

# Design and Fabrication of a Detachable Motorcycle Trailer Combination

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## ABSTRACT

The study was aimed at designing and fabricating a motorcycle trailer capable of carrying 100 kg of load. The design scope was that the trailer must be detachable, the drag bar or hitch should be designed in such a way to that it should have freedom of motion for vertical, horizontal and tilting movement. A motorcycle trailer was designed and fabricated with mild steel plate, angle bar and rod. The weight of the trailer was 50kg, giving a gross weight of 150kg for trailer and load.

Experienced motorcyclists tested the trailer driven with 125 cc Daylong Motorcycle to assess its performance. The results showed that the trailer cruised well for the designed load and the fabricated hitch provided easy attachment between the trailer and the motorcycle. The trailer towed best in the presence of a limited amount of load of less than 60 kg but had less cruising effect when the trailer was empty. In the case of driving through a bend, the trailer could perform well at not-so-constrained zone of radius not less than 900 mm. The design and fabrication of a prototype trailer was satisfactorily achieved in this study.

(Keywords: motorcycle, trailer, drawbar pull, tongue, hitch rod).

## INTRODUCTION

Motorcycles are one of the most affordable forms of motorized transport in many parts of the world and are also the most common type of motor vehicle because of their easy accessibility to most local communities (Nakata, 2008; Ibitoye, et al., 2018). Commercial motorcycling in Nigeria started in Benin City in 1988, as a result of the crippling unemployment brought about by Federal Government Structural Adjustment Program

(SAP). Owing to the recent ban on commercial motorcycling in some Nigerian cities (Olubomehin, 2012; Asekhome and Oisamoje, 2013), an alternative use of the motorcycles that could make it useful to rural dwellers was developed.

Ordinarily, motorcycles, by design, are unable to carry many luggage or cargo, hence the need to increase the loading capacity with equitable distribution of its weight. This necessitated the introduction of trailer at the rear end on the motorcycle for the purpose. A trailer is generally an unpowered vehicle pulled by a powered vehicle (tractor). The vehicle must have enough traction force that is commensurate with the load for proper trailering. Traction force is the force needed to simply move the powered vehicle with the trailer. This concept could be applied to so many other areas of use. It can be used for cargo trailers or cart for carrying loads or extra luggage and avoid over loading of the motorcycle. It can also be used to pull behind some farm accessories like a rake or reel mower.

A motorcycle trailer is either a trailer used to carry motorcycles or one to be pulled by a motorcycle. Different motorcycles are manufactured with specific loading capacity and an evaluation of this would give an idea of the kind of loading expected from a given design. Load capacity for a motorcycle is determined by several factors including displacement (engine size and by extension, power), chassis design, tire and suspension, and the way the motorcycle is designed.

## MATERIALS AND METHODS

The research methodology covers research design, selection of materials and manufacturing specification.

## Research Design

The research design consists of the design of the trailer, design of rod on the hitch and selection of the bearing.

## Design of the Trailer

In designing a pull-behind-trailer that will be towed by a motorcycle, the towing force of the motorcycle exerted at its coupler in the direction of motion of the coupling point can be calculated from motorcycle specifications. This towing force is the drawbar pull (DP) (Mercury, 2005). Some motorcycle specifications include:

Gross Vehicle Weight (GVW)  
Wheel Tire Size  
Max. Torque of Engine  
Max. Power of Engine

(a) The Drawbar Pull (DP) is given as:

$$DP = \frac{T \times R \times 1000}{r} - RR \quad (1)$$

where,  $DP$  = drawbar pull in (N)

$T$  = motor torque (Nm)

$R$  = overall gear reduction including both axle and transmission

$r$  = Rolling radius of loaded driving tyre (mm)

$RR$  = Road rolling resistance in (N)

The overall gear reduction ( $R$ ) (Webtec, 2008) can be determined as thus:

$$R = \frac{RPM \times r}{2651.51 \times KPH} \quad (2)$$

where,  $RPM$  = Engine speed in revolutions per minute

$KPH$  = Vehicle speed in km/hr.

The Road Rolling Resistance ( $RR$ ) can be determined by:

$$RR = \frac{GVW \times R}{100} \quad (3)$$

Where,  $GVW$  = Gross vehicle weight (kg).

In computing the drawbar pull for the design, the following standard values were used:

Motor torque ( $T$ ) = 7.6Nm (obtained from motorcycle catalog)

Rolling radius of loaded driving tyre ( $r$ ) = 304.8mm

Engine speed in revolutions per minute = 750rpm (obtained from motorcycle catalog)

Vehicle speed in km/hr = 10 (speed chosen for design),

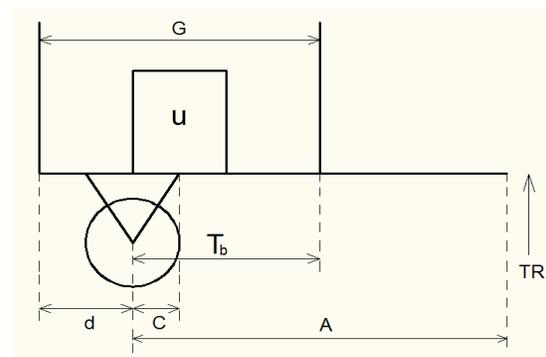
Rolling resistance ( $RR$ ) = 80N (Sand, level soft sand)

Gross vehicle weight,  $GVW$  = 190kg (mass of motorcycle plus mass of one passenger).

It was therefore observed that the drawbar pull is dependent on the overall gear reduction ratio. The speed can be varied to give it corresponding drawbar pull. Due to iteration of drawbar pull at different speeds, the excel spreadsheet was used to calculate the drawbar pull of the motorcycle at different speeds.

## Length of Trailer Tongue

Figure 1 shows the free body diagram of the lever representation of the trailer.



**Figure 1:** Lever Representation of the Trailer.

From Figure 1:

$U$  = the load

$TR$  = the load's reacting force on the bike

$T_b$  = the distance from the center of the wheel to the edge of the trailer towards the tongue

$C$  = the distance from the center of the wheel to the load's center of gravity

$A$  = the distance between the ground supports (center of wheel to the bike)

$d$  = wheel center from rear

$G$  = base length

$$U \times C = TR \times A \quad (4)$$

$$\text{Also, } C = \frac{G}{2} - d \quad (5)$$

$$A = \frac{U \times C}{TR} \quad (6)$$

Assuming, load is uniformly distributed about the trailer base and also the load's reaction on the motorcycle is about 10% of the trailer weight, and wheel center is located 40% from the rear of the base length,

Then, the reaction load on the bike ( $TR$ ) will be 10% of the load ( $U$ ).

But the length of trailer tongue ( $L_t$ ) is the difference between the distance between the ground support ( $A$ ) and the distance from the center of the wheel to the edge of the trailer towards the tongue ( $T_b$ ) as shown in Figure 1. That is:

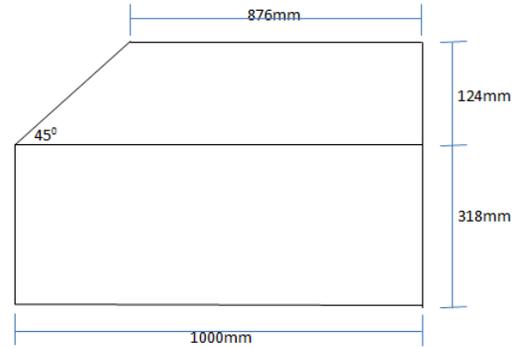
$$T_b = G - d \quad (7)$$

$$\therefore, \text{ length of trailer tongue, } L_t = A - T_b \quad (8)$$

### Design of the Carriage

The decision for the dimensions of the bucket was made with respect to the width of the rear of the motorcycle for the purpose of balancing and aerodynamics. For ease of determining the volume of the trailer carriage (or bucket), it was divided into two shapes of trapezium and rectangle as shown in Figure 2.

The total volume or capacity of the carriage ( $V_T$ ) was determined from the summation of the volume of the trapezium ( $V_t$ ) and volume of the rectangle ( $V_r$ ) in Figure 2.



**Figure 2:** The Side View of the Trailer Carriage.

The total volume or capacity of the carriage ( $V_T$ ) was determined from the summation of the volume of the trapezium ( $V_t$ ) and volume of the rectangle ( $V_r$ ) in Figure 2.

$$\text{i.e. } V_T = V_r + V_t \quad (9)$$

But, the volume of rectangle is:

$$(V_r) = l \times b \times h_r \quad (10)$$

$$\text{and volume of trapezium } (V_t) = \frac{1}{2} (\text{sum of parallel sides}) \times h_t \times b \quad (11)$$

where,  $l$  = length of trailer = 1000mm

$b$  = breadth of trailer = 635mm

$h$  = height of trailer both rectangle and trapezium = 442mm.

For balancing and aerodynamic considerations, the dimension used for the motorcycle width and length should be made to accommodate the effect of taking a bend by the carriage. The total volume or capacity of trailer bucket ( $V_t$ ) was determined to be  $2.75 \times 10^8 \text{ mm}^3$ .

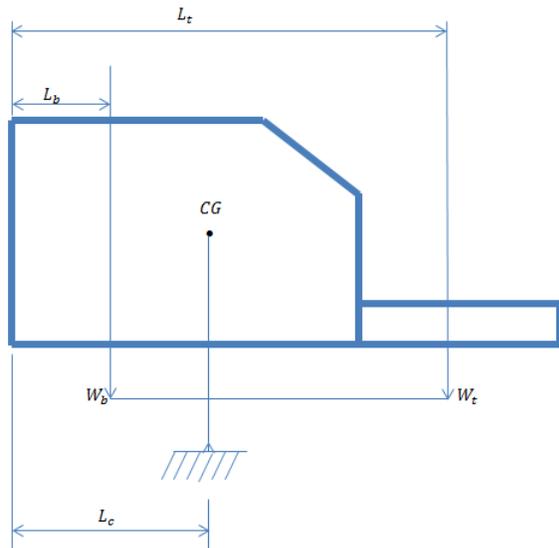
### Trailer Balance

In order to maintain the trailer balance, the location of center of gravity along the length of the trailer was considered. In estimating the center of gravity of a trailer, two perspectives were considered.

a) the weight of each part of the trailer, and

b) the center of gravity of each part of trailer.

So if we take moments about the rear of the trailer, we know that the combined effect of the two weights must be the same as all the weight concentrated at the center of gravity of the trailer as shown in Figure 3.



**Figure 3:** Free Body Diagram of Combined Effect of Weight on the Trailer.

From Figure 3, which showed the trailer's load distribution, the longitudinal center of gravity can be determined as:

$$(W_b + W_t) \times L_c = (W_b \times L_b) + (W_t \times L_t) \quad (11)$$

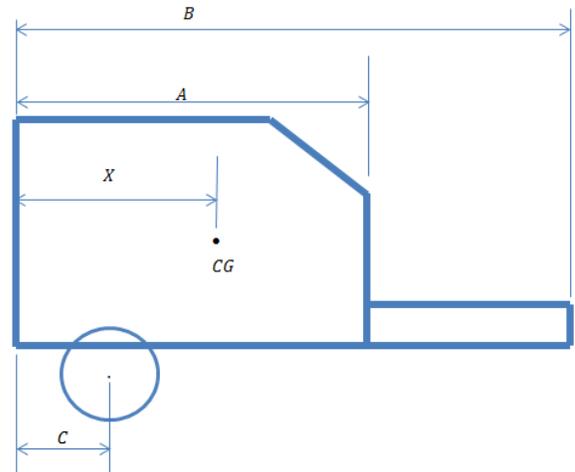
where,  $W_b$  = body weight of trailer

$W_t$  = weight of tongue

$L_c$  = length of center of gravity from the rear

$L_b$  = length of trailer

$L_t$  = length of tongue



**Figure 4:** Schematic Diagram showing the Trailer Balance Design.

The various weight distribution and distances from the rear are shown in Figure 4. The following dimensions were used:

- (i)  $A = 1000\text{mm}$  (i.e. body length from rear)
- (ii)  $B = 1450\text{mm}$  (i.e. hitch center from rear)
- (iii)  $C = 400\text{mm}$  (i.e. wheel center from rear).

In order to determine the body weight of the trailer ( $W_b$ ) the following parameters were assumed:

- (i) Basic trailer weight,  $W_{bt} = 500\text{N}$
- (ii) Weight of tongue, ( $W_t$ ) = 35N
- (iii) Weight of axle/suspension, ( $W_a$ ) = 145N

Then,  $W_b$  which is =  $W_{bt} - (W_t + W_a)$  was designed to be 320N.

From Figure 4, the following were determined:

For lever about rear,

$$\text{the tongue} = A + \frac{B-A}{2} = \frac{B+A}{2} \quad (12)$$

For lever about rear, axle/suspension = C For lever about rear, body weight,

$$W_b = \frac{A}{2} \quad (13)$$

Taking moment about rear,

$$\text{the tongue} = W_t \times \frac{B+A}{2} \quad (14)$$

Taking moment about rear,

$$\text{axle/suspension} = W_a \times C \quad (15)$$

Taking moment about rear,

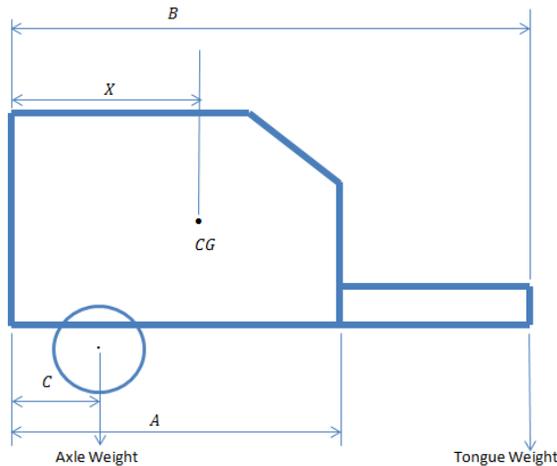
$$\text{the body weight, } W_b = W_b \times \frac{A}{2} \quad (16)$$

### Longitudinal Center of Gravity (LCG or X)

The longitudinal center of gravity (LCG) was determined as follows:

$$\text{LCG} = \frac{W_t \times \frac{B+A}{2} + W_a \times C + W_b \times \frac{A}{2}}{(W_t + W_a + W_b)} \quad (17)$$

Using Equation (17) the longitudinal center of gravity (LCG or X) was computed to be 521.8mm.



**Figure 5:** Weight Distribution on the Axle and Tongue.

To determine the total trailer weight, moment about the tongue was taken as shown in Figure 5.

$$W_{TR}(B - X) = W_A(B - C) \quad (18)$$

$$W_A = \frac{W_{TR}(B-X)}{(B-C)} \quad (19)$$

where,  $W_A$  = reaction of axle weight,

$W_{TR}$  = reaction of trailer weight,

$W_{TG}$  = reaction of tongue weight.

Using Equation (19) the reaction of axle weight ( $W_A$ ) was computed to be 442mm.

Therefore, weight reaction on tongue which is  $W_{TG} = W_{bt} - W_A$ , was determined to be 58N.

Recall that,  $B = 1450\text{mm}$ ,  $X$  or  $\text{LCG} = 521.8\text{mm}$  and  $C = 400\text{mm}$ . Equation (19) gave the weight on the axle ( $W_A$ ).

$$\therefore \text{Weight on each wheel} = \frac{W_A}{2} \quad (20)$$

Using Equation (20), weight on each wheel,  $W_h$  was calculated to be 221mm.

$$\therefore \text{The \% total trailer weight on tongue will be} = \frac{W_{TG}}{W_{TR}} \times 100$$

which was computed to be 11.6%, and the % of total trailer weight on axle =  $100 - W_{TG}$  was calculated to be 88.4%.

### Hitch Rod Design

The rod on the hitch was designed for, in terms of its principal stresses [i.e. maximum principal stress,  $s_n(max)$  and minimum principal stress,  $s_n(min)$ ] at critical points which occur on planes that are  $90^\circ$  to each other, called the principal planes. The maximum and minimum principal stresses were determined using the following formula:

$$s_n(max) = \frac{s_x + s_y}{2} + \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + \tau_{xy}^2} \quad (21)$$

$$s_n(min) = \frac{s_x + s_y}{2} - \sqrt{\left(\frac{s_x - s_y}{2}\right)^2 + \tau_{xy}^2} \quad (22)$$

where,  $S_x$  = stress at a critical point in tension or compression,

$S_y$  = stress at the same critical point and in a direction normal to the  $S_x$  stress,

$\tau_{xy}$  = shear stress at the same critical point acting in the plane normal to the x and y-axes.

The stress at a critical point in tension or compression ( $S_x$ ) normal to the cross section under consideration may be due to either bending or axial loads, or to a combination of the two, and it is determined by the following relationship:

$$S_x = \frac{Mc}{I} + \frac{P}{A} \quad (23)$$

where, M = bending moment,

c = distance from neutral axis to outer surface,

I = rectangular moment of inertia of cross section,

P = axial load,

A = area of cross section.

The shear stress ( $\tau_{xy}$ ) at the same critical point acting in the plane normal to the y-axis (which is the x-z plane) and in the plane normal to the x-axis (which is the y-z plane), may be due to a torsional moment, a transverse load, or a combination of the two. It is determined as:

$$\tau_{xy} = \frac{Tr}{J} + S_v \quad (24)$$

where, T = torsional moment,

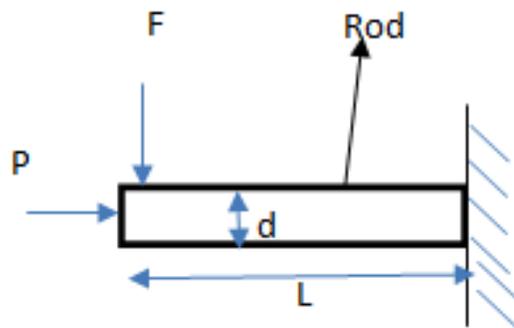
r = radius of circular cross section,

J = polar moment of inertia of cross section,

$S_v$  = transverse shear.

Therefore, the maximum shear stress ( $\tau_{max}$ ) at the critical point being investigated is equal to half of the greatest difference of any two of the three principal stresses given by:

$$\tau_{max} = \frac{s_n(\max) - s_n(\min)}{2} \quad (25)$$



**Figure 6:** Rod Subjected to Bending and Axial Loading.

Figure 6 shows the rod as subjected to axial and bending loading. According to Sharma and Aggarwal, (2012) for a mild steel or low carbon steel rod with code AISI 1018, the ultimate tensile strength ( $S_t$ ) of 440 MPa and tensile yield strength ( $S_b$ ) of 370 MPa were selected for the axial loading and bending moment, respectively.

#### Under Axial Loading

The diameter of the circular hitch rod was determined using the relationship below:

$$S_t = \frac{P}{A} \quad (26)$$

where,  $S_t$  = ultimate tensile stress, (N/m<sup>2</sup>)

P = axial load, (N)

A = area of cross section, (m<sup>2</sup>)

$$\text{But, } A = \frac{\pi d^2}{4} \quad (27)$$

where, d = diameter of rod (m)

Substituting Equation (26) into Equation (25) we have,

$$d = \sqrt{\frac{4P}{S_t \pi}} \quad (28)$$

From Equation (27), with an axial load of 2000N and 440MPa ultimate tensile stress, the diameter, d of the rod was determined to be 2.4mm.

### Under Bending Loading

The diameter of the hitch rod was determined with a bending loading of 1500N and tensile yield strength ( $S_b$ ) of 370 MPa from the following relationship:

$$S_b = \frac{Mc}{I} = \frac{Md}{2I} \quad (29)$$

where,  $I$  = moment of inertia,

$M$  = bending moment,

$d$  = diameter of rod,

$c$  = distance from neutral axis to outer surface,

$S_b$  = tensile yield strength.

The diameter of rod can be determined according to Hall, Holowenko, and Laughlin (2002):

$$S_b = \frac{32M}{\pi d^3} \quad (30)$$

where,  $M = F \times L$  (31)

$F$  = bending load (i.e. 11% of 1500N)

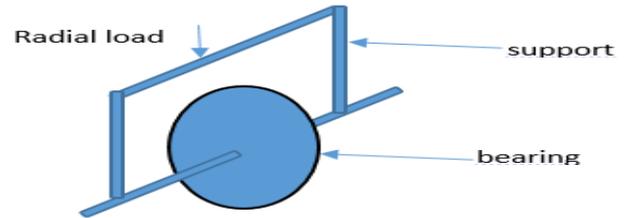
$L$  = length of the rod.

For a length of rod of 120mm (i.e., 0.12m), the diameter of the rod ( $d$ ) was calculated using Equations. (29) and (30) to be 8.3mm.

From the above calculation, it can be seen that the rod on the hitch should be designed according to the bending loading and not due to axial loading. Again, the diameter gotten from the bending loading is quite small which suggests that any diameter bigger than it can be used. Using a factor of safety of 4, the diameter of the rod is 13.2mm.

### Bearing Selection Analysis

Figure 7 shows schematic diagram of bearing arrangement subject to the radial load.



**Figure 7:** Bearing Arrangement.

The equivalent static loading (Hall, Holowenko, and Laughlin, 2002) is obtained as:

$$P_o = X_o F_r + Y_o F_a \quad (32)$$

where,  $P_o$  = equivalent static load,

$X_o$  = radial factor,

$Y_o$  = thrust factor,

$F_r$  = radial load,

$F_a$  = thrust load.

To determine the reactions at the bearing supports ( $R$ ), the reactions at any of the axle is used to determine the selection of bearings for the design.

$\therefore R = 1500 \times 88.4\%$  (i.e. the bending loading multiplied by the % weight on the axle)

$\therefore$  the reactions at the support was 1326N. However, there are two supports (Figure 7).

$$\therefore R_a = \frac{1326}{2} = 663 \approx 670\text{N}$$

Also, from the design, only a radial load effectively acts on the bearing, therefore,  $F_r = 670\text{N}$

From the SKF catalogue for radial contact groove single row ball bearings  $X_o = 0.6$  and  $Y_o = 0.5$  (Hall, Holowenko, and Laughlin, 2002). The equivalent static load ( $P_o$ ) was calculated to be 402N. But  $P_o$  must be equal to or greater than  $F_r$ ; hence the equivalent radial load for this design was 670N.

## **Fabrication of the Trailer**

The typical consideration in designing pull-behind motorcycle trailer is to produce a trailer that can be easily fabricated, assembled, and disassembled. The trailer was made of mild steel and the fabrication and assembly processes included welding, drilling and mechanical joining processes. The trailer frame was made of mild steel plates and angle bars. The tongue was also made of mild steel. The tires were made of rubber and selected for maximum loading. Suspensions were also selected for shock absorption. Figure 10 shows the CAD drawing of the produced trailer.



**Figure 8:** Pictorial View of the Produced Trailer.

## **Performance Evaluation**

Performance test was carried out on the trailer to ascertain its performance under actual condition of loading. Test was carried out to determine ease of cruising, climbing, ascending, cornering, reversing, and pushing by attaching the trailer to 125cc Daylong Motorcycle. Persons with various driving experience were used in carrying out the test.

The test was carried out on a level soft sand road surface as encountered in rural areas with slopes and steeps as the road type and the distance covered was about 400m. The loading condition was varied and put in the trailer at different instances to ascertain its performance. Concrete blocks weighing 20kg each were used as loads.

## **RESULTS AND DISCUSSION**

The performance evaluation results showed that the motorcycle can sufficiently tow a trailer given the drawbar pull which varies with speed as seen in the design. At a speed of 10km/hr, the drawbar pull was 1998N which was sufficient for pulling the trailer weight of 50kg. The design met with the standard theory being able to distribute its weight of about 10-15% on the trailer tongue, which was obtained from the trailer balance design of 11.6%. The center of gravity was also obtained to be 521.8mm from rear of the trailer which helped in the analysis of proper loading of the trailer. The trailer towed best in the presence of a limited amount of load of less than 60kg but had less cruising effect when the trailer was empty. In the case of driving through a bend, the trailer could perform well at not-so-constrained zone of radius not less than 900mm.

## **CONCLUSION**

The issue of the motorcycle being able to tow equipment has been shown to be possible by this study which was aimed at designing and fabricating a detachable trailer combination. The fabricated hitch provided the ease required for assembling and disassembling of the trailer from the motorcycle. The hitch operated satisfactorily during the performance evaluation, however there were some concerns on its performance with respect to its ability to reverse smoothly when the trailer is carrying load. The design and fabrication of a prototype trailer was satisfactorily achieved.

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