Development of an Automobile Tyre Remover from Rim

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ABSTRACT

In order to circumvent the difficulty in disengaging an automobile tyre from its rim, using manually operated devices, an automobile tyre remover from rim was developed in this research work. The design of the major components such as the vertical link, horizontal link, determination of the capacity of the required screw jack ,the exerted stress on the tyre by the ring press and bending stress on the load carrying frame member were carried out. On performance evaluation, the automobile tyre remover was able to disengage tyre of sizes of 165/65,175/65 and 185/65 from rim sizes of R13, R14 and R15 in 10, 13 and 17 seconds respectively. The efficiency was found to be 14%.

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1.0 INTRODUCTION

Flat tyre occurs when a tyre deflates ,as a result of a leak or normal tear and wear or more serious damage. If a vehicle with a flat tyre is driven continuously, the occupants and other vehicles are put in danger and the flat tyre as well as its rim perhaps be damaged beyond repair. A tyre that has lost sufficient pressure will impair the stability of the vehicle A flat tyre or low pressure tyre should be considered as emergency situation, requiring immediate attention. (Abcnews,2008).

The damage done to the tyre determines the type of maintenance to be carried out. If there is a leak or leakage in the tubed or tubless tyre, it can be sealed. If the tyre is burst , torn or worn, it can be replaced. Whatever type of maintenance is to be carried out,the tyre must be removed or disengaged from its rim. This is being achieved by using powered or manually operated devices or equipment. The manually operated devices which are commonly used, are strenuous and requires a lot of human effort in operation, most times, the vulcanizers using the manually operated device would resort back to using a hammer, which may damage the tyre, as a result of frustration over not being able to apply the required effort on the tyre removing epuipment. This technique led to the nickname to a tyre removing worker to be called a tyre buster and the act of changing a tyre to be known as tyre busting. (http://www.wisegeek.com/tyrechanger)

In order to circumvent the problem of applying so much effort in the operation of the manually operated type, development of powered screw automobile tyre remover from rim was embarked on.

2.0 MATERIALS AND METHODS

The material selected for the design of an automobile tyre remover from rim is mild steel (AISI 1080) having 0.15 to 0.3% Carbon. It is ductile, tough and hard. The magnitude of the yield stresses and ultimate strength are: Yield stress in tension = 370MPa, Ultimate tensile strength =440MPa, Yield stress in shear = 210MPa and Yield stress in compression = 407.7MPa (http://www.answers.com)

2.1 DESIGN THEORY AND CALCULATIONS

The automobile tyre remover consists of the following parts.

(1) Screw Jack (2) Vertical links (Columns) (3) Horizontal link (Beam) (4) Frame and

(5) Ring press

2.1.1 Determination of the Capacity of the Required Screw Jack

The screw jack is the component of the tyre remover that supplies the effort needed to disengage the vehicle tyre from its rim by transmitting the rotary motion of the screw into translatory motion. Basic dimensions of the screw according to IS4694-1968 (Reaffirmed1996) cited by Khurmi and Gupta (2010) are as follows:

Major diameter, d_{maj} of screw = 40mm and Pitch of screw, p = 7mm

Therefore, mean diameter of screw is determined as $d_m = d_{maj} - \frac{p}{2} = 40 - \frac{7}{2} = 36.5$mm

Core diameter d_c = 33mm, Major diameter of nut = 40.5mm and the Length of lever, L = 185mm

According to Hannah and Hillier (1995), the torque (T) required to rotate the screw is given as

$$T = W\frac{d_m}{2}[Tan(\emptyset + \theta)]$$

Where W=Axial load on the screw, d_m= Mean diameter, $\theta = Helix$ angle, $\emptyset$ = Friction angle and μ = coefficient of friction between screw and nut which is given as 0.15

Then $\mu = Tan\emptyset$

$\emptyset = Tan^{-1}\mu = Tan^{-1}0.15 = 8.5^{\circ}$

and Tan $\theta = \frac{p}{\pi d_m}$

Tan $\theta = \frac{7}{40.5\pi} = 0.0610$, $\theta = Tan^{-1}0.0610 = 3.5^{\circ}$

Turning moment of screw T = Maximum Effort (F) applied on moment arm ×length (L) of lever i.e T = F × l

For F = 50N and l = 185mm,

$$Fl = W\frac{d_m}{2}\tan(\theta + \emptyset)$$

$$\therefore\ W = \frac{Fl}{\frac{d_m}{2}\tan(\theta+\emptyset)} = \frac{50 \times 185}{\frac{36.5}{2}\tan(8.5+3.5^{\circ})} = 2384.5N$$

Since the maximum axial load (W) the screw jack is subjected is calculated to be

2384.5N, a 650kg capacity jack which is the minimum capacity of available screw jack was selected.

2.1.2 Determination of the Thickness of the Vertical link(Column)

The vertical link is a flat bar which is rectangular in shape is made of mild steel material. It is the vertical member responsible for supporting compressive load and transmitting vertical force to the tyre as supplied by the screw jack. The vertical link is bound to fail when it is subjected to compression above its load carrying capacity.

The load on each vertical link $F_1 = \frac{w}{2} = \frac{2384.5}{2} = 1192.3N$

Assuming a factor of safety (F.S) of 4, the two columns must be designed for buckling load.

Buckling load $W_{cr} = F_1 \times F.S = 1192.3 \times 4 = 4{,}769N$

Area of Cross-section of link = b× t

Where b = width of link = 35mm

t = thickness of link

Compressive stress σ_c of the vertical link $= \frac{Yield\ stress\ \sigma_y}{Factor\ of\ safety} = \frac{407.7}{4} = 101.93N$

Rankine formula for buckling load is given as

$$W_{cr} = \frac{\sigma_c A}{1 + a(\frac{L}{K})^2} \tag{2.1}$$

$$4769 = \frac{101.93 \times 35 \times t}{1+\frac{1}{750}(\frac{L}{K})^2} \tag{2.2}$$

Where the length of the link L = 310mm

The moment of inertia of the cross-section of the link is given as

$$I = \frac{1}{12}t \times 35^3 = 3572.92t \tag{2.3}$$

The radius of gyration K = $\sqrt{\frac{I}{A}} = \sqrt{\frac{3572.92t}{35t}} = \sqrt{102.08333} = 10.1mm$

From equation (3.2)

$$4769 = \frac{101.93 \times 35 \times t}{1+\frac{1}{750}(\frac{L}{K})^2}$$

Substituting for $\frac{L}{K} = 30.7$

t = 3mm

So standard thickness of 4mm flat bar was selected for the fabrication of the link.

2.1.3 Determination of the critical buckling load on the vertical link

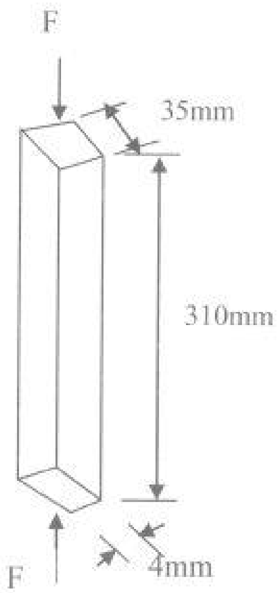


Figure 1. Vertical link

T ve stress. In order to know whether the column is safe or not, the critical crippling load on the column must be determined. In determining the crippling load, there is comparison between the value of $\frac{L}{K}$ and $\sqrt{\frac{2C\pi^2E}{S_y}}$ which shows the appropriate column criteria needed for analyzing the crippling load.

From equation (2.3)

$I = 3572.92t = 3572.92 \times 4 = 14{,}291.68mm^4$

Since the column has a rectangular cross-section, the area is determined as, **Area (A)** = bh

Where b = thickness = 4mm and h = width = 35mm

$\therefore bh = 4 \times 35 = 140mm^2$

For L = 310mm, and K = 10.1mm

$\frac{L}{K} = \frac{310}{10.1} = 30.7$ and

$\sqrt{\frac{2C\pi^2E}{S_y}}$ is computed with the following parameters:

E = Young's modulus of elasticity of material = $210 \times 10^3 Nmm^{-2}$, C = Constant representing end fixity coefficient of the column = 1 and S_y = Yield stress of column material = 407.7MPa

Hence, $\sqrt{\frac{2C\pi^2E}{S_y}} = \sqrt{\frac{2\pi^2 \times 1 \times 210 \times 10^3}{407.7}} = 100.8$

Since the value of $\frac{L}{K}$ is below $\sqrt{\frac{2C\pi^2E}{S_y}}$, therefore, J.B Johnson's equation is suitable for determining the crippling load on the column(Hall et al, 2004)

The J.B Johnson's equation for determining the crippling load of a column is given as

$$W_{cr} = S_y A\left(1 - \frac{S_y (\frac{L}{K})^2}{4C\pi^2 E}\right)$$

Where S_y = yield stress of mild steel in compression 407.7MPa, A = Cross-sectional area of column = $140mm^2$, L = Length of column = 310mm, K = Radius of gyration = 10.1mm, C = Constant representing end fixity coefficient of the column = 1 and E = Modulus of elasticity of column material = $210 \times 10^3 Nmm^{-2}$

Therefore, $W_{cr} = 407.7 \times 140\left(1 - \frac{407.7 \times (\frac{310}{10.1})^2}{4\pi^2 \times 1 \times 210 \times 10^3}\right) = 5{,}556{,}657N$

2.1.4 Determination of the thickness of the horizontal link(Beam)

The horizontal link is the straight horizontal member that provides surface for attachment of the vertical links (column). It is a circular shaped hollow pipe.

According to Rajput (2012), the bending equation is given as $\frac{M}{I} = \frac{\sigma}{y}$

Where M = Maximum bending moment, I = Moment of inertia, σ_b = Bending stress of beam in tension and y = Distance from the neutral axis of beam to the outer.most fibre

$M = \sigma_b Z$, $\quad Z = \frac{I}{y_{max}}$, $\quad I = \frac{\pi}{64}(d_o{}^4 - d_i{}^4)$ and $y_{max} = \frac{d_o}{2}$

Therefore, section modulus, $Z = \frac{\pi}{64}(d_o{}^4 - d_i{}^4) \times \frac{2}{d_o}$

$$M = \sigma_b \frac{\pi}{32}(d_o{}^4 - d_i{}^4) \times \frac{1}{d_o} \qquad (2.4)$$

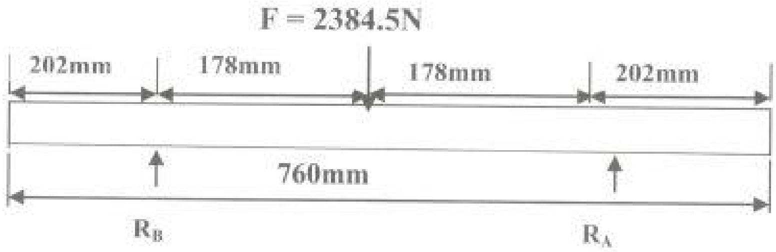


Figure 2 horizontal link with the position of the acting load.

link and the maximum bending moment is determined by taking moment about A,

$M_A = 2384.5 \times 178 = 424{,}441$Nmm

Taking bending stress σ_b to be limited to 191MPa, Outer diameter $\boldsymbol{d_o}$ of the beam to be 35mm, internal diameter $\boldsymbol{d_i}$ of the pipe can be determined from equation (2.4) as

$$\left(\frac{d_o{}^4 - d_i{}^4}{d_o}\right) = \frac{M_{max}}{\sigma_b \frac{\pi}{32}} = \frac{424{,}441}{191 \times \frac{\pi}{32}} = 22{,}635$$

$$d_o{}^4 - d_i{}^4 = 22{,}635 d_o$$

$$35^4 - d_i{}^4 = 22{,}635 \times 35$$

$$d_i = \sqrt[4]{708{,}400} = 29\text{mm}$$

Thickness t, of pipe $= d_o - d_i = 35 - 29 = 6$mm

So a standard thickness of 6mm hollow pipe was selected for the fabrication of the horizontal link.

2.1.5 **Determination of compressive stress exerted on the tyre by the ring press**

The ring press is a pair of semi-circular shaped components. Each half of the ring is fitted to the column to transmit compressive force to the tyre. The compressive stress exerted by the ring press on the tyre, is determined as follows:

σ_{cr} = compressive stress, F_2 = weight of each vertical link = 1N, F_1 = weight of beam = 2N

F = total weight on the ring = $F_1 + F_2 + W = 2 + 1 + 2384.5 = 2387.5N$

d = diameter of ring = 355.4mm and t = ring thickness = 6.5mm

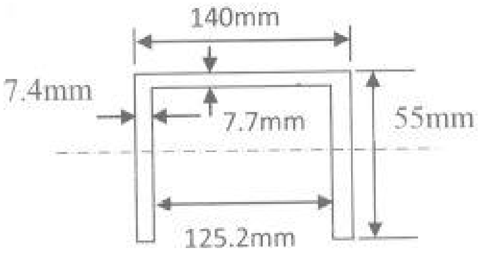
$$\sigma_{cr} = \frac{F_1+F_2+w}{\pi dt} = \frac{2387.5}{\pi \times 355.4 \times 6.5} = 0.33\text{MPa}$$

2.1.6 **Determination of Bending stress on the frame member that carries load**

The frame is the component of the automobile tyre remover which serves as the body and an enclosure to all other components of the design. The U-channel which form the top of the frame carries the load that is exerted on the tyre to be removed from the rim and it is subjected to bending stress. Its dimensions are as follows:

B = Width of U-channel = 140mm, b = internal width = 125.2mm, t = base thickness = 7.7mm,

a = top thickness = 7.4mm, H = height = 55mm, h = distance from top of U-channel to mid section = $\frac{H}{2}$ =27.5mm and $h_{1=}$ vertical distance from base mid section = h = 27.5mm



Area (A) = Figure. 3 Cross-section of U- channel

Moment of inertia (I) = $I_{XX} = \frac{Bh^3 - b(h-t)^3 + ah_1{}^3}{3}$

$$= \frac{140 \times 27.5^3 - 125.2(27.5-7.7)^3 + 14.8(27.5)^2}{3} = 650301mm^4$$

Distance (y) from neutral axis to the extreme fibre is given as

$$y = \frac{aH^2+bt^2}{2(aH+bt)} = \frac{14.8 \times 55^2 + 125.2(7.7)^2}{2(14.8 \times 55 = 125.2(7.7))} = 14.7\text{mm}$$

Section Modulus (Z) = $Z_{XX} = \frac{2I_{XX}(aH+bt)}{aH^2+bt^2} = \frac{2 \times 650301((14.8 \times 55)+(125.2 \times 7.7))}{(14.8 \times 55^2)+(125.2 \times 7.7)} = 23{,}148.9mm^3$

Radius of gyration (K) = $K_{XX} = \sqrt{\frac{I_{XX}}{Bt+(h-t)a}} = \sqrt{\frac{650301}{140 \times 7.7+(55-7.7)14.8}} = 19.1\text{mm}$

M = moment of resistance of each member of the frame = $\frac{I}{y_{max}}$ = 14,238Nmm

Z = section modulus = $23{,}148.9mm^3$

The bending stress (σ_b) on the frame members is computed as $\sigma_b = \frac{M}{Z} = \frac{14238}{23148.9} = 0.62\text{MPa}$

Therefore,

Based on the design theory and calculations, the automobile tyre remover from rim was produced. The component and assembly drawings are shown in figures 4 and 5 respectively.

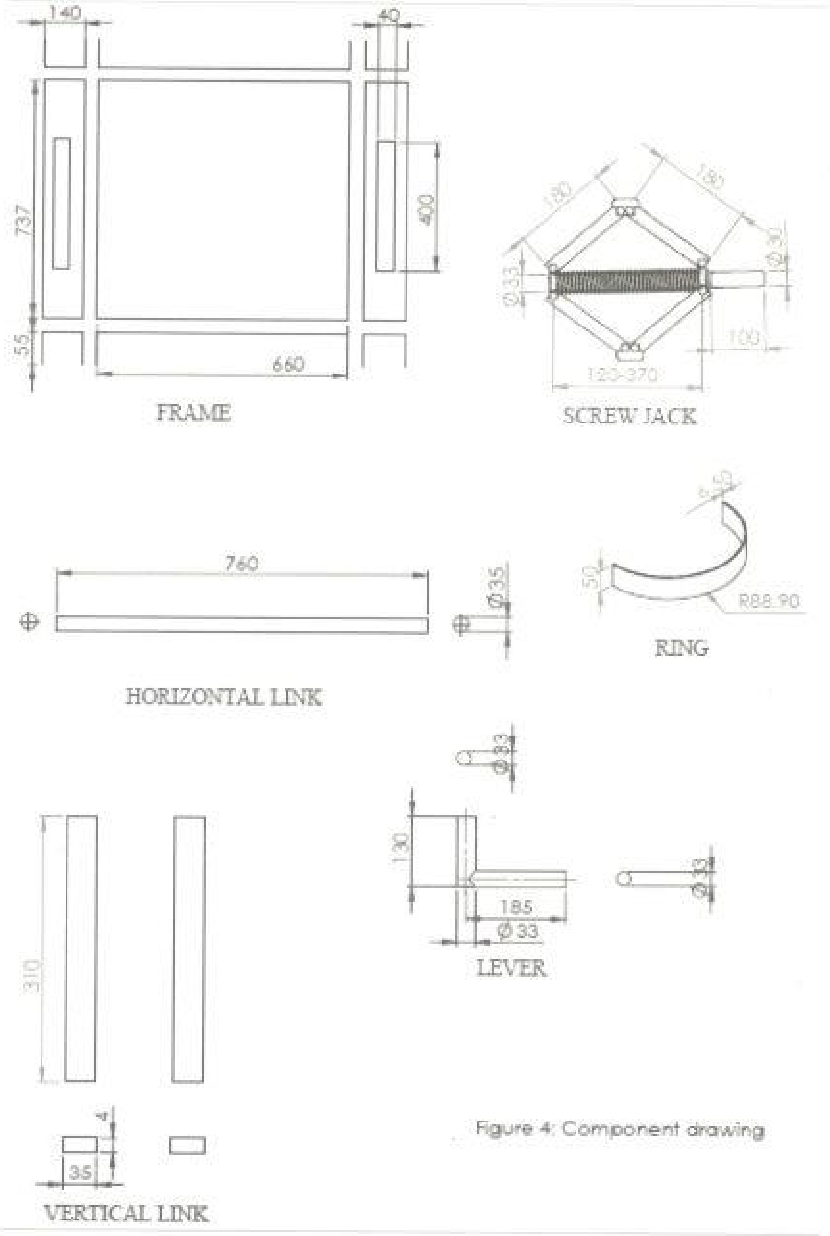
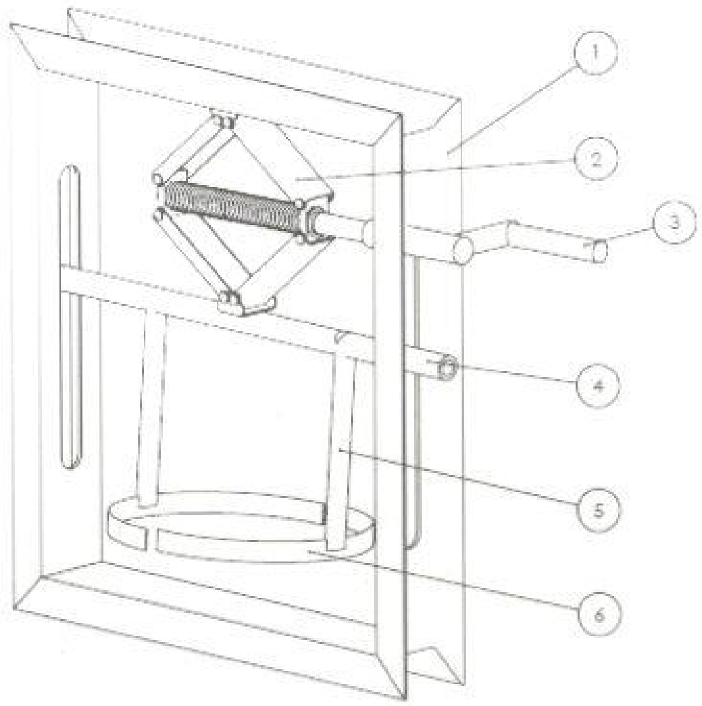


Figure 4: Component drawing



S/N	PART	MATERIAL	QTY
1	FRAME	MILD STEEL	1
2	SCREW JACK	MILD STEEL	1
3	LEVER	MILD STEEL	1
4	HORIZONTAL LINK	MILD STEEL	1
5	VERTICAL LINK	MILD STEEL	2
6	RING	MILD STEEL	2

Figure 5: ASSEMBLY DRAWING

2.4 Performance Evaluation

Tyre and rim sizes of 165/65R13,175/R14 and 185/R15 were individually placed on the horizontal floor of the frame of the developed tyre remover and the ring was aligned with the edges of the tyre. The lever was wound in clockwise direction making the screw jack to move vertically downward. As a result, all other components joined to its base also moved downward in a continuous linear path. As the operator continued to wind the lever, the tyre got disengaged from the rim.

2.5 Determination of Efficiency of the automobile tyre Remover

The efficiency of the Tyre Remover is determined as the ratio of the mechanical advantage (M.A) to velocity ratio(V.R)

That is $\eta = \frac{MA}{VR} \times 100 = \frac{WP}{2\pi LF} \times 100 = \frac{2384.5\times 0.007}{50\times 2\square \times 0.370} \times 100 = 14.4\%$

3.0 RESULTS AND DISCUSSION

3.1 Results

The time taken to disengage different sizes of tyre from rim is tabulated below.

Table 3.1 Results of Time taken to remove tyre from rim.

S/N	Tyres and Rim sizes	Numbers of revolutions of the lever	Time taken to disengage tyre (seconds)
1	165/65R13	12	10
2	175/65R14	15	13
3	185/65R15	17	17

3.2 Discussion of Results

The results obtained from the performance evaluation of the tyre remover showed that, tyres of small sizes are more easily disengaged from rim than those of larger sizes owing to the fact that, the tyres with larger sizes have larger contact surfaces with the rim than smaller ones. It is evident in table 3.1 that, the larger the diameter of a tyre, the more the number of revolutions required on the lever and the more the time spent to disengage the tyre from rim

4.0 CONCLUSION

The use of automobile tyre remover from rim cannot be overemphasized because, prior to the maintenance of tyre, it has to be disengaged from its rim . Due to the high cost of imported means of removing tyres from rim and much effort required to operate the manually operated tyre remover, the developed automobile tyre remover from rim is advantageous in terms of ease of operation and less labour intensive because of the incorporation of screw mechanism.

5.0 REFERENCES

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