A STUDY OF EFFECT OF ENHANCED ENERGY ABSORBER ON PERFORMANCE OF AUTOMOTIVE BUMPER USING FINITE ELEMENT ANALYSIS

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Abstract

A normal automotive bumper system is composed of two energy absorbers that are joined together by a beam of the bumper and the two energy absorbers that are important in the reduction of the impact forces during collisions. In general, the absorptions in the case of the metallic tubular absorber are dissipated by progressive folding and it is observed that most crashes which happen in the real world are at oblique angles of the axis of the absorber. In order to respond better to such conditions, the researchers have proposed buckling initiators to increase energy absorption in oblique impacts. This paper explores the behavior of the conventional absorbers (when not modified) and enhanced absorbers (when buckles initiators are employed) through the finite element analysis in LS-DYNA. The findings imply that, in the bumper assembly, the improved absorbers are not a big difference in the overall energy absorption. Rather, the bumper beam which forms the structural connection between the two absorbers, is the primary determinant of the deformation trend and energy absorption properties of the system. These results indicate that there is need to optimize the design of the bumper beam and improvements in the absorber to realize significant performance in terms of crash.
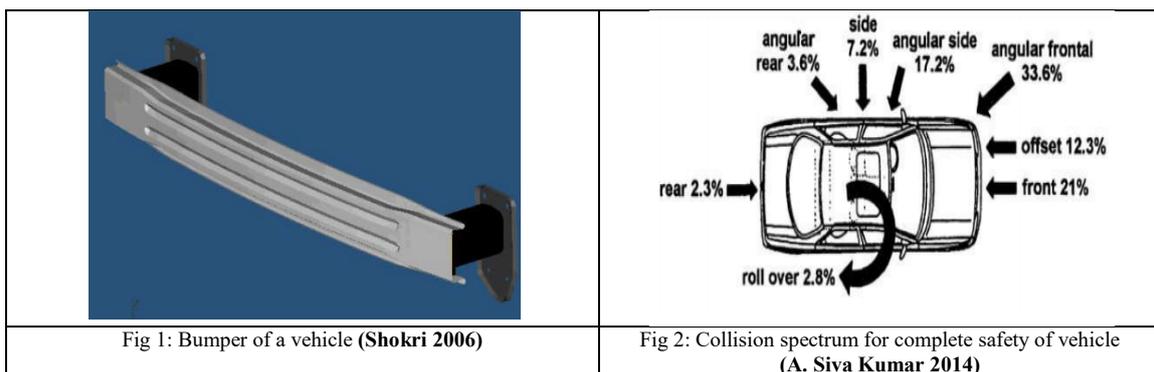
Keywords: Bumper, energy absorbers, bumper beam, oblique impact, angle of impact, buckling initiators, mean crushing load.

INTRODUCTION

Fig. 1 illustrates a vehicle bumper, which is made up of two energy absorbers attached by a beam of the bumper. Bumper system is important in the reduction of damages during collisions. In the normal crash conditions, the energy absorbers can absorb most of the energy of impact by progressive folding of their tubular structures. The main roles of a bumper of an automotive car are:

• Preventing structural and visible damage during low-speed impacts

• Reducing repair costs, particularly for medium-speed impacts around 15 km/h

• Managing load paths and maintaining structural integrity during high-speed impacts to ensure occupant protection

A lot of literature on different forms of energy absorbers during axial loading has been done. In actual car conditions, however, there is frequently oblique, not necessarily axial, impact. Furthermore, little focus has been directed on the behavior of the energy absorbers when incorporated in a full bumper assembly. This implicates the fact that the performance that is offered by individual absorbers should be considered but the interactions of the system at the system level which dictate the behavior of the crash.

Fig 1: Bumper of a vehicle **(Shokri 2006)**	Fig 2: Collision spectrum for complete safety of vehicle **(A. Siva Kumar 2014)**

In order to improve the safety of frontal vehicles, there is need to develop structures that can absorb adequate energy in a broad variety of real-world crash conditions. Even though cars must undergo regular regulatory crash tests, these tests do not always guarantee safety in any crash situations that do not pose as per the recommended approval **Wittman, (1999).** Hence, the spectrum of the collisions that a vehicle has to operate in safely has to be studied more broadly as

depicted in Fig. 2.

In practice, loading conditions are not often either axial or bending. In their place, they are usually off-axis or oblique loads. Furthermore, the energy absorption dynamics of a crash event that occurs in practice is majorly dynamic or impulsive in essence **Nagel, (2005).** These aspects explain why the research of energy absorbers and bumper systems when loaded under realistic and non- idealized conditions is important.

Motivation: Limited research has been conducted on the influence of buckling initiators in improving the energy absorption characteristics of tubes under oblique loading conditions **Chang Qi (2012).** Furthermore, only a small number of studies have examined the performance of energy absorbers within complete bumper assemblies, despite their critical role in real-world crash scenarios.

Objectives:

i) To investigate the energy absorption characteristics of the existing bumper system under various angular impact conditions.

ii) To evaluate the energy absorption performance of a bumper system equipped with enhanced energy absorbers incorporating buckling initiators, subjected to the same range of angular impacts.

Data:

This study uses the geometric dimensions of a Jeep-class LCV/SUV bumper as the baseline for modeling. All additional parameters are selected in accordance with this reference configuration.

Softwares used: Hyperworks & LS Dyna.

Parameter for energy absorption calculation: Mean crushing load: It is the average force or load over which the energy absorber deforms in a stable manner, and is obtained for a given deflection by dividing the energy absorbed by the crush distance. This should be as high as possible.

ANALYTICAL APPROACH

The analytical estimation of the mean crushing load was done before carrying out the finite element analysis process. The assumed energy absorber is a rectangular metal pipe, which has a

thickness of the wall as 2.5 mm, a cross-sectional size of 110 mm × 60 mm, and a length of 250 mm. The material features that will be analyzed are ultimate tensile strength of 452.53 MPa and a yield strength of 293.8 MPa. In a perfect energy absorber, the load-deflection response must show a long flat plateau in the response, which is indicative of a stable progressive folding and constant energy dissipation.

Quasi-Static Analysis:

The mean crushing load of quasi-static deformation on the rectangular tube is determined by taking the equation suggested by **Abramowicz and Jones (1984)**. Though this formula makes precise predictions when dealing with square tubes, it has been established to give relatively good predictions when dealing with rectangular sections too **Reid and Reddy (1986)**. Thus, the given analysis model will be utilized in order to approve the results of the finite element simulation in the current paper.

$$\frac{P_m}{P_o} = \left(\frac{c}{h}\right)^{\frac{1}{3}} \quad \text{--} \quad (1)$$

Where $c =$ side length of tube. In this case, $c = 85$ mm (average of 110 mm and 60 mm).

$h =$ thickness of rectangular tube. Hence, $h = 2.5$ mm

Here $M_o =$ fully plastic bending moment per unit length for sheet metal

$$M_o = \sigma_o \times \frac{h^2}{4} \quad \text{--} \quad (2)$$

Here σ_o is yield stress or flow stress of the tube material.

Thus,

$$\sigma_o = \sigma_{\text{yield}} = 293 \text{ MPa} \quad \text{--} \quad (3)$$

Hence, $M_o = 293.8 \times \frac{(2.5)^2}{4} = 459$ MPa

Thus $\dfrac{P_m}{P_o} = 52.22 \left(\dfrac{c}{h}\right)^{\frac{1}{3}} = 459 \times 52.22 \times \left(\dfrac{85}{2.5}\right)^{\frac{1}{3}} = 77.65$ kN

FINITE ELEMENT ANALYSIS OF ENERGY ABSORBER

1) Quasi-Static Analysis: To evaluate the performance of the rectangular tube under oblique

impact loading, a finite element analysis was performed using Belytschko–Tsay shell elements. The model consisted of 3,760 elements and 3,873 nodes, with a 3 mm corner fillet and an element size of 5 × 5 mm. A rigid plate was used to apply the load, and all translational and rotational degrees of freedom at the tube's base were constrained as shown in Fig. 3. The plate was driven downward at 10 mm/min to simulate quasi- static loading. The resulting load–deflection response is presented in Fig. 4, and the mean crushing load was determined from the area under the curve and the corresponding displacement.

2) Dynamic Analysis:

For the dynamic analysis of the rectangular thin tubes, the same meshed model shown in Fig. 3 was used, with boundary conditions identical to those in the quasi-static study. A rigid plate of 90 kg mass was assigned an initial downward velocity of 15 m/s to simulate impact loading. Strain-rate effects were incorporated through the Cowper–Symonds constitutive model to account for material strain hardening under dynamic conditions.

$$\varepsilon_p = D\left[\frac{\sigma_d}{\sigma_s} - 1\right] \text{ and } D = 6844 \ s^{-1} \text{ and } q = 3.91 \ \text{-- (4)}$$

where, constants D and q are material parameters, σ_d is dynamic flow stress at a uniaxial plastic strain rate σ_s is associated with static flow stress as is taken by **Nagel (2005).**

Furthermore, the behaviour of the tube under oblique loading is studied using the finite element method. The mass of impact was 125 kg and velocity of impact was 15 m/s.

3) Use of Buckling Initiators for Enhancing Energy Absorption Under Oblique Impact:

Buckling initiators, introduced as corner cut-outs on the energy absorber, were employed to improve its performance during oblique impacts. Three configurations were evaluated, with the first initiator positioned at 35 mm, 40 mm, and 45 mm from the top. Among these, the 35 mm configuration delivered the most effective energy absorption, and the corresponding results are summarized in Table 2.
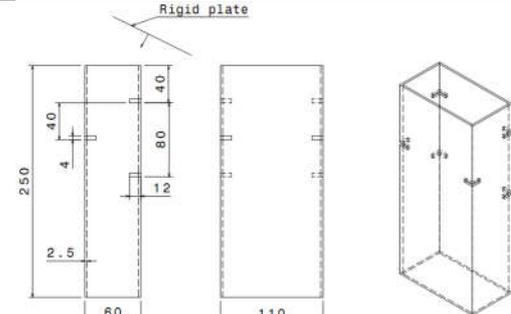
Fig 3: Meshed model of rectangular tube	Fig 4: Buckling Initiators on the energy absorber

CASE STUDY: APPLICATION OF THE IMPROVED ENERGY ABSORBER IN A BUMPER SYSTEM

The behaviour of the existing bumper system was first evaluated using finite element analysis for a 15 m/s impact at various oblique angles ranging from 0° to 30° in increments of 5°. A total impact mass of 250 kg was used to represent the two energy absorbers in the bumper assembly. Subsequently, the same analysis was repeated with the bumper equipped with enhanced energy absorbers incorporating buckling initiators, while keeping all other parameters unchanged.
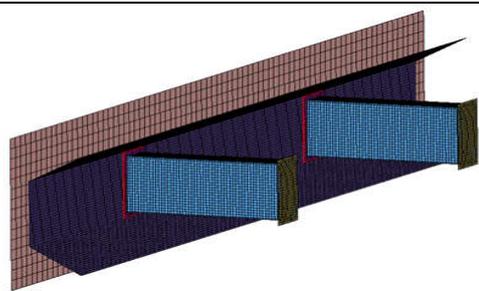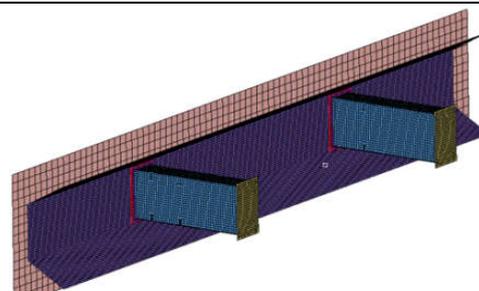
Fig. 5: Meshed model of present bumper	Fig. 6:Bumper with improved energy absorber (with absorber arrangement as mirror image of each other)

RESULTS AND DISCUSSION

1) Quasi-Static Loading:

The load–deflection response obtained under quasi-static loading is shown in Fig. 7. The mean crush load, calculated from the area under the curve, was found to be 80.58 kN. When compared with the analytically predicted value, the error was only 3.77%, confirming the validity of the finite element model.
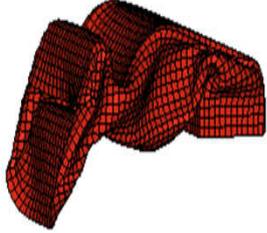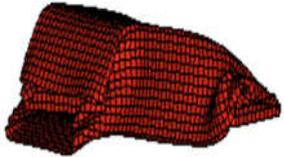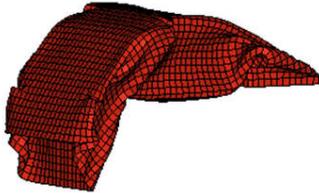
Fig 7: Load-Deflection curve for quasi-static analysis	Fig 8: Deformed shapes of tube
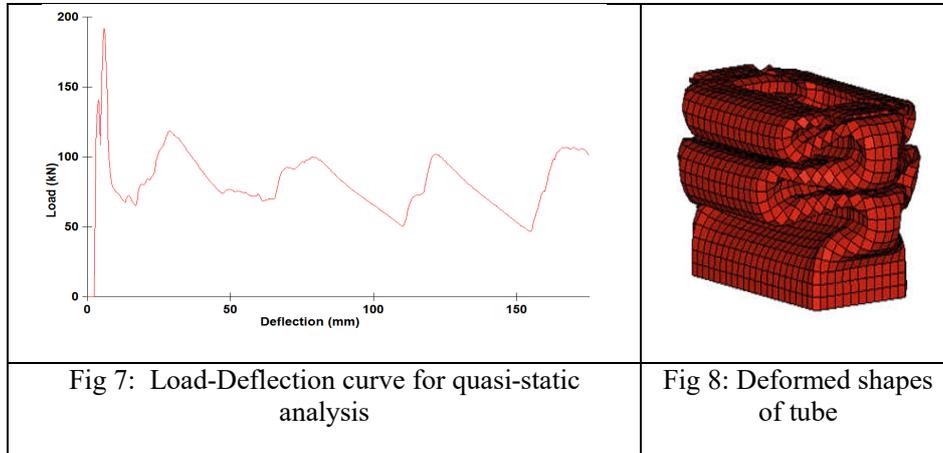
2) Dynamic analysis of energy absorber with & without bucking initiators:

Table 1 shows the comparison of mean crush load. Fig 8 shows deformed shapes of energy absorbers.

Table 1. Comparison of energy absorption by tubes with & without buckling initiators

Angle of impact (degrees)	Without initiators	With initiator at 35 mm	
	Mean load (100%)	Mean load	% Increase in mean load
0	145.57	108.17	-25.69
5	99.82	104.91	5.10
10	82.37	98.99	20.18
15	62.61	90.87	45.14
20	54.7	70.11	28.17
25	43.34	55.59	28.26
30	35.87	52.73	47.00

a) 15° Impact angle	a) 15° Impact angle
b) 30° Impact angle	b) 30° Impact angle
Fig 9: Energy absorber without buckling initiators	Fig 10: Energy absorber with buckling initiators

3) Dynamic analysis of bumpers with & without enhanced energy absorber

Mean crushing load for present bumpers:

Table 2 shows mean load for various angles of impact for present bumper. Here for the calculation of the mean load, deflection of the energy absorber of the bumper up to 2/3rd of its length is considered. Also mean load is calculated for energy absorber only as contribution of bumper beam is very less (nearly 225 Joules for all cases) and its deflection is more. The mean load goes on decreasing with increasing impact angle. But the rate of decrease is different. Initially it is higher, but afterwards it decreases. For impact at 30° angle, it is only 40 % of the mean load for axial loading.

Table 2. Comparison of present bumper and bumper with improved energy absorbers

Angle of impact in degree	Mean load (KN) of bumper with improved energy absorber	Mean load (KN) of present bumper	% improvement
0	156.15	162.64	- 3.99
5	90.54	120.97	- 25.15
10	70.81	97.28	- 27.21
15	69.39	78.90	- 12.05
20	65.33	76.90	- 15.05
25	61.05	72.99	- 16.36
30	56.40	65.38	- 13.74
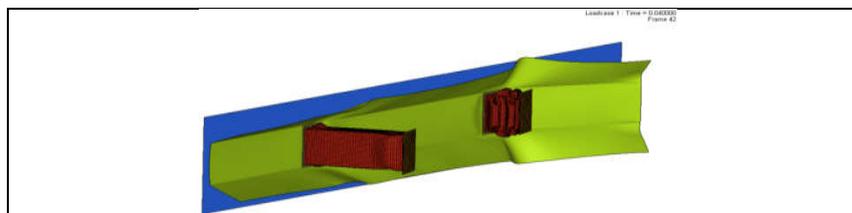
Fig 11: Deformed shape for 15º impact with normal energy absorber
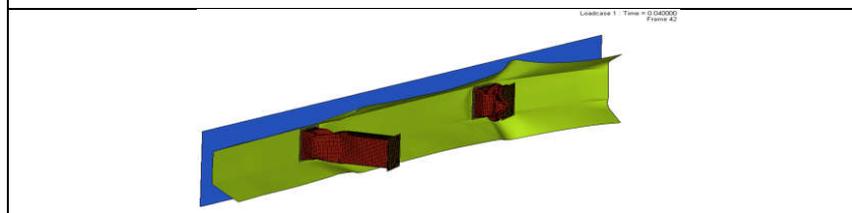
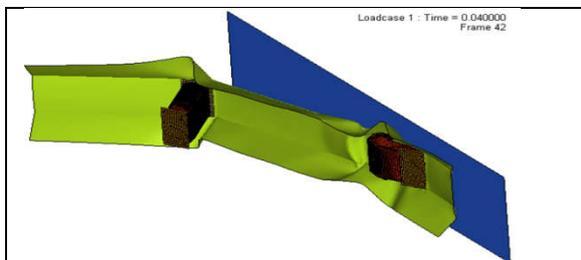Fig 12: Deformed shape for 15º impact with enhanced energy absorber

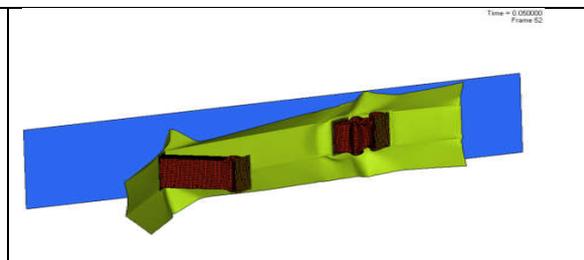| Fig 13: Deformed shape for 15º impact with normal energy absorber	Fig 14: Deformed shape for 15º impact with enhanced energy absorber

Deformed shapes:

Fig 11 to Fig 14 shows deformed shapes for 15º & 30º impact for bumpers with & without enhanced energy absorbers.
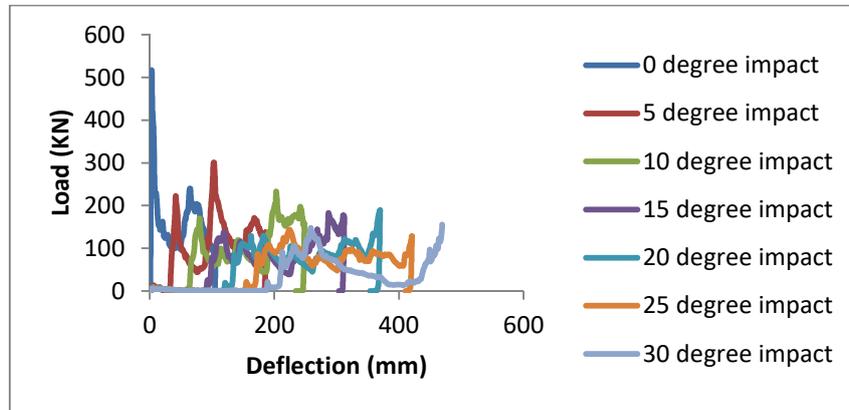

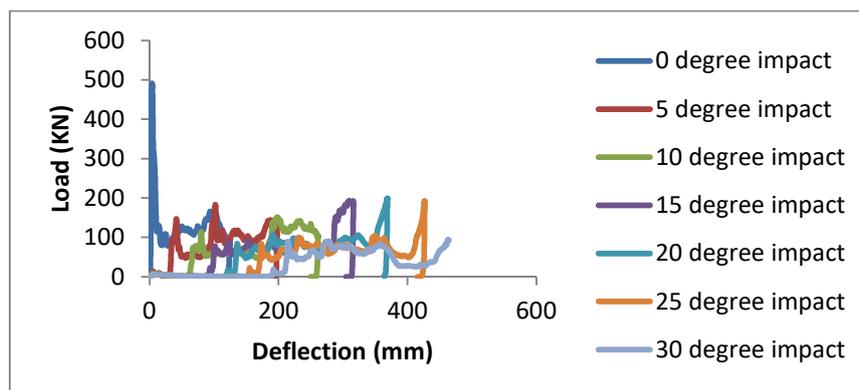Fig 15: Load deflection curves for present bumper

Fig 16: Load –displacement curves of bumper with improved energy absorbers

CONCLUSION

➢ The energy absorption capacity of rectangular tubes decreases significantly under angular impacts. At a 30° impact angle, the mean crushing load is reduced to nearly 25% of that under pure axial loading. At higher angles, a substantial portion of the tube remains unfolded, resulting in lower energy absorption.

➢ Buckling initiators can effectively enhance the energy absorption of rectangular tubes during oblique impacts. The position of the initiators plays a key role in determining the percentage improvement. However, while they improve performance under oblique loading, they may slightly reduce energy absorption during axial loading.

➢ For bumper systems, energy absorption also decreases with increasing impact angle. At 30°, the mean load is only about 40% of that obtained under axial impact conditions

➢ The bumper beam significantly influences the deformation pattern of the energy absorbers. Its presence alters the folding mechanism and limits the effectiveness of buckling initiators, leading to reduced overall energy absorption in the bumper assembly.

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