DEVELOPMENT OF SOFTWARE FOR THE DESIGN OF CONVENTIONAL SELF TILTING WHEELBARROW

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ABSTRACT: *This research presents the software approach to the design of self-tilting wheelbarrow. A total of Ninety Five (95) variables were used to form the basis of the computer input. A compact and clear flowchart was presented from which the written command using 'If' statement is formed. The output from the printed computer interface shows an accurate, compact, fast and easy design which of course could be easily modified, adapted or manipulated for further design and interchangeability.*

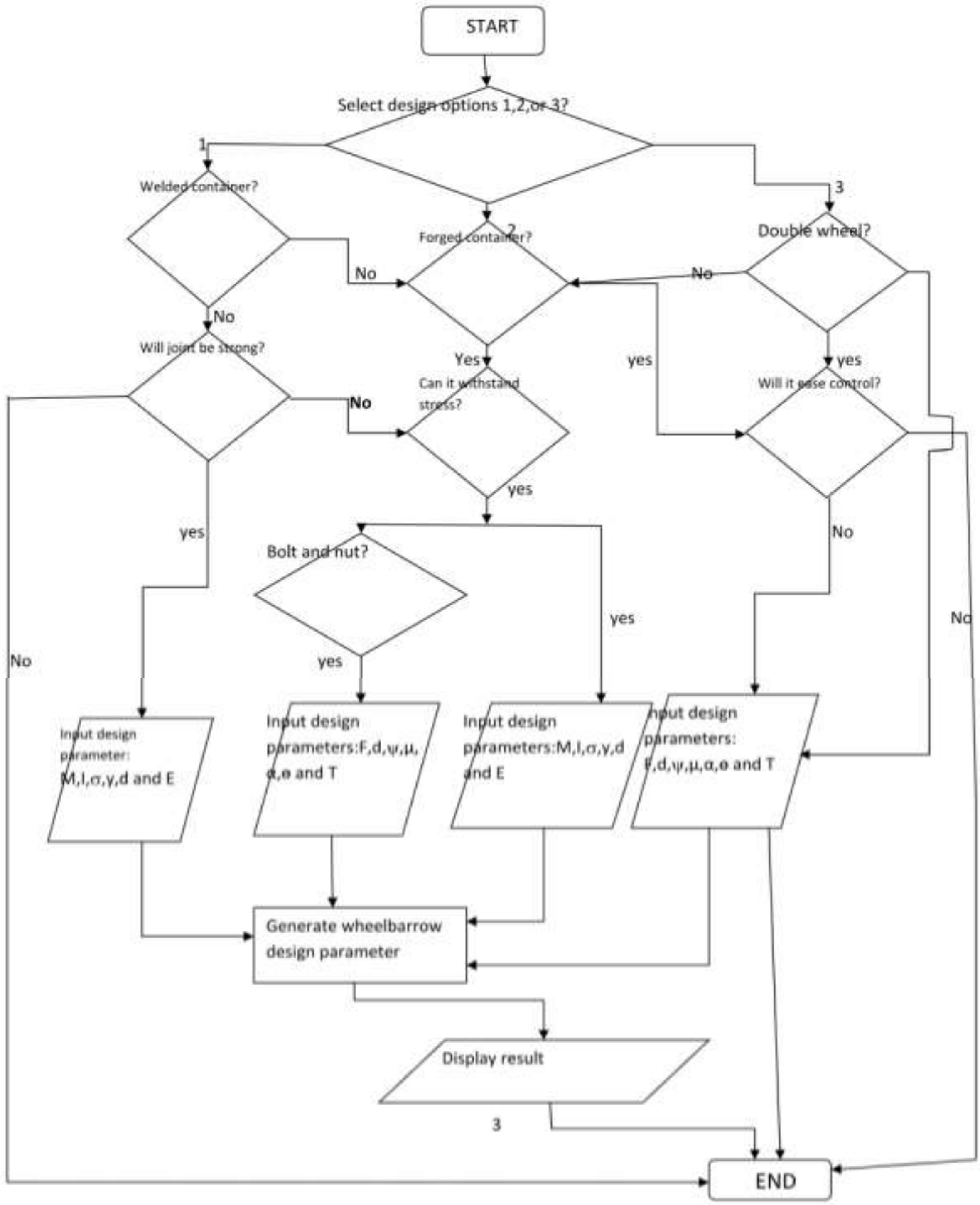
KEYWORDS: Software Design, Self Tilting Wheelbarrow, Development

INTRODUCTION

The wheelbarrow is designed to distribute the weight of its load between the wheel and the operator so enabling the convenient carriage of heavier and bulkier loads than would be possible if the weight is to be carried entirely by the operator. As such it is a second-class lever. Traditional Chinese wheelbarrows, however, had a central wheel supporting the whole load. Use of wheelbarrows is common in the construction industry and in gardening. Typical capacity is approximately 100 liters (4 cubic feet) of material (Temple 1986).

METHODOLOGY

In order to get all involving parameters relevant, literatures and encyclopedia where searched and a total of ninety five (95) of them were enlisted. This form the basis of the database and input were made. The users interface was presented. The value of each parameter were carefully selected from code of engineering practice for the purpose of relevancy.



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WRITTEN COMMANDS

f = stress constant
μ = coefficient of friction between wheels and road

S = stress at the cross section (N/mm^2)

W = total load on the pipe (N)

X = distance used for analysis (mm)

Z = section modulus of the cross of the pipe

L = length of pipe (mm)

y = deflection inches (mm)

E = Modulus of Elasticity (Mpa)

I = Moment of Inertia

d = diameter of the pipe (mm)

f_y = Applied load vertically (N)

f_x = Applied load horizontally (N)

δa = stress, tensile or compressive (N/mm^2)

F = Applied force (N)

W = weight of the load (kg)

K= constant, pounds of load per inch of deflection

G = modulus of rigidity of spring material (Gpa)

D = mean coil diameter or outer diameter (mm)

lf = uncompressed length of spring (mm)

n_C = number of coils

Cp = pitch of the spring coil(mm)

pd= pitch circle diameter of the spring(mm)

W= section modulus of spring

d=inner diameter of spring(mm)

v=volume of the spring(mm^3)

L= required rating life of the bearing (hours)

L_{10}= life of the bearing for 90% survival

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L_{10} = 1 x 10^6rev for pillow bearing

W_e = weight of load (N)

K = bearing constant

x = radial factor

y= thrust factor

s= service factor

C= dynamic capacity(Wh)

P = equivalent dynamic load (N/s)GVB

N = no of revolutions (rpm)

f = coefficient of friction

W = load (N)

R = outer radius of the wheel (mm)

r = inner radius of the wheel (mm)

n = revolution per seconds (rpm)

m = mass of bucket (kg)

v = volume of bucket (mm^3)

ρ = density of bucket material (kg/mm^3)

A = area of the trapezium (mm^2)

h = height of the trapezium (mm)

a = longest side (mm)

b = shortest side (mm)

ρ= density of wheel barrow material (kg/m^3)

a= the thickness of the nut (mm)

h= height or thickness of the base of the thread (mm)

t= depth of thread is 0.75p (mm)

b= length of thread taking the turn (mm)

p= pitch of threads (mm)

L= thread length (mm)

D= nominal diameter (mm)

d_i= root diameter = D-1.5p (mm)

d= pitch diameter (mm)

Θ = helix angle (radians)

μ= coefficient of friction between bolt threads and nut threads

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R_A, R_B, R_C, R_D = reactions at points A, B C, and D respectively (N)

X_1 , X_2, and X_3 = perpendicular distances (mm)

$T_h = T_r$ cosØ = horizontal component of force(Nm)

$T_v = T_r$ sin Ø= vertical component of force(Nm)

L = load (N)

E = effort (N)

x = distance moved by load (mm)

y = distance moved by effort (mm)

d =pitch circle diameter of the bolt(mm)

T =torque(Nmm)

Ps=Torque or power required to overpower friction of shaft in bearing(Nm)

W_0= work output

W_1= work input

α = face angle of thread (^{o})

F= downward thrust(turning)(Nmm)

Ø= flange angle(^{o})

P_b= stress in the nut thread(N/mm^2)

F_t = tangential force on the gear (N)

F_n = normal force on the gear (N)

F_r = radial force on the gear (N)

L and V = the lateral and vertical forces acting upon the road and wheel(N)

δ = the angle made when the wheel flange is in contact with the road face(^{o})

Q = the charge (coulombs)

I = the current (amps)

t = the time (seconds).

E = the energy stored (watt-hours)

C = the capacity (amp-hours)

Vavg = the average voltage during discharge (volts)

C =battery capacity (Wh)

C' = battery lifetime (years)

α ′ = gear tooth angle

If the load in the wheelbarrow = known, **Then**

Compute the deflection at any point on the pipe using (Y = $\frac{wx(1-x)}{24EIL}(L^2 + x(1-x))$)

Compute the stress that may affect deflection δ_a using ($\delta_a = \frac{1.273}{d^2}(\frac{2lfy+fx}{d})$

Compute the force required to propel the wheelbarrow F using (F = $\frac{WX}{L}$)

If the computed stress that may affect deflection < the computed force required to propel the wheelbarrow, **Then**

Design is satisfactory

Else, select a better load capacity and redesign until:

the computed stress that may affect deflection ($\delta_a = \frac{1.273}{d^2}(\frac{2lfy+fx}{d})$ < the computed force required to propel the wheelbarrow (F = $\frac{WX}{L}$)

End If

Having **known** the mean coil diameter of the spring, **Then**

Compute the stiffness K of the spring using (K = $\frac{ad^4}{8nD^3}$)

Compute the pitch of the spring coil Cp, using (Cp = $\frac{lf}{nc}$)

Compute the angle of the spring coil relative to the mounting base using (F = $\tan^{-1}(\frac{cp}{pd})$)

Compute the section modulus of spring using (W = $s^2(\frac{l-2s}{6})$)

If the computed pitch of the spring coil (Cp = $\frac{lf}{nc}$) > the computed angle of the spring coil relative to the mounting base (F = $\tan^{-1}(\frac{cp}{pd})$), **Then**

Design is satisfactory

Else select a better mean coil diameter and redesign until:

the computed angle of the spring coil relative to the mounting base > computed angle of the spring coil relative to the mounting base

End If

If the radial and axial forces = known **Then**

Compute the equivalent load P_e on the bearing using (P_e = (xF_r+ yF_a)s)

Else If the equivalent load on the bearing = unknown **Then**

Compute the dynamic capacity C of the bearing using (C = $\frac{L}{K}$ x P_e and L = $\frac{10}{60N}$ $(\frac{6}{We})^k$)

Compute the torque required to overpower friction of shaft in bearing using eqn(P = $\frac{2}{3}fw(\frac{R^3-r^3}{R^2-r^2})$)

Compute the Power p lost by friction using eqn (P = $\{\frac{cfn\pi}{36}\}$)

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End If

ع Check for the total reactions(ع $f = R_A + R_D = R_B + R_C$) in the shaft due to rotational motion using eqn(3.14)

Compute the bending moment of the shaft using ($M = \frac{\Pi}{32}$ x δb (d3))

If $M = \frac{\Pi}{32}$ x δ_b (d^3) < $T = \frac{\pi}{16} i D^3$, Then

the shaft is save under tensional load

design is satisfactory

Else select a better diameter and redesign until:

Bending moment of the shaft $M = \frac{\Pi}{32}$ x δ_b (d^3) < its rotational torsion $T = \frac{\pi}{16} i D^3$

End If

If the maximum load = known, **Then**

Compute the mechanical advantage using eqn($M.A = \frac{L}{E}$)

Compute the compute the velocity ratio using eqn($V.R = \frac{x}{y}$)

Compute the overall efficiency using eqn($E = \frac{M.A}{V.R}$ x 100 %)

If the computed mechanical advantage ($M.A = \frac{L}{E}$) > the computed velocity ratio ($V.R = \frac{x}{y}$),

Design is satisfactory

Else, find a better maximum load capacity and redesign until:

the mechanical advantage > the computed velocity ratio,

End If

Having **known** mass of the bucket, **Then**

Compute the volume of the bucket using eqn(m = ρv)

Compute the area of the bucket using eqn($A = \left(\frac{1}{2}(a+b)l\right)^4$)

If the computed area ($A = \left(\frac{1}{2}(a+b)l\right)^4$ < the computed volume(m = ρv)

Design is satisfactory

Else, select a better mass (m) and redesign until:

the area of the bucket < the volume of the bucket,

End If

If the bolt diameter = known, **Then**

Compute the pitch circle diameter of the bolt using eqn($d = \frac{D+\mathrm{di}}{2}$)

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Compute the helix angle of thread on the bolt using eqn($\tan\Theta = \frac{L}{\Pi d}$)

Compute the efficiency of the bolt using eqn($Ŋ0 = \frac{W0}{W1} = \frac{FL}{2\Pi T}$)

If the computed pitch circle diameter of the bolt ($d = \frac{D+\mathrm{di}}{2}$) > the computed helix angle of thread ($\tan\Theta = \frac{L}{\Pi d}$) on the bolt,

Design is satisfactory

Else, select a better bolt diameter and redesign until;

the pitch circle diameter of the bolt > the computed helix angle of thread on the bolt.

If the angular velocity of the shaft = known, **Then**

Compute the tangential force on the gear using eqn ($F_t = F_n \cos\alpha$ ')

Compute the radial force using eqn ($F_r = F_n \sin\alpha$ ')

If the computed radial force ($F_r = F_n \sin\alpha$) > the computed tangential force ($F_t = F_n \cos\alpha$)

Design is satisfactory

Else, select a better angular velocity and redesign until:

the radial force > the tangential force acting on the gear

Check for the battery capacity using eqn (C = xT)

Compute battery life using eqn ($C' = \frac{C}{0.8}$)

If the angle made when the wheel flange is in contact with the road face = **known**

Compute the lateral and vertical forces acting upon the road and wheel using eqn ($\left(\frac{L}{V}\right) = \left(\frac{\tan(\delta) - \mu}{1+\mu * \tan(\delta)}\right)$)

Compute the diameter of the wheel using eqn ($D = \sqrt{\frac{4L}{\pi V}} = \left(\frac{\tan(\delta) - \mu}{1+\mu * \tan(\delta)}\right)$)

If the computed lateral and vertical forces ratio $\left(\frac{L}{V}\right) = \left(\frac{\tan(\delta) - \mu}{1+\mu * \tan(\delta)}\right)$ < the computed diameter of the wheel $D = \sqrt{\frac{4L}{\pi V}} = \left(\frac{\tan(\delta) - \mu}{1+\mu * \tan(\delta)}\right)$, Then

Design is satisfactory

Else, select a better angle until:

the lateral and vertical forces ratio < the diameter of the wheel

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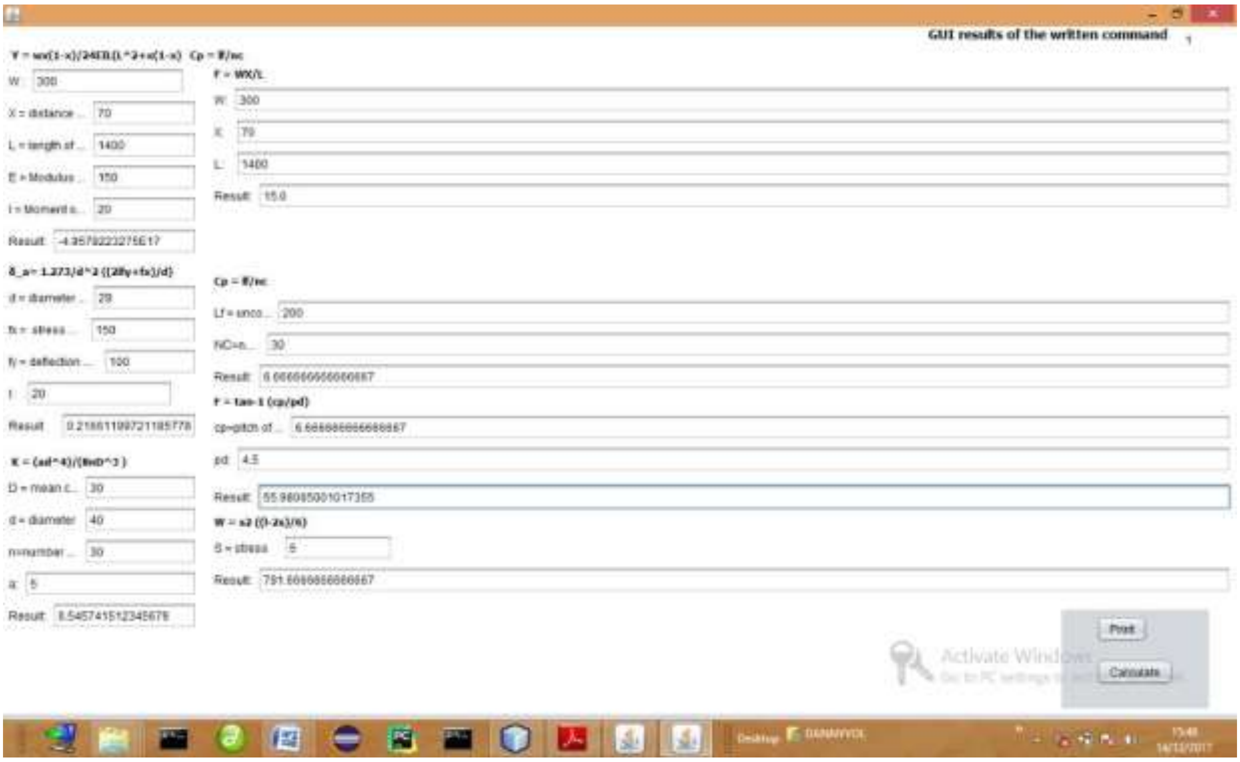
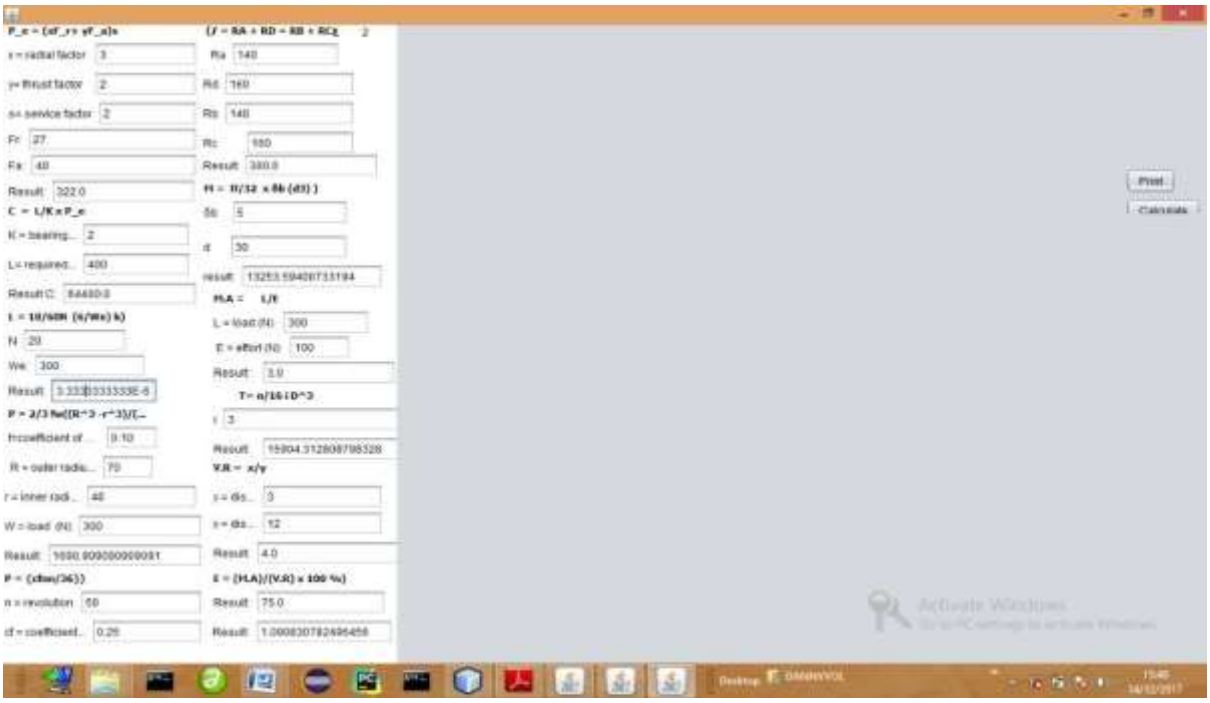
End





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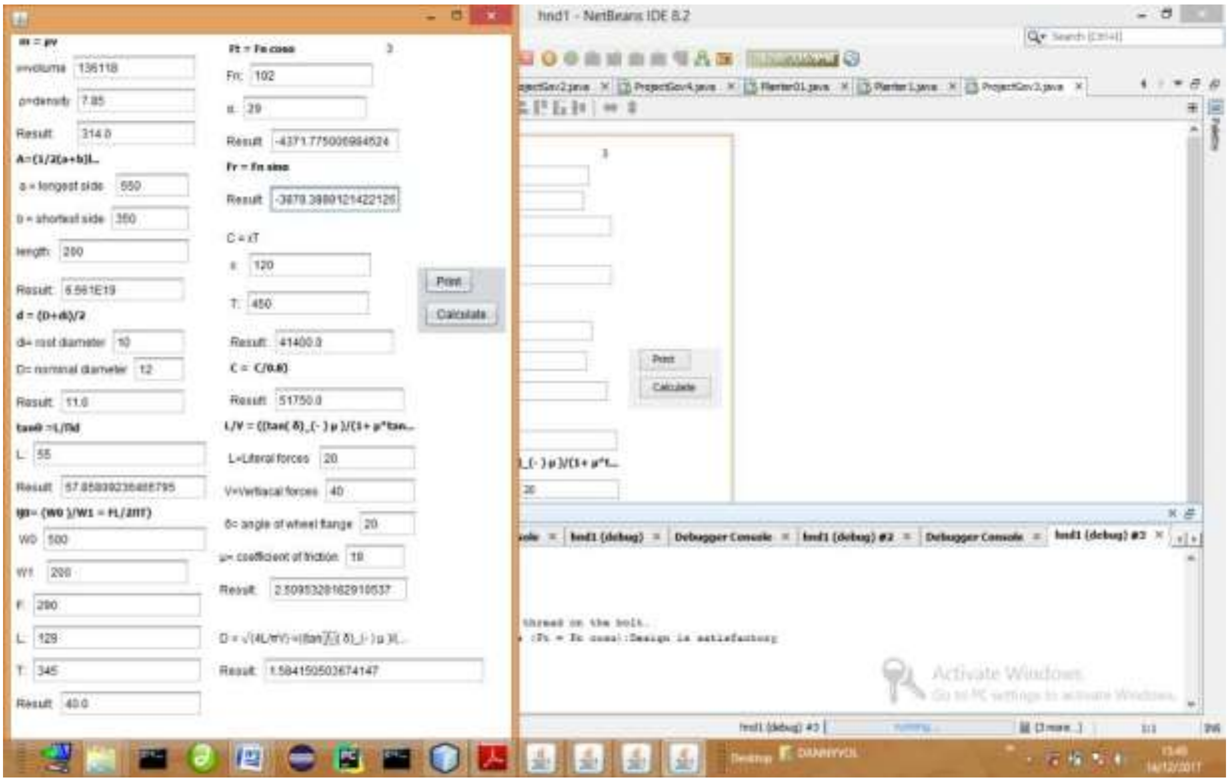

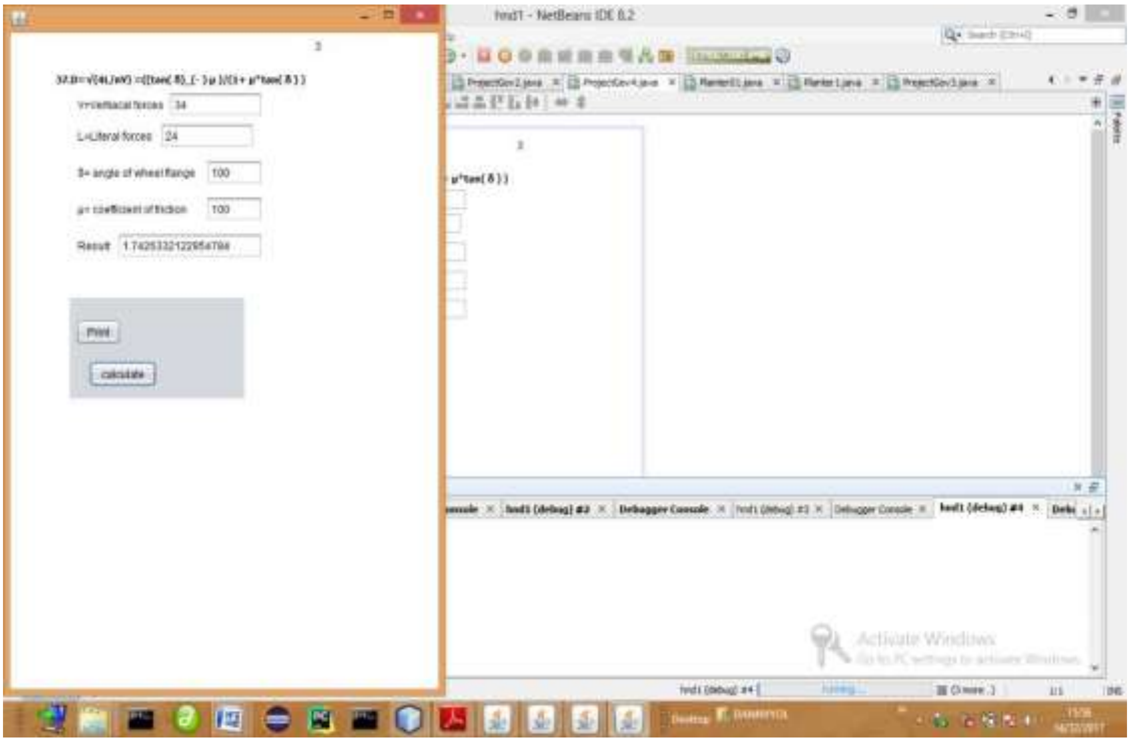

TABLE 1: Result Table

S/N	COMPONENT	PARAMETER	RESULT/VALUE
1	SHAFT	Diameter(d) Length(l), Material,	30mm 180mm Medium carbon steel
2	WHEEL	Diameter Type	250mm Perforated wheel
3	BUCKET	Volume Area Mass	1336118mm^3 6561mm^2 314g
4	BEARING	Size Outer diameter Inner diameter Dynamic capacity Friction coefficient	Hub size 40mm 30mm 64400.0 0.10
5	SPRING	Stiffness Pitch of the spring coil Spring coil angle	0.54574151234567 40mm 30mm
6	PIPE/ROD	Diameter Thickness Length Deflection on the pipe	50mm 2mm 1400mm -0.00000000049
7	BOLT AND NUT	Diameter Helix angle Pitch circle diameter	M12 60^o 11.025
8	DRIVING GEAR	Diameter Number of tooth Type	120 15 Spur gear
9	DRIVEN GEAR	Diameter Number of tooth Type	30mm 60 Spur gear
10	ROTOR	Capacity Speed	300rpm
11	BATTERY	Capacity type	12V Bike type
12	TYRE	Diameter	10inches

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CONCLUSION

This paper is conscience but comprehensive presentation of the database for the design of self-tilting wheelbarrow. The beauty of this product is its adaptability, machinability, interchangeability and modification.

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