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Conceptual Design of an Instant Plantain Dough Meal Machine

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Abstract **Original Research Article**

Plantain (Musa Paradisiaca) is one of the major staples widely cultivated in the tropics and subtropics, with Nigeria being a significant producer worldwide. Plantain dough is a meal common in the southern part of Nigeria, known for its pharmacological benefits, which include serving as an antihypertensive, hypoglycemic, anti-cholesterol, antioxidant, and anti-allergic agent. The annual plantain production suffers a significant loss of over 50% due to post-harvest issues, resulting from inadequate facilities and food processing machines. This thesis addresses one of the major challenges of the plantain (food) industry. A conceptual design of a plantain processing machine was created using a stainless-steel grade of austenitic 304, and SolidWorks CAD was utilized to analyze the behavior and performance of the conceptual design. Employing a solid curvature-based mesh of high-quality settings with 195,699 nodes and 102,854 elements. The combined effects of applied torque, normal forces, gravity, and thermal loading do not compromise the structural integrity of the assembly. Stress, strain, and displacement values remain well below critical thresholds, demonstrating that the current design is mechanically sound and suitable for the intended operating design conditions.

Keywords: Conceptual design, plantain dough, milling, pharmacological, SolidWorks CAD, Austenite 304.

1. Introduction.

Plantain *(Musa paradisiaca)* is considered one of the common foods in the south western part of Nigeria, (Olanrewaju & Abidemi, 2017) rich in starch and a low sugar content, coupled with an abundant, bitter-tasting latex (Abiodun-Solanke & Falade, 2011), harvested freshly at the mature (unripe) stage and allowed to ripen within two to seven days, consequently, making plantain a highly perishable fruit. The edible fruit can be prepared by steaming, boiling, baking, roasting, or frying. It is consumed in different varieties alongside various dishes in western Nigeria, such as roasted plantain (booli), diced and fried plantain (dodo), spiced with pepper (dodoikire), plantain beer, plantain chips, moi moi, plantain cake, biscuits, etc. (Akinyemi et al., 2010; Ndayambaje et al., 2019). Plantain is highly rich in nutrients, which are beneficial to human health as it promotes healing, and it’s recommended in the treatment and cure for diseases such as diabetes mellitus, tonsillitis, diarrhea, stomach ulcer, constipation, etc. (Akinola & Okeniyi, 2021; Oyejide et al., 2018).

Over 50% of plantain production in the world comes from Africa, with Nigeria as one of the major producers of plantain in West Africa, with an estimated of over 12 million metric tonnes harvested annually from Africa out of the world's total plantain production. Due to its highly perishable nature with low shelf life, plantain suffers a post-harvest loss of about 50% of the total annual harvest. (Oloyede et al., 2013), with contributory factors such as inefficient storage facilities, inadequate post-harvest processing equipment, and methods of preservation(Morris et al., 2019).

Literature reveals upgrades and improvements on existing plantain storage and processing innovations, (Ayodeji, 2016) Designed a plantain flour processing mill, which in turn is an additional facility to mitigate the post-harvest loss in the plantain industry, with a complete plantain flour processing mill that is edible for human consumption. The fabrication of an instant plantain dough meal machine will also help in mitigating the loss of plantain fruits and, hence, help to boost the food industry in Nigeria. The machine is designed to process raw plantain by boiling and turning it into a dough meal. The design material is stainless steel, which is an edible food.

2. Materials and methods.

2.1 Description of the machine.

The instant plantain dough machine consists of two sections in a single framework, viz are the boiling section and the grinding section, with an overall dimension of 370mm × 400mm × 990mm. The boiling section consists of: a stainless pot for boiling the raw fruit (plantain), which is also incorporated with a drain tap connected to a reservoir, an electric heater plate attached to the bottom plate of the stainless pot, which is supported by the framework. Two metal ball bearings are attached to the pot at the point where it is connected to the frame support to allow the pot to turn at an angle without constraints. In the grinding section, a hopper is connected to a cylindrical pounding chamber and carries a grinding disc at the end. A driveshaft runs through the pounding chamber and carries a grinding disc, which is adjustable, while the one at the end of the pounding chamber is permanently fixed.

The drive shaft carries an auger and is connected to a v-grooved pulley, which is fastened to the shaft with the aid of a key. The auger facilitates the movement of the deposited pulp to the grinding disc. The pulley is connected to the electric motor via a V-belt through which motion is transmitted to the shaft. The driveshaft is supported by ball bearings in the drive mechanism. This assembly sits on a solid framework that aids the rigidity of the dough machine. Figure 2.1 (a-b) shows the detailed machine drawing and exploded view of the machine design drawing.

2.2 Materials Selection

The selection of material in a design is a crucial aspect, since it is an edible food processing machine; hence, the machine components achieving the desired performance are of utmost importance.

The material design consideration was based on the following criteria:

i. Material availability
ii. Mechanical and physical properties
iii. Reliability of components and
iv. Cost-effective.

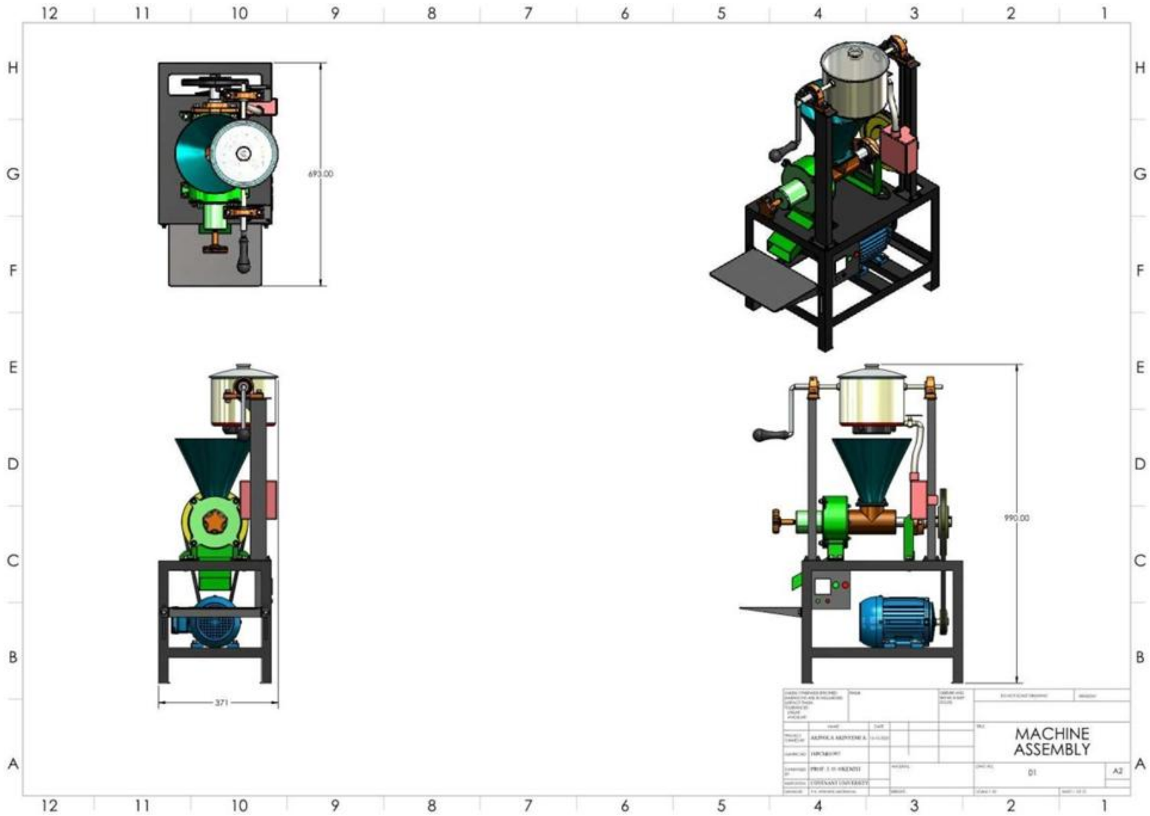
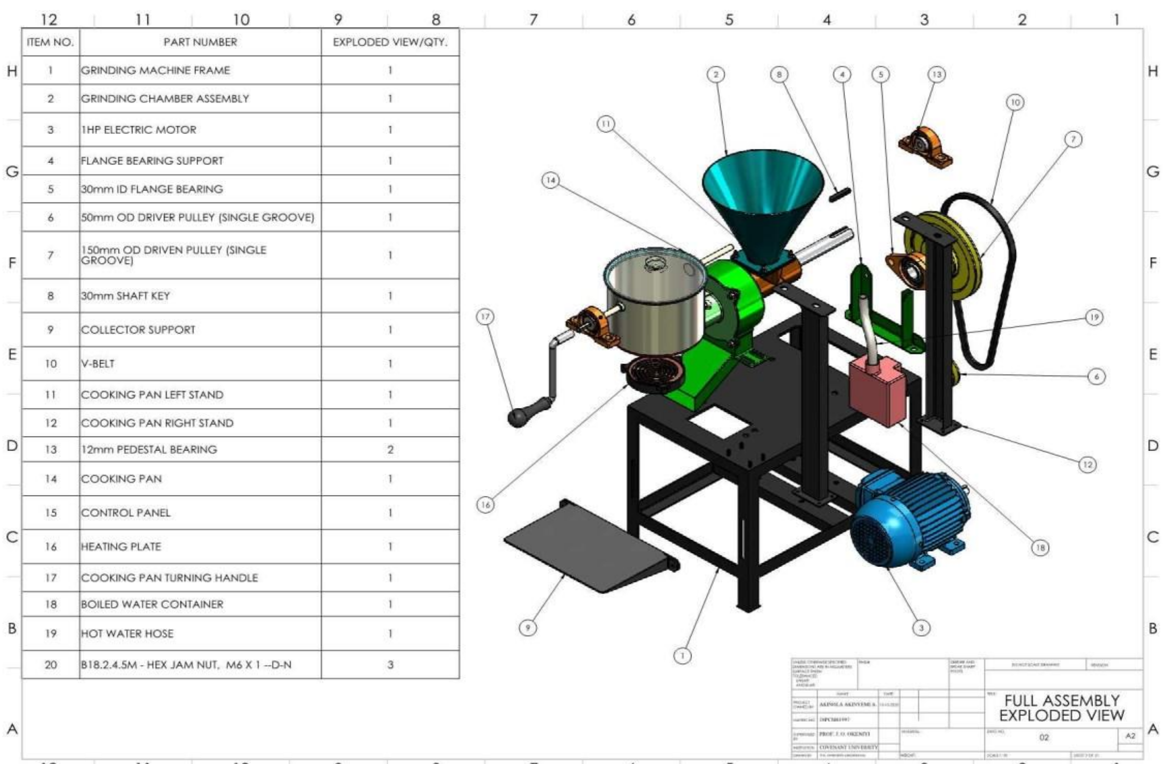
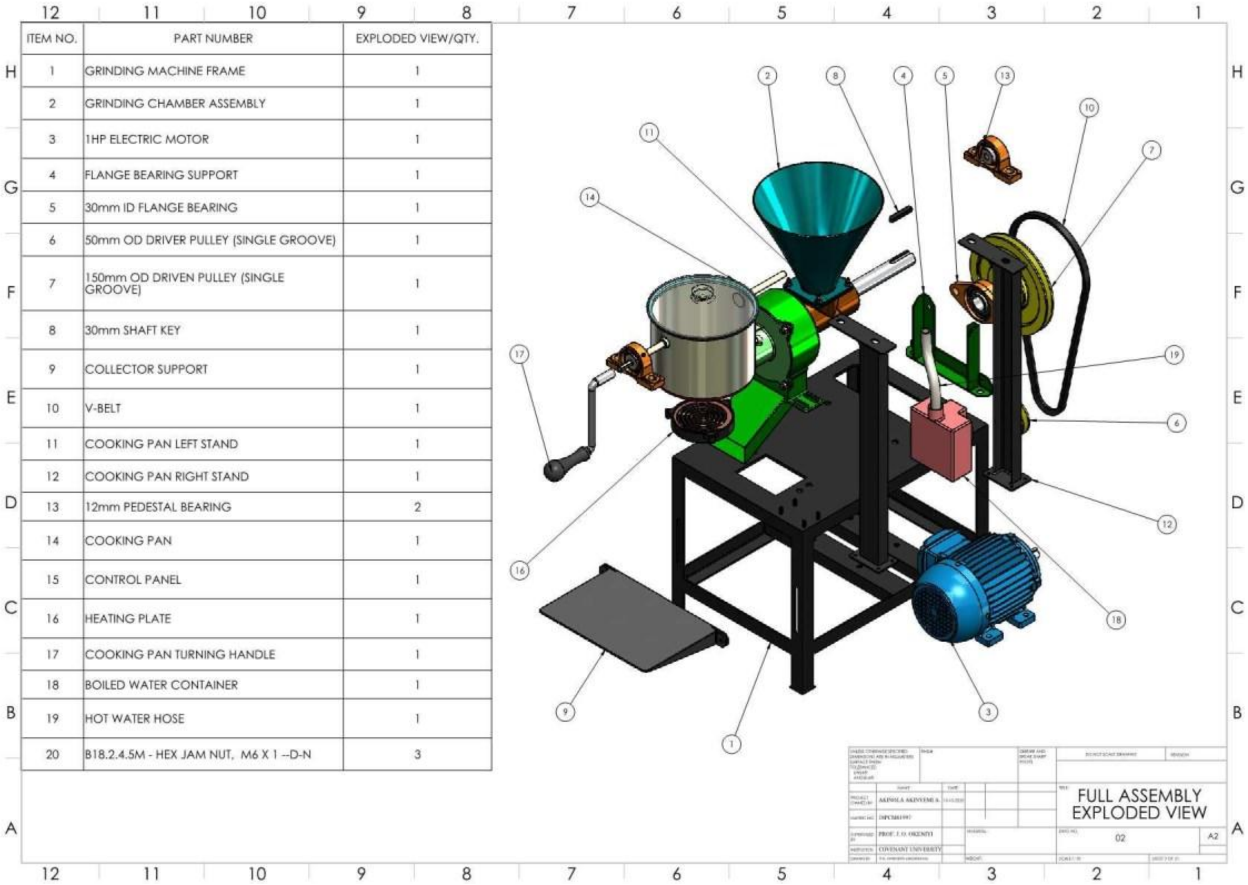


Figure 2.1 (a): Conceptual Machine design drawing.



ITEM NO.	PART NUMBER	EXPLODED VIEW/QTY.
1	GRINDING MACHINE FRAME	1
2	GRINDING CHAMBER ASSEMBLY	1
3	1HP ELECTRIC MOTOR	1
4	FLANGE BEARING SUPPORT	1
5	30mm ID FLANGE BEARING	1
6	50mm OD DRIVER PULLEY (SINGLE GROOVE)	1
7	150mm OD DRIVEN PULLEY (SINGLE GROOVE)	1
8	30mm SHAFT KEY	1
9	COLLECTOR SUPPORT	1
10	V-BELT	1
11	COOKING PAN LEFT STAND	1
12	COOKING PAN RIGHT STAND	1
13	12mm PEDESTAL BEARING	2
14	COOKING PAN	1
15	CONTROL PANEL	1
16	HEATING PLATE	1
17	COOKING PAN TURNING HANDLE	1
18	BOILED WATER CONTAINER	1
19	HOT WATER HOSE	1
20	B18.2.4.5M - HEX JAM NUT, M6 X 1 --D-N	3



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20	B18.2.4.5M - HEX JAM NUT, M6 X 1 --D-N	3

Figure 2.1(b): Exploded view of the machine design.

2.3 Design Analysis

2.3.1 Speed Ratio and Pulley Size.

Speed of motor pulley, N_1 = 1440rpm Speed of shaft pulley, N_2 =?

The diameter of the driver pulley, D_1 =

The diameter of driven pulley, D_2 = 150mm Using a velocity ratio of;

$$V.R. = \frac{N1}{N2} = \frac{D2}{D1} = \frac{3}{1}$$ ……………. (Khurmi & Gupta, 2005)

2.3.2 Power Required to Shear Boiled Plantain

To calculate the power required to grind boiled plantain, the shear force of plantain is used, that is;

Shearing force of plantain,

$\tau_p = \frac{F_p}{A}$ ……………………. (2)

Where;

τ_p = the shear stress of boiled plantain (N/m^2) F_p = force needed to grind boiled plantain (N) A_p = Area of plantain under shear (m^2) Hence from equation (2)

$F_p = \tau_p \times A_p$ …………………………. (3) But,

$A_p = \frac{\pi D^2}{4}$ ………………………………. (4) Where;

D = diameter of plantain

2.3.3 Velocity of Belt

Velocity V, = $\frac{\pi D_1 N_1}{60}$ ……………………… (5)

2.3.4 Co-Efficient of Friction

$\mu = \frac{0.5 - 42.6}{152.6 + V}$ ……………………………… (6)

2.3.5 Length of Belt

The length of the belt over the pulleys is calculated using:

$L = \pi (r_1 + r_2) + 2x + (\frac{r1+r2}{2})$ ………… (7)

2.3.6 Shaft Design

A shaft is a rotating device that is used to transmit power and motion from one member to another. The basis for the design of shafts is;

i Strength

ii. Rigidity and stiffness.

Conditions for shaft considerations are;

a) Shafts subjected to twisting moments only
b) Shafts subjected to bending moments only
c) Shafts subjected to axial loads, combined torsional and bending loads.

2.3.7 Minimum Diameter of Shaft

The shaft diameter was calculated using the torsion equation (Khurmi & Gupta, 2005), thus;

$\frac{\tau}{r} = \frac{T}{J} = \frac{G.\theta}{L}$ ……………………… (8)

Where;

G = Modulus of rigidity for the shaft material

L = length of the shaft

J = polar moment of inertia

τ = torsional shear stress

r = radius of shaft

θ = angle of twist in radians

3. Performance Evaluation

SolidWorks CAD was used to run the behavioral pattern of the design, and the results of the structural and displacement patterns of the machine were observed. Figures 3.1, 3.2, and 3.3 show the static, displacement, and strain simulations, respectively.

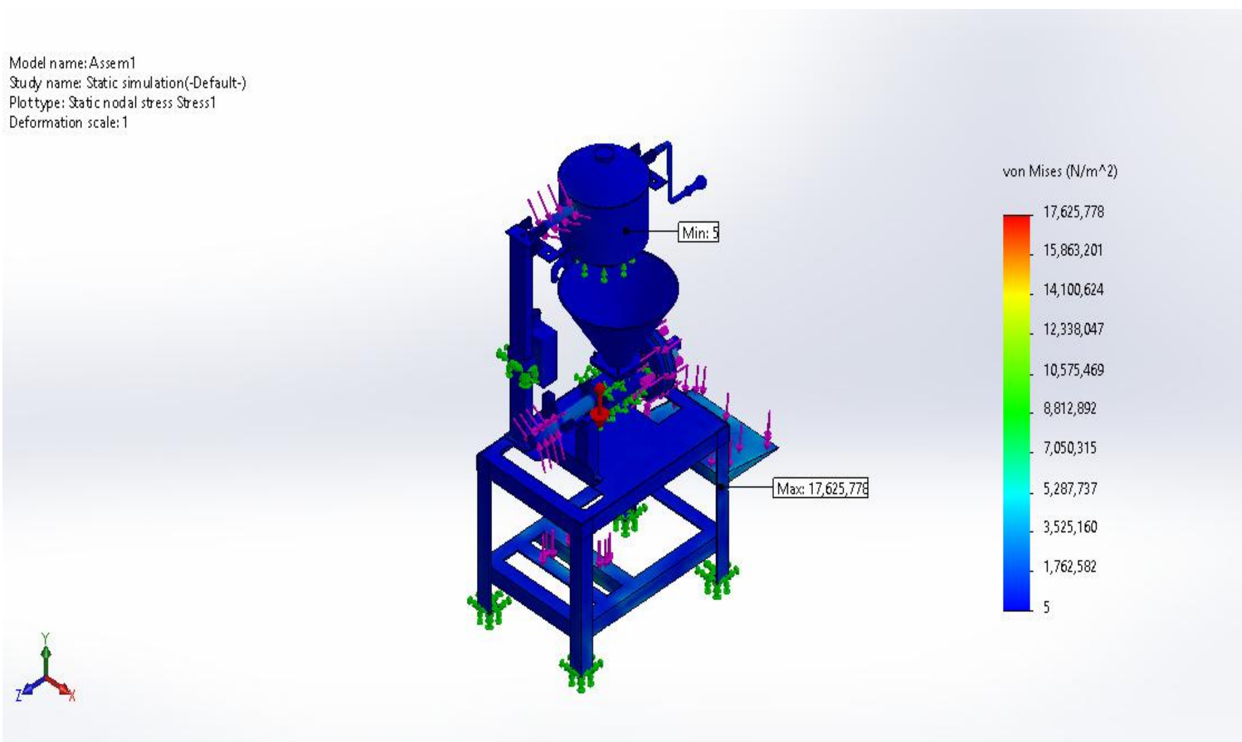

Fig. 3.1: Performance evaluation (Static Simulation) of the Machine design.

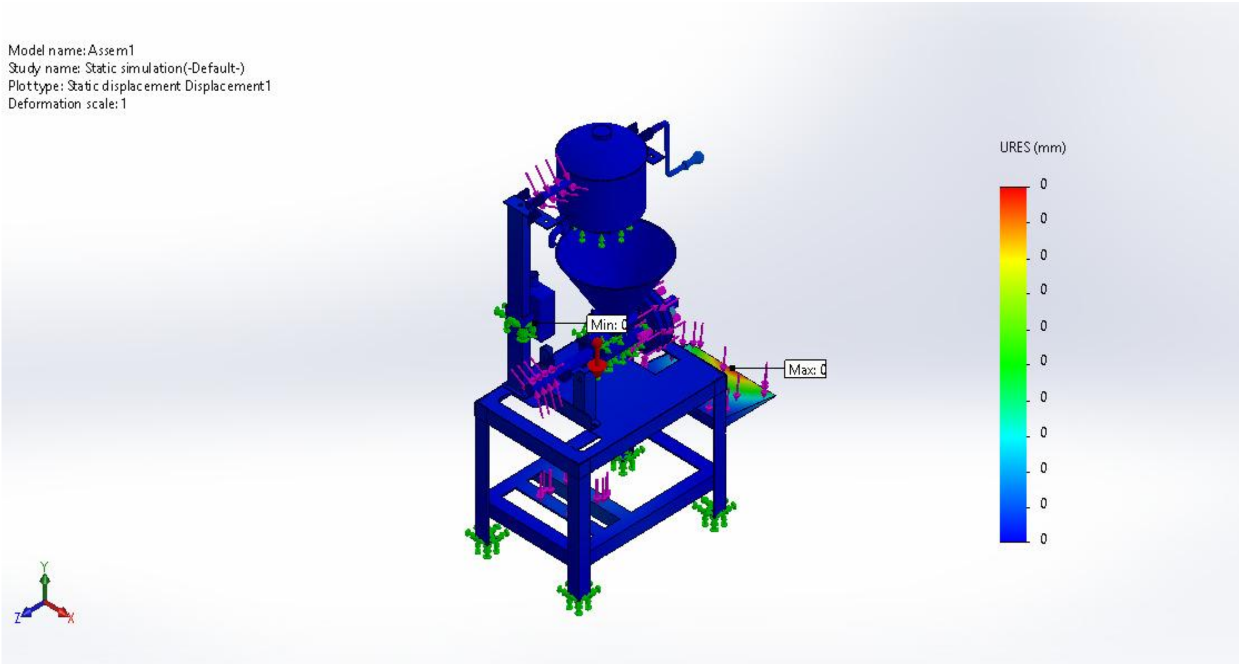

Fig. 3.2: Performance evaluation (Static Displacement Simulation) of the Machine design.

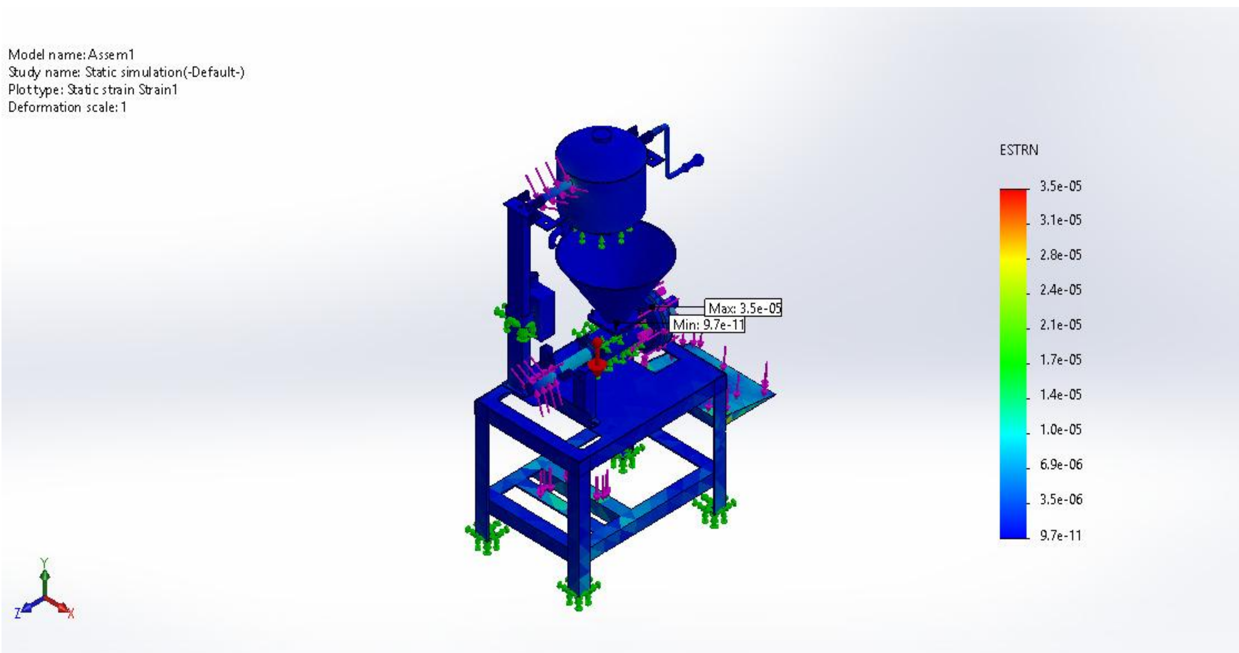

Fig. 3.3: Performance evaluation (Strain Simulation) of the Machine design.

4. Results and Discussions

The performance of the machine design was completed using Solid Works CAD, and the machine structural analysis was carried out using the finite element analysis of the same, hence evaluating the structural integrity under different loading (forces) conditions. Employing a solid curvature-based mesh of high-quality settings with 195,699 nodes and 102,854 elements. It was deduced that the mesh density was sufficient not only to accommodate but also capture the stress concentrations in critical regions.

The von Mises stress results indicated a maximum of 17.6MPa with an effectively negligible minimum stress. The peak stress occurs locally closer to constrained and loading regions, particularly near bearing supports and interfaces subjected to applied torque and normal forces. The maximum induced stress is considered significantly lower when compared to the material yield strength -220MPa for plain carbon steel and 172MPa for austenitic stainless steel. This indicates a factor of safety of greater than 9, confirming the satisfaction of a well-behaved result during operation within the elastic limit of the selected material with no yielding or risk of any form of deformation under the specified loading conditions.

The design result exhibits a relatively small displacement behavior with a maximum value of 0mm, which is indicative of rigid behavioral structure under operational loads, and this is desirable for maintaining alignment of rotating components, such as the screw conveyor (auger) shaft and the supporting bearing elements. Equivalently, the strain distribution follows the same trend as the stress results, with a maximum strain record of $3.5 \times 10^{-5,}$ suitably within the elastic strain range of the material used. However, the low strain levels further indicate that the design assembly is not subjected to any excessive deformation with localized stress concentrations and is non-critical.

The majority of the load is transferred through the main fixed supports and bearing connectors, as shown in the reaction force result. The highest reaction force is observed as approximately 1,961N, primarily in the vertical direction, aligning with the force of applied gravity and normal forces, all within acceptable limits, negligible bending moments, suggesting proper alignment and load sharing among the supports.

Overall, the combined effects of normal forces, gravity, applied torque, and thermal loading do not compromise the structural integrity of the assembly. Stress, strain, and displacement values remain well below critical thresholds, demonstrating the current design is mechanically sound and suitable for the intended operating conditions.

5. Summary

To summarize it all, the static simulation confirms that;

i. The maximum von Mises stress on the machine is extremely below the material yield limits.
ii. Displacements are negligible, indicative of structural stability.
iii. A purely elastic behavior is exhibited in the strain levels.
iv. Load paths via bearings and fixtures are well distributed.

6. Conclusions

The performance evaluation results indicate that the machine design is structurally safe, stable, and adequate for operation under the various loading conditions analyzed. The induced stresses, strains, and displacements remain well within allowable limits, and therefore, no immediate material substitution or geometric modification is required based on static strength considerations.

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