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# Design and Stress Analysis of Tow Bar for Medium Sized Portable Compressors

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**Abstract**

In this paper, design and analysis of a tow bar for a portable compressor unit has been presented. The design is limited to medium sized portable compressors. The key features of this project are selection of ideal length and offset of tow bar, calculation of various dimensions of tow bar including the mounting accessories and pulling tongue, evaluating the actual results by finite element analysis for different prototypes using FEA software. The final design is selected based on design as well as FEA results. The parameters considered for selecting the design are maximum equivalent stress value, ease of fabrication, material saving and factor of safety.

***Keywords:*** *Tow bar, tongue, design, FEA model, stress analysis.*

## 1. Introduction

The need for portability of equipment has increased manifold times today. In today's scenario due to competition it is essential to have equipment with lowest cost of design, production and operation. Design of the equipment plays a vital role; hence the handling and utilization of the equipment should be quicker and user-friendly. In case of portable compressors, tow bar acts as a connecting link between the compressor unit and the towing vehicle. The tow bar does a couple of important jobs apart from being the part that keeps the portable unit attached and at a distance from your tow vehicle. The tow bar keeps the portable unit in balance when towing and assists in keeping weight on the tow bar which is required for controlled towing. It also adds stiffness to the chassis and depending on the axle and load placement will assist in keeping the trailer running true and stable.

The tow bar does a lot of work while towing. Every little bump in the road and every turn you make transfers stress through the tow bar and compresses, twists and stretches the tow bar material constantly. If the trailer has been built with an undersized tow bar or the trailer is constantly overloaded or unbalanced, this repeated loading and unloading (cycling) of stresses on the tow bar can create microscopic cracks within the grain structure of the tow bar material. Tow bars are subjected to both constant fatigue loading and heavy application specific loads, with each putting different demands on the structure and steel used in the tow bar construction. It is vital that the correct steel grade and section type is used to ensure that each part will be able to endure the load it will experience. The type of material to be used for tow bar depends on its application requirements and operating conditions. The material is decided on the basis of its rigidity, strength, cost, durability and reliability. The material selected should also be easy to fabricate. Fabrication plays an important role in material selection. As strength of material increases, cost of fabrication increases. The commonly used tow bars are shown below.

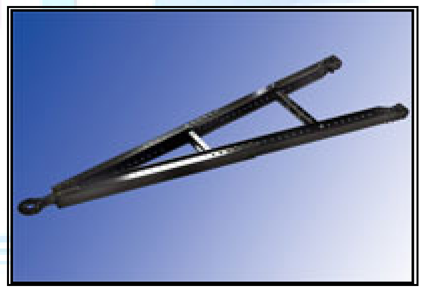


Fig. 1: A-frame tow bar

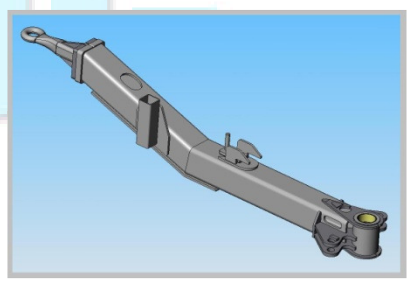


Fig. 2: Single frame tow bar

In the present paper, design and stress analysis of a tow bar for medium sized portable compressor has been discussed in detail.

## 2. Literature review

A very little research on cross section and shape of tow bar has been reported in the earlier literature. A tow bar helps in transporting trailers, industrial equipment from one place to another. A tow hitch (or tow bar) is attached to the chassis of a vehicle for towing. A tow bar needs to have good vertical and sideways strength and depending on the size of your trailer and the loads you wish to carry, this will determine the size of the material required and the design that you should use. The cross section of tow bar can be circular, square or rectangular. Square and rectangular sections make the structure more robust, however, it increases the stress concentration at edges and is likely to fail after a certain number of stress cycles. Circular section gives uniform stress concentration throughout, however, it has lesser load pulling capacity as compared to the other sections. The tow bar frame can be a simple curved rod type structure or a rigid A-frame type structure. These are the two most widely used tow bar frames in the industry, with both having their advantages and disadvantages. In the case of rigid A-frame tow bars, the tow bar is mounted on the frame of the towed unit, with a receiver mounted under the frame of the car. The A-frame tow bar is a solid, welded tow bar is bulky. The overall cost is more as more material is needed..

The single frame tow bars are used for light and medium weight units. The main advantage of this frame is its low price and weight. Also the structure is compact and easy to fit. Mostly the cross section is hollow. Accessories like safety chains and stand can be mounted on this type of frame.

## 3. Design philosophy

The design calculations are based on application need. Suitable design considerations are made before starting with the actual work. These considerations are assumed to be the same throughout the calculations. Based on these considerations, the various parameters of the tow bar are evaluated. This includes selecting the type of tow bar as well as its frame, the length and the vertical offset of the tow bar and the type of tongue to be attached to it. This too is selected from application need, cost and practical conditions point of view. For medium sized portable units the weight is usually less than three ton. In this case, the mass of portable unit assembly is taken as 1395kg.

The following design considerations have been taken.

a. The entire assembly including the tow bar, the towed vehicle and the towing vehicle is moving with constant acceleration.
b. The tow bar is acted upon by a pulling force of constant magnitude in horizontal direction only.
c. The pulling force acts along the inner semi-circumferential area of the tow bar tongue and is uniformly distributed about the same.
d. All the welded joints are strong enough to withstand the load and will not fail under operating conditions.
e. The static coefficient of friction acting between the road-tyre surface has a constant value. This value is taken on the higher side for design safety.

The following material properties are selected for normal steel.

Table 1: Material properties

| ***Property*** | ***Value*** |
|---|---|
| *Young's Modulus* | *210000 MPa* |
| *Shear Modulus* | *7900 MPa* |
| *Density* | *7800 kg/m$^3$* |
| *Yield limit* | *240 MPa* |
| *Ultimate limit* | *360 MPa* |

### 3.1 Determination of tow bar length and offset distance

The overall length of the tow bar plays a very important role in design calculations. The maximum length is restricted to 3.5m by Australian standards while the minimum is determined by the criteria of maximum dead weight on the tow bar tongue. This maximum dead weight on the tow bar tongue should not exceed 10% of the total portable assembly weight. In this case the lower limit for overall tow bar length comes to be 1.275m. For this case, because of physical constraints and application demand the length is taken as 1750mm.

Offset distance is the vertical distance between the front bracket mounting and the towbar tongue mounting. In other words, it is the relative distance between the chassis of towed vehicle to the clipping device of towing vehicle. The chassis height of the portable unit is 523.5mm and normally the tow ball or the clipping device is mounted on the towing vehicle at a height of 350mm to 750mm. For safety and ease of fabrication the value of vertical offset distance is taken as 150mm.

### 3.2 Force subjection on tow bar assembly

The tow bar assembly including the portable unit is subjected to various forces. These forces are calculated considering two separate conditions, namely, static and kinematic. Under static conditions, the forces acting are the self-weight of portable unit in vertically downward direction, the normal reaction between the road tyre surface in vertical direction, the static nose dead weight acting on tow bar tongue in vertical direction and the frictional force between road and tyre surface. Under kinematic conditions, the forces acting are the self-weight of portable unit in vertically downward direction, the normal reaction between the road tyre surface in vertical direction, the horizontal pulling force acting on tow bar tongue and the frictional force between road and tyre surface acting in horizontal direction opposite to pulling force.

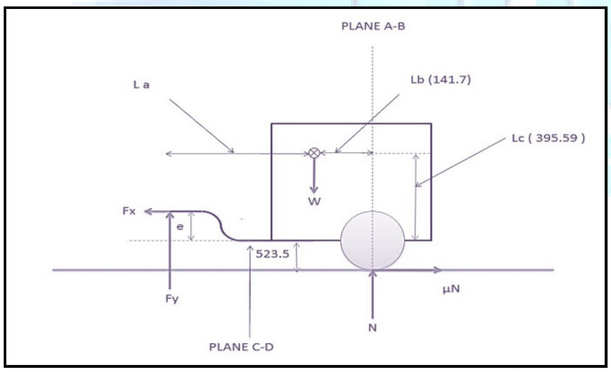



Fig. 3: Forces acting on tow bar assembly

Calculation of vertical force on tow bar tongue (Eye):

Assumption: Entire assembly is at rest. (Static Conditions)
Taking moment about C.G,

$F_y \times l_a = N \times l_b$
$F_y \times 1750 = N \times 142$
$\therefore F_y = 0.08114N$ (1)
Also,

$\sum F_y = 0$ (Equilibrium of forces in vertical direction)
$\therefore F_y + N - W = 0$
Where
$W = m \times g$
$W = 1395 \times 9.81$
$W = 13684.95$ N
$\therefore F_y + N - 13684.95 = 0$
$\therefore F_y + N = 13684.95$ (2)

On solving (1) & (2),

$N = 12659.86$ N
$F_y = 1025.08$ N

Calculation of horizontal force on tow bar tongue (Eye):

$\sum F_x = 0$ (Equilibrium of forces in horizontal direction)
$F_x - \mu.W = 0$
Where
$\mu$ - Static friction coefficient
$\mu = 0.9$ [4]
$\therefore F_x = \mu.W$
$F_x = 0.9 \times 13684.95$
$F_x = 12316.45$ N
$\therefore F_x \cong 12316$ N

### 3.3 Calculation of tow bar dimensions

Forces under motion-
Taking moment about C.G of portable compressor unit.
Assumptions:

1. Motion is under constant acceleration.
2. Only Fx is acting on the towbar tongue.

Bending Moment = Fx × 258
Bending Moment = 12316 × 258
Bending Moment = 3177528 N-mm
Now,

$$\text{Section Modulus } (Z) = \frac{\text{Bending Moment}}{\mathit{Allowable\ Stress}} \quad [7]$$

$$= \frac{3177528}{240}$$
$$= 13239.7\ mm^3$$
$$= 13.2397\ cm^3$$

Corresponding cross-section dimensions from table,

Circular hollow section:
Outer diameter ($D_o$) = 76.1 mm and thickness (t) = 4 mm

Square hollow section:
Side (a) = 60 mm and thickness (t) = 4 mm

Rectangular hollow section:
Length (l) = 90 mm, Breadth (b) = 50 mm and thickness (t) = 3.6 mm

3.4 Design of tow bar tongue

Assumptions:

1. Diameter of pin (d) is taken as 50 mm [2]
2. Factor of safety = 5 [2]
3. Overall length of the tongue = 2d [2]
4. Width of the tongue = 2d [2]

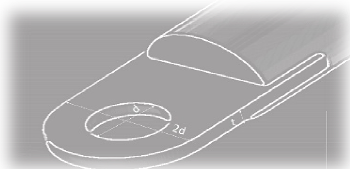


Fig. 4: Sketch of tongue

Now,

$$\text{Allowable tensile stress } (\sigma_a) = \frac{\text{Permissible Stress}(\sigma)}{\text{Factor of safety(FOS)}}$$

$$\text{Allowable tensile stress } (\sigma_a) = \frac{240}{5}$$

$$\text{Allowable tensile stress } (\sigma_a) = 48\ \text{N/mm}^2$$

$$\text{Allowable shear stress } (\sigma_s) = \frac{\text{Allowable tensile Stress}(\sigma_a)}{2} \quad [7]$$

$$\text{Allowable shear stress } (\sigma_s) = 24\ \text{N/mm}^2$$

Calculation of tongue thickness (t):
Calculations based on failure in tension:

$$\text{Allowable tensile stress } (\sigma_a) = \frac{\text{Force}}{\text{Area}}$$

$$80 = \frac{Fx}{(2d-d)\times t}$$

$$t = \frac{12316}{(d)\times 48}$$

$$t = \frac{12316}{(1\times 50)\times 48}$$

$$t = 5.13\ \text{mm}$$

$$\therefore t \cong 5\ \text{mm}$$

Calculations based on failure in shear:

Assumption:
Area in shear failure is taken as 0.8 times the area in tensile failure.[2]

$$\text{Allowable shear stress } (\sigma_s) = \frac{\text{Force}}{\text{Area}}$$

$$40 = \frac{Fx}{(2d-d)\times 0.8\times t}$$

$$t = \frac{12316}{(d)\times 0.8\times 24}$$

$$t = \frac{12316}{(1\times 50)\times 0.8\times 24}$$

$$t = 12.829\text{mm}$$

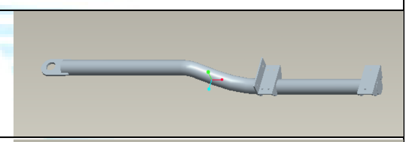
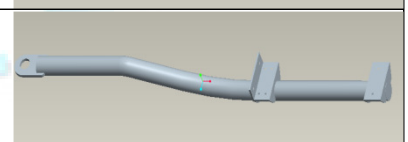
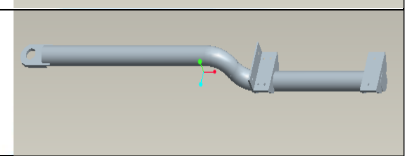
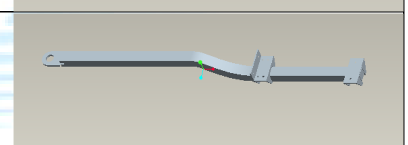
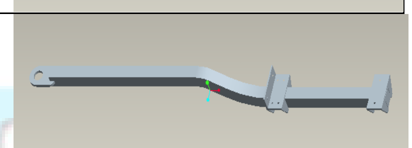
$$\therefore t \cong 13\ \text{mm}$$

Consider higher value of thickness for design,

$$\therefore t = 13\ \text{mm}$$

3.5 Proposed 3-D models

Table 2: Figure of 3D models of tow bar

| *Sr. No* | *Type of cross-section* | *3D-model as per calculations* |
|---|---|---|
| *1* | *Ideal Circular* |  |
| *2* | *Increased Curvature* |  |
| *3* | *Decreased Curvature* |  |
| *4* | *Rectangular* |  |
| *5* | *Square* |  |

## 4. Stress analysis

The tow bar was designed using conventional design equations. However, to check the validity of these calculations, finite element analysis was performed and conventional calculated results are found in good agreement with the FEA results. Stress analysis for hollow circular, hollow rectangular and hollow square cross sections was performed. However the analysis of hollow circular section has been presented in this paper as safer results are obtained as compared to the results of other cross section analysis.

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## 4.1 Stress analysis for ideal curvature

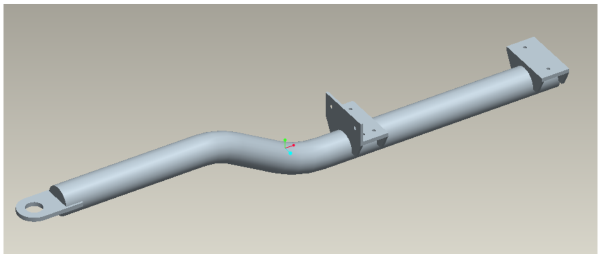

Fig. 5: Tow bar assembly

The towbar assembly was made in assembly mode of Pro-e by combining three parts namely towbar, front bracket and rear bracket. The towbar tongue was considered as an integral part of towbar and was made as a single unit. The vertical offset between the two ends was taken as 150mm. The curvature was given an angle of inclination of 30°. This inclination gives lower value of localized stress concentration near bends. As per the calculations, the outer diameter of circular cross section was taken as 76.1mm and thickness, 4mm.

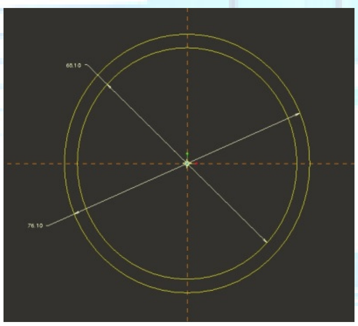

Fig. 6: Cross section of towbar

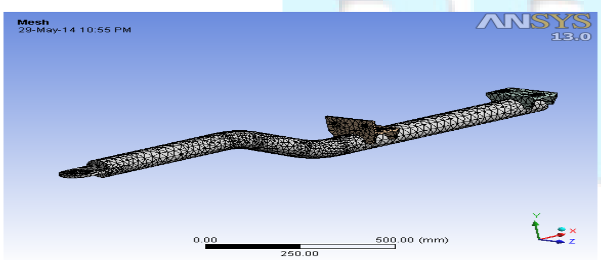


Fig. 7: Meshing of towbar assembly

Fine meshing was carried out with default element size. Fast transition mode was selected with high smoothing and fine span angle center. Minimum edge length was taken as 5mm.

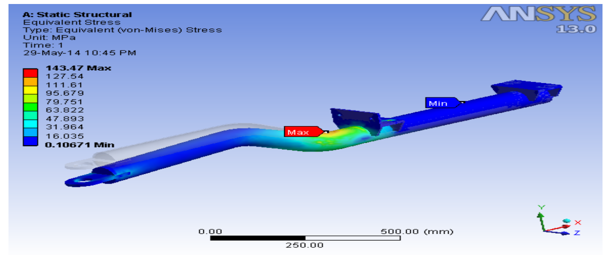


Fig. 8: Equivalent stress (Von-Mises) in Towbar assembly (Circular cross section)

The maximum stress was induced near the rear bend of the towbar. The stress was localized having maximum value of 143.47MPa. The average stress value was safely below 120Mpa. So this design is safe.

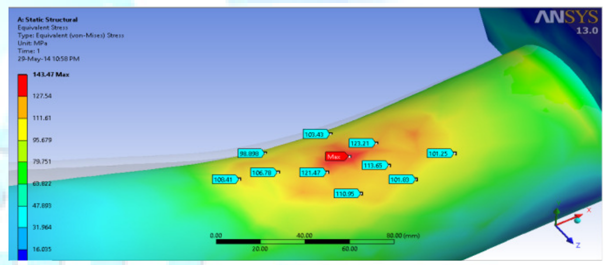

Fig. 9: Stress concentration around maximum equivalent stress (Von-Mises) area (circular cross section)

The localized stress concentration shows the variation of stress around the maximum stress induced region. It is well below the 130Mpa and can be considered safe.

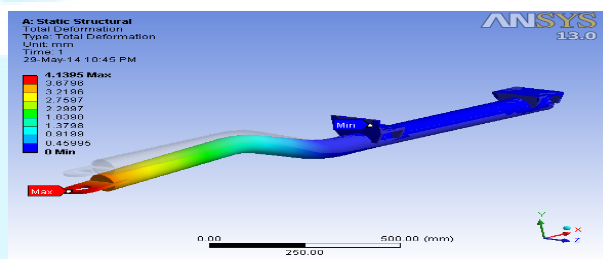


Fig. 10: Total deformation in towbar assembly (Circular cross section)

The total deformation shows the deflection of the towbar under the action of loads. In this case we get a deformation of 4.1395mm. However under practical operating conditions, this deformation will be much less as the towball will restrict its motion.

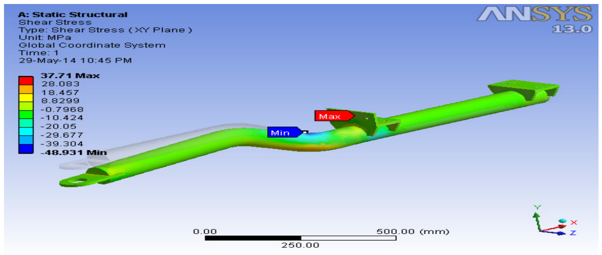


Fig. 11: Shear stress in towbar assembly (Circular cross section)

The maximum shear stress was induced near the front bracket of the towbar. The stress was localized having maximum value of 37.71MPa. The average stress value was well below limiting stress value of 80Mpa. So this design is safe.

### 4.2 Results of stress analysis of the tow bar for three cross sections

Table 3: Results of FEA

| Sr. No | Type of cross section | | Equivalent stress (MPa) | Total Deflection (mm) | Shear stress (MPa) |
|---|---|---|---|---|---|
| 1 | Circular | Ideal Curvature | 143.47 | 4.1395 | 37.71 |
| | | Increased Curvature | 166.49 | 3.6658 | 50.83 |
| | | Reduced Curvature | 220.33 | 6.0013 | 72.19 |
| 2 | Rectangular | | 282.45 | 9.8347 | 75.82 |
| 3 | Square | | 190.15 | 4.9555 | 63.30 |

From the result table 3, it can be seen that circular cross section gives the minimum value of equivalent stress. Also the values of total deformation and directional deformation are comparatively less. These values are below the allowable limit and can be considered. Thus, this has verified the design calculations and has found the analysis in line with the design calculations. From the results tabulated in the above table, the ideal circular cross section dimensions are safely chosen as final towbar dimensions.

The required factor of safety is calculated by dividing the yield stress value of material (240 MPa) by the equivalent stress of the particular design obtained from the FEA analysis. The table below clearly shows that the factor of safety is highest for the circular cross-section. Thus ideal circular cross-section is chosen as the design for the towbar

Table 4: Factor of safety for various design cross-sections

| Sr. No | Type of cross section | | Factor of safety |
|---|---|---|---|
| 1 | Circular | Ideal curvature | 1.6725 |
| | | Increased Curvature | 1.4415 |
| | | Reduced curvature | 1.089 |
| 2 | Rectangular | | 0.8496 |
| 3 | Square | | 1.2615 |

## 5. Conclusion

From design and manufacturing point of view, circular hollow cross-section is found safer and suitable for the tow bar of the portable compressor.

## Acknowledgement

We express our deep sense of gratitude to our guide and Prof. Amar Pandhare, Head of Mechanical Engineering Department for their valuable support.
Also, we thank to all the teaching and non-teaching staff for their cooperation.

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