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Design and Finite Element Analysis of a Thresher for Palm Oil (*Elaies guineensis)* Extraction Plant

Olukunle Elijah ITABIYI[1], Kehinde Monsuru ADELEKE[2], Samson Ayorinde AKANGBE[3], Abiodun Ayodeji OJETOYE[2], Fatai Ayofe BUHARI[4]

[1] *Department of Mechanical Engineering, Ladoke Akintola University of Technology, Ogbomoso, Nigeria*
oeitabiyi@lautech.edu.ng

[2] *Department of Mechanical Engineering, Adeleke University, Ede, Nigeria*
adeleke.kehinde@adelekeuniversity.edu.ng/abiodun.ojetoye@adelekeuniversity.edu.ng

[3] *Department of Electrical and Electronics Engineering, Adeleke University, Ede, Nigeria*
samson.akangbe@adelekeuniversity.edu.ng

[4] *Department of Agricultural Engineering, Adeleke University, Ede, Nigeria*
fatai.fatai@adelekeuniversity.edu.ng

Corresponding Author: *adeleke.kehinde@adelekeuniversity.edu.ng, +2348064647667*
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***Abstract**: This study presents the design and Finite Element Analysis (FEA) of a thresher used in palm oil (Elaeis guineensis) extraction plants. The FEA was performed to ensure safe and cost effective of the thresher before fabrication. The analytical design of the threshing shaft and drum of the thresher was validated using SolidWorks (2021) CAD software for static simulation, employing plain carbon steel as the material. For the threshing shaft, forces of* $2295.58\ N$ *and* $4483.14\ N$ *were applied at strategic points, resulting in a maximum bending stress of* $6.97 \times 10^7\ N/m^2$*, significantly below the yield strength of* $2.206 \times 10^8\ N/m^2$*. The shaft's diameter of 50 mm was confirmed as adequate with a factor of safety (FOS) ranging from 3.17 to 142.42, validating the shaft design's safety for fabrication. Similarly, the drum unit, supported by a spider arm and cylindrical bars, was subjected to an equivalent twisting moment of 861.25 Nm and a batch weight of 1226.25 N. The maximum von Mises stress of* $1.709 \times 10^7\ N/m^2$ *was well within safe limits, indicating robustness under operational loads. The maximum resultant displacement and equivalent strain were* $5.820 \times 10^{-4}\ mm$ *and* 3.507×10^{-5} *respectively which can be said to be minimal, reinforcing the drum's structural integrity. A minimum FOS of 20.45 further highlighted the drum's durability and resistance to fatigue. These results confirm the reliability and safety of the designed thresher components, ensuring efficient and sustainable palm oil extraction.*

Keywords: *Thresher, Finite Element Analysis, Palm Oil, Bunch Stripper, Static Simulation*

1. INTRODUCTION

Palm oil is a highly prevalent and extensively used vegetable oil on a global scale, with a substantial impact on the economics of numerous tropical nations, especially in Southeast Asia and Africa. Nigeria was the highest exporter until 1934 when the country was surpassed by Malaysia. Nigeria is currently the third highest producer of palm fruits only after Malaysia and Indonesia [1, 2]. Apart from palm oil and palm kernel which are direct products from the processing of palm fruit, there are other by-products such as the palm kernel oil and palm kernel cake [3]. The palm kernel oil is edible oil obtained from the pulverized palm kernel after application of pressure [4-6]. Traditional or native processing of palm fruits is very tedious, involves a lot of drudgery and is relatively unhygienic. Palm oil is in high demand due to its adaptability and extensive use in various industries such as food, cosmetics, medicines, and biofuels [7, 8]. The palm oil manufacturing process encompasses various crucial stages, starting from the growth of oil palm trees to the extraction and refining of crude palm oil [9-11]. The threshing process is crucial among these processes as it effectively separates the oil-bearing fruits from the bunches, hence significantly impacting the overall efficiency and yield of the oil extraction process.

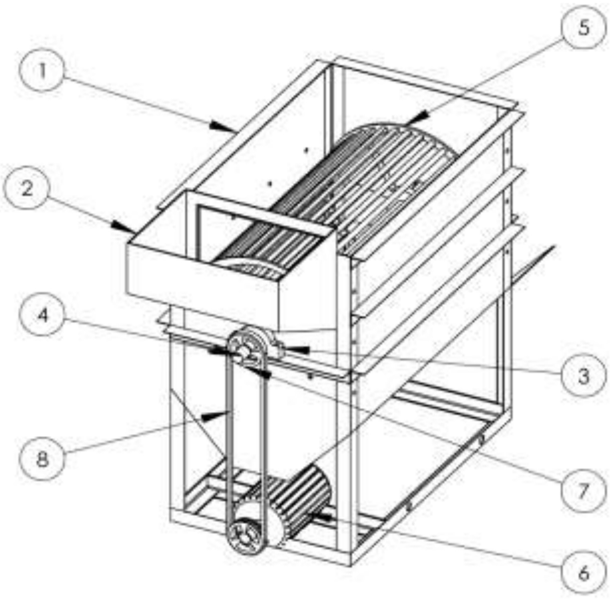
Threshing is an essential and pivotal stage in the extraction process of palm oil. Palm fruit detachment from the Fresh Fruit Bunches (FFBs) is a crucial stage in the process of extracting oil from the fruits [8]. The failure of spider arms within thresher drums in palm oil processing plants is a significant problem that can result in maintenance difficulties and higher expenses. Research has indicated that the spider arms within thresher drums are susceptible to developing fractures and experiencing breakdowns as a result of the impact of loads, which negatively impacts the overall efficiency of the operation [12, 13]. The effectiveness of the threshing process has a direct impact on both the quality and quantity of the produced oil [14]. Insufficient threshing can result in elevated oil losses and heightened levels of Free Fatty Acids (FFAs) in the oil [15], so adversely affecting its quality and shelf life [16].

Manufacturing industries who make use of palm oil as raw material and cannot afford the scale large scale production cost find it difficult to make use of the available local palm oil due to the level of impurities present. They however further carried out further sterilization on it. The judicious use of this design will however not only improve the level hygiene but also its availability for small scale production. Consequently, the development of an effective thresher is crucial for improving the efficiency and financial success of palm oil processing facilities.

2. MATERIALS AND METHODS

2.1 Conceptual Design

The thresher is a tank containing with V-shaped bottom and a rotating circular disc driven by electrical motor. It function is to remove palm fruit from its husk. This is done by introducing harvested palm husk into it while rotating disc move it against the wall surface and threshes the fruit from the husk at a low rotating speed. It is designed to allow the fruits chute out of the compartment; the fruits are obtained and separated from the husk. Figure 1 shows the isometric view of the design with parts list under consideration.



ITEM NO.	PART NUMBER	QTY.
1	Supporting Frame	1
2	Housing Plate	1
3	Bearing with Housing	2
4	Threshing Shaft	1
5	Threshing Drum with Spider Arm	1
6	Electric Motor	1
7	Pulley 1	2
8	Power Transmitting Belt	1

Figure 1: Isometric view of the conceptual design with part list

2.2 Design Analysis

The palm fruit thresher's conceptual design aims to provide efficient and dependable separation of palm fruits from their husks. The design incorporates mechanical robustness, material durability, operational efficiency, and safety measures to ensure that the thresher can handle the necessary capacity and work continuously with low maintenance requirements [17]. The design assumptions guarantee that the thresher is tailored to efficiently handle the "*tenera*" kind of palm fruits, which is crucial for attaining the necessary performance and oil quality. The parameters used for this analysis are presented in this section.

2.2.1 Design considerations

The factors to be considered in the design of the palm fruit thresher are;

i. Mechanical Design: The mechanical elements, such as the electric motor, drum, and screen, need to be built in a manner that guarantees effective threshing while upholding structural robustness.
ii. Material Selection: Selecting suitable materials capable of enduring the rigorous conditions of the threshing process, including high resistance to both wear and corrosion.
iii. Operational Parameters: Establishing the most effective operational parameters, such as the speed of the electric motor, the diameter of the drum, and the size of the screen, in order to attain optimal performance.
iv. Ensuring safety and optimizing ergonomics: Ensuring the thresher is both safe and ergonomically constructed to facilitate ease of usage by personnel.

2.2.2 Assumptions in the design

The following assumptions were made in the design analysis;

i. Palm fruits are all of the same variety (*"tenera")*. The variety is produced by crossing the *"dura"* and *"pisifera"* varieties and possesses a thin shell and higher oil content. The *"tenera"* variety has a relatively thin shell, thereby, making cracking easy [18].

ii. Feedstock (palm fruit) are processed fresh after harvest. Traditionally, FFBs are left for about 2-3 days before processing to promote fermentation. Fermentation enhances the taste of the extracted oil. However, delayed storage before processing promotes the formation of free fatty acids, which are undesirable in processing. Thus, the FFBs are processed immediately after harvest [19, 20].

iii. Plant will work for 8 hr/day, 5 days/week. It is assumed that the plant works continuously for 8 hours a day and five days in a week.

2.2.3 Determining the plant capacity

The plant is to be designed to handle 0.5 Ton of fresh fruit bunch (FFB) per hour

Poku [21] confirmed that for an average FFB, Bunch weight, 23- 27 kg

Average weight of the FFB = $\frac{23+27}{2} = 25\ kg$

Fruit mass in FFB = $60\%\ \times average\ mass\ of\ FFB$

Fruit mass in FFB $= 0.6 \times 25 = 15\ kg$/FFB

Required Production Capacity = 0.5 Ton FFB/hr

Number of FFB required for optimum production = $\frac{1 \times 500}{25} = 20\ FFBs$

But, Number of labour hours = $8\ hours$

Therefore, plant capacity = $20 \times 8 = 200\ FFB/day$

Fruit/bunch, 60-65% = 0.6 × 25 = 15 kg

Oil per bunch, 21-23% = $\frac{22}{100} \times 25 = 5.5\ kg$

Kernel per bunch, 5-7% = $\frac{6}{100} \times 25 = 1.5\ kg$

Mesocarp/bunch, 44-46% = $\frac{45}{100} \times 25 = 11.25\ kg$

Kernel/ 15 kg of fruit, 20-22% = $\frac{2}{100} \times 15 = 3\ kg$

Shell/15 kg of fruit, 10-11% = $\frac{10}{100} \times 15 = 1.5\ kg$

Mass of FFB in thresher/batch = 25× 5 = 125 kg

Mass of fruits in press/batch = 125 kg

2.2.4 Design of Thresher (Bunch Stripper)

The thresher is designed to strip the palm fruits from the sterilized bunches. A cylindrical drum is keyed along the cross section of the shaft to produce a rotating effect on bunches as they get lifted, dropped and conveyed away from the hopper as presented in Figure 1. The thresher shaft is designed to transmit $15\ kW$ at a speed of $2500\ r.p.m.$ The shaft is supported at both ends with bearings. The supported length is $1.30\ m$ (housing the threshing drum). A cast iron pulley with a coefficient of friction μ, equal to 0.25, is mounted on the shaft vertically above the electric motor such that the centre point on the pulley is $0.06\ m$ from the left end of the shaft.

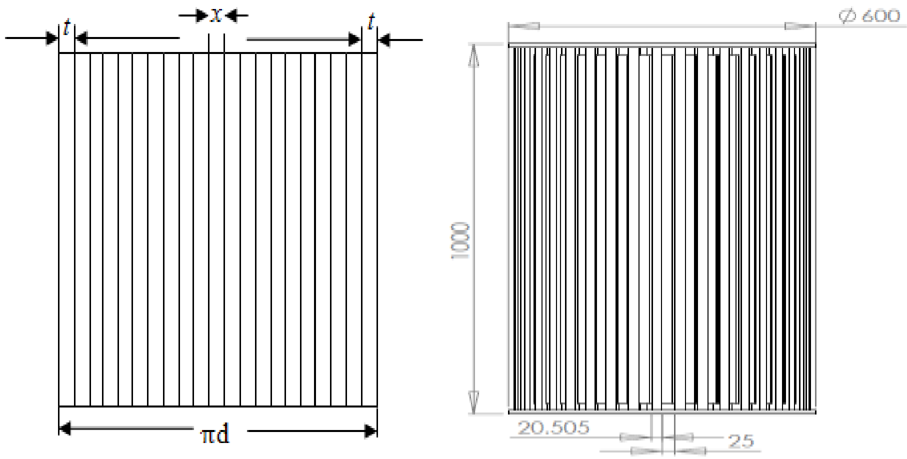


Figure 2: Cross section of the cylindrical thresher drum

i. Determination of the volume of the cylindrical thresher drum

The inner volume, V_i, of the drum is derived and presented in Equation (1)

$$V_i = \pi(r - y)^2 h\ (m^3) \quad (1)$$

If the total length of the cylinder is stretched into sheet metal of height 'h' and length "πd"; then removing alternate rectangular cross sections from the sheet metal to make allowance for fruit dropping such that:

If the width of the fruit = $a\ (mm)$

The volume of a single rectangular section removed, V_i, is given in Equation (2)

$$V_i = a \times h \times y \, (m^3) \tag{2}$$

Where h = the height of the cylinder and
y = thickness of the sheet metal

Number of rectangular sections of the drum, n, removed from along the sheet metal can be calculated from equation (3) as shown in Figure 2.

$$n = \pi \frac{d-2t}{2a} \tag{3}$$

Where t = edge clearance for joining sheet metal

From the design in Figure 2;

$$h = 1.0\,m, \quad r = 0.3\,m, \quad y = 0.01\,m, \quad a = 0.025\,m, \quad t = 0.01\,m, \quad \pi = 3.142 \text{ and } \theta = 90°$$

Hence, the number of rectangular sections removed, $n = 36.44$

Also, six sectorial shaft supports on the two bearings each at interval angle, θ, to the center and thickness, t, from both ends. Then, the volume of the sectorial support is $\frac{\theta}{360} 6\pi r^2 t$

Therefore, volume of the cylindrical drum, V_{cd}, is determined from Equation (4).

$$V_{cd} = \pi r^2 h - \left(\frac{\theta}{360} 6\pi (r-y)^2 t + \pi (r-y)^2 h + nahy\right) \tag{4}$$

Then, $V_{cd} = 0.0139\, m^3$

ii. Determination of the weight of the cylindrical threshing drum

The mass of the cylindrical threshing, m, is determined from Equation (5)

$$m = \rho V_{cd} \tag{5}$$

Where, ρ *is density in* kg/m^3

If the cylindrical drum is made of mild steel of density 7850 kg/m^3; then the mass of the drum can be determined as:

$m = 109.12\, kg$

Therefore, weight of the thresher drum, W_{cd}, becomes 1069.33 N (taking g = 9.81 m/s^2).

iii. Belt design for the thresher shaft

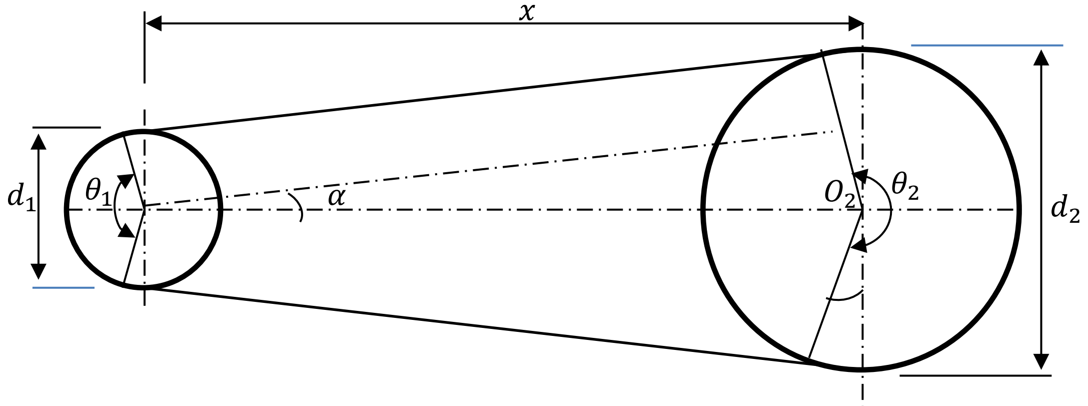


Figure 3: Pulley – belt arrangement of the power transmission

Power is transmitted by means of flat belts and pulleys from a 15 kW rated electric motor and a desired speed of 2500 r.p.m. of the shaft [22].

From Figure 3, the speed ratio of the pulley is determined using Equation (6);

$$\frac{N_1}{N_2} = \frac{d_2}{d_1} \tag{6}$$

Where N_1 = speed of the Motor
N_2 = speed of the thresher pulley
d_2 = diameter of the thresher pulley
d_1 = diameter of the motor pulley

For V-belts, we know that the minimum pitch diameter for *B-type* belts (2 – 15 kW) is 125mm [17]; and our desired output speed for the shaft, N_1, is 2500 r.p.m.; using a medium speed electric motor of 1425 $r.p.m.$;

d_1= 219 mm

The minimum and maximum possible centre distance between the pulleys, according to Mohammed (2012) is given by Equations (7) and (8) respectively;

$$x_{max} = 3(d_1 + d_2) \tag{7}$$

$$x_{min} = 0.55(d_1 + d_2) + d_2 \tag{8}$$

$$x_{max} = 1032\ mm$$

$$x_{min} = 408.2\ mm$$

The design centre distance is the average of the minimum and maximum possible centre distances

$$x = \frac{x_{max}+x_{min}}{2}$$

$$x = 720.1\ mm$$

Also, the lap, α, of the small pulley on the larger pulley is calculated from Equation (9);

$$\sin\alpha = \frac{d_1-d_2}{2x} \tag{9}$$

$$\alpha = \sin^{-1}\left(\frac{d_1-d_2}{2x}\right)$$

$$\alpha = 3.74°$$

$$\alpha = 0.0653\ rad$$

Angle of contact on larger pulley is given by equation (10)

$$\theta = 180 + 2\alpha \tag{10}$$

Angle of contact = 3.27 rad

And that of the smaller pulley is given by

$$\theta = 180 - 2\alpha = 3.01\ rad$$

It is known that the power input, in Watts, from the electric motor is determined from Equation (11)

$$P = \frac{2\pi N_1 T}{60} \tag{11}$$

Where, T, is the driving torque in Nm

$$T = \frac{60P}{2\pi N_1}$$

By substituting the power requirement of *B-type* (15 kW) [17]; and the chosen desired output speed for the shaft, N_1, is 2500 r.p.m, the driving torque becomes

$$T = 100.52\ N$$

The power transmitted by the belt and Torque transmitted as related to belt tensions [17] has the relations in Equations (12) and (13) respectively;

$$P = (T_1 - T_2)v \tag{12}$$

T_1 = tension in the tight side of the belt (N)
T_2 = tension in the slack side of the belt (N)
v = velocity of the belt (m/s)

$$T = (T_1 - T_2)R \tag{13}$$

Where T = torque transferred (Nm) and R = radius of the shaft pulley (m).
Combining equations (12) and (13), resulting to Equation (14)

$$(T_1 - T_2) = \frac{P}{v} = \frac{T}{R} \tag{14}$$

Given that $P = 15\ kW, R = 62.5\ mm = 0.0625\ m\ and\ T = 100.52\ Nm$; thus the velocity of the belt can be calculated;

$$v = \frac{RP}{T} = 9.33\ ms^{-1}$$

Thus, since the velocity is less than 10 ms^{-1} , the effect of centrifugal tension is negligible [17].

From Equation (14), the ratio $\frac{P}{v}$ becomes

$$(T_1 - T_2) = 1600\ N$$

The belt tension ratio is given by equation (15) according to Khurmi and Gupta [17].

$$\frac{T_1}{T_2} = e^{\mu\theta} \tag{15}$$

Taking the coefficient of friction between the pulley, μ, as 0.25 and smaller angle of contact, θ as calculated using Equation (10).

Then, $\frac{T_1}{T_2} = e^{0.25\times 3.01} = 2.12$
Therefore $T_1 = 2.12T_2$
$2.12T_2 - T_2 = 1.12T_2 = 1600$
Therefore T_2 = 1428.57 N,
Hence, $T_1 = 2.12T_2$ = 3028.57 N

iv. Determination of the torque and maximum bending moment of the shaft

In order to determine the maximum bending moment acting on the shaft, the forces acting on the system need to be analysed. For purposes of analysis the weight of the shaft, coupling and support bars screwed to the metal frame is assumed negligible.

The material made of the pulley is cast iron and its density is taken to be 7200 kg/m^3, the weight of the pulley, W_p, is thus determined using Equation (16).

$$\text{Volume of pulley} = \pi\frac{d^2}{4} \times t \tag{16}$$

Where d = diameter of pulley = 125 mm = 0.125 m

Since the diameter of the pulley is less than 200 mm, the diameter is made with solid disc rather than arms [17] assuming a thickness of 30 mm.

Since, density = $\frac{mass}{volume}$
Then mass of pulley = density× volume
Mass of pulley = 2.65 kg

By converting the mass to weight, weight of the pulley, W_p, becomes 26.0 N

The total vertical load acting on the shaft through the pulley at point C (see Figure 4) is given by Equation (17).

$$W_C = W_p + T_1 + T_2 \tag{17}$$
$$W_C = 4483.14\ N$$

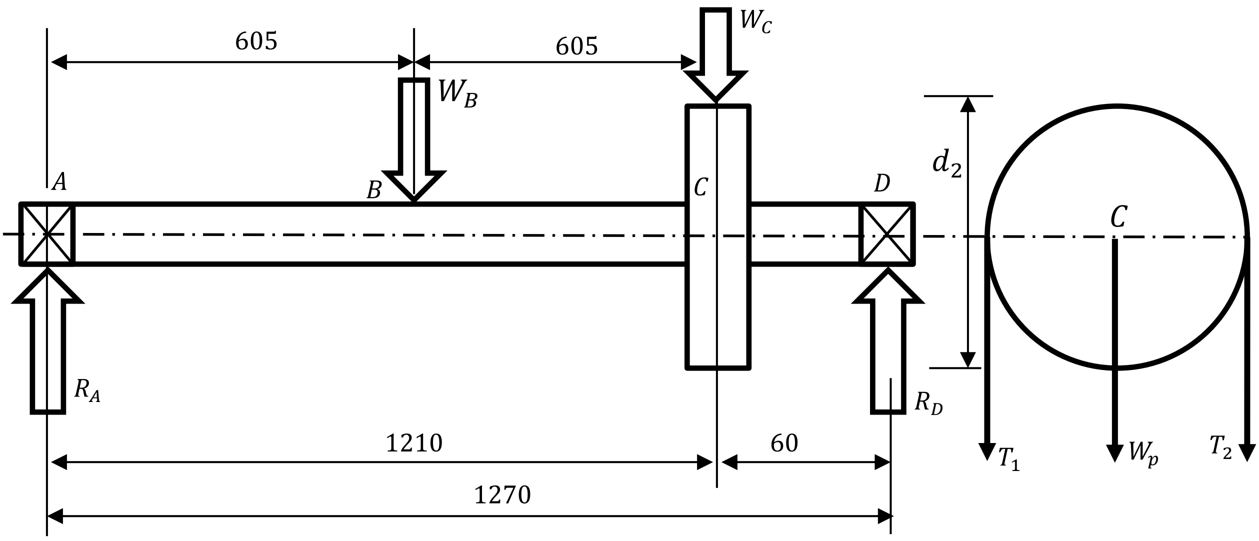


Figure 4: Shaft space diagram of threshing shaft

Having known all the forces acting on the system as shown in Figure 4, then the bending moment can be determined as follows:

R_A = reaction at bearing A
R_D = reaction at bearing D
W_B = combined weight of cylinder drum with FFB per batch acting at point B
W_C = total weight acting downwards at point C

Total upward forces is equal to total downward forces as stated in equation (18).

$$R_A + R_D = W_B + W_C \tag{18}$$

From the space diagram in Figure 4, $L_{AD} = 1.270\ m,\ L_{CD} = 0.060\ m\ and\ \ L_{BD} = 0.665\ m$. Also, $W_C = 4483.14N\ and\ \ W_B$.

$W_B = W_{cd} + W_{FFB/batch} = 1069.33 + 125(9.81)$

$W_B = 2295.58\ N$

$R_A + R_D = 6778.72\ N$

Taking moment about point D as stated in equation (19)

$$(R_A \times L_{AD}) = (W_C \times L_{CD}) + (W_B \times L_{BD}) \tag{19}$$

Hence,

$R_A = 1413.82\ N$

From equation (18),

$R_D = 5364.90\ N$

Bending moment, M, at point $A\ and\ D = 0$

M at B = $R_A \times L_{AB} = 1413.82 \times 0.605 = 855.36\ Nm$

M at point C = $R_D \times L_{CD} = 5364.90 \times 0.06 = 321.89\ Nm$

Since there are no horizontal loads, therefore individual vertical bending moment gives the resultant bending moment at each point and its maximum value is at point B as shown in Figure 5.

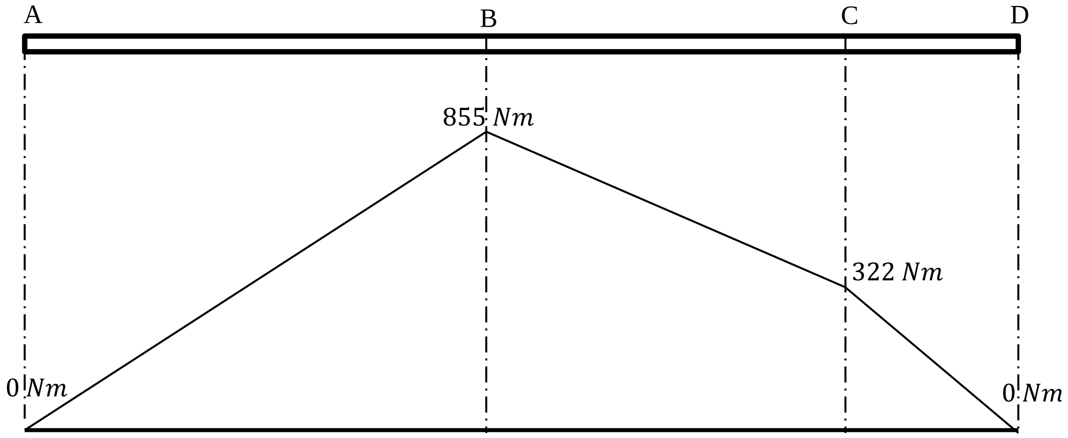


Figure 5: Bending moment diagram of threshing shaft

The equivalent twisting moment, T_e, and the diameter, d, of the shaft are determined using Equations (20) and (21) respectively according to Khurmi and Gupta [17].

$$T_e = \sqrt{{M_{max}}^2 + T^2} \tag{20}$$

Where, M_{max} is the maximum bending moment

$T_e = 861.25\ Nm$

To obtain diameter, d, of the shaft, the equivalent bending moment is given by Equation (21)

$$T_e = \frac{\pi}{16} \times \tau \times d^3 \tag{21}$$

Where,

τ is the allowable shear stress of the shaft material

But for shaft for allowance for keyways, the maximum permissible shear stress is 42 MPa [17].

Hence,

$$d = \sqrt[3]{\left(\frac{16T_e}{\pi\tau}\right)}$$

$$d = \sqrt[3]{\left(\frac{16 \times 861.25}{3.143 \times 42 \times 10^6}\right)}$$

$d = 0.0471\ m$

$d = 47.1\ mm$

For higher factor of safety, the shaft diameter is taken as $50\ mm$.

3. RESULTS AND DISCUSSION

3.1 Finite Element Analysis Shaft

The results of the analytical design of threshing shaft were validated with Finite Element Analysis (FEA) through SolidWorks 2021 CAD software static simulation [23]. The chosen material for the design is plain carbon steel which was used for the (FEA). The parameters observed include the upper bound axial and bending stress, bending moment as well as factor of safety. Figure 6 shows result of threshing shaft simulation by applying a force of $2295.58\ N$ and $4483.14\ N$ at points B and C respectively with reference to Figure 4. A maximum bending stress of $6.97 \times 10^7\ N/m^2$ was obtained at point B with the material yield strength of $2.206 \times 10^8\ N/m^2$.

The shaft diameter of $50\ mm$ obtained from section 2.2.4 (d) was used to design the threshing shaft that was subjected for FEA. The shaft designed showed adequate diameter through the factor of safety (FOS) plot as shown in Figure 7. A minimum FOS of 3.17 was observed at point B and a maximum value of 142.42 at point A. This nearly follows the research conducted by Mokhtar *et al.*, [12]. The load This result confirms that the design of shaft with a diameter of $50\ mm$ is safe for fabrication.

The bending moment analysis was also validated from FEA simulation by applying the calculated amount of load of $2295.58\ N$ and $4483.14\ N$ at points B and C respectively. A maximum bending moment of $855.36\ Nm$ was obtained at point B as shown in Figure 8. This confirms that the design analysis of the bending moment can be said to be in correct order. It also further confirms the reliability of design analysis result obtained.

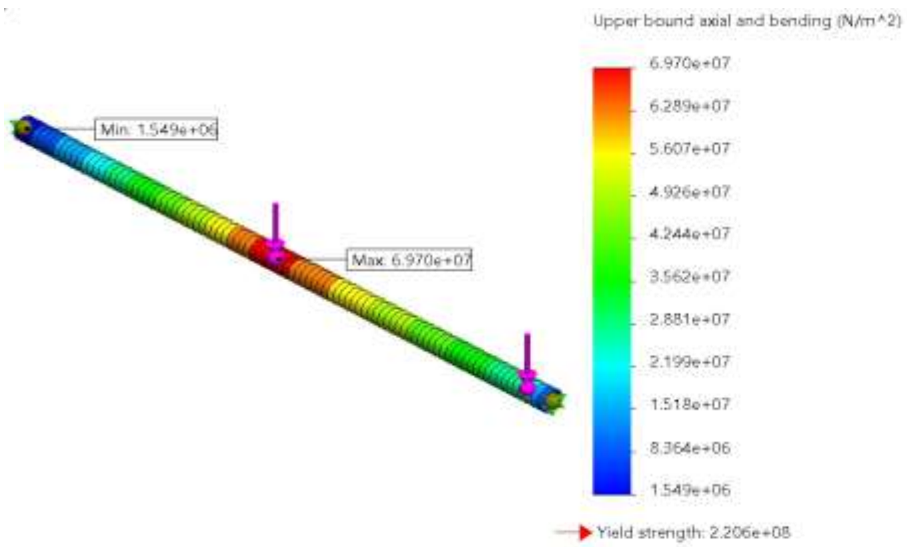


Figure 6: Upper bound axial and bending stress FEA result of threshing shaft

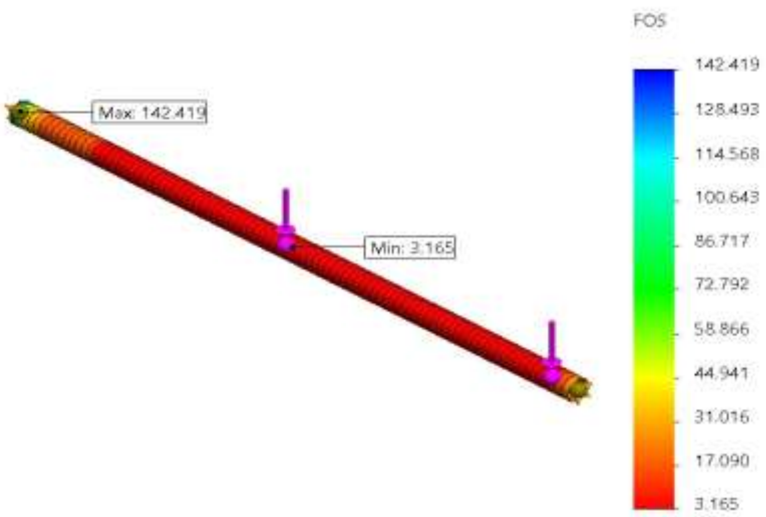


Figure 7: Factor of safety FEA result of threshing shaft

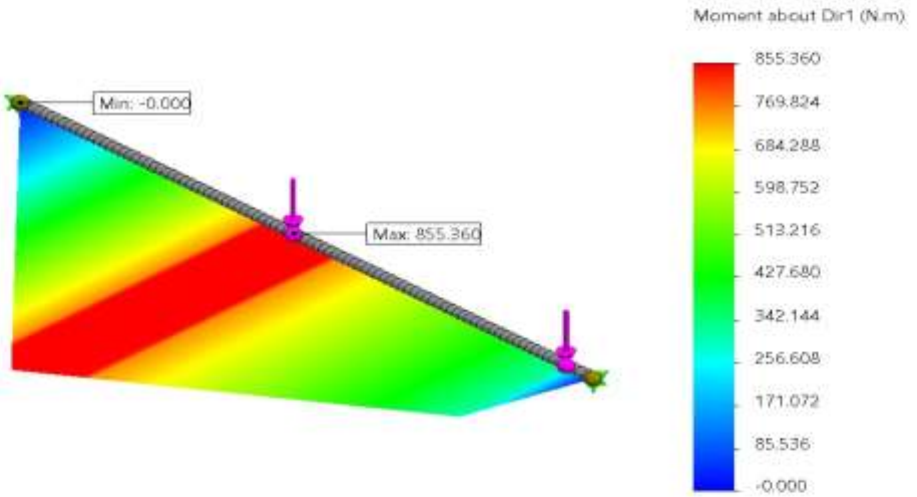


Figure 8: Bending moment FEA result of threshing shaft

3.2 Finite Element Analysis of the Drum

The drum unit of the machine was simulated with static FEA using plain carbon steel as material. The drum is held by spider arm that is joined with cylindrical bars. A calculated equivalent twisting moment, T_e , of $861.25\ Nm$ (obtained from section 2.2.4 (d)) was applied at both ends that rests directly on the threshing shaft and an average weight of FFB, $W_{FFB/batch}$, of $1226.25\ N$ was applied at the cylindrical surface of the drum. The von Mises stress, resultant displacement (URES), equivalent strain (ESTN) as well as FOS results were obtained from the simulation.

The von Mises stress simulation result is shown in Figure 9. It can be seen that the maximum value of $1.709 \times 10^7\ N/m^2$ was obtained at the contact surface of the spider arm and threshing shaft. It can be said that the drum design has not reached the yield strength of the material ($2.208 \times 10^8\ N/m^2$) and hence safe under the action of the designed loads.

A maximum value of $5.82 \times 10^{-4}\ mm$ was observed from resultant displacement of the drum unit under static loading around the centre of the spider arm as shown in Figure 10. This is below the value which was obtained in the spider arm analysis by Mokhtar et al. [12]. The maximum value of the resultant displacement of the drum obtained is negligible and indicates that the material thickness of the drum is good enough to resist the action of forces around it. This negligible of the deformation of the threshing drum also results in very small value of equivalent strain around the centre of the spider arm as shown in Figure 11 with as maximum value of 3.507×10^{-5}.

To validate the design of the threshing drum, a minimum factor of safety of 20.45 was obtained around the cylindrical surface of the drum as presented in Figure 12. This can be said to be due to action of average weight of FFB applied the design simulation would be moving around the inner part of the drum unit. This indicate that the design thickness would be large enough to reduce the effect of the fatigue in the equipment over a long period of time.

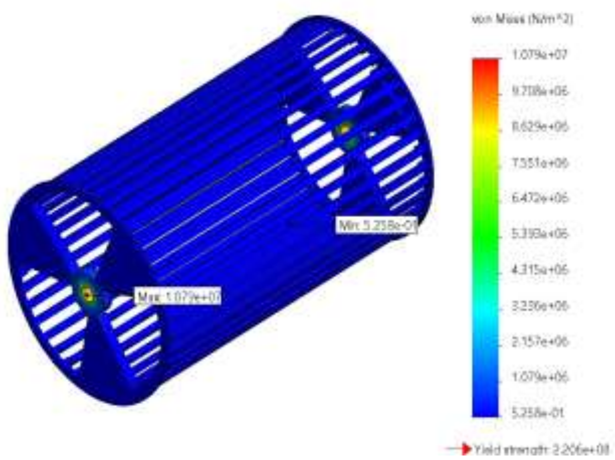


Figure 9: Von mises Stress FEA of threshing drum

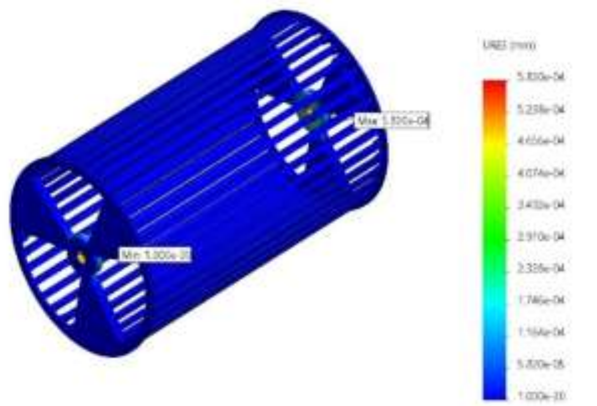


Figure 10: Resultant displacement FEA of threshing drum

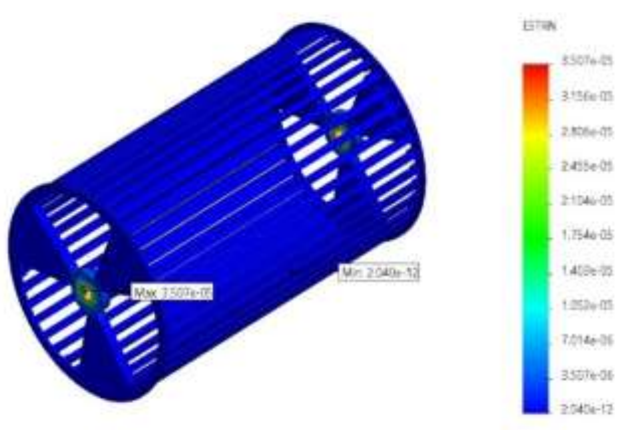


Figure 11: Equivalent strain FEA of threshing drum

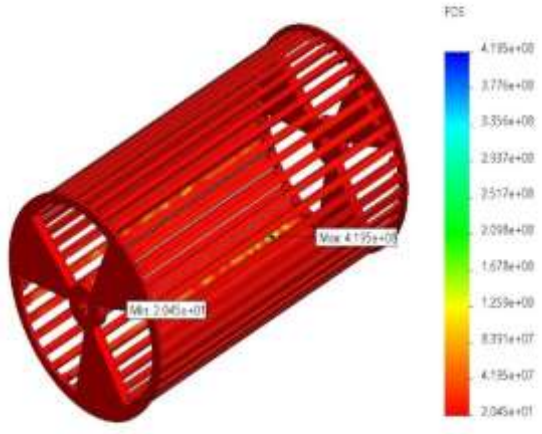


Figure 12: Factor of safety FEA of threshing drum

4. CONCLUSION

The Finite Element Analysis (FEA) conducted on both the threshing shaft and drum of the machine confirms the robustness and safety of the designs. For the threshing shaft, the application of forces at designated points resulted in a maximum bending stress significantly below the material's yield strength, with a factor of safety (FOS) indicating more than adequate reliability. The design shaft diameter proved to be safe for fabrication, as evidenced by the FEA validation of bending moments and stress distribution.

Similarly, the drum unit, supported by a spider arm and subjected to calculated twisting and weight loads, demonstrated excellent performance under static conditions. The von Mises stress remained well below the yield strength of the plain carbon steel, ensuring safety against failure. Minimal displacement and strain values further underscored the drum's capacity to withstand operational loads, and the high FOS around the drum's cylindrical surface suggested long-term durability with minimal fatigue.

Overall, the design and material selection for both components effectively meet the mechanical and operational requirements, ensuring the machine's reliable performance and longevity.

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