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Design and Development of a Mobile Grain Grinding Machine for Small-Scale Feed Production

NNAM Ikechukwu Onwuka[1], Dumpe Koba Barinem[2], Sampson, Nsimoh Michael[3]

[1,2,,] Department of Mechanical Engineering Akanu Ibiam Federal Polytechnic, Unwana, Afikpo Ebonyi State, Nigeria,
[3,] Department of Agricultural and Bio-Environmental Technology, Federal College of Agriculture Isiagu, Ivo L.G.A, Ebonyi State
[1]nionwuka@polyunwana.edu.ng, [2]dkb4life.bk@gmail.com, [3]Nsimoh@fcaishiagu.edu.ng

ABSTRACT

This study focuses on improving a grain grinding machine used for processing maize, millet, sorghum, and wheat to support small-scale livestock feed production. The aim was to overcome common issues such as low efficiency, limited mobility, excessive noise, and reduced throughput found in the existing equipment at the Akanu Ibiam Federal Polytechnic livestock farm. Using locally available engineering materials, several design modifications were made, including lowering the machine's height for better ergonomics, enlarging and tilting the hopper (40° angle of repose) to enhance feed flow, fitting rubber tires to aid movement and reduce vibration noise, changing the pulley ratio from 3:1 to 2:1 to boost speed, upgrading the engine from 1 hp to 4 hp for greater output, and minimizing the grinding chamber clearance for finer grinding. The redesigned machine was modeled in SolidWorks 2015 and fabricated using arc welding and bolted joints. A 4 hp petrol engine operating at 3000 rpm and delivering 559.48 Nm of torque powered the 38 mm diameter shaft, which rotated the grinding discs. The grinding process operates on a frictional principle, where grains fed through the hopper are crushed between a rotating toothed wheel (driven by the engine) and a fixed toothed plate until the desired fineness is achieved. The discharge is controlled via an adjustable outlet gate. Overall, the modifications significantly improved the grinder's ergonomics, efficiency, mobility, and usability, offering a reliable and adaptable solution for small-scale grain processing.

Keywords: Ergonomics, Mobility, Development, Grinding, Grain

1.0 INTRODUCTION

The traditional grain grinding process dates back to the Neolithic era, around 4,500 BC and earlier, when quern-stones were commonly used as household tools for grinding or milling grains and various food items. The grinding stone typically consists of two components: a stationary lower stone, known as the quern-stone, and a mobile upper stone, referred to as the hand stone. The earliest known devices for milling grains or flour are saddle stones (Williams and Rosentrater, 2007). A saddle stone is a cradle-shaped piece of hard stone designed to hold the grain. The hand stone could either be a cylindrical stone, gripped with both hands and moved across the grain like a rolling pin, or a disc-shaped stone with a vertical handle resembling an upside-down mushroom, held in one hand. These hand stones were used to crush grain and produce coarse flour (Thomas and Filippov, 1999).

An early method of grain grinding involved a device similar to a pestle and mortar, used to produce meal for human consumption (Culpin, 1992). The first mills were adaptations of this basic device, where grains were fed through an opening in a disc-shaped stone that rotated against another. Over thousands of years, these simple mills gradually evolved into the burr stone mill. Historical records of food production in Africa indicate that indigenous grains have been milled for centuries to produce coarse flour for cooking. Traditional crops such as sorghum, millet, and maize were ground using either a crude mortar and pestle made from a tree stump and branch or with flat stones or rubbing stones. These traditional grinding methods remain widely used across Africa today (Nwaigwe et al., 2012).

In 2006, Brain and Rottger noted that during the mid-nineteenth century, the invention of electric motors led to the development of faster machines, such as hammer and plate mills, which began to replace traditional stone grinders. For instance, a relatively low-speed, water-cooled diesel engine was used to power a hammer mill that produced high-quality maize flour. These mills are commonly used in rural regions where there is no electricity grid. However, diesel-powered grain mills are only feasible in areas with access to fuel and spare parts. Many people are unable to afford commercial grain-grinding services and resort to manual grinding using traditional methods. Consequently, pounding grains remains a familiar sight and sound in many regions. This activity, which is mainly carried out by women, is often social in nature and requires significant time and effort. The pestle can weigh as much as 4 kg, making the pounding task physically demanding (FAO, 1983).

Suresh et al. (2018) observed that fish weight fluctuates depending on the type of feed nutrients provided. These high-quality diets typically consist of protein-rich grains such as wheat, barley, and oats, with protein content ranging from 11–14%. While most grain grinding technologies today are geared towards large-scale feed and particle production, small-scale operators often face challenges in expanding their businesses. This situation highlights the pressing need for specialized small-scale grain grinding equipment (Adekomaya et al., 2014).

This work aims to solve the problems faced by small-scale operators in grinding grains for livestock feed by modifying and developing a grain grinding machine (for maize, millet, sorghum, and wheat) using locally sourced engineering materials. The objective is to address key challenges such as limited mobility, excessive noise, reduced operational efficiency, and low throughput issues associated with existing grain grinding machines at the institution's livestock feed production farm. By tackling these challenges through innovative design and rigorous testing, the upgraded machine is expected to deliver superior performance and adaptability.

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2.0 MATERIALS AND METHODS

A motorized grain grinding machine, which is efficient and economically viable, was modified and developed with available and cheap engineering materials that could give optimum service performance. The materials used in development the machine was chosen based on their availability, suitability, economic consideration, viability in service (Ugwu & Omoruyi, 2016). But the Selection of grinding plate was done based on four different grinding plates fell within the speed limit of the grinding machine. A233 is for medium fine for small grains, A434 is for medium fine for small grains of high capacity, B41 is for uniform coarse for small grains, and B43 is for extremely fine for small dry grains. Based on the description above, B43 was selected for the particular grinding process (Henderson & Perry 1986).

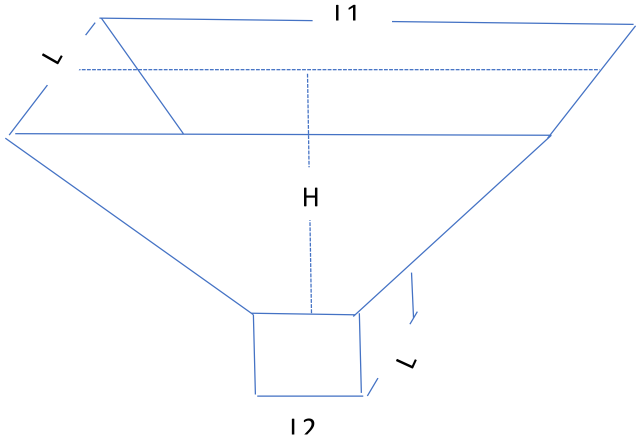
2.1 Modification of the machine

The height of the machine was reduced from 800mm mm to 420 mm to accommodate everybody in the operation of the machine. The reduction was to eliminate the stress and fatigue to be incurred in stretching or bending loading the materials into the hopper. A well-designed hopper was incorporated to avoid clogging of material during loading and prevent either splashing or flying back of materials from the hopper. The 3:1 pulley ratio which reduces grinding speed thereby limiting productivity was changed to 2:1, while the Petrol engine of 1hp (1450rpm) was changed to 4hp(3000rpm) these changes contributed to the high efficiency of the machine. The large clearance between the mechanism and grinding chamber was lowered to increase the machine’s efficiency. Four (4) rubber wheels were attached to the feets of the machine including a handle to take care of easy of movement and noise reduction.

2.2 Principle of Operation of the machine

The grinding machines have hopper, grinding chambers. The outlet, the frames. The grinding chamber houses the transmission shaft and the grinding wheels. A petrol engine of 4hp provides drive through belt connections to drive the pulley on the grinding chamber. The grains were introduced to the machine through the truncated hopper and were received by the grinding chamber where the grinding is done and after grinding collected through the outlet with a small bucket.

2.3 DESIGN CONSIDERATIONS



Volume of rectangle

$V = L_{RH} \times B_{RH} \times H_{RH}$

Since $B_{1RH} = B_{2RH}$, $L_{1RH} = L_{2RH}$

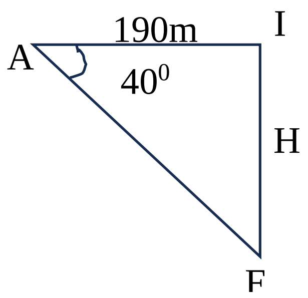
Total Volume of Hopper $= H\,\frac{(2L_1B_1 + L_1B_2 + L_2B_1 + 2L_1B_2)}{6} + L_{RH} \times B_{RH} \times H_{RH}$

Taking the angle of repose of grain, that ranges from $35^0 - 40^0$, but for the sake of this design 40^0 is to be used, similar to other agricultural seed into consideration to slant the hopper. 40^0 is used

$$\text{Tan}\Theta = \frac{opp}{adj} = = \frac{opp}{190}$$

$\text{Tan}40^0 = \frac{OPP}{190}$, $H_0 = 190\ \text{Tan}40^0 = 159.94^0$

+



$$H_T = 0.16\ \frac{(2x0.38x0.39 + 0.38x0.080 + 0.090x0.39 + 2x0.38x0.080)}{6} + 0.080 \times 0.090 \times 0.080$$

$$H_T = 0.7444m^3$$

2.3.2 Determination of Speed of Driven Pulley

Where N_1 is speed of driving pulley (3000rpm); D_1 is Diameter of driving pulley (Engine) (3´´ x 0.0245 = 0.0735m); D_2 is Diameter of driven pulley (Shaft) (6´´ = 6 x 0.0245 = 0.147m). then the speed of the driven pulley can be calculated using equation 2.0 (Zakaria, 2019).

$$\mathbf{N_m D_m = N_s D_s} \qquad \textbf{(2.0)}$$

$$3000x0.0735 = N_s x0.147 = 1500\text{rpm}$$

2.3.3 Determination of Belt Speed

The calculation of the V-belt circumferential speed is influenced by the diameter of the pulley and its rotation. Then the calculation can be done using Equation 3.0 (Innocent, 2023)

$$\mathbf{x = \frac{\pi N D_s}{60}} \qquad \textbf{(3.0)}$$

where x is the velocity of V-belt, N is the speed of driven pulley, D_2 is the diameter of driven pulley and π is a constant.

$$x = (3.142 \times 3000 \times 0.147)/60 = 11.55\text{m/s}$$

2.3.4 Determination of belt length

According to Khurmi and Gupta, (2005), the belt length is determined by equation 7.0

$$L_{vb} = 2P_{vb} + \frac{\pi}{2}(D_m + D_s) + \frac{(D_m - D_s)^2}{4P_{vb}} \quad \textbf{(4.0)}$$

And the center-to-center distance between driving pulley and driven pulley is $\mathbf{P_{vb}}$ calculated with equation 8.0 (Sharma and Aggarwal, 2006)

$$P_{vb} = \left(\frac{D_m + D_s}{2}\right) + D_s \quad (5.0)$$

$$P_{vb} = \left(\frac{0.147+0.0735}{2}\right) + 0.0735 = 0.18375\text{m}$$

$$L = 2(0.18375) + \frac{\pi}{2}(0.147 + 0.0735) + \frac{(0.147-0.0735)^2}{4(0.18375)} = 0.722\text{m} = 722\text{mm}$$

From **Table 1.0:** Standard pitch length of V-belt according IS: 2494-1974 (Yekinni et al, 2017)

Types of Belts	Standard Pitch Length of V-belt in mm
A	645, 696, 747, 823, 848, 925, 950,1001, 1026, 1051, 1102, 1128, 1204, 1255, 1331, 1433, 1458, 1509, 1560, 1636, 1661, 1687, 1763, 1814, 1941, 2017, 2068, 2093, 2195, 2322, 2474, 2703, 2880, 3084, 3287, 3693.
B	932, 1008, 1059, 1110, 1212, 1262, 1339, 1415, 1440, 1466, 1567, 1694, 1770, 1821, 1948, 2024, 2101, 2202, 2329, 2507, 2583, 2710, 2888, 3091, 3294, 3701, 4056, 4158, 4437, 4615, 4996, 5377.
C	1275, 1351, 1453, 1580, 1681, 1783, 1834, 1961, 2088, 2113, 2215, 2342, 2494, 2723, 2901, 3104, 3205, 3307, 3459, 3713, 4069, 4171,4450, 4628, 5009, 5390, 6101, 6863, 7625, 8387, 9149.
D	3127, 3330, 3736, 4092, 4194, 4473, 4651, 5032, 5413, 6124, 6886,7648, 8410, 9172, 9934, 10696, 12220, 13744, 15268, 16792.
E	5426, 6137, 6899, 7661, 8423, 9185, 9947, 10709, 12233, 13757, 15283, 16805.

From Table 1.0 the length of 722mm. so the nearest belt length of 747mm is selected, this length falls under type A belt.

2.3.5 Determination of the Wrapping angle (α)

Wrapping angle of a v-belt with the pulley is calculated using equation 6.0 (Oluwaseun et al, 2019)

$$\text{Sin}\alpha = \frac{D_m - D_s}{2\mathbf{P_{vb}}} \quad (6.0)$$

$$\text{Sin}\alpha = \frac{0.147-0.0735}{2(0.18375)} = 0.1,\ \alpha = \sin^{-1}(0.1) = 5.74^o$$

$$\text{Angle of } \theta = 180 - 2(\alpha);\ \theta = 180 - 2(5.74) = 168.53\text{deg or } \frac{\pi}{180} \text{ x } 168.53 = 2.94\text{rad.}$$

2.3.6 Determination of Belt Area

The data for the calculation of the cross-sectional area of the belt is obtainable from table 2.0. Check for belt A—IS:2494

Table 2.0 Dimensions of standard V-belts according to IS: 2494-1974 (Kurmi & Gupta, 2014)

Type of belt	Power ranges	Minimum pitch diameter of	Top width(b) mm	Thickness(t) mm	Weight per meter length in

	in Kw	pulley(D) mm			newton
A	0.7-3.5	75	13	8	1.06
B	2-15	125	17	11	1.89
C	7.7-75	200	22	14	3.43
D	20-150	355	32	19	5.96
E	30-350	500	38	23	-

The groove angle 2β is 32 from table 2. From trigonometry. Other two angles are 74° each.

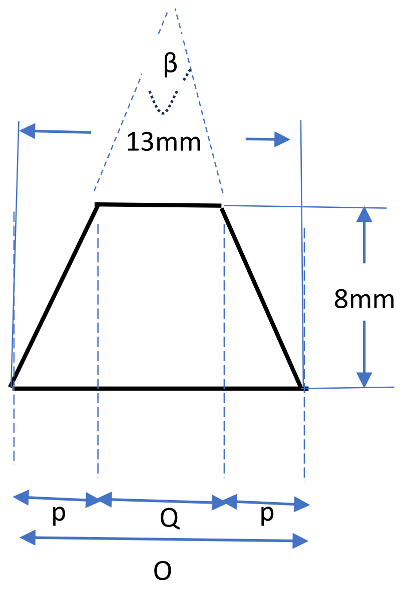


Fig.1.0. V-belt cross-sectional area (Putra, 2022).

Where O = 13mm from trigonometric ratio with α = 168.53deg, and R=8mm. P and Q is calculated with equation 10.0 and 11.0 respectively (Rahman et al., 2022). Therefore, belt cross-sectional area and mass of the belt is then calculated using equation 12.0 and 13.0 (Enyi, 2022).

$$\mathrm{Tan}\beta = \frac{\mathrm{Opp}}{\mathrm{Adj}} \quad (7.0)$$

$$Q = b - 2p \quad (8.0)$$

From calculation using equation 10.0, Q =8.41mm

The belt cross sectional area is obtained from the area of a trapezium

$$A = \left(\frac{O+Q}{2}\right)R \quad (9.0)$$

$$A = \left(\frac{13+8.41}{2}\right)x8 = 85.64mm^2 = 85.64x10^{-6}m^2$$

Mass of belt per meter length is calculated using equation 10.0 (Rahman et al, 2022)

$$M = AxLx\rho \quad (10.0)$$

Where Mass = Mass of Belt, A = Area of belt, L= length of belt, ρ = density of belt, which is 1140kg/m^3 for rubber belt.

$$85.64x10^{-6} \text{ x } 0.79183 \textbf{ x} 1140 = 0.0773kg$$

2.3.7 The maximum tension on the belt

The maximum tension on the belt is given by equation 11.0 (Tampubolon et al., 2022)

$$T_{max} = \sigma \text{ x } A \quad (11.0)$$

where, Ϭ is the stress in the belt which depends on the material that has been selected. For this design purpose is 7mpa and A=cross sectional area of belt

$$T = 7 \text{ x } 85.64mm^2 = 599.48N$$

Centrifugal tension is given by equation 12.0 ()

$$T_c = mx^2 \quad (12.0)$$

$$T_c = 0.0773kg \text{ x} 11.55^2 = 10.31N$$

2.3.8 Determination of Tension on belt.

The expression in equation 13.0 is used to calculate the tension on the tight side of the belt (Rahman et al, 2022)

$$T_1 = T_{max} - T_c \quad (13.0)$$

$$T_1 = 599.48N - 10.31N = 589.17N$$

tension in the stack side of the belt is found by the general formula of 14.0 (Kurmi & Gupta, 2005)

$$2.3\log\left(\frac{T_t}{T_s}\right) = \mu\theta \operatorname{cosec}\beta \quad (14.0)$$

$$2.3\log\left(\frac{T_t}{T_s}\right) = 0.3\text{x } 2.94 \text{ x cosec}16$$

$$\frac{T_t}{T_s} = \log\text{-}\frac{0.3\text{x}2.94\text{x}3.612}{2.3} = 24.26N,\ T_s = \frac{589.17}{24.26} = 24.29N$$

Therefore, power transmitted by each belt is calculated using equation 15.0 (Kurmi & Gupta, 2005)

$$P_{tb} = (T_t - T_s)x \quad (15.0)$$

$$P_{tb} = (589.17N - 24.29)\text{x}11.55 = 6.5kw$$

2.3.9 Determination of Safe Life Span of the Selected V-belt

To find out that the selected V-belt is safe to use, it can be calculated using V-belt tension to the move of the pulley on the grain grinder

$$T = T_t - T_s \quad (16.0)$$

T_t = 589.17N **T_s** = 24.29, Then T = 564.83Nm, with a value of 30.61N, it can be concluded that the belt is declared safe because the maximum tension of the belt is 599.48N

2.3.10 Determination of Bending and Twisting Moment

Bending Moment, M = ($T_t + T_s + 2T_C$)

M =589.17+ 24.29+ 2(10.31)= 634.08Mpa

$$T_e = \sqrt{(K_m M)^2 + (K_t T)^2}$$

$$\sqrt{(1.5 \text{ x}634.08)^2 + (1.0 \text{ x } 564.83)^2} = 1106.2Nm$$

2.3.11 Determination of Shaft Diameter

$$d = \sqrt[3]{\frac{16}{\pi S_s} T_e} \quad \text{where, } S_s = \text{Shear stress} \quad (18.0)$$

Taking allowable shear stress of steel (S_s) = 100MPa (Khurmi and Gupta, 2005)

$$1MPa = 1X10^6$$

$$d = \sqrt[3]{\left(\frac{16}{\pi x 100X10^6} \text{ x } 1106.2\right)} = 0.03826m = 38mm$$

using table of standard shaft diameters to select the appropriate shaft to be used

2.4 DESIGN OF TOOTH ON THE PLATE

These teeth are the main part of the grinding machine which is welded on rotational plate. The material used for these teeth must be high strength which is steel.4SC8-steel with yield strength of 350mpa and ultimate strength 610-730mpa.

Design analysis

Total power transmitted by the plate is 6.5kw and 1500rpm

$$\text{Angular velocity } (\omega) = \frac{2\pi N}{60} \quad (19.0)$$

$$= \frac{2 \text{ X } \pi \text{ X } 1500}{60} = 157.1\text{Rad}$$

Total torque exacted by the teeth is T= 564.83Nm, determined using equation 14.0

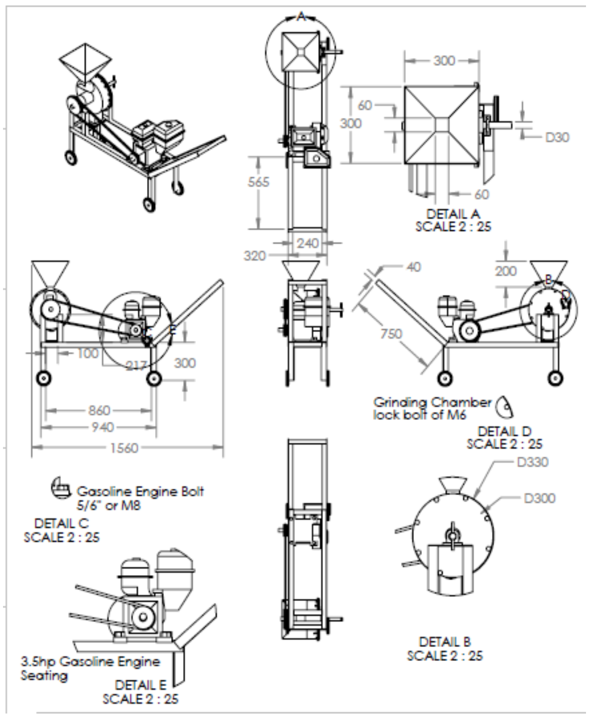
Number of teeth on the plate will be 18

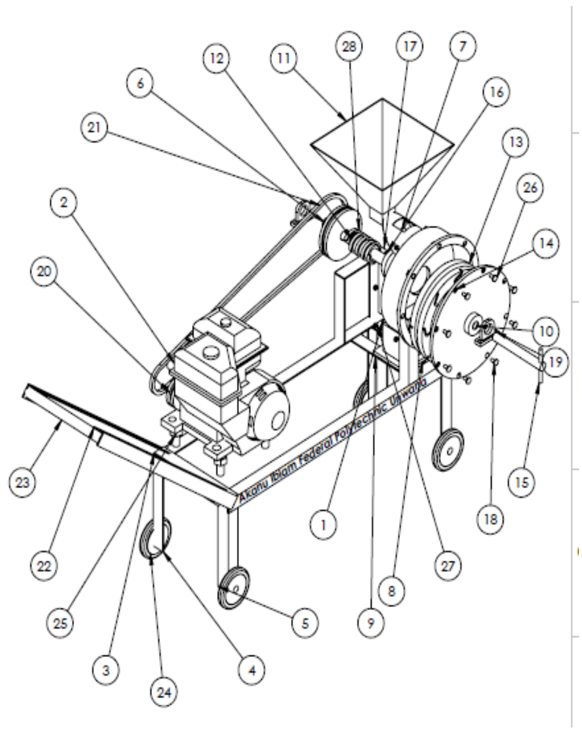
$$T_{tp} = \frac{P}{\omega} \quad (20.0)$$

$$T_{tp} = \frac{6.5 \text{ x}1000}{157.1} = 41.37\text{Nm}$$

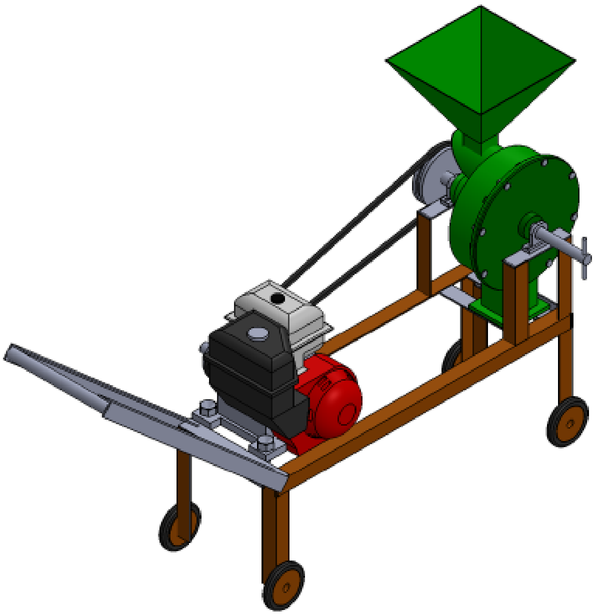
Torque transmitted for each tooth are calculated as: -

$$T_{each} = \frac{\text{total torque}}{\text{number of teeth}} = = \frac{41.37}{18} = 2.3\text{Nm}$$





ISOMETRIC VIEW OF A GRAIN GRINDING MACHINE



3.0 OPERATION PRINCIPLES OF THE MACHINE

This machine operates based on the frictional interaction between two toothed wheels that grind the grains by rubbing them together. One of the wheels is powered directly by the petrol engine’s output shaft. As grains are fed from the hopper into the grinding chamber, they are pressed and crushed between the rotating wheel and a fixed circular toothed wheel. The longer the grains remain in the chamber, the finer the resulting flour becomes, ensuring that all grains are completely ground according to the operator’s preference. The discharge of flour is regulated by a flap or rag attached to the outlet spout, which can be opened or closed as needed.

4.0 DISCUSSION

The grinding wheels were designed and constructed based on the specified design considerations. The machine functions as follows: A measured quantity of grains is poured into the hopper, from where it flows by gravity through the feed control gate into the grinding chamber. Once the prime mover is started, its output shaft directly drives the grinding wheel into rotation. The rotating wheel subjects the grains to centrifugal force, pressing them against the grinding rings, where they are finely ground through frictional action.

5.0 CONCLUSION

The modification and development of a mobile grain grinding machine using locally sourced materials were successfully carried out to address the limitations of existing grain milling equipment, particularly those used in small-scale livestock feed processing. The redesigned system significantly improved mobility, operational efficiency, and throughput capacity. Major design alterations including the reduction of machine height, modification of the hopper geometry to increase throughput and prevent clogging, adjustment of the pulley ratio from 3:1 to 2:1, and the upgrade of the prime mover from 1 hp to 4 hp collectively enhanced machine performance.

The resulting machine demonstrated stable operation, reduced noise and vibration through the incorporation of rubber wheels, and achieved finer and more uniform grinding. The machine's performance showed that the design is ergonomically efficient, easy to operate, and cost-effective for small and medium-scale agricultural applications. Overall, the project successfully achieved its goal of developing an efficient, durable, and portable grain grinding machine suitable for rural and semi-urban communities.

6.0 RECOMMENDATIONS

Future designs can adopt hardened alloy steel for the grinding teeth to increase wear resistance and prolong service life. Incorporating a protective guard around the grinding chamber and pulley system is recommended to prevent operator injury. To enhance flexibility, the machine could be redesigned to operate using multiple power sources such as electric motors or renewable energy (solar) in areas with fuel scarcity. Further studies should evaluate grinding efficiency for different grain types under varying speeds and feed rates to optimize the process parameters. Also additional vibration-damping materials or rubber isolators may be introduced to further reduce noise and enhance user comfort. The design should be standardized for mass production, focusing on modular construction for easy maintenance and component replacement. Finally, Extended field testing under real farm conditions is necessary to evaluate durability, maintenance needs, and long-term economic benefits.

Author's information

Nnam Ikechukwu Onwuka born on November 23rd 1985 is a Lecturer in the Department of Mechanical Engineering Technology at Akanu Ibiam Federal Polytechnic Unwana, Afikpo L.G.A of Ebonyi State, Nigeria, and have authored many articles in both Local and international journals. Engr. Nnam is a Ph.D scholar in Design and Production Engineering; he is a registered Engineer in Nigeria and a member of many professional bodies.

ENGR. Barinem Koba Dumpe born on 22nd September, 1974. Holds a Masters degree in Mechanical engineering (Production Option), Currently Lecturer at Akanu Ibiam Federal Polytechnic, unwana, Afikpo, Ebonyi State. Has several publications in journal's , an ardent scholar, married with children.

Sampson, Nsimoh Michael born on September 16th 1982. He is a lecturer in the Department of Agricultural and Bio-Environmental Engineering Technology at Federal College of Agriculture Ishiagu Ivo L.G.A Ebonyi state, Nigeria, and have authored many articles in both local and International Journals. Engr. Nsimoh is a Ph.D scholar in Design and Production Engineering; he is a registered Council member of the Council for the Regulation of Engineering in Nigeria (COREN) and a certified Engineering Failure and Forensic investigator (EFFI) in Nigeria.

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