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Fatigue Life Enhancement of Three Point Hitch System Brackets in the Garden Series Tractors

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ABSTRACT

The main objective of this research was to enhance the fatigue performance of the brackets found in the three point hitch system used in garden series tractors. This was achieved by using experimental tests and finite element analysis. The manufactured brackets were validated with fatigue rig tests, namely a lifting capacity test, a push-pull test and a lifting-lowering test. The lifting capacity test of three point hitch mechanism was established according to ISO 730-1 standards. In addition to the lift capacity test, problems were also experienced with the cylinder clamping brackets during the push-pull tests. The bracket brakeage occurred during the 11,218th test cycle. According to the test results and finite element analysis, the brackets were strengthened at critical damage points. The thickness of the bracket connection surface was increased from 12 mm to 19 mm and the bracket material was changed from GG25 to GG35. The enhanced brackets passed the tests without any breakage.

Keywords: Hydraulic lift; Three point hitch mechanism; Push-pull tests; Lifting-lowering tests; Fatigue analysis; Stress analysis

Bahçe Tipi Traktörlerin Üç Nokta Askı Sistemi Braketlerinin Yorulma Ömürlerinin İyileştirilmesi

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ÖZET

Bu araştırmanın temel amacı bahçe tipi traktörlerde kullanılan hidrolik kaldırıcılı üç nokta askı sistemi bağlantı braketlerinin yorulma performanslarını iyileştirmektir. Bu amaca deneysel ve sonlu elemanlar analizi yardımıyla ulaşılmıştır. Üretilmiş olan braketlere kaldırma kapasitesi testi, çek bırak testi ve indir kaldır yorulma testleri uygulanmıştır. Kaldırma kapasitesi testleri ISO 730-1 standartlarına göre gerçekleştirilmiştir. Kaldırma kapasitesi testlerine ek olarak uygulanan çek bırak testlerinde silindir bağlantı braketlerinde problemler gözlenmiştir. Denemelerin

11218. çevriminde braketlerde kırılmalar görülmüştür. Test sonuçları ve sonlu elemanlar analizlerine göre braketlerin kritik hasar bölgeleri güçlendirilmiştir. Braket bağlantı yüzeyinin kalınlığı 12 mm'den 19 mm'ye çıkartılmıştır ve GG25 olan braket malzemesi GG35 olarak değiştirilmiştir. İyileştirilen braketler, deneysel testleri kırılma olmadan tamamlamıştır.

Anahtar Kelimeler: Hidrolik kaldırıcı; Üç nokta askı sistemi; Çek bırak testi; İndir kaldır testi; Yorulma analizi; Gerilme analizi

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1. Introduction

The attachments of three point hitch (TPH) systems are mainly implemented in hydraulic lift design (Keçecioğlu & Gülsoylu 2005). The design of an adjustable three point hitch system (TPH) affects the force imposed on lift and tractor equipment (Al-Jalil et al 2001). Previous studies on TPH mechanisms were mainly focused on the transition of tillage forces between tillage implements and tractors, a factor which is extremely important for both operational efficiency and energy management (Alimardani et al 2008; Askari et al 2011).

Otmianowski (1983) expressed that all agricultural machines working on farms are exposed to dynamic loads during tillage especially at bed-like farmlands. Sule et al (2007) studied three-point lift system oscillations during the drafting and implementation of transport, resulting in partial parts failure of the lift system such as the stabilizer bracket. The dynamic behavior of a tractor lift system was modelled by many researchers (Laceklis-Bertmanis & Kronbergs 2010; Laceklis-Bertmanis et al 2013). In these works, the Working Model software for a tractor three point hitch-system with small amplitude oscillation simulations was used. Kolator & Białobrzewski (2011) developed a 2D model for working tractors which was implemented on various types of soil. Portes et al (2013) developed a model which transformed the load on the three-point linkage to the tractor driving wheels. Computer aided engineering analysis led to significant cooperation between design and testing departments. Agricultural machines are subjected to various loads according to different parameters, such as soil conditions and operation types throughout their working life. In order to asses fatigue life, strain data from maximum stressed locations must be obtained or estimated (Chisholm & Harral 1989). Koike & Tanaka (1976) measured the strains at tractor's rear axle housing, to calculate fatigue strength under random load conditions. Mattetti et al (2012) collected strain data from an 80 kW tractor and used field data for accelerated life tests. Paraforos et al (2013) investigated the fatigue properties of tractor chassis and axle housing under random vibration conditions based on dynamic road loads.

Khan et al (2007) expressed that measurement studies increased as different types of transducers were developed and implemented on test samples. Goodwin et al (1993) developed a tri-axial dynamometer for measuring the forces and moments along orthogonal axis. Data acquisition and varying strain-gauge systems were also used by many researchers to evaluate the effects of on the three-point hitch mechanism (Al-Janobi 2000).

A prediction of fatigue failure is important for determining effective design parameters for final production. The best-known fatigue analysis procedures are stress-life, strain-life and crack propagation methods (Socie & Marquis 2000). The stress-life method is generally accepted in the case of high-cycle fatigue problems. Using this method, the stress amplitude can be evaluated with an S-N or Wöhler curve (Radaj 1995). The stress life approach assumes all stresses occur below the elastic limit at all times. Strain life approach is applicable to low cycle fatigue problems.

In this study, the fatigue life of the tractor lift bracket was studied using experimental tests. The push-pull and lifting-lowering tests were additional tests to the standard lifting capacity test. The lifting capacity test results were compared with theoretical calculations based on ISO-

730-1 standards (ISO 1994). The parameters of push-pull and lifting-lowering tests, the leading sources of stress on the brackets, were calculated from the static equilibrium of mechanism. Then, strain life-based fatigue life predictions (Coffin-Manson approach) were calculated using finite element analysis. In experimental studies, broken brackets in the critical equipment were observed using push-pull tests and lifting-lowering tests. Critical damage points of the brackets were determined by cyclic loading. The brackets were then redesigned, dependent on the results.

2. Material and Methods

2.1. Design criteria of hydraulic lift

For this research, the brackets of the hydraulic lift cylinder in garden series tractors were investigated. Pistons were attached directly to the external lifting arms, as can be seen in Figure 1. Hydraulic lifts with external cylinders were preferred for high lifting capacity, easy assembly and servicing capabilities. The hydraulic lifts for the three point hitch mechanisms were designed according to ISO 730-1 standards. The technical properties of the hydraulic lift are given in Table 1. The three point hitch system and hydraulic lift mechanism can be seen in Figure 2.

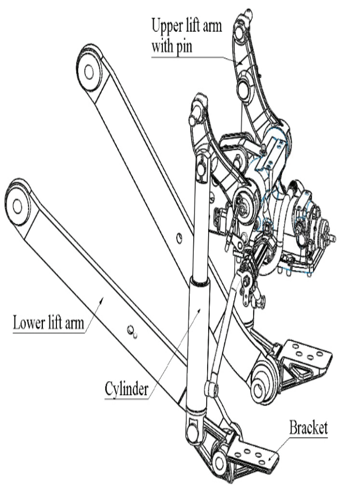

Figure 1- Externally cylindered hydraulic lifts

Şekil 1- Dıştan silindirli hidrolik kaldırıcı

Table 1- Three point hitch system and hydraulic lift properties

Çizelge 1- Üç nokta askı sistemi ve hidrolik kaldırıcı özellikleri

Property	*Value*
TPH category	2
Lifting capacity (kg)	3200
System safety pressure (bar)	195
Flow rate (L min^{-1})	30
Cylinder diameter (mm)	60
Upper lift arm (A)	280
Side suspension arm length (B)	650
Lower lift arm seam (C)	405
Lower lift arm length (D)	900
Top link (E)	750
Equipment height (F)	610
Equipment from the center of gravity (G)	610

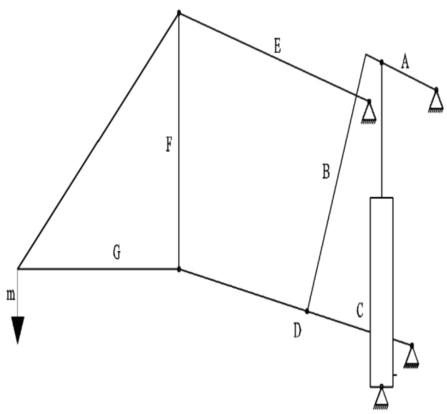

Figure 2- Three point hitch system and hydraulic lift mechanism

Şekil 2- Üç nokta askı sistemi ve hidrolik kaldırıcı mekanizması

2.2. Lifting capacity test

The theoretical lifting capacity was calculated after the first design sketches. Lifting capacity for the final design was tested in line with ISO 730 static capacity test procedure. Lifting capacity was tested with a hydraulic lift test setup that can be seen in Figure 3.

The hydraulic lift cylinders were pressurized with an externally integrated pump and the rods were raised. The pressure was controlled by a manometer installed between the hydraulic lift and pump. The lifting rods were connected via suitable apparatus to the load cell. Lifting capacity was measured with an Esit TB S-Type 5000 kg capacity load cell with an accuracy class of C3, in accordance with OIML R60. Hydraulic pressure of 190 Bar was applied to the cylinders. Per-second data from the load cell and sensors was collected during the experiments by a National Instruments PCIe-6363 data acquisition system and a LabVIEW-based data logger program. The results were compared with theoretical lifting capacity calculations.

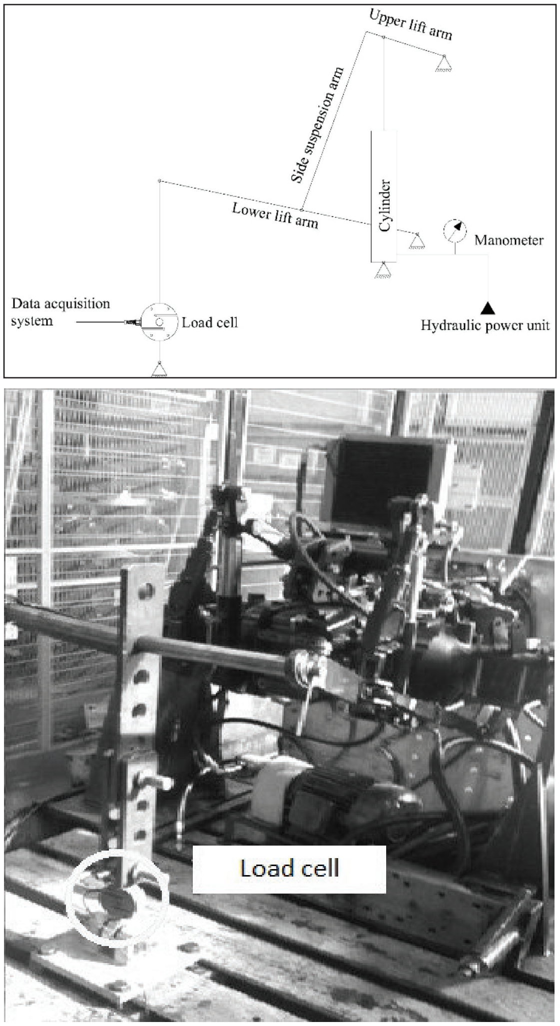

Figure 3- Lifting capacity test setup

Şekil 3- Kaldırma kapasitesi test düzeneği

During the cycle, effective loads for the TPH system were also established using static equilibrium calculations of the mechanism, which can be seen below. Loads placed on the TPH mechanism were calculated using the Equation 1-3, with the angles given in Figure 4.

$$F_s = P_{hyd.p.}.A \quad (1)$$

The equilibrium of AB rod and ED rod are expressed as follows.

$$\Sigma M_A = 0, \quad F_s\, Cos(\theta_p)\, l_{AP} - F_{bc}\, Cos\,(\theta_2)\, l_{AB} = 0 \quad (2)$$

$$\Sigma M_D = 0, \quad F_k\, Cos\,(\theta_4)\, l_{ED} - F_{bc}\, Cos(\theta_3)\, l_{CD} = 0 \quad (3)$$

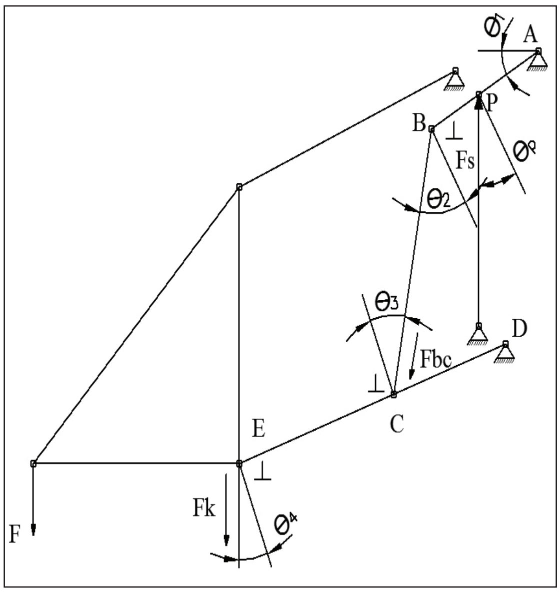

Figure 4- Three point hitch mechanism

Şekil 4- Üç nokta askı sistemi mekanizması

The parameters used in the calculations are given at Table 2. These parameters were determined from the experiments. The angles of the TPH rods were varied according to external rod angle (θ_1).

The lengths used in calculations are taken as $l_{AP} = 157.88$ mm, $l_{AB} = 275$ mm, $l_{CD} = 405$ mm, $l_{DE} = 900$ mm.

Table 2- Variables of TPH mechanism used in equilibrium calculations

Çizelge 2- Hesaplamalarında kullanılan TPH mekanizması değişkenleri

Hydraulic cylinder data	θ_1	θ_p	θ_2	θ_3	θ_4
	-30	33.09	45.9	36.34	20.44
	-20	20.10	35.19	29.59	14.40
	-10	9.04	24.85	22.82	7.97
160 mm diameter	0	1.23	14.82	16.14	1.33
150 bar pressure	10	10.84	5.06	9.55	5.41
	20	19.93	4.44	3.45	12.11
	30	28.58	13.68	2.37	18.69
	40	35.89	22.64	7.68	25.04
	50	44.94	31.28	12.38	31.07

2.3. Push-pull test

Hydraulic lift test apparatus was used for the push-pull test. The external bars of a hydraulic lift TPH mechanism were lifted up to maximum angle of 47° and fixed at that position. Pressure transducers were placed between the hydraulic pipe and the hydraulic lift for measuring pressure variations. The lower lifting arms of the TPH system were equipped with hydraulic lift apparatus at a 610 mm distance. A 1000 kg impulsive load was applied at the tip of TPH mechanism. The load was calculated using load cells at the tip of the piston, as seen in Figure 5. In these experiments, 40,000 cycles were found to be the optimum cycle count-where a bracket completed 40,000 cycles without any breakage, and passed load capacity tests and lifting-lowering tests. The push-pull test periods of 40,000 cycles were completed in approximately 55 hours (one cycle takes 5 seconds).

2.4. Lifting-lowering test

This was carried out in order to study the hydraulic lift system functions and to test the strength of the equipment, as seen in Figure 6. The test was also used for validation of the push-pull test at the final design stage of the brackets.

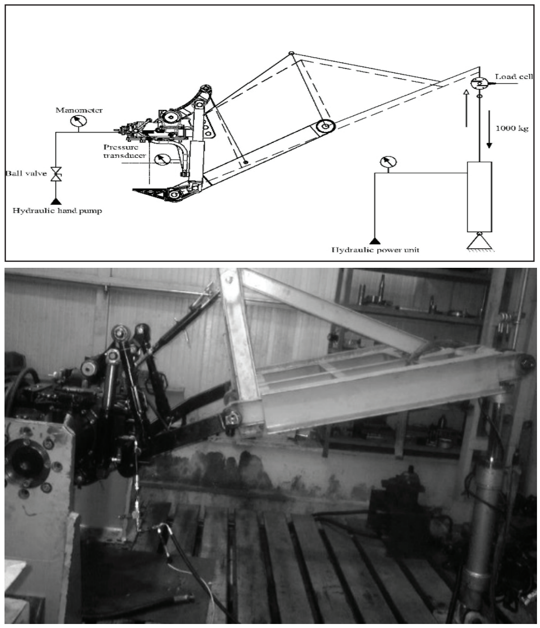

Figure 5- Push-pull test setup

Şekil 5- Çek bırak test düzeneği

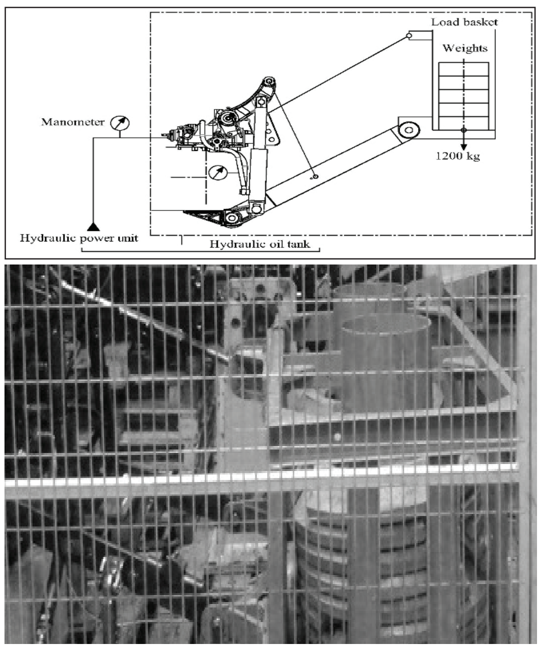

Figure 6- Lifting-lowering test setup

Şekil 6- İndir kaldır test düzeneği

A steel load basket connected to the three-point hitch system was used for the lifting-lowering tests. The load basket at the tip of TPH system was raised and lowered to the highest and lowest lifting points respectively. The gravity center of the basket was positioned 610 mm in height from lower arms. The basket was filled with 1200 kg of cast iron weights. One cycle was completed, with the lifting and lowering of the load basket taking 4 seconds. The position of the lower arms was determined by digital output proxy sensors. This process was repeated until a cycle count of 40,000 was achieved.

2.5. Finite element analysis of bracket

Finite element analysis was performed using ANSYS general finite element modeling software for study the stress and fatigue performance of brackets. The final design was achieved according to this analysis. The critical damage points of the brackets were determined under cyclic loading. Model mesh density was determined by convergence study of the bracket, as seen in Figure 7. The lifting-lowering test load procedure was used to convergence study. The finite element approximation of stress analysis was obtained for a number of 50,000 elements. The boundary conditions and applied mesh at the finite element model are given in Figure 8. Calculated loads at Equation 1-3 were applied to lower rod joints and cylinder rod joints, respectively and then the connections of the brackets were fixed.

At first, stage stress analyses of the brackets were performed. Nonlinear analysis with a multi-linear isotropic hardening material model was applied to the static stress analyses. Then strain life, based on the Coffin-Manson approach, was used to predict the fatigue life of the brackets. The stress of the critical breakage planes was evaluated according to the maximum principle stresses of the finite element analysis. The maximum principle stress results were used for the calculation of fatigue life because of the unyielding nature of GG materials.

Strain life-based, fatigue life-approximation was preferred because of the low cycle fatigue damages found in the experiments. In Table 3, the strain life parameters of the bracket materials GG25 and GG35 are given (ASM 1996).

The strain life was calculated using the Equation 4.

$$\frac{\Delta\varepsilon}{2} = \acute{\varepsilon}_f (2N_f)^c + \frac{\acute{\sigma}_f}{E} (2N_f)^{\mathrm{b}} \quad (4)$$

The results of the finite element analysis are given in the related test sections.

Table 3- Strain life parameters of bracket materials

Çizelge 3- Braket malzemelerinin gerilme ömür parametreleri

Material	$\sigma_y - \sigma_U$ *(MPa)*	*E (GPa)*	$\acute{\sigma}_f$*(MPa)*	*b*	$\acute{\varepsilon}_f$	*c*
GG25	215-260	90	353	-0.115	0.037	-0.582
GG35	345-438	134	696	-0.114	0.016	-0.383

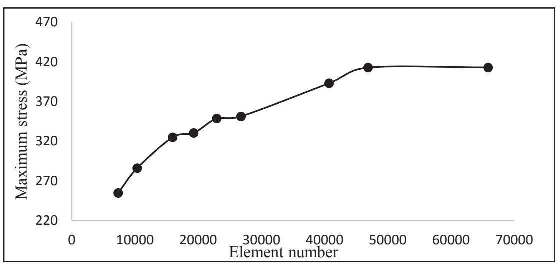

Figure 7- Convergence study of bracket finite element model

Şekil 7- Braket sonlu elemanlar modelinin yakınsama çalışması

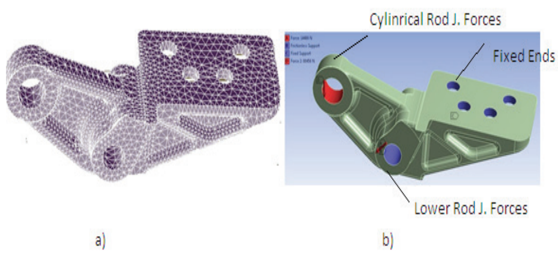

Figure 8- a, mesh detail at finite element model; b, boundary conditions

Şekil 8- a, sonlu elemanlar modeli mesh detayı; b, sınır koşulları

3. Results and Discussion

3.1. Lifting capacity test Results and bracket reaction forces for additional tests

The experimental results of the lifting capacity were compared with the static equilibrium calculations (Figure 9). According to the results of the Chi-Square test, there is no significant difference between the theoretical calculations and experimental test results (P= 0.056>0.05).

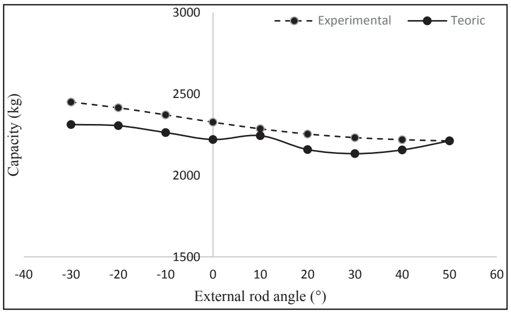

Figure 9- Comparison of experimental and theoretical calculation result of the lifting capacity

Şekil 9- Kaldırma kapasitesinin deneysel ve teorik hesaplama sonuçlarının karşılaştırılması

The TPH system cycle time was obtained from experimental tests. Cyclic loads applied to the TPH mechanism brackets, relative to the test loads, were calculated from equilibrium equations at Equation 1-3 as seen in Figure 10-12. It can be seen here that the theoretical maximum reaction force results of the lifting-lowering tests, obtained for the 0^0 external rod angle, were applicable at lower rod and cylinder rod joints. Reaction forces of the bracket joints on the push-pull test varied, according to the load placed at the tip of the TPH mechanism. Reaction forces acting on cylinder rod joints were found to be higher than on lower rod joints because of hydraulic cylinder reactions. In push-pull tests, an impulsive load factor of 1.3 was determined from test data and fatigue life calculations.

3.2. Additional test results

During the experimental push-pull tests, the breakage points of the brackets designed are shown in Figure 13. Breakage occurred after 11,218 cycles.

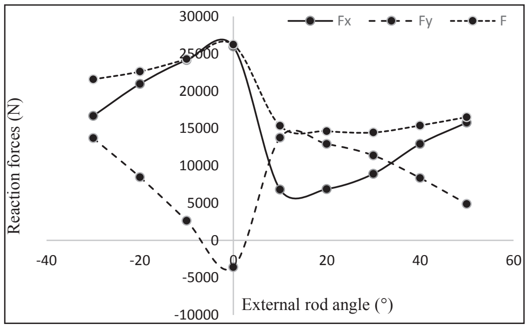

Figure 10- Reaction forces acting on brackets lower rod joint (LRJ) at the lifting-lowering test

Şekil 10- İndir kaldır testinde, braket alt rod bağlantılarında oluşan reaksiyon kuvvetleri

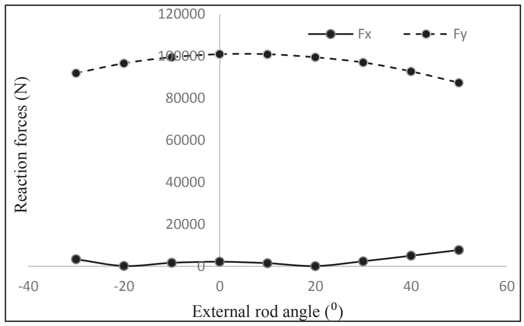

Figure 11- Reaction forces acting on brackets cylinder rod joint (CRJ) at the lifting-lowering tests

Şekil 11- İndir kaldır testlerindeki braket silindir rod bağlantılarında oluşan reaksiyon kuvvetleri

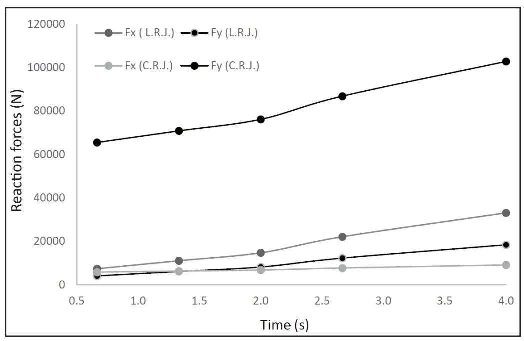

Figure 12- Reaction forces acting on brackets joints at the push-pull test

Şekil 12- Çek bırak testindeki braket bağlantılarında oluşan reaksiyon kuvvetleri

When the final design of the brackets was tested, it completed a full 40,000 cycles test period. During the lifting-lowering tests of the original bracket design, breakage occurred after 12,500 cycles. In comparison, the final bracket design completed a full 50,000 cycles test period without breakage. The difference between the two tests was the effect of impulsive load at the tip of the TPH mechanism. It was therefore assessed that push-pull tests give additional information about TPH mechanisms under impulsive load conditions.

Figure 13- Brackets breakage points at the push-pull tests

Şekil 13- Çek bırak testinde braketlerde oluşan kırılma noktaları

3.3. Finite element results

The analysis results of the failure zone, estimated by the finite element analysis, was compatible with the experimental results seen in fatigue life results, as seen in Figure 14-15. According to the test and analysis results, the breakage planes and maximum stress regions were determined, and the thickness of the connection surface of the brackets was increased from 12 mm to 19 mm. This modification increased

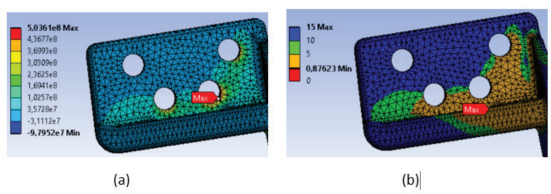

Figure 14- Finite element analysis results of first design brackets at push-pull tests; a, the maximum stresses at the bracket; b, the results of the safety factor

Şekil 14- İlk tasarım braketlerin çek-bırak testi için sonlu elemanlar analizi sonuçları; a, braketlerdeki maksimum gerilmeler; b, emniyet faktörü sonuçları

the weight of the bracket from 4,540 to 4,800 g. The breakage section of the bracket was redesigned, as seen in Figure 16, and the material of the brackets was changed from GG25 to GG35. As a result of the new design, the stress placed on the fracture plane was reduced from 503 MPa to 371 MPa for the push-pull tests, as seen in Figure 17. It can be seen that the maximum stress variation across the thickness was reduced by the new design of the brackets.

The analysis results of redesigned brackets at the lifting-lowering test are given in Figure 18. It can be seen here that maximum stresses were only 274 MPa and were reduced along the cross section at brackets. These results correlated with the push-pull tests. Similar maximum stress regions were found

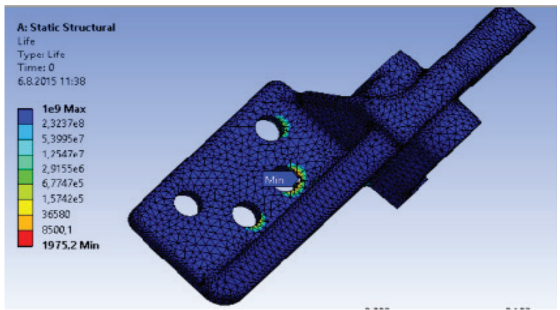

Figure 15- Brackets fatigue life results at finite element analysis

Şekil 15- Sonlu elemanlar analizindeki braket yorulma ömrü sonuçları

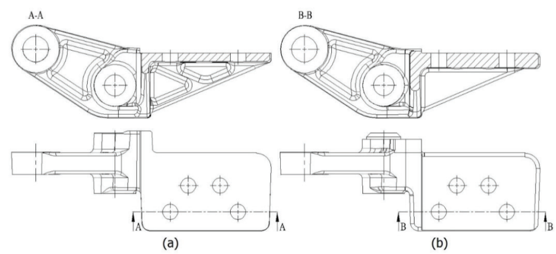

Figure 16- Bracket enhancements according to push-pull test results; a, first designed bracket; b, final design enhanced bracket

Şekil 16- Çek-bırak test sonuçlarına göre braket iyileştirmeleri; a, ilk tasarlanan braket; b, iyileştirilmiş son tasarlanan braket

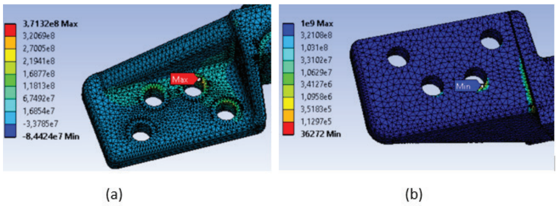

Figure 17- Finite element analysis results of optimized brackets at push-pull tests; a, the maximum stresses at the bracket; b, the results of the fatigue life

Şekil 17- Optimize edilen braketlerin çek-bırak testi için sonlu elemanlar analizi sonuçları; a, braketlerdeki maksimum gerilmeler; b, yorulma ömür sonuçları

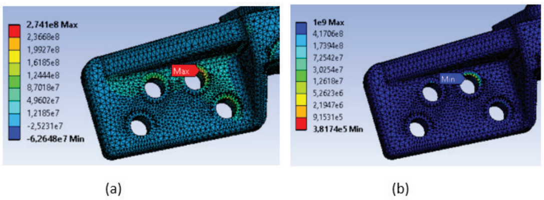

Figure 18- Finite element analysis results of optimized brackets at lifting-lowering tests; a, the maximum stresses at the bracket; b, the results of the fatigue life

Şekil 18- Optimize edilen braketlerin indir-kaldır testi için sonlu elemanlar analizi sonuçları; a, braketlerdeki maksimum gerilmeler; b, yorulma ömür sonuçları

for both tests. However, stress values obtained from the lifting-lowering tests were lower than those from the push-pull tests where the impulsive load effect on the TPH tip was a factor.

TPH mechanisms are generally tested according to Iso Standart Static Test methods. Some studies are focused on calculations and failure analysis of oscillations or dynamic loads acting on TPH mechanisms during drafting and implementations Otmianowski (1983), Sule et al (2007). There is not any test procedure is developed according to these dynamic loads. In our study newly developed tests on TPH mechanisms under impulsive dynamic loads are presented implemented to standard tests.

4. Conclusions

In the analysis, the boundary conditions were determined from system operating conditions. The maximum stress regions were examined comprehensively.

Improved safety factor results at the critical points were achieved by the material, shape and thickness modifications of the brackets. For this paper, three experimental test methods and finite element modeling were used in order to establish a methodology that could assess fatigue life in agricultural machine parts, namely the push-pull test, the lifting-lowering test and the lifting capacity test. In these experiments, it was seen that the push-pull test in particular was a very effective tool in determining the fatigue life of three point hitch system components. The results show that if any part of the TPH mechanism could complete 40,000 cycles of the push-pull test, it was also capable of passing the other tests. Therefore it was concluded that push-pull test data was implemented the standard tests for TPH mechanisms in order to assess impulsive loading.

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Abbreviations and Symbols	
TPH	Three point hitch
OIML R60	International organization of legal metrology recommendations 60
F_s	Hydraulic force
$P_{hyd,p}$	Hydraulic pressure
A	Piston cross section
F_k	Lifting load
l_{AP}, l_{AB}, l_{ED}, l_{CD}	Length of rod parts
θ_1, θ_2, θ_3, θ_4, θ_p,	TPH mechanism rod angles
F_x, F_y, F	Reaction forces
$\acute{\varepsilon}_f$	Fatigue ductility coefficient
σ_y	Yield strength
σ_u	Ultimate
σ_f	Fatigue strength coefficient
N_f	Cycles to failure
b	Fatigue strength exponent
c	Fatigue ductility exponent

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