming to be significantly higher than the value from the study done by Hao Chen et al [4] as instead of detailed model simplified model is designed for rapid estimate of results. [6]

7. INTEGRATION STUDY:

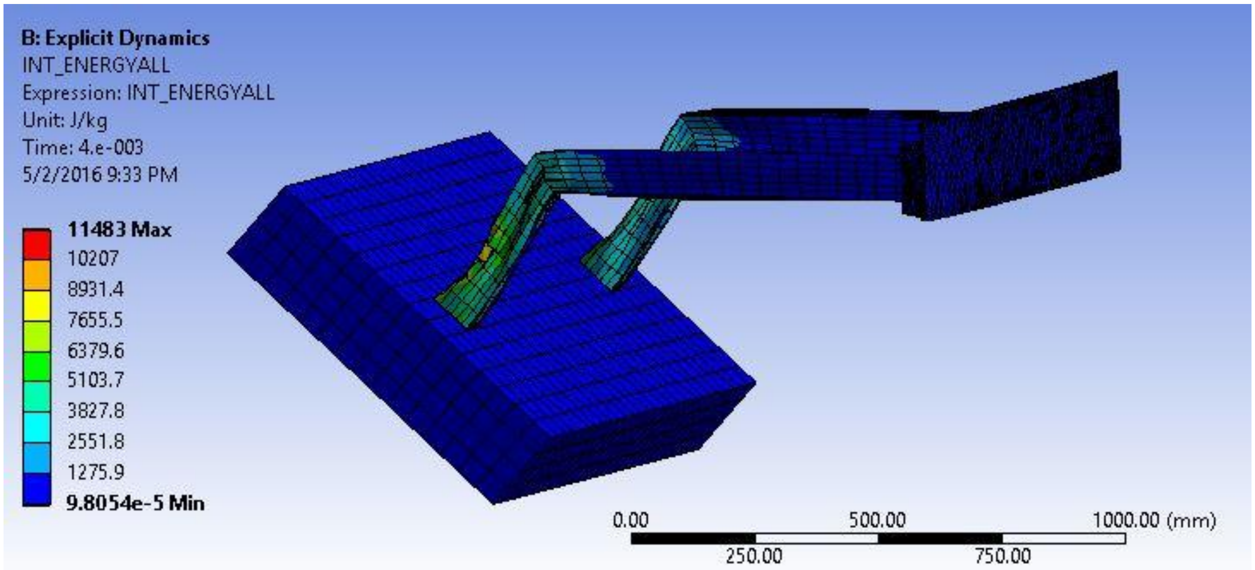


Fig.33: Internal energy analysis of assembly in ANSYS Workbench

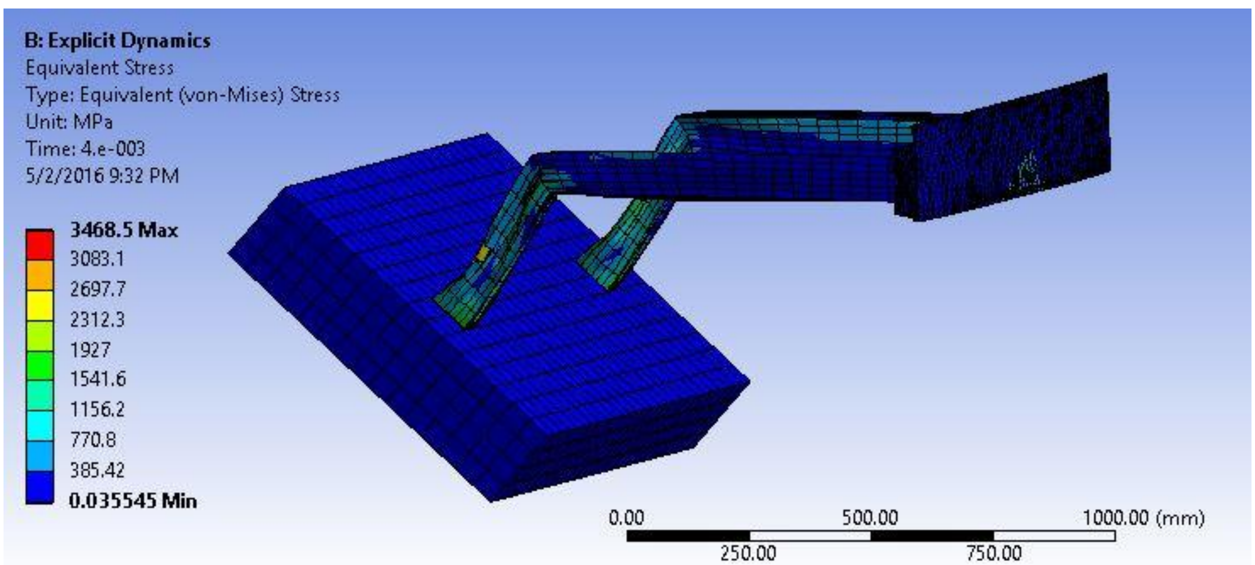


Fig.34: Von Mises stress analysis of assembly in ANSYS Workbench

Shouldn't the wall be on this side?

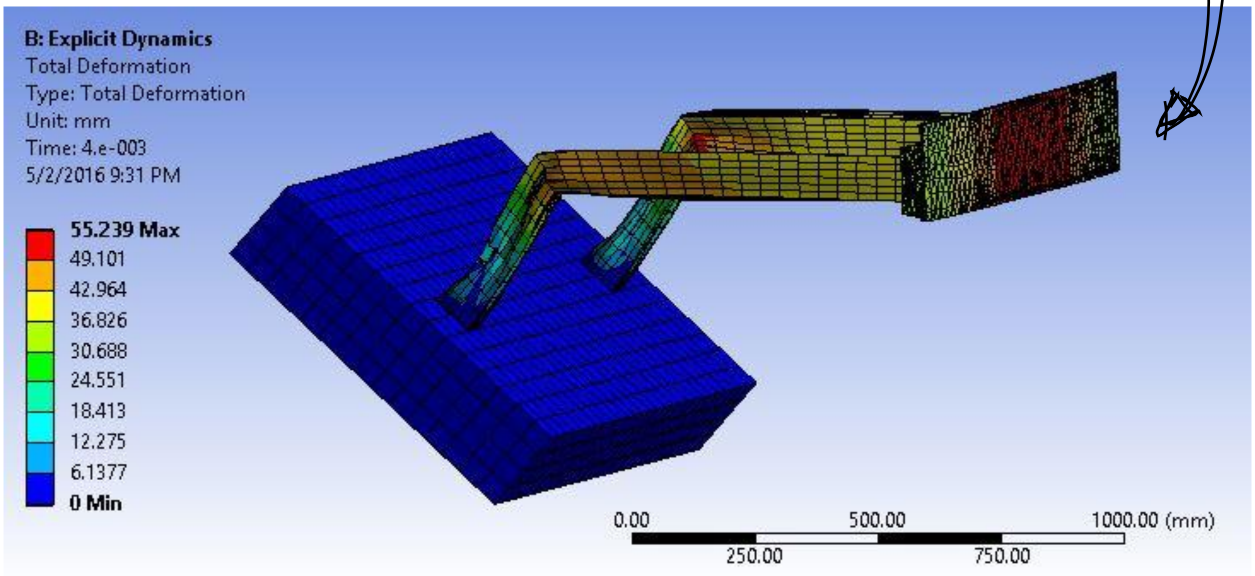


Fig.35: Total deformation analysis of assembly in ANSYS Workbench

The above three figures show the results of optimized assembly under the crash test with standard conditions. From the results it is evident that for the same run time of analysis as the individual components, energy absorbed values are scaled down for chassis as the deformation is less as compared to component analysis. And energy absorbed for bumper assembly is close to the component analysis.

Appendix 1

Iterative selection of boundary condition for Bumper beam

In order to effectively use the computation resource and reduce nodes in FEM calculation rigid constraint were considered for bumper beam simulation. Following is the summary of iterations conducted with results –

Displacement constraint at interface of bumper beam and chassis long member

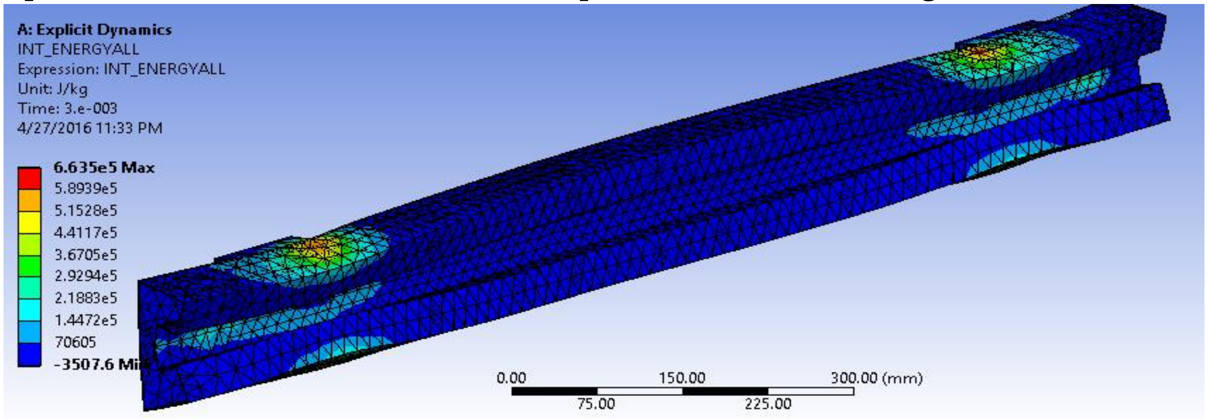


Fig. 1: Energy plot from case 1 constraint

Max. Energy is calculated to be 6.6e5 J/Kg and max. Von-Mises stress as 39860 MPa. These values are unrealistically large due to stress concentration near the constraints as shown in Fig. 1.

This due to fact that in service beam will never undergo such a severe impact as the energy is simultaneously transmitted to chassis long member. Using this result for further will lead to optimization to dimensions of the beam to maximize this concentration and will lead to further unrealistic and incorrect values.

Displacement constraint at extracted surface on rear of beam corresponding to assembly of chassis long member

Max. Energy is calculated to be 1.4e5 J/Kg and max. Von-Mises stress as 48674 MPa. These values are unrealistically large due to stress concentration near the constraints as shown in Fig. 2. And further pose similar problem as case 1 constraint.

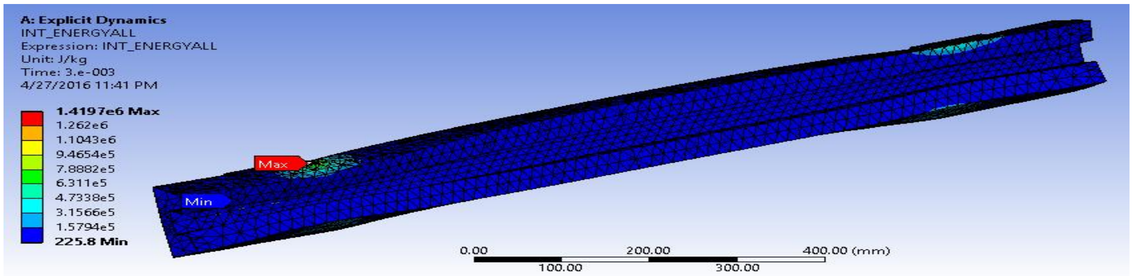


Fig. 2: Energy plot from case 1 constraint

Displacement constraint at extracted surface (1mmX1mm square) on rear of beam at connection with chassis long member

Max. Energy is calculated to be 7.61e5 J/Kg and max. Von-Mises stress as 629 MPa. Energy value is unrealistically large due to stress concentration near the constraints. And further pose similar problem as case 1 constraint.

Displacement constraint at edges (similar to bumper stiffener)

Max. Energy is calculated to be 1.6e5 J/Kg and max. Von-Mises stress as 700 MPa. Energy value is unrealistically large due to stress concentration near the constraints. This case doesn’t simulate crushing but only bending and further pose similar problem as case 1 constraint.

Reference

[1] NHTSA. "Federal Motor Vehicle Safety Standards: Occupant Protection in Interior Impact". 49 C.F.R., Part 571.201.

[2] Ross, Hayes E., Sicking, D.L., Zimmer,R.A., and J.D. Michie. *"Recommended Procedures for the Safety Performance Evaluation of Highway Features"*. NCHRP Gabauer 9 Report 350, TRB, National Research Council, Washington, D.C., 1993.

[3] Douglas Gabauer, Robert Thomson "*CORRELATION OF VEHICLE AND ROADSIDE CRASH TEST INJURY CRITERIA"* 05-0283

[4] Zhida Shen, Xin Qiao and Haishu Chen *"BIW Safety Performance Research Based on Vehicle Frontal Crash"*

[5] Hao Chen*, Yali Yang and Liangjie Wang "*Vehicle Front Structure Energy Absorbing Optimization in Frontal Impact"*

[6] Yucheng Liu *"Development of simplified models for crashworthiness analysis." University of Louisville*