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Design and Analysis of Electric Crane with 500Kg Material Handling Capacity

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***Abstract:** In industry material handling is very important. For that purpose efficient machine is needed. In modern days the electric crane plays important role to fulfil that requirement. It is economical, effective, less time consuming, more safety, and flexible for uses. In industry the safety is most important factor for any material handling machine which deals with the heavy load depending on the capacity. In electric crane the safety keeps on first priority. In this paper we design the electric crane which is more economical, less time consumable, with more safety and the easy to operate. It has two control units, one is at crane frame and one is at the remote location. This crane is able to operate by the single operator far from the crane and also can operate directly from the crane frame itself. Its lifting capacity is up to 500kg. according to load we carried out the load calculations for the crane frame or body & we created a 3D CAD model. Every component should able to deal with that load. To ensure that we carried out the analysis of the whole assembly in ANSYS 19.2. In analysis we found that the whole assembly is able to handle the load which we suppose to apply on it. We implement the pulley reduction to reduce the load on motor and gearbox as well. We tried to keep crane more compact in size so that it consume less space in industry.*
***Keywords:** Electric Crane, Design, Analysis, Material Handling, 500kg, Load*

I. INTRODUCTION

In this paper we design the electric crane which is more economical, less time consumable, with more safety and the easy to operate. It has two control units, one is at crane frame and one is at the remote location. This crane is able to operate by the single operator far from the crane and also can operate directly from the crane frame itself. Its lifting capacity is up to 500kg. For safety it has a ratchet to lock the reverse motion in case of any kind of machine failure. It has a pulley reduction to reduce the load on the motor and the gearbox. That pulley reduction also helps to improve the safety of the machine.

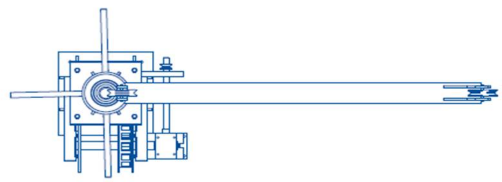
This crane is operated by the 1HP 3Phase induction motor. So it is more power efficient as it consumes less electricity that the other crane from the market. To reduce the speed of the motor and to increase the torque value we used a worm reduction gearbox. It has the reduction of 25. As we use the rope pulley reduction in crane It results it half load on the motor and the gearbox. The motor is connected to the gearbox and the gearbox is connected to the hoist directly. As motor start, the hoist shaft is starting to wound the rope so that the object starts lifting upward.

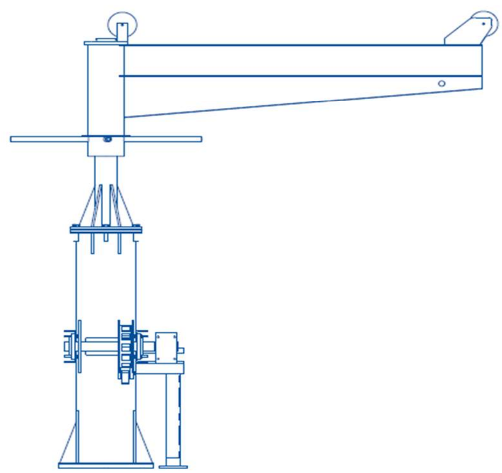
A. Parts of Crane

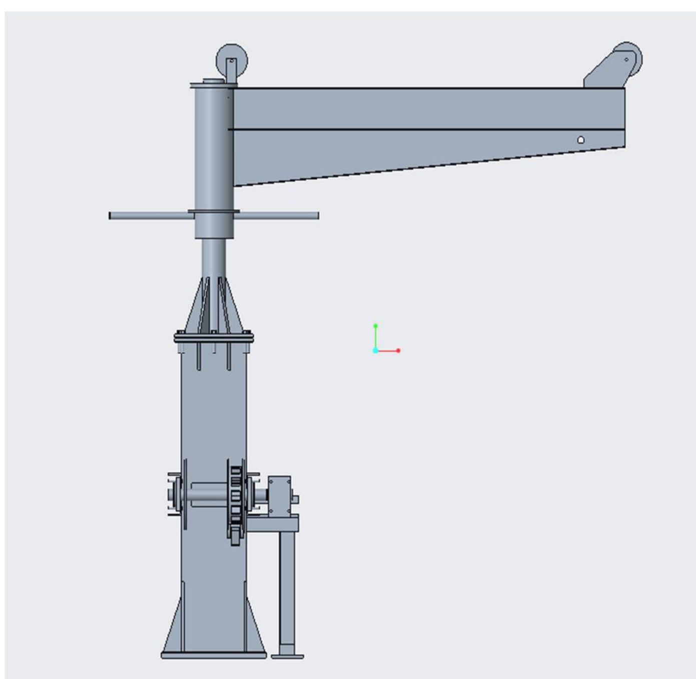
1) C-channel
2) Cylinder 1
3) Cylinder 2
4) I channel
5) Rib 1
6) Rib 2
7) Rib 3
8) Rib 4
9) Flange bearing
10) Ratchet
11) Hoist shaft
12) Bolt
13) Pulley
14) Gearbox

15) Support plate
16) Taper roller bearing
17) L channel
18) Hoist shaft
19) Gearbox input shaft

B. Crane







II. LITERATURE REVIEW

Mulugeta Tadesse, Tesfahun Meshesha[1] has described for many years cranes have been designed for lifting heavy objects with different capacities in different work sites. Portable cranes are one type of cranes designed for lifting objects which are beyond the capacity of human beings. Of this lifting operation by using portable and moveable cranes which has not been used before, they have identified that there is the need for using portable cranes to lift up objects that are beyond the capacity and difficult of human power. Thus this paper provides the design of each part of the portable crane. And the design analysis for each part is checked so that it is safe accordingly the size of each part of the crane.

Y. Torres et. [2], initially studied the probable causes of failure of crane hook. It includes the manufacturing and lifting of crane hook, experimental analysis mechanical behaviour of material of crane hook. It was concluded that the brittle fracture was originate from crack in the material.

E. Narvydas et. al [3], investigated circumferential stress concentration factors with shallow notches of the lifting hooks of trapezoidal cross-section employing finite element analysis (FEA).The stress concentration factors were widely used in strength and durability evaluation of structures and machine elements. The FEA results were used and fitted with selected generic equation. This yields formulas for the fast engineering evaluation of stress concentration factors without the usage of finite element models.

Rashmi Uddanwadiker [4], studied stress analysis of crane hook using finite element method and validated results using Photo elasticity. Photo elasticity test is based on the property of birefringence. To study stress pattern in the hook in a loaded condition analysis was carried out in two steps firstly by FEM stress analysis of approximate model and results were validated against photo elastic experiment. Secondly, assuming hook as a curved beam and its verification using FEM of exact hook.

Spasoje Trifkovic et. al [5], this paper analysis the stress state in the hook using approximate and exact methods. They calculated stresses in various parts of the hook material firstly by assuming hook as a straight beam and then assuming it as curved beam Analytical methods were used with the help of computers, using FEM.

Bernard Ross et. al [6], this paper describes the comprehensive engineering analysis of the crane accident, Under taken to disprove the Mitsubishi theories of failure as confirmed by jury verdict. Crucial role of the SAE J1093, 2% design side load criterion and Lampson's justification or an 85%crawler crane stability criterion were presented.

III. PROJECT OUTLINE

A. *Material Selection*

The selection of material is very important thing in order to design any mechanical components. The recent trends towards optimizing the mechanical components through continuous design modification needs lots of data to maintained, also during the design proper material selection is also needed. For our design we have used Mild steel with IS 2062 grade. The basic mechanical properties are mentioned in below table.

Table1.1 Material properties

Material	Tensile strength	Yield strength	Elongation
IS 2062	410 Mpa	230 Mpa	23%
EN8	850 Mpa	465 Mpa	16%

Table 1.2 Bearing selection

D	d	T	Static load rating (KN)	Dynamic load rating (KN)
80	125	23	115	156.3
90	125	18	17	20.2

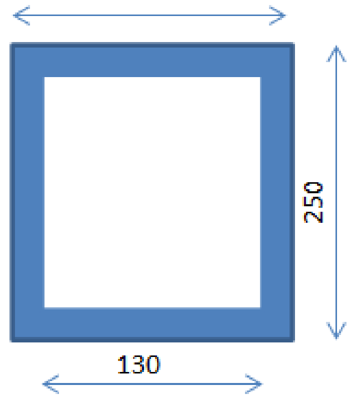
B. *Basic Calculation of Crane*

1) Total Lifting Capacity (W) = 0.5 ton

$$= 0.5*10000$$

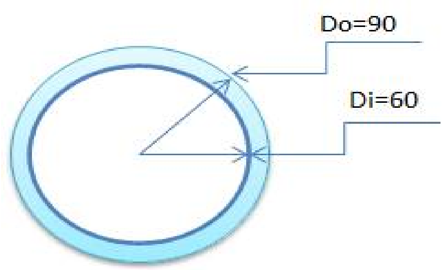
$$= 5000 \text{ N}$$

2) Stationary part 1 (Cylinder 1) [10]



Deflection calculations,
Ixx = (BD^3)/12 - (bd^3)/12
Ixx = (160*250^3)/12 - (130*210^3)/12
Ixx = 108005833.33 mm^4.
EI = 22681225.3 Nm^2.
Deflection is 0.5 mm.

3) Stationary part 2 (Cylinder 2) [10]



Deflection calculations,
Ixx $=\pi/64*(do^4 - di^4) = \pi/64*(90^4-60^4)$
Ixx = 2584450.83 mm^4.
EI = 542734.5 Nm^2.
Deflection is 6 mm.

4) I channel
Deflection calculations are,
Ixx1 = $(BD^3)/12 + A_1h_1^2$
Ixx1 = $(100*5^3)/12 + 500*72.5^2$
Ixx1 = 2629166.6 mm^4
Ixx2 = $(BD^3)/12$ = 2286666.66 mm^4
Ixx3 = $(BD^3)/12 + A_3h_3^2$ = 2629166.6 mm^4
Ixx = 7545000.01 mm^4
EI = 1584450.002 Nm^2.
Deflection = 7 mm.

5) V-belt drive [8] [Page no. 12.13]
Power= 1 hp = 0.7465kw
N1=N2= 1420rpm
Design power (PD)= P*Fa = 0.7465KW

Take Z-type of v-belt cross section from design power and highest RPM
Dimensions of Z-type belt
Wp=8.5 mm
W=10 mm
T=6 mm
A=40 degree
D1=D2=50mm
C=center distance=350 mm (assumed)

Calculation for pitch length (Lp)
$Lp=2C+Pi/2(D1+D2)+(D2-D1)^2/4C$
Comparing with standard pitch length
Lp= 780 mm

No of belts=PD/pr*Fc*Fd
= 0.9
Therefore,
We will required 1 no of v belt.

6) Bolts [9]
P = 7500 N , l = 1500mm
Permissible tensile stress = (ρt max)= 60N/mm²
l1=50 mm, l2 =200mm , l3 =350mm
As we know : δ1 α l1 , δ2 α l2
P1 = cl1= P2=P3
P4=P5=cl2
P6=P7=P8=cl3
C= load per unit in bolt
Equating the moment of resisting forces
$C= p \times l/(3l_1^2+2l_2^2+3l_3^2)$

Maximum force will act l3 distance its
On P6, P7, and P8
$P6= p \times l/(3l_1^2+2l_2^2+3l_3^2) \times l3$
P6= 8653.84N
P6 = A x (ρt max)
$A=144.1307 \text{ mm}^2$
Comparing with standard chart,
Bolt selected is M16.

7) Shaft [ASME Code] [9]
$\tau_{max} = (16/pi*d^3) \times \sqrt{((kb \times Mb)^2 + (kt \times Mt)^2)}$
$\tau_{max}= 0.2$
=0.18
$Sut=850 \text{ N/mm}^2$
$Syt=650 \text{ N/mm}^2$
$\tau_{max}= 0.3 \times 650 = 195 \text{ N/mm}^2$
$\tau_{min}=0.18 \times 850= 153 \text{ N/mm}^2$
$\therefore \tau_{per} =114.75 \text{ N/mm}^2$
Torque (Mt)= 4 Nm

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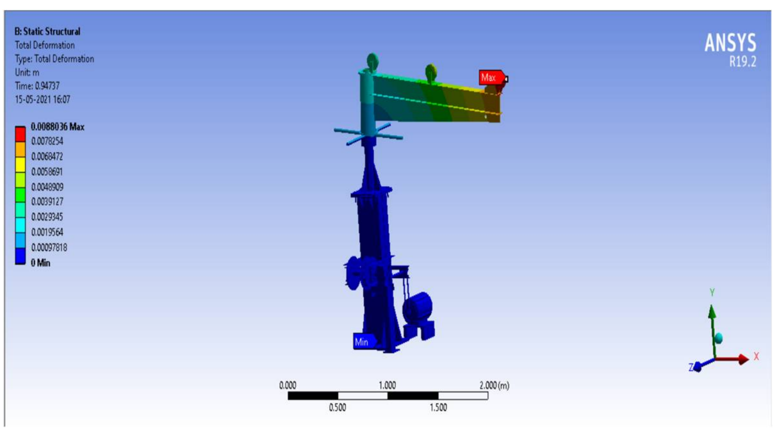
kb= 1.5, kt= 1 (from table 9.2 V.B. Bhandari)
$d^3 = (16/ \tau_{per} *pi) \times \sqrt{((kb \times Mb)^2 + (kt \times Mt)^2)}$

Bmd md solids
Mbmax = -4
$d^3 = 320.05$
d=6.8
d≈10mm

BMd md solid
Mt = 120 Nm
Mbmax = 225.88 Nm
$d^3 = (16/ \tau_{per} *pi) \times \sqrt{((kb \times Mb)^2 + (kt \times Mt)^2)}$
$d^3=5326.03$
d= 17.46mm
D ≈20 mm

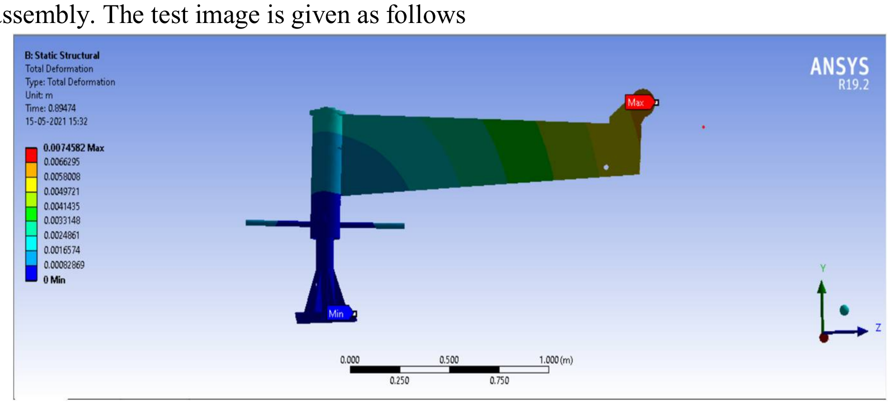
C. Finite Element Analysis

In this chapter we carried out the total assembly analysis of the machine. The aim of this test is to find out our design is safe or not. For this analysis we use ANSYS 19.2 software. For this machine the main factor is weight handling. So we carried out deformation test for the same. For this test first we assign the fixed support which is on the ground. The load is applied on the pulleys. The results are as we expected the machine is safe and has better strength to handle 500kg load.
The test image is as follows.



1) Total Deformation test

For the individual subassembly deformation test we carried out deformation test on upper assembly as it get more affected by the load than the lower subassembly. The test image is given as follows



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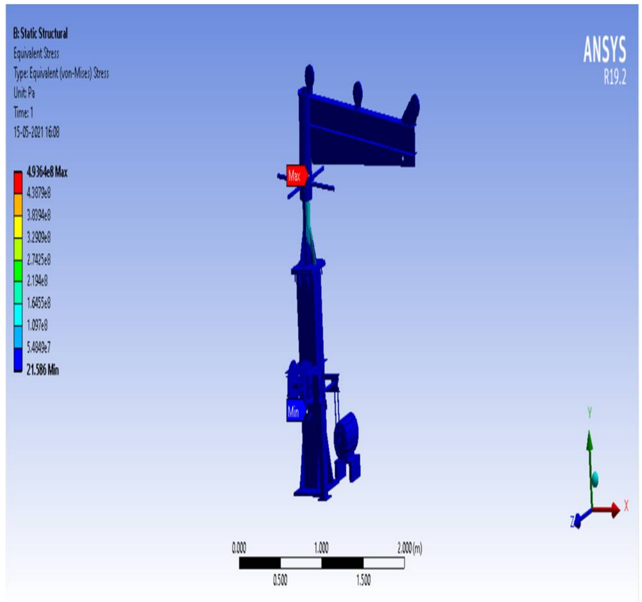
2) Upper Assembly Deflection Test

Along with the deformation test we carried out the equivalent stress test to find the part which is under the maximum stress. We carried out test by the Von-Mises stress theory. The test image is as given below.



IV. RESULT AND DISCUSSION

As we carried out the different calculations for the various parts of the machine to verify the safety. From the calculations we get the deformation value for the I channel is 7 mm (without rib). after that we did the ansys of the whole assembly as well as upper subassembly consisting of I channel. The result for the Upper subassembly or the I channel from the both test is approximately same which are 7 mm by calculations and 7.4mm by the ansys. But in ansys we include the rotating cylinder and the rib for analysis and in numerical calculations we carried out the calculations only on the I channel. The maximum deformation considering the complete assembly in ansys is 8.8mm.

V. CONCLUSIONS

Deformation (maximum) of I channel is 7mm and the deformation (maximum) of Upper subassembly (including I channel, Rotating cylinder, rib for I channel, Stationary part 2 with ribs, etc.) is 7.4mm. the deformation is very low and negligible. The maximum deflection in the whole assembly is 8.8mm which is also very low as compare to the size of the machine.

From the result we get the deformation is very low and safe so we conclude that the design can be consider for the manufacturing.

REFERENCES

[1] R.M. Maccrimmon ,Crane-supporting steel structures.

[2] Mr. A. Gopichand, Ms. R. V. S. Lakshmi, Mr. B.Maheshkrishna "Optimization of design parameter for crane hook using taguchi method" in international journals of innovative research in science ,engineering and technology, vol. 2, Dec 2013, ISSN: 2319-8753.

[3] Chetan N. Benkar, Dr. N. A. Wankhade "Finite Element stress Analysis of Crane Hook With Different Cross Sections" in International Journal For Technological Research In Engineering, volume 1, Issue 9,May-2014, ISSN 2347- 4718.

[4] M. Shaban, M. I. Mohamed, A. E. Abuelezz and T. Khalifa,"Determination Of Stress Distribution in Crane Hook by Caustic" in International Journal of Innovative Research in science, Engineering and Technology, Vol. 2 Issue 5, May 2013, ISSN: 2319-8753.

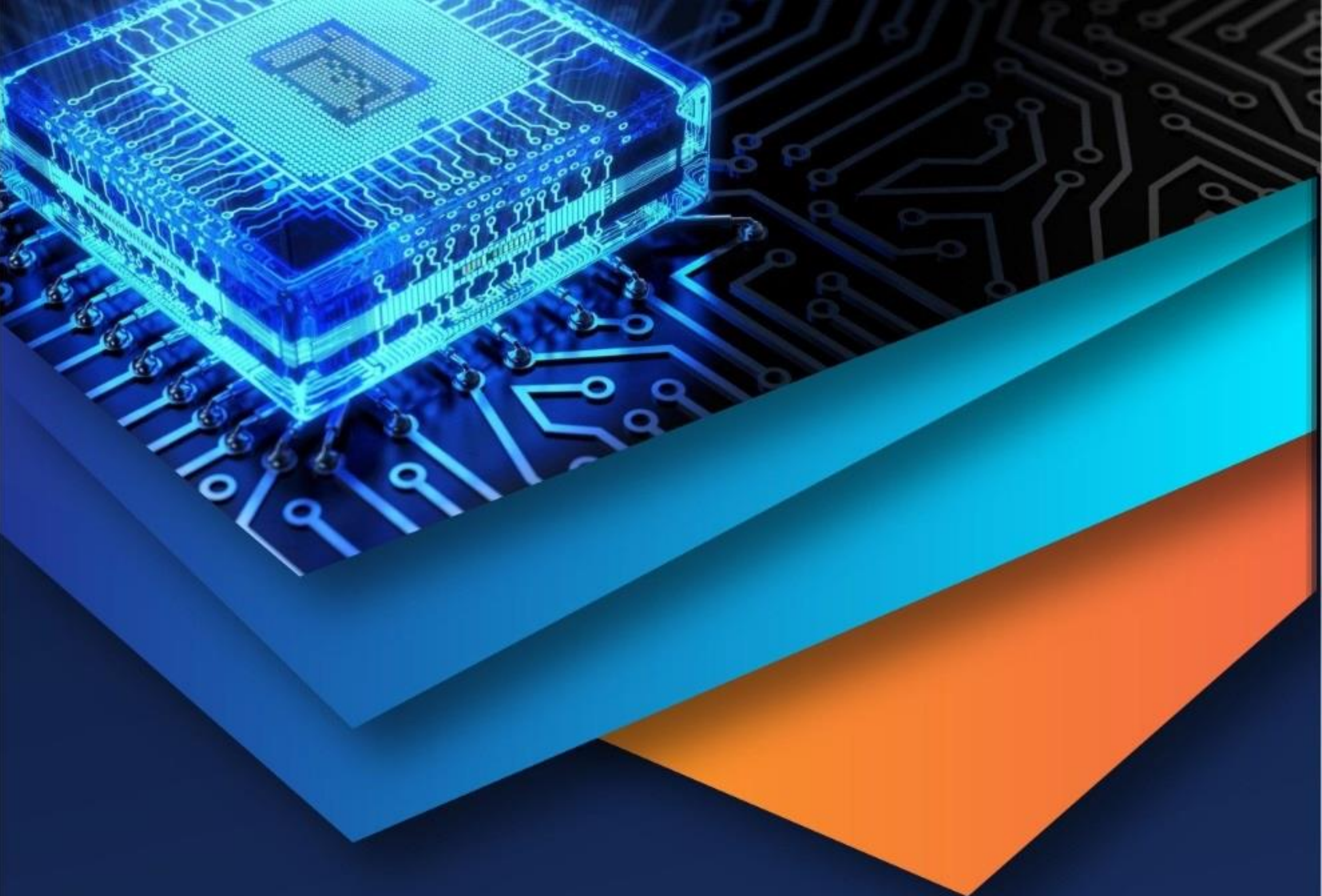
[5] Apeksha. K. Patel, Prof. V. K. Jani, "Design and Dynamic Analysis of 70T Double Girder Electrical Overhead Crane" in Journal of Information, Knowledge and Research in Mechanical Engineering Vol.2, Oct-2013, ISSN- 975-668X.

[6] Pradyumna Kesharimaharana, "Computer Aided Analysis and Design of Hoisting Mechanism of an EOT Crane", .Thesis, National Institute of Technology Rourkela, May-2012.

[7] V. B. Bhandari , Machine Design Data Book, MacGraw Hill Education Pvt Ltd , ISBN- 978-93-5134-284-7, page no- 13.17

[8] V. B. Bhandari, Design of Machine Elements , MacGraw Hill Education Pvt Ltd.

[9] S. S. Ratan , Strength of Material, MacGraw Hill Education Pvt Ltd.



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