

The Design and Construction of Maize Threshing Machine

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Abstract

Many farmers grow maize but could not afford the cost of acquiring some of the imported threshing machines because of their cost. Such people resort to manual means of threshing which results into low efficiency, high level of wastage and exerting of much labor. This machine was constructed to shell maize and separate the cob from the grains. It was constructed from locally available materials and its cost is very low and affordable. Its threshing efficiency is 99.2% and breakage is very insignificant, as well as losses.

Keywords: Hopper, shutter, threshing chamber, breakage, collector, wastage.

Introduction

Grains, according to Okaka (1997) are fruits of cultivated grasses belonging to the monocotyledonous family, Gramineae. The principal cereal grains of the world include wheat, barely, rye, sorghum, rice and maize. The last has become a popular staple in West Africa. Maize is another world's most versatile seed crop. Its cultivation originated from Europe but was soon brought to Africa by explorers early in the sixteenth century. Within hundreds of years, it was well established as a staple food in areas around the north and south shores of Mediterranean Sea. In later years, maize cultivation spread widely into Africa down to Nigeria as well as many parts of Asia all at the same span of time. Its production in the southern states of the United States of America also expanded greatly just as it was in Africa and Asia (Adaokoma 2001). The use of sticks for threshing was predominant in the pre-historic era. In Egypt, livestock was earlier employed for threshing out grains after which it was winnowed. In Palestine, threshing sledge was used 3,000 years earlier. In Nigeria, maize was threshed originally by bare hands. Other popular method was the use of pestle and mortar. This method is still used in the rural areas today. The above methods became

unsatisfactory because of their low output, tediousness and their requirement of extra strength. According to Kaul and Egbo (1985), the performance of a thresher depends upon its size, cylinder speed, cylinder concave clearance, fan speed and the sieve shaker speed. Oni and Ali (1986) reported that the factors influencing threshability of maize in Nigeria are field drying, maize varieties, ear size cylinder speed and feed rate. The properties of the crop that affect the thresher performance are crop variety, shape and size, hardness of the seed, the moisture content of the seed and the density.

Design Analysis

The Hopper

The hopper (Fig. 1) is designed to be fed in a vertical position only. The material used for the construction is mild steel sheet metal, which is readily available in the market and relatively affordable. The hopper has the shape of a frustum of a pyramid truncated at the top, with top and bottom having rectangular forms.

Calculation of the Overall Height

Lengths AC and EG are calculated by

using the Pythagorean theorem,
 $AC^2 = AB^2 + BC^2$. Also, $AC = (AB^2 + BC^2)^{1/2}$,
 and $OC = (AB^2 + BC^2)^{1/2}/2$,
 $EG^2 = EH^2 + HG^2$, $EG = (EH^2 + HG^2)^{1/2}$,
 and $MG = [(EH^2 + HG^2)/2]^{1/2}$.

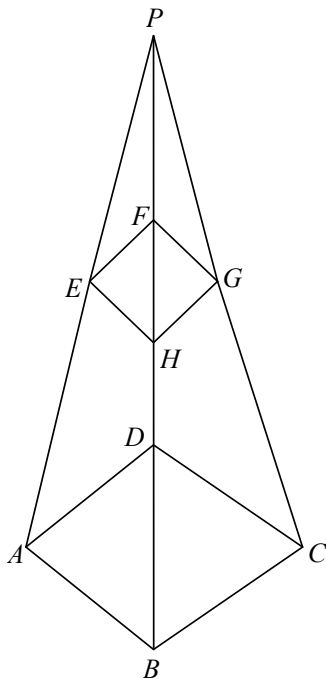


Fig. 1. Schematic diagram of the hopper.

From the principle of similar triangles, for triangles PMG and POC :

$$PM/MG = PO/OC, \text{ or } PM = PO \times MG/OC.$$

Then the volume of the hopper is given by:

$$V_{\text{hopper}} = [(\text{area of base}) \times \text{height}]/3 \\ = [(AB \times BC) \times h - (EH \times HG) \times x]/3,$$

where:

h = overall height;

x = height of the truncated top.

The Main Frame

The main frame supports the entire weight of the machine. The total weights carried by the main frame are:

- weight of the hopper and housing;
- weight of the threshing chamber;
- the collector and pot; and
- the bearings and pulleys.

The two design factors considered in determining the material required for the frame are weight and strength. In this work, angle steel bar of $1\frac{1}{2}$ " by $1\frac{1}{2}$ " and 2mm thickness is used to give the required rigidity.

The Threshing Bars

Weight, W , of threshing bar is given by:

$$W = mg,$$

where:

m = mass of threshing bar;

g = acceleration due to gravity.

Mass, m , of threshing bar:

$$m = \rho \times V,$$

where:

ρ = density of mild steel;

V = volume of threshing bar.

Volume, V , of threshing bar:

$$V = l \times b \times h,$$

where:

l = length;

b = breadth;

h = height.

Shaft Design

A shaft is a rotating or stationary member, usually of circular cross-section having such elements as gears, pulleys, flywheels, cranks, sprockets and other power transmission elements mounted on it Shigley (1986). The shaft of this machine has a threshing tool attached to it (by welding) at two opposing sides and a pulley mounted on it. It is supported on bearings. Shaft design consists primarily of the determination of the correct shaft diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading conditions. Shafts are either solid or hollow. The following presentation is based on shafts of ductile materials and circular cross-section. The length of the shaft has been pre-determined at 770mm.

Power Delivered By Shaft

Power, P = work done per second:

$$P = (\text{work done})/\text{time} = (\text{force} \times \text{distance})/\text{time} \\ = \text{force} \times \text{velocity}.$$

Calculation of Reactions R_B and R_D

From Fig. 2:

X_T = total length of shaft;

X = distance along the shaft;

W_p = weight of the pulley;

T_1 & T_2 = tensions of belt.

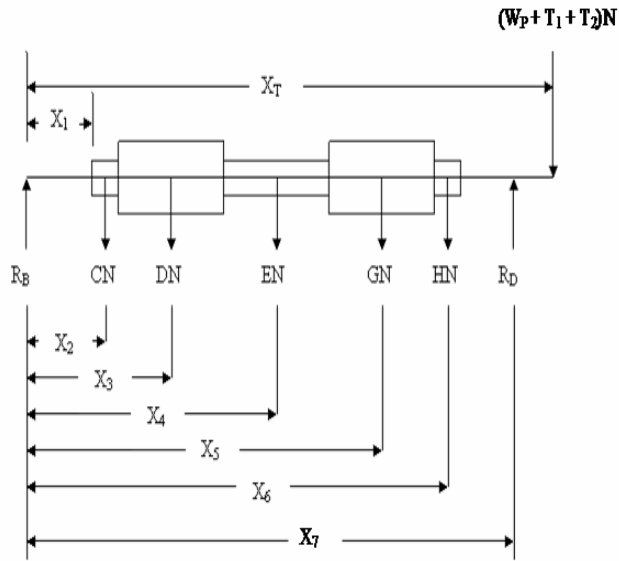


Fig. 2. Distributed weights acting on the shaft at different sections of the shaft.

Taking moment about R_B :

sum of clockwise moments equal = sum of anticlockwise moments, i.e.,

$$(W_P + T_1 + T_2)X_T + Hx_6 + Gx_5 + Ex_4 + Dx_3 + Cx_2 = R_Dx_7,$$

$$R_D = [(W_P + T_1 + T_2)X_T + Hx_6 + Gx_5 + Ex_4 + Dx_3 + Cx_2]/x_7.$$

But sum of upward forces = sum of downward forces:

$$R_D + R_B = W_P + T_1 + T_2 + C + D + E + G + H,$$

$$R_B = (W_P + T_1 + T_2 + C + D + E + G + H) - R_D.$$

Calculation of the Shearing Force and Bending Moment of the Shaft at Different Sections of the Shaft

Here:

S.F. = upward forces – downward forces;

B.M = forces x perpendicular distances.

Force Required to Thresh Maize along the Length of the Threshing Bars

The threshing bars, which are attached to the shaft contained in the threshing chamber, rotate with the shaft, giving rise to centripetal force:

$$F = m \omega^2 r,$$

where:

F = centripetal force;

m = mass of threshing bars;

ω = angular velocity;

r = radius of the arm of the threshing bar.

To determine the mass, m , of the threshing bars:

mass (m) = density (ρ) x volume (V), and volume (V)

= length (L) x breadth (B) x thickness (t).

Determination of Angular Velocity, ω

The angular velocity, ω , is given by:

$$\omega = 2\pi N/60,$$

where:

N = speed of the shaft in r.p.m.

The Radius, r , of the Threshing Arm

The radius, r , of the threshing arm increases along the length of the shaft and also decreases towards the other end of the shaft, where:

r = radius of threshing arm and it is given as

$r_{max} = 0.082m$ (assumed),

$r_{min} = 0.058m$ (assumed), so that

centripetal force at r_{max} (F) = $m\omega^2 r_{max}$,

centripetal force at r_{min} (F) = $m\omega^2 r_{min}$.

Determination of Threshing Torque

The torque, T , is given by:

$$T = F \times r,$$

where:

F = Force available along the threshing bar;

r = threshing radius.

Determination of Power Delivered by Shaft along the Length of Threshing Bars

The power is given by

$$power = energy/time = (work done)/time = (force \times distance)/time = force \times velocity,$$

$$velocity = \omega r,$$

where:

ω = angular velocity;

r = radius.

$$\text{Therefore, } power = F \omega r.$$

Determination of Torsional Moment, M_T

The torsional moment, M_T , is given by

$$M_T = 9,550 \times KW/N,$$

where:

KW = power delivered;

N = revolutions per minute.

Determination of Maximum Bending Moment

The maximum bending moment, M_{bmax} , is given by $M_{bmax} = (MV^2 + MH^2)^{1/2}$.

The Prime Mover

The purpose of an electric motor is to develop the necessary power required for a task.

The most common types of motors used for industrial purpose are: the squirrel cage induction motors where a continuous supply of power is required: and the split-ring induction motors being used for intermitted supply of power. The squirrel type motor has its current induced in the motor bars by induction thereby dispensing with commutators. For the purpose of this work, a squirrel induction motor is selected. The rating however will be determined by calculation results.

Pulley

i. Driver Pulley: Williams (1953), gives the horse power rating at a maximum pitch diameter of pulleys and the corresponding speeds. The horsepower rating of the electric motor will therefore determine the diameter of the driver pulley.

ii. Driven Pulley: The spindle speed and the speed of the prime mover are related by the expression:

$N_1D_1 = N_2D_2$, or $N_1/N_2 = D_2/D_1$, i.e.,
 (speed of driver)/(speed of driven)
 = (diameter of driven)/(diameter of driver).

The following factors determined the centre distance of pulleys:

- the class of V-belt used;
- the configuration of the machine;
- the space available.

However, the two pulleys must be near to each other.

iii. Weight of Pulley: The weight of pulley on a shaft can be determined as follows:

weight of pulley, $W_p = mg$, with

$m = \rho V$,

$V = A \times L_p = (\pi d^2/4) \times L_p$,

$m = \rho \times (\pi d^2/4) \times L_p$,

$W_p = \rho \times (\pi d^2/4) \times L_p g$,

where: L_p = length of pulley.

Belt Design

A belt provides a convenient means of transferring power from one shaft to another. Belts are frequently necessary to reduce the higher rotational speeds of electric motors to lower values required by mechanical equipments (Spotts 1985). The belt driver relies on frictional effects for its efficient operation. When the belt connecting two pulleys is stationary, the tensions in the two portions of the belt are equal but when torque is applied to the driving pulley, one portion of the belt is stretched and the other portion becomes slack. The procedure for selecting a V-belt drive is dependent on the motor horse power and the speed (rpm) rating. V-belts are rated from class A to E (Table 1).

The effective pull on belt:

$T = T_1 - T_2$,

where:

T_1 = tension on tight side;

T_2 = tension on slack side;

Torque, T_s , on shaft = $F \times r$, and $F = T_s/r$.

Table 1. V-belt horse power*.

Cross section	Load of drive (Kw)	Recommended min. pulley pitch diam., d (mm)	Nominal top width, w (mm)	Nominal thickness, T (mm)	Weight per meter (kg f)
A	0.75 – 5	75	13	8	0.106
B	2-15	125	17	11	0.189
C	7.5-75	200	22	14	0.343
D	22-50	355	32	19	0.596
E	30-190	500	38	22	—

* Design data (PSG Tech 1989)

But $T = F$ and $T = T_s/r$.

Also, $T_s = T_M$, and $T = T_M/r$.

But $P_M = \omega T_M$, and $T_M = P_M/\omega$.

Therefore, $T = P_M/(\omega r)$, and $M_T = (T_1 - T_2)r_1$, where:

T = effective pull on bell;

F = Force on driver pulley;

T_s = torque on shaft;

T_M = Torque on the motor;

r_1 = radius of driver pulley;

T_1 - Tension on the tight side of belt;

T_2 = Tension on the slack side of belt;

M_T = Torsional moment.

Power Transmitted by Belt

According to Hannah and Stephens (1970), the power transmitted by belt is given by $P = (T_1 - T_2)V$.

But $V = (\pi DN)/60$. Also,

$T_1/T_2 = \exp(\mu\theta \operatorname{cosec} \beta)$,

where:

β = groove semi-angle;

θ = angle of lap;

α = angle of contact at the smaller pulley;

μ = coefficient of friction.

The coefficient of friction for rubber belt on cast iron or steel operating on dry surface is $\mu = 0.3$ (Ogunwede 2003). The angle of lap for open V-belt drive is given as:

$\theta = (180 - 2\alpha) \times \pi/180$ rad. Also,

$\sin \alpha = (r_2 - r_1)/x$,

where:

x = distance between pulleys;

r_1 = radius of smaller pulley;

r_2 = radius of bigger pulley.

Centre Distance of Belt

The maximum centre distance of belt in use depends on the size of the two pulleys and is given as:

$X_{max} = 2(D + d)$.

Length of Belt

Belt length can be calculated if the diameters of both the bigger pulley and the smaller pulley and the belt centre distance are known:

$L = 2x + (\pi/2)(D + d) + (D - d)^2/(4x)$.

Groove Dimension

The groove for pulleys has been standardized depending on the type of class of belt intended for use (Williams 1953). Classes of belts, pitch diameters of pulleys, and the groove angles have been listed out by Williams (1953).

Determination of Minimum Diameter of Shaft

Design of shafts of ductile material based on strength is controlled by maximum shear theory. Shafting is usually subjected to torsion, bending and axial loads. For a solid shaft having little or no axial loading, the ASME code equation is given as (ASME 1995):

$d^3 = [16/(\pi S_s)] \times [(K_b M_b)^2 + (K_t M_t)^2]^{1/2}$,

where:

d = diameter of the shaft;

M_t = torsional moment;

M_b = bending moment;

K_b = combined shock and fatigue factor applied to bending moment;

K_t = combined shock and fatigue factor applied to torsional moment;

S_s = Allowable Stress.

For rotating shafts, when load is suddenly applied (minor shock):

$K_b = 1.5$ to 2.0 ;

$K_t = 1.0$ to 1.5 .

For shaft without key-way:

S_s (allowable) = 55 MN/m^2 .

For shaft with key-way:

S_s (allowable) = 40 MN/m^2 .

A shaft with key-way was used for this work.

Bending Stress

According to Hall *et al.* (1988), for bending load, bending stress (tension or compression) is:

$S_b = M_b r/I$.

Hence,

$S_b = (32 M_b)/(\pi d^3)$,

where:

S_b = bending stress;

M_b = bending moment;

d = shaft diameter;

I = moment of inertia.

Also, $I = \pi d^4/64$, for a cylindrical shaft (PSG Tech 1989).

Torsional Stress

According to Black and Adams (1968), torsional stress is determined using

$$\tau_{xy} = M_T r/J.$$

But $J = \pi d^4/32$, hence, $\tau_{xy} = (16M_T)/(\pi d^3)$, where:

τ_{xy} = Torsional stress N/m²;

M_T = torsional moment;

r = radius of shaft;

J = Polar moment of area;

d = diameter of shaft.

Torsional Rigidity

Torsional rigidity of a shaft is based on permissible angle of twist. The amount of twist permissible depends upon the particular application and varies from 0.3 degree/m for a machine tools shaft to about 3 degree/m for line shafting as given by:

$$\theta = (584 M_T L)/(Gd^4),$$

where:

θ = angle of twist (degree);

L = length of shaft (m);

M_T = torsional moment (Nm);

G = Torsional modulus of rigidity (N/m²);

d = diameter of shaft (m).

Lateral Rigidity

The lateral rigidity of a shaft is based upon the permissible lateral deflection for proper bearing operation, accurate machine tool performance, shaft alignment etc. Amount of deflection can also be calculated by two successive integrals of the formula:

$$d^2y/dx^2 = (M_b)/(EI),$$

where:

M_b = bending moment N/m²;

E = Modulus of elasticity (N/m²);

I = Moment of inertia (m⁴).

Shear Stress

The shear stress on the shaft is determined by the formula below (Black and Adams 1968):

$$\tau = (16T)/(\pi d^3),$$

where:

τ = Shear stress N/m²;

T = Torque (Nm);

d = Shaft diameter (m).

Bearing Selection

Bearing must be selected based on its load carrying capacity, life expectancy and reliability (PSG Tech 1989). The relationship between the basic rating life, the basic dynamic rating and the bearing load is:

$$C = [L/L_{10}]^{1/K} P, \text{ or } C/P = [L/L_{10}]^{1/K}, \text{ i.e.,}$$

$$[C/P]^K = L/L_{10}, \text{ or } L_{10} = [C/P]^K/L.$$

But $L = 60n/10^6$ million revolutions, therefore, $L_{10} = (10^6/60n) \times [C/P]^K$, where:

L_{10} = life of bearing for 90% survival at one million revolutions;

L = required life of bearing in million revolutions (mr);

n = rotational speed (rev/min);

C = basic dynamic load rating (N);

P = equivalent dynamic bearing load (N);

K = exponent for life equation with:

$K = 3$ for ball bearing;

$K = 10/3$ for roller bearing.

Also, $P = \text{radial load} + \text{axial load}$,

$$P = (XF_r + YF_a),$$

where:

X = radial load factor for the bearing;

Y = axial load factor for the bearing;

F_r = actual radial bearing load (N);

F_a = actual axial bearing load (N).

Table 2 shows the recommended life value in operation. It is assumed that this machine is designed to operate for 8 hours per day intermittently and whose breakdown will have serious consequences.

The bearing life in operating hours is chosen to be 8,000.

Table 2. Recommended life value of bearings.

Type of operation	Life in operation
Infrequently operated	500
Brief operation only	4,000-8,000
Intermittent operation	8,000-15,000
One shift operation	15,000-30,000
Continuous operation	30,000-60,000
Continuous operation with high production capacity	100,000

* Design data (PSG Tech 1989)

Table 3. Deep groove bearing selection factor.

Bearing	Basic dynamic load, <i>C</i> (N)	Inner race max. diam., <i>d</i> (mm)	Outside diameter, <i>D</i>	Width, <i>B</i>	Max. speed, rpm
6,406	3,350	30	90	23	6,300
6,407	4,300	35	100	25	5,600
6,408	5,000	40	110	27	5,000
6,409	5,850	45	120	29	4,500

* Design data (PSG Tech 1989)

A deep grooved bearing will be used for this work due to its ability to withstand heavy radial loads at high speed (Table 3).

Screen

Purpose of the screen is to act as a separator between the threshed maize and the cobs. Screens are either woven-wire or plate with perforated round holes. The plate with perforations was selected for this work. This will allow threshed maize grains to escape into collector pot.

Key

The function of a key is to prevent relative rotation of a shaft and the member to which it is connected, e.g., hub of a gear, pulley or crank. Different types of keys are available, the choice of which is dependant on power requirements, tightness of fit, stability of connection and cost. For a light transmission, a set screw may be employed. But for this work, a flat key was adopted, as this is used where added stability is desired, e.g. machine tools.

Proportion of a Key

For a good result, the width of a key is made one-quarter the diameter of the shaft. The thickness of a key for equal strength of the key in failure by shearing of the key, and compression on the key may be determined by the corresponding allowable stresses in shear and compressions.

The length of the key can be calculated as $L = \pi d/2 = 1.57d$.

The forces on the top and bottom of the key resist tipping of the key, and the force, *F*, between the side of the key and the key way in the hub is due to the resisting torque, *T'*:

$$T' = Fd/2 = FL/\pi,$$

where:

T' = resisting torque;

F = resisting force;

d = diameter of shaft;

L = length of key.

Design Calculation and Results

The average cylinder speed for high and efficient threshing output was estimated as 830 rpm. This cylinder speed was used as the basis for the calculation. Then:

volume of hopper, $V = 0.004797m^3$;

overall height, $H = 0.511m$;

mass required to thresh maize along the length of the threshing bars = 0.837 kg;

angular velocity, $\omega = 86.92 \text{ rad/sec}$;

maximum force available along the threshing arm, $F_{max} = 518.88N$;

threshing torque = 42.55Nm;

power delivered by shaft along the length of threshing bars = 3.698KW;

prime mover (electric motor) = 5Hp.

According to Spotts (1985), for a 5 horse power motor and a speed of 870 rpm, the recommended minimum sheave pitch diameter is 3.8", which is equivalent to 96.52mm or 100mm diameter of pulley. For this range of speed, V-belt is the best. Referring to Table 2, a class B V-belt was selected for this work. According to Williams (1953), a groove angle of 38° was chosen for a pulley of 100mm diameter using class B V-belt. Then:

weight of pulley, $W_p = 10.01N$;

speed of driver pulley, $N_1 = 870 \text{ r.p.m}$;

speed of driven pulley, $N_2 = 830 \text{ r.p.m}$;

diameter of driver pulley, $D_1 = 100mm$;

diameter of driven pulley, $D_2 = 105mm$;

effective pull on belt, $T = 518.9N$;

tension on tight side, $T_1 = 550.2N$;

tension on slack side, $T_2 = 31.33N$;

maximum centre distance of belt, $X_{max} = 102.5\text{mm}$;
length of belt, $L = 537\text{mm}$;
reaction, $R_D = 647.9\text{N}$;
reaction, $R_B = -79.44$, negative sign means one should reverse the direction of R_B ;
maximum bending moment $M_{bmax} = -50.25\text{Nm}$;
torsional moment, $M_t = 42.55\text{ Nm}$;
maximum shear force = -87.43N ,
negative sign implies the direction of the force;
shock and fatigue factor (bending moment),
 $K_b = 1.5$.

Performance Test

The completed work was subjected to test and it was found to thresh maize very effectively. Losses and breakages were found to be very negligible. The efficiency of the machine was found from the equation below to be equal to 99.2%:

$$\text{Efficiency} = [(W_1 - W_2)/W_1] \times 100,$$

where:

W_1 = weight of unthreshed cobs;

W_2 = weight of cobs not well threshed.

Conclusion

Self-reliance is the major drive of development and vibrant economy. This machine has been fabricated with the use of locally available materials. The machine is simple, less bulky and effective with its self-cleaning ability. Grains loss and mechanical visible damage have been very minimal. Performance test has revealed that the efficiency of the machine is 99.2%. The machine threshes 200kg of maize within an hour. The machine can either be powered by an electric motor or engine (diesel or petrol). The electric motor seat provides adjustment so that the V-belt can be fixed easily. There is no doubt that the machine will ease the long term scourge of youth unemployment in our land.

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