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Improvement in the Design of Engine Crane for Modern Industries

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Abstract *- This paper presents an improvement in the design of engine crane, introducing a rotating mechanism comprising a gearing system, electric motor, solid shaft with flange, sleeve bearing, and a support bearing. The precision cut gears used are made of steel with endurance strength of 55 MN/m^2 (according to the American Gear Manufacturers Association (AGMA). In the design, the limiting endurance load of 17.06 KN and the limiting wear load of 22.96 KN are greater than dynamic load of 10.83 KN which implies the design is satisfactory from the standpoint of wear, dynamic and endurance loads. Since the use of cranes is indispensable in the industry, the design will aid productivity, safety of workers, ergonomics, efficiency, effectiveness and versatility of the cranes, for which they are designed and manufactured.*

Keywords*: design, dynamic load, engine crane, limiting endurance, rotating mechanism*

1. INTRODUCTION

Cranes are a complex combination of simple machines. Basically, a lever is used to crank a line of cable through a pulley, or system of pulleys, to lift heavy objects. The first cranes were invented by the Ancient Greeks and were powered by men or animals, such as donkeys. The cranes built by the Ancient Greeks were typically made with wood and rope, and anchored into the ground with large stakes. These cranes were used for the construction of tall buildings. The archaeological record shows that no later than 515 BC, distinctive cuttings for both lifting tongs and Lewis irons began to appear on stone blocks of Greek temples. Since these holes point at the use of a lifting device, and since they are to be found either above the centre of gravity of the block, or in pairs equidistant from a point over the centre of gravity, they are regarded by archaeologists as the positive evidence required for the existence of the crane. The heyday of crane in ancient times came under the Roman Empire, when construction activity soared and buildings reached enormous dimensions. The Romans adopted the Greek crane and developed it further. The simplest Roman crane consisted of a single-beam jib, a winch, a rope and a block containing three pulleys. Having thus a mechanical advantage of 3:1, it has been calculated that a single man working with the winch could raise 150 kg (3 pulleys x 50 kg = 150 kg), assuming that 50 kg represents the maximum effort a man can exert over a longer time period.

Today crane parts are made of different types of metals, heavy-duty cables, and can lift and move objects many times heavier than the loads of ancient times. In olden times, thousand of slaves had to be arranged whenever a heavy load had to be lifted or dragged Khurmi, (2009). Moving these objects involves some interesting physics. The load is attached to the end of the cable, and then cranked along pulleys to whatever height necessary. The most fun lies in the smallest piece, the pulley. This wonderful, simple machine reduces the force of the weight of an object, and allows less force to be used to move it than would be necessary with direct force, such as pushing it up a ramp. The pulley divides the weight, spreads it out along the cables, and multiplies the force being used to lift the object. The perfect condition for this is a pulley with zero friction and cables that do not stretch. While currently impossible to reach these conditions, technology has gotten much closer than the Ancient Greeks with their ropes and wooden pulleys. Now, with new materials and powerful motors, cranes can be used to lift much heavier objects than ever before. In any industry, time spent in doing a particular work is very important to management. The safety of workers can never be ruled out. Crane accidents and emergencies are occurring with increasing frequencies in ports around the world. This is understandable due to rapidly increasing population of cranes, increasing crane dimensions resulting in reducing visibility and operator control, frequent adverse weather conditions, and also crane maintenance and operating procedures not keeping up with increasing risks and demands of a fast paced modern terminal Larry, (2007). When all the forces that act on a given part are known, their effect with respect to the physical integrity of the part still must be determined Leonardo and George, (1995). It would therefore be reasonable to suppose that fatigue failure due to lack of allowance do not occur Raymond, (1990). On 21st May 2000 the top of a tower crane collapsed at Canada Square in the Canary Wharf area of East London. Tragically, three of the erection crew died in the collapse. The collapse occurred near the end of an operation to raise the height of the tower crane with an external climbing or jacking frame. The Health and Safety Executive (HSE) understands that at that time this was only the second collapse of a tower crane during climbing anywhere in the world. The first occurred in San Francisco on 28th November 1989. In the automobile industry for example, the engine crane is used to lift and mount car engines and other dexterous components during repair works or servicing. In a modernized automobile workshop, damaged engines are kept in a spacious room allocated for such purposes until vehicle owners approve of their state. Dismantling and cleaning of the entire engine is also carried out in this same room. As the damaged engines pile up in the room, there is the need to sell them to scrap dealers. Some technicians are made to halt their work in order to help load these damaged engines into a light duty truck. Lifting of such heavy objects, whose edges are sharp, with the hands, is very dangerous. The involvement of technicians who are busy on their job, are deplored to the loading, which is time wasting. The objective of this paper is therefore to create an improvement on the existing engine crane in order to aid productivity, safety of workers, ergonomics, efficiency and effectiveness, usefulness and versatility of the cranes, for which they are designed and manufactured.

II. MATERIALS AND METHOD

2.1 Design Features of Existing Crane

Figure 1 shows the existing design of the engine crane. For some designs, the upright support or column is inclined with reference to transverse load supporting member.

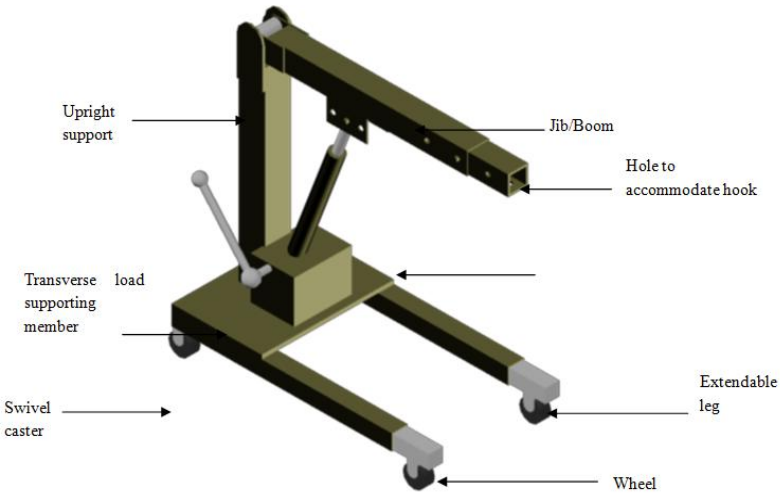

Fig.1 Existing Design of an Engine Crane

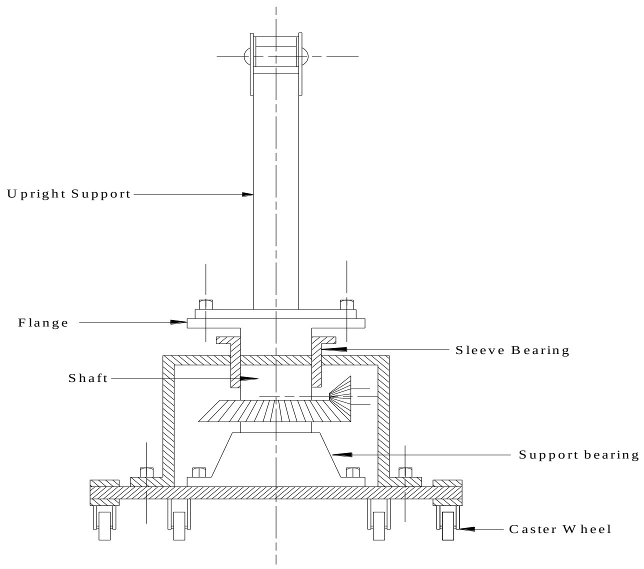

Fig. 2 Front View of the Rotating Mechanism of the Proposed Design

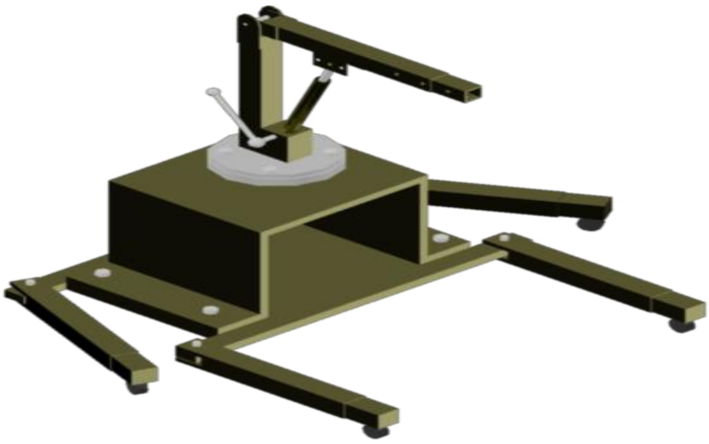

Fig. 3 Pictorial View the Proposed Design

2.2 Design Calculations and Analysis

2.3 Rotating Mechanism (design of gears employed)

Two cast iron bevel gears of endurance strength of 55 MN/m^2 having pitch diameters of 100 mm and 600 mm respectively were used. The tooth profiles are of $14\frac{1}{2}°$ composite form are selected to avoid interference of the meshing gears.

2.4 Design for strength for the bevel gears

The pinion is the weaker gear since both the gear and pinion are of the same material. Therefore design will be based on the pinion. Since the diameters of the gears are known, the Lewis equation (1) below will be applied

$$\frac{1}{my} = \frac{sb\pi}{F}\left(\frac{L-b}{L}\right) \qquad (1)$$

where

m = module based on the largest tooth cross section.

y = form factor based on the formative number or the teeth and type of tooth profile.

b = the face width of the gear,

s = allowed stress

F = the permissible force that may be transmitted

$$L = \text{cone distance} = \frac{1}{2}\sqrt{R_p^2 + R_g^2} \qquad (2)$$

R_P = pitch radius of pinion, R_g = pitch radius of gear

$\therefore L = \frac{1}{2}\sqrt{100^2 + 600^2} = 304.14$ mm

Face width $b = \frac{L}{3} = \frac{304.1}{3} = 101.3.$

Pitch line velocity $V = r\omega$

Where, $v = 0.1\left(\frac{30 \times 2\pi}{60}\right) = 0.157$ m/s

Power $=$ Torque $\times$ angular velocity

$= 450 \times \left(\frac{30 \times 2\pi}{60}\right) = 1413.7$ W

Force transmitted $F = \frac{power}{V}$ $\quad V = 1413.7/0.157 \quad F = 9004.56$ N

Endurance strength $s = s_o\left(\frac{6}{6+V}\right) = 55 \times 10^6\left(\frac{6}{6+0.157}\right) = 53.598$ MN/m^2

From equation 1, $\quad \Rightarrow \frac{1}{my} = \frac{53.598 \times 10^6 \times 0.1014 \times \pi}{9004.56}\left(\frac{304.14 - 101.38}{304.14}\right)$

$\therefore \frac{1}{my} = 1.264 \times 10^3$ N/m^2, allowable

Assuming form factor $y = 0.1$

$\Rightarrow m = (0.1 \times 1264)^{-1} = 5.259 \times 10^{-3} mm = 5.259\ m$

Number of teeth of pinion $N_p = \frac{\text{diameter of pinion } (D_p)}{\text{module}(m)}$

Formative number of pinion teeth $N_{f\,(pinion)} = N_p/\cos \alpha_p$;

$\cos \alpha_p = \frac{R_g}{L} = \frac{300}{304.14} = 0.9864$. Also,

Number of teeth of gear $N_g = \frac{\text{diameter of gear } (D_g)}{\text{module}(m)}$

Formative number of gear $N_{f\,(gear)} = N_g/\cos \alpha_g$

$\cos \alpha_g = \frac{R_p}{L} = \frac{50}{304.14} = 0.1644$

Values of standard modules in millimeters, taken from ISO/R54:

Preferred: 1, 1.25, 1.5,2, 2.5,3, 4,5,6,8,10,12,16,20,25,40,50,

Second Choice: 1.125,1.375,1.75,2.25,2.75,3.5,34.5,5.5,7,9,11,14,18,22,28,36,45,

From the values of the standard modules, we will try preferred module m = 1 mm

$$N_p = \frac{100}{1} = 100;\ N_{f(pinion)} = \frac{300}{304.14} = 101.379$$

Table 1 shows the form factor of a particular number of teeth for different pressure angles for use in the Lewis equation

Table 1: Form Factor (y) for Involute Gears

Numbers of teeth	**$14\frac{1}{2}^0$ Full-Depth Involute or Composite**	**20^0 Full-Depth Involute**	**20^0 Stud Involute**
12	0.067	0.078	0.099
13	0.071	0.083	0.103
14	0.075	0.088	0.108
15	0.078	0.092	0.111
50	0.110	0.130	0.151
60	0.113	0.134	0.154
75	0.115	0.138	0.158
100	0.117	0.142	0.161
150	0.119	0.146	0.165
300	0.122	0.150	0.170
Rack	0.124	0.154	0.175

With module m = 12

$$\Rightarrow N_p = \frac{100}{12} = 8.333$$

$$N_{f(pinion)} = \frac{8.333}{0.9864} = 8.4482$$

From Table 1 form factor y = 0.067 is chosen since N_p is less than 12

$\therefore \frac{1}{my} = \frac{1}{12 \times 0.067} = 1.244 \times 10^3$

The pinion is satisfactory; 1244.0 < 1264.0 (allowable)

$\Rightarrow N_g = \frac{600}{12} = 50$

$N_{f(gear)} = \frac{50}{0.1644} = 304.14$

Check for wear and dynamic effect

Checking for wear and dynamic effects using m = 12

The limiting wear load $F_w = \frac{0.75 D_p b K Q}{\cos \alpha_{(pinion)}}$ ……………………………………………………………………………(3)

where, $K = s_{es}^{\,2} \frac{(\sin \emptyset)(1/E_p + 1/E_g)}{1.4}$

s_{es} = surface endurance limit of the gear pair, N/m^2

E_P = modulus of elasticity of the pinion material, N/m^2

E_g = modulus of elasticity of the gear material, N/m^2

From Table 2, K for the gear material = 1330 kN/m^2

Table 2 is a tabulation of surface endurance limit and the stress fatigue factor for a combination of gear and pinion material

Table 2 Values for Surface Endurance Limit and Stress Fatigue Factor

Average Brinell Hardness Number of Steel pinion and steel gear		Surface Endurance Limit S_{es} (MN/m^2)	Stress Fatigue Factor K (kN/m^2) $14\frac{1}{2}^0$	Stress Fatigue Factor K (kN/m^2) 20^0
150 300 400		342 755 1030	206 1004 1869	282 1372 2553
Brinel Hardness Number, BHN				
Pinion	Gear			
150 250 200	C.I. C.I. Phosphor Bronze	342 618 445	303 1000 503	414 1310 689
C.I. Pinion	C.I. Gear	619	1330	1960

$$Q = \frac{2N_{f(pinion)}}{N_{f(pinion)} + N_{f(gear)}} = \frac{2 \times 8.4482}{8.4482 + 304.14} = 0.0540$$

Substituting the values of K, Q and D_P into equation (3)

$$F_w = \frac{0.75 \times 0.1 \times 1330000 \times 0.050}{0.164} = 32.8532 \text{ KN}$$

Endurance Load $F_o = s_o b y \pi \left(\frac{L-b}{L}\right) m$ …………………………………………………… (4)

$$F_o = 55 \times 10^6 \times 0.067 \times 0.1013 \times \pi \left(\frac{304.14 - 101.14}{304.14}\right) \times 0.012 = 9394.4 \text{ N}$$

Dynamic load $F_d = F + \frac{21V(Cb+F)}{21V + \sqrt{Cb+F}}$ ……………………………………………………….(5)

Where

F = transmitted force

C = a constant based on the tooth form, material and the degree of accuracy with which the tool is cut

From Table 3 below, C = 220 kN/m

Table 3 displays values of deformation factor for dynamic load check for the material for the gear and pinion in mesh

Table 3 Values of Deformation Factor C in KN/m

Materials		Involute Tooth form	Tooth Error-mm				
Pinion	Gear		0.01	0.02	0.04	0.06	0.08
Cast iron	Cast iron	$14\frac{1}{2}^0$	55	110	220	330	440
Steel	Cast iron	$14\frac{1}{2}^0$	76	152	304	456	608
Steel	Steel	$14\frac{1}{2}^0$	110	220	440	660	880
Cast iron	Cast iron	20^0 Full depth	57	114	228	342	456
Steel	Cast iron	20^0 Full depth	79	158	316	474	632
Steel	Steel	20^0 Full depth	114	228	456	684	912

From table 2, K = 1004 KN /m$^{2.}$

The limiting wear load $F_w = \frac{21V(Cb+F)}{21V + \sqrt{Cb+F}} \Rightarrow F_w = \frac{0.75 \times 0.1 \times 1004000 \times 0.050}{0.164} = 22.96 \text{ KN}$

Endurance Load $F_o = s_o b y \pi \left(\frac{L-b}{L}\right) m$

$$F_o = 100 \times 10^6 \times 0.067 \times 0.1013 \times \pi \left(\frac{304.14 - 101.4}{304.14}\right) \times 0.012 = 17.056 \text{ kN}$$

$$F_d = F + \frac{21 \times 0.154(220000 \times 0.1014 + 9004.5)}{21 \times 0.154 + \sqrt{(220000 \times 0.1014 + 9004.5)}} = 9004.5 + 1817.4$$

which gives $F_d = 10.827$ KN

Since the limiting endurance load $F_o = 17.056$ KN and the limiting wear load $F_w = 22.96$ KN are greater than the dynamic load, $F_d = 10.827$ KN, it implies that the design is satisfactory from the standpoint of wear, dynamic and endurance loads by American Gear Manufacturing Association (AGMA) standard.

2.5 Calculation of Shaft Parameters

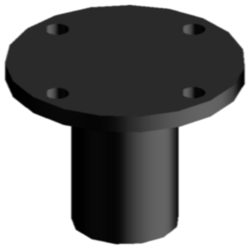

Fig. 4 Shaft for the Lifting Mechanism of the Crane

$K = \frac{GJ}{l}$

K = torsional stiffness;

l = length of shaft = 550 mm

G = Modulus of rigidity;

J = Polar moment of inertia, $J = \frac{\pi D^4}{32}$;

D = diameter of shaft = 160 *mm*

Polar moment of inertia of the shaft

$\Longrightarrow \quad J = \frac{\pi 0.16^4}{32} = 2.5133 \times 10^{-3}\ m^4$.

Table 4 shows the modulus of rigidity of alloy steel G = 81 GN/m^2

Table 4 Properties of Some Ferrous Metals

Materials	Modulus of Rigidity GN/m^2	Modulus of Elasticity GN/m^2	Yield stress MN/m^2
Malleable Cast Iron	61	150	420
Steel (mild)	83	207	308
Steel (Low carbon)	80	210	420
Steel (alloy)	81	200	600
Steel (stainless)	79	190	700

$\Longrightarrow \quad K = \frac{81 \times 10^9 \times 2.5133 \times 10^{-3}}{0.16} = 1.272 \times 10^9$ Nm

2.6 Surface speed of shaft

Surface speed of the shaft is defined as the number of feet travelled per minute by the shaft circumferentially. To calculate this value for the shaft, the following formula will be used:

Surface speed = π x d x rpm

where:

π = 22/7; d = Diameter of the shaft in inches; rpm = revolutions per minute

The shaft will be running at the speed of the gear keyed to it.

$$= \frac{\text{angular velocity of gear} \times 60}{2\pi} = \frac{0.5333 \times 60}{2\pi} = 5.0898 \text{ rpm}$$

Surface speed = π x 130 x 5.0898 = 2078.704 m/s

2.7 Maximum Load to be carried by the Base Plate

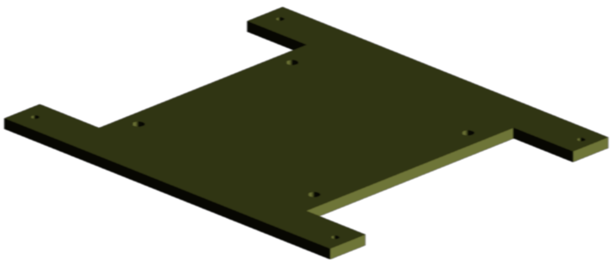

Fig. 5 Isometric View of the Base Plate

The maximum load that the base plate can carry should not exceed the allowable stress of the plate material

Shear stress = $\frac{\text{Force}}{\text{Area}}$

Since the maximum load to be carried by the crane = 900×9.81= 8.84 KN, it will be assumed the maximum load resting on the base plate is three times the load to be carried by the crane.

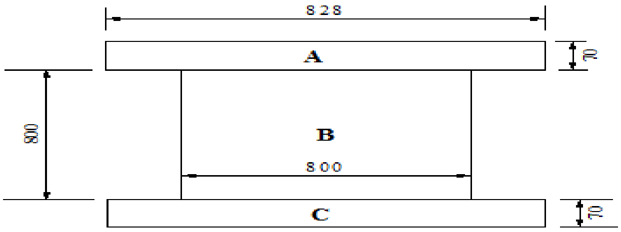

Fig. 6 Schematic Top View of Base Plate with Dimensions (in mm)

Total area of base plate is equal the sum of areas A + B + C

= (8.28×0.70) + (8.00×8.00) + (8.28×0.70) = 755.92 m^2

Shear stress =$\frac{8839 \times 3}{755.92}$= 35.079 N / m^2

Allowable shear stress of plate material (medium carbon steel)

= 0.4 × Yield Stress (Hamrock et al, 2003)

Table 5 shows some properties of medium carbon steel

Table 5 Properties of Medium Carbon Steel

Material	**Density kg / m^2**	**Modulus of Elasticity GPa**	**Yield Strength MPa**
Medium Carbon Steel	7850	207	520

(Source: Hamrock et al, 2003)

Allowable shear stress of plate material = 0.4 × 520 MPa = 208 MPa

Since the allowable shear stress of the plate is greater than that of the load, it implies that the plate can carry the entire load without fracture.

2.8 Bolt Design

The arrangement in Fig 6 can be classified as eccentricity load in bolted joints where the bolt load is parallel to the axis of the bolt. The load P will tend to rotate or turn the bolted joint about the edge O. The elongation and hence the stress in the bolt will be proportional to its distance from O, so that if all the bolts have equal cross-sectional areas, the load in the bolt will also be perpendicular to its distance from the turning edge.

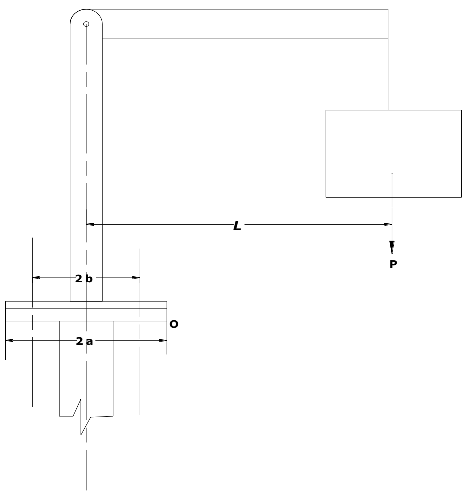

Fig. 7 Schematic Diagram of the Existing Crane Bolted to the Flange of the Shaft

Maximum force or load which may come on a bolt is given by

$$F_{max}=\frac{2P(1-a)(a+bv)}{z(2a^2+b^2)}$$ Sharma, (2003) …………………………(6)

where
a = radius of flange on which the pillar of the crane will be bolted;
b = radius of bolt circle.
Also

$$F_{max}=\frac{\pi}{4}d_c^{\ 2}\times f_t$$

where
d_c = crest diameter, f_t = tensile force
Six bolts will be employed to bolt the pillar of the crane to the flange of the shaft. The higher the number of bolt the lesser the force or load which may come on a bolt.

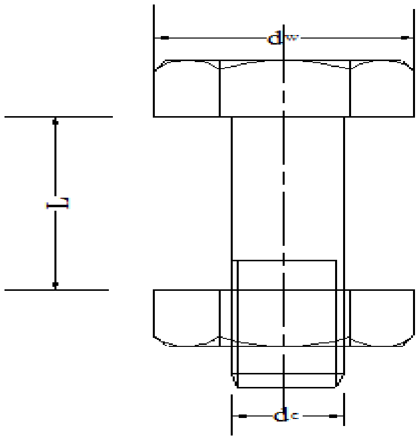

Fig. 8 Bolt and Nut Assembly

Maximum load to be lifted by crane P = 900 x 9.81 N
a = 0.4 *m*; b = 0.3 *m*
Substituting these values into equation 6

$$F_{max}=\frac{2\times 8.829\times 10^3(1.93-0.2)(0.15+0.2)}{6(2\times 0.2^2+0.15^2)} = 21.2775\times 10^3\ N$$

We use steel bolts having elastic stress of 280N/mm^2 and a factor of safety 4.

From equation two $d_c=\sqrt{\frac{4}{\pi f_t}F_{max}}$

where

$$f_t = \frac{280}{4}=70\ N/mm^2$$

$$\Rightarrow \square_{\square} = \sqrt{\frac{4}{70\times\square}\times 21.569\times 10^2}\ \therefore d_c=19.81\ mm$$ Approximately d_c = 20 mm

Frustum cone diameter $d_w = 1.5\times d_c = 1.5 \times 20$ mm = 30 mm

2.9 Selection of the Material of the Components of the Crane
Material Selection

Some typical failure mechanisms in lifting equipment are due to fatigue, wear, corrosion, and ductile or brittle fracture. Components may fail from one or more of these mechanisms or due to other failure mechanisms. For this reason, a suitable material which is cheap but capable of withstanding every failure mechanism is required. Steel is suitable for this design. Medium carbon steel will be employed in the design for the legs, bedplate and the housing. The versatility of steel as an engineering material is due to the range of good mechanical properties and forming processes. Steels, generally combines most properties as high tensile, compressive and shear strength, good ductility and malleability, good machinability, toughness and hardness, moderately high thermal and electrical conductivity, high melting point and ease of heat treatment.

III. DISCUSSION OF RESULTS

The improvement in the design of the engine crane introduced of a rotating mechanism comprising of a gearing system, electric motor, a solid shaft with flange, sleeve bearing, and a support bearing could be seen in Fig.1 showing existing design of an engine crane and Fig. 3 showing the pictorial view the proposed design. The base of the existing design changed in other to accommodate the rotating mechanism and also give the whole system the needed stability. Fig. 4 shows the shaft for the lifting mechanism of the crane. A base steel plate was employed to accommodate the rotating mechanism and its housing as shown in Fig. 5 showing the isometric view of the base plate. The legs increased from two to four i.e. two in front and another two at the rear, and bolted to the plate at its four corners, such that they can be adjusted to form an x-shape during operation. This will give maximum stability to the crane at any angle. The legs can also be extended. Four extra casters are fixed to the bottom of the plate which will accommodate the rotating mechanism. Two of these casters, which is positioned in front, can swivel. The pillar is bolted securely onto the flange of the shaft. Both gears made of steel with endurance strength of 55 MN/m^2 (according to the American Gear Manufacturers Association (AGMA). The gears were precision cut gears. A new fatigue factor of 4 is selected. The condition of the limiting endurance load of 17.056 KN and the limiting wear load of 22.96 KN are greater than dynamic load of 10.827 KN implies the design is satisfactory from the standpoint of wear, dynamic and endurance loads

IV. CONCLUSION

The redesign of the engine crane will replace the manual means of lifting car engine from one point to the other in a modern workshop. The construction is costly because of the introduction of other components into the existing design, but it will eliminate the risk in lifting the engines with the hands.

V. RECOMMENDATION

The design of the support bearing should be looked at for improvement. Design of the gears employed to reduce the speed from the motor can be improved upon to reduce their sizes.

REFERENCES

[1] Hamrock J. Bernard, Schmid R. Steven, Jacobson O. (2003), "Fundamentals of machine element",2nd Edition. McGraw Hill. New York. pp 726 – 729.

[2] Khurmi, R.S. (2009) "A Textbook on Engineering Mechanics" S. Chand and Company Ltd, Ram Nager, New Delhi 20th Revised Edition, pg171. ISBN 8121926165.

[3] Larry Lam, (2007) "Crane Accidents and Emergencies –Causes, Repairs, and Prevention" Presented at TOC ASIA 2007 Hong Kong

[4] Leonardo Spiegel, P.E and George F. Limbrunner, P.E (1995) "Applied Static and Strength of Materials" Prentice Hall, 2nd edition, 754pp. ISBN 0-02-414961-6.

[5] Raymond, A. Higgins (1990), "Properties of Engineering Materials" 4th Edition, Edward Arnold pp 348 ISBN 0340380349.

[6] Ryder G.H., (1969) "Strength of Materials",3rd Edition, Rajiv Beri, New Delhi. pp131.