MODIFICATION AND PERFORMANCE EVALUATION OF IAR MULTI CROP THRESHER FOR SORGHUM THRESHING

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ABSTRACT

A comprehensive performance evaluation was conducted on an existing IAR multi crop thresher to identify its capabilities and limitations on sorghum threshing. The threshing unit was identified to have some deficiencies in the form of inadequate number of beaters and the absence of cutting knives on the cylinder cover. Test result showed that its performance parameters were low. The output capacity, threshing efficiency, cleaning efficiency, grain damages and scatter losses were found to be 76.32 kg/hr, 90.87\%, 88.95\%, 5.17\% and 4.30\% respectively. The thresher was modified by redesigning the threshing, cleaning and feeding units. To increase threshing effectiveness; the number of pegs around the threshing cylinder was increased, three rows of cutting knives were mounted on the cylinder cover and the size of the sieve shaker holes were redesigned by considering sorghum grain size characteristics. The redesigned thresher was fabricated using locally available materials. Its performance was evaluated under four feed rates; 3, 4, 5 and 6 kg/min, four cylinder speeds; 600, 700, 800, and 900 rpm and two levels of moisture content (8.93 and 10.38\%). Test result of the modified thresher indicate an average output capacity, threshing efficiency, cleaning efficiency, grain damages and scatter losses values of 219.92 kg/hr, 98.00\%, 95.58\%, 1.78\% and 4.36\% respectively.

Keywords: Threshing unit, Cylinder cover, Output capacity, sieve shaker and Sorghum

1. INTRODUCTION

Nigeria is the third largest sorghum producer in the world today with an annual production output of about 6.7 metric tons (FAOSTAT, 2014). Sorghum and millet are the major staple food in northern part of the country. From 1979 to 1994, the production trend of sorghum shows a consistent increase, about 4.62 and 2.5 metric tons of sorghum was produce in 1994 and 1979 respectively (FAO, 2010). With the recent change in government policy, which emphasize the need for national food security, the production output of sorghum is expected to keep growing. However, for a sustainable food security, the increase in food production has to be in pace with improvement in post-harvest operations such as threshing. Rooney (2010) has identified lack of consistent supply of good quality grain as the major constraint in producing excellent food products from sorghum and to produce quality grain, a good threshing method is necessary.

In Nigeria, sorghum is predominantly threshed using the traditional method, which is associated with high losses, grain contamination by extraneous materials, low productivity and high drudgery (Sale, 2015). To address grain threshing problems, different kinds of threshers were imported. However, the imported threshers are expensive, complex in design, require skillful technical personnel and lack spare parts for repair and maintenance. The Institute for Agricultural Research (IAR) developed a multi crop thresher for
sorghum and millet. The performance of the thresher was not satisfactory as it has low output and performance efficiency for sorghum threshing. The current study reports the modification of the IAR multi
crop thresher. This is anticipated to improve timeliness of threshing operations and other performance parameters.

2. MATERIALS AND METHODS

2.1. Design Considerations
A cylinder-concave clearance of 10 mm was used for the thresher. This is in accordance with the recommendations of Kamble et al. (2003) and Saeidirad et al. (2013). To keep the mechanical grain damage low, 9.21 m/s (800 rpm) was chosen as the threshing cylinder peripheral speed. This is within the range of values recommended by Joshi (1981) for sorghum threshing. Sieve hole size of 6 mm was used, this is based on IRRI (1978) recommendation. The shaking frequency and amplitude of oscillation were 360 rpm and 7 mm respectively. A threshing cylinder diameter of 220 mm and length of 230 mm was maintained from the previous machine. To increase threshing capacity, the number of pegs around the cylinder was increased from 4 to 5 and cutting knives were mounted on the cylinder cover.

2.2. Design Calculations

Determination of concave sieve radius: The radius of curvature of concave sieve \( r_c \), was determined by the following expression as used by Dangora et al. (2006):

\[
r_c = r_d + h_p + C_c \quad \ldots (1)
\]

Where: \( r_c \) = Radius of concave sieve (mm); \( r_d \) = Radius of cylinder drum (110 mm); \( h_p \) = Height of peg above the drum (78 mm); \( C_c \) = Cylinder concave clearance (10 mm); \( r_c = 198 \text{ mm} \).

Determination of the volume of cylinder and spike tooth: This was obtained by summing up the volume of the threshing cylinder which is hollow and peg teeth volume which is solid

\[
V_c = \pi (r_1^2 - r_2^2) L + (\pi r_3^2 h)n \quad \ldots (2)
\]

Where: \( V_c \) = Cylinder and peg tooth Volume (m\(^3\)); \( r_1 \) = Outer radius of the cylinder (110 mm); \( r_2 \) = Inner radius of the cylinder (105 mm); \( r_3 \) = Radius of the peg tooth (10 mm); \( L \) = Length of the cylinder (230 mm); \( h \) = Height of the peg tooth (78 mm); \( n \) = Number of peg tooth (40); \( V_c = 1.757 \times 10^{-3} \text{ m}^3 \).

Fan design: Actual discharge: According to Joshi (1981), the actual flow rate of fan can be estimate as

\[
Q_a = V_t x D x W \quad \ldots (3)
\]

Where: \( Q_a \) = Actual flow rate (m\(^3\)/s); \( V_t \) = Terminal velocity of air (m/s); \( D \) = Depth of air stream over the screen (400 mm); \( W \) = Width over which air is required (450 mm).

An air stream terminal velocity \( V_t \) is selected such that it is greater than the terminal velocity of the light contaminants and smaller than that of the principal grain materials. At 10 – 20% moisture content, The terminal velocity of sorghum seeds ranges from 3.73 – 5.13 m/s (Ajav and Ojediran, 2006). Therefore 3.70 m/s was chosen as the terminal velocity of air; \( Q_a = 0.670 \text{ m}^3 /\text{s} \).
Theoretical discharge: Theoretical discharge was defined as

\[ Q_t = \frac{Q_a}{\eta} \]  

(5)

Where: \( Q_t \) = Theoretical discharge (m\(^3\)/s) ; \( \eta \) = Volumetric efficiency of the fan which is usually assumed as 30% (Joshi, 1981) \( Q_t = 2.23 \text{ m}^3/\text{s} \).

Calculating outer diameter of impeller: Theoretical discharge was also defined by Joshi (1981) as:

\[ Q_t = \pi x \frac{d_1}{b_1} \times x v_1 = \pi x d_2 x b_2 x v_2 \]  

(6)

Where: \( d_1 \) = Inner diameter of fan impeller (mm); \( d_2 \) = Outer diameter of the fan impeller (mm); \( b_1 \) = Width of the blade at inner impeller diameter (mm); \( b_2 \) = Width of the blade at outer impeller diameter (mm); \( N_t \) = Rotational speed of the fan shaft (1500 rpm); \( v_1 \) = Tangential component of absolute velocity of the impeller at inner diameter; \( v_2 \) = Tangential component of absolute velocity of the impeller at outer diameter.

According to Joshi (1981), \( v_2 \) can be approximate as 20% of the peripheral velocity of the impeller tip for the design. Therefore,

\[ v_2 = 20\% \times f \left( \frac{\pi x d_2 x N_t}{60} \right) \]  

(7)

From equation (6) and (7), \( d_2 \) is calculated as 520 mm.

Dynamic head of the fan: From Bernoulli’s principle as reported by Schobeiiri (2010), the dynamic head is defined as

\[ H = \frac{v_t^2}{2g} \]  

(9)

Where: \( H \) = Dynamic head; \( V_t \) = Terminal velocity of air (3.70 m/s); \( g \) = Acceleration due to gravity (9.81 m/s\(^2\)); \( H = 0.709 \text{ m} \).

Power required to operate the fan: This was calculated by the following equation (Korpella, 2011):

\[ P_1 = \frac{\rho Q_a g H}{\eta} \]  

(8)

Where: \( P_1 \) = Power to be consumed by the fan (kW); \( \rho \) = Mass density of air (1.16 kg/m\(^3\) at 30°C); \( Q_a \) = Actual flowrate of the fan (m\(^3\)/s); \( g \) = Acceleration due to gravity (9.81 m/s\(^2\)); \( H \) = Dynamic head (m); \( \eta \) = Volumetric efficiency of the fan which is usually assumed as 30% (Joshi, 1981); \( P_1 = 0.0187 \text{ kW} \).

Power required to turn the unloaded threshing cylinder: This was calculated by the formula used by Olaoye et al. (2011):

\[ P_1 = \frac{2\pi N_t \times r \times Mc \times 60 \times 75 \left( g + \frac{V_{tp}^2}{r} \right)}{60} \]  

(10)

Where: \( N_t \) = Speed of the threshing cylinder (800 rpm); \( Mc \) = Mass of threshing cylinder (13.79 kg); \( r \) = Radius of cylinder (110 mm); \( V_{tp} \) = Peripheral velocity of the threshing mechanism (9.21 m/s); \( P_1 = 1.32 \text{ kW} \).

Power requirement due to air resistance: This was calculated by the following expression (Ndrika, 1997):

\[ P_2 = k_1 \times F_r \times V_t^2 \]  

(11)

Where: \( k_1 \) = 0.06 for sorghum (Ndrika, 1997); \( F_r \) = Feed rate (Assume 240 kg/hr), \( P_2 = 1.22 \text{ kW} \).

Power required to detach grain: It was calculated by the following formula (Olaoye et al., 2011):
\[ P_3 = \frac{3}{2} K_e \left( \frac{V_s^2 \times f_r}{\rho \omega^2 L_c} \right) \] …… (12)

Where: \( K_e = \) A constant (grain size characteristics), equals to 0.26 for sorghum (Ndrika, 1997); \( L_c = \) concave length (230 mm); \( V_s = \) Speed of the grain crop which is approximately equal to the peripheral velocity of threshing mechanism (9.21 m/s); \( f_r = \) feed rate (240 kg/h); \( \rho_w = \) Bulk density in wet basis (for sorghum, it is 30.92 at 12% moisture content (Ndrika, 1997)); \( P_3 = 0.065 \) kW.

Power required by shaking mechanism: The power was calculated using the equation given by Joshi (1981);

\[ P_4 = \left( \frac{W_s \times N \times 2x}{4500} \right) + \left( \frac{2u \times W_s \times N \times 2x}{4500} \right) \] …… (13)

Where: \( W_s = \) Weight of sieve component along with threshed material (0.4 kN); \( u = \) Coefficient of friction of the moving component (0.25); \( x = y = 7 \) mm; \( P_4 = 0.058 \) kW.

Total power requirement of the thresher:

\[ P_5 = P_1 + P_2 + P_3 + P_4 + P_f \]

\[ P_5 = 2.68 \) kW

Corrected power: Considering transmission and other losses a factor of safety of 1.2 was assumed. Therefore the design power was \( P_5 \times 1.2; P = 3.21 \) kW.

Estimation of pulley dimensions: Pulleys dimensions were determined by using expression given by Sanjay (2010);

\[ N_1 D_1 = N_2 D_2 \] …… (14)

Where: \( N_1 = \) Speed of drive pulley; \( D_1 = \) Diameter of drive pulley; \( N_2 = \) Speed of driven pulley; \( D_2 = \) Diameter of driven pulley

For cylinder and prime mover: \( N_1 = \) Speed of the prime mover (1400 rpm); \( D_1 = \) Diameter of the prime mover pulley (70 mm); \( N_2 = \) Speed of the threshing cylinder (800 rpm); \( D_2 = \) Diameter of the threshing cylinder pulley. Therefore, \( D_2 = 122.5 \) mm

For Cylinder and fan: \( N_2 = 800 \) rpm, \( D_2 = 122.5; N_3 = \) Speed of the fan (1500 rpm); \( D_3 = \) Diameter of the fan pulley (mm). Therefore, \( D_3 = 144 \) mm.

For Cylinder and shaker: \( N_3 = N_2 = \) Speed of cylinder shaft (800 rpm); \( D_3 = \) Diameter of pulley on cylinder (84 mm); \( N_4 = \) Speed of shaker (360 rpm); \( D_4 = \) Diameter of shaker pulley. Therefore, \( D_4 = 186 \) mm.

Torsional and bending moment: The following equations which was given by Khurmi and Gupta (2005) was used in determination of torsional and bending moments. The torsional moment \( (M_t) \) was calculated from equation (15) while values from equation 15 and 16 were used in plotting shear force and bending moment diagrams. The bending moment diagram was used in the determination of maximum bending moment \( (M_b) \).

\[ M_t = \frac{P_5 \times 60}{2\pi \times N} = \left( S_1 - S_2 \right) r \] …… (15)

\[ \frac{S_1}{S_2} = e^{60 \theta \cot \beta} \] …… (16)

Where: \( M_t = \) Torsional moment (38.60 kNmm); \( M_b = \) Maximum bending moment (from bending moment
Design of threshing cylinder shaft: The selected material used for the shaft in this machine is medium carbon steel (C1040). It has a tensile stress $S_t$ of 668.8 MN/m$^2$ and yield stress $S_y$ of 568.7 MN/m$^2$. Maximum allowable shearing stress ($S_a$) from Khurmi and Gupta (2005) is the lower value of 18% ultimate tensile stress ($S_t$) and 30% yield stress ($S_y$) and when there is keyways the value is reduced by 25%. $S_a = 90.3 \text{ MN/m}^2$

To get the shaft diameter (d), the following relationship by (Sanjay, 2010) was used.

$$d^3 = \frac{16}{\pi S_a} \sqrt{(k_b M_b)^2 + (K_t M_t)^2} \quad \ldots \quad (17)$$

Where: $K_b$ and $K_t$ are combine shock and fatigue factors apply to torsion and bending respectively; $K_b = 1.5 - 2$ (Suddenly applied load with minor shock); $K_t = 1.5 - 2$ (Suddenly applied load with minor shock).

$$d \approx 25 \text{ mm}$$

2.3. Major components of the Modified Thresher

**Hopper:** The hopper is designed to be fed in a vertically inclined at an angle of 28°. This was due to consideration of the earhead angle of repose which was measured and found to range from 26 to 29°. The hopper has the shape of a frustum of a pyramid truncated at the top, with top having rectangular forms and bottom having square form. It was constructed using mild steel metal sheet of gauge 18.

**Threshing Unit:** The threshing unit consists of the cylinder, perforated concave sieve, pegs on the cylinder and knives on the cylinder cover. The cylinder is made of mild steel sheet metal formed into 220 mm diameter. The concave is made of mild steel square iron rod and it is of 250 mm in diameter. The threshing cylinder was placed inside the concave. It has eight rows of beaters welded strongly on its surface and at right angle to the cylinder axis. For effective grain detachment, the spike tooth rods were space at 4.6 mm apart. The cylinder cover has three rows of knives. Clearance between the free ends of the beaters on the threshing cylinder and the concave is 10 mm respectively.

**The blower:** The fan is stationed besides the threshing cylinder and 300 mm below it. It is an aspirator type centrifugal fan. It suck up the chaffs and pieces of broken stalks out of the cleaning chamber. It was constructed with four blades of 520 mm diameter. Materials of construction was mild steel metal sheet of gauge 18. To regulate air flow, shut-off was provided by the side of the fan casing.

**The separation unit:** This unit consists of the sieves, shaking mechanism and grain collection tray. The shaking mechanism consists of crankshaft and connecting rod mechanism. It reciprocate at 6 cycles per second at an amplitude of 7 mm. Depending on the type of grain being threshed, the sieve can be interchange. To aid delivering of threshed grain, the grain collection tray was tilted at 30° angle which was also interchangeable. The sieve plate was constructed from mild steel sheet of gauge 18.

Other components that are critical to the operation of the thrasher are prime mover (5 hp diesel engine), two different size of shafts (20 mm and 15 mm), five different sizes of pulleys (70 mm, 125 mm, 144 mm, 272 mm and 48 mm) and three set of belts, (90, 27 and 32 inch).
2.4. Principle of operation of the improved thresher

The thresher work on the principle of gravity. The sorghum ear head are feed into the hopper from where it flows under gravity to the threshing chamber. Due to the action of beaters (peg tooth) on the ear head the grains are separated from the chaffs and the stalks are broken down into smaller pieces. From the threshing chamber, the grains, chaffs and pieces of stalks pass through the concave sieve opening to the cleaning chamber. Chaffs and pieces of stalk are sucked out of the cleaning chamber by the fan while the clean grain pass through the concave sieve opening to the collection compartment. Plate 1 shows sorghum threshing operation.

Plate 1: Sorghum threshing operation

2.5. Performance Evaluation

2.5.1 Materials and Instrumentation: The experiment was conducted in accordance with the method detailed in FAO (1994) and the following materials were used; Kaura variety of sorghum, Stop watch (Precista), Tachometer (HW2), Electronic weighing balance (OPH-T300) and drying oven (Agrisearch equipment UN30).

2.5.1 Performance Parameters: The following equations, which were introduced by FAO (1994) were used in calculating the performance parameters:

**Threshing efficiency, \( T_e \) (%):**

\[
T_e = 100 - \left( \frac{Q_u}{Q_t} \right) \times 100 \quad \ldots \quad (18)
\]

Where: \( Q_u \) = quantity of unthreshed grains in sample (kg); \( Q_t \) = Total quantity of grain in sample (kg).

**Grain damage, \( M_D \) (%)**: The Mechanical grain damage was determined by the relationship:

\[
M_D = \left( \frac{Q_b}{Q_t} \right) \times 100 \quad \ldots \quad (19)
\]

Where: \( Q_b \) = quantity of broken grains in sample (kg); \( Q_t \) = Total quantity of grain in sample (kg).

**Cleaning efficiency, \( C_e \) (%):** The Cleaning efficiency was determined by the relationship:

\[
C_e = \frac{W_t - W_c}{W_t} \quad \ldots \quad (20)
\]

Where: \( W_t \) = Weight of total mixture of grain and chaff received at the grain outlet (kg); \( W_c \) = Weight of chaff at the main outlet of the thresher (kg).

**Scatter loss, \( S_L \) (%)**: The Scatter loss was determined by the relationship:

\[
S_L = \left( \frac{Q_l}{Q_t} \right) \times 100 \quad \ldots \quad (21)
\]
Grain throughput capacity, $T_c$ (kg/hr)

$$T_c = \frac{Q_s}{T}$$

Where: $Q_s = \text{Quantity of threshed grain collected after a threshing operation (kg)}$; $T = \text{Time taken for a complete threshing operation (hr)}$.

3. RESULTS AND DISCUSSION

3.1. Threshing efficiency

Figure 1, depicts that sorghum threshing efficiency increased from 95.47% to 98.38% as the cylinder speed increased from 600 to 900 rpm at a moisture content of 8.93%. It increased from 94.29% to 98.48% as the cylinder speed increased from 600 to 900 rpm at a moisture content of 10.38%. The results shows that threshing efficiency increased linearly as the drum speed increases. The coefficient of determination is 0.9244 at 8.93% moisture content and it is 0.8080 at 10.38%. Both values are close to one, these showed good agreement between the two variables considered. This is consistent with the findings of Simonyan and Imokheme (2006) who reported that the threshing efficiency increased as cylinder speed increased.

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3.2. Cleaning efficiency

The results in Figure 2, indicated that sorghum cleaning efficiency increased with an increase in cylinder speed. A high cleaning efficiency of 97.10% was obtained at a cylinder speed of 900 rpm and moisture content of 8.93% and a lower value of 91.93% was obtained at a speed of 600 rpm and moisture content of 10.38%. This showed that cleaning efficiency increased as the cylinder speed increased but decreased as the moisture content increased. From the results, the coefficient of determinations were 0.8805 and 0.7046 for the two moisture content considered. This showed that the two variables were highly correlated.

Scatter losses

A high sorghum scatter loss of 3.49% was obtained at the speed of 900 rpm and moisture content of 8.93% level while a low scatter loss of 1.65% was at the
lowest speed of 600 rpm (Figure 3). This showed that scatter loss increased in proportion to drum speed. This could be due to overloading of the sieve at higher speed which is always associated with increased in quantity of the materials being threshed. Moisture content also affected the scatter loss in that at higher moisture content the scatter seed was lower. Correlation coefficient values of 0.9863 and 0.9781 were obtained at the two moisture content of 8.93% and 10.38% respectively. This implied that speed increase has positively affected the scatter losses. A similar trend was reported by Sale (2015).

3.3. Mechanical Grain Damages

From figure 4, the percentage weight of damaged sorghum seeds increased from a value of 2.06% at 8.93% moisture content to a value of 3.80% at 10.38% moisture content as the cylinder speed increased from 600 to 900 rpm. This might be attribute to the fact that with the increase of cylinder speed the collision energy between grains and spike tooth was increased and the seeds were loaded with greater forces. For cumin threshing (Saedirad et al., 2011) and chickpea...
threshing (Khazaei, 2003), similar trend of results were obtained. The damage was also found to be higher at the highest moisture content. The reason for this can be that, the increase in moisture content makes the grain softer enabling it to undergo deformation easily.

3.4. Throughput capacity

Figure 5 shows that at 8.93% moisture content and a cylinder speed range of 600 to 900 rpm, the throughput capacity increased from 91.01 kg/hr, to 135.19 kg/hr while at 10.38% moisture content, the throughput capacity increased from 79.26 kg/hr to 122.92 kg/hr. These indicated that, throughput capacity increased with speed and decreased with moisture content. The reason for that, could be as a result of greater speed of cylinder which made more impact on the ear head thereby increasing the rate of grain detachment from the ear head. The coefficient of determinations were found to be 0.9821 and 0.9304 for the two level of moisture content considered. This showed a good agreement between the dependent and independent variables.
4. CONCLUSION

The performance of an existing IAR multi crop thresher on sorghum was investigated. Based on identified shortcomings, it was modified through redesigning some of the components and reconstructed. The redesigned thresher was tested to determine its performance parameters. From the results of the test carried out, it was concluded that, threshing efficiency, cleaning efficiency, scatter losses and throughput capacity increased with increase in cylinder speed but decreased as the moisture content increased. It was also, observed that the mechanical grain damage increases with increase in speed and moisture content respectively.

REFERENCES


