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A STATIC STRUCTURAL ANALYSIS OF KNUCKLE JOINT USED FOR SUGARCANE MILL

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Abstract: Sugarcane crushing is an important activity and maximum insistence nowadays is on speed, maximum extraction without losses, and easy collection of residual material which can be used as organic compost. Sugarcane crushing is carried out in sugar mills with large capacity rollers mounted on shafts and this complete assembly called as Headstock. Knuckle joint is a joint between two parts allowing movement in one plane only. However the entire assemble is manufactured using casting, and typically these brittle materials have a high chance of failure if there is a stress concentration as certain locations. Knuckle joint is a major component of Headstock of sugar mill which is used to two rods between each other. It experiences maximum shear and tensile stresses due to heavy weight of Headstock. Hence in order to find stresses in headstock, one must find out stresses in knuckle joint. The FEA Analysis of Knuckle joint is done and various shear and tensile stresses results are plotted. The Analytical solution of knuckle joint is found out using standard calculations. Now, these results are validated by theoretical calculations available for knuckle joint. The force applied knuckle joint is 50 KN .The diameter of pin is proposed to be around 30 mm. The stress results by theoretical calculations and FEA software are validated. This proves that the FEA software results are correct. Secondly it also shows that certain high stresses are generated near knuckle joints and this result are helpful for further analysis of Headstock for reducing the stresses, increasing life and reliability of headstock.

Keywords: FEA, Knuckle joint analysis, eye, pin, fork, knuckle joint stress analysis by FEA.

I. Introduction

Knuckle joint is a joint which is situated between two elements allowing movement in only one plane. It is a kind of hinged joint between two rods, just like a ball and socket joint. There are many situations where two parts of machines are required to be restricted, for example two rods may be joined coaxially and when these rods are pulled apart they should not separate i.e. should not have relative motion and continue to transmit force.

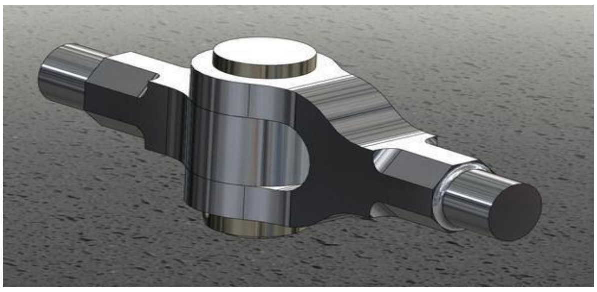

Figure 1: 3D View of knuckle joint [2]

A knuckle joint is used to connect two rods which are under tensile loads whereas, if the joint is guided, compressive load may be supported by rods. A knuckle joint can be easily disconnected when required.

Finite element analysis consists of a model made on computer of a material or design that is stressed and is then analyzed for specific results like stress, deformation, etc. It is used in new product design, and refinement of existing product. A company is able to verify a proposed design will be able to perform to the client's specifications prior to manufacturing or construction.

Knuckle joints are most common in steering and drive train applications where it needs to move something but also need to allow for offset angles. A knuckle joint is used when two or
More rods subjected to tensile and compressive forces are fastened together such that their axes are not in alignment but meet in a point. The joint can be easily connected and disconnected. Knuckle joint is found in valve rods, braced girders, suspension chain links, elevator chains, etc. Figure 1 shows the 3D view of a knuckle joint.

Figure 2 shows the sectional view of knuckle joint. The Knuckle joint assembly consists of components such as Single eye, Fork (Double eye), Knuckle pin, Collar, Tapper pin. The end of one of the rods is forged in the form of a fork while the end of the other rod has an eye, which can be inserted in the jaws of the fork. A cylindrical pin is passed through the holes in the forks and the eye. The pin is secured in position by a taper pin, split pin or a thin nut screwed up to shoulder on the end of the pin. The ends of the rods are made octagonal for good hard grip. There is always axial or linear line of action of load.

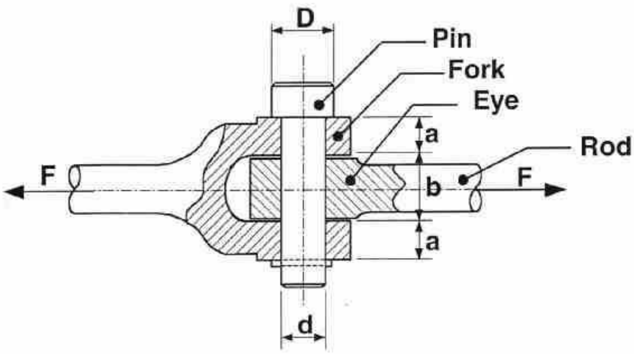

Figure 2. Sectional view of knuckle joint [1]

At one end of the rod the single eye is formed and double eye is formed at the other end of the rod. There is a small pin which is inserted through eyes and both, single and double eyes are connected by it. The pin has a head at one end and at other end there is a taper pin. Now, when the two eyes are pulled apart, the pin holds them together. The solid rod portion of the joint in this case is much stronger than the portion through which the pin passes [11].

A knuckle joint may be failed on the following three modes
1. Shear failure of pin (single shear).
2. Crushing of pin against rod.
3. Tensile failure of flat end bar.

II. Literature Review

The research work carried out in the area of analysis and feature driven optimization of knuckle joint of Sugar mill, it is found that an evolutionary research has been done and keeps on expanding in this topic. The failure mechanism of knuckle joint has been studied by several investigators (3-6). Jones [3] has reported that shear failure due to torsional loading is the normal failure mechanism in many engineering components. Pantazopoulos et.al [4] has studied the failure of a knuckle joint of a universal coupling system. It was mentioned, that overload of torsional stress of the knuckle joint is the major cause of failure. However, in many cases it was reported that wear of material due to severe friction leading to delaminating wear [5][6].

ANSYS 13 software has been used for analysis of knuckle joint with modified material and changing loads. Many systems used in industries use knuckle joint which is combination of two materials i.e. cast iron and stainless steel. The modification of one of the material that is changing cast iron into a composite polymer material.[8] The proposed system has many advantages over other system such as making the device, simpler and having maximum safety and is eco friendly. The reason for considering polymer is that property of polymer is mostly similar to the property of metal. Composite polymers are mainly characterized by a high flexibility material. The revolutionary evolution in technologies in last year allowed reducing stress and strain. [9]
The frequent failure analysis of output shaft of gear motor used for cold rolling mill to drive the Pay-off Four-HI (Horizontally inserted). One end of that shaft is connected to the claw coupling to the drive of the Pay-off four-HI .[10]By considering the drive system forces and torque acting on the output shaft are determine using which the stresses occurring at the failure section are calculated .[12] Stresses analysis is also carried out with analytical method and comparing these results with the software results that is done using the ANSYS software. After getting all the different parameter redesign of shaft is done and again analyzing the shaft using ANSYS and it is expected that shaft may be torsionally Rigid. [7]

III. Theoretical Calculations For Stress Of Knuckle Joint Components

Material selection

The rods are subjected to tensile force. Therefore, yield strength is criteria for selection of material of rod. The pin is subjected to shear stresses and bending stresses. Therefore strength is criteria for selection of pin. On strength basis, material for rod and pin is selected as plain carbon steel of Grade 30C8 ($Syt=405N/mm^2$).Assumption –yield strength in compression is equal to yield strength in tension.
Axial tensile force applied= 50 KN.

Mechanical Properties-

Yield strength (Syt) - 405 Mpa
Grade-30C8
Carbon- 0.25 to 0.35 %
Manganese- 0.30 to 0.60%
Silicone: 0.10 – 0.35%
Phosphorus: 0.030, S: 0.035
Used as per IS: 1871 (PART II) Reaffirmed 1993

1) Calculation of permissible stress-

$$\sigma_t = \frac{Syt}{FOS} = \frac{405}{5} = 81 N/mm^2 \quad (1)$$

$$\sigma_c = \frac{Syc}{FOS} = \frac{Syt}{FOS} = \frac{405}{5} = 81 N/mm^2 \quad (2)$$

$$\tau = \frac{Syc}{FOS} = \frac{0.5Syt}{FOS} = \frac{0.5X405}{5} = 40.5 N/mm^2 \quad (3)$$

2) Calculation of Dimensions

The dimensions of knuckle joint are calculated by procedure

Step 1- Diameter of rods (D)

$$D= \frac{\sqrt{4P}}{\pi\sigma} = \sqrt{\frac{4(50X1000)}{\pi(81)}} \quad (4)$$

=28.11or 30mm.

Step 2- Enlarged diameter of rods (D_1)

$$D_1=1.1D \quad (5)$$

=1.1(30)

= 33 Or 35 mm.

Step 3-Dimentions of a & b

a =0.75D (6)

=0.75(30)

=22.5 or 25 mm

b =1.25 D (7)

= 1.25(30)

= 37.50 or 40 mm

Step 4- Diameter of Pin (d)

$d=\sqrt{\frac{2P}{\tau\pi}}$ (8)

$= \frac{\sqrt{2(50X1000)}}{\pi(4)}$

= 28.21 or 30mm

Also,

$d=\sqrt[3]{(32/\pi\sigma_b)(P/2)[b/4 + a/3]}$ (9)

$d= \sqrt[3]{32(50x10^3)/\pi(80)2[(40/4 + 25/3)]}$

= 38.79 or 40 mm

d= 40mm

Step 5- Dimensions of d_0 & d_i

$d_o=2d=2(40.5) =81mm$ (10)

$d_i=1.5d=1.5(40.5) =61mm$ (11)

Step 6-Check for stresses in eye

$\sigma_t=\frac{P}{b(do-d)} = \frac{(50x103)}{[40(80-40)]}$ (12)

$= 31.25\ N/mm^2$

$\sigma_t < 81\ N/mm^2$

$\sigma_c=\frac{P}{bd} = \frac{50x1000}{(40x40)}$ (13)

$= 31.25\ N/mm^2$

$\sigma_c<81N/mm^2$

$\tau = \frac{P}{b(do-d)} =\frac{50x10000}{[40(81-40)]}$ (14)

$=31.25N/mm^2$

$\tau <40N/mm^2$

Step 7-Check for stresses in Fork

$\sigma_{t=}\frac{P}{[2a(do-d)]}$ (15)

$= \frac{(50x103)}{[2x25x(81-40)]}$

$=25N/mm^2$

$\sigma_t < 80\ N/mm^2$

$\sigma_c=\frac{P}{2ad}$ (16)

$= \frac{(50x103)}{(2x25x40)}$

$=25N/mm^2$

$\sigma_c<81N/mm^2$

$\tau =\frac{P}{b(do-d)}$ (17)

$= \frac{(50x103)}{(2x25x40)}$

$= 25N/mm^2$

$\boxed{\tau =25N/mm^2}$

IV. Methodology

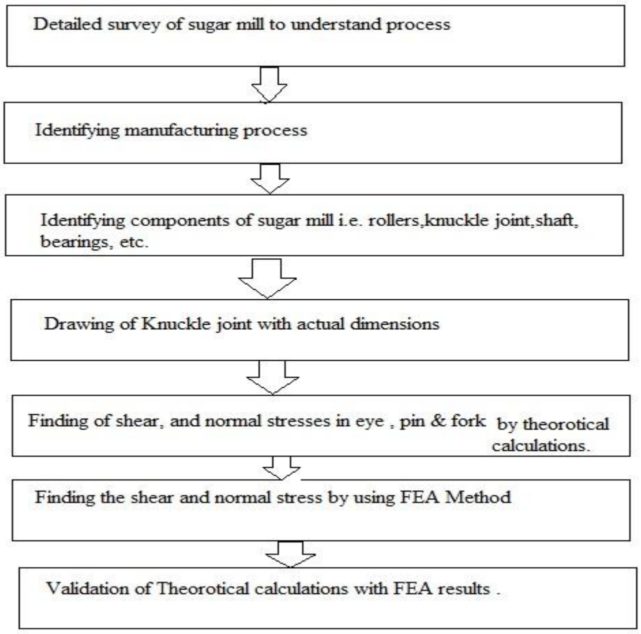

Figure 3. Flowchart for Methodology of complete work

In the flowchart shown in figure 1, complete methodology is stated. The FEA analysis of Knuckle joint is done and various shear and tensile stresses results are plotted. The Analytical solution of knuckle joint is found out using standard calculations. Now, these results are validated by theoretical calculations available for knuckle joint.The force applied knuckle joint is 50 KN .The diameter of pin is proposed to be around 30 mm. The stress results by theoretical calculations and FEA software are validated.This shows that certain high stresses are generated near knuckle joints and these results are helpful for further analysis of Headstock for reducing the stresses, increasing life and reliability of headstock.

V. Results from Theoretical Calculations

Table 1.Stress calculations by theoretical method

Element	Type of stress	Hand Calculations stress result(N/mm^2)	Maximum permissible limits (N/mm^2)
Fork	Shear stress	25	40 .5
	Tensile stress	25	81
Eye	Shear stress	31.25	40.5
	Tensile stress	31.25	81
Pin	Shear stress	40.5	40.5
	Tensile stress	40.5	40.5

Table 1 shows the results of shear stress and tensile stress for fork, pin and eye by theoretical calculations with their maximum permissible limits.

VI. 3d Model & Geometry

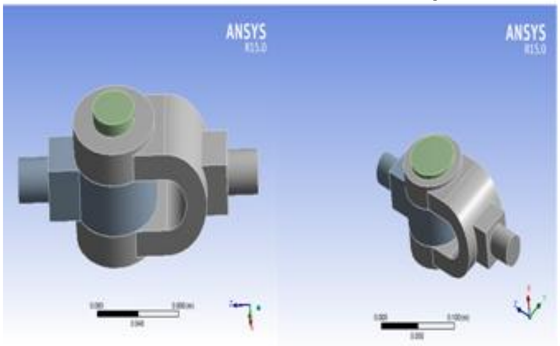

Figure 4.3D Model of Knuckle Joint using ANSYS R 15.0 Software .

Figure 4 shows the 3D model of knuckle joint with actual dimensions and figure 5 shows meshing of knuckle joint assembly with following details.

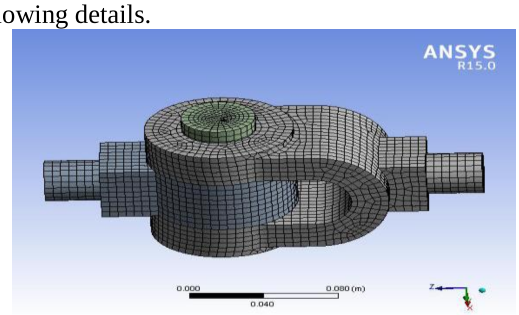

Figure 5. Meshing of Knuckle Joint

Mesh Details:
Method used- Hex dominant method
Force applied- 50 KN
No of nodes- 70337
No of elements- 18289
Element size- 4.5 mm for all bodies i.e. eye, pin and fork.

VII. Boundary Conditions

The load of 50 KN is applied on one end and one end is fixed as shown in figure 6 below. It shows boundary conditions for knuckle joint with one end fixed and one on end force of 50 KN is applied.

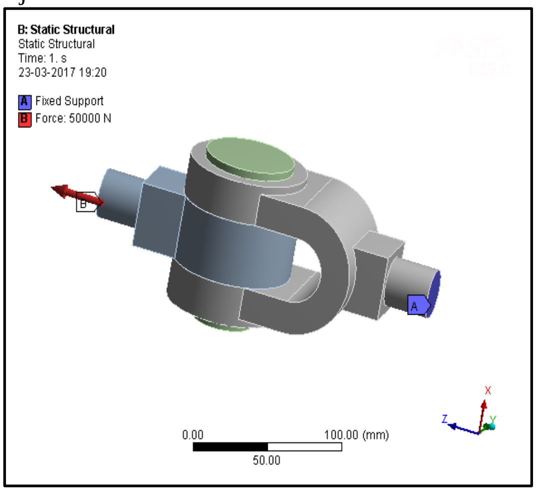

Figure 6. Boundary conditions for Knuckle Joint

VIII. Results and Discussion

a) Static structural analysis of Pin

In figure 7, the shear stress of Pin with analytical stress as 40.5 MPa, which is validated by FEA Result i.e. 43.72 MPa which is nearly similar. In figure8, normal stress for pin which is 40.5 MPa by theoretical calculation and is 20.012 MPa i.e. less than allowable stress by FEA. Hence pin is safe in tension. In figure 9,the maximum shear stress for pin by ANSYS is 43.414 MPa, which is nearly similar to maximum permissible limit i.e. 40 MPa. Hence pin is safe.

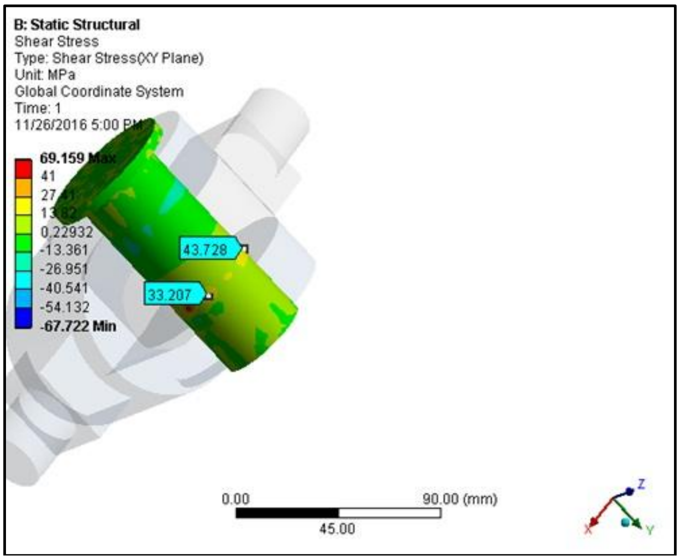

Figure 7. Shear stress analysis for pin with 50 KN Force

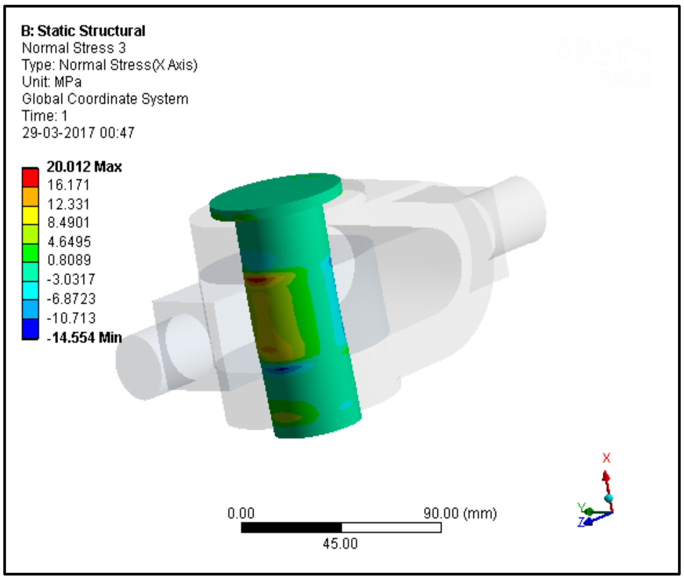

Figure 8. Normal stress for pin with 50 KN Force.

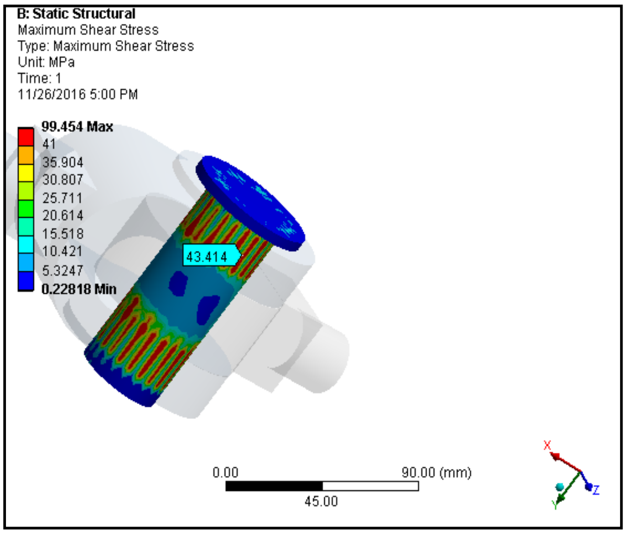

Figure 9. Maximum shear stress for pin with 50 KN force.

b) Static Structural Analysis for eye

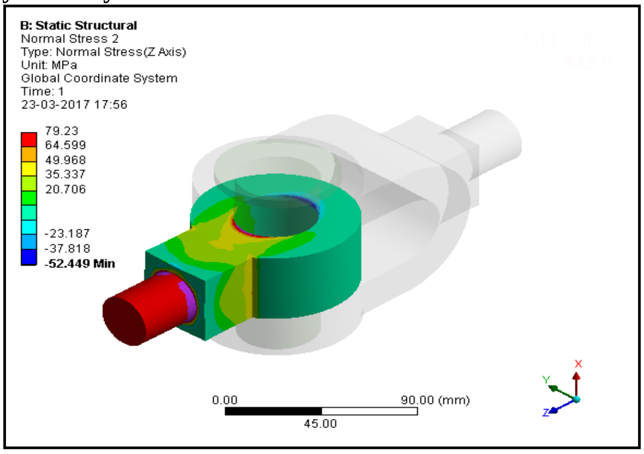

Figure 10.Normal stress for eye with 50 KN Force.

In figure 10,the normal stress for eye is 79.23 MPa which is less similar to theoretical normal stress 31.25 MPa, but is below maximum permissible limit. i.e. 80 MPa. Hence eye is safe for tensile stress.Figure 11 explains the shear stress for eye by FEA is 31.898 MPa which is exactly similar to theoretical calculations i.e. 31.25 MPa. Hence eye is safe for shear stress. Figure 12 shows the maximum shear stress for eye with 50 KN force, i.e. 48.465 MPa as maximum shear stress which is near to 40 MPa by theoretical calculations. The normal stress for fork by FEA is 78.786 MPa which is less similar to theoretical stress i.e. 25 MPa but it is less than maximum permissible limit i.e. 80 MPa as seen in figure 13 . Hence fork is safe for shear stresses**.** Figure 14 explains, the shear stress by FEA is 34.97 MPa, which is nearly similar to theoretical stress value i.e. 25 MPa, but less than maximum permissible limits i.e. 40 MPa. Hence fork is safe for shear stress. Maximum shear stress for fork with 53.494 MPa,as seen in figure 15.

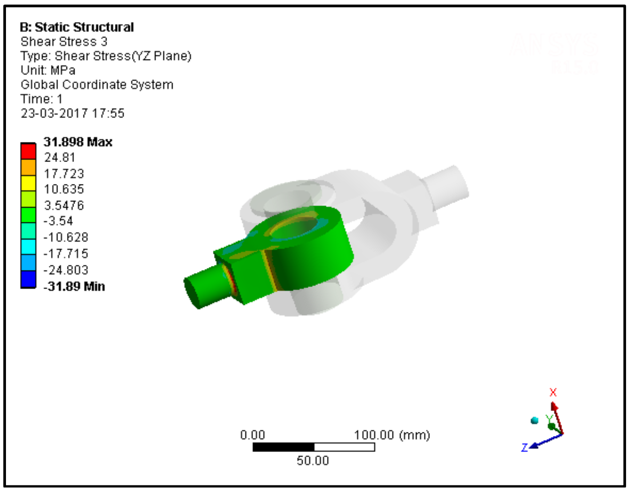

Figure 11. Shear stress for eye with 50 KN force.

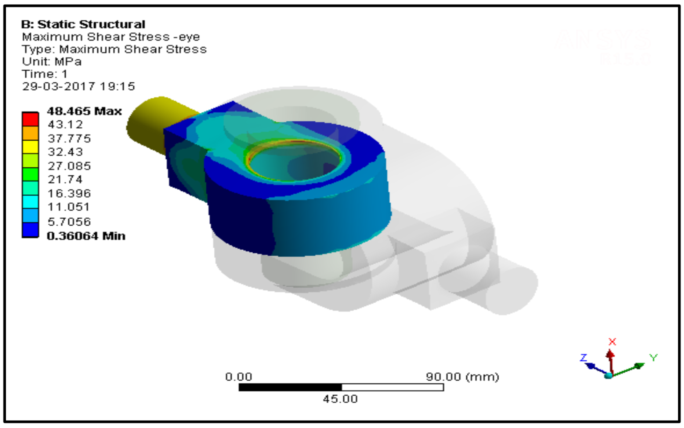

Figure 12.Maximum shear stress for eye with 50 KN force.

c) Static Structural Analysis of Fork

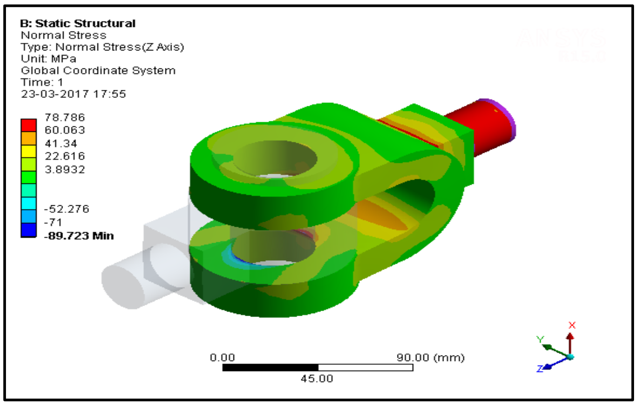

Figure 13. Normal stress for fork with 50 KN force.

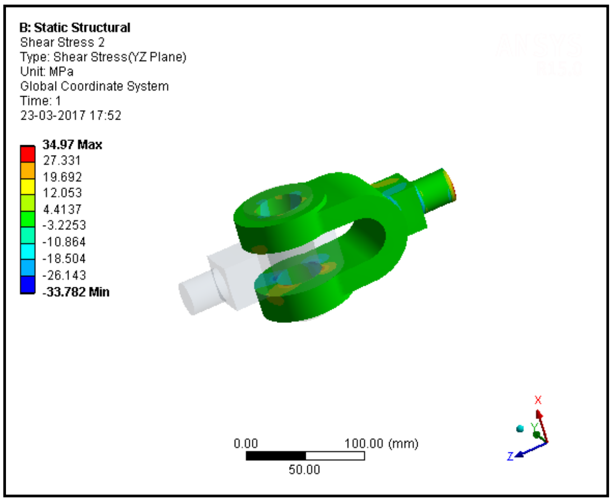

Figure 14. Shear stress for fork with 50 KN force.

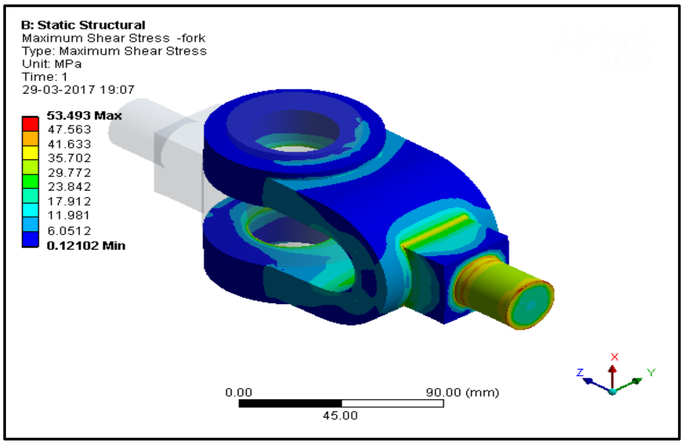

Figure 15.Maximum shear stress for fork with 50 KN force

d) Strain energy distribution

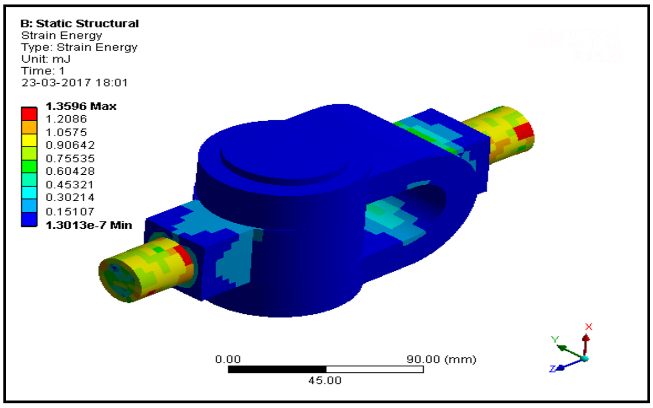

Figure 16. Strain energy distribution for knuckle joint assembly.

Figure 16 shows strain energy distribution which is maximum at the ends i.e. 1.3596 m J. Figure 17 depicts Total deformation of knuckle joint assembly. Figure 18 shows Maximum shear elastic strain for knuckle joint assembly e)Total deformation

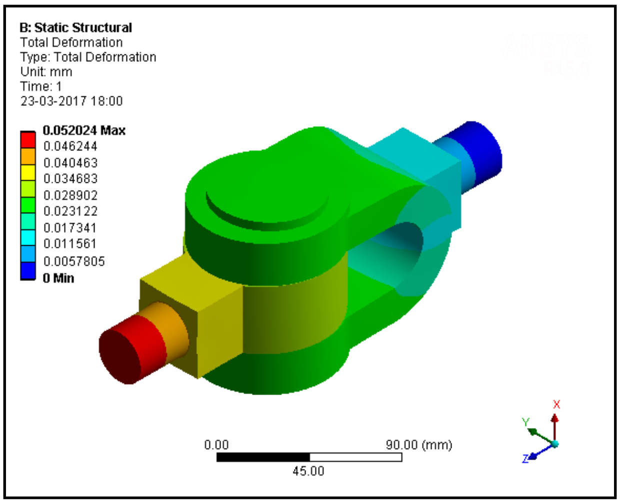

Figure 17. Total deformation of knuckle joint assembly.

f) Maximum shear elastic strain

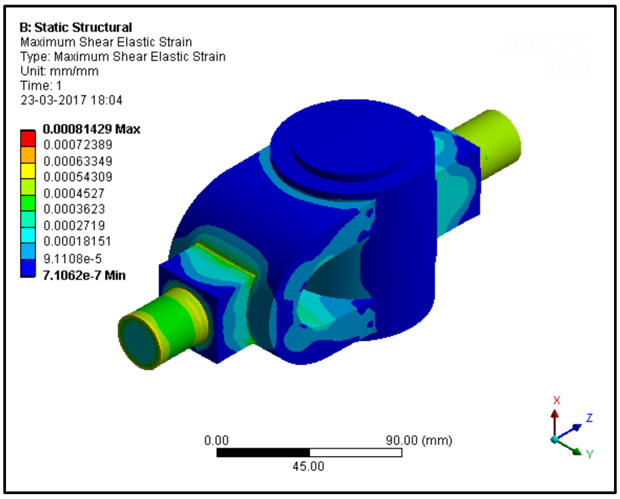

Figure 18. Maximum shear elastic strain for knuckle joint assembly

IX. Result Table

Table 2. Theoretical results validation with FEA Result.

Element	**Type of stress**	**Hand Calculations result(N/mm^2)**	**FEA Results (N/mm^2)**	**Maximum permissible limits (N/mm^2)**
Fork	Shear stress	25	34.97	40.5
	Tensile stress	25	78.78	81
Eye	Shear stress	31.25	31.89	40.5
	Tensile stress	31.25	79.23	81
Pin	Shear stress	40.5	43.728	40.5
	Tensile stress	40.5	20.02	40.5
	Max.shear stress	40.5	43.414	40.5

Table 2 shows the comparison of shear and tensile/normal stress values for fork, eye and pin by both theoretical and FEA method.

Table 3. Results of various parameters for knuckle joint assembly by FEA.

Sr. no	Mechanical parameter for knuckle joint assembly	FEA analysis value
1.	Strain energy distribution	1.3596 m J
2.	Total deformation	0.052024mm
3.	Maximum shear elastic strain	0.00081429

X. Conclusions

The knuckle joint proposed to develop in the present study is for an applied force of 50 KN. The diameter of the pin is proposed to be around 30 mm. The material of the knuckle joint is considered as mild steel grade 30C8. A 3-D CAD model of knuckle joint based on calculated dimensions is designed. In order to carry out the stress analysis, mesh is developed for the knuckle joint. The mesh consists of 70337 nodes and 18289 elements. ANSYS software was run and the stress contour, tensile stress, displacement contour, strain energy, maximum shear elastic strain, total deformation contour for eye pin and fork are obtained.

Based on the ANSYS analysis, it shows that a pin of 30mm diameter can withstand a load of 50 kN, if a factor of safety of 5 is used. Thus it can be seen that the theoretical stress values of shear, tensile and maximum allowable stresses nearly matches with the FEA Stress results. Hence as the theoretical and FEA results are nearby same and below the maximum allowable stress for all components, we can say that all three components are safe. Further due to this validation we can state that the FEA software results are correct as they are similar to theoretically calculated results. Thus we can also conclude that as the regions of eye where force of 50 KN was applied, has maximum tensile stresses as in figure 10, and total deformation on the same area is also maximum i.e.0.052024mm as shown in figure 17.It shows that these highly stressed areas can be main reasons of cracks or fatigue in Sugarmill headstock.

XI. Future Scope

Thus similar kind of FEA analysis can be done for sugar mill headstock. On the basis of this knuckle joint analysis, we can predict that the regions where knuckle joints are located in headstock will be highly stressed. Thus, if stress in headstock is found to be high, then measures can be taken to reduce that stress in knuckle joint and ultimately in headstock .This will increase life and reliability of Sugar mill Headstock.

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