

Research Article

Comparative Prediction of the Fatigue Life of Traction Arms on the New Holland TT75 Tractor in Chad: Miner, Chaboche-Lemaitre and Tikri Approaches

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Abstract

This study presents a comparative fatigue-life assessment of two traction-arm designs used on the New Holland TT75 tractor operating under the demanding agricultural conditions found in Chad: the current production arm (BA) and a geometrically optimized alternative (BP), both manufactured from S235 steel. Finite element simulations performed in ABAQUS/CAE 2017 were used to determine the maximum and equivalent stresses generated in each arm under applied pressures ranging from 90 to 125 MPa, the equivalent stresses being obtained through the Goodman mean-stress correction. Fatigue life was then predicted using three complementary damage models of increasing complexity: Miner's linear cumulative-damage law, the Tikri model, and the Chaboche-Lemaitre continuous damage model, applied first to constant-amplitude loading and then to loading sequences that include periodic overload cycles (one overload every 50 cycles, with overload ratios $R = 1.2, 1.4$ and 1.6) representative of the irregular soil, stones and roots encountered during ploughing and traction work in Chadian fields. Reference life values were obtained by digitizing a published Wohler S-N curve for S235JR steel. Across every applied pressure and every overload scenario, the optimized arm (BP) developed systematically lower equivalent stresses and achieved a longer predicted fatigue life than the current arm (BA), regardless of the damage model used. Among the three models, the Chaboche-Lemaitre formulation consistently gave the most accurate and most conservative life estimates, capturing the progressive, sequence-dependent nature of damage accumulation more realistically than Miner's rule, which tended toward optimistic, non-conservative predictions. These findings confirm that modest geometric optimization of the traction arm can meaningfully extend component durability under the harsh tropical operating conditions typical of Chadian agriculture, and they point to the Chaboche-Lemaitre model as the preferred tool for life prediction in this application, pending confirmation through physical fatigue testing.

Keywords

Fatigue, Traction Arms, Finite Elements, Miner, Chaboche-Lemaitre, Tikri

1. Introduction

Traction arms are among the most heavily loaded parts of a farm tractor. During ploughing and the towing of heavy

implements they carry cyclic loads on which random overloads are superimposed, and this combination governs their

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Received: 17 September 2026; Accepted: 27 September 2026; Published: 20 October 2026



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fatigue behaviour [1, 2]. In Chad, field conditions are particularly harsh: stones, roots and uneven soil, together with the tropical climate, speed up fatigue damage, cause premature failures and cost farmers valuable working time [3]. In an earlier study we showed, through two- and three-dimensional finite element analysis, that the traction arm of the New Holland TT75 tractor in service in Chad has marked stress concentrations and that a change in its geometry could reduce them [3].

The present work extends that analysis to fatigue life. It compares the current arm (BA) with a geometrically optimized arm (BP), both made of S235 steel, and assesses how well three damage models predict their life under uniaxial loading, with and without overloads: the linear rule of Palmgren and Miner [4, 5], the Tikri model [6] and the Chaboche-Lemaitre continuous damage model [7, 8]. Stresses are computed with ABAQUS/CAE 2017. The reference (“experimental”) lives come only from the digitization of a published S-N curve [9], not from independent tests, and they should be read with that limit in mind. The overall approach is one of structural optimization suited to the constraints of tropical farming.

2. Literature Review

Fatigue of metals under cyclic loading has been studied since Wöhler’s tests on railway axles, which introduced the stress-life (S-N) curve [10]. Basquin later gave the high-cycle part of this curve its power-law form [11], and Goodman’s correction is still a standard way of accounting for mean stress when the loading is not fully reversed [12, 13]. For cumulative damage, the linear rule first proposed by Palmgren for ball bearings [4] and generalized by Miner [5] remains the most widely used in engineering because it is simple. It ignores, however, the order in which load levels are applied and the interaction between them, so it can be seriously wrong under variable-amplitude loading [14-16].

Several nonlinear approaches were developed to address these shortcomings. Manson and Halford proposed the double linear damage rule and the damage curve approach [17]. Chaboche and Lesne formulated a nonlinear continuous damage model in which the damage rate depends on both the current damage state and the stress level [7], within the continuum damage mechanics framework set out by Lemaitre and Chaboche [8]. The Tikri model, developed for spot-welded steels subjected to overloads, also takes the loading history into account [6]. Recent studies continue in this direction, with nonlinear accumulation models that include loading-sequence effects [18] and multi-level models that correct the Manson-

Halford formulation for load interaction [19]. Overloads need particular attention. A single high peak can slow down subsequent crack growth, as described by Wheeler’s model [20], but it also uses up part of the life, and the net effect depends on the load spectrum. Gassner’s variable-amplitude tests introduced the spectrum-based fatigue curves that we use later to present our results [21].

In agricultural and off-road machinery, finite element analysis is now the usual tool for locating critical zones and predicting fatigue life. It has been applied to tractor frames [22], transmission gears [23], the weak points of an electrically driven multi-purpose tractor [24], wheel discs under cornering loads [25] and excavator turntables [26]. Coupling multibody dynamics with finite element analysis gives more realistic load histories [27], and digital twins are beginning to be used to predict damage and plan maintenance [2]. A recent review of fatigue evaluation in agricultural machinery nevertheless points to the lack of common evaluation standards and to the difficulty of reproducing real field loads in the laboratory [1].

The fatigue properties of mild structural steels such as S235JR have been characterized in several studies, including high-cycle design curves for offshore applications [28] and tests on corroded and uncorroded specimens [29]. Our previous work on the drawbars of New Holland TT75 tractors in service in Chad showed that their geometry needs to be optimized for the severe local conditions [3]. The present study builds directly on those results.

3. Materials and Methods

Two geometric models were considered: the existing arm (BA) of the New Holland TT75 tractor and a proposed optimized arm (BP), whose modified geometry is designed to reduce stress concentrations and distribute forces more evenly [3].

Both arms are made of S235 steel, equivalent to S235JR ($E = 200$ GPa, $\nu = 0.3$) [28, 29].

Modeling: the finite element simulations were run in ABAQUS/CAE 2017 with reduced-integration eight-node brick elements (C3D8R). An approximate global mesh size of 0.2 was used for the structure.

3.1. Wohler S-N Curve for S235JR Steel

The applied pressures, from 90 to 125 MPa, were chosen so that the equivalent stresses fall within the finite-life region of the S-N curve for S235JR steel, which covers a limited stress range. The load ratio is $R = 0$. Figure 1 shows the reference S-N curve for S235JR steel [9], digitized with PlotDigitizer 2.6.8.

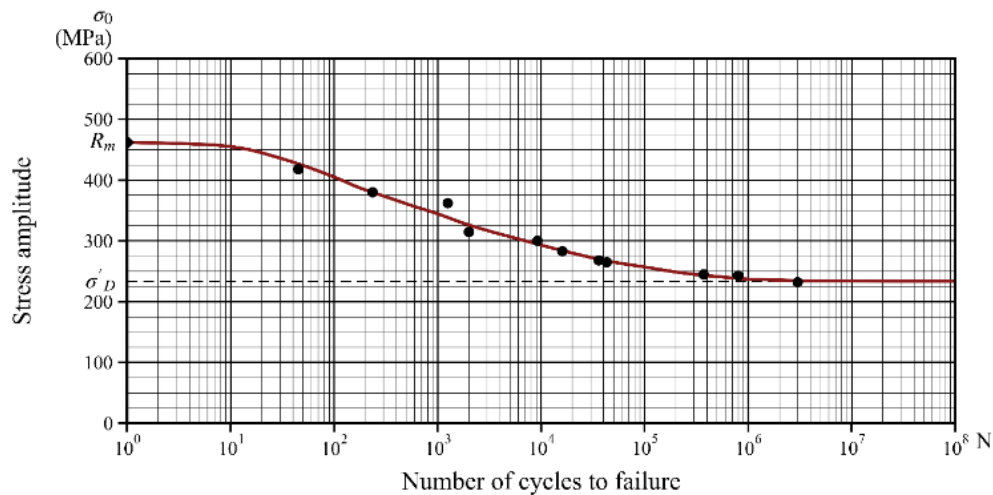


Figure 1. Wöhler curve for S235JR steel [9].

3.2. Calculation of Equivalent Stresses Under Different Applied Pressures

Table 1 lists the maximum stresses ($S_{11 \max}$) obtained by simulation for each traction arm, together with the equivalent stresses (σ_0) calculated with the Goodman mean-stress correction [12, 13]. The Goodman equation is:

$$\sigma_0 = \frac{\sigma_a}{\left(1 - \frac{\sigma_m}{R_m}\right)}$$

R_m : the maximum traction stress indicated by the SN curve for S235JR steel;

σ_m : the mean stress (in MPa)

σ_a : the stress amplitude (in MPa)

Table 1. Equivalent stresses under different applied pressures for the actual arm (BA) and the proposed arm (BP).

Pressure applied in MPa	Constraints of the actual Arm (BA)		Constraints of the Proposed Arm (BP)	
	$S_{11 \max}$ (MPa)	Σ_0 (MPa)	$S_{11 \max}$ (MPa)	σ_0 (MPa)
90	314,7	239,16	296	218,20
95	332,2	259,97	312,4	236,51
100	349,7	282,07	328,9	255,95
105	367,2	305,56	345,3	276,38
110	384,7	330,58	361,8	298,15
115	402,2	357,30	378,2	321,10
120	419,7	385,89	394,7	345,63
125	437,1	416,37	411,1	371,60

$\sigma_{\max}$ (or $S_{11 \max}$): the maximum stress in MPa.

3.3. Data Extracted by PlotDigitizer 2.6.8 from the Wöhler S-N Curve

Table 2 presents the equivalent stress values extracted from

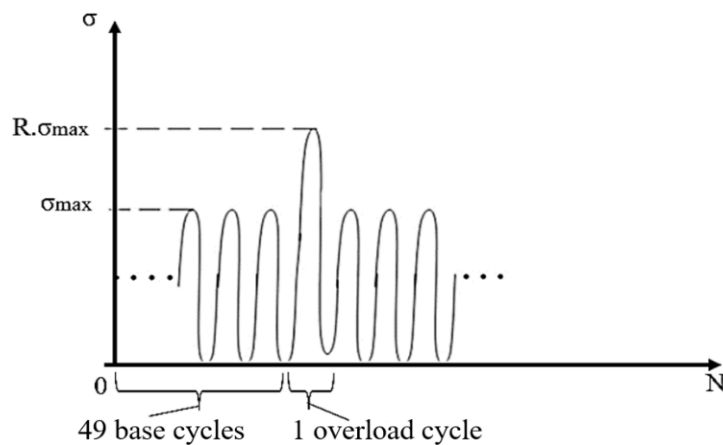
the experimental Wöhler S-N curve, which are approximated by the equivalent stress values calculated from the simulation results for the two traction arms under the same applied pressure; this table also includes the number of experimental cycles obtained through the extraction.

Table 2. Stresses and life cycles extracted from the Wöhler S-N curve for S235JR steel using PlotDigitizer 2.6.8 software.

Applied pressure in MPa	Equivalent stress in MPa (BA)	Experimental Life Cycle of BA (extract)	Equivalent stress in MPa (BP)	Experimental Life Cycle of BP (extract)	$\frac{N_{BA}}{N_{BP}}$ %
90	239	132756	218	2262300	5,87
95	260	35229	237	132756	26,54
100	282	13174	256	43638	30,19
105	306	4575	276	14169	32,29
110	331	1475	298	6140	24,02
115	357	614	321	2240	27,41
120	386	222	345	896	24,78
125	416	74	372	296	25

3.4. Overload Adopted Under Repeated Traction for Life Cycle Prediction

To reproduce field conditions in Chad, an overload cycle was inserted every 50 cycles (49 nominal cycles followed by one overload cycle, with overload ratios $R = 1.2, 1.4$ and 1.6), as shown in Figure 2. Periodic overloads of this kind change the way damage accumulates compared with constant-amplitude loading [18, 20].

**Figure 2.** Cycle variation with an overload cycle every 50 cycles.

3.5. Model Equations

Three models were applied: Miner's law [4, 5], the Tikri model [6] and the Chaboche-Lemaitre model [7, 8].

1) Miner's law

$$N_r = 10 \cdot \left(\frac{\sigma_0}{0,9R_m} \right)^{\frac{\log 5 \cdot 10^6 - 1}{\log \frac{\sigma_D}{0,9R_m}}}$$

N_r : The number of life cycles required for a crack to initiate.

2) Tikri Model

$$N_{rj} = \frac{1}{aM_0^{-0,5\beta}(1+\beta)} \cdot \left(\frac{\sigma_{\max}}{\sqrt{1 - \frac{\sigma_{\max}}{2AR_m}}} \right)^{-\beta}$$

N_{rj} : Estimated service life at the start of crack initiation.

3) Chaboche-Lemaitre model

$$N_{ij} = \frac{1}{aM_0^{-\beta}(1+\beta)K_j} \left(\frac{1 - b\sigma_{mj}}{\sigma_{aj}} \right)^{\beta}$$

N_{fi} is the number of cycles required for a crack to initiate in the material under a constant-amplitude load defined by (σ_{aj} , σ_{mj}).

The Relative Prediction Errors (ERP) used to evaluate the performance of each model are calculated as follows:

$$ERP(\%) = \frac{N_{exp} - N_{loi}}{N_{exp}} \times 100$$

N_{exp} : Number of life cycles obtained by extracting the “experimental S235JR” curve;

N_{loi} : Life cycle number calculated by the law or by the model.

- 4) If $ERP(\%) > 0$: The life cycle obtained through extraction exceeds that predicted by the law or model, indicating that the law or model is considered more safe.

- 5) If $ERP(\%) < 0$: The life cycle obtained through extraction is shorter than that specified by the law or the model, suggesting that the law or the model is not safe.

This methodology follows current practice in fatigue assessment of agricultural equipment [1, 27] and in the comparison of cumulative damage models [16].

4. Results and Discussion

4.1. Prediction Without Overload

Tables 3 and 4 give the calculated service life of the current arm (BA) and the proposed arm (BP) under nominal load. Figures 3 to 6 show the corresponding Gassner-type S-N curves [21] and histograms of the relative prediction errors (ERP%).

Table 3. Life cycle of the BA as determined by Miner’s law, the Tikri model, and the Chaboche model (without overload).

Equivalent stress	(MPa) Number of Experimental Life Cycles (extract)	Miner's Law		Tikri Model		Chaboche Model	
		Life Cycle	ERP (%)	Life Cycle	ERP (%)	Life Cycle	ERP (%)
239	132756	744627	-461	137783	-3,78	111520	15,99
260	35229	175910	-399	45576	-29,37	33532	4,81
282	13174	43743	-232	15694	-19,12	10522	20,13
306	4575	10794	-136	5378	-17,55	3281	28,28
331	1475	2811	-90,55	1922	-30,30	1070	27,45
357	614	769	-25,31	714	-16,28	364	40,71
386	222	202	9,07	257	-15,76	119	46,39
416	74	56	24,33	97	-31,08	41	44,59

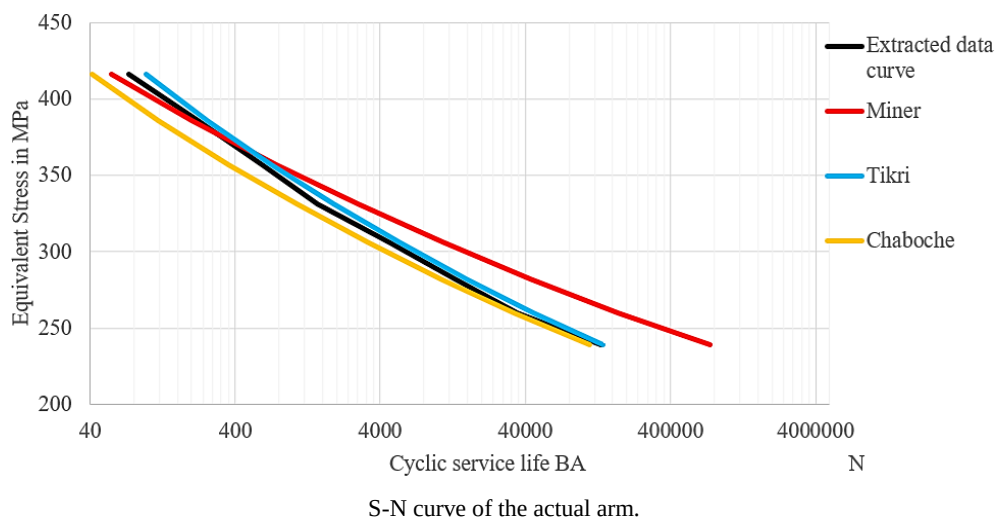


Figure 3. GASSNER S-N curve for the actual arm model.

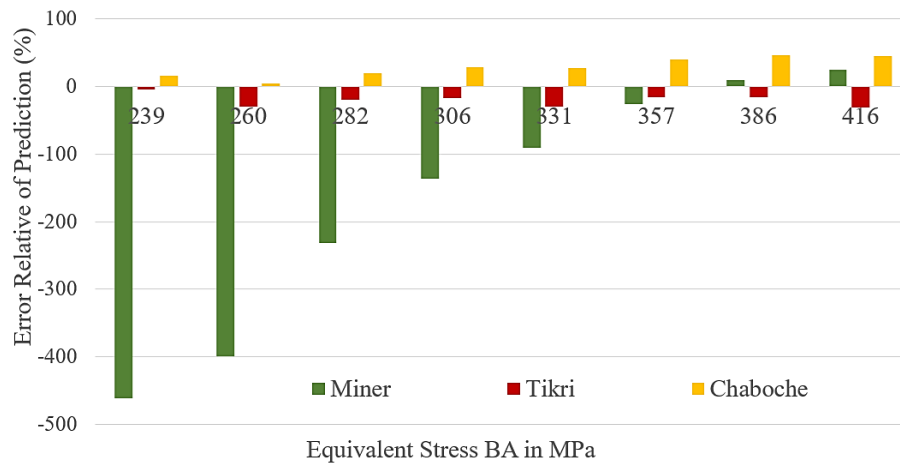
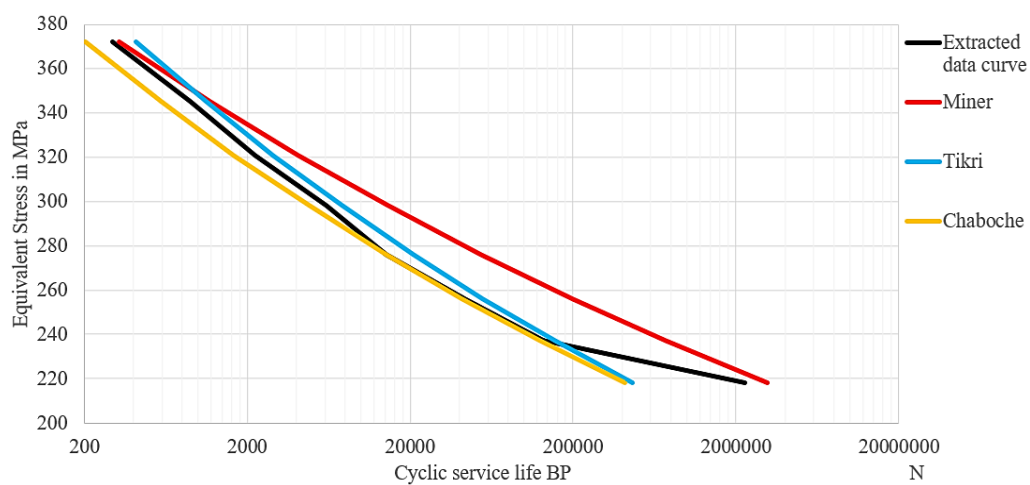


Figure 4. Histogram of ERP (%) as a function of the applied pressure of the actual arm.

Table 4. Life cycle of the BP as determined by Miner's law, the Tikri model, and the Chaboche model (without overload).

Equivalent stress	(MPa) Number of Experimental Life Cycles (extract)	Miner's Law		Tikri Model		Chaboche Model	
		Life Cycle	ERP (%)	Life Cycle	ERP (%)	Life Cycle	ERP (%)
218	2262300	3099342	-36,99	461648	79,59	414264	81,68
237	132756	740425	-458,7	153871	-15,90	125727	5,29
256	43638	197546	-352,7	55867	-28,02	41835	4,13
276	14169	54444	-284,24	20811	-46,87	14302	-0,93
298	6140	14632	-138,30	7610	-23,94	4788	22,01
321	2240	4093	-82,74	2873	-28,25	1657	26,02
345	896	1190	-32,82	1118	-24,77	592	33,92
372	296	327	-10,57	417	-40,87	202	31,75



S-N curve for the proposed arm.

Figure 5. GASSNER S-N curve for the proposed arm model.

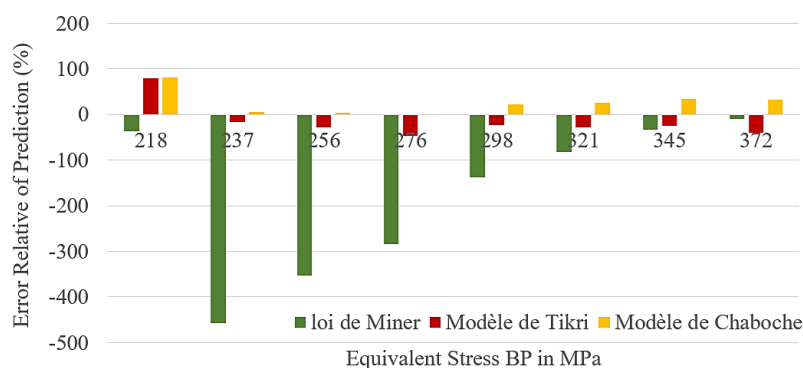
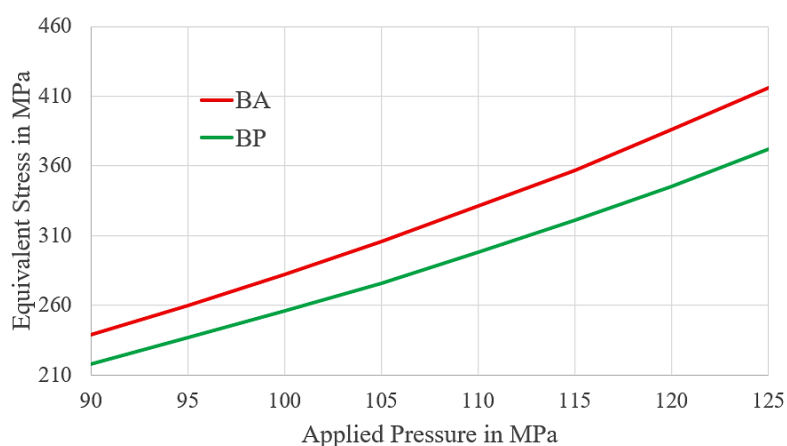


Figure 6. Histogram of ERP (%) as a function of the equivalent stress of the proposed arm.

The proposed arm (BP) has a markedly longer service life because its equivalent stresses are lower. The Chaboche-Le-maitre model [7, 8] gave the most conservative and most accurate predictions, while Miner's rule was often optimistic [5, 14]. This agrees with published comparisons between linear and nonlinear damage models [16, 19].

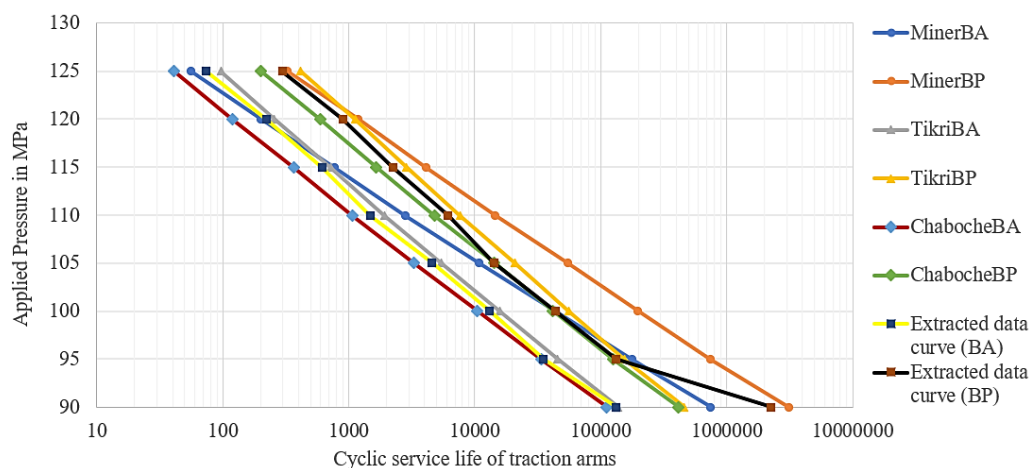
4.2. Comparison with and Without Overload

Figures 7 and 8 compare stresses and service lives without overload, while Figures 9 through 14 present the results with overloads ($R = 1.2, 1.4$, and 1.6).



Equivalent stress curve as a function of applied pressure—without overload.

Figure 7. Equivalent stress as a function of applied pressure for BA and BP (without overload).



S-N curve of applied pressure as a function of life cycle—without overload.

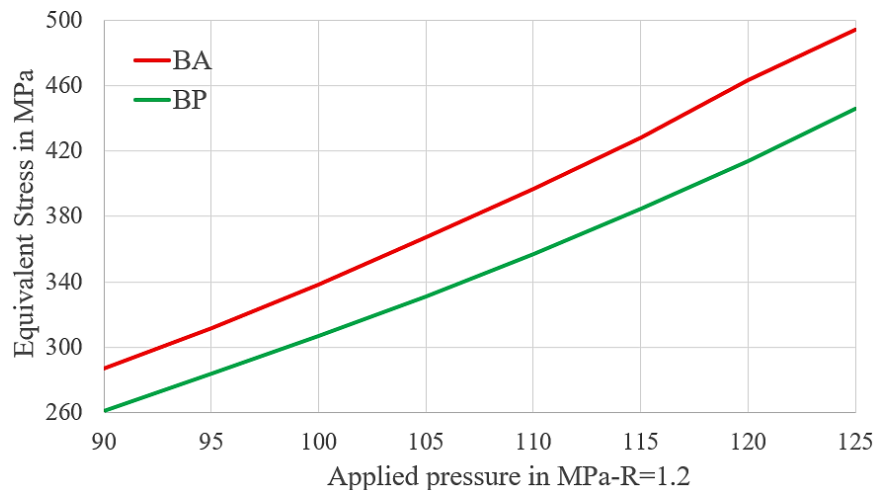
Figure 8. Applied pressure as a function of the BA and BP service life—without overload.

4.3. Comparison of Stresses and Service Life of Arms-With Overload

Predicting the life of the traction arms under overload

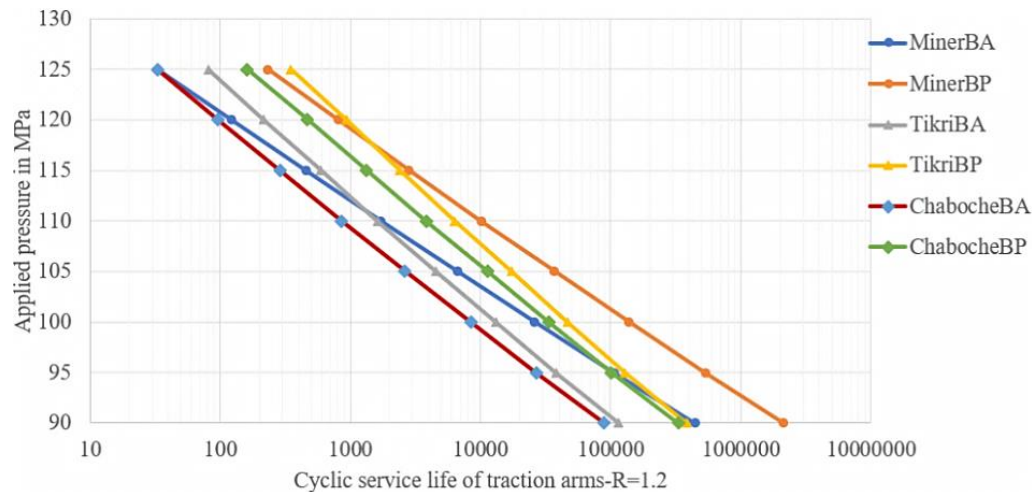
means estimating their life when they carry loads higher than those they were designed for. This happens under intensive or abnormal use and raises the risk of premature failure by fatigue or other forms of damage [13, 20].

4.3.1. Comparison of Stresses and Life Cycles of Arms-with $R=1.2$



Equivalent stress curve as a function of applied pressure-R=1.2.

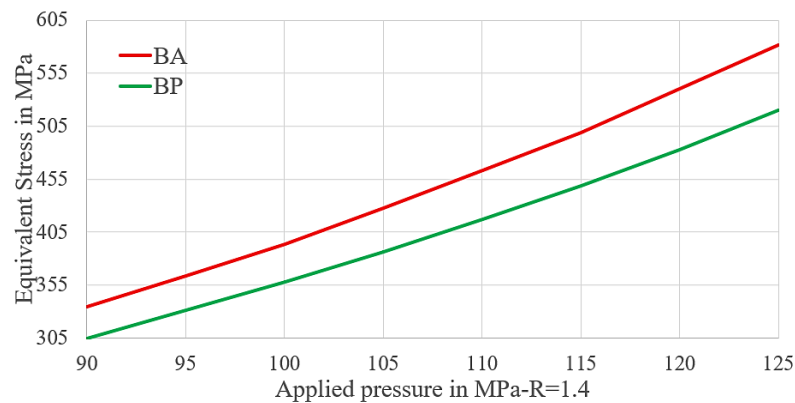
Figure 9. Equivalent stress as a function of applied pressure for BA and BP-R=1.2.



S-N curve of applied pressure as a function of life cycle-with $R = 1.2$.

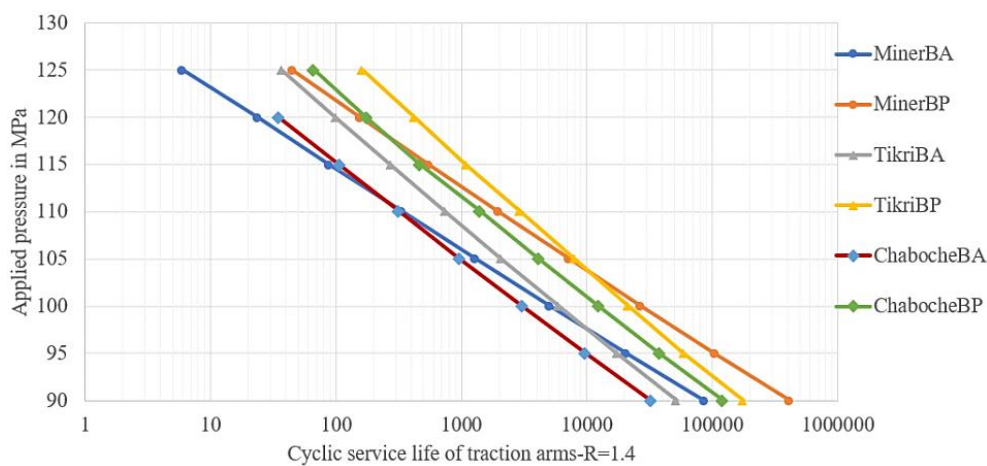
Figure 10. Applied pressure as a function of the life cycle of the BA and BP-R=1.4.

4.3.2. Comparison of Stresses and Life Cycles of Arms-with $R = 1.4$



Equivalent stress curve as a function of applied pressure- $R=1.4$.

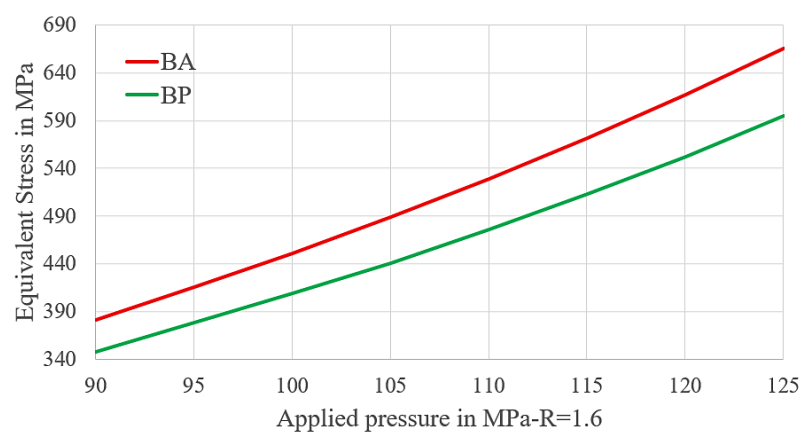
Figure 11. Equivalent stress as a function of applied pressure for BA and BP- $R=1.4$.



S-N curve of applied pressure as a function of life cycle-with $R = 1.4$.

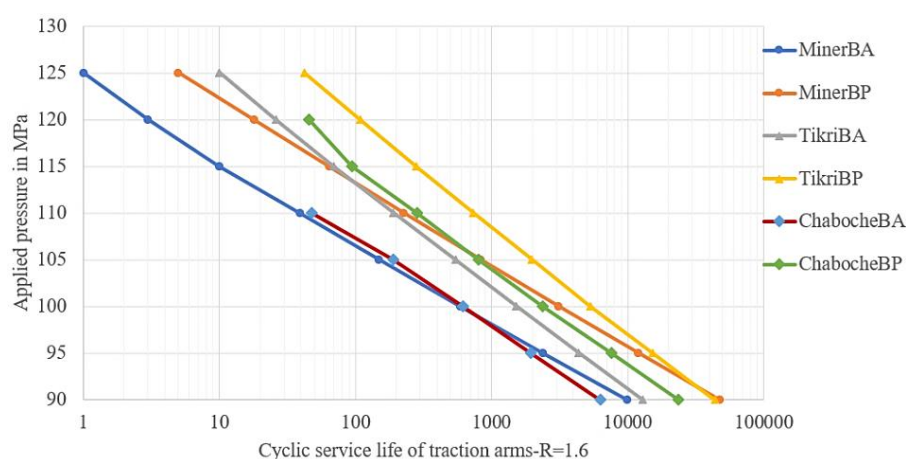
Figure 12. Applied pressure as a function of the life cycle of the BA and BP- $R=1.4$.

4.3.3. Comparison of Stresses and Life Cycles of Arms-with $R = 1.6$



Equivalent stress curve as a function of applied pressure- $R=1.6$.

Figure 13. Equivalent stress as a function of applied pressure for BA and BP- $R=1.6$.



S-N curve of applied pressure as a function of life cycle-with R = 1.6.

Figure 14. Applied pressure as a function of the life cycle of the BA and BP-R=1,6.

The proposed arm (BP) performs better in every configuration, and the Chaboche-Lemaitre model again gives the most reliable predictions [7].

These results confirm and extend those of Djondang et al. [3], and they support the benefit of geometric optimization reported for other agricultural components [22-25].

General analysis: the advantage of the proposed arm comes from a better stress distribution, which reduces local stress concentrations and delays crack initiation. This is consistent with recent work on design optimization and damage prediction for agricultural equipment [1, 2, 27]. Geometric optimization appears to be an effective and inexpensive way to improve durability in tropical environments.

5 Conclusions

This comparative study shows that the proposed arm (BP) resists fatigue better than the current arm (BA). The geometric optimization lowers stresses appreciably and increases service life, including under overloads. Of the three damage models, the Chaboche-Lemaitre model stood out for its accuracy and its conservative predictions. These results confirm and extend the earlier work of Djondang et al. [3] and show the value of combining finite element simulation with advanced damage models when designing agricultural equipment for use in Chad. Fatigue tests on real arms, under loads measured in the field, are still needed to validate these numerical predictions [1].

Author Contributions

Djondang Sindang: Conceptualization, Data curation, Formal Analysis, Funding acquisition, Investigation, Methodology, Project administration, Resources, Software, Supervision, Validation, Visualization, Writing – original draft, Writing – review & editing

Kinet Ouinra: Funding acquisition, Methodology, Validation, Writing – review & editing

Tchoudang Sehoret: Funding acquisition, Methodology, Validation, Writing – review & editing

Baimi Badjoua: Software, Validation, Writing – review & editing

Conflicts of Interest

The authors declare no conflicts of interest.

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