

Design Analysis of a Portable Manual Tyre Changer

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Abstract — In this paper, an efficient, affordable, and portable manual automobile tyre changing tool was designed. The tool operates on the principle of second-class levers, where the load to be overcome is situated between the fulcrum and the effort point. The mechanical advantage (MA) of the bead breaker arm and pry bar assembly of the tool was determined to be 11.5. The standard tube size of the bead breaker arm that can withstand stress due to bending moment was determined to be a 2" (50mm) diameter Schedule 80 steel tube. Also, the dimensions of the standard hexagonal bolts to be used at the fulcrum and load point of the bead breaker arm are M12 × 1.5 × 75 mm, while the standard size of hexagonal nuts to be used with the bolts is M12 × 1.5 mm with a height of 10mm. Moreover, the mechanical advantage (MA) of the mount/demount arm and pry bar assembly of the tool was determined to be 5.22. Also, by comparing the angles of twist of two sizes of the mount/demount arm when the tool is used in mounting or demounting a tyre on a 16" × 7" wheel rim, the standard size of the mount/demount arm that can withstand stresses due to bending moment and torsional moment was determined to be a 60mm × 60mm × 5mm square tube. The cost of materials needed to fabricate the tyre changer summed up to ₦21,000. Ergo, the design provides an alternative portable and relatively affordable tyre changing tool that can be afforded by tyre technicians across Nigeria, and other developing or underdeveloped countries.

Index Terms — Bead Breaker, Design, Lever, Tyre Changer.

I. INTRODUCTION

A tyre is a torus-shaped rubber covering that surrounds the wheel rim and transfers the load of a vehicle from the axle, through the wheel rim to the ground, thereby providing traction on the surface over which the tyre-rim combination travels. Tyres consist of a tread which provides traction, and a body which contains a certain quantity of compressed air. Therefore, majority of tyres are air inflated structures, which provide an elastic cushion that reduces the effect of shock as the tyre rolls over rough features on the ground. Studies show that it is important to maintain the correct inflation pressures in tyres in order to keep vehicle dynamics, handling performance and braking at their best [1]-[4]. The basic functions of the tyres on a vehicle are supporting the weight of the vehicle, guiding the vehicle along trajectories decided by the driver, transmitting braking or acceleration forces and absorbing irregularities on the road. It is for these reasons that the tyre-rim combination has been described as one of the most important and critical components of an automobile in

several studies [5]-[10]. Basically, modern pneumatic tyres are made from natural rubber, synthetic rubber, carbon black, fabric and wire, as well as other chemical compounds. The structure and layers of a typical tyre are shown in Fig. 1.

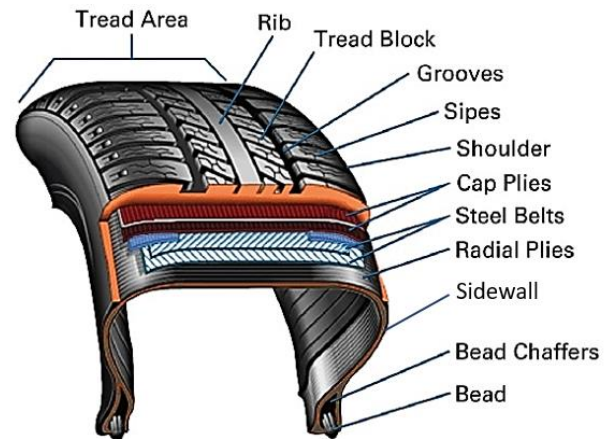


Fig. 1. Tyre structure and engineered layers [11].

As tyres carry out their functions over time, they fail due to wear from friction or damage in the form of blowouts, and the subsequent replacement of these tyres becomes necessary. After the wheel rim and tyre assembly are removed from the vehicle, tyre changing tools and machines help tyre technicians demount and mount tyres on the wheel rim. It is pertinent to state that the tyre changing process can be extremely laborious, thereby making the technician prone to injury and the wheel rim susceptible to damage, if proper tyre changing tools, machines or techniques are not used during the tyre changing process. Over the years, studies have shown that most of the injuries that occur while servicing tyres happen during the tyre inflation process, with the head and face of the tyre servicing personnel being the most commonly injured regions [12]-[15].

The tyre bead is the innermost diameter of the tyre that interfaces with the rim of the wheel. The surface of the bead creates a seal between the tyre and the wheel rim to prevent leakage of air. Unfortunately, the bead can become stuck to the rim after corrosion occurs. Therefore, tyre changing requires a process known as bead-breaking. Bead-breaking is the process of separating the tyre beads from the rim, by applying force on the sidewall of the tyre, in close proximity to the wheel rim flange.

Even though various designs of tyre changing machines are available for loosening beads and changing tyres automatically, these machines are usually ponderous and not affordable by the majority of tyre technicians across Nigeria and most parts of other developing or underdeveloped countries. Moreover, the locally fabricated bead-breaking tool which is the sole tyre changing tool

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commonly available at the tyre service centres of tyre technicians across Nigeria is not portable, barely useful for the bead-breaking operation, and completely inadequate for the tyre mounting and demounting operation. This makes the tyre changing process at these tyre service centres characterised by drudgery and waste of time. The commonly available, inefficient locally fabricated bead breaker is shown in Fig. 2.

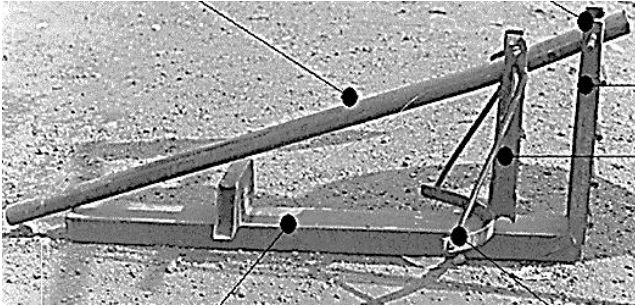


Fig. 2. Ineffective locally fabricated bead breaker [7].

From the foregoing, there is the need for more portable, complete and affordable tools or machines that can carry out the tyre changing process efficiently. The portable manual tyre changing tool designed in this paper works based on the second-class lever principle, in which the load to be overcome is situated between the fulcrum and the effort point. The design is based on standard tube sizes, making it relatively affordable and easy to fabricate.

II. LITERATURE REVIEW

There are various tools, machines, and methods for changing the tyre on a wheel rim. When the appropriate tools, machines and methods are not utilized, the tyre changing process can be both hectic and hazardous. Apparently, the first step of replacing a bad tyre on an automobile is to remove the wheel from the wheel hub of the automobile. The conventional method of carrying out this operation is to use a lug wrench to loosen the lug nuts attaching the wheel to the wheel hub of the vehicle, one lug nut at a time. However, machines that can loosen and tighten all lug nuts on the wheel simultaneously have been developed in several studies [16]-[23], thereby saving the tyre servicing personnel valuable time and energy.

The next step of the tyre changing process is to deflate the tyre and separate it from the rim. Adetan *et al.* [6] described various tools, machines and techniques used in separating tyres from wheel rims. They described the least expensive technique as a process of using a rubber mallet to repeatedly hit the sidewall of a tyre until it separates from the rim. Clearly, this method takes time and is labour-intensive. Similarly, another crude method described by Adetan *et al.* is the drive-over technique. In this technique, a vehicle with a good fully inflated tyre is driven over the sidewall of a bad deflated tyre to be changed, in order to separate the tyre from the rim. This method is dangerous because if it is not done properly, the deflated tyre can stand upright and hit the side of the vehicle, or worse still, injure bystanders. Another dangerous method of bead-breaking described by Adetan *et al.* is the high-lift jack technique, where the deflated tyre is placed under the bumper of a

vehicle, and the high-lift jack is positioned between the tyre and the bumper. The jack is then activated. This method is dangerous because the tyre can lift up on the side opposite the jack, and roll in unanticipated directions. The design proposed by Adetan *et al.* in their paper, is a bead breaker that utilizes compound levers to carry out the bead-breaking task. Though the tool designed by Adetan *et al.* is portable and affordable, the tool lacks the ability to mount or demount a tyre on a wheel rim.

After the bead-breaking step, the tyre is demounted from the wheel rim. Conventionally, the demounting operation is carried out with the aid of tyre bars, and can be a very tedious task. Finally, a new tyre is mounted on the wheel rim, inflated and replaced on the wheel hub of the vehicle.

Scrap tyres which have lost their useful life need to be recycled. Recycling and recovery of waste tyres is a major environmental problem because vulcanized rubbers usually take several years to decompose naturally and thus, remain for long periods of time in the environment. About 78 percent of scrap tyres end up in overcrowded landfills, and thousands more are strewn across empty car lots, highways and illegal tyre dumps [24]. Therefore, options such as discarding end-of-life tyres as waste in landfills or incineration for energy recovery are no longer acceptable for the environment as they pose numerous environmental and health risks.

Some of the present usages of scrap tyres are tyre-derived fuels, artificial reefs, erosion control and crumb rubber, an asphalt additive. However, all of the recycling, re-use and recovery practices combined only consume about 22 percent of discarded tyres [24]. Various studies [24]-[29] have shown that old tyres can be broken down into smaller aggregates known as ground tyre rubber (GTR) that can be used as fillers in civil engineering applications, as well as blended with polymeric matrices such as thermoplastics, thermosets or virgin rubber. Though rubber from scrap tyres can be added to concrete as coarse aggregates to achieve durability, less ductility and greater crack resistance in the concrete, it was discovered that rubber reinforced concrete has a low compressive strength when compared with ordinary concrete [29].

Apart from the ecological advantages of utilizing tyre-derived fuels, there has been a growing interest in using these fuels as a result of the fact that tyre-derived fuels have a higher heating value than both coal and wood [30], [31]. Moreover, waste tyre pyrolysis which is the chemical decomposition of waste tyres at elevated temperatures in the absence of oxygen, produces important chemical substances such as synthesis gas, benzene, toluene and xylenes. Synthesis gas can be used for fuel, electricity and chemicals, while benzene, toluene and xylenes are used as primary feedstocks to produce plastics, resins, fibres, surfactants and pharmaceuticals.

III. MATERIALS AND METHODS

The material chosen for the tyre changing tool is plain carbon steel 30C8. Plain carbon steel 30C8 was selected because it possesses good strength and stiffness to resist applied forces and stresses.

A. Calculation of Permissible Stresses

The allowable bending stress, σ_b , and allowable shear stress, τ , were obtained from [32] as:

Allowable Bending Stress,

$$\sigma_b = \frac{S_{yt}}{(fs)} \quad (1)$$

where S_{yt} = Yield strength in tension (N/mm²).

fs = Factor of safety.

Allowable Shear Stress,

$$\tau = \frac{S_{sy}}{(fs)}$$

where S_{sy} = Yield strength in shear (N/mm²).

But from [32]

$$S_{sy} = 0.5S_{yt} \\ \therefore \tau = \frac{0.5S_{yt}}{(fs)} \quad (2)$$

B. Design Calculations for the Bead Breaker Section

The bead breaker section of the tyre changing tool has a bead breaker arm which is basically a circular tube into which a telescoping pry bar can be fitted. The assembly was modelled as a second-class lever as shown in Fig. 3.

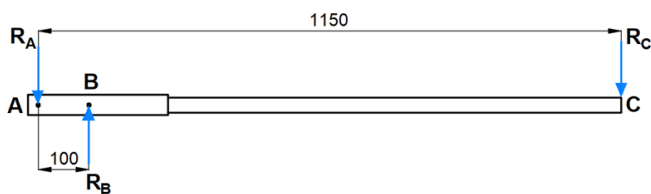


Fig. 3. Free body diagram of the bead breaker arm and pry bar assembly of the tyre changing tool.

From Fig. 3, point A is the fulcrum, while points B and C are the load and effort points respectively. The load arm, l_{1B} , is 100mm long while the effort arm, l_{2B} , is 1150mm long. Since, l_{2B} is greater than l_{1B} , the mechanical advantage of the lever is greater than 1.

1. Determining the Mechanical Advantage of the Bead Breaker Arm and Pry Bar Assembly

The mechanical advantage of the bead breaker arm was obtained from [33] as:

Mechanical Advantage,

$$MA = \frac{l_2}{l_1} \quad (3)$$

where l_2 = length of effort arm (mm).

l_1 = length of load arm (mm).

2. Determining the Load, R_B

The force applied at the effort arm of the lever by the operator, $R_C = 733N$, which is the maximum power grip strength of a man working with two hands [34]. This value was obtained by assuming that the maximum power grip on

the effort arm of the lever was performed using two hands, with an optimal grip span (50 ± 20 mm), neutral wrist posture and while wearing heavy heat-treated gloves. It was also assumed that the fingers of the operator can curl around the pry bar being grasped and that the surface of the bar is not slippery.

Taking moments about the fulcrum, A

$$R_B \times l_{1B} = R_C \times l_{2B}$$

$$R_B = \frac{R_C \times l_{2B}}{l_{1B}} \quad (4)$$

3. Determining the Reaction at the Fulcrum, R_A

Considering equilibrium of forces and summing vertical forces, taking upward forces as positive and downward forces as negative:

$$R_A = R_B - R_C \quad (5)$$

4. Determining the Tube Diameter of the Bead Breaker Arm

The bead breaker arm of the tyre changing tool was designed to be made from a plain carbon steel 30C8 circular tube. Therefore, it was necessary to determine the appropriate size of the circular tube considering the forces acting at various points on the tube. The bending stress, σ_b , was obtained from [35] as:

$$\sigma_b = \frac{M_b y}{I} \quad (6)$$

where M_b = Applied bending moment (N·mm).

y = Distance from the neutral axis to the most extreme fibre of the shaft (mm).

I = Area moment of inertia (mm⁴).

The area moment of inertia, I , of a circular section was obtained from [36] as:

$$I = \frac{\pi r^4}{4}$$

where r = Radius of the circular section (mm).

Therefore, the area moment of inertia, I , of a circular hollow section is given by

$$I = \frac{\pi(r_o^4 - r_i^4)}{4} \quad (7)$$

where r_o = Outer radius of the circular hollow section (mm).

r_i = Inner radius of the circular hollow section (mm).

Also,

$$y = \frac{d_o}{2} = r_o \quad (8)$$

where d_o = Outer diameter of the circular hollow section (mm).

By substituting (7) and (8) into (6)

$$\sigma_b = \frac{4M_b r_o}{\pi(r_o^4 - r_i^4)} \quad (9)$$

Assuming

$$r_o = 1.2r_i \quad (10)$$

By substituting (10) into (9)

$$\sigma_b = \frac{4.4709M_b}{\pi r_i^3}$$

$$\therefore r_i = \left(\frac{4.4709M_b}{\pi \sigma_b} \right)^{\frac{1}{3}} \quad (11)$$

Tube wall thickness,

$$t_B = r_o - r_i \quad (12)$$

5. Determining the Sizes of Hexagonal Bolts and Nuts

The shanks of the bolts at A and B are subjected to double shear stress. The shear stress, τ , was obtained from [37] as

$$\tau = \frac{P}{A}$$

where P = Load (N).

A = Area of the shank of the bolt (mm^2).

Since the bolts are subjected to double shear stress,

$$\tau = \frac{P}{2A} = \frac{P}{2 \left(\frac{\pi}{4} d^2 \right)}$$

$$\therefore d = \sqrt{\frac{2P}{\pi \tau}} \quad (13)$$

The diameter of the shank of the bolt at A, d_A , and that of the shank of the bolt at B, d_B , were calculated from (13).

The height or thickness, h, of the standard nut to be used with the bolts at A and B was obtained from [32] as

$$h = 0.8d \quad (14)$$

where d = Nominal diameter of the bolt (mm).

C. Design Calculations for the Mount/demount section

The mount/demount section of the tyre changing tool has a duckhead bar on which a plastic duckhead is attached. The duckhead bar was designed to be adjustable on a mount/demount arm between 40 mm and 330 mm from the centre of the centring post sleeve, which serves as the fulcrum. It was assumed that the tyre changing tool was being used to demount or mount a tyre on a 16-inch wheel rim of 7 inches width. So, the duckhead bar was adjusted on the mount/demount arm to about 200mm from the fulcrum, which is the centre of the centring post sleeve. The duckhead bar was also adjusted to a length of about 300mm from the centre of the mount/demount arm to the surface of the wheel rim. Therefore, the assembly of the mount/demount arm and the pry bar was modelled as a second-class lever as shown in Fig. 4.

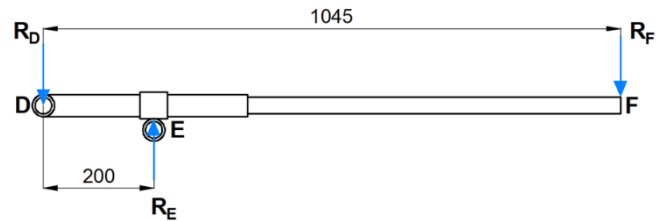


Fig. 4. Free body diagram of the mount/demount arm and pry bar assembly of the tyre changing tool.

While mounting or demounting a tyre, the assembly of the mount/demount arm and the pry bar is rotated clockwise or anticlockwise about the centre of the centring post sleeve which serves as the fulcrum. The rotation of the mount/demount arm causes a rotation of the duckhead bar, which carries the duckhead, in the same direction. The clockwise or anticlockwise rotation of the duckhead between the tyre and the wheel rim subjects the mount/demount arm to stresses due to bending moment, as well as stresses due to torsional moment. Therefore, it was necessary to use a square tube for the mount/demount arm, in order to prevent rotation of the duckhead arm about the longitudinal axis of the mount/demount arm.

1. Determining the Mechanical Advantage of the Mount/demount Arm and Pry Bar Assembly

From Fig. 4, point D is the fulcrum, while points E and F are the load and effort points respectively. The load arm, l_{1M} , is 200mm long while the effort arm, l_{2M} , is 1045mm long. Since, l_{2M} is greater than l_{1M} , the mechanical advantage of the lever is greater than 1. The mechanical advantage was calculated from (3).

2. Determining the Load, R_E

Again, the force applied at the effort arm of the lever by the operator, $R_F = 733\text{N}$, which is the maximum power grip strength of a man working with two hands [34]. This value was obtained by assuming that the maximum power grip on the effort arm of the lever was performed using two hands, with an optimal grip span (50 ± 20 mm), neutral wrist posture and while wearing heavy heat-treated gloves. It was also assumed that the fingers of the operator can curl around the pry bar being grasped and that the surface of the bar is not slippery.

Taking moments about the fulcrum, D,

$$R_E \times l_{1M} = R_F \times l_{2M}$$

$$R_E = \frac{R_F \times l_{2M}}{l_{1M}} \quad (15)$$

3. Determining the Reaction at the Fulcrum, R_D

Considering equilibrium of forces and summing vertical forces, taking upward forces as positive and downward forces as negative,

$$R_D = R_E - R_F \quad (16)$$

4. Determining the Tube Size of the Mount/demount Arm

The mount/demount arm of the tyre changing tool was designed to be made from a plain carbon steel 30C8 square tube. Therefore, it was necessary to determine the

appropriate size of the square tube considering the forces acting at various points on the tube.

a. Determining the Tube Size of the Mount/demount Arm based on Stresses due to Bending Moment

The bending stress, σ_b , is given by (6). The area moment of inertia of a rectangular section about a centroidal axis, I , in (6) was obtained from [36] as

$$\bar{I}_{x'} = \frac{bh^3}{12} \text{ and } \bar{I}_{y'} = \frac{b^3h}{12} \quad (17)$$

where $\bar{I}_{x'}$ = Area moment of inertia about the centroidal x-axis (mm^4).

$\bar{I}_{y'}$ = Area moment of inertia about the centroidal y-axis (mm^4).

b = Length of rectangular section (mm).

h = Width of rectangular section (mm).

For a square section, b = h, therefore the area moments of inertia, I , in (17) become

$$I = \bar{I}_{x'} = \bar{I}_{y'} = \frac{b^4}{12} \quad (18)$$

where b = Length of a side of the square section (mm).

While for a square hollow section, the area moments of inertia, I , in (17) become

$$I = \frac{b_o^4 - b_i^4}{12} \quad (19)$$

where b_o = Length of a side of the outer square section (mm).

b_i = Length of a side of the inner square section (mm).

Also

$$y = \frac{b_o}{2} \quad (20)$$

By substituting (19) and (20) into (6)

$$\therefore \sigma_b = \frac{6M_b b_o}{b_o^4 - b_i^4} \quad (21)$$

Assuming

$$b_o = 1.2b_i \quad (22)$$

By substituting (22) into (21)

$$\sigma_b = \frac{6.7064M_b}{b_i^3}$$

$$\therefore b_i = \left(\frac{6.7064M_b}{\sigma_b} \right)^{\frac{1}{3}} \quad (23)$$

Tube wall thickness,

$$t_{MB} = \frac{b_o - b_i}{2} \quad (24)$$

b. Determining the Tube Size of the Mount/demount Arm based on Stresses due to Torsional Moment

The torque, T , setup in the mount/demount arm by the clockwise or anticlockwise rotation of the duckhead between the tyre and the wheel rim was calculated from

$$T = R_E \times L \quad (25)$$

where R_E = The force acting on the duckhead arm (N).

L = The perpendicular distance from the longitudinal axis of the mount/demount arm to the force R_L (mm).

The shearing stress at any given point of the mount/demount arm wall, τ , was obtained from [38] as

$$\tau = \frac{T}{2t\alpha} \quad (26)$$

where T = Torque applied to the hollow member ($\text{N}\cdot\text{mm}$).

t = Tube wall thickness (mm).

α = Area bounded by the centre line (mm^2).

Assuming

$$l_o = 1.2l_i, \quad (27)$$

where l_o = Length of a side of the outer square section (mm).

l_i = Length of a side of the inner square section (mm).

Then, tube wall thickness,

$$t_{MT} = \frac{l_o - l_i}{2} = \frac{1.2l_i - l_i}{2}$$

$$t_{MT} = 0.1l_i \quad (28)$$

Also, length of a side of the square section bounded by the centre line,

$$l_c = l_o - t_{MT}$$

$$l_c = 1.2l_i - 0.1l_i$$

$$l_c = 1.1l_i \quad (29)$$

By substituting (27), (28) and (29) into (26)

$$\tau = \frac{T}{2 \times 0.1l_i \times 1.1l_i \times 1.1l_i}$$

$$\therefore l_i = \left(\frac{T}{0.242\tau} \right)^{\frac{1}{3}} \quad (30)$$

c. Determining the Angle of Twist of the Mount/demount Arm

The angle of twist of a thin-walled hollow noncircular shaft, ϕ , was obtained from [38] as

$$\phi = \frac{TL}{4\alpha^2 G} \oint \frac{ds}{t} \quad (31)$$

where T = Torque ($\text{N}\cdot\text{mm}$).

L = Length of shaft (mm).

α = Area bounded by the centre line (mm^2).

G = Modulus of rigidity (N/mm^2).

ds = An element of the centre line of the wall cross section (mm).

t = Wall thickness (mm).

Since the mount/demount arm has constant wall thickness, (31) becomes

$$\phi = \frac{TLs}{4\alpha^2Gt} \quad (32)$$

where s = Length or perimeter of the centre line (mm).

IV. RESULTS AND DISCUSSION

The design equations were applied, and with the required inputs, results were obtained. The design inputs, results and discussions are presented in this section.

A. Design Input Parameters

Some design inputs were required for computing the results of the design. These input parameters and their corresponding values are shown in Table I.

TABLE I: DESIGN INPUT PARAMETERS

Input Parameter	Value	Unit
Yield strength in tension, S_{yt}	400	N/mm ²
Factor of safety, (fs)	5	-
Yield strength in shear, S_{sy}	200	N/mm ²
Length of bead breaker load arm, l_{1B}	100	mm
Length of bead breaker effort arm, l_{2B}	1150	mm
Force applied at effort point of bead breaker arm and pry bar assembly, R_C	733	N
Length of mount/demount arm load arm, l_{1M}	200	mm
Length of mount/demount arm effort arm, l_{2M}	1045	mm
Force applied at effort point of mount/demount arm and pry bar assembly, R_F	733	N
Perpendicular distance from the longitudinal axis of the mount/demount arm to R_L, L	300	mm
Modulus of Rigidity, G	77×10^9	N/mm ²
Length of mount/demount arm subjected to torsion, L_T	200	mm

B. Design Calculations and Results

The design input parameters were substituted into the design equations and results for component parts of the manual tyre changer were obtained. The calculations are shown below.

1. Calculation of Permissible Stresses

Equations (1) and (2) were used to calculate the allowable bending stress, σ_b , and allowable shear stress, τ , as follows:

$$\text{Allowable Bending Stress, } \sigma_b = \frac{400}{5} = 80\text{N/mm}^2$$

$$\text{Allowable Shear Stress, } \tau = \frac{0.5 \times 400}{5} = 40\text{N/mm}^2$$

2. Determining the Mechanical Advantage of the Bead Breaker Arm and Pry Bar Assembly

Equation (3) was used to calculate the mechanical advantage, MA_B , of the bead breaker arm and pry bar assembly as follows:

$$\text{Mechanical Advantage, } MA_B = \frac{1150\text{mm}}{100\text{mm}} = 11.5$$

3. Determining the Load, R_B

By taking moments about the fulcrum, A, the load, R_B was determined from (4) as follows:

$$R_B = \frac{733\text{N} \times 1150\text{mm}}{100\text{mm}} = 8430\text{N}$$

4. Determining the Reaction at the Fulcrum, R_A

The reaction at fulcrum A, R_A was also determined from (5) as follows:

$$R_A = 8430\text{N} - 733\text{N} = 7697\text{N}$$

5. Developing the Shear Force Diagram of the Bead Breaker Arm and Pry Bar Assembly

The shear force diagram of the bead breaker arm and pry bar assembly is shown in Fig. 5, and the values were calculated as follows:

$$F_A = -R_A = -7697\text{N}$$

$$F_B = -7697\text{N} + 8430\text{N} = +733\text{N}$$

$$F_C = +733\text{N} - 733\text{N} = 0$$



Fig. 5. Shear force diagram of the bead breaker arm and pry bar assembly.

6. Developing the Bending Moment Diagram of the Bead Breaker Arm and Pry Bar Assembly

The bending moment diagram of the bead breaker arm and pry bar assembly is shown in Fig. 6, and the values were calculated as follows:

$$M_A = 0$$

$$M_B = -R_F \times l_1$$

$$M_B = -7697\text{N} \times 100\text{mm} = -769700\text{N} \cdot \text{mm}$$

$$M_C = 0$$

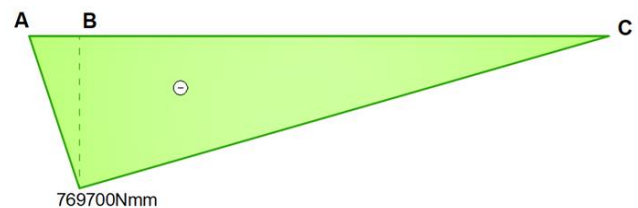


Fig. 6. Bending moment diagram of the bead breaker arm and pry bar assembly.

The load, shear force and bending moment values of the bead breaker arm and pry bar assembly are listed in Table II.

TABLE II: LOAD, SHEAR FORCE AND BENDING MOMENT VALUES AT VARIOUS POINTS ON THE BEAD BREAKER ARM AND PRY BAR ASSEMBLY

Points	Distance from Fulcrum (mm)	Load (N)	Shear Force (N)	Bending moment (N·mm)
A	0	7697	-7697	0
B	100	8430	733	-769700
C	1150	733	0	0

Therefore, the maximum bending moment, M_b is 769700N·mm, and it occurs at point B on the bead breaker arm.

7. Determining the Tube Diameter of the Bead Breaker Arm

Equation (11) was used to calculate the inner radius of the circular hollow section of the bead breaker arm, r_i , as follows:

$$r_i = \left(\frac{4.4709 \times 769700 \text{N}\cdot\text{mm}}{\pi \times 80 \text{N}/\text{mm}^2} \right)^{\frac{1}{3}} = 23.92 \text{mm}$$

Equation (10) was used to calculate the outer radius of the bead breaker arm, r_o as follows:

$$r_o = 1.2 \times 23.92 = 28.70 \text{mm}$$

Equation (12) was used to calculate the bead breaker arm tube thickness, t_B , as follows:

$$t_B = 28.70 \text{mm} - 23.92 \text{mm} = 4.78 \text{mm}$$

However, from the ASME B36.10M standard for tube sizes [39], the standard tube size for the bead breaker arm was selected to be a 2-inch (50mm) diameter Schedule 80 steel tube.

8. Determining the Sizes of Hexagonal Bolts and Nuts

On the bead breaker arm, the bolt at A was assumed to be subjected to a force of 7697N while the bolt at B was assumed to be subjected to a force of 8430N. Therefore, it was necessary to determine the sizes of bolts that can withstand the forces at those points. Equation (13) was used to calculate the diameter of the shank of the bolt at A, d_A , as follows:

$$d_A = \sqrt{\frac{2 \times 7697 \text{N}}{\pi \times 40 \text{N}/\text{mm}^2}} = 11.07 \text{mm}$$

However, from the ASME B18.2.6-2006 standard for hexagonal bolt sizes [40], the standard size of the bolt at A was selected to be M12 × 1.5 × 75mm.

Again, equation (13) was used to calculate the diameter of the shank of the bolt at B, d_B , as follows:

$$d_B = \sqrt{\frac{2 \times 8430 \text{N}}{\pi \times 40 \text{N}/\text{mm}^2}} = 11.58 \text{mm}$$

However, from the ASME B18.2.6-2006 standard for hexagonal bolt sizes [40], the standard size of the bolt at B was selected to be M12 × 1.5 × 75mm. Therefore, the size of the bolt at A is the same as that of the bolt at B, this reduces variety and facilitates interchangeability of parts. The dimensions of the hexagonal bolt are shown in Fig. 7.

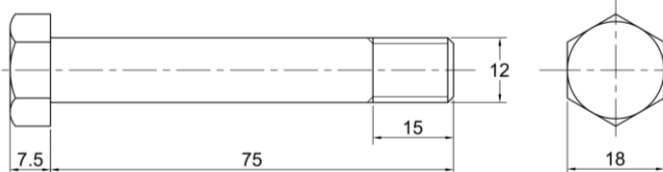


Fig. 7. Dimensions of the hexagonal bolts at A and B in mm.

Equation (14) was used to calculate the height or thickness, h , of the standard nut to be used with the bolts at A and B as follows:

$$h = 0.8 \times 12 \text{mm} = 9.6 \text{mm}$$

However, from the ASME B18.2.6-2006 standard for hexagonal nut sizes [40], the standard size of nuts to be used with the bolts at A and B is M12 × 1.5 with a height or thickness of 10mm. The hexagonal nut is shown in Fig. 8.

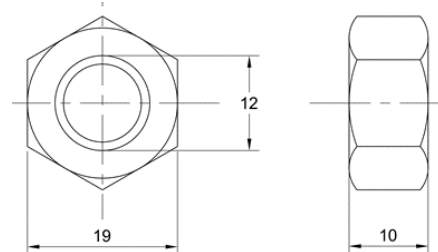


Fig. 8. Dimensions of the hexagonal nuts at A and B in mm.

9. Determining the Mechanical Advantage of the Mount/demount Arm and Pry Bar Assembly

Equation (3) was used to calculate the mechanical advantage, MA_M , of the mount/demount arm and pry bar assembly as follows:

$$\text{Mechanical Advantage, } MA_M = \frac{1045}{200} = 5.22$$

10. Determining the Load, R_E

By taking moments about the fulcrum, D, the load R_E on the mount/demount arm and pry bar assembly was calculated from (15) as follows:

$$R_E = \frac{733 \text{N} \times 1045 \text{mm}}{200 \text{mm}} = 3830 \text{N}$$

11. Determining the Reaction at the Fulcrum, R_D

The reaction at the fulcrum, R_D , was also determined from (16) as follows:

$$R_D = 3830 \text{N} - 733 \text{N} = 3097 \text{N}$$

12. Developing the Shear Force Diagram of the Mount/demount Arm and Pry Bar Assembly

The shear force diagram of the mount/demount arm and pry bar assembly is shown in Fig. 9, and the values were calculated as follows:

$$F_D = -R_D = -3097 \text{N}$$

$$F_E = -3097 \text{N} + 3830 \text{N} = +733 \text{N}$$

$$F_F = +733 \text{N} - 733 \text{N} = 0$$



Fig. 9. Shear force diagram of the mount/demount arm and pry bar assembly.

13. Developing the Bending Moment Diagram of the Mount/demount Arm and Pry Bar Assembly

The bending moment diagram of the mount/demount arm and pry bar assembly is shown in Fig. 10, and the values were calculated as follows:

$$M_D = 0$$

$$M_E = -R_F \times l_1$$

$$M_E = -3097N \times 200mm = -619400N \cdot mm$$

$$M_F = 0$$

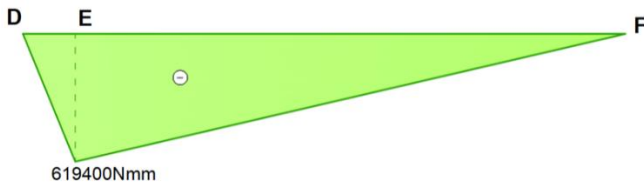


Fig. 10. Bending moment diagram of the mount/demount arm and pry bar assembly.

The load, shear force and bending moment values of the mount/demount arm and pry bar assembly are listed in Table III.

TABLE III: LOAD, SHEAR FORCE AND BENDING MOMENT VALUES AT POINTS ON THE MOUNT/DEMOUNT ARM AND PRY BAR ASSEMBLY

Points	Distance from Fulcrum (mm)	Load (N)	Shear Force (N)	Bending moment (N·mm)
D	0	3097	-3097	0
E	200	3830	733	-619400
F	1045	733	0	0

It was necessary to determine the appropriate size of the mount/demount arm considering the forces acting at various points on the tube, based on stresses due to bending moment, as well as stresses due to torsional moment.

14. Determining the Tube Size of the Mount/demount Arm based on Stresses due to Bending Moment

From Fig.10, the maximum bending moment occurs at point E. Equation (23) was used to calculate the length of a side of the inner square section of the mount/demount arm, b_i , as follows:

$$b_i = \left(\frac{6.7064 \times 619400N \cdot mm}{80N/mm^2} \right)^{\frac{1}{3}} = 37.31mm$$

The length of a side of the outer square section, b_o , was calculated from (22) as follows:

$$b_o = 1.2 \times 37.31mm = 44.77mm$$

Next, the tube thickness of the mount/demount arm based on bending stress, t_{MB} , was calculated from (24) as follows:

$$\text{Tube wall thickness, } t_{MB} = \frac{44.77mm - 37.31mm}{2} = 3.73mm$$

From ASTM A500 tables for standard square tube sizes [41], the standard square tube to be used for the mount/demount arm of the tyre changing tool in order for

the arm to withstand stresses due to bending moment is a 50mm × 50mm × 4mm square tube.

15. Determining the Tube Size of the Mount/demount Arm based on Stresses due to Torsional Moment

The tube size of the mount/demount arm was also determined based on stresses due to torsional moment. Equation (25) was used to determine the torque setup in the mount/demount arm as follows:

$$T = 3830N \times 300mm = 1149000N \cdot mm$$

Equation (30) was used in determining the length of a side of the inner square section of the mount/demount arm based on torsional stress, l_i , as follows:

$$l_i = \left(\frac{1149000N \cdot mm}{0.242 \times 40N/mm^2} \right)^{\frac{1}{3}} = 49.15mm$$

Equation (27) was used to determine the length of a side of the outer square section of the mount/demount arm based on torsional stress, l_o , as follows:

$$l_o = 1.2 \times 49.15mm = 58.98mm$$

Also, the wall thickness of the mount/demount arm based on torsional stress, t_{MT} , was calculated from (28) as follows:

$$t_{MT} = 0.1 \times 49.15mm = 4.92mm$$

Finally, the length of a side of the square section bounded by the centre line, l_c , was calculated from (29) as follows:

$$l_c = 1.1 \times 49.15mm = 54.07mm$$

From ASTM A500 tables for standard square tube sizes [41], the standard square tube to be used for the mount/demount arm of the tyre changing tool in order for the arm to withstand stresses due to torsional moment is a 60mm × 60mm × 5mm square tube.

16. Determining the Angle of Twist of the Mount/demount Arm

The angle of twist of the 50mm × 50mm × 4mm square tube while mounting or demounting a tyre on a 16" × 7" rim was determined using (32) as follows:

$$\phi_{50} = \frac{1149000N \cdot mm \times 200mm \times 184mm}{4 \times (46mm \times 46mm)^2 \times 77 \times 10^9 N/mm^2 \times 4mm}$$

$$\phi_{50} = 7.67 \times 10^{-9} rad$$

The angle of twist of the 60mm × 60mm × 5mm square tube, while mounting or demounting a tyre on a 16" × 7" rim was also determined using (32) as follows:

$$\phi_{60} = \frac{1149000N \cdot mm \times 200mm \times 220mm}{4 \times (55mm \times 55mm)^2 \times 77 \times 10^9 N/mm^2 \times 5mm}$$

$$\phi_{60} = 3.59 \times 10^{-9} rad$$

ϕ_{60} is less than ϕ_{50} , therefore, the 60mm × 60mm × 5mm square tube was selected as the appropriate size of the mount/demount arm. The results are presented in Table IV.

TABLE IV: DESIGN RESULTS

Parameter	Value	Unit
Allowable bending stress, σ_b	80	N/mm ²
Allowable shear stress, τ	40	N/mm ²
Mechanical Advantage of bead breaker arm, MA_B	11.5	-
Outer radius of bead breaker arm, r_o	28.7	mm
Inner radius of bead breaker arm, r_i	23.92	mm
Wall thickness of bead breaker arm, t_B	4.78	mm
Shank diameter of bolt at A, d_A	11.07	mm
Shank diameter of bolt at B, d_B	11.58	mm
Height/Thickness of nut at A and B, h	9.6	mm
Mechanical Advantage of mount/demount arm, MA_M	5.22	-
Length of a side of the outer square section of mount/demount arm based on bending stress, b_o	44.77	mm
Length of a side of the inner square section of mount/demount arm based on bending stress, b_i	37.31	mm
Wall thickness of mount/demount arm based on bending stress, t_{MB}	3.73	mm
Torque on the mount/demount arm	1149000	N·mm
Length of a side of the outer square section of mount/demount arm based on torsional stress, l_o	58.98	mm
Length of a side of the inner square section of mount/demount arm based on torsional stress, l_i	49.15	mm
Wall thickness of mount/demount arm based on torsional stress, t_{MT}	4.92	mm
Length of centre line of mount/demount arm, s	220	mm
Area bounded by centre line of mount/demount arm, α	3,025	mm ²
Angle of twist of a 50mm × 50mm × 4mm mount/demount arm while mounting/demounting a tyre on 16" × 7" rim, ϕ_{50}	7.67×10^{-9}	rad
Angle of twist of a 60mm × 60mm × 5mm mount/demount arm while mounting/demounting a tyre on 16" × 7" rim, ϕ_{60}	3.59×10^{-9}	rad

The isometric projection of the portable manual tyre changer is shown in Fig. 11. The orthographic projection of the tool is shown in Fig. 12, while the assembly drawing and bill of materials are shown in Fig. 13.

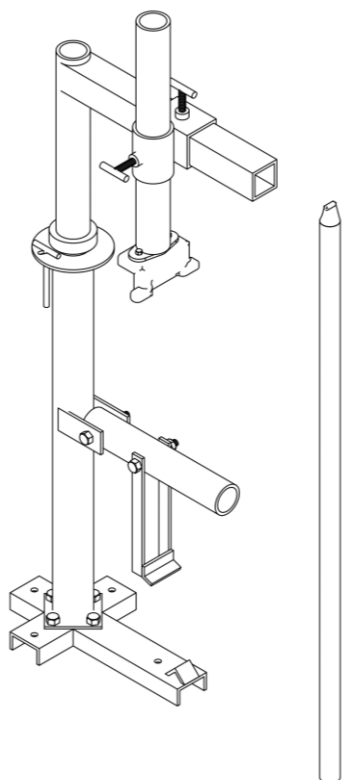


Fig. 11. Isometric projection of the tyre changer.

The tool is efficient, relatively affordable, light-weight and can be easily dismantled and its parts stored away. The mode of operation of the tool is described as follows:

1. Bead-breaking Operation

To replace a tyre on a wheel rim with the tyre changing tool, the tyre is first deflated and placed flat on the base of the bead breaker section of the tool. The bead breaker shoe is then placed on the sidewall of the tyre near the edge of the rim, and a downward force is applied at the free end of the bead breaker arm and pry bar assembly. The tyre-rim assembly is then rotated through 180° about the vertical axis and the bead breaker shoe is again placed on the sidewall, close to the edge of the rim, and a downward force is applied at the free end of the bead breaker arm and pry bar assembly. The tyre is then flipped over and the bead-breaking process is repeated for the underside of the tyre-rim assembly.

2. Demounting a Tyre from a Wheel Rim

After carrying out the bead-breaking operation, the tyre-rim assembly is placed on the mount/demount section of the tool, centred with the aid of the locking pin and the centring post and fastened with the aid of the centring cone and the centring post sleeve. The area where the edge of the wheel rim meets the tyre is properly lubricated to allow for easy demounting of the tyre. Next, the duckhead is moved into position, between the rim flange and the tyre bead. The mount/demount arm and pry bar assembly are then rotated clockwise or anticlockwise until the tyre bead completely lifts off from the wheel rim. The process is repeated for the underside of the tyre-rim assembly, the tyre can then be easily separated from the wheel rim.

3. Mounting a Tyre on a Wheel Rim

The wheel rim is placed on the mount/demount section of the tool, centred with the aid of the locking pin and the centring post and fastened with the aid of the centring cone and the centring post sleeve. The tyre bead edge is properly lubricated to allow easy mounting of the tyre on the wheel rim. Next, the duckhead is moved into position, between the rim flange and the tyre bead. The mount/demount arm and pry bar assembly are then rotated clockwise or anticlockwise until the flange of the wheel rim is completely above the tyre bead. The process is repeated for the underside of the tyre-rim assembly, until the tyre bead is completely above the flange of the wheel rim.

V. CONCLUSION AND RECOMMENDATIONS

The design of a portable manual automobile tyre changer has been developed in this paper. Loads, stresses, shear forces, bending moments and torsional moments were considered in selecting the component parts of the tool. The bill of materials in Fig. 13 shows that it costs only ₦21,000 to purchase the materials needed to fabricate the tool. Therefore, the design provides an alternative portable and relatively affordable tyre changing tool that can be afforded by tyre technicians across Nigeria, and other developing or underdeveloped countries.

Obtaining a higher mechanical advantage of the mount/demount arm and the bead breaker arm of the tyre

changing tool will improve the overall effectiveness of the tool. Thus, if the tool is installed in an area with ample space, the overall mechanical advantage of the manual tyre changer can be further increased by increasing the length of the effort arm or pry bar. However, if there are space constraints, compound levers should be utilized instead.

Therefore, it is recommended that more designs of tyre changers that utilize compound levers could be developed, in order to produce a cost-effective tyre changing tool that possesses an improved overall mechanical advantage.

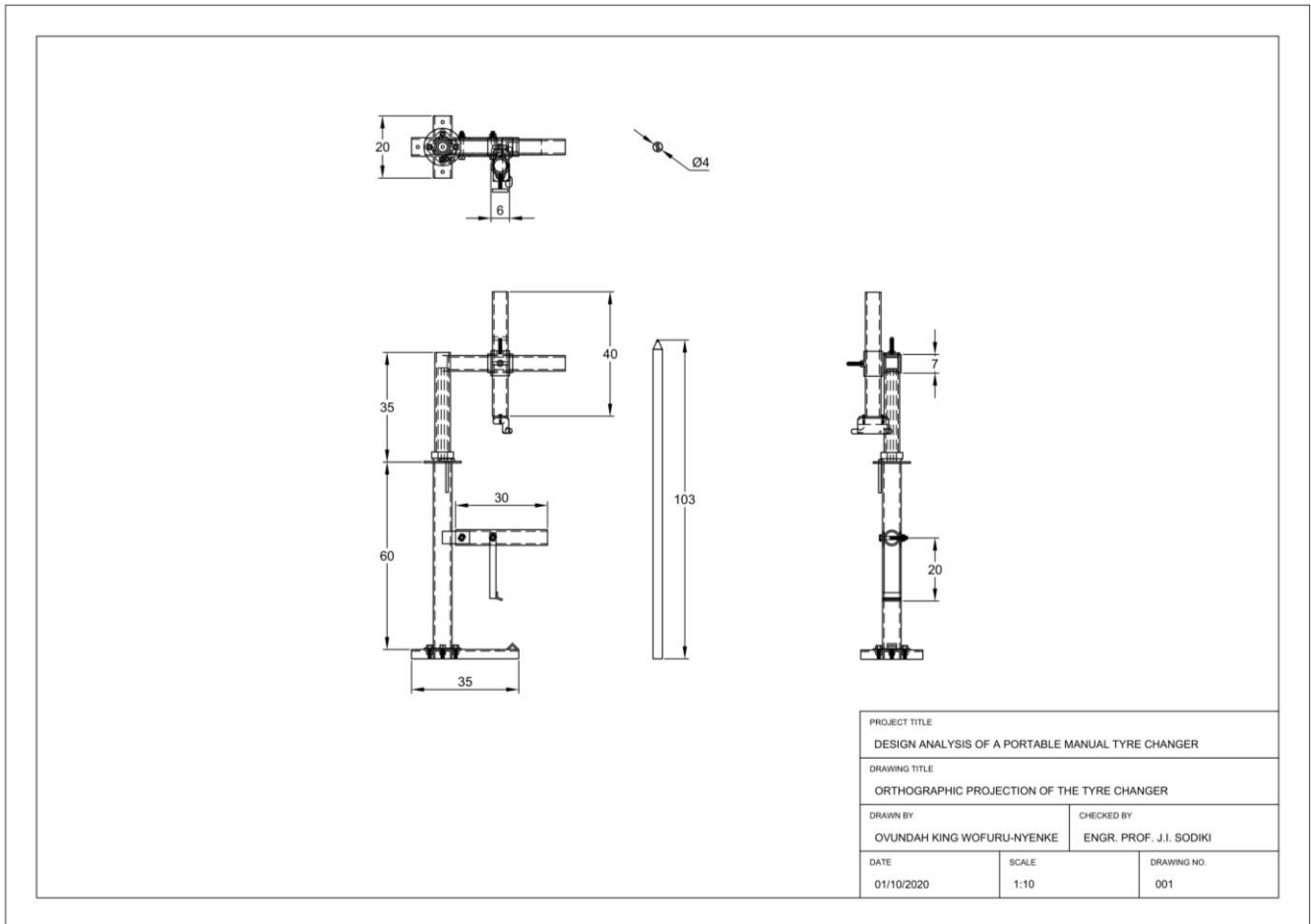


Fig. 12. Orthographic projection of the portable manual tyre changer.

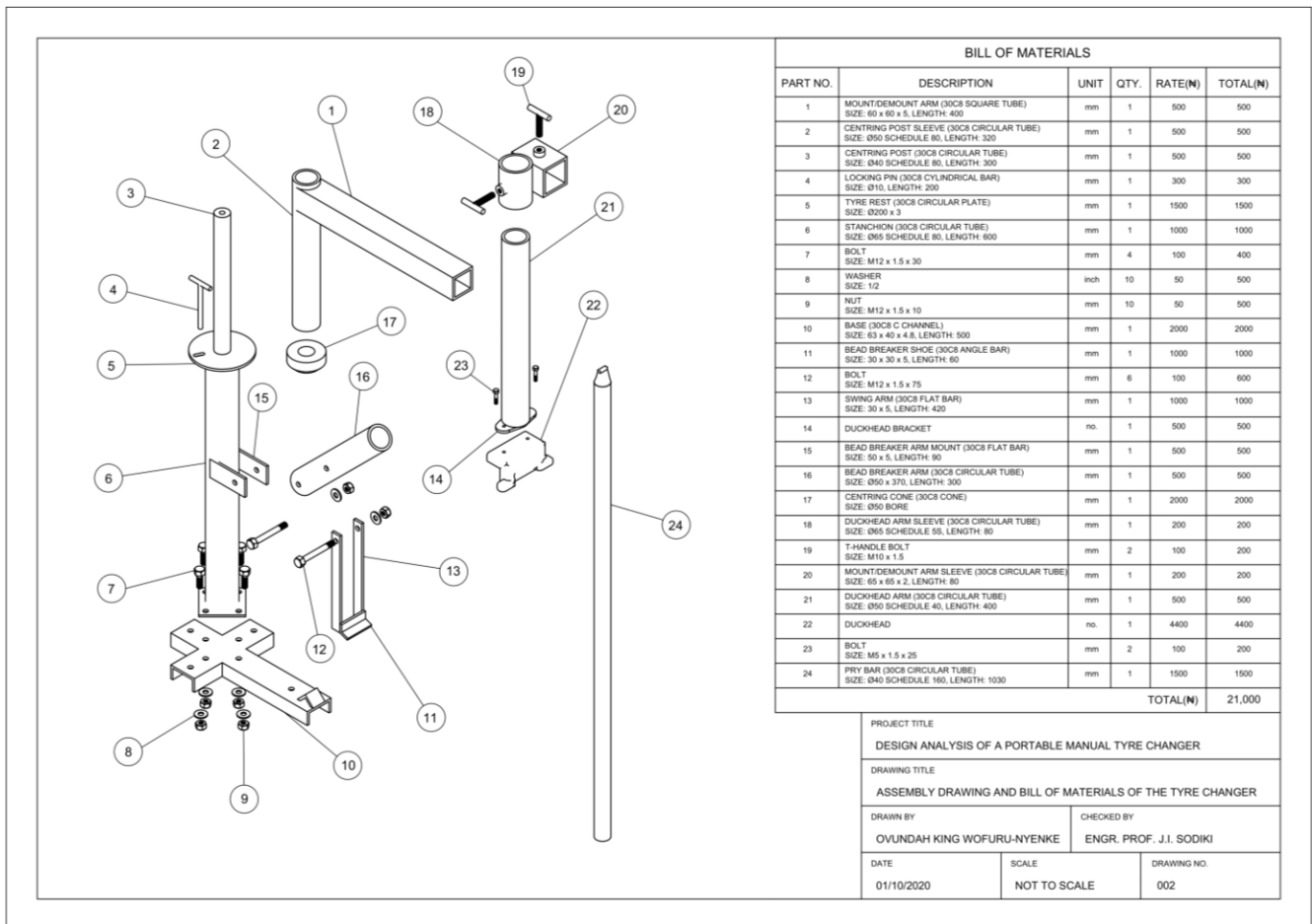


Fig. 13. Assembly drawing and bill of materials of the portable manual tyre changer.

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