Design Optimisation of Four Wheel Drive Tractor Front Axle Housing

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Abstract—Four wheel drive tractors are used for high 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. Compare the 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,

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 housing with 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.

II. 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

<table>
<thead>
<tr>
<th>Failure analysis</th>
<th>Analytical approach / Calculations</th>
<th>FEA</th>
<th>Field Test</th>
<th>Comparison and improvement</th>
</tr>
</thead>
</table>

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III. 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/mm²,
Yield Strength: 380 N/mm²,
Elongation: 6%

<table>
<thead>
<tr>
<th>TABLE I. INSPECTION REPORT</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td># CHARACTERISTIC</td>
<td>SPECIFICATION</td>
</tr>
<tr>
<td>1</td>
<td>GRADE - 550/6 or MS-75B Chemistry</td>
</tr>
<tr>
<td></td>
<td></td>
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<tr>
<td></td>
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<td></td>
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</tr>
<tr>
<td>2</td>
<td>Surf Hardness</td>
</tr>
<tr>
<td>3</td>
<td>Tensile Strength</td>
</tr>
<tr>
<td></td>
<td>Yield Strength</td>
</tr>
<tr>
<td></td>
<td>Elongation %</td>
</tr>
<tr>
<td>4</td>
<td>Matrix Structure</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Internal Soundness</td>
</tr>
</tbody>
</table>

B. Component Geometry

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>
<thead>
<tr>
<th>TABLE II. MECHANICAL PROPERTIES OF HOUSING</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>PARAMETER</td>
<td>BEFORE</td>
</tr>
<tr>
<td>Section ID</td>
<td>26 mm</td>
</tr>
<tr>
<td>Section OD</td>
<td>35 mm</td>
</tr>
<tr>
<td>Step OD</td>
<td>41 mm</td>
</tr>
<tr>
<td>Transition radius</td>
<td>3 mm</td>
</tr>
<tr>
<td>Stress concentration factor</td>
<td>1.68 N</td>
</tr>
<tr>
<td>Young modulus</td>
<td>210000 N/mm²</td>
</tr>
<tr>
<td>Yield of material</td>
<td>380 N/mm²</td>
</tr>
</tbody>
</table>

Fig.1 Housing geometry

Fig.2 Failed housing

Fig.3 Re-action diagram
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,

\[
\text{Section modulus (Z)} = \frac{\text{Bending moment (M)}}{\text{Ultimate tensile strength of the material (F)}}
\]  
(1)

Where, 
- \(Z\) = Section modulus
- \(M\) = Bending Moment
- \(F\) = Ultimate tensile strength of material

E. Original Design + old Loads (18KN) on axle

For the area of the spindle we can find out using,
\[
A = \frac{\pi}{4} \times (D^2 - d^2)
\]  
(2)

Moment of inertia (I)
\[
I = \frac{\pi}{4} (D^4 - d^4)
\]  
(3)

Section Modulus at lower Bearing \(= \frac{I}{Y}\)
\[
= \frac{\pi (35^4 - 26^4)}{32 - 35}
\]  
= 2925.245 mm³

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

For the area of the spindle we can find out using,
\[
A = \frac{\pi}{4} \times (D^2 - d^2)
\]  
(2)

Moment of inertia (I)
\[
I = \frac{\pi}{4} (D^4 - d^4)
\]  
(3)

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

For the area of the spindle we can find out using,
\[
A = \frac{\pi}{4} \times (D^2 - d^2)
\]  
(2)

Moment of inertia (I)
\[
I = \frac{\pi}{4} (D^4 - d^4)
\]  
(3)

Section Modulus at lower Bearing \(= \frac{I}{Y}\)
\[
= \frac{\pi (40^4 - 26^4)}{32 - 35}
\]  
= 5158.98 mm³

H. Endurance strength & Stress Concentration factor

Laboratory endurance strength (\(S_e'\)) of the materials obtained from S-N diagram are therefore corrected for actual conditions by using correction factors,
\[
S_e = K_a x K_b x K_c x K_d x K_t x K_f x S_e'
\]  
(5)

Where,
- \(K_a\) = Surface Correction factor
- \(K_b\) = Size Correction factor
- \(K_c\) = Loading factor
- \(K_d\) = Temperature Correction factor
- \(K_t\) = Stress concentration Correction factor
- \(K_f\) = Miscellaneous Correction factor
- \(S_{e'}\) = Endurance Strength of material specimen under laboratory condition

Stress concentration factor
\[
K_t = \text{Normal Stress}
\]
\[
K_{ts} = \text{For Shear Stresses}
\]
\[
K_t = \frac{\sigma_{max}}{\sigma}
\]  
(6)
\[
K_{ts} = \frac{\tau_{max}}{\tau}
\]  
(7)

From table A-15[8]

Size factor \((K_b)\) from 2.79 \(\leq d \leq 51\) mm

So for 35 mm dia.
\[
= (\frac{d}{7.62})^{-0.107}
\]  
(8)
\[
= (35/7.62)^{-0.107}
\]  
= 0.85

So for 40 mm dia.
\[
= (\frac{d}{7.62})^{-0.107}
\]  
(8)
\[
= (40/7.62)^{-0.107}
\]  
= 0.837

Surface factor \((K_a)\)
\[
K_a = 4.51 \times \frac{S_{ut}}{\sigma}
\]  
(9)

Sut - Minimum tensile strength, \(\alpha\) and \(b\) from ref Table[8]

For machined component
\(a = 4.51\) Mpa, and \(b = -0.265\) (it is an exponent)
\[
K_a = 4.51 \times (580)^{-0.265}
\]

Loading factor \(K_c\) for bending is 1

I. Stress calculation

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

\[
N_f = \frac{[\sigma_a/a]^{1/b}}{[\sigma_a/a]^{1/b}}
\]

(15)

Where

\[
a = (0.9*Sut)^2 / Se
\]

And

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

(16)

(17)

**TABLE V. LIFE CALCULATIONS**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Original Design + Original loads</th>
<th>Original design + Revised load</th>
<th>New design +Revised load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torsion</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Bending</td>
<td>375</td>
<td>459</td>
<td>213</td>
</tr>
<tr>
<td>Axial</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Alternating stress</td>
<td>375</td>
<td>459</td>
<td>213</td>
</tr>
<tr>
<td>Mean stress</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Max stress</td>
<td>375</td>
<td>459</td>
<td>213</td>
</tr>
<tr>
<td>Min Stress</td>
<td>-375</td>
<td>-459</td>
<td>-213</td>
</tr>
<tr>
<td>Stress Amplitude</td>
<td>375</td>
<td>459</td>
<td>213</td>
</tr>
<tr>
<td>Reversible</td>
<td>375</td>
<td>459</td>
<td>213</td>
</tr>
<tr>
<td>Slope</td>
<td>6</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Fatigue limit</td>
<td>165</td>
<td>165</td>
<td>163</td>
</tr>
<tr>
<td>Life (No of cycles)</td>
<td>728</td>
<td>216</td>
<td>20225</td>
</tr>
<tr>
<td>Ultimate Stress Nmm</td>
<td>580</td>
<td>580</td>
<td>580</td>
</tr>
<tr>
<td>Yield Stress N mm</td>
<td>380</td>
<td>380</td>
<td>380</td>
</tr>
<tr>
<td>Endurance ratio</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>Size factor Kb</td>
<td>0.85</td>
<td>0.85</td>
<td>0.837</td>
</tr>
<tr>
<td>Surface Ka</td>
<td>0.67</td>
<td>0.67</td>
<td>0.67</td>
</tr>
<tr>
<td>Load</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>
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.

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.
L. Other Options of design

There are few more options for this design
- Case hardening of current housing shoulder
- Use of high tensile forged steel material

I. 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 grams. 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.

ACKNOWLEDGMENT

I would like to give my sincere thanks to my guide Prof. M.S. Ramgir, who accepted me as his student and being a mentor for me. He offered me so much advice, patiently supervising and always guiding in right direction. I have learnt a lot from him and he is truly a dedicated mentor. His encouragement and help made me confident to fulfill my desire and overcome every difficulty I encountered. Also I would like to mention my sincere thanks to senior Mr. Idris Poonawala who guided me thru all steps of design and helped to complete the project and understand the design in detail.

I also highly obliged to the organization and management, for giving me an opportunity to continue my education and enhance my knowledge.

REFERENCES