Design Analysis of Bumper Support Beam

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Dedicated to my parents

"For their love, endless support

And encouragement"

ACKNOWLEDGEMENTS

In the name of Allah, the Magnificent, the Merciful.

Research work is a never ending struggle, and I am happy to have support from family and friends. I have had my fair share of mistakes along the way, but as Einstein once said, “Anyone who has never made a mistake has never tried anything new.” After each road block, Dr. Shahid Ikramullah Butt has been a beacon of hope, enlightening the path and guiding me through it. I am forever in gratitude to him. Also the encouragement and guidance of my Guidance and Examination Committee is highly appreciated.

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ABSTRACT

Provision of safer vehicles with high fuel efficiency at competitive prices is a prime challenge for the automotive industry. In providing safety, bumper of the car plays a crucial role. It tolerates high velocity impacts. "Crash-Worthiness" is the vehicle response to impact force; less damage means good "Crash-Worthiness". Bumper beam generally defends vehicle vital parts from damage. Commercial front bumper beam of one of the best-selling cars in Pakistan is studied in this thesis. Efforts were taken to model the bumper beam precisely closer to the reality. Bumper Beam material, mild steel, was identified by Energy-dispersive X-ray spectroscopy (EDX) technique and mechanical testing. The barrier and bumper beam was designed and analyzed using RCAR standards. Front bumper beam test according to the RCAR standards were performed and the bumper beam completely fractured and hence failed the impact test. The bumper beam then optimized using the surface optimization. Radius of curve in the beam and thickness of the beam were our input design parameters. Experimental analysis was performed with 9 design points of radius and thickness. The results were inspected on the basics of total deformation, von Misses stress and directional deformation and energy absorbed. The optimum design presented in this paper shows significant improvement over the one in use and meets all the requirements of the RCAR front test.

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CHAPTER 1

Introduction

Provision of safer vehicles with high fuel efficiency at competitive prices is prime challenge for the automotive industry [1]. Automobile bumper system consists of frontal and rear structure of the vehicle that has the fundamental purpose of protection of passengers (/pedestrians) and vehicle body during low velocity impact [2]. Three main components: fascia, the absorber and the bumper beam constitutes the frontal bumper beam, shown in figure 1[4]. Fascia is generally considered to minimize the drag force and to enhance the elegance. It is generally considered as the non-structural component as it cannot bear the impact. The absorber damps a light portion of the energies therefore bumper beam becomes the crucial component to tolerate high velocity impact energies. The bumper beam is the key structural element; enough deformation of bumper beam is conventional to dampen the impact [11]. Therefore, Bumper beam deforms and absorbs energy, to protect the vehicle components and to minimize the injury risks of pedestrians [2-4]. The arduous task in designing the bumper beam is to design for optimal characteristics. Generally, trade-off between deformation and strength is made.

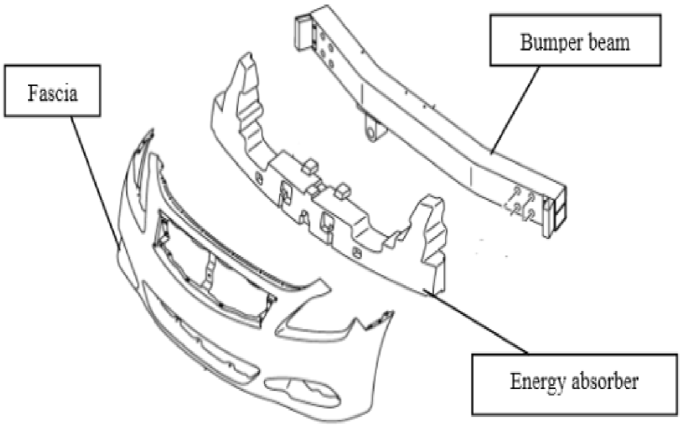

Figure 1- Components of Bumper System

In collision (impact) scenario two types of impacts occurs: elastic and plastic. Elastic loses less energy and plastic loses considerable amount like the vehicle to vehicle or collision with solid body. In the process of impact of bumper beam with the barrier, the law of energy conservation is preserved. “Crash-Worthiness” concept suggests that the energy generated during the impact is converted into the displacement (deflection) [39-40]. The equation of energy conservation in case of bumper impact is:

$$\frac{1}{2}mv^2 = E_{vke} + E_{bar} + E_{vint} \qquad (1)$$

Where $\frac{1}{2}mv^2$ is the kinetic energy of the moving body before impact and E_{vke} is the kinetic energy of the entire system after the impact. Whereas energy consumed the vehicles is E_{vint} and the energy to deform the barrier is represented by E_{bar}.

Due to utmost importance of safety structural “Crash-Worthiness” of automotive parts becomes decisive factor [4-8]. ““Crash-Worthiness”” is the vehicle response when it experiences impact force. Less damage means good “Crash-Worthiness”. Structural members are analysed for “Crash-Worthiness” before executing. Passengers’ safety during vehicle crashes can be assured by a good bumper, albeit to a certain limit. The manufactures make sure that material does not have crash or failure [10].

Bumper beam generally defends vehicle parts like cooling systems, radiators, hoods, fenders headlights, and exhaust system from damage. At high velocity bumper beam guides the crash forces into the vehicle's body structure so that the survival of the passengers can be ensured and the collateral damage minimized [9]. Simulation models for bumper beams under impact conditions are devised by many researchers. To get more practical finite element model, effect of strain rate during transverse load impact analysis for the bumper beam was studied by Kokkula et al. [28]. Marzbanrad et al. [12] considered high strength sheet molding compound (SMC), glass-mat- reinforcement thermoplastics (GMT) and aluminum and studied the effect of thickness, material, shape and impact conditions on bumper-beams when subjected to low- velocity impact [12]. Farkas et al. [13] collide bumper beams at 4.444 m/s for frontal offset test and found optimal geometry for dual channel bumper beam. Beams were also analyzed for pole test at 4.166 m/s.

Eight bumper beams of different cross-sectional shapes and having same material and under same low velocity impact test conditions were studied by Davoodi et. al [37]. The work mainly focuses on the material and manufacturing optimization along with improvement of energy optimization by different cross-sectional profiles. Their work further [38] shows that during low velocity impact bumper beams absorb energy by deflection whereas in high velocity impact energy is absorbed by deformation.

Hosseinzadeh et al. [42] proposed model of the research has equal strength and rigidity of the structure, reduction of material, ease of manufacturing by simplifying geometrical shape and reduction of production cost are studied and proved. There is a lot of work on the optimization of energy absorbers but the effect of transverse load on bumper beams is comparatively very low [14-27]. Environmental protection and energy conservation are becoming more prominent factors in the automotive structure design. Due to these factors, lightweight structure design of the vehicles has grown popular in the automotive industry [41]. The competent process for optimal design of bumper beam is to maximize the "Crash-Worthiness" at higher speeds. Therefore, equipping for the better protection of passengers, and in the meantime applying limits on performance of lower-speed crash. In case of low velocity impacts, all the energy other than the absorbed by bumper cover, radiator, body panel and reinforcement should be absorbed by bumper beam. Most of the bumper-beams are box shaped but their profile can be modified to enhance their impact performance [9].

This study examines and optimizes a commercially endorsed Bumper Beam. The commercially available beam material was identified using EDX and mechanical testing then it after modelling the impact was analyzed using the Ansys Work bench [Explicit Dynamics]. The proposed optimized design proved to be more crashworthy owing to the simulated and experimental results. The proposed profiles are designed in such a way that they can replace the existing structure without any alteration.

CHAPTER 2

Methodology

This chapter details out the research methodology for the present study. It illustrates the research objectives and methodology adopted to achieve objectives. The objectives of the study were to model the commercial bumper and the barrier according to the Research Council for Automotive Repair (RCAR) and analysis of the commercial bumper beam. The commercial beam was then optimized to achieve better results. These objectives were achieved by using the methodology illustrated in Figure 2.

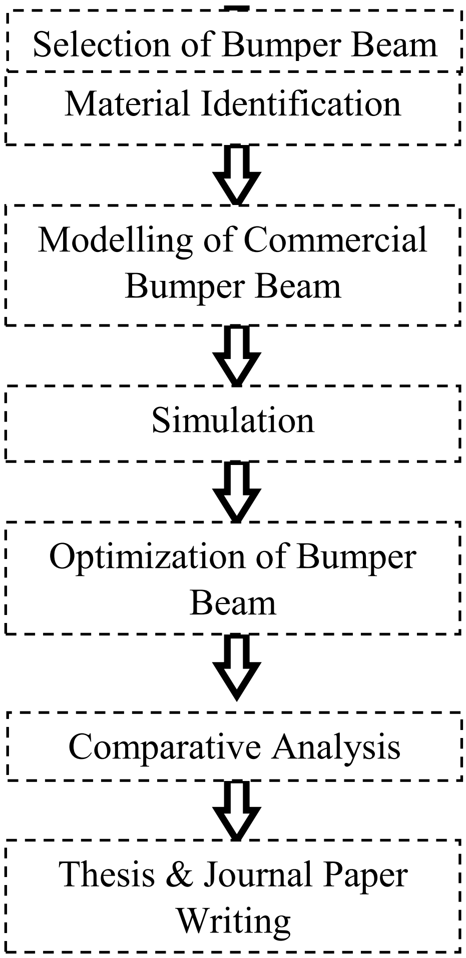

Figure 2- Research Methodology

Bumper beam of one of the best-selling cars in the last decade (average 20.91 % from 2007-17) was selected to study. To acquire realistic results, beam material should be known. Energy-dispersive X-ray spectroscopy (EDX) technique and mechanical tests were performed to identify the material. The commercial beam was modelled and analyzed according to the RCAR standards. The bumper beam failed to achieve the required results from the test. Then the bumper beam was optimized using the surface optimization. Radius and the thickness were the input design parameters. Then the results were interpreted based on total deformation, von misses stress and directional deformation [12-13]. The final optimized structure shows significant improvement above the original design

CHAPTER 3

Modeling, Simulation & Experimentation

3.1 Material Identification

Beam material should be known to acquire realistic results. Energy-dispersive X-ray spectroscopy (EDX) technique was used to determine chemical properties. It is a quantitative analysis method, the specimen in figure 5 was excited by explosion to X-ray spectrum. When atoms are energized they emit unique electromagnetic spectrum. This spectrum was observed [as shown in figure 6] which reveals the chemical properties of the material. Table 2-1 shows the chemical properties of our bumper beam material. Mechanical properties were determined by performing tensile and hardness tests. ASTM standards E8/E8M−16a and A370 – 17 were used, Table 2-2 shows the mechanical properties. After analyzing the chemical and mechanical properties the material was identified as AISI 1006 (Mild steel).

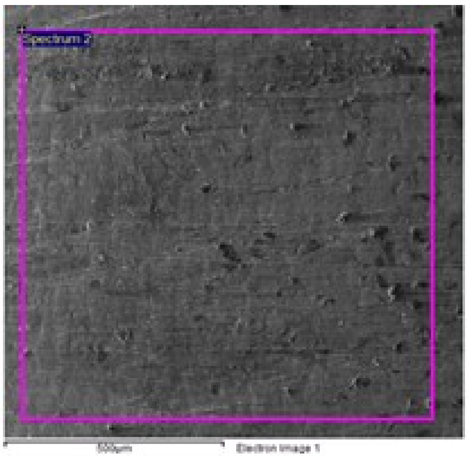

Figure 3 - EDX specimen

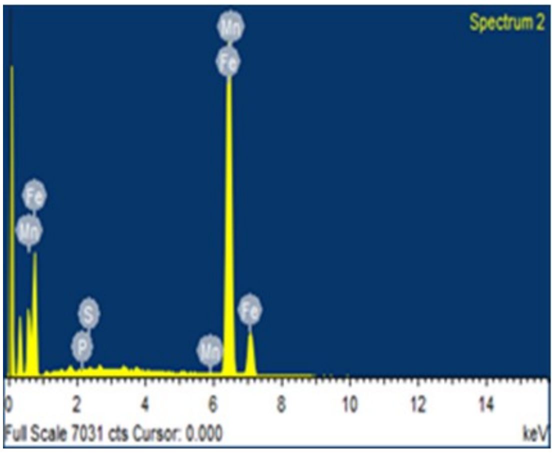

Figure 4 - Electromagnetic Spectrum

Table 1 Chemical properties

Element	P	S	Mn	Fe	Total
Weight%	**0.01**	**0.11**	**0.20**	**99.68**	**100**

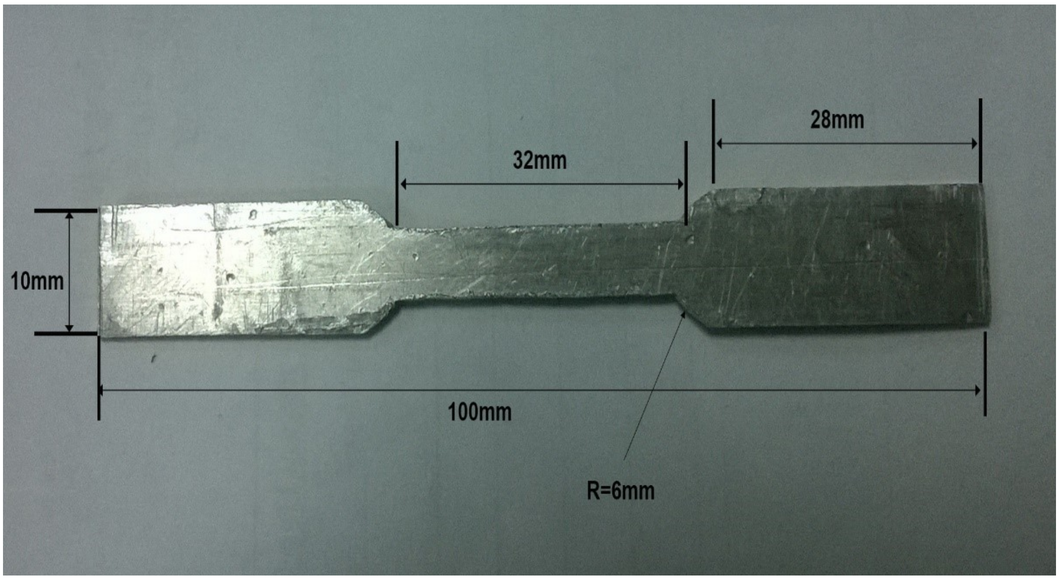

Figure 5 - Tensile test sample

Table 2 Mechanical Properties

Ultimate Tensile Strength	Yield Strength	Modulus of Elasticity	Brinell Hardness
287 MPa	163 MPa	200 MPa	85 HB

3.2 Numerical Simulation

3.2.1 Modeling of Commercial Bumper Beam

In this paper, a commercial front bumper beam of one of the best-selling cars in Pakistan during the last decade (average 20.91 % from 2007-17) was selected to be studied [43]. Efforts were taken to model the bumper beam precisely closer to the reality. The beam and its supports were designed using the Ansys design modeler. The barrier was designed as specified in the Research council for automotive repair (RCAR) manual for the impact of the beam. Figure 6 (a & b) shows the bumper beam top and side view whereas c & d represents top and size view of the barrier. Figure 7 shows the impact mechanism of bumper and barrier (all dimensions are in mm).

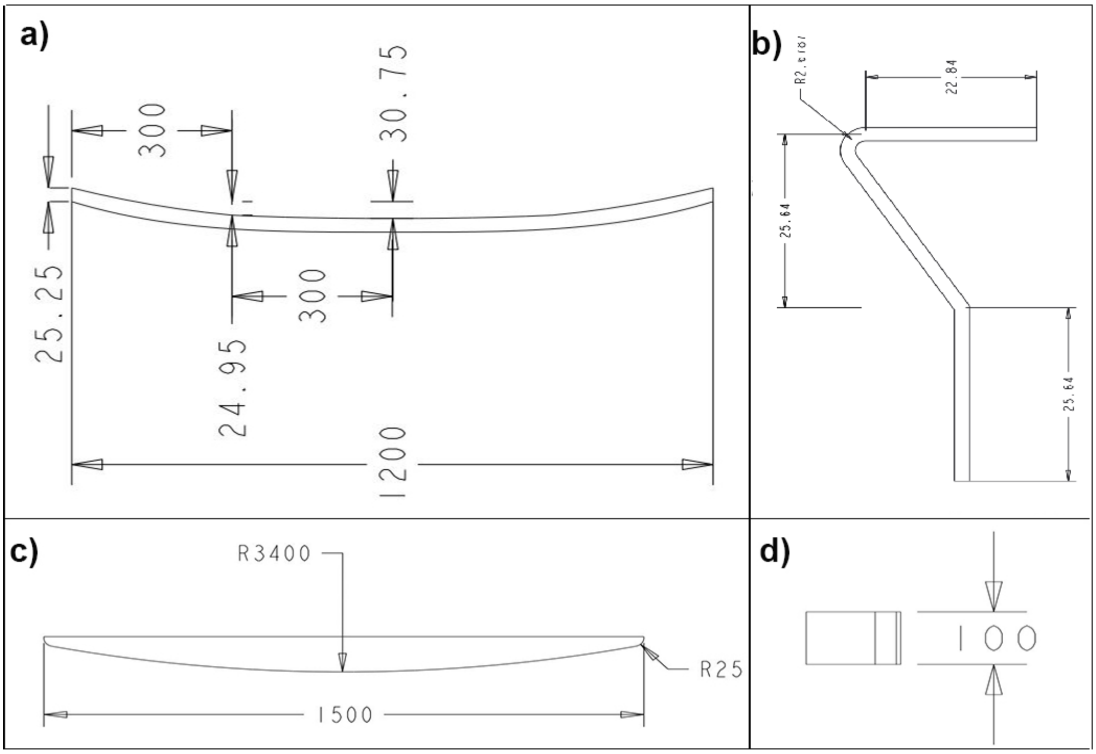

Figure 6 Drawing of Beam and Barrier

Figure 7 - Impact Mechanism

Model of the whole vehicle body requires extremely prolonged computational time and requires precise substantial number of iterations to optimize the process. Therefore, the whole vehicle model is not a reasonable choice and so, it is rational to model bumper beam and barrier [9]. To further minimize the simulation time symmetry condition was applied on both beam and the barrier.

The figure 8 shows the symmetric model used for the simulation. Point mass was applied to the barrier to which is equal to the mass of the vehicle. Therefore, the total mass of the barrier and moving with the velocity of 2.7777m/s (10 Km/h) striking the fixed beam relatively represents the actual car crash at 2.7777 m/s to the barrier mounted to a support.

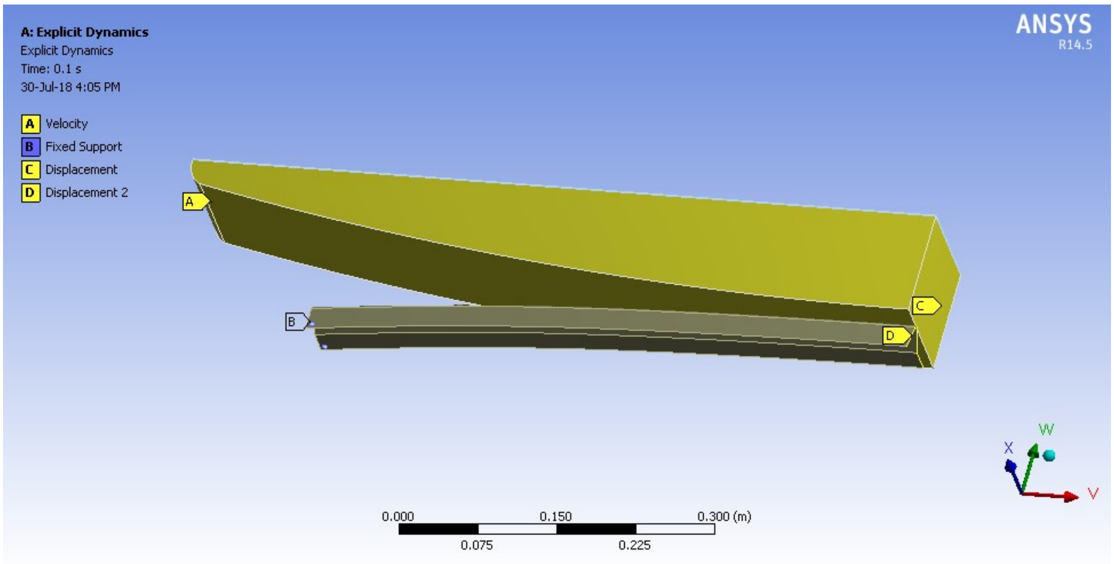

Figure 8 - Schematics of Proposed Profiles

3.3 Meshing

The model created in the design module was meshed using the advance size function. The beam was meshed with element size of 5mm with bias factor 6. The biasness was towards the center were the barrier hits first. Precise meshing of the barrier was also important for the proper results of analysis. In horizontal direction the barrier was meshed using the element size 10mm with bias factor of 10 and in vertical direction the element size was 2mm with bias factor of 2. Then mapped face meshing was applied resulting 19946 elements of whole structure (as shown in figure 9 and 10). The beam was fixed from one end and the symmetry conditions were applied to 2nd end.

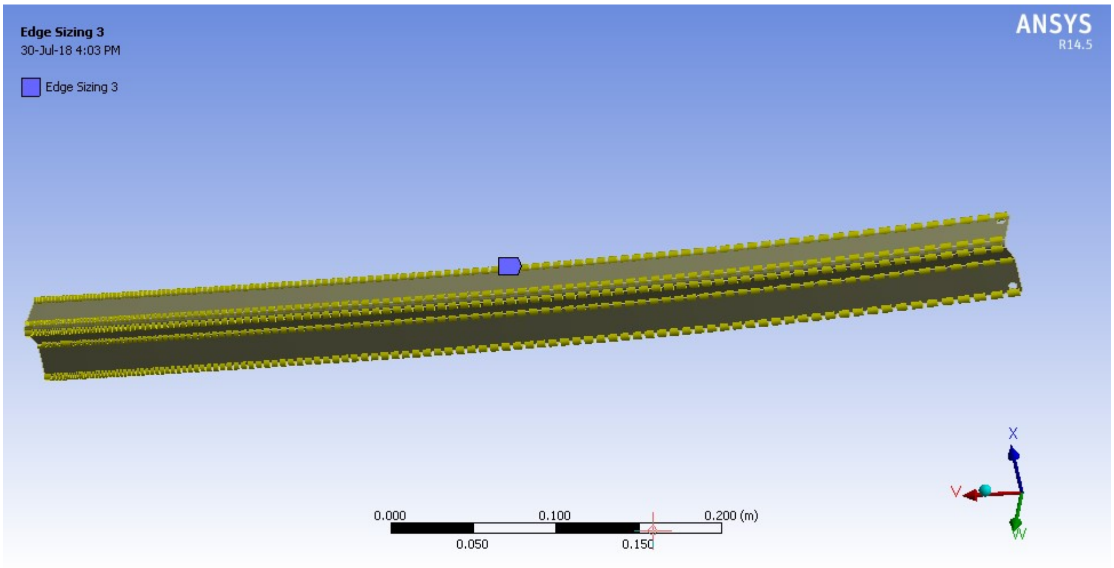

Figure 9 – Meshing of Beam

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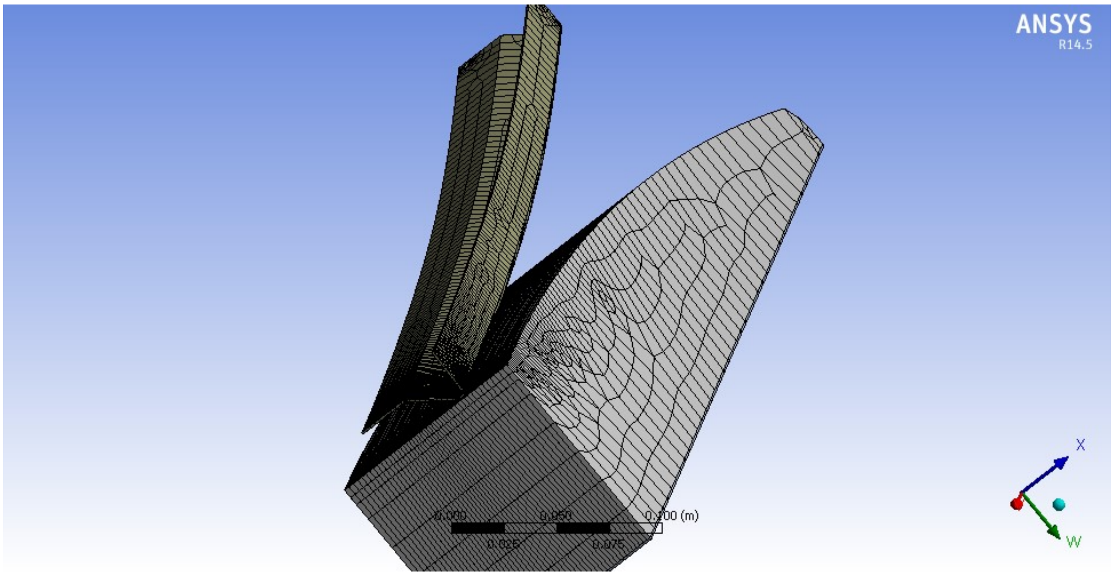

Figure 10- Meshed Impact Mechanism

3.3 Simulation

Figure 11 shows the velocity versus impact time graph for the crash test. The impact time in case of front test was 0.1 sec. The barrier moves forward for 0.05 sec with constant velocity of 2.7777 m/s and then the barrier starts to deaccelerate at the rate of 55.554 till the end.

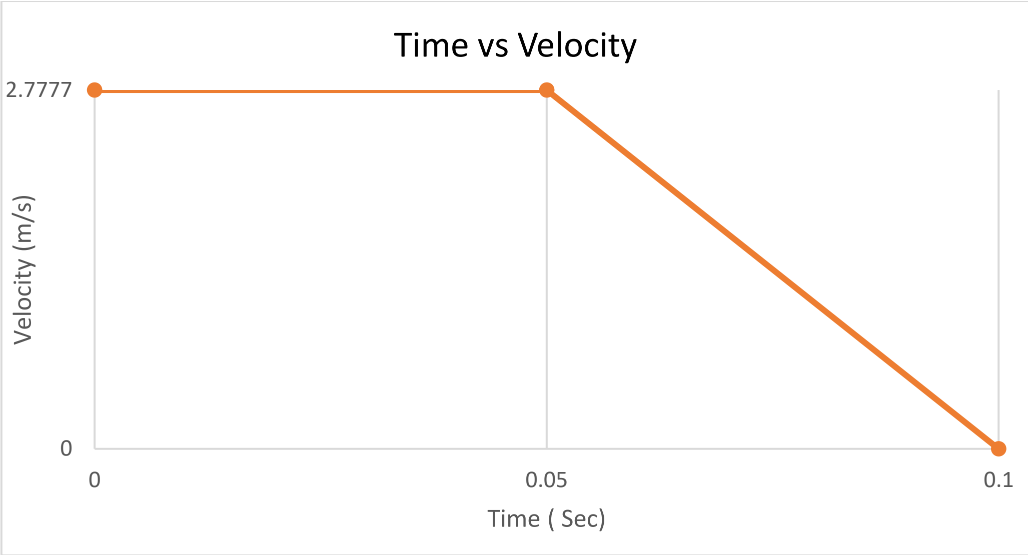

Figure 11- Velocity vs Time Relation

The Explicit dynamics module was used to perform the front test. The model was designed to replicate the actual test. Both the beam and barrier were divided symmetrically to minimize the simulation time without violating any fundamental law. According to the RCAR standards the bumper beam tolerates the impact of the applied momentum, which is equivalent to the 2.7777 m/s velocity into the mass of the car. The impact time in this case is this case is 0.1 sec as shown in figure 9. The barrier moves forward for 0.05 sec with constant velocity of 2.7777 m/s and then the barrier starts to deaccelerate at the rate of 55.554 till the end.

The figure 8 shows the structural limitations applied in the simulation, the beam is fixed adjacent to the end through holes precisely in the way the beam is fixed to the vehicle. The barrier with mass proportional to the vehicle mass and velocity 2.7777 m/s was set to impact the barrier for the impact time of 0.1 sec.

The Crash-Worthiness of a bumper is characterized by its deformation to prevent the vital components of the beam. Deformation of bumper beam is acceptable [5] so long as it ensures the safety of the vehicle components. The numerical results generated by the Ansys clearly presented that the bumper beam completely fractured, which means that the beam failed to meet the RCAR crash test standards.

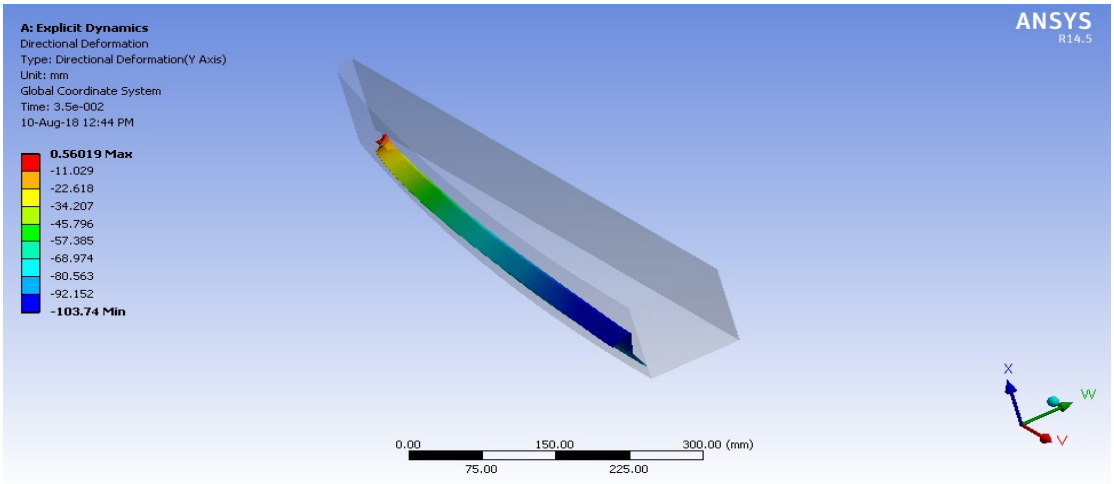

Figure 12 – Directional Deformation of the Beam

The bumper beam completely fractured at 0.035 sec which means after 0.035 seconds the barrier will collide with the vital components of the vehicle. And the crash will result in severe damage to the vehicle and the passengers.

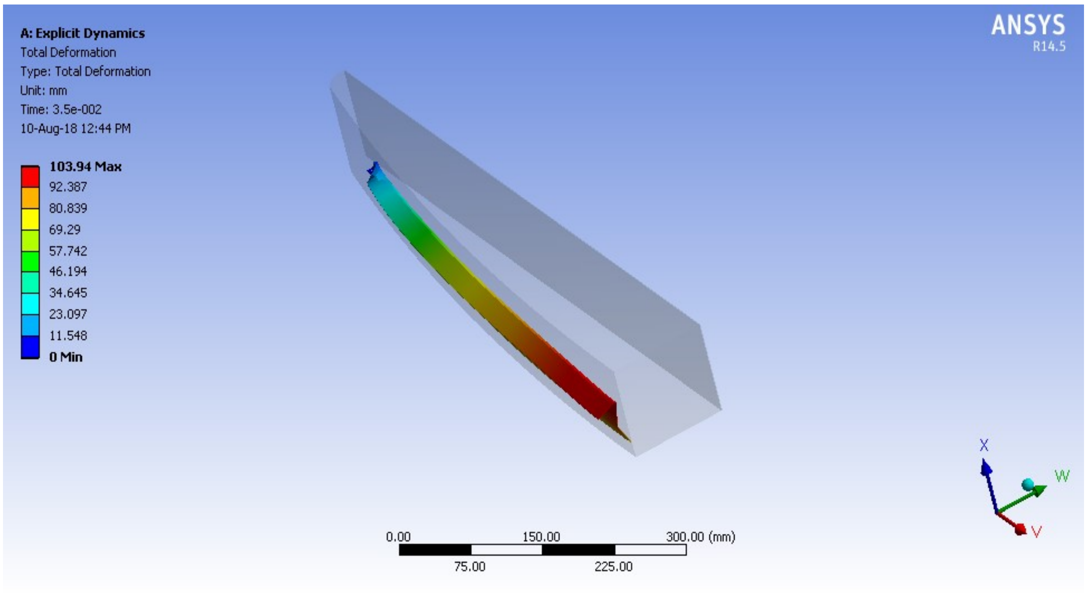

Figure 13 – Total Deformation of Beam

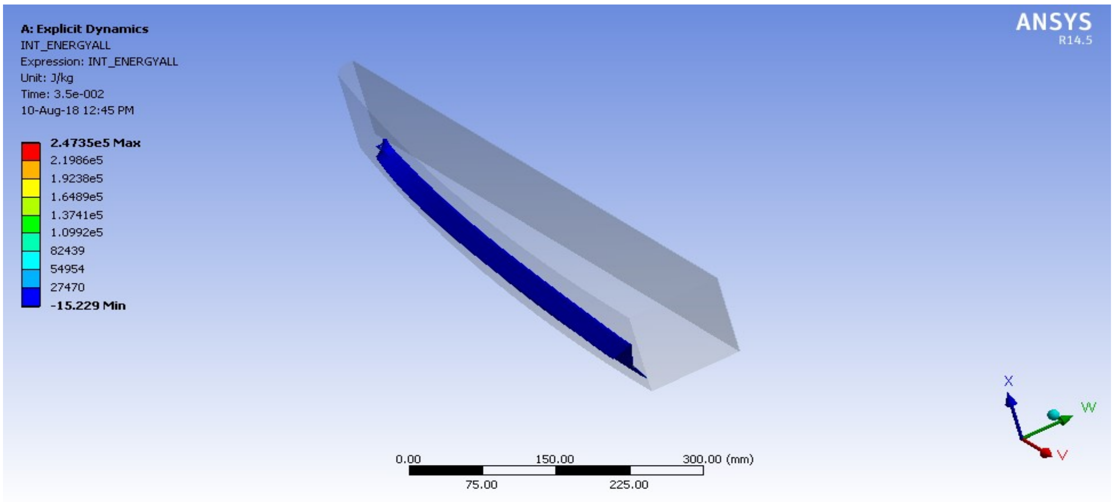

Figure 14 – Energy Absorbed by Beam

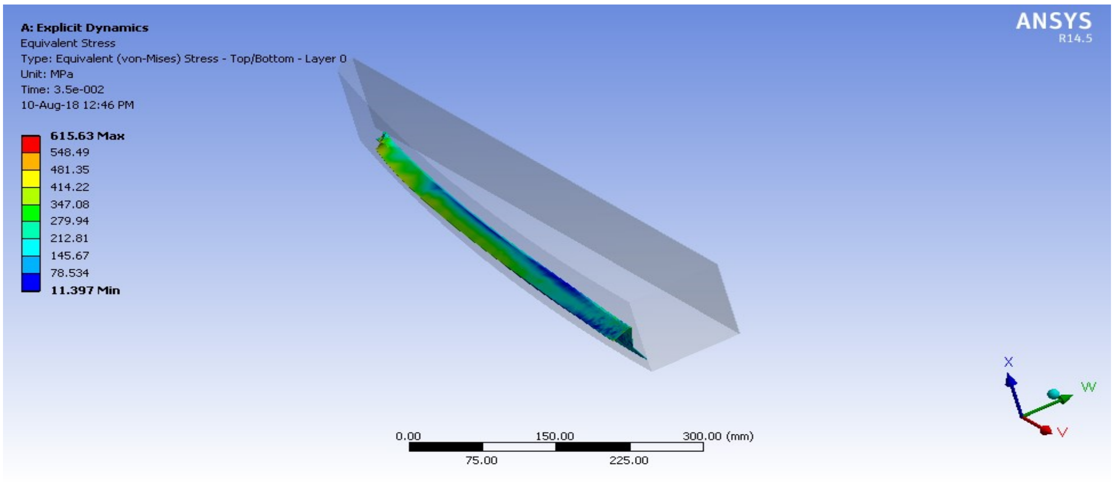

Figure 15 – Stress Endured by Beam

Bumper beam with insignificant impact strength cannot trusted to be a part of the most sold vehicle in Pakistan. Therefore, the beam should be optimized to endure maximum impact for the safety of passengers and the vehicle components. Our objective was to optimize the beam in such a way that the new beam design can be readily integrated without any modification to the vehicle structure.

3.4 Experimentation

Front and rear components of the cars usually collapse in bending mode. So, bending collapse must be considered in designing the vehicle structure. Literature suggests that three-point bend test can be studied for Crash-Worthiness. Kroger studied the Crash-Worthiness of the beams by three-point bend test [37-43]. Three-point bending test was done considering ASTM standard E290 − 14 for the profile shown in Figure 25. The results of three-point bend test were further processed and then compared with numeric simulation.

Figure 16 Bend Test Sample

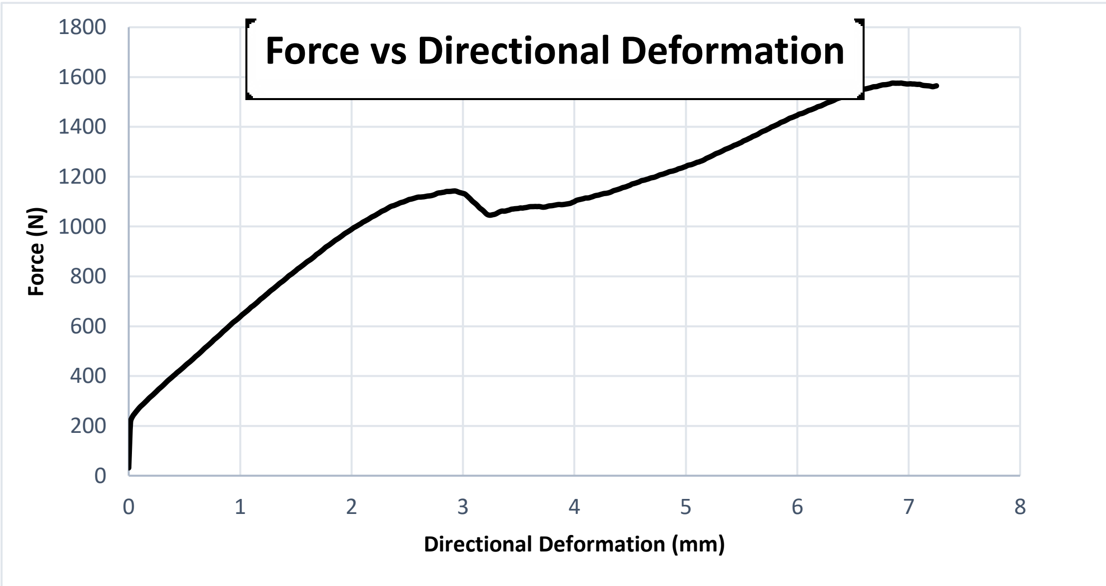

Figure 17 Experimental Results

Force is product of object's mass and its acceleration. Acceleration is the rate of change of velocity. So, force can be described as:

$$F = m * \frac{v}{t}$$

By using this equation, the force bearded by the beam was calculated and the force trend is shown in figure.

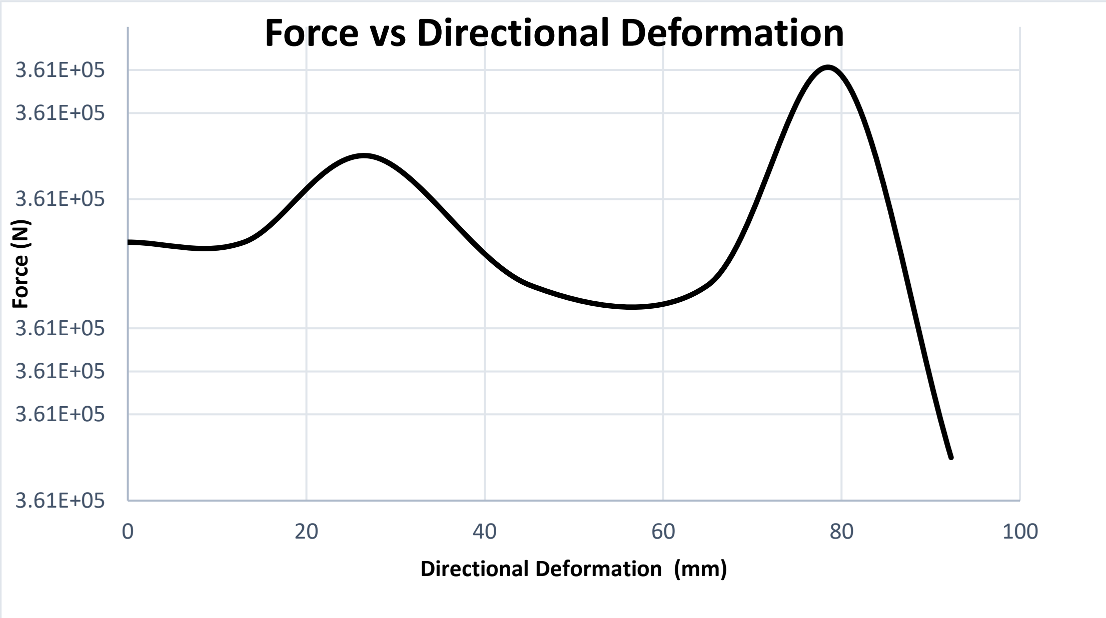

Figure 18 Numerical Simulation Results

The figure 17 and 18 show experimental and numerical results of the bumper beam. The trend of experimental results equivalent to that of the numerical results till 7 mm deflection, the maximum deflection of experimental results.

3.5 Modeling of the Propose Profiles

Most of the bumper-beams are box shaped but their profile can be modified to enhance their impact performance [9]. Novel design for the bumper beam profile was considered to design. Three-point bend test was performed on the new profile and its results were compared with the bending test of the commercial bumper beam.

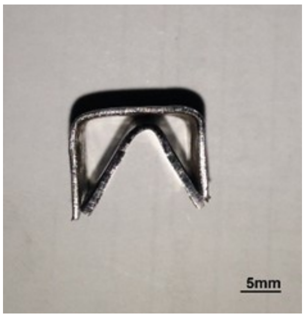

Figure 19 Bend Test Sample of Proposed Profile

The experimental results of the proposed bumper beam profile were very promising. Experimental result comparison of both profiles is under.

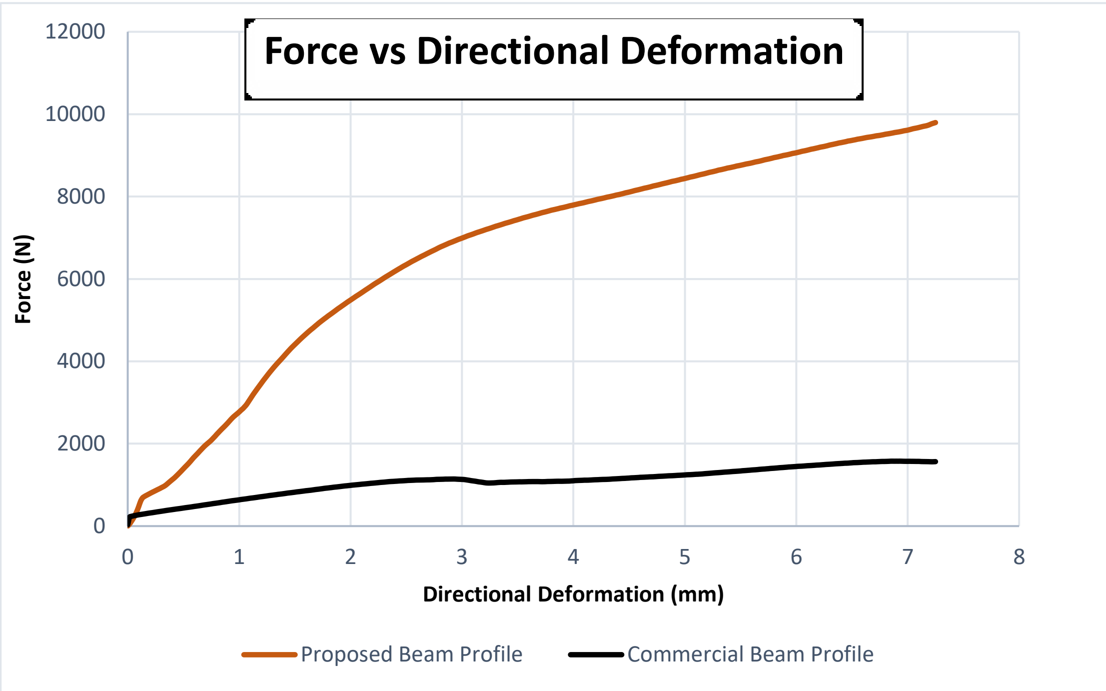

Figure 20 Experimental results- Relative study of profiles

Then the proposed profile was also modeled in Design Modular similar to the commercial profile.

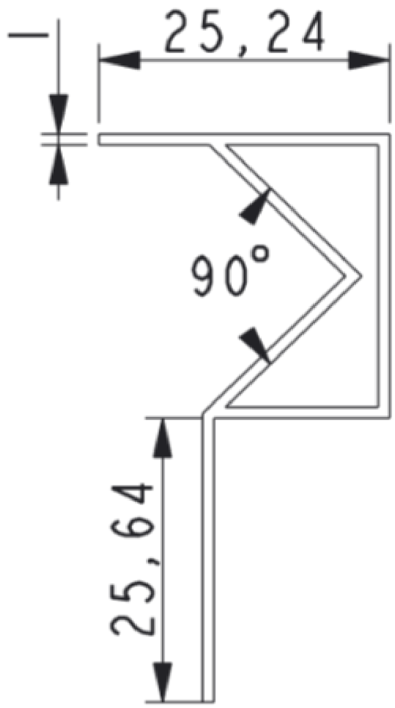

Figure 21 Drawing of Proposed Profile

CHAPTER 4

Optimization of the Bumper Beam

RCAR standards emphasizes that the bumper beam must withstand front impact test to be considered to use in a vehicle. The bumper beam of our car failed to sustain the impact so, it was decided to optimize the bumper beam for better results. The two most common ideas for the optimization bumper are: to use composite materials for better results and second is to optimize the geometry to absorb more impact energy. There is a lot of work on the optimization of energy absorbers but the effect of transverse load on bumper beams is comparatively very low [14-27]. Therefore, the effect of momentum on the shape of bumper beam was studied. The beam was optimized using the Response surface optimization.

4.1 Parametric Constrains & Objective

The first task for optimization was to identify the guanine constrains of parameters and objectives. By carefully examining the bumper beam profile and considering that the resultant bumper must be readily attached to the bumper beam two design parameters were opted for the optimization.

Thickness and radius of the curve were opted to be parameterize for optimization (shown in figure 6). As alteration to these parameters under limits will be easily adjusted in the vehicle without any structural change. The radius of the curve can be varied from 2.4 to 10 mm and maximum beam thickness which can readily be incorporated in the vehicle is 8mm. Therefore, the limits for thickness were up to 8mm and for radius 10 mm. The design point generated for the optimization matrix were:

Table 3 Optimization Matrix

Sr #	Thickness (mm)	Radius (mm)
1	4.45	6.205
2	4.45	2.410
3	4.45	10
4	0.9	6.205
5	8	6.205
6	0.9	2.410
7	0.9	10
8	8	2.410
9	8	10

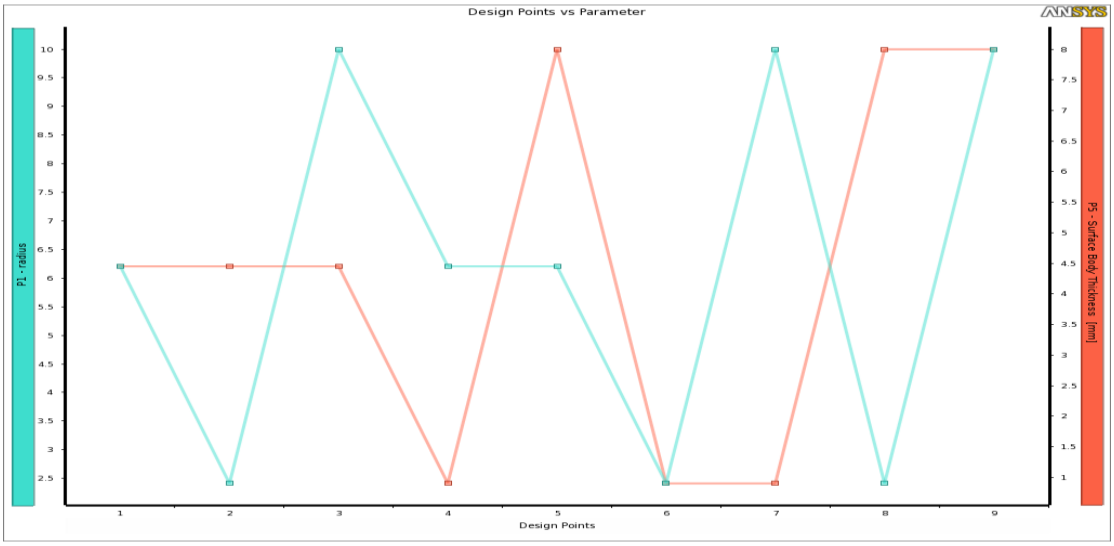

Figure 22 – Parameters vs Design Points

The important output parameters which were to be observed carefully were directional deformation, total deformation, stress and energy absorbed by the beam. The constrains applied and the parameters were:

Table 4 Parameters, Objective and Constrains

Sr #	Parameter	Constrain	Objective
1	P1 - Radius	2<= R => 10	Maximize
2	P5 - Thickness	0.9<= T =>8	Maximize
3	P3 - Directional Deformation	DD <= 30	Minimize
4	P4 - Equivalent Stress		Minimize
5	P6 - Total Deformation		Minimize
6	P7 – Energy Absorbed		Maximize
7	P8 - Maximum Shear Elastic Strain		Minimize

The response generated for the design points was:

Table 5 Optimization Results

Sr #	Radius (mm)	Thickness (mm)	Directional Deformation mm)	Equivalent Stress (MPa)	Total Deformation (mm)	Energy Absorbed (KJ kg^-1)	Equivalent Plastic Strain (mm mm^-1)
1	6.205	4.45	445.440	24601.093	793.653	263836.945	6.802
2	2.410	4.45	73.062	584.820	73.082	883.967	0.420
3	10	4.45	109.702	1120.257	109.735	760.560	4.508
4	6.205	0.9	1492.193	11662.617	5207.236	34750.719	7.846
5	6.205	8	97.801	614.376	98.024	114.038	0.831
6	2.410	0.9	432.065	1983.301	492.847	24266.697	5.635
7	10	0.9	110.249	615.373	110.347	2120.976	4.173
8	2.410	8	17.733	461.258	17.764	370.643	0.154
9	10	8	75.216	512.323	75.249	177.008	0.232

The maximum deformation beam can bear before facture is 103mm. All the design points having directional deformation greater than 103 mm are failed design points.

4.2 Proposed Profile Optimization

Due to limited computational power the proposed bumper beam was optimized for 2D structure. Therefore, only angle is optimized for better results.

Table 6 Optimization Matrix for Proposed Beam

Sr #	Angle
1	129
2	81
3	180
4	88.895
5	170.5

Table 7 Optimization Results of Proposed Profiles

Sr #	Angle	Directional Deformation mm)	Equivalent Stress (MPa)	Total Deformation (mm)	Energy Absorbed (KJ kg^-1)	Equivalent Plastic Strain (mm mm^-1)
1	129°	0.24806	187.84	0.4747	35.04	0.57714
2	81°	0	0	0	0	0
3	180°	4.3568	69.522	8.3044	48.251	1.6905
4	88.895°	1.1459	125.4	4.3956	29115	2.7338
5	170.5°	0.895	187.44	3.459	40.495	1.739

4.2 Deformation

After carefully analyzing the geometry it was concluded that the radius of the curve can be varied from 2.4 to 10 mm and thickness of the beam can be varied up to 8mm. The maximum deformation beam can bear before facture is 103mm. All the design points having directional deformation greater than 103 mm are failed design points. So, the design points with valid results were:

Table 8 Deformation Results

Sr #	Radius (mm)	Thickness (mm)	Directional Deformation mm)	Total Deformation (mm)

1	2.410	4.45	73.062	73.082
2	6.205	8	97.801	98.024
3	2.410	8	17.733	17.764
4	10	8	75.216	75.249

Deformation is any change in the shape or size. Total deformation is the vector sum of all directional displacements. Total Deformation is the root mean square of directional deformations in global X, Y and Z coordinates. Directional deformation is the displacement of the system in a particular axis, in this scenario the deformation towards the vehicle components was observed.

The results trend showed that by increasing the thickness directional deformation and total deformation decreases. When the radius was less than 3 mm the beam can absorb the impact without fracturing but for the radius more than 3mm and less than 8mm the beam will fracture completely. So, do to manufacturing accuracy beam with radius more than 8 mm would be reliable.

These results clearly state that the minimum thickness of the beam must be greater than 4 mm.

For the proposed beam profile, buckling was observed when the angle was increased more than 100°. The profile showed stable results for the angle less than 100°.

Table 9 Proposed Profile- Deformation Results

Sr #	**Angle**	**Directional Deformation mm)**	**Total Deformation (mm)**
1	129°	0.24806	0.4747
2	81°	0	0
3	180°	4.3568	8.3044

4	88.895°	1.1459	4.3956
5	170.5°	0.895	3.459

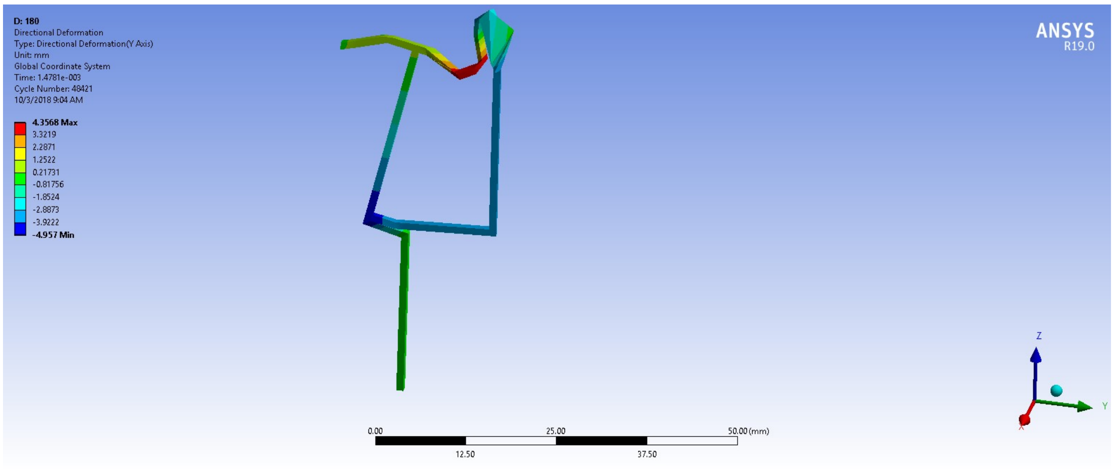

Figure 23 Directional Deformation for 180° Angle

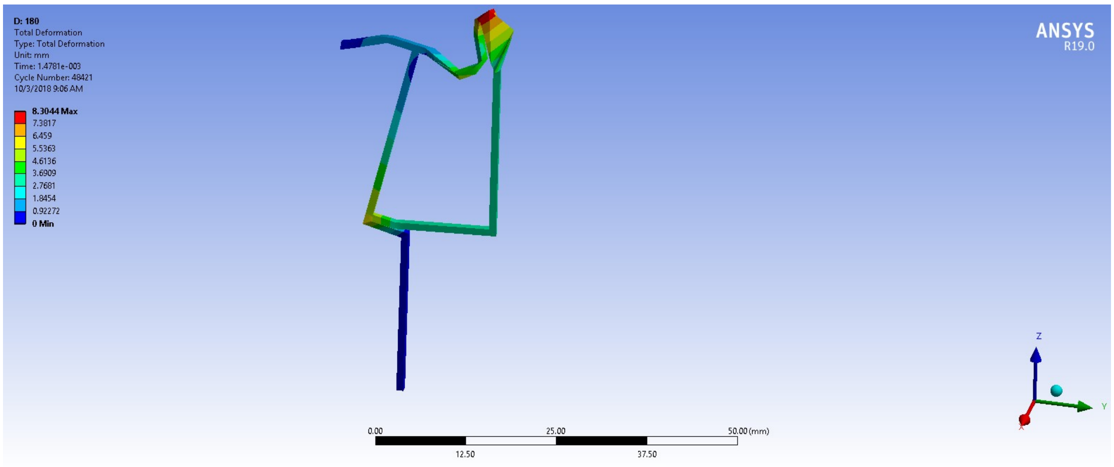

Figure 24 Directional Deformation for 170.5° Angle

4.3 Stress & Strain

Force per unit area is defined as stress. The equivalent stress (von misses stress) designates the plastic deformation of the material. When the Von Misses exceeds the material strength, it causes the structural failure. Higher value of Von Misses indicate that the object is more prone to failure. When the Von Misses strength exceeds the material strength, it causes the structural failure that means the higher value of Von Misses indicates that the material is more prone to failure. Strain is the measure of the deformation produced in a member by the applied Load. When the deformation produces an angle between the originally perpendicular faces, such deformation is the result of the shear strain. If body does not return to the original shape after removal of shear stress its known as shear plastic strain. Mega pascal (MPa) is the unit stress and strain have no unit.

Table 10 Stress Strain Results

Sr #	Radius (mm)	Thickness (mm)	Equivalent Stress (MPa)	Equivalent Plastic Strain (mm mm^-1)
1	2.410	4.45	584.820	0.420
2	6.205	8	614.376	0.831
3	2.410	8	461.258	0.154
4	10	8	512.323	0.232

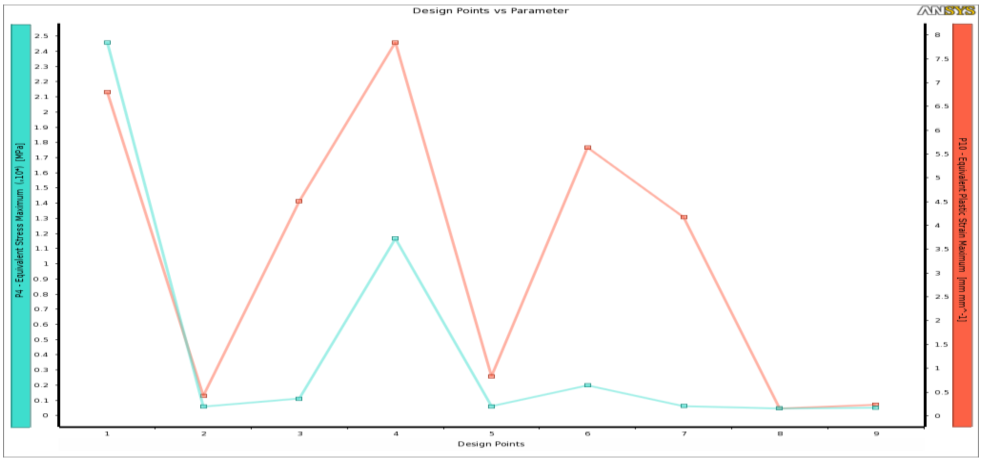

Figure 25 Design Point vs Equivalent Stress and Equivalent Plastic Strain

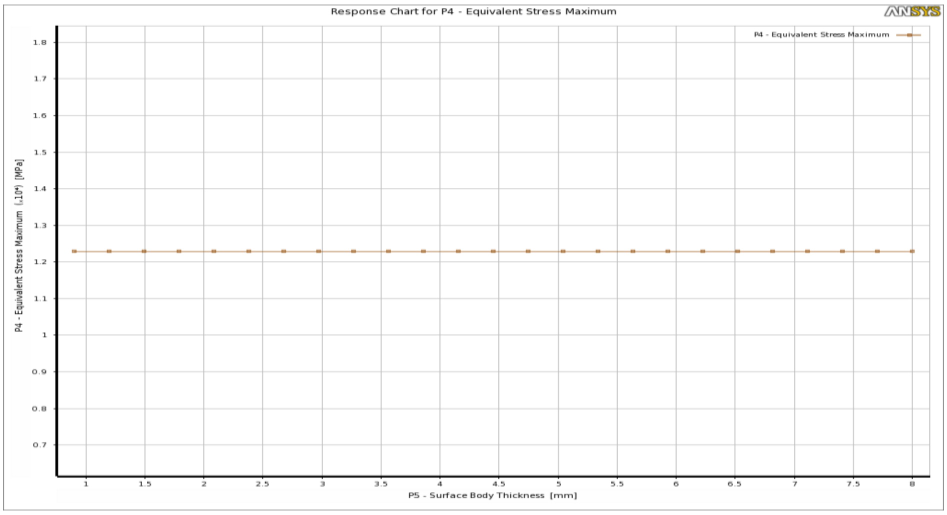

Figure 26 Thickness vs Equivalent Stress

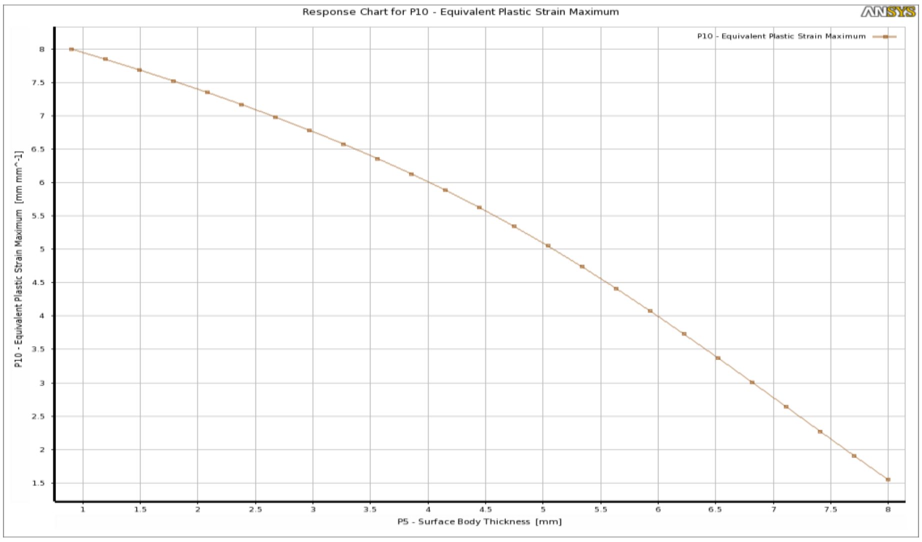

Figure 27 Thickness vs Equivalent Plastic Strain

The figure 23 and 24 clearly state that stress is not affected by the thickness and the strain has inverse relation with the thickness.

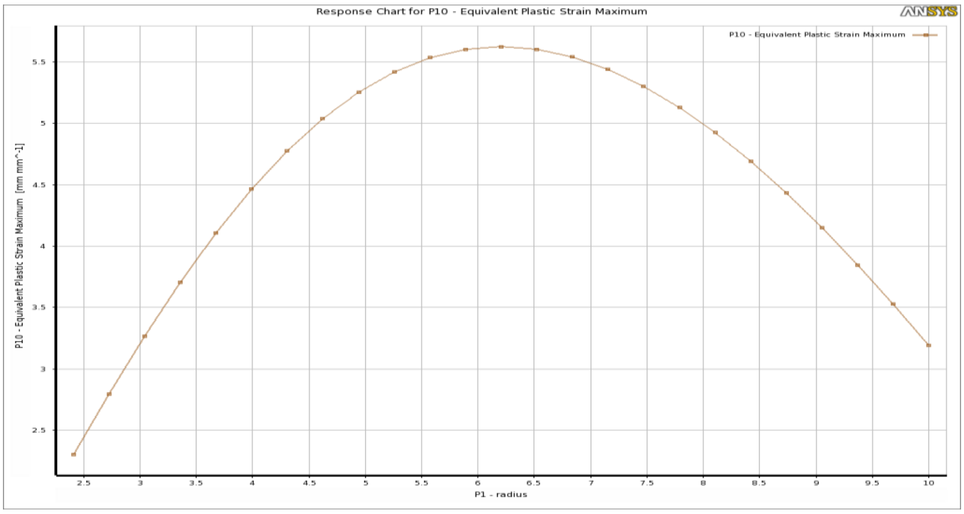

Figure 28 Radius vs Equivalent Plastic Strain

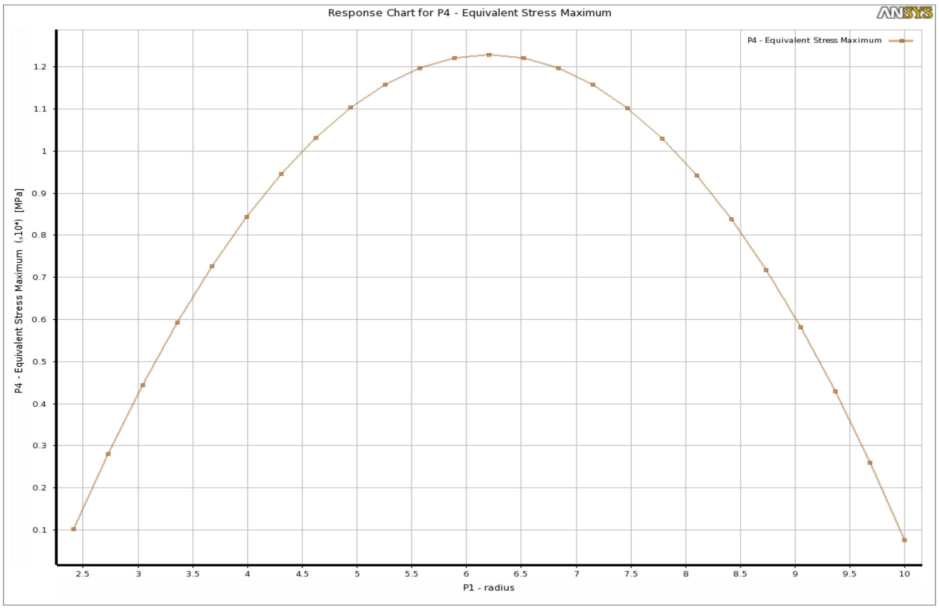

Figure 29 Thickness vs Equivalent Stress

By increasing the radius, the stress strain increases first and then starts decreasing. Which means greater thickness with greater radius will be best for the beam to exhibit less deformation.

For the proposed profile, the profile has minimum stress and strain at 81°. For 170.5° angle the profile has maximum stress and strain.

Table 11 Proposed Profile- Stress Strain Results

Sr #	Angle	Equivalent Stress (MPa)	Equivalent Plastic Strain (mm mm^-1)
1	129°	187.84	0.57714
2	81°	0	0
3	180°	69.522	1.6905
4	88.895°	125.4	2.7338
5	170.5°	187.44	1.739

4.2 Energy Absorption

Energy absorption one of the most important parameter. The beam must absorb maximum energy to protect the vital components of the vehicle. The energy absorption at different design points is:

Table 12 Energy Absorption Results

Sr #	Radius (mm)	Thickness (mm)	Energy Absorbed (J kg^-1)
1	2.410	4.45	883.967
2	6.205	8	114.038
3	2.410	8	370.643
4	10	8	177.008

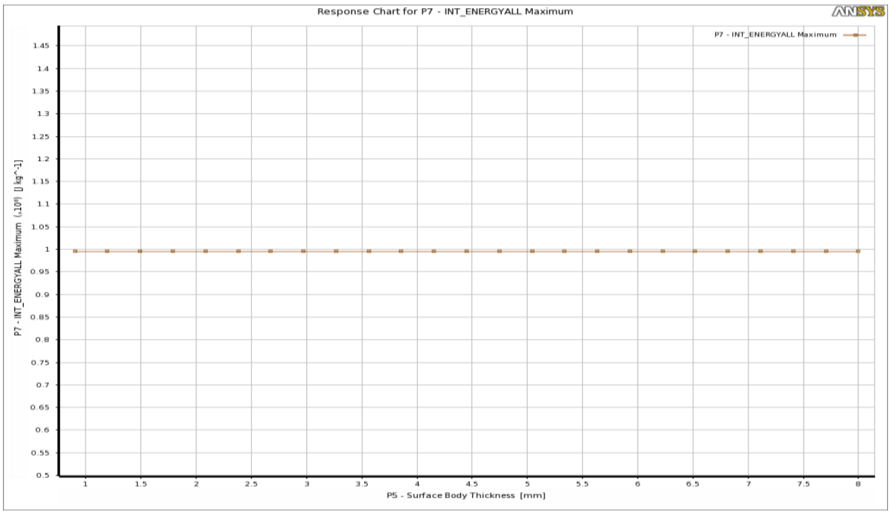

Figure 30 Thickness vs Energy Absorption

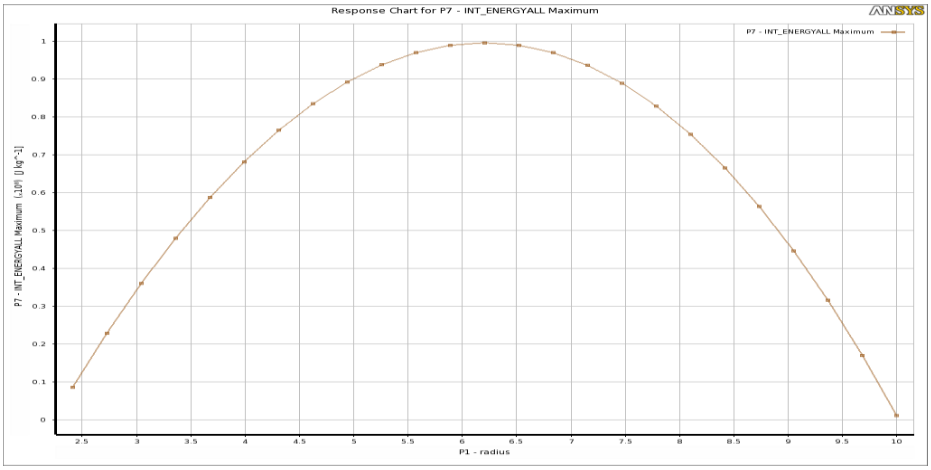

Figure 31 Radius vs Energy Absorption

The thickness has no effect on the energy absorption, but the radius shows same trend here. The maximum energy was for radius 6.2054. As it is energy absorbed for kg so by increasing the radius and thickness the mass increases but the energy to be absorbed was the same. So, for higher radius and thickness the energy absorption per kg was minimum.

For proposed profile, maximum energy is absorbed by the 180° angled profile.

Table 13 Proposed Profile- Energy Absorption Results

Sr #	Angle	Energy Absorbed (KJ kg^-1)
1	129°	35.04
2	81°	0
3	180°	48.251
4	88.895°	29115
5	170.5°	40.495

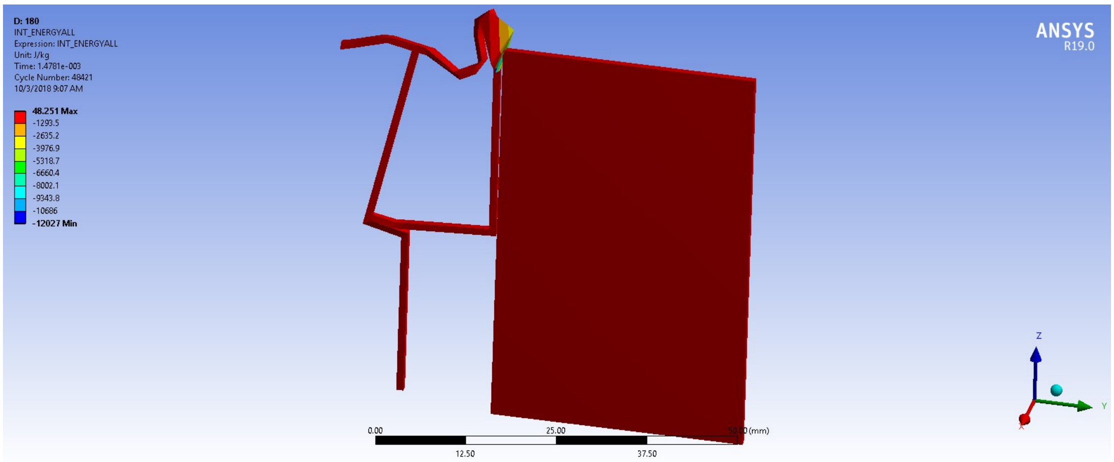

Figure 32 Energy Absorbed by Profile at 180° Angle

4.3 Weight

Environmental protection and energy conservation are becoming more prominent factors in the automotive structure design. Due to these factors, lightweight structure design of the vehicles has grown popular in the automotive industry. So, weight of the beam should be considered while designing the bumper beam.

Table 14 Weight Results

Sr #	Thickness	Radius	Weight
	mm	mm	Kg
1	4.45	2.410	1.696
2	8	6.2054	2.814

3	8	2.410	3.049
4	8	10	2.579

The design point having 4.45 mm thickness and 2.4108 mm radius have the lowest weight. The arduous task in designing the bumper beam is to design for optimal characteristics. Generally, trade-off between parameters is made. Though the design point 1 has the least weight and design point 3 with least directional and total deformation but the trends suggested that the thickness and radius should be maximum. The best trade-off suited between design parameter with acceptable; deformation, stress, energy absorption and least possible mass will be the design point 4. The properties of design point 4 are:

Table 15 Optimization Properties

Design point	**Directional Deformation**	**Total deformation**	**Stress**	**Strain**	**Energy Absorbed**	**Weight**
(thickness, radius) (mm , mm)	**mm**	**mm**	**MPa**		**J/Kg**	**Kg**
8, 10	75.249	75.25	512.32	0.232	177008.61	2.579

The proposed profile showed the best results for the 81° angle.

CHAPTER 5

Conclusions & Future Recommendations

5.1 Conclusion

Bumper beam is a crucial component to absorb energies and mitigate damages in case of an accident. In this paper, a commercial bumper beam was designed and analyzed after identifying the beam material. Efforts were made to make it realistic. The beams were analyzed for the RCAR front test standards for velocities from 10km/h. The beam of commercial vehicle fracture at 0.035 sec which means the beam failed to achieve the RCAR standards. Therefore, the beam was optimized using surface optimization feature. Radius of curve and the thickness were the input design parameters. The radius was varied up to 10 mm and the thickness of the beam was maximized to 8 mm. The beam was optimized by using 9 design point parameters and then the results of these design points were analyzed. After considering the manufacturability of the beam the optimized beam with thickness 8 mm and radius 10 mm was selected. The properties of the optimized beam were, 75.25mm Total deformation and 75.24 mm directional deformation with 177008.61 J/Kg energy absorption. The weight of the beam was 2.579 kg.

Novel profile is proposed which was molded and tested on the same standards as the commercial profile. Three-point bend test was performed on the proposed profile and then optimized for the 2D structure. The proposed profile showed the promising results.

5.2 Future Recommendations

In the course of this project, various aspects were identified where further research is needed. In this study, beam shape is not altered. It should be changed and optimized for better Crash-Worthiness. The effect of different materials with varying thickness on the Crash-Worthiness must be investigated. New novel profile designs should be considered and analyzed, and these profiles should be tested by using other standards like Federal Motor Vehicle Safety Standards (FMVSS) and Insurance Institute for Highway Safety (IIHS) standards.

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