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DESIGN CALCULATION FOR MULTIPLE PULLEY CRANE

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Abstract—Crane is a reliable component for lifting load in industries. Cranefails due to high friction in between wire rope and pulley. It leads to failure in gear box or it may increase power requirement of crane to lift loads. It is necessary for the crane to lift the load with minimum effort and minimum friction between the mating surfaces. Multiple pulleys in a crane reduce the friction; it is economical and effective. This work focus on design, and fabrication of mini crane with fixed multiple pulleys. Based on the design calculations and analysis, a prototype crane was fabricated and analyzed. Multiple pulley crane design calculations, part and assembled drawings are presented in this paper.

Keywords—Crane, friction, multiple pulleys

I. Introduction

Cranes load and move materials, multiple pulley cranes are the modified model of a tower crane with fixed and movable pulleys which helps in reducing the effort to lift the load. The fixed pulleys help in uniform distribution of load throughout the arm of the crane.

Cranes are a central component in engineering industries and are related with large number of threats with its operations in workshop and also there are numerous types of hoisting machine available depending upon basic design, mechanism and working, so it is also make problem to handle these type of hoisting machinery in workshops. A floor crane is equipment, generally equipped with a lifting mechanism such as a hoist with wire Ropes or hydraulic cylinder that can be used both to lift and lower materials and prior to the horizontal movement of the crane. A mobile floor crane is equipment with portable features which makes it admirable and recommended for both indoor (workshop/ warehouse) and outdoor purposes, for the sole aim of lifting and moving heavy materials from one place to another .The cranes reduce the workers fatigue and increase the overall efficiency of production processes with good safety. Material Handling includes movement of material, goods and products for storage, control and protection. Jib crane consists of an inclined member that can rotate about pillar at center and suspend the load from the outer end of the inclined member.

Table 1. Material Selection

Name	Material
Working arm & counter arm	Mild steel
Pulley	Nylon plastic
Supporting beam	Mild steel
Winding drum	Mild steel
Wire rope	Rope steel
Hook	Forged iron [4]

Electric motor is power source for loading operation[5].
It has following specifications

Electric Motor

Phase	Single phase
Voltage operation	115-230
RPM	1420
H.P	1/3

II. Structure Design

Beam Design

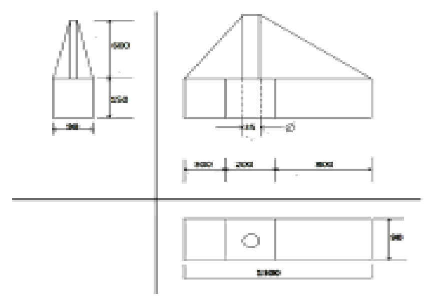

Figure 1: Beam Design

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[1]Corresponding Author

Column beam:

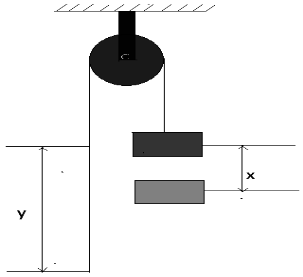

Figure 2: Design of Column

Base Design:

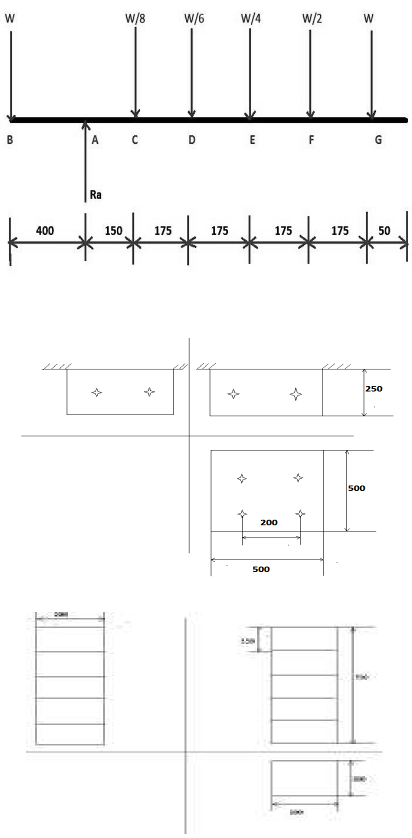

Figure 3: Base Design

Design Calculations:

Velocity ratio:

Velocity ratio defined as the ratio between length of rope pulled into height at which weight is lifted.

$$R=y/x \quad (1)$$

Here R=2

Weight/mm:- 0.8 g/mm

Self weight of the beam structure

= 4160(length rod) + 480(cross rod) + 288

= 4.9 kg

Mass of the pulley	= 290 g / pulley
Six no pulley	= 1740 g
Motor and roller	= 1000 g

Now here to calculate the counter balancing weight of the structure under the no loading condition.

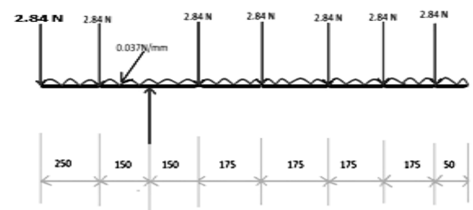

Fig 5: Loads Acting On The Beam

Under balancing or equilibrium condition,

$$\Sigma Ma=0 \quad (2)$$

X= 36.93N Mx=36.93/9.81=3.76Kg

Total mass of beam (M)= 11.4 kg

Stability or equilibrium: By adding the counter balancing weight 4 kg on left side of the structure.

Calculation of beam moment – moment area method.

W – Weight is to be liftedC, D, E, F – point at which this weight is distributed.

We know that,

$$\sum V=0 \quad (3)$$

$$RA=W+W/8+W/4+W/2+W+40 \quad (4)$$

$$RA = 3.04W\ N$$

Bending moment calculation at	Value of the Bending moment
G	0
F	-175W
E	-437.5 W
D	-743.75W0
C	-1129.17 W
A	-1385.42 W
B	0

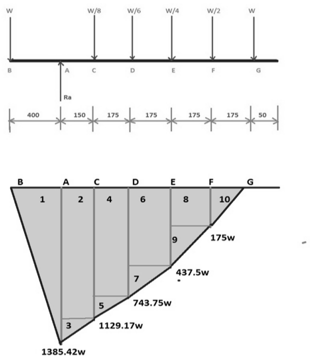

Fig 4: Bending Moment Diagram

Area of the beam diagram:

$$A=A1+A2+A3+A4+A5+A6+A7+A8+A9+A10 \quad (5)$$

Area of the triangle A = ½ bh(6)

Area of the rectangle

$$A = lb \quad (7)$$

Triangle	Area of Triangle
A1	277084W N mm^2
A2	169375.5W N mm^2
A3	19218.75W N mm^2
A4	130156.25W N mm^2
A5	33724.26W N mm^2
A6	76562.5W N mm^2
A7	267896.875 N mm^2
A8	30625W N mm^2
A9	22968.75W N mm^2
A10	15312.5W N mm^2

Using the above calculated individual area values final area is A = $1.043 * 10^6$N mm^2

Young's modulus for mild steel is noted from its standard value

$$E_{mild}= 2.1* 105 \text{ N/mm2} \quad (8)$$

Moment of inertia of rectangular hollow section is given by

$$I = \frac{1}{12}(BH^3 - bh^3) \quad (9)$$

$$I = 1.97*107 \text{ mm}^4$$

The center of gravity of the section

$\bar{x}$= 90/2

$\bar{x}$ = 45mm

According to "MOMENT AREA METHOD":

$$y=A\bar{x}/EI \quad (10)$$

$$y=1.043*45/2.1x10^5x1.98x10^7$$

$$y=1.63x10^{-5}mm$$

$$y = \frac{1.043W * 45}{2.1 * 10^5 * 1.98 * 10^7}$$

Maximum load carrying capacity of the beam:

w_{max} = 180 kgw_{min} = 1 kgw_{normal} = 100 kg

Factor of safety is calculated using the maximum load to working load condition

Fos = 1.8

B. Column design:

Factors to be considered while column design:

- Crushing stresses
- Normal compression stresses
- Crippling (or) buckling stresses
- The stress on the column should be within limits otherwise failures occur.

Short column:

For short column the crippling or buckling stress in equal to that of crushing stresses. Here failure occurs due to normal stresses.

Long column:

Here failure occurs due to buckling loads only.

Static condition:

Load acting on column:

P= 11.4 X 9.81

P= 111.83 N

Normal stress is calculated using the ratio load to area

$$\sigma = P/A \quad (11)$$

σ = 0.1016 N/mm^2.

σ_c for mild steel = 320 N/mm^2

Since the induced stress is less than crushing stress the design is safe.

c. Base design:It should be rigid from which the structure of the crane is mounted along with material[1]

Material : Reinforced concrete

Density : 24 kN/ mm^3

Maximum working load: 180 kg

W_{max} : 1765.8N

De-stabilizing moment = moment at center point A due to lifting load + Moment (12)

De-stabilizing moment = $4.97 * 10^6$

Stabilizing moment= (self weight + vertical load) * L/2 (13)

$$b = 3.75 * 10^{14} \text{ N mm}$$

From this,

De-stabilizing moment < stabilizing moment

Hence the design is safe condition.

Bearing calculation:

Type: ball bearing

Speed: 250 rpm

Life: 4000 hrs

From PSG design data book of page no 4.6 at 250 rpm & 4000hrs

$$c/p = 4.23$$

To select the a series 60 SKF 6007

c0 = 8800 N/ mm2

c = 12500 N/ mm2

Total load acting on bearing,

Axial load = 115 N

Radial load = 5368.03 N (under working condition)

From data book, the ratio of axial to radial load

$$Fa/Fr = 46 > e \tag{14}$$

$F_a/F_0 = 115/8800 = 0.013$

X = 0.56, y = 2

$$P = (X\,Fr + Y\,fa) \tag{15}$$

$$\frac{C}{P}\frac{12500}{3236.08} = 3.86$$

$$\left(\frac{C}{P}\right)_{calculated} < \left(\frac{C}{P}\right)_{designed}$$

Design is safe. SKF 6007 bearing is to be selected.

7.3 Gear design:

P = 17W, N = 60 rpm

1. Type: spur gear
2. GearRatio : i= 3

The number of teeths in driver and driven are

$$Z_1 = 18$$

$$Z_2 = 54$$

3. Material: CI
4. Gear life: assume 15000 hrs

$$N = 15000*60*60 = 54*10^6 \text{ cycles}$$

5. Internal design torque:

$$Mt = mt*k*kd \tag{16}$$

$$M_t = 3516.5 \text{ N.mm}$$

6. Calculation of young's modules & bending stress[2],

Young's modules $E_{eq} = 1.46*10^5$ N/mm^2

Design bending stress,

$$\sigma b = \frac{1.4\,Kbl}{n*K\sigma}\sigma\text{-}1 \tag{17}$$

$$Kbl = \sqrt[9]{10^7/N} \tag{18}$$

K_{bl}= = 0.83

N = 2 (cast tempered)

$K\sigma = 1.2$

$\sigma_u = 250$ N/mm^2

$$\sigma_{-1} = 0.45\,\sigma_u \quad = 0.45*250$$

$$\sigma_{-1} = 112.5 \text{ N/mm}^2$$

$$\sigma_b = 54.47 \text{ N/mm}^2$$

Crusting temperature:

$$\sigma c = Cb*Hb*Kclkgf/cm2 (19)$$

$H_b = 200, C_b = 23$, $K_{cl} = 0.83$ for CI grade 30,35

B = 54 mm

Pitch diameter of pinion $d_1 = mZ_1 = 5*18$, $D_1 = 90$ mm

Pitch line velocity

$$v = (\pi * d*N1)/60 \tag{20}$$

$$v = 54/90 = 0.6$$

11. Quality of gear:

IS Quality of this gear is 8.

12. Revision of design torque:

$M_t = 2.705*10^3*1.03*1.3$

K = 1.03 for $\varphi_p = 0.6$, Kd = 1.3

Mt = 3621.99 N.mm

13. Check for bending:

$$\sigma b = (i+1) / \alpha mby \ *Mt \tag{21}$$

$$\sigma b = o.82 \text{ N/mm2}$$

σb(revision) <σb(design)Design is safe condition.

14.Check for crushing:

$$\sigma c = 0.74\ (i+1)/d \sqrt{\frac{(i+1)}{b} E_{eq}(M_t)} \sqrt{\frac{(i+1)}{b} E_{eq}(M_t)} \qquad (22)$$

σc = 102.92 n/mm2σc(revision) < (σc)

Design is safe.

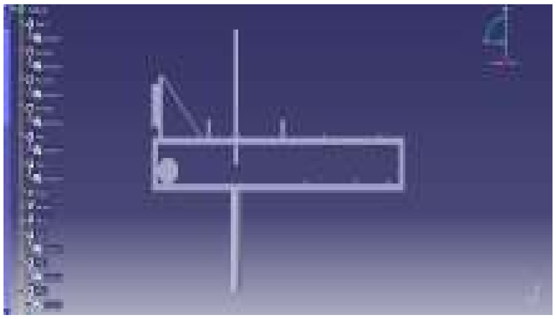

Fig 6. Working arm and counter arm

Fig 7: Design of base structure

Fig 8: Design of Gear Box

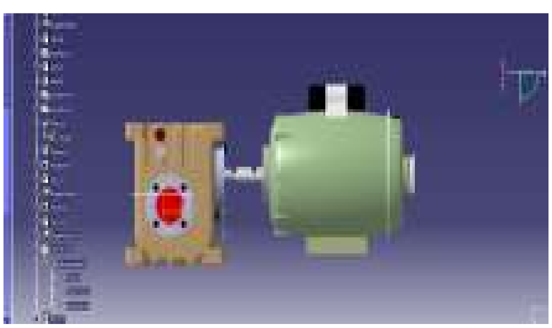

Fig 9: Design of Electric Motor

Fig 10:Designing of gear box

Fig 11. Design of motor and gear box assembly

Fig 12: Design of sprocket

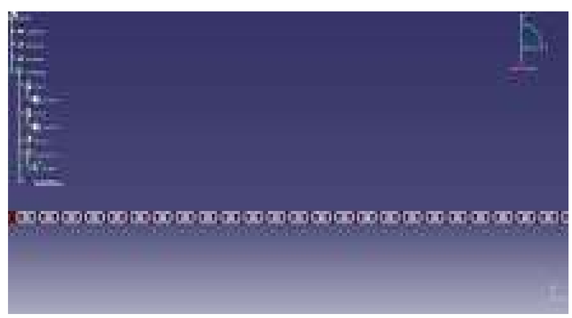

Fig 13. Design of chain

Fig 14. Design of sprocket

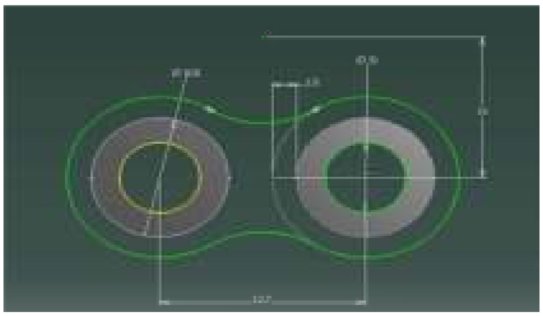

Fig 14: Design of chain

Fig 15: Assembly of pulley and hook

Fig 16.Assembly of multiple pulley crane

III.Conclusion

Design assures that the usage of multiple pulleys in crane reduces the friction between mating parts. Load carrying capacity also increases due to the incorporation of multiple pulleys. This design reduces accidents and ensures safety in operation. Life span of crane also is extended and stability is in control.

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