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 developing Agriwaste Briquette making machine, various component design of chain drive, design of connecting rod, design of ratchet and pawl, design of rack and pinion, pedal mechanism. 

Keywords: Were Designed And Fabricated chain Drive, Connecting Rod, Ratchet And Pawl, Rack And Pinion, Pedal Mechanism.

I. INTRODUCTION

The Agriwaste 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, \( Z_1 = 16 \)

Standard minimum pitch, \( p = 12.7 \text{mm} \)

C. C. distance between two shafts, \( a = 152 \text{mm} \)

Calculations

Velocity Ratio, \( i = \frac{Z_2}{Z_1} \)  

Teeth on Driven Sprocket, \( Z_2 = 32 \)

Pitch Diameter, \( D = \frac{p}{\sin\left(\frac{180}{Z_1}\right)} \)  

Pitch Diameter of driver sprocket, \( D_1 = 65 \text{mm} \)

Pitch Diameter of driven sprocket, \( D_2 = 129.5 \text{mm} \)

Number of links in the chain, \( L_n = 48.4788 \) i.e. \( L_n = 49 \) Numbers of links.

Length of chain, \( L = L_n \times p \)  

\( L = 622.3 \text{mm} \)

Now, From Table No.14.1

Selecting chain, ISO chain number 08B

Pitch, \( p = 12.7 \text{mm} \)

Roller diameter, \( d_{1 (\text{max})} = 8.51 \text{mm} \)

Width, \( b_{1 (\text{min})} = 7.75 \text{mm} \)

Transverse pitch, \( p_t = 13.92 \text{mm} \)

Power rating of chain = \( \frac{\text{POWER TRANSMITTED} \times K_s}{K_1 \times K_2} \)

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 = \( \frac{\text{POWER TRANSMITTED} \times K_s}{K_1 \times K_2} \)

Power to be transmitted = 0.2234 K

Top diameter (Addendum Diameter)- \( D_a \)

From Table No.14.6

\( D_a (\text{max}) = D_2 + 1.25p - d_1 \)

\( D_a (\text{max}) = 136.865 \text{mm} \)

\( D_a (\text{min}) = D_1 + p(1 - \frac{1}{Z_1}) - d_1 \)

\( D_a (\text{min}) = 67.92 \text{mm} \)

Top diameter of driven sprocket, \( D_{a2} = 136.865 \text{mm} \)

Top diameter of driver sprocket, \( D_{a1} = 67.92 \text{mm} \)

Root diameter (Dedendumdia), \( D_f \)

\( D_f = D - 2ri \)

Roller seating radius, \( r_i \)

\( (r_i)_{\text{max}} = 0.505d_1 + 0.069 \times 3\sqrt{d_1} \)

\( (r_i)_{\text{max}} = 4.438 \text{mm} \)

\( (r_i)_{\text{min}} = 0.505d_1 \)

\( (r_i)_{\text{min}} = 4.297 \text{mm} \)

\( D_f (\text{min}) = D_1 - 2r_{i\text{min}} \)

\( D_f (\text{min}) = 56.406 \text{mm} \)

\( D_f (\text{max}) = D_2 - 2r_{i\text{max}} \)

\( D_f (\text{max}) = 120.624 \text{mm} \)

Tooth flank Radius, \( r_e \)
(r_D) Driven = 0.008 d_1(Z_2^2+180)
(r_D) Driven =81.968 mm
(r_D) Driver = 0.12d_1(Z_1 + 2)
(r_D) Driver = 18.38 mm
Roller Seating Angle, α

For, Driven Sprocket, α_Driven

α_Driven=(120 \ \frac{90}{Z_2})
α Driven=117.2°

For, Driver Sprocket, α_Driver

α_Driver=(140 \ \frac{90}{Z_1})
α Driver= 134.4°

Tooth height above the pitch polygon, h_a
For Driven Sprocket, (h_a)_Driven

(h_a) Driven = 0.625p – 0.5d_1 + \frac{0.1p}{Z_2}

(h_a) Driven = 4 mm

For Driver Sprocket, (h_a) Driver

(h_a) Driver = 0.5(p – d_1)

(h_a) Driver = 2 mm

Tooth side radius, r_x

r_x (min) = p = 12.7 mm

Tooth width, b_f

b_n = 0.93 b_1 if p \leq 12.7 mm
b_n = 0.95 b_1 if p > 12.7 mm
b_n = 0.93 b_1
b_n = 7.2 mm

Tooth side relief, b_a

b_a= 0.1p to 0.15p

Taking b_a= 0.15p

= 1.9 mm

Power transmitted by chain drive

\text{Power transmitted by chain drive} = \text{Force caused by driver shaft rotation} \times \text{average (minimum) velocity of chain drive}

= 50 \times 9.81 = 490.5 N

223.4 = 490.5 \times \text{minimum velocity of chains drive, } V

= 0.455 \text{ m/sec.}

\text{2. Design Calculations Of Rack And Pinion}

Selecting profile type of tooth is 20° full depth

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

\text{A) Pinion}

Pitch diameter of Pinion, D_P = T_P \times m \text{ mm.}

Since, N_P = 2880 rpm.

\text{Figure 4. Proposed Rack and Pinion}

\text{Figure 5. Actual diagram of Rack and Pinion}

V_P = \pi \times T_P \times \frac{m}{1000} \times N_P \times s

V_P = \pi \times T_P \times \frac{m}{1000} \times N_P \times s

V_P = 1.055 \text{ m/sec}

Design Power, P_D

From Table XVI – 1

P_D = P_R \times K_1

P_R= 1 \text{ HP} = 746 \text{ Watt}

From table XVI – 2

K_1 = 1.80

P_D = 746 \times 1.80 = 1342.8 \text{ 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, \( b = 10mm \) or \( b = D_p mm \)
Assuming the Velocity Factor, \( C_v = 0.25 \)
Bending Strength,
\[ F_B = 36.75 \times m^2 \]
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, \( D_p = T_p \times m \)
\[ D_p = 7 \times 5 = 35mm \]
Face Width, \( b = D_p = 35mm. \)
Tangential tooth load, \( F_T \)
\[ F_T = \frac{1272.19}{m} \]
\[ F_T = 254.438N \]
Bending Strength, \( F_B \)
\[ F_B = 36.75 \times m^2 = 36.75 \times 5^2 \]
\[ F_B = 918.75 \] N
Pitch line velocity, \( V_p \)
\[ V_p = 1.0555m = (1.0555 \times 5)/1000 \]
\[ V_p = 0.005277 \text{ m/sec} \]
Pitch Peripheral Length = \( \pi \times D_p \)
\[ \pi = 109.955 \approx 110 \text{ mm} \]
B) RACK SHAFT
Total length of the rack shaft, \( L_T \)
\[ L_T = 975 \text{ mm} \]
Load applied on the rack shaft 75Kgf
Applied load = 75 \times 9.81 = 735.75 N
From table II – 7
Selecting material of rack is carbon steel, SAE1030
\[ S_m = 527 \text{ MPa; } S_{yt} = S_{yc} = 296 \text{ MPa} \]
\[ E = 204 \text{ MPa} \]
Effective length, \( L_e = 270mm \)
Total load, \( W = 735.75 \text{ N} \)
Factor of safety, \( FOS = 2 \) (Assumed)
Area moment of Inertia, \( I \)
\[ I = \frac{\pi}{64} \times D^4 \]
Crippling load,
\[ W_{cr} = FOS \times W = 2 \times 735.75 \]
\[ W_{cr} = 1471.5 \text{ N} \]
According to Euler formula
\[ W_{cr} = \frac{n \pi^2 E I}{L^2} \]
\[ (n=2 \text{ – taking one end fixed and other hinged}) \]
\[ I = 26639.61 \text{ mm}^4 \]
\[ D = 27.14 \text{ mm} \]
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 = 746 Watt
Rotation = 1440rpm
Design power , \( P_D = 1342.8 \text{ Watt} \)
Considering shaft subjected to twisting moment only.
So, neglecting the bending moment on the shaft.
From Table II – 7,
Selecting Carbon Steel, SAE 1030
\[ S_m = 527 \text{ MPa; } S_{yt} = S_{yc} = 296 \text{ MPa} \]
\[ S_{sy} = 183 \text{ MPa; } E = 204 \text{ MPa} \]
Working Shear Stress, \( \Sigma = \frac{S_{sy}}{FOS} \)
Taking FOS = 2
\[ \Sigma = 91.5 \text{ MPa} \]
Torque transmitted by shaft, \( T \)
\[
T = \frac{P_D \times 60}{2\pi N}
\]
\( T = 8.9047 \) Nm

Torque transmitted by shaft, \( T \)
\[
T = \frac{\pi}{16} \times \bar{T} \times d^3
\]
\( d = 7.91 \) mm

From Table XI – 7,
Selecting Dia. of Shaft, \( d = 10 \) mm

**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
\( S_{eq} = 527 \) MPa; \( S_{yt} = 296 \) MPa
\( S_{ys} = 183 \) MPa; \( E = 204 \) MPa

Working Shear Stress, \( \bar{T} = \frac{S_{eq}}{\text{FOS}} \)
Taking FOS = 2
\( \bar{T} = 91.5 \) MPa

Torque Transmitted by Shaft, \( T \)
\[
T = \frac{P_D \times 60}{2\pi N}
\]
\( T = 4.45 \) Nm

Torque transmitted by shaft, \( T \)
\[
T = \frac{\pi}{16} \times \bar{T} \times d^3
\]
\( d = 6.28 \) mm

From Table XI – 7,
Selecting Dia. of Shaft, \( d = 10 \) mm

### 4. Designcal. Of Pedal Mechanism

Applying Budhayan–Pythagoras Theorem

\( AC^2 = AB^2 + BC \)

\( AC = 854.28 \) mm

\( .\cos \theta = AB/AC \)

\(. \cdot \theta = 25.63^\circ \)

\( 180 = \alpha + \theta + 90 \)

\( . \cdot \alpha = 64.37^\circ \)

\( W = 7350.75 \) N

\( W_T = \text{Tangential Load} \)

\( W_P = \text{ParallelLoad} \)

\( W_{TA} = W \sin \theta \)

\( W_{TA} = 318.25 \) N

Using Equilibrium Condition of Couple

\( W_{TA} \times 854.28 = W_{TB'} \times 245 \)

\( W_{TB'} = 1109.7 \) N

Verticle Force Acting on Point B

\( F_{TV} = W_{TB'} \cos \alpha \)

\( F_{TV} = 479.71 \) N = \( W_{VB'} \)

\[
W_{VB'} = W_{VD} = 479.71 \text{ N}
\]

\( T_{VD} = W_{VD} \times OD \)

\( T_{VD} = 103136.9 \) Nmm

\( . \cdot X = 33579.48 \) Nmm
Torque Developed at the Periphery of Ratchet

\[ T_{RATCHET} = T_{VD} = X = 33579.48 \text{ Nmm} \]

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, \( F = 479.71 \text{ N} \)

Material Selected; Carbon Steel; SAE 1030

\( S_y = 183 \text{ MPa} \) Taking FOS = 6

Working Shear Stress,

\[ \tau = \frac{S_y}{FOS} = 30.5 \text{ MPa} \]

Figure 11. Force on Ratchet Tooth

Required Area of Base of Ratchet Tooth

\[ A_R = \frac{F}{\tau} \]

\[ A_R = 15.73 \text{ mm}^2 \]

Available Thickness, \( t = b = 4\text{mm} \)

Available Width, \( b = 12\text{mm} \)

\( \therefore \) Available Area, \( A_A = 120 \text{ mm}^2 \)

So, Ratchet Design is Safe.

B) PAWL

Force Applied on Ratchet Teeth, \( F = 479.71 \text{ N} \)

Material Selected; Carbon Steel; SAE 1030

\( S_y = 296 \text{ MPa} \) Taking FOS = 6

\( \sigma_c = \frac{S_y}{FOS} \)

\( \therefore \sigma_c = 49.33 \text{ MPa} \)

Figure – Force on Pawl

Required Cross Section Area,

\[ A_R = \frac{F}{\sigma_c} \]

\( \therefore A_R = 9.72 \text{ mm}^2 \)

\( t = b = 3.11 \approx 4 \text{ mm} \)

Available Thickness, \( t = 7 \text{ mm} \)

Available Width, \( b = 12 \text{ mm} \)

\( \therefore \) Available Area, \( A_A = 84 \text{ mm}^2 \)

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.

IV. REFERENCES


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