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DESIGNING AND MODELING OF A FLOOR-MOUNTED JIB CRANE

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ABSTRACT

A jib crane is a type of crane with a cantilever beam, hoist, and trolley that is either attached to a building column or cantilevers vertically from an independent floor-mounted column. This study will primarily focus on floor-mounted jib cranes, in which the trolley hoist moves down the length of the boom and the boom spins, allowing the hoisted load to be skillfully moved in a generally circular area. When building a jib crane, numerous aspects must be considered. The most essential are the crane's own weight and the weight of the items. The purpose of this thesis is to carry out detailed design and analysis of a jib crane. This study studies the stress regions in the jib crane using various materials, and the job is completed by designing reinforcement to overcome such stresses in the component. Models are created in modeling software using the analytical design dimensions, and the models are analyzed using a finite element solver under appropriate conditions, with the results compared.

1. INTRODUCTION

Jib cranes are versatile lifting machines with three degrees of freedom: vertical, radial, and rotating. However, they are limited to particular positions and cannot reach corners. The lifting capacity ranges from 0.5 to 200 tons, with an outreach of up to 50 meters. Jib cranes are often utilized at ports, construction sites, and other elsewhere areas. Jib cranes with grabbing facilities typically have a capacity of 3 tons and operate at 50 to 100 cycles per hour. Lifting heights may be 30 meters or more. Jib cranes used in shipyards to lift heavy mechanical machinery weighing 100 to 300 tons are often positioned on pontoons. These cranes are frequently outfitted with two main hoisting winches that can be used to lift a load either separately or in tandem. Hand auxiliary arrangements, such as those found in machine shops, can be used to handle light weights. Column mounted jib cranes have a common use in the packaging business. These cranes have the ability of lifting loads of up to one ton [1].

Jib cranes are either fixed to a building column or cantilever vertically from a separate floor-mounted column. Figure 1 depicts a column-mounted jib crane. A jib crane is just a boom that includes a mobile trolley hoist. The trolley hoist runs down the length of the boom, which swivels, allowing the lifted cargo to be moved around in a relatively compact semi-circular region. Jib crane pulls and trolleys are usually slow moving and can be handled manually or by radio. The arc swing is usually carried out manually, however it can be mechanized as necessary. Typically, column-mounted jib booms come in two varieties. The primary distinction between the two is how the vertical column force is distributed. The suspended boom shown is examined as if it transfers 100% of the vertical load to the column at the top hinge. The cantilevered boom evenly distributes the vertical stress between the two hinges [10].

2. LITERATURE REVIEW

In today's industry, material handling equipment must be versatile, efficient, and cost-effective. It should also provide flexibility and boost productivity to save money. The necessity for continuous development in material handling technology is a recurrent theme in many modern engineering projects. Engineering structures currently comprises a diverse set of technologies, including structural creation, analysis, design, testing, production, and maintenance. Advances in material handling technologies have been substantially responsible for significant performance gains in many engineering structures, and they continue to play an important role in determining their dependability, performance, and effectiveness.

Figure 1: Fabricated Floor mounted jib crane [4]

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(Krunal Gandhare, Prof. Vinay Thute, et al., 2015) Discovered that optimizing the jib crane is a nonlinear problem; if this problem is solved using the classical method under the Kuhn-Tucker condition, it becomes too complex and difficult to solve, necessitating the use of automated programming. In this research, it was discovered that the evolutionary method produces excellent results and may be utilized to generate the optimum parameters for the crane, as well as parameter values that are possible and within constraints. There are some limitations to the provided optimization model, such as the weight applied being fixed, which really changes with changes in the cross sectional area of the boom, which when removed may yield positive results [1].

(Amit S. Chaudhary, 2015) Conducted a promising structural analysis of a jib crane's cantilever beam. A new beam shape design approach is proposed to address the issues of cantilever beam deflection, shear capacity, and lateral tensional buckling caused by loading. The discussions in this work demonstrate how the web tapered cantilever beam is better capable of resisting lateral torsional buckling and bending with high shear capacity for a given load when compared to the normal I section cantilever beam [2].

(Chirag A. Vakani, Shivang S. Jani, et al, 2014). Conducted an evaluation of jib crane utilization for radius-type operations. Two planar degrees of freedom each demand a different amount of force input. Fibers are elasticity, and some additional load cases not contemplated in the norm have been established, and they have a strong interest in the correct design of the mechanical set, primarily because they simulate some maneuvers that, while discouraged or prohibited, can occur during the use of the crane jib [3].

(K Suresh Bollimpelli, V Ravi Kumar, et al. , 2015) Performed static, model, and harmonic analyses of a column-mounted jib crane under the appropriate load circumstances. The static analysis on the jib crane gave a maximum von-Misses stress of 156.8N/mm2, which is the material's yield stress limit (250MPa). The manual calculations, which assume a simpler model, are likewise compatible with the program results. Analysis provides information about the jib crane's inherent frequencies.

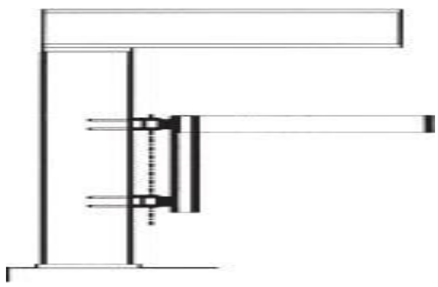

Figure 2: Column mounted jib crane [11]

2.1 Global jib crane loads

Jib cranes apply vertical gravity loads and horizontal thrust loads to the supporting column. Hinge forces supplied by the crane manufacturer should be used when available. If statics are lacking, the loads can be approximated from them, as demonstrated in Figures 2 and 3. lacking,

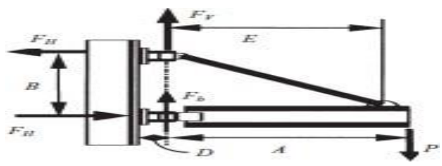

Figure 3: suspended boom jib crane [11].

$$F_v = W_{Lifted} + \frac{1}{2} W_{Boom} + W \frac{Trollf}{Hoist} J \frac{A}{E}$$

Acting @top hinge only

$$F_H = W_{Lifted} + \frac{1}{2} vv_{Boom} + vv \frac{Trollf}{Hoist} I \frac{A}{B}$$

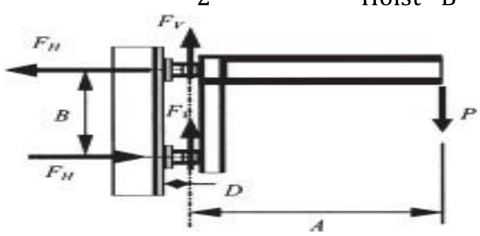

Figure 4: cantilever boom jib crane [11].

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2.2 Pivoting Jib Cranes.

A study about jib cranes would be incomplete devoid mentioning the pivoting type jib crane. This device is made up of a vertical column (generally a WF form) with a thrust bearing at the bottom and a ring-type rollers or ball bearing, or a bearing, at the highest point, as well as one or more horizontally arms (often a conventional beams section for supporting the trolley's wheels). The pivot jib's characteristics include the following:

- **Easy assembly:** Components can be assembled in most fabricating shops.
- **No hinge assemblies:** Direct attachment of boom to jib column flange.
- **Reduced friction:** Bearings enhance ease of swinging.
- **No axial load:** Upper bearing/bushing eliminates axial load.
- **No torsional considerations:** Single boom simplifies design.
- **Full 360-degree swing:** Unlike hinge-mounted jibs.

2.3 Types of Jib Cranes.

There are hundreds of different jib crane types available, depending on the crane manufacturer. However, for the sake of this blog, they all differ to match the needs of your business. Furthermore, each jib type has a unique collection of sub-characteristics that can be tailored to a given application. These make creating the ideal solution for your application simple and efficient. We'll look at three common jib cranes.

A. Wall-mounted jib crane

B. Floor-mounted jib crane.

C. Jib articulates.

A. Wall-mounted jib crane

Rotate 270° for a circular coverage area. With capacities of up to 5 tons, these systems are not lightweight. However, they are not equally strong as a freestanding system. Wall-mounted jibs require no floor space.

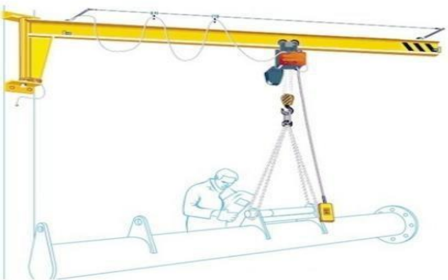

Figure 5: wall mounted jib crane [5]

This system is reinforced with an overhead mount and a tie rod-supported boom. To overcome the problem about off-centered launching, the design features a standard I, T-beam and a single tie rod. Wall-mounted jib cranes are an ideal material handling option for busy industries because they maximize space efficiency and allow for simple installation of individual workstations.

- Cantilever
- tie rod supported

The cantilever wall-mounted design provides the most clearance above and below the boom. It also transmits less direct stress to building columns, making it simple to place on nearly any wall or column in your structure.

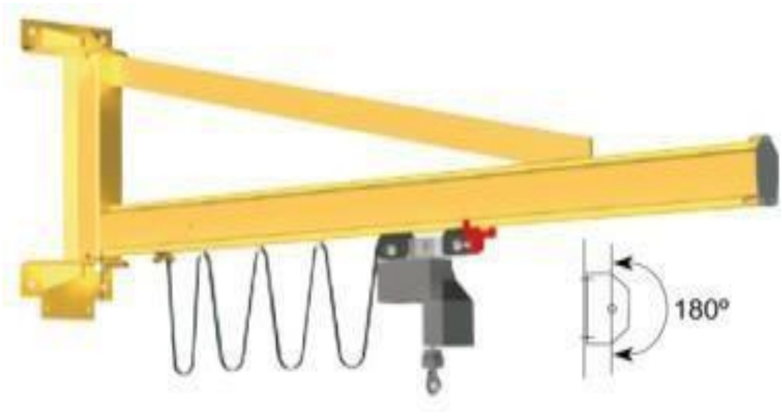

Figure 6: Wall-mounted cantilever jib crane [11].

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The **tie rod-supported** wall-mounted jib crane is exceptionally cost-effective. It has no support structure under the boom, allowing the trolley hoist to simply travel the entire length of the boom.

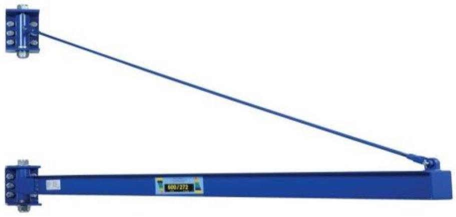

Figure 7: tie rod supported wall mounted Jib crane

B. Floor-mounted Jib Crane

Floor-mounted jib cranes are classified into several varieties, each of which serves a distinct purpose. For example, a freestanding (also known as a stand-alone) jib crane is foundation-mounted, allowing it to be erected practically anyplace inside or outside. Freestanding systems have bigger capacities, longer spans, and 360° and 270° rotation to cover a vast circular area of your business.

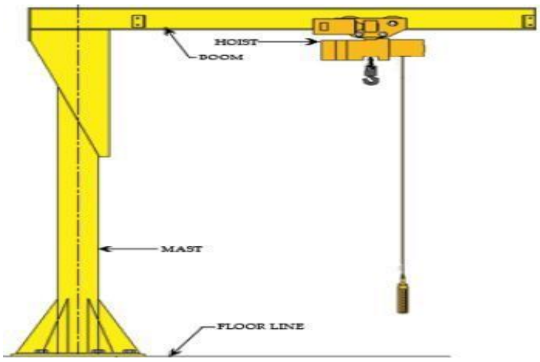

Figure 8: General Floor mounted Jib crane [11].

C. Articulated Jib Crane

These adaptable material handling solutions can lift and move items around corners and columns, reach into machinery, and serve almost any place between the pivot anchor and the boom's long reach. They provide a variety of installation choices, including floor mounting, wall mounting, ceiling mounting, and even bridge mounting. Articulating jib cranes are equipped with an inner and outer arm, allowing them to actually articulate around various areas of a facility, in and out of machinery, and over and under virtually any other impediment. The inner arm offers 270° of rotation, while the outside arm offers 360° of revolution.

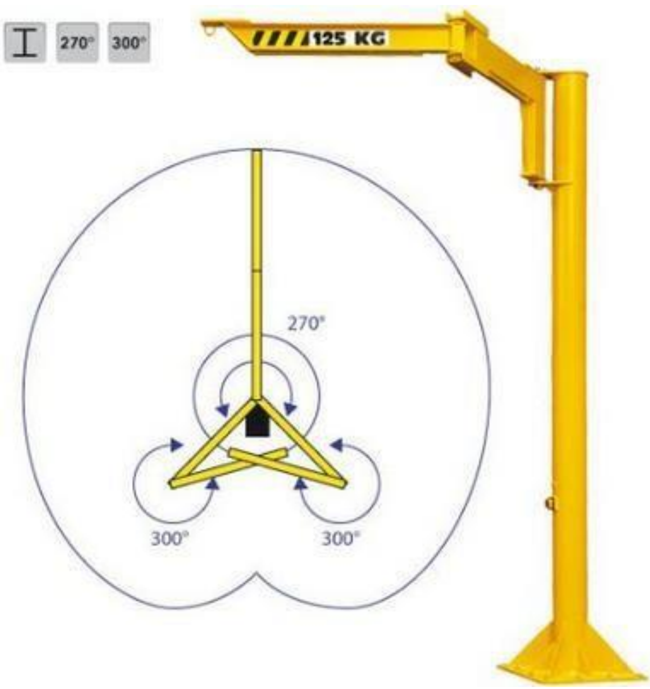

Figure 9: articulating jib crane [11]

2.4 Working with Jib Cranes

The underlying design of a jib crane is a sturdy boom chained to a fixed pivot point. This pivot is then securely affixed to a wall or the top of a freestanding column. This pivot can rotate 180 or

360 degrees and has a large operating range. The hoisting is done with an integrated pulley or motorized chain hoist that can glide along the book and has a broad operational area. Jib cranes, both freestanding and mast-type, can rotate 360 degrees. Wall-mounted options provide 200 degree rotation.

3. DESIGN ANALYSIS

The performance parameters of the Floor Mounted Jib Crane are dependent on its movement. Here, we cover mathematical calculations and an investigation of the performance of a floor-mounted jib crane with a practical capacity of 0.5 tons. Jib cranes have a variety of performance characteristics; in this case, we are considering experimental research of a 0.5ton floor mounted jib crane.

Table 1: Specification of 0.5 Ton FMJC

TYPE	Floor mounted
SWL	0.5Ton
Height of Lift	6000mm
Span	5000mm
Angles of swivel	270
Lifting speed	4m/min.(HT) and 18m/min.(LT)

Where: *SWL = Safe Working Load, *HT = Hoisting Travelling, *CT = Cross Travelling

3.1. Conceptual Design Major Components of Jib Crane

A floor mounted jib crane is made up of the following parts: (a) the jib, (b) the column, (c) the base plate, and (d) the tie rod. Figure depicts the various components of a jib crane. The jib crane has a base that is fastened to the ground at the bottom. A fixed support connects the bottom of the column to the ground. The trolley's motion is powered by an electric motor positioned inside the trolley. The wheels run on the flanges of the "I" sectioned jib along its length. The trolley is made up of hoisting machinery that lifts and lowers the weight using a hook. The load hook has three different motions: lifting, longitudinal traverse of the trolley, and crane swiveling through 270°. Each motion is controlled independently of the others by separate controllers located in a control cage.

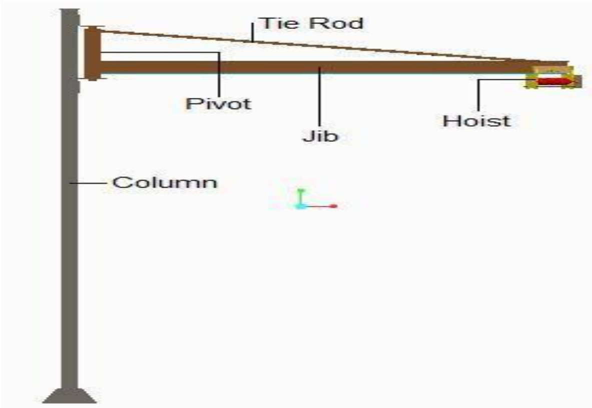

Figure 10: A Model of FMJC 0.5ton

3.2 Structure Design of Jib Crane

3.2.1 Design of Tie Rod

The tie is inclined at an angle of 80 degrees to the jib and secured at a location 5 meters from the center line of the crane column, giving a clear radius of 5 meters from the buckle for ease of erection and to make any necessary changes due to defective workmanship. It is also believed that the crane post exerts no fixing moment on the jib, which would be exceedingly minor in this configuration. The greatest load in the tie will occur when the hook block is at the extreme of our position, which is a radius of 5 meters. Drawing the triangle of forces yields tension in the tie of 10,000N and compression in the jib of approximately 8660N.

The weights of the jib, tie rod, and trolley have been ignored for the time being; actual loads will be evaluated later, once certain measurements have been allocated to these components. A round tie bar of M.S. with the top end forged into the shape of an eye for pin joint is proposed for attachment to a crane post. In slow motion hand cranes, an essential allowance of 15% to 22% must be made. As part of the course work, the following problem design is included as project work. In other words, all actual operating loads and stresses in various structural elements should be increased by 30% to 40% to achieve their static equivalents. To calculate the maximum acceptable working stress, a factor safety equal to four should be added to these static values based on the maximum possible stress.

Assume the actual working load in particular parts is 1000N. The static number after applying a 40% impact factor would be 1200N. M.S. tensile strength is approximately 350N/mm2 (static) and 62.5N/mm2 (dynamic), assuming simple calculations without the impact factor.

Cross sectional area of the bar is required. $\frac{30000}{62.5} = 480mm$

$A = 482mm^2\ approximately$

$$= \frac{\pi}{4} D^2 = 480mm^2$$

$D =$ 24. 72mm, Therefore **D** ~ **25mm**

This diameter is at the bottom of the thread.

Core diameter = **25mm**

$$full\ diameter = \frac{25}{0.85} mm = 29.411mm$$

Therefore, the diameter of the tied rod needed to be$\sim 30mm$

3.2.2 Design of Jib

The maximum bending moment on the jib occurs when the load is at the free end of the jib which is 5 meters from the fixed end. The shearing force will be 3334N only and may not be taken in to consideration.

Maximum bending moment $= 9467 * 5000 = 47.34 * 106$ Allowing a maximum allowed bending stress of $250\, {}^{N}/_{mm^2}$

Section modules of jib required$= \frac{47.34}{165} * 10^6 = 28.7 * 10^6$

Table 2: Specification of ISMB 250

Designation	Area (cm^2)	Depth (mm)	Width (mm)	Web Thk. (mm)
	Ar	D	B	Tw
ISMB 250	47.5	250	125	25.5

Table 3: ISMB with tapered flanges actual load carried by the boom

Designation	Root Thk.(mm)	Root Radius (mm)	Toe Radius (mm)	MI (cm^4)
	Tw	R	r	Iz
ISMB 250	6.9	13	6.5	5131.6

Dead load: (DL) = 315Kg

Hoist Load: (HL) = 150Kg

The lifted load is (LL) = 500$\boldsymbol{W}g$.

The entire load upon the boom has been determined as $\boldsymbol{PX}$ + $\boldsymbol{TX}$ + $\boldsymbol{XX}$ = 315 + 150 + 500 = 965 $\boldsymbol{W}c$. Dead load (DL): The weight of the beam plus any other fixed item supported by it.

Hoist load (HL) refers to the weight of the hoist and any other equipment attached to it. Lifted load (LL): The weight of the item being lifted, including any accompanying lift devices such as slings and shackles.

The column and boom of the self-supported jib crane should be designed so that the overall support of the jib boom does not go over the following limit, whenever the lifting mechanism with full load is at the extreme boom radius:

$\frac{column\ height\ H + boom\ length\ l}{300}$ 43.33 To get maximum deflection

A reference to hand book for structural engineers, "Structural Steel Section" shown that depth of nearest standard beam is either 225mmor 250mm. We select **250x125.**

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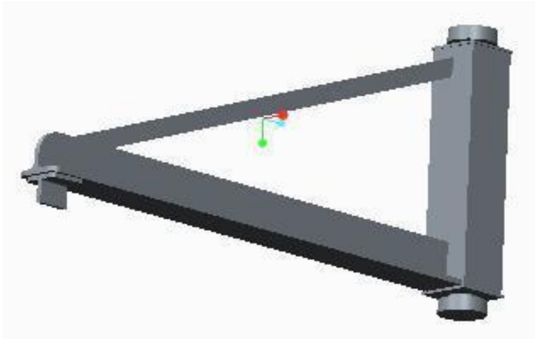


Figure 11: Jib

3.2.3 Design of Rope

Z = No. Of ropes = 1

Q = Lifting capacity = 500kg (IS: 2266 − 1963)

□ = pulley efficiency = 0.94% (IS: 2266 − 1963)

Breaking Strength of Rope, $\boldsymbol{P = \frac{W}{0.94\% * Z} = 5319N}$

Choice of rope: As per IS 3938: 1967 for D/d = 16

Selecting the Rope

$$A = \frac{P}{\left(\frac{\sigma_u}{\eta_f}\right) - \left(dw * \frac{Er}{D_{min}}\right)}$$

Where

s_u = Tensile of the wire = 1600 N / $mm2$ (IS: 2266 − 1963)

$Er = Corrected\ modulus\ of\ elasticity\ = 8 * 10^4\ N/mm$

f = Design factor = 4 η

D = Diameter of drum

d = Diameter of wire rope

dw = Diameter of wire = 0.045 d

P = Breaking strength of rope = 5319N

Now, all values are put in this equation,

$$Now, \quad A = \frac{5319N}{\frac{1600}{4} - 0.04d * \frac{8 * 10^4}{16 * d}}$$

$A = 30.39mm^2$

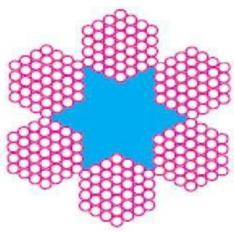
$A = 0.4d^2$

$30.39 = 0.4d^2$

$d = 8mm$ As per IS: 2266 − 1963, acceptable standard rope diameter is $8mm$.

So, the selected rope is 6x37 - 8 - 1600. Here, 6 = strands. 37 = wires.
8 = rope diameter.

1600 = bending strength ($N/mm2$).



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Minimum diameter of pulley = $16d$ (5) = $128mm$

It is recommended to take pulley diameter = 27 (5) = 216 mm. Compensating pulley $D1$ = 0.6 * 216 = 129. $6mm$ $D1$ = $130mm$

3.2.4 Hoisting Drum

Hoisting drum with one coiling rope has only one helix, while the drums with two coiling ropes are provided with helices, right hand & left hand. A design procedure of hoisting drum is as under:

A. Number of turns on a drum for one rope member

$$n = \frac{hi}{\pi d} + 2_{PSG_{9.2}} = \frac{6000 * 2}{\pi * 200} + 2$$

$n = 21.01 turns = 22 turns$

Where,

h = height of load to which it is raised

i = ratio of pulley system = $2_{(PSG\_9.3)}$

D = drum diameter = $25d$ = 25(8) = 200 mm

B. Length of drum

$$L = \left(\frac{2Hi}{\pi D} + 7\right) * Pi_{PSG} = \left(\frac{2 * 6000 * 2}{\pi * 200} + 7\right) * 9.5$$

L = 429.37mm = $430mm$ the pitch of two rope grooves (p) equals 9.5 mm

C. Drum thickness is calculated as

t = 0.02D + 10(PSG_9.3) = (0.02 * 200) + 4 = $14mm$.

D. Outside diameter of drum

Do = D + 6(PSS_9.3) = 200 + 6(8) = 248 mm

E. Drum inner diameter

Di = D − 2t = 200 − 2(14) = $172mm$

3.2. 5 Design of Rope Sheave

Sheaves are usually made of steel, grey cast iron. Small sheaves are made by casting. Sheaves are mounted on axles with antifriction bearings or bronze bushes.

1. Sheave diameter (D = 27d = 216 mm).
2. $width$ that $sheave$ = $2.75d$ = $22mm$ = 2 * $Holediameter$ = $2.7d$ to $3.6d$
3. $Depth$ for the $groove$ = $1.6d$ = $12.8mm$
4. $Radius$ for the $groove$ = $0.53d$ = $4.24mm$
5. $Thickness$ for the web = $0.75d$ = $6mm$
6. $width$ for the $groove\ at\ the\ top$ = $2.1d$ = $16.8mm$
7. $Radial\ thickness$ = $0.4d$ = $3.2mm$
8. $Boss\ diameter$ = 2 (Hole diameter) = 2.7 to 3.6

3.2.6 Design for the Hook

General purpose cranes that carry loads that have different shapes carry the loads via chains or rope slings attached to hooks. Standard (single) ram hooks are the most commonly utilized design for this function. Solids are sometimes used to construct triangle hooks, which can be flat-die or closed-die forged, or composed of a sequence of shaped plates.

$$C = \sqrt[k]{W}$$

$w = load\ to\ be\ lifted$ = 5000 N

k = 12

$$C = \sqrt[12]{\frac{5000}{1000}} = 26083mm$$

$take\ inner\ hook\ diameter\ of\ hook\ (C)$ = 26. 83

$Depth, H$ = 0.93 * C = 0.93 * 26.83 = 24. $95mm$

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$Base\ section, M = 0.6 * C = 0.6 * 26.83 = 16mm$

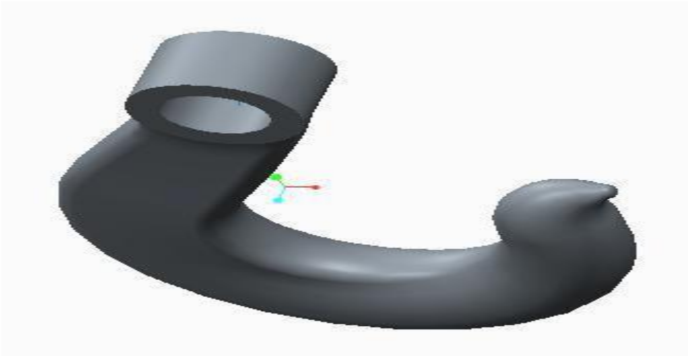

Figure 13: Hook

3.2.5 Design of Gear Box

Design for both the pairs is similar in fashion as per the design for first gear and hence the dimensions or both pairs are same.

$speed\ ratio = pinion\ \frac{RPM}{Gear\ RPM} = \frac{387}{54} = 7.$ Material referring to the standard table given in P.S.G. Data –Book for i = 8 (PSG _8.4, Table_5)

(Surface hardness HB ≤ 350) For Pinion   C45

For Wheel............ C35Mn75

$For\ = C45\ \sigma_1\ = 270N/mm^2$

(I) For Pinion,

$\sigma 1 = Endurance\ strength = 270\ mm2\ (PSG\_8.18).$

$Kb1 = life\ factor = 1(\leq 350HB\ surface\ hardness)(PSG8.17, Table17)$

$Ks = surface\ finish\ factor = 1.6\ PSG\_8.15)$

$\mu f = factor\ of\ safety = 2\ (PSG8.19, Table20, Normalised)$

$YI = form\ factor = 0.3\ (PSG8.19, Table18)$

$$Y_1 * \sigma = \frac{0.3 * 270 * 1 * 1.4}{1.6} = 35.43\ N/mm^2$$

Table 3. Dimension of 2nd Pinion

No. Of Teeth (Z)	23
Module (m)	2.0 mm
P.C.D	46 mm
O.D	50 mm
Addendum	2.0 mm
Dedendum	2.5 mm

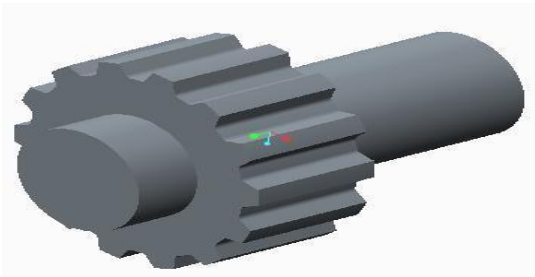

Figure 14: Pinion

(II) For Gear

$\mu_f = 2(PSG_{8.19}, Table_{20}, Normalised)$

$K_s = 1.6(PSG_{8.15})$

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$Y_2 = 0.05(PSG_{8.18}, Table_{18})$

$K_b = 1(\leq 350HB\ surface\ hardness)\ (PSG_{8.17}, Table_{17})$

$$Y_2 * \sigma_{B2} = \frac{0.5 * 1.6 * 160}{2 * 1.6} = 40\ N/mm^2$$

$here,\ Y_2 * \sigma_{b2} > Y_1 * \sigma_{b1}$

$so,\ gear\ is\ stronger\ than\ pinion\ and\ hemce\ pinion\ is\ to\ be\ designed\ center\ distance$

$$a \geq (i \pm 1)\sqrt{\frac{0.74^2}{\sigma_e}} * \sqrt{\left(\frac{E * M_i}{i * \in}\right)}\ (PSG_{8.13}, Table_8)$$

$E\ =\ Modulus\ of\ elasticity\ = 2.15 * 10^5\ N/mm^2 (PSG_{8.14}, Table_9)$

$$center\ distance, a = \frac{Z_g + Z_p}{2}$$

Number of teeth on pinion = 23

Number of teeth on wheel = 11 * i = 23 * 7.1 = 163

$$a = \frac{163 + 23}{2} = 93mm$$

$Z_1\ = number\ of\ teeth\ on\ pinion\ = 23$

$Z_2\ = number\ of\ theeth\ on\ gear\ = 163$

$face\ width, b\ = \in * \ a\ = 0.2 * 93\ = 18.6mm$

$$now\ checking\ in\ bending\ stress, \sigma_{b1}\ = \frac{i + 1}{a * m * b * Y} * M_I (PSG_{8.18})$$

$$= \frac{7.1 + 1}{93 * 2 * 18.6 * 0.3} * 734022 = 5728\ N/mm^2$$

$other\ dimensions\ of\ pair\ Module\ m\ = 2mm$

$center\ distance\ a\ = 93mm$

$tooth\ debth\ h\ = 2.25 * 2\ = 4.5mm\ \ (PSG\_8.22, Table\_26)$

Pitch diameter, $d_2 = m * Z_1 = 23 * 2 = 46m(PSG\_8.22, Table\_26)$

$d_2\ = m * Z_2\ = 163 * 2\ = 326m(PSG\_8.22, Table\_26)$

$Tip\ diameter, d_{s1} = (Z_1 + 2) * m\ = 50 * 2\ = 100m(PSG\_8.22, Table\_26)$

$d_{s2}\ = (Z_2 + 2) * m\ = 165 * 2\ = 330mm\ (PSG\_8.22, Table\_26)$

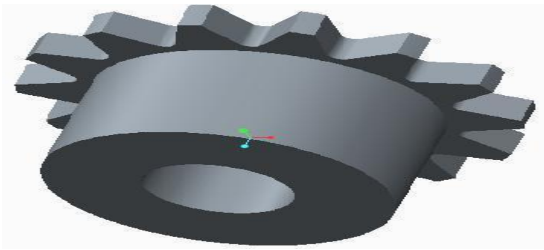

Figure 15: Gear

3.2.6 Design of Shaft

Table 4. Force on Shaft

Shaft No.	Torque (M_t)	Tangential Force (P_t) = M_t/d	Radial Force (P_r) = $P_t * Tan\alpha$
A	734022 N /mm	101.19 N	36.83 N
B	734022 N /mm	715.6 N	260.48 N

$\alpha\ = Pressure\ Angle\ = 20^0$

$R_x + R_y = 869.20\ N$

$Taking\ moment\ about\ x;\ (170.68 * 11.5) + (761.52 * 21.5)\ = 33 * R_y$

$R_y\ = 555.62\ \&\ R_x\ = 313.58/8$

$Moment\ at\ A.\ M_A = 313.58 * 11.5 = 3606.17\ N/mm$

$Moment\ at\ B.\ M_B = 555.62 * 115 = 6389.63 N/mm$

$so\ maximum\ moment\ is\ at\ B,\quad M_{max} = 6389.63\ N/mm$

$Torque, T_q = 734022\ N/mm$

$so\ equvalent\ torque = \sqrt{}\ ((K_b * M_{max})^2 + K_t * T)$

$K_b = 1.3\ and\ T_{eq.} = \sqrt{}\ ((6389.63 * 1.5)^2 + (734022 * 1.3)^2) = 954276.73\ N/mm$

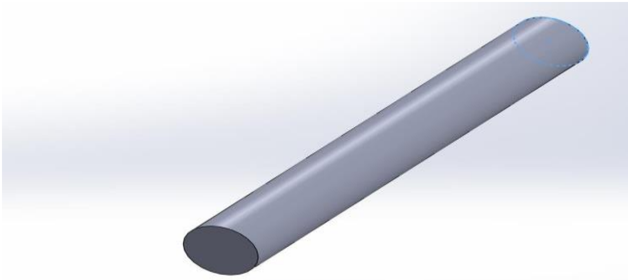

Figure 16: Shaft

3.2.7 Design of Cross-Travel

Cross travel consists of wheel and motor.

- d = Journal diameter
- D = Outer race diameter First consider friction at inner race

$Total\ load\ on\ wheels\ of\ cross\ travel = 9467N$

9467

$For\ one\ wheel\ P = \quad = 2367\ N\,d$

4

$Frictional\ resistance\ moment\ at\ journal, = p * \frac{d}{2} * f$

$f = coeffieient\ of\ friction\ for\ bearing\ from\ standard\ table\ Alexandrov$

$f = 0.015\ (for\ ball\ and\ roller\ bearing)$

$10\ Frictional\ resistance\ moment = 2367 * \frac{10}{2} * 0.015 = 178\ N/mm$

$Frictional\ resistance\ momentat\ contact\ face\ of\ wheel\ and\ plate, = p * \frac{35}{2} * \mu$

$Total\ moment\ of\ resistance\ = 2367 + 178 = 1421\ N/mm$

$Speed\ of\ cross\ travel\ (linear\ velocity)\ = 18\ m/min = 0.3\ m/s$

$Speed\ of\ cross\ travel\ (linear\ velocity)\ = 18\ m/min = 0.3\ m/s$

$$Angular\ speed\ w\ =\ linear\ \frac{velocity}{radi\ of\ wheel} = \frac{0.3 * 2}{0.035} = 17.14\ rad/sec$$

$$No. of\ revolutions\ = \frac{w * 60}{2\pi} = \frac{17.14 * 60}{2\pi} = 163.75 = 164RPM$$

3.2.8 Design of Column

For the case at hand, the column represents an important crane number. It has been observed that when a column or strut is subjected to a progressive rise in compressive load, a point occurs at which the column is subjected to ultimate load. Beyond this, the column will fail due to crushing, and the load will be referred to as the crushing load. It has also been observed that a compression member may fail by bending rather than crushing. Beam columns are well named because they can sometimes act fundamentally like restrained beams, producing plastic hinges, and under other conditions fail by buckling in a similar fashion to axially loaded columns or by lateral tensional buckling in the same way that uncontrolled beams do. Columns must be constructed for both compressive and bending loads. In a bending moment-based design, the moment owing to total load 9467N is communicated to the column's base.

The weight of beam = 367.87N Hence total moment,

$\mathrm{M} = \mathrm{Eccentric\ moment} + \mathrm{Moment\ due\ to\ load} = (0.25 * 367.887) + (5 * 9467) = 47426.96\ \mathrm{N/mm}$

The maximum bending moment for column fixed at one end and other end free with eccentric loading is given by,

$$\mathrm{M} = \mathrm{w} * \mathrm{e} * \sec(\mathrm{L}/2)\sqrt{\frac{\mathrm{W}}{\mathrm{EI}}}$$

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Where

- M = Load on column
- e = Eccentricity of load
- L = Equivalent length of column = L/4
- E = Module of elasticity = $0.215\ \text{N/mm}$
- I = Moment of inertia

$$\text{M} = 5000 * 5 * \text{Sec}\left(\frac{5}{4}\right)\sqrt{\left(\frac{5000}{0.125}\right)}\text{I} = 6.48 * 10^{-2}\text{m}^4$$

Then choose the section for the column, and the dimensions are as follows: $b = 125mm$, $h = 75mm$

$r(cm) = 410$, $ry(cm) = 2.65$ $Zx(cm3) = 410$, $Zy(cm3) = 53.5$

4 RESULT AND DISCUSSION

4.1 Result

The design parts of the floor mounted jib crane components and show the parts 3D modeling using solid work software, assemble the parts of the jib crane and show an animation of the working of the floor mounted jib crane using solid work software, we found the lifting height to be up to 6 meters, lifting load to be up to 0.5 tons, degree of rotation to be from 180 to 270, and lifting speed to be 4 m/min up to 18 m/min.

4.2 Discussion

The components of the floor mounted jib crane were designed and selected to comply with the specifications provided in the literature review, and the results were assessed and verified. The designed components required for the machine were identified, their availability and manufacturability were checked, and the appropriate materials for the components were selected. We analyzed the strength of each component, modeled them using Solid Work, and assembled them in order.

5 CONCLUSION

More study will result in the automation of all lifting and hoisting firms, allowing for higher production quality and quantity while also ensuring the well-being and safety of workers. We feel that the floor mounted jib crane we built is better suited to heavy lifting up to 0.5 tons of weight and can raise automobile engines in workshops to a maximum height of 6 meters. I prepared 3D models using Solid Works.

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