Design, Fabrication, and Testing of Motor Operated Multi-Grain Cleaning Machine

Sara Addisu Birhanu

A Thesis Submitted to the Department of Mechanical Systems and Vehicle Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirement for the Degree of Master's in Agricultural Machinery Engineering

Office of the Graduate Studies

Adama Science and Technology University

July, 2021

Adama, Ethiopia

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DECLARATION

I hereby declare that this MSc thesis entitled "Design, Fabrication, and Testing of Motor Operated Multi-Grain Cleaning Machine" is my original work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

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Name of the student	Signature	Date

RECOMMENDATION

I, the advisor of this thesis, hereby certify that I have read the revised version of the thesis entitled "Design, Fabrication, and Testing of Motor Operated Multi-Grain Cleaning Machine" prepared under my guidance by Sara Addisu submitted in partial fulfillment of the requirements for the degree of Mater of Science in Agricultural Machinery Engineering.

Therefore, I recommend the submission of revised version of the thesis to the department following the applicable procedures.

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APPROVAL SHEET

I, the advisor of the thesis entitled "Design, Fabrication, and Testing of Motor Operated Multi-Grain Cleaning Machine" and developed by Sara Addisu, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

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Advisor	Signature	Date

We, the undersigned, members of the Board of Examiners of the thesis by Sara Addisu have read and evaluated the thesis entitled "Design, Fabrication, and Testing of Motor Operated Multi-Grain Cleaning Machine" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Agricultural Machinery Engineering.

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Prof. (Dr) Kishor Kolhe		26/07/2021
External Examiner	Signature	Date

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ACKNOWLEDGMENT

First, I would like to praises and thank Almighty God, for the most merciful, and the most blessing that makes me stronger and stay healthy throughout my life and to do this project successfully.

I gratefully acknowledge Adama Science and Technology University for granting me a scholarship to pursue my study and for funded my research by grant number ASTU/SM-R/105/19.

I would like to express my deepest gratitude to my advisor Amana Wako (Ph.D.) for serving as supervisor and for his support, dedication, timely advice, patience, and experience he shared with me. Thanks to you for the confidence and encourage me that led this thesis successfully.

My gratitude and acknowledgment go to Melkassa Agricultural Research Center for permission to use the workshop, facilities, and supply materials to do my project. Also, my honest gratitude to researcher Mr. Tamrat Lema and technician Mr. Besufikade Getachew. My special appreciation and acknowledgment also go to all staff members of the Melkassa Agricultural Research Center AIRC department.

Lastly, but not least great gratitude goes to my beloved family; mom, sisters, brothers for their love, prayers, caring, support, and sacrifices for my education throughout my life and to complete this research work. Also, my special thanks go to my best friends who give love, appreciation, and understanding.

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LIST OF ACRONYMS

USAID	United State Agency for International Development
ASABE	American Society of Agricultural and Biological Engineering
ASME	American Society of Mechanical Engineers
CSA	Central Statistical Agency
AARC	Assela Agricultural Research Center
MARC	Melkassa Agricultural Research Center
EIAR	Ethiopian Institute of Agricultural Research
HP	Horse Power
MOG	Material Other than Grain
Std.Er	Standard Errors of Means
Cv	Coefficient of Variation
ANOVA	Analysis of Variance
Rpm	Revolution per minute
AoR	Angle of Repose
CE	Cleaning Efficiency
SE	Separation Efficiency
SA	Sieve Angle
FR	Feed Rate
AV	Air Velocity

ABSTRACT

Post harvesting for cereal grains following a chain of activities which starting from farmer fields leading eventually to cereal being supplied to the consumer as they prefer. As the above-mentioned unit of operation cleaning is one of the most important processes to separate the material other than grain which reduces the value of the crop. Even though, in Ethiopia farmers use the traditional way of cleaning system which causes too much physical drudgery; time consumption; and also labor-intensive. Furthermore, there is grain loss during cleaning, and also during storing the uncleaned grain cause the increase of mold and insects. To alleviate the aforementioned problem, an electric motor-operated multi-grain cleaning machine was designed, fabricated, and tested. Therefore, the study was conducted at MARC. The cleaning machine component was designed and fabricate by using different methods; it consisted of mainframe, sieve frame, fan, sieve, brush, outlets, motor, hopper, and power transmission units. The physical and mechanical properties were determined to the selected crops with three different varieties including wheat with Dandaia, kingbird, and Lemu; maize with Melkassa 2, Melkassa 4, and MH – 104; and sorghum with Melkame, Tillahune, and Dagim. The performance evaluation was carried out at three levels of feed rate (10, 12, 15kg), sieve angle (2, 4, 6°), and air velocity (3, 2.1, and 1.3m/sec) to determine the optimum combination of the factor. Therefore, the experimental design laid was factorial, in a split – split-plot design with three replication. The cleaning and separation efficiency of the machine for wheat, maize, and sorghum was determined from the tested results. Hence, the result indicates that the cleaning efficiency was increased with decreasing in the value of feed rate, sieve angle, and air velocity. Although, the separation efficiency of the machine increase with decreasing of the feed rate but it decreases with the decrease of sieve angle and air velocity. The maximum separation efficiency of 100% found for the wheat crop was at sieve angle 2 and 4°, air velocity 3m/sec and feed rate10kg; for maize crop, 97% was found at sieve angle 4°, feed rate 10 and 12kg, air velocity 3m/sec; and also for sorghum 100% estimated from sieve angle 2°, feed rate 10kg, and air velocity 3m/sec. accordingly, it can be concluded that the performance of the prototype machine is very much acceptable with a high prospect for extending the technology for all types of farmers.

Key Word: cleaning efficiency, separation efficiency, wheat, sorghum, maize

CHAPTER ONE

1. INTRODUCTION

1.1. Background of the Study

Agriculture is the backbone of the Ethiopian economy. Therefore, it generates approximately 36.5% of its Gross Domestic Product (GDP), 82% of total exports, and provides 80% of employment (CSA, 2018). Frequently, agriculture is dominated by small-scale farmers who practice rain-fed mixed farming by employing traditional technology, adopting a low-input and low-output production system. The land tilled by the Ethiopian small-scale farmer accounts for 95% of the total area under agricultural use, and these farmers are responsible for more than 90% of the total agricultural output (Gebre-selassie & Bekele, 2010). Also, the smallholding is with 85% of landholdings less than 2ha, 60% less than 1ha, and a staggering 40% less than 0.5ha (USAID, 2018).

In Ethiopia cereals are the major food crops both in terms of the area they are planted and the volume of production obtained. Because of produced in a larger volume they are principal staple crops when it is compared with others. So, there are five main kinds of cereal teff, wheat, maize, sorghum, and barley (Seyoum et al., 2011).

Out of the total grain crop area, 80.78% (10,144,252.30 hectares) is under cereals. Teff, maize, sorghum and wheat took up 24.02% (about 3,016,053.75 hectares), 16.80% (about 2,110,209.61 hectares), 14.58% (1,831,600.45 hectares) and 13.25% (1,663,837.58 hectares) of the grain crop area, respectively. Cereals contributed 87.31% (about 236,076,624.39 quintals) of the grain production. Maize, teff, wheat and sorghum made up 26.76% (72,349,551.02 quintals), 17.57% (47,506,572.79 quintals), 15.65% (42,315,887.16 quintals) and 16.05% (43,391,342.61 quintals) of the grain production (CSA, 2015).

Harvesting and Postharvest operations for cereal grains follow a chain of activities starting in farmers' fields and leading eventually to cereals being supplied to consumers in a form they prefer (Goletti & Samman, 2002). When determining the losses that may occur in this chain it is conventional to include drying in the field and/or on platforms, threshing, winnowing, transport to store, farm storage, losses incurred in transport to market, and market storage

(Dubale, 2018). So the losses can be broadly categorized as weight loss due to spoilage, quality loss, nutritional loss, seed viability loss, and commercial loss (Boxall, 2001).

The magnitude of postharvest losses in the food supply chain varies greatly among different crops, areas, and economies. In developing countries, people try to make the best use of the food produced; however, a significant amount of produce is lost in postharvest operations due to a lack of knowledge, inadequate technology, and/or poor storage infrastructure (FAO, 2014). In sub-Saharan Africa cumulative loss of cereal production is 10-23% of total production (Hodges et al., 2014).

Ethiopian Institute of Agricultural Research (2016) reported harvesting and postharvest losses in Ethiopia are 15% for food grains and 20–25% for fruits and vegetables. For highly perishable fruits and vegetables, these losses may go as high as 40%. One earlier report estimated that crop losses of 2% to 3%, 1 to 2%, 4 to 6%, 2 to 5%, and 1 to 3% occur in cereals during cutting, drying, threshing, winnowing, and transportation, respectively (Mohammed & Abreaham, 2018).

Crops on harvest usually contain impurities that have to be removed for the efficient and safe operation of processing equipment and production of good quality products. Examples of such impurities are unwanted seeds, plant residues, fungal impurities, rodent excreta, mud, dust, and stone. These impurities reduce the value of the crop (Akinoso et al., 2010).

Therefore, the cleaning process is one of the important postharvest processes performed after the threshing to separate whole grains from broken grains and other foreign materials. Winnowing is the most common method used for cleaning in developing countries. Screening/sifting is another common method of cleaning, which can be performed either manually or mechanically. Inadequately cleaned grains can increase insect infestation and mold growth during storage, add unwanted taste and color, and can damage the processing equipment. A large number of grains are lost as spillage during this operation, and grain losses during winnowing can be as high as 4% of the total production (Debashis et al., 2013).

In Ethiopia cleaning activates done by using a wooden fork, usually with three or four prongs, are used to separate the grain and straw. A further way of winnowing is carried out by raising a container full of grain and chaff above the farmer’s height and letting it drop smoothly so that the blowing wind will separate the seeds from the chaff. Lighter-weight broken grains, straw, and weed seeds are carried by the wind to one side, as the whole and the sound grain falls to the bottom of the winnowing device. The operation requires a continuous brisk wind and several repetitions.

The traditional way of threshing and winnowing leads to contamination of grains with foreign matter (pebbles, dirt, and cow dung) and loss of grains due to the foreign matter dropping into grains and the wind will blow grains away with chaff. Some innovative farmers have started to use canvas or plastic sheeting on their threshing and winnowing floors to minimize these risks. After cleaning, farmers pack their grain in jute bags, silcha, or bags made of leather (Mohammed & Abreaham, 2018).

Abayineh researched *tef* grain separating and cleaning machine at Haramaya University. The separation efficiency, cleaning efficiency, separation loss, and cleaning loss were 97.94, 98.58, 0.71, and 0.7%, respectively at sieve oscillation of 5 Hz, a feed rate of 3 kg/min, and a sieve slope of 5 degrees. The separation efficiency, cleaning efficiency, separation loss, and cleaning loss were 52.71, 39.88, 25.62, and 34.5%, respectively, at sieve oscillation of 20 Hz, at a feed rate of 12 kg/min, and a sieve slope of 5 degrees. It is only used for one type of cereal; didn't have a hopper to feed the system and also uses neither Brush nor other material to increase the performance of the machine.

Besides there all motivation, this research aimed at the fabricated machine which collaborates different parts and mechanisms to reduce labor number in addition to increase the cleaning efficiency and separation efficiency. Furthermore, this fabricated machine had a notable advantage for reduction of quantity loss during cleaning and also a quality loss for over and above processing operations. What is the most prosperous part of the machine is it can be used for different cereal crops in a different place as a consequence it is amiable to use by the farmer and another user.

1.2. Problem Statement

In Ethiopia cereal crop farming system most of the farmer has been suffering as a result of tradition cleaning system due to the irregularity of Agroecology. Also, most of rural community's has no access to use combine harvester and there is no enough technology's to use for small scale grain cleaning system because of this farmers are obligated to use traditional cleaning system. Even though, they custom combine harvester and threshing machines it requires further cleaning to use the cereal for another processing operation.

These traditional systems require a lot of energy and time to complete the cleaning process. Furthermore, the whole process required repetition which led the farmer to drudgery. Muhammad et al. (2013) indicated the time taken to clean a batch of 1 kilogram of un-cleaned seeds ranges between seven to twelve minutes depending on the winnower's skill, the required cleanliness, grain/non-grain ratio, amount, and stability of the natural air current and other environmental factors.

Additionally, traditional cleaning operation which is done in an open space with free flow of natural wind winnowing activity is only effective when there is a good climate condition. Besides, if there is a shortage of speed of the wind the farmer should wait for hours until the wind speed increase. On the other hand, the grain takes with the chaff if the wind speed is excessive than required. These all difficulties led to grain loss and reduction of the grain quality.

According to Sinha (1975), the presence of dockage that includes damaged grain, dust, weed seeds, and other foreign materials from unclean grain increases the chance of heating and deterioration of stored grain. Chaff and grain dust tend to absorb moisture, it is a more suitable substrate for mold development than entire grains. Many insects that are unable to penetrate sound grain can develop well on this material so deterioration is caused by insect infestation, microbial infection, or both. As a result of this excessive grain, the loss can be occurs

Hence, mechanized cleaning system design and fabrication is crucial to alleviate the aforementioned problem

1.3. Research Objectives

1.3.1. General Objective

The general objective of this research is to design, fabricate, and testing a motor-operated multi-grain cleaning machine.

1.3.2. Specific Objective

The specific objective of the study is to:

- Measure the physical and mechanical properties of the grain (Wheat, Maize, and Sorghum).
- Design and fabricate different parts of the machine.
- Evaluate the performance of the machine with respect to separation efficiency, cleaning efficiency by using three parameters.

1.4. Significance of the Study

Ethiopia heavily relies on an agriculture production system to fulfill the people's needs. Therefore, Post harvesting operation is one of the important parts of the agricultural production system, which perform from the farmer field up to the consumer. Therefore, Cleaning is one the most significant part of the activity to do before consumption and take to the market. Even though, most Ethiopian farmer uses traditional cleaning system which requires too much human power and time; also which is drudgery working cycle. Besides, the loss of quality and quantity is high during cleaning and storage.

This research has major significance in assuring the development of an alternative, suitable, consistent, simple to maintain, and also the components can be adjusted and replace merely by unscrewed bolt and nuts of machine for the farmer. The other major significance of this study is to reduce the quality and quantity loss of the product. Furthermore, reduce the requirements of human power to do cleaning operation and increase the market incomes for the farmer.

1.5. Scope of the Study

The scope of thesis work within the given time and resource includes measuring the physical properties of the three cereal crops within three different varieties for each crop (Maize, Sorghum, and Wheat) by using different instruments and methods to determine the sieve perforation type and size also angle of inclination for hopper design; design the machine parts and drawing by using CATIA V5 software before starting the fabrication; construction of the machine beyond to the design; finally test the performance of the machine on three different crops after all the data analyzed in the form of table and graph, which is found from the measured and tested value by using Excel, Statistix 10, and SPSS software.

1.6. Limitation of the Study

The thesis work does not cover some of cereal crop properties including the terminal velocity of crops which is important to determine fan speed. Even though, it was difficult to get the instrument as well as design and fabricate due to time limitations. Hence, the study uses the value of terminal velocity from other researcher results.

1.7. Organization of the Study

The remaining part of the thesis is organized as follow:

Chapter 2: Discuss the relevant literature and related work regarding the physical properties of crops; factors that affect the performance of cleaning machines; and also the consideration of machine design.

Chapter 3: Feature the material and methods for the research, including instruments for measuring the physical properties of crops and machinery to fabricate the machine. And also it includes different methods to design the machine.

Chapter 4: Explain the result from measuring the crop physical properties and performance of the machine and discuss the result of cleaning efficiency, separation efficiency, followed by the analyzed data.

Chapter 5: Concludes on every proposed specific objective and recommend different possible direction for further study.

CHAPTER TWO

2. LITERATURE REVIEW

2.1. Post Harvesting

The post-harvest system is a crucial and dynamic component of the agricultural complex. Increasingly, agricultural products are not consumed in their raw form, and post-harvest activities such as transport, storage, processing, and marketing account for a growing part of their final value. In the poorest countries, these activities often provide the first step for the development of manufacturing. Processed products can generate export earnings, and even if consumed domestically, can free up a foreign exchange to be used for other imports and time traditionally spent preparing food for more productive uses. An efficient post-production system can help to ensure a safe and sufficient food supply in growing cities and can give impetus to the development of infrastructure and institutions that are vital for further economic growth (Goletti & Samman, 2002).

Postharvest Operations

The main stages within the agricultural supply chain during which losses occur in grain crops and the associated causes of losses may be distinguished as indicated below.

Staking and Field Drying

Grains should be dried in such a manner that damage to the grain is minimized and moisture levels are lower than those required to support mold growth during storage (usually below 13–15 percent). This is necessary to prevent further growth of several fungal species that may be present on fresh grains. To achieve this, the harvested crop may be left standing in the field, cut and left drying on the ground, or stoked. In some places, the crop may be moved immediately from the field to a swept area of ground at the homestead or to racks or cribs that are specifically designed to promote drying (Zorya et al., 2011).

Insufficient drying - Grain that is not dried to a safe moisture content shortly after harvest will be attacked by molds. Molds may develop on the surface of grain that is above the safe moisture content, which under hot tropical conditions is around 14%. High moisture content is also favorable for the development of insect infestation and may also lead to chemical browning reactions that result in grain discoloration (Hodges, 2013).

Threshing/shelling

Losses at threshing may arise because the threshing is incomplete (i.e. some grain remains on the seed head), the grain is scattered and spilled, or the grain becomes damaged in the process (Hodges et al., 2014).

In the Ethiopian context, the common methods of threshing under small-scale farmers' conditions are: (i) manually using hands, mainly for maize (shelling); (ii) manually beating with sticks; (iii) animal trampling. However, the most common method is animal trampling (Dawit, 2014).

According to Dereje & Getachew (1989) in crop production, where the traditional way respectively per annum of production is followed, the lion-share of labor requirement goes to threshing and harvesting which accounts for about 40% of its total man-day. Moreover, in the traditional method (manual and animal trembling methods) of harvesting and threshing, huge amounts of harvest and post-harvest loss are recorded which was estimated to about 12.9%,13.6%, and 10.9% for teff, wheat, and maize (Tamrat, 2016).

2.2. Cleaning/Winnowing

The function of grain separation is to remove/separate the grain from unwanted materials such as broken and yet short straw, pieces of broken ears (cobs), and chaff. The function of grain cleaning is to eliminate foreign or undesirable material such as twigs, leaf and soil particles, empty and weed seeds, dust, chaff, and the like to reduce bulk, improve storability and make grains easier to handle during subsequent processes. The ideal cleaned grain is free from any other materials. The degree to which this is achieved is called the purity (cleaning efficiency), usually measured in percentage (Drambi et al., 2017).

2.3. Method of Grain Cleaning

2.3.1. Traditional Grain cleaning

Cleaning/winnowing involves separating the grain from dirt and chaff/straw by throwing/tossing the chaff-grain mixture in the air to separate the seeds from the chaff. A wooden fork, usually with three or four prongs, is used to separate the grain and straw. A further way of winnowing is carried out by raising a container full of grain and chaff above the farmer's height and letting it drop smoothly so that the blowing wind will separate the seeds from the chaff. Lighter-weight broken grains, straw, and weed seeds are carried by the wind to one side, as the whole and the sound grain falls to the bottom of the winnowing device. The operation requires a continuous brisk wind and several repetitions. The remaining chaff and dirt are removed by fanning with a "sephed" (a flat plate made of grass and used for separating grain from the chaff), by sieving and handpicking (Mohammed & Abreaham, 2018).

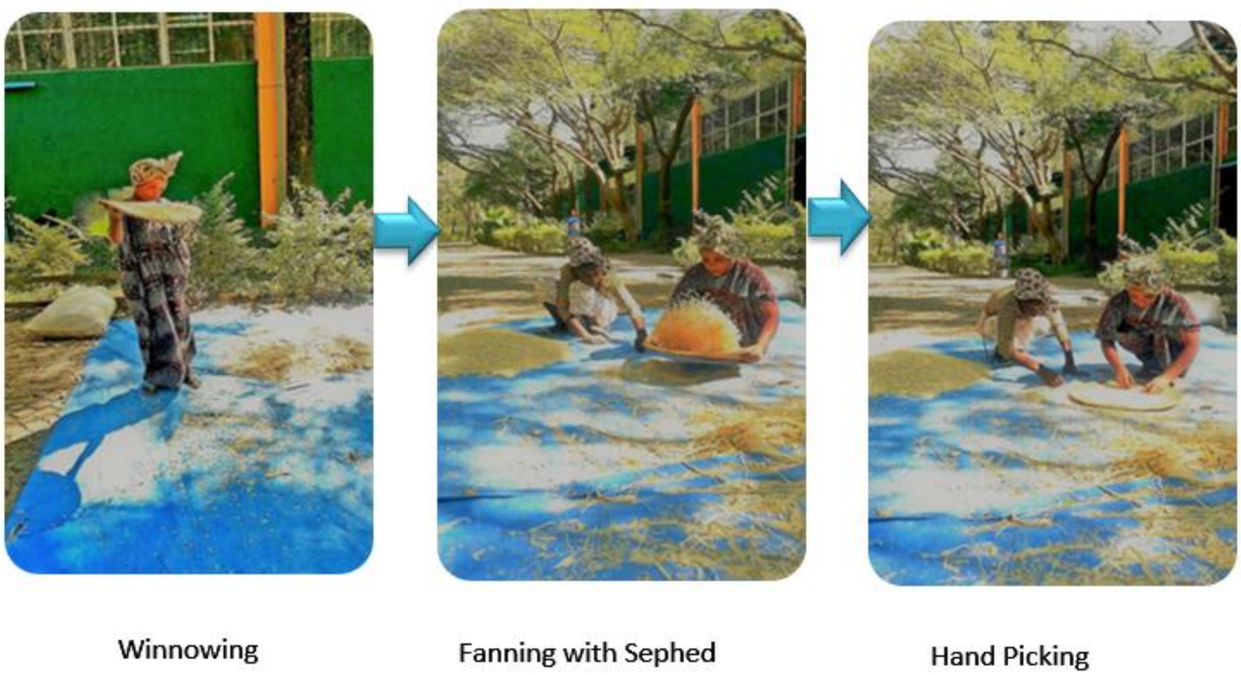


Figure 2.1. Traditional cleaning system.

2.3.2. Pedal Operated Grain Cleaning

The threshed cowpea to be winnowed is firstly fed into the hopper before operating the machine. The container for the clean seeds is then put in the clean seed outlet. Furthermore, a tarpaulin is spread on the floor below the chaff outlet for collecting the chaffs. The operator then sits in position to crank and drive the pedals. Power is transmitted via the chain and sprocket to the fan unit, whose shaft contains the driven sprocket and a v-grooved pulley. A v-belt transmits power from the fan shaft to the feeder shaft which controls the feeder blades via another v-grooved pulley, thereby controlling the feed rate. Cleaned seeds and chaffs are collected through separate outlets (Agbetoye, 2004).

Agbetoye (2004) developed a pedal-operated cowpea winnower machine, which requires 0.05 kW power and it evaluated for performance under three different speeds of feeder rotation of 54 rpm, 108 rpm, and 216 rpm but at a constant fan speed of 108 rpm. Results show that the machine has a maximum feed rate of 374 kg/h and 90% cleaning efficiency. Furthermore, results indicated that the optimum values of feeder speed, feed rate, and cleaning efficiency were 128 rpm, 280 kg/h; and 70% respectively.

According to Yamba et al. (2017), the cleaning effect of the winnower is achieved through the cranking of the prime mover (Bicycle) which drives the fan. The machine was developed to be driven by human muscle power through the pedal on the bicycle. The threshed grain with a moisture content of 13.28% (wb) was fed into the hopper manually and the final cleaned grain was collected under the sieve on a collecting pan, on the other hand, the performance test indicates a cleaning capacity of 90 – 100kg/hr of threshed paddy rice, with a cleaning efficiency of 84.04% and a loss of not more than 13.82%.

2.3.3. Mechanical Grain Cleaning

Air-Screen Cleaner: It is widely used and achieves a high level of physical purity. It is based on screens and a stream of air and separates the seed lot into various quality fractions based on size (length, width, and thickness), weight, and density (Kugbei et al., 2018)

The process entails three main steps

- **Aspiration by suction air** - removal of light particles from the mass of seed. Aspiration can be before or after screening - or both before and after.
- **Scalping** - removal of coarse impurities. They are channeled to an outlet spout, while the remaining seed moves onto grading screens.
- **Fine cleaning** - removal of oversized and undersized seeds and other particles (e.g. stones and sand). The cleaned seed goes into the outlet for bagging or further processing.

There are many makes, sizes, and models of air-screen cleaners ranging from the small, one-fan, single-screen machine to the large, multiple-fan, six- or eight-screen machine with several air columns. Screens are manufactured with many sizes and shapes of openings (Kugbei et al., 2018)

Akinoso et al. (2010) developed the machine whose capacity is determined to be 215 kg/h and it can be approximated to 2 tons/day of 8 working hours and this machine consist of a hopper fitted above a screen box. The screen was made of a steel mesh of varied in size. The top screen prevents foreign material bigger than sesame while the bottom screen retains sesame and allows passage of material smaller than sesame. The screen box was tilted at about 5 degrees to the horizontal and suspended by four tension springs and sitting on four compression springs which were oscillated by an eccentric unit. Next to the screen, there is a vertical separation column connected to a centrifugal fan also power transmission by V belt and pulley arrangement.

Specific gravity separator: The specific gravity separator classifies material according to density or specific gravity. It comes at the end of the seed-cleaning process, grading the cleaned seeds into quality fractions according to their specific weight (USDA, 1968).

The gravity separator comprises an inclined deck made of special square wire mesh mounted on a sturdy support frame. For some light seeds, decks are made of special cloth. The deck is fitted with a fully balanced eccentric drive that causes the deck to oscillate. The deck may be rectangular or triangular. In general, all kinds of decks separate both lighter and heavier components of the seed lot. However, the triangular models tend to remove lighter seed more efficiently, while rectangular models are better at removing the heavier seed below the deck is a fan with a pressurized air supply system. Adjust the angle of inclination of the deck, the oscillation speed, the quantity of air, and the pressure to suit the type of seed (Kugbei et al., 2017).

2.4. Factors Consider in Design of Grain Cleaner

2.4.1. Physical Properties of Grain

Crop separation from unwanted materials is based on differences in physical properties between the crop and materials (Seyoum et al., 2011). The Physical and Engineering Properties of the grain and non-grain material were considered in the determination of the design parameters of the cleaning device. These properties include size, shape, angle of repose, coefficient of friction, and moisture content of the grain be measured or estimated to design separating, cleaning, sizing, and grading machines. Also considered were moisture content, density, and terminal velocity of the grain and non-grain material (Muhammad et al., 2013).

Diameter of the Grain

According to Hamdani et al. (2014), the grains were harvested separately, cleaned manually to remove all foreign matter such as dust, dirt, stones, and chaff as well as immature, broken grains. To determine the dimensions of the grain 100 individual grains were randomly selected and their three principal linear dimensions namely length (L), width (W), and thickness (T) were measured by a digital caliper reading to an accuracy of 0.01mm. The geometric mean diameter (D_g) is calculated as (Sharma et al., 1985).

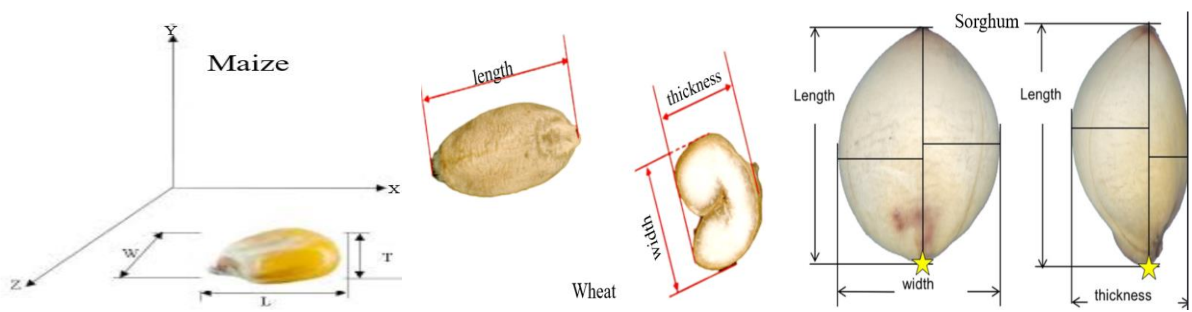


Figure 2.2. Measuring the dimensions of different crops

$D_g = (LWT)^{1/3}$ .... (1)

The arithmetic mean diameter is determined according to the formula given by (Mohsenin & Nuri, 1986):

$D_a = \frac{(L+W+T)}{3}$ ....(2)

Where: L = length of the grain (mm),

W = Width of the grain (mm),

T = Thickness of the grain (mm)

Hamdani et al. (2014) indicated geometric mean diameter was found to be 4.33 ± 0.27, 4.53 ± 0.24, 4.22 ± 0.21, and 4.01 ± 0.24 mm for hulled barley, hulless barley, Sabzaar oats, and

SKO-20 oats respectively. The arithmetic mean diameter was found to be 4.96 ± 0.50 mm for hulled barley, 5.34 ± 0.31 mm for hulless barley, 6.00 ± 0.26 mm for Sabzaar oats, and 5.41 ± 0.44 mm for SKO-20 oats. The mean of geometric wheat is 7.08, 3.27, 2.98, and 4.26 for Length, Width, thickness, and Geometric mean diameter (Tabatabaeefar, 2003).

Simonyan et al. (2007) compiled the engineering properties of sorghum. The average values obtained for the arithmetic mean, geometric mean, and equivalent diameters were 3.32 mm, 3.31 mm, and 3.31 mm, respectively at a moisture content of 8.89 % w.b. At a moisture content of 16.50 %wb, the values were 4.20 mm, 4.16 mm, and 4.18 mm respectively. The average diameter of the sorghum grains calculated by the arithmetic mean, geometric mean, and equivalent diameter methods in the moisture range of 8.89 – 16.50 % w.b are similar.

Brar et al., (2017) Measure a range of major, intermediate and minor dimensions of the seeds of three different varieties maize PMH-1, PMH-10, and PIONEER-3396. For PMH-1 seeds the major dimension varied from (8.18-10.90 mm), intermediate dimension (5.67-8.87 mm), and the minor dimension (4.45-6.98 mm) having mean values of 9.32, 7.22, and 5.89 mm respectively. And the mean geometric diameter of 50 seeds as calculated from these dimensions are found to be (7.33, 7.06, and 7.68 mm) for PMH-1, PMH-10, and PIONEER-3396 maize seeds respectively.

Sphericity of Grain

The sphericity of the grain is used to indicate the shape of the grain. The equivalent diameter for irregular shape material is accepted as the average dimension. Differences between average diameter and measured dimension are determined by the sum of the square of differences. When this difference is divided by the square of the product of average diameter and number of measurements, it gives a fraction for the approach of the slop to an equivalent sphere. Sphericity (φ) was calculated by using the following formula (Soyoye et al., 2018).

$$\varphi = \frac{(L \times W \times T)^{1/3}}{L} \quad \text{.........(3)}$$

Where: φ = Sphericity of grain (%),

L = Length of grain (mm),

W = Width of grain (mm)

T = Thickness of grain (mm)

Kheiralipour et al., (2008) found the sphericity by measured the dimensions of wheat grain using the above equations. The result is in the range of 58 to 60% as the moisture content of the grain increase from 8 to 18%. Also, Gürsoy & Güzel, (2010) were calculated the sphericity to be in the range of 53.3 to 58.7% to wheat varieties. For maize, it is between 65 and 86% (Brar et al., 2017), and for sorghum according to some researchers the range is between 67 to 94% (Kumaran et al., 2018; Simonyan et al., 2007).

Moisture Content of Grain

Grain moisture content is expressed as a percentage of moisture based on wet weight (wet basis) or dry matter (dry basis). Wet basis moisture content is generally used (Hellevang, 1995).

$$Mw(wet\ basis) = \frac{w-d}{w} \times 100 \quad \text{...}(4)$$

$$Md(dry\ basis) = \frac{w-d}{d} \times 100 \quad \text{...}(5)$$

Where: w = wet weight (Kg),

d = dry weight (Kg),

M = moisture content on a percent basis.

Table 2.1. Water in corn and wheat at various moisture levels

	% of moisture content on a wet basis								
Grain	10	11	12	13	14	15	16	17	18
Corn	5.3	5.9	6.5	7.1	7.8	8.4	9.1	9.8	10.5
Wheat	5.8	6.4	7.1	7.8	9.2	9.9	10.6	11.4	12.2

(source: Mcneill and Overhults, 1999)

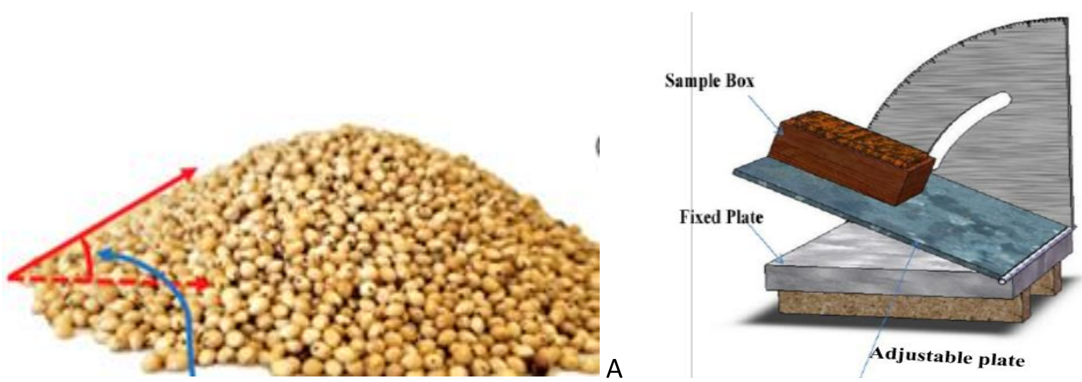
2.4.2. Mechanical Properties

Angle of Repose

The angle of repose is another important physical property used for the characterization of the bulk of particulate foods such as seeds, grains, flours, grits, and fruits. When granular solids are piled on a flat surface, the sides of the pile are at a definite reproducible angle with the horizontal leveled surface (Teferra, 2019).

According to Serpli & Servet (2005), the angle of repose is important for the design processing, storage, and conveying system of particulate material. When the grain is smooth and rounded, the angle of repose is low for very fine and sticky material the angle of repose is high.

Some researchers measure the angle of repose whether by pile the grain on a flat surface or by use apparatus which have an angle on it. Kumaran et al., (2018) used pile type measuring system to identify the angle of repose it is done by bulk material of samples are poured onto a horizontal surface, a conical pile will be formed. The internal angle between the surface of the pile and the horizontal surface is known as the angle of repose. Tabatabaeefar, (2003b) used apparatus which is constructed by an aluminum cylinder of 75 mm diameter and 50 mm height was placed on an adjustable tilting plate faced with the test surface and filled with about 150 g of the sample. The cylinder was raised slightly so as not to touch the surface. The structural surface with the cylinder resting on it was inclined gradually until the box just started to slide down. The angle of tilt was read from the graduated scale.



A B

Figure 2.3. Angle of repose measuring by A) pile on the surface and B) using apparatus

The angle of repose was calculated by measuring the distance between the end of the flow and the height of the box (Adebowale et al, 2012).

$$angle\ of\ repose = \frac{tan^{-1}(horizontal\ distance)}{height\ of\ box} \quad (6)$$

Table 2.2. Angle of repose for different cereal crops

Grain	Average AoR	Minimum AoR	Maximum AoR
Barley	28	24	34
Corn	23	21	26
Oats	28	24	32
Soybeans	25	22	33
Sorghum	29	27	29
Wheat	25	19	38

(Source: Bhadra et al., 2017)

Coefficient of Static Friction

The coefficient of static friction can be determined by different surface areas of a material like plywood, galvanized iron, mild steel, stainless steel, and glass. Most researchers use plywood, galvanized iron, and glass (Tabatabaeefar, 2003a; Soliman et al., 2009; Brar et al., 2017). The friction coefficient for the different sample it was obtained as follows:(El Fawal et al., 2009).

$$Friction\ coefficient = \tan\alpha \quad (7)$$

Where: α = Friction angle

According to Soyoye et al., (2018) coefficient of friction for maize over four surfaces stainless steel, galvanized steel, mild steel, and plywood has also been calculated and ranges from 0.577 to 1.072, 0.364 to 1.111, 0.466 to 1.036, and 0.466 to 0.900 respectively. As well, the average values of static coefficient of friction for sorghum against steel and galvanized iron sheet and wooden plate were 0.20, 0.238, and 0.24 respectively (Surpam et al., 2019). Besides, El Fawal et al., (2009) determined the result for wheat by use glass, plywood, galvanized steel wood, and stainless steel therefore it is 0.28, 0.34, 0.41, 0.49, and 0.33 respectively.

2.4.3. Aerodynamic Properties of Grain and Straw

Mohsenin (2010) indicated that separation of a product from its associated and yet undesirable materials, such as straw and chaff, in an air stream, requires knowledge of aerodynamic characteristics of all the particles involved. Furthermore, it was noted that a range of air velocities, for effective separation of the grain from MOG need to be estimated.

The proper airspeed can be determined from the aerodynamic properties of agricultural materials. These properties are terminal velocity and drag coefficient. If an object is dropped from an appropriate height, the force of gravity will accelerate it until the drag force exerted by the air balances, the gravitational force. It will then fall at a constant velocity called the terminal velocity (Khoshtaghaza & Mehdizadeh, 2006). Consequently, by measuring the terminal velocity of various threshed materials, it was possible to determine and set the maximum possible air velocity in which MOG could extract without losses (Wu et al., 1999).

Some straws tended to float at an oblique angle across the air stream, while others floated slantingly, and this gave them a higher terminal velocity than when floating horizontally. Straws greater than 6cm long and all straws with the node at one end assumed a more vertical orientation in the air stream than others (Gorial & O'Callaghan, 1990).

Terminal velocity and aerodynamic drag resistance may be estimated or measured in the laboratory. Two widely used methods of calculating the terminal velocity experimentally are the drop and suspension methods. The drop method involves lowering the particle from some height by which the particles can attain their terminal velocity after dropping a certain amount. Terminal velocity can be taken from the distance versus time curves where it starts to become linear (Çarman, 1996).

Table 2.3. Terminal velocity for different cereal crops

Grain	Mass (mg)	Terminal velocity (m/s)	Drag coeff C_d
Wheat	30.2	7.8	0.85
Barley	37.6	7.5	0.98
Soybean	197	14	0.5
Rye	33.4	7.8	0.87
Maize	321.2	11.6	0.81
Sorghume	30.9	9.7	0.59

(Source: Gorial and O'Callaghan, 1990)

2.4.4. Machine Unit Operation

Sieve Oscillation and Till of Sieve

Sieve slope, sieve oscillation, and airspeeds were the main factors that affect separating, cleaning and total losses. Both separating and leaning efficiency and total loss increased with increasing airspeed, sieve tilt angle, and sieve oscillation (Ebaid, 2005). The separation efficiency increased significantly with an increase in sieve slope at small stroke length and decreased at high stroke lengths. In general, there was a decrease in cleaning efficiency with increasing sieve oscillation frequency along the sieve length. The decrease in cleaning efficiency with increasing sieve oscillations may be due to less resident time of materials to be separated on the sieve (Awady et al., 2003).

Okunola et al., (2015) described that total efficiency was highest at a 3° tilt angle. It decreases initially with a gentle gradient at successive tilt angle until at 6°where the gradient becomes steep to 8°. This implies that the higher the tilt angle the lower the total separation efficiency. The product purity was highest at 98% at 3° tilt angle and the curve decreases gradually to 93% at 5° and then increases to 96% at 6° and remains constant till 8° tilt angle machine performance occurred at 3^o and 4^o tilt angles having the highest total efficiency and product purity values.

Harrison & Blecha (1983) described that the transport of particles along with oscillating sieves, which is a function of sieve oscillation frequency, affects the efficiency of the process and affects the metering of particulate substances along with the sieve.

Feeding Rate of the Grain

Inverse relationships had been reported between separating and cleaning efficiencies and feed rate. The decline of inefficiency has been a common phenomenon with increasing feeding rates. The inverse relationship between cleaning efficiency and feed rate was indicated to be due to the increasing load on the sieve that restricted the free movement of grains and undesired materials (Abayineh, 2015).

Muhammad et al., (2013) indicated the effect of feed rate on cleaning efficiency and grain loss of sorghum, soybean, and millet. Cleaning efficiency for each of the three crops with the coefficient of determination of 0.95, 0.93, and 0.96 respectively, and also they attained 100 % cleanliness at feed rates of 13.5, 11.5, and 5.8 kg/s respectively. Since as the feed rate increased, the material flowing across the air current forms a thicker blanket and, therefore, increasingly more difficult for the air current to penetrate and flush out the unwanted material and subsequently decreased the cleanliness as more of the debris passed through the sieve holes together with the grain.

Cleaning efficiency and total losses were affected by increasing feed rate when cleaning rice (at a higher feed rate), it required greater sieve length and took a long time for the grain to be separate from MOG. This was considered to be due to the denseness of the MOG, which makes diffusion of the grain through it very difficult. The increase in cleaning loss with increasing feed rate was indicated to be due to high intensity of load with MOG on the sieve that had a matting effect on the sieve and possibly blocked sieve holes further indicated that if the feed rate is increased (higher), the materials are no longer supported aerodynamically, and form a mat on the sieve, increasing grain losses (Awady et al., 2003).

Air Flow Velocity

Afolabi et al., (2019) reported the effect of air velocity on cleaning loss, cleaning efficiency, and separation loss was significant, but its effect on the separation efficiency was not significant as analyzed by using 9.12-12.30$^{\mathrm{m}}/_{\mathrm{sec}}$. Separation efficiency cleaning loss increased a corresponding increase in air velocity but cleaning efficiency, separation loss increase with the decrease of air flow velocity.

Pawankumar et al., (2018) used three levels of air velocity (2.9, 2.4, and 2.0$^{\mathrm{m}}/_{\mathrm{sec}}$) when fully open, half-open, and fully closed to determine the influence on the cleaning efficiency of grain. Therefore, the result determined that increase in cleaning efficiency with an increase in air velocity since the larger quality of chaff was sucked by the blower at the higher air velocities, thus reducing the quality of the chaff for further separation.

2.5. Design and Component Consideration

2.5.1. General Consideration in Machine Design

Machine design factors should be considered before further design and start production of the machine. Therefore, design factor consider to be important were as (Khurmi & Gupta, 2005b);

- **The motion of the parts or kinematics of the machine.** The successful operation of any machine depends largely upon the simplest arrangement of the parts which will give the motion required.

- **Selection of materials**. A designer must have a thorough knowledge of the properties of the materials and their behavior under working conditions. Some of the important characteristics of materials are strength, durability, flexibility, weight, resistance to heat and corrosion, ability to cast, welded or hardened, machinability, electrical conductivity, etc.
- **Frictional resistance and lubrication**. There is always a loss of power due to frictional resistance and it should be noted that the friction of starting is higher than that of running friction. It is, therefore, essential that careful attention must be given to the matter of lubrication of all surfaces which move in contact with others, whether in rotating, sliding, or rolling bearings.
- **Workshop facilities**. A design engineer should be familiar with the limitations of his employer's workshop, to avoid the necessity of having work done in some other workshop. It is sometimes necessary to plan and supervise the workshop operations and to draft methods for casting, handling, and machining special parts.
- **Cost of construction.** The cost of construction of an article is the most important consideration involved in the design

2.5.2. Design of Cleaning Machine

Sieve open type and size

Screening is the common method for the separation of particulate material according to size and is widely used for cleaning and sorting agricultural products. Screening results depend on a large number of variables (Feller & Foux, 1975).

During cleaning the screen is two types of screens flat and rotary, are generally used. Besides, the screens could be rotated or vibrated to bring all the particles into contact with the openings. For flat screen, three different mechanical motions are involved in the separation process. These motions are oscillation, vertical vibration, and gyration.

According to Abiodun et al. (2018), the basic requirement for grain separation is that grain movement on the sieve surface towards the lower end of the sieve in a vibration mode. The sieve to and for motion is affected by an eccentric connection to the power source also The sieves as a result of the eccentric drive connection to the casing experience horizontal oscillation and small vertical motion. The combined motion ensures that the grains move or slides down from the screen and at the same time bounce to stir the whole bulk.

The basic sieve for cleaning elongated grain is a slotted top sieve and a slotted bottom sieve. In special separations, it may be necessary to pass such grain through a round-hole top sieve or over some sieve other than a slotted bottom sieve, but generally, slotted top and bottom sieves are used (Tabatabaeefar et al., 2003).

Table 2.4. Recommended sieve openings sizes for different grains

Screen opening in mm			
Crop	**Top screen**	**Middle screen**	**Bottom screen**
Wheat	10r, 9r, 8r, 6.5r	4s, 4.5s, 6.3s	2s, 2.25s
Barley	14r, 10r	4.5s, 3.5s	1.7s, 1.4s
Soybean	6r, 7r	8r	1.7s, 2s
Sorghum	10r	4.25s, 4.5s	1.7s, 2s
Haricot bean	10r	6.5s	2.75s

(Source Abayineh, 2015)

Fan type and selection

There are two primary types of fans: centrifugal and axial. These types are characterized by the path of the airflow through the fan. Centrifugal fans use a rotating impeller to increase the velocity of an airstream. As the air moves from the impeller hub to the blade tips, it gains kinetic energy. This kinetic energy is then converted to a static pressure increase as the air slows before entering the discharge. Centrifugal fans are capable of generating relatively high pressures. Axial fans, as the name implies, move an airstream along the axis of the fan. The air is pressurized by the aerodynamic lift generated by the fan blades, much like a propeller and an airplane wing. Although they can sometimes be used interchangeably with centrifugal fans, axial fans are commonly used in "clean air," low-pressure, high-volume applications. Axial fans have less rotating mass and are more compact than centrifugal fans of comparable capacity. Additionally, axial fans tend to have higher rotational speeds and are somewhat noisier than in-line centrifugal fans of the same capacity (LBNL, 2003)

2.5.3. Sliding Crank Mechanism

Slider-crank Mechanism is an arrangement of mechanical parts designed to convert straight-line motion to rotary motion, as in a reciprocating piston engine, or it is used to convert rotary motion to straight-line motion as in a reciprocating pump. Internal combustion engines are a common example of this mechanism, where combustion in a cylinder creates pressure that drives a piston. The linear motion of the piston is converted into rotational motion at the crank through a mutual link, referred to as the connecting rod. As the geometry of the crank forces the conversion of linear motion to rotational, shaking forces are generated and applied to the crank's housing. The shaking forces result in vibrations that impede the operation of the engine (Mukkawar et al., 2018).

The precise relationship between the input and output is important for the slider-crank mechanism. For example in certain applications, the rotation of the crank is considered as the input and the displacement of the slider is considered as the output. Since the mechanism is not manufactured perfectly and cracks always exit which is known to be the source to reduce system reliability and precision (Cheng & Liu, 2018).

Besides, the Slider-crank mechanism is used to transform rotational motion into translational motion through a rotating driving beam, a connection rod, and a sliding body to get great output by using a flexible body for the connecting rod. The sliding mass is not allowed to rotate and revolute joints are used to connect the bodies. (Ashok, 2007).

An in-line slider-crank mechanism has the crank pivot coincident with the axis of the sliding motion of the piston pin. An in-line slider-crank mechanism is illustrated in the figure below. The stroke, $|\Delta R4|max$, is defined as the linear distance that the sliding link exhibits between the extreme positions. Because the motion of the crank (L2) and connecting arm (L3) is symmetric about the sliding axis, the crank angle required to execute a forward stroke is the same as that for the return stroke. For this reason, the in-line slider-crank mechanism produces balanced motion.

The book assuming that the crank is driven with a constant velocity source, as an electric motor, the time consumed during a forward stroke is equivalent to the time for the return stroke. Although the design of an in-line slider-crank mechanism involves determining the appropriate length of the two links, *L*2 and *L*3, to achieve the desired stroke, |Δ**R**4|max.

As can be seen from the figure above the stroke of the in-line slider-crank mechanism is twice the length of the crank. That is, the distance between *B*1 and *B*2 is the same as the distance between *C*1 and *C*2. Therefore, the length of crank, *L*2, for an in-line slider-crank can be determined as follows (Myszka, 2012):

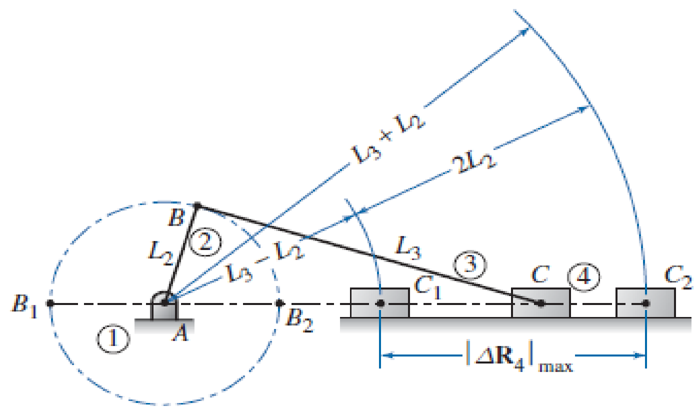


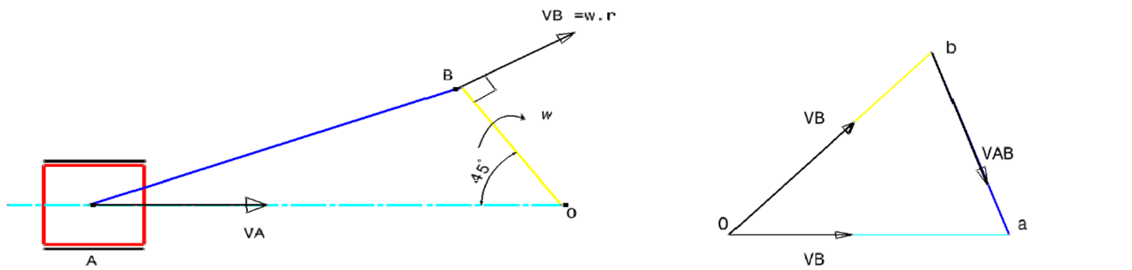
Figure 2.4. In-Line slider-crank mechanism.

$$L_2 = \frac{|\Delta R_4|_{max}}{2} \quad \text{...} (8)$$

Velocity in Slide Crank Mechanism

Velocity analysis involves determining "how fast" certain points on the links of a mechanism are traveling. Velocity is important because it associates the movement of a point on a mechanism with time (Myszka, 2012).

Slid crank mechanism is showing in Figure 2.5 (A) slider A is attached to connecting rod AB. Let the result of crank OB be r and let it rotate in a clockwise direction about the point O with uniform angular velocity ω (rad/s) there for the velocity of B i.e. V_B is known in magnitude and direction. The slider reciprocates along the line of stroke AO (Khurmi & Gupta, 2005a).



A B

Figure 2.5. A) Slider crank mechanism and B) velocity diagram.

The velocity of slider A (i.e.V_A) may be determined by relative velocity as discussed below:

1. From any point *o*, draw vector ob parallel to the direction of V_B (or perpendicular to OB) such that ob = V_B =ω. r, to some suitable scale, as shown in Fig (*b*).
2. Since AB is a rigid link, therefore the velocity of A relative to B is perpendicular to AB. Now draw vector ba perpendicular to AB to represent the velocity of A with respect to *B i.e.* V_{AB}.
3. From point o, draw vector oa parallel to the path of motion of the slider *A* (which is along with AO only). The vectors ba and oa intersect at *a*. Now oa represents the velocity of the slider A *i.e.*V_A to the scale.

The angular velocity of connecting rod AB (ω_{AB}) may be determined as follow:

$$\omega_{AB} = \frac{V_{BA}}{AB} = \frac{ab}{AB} \qquad \text{(Anticlockwise about A)} \quad (9)$$

The direction of vector ab (or ba) determines the sense of ω_{AB} which shows that it is Anticlockwise.

Acceleration in the Slider Crank Mechanism

Acceleration analysis involves determining how certain points on the links of a mechanism are either “speeding up” or “slowing down.” Acceleration is a critical property because of the inertial forces associated with it. Besides an important mechanism design is to ensure that the strength of links and joints is sufficient to withstand the force imposed on them (Myszka, 2012).

A slider-crank mechanism is shown in figure 2.6 (A). Let the crank OB makes an angle with the inner dead center (I.D.C) and rotates in a clockwise direction about the fixed point O with uniform angular velocity ω_{BO} rad/s (Khurmi & Gupta, 2005a).

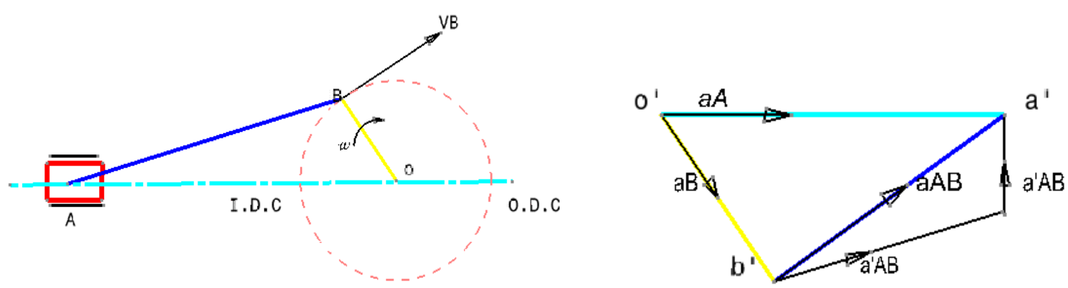
The velocity of B with respect to O or velocity of B (because O is a fixed point),

$V_{BO} = V_B = \omega_{BO} \times OB$, acting tangentially at B .... (10)

The centripetal or radial acceleration of B with respect to O or acceleration of B (because O is a fixed point),

$a^r_{BO} = a_B = \omega^2_{BO} \times OB = \frac{V^2_{BO}}{OB}$ .... (11)

- A point at the end of a link that moves with constant angular velocity has no tangential component of acceleration.



A B

Figure 2.6. A) Slider crank mechanism and B) acceleration diagram

The acceleration diagram, as shown in Fig. 8.3 (*b*), may now be drawn as discussed below:

1. Draw vector o' b' parallel to BO and set off equal in the magnitude of $a^r_{BO} = a_B$, to some suitable scale.
2. From point b', draw vector b'x parallel to BA. The vector b'x represent the radial component of the acceleration of A with respect to B whose magnitude is given by:

$a^r_{AB} = {V^2_{AB}}/{BA}$ .... (12)

Since point B move with constant angular velocity, therefore there will be a not tangential component on acceleration.

3. From point x, draw vector xa’ perpendicular to b’x (or AB). The vector xa’ represent the tangential component of the acceleration of A with respect to B i.e. a_{AB}^{t}.

   When a point moves along a straight line, it has no centripetal or radial component of acceleration.

4. Since point A reciprocates along with AO, therefore the acceleration must be parallel to velocity. Therefore from o’, draw o’a’ parallel to AO. Intersecting the vector xa’ at a’.

   Now the acceleration of the slider A (a_A) and a_{AB}^{t} maybe measured to the scale.

5. The vector b’a’, which is the sum of the vector b’x and xa’, represent the total acceleration of A with respect to B i.e. a_{AB}. The vector b’a’ represents the acceleration of the connecting rod AB.

6. The angular acceleration of the connecting rod AB may be obtained by dividing the tangential component of acceleration of A with respect to B (a_{AB}^{t}) to the length of AB. In other words, the angular acceleration of AB.

   $\alpha_{AB} = a_{AB}^{t}/AB$ (Clockwise about B) .... (13)

2.6. Consideration of Power Transmission Components

2.6.1. Belt and Pulley Design

The power transmission systems used in the design of the machine are belts and pulleys, sprocket wheels and chains, shafts, gears, universal joints, and clutches. Though, *V*-belt is mostly used in factories and workshops where a great amount of power is to be transmitted from one pulley to another when the two pulleys are very near to each other. And also, flexible machine elements, perhaps V-belt drives have the widest industrial application. Hence, one or more V belts may be used on a drive, as required, to transmit power because of their simplicity and ease of design, selection, installation, maintenance and repair, and their low cost (Richard & Kelth, 2011).

Selection of belt

The nominal pitch length of the belt from an electric motor/engine to the fan from the fan to eccentric wheel and also from eccentric wheel to brush have to determine to know the actual belt size. Therefore, nominal pitch length is calculated by using the following formula (Khurmi & Gupta, 2005b).

$$L = 2C + \frac{\pi}{2}\left(D_1 + D_2 + \frac{(D_2 - D_2)}{2C}\right) \quad \text{.......} \quad (14)$$

Where: D_1 and D_2 = diameters of driving and driven pulleys (m),

C = center distance between two adjacent pulleys (m),

L = length of belt (m),

n_1 = Speed in RPM of the smaller pulley

n_2 = Speed in RPM of the bigger pulley

The center distance between the two pulleys must not be less than twice the diameter of the larger pulley (Richard &.Kelth, 2011).

$$C = 2D_2 \quad \text{.......} \quad (15)$$

The wrap angle on the smaller pulley is determined using the below equation.

$$\alpha_1 = 180 - 2\,sin-1\left(\frac{D_2 - D_1}{2C}\right) \quad \text{.......} \quad (16)$$

$$\alpha_2 = 180 + 2\,sin-1\left(\frac{D_2 - D_1}{2C}\right) \quad \text{.......} \quad (17)$$

Where: - α_1 = angle of wrap on smaller pulley (degree).

α_2 = angle of wrap on larger pulley (degree).

The tension of the belt determines by using the following equations to determine the power transmitted by the belt to the pulley from the primary power source. Therefore there are two types of tension the belt tight side and the slack side (Khurmi & Gupta, 2005b).

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\frac{\mu\theta}{\sin\alpha}} \quad \text{.......} \quad (18)$$

Where: m = mass of belt (kg.m-1),

σ = maximum allowable stress of belt (MPa),

T_1= tension of the belt on the right side (N),

T_2= tension of the belt on the slack side (N),

μ = coefficient of friction between the belt and the pulley (0.43),

θ = lap angle on smaller pulley (rad) and

α= groove angle (34°).

Tension on the tight side of the belt can be calculated using the following formula from Khurmi & Gupta, (2005b).

$$T_1 = T_{Max} - T_C \quad \text{.......... (19)}$$

Where: T_{Max}= maximum of the belt (N).

T_C = centrifugal tension of the belt (N).

The maximum tension of the belt, centrifugal tension, mass per unit length, and cross-sectional area can be found by using Eqn (20) (Richard & Kelth, 2011).

$$T_{Max} = A.\sigma \quad \text{.......... (20)}$$

$$T_c = mv^2 \quad \text{.......... (21)}$$

$$m = \rho A \quad \text{.......... (22)}$$

$$A = \frac{(b-x)}{2}t + xt \quad \text{.......... (23)}$$

Where: A= cross-sectional area of a belt

m = mass per unit length of a belt (kg/m),

σ = maximum allowable stress of belt (MPa),

v = speed of belt (mm.s-1),

b = top width of the belt (mm),

x = bottom width of the belt (mm),

t = thickness of the belt (mm) and

ρ = density of belt material (Rubber) (kg.m-3).

The power requires to operate the grading machine is the total power required to operate the cleaning machine units. The total power required for cleaning can be determined by using Eqn. (25) (L Robert, 2004).

$P = (T_1 - T_2) \times V$ .... (24)

$P_t = P + 0.1P$ .... (25)

Where: Pt = total power required to drive the machine (Kw).

P = power required to drive the system (Kw).

V = the velocity of the belt (m.s-1) and

P = the power transmitted by a belt, (watts).

Design of pulley

As stated in Khurmi & Gupta, (2005b) the pulleys are used to transmit power from one shaft to another using flat belts, V-belts, or ropes. Since the velocity ratio is the inverse ratio of the diameters of driving and driven pulleys, therefore the pulley diameters should be carefully selected to have the desired velocity ratio. The pulleys must be in perfect alignment to allow the belt to travel in a line normal to the pulley faces.

The speed of the cleaning machine unit was related by using Eqn. (26) (Richard & Kelth, 2011).

$\frac{N_1}{N_2} = \frac{D_2}{D_1}$ .... (26)

Where: N_1 = maximum angular speed of driven pulley (rpm),

N_2 = maximum angular speed of driving pulley (rpm).

The width of the pulley (face width) 'B' consider being greater than the width of the belt by 25% and the Eqn. (27) suggested by Richard & Kelth, (2011) can be used to determine the width of the pulley.

$B = b + 0.25b = 1.25b$ .... (27)

2.6.2. Shaft Design

A shaft is a rotating machine element that is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. Also, to transfer the power from one shaft to another, the various members such as pulleys, gears, etc., are mounted on it. These members along with the forces exerted upon them cause the shaft to bend (Khurmi & Gupta, 2005b).

The diameter of the shaft can be estimated by using the following Eqn.

$$d^3 = \frac{16}{\pi S_a}\sqrt{(C_b K_b M_b)^2 + (C_t K_t T)^2} \quad \text{.......} \quad (28)$$

Where: d= diameter of the shaft (mm),

K_b= combine shock and fatigue factor applied to bending moment,

M_b= bending moment (N-m),

K_t= combine shock and fatigue factor applied to torsional moment,

T = torsional moment (N-m),

C_b= stress concentration factor for bending,

C_t= stress concentration factor for a torsional moment and

S_a = shear strength (N/m2).

The resultant bending moment on the shaft can be determined by using the following Eqn.

$$M_b = \sqrt{(M_V)^2 + {M_H}^2} \quad \text{.......} \quad (29)$$

Where: Mb = resultant bending moments (N),

MV = maximum bending moments on a vertical plane (N) and

MH = maximum bending moments on a horizontal plane (N).

Torque on the shaft can be determined by using the following Eqn. which determine by (Khurmi & Gupta, 2005b).

$T = (T_1 - T_2) \times \frac{D}{2}$ .... (30)

Where: T = torque on a shaft (N-m),

T1 = tight side tension of a belt (N),

T2 = slack side tension of a belt (N) and

D = diameter of the driven pulley (m).

2.6.3. Selection of Bearing

A bearing is a machine element that supports another moving machine element (known as a journal). It permits a relative motion between the contact surfaces of the members while carrying the load (Khurmi & Gupta, 2005b). Bearing size can be selected by determining using maximum resultant force on it, bore size, and desire maximum lifespan.

$R = (\sqrt{R_{BX}^2 + R_{BY}^2}\ )$ .... (31)

The design load of a bearing can be calculated using the following formula (L Robert, 2004).

P_d=VR .... (32)

Where: P_d= load of the bearing (N).

V= rotation factor 1.0 if the inner race of the bearing rotates and 1.2 if the outer race rotates.

R = radial load (N)

The design life of a bearing, L_d can be estimated using Eqn. (33)

L_d= operation hour × rotational speed .... (33)

Dynamic load rate can be determined by using Eqn. (34) (Richard G & J.Kelth, 2011).

$C = P_d \left(\frac{L_d}{10^6}\right)^{\frac{1}{k}}$ .... (34)

Where: C= basic dynamic load of bearing (N).

R_{BX}= maximum resultant force acting on a bearing at point B on a horizontal plane (KN),

R_{BY}= maximum resultant force acting on a bearing at point B on a vertical plane (KN),

R = maximum resultant reaction force on the bearings at point B (KN),

N = revolution of the shaft (rpm),

K= 3 for ball bearing and 10/3 for roller bearings,

L_d= design life of the bearing.

2.7. Research Gap

In post-harvesting operation done different agricultural activity, one of them is which an eventual process cleaning activity. Even though, in Ethiopia, most of the country parts use traditional cleaning systems, mostly this work done by women.

Different researchers in the world indicate that cleaning is one of the important operations performed before reached to storage and also further processing operation to increase the quality of the crop. So cleaning operation can be done by traditional, pedal-operated machine and mechanical machine with some efficiency of cleaning.

Agbetoye, (2004) develop pedal operated cowpea winnower machine which as maximum feed rate of 374kg/h and 90% cleaning efficiency. In addition, Yamba et al. (2017) develop pedal operated cleaning machine which working with cranking of the prime mover (Bicycle)which drive the fan which as 84% cleaning efficiency. Akinoso et al. (2010) developed the machine whose capacity is determined to be 215 kg/h and it can be approximated to 2 tons/day of 8 working hours.

Different cereal crop cleaning and grading machines develop in the world. Different researchers study the performance of the cleaning machine. The researcher uses different factors to determine cleaning efficiency, cleaning loss, separation efficiency, and separation loss. These factors including feeding rate, angle of inclination, and air velocity.

CHAPTER THREE

3. MATERIALS AND METHOD

3.1. Study Area Description

The grain cleaning machine prototype was fabricated and test in Melakasa agricultural research center which was found 117 km from Addis Ababa. The center was established in 1969 as a national horticulture research center with staff members of five personnel and land area not exceeding five hectares now it covers 275ha. Over the years, other important research programs and commodities have been included in the research program that contributed to improved rain-fed and irrigated agricultural development with the main focus on dry and lowland areas of moisture stress and irrigable areas of the country. The center is located at (Latitude/longitude) 8º24'N and 39º21'E having an Altitude of 1550 m.a.s.l.

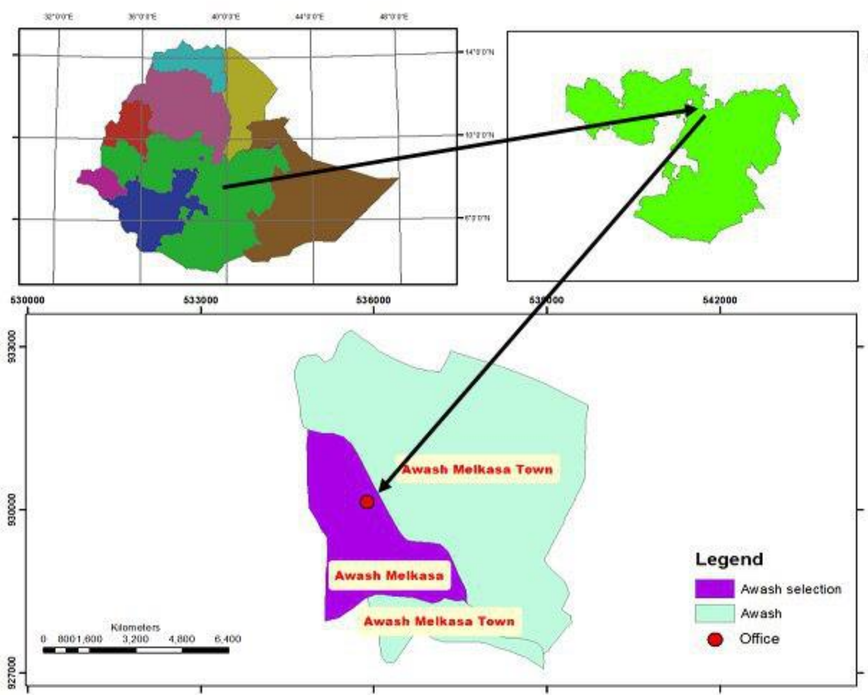


Figure 3.1. Location map of the study area

3.2. Material, Machinery, and Instruments Used

3.2.1. Material Used

The following materials used for the accomplishments fabrication of the grain cleaning prototype based upon their strength, availability, and flexibility:

Table 3.1. The material used in the development of the cleaning machine

No	Materials		Quantity	Specification
1	Square Hollow Section Steel		9m	The dimension of 40×40×3mm to compute the mainframe to get great strength.
2	Angle Iron		9.2m	The dimension of 30×30×3mm for the sieve holder frame.
3	Sheet Metal	1.5 thickness	$1.3m^2$	Used to hopper and fan case.
		1.25 thickness	$2m^2$	Used to outlets and fan blade.
4	U channel		1.4m	Used to sliding operation of the brush
5	Flat iron		3.7m	The dimension of 30×5mm for sieve and mainframe linkage and hopper holder.
6	Shaft		3pcs	Machined from mild steel. Used to a fan, sieve oscillation, and brush slider.
7	V-belt		3pcs	Made from rubber to transmitted power to fan, sieve oscillation, and brush slid.
8	Pulley		5pcs	Made from Aluminum Cast with a diameter of two 80mm, two 350mm, and 170mm.
9	Bearing		6pcs	Made from chromium alloy steel which is attached to rotational parts.
10	Electrode		3pack	Used to welding different parts.
11	Anti-rust		1kg	Used to prevent corrosion.
12	Paint		1gallon	Used after applying anti-rust to make the machine attractive.

12	Cut-off disc (350mm)	2pack	Used to cut parts to appropriate size
13	Sieve	$3.9m^2$	Used to separate the grain from impurity

3.2.2. Machinery and Instrument Used

Machinery used to develop the prototype

Table 3.2. Machinery used to developed prototype

No	Machinery	The function of the machinery
1	Cutter	Cutting
2	Grinder	face the welding parts and cut
3	Lathe	turning and facing
4	Shaper	make shape
5	Drill	make a hole
6	Welding	Weld
8	Rolling	roll sheet metals
9	File	Smooth rough edge
10	Bending	Sheet metal bender

The instrument has been used to measure the different characters of grain and prototype activity.

Table 3.3. Instruments to measure grain properties and prototype components activity

No	Instrument	The function of the instrument
1	Measuring tape	Measure the vertical and horizontal distance
2	Stopwatch	Measure time during an experiment
3	Anemometer	Measure the speed of air
4	Caliper	Measure the size of the grain
5	Digital balance	Measure Weight of the grain
6	Grain moisture meter	Measure the level of moisture

3.3. Determination of Physical and Mechanical Properties of Grain

3.3.1. Size and Shape

To determine the physical properties of wheat, sorghum and maize, took 100 randomly selected three varieties from each crop type which is free from any damage. Besides, the sample was get from the Melkassa Agricultural Research center and Asela Agricultural Engineering Research Center.

The moisture content of each crop is measured by using a grain moisture meter before measuring the size. Therefore, the three principal dimensions, namely length, width, and thickness of each grain were measured by using a digital caliper with an accuracy of 0.01 mm and measure the range of 0.00-150.00 mm.

Figure 3.2. Digital Vernier Caliper

Geometric Mean Diameter

Geometric mean diameter was determined by using the three principal dimensions of the grain by using Eqn. (1), this is beyond to determine the sieve perforated size for the upper, intermediate, and bottom of the machine cleaning units.

Sphericity

The sphericity of cereal crops was determined by the set of standard shapes. The shape of agricultural products usually expressed in sphericity. Therefore, it can be determined by using the size of length, width, and thickness of the sample of grain in Eqn. (3).

Moisture Content

The moisture content of the grain was conducted by using a grain measuring meter device with the highest accuracy measurement.

Figure 3.3. Grain moisture content measuring device

3.3.2. Mechanical Properties of the Grain

Angle of Repose

The angle of repose of cereal crops is important to the design of hoppers for cleaning machines. Hence, it was determined by using a device that has been constructed in Melkassa research center. The device consists of galvanized sheet metal, wood, angle meter, and rope with dimensions of 300×200×50 mm. The operation of measuring is taken by 10gm samples randomly selected from each crop and fill on the surface of the device after that by pulling the rope slowly which connected with galvanized sheet metal the grain start to flow down there for the data were a recorded at which angle the grain start flow down.

Coefficient of Friction

The coefficient of friction between the surface of the galvanized metal and the grain was determined by using Eqn. (7).

3.4. Description and Principle Operation of Grain Cleaning Machine

3.4.1. Description of the Grain Cleaning Machine

The parts of the prototype of multi-grain cleaner comprised; hopper, cleaning units, grain and impurities outlet, mainframe, sieve frame, brush slider frame, pulley, v-belt, bearing, and electric motor. Each part was constructed and assembled step by step to facilitate the construction of the system.

The cleaning unit is a fan and sieve, the sieve has three regions upper, middle, and lower according to the size and geometric and sphericity of each grain. Thus, the upper sieve selected has a round hole at 11 mm, 5 mm, and 4.5 mm for maize, sorghum, and wheat respectively to remove larger particles; the middle one has round type perforation for sorghum and maize but wheat, it has a slot type at 3 mm, 7mm and 3.5 perforation size. The lower sieve has small types of perforation to remove smaller objects including; sand and stone also damaged (broken) grain. The type of perforation is round for all of the grain, although it has a 2 mm size for both wheat and sorghum plus 4 mm to maize.

Besides, the cleaner has an electrical motor which is a prime mover, and power transmission (pulley and v-belt) assemble vertically aligned with the shafts of a fan, sieve oscillation, and brush slider which is mounted on the mainframe. Hence, each rotates at speeds of 687, 313, and 98 rpm respectively which get from the speed of a motor at 1140 rpm.

The mainframe of the machine dimension is 800 mm width, 1000 mm height, and 800 mm length, and also square hollow section steel with high strength was used. This frame embraces all parts of the machine including fan, sieve frame, hopper, brush frame, chaff outlet, grain and other impurities outlet, and power transmission units.

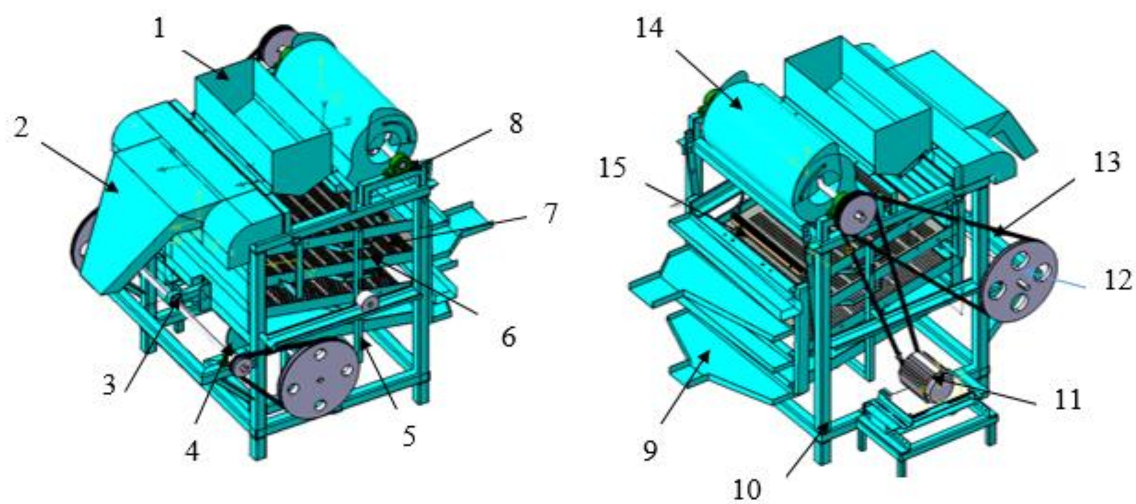


Figure 3.4. Isometric view of grain cleaning machine

The key component of the machine

1- Hopper	6- Sieve frame	11- Motor
2- Chaff outlet	7- Sieve	12- Pulley
3- Eccentric wheel	8- Bearing	13- Belt
4- Crank	9- Outlet	14- Fan house
5- Brush slider frame	10- Main frame	15- Brush

3.4.2. Principle of Operation

The prototype of the cleaner was designed and fabricated to be operated on the principles of rotating a fan, eccentric wheel, and slid crank units. For that reason, the operation of the machine began with feed the grain into the hopper that has been fully closed. Beyond that, the rotational motion of the cleaning units started from the primary mover (motor) to the fan shaft by pulley and v-belt then the rotational motion cycle continues to eccentric wheel from fan and goes to brush slider by eccentric wheel shaft. After all, by releasing the closed hopper the grains with impurities started to drop down; the air from the fan float the light particles horizontally to the chaff outlet. Though, the heavier objects including grain fall over to the upper sieve: the upper sieve remove the larger objects through the outlet at that moment the grain that is smaller in size pass through the perforation to the middle sieve: ahead the middle sieve receives and isolated and pass the smaller grain to the lower sieve as well as eliminate the larger grains over the outlet. After all the lower sieve separates the larger grain from the smaller grains. Which is not removed by a fan as a result of the mass of the particle then it passes through the sieve to the plate which is placed underneath all of the sieves. Furthermore, during the operation of all of this, the brush slides forward and backward under each sieve to prevent the clogging of the sieve hole.

3.5. Design the Components of Cleaning Machine

3.5.1.Hopper Design

The hopper was designed to properly feed the mixture of grain with impurity into the cleaning units. Therefore, the material used in the construction of the hopper was 1.5 mm thick sheet metal. Besides, the hopper has a shape of rectangular section and a trapezoidal section. The trapezoidal section was designed depends upon the average value of an angle of repose 25° for all crops. Hence, to control the followability of the grain there is a gate to open and close the hopper.

The volume of the hopper is determined by using the following equation (Sharma & Mukesh ., 2010).

$$V_H = V_A + V_B \quad \text{.......} \quad (35)$$

$V_H = 0.2 + 1.7\ m^3 = 0.23\ m^3$

Where: V_H = volume of hopper (m^3).

V_A = volume of rectangular section (m^3).

V_B = volume of trapezoidal section (m^3).

$$V_B = \frac{(a+b)}{2} \times h \times l_b \quad \text{......(36)}$$

Where: a = bottom width of the hopper (m).

b = top width of hopper (m), b = a + 2l

l_b = length of the hopper (m),

h = height of trapezoidal section (m),

l = width of a trapezoidal section (m), where: $l = h \cot \theta$

θ = angle of repose (degree).

Putting the value of b in Eqn. (36)

$$V_B = \frac{(a+a+2l)}{2} \times h \times l_b \quad \text{...... (37)}$$

Also

$$V_B = \frac{(2a+2h \cot \theta)}{2} \times h \times l_b \quad \text{...... (38)}$$

$$V_B = (a + h \cot \theta) \times h \times l_b \quad \text{...... (39)}$$

$V_B = (0.02 + (0.06 \cot 25°)) \times 0.06 \times 0.6$

$V_B = 0.2\ m^3$

$$V_A = h_r \times l_r \times w_r \quad \text{...... (40)}$$

$V_A = 0.2 \times 0.6 \times 0.27$

$V_A = 0.03\ m^3$

Where: - h_r = height of rectangular section (m).

l_r = length of rectangular section (m).

w_r = width of rectangular section (m).

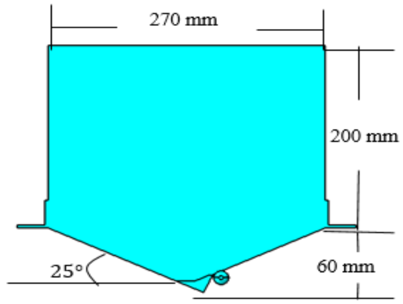


Figure 3.5. Side view of a hopper with dimension

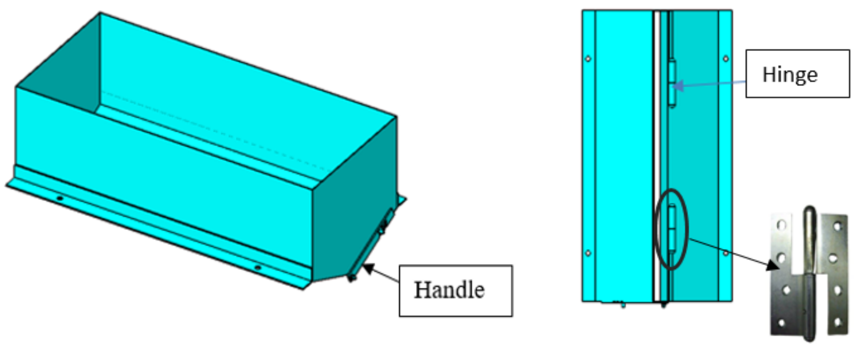


Figure 3.6. Isometric and bottom view of a hopper

3.5.2. Frame Design

The frame is constructed with 40 × 40 × 3 mm square hollow section steel which is high in strength. The dimension of the frame is as indicated in Figure 3.7; height is 1000 mm, length 700 mm, and 800 mm width.

Besides, it supports the whole weight of the cleaning machine. Then, the total weight carried by the frame includes, sieve frame, sieve, fan, hopper, chaff outlet, shaft, pulley, belt, bearing grain outlet.

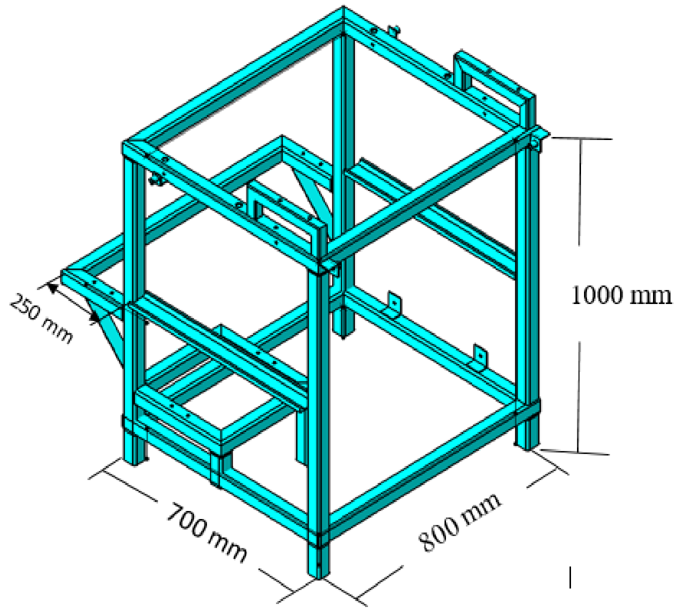


Figure 3.7. Isometric view of a frame

3.5.3. Design and Selection of Sieve

The selection of each sieve had 800 × 692 × 1.5 mm dimensions. Moreover, perforation size and shape were depended upon the average value of the geometric mean diameter and sphericity of the crops respectively. The sieve type for each crop is different from one another since they are not the same in shape and size. Consequently, the three-section of sieve for wheat is 4.5 mm round type, 3.5 mm slot type, and 2 mm round type for upper, middle, and lower section correspondingly; for sorghum is all perforation type is round than the size is 5 mm, 3 mm and 2 mm likewise the size of perforation for maize is 11 mm, 7 mm and 4 mm. The coefficient of opening of sieve is 0.44, 0.58, 0.68, 0.71, 0.75, 0.84, and 0.94 for 2, 3, 4, 4.5, 5, 7 and 11 of opening diameter individually which determined from Eqn. (41).

The sieve angle of tilt is determined by using a coefficient of friction of those three crops and also by considering the other researcher's assessment. Hence, the recommended angle of tilt is 0 –13 degrees (Okunola et al., 2015; Ebaid, 2005; Awady et al., 2003). As a result, the angle of tilt is arranged by using the hole on the hanger of the sieve frame linked with the mainframe in 2, 4, and 6 degrees.

The coefficient of the opening of the sieve was determined by using Eqn. 41 (Okunola et al., 2015).

$$C_o = \frac{3\pi}{2} \times \frac{D^2}{(D+d)^2} \quad \text{.......} (41)$$

Where: C_o= coefficient of opening

D= Diameter of a hole (mm),

d= distance between two successive holes (mm).

Besides, the sieve frame designed by 30 × 30 × 3 mm angle iron with a dimension of 800 × 700 × 450 mm. additionally, it hanging with four linkages to provide smooth forward and backward motion. Thus, used to bolt all of the three sieves and plate through their precise arrangement. As seen in Figure 3.8 it has four partitions with a 150 mm gap between each partition to make sieve adjustment simplest way and also to get proper operation of brush under the sieve of cleaning unit.

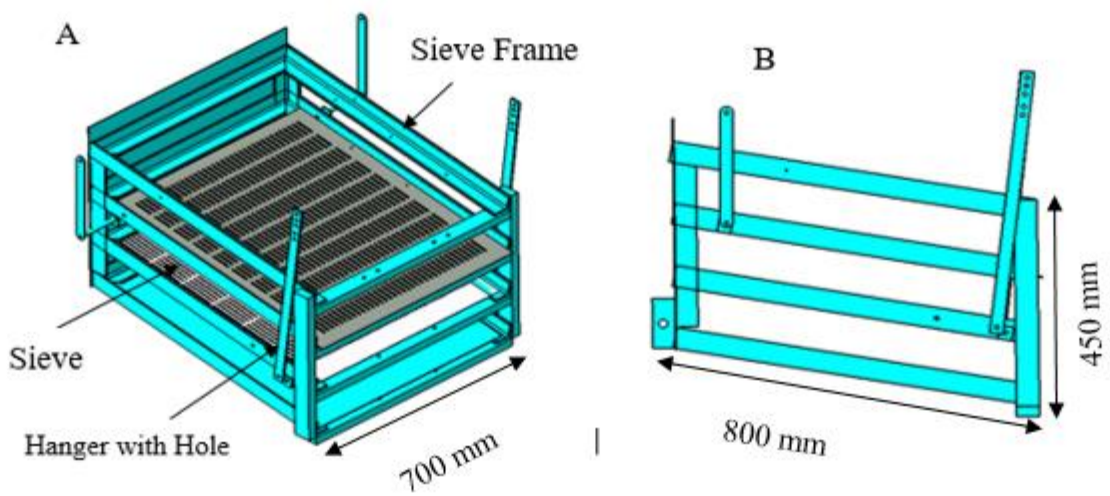


Figure 3.8. Sieve frame with a sieve; A) isometric view and B) side view

3.5.4. Design of Grain and Chaff Outlets

The grain, impurities, and chaff remove by the sieve and fan; this removal of particles released by the outlets constructed by 1.25 thickness sheet metal. Further, parts are important for increase the precisions of the separation and cleaning efficiency. The chaff outlet was put in a parallel position to the fan consequently it can remove light particles released by the fan. Also, the grain outlet has three sections to remove the larger impurities, grain, and smaller particles separately.

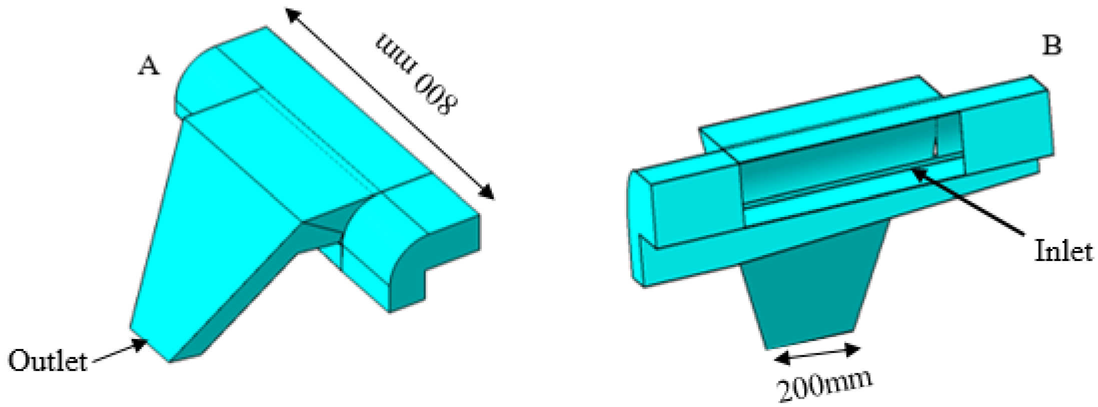


Figure 3.9. Chaff outlet A) rear isometric view and B) front isometric view

3.5.5. Design and Selection of Brush Slider

The brush is made from wood to prevent a sieve from clogging by grain likewise other impurities over by sliding forward and backward. There is six total number of a brush to done the operation from the first sieve up to the last one. Therefore, a slider-crank mechanism was designed to slide the brush under the sieve to open the clogged hole. The design of the length of a crank is determined 200 mm from the Eqn. (8).

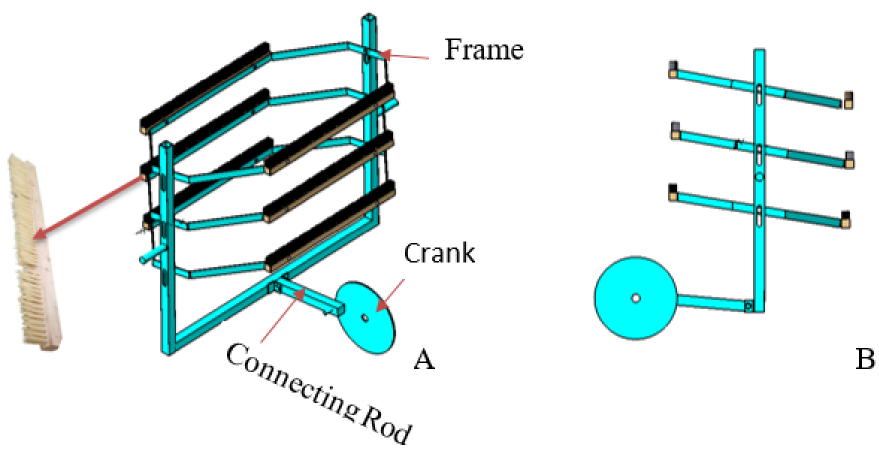


Figure 3.10. Brush slide mechanism A) isometric view and B) side view

The velocity of the slider link was determined by how fast certain points of a link of mechanism traveling. By using section (2.5.3.) procedure the diagram is sketched and showed in Figure 3.11. Therefore, the diagram showed the crank rotates in a clockwise direction at the angular velocity of 10 rad/sec. Depending on the angular velocity of the link B velocity is determined as 2 m/sec. Besides, the velocity of the sliding brush is 168 m/sec.

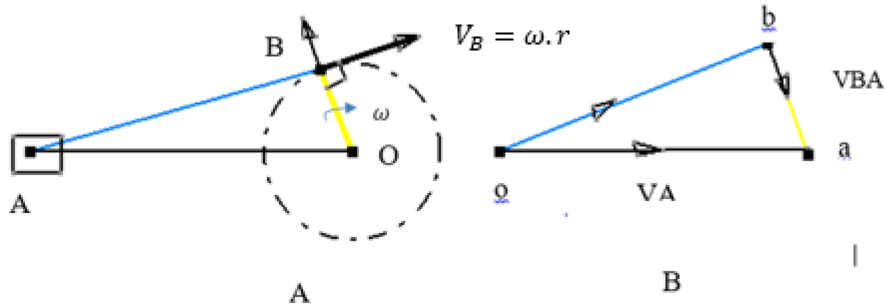


Figure 3.11. A) Slid crank mechanism and B) velocity diagram

The acceleration of the slider was determined by how certain points on the link of the mechanism are either speed up or slow down. Therefore, radial acceleration of B with respect to O is determined as 0.4 $\mathrm{m}^2/\mathrm{sec}$ by using Eqn. (11). Similarly, the radial component of the acceleration of A with respect to B was determined as 592 $\mathrm{m}^2/\mathrm{sec}$ also the total acceleration of A with respect to B located as 279.4 $\mathrm{m}^2/\mathrm{sec}$ from Eqn. (12). Furthermore, the acceleration diagram of the slid crank mechanism was outlined as in Figure 3.12 by used the procedure in section (2.5.3).

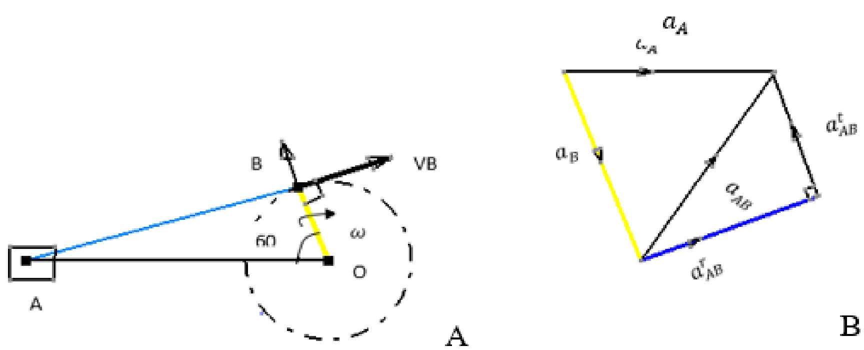


Figure 3.12. A) Slide crank mechanism and B) acceleration diagram

3.5.6. Fan and Fan Case Design

Centrifuge fan was selected as a result of the pressure of an incoming airstream is increased by a fan wheel, four-blade mounted on a circular hub in 45° from each blade direction. Also, the centrifugal fan moves air radially, and the direction of the outward flowing air is changed by 90° from the direction of the income air. Although, the fan has a fixed rotational speed at 687 rpm which gets from the motor by an arrangement of the pulley and belt. While there is a gate on the fan case to adjust the income air flow rate through the axis of the shaft.

The fan case was constructed by sheet metal with 1.5 mm thickness and a diameter of 300 mm and 150 mm opening to inlet air through it. Besides, the cross-sectional area of the fan is estimated to be 0.06 m^2 straight toward the hopper lower part and in a parallel position to chaff outlet to released light particles. Besides, the four-blade which is made of 1 mm thick sheet metal was mounted on the fan shaft inside the fan case with a dimension of 600 mm length and height of 177.5 mm which was determined by using the following formula (Chukwulozie et al., 2018).

$$R = 1 + \frac{\theta}{360} \quad \text{.......} \quad (42)$$

Where: r = radius of fan case, mm,

R = distance from center of shaft to the tip of the blade, mm.

θ = angle between blades, deg.

The volume flow rate of air at fan exist was calculated to be 0.18 $\mathrm{m^3/sec}$ which was found by using Eqn. (43) (Ojediran et al., 2018).

$$Q = A \times V \quad \text{.......} \quad (43)$$

Where: Q = volumetric flow rate, ($\mathrm{m^3/sec}$),

A= cross-sectional area,($\mathrm{m^2}$),

V= air velocity,($\mathrm{m/sec}$).

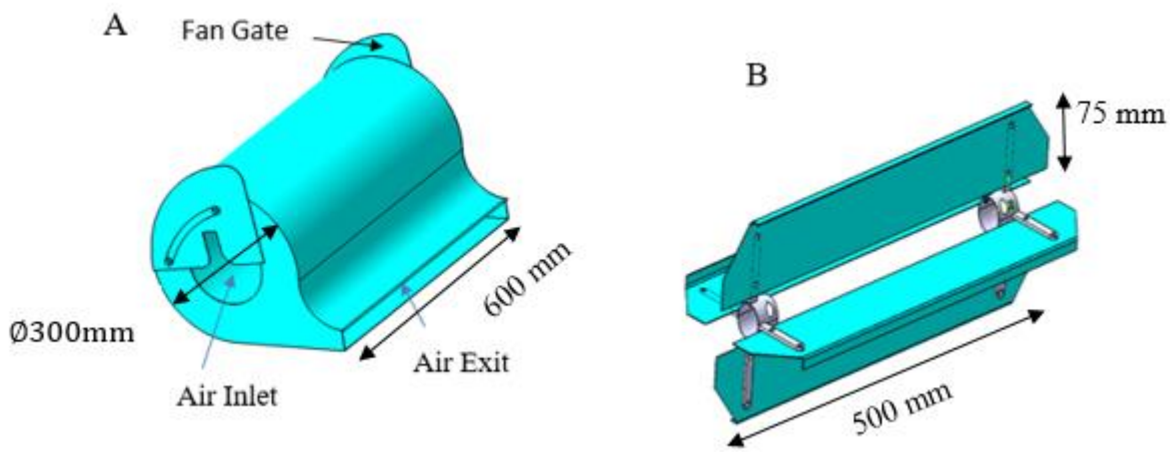


Figure 3.13. Isometric views of fan A) fan case and B) blade

3.6. Power Transmission Components Design and Selection

3.6.1. Pulley Design

The cleaning machine mandatory had five pulleys with a groove angle of 34° for proper arrangement of one to another. Besides, based on the speed required by each part operation including motor, fan, eccentric wheel, and brush. Therefore, the diameter of each pulley was predicted by using Eqn. (26) the diameter value of each pulley is 80 for motor and power transmission to brush slider; 170 for the fan, 350 for both eccentric wheel and brush slider. Besides the width of the pulley was found to be 22 mm from Eqn. (27).

3.6.2. Selection of V-belt

There are three belts used in power transmission of cleaning operation one from a prime mover to the fan; the other to eccentric wheel from the fan and also from eccentric wheel to brush slider. Therefore, the selection of the belt was made by using the pitch length and center distance determined from Eqn. (14) and the nearest belt length was 2108 mm, 2565 mm, and 1346 mm individually for the above cleaning operation. Beyond this the wrap angle for smaller and larger pulleys was found to be 179.9 and 180.1 degrees for motor and fan; 179.8 and 180.2 degrees for fan and eccentric wheel; 137.18 and 222 degrees for eccentric wheel and brush slider by using Eqn. (16) and (17).

3.6.3. Shaft Design

All of the three shafts were designed based on the strength and rigidity criterion when transmitting power under various operating power and load condition. Hence, the diameter of the shaft was determined for fan shaft, eccentric wheel shaft, and brush slider shaft; were all of the force acting on the shaft (weight of the fan, weight of the eccentric wheel, weight of crank, tension in the belt, tight side, slack side and weight of pulley) was considered.

Fan Shaft Design

The diameter of the fan shaft was determined by computing the force acting on the shaft in a vertical and horizontal direction. Also, the total belt tension 334 N is determined by the summation of the tight side (T_1) 308 N and slack side (T_2) 26 N force from Eqns. (19 and 20).

The force acting at each location of the shaft were computed separately and the force acting on the shaft shown in Figure 3.14 and 3.15 so that the computation of resultant bending moment was carried out by resolving the force at the pulley, fan, and reaction force at bearing.

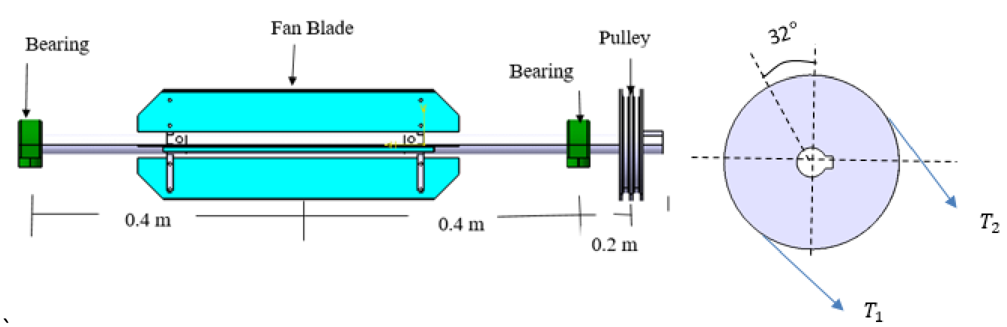


Figure 3.14. Lay out of shaft components

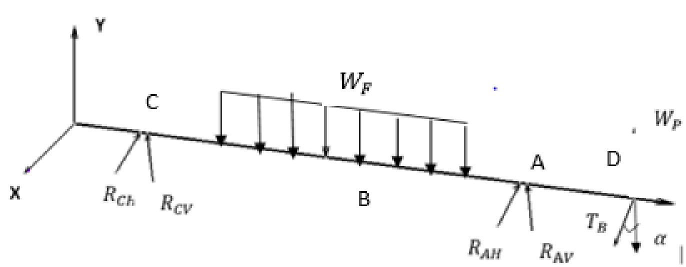


Figure 3.15. Free body diagram of all forces acting on the shaft

Where: R_{AH} = horizontal bearing reaction force at A (N),

R_{AV} = vertical bearing reaction force at A (N),

R_{CH} = horizontal bearing reaction force at B (N),

R_{CV} = vertical bearing reaction force at C (N),

W_P = weight of pulley (N),

W_F = weight of fan (N),

T_B = total belt tension (N).

Analysis of horizontal force on the fan shaft as shown in Figure 16.

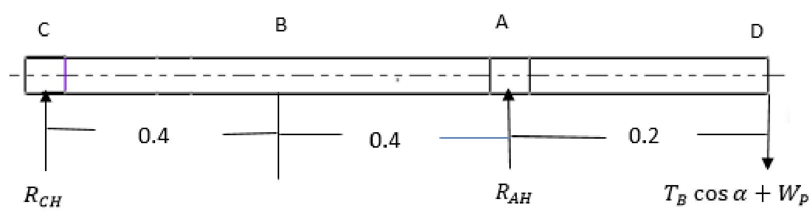


Figure 3.16. Free body diagram for fan shaft on the horizontal plane

The value of the reaction force was determined by taking the summation of a moment at point A: -

$T_B \cos \propto + W_P$ = 303 N

$\sum BM_A = 0$

$(T_B \cos \propto + W_P) \times 0.2 + R_{CH} \times 0.8 = 0$

(303) × 0.2 + R_{CH} × 0.8 = 0

$R_{CH} = 75N$

$\sum FH = 0 \quad \downarrow$-

$-(T_B \cos \propto + W_P) + R_{AH} + R_{CH} = 0$

- (303 N) + R_{AH} +75= 0

$R_{AH} = 228N$

The bending and the shear force diagram is drawn in the following figure

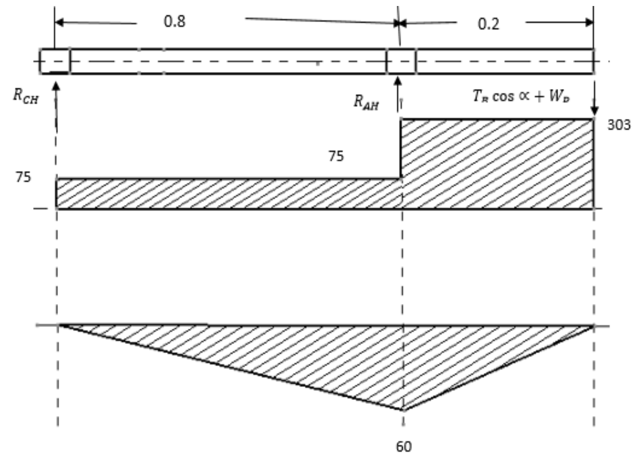


Figure 3.17. Free body diagram for shear force and bending moment

Analysis of vertical force on the fan shaft is shown in the figure below.

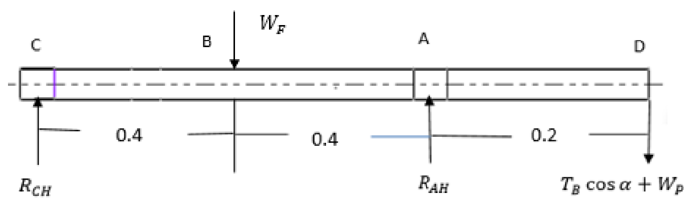


Figure 3.18. Free body diagram for fan shaft on the vertical plane

$T_B \sin \propto = 333.8 \times \sin \propto = 177$ N

$W_F = 39$ N

$W_P = 20$ N

$\sum BM_A = 0$

$\therefore T_B \sin \propto + W_P = 196.88$

$0.2 \times (T_B \sin \propto + W_P) - (0.4 \times W_F) + 0.8 \times R_{CV} = 0$

$0.2 \times 196.88 - 0.4 \times 39 + R_{CV} \times 0.8 = 0$

$R_{CV} = 29.75$ Downward

$\sum F_V = 0$ ↓-

$-T_B \sin \propto + R_{AV} - W_F + R_{CV} = 0$

-196.88 - 39 + 29.75 + R_{CV} = 0

$R_{AV} = 206.1$ N

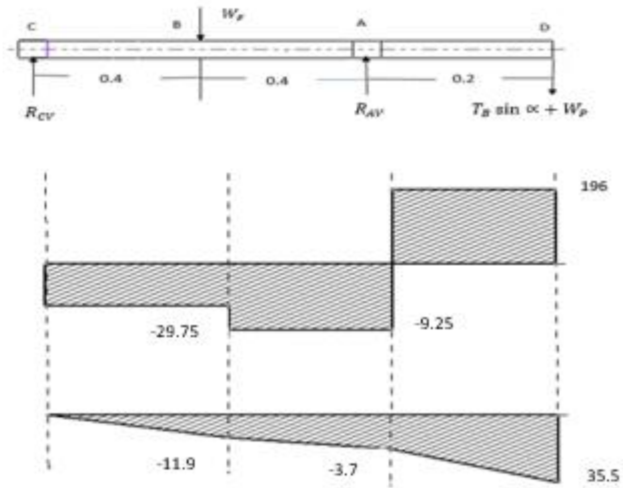


Figure 3.19. Free body diagram for shear force and bending moment

The resultant maximum bending moment on the shaft was calculated using Eqn. (29). The resultant bending moment with the highest value of 60 Nm was selected by taken the maximum Bending moment of M_V and M_H. And the torque of the shaft was estimated by using Eqn. (30) finding out 48.28 Nm.

A bearing catalog was used to determine the ratio of D/d, and the value was found to be between 1.2 and 1.5. And also the value of the constant C_b and $C_t$is used in Eqn. (28) were obtained from Appendix Table (18, 19) the value of C_t is 1.98. as recommended by Richard & Kelth, (2011). Besides the value of K_t is 1 and

the value of K_b is 1.2 were used in equation (28) to estimate the value of the shaft diameter. In addition allowable shear stress (S_a) of 40Mpa for shaft diameter. As a result, the minimum diameter of the shaft was found to be 25 mm. therefore, a 25 mm diameter stainless steel shaft was used for the operation of the fan.

Eccentric Wheel Shaft Design

The diameter of the eccentric wheel shaft was determined by computing the force acting on the shaft in a vertical and horizontal direction. Also, the total belt tension 330 N is determined by the summation of the tight side (T_1) 305.97 N and slack side (T_2) 24.59 N force from Eqns. (19 and 20).

The force acting at each location of the shaft were computed separately and the force acting on the shaft shown in Figure 3.20 and 3.21.

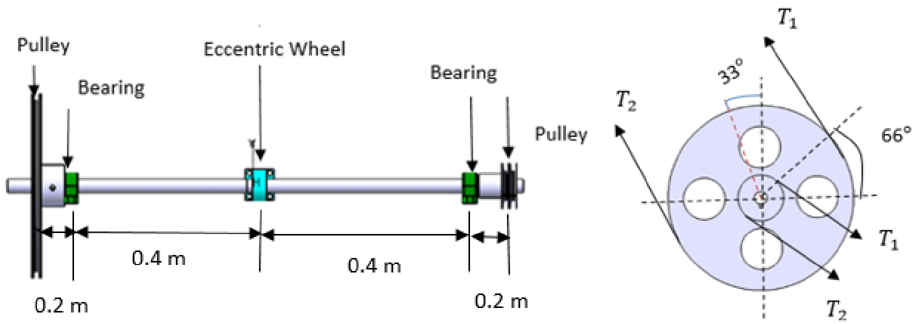


Figure 3.20. Lay out of shaft components

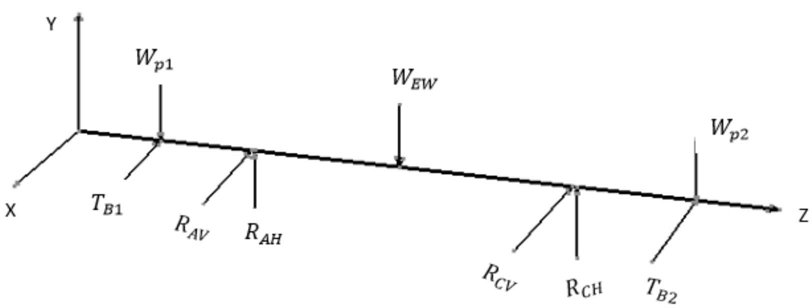


Figure 3.21. Free body diagram of all forces acting on the shaft

Where: W_{EW} = weight of eccentricity wheel

Analysis of horizontal force on the eccentricity wheel shaft as shown in Figure 3.22.

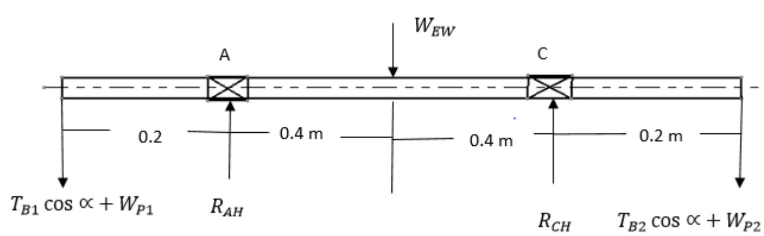


Figure 3.22. Free body diagram for eccentric wheel shaft on horizontal plane

The determined torsional moment on both pulley one and pulley two was 49 Nm and 4.426 Nm respectively and by the summation of that moment is 53.4 Nm.

Where:

$T_{B1} \cos \propto_1 + W_{P1} = 178 \text{ N}$

$T_{B2} \cos \propto_2 + W_{P2} = 264.8 \text{ N}$

$W_{EW} = 12.8 \text{ N}$

$\sum BM_A = 0$

$(T_{B2} \cos \propto + W_{P1}) \times 0.2 - W_{EW} \times 0.4 + R_{CH} \times 0.8) - (T_{B1} \cos \propto + W_{P2}) \times 1 = 0$

$R_{CH} \times 0.8 = W_{EW} \times 0.4 + (T_{B1} \cos \propto_2 + W_{P2}) - (T_{B2} \cos \propto + W_{P2}) \times 0.2$

$$R_{CH} = \frac{12.8 \times 0.4 + 178 - 264.8 \times 0.2}{0.8}$$

$R_{CH} = 162.7 \text{ N}$

$\sum F_H = 0 \quad \downarrow$-

$-(T_{B1} \cos \propto + W_{P1}) + R_{AH} - W_{EW} + R_{CH} - (T_{B2} \cos \propto_2 + W_{P2}) = 0$

$R_{AH} = 178 + 12.8 - 162.7 + 264.8$

$R_{AH} = 292.9 \text{ N}$

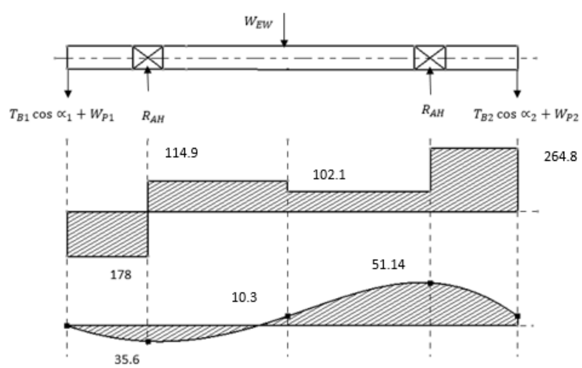


Figure 3.23. Free body diagram for shear force and bending moment on horizontal plane

Analysis of vertical force on the Eccentricity Wheel shaft shown on the figure below;

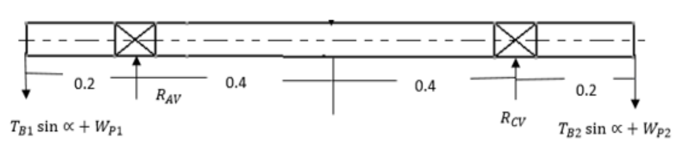


Figure 3.24. Free body diagram for Eccentricity wheel Shaft on vertical plane

$\mathrm{T_{B1}} \sin \propto + \mathrm{W_{P1}} = 345.47$

$\mathrm{T_{B2}} \sin \propto + \mathrm{W_{P2}} = 208.5$

$\sum \mathrm{BM_A} = 0$

$-\mathrm{T_{B1}} \sin \propto + \mathrm{W_{P1}} \times 0.2 - \mathrm{R_{CV}} \times 0.8 + 1 \times \mathrm{T_{B2}} \sin \propto + \mathrm{W_{P2}} = 0$

$$\mathrm{R_{CV}} = -\frac{(345.47 \times 0.2) - (208.5 \times 1)}{0.8}$$

$\mathrm{R_{CV}} = 174.5 \text{ N}$

$\sum \mathrm{F_V} = 0 \quad \downarrow$-

$(-\mathrm{T_{B1}} \sin \propto + \mathrm{W_{P1}}) + \mathrm{R_{AV}} + \mathrm{R_{CV}} - \mathrm{T_{B2}} \sin \propto + \mathrm{W_{P2}} = 0$

$\mathrm{R_{AV}} = 345.47 - 174.5 + 208.5 - 12.8$

$\mathrm{R_{AV}} = 366.6 \text{ N}$

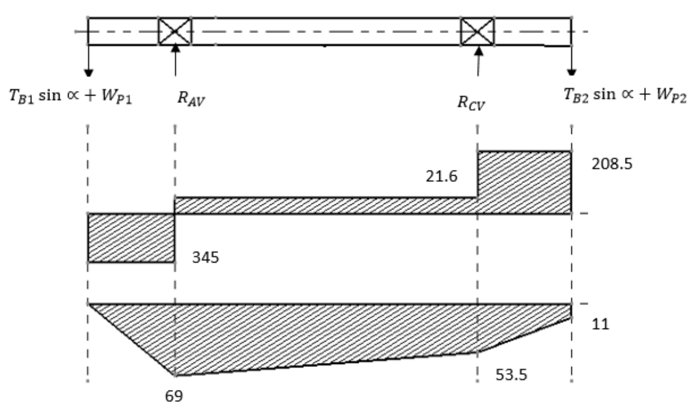


Figure 3.25. Free body diagram for shear force and bending moment

The resultant maximum bending moment on the shaft was calculated using Eqn. (29). Then, the highest value of 72 Nm was selected by taken the maximum Bending moment of $\mathrm{M_V}$ and $\mathrm{M_H}$. And the torque of the shaft was estimated by using Eqn. (30) finding out 53.4 Nm. Further, the diameter of the shaft was determined to be 30 mm by using Eqn. (28).

Brush Slider Shaft

The diameter of the brush slider shaft was determined by computing the force acting on the shaft in a vertical and horizontal direction. Also, the total belt tension is 318 N determined by the summation of the tight side (T_1) 311.9 N and slack side (T_2) 6.608 N force from Eqn. (19 and 20).

The force acting at each location of the shaft were worked out separately and the force acting on the shaft show in Figure 3.26 and 3.27.

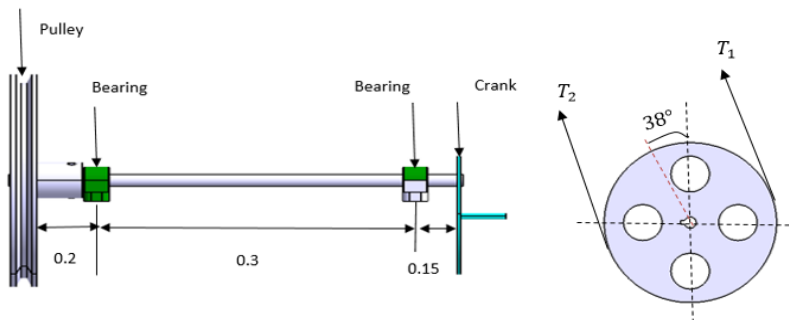


Figure 3.26. Lay out of shaft components

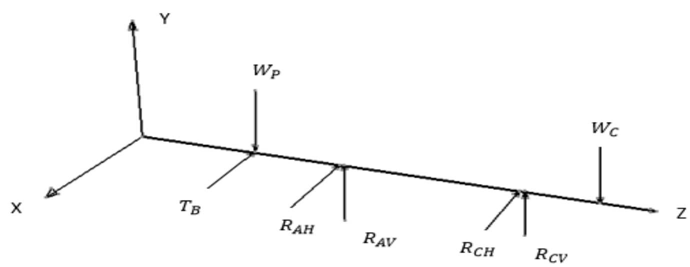


Figure 3.27. Free body diagram of all forces acting on the shaft

Where: - W_C = weight of the crank

Analysis of horizontal force on the eccentricity wheel shaft as shown in Figure 3.28.

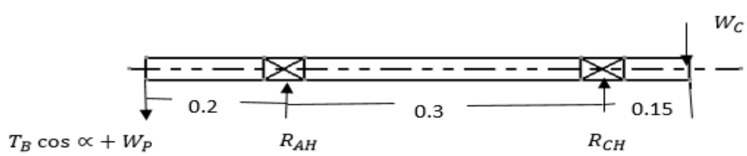


Figure 3.28. Free body diagram for brush slider shaft on horizontal plane

Where: -

$T_B \cos \propto = 250$ N

$W_c = 9.81\ \mathrm{N}$

$Wp = 44\ \mathrm{N}$

$T_B \cos \propto + W_P = 294\ \mathrm{N}$

$\sum BM_A = 0$

$0.2 \times T_B \cos \propto + W_P - R_{CH} \times 0.3 = 0$

$$R_{CH} = \frac{294 \times 0.2}{0.3}$$

$R_{CH} = 196\ \mathrm{N}$

$\sum F_H = 0 \quad \downarrow$ -

$-(T_B \cos \propto + W_P) + R_{AH} + R_{CH} = 0$

$R_{AH} = T_B \cos \propto + W_P - R_{CH}$

$R_{AH} = 294 - 196N$

$R_{AH} \quad = 98\ \mathrm{N}$

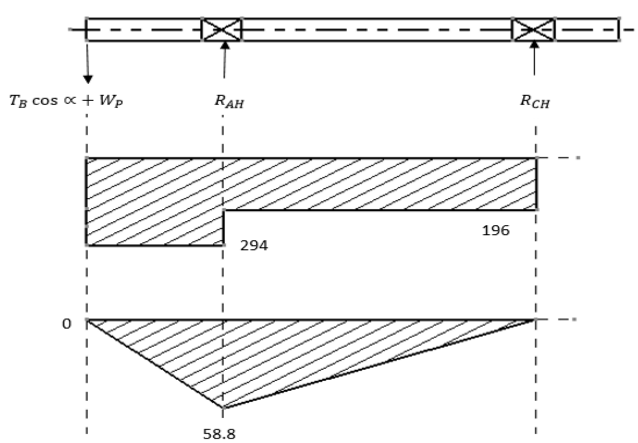


Figure 3.29. Free body diagram for brush slider shaft on horizontal plane

Analysis of vertical force on the Brush Slider shaft was determined as shown on the Figure below;

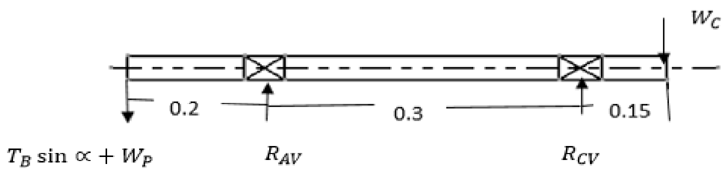


Figure 3.30. Free body diagram for eccentricity wheel shaft on vertical plane

$$\mathrm{T_B} \sin \propto + \mathrm{W_P} = 239$$

$$\sum \mathrm{BM_A} = 0 \ \downarrow +$$

$$0 = \big((-\mathrm{T_B} \sin \propto + \mathrm{W_P})0.2\big) - (\mathrm{R_{CV}} \times 0.3) + \mathrm{W_C} \times 0.45)$$

$$\mathrm{R_{CV}} = \mathrm{W_C} \times 0.45 - (\mathrm{T_B} \sin \propto + \mathrm{W_P})\ (0.2)$$

$$\mathrm{R_{CV}} = \frac{(9.81 \times 0.45) - (239 \times 0.2)}{0.3}$$

$\mathrm{R_{CV}} = 144$ Downward

$\sum \mathrm{F_V} = 0 \downarrow$-

$$-(\mathrm{T_B} \sin \propto + \mathrm{W_P}) + \mathrm{R_{AV}} - \mathrm{R_{CV}} - \mathrm{W_C} = 0$$

$$\mathrm{R_{AV}} = 239 + 144 + 9.81N$$

$$\mathrm{R_{AV}} = 392.9\mathrm{N}$$

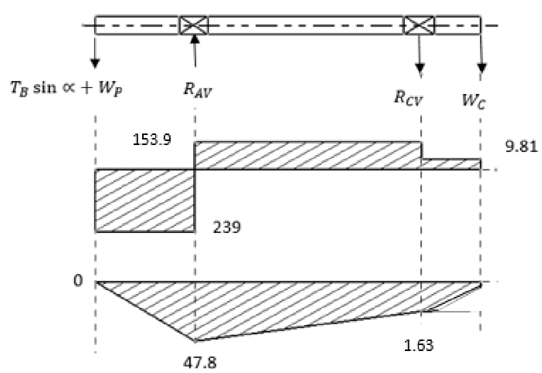


Figure 3.31. Free body diagram for shear force and bending moment

The diameter of the shaft was determined to be 22 mm by using the Eqn. (28). And also, the resultant maximum bending moment on the shaft was estimated to the highest value of 75 N by taken the maximum bending moment of M_V and M_H.in addition the torque of the shaft was calculated to be 49 N by using Eqn. (30).

3.6.4. Bearing Selection

Bearing Selection for Fan Shaft

The selection of the bearing was depending upon the resultant radial load, bore size (shaft diameter), and bearing carrying load capacity. Therefore, the maximum radial load on the fan shaft occurs at point ‘A’ hence the resultant load at this point is estimated to be 313 N by using Eqn. (31). Beyond this, the load of bearing was calculated to be 313 N by using Eqn. (32).

Besides, the bearing number 205 by the bore of 25 mm with a width of 15 mm was selected Appendix Table (20). Since the diameter of the shaft has been determined to be 25 mm. the design life of the bearing was found to be 247 $\times 10^6$ revolution from Eqn. (33). Consequently, the basic dynamic load rating of the bearing was estimated using Eqn. (34) and found to be 1878 since the basic dynamic load rating of the bearing selected is less than the dynamic load rating by the manufacturers. That is 1878 < 11000 N Appendix Table (20).

Bearing Selection for Eccentric Wheel Shaft

The maximum radial load on the eccentric wheel shaft occurs at point ‘A’ is that so the resultant load calculated to be 471 N by using Eqn. (31). Therefore, the load of the bearing is calculated to be 471 N by using Eqn. (32).

The bearing number 206 by bore 30 mm with width 16 was selected which depends on the diameter of the shaft that was 30 mm. As a result, the design life of bearing was found to be 112.7 $\times 10^6$ revolution. Thus, the basic dynamic load was calculated to be 2260 N which is less than the value rated to be 15300 N. which means the design is safe to select the bearing.

Bearing Selection for Brush Slider

The bearing number 204 by the bore of 20 mm with a width of 14 mm was selected from the shaft diameter of 20 mm. the maximum radial force occurs at point ‘A’ then the resultant load was calculated to be 308 N hence, the total load-bearing was estimated to be 308 N.

Besides, the design life of bearing was calculated to be 35×10^6 revolution hence, the basic dynamic load was determined to be 1007 N which is less than 10000 N which is rated by the manufacture.

3.6.5. Power Required to Operate the Cleaning Machine

The total power required to operate the cleaning machine was determined by the summation of each part of the machine which needs power. Therefore, this part is fan rotation, sieve oscillation, and brush slider. The power was determined as 1 Kw, 1.7 Kw, and 0.45 Kw for each operation respectively by using Eqn. (24) then the total power required by the machine is 4.2 hp determined by Eqn. (25). Therefore, a 4 hp electric motor with a desired rotational speed of 1440 rpm was selected to operate the cleaning machine during test and performance evaluation.

3.7. Cost Analysis

The grain cleaning machine economic analysis includes the cost of ownership and operation. Ownership costs to be independent of the amount a machine is used and are often called fixed or overhead costs. Conversely, operating costs or variable costs include the cost of repair and maintenance, electricity, and labor wage which, increase in proportion to the amount of the machine is used.

3.7.1. Ownership Cost (Fixed Cost)

Fixed costs include deprecation of the machine, interest on the investment, and cost of taxes, insurance, and housing of a machine.

Depreciation

Depreciation is a cost resulting from the wear, obsolescence, and age of a machine. The degree of mechanical wear may cause the value of a particular machine to be somewhat above or below the average value of similar machines when it is traded or sold. The introduction of new technology or a major design change may make an older machine suddenly obsolete, causing a sharp decline in its remaining value. But age and accumulated hours of use are usually the most important factors in determining the remaining value of a machine. Therefore the depreciation cost is estimated using the following formula (Kepner et al., 2005)

$$D = \frac{P-S}{L} \quad \text{.......... (44)}$$

Where: D = depreciation per hour (ETB/hr.),

P = capital investment cost (ETB),

S = salvage value, 10% of the capital investment (ETB),

L = life of the machine in hours or year.

Interest

The cost of ownership includes the interest on the money that is invested in the machine. If a loan is used to purchase a machine, the interest rate is known. If a machine is purchased for cash, the relevant interest rate is the prevailing rate that could have been obtained if the money had been invested instead of being used to purchase the machine. The principle on which the interest is assessed is equal to the remaining value of the machine in any given year. The interest was determined by using Eqn. (45) (Kepner et al., 2005).

$$I = \frac{P+S}{2} \times i \quad \text{.......... (45)}$$

Where: I = interest per hour (ETB/hr.),

i = interest rate (%).

Taxes, Insurance, and Housing

Manly the cost value of the three of them is much smaller than depreciation and interest, even though they need to be concerned. Also, it’s was hard to determine the exact value of the rate for tax, insurance, and housing. According to Srivastava et al., (2006) the tax charge estimated 1% of the capital cost investment of the machine; for insurance cost 0.25% of the capital cost of investment of machine; and also the shelter cost as 0.75% of the capital cost of the investment of the machine. Therefore the total cost of tax, insurance, and housing can be estimated at 2% of the capital cost of investment of the machine.

Total fixed cost (ETB/hr.) = Deprecation + Interest + Insurance + Taxes + House cost

3.7.2.Operating Cost

Operating costs include repair and maintenance, fuel/ electricity costs, and labor wages. A constant hourly labor cost can be determined for hired operators. If the owner operates the machine, the labor cost is determined from alternative uses of the owner's time (Ojha & Michael, 2009).

Repair and Maintenance Cost

Costs for repairs and maintenance are highly variable depending on the care provided by the manager of the machine. Some expenditures will always be necessary to replace worn or failed parts and/or to repair damage from accidents. Repair and maintenance costs tend to increase with the size and complexity (Kepner et al., 2005)

RM = 2.5% × purchase price or capital investment per year

Electricity Cost

Electric motors are devices that convert electric power into mechanical power. They are often chosen over the engine for application of which quieter, more readily adaptable to automatic or remote control, and do not produce exhaust emissions (Srivastava et al., 2006).

From Ethiopia, the electricity price or tariff is 0.816 Birr/Kwh. Therefore, the total electricity consumption is 2640 KWh, and also the cost of electricity for one year is 2154.24 Birr/ Kwh.

Labor Cost/ Wage

The cost of labor or the wage rate varies. For hired operators, a constant hourly rate is (W), appropriate; for owner-operators, the labor charge should be the opportunity cost. In no instance should the charge be less than the typical community labor rate.

Total operating cost = repair and maintenance + electricity + labor cost

To determine the cost of operation the following assumption was made;

1. Useful working hours = 3000 for small and stationary machinery
2. Useful life year = 8
3. No of labor per day = 1
4. No of a working hour per year = 806 hr. (cleaning take place from October to February)

5. Salvage value = 10%
6. Ret of interest on fixed capital = 10% of initial investment
7. Taxes per annum = 1% of initial investment
8. Insurance rate = 0.25 % of initial investment
9. Housing = 0.75% of initial investment

3.8. Performance Evaluation of the Prototype Machine

3.8.1. Performance Evaluation of Grain Cleaning Machine

The performance of the cleaning machine was carried out on one variety from each crop type (wheat, maize, and sorghum) such that Dandaia, Melkassa 2, and Melkame respectively. During each test run, the crop was cleaned manually and weighted by digital balance before starting the testing, and also mixed with MOG. A weighted mixture of grain was feed into the machine at a selected sieve angle and air velocity for an evaluation of the level of cleaning efficiency, and separation efficiency.

The equation below was determined to calculating the cleaning efficiency, and separation efficiency (Afolabi et al., 2019).

$$CE = \frac{M_{cas}}{M_{cbs}} \times 100\% - CL \quad \text{..........} \quad (46)$$

$$CL = \frac{S_l}{S_o} \times 100\% \quad \text{..........} \quad (47)$$

Where: CE = cleaning efficiency (%),

CL = cleaning loss (%),

M_{cas} = the mass of clean grain sample after separation and cleaning (kg),

M_{cbs} = the mass of the sample before cleaning and separation (kg),

S_l = the grain loss behind the machine ($M_4 - M_3$) (kg),

S_o = the grain output (kg).

$$SE = \frac{M_1}{M_2} \times 100\% \quad \text{..........} \quad (48)$$

Where: SE = separation efficiency (%),

M_1 = mass of impurities after separation (kg),

M_2 = mass of impurities before separation (kg),

M_3 = mass of grain after separation and cleaning (kg),

M_4 = mass of grain before separation and cleaning (kg).

3.8.2. Experimental Design

The experiment was designed in a split-split plot design for three factorial as recommended by Gomez & Gomez, (1984). Hence, those three factors taken in the evaluation are federate, sieve angle, and air velocity within main – plot, sub-plot, and sub-sub plot respectively with three replications. The design had the form of $3 \times 3 \times 3$ equal to 27 factorial combinations with three replication as a block which result in a total of 81 test runs.

3.8.3.Statistical Analysis

The evaluation of the machine was determined by the effect of feed rate at 10, 12, and 15 kg; sieve angle at 2, 4, and 6 degrees; and also air velocity 3, 2.1, and 1.3 on the grain mixture cleaning and separation of from grain and MOG were analysis. Thus, data collected at each test run was subjected to analysis of variance using the statistical procedure as described by Gomez & Gomez, (1984). Therefore, the analysis was done by using SPSS statistical software. When the effects of the treatment were found significant, an LSD test was performed to assess the difference among the treatments at a 5% level of significance.

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1. Physical and Mechanical Properties of Grain

4.1.1. Physical Properties of Grain

The physical properties of the grain were obtained by taken a 100 seed size sample to decide the perforation size and shape of the sieve. Thus, has length, width, thickness, geometric mean diameter, arithmetic mean diameter, and sphericity of the selected varieties of each crop (wheat, Sorghum, and maize) were display in the Table shown below. Therefore, results show that the min, max, range, mean, standard deviation, and variance.

Therefore, Table 4.1 presents the physical properties of wheat varieties which are Lemu, Dandaia, and kingbird at the moisture content of 11.2 %, 10.98 %, and 12.4 % respectively. Hence, the mean value was determined for length, thickness, width, geometric mean diameter, arithmetic mean diameter, and sphericity as 5.97, 2.8, 3.31, 3.8, 4.03, 63.78 for Lemu; 5.77, 2.64, 3.12, 3.62, 3.84, and 62.7 for Dandaia; 5.83, 2.72, 3.35, 3.76, 64.4 for Kingbird variety in the above order.

Besides, the physical properties of maize varieties Melkassa 2, Melkassa 4, and MH – 104 at moisture content 13.2 %, 12.11 %, and 12.75 % individually for instance determined in Table 4.2. Consequently, the result of the mean value of length, thickness, width, geometric mean diameter, arithmetic mean diameter, and sphericity was estimated as 11.78, 4.83, 8.91, 7.74, 8.39, and 66.51 for Melkassa 2; 11.03, 4.83, 8.69, 7.69, 8.18, and 70.3 for Melkassa 4; also 11.75, 4.47, 9.4, 7.86, 8.34, and 67.4 for MH – 104 Variety respectively.

The other crop is sorghum and its physical properties of three crop varieties namely Melkame, Dagim, Tillahune were determined in Table 4.3 at 9.89 %, 10.1%, and 12 % respectively. Thus, the length, thickness, width, geometric mean diameter, arithmetic mean diameter and sphericity mean value was estimated such as 4.7, 2.69, 4.2, 3.76, 3.80, and 80 for Melkame; 4.03, 2.67, 3.36, 3.3, 3.35, 81 for Dagim; 4.59, 2.67, 4.54, 3.79, 3.91, 82.56 for Tllahune individually.

Hence, from the Appendix table (21, 22, and 23), all of the physical properties of those three crops had been significantly different at a 5 % level of significance. In addition to this Lemu and Kingbird has higher physical properties than Dandaia of a wheat variety; likewise, Melkassa 2 and MH - !04 had higher physical properties except for sphericity than Melkassa 4 for maize variety; in sorghum varieties, Dagim and Melkame had higher physical properties than Tillahune except that sphericity was smaller for Melkame.

Table 4.1. The physical properties of wheat

Treatment	Varieties	Min	Max	Range	Mean	Std.Er	S.D	Variance
Length (mm)	Lemu	5.17	6.90	5.17 – 6.90	5.97	0.034	0.34	0.113
	Dandaia	4.73	6.60	4.73 – 6.60	5.77	0.031	0.31	0.099
	Kingbird	4.9	6.59	4.9 -6.59	5.83	0.031	0.35	0.102
Thickness (mm)	Lemu	2.09	3.36	2.09-3.36	2.8	0.025	0.25	0.06
	Dandaia	2.04	3.26	2.04 -3.26	2.64	0.026	0.26	0.065
	Kingbird	2.19	3.21	2.19 – 3.21	2.72	0.022	0.22	0.048
Width (mm)	Lemu	2.69	3.71	2.69-3.71	3.31	0.025	0.25	0.062
	Dandaia	2.35	3.85	2.35 – 3.85	3.12	0.032	0.32	0.1
	Kingbird	2.71	5.59	2.71 -5.59	3.35	0.035	0.35	0.121
Geometric mean dia (mm)	Lemu	3.15	4.28	3.15 -4.28	3.8	0.023	0.23	0.053
	Dandaia	2.83	4.14	2.83 – 4.14	3.62	0.026	0.26	0.066
	Kingbird	3.20	4.62	3.2 – 4.62	3.76	0.24	0.24	0.058
Arithmetic	Lemu	3.39	5.35	3.39 – 5.35	4.03	0.026	0.26	0.069
	Dandaia	3.04	4.36	3.04 – 4.36	3.84	0.024	0.24	0.059

mean dia (mm)	Kingbird	3.37	4.87	3.37 – 4.87	3.97	0.024	0.24	0.058
Sphericity (%)	Lemu	56.5	70.6	56.5 -70.6	63.78	0.27	2.73	7.5
	Dandaia	55.87	71.48	55.9 – 71.5	62.7	0.32	3.24	10.51
	Kingbird	57.8	75.3	57.8 – 75.3	64.4	0.024	0.24	7.88

Table 4.2. The physical properties of maize

Treatment	Varieties	Min	Max	Range	Mean	Std.Er	S.D	Variance
Length (mm)	Melkassa 2	8.51	14.43	8.51 – 14.4	11.78	0.12	1.22	1.49
	Melkassa 4	5.73	8.18	5.73 – 8.18	11.03	0.11	1.12	1.25
	MH - 104	9.22	14.75	9.72 – 14.75	11.75	0.12	1.16	1.35
Thickness (mm)	Melkassa 2	3.08	7.77	3.08 – 7.77	4.6	0.099	0.99	0.987
	Melkassa 4	3.59	7	3.59 - 7	4.83	0.069	0.69	0.477
	MH - 104	3.54	7.01	3.54 – 7.01	4.47	0.072	0.72	0.52
Width (mm)	Melkassa 2	6.41	10.85	6.41 – 10.85	8.81	0.096	0.96	0.925
	Melkassa 4	5.42	10.52	5.42 – 10.52	8.69	0.096	0.96	0.921
	MH - 104	6.70	11.26	6.70 – 11.26	9.4	0.072	0.723	0.52
Geometric mean dia (mm)	Melkassa 2	6.44	8.8	6.44 – 8.8	7.74	0.046	0.46	0.211
	Melkassa 4	6.57	8.91	6.57 – 8.91	7.69	0.044	0.44	0.197
	MH - 104	6.61	9.26	6.61 – 9.26	7.86	0.48	0.48	0.234
Arithmetic	Melkassa 2	2.34	7.07	2.34 – 7.07	8.39	0.80	8.09	0.223
	Melkassa 4	6.9	9.36	6.9 – 9.36	8.18	0.048	048	0.232

mean dia (mm)	MH - 104	7.13	9.68	7.13 – 9.68	8.54	0.05	0.50	0.25
	Melkassa 2	53.03	98.52	53 – 98.52	66.51	0.809	8.1	65.56
Sphericity (%)	Melkassa 4	56.1	87.24	56.1 – 87.24	70.3	0.68	6.84	46.79
	MH - 104	54.46	84.49	54.46 – 84.5	67.4	0.63	6.34	40.02

Table 4.3. The physical properties of sorghum

Treatment	Varieties	Min	Max	Range	Mean	Std.Er	S.D	Variance
Length (mm)	Melkme	4.08	5.71	4.08 – 5.71	4.7	0.03	0.303	0.092
	Dagim	3.30	4.54	3.30 – 4.54	4.03	0.023	0.23	0.053
	Tillahune	3.97	5.22	3.97 – 5.22	4.59	0.024	0.248	0.057
Thickness (mm)	Melkme	2.23	3.14	2.23 – 3.14	2.69	0.021	0.206	0.042
	Dagim	2.28	3.18	2.28 – 3.18	2.67	0.017	0.167	0.028
	Tillahune	2.23	2.97	2.23 – 2.97	2.67	0.014	0.137	0.019
Width (mm)	Melkme	2.32	4.2	2.32 – 4.2	4.2	0.033	0.33	0.11
	Dagim	2.63	3.98	2.63 – 3.98	3.36	0.027	0.27	0.074
	Tillahune	3.96	5.2	3.96 – 5.2	4.54	0.025	0.248	0.061
Geometric mean dia (mm)	Melkme	3.18	4.32	3.18 – 4.32	3.76	0.022	0.22	0.05
	Dagim	2.88	3.69	2.88 – 3.69	3.3	0.017	0.17	0.028
	Tillahune	3.35	4.20	3.35 – 4.20	3.79	0.017	0.171	0.029
Arithmetic mean dia (mm)	Melkme	3.37	4.44	3.37 – 4.44	3.86	0.022	0.22	0.049
	Dagim	2.91	3.71	2.91 – 3.71	3.35	0.0169	0.169	0.028
	Tillahune	3.47	4.37	3.41 – 4.37	3.91	0.23	2.301	0.032
	Melkme	60.49	91.8	60.49 – 91.8	80	0.39	3.89	14.92

Sphericity (%)	Dagim	73.83	98.3	73.83 – 98.3	81	0.38	3.83	14.65
	Tillahune	75.69	88.64	75.7 – 88.6	82.56	0.23	2.3	5.298

4.1.2.Mechanical Properties of the Crops

Mechanical properties of grain include angle of repose and coefficient of friction, these properties observed by randomly selected 10 gm of grain for three different crops which are wheat, maize, and sorghum, each group of crops had three varieties.

Therefore, the angle of repose and coefficient of friction of wheat was described in Table 4.4 for Kingbird, Lemu, and Dandaia varieties according to min, max, range, mean, standard deviation, and variance. Hence, the mean value of angle of repose was 26, 23.5, and 25.2 degrees and for each variety respectively. Additionally, the mean coefficient of friction was 0.488, 0.434, and 0.47 for each variety.

Furthermore, Table 4.5 shown the angle of repose and coefficient of friction for maize concerning its varieties. Hence, the value of mean determined as 22.4, 22.4, and 22.8 degrees of angle of repose with respective to Melkassa 2, Melkassa 4, and MH-104 and also the coefficient of friction mean was 0.412 for Melkassa 2 and Melkassa; 0.42 for MH – 104.

The angle of repose and coefficient of friction for sorghum was determined in Table 4.5 as 24.1 for Melkame; 24.6 for both Dagim and Tillahune. Besides, the mean coefficient of friction was determined as 0.447 for Melkame; 0.458 for both Dagim and Tillahune.

Beyond this, both angles of repose and coefficient of friction had a significant difference at 5 % of significance only between wheat Variety, but there is no significant difference for maize and sorghum shown in Appendix table (24). The comparison of the mean for three crops predicts that wheat and sorghum had a higher angle of repose and coefficient of friction than maize crops.

Table 4.4. Angle of repose and coefficient of friction for wheat crop

Treatment	**Varieties**	**Min**	**Max**	**Range**	**mean**	**Std.Er**	**S.D**	**Variance**
	Lemu	21	27	21 - 27	23.5	0.563	1.779	3.167
AoR	Dandaia	23	28	23 - 28	25.2	0.593	1.874	3.511

(degree)	Kingbird	24	29	24 – 29	26	0.596	1.885	3.556
	Lemu	0.38	0.51	0.38 – 0.51	0.434	0.119	0.377	0.001
CoF	Dandaia	0.42	0.53	0.42 – 0.53	0.47	0.013	0.042	0.002
	Kingbird	0.45	0.55	0.45 – 0.55	0.488	0.013	0.043	0.002

Table 4.5. Angle of repose and coefficient of friction for maize

Treatment	Varieties	Min	Max	Range	mean	Std.Er	S.D	Variance
	Melkassa 2	21	24	21 – 24	22.4	0.306	0.966	0.933
AoR	Melkassa 4	20	24	20 – 24	22.4	0.371	1.174	1.378
(degree)	MH - 104	22	24	22 - 24	22.8	0.200	0.632	0.400
	Melkassa 2	0.38	0.45	0.38 – 0.45	0.412	0.006	0.019	0.000
CoF	Melkassa 4	0.36	0.45	0.36 – 0.45	0.412	0.008	0.024	0.001
	MH - 104	0.40	0.45	0.40 – 0.45	0.420	0.004	0.013	0.000

Table 4.6. Angle of repose and coefficient of friction for sorghum

Treatment	Varieties	Min	Max	Range	mean	Std.Er	S.D	Variance
	Melkame	23	26	23 – 26	24.1	0.314	0.994	0.989
AoR	Dagim	23	27	23 - 27	24.6	0.4	1.265	1.6
(degree)	Tillahune	23	26	23 - 26	24.6	0.371	1.173	1.378
	Melkame	0.42	0.49	0.42 – 0.49	0.447	0.007	0.021	0.000
CoF	Dagim	0.42	0.49	0.42 – 0.49	0.458	0.008	0.025	0.001
	Tillahune	0.42	0.51	0.42 – 0.51	0.458	0.008	0.027	0.001

4.2. Performance of Grain Cleaning Machine

The influence of feed rate, sieve angle, and air velocity on the performance of the cleaning machine for wheat, maize, and sorghum crops was estimated. This performance including cleaning efficiency and separation efficiency of each crop concerning feed rates 1, 2, and 3

(10, 12, 15kg); sieve angles 1, 2, and 3 (2, 4, and 6 degrees); and also air velocity (3, 2.1, 1.3m/sec) was determined below.

4.2.1. Performance Evaluation on Wheat Crop

Cleaning Efficiency (%)

The mean cleaning efficiency and analysis of the variance were presented in the Figure and Appendix Table individually. Hence, the statistical ANOVA shows that the cleaning efficiency of the machine was high significantly affected by the feed rate, sieve angle, and air velocity at $p < 0.05$ beyond that the cross of feed rate with sieve angle; feed rate with air velocity; and sieve angle with air velocity had a significant effect and also the cross effect of the three-factor had a less significant effect on the cleaning efficiency. Furthermore, the response plot of the cleaning efficiency versus the three independent variables in Figure 4.1 illustrated that 100% cleaning efficiency was found on sieve angle one, feed rate one, and air velocity three also lowest cleaning efficiency 55% found at feed rate three and air velocity one.

Besides, the evaluation shows that the cleaning efficiency decreased when the feed rate, sieve angle, and air velocity increase. A similar result was reported by Muhammad et al., (2013) and Simonyan, (2008) that cleaning efficiency decreased with increasing feed rate. Also, Okunola et al., (2015) and Afolabi et al., (2019) indicated that the sieve angle and air velocity increased in reverse the cleaning efficiency of the machine decreased.

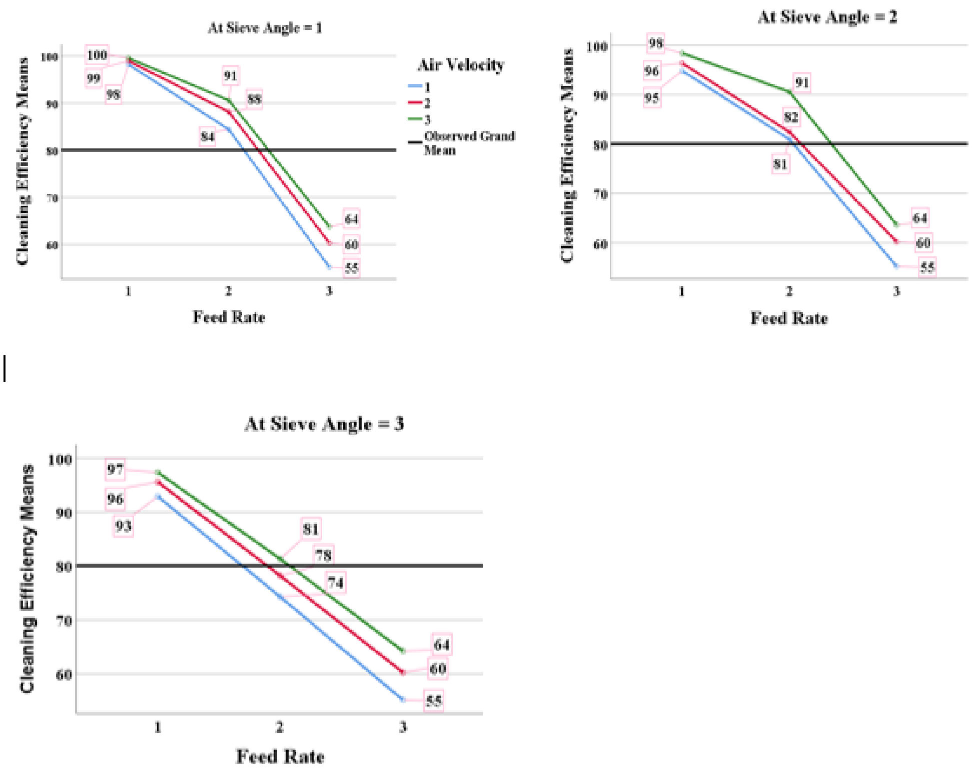


Figure 4.1. Effect of FR, SA, and AV on cleaning efficiency for wheat

Separation Efficiency (%)

The effect of feed rate, sieve angle, and air velocity on the separation efficiency of the cleaning machine was reported in Figure and Appendix Table. The response surface plot showed in Figure 4.2 indicated the highest separation efficiency of the machine on the wheat crop was 100 % obtained at feed rate one, sieve angle three, and air velocity while the lowest separation efficiency 60% found at feed rate one, sieve angle two and air velocity three. Additionally, the interaction of feed rate, sieve angle, and air velocity were significant at the 95% level determined in the Appendix Table (15). Therefore, the effect of feed rate, sieve angle, and air velocity is highly significant ($p < 0.05$). Plus the cross effect of feed rate and sieve angle; feed rate and air velocity; sieve angle and air velocity; and also the three-factor were significant at $p < 0.05$.

The separation efficiency of the machine decrease with the increase of the feed rate from one up to three as a result of massive properties of the grain mixture which reduce the air velocity capacity to remove the MOG but it increases within the increasing of the sieve angle and air velocity from one to three. Some researchers evaluate a machine according to feed rate sieve angle and air velocity within it determined the increase of feed rate lead to decreasing separation efficiency of the machine and also increase of sieve angle and air velocity (Ojediran et al., 2018; Okunola et al., 2015; Eissa, 2009)

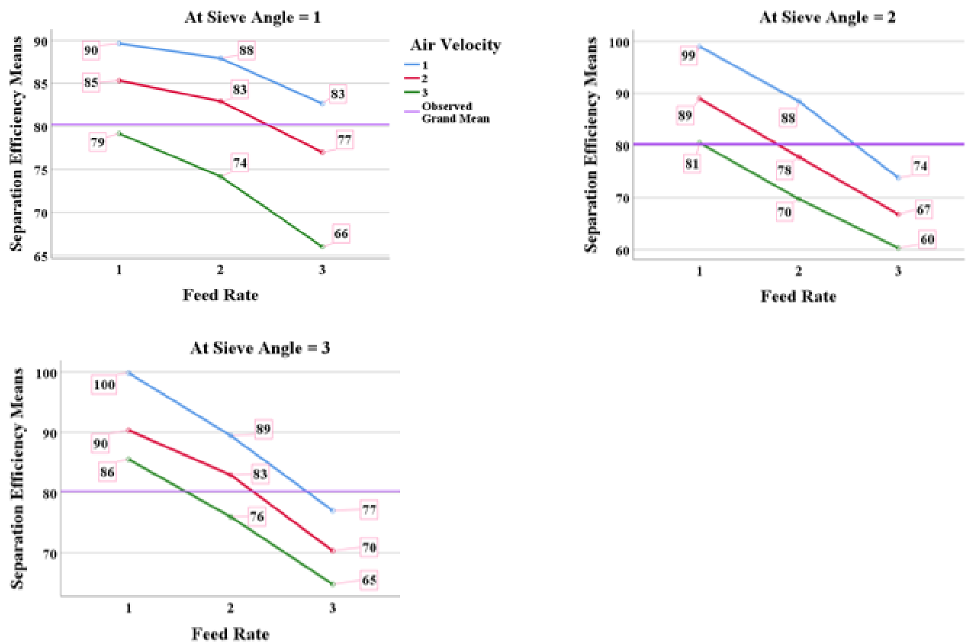


Figure 4.2. Effect of FR, SA, and AV on the separation efficiency for wheat

4.2.2. Performance Evaluation of Maize Crop

Cleaning Efficiency (%)

The analysis of the cleaning efficiency of the cleaning machine on the maize crop is given in the Figure and Appendix Table. Analysis of the variance revealed that feed rate, sieve angle, and air velocity had a significant effect on the cleaning efficiency. The interaction of feed rate and sieve angle; feed rate and air velocity; sieve angle and air velocity; also the cross interaction of the three factors have a significant effect at $p < 0.05$ (Appendix Table 13). Therefore, Figure 4.3 illustrates the maximum cleaning efficiency of 100% was found at feed rate one, sieve angle one, and air velocity three but the result shows that on feed rate one and feed rate two the result of cleaning efficiency was similar at sieve angle one and three. Even though, the lowest cleaning efficiency of the machine for maize crop of 37% was determined at sieve angle three, feed rate three, and air velocity one.

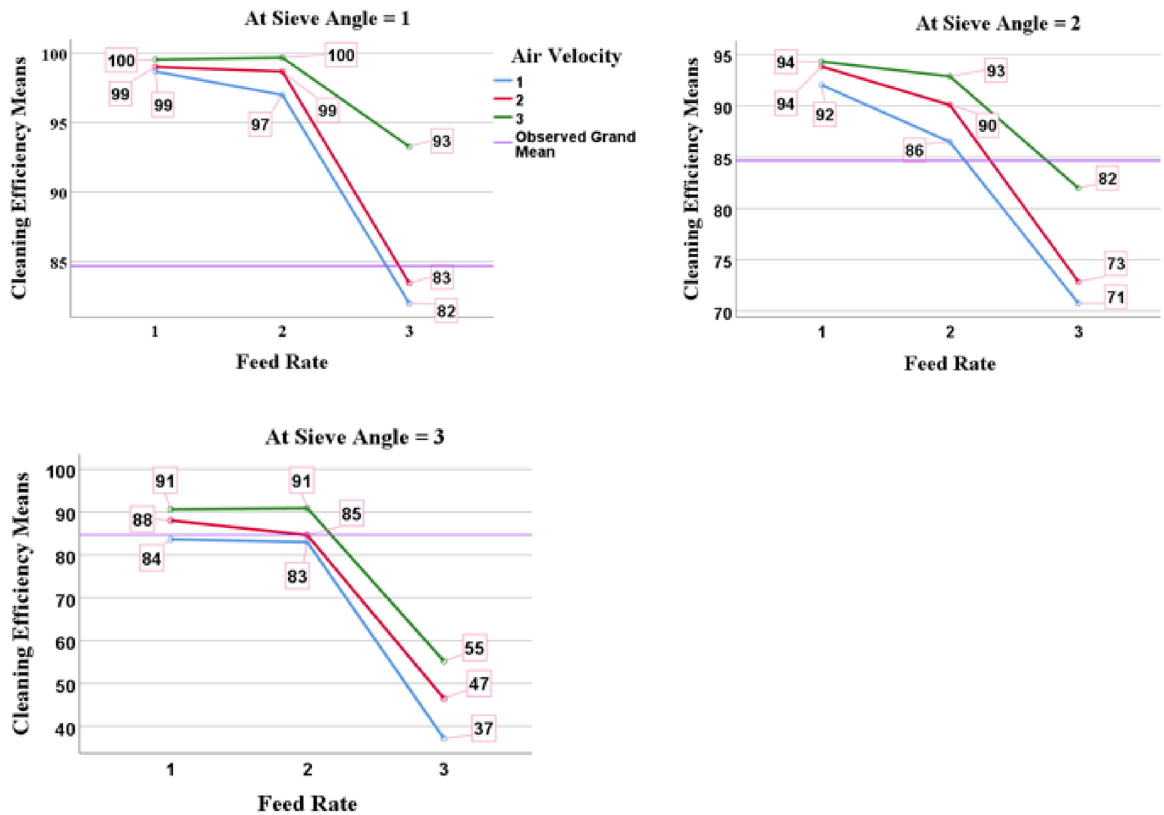


Figure 4.3. Effect of FR, SA, and AV on cleaning efficiency for maize

Generally, the result shown in Figure 4.3 illustrates that an increase in feed rate leads to decreasing the cleaning efficiency of the machine as a result of reducing the chance of passing through the perforation of the sieve as well as reduce the capacity of the air velocity to remove the particle. Besides, it decreases with the increase of the sieve angle which lessens the time staying on the sieve; likewise, it decreases with the increase of the air velocity.

Separation Efficiency (%)

The influence of feed rate, sieve angle, and air velocity on the separation efficiency of the cleaning machine for maize products are presented in the Appendix Table and Figure. The highest separation efficiency as shown in Figure 4.4 was 97% found at feed rates one and two; sieve angle two, and air velocity one. Hence, the lowest separation efficiency occurred at feed rate three, sieve angle one, and air velocity three. Besides, the analysis of variance in the Appendix Table (16) revealed that the effect of feed rate, sieve angle, and air velocity was significant on a percentage of separation efficiency. Additionally, the interaction of the feed rate with sieve angle; feed rate with air velocity; and also sieve angle with air velocity has a significant effect but the cross effect of the three-factor was not significant at $p < 0.05$.

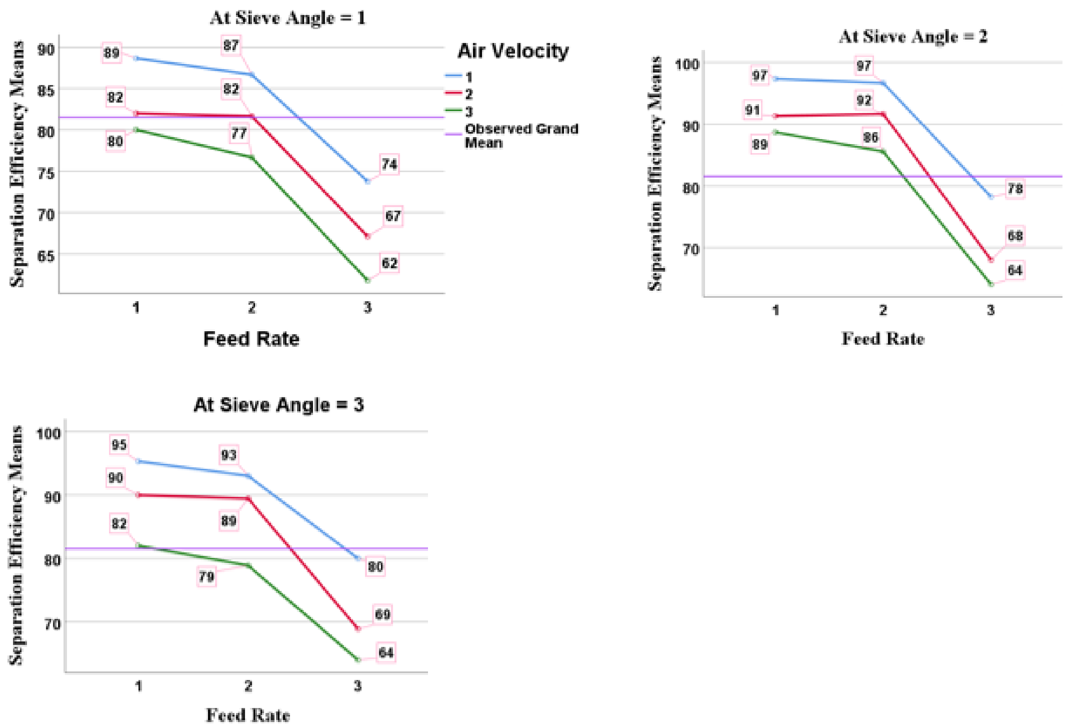


Figure 4.4. Effect of FR, SA, and AV on the separation efficiency for maize

The separation efficiency of the machine decrease with the increase of feed rate. This was recognized by increasing the amount of the MOG that came out through the cleaned grain outlet. Along with, the separation efficiency was almost the same at feed rate one and feed rate two as shown in Figure 4.4; which predicts that at a given feed rate the effect was almost negligible. Beyond that, the separation efficiency decrease with the decrease of sieve angle and air velocity.

4.2.3. Performance Evaluation of Sorghum Crop

Cleaning Efficiency (%)

The mean and relations of cleaning efficiency of the cleaning machine at different feed rates, sieve angle, and air velocity as illustrated in Figure 4.5. Furthermore, the analysis of variance on the cleaning efficiency revealed that each feed rate, sieve angle, and air velocity were significantly affected the cleaning efficiency at $p < 0.05$ shown in Appendix Table 14. Hence, the interaction of feed rate with sieve angle; feed rate with air velocity; sieve angle with air velocity, and the cross of the three factors had a highly significant ($p < 0.05$) effect on cleaning efficiency. Therefore, the maximum cleaning efficiency of 100% occurred at feed rate one, sieve angle one, and air velocity one; while the lowest cleaning efficiency of 75% was recorded at feed rate three, sieve angle three, and air velocity three.

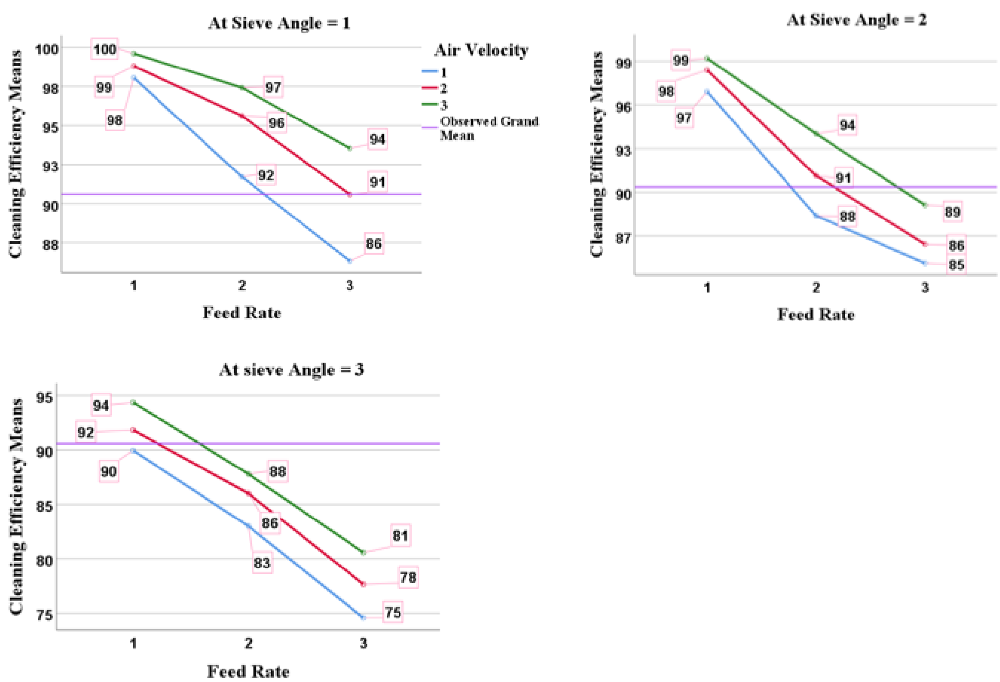


Figure 4.5. Effect of FR, SA, and AV on cleaning efficiency for sorghum

The cleaning efficiency of the machine has been increased as the feed rate, sieve angle, and air velocity decrease. From Figure 4.5, it is seen that percentage of cleaning efficiency has no significant differences at sieve angle one and sieve angle two within feed rate one but there are significant differences with sieve angle three.

Separation Efficiency (%)

The separation efficiency analysis was present in the Figure and Appendix Table with the influence of the feed rate, sieve angle, and air velocity. The analysis is used to define the impact of each factor on the separation. Therefore, the statistical analysis, ANOVA has clearly shown that the cleaning efficiency of the cleaning machine was significantly affected by feed rate, sieve angle, and air velocity. Also, the combined effect of feed rate with sieve angle; feed rate with air velocity; sieve angle with air velocity; and the cross of the three-factor were significant at $p < 0.05$(Appendix Table 17). The maximum separation efficiency of 97% was found at feed rate one, sieve angle three, and air velocity one; while minimum separation efficiency of 55% was found at feed rate three, sieve angle three, and air velocity three as illustrated in Figure 4.6.

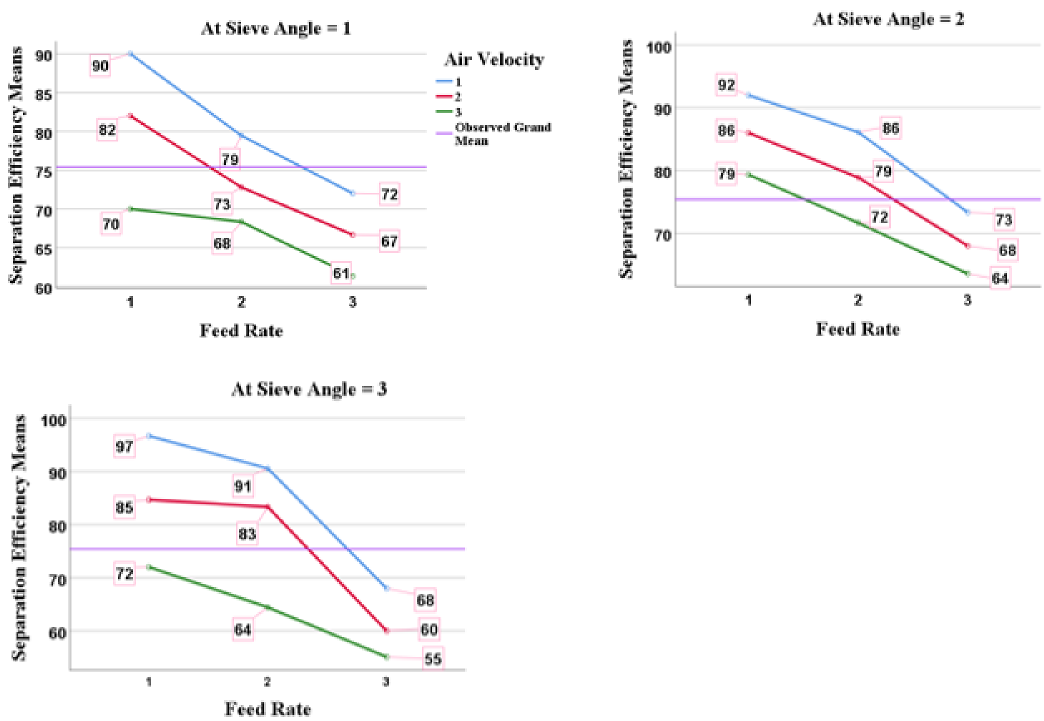


Figure 4.6. Effect of FR, SA, and AV on the separation efficiency for sorghum

The separation efficiency of the cleaning machine is mention in sections (4.2.1.) and (4.2.2.) decrease with an increase in feed rate as shown in Figure 4.6. Besides, the separation efficiency increase with the air velocity and sieve angle increasing; the percent of the separation efficiency shown a significant difference between air velocity one and three at sieve angle three.

4.2.4. Multiple Regression Analysis Factor Combination for Three Crops

The regression analysis was established to determine the relationship between feed rate, sieve angle, and air velocity on the cleaning efficiency and separation efficiency of the multi-cleaning machine. Hence, it can be seen in Table 4.7 the percentage of cleaning efficiency and separation efficiency is highly affected by feed rate, sieve angle, and air velocity. Besides, as shown in the table feed rate, sieve angle and air velocity had a negative effect on cleaning efficiency as a result of that the when the factors increasing the cleaning efficiency also decreased. And also, feed rate and air velocity had a negative effect but sieve angle has a positive effect on the separation of the efficiency.

Table 4.7. Multiple regression equation on cleaning efficiency and separation efficiency

Crops	Regression equation	R^2
Wheat	CE = 115.1 – 18.57FR – 2.24SA - 3.24AV	0.96
	SE = 111.3 – 8.87 FR + 0.62 SA – 7.31AV	0.90
Maize	CE = 122 - 12.03FR – 10.63SA - 3.76AV	0.73
	SE = 107.7 - 9.42FR + 2.6SA – 6.2AV	0.78
Sorghum	CE = 106.9 – 5.73FR – 4.77SA - 2.31AV	0.94
	SE = 108.2 – 9.15FR + 0.68SA – 7.9AV	0.85

CHAPTER FIVE

5. CONCLUSION AND RECOMMENDATION

The current study aimed to design, fabricate, and testing of the machine on wheat, maize, and sorghum crops. Therefore, the study was carried out in Melkassa Agricultural Research Center to fabricate and testing, and evaluating the performance of the machine. The machine has a hopper, sieve, brush, outlets, fan, power transmission units, and electric motor as the major components.

Besides, the testing of the machine was prepared as per statistical procedures; data was collected and analysis was done by using computer software. The study includes the evaluation of the effect of feed rate (10, 12, and 15kg), sieve angle (2.4, and 6 degrees), and air velocity (3, 2.1, and 1.3m/sec) as an independent variable on cleaning efficiency and separation efficiency. Also, the cost of the machine was the part of the study; the total production cost of the machine was found to be 39,838.00 Birr and the annual cost of the operation was estimated to be 55,447 birrs. The performance of the machine concludes in the following stopping point and recommendation of the study.

5.1. Conclusion

The current study discusses the testing and performance evaluation of the machine for wheat, maize, and sorghum. Consequently, the performance evaluation was made to determine the effect of feed rate, sieve angle, and air velocity on the cleaning efficiency and separation efficiency. Therefore, the optimum operating condition was estimated based on measurements made and analysis carried out. Accordingly, the following conclusions were prepared.

- The optimum combination of the factors at feed rate 10kg, sieve angle 4°, and air velocity 3m/sec for both cleaning and separation efficiency but sieve angle was different for three crops within 4° for wheat, 2° for maize, and 6° for sorghum accordingly was considered.
- The decrease in cleaning efficiency and separation efficiency was highly dependent on the feed rate rather than other factors.

- The cleaning efficiency was increased with the decrease of feed rate, air velocity, and sieve angle. Besides, the separation of efficiency increase with the increase of air velocity and sieve angle but it decreases with the increase of feed rate.
- Based on the results obtained, it can be concluded that the performance of the prototype of the machine is very much acceptable with the high prospect for extending the technology.

5.2. Recommendation

The traditional cleaning mechanism in Ethiopia is consuming much time, drudgery, and requires too much labor to complete an activity as a result of all of this needs to modernize through the introduction of manageable technology to the farmer and also other processing industries. Therefore, to fulfill the reduction of the impact of traditional cleaning systems multi-grain cleaning machines developed. The following are also recommended as potential research recommendations based on the result of the study.

- The low-income farmers should think of the systematic, coordinated, and relentless effort to buying and owning the machine jointly with each other to make it affordable.
- From the result, there was noticed that the machine should be evaluated again with a variation of the factor at different sieve oscillation and also fan speed.
- Instead of using three phases, the electric motor must consider using a single-phase motor as a result of not easily finding the source of a three-phase installation line to connect.
- The cleaning machine needs a transport wheel as a means of transportation of the machine from one place to another place within draft requirement low and easily transportable.
- The design of the fan was depending on the works of literature of what the other researchers recommended but for further study, there should be consideration of measuring aerodynamic properties of the crops.
- The design of the hopper should be modified by using the stirring mechanism to avoid clogging.

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APPENDICES

Table 1.Parameter for v - belt section selection

Section	Kw Range	Min pulley pitch diameter (mm)	Width (mm)	Thickness (mm)
A	0.4 – 4	125	13	8
B	1.5 -15	200	17	11
C	10 – 70	300	22	14
D	35 - 150	500	32	19

Source: (EA854, 2000)

Table 2.Density of belt material

Material of belt	Mass density in kg/m^2
Leather	1000
Convass	1220
Rubber	1140
Balata	1110
Single woven belt	1170
Double woven Belt	1250

Source: (Khurmi & Gupta, 2005b)

Table 3.Recommended value for k_m and k_t

No	Nature of load	k_m	k_t
1	Stationary shafts		
	(a) Gradually applied load	1.0	1.0
	(b) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2	Rotating Shafts		
	(a) Gradually applied load or steady load	1.5	1.0
	(b) Suddenly applied load with minor shocked only	1.5 to 2.0	1.5 to 2.0
	(c) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

Source: (Khurmi & Gupta, 2005b)

Table 4.Theoretical stress concentration factor (c_b) for a stepped shaft with a shoulder fillet (of radius r) in bending

$\frac{D}{d}$	Theoretical stress concentration factor (c_b)									
	r/d									
	0.02	0.04	0.08	0.1	0.12	0.16	0.20	0.24	0.28	0.30
1.01	1.85	1.61	1.42	1.36	1.32	1.24	1.20	1.17	1.15	1.14
1.02	1.97	1.72	1.50	1.44	1.40	1.32	1.27	1.23	1.21	1.20
1.05	2.20	1.88	1.60	1.53	1.48	1.40	1.34	1.30	1.27	1.25
1.10	2.36	1.99	1.66	1.58	1.53	1.44	1.38	1.33	1.28	1.27
1.20	2.52	2.10	1.72	1.62	1.56	1.46	1.39	1.34	1.29	1.28
1.50	2.75	2.20	1.78	1.68	1.60	1.50	1.42	1.36	1.31	1.29
2.00	2.86	2.32	1.87	1.74	1.64	1.53	1.43	1.37	1.32	1.30
3.00	3.00	2.45	1.95	1.80	1.69	1.56	1.46	1.38	1.34	1.32
6.00	3.04	2.58	2.04	1.87	1.76	1.60	1.49	1.41	1.35	1.33

Source: (Khurmi & Gupta, 2005b)

Table 5.Theoretical stress concentration factor (c_t) for a stepped shaft with a shoulder fillet (of radius r) in torsion

$\frac{D}{d}$	Theoretical stress concentration factor (c_t)									
	r/d									
	0.02	0.04	0.08	0.10	0.12	0.16	0.20	0.24	0.28	0.30
1.09	1.54	1.32	1.19	1.16	1.15	1.12	1.11	1.10	1.09	1.09
1.20	1.98	1.67	1.40	1.33	1.28	1.22	1.18	1.15	1.13	1.13
1.33	2.14	1.79	1.48	1.41	1.35	1.28	1.22	1.19	1.17	1.16
2.00	2.27	1.84	1.53	1.46	1.40	1.32	1.28	1.22	1.19	1.18

Source: (Khurmi & Gupta, 2005b)

Table 6.Basic number of bearing and capacity of bearing

Bearing No.	Bore	Outside dia	Width (mm)	Basic capacity in KN			
				Single row deep groove ball bearing		Single row angular contact ball bearing	
				Static	Dynamic	Static	Dynamic
202	15	35	11	3.55	6.10	3.75	6.30
203	17	40	12	4.4	7.5	4.75	7.8
204	20	47	14	6.55	10	6.55	10.4
205	25	52	15	7.1	11	7.8	11.6
206	30	62	16	10	15.3	11.2	16
207	35	72	17	13.7	20	15.3	21.2

Source: (Khurmi & Gupta, 2005b)

Analysis of variance on physical properties

Table 7.ANOVA on physical properties of wheat (between varieties)

Treatment	SS	df	MS	F	P - value	F - crit
Length	2.094	2	1.047	10.029	6.098E - 05	3.0262
Width	3.1903	2	1.5952	16.836	1.185E – 07	3.0261
Thickness	1.9369	2	0.9684	9.1112	1.443E - 04	3.026
Geometric mean dia	2.1339	2	1.067	16.324	1.88E - 07	3.026
Arithmetic mean dia	1.9645	2	0.9822	15.857	2.86E - 07	3.026
Sphericity	165.4	2	82.701	7.709	5.44E - 04	3.026

Table 8.ANOVA on physical properties of maize (between varieties)

Treatment	SS	df	MS	F	P - value	F - crit
Length	36.083	2	18.042	13.208	3.2E - 06	3.026
Width	28.342	2	14.174	17.939	4.41E - 08	3.026
Thickness	6.638	2	3.319	5.023	7.16E - 03	3.026
Geometric mean dia	1.524	2	0.762	3.558	2.97E - 02	3.026
Arithmetic mean dia	6.407	2	3.204	13.598	2.24E - 06	3.026
Sphericity	802.01	2	401.01	7.886	4.6E - 04	3.026

Table 9.ANOVA on physical properties of sorghum (between varieties)

Treatment	SS	df	MS	F	P - value	F - crit
Length	25.75	2	12.872	191.3	4.09E - 54	3.026
Width	74.675	2	37.337	454.57	4.13E - 91	3.026
Thickness	0.39	2	0.197	6.619	1.54E - 03	3.026
Geometric mean dia	14.923	2	7.462	209.53	1.7E - 57	3.026
Arithmetic mean dia	19.495	2	9.747	267.79	3.32E - 67	3.026
Sphericity	341.769	2	14.704	14.704	8.14E - 07	3.026

Table 10.ANOVA on angle of repose of crops (between varieties)

Crops	SS	Df	MS	F	P - value	F - crit
Wheat	32.6	2	16.3	4.779	1.67E – 02	3.354
Maize	1.067	2	0.533	0.59	5.6E – 01	3.354

Sorghum	1.667	2	0.833	0.63	5.4E – 01	3.354

Table 11.ANOVA on coefficient of frictions of crops (between varieties)

Crops	SS	Df	MS	F	P - value	F - crit
Wheat	0.015	2	0.008	4.861	1.57E - 02	3.354
Maize	0.0004	2	0.0002	0.569	5.73E – 01	3.354
Sorghum	0.0008	2	0.0004	0.639	5.39E – 01	3.354

Table 12.ANOVA on cleaning efficiency of machine on wheat crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	19100.1	2	95550.03	89012.77	0.000	1.000
SA	273.621	2	136.811	920.657	0.001	0.998
AV	568.568	2	284.284	3111.74	0.000	0.999
FR * SA	244.761	4	61.19	302.811	0.003	0.993
FR * AV	85.88	4	85.88	110.9	0.009	0.982
SA * AV	16.45	4	12.197	21.78	0.023	0.916
FR * SA * AV	21.55	8	17.609	22.495	0.027	0.918

Table 13.ANOVA on cleaning efficiency of machine on maize crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	9685.86	2	4842.93	1207.5	0.000	1.000
SA	6192.2	2	3.096.1	10042	0.000	1.000
AV	775.89	2	775.89	1253.85	0.000	0.998
FR * SA	2782.66	4	695.415	2817.405	0.000	0.999

FR * AV	281.629	4	70.407	262..647	0.004	0.992
SA * AV	90.148	4	22.537	80.783	0.012	0.76
FR * SA * AV	43.198	8	5.4	16.343	0.056	0.891

Table 14.ANOVA on cleaning efficiency of machine on sorghum crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	1775.19	2	887.59	333355.1	0.000	1.000
SA	1319.83	2	659.9	18644.9	0.000	1.000
AV	288.98	2	144.49	2957.97	0.000	0.999
FR * SA	91.514	4	22.87	848.48	0.001	0.998
FR * AV	24.908	4	24.908	62.579	0.016	0.69
SA * AV	4.515	4	4.515	72.323	0.014	0.973
FR * SA * AV	14.533	8	14.533	43.67	0.022	0.956

Table 15.ANOVA on separation efficiency of machine on wheat crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	4274.6	2	2137.32	15511.4	0.000	1.000
SA	131.326	2	159.8	794.1	0.001	0.997
AV	2888.1	2	1444.1	31594.8	0.000	1.000
FR * SA	468	4	117	1164.7	0.000	0.998
FR * AV	9.269	4	8.15	16.8	0.02	0.892
SA * AV	63.28	4	63.28	60.06	0.016	0.968
FR * SA * AV	60.529	8	60.52	31.27	0.031	0.940

Table 16.ANOVA on separation efficiency of machine on maize crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	5870.27	2	2935.14	1307.71	0.000	0.998
SA	694.22	2	347.11	118.5	0.008	0.983
AV	1954.68	2	977.34	290.21	0.003	0.993
FR * SA	144.72	4	36.18	6.822	0.011	0.77
FR * AV	74.516	4	18.62	4.2	0.04	0.678
SA * AV	57.078	4	14.27	4.357	0.037	0.685
FR * SA * AV	27.704	1.649	16.8	1.27	0.376	0.388

Table 17. ANOVA on separation efficiency of machine on sorghum crop

Treatment	SS	Df	MS	F	Sig	Partial Eta Squared
FR	4660.6	2	2330.3	2501.9	0.000	0.999
SA	277.87	2	113.93	25.903	0.005	0.928
AV	3382.9	2	1691.46	321.53	0.000	0.994
FR * SA	352.58	4	88.145	30.218	0.01	0.938
FR * AV	173.19	4	43.29	15.79	0.001	0.888
SA * AV	211.60	4	52.90	9.133	0.004	0.820
FR * SA * AV	153.92	8	19.24	4.218	0.007	0.678

Table 18.Raw material costs

No	Material	unit	Quantity	Unit cost (birr)	Total cost
1	40 × 40 × 3 mm SHS	m	9	1800	2100
2	30 × 30 × 3 mm angle iron	m	9.2	400	613

3	Sheet metal = 1.5 mm thickness	m^2	1.3	1450	943
	= 1.25 mm thickness		2	1100	1100
4	Mils steel shaft; 30 mm	m	1	2500	420
	25mm		1.2	1700	340
	20 mm		0.5	1350	120
5	V – belt = B 101	pcs	1	460	460
	= B 83		1	250	250
	= B53		1	130	130
6	Bearing With housing	pcs	6	320	1920
7	Bearing without housing	pcs	6	110	660
8	Flat iron = 30 × 5 mm	m	3.6	700	420
	= 30× 3 mm		6	400	400
9	Hollow pipe ∅ 15 mm	m	1.2	1200	240
10	Aluminum pulley = ∅ 350 mm	pcs	2	950	1900
	= ∅ 170 mm		1	700	700
	= ∅ 80 mm		2	500	1000
11	U channel 15 × 50 mm	m	1.4	2500	583
12	Square pipe 25 × 40 mm	m	1.74	800	232
13	Brush 50 × 600 mm	pcs	6	200	1200
14	Sieve 700 × 800 × 1.25 mm	pcs	7	750	5250
15	M6 × 20	No.	24	4	96
16	M10 × 80	No.	20	5	100
17	M8 × 10	No.	16	6	96
18	M20 × 80	No.	2	15	30
19	Electric motor	No.	1	6000	6000
20	Anti-rust	kg	1	70	70
21	Cutoff disc	pack	2	250	500
22	Electrode ∅ 2.5	pack	2	250	500
23	Paint	Gallon	1	650	650
Total					28, 827

Table 19.Machine and labor costs

No.	Type of machine	Machine cost/hrs.	Working hours	Total cost	Labor cost/hrs.	Labor total cost/hrs.
1	Lathe machine	20	3	60	25	75
2	Grinding machine	7	10	70	25	250
3	Universal shearing	65	3	195	25	75
4	Welding machine	30	15	450	25	375
5	Rolling machine	15	2	30	25	50
6	Bending machine	5	1	5	25	25
7	Power hack saw	11	3	33	25	75
8	Drilling machine	3	3	9	25	75
	sub total			854		1000
	Total					1854

Table 20.Cost summary

No.	Cost variables	Cost
A	Raw materials cost	28,827
B	Material wastages (2.5%)	720.65
C	Machine cost	854
D	Labor cost	1000
E	Overhead cost 5% (C+D)	92
F	Profit 10% (A+B+C+D+E)	3149.3
G	Selling tax 15% (A+B+C+D+E+F)	5196.3
	Selling price	39838

Table 21.Cost analysis for the machine

Cost of the machine	39838 birr
Fixed cost	
Salvage value (S)	3983.8 birr
Depreciation (D)	4481.8 birr
Interest	2191 birr

Tax	398.4 birr
Insurance	99.6 birr
Housing	298.8 birr
Total fixed cost	51290 birr
Variable cost	
Repair and maintenance	995 birr
Electricity cost	2154.8 birr
Labor cost	1007.5 birr/yr
Total variable cost	4157.3 birr
Total cost	51290 + 4157 = 55447 birr

ANNEES

Measuring Physical dimension, Angle of repose and moisture content of a crops

Manufacturing of the machine

Preparation the sample and testing the performance of the machine