

# Design and Analysis of Hook Nut for a 75 Tonne EOT-Crane

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## ABSTRACT

Electric Overhead Travelling (EOT) Crane is used to carry a heavy load in the industry. Due to the convention of heavy metal sat very high temperature, a Foundry requires cranes to transfer loads from one point to another. This makes the EOT cranes most widely and extensively used element in the industry. Every year the Crane is tested of its load caring capacity and only then it is inducted for use in the Foundry Shop. All the accidents that have taken place in the past were due to the negligence of the crane operator but, no design failure has been recorded until now. This article comprises of all the study, design calculations and load analysis of forces exerted on the Hook nut of the 75 Tonne EOT Crane.

**KEYWORDS:** Hook Nut; Load Analysis; EOT Crane; Design Failure; Forces

**How to cite this paper:** Neeladri S Bose | Dadaso D Mohite | Vibhor Chaturvedi | Harsh Toraskar "Design and Analysis of Hook Nut for a 75 Tonne EOT-Crane" Published in International Journal of Trend in Scientific Research and Development (ijtsrd), ISSN: 2456-6470, Volume-5 | Issue-1, December 2020, pp.536-540, URL: www.ijtsrd.com/papers/ijtsrd35840.pdf



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## I. INTRODUCTION

Cranes are industrial machines that are mainly used for material movements in construction sites, production halls, assembly lines, storage areas, power stations and similar places. The design features of crane vary widely according to their major operational specifications such as type of motion of the crane structure, weight and nature of the load, location of the crane, geometric features, operating regimes and environmental conditions.

The Comparative study by G.Bhagyaraj et al.[1] has given an overview of the CAD-CAM Model of the Nut part that is to be studied. The Study from V.B. Bhandari[6] gives the exact method for calculating stresses in Threads. This reference helped to locate and calculate the stresses of threads of the Hook Nut. Chetan N.Benkar et al. [11] worked on the crane hook for the optimization and its studies. He had estimated the stress pattern of crane hook in its loaded condition by preparing a solid model with the help of ANSYS14 workbench. The study by Osman Ashraf Ansari et al.[12] shows the stress pattern of crane hook in its loaded condition, a solid model of crane hook is prepared with the help of CMM and CAD software, then the real-time pattern of stress concentration in the 3D model of the crane hook is obtained. But all the Research articles were mainly focused over Crane Hook Block design.

The need for the design of the hook nut arises because of the excessive loading applications that are performed on the EOT Crane due to which it can be determined whether the design is safe to proceed or not. Thus, by designing this

hook nut, it can be recognized about different forces that are acting on the Hook Block. The Hook block design deals with the forces acting on the Hook nut which tells about the loading structure of the whole design.

The nut is needed to design in such a way that the loading is done safely, and no fatigue occurs in the whole system. Issue of external factors on hook nut: Hook nut comprises of various stresses acting on it when a load is applied. Compressive stresses on screw threads also make a factor for its load-carrying capacity. Therefore, the design of a hook nut will be discussed in this paper. The Hook Nut Analyzed is of the 75 Tonne EOT Crane of Foundry at Bhilai Steel Plant.

## II. Methodology

This part will discuss about the process of including the fabrication of the design of Hook Nut. It includes the concept behind design and its sketches. Thus, it gives the Final concept behind the design. Drawings were made in AUTOCAD which gave a clear picture on the design part. After this Final Concept modeling can easily be done on SOLIDWORKS. After then Analysis part can be proceeded in the ANSYS WORKBENCH. The Calculation part then deals with the conventional approach towards the parameters of the Hook Nut. After the calculation and the design, we can obtain the result whether the design is sustainable for industrial use or not. For 2-D drafting AUTO-CAD is used, for 3-D Modeling, SOLIDWORKS is used and for the Analysis, ANSYS is used. Methodology thus

includes all the important parts of the design and its application which make the research on design possible.

$$\text{Compressive Stress} = \frac{\text{Force}}{\text{Area}} = 67.312 \frac{N}{mm^2} = 67.312 \text{ MPa.}$$

Allowable Stress,  $\sigma_t = 140 \text{ MPa}$ .

Since the Compressive Stress is less than Allowable Stress, Therefore Design is safe.

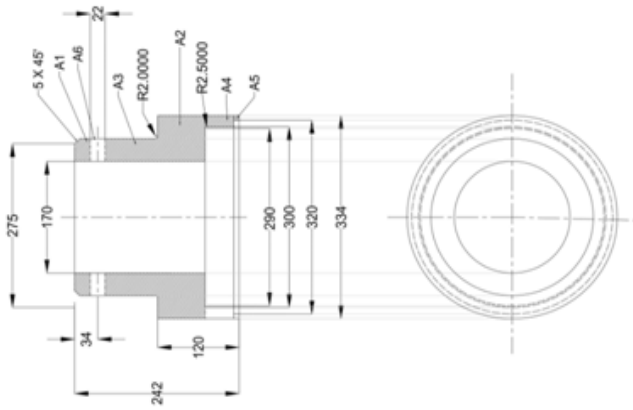


Fig. 1: Sketch of Hook NUT of a 75 Tonne EOT Crane

**Design of Hook Nut**

Design of Hook Nut includes the analysis of the Hook Block which shows where the forces are acting in the whole system. The Design will consist of both calculations and the CAD-CAM Model. The Design includes all the parameters and dimensions required for the analysis. Thus, the role of ANSYS comes into the picture through which analysis of the Nut will be done.

**Material Used:** Steel 45C8 (IS 2004-91)

**Stresses on nut:** There are two stresses

- A. Compressive Stress
- B. Torsional Stress

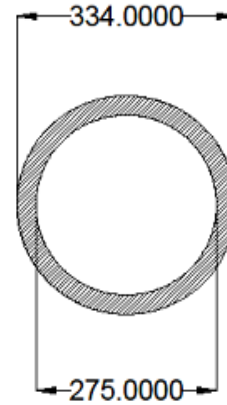


Fig. 3: Area under Compressive Stress (Half Scale)



Fig. 4: Nuts for a different type of Hook Crane Capacity



Fig 2: Hook Block of 75 Tonne EOT Crane (BSP, SAIL)

**III. Design Calculations**

**A. Compressive Stress:**

Mass = 75 Tonne = 75000 Kg  
 Force = mg = 75000 × 9.81 = 735750 N = 735.75 KN

D1=334mm, D2= 275mm

Diameter under Compressive Stress = D1 – D2  
 = 334mm – 275mm = 59 mm

Since, scaling in drawing=1:2.

Therefore, diameter= 118 mm.

Area of shaded region shown(fig. 3) is calculated.

$$\text{Area} = \frac{\pi}{4} D^2 = 10930.34 \text{ mm}^2.$$

**B. Load Analysis of Screw Threads**

It consists of two factors i.e. internal and external forces. When a load is applied on the Hook Block, Hook nut comprises of the various load acting as it is already discussed. Under this analysis due to Internal Forces Shear failure and Crushing Failure may be evaluated. And due to External forces Torsional and Tensile Forces are calculated. As we know that the internal threads of the Hook nut are connected to the outer threads of the Hook. Here the Load on screw threads of Hook nut especially is analysed. This gives an overview of the overall design safety or failure of the screw threads.

- |                            |                            |
|----------------------------|----------------------------|
| A. Due to Internal Forces: | B. Due to External Forces: |
| 1. Shear Failure           | 1. Torsional Stress        |
| 2. Crushing Failure        | 2. Tensile Stress          |

**1. Shear Failure:**

$$\text{Shear Stress, } \tau = \frac{4P}{(\pi d c)} = \frac{4Pb}{h * \pi d c}$$

(Height of thread,  $h = n \times b$ ); Where, Number of threads,  $n=32$ ; Pitch,  $b=12 \text{ mm}$ ;  $h=384 \text{ mm}$ ; Diameter to be considered,  $dc= 340 \text{ mm}$ ;

$$\therefore \text{Shear Stress, } \tau = \frac{4 \times 735750}{34163.2} = 86.145 \text{ MPa}$$

As the shear stress is less than Allowable Stress, the Design is safe.

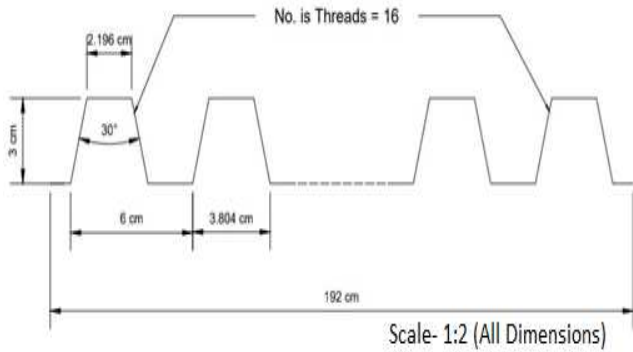


Fig. 5: Trapezoidal Threads (Full Scale)

**2. Crushing or Compression Stress ( $\sigma_{crushing}$ ):**

The Crushing or Compression Stress can be calculated by;

$$\sigma_{crushing} = P/A_c'$$

Where, Applied Load,  $P = 735750 \text{ N}$ ; Outer diameter,  $d_1 = 320 \text{ mm}$  (Half scale); and Inner diameter,  $d_2 = 300 \text{ mm}$  (Half scale)

Area of Shaded resion (Fig. 6),  $A_c = \frac{\pi(d_1^2 - d_2^2)}{4}$

$$\therefore \sigma_{crushing} = \frac{4P}{\pi(d_1^2 - d_2^2)} = 75.585 \text{ MPa.}$$

$\therefore$  Crushing stress that is applied on hook nut is  $75.585 \text{ MPa}$ . Since the Crushing stress is less than Allowable Stress, the Design is safe.

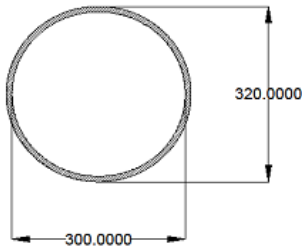


Fig. 6: Crushing Area (Half Scale)

**3. Torsional Stress:**

The Torsional Stress can be calculated by;

$$\frac{\tau_{max}}{R} = \frac{T}{J} = \frac{G\theta}{L}$$

But, Polar Moment of Inertia,

$$J = \frac{\pi(R_o^4 - R_i^4)}{2} = 527120742.7 \text{ mm}^4;$$

Outer radius,  $R_o = 167 \text{ mm}$ ; and Inner radius,  $R_i = 145 \text{ mm}$ ; the angle of twist,  $\theta = 2^\circ$  (Std. value);

Torsion,  $T = [(80 * 2 * 527120742.7) / 192]$

$T = 439267285.6 \text{ N-mm}$ ;

Radius of Curvature,  $R = 170 / 2 = 85 \text{ mm}$ .

$$\therefore \text{Max. shear stress, } \tau_{max} = T * \frac{R}{J} = 70.833 \text{ MPa}$$

Since, Maximum Torsional Shear Stress is less than Allowable Torsional stress, Design is safe.

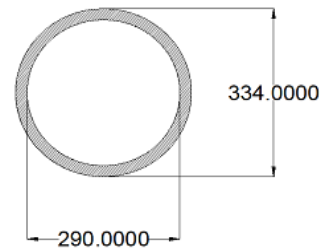


Fig. 7: Torsional Area (Half Scale)

**4. Tensile Stress:**

$$\text{Max. Tensile Stress, } \sigma_{tmax} = \frac{\sigma_t}{2} + \frac{\sqrt{(\sigma_t)^2 + 4(\tau_{max})^2}}{4}$$

Max. Tensile Stress,  $\sigma_{tmax} = 119.79 \text{ MPa}$ .

Since the Tensile Stress is less than Allowable Stress, Therefore the Design is safe.

**5. Stress due to Total Shear Area**

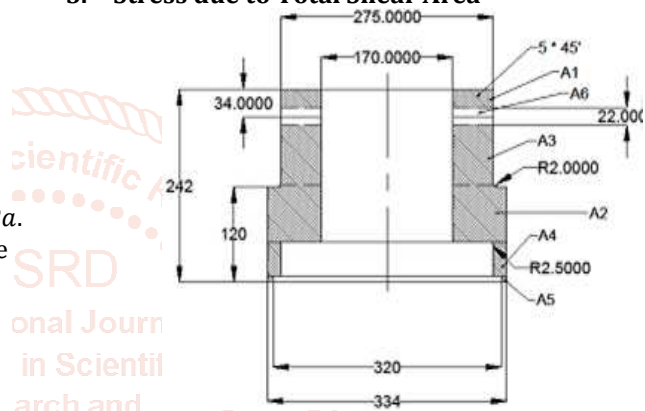


Fig. 8: Total Shear Area (Half Scale)

It is necessary to calculate the shear area and for that, The cross-sectional area is divided into 6 different areas and their sum will make up the total shear area into consideration. Thus, the shear stress can be calculated and to get the output.

$$A_1 = 52.5 \times 23 = 1207.5 \text{ mm}^2, \quad A_4 = 1428 \text{ mm}^2, \\ A_2 = 5740 \text{ mm}^2, \quad A_5 = 112 \text{ mm}^2, \\ A_3 = 4042.5 \text{ mm}^2, \quad A_6 = 1155 \text{ mm}^2$$

Total Area taken into consideration =  
Sum of area at all section =  $11375 \text{ mm}^2$

$$\therefore \text{Total Shear Area} = 11375 \text{ mm}^2$$

shear stress is Calculated by dividing Load by Total Shear Area.

$$\therefore \text{Shear Stress} = \frac{P}{\text{TotalArea}} = 64.68 \text{ MPa}$$

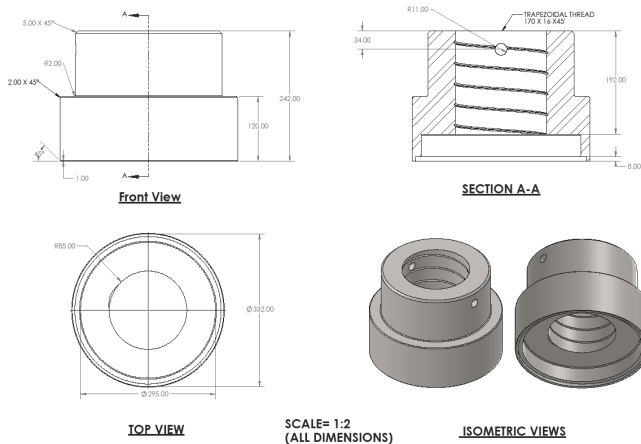
Since the Total Stress by Shear Area is less than the Allowable Stress, the design is safe.

**IV. ANALYSIS OF HOOK NUT ON CAD-CAM**

The Hook nut of 75 EOT Crane is developed in SOLIDWORKS. Further analysis of this model is to be done on ANSYS. The analysis will show the points or region where the design will failure break. The CAD-CAM Model gives us theraw idea of the study. The model is having Full-scale dimensions. This will give us a basic understanding of

“How a Hook nut meshes with a Hook?”, which will further give the basic understanding of Hook Block in a nutshell. Thus, comparing theoretical calculations with that of CAD-CAM and ANSYS model will become possible.

The Orthographic and Isometric views are shown in fig. 9, which gives picture about the Hook Nut Design and its depiction shows the nature of the same.



**Fig.9: Orthographic and Isometric Views of Hook Nut**

**A. Finite Element Analysis**

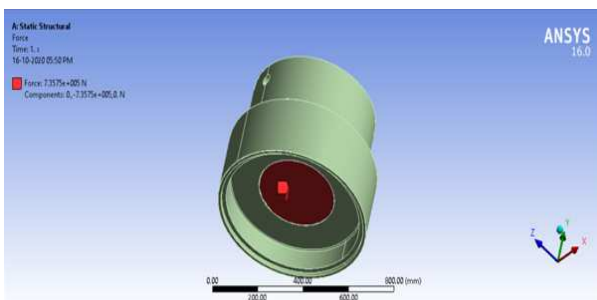
Finite element Analysis (FEA) is a computerized method for determining how a product reacts to real-world forces, vibration, heat, fluid flow, and other physical effects. FEM is the most widely used method for solving problems of engineering and mathematical models. It includes the use of mesh generation techniques for dividing a complex problems into small elements. It shows whether a product will break, wear out or work the way it was designed. It is called analysis, but in the product development process, it is used to predict what is going to happen when the product is used. The finite element analysis of the product is initially carried out before the fabrication of the final product or system.

**1. Stress test:**

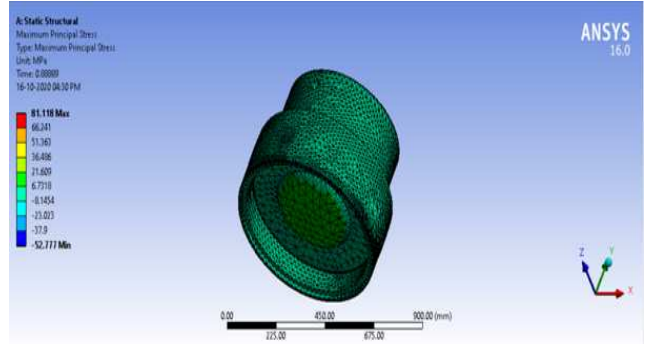
Values Taken:

- Load/Force applied : 735.75 KN.
- Type of support : Frictionless.

In the process of FEA, first subject the model (nut) by a load of 735.75 KN for it being the maximum amount of load the crane hook will bear under maximum loading condition. The threads of both the bolt and the nut are secured with the help of bonded contacts between the two surfaces. Applying respective friction-less support and the load supposedly in the area (on the bolt) highlighted in the figure. Thereafter record Maximum Principal Stresses and Total deformation of the nut.

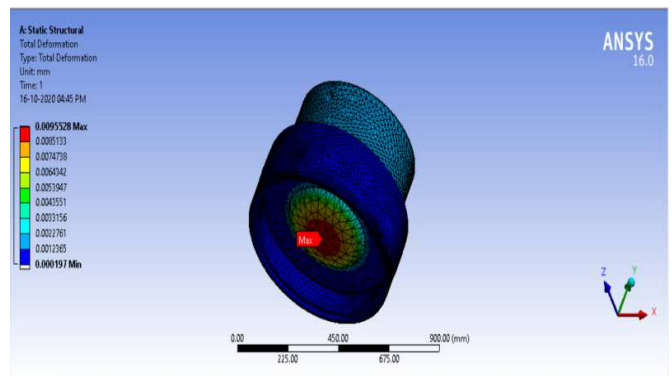


**Fig. 10: Occurrence of Compressive Stress**



**Fig. 11: Occurrence of Maximum Principal Stress in the Nut**

The recorded Maximum Principal Stress is 81.118MPa. Being close to the calculated value of 75.585 MPa. Yielding 7% error between the two values. The total deformation observed is displayed in the analysis figure 12.



**Fig. 12: Total Deformation**

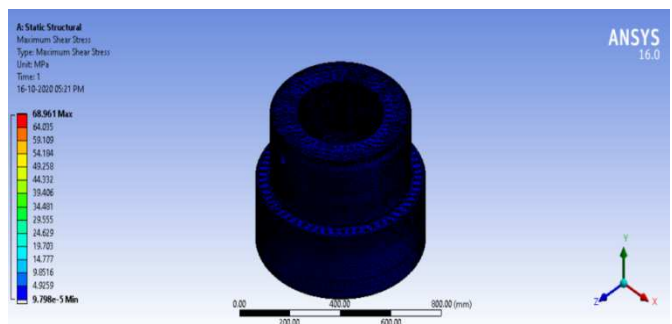
The maximum deformation is observed mainly in the region of the bolt where the load was applied.

**2. Shear Test:**

Values taken:

- Moment: 4.3927e+008 N-mm.
- Type of support: Fixed.

By applying a moment force along the outer surface of the nut body, one can get results for shear stress. It is shown in fig. 13.



**Fig. 13: Shear Stress in Nut**

The simulated generated value is 68.961 MPa. Compared to the calculated value of 70.833 MPa. Making the error as low as 2% between the two.

**V. RESULT AND DISCUSSION**

From all the above calculations it is clear that the hook nut is completely safe and it has no design complications in it. After analyzing the Hook Nut based on every subject to its

design. It has to be noted that though the Design is safe for Hook but it can fail because of some external factors. In the calculations, all external factors such as Operator's inaccuracy have been neglected, Failure of limit switches etc. Therefore this study will benefit only in the design point of view. Various sketches are also provided in this study to refer to the design calculations. This study is fully based on the theories of failure of a particular material. Thus making this a unique study on the concepts and applications in an Industry. When an E.O.T Crane carries the load the major or maximum stress is generated on the Hook Nut of the crane which is supporting the Hook block. In the Calculations, it is observed that the main load was acting on the Threads of the nut, which symbolizes it may be the first component to fail if the load is increased.

## VI. CONCLUSION

It is concluded that the Hook Nut of a 75 Tonne EOT Crane is completely safe to use in design point of view. But due to external factors, it may still be subjected to failure. Thus the hook nut can be used in the Industry.

## ACKNOWLEDGEMENT

We would like to express our special thanks of gratitude to all the employees at Bhilai Steel Plant as well as to Mr. K. K. Thakur (GM, Foundry Division, BSP) who gave us the golden opportunity to do this wonderful project on the topic "DESIGN AND ANALYSIS OF HOOK NUT OF 75 TONNE EOT-CRANE", which also helped us in doing a lot of Research and We came to know about so many new things, we are thankful to them.

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