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Design and Development of Portable Crane in Production Workshop: Case Study in BISHOPTU AUTOMOTIVE INDUSTRY, Ethiopia

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Abstract This portable crane uses a hydraulic system to lift a heavy loads applying only small force. The main advantage of the project is having detail design of the mechanism in the production workshop of BISHOFTU AUTOMOTIVE INDUSTRY; Ethiopia is that it is portable, moveable, and easy for operation. In this project we designed and produced a portable crane which can lift a heavy load with a maximum capacity of 3 ton. The crane has two loaded side bars to make the base and two links (i.e. Vertical column and boom) connected each other by using pin joint. The Vertical column is secured on the cross bar that is welded to the side bars making the base using bolt connection. There are also other bars for supporting purpose which are connected to the basic link (vertical column) using bolt connection. The crane uses four wheels, of which two of them in the front are connected to the base using permanent joint and the rear wheel is connected to the base using roller. Since the crane operates hydraulically there is piston cylinder device which is connected to the vertical column and boom for lifting up and down the objects. The maximum carrying capacity 3 ton, and maximum lifting height is estimated as greater than greater than 2.96m from the ground run by using 3KW electric motor rotating 2830/3620 rpm.

Keywords: *portable crane, lifting, vertical column bar, hydraulic cylinder, crane wheels*

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1. Introduction and Background

The development of lift machine or crane has reached through different time starting the first crane for lifting heavy load was invented by ancient Greeks in the late 6th century BC. The heydays of crane in ancient times come during Roman Empire when construction activity soared and buildings reached enormous dimensions. The Romans adapted the Greeks cranes and developed it further.

The simplest Romans crane is the trespasses, which consists of a single beam jib, a winch, a rope, and a block containing three pulleys. Having this mechanical advantage of 3:1, it has been calculated that a single man working the winch could raise 150kg [3pulles*50kg=150kg], assuming that 50kg represent the maximum effort of a man can exert. Over a long time period heavier crane type featured five pulleys (pentaspastos) or in case of the largest one a set of three by five pulleys (polyspestos) and came with two, three, or four masts depending on the maximum loads.

The polyspestos when operated by four men at both side of the winch could readily lifts 3000kg[3 ropes*5 pulleys*4 men*50kg=3000kg] If the winch was replaced by a trade wheel, the maximum load could be doubled to 6000kgs, because the trade wheel have much bigger mechanical advantages due to its higher diameters. This means comparing with the construction of Egyptian pyramid where in 50 men are needed to move 2.5 tons of stone up the ramp (50kg per persons). The lifting capability of the Roman polyspastos is proved to be 60 times higher than the Egyptian system of lifting stones.

During the high middle age the trade wheel was introduced on large scale after the technology had fallen in the Western Europe with dismiss of Western Roman Empire.

The earliest reference to the trade wheel reappears in the archival literature in France about 1225. Generally vertical transport could be done more safely and inexpensively by crane than customary method. Typical areas of application were harbors, mines, and in particular building sites where the trade wheel crane played a pivoted role in the construction of lofty Gothic cathedrals.

In contrast to the modern cranes, middle age cranes and hoists –much like to their counter parts in Greece and Rome were primarily capable of a vertical lift, not used to move loads for considerable distance horizontally as well. It is not worthy that middle age cranes rarely featured ratchets or brakes to forestall the loads from running backwards. This curious absence is explained by a high friction force exercised by middle age trade wheels, which normally prevented the wheel from accelerating beyond control.

With the onset of the industrial revolution, the first modern cranes were installed at harbors for loading cargo.

In 1838, the industrialist and business man sir William Armstrong designed a hydraulic water powered crane .His design used a ram in a closed cylinder that was forced down by a pressurized fluid entering the cylinder. Thus the valve on the cylinder regulates the amount of fluid intake relative to the load on the crane [1].

A. Types of portable crane

Thus cranes are broadly categorized into three

1) Overhead crane
2) Fixed cranes and
3) Mobile crane

Overhead crane

It is being used in a typical industrial shop. The hoist is being operated via a wired push button station to move system and load along any direction .un overhead mechanism crane is also known as abridge crane where in the hook and by line mechanisms runs along two widely separated rails. Overhead crane is typically consists of lifter single beam or double beam construction. It can be built by using typical steel beam or more complex box grinder type. Double Grinder Bridge is more typical when there is a need for heavy capacity system of 10tons and above.

The basic components of overhead cranes are the hoist to lift the item, the bridge, a trolley to move along the bridge. Most of the time over head cranes are applicable in steel manufacturing industry and vehicle/truck production industry

Fixed cranes

Fixed cranes are preferable in order to insure the ability to carry heavy reaches greater heights due to increased stability. These types of cranes are characterized by the fact that their main structures does not move during the period of use. However many can still be assembled and disassembled. The structure basically is fixed in one place. There are many different types of fixed cranes such as Tower cranes, self-erecting, telescoping, Hammerhead, Gantry cranes, Deck cranes, jib cranes, and bulk handling cranes. However most of them are used under construction sites. Only few of them like jib cranes Gantry and deck cranes are used in mechanical engineering operations.

Mobile cranes

For effective and versatile operation cranes can also made to be mobile. Mobile cranes are designed in different manner in order to be used on the road, rail, water and air. There are many different kinds of mobile cranes. That are truck mounted, side lifter, rough terrain, all terrain cranes, pick, and carry cranes, telescopic handler crane crawl crane, pail road cranes, floating Aerial crane and also portable crane.

B. Definition of portable crane

The portable crane is a product selected into the application of design axiom for this project. Portable crane is a small crane that can be broken down into several parts for ease of transportation. It must be assembled and bolted into a place where to be used effectively. Portable crane designed for a whole range of floor lifting job. They are transportable and require no external power. They can be used where no overhead lift is available and relatively short lift required. It also increase efficiency and productivity while decreasing operator bending and risk of back injury. The portable crane is widely used in mechanical engineering filed for assist any heavy duty job. There are various types portable crane that are available in the market with a various function and features.

C. Statements of problems

In BISHOFTU AUTOMOTIVE INDUSTRY there are seven factories .From those factories we have visited most of them and there we have observed many drawbacks. For example in light duty vehicles production factory there are only two overhead cranes and also in bus production factory there is only one overhead crane. In addition in heavy duty truck production factories some components like lower torque rod, v-torque rod, fuel tank …etc are carried by man power to be assembled.

All of the components are transported from store to the factories by using fork lift and this needs cost for fuel consumption. In addition to this electrical power is needed to operate the crane ,this means if there is no electrical power the factory is obliged to wait for the electrical service during this time workers are expected that they are working and this is another loss of cost for the man power.

D. Objectives of the research project

The objective of this project is categorized in to general and specific objectives.

General objectives

The general objective of the research project is to design and produce portable and moveable lifting crane to lift heavy loads that is beyond the capacity of human being applying only small force in the production machine shop.

Specific objectives

The details of the objectives of the research project are listed as follows;

- To design analytically the portable lifting machine lift slightly heavy objects that can't be carried by single worker
- To minimize cost expenditure for fuel that is made for operating fork lift in transporting every component in the production shop
- Specifically in bus production factory to minimizes wastage of time due to each station should wait for a single crane for lifting
- To minimize risk of life and property
- To produce the working prototype of portable, moveable crane for the production shop
- To determine the overall cost of the crane production
- Finally, to documentation of the research project

E. Methodology

The methodology incorporated in design and development of portable crane in the production shop of Bishoft Automotive Industry is indicated as follows;

- Data source identification, collection and analyze;
  Observation in different factories and shops such as; Heavy duty truck production factory, Light duty vehicles production factory, After sale service shop under heavy duty truck, Production factory, Bus production factory, Power house
- Development of suitable system configuration
  It should have moveable wheel attached base, hydraulic link, basic vertical column, and connections
- Design analysis of each components of the portable crane

- Development of working crane prototype for the application

F. Material Selection

Material selection is a means of selecting material that best suited for the member of machine to be designed. As a designer material selection should be done carefully in order to design each components, that will serve till the end of service life. The basic considerations done in selecting materials are; Strength, Machinability, Toughness, Ductility, Hardness…etc. for each components of the crane.

Material for vertical column

Load that the vertical column subjected to is compressive, and material for the column is selected to be cast iron, because it has low cost, good casting, and high compressive stress. It is primarily made of carbon and iron with carbon content of 1.7% to 4.5%

Material for the boom and base plate

Since the load applied on these components is high the material used to make these components should be strong and hard, thus the best material suited for is steel which has carbon content of up to 1.5% which results in an increased strength and hardness.

Material for the crane hook

Thus the crane hook is subjected to both tensile and compressive stress, and wrought iron is selected for it, because it is malleable tough and ductile material. It has carbon content of0.02%, 0.12% silicon, 0.018% of phosphorus, 0.07%of slag, and the remaining is iron [2].

Material for Bolt

The material for the bolt is subjected to tensile and shearing stress and mild steel best fit the criteria to resist these stresses and is selected as a material for bolt.

Material for the Pins

In our project pins are subjected to tensile and shearing. The material selected for the pin should be ductile and we have selected mild steel.

2. Design Analysis the Parts

A. Design of vertical column

Vertical column is modeled as a strut or short compression member thus it is exposed to a compressive stress and this stress is the sum of simple stress component and flexural (bending) components [2].

$$\sigma_c = P/A\ + MC)/I$$
$$= P/A + (PecA)/IA$$

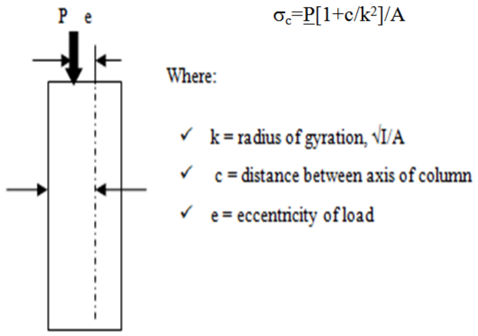

Figure 1. Vertical column

Therefore: c=0.15m/2, e=1.6m

The column cross section subjected to compression stress is

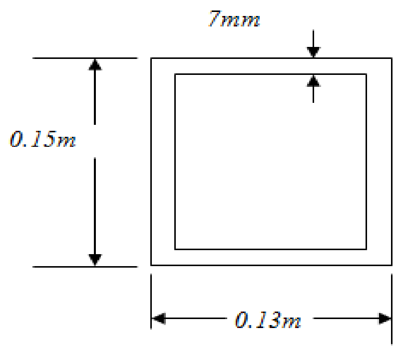

Figure 2. Cross Section of Vertical Column

The area of column cross section is

$$A = [0.15*0.13) - (0.116*0.136)]m^2 = 3.724*10^{-3}m^2$$

• **Moment of Inertia**

$$I_{xx} = \left[(0.15*0.13)^3 - (0.136*0.116)^3\right]/12$$
$$= (2.74625 - 1.769)*10^{-5}m^4$$
$$\mathbf{I_{xx} = 0.97725*10^{-5}m^4}$$
$$I_{yy} = \left[(0.13*0.15)^3 - (0.116*0.136)^3 *10^{-5}\right]/12$$
$$\mathbf{I_{yy} = 1.22089*10^{-5}m^4}$$

NOTE: Buckling always occur about the axis having minimum radius of gyration or least moment of inertia, therefore in our case buckling occur along horizontal direction (I_{xx}).

$$K = \sqrt{Ixx/A}\ = \sqrt{0.97725*10^{-5}/3.724*10^3}$$
$$= 0.051226891m$$
$$\sigma c = P_{vc}/A(1 + ec/k2)$$
$$P = (3000)\left(9.81m/s^2\right)$$
$$= 29.43KN$$
$$P_{vc} = P/(\cos 12°)$$
$$= 30.1KN$$

$$\sigma_c = 30.1KN/1.911*10^{-3}m^2$$
$$[1 + (1.6m)(0.075m)]/(0.051226891)m^2$$
$$= 15.751*106N/m^2[1 + 45.72832]$$
$$= 794.024*106N/m^2$$
$$\boldsymbol{\sigma_c = 736.013MPa}$$

→ Referring to text book of machine design by KHURMI and GUPTA, The maximum value of crushing stress (σc) that will develop in cast iron is given to be 400MPa to 1000Mpa. Since the induced compressive stress due to the applied load is not greater than the maximum crushing stress developed in the cast iron. Therefore the vertical column is designed safe.

B. Design of Boom

The boom is modeled as simply supported beam, and it is subjected to a bending stress due to bending moment developed at the fixed end where it is pinned with the vertical column.

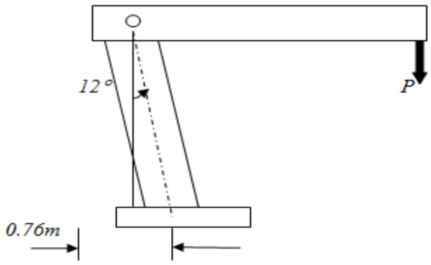

Figure 3. Boom connected with

$$\sigma_b = M / Z$$

Where:

- Z = section modulus
- M = Bending Moment at point A
- σ_b = Bending stress

Since the boom is hollow rectangular cross section, the area of boom to which the effect of load P induces the stress is:

$$A = 1.6(0.12 - 0.108)$$
$$A = 0.0192 m^2$$

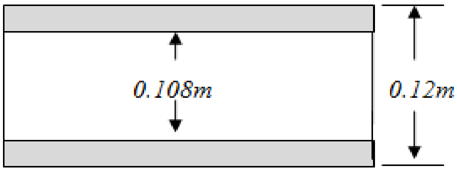

Figure 4. Boom section vie

• Moment of Inertia

$$I_{xx} = 1.6(0.12 * -0.108)/12 \; m^4$$
$$I_{xx} = 6.24384 * 10^{-5} m^4$$

- Distance from neutral axis extreme fiber (c), is;

$$c = (0.12)/2m = 0.06m$$

- Section Module(z)

$$z = I/c = \left[6.24384 * 10^{-5} / (0.06)\right] m^4$$
$$z = 1.04064 * 10^{-3} m^3$$

- Bending Moment , M is;

$$M = P * L, \; L = \text{length of boom}$$
$$= 29.43 KN \; * 1.6m = 47.088 * 10^3 Nm$$
$$\sigma_c = M / Z$$
$$= 47.088 * 10^3 Nm / 91.0406 * 10^{-3} m^3)$$
$$\boldsymbol{\sigma_c = 45.25 Mpa}$$

NOTE: The material selected for the boom is standard steel Fe E-520-Indian standard designation, and it has a minimum yield stress of 520Mpa and the allowable bending stress is;

$$\sigma_b = Sy / n,$$
$$n = \text{factor of safety – which is assumed to be 4}$$
$$= 520 Mpa / 4$$
$$\sigma_b = 130 Mpa, \text{ Allowable bending stress.}$$

Since the allowable bending stress developed is greater than the induced bending stress due to the load applied, then the boom is designed safe.

C. Design of crane hook

Hook is the component which is fixed with the boom and it is used for hanging the load on the boom which moves up and down in lifting the load.

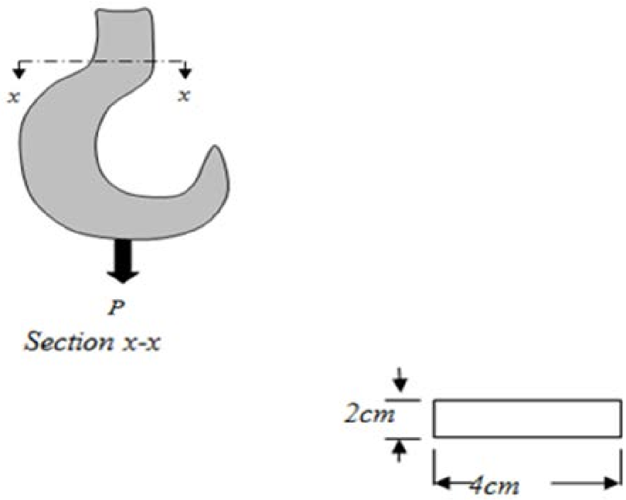

Figure 5. Plane view of crane hook

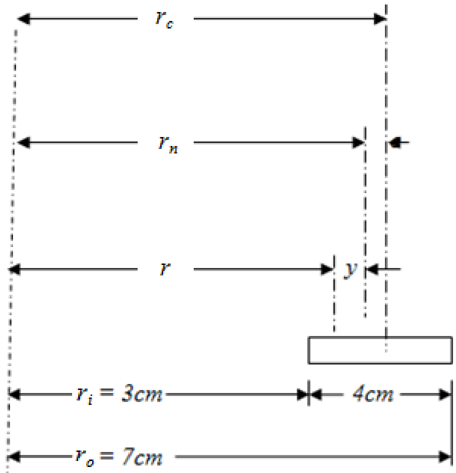

Figure 5.1. Cross section of crane hook

(From Shigley's "Mechanical Engineering design", 8^{th} edition –Table 3.4)

$$r_c = r_i + h/2$$

and

$$r_n = h / \ln(r_o / r_i).$$

Due to the applied load the hook is subjected to tensile and compression stress with moment M=P*r_c.

- **Normal stress**

$$\sigma = P / A + My / \left(Ae\left(r_n - y\right)\right),\ r_n - y = r.$$

But r_c = 3cm+4cm/2 = 5cm

$$r_n = 4 / \ln\ (7/2) = 4.72\text{cm}$$

$$e = r_c - r_n \Rightarrow (5 - 4.72)\text{cm} = 0.0028\text{m}$$

Cross sectional area at which the load act is,

$$A = 2\text{cm} * 4\text{cm} = 8\text{cm}^2 = 8*10^{-4}\text{m}^2.$$

Moment due to the load applied, M = P * r_c, => 29,430N*0.05m = 1471.5 Nm.

→ Substituting r from 3 to 7 we can determine the stress developed in compression and tensile side.

- For r = 3cm

$$\sigma = 29430\text{N} / 8*10^{-4}$$
$$+1471.5\text{Nm}(0.0472 - 0.03) / (\ 8*10^{-4}(0.0028)(0.03))$$
$$\sigma = 413.4215\text{Mpa}.$$

- For r = 4cm

$$\sigma = 29,430\text{N} / (8*10^{-4})$$
$$+1471.5(0.0472 - 0.04) / (\ 8*10^{-4}(0.0028)\ (0.04))$$
$$\sigma = 155.0255\text{Mpa}.$$

- For r = 5cm

$$\sigma = 29,430 / 8*10^{-4}$$
$$+1471.5(0.0472 - 0.05) / \left(8*10^{-4}(0.0028)(0.05)\right)$$
$$= 0\text{Mpa}$$

- For r = 6cm

$$\sigma = 29,430 / 8*10^{-4}$$
$$+1471.5(0.0472 - 0.06) / \left(8*10^{-4}(0.0028)(0.06)\right)$$
$$\sigma = -103.3557\text{Mpa}$$

- For r = 7cm

$$\sigma = 29,430 / 8*10^{-4}$$
$$+1471.5(0.0472 - 0.07) / \left(8*10^{-4}(0.0028)(0.07)\right)$$
$$\sigma = -177.1806\text{Mpa}.$$

NOTE, since the induced tensile and compressive Stress are less than for the crane hook, i.e. wrought iron which have a stress of 250MPa to500MPa and 300MPa under tensile and compressive respectively, then hook is designed safe. The ultimate tensile and compressive stress of the material that is selected

D. Design of base plate

Base plate /Truck serve as a base for carrying all the weight of the proposed design project. In addition it carries the load by all the components of the crane .It is composed of four bars and each of them is modeled as a beam.

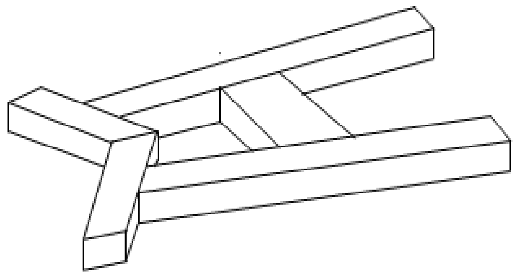

Figure 6. Base plate

Bar 1 and Bar 2 are side bars on which the center connecting bar is supported over. And the other is end bar which is provided at the back end of the two side bars to give extra strength.

To know the stress developed in these components first we should know the all loads applied and let's first calculate all the masses of all the other components.

Masses of vertical column:

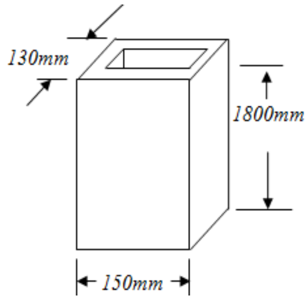

Figure 6.1. Vertical column

- Volume vertical column

$$V = A*L$$
$$= \left[(150*130) - (136*116)\right](1800)$$
$$= 6,703,200\text{mm}^3$$
$$V = 0.0067032\text{m}^3.$$

Since the material for the vertical column is made from cast iron, the mass density of cast iron is 7250kg/m ^{3}.

$$\rho = m / V,\ \ m = \rho V$$
$$= \left(7250\text{kg} / \text{m}^3\right)\left(0.0067032\text{m}^3\right)$$

m_v = 48.6kg, mass of Vertical column

Mass of Boom:

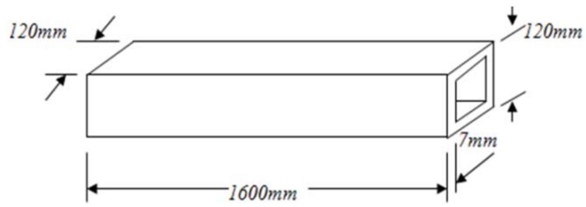

Figure 6.2. Boom cross section

The volume of the boom is

$$V = A*L$$
$$= \left[(1200*1200) - (1060*1060)\right](1600)$$
$$= 0.00506062\text{mm}^3.$$

The boom is made of standard steel, and the mass density of steel is 7850kg/m^3

$$\rho = m / V, m = \rho V$$
$$= \left(7850kg / m^3\right)\left(0.00506062m^3\right)$$
$$m = 39.7kg, \text{ mass of boom}$$

Mass of vertical column support:

$$V = A * h$$
$$= \left[(60*70)-(52*62)\right]1200$$
$$= (4200-3224)1200$$
$$V = 0.0006048mm^3.$$

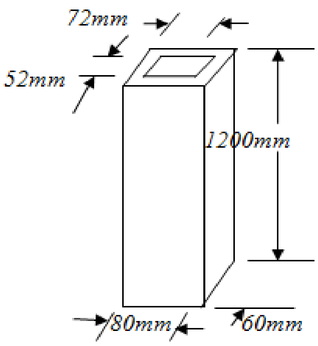

Figure 6.3. Vertical Column support

The material for the support is selected to be cast iron, which have mass density of 7250kg/m^3.

$$\rho = m / V, \; m = \rho V = 7250 * 0.0006048,$$
$$m = 4.4kg, \text{ mass of vertical column support.}$$

Mass of crane hook

The cross sectional area of crane hook is assumed to be equal to thin circular plate of 6cm inner diameter and 14 cm outer diameter having a thickness of 2cm.

$$V = A * t$$
$$= [\pi\left(d_o^2 - d_i^2\right)] / 4(t)$$
$$= \pi / 4\left(7cm^2 - 3cm^2\right)(2cm)$$
$$V = 62.832 * 10^{-6} m^3.$$

Since the material for crane hook is wrought iron, which have mass density of 7780kg/m^3,

$$m = \rho * V = (7780)\left(62.832*10^{-6}\right)kg, \; m = 0.5kg.$$

The mass of other components like hydraulic tank with the fluid, hydraulic cylinder , pin , bolts with nut, all in one are estimated to be 25kg.Additionally the design is proposed to lift a load of 3000kg ,and the total mass applied on the base plate is,

$$m_T = (48.6+39.7+4.4+0.5+25+3000)kg$$
$$= 3,118.2kg$$
$$P_T = m_{T*}g$$
$$= 3,118.2kg * 9.81m / s^2 = \mathbf{30.58954kN}.$$

Center connecting bar:

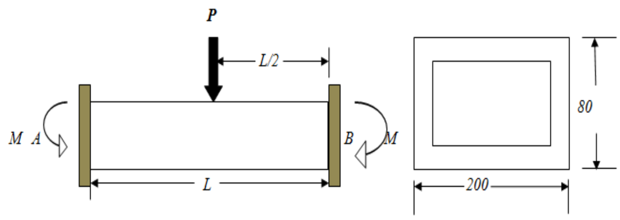

Figure 6.4. Center connecting Bar

Due to the load **P** the bar is subjected to equal bending moment at both end of the plate. At both end there are two reaction forces to encounter the bending of beam due to the load applied. And these are equals to half the load applied.

$$R = 15.294771kN$$
$$A = (200*80)-(193*73)mm^2$$
$$A = 1900mm^2 = 0.0019m^2.$$

Section Modules

$$Z = \left(200*80^3 - 186*66^3\right)$$
$$= 5.69*10^{-5} m^3$$

- Moment at the two end of bar

$$M_{A,B} = (P_T * L) / 4$$
$$= 30.58954KN * 0.516\_m) / 4$$
$$M = 3.946051KNm$$
$$\sigma_b = M / Z$$
$$= 3,940.51Nm / 5.96*10^{-5} m^3$$
$$= 661.159*10^5 N / m^2$$
$$\sigma_b = 66.1159MPa, \text{ Induced stress.}$$

The material selected for the center bar of the base plate is standard steel of Indian standard designation FeE520, having yield strength of 520N/mm^2. Taking the factor of safety equals 4.

$$\sigma = Sy / n = 520 / 4 = 130 \; N / mm^2.$$

Since the induced stress is less than the ultimate bending stress, then the bar is designed safely.

E. Design of pins

Pin 1

Due to the load applied at the end of the boom and the load by the boom itself, the pin that connect the vertical column and the boom is subjected to shearing stress. The shearing force applied on the pin that connects the vertical column with the boom can be obtained using the principle that summation of moment on the pin that connects the actuator and boom is zero.

$$\Sigma M_B = 0$$
$$0.7(R_C) = 0.1(W) + 0.9(P)$$

- W = (39.7Kg)(9.81m/s^2)

$$W = 389.457N$$

- P =29.43KN

$$R_C = (38.9457 + 26,487) / 0.7$$
$$= 37,894.208\text{N}$$
$$R_C = 37.894208\text{KN, Shear force on pin 1.}$$

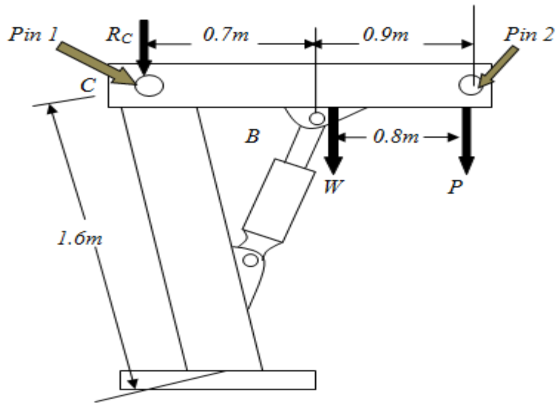

Figure 7. Pins of portable crane

Since the pin is subjected to high tensile and shearing stress, the material for the pin should be ductile. And the material selected is mild steel. From Indian standard designation of steel (Mechanical Engineering design by KHURMI and GUPTA, 8th edition) - Table 2.5 we have selected Fe 690 which has minimum tensile strength of 690N/mm^2 and minimum yield strength of 410N/mm^2.

$$\tau_{Max} = \sigma y / 2(n)$$
$$\tau_{Max} = 25.625\text{MPa.}$$

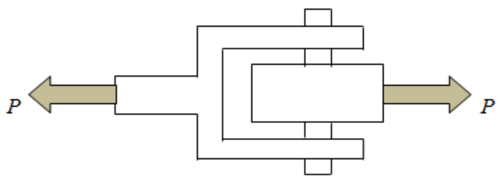

Figure 8. Pin supporting vertical column and Boom

Thus the pin will be subjected to double shear and then the pin is designed as follow.

$$\tau = P / 2A, \text{ P = Shear force,}$$
$$\text{A = Area subjected to shear stress}$$
$$A = R_C / 2\tau,$$
$$\pi d^2 / 4 = Rc / 2\tau,$$
$$d = \sqrt{2Rc / \pi\tau}$$
$$= (2 * 37.894208 * 1000\text{N})^{0.5} / (\pi * 25.625\text{N/mm}^2)$$
$$= 30.6827\text{mm},$$
$$\text{say d = 32mm, Diameter of pin at lower extreme.}$$

When the actuator is at its upper extreme, the pin that is used to connect the vertical column with the boom is designed as follow. Reaction at point C is obtained by applying the principle that summation of moment about point B is zero.

$$R_C (0.7) \cos 71.45°$$
$$= W(0.1)\cos 71.45° + P\ (0.8)\cos 71.45°$$

$$R_C = 389.45(0.1)(0.31813)$$
$$+ (29,430)(0.8)(0.31813) / 0.7((0.31813)$$
$$= (12.389 + 7490.0527) / 0.222691$$
$$R_C = 33,689.92\text{N}$$

$$d = [2 * R_C]^{1/2} / \pi * t$$
$$= \sqrt{(2 * 33,689.92) / (\pi * 25.625)}$$

d = 28.93mm, diameter of the pin when the actuator is at its upper extreme.

Taking the larger of the two is safe, and the pin that connects the vertical column and the boom should be **32mm** of diameter. The length of pin is slightly made to exceed the width of vertical column, that is

$$L = (150 + 15)\text{mm}$$
$$L = 165\text{mm}$$

Similar way;

For pin 2, d = 28mm

Pins for the Actuator, d = 74mm

F. Design of Hydraulic system

Hydraulic cylinder or also known as linear hydraulic motor is a mechanical actuator that is used to give a unidirectional force through a unidirectional stroke. Hydraulic cylinders get their power from pressurized hydraulic fluid, which is typically hydraulic oil. Design of hydraulic cylinder consists of design of cylinder, design of piston rod, hinged pin, design of flat end cover design of piston,

Design of Cylinder

The main function of cylinder is to retain the working fluid and to guide the piston. Hydraulic cylinder usually made of cast steel or cast iron, But for our design purpose we select cast steel because of high heat resistance, easily machine able and low cost.

The values taken for our design are;

$$\sigma_t = \text{permissible tensile stress}$$
$$= 80\text{N/mm}^2$$
$$p = \text{maximum pressure inside the Cylinder}$$
$$= 3.5\text{N/mm}^2$$
$$F = \text{load that is applied on the actuator}$$
$$= 212,615.4189\text{N.}$$

Then, inner diameter of the cylinder is (d_i):-

$$F = p * d_i^2 / 4 * p$$
$$d_i = \sqrt{4P / p * p} = \sqrt{4 * 212615.4189\text{N} / p * 3.5\text{N/mm}^2}$$
$$= 280\text{mm, say,}$$

- **Thickness of the cylinder;**

Let, t= thickness of the cylinder

$$t = r_i\{(\sqrt{\sigma_t} + p) / (\sigma_t - p) - 1\}$$

Where, $r_i = d_i/2$= 280mm/2=140mm

t= {[140mm {(√80N/mm² + 3.5N/mm²)/ 80N/mm² - 3.5N/mm²] - 1}

t = 9.22mm.

- **Outer diameter of the cylinder**

Let, d_o = Outer diameter of the cylinder

$$d_o = d_i + 2t = 280mm + 2(9.22mm)$$
$$= 298.44mm.$$

The cylinder head may be taken as flat circular plate whose thickness may be determined from the following relation.

Let, t_h = is the thickness of the head

$$\sigma_c = \text{allowable circumferential stress in N / mm}^2$$
$$= 30 \text{ to } 50Mpa.$$

Take, σ_c = 40Mpa (for our design purpose)

$$t_h = d_i (c * p / \sigma_c)$$

Where, C = constant = 0.1

t_h = 280mm ($\sqrt{}0.1*3.5N/mm^2$)/ 40N/mm^2 = 26.2mm.

- **And the length of cylinder is:-**

$$L = 2 * d_o$$
$$= 2 * 298.44mm = 596.88mm$$

Design of Piston:

Piston is moved by a fluid, or it moves the fluid which enters the cylinder. The most commonly material used for piston is cast iron, cast aluminum, forged aluminum, cast steel, and forged steel. But the material we selected for our design purpose is cast iron.

Piston is consists of the following parts, Head or crown, Piston ring, Skirt, Piston pin, Piston head:-

The thickness of piston head (t_H), according to Grashoff's formula is given by:-

$$t_H = \sqrt{(3P)(Do)} / 16\sigma_t, \text{ p = Maximum pressure,}$$
$$D_o = \text{Outer diameter of the piston,}$$
$$\sigma_t = \text{tensile stress,} = 38MPa.$$

Note: The outer diameter of the piston is assumed to be equal to the inner diameter of the cylinder. Therefore the outer diameter of the piston is 280mm.

$$t_H = \sqrt{(3*3.5*280)} / (16*38)$$
$$= 4.8mm, \text{say, } t_H = 5mm$$

Piston rings:

The piston rings are used to impart the necessary radial pressure to maintain the seal between the piston and cylinder bore. The radial thickness (t_r) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring.

$$t_r = D_o \sqrt{[(3*p_w) / (\sigma_t)]}$$

Where

- D_o = Outer diameter of piston
- Pw = pressure on the wall of cylinder, limited to 0.025 to 0.042N/mm^2
- σ_t = Allowable tensile stress in Mpa, it may be taken 85MPa to 110MPa for cast iron rings.

$$t_r = 280mm \sqrt{\left[(3*0.035)/(90)\right]}$$
$$t_r = 9.56mm.$$

And also the axial thickness of the piston ring (t_a) is:-

$$t_a = 0.7t_r \text{ to } t_r$$
$$= 7mm \text{ to } 9.56mm, \text{ adopting}$$
$$t_a = 8.5mm.$$

We also know that the minimum axial thickness of the piston ring is:

$$t_a = D_o / 10n_r,$$
$$\text{Where } n_r = \text{No of rings} = 4$$
$$t = 280mm / 10(4) = 7mm.$$

Thus the axial thickness of the piston ring as calculated (i.e. t_a = 7mm) is satisfactory.

The distance from the top of the piston to the first ring groove, i.e. the width of the top hand:-

$$b_1 = t_H \text{ to } 1.2\ t_H$$
$$= 5mm \text{ to } 6mm, \text{ taking, } b_1 = 5.5mm.$$

And the width of the other hand of the ring

$$b_2 = 0.75t_a \text{ to } t_a$$
$$= 6.63mm \text{ to } 8.5mm, \text{ taking, } b_2 = 7.5mm.$$

We know the gap between the free ends of the rings:

$$G_1 = 3.5t_r \text{ to } 4t_r$$
$$= 33.46mm \text{ to } 38.24mm.$$

And the gap when the ring is in the cylinder,

$$G_2 = 0.002D_o \text{ to } 0.004D_o$$
$$= 0.56mm \text{ to } 1.12mm, \text{ let's adopt}$$
$$G_1 = 36mm, \text{ and } G_2 = 1mm.$$

Piston Barrel:

It is a cylindrical portion of the piston. The maximum thickness (t_3) of the piston barrel may be obtained from empirical relation:

$$t_3 = 0.03D_o + b + 4.5,$$

Where

b = radial depth of piston, and it is taken to be 0.4mm greater than the radial thickness of piston ring

I.e. b = t_r + 0.4mm

Therefore, $t_3 = 0.03D_O + t_r + 4.9$

$$t_3 = 0.03(280) + 9.56 + 4.9$$
$$= 22.86mm,$$

say

$$t_3 = 23mm.$$

And the piston wall thickness (t_4) towards the open end is decreased and should be taken as:-

$$t_4 = 0.25t_3 \text{ to } 0.35t_3$$
$$= 0.25(23) \text{ to } 0.35(23)mm$$
$$= 5.75 \text{ to } 8.05mm, \text{ Adopting}$$
$$t_4 = 8mm.$$

G. Lifting system

Here the system is used to lift the load up and down as required. Thus the system works by allowing a certain amount of fluid to pass through the hose. The system is made of three basic components.

1. Electrical system box
2. Solenoid coils
3. Electric motor

Electrical system box:

This system box include power indicator, switch indicators, raising indicator, switch, and push buttons for raising the piston rod up and down. When the push buttons are pressed, electrical energy will turn the solenoid and the respective valves will be opened forcing the pressurized oil to pass through the hose.

Solenoid coils:

It is used to close and open the rising and lowering valve, when we press up and down push button. There are different types of solenoid; of these we used electrically actuating double acting coil.

Electric motor:

The motor is used to compress and pump the oil from the oil tank to the actuator through the hose.

Motor specification:

Power 3000W

Speed 2830/3620rpm

H. Lifting height

- The maximum height that the crane can lift is obtained as follow

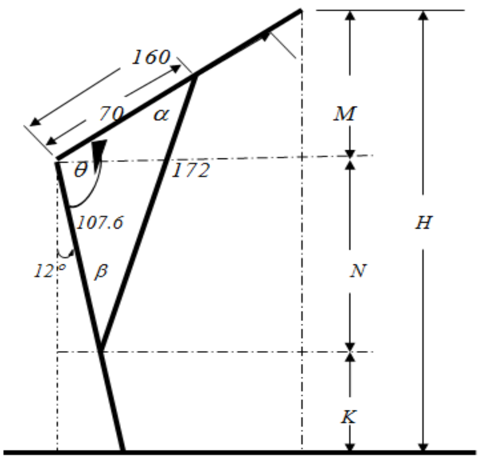

Figure 9. Height of the crane at maximum height

From cosine low:

$$107.6^2 = 172^2 + 70^2 - 2(172)(70)\cos(a)$$

$$\alpha = \cos^{-1}[0.9512]$$

$$\alpha = 17.96°$$

From sin low:

$$\text{Sin}17.96° / 107.6 = \text{Sin}\beta / 70$$

$$\beta = \text{Sin}^{-1}[0.20056]$$

$$\beta = 11.57°$$

$$\theta = 180° - (17.96 + 11.57)°$$

$$\theta = 150.46°$$

$$\text{M} = 160\text{Sin}(60.46)° = 139.2\text{cm}$$

$$\text{N} = 107.6\cos 12° = 105.24\text{cm}$$

$$\text{K} = 53\cos 12° = 51.84\text{cm}$$

$$\text{H} = \text{M} + \text{N} + \text{K} = 296\text{cm}$$

$$\text{H} = 2.96\text{m}$$

Figure 10. Designed portable crane at maximum height

- Minimum height that the crane can pick up objects from is obtained as:

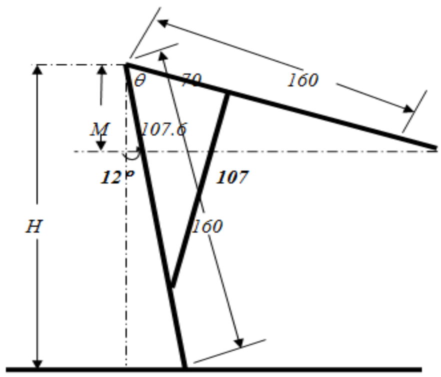

Figure 11. Height of the crane at lowest height

From Cosine law:

$$\theta = \text{Cos}^{-1}\left[(107.6)^2 + (70\ )^2 - (107)^2\right] / 2(107.6)(107.6)$$

$$\theta = 70.49°$$

$$\text{M} = 160 * \cos(82.49)° = 20.99$$

$$\text{h} = 160\cos 12° - \text{M}$$

$$\text{h} = 135.5\text{cm}$$

Figure 12. Designed portable crane at lowest height

I. Result, Advantage, and Disadvantage

A portable crane designed and developed its prototype here in BISHOPTU ABUTOMOTIVE INDUSTRY; Ethiopia is electric motors driven and moveable by wheels in the flours of the machine shop. It can also use as transporting objects from place to place in a shop replacing overhead crane. The operation is simple press to make the motor ON, very simple and suitable for the purpose.

Disadvantage of the portable crane may its using of electric power that its application impossible in the area where no electricity.

3. Conclusion and Recommendations

For many years cranes have been designed for lifting heavy objects with different capacity in different work sites. Portable cranes are of one type of cranes designed for lifting objects which are beyond the capacity of human being. As BISHOFTU AUTOMOTIVE INDUSTRY is one of the companies where in many different lifting operation is practicable.

Of this lifting operation by using portable and moveable crane which is not being use before, we have identified that there is the need for using portable crane to lift up objects these are beyond the capacity and difficult of human power. Thus this paper provides the design of each part of portable crane. And the design analysis for each part is checked that it is safe accordingly the size of each parts of the crane.

The maximum carrying capacity 3 ton, and maximum lifting height is estimated as greater than greater than 2.96m from the ground run by using 3KW electric motor rotating 2830/3620 rpm.

We recommend that anybody who interested to modify the current design of the portable crane for the production workshop of our design can make this work base study or can use as reference.

Symbols and Descriptions

A Cross sectional area
Aw Area of wire
b Width / depth
C Stiffness constant
c Distance b/n axis and end fiber
d nominal diameter
dc Core diameter
dh Diameter of hinged pin of piston
di Inner diameter
dp Diameter of piston
do Outer diameter
dr Diameter of rope
dw Diameter of wire
E Modulus of elasticity
e Eccentricity
G GAP
M Bending moment
m Mass
n Factor of safety /load factor
P Pressure
r_c Radius at neutral axis
r_n Radius at eccentricity
Sut Ultimate strength
Sy Yield strength
T Torque
t Thickness
V Volume
Wcr Buckling load
Z Section modulus
σb Bending stress
σc Compressive stress
σc Circumferential stress
σt Tensile stress
τ Shear stress
π Pie≈3.14159265
μ Coefficient of friction
ρ Density

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