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Investigation of Crash Performance of Multi-Cell Crash-Boxes

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Abstract

Highway transportation is the most used type of transportation and logistics way of the 20th century. Today, the increasing need for transportation and logistics has caused a great interest in the number of vehicles in traffic. In this direction, automotive manufacturers continue their efforts to develop new designs and production methods in order to make vehicles safer, reduce the damage they cause to the environment, and increase their comfort. This article examines crash boxes within the scope of vehicle security systems. Multi-cell crash boxes, which are innovative crash box designs, are investigated for the effects of the cross-sectional area on energy dissipation performance. Within the scope of the study, four groups of crash boxes with circular, square, hexagonal, and octagonal outer sections with different inner and outer wall variations were designed. Then, the crash performances of the designed crash boxes were determined. As a result of the studies, the developed O6 model is the best crash performance as developed 23 model. The specific energy absorption(SEA) and energy absorpstion (EA)of the O6 model are 20.37 kJ/kg and 9.98 kj respectively. The results of the O6 are the best value compared with the other 23 designs developed in this research.

Keywords: Crashworthiness, Multi-cell crash box, Thin-walled structures, Energy absorption

Çok Hücreli Çarpışma Kutularının Çarpışma Performansının İncelenmesi

Öz

Karayolu taşımacılığı 20. yüzyılın en çok kullanılan ulaşım ve lojistik yoludur. Günümüzde artan ulaşım ve lojistik ihtiyacı, trafikteki araç sayısında büyük bir artışa neden olmuştur. Bu doğrultuda otomotiv üreticileri, araçları daha güvenli hale getirmek, çevreye verdikleri zararı azaltmak ve konforlarını artırmak amacıyla yeni tasarım ve üretim yöntemleri geliştirmek için çalışmalarını sürdürmektedir. Bu makalede araç güvenlik sistemleri kapsamında yer alan çarpışma kutuları incelenmiştir. Yenilikçi çarpışma kutusu tasarımları olan çok hücreli çarpışma kutuları, kesit alanının enerji yayılım performansı üzerindeki etkileri açısından incelenmiştir. Çalışma kapsamında, farklı iç ve dış duvar varyasyonları ile dairesel, kare, altıgen ve sekizgen dış kesitli dört grup çarpışma kutusu tasarlanmıştır. Daha sonra tasarlanan çarpışma kutularının

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çarpışma performansları belirlenmiştir. Yapılan çalışmalar sonunda geliştirilen 24 farklı çarpışma kutusundan en iyi çarpışma performansına sahip yeni bir çok hücreli çarpışma kutusu geliştirilmiştir. Yapılan çalışmalar sonucunda O6 modelinin 20.37 kJ/kg ile diğer 23 tasarımdan daha iyi spesifik enerji sönümleme değerine sahip olduğunu ortaya koymuştur.

Anahtar Kelimeler: Çarpışma dayanımı, Çok hücreli çarpışma kutusu, İnce cidarlı yapılar, Enerji emilimi

1. INTRODUCTION

Today, road transportation is preferred more than other types of transportation, and therefore the number of vehicles used is increasing daily. Due to the increase in the number of vehicles, traffic accidents are more common, and this causes many injuries and deaths. According to the Turkish Statistical Institute (TUIK) and the General Directorate of Security (EGM) data, there were 1.168.144 traffic accidents in Turkey in 2019, resulting in 5.473 deaths and 283.234 injuries. Due to the increase in traffic-related deaths and injuries, automotive designers give more importance and make efforts to develop new safety systems in vehicles and improve existing safety systems. These security systems can be examined under two headings active and passive security systems. Active safety systems are designed to prevent accidents before they happen. Passive safety systems aim to reduce the effects of accidents on human health and to minimize the material damage that may occur when active safety systems are insufficient to prevent accidents. In this study, crash boxes under the heading of passive safety systems will be examined. The positions of the crash boxes on the vehicle are shown in Figure 1.

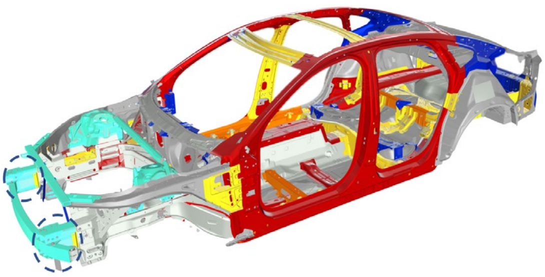

Figure 1. Positions of the energy absorbers

Recent research in the development of shock-absorbing structures is more about multicellular thin-walled tubes. Ahmad and Thambiratnam examined the effects of the geometry and loading angles of the tubes on the behavior of these structures in their study, in which they examined the energy absorption ability of conical tubes. According to their results, adding foam filling to the structures increased energy absorption by stabilizing the crushing process [1]. Mehdi et. al. applied crush tests to square and rectangular hollow and polyurethane foam-filled pipes to develop analytical formulas to be used to predict folding load and examined the interactions between model walls and polyurethane foam [2]. Meran investigated the crash resistance of thin-walled pipes by designing them in different geometric sections and examined the effects of the section geometry of these structures on the crash resistance parameters [3]. Eyvazyan et. al. examined the square-shaped pipes with vertical grooves in the axial loading condition [4]. They concluded that the vertical wave height affects the maximum load. Taghipoor et. al. examined the crash resistance properties of different crash boxes at low-speed loading [5]. Then, they continued their experiments by creating deformations on the model surfaces to reduce the models' weight and the maximum crushing force. They observed that the square-shaped tube filled with aluminum foam showed better crash resistance performance in different configurations. Alavi Nia et. al. and Vimal et. al. examined multi-cell crash boxes of different cross-sections in their study [6,7]. They concluded that hexagonal and octagonal multi-cell crash boxes had better crash resistance performance than square-shaped multi-cell crash boxes. Xu et al. investigated the factors affecting the energy absorption parameters with the crash strength analysis they conducted on hexagonal pipes with a hierarchical structure [8]. In general, it has been examined multi-cell tubes [9]. El-Hage et. al. studied the load-displacement properties of square aluminum tubes subjected to quasi-static axial crushing and the effects of the trigger mechanism on the folding pattern [10]. They preferred chamfering, drilling triangular holes, creating geometric imperfections in the model, and combining these as the triggering mechanism. The study showed that the trigger mechanism could significantly control the maximum crush force but did not cause significant effects on the average crush force. Alkhatib et. al., in their study examining the impact of corrugation on the crash resistance behavior of corrugated conical pipes, concluded that reducing the maximum crushing force during the crushing of corrugations is possible with corrugations with long wavelengths [11]. Arnold et. al. studied the crash resistance in the presence of circular discontinuities in pipes and concluded that an increase in hole diameter could reduce the maximum crushing force but can cause cracks in the pipe walls during axial crushing if the hole diameter is larger than 32 mm. [12]. Bodlani et. al., in their study examining the effect of circular perforated pipes on crash resistance under axial loading, concluded that drilling holes in the side walls of shock absorbers could reduce the maximum pick force but found that if the number of holes exceeded two, the maximum pick force did not decrease reasonably [13,14].

Marzbanrad et. al. investigated the crash performance of a circular pipe [15]. Yamashita et. al. examined hollow cylindrical pipes in different polygonal sections and showed that the increase in the number of corners in the sections increases the breaking strength, while it almost maximizes it beyond the 11 corners [16]. Yang et. al. examined the design optimization of multi-cell crash boxes and proposed three models with origami patterns to reduce the maximum pick force and increase the average pick force [17]. According to the results they obtained, it was seen that the five-cell origami crash boxes met the intended values. Bigdeli et. al. used DoE and multi-lens particle swarm optimization (MOPSO) algorithm for the optimization of multi-cell crash boxes [18].

Albak has worked on twenty-four different models and modeled the inner wall and connection walls to be shorter than the length of the outer wall in the round, square, and hexagonal cross-section multi-cell crash boxes he designed [19]. Li et. al. proposed a new design method combining multi-slot and multi-cell configurations and designed multi-cell shock absorbers with triangular, circular, and square flutes [20]. Based on the results, they determined that the three-row and three-cell model was the best model. They stated that due to the multi-objective optimization study on the best model, the specific energy absorption value increased by 2.39%, and the crush force efficiency increased by 7.69%. The study confirms that multi-groove and multi-cell energy absorber designs positively affect energy absorption parameters and can be used in the automotive industry. Ma et. al. propose a multi-cell corrugated energy absorber model inspired by natüre [21]. The model includes two parts, the first part is a skeleton structure located inside, and the second part is a corrugated tube into which the skeleton structure is placed. Kim modeled multi-cell crash boxes with square sections and trigger mechanisms at each corner [22]. It was observed that these models performed much better than flat rectangular crash boxes regarding the amount of energy dissipated and specific energy absorption. It was made optimization studies to maximize the specific energy absorption and addressed this process's gradual folding problem. As a result, it has been shown that multi-cell crash boxes are more advantageous than traditional single-cell energy absorbers. Sun et. al. aim to examine the effects of axial configurations of multi-cell crash boxes as hollow and foam-filled on the energy absorption parameters and optimize wall thickness and foam density [23]. As a result of the study, it was seen that the model with five cells and four cells at the corners filled with foam had the best values when compared to other topological configurations. Finally, optimization studies were carried out on the five-cell model with four foam-filled cells in the corners, keeping the wall thickness and foam density parameters variable. The specific energy absorption value of the new model designed as a result of the optimization study is 6.15% higher than the model designed before the optimization. Gao et al. proposed a foam-filled shock absorber model with an elliptical cross-section and compared this model with foam-filled crash boxes by applying different load directions [24]. Their analysis concluded that the foam-filled tube gave better results in different load directions. In the continuation of the study, the experimental design method was preferred to examine the radial ratio, wall thickness, and foam density values. Finally, optimization studies were carried out to maximize the specific energy absorption value and minimize the maximum crushing force under different load directions with the NSGA-II method. After the optimization studies, it was observed that the specific energy absorption value of the optimized foam-filled ellipse-section energy absorber was increased by 35%, and the maximum crushing force value decreased by 20% compared to the model before the optimization. Albak worked on multi-cell, polygonal, and vertex-coupled energy absorbers [25]. In his study, he examined the energy absorption parameters of twenty-one different crash boxes containing these three configurations under axial and angular loads.

2. ANALYSIS OF THE MULTI-CELL CRASH BOXES

Within the scope of the study, crash box models with 24 different cross-section geometries were created, and the crash strength parameters of these models were examined. The developed models were examined in four primary categories: circle, square, hexagonal, and octagonal cross-sectional, according to the outer wall section geometries. The cross-sectional configurations of models are shown in Figure 2.

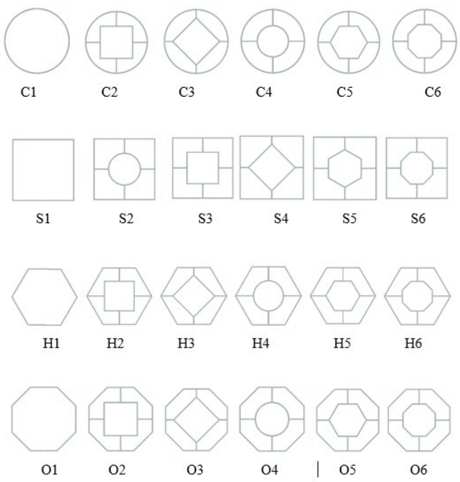

Figure 2. Cross-sectional configurations of models

Various parameters are mentioned in the literature to examine the performance of energy absorbers. In this study, the performance of crash boxes was investigated by examining parameters such as energy absorption (EA), specific energy absorption (SEA), peak crushing force (PCF), average crushing force (MCF), and crushing force efficiency (CFE).

Specific energy absorption can be calculated from Equation 1 [26].

$$SEA = \frac{EA}{m} \tag{1}$$

Where EA represents the energy absorption and m represents the total mass of the energy absorber.

The ratio of the absorbed energy to the total crushing length in the axial direction is expressed as the average crushing force. The mean crushing force is obtained using Equation 2 [27].

$$MCF = \frac{EA}{d} \tag{2}$$

Here, given by EA is the energy absorption, and d is the total length of the energy absorber.

Peak crushing force (PCF) is the name given to the highest value force that occurs due to the axial loads on the crash absorber at the time of the crash. It is calculated by Equation 3 [28].

$$PCF = max[F(x)] \tag{3}$$

It is a value that automotive designers want to keep at low values, as the peak crushing force can cause the crash absorber to transmit the axial force directly to the other components of the vehicle without functioning.

The crush force efficiency (CFE) is expressed as the ratio of the average crush force to the maximum crush force and is calculated by Equation 4.

$$\boldsymbol{CFE} = \frac{\boldsymbol{MCF}}{\boldsymbol{PCF}} \tag{4}$$

A high value of crush force efficiency in an energy absorber design does not indicate that the amount of absorbed energy will be high under all conditions. The desired feature of crash boxes is to have a high average crushing force and a low maximum crushing force value.

The material of the crash boxes was determined as Al 6063-T5 by using the studies on this subject in the literature. Wu et al. (2016) tested the Al 6063-T5 specimen in MTS 322 material testing device according to ASTM E8M-04 standard in their study and reached the material's stress-strain curve. The stress-strain curve is given in Figure 3. The poison ratio, density, elastic modulus, initial yield stress, and final stress values of the material are given in Table 1.

Table 1. The material properties of Al 6063-T5 [34]

Material	AA6063-T5
Density [g/cm^3]	2.70
Elastic Modulus [GPa]	68.2
Poisson's Ratio	0.3
Initial Yield Stress [MPa]	180
Ultimate Stress [MPa]	206

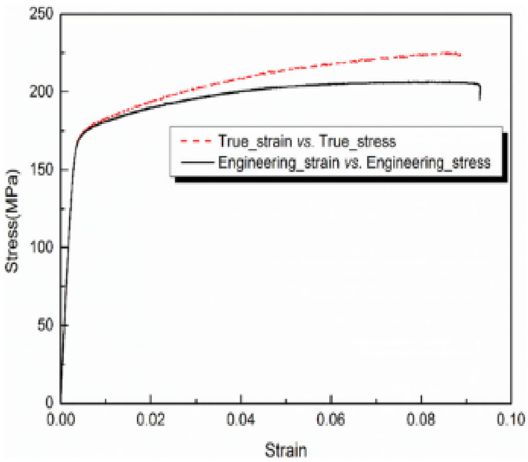

Figure 3. The stress-strain graph of Al 6063-T5 [29]

2.1. Analysis Method

In the study, non-linear explicit analyzes of crash boxes were made. The formulation type of the models was defined as 'QEPH Shell Formulation' in the software library. In the finite element structure, quad elements are used.

The integration number was determined as '5' along the thickness and was modeled to 1.5 mm thickness by using 2mmx2mm surface elements. The length of the crash boxes used in this paper is 200 mm, and the diameter is 98.5 mm. 24 crash boxes developed in this paper have the same length and diameter as given in Figure 4. The speed of the rigid wall is defined as 10 m/s, and its mass is defined as 263 kg, as shown in Figure 4. The "Type7" contact card has been used for contact situations of crash boxes and rigid walls. The research studies in the literature were considered to determine the speed and weight of the rigid wall [8,28].

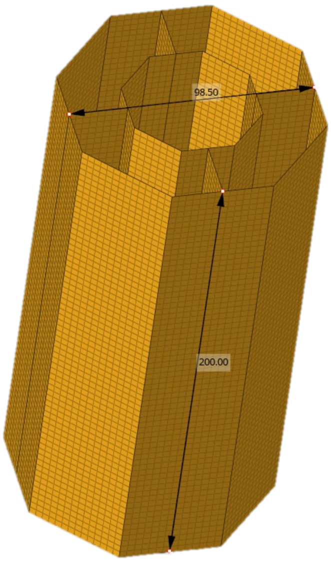

Figure 4. Dimension of the O6 crash boxes

In order to prove the accuracy of the analysis, we considered Wu et. al. [29]. They developed the crash boxes and performed physical crash tests and compared the results, and confirmed the analysis studies. Based on the test study performed in this study, the physical test was modeled and solved in the analysis software.

Wu et. al. [29] determined the amount of energy absorbed as 9.55 kJ in their second physical test study with the C5 design. Analysis results in this study were compared, and it was observed that the results were close to each other. The amount of absorbed energy obtained in the verification study carried out within the scope of this study is 9.75 kJ. The comparison of the verification

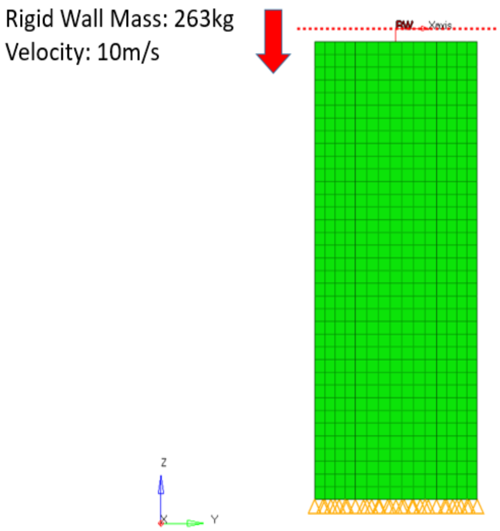

Figure 5. Symbolic representation of the FEA model

Table 2. Comparison of FE analysis and psychical test results

	MCF [kN]	PCF [kN]	CFE	SEA [kJ/kg]	EA [kJ]
C5 Model Psychical Test 2	79.64	181.99	0.44	22.98	9.55
Validation Analysis	81.25	137.52	0.59	25	9.75
Failure Rate [%]	-2.02	24.41	-34.97	-8.79	-2.09

Work performed with the physical test and the energy dissipation parameters values are shown in Table 2.

As seen in Table 2, there are differences in the PCF parameter between the analysis and physical test results. The reason for this difference is that the initiators modeled to keep the starting point of the buckling under control in the analysis model. In Figure 6, the crash behavior of the energy absorber in the finite element analysis results and the crash behaviour in the test results in the literature study are shown. The analysis results and the test results confirm each other.

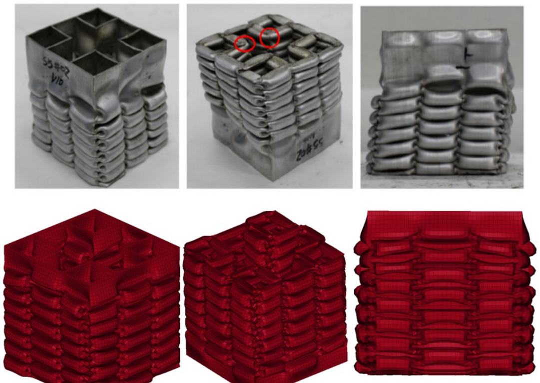

Figure 6. The comparison of FE analysis and test results

The analysis results of the models with circular, square, hexagonal, and octagonal sections on the outer wall are shown in Table 3. it is concluded that all multi-cell crash box models with an inner wall and connecting walls outperform single-cell models. When comparing the analysis results, the best models of all groups were compared. When the graph shown in Figure 7 is examined, the maximum crushing force values of the S6 and H6 models are at higher levels than other models. When the performance parameters in Table 3 and the crash force-displacement graph in Figure 7 are examined, it is seen that the model with the highest amount of absorbed energy is the O6 model. The absorbed energy amounts of other models are also exceptionally high. However, the crush force efficiencies are lower than the O6 model due to the high crushing force level.

At the same time, the mass of the S6 model is more than other models, which is a significant disadvantage for this model. It can be said that the O6 model is the most efficient. The crash behavior of this model is shown in Figure 8. It is seen that the multi-cell energy absorbers are more efficient than the classical single-cell energy absorber models, regardless of the cross-section geometry. In all four groups investigated throughout this research, the models with an octagonal inner wall were selected as the best models. Based on this, it can be concluded that crash boxes with octagonal inner wall geometry are more efficient than crash boxes with circular, square, and hexagonal inner walls.

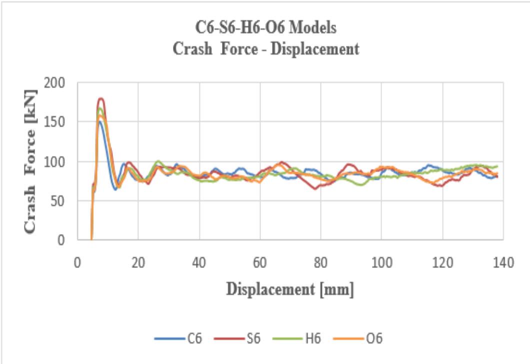

Figure 7. Crash force - displacement of C6-S6-H6-O6 models

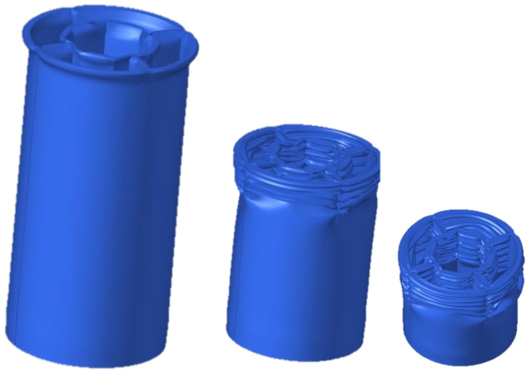

Figure 8. The crash behavior of the O6 model

Table 3. Performance parameters of energy absorbers

	Model Name	Mass [kg]	MCF [kN]	PCF [kN]	CFE	% CFE Gain	SEA [kJ/kg]	% SEA Gain	EA [kJ]
Circular Models	C1	0.25	28.44	77.4	0.37	-15.02	15.36	-12.17	3.84
	C2	0.49	69.19	160.6	0.43	-1.24	19.06	8.98	9.34
	C3	0.46	70.33	150.04	0.47	7.94	18.35	4.92	8.44
	C4	0.55	75.92	147.33	0.52	19.43	16.56	-5.31	9.11
	C5	0.46	76.75	150.87	0.51	17.13	20.02	14.47	9.21
	C6	0.47	79.08	150.47	0.53	21.72	20.19	15.44	9.49
Square Models	S1	0.39	23.63	107.09	0.22	-49.47	8.18	-53.23	3.19
	S2	0.53	78.17	176.13	0.44	1.05	17.7	1.21	9.38
	S3	0.56	78.75	189.07	0.42	-3.54	16.88	-3.48	9.45
	S4	0.52	70.92	176	0.4	-8.13	16.37	-6.40	8.51
	S5	0.53	78.83	178.74	0.44	1.05	17.85	2.06	9.46
	S6	0.52	78.75	178.64	0.44	1.05	18.17	3.89	9.45
Hexagon Models	H1	0.28	28	91.84	0.3	-31.10	13.5	-22.81	3.78
	H2	0.53	69.04	177.33	0.39	-10.43	17.58	0.52	9.32
	H3	0.49	70.42	165.34	0.43	-1.24	17.24	-1.42	8.45
	H4	0.52	71.58	176	0.41	-5.84	17.62	0.75	8.59
	H5	0.5	75.75	167.63	0.45	3.35	18.18	3.95	9.09
	H6	0.5	79	167.49	0.47	7.94	18.9600	8.41	9.48
Octogon Models	O1	0.26	28	86.66	0.32	-26.51	14.54	-16.86	3.78
	O2	0.51	80	168.19	0.48	10.24	18.82	7.61	9.6
	O3	0.47	73.42	175.11	0.47	7.94	18.74	7.15	8.81
	O4	0.47	79.17	156.23	0.51	17.13	20.21	15.56	9.5
	O5	0.49	78.75	158.32	0.5	14.83	19.29	10.30	9.45
	O6	0.49	83.17	158.32	0.53	21.72	20.37	16.47	9.98

3. CONCLUSION

In this paper, the crash performance of 24 different multi-cell crash boxes is investigated. For this aim, firstly, the C5 model taken from the literature is investigated. After correcting the C5 model with experimental results, we have developed 23 different multi-cell crash box designs, which have different cross-sections. We investigated the crash performance of the mentioned models considering mass, the amount of energy absorption, and specific energy absorption values. The results given in Table 3 show that the O6 model has the best performance with 20.37 kj/kg specific energy absorption and 9.98 kj energy absorption. We concluded that multi-cell crash boxes are better than single-cell crash boxes, and the cross-section of the multi-cell crash boxes is important to improve the performance of the crash boxes. As future work, optimization of the multi-cell crash boxes can be made.

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