Numerical and Experimental Investigation of the Effect of Fiber Orientation on Crash Behavior of Composite Hat Shape Energy Absorber

N. Tavassoli[1,*], A. Darvizeh[2], M. Darvizeh[3], S. A. Sabet[4] and H. Ganjgahi[5]

[1] Ph.D Student, [2] Prof ., [3] Prof., Department of Mechanical Engineering, Faculty of Engineering, University of Guilan, Rasht, Iran.
[4] Assistant Professor, Iran Polymer and Petrochemical Institute.
[5] MSc.Student ,Department of Management, Mehr Alborz University.

[*] naser_tavassoli@yahoo.com

Abstract

Car body lightening and crashworthiness are two important objectives of car design. Due to their excellent performance, composite materials are extensively used in the car industries. In addition, reducing the weight of vehicle is effective in decreasing the fuel consumption. Hat shape energy absorber is used in car's doors for side impact protection.

Numerical and experimental studies are conducted in this work to understand the characteristics of energy absorption through the composite hat shape, and also to show the effect of fibers orientation angles on the amount of energy absorption. The effects of different fiber orientation on crushing behavior of hat shape structure are presented.

Keywords: *Crash, Composite materials, Side impact, Hat shape structure, Experimental test.*

1. INTRODUCTION

Due to excellent performance of composite materials, they are extensively used in the industry. Vehicle weight reduction has been dominantly considered in the automotive industry to reduce the fuel consumption. Meanwhile, design of structural components for the purpose of absorbing more kinetic energy is very important to ensure the occupant safety in the event of a crash. Therefore, the energy absorbing capacity and the weight of energy absorbing components are important objective functions to be optimized simultaneously [1].

To improve the passenger's safety during the side impact, side-door impact beams widely are used in car's doors. High static strength and high impact energy absorption are important factors during the design of the impact beams.

These properties are rarely possessed simultaneously by conventional metals because high strength metals have low toughness and vice versa. To satisfy the high strength and high toughness criteria for the design of the impact beams, conventional impact beams are made of high strength alloy steel with several heat treatments [2].

However, the heat-treated steel impact beam usually has a low nil-ductility temperature. Recently, because of the international movement of regulation of the fuel efficiency; the car body lightening has become one of the most important factors in designing a car. Therefore, reducing the weight of impact beam in passenger cars has been studied by many researchers. Fiber reinforced composite materials have high specific strength and can be mass produced by employing automated manufacturing equipment [2].

During the last decade, using fiber reinforced composite materials in various parts of passenger cars instead of conventional material attracted many researchers [1,3]. High specific strength (strength/density) and impact energy absorption characteristics of glass fiber reinforced polymer composites caused that this material to be used in load bearing structures such as bumpers and leaf springs [4–6].

Nowadays, Hat sections are used in road vehicles to absorb side impacts. Also, weight reduction has dominantly been considered in the automotive industry to reduce the fuel consumption. Hofmeyer [7] studied that 2D behavior of the cross-section. He also presented an analytical answer for the behavior of the yield line patterns [8].

In this study the hat shape structure made from fiber glass-polyester were investigated using numerical and experimental methods under quasi static and impact loads. The effects of different fiber orientation on the crushing behavior of the hat shape structure and the value of absorbed energy of this structure are investigated.

2. FABRICATION OF COMPOSITE HAT SHAPES

The hat-shape which is an idealized model of the side member of a vehicle body is depicted in figure 1 four unidirectional (UD) fiber glass-polyester layers are used in the hat-shape energy absorber. For fabrication of the prototype composite hat shapes in laboratory, the hand layup process was used. The length of the composite hat shape was 1m, which was the approximately length of an actual rear door impact beam of a compact passenger car, and the mass of beams were about 1 kg.

There are different material parameters which generally are described as follow. These parameters are namely, fiber orientation of layer 1(θ1-inner layer), fiber orientation of layer 2(θ2), fiber orientation of layer 3(θ3), fiber orientation of layer 4(θ4-outer layer). Evidently, different designs of such structures are considered by variation of material parameters θ1, θ2, θ3, θ4. The average of wall thickness was equal to t=3mm. Figure 2 shows the dimensions of cross section.

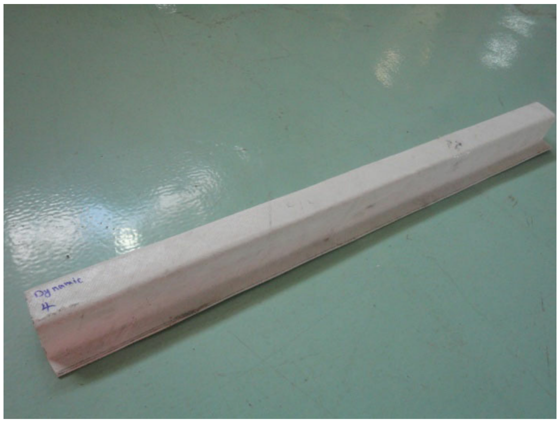

Fig. 1. Composite hat-shape

Table 1. Composite properties

Density (ρ)	$1600\frac{Kg}{m3}$
Young's modulus in the fiber direction (E_{11})	16.5GPa
modulus in the matrix-transverse direction (E_{22})	0.91 GPa
In-plane shear modulus (G_{12})	0.38 GPa
Major poisson's ratio (Nu_{12})	0.32
Tensile strength in the fiber direction (X_t)	240 MPa
Tensile strength in the transverse direction (Y_t)	6 MPa

The hat-Shape is made of UD composite layers with certain mechanical properties. The tensile properties of composite were measured according to ASTM D 3039, using a static universal testing machine.

3. MODELING OF COMPOSITE HAT SHAPES

In order to find the best combination of orientation angles in four layers with the highest amount of energy absorption in quasi static and impact loading, 366 various finite element analyses with ABAQUS software have been performed with different fiber orientation

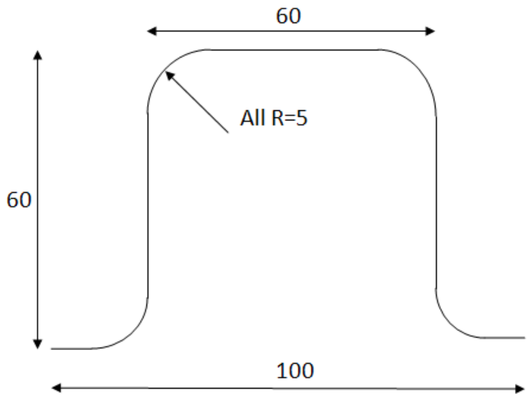

Fig. 2. Dimensions of cross section

under the same boundary conditions.

In order to simulate the crushing condition, the fully clamped condition was applied in 400mm of edges in both heads of model, and middle of specimen was continuously pushed by rigid body as shows in figure 3. The rigid body was a 305mm diameter half cylinder with 10mm in width. The velocity of rigid body in quasi static loading was 5mm/min and in impact loading it was 4m/s. The explicit analysis runs for total of 0.4s in quasi static loading and 0.05s in impact loading.

The element that was used in modeling is S4R (a 4-node doubly curved thin shell, reduced integration). The best combinations of the orientation angles in order to maximize energy absorption in quasi static and impact loading between those 366 models are presented in table 2.

4. EXPERIMENTS

The quasi static test performed according to the conditions that were presented in the Federal motor vehicle safety standard 214 (FMVSS 214) regulation [9].

Figure 4 shows the stand of quasi static bending tests. As shown in Figure 4, the fully clamped condition was applied in 400 mm of head and end of

Table 2. The optimized combinations of orientation angles in four layers

Best states	The orientation angles of layers									
	Quasi static loading					Impact loading				
	θ_1	θ_2	θ_3	θ_4	Absorbed energy (J)	θ_1	θ_2	θ_3	θ_4	Absorbed energy (J)
1	75	0	0	-75	384.8	60	45	-45	-60	189.34
2	-75	0	0	75	330.46	45	-45	-60	60	187.26
3	60	-30	30	-60	321.44	75	60	-60	-75	186.01
4	-75	-30	75	30	316.85	-30	30	45	-45	184.38
5	30	60	-30	-60	315.7	-75	-60	75	60	184

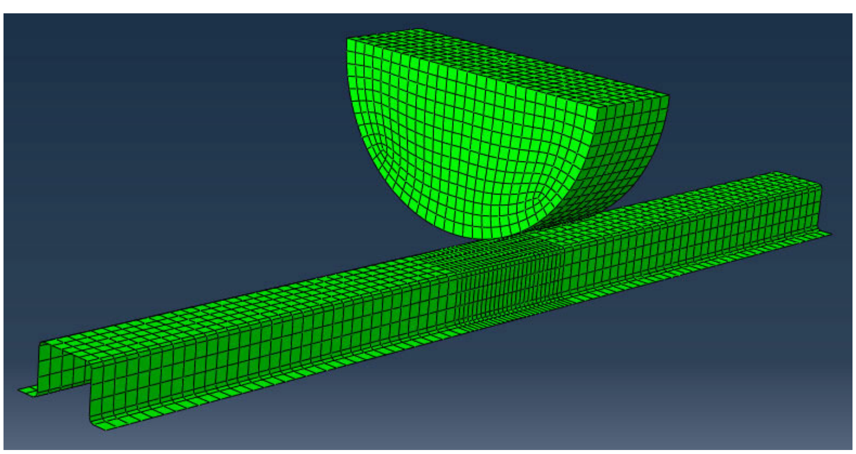

Fig. 3. The numerical model

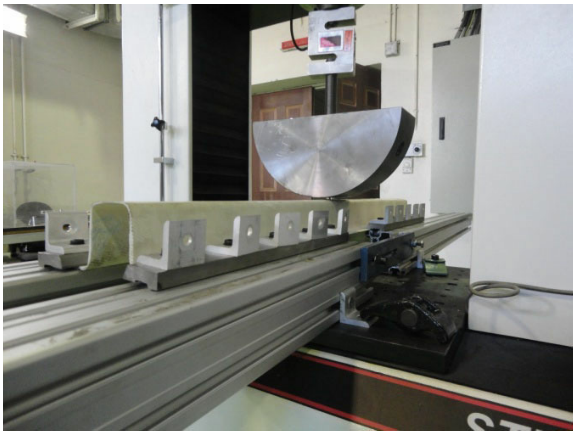

Fig. 4. Quasi static bending test machine

specimens were used to support it.The load was applied by an universal tensile machine (SANTAM STM150). The diameter of half cylinder that used for this test was 305 mm and the width of half cylinder was 10 mm. The half cylinder applied the load in the middle of the part with speed of 5 mm/s.

Figure 5 shows the pendulum that was used for impact tests. For comparing impact and quasi static test results; load was applied by a 305 mm diameter half cylinder at the midpoint of the specimens. The velocity of half cylinder in impact test was 4 m/s. This velocity is the highest value that the equipment can produce. Impact energy applied in impact test was 240J.

Fig. 5. Impact loading bending test machine

5. RESULTS

The numerical models show that in quasi static loading, the maximum energy-carrying capacity of the composite impact beams was 384.8 J and in impact loading, the maximum energy-carrying capacity of the

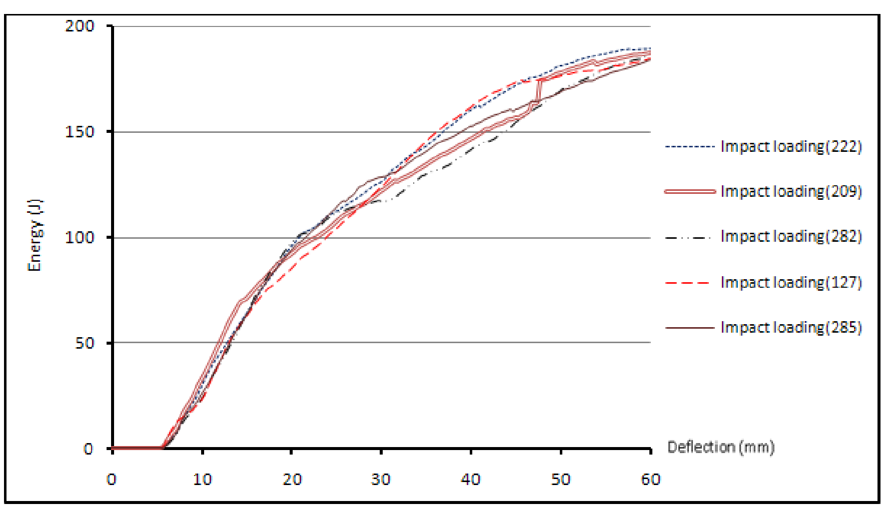

Fig. 6. Energy-Deflection curves from the impact bending models of the composite hat-shapes

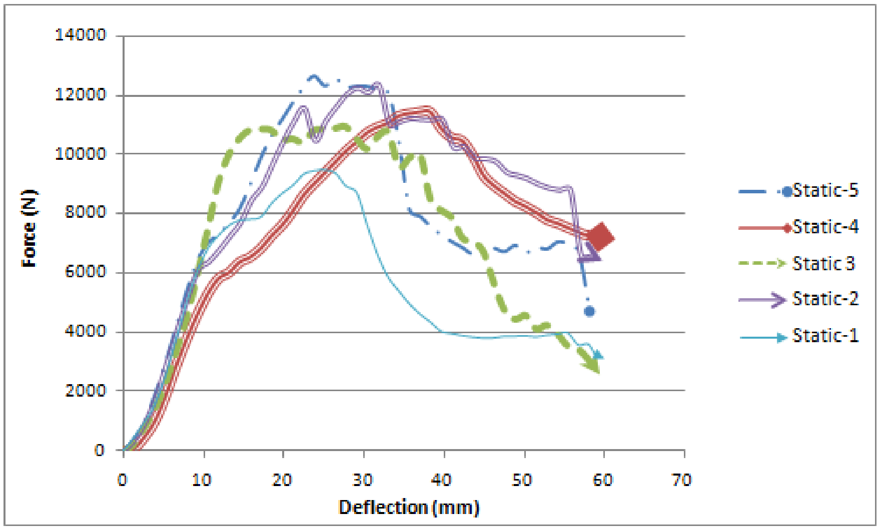

Fig. 7. Load-deflection curves from the quasi static bending tests of the composite hat-shapes

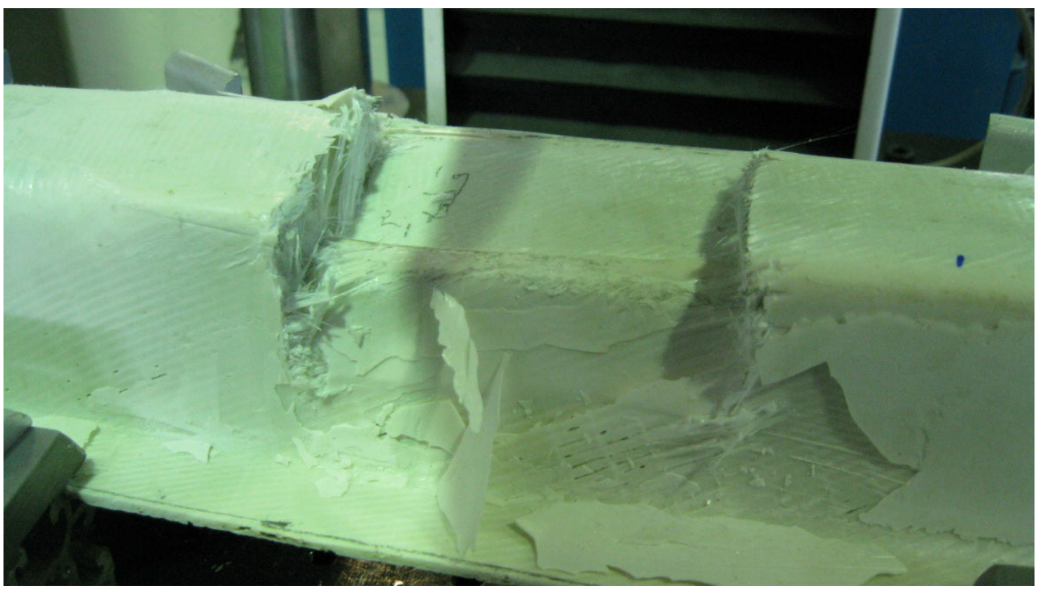

Fig. 8. Specimens after quasi static tests.

composite impact beams was 189.34 J (as shown in table 2). Figure 6 shows the energy-deflection curves from the impact models of the composite hat-shapes. The results show that all combination of The optimized combinations had the same axial stiffness but first combination of orientation angles had the highest energy absorption. Although the second combination had the best energy absorption in low deflection, but it had lower than the first one at the end of loading. Also the fifth combination had good energy absorption in low deflection, but it had the lowest energy absorption at the end of loading.

The experiments show that in quasi static loading, the maximum energy absorption was in second combination with the amount of 344.66 J. Also the energy absorption in first state was 330.3 J, in third state was 328.7 J, in fourth state was 304.67 J and in fifth state was 325.18 J.

In impact loading, the maximum energy absorption was in fourth combination with the amount of 183.18

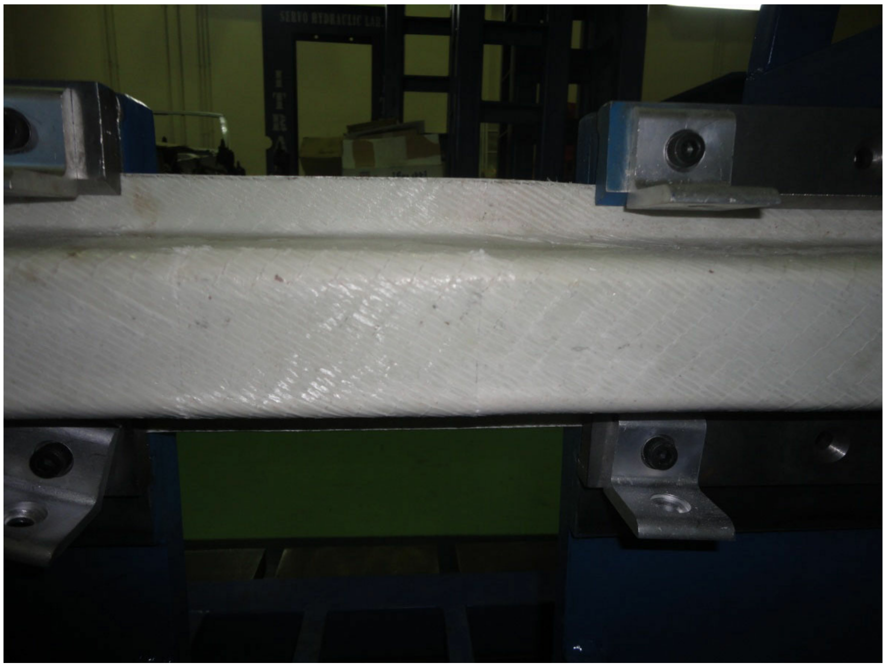

Fig. 9. Specimen after impact loading test.

J. Also the energy absorption in first state was 171.67 J, in second state was 181.05 J, in third state was 175.75 J, and in fifth state was 169.11 J.

Figure 7 shows the load-deflection curves from the quasi static bending tests of the composite hat-shapes. Figure 8 shows the fracture mode of the composite hat-shape in quasi static test and figure 9 shows it in impact loading test. It shows that in impact loading test conditions there are no big deformations in specimens.

6. CONCLUSION

In this study quasi static and impact loading behavior of fiber reinforced side-door impact hat shape beam for passenger cars were investigated and the value of energy absorption in different cases, were measured.

From the quasi static bending tests, it was found out that the maximum variance between numerical models and experimental tests is about 17%. The maximum energy absorption in experimental tests was reported in second combination with the amount of 344.66 J.

From the impact loading tests, it was found out that the maximum variance between numerical models and experimental tests is about 10%. The maximum energy absorption in experimental tests was reported in first combination with the amount of 189.34 J.

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