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DESIGN AND CONSTRUCTION OF UNIVERSAL GRAIN THRESHER

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ABSTRACT

Nigeria as an agricultural country has many farm products such as maize, millet, sorghum, rice, groundnut, cowpea and many others which are always threshed in traditional ways. Labour and time are wasted in the processing and recovering of the end products and therefore sometimes yields are low.

As most universal grain threshers in Nigeria are imported, their costs are high and many farmers cannot afford them.

The need to reduce cost, labour and time also to use locally available raw materials necessitated the design and construction of the universal grain threshing machine being presented in this paper.

The major components of the machine are the threshing unit, sieves, blower and the prime mover, each fabricated separately and assembled on a rigid frame.

The threshing action is accomplished by shredding and
rubbing action of the rotating stub spike toothed drum on a stationary concave with matching tooth. The harvested and dried crops are fed into the threshing chamber through the hopper.

The threshed grains and the chaff escape through the perforated concave plates on to a reciprocating sieve housing. The grains are cleaned by a blast of air being issued by the blower. The threshed and cleaned grains are then collected through the outlet channel.

Standard evaluation were done by loading 20kg of sorghum head. The procedure was repeated for millet and maize.

The results showed that the machine has an overall efficiency of 85.5%.

The machine can thresh from 2.5 to 5 tonnes daily depending on the type of crops.
INTRODUCTION

The types of crop grown in many parts of Nigeria are maize, millet, sorghum, rice, wheat and grain legumes such as cowpea, bambara nut and groundnut. Processing of such crop after the harvest is of utmost importance because they must be processed to some extent before being sold or stored. Most of the processing are done traditionally. Labour, time and expenses involved in this traditional process are so huge that either the prices of such commodities rise high in the open market or profits to the producer become low.

Conventionally, there are different types of threshing machine which are imported to the country. However their prices are very high, such as much as N110,000.

The needs to reduce costs, labour and time as well as to use locally available raw materials have necessitated the design and construction of universal grain threshing machine reported in this paper. This work will entail these following design procedures. While a few important design methods will be outlined in this paper. The details of the design methods are described elsewhere. [1]

2.0 DESIGN METHODOLOGY

Synthesis of various new and old ideas are employed in this design so as to produce an overall new idea.
2.1 Functional Requirements

The machine is designed to shell maize and groundnut as well as thresh and winnow grain such as millet, sorghum, wheat, rice, cowpea etc.

To satisfy these requirements the machine has the following components:

(i) Threshing drum (ii) Concave (iii) Blower (iv) Group of Sieves (v) Frame and various drives (vi) Hopper (vii) Source of power

The machine derives its power from a five (5) horsepower (3.725 kw) single phase electric motor.

See Fig. 1a, b, c for the isometric drawing of the machine.

2.2 Design Features

Some of the important factors considered in design of this machine are

(i) High efficiency (ii) Minimum cost of construction (iii) Ease and Safety of use (iv) Ease of Inspection (v) Incorporated standard consumable parts (vi) Ease of maintenance (vii) Adaptability to a wide variety of crops and (viii) Durability.

2.3 Power Transmission

In the design of this machine, power is first transmitted from the prime mover to the blower shaft by a flat belt drive.
2.4 Analysis and Design of flat belt drive

Transmission of power from output shaft to blower shaft.

Power generated by motor \( P_1 \) = 3.725 kW

Motor Speed \( N_1 \) = 1440 rpm

Angular Speed of motor \( w_1 \) = \( \frac{2\pi N_1}{60} \) = \( \frac{2\pi \times 1440}{60} \) = 150.79 rad/sec.

Motor pulley diameter \( d_1 \) = 180mm

Blower speed \( N_2 \) = 1000 rpm

Angular speed of blower \( w_2 \) = \( \frac{2\pi N_2}{60} \) = \( \frac{2\pi \times 1000}{60} \) = 104.73 rad/sec.

Efficiency of flat belt drive

\[ n = 0.96 \]  \[ \text{[1]} \]

Power at the blower shaft \( P_2 \) = \( P_1 \times n \)  \[ \text{[2]} \]

\[ P_2 = 3.725 \times 10^3 \times 0.96 = 3576 \text{ Watts.} \]

\[ P_2 = 3576 \text{KW} \]

The diameter of the blower pulley \( d_2 \) can thus be computed as follows:

\[ d_2 = \frac{d_1 w_1}{w_2} = \frac{180 \times 150.79}{104.73} = 259.16 \text{mm} \]

\[ d_2 \approx 250 \text{mm} \]

To refine the angular speed of the blower shaft, assuming a relative speed loss factor \( E = 0.01 \), \[ \text{[1]} \] the angular speed of the driven shaft is now

\[ w_2 = \frac{d_1 w_1 (1-E)}{d_2} = \frac{180 \times 150.79 (1-0.01)}{250} = 107.48 \text{rad/sec.} \]
To determine the belt speed \( (V_1) \)
\[
V_1 = \frac{w_1 d_1}{2} = \frac{150.79 \times 180}{2 \times 1000} = 13.57 \text{ m/s}
\]
The belt speed lies within the recommended limit of 10 to 30 \( \text{m/s} \).

To determine the centre distance \( (a_1) \) of the pulleys
\[
a_1 \geq 2(d_1 + d_2) \quad [3]
\]
\[
a_1 = 2(180 \times 250) = 850 \text{ mm}
\]
\[
a_1 = 900 \text{ mm} \quad [3]
\]
To determine the required belt length \( (L_1) \)
\[
L_1 = 2a_1 + \left( \frac{\pi}{2} \right) (d_1 + d_2) + \left( \frac{(d_2 - d_1)^2}{4a_1} \right) \quad [3]
\]
\[
= 2(900) + \frac{(1.57)(180+250)+(250-180)}{4 \times 900} = 24.76 \text{ mm}
\]
\[
L_1 = 24.76 \text{ mm}
\]
The angle of lap of the belt on the smaller pulley is given as thus,
\[
\alpha_1 = 180^\circ - 57^\circ \frac{(d_2 - d_1)}{a} = 180 - 57 \frac{(250 - 180)}{900} = 175.56^\circ
\]
The angle of lap for smaller pulley should not be less than 150\(^\circ\) for an effective drive.

The angle of lap of the belt on the larger pulley is also given as thus:
\[
\alpha(a_1) = 180^\circ + 57^\circ \frac{d_2 - d_1}{a} \quad [3]
\]
\[
a_1 = 180^\circ + 57^\circ \frac{(250 - 180)}{900} = 184.44
\]

By using the rubberised fabric belt, the thickness must meet its requirement.
The belt thickness \( b_1 = \frac{d_1}{20} = \frac{180}{30} = 6.0 \text{mm} \)

The dimension for a ply of a rubberished belt including the interlayer \( (d_y) \) is 1.5mm \([3]\) and should not exceed 7 plies. Therefore the number of plies is thus computed:

\[
N_{\text{ply}} = \frac{b_1}{d_y} = \frac{6.0}{1.5} = 4 \text{ plies}
\]

The belt width \((b_0)\) can be computed from the following formula:

\[
b_0 = \frac{F_t}{\left[\sigma\right]} \quad \text{[4]}
\]

Where \( F_t = F_{\text{stress}}, N \)

\( b_1 = \text{Belt thickness} \)

\( \left[\sigma\right] = \text{allowable effective stress} \)

\( F_t = \frac{P_1}{V} = \frac{3.725 \times 10^3}{13.57} = 274.5 \text{N} \)

\( \left[\sigma\right] = \delta_{\text{eff}} \times C_{\text{c}} \times C_{\text{v}} \times C_{\text{s}} \quad \text{[3]} \)

where \( \delta_{\text{eff}} = \text{nominal effective stress determined under standard condition. For rubberised fabric belt with tensile pre-stress} (\delta_0) = 1.8 \text{ MPa.} \quad \text{[3]} \)

The nominal effective stress \( \delta_{\text{eff}} \) is determined according to the ratio \( \left( \frac{d_1}{b_1} \right) \):

\[
\frac{d_1}{b_1} = \frac{180}{6} = 30
\]

\[
\delta_{\text{eff}} = 2.17 \text{ MPa} \text{ if } \frac{d_1}{b_1} = 30. \quad \text{[3]}
\]
The coefficient \( C_0 \) takes care of the mutual arrangement for the shafts. As reported by \([1]\), when the drive from the prime mover to the blower shaft lies within the range of 80° to 90° to the horizontal, \( C_0 = 0.8 \).

The coefficient \( C_\ell \) accounts for the effect for the angle of contact for the smaller pulley. If \( \ell \) = 175.56°, we can round it up to 180° which corresponds to the value of \( C = 1 \) \([1]\).

The coefficient \( C_v \) accounts for the influence of the tension due to centrifugal force which tends to reduce traction. It depends on the peripheral speed. When the speed \( \ell \) = 13.57 ms\(^{-1}\), it can be rounded up to 15 ms\(^{-1}\) which corresponds to the value of \( C_v = 0.99 \) \([1]\).

The coefficient \( C_s \) is the service factor which takes care of the type of driven machine and the daily times in operation. The value is to be reduced by 0.1 in two shift operation and by 0.2 in three shift operation \([1]\).

\[
C_s = 0.99 - 0.1 = 0.8
\]

Thus \( f = 2.17 \times 0.8 \times 1 \times 0.99 \times 0.8 = 1.375 \) MPa.

The belt width can thus be computed as

\[
b_o = \frac{F_t}{[f] b_1} = \frac{274.5}{1.375 \times 6} = 33.27 \text{mm}
\]

As reported by \([3]\), the appropriate standard belt width is 32 mm. However, such dimension is not available in our market here. Therefore the standard size available is 50 mm. For this design, it was accepted.
Belt width = 50mm

and the pulley width (B) = 63mm

The contact surface of the run of the pulley is crowned to prevent the flat belt running off in operation. As a rule a crown is provided by an arc of a circle. The camber can be determined from the approximately standard table. [1]

Camber for the pulley is 0.8mm.

Determination of the pre-tension in the belt

\[ F_0 = \sigma_{bob} b = 1.8 \times 50 \times 6 = 540N \quad [4] \]

Where \( \sigma_0 \) = the tensile pre-stress in the belt

To find the force acting on the shaft and the bearings,

\[ F_a = 2F_0 \sin \frac{\alpha_1}{2} \quad [4] \]

\[ F_a = 2 \times 540 \times \sin \frac{175.6}{2} \]

\[ = 1079.2N \]

To find the maximum force applied to the shaft due to belt tensioning, \( F_{bmax} = 1.5F_0 = 1.5 \times 540 = 810N \)

2.5 Analysis and Design of Flat belt drive power transmission to the thresher drum

The thresher drum derives its power from the blower shaft, with belt drive inclination of 20° to the horizontal.

Power available from the blower shaft \( (P_2) = 3.576KW \)

Flat belt drive efficiency \( \eta = 0.96 \)

Power available at the drum shaft \( (P_3) = P_2 \times \eta \quad [2] \)

\[ P_3 = 3.576 \times 0.96 = 3.433 \text{ KW} \]
The drum is designed to run at a speed of not more than 600 rpm. Higher drum speeds damage grain in the threshing process.

Angular speed of drum \( (W_3) = \frac{2\pi \times 600}{60} \)

\( W_3 = 62.8 \text{ rad/sec.} \)

To determine the drum pulley diameter \( (d_3) \)

\[
d_3 = \frac{w_2 d_2}{w_3} = \frac{107.48 \times 250}{62.8} = 427.86 \text{mm}
\]

Select standard value for the pulley diameter \( (d_3) \).

The nearest standard size is 400mm.

\( d_3 = 400 \text{mm} \)

To refine the angular speed of the drum shaft assuming a relative speed loss factor \( E = 0.01 \).

The angular speed of the driven shaft is

\( W_3 = \frac{d_3 w_2 (1-E)}{d_3} = \frac{250 \times 107.48 (1-0.01)}{400} = 66.5 \text{ rad/sec.} \)

To determine the belt speed \( (V_2) = \frac{w_2 d_2}{2} \)

\[
= \frac{250 \times 107.48}{2 \times 10^3} = 13.44 \text{ m/s}
\]

The belt speed lies within the design limits of 10 to 30 m/s.

To determine the centre distance of the two pulleys

\[
a_2 \geq \frac{2(d_2 + d_3)}{3}
\]

\[
a_2 \geq \frac{2(250 + 400)}{1300} = 1300 \text{mm}
\]
To determine the required belt length \( L_2 \)

\[
L_2 = 2a_2 + (\pi/2) (d_2+d_3)+(d_3-d_2)^2
\]

\[
= (2 \times 1300) + (1.57)(250+400) + (400-250)^2
\]

\[
= 3625\text{mm}
\]

\[
L_2 = 3.625\text{m}
\]

Due to the compactness of the machine this dimension is not feasible.

The angle of lap of the belt on the smaller pulley

\[
\alpha_2 = 180^\circ - 57^\circ \frac{(d_3-d_2)}{a}
\]

\[
= 180^\circ - 57^\circ \frac{(400-250)}{1300} = 173.4^\circ
\]

The angle of lap on the larger pulley is

\[
\alpha_{2a} = 180^\circ + 57^\circ \frac{400-250}{1300} = 186.6^\circ
\]

Using the rubberized fabric belt, the thickness meet its requirement.

The belt thickness \( b_2 \)

\[
b_2 = d_2/30 = 250/30 = 8.33\text{mm}
\]

The dimension for a ply of a rubberized belt including the inner layer is 1.5mm and should not exceed 7 plies. Therefore the number of plies of the belt can be determined

\[
N_{\text{ply}} = \frac{8.33}{1.5} = 5.5
\]

\[
N_{\text{ply}} = 6 \text{ plies}
\]

The belt width can be computed from the following formula

\[
b_0 = F_L/((\delta) \times b1)
\]

The terms involved had been defined earlier.
\[
F_b = \frac{P_2}{V_2} = \frac{3.433 \times 10^3}{13.44} = 255.4 \text{N}.
\]

The above terms are defined earlier and their corresponding values can be obtained from [1].

\[
d_2/b_2 = \frac{250}{8.33} = 30.012
\]

\[
\text{Thus} \quad \frac{C_0}{C_1} = \frac{2.17 \times 10^{-1}}{0.97} = 1.67 \text{MPa}
\]

The belt width can thus be computed

\[
b_0 = \frac{F_t/[C_1]}{b_1} = \frac{255.4}{1.67 \times 8.33} = 18.36 \text{ mm}
\]

This dimension of the belt is not realistic compared to its duty and moreover such width for a flat belt is not available around it. In this case the belt width to be assumed is 50 mm.

Therefore belt width \( (b_0) = 50 \text{ mm}\)

To determine the pretension on the belt

\[
F_o = \rho \rho b_0 b_1 = 1.8 \times 50 \times 8.33 = 750 \text{ N}
\]

To find the force acting on the shaft and bearings

\[
b_0 = 2F_o \sin \frac{\alpha_2}{2} = 2(750) \sin \frac{17.3.4}{2} = 1499.5 \text{ N}
\]

To find the max force applied to the shaft due to belt tensioning.

\[
F_{b\text{max}} = 1.5F_o = 1.5 \times 750 = 112.5 \text{ N}
\]
Diameter of pulley = 400 mm.
Corresponding camber for run width for upto 125 mm is given 1.0mm. [2]

3.0 DESIGN AND ANALYSIS OF POWER TRANSMISSION SHAFT

3.1 Determination of Blower Shaft Diameter
The blower shaft diameter \( d_s \) can thus be determined as follows:

\[
\frac{d_s^2}{2} = \left( \frac{16(K_t M_t)^2 + (K_b M_b)^2}{s} \right)^{\frac{1}{2}}
\]

Where \( K_t \) and \( K_b \) = combine shock and fatigue factors.
\( s \) = allowable stress for shafts having keyways = 40 MN/m\(^2\) [4]
\( M_t \) and \( M_b \) = the torsional and resuctact bending moments respectively

\[
M_t = \frac{9500 \times kw}{rpm}, \quad N_m [4]
\]
\[
M_b = 73.38 \text{ Nm from the bending moment diagrams} [1]
\]

\[
d^3_s = \frac{16}{40 \times 10^6} \left[ (1 \times 34.15)^2 + (1.5 \times 73.03)^2 \right]^{\frac{1}{2}} [4]
\]

\[
d_s = 0.024\text{m}
\]
\[
d_s = 24.45\text{mm selecting a factor of safety 1.23} [1]
\]
\[
d_s = 24.45 \times 1.23 = 30.06\text{m}
\]

The thresher shaft diameter \( d_T \) can be computed as follows

\[
\frac{d_T^3}{s} = \left\{ \frac{(K_t M_t)^2 + (K_b M_b)^2}{s} \right\}^{\frac{1}{2}} [4]
\]

\[
d_T = \frac{16}{40 \times 10^6} \left\{ (1 \times 54.64)^2 + (1.5 \times 101.18)^2 \right\}^{\frac{1}{2}} [4]
\]

\[
d_T = 0.0274\text{m}
\]
\[
d_T = 27.4\text{mm}
\]
Now selecting a power of safety of 1.28

\[ d_T = 27.4 \times 1.28 = 35.05 \text{mm} \]

The permissible angle of twist for the shaft can be determined as follows

\[ \theta = \frac{584 \times M_t \times L}{G_d^4 \times d_s^4} \]  \[ \text{[4]} \]

Where \( \theta \) = angle of twist (degree)
\( L \) = Length of shaft m
\( M_t \) = Torsional moment, Nm
\( G \) = Torsional modulus of elasticity, N/m²
\( d_s \) = Shaft diameter (blower shaft)

\[ \theta = \frac{584 \times 24.15 \times 0.72}{80 \times 10^9 \times 0.03^4} = 0.22^0/\text{m} \]

It can be seen that the shafts are torsionally rigid.

The bending stress on the blower shaft can be computed as follows

\[ \sigma_b = \frac{32 \times M_b}{d_s^3} \]  \[ \text{[4]} \]

\[ \sigma_b = \frac{32 \times 73.03}{0.03^3} = 27.55 \times 10^6 \text{ N/m}^2 \]

The bending stress of the threshing drum shaft is

\[ \sigma_T = \frac{32 \times M_t}{d_T^3} = \frac{32 \times 104.18}{0.035^3} = 24.03 \times 10^6 \text{ N/m}^2 \]

The combined torsional stress can also be determined as follows

\[ \sigma_{xy} = 16 \frac{M_t}{d^3} \]  \[ \text{[3]} \]

The torsional stress in blower shaft is \( \sigma_{xy} \).
The torsional stress for the threshing drum shaft \( \tau_{xy} \) is given by:

\[
\tau_{xy} = \frac{16 \times 91.62}{0.0353^3} = 10.88 \times 10^6 \text{ N/m}^2
\]

There are no axial loads on the shafts. The calculated stress are well below the yield strength of steel.

3.3 Grain Threshing

The theoretical principle applicable to the threshing processes are:

(i) Aerodynamic separation based on the terminal velocities.
(ii) Movement of the crop through the sieves
(iii) Escape of the grain through the opening of the sieves.

Aerodynamic separation is based on the pneumatic conveying of chaff and straw in turn depends upon the terminal velocities and the drag coefficient of the different components in the crop oscillating conveyors.

Grain motion, through the chaff and straw mat is due to the gravity and the resistive force caused by the straw. The escape of grain through the sieve opening is based on the theory of probability.

The following assumptions are made:

(i) The drag coefficient is independent of the air velocity.
(ii) The particles (grains) are accelerated as free bodies and not as a mat.
(iii) The velocity of air through the sieve housing is constant.
(iv) Air flow above the upper screen is streamlined and parallel to the orientation.
Result and Discussion

After the design and construction of the machine to ascertain whether the machine is functioning accordingly.

The standard evaluation were done by loading 20kg of sorghum head. Few heads were fed into the machine at a time and it is noticed that the machine is functioning accordingly.

The following apparatus were adopted during the process:
(a) Universal threshing machine
(b) Weighing scale
(c) Stop watch
(d) Sorghum heads

See Table 1 for the result.

<table>
<thead>
<tr>
<th>Crop Type</th>
<th>Qty before threshing</th>
<th>Weight of threshed</th>
<th>Weight of husks/chaffs</th>
<th>Time taken</th>
<th>Loss of grain/husks</th>
<th>Power consumed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sorghum</td>
<td>1kg</td>
<td>0.7kg</td>
<td>0.28kg</td>
<td>15sec</td>
<td>0.02kg</td>
<td>3.048kW</td>
</tr>
<tr>
<td></td>
<td>2kg</td>
<td>1.5kg</td>
<td>0.45kg</td>
<td>20sec</td>
<td>0.05kg</td>
<td>3.075kW</td>
</tr>
<tr>
<td></td>
<td>3kg</td>
<td>1.0kg</td>
<td>0.96kg</td>
<td>26sec</td>
<td>0.04kg</td>
<td>3.10kW</td>
</tr>
<tr>
<td></td>
<td>4kg</td>
<td>2.7kg</td>
<td>1.02kg</td>
<td>32sec</td>
<td>0.28kg</td>
<td>3.127kW</td>
</tr>
<tr>
<td></td>
<td>5kg</td>
<td>3.3kg</td>
<td>1.5kg</td>
<td>36sec</td>
<td>0.20kg</td>
<td>3.154kW</td>
</tr>
</tbody>
</table>

The procedure followed in the sorghum trial is repeated on this experiment for millet and maize and the following results are also obtained.
Table 2: Result obtained from Millet Trial

<table>
<thead>
<tr>
<th>Crop Type</th>
<th>Qty before threshing</th>
<th>Weight of threshed</th>
<th>Weight of husks/chaffs</th>
<th>Time taken</th>
<th>Loss of grain/ husks</th>
<th>Power consumed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Millet</td>
<td>1kg</td>
<td>0.6kg</td>
<td>0.35kg</td>
<td>12sec</td>
<td>0.05kg</td>
<td>2.28KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>2kg</td>
<td>1.3kg</td>
<td>0.6kg</td>
<td>21sec</td>
<td>0.1kg</td>
<td>2.306KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>3kg</td>
<td>1.8kg</td>
<td>1.0kg</td>
<td>28sec</td>
<td>0.2kg</td>
<td>2.328KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>4kg</td>
<td>2.2kg</td>
<td>1.5kg</td>
<td>30sec</td>
<td>0.3kg</td>
<td>2.344KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>5kg</td>
<td>2.7kg</td>
<td>2.1kg</td>
<td>36sec</td>
<td>0.2kg</td>
<td>2.36KW</td>
</tr>
</tbody>
</table>

Table 3: Result obtained from Maize Trial

<table>
<thead>
<tr>
<th>Crop Type</th>
<th>Qty before threshing</th>
<th>Weight of Grain</th>
<th>Weight of husks/chaffs</th>
<th>Time taken</th>
<th>Loss of grain/ Chaff</th>
<th>Power consumed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maize</td>
<td>1kg</td>
<td>0.6kg</td>
<td>0.3kg</td>
<td>13sec</td>
<td>0.1kg</td>
<td>3.354KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>2kg</td>
<td>1.3kg</td>
<td>0.6kg</td>
<td>18sec</td>
<td>0.1kg</td>
<td>3.392KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>3kg</td>
<td>2.1kg</td>
<td>0.8kg</td>
<td>26sec</td>
<td>0.1kg</td>
<td>3.411KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>4kg</td>
<td>2.6kg</td>
<td>1.2kg</td>
<td>31sec</td>
<td>0.2kg</td>
<td>3.406KW</td>
</tr>
<tr>
<td>&quot;</td>
<td>5kg</td>
<td>3.0</td>
<td>1.8</td>
<td>37sec</td>
<td>0.2kg</td>
<td>3.422KW</td>
</tr>
</tbody>
</table>

Machine Efficiency

Overall machine efficiency

\[
\text{Efficiency} = \frac{\text{Average Power Output(Consumed)}}{\text{Average Power Input}} \times \%
\]

Output power (average) = \(\frac{3.101+2.325+3.375}{3}\)

\[= 2.933KW\]
Average power input = 3.433KW

\[ \text{Efficiency} = \frac{2.833}{3.433} \times 100 = 85.5\% \]

Average machine output/sec = \( \frac{9.6+8.6+10.2}{125+127+131} \)

= 0.07415kg/sec.

Average output/hour = 0.7415x3600 = 266.95kg

Average output/day at 9 hrs operation period =

= 267x9 = 2.4 Tonnes.

**CONCLUSION**

The imported threshing machine at present market value cost (₦110,000) while the designed and constructed universal grain thresher using locally available material is estimated at (₦51,074).

The standard evaluation was done by loading 20kg of sorghum heads and the procedure was repeated for millet and maize.

The result showed that the machine has an overall efficiency of 85.5%.

The machine can thresh from 2.5 to 5 tonnes daily depending on the type of crops.
REFERENCE


### Notes:

<table>
<thead>
<tr>
<th>S/No</th>
<th>Descriptions of Parts</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Hopper</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Hood</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Threshing Chamber</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>Sieve Housing</td>
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<tr>
<td>5</td>
<td>Blower</td>
<td>1</td>
</tr>
<tr>
<td>6</td>
<td>Motor</td>
<td>1</td>
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<td>7</td>
<td>Frame</td>
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</tr>
<tr>
<td>8</td>
<td>Sieve</td>
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<tr>
<td>9</td>
<td>Thresh Cyl Pulley</td>
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<tr>
<td>10</td>
<td>Flat Belt</td>
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<tr>
<td>11</td>
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<tr>
<td>12</td>
<td>Motor Pulley</td>
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<tr>
<td>13</td>
<td>Grain Outlet</td>
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<tr>
<td>14</td>
<td>Con. Rod</td>
<td>1</td>
</tr>
</tbody>
</table>

Scale: 1" = 12" on 3/4"" = 144" All dim. in mm

Fig. 1c