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Design of pedal operated maize sheller

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Abstract
Maize shelling manually is a very exhausting and time-consuming task. In this operation human palm and fingers got injured during maize shelling which decreases the efficiency of the farm workers. An appropriate technology is urgently needed for the tribal people to maximize profit. The main aim of this research is to design and fabricate various parts of pedal operated maize sheller to improve design quality and durability of the product. The performance of the machine was evaluated in terms of shelling efficiency, damage percentage and output capacity. Overall, this research includes processes like design, fabrication and assembling of various components etc.

Keywords: Maize sheller, design calculations, maize

Introduction
Maize (*Zea mays*) is the second most-sought crop after paddy preferred by farmers in Chhattisgarh. In tribal districts of Chhattisgarh, the farmers grow maize and just after harvesting early maize, taking up oilseed crops. At present in Chhattisgarh total area, production, productivity was 119.6 thousand ha, 306.98 metric tonnes and 2.566 t/ha, respectively (Annon, 2020). After maturity the maize cobs are harvested manually by hand plucking. It was dried to a moisture level of 15-21% before shelling. In Chhattisgarh, maize is generally cultivated in the tribal area like Bastar and Surguja. In these areas proper electricity is not available. Hence, there is need to design and fabricate a non-electric pedal operated maize sheller in this project.

Design Calculations
The following design procedure were considered in manufacturing and assembling of pedal operated maize sheller.

Pitch circle diameter
The relation between pitch and pitch circle diameter was calculated by following formula (Khurmi *et al.*, 2008) [2]

$$p = d sin\frac{180}{T} - - - (1.1)$$

Pitch circle diameter of driving sprocket was calculated as follows:

$$p = d sin\frac{180}{T_1}$$

$$13 = d sin\frac{180}{48}$$

Hence, d = 200 mm

Pitch circle diameter of driven sprocket

$$p = d sin\frac{180}{T_2}$$

$$13 = d\sin\frac{180}{15}$$

Hence, d = 62.5 mm

Velocity ratio of the chain drives

The velocity ratio of the chain drive is given by following expression (Khurmi *et al.* 2005)

$$\mathrm{VR} = \frac{N_1}{N_2} = \frac{T_2}{T_1} - - - (1.2)$$

The number of teeth in driving and driven sprocket was 44 and 18, respectively.

Therefore, velocity ratio

$$\mathrm{VR} = \frac{T_2}{T_1} = \frac{48}{15} = 3.20$$

Given speed of driving sprocket, N_2 = 35 rpm

So,

$$\frac{N_1}{35} = 3.20$$

N_2 = 3.20 X 35 =112 RPM

Centre distance between the chain drives

The centre distance between the centers of two sprockets is given by

$$X = \frac{p}{4}\left[K - \left[\frac{T_1 + T_2}{2}\right] + \sqrt{\left\{K - \frac{T_1 + T_2}{2}\right\}^2 - 8\left\{\frac{T_2 - T_1}{2\pi}\right\}^2}\,\right] - - - (1.3)$$

$$K = \left\{\frac{T_1 + T_2}{2}\right\} + \left\{\frac{2X}{p}\right\} + \left\{\left(\frac{T_2 - T_1}{2\pi}\right)^2 \frac{p}{X}\right\} - - - (1.4)$$

$$K = \left\{\frac{48 + 15}{2}\right\} + \left\{\frac{2 \times 980}{13}\right\} + \left\{\left(\frac{15 - 48}{2\pi}\right)^2 \frac{13}{980}\right\}$$

K = 182

The length of the chain was calculated by following formula

$L = K \times p - - - (1.5)$

$L = 182 \times 13 = 2366$ mm

Calculation of power, force, torque, and angular velocity on the pedal powered unit

The angular velocity from the sprocket will be calculated as follow,

$$\omega = \frac{2\pi N}{60} - - - (1.6)$$

$$\omega = \frac{2\pi \times 112}{60} = 11.72 \text{ rad/s}$$

Peripheral velocity of sprocket is given by

$$V = \frac{\pi d N}{60} - - - (1.7)$$

$$V = \frac{\pi \times 0.063 \times 112}{60} = 0.370\ m/s$$

And the power on the sprocket will be given as

$P = F \times V - - - (1.8)$

$P = 200 \times 0.370 = 74$ watts

Torque delivered by chain sprocket system can be calculated by following equation

$$T = \frac{P \times 60}{2\pi N} - - - (1.9)$$

$$T = \frac{74 \times 60}{2\pi \times 112} = 6.31\ N - m$$

Design of threshing element (flail)

The following parameters were selected during design of the threshing element i.e. flail – (1). Force required for detaching the maize grains from maize cob ranges between 15 to 21 N (Sharma *et al.*, 2007) [3]. (2) Average pedalling speed was considered to be 35 rpm. (3) The number of teeth in driving and driven sprocket was 48 and 15, respectively.

Therefore, velocity ratio, VR

VR = 48/15= 3.20

Given speed of driving sprocket, N_1 = 35 rpm

So, N_2/N_1 = 3.20

N_2 = 3.20 X 35 = 112 RPM

Since driven sprocket is attached to bearing and shaft therefore speed of shaft is equal to speed of driven sprocket i.e., 112 RPM.

Power is given by,

$$P = \frac{2\pi N_2 T}{60}$$

$$T = \frac{74 \times 60}{2\pi \times 112}$$

T = 6.31 N-m

Since angular velocity

$$\omega = \frac{2\pi N}{60} = \frac{2 \times 3.14 \times 112}{60}$$

ω = 11.72 rad/s

The iron block strikes with the unshelled cobs with tangential force F at an angle θ can be divided into two components i.e., vertical component F_N and horizontal component F_H and can be expressed as

$$F_N = F.\sin\theta - - - (1.10)$$

$$F_H = F.\cos\theta - - - (1.11)$$

Tangential force F causes impact force to shell the maize cobs. According to Newton's second law

$$F = m.a_t - - - (1.12)$$

And

$$a_t = m.R.\varepsilon - - - (1.13)$$

By resolving the force,

$$F = F_c.\tan\theta - - - (1.14)$$

and we know that centrifugal force F_c is given by

$$F_c = m\omega^2 R - - - (1.15)$$

If, $\Theta = 45^{o}$

Then,

$$F = F_c = m\omega^2 R - - - (1.16)$$

Assuming that the force required to detach the maize grains from the cob was 21 N

$$\text{So}, 21 = m \times (11.72)^2 \times 0.2$$

$$m = \frac{21}{11.72 \times 11.72 \times 0.2}$$

m = 0.910 kg

Rectangular section was taken for the further design. Since there are 21 flails in the shaft. Therefore, the mass of single block

$$= \frac{0.910}{21} = 0.043 \text{ kg}$$

= 50 g (aprox)

Size of the square block
Mass = density × volume
0.050 = 7850 × V
V = 6.37 × 10^{-6} m^3
Assuming length of the square block, l = 2b
V = l × b × b
V = 2 b^3
b = 0.014 m = 1.5cm
l = 2.5 b = 3 cm

Fig 1(a): Pictorial view of the inside of the shelling drum of maize sheller

Design of hopper
Upper length, L = 40, Lower length, L = 40
Upper width, A =38, Lower width, B = 26
Height of the hopper, H = 20

The volume of the trapezoidal hopper is given by following formula

$$\text{V} = \frac{1}{2}(\text{A} + \text{B}) \times \text{L} \times \text{H} - - - - - - (1.17)$$

$$\text{V} = \frac{1}{2}(0.26 + 0.38) \times 0.40 \times 0.20$$

V = 0.0256 m^3

Then the capacity of the hopper can be calculated as

Capacity, kg = $Density \times Volume$ = 750 × 0.0256 = 19.2 kg

Design of shaft
According to maximum shear stress theory, the maximum shear stress in the shaft

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2} - - - (1.18)$$

Torsional stress is given by

$$\tau = \frac{T \times r}{J} - - - (1.19)$$

$$\tau = \frac{T \times \left(\frac{d}{2}\right)}{\left(\frac{\pi}{32}\right) d^4} = \frac{16 \times T}{\pi \times d^3}$$

$$T = \frac{\pi \times \tau \times d^3}{16} - - - (1.20)$$

And,
Bending stress is given by

$$\sigma_b = \frac{M \times y}{I} - - - (3.45)$$

$$\sigma_b = \frac{M \times \left(\frac{d}{2}\right)}{\left(\frac{\pi}{64}\right) d^4} = \frac{32 \times M}{\pi \times d^3}$$

$$M = \frac{\pi \times \sigma_b \times d^3}{32} - - - (1.21)$$

Substituting the value of τ and σ_b in equation (1.18)

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_b^2 + 4\tau^2}$$

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2 + T^2} - - - (1.22)$$

$$\frac{\tau_{max} \times \pi \times d^3}{16} = \sqrt{M^2 + T^2} - - - (1.23)$$

The expression $\sqrt{(M^2+T^2)}$ is known as equivalent twisting moment and denoted by T_e.

$$T_e = \frac{\tau_{max} \times \pi \times d^3}{16} = \sqrt{M^2 + T^2} --- (1.24)$$

Bending moment (M) acting on the different point of shaft can be calculated as follows:

1. Bending moment at length BC

$$M_{BC} = \frac{(W_c \times L_c)}{8} --- (1.25)$$

$$M_{BC} = \frac{(66 \times 0.86)}{8}$$

$$M_{BC} = 56.58\ N - m$$

2. Bending moment on shaft at point B due to chain tension

The bending moment at point B due to chain

M_{AB} = 155.69 x 0.140

M_{AB} =21.02 N-m

For safe design of shaft, the higher bending moment was considered.

Therefore,

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2 + T^2} --- (1.26)$$

$$72 \times 10^6 = \frac{16}{\pi d^3}\sqrt{(58.58)^2 + (6.31)^2}$$

d = 0.016 m = 16 mm

factor of safety [FOS]= 3

Hence the shaft diameter of 50 mm was chosen.

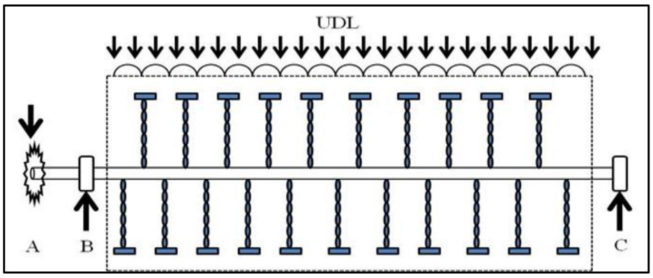

Fig 1(b): Sketch diagram of the shelling shaft with flails

Determination of pedalling force

The total mass act on the threshing unit is:

$M = M_{sp} + M_s + M_f + M_o$

Where, M_{sp} =mass of smaller sprocket, M_s = mass of shaft, M_f = Total mass of flails and M_o = Other consideration of mass

M = 1.2 + 15.50 + 6.7 + 2 = 25.40 kg

The pedalling force is than calculated as follows:

$$F = Mw^2R --- (1.27)$$

$$w = \frac{\pi d N_2}{60}$$

$$w = \frac{\pi \times 0.4 \times 112}{60}$$

$w = 2.34\ m/s$

Now,

$F = 25.40 \times 2.34^2 \times 0.4$

$F = 55.63$ N

Fabrication

Hopper

Shaft

Shelling element (Flail)

Pedaling Unit

Mainframe

Working principle of the machine

The developed maize sheller in the present study is worked on the principle of impact force which was applied through revolving flails i.e., an iron block was attached to the rotating shaft by chain with oval links. The maize sheller is powered by pedal operated power transmission system by applying force to rotate the pedal. The motion of the pedal provides angular velocity to the threshing shaft through chain and sprocket system. So, when cycle pedal is rotated the freely revolving flails revolve around the shaft and give the impact force for shelling of the maize cobs.

Conclusion

The design of different components such as; the shelling drum size, shelling unit mechanism, power required for shelling, power transmission system and design of hopper was considered for easy operation of the machine. According to this design procedure, we can fabricate pedal operated maize sheller which is useful, feasible, safe, cost effective, eco-friendly and productive for the tribal farmers of Chhattisgarh. After complete design, the pedal operated machine was fabricated at workshop of the Department of Farm Machinery and Power Engineering, IGKV, Raipur in the year 2018. This implement may be useful towards saving of time and labour requirement. Since the pedaling force of the human being was 150 N as cited in literature and the calculated pedaling force of the machine was found to be 55.63 N. Hence, it was concluded that the machine could be easily operated by human being easily.

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