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Design Calculations for Agriwaste Briquette Making Machine

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ABSTRACT

This paper presents the design calculation of Agriwaste Briquette making machine. Assumptions and references are taken for designing the Agriwaste Briquette making machine. The design calculation of Agriwaste Briquette making machine is done. For developingAgriwaste Briquette making machine, various componentdesign of chain drive, design of connecting rod, design of ratchet and pawl,design of rack and pinion, pedal mechanism.
Keywords : Were Designed And Fabricatedchain Drive, Connecting Rod, Ratchet And Pawl, Rack And Pinion, Pedal Mechanism.

I. INTRODUCTION

TheAgriwaste Briquette making machine shown in the figure no.1 other than the frame consist of chain drive, sprocket, gear, rack and pinion, ratchet and pawl, rack shaft, press pedal etc.

Figure 1. Agriwaste Briquette Making Machine

II. DESIGN CALCULATIONS FOR AGRIWASTE BRIQUETTE MAKING MACHINE

The aim of this is to give the complete design information about the Agriwaste Briquette making machine. In this, the explanations and some other parameters related to the project are included. With references from various sources as journal, thesis, design data book, literature review has been carried out to collect information related to this project.

A. Design consideration

- Capacity of Agriwaste Briquette Making Machine = 3 Kg
- Force Applied on Pedal = 7350.75 N
- Average Height of Machine According to Human Ergonomics = 165 cm

Design calculations

Determination of agriwaste briquette making machine force experimentally.

1.Design Calculations Of Chain Drive

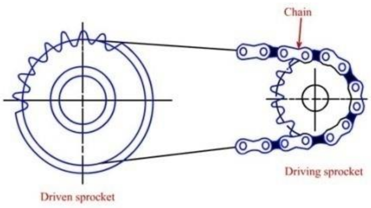

Figure 2. Proposed diagram of Chain Drive

Figure 3. Actual diagram of Chain Drive

Velocity Ratio, **i = 2**
Minimum teeth on driver sprocket, $\mathbf{Z_1 = 16}$
Standard minimum pitch, **p = 12.7**mm
C. C. distance between two shafts, **a = 152**mm

Calculations

Velocity Ratio, $i = \frac{Z_2}{Z_1}$
∴Teeth on Driven Sprocket, $\mathbf{Z_2} = 32$
Pitch Diameter, $D = \frac{P}{\sin\frac{180}{Z}}$
∴Pitch Diameter of driver sprocket, D_1
$D_1 = P/\sin(180/Z_1) = 65$mm
∴Pitch Diameter of driven sprocket, D_2
$D_2 = P/\sin(180/Z_2) = 129.5$mm
Number of links in the chain, L_n
$L_n = 2(\frac{a}{p}) + (\frac{Z_1 + Z_2}{2}) + (\frac{Z_2 - Z_1}{2\pi})^2 \times (\frac{p}{a})$
$L_n = 48.4788$ i.e.
$L_n = 49$ Numbers of links.
Length of chain, L
$L = L_n \times p$
L=622.3mm
Now, From Table No.14.1
Selecting chain, ISO chain number 08B
Pitch, p=12.7mm
Roller diameter, $d_{1(max)} = 8.51$mm
Width, $b_{1(min)} = 7.75$mm
Transverse pitch, $p_t = 13.92$mm
Breaking load (minimum)=17800N
Power rating of chain $= \frac{\text{POWER TRANSMITTED} \times Ks}{K_1 \times K_2}$
Where,
K_S=Service Factor
K_1=Multiple Strand Factor
K_2=Tooth Correction Factor

From Table No.14.2

Taking pinion speed 50RPM, & for the ISO chain number 08B
Power rating of chain=0.34KW

From Table No.14.3

For heavy shock & electric drive
Service factor K_S=1.4

From Table No.14.4

For single strand,
Multiple strand factor K_1=1.0

From Table No.14.5

For 16 teeth,
Tooth correction factor, K_2=0.92
Power rating of chain $= \frac{\text{POWER TRANSMITTED} \times Ks}{K_1 \times K_2}$
Power to be transmitted = 0.2234 K
Top diameter (Addendum Diameter)-D_a
From Table No.14.6
$D_{a\,(max)} = D_2 + 1.25p - d_1$
$D_{a\,(max)} = 136.865$ mm
$D_{a\,(min)} = D_1 + p \times (1 - \frac{1.6}{Z_1}) - d_1$
$D_{a\,(min)} = 67.92$mm
Top diameter of driven sprocket, D_{a2}=136.865mm
Top diameter of driver sprocket, D_{a1}=67.92mm
Root diameter (Dedendumdia), D_F
$D_F = D - 2ri$
Roller seating radius, r_i
$(r_i)_{max} = 0.505d_1 + 0.069 \times \sqrt[3]{d_1}$
$(r_i)_{max} = 4.438$ mm
$(r_i)_{min} = 0.505d_1$
$(r_i)_{min} = 4.297$ mm
$D_{F(min)} = D_1 - 2\,ri_{min}$
$D_{F\,(min)} = 56.406$ mm
$D_{F\,(max)} = D_2 - 2ri_{max}$
$D_{F\,(max)} = 120.624$ mm
Tooth flank Radius, r_e

$(r_e)_{Driven} = 0.008\, d_1(Z_2^{\,2}+180)$

$(r_e)_{Driven} = 81.968$ mm

$(r_e)_{Driver} = 0.12 d_1(Z_1 + 2)$

$(r_e)_{Driver} = 18.38$ mm

Roller Seating Angle, α

For, Driven Sprocket, α_{Driven}

$\alpha_{Driven} = (120 - \frac{90}{Z_2})$

$\alpha_{Driven} = 117.2^{o}$

For, Driver Sprocket, α_{Driver}

$\alpha_{Driver} = (140 - \frac{90}{Z_1})$

$\alpha_{Driver} = 134.4^{o}$

Tooth height above the pitch polygon, h_a

For Driven Sprocket, $(h_a)_{Driven}$

$(h_a)_{Driven} = 0.625p - 0.5d_1 + \frac{0.8p}{Z_2}$

$(h_a)_{Driven} = 4$ mm

For Driver Sprocket, $(h_a)_{Driver}$

$(h_a)_{Driver} = 0.5(p - d_1)$

$(h_a)_{Driven} = 2$ mm

Tooth side radius, r_x

$r_{x\,(min)} = p = 12.7$mm

Tooth width, b_{f1}

$b_{f1} = 0.93\, b_1$ if $p \le 12.7$ mm

$b_{f1} = 0.95\, b_1$ if $p > 12.7$ mm

$b_{f1} = 0.93\, b_1$

$b_{f1} = 7.2$ mm

Tooth side relief, b_a

$b_a = 0.1p$ to $0.15p$

Taking $b_a = 0.15p$

$\therefore b_a = 1.9$ mm

Power transmitted by chain drive

Power transmitted by chain drive = Force caused by driver shaft rotation x average (minimum) velocity of chain drive

$\therefore$Force caused by driver shaft rotation

= 50 x 9.81 = 490.5 N

223.4 = 490.5 x minimum velocity of chains drive, V

$\therefore V = 0.455$ m/sec.

2.Design Calculations Of Rack And Pinion

Selecting profile type of tooth is 20^{o} full depth

Assuming preferred no. of teeth on pinion, $T_P = 7$

A) Pinion

Pitch diameter of Pinion,

$D_P = T_P \times m$ mm.

Since, $N_P = 2880$ rpm.

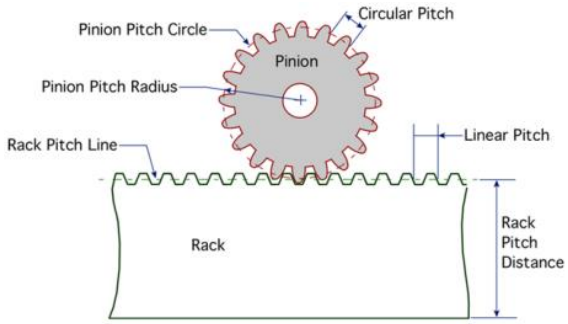

Figure 4. Proposed Rack and Pinion

Figure 5. Actual diagram of Rack and Pinion

$V_P = \pi \times T_P \times \frac{m}{1000} \times N_{P8}$

$V_P = \pi \times T_P \times \frac{m}{1000} \times N_{P8}$

$V_P = 1.055m$ m/sec

Design Power, P_D

From Table XVI – 1

$P_D = P_R \times K_1$

$P_R = 1$ HP = 746 Watt

From table XVI – 2

$K_1 = 1.80$

$P_D = 746 \times 1.80 = 1342.8$ Watt

Tangential tooth load ,F_T

$F_T = \frac{P_d}{V_p}$

F_T = 1272.19/ m N

From Table XVI – 10

Selecting material is Cast Steel

0.20% Carbon Heat Treated

Basic Stress of Pinion,

$S_o = S_{yt}$ = 196 N/mm^2

From Table XVI – 1

Bending Stress by Lewis Equation,F_B

$F_B = S_O \times C_V \times b \times y \times m$

From table XVI – 5

For Involute Gear, 20° full depth

Modified Lewis from factor, $Y = 0.485 - \frac{2.8}{T_p}$

Y = 0.075

From table XVI – 7

Face width, b = 10m mm or b = $D_P$mm

Assuming the Velocity Factor,C_V = 0.25

Bending Strength,

$F_B = S_O \times C_V \times b \times Y \times m$

$F_B = 36.75m^2$ (2)

Equation (1) and (2) i.e.

$F_T = F_B$

m^3 = 34.6175

Module m = 3.259 mm

$m^3 = \frac{1272.19}{36.75}$

F_T = 254.438N

From Table XVI – 7

Selecting Standard Module, m = 5 mm

Pitch Diameter, $D_P = T_P \times m$

D_P = 7 x 5 = 35mm

Face Width, b = D_P = 35mm.

Tangential tooth load, F_T

$F_T = \frac{1272.19}{m}$

F_T = 254.438N

Bending Strength, F_B

$F_B = 36.75m^2 = 36.75 \times 5^2$

F_B = 918.75 N

Pitch line velocity, V_P

V_P = 1.0555m = (1.0555x 5)/1000

V_P = 0.005277 m/sec

Pitch Peripheral Length= $\pi \times D_P$

= 109.955 ≈ 110 mm

B) RACK SHAFT

Total length of the rack shaft, L_T

L_T = 975 mm

Load applied on the rack shaft 75K_gF

Applied load = 75 x 9.81 = 735.75 N

From table II – 7

Selecting material of rack is carbon steel, SAE1030

S_{ut} = 527 MPa; $S_{yt} = S_{yc}$ = 296 MPa

E = 204 MPa

Effective length, L_E = 270mm

Total load,W= 735.75 N

Factor of safety, FOS = 2 (Assumed)

Area moment of Inertia, I

$I = \frac{\pi}{64} \times D^4$

Crippling load,

W_{Cr} = FOS x W = 2 x 735.75

W_{Cr} = 1471.5N

According to Euler formula

$W_{Cr} = \frac{n\pi^2 EI}{L^2}$

(n=2 –taking one end fixed and other hinged)

I = 26639.61 mm^4

D = 27.14 mm

From table XI – 4

Selecting available size, D = 28 mm

3. Design Calculations Of Shafts

A) SHAFT 1 (ratchet and larger sprocket)

Total Length = 225 mm

Power rated = 1 HP = 746Watt

Rotation = 1440rpm

Design power , P_D = 1342.8 Watt

Considering shaft subjected to twisting moment only.

So, neglecting the bending moment on the shaft.

From Table II – 7,

Selecting Carbon Steel, SAE 1030

S_{ut} = 527MPa; S_{yt} = 296MPa

S_{ys} = 183 MPa; E = 204MPa

Working Shear Stress, $\tau = \frac{S_{ys}}{FOS}$Taking FOS = 2

τ= 91.5 MPa

Torque transmitted by shaft, T

$T = \frac{P_D \times 60}{2\pi N}$

T = 8.9047 Nm

Torque transmitted by shaft, T

$T = \frac{\pi}{16} \times \tau \times d^3$

d = 7.91 mm

From Table XI – 7,

Selecting Dia. of Shaft, d = 10mm

B) SHAFT- 2 (small sprocket to pinion)

Total Length = 140 mm

Design power , P_D = 1342.8 Watt

Rotation = 2880rpm

From Table II – 7,

Selecting Carbon Steel, SAE 1030

S_{ut} = 527MPa; S_{yt} = 296MPa

S_{ys} = 183 MPa; E = 204MPa

Working Shear Stress, $\tau = \frac{S_{ys}}{FOS}$ Taking FOS = 2

τ = 91.5 MPa

Torque Transmitted by Shaft, T

$T = \frac{P_D \times 60}{2\pi N}$

T = 4.45 Nm

Torque transmitted by shaft, T

$T = \frac{\pi}{16} \times \tau \times d^3$

d = 6.28 mm

From Table XI – 7,

Selecting Dia. of Shaft, d = 10mm

4.Designcal. Of Pedal Mechanism

Applying Budhayan–Pythagoras Theorem

$AC^2 = AB^2 + BC$

AC = 854.28 mm

.cos θ = AB/AC

∴θ = 25.63°

180 = α + θ+ 90

∴α = 64.37°

W = 7350.75 N

W_T = Tangential Load

W_P= ParallelLoad

W_{TA} = W sin θ

W_{TA} = 318.25 N

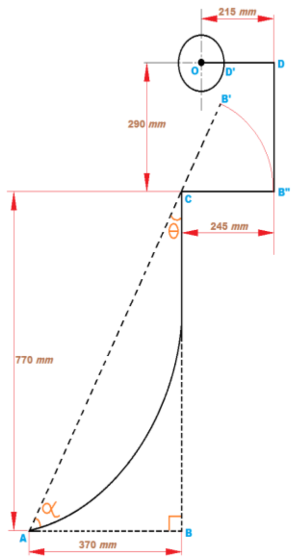

Figure 6. Line Diagram of Pedal Mechanism

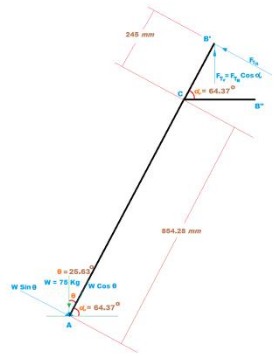

Figure 7. Forces on Pedal Link

Using Equilibrium Condition of Couple

WTA x 854.28 = WTB' x 245

WTB' =1109.7N

VerticleForce Acting on Point B

$F_{TV} = W_{TB'} \cos \alpha$

F_{TV} = 479.71 N= $W_{VB''}$

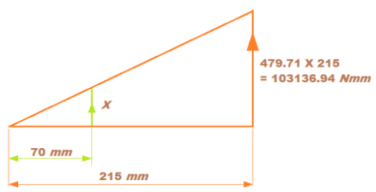

Figure 8. Torque Diagram

$W_{VB''} = W_{VD}$ = 479.71N

$T_{VD} = W_{VD} \times OD$

T_{VD} = 103136.9 Nmm

∴X = 33579.48 Nmm

Torque Developed at the Periphery of Ratchet

$T_{RATCHET} = T_{VD} = X = 33579.48$ Nmm

5.Ratchet And Pawl Mechanism

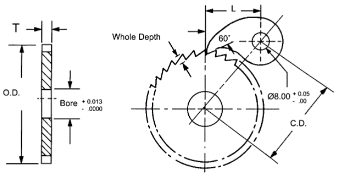

Figure 9. Ratchet and Pawl

Figure 10. Actual Diagram of Ratchet and Pawl

A) Ratchet

Force Applied on Ratchet Teeth, F = 479.71 N

Material Selected; Carbon Steel; SAE 1030

S_{ys} = 183 MPa   Taking FOS =6

Working Shear Stress,

$\tau = \frac{S_{ys}}{FOS} = 30.5$ MPa

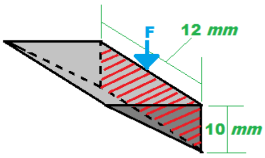

Figure 11. Force on Ratchet Tooth

Required Area of Base of Ratchet Tooth

$A_R = \frac{F}{\tau}$

$A_R = 15.73$ mm²

t = b = 4mm

Available Thickness, t = 10mm

Available Width, b = 12m

∴Available Area, $A_A = 120$ mm²

So, Ratchet Design is Safe.

B) PAWL

Force Applied on Ratchet Teeth,

F = 479.71 N

Material Selected; Carbon Steel; SAE 1030

S_{yc} = 296 MPa   Taking FOS = 6

$\sigma_c = \frac{S_{ys}}{FOS}$

∴σ_c = 49.33 MPa

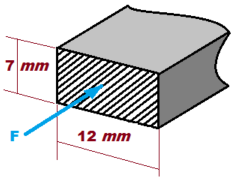

Figure – Force on Pawl

Required Cross Section Area,

$A_R = F/\sigma_c$

∴A_R = 9.72 mm²

t = b = 3.11 ≈ 4 mm²

Available Thickness, t = 7 m

Available Width, b = 12 mm

∴Available Area, A_A = 84 mm²

So, Ratchet Design is Safe.

III. CONCLUSION

The designs of various parts and parameters are taken into consideration and above values obtained successfully. These values are implemented of fabrication of same machine which is working successfully.

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