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Design and Fabrication of a Tricycle for Municipal Waste Collection

Kayode Oyindamola
Department of Mechanical Engineering,
School of Engineering and Engineering Technology,
Federal University of Technology, Akure, Ondo State, Nigeria.

ABSTRACT: *The use of inappropriate technology for the primary collection of municipal wastes in general is a common problem. A tricycle was designed and fabricated with a carriage to be used for doorstep collection of waste. The tricycle was designed to be human powered for the purpose of primary collection of municipal solid waste in locations and communities where the existing collection trucks could not cover due to narrow, poor quality roads and a high density population and congested areas. The final result shows that the tricycle is stable, strong, durable, and the dynamic test conducted has confirmed the stability, easy maneuvering, and the effectiveness of the braking system. This research work is applicable to the reduction in the inefficiency with the existing methods and equipment used in the collection of municipal waste.*

KEYWORDS: *Design and Fabrication, Human powered, Municipal Waste Collection, Tricycle*

I. INTRODUCTION

Waste is a general term covering all types of refuse resulting from living activities of humans and other animals. Wastes referred as rubbish, trash, garbage or junk are unwanted material which could be toxic and harmful to the environment [1]. Solid waste is the unwanted or useless solid materials generated from combined residential, industrial and commercial activities in a given area. Municipal or urban solid waste consists of organic and inorganic waste materials produced by households, commercial and institutional establishments that have no-economic value to the owner [2]. The inorganic wastes are of two types: recyclable and non-recyclable. General characteristic of solid wastes may be Garbage, Ash, Paper, Plastic, Leather, metals etc. Solid Waste Management is a science associated with the management of generation, storage, collection, transportation, processing and disposal of solid waste using the best principle and practices of public health, economics, engineering, conservation, aesthetics and other environmental conditions [3]. Primary collection of waste is an essential step of solid waste management activity. Primary collection system is necessary to ensure that waste stored at source is collected regularly and it is not disposed of on the streets, drains, water bodies, etc. [4]. The common arrangement in the few urban communities where a system is in place, is for waste management authorities to collect refuse from households and public containers on a regular basis using collection trucks. Unfortunately, operations managed by the waste management authorities has mostly been inefficient and ineffective as evidenced by mounds of decomposing refuse that have become a regular site in many urban areas [5]. The waste collection vehicle that are designed to operate in low-density urban areas with wide, well paved roads do not perform to the same level of service and in the same manner in locations with narrow, poor quality roads and with a high density population. This research work overcomes all those short comings.

II. DESIGN ANALYSIS AND CALCULATIONS

In this research work it was assumed that no friction exist between parts and the force on the pedal to be constant. In the tricycle frame, the ride's weight is assumed to be 1177N and that all the weight is placed on the seat. It was assumed that all the reaction forces are equal to the weight. In the carriage, the weight of the waste to be collected it is assumed to be 785N and uniformly distributed. The back shaft is assumed to be simply supported and carrying a uniformly distributed load of 931N carriage weight inclusive.

Determination of force applied on pedal.

$$I_m = ma = \frac{W}{g}a \tag{1}$$

$$I_m = ma = \frac{W}{g}a = \frac{1962}{9.81} \times 5 = 1000\text{N}$$

Where; I_m = inertial force (N)

M = mass of rider + mass of waste = 120Kg + 80Kg = 200Kg

a = acceleration of the tricycle = 5m/s²

g = acceleration due to gravity (m/s^2) = $9.81 m/s^2$
Since this is the resisting force of the motion of the tricycle, this is also the tension in the chain.
Now the required force applied on the pedal is found using:

$$\sum M_c = 0 = FL_c - TR \quad (2)$$

Where; M_c = moment about the crank (Nm)
F = applied force on the pedal (N)
T = tension in the chain = Inertial force (I_m)
L_c = length of the crank arm = 205mm = 0.205m
R = radius of the crank = 85mm = 0.085m

$$F = \frac{TR}{L_c} = \frac{1000\text{N} \times 0.085\text{m}}{0.205\text{m}} = 414.63\text{N}$$

Stresses Analysis of the Tricycle Chain
The tensile force in the chain, which is equal to the inertial force of the tricycle, was found to be
$T = I_m = 1000\text{N}$
Since the links contain holes, the stress in the links is not evenly distributed, but reaches a maximum near the hole. The maximum stress acting on the outer links in the chain is found by;

$$\sigma_{max} = K.\sigma_{nom} = \frac{K.F}{A} \quad (3)$$

Where; σ_{max} = maximum stress in the chain (MPa)
K = stress concentration factor = 2.4
σ_{nom} = normal stress in the chain (MPa)
A = cross sectional area of the chain (m^2)
$F = \frac{T}{2}$ = tension in one arm of the chain (N)

Therefore; $\sigma_{max} = 2.4 \ \frac{1000/2}{(1.59 \times 10^{-3})(6.35 \times 10^{-3})} = 118.85\text{MPa}$

The shear stress (τ) acting on the pin holding the links together is given by;

$$\tau = \frac{F}{A} = \frac{T}{2\pi r^2} \quad (4)$$

$$\tau = \frac{F}{A} = \frac{T}{2\pi r^2} = \frac{1000/2}{3.142\ (1.5875 \times 10^{-3})^2} = 63.14\text{MPa}$$

Stresses Analysis of the Chain Wheel
The spokes on the chain wheel which pull on the chain links, experience a shear stress on the area between the base of the spoke and the chain wheel. The area on the base of each spoke is roughly 3.175×10^{-3}m by 9.525×10^{-3}m.

$$\tau_{xy} = \frac{F}{A} \quad (5)$$

$$\tau_{xy} = \frac{F}{A} = \frac{1000}{(3.175 \times 10^{-3})(9.525 \times 10^{-3})} = 33.07\text{MPa}$$

Determination of length of Chain
The length of chain was calculated using:

$$L = K.P \quad (6)$$

$$K = \frac{T_1 + T_2}{2} T_1 + \frac{2x}{P} + \left[\frac{T_2 - T_1}{2}\right]^2 . \frac{P}{x} \quad (7)$$

Where; L = length of chain (m)
x = center distance between the two sprockets = 0.047m
P = pitch of the chain = 8mm = 0.008m
T_1 = no of teeth on the smaller sprocket = 23
T_2 = no of teeth on the bigger sprocket = 39

$$K = \frac{23+39}{2} + \frac{2(0.047)}{0.008\text{m}} + \left[\frac{39-23}{2}\right]^2 . \frac{0.008}{0.047} = 53.64$$

Therefore; The length of the chain:
$L = K \times P = 53.64 \times 0.008 = 0.429\text{m}$

Pedal Crank Assembly
To analyze the maximum state of stress acting on the crank, a point on the surface of the crank is chosen for maximum shear stress, and a point at the farthest distance away from the pedal is chosen for maximum bending stress. Since the stress acting on the crank are a combination of the torsion and bending moment,

The total stress is given by:

$\sigma_x = 0$

$$\sigma_y = \frac{M}{S} = \frac{FL_c}{\pi d^3/32} \tag{8}$$

$$\sigma_y = \frac{M}{S} = \frac{FL_c}{\pi d^3/32} = \frac{(414.63N)(0.205)}{3.142(0.0127)^3/32} = 422.62MPa$$

$$\tau = \frac{T_r}{I_P} = \frac{16FL_p}{\pi d^3} \tag{9}$$

$$\tau = \frac{T_r}{I_P} = \frac{16FL_p}{\pi d^3} = \frac{16(414.63N)(0.102m)}{3.142(0.0127)^3} = 105.14MPa$$

Where; σ_x, σ_y = bending stresses on the pedal

τ = torsional stress

With the state of stress found at this point on the crank, the principal stress and maximum shear stress are calculated as follows:

$$\sigma_{1,2} = \frac{(\sigma_x+\sigma_y)}{2} \pm \sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2 + \tau^2 xy} \tag{10}$$

$$\sigma_{1,2} = \frac{(422.62)}{2} \pm \sqrt{[\frac{(-422.62)}{2}]^2 + 105.14^2}$$

$\sigma_{1,2}$ = 211.31MPa ± 236.02MPa

σ_1 = 447.33MPa

σ_2 = −24.71MPa

$$\tau_{max} = \sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2 + \tau^2 xy} \tag{11}$$

$$\tau_{max} = \sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2 + \tau^2 xy} = \sqrt{[\frac{(-422.62)}{2}]^2 + 105.14^2} = 236.02MPa$$

Stress Analysis of the Tricycle Frame

The point chosen on the tricycle frame is on the seat tube with the assumptions that the total weight of the rider (1177N) acts on this segment.

$$\sigma_y = \frac{M}{S} = \frac{WL_{AB}}{\frac{\pi}{4}(r_2^4 - r_1^4)/r_2} \tag{12}$$

Where; W = weight of the rider (N)

L_{AB} = length of the seat tube (m)

r_2 = external diameter of the tube (m)

r_1 = internal diameter of the tube (m)

$$\sigma_y = \frac{M}{S} = \frac{1176 \times 0.6096}{\frac{3.142}{4}(0.05^4 - 0.025^4)/0.05} = 7.79MPa$$

$\sigma_x = 0$

$\tau = 0$

$$\tau_{max} = \frac{\sigma_y}{2} = \frac{7.79}{2} = 3.895MPa$$

Stress Analysis of the Handle Bar

$$\sigma_x = \frac{M}{S} = \frac{FL_H}{\frac{\pi}{4}(r_2^4 - r_1^4)/r_2} \tag{13}$$

Where; L_H = length of handlebar (m)

F = applied force on the handlebar (N)

r_2 = outer diameter of the handlebar (m)

r_1 = inner diameter of the handlebar (m)

$$\sigma_x = \frac{M}{S} = \frac{0.18415 \times 90}{\frac{3.142}{4}(0.03175^4 - 0.025^4)/0.03175} = 1.12MPa$$

$\sigma_y = 0$

$$\tau = \frac{T_r}{I_P} = \frac{Tr_2}{\frac{\pi}{2}(r_2^4 - r_1^4)} \tag{14}$$

$$\tau = \frac{T_r}{I_P} = \frac{Tr_2}{\frac{\pi}{2}(r_2^4 - r_1^4)} = \frac{(90)(0.127)(0.03175)}{\frac{3.142}{4}(0.03175^4 - 0.025^4)} = 0.39MPa$$

$$\sigma_{1,2}=\frac{(\sigma_x+\sigma_y)}{2}\pm\sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2+\tau^2xy} \quad (15)$$

$$\sigma_{1,2}=\frac{1.12}{2}\pm\sqrt{[\frac{1.12}{2}]^2+0.39^2} = 0.56\text{MPa} \pm 0.682\text{MPa}$$

$\sigma_1 = 1.24$MPa

$\sigma_2 = -0.122$MPa

$$\tau_{max}=\sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2+\tau^2xy} \quad (16)$$

$$\tau_{max}=\sqrt{[\frac{(\sigma_x-\sigma_y)}{2}]^2+\tau^2xy} = \sqrt{[\frac{1.12}{2}]^2+0.39^2} = 0.682\text{MPa}$$

Determination of Maximum Torsional Moments acting on the shaft.
The Shaft is to carry a uniform distributed load of 932N and supported by two wheels at the ends.

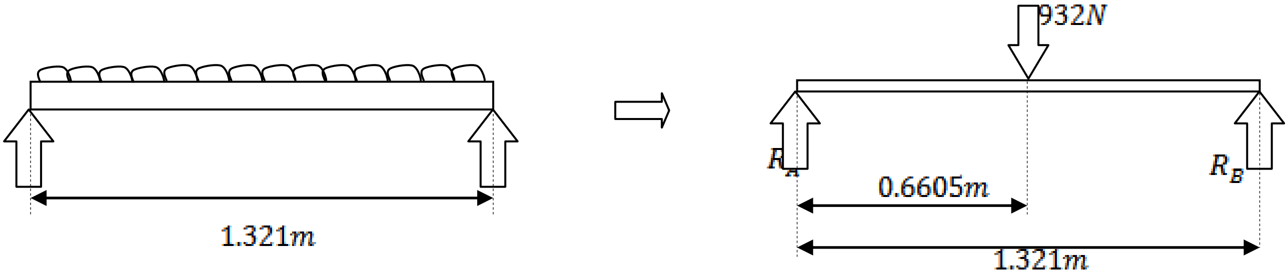


Figure 1: Vertical load diagram

Considering vertical forces in Figure 3.5(b) we obtain

$R_A + R_B = 931\ N$

Taking moment about R_A we obtain

$(932 \times 0.6605) - 1.321R_B = 0$

$\therefore R_B = 465.5N$

The maximum bending moment M_{bmax} occurs at the midway of the shaft with a value of 307.5Nm

M_{bmax}= 307.5Nm

The maximum torsional moment is given [6] as;

$$M_{tmax}=\frac{60P}{2\pi N} \quad (17)$$

Where; P = power transmitted to the shaft (KW)

N = number of revolutions = 100rpm

P = F.V (18)

Where; F = applied force on the pedal = 414.63N

V = speed = 35.24m/s

P = 414.63N × 35.24m/s = 14.56KW

N = number or revolutions per minute

$$M_{tmax}=\frac{60P}{2\pi N} = \frac{60 \times 14611.56}{2 \times 3.142 \times 100} = 1395.12\text{Nm}$$

Determination of the shaft diameter.
The diameter of the shaft is given by the ASME code equation for a solid shaft having little or no axial loading [6];

$$d^3 = \frac{16}{\pi S_s}\sqrt{(K_b M_{bmax})^2 + (K_t M_{tmax})^2} \quad (19)$$

Where; d = shaft diameter

S_s = allowable shear stress (N/m²)

K_t= combined shock and fatigue factor applied to torsional moment

K_b= combined shock and fatigue factor applied to bending moment

K_b and K_t for rotating shaft with gradually applied load are given as 1.5, and 1.0 respectively. For shafts without keyways, the allowable shear stress S_s is $55\ MN/m^2$[6]. Hence:

$$d^3 = \frac{16}{3.142 \times 55 \times 10^6}\sqrt{(1.5 \times 307.5)^2 + (1.0 \times 1395.12)^2} = 1.25 \times 10^{-4}$$

d = 0.05m.

III. TESTING

The various parts of the machine were assembled, the tricycle to see if desired result is attained. Testing of machine materials used basically in the mechanical design and construction of the tricycle requires various tests and balance in its static and dynamic as well as its physical and mechanical properties to enable their state standard and efficient use in load bearing with or without load application.

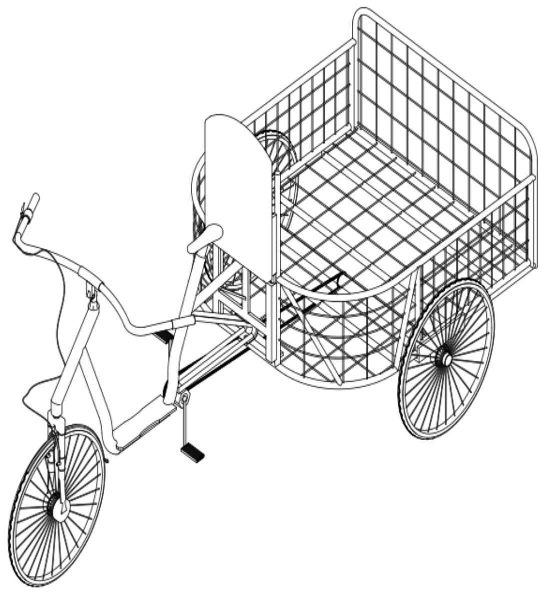
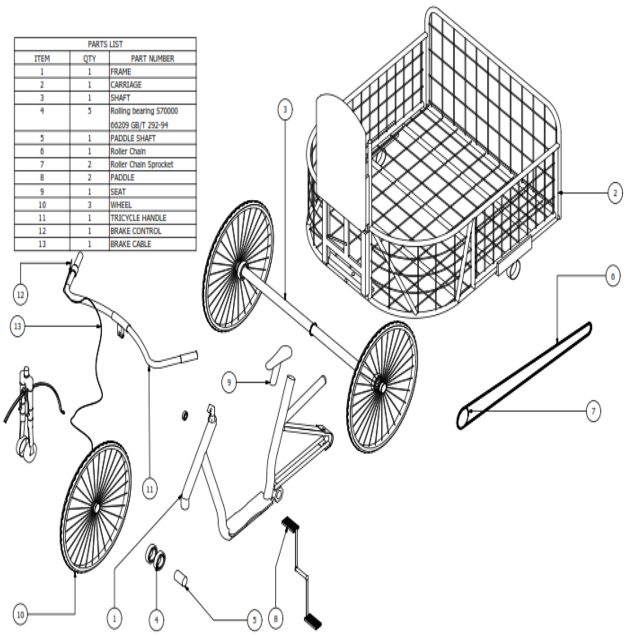


Figure 2: Human powered tricycle



PARTS LIST		
ITEM	QTY	PART NUMBER
1	1	FRAME
2	1	CARRIAGE
3	1	SHAFT
4	5	Rolling bearing S70000 66209 GB/T 292-94
5	1	PADDLE SHAFT
6	1	Roller Chain
7	2	Roller Chain Sprocket
8	2	PADDLE
9	1	SEAT
10	3	WHEEL
11	1	TRICYCLE HANDLE
12	1	BRAKE CONTROL
13	1	BRAKE CABLE

Figure 3: Exploded view of various sub-assemblies

The tricycle was tested to see if desired result is attained. The test was done to check its balance in its static and dynamic state with or without load application. Static test confirmed the tricycle stability, properly aligned, since all the extra weights were installed as low as possible to the center of gravity of the tricycle. Dynamic test conducted confirmed also the stability, easy maneuvering, the effectiveness of the braking system, the approximate distance covered per kilometer hour. The tricycle performed efficiently and effectively when the tricycle was loaded with the actual load capacity of 785N and pedaled for 1.5km. It also moved effectively without tilting or observation of any inconvenience when tested with a load capacity of 1750N at the same distance.

IV. CONCLUSION

This research work proffers a means of effective collection of municipal solid waste by designing and fabricating a tricycle with a carriage that will be human powered for the purpose of primary collection of municipal solid waste in locations and communities where the existing collection trucks could not cover due to narrow, poor quality roads and a high density population and congested areas such as markets.

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