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Development and Performance Evaluation of a Coffee Cherry Size Grading Machine

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ABSTRACT

The coffee cherry processing industry traditionally relies on the dry method, involving harvesting, drying, and dehulling. Conventional dehulling machines with fixed drum clearances struggle to handle natural size variations, leading to inefficiencies and increased cherry breakage. To address these challenges, an innovative coffee cherry-size grading machine has been developed, utilizing an inclined oscillating sieve technique powered by a diesel engine through a belt drive. Key components include a feeding hopper, reciprocating grading sieve assembly, support frame, and power transmission system. A comprehensive performance evaluation focused on grading efficiency and capacity, exploring various operational parameters: feed rates (5, 10, 15 kg min^{-1}), sieve angles (7°, 9°, 11°), and speeds (80, 140, 200 rpm). Using a split-split-plot block design for data analysis, the study yielded promising results. Maximum grading efficiency of 88.40% was achieved at 15 kg min^{-1} feed rate, 8 rpm speed, and 7° sieve angle, with a capacity of 137.11 kg h^{-1} and 4.96% sieve clogging rate. ANOVA revealed significant influences of operational speed and sieve inclination angle on performance parameters. These findings offer valuable insights for optimizing coffee cherry processing, potentially enhancing efficiency and quality in the industry.

Keywords: *Coffee, Dehulling, Eccentric, Grading, Oscillating sieve, Machine*

INTRODUCTION

The nonalcoholic stimulating beverage crop, Coffee Arabica, belongs to the Rubiaceae family and the genus Coffea (Alemnew, 2020). This coffee is the backbone of the Ethiopian economy, accounting for 25-30% and 10% of the total foreign currency earnings and the government revenues of the country, respectively (Adem, 2023). Approximately 200 thousand tons are shipped to the central market in 2023, with the remaining consumed locally (Gofe Balami *et al.,* 2024).

During the 2018-19 main production season coffee covered 764863.16 ha of land in Ethiopia and yielded almost 500,000 tons, with an average productivity of 0.64 tons per hectare. South Nation Nationalities and Peoples Regional State contributed 30% of the total production (CSA, 2019).

Coffee quality may be impacted by a variety of factors, including gene type, climate, soil characteristics, and agricultural practices, time of harvest, processing following harvest, grading, packaging, storage, and transporting conditions (Adugna, 2020). Among these, processing is critical in determining the quality of coffee. Wet or dry methods can be used to prepare coffee, and the intricacy and desired level of quality can vary (Freitas *et al.,* 2024). Ethiopia uses both sun-drying and wet-processing methods, accounting for 70% and 30% of the country's coffee production, respectively.

The simplest, most affordable, and oldest method is the dry method. It is primarily popular in Brazil and Africa and yields "natural" coffee (Godwill, 2013). The picked cherries are first cleared of dirt, twigs, and leaves, as well as overripe, damaged, and unripe cherries. This is done manually with a large sieve. Unnecessary berries or other materials that have not been sprayed are collected from the top of the sieve.

Grading is an agricultural processing operation that is undertaken to categorize grains into different grades based on the desired quality parameters for storage, seed preparation, and commercialization or further processing (Yayock and Ishaya, 2020). An essential value-adding method that raises the market value and processing standards is grading agricultural products according to their size (Umani and Markson, 2020). Hand grading is costly, and personnel shortages make the process difficult during peak seasons. Human operations can be laborious, inconsistent, and inefficient at times. Farmers are hopeful that appropriate agricultural product grading machinery can assist in alleviating manpower shortages, save time, and enhance the overall quality of items that have been rated (Mostafa, 2003).

This study addressed the issue at hand and improved the technology used in the processing of micro to medium-sized coffee processing, benefiting all parties involved in the industry like farmers by increasing their income by value adding to their product and by decreasing the breakage losses that occurs during hulling operation. The objective to determine the physical properties of Coffee cherries, to design, manufacture/fabricate and assemble the machine and to test and evaluate the performance of the machine (study the effect of sieve inclination angle, speed of oscillation of the sieve and feeding rate on capacity and efficiency of the machine).

MATERIALS and METHODS

Description of the Study Area

The prototype was manufactured at the Jimma Agricultural Engineering Research Centre workshop. It lies at latitude of 7°40'51.5"N and a longitude of 36°50'38.1"E, and an assessment of the machine's performance was carried out at Goma Woreda.

Determination of Coffee Cherry Properties

The key physical and mechanical properties that influence the design of the grading machine like size, shape, density, friction coefficients, and angle of repose were experimentally determined for coffee Arabica cherries. These are described below:

Geometric mean diameter

Equation (1) can be used to determine the geometric mean diameter (D_g) (Garnayak, 2008).

$$D_g = \sqrt[3]{(LWT)}\ (\text{cm}) \tag{1}$$

Where; L: length, cm,

W: width, cm,

T: thickness, cm.

Arithmetic mean diameter

The arithmetic mean diameter (D_a) of the bean was calculated using Equation (2) (Garnayak, 2008).

$$D_a = (LWT)/3,\ (\text{cm}) \tag{2}$$

Aspect ratio

The aspect ratio (R_a) of the bean can be calculated using Equation (3) (Bizimungu *et al.,* 2022).

$$R_a = \frac{W}{L},(\text{cm}) \tag{3}$$

Size and shape

The three perpendicular dimensions of 100 randomly selected cherries were measured using digital calipers. The length, width, and thickness gave intermediate (*b*), major (*a*), and minor (c) diameters respectively.

Sphericity was calculated using the formula:

$$\varphi = \frac{\sqrt[3]{abc}}{a} \tag{4}$$

The shape index was determined as:

$$Shape\ Index = \frac{W}{\sqrt{LT}} \tag{5}$$

Bulk density

This indicates the mass per unit volume of the coffee cherries useful for hopper sizing. By adding a sample mass of cherries to a container with a known capacity, it became known,

$$\rho b = \frac{Ms}{Vc} \ ,(\frac{kg}{m^3}) \tag{6}$$

Where; ρb: bulk density,

Ms: total mass of bean in the container, kg,

Vc: volume of the container, m^3.

Angle of repose

The angle of repose of the coffee bean was measured by emptying method by using an open-ended cylinder of 15 cm diameter and 30 cm height. The cylinder was placed at the center of a circular plate having a diameter of 70 cm and filled with coffee cherries. The cylinder was raised slowly until it formed a cone on the circular plate. The angle of repose was calculated using Equation (7) (Karababa, 2006),

$$\theta = tan^{-1}\frac{2H}{d} \tag{7}$$

Where; θ: Angle of repose, empty or filling (deg.),

H: Height of the cone (cm),

d: Diameter of the cone (cm).

Moisture content

A small sample was oven dried and the moisture percentage determined by using Equation (8),

$$MC(\%) = \frac{(w-d)}{w} * 100 \tag{8}$$

Where; MC: moisture content,

w: weight before drying,

d: weight after drying.

Overall Structure and Design of the Machine

Design consideration

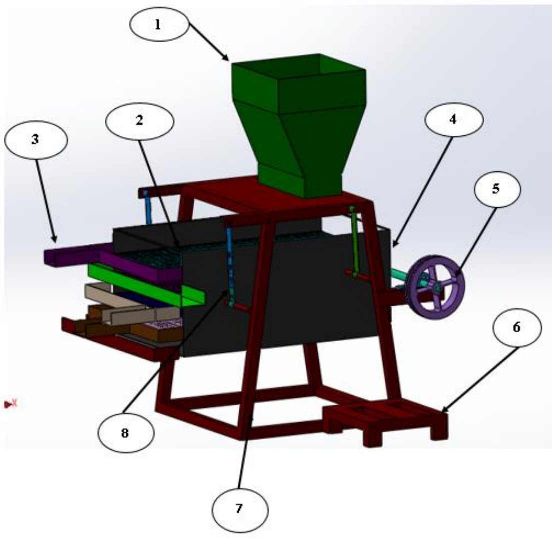
When designing the machine, it is crucial to evaluate the available alternatives based on several key criteria. The ease of assembly and handling should be a primary consideration, ensuring that the machine can be put together and managed without excessive difficulty. Additionally, the design should prioritize low maintenance requirements and minimal running costs, making it economically viable for long-term use.

Efficiency and durability are equally important factors, as they directly impact the machine's performance and lifespan. The design should also emphasize ease of operation, allowing users to interact with the machine intuitively and effectively. Lastly, market availability and suitability for operation in rural areas must be taken into account, ensuring that the machine is accessible and practical for its intended users and environments.

Overall structure of the machine

As indicated in Figure 1, the machine consists of the following parts: (i) a feeding hopper; (ii) a grading unit; (iii) a power transmission unit; and (iv) a supporting frame.

Description and design of machine components



Part no	Part name
1	Hopper
2	Sieve
3	Outlet chute
4	Eccentric
5	Pulley
6	Engine seat
7	Frame
8	Angle adjuster

Figure 1. *Pictorial view of machine.*

Hopper design

This machine is designed to accommodate 20 kg of coffee cherries at a time. According to Tafa and Olaniyan (2023), the actual capacity was estimated in accordance with the requirement that the hopper's capacity be 1% higher than the anticipated capacity.
Then,

$$V_h = 1.1\, V_E \tag{9}$$

$$V_E = \frac{w_C}{\rho_d} \tag{10}$$

$$= \frac{20\ kg}{650\ kg\ m^{-3}};\ V_E = 0.03\ m^3$$

Then, V_h=1.1*0.03= 0.033 m^3 this volume of the hopper is needed for our machine. The form of the hopper is trapezoidal. To limit the flow of coffee to the grading equipment, a feed control mechanism was built into the bottom of the hopper. Equation (11) was used to calculate the design hopper's overall volume.

$$V = \frac{1}{2}(a+b) \times h\ \times L \tag{11}$$

$V = \frac{1}{2}(0.4 + 0.3) \times 0.35 \times 0.5$; $V = 0.07\ m^3$

Where; V_h: Actual volume of hopper

V_E: estimated volume of hopper,

V: volume of hopper, (m^3),

h: height of the hopper, (0.35 m),

a: base width (0.3 m),

b: top width (0.4 m),

L: Length of the hopper (0.55 m).

Frame design

The material used for the frame was mild steel which has a Modulus of Elasticity = 210 GPa, and yield strength, σy = 300 MPa. The ability of a frame structure to support a given load without a sudden change in configuration was assessed by determining its crushing stress and criticality using the equations below (Khurmi and Gupta, 2005).

$$\sigma_{cr} = \frac{\pi^2 E}{(\frac{Le}{r})^2} \tag{12}$$

$$r^2 = \frac{I}{A} \tag{13}$$

$$P_{cr} = \frac{\pi^2 EI}{{L_e}^2} \tag{14}$$

By using the standard yield strength and comparing it with a critical load on the frame, we can decide whether the frame is safe or not. Let's determine the specification of the materials utilized for the frame based on the information at hand:

Modulus of elasticity, E = 210 GPa = 210 * 103 MPa,

Yield strength, σy = 300 MPa

The actual length of the frame, L = 1 110 mm,

For the frame fixed at both ends equivalent length, Le was calculated as,

For the frame fixed at both ends equivalent length, Le was calculated as,

$$Le = \frac{L}{2} \tag{15}$$

$$= \frac{1\,100}{2} = 550 \text{ mm}$$

The ratio of equivalent length to the radius of gyration can be determined by,

$$\frac{Le}{r} = \pi\sqrt{\frac{E}{\sigma_y}} \tag{16}$$

$$= \pi\sqrt{\frac{210 * 10^3 MPa}{300\ MPa}} = 83.12$$

The crushing stress is given by, $\sigma_{cr} = \frac{\pi^2 E}{\left(\frac{L_e}{r}\right)^2}$ (17)

$= \frac{\pi^2 * 210 * 10^3 MPa}{83.12^2}$ = 299.99 MPa= 300 MPa

The aforementioned outcome indicates the yield strength of the material used to create the frame is equal to the amount of crushing stress. From Equation (16), a radius of gyration can be calculated as:

$\frac{L_e}{r} = 83.12$, Le= 625 mm r= $\frac{L_e}{83.12} = \frac{550\ mm}{83.12}$ = 6.62 mm and r^2= $(6.62\ mm)^2$= 43.80 mm^2

Also, from the radius of gyration can be calculated the breadth or width of the mild steel pipe material used for the frame as follows:

$r^2 = \frac{I}{A}$ (18)

$= \frac{bh^3}{12hb} = \frac{h^2}{12}$, then $h^2 = 12 * r^2$ = 12 * 43.80 mm^2 = 525.6 mm^2

h = 24.04mm, for standardizing let take h = 25 mm.

Assuming thickness = 3mm and depth = 50mm.

The critical load can be determined as:

$P_{cr} = \frac{\pi^2 EI}{{L_e}^2}$ (19)

$= \frac{\pi^2 * 210\ (\frac{bh^3}{12})}{{L_e}^2} = \frac{\pi^2 * 210 * 10^{11}\ (\frac{25^3 * 50}{12})}{525^2}$ = 3454.36 $*10^8$ kN

Hence, critical stress was computed using:

$\sigma_{cr} = \frac{P_{cr}}{A}$ (20)

$= \frac{P_{cr}}{b * h} = \frac{3{,}454.36 * 10^8}{50mm * 25\ mm} = 276.34\ Mpa$

Where; σ_{cr}: Crushing stress (MPa), *r:* Radius of gyration (mm), *Le:* Frame equivalent length, (mm), P_{cr}: The critical load on the frame (kN), *I:* Polar moment of area for a hollow square shape (mm^4), *A*: Cross-section area for a hollow square shape (mm^2).

Therefore, when compared to the material's critical stress with yield strength, the estimated critical stress was less than the latter ($\sigma_{cr} < \sigma_y$). According to Euler's theory of buckling, the critical buckling stress for thin columns is frequently lower than the yield stress. As a result, the designed proportions of the frame materials are safe.

Design of the grading unit

The shape of the sieve hole is selected based on the shape index of coffee cherries which is the spherical shape and circular shape was selected since the shape index of the coffee cherries is less than 1.5 and considered as spherical shape. The size of the sieve was selected based on the result of the physical properties of the coffee bean and the diameter of the holes was chosen to be 12 mm, 10 mm, and 8 mm based on the physical characteristics of coffee cherries.

Design of Power Transmission Unit

Shaft selection

For a solid shaft with little or no axial load, the diameter of the shaft was determined using Equation (21) (Khurmi and Gupta, 2005).

$$d^3 = \frac{16Te}{\pi\tau}[(KbMb)^2 + (KtMt)^2]^{1/2} \quad (21)$$

Where: *d*: diameter of the shaft, mm,

Mt: torsional moment, N m,

Mb: bending moment, N m,

Kb: combined shock and fatigue factor applied to bending moment,

Te: equivalent twisting moment, N m,

Kt: combined shock and fatigue factor applied to torsional moment,

τ: maximum allowable shear stress, N m^{-1},

The vertical force acting on the shaft is shown in Figure 2 below.

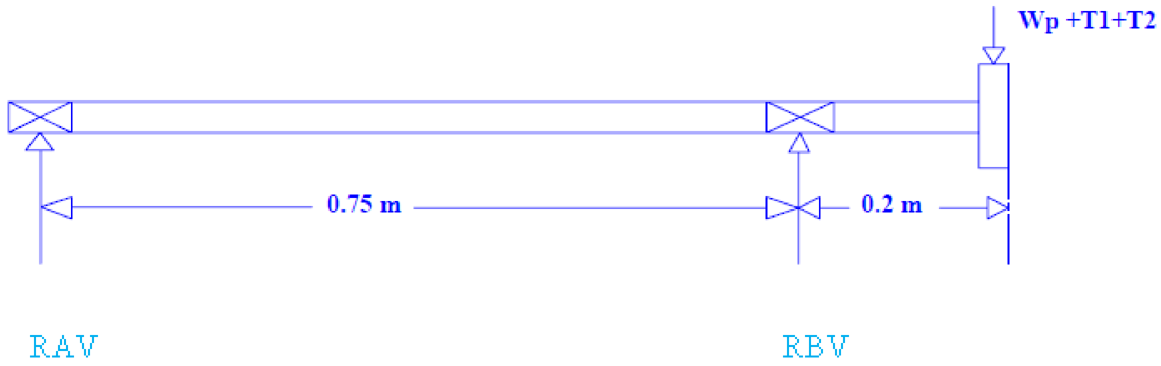


Figure 2. *Free body diagram of force acting vertically on shaft.*

Then we can find the moment at point A assuming that, the summation of the moment at point A is zero.

$$\sum MA = 0$$

$$RBV(0.75) - (WP + T1 + T2)(0.95) = 0$$

= RBV (0.75) - (316.87) (0.95) = RBV (0.75)-301; RBV (0.75) = 301 N

RBV= 401.37 N

Then, calculating for moment at B assuming that the summation of moment at B is zero.

$$\sum MB = 0$$

$RAV(0.75) - (WP + T1 + T2)(0.2) = 0$

= RAV (0.75) - (316.87) (0.2) = RAV (0.75)-163.37; RAV (0.75) = 163.37

RAV= 84.49 N

Then, we can calculate Maximum bending moment:

BM at AC = RAV (0.2) = 401.37(0.2) N; BM= 80.27 N

Horizontal force acting on the shaft was showed in Figure 3 as below.

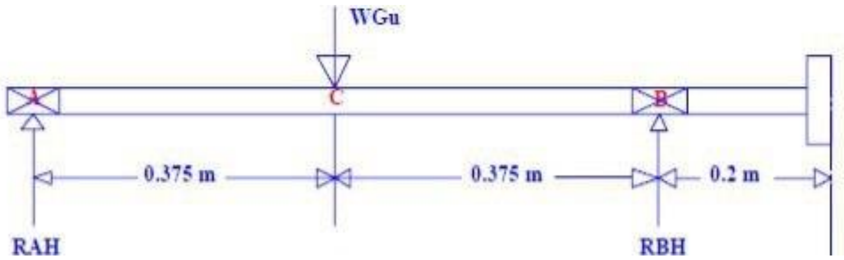


Figure 3. *Force acting horizontally on shaft.*

Then we can find the moment at point A assuming that, the summation of the moment at A is zero.

$\sum MA = 0$

$RBH(0.75) - (WGU)(0.375) = 0$

= RBH (0.75) - (566.31) (0.375) = RBH (0.75)-212.37; RBH (0.75) = 212.37

RBH= 283.16 N

$\sum MB = 0$

$RAH(0.75) - (WGU)(0.375) = 0$

= RAH (0.75) - (566.31) (0.375) = RAH (0.75)-212.37; RAH (0.75) = 212.37

RAH= 283.16 N

The maximum bending moment on the shaft was calculated:

BM= (283.16) (0.375) = 106.185 N m

The shear force and bending moment diagram for the shaft was shown in Figure 4 below.

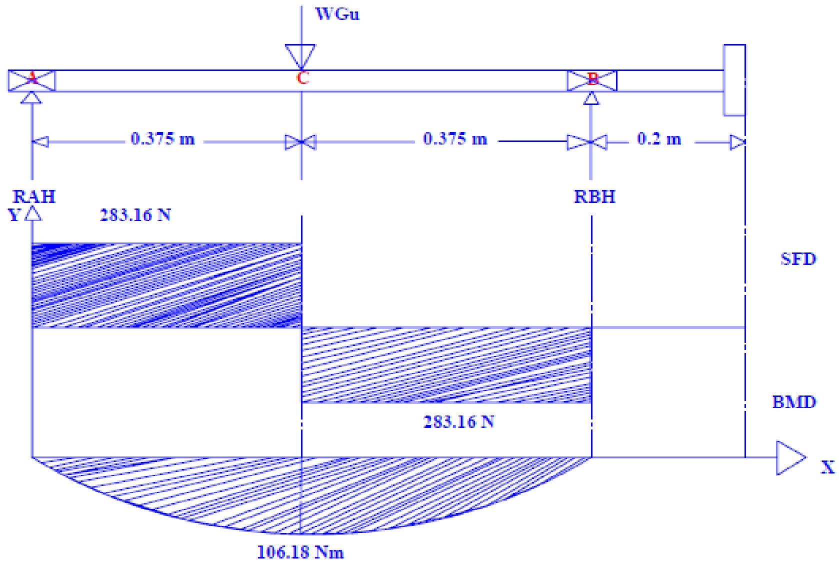


Figure 4. *Shear force and bending moment of force acting horizontally on shaft.*

The resultant bending moments on the shaft can be calculated using Equation 22.

$$M_b = \sqrt{(M_V)^2 + (M_H)^2} \tag{22}$$

$$= \sqrt{(80.27)^2 + (106.19)^2} = \sqrt{6443.91 + 11275.67} = \sqrt{17719} = 133.15\ N$$

Where; M_b: Resultant bending moments (N),
M_V: Maximum bending moments on a vertical (N),
M_H: Maximum bending moments on the horizontal (N).

Then we can calculate the diameter of the shaft by using Equation (20) as follow.

$$d^3 = \frac{16}{\pi\tau}\sqrt{(KbMb)^2 + (KtMt)^2}$$
$$= \frac{16}{3.14 * 45}\sqrt{(1.3 * 133.15 * 1000)^2 + (1.2 * 24.53 * 1000^2}$$

$= 0.11\sqrt{30828357121}$= 0.11*175580 $d^3 = 19313$

D= 26.83 mm, we can use a 30 mm diameter shaft (Figure 14).

Belt selection

In this design open type v-belt was used. According to Khurmi and Gupta (2005), the nominal pitch length (*L*) can be determined using Equation (23).

$$L = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_2+D_1)^2}{4C} \tag{23}$$

Where; D_1: Diameter of smaller pulley (mm),
D_2: Diameter of larger pulley (mm),

C: The center distance between the grading cylinder shaft pulley and the motor pulley (mm).

The least center distance of two pulleys, one acting as the driver and the other as the follower, can be calculated by using Equation (24) (Khurmi and Gupta, 2005).

$$C = \left(\frac{D_2+D_1}{2}\right) + D_1 \tag{24}$$

$$= \left(\frac{0.25+0.07}{2}\right) + 0.07 = 0.23\ m$$

The belt length is computed by using Equation (25):

$$L = 2C+\frac{\pi}{2}(D_1 + D_2) + \frac{(D_2+D_1)^2}{4C} \tag{25}$$

$$= 2 * 0.23+\frac{\pi}{2}(0.07 + 0.025) + \frac{(0.07+0.25)^2}{4*0.23}$$

=1.24 + 1.57 (0.32) + 0.11= 1.24 +0.5 +0.11= 1.85 m = 72.83 inch

Determination of belt tension

Equations (26) and (27) were used to calculate the belt's peripheral velocities (V_1) on the driver pulley, and (V_2) on the driven pulley (Khurmi and Gupta, 2005).

$$V_1 = \frac{\pi D_1 N_1}{60}(1 - \frac{S}{100}) \tag{26}$$

$$V_2 = \frac{\pi D_2 N_2}{60}(1 - \frac{S}{100}) \tag{27}$$

Where: V_1: The belt's peripheral velocity on the driver pulley (m.s^{-1}), V_2: The belt's peripheral velocity on the driven pulley (m.s^{-1}), N_1: Speed of the driver (rpm), N_2: Speed of the driven (rpm), S: Slip (%) (1° to 2°) (KhurmiandGupta2005).

So, $V_1 = \frac{3.14*0.07*450}{60}(1 - \frac{0.015}{100})$= 1.63 m s^{-1}

$V_2 = \frac{3.14*0.25*140}{60}(1 - \frac{0.015}{100})$=1.8 m s^{-1}

The following formula was used to calculate the mass of the belt:

$m = b * t * l * \rho$= 0.22 kg

Where; b: Top width of belt (0.013), M: Mass of the belt, t: Thickness of belt (0.008), l: Length of the belt, ρ: $Density\ of\ the\ belt$ $(\frac{1140kg}{m3})$

The tension on the open belt's two sides was determined using the relationship below (Khurmi and Gupta, 2005).

$$\frac{T_1-mv^2}{T_2-mv^2} = e^{\frac{\mu\theta}{sin\alpha}} \tag{28}$$

Where; m: Mass of the belt (kg m^{-1}), σ: Maximum allowable stress of the belt (MPa), T_1: Belt tension on the tight side (N), T_2: Belt tension on the slack side (N), μ: Friction coefficient between the pulley and the belt (0.43), θ: lap angle on smaller pulley (rad) and groove angle (34°).

Equation 28, can be used to determine the tension on the tight side of a power transmission belt (Khurmi and Gupta, 2005),

$$T_1 = T_{max} - T_C \quad (29)$$

Where, T_1: Tension in the tight side (N), T_C *and* T_{max}: The centrifugal and maximum tension of the belt (N)

The belt's maximum tension (T_{max}), the centrifugal tension (T_C), the belt's bulk relative to its length, and the belt's cross-sectional area are determined by using Equation 30, 31, and 32 (Khurmi and Gupta, 2005) as follow.

$$T_{max} = A\sigma \quad (30)$$

$$T_c = mv^2 \quad (31)$$

$$A = \frac{(b-x)}{2}t + xt \quad (32)$$

Then, A=$\frac{(0.013-0.011)}{2} * 0.008 + (0.011 * 0.008)$= 96 mm^2

Then, $T_c = mv^2$= 0.55N

$T_{max} = A\sigma$= 96 mm^2 * 2.5N mm^{-2} = 240 N

As a result, the tight side of the belt's tension has been determined by

$$T_1 = T_{max} - T_C \quad (33)$$

$$= (240 - 0.55)N = 239.45\ N$$

Where; A: Belt's cross-sectional area (mm^2), m: Mass per unit length of a belt (kg m^{-1}), σ: Maximum allowable stress of belt (2500N m^{-2} = 2.5N mm^{-2}), v: belt speed (mm.s^{-1}), b: Top width of the belt (mm) x: Bottom width of the belt (mm), t: belt thickness (mm) and ρ: Density of Rubber (kg m^{-3}).

The tension on the slack side of the belt,

$$ln(\frac{T_1}{T_2}) = \mu * \theta_1 + cosec\beta \quad (34)$$

$$ln(\frac{239.45\ N}{T_2}) = 0.25 * 2.12 + cosec\ 18$$

$$=\frac{239.45}{T_2} = e^{1.715} = \frac{239.45}{T_2} = 5.557T_2 = 43.089\ N$$

According to Khurmi and Gupta (2005), torque on a shaft can be determined using Equation (35).

$$T = (T_1 + T_2) * \frac{D_4}{2} \quad (35)$$

$$= (239.45\ N + 43.09\ N) * \frac{0.25\ m}{2} = 24.54\ \text{Nm}$$

The power required to rotate the main shaft was determined by using the Equation (36).

$$P = \frac{2\pi NT}{60} \quad (36)$$

$= \frac{2*3.14*80*24.54\ Nm}{60}$= 205.48 watts

Where; *P:* Power required to rotate the main shaft, *T:* Torque (Nm), *N:* Rotational speed (rpm), D_4: Diameter of the driven pulley, T_1: Tight side tension of the belt, T_2: Tension slack side of the belt.

Pulley selection

The grading unit's main shaft speed and the prime mover-driven pulley's speed are computed using Equation (37) (Khurmi and Gupta, 2005).

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \quad (37)$$

In this design, the pulley is needed to reduce the engine speed which has the highest speed of 700 rpm, medium speed of 500 rpm, and 300 rpm lower speed. The pulleys needed to reduce from the highest engine speed 700 rpm to 200 rpm of shaft speed and the other speed was adjusted by the speed adjuster on the engine. Thus, the driven pulley's diameter was calculated using Equation (34) as follows:

$$\frac{700}{200} = \frac{D_2}{70} \quad D_2 = \frac{700*70}{200} = 245 \text{ mm}$$

Where; N_1: driven pulley speed (rpm), N_2: Driver pulley speed (rpm), *B:* Width of pulley (mm), *b:* Width of belt (mm), t_1: rim thickness of the driving pulley (mm) and t_2: rim thickness of the driven pulley (mm).

Power required

The machine needed the power to rotate the eccentric shaft, run the sieve when the material was on it both horizontally and vertically, and overcome friction. This totaled the power needed for the machine. The power needed to run the sieve was computed using the formula below (Okunola *et al.*, 2015).

Power required to operate the sieve horizontally;

$$P_1 = \frac{W_S+N+2*X*\mu}{4500} = \frac{66.7\ kg+200\ rpm+2*0.09m*0.1}{4500} = 0.05 \text{ Hp} = 39.52 \text{ watt}$$

Power required operating the sieve vertically;

$$P_2 = \frac{W_S+N+2*Y}{4500} = \frac{66.7\ kg+200\ rpm+2*0.02}{4500} = 0.1 \text{ hp} = 74.57 \text{ watt}$$

From Equation 36, we have P_3 205.48 watts. So, we can determine the total power required by using Equation (38) as follows:

$$P = P_1 + P_2 + P_3 \quad (38)$$

$$P = 39.52 + 74.57 + 205.48 = 319.57 \text{ watt}$$

The total power required was calculated as the power required and power loss due to friction (10% of total power):

$$P_T = P + P(0.1) = 319.57+319.57(0.1) \ P_T = 351.53\ watt$$

Where; W_s: Weight of grading unit with material on it (66.7 kg with material on it), *N:* Speed (200 rpm), *X:* Horizontal movement of sieve (diameter of eccentric= 0.09m), *Y*: Vertical movement of sieve (maximum angle of sieve 11°), μ: Coefficient of friction of bearing (0.1)

Working Principles of the Machine

The coffee cherries are loaded into the hopper while the main power source is switched on and the machine is operating. The feed regulating is adjusted to the proper position and the engine to the proper speed for optimum performance. Coffee cherries flow under gravity to the grading sieve and fall on the reciprocating screen through the hopper. The horizontal reciprocating motion of the screen causes the materials on it to move towards the front end and the beans that are below the diameter of the hole pass through the hole of the screen to the second level screen which then repeats the first step until the last level sieve and graded was collected through the grain outlet for each sieve level.

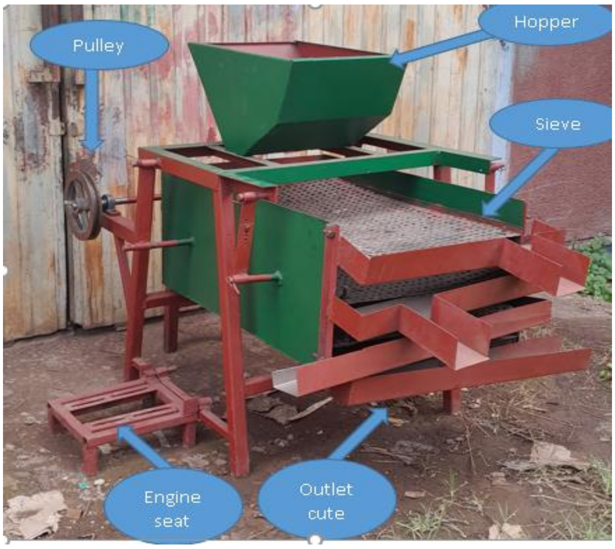


Figure 5. *Pictorial view of the developed grading machine.*

Performance Evaluation

Experimental design

The experimental design was a split-split plot with three replications. Treatments consisted were combinations of three levels of oscillating speed (80 rpm, 140 rpm, and 200 rpm), three levels of feed rate (5 kg min^{-1}, 10 kg min^{-1}, and 15 kg min^{-1} and three levels of inclination of the sieves (7°, 9°, 11°) (Yayock and Ishaya, 2020). The experiment design was laid as 33 with 3 replications having an entire test run of 81.

Variables and Data Collection

Independent variables

During the evaluation, three independent variables were employed. These were feeding rates (5, 10, and 15 kg min^{-1}), the grading unit oscillating speed (80, 140, and 200 rpm), and sieve inclination angle (7°, 9° and 11°).

Dependent variables and collected data

Grading efficiency (GE, %)

Grading capacity (kg h^{-1})

Fuel consumption (l kWh^{-1})

Performance indicators

Data were collected through the experiment that was undertaken during testing and, then performance evaluation of the machine was made. The assessment of the performance of the machine followed the method of Chungcharoen *et al.* (2019).

Size grading efficiency (E_W)

$$E_W = \frac{\sum P_g W_i G_i}{Q P_i} \tag{39}$$

Where; P_g: fraction of coffee cherries size I to overall coffee cherries dropped on receipt plate

Wi: fraction of coffee cherries size I at the start of sizing to total coffee cherry at the beginning of sizing

Gi: flow rate of coffee cherry size I (kg h^{-1})

Pi: fraction of size i to total coffee cherries at the beginning of sizing

WC: weight of coffee cherry with a clogging sieve in all sieves (kg)

Wt: total weight of cherry

Capacity of the machine

$$C = \frac{Total\ output}{Time\ taken} \tag{40}$$

Fuel consumption

Fuel consumption was assessed by refilling the fuel tank after each test run and determining the amount required to refill, recording the time of a test run.

Data Analysis Method

An analysis of variance was performed on the experiment's data using a protocol that was suitable for its design, i.e. oscillating speed as main plot, angle of sieve as subplot, and feed rate as sub-sub plot. The treatment means were different at a 5% level of significance and were separated using LSD. R-statistics 4.1.0 version software was used to analyze the data.

RESULTS AND DISCUSSION

This chapter reports and discusses the findings of the tests done as part of the research study, including the results of the physical and mechanical features. Important physical and mechanical characteristics of coffee cherries were identified in order to build the various components of the grading machine.

Physical and Mechanical Properties of Coffee Cherries

The linear dimensions, geometric mean diameter, arithmetic mean diameter, shape index, surface area, moisture content, bulk density, and unit mass of coffee cherries, as well as their mechanical characteristics, including static friction coefficient and angle of repose, were determined. Based on their applicability to the design and performance evaluation of the coffee bean grading machine, some of the examined physical and mechanical parameters were taken into consideration.

Major diameter, minor diameter, intermediate diameter, shape index, and sphericity.

In order to determine the size of the hole of the sieve and the physical properties of coffee cherries, major diameter, minor diameter, intermediate diameter, shape index, and sphericity were calculated and considered in the design on the coffee bean grading machine. The result of the determination of coffee physical properties revealed that the coffee cherries have major diameter, minor diameter, intermediate diameter, shape index, and sphericity of 13.44 mm, 8.7 mm, 10.26 mm, 0.9, and 0.77, respectively. Those parameters were used in the design of the size and shape of the sieve of the grading machine. The mean value of the shape index of coffee cherries was 0.98, and this showed that the coffee cherries are spherical in shape. In this regard, the shape of the sieve was selected as spherical.

The measurements of coffee cherry dimensions (major diameter: 13.44 mm, minor diameter: 8.7 mm, intermediate diameter: 10.26 mm) provide a comprehensive understanding of the fruit's size variability. This information is vital for designing sieves that can effectively separate cherries based on size, ensuring proper grading and sorting.

The shape index of 0.9 and sphericity of 0.77 indicate that coffee cherries are not perfectly spherical but close to it. This near-spherical nature is further confirmed by the mean shape index of 0.98. This finding has significant implications for the sieve design, as it justifies the selection of shape for the sieve holes. A circular sieve design aligns well with the natural form of the coffee cherries, potentially improving the efficiency and accuracy of the grading process.

Bulk density and moisture content

When designing the hopper, storage, and grading equipment, bulk density is crucial. The bulk density of the coffee cherries and the moisture of the coffee to be graded were taken into consideration when building the hopper for the grading equipment. Accordingly, the cherries had a bulk density of 650 kg m^{-3} and a moisture content of 12.69%, respectively.

Angle of repose, surface area, arithmetic mean, geometric mean, and aspect ratio of coffee cherries. Thus, these parameters are more important for setting up the coffee grading machine sieve size. The coffee cherry has a mean average of 349.05 m^2 of surface area and 77.38 mm and 10.42 mm of Arithmetic mean and geometric mean respectively.

Coefficient of static and dynamic friction

The roughness of the frictional surface is determined by the coefficient of friction. In line with this, the value of the coefficient of static friction increases with surface roughness. In the experimental test, the coefficients of static friction of coffee cherries were found to be 0.21, 0.29, and 0.37 on surfaces made of galvanized mild steel sheet, mild steel sheet, and plywood, respectively. Similarly, the coefficients of dynamic friction in coffee cherries were found to be 0.34, 0.44, and 0.51 on surfaces made of galvanized mild steel sheet, mild steel sheet, and plywood, respectively.

General design specification of the coffee grading machine

The general specifications of the coffee grading machine describe the functional components such as the feeding hopper, grading mechanism, sieve, shaft, outlet chute, power source, and supporting frame of the machine are listed in Table 1 below.

Table 1. *An overview of the coffee grading machine's design specifications.*

S/N	Components	Functions	Specifications
	Frame	Serves as the platform for mounting additional	0.9 m, 1.33 m and 1.11 m Height, length, and width respectively
	Hopper	Serve as temporary storage	0.35 m, 0.4 m, 0.55 m Height, width and length respectively
	Grading Unit	To hold grading sieve	0.40 m, 0.95 m, and 0.60 m in Height, length, and width respectively
	Grading Sieve	To grade Coffee cherries into different sizes	0.95 m and 0.60 m, length, and width respectively
	Shaft	To transmit power	0.03 m diameter, 0.95m length
	Outlet chute	To collect graded cherry	0.10 m, 0.70 m, and 0.10 m in Height, length, width respectively
	Pulley	To transmit power	245 mm diameter, 12.5 cm width

Performance evaluation of the machine

The outcomes of the data analysis are summarized in Tables 2 and 3. These comprised of the combination mean values of the performance parameters.

Table 2. *The combination effect of operating parameter on capacity, efficiency, sieve clogging, and fuel consumption of machine.*

Feed rate (kg min^{-1})	Angle (°)	Speed (rpm)	Efficiency (%)	Capacity (kg h^{-1})	Clogging (%)	FC (l kWh^{-1})
5	7	80	90.40 a	147.12 f	4.86 a	0.60 j
		140	88.47ab	161.36 e	4.37 b	0.70i
		200	86.40ab	1 72.54 c	3.57 c	0.80gh
	9	80	83.21bc	149.10 f	4.18 b	0.61 j
		140	77.87d	161.77 d	3.68 c	0.71i
		200	73.64de	218.27 b	3.24 d	1.00 h
	11	80	78.29cd	191.34 c	3.69 c	0.77 j
		140	73.25de	213.65 b	3.27 d	0.93i
		200	67.66de	277.33 a	2.69 e	1.27 h
10	7	80	89.43 a	139.17 f	4.91 a	0.65 g
		140	88.01ab	157.39 e	4.42 b	0.60 f
		200	87.40ab	177.56 c	3.63 c	0.96 de
	9	80	83.11bc	142.10 f	4.09 b	0.66gh
		140	77.57d	169.77 d	3.68 c	0.64 f
		200	73.84de	228.35 b	3.25 d	1.23 de
	11	80	78.20cd	181.27 c	3.73 c	0.83 h
		140	73.15de	220.37 b	3.23 d	1.12 f
		200	67.76de	278.52 a	2.70 e	1.50 de
15	7	80	90.50 a	137.11 f	4.96 a	0.74 de
		140	88.00ab	151.37 e	4.32 b	0.90 c
		200	86.90ab	175.51 c	3.65 c	1.09 ab
	9	80	83.21bc	146.00 f	4.13 b	0.80 d
		140	78.00d	165.72 d	3.70 c	0.99 c
		200	74.01de	223.26 b	3.22 d	1.38 b
	11	80	78.22cd	181.27 c	3.70 c	0.97 e
		140	73.35de	223.26 b	3.20 d	1.34 c
		200	67.76de	279.32 a	2.66 e	1.75 a
SD			2.65	3.74	0.14	0.05
CV			5.32	2.01	2.01	0.50

Where; SD is standard deviation, CV is coefficient of variance and Mean values with the same letter in a column are not significantly different at 5% level of significance.

Table 3. *The main effect of operating parameters is on the capacity, efficiency, sieve clogging, and fuel consumption of the machine.*

Parameters		Capacity (kg h^{-1})	Efficiency (%)	Sieve clogging (%)	Fuel consumption (l kW h^{-1})
Feed rate (kg/min)	5	186.76 [a]	79.93 a	3.73 a	0.81 c
	10	196.16 [a]	79.88 a	3.72 a	0.99 b
	15	198.76 [a]	79.82 a	3.68 a	1.17 a
Sieve inclination angle (°)	7	154.66 [c]	88.42 a	4.31 a	0.79 a
	9	178.33 [b]	78.24 b	3.68 b	0.91 a
	11	227.28 [a]	71.95 c	3.13 c	1.15 a
Oscillating speed (rpm)	80	154.79 c	82.84 a	4.21 a	0.73 c
	140	211.11 b	79.56 b	3.74 b	1.09 b
	200	225.36 a	75.90 c	3.18 a	1.22 a
SD		1.24	0.88	0.04	0.05
CV		2.50	1.77	2.00	2.00

Where, SD is standard deviation, CV is coefficient of variance and Mean values with the same letter in a column are not significantly different at 5% level of significance.

Efficiency of machine

The effect of operation parameters on the grading efficiency of machines was presented in Tables 2 and 3. The result shows that when the oscillating speed of operation increased from 80 rpm to 140 rpm the grading efficiency decreased from 82.84% to 76.56%, whereas it decreased from 76.56% to 75.90% when the oscillating speed increased from 140 rpm to 200 rpm. The grading efficacy decreased from 88.42% to 78.24% and from 78.24% to 71.95% when the sieve inclination angles were increased from 7˚ to 9˚ and from 9˚to 11 ˚, respectively. This revealed that the greater inclination angle would cause the coffee cherries to move too quickly, and lead to the wrong category of size grading.

The grading efficiency decreased from 79.93% to 78.88% and from 78.88% to 79.82% when feed rates increased from 5 kg min^{-1} to 10 kg min^{-1} and from 10 kg min^{-1} to 15 kg min^{-1}, respectively (Table 3). The analysis of variance (ANOVA) revealed that the inclination angle and oscillating speed had a significant effect ($p<0.05$) on the grading efficiency. This agrees with the findings of *Chungcharoen et al.* (2019), who concluded that too much-inclined angle and oscillating speed would lead to excessive movement of coffee cherries, leading to the incorrect size category classification. The combination of feed rate and inclination angle, feed rate and oscillating speed, inclination angle and oscillating speed, and the interaction of inclination, feed rate, and oscillating speed had no significant effect ($p < 0.05$) on the grading efficiency.

Grading capacity of the machine

Main and combined effects of inclination angles, feed rates, and sieve cycles on the grading capacity of the machine

The main and interaction effects of oscillating speed, inclination angle, and feed rate on the grading capacity are shown in Tables 2 and 3. The result revealed that the grading capacity of the machine increased from 154.66 kg h^{-1} to 178.33 kg h^{-1} and from 176.33 kg h^{-1} to 227.28 kg h^{-1}, as the sieve inclination angles increased from 7˚ to 9˚ and from 9˚ to 11˚, respectively. Similarly, when the oscillating speed was increased from 80 rpm to 140 rpm the grading capacity increased from 154.79 kg h^{-1} to 211.11 kg h^{-1}. In addition, it increased from 211.11 kg h^{-1} to 225.36 kg h^{-1}, as the oscillating speed increased from 2.33 Hz to 3.33 Hz (Table 5). This happened because the coffee cherries' retention period on the sieves reduced due to increased sieves' inclination angles and sieves' oscillations increase.

The feed rates were also used to evaluate the machine's capacity. Results show that the grading capacity increased from 186.76 kg h^{-1} to 196.16 kg h^{-1} and from 196.16 kg h^{-1} to 198.76 kg h^{-1} as feed rates increased from 5 kg min^{-1} to 10 kg min^{-1} and from 10 kg min^{-1} to 15 kg min^{-1}, respectively (Table 3).

The machine's highest capacity of 277.32 kg h^{-1} of grading was recorded at 200 rpm oscillating speed and 11˚ sieve inclination. While the machine's minimum grading capacity of 137.11 kg h^{-1} was observed at a feed rate of 15 kg h^{-1} inclination angle of 7˚ and at oscillating speed of 80 rpm.

The analysis of variance revealed that, the inclination angle and oscillating speed had a significant effect ($p<0.05$) on the machine's grading capacity. This result matches with the results of Ansar *et al.* (2021), and Chungcharoen *et al.* (2019) hich indicated that the grading capacity of coffee cherries machines is significantly affected by the inclination angle and oscillating speed. Feed rate, the combination of feed rate and inclination angle, feed rate and oscillating speed, inclination angle and oscillating speed and the interaction of inclination, feed rate, and oscillating speed all had no significant effect ($p<0.05$) on the machine's grading capacity.

The results demonstrate the significant impact of oscillating speed and sieve inclination angle on the coffee grading machine's capacity. As these parameters increased, the grading capacity improved substantially, with the highest capacity of 277.32 kg h^{-1} achieved at 200 rpm and 11° inclination. This improvement can be attributed to the reduced retention time of coffee cherries on the sieves at higher angles and oscillation speeds. While feed rate also influenced capacity, its effect was less pronounced and not statistically significant. These insights provide valuable guidance for optimizing grading machine design and operation, suggesting that focusing on these two parameters could lead to significant improvements in processing efficiency without the need for increased feed rates.

Fuel consumption

Main and combined effect operating parameters on fuel consumption of the machine. Tables 4 and 5, illustrate the main and combined effects of the inclination angle, feed rate, and speed on fuel consumption. The maximum fuel consumption was 1.75 l kWh^{-1} at an inclination angle of 11˚, feed rate of 15 kg min^{-1}, and speed of 3.33 Hz. On the other hand, the minimum fuel consumption was 0.60 l kWh^{-1} at 7˚ inclination angle, feed rate of 5 kg min^{-1}, and oscillating speed of 1.33 Hz.

As shown in Table 3, fuel consumption increases with increasing feed rate and speed. Fuel consumption increased from 0.81 l kWh^{-1} to 0.99 l kWh^{-1} and from 0.73 l kWh^{-1} to 1.09 kWh^{-1} when the feed rate and oscillating speed were increased from 5 kg min^{-1} to 10 kg min^{-1} and from 80 rpm to 140 rpm, respectively. It also increased from 0.99 l kWh^{-1} to 0.91 l kWh^{-1} and from 1.09 l kWh^{-1} to 1.22 l kWh^{-1} when the feed rate and oscillating speed were increased from 10 kg min^{-1} to 15 kg min^{-1} and from 140 rpm to 200 rpm, respectively.

The analysis of variance (ANOVA) revealed that feed rate and speed had a significant effect ($p<0.05$) on fuel consumption. However, the combination of feed rate and inclination angle, feed rate and speed, inclination angle and speed, and the interaction of inclination, feed rate, and speed had no significant effect ($p<0.05$) on fuel consumption.

CONCLUSION

A coffee cherry grading machine was designed and developed taking into consideration the physical and mechanical properties of Coffee Arabica. The machine was fabricated and tested at the Jimma Agricultural Engineering Research Center. From the study, the average measured values of the major diameter, minor diameter, and intermediate diameter of the coffee cherries were found to be 13.44 mm, 7.00 mm, and 10.26 mm, respectively. From the analysis, the shape of the sieve was found to be spherical; its sphericity was 0.77. To fix the size of the hopper, the moisture content and bulk density are critical, and the results of those properties were 650 kg m^{-3} and 12.69%, respectively. The static and dynamic friction coefficients were 0.37 and 0.51 on plywood, 0.29 and 0.44 on mild steel sheet metal, and 0.21 and 0.34 on galvanized metal, respectively. Also, the angle of repose was found to be 35.96. The designed machine was operated with a 3 hp gasoline engine as a power source. The sieve of the grading machine was circular based on the shape index of the coffee cherries and the machine has three sieves. The reciprocating action of the sieve was created by using the eccentric method.

The performance evaluation indicated that the machine's grading efficiency depends on the sieve's inclination angle and the sieve's oscillating speed. In this regard, the grading efficiency decreased when the inclination angle and the oscillating speed increased. The highest grading efficiency was recorded to be 82.84% at an oscillating speed of 80 rpm.

The grading capacity of the machine mainly depended on the inclination angle and oscillating speed of the sieve. The capacity of machine increased with the increase of the inclination angle and oscillating speed. The highest machine capacity of 279.32 kg h^{-1} was recorded at an 11˚ inclination angle of the sieve, 15kg min^{-1} feed rate, and 20 rpm.

The lowest value of the sieve clogging recorded was 3.18 % and 3.13 %at an oscillating speed of 200 rpm and 11˚ inclination angle of the sieve, respectively. Sieve clogging decreased with increasing oscillating speed and inclination angle of the sieve.

In general, the following conclusions can be made:

-The machine has the highest grading capacity when it is operated at 15 kg min^{-1} feed rate, and 11° inclination angle, and a 200 rpm oscillating speed.

-The machine has the highest grading efficiency when it is operated at a 15 kg min^{-1} feed rate, 7˚ inclination angle, and a 80 rpm oscillating speed.

-The machine has the lowest sieve clogging when operated at a 15 kg min^{-1} feed rate, 11˚ inclination angle, and 200 rpm oscillating speed.

-The current grading machine has been developed and tested on a small scale. In this regard, the following recommendations are made.

-The grader should be operated at 80 rpm oscillating speed, 15 kg min^{-1} feed rate, and inclination angle of 7° to get high grading efficiency.

-The current machine should be further modified and tested at farmers' level.

-The machine needs to have wheels for easy transportation from place to place.

-The current machine uses a diesel engine better to design it for other alternative sources of energy.

-To make it more versatile, the grading machine should be redesigned with replaceable sieves to grade green coffee bean.

DECLARATION OF COMPETING INTEREST

The authors declare that they have no conflict of interest.

CREDIT AUTHORSHIP CONTRIBUTION STATEMENT

The authors declared that the following contributions are correct.

Tolasa Berhanu: Proposal writing, methodology, manufacturing prototype, writing original draft,
Adesoji Matthew Olaniyan: Advising, review, and editing,
Habtamu Alemayehu: Advising, review, and editing.

ETHICS COMMITTEE DECISION

This article does not require any Ethical Committee Decision.

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