International Journal of Research in Advent Technology (IJRAT)
(E-ISSN: 2321-9637)
Special Issue National Conference "CONVERGENCE 2017", 09th April 2017

Design and Stress Behaviour Analysis of Cab Tilting Arrangement of Crane Cab assembly

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Abstract- Crane industries are very innovative and advance in engineering development. This work gives new tilting pivot assembly for CAB to be installed in crawler type mobile crane. Cab is having multiple loads for operating it from one fixed location to the end of Boom and jibs. So tilting and turning joint must be capable to hold the all loads coming on it, so pivot pin joint is designed here with unique rod insertion technique. To accommodate and balance forces on two pivots called tilting pivots.

Keywords:-Crane cab, CAE, Joints.

1. INTRODUCTION

Work is in the fabricated arrangement of tilting cab with 30 degree positions maximum. The design of the cabin to a major extent should be driven by the need for optimum field visibility where there is an unrestricted and reliable view of the ground situation and working surroundings .Ergonomic guidelines require that a machine operator should have a free view of the operating zone without have to adjust posture .The guideline states that the operator should not have to turn their more than 30˚ to either side and that head should not tilt more than 5˚ up and 25˚ down maintain comfort. Occasional head movements of 50˚ to the sides and 40˚ up and 50˚ down are acceptable. Seats, windows and cabin features should be designed with these guidelines in perspective. The operator should inherit a clear view as possible free from large blind areas caused from window frames and obstructions due to the cabin structure such as cabin pillars. Tilting is done by hydraulics arrangement for making smooth and powerful movement. Tilting loads are to be considered like self weight of the cab fabricated body, operator's weight, internal consoles and dashboards, glass panel work etc. Tilting is exist in crane cab for direct visibility of lifting object when it catches at different angle of jib or boom so for operator it's very important to control slewing and tilting of boom by in hand control unit its possible only when the object is visible.

2. LITERATURE REVIEW

D. Chen and S. Cheng have made the analysis of the stress distribution in adhesive bonded tubular lap joints subjected to torsion. The analysis was based on the elasticity theory in conjunction with variational principle of complimentary energy, with two adhered may be having different materials and different thickness. The closed form solution so obtained was used to determine the stress intensities in adhesive layer and stress concentration factor.

Chon T. Chon also have made analysis of the stress distribution of tubular lap joint in torsion whose adherents were of composite materials and obtained a closed such as wrap angles form solution. The stress concentrations at and near the end was studied as function of various parameters, overlap length and thickness of adhesive layer.

N. Pug no and G. Saracen make the analysis of the problem of torsion in adhesive bonded tubular joint. The stress field in the adhesive layer was obtained based on the elasticity theory and pure torsion theory. A special type of tubular joint was proposed by optimizing the tubular joint for torsion strength.

Kozo Ikegami make the investigation of the strength of adhesive shaft joint under combined axial tensile and torsion load analytically and experimentally to study effect of overlap length and cross sectional ratio. The joint considered was consisting of two steel shafts and a coupling bonded with adhesive. The stress strain distributions were computed by the elastic Finite Element Method with the assumption of symmetric three dimensional conditions. For strength evaluation Von-Moses conditions were applied to the outer and inner adherents and to the adhesive layer.

2.1Hydraulic Cab Tilt Systems

Fig.1 *Hydraulic cab tilt system*

Power-Packer's complete range of cab tilt systems and hood assist units make it possible for us to design a cab tilting system that is perfectly matched to your application. Combining our hydraulic cylinders with any of our hydraulic pumps whether electric, air or manually powered – we can offer you a better fitting system which results in a better performing system.

2.1 Features

- Internal spring mechanism in hood assists units. Self-contained design.
- Cab Tilt systems consist of a very simple design: cylinder, pump and latch make a fully functioning system.

2.2 Benefits

- Many customization options available – choose from electric, air, or manually powered pump and we can design the system for you.
- Many standard options available to create the best fit possible.
- Enhance safety by allowing for easy access to any repairs on job site.
- Hood assist systems substantially reduce effort required to open and close hoods.
- Internal spring mechanism in hood assist units holds the hood in open position even in wind or uneven terrain

3. PROBLEM IDENTIFICATION

Existing method to tilt it is by hydraulic cylinder it lifts the cab through hinges and main issues here is to maintenance multiple joints and big hydraulic arrangement, we need to replace it by single pivot rod connection .Metal to metal contact joint may form extra tightening and maintenance issue in joint parameters here we are planning to make bushing tube between holder and long pin.

4. OBJECTIVES

- To design new tilting arrangement with strength analysis for crane cab.
- To validated the new joint with strength analysis with examining real life boundary conditions.

4.1 The work includes

- Redesign of lifting bracket pivot assembly
- Design of pivot rod between superstructure body and the cab
- Design of weldment
- Stress analysis of pivot joint
- Stress analysis of structure
- Validation of the system using ansys workbench

4.2 Outcomes

The new cab allows the operator to adjust the angle of the cab from 0 degrees to 20 degrees above horizontal, with hydraulic power, the company said. The new tilt cab will allow operators to have better vertical visibility and less neck strain during certain work applications, such as setting cross arms on transmission structures, the company said. Additionally, the tilt cab is located forward of the centerline of rotation, providing better horizontal visibility when rotating the crane clockwise. It's a first boom truck crane manufacture to offer a tilt cab option.

4.3 Proposed Method of work

1. Design of pivot joint, development will be done once design get finalised.
2. Structural calculations and deflection analysis.
3. CAE on Ansys
4. FEA approach to make feasible working conditions.

5. DESIGN PARAMETER

5.1Design Factors

Design factors are essentially "safety factors" that allow us to design safe, reliable sheet metal components. Each operator may have his own set of design factors, based on his experience, and the condition of the raw material used.

- Worst Possible Conditions
- Effect of Axial Tension on Collapse Strength
- Design for Burst, Collapse and Tension

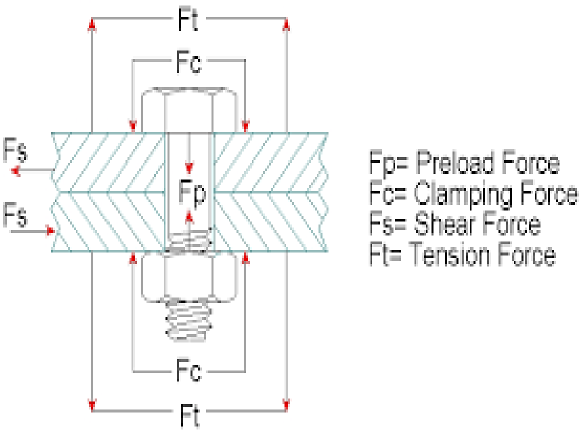
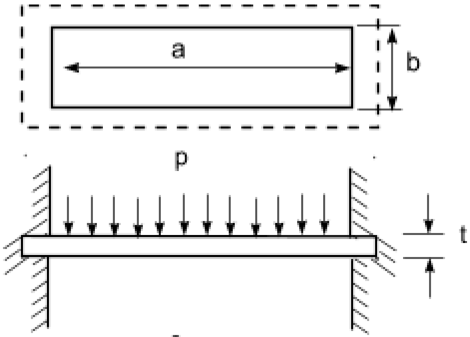


Fig.2 coupling joint behavior in frames component

5.2 *Sheet metal fabricated components*

Bending of plate components, or plate bending, refers to the deflection of a plate perpendicular to the plane of the plate under the action of external forces and moments. The amount of deflection can be determined by solving the differential equations of an appropriate plate theory. The stresses in the plate can be calculated from these deflections. Once the stresses are known, failure theories can be used to determine whether a plate will fail under a given load.

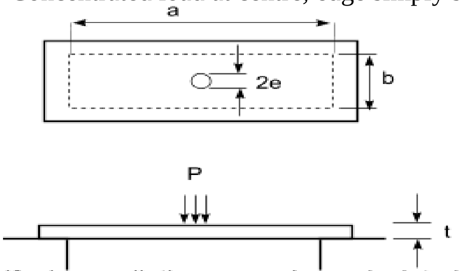
Uniform load, edge simply supported.



$$\sigma_m = \frac{pb^2}{2t^2\,[0{,}623(b/a)^6 + 1\,]} \quad \text{At mid edge a}$$

$$y_m = \frac{0{,}0284\,pb^4}{Et^3[1{,}056(b/a)^5 + 1]} \quad \text{At centre}$$

Concentrated load at centre, edge simply supported.



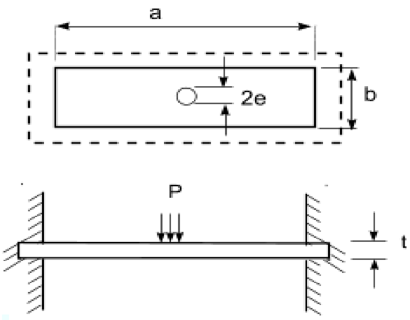
If e is small then use e' as calculated below

$$e' = \left(\sqrt{1{,}6e^2 + t^2}\right) - 0{,}675t \quad \text{if } e < 0{,}5t \text{ else use } e' = e$$

$$\sigma_m = \frac{1{,}5P}{\pi t^2}\left((1+\nu)\ \ln\left(\frac{2b}{\pi e}\right) + k_2\right) \quad \text{At centre}$$

$$y_m = k_1\,\frac{Pb^2}{Et^3} \quad \text{At centre}$$

	a/b								
	1,0	1,1	1,2	1,4	1,6	1,8	2,0	3,0	4->
k_1	0,127	0,138	0,148	0,162	0,17	0,177	0,180	0,185	0,185
k_2	0,435	0,565	0,650	0,789	0,875	0,927	0,958	1,000	0,000



Concentrated load at centre, edge clamped

If e is small then use e' as calculated below

$$e' = \left(\sqrt{1{,}6e^2 + t^2}\right) - 0{,}675t \quad \text{if } e < 0{,}5t \text{ else use } e' = e$$

$$\sigma_{mc} = \frac{1{,}5P}{\pi t^2}\left((1+\nu)\ \ln\left(\frac{2b}{\pi e}\right) + k_3\right) \quad \text{At centre}$$

$$\sigma_m = k_2\,\frac{P}{t^2} \quad \text{Middle of edge a}$$

$$y_m = k_1\,\frac{Pb^2}{Et^3} \quad \text{At centre}$$

	a/b						
	1,0	1,2	1,4	1,6	1,8	2,0	3 ->
k_1	0,061	0,071	0,076	0,078	0,0786	0,0788	0,0791
k_2	0,754	0,894	0,962	0,991	1,000	1,004	1,008
k_3	-0,238	-0,0078	0,011	0,053	0,068	0,067	0,067

5.3 *Optimization stages*

Material selection on raw material availability, Strength behavior analysis and accordingly material and sizes to be finalized. Size, weight reduction will be done once higher side of strength we can get in mechanical components. Multiple joint options and metal flame cut designs to be made for option and optimized solution will be finalized.

5.4 CAD IMPLIMENATTION

5.4.1Design implementation

1. Design of pivot joint, development will be done once design get finalised.
2. Structural calculations and deflection analysis.
3. CAE on ansys
4. FEA approach to make feasible working conditions.

5.4.2 *Design strategy of joint*

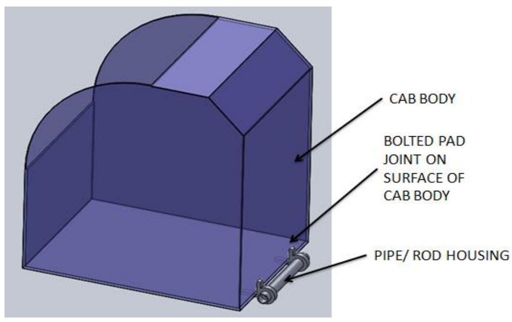


Fig.3 CAB tilting joint Design

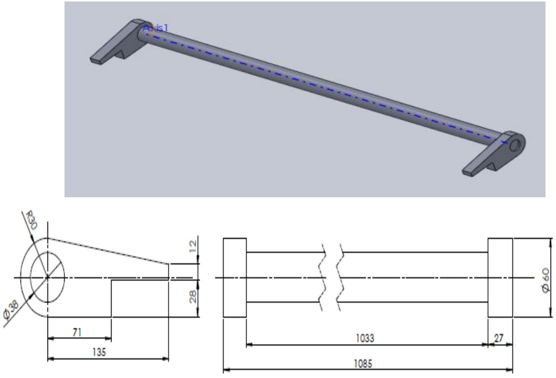


Fig.4 Tilting Rod

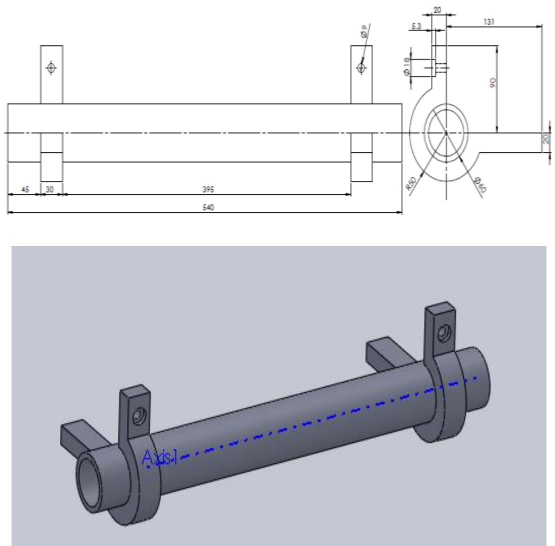
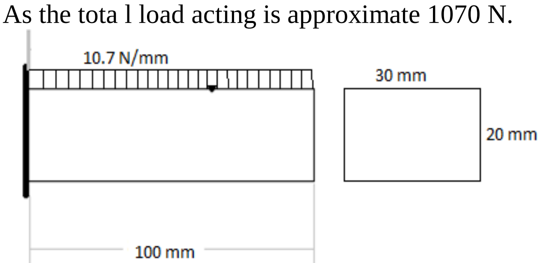


Fig.5 Rod covering bushing pipe assembly

5.5 Formulation of cab holder

Considering the bracket as a cantilever beam of rectangular cross section as shown in fig.
As the tota l load acting is approximate 1070 N.

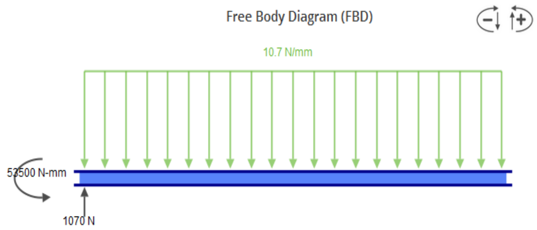


The load is acting on entire length,
Load acting per unit length is 10.7 N/mm
Hence, Load, P = 10.7 N/mm
We have,

$\frac{M}{I} = \frac{\sigma}{y}$............ (i)

Where,
M = bending moment
I = moment of inertia
σ= Stress

From bending moment diagram, maximum bending moment is 53500 Nmm.



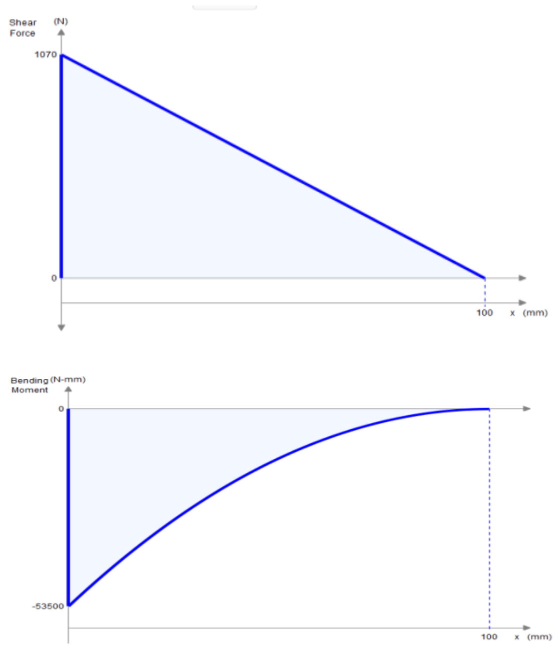


Fig.6 Shear force Diagram and bending moment Diagram

Moment of Inertia (I)

$$I = \frac{b\,d^3}{12}$$

$$I = \frac{30 \ x\ 20^3}{12}$$

I = 20000 mm^4

y = 20 /2
y = 10 mm
From equation (i),

$$\frac{M}{I} = \frac{\sigma}{y}$$

$$\frac{53500}{20000} = \frac{\sigma}{10}$$

$\sigma = 26.75\ N/mm^2$

Hence maximum bending stress developed is 26.75 MPa.

Maximum Deflection

$$\delta = \frac{PL^4}{8EI}$$

Where,

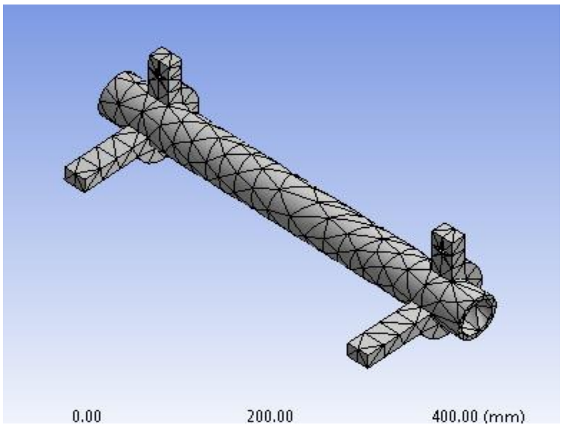
E = Modulus of Elasticity = $2 \times 10^5\ N/mm^2$

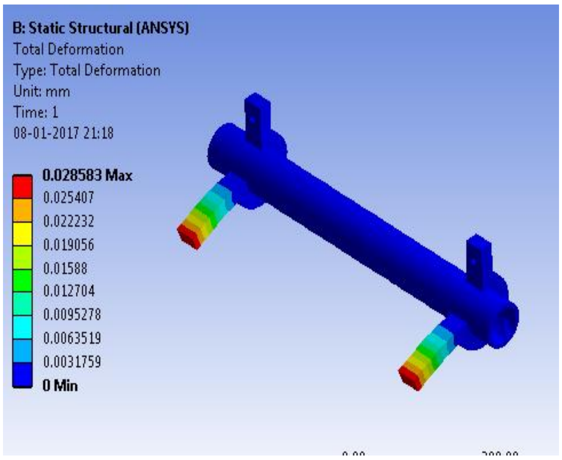
$$\delta = \frac{10.7 \; x \; 100^4}{8 \; x \; 2 \; x \; 10^5 x \; 20000}$$

$$\delta = \; 0.0033 \; mm$$

Hence Maximum Deflection is 0.0033 mm.

6. CAE Validation



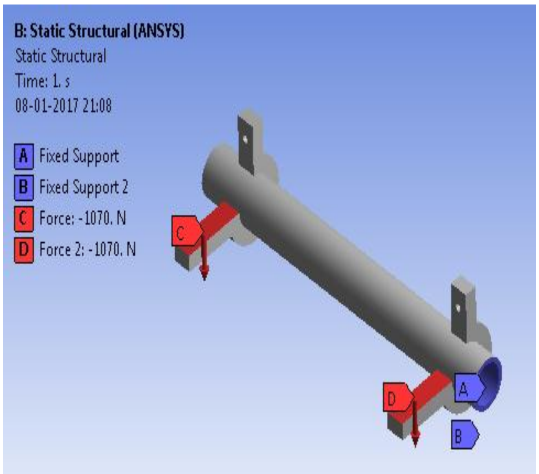


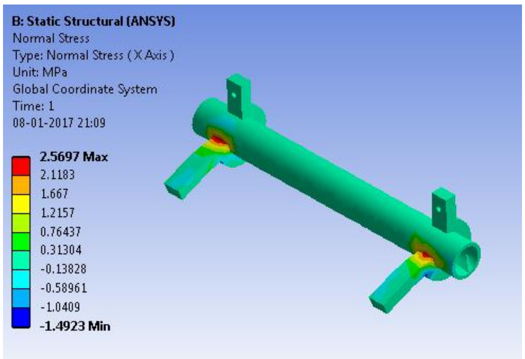
Details of "Mesh"

Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Si...	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge L...	20.0 mm
Inflation	
Use Automatic Te...	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorit...	Pre
View Advanced ...	No
Advanced	
Shape Checking	Standard Mechanical
Element Midside ...	Program Controlled
Straight Sided El...	No
Number of Retries	Default (4)
Rigid Body Beha...	Dimensionally Reduced
Mesh Morphing	Disabled

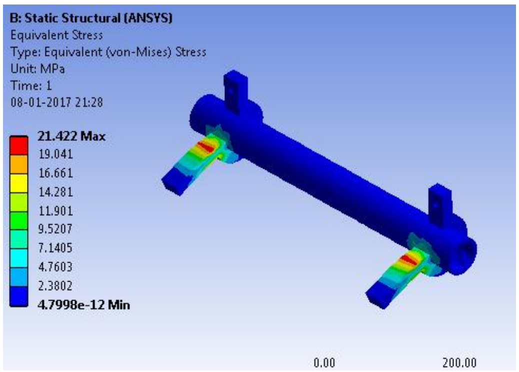
Meshing details

214 kg load is coming vertically,
114 kg fabricated and other modules weight and 100 kg of operator's weight.
So load = **2140 N**
On one side holder bracket consider half load as two holder brackets we have used as shown





Deformation 0.02 mm



Stress we got 21.42 mpa

6.1 *Material selection*

Material to be used is AISI316 as cab can work in fluid medium so other parameters of supporting machineries are also get contact with wet medium hence here anticorrosive material to be applied everywhere. Also ergonomically it should be usable so SS medium is only the option.
AISI316 grade of stainless steel

Material selected

Grade	Design strength (N/mm²)	Ultimate tensile strength (N/mm²)	Young's Modulus (N/mm²)	Elongation (%)
Stainless steel				
304 (1.4301)	210	520	200 000	45
316 (1.4401)	220	520	200 000	40
Carbon steel				
S275	275	410	205 000	22
S355	355	490	205 000	22

Material selected SS 316

No limitations on thickness in relation to brittle fracture apply to stainless steel; the limitations for carbon steel are not applicable due to the superior toughness of stainless steel. The austenitic stainless steel grades do not show a ductile-brittle impact strength transition as temperatures are lowered. Stainless steels can absorb considerable impact without fracturing due to their excellent ductility

ACKNOWLEDGMENT

We would like to thanks Dr. P.M. Jawandhiya (Principal PLITMS Buldana) for his valuable guidance, which leaded us to study hard and effectively. Also thanks to Prof K. R. Sontakke, who contributed greatly to the setting up the extent and success of this study, for his support in providing Necessary literature also thanks to Prof.A.P.Bawane, for giving their valuable Suggestions and time. We are also grateful to Mr. Ashish Dhakole (senior engineer Y.D.CORE Technology, Pune) for giving me great opportunity to work on this project. Also thanks to all my colleagues for their significant contributions also thanks to all authors for giving me best past technical knowledge and reference about this topic. Finally, our special thanks go to parent for their vast support for encouraging us, and great patience along the study.

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