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(Research Article)

Design and Implementation of a High-efficient Chicken Defeathering Machine for Commercial Utilization

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Abstract

Due to the challenges of poor quality and efficiency associated with the traditional method of defeathering chickens, in addition to the highest efficiency value of 95% that has so far been achieved in reviewed studies, the need to further improve on the efficiency of the chicken defeathering (CDF) machine necessitated this study. Therefore, this study employed a systemic approach to design and develop a high-efficient chicken defeathering machine (CDF) for commercial utilization. The frame material was optimized using Granta software in quest of getting a material of high strength and low cost at minimal mass design. The selected material was further subjected to stress and deformation analyses using finite element analysis (FEA) technique in order to ensure a non-plastic failure of the material during service conditions. The other components of the CDF machine were designed and analyzed for optimal performance assurance. The results obtained from the study showed that low carbon steel material was suitable for the CDF frame design and all the CDF components design assured the safety of the design. An average efficiency of 98.1% was obtained at an average operating time of twenty seconds. Hence, this machine can be commercialized for low-time and high-efficient processing of poultry birds.

Keywords: Material optimization, Finite element analysis, Defeathering machine, Mechanical design

1. Introduction

Defeathering of poultry birds is one of the stages involved in poultry meat processing before it reaches the end product for cooking. Therefore, the concerns are on the slaughtering and defeathering while still maintaining the high standard of hygiene and fast product delivery [1]. Sequel to that, the increased preference for chicken meat over other meat types has generated major interests in poultry farming and processing industry. Considering the rapid growth of the 'world population, livestock consumption rate may likely be increasing correspondingly to meet the effective protein requirements of the world [2]. According to [3] and [4], chicken processing has faced challenges that are of safety and health concerns in the developed and developing countries. Scalding and manual defeathering operations of chicken have an associated level of human exposure to occupational risk and other health hazards [2]. This challenge 'calls for effective mechanization of the process which will support quality, safe, ergonomic and economic operation. The mechanization of the process makes the operation an economical practice that saves time, cost, and increases number of productions per day compared to the manual approach of removing poultry feathers. De-feathering or plucking refers to the process of removing feathers from scalded fowls [5] . The design and production of a chicken de-feathering (CDF) machine requires detailed and accurate planning and consideration of various design variables that would ensure an optimal performance of the machine. In manual de-feathering operations, an optimal condition of these variables: scalding temperature, duration of immersion, age of the chicken and feather retention force determine the quality of the final product [6] . There is need for the poultry processing machine to be user friendly, reliable and efficient in order to avoid accidents and infections from poultry carcass which may occur during usage operations [7].

On the basis of these concerns, some studies have been carried out by researchers in quest of designing and developing an efficient chicken defeathering machine. Also, some works were done by employing various methodologies towards the modification and improvement of the efficiency of the system. Locally sourced materials were used by [5] in the design and development of a household poultry de-feathering machine. The developed machine removed the feathers of the chicken

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without any serious damage to the chicken and the average efficiency of the machine was approximately 95%. The de-feathering efficiency of the machine was studied using two different species of chicken (Isa brown layer and cockerel) in the work of [8]. They designed the de-feathering machine and produced it with the three primary units: frame structural assembly, plucking unit and power transmission unit present in the machine. The efficiency was machine was studied at speeds of: 225, 312, 369, and 426 RPM and scalding time of 30, 60 and 90 seconds. It was deduced that as the machine speed and scalding temperature increased, the plucking efficiency dropped. Also, cockerel gave the highest machine efficiency. Also, [9] designed and developed a chicken de-feathering machine with a boiler system for scalding the chicken before de-feathering was incorporated into the design. The average feather removal time was estimated as 32.5 seconds. This value indicated that the feather removal rate for the design was 107 birds/hour. The developed machine was designed to process two birds at a time. The scalding temperature is very imperative in the processing of poultry birds. The ideal scalding temperature to be employed is determined by the type of bird being processed [10]. In addition, [11] designed, developed and evaluated the performance of a chicken de-feathering machine for small scale farmers. The design employed a food-grade plastic drum in the de-feathering chamber, the frame structure was made of wood and the plucking fingers were made out of tapered rubber. It was observed that at a running speed of 300 RPM, the mass of removed feathers totaled 126.97g and the machine capacity was found to be one bird per 25 seconds with an efficiency of 95%. This study was centered on detailed design and' development of a high-efficient chicken de-feathering machine. Computer aided design (CAD) software tool was employed in the virtual design of the system and the material for the design of some significant parts of the machine were selected with the aid Granta software. The machine was analyzed for structural strength to ascertain its suitability during service.

2. Material Selection

Material selection is an imperative aspect of engineering design. In the design of a product, the key goal of material selection is to reduce cost while meeting product performance goals [12]. The selection of the materials used in design of the CDF machine was solely based on strength, availability, applicability and cost requirements. Table 1 shows the various CDF machine members and the material type used.

Table 1. CDF machine members and the material type used

Component	Materials
De-feathering chamber	316 Stainless steel
Pluckers/fingers	Silicone rubber
Belt	Rubber
Shaft	304 Stainless steel
Rotating plate	304 Stainless steel
Pulley	Mild steel

The structural frame of the CDF machine is a very crucial component of the system as it carries the entire weight of the other machine components. Therefore, its failure during service operation would definitely affect the optimal performance of the system. Based on this concern, the structural frame material was selected by optimizing the material's strength and cost factors using Granta software 2009 version. According to [13], the performance matrix of a material is given by equation 1.

$$P = \left[\begin{pmatrix}\text{functional}\\ \text{requirement, f}\end{pmatrix}, \begin{pmatrix}\text{Geometric}\\ \text{parameters, G}\end{pmatrix}, (\text{material properties, m})\right] \quad (1)$$

Where: P = the performance matrix which describes the essential features of the material . Optimum product design entails the selection of a material and geometry that either maximize or minimize P using the index of desirability [12]. Modelling the performance matrix of the structural frame, the used functional criteria was a material that can withstand the compressive loads of the other components of the machine, while the constraint was a material with high yield strength, low weight and low cost. The material choice is also expected to be ductile in order to accommodate impact loads and non-linear load effects. Hence, yield stress is given as,

$$\sigma_y = \frac{F}{A} \quad (2)$$

Area of the material is given as $A = \frac{F}{\sigma_y}$ (3)

The mass of the frame material is expressed as,

$$m = \rho A t \quad (4)$$

Therefore, substituting equation 3 into equation 4, we have,

$$m = F.t.\frac{\rho}{\sigma_y} \quad (5)$$

Where: σ_y = elastic yield strength, m = mass of the frame, ρ = density of the frame material, A = frame area, t = material thickness and F = applied load. Therefore, the mass of the structural frame is minimized by selecting material with the lower value of M_1 and higher value of M_2

$$M_1 = \frac{\sigma_y}{\rho} \quad (6)$$

$$M_2 = \frac{\sigma_y}{C_a \rho} \quad (7)$$

Where: C_a = material cost.

The output from the Granta environment was further subjected to stress and deformation analyses using finite element analysis (FEA) method. In addition, the defeathering drum (DD) was analyzed using finite element software (ANSYS

17.2 version) to ascertain the suitability of AISI 316 stainless steel material for the DD production.

3. Method

The method employed in this study entailed: the description of the operational principle of the CDF machine, component design of the CDF machine using computer aided design (CAD) procedures, FEA validation, design implementation, testing and performance evaluation. The CDF machine was designed with Autodesk Inventor Professional software 2021 version.

3.1 Operational principle of the CDF machine: The chickens used in this study died by natural cause within 30-48 minutes in the poultry farm situated at Ihiala, Anambra state, Nigeria before they were purchased. The chickens were scalded at a temperature of 80℃, and then fed directly into the machine through the de-feathering chamber, where sets of de-feathering rubber fingers or pluckers were arranged to aid easy and effective de-feathering process. The rotating plate attached to a drive shaft that is connected to an electric motor through a system of pulleys is responsible for the rotation of the chicken. The rotation of the chicken about the spaces between the adjacent pluckers brings about the defeathering of the chicken. The principle of the CDF machine operation is solely based on the concept of rubbing, impact and collision actions. The impact force on the chicken is actuated by the rotating shaft, which consequently results to rubbing action. The surface roughness and area of contact of the rubber pluckers determine the severity of rubbing action. Also, the collision action between the rubber fingers and the chicken contributes greatly to the defeathering action of the chicken. Therefore, the impact, rubbing and collision actions cause the defeathering of the chicken.

3.2 Components design: The design of the various components of the CDF machine is expressed in this section of the study.

3.2.1 Structural frame: The structural frame was fabricated using rectangular angle iron of dimension 40×40×4.0mm. The area of the frame was computed using,

$$\text{Area, } A = \text{length} \times \text{width} \tag{8}$$

The frame's box volume was determined using,

$$V = \text{length} \times \text{width} \times \text{height} \tag{9}$$

The height of the frame = 0.5m. Figure 1 shows the CAD model of the frame structure and table 2 depicts the design specifications of the frame.

Figure 1. CAD model of the frame structure

Table 2. Design specifications of the frame

Properties	Design specifications
Mass	16.948kg
Surface area	$1.098m^2$
Volume	$0.00216m^3$
Material density	$7850\ kg/m^3$
Length of the material	$0.4m$
Width of the material	$0.4m$
Height of the material	$0.5m$

3.2.2 Drive shaft: The drive shaft was designed by considering the loads on the two support bearings, the speed of the electric motor, and the compressive load of the entire de-feathering chamber and rotating plate. All these are the variables that impact on the drive shaft. Hence, for design safety, the shear force, shear stress, bending moment, deflection and bending stresses of the drive shaft were thoroughly examined. Figures 2-3 delineates the drive shaft in loaded condition. The diameter of the drive shaft was 40mm.

Figure 2. The drive shaft in loaded condition

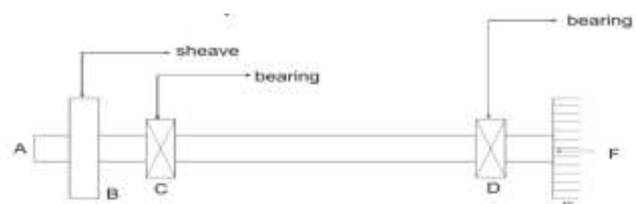

Figure 3. Components on the drive shaft

According to [5], in designing a drive shaft based on the factor of strength, the axial load, the combined torsional and bending loads must be considered, else the safety of the design is not guaranteed. Therefore, the axial load F1 which comprised of the rotating plate that was coupled to the drive shaft, the de-feathering chamber and the weight of the chicken was considered in the drive shaft design. Furthermore, when a shaft is under torsion and bending loads, [14] gave the models for computing some of the dynamic variables of the shaft. These models are given in equations 10 and 11.

$$T_m = \sqrt{M^2 + T^2} = \frac{\pi}{16}\tau d^3 \tag{10}$$

$$M_b = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] = \frac{\pi}{32}\sigma_b d^3 \tag{11}$$

Where: T_m = equivalent twisting moment factor, M = maximum moment (Nm), T = transmitted torque (Nm), σ_b = bending stress factor (N/m^2), d = shaft diameter (m), τ = shearing stress (N/m^2), M_b = equivalent bending moment. In addition, the drive shaft which is under coupler effect is subjected to torsional load that initiates torsional shear stress in the drive shaft [14]. The torsional shear stress of the drive shaft was designed using equation 12.

$$\frac{\tau}{r} = \frac{T}{J} = \frac{C.\theta}{l} \tag{12}$$

Where: τ = Shear stress due to torsional load, r = Drive shaft radius, T = Torque moment, J = Polar moment of inertia, C = Shear modulus of the shaft material, l = Shaft length, and θ = Twist angle in radians

3.2.3 Driving pulley: The design specifications of the driving pulley are shown in figure 4 and table 3. The driving pulley was coupled to the rotor of the electric motor which drives the system.

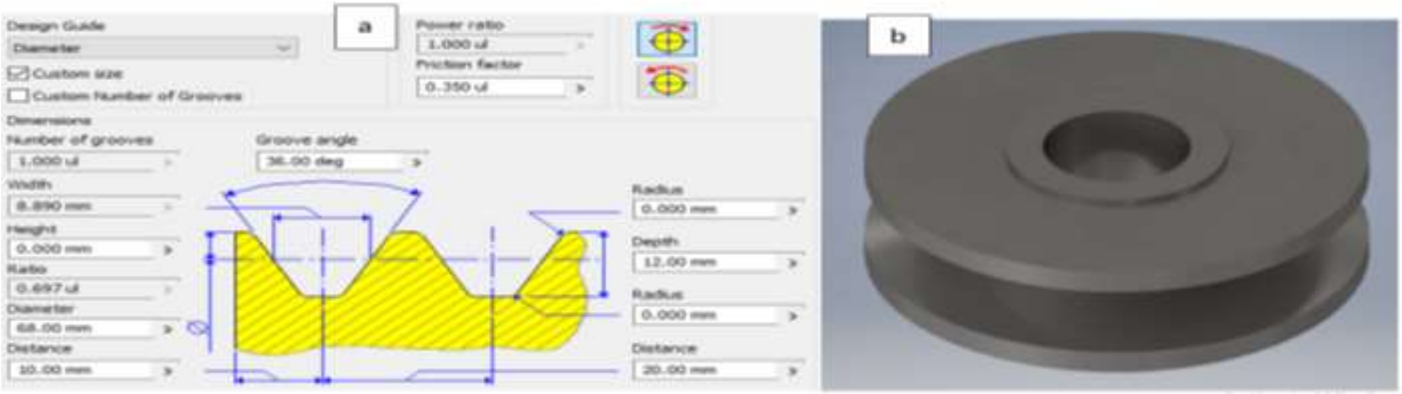

Figure 4. (a) Design geometry of the driving pulley (b) CAD model of the driving pulley

Table 3. Driving pulley specification

Properties	Specifications
Mass	0.389 kg
Volume	49.514 cm^3
Surface area	153.596 cm^2
Density	7.850 g/cm^3
Material	Mild steel

3.2.4 Driven pulley: The design geometry and specifications of the driven pulley are shown in figure 5 and table 4.

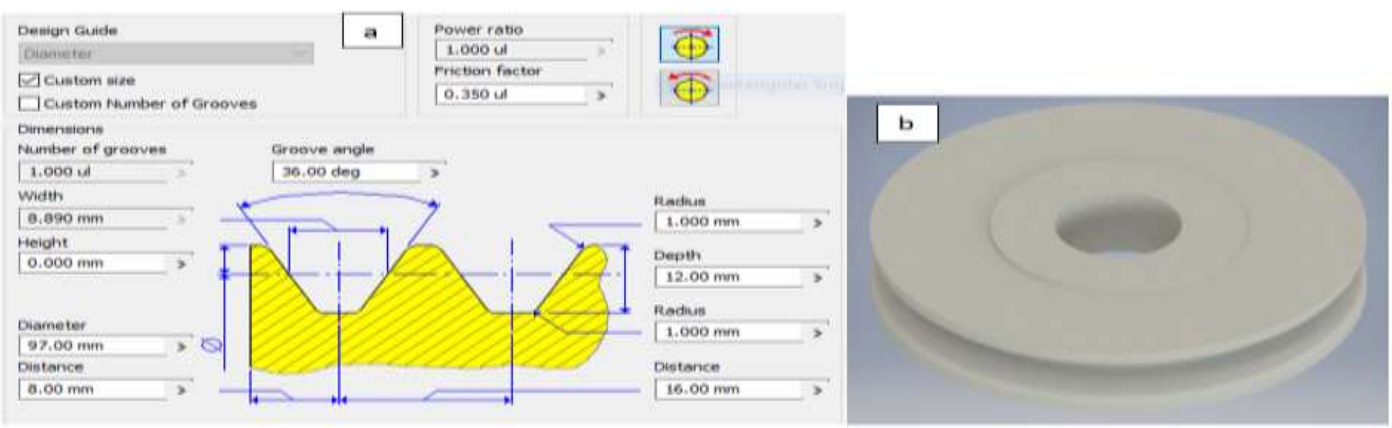

Figure 5. (a) Design geometry of the driven pulley (b) CAD model of the driven pulley

Table 4. Driven pulley specifications

Properties	Specifications
Mass	0.747 kg
Volume	95.096 cm^3
Surface area	231.601 cm^2
Density	7.850 g/cm^3
Material	Mild steel

The driven pulley was coupled to the drive shaft which transmits power to the rotating plate of the system. The speed of the driven pulley was obtained using the speed ratio formula expressed as,

$$\frac{N_2}{N_1} = \frac{d_1}{d_2} \tag{13}$$

Where: $d_1 =$ driving pulley diameter (m), $D_2 =$ driven pulley diameter (m), $N_1 =$ speed of the driving pulley in rpm , $N_2 =$ speed of the driven pulley in rpm

3.2.5 De-feathering chamber: The de-feathering chamber was fabricated using AISI 316 stainless steel sheet of thickness 1.5mm. The diameter and height of the chamber were 400mm and 500mm respectively. The area of the chamber was computed using equation 14. Figure 6 shows the 'image of the de-feathering chamber.''

$$\text{Area of the defeathering chamber} = \pi dh \tag{14}$$

Figure 6. De-feathering chamber

3.2.6 Rotating plate: The rotating plate was fabricated with 304 stainless steel plate of 1mm thickness. The rotating plate had a circular cross-section with a diameter of 320mm. The area of the plate was determined using equation 15.

$$Area = \pi r^2 \tag{15}$$

3.2.7 Electric motor: The electric motor is the prime mover that generates the rotary motion of the rotating plate. The specifications of the prime mover used in this design is shown in table 5.

Table 5. Specifications of the electric motor

Model	AM 80KY6
Voltage	220v
Efficiency	78
Power factor	0.95
Rated current	1.22 A
Starting current	2.15 A
Frequency	50Hz
Horse power	0.49Hp
Rated power	0.37kw
Rated speed	910 rpm
Torque	2.82Nm

3.2.8 Belt design: Figure 7 shows the belt connection between the driving and driven pulleys, and the angle of contact.

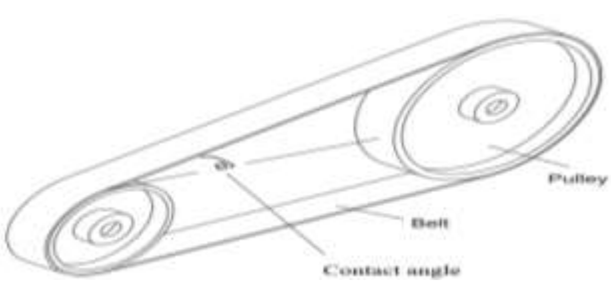

Figure 7. Belt connection assembly

The angle of contact between the belt and pulley was measured to be approximately 3.08°. Equation 16 was used to determine the centre distance of the two pulleys.

$$\sin\theta = \frac{r_1 - r_2}{c} \tag{16}$$

The length of the belt was determined using equation 17.

$$L = \frac{\pi}{2}(D_1 + D_2) + 2C + \frac{(D_1 - D_2)^2}{4C} \tag{17}$$

Where: $L =$ length of belt,
$D_1 =$ driven pulley diameter, 97mm.
$D_2 =$ driving pulley diameter, 68mm.
$C =$ centre distance of the pulleys
r_1 & r_2 = radii of the bigger and smaller pulleys respectively.
θ = angle of contact.

Tables 6-8 show the specifications, properties and strength check analysis respectively of the belt used in power transmission from the electric motor to the rotating plate pulley.

Table 6. Belt specifications

Model	SKU 9002-2045
Length (mm)	775mm
Length type	Outside circumference
Top belt width per strand (m)	0.13
Nominal thickness (m)	0.08

Table 7. Belt properties

Design variables	Description/value
Name	Narrow V-belt ANSI/RMA IP-22
Size	1.000ul
Number of belts	40.00deg.
Wedge angle	9.525mm
Height	7.938mm
Effective width	8.890mm
Effective length	800.100mm
Internal length	750.227mm
Length correction factor	0.870ul
External line offset	0.000mm
Pitch line offset	-0.635
Minimum recommended pulley effective diameter	67.310mm
Maximum flex frequency	100.00Hz
Maximum belt speed	42.062mps
Specific mass	0.107kg/m
Base power rating	1.145kw

Table 8. Strength check analysis

Design variables	Description/value
Power	0.370kw
Torque	3.883Nm

Speed	910.000rpm
Efficiency torque factor	0.980ul
Efficiency	0.962ul
Belt slip	0.019ul
Arc of contact correction factor	0.986ul
Service factor	1.200ul
Resultant service factor	2.654ul
Length correction factor	0.870ul
Number of belts correction factor	1.000ul
Number of pulleys correction factor	1.000ul
Modify friction with speed factor	0.012s/m
Tension factor	1.300ul
Belt speed	4.561mps
Belt flex frequency	11.402Hz
Number of belts required	0.452ul
Effective pull	81.117N
Centrifugal force	2.229N

3.2.9 Power requirement of the CDF machine: According to [15], the total amount of power (P_t) required to optimally run the CDF machine is the sum of the power needed to turn the rotating plate (P_r) and the power for the de-feathering of the bird's feathers (P_f). This is mathematically given in equation 18.

$$P_t = P_r + P_f \tag{18}$$

The formulae given by [15] for computing the values of P_r and P_f are expressed in equations 19 and 20 respectively.

$$P_r = W_p R_P \omega_P \tag{19}$$

$$P_f = T_s \omega_f \tag{20}$$

$$\text{But, } T_s = \frac{\pi d^3 \tau}{16} \tag{21}$$

$$\text{But, } \omega_P = \frac{2\pi N}{60} \tag{22}$$

Where: W_p = Weight of the rotating plate, R_p = Rotating plate radius, ω_P = Angular speed of the rotating plate, P_f = Required defeathering power, T_s = Defeathering chamber torque, ω_f = Angular speed of the defeathering chamber, d = Mean diameter of the defeathering unit, τ = Shearing stress of the defeathering chamber.

To compute the value of the motion torque expressed in equation 21, the shear stress experienced by the defeathering chamber should be determined. The centripetal force acting on the chicken during the defeathering operation ensured that the chicken revolved within the curvilinear path of the chamber and as well directs its acceleration towards the central axis of the chamber. This force in addition to centrifugal force are responsible for the removal of the chicken feathers. Therefore, the shear stress was determined using equation 23.

$$\tau = \frac{\text{Centripetal force}}{\text{Area of the defeathering chamber}} = \frac{m\omega_f^2 r}{A} \tag{23}$$

Where: r = the radius of the defeathering chamber/drum, m = the average mass of a chicken = 2.585kg.

3.2.10 Efficiency of the CDF machine: The efficiency of the CDF machine as expressed in the study of [16] is given in equation 24.

$$\eta_{df} = \frac{W_m}{W_m + W_n} \times 100 \tag{24}$$

Where: η_{df} = CDF efficiency, W_m = weight of feather plucked by the CDF machine (kg), W_n= weight of feather manually plucked out after using the CDF machine, (kg).

The CAD model of the assembled CDF machine and the side component view are shown in figures 8 and 9.

Figure 8. CAD model of the CFT machine

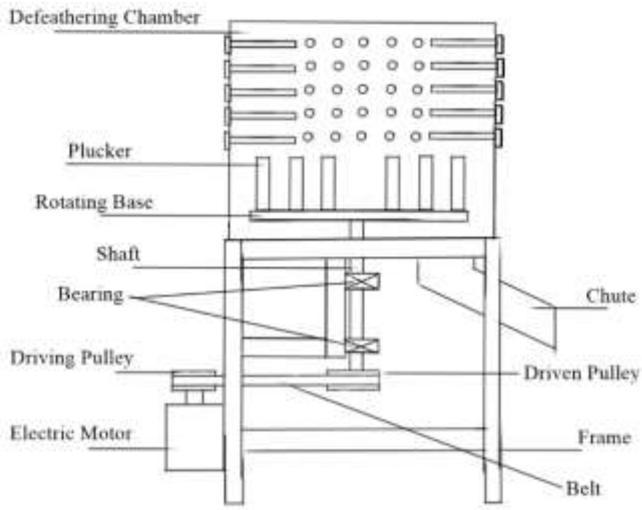

Figure 9. Side component view of the CDF machine

Table 9. Summary of the design variables of the CDF machine

S/N	Design variables	Values
1	Speed of the driving pulley'	910 rpm
2	Speed of the driven 'pulley'	637.9rpm
3	Area of the frame	$0.16m^2$
4	Box volume of the frame	$0.08m^3$
5	Area of the de-feathering drum/chamber	$0.6284m^2$
6	Area of the rotating plate	$0.0804m^2$

7	Belt length	0.8m
8	Speed of the rotating plate	637.91rpm
9	Power required to turn the rotating plate	320.64 watts
10	Shear stress in the defeathering chamber	3679.99N/ m^2
11	Torque of the defeathering chamber	46.24Nm
12	Power required for defeathering operation	3089watts
13	Centripetal/centrifugal force for feather removal	2306.98N
14	Centre distance of the pulleys	270mm

4 Results and Discussions

4.1 Frame material optimization: The result of the material optimization for the CDF frame revealed a selection of eight smaterials out of ninety-eight materials that were used in the optimization operation. Figure 10 depicts the graphical plot of material's yield strength against the density of the material. The box-selection method was used to obtain the eight materials that maximizes the yield strength/density parameter for frame mass minimization. Figure 11 delineates the cost minimization plot of yield strength against the product of density and price.

The material optimization was concentrated on the family of metals and alloys as depicted in figure 10. The candidate materials which maximized the yield strength/density factor for frame mass minimization and high strength design value are shown in figure 10 as well. Also, as cost is an imperative aspect of design, figure 11 shows the material selection chart based on the cost of materials. Some metals and alloys passed the minimum cost requirement which was still based on the performance indices of the material. Table 10 gives the summary of the candidate materials selected for the frame design of the CDF machine using their performance indices.

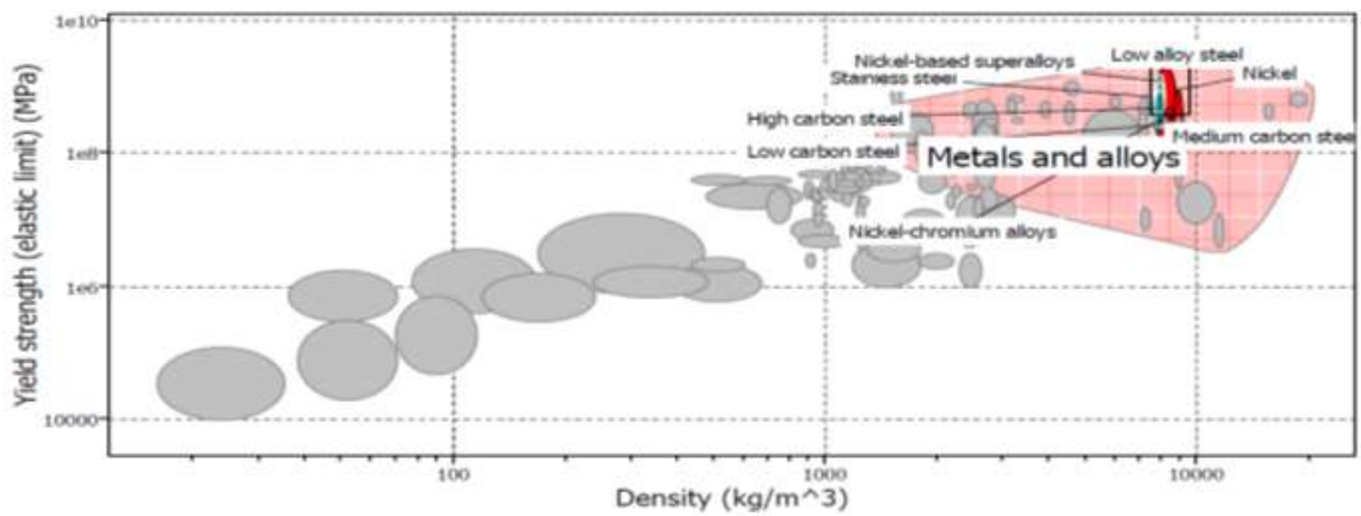

Figure 10. Graph of yield strength against density of material

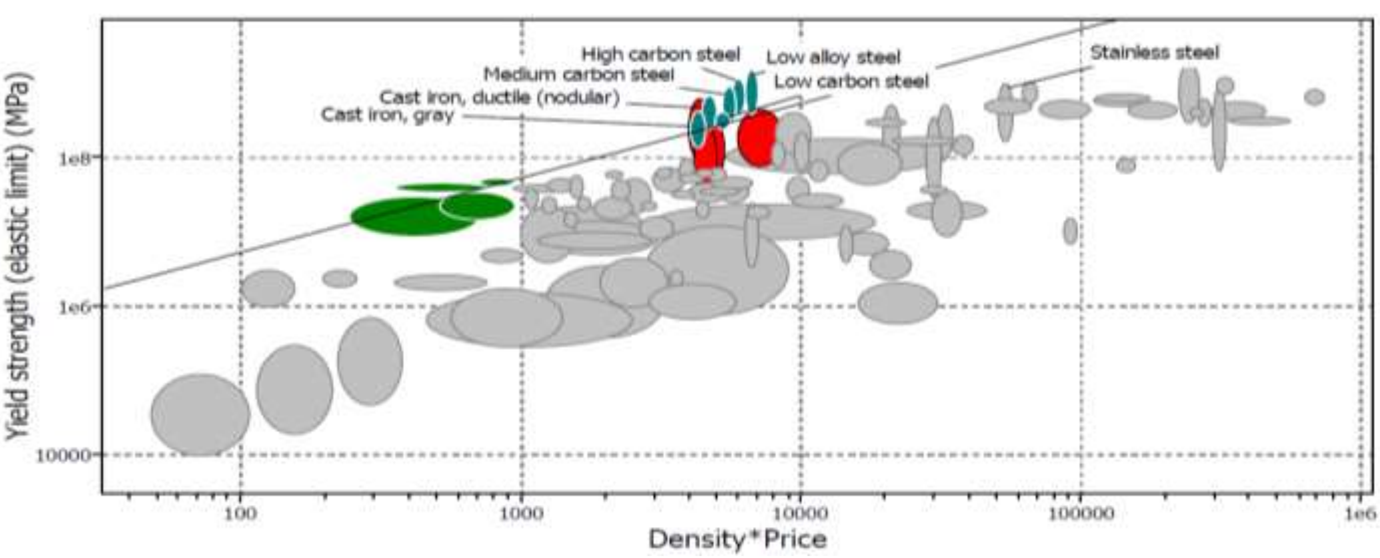

Figure 11. Material selection chart based on cost

Table 10. Possible materials for high strength and minimum frame mass design

S/N	Material Type	Density (Kg/m^3) (ρ)	Price (Naira/Kg) (C_a)	Yield Strength (MPa) (σ_y)	$M_1 = \frac{\sigma_y}{\rho}$ ($\times 10^6$)	$M_2 = \frac{\sigma_y}{C_a \rho}$
1	Stainless steel	7848	2483.125	584	0.0744	29.968
2	Low carbon steel	7845	242.875	322.4	0.0411	169.207
3	Low alloy steel	7848	308.125	952	0.1213	393.687
4	Nickel-based superalloys	8195	10766.250	1098	0.1340	12.445
5	Nickel-chromium alloy	8395	11128.750	412.4	0.04912	4.4142

Low carbon steel material was employed for the frame design owing to the factors of easy accessibility, cost and it is still among the suitable materials for the design implementation'. Also, low carbon steel material had the lowest M_1 value and high M_2 value compared to the various materials outputted from the material optimization process. Table 11 presents the supporting information on the choice of material.

Table 11. The supporting information about low carbon steel blade material

S/N	Properties	Units	Value
1	Mass density	Kg/m^3	7845
2	Elastic modulus	GPa	206
3	Poisson's ratio	-	0.3
4	Compressive yield stress	MPa	322.4
5	Rupture stress	MPa	335

4.2 FEA simulation result of the frame using Low carbon steel for its design: Figures 12 and 13 present the bending stress and deformation analyses result respectively of the CDF machine frame using a beam mesh of 338 elements and 348 nodes. The frame had a total load of 250N from the defeathering unit and 180N from the unit of the prime mover.

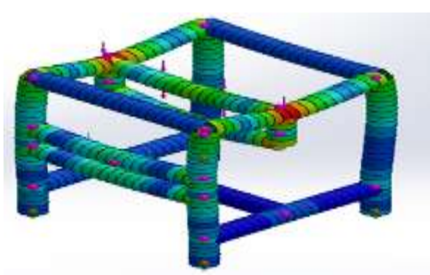

Figure 12. Frame's axial and bending stress result

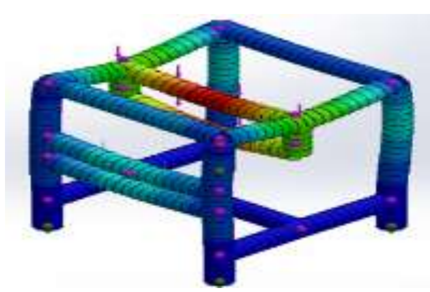

Figure 13. Resultant frame displacement

From figure 12, a maximum bending stress of 1.703e7 (17.03MPa) could be observed on the central axis of the frame. This could be attributed to the load concentration of the defeathering unit and the operational impact load effect on that section of the frame. Also, a minimum stress value of 3.29e4 (32.9KPa) was noted at most of the other members of the frame that are not situated at the central axis. Considering the obtained maximum stress value, it is far lower than the yield point stress of low carbon steel material (322.4MPa). Hence, the frame design using low carbon steel is very safe as the material will not experience plastic failure during service operations. In addition, the deformation degree of the frame under various load effect as presented in figure 13 revealed a maximum value of 4.738e-02mm (0.04738mm) at the central axis of the frame and the lowest value of 1000e-30mm (very negligible) at most of its other member units. These values attest that thelow carbon steel material design is suitable for the frame as the deformation values are within the permissible design boundary and no plastic deformation is envisaged.

4.3 Result of the defeathering drum analysis: The safety of the defeathering drum was analyzed using ANSYS 17.2 software. The total deformation and equivalent von-mises stress produced on the drum while in service condition are respectively shown in figures 14 and 15.

The drum was subjected to a pressure load of 4.774E-05Pa on its interior area for a chicken mass of 3kg and a rotational velocity load of 66.81rad/s from the rotating plate coupled to the prime mover of speed 637.91rpm. From figure 14, the maximum deformation of the drum was 0.28993mm and it occurred at the base of the drum. This value attests that the drum is safe for its intended application since the obtained maximum deformation did not exceed the permissible strain value of 50mm for AISI 316 stainless steel material [17]. The occurrence of the maximum deformation at the base of the drum could be as a result of the velocity load of the rotating plate on the drum. Also, figure 15 revealed that the maximum von-mises stress of 68.292MPa occurred at the circular hole positions of the pluckers. This could be caused by the pressure load of the chicken that was impacted on the interior area of the drum. This stress value suggests that the drum is safe since the obtained maximum stress did not exceed the maximum yield strength (290MPa) of the drum material.

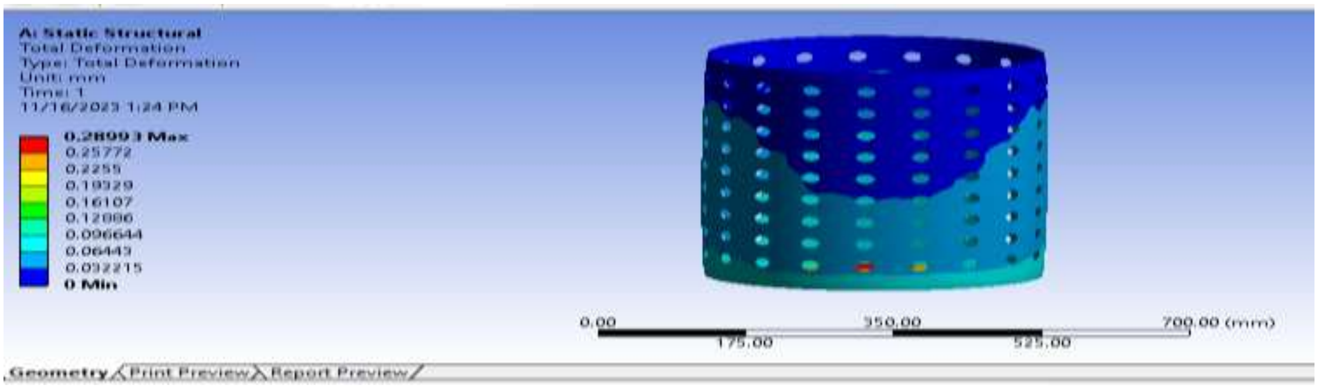

Figure: 14. Total deformation produced on the drum

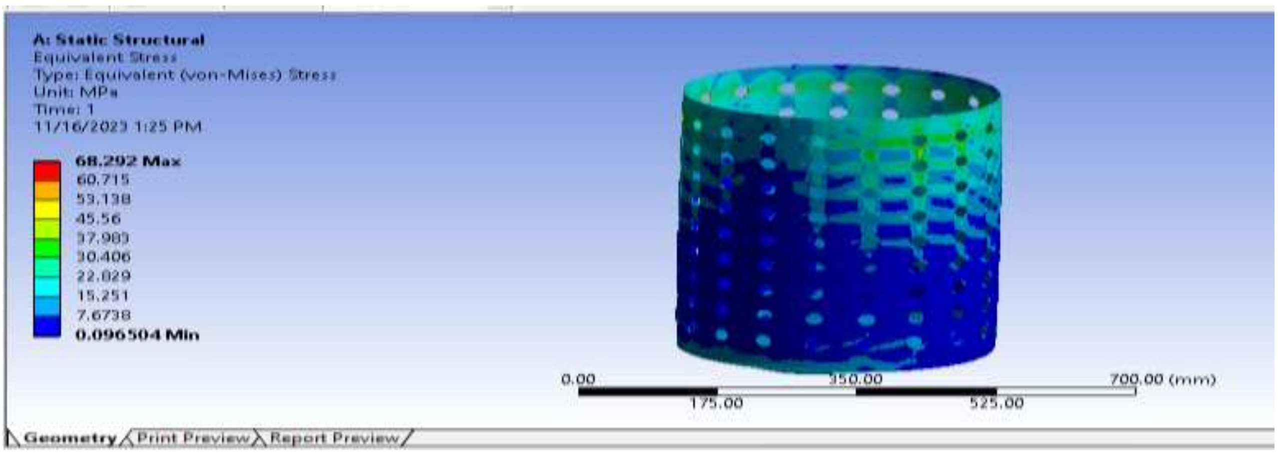

Figure 15. Equivalent von-mises stress of the drum

4.4 The drive shaft safety analysis result: The safety of the drive shaft used to transmit power from the electric motor to the rotating plate was accessed using shear force (SF), bending moment (BM), deflection, bending stress (BS) and shear stress diagrams as shown in figure 16.

From figure 16, the safety of the drive shaft is guaranteed since the maximum: SF, BM, deflection, BS, and SS are respectively 6.11233N, 0.156553Nm, 0.201162μm, 0.102057MPa and 0.00708708MPa. These values are too small to cause a plastic failure of the shaft during service operations.

4.5 Performance evaluation of the CDF machine: The performance evaluation of the CDF machine was conducted thrice at a fixed motor speed of 600 rpm using chickens of varying weights as shown in table 12. Scalding temperature of 80℃ was maintained throughout the tests period and the average efficiency of the machine was computed to be 98.1% at an average time of 20 seconds.

5. Conclusions

This study presents a systemic approach that utilizes the top-bottom design technology to effectively design and implement a high-efficient CDF machine for commercial purposes. The various component designs following the analytical and simulation test results, performed at optimal degrees and are very safe based on their respective design safety values. The CDF machine performed optimally at an efficiency value of 98.1% for an average time of 20 seconds.

Table 12. The summary of the test results

Initial mass (kg)	Final mass after defeathering (kg)	Mass of the feather removed after defeathering (kg)	Mass of the feather remained after defeathering (kg)	Mass of the total feather on the chicken (kg)	Time of defeathering (s)
1.32	0.98	0.35	0.01	0.36	16
1.72	1.31	0.435	0.005	0.44	20
1.91	1.40	0.53	0.01	0.54	23

$$\text{Efficiency} = \frac{\text{Mass of plucked feathers}}{\text{Total mass of feathers}} \times 100 \tag{24}$$

$$E_1 = \frac{0.35}{0.36} \times 100 = 97.22\%$$

$$E_2 = \frac{0.435}{0.44} \times 100 = 98.86\%$$

$$E_3 = \frac{0.53}{0.54} \times 100 = 98.15\%$$

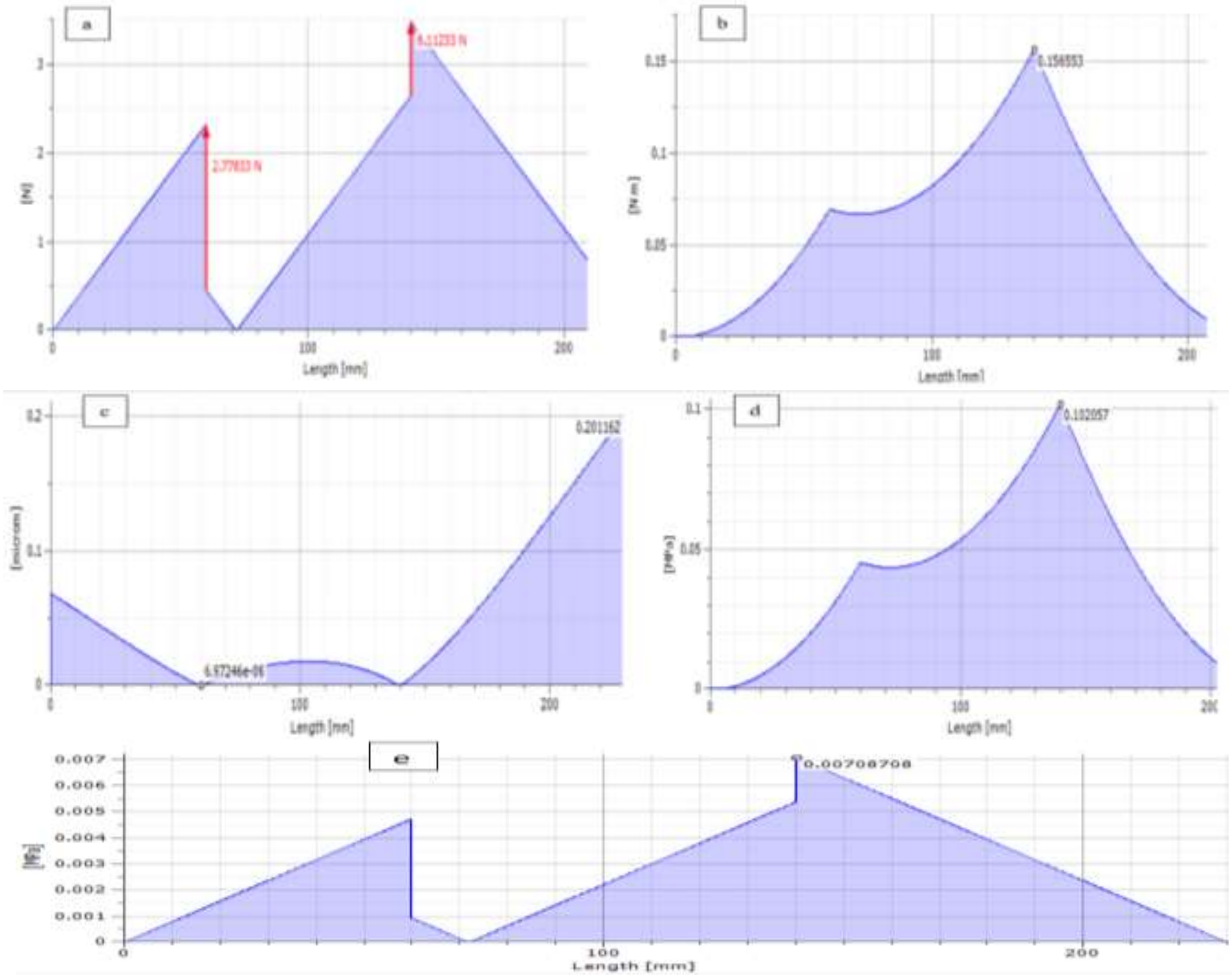

Figure 16. Drive shaft safety analysis using (a) SF diagram (b) BM diagram (c) deflection diagram (d) BS diagram (e) SS diagram

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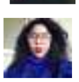

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