Design of Tire Changer

Melku Abebe[1], Kirubel Akele[1], Melaku Solomon[1], Meron Bancha,[1] Teshome Dengiso Megiso[2]
≠1Faculty of Mechanical and Production Engineering, Arba Minch Institute of Technology Arba Minch University
≠2Asst. Prof. Arba Minch University, Arba Minch Institute of Technology, Faculty of Mechanical and Production Engineering, P.O. Box: 21, Arba Minch, Ethiopia

Abstract

In present situation in case of tire worn-out or air leakage in heavy and light vehicles, conventional way of mounting and dismounting of tire from the rim plays a great role. The conventional method which involves hammers and rods for removing the tire from heavy wheels leads to the chances of tire wear, could lead to serious injuries, consumes more efforts, time and wasting of different equipment's. In the present context of market situation, there is always a need for better design of the equipment with the maximum reduction in cost. Additionally, the human comfort which enables a person to operates it with ease and least consumption of human effort the tire changer equipment is used by vulcanizing shops, retailers of tires, tire re-treading works and servicing stations for easy tire demounting and mounting operation from and/or on to the wheel disc/rim. This paper presents the tire changer machine with no need of manpower and only a single person can do the entire job in a short period of time. The main objective of this paper is to design of tire and rim inserter and separator machine, which is operated electrically, mechanically, pneumatically, and less human effort.

Key words — *Tire changer, tire removing method, tire press, design of tire changer*

I. INTRODUCTION

Tires are used for certain period and they need to be maintained. Tire maintenance is occupied in several methods based on the tire failure such as: changing tread, removing leakage, changing inner tube and changing wheel (rim). This kind of works are done, but to do this work there are two primary tasks to be done and this will be separating tire and rim and reinserting the tire into the rim. This task is done by tire changing machine. Tire changing machine can be either manually or automatically operated. Manual tire changers are good and considerable for small automobiles with small tire size, the labour effort required is affordable but for higher truck, buses and heavy-duty trucks the tire is larger in size and the labour effort required is relatively higher and very difficult task. Therefore, automatic tire changing machines are used, to gain power from several system components such as: hydraulic system, pneumatic system, lever system, electrical motors and several accessories [1]. In Anbessa city Bus Service Enterprise (ACBSE)Addis Ababa, Ethiopia, In the maintenance department repairing of the bus tires fundamental task. As observed, the maintenance of tire before repair from the bus is somewhat easy. But there is a difficulty of extract and insert the tire from rim due to the absence of tire inserter and extractor machine. That is why this paper focused to design a pneumatic tire inserter and extractor machine for the tire to reduce labour cost, to reduce repair time and to reduce physical load required. The design enables the technicians to handle this job easily and quickly without consuming high effort.

II. METHODOLOGY AND MATERIALS

This paper started from observation, gather information and discussion about the design of machine. After gather and collect all related information and obtain new idea and knowledge about the project would continue with design process. After numerous analysis and sketches the best design proposed. Then the selected design transferred using AutoCAD and Solid Work.

The data collection performed by using interview, field observation, concept design, and detail design, analysis/ calculation, checking, redesign, parts and assembly drawing

The tire changer physical properties such us length, thinness, diameter of different components. Based on maintenance shop ergonomic style, selection done considering the overall length or diameter which the tire changer is operated.

III. DESIGN ANALYSIS

A. Design of bolts for machine fixing to ground

Mass of tire 150Kg (maximum)[3]

Total load = pressing force + weight of tire

Total load = 1524.7+(150*9.81) = 2996.2 N

Bolt material ASTM A307 grade A carbon steel

Tensile strength = 250MPA Shear strength = 145MPA

Young's modulus = 200MPA

Diameter of the bolt is assumed standard M20

B. Selection of motor

Single phase electrical motor that operates with in the range of 220 – 230/400volt, 50/60 Hz electrical power supply and according to the tire size, most tire changing machines uses over 1000-1800Nm torque for separation operation for automobiles and 2000 Nm – 3000 Nm used for heavy duty. Factors considered while selecting the motor:

i. Design of speed reducer gearbox- spur gear

The number of input shaft gear teeth is 22.

First stage gear (spur gear)

We know V.R = 3:1 = (No of teeth on the driven gear)/(No of teeth on the driver gear)

T2=V.R*T1 = 22* 3 = 66

Second stage gear

Second stage driver gear number of teeth is 15.

V.R = 4:1 = (No of teeth on the driven gear)/(No of teeth on the driver gear) => V.R2 = T4/T3 ➔ T4 = T3*4 = 15*4 = 60

Third stage gear

Third stage driver gear number of teeth is 15.

V. R3=3:1 = T6/T5 ➔T6 = 15*3 = 45

Gear Design

Let gear module (m) is 3 mm,

Gear – 1 and since, m = D1/T1

D1 = T1*m = 3*22 = 66mm

Circular pitch (Pc1) = πD1/T1 = 3.14*66/22= 9.42mm

Addendum = m = 3mm Dedendum =1.2m = 3.6 mm

Centre distance between the shaft is

= m (N1+N2)/2 = D1+D2/2 = 132mm

Do = D1+2m = 66+6=72mm

Root diameter (Dr) =D1-2m=66-6=60mm

Gear -2 **=>,** m=D2/T2 ➔ D2=m*T2 = 3*66=198mm

Circular pitch (Pc2) = πD2/T2 =π*198/66= 9.42mm

Addendum = 1m = 3mm

Dedendum =1.2m=3.6mm

Gear – 3➔ M= D3/T3 => D3=T3*m =15*3=45mm

Circular pitch (Pc3) = πD3/T3=3.14*45/15=9.424

Addendum = 1m = 3mm Dedendum = 1.2m 3.6mm

Outside diameter = Do=D3+2m=45+6=51mm

Root diameter = Dr =D3-2m =45-6 = 39mm

Gear 4➔ M=D4/T4 => D4=m*T4 =60*3=180mm

Circular pitch (Pc4)=πD4/T4= 3.14*180/60= 9.424mm

Addendum = 1m= 3mm

Dedendum =1.2m=3.6mm

Outside diameter = Do=D4+2m=180mm

Root diameter = Dr=D4-2m=180 – 6=174mm

Centre distance between the shafts is

=D4+D3/2=45+180/2 = 112.5mm

Gear 5➔ M=D5/T5 => D5=mT5 =15*3=45mm

Circular pitch (Pc5)=πD5/T5 = 3.14*45/15=9.424mm

Addendum = 1m= 3mm

Dedendum =1.2m=3.6mm

Outside diameter = Do=D5+2m=45+6=51mm

Root diameter = Dr=D5-2m=45-6=39mm

Gear 6 ➔ M=D6/T6 , D6=mT6 =45*3=135mm

Circular pitch (Pc6) =πD6/T6 = 3.14*135/45=9.424mm

Addendum = 1m= 3mm

Dedendum =1.2m=3.6mm

Outside diameter = Do=D6+2m=135+6=141mm

Root diameter = Dr=D6-2m=135-6=129mm

ii. Force analysis (spur gear)

Only tangential force is considered

Ft (1,2) = P/V1-2 Where; V1-2= pitch line velocity

V1-2 = π*D1N1/60 =3.14*66*1800/60 = 6.22m/s

Ft1-2 = P/V1-2 =0.75*1000/6.22=120.57N

Pitch line velocity of input shaft gear; Ft (3-4) = P/V3-4

V3-4 = πD3N3/60=3.14*45*600/60=1.4137m/s

Ft (3-4) = 0.75*1000/1.4137=530.5N

Ft (5-6) =P/v5-6 => V5-6=πD5N5/60 =3.14*45*150/60 =0.3534m/s

Ft (5-6) =0.75*1000/0.3534=2122.24N

iii. Stress Analysis (Spur Gear)

Ft=2122.24N, Since, Pd = T5/D5=π/Pc5 =0.333mm

Material for gear C-30, 20° full depth in volutes teeth and safety factor of 1.5[4][5]

Allowable stress $(\delta all) = \frac{bF_t l}{bt^2} = \frac{F_t P_d}{b}\Upsilon$ (for only static load)

Where ϒ is Lewis factor, since ϕ is 20° full depth involutes

$$\Upsilon = 0.154 - 0.912/T_5 \quad \Upsilon = 0.154 - \frac{0.912}{15} = 0.0932$$

$$\delta_{all} = \frac{F_t P_d}{b\Upsilon} * Kv * Kf * Ko * Km$$

$$\Upsilon/K_f = j$$

$$\delta_{all} = \frac{F_t P_d}{bj} * Kv * Kf * Ko * Km$$

Where;

Kv =1.602 for only one-way bending

Ko =1 for uniform shock

Km =1.2 load distribution factor

J = 0.225(geometry factor)

From the above, face width of the gear (b)

$$b = \frac{FtPd}{\delta(all)} * kvkokm = 2122.2 * 0.333 * 1.602 * 1 * \frac{1.22}{300} * 0.225 = 20.12mm$$

Since tangential force on the gear 5 & 6 are same, based on this tangential force calculate face width of the gear then we get b_5=20.12mm. Use face width b6' same value as b5 so b6 = 20.12mm. Other gear face width we find similar step like b5&b6. B5 will be get 1.14mm, but this much dimension difficult to manufacture and for the sack of standard we use 5mm face width. Since b1 and b2 are same size. Similarly, find gear face width 3, then we get 5mm. like the above method we use face width for gear 4 is 5mm [1].

v. Shaft Design (For Spur Gear)

To determine that of shaft for spur gear, we followed the following procedure, finally we get diameter of e shaft.

- Normal load (Wn), acting between the tooth surface y

$Wn = Wt/cos\phi = 120.57/cos20 = 128.3N$

Where; Wt = tangential load, Φ=pressure angle

The weight of the gear

$Wg = 0.001 * 18Tgbm^2(N) = 0.001 * 18 * 22 * 5 * 5^2$

=3.245N

- resulting load on the gear is given by

$Wr = \sqrt{Wn^2 + Wg + 2Wn + Wgcos\phi}$

$Wr = \sqrt{128.3^2 + 3.245^2 + (2 * 128.3) + 3.245cos20}$
=129.34N

- Bending moment due to the resultant load

$M = Wr * x$

X=the distance between the center of gear and the center of bending

Assuming that the pinion is overhung on the shaft and taking overhang us 100mm, therefore bending moment on the shaft due to resultant load.

$M = 129.34 * 100 \quad M = 12934\ Nmm$

- Combined effect of torsion and bending

$Te = \sqrt{M^2 + T2}$

$T = twisting\ moment = Wt * Dp//2$

$= 120.57 * \left(\frac{66}{2}\right) = 3978.81N - mm$

Equivalent twisting moment,

$Te = \sqrt{12934^2 + 3978.81\text{\^{}}2} = 13532.157N - mm$

$Te = \frac{\pi}{16} * \tau * d^3$

Where;

τ=shear stress (Indian standard designation) properties of steel used for shaft. For a shaft strength an alloy steel line nickel- chromium steel is used with ultimate tensile strength (δy) =320mpa.

Now shear stress of the material of the gear shaft

$\tau = 0.18\delta u = 0.18 * 560 = 100.8mpa$

$Te = \frac{\pi}{16} * \tau * d^3$

$d^3 = \frac{Te * 16}{\pi * \tau} = \frac{13532.157 * 16}{\pi * 100.88}$

$D = 8.8mm\ take\ 10mm$

vi. Shaft Design (For Spur Gear)

To determine that of shaft for spur gear, we followed the following procedure, finally get diameter of shaft

(1) Normal load (Wn), acting between the tooth surface

$Wn = Wt/cos\phi = 530.5/cos20 = 564.546N$

(2) The weight of the gear is given by

$Wg = 0.00118 * Tg * b * m^2(N) = 0.00118 * 15 * 5 * 5^2$
$= 2.2125N$

(3) resulting load on the gear is given by

$Wr = \sqrt{Wn^2 + Wg^2 + 2WN + WGcos\phi}$
$= \sqrt{319848.2523} = 565.55$

(4) Now bending moment due to the resultant load

$M = Wr * x$

Assuming that the pinion is overhang on the shaft and taking overhang as 20mm,

$M = Wr * 20 = 565.55 * 20 = 11311.0256N$

(5) Since the shaft is under the combined effect of torsion and bending, therefore we shall determine the equivalent torque, we know that equivalent torque,

$Te = \sqrt{M^2 + T^2}$

T=twisting moment=Wt*D3/2

$T = 530.5 * \frac{45}{2} = 11936.25N$

Therefore equivalent twisting moment

$Te = \frac{\pi}{16} * \tau * d^3 2 \qquad d^3 2 = Te * \frac{16}{\pi} * \tau$

$\sqrt[3]{d}2 = Te * \frac{16}{\pi} * \tau$

$d2 = \sqrt[3]{(16444.25 * 16)/(\pi * 100.8)}$

$d2 = 9.4 = 10mm$

vii. Shaft design (for spur gear)

To determine of shaft for spur gear, followed the following procedure, finally get diameter of shaft.

(1) First of all, find the normal load (Wn), acting between the tooth surface. It is given by

$Wn = Wt/cos\phi = 2122.24/cos20 = 2258.44N$

(2) the weight of the gear is given by

$Wg = 0.00118 * T5 * b * m^2 N = 0.00118 * 15 * 20 * 5^2(N)$
$= 8.85N$

(3) resulting load on the gear is given by

$Wr = \sqrt{Wn^2 + Wg^2 + 2Wn + Wgcos\phi}$

$Wr = 2259.4589\ N$

(4) Now let us calculate the bending moment on the shaft due to resultant load.

$M = Wr * x$

Assuming that the pinion is overhang on the shaft and taking overhang as 35mm, bending moment due to resultant load.

$M = Wr * 35 = 79081.06329N$

Since the shaft is under the combined effect of torsion and bending, the equivalent toque[4][5]

$Te = \sqrt{M^2 + T^2}$

$T = twisting\ moment = Wt * \frac{D5}{2} = 47750.4N$

Therefore, equivalent twisting moment

$Te = \sqrt{M^2 + T^2} = 92379.1928N$

Now the diameter of the gear shaft (d) is determine by using the following relation, i.e

$$Te = \frac{\pi}{16} * (\tau) * d^3 3$$

$$d^3 3 = Te * \frac{16}{\pi * \tau}$$

$$d3 = \sqrt[3]{Te * \frac{16}{\pi * \tau}} \qquad D3 = 16.7mm = 17mm$$

viii. Bolt for fix gear box

4xM5 with material is the ASTM low carbon steel selected.

ix. Radial ball bearing

From various type of radial ball bearing, we select angular contact bearing is selected.

Since in 6^{th} gear there will be high radial and tangential load or force or maximum amount. Therefore, the bearing will be selected by, the maximum load and it is used for all shafts for gearbox.

The 6^{th} Gear's tangential force (axial) = 2471.5N

Dynamic equivalent load for rolling contact bearing

The dynamic equation load may define as the constant stationary radial load which is applied to a bearing under the actual condition of load and rotation.

$W = X \times V \times W_R - Y \times W_A$ Where;

V = A rotation factor = 1; for all bearing when the inner race is rotating

W = equivalent radial (W) W_R = radial load

W_A = axial or thrust load X = radial load factor

Y = dynamic (thrust factor)

Single row angler constant $\frac{W_A}{W_R} \le e$

Y = 0 and X =1 ball bearing from design data

Dynamic load rating for rolling contact bearing variable load

$L = (\frac{C}{W})^2 = 10^6$ L = rate lift

C = basic dynamic load rating W = equivalent dynamic load

K = 3 for all ball bearing

⇨ Single row angular bearing; lifetime of bearing

$L = (\frac{C}{W})^k \times 10^6$ where C = 7.8 selected

$W = X \times V \times W_R - Y \times W_A$ $\frac{W_A}{W_R} \le e = 1.14$

$W = d \times d \times 2873.8 - 0.73 \times 2471.5$

$W_A = W_R$ 0.86 W = 1861N $W_R = \frac{W_A}{1.14}$ = 1861N

Bearing subjected to shock load our service factor (k_s) will taken design data

$k_s = 1.5$ ∴ $W = k_s \times W$ = 1.5 × 2873.8 2792.42N

$L = (\frac{C}{W})^K \times 10^6$ = 21.792 × 10^6 revolution

Thus, bearing selection is safe

Dynamic equivalent load for rolling contact bearing

$W = \times X V \times W_R - Y \times W_A$ X = 1 Y = 0

Where; V = rotation factor = 1, for all bearing when it is inner race rotating

W = equivalent radial load, W_R = radial load

W_A = axial or thrust load, X = radial load factor

Y = axial load factor (thrust) (dynamic load factor)

Dynamic load rate for rolling contact bearing under variable load

$L = (\frac{C}{W})^k \times 10^6$ $\frac{W_A}{W_R} = e$

$W = X \times V \times W_R - Y \times W_A$ $W_R = \frac{W_A}{e}$

W = 465.35N W_R = 465.35N

Let the bearing subjected to light shock load service factor (k_s)

$\therefore W = k_s \times W = 1.5 \times 105.76$ **= 698N**

Know we us calculate the life time (L)

$L = (\frac{C}{W})^k \times 10^6$ C = 4

$= (\frac{4 \times 10^3}{698N})^3 \times 10^6$ = 188 × 10^6 rev

∴ *The bearing section safe and the standard material selected for bearing is Chrome steel..*

x. Design of key

A key is mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always inserted parallel to the axis of the shaft [1][4][5][6].

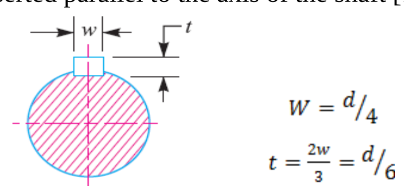


Figure1 rectangular sunk key

Key with shaft diameter of 4mm, shearing stress for mild steel key material is $\tau = 42Mpa$

Crushing stress is σ_c = 70Mpa

- Width of key = > W = 4mm
- Thickness of key => t = 4 mm

$T = l * w * \tau * d/2 = l * 4 * 42 * 10/2 =$

$T = 840 * l\ Nmm$

And tensional shearing stress of the shaft.

$T = \frac{\pi}{16} * \tau * d^3 = \frac{\pi}{16} * 42N/mm^2 * 10^3$

= 8246.68 Nmm.. (2)

From above equation (1^{st}) we have:

$T = 840 * l\ Nmm$

$8246.68\ Nmm = 840 * l\ Nmm$

$l = \frac{8246.68}{840} = 9.8 \sim 10\ mm$

Strength (torque transmitted) of the key.

$$T = l * \frac{t}{2} * \sigma_c * \frac{d}{2}$$

$= l * \frac{4}{2} * 70 * \frac{10}{2} = 700 * l\ Nmm$........................... (3)

From eqn (2 & 3) we have

$700 * l\,Nmm = 8246.68\,Nmm$

$l = \frac{8246.68}{700} = 11.7\,mm$

taking the larger of the two values, we have length of key

$l = 11.7\,mm$

xi. Design of bevel gear

The bevel gears transmit power at a constant velocity ratio between two shafts whose axes intersect at a certain angle. The pitch surfaces for the bevel gear are frustums of cones [1]. The two pairs of cones in contact is shown in Fig 2.

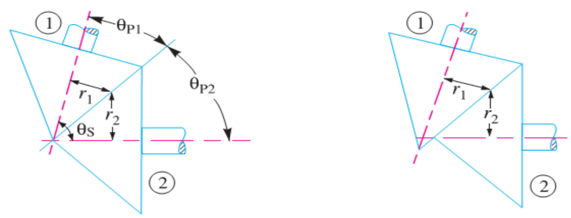


Figure 2 Pitch surface for bevel gear

The velocity ratio,

$V.R = DG/DP = TG/TP = NP/NG$

Where; $NP\ (speed\ of\ pinion) = 50\ RPM$

$NG\ (speed\ of\ gear) = 20\ RPM$

$V.R = \frac{NP}{NG} = \frac{50}{20} = 2.5$

We know, $V.R = \frac{DG}{DP}$

$DG = V.R * DP = 2.5 * DP$

Assume Pinion Number of Teeth Will Be 25

$TG = V.R * TP = 2.5 * 25 = 62.5 = 63$

We Know That Pitch Line Velocity

$V = \frac{\pi DPNP}{1000} = \frac{\pi DPNP}{1000}$

LET $DP = 50\ MM$

$V = \pi * 50 * \frac{50}{1000} = 7.854 mm/min$

Let ,$Wt = 2471.5N$

$= Force\ Due\ To\ Tire\ Load + Pressing\ Forces$

$150 * 9.81 + 1000N = 2471.5N$

Since, Assumption for $Tire\ Weight = 150kg$

$Pressing\ Forces = 1000N$

Now We Can Calculate Torque Acting On the Pinion

$T = P * \frac{60}{2\pi NP} N/M$

Where; P – Power Transmitted From Motor

Np – Speed of the Pinion in Rpm

$T = 0.75KW * \frac{60}{2\pi} * 50 = 0.75 * 1000 * \frac{60}{2\pi} * 50$

$T = 143.239N/M$

We Now Tangential Load on the Pinion

$Wt = 2471.5N \qquad Wt = \frac{2t}{Dp}$

Where Dp – Pinion Diameter;

Wt- Tangential Load on the Pinion Gear

T - Torque

$Dp = \frac{2t}{Wt} = 2 * \frac{143.239}{2471.5}$

$Dp = 115.9mm = 116mm$

$Vr = \frac{DG}{DP} \qquad Dg = Dp * V.R$

Where; $V.R = 2.5$

$Dg = 2.5 * 116mm = 290mm$

Outside Or Addendum Cone Diameter Knows As

$Do = DP + 2a\cos\theta p$

Where; a – addendum

θp – pitch angle

When the angle between the shaft axis is 90° , then the shaft axis is 90°

i.e. $\theta s = 90°$, then $\theta p1$ and $\theta p2$ became

$\theta p1 = tan^{-1}(1/VR) = tan^{-1}\left(\frac{DP}{DG}\right) = tan^{-1}\left(\frac{TP}{TG}\right) = tan^{-1}\left(\frac{NG}{NP}\right)$

$\theta p2 = tan^{-1}(V.R) = tan^{-1}\left(\frac{DG}{DP}\right) = tan^{-1}\left(\frac{TG}{TP}\right) = tan^{-1}\left(\frac{NP}{NG}\right)$

Pitch angle of the pinion becomes;

$\theta p1 = tan^{-1}(1/VR) = 0.38 = 21.8 = 22°$

Pitch angle of the gear becomes;

$\theta p2 = tan^{-1}(VR) = 68.2° = 68$

We know the formative number of teeth for pinion.

$T_{EP} = Tp\sec\theta p1 = 25\sec 22° = 26.963$

We know the formative number of teeth for the gear

$T_{Eg} = Tg\sec\theta p2 = 63\sec 68° = 168.18$

Pressure angle of bevel gear is 20° stub involute system.

Therefore, tooth form factor for the pinion

$Y'p = 0.175 - \frac{0.841}{T_{Eg}} = 0.175 - \frac{0.841}{168.18}$

$= 0.1699$

Tangential load on the pinion

$Wt = \frac{2T}{Dp} = \frac{2T}{mTp} = 2 * \frac{143.8239}{m} * 25$

$= \frac{11.459}{m} = \frac{11459.12}{m} N$

Where; m = pitch diameter /number of teeth

$m = \frac{Dp}{Tp} = \frac{116}{25} = 4.64$

Therefore; $M = 4.64\ say\ 5mm$

$Wt = \frac{11459.12}{5mm} = 2291.824N$

Calculated tangential load on the pinion equal to 2291.824 N, but the assumed tangential load was 2471.5N

We know that length of the pitch come element or slant height of the pitch cone,

$L = \frac{Dp}{2\sin\theta p1} = m * \frac{Tp}{2\sin\theta p1} = \frac{116mm}{2\sin 22}$

$L = 154.8mm$

Since the face width (b) is ¼th of the slant height of the pitch cone, there fore

$B = \frac{L}{4} = \frac{154.8}{4} = 38.7mm$

The proportion of the bevel gear may be taken as follows;

$Addendum, e = 1m = 5mm$

$deddendum, d = 1.2m = 6mm$

$clearance, = 0.25m = 1mm$

$working\ depth, = 2m = 10\ mm$

$thickness\ of\ tooth = 1.5708m = 7.854mm$

For pinion

Outside or addendum cone diameter is known as

$Do = Dp + 2\,a \cos\theta p1 = 116mm + 2 * 5 \cos 22°$
$= 125.27\ mm$

Inside or addendum cone diameter is known as

$Dp = Dp - 2d \cos\theta p1 = 116 - 2 * 6\cos22$
$= 104.87mm$

Addendum angle know as

$(\alpha) = \tan^{-1}\left(\frac{a}{op}\right) = \tan^{-1}\left(\frac{5}{155.6}\right) \qquad \alpha = 2°$

Dedendum angle known as

$(\beta) = \tan^{-1}\left(\frac{d}{op}\right) = \tan^{-1}\left(\frac{6}{155.6}\right) \qquad \beta = 2.2°$

For gear

Outside or addendum con diameter of gear is

$Do = Dg + 2a \cos\theta p2 = 290 + 2 * 5\cos68°$
$= 293.74mm$

Inside or addendum cone diameter of gear is

$Dd = Dg - 2d \cos\theta p2 = 290 - 2 * 6 \cos 68$
$= 285.5\ mm$

Analysis of Strength of Bevel Gear

From the modified Lewis equation for the tangential tooth load is given as follows [5][6]

$Wt = (\delta o * Cv)b.\pi m.y\left(L - \frac{B}{L}\right) = 2471.5N$

$Where;\ cv = 3/3 + V = 0.1413$

$V = \pi DpNp/1000 = pitch\ line\ velocity$

$B = face\ width = 38.7mm$

$M = module = 5mm$

$Y' = lewis\ factor = 0.175 - 0.841/Tep$

$L = slant\ height\ of\ pitch\ cone = 154.8mm$

$L - b/l = bevel\ factor = 0.75$

$\delta o = allowable\ static\ stress$

$V = \pi * 116 * 50/1000 = 18.22\ m/min$

$Cv = 0.1413\ min/m$

$L - b/L = 0.75$

$\delta o = WT/Cvb\pi my^{\wedge\prime}(L - b/l)$

$\delta o = 2471.5/(0.1413 * 38.7 * \pi * 5 * 0.1418 * 0.75)$

$\delta o = 26.67\ N/mm^2$

$Allowable\ static\ stress\ (\delta o) = 26.67\ N/mm^2$

$Taking\ factor\ of\ safety\ 2$

$Factor\ of\ safety = operating\ stress/Allowable\ stati$

$Operating\ stress = 2 * 26.67 = 53.34\ N/mm^2$

Alloy steel which has high tooth strength and low tooth wear, yield strength and tensile strength is 250N/mm^2 and 400N/mm^2 respectively selected. Therefore, design is safe.

xii. Design of shaft (bevel gear) (pinion)

In designing a pinion shaft,

It Is given by $T = \frac{P*60}{(2\pi Np)N} - m$

Np=speed of the pinion in rpm.

$T = \frac{0.75*10^3*60}{2*3.14*50} = 143.239N - m = 143239\text{N} - \text{mm}$

Find the tangential force (Wt) acting at the mean radius (Rm) of the pinion;

$Wt = \frac{T}{Rm}$

$Rm = \frac{\left(L - \frac{b}{2}\right)Dp}{2L} = \left(L - \frac{b}{2}\right)\sin\theta p$

$= (154.8 - 38/2)\sin 22 = 50.87mm$

$Wt = \frac{T}{Rm} = 2815.78N$

Axial and radial forces (i.e Wrh&Wrv) acting on the pinion shaft, as given as bellow

The axial force acting on the pinion shaft

$Wrh = Wr \sin\theta p1 = Wt * \tan\phi - \sin\theta p1 = 2815.78 * \tan 20 * \sin 22$
$= 383.92N$

The radial force acting on the pinion shaft

$Wrv = Wr\cos\theta p1$

$Wrv = 2815.78\tan 20 * \cos 22 = 950.23N$

Find resultants bending moment on the pinion shaft as follows;

The bending moment due to Wrh and Wrv

$M1 = Wrv * overhung - Wrh * Rm$

$Overhung = 100mm$

$M1 = 75493.36N$

The bending moment due to Wt'

$M_2 = Wt * overhang = 281578N$

Therefore, resultant bending moment

$M = \sqrt{M^2{}_1 + M^2{}_2} = 291522.58N$

Since the shaft is subjected to twisting moment (T) And resultant bending moment (M) therefore equivalent twisting moment.

$Te = M^2 + T^2 = 324811.46N - mm$

Now the diameter of the pinion shaft may be obtained by using the torsion equation. We know that

$Te = \frac{\pi}{16} * \tau\,(dp^3)$

$dp^3 = \frac{16 * Te}{\pi * \tau}$

$Dp = 25.41mm$

Where;

Dp = diameter of the pinion shaft, and

T= shear stress for the material of the pinion shaft.

According to Indian standard, material for shaft, alloy steel like nickel-chromium steel is used with the properties of

Ultimate tensile strength $(\delta u) = 560 - 670Mpa'$

Yield strength $(\delta y) = 320mpa$

Therefore, now let us calculate shear stress of the material of the pinion shaft.

$\tau = 0.18\ \delta u = 0.18 * 560 = 100.8mpa$

$therefore;$

$$dp = \sqrt[3]{\frac{16 * Te}{\pi * \tau}} =$$

$dp\ =\ 25.41mm\ = 26mm$

xiii. Design of Gear Shaft (Bevel Gear)

In designing a gear shaft, the following procedure may be adopted;

First of all, find the torque acting on the gear

It is given by

$$T = \frac{p * 60}{2\pi Np} Nm\ = \frac{0.75 * 10^3 * 60}{2\pi 20}$$

$= 358098.622N - mm$

Find the tangential force (Wt) acting at the mean radius (Rm) of the fear. We know that

$$Wt = \frac{T}{Rm} = 2844.048N$$

$$Rm\ = \left(L - \frac{b}{2}\right)\left(\frac{DG}{2L}\right) = \left(L - \frac{b}{2}\right) sin\theta p2$$

$$= \left(154.8 - \left(\frac{38}{2}\right)\right) sin68\ \ = 125.911mm$$

Now find the axial and radial forces (i.eWrh & Wrv) acting on the shaft as given as given bellow

The axial force acting on the gear shaft

$Wrh\ = Wrsin\theta p2\ = Wt * tan\phi * sin\theta p2$

$Wrh\ = 956.773N$

$The\ radial\ force\ acting\ on\ the\ gear\ shaft$

$Wrv\ = Wrcos\theta P2\ = Wt\ \ tan\phi * cos\theta p2$

$Wrv\ = 387.77N$

Find resultant bending moment on the gear shaft as follows;

The bending moment due to Wrh and Wrv is given by

$M1 = Wrv * overhang - Wrh * Rm$

Where;

$Overhang\ = 515mm$

$M1 = 78069.99N$

Bending moment due to Wt,

$M2 = \sqrt{M^12 + M^22}\ = 1458243.892N$

Since the shaft is subjected to twisting moment (T) and resultant bending moment (M),

Therefore equivalent twisting moment

$Te = \sqrt{M^2 + T^2}\ \ = 101569.136N$

Now the diameter of the gear shaft may be obtained by using torsion equation, we know that

$$Te = \left(\frac{\pi}{16}\right) * \tau d G^3$$

$$dG^3 = \frac{Te * 16}{\pi * \tau}$$

$$dG = \sqrt[3]{\frac{Te * 16}{\pi * \tau}}$$

$dG = 42.33mm\ = 43mm$

xiv. Design of knuckle joint

Knuckle joint is type of joint that is basically used for connecting two shaft and at the same time allows the shafts to swing or rotate at angle from the shaft-rotating axis. The knuckle joint is used to transmit rotation at the same time to allow the upper shaft to turner to 90° towards the axis of rotation.

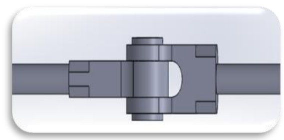


Figure 3 knuckle joint

Knuckle joint specifically the dimensions are given in standard

If cotter and rod are made up of steel and wrought iron then[1][5][6]

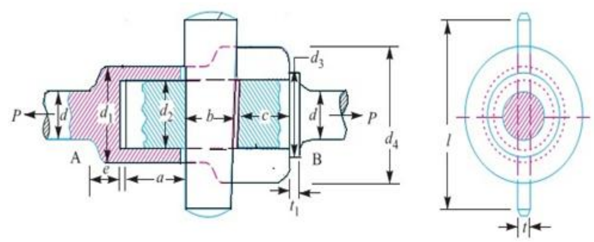


Figure 4 knuckle term

Where a, b, c taken

The dimension of the knuckle joint d =43mm and the shaft materialused is the same for both shafts σ_c , σ_t , τ will be the same

The material used for the knuckle joint is nickel chromium steel with property of

$\sigma_u\ = 560mpa\ \ \ \sigma_y\ =\ 320mpa$

$\tau = 0.8\sigma_t$, where; $\sigma_t = 320$mpa, $\sigma_c = 2\ \sigma_t$

Since the tensile (compressible) stress is assumed to be equal with the yield stress of the material

$\sigma_t = \sigma_y = 320mpa$

$\tau = 0.8(320) = 256mpa$

$\sigma_{cr} = 2(320Mpa)\ \ = 640Mpa$

IV. FAILURE ANALYSIS

A. Failure of the rods in tension (compression)

Since the area of the radius[1][2][5][6]

$$A = \frac{\pi}{4} d^2 = \frac{\pi}{4}(43)^2 = 145.2mm^2$$

Tearing strength of the rods equation with tangential load

$W_T = P = \frac{\pi}{4} \times d^2 \times \sigma_c$

Where W_T = 2471.5N

$\sigma_{com} = \frac{4W_T}{d^2\pi}$

$= \frac{4(2471.5)}{(43)^2\pi} = 1.7\, ^N/_{mm^2}$

Therefore, the design is safe according to tension.

B. Failure of spigot in tension for slot

First, let as calculate the area resistance-tearing spigot across the slot

$A = \frac{\pi}{4}(d_2^2)$ -$d_{2\times}$t

The tearing strength of the spigot across the slot equation this to load (P) which is tangential load W_T

$W_T = \left[\frac{\pi}{4}(d^2)^2 - d_{2*}t\right]\sigma_c$

$\sigma_{com} = \frac{W_T}{\frac{\pi}{4}d_2^2 - d_2 \times t} = \frac{2471.5}{\frac{\pi}{4}(52.03)^2 - 52.03 \times 13.3}$

$= \frac{2471.5}{2126.16 - 693.56}$

$\sigma_{com} = 1.725 mpa$

C. Failure of the rod or cotter in crushing

As we know the area that resists crushing of a rod or cotter

$A = d_2 \times t$

The crushing strength will be equal with tangential load on the gear (W_T) = P

$P = \frac{P}{d_2 \times t} = \frac{2471.5}{52.03 \times 13.33} = 3.56 mpa$

D. Failure of the socket in tension across the slot

Area of resisting area of the socket a cross the slot

$P = \{\frac{\pi}{4}[(d_2)^2 - (d_2)^2] - (d_1 - d_2)\, t\}\sigma_t$

$\sigma_t = \frac{P}{\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t}$

$\sigma_t = \frac{2471.5}{\left[\frac{\pi}{4}(75.25^2 - 52.03^2) - (75.25 - 52.03)13.33\right]}$

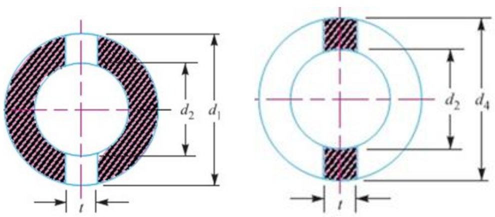


Figure 5 socket tension cross section area (left) and cotter section in crushing (right)

$\sigma_t = 1.228 mpa$ This is very lower than the material working stress and safe. [2][5][6]

E. Failure of cotter in shear

Considering the failure of the cotter in shear, since the cotter is in double shear, therefor area of the cotter

A= 2b × t.

Shearing strength of the cotter = P

P = 2b × t × τ

$\tau = \frac{P}{2b \times t} = \frac{2471.5}{2(55.9) \times 13.33}$ = ***1.658mpa***

Therefore, the design is safe according shear.

F. Failure of the socket cotter in crushing

Crushing the area that cursing of socket collar

= ($d_4 - d_2$) t and crushing strength = ($d_4 - d_2$) t × σ_c = P

$\sigma_{cr} = \frac{P}{(d_1 - d_2)t} = \frac{2471.5}{(103.2 - 52.03)13.33}$

$\sigma_{rc} = 3.623 Mpa$ *therefore the disign is safe*

Figure4.0.10 Socket cotter section in crushing

G. Failure of socket end in shearing

Since the socket in double, shear the area resisting shearing of socket collar

$A = 2(d_4 - d_2)c$

Shearing strength of socket collar = P

$P = 2(d_4 - d_2)c * \tau$

$\tau = \frac{p}{2(d_4 - d_2)c} = \frac{2471.5}{2(103.2 - 52.03)32.25}$ = 0.75mpa

H. Failure of rod end in shear

The area of shear is double therefore the area resisting shear of the rod end   A = 2a × d_2

The shear strength of the load end (P)

P =2a × $d_2 \times \tau$

$\tau = \frac{P}{2a \times d_2} = \frac{2471.5}{2(32.25) \times 52.03}$ = 0.736mpa

I. Failure of spigot collar in crushing

Crushing of the collage A = $\frac{\pi}{4}[d_3^2 - d_2^2]$

∴ Crushing strength of the collage = $\frac{\pi}{4}[d_3^2 - d_2^2]\, \sigma_c$ equating this to load (P), we have

P = $\frac{\pi}{4}(d_3^2 - d_2^2)\, \sigma_c$

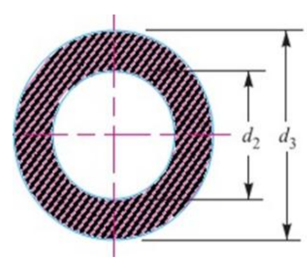


Figure 6 spigot collar in crushing

$\sigma_c = \frac{P \times 4}{\pi(d_3^2 - d_2^2)} = \frac{2471.5 \times 4}{\pi(645^2 - 52.03^2)}$ = 2.1655mpa

J. Failure of the spigot collag in shearing

We know that area that resist shearing of the collage $A = \pi\, d_2 \times t_1$

And shearing strength of the collage $= \pi d_2 \times t_1 \times \tau$ equation this to load (P) we have

$P = \pi\, d_2 \times t_1 \times \tau$

$\tau = \frac{P}{\pi d_2 \times t_1} = \frac{2471.5}{\pi \times 52.03 \times 19.35}$ = 0.7814mpa

K. Failure of cotter in bending

The maximum bending moment occurs at the center of the cotter and is given by

$M_{max} = \frac{P}{2}\left(\frac{1}{3} \times \frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2} \times \frac{d_2}{4}$

We now that section modulus of cotter $Z = t \times {}^{b^2}/_{6}$

∴ Bending stress in the cotter,

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times {}^{b^2}/_{6}} = \frac{P(d_4 + 0.5 d_2)}{2t \times b^2}$$

$\sigma_b = \frac{2471.5(103.2 + 0.5 \times 52.03)}{2 \times 13.55 \times 55.9^2}$ = 3.7712mpa

xv. Design of turntable

To design this table, we need to know the dimension of the tier

Figure 7 turn table

From our information over all dimension of the given is as follows

Outer diameter = 584.2mm

Rim width = 330.2mm

Bolt circle diameter = 2.75mm

Bolt number =8

Back up diameter =12.4 – 12.6 inch

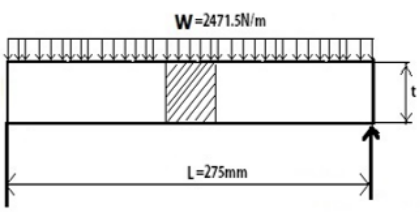


Figure 8 force distribution on turn table

The turner table must be fit with the bolt circle of rim. The tire load and pressing load directly transmitted to the turntable. And the load is uniformly distributed over the circular plate.

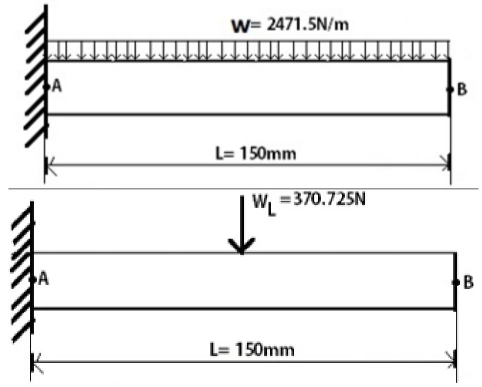


Figure 9 support/assumption

1. load apply on the plate is uniformly distributed load.

First, let us find reaction force R_e

$\sum F_Y = 0$ $\quad -370.725 + R_e = 0$

$R_e = 370.725$

Let us find bending moment (M) and shear force at any point

We cut the beam at a point C between A and B

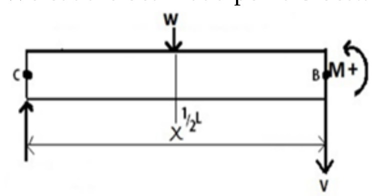


Figure 10 Beams between A and B

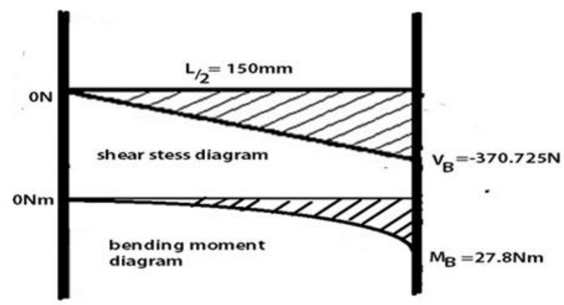


Figure 11 shear and bending moment diagram

$\sum F_Y = 0$ $\qquad \sum M_C = 0$

$-Wx - v = 0$ $\qquad Wx\,(\frac{1}{2}x) + M = 0$

V = - wx $\qquad$ M = $\frac{\omega}{2}$

$X^2 = 27.8$ Nm

V = - 370.725N

The design of turntable uses ASTMGRAND 450 carbon steel that has mechanical properties

$\sigma_{u_1 t}$ =550mpa $\qquad$ G = 77Gpa

$\sigma_{y_1 d}$ =450mpa $\qquad$ E = 200Gmpa

Since our maximum deflection of the plate due to the applied moment

$\frac{1}{\rho} = \frac{M_{max}}{EI}$

Where I → moment of inertia

I → will be taken for rectangular cross section (assumption)

$I = \frac{BH^3}{12}$

B → equivalent to the radius of plate H→ thickness of the plat

B = 150mm = 0.15m H = 10mm = 0.01m

$I = \frac{0.15 \times (0.01)^3}{12} = 1.25 \times 10^{-8} m^4 \quad \frac{1}{\rho} = \frac{M_{max}}{EI}$

$$\frac{1}{\rho} = \frac{27.8Nm}{200 \times 10^6 \,{}^{N}/_{m^2} \times 1.25 \times 10^{-8}}$$

$\frac{1}{\rho} = 11.12\frac{1}{m} \quad \rho = 0.089\text{m} \approx 0.00089\text{mm}$

This deflection indicates that the bending moment on the negligible (or) the plate is safe.

xvi. *Design of welding*

Welding design is many concerned with the strength of the welding and that led us to determine the size of welding.

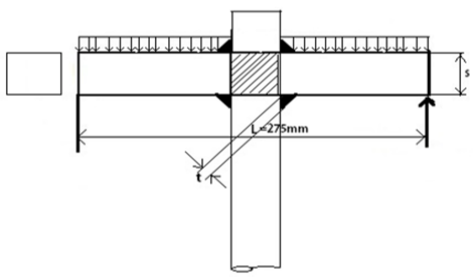


Figure 12 force distribution over the welding

Conceder half of the plate section

W = 2471.5N⁄m

e = 0.075m

Since the bar is welded in both ends to the circular plate then the load is supported by the shaft that are welded at the both

The load is assumed to be distributed uniformly we can apply the load by multiply by the length of the plate form center

S = size of the weld

t= throat thickness

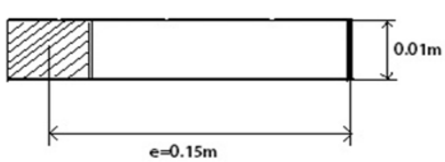


Figure 13 load on the plate length

The joint as shown in the figure is subjected to direct shear and the bending stress. As we know the throat are for rectangle fillet weld but the welding is in the circular perimeter of the shaft and plate. [1][2][5][6]

∴ the Area will be considered as

$A = \pi D_t$ where t = 0.707.s

$= \pi \times 43mm \times tmm$

$= \pi \times 43mm \times 0.707s$ = 95.51mm.s

Now let as find the bending moment as we know

M = P × perpendicular distance = 2471 × 75

= 185.36KN.mm

Polar moment of inertia

$$\tau = \frac{t(b+l)^3}{6}$$

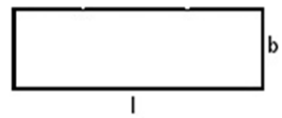


Figure 14 rectangular section

Section modulus for circular section (Z)

$Z = \frac{\pi t D}{4} = \frac{\pi(0.707.S)\times 43^2}{4}$ =1026.7× S mm^3

Then now let as determine bending stress (σ_b)

$$\sigma_b = \frac{M}{Z} = \frac{185.36\,KNmm}{1026.26.Smm3} \quad \sigma_b = 180.62 \,{}^{N}/_{mm^2}$$

Know let as determine direct shear stress

$$\tau = \frac{P}{A} = \frac{2471.5N}{95.5mm^2} = 25.88\frac{N}{mm^2}$$

We know the maximum shear stress (τ_{max})

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b^2) + 4\tau^2}$$

τ_{max} Is selected from the date table. The frame is welded with electrode specification E60.

Since our plate is ASTMGRAND 450 CARBON steel.

Maximum shear from the design data, $\tau_{max} = 70mpa$

$$\tau_{max} = \frac{1}{2}\sqrt{(\frac{180.62}{S})^2(\frac{NN}{mm^2})^2 + 4(\frac{25.88}{S}\frac{N}{mm^2})^2}$$

$$\frac{93.945}{70 \,{}^{N}/_{mm^2}} \,{}^{N}/_{mm} \approx 2\,mm$$

xvii. *Design of flat plate support*

Force analysis over the plate, Let points ‘C’ and ‘D’ taking point ‘A’ as a reference

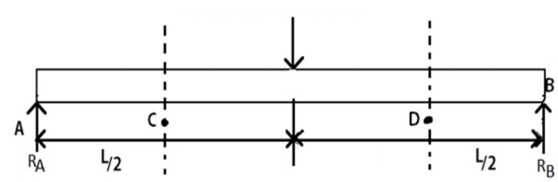


Figure 15 left of center (A-C)

$\sum M = 0 \quad -2347.5N \times \frac{L}{2} + R_B \times L = 0$

$R_B = \frac{P}{2} = 1235.75N$

Taking point ‘B’ as a reference

$R_A = -\frac{P}{2} = -1235.75N$

Taking left of center (A-C)

Where $X \to 0 < X < \frac{1}{2}(OR)540mm$

Take x =0

$V_1 = \frac{1}{2} W = R_A \quad = \frac{1}{2} \times 2471.5N =$ 1235.75N

$\sum M = 0 \quad \rightarrow M = \frac{PX}{2}$

Where x=0. M=0

Taking right of center (D - B)

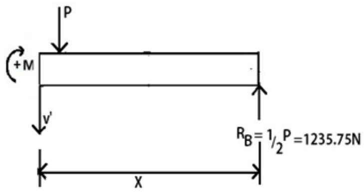


Figure 16 right of centre (D - B)

$\rightarrow \quad 0 < X < 540mm$ Take x = 0

$R_B = \frac{P}{2} = 1235.75N$

$V_2 = -\frac{P}{2} = -1235.75N$

$\sum M = 0 \quad \rightarrow M = \frac{PX}{2} \rightarrow X = 0$

M = 0

Selection from A to B

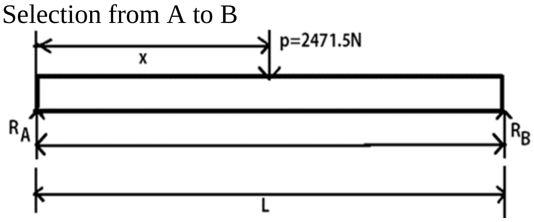


Figure 17 from A to B

Taking point A as reference

V = $^P/_2$ $\quad V = 2471 - 1235.75$

$V' = -^P/_2 \quad = 1235.75N$

$V' = -2471.5 + 1235.75$ = - 1235.75N

M = $\frac{P(L-X)}{2} = \frac{PL}{4} = \frac{2471.5 \times 1080}{4}$

$M_{max} = 667.305N.m$

$M' = 0 \text{ and } X = L$

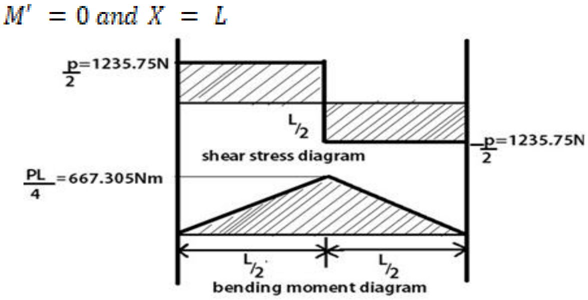


Figure 18 shear and bending diagram

The dimensional of the flat plate support is rectangular shape (1080*480) with thickness of 10mm.

The material selected for this support is $ASTM = A709 \text{ grand } 690(\text{quinchied and tempered})$

$\sigma_{ultimate} = 760Mpa \quad \sigma_{yield} = 690Mpa$

$G = 77GPa \quad E = 200GMpa$

The maximum deflection Y=maximum deflection

$\frac{1}{y} = \frac{M_{max}}{EI}$

I=moment inertia for rectangular shape

$I = \frac{BH^3}{12} = \frac{0.48m * 0.01m^3}{12}$

$I = 4 * 10^{-8} m^4$

$\frac{1}{y} = \frac{667.305Nm}{200 * 10^6 \left(\frac{N}{m^2}\right) * 4 * 10^{-8} m^4} = 83.41 \frac{1}{m}$

$Y = \frac{1}{83.41} = 0.012m$

The maximum bending moment is 667.305Nm on

$\sigma_{max} = 667.305Nm$

It is less than the allowable stress and the frame is safe.

xviii. *Design of welding for plate*

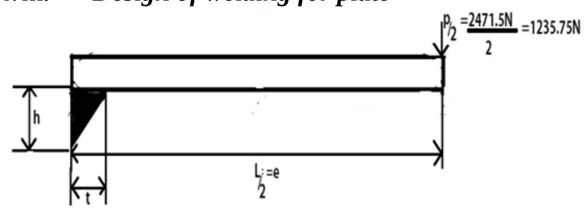


Figure 19 free body diagram of welding

Where S=size of the weld t = throat thickness

$A = 2t * l \quad 2 * 0.707.s * 40 = 56.56.smm^2$

$A = 2t * l = 2 * 0.070.s * 40 = 56.56.smm^2$

Now let as find the bending moment knows as

$M = P * perpendicular\ distance$

$= P * e \quad = 1235.75 * 240 = 296580Nmm$

Know find polar moment of inertia (j)

$j = \frac{t(b+l)^3}{6}$

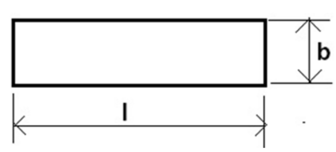


Figure 20 Cross section of plate

Section modulus for rectangular section (z)

$Z = t\left(b.l + \left(\frac{b^2}{3}\right)\right)$

Then now let us determine bending stress(σ_b)

$\sigma_b = \frac{M}{z}$

To determine bending stress, we need to determine section modulus(z)

$z = t\left(b * l + \left(\frac{b^2}{2}\right)\right) = 0.707.s(10 * 140) + (\frac{10^2}{2}))$

$318.15smm^3$

$$\sigma_b = \frac{M}{z} = \frac{296580}{318.15 * 5} = \frac{932.2}{5} N/_{mm^2}$$

Now let as determine direct shape stress

$$\tau = \frac{p}{A} = \frac{1235.75N}{56.56s} = \tau = \frac{21.84}{s} N/_{mm^2}$$

We know maximum shape stress(τ_{max})

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_b{}^2 + 4z^2}$$

τ_{max} is selected, the frames are welded with electrode specification E60.since our plate is ASTM-A790 grand 690(quenched and tempered)we can find maximum shear

$$T_{max} = 124Mpa$$

$$\tau_{max} = \frac{1}{2}\sqrt{(\frac{932.2}{5})^2 + 4(\frac{21.84}{5})^2}$$

$$\frac{1}{2}\sqrt{\frac{868996.84}{s^2} + \frac{1907.9424}{s^2}} = \frac{1}{2}\sqrt{\frac{870904.7814}{s^2}}$$

$$\tau_{max} = \frac{466.6}{s} N/_{mm^2}$$

$$s = \frac{466.6}{\tau_{max}} = \frac{466.6}{124} = 3.76mm$$

$$t = 0.707.s = 2.66mm$$

Using electrode type (covered electro) with max shape strength 70Mpa with welding thickness 10mm. The design of welding will be safe with this electrode

.

xix. Dynamic equivalent thrust load (pa)

To calculate a dynamic equivalent thrust load pa the, for thrust ball, thrust spherical loads introduces complex load calculations that must be carefully considered. If the radial load (fr) is zero, the dynamic equivalent thrust lead will be equal to the applied thrust load (fa). For thrust ball bearing, the dynamic equivalent thrust load is determined by

$$p_a = xf_r + yf_a$$

For standard TVL and DTVL bearing having a 50^0 contact angle, x=0.76 and y=1.00 minimum$\frac{f_a}{f_r}$ Ratio to maintain proper operation for these applications is 1.56.

$P_a = 0.76 \times 0 + 1 \times 2471.5N$ =2471.5N

xx. Bearing Ratings- dynamic and static load

For combined loading, the L_{10} life has been calculated as follows for bearing under radial or, where the dynamic equivalent radial load, P_r has been determined and the dynamic load rating is based on one million cycles;

$L_{10} = (\frac{C}{P_r})^e(\frac{1*10^6}{60n})$ hours

$L_{10} = (\frac{C}{P_r})^e(1 * 10^6)$ revolution

For thrust bearing the above equation change to the following

$L_{10} = (\frac{C_a}{P_a})^e(1 * 10^6)$ Revolutions

$L_{10} = (\frac{C_a}{P_a})^e(\frac{1*10^6}{60n})$Hours

E=3 for ball bearing

$= \frac{10}{3}$ for tapered, cylindrical and spherical roller bearing tapered roller bearings typically used a dynamic load rating based on 90million cycles, denoted as C_{90}, changing the equation as follows and for tapered thrust bearing number 51109, the basic dynamic capacity

$C_a = 28035N$

$$L_{10} = (\frac{28035}{2471.5})^{\frac{10}{3}} * (90 * 10^6)$$

xxi. Design of I-Beam

A beam of alloy steel that is with E=200KN/mm² and an I-section 80mm*202mm*10mm and 1.6m long is used as Hold the pneumatic cylinder and lever and it is fixed on the body of tire changer with bolt.

Using Euler formula, crippling load or blocking load of the I-beam

Input data

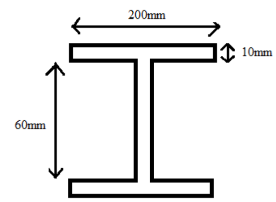
D=80mm

B=200mm

T=10mm

L=1600mm

E=200kN/mm²



M_{max} @ x = $\frac{1}{2} \rightarrow M_{max} = \frac{PL}{4}$

Where; B = width of I-beam T = thickness of the flange L = length of beam E = young's modulus

According to Euler's theory, the crippling or buckling load (Wcr) under various and conditions is represented by a general equation.

$$Wcr = \left(\frac{c\pi^2 EI}{l^2}\right) = \frac{c\pi^2 EAK^2}{L^2} = \frac{(c\pi^2 EA)}{(L/k)^2}$$

E= young's modulus, A =area of cross section

K=least radius of gyration of the cross section,

L=length of the column, C=constant fixing coefficient.

Slenderness ratio

In Euler's formula, the ratio l/k is known as slenderness ratio, Assuming the slenderness ratio l/k, that the failure of the column occurs only due to bending, the effect of direct stress (i.e w/A) bending negligible. A little consideration will show that the crippling stress will be high, when the slenderness ratio Is small. The moment of inertia of I-section about x-xI

$$Ixx = \frac{BD^3}{12} - \frac{bd^3}{12} = \frac{200 * 80^3}{12} - \frac{(200-10)(60-20)^3}{12}$$

$Ixx = 7520000\,mm^4$ $=7.52 * 10^6 mm^4$

And moment of inertia of the I- section about y-y

$$Iyy = \frac{2(t.b^3)}{12} + \frac{dt^3}{12} = 2\left[\frac{10 * 200^3}{12}\right] + \frac{40 * 10^3}{12}$$

$=13.34 * 10^6 mm^4$

Since Ixx is less than Iyy, therefore the section will tend to buckle about x-x axis. Thus, we shall take I as Ixx=$7.52 * 10^6$ mm^4

Since the column is fixed at one end and free at other end, therefore equivalent length

$L = l/2 = 1600/2 = 800mm$

We know that the crippling load

$$Wcr = \frac{\pi^2 EI}{L^2} = \frac{(\pi^2 * 200 * 10^3 * 7.5 * 10^6)}{800^2}$$

$$= 23193570.34N$$

xxii. *Design of bolt for up – stand (I-beam)*

Bolts are designed on the basis of direct stress with a large factor of safety in order to account for the indeterminate stress. The relation may find the initial tension in bolt, based on experiment.

$$Stress\ area = \left(\frac{\pi}{4}\right)\left(\frac{dp + dc}{2}\right)^2$$

$$= \left(\frac{\pi}{4}\right)\frac{(9.026 + 9.858)^2}{2}$$

$Stress\ area = 70mm^2$

$Pi = 2840(10)N = 28400N$

$Safe\ tensile\ load$

$ps = stress\ area * \delta t$

$ps = 70mm^2 * 250N/mm^2$

$ps = 17{,}500N$

for single bolt and for 8 bolts the load is distributed equally to these bolts. Ps on single bolt will be (17500N/8) = 2187.5N

The average threads shearing stress for the threads (τs) is obtained by using the relation.

$$\tau s = \frac{p}{\pi * dc * b * n}$$

$$\tau s = \frac{17500}{\pi * 9.888 * 8 * 2.147}$$

$\tau s = 32.89mpa$

This shear strength applied on single bolt is much less than the material properties. Therefore, the bolts are safe with respect to shear. Check for bending

$$\delta b = \frac{(x * E)}{(2l)}$$

L = distance of shank of the bolt

E = young's modulus of the bolt material

X= difference in height between the extreme corners of the nut or head.

$$\delta b = \frac{(30 * 200)\left(\frac{N}{mm^2}\right)}{2 * 50mm}$$

δb=60N/mm² = 60Mpa

Bending stress is for total no of bolts 8, will be δb/2 = 60/8 =7.5mpa and the bolt is safe either in bending or shear strength

M10 – ASTM – 36Sstud bolt = for up stand (I - beam)

xxiii. *Design of Arm Support and Lever*

The armis welded with the up stand (or) I- beam. Therefore, the arm forms cantilever beam support type. The force exerted on the arm will be divide in to two. The total load the at are exerted in the pin

$Ftotal = 2524.6N$

Therefore, the force exerted on the single arm support will be;

$$Farm = \frac{Ftotal}{2} = 1262.3$$

Assumption

The arm support is fixed in one end and free on the other side; it can be considered as cantilever beam

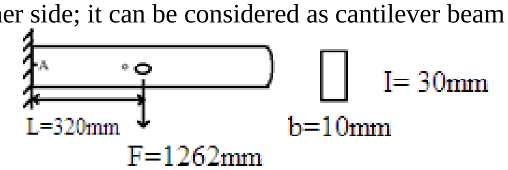


Figure 21 Free body diagram of arm

Therefore the maximum moment on the arm will be;

M= $-f * e = 1262 * 0.32$

$= -403.84Nm\ clockwise$

The material used in the arm support is ASTM grand 450 carbon steel, same material used for base support (box)

$\delta ulti = 550Mpa \quad \delta yield = 450Mpa$

$E = 200Gpa \quad G = 77Gpa$

Since the cross-section area is rectangular shape;

$$I = \frac{bd3}{12}\ and\ Y = \frac{d}{2}$$

$I = (0.01 * 0.04^3)/12 = 5.33 * 10^{-8}m^4$

$Y = 40/2 = 0.04/2 = 0.02m$

Now let as determine the bending stress on the arm support

$$\delta b = \frac{Mby}{I} = \frac{403.84 * 0.02m}{5.33 * 10^{-8}m^4} = 151.535Mpa$$

The material used for arm support is with maximum bending stress is much less than the yield stresses the design is safe.

xxiv. *Design of lever*

Length of the lever portion from the cylinder rod pin to the fixed hinged is L1. whereas the length of the lever portion from the load rod in to the fixed hinged is L2. To determine the cylinder force Fcyl required to drive a load force Fload, force Fload that drive by cylinder force Fcyl and length L2 it is from load road pin to the fixed hinge.

Counter clockwise moment = clockwise moment

$$Fcyl\ (L_1\ cos\theta) = Fload\ (L_2\ cos\theta)$$

From our design of pneumatic system cylinder force is we considered not excessed 1000N. ➔

$Fcyl = 1000N$

Since for pressing our tire to separate tire and rim we need up to 1000N. and our lever length from cylinder to hinge pin[L_1] and from hinge pin and load (L_2)is

$L_1 = 340mm \qquad L_2 = 223mm$

$Therefore\ fcyl(L1) = Fload(L_2)$

$Fload = Fcyl(L_1/L_2) = 1000(340/223)$

$Fload = 1524.6N$

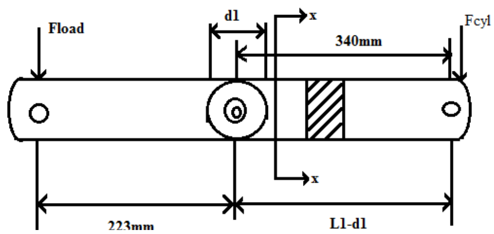


Figure 22 Free body diagram of lever

Lever consists of two aspects

F by means of an effort 'p', but our length of lever we decide by considering geometry of our tire changer machine and the force that required to separate tire and rim. That is mentioned on the previous calculation.

L_1 =340mm and L_2 =223mm

Since, the force calculated on previous page, next reaction force at the fulcrum pin, since it is the sum of vertical forces acting on the lever.

$\sum Fcyl = 0 = Rb - Fcyl - Fload = 0$

$Rb = Fcyl + Fload = 1000 + 1524.6$

$Rb = 2524.6N$

The cross section of the lever is subjected to bending moment. The cross-section at which the bending moment is maximum can be determined by constructing bending moment diagram. In the next figure 28 the bending moment is maximum at section xx as it is given by[1][2][5][6]

$Therefore;\ Mb = p(L_2 - d_1)$

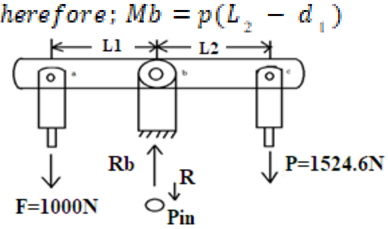


Figure 23 Cross section of lever

In order to find maximum bending moment first find the diameter of pin, so first we need design a pin.

xxv. Design of Pin

The area of shear is one therefore the cross section are of pin under shearing will be

$As = \pi/4Dp^2$

$Shear\ strength\ of\ the\ pin = \pi/4Dp^2\tau$

Now equating this to the load (W) acting on lever, we will find or determine the diameter of the pin.

Now let us determine the value of maximum bending moment.

$M = \left(\frac{w}{2}\right) * t - \left(\frac{w}{2}\right) * t$

Therefore M=D ➔ there is not bending moment for the pin only the shear stress is applied to the pin

$A = \left(\frac{\pi}{4}\right) D^2 = \left(\frac{\pi}{4}\right) 20^2 = 314.15mm^2$

$\tau = \frac{F}{A} = \frac{2524.6}{314.15} = 8.036MPA$

Taking factor of safety (fos=3)

$Fos = maximum\ stress/allowable\ working\ stress$

$Maximum\ stress = fos * working\ stress$

$= 3 * 8.036 = 24.1N/mm^2$

$\delta max = 24.1mpa$

Using Indian standard designation of steel according to Is; 1570(part)-1978 (reaffirmed 1993) (Fe290) choose and has properties of;

$Tensile\ strength = 290mpa$,

$Yield\ strength = 170mpa$

The maximum stress is much less than the yield stress Fe290 with diameter of 20mm used. Thus, effective diameter of the pin, so we can find the maximum bending moment of lever.

$Mb = P(l_2 - d_1)$

$= 1000(340 - 20) = 320Nm$

The cross section of the lever can be rectangular

$F = \frac{bd^3}{12} \quad and \quad y = \frac{d}{2}$

$I = 563 * \frac{40^3}{12} = 0.563 * \frac{0.04^3}{12}$

$I = 3 * 10^{-6} m^4$

Using the above-mentioned proportions, the dimensions of the cross section of the lever can be determined by

$\delta b = \frac{MbY}{I} \quad where;\ Y = \frac{d}{2}$

$Y = 20 = 0.02m$

$\delta b = 320N - M * \frac{0.02m}{3} * 10^{-6} m^4$

$\delta b = 2.1314MPa$

xxvi. Design of Pneumatic System

In ACBSE pneumatic system normally work with low pressure (p≤ 1Mpa) and a temperature close to $+20^0$ c (with the deviation from a perfect gas can be neglected for practical calculation the air can be treated as a perfect gas, with the following data [8]

Gas constant $R = 287\ J/(KG.K)$

Molar mass $M = 29\ kg/kmol$

Specific heat at Constant pressure $cp = 1005\ J/(KG.K)$

Specific heat at Constant volume $cv = 718\ J/kg.k$

Theory

$F = PA$

Where: $p -$ *is the pressure in* N/mm^2
$A -$ *is the area that the pressure acts on in* m^2.
$F = PA$ – On the full area of piston.
$F = P(A - a)$ on the rod-side.

Speed
The speed of the piston and rod depends up on the flow rate of fluid. The volume per second entering the cylinder must be the change in volume per second inside. It follows then that.
$Q(m^3/s) = area * distance\ moved\ per\ second$
$Q(m^3/s) = A * velocity\ (full\ side)$
$Q(m^3s) = (A - a) * velocity\ (rod\ side)$
In the case of air cylinder, Q is volume of compressed air and this changes with pressure so any variation in pressure will cade a variation in the velocity.
Power- mechanical power = $P = F.V\ watts$
$Max - pressure - 10bar$
$Displacement\ (flow\ rate) = 0.58L/min = 3.48m^3/s$

Pneumatic Cylinder
This type of pneumatic cylinder is used in this design not excessed 20 kg. In this design double acting pneumatic cylinder, maximum pressure is not excessed 10bar used, and the force we need apply on the tire is not excessed 1000N.
=>operating pressure
$(p) = 10bar = 10 * 10\text{^}5N/m^2$
Max force that we need apply on the tire [7][8]
$(F) = 1000N$
Using the basic theory of force of pneumatic jack double acting cylinder force in forward stroke is

$$F = \left(\frac{\pi}{4}\right) D^2 * p$$

$$1000 = \left(\frac{\pi}{4}\right) D^2 * \frac{1000000N}{m^2}$$

$$D^2 = \frac{1000 * 4}{\pi} * 1000000 \quad = \frac{4000}{3140000}$$

$$D = \sqrt{0.00127388} = 0.0356m = 35.6mm$$

The piston rod in forward stroke diameter is find from force basic theory (D=36mm) now consider our piston rod diameter in return stroke less by 16mm, so our piston rod on return stroke diameter is 20mm (d=20mm)

Design of the cylinder
The cylinder selected with wall thickness (t) less than 1/10 of the diameter of shell vessels having the circumferential of hoop stresses are induced by the fluid pressure. In case of cylinder of ductile material, the value of circumferential stress (δy)
$\delta t1 = 0.8\delta y$[7][8]
Selected cylindrical pressure vessel considered to be outer diameter is 40mm and it is subjected to an internal pressure of 10bar = 1mpa. If the thickness of the cylinder is 3mm.
Calculating the hoop stress
$\delta t1 = (p * d)/2t = (1 * 40)/(2 * 3)$
$= 6.61mpa$
Therefore,
$D = outer\ diameter = 40mm$
$T = thickness\ of\ cylinder = 3mm$
$L = length\ of\ cylinder = 300mm$
$\delta t1 = circumferential\ or\ hoop\ stress\ of\ the\ material$
.

Air consumption
$Free\ air\ consumption = (piston\ area * operating\ pressure + 1.013) * stroke$
Let D = diameter of cylinder in mm
d= piston rod L=stroke in mm, P= air pressure in bar
Free air consumption for forward stroke;
C = $\left(\frac{\pi}{4}\right) * D^2 * (p + 1.013) * \left(\frac{L}{1000}\right)$
C= $\left(\frac{\pi}{4}\right) * 3.6^2 * (10 + 1.013) * 30$ $C = 3.36ltrs$
Free air consumption for return stroke
C = $\left(\frac{\pi}{4}\right) * (D - d^2) * (p + 1.013) * \left(\frac{L}{1000}\right)$
C= $\left(\frac{\pi}{4}\right) * (3.6 - 2^2) + (10 + 1.013) * \frac{L}{1000}$
$C = 0.32litr$
For one complete cycle air consumption will be
$(3.36 + 0.32 = 3.68\ liters$
$Q(m^3/s) = A * v(full\ side)$
$Q(m^3/s) = (A - a) * v(rod\ side)$
Speed for forward stroke $Q = A * v$
$V = Q/A = 3.45mm^3/s / \left(\frac{\pi}{4}\right)^2$
$= 3.48/(\pi/4 * 0.036^2 m^2)3.41m/s$

Valve selection/Pneumatic valve sizing
The formula will give the C_v (valve flow) required for operating a given air cylinder at a specific time period.

$$Cv = \frac{area * stroke * A * Cf}{time * 29}$$

Where Area = $\pi * r^2$
Stroke = cylinder travel (in) A = pressure drop constant
C_f = compression factor Time = in second
Inlet pressure (psi) = 120
A = $\pi * (\frac{1.42}{2})^2$ = 1.5836 in^2 , Stroke = 11.81 in^2
Use A constant at 5psi ▲P for most application
A = 0.039, C_f = 9.2 Time = 2 s
Therefore; $C_v = \frac{1.5836 * 11.81 * 0.039 * 9.2}{2 * 29}$ = 0.175= 0.18
Using this value of C_v pneumatic system 4-way 3 position LTV type valve selected.

Selection of pressing dye
The geometry of pressing die as shown in the figure 24 left.

Figure 24 pressing dye(left) and pulling dye(right)

xxvii. Design of pulling die

A pulling die shown figure 29 right. It is made of plain carbon steel with a yield strength of 380Mpa in tension. Know let us find the load capacity of the pulling die, for a factor of safety

To find load capacity for trapezoidal section

$$R = r_i + \frac{h(bi+2bo)}{3(bi+bo)} = \frac{70\ (60+(2*20))}{3\ (60+20)} = 54.16\text{mm}$$

$$r_n = \frac{\frac{1}{2}(bi+bo)h}{\frac{(bi*ro-bo*ri)}{h}\ln\left(\frac{ro}{ri}\right)-(bi-bo)}$$

$$\frac{\frac{1}{2}(20+60)70}{\left(\frac{(60*100-20*25)}{70}\right)\ln\left(\frac{100}{25}\right)-(60-20)} = 40.22mm$$

$e = R - r_n = 54.16 - 40.6 = 13.56$ mm

$h_i = r_n - r_i = 54.16 - 25 = 29.16$ mm

$A = ½ (b_i + b_o) h = ½ (60 + 20)70 = 2800mm^2$

Therefor moment, $M_b = P * R = P * 54.16 = 54.16$ P N-mm

Bending stress (occurring at inner surface)

$\sigma_{b,max} = \frac{Mb*hi}{A*e*ri} = \frac{54.16\ P*29.16}{2800*13.56*25} = 1.66 * 10^{-3}$ P N/mm^2

Direct stress,

$\sigma_b = \frac{P}{A} = \frac{P}{2800} = 3.57 * 10^{-4}$ P N/mm^2

Total stress,

$(1.66 * 10^{-3} + 3.57 * 10^{-4})$ P N/mm^2

$= 2.017 * 10^{-3}$ P N/mm^2

Since $\sigma all = \frac{380}{3} = 126.6$ N/mm^2

Equating the equation $\sigma_{b,max} = \sigma all$

$2.017 * 10^{-3}$ P N/mm^2 = 126.6 N/mm^2

P = 62964.79 N = 62.96 KN, It is safe to work.

V. CONCLUSION

This machine is designed to mount and demount tire from rim. In addition to this shaft design, design of set of spur gear, pneumatic system analysis has been done. In shaft and pneumatic cylinder and ram design diameter is determine and appropriate materials were selected. In the box and turn table design for tire changer suitable material for the operation selected and the dimension identified.

REFERENCE

[1] S.Kurmi & J. K. Gupta, A Textbook of Machine Design

[2] Tom Duke Contributing Editor Tire Mounting and Demaunting Tire Review, May 2012.

[3] Pornima T. Godbole, C. C. Handa , S. P. Bhorkar " Machines For Tire Removing And Fitting On Rim Of Wheel." IJSRD- International Journal for Scientific Research & Development, Vol. 4, Issue 05, 2016

[4] Dr. R.K. Bansal ,"Applied Mechanics And Strength of Materials"

[5] Ferdinand P.beer & John T.Dewolf , A Textbook of Mechanics of Material

[6] G.K Vijayaraghavan & S. Vishnupriyan, A Textbook of Design of Machine Element

[7] M.J. Neale Contributing Editor Lubrication and Reliability Handbook, Sep 2014.

[8] Monkmann J.H, on pneumatic system, 6th international symposium on industrial machinery, Tokyo, 1985.

[9] http://www.pneumaticcylinder.com

[10] http://www.tyerchangerautomatic.com

[11] http://www.hunterengineering.com

[12] http://www.ebay.co.uk

[13] http://www.Timer.com

[14] http://www.humberg.com