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A novel design procedure for tractor clutch fingers by using optimization and response surface methods †

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Abstract

This paper presents a methodology for re-designing a failed tractor transmission component subjected to cyclic loading. Unlike other vehicles, tractors cope with tough working conditions. Thus, it is necessary to re-design components by using modern optimization techniques. To extend their service life, we present a design methodology for a failed tractor clutch power take-off finger. The finger was completely re-designed using topology and shape optimization approach. Stress-life based fatigue analyses were performed. Shape optimization and response surface methodology were conducted to obtain optimum dimensions of the finger. Two design parameters were selected for the design of experiment method and 15 cases were analyzed. By using design of the experiment method, three responses were obtained: Maximum stresses, mass, and displacement depending on the selected the design parameters. After solving the optimization problem, we achieved a maximum stress and mass reduction of 14% and 6%, respectively. The stiffness was improved up to 31.6% compared to the initial design.

Keywords: Clutch; Response surface method; Shape optimization; Topology optimization

1. Introduction

The heavy vehicle industry has reflected a steady growth over the last decade by constantly trying to upgrade their technology and production processes in the world. These vehicles have a significant effect on the commercial activity and transportation. Among these vehicles, tractors are the most common investment in the agricultural sector, and unlike the other vehicles they have to cope with tough working conditions. Tractors have to efficiently transfer power from the engine to the drive wheels in order to move, as well as the power required for Power take-off (PTO) in order to actuate the farming equipment's must be provided. The power transfer can be achieved through an appropriate transmission system. Transmission system components of a tractor take about 25-30% of its total costs. Breakdown of the transmission component leads to large production losses and to loss of reputation for the companies [1]. Especially for the clutch, an unexpected failure occurring in this component can, therefore, lead to a total breakdown of the tractor transmission system and possibly cause long-term tractor down times.

Automotive part manufacturers face major competition to design lightweight, cost-effective and more efficient components that exhibit more precise dimensions, less machining, and processing costs. New tools are necessary to help the designer at the initial design phase for these purposes. Computer-aided engineering (CAE) provides a means of verifying the design in terms of the durability of structures without making prototypes, thus reducing the design cycle time and minimizing costs [2, 3].

Since fatigue behavior is a key constraint in design and performance evaluation of vehicle components, engineers need to model and design for fatigue behavior early in the product design stage with the help of the CAE. The fatigue performance of tractor parts is investigated in the literature. Nanaware and Pable (2003) determined the optimum value of the spline root radius for tractor rear axle shaft under the fatigue criteria [4]. Fourlaris et al. (2007) investigated gauge and material strength effect on the fatigue performance of automotive suspension arm. They stated that FE modeling accelerated the service life testing [5]. Palma et al. (2009) determined the fatigue damage of rear trailer hook town assembly using the local material response measured during experimental tests. They reported that increased stiffness is a parameter that indicates ability to withstand durability tests without fail [6]. Shim

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and Kim (2009) performed shape optimization to get optimal design to increase the fatigue life of the pulley while minimizing the weight of it [7]. Talemi et al. (2015) investigated the fretting fatigue behavior of shock absorber washer-disc contact configuration. They indicated that numerical modeling increased the possibility to easily investigate more details compared to performing experimental tests [8].

As well as CAE, structural optimization techniques also play a very important role in the product development phase. Structural optimization techniques consist of size, shape and topology optimization. Size optimization allows the variation of predefined dimensions of the product. While shape optimization is more flexible than size and allows more freedom in the design of the geometry. Topology optimization allows the maximum freedom in the design space by a possible change of a structure [9]. In the topology optimization process, it is only necessary to know the design space, the boundary conditions and the loads. With the obtained results, the designer is able to define a detailed design. Topology optimization has proven very effective in determining the topology of initial design structures for component development in the conceptual design phase [10]. Shape optimization also has received attention from automotive part manufacturers. The aim of shape optimization is to find the best profile of a structural system that improves its performance. To reduce the weight and prevent fatigue failure of automotive components, shape optimization techniques are generally used. Response surface method (RSM) and Design of experiment (DOE) methods are also the indispensable tools in the product development process. To investigate the combined response of design variables, RSM is needed. It involves regression surface fitting to obtain approximate responses, design of experiments to obtain minimum variances of the responses and optimizations using the approximated responses [11].

There are various studies in the literature applying optimization techniques for product development [12-14]. Shenoy and Fatemi (2005) did a shape optimization study on a connecting rod by considering the improvement in weight under a fatigue life constraint. They stated that the optimized connecting rod was 10% lighter and 25% less expensive, as compared to the existing connecting rod [15]. Kaya et al. (2010) developed a new design proposal for a failed clutch fork with the topology optimization approach, and then shape optimization by the response surface method. The study results showed that 24% mass reduction was obtained. And also the stiffness improved up to 37% in comparison with the original clutch fork [16]. Heo et al. (2013) presented shape optimization of lower control arm considering multi-disciplinary constraint conditions, which are stiffness, strength and durability [17]. Tanlak et al. (2015) investigated the shape optimization of the bumper beam to maximize the crashworthiness of the beam. They indicated that significant improvement in the crashworthiness of the bumper beam currently in-use and also resistance to low-velocity impact improved [18].

Tractor clutch, which is usually of the friction type, is the essential and critical part of the tractor transmission system. Unlike automobile clutches, it operates the torque transfer to PTO and wheels in the same clutch system. The torque passes from the engine through a friction clutch which is frequently operated with the same pedal as the transmission clutch. Some works dealing with transmission systems and clutch mechanism are available in the literature. The majority of these studies are related to automobiles [19-21]. Li-Jun et al. (2008) established an optimum mathematical model of a diaphragm spring clutch in order to meet the design requirements. They reported that the design period can be shortened, the clutch size can be decreased and the production cost can be lowered [22]. Danev et al. (2014) determined the possibility of decreasing finger backlash that may be achieved by profiling of the fingers of the diaphragm spring. The research results showed that there is a significant capacity for contribution to the diaphragm spring fingers' stiffness, which can be achieved by changing their cross-section [23]. Karpat et al. (2014) investigated the effect of thickness on stress and deformation of the PTO finger by using Finite element analysis (FEA). They observed that approximately 13% stress decreasing with the increment of the 0.5 mm for the finger thickness [24].

Although there are various research works related to failed automotive components, there is a lack of transmission parts, especially heavy vehicle clutch components in the literature. Whereas for transmissions systems one of the biggest parts of these vehicles costs, breakdowns often arise due to tough working conditions. Therefore, our aim was to design optimum PTO finger of the tractor clutch dimensions considering the service life. Stress-life based fatigue analyses were conducted and possible crack locations were determined. A new design proposal for a failed tractor clutch PTO finger was determined with topology optimization and then shape optimization by the response surface method was effectively used to improve the new PTO finger mechanism.

1.1 Introducing the tractor clutch and failures

Tractor clutches provide a way for the tractor operator to start a smooth delivery of power to the transmission, to interrupt power while the transmission gear ratio is being changed, and to interrupt power when the tractor is to be stopped. Tractor clutches mainly consist of cover, inner and outer pressure plates, diaphragm disc and finger mechanism (Fig. 1). Unlike the automobile clutches, tractor clutches need two different disks due to the working of the PTO and wheels simultaneously. Because of the two different disks, disk allocation process cannot be done applying courses directly onto the diaphragm spring. The separation process is accomplished by using the finger mechanism. This mechanism consists of two different parts which can be work simultaneously or independently. Both parts are activated with the bearing courses on their fingers. Thus, fingers are one of the most crucial components of the tractor clutch.

The working principle of the PTO finger mechanism can be

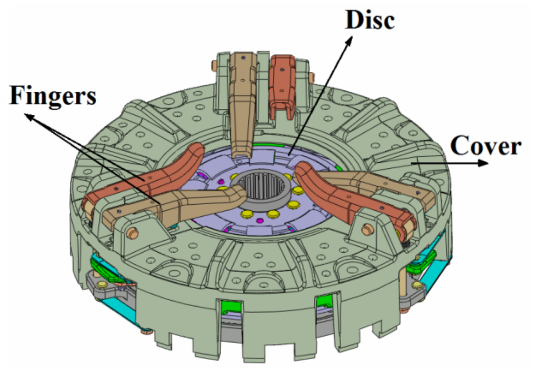


Fig. 1. A schematic illustration of the tractor clutch.

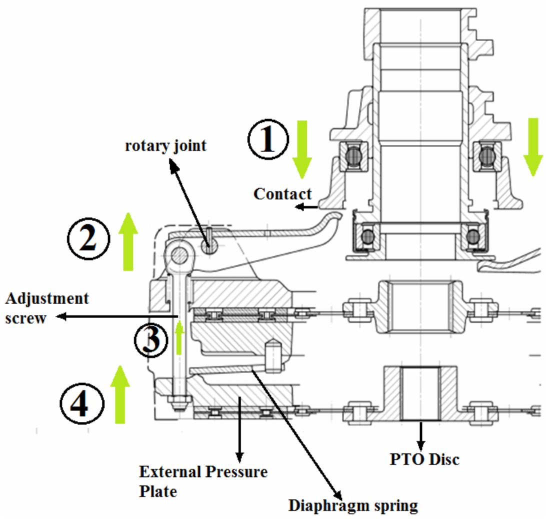


Fig. 2. Working principle of the PTO finger mechanism.

Fig. 3. Damaged PTO finger images.

seen in Fig. 2. In the first position (1) roller bearing is giving the course on the PTO finger. The PTO finger is a kind of lever; it converts the movements of the bearing to adjustment bolt, which is fixed on the external pressure plate (2). The adjustment bolt moves upward with the pressure plate; thus, it removes the pressure, which is applied by the diaphragm disk, on the PTO disk, and therefore the torque transmission ends.

In the tractor transmission system, due to tough working conditions and cyclic loadings, PTO fingers fail before completing their design life (Fig. 3). Most of the damage is caused

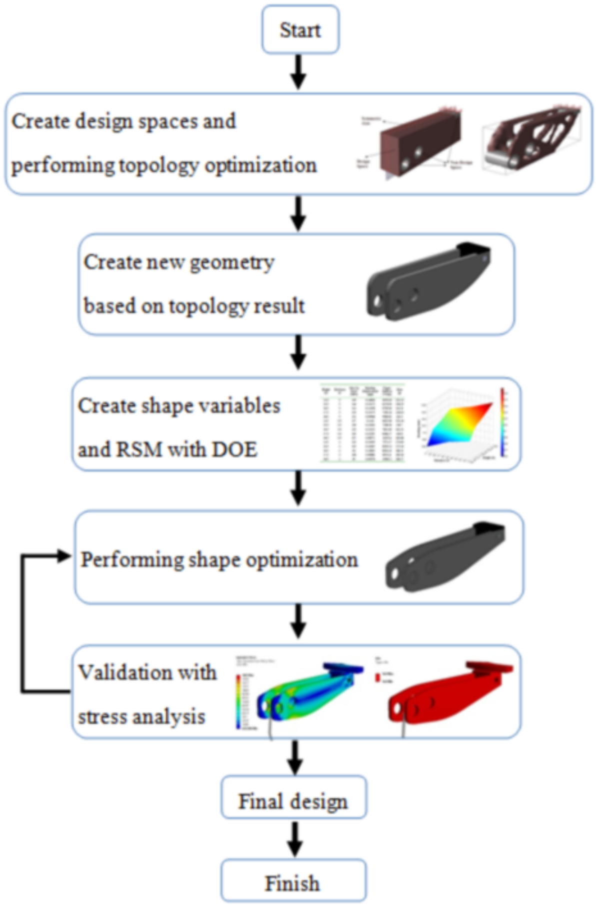
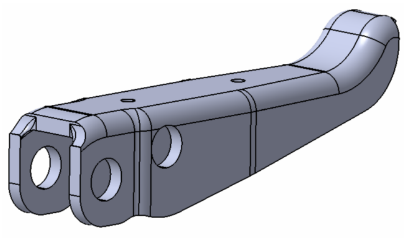


Fig. 4. Design and validation process flowchart.

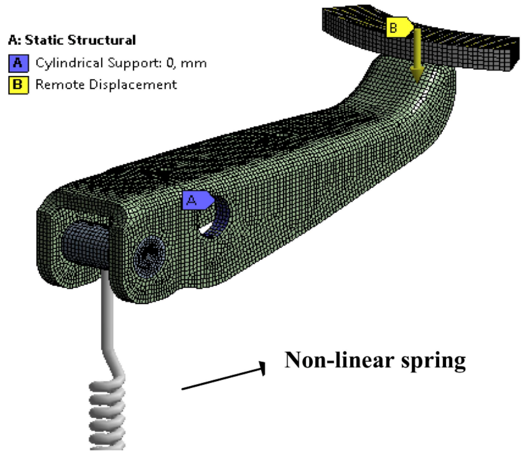
by a failure. Therefore, the optimum size of the finger should be determined using modern optimization tools. So it can perform its functions without damaging during the warranty period.

2. Methodology

Computer aided design (CAD) and Finite element analysis (FEA) are accepted across a wide range of industries as crucial for product design and optimization and validation phases. They are essential in order to predict accurately the safety performance of automotive parts. FEA techniques allow design engineers to predict the maximum stress locations and life of the parts before making a prototype. Traditionally, bench tests have been used to investigate the structural behavior of prototype products and to develop new products. However, such tests are time consuming and expensive. FEA reduces the prototype costs and makes the correct design possible before the bench test. This study consists of three main phases. At the beginning, a stress-life based fatigue analysis of current clutch PTO finger was undertaken for determining the failure locations. Topology optimization was conducted for obtaining low mass and most rigid geometry. And then shape optimization and RSM with DOE was conducted to obtain



(a)



(b)

Fig. 5. (a) CAD model; (b) FEA model and boundary conditions of the PTO finger.

optimum dimensions of the PTO fingers. Fig. 4 summarizes the whole design and validation process used in this study.

2.1 Stress-life based fatigue analysis of current clutch PTO finger

Stress-based fatigue analysis is typically used for life prediction of components subject to high-cycle fatigue where stresses are mainly elastic. Fatigue analyses are inevitable for the design of the tractor clutch components which have cyclic operating conditions. Moreover, by using fatigue analysis, crack initiation locations could be found without fatigue tests; thus time savings can be ensured. The starting point of fatigue analysis is the response of a structure to the input loading which is generally expressed as a stress or strain time history using FEA. The maximum stress-based fatigue analysis is generally used for the life prediction of the parts in the automotive sector. Once the stress histories are obtained, the fatigue damage and life can be calculated for each loading condition, and the total fatigue damage can be approximated as the sum of damages from all the loading conditions.

The first step of this study involves modeling the failed PTO finger to determine the current situation. CAD model of the PTO finger was obtained using CATIA software (Fig. 5(a)). Finger length and height were determined according to the failed geometry, which was 130 mm and 35 mm, respectively. The CAD model of the failed PTO finger was exported

Table 1. Material properties of current PTO fingers.

Material	DD11 Steel
Modulus of elasticity	215 GPa
Yield strength	210 MPa
Tensile strength	340 Mpa

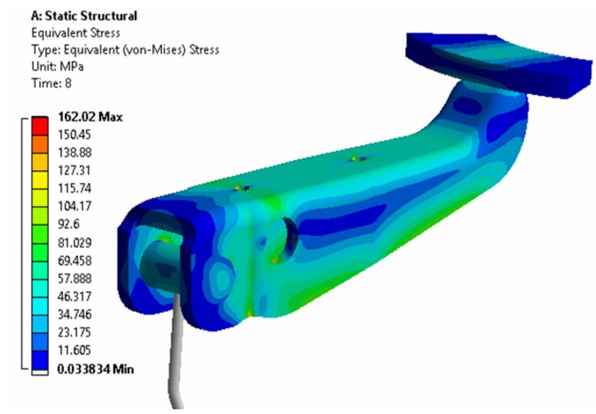


Fig. 6. Von-Mises stress distribution of current PTO finger.

from CATIA to ANSYS 16 Workbench for the creation of the finite element model. Due to the complexity of the geometry (corners, holes and the rounds) quadratic, both hexahedron and tetrahedron elements were used to create mesh structure which consisted of 43700 elements and 150000 nodes (Fig. 5(b)). Additionally, to represent the non-linear operating characteristics of diaphragm spring, COMBIN39 element type was chosen. Bearing apparatus was also modeled. A bonded contact was defined between pin and finger. A cylindrical support was located in the second pin region, and this support constrained five degrees of freedom; it only allows rotation of the tangential axis. Remote displacement was applied on the bearing; the applied course was measured as 18 mm on the axial from the test bench and also maximum reaction force from bearing to finger was measured as 980 N. We did a contact analysis between bearing apparatus and the PTO finger. A frictional contact was defined to simulate the operations conditions of the finger. The friction coefficient between finger and bearing was determined as 0.1.

Tensile tests were performed on sample materials to obtain true stress-strain curves for sheet metal (DD11) in the Uludag University Laboratory. The mechanical properties of the material are shown in Table 1. After defining the boundary conditions, loads and material properties, static stress analyses were conducted in ANSYS software.

Linear static stress analyses of failed clutch finger were performed in ANSYS software. The highest stresses were observed at the pin region whose value was 162 MPa (Fig. 6). The maximum displacement was found as 0.168 mm.

The results of the stress analyses were used in stress-life based fatigue analyses. The fatigue life simulation of the failed PTO finger was conducted using ANSYS fatigue tool [25].

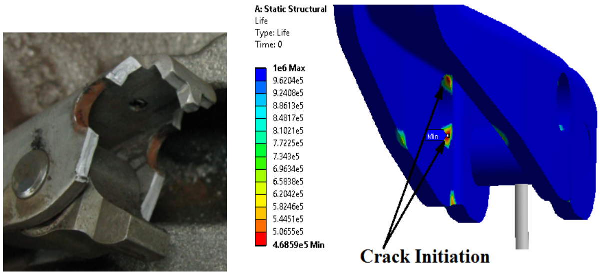


Fig. 7. Fatigue life plot of the finger and failed PTO finger.

The maximum stress results were used to calculate the fatigue life of the PTO finger. Cyclic loads were applied on the finger between 0 - 980 N. The fatigue analysis type was chosen as zero-based stress life. Life of the PTO finger was calculated according to the Goodman mean stress correction. Contour plot of the fatigue life result is given Fig. 7. Crack initiation began in 468000 cycles at the pin region. The correlation of results between the failed finger and the fatigue analyses results in terms of crack initiation location is shown Fig. 7. The results show that the stress-life approach predictions are very reasonable and show good correlation with the area of failure.

2.2 Topology optimization of clutch PTO finger

Topology optimization method allows finding optimum material distribution in structures before size or shaping optimization. The objective of topology optimization is to minimize the compliance, which is equivalent to maximizing the stiffness. In this study, topology optimization is used in a variety of circumstances to help identify potential improvements to PTO finger design. Generally, homogenization and material distribution methods are used for topology optimization. The material definition is as void or solid.

Manufacturability is the main constraint for topology optimization. Therefore, it needs to be revised in the topology design under the manufacturability criteria. The PTO finger is to be designed as a sheet metal cutting process by a punch. Cutting operations involve the separation of the metal of the sheet in certain areas.

To perform topology optimization, a design space was defined according to the dimensions of the current PTO finger. The design and non-design spaces were defined in the solidThinking Inspire software (Fig. 8). The boundary conditions of the PTO finger were applied on the design space. Symmetry axis was defined as a manufacturing constraint. Optimization target was defined to make stiffness maximization and volume reduction of the PTO finger defined as 50%.

The result of the topology optimization for 50% volume reduction is given Fig. 9. It can be seen that this design is impossible for manufacturing. Therefore, some design changes were performed.

Topology optimization serves as a design draft for the creation of a new FE model for the subsequent simulation calculation and shape optimization procedure. The new PTO finger

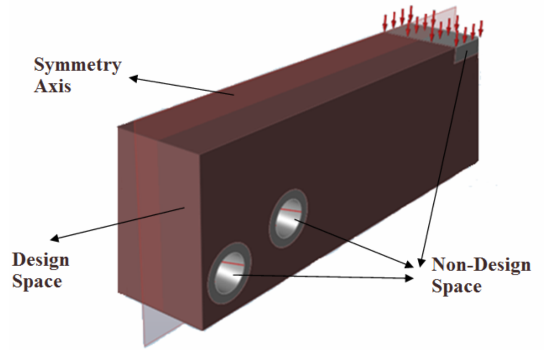


Fig. 8. Design and non-design spaces of the PTO finger.

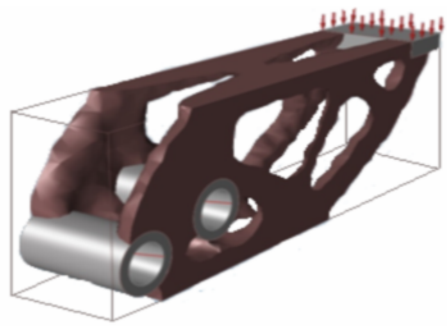


Fig. 9. Result of the topology optimization.

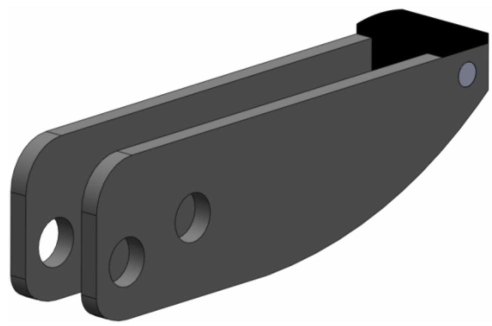


Fig. 10. New design of PTO finger.

was developed based on the results of the topology optimization (Fig. 10).

2.3 Response surface method and shape optimization of new PTO finger

The dimensions of the structure after topology optimization are not precise. Therefore, local developments and upgrades are compulsory. In the design process of the PTO finger, the last step is shape optimization. To define the optimization problem, we used design of experiment and response surface methodologies.

Approximation methods are the key way for reducing the computational time of the structural optimization studies. These methods utilize a variety of the optimization problems due to obtaining the global behavior of the original function and shortening the computational time. In this study, due to the complex mechanical behavior of the clutch PTO finger, it is hard to describe analytically of the objective function. Therefore, RSM was conducted for obtaining the functions. First, response of interest was obtained by the design of experiments. Then a mathematical model was determined which was appropriate with collected data from the experiments. After the curve fitting, the RSM defines surfaces which include the behavior of the objective function inside the certain design space. Finally, the optimum setting of the experimental factors was found.

In this study, cubic polynomials are used as fitting curves. Thus, the response surface can be described in terms of two design variables as x_1 and x_2, which are height and thickness of the finger respectively [16]:

$$y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \beta_3 x_1 x_2 + \beta_4 x_1^2 x_2 + \beta_5 x_2^2 x_1 + \beta_6 x_1^3 + \beta_7 x_2^3 . \quad (1)$$

New variables x3, x4, x5, x6, x7, which are used to Simplify Eq. (1) can be described as:

$$x_3 = x_1 x_2 , \quad (2)$$
$$x_4 = x_1^2 x_2 , \quad (3)$$
$$x_5 = x_2^2 x_1 , \quad (4)$$
$$x_6 = x_1^3 , \quad (5)$$
$$x_7 = x_2^3 . \quad (6)$$

Eq. (1) can be written as a linear regression.

$$y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \beta_3 x_3 + \beta_4 x_4 + \beta_5 x_5 + \beta_6 x_6 + \beta_7 x_7 . \quad (7)$$

The coefficients β_i are determined through the least square error method. Using two design variables and *n* experiments the coefficients are calculated using Eq. (7)

$$\beta = (X^T X)^{-1} X^T Y , \quad (8)$$

where

$$Y = \begin{Bmatrix} y_1 \\ y_2 \\ y_3 \\ . \\ . \\ . \\ y_n \end{Bmatrix} \quad x = \begin{bmatrix} 1x_{11} & \ldots & x_{71} \\ 1x_{12} & \ldots & x_{72} \\ 1x_{13} & \ldots & x_{73} \\ .\,. & \ldots & . \\ .\,. & \ldots & . \\ .\,. & \ldots & . \\ 1x_{1n} & \ldots & x_{7n} \end{bmatrix} \quad \beta = \begin{Bmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \\ \beta_5 \\ \beta_6 \\ \beta_7 \end{Bmatrix} . \quad (9)$$

In this study, we used design of experiment to determine the combination of design variables which must be tested. The full factorial DOE method was applied. This method is only useful when the number of the factors and levels are few. The full factorial DOE investigates all possible combinations of the factor levels. Such a design is beneficial for calculating all main and interaction effects.

Fig. 11. Design variables of the new PTO finger.

Design variables of this study were determined as finger height (h) and thickness (t) (Fig. 11). Finger thickness is a variable that has the greatest impact on the finger mass. Therefore, sheet thickness values were taken as three levels, such as 3 mm - 3.5 mm - 4 mm, because of the standard sheet thickness. The second variable is height. Finger height is an important parameter on the finger stiffness. Because of the geometrical constraints of the clutch house it was ranged between 14.5 mm to 18.5 mm due to PTO finger height minimum and maximum size.

The aim of this study was to design lightweight PTO finger under the durability conditions. Therefore, minimizing the maximum von-Misses stress function was defined as the objective function. It is possible to design more safely a PTO finger with thicker steel sheet, but structural mass will be increased. So it is not an optimum design solution. Therefore, structural mass is one of the constraints. Due to the failed clutch, PTO finger mass was 142 g mass criteria was defined at least the same weight or lighter than the initial design.

The second constraint was defined as the maximum displacement occurs on the finger. It means the stiffness of the PTO finger. In the first design, the maximum bending displacement measured 0.168 mm. After the shape optimization, the rate of increase in stiffness is targeted at least 30% so the constraint was defined as 0.12 mm. In this way, the optimization problem was created. The optimization problem was defined as follows:

Objective Function:

- Min. [f(h,t)Stress] .

Subject to

- f(h,t)Mass ≤ 142 g,
- f(h,t)Displacement ≤ 0.12 mm,
- 14.5 ≤ h ≤ 18.5 , 3 ≤ t ≤ 4.

The optimization problem is solved by using fmincon function in MATLAB.

3. Results and discussion

We investigated a failed clutch PTO finger from the fatigue life point of view and then determined a new finger design

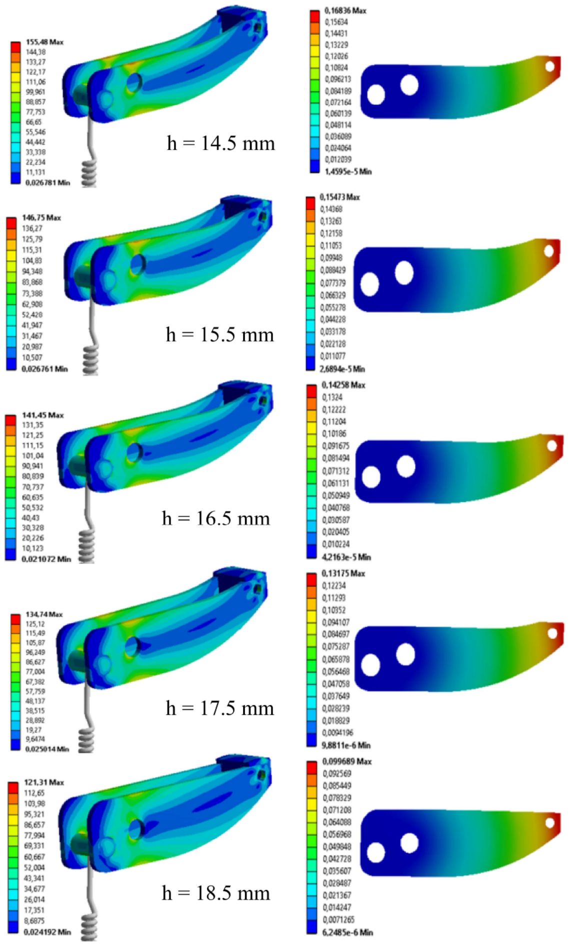


Fig. 12. Stress and displacement FEA results for t = 3 mm.

using structural optimization tools. The results of the stress analyses were used in stress-life based fatigue analyses. Crack locations of the failed finger and simulation model were correlated. Primarily in the design stages, stress analysis verification was performed to accurately predict the performance of the new design to obtained topology optimization. The same element size, element type and boundary conditions were applied on the designed PTO finger. Stress analysis results showed that the maximum equivalent von-Mises stress values were decreased considerably. However, this design failed to meet the optimization problem criteria. These results showed that new design which obtained topology optimization had some improvements according to the failed finger. But local shape improvements are still necessary after topology optimization. Especially in terms of structural mass of the component should be reduced.

The last step of the design process was response surface shape optimization procedure. Based on the DOE results in Table 2, the response surface models were generated. FE analyses were conducted for each case in the DOE table (Fig. 12).

To create a shape optimization problem, these outputs

Table 2. Full factorial design table and FEA results.

Height (h) (mm)	Thickness (t) (mm)	Max. Eq stress (MPa)	Bending displacement (mm)	Stiffness (N/mm)	Mass (g)
14.5	3	155	0.16836	5678.30	134.44
15.5	3	146	0.15473	6178.50	138.25
16.5	3	141	0.14258	6705.00	142.11
17.5	3	134	0.13175	7256.16	146.04
18.5	3	121	0.09968	9589.82	161.5
14.5	3.5	137	0.1443	6625.08	152.26
15.5	3.5	136	0.13262	7208.56	156.7
16.5	3.5	125	0.12221	7822.60	161.21
17.5	3.5	119	0.11292	8466.17	165.8
18.5	3.5	107	0.08571	11153.6	183.88
14.5	4	122	0.12625	7572.27	170.09
15.5	4	115	0.11603	8239.24	175.16
16.5	4	111	0.10693	8940.42	180.31
17.5	4	106	0.09880	9676.11	185.56
18.5	4	95	0.05978	15992.2	206.17

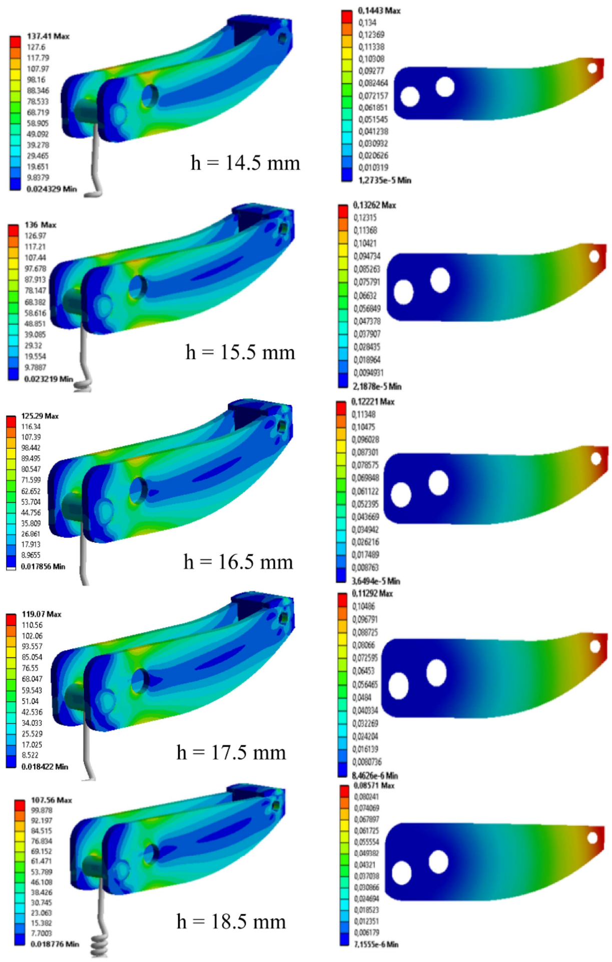


Fig. 13. Stress and displacement FEA results for t = 3.5 mm.

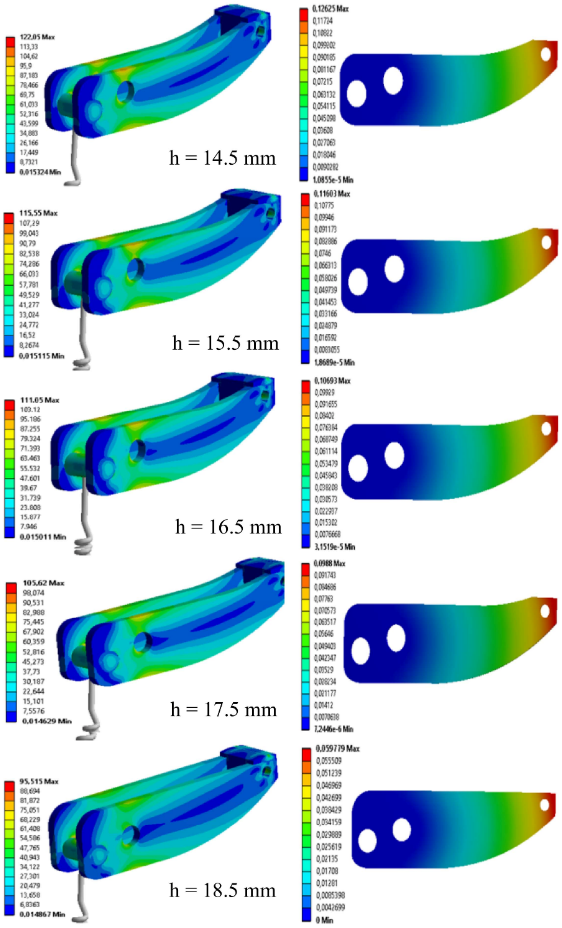


Fig. 14. Stress and displacement FEA results for t = 4 mm.

should be defined by polynomials. Thus, a model which is cubic polynomial was developed below; and according to the full factorial design table, constant coefficients (a_1, a_2, a_3, a_4, a_5, a_6, a_7, a_8) of the cubic polynomials were found by using MATLAB according to the least squares method.

The response surface models were developed. According to the DOE results, response surfaces of the maximum von-Mises stress, maximum bending displacement, and minimum finger mass are given in Fig. 15.

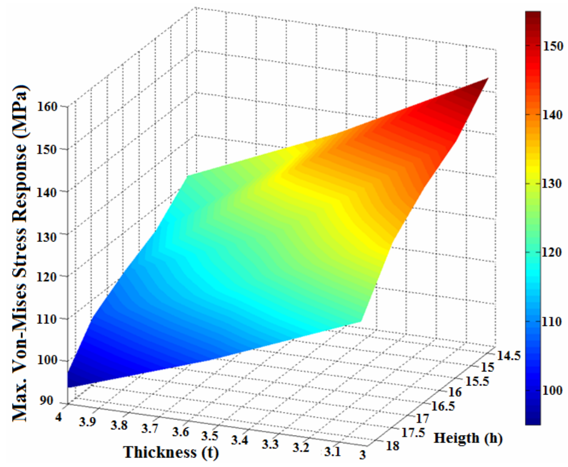
$$\text{Model} = a_1t^3 + a_2h^3 + a_3t^2h + a_4h^2t + a_5th + a_6t + a_7h + a_8.$$

Maximum von-Mises stress:

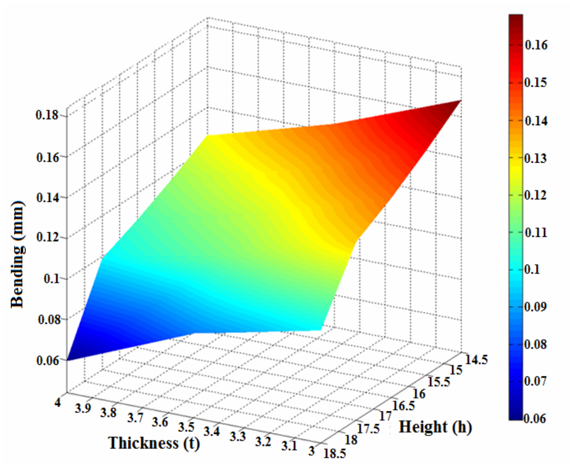
$$Stress(h,t) = -183.98 + 42.24h + 0.0071h^3 + 176.72t - 25.4ht - 0.26h^2t + 5.08ht^2 - 8.24t^3 . \quad (10)$$

Bending displacement:

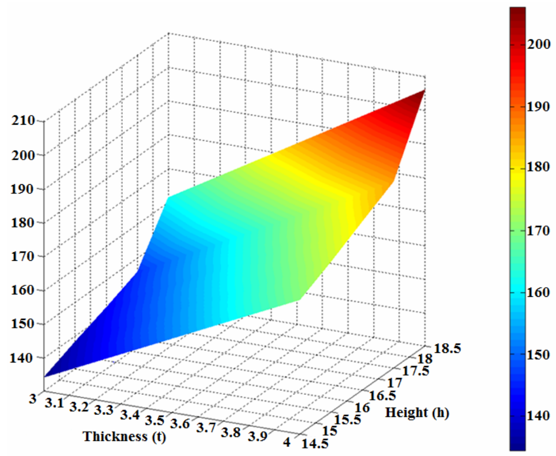
$$u(h,t) = 1.42 - 0.09h - 0.00001h^3 - 0.66t + 0.072ht - 0.0006h^2t - 0.0073ht^2 + 0.012t^3 . \quad (11)$$



(a)



(b)



(c)

Fig. 15. Response surfaces of the: (a) Maximum von-Mises stress; (b) maximum bending stress; (c) minimum finger mass.

Total mass:

$$m(h,t) = 457.76 - 33.812h - 0.045h^3 - 52.46 + 5.574ht + 1.14h^2t - 5.88ht^2 + 9.88t^3 . \quad (12)$$

Table 3. Results of the shape optimization.

Max von-Mises stress (MPa)	133.3
Height (h) (mm)	17.7
Thickness (t) (mm)	2.9
Mass (g)	141.9
Bending displacement (mm)	0.012

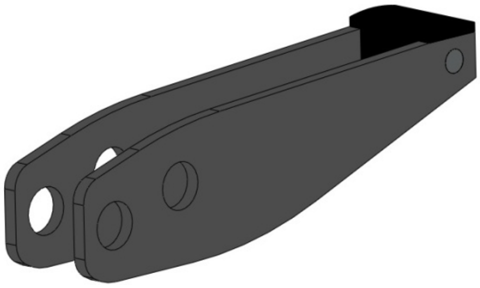


Fig. 16. Final design of PTO finger after shape optimization.

For shape optimization, constraint functions were defined as the total mass and bending displacement. Minimization of maximum von-Mises stress function was defined as the objective function. The results of the optimization problem can be seen in Table 3.

The optimum ultimate dimensions are obtained by using shape optimization. However, the standard sheet thickness of the PTO finger cannot be selected as 2.9024 mm. Instead of 2.9 mm, the finger thickness should be taken 3 mm from the standard sheet metal thickness. According to the shape optimization results the design new PTO finger is re-designed in CATIA (Fig. 16). The new design has some minor trims for getting weight reduction.

After the precise dimensions of the PTO finger are determined, this new optimized finger must be validated, in terms of maximum stress and fatigue life. First, this finger was analyzed in the ANSYS software to find the maximum stress and displacements. In this analysis the mesh type and size remained the same with the first design. The boundary conditions of the analysis were also the same as the initial model. As a result of the stress analyses of the new optimized PTO finger, maximum stress occurred in the pin region and its value is 140 MPa (Fig. 17). While the maximum stresses are compared with the first failed design, they are decreased about 14%. Maximum displacement values also reduced from 0.168 to 0.11487 mm, nearly 31% reductions achieved in the displacement values. Thus, the stiffness of the PTO finger was increased.

After the stress analysis of the PTO finger, fatigue analysis was performed. The fatigue analysis was also conducted under the same boundary conditions. New optimized PTO finger fatigue analysis results show that the new optimized design of the PTO finger passes the analysis successfully. The contour plot of the PTO finger fatigue analysis result is given in Fig. 16. This new finger completes its design life, which is 10^6 cycles without any problem.

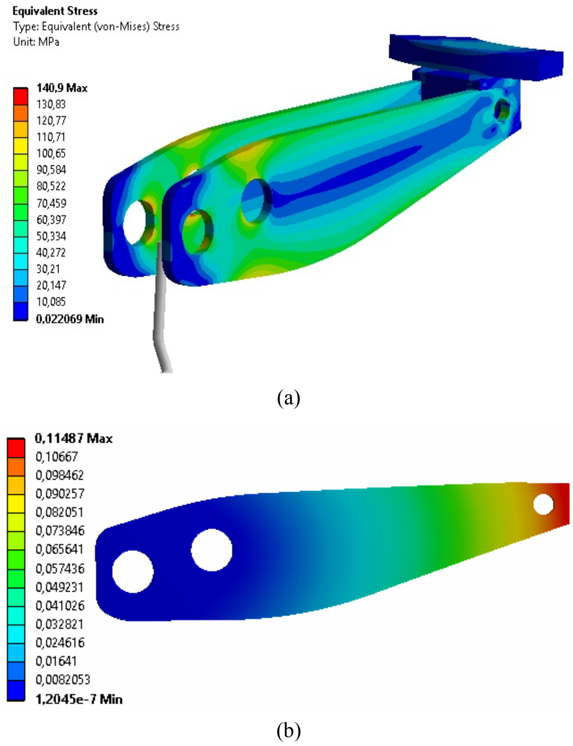


Fig. 17. (a) Von-Mises stress distribution; (b) displacement of the final design PTO finger.

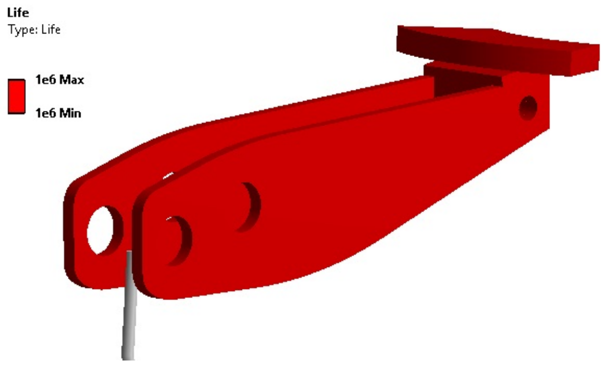


Fig. 18. Fatigue analysis result of the final design.

It is seen that the new optimized design of the PTO finger was validated using the stress-life based fatigue analysis successfully. As a result of the optimization study the finger mass was decreased 6%. Maximum equivalent von-Misses stress reduction of 14% was achieved. The stiffness was improved up to 31.6% compared to the initial design (Table 4).

4. Conclusions

In this paper, a failed tractor clutch PTO finger was investigated and modern optimization techniques to re-design of product were applied. According to the results, the new PTO finger more stiffness than failed finger. As well as it provides

Table 4. Comparison of failed and new PTO finger.

	Failed finger	New finger	Change (%)
Max von-Mises stress (MPa)	162	140.9	-14
Max deformation (mm)	0.168	0.11487	-31.6
Mass (g)	142	133.31	-6.1
Fatigue life (Cycle)	468900	1000000	+53.2

the fatigue life expectations. Through topology and shape optimization, a maximum stress reduction of 14% is achieved and the stiffness is improved up to 31.6% compared to the initial design. Besides structural mass is reduced nearly 6%. As along with stress-life based fatigue analysis the results showed that new design can be conducted 1000000 cycles without any failure. The optimal results show that the developed method can be used to design durable, low manufacturing cost and lightweight clutch parts.

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