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Design Optimisation of Four Wheel Drive Tractor Front Axle Housing to Address Field Failure

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ABSTRACT

Four wheel drive tractors are used forhigh torque demand applications in field and subjected to severe load conditions.Front axle is one of the most critical aggregate of the Tractor. Design of front axle is more important and critical in application stand point. Specific applications like front bucket, bund preparation and paddy field demand very rigid axle design. Front axle endures the most in tractor aggregate.Front axle housing has failed in field from housing shoulder location in initial proto test axles. The objective of the study is to analyse and optimise the design of the axle housing.Comparethe modified design with old design for improvement. Comparison study between hand calculations, FEA and test results. Improvement of the shoulder of the axle is a major area to address the failure of the shoulder in field testing. The housing shoulder required attention during design for a fail safe operation in service. The various design formulas of mechanical elements of shafts and beam are used for design and analysis of the shoulder.

Keywords— **MFWD- Mechanical four wheel drive,4WD- Four wheel drive, Stress concentration factor, Axle housing, structural strength, Von misses stress, Goodman equation**

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I. INTRODUCTION

During field testing of the front axle there was a shoulder failure reported at the lower king pin bearing area. As per the analysis the design was acceptable for the given load goals and no metallurgical non-conformance was reported. This led to a data acquisition activity on the front axle which reported more severe loading than initially specified.

The main purpose of the project is to analyze the existing design of the tractor front axle housing for service load conditions and redesign the axle housingwith the updated load conditions. The existing geometry of the front axle is modified to the optimum size which suits for functional life requirements. In this analysis, the geometry of the front axle is modified and a new design is proposed. The objective of this study is to improve the existing design with a higher cross section at the spindle of the axle housing resulting in better performance of the tractor. Finite element simulation is carried out for the existing front axle. The critical location identified and redesigned to ensure life goals are met for the structural components.

In this analysis, the spindle is a critical structural member and the complete load passes through the spindle. In this paper we will design the axle housing. The major advantage in using MFWD tractor is that it can deliver 10 - 15 % more power for the same fuel consumption.

I. METHODOLOGY

The load on the axle is transferred through the lower king pin bearing. The vertical load and the tractive effort put a combined load on the axle of reversible nature. The stress in the shoulder reverses from tensile to compression in certain sections with varying severity depending on the track of the vehicle and the tire radius. The spindle and the kingpin bearing bearings support the complete load of the tractor. If the load on the spindle is within the range of 66% of the yield of the material the housing will give infinite life. In case the stress values in the spindle exceeds the 66% limit the life of the component needs to be calculated for the life and determined if sufficient life is available on the part. Similarly when the stress exceeds its ultimate tensile strength the housing will fracture.

The process begins with preliminary analysis of failed part followed by baseline design analysis, new design of housing, stress /strain analysis, FEA approach, Experimental data collection and validation of the of the front axle housing

A. Process

Failure analysis
Analytical approach / Calculations
FEA
Field Test
Comparison and improvement

II. Failure analysis

A. Material for axle housing,

Material specification ductile iron, grade SAE D5506
Mechanical Properties-
Hardness: - 187-241 BHN(MS 75 B),
Tensile strength: - 550 N/mm^2
Yield Strength: - 380N/mm^2
Elongation: 6%

TABLE I
INSPECTION REPORT

#	CHARACTERISTIC	SPECIFICATION	OBSERVED
1	GRADE - 550/6 or MS-75B Chemistry	C: 3.20 - 4.10 %	C:3.8 %
		Mn :0.10 -1.00%	Mn : 0.6%
		Si : 1.80 - 3.00%	Si : 2.55%
		S : 0.005 - 0.035%	S : 0.004%
		P : 0.015 - 0.10%	P : 0.025%
2	Surf Hardness	187 - 255 BHN	210 BHN
3	Tensile Strength	> 552 N/mm^2	637 N/mm^2
	Yield Strength	> 379 N/mm^2	430 N/mm^2
	Elongation %	> 6 %	> 8.50 %
4	Matrix Structure	Pearlite: 80 +/-15%	57.72%
		Ferrite : 20 +/- 15%	20.62%
		Graphite %	21.66%
		Nodularity > 85 %	88.30%
5	Internal Soundness	Radiography level - II accept	ok

B. Component Geometry

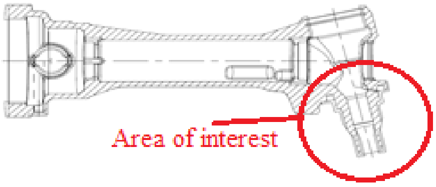

Fig.1 Housing geometry

Fig .2Failed housing

Failure observed at neck of the shoulder where cross section of the housing changes. (Ref Fig. 1). It is observed that low cycle fatigue phenomenon of the failure.

C. Application / loading cycle

In a front axle the forces are applied through the rear and front Trunnion and transmitted through the housing to the kingpin bearings and to the knuckles.These forces give rise reversible stress in the shoulder spindle which leads to fatigue failure. The spindle is a critical structural member and the complete load passes through the spindle.

The existing geometry of the front axle is modified to the optimum size which suits for functional life requirements. In this analysis, the geometry of the front axle is modified and a new design is proposed. The objective of this study is to improve the existing design with a higher cross section at the spindle of the axle housing resulting in better performance of the tractor.

D. Analytical calculations

The section modulus is directly related to the strength of a corresponding housing. It is expressed in units of volume m3, mm3. For design, the Elastic section modulus is used, applying up to the Yield point for most metals and other common materials. The elastic modulus is denoted by Z. Now Section modulus is calculated by using following formula,

TABLE II
MECHANICAL PROPERTIES OF HOUSING

PARAMETER	BEFORE	AFTER
Section ID	26 mm	26 mm
Section OD	35 N/mm²	40 N/mm²
Step OD	41 mm	46 mm

Transition radius	3 mm	3 mm
Stress concentration factor	1.68 N	1.68 N
Young modulus	210000 N/mm²	210000 N/mm²
Yield of material	380 N/mm²	380 N/mm²

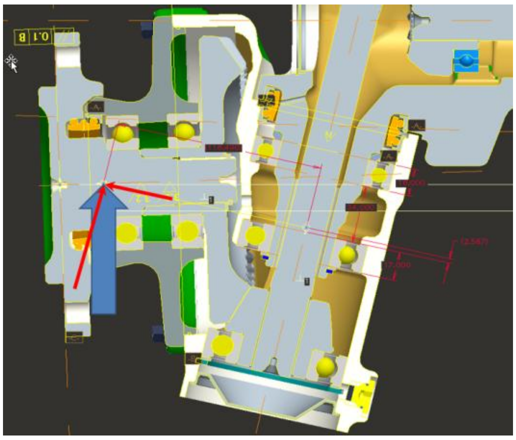

Fig.3 Reaction diagram

1) Vertical load is divided into the resulting components at tractor tire center line.

By drawing Free body diagram, Shear force diagram and Bending moment diagram as like below Free body diagram and calculating the forces,

$$\text{Section modulus (Z)} = \frac{\text{Bending moment (M)}}{\text{Ultimate tensile strength of the material (F)}}$$

Where, Z= Section modulus

M = Bending Moment

F = Ultimate tensile strength of material

E. OriginalDesign + oldLoads (18KN) on axle

For the area of the spindle we can find out using,

$$A= \frac{\pi}{4} \text{ x } (D^2 - d^2)$$
$$= \frac{\pi}{4} \text{ x } (35^2-26^2)$$
$$= 431.18 \text{ mm}^2$$

Moment of inertia (I)

$$I= \frac{\pi}{64} (D^4 - d^4)$$
$$= \frac{\pi}{64} (35^4-26^4)$$
$$= 51204.03 \text{ mm}^4$$

Section Modulus at lower Bearing $= \frac{I}{Y}$

$$= \frac{\pi(D^{\wedge}4 - d^{\wedge}4)}{32D}$$
$$= \frac{\pi(35^{\wedge}4 - 26^{\wedge}4)}{32*35}$$
$$= 2925.245 \text{ mm}^3$$

F. Original design + Revised load goals (22KN) on axle

For the area of the spindle we can find out using,

$$A= \frac{\pi}{4} \text{ x } (D^2 - d^2)$$
$$= \frac{\pi}{4} \text{ x } (35^2-26^2)$$
$$= 431.18 \text{ mm}^2$$

G. New Design + Revised loads (22KN) on axle

For the area of the spindle we can find out using,

$$A= \frac{\pi}{4} \text{ x } (D^2 - d^2)$$
$$= \frac{\pi}{4} \text{ x } (40^2-26^2)$$
$$= 725.34 \text{ mm}^2$$

Moment of inertia (I)

$$I= \frac{\pi}{64} (D^4 - d^4)$$
$$= \frac{\pi}{64} (40^4-26^4)$$
$$= 103179.6 \text{ mm}^4$$

Section Modulus at lower Bearing $= \frac{I}{Y}$

$$= \frac{\pi(D^{\wedge}4 - d^{\wedge}4)}{32D}$$
$$= \frac{\pi(40^{\wedge}4 - 26^{\wedge}4)}{32*35}$$
$$= 5158.98 \text{ mm}^3$$

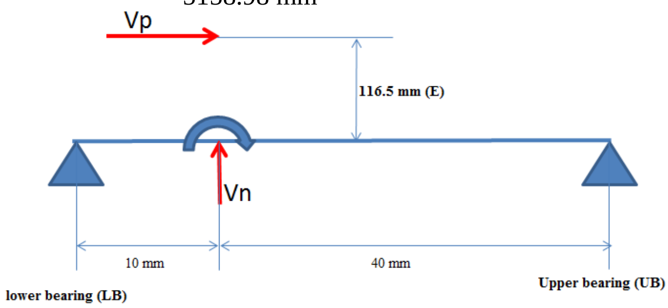

Fig. 4 Stress amplitude Free Body diagram

H. Endurance strength &Stress Concentration factor

Laboratory endurance strength (Se') of the materials obtain from S-N diagram are therefore corrected for actual conditions by using correction factors,

Se = Ka x Kb x Kc x Kd x Kt x Kf x Se'

Where,

Ka = Surface Correction factor

Kb = Size Correction factor

Kc = Loading factor

Kd = Temperature Correction factor

Kt = Stress concentration Correction factor

Kf = Miscellaneous Correction factor

Se' = Endurance Strength of material specimen under laboratory condition

Se = Endurance Strength of material

Stress concentration factor

Kt= Normal Stress

Kts = For Shear Stresses

Kt= $\sigma max/\sigma 0$

Kts=$\tau max/\tau 0$

From table A-15[8]

Size factor (Kb) from $2.79 \leq d \leq 51$ mm

So for 35 mm dia.

$$= (d/7.62)^{-0.107}$$

$= (35/7.62)^{-0.107}$
$= 0.85$

So for 40 mm dia.

$= (d/7.62)^{-0.107}$
$= (40/7.62)^{-0.107}$
$= 0.837$

Surface factor (Ka)

$Ka = a*Sut^{b}$

Sut- Minimum tensile strength, a and b from ref Table[8]

For machined component

a = 4.51 Mpa, and b = -0.265 (it is an exponent)

$Ka = 4.51 *(580)^{-0.265}$
$Ka = 0.8353$

Loading factor Kc for bending is 1

I. Stress calculation

Moment (Pe) = Resultant load * Load distance from flanges
=8769.3 N *116.5 mm
= 1021627 N-mm

TABLE III
PARAMETERS

Dist. of Horizontal load (mm)	262.49
Span of bearings (mm)	50.5
coefficient	0.7
Vertical Load (N)	9000
Horizontal Load (N)	6300
King pin angle (deg)	13
Parallel component Vp (N)	8769.3
Normal component Vn (N)	2024.6
Check	TRUE
Vertical load Couple (N-mm)	1021627
SLR (mm)	376

TABLE IV
STRESS CALCULATIONS

Bending Moment @ corner radius		
Dist of radius point from Bearing (mm)	15	Lower Brg
OLD DESIGN		
Bending Moment	653957	Nmm
Section ID	26	mm
Section OD	**35**	mm
Moment of Inertia	51230	mm^4
Stress	223	Mpa
Size corr. Factor Kb	1.68	
Actual stress	**375**	**Mpa**
NEW DESIGN		
Bending Moment	653957	Nmm
Section ID	26	mm
Section OD	**40**	mm
Moment of Inertia	103232	mm^4
Stress	127	Mpa
Size corr. Factor Kb	1.68	
Actual stress	**213**	**Mpa**

J. Using modified Goodman equation

After calculating the maximum and minimum for each stresses the alternating and mean effective stresses calculated. The following equations are used.

Mean Stress $\sigma_m = (\sigma_{max} + \sigma_{min}) / 2$

Range of Stress $\sigma_r = (\sigma_{max} - \sigma_{min})$

Stress Amplitude $\sigma_a = \sigma_r/2 = (\sigma_{max} - \sigma_{min}) / 2$

Stress Ratio $R = \sigma_{min} / \sigma_{max}$

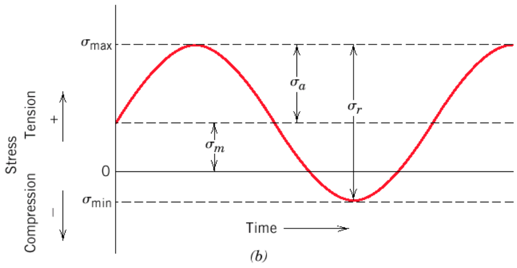

Fig. 5 Stress amplitude curve[8]

The alternating stress must then have various size, load, and stress concentration factors applied to it. This is necessary because these values are different for each loading mode. In addition, because these factors are applied to each stress they are not factored into endurance limit in the Marin equation

The no of cycle for failure are calculated by using through calculations using the following equations

$Nf = [\sigma a`/a]^{1/b}$

Where $a = (0.9*Sut)^2 /Se$

And $b = -1/3\log[0.9*Sut/Se]$

TABLE V
LIFE CALCULATIONS

Life Comparison			
Parameter	Original Design + Original loads	Original design + Revised load	New design +Revised load
Torsion	0	0	0
Bending	375	459	213
Axial	0	0	0
Alternating stress	375	459	213
Mean stress	0	0	0
Max stress	375	459	213
Min Stress	-375	-459	-213

Stress Amplitue	375	459	213
Reversiable	375	459	213
Slope	6	6	6
Fatigue limit	165	165	163
Life (No of cycles)	**728**	**216**	**20225**
Ultimate Stress Nmm	580	580	580
Yield Stress N mm	380	380	380
Endurance ratio	0.5	0.5	0.5
Size factor Kb	0.85	0.85	0.837
Surface Ka	0.67	0.67	0.67
Load	1	1	1

It is observed the life, No of cycles has increased drastically to cater the field application requirement. The design life of housing has improved by multiple times. This is because the next available suitable bearing is of dia.40 mm. so by changing the section modulus with the same material of housing the duty cycle of axle accomplished.

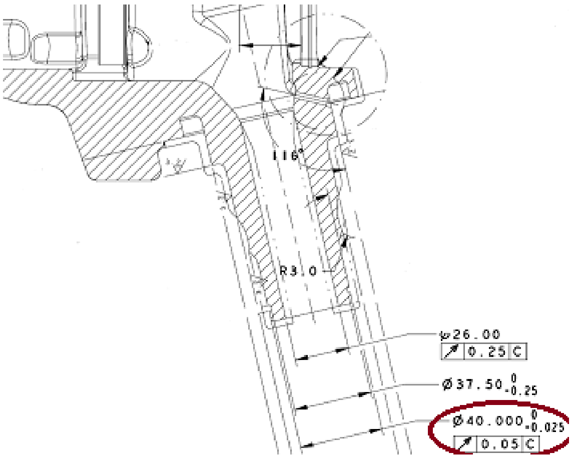

Fig.6 New Housing with 40 mm diameter

Fig.7 Optimisation geometry comparison

K. FEA analysis

FEA analysis is carried out using Abaqus 6.13-1 with SimLAb 9.0. The results of analytical and FEA compared with each other and it is found that both correlate to each other. Tie rod, hydraulic cylinder and axle shaft are simulated with beam elements. Beam release is provided to simulate the ball joint. It was observed that panic stop and breakout are causing the maximum damage to the housing. With new design the panic stop stress 358 Mpa reduced to 300 Mpa. And loader breakout application stresses 313Mpa to 294 Mpa.

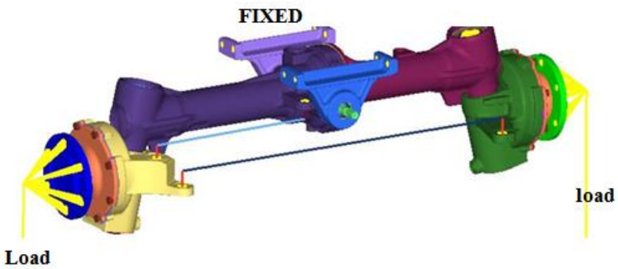

Fig.8 Boundary conditions of FEA

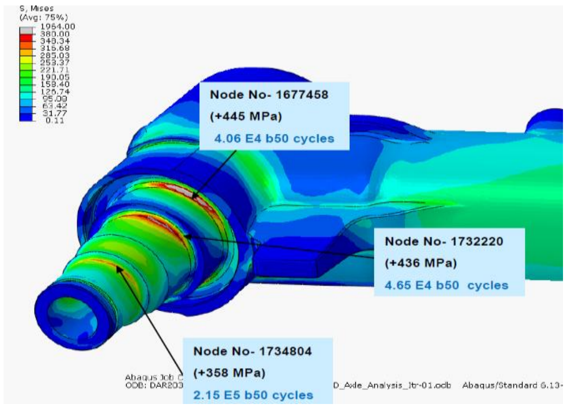

Fig.9 Panic Stop Old housing

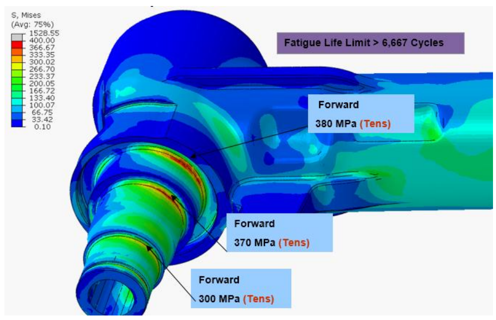

Fig.10 Panic Stop New housing

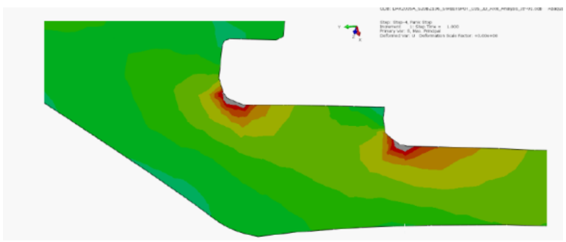

Fig.11 Enlarged view of stress damage

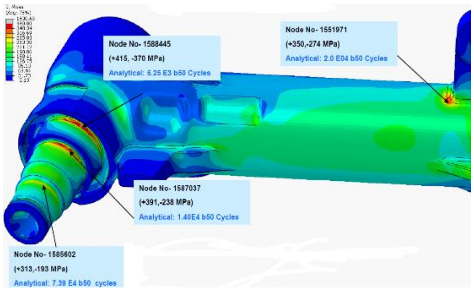

Fig.12 Loader Breakout Old Housing

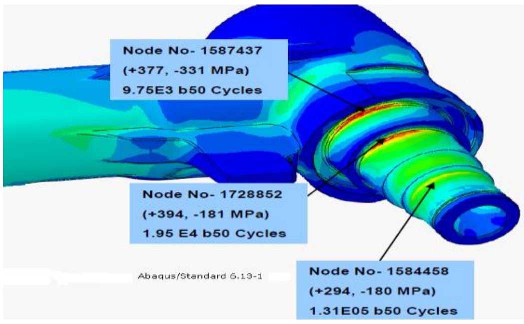

Fig.13 Loader Breakout New Housing

Fig.14Field testing of axle with New Housing

L. *Other Options of design*

There are few more options for this design

- Case hardening of current housing shoulder
- Use ofhigh tensile forged steel material

Both the options are validated but not economical and ease of design for manufacturing.

III. Conclusions

The calculated life of the housing is better in revised design with revised load goals. The section modulus change has improved the life of housing. Panic Stop and loader break out are the two critical applications that need to be withstand by the new design. It was observed that panic stop and breakout are causing the maximum damage to the housing. With new design the panic stop stress 358Mpa reduced to 300Mpa and breakout stresses 313Mpa to 294Mpa. The next bearing available to accommodate the best option to choose and opt for the application is of 40 mm. the fatigue life of the housing increased from 728 cycle to 20225 cycles. The New Housing weight is increased by 150 gms.New Axle Tested in field for loader application and found no damage to the housing. Field application data is very crucial to have robust design.Product performance has improved drastically than the required no of cycles. Close co-relation of design factors and application study decides the life of a product. Factor of safety plays vital role in applications where application data is not precise.

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