Design of Tire Changer

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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 as length, thickness, 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.R = T4/T3

\[ T4 = T3 * m = 15 * 3 = 45 \]

**Third stage gear**

Third stage driver gear number of teeth is 15.

V. R3 = 3:1 = T6/T5

**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 \Rightarrow D2=m*T2 = 3*66=198mm**

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

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

Outside diameter = Do = D2+2m = 198+6 = 204mm

Root diameter = Dr = D2-2m = 198-6 = 192mm

**Gear - 3 \Rightarrow M = D3/T3 = D3*T3*m =15*3=45mm**

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

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

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

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

**Gear -4 \Rightarrow 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 = 3.6 mm

Outside diameter = Do = D4+2m = 180+6 = 186mm

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

Centre distance between the shafts is

\[ = D4+D3/2 = 45+180/2 = 112.5mm \]

**Gear 5 \Rightarrow M = D5/T5 = D5*m*T5 =15*3=45mm**

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

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

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

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

**Gear 6 \Rightarrow M = D6/T6 , D6=m*T6 =45*3=135mm**

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

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

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

\[ Ft(1,2) = P/V1*2 \]

\[ V1-2 = \pi*D1N1/60 = 3.14*66*1800/60 = 6.22m/s \]

\[ Ft(1,2) = P/V1*2 = 0.75*1000/6.22 = 120.57N \]

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

\[ V3-4 = \pi*D3N3/60 = 3.14*45*600/60 = 1.4137m/s \]

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

\[ Ft(5-6) = P/v5-6 \Rightarrow V5-6 = \pi*D5N5/60 = 3.14*45*150/60 \]

\[ = 0.3534m/s \]

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

**iii. Stress Analysis (Spur Gear)**

\[ Ft = 2122.24N \]

Since, \( Pd = T5/D5 = \pi/Pc5 = 0.333mm \)

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

Allowable stress \( \sigma \) (for only static load)

Where \( Y \) is Lewis factor, since \( \phi \) is 20° full depth involutes

\[ Y = 0.154 - 0.912/T_5 \]

\[ Y = 0.154 - 0.912/15 = 0.0932 \]

\[ \delta_{a} = \frac{Ft}{b} \]

\[ \frac{Kv * Kf * Ko * Km}{J} \]

Where;

\[ Kv = 1.602 \]

for only one way bending

\[ Ko = 1 \]

for uniform shock

\[ Km = 1.2 \]

load distribution factor

\[ J = 0.225 \]

(geometrical factor)

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

\[ b = \frac{Ft}{\delta_{a} * kvkokm} \]

\[ 2122.24 * 0.333 * 1.602 * 1.22 * 0.225 = 20.12mm \]

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 sake 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 shaft.

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

\[
W_n = W_t / \cos \phi = 120.57 / \cos 20 = 128.3N
\]

Where; \( W_t \) = tangential load, \( \Phi \) = pressure angle

The weight of the gear

\[
W_g = 0.001 \times 187 \times 5m^2(N) = 0.001 \times 18 \times 22 \times 5 \times 5^2 = 3.245N
\]

• result load on the gear is given by

\[
W_r = \sqrt{W_n^2 + W_g^2 + 2W_n + W_g \cos \phi}
\]

\[
W_r = \sqrt{128.3^2 + 3.245^2 + (2 \times 128.3) + 3.245 \cos 20} = 129.34N
\]

• Bending moment due to the resultant load

\[
M = W_r \times x
\]

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

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

\[
M = 129.34 \times 100 \quad M = 12934 Nmm
\]

• Combined effect of torsion and bending

\[
T_e = \sqrt{M^2 + T^2}
\]

\[
T = \text{twisting moment} = \frac{W_t \times D_p}{2}
\]

\[
= 120.57 \times \frac{66}{2} = 3978.81N \quad mmm
\]

Equivalent twisting moment.

\[
T_e = \sqrt{12934^2 + 3978.81^2} = 13532.157N \quad mmm
\]

\[
T_e = \frac{\pi}{16} \times \tau \times d^2
\]

Where;

\[
\tau = \text{shear stress (Indian standard designation) properties of steel used for shaft. For a shaft strength an alloy steel line nickel- chromium steel is used with ultimate tensile strength (δy)} = 320mpa.
\]

Now shear stress of the material of the gear shaft

\[
\tau = 0.18\delta u \quad \delta u = 0.18 \times 560 = 100.8mpa
\]

\[
T_e = \frac{16}{\pi} \times \tau \times d^2
\]

\[
d^2 = \frac{13532.157 \times 16}{\pi \times 100.88}
\]

\[
D = 8.8mm \text{ take } 10mm
\]

vi. **Shaft Design (For Spur Gear)**

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

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

\[
W_n = W_t / \cos \phi = 530.5 / \cos 20 = 564.54N
\]

(2) The weight of the gear is given by

\[
W_g = 0.00118 \times 75 \times b \times m^2(N) = 0.00118 \times 15 \times 5 \times 5^2 = 2.2125N
\]

(3) result load on the gear is given by

\[
W_r = \sqrt{W_n^2 + W_g^2 + 2W_n + W_g \cos \phi}
\]

\[
= \sqrt{3198 \times 8.2523 = 565.55}
\]

(4) Now bending moment due to the resultant load

\[
M = W_r \times x
\]

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

\[
M = 565.55 \times 20 = 11311.0256N
\]

(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,

\[
T_e = \frac{\sqrt{M^2 + T^2}}{2}
\]

\[
T = \text{twisting moment}=W_t \times D_p / 2
\]

\[
= 530.5 \times \frac{66}{2} = 11936.25N
\]

Therefore equivalent twisting moment

\[
T_e = \frac{\pi}{16} \times \tau \times d^2
\]

\[
d^2 = \frac{16}{\pi} \times \tau \times d^2
\]

\[
= \frac{16}{\pi} \times 16 \times \tau \times d^2
\]

\[
d^2 = \frac{(1644.25 \times 16)/(\pi \times 100.8)}{10mm}
\]

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

\[
W_n = W_t / \cos \phi = 2122.24 / \cos 20 = 2258.44N
\]

(2) the weight of the gear is given by

\[
W_g = 0.00118 \times 75 \times b \times m^2(N) = 0.00118 \times 15 \times 20 \times 5^2(N) = 0.85N
\]

(3) result load on the gear is given by

\[
W_r = \sqrt{W_n^2 + W_g^2 + 2W_n + W_g \cos \phi}
\]

\[
W_r = 2259.4589 N
\]

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

\[
M = W_r \times x
\]

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

\[
M = 79081.06329N
\]

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

\[
T_e = \frac{\sqrt{M^2 + T^2}}{2}
\]

\[
T = \text{twisting moment}=W_t \times D_p / 2
\]

\[
= 47750.4N
\]

Therefore, equivalent twisting moment

\[
T_e = \frac{\sqrt{M^2 + T^2}}{2} \quad 92379.1928N
现在齿轮的直径（d）的确定由使用下列关系，即

\[ T_e = \frac{\pi}{16} \times (\tau) \times d^3 \]

\[ d^3 = \frac{16}{\pi \times \tau} \times T_e \]

### viii. Bolt for fix gear box

4xM5 material is 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 6th 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 6th 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.

\[ W = X \times V \times W_A \]

Where:
- \( V \) = A rotation factor = 1; for all bearing when the inner race is rotating
- \( W = \text{equivalent radial load} \)
- \( W_A = \text{axial or thrust load} \)
- \( Y = \text{radial load factor} \)

**Single row angular constant**

\[ Y = 0 \text{ and } X = 1 \text{ ball bearing from design data} \]

Dynamic load rating for rolling contact bearing variable load

\[ L = \left( \frac{C}{W} \right)^2 \times 10^6 \]

\[ L = \text{rate lift} \]

\( C = \text{basic dynamic load rating} \)

\[ K = 3 \text{ for all ball bearing} \]

- Single row angular bearing; lifetime of bearing

\[ L = \left( \frac{C}{W} \right)^2 \times 10^6 \]

Where \( C = 7.8 \text{ selected} \)

\[ W = X \times V \times W_A - Y \times W_A \]

\[ W_A = \frac{W}{W_e} \leq e = 1.14 \]

\[ W = d \times d \times 2873.8 - 0.73 \times 2471.5 \]

\[ W_A = 0.86 \text{ W} = 1861N \]

Bearing subjected to shock load our service factor (\( k_2 \)) will take design data

\[ k_2 = 1.5 \times \frac{W}{k_2} = 1.5 \times 2873.8 \times 2792.42N \]

\[ L = \left( \frac{C}{W} \right)^2 \times 10^6 = 21.792 \times 10^6 \text{ revolution} \]

Thus, bearing selection is safe

Dynamic equivalent load for rolling contact bearing

\[ W = X \times V \times W_A - Y \times W_A \]

\[ X = 1 \quad Y = 0 \]

Where; \( V = \text{rotation factor} = 1 \), for all bearing when it is inner race rotating

\[ W = \text{equivalent radial load} \]

\[ W_A = \text{axial or thrust load} \]

\( X = \text{radial load factor} \)

\( Y = \text{axial load factor (thrust)} \)

Dynamic load rate for rolling contact bearing under variable load

\[ L = \left( \frac{C}{W} \right)^2 \times 10^6 \]

\[ \frac{W_A}{W_e} = e \]

\( W = X \times V \times W_A - Y \times W_A \)

\( W = 465.35N \)

\( W_A = 465.35N \)

Let the bearing subjected to light shock load service factor (\( k_2 \))

\[ \times W = k_2 \times W = 1.5 \times 105.76 = 698N \]

Know we us calculate the life time (L)

\[ L = \left( \frac{C}{W} \right)^2 \times 10^6 \]

\[ C = 4 \]

\[ = \left( \frac{698N}{3} \right)^2 \times 10^6 = 188 \times 10^6 \text{rev} \]

\[ ^{\ddagger} \text{The bearing section safe and the standard material selected for bearing is Chrome steel.} \]

### 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].

**Figure 1 rectangular sunk key**

Key with shaft diameter of 4mm, shearing stress for mild steel key material is \( \tau = 42 \text{MPa} \)

Crushing stress \( \sigma_c = 70 \text{MPa} \)

- Width of key => \( W = 4 \text{mm} \)
- Thickness of key => \( t = 4 \text{mm} \)

\[ T = l \times W \times \tau \times d/2 = l \times 4 \times 42 \times 10^2 \times 2 \]

\[ T = 840 \times l \text{ Nmm} \]

And tensional shearing stress of the shaft.

\[ T = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 42 \text{N/m} \times 4 \times 10^3 \]

\[ = 8246.68 \text{Nmm} \]

From above equation (1) we have:

\[ T = 840 \times l \text{ Nmm} \]

\[ 8246.68 \text{Nmm} = 840 \times l \text{ Nmm} \]

\[ l = \frac{8246.68}{840} = 9.8 \sim 10 \text{mm} \]

Strength (torque transmitted) of the key.

\[ T = l \times \frac{\pi}{2} \times \sigma_c \times \frac{d^2}{2} = l \times 4 \times 70 \times \frac{10}{2} = 700 \times l \text{Nmm} \]

From eqn (2 & 3) we have
taking the larger of the two values, we have length of key.

\[ l = 11.7 \text{ mm} \]

**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](image)

**Figure 2 Pitch surface for bevel gear**

The velocity ratio,

\[ V \cdot R = \frac{DG}{DP} = \frac{TP}{TP} = \frac{NP}{NG} \]

Where; \( NP \) (speed of pinion) = 50 RPM  
\( NG \) (speed of gear) = 20 RPM

\[ V = \frac{NP}{NG} = 2.5 \]

We know, \( V \cdot R = \frac{DG}{DP} \)

\[ DG = V \cdot R \cdot DP = 2.5 \cdot DP \]

Assume Pinion Number of Teeth Will Be 25  
\( TG = V \cdot R \cdot TP = 2.5 \cdot 25 = 62.5 = 63 \)

We Know That Pitch Line Velocity

\[ V = \frac{1000}{DP} \]

\[ V = \pi \cdot 50 \cdot \frac{1000}{DP} = 7.854 \text{ mm/min} \]

Let, \( Wt = 2471.5N \)

= Force Due To Tire Load + Pressing Forces

\[ 150 \cdot 9.81 + 1000N = 2471.5N \]

Since, Assumption for Tire Weight = 150 kg  
Pressing Forces = 1000N

Now We Can Calculate Torque Acting On the Pinion

\[ T = \frac{P \cdot 60}{2 \pi NP} \]

Where; \( P \) – Power Transmitted From Motor  
\( Np \) – Speed of the Pinion in Rpm

\[ T = 0.75KW \cdot \frac{60}{2 \pi} \cdot 50 = 0.75 \cdot 1000 \cdot \frac{60}{2 \pi} \cdot 50 \]

\[ T = 143.239N \cdot M \]

We Now Tangential Load on the Pinion

\[ Wt = 2471.5N \]

\[ Wt = \frac{2t}{DP} \]

Where; \( DP \) – Pinion Diameter;

\[ Wt \text{- Tangential Load on the Pinion Gear} \]

\[ T \text{- Torque} \]

\[ Dp = \frac{2t}{Wt} = 2 \cdot \frac{143.239}{2471.5} \]

\[ Dp = 115.9mm = 116mm \]

\[ Vp = \frac{DG}{DP} \]

\[ g = Dp \cdot Vp \]

Where; \( Vp = 2.5 \)

\[ Dp = 2.5 \cdot 116mm = 290mm \]

Outside Or Addendum Cone Diameter Knows As

\[ Do = Dp + 2a \cdot \cos \theta_p \]

Where; \( a \) – addendum  
\( \theta_p \) – pitch angle

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

\[ \theta_s = 90^{\circ} \]

\[ \theta_p1 \]

\[ \theta_p2 \]

\[ \theta_p1 = \tan^{-1}(1/Vp) = \tan^{-1}(\frac{Dp}{TP}) = \tan^{-1}(\frac{115.9}{290}) \]

\[ \theta_p2 = \tan^{-1}(Vp, R) = \tan^{-1}(\frac{DG}{TP}) = \tan^{-1}(\frac{115.9}{290}) \]

Pitch angle of the pinion becomes;

\[ \theta_p1 = \tan^{-1}(1/Vp) = 0.38 = 21.8 = 22^{\circ} \]

Pitch angle of the gear becomes;

\[ \theta_p2 = \tan^{-1}(Vp, R) = 68.2^{\circ} = 68 \]

We know the formative number of teeth for pinion.

\[ Tp = \frac{TP}{2 \pi} \cdot \sec \theta_p1 = 25 \cdot \sec 22^{\circ} = 26.963 \]

We know the formative number of teeth for the gear

\[ Te = \frac{Tp}{2 \pi} \cdot \sec \theta_p2 = 63 \cdot \sec 68^{\circ} = 168.18 \]

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

Therefore, tooth form factor for the pinion

\[ Y_p = 0.175 - \frac{0.841}{Tg} = 0.175 - \frac{0.841}{168.18} = 0.1699 \]

Tangential load on the pinion

\[ Wt = \frac{2T}{Dp} = 2 \cdot \frac{143.239}{115.9} \cdot 25 = \frac{11459.12}{11459.12} \cdot N \]

Where; \( m \) = pitch diameter /number of teeth

\[ m = \frac{Dp}{25} = 4.64 \]

Therefore; \( M = 4.64 \text{ say 5mm} \)

\[ Wt = \frac{5mm}{2291.824N} \]

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 cone element or slant height of the pinion cone,

\[ L = \frac{2p}{2 \sin \theta_p1} = m \cdot \frac{7p}{2 \sin \theta_p1} = \frac{116mm}{2 \sin 22^{\circ}} \]

\[ L = 154.8mm \]

Since the face width (b) is 1/4th of the slant height of the pitch cone, there fore
The proportion of the bevel gear may be taken as follows:

- **Addendum,** \( e = 1m = 5\text{mm} \)
- **Dedendum,** \( d = 1.2m = 6\text{mm} \)
- **clearance,** \( = 0.25m = 1\text{mm} \)
- **working depth,\( = 2m = 10\text{mm} \)
- **thickness of tooth = 1.5708n = 7.854\text{mm} \)

### For pinion

Outside or addendum cone diameter is known as

\[
Do = Dp + 2a \cos \Theta p1 = 116\text{mm} + 2 \times 5 \cos 22^\circ = 125.27\text{mm}
\]

Inside or addendum cone diameter is known as

\[
Dp = Dp - 2d \cos \Theta p1 = 116 - 2 \times 6 \cos 22 = 104.87\text{mm}
\]

Addendum angle known as

\[
(\alpha) = \tan^{-1} \left( \frac{e}{Dp} \right) = \tan^{-1} \left( \frac{5}{155.6} \right) \quad \alpha = 2^\circ
\]

Dedendum angle known as

\[
(\beta) = \tan^{-1} \left( \frac{d}{Dp} \right) = \tan^{-1} \left( \frac{6}{155.6} \right) \quad \beta = 2.2^\circ
\]

### For gear

Outside or addendum cone diameter of gear is

\[
Do = Dg + 2a \cos \Theta p2 = 290 + 2 \times 5 \cos 68^\circ = 293.74\text{mm}
\]

Inside or addendum cone diameter of gear is

\[
Dg = Dg - 2d \cos \Theta p2 = 290 - 2 \times 6 \cos 68 = 285.5\text{mm}
\]

### Analysis of Strength of Bevel Gear

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

\[
Wt = (\delta o \times Cv) \times 3.3 \times (L - \frac{B}{L}) = 2471.5\text{N}
\]

Where; \( Cv = 3/3 + V = 0.1413 \)

\( V = \pi Dp N /\text{pitch line velocity} \)

\( B = \text{face width} = 38.7\text{mm} \)

\( M = \text{module} = 5\text{mm} \)

\( Y' = \text{lewis factor} = 0.175 - 0.841/\text{Tep} \)

\( L = \text{slant height of cone} = 154.8\text{mm} \)

\( L - b/l = \text{bevel factor} = 0.75 \)

\( \delta o = \text{allowable static stress} \)

\( V = \pi \times 50 /1000 = 18.22 \text{m/min} \)

\( C v = 0.1413 \text{min/m} \)

\( L - b/L = 0.75 \)

\( \delta o = Wt / (C v \times Y') (L - b/l) \)

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

\( \delta o = 26.67 \text{N/mm}^2 \)

**Allowable static stress (\( \delta o \)) = 26.67 \text{N/mm}^2**

**Taking factor of safety 2**

**Factor of safety = operating stress/Allowable static stress**

**Operating stress = 2 \times 26.67 = 53.34 \text{N/mm}^2**

Alloy steel which has high tooth strength and low tooth wear, yield strength and tensile strength is 250N/mm² and 400N/mm² respectively selected. Therefore, design is safe.

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

In designing a pinion shaft,

\[
T = \frac{P \times 60}{(2\pi Np)} - m
\]

\( P = 0.75 \times 10^3 \times 60 = 143.239\text{N} - m = 143239\text{N} - mm \)

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

\[
Wt = \frac{Rm}{I}
\]

\[
Rm = \frac{L - b}{2L} \times Dp = \left( \frac{L - b}{2} \sin \Theta p \right)
\]

\[
= (154.8 - 38/2) \sin 22 = 50.87\text{mm}
\]

\[
Wt = \frac{Rm}{I} = 2815.78\text{N}
\]

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

\[
Wrh = Wrcos \Theta p1 = Wt \times \tan20 \times \sin 22 = 2815.78 \times 20 \times 22 = 383.92\text{N}
\]

The radial force acting on the pinion shaft

\[
Wrv = Wrcos \Theta p1 = 2815.78\text{tan 20} \times \cos 22 = 950.23\text{N}
\]

Find resulting bending moment on the pinion shaft as follows;

The bending moment due to \( Wrh \) and \( Wrv \)

\[
M_1 = Wrv \times \text{overhang} - Wrh \times Rm
\]

\( \text{Overhang} = 100\text{mm} \)

\( M_1 = 75493.36\text{N} \)

The bending moment due to \( Wt' \)

\[
M_2 = Wt' \times \text{overhang} = 281578\text{N}
\]

Therefore, resultant bending moment

\[
M = \sqrt{M_1^2 + M_2^2} = 291522.58\text{N}
\]

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

\[
T_e = M^2 + T^2 = 324811.46\text{N} - mm
\]

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

\[
Te = \frac{16}{\pi} \times \tau (dp^2)
\]

\[
dp^2 = \frac{16 \times Te}{\pi \times \tau}
\]

\[
Dp = 25.41\text{mm}
\]

Where;

\( Dp = \text{diameter of the pinion shaft, and} \)

\( T = \text{shear stress for the material of the pinion shaft.} \)

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 - 670\text{Mpa} \)

**Yield strength (\( \delta y \)) = 320\text{mpa} \)
Therefore, now let us calculate shear stress of the material of the pinion shaft.

\[ \tau = 0.18 \delta u = 0.18 \times 560 = 100.8 \text{mpa} \]

Therefore:

\[ dp = \frac{3}{\pi} \frac{16 \times Te}{\pi \times \tau} \]

\[ dp = 25.41 \text{mm} = 26 \text{mm} \]

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

\[ T = \frac{p \times 60 \times N \times m}{2 \pi N p} = 0.75 \times 10^3 \times 60 \times \frac{2 \pi 20}{15} \]

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

\[ Wt = \frac{T}{Rm} = 2844.048N \]

\[ Rm = \left( \frac{b}{L} \right) \left( \frac{DG}{2L} \right) = \left( \frac{b}{L} \right) \sin \theta p 2 \]

\[ = \left( 154.8 - \frac{38}{2} \right) \sin 68 \times 2 = 125.911 mm \]

Now find the axial and radial forces (i.e Wrh & Wrv) acting on the shaft as given as given bellow
The axial force acting on the gear shaft

\[ Wrh = Wrsin\theta p 2 = Wt \times \tan \phi \times \sin \theta p 2 \]

\[ Wrh = 956.773N \]

The radial force acting on the gear shaft

\[ Wrv = Wrcos\theta p 2 = Wt \times \cos \phi \times \cos \theta p 2 \]

\[ Wrv = 387.77N \]

Find resultant bending moment on the gear shaft as follows;
The bending moment due to Wrh and Wrv is given by

M1 = Wrh \times overhang + Wrv \times Rm

Where;

Overhang = 515mm

M1 = 78069.99N

Bending moment due to Wt,

M2 = \sqrt{M1^2 + M2^2} = 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

\[ Te = \left( \frac{\pi}{16} \right) \times \tau dG^3 \]

\[ dG^3 = \frac{3Te \times 16}{\pi \times \tau} \]

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 material used is the same for both shafts \( \sigma_c, \sigma_t \), \( \tau \) will be the same

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

\( \sigma_u = 560\text{mpa}, \sigma_y = 320\text{mpa} \)

\( \tau = 0.8\sigma_t, \) where; \( \sigma_t = 320\text{mpa}, \sigma_c = 2\sigma_t \)

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

\( \sigma_t = \sigma_y = 320\text{mpa} \)

\( \tau = 0.8(320) = 256\text{mpa} \)

\( \sigma_cr = 2(320\text{MPa}) = 640\text{Mpa} \)

IV. FAILURE ANALYSIS

A. Failure of the rods in tension (compression)

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

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

Tearing strength of the rods equation with tangential load
\[ W_T = \frac{P}{\pi} \times d^2 \times \sigma_c \]

Where \( W_T = 2471.5 \text{N} \)

\[
\sigma_{com} = \frac{4}{\pi} = \frac{4(2471.5)}{(43.2)^2 \pi} = 1.7 N/mm^2
\]

Therefore, the design is safe according to tension.

**B. Failure of spigot in tension for slot**

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

\[
A = \frac{\pi}{4} (d_2^2 - d_1^2) \times t
\]

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

\[
W_T = \frac{\pi}{4} (d_2^2 - d_1^2) \times t \sigma_c
\]

\[
\sigma_{com} = \frac{\pi}{4} (d_2^2 - d_1^2) \times t = \frac{2471.5}{(52.03)^2 - 52.03 \times 13.33}
\]

\[
= \frac{2126.16 - 693.56}{2471.5} = 1.725 \text{mpa}
\]

**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

\[
P = \frac{W_T}{d_2 \times t} = \frac{2471.5}{52.03 \times 13.33} = 3.56 \text{mpa}
\]

**D. Failure of the socket in tension across the slot**

Area of resisting area of the socket a cross the slot

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

\[
\sigma_t = \frac{\pi}{4} \left( d_2^2 - (d_2^2) \right) - (d_1 - d_2) t = \frac{2471.5}{(52.03)^2 - 52.03 \times 13.33}
\]

\[
\sigma_t = \frac{\pi}{4} (75.25^2 - 52.03^2) - (75.25 - 52.03 \times 13.33)
\]

\[
= \frac{1.228 \text{mpa}}{2.1655 \text{mpa}}
\]

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

\[
A = 2b \times t
\]

Shearing strength of the cotter = P

\[
P = \frac{P}{2b \times t} = \frac{2471.5}{(55.9) \times 13.33} = 1.658 \text{mpa}
\]

Therefore, the design is safe according shear.

**F. Failure of the socket cotter in crushing**

Crushing the area that cursing of socket collar

\[
= (d_4 - d_2) t \text{ and crushing strength } = (d_4 - d_2) t \times \sigma_c
\]

\[
P = \frac{P}{2(d_4 - d_2) t} = \frac{2471.5}{(103.2 - 52.03) \times 13.33}
\]

\[
\sigma_{cr} = \frac{3.623 \text{mpa}}{2(52.03) \times 13.33} \text{ therefore the design is safe}
\]

**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

\[
P = \frac{P}{2(d_4 - d_2) c} = \frac{2471.5}{2(103.2 - 52.03) \times 13.33} = 0.75 \text{mpa}
\]

**H. Failure of rod end in shear**

The area of shear is double therefore the area resisting shear of the rod end \( A = 2a \times d_2 \)

The shear strength of the load end (P)

\[
P = 2a \times d_2 \times \tau
\]

\[
\tau = \frac{P}{2a \times d_2} = \frac{2471.5}{2(52.25) \times 52.03} = 0.736 \text{mpa}
\]

**I. Failure of spigot collar in crushing**

Crushing the of the collage \( A = \frac{\pi}{4} (d_2^2 - d_2^2) \)

\[
\sigma_c = \frac{\pi}{4} (d_2^2 - d_2^2) \frac{P}{(d_2^2 - d_2^2) \times \pi}
\]

\[
equating this to load (P), we have
\]

\[
P = \frac{\pi}{4} (d_2^2 - d_2^2) \text{ safe}
\]

\[
\sigma_c = \frac{2.1655 \text{mpa}}{2.1655 \text{mpa}} = 2.1655 \text{mpa}
\]
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 \( P = \pi d^2 \times t_1 \times \tau \)
equation this to load (P) we have
\[ P = \frac{2471.5}{\pi \times 52.03 \times 19.25} = 0.781 \text{ mpa} \]

K. Failure of cotter in bending
The maximum bending moment occurs at the center of the cotter and is given by
\[ M_{\text{max}} = \frac{P(d_4 - d_2)}{6} + \frac{d_4 \times d_2}{4} \]

We now that section modulus of cotter \( Z = t \times \frac{b^2}{6} \)

Bending stress in the cotter,
\[ \sigma_b = \frac{M_{\text{max}}}{Z} = \frac{P(d_4 - d_2)}{6} + \frac{d_4 \times d_2}{4} \]
\[ \sigma_b = \frac{2471.5(101.2+0.5\times52.03)}{2 \times 12.55 \times 55.91} = 3.771 \text{ mpa} \]

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.
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 plate
\[ B = 150mm = 0.15m \quad H = 10mm = 0.01m \]
\[ I = \frac{0.15 \times (0.01)^3}{12} = 1.25 \times 10^{-8} m^4 \]
\[ \frac{1}{\rho} = \frac{27.8Nm}{200 \times 10^5 \cdot \frac{N}{m^2} \times 1.25 \times 10^{-8}} \]
\[ \frac{1}{\rho} = 11.12 \frac{1}{m} \quad \rho = 0.089m \approx 0.00089mm \]
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
\[ S = \text{size of the weld} \]
\[ t= \text{throat thickness} \]

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]
\[ \Delta \text{ the Area will be considered as} \]
\[ A = \pi D_t \quad \text{where} \quad t = 0.707.s \]
\[ = \pi \times 43mm \times 0.01mm \]
\[ \pi \times 43mm \times 0.707s = 95.51mm.s \]
Now let as find the bending moment as we know
\[ M = P \times \text{perpendicular distance} = 2471 \times 75 \]
\[ = 185.36KN.mm \]
Polar moment of inertia
\[ \tau = \frac{t(b + l)^3}{6} \]

Figure 14 rectangular section
Section modulus for circular section (Z)
\[ Z = \frac{\pi d}{4} = \frac{1026.7 \times S}{mm^3} \]
Then now let as determine bending stress (\( \sigma_b \))
\[ \sigma_b = \frac{M}{Z} = \frac{185.36KNmm}{1026.7mm^3} \quad \sigma_b = 0.180.62 \frac{N}{mm^2} \]
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 (\( \tau_{max} \))
\[ \tau_{max} = \frac{1}{2} \sqrt{\left(\sigma_b \right)^2 + 4\tau^2} \]
\[ \tau_{max} \quad \text{Is selected from the date table. The frame is welded with electrode specification E60.} \]
Since our plate is ASTMGRAND 450 CARBON steel. Maximum shear from the design data,
\[ \tau_{max} = 70mpa \]
\[ \frac{180.62}{S} \left( \frac{NN}{mm^2} \right) + 4 \left( \frac{25.88}{S} \frac{N}{mm^2} \right)^2 \]
\[ \frac{99.445}{70N/mm^2} N/mm^2 \approx 2 \text{ mm} \]

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)
\[ \Sigma M = 0 \quad -2347.5N \times \frac{l}{2} + R_2 \times L = 0 \]
\[ R_2 = \frac{P}{2} = 1235.75N \]
Taking point ‘B’ as a reference
\[ R_A = -\frac{P}{2} = -1235.75N \]
Taking left of center (A-C)
\[ \text{Where} \quad X \to 0 < X < \frac{l}{2}(OR)540mm \]
Take $x = 0$

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

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

Where $x = 0$, $M = 0$

Taking right of center (D - B)

Figure 16 right of center (D - B)

$0 < X < 540mm$ Take $x = 0$

$$R_B = \frac{p}{4} = 1235.75N$$

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

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

Taking point A as reference

$$V = \frac{p}{2} = 2471 - 1235.75$$

$$V = \frac{-p}{2} = -1235.75N$$

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

$$M_{\text{max}} = 667.305Nm, m$$

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

Figure 17 from A to B

Selection from A to B

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 $\text{ASTM} = A709\ \text{grand} \ 690\ (\text{quenched \ and \ tempered})$

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

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

The maximum deflection $Y=\text{maximum deflection}$

$$\frac{1}{E} \frac{M_{\text{max}}}{I} = \frac{1}{12} \frac{667.305Nm}{\frac{480 \times 1080}{12}} = \frac{0.48m}{12} \times \frac{0.01m^2}{12}$$

$$l = \frac{1}{y} \frac{667.305Nm}{200 \times 10^4 \left(\frac{N}{m^2}\right)} = \frac{83.41 \times 1m}{m}$$

$$Y = 0.012m$$

The maximum bending moment is 667.305Nm on $\sigma_{\text{max}} = 667.305Nm$

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 $S = \text{size of the weld} \quad t = \text{throat thickness}$

$$A = 2t \times l \times 0.707 \cdot s \cdot 40 = 56.56.smm^2$$

$$A = 2t \times l \times 2 \times 0.707 \cdot s \cdot 40 = 56.56.smm^2$$

Now let us find the bending moment knows as $M = P \times \text{perpendicular distance}$

$$= P \cdot e = 1235.75 \times 240 = 296580Nm$$

Know find polar moment of inertia (j)

$$J = \frac{t(b+l)^4}{6}$$

Figure 20 Cross section of plate

Section modulus for rectangular section (z)

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

Then now let us determine bending stress ($\sigma_b$)

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

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

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

$$318.155\text{mm}^2$$
Now let us determine direct shape stress
We know maximum shape stress 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

\[ \tau_{\text{max}} = \frac{1}{2} \sqrt{\sigma_b^2 + 4\tau_e^2} \]

\[ \tau_{\text{max}} = \frac{1}{2} \sqrt{932.2^2 + 4 \times 21.64^2} \]

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

\[ \sigma_b = \frac{M}{z} = \frac{296580}{318.15 \times 5} = \frac{932.2}{5} \text{ N/mm}^2 \]

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 load will be equal to the applied thrust load (fa).

For thrust ball bearing, the dynamic equivalent thrust load is determined by

\[ p_a = \frac{f_r}{E} + \frac{f_a}{E} \]

For standard TVL and DTVL bearing having a contact angle, \( \alpha = 0.76 \) and \( \beta = 1.00 \). Minimum Ratio to maintain proper operation for these applications is 1.56.

\[ \beta = 0.76 \times 0 + 1 \times 2471.5 \text{ N/mm}^2 \]

**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;

\[ L_{10} = \left( \frac{C}{p_r} \right)^6 \left( \frac{10^6}{60} \right) \text{ hours} \]

\[ L_{10} = \left( \frac{C}{p_r} \right)^6 (1 \times 10^6) \text{ revolutions} \]

For thrust bearing the above equation change to the following

\[ L_{10} = \left( \frac{C}{p_e} \right)^6 (1 \times 10^6) \text{ Revolutions} \]

\[ L_{10} = \left( \frac{C}{p_e} \right)^6 \left( \frac{1 \times 10^6}{60n} \right) \text{ Hours} \]

E = 3 for ball bearing
E = \( \frac{10}{3} \) for tapered, cylindrical and spherical roller bearing. Tapered roller bearings typically used a dynamic load rating based on 90 million cycles, denoted as \( C_{90} \), changing the equation as follows and for tapered thrust bearing number 51109, the basic dynamic capacity \( C_a = 28035N \)

\[ L_{10} = \left( \frac{28035 \times 10}{2471.5} \right)^{\frac{3}{2}} \times (90 \times 10^6) \]

xxi. Design of I-Beam
A beam of alloy steel that is with \( E=200 \text{KN/mm}^2 \) and an I-section 80mm*202mm*10mm and 1.6m long is used to 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

\[ M_{\text{max}} @ x = \frac{F L}{I} \]

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 \( (W_{cr}) \) under various and conditions is represented by a general equation.

\[ W_{cr} = \left( \frac{3\pi^2 EI}{L^2} \right) = \frac{3\pi^2 E A K^2}{L^2} = \frac{3\pi^2 E A}{(L/K)^2} \]

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-x

\[ I_{xx} = \frac{BD^3}{12} - \frac{bh^3}{12} = 200 \times 80^3 - (200 - 10)(60 - 20)^3 \]

\[ I_{yy} = 7520000 \text{mm}^4 \]

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

\[ I_{yy} = 2 \int (x^2 - \bar{x}^2) \ dx = 2 \int \left[ 10^2 + 200^2 \right] - \frac{100}{12} + \frac{400}{12} \]

\[ = 13.34 \times 10^6 \text{mm}^4 \]
Since $I_{xx}$ is less than $I_{yy}$, therefore the section will tend to buckle about $x$-$x$ axis. Thus, we shall take $I$ as $I_{xx} = 7.52 \times 10^6 \text{mm}^4$

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

$L = l / 2 = 1600 / 2 = 800 \text{mm}$

We know that the crippling load

$W_{cr} = \frac{\pi^2 E I}{L^2} \left( \frac{\pi^2 \cdot 200 \times 10^4 \cdot 7.5 \times 10^6}{800^2} \right) = 23193570.34 \text{N}$

**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.

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

$\text{Stress area} = 70 \text{mm}^2$

$P_i = 2840 (10) \text{N} = 28400 \text{N}$

$\text{Safe tensile load}$

$ps = \text{stress area} \times \delta t$

$ps = 70 \text{mm}^2 \times 250 \text{N/mm}^2$

$ps = 17,500\text{N}$

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

The average threads shearing stress for the threads ($\tau_s$) is obtained by using the relation.

$\tau_s = \frac{p}{\pi \cdot d \cdot b \cdot \sigma_y}$

$\tau_s = \frac{9.988 \times 8 \times 2.147}{32.89 \text{MPa}}$

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

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

$L = \text{distance of shank of the bolt}$

$E = \text{young’s modulus of the bolt material}$

$X = \text{difference in height between the extreme corners of the nut or head}$

$\delta b = \frac{(30 \times 200 \times N}{2 \times 50 \text{mm}^2}$

$\delta b = 60 \text{N/mm}^2 = 60 \text{MPa}$

Bending stress is for total no of bolts 8, will be $\delta b / 2 = 60 / 8 = 7.5\text{MPa}$ 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 arms 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 two. The total load the at are exerted in the pin

$F_{total} = 2524.6\text{N}$

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

$F_{arm} = \frac{F_{total}}{2} = 1262.3\text{N}$

**Assumption**

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

![Free body diagram of arm](image)

Therefore the maximum moment on the arm will be;

$M = -f \times e = 1262.3 \times 0.32$

$= -403.84\text{N/m clockwise}$

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

$\delta_{ult} = 550\text{MPa}$

$\delta_{yield} = 450\text{MPa}$

$E = 200\text{Gpa}$

$G = 77\text{Gpa}$

Since the cross-section area is rectangular shape;

$I = \frac{bx^3}{12}$ and $Y = \frac{d}{2}$

$I = (0.01 \times 0.04) / 12 = 5.33 \times 10^{-8}\text{m}^4$

$Y = 40 / 2 = 0.04 / 2 = 0.02\text{m}$

Now let as determine the bending stress on the arm support

$\delta b = \frac{Mby}{I}$

$\delta b = \frac{403.84 \times 0.02}{5.33 \times 10^{-8} \times 151.535 \text{MPa}}$

The material used for arm support is with maximum bending stress is much less than the yield stress 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 $F_{cyl}$ required to drive a load force $F_{load}$, force $F_{load}$ that drive by cylinder force $F_{cyl}$ and length $L2$ it is from load road pin to the fixed hinge.

Counter clockwise moment = clockwise moment

$F_{cyl} \left( L_2 \cos \theta \right) = F_{load} \left( L_2 \cos \theta \right)$

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

$F_{cyl} = 1000\text{N}$
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

\[
L_1 = 340 \text{mm} \quad L_2 = 223 \text{mm}
\]

Therefore \( F_{cyl} (L_1) = F_{load} (L_2) \)

\[
F_{load} = F_{cyl} (L_1 / L_2) = 1000 (340/223)
\]

\[
F_{load} = 1524.6 \text{N}
\]

**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 required to separate tire and rim. That is mentioned on the previous calculation.

\[
L_1 = 340 \text{mm} \quad L_2 = 223 \text{mm}
\]

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.

\[
\sum F_{cyl} = 0 = R_b - F_{cyl} - F_{load} = 0
\]

\[
R_b = F_{cyl} + F_{load} = 1000 + 1524.6
\]

\[
R_b = 2524.6 \text{N}
\]

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\([1][2][5][6]\)

\[
\text{Therefore: } M_b = P(L_2 - d_1)
\]

**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

\[
A_s = \pi / 4D^2
\]

Shear strength of the pin = \( \pi / 4D^2 \tau \)

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 \( M=D \rightarrow \) there is not bending moment for the pin only the shear stress is applied to the pin

\[
\tau = \frac{F}{A} = \frac{2524.6}{314.15} = 8.036 \text{MPA}
\]

Taking factor of safety \( (f_s=3) \)

\[
Fos = \text{maximum stress/allowable working stress}
\]

\[
\text{Maximum stress} = f_s * \text{working stress}
\]

\[
= 3 * 8.036 = 24.1 \text{N/mm}^2
\]

\[
\delta \text{max} = 24.1 \text{mpa}
\]

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.

\[
M_b = P(L_2 - d_1)
\]

\[
= 1000 (340 - 20) = 320 \text{Nm}
\]

The cross section of the lever can be rectangular

\[
F = \frac{bd^2}{12} \quad \text{and} \quad y = \frac{d}{2}
\]

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

\[
I = 3 * 10^{-2} m^4
\]

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

\[
\delta b = \frac{MbY}{I} \quad \text{where; } Y = \frac{d}{2}
\]

\[
Y = 20 = 0.02m
\]

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

\[
\delta b = 2.1314 \text{MPa}
\]

**xxvi. Design of Pneumatic System**

In ACBSE pneumatic system normally work with low pressure \((p \leq 1 \text{Mpa})\) and a temperature close to \(+20^\circ C \) (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 \([8]\)

<table>
<thead>
<tr>
<th>Gas constant</th>
<th>( R = 287 \text{ J/(K.G.K)} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molar mass</td>
<td>( M = 29 \text{ kg/kmol} )</td>
</tr>
</tbody>
</table>

Specifc heat at Constant pressure \( c_p = 1005 \text{ J/(K.G.K)} \)

Specific heat at Constant volume \( c_v = 718 \text{ J/kg.k} \)

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 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.

$Q(m^3/s) = \text{area} \times \text{distance moved per second}$

$Q(m^3/s) = A \times \text{velocity (full side)}$

$Q(m^3) = (A - a) \times \text{velocity (rod side)}$

In the case of air cylinder, $Q$ is volume of compressed air and this changes with pressure so any variation in pressure will cause a variation in the velocity.

Power: mechanical power = $P = F \cdot V$ 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.

$=\text{operating pressure}$

$p = 10\text{bar} = 10 \times 10^5\text{N/m}^2$

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

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

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

$D^2 = \left(\frac{1000 \times 4}{\pi \times 1000000}\right) = 4000$

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

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 ($\delta t$)

$\delta t = 0.86(t)$ $[7][8]$

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

$\delta t = \frac{P \times d}{2t} = \frac{(1 \times 40)}{(2 \times 3)} = 6.61\text{mpa}$

Therefore,

$D = \text{outer diameter} = 40mm$

$T = \text{thickness of cylinder} = 3mm$

$L = \text{length of cylinder} = 300mm$

$\delta t = \text{circumferential or hoop stress of the material}$

**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;

$C = \left(\frac{\pi}{4}\right) \times D^2 \times (p + 1.013) \times \left(\frac{L}{1000}\right)$

$C = \left(\frac{\pi}{4}\right) \times 4^2 \times (10 + 1.013) \times 30 = 3.36\text{litr s}$

Free air consumption for return stroke

$C = \left(\frac{\pi}{4}\right) \times (D - d^2) \times (p + 1.013) \times \left(\frac{L}{1000}\right)$

$C = \left(\frac{\pi}{4}\right) \times (3.6 - 2^2) \times (10 + 1.013) \times \frac{L}{1000}$

$C = 0.32\text{litr s}$

For one complete cycle air consumption will be

$(3.36 + 0.32) = 3.68\text{ liters}$

$Q(m^3/s) = A \times v(\text{full side})$

$Q(m^3/s) = (A - a) \times v(\text{rod side})$

Speed for forward stroke $Q = A \times v$

$V = \frac{Q}{A} = 3.45mm^3/s \times \left(\frac{\pi}{4}\right)^2$

$= 3.48/(\pi/4 \times 0.036^2mm^2)3.41m/s$

**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.

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

Where Area = $\pi \times r$

Stroke = cylinder travel (in)  A = pressure drop constant

$C_t = \text{compression factor}$  Time = in second

Inlet pressure (psi) = 120

A = $\pi \times \left(\frac{1.5836}{2}\right)^2 \times 1.836 \text{ in}^2 \cdot \text{Stroke} = 11.81 \text{ in}^2$

Use A constant at 5psi $\Delta P$ for most application

$A = 0.039, C_t = 9.2 \cdot \text{Time} = 2 s$

Therefore; $C_v = \frac{1.5836 + 11.81 \times 0.039 + 9.2}{2 \times 29} = 0.175 = 0.18$

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.
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

\[ R = r_i + \frac{h(b_i+2b_o)}{2(b_i+b_o)} = \frac{70}{3} (60+20) \] 

\[ = 54.16 \text{mm} \]

\[ e = R - r_n = 54.16 - 40.6 = 13.56 \text{ mm} \]

\[ h_i = r_n - r_i = 54.16 - 25 = 29.16 \text{ mm} \]

\[ A = \frac{1}{2} (b_i + b_o) h = \frac{1}{2} (60 +20 )70 = 2800 \text{mm}^2 \]

Therefor moment, \( M_b = P * R = P * 54.16 = 54.16 \text{ P N-mm} \)

Bending stress (occurring at inner surface)

\[ \sigma_{b,max} = \frac{M_b}{A e + r_i} = \frac{54.16 \times 29.16}{2800 \times 13.56 + 25} = 1.66 \times 10^{-3} \text{ P N/mm}^2 \]

Direct stress,

\[ \sigma_b = \frac{P}{h_i} = \frac{P}{29.16} = 3.57 \times 10^{-4} \text{ P N/mm}^2 \]

Total stress,

\[ (1.66 \times 10^{-3} + 3.57 \times 10^{-4}) \text{ P N/mm}^2 \]

\[ = 2.017 \times 10^{-3} \text{ P N/mm}^2 \]

Since \( \sigma_{all} = \frac{2800}{2} = 126.6 \text{ N/mm}^2 \)

Equating the equation \( \sigma_{b,max} = \sigma_{all} \)

\[ 2.017 \times 10^{-3} \text{ P N/mm}^2 = 126.6 \text{ N/mm}^2 \]

\[ P = 62964.79 \text{ N} = 62.96 \text{ KN} \]

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