DESIGN AND FABRICATION OF A PEDAL OPERATED GUINEA CORN THRESHER

BY

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2004/18441EA

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:ING A FINAL YEAR PROJECT REPORT SUBMITTED IN PARTIAL FULFILLMENT F THE REQUIREMENTS FOR THE AWARD OF BACHELOR OF ENGINEERING (B. G.) DEGREE IN AGRICULTURAL AND BIORESOURCES ENGINEERING, FEDERAL UNIVERSITY OF TECHNOLOGY, MINNA, NIGER STATE.

FEBRUARY, 2010

DECLARATION

I hereby affirm that this project titled "Design and Construction of a Pedal Operated Guinea Corn Thresher with Blower" is an original work and has never been submitted anywhere else before, neither has it been wholly or partially presented for any other degree. All sources of information have been duly acknowledged by means of reference.

ISMAILA ISMAILA

Name of student

02 - 03 - 2010

Date

CERTIFICATION

his is to certify that "Design and Fabrication of a Pedal Operated Guinea Corn Thresher" by smaila ismaila, meets the regulations governing the award of the degree of Bachelor of ngineering (B. ENG.) of the Federal University of Technology, Minna, and it is approved or its contribution to scientific knowledge and literary presentation.

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DEDICATION

This Project is solely dedicated to Almighty Allah, who in His mercy I started this project and ended it in His infinite mercy "Say surely my prayer, my sacrifice my living and my death are for Allah, the Lord of the world Q6: 162.

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ABSTRACT

'his project is the design and construction of a pedal operated guinea corn thresher with ower. Guinea corn threshing involves detaching the grains from the ear heads. This is done hen it is dried to a moisture content of 17%. It is one of the most important post harvest ›erations in guinea corn production. The materials used for the construction include mild ɜel which is available locally. The machine is manually operated. The required power to ›erate the machine is calculated to be 80W. The maximum weight that can be carried by the at is 523kg. Performance evaluation of the machine was done with the kaura variety of ıinea corn at two levels of feed rates. It was observed to have a threshing efficiency of 94% d cleaning efficiency of 92% at 27.6kg/hr feed rate. A maximum throughput capacity of kg/hr was recorded.

TABLE OF CONTENTS

TABLE	LIST OF TABLES	PAGE

FIGURE	LIST OF FIGURES	PAGE

PLATE	LIST OF PLATES	PAGE

LIST OF APPENDICES

CHAPER ONE

1.0 INTRODUCTION

1.1 Background of the Study

Sorghum bicolor is commonly called guinea-corn or sweet sorghum. Guinea corn is called dawa in Hausa.It is an African crop, which is widely distributed throughout the world. Different cultivars are found in different regions depending on the climate. It is adapted to a wider range of ecological conditions. It is mostly a plant of hot, dry regions; still survive in a cool weather as well as waterlogged habitat (Kebede and Mishra, 1990).

In Nigeria, Guinea corn is grown under a wide variation of soil and climate conditions and can withstand fairly severe drought. This makes Nigeria to be one of the major producers of guinea corn in the world.

Guinea corn, an important crop providing food and fodder in the semi-arid tropics of the world. It is a staple food for more than 500 million people in more than 30 countries. It has been used in the production of alcohol. The whole plant is used for forage, hay or silage. The stem of some types is used for building; fencing, weaving, broom making and firewood. Industrially it can be used for vegetable oil, waxes and dyes. Until now, guinea corn has been used in South Western Nigeria for many generations, to treat several diseases including sickle-cell anemia, leukemia, multiple myeloma, heart and other blood- related problems as well as for headaches

Guinea corn is harvested almost exclusively by hand with a knife after bending the taller stems to reach the spikes. Harvested ears are stored in traditional granaries after which they are ready for threshing.

Guinea corn threshing involves detaching the grains from the ear heads. This is done when it is dried to the required moisture content. It is one of the most important post harvest operations in guinea corn production.

Threshing of the Guinea corn after harvest has posed a serious threat to the farmers. This is as a result of high cost and non availability of uncomplicated threshing machine in the market.

The combine harvester thresher, and other threshers are available or could be imported, but their high cost and also maintenance cost has been a discouraging factor to farmers. Most farmers in the rural area still thresh their guinea corn using the local method; this has resulted to untimely harvest and serious loss to the farmers. As such there is need to construct a thresher that is capable of increasing the threshing output of guinea corn and at an affordable price.

In view of the above stated problems, it has become necessary to construct a simple guinea corn threshing machine that is capable of solving these problems for the farmer.

1.2 Statement of Problem

In Nigeria, threshing of guinea corn is either done traditionally by hand-beating or animal treading. This method is time wasting, energy consuming and most often; the grains are broken and sometime pick up foreign materials such as sand, stones and even metallic debris (Olugboji, 2004).

From the viewpoint of milling, the above process is counterproductive as the stone cause wear of the mill husker, which are seldom provided with pre-cleaners.Also, due to late harvesting, shattering and cracking of grains could ensue leading to loss (Glaszmann, 1987).

A number of small, medium and large-scale threshers have been in existence for some time now, but some of them are large, expensive with low performance in comparison with the traditional method and often time fragile, they have not been adapted to a significant extent.

The portable ones require winnowing after threshing and the threshing teeth needed replacement up to twice in a single season (Glaszmann, J.C.1987).

1.3 Justification

Base on the above stated problem, it is therefore imperative to develop a guinea corn thresher (pedal-operated) which will enable the farmer who is the main target conveniently carryout threshing operation on the farm with less stress, grain damage, grain loss, labour cost, time spent and most importantly cost compared to other threshing machines.

1.4 Objectives of the Study

- To design a pedal operated guinea corn threshing machine.
- To fabricate the designed threshing machine.
- To carryout performance evaluation of the designed machine.

CHAPTER 2

2.0 LITERATURE REVIEW

The best way to present crops to the market is in its processed form, the threshed seeds are not only marketable but they last longer and do not perish easily. Threshing which is the detachment of grain seeds from plant cobs, involves different methods and machine types in use today. It could be done by stripping, impacting, rubbing or combination of any of these. There are basically two modes of threshing, manual and mechanical.

2.1 Manual Threshing Methods

This was the first threshing method before the invention of machines. It is still practiced greatly in undeveloped countries all over the world including Nigeria. As reported by the FAO (1996), the average output of this method is 12-25kg per hour.

2.1.1 Stripping

This is the use of stick to beat off the paddy of stalks laid upon a surface (Juo and lowe, 1996). The logs of the paddy are laid on a clean surface or tarpaulins and then beaten with sticks thereby removing the grains from their cobs. The threshed cob is then winnowed using a calabash. This results in high grain loss, time consumption, labour intensive and tedious. The output of this method is between 13-20kg per hour (Smith and Wilcox, 1997).

2.1.2 Mortar and Pestle

This method is practiced by women on small scale farm. The stalks are packed into the mortar and a pestle is used to pound it. The impact of the pestle removes the grain from the stalk. Cleaning is done by winnowing using calabash. Threshing output ranges from 20-25kg per hour. (FAO, 1995).

2.1.3 Treading

This is accomplished by treading the grain under the foot of men or hooves of animals. It is time wasting and also requires lots of hard labour.Threshing performance as reported by Stone/Gulvin (1997) is 17-22kg per hour.

2.2 Mechanical Threshing Methods

As reported by Smith and Walkos (1997), mechanical threshing is practiced on large scale farms. The stalks are threshed and cleaned with one machine requiring little time and labour.

The first thresher was constructed in the middle of the 19^{th} century before the advent of the automobile. A rice thresher was used in the South Texas in 1845.A combine was first built in Michigar in 1836 and later shipped to California where it was successfully used.(Smith/Wilcox,1997).With advance in technology, mechanical threshers replaced the common manual methods. This method includes:

2.2.1 Foot Operated Threshing Machine

Foot pedal operated threshing machines are less popular to those operated by electrical motors. Here the operator either grip the bundle of stalks and hold the panicle end against the rotating cylinder or throw the crop into the machine where the major portion is then threshed by the initial impact of the bars and spikes on the revolving cylinder. It was originated in Japan and is now been used in most of the sorghum growing countries. The threshing output is 60-80kg/hr (I.A.R. Zaria, 1988).

2.2.2 Disc Roller Thresher

(FAO, 1995) reported that horses are used to pull rollers across a spread of rice stalks on the ground. The roller is mounted on a frame and pulled over the laid crops. The disadvantage here is that the grains are easily broken and eaten by animals. The threshing capacity is 30-35kg/hr.

2.2.3 Ground Hog Thresher

(Smith and Wilcox, 1997) reported that "Ground hog" spikes tooth cylinder employs animal driven power or feed mill to operate.

2.2.4 ST – 45 Thresher

This is designed and constructed by I A R, Zaria in 1988 to thresh wide variety of cereals by selection of appropriate speeds, sieves and cylinder concave clearance. A blower and set of reciprocating sieves achieved the cleaning process. Two men can ensure continuous operation. A power unit of about 3kw runs the machine. The threshing capacity for rice is 400-500kg/hr.

2.2.5 Bamba Thresher

Bamba thresher is designed in such a way that it can be moved to all part of the farm. It can thresh with full separation and cleaning. It is simple to operate and maintain. However, it is expensive. It has a power unit of 9Hp and threshing capacity of 500-600kg/hr (A.N.P. Kaduna.1988).

2.2.6 Combine Harvester Thresher

This can be pull-type where it is drawn by a tractor, or the self propelled type which is driven by it own engine of 25-30Hp (Smith and Wilcox,1997).It can be operated by one man, it is capable of cutting,threshing,cleaning e.t.c. The capacity is 1200-1500kg/hr.

2.3 Factors Influencing Threshing Performance

The threshing performance of a crop depends largely on the moisture content of the crop at the crop at the time of threshing. At high moisture content, (22%), the crop requires high threshing force in order to separate the grains but at low moisture content, there is increased breakage of earhead leading to increased amount of unthreshed grains (Dash and Dash, 1989).

Threshing speed is seen to affect the threshing efficiency as well as mechanical grain damage. At high threshing speed, due to high impact force, most of the grains end up damaged, which invariably affect threshing quality. At low threshing speed, most of the earhead are unthreshed since the impact force for threshing is not available (Dash and Dash, 1989).

2.4 Criteria for Evaluating Threshing Performance

Various criteria are considered in evaluating the performance of a thresher. Some of these criteria include threshing efficiency, mechanical grain damage, scatter losses, throughput capacity and power requirement.

The threshing efficiency gives the percentage of threshed material to the total material handled by the machine. It is a function of the cylinder speed, feed rate and crop variety respectively (Ndirika, 1994).

Dash and Dash (1989) showed that threshing efficiency decrease with increase in peripheral velocity since at high peripheral velocity of the cylinder, the impact force exacted by the cylinder on the crop detaches the ear alongside the grain.

Mechanical grain damage which is a function of moisture content of the grain. Peripheral velocity of the cylinder is said to increase with increase in peripheral speed as greater impact force is exacted on the crop causing damage (Dash and Dash, 1989).

The concaves primary function is to hold the grain in position for threshing. Threshing improves as concave clearance decreases. Grain damage also reduces as concave clearance increases. This is because, as concave clearance increases, the impact force on the grain also reduces (www.agric.gov.ab.ca).

Scatter loss is said to decrease with increase in feed rate since at low feed rate, some of the threshed grains bounce on the stationary sieve plate and are taken out easily (Ndirika, 1994).

Cleaning efficiency decreases with increase in fan speed and air velocity as most of the grains are blown off at greater air velocity (Dauda, 2001).

CHAPTER 3

3.0 MATERIALS AND METHODS

3.1 Description of the guinea corn thresher

3.1.1 Description

The thresher involved, comprises of three main units viz: the hopper, threshing unit and the cleaning unit.

Hopper

The hopper is trapezoidal in shape and is made of 18 gauge mild steel with all sides slanting inwards. It forms the feeding chute through which guinea corn heads are fed into the threshing unit.

The Threshing Unit

The threshing unit comprised of a 100mm diameter and 420mm length cylinder, beater and a concave, made of mild steel metal. The cylinder is made from a steel shaft. It has six rows of beaters strongly welded on the surface of a pipe at an angle to the vertical so as to aid conveyance of the stalk to the outlet. The beaters are made from small rods of 10mm diameter and spaced 30mm apart. It also has curved rods which serves as the concave and spaced at 20mm apart, so as to enhance the threshing action. The clearance between the beaters and the concave is maintained at 31mm. the threshing unit is covered with a semi-circular steel plate to prevent loss of grains through scatterings.

Frame

The frame makes the mounting support for all other units and is made of 25 × 25mm angle bar. The overall dimension is 460mm ×320mm ×750mm.

The Cleaning Unit

The blower/ fan is located just below the threshing unit and opposite the chaff exit. It is a centrifugal fan comprising of four straight impellers attached to the shaft inclined at 90° to one another, all in an involute casing. A pinion, which is driven by a chain drive, is attached to the blower shaft at one of the ends.

3.1.2 Description of the Pedal Power

The pedal power consists of the following parts:

Saddle

This is the part on which the operator seats to provide power to the thresher. It is made adjustable so as to provide full extension of the human muscle when pedaling. The average human power is estimated to be 0.08kw.

Sprocket

The sprockets use are two in number, the driven sprocket and the driving sprocket link to produce drive via chain.

Pedal

It is an acceleration, which produces inductive energy for motion. It employs a heavy based semi conductor to reduce heat energy and reduce pressure inherent in the process of operating.

Chain

It is used to transmit power from input to output source, connected via the sprockets for a selected type of chain.

3.2 Design Calculations

3.2.1 Determination of Size of Guinea Corn:

A venier caliper and a micrometer screw gauge were used for all dimensions and a beam balance was used for all weighing. All measurement was taken in the laboratory. The mean values of the major, intermediate and minor diameter of the grains were obtained from calculation.

Major Diameter (mm)	Intermediate Diameter (mm)	Minor Diamter (mm)
4.97	3.09	4.34

3.2.2 Determination of Terminal Velocity

The terminal velocity of the grains were obtained from calculation, formula used is given as:

$$V_t = 1.74\left[g.dp\frac{(\rho p - \rho f)}{\rho f}\right]^{1/2} \qquad (1)$$

For Guinea corn,

$M_D = 4.66mm$

$I_D = 3.08mm$

$M_d = 4.34mm$

Where,

M_D = Major diameter

I_D = Intermediate diameter

M_d = Minor diameter

Diameter of particle, $d_p = [M_D \times I_D \times M_d]^{1/3}$ (Sitkei, 1986) (2)

$d_p = [4.66 \times 3.08 \times 4.34]^{1/3}$

$d_p = 3.9641 \times 10^{-3}m$

Volume of particle, $V_p = \frac{1}{6\pi}(de)^3$ (Sitkei, 1986) (3)

$V_p = \frac{1}{6\pi}(3.9641 \times 10^{-3})^3$

$= 3.2641 \times 10^{-8}m^3$

$$\text{Density} = \frac{mass}{volume} \quad (4)$$

$\text{Mass} = 4.1 \times 10^{-5}kg$

$$\rho = \frac{4.1 \times 10^{-5}}{3.2614 \times 10^{-3}} = 1257.13 kg/m^3$$

$$V_t = 1.74\left[\frac{9.81 \times 3.9641 \times 10^3 (1257.13 - 1.293)^{1/2}}{1.293}\right]$$

$V_t = 8.11m/s$

3.2.3 Design of Sieve

The basic factor taken into consideration in the design of sieve is the size / diameter of the grain in both longitudinal and transverse plane, which actually was used to decide the diameter of the holes on the sieve used for cleaning.

The diameter of the holes of the sieve was determined based on the geometric mean of the particle diameter of the grain.

The particle diameters of the grain were obtained from the formula given as:

$$d_p = (\varphi_1 \times \varphi_2 \times \varphi_3)^{1/3} \quad (5)$$

Where,

d_p = particle diameter

φ_1, φ_2, and φ_3 = major, intermediate and minor diameter.

For Guinea Corn,

$d_p = (4.66 \times 3.08 \times 4.34)^{1/3}$

$= 3.96$mm

3.2.4 Design of Steel Seat

The saddle consists of a hollow circular pipe. Data of hollow circular pipe are given below.

Material: low carbon steel (Fe 290)

Ultimate stress, $\delta_{ult} = 290$Mpa

Yield stress, $\delta_y = 160$Mpa

Allowable stress of the material, $\delta_{all} = 47.25$Mpa

Young modulus constant, $\varepsilon = 2 \times 10^5$Mpa

Length of hollow circular pipe, $L_{sq} = 700$mm

Acceleration due to gravity, g = 9.81m/s

- **Determination of design length, l_{eff} using equation below**

$l_{eff} = \mu Ls$ (6)

$l_{eff} = \frac{1}{4} \times 700 = 175mm$

- **Determination of moment of inertia, I_{yy} using the equation**

$I_{yy} = I_{xx} = \frac{\pi}{64}(d^4 - d_1{}^4)$ (7)

$= \frac{\pi}{64}(38^4 - 36^4)$

$= 19908.6mm^4$

- **Determination of cross-sectional area, A_s, using equation**

$A_s = \frac{\pi}{4}(d^2 - d_1{}^2)$ (8)

$A_s = \frac{\pi}{4}(38^2 - 36^2)$

$= 116.3mm^2$

- **Determination of radius of gyration, L_s, using equation**

$L_s = \sqrt{\frac{I\rho}{As}}$ (9)

$= \sqrt{\frac{19908.5}{116.3}} = 13.1mm$

- **Determination of slenderness ratio, λ, of the hollow circular pipe using equation**

$\lambda = \frac{Leff}{Ls}$ (10)

$= \frac{175}{13.08} = 13.38 \simeq 13$

- **Determination of slenderness ratio, λm, of the material using equation**

= 271.5 sin 71

= 88.4N

Horizontal component of W

= W cos 71 (46)

= 271.5 sin 71

= 26.0N

Thus, the shaft is subjected to the vertical and horizontal loads.

First of all considering vertical loading on the shaft. Let R_{av} and R_{bv} be the reactions at bearing A and B.

$R_{av} + R_{bv} = 190.5 + 88.4 = 278.9N$ (47)

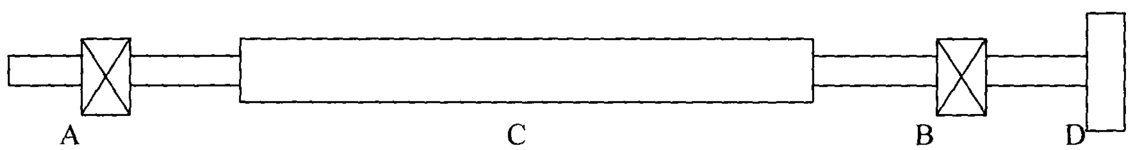
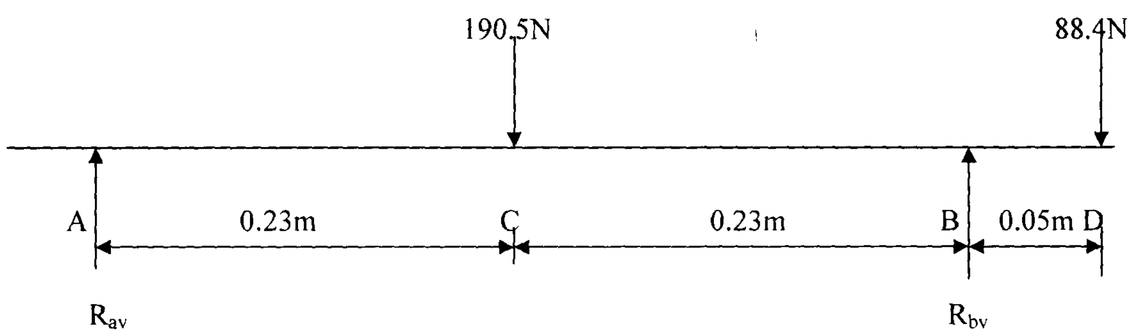


Figure 3.1 Free body diagram



$= -14.03N$

The bending moment diagram for horizontal loading is shown in fig 3.5.

The resultant bending moment M_b is given as,

Resultant B.M. $M_b = \sqrt{(19.7)^2 + (14.03)^2}$

$= \sqrt{388 + 196.8}$

$= 24.19N\text{-}m$

For a shaft subjected to combined and torsion, the equivalent twisting moment,

$$T_e = \sqrt{(Km \times Mb)^2 + (Kt \times Mt)^2} \quad (48)$$

Where,

Km = Combine shock and fatigue factor for bending, and

Kt = Combine shock and fatigue factor for torsion.

Rotating Shaft	Load gradually applied
K_b	1.5
K_t	1:0

$T_e = \sqrt{(1.5 \times 24.19)^2 \times (1 \times 9.5)^2}$

$= \sqrt{36.3^2 \times 9.5^2}$

$= 37.5N\text{-}m$

Equivalent twisting moment (T_e),

$$\lambda m = \sqrt{\frac{\pi^2 E}{\delta all}} \quad (11)$$

$$= \sqrt{\frac{\pi^2 \times 2 \times 10^5}{47.25 \times 10^6}} = 204.39$$

$$\lambda < \lambda m$$

- **Determination of reduction factor, U, from table (Kinasoshville, 1972)**

$$U = 0.94 + \frac{37-30}{40-30}(0.9 - 0.94) \quad (12)$$

$$= 0.934$$

- **Determination of allowable stress, δ_a, on the hollow pipe, using equation**

$$\delta a = \delta_{all} \times U \quad (13)$$

$$= 47.25 \times 0.934$$

$$= 44.13\text{Mpa} \simeq 44{,}131{,}500\text{n/m}^2$$

- **Determination of critical load, F_u, on the hollow circular pipe using equation**

$$F_u = \delta a \times A_s \quad (14)$$

$$= 44.1315 \times 116.3$$

$$= 5132.5\text{N}$$

Maximum weight that can be carried by the seat is

$$\frac{5132.5}{9.81} = 523.2kg \simeq 523kg$$

3.2.5 Design of operator seat base support

The force is the weight of the operator, assume to be 100kg i.e. 981N using a factor of safety 4, the weight then is

$981N \times 4 = 3924N$

W = weight of the operator + weight of saddle bar (part A).

$= 3924N + V_{ą} \times ρ_{ą} \times 9.81$

$= 3924 + (2.636 \times 10^{-3} \times 7854 \times 9.81)$

$=3924 + 203.1$

$= 4127.1N$

Using the working stress of 58.25Mpa,

$Б = F/A$ (15)

$A = F/б$

$A= 4127.1N/58.25 \times 10^{6}Nm^{-2}$

The support is double legged as shown in the figure. Each leg then is supposed to have an area (A).

$A = 7.1 \times 10^{-5}/2$

$A = 3.54 \times 10^{-5}m^{2}$

A 38mm × 38mm angle bar has a cross-section as shown below.

Cross-section $= (0.038 \times 0.002) \times 2$

$=1.52 \times 10^{-4}m^{2}$

This cross-sectional area is larger than the required cross-sectional area calculated above hence, design is alright.

3.2.6 Design of Chain and Sprocket Drive

The data below are used for chain/sprocket drive during the design and fabrication. Chain type ANAI No (Baumeister et al, 1978).

Chain pitch - $P_c = 12.7mm$

Roller width - $W_R = 7.935mm$

Roller diameter - $D_R = 7.9248mm$

Pin diameter - $P_{in} = 3.96mm$

Roller link plate height, $H_R = 11.48mm$

Roller thick plate thickness, $P_R = 1.524mm$

Speed generated by human (driving sprocket speed)

$N_C = 50rpm$ (konz 1983)

P_s, power generated by human leg = 74.57W (konz, 1983)

Number of teeth on the driven sprocket, $Z_1 = 44$

Centre distance, $C_c = 825mm = 65$ pitches

Transmission Ratio $U_c = 2.2$

- **Pitch dia of S_1 driving sprocket**

$$d_{p1} = \frac{P}{Sin(180/Z_1)} \tag{16}$$

$$= \frac{12.7}{Sin(180/44)} = 178mm$$

- Bottom dia D_B of S_1

So that $D_B = d_{p1} - D_R$ (17)

$D_B = 178 - 7.9 = 170.1mm$

- **Outside dia. D_O of S_1(driving sprocket)**

$D_o = P_c\ [0.6 + Cot\ (\frac{180}{Z_1})]$ (18)

$= 12.7\ [0.6 + cot\ (\frac{180}{44})]$

$= 185.19mm$

- **Caliper dia D_{c1} of S_1 driving sprocket**

$Dc_1 = [dp_1 \times cos\ (\frac{90}{Z_1})] - Dr$ (19)

$= [178.02 \times cos\ (\frac{90}{44})] - 7.9258$

$= 169.98mm$

- **Number of teeth Z_2 driven sprocket S_2**

$Z_2 = \frac{Z_1}{Uc}$ (20)

$= \frac{44}{2.2}$

$Z_2 = 20$

- **Chain length L_c**

$$L_c = 2C_c + \frac{(Z^1 + Z_2)}{2} + \frac{(Z_1 - Z_2)^2}{\pi \times 4\, Cc} \tag{21}$$

$$= 2 \times 65 + \frac{(44 + 20)}{2} + \frac{(44 - 20)^2}{\pi \times 4 \times 65}$$

$= 163.75 \quad \simeq 163$ pitches on the chain when measured

Given = 2082.8mm

- **Velocity of chain V_c**

$$V_c = \frac{\pi \times dp \times Nc}{60} \tag{22}$$

$$= \frac{\pi \times 178 \times 10^{-3} \times 50}{60}$$

= 0.466m/s

- **Tangential Force, F_r transmitted by chain**

$$F_r = \frac{Ps}{Vc} \tag{23}$$

Ps, power generated by human leg = 74.57w

$$= \frac{74.57}{0.47}$$

= 158.66N

Where torque $T_1 = \frac{Ps}{W}$ or $T_1 = \frac{Fr \times dp}{2}$ (24)

$$= \frac{158.7 \times 178 \times 10^{-3}}{2}$$

= 14.24Nm

Angular speed ω_1 of driving sprocket S_1

$$\omega_1 = \frac{2 \times \pi \times Nc}{60} \tag{25}$$

$= 5.24$rad/s

- **Tension caused by centrifugal force**

$$F_o = qVc^2 \tag{26}$$

Where q, chain density = 0.44kg/m (Roslar 1983)

$= 0.44 \times 0.47^2$

$= 0.097\text{N} \simeq 0.1\text{N}$

- **Total force, F_t on chain**

$$F_t = Fo + F_r \tag{27}$$

$= 0.1 + 158.66$

$= 158.76\text{N}$

- **Chain bearing pressure/Unit hinge area P_x**

$$P_x = \frac{Fr \times Ks}{Ac} \leq [P] \tag{28}$$

$K_s = K_d \times K_y \times K_t \times K_l$

$K_d = 1$

$K_y = 1$ where, $A_c = 47.8$mm

$K_t = 1.25$

$K_l = 1$

$K_s = 1 \times 1 \times 1.25 \times 1 = 1.25$

$P_x = \frac{158.66 \times 1.25}{47.8} = 4.15mpa \simeq 41.5$bar (kg/cm²)

P = 12.34mpa (Berezovtsky et al 1988) = 123.4bar (kg/cm²)

$P_x < P$

- **Maximum Tangential force F_k that can be transmitted by the proposed chain without causing excessive wear of the link hinge**

$$F_k = [P] \times A_c \tag{29}$$

$= 2.34 \times 10^6 \times 47.8 \times 10^{-6}$

$= 589.85N$

- **Chain actual bearing pressure P_{bc}**

$$P_{bc} = \frac{Ft}{Ac} \tag{30}$$

$= \frac{158.76}{47.8 \times 10^6} = 3.32mpa$

- **Factor of safety (s)**

$$s = \frac{Fb}{Ft} \geq [S] \tag{31}$$

$F_b = \frac{16464}{158.76} = 103.7$

S = 8.33 (Berezovsky et al 1988)

$S > [S]$

The chain is safe.

- **Design power P_{DS} for a chain drive**

$P_{DS} = C_d \times P_S$ (32)

C_d = (Berezovsky et al 988)

$p_{DS} = 1 \times 74.5 + W \quad p_{DS} > PG$

- **Pitch dia. d_{p2} of the driven sprocket**

$$d_{p2} = \frac{P}{Sin(180/Z_2)} \quad (33)$$

$$= \frac{12.7}{Sin(180/20)} = 81.18mm$$

- **Bottom diameter D_{b2} of the driven sprocket**

$D_{b2} = d_{p2} - D_r$ (34)

$= 81.18 - 7.725$

$= 73.26mm$

- **Outside diameter D_{o2} of driven sprocket**

$$D_{o2} = Pc\ [0.6 + Cot\ (\frac{180}{Z_2})] \quad (35)$$

$$= 12.7\ [0.6 + Cot\ (\frac{180}{20})]$$

$= 87.81mm$

- **Caliper diameter Dc_2 of the driven sprocket**

$$Dc_2 = [dp_2 \times \text{Cos}\,(\frac{90}{Z_2})] - D_r \quad (36)$$

$$= [81.18 \times \text{Cos}\,(\frac{90}{20})] - 7.925$$

$$= 73.01\text{mm}$$

- **Velocity Vc_2 of driven sprocket**

$$Vc_2 = \frac{\pi \times dp^2 \times Nc_2}{60} \quad (37)$$

$$Nc_2 = \frac{44 \times 50}{20} = 110rpm$$

$$Vc_2 = \frac{\pi \times 81.18 \times 10^{-3} \times 110}{60}$$

$$= 0.47\text{m/s}$$

- **Torque τ, on driven sprocket**

$$\tau^2 = \frac{Fk \times dp_2}{2} \quad (38)$$

$$= \frac{158.66 \times 81.18 \times 10^{-3}}{2}$$

$$= 6.44\text{Nm}$$

- **Angular speed ω_2, of the driven sprocket**

$$\omega_2 = \frac{2 \times \pi \times Nc_2}{60} \quad (39)$$

$$= \frac{2 \times \pi \times 110}{60}$$

$$= 11.52\text{rad/s}$$

3.2.7 Hopper Design

The ability of seed flow of most agricultural materials from the hopper depends on;

- Angle of repose
- Angle of internal friction (Hyetson,2000)

In determining the angle of repose, the material was allowed to fall freely under gravity from a height onto a platform. The angle of repose is then measured thus;

$$\tan \theta = y/x \qquad (40)$$

Where θ = Angle of repose

Y = Height of material tip from bottom of platform

X = Radius of material from centre to edge

The feed hopper was designed with the view that the whole guinea – corn stalk will be fed through the hopper for threshing. A hopper with a trapezoidal cross-section was considered with a length of 260mm and breath of 260mm inclined at 40.5 to the horizontal.

3.2.8 Shaft design

Designing shaft requires the determination of the shaft diameter and length such that it will be strong and rigid enough to transmit power under stated operating and loading conditions. It involves material selection so that the stated condition is efficiently and cheaply met.

The material could either be brittle or ductile and when designing shaft for strength, we make distribution based on the material type.

3.2.8.1 Consideration on Allowable Maximum Stress

Using ASME code for steel which gives the steel allowable stress (S_a), the lesser value between both critical points is chosen; 30% of the elastic limit and 18% of ultimate strength while in tension for shaft without keyway i.e. 30% S_x and 18% S_y.

Where,

S_x = stress at a critical point in tension or compression normal to the cross-section under condition.

S_y = stress at the same critical point and in a direction normal to the S_x stress. This further reduced by 25% when there is stress concentration on the shaft if there is keyway.

3.2.8.2 Allowance for Shock and Fatigue

The tentative code for the design of transmission shaft of ASA (American Standard Association) (ASME, 1948) recommends that the values of bending moment (M) and torque (T) action on the shaft must be multiplied by certain shock and fatigue (K_b) and (K_t) respectively depending on the service conditions. The suddenly applied load with maximum shocks is the conditions for which the shaft is designed in this machine. The value of K_b and K_t for this condition ranges from 1.5 to 2.0 and 1.0 to 1.5 respectively.

3.2.8.3 Loads on the Shafts

Shafts are generally subjected to one or more combination of loads. Loads generally experienced by shafts are torsional load, bending load, in dynamic and static situation and axial load.

A solid shaft was used for this machine and the equations used in calculating the various loads are as follows;

Torsional load: $\tau_{xy} = \frac{16Mt}{\pi d^3}$ (41)

Bending load: $\sigma_b = \frac{32Mb}{\pi d^3}$ (42)

Axial load: $\delta = \frac{4Fa}{\pi d^3}$ (43)

3.2.9 Determination of Weight on Thresher Shaft

Area of side disc = $\pi \frac{(D^2 - d^2)}{4} = \pi \frac{(0.08^2 - 0.025^2)}{4} = 6099 \times 10^{-3} m^2$

Where,

D = 8.2cm

d = 2.5cm

Volume of each side disc (V) = t × A = 0.0025 × 6099 × 10^{-3}

= $1.525 \times 10^{-5} m^3$

Weight of disc (W) = $\rho \times v \times g$

= $7.83 \times 10^3 \times 1.525 \times 10^{-5} \times 9.8$ =1.170N

Total weight of disc (2) = 2.340N

Spikes: Area of each rod = $\frac{\pi d^2}{4} = \frac{\pi(0.1)^2}{4} = 7.854 \times 10^{-3} m^2$

Where,

d = 1cm

Volume of each spikes (V) = L × A = $4.084 \times 10^{-4} m^3$

L= 5.2cm

Weight of spikes (W) = $\rho \times v \times g$

$\rho = 7830$

W = 7830 × 0.000408 × 9.8

= 0.3138

Total weight of spikes (54) = 16.95N

Curved plates: Area of curved plates = L × b = 0.43 × 0.23

= $9.89 \times 10^{-2} m^2$

Volume of each curved plates (V) = t × A = $9.89 \times 10^{-2} \times 0.0025$

= $2.47 \times 10^{-4} m^3$

Weight of curved plates (W) = $\rho \times v \times g$

W = $7830 \times 2.47 \times 10^{-4} \times 9.8$

= 18.97N

Total weight acting on the shaft = 2.340 + 169.3 + 18.97

= 190.5N

3.2.10 Determination of Thresher Shaft Diameter

The sizes of the shaft are normally determined by the limits that are placed on the deflection, bending and torsion. Similarly the sizes of the shaft must be sufficient to prevent induced stress from exceeding the allowable stress of the material.

Torque on shaft drum, τ = force acting on the drum × radius of its neutral axis

$= 190.5 \times 0.05$

$= 9.5$N-m

Torque transmitted by pulley,

$$\tau = (T_1 - T_2)\ R \qquad (44)$$

$$9.5 = (T_1 - T_2)\ 0.07$$

$T_1 - T_2 = \frac{9.5}{0.07}$

$T_1 - T_2 = 135.7$

$T_1 = 3T_2$

$3T_2 - T_2 = 135.7$

$T_2 = \frac{135.7}{2} = 67.9$N-m

$T_1 = 3\ (67.9) = 203.6$N-m

Total load acting on pulley (W)

$W = T_1 + T_2 = 203.6 + 67.9$

$= 271.5$N

This load acts at 71° to the horizontal

Resolving the load W into vertical and horizontal components,

Vertical component of W

$$= W \sin 71 \qquad (45)$$

$$T_e = \frac{\pi}{16} \times \tau \times d^3 \quad (49)$$

$$= \frac{\pi}{16} \times 40 \times 10^6 \times d^3$$

$$d^3 = \frac{601.6}{3.142 \times 40 \times 10^6}$$

$$d = \sqrt[3]{4.79 \times 10^{-6}}$$

$$= 0.0169\text{m}$$

$$= 16.9\text{mm}$$

A 25mm diameter shaft is selected. According to Khurmi and Gupter (2008), shafts are in standard sizes of 5mm in steps.

3.2.11 Bearing Selection

Equivalent load, W

$$W = X \cdot V \cdot Fr + Y \cdot Fa \quad (50)$$

Where,

X = radial factor

V = rotation factor

Fr = radial load (N)

Fa = thrust load (N)

Y = thrust factor

In order to determine the radial load factor (X) and axial load factor (Y), $\frac{Fa}{Fr}$ and $\frac{Fa}{C}$ are required. Since the value of basic static load capacity (C) is not know, therefore $\frac{Fa}{C}$ = 0.5.

From table 27.4, the values of X and Y corresponding to $\frac{Fa}{C}$ = 0.5 and $\frac{Fa}{Fr} = \frac{56.5}{85.6} =$ 0.67 $(which\ is\ greater\ than\ e = 0.44)\ are$

X = 0.56 and Y = 1 (from table)

Since the rotational factor (V) for most of the bearings is 1, therefore basic dynamic equivalent radial load,

$W = 0.56 \times 1 \times 85.6 + 1 \times 56.5$

$= 104.4N$

Basic dynamic load rating

$C = W \left(\frac{L}{10^6}\right)^{1/3}$ (51)

(k = 3, for ball bearing)

Life of bearing in hours,

$L_H = 10yrs \times 300 \times 10hrs$ ... (Assuming 300 working days per year)

$= 30{,}000hrs$

Life of bearing in revolutions,

$L = 60N \times L_H$ (52)

$= 60 \times 600 \times 30000$

$= 1080 \times 10^6$

$C = 104.4\ (\frac{1080 \times 10^6}{10^6})^{1/3}$

$= 1071.1N$

$= 1.071KN$

From Table 27.1, with diameter of shaft 25mm, bearing No 205 with outside diameter of 52 and width 15mm of radial load capacity of 85.6N and thrust load 56.5N will be suitable and was selected.

3.2.12 Design Consideration of a Blower

The factors considered in the design of the blower unit were the blade length and width, number of blades and weight of the fan blades, the requirement of air discharge through the blower or the velocity of air required for separation.

3.2.13 Determination of Blade Weight

Area of the blade = $L \times b = 400 \times 90$

$= 36000mm^2 \simeq 0.036m^2$

Blade constructed from a guage 18 mild steel sheet metal of thickness 0.02mm,

Volume of blade= area × thickness

$= 0.036 \times 2 \times 10^{-3}$

$= 7.2 \times 10^{-5}m^3$

Density of mild steel = $7850kg/m^3$

Mass of blade = volume × density

$= 7.2 \times 10^{-5} \times 7850$

$= 0.57kg$

Total mass of the blade = $0.562 \times 2 = 1.124kg$

Weight of the blade = 11.0N

3.2.14 Determination of the Air Discharge Through the Blower

The requirement of air discharge through the blower can be estimated on the basis of velocity of air Required for cleaning (V), depth of air stream over the sieve (D) and width over which air is required (W). Therefore, the actual air discharge rate (Q) can be estimated from the formula: Since fan speed (N) is 146rpm, the velocity, V, of air required for cleaning is given as:

$$V = \frac{2\pi N}{3600} \text{ m/s} \qquad (53)$$

$$= 2\pi \times \frac{146}{3600}$$

$V = 0.26$ m/s

$D = 5cm = 0.04m$

$W = 34cm = 0.34m$

Therefore,

$Q = 0.26 \times 0.04 \times 0.34$

$= 0.004m^3/s$

3.2.14 Fan Case Design

The radius of the fan case calculated from

$$r = R\,(1 + \frac{\theta}{360})\ \text{(Hyestson, 2000)} \qquad (54)$$

Where r = radius of fan case

$R = \frac{distance\ from\ center\ of\ shaft\ to\ tip\ of\ blade}{Impeller}$. This is assume to be 9.75cm

θ = angle between impellers; (90°)

$r = 9.75\,(1 + \frac{90}{360})$

r = 12.19cm or 1.22mm

3.2.15 Blower shaft design consideration

Torque transmitted by pulley,τ

$\tau = (T_1 - T_2)\,R$

$6.44 = (T_1 - T_2)\,0.04$

$T_1 - T_2 = \frac{6.44}{0.04}$

$T_1 - T_2 = 161$

$T_1 = 3T_2$

$3T_2 - T_2 = 135.7$

$T_2 = \frac{161}{2} = 80.5$N-m

$T_1 = 3\ (80.5) = 241.5$N-m

Total load acting on pulley (W)

$W = T_1 + T_2 = 241.5 + 80.5$

$= 322$N

This load acts at 71° to the horizontal

Resolving the load W into vertical and horizontal components,

Vertical component of W

$= W \sin 71 = 322 \sin 71$

$= 104.8$N

Horizontal component of W

$= W \cos 71 = 322 \sin 71$

$= 99.28$N

Tangential force on the gear E,

$$F_{te} = \frac{TE}{R} \tag{55}$$

$$= \frac{6.44}{0.04} = 0.064$$

Normal loading acting on the gear teeth,

$$W_e = \frac{Fte}{COS\ 71} \tag{56}$$

$$= \frac{0.064}{0.33} = 0.20$$

Resolving load vertically and horizontally,

Vertical component of W_e

$= W_e \cos 71 = 0.20 \cos 71 = 0.07$

Horizontal component of W_e

$= W_e \sin 71 = 0.20 \sin 71 = 0.19$

Thus, the shaft is subjected to the vertical and horizontal loads.

First of all considering vertical loading on the shaft. Let Ra_v and R_{bv} be the reactions at bearing A and B.

$R_{av} + R_{bv} = 11.0 + 104 + 0.07 = 115.8N$

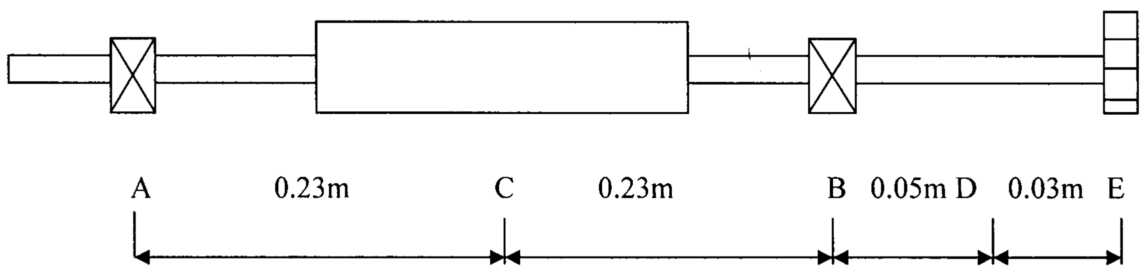


Figure 3.6 Free body diagram

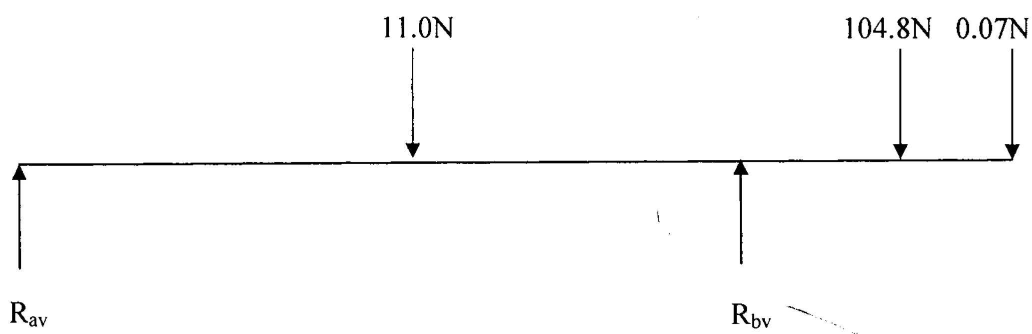


Figure 3.7 Vertical load diagram

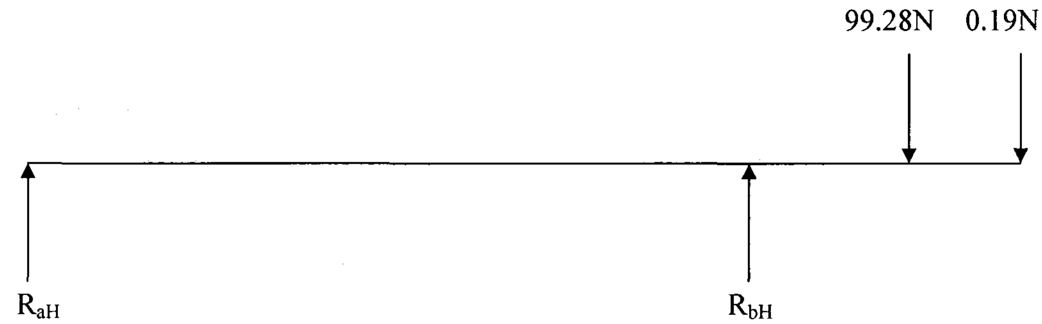


Figure 3.8 Horizontal load diagram

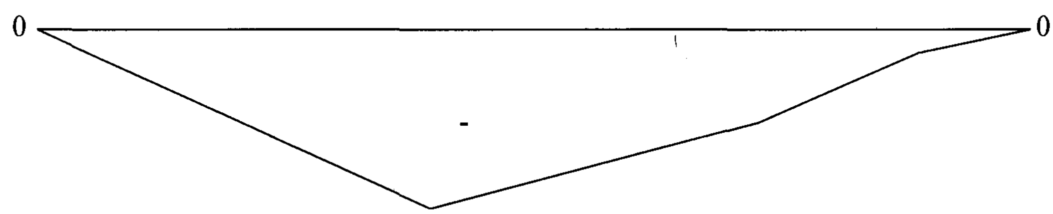


Figure 3.9 Vertical B.M. diagram

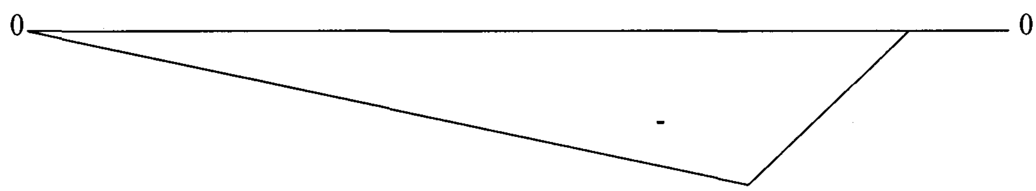


Figure 3.10 Horizontal B.M. diagram

Taking moments about A,

R_{bv} (0.46) = 11.0(0.23) + 104.8(0.51) + 0.07(0.54)

= 56.07

$R_{bv} = \frac{56.07}{0.46} = 121.9N$

$R_{av} + R_{bv} = 115.87$

$R_{av} + R_{bv} = 115.87$

$R_{av} = 115.87 - 121.9$

$= -5.99N$

Bending moments (B.M.) at A and B,

$M_a = M_E = 0$

B.M. at B, $M_b = -5.99(0.46) - 11.0(0.23) = -5.29N\text{-}m$

B.M. at C, $M_c = -5.99(0.23) = -1.38N\text{-}m$

B.M. at D, $M_d = -5.99(0.51) - 11(0.28) + 121.9(0.05)$

$= -0.1$

The bending moment diagram of the vertical loading is shown in fig 3.9.

Considering horizontal loading. Let R_{aH} and R_{bH} be the reactions on the bearings A and B respectively for horizontal loading.

$R_{aH} + R_{bH} = 99.28 + 0.19 = 99.47N$

Taking moments about A,

$R_{bH}(0.46) = 99.28(0.51) + 0.19(0.54)$

$RbH = \frac{50.73}{0.46} = 110.29N$

$R_{aH} = 99.47 - 110.29$

$= -10.82N$

Bending moments (B.M.) at A and B,

$M_a = M_d = 0$

B.M. at B, $M_b = -\ 10.82(0.46)$

$= -4.98N$

B.M. at D, $M_d = -\ 10.82(0.51) + 110.29(0.05)$

$= -\ 0.005$

The bending moment diagram for horizontal loading is shown in fig 3.10.

The resultant bending moment M_b is given as,

Resultant B.M. $M_b = \sqrt{(-1.38)^2 + (-4.98)^2}$

$= \sqrt{1.904 + 24.8}$

$= 5.17N\text{-}m$

For a shaft subjected to combined and torsion, the equivalent twisting moment,

$T_e = \sqrt{(Km \times Mb)^2 + (Kt \times Mt)^2}$

Where,

Km = Combine shock and fatigue factor for bending, and

Kt = Combine shock and fatigue factor for torsion.

$T_e = \sqrt{(1.5 \times 5.17)^2 \times (1 \times 6.44)^2}$

$= \sqrt{7.75^2 \times 6.44^2}$

$= 10N\text{-}m$

Equivalent twisting moment (T_e),

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

$$= \frac{\pi}{16} \times 40 \times 10^6 \times d^3$$

$$d^3 = \frac{160}{3.142 \times 40 \times 10^6}$$

$$d = \sqrt[3]{1.28 \times 10^{-6}}$$

$= 0.011m$

$= 11mm$

A 20mm diameter shaft is selected.

3.2.16 Bearing Selection

Equivalent load, W

$$W = X \cdot V \cdot Fr + Y \cdot Fa$$

Where,

X = radial factor

V = rotation factor

Fr = radial load (N)

Fa = thrust load (N)

Y = thrust factor

In order to determine the radial load factor (X) and axial load factor (Y), $\frac{Fa}{Fr}$ and $\frac{Fa}{Cr}$ are required. Since the value of basic static load capacity (Cr) is not know, therefore $\frac{Fa}{Cr} = 0.5$.

From table 27.4, the values of X and Y corresponding to $\frac{Fa}{Cr} = 0.5$ and $\frac{Fa}{Fr} = \frac{110.29}{121} = 0.91\ (which\ is\ greater\ than\ e = 0.44)\ are$

X = 0.56 and Y = 1 (from table)

Since the rotational factor (V) for most of the bearings is 1, therefore basic dynamic equivalent radial load,

$W = 0.56 \times 1 \times 121 + 1 \times 110.29$

$= 178.05N$

Basic dynamic load rating

$C = W\,(\frac{L}{10^6})^{1/3}$ (k = 3, for ball bearing)

Life of bearing in hours,

L_H = 10yrs × 300 × 10hrs ... (Assuming 300 working days per year)

= 30,000hrs

Life of bearing in revolutions,

$L = 60N \times L_H$

$= 60 \times 110 \times 30000$

$= 198 \times 10^6$

$C = 178.05\,(\frac{198 \times 10^6}{10^6})^{1/3}$

= 1037.8N

= 1.04KN

From Table 27.1, with diameter of shaft 20mm, bearing No 204 with outside diameter of 47 and width 14mm of radial load capacity of 121N and thrust load 110.3N will be suitable and was selected.

3.2.17 Power Required for Threshing

Torque of shaft drum, τ = force acting on the drum × radius of its neutral axis

= 190.5 × 0.05

= 9.5N-m

Power (p) = torque (τ) × angular velocity (ω)

$$P = 9.5 \times \frac{2\pi N}{60}$$

$$P = 9.5 \times \frac{2 \times 3.142 \times 50}{60}$$

= 49.75W

= 0.050Kw

3.2.18 Pulley size determination

The factors considered in the selection of pulley sizes was based on the speed of the drive and the driven Pulleys, the type and arrangement of belt drive (grooved or flat) and chain drive (roller or silent)) of the pulley or sprocket so as to minimize energy loss during Power transmission. The groove of the pulley was determined based on the driver.

- Estimated pulley diameter for thresher

$N_1D_1 = N_2D_2$ (Kurmi and Gupta, 2006) (57)

Where,

D_1 = diameter of drive pulley = 0.3m

N_1 = speed of drive pulley = 146rpm

D_2 = diameter of thresher pulley (m)

N_2 = speed of thresher pulley = 600rpm

$D_2 = \frac{N_1 D_1}{N_2}$

$= 146 \times \frac{0.3}{600}$

D_2 = 0.073m

3.2.19 Belt Selection

Belt selection was based on ASAE (1979) standard, given as

$L = 2C + 1.57\,(D - d) + \frac{(D-d)^2}{4C}$ (Kurmi and Gupta, 2006) (58)

Where,

L = Effective length of belt (mm)

C = Centre distance from drive to driven pulley (mm)

D = Diameter of driven pulley

d = Diameter of drive pulley

For belt length from blower to thresher pulley

$$L = 2 \times 0.41 + 1.57\,(0.3 - 0.07) + \frac{(0.3-0.07)^2}{4 \times 0.41}$$

$= 1.1811 + 0.14$

$= 1.32m$

D = 0.3m

d = 0.07m

C = 0.41m

L = 1.32m

3.2.20 Determination of Angle of contact

From the expression, the formula for calculating the angle of contact θ is given as:

$$\theta = 180 - \frac{[(D-d) \times 60]}{C} \quad (59)$$

I. Estimation of angle of contact for the thresher pulley:

$$\theta_1 = 180 - \frac{[(0.3 - 0.07) \times 60]}{0.41}$$

$= 146°$

3.3 Experimental Procedure

The experiment was carried out on one variety of guinea corn (Kaura). A combination of feed rates, f, of 230g/min and 30g/min with five levels of cylinder speeds, N, 400rpm, 500rpm, 600rpm, 700rpm were used. In all, 10 combinations of f and N were obtained and replicated. The harvested guinea corn material was weighed using a lysimeter before being fed into the thresher. After threshing, the materials collected from the chaff and grain outlet where weighed and recorded. The parameters measured were cylinder speed, fan speed, threshing efficiency, mechanical grain damage, scatter loss, throughput capacity, cleaning efficiency.

The moisture content was determined on dry basis. The grains were fed into the protemeter, and ground completely. The instrument was switched on and the moisture content is read from the screen after the grain was selected.

3.4 Instrumentation

The instruments used include:

Protemeter: It is a digital instrument used to measure moisture content of grains. The Grain master I-s protemeter Moisture meter was manufactured by Martin Lishman and has a range of 0 – 100%.

Meter Rule: This is used to obtain linear measurement in centimeters or millimeters.

Lysimeter: This is used to measure the weight in kilograms of the material. The lysimeter was manufactured by Norwood Instruments limited and has a range of 30kg maximum weight.

Stop watch: This is used to obtain time for each complete experiment in seconds. The stop watch model KK – 1045, and manufactured by Kenko and has a range of 0.00 – 9.59.59

3.5 Performance Evaluation

The experiment was carried out with two levels of feed rate, one crop variety and five levels of cylinder speed. Total combination for each experiment is 10. The formulae for evaluating the various performance parameters of the thresher were obtained from (Hyetson, 2000).

3.5.1 Threshing Efficiency, Tc (%): This can be defined as the ratio of quantity of threshed grain in sample to the quantity of grains in sample; to determine the threshing ability of the thresher.

$$Tc = 100 - \left(\frac{QU}{QT} \times 100\right) \quad (60)$$

Where Qu = quantity of unthreshed grain sample (kg)

QT = Total quantity of grain sample (kg)

3.5.2 Cleaning Efficiency Ce (%): This is defined as the ratio of grain collected to the total mixture of grain and chaff received at the main outlet.

$$Ce = \frac{WT - WC}{WT} \times 100 \quad (61)$$

Where WT = Total mixture of grain and chaff received at the main outlet (kg)

Wc = Weight of the chaff at the main outlet of the thresher (kg)

3.5.3 Mechanical Grain Damage, Md (%): This can be determined by collecting the broken grains in sample in order to determine the grain damaged in the threshed sample.

$$Md = \frac{Qb}{QT} \times 100 \quad (62)$$

Where Qb = Quantity of broken grains in the sample (kg)

3.5.4 Scatter Loss, SL (%): This is the loss acquired due to grain scattering around the thresher during operation.

$$SL = \frac{QL}{QT} \times 100 \quad (63)$$

Where QL = Quantity of grains scattered around the threshing operation

3.5.5 Throughput Capacity, Tc (kg/h): This is the total material quantity that passes through thresher in a given time.

$$Tc = \frac{QS}{T} \quad (64)$$

Where Qs = Quantity of material that pass through the threshed grain collector (kg)

T = Time taken to complete operation

3.6 Statistical Analysis

Tabular and graphical methods were used to compare the effects of speed and feed rates on threshing efficiency, mechanical grain damage, cleaning efficiency, scatter loss and throughput capacity for the guinea corn variety tested.

CHAPTER FOUR

4.0 RESULTS AND DISCUSSION

4.1 Results

The following is a summary table showing the performance parameters (threshing and cleaning efficiencies, scatter loss, mechanical damage and throughput capacity) obtain from the test carried out to ascertain the performance of the machine at two levels of feed rates. Details of the result are presented in appendix A (A_1, A_2, A_3, A_4, A_5). The data were obtained at 17.5% moisture content dry basis and 5cm concave clearance.

Table 4.1: Summary of various performance parameters at two levels of feed rates

Speed	T_c (%)		C_c (%)		S_L (%)		M_d (%)		T_c (kg/hr)	
	27.6	36	27.6	36	27.6	36	27.6	36	27.6	36
	Kg/hr	kg/hr	kg/hr	kg/hr	kg/hr	kg/hr	kg/hr	kg/hr	kg/hr	kg/hr
300	82	79	75	72	0.90	0.97	0.93	0.99	18.43	20.07
400	85	81	79	77	1.10	1.25	1.50	1.90	19.89	21.92
500	88	84	85	83	1.34	1.50	1.70	2.00	21.40	27.07
600	94	92	89	87	1.42	1.60	1.72	2.10	22.00	27.15
700	91	89	92	90	2.00	2.30	2.30	2.60	23.55	28.85
Mean	88	85	86	81.5	1.35	1.52	1.65	1.92		
SD	4.24	4.86	5.93	6.55	0.37	0.45	0.45	0.52		

Figures 4.1, 4.2, 4.3 and 4.4 show the effect of cylinder speed on threshing and cleaning efficiencies, scatter loss, mechanical damage and throughput capacity.

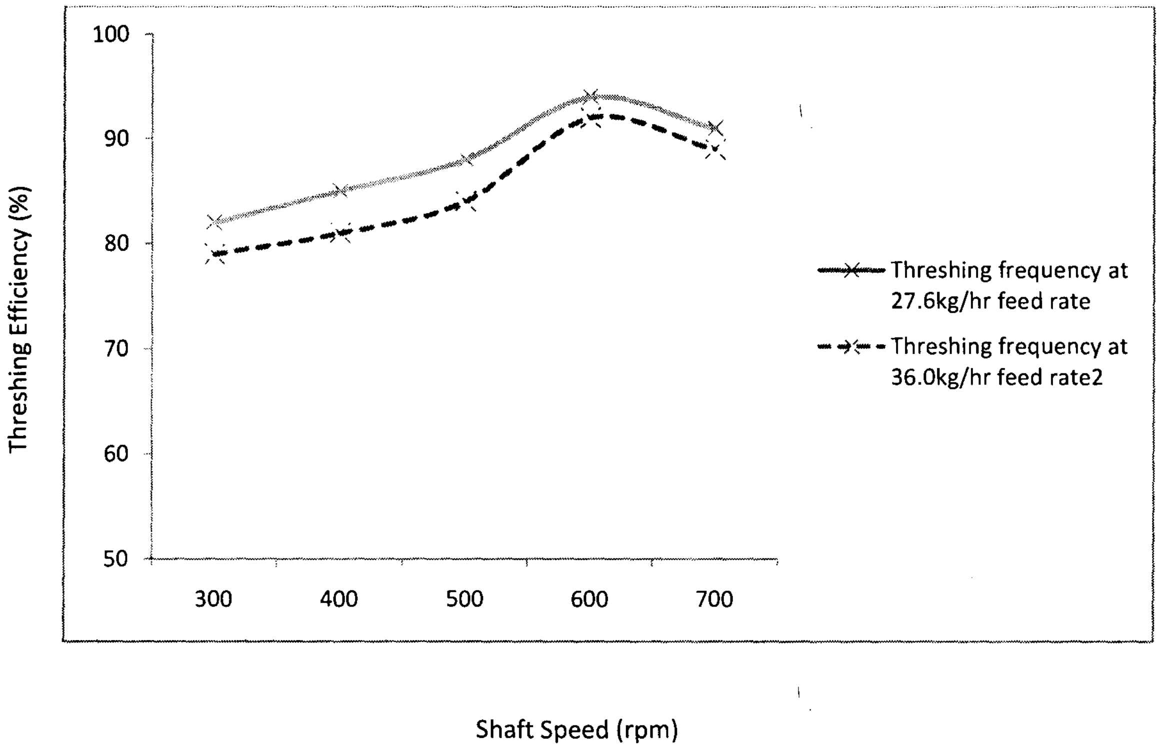


Fig. 4.1: Effect of shaft speed on threshing efficiency at different feed rates

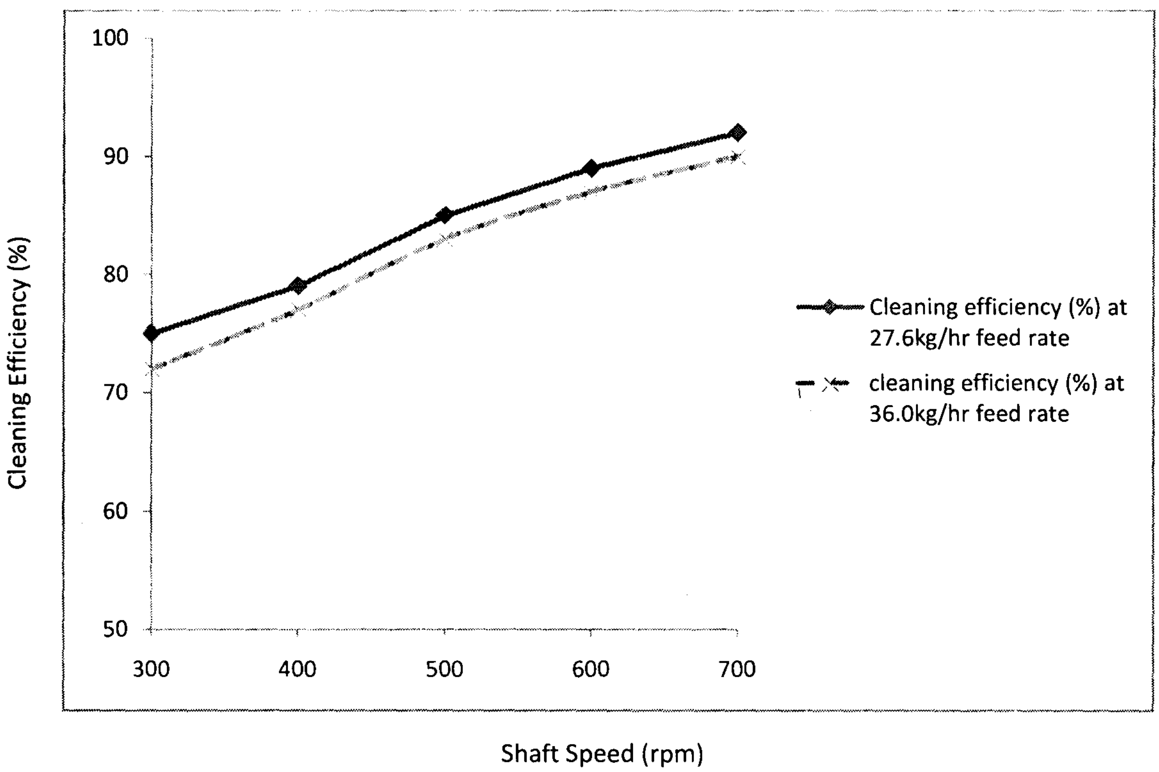


Fig. 4.2: Effect of shaft speed on cleaning efficiency at different feed rates

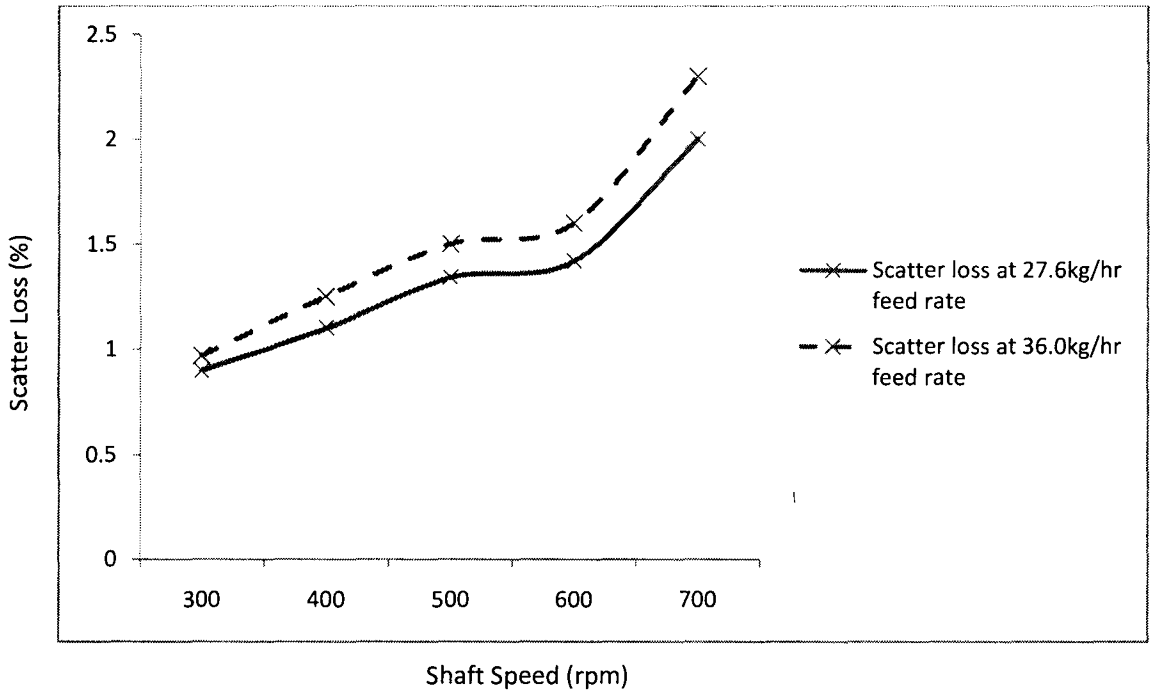


Fig. 4.3: Effect of shaft speed on scatter loss

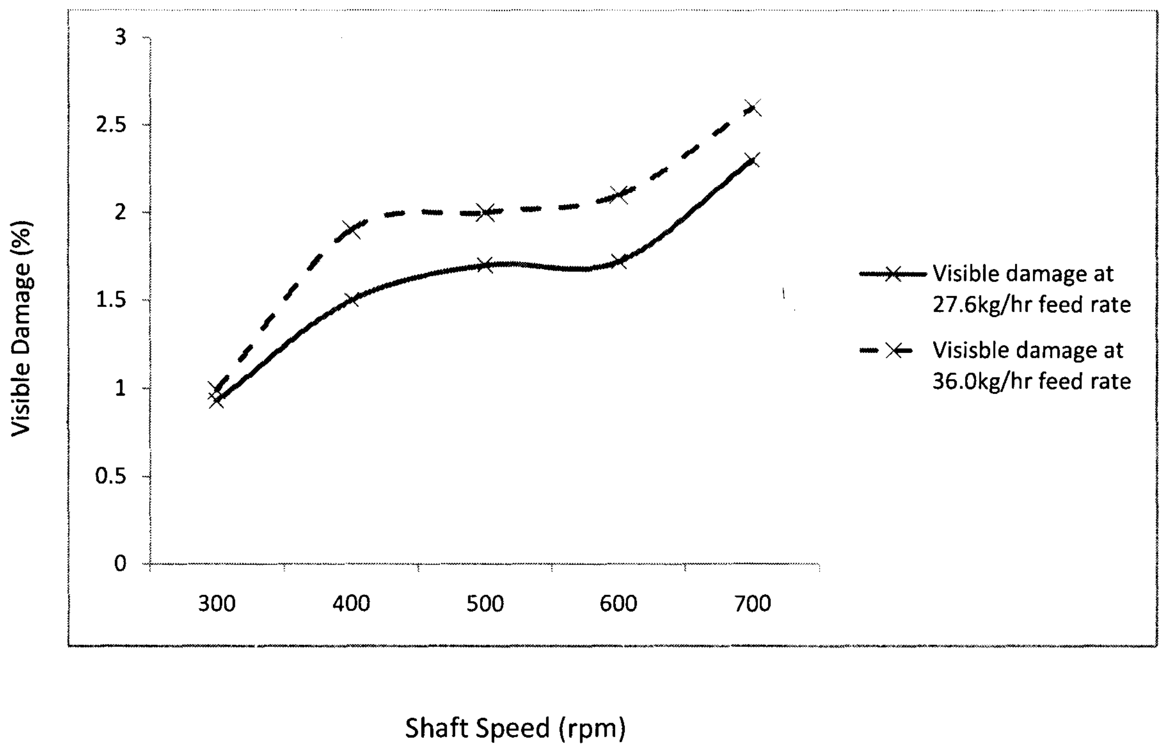


Fig. 4.4: Effect of shaft speed on mechanical (Visible) damage

4.2 Discussions

From table 4.1 and fig. 4.1, it can be seen that the effect of cylinder speed on threshing efficiency follows a linear pattern with threshing efficiency increasing from 82% to a maximum of 94% (at 27.6kg/hr feed rate) and 79% to a maximum of 92% (at 36kg/hr feed rate) as shaft speed increases from 300 to 700rpm regardless of the feed rate. This is because at higher threshing speed, the impact force is greater causing more grains to be threshed. But as cylinder speed increased, threshing efficiency dropped due to high impact which causes the stalk to be fall with the grain. Maximum (94%) and minimum (79%) threshing efficiencies were recorded at 700rpm (13.8kg/hr) and 300rpm (18kg/hr) respectively. Also as the feed rate increases, the efficiency decreases as seen in the figure, with the efficiency at 460g/min feed rate having higher efficiencies than that at 600g/min.

From table 4.1 and fig. 4.2, cleaning efficiency increases linearly with shaft speed with maximum cleaning efficiency recorded at 92% (27.6kg/hr) and minimum at 72% (36kg/hr) at 700rpm and 300rpm respectively. This is because at higher shaft speed, the velocity of the fan increase causes the efficiency of cleaning to increase. At higher feed rate 600g/min, the volume of materials to be handled increased allowing for some of the chaff to find its way through the grain outlet. This explains why the cleaning efficiency decreased with increase in feed rate.

From table 4.1 and fig. 4.3, the grain scatter also increases with increase in speed and feed rates, with maximum scattering (1.52%) recorded at 700rpm and 36kg/hr feed level.

Mechanical (visible) grain damage as seen from fig. 4.4 increase linearly from 0.93% and 0.99% to 2.30% at 460g/min and 600g/min respectively as cylinder speed increases from 300rpm to 700rpm. This is because as the speed increases at particular moisture content, the impact forces increases, causing more grain damage. Maximum grain damage (1.92%) was recorded at a cylinder speed of 700rpm and 36kg/hr feed rate.

Threshing (Throughput) capacity as seen from table 4.6 increased (from 19.89kg/hr and 21.92kg/hr to 23.55kg/hr and 28.85kg/hr at 460g/min and 600g/min feed rates respectively) with increase in speed and feed rate. The reason for this is that at higher speed, the threshing time decreases giving the machine room to handle more grains.

4.2.1 Working Principles

The thresher works on the rotary impact principle. The cylinder is designed to rotate at 600rpm during operation. The harvested ear heads of guinea corn are fed uniformly into the hopper. The ear falls by gravity onto the rotary cylinder, and whirled round between the concave and the rotating cylinder. This impact thus, brings about the threshing. The chaff and grains fall through the concave openings by gravity onto the stationary sieve while the straw exit through the straw exit. After falling onto the sieve, the air stream blows the chaff and other lighter materials through the chaff outlet by principle of buoyancy. The threshed guinea corn is then directed to a collecting pan by a slanted metal sheet beneath the sieve.

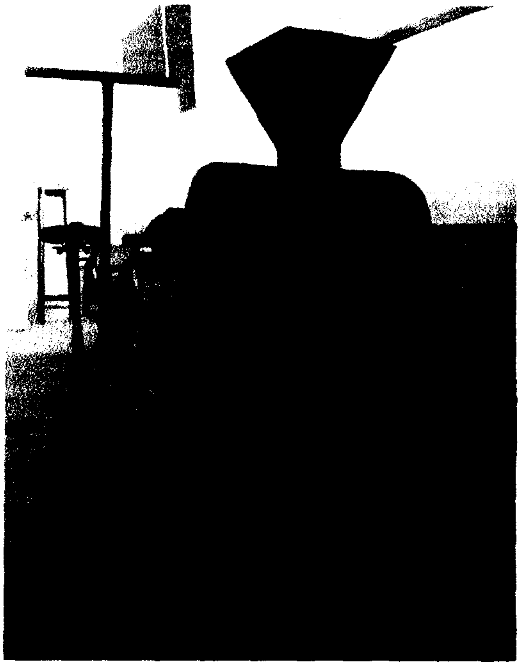


Plate 5.1: The Guinea corn Thresher after Fabrication

Plate 5.2: Test Material (ear head)

Plate 5.3: Test Product (Grains)

Plate 5.4: Test product (Chaff)

CHAPTER FIVE

5.0 CONCLUSION AND RECOMMENDATION

5.1 Conclusion

In conclusion, the effect of speed on the performance parameters of the designed thresher as summarized as follows:

- Threshing and cleaning efficiencies increases with increase in cylinder speed and feed rate.
- Grain damage and scatter loss also increased with increase in cylinder speed and feed rate.
- Throughput capacity increase with increase in cylinder speed and feed rates.

Best performance

The guinea corn thresher was designed, fabricated and performance evaluation carried out. With best performance combination at 600rpm cylinder speed, 27.6kg/hr feed rate, 5cm concave clearance and at moisture content of 17.5% dry basis. The threshing, cleaning, grain damage and scatter loss corresponding to these are 94%, 92%, 2.9%, and 2.30% respectively. The throughput at this performance combination was 22.0kg/hr.

Also, the designed guinea corn thresher is relatively affordable to local farmers.

5.2 Recommendations

To adequately explore the machine and improve its performance, the following are recommended for further study:

- The concave arrangement should be made adjustable so as to vary the cylinder concave clearance in order to ascertain its effect on threshing efficiency.

- A more comprehensive test should be carried out to ascertain the effect of moisture content on threshing performance of the machine.

Table 4.2: Bill of materials and cost Estimate

S/N.	Material	Dimension (mm)	Quantity	unit cost (N)	Total cost
1.	Angle iron	38×38×2	2	1500	3000
2.	Mild steel plate	2440×1220	1	3500	3500
3.	Bearing carbon steel	F 207	2	700	1500
4.	Shaft (m.s)	Ø25, L= 184	1	1200	1200
.	Smooth rod (m.s)	Ø10, L=600	2	1100	2200
.	Pulley (m.s)	Ø300	1	800	800
.	Belt rubberized	A-58	1	250	250
	Bolt & Nut (steel)	m8 × 25	20	20	400
	Paint (car paint)	Green	2lt	1000	2000
		Aluminum	1lt	850	850
.	Screen (m.s)	$\frac{1}{2}$ sheet	$\frac{1}{2}$	5000	5000
.	Electrode	G12	$\frac{1}{2}$ pkt	700	700
.	Pipe (m.s)	Ø100, L=560	1	1300	1300
	Saddle	Rubber & (m.s)	1	300	300
	Pedal	Rubber & (m.s)	1 set	200	200
	Crank	Mild steel	1	400	400
	Chain	Mild steel	$1\frac{1}{2}$	200	450
	Sprocket	Mild steel	1	200	200
	Angle iron	50×50×3	1	3500	3500
	Pipe(m.s)	Ø100	1	1000	1000

Material cost	28,970

Labour cost

The cost of labour was done considering the prevailing hourly rate of labour charged. The labour cost is the cost which can be traced to the threshing of the product. The labour cost was estimated to about 20% of the material cost.

Labour cost = $\frac{20}{100} \times 28{,}970 = N\ 5{,}794$

Overhead cost

A widely used method of calculating the overhead cost is by applying th overhead application rate. This is achieved by expressing overhead cost as % of direct labour cost. That is; they can't be traced directly with the product. Such cost include electricity, transportation, feeding. In this work, the overhead application rate was taken to be 15% of the material cost.

Overhead cost $= \frac{15}{100} \times 28970 = N\ 4{,}345.5$

Total cost of work = material cost + labour cost + overhead cost

= 28,970 + 5,794 + 4,345.5

= N39109.5 $\simeq$ N 40,000

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LIST OF APPENDICES

APPENDIX A

A1: Performance of the Guinea Corn Thresher at 300rpm Shaft Speed

eed Rate (g/min)	weight of Earhead	Q_U (g)	Q_O (g)	Q_L (g)	Q_b (g)	W_C (g)	T_E (g)	C_E (g)	S_L (g)	M_d (g)	T_C (kg/hr)
460	230	31.22	131.59	1.48	1.52	33.27	81	75	0.90	0.93	18.43
600	300	45.00	154.29	2.08	2.12	60.00	79	72	0.97	0.99	20.85
ean	265	38.11	142.94	1.78	1.82	46.64	80	73.5	0.94	0.96	
	35	6.89	11.35	0.30	0.30	13.37	1	1.5	0.04	0.03	

A2: Performance of the Guinea Corn Thresher at 400rpm Shaft Speed

ed Rate g/min)	weight of Earhead	Q_U (g)	Q_O (g)	Q_L (g)	Q_b (g)	W_C (g)	T_E (g)	C_E (g)	S_L (g)	M_d (g)	T_C (kg/hr)
60	230	27.93	135.65	1.81	2.46	28.64	83	79	1.10	1.50	19.89
00	300	40.72	174.39	2.67	4.07	39.92	81	77	1.25	1.90	21.92
n	265	34.33	155.02	2.24	3.27	34.28	82	78	1.18	1.70	
	35	6.40	19.37	0.43	0.81	5.64	1.0	1.0	0.08	0.20	

A3: Performance of the Guinea Corn Thresher at 500rpm Shaft Speed

Feed Rate (g/min)	weight of Earhead	Q_U (g)	Q_O (g)	Q_L (g)	Q_b (g)	W_C (g)	T_E (g)	C_E (g)	S_L (g)	M_d (g)	T_C (kg/hr)
460	230	21.36	142.85	2.20	2.79	21.44	87	85	1.34	1.70	21.4
600	300	34.29	183.69	3.21	4.29	30.60	84	83	1.50	2.0	27.0
ean	265	27.83	163.27	2.17	3.54	26.02	85.5	84	1.42	1.85	
)	35	6.47	20.42	0.51	0.75	4.58	1.5	1.0	0.08	0.15	

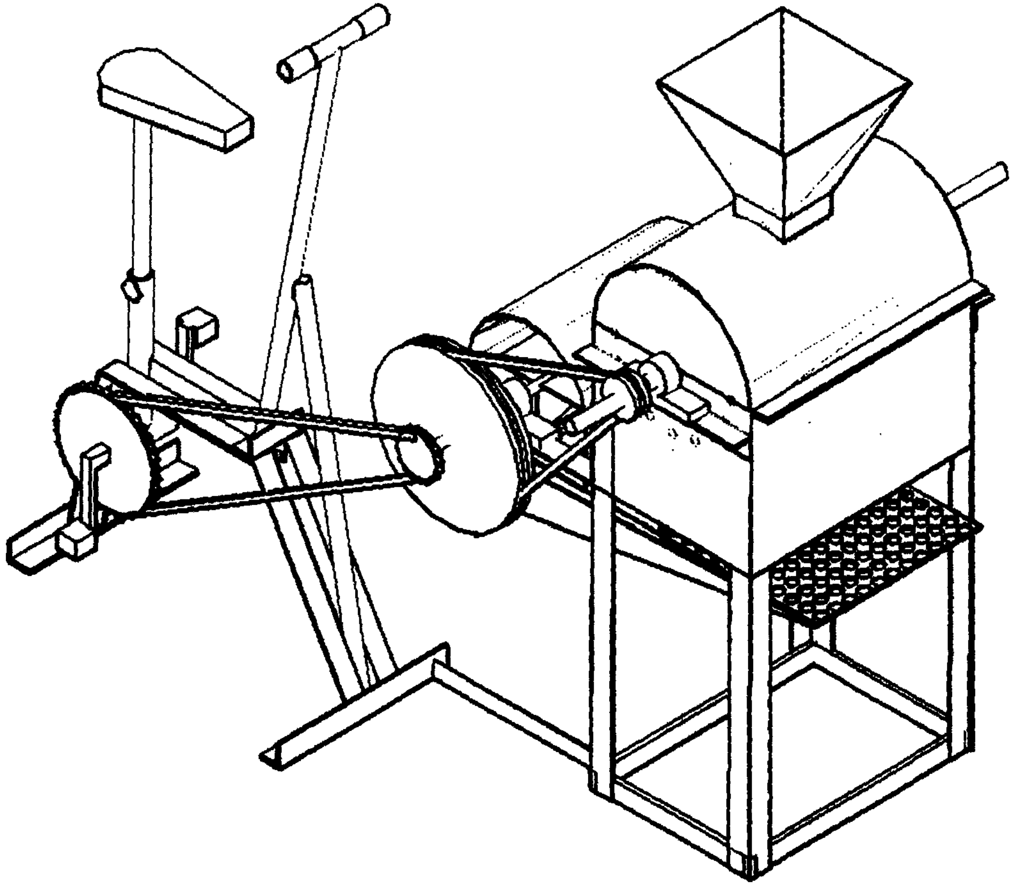
A4: Performance of the Guinea Corn Thresher at 600rpm Shaft Speed

ed Rate (g/min)	weight of Earhead	Q_U (g)	Q_O (g)	Q_L (g)	Q_b (g)	W_C (g)	T_E (g)	C_E (g)	S_L (g)	M_d (g)	T_C (kg/hr)
60	230	9.85	142.08	3.29	3.78	12.35	94	92	2.0	2.3	22.0
00	300	17.14	175.46	4.93	5.57	21.69	92	89	2.3	2.6	27.1
an	265	13.50	158.77	4.11	4.68	17.02	93	90.5	2.15	2.45	
	35	3.64	16.69	0.82	0.90	2.34	1.0	1.5	0.15	0.15	

A5: Performance of the Guinea Corn Thresher at 700rpm Shaft Speed

-eed Rate (g/min)	weight of Earhead	Q_U (g)	Q_O (g)	Q_L (g)	Q_b (g)	W_C (g)	T_E (g)	C_E (g)	S_L (g)	M_d (g)	T_C (kg/hr)
460	230	13.14	134.52	2.33	2.83	16.63	92	89	1.42	1.72	23
600	300	21.43	167.79	3.43	4.50	25.7	90	87	1.60	2.10	28
ean	265	17.29	151.16	2.88	3.67	20.85	91	88	1.51	1.91	
)	35	4.15	16.64	0.55	0.55	4.22	1.0	1.0	0.09	0.19	

)TE: Q_O = clean grains



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DEPARTMENT OF AGRIC. AND BIORESOURCES ENGINEERING					
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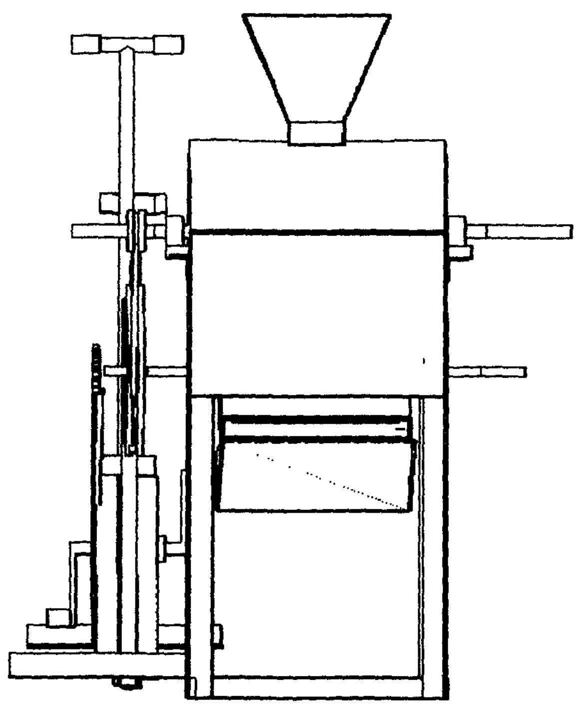

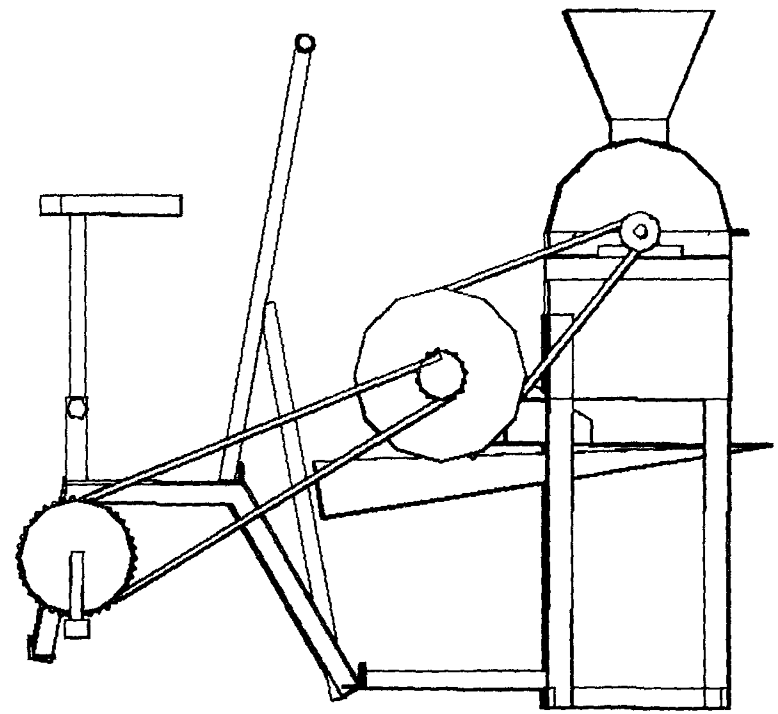

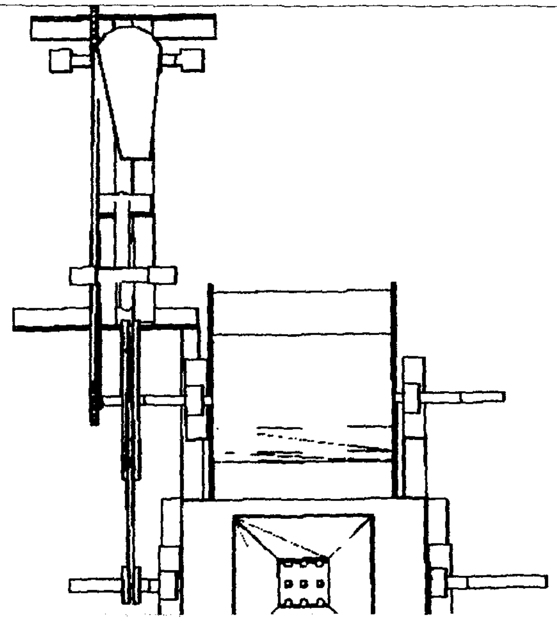

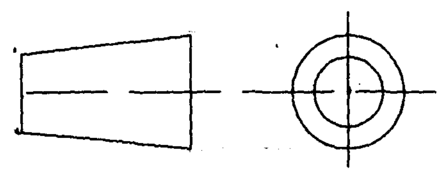

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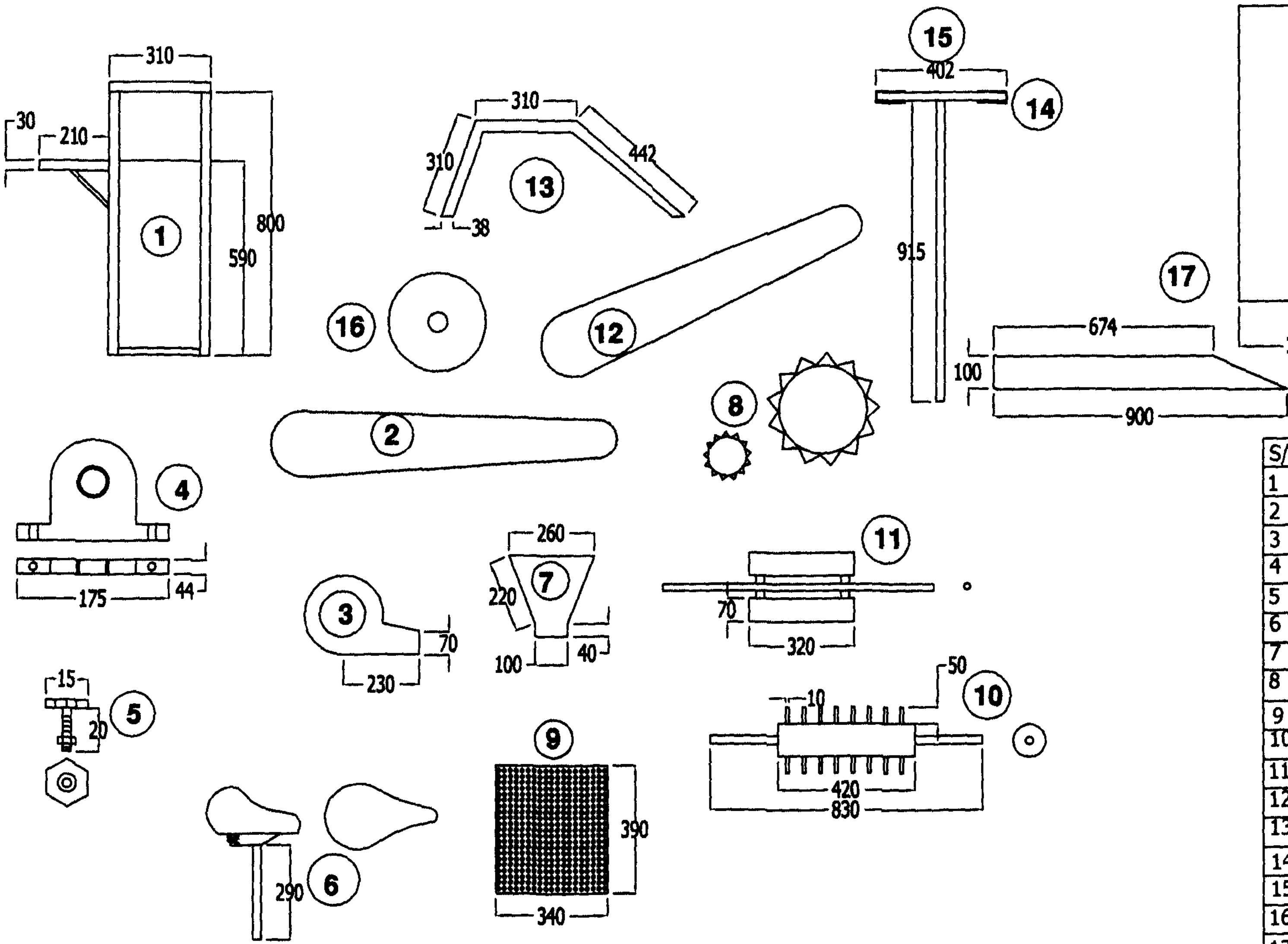

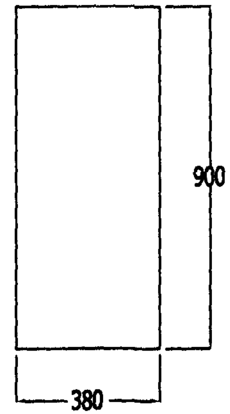

S/N	DESCRIPTION	MATERIAL
1	FRAME	MILD STEEL
2	CHAIN	MILD STEEL
3	FAN CASE	MILD STEEL
4	BEARING	CARBON STEEL
5	BOLT & NUT	ALLOY STEEL
6	SADDLE (SEAT)	RUBBER & MILD STEEL
7	HOPPER	MILD STEEL
8	SPROCKET(S)	ALLOY STEEL
9	SIEVE	MILD STEEL
10	THRESHING MECHANISM	MILD STEEL
11	BLOWER MECHANISM	MILD STEEL
12	BELT	RUBBER
13	PEDDAL STAND	MILD STEEL
14	HANDLE BAR RUBER	RUBBER
15	HANDLE BAR	MILD STEEL
16	PULLEY	MILD STEEL
17	TROUGH	MILD STEEL

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