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Original Research Article

Weight reduction of the backhoe arm of a backhoe loader

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ABSTRACT

Today, backhoe loader machines are widely used in different areas such as the construction industry, mining industry, arrangement of agricultural areas, maintenance and repair activities in settlements. One of the most important reasons why the usage areas of these machines are so wide is that they have a design that includes both excavation and loading capabilities. With different attachments attached to both the backhoe and loader mechanisms, backhoe loader machines can have a wide variety of operating capabilities. In this study, weight reduction of the backhoe arm of a backhoe loader is carried out. The results obtained from the analyzes made on the existing design gave direction to the weight reduction process. While creating alternative models within the scope of the weight reduction process, the structures of the connection points of the mechanisms, their positions and the distances between these connection points were kept constant, allowing all alternative designs to be mounted on the machine without the need to make any changes on the existing machine. With this weight reduction study, it is aimed to reduce the weight and fuel consumption of the machine as well as to increase the operating efficiency of the machine. Structural analyzes were made on the optimum structure obtained, and it was examined whether the new model was suitable for working conditions in terms of strength. The current design and the optimized design are compared in terms of strength and operating performance.

Keywords: Backhoe arm, weight reduction, backhoe loader machines, FEM analysis, thickness reduction.

1. Introduction

Backhoe loader is one of the most used types of construction machines all over the world. Since they have both excavation and loading capabilities, these machines can be used in various fields such as construction, mining, agricultural areas and urban planning. As the name suggests, backhoe loaders are machines that have both digging and loading capabilities. Thanks to these capabilities, backhoe loader machines are preferred in many different areas such as construction, landscaping, mining, urban regulations and agricultural activities. The basic structures that make up the backhoe loader work machines are the chassis, cabin, chiller, power transmission elements, the loader mechanisms at the front and the backhoe mechanisms at the back of the machine. The mobility of these mechanisms is provided by the hydraulic systems on the machine and driven by the engine. The basic structural elements of backhoe loader machines are shown in Figure 1.1.

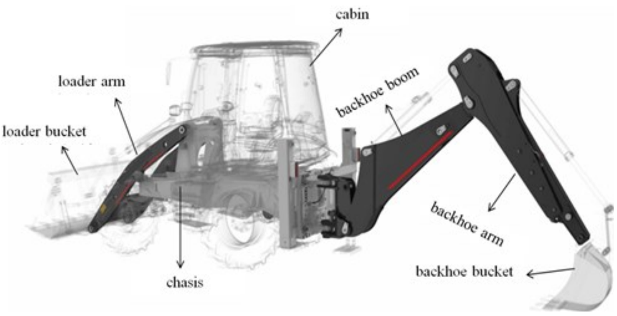

Figure 1.1. Structural elements of backhoe loader machines [1]

With the excavator group located at the back of the machine, operations such as digging pits, trenching, carrying and lifting various loads, breaking hard floors and materials can be performed. The excavator group generally consists of the kingpost, the excavator boom, the excavator arm, the bucket-arm linkage mechanism and the bucket. When necessary, various attachments such as ripper, polyp and breaker can be installed instead of the bucket, and different operations can be performed with the excavator group. The hydraulic pipes, hoses and cylinders on the excavator group provide power and mobility to these attachments, backhoe boom, backhoe arm and bucket. The hydraulic cylinders in the excavator group are shown in Figure 1.2.

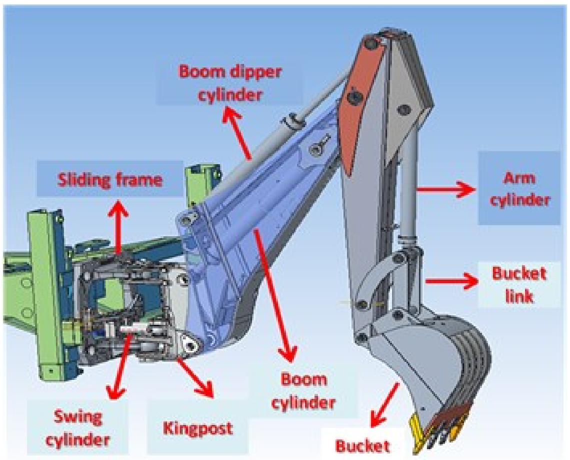

Figure 1.2. Backhoe hydraulic cylinders

The swing cylinders located between the kingpost and the sliding frame provide the right and left rotation of the excavator group. While the boom cylinder allows the boom to move up and down, the boom dipper cylinder also allows the excavator arm to move up and down. The bucket cylinder on the backhoe arm provides the movement of the bucket. The force that can be obtained from the bucket cylinder and the boom digger cylinder is one of the most important determining factors in the digger performance. Figure 1.3 shows the movements of the excavator group [1].

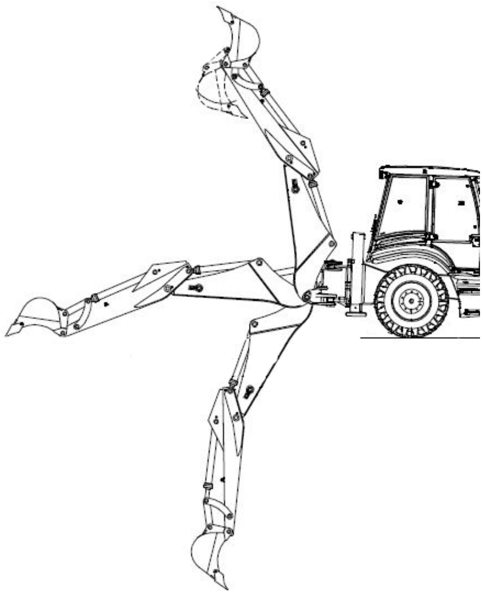

Figure 1.3. Backhoe boom and bucket movements [1]

At the beginning of the design process of a commercially produced backhoe loader machine, system requirements are determined by taking into account the demands and needs from the sector and potential customers. These requirements are used as design parameters in the conceptual design phase and the design process proceeds in this direction. Examples of these parameters are digging depth, backhoe bucket and loader bucket capacities, and backhoe bucket breakout force values. The design should also have a durable structure that can withstand working loads. For this reason, the design and analysis process are largely intertwined. Many studies in the literature also exemplify this situation.

In a study by Khan et al. in 2016, designs with different weights and features were obtained by changing the materials and part thicknesses used to reduce the weight of an existing excavator arm. First of all, the weight of the model was increased by increasing the thickness of the parts used in the current design. Afterwards, the part thicknesses used in the current design were reduced in two stages and lighter designs were obtained. In addition to these, the material used in the building was examined in two different ways as steel alloy and aluminum alloy, and a weight comparison was made. Considering the calculated working loads in these designs, analyzes were made using the finite element method. By comparing the critical von Mises values obtained in the analyzes and the strength values of the materials used, it was checked whether the designs are suitable for safe operation. The analyzes show that a safe structure can be obtained if the model with reduced weight is produced with steel alloy material in the first stage [2].

In a study conducted by Solazzi in 2010, it was aimed to reduce the weight of the excavator arm and boom by using aluminum alloy instead of steel alloy in the excavator arm and boom and by making design changes. Thus, it was stated that the capacity of the excavator bucket and the working efficiency of the machine could be increased. The loads on the mechanism components were investigated for different load conditions. These loads were used as parameters of the analysis modeling to be done on the new arm design. In the analyzes, safety coefficients were also taken into account in order to check whether the mechanism is suitable for safe working conditions. The results of the analyzes showed that the model using aluminum alloy is suitable for safe operation. [3] Weight comparisons before and after the optimization process are shown in the Table 1.1.

Table 1.1. Weight comparisons before and after the optimization process [3]

Part	Arm components	Pins	Bucket
Original model weight (kg)	2050	335	2150
Optimized model weight (kg)	1080	195	2785
Weight change (%)	-47.3	-41.8	29.5

When the new design is compared with the existing design, it is stated that the total weight of the excavator arm decreases by 50%, while the total cost increases in the range of 2500-3500 €. In addition, it was emphasized that working efficiency increased by 35% [3].

In their 2012 study, Qureshi and Sagar aimed to help both the development of the boom structure and the understanding of the finite element analysis method with their definitions and research by performing strength analysis on the boom of an existing backhoe loader machine with the finite element method. The finite element method is simply based on first separating a structure into different elements and then joining them with nodes. After this process, the simultaneous algebraic equations solved with mathematical formulas are established on the model and the stress and deformation values are obtained. In this study of Quershi and Sagar, a three-dimensional solid model of the existing excavating boom was first created in the computer environment, and then the necessary parameters were introduced to the program. Then, the tetrahedral mesh structure, which is frequently preferred in complex three-dimensional structures, is used to carry out finite element analysis of this structure. The results obtained from the analysis process carried out within the scope of the study showed that the existing structure is safe in terms of strength [4].

In another study, Kachave et al. analyzed a ladder frame truck chassis for size optimization. [5] The chassis used in this study consists of beams with type C cross-sectional area. Different chassis alternatives were produced by making partial thickness changes in the C profile and these alternatives were analyzed using CATIA V5R117 and ANSYS programs under the same load conditions. These chassis alternatives are shown in Figure 1.4.

In the study, the maximum stress, bending, bending strength and the ratio of resistance to weight seen in these different alternatives were compared. This comparison is shown in Table 1.2.

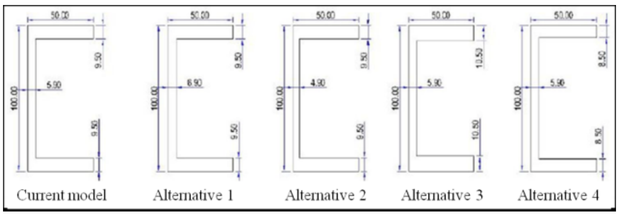

Figure 1.4. Different chassis alternatives with different dimensions [5]

These results showed the effects of thickness on weight and bending resistance and guided the optimum dimensions in design [5].

2. Materials and Methods

2.1. Backhoe arm weight reduction process

Achieving the purpose of the weight reduction

Table 1.2. Comparison of different chassis alternatives [5]

	Bending (mm)		Stress (N/mm²)	Weight (kg)	Bending strength (N/mm²)		Bending/Weight ratio	
	Analytical	Numerical			Analytical	Numerical	Analytical	Numerical
Current model	0.853	10.515	111.04	544.17	13716.30	11126.96	25.21	20.45
1. alternative	0.846	10.095	105.16	550.96	13829.79	11589.90	25.10	21.04
2. alternative	0.859	1.201	124.02	537.38	13620.49	9741.88	25.35	18.13
3. alternative	0.831	0.97404	100.31	557.56	14079.42	12011.83	25.25	21.54
4. alternative	0.877	1.0468	108.36	533.35	13340.94	11176.92	25.01	20.96

process is possible by achieving sufficient strength values of the optimized design as in the old design. In order to compare the existing and new structures in terms of strength, the critical strength values of the existing structure should be known. For this purpose, first of all, strength analysis was performed on the existing design. In this analysis, the backhoe breakout forces were taken into account while determining the force applied to the backhoe arm, because it is accepted that the most critical loadings on the backhoe group under normal operating conditions are carried out under the conditions in which the shearing forces are determined, and these conditions are also taken into account during the design phase. The ISO 6015 standard specifies standardized methods for calculating, testing and reporting the backhoe performance of excavator and backhoe loader machines. When calculating the breakout force according to the ISO 6015 standard, the backhoe group should be in the position indicated in the Figure 2.1 [6].

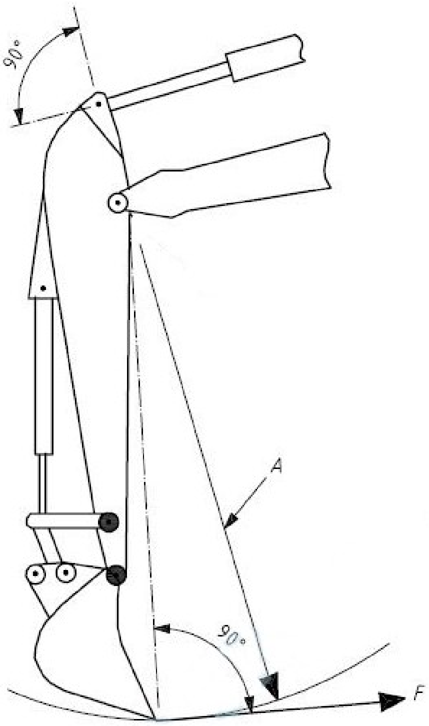

Figure 2.1. Backhoe breakout force position [6]

In Figure 2.1, the expression A denotes the radius line of the arc that the digging arm will draw during digging, and F denotes the shear force. In the current design, the breaking force is calculated according to the dimensions indicated in the Figure 2.2.

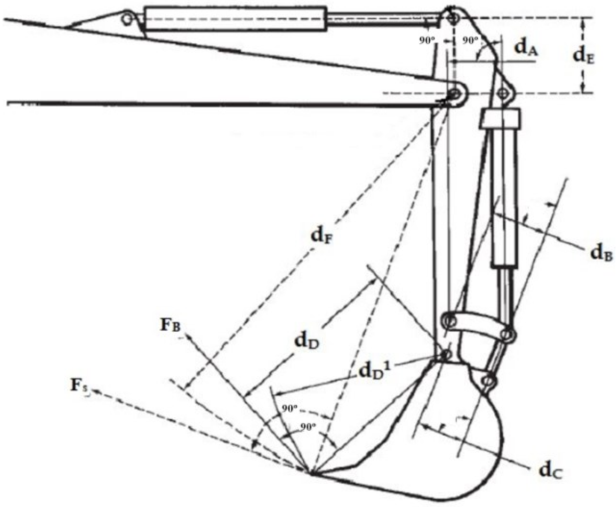

Figure 2.2. Backhoe group dimensions [7]

In Figure 2.2, Fb is the bucket breaking force, Fs is the arm breaking force, and dA, dB, dC, dD, dE and dF are the lengths indicated on the figure. When the values are placed in Equation 2.1, the bucket breakout force (FB) is obtained [7].

$$F_B = \frac{p\times(\pi/4)D_B^2}{d_D}\left(\frac{d_A\times d_C}{d_B}\right) \quad (2.1.)$$

In Equation 2.1, the value of p represents the system working pressure in MPa, and DB the diameter of the bucket cylinder in mm. The arm breaking force is calculated as shown in Equation 2.2 [7].

$$F_S = \frac{p\times(\pi/4)D_A^2\times d_E}{d_F} \quad (2.2.)$$

In equation 2.2, DA refers to the piston diameter of the lever cylinder in mm. Considering the calculation figures, it is understood that the arm breakout force obtained by moving the backhoe arm of the backhoe arm cylinder on the boom is more critical in terms of the strength of the backhoe arm. For this reason, arm breakout force was applied to the structure in the strength analysis made on the backhoe arm. The mechanical properties of the materials used in the building also have a decisive role in the strength of the structure. While S335J2 structural steel is used in the body of the said backhoe arm, C45 high carbon steel is used in the bushings in the cylinder and mechanism connections. The mechanical properties of these materials vary according to their thickness. Considering the material thicknesses used in the current design, the mechanical properties of these materials according to EN 10025 and EN 10277 standards are given in the Table 2.1 [8, 9].

Table 2.1. Mechanical properties of the materials used

Material	Yield strength (MPa)	Tensile strength (MPa)
S355J2	355	510-680
C45	310	565

Considering the loading scenarios that the backhoe arm will encounter in real working conditions, the arm breakout force was applied in two different ways, symmetrical and asymmetrical from the bucket tip as shown in Figure 2.3.

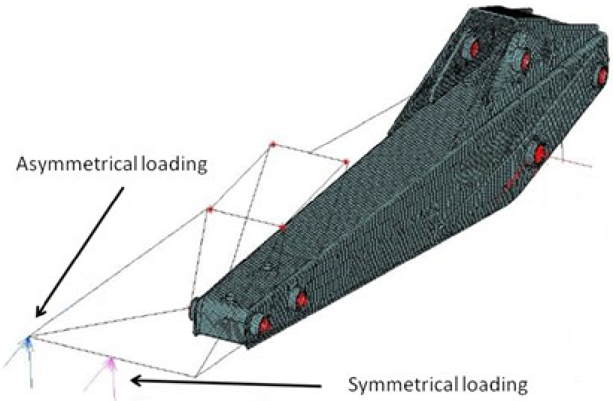

Figure 2.3. Asymmetrical and symmetrical loading

Necessary adjustments and meshing on the existing solid model were made in the MSC Apex program. The definition of boundary conditions and material identification were performed with the MSC Mentat interface, and the analysis was solved with the MSC Marc program. The backhoe bucket, bucket connection links, arm cylinder and bucket cylinder are modeled as line elements to reduce the complexity of the analysis model. The backhoe arm body is fixed from the bucket, bucket cylinder and boom connection areas. During the analysis process, the Equivalent Von Mises stresses and the maximum principal stresses formed in the existing backhoe arm model were examined separately in both symmetrical loading and eccentric loading scenarios. As can be seen in Figure 2.4 and Figure 2.5, in both stress approaches, maximum stresses occur in the areas where the boom connection is provided on the backhoe arm and the end regions of the reinforcement plates on both sides of the backhoe arm body.

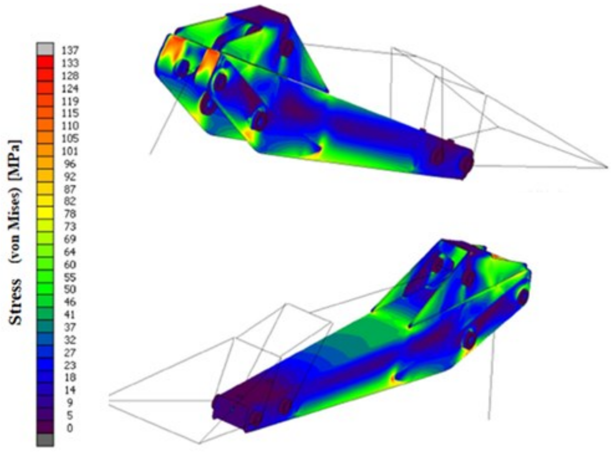

Figure 2.4. Von Mises stresses in symmetrical loading scenario

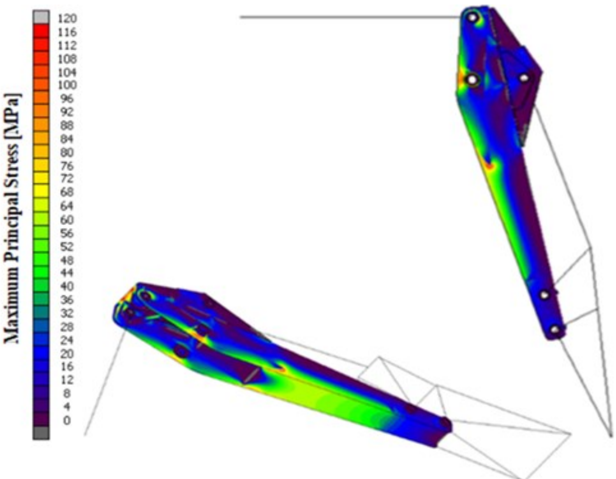

Figure 2.5. Maximum principal stresses in symmetrical loading scenario

While working with the backhoe group of the machine, the possibility of the backhoe arm to be subjected to eccentric loading is also quite high. The scenario implied by the concept of off-loading is that the load is applied asymmetrically from the outer corner of the backhoe bucket. This scenario was also considered during the analysis, as this type of loading would cause higher stress values on the backhoe arm. Figure 2.6 and Figure 2.7 show the stresses on the backhoe arm under eccentric loading conditions.

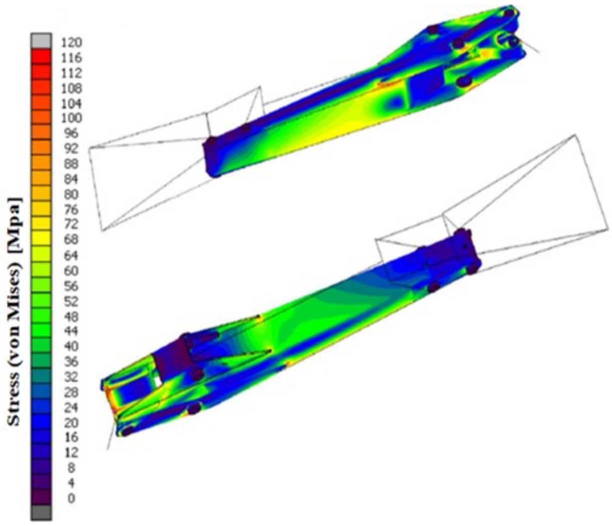

Figure 2.6. Von Mises stresses in the eccentric loading scenario

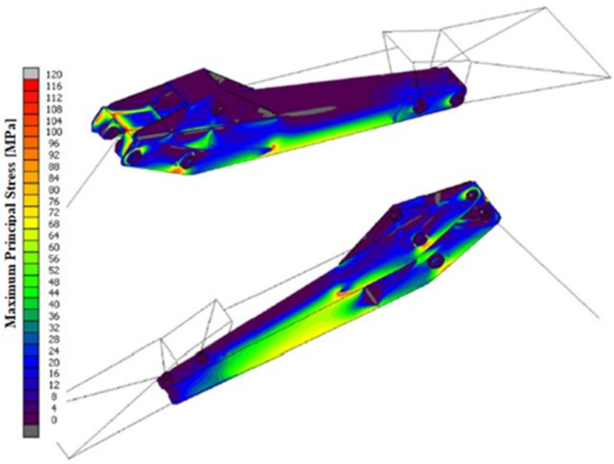

Figure 2.7. Maximum principal stresses in the eccentric load scenario

While examining both the analysis results of the current model and the analysis results of the alternative designs to be created, since the tensile force in the weld regions is critical, especially the maximum principal stress values were taken into account while evaluating these regions. The critical upper value when examining the weld regions is 120 MPa. This value has been determined by the experience gained from previous analyzes and tests on the existing machine. Since the shear stress is critical in the regions outside the welded regions in the model, especially von Mises stress values are taken into account in these regions. In these regions, the yield strength values of the materials were taken into consideration as the upper critical value. The fact that the values below the critical values are mostly found in the backhoe arm body in the analysis results show that the structure is suitable for weight reduction applications.

2.1.1. Process of reducing the thicknesses of parts

A box-section design is seen in the current backhoe arm design. Basically, this structure is obtained by welding steel sheets of different thicknesses to each other. The current structure and the sheet metal parts that make up the structure are shown in the Figure 2.8.

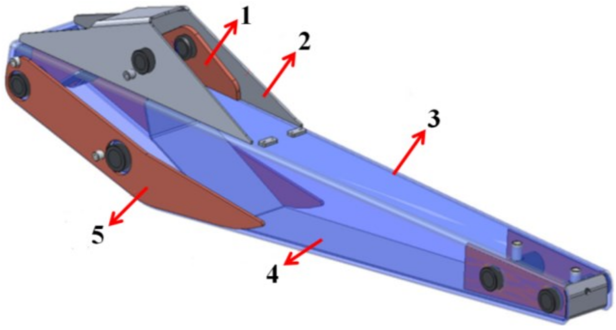

Figure 2.8. Structure of backhoe arm

The low stress values seen in the overall structure in the strength analyzes made on the existing design show that a lighter structure can be obtained by determining the optimum thicknesses by changing the thickness of these sheets. In the thickness reduction process applied in this direction, different thickness combinations were foreseen for the optimum design by considering the stress distributions on the existing design. While determining the alternative thickness values of the parts, standard thicknesses, which are frequently found in the market, were chosen in order to keep the production costs low. These combinations are as in Table 2.2 and the numbering in Figure 2.8 is taken as reference for the part numbers.

Table 2.2. Thickness combinations for thickness reduction process

	Part no	1	2	3	4	5
Part thicknesses (mm)	**Current design**	15	8	8	6	12
	1. combination	12	6	6	5	10
	2. combination	10	5	5	5	8
	3. combination	8	5	5	5	6

2.1.2. Structural analysis of combinations

Structural analyzes were made on 3 different combinations created within the scope of weight reduction process. In the analysis made, the same boundary conditions were determined as the analysis made on the existing design and the same forces were applied in the same way. The results of the structural analyzes made on the alternative designs created with these different combinations are shown in the Table 2.3. While evaluating these results and deciding whether the combination is safe or not, the maximum principal stress value of 120 MPa in the weld region was determined as the critical limit, as was the case while evaluating the analysis results of the current design. In other regions, the von Mises stress value of 235 MPa, which is the 66% (1.5 factor of safety) of yield value of the material, was determined as the critical limit. In addition, in Figure 2.9 and Figure 2.10, the distribution of loads according to combinations in symmetrical and eccentric loading situations, respectively, is shown with safe operating limits.

Table 2.3. Structural analysis results of alternative combinations

		Current structure	1.combination	2. combination	3. combination
Symmetrical loading	Maximum Stress (von Mises) [MPa]	115	120	130	145
	Maximum Principal Stress [MPa]	80	100	125	135
Eccentric loading	Maximum Stress (von Mises) [MPa]	115	130	140	145
	Maximum Principal Stress [MPa]	90	115	135	150

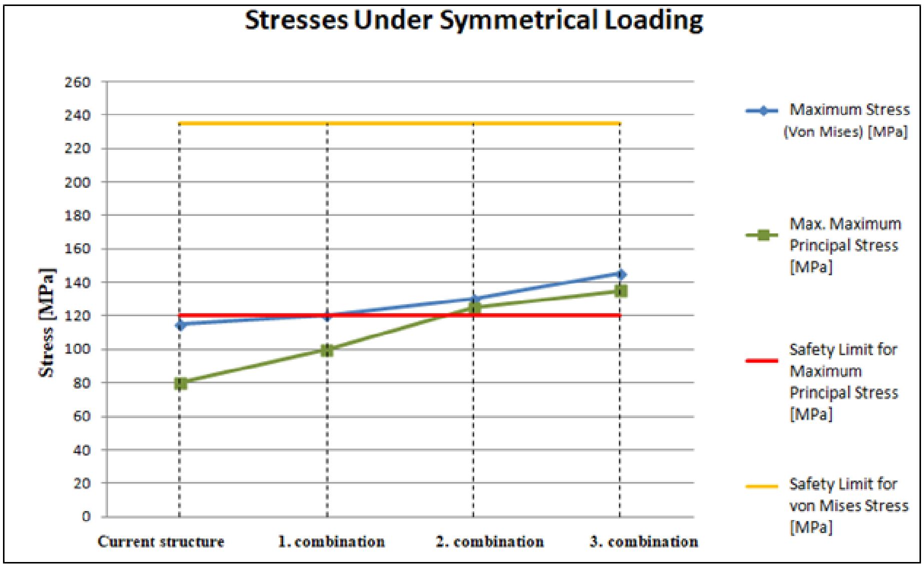

Figure 2.9. Distribution of stresses according to combinations under symmetrical loading

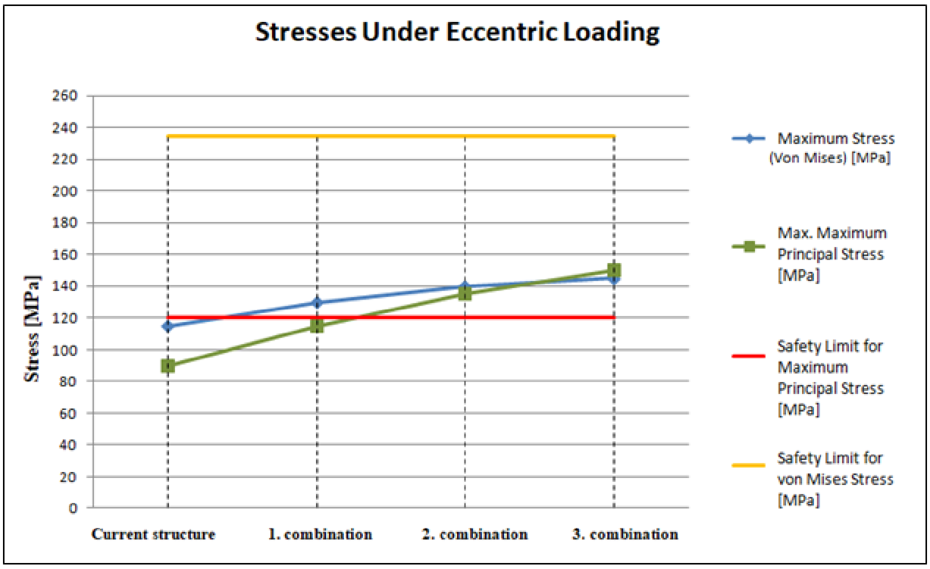

Figure 2.10. Distribution of stresses according to combinations under eccentric loading

While evaluating the data specified in the tables, the safety factor was taken as 1.5, as in the analysis made on the current design. In this case, it is seen that the maximum principal stresses occurring in the second and third combinations are above the safe values. The values obtained in the first combination, on the other hand, are close to the maximum safe stress values, but do not exceed the values considered safe. The stress values obtained as a result of the strength analysis performed on the first combination in symmetrical loading and eccentric loading scenarios are shown in Figure 2.11, Figure 2.12, Figure 2.13 and Figure 2.14.

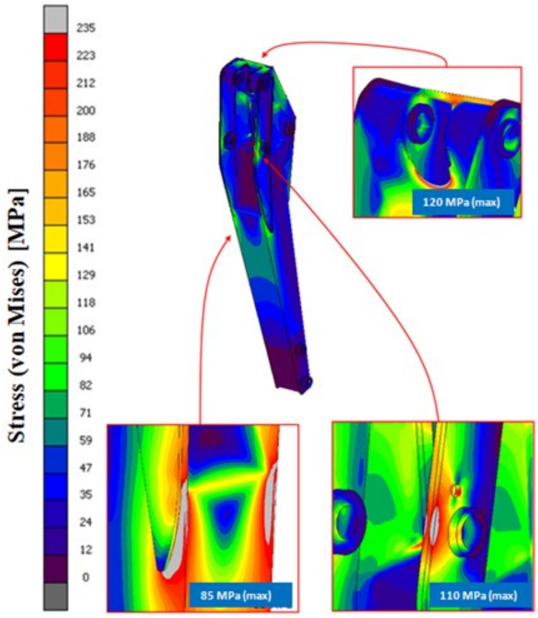

Figure 2.11. Von Misses stress distribution on the first combination under symmetric load

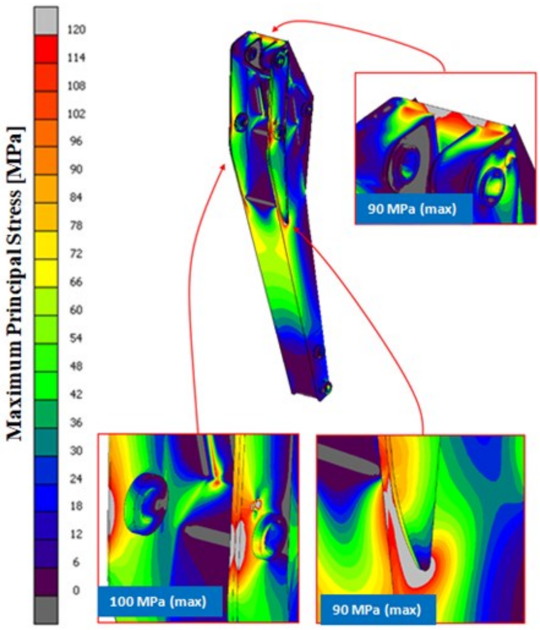

Figure 2.12. Maximum principal stress distribution on the first combination under symmetric load

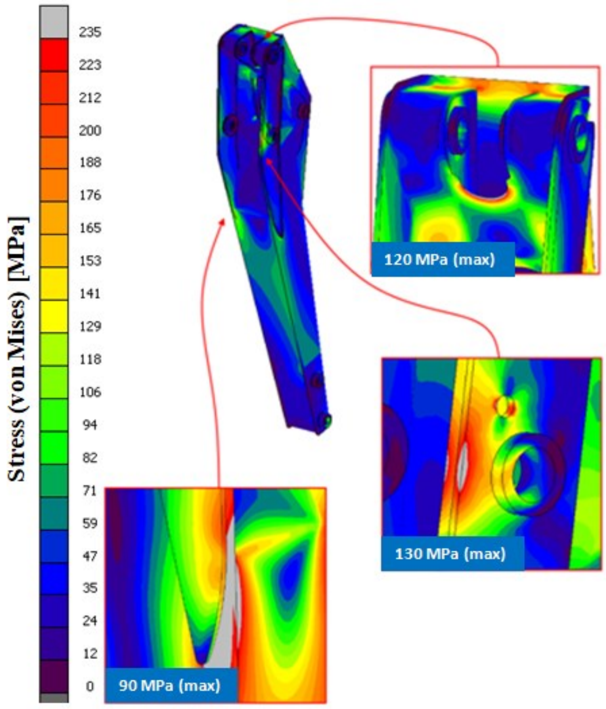

Figure 2.13. Von Misses stress distribution on the first combination under eccentric load

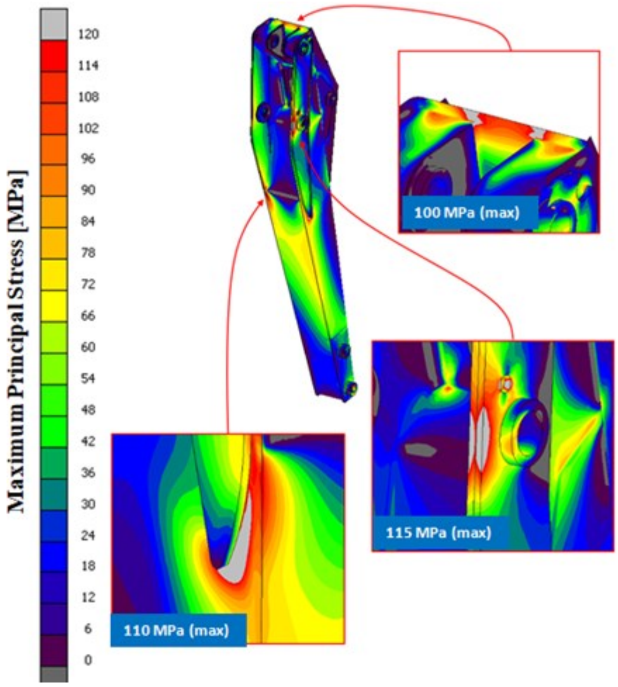

Figure 2.14. Maximum principal stress distribution on the first combination under eccentric load

In line with the results of the structural analysis, the sheet thicknesses in the first combination were evaluated as optimum and applicable, and the design was revised accordingly. In the revised new design, the structure has become lighter as a result of the optimum reduction of the thickness of the sheet metal parts in the backhoe arm. In Table 2.4, weight comparison between current model and optimized model is shown.

Table 2.4. Weight comparison between current model and optimized model

Part no	Quantity	Weight of parts (kg) Current model	Lightened model	Weight difference (kg)	Weight difference (%)
1	2	5.2	4.1	2.2	21
2	1	45.3	33.9	11.4	25
3	1	110.8	83.1	27.7	25
4	1	21.7	18.1	3.6	16
5	2	13.8	11.5	4.6	17
			Total weight difference (kg)	49.5	

As can be seen in Table 2.4, a total of 49.5 kilograms of lightening was achieved in the backhoe arm after the weight reduction process. This amount means a reduction of approximately 20%, based on the total weight of the backhoe arm. This reduction in weight will also reflect on the fuel consumption of the machine as a decrease, thus increasing the efficiency of the machine in terms of fuel consumption.

3. Results and Discussion

In this study, the weight of the backhoe arm was reduced by making thickness reduction on the backhoe arm of a backhoe loader by finding optimum thicknesses of main structural parts of the backhoe arm.

With the lightened backhoe arm, approximately 20% compared to the current design, the backhoe group has lightened approximately 49.5 kg in total. The fact that the excavator arm design obtained as a result of this study is lighter than the existing design will reduce the fuel consumption of the machine. At the same time, it is foreseen that the working efficiency of the machine will increase thanks to this study. Reducing the thickness of the parts that make up the backhoe arm reduced the costs of these parts. This study also shows how the weight reduction process applied during the design phase affects key factors such as the cost and performance of both the part and the machine. The analysis and weight reduction processes carried out within the scope of this study are applicable to many other heavy equipment in the industry.

Based on this study, similar weight reduction and design processes can be applied to the structural mechanism elements of the machines commonly used in the construction equipment sector such as backhoe loaders, loaders and excavators, which are moved by hydraulic cylinders such as backhoe boom and loader arm, apart from the backhoe arm. In this way, the machine-wide cost can be further reduced and machine efficiency can be further increased. In addition to the weight reduction and strength analysis processes, fatigue analyzes can be applied on critical mechanism elements and the useful life of the mechanisms can be predicted with the results of fatigue analysis.

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CRediT authorship contribution statement

Fazıl Anıl ÖZCAN: Literature review, performing the analyzes and calculations, design of the new alternatives and new cylinder
Mesut DÜZGÜN: Manuscript editing, guiding the writing processes, journal correspondence

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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