Design, Development, and Performance Evaluation of Carrot Washing and Grading Machine

Mustefa Jarso Godana

A Thesis Submitted to

Department of Mechanical Engineering

School of Mechanical, Chemical, and Material Engineering

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

Office of Graduate Studies

Adama Science and Technology University

December, 2021

Adama, Ethiopia

Design, Development, and Performance Evaluation of Carrot Washing and Grading Machine

Mustefa Jarso Godana

Advisor

Dr. Amana Wako Koroso (Ph.D.)

A Thesis Submitted to

Department of Mechanical Engineering

School of Mechanical, Chemical, and Material Engineering

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

Office of Graduate Studies

Adama Science and Technology University

December, 2021

Adama, Ethiopia

STATEMENT OF THE AUTHOR

I hereby declare that this Master Thesis entitled “Design, Development, and Performance Evaluation of Carrot Washing and Grading 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.

Mustefa Jarso		
Name of the student	Signature	Date

DEDICATION

Thank you, my beautiful parents, for everything you've done! This thesis would have remained an incomplete process if it hadn't been for your unending love, patience, support, and encouragement. Know that through the act of this dedication, I express enormous thanks and gratitude for your vital role in the realization of this achievement with all my love.

ACKNOWLEDGMENT

All praise is due to Allah, the world's Cherisher and Sustainer. Thank you, Allah! First and foremost, my heartfelt gratitude. Your support has never differentiated between day and night in my work. You are providing me with strength and health. Everything is made easier for me because of you. All honor and glory to you! Thank you, Allah! I respectfully suggest and hope that you regard all of my efforts to be only for your benefit.

I will never be able to adequately convey my gratitude to my respected adviser, Dr.Amana Wako, for his invaluable counsel, continual direction, and assistance throughout the research study. I gratefully welcome the helpful recommendations obtained from past ASTU Agricultural Machinery MSc students, particularly Mr. Alamayo Teshoma and Mr. Yemane Woldeyesus, who assisted me during the project. I'd like to express my gratitude to the Waltai Development Association members for their contributions of additional resources and funds for this study, particularly to Mr. Kemal Jewar, a Ph.D. student at Turkey Istanbul University.

Furthermore, I express my gratitude to Mr. Eyasu Tsagaye and Sultan Tuna for their invaluable assistance in completing the machine's construction and for allowing their workshop materials and tools to be used. For their assistance in providing the various facilities, I am grateful to MoSHE, HU, ASTU, Department of Mechanical System and Vehicle Engineering at ASTU, and Agricultural Engineering Department at HU.

No words can adequately describe my gratefulness to my loving parents and brothers, Mr. Jarso Godana, Mrs. Jamila Habib, Mr. Kadir Jarso and all my brothers, and my cousin and close buddy Husein Jemal, for their unwavering support and efforts in molding my personality throughout my life.

It's impossible for me to express how grateful I am to Mrs. Safiya Abdurhman, the mother of my children and my beloved wife, for her unwavering support and encouragement during this journey.

Finally, because emotions cannot be adequately stated in words, my gratitude extends well beyond what I have expressed here.

TABLE OF CONTENTS

CONTENTS **PAGES**

LIST OF TABLE

TABLES **PAGES**

TABLES IN APPENDICES

LIST OF FIGURES

FIGURES **PAGES**

ACRONYMS AND ABBREVIATION

ABMA	American Bearing Manufacturers Association
AIChE	American Institute of Chemical Engineers
ANOVA	Analysis of Variance
ASTU	Adama Science and Technology University
CATIA	Computer-Aided Three-dimensional Interactive Application
CFD	Computational Fluid Dynamic
CISA	Cyber security and Infra srtucture Security Agency
CNC	Computer Numerical Control
CRD	Completely Randomized Design
CSA	Central Statistical Agency
CV	coefficient of variation
DRDGBB	Double row deep groove ball bearing
Eqn	Equation
et. al.	And other
Etc.	Etcetera
FAO	Food and Agriculture Organization of the United Nations
GDP	Gross Development Production
GMD	Geometric Mean Diameter
HDPE	High-Density Polyethylene
HU	Haramaya University
I.D.C	inner dead center
ISO	International Organization for Standardization
LSD	least significance difference
Masl	The meter above sea level
MC	Machine capacity
MHPR	Mean heart pulse rate
Min	minute
MoSHE	Ministry of Science and Higher Education
NA	Needle Roller Bearings

O.D.C	outer dead center
OCR	oxygen consumption rate
PVC	Polyvinyl Chloride
Pvt Ltd	Private Limited Company
RCBD	Randomized Complete Block Design
Rpm	Revolution per minute
SAS	Statistical Analysis System
SKF	Svenska Kullager Fabriken
SNNPR	South Nation Nationality People Region
SPD	Split Plot Design
SPSS	Statistical Package for the Social Sciences
SRDGBB	Single row deep groove ball bearing
UCF	Bearing Code and its element
UCP	Bearing Code and its element
We	Washing efficiency

ABSTRACT

Carrot washing and grading, which is done after harvesting and before delivering to the market, has been done by farmers in Ethiopia using conventional methods that are too laborious, time-consuming, and labor-intensive, resulting in decreased carrot productivity and health problems. The major goal of this study was to design, develop, and test a carrot washing and grading machine that was not utilized for cooking purposes after harvesting before delivering to market in order to solve the concerns mentioned above. Based on literature reviews, the pump's power source, beginning speed, and discharge capacity were set at 50W, 60rpm, and 0.036l/s, respectively, during design. The bicycle chain, pedal, pulleys, belts, and shafts were chosen to transfer human power to the planned machine. Before building, all components were modeled using Computer-Aided three-dimensional Interactive Application software (CATIA V5). The Nantey variety was used to evaluate the machine's performance at two levels of feeding loads: 10kg and 15kg, as well as three distinct drum speeds: 14rpm, 21rpm, and 28rpm. The drum speed and feeding load were assigned as treatments in a completely randomized design (CRD). The data was collected using both qualitative and quantitative approaches, and the results were analyzed using SPSS software at a 5% level of significance. Overall, the mean findings for time required to complete washing and grading activities, washing efficiency, grading efficiency, percentage of damaged tuber, and throughout put capacity were 2.22 minutes, 98.70 percent, 92.23 percent, 1.21 percent, and 242.17 kg/hr, respectively. For the range of the variable of drum speed between 14-28 rpm and for the range of the variable of feeding load 10-15kg, the evaluation findings revealed time requirements of 2 to 10.88 minutes and washing efficiencies of 84.87 to 98.70 percent. As the speed rose (14 to 28 rpm), the grading efficiency increased 72.89 to 92.23 percent, and the feeding load increased from 10 to 15kg. The carrot washing and grading machine require no formal education to use and maintain, and it was built with locally sourced materials. This equipment cost birr 16363.52 per unit, which is a low price that most Ethiopian farmers can afford.

Keywords: Carrot, Carrot Washer, Grading, Pedal-Operated, Washing, Human Power, Pump

CHAPTER ONE

1. INTRODUCTION

Carrots are a root crop whose roots are consumed by millions of people throughout the world (Jahanbakhshi, 2018). It has consumed fresh, ripe, and mixed crops. Carrots can also be ingested as oil, juice, dehydrated items, and so on. It is one of the most significant and beneficial vegetables for the human body since it contains vitamins A, B, C, -carotene, thiamine, riboflavin, and other minerals, among others (Surbhi et al., 2018; Ahmad et al., 2019; Bhattarai et al., 2017; & Bender et al., 2020). Because of these nutritional values and the reduced incidence of night blindness in mothers and children caused by vitamin-A deficiency, as well as the risk of heart disease, stomach diseases, and many types of cancer, the carrot is cultivated in many countries such as America, Europe, South-west Asia, and Africa (Bhattarai et al., 2017 & Yadav, 2020).

Carrot grows best in loose, sandy loam soils that are well-drained, above 1000 masl where the temperature is between 16 and 21^{o}C and rainfall is between 800 mm and ready for harvest 100 to 120 days after planting depending on the agro ecology of the growing area (Rashid et al., 2018; Hussain & Ali, 2020). In Ethiopia, carrots are cultivated at an altitude range of 300-3000masl mainly in Oromia, Amhara, Addis Ababa, Tigray, and SNNPR (CSA, 2020). This means it is produced in a wide range of agro-ecologies from the lowlands to the highlands of Ethiopia. Even though it can cultivate in a varied range of agro-ecologies, Ethiopia produced the very lowest production per year compared to developed countries (FAO, 2018).

This results in a scarcity of carrots, which is one of the causes of various maladies in the country, such as vitamin A deficiency, which causes Bitot's spots and night blindness. Bitot's spots, a medical symptom caused by a deficiency of this vitamin in the country, were seen in 9 percent of girls and 2.2 percent of boys among 7,000 preschool and school-aged children (Tabor & Mohammed, 2012).These reveal that Producing and promoting the carrot for consumption in Ethiopia is a fundamental issue to reduce the above stated and other diseases.

Post-harvest actions such as washing and grading have a significant impact on carrot production and promotion. Farmers do not nurture the carrot well when these actions are not mechanized because they anticipate the dull activities that will await them in the future. Not only that, but due to the boring duties like washing and grading that await them in the next phase, the farmed crop is not properly gathered. If the actions after harvesting are not well mechanized, these and similar conditions render farmers immoral for producing carrots. Furthermore, washing and grading carrots after harvest before shipping them to market has a significant impact on improving product quality and marketability, reducing post-harvest loss, removing soil and debris, and lowering product temperature and microbial load (Alam et al., 2018).

After harvesting, farmers washed and graded their crops before transporting them to roadsides where traders may readily reach them or to neighboring markets in the rural region. They can be done in a traditional or mechanical way. The old process of washing and grading is too time-consuming and labor-intensive, resulting in health issues and significant post-harvest losses. (Bhattarai and colleagues, 2017; Abdul and colleagues, 2020).

Different carrot and fruit washing machines powered by electric motors, engines, and manually cranked by hands were conceived and developed in various countries to alleviate the drudgery of the conventional approach. Electric motor machines are difficult to use in remote areas due to a lack of electric power distribution. Engine-driven devices are energy-intensive and costly, necessitating a large initial capital investment that is unsuitable for small-scale farmers. In rural locations, according to Seweh et al. (2016), installing electric motor and engine machines quickly and easily is problematic. Manually operated or hand-cranked machines are one of the possibilities for rural areas, but they must be replaced with other appropriate techniques such as pedal operating machines because anyone can generate around 2-4 times the power of a hand crank using leg pedaling (Mushiri et al., 2017 and Jadoun & Choudhary, 2016).

Despite the fact that manually operated, electrical motor driven, and diesel engine powered carrot washer machines have been designed and developed around the world, none of them have been adopted and introduced to Ethiopia. Furthermore, no grader was created to assist farmers in grading carrots according to market criteria. As a result, the carrot is Ethiopia's forgotten crop that has yet to be automated. Because it is washed and sorted by a boring mechanism, its post-harvesting is quite high and its quality is very low.

The main goal of this research is to build and construct a pedal-operated, low-cost, low-energy machine using locally accessible materials in order to keep the design simple and economical. The created machine can wash and grade a small-capacity carrot appropriate for small-scale farmers, improving quality, saving time, and labor, and can be utilized anywhere.

1.1.Statement of the Problem

Carrot washing and grading are typical post-harvest tasks undertaken before sending carrots to the market in Ethiopia's rural areas. This strategy involves activities carried out beside canals and waterways, which are laborious, time-consuming, labor-intensive, expensive, and inefficient (Rashid et al., 2018 & Siddique et al., 2017). As a result, this method of washing is immoral and unhygienic, demotivating farmers to grow carrots and increasing post-harvest losses such as carrot contamination and damage, river water pollution, fertile soil erosion, human health issues, and low washing and grading efficiency, all of which lead to lower market appeal (Abdul et al., 2020; Adegbite et al., 2019).

A few researchers were attempting to design and modify various machines such as manually operated carrot washers, diesel engine operated carrot washers, and electrical motor-operated carrot washers in order to alleviate the aforementioned problems and mechanize this traditional system of carrot washing. Hand-cranking machines that are manually controlled require a lot of work and time to operate, which results in low machine efficiency and a lot of discomforts (Amin & Hossain, 2021; Jatobá et al., 2016). The initial investment for machinery powered by diesel engines and electric power is extremely high, making them unaffordable for farmers or individuals. These machines have become outdated because to a lack of electric power and fuel in the ruler area. As a result, traditional carrot washing and grading is currently the sole alternative available in rular Ethiopia. No one has built a carrot grader that aids farmers in the field until now.

Based on the stated facts, pedal-operated carrot Washer and Grader were designed and constructed which was reduce the time and human power consumption, be affordable by anyone and reduce the drudgery and constraints faced by rural farmers concerning carrot washing and grading.

1.2.Objectives

1.2.1.General Objective

This study was conducted to design and construct a carrot tubers washer and Grader and evaluate its performance to improve the traditional practice of washing and grading practice.

1.2.2.Specific Objectives

- Improving the efficiency of washing and grading

- to choose machine components (Chain Drive, Pulleys, V-Belt, Shafts, and Bearings)
- to design (Pump Cylinder Bore and Length, Piston Assembly, Connecting Rod, Shafts, Drum, and Frame)
- to model and create the carrot washing and grading machine parts using CATIA V5 software.
- to put together the pieces of the carrot washing and grading machine
- to assess the washing and grading capability of the carrot washing and grading machine.

1.3.Significance of the Study

This equipment primarily benefits carrot farmers by lowering washing and grading time, labor costs, and improving quality as well as market appeal. Farmers aren't the only ones that benefit from this equipment; wholesalers and retailers are as well.

1.4.Scope of the Study

According to the objectives stated, the design and development of pedal operated carrot washing and grading machine includes all the processes needed to design and develop an efficient washer and grader. It is applicable for farmers in rural areas who could not get tap water for washing. The machine uses water from the river by the developed a reciprocating pump. It was designed for carrot washing that the farmers perform after harvesting but not to include the washing of carrots for consumption or it was not intended to clean and grade the carrots sufficiently for human consumption.

CHAPTER TWO

2. LITERATURE REVIEW

2.1.Origin, Taxonomy, Morphology, and Ecology of Carrot

Carrots were first thought to have originated in Asia before 50 centuries, and later in the 12th century (Stolarczyk & Janick, 2011). Although the exact date of carrot introduction to Africa and Ethiopia is unknown, this crop has been known in Ethiopia since the early 1960s in the research system at Alemaya College of Agriculture (now Haramaya University) by importing eight types of seeds from Kenya (Biratu, 2020). Carrots are scientifically known as Daucus carota subsp. sativusis. It belongs to the Apiaceae family and is a popular root vegetable. The leaf, stem, and root components of the carrot make up the majority of the plant.

2.2.Carrot Growing Regions in Ethiopia

Carrots are grown in Ethiopia in a variety of agro-ecologies ranging from the lowlands to the highlands, primarily in the central highlands of Oromia, Amhara, Addis Ababa, Tigray, and the SNNPR (CSA, 2020).

Currently, the country produces 18225.4t of carrots on 4073.63 ha of land (CSA, 2020). Despite the fact that the production trend is not continuous from year to year, as farmers and other communities become more aware of carrot production and nutritional advantages, carrot output increases from year to year (Tabor & Mohammed, 2012). The total area covered by carrots (in hector) and production (in quintal) went from 2556.05 to 4073.63 and 101482.29 to 182254.14 in just one year, from 2018/19 to 2019/20. Carrot yields range from 40 to 45 quintal/ha under farmers' management depending on agro ecology in the indicated year, with a percentage change of 59.32 and 79.59, respectively (CSA, 2020).

In other cases, approximately 201,111 Ethiopian farmers grew carrots primarily as a commercial crop for selling in urban marketplaces and to earn foreign money by exporting carrots to neighboring Djibouti and Somalia (Mohammed et al., 2014). Furthermore, a large number of people earn money through brokering, trading (wholesale or retail), transporting carrots, and producing transportation supplies such as bags for packing. Not only that, but the cultivation of this crop can boost GDP values; in 2018, the carrot contributed $3412.175 to GDP (FAO, 2018). According to Tabor and Yesuf (2012), 15 percent of the urban population consumes carrots. It is feasible to conclude from the given facts that the demand for carrots and their production in Ethiopia is not proportional. Because it was created and processed using hand tools in the traditional manner, it led to poor quality and production. According to Tabor & Yesuf, (2012), at all production sites of the country, they are topped, washed, and filled in poly-bags and transported to roadsides where merchants can easily access products or to nearby markets.

2.3.Methods of Carrot Washing and Grading

Farmers wash and grade carrots after harvesting them in all places where they are grown before transporting them to the roadside (Ambrose & Annamalai, 2013; Choi et al., 2014; Ganesh et al., 2018). These activities are carried out to avoid contamination caused by soil drying on the crop surface, to reduce unwanted load during transportation caused by soil and impurities, to prevent disease transmission into storage and infecting other carrots, to prepare market recommended size, to attract customers, and to obtain a good market price (Naresh et al., 2018; Oyeleke et al., 2014; Rashid et al., 2018). Farmers and retailers have traditionally washed and graded carrots after harvesting in all producing zones in Ethiopia before transporting them to market (Tabor & Mohammed, 2012).

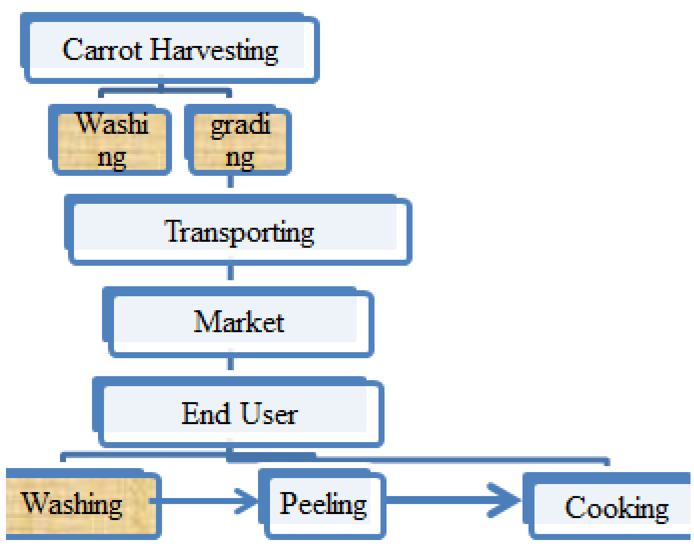


Figure 2.1. Carrot post-harvest processing trends in Ethiopia

In Ethiopia, women and children have been performed these activities at the farm level using traditional methods. Traditionally farmers in the country who cultivate carrots often wash Women and children in Ethiopia have traditionally carried out these tasks on the farm using traditional ways. Farmers that raise carrots in the country have traditionally washed them by hand or feet in rivers or ponds, then graded them by hand into selected and rejected sizes. Farmers in Ethiopia's central highlands, such as Shirkaa, Leemmuu fi Bilbilloo, and Hanqoolo Wabe Districts, transported harvested carrots from the farm to the river using various transportation methods, first soaking them in water to remove soil and impurities, then dividing them into sacks and pushing them perpendicular to the running water. When there isn't enough water in the river, they wash freshly obtained carrots with their feet and hands on the river's edge rather than soaking them in it.

Farmers in east Hararghe's Haramaya neighborhood wash it in the pond. They add water to the pond and then soak the carrot in it. They collect carrots with impurities from the pond and wash them with their hand fit after they have been wet and loosened. Carrots are washed and evaluated at every stage of manufacturing. Depending on location, polybags with a capacity of 100 kilograms (locally known as Tor) or 150 to 170 kg (locally known as Wosla) are used. The carrots are washed and packed in 50 kg polybags by shopkeepers in Tigray (Tabor & Yesuf, 2012). According to a non-structured discussion with farmers and a visit of their working area, traditional washing involves more laborers or employees, more time, and exposes them to various health concerns as a result of the extended duration of work.

Type 2 diabetes, cancer, back pain, neuropathy, stomach disease, and kidney stones are all linked to overwork (Basiska-Zych & Springer, 2017; Amin & Hossain, 2021). According to information obtained from the work area, washing 100 kg of carrots takes roughly one hour, including soaking in water. When two individuals conduct the operations with significant rainfall and sufficient water in the river from June to the end of August, it takes an extra 20 minutes for grading. Another issue with carrot washing is the amount of water required in the process. Because the spring season in Ethiopia is the dry season, washing carrots are extremely tough owing to a lack of water in the river.

During the dry season, they are compelled to wash using dirty water from nearby ponds or rivers. These cleansed items are hazardous to human health, causing a variety of ailments (Amin & Hossain, 2021). Even when the river had ample water, most Ethiopian rural regions used river water for drinking, agriculture, and other purposes. As a result, washing carrots in the river exposed people to a variety of ailments caused by water contamination. The principal drawbacks of this procedure include low washing efficiency, erosion of fertile soil during washing with flowing water, a high percentage of contamination, and a significant amount of damage.

Figure 2.2 Traditional carrot washing process in the river, as seen in the Arsi zone

Figure 2.3. Traditional Carrot Grading Technique after Washing

To tackle the abovementioned disadvantages in the traditional washing process, Ambrose & Annamalai, (2013) construct a manually operated root carrot washer, as shown in figure 2.4. The revolving drum of this machine is made of steel metal and revolves at 10 to 12 rpm by hand. It was able to wash 10kg of carrots in 5 minutes or 100kg of carrots in one hour with around 100L of water. Stopping washing processes allows materials to be loaded and unloaded. It takes at least two individuals to operate the equipment, with one person operating the machine and the other loading and unloading the carrot. The average bruise index utilizing 3.5mm plastic matting of the carrot was 1.25, with washing and cleaning efficiency of 97, 91, and 90 percent for carrot, 96, and 90 percent for radish, respectively.

The water source for washing was gravitationally dropped from the plastic barrel to the drum. Filling the barrel with water in a rural region by transporting it from the river, on the other hand, is a highly inconvenient task. They discovered a high bruising index during the performance test. As a result, the time and labor-intensive problem of excessive crop breakage remain unsolved. Tuber grading in order to make them marketable was not considered.

Figure 2.4. Manually operated carrot washing machine

1- Freshly harvested carrot; 2- loading; 3- washing; 4-unloading; 5- washed carrot.

Jatobá et al. (2016) created and produced a manually operated carrot washer, the spinning drum of which was made of wood, as shown in figure 2.5. To achieve a minimum bruising index, it was manually rotated between 12 and 13 rpm. It was able to wash 0.61 kilograms in 69 seconds, or less than 32 kilograms of unwashed carrots in one hour. The amount of water utilized in each batch of the sample, however, was not specified. Material load and unloading by stopping washing activities, similar to the machine designed by Ambrose & Annamalai, (2013). With the indicated sample, the washing efficiency of this machine was 94.3 percent. The only water available for washing was tap water, which was only appropriate for use around the house.

Figure 2.5. A carrot washing machine that is controlled manually.

Manually operated hand-cranking carrot washing machines, in general, require a significant amount of work and time to operate, resulting in low machine efficiency, high degrees of pain, and a considerable volume of water (Dereje, 2019; Jatobá et al., 2016). At the farmer level, there has never been a manually operated hand-cranking carrot grading machine.

Different researchers (Anjali & Ikhar, 2018a; Batara et al., 2014; Ganesh et al., 2018; Ghuman et al., 2014; Hussain & Ali, 2020; Minh et al., 2020; Naresh et al., 2018; Oyeleke et al., 2014;

Rashid et al., 2018) devised and manufactured carrot washing machines with greater efficiency that were operated using an electric motor, diesel engine, and tractor PTO. They are, however, energy-intensive and costly, requiring a significant initial capital investment that is not suitable for small-scale farmers and difficult to install in remote locations (Seweh et al. 2016).

Neither manual hand cranking nor other sophisticated machines consider carrot grading to alleviate the drudgery faced by farmers after harvesting before transporting to market, i.e. no grader designed to assist farmers in grading carrots according to market requirements anywhere in the world except very complex machines in the food industry.

Nothing of the above-mentioned machines, whether manually operated, electrically powered, tractor PTO shaft powered, or diesel engine powered, was designed, produced, adopted, or introduced in Ethiopia. As a result, the carrot is Ethiopia's forgotten crop that has yet to be automated. As a result, there is a need for a machine that can be used for carrot washing and grading that is affordable to country farmers, simple to operate, time-saving, has a pump that draws water from the river for washing, and can be installed in rural areas.

2.4. Power Source Selection

Agricultural machinery can be powered by an engine, electric power, human power (manual), and so on, depending on the circumstances. Agricultural mechanization in Ethiopia is on the verge of transitioning to a semi-mechanized farming paradigm (Koroso, 2016). This is related to a fear of job loss and the belief that the costs of living are prohibitively high (Amare & Endalew, 2016). As a result, choosing a power source that is compatible with the phycology and economics of farmers is essential. Electric motor machines are difficult to use in remote areas due to a lack of electric power distribution. Engine-driven devices are energy-intensive and costly, necessitating a large initial capital investment that is unsuitable for small-scale farmers (Seweh et al., 2016). As a result, installing both electric motor and engine devices in rural areas is challenging. In Ethiopia, one of the choices for rural areas is manually driven hand-cranked and pedal-powered equipment.

When compared to the pedal operation method, which needs to be replaced with other acceptable methods, the manual approach requires a lot of human energy to crank the machine. Anyone can generate roughly 2-4 times the power of a hand crank using leg pedaling (Mushiri et al., 2017 and Jadoun & Choudhary, 2016). Pedal power is the energy generated by humans via the use of a foot pedal and a chain with a sprocket to perform tasks like washing and grading fruits and vegetables. It is the ideal alternative for remote locations if there is a power outage or a shortage of electricity.

A person can generate 50-150 watts using leg pedaling based on different conditions such as Maturity, healthy, nutrition, age, sex, human pedaling ability, a physicality that may be 2-4 times hand-generated power(Mushiri et al., 2017; Jadoun & Choudhary, 2014; Krishnamurthy et al., 2017). Additionally, any person can exert Pedal forces of up to three times his/her body weight with 50-100 pedal rpm under different conditions and loads (Soden & Adeyefa, 2018; Bhatawadekar et al., 2015). Therefore, using pedal rather than arm has reasonable advantages that stated above and others.

2.5.Research Gap

The carrot washing and grading activities have not achieved enough attention in Ethiopia, however, various Engineers and Researchers worldwide have looked into ways of improving the traditional washing carrot after harvesting before transporting it to the roadside by designing manually or automatically operated carrot washers with some constraints.

As stated in the research gab under each title, all created devices require tap water to wash the carrot tuber, which is inconvenient in rural locations. The majority of them used diesel engines or electric power, which were not available or affordable to farmers, while pedal-operated carrot washers and graders use their pump to pull water from any water tank or river and use human power, which is readily available in rural regions.

Ambrose & Annamalai (2013) designed a manually controlled carrot washer that has a maximum washing rate of 97.91 percent and a drum that rotates by hand, putting the operator in danger. The water for washing comes from a waterfall that is fed by gravity. As a result, it must elevate the water over the machine and fill it with water, which is a time-consuming process. The highest washing efficiency of the pedal-operated carrot washer was 98.7% with the operator in a comfortable position and with minimal effort, and water was easily sucked and delivered to the drum as the user rotated the pedal.

Jatobá et al. (2016) constructed a manually operated carrot washing machine that can wash less than 32kg of unwashed carrot samples in one hour. With the indicated sample, the washing efficiency of this machine was 94.3 percent. Tap water was used for washing, and it was only acceptable for use in metropolitan areas. The pedal-operated carrot washer and grader can wash 242.17 kg of carrots each hour.

Finally, the pedal-operated carrot washing and grading machine is the only machine that grades the carrot tubers at the farmer level. There is no machine developed on the grading at the farmer level. Additionally, it is the first machine in Ethiopia that washes and grades carrot tubers.

CHAPTER THREE

3. MATERIALS AND METHODS

3.1.Description of the Research Area

In the Oromiya Region's Arsi Zone Sirka District, performance evaluations were done in the farmers' field. This location is regarded as the country's primary carrot-growing region (Tabor & Mohammed, 2012). The district is bordered on the east by Arsii Robe, on the north by "Xiichoo," on the north and northwest by "Digaluu fi Xiijoo," on the west and southwest by "Lemmuu fi Bilbilloo," on the south by "Hanqoloo Waabee," and on the southeast by the Waabee Shebelle River, which separates it from the East Baalee zone. The administrative hub of the wereda is Gobessa town with a latitude and longitude of 07°37′N 39°30′E and an elevation of 2353 meters above sea level. It is 89.5 kilometers from Asellaa, Arsii's capital, and 253.1 kilometers from Addis Ababa (Finfinnee).

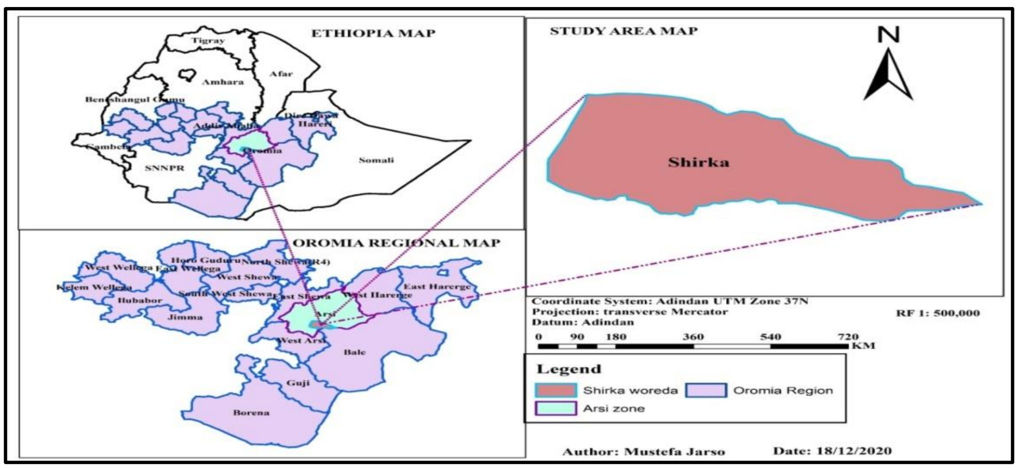


Figure 3.1. Map of the research area chosen for machine performance evaluation.

3.2.Assumptions and Design Considerations

Assumption Made

Based on literature reviews, the following assumptions were made during the design of the carrot washing and grading machine.

- Because most individuals pedal efficiently and continuously for an hour or longer at a pace of 50 to 70 revolutions per minute (rpm), 60 rpm is utilized as the pedal speed for simplicity's sake.

- In order to make machines accessible to people of all ages, 50 watts of electricity was chosen as the input power.
- Because a 7 to 20 kg carrot sample should be washed in 5 to 7 minutes with 11 to 15L of water, the average discharge rate for washing was assumed to be 0.036 l/s.
- The diameter of the pulley was calculated using the velocity ratio.
- Drum speeds of 10-28 rpm were recommended for washing carrot tuber samples weighing 10kg-15kg, hence 28 rpm and 10 kg were taken into account when designing this machine.

Design Considerations

Before beginning the design and construction of the Carrot washer and grader, the following issues were considered for proper material selection, high efficiency, low cost, and post-harvest losses, and to improve product quality.

- Carrot tuber physical qualities and damage
- Waterproofing ability, durability, and corrosion characteristics of construction materials
- Construction material availability and cost
- the design's flexibility, mobility, adaptability, stability, ease of construction, maintenance, loading and unloading, and operability by village craftsmen

3.3.Material Selection

Samples of dominant and popular carrot cultivars produced in Ethiopia's "Arsii" Zone "Sirkaa", "Hanqooloo Waabee", and "Leemmuu fi Bilbilloo" Districts were gathered to evaluate tuber physical properties. Farmers in the Sirka wereda were contacted to obtain nantey Variety for performance evaluation. The construction Materials selection is based on the material's function and related to the carrot's strength in order to reduce and manage the breakage of its pieces during washing and grading, as shown in Table 3.1. Table 3.2 lists the machines, tools, and equipment employed in the creation of washing and grading machines, along with their functions.

3.4. Carrot Tuber Physical Properties

The physical features of carrot tubers were measured in order to develop and choose appropriate pieces, such as a drum for easy loading and unloading, and so on. 100 freshly harvested carrot samples were chosen at random from local types produced around the Arsi Zone to assess various physical attributes. And, using a process described by, the major diameter (a), intermediate diameter (b), and minor diameter (c) were measured using a vernier caliper with an accuracy of 0.05 mm and a measuring range of 0.00 –150.00 mm (Gautam et al., 2016).

Table 3.3.1. Materials used for Construction of Carrot Washing and Grading Machine

Parts	Material Used	Specification	Quantity
Washing and Grading Drum	Waterproof wood	3000×200×2mm	1
Water Pump Cylinder	Aluminum	-	
Connecting Rod	Flat iron	400 ×40 ×2mm	1
Piston	Aluminum	-	
Piston Rod	Aluminum	-	
Crank	Flat Iron	200×20×2mm	1
Pedal	Cast Iron	320mm dia	1
Shafts	Round mild steel	20mm dia × 6000mm	1
Handle for Drum rotation	Galvanized pipe	18mm Dia ×1000mm length	1
Bicycle sprocket	Cast Iron	85mm dia with 20 teeth	1
Water Delivery and Suction	Plastic	1inch dia×3000mm	1
Chain	Cast iron	06B with 1300 mm length	1
Standard finished items	Roller Chain, Sprockets, Bolts and Nuts, and Washers		
Pulleys	Cast Iron	75mm, 140mm, 75mm, and 150mm dia	4
Belts	Rubber	28 and 29 code numbers	2

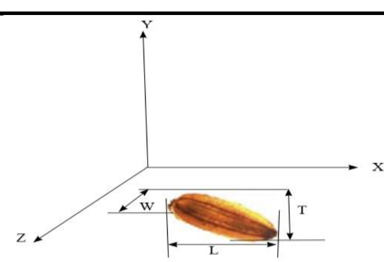


Figure 3.2. Measured dimension of carrot tuber

Table 3.2. Machines, Tools, and equipment used for machine development

Machine	Purpose
Welding machine	To weld together fixed parts
Grander machine	For Cutting metals and finishing the body of components
Lathe machine	For boring, threading, finishing, etc.
Drilling machine	For Making holes
Tools and Equipment	Purpose
Steel tape	Measurement of linear distance
Vernier calipers	Measurement of the outer and inner diameter
Hammer	Used to strike an object
Screwdriver	Tighten screw
Spanner	Tighten nut and bolt
Snip	Cutting sheet metal
File	Smooth rough edge

3.4.1.Geometric Mean Diameter and Sphericity

Equations (1) and (2) were used to calculate GMD and sphericity from the product's length (L), width (W), and thickness (T) (Jahanbakhshi, 2018).

$$D_g = \sqrt[3]{a.b.c} \tag{1}$$

$$\emptyset = \frac{D_g}{a} \tag{2}$$

Where, D_g=Geometric Mean Diameter and (mm) and $\emptyset$ =Sphericity (%)

3.4.2.Bulk density

The volume of the washing and grading drum was calculated using the bulk density of carrot tubers. After harvesting three fields, it was determined from a box with dimensions of 200mm, 152mm, and 71mm in length, width, and height, with a sample mass of 1375g, and the average bulk density was calculated using equation (3). (Bashar et al., 2014; and Soyoye et al., 2018).

$$Bulk\ density\ \left(\frac{g}{cm3}\right) = \frac{weight\ of\ Sample\ (g)}{Volume\ occupied\ (cm^3)} \tag{3}$$

3.4.3.Coefficient of Friction

During the washing and grading of carrot tubers, the coefficient of static friction plays an essential role in transporting (unloading). i.e. the carrots are usually moving or sliding against the drum. It's the ratio of the frictional force that prevents carrot surfaces from moving when they come into touch with a drum's wood board (Jahanbakhsh et al., 2018). Both static and dynamic Coefficients of Friction aid in the design of the washing and grading unit's inclination angle. The inclined plane approach was used to calculate the static coefficient using wood surfaces. With an angle of, the timber was raised from the bench table. The static friction coefficient can then be calculated using equation (4) (Jahanbakhsh et al., 2018) by solving the free body diagram of static conditions.

$$\mu_s = \frac{sin\theta}{cos\theta} = tan\theta \quad (4)$$

Where, μ_s – static coefficient of friction

Similarly, the kinetic friction coefficient determined by equation (5) is always smaller than the static friction coefficient when kinetic circumstances and uniform acceleration equations are taken into account (Bayram, 2018, 2019; Boukouvalas et al., 2010; and Jahanbakhsh et al., 2018).

$$\mu_k = \frac{(g\ x\ sin\theta) - a_x}{gxcos\vartheta},\ a_x = {2d}/{\Delta t^2} \quad (5)$$

Where, μ_k= kinetic or dynamic coefficient of friction, d=sliding distance of the timber, Δt^2=time for the object to slide along distanced, and a_x= uniform acceleration of the object

3.4.4. Angle of Repose

The angle of repose is a popular parameter for determining antiparticle force. A bottomless box was used to determine it. After placing the box on a horizontal surface and filling it with a sample, it was progressively elevated. Allowing the sample to gather on the surface and form a conical mound. The angle of repose (ϕ) was calculated using equation (6) by measuring the diameter of the conical shape formed by the carrots sample (D) and two opposing sides (L1 and L2) (Teferra, 2019).

$$\phi = \cos^{-}1\ \left[\frac{D}{l1 + l2}\right] \quad (6)$$

3.5.Carrot Washing and Grading Machine Description

3.5.1.Carrot Washing and Grading Machine Components

Operator seat, pedal, chain, (transmission, Idler, and drum) shafts, sprocket, pulleys and belts, crank, water pump, suction, and delivery pipes, nozzle, drum (cylindrical washing and grading unit), handle, bearings, and mounting frame were among the components of the carrot washing and grading machine, as shown in figure 3.3.

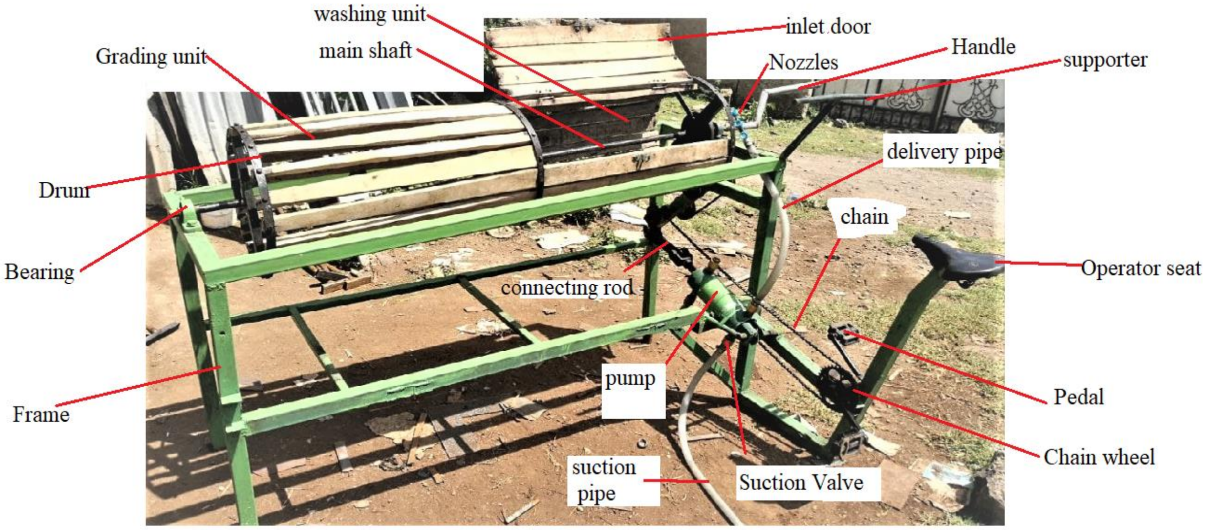


Figure 3.3.Carrot Washer and Grader Machine

The washing unit is of the cylindrical form, with eighteen (18) rectangular wood boards that were calculated. According to Ambrose & Annamalai (2013), these timbers should be spaced 2mm apart to allow just water and pollutants to flow through, and they should be 70cm long to allow for effective washing time (Jatobá et al., 2016).

The grading unit is a trapezoidal timber expansion of the washing unit. GMD and the sphericity of carrots were used to create these trapezoidal timbers. This device was limited to an 80cm length, which was determined by trial and error. Three size-based gradings are performed by the grading unit. The undersized and broken carrots with top diameters less than 20cm are assessed in the first trapezoidal slots. These are little, shattered pieces that are used to feed animals. The quality of carrots on the marketplaces with 21cm to 30 cm graded that are utilized for local markets is reduced by medium-sized and some broken carrots in the next slot. The outflow port for large-sized carrots is used for the final grading procedures. These are utilized for exports and marketplaces that are far distant from the production location.

The washing and grading drum is tilted to allow the washed and graded carrots to flow freely to the outflow port without interrupting the process. By opening the door hinged on the washing portions, the unwashed and ungraded carrots are introduced into the machine.

A pedal on the machine turns a chain connected to the sprocket and transmission shaft by 60^0 degrees from horizontal. Transmission and Idler pulleys vertically aligned the power transmission shaft with the shaft of the idler pulley, which was installed on the vertical frame. To reciprocate connecting and piston rod for pump operation, the crank is welded to the transmission shaft on the other side.

3.5.2. Operational Principle

The procedure begins with the pedal being rotated, which rotates the transmission shaft through the sprocket and chain. The motion of the transmission shaft is transmitted simultaneously to the idler shaft and the cranks. The Idler shaft rotates the washing and grading unit through the belt and pulleys, reducing the speed of the transmission shaft with 27:7 ratios. The inlet door fed carrot roots into the inclined and revolving cylinder, which proceeded along the length of the cylinder. The trapezoidal slot pushed the carrot roots at the bottom of the cylinder upwards, and then they were lifted along with it and slid down. They gradually travel toward the cylinder's opposite end from the inlet port and, due to their diameters, are forced to pass through the slots as the cylinder rotates.

The transmission shaft motion is transferred to the connecting rod by the crank. The connecting rod transfers the rotating motion of the crank to the piston rod as a reciprocating motion. The piston rod is attached to the piston by rubber piston rings. These pieces reciprocate through the cylinder and create a vacuum, sucking water from the river or any other water reservoir and delivering it to the washing and grading unit via a delivery pipe and nozzle. In addition to the pedal, a handle was installed that could do all of the above activities if the pedal or chain were to fail.

3.5.3.Selection of Chain Drive

The 150mm diameter bicycle chain wheel with 36 teeth, the 85mm diameter sprocket with teeth number 20, and the 320mm diameter pedal were chosen and purchased at random from the market. Because the power delivered by a chain from wheel shaft to sprocket shaft is equal to input power, the rpm of the smaller sprocket (108rpm) was calculated using equation (7) from the relationship between the number of teeth on the chain wheel and sprocket (Adegbite et al., 2019)

$$N_{sprocket} = \frac{N_{wheel}T_{wheel}}{T_{sprocket}} \tag{7}$$

Where, $N_{sprocket}$=revolution per minute of the sprocket in rpm, N_{wheel}=revolution per minute of chain wheel in rpm, T_{wheel}=teeth numbers on the chain wheel, and $T_{sprocket}$=teeth numbers on the sprocket

Design power and service parameters determined the optimum number of chains as well as criteria such as pitch, roller diameter, minimum breadth of the roller, and breaking load. To estimate rate power, design power was determined from input power (Pi=50W) as indicated by equation (8), and service factors (Ks=1.5) are the product of several variables, including a factor for variable load with heavy shock (K1), a factor for lubrication (K2), and a factor for rating (K3). The values of these variables are determined by the machine's features (Khurmi & Gupta, 2005).

$$P_D = P_i \times k_s \tag{8}$$

$k_1 = 1.5$, $k_2 = 1.0$, and $k_3 = 1.0$, $K_s = k_1.k_2.k_3 = 1.5 \times 1 \times 1 = 1.5$

The rating power 0.25kW and the chain designation 06B were chosen based on the above computed design power (PD=75W) cross ponding with pinion (smaller sprocket) speed of 108 rpm and the appendix table 1.1. Pitch, roller diameter, minimum width of the roller, and breaking load were computed based on rated power and the number of chains indicated above (06B) and from the appendix table 1.2.

Based on the values stated in appendix table 1.2 and equations (9), (10), and (11) the chain's center distance (x), number of chain links (k), and length (l) were calculated (Adegbite et al., 2019).

$$x = 30\ to\ 50p \tag{9}$$

i.e., the minimum center distance should be 30 to 50 times the pitch for best outcomes. In appendix table 2, the pitch diameter is 9.525mm, which is 30 times the pitch value used in this design.

Therefore, $x = 30p = 285.75mm$

The center distance is shortened by 2 to 5mm to accommodate the early sag in the chain. Let's pretend it was shrunk by 5mm. (x=280.75mm) is the correct center distance.

$$k = \frac{T_2+T_1}{2} + \frac{2x}{P} + \left[\frac{T_2-T_1}{2\pi}\right]^2 \frac{p}{x} \tag{10}$$

$$k = \frac{36+20}{2} + \frac{2\times 280.19mm}{9.525mm} + \left[\frac{36-20}{2\pi}\right]^2 \frac{9.525mm}{280.19mm} = 87.05$$

Where, T_1=Number of teeth on the smaller sprocket, T_2 =Number of teeth on the larger sprocket, x=Center distance between the two sprockets, k =Number of chain links, p=Pitch of the chain (mm)

$$L = kp \tag{11}$$

$$L = 87.05 \times 9.525mM = 829.16m = 0.9m$$

Where, L Length of the chain (m)

As a result of the rating power cross ponding with pinion speed of 108 rpm and appendix table 1.2, minimum center distance, chain link, and computed chain length, a chain with a length of 90cm and a standardized chain designation of 06B was chosen.

3.5.4.Selection of Pulley

The drum speed was considered to be 28 rpm for the design of this machine, while the smaller sprocket's rpm was determined to be 108 rpm, as stated in section 3.2. As a result, with a speed reduction ratio of 27:7, power from the transmission shaft pulley is transmitted to the drum shaft pulley. The minimum pulley size for "A" type V-belt 75mm was chosen for driving pulleys on the crankshaft and idler shaft to achieve this ratio. The "A" type V-belt was chosen for this application because it delivers power between two pulleys that are close together (Jis et al., 2014).By considering the output rpm, the driven idler pulley was also arbitrarily set at 140mm. According to Equation (12), the diameter of the drum shaft pulley is projected to be 155mm based on the speed of the crank (transmission) shaft pulley, the speed of the drum shaft pulley, the diameter of the driver pulleys, and the diameter of the idler driven pulley (Khurmi & Gupta, 2005; Macmillan, 2010).

$$\frac{N_4}{N_1} = \frac{D_1 \times D_3}{D_2 \times D_4} \tag{12}$$

Where, N_1=Angular Speed of the first driver in r.p.m. on the first belt, N_4 =Angular Speed of the last in r.p.m. on the second belt, D_1=Diameter of the first driver, D_2=Diameter of the first follower, D_3=Diameter of the last driver, and D_4=Diameter of the last follower

3.5.5.V-Belt selection

The nominal pitch lengths (L_1=708.51mm) and (L_2=735.93mm) and center distances (x_1=185.5mm) and (x_2=187.5) from transmission shaft pulley to idler shaft pulley and from idler shaft pulley to drum shaft pulley, determined by equations (13) and (14) respectively, were used to select the belt (Charles & Etiese, 2020).

$$x = \frac{D_1 + D_2}{2} + D_2 \tag{13}$$

$$L = \frac{\pi}{2}(D_1 + D_2) + 2x + \frac{(D_1 - D_2)^2}{4x} \tag{14}$$

Where, D_1=Diameter of the larger pulley,D_2=Diameter of the smaller pulley, and x=Distance between the centers of two pulleys

The nearest standard pitch lengths for L_1 and L_2 are 710mm and 730mm, respectively, according to Appendix Table 1.4. So, based on the minimum pulley diameters (75mm) recommended for standard V-belts (Appendix Table 1.3), the nearest standard pitch lengths708.51mm) and (735.93mm), and the minimum center distance x_1 andx_2, belt sizes A-28 and A-29 were selected to transmit power from transmission shaft to idler shaft and from idler shaft to washing and grading cylinder. The pulley's groove angle was set to 40^0, which is the average of the range.

Section 2.7.2 discussed the breadth of the pulley or face of the pulley (B), which was calculated to be 20 mm using equation (15) and appendix table 6. (Khurmi & Gupta, 2005).

$$Face\ width\ (B) = (n - 1)\,e + 2f \tag{15}$$

3.5.6.Cylinder Bore and Length Design for Reciprocating Pumps

In rural places, tap water is scarce, hence a water pump was created for washing needs. According to literature, the pump cylinder discharge rate for washing is considered to be 0.036 l/s in this design, and the water discharge through the pump is given by equations (16). (Kumar & Sumalatha, 2018; Shankar, 2016; Thorat & Kanase-Patil, 2015, and Ashiedu et al., 2016).

$$Q = \frac{AlN}{60} \tag{16}$$

$$since\ Al = V,$$

$$V = \frac{60Q}{N} \tag{17}$$

$$V = \frac{60Q}{N} = \frac{60 \times 3.6 \times 10^{-5}\,\frac{m^3}{s}}{108rpm} = 2 \times 10^4 mm^3$$

Where, Q=water discharge through the pump (m^3/s), A=Area of the piston (m^2), l=stroke length (m), N=108 rpm (transmission shaft/crank rotation per minute), V=volume of the pump (m^3)

The length of the stroke is usually 1.25 to 2D mm, and because there is clearance on both sides of the cylinder, the length of the cylinder is 15 percent longer than the length of the stroke, as stated by equations (18) and (19). (Khurmi and Gupta, 2005).

$$l = 2\mathrm{D} \tag{18}$$

$$L = 1.5l \tag{19}$$

Where D=Inner Cylinder diameter, l=Stroke of the piston, and L= length of the cylinder

The volume (V) is also the product of the horizontal cross-sectional area (A) and the length of the piston stroke (l) as given by equation (20) (Yang *et al.*, 2011).

$$V = \frac{\pi D^2}{4} \times l \tag{20}$$

Where, V =volume of pump cylinder (m^3), D=Diameter of pump cylinder (m), and l=stroke length (m)

$$V = \frac{\pi D^2}{4} \times 2D = \frac{\pi D^3}{2}$$

$$D = \sqrt[3]{\frac{2V}{\pi}} = \sqrt[3]{\frac{2 \times 2 \times 10^4 mm^3}{\pi}} \quad \therefore D = 23.37mm$$

The crank radius is the half of the stroke given by equation

$r = \frac{l}{2} = 11.69mm$ where

r=the radius of the crank (m), l=stroke length (m),

The area of the cylinder is determined from the cylinder's equation which is given by equation (18) (Khurmi and Gupta, 2005).

$$A = \frac{\pi D^2}{4} \tag{21}$$

$$A = = 428.95mm^2 \approx 0.0004m^2$$

Where, D=inner diameter of the cylinder (m), and A=area of the cylinder (m^2)

The theoretical power required to drive the pump (P_{th}) is also given by equations (22) (Kumar & Sumalatha, 2018; Shankar, 2016; Thorat & Kanase-Patil, 2015, Ashiedu et al., 2016).

$$P_{th} = pQ \tag{22}$$

$$p = 0.001\ MPa$$

Where, P_{th}= theoretical power required to drive the pump and p=net hydraulic pressure (MPa)

The value of _t may be taken as 0.8 times the yield point stress(y) as represented by Eqn (24) and the cylinder wall thickness (t) determined according to Birnie's equation for the wall thickness of a cylinder as given by (23) and the value of _t may be taken as 0.8 times the yield point stress(y) as represented by Eqn (24) (Khurmi and Gupta, 2005).

$$t = r_i\sqrt{\frac{\sigma_t+(1-\mu)p}{\sigma_t-(1+\mu)p}} - 1 \tag{23}$$

$$\sigma_t = 0.8\sigma_y \tag{24}$$

Where t= cylinder wall thickness, r_i=inner radius of the cylinder, p=Intensity of internal pressure, σ_t = Tangential stress ($\sigma_t = 50MPa$ (Alloy *et al*., 2017)), and μ = Poisson's ratio (μ = 0.31 For aluminum (Khurmi & Gupta, 2005)

$$t = 11.69mm\sqrt{\frac{50MPa+(1-0.31)0.001\ MPa}{50MPa-(1+0.31)0.001\ MPa}} - 1 = 10.69mm$$

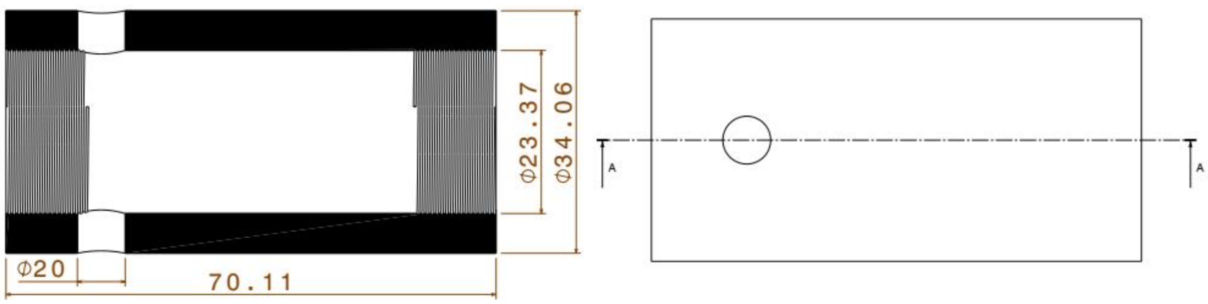


Figure 3.4.Pump Cylinder Cross-sectional View

(all dimensions are in mm)

3.5.7.Piston Assembly of a Reciprocating Pump Design

The thickness of the piston head (tH), the radial thickness (t1), the axial thickness (t2) of the piston rings or cups, the width of the top land (b1), and the width of other ring lands are all included in the piston assembly (b2). According to Grashoff's formula, they were designed as equations (25), (26), (27), (28), and (29) accordingly (Gopal et al., 2016).

$$t_H = \sqrt{\frac{3pD_0{}^2}{16\sigma_t}} \tag{25}$$

$$D_o = D_i + t = 34.06mm$$

$$t_H = \sqrt{\frac{3\times 0.001\ N/mm^2\times 34.06^2 mm^2}{16\times 50\ N/mm^2}} = 0.004mm \approx 0$$

Where, p= Maximum water pressure in N/mm^2, D_0=Cylinder bore or outside diameter of the piston in mm, σ_t= Permissible bending (tensile) stress for the Aluminum in MPa or N/mm^2, It may be taken as 50 to 90 MPa for nickel cast iron and aluminum alloy

$$t_1 = D_0\sqrt{\frac{3p_w}{\sigma_t}} \tag{26}$$

$$t_1 = 34.06mm\sqrt{\frac{3\times0.042N/mm^2}{90N/mm^2}} = 1.27mm$$

Where, D_0 = Cylinder bore in mm, p_w = Pressure of water on the cylinder wall in N/mm2. Its value is limited from 0.025 N/mm^2 to 0.042 N/mm^2, σ_t = Allowable bending (tensile) stress in MPa Its value may be taken from 85 MPa to 110 MPa for cast iron rings.

$$t_2 = \frac{D_0}{10n_R} \tag{27}$$

$$t_2 = \frac{34.06mm}{10\times1} = 3.41mm$$

Where n_R = Number of rings

$$b_1 = 1.2t_H \tag{28}$$

$$b_2 = t_2 \tag{29}$$

Where, p= Maximum water pressure in N/mm^2, D_0=Cylinder bore or outside diameter of the piston in mm, σ_t= Permissible bending (tensile) stress for the Aluminum in MPa or N/mm^2, It may be taken as 50 to 90 MPa for nickel cast iron and aluminum alloy and n_R= Number of rings.

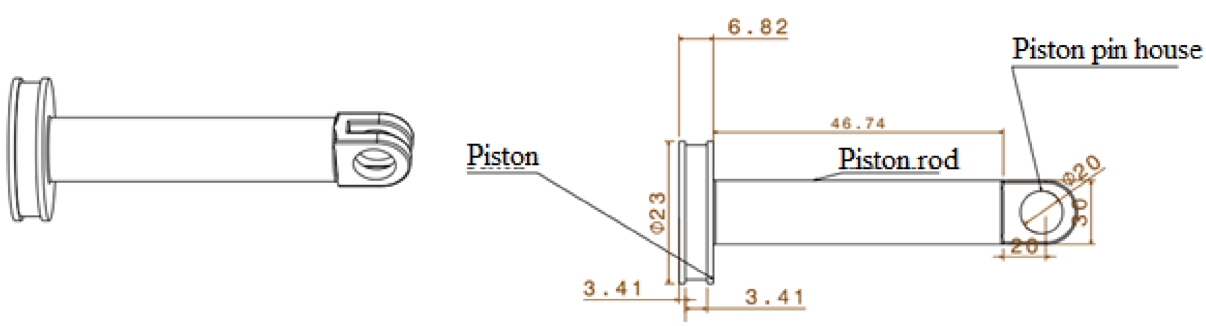
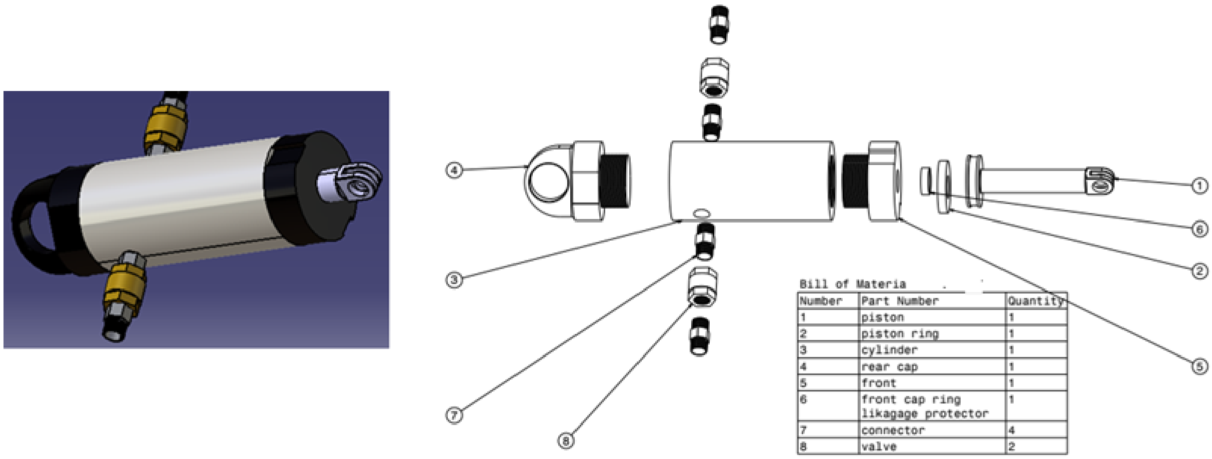


Figure 3.5. The piston assembly of the reciprocating pump

(all dimensions are in mm)



Number	Part Number	Quantity
1	piston	1
2	piston ring	1
3	cylinder	1
4	rear cap	1
5	front	1
6	front cap ring likagage protector	1
7	connector	4
8	valve	2

Figure 3.6.Pump and pump components

3.5.8.Connecting Rod Design

According to Kumar & Sumalatha,(2018), the maximum force ($F_L = 428.95$N) on the piston due to pressure (p) is calculated as equation (30)

$$F_L = \mathrm{Ap} \tag{30}$$

Where, p = Maximum pressure of the fluid, D = Diameter of the piston, and A = Cross-section area of the piston

The connecting rod is built as a strut or column and is subjected to an axially compressive force equal to the maximum fluid force (F L) (Sebastian Antony et al., 2016). The buckling loads about the X-axis (W cr about X-axis) and the Y-axis (W cr about Y-axis) are computed using appendix table 1.6 and Rankine's formula (equations (31) and (32). (Nitturkar et al., 2020; Bharath Kumar et al., 2015; Mukkawar & Kadam, 2015).

$$W_{cr}\ about\ X - axis = \frac{\sigma_c A}{1+a\left(\frac{l}{k_{xx}}\right)^2},\ \text{for both end hinge } L = l \tag{31}$$

$$W_{cr} about\ Y - axis = \frac{\sigma_c A}{1+a\left(\frac{l}{2k_{yy}}\right)^2},\ \text{for both end fixed } L = \frac{l}{2} \tag{32}$$

Where A = Cross-sectional area of the connecting rod, l = Length of the connecting rod, σ_c = Compressive yield stress, σ_c is 330Mpa For steel, W_{cr} = Crippling or buckling load, I_{xx} and I_{yy}= Moment of inertia of the section about X-axis and Y-axis respectively, k_{xx}, and k_{yy}= Radius of gyration of the section about X-axis and Y-axis respectively, a =constant= $\frac{1}{7500} = 1.33 \times 10^{-4}$, for mild steel.

Because the connecting rod is constructed with the force in the connecting rod (F_C) equal to the maximum force on the piston due to water pressure (F_L), $F_L = F_C = 428.95N$ is the force in the connecting rod. If both ends of the connecting rod are hinged and the connecting rod is built for buckling about the X-axis (i.e. in a plane of motion), the buckling load is specified by equation (33) (Khurmi & Gupta, 2005).

$$W_{cr} = F_C \times F.O.S. \tag{33}$$

Where, $F.O.S.$ =factor of safety. Let's consider F.O.S. as 10, then $W_{cr} = 428.95\text{N} \times 10 = 4289.5N$

To have a connecting rod that is equally strong in buckling about both axes, the buckling loads must be equivalent. If equations (31) and (32) are equal, the connecting rod is four times stronger in buckling about the Y-axis than the X-axis as mentioned by equation (34).

$$I_{xx} = 4\, I_{yy} \tag{34}$$

$$\frac{bh^3}{12} = 4 \times \frac{b^3 h}{12}\ ,\ h^2 = 4b^2 \text{ or } h = 2b$$

The value of b=3mm was determined from equations 34 and 35 then h=6mm

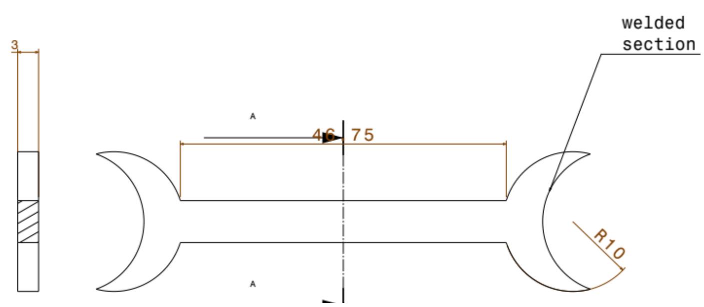


Figure 3.7. Connecting rod section view

(all dimensions are in mm)

3.5.9.Shaft Selection and Drum Design

3.5.9.1. Selection of Transmission Shaft

The transmission shaft is the shaft that gets power from the pedal via the chain and transmits it to the drum shaft via belts for simultaneous washing and grading water pumping. As illustrated in figure 3.8, the smaller sprocket (driven pinion) was equipped with a bearing using an interference fitting technique, while the transmission shaft was fitted with a bearing using a clearance fitting approach. Because it is bent and twisted by chain tension, belt tension, and Crank tangential force, this shaft is subjected to torsion and bending pressures.

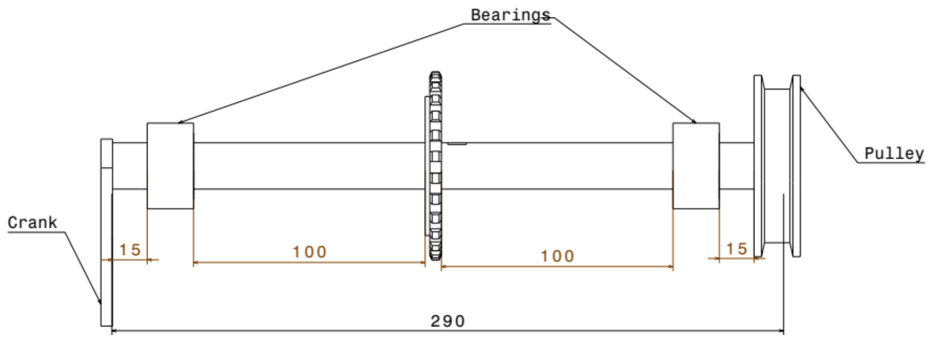


Figure 3.8. Transmission assembly

(all dimensions are in mm)

The torque exerted by the sprocket is equal to that exerted by the transmission shaft as related by equation (35) (Gujar *et al.*, 2013 and Babu et al., 2016).

$$\tau_{Sprocket} = \tau_{Transmission\ shaft} \tag{35}$$

Where, $\tau_{Sprocket}$=torque exerted by sprocket and $\tau_{Transmission\ shaft}$=torque exerted by transmission shaft.

The value of transmission shaft torque ($= 4.42Nm$) was obtained from the power formula developed by equation (36) (Sharma & Mukesh, 2010).

$$\tau_{Sprocket} = \frac{60P}{2\pi N_{sprocket}} \tag{36}$$

Where, P=input power in W, $\tau_{Sprocket}$=torque exerted by sprocket (Nm), and $N_{sprocket}$=angular speed of sprocket (rpm)

The transmission shaft torque and turning moment (crank effort in Nm) for $11.69mm$ crank radius, its belt tension, chain tensions, and crank tangential were calculated as given by equations (37) and (38) (Sharma & Mukesh, 2010).

$$T_{Transmission} = T_{Belt} - T_{crank} \tag{37}$$

$$T_{crank} = F_T r \tag{38}$$

Where, $T_{Transmission}$ =torque exerted by transmission shaft, T_{Beltt}=torque exerted by belt, T_{crank}=torque exerted by the crank, F_T =Crank tangential force, and r=crank radius.

The net force applied on the crankpin or Crank tangential force was determined by equation (39) (Guru et al., 2019).

$$\therefore F_T = F_C \sin(\theta + \phi) \tag{39}$$

Where F_C =Force acting along the piston rod, ϕ=Angle of inclination of the connecting rod with the line of stroke, and θ = Angle of inclination of the crank from top dead center

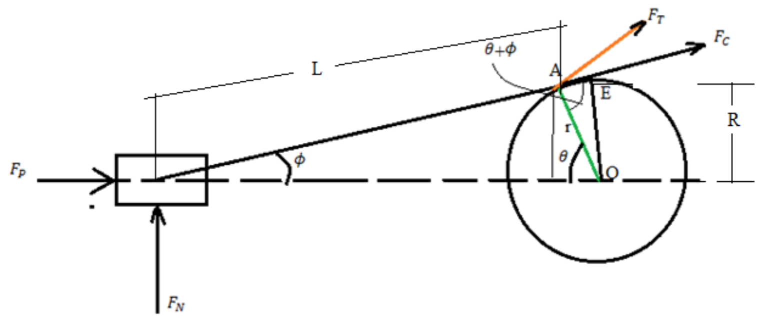


Figure 3.9.Net force applied on the crankpin and piston effort

As previously stated, the maximum value of Fc force along the connecting rod is used to construct the connecting rod. The angle of inclination of the connecting rod with the line of stroke (ϕ) obtained using sine law as given by equation (40) for the maximum FC, =900, r=OE, and OE=R as seen from the geometry of the picture above (Nicholas & Adamson, 2006).

$$\frac{\sin\phi}{R} = \frac{sin\theta}{L} \tag{40}$$

$$\phi = \sin^{-1}\left(\frac{R\times sin\theta}{L}\right)$$

$$\phi = \sin^{-1}\left(\frac{11.69mm\times sin90^0}{23.37mm}\right) = 30^0$$

Where L=length of connecting rod and R=radius of crank when $\theta = 90^0$

$\therefore F_T = 428.95N \times \sin(90^0 + 30^0) = 397.46N$

$T_{crank} = 4289.5NN \times 0.14mm = 26.28\text{Nm}$

$T_{Transmission} = T_{Belt} - T_{crank},\ T_{Belt} = 4.42Nm + 26.28\text{Nm} = 30.70Nm$

In another scenario, the torque exerted on the driving pulley was determined as equation (41) because the effective turning (driving) force at the circumference of the driven pulley or follower is the difference between the two tensions (Field & Solie, 2007; Khurmi & Gupta, 2005; Shigley & Mischke, 1996; Srivastava et al., 2006).

$$T_{Belt} = (T_1 - T_2)r_1 \tag{41}$$

Where, r_1 The radius of the driving pulley (radius of transmission shaft pulley) (m)

T_{Belt} The torque exerted on the driving pulley (Nm)

T_1 Tension in the tight side of the belt

T_2 Tension in the slack side of the belt

Equation (42) was used to calculate the relationship between the tight side (T1) and slack side (T2) tensions of a V-belt belt in terms of coefficient of friction (μ) and angle of contact (θ). (Khurmi & Gupta, 2005, Charles & Etiese, 2020; Macmillan, 2010; Sharma & Mukesh, 2010; and Shigley & Mischke, 1996).

$$\frac{T_1}{T_2} = e^{\mu\theta} \tag{42}$$

The angle of contact or lap at the Smaller pulley (θ) was computed using equation (48) for an open belt with both pulleys made of the same material (Khurmi & Gupta, 2005; Shigley & Mischke, 1996).

$$\theta = \left(180^0 - 2 \times \sin^{-1}\left(\frac{D-d}{2C}\right)\right)\frac{\pi}{180} rad \tag{43}$$

Where,θ= the angle of contact or lap at the smaller pulley, D=Diameter of larger pulley (Diameter of Idler Shaft Pulley) (m), d=Diameter of smaller pulley (Diameter of Transmission Shaft Pulley)(m), and C =The center distance between transmission shaft pulley and idler shaft pulley

$$\theta = \left(180^0 - 2 \times \sin^{-1}\left(\frac{0.14m - 0.075m}{2 \times 0.183m}\right)\right)\frac{\pi}{180} rad = 2.78\, rad$$

Because the chosen pulleys were made of cast iron and the belts were made of rubber, the coefficient of friction between cast iron and rubber is 0.30, or 0.30, as shown in appendix table 1.7. Another significant component of the transmission shaft is the chain tension, which was calculated using equation (44) and the Chain velocity (Pitch line velocity of the smaller sprocket), which was measured using equation (45). (Adzimah & Agbovi, 2015).

$$P = T_C V_C \tag{44}$$

Where, T_C =chain tension and V_C =Velocity of chain

$$V_C = \frac{TpN}{60} \tag{45}$$

Where, N=speed of rotation of smaller sprocket in rpm, T = Number of teeth on the sprocket, and p = Pitch of the chain in meters.

$$V_C = \frac{20 \times 0.009525m \times 108\, rpm}{60} = 0.34\frac{m}{s} \quad \therefore T_C = \frac{50W}{0.34\frac{m}{s}} = 145.82N$$

The chain attached to the shaft with 30^0 from vertical line, therefore the chain tension applied to the shaft in vertical and horizontal components as following

$$T_{Cv} = T_C \cos 30 \qquad T_{Ch} = T_C \sin 30$$

$$T_{Cv} = 126.28N \qquad T_{Ch} = 72.91N$$

Where, T_{Cv}=chain tension of vertical component in N and T_{Ch}= chain tension of horizontal component in N.

The mass of the transmission shaft pulley, smaller sprocket, crank, and chain were measured using beam balance at 0.25kg, 0.12kg, 0.02kg, and 0.41kg, respectively. These weights were calculated by multiplying each mass by the acceleration of gravity: 2.45N for the pulley (W p), 1.18N for the sprocket (W s), 0.20N for the crank (W crank), and 1.37N for the chain. At the same time, the vertical component of chain tension caused by sprocket weight and the crank tangential force caused by crank weight push on the shaft vertically downward. As a result, they were treated as a single force in the following way.

$$T_{cvs} = T_{Cv} + W_s = 126.28N + 1.18\text{N} = 127.46\text{N}.$$

$$F_{TW} = F_T + W_{Crank} = 187.71N + 0.2N = 187.91N$$

Where, T_{cvs}=the summation of the vertical component of chain tension and sprocket weight

F_{TW}= the summation of crank tangential force and weight of the crank

The belt tensions and the pulley's weight both act at the same location but in opposite ways in this situation. The tensions are increasing and the pulley weight is decreasing. Because the stresses were more than the pulley's weight, they were reinterpreted as succeeding.

$$T = T_1 + T_2 - W_p = 2053.06N - 2.45N = 2050.61N$$

The shaft was supported by two bearings at locations B and D, as indicated in figure 3.10. The bearing B is 15mm distant from the crank, and the bearing D is 15mm apart from the pulley. However, because the forces and weights are applied from the center of the pulley, sprocket, and crank, the bearing B is 30mm and 113.5mm apart from the crank center and sprocket center, respectively. In the same way, the bearing B is 35mm and 113.5mm distant from the center of the pulley and the center of the sprocket, respectively. There is a response force at each support that balances the shaft's twisting and bending. As a result, the response at each support must be determined before twisting and bending moments can be calculated.

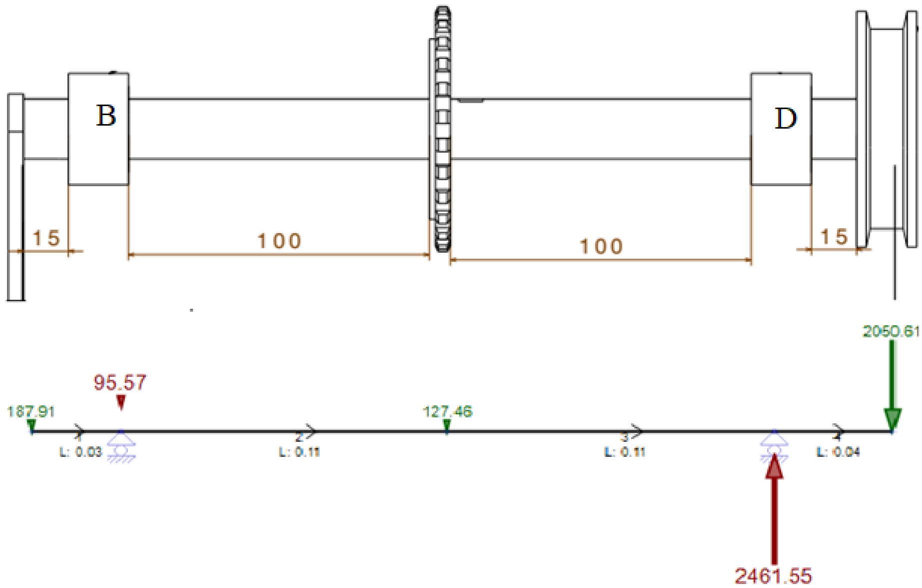


Figure 3.10.Transmission shaft loading diagram on the vertical plane

(All Dimensions in m and all Forces are in N)

To determine the reaction at B point for vertical components, the moment about point D was taken as M_{Dy} according to equation (46) (Jatobá et al., 2016).

$$+\circlearrowleft \sum M_{Dy} = 0 \tag{46}$$

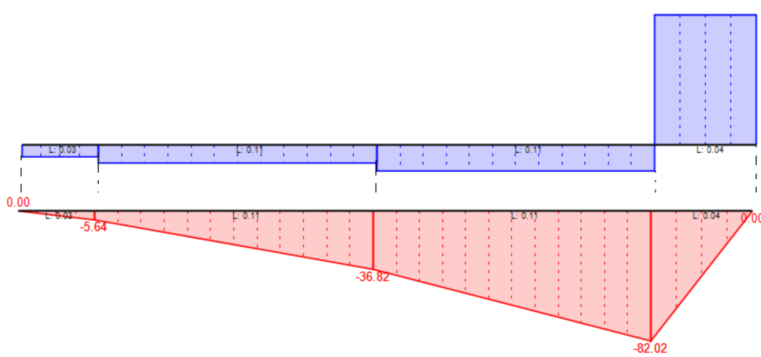


Figure 3.11.Shear force and bending moment diagram on the vertical plane

The moment around point D is taken as MDx in the same scenario to determine the reaction at B point for horizontal components (Jatobá et al., 2016).

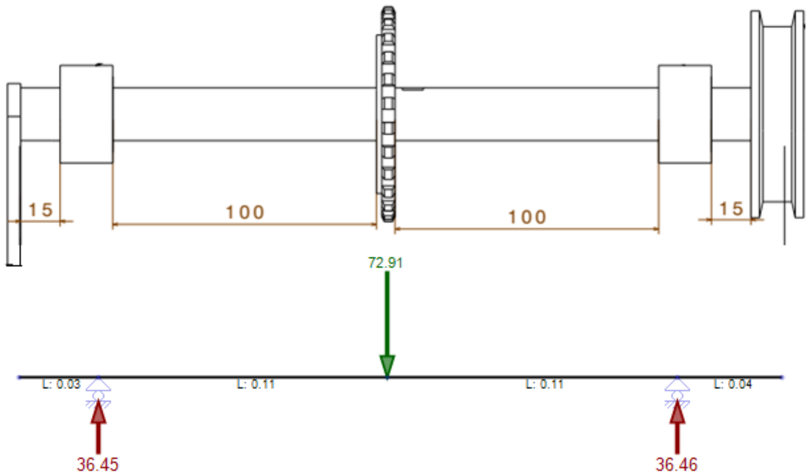


Figure 3.12. Transmission shaft loading diagram on the horizontal plane

(All dimensions are in m and all forces are in N)

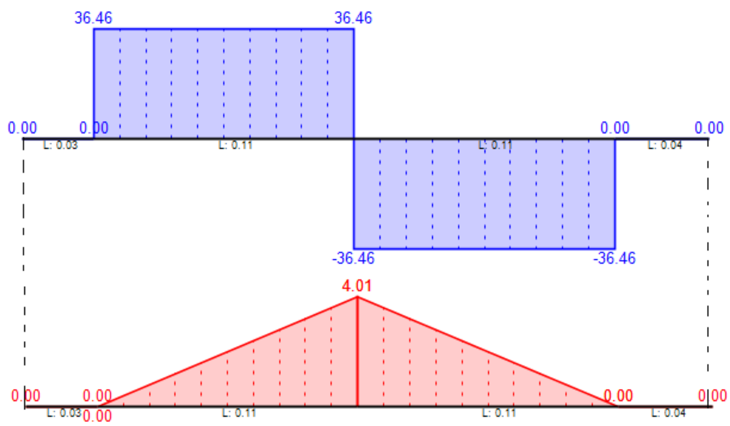


Figure 3.13.Transmission shaft shear force and bending moment diagram on the horizontal plane

The resultant bending moments on the shaft can be determined according to equation (47) as detailed in table 3.3.

$$M_b = \sqrt{M_y^2 + M_x^2} \tag{47}$$

Where, M_b = resultant bending moments (N), M_y = maximum bending moments on the Vertical plane (N), and M_x = maximum bending moments on the horizontal plane (N)

Table 3.3.Transmission Shaft Resultant Bending Moments

At	A	B	C	D	E
$\boldsymbol{M_y}$	0	$5.64Nm$	$36.82Nm$	$82.02Nm$	0
$\boldsymbol{M_x}$	0	0	$4.01Nm$	0	0
$\boldsymbol{M_b = \sqrt{M_y^2 + M_x^2}}$	0	$5.64Nm$	37.04Nm	$82.02Nm$	0

It was seen that bending moment was maximum at C

∴Maximum bending moment, $M_m = M_D = 82.02Nm$

As previously stated, a transmission shaft is a shaft that is subjected to varying torque and bending moments. Equations (48), (49), (50), and (51) and Km & Kt 1 to 3 were used to compute the equivalent twisting and bending moments for rotating shafts (Guru et al., 2019).

$$.T_e = \sqrt{\mathrm{K_t T^2 + k_m M_m^2}} \quad (48)$$

$$\mathrm{T_e = \tau d^3 \frac{\pi}{16}} \quad (49)$$

and equivalent bending moment,

$$\mathrm{M_e} = \frac{1}{2}(\mathrm{k_m M_m} + T_e) \quad (50)$$

$$\mathrm{M_e = \frac{\pi d^3}{32} \sigma_b} \quad (51)$$

Where, Te=equivalent twisting moment, Km = Combined shock and fatigue factor for bending τ= Torsional shear stress, σ_b=Bending stress and Kt = Combined shock and fatigue factor for torsion. T=Twisting moment of the shaft M=Bending moment, Me=equivalent bending moment

$$\mathrm{T_e} = \sqrt{\mathrm{K_t T^2 + k_m M_m^2}}$$

$\tau_{Transmission\ shaft} = T = 4.42Nm$, $k_t = 3$, $k_m = 1$, and $M_m = M_D = 82.02Nm$

$$\mathrm{T_e} = \sqrt{3 \times 19.54\mathrm{N^2m^2} + 1 \times 6727.28N^2m^2} = 82.38\mathrm{Nm}$$

According to the American Society of Mechanical Engineers (ASME) code for the design of transmission shafts, $\tau = 42$ MPa=42N/mm^2 and $\sigma_b = 84\ MPa = 84N/mm^2$ for shafts with allowance for keyways (Hakizimana et al., 2020).

$\mathrm{T_e = \tau d^3 \frac{\pi}{16}}$ d = $\sqrt[3]{\frac{16\times T_e}{\tau\pi}} = 21.54mm \approx 22mm$

$\mathrm{M_e} = \frac{1}{2}(\mathrm{k_m M_m} + T_e) = 82.20\mathrm{Nm}$

$\mathrm{d} = \sqrt[3]{\frac{32\times \mathrm{M_e}}{\sigma_b \pi}} = 21.51mm \approx 20mm$

The diameter of the transmission shaft was determined to be 20mm which is the standard bearing size near to calculated value.

3.5.9.2. Selection of Idler Shaft

The idler shaft was necessary to reduce the rotational speed of the transmission shaft (108 rpm) to 14 rpm, as advised by various publications. It is supported by two pulleys and two supporting bearings. A rubber belt connects the driver transmission shaft pulley (75mm) to the idler shaft-driven pulley (140mm), and the idler shaft driver pulley (75mm) to the main (drum) shaft-driven pulley (155mm). As illustrated in figure 3.14, the two bearings were attached on the frame column, allowing the shaft to pass through the frame column and the bearings to hold the pulleys at either edge.

The torque exerted by the idler shaft on other assemblies is the first parameter used to compute its diameter (equation 52). (Sharma & Mukesh, 2010).

$$\frac{D_2}{D_1} = \frac{\tau_{Idler}}{\tau_{Transmission\ shaft}} \tag{52}$$

Where, D_1 =transmission shaft driver pulley, D_2 = Idler shaft-driven pulley, τ_{Idler}=torque exerted by idler shaft, and $\tau_{Transmission\ shaft}$=torque exerted by the transmission shaft

$$\frac{0.140m}{0.075m} = \frac{\tau_{Idler}}{4.42Nm},\ \tau_{Idler} = 8.25Nm$$

The torque imparted on the driven idler shaft pulley was computed using equation in another instance (41).

Equation (53) was used to compute the angle of contact (θ_1=3.50 rad) at the bigger pulley of the idler attached to the transmission, shaft pulley (Khurmi & Gupta, 2005; Shigley & Mischke, 1996).

$$\theta_1 = \left(180^0 + 2 \times \sin^{-1}\left(\frac{D-d}{2C}\right)\right)\frac{\pi}{180} rad \tag{53}$$

Where, θ_1= the angle of contact or lap at the larger pulley, D=Diameter of larger pulley (m), d=Diameter of smaller pulley (m), C =The center distance between transmission shaft pulley and idler shaft pulley

$$\theta_2 = \left(180^0 - 2 \times \sin^{-1}\left(\frac{D_5-D_4}{2C}\right)\right)\frac{\pi}{180} rad$$

Where, $\theta_2 = 2.71\ rad$ which is the angle of contact at the smaller pulley of the idler shaftD_5 = Drum shaft pulley diameter (m), D 4 = Idler shaft driver pulley diameter (m), C=Idler shaft pulley and Drum shaft pulley center distance.

As shown in appendix table 1.7, the coefficient of friction between cast iron and rubber is 0.30, i.e. μ=0.30.

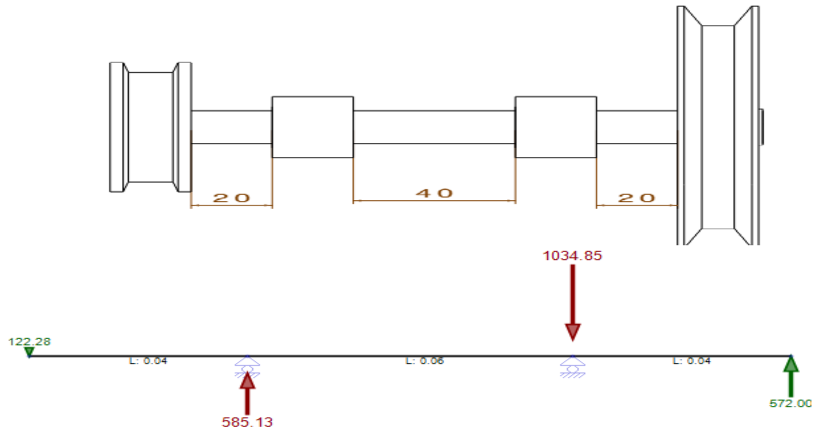


Figure 3.14.Idler shaft loading diagram on the vertical plane

(all dimensions in M and all forces in N)

The instant about point 3 was taken as M3y according to equation (46) to determine the reaction at two points (R_2)

$$+\circlearrowright \sum M_{3y} = 0$$

$$R_2 = \frac{35.11Nm}{0.06m} = 585.13N$$

The total of forces in the vertical direction (FV) is zero at equilibrium. As a result, the equilibrium equation was used to determine the reaction at point 3 (R_3).

$$\uparrow + \sum F_v = 0$$

$$R_3 = -T + R_2 + T_i$$

$$R_3 = -122.28N + 585.13N + 572N = 1034.85N$$

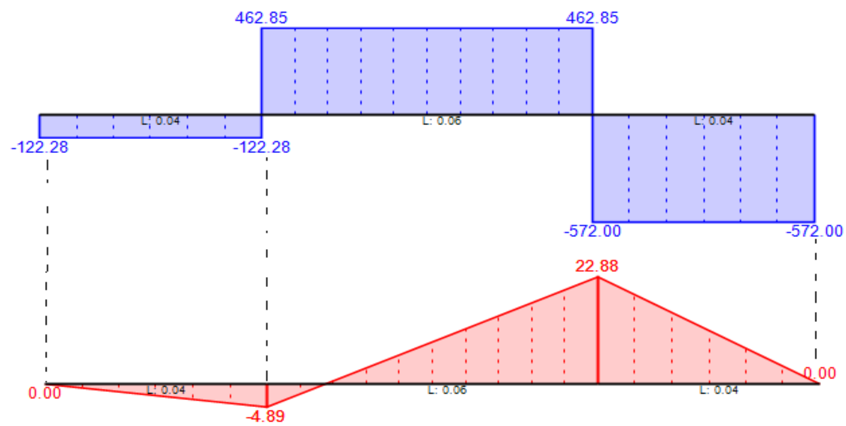


Figure 3.15. Shear force and bending moment diagram of idler shaft on the vertical plane

The bending moment was highest at point 3, which was 22.88Nm.The idler shaft is another type of shaft that experiences variable torque and bending moments. Using equations (48), (49), (50), and (51), the corresponding twisting (Te=83.26Nm) and bending moments of the idler shaft were determined in the same way as the transmission shaft. To design the shaft with allowance for keyways and subject to twisting moment, the shear and bending stresses are 42MPa and 84MPa, respectively, as indicated in the section above.

$$\mathrm{T_e} = \tau \mathrm{d}^3 \frac{\pi}{16} \qquad \mathrm{M_e} = \frac{1}{2}(\mathrm{k_m M_m} + T_e)$$

$$\mathrm{d} = \sqrt[3]{\frac{16 \times T_e}{\tau\pi}} = 21.59mm \approx 22mm \qquad \mathrm{d} = \sqrt[3]{\frac{32 \times \mathrm{M_e}}{\sigma_b \pi}} = 21.56mm \approx 20mm$$

Therefore the diameter of the idler shaft was also determined to be 20mm because 20mm is the standard bearing size near 21.56mm.

3.5.9.3.Carrot Washing and Grading Cylinder (Drum) Design

The drum is a cylinder into which water is pumped through a nozzle into which washed carrots are sorted into desired sizes. It rotates at the desired speed for the purpose using a sprocket and chain configuration. Carrots could be loaded through the door.

The weight of the drum (WD), the mass of unwashed and ungraded carrot (WC), the drum diameter (D), and the actual volume of the drum (VD) are all critical for optimal operation and achieving the desired outcomes. The weight (WD) of the washing and grading unit was calculated using equation (54) using the volume of the washing and grading cylindrical unit (V D) and the density of the Eucalyptus tree (= 398kg/m3) used to construct the Drum (Frederick et al., 1982).

$$W_D = \rho g V_D \tag{54}$$

$$W_D = 3904.38 \frac{N}{m^3} \times V_D$$

Where $g = 9.81\ m/s^2$

Dereje (2019) computed the volume of the washing and grading cylindrical unit (VD) by subtracting the total volume of the holes or slots (Vs) from the total volume of the solid cylindrical washing and grading unit (Vsolid) using equation (55).

$$V_D = V_{solid} - V_s \tag{55}$$

The complete solid drum should be filled 1/4 to 1/6, or 4 to 6 times the volume of unwashed and ungraded carrots, for better washing and grading (Anjali & Ikhar, 2018b). For hand cranking machines, the ideal capacity of the drum for washing and grading was around 10kg to 15kg per per charge, as reported by many scholars in various types of literature. For developing this machine based on the recommendation above, WC=98.1N, the volume of the drum was taken as 6 times the volume of unwashed and ungraded, and its capacity was taken as 10kg per one charge.

Because the bulk density of the observed carrot samples was 500 kg/m3, the volume of unwashed and ungraded carrots, as well as the inner diameter of the washing and grading drums, were calculated using the aforementioned data and the volume, density, and mass connection as indicated by equations (56) and (57).

$$V_{solid} = 6 \times V_{Carrot} \tag{56}$$

Where, V_{solid} =volume solid washing and grading drum

V_{Carrot}= the volume of unwashed and ungraded carrots

$$V_{Carrot} = \frac{m}{\rho} \tag{57}$$

$$V_{Carrot} = \frac{10kg}{500kg/m^3} = 0.02m^3$$

Where, m =mass of unwashed and ungraded carrots (kg)

ρ =bulk density of unwashed and ungraded carrots ($\frac{kg}{m^3}$)

$$\therefore V_{solid} = 6 \times 0.02m^3 = 0.12m^3$$

The volume of the slot is calculated using equation (58), which is equal to the volume of the slot in the washing unit (Vws) plus the volume of the slot in the grading unit (Vsg) (Stroshine and Hammand, 1994).

$$V_s = V_{ws} + V_{sg} \tag{58}$$

In the washing unit, the space between the wood planks was 2mm to allow only water with impurities to get through, while in the grading unit, the spacing between timbers was set depending on the GMD of the selected carrot sample. The sample was divided into four groups based on the geometric mean. At an initial spacing of the timbers in the grading unit, smaller diameters less than 20mm were graded. The carrots with diameters of 20mm to 30mm were graded in the next gap. Carrots with a diameter of 30mm to 40mm were graded at an intermediate section of the grading unit, while carrots with diameters greater than 40mm were graded at the cylinder's out port.

Based on these findings, wood planks with a length of 1.5 meters and a thickness of 20 millimeters were chosen to allow enough time for washing and grading. The lengths of the grading and washing units were 0.7m and 0.8m, respectively (see figure 3.16 and 3.17).

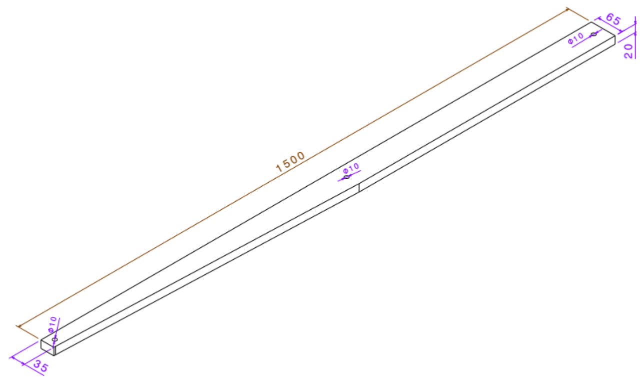


Figure 3.16.Isometric View of Timber

(all dimensions are in mm)

The volume of slot (V_{ws}) in washing unit determined using equation (60) from the geometry of figure 3.17

$$V_{ws} = LWTn \qquad (59)$$

$$V_{ws} = 700mm \times 1mm \times 20mm = (1.40 \times 10^5 mm^3) \times n$$

Where, L =Length of slot in washing unit (mm), W =Width of slot in washing unit (mm),

T=thickness of timber in the washing unit (mm), and n = number of Tiber

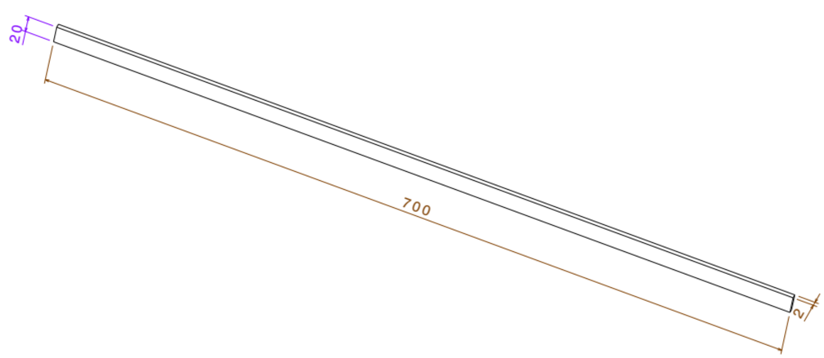


Figure 3.17.Isometric view of slot in washing unit

(all dimensions are in mm)

As illustrated in figure 3.18, the slots are trapezoidal in shape, with a width of 2mm at the end of the washing section and 30mm at the outport. As a result, its volume (V sg) is calculated using the trapezium volume formula (equation) (60).

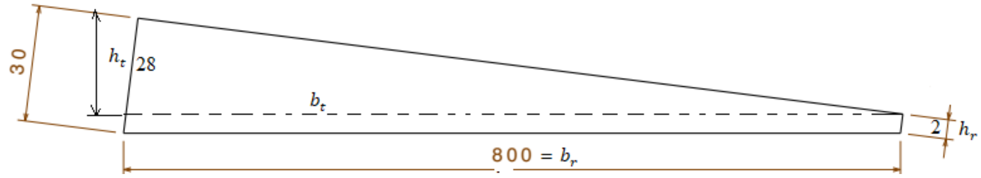


Figure 3.18. The shape of slot in grading unit

(all dimensions are in mm)

$$V_{sg} = \left(\frac{1}{2} b_t h_t t + b_r h_r t\right) n \quad (60)$$

Where bt is the base of the triangular parts of the trapezoidal slot (mm), br is the base of the rectangular parts of the trapezoidal slot (mm), ht is the height of the triangular parts of the trapezoidal slot (mm), hr is the height of the rectangular parts of the trapezoidal slot (mm), t is the thickness of the grading cylinder that makes the Timber (mm), and n is the number of slots

V sg=(1/2×800mm×29mm×20mm+800mm×2mm×20mm)

n V sg = (3.23104 mm3)n

The number of Slots was calculated by dividing the solid circumference of the washing and grading cylinder by the sum of the width of the timber and the gap between the timber at the washing unit, as indicated by equation (61). Because the width of the timbers in the washing machine is the maximum and the slots are the smallest, the width of the timbers is the maximum and the slots are the smallest. $n = \frac{\pi d_i}{w_i + c_i}$

(61)

Where, n =total number of timber, d_i = Inner Diameter of washing and grading unit (Drum) (mm), w_i = width of timber at washing unit (mm), c_i = gap size in washing unit (mm)

$$n = \frac{\pi d_i}{65mm + 2mm} = 0.05mm \times d_i$$

The cylinder volume formula (equation 62) is used to calculate the volume of the solid hollow cylinder (V solid) from the inside diameter (di) and length of the cylinder (L).

$$V_{solid} = \frac{\pi {d_i}^2 L}{4} \quad (62)$$

$$\therefore V_{solid} = \frac{\pi \times {d_i}^2 \times 1500mm}{4} = 1178.10mm \times d_i^2 = 1.18m \times d_i^2$$

Therefore, the diameter of washing and grading cylinder determined equalizing the equation (58) and (62) as following.

$$0.12m^3 = 1.18m \times d_i^2, d_i = 0.32m = 320mm \therefore n = 0.05mm \times 320mm = 15$$

After the value of n determined, V_{ws} and V_{sg} calculated according to equations (59) and (60) as follow

$$V_{ws} = (1.40 \times 10^5 mm^3) \times 15 = 2.10 \times 10^6 mm^3 = 2.10 \times 10^{-3} m^3$$

$$V_{sg} = (3.23 \times 10^4 mm^3) \times 15 = 4.85 \times 10^5 mm^3 = 4.85 \times 10^{-4} m^3$$

Therefore, using equations (60),(57),(56) respectively, the value of $V_s = 2.59 \times 10^{-3} m^3$, $V_D = 0.117m^3$, and $W_D = 458.43N$ were estimated.

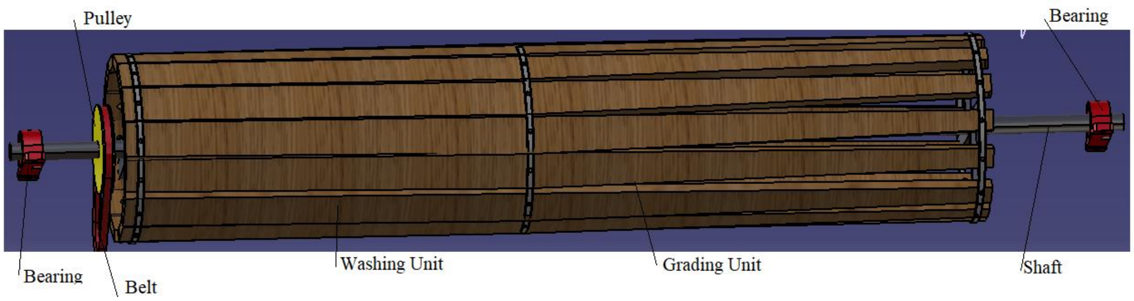


Figure 3.19.Carrot washing and grading machine drum assemblies modeled by CATIA V5

3.5.9.4. Drum Rotating Shaft Selection

The weight of the washing and grading cylinder (WD), the maximum weight of carrot tuber feed (WC), the weight of the pulley, and belt tensions all operate on the drum shaft.

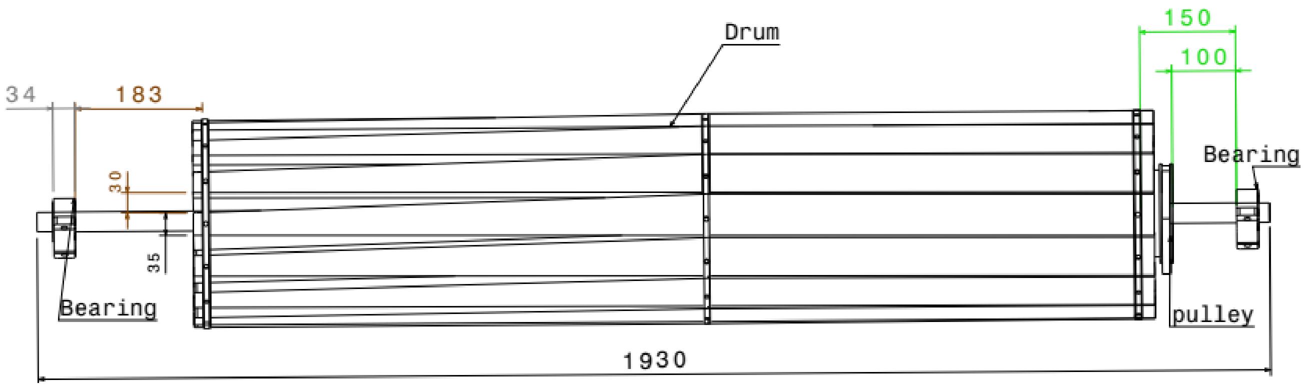


Figure 3.20.Washing and grading drum

(all dimensions are in mm)

The tensions (T=572N) and the weight of the pulley (WP= 4.91N) act downward in a single point, and WC and WD act as a single force. Wp+T= 576.91N and WD+WC= 556.53N, respectively. The load applied to the shaft by the product and Drum is triangular uniform because the load applied by the product is high in the washing unit and low in the grading unit. According to equation (63) (Jatobá et al., 2016), the load operating on the Shaft in Washing and grading units is the area under the triangle, which is the distributed load (w) divided by two, that is (wL/2) (Jatobá et al., 2016). As the triangular load (w) is acting straight down, the horizontal reaction (A_x)is zero.

$$\frac{wL}{2} = W_D + W_C \tag{63}$$

$$w = \frac{2(W_D+W_C)}{1.5m} = 742.04\frac{N}{m}$$

Where w=he distributed load triangular load (N/m)

$W_D + W_C$=the sum of the weight of the drum and unwashed and ungraded carrot (N)

L=length of Drum (m)

The center of gravity (C.G.) of the triangle lies a distance of $\frac{2}{3} \times L$ from the apex.

$$i.e. C.G. = \frac{2}{3} \times 1.5m = 1m$$

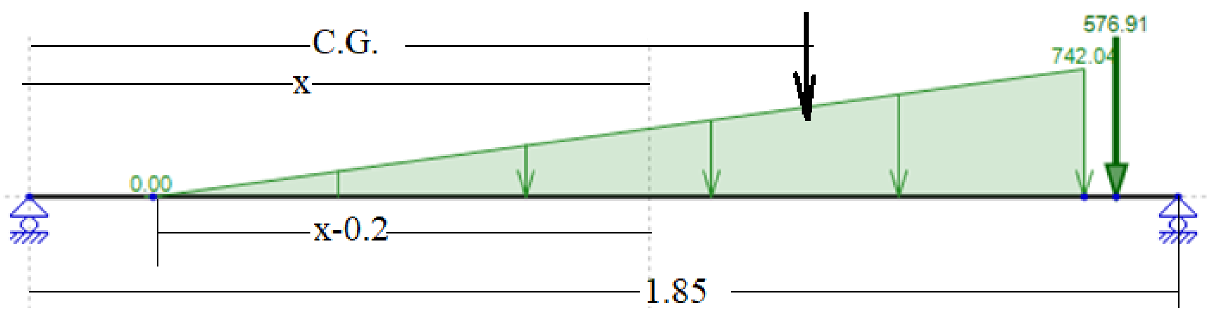


Figure 3.21.Drum shaft load diagram

To determine the reaction at E point (R_E), the moment about point A was taken as M_{Ay} according to equation (47).

$$+\circlearrowright \sum M_{Ay} = 0$$

$$(-1.85m \times R_E) + (1.75m \times 576.91N) + ((0.2m + 1m) \times 556.53N) = 0$$

$$R_E = 906.72N$$

At equilibrium, the summation of forces on the vertical direction (F_V) is zero. Consequently, the reaction at point 1 (R_1) was determined from the equilibrium equation.

$\uparrow + \sum F_v = 0$

$$R_A = 556.53N + 576.91N - 906.72N = 226.72N$$

Shear forces at each point were calculated as follows

$\uparrow +V_y$ = shear force at x distance from point 1 to the required point.

$$V_A = 226.72N, V_B = 226.72N, V_C = 226.72N - 556.53N = -329.81N$$

$$V_D = -329.81N - 576.91N = -906.72N, V_E = -906.72N + 906.72N = 0$$

The direction of V_B and V_C are opposite, that means there is a point where the Shear force becomes zero between point B and C. Let's take section xx at distance x from point A between B and C at which V_x=0.

$$V_x = R_A - area\ of\ section$$

$$0 = R_A - \frac{1}{2}bh, b = x - 0.2m$$

$$0 = 226.72N - \frac{1}{2}(x - 0.2m)h \quad \text{(k)}$$

By similar triangle law

$$\frac{742.04N}{1.5m} = \frac{h}{x-0.2m}$$

$$h = 494.69N(x - 0.2m) \quad \text{(l)}$$

By substituting equation (l) into (k)

$$247.35N(x - 0.2m)(x - 0.2m) = 226.72N, x = 1.16m$$

Therefore, shear force is zero at a distance of 1.16m from supporter A or $x - 0.2m = 0.96m$ from point B.

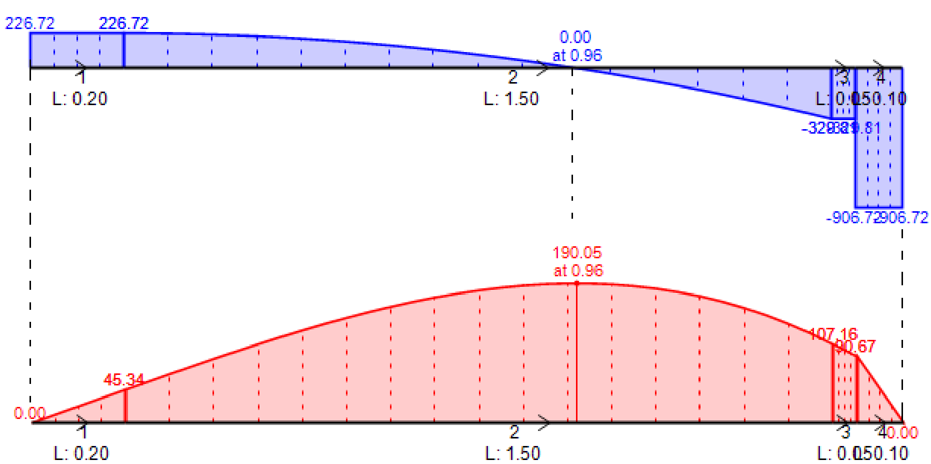


Figure 3.22.Shear force and bending moment diagram of the drum shaft on the vertical plane

The bending moment on the drum shaft as a result of the equation was also calculated (47). And the maximum bending moment was 190.05Nm at point x, i.e. M m=190.05Nm. Drum shafts are hollow shafts with perforated holes that are used to transmit water from the washing drum and are subjected to variable torque and bending forces. Equations (47), (49), (50), and (51), respectively, were used to compute the equivalent twisting and bending moments of the Drum shaft in the same way that they were calculated for transmission and idler shafts.

$$\tau_{Drum} = 8.25Nm \times \frac{0.155m}{0.075m} = 17.05Nm,$$

$$\mathrm{M_e} = \frac{1}{2}(\mathrm{k_m M_m} + T_e),$$

$$\mathrm{d} = \sqrt[3]{\frac{16 \times T_e}{\tau\pi}} = 28.57mm \approx 30mm$$

$$\mathrm{d} = \sqrt[3]{\frac{32 \times \mathrm{M_e}}{\sigma_b \pi}} = 28.51mm \approx 30mm$$

Therefore the diameter of the idler shaft was also determined to be 30mm according to the above procedures.

3.5.10. Bearings Selection

A bearing is a machine component that limits relative motion between moving elements to to that which is desired. The static load carrying capacity, bearing life, dynamic load-carrying capacity, equivalent dynamic load, and load life relationship are some of the elements that should be considered during bearing selection, as indicated in section 2.10. The shafts diameter serve to define the bore size of the bearings; and resulting axial and radial force acting on the shaft was estimated for selection of the kind of bearings from axial to radial load ratio according to Khurmi and Gupta, (2005) and Richard et al, (2011) using equation (64).

$$F_R = \sqrt{R_{BX}^2 + R_{BY}^2} \tag{64}$$

For transmission shaft

$$F_{RA} = \sqrt{R_{BX}^2 + R_{BY}^2} = \sqrt{(36.46N)^2 + (95.56N)^2} = 102.28N$$

$$F_{RB} = \sqrt{R_{DX}^2 + R_{DY}^2} = \sqrt{(36.46N)^2 + (2461.54N)^2} = 2461.81N$$

For Idler shaft

$$F_{R2} = \sqrt{R_{2X}^2 + R_{2Y}^2} = \sqrt{0 + (585.13N)^2} = 585.13N$$

$$F_{R3} = \sqrt{R_{3X}^2 + R_{3Y}^2} = \sqrt{0 + (1034.85N)^2} = 1034.85N$$

For Drum shaft

$$F_{RA} = \sqrt{R_{AX}^2 + R_{AY}^2} = \sqrt{0 + (226.72)^2} = 226.72N$$

$$F_{RE} = \sqrt{R_{EX}^2 + R_{EY}^2} = \sqrt{0 + (906.72N)^2} = 906.72N$$

Where, R_{BX} denotes the maximum resultant force acting on a bearing at point B in the horizontal plane (kN), R_{BY} denotes the highest resultant force acting on a bearing at point B in the vertical plane (kN), and F_R denotes the greatest resultant radial reaction force acting on the bearings at point B. (kN).

Because the axial reaction was not applied to the three shafts, all of them had Fa=0 and F a(FR0.7). A single row deep groove ball bearing (SRDGBB) was used for the three shafts.The bearing dynamic capacity of each bearing was estimated in the second phase. Associations such as ABMA and ISO have specified the method of measuring bearing dynamic capacity as equation (65) (Edinger, 2003).

$$C = P_D \left(\frac{L}{10^6}\right)^{\frac{1}{K}} \tag{65}$$

Where, L = Rating life, C = Basic dynamic load rating, P_D = design basic dynamic equivalent radial load, and $k = 3$, for ball bearings,$k = \frac{10}{3}$, for roller bearings. Since the selected bearings were SRDGBB, therefore, $k = 3$.

Basic dynamic equivalent radial load (P_D) was determined according to equation (66) (NTN, 2011).

$$P_D = PK_s \tag{66}$$

$$P_D = 1.5P$$

Where, K_s= service factor**,** p=equivalent dynamic load

The machine loads were considered as light shock load therefore the value of K_s determined to 1.5 according to (Khurmi & Gupta, 2005).

The Equivalent dynamic load was estimated as Equation (67) according to (Ashahi, 2016).

$$P = X.V.F_R + Y.F_a \tag{67}$$

The axial or thrust load factor is denoted by Y, and the rotation factor is denoted by V. For all types of bearings, v = 1 when the inner race is rotating, v= 1 when the inner race is stationary for self-aligning bearings, and v= 1.2 for all types of bearings except self-aligning bearings.
The radial load factor is denoted by the letter X. The rotation factor was considered to be 1 because the shaft rotates the inner race (see appendix table1), i.e. v=1. For Fa/FR =0,X=1 and Y=0, the values of radial load factor (X) and axial or thrust load factor (Y) for dynamically

loaded bearings were obtained from appendix table 1.8. As a result, we estimated the equivalent dynamic loads for the transmission, idler, and drum shafts as follows.

For transmission shaft

$P_{RA} = 102.28$ $\quad$ $P_{RB} = 2461.81N$

For Idler shaft

$P_{R2} = 585.13N$ $\quad$ $P_{R3} = 1034.85N$

For Drum shaft

$P_{RD} = 226.72N$ $\quad$ $P_{RE} = 906.72N$

P_{RA}= Equivalent dynamic load of transmission shaft bearing A, P_{RB}= Equivalent dynamic load of transmission shaft bearing B, P_{R2}= Equivalent dynamic load of Idler shaft bearing 2, P_{R3}= Equivalent dynamic load of Idler shaft bearing 3, P_{RE}= Equivalent dynamic load of Drum Shaft bearing E, and P_{RD}= Equivalent dynamic load of Drum shaft bearing A.

Because the life of a bearing in hours (LH) for stationary machines ranges from 12, 000 to 20,000, the rating life in revolution was computed using equation (68) and the estimated life in working hours of 12,000 from appendix table 1.10. (Bsen-ds, 2018).

$$L = 60N \times L_H \tag{68}$$

$$L = 60N \times 12{,}000\ Hr = 7.2N \times 10^5$$

Where, L =Rating life in revolution, L_H = is the life in working hours estimated, and N = the shaft speed in r.p.m.

The speeds of the Transmission, Idler, and Drum Shafts are 108 rpm, 57.86 rpm, and 28 rpm, respectively. Thus, the bearings for Transmission shafts, Idler shafts, and Drum shafts have a rating life of 77.8×10^6, 41.70×10^6, and 20.20×10^6 revolutions, respectively. P_D for transmission shaft bearing A and B, Idler shaft bearing 2 and 3, and Drum shaft bearing E and A were obtained as 153.42N, 3692.72N, 877.70N, 1552.28N, 340N, and 1360N, respectively, by multiplying each Equivalent dynamic load of the shafts by ks factor or 1.5 using equation (66). Equation (65) was used to calculate the dynamic load-carrying capacity (C) for each bearing using the values P_D, and the result of equation (68) was as follows.

For transmission shaft

For Idler shaft

$C_{R2} = 3.04kN$, $C_{R3} = 5.38kN$, $C_{RA} = 0.65kN$, and $C_{RB} = 15.76kN$

For Drum shaft

$C_{RD} = 0.93kN$ and $C_{RE} = 3.70kN$

Where, C_{RA}= Bearing A of the Transmission shaft's dynamic load-carrying capability C_{RB}= Bearing B of Transmission shaft B's dynamic load-carrying capacity C_{R2}= The dynamic load-carrying capability of Idler shaft bearing 2, C_{R3}= Bearing 3 of the Idler shaft's dynamic load-carrying capacity C_{RE}= The E drum shaft bearing's dynamic load-carrying capability C_{RD}= the dynamic load-carrying capacity of Drum shaft bearing A.

Bearing number 204 was chosen for supporter A of the Transmission Shaft and both supporters of the Idler shaft after comparing the computed C for each bearing with the value given by the catalog based on the bore diameter in appendix table 11. The 404 bearing was used for the transmission shaft's supporter B. Finally, bearing number 206 was chosen for both the drum shaft's supporter E and A.

3.5.11. Design of the Frame

The frame is the machine's chaises, on which all other components are mounted, such as the drums, seat, handlebar, sprocket, chain, and so on. It was designed to withstand the largest magnitudes of compressive, tensile, and impact forces while also supporting the other machine elements attached to it. This machine had frame components such as horizontal pedal support, vertical Seat and handle support, horizontal water pump support, inclined transmission shaft support, horizontal idler shaft support, horizontal Main shaft support, and machine base, as illustrated in Figure 3.23. To create a sturdy and efficient frame, these vertical, horizontal, and inclined components were welded together. Because of its availability and relatively inexpensive cost, hollow square steel was used for all frame components. Except for the transmission shaft and idler shaft supports, which were made of 60 mm 60 mm 4 mm steel because they support the bearing that fits in a drilled 50 mm sized shaft welded to the frame, the remaining frame components were made of mild steel with a diameter of 40 mm 40 mm 3 mm.

Modifying an existing bicycle frame provided the dimensions for the operator seat and base frame components. The dimensions of the other frame components were determined by the Drum's size and location. The horizontal distance between the vertical columns was calculated using the cylinder's diameter, which came out to be 0.4m. The length of the frame (1.93m) from intake to outlet vertical columns was calculated using the length of the drum and the unloading space between the drum and the drum shaft.

The length of vertical columns at the intake (1.1m) was chosen based on their suitability for loading, operating, and pumping. The vertical columns at the outlet length (0.7m) were shorter than the intake components to allow for gradual product migration from the inlet port to the cylinder's outlet port depending on the measured static friction angle (18.50). Sample loading and unloading would be easier with an elevated frame.

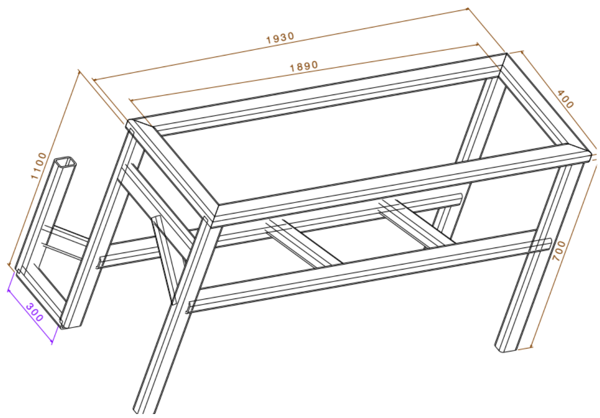


Figure 3.23. The frame

3.6.Carrot Washing and Grading Machine Construction

With the help of CATIA V5 software, the machine was built according to preset measurements for each component. The Machine's many components, as explained in the subheadings above, were fabricated and assembled in the local workshop. For the pump cylinder and piston, the aluminum alloy was molded and bored to the dimensions specified on the design paper. For leakage protection, the rubber ring is attached to the piston and front cap.

Pump assemblies were joined to the frame components after they were welded together in the desired shape. Wood that is relatively waterproof was chosen and machined to predefined forms and dimensions in the workshop. Because attaching the wood to ring was straightforward, the drum rings that tie the wood planks together were drilled with an equal number of timber/wood planks.The chosen chain and sprockets, belts and pulleys, and bearings were installed on the machine at the positions defined during the design process. The diameter and length of the designed shafts were prepared and assembled in their final positions.

3.7.Data Gathering and Organization

Local farmers in the Arsi zone of Sirka District provided 225 kg of carrot tuber. The tubers were separated into eighteen samples (each including tiny, medium, and large tubers when the total drum volume was filled with 1/6 and 14 accordingly, as repeatedly advocated by several academics for safe carrot tuber washing). The machine was put through its paces to see how efficient it was in washing, grading, and detecting mechanical damage. CRD did the performance test with 10 kg and 15 kg carrot samples with drum speeds of 14 rpm, 21 rpm, and 28 rpm, which were indicated in the literature for optimum carrot washing performance. Each experiment was carried out three times.

3.8.Carrot Washing and Grading Machine Evaluation

3.8.1. Preliminary Assessment

Visual observations and specification comparisons were used to conduct a preliminary evaluation in the local workshop without a load. During visual inspection, the machine was placed on a level area of the workshop and the operating state was inspected, as well as moving and rotating parts, sound, and any other elements such as vibration and shock. Similarly, during the specification review, the machine was placed on a level surface and the completed and manufactured elements were meticulously measured. The machine parts were then compared to the design specifications.

3.8.2.Indicator of performance

Equations 69 - 75 were used to calculate washing capacity, grading capacity, washing efficiency, grading efficiency, and percentage of mechanical defects (Ambrose & Annamalai, 2013; Siddique et al., 2017; El-Rahman & Mosa, 2011; Charles & Etiese, 2020: Valentin et al., 2017; Dereje, 2019). The machine capacity of the carrot tubers washer and grader was calculated using equation 69, which was based on the number of carrot tubers washed and graded in a certain amount of time.

$$C = \frac{m_{total}}{t} \tag{69}$$

Where, C=machine capacity (kg/h), m_{total}=total mass of unwashed and ungraded sample (kg), and t= the time consumed in operation (h).

The Washing efficiencies of the machine were determined by dividing the mass of the sample after washing by total mass sample by the total number of the sample as given by equation (70)

$$we = \frac{m_{after}}{m_{total}} \times 100 \qquad (70)$$

Where, we =washing efficiency (%) and m_{after}=average mass of washed sample received at all outlets in kg

Grading system efficiency was determined by taking the products of the efficiencies of grading small, medium, and large tubers and Equations (71), (72), (73), and (74) according to

$$Ge\ (\%) = (\eta_s \times \eta_m \times \eta_L) \times 100 \qquad (71)$$

$$\eta_s = \frac{mass\ of\ corectly\ graded\ small\ tuber}{Total\ mass\ of\ small\ tuber\ in\ the\ sample} \qquad (72)$$

$$\eta_m = \frac{mass\ of\ corectly\ graded\ medium\ tuber}{Total\ mass\ of\ medium\ tuber\ in\ the\ sample} \qquad (73)$$

$$\eta_L = \frac{mass\ of\ corectly\ graded\ larger\ tuber}{Total\ mass\ of\ large\ tuber\ in\ the\ sample} \qquad (74)$$

where Ge denotes the effectiveness of the grading system (in percent), η_s denotes the efficiency of grading smaller tubers, η_m denotes the efficiency of grading medium tubers, and η_L denotes the efficiency of grading big tubers.

Abrasion damage to tubers after washing and grading was taken into account. As shown in Equation 1, the percentage of damaged tubers was calculated by comparing the total number of tubers with abrasion after the procedure to the total number of tubers in the sample (75)

$$\text{Damaged tuber}(\%) = \frac{\text{Total mass of tuber with abrasion}}{\text{Total mass of tuber in the sample}} \times 100 \qquad (75)$$

3.8.3.Statistical Analysis

SPSS statistical software was used to conduct the analysis. When the treatment's effects were determined to be significant, an LSD test was used to determine the difference between the treatments at a 5% level of significance.

CHAPTER FOUR

4. RESULTS AND DISCUSSION

The machine's performance was measured, including the time it took to finish the washing and grading activities, throughput capacity, washing efficiency, grading efficiency, and mechanical damage.

4.1.The amount of time it takes to do the washing and grading tasks

The goal of this study was to see if the drum speed and feeding load had an impact on the time it took to complete the carrot washing and grading tasks.

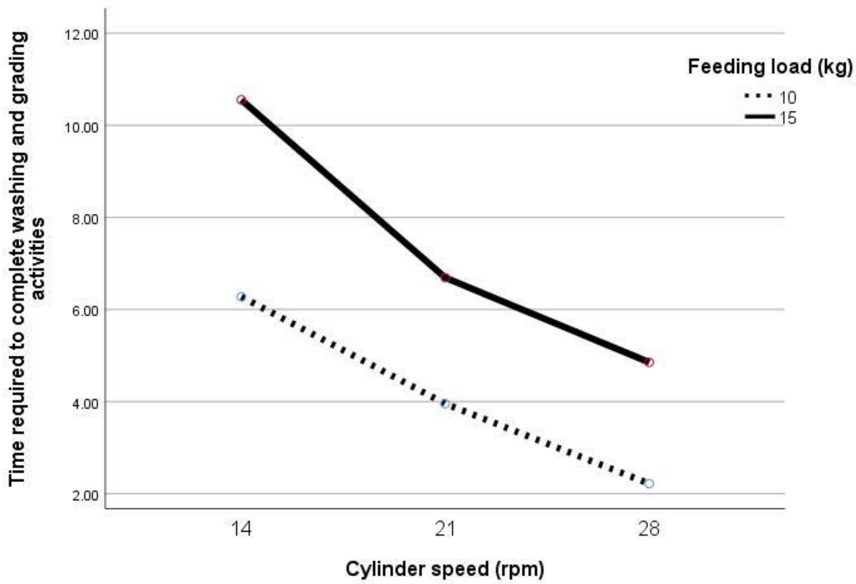


Figure 4.1.Time required to perform washing and grading processes as a function of feeding load and cylinder speed

The cylinder speed has a negative relationship with time, as seen in figure 4.1. The loaded carrot is forced to exit port fast as the drum rotation increases, reducing the duration. As a result, as the cylinder size was raised, there was a significant reduction in time. The feeding burden, on the other hand, has a positive relationship with time. Because the drum speed drops as the feed weight increases, the time is elongated. As a result, as the feeding load was raised, there was a significantly substantial rise in time. As indicated in figure 4.1, the average minimum and maximum time to complete the washing and grading procedure was 2 minutes and 10.88 minutes, respectively. When compared to traditional washing and grading processes, which take at least 12 minutes for the same samples, this method is superior.

4.2.Washing Efficiency

The washing efficiency increased from 91.13 to 98.70 percent for the range of variable speed between 14 and 28 rpm when the machine was operated with 10 kilogram feeding materials, and from 84.87 to 95.24 percent when the machine was operated with 15 kg feedings, according to the evaluation data. As indicated in figure 4.2, the influence of drum speed and feeding load on washing efficiency was observed. Due to the tremendous impact of the washing drum on the carrot tubers, the washing efficiency steadily increases as the speed increases. Furthermore, when the feeding load increases, it is lowered because the drum-free area becomes occupied with carrot tubers, which prevent water from reaching each of the single products. In addition, the water added to the drum boosted the washing efficiency based on pedaling power. Crank rotation and cylinder speed increased as the operator increased pedaling force. The high impact of the washing drum applied to the tuber was the reason for the high washing efficiency. As the crank rotation and cylinder speed increased, enough water was induced to the washing drum, and the high impact of the washing drum applied to the tuber was the reason for the high washing efficiency.

Due of the high impact and high water fed to the drum, the effect analysis for washing efficiency was maximum (98.70 percent) at 28rpm and feeding load 10kg, respectively. For the effect of speed drum on washing efficiency, our finding outperforms those of (Ambrose & Annamalai, 2013) and (Amin & Hossain, 2021).

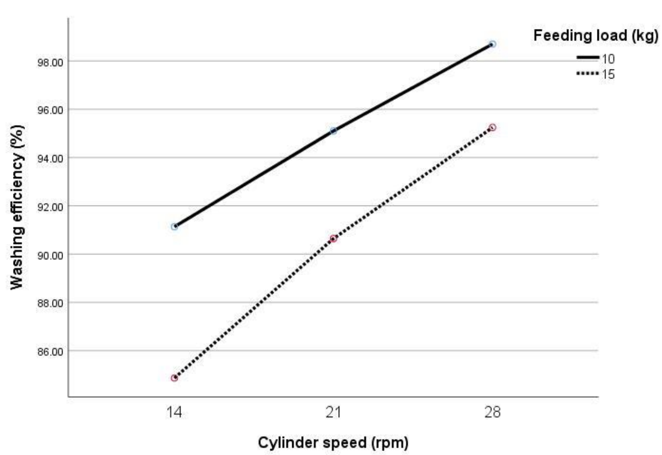


Figure 4.2.Effects of feeding load and cylinder speed on the washing efficiencies

4.3.Grading Efficiency

When the machine is operated with 10 kilogram feeding materials, the grading efficiency ranges from 81.87 to 92.23 percent, and when the machine is operated with 15 kg feeding materials, the grading efficiency ranges from 72.89 to 80.86 percent. With the effect of cylinder speed, the grading efficiency inclined, but dropped with the effect of feeding load, as shown in Figure 4.3. This pattern is the outcome of the grading efficiency evaluation technique, which is based on the ratio of graded tuber mass acquired from correctly graded tubers in their right outflow to total tuber mass at the drum. As the speed increased from 14rpm to 28rpm, more tubers were retrieved at the larger outlet (out port) in the test. As a result of the high drum speed, there's a chance that tiny tubers will pass through their right outlet and into the larger outlet. Unlikely, the grading efficiency declined steadily as the feeding load rose, as shown in figure 4.3, due to the growing quantity of tubers covering the right outlet and forcing it to the outer port. As a result, there's a good chance you'll grade things erroneously.

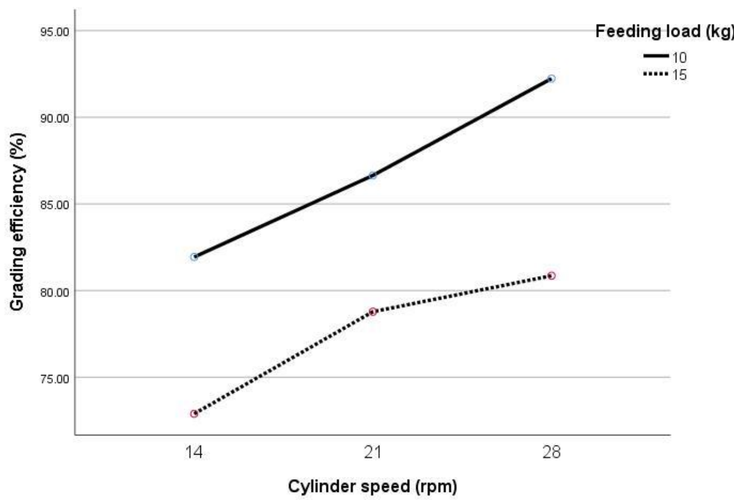


Figure 4.3. Effects of feeding load and cylinder speed on the grading efficiencies

4.4.Tuber damage

At various feeding loads, the effect of drum speed on tuber damage is shown in Figure 4.4. The proportion of tuber damage grew steadily from 1.05 percent to 3 percent as the drum speed increased from 14 rpm to 28 rpm and the feeding load increased from 10 kg to 15 kg, according to the results. By delivering a high impact on the tubers, the cylinder speed causes tuber damage. Tuber damage became more worse as the tubers crowded together due to the abundance of feeding supplies.

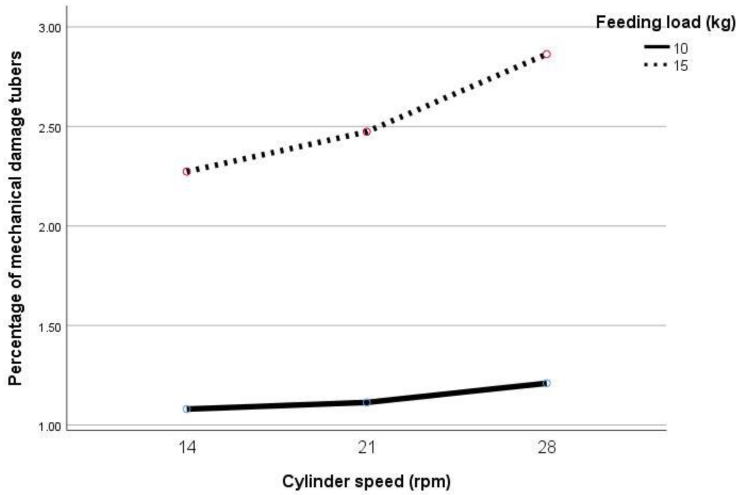


Figure 4.4.Effect of feeding load and cylinder speed on the percentage of damage tuber

4.5.Throughout put Capacity

Figure 4.5 shows that overall put capacity was related to drum speed and inversely proportional to feeding load, respectively. At the greatest cylinder speed (28rpm) and lowest feeding load (10 kg), the highest throughput capacity of 242.17 kg/hr was achieved.

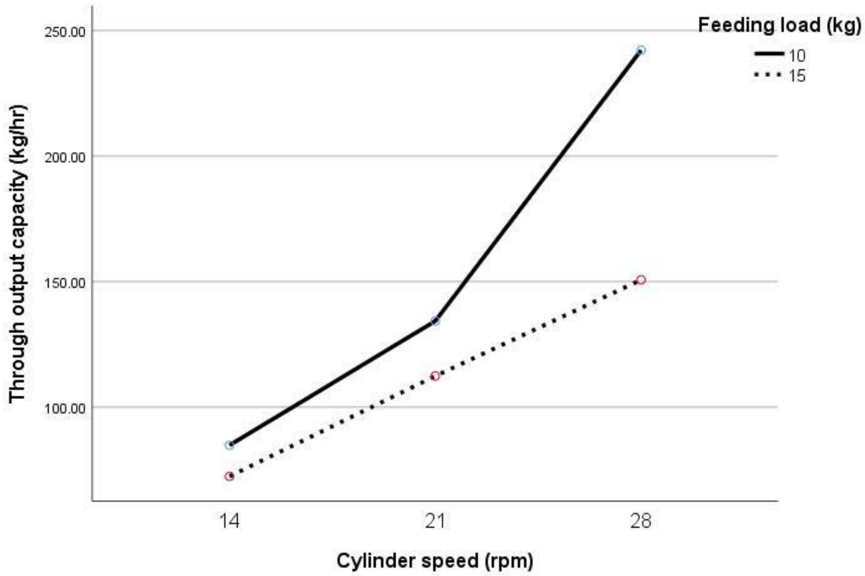


Figure 4.5.Effects of feeding load and cylinder speed on the throughout put

This is due to the fact that a small number of materials with the maximum cylinder speed can be cleaned and graded in a short amount of time, resulting in better throughput capacity. As a result, this machine can wash and grade over 242.17 kg carrot tubers per hour on average, which is better than (Ambrose & Annamalai, 2013) and the traditional approach, which takes over an hour to wash and grade 100 kg carrot tubers.

As previously stated, the average minimum required time (2.22 min), maximum washing efficiency (98.7%), maximum grading efficiency (92.23%), maximum throughput capacity (242.17 kg/hr), and optimum percentage of damage (1.21%) were all recorded at 28rpm and 10kg feeding load, which is recommended as the machine's best working condition.

4.6.Human power consumption

To estimate human power consumption, determine how long the selected subjects operate the machine continuously, and determine the appropriate load for all subjects, a comparison means test was conducted to compare the mean heart pulse rate of the subjects after one hour of washing and grading activities. For the 10kg and 15kg feeding loads, the mean cardiac rate was 106.90 and 127.0313 beats/min, respectively, which was a significant difference.

CHAPTER FIVE

5. CONCLUSION AND RECOMMENDATIONS

5.1.Conclusion

We designed, built, and tested a carrot washing and grading machine. The pedal assembly (power transmission unit), the reciprocating pump assembly, the washing and grading cylinder, and the supporting frame are the four basic components. The machine was tested with the Nantay type at three different cylinder speeds (14, 21, and 28 rpm) and two different feeding loads (10 kg and 15 kg). The results of the performance review were as follows:

1.The average amount of time it took to wash was 2.22 minutes. It tends to rise with increased drum speed and fall with increased feeding load.

2.Overall, the average washing and grading efficiency was 98.70% and 92.23 percent, respectively. Both tend to grow as the drum speed increases and decrease as the feeding load increases..

3. Lowering both the cylinder speed and the feed load reduces the percentage of damage, whereas a low feed load and a high cylinder speed reduces the time necessary to accomplish tasks.

4. The independent factors, cylinder speed and feeding load, have a substantial impact on the washing and grading efficiency of carrots in a washing and grading machine.

5. As the speed increases, the throughput capacity increases as well. The machine has a throughput capacity of 242.17 kg/hr on average.

6. The washing and grading equipment cost birr 16363.52 per unit, which is a low and affordable price for an average Ethiopian farmer.

REFERENCE

Abdul, M., Kanoje, L., & Dhandre, H. S. (2020). *3D Modeling of vegetable and fruit washing machine*. *8*(7), 3227–3229.

Adegbite, S. A., Adeyemi, S. K., Komolafe, A. O., Salami, M. O., Nwaeche, C. F., & Ogunbiyi, A. A. (2019). Design and Development of Fruit Washer. *Journal of Scientific Research and Reports*, *21*(6), 1–11. https://doi.org/10.9734/jsrr/2018/46041

Adeyefa, B. adegboyega. (2018). Forces applied to a bicycle during normal cycling *. *Journal of Biomechanics ·*, *12*, 527–541. https://doi.org/10.1016/0021-9290(79)90041-1

Adzimah, S. K., & Agbovi, H. E. (2015). Foot-Operated Device for Controlling the Flow of Water into Plumbing Fixtures. *Innovative Systems Design and Engineering*, *6*(5), 18–36.

Ahmad, T., Cawood, M., Iqbal, Q., Ariño, A., Batool, A., Sabir Tariq, R. M., Azam, M., & Akhtar, S. (2019). Phytochemicals in daucus carota and their health benefits—review article. *Foods*, *8*(424), 1–22. https://doi.org/10.3390/FOODS8090424

Alam, S., Chavan, P., & Sharma, R. (2018). Post Harvest Value Chain of Carrot--A Review. *Agricultural Engineering Today*, *42*(4), 1–11.

Alloy, A., En, T., & Tori, N. (2017). Experimental Analysis of the Behaviour of Aluminium Alloy EN 6082AW T6 at High Temperature. *Metals*, 1–15.
https://doi.org/10.3390/met7040126

Amare, D., & Endalew, W. (2016). Agricultural Mechanization : Assessment of Mechanization Impact Experiences on the Rural Population and the Implications for Ethiopian Smallholders. *Engineering and Applied Sciences*, *1*(2), 39–48. https://doi.org/10.11648/j.eas.20160102.15

Ambrose, D. C. P., & Annamalai, S. J. K. (2013). Development of a manually operated root crop washer. *African Journal of Agricultural Research*, *8*(24), 3097–3101. https://doi.org/10.5897/AJAR2013.7012

Amin, N., & Hossain, A. (2021). Development and evaluation of a mechanical carrot washing machine. *Agricultural Engineering International: CIGR Journal*, *23*(3), 111–120.

Anjali, M., & Ikhar, M. S. R. (2018a). Design and Development of Rotary Type Vegetable Cleaner. *International Journal for Scientific Research & Development*, *6*(03), 878–880.

Anjali, M., & Ikhar, M. S. R. (2018b). Design and Development of Rotary Type Vegetable Cleaner. *International Journal for Scientific Research & Development*, *6*(03), 878–880.

ASHAHI. (2016). Bearing unit. In *Sealing Technology* (Vol. 2016, Issue 4, pp. 12–13).

https://doi.org/10.1016/s1350-4789(16)30111-8

Ashiedu, F. I., Nwaoha, T. C., & Izelu, C. O. (2016). Design and development of a manual pump for agrarian communities. *International Journal of Current Research*, *8*(03), 27378–27382. http://www.journalcra.com

Bashar, Z. U., Wayayok, A., Mohd, A., & Mohd, S. (2014). Determination of some physical properties of common Malaysian rice MR219 seeds. *Austrian Journal of Crop Science*, *8*(3), 332–337.

Basińska-Zych, A., & Springer, A. (2017). The concept of preventing negative consequences of work overload based on positive interactions between work and life , and promotion of workers ' healthy behaviors. *The Central European Review of Economics and Management*, *1*(2), 45–75. https://doi.org/10.29015/cerem.408

Batara, G. B., Aguinaldo, T. G., Cuaresma, F. D., & Gavino, H. F. (2014). Design , fabrication , and performance evaluation of a small-scale barrel type potato (Solanum Tuberosum L .) WASHER. In *Conference.UPD, 11th Engineering Research and Development for Technology Conference*. Central Luzon State University, Science City of Munoz, Nueva Ecija.

Bayram, M. (2018). Determination of the sphericity of granular food materials Determination of the sphericity of granular food materials. *Journal of Food Engineering*, *68*, 385–390. https://doi.org/10.1016/j.jfoodeng.2004.06.014

Bayram, M. (2019). Sphericity Measurement Techniques for Solid Materials Sphericity Measurement Techniques for Solid Materials. *International gap agriculture & livestock congress*.

Bender, I., Edesi, L., Hiiesalu, I., Ingver, A., Kaart, T., Kaldmäe, H., Talve, T., Tamm, I., & Luik, A. (2020). Organic Carrot (Daucus carota L .) Production Has an Advantage over Conventional in Quantity as Well as in Quality. *Agronomy*, *10*(1420), 2–15. doi:10.3390/agronomy10091420

Bharath Kumar, K., Thomas, B. G., Thadhani, P. J., & Pandiyan, A. (2015). Design and analysis of connecting rod using composite materials. *International Journal of Applied Engineering Research*, *10*(65 Special Issue), 236–241.

Bhatawadekar, G., Salman, B., Chiplunkar, N., Devrukhakar, S., & Akashdeep, S. (2015). Design and Fabrication of Pedal Powered Washing Machine. *International Journal of Engineering*

Research and General Science, *3*(1), 1307–1311. www.ijergs.org

Bhattarai, D. R., Subedi, G. D., Gautam, I. P., & Chauhan, S. (2017). Posthavest Supply Chain Study of Carrot in Nepal. *International Journal of Horticulture*, *March 2018*. https://doi.org/10.5376/ijh.2017.07.0026

Biratu, W. (2020). Evaluation of Carrot (Daucus carrota) Cultivarss for Growth, Root Yield and Yield Related Traits at Adigrat and Atsbi, Eastern Zone of Tigray Region. *Journal of Natural Sciences Research*, *10*(3), 14–18. https://doi.org/10.7176/jnsr/10-3-02

Boukouvalas, C. J., Bisharat, G. I., Krokida, M. K., Boukouvalas, C. J., Bisharat, G. I., & Krokida, M. K. (2010). Structural Properties of Vegetables During Air Drying. *International Journal of Food Properties ISSN:*, *2912*.

https://doi.org/10.1080/10942910903079267

Bsen-ds, C. A. T. N. O. (2018). *Insert Bearing Units* (pp. 1–196). JTEKT corporation (www.jtekt.co.jp).

Budynas, R. G., & Nisbett, J. K. (2011). Mechanical Engineering Design. In *Shigley's Design Engineering Mechanical* (Ninth, pp. 1–5). McGraw-Hill Companies, Inc.,. www.mhhe.com

Charles, K., & Etiese, I. (2020). Development and performance evaluation of a manually operated onions grading machine. *Journal of Agriculture and Food Research*, *2*(April), 100070. https://doi.org/10.1016/j.jafr.2020.100070

Chattopadhyay, S. (2010). Selection of material, shape and manufacturing process for a connecting rod. *ASEE Annual Conference and Exposition, Conference Proceedings*, 1–12. https://doi.org/10.18260/1-2--16217

Choi, M., Han, I., Mohtashami, K., Walker, K., & Clark, G. (2014). *Design of a Small Scale Root Crop Washer*.

CSA. (2020). *the Federal Democratic Republic of Ethiopia Central Statistical Agency Report on Area and Production of Crops* (Vol. 2008, Issue April, 2020).

Dereje, A. A. (2019). *Design, Manufacturing and Performance Evaluation of Potato Grading Machine , Unpublished" MSc*. Thesis Submitted to Haramaya University.

Edinger, T. (2003). General Catalogue. *Paleoneurology*, 12–182. https://doi.org/10.1007/978-3-642-66029-0_1

El-Rahman, A., & Mosa, M. (2011). Development and Evaluation of a Simple Grading Machine for Okra Pods. *Journal of Soil Sciences and Agricultural Engineering*, *10*(4), 227–232.

https://doi.org/10.21608/jssae.2019.36737

FAO. (2018). *Crops_Carrot Production_2016*.

Field, H. L. ., & Solie, J. B. (2007). *Introduction to Agricultural engineering technology* (third). Springer Science+Business Media, LLC. https://springer.com

Frederick, D. J., Madgwick, H. A. I., & Oliver, G. R. (1982). Wood basic density and moisture content of young eucalyptus regnans grown in new zealand. *New Zealand Journal of Forestry Science*, *12*(3), 494–500.

Ganesh, M. R., Akhil, V. B., Hareesh, V. P., Harish, M. U., & Mallikarjun, C. G. (2018). Fabrication of Root Vegetable Washer. *International Journal of Information and Computing Science*, *5*(5), 70–75. http://ijics.com/gallery/13-may-666.pdf

Gautam, A., Khurana, R., Manes, G. S., & Dixit, A. K. (2016). Studies on Engineering Properties of Pelleted Carrot (Daucus carota L.) Seeds. *International Journal of Bio-Resource and Stress Management*, *7*(5), 1044–1048. https://doi.org/10.5958/0976-4038.2016.00186.X

Ghuman, R. S., Khanna, R., Singla, S., Singh, P., & Singh, H. (2014). Designing and Fabrication of Automatic Root Crop Washer. *InternatIonal Journal of Research In MechanIcal EngIneerIng & Technolog*, *5762*(2), 222–224. http://www.ijrmet.com/vol4issue2/spl2/49

Gopal, G., Kumar, L. S., Gopinath, D., & Maheshwara, U. (2016). Design and analysis of assembly of Piston , Connecting rod and Crank shaft. *International Journal of Current Engineering and Technology*, *6*(1), 235–242.

Gujar, R. A., Bhaskar, S. V, Zappalorto, M., & Berto, F. (2013). Shaft Design under Fatigue Loading By Using Modified Goodman Method. *International Journal of Engineering Research and Applications*, *3*(4), 1061–1066.

Guru, D. K., Singh, S. P., & Shukla, S. T. | S. P. | S. K. (2019). Design and Working of Pedal Operated Flour Mill. *International Journal of Trend in Scientific Research and Development*, *Volume-3*(Issue-3), 1279–1280. https://doi.org/10.31142/ijtsrd23297

Guyer, J. P. (2012). Introduction to pumping stations for water supply systems. *Continuing Education and Development*, *9*(877), 42.

Hakizimana, E., Masengesho, P., Cyusa, O., & Niyigena, M. (2020). Design and Analysis of a Pedal Operated Washing and Drying Machine. *Journal of Applied Mechanical Engineering*, *9*(:338.), 1–8.

Hussain, G., & Ali, H. S. (2020). *Development and performance evaluation of carrot*. *January*

2017. https://doi.org/10.22194/JGIASS/5.1.772

Jadoun, R. S., & Choudhary, S. K. (2016). Design & fabrication of manually driven pedal powered washing machine Design & Fabication of Manually Driven Pedal Powered Washing Machine. *Innovative Systems Design and Engineering*, *5*(6), 1–66.

Jahanbakhsh, A., Abbaspour-Gilandeh, Y., & Tarahom Mesri Gundoshmian. (2018). Determination of physical and mechanical properties of carrot in order to reduce waste during harvesting and post- harvesting. *WILEY Food Science and Nutrition*, 1–6.

Jahanbakhshi, A. (2018). Determination of physical and mechanical properties of carrot in order to reduce waste during harvesting and post- - harvesting. *Wiley Food Science and Nutrition*, *June*, 1–6. https://doi.org/10.1002/fsn3.760

Jatobá, A., Bellas, H., Marinho, A. R., Amaral, F. G., Salazar, M., Mateus, S., Lossio, C., Jatobá, A., Bellas, H., Koster, I., Bonfatti, R., Vidal, M. C. R., Carvalho, P. V. R., Rodrigues, Y. W., Medeiros, A. P., Reinert, F., Veiga, R. K., Gontijo, L. A., Federal, U., … Brito, M. (2016). Design & fabrication of a manually operated carrot washing machine for small scale farmers. In *Revista Brasileira de Ergonomia* (Vol. 9, Issue 2). https://doi.org/10.5151/cidi2017-060

Khurmi, R. S., & Gupta, J. K. (2005). *A textbook of Machine Design* (1st ed., Issue I). EURASIA PUBLISHING HOUSE (PVT.) LTD.

Koroso, A. W. G. (2016). *A Dissertation for the Degree of Doctor of Philosophy Farm Mechanization of Small Farms in Ethiopia : A Case of Cereal Crops in Hetosa District Graduate School of Seoul National University*. Seoul National University.

Krishnamurthy, M., Rakshith, K. K., Harshaa, R., & Rakesh, N. (2017). Pedal Operated Washing Machine. *International Journal of Latest Engineering Research and Applications (IJLERA) ISSN*, *02*(05), 60–66.

Kumar, P. K. K., & Sumalatha, T. (2018). Design and Analysis of Automatic Double Acting Multi Cylinder Pneumatic Reciprocating Pump. *International Journal of Innovative Technologies*, *06*(02), 491–498.

Macmillan, R. . (2010). *The Mechanics of Tractor - Implement Performance Theory and Worked Examples a textbook for students and engineers*. http://www.eprints.unimelb.edu.au.

Man, N., Kadhim, Z. R., Abd, I., Wong, K., & Seng, K. (2016). Transactions Costs Analysis of Agricultural Machinery Hiring Decision in Iraq. *Management Science and Engineering*, *10*(04), 75–86. https://doi.org/10.3968/8913

Minh, D. H., Phuong, B. Van, Binh, P. Van, & Duc, N. V. (2020). Design, development and performance evaluation of a new-type fruit vegetable washer. *Journal of Mechanical Engineering Research and Developments*, *43*(4), 265–274.

Mohammed, W., Bezu, T., Dechassa, N., Woldetsadik, K., Hailu, M., & Abebie, B. (2014). Registration of " Haramaya I " Carrot (Daucus carota L .) Variety. *East African Journal of Sciences*, *8*(1), 65–70.

Mukkawar, V. V, & Kadam, A. P. (2015). *Design , Analysis and Optimization of Connecting Rod*. *2*, 38–41.

Mushiri, T., Mugova, T. J., & Mbohwa, C. (2017). Design and Fabrication of a Pedal Powered Washing Machine. In C. Mbohwa (Ed.), *Proceedings of the International Conference on Industrial Engineering and Operations Management* (pp. 355–366). IEOM Society International.

Narayana, K. ., Kannaiah, P., & Venkata Reddy, K. (2006). *Machine Drawing* (3rd ed.). New Age International (P) Ltd. www.newagepublishers.com

Naresh, Jain, M., Narender, Rani, V., & Kumar, A. (2018). Design and development of tractor operated carrot digger. *AMA, Agricultural Mechanization in Asia, Africa and Latin America*, *49*(3), 79–85.

Nicholas, J., & Adamson, P. (2006). *Introduction to Trigonometric Functions*. University of Sydney.

Nitturkar, H. D., Kalshetti, S. M., & Nadaf, A. R. (2020). Design and Analysis of Connecting Rod using Different Materials. *International Journal of Scientific & Engineering Research*, *4*(6), 1112–1117.

NTN. (2011). Bearing units. In *NTN* (Vol. 5, Issue 4, p. 150). https://doi.org/10.1016/0041-2678(72)90056-5

Oyeleke, F. I., Olaniyan, A. M., Sunmonu, M. O., & Oyeniyi, S. K. (2014). Development of a fruit washing machine. *Journal of Agricultural Engineering and Technology (JAET)*, *22*(1), 29–36.

Rashid, T., Siddique, G., Aleem, A., & Mechanization, A. (2018). *Fabrication and performance evaluation of a tractor*. *30*(2), 329–332.

Sebastian Antony, Arjun A., Shinos T. K, & Anoop P. (2016). Design and Analysis of a Connecting Rod. *International Journal of Engineering Research And*, *V5*(10), 188–192.

https://doi.org/10.17577/ijertv5is100142

Seweh, E. A., Darko, J. O., Addo, A., Asagadunga, P. A., & Achibase, S. (2016). Design, construction and evaluation of an evaporative cooler for sweet potatoes storage. *Agricultural Engineering International: CIGR Journal, 18*(2), 435–448.

Shankar, R. (2016). Overhauling of Two Stage Reciprocating Air Compressor of a Conventional Locomotive-A Case Study. *International Research Journal of Engineering and Technology (IRJET)*, *03*(04), 653–664.

Sharma, D. ., & Mukesh, S. (2010). *Farm Machinery Design Principles and Problems I* (2nd ed.). Sunilkumarjain. www.thejainbros.com

Shigley, J. E., & Mischke, C. R. (1996). *Standard Hand book of Machine Design* (Second). McGraw-Hill Companies, Inc.

Siddique, G., Aleem, A., Hussain, G., & Raza, H. A. (2017a). Development and Performance Evaluation of Carrot. *J. Glob. Innov. Agric. Soc. Sci.*, *5*(1), 28–31. https://doi.org/10.22194/JGIASS/5.1.772

Siddique, G., Aleem, A., Hussain, G., & Raza, H. A. (2017b). *Development and performance evaluation of carrot washer*. *5*(1), 28–31.

Soyoye, B. O., Ademosun, O. C., & Agbetoye, L. A. S. (2018). Determination of some physical and mechanical properties of soybean and maize in relation to planter design. *AgricEngInt: CIGR Journal Open*, *20*(1), 81–89.

Srivastava, A. K., Goering, C. E., Rohrbach, R. P., & Buckmaster, D. R. (2006). *Engineering Principles of Agricultural Machines* (P. McCann & M. Miller (eds.); 2nd ed.). American Society of Agricultural and Biological Engineers.

Stolarczyk, J., & Janick, J. (2011). Carrot: History and Iconography. *Chronica Horticulturae*, *51*(2), 13–18. http://www.hort.purdue.edu/newcrop/pdfs/ch5102-carrot.pdf

Surbhi, S., Rc, V., Deepak, R., Hk, J., & Kk, Y. (2018). *A review : Food , chemical composition and utilization of carrot (Daucus carota L .) pomace*. *6*(3), 2921–2926.

Tabor, G., & Mohammed, Y. (2012). Mapping the Current Knowledge of Carrot Cultivation in Ethiopia Table of Contents. In *Carrot Aid, Denmark* (Issue August).

Teferra, T. (2019). *Engineering Properties of Food Materials* (3rd ed.). Elsevier B.V.

Thorat, S. D., & Kanase-Patil, A. B. (2015). Study of Performance Analysis of Reciprocating Pumps using CFD. *International Journal on Theoretical and Applied Research in Mechanical*

Engineering (IJTARME), *4*(1), 40–46.

Yadav, P. (2020). *A Review on different types of carrot and its chemical compositions. 10*(5), 32–39.

Yang, K., Guan, S., & Wang, C. (2011). *The design & calculation for hydraulic cylinder of workpiece hydraulic clamping system of a special CNC machine for guide disc*. *16*, 418–422. https://doi.org/10.1016/j.proeng.2011.08.1105

APPENDICES

Table1.1. Power rating of simple roller chain

Speed of smaller sprocket or pinion (r.p.m.)	**Power (kW)**				
	06B	08B	10B	12B	16B
100	0.25	0.64	1.18	2.01	4.83
200	0.47	1.18	2.19	3.75	8.94
300	0.61	2.70	3.15	5.43	13.06
500	1.09	2.72	5.01	8.53	20.57
700	1.48	3.66	6.71	11.63	27.73
1000	2.03	5.09	8.97	15.65	34.89
1400	2.73	6.81	11.67	18.15	38.47
1800	3.44	8.10	13.03	19.85	-
2000	3.80	8.67	13.49	20.57	-

Source: Khurmi and Gupta, 2005 and Beratung, 2018

Table 1.2. Characteristics of roller chains

ISO Chain number	***Pitch (p) mm***	***Roller diameter (d_1) mm Maximum***	***Width between inner plates (b_1) mm Maximum***	***Transverse pitch (p_1)mm***	***Breaking load (kN) Minimum***		
					Simple	Duplex	Triplex
05B	8.00	5.00	3.00	5.64	4.4	7.8	11.1
06B	9.525	6.35	5.72	10.24	8.9	16.9	24.9
08B	12.70	8.51	7.75	13.92	17.8	31.1	44.5
10B	15.875	10.16	9.65	16.59	22.2	44.5	66.7
12B	19.05	12.07	11.68	19.46	28.9	57.8	86.7
16B	25.4	15.88	17.02	31.88	42.3	84.5	126.8
20B	31.75	19.05	19.56	36.45	64.5	129	193.5
24B	38.10	25.40	25.40	48.36	97.9	195.7	293.6
28B	44.45	27.94	30.99	59.56	129	258	387
32B	50.80	29.21	30.99	68.55	169	338	507.10
40B	63.50	39.37	38.10	72.29	262.4	524.9	787.3
48 B	76.20	48.26	45.72	91.21	400.3	800.7	1201

Source :(Khurmi & Gupta, 2005) according to IS: 2403 — 1991.

Table 1.3. Dimension of standard V-belts

Types of belts	**Power ranges in kW**	**Minimize pitch diameter of pulley (D)mm**	**Top width (b)mm**	**Thickness (t)mm**	**Weight per meter length in newton**
A	0.7-3.5	75	13	8	1.06
B	2-15	125	17	11	1.89
C	7.5-75	200	22	14	3.43
D	20-150	355	32	19	5.96
E	30-350	500	38	23	-

Source :(Khurmi & Gupta, 2005) according IS: 2494-1974

Table 1.4. Classical V-Belts

A Type V-belt code and inner length,			
Belt code	Inner length Li (mm)	Belt code	Inner length Li (mm)
20.	508	46.	1168
21.	535	47	1200
22.	560	48	1220
23.	585	49.	1250
24.	610	50.	1270
25.	630	51.	1300
26.	660	52.	1320
27.	686	53.	1350

28.	710	54.	1375
29.	730	55.	1400
30.	767	56.	1422
31.	787	57.	1450
32.	813	58.	1475
33.	838	59.	1500
34.	864	60.	1525
35.	889	61.	1550
36.	914	62.	1575
37.	940	63.	1600
38.	965	64.	1625
39.	990	65.	1650
40.	1016	66.	1676
41.	1041	67.	1700
42.	1060	68.	1725
43.	1100	69.	1750
44.	1120	70	1775
45.	1143		

for DIN 2015/ISO 4184

Table 1.5. Dimension of Standard V-grooved pulleys

Types of belt	**w (mm)**	**d (mm)**	**a (mm)**	**c (mm)**	**f (mm)**	**e (mm)**	**N0. Of sheave grooves (n)**	**Groove angle (2β)**
A	11	12	3.3	8.7	10	15	6	32, 34,38
B	14	15	4.2	10.8	12.5	19	9	32,34,38
C	19	20	5.7	14.3	17	25.5	14	34, 36,38
D	27	28	8.1	19.9	24	37	14	34, 36, 38
E	32	33	9.6	23.4	29	44.5	20	-

Source: Khurmi and Gupta, 2005 according to IS: 2494-1974.

Table 1.6. Relation between equivalent length and actual length

S.No.	**End Conditions**	**Relation between equivalent length (L) and actual length (l)**
1.	Both ends hinged	$L = l$
2.	Both ends fixed	$L = \frac{l}{2}$
3.	One end fixed and the other end hinged	$L = \frac{l}{\sqrt{2}}$
4.	One end fixed and the other end free	$L = 2l$

Source: Khurmi and Gupta, 2005

Table 1.7. Coefficient of friction between belt and pulley

Speed of smaller sprocket or pinion (r.p.m.)	**Power (kW)**				
	06B	08B	10B	12B	16B
100	0.25	0.64	1.18	2.01	4.83
200	0.47	1.18	2.19	3.75	8.94
300	0.61	2.70	3.15	5.43	13.06
500	1.09	2.72	5.01	8.53	20.57
700	1.48	3.66	6.71	11.63	27.73
1000	2.03	5.09	8.97	15.65	34.89
1400	2.73	6.81	11.67	18.15	38.47
1800	3.44	8.10	13.03	19.85	-
2000	3.80	8.67	13.49	20.57	-

Table 1.8. The radial load factor and axial or thrust load factor

Type of bearing	**Specifications**	$\frac{W_A}{W_R} < e$	$\frac{W_A}{W_R} > e$	e

		X		X	Y	
Deep groove ball bearing	$\frac{W_A}{C_0} = 0.025$ $= 0.04$ $= 0.07$ $= 0.13$ $= 0.25$ $= 0.50$	1	0	0.56	2.0 1.8 1.6 1.4 1.2 1.0	0.22 0.24 0.27 0.31 0.37 0.44
Angular contact ball bearings	Single row Two rows in tandem Two rows back to back Double row	1	0 0 0.55 0.73	0.35 0.35 0.57 0.62	0.57 0.57 0.93 1.17	1.14 1.14 1.14 0.86
Self-aligning bearings	Light series: for bores $10 - 20\ mm$ $25 - 35$ $40 - 45$ $50 - 65$ $70 - 100$ $105 - 110$	1	 1.3 1.7 2.0 2.3 2.4 2.3	6.5	 2.0 2.6 3.1 3.5 3.8 3.5	 0.50 0.37 0.31 0.28 0.26 0.28
	Medium series: for bores $12\ mm$ $15 - 20$ $25 - 50$ $55 - 90$	1	 1.0 1.2 1.5 1.6	0.65	 1.6 1.9 2.3 2.5	 0.63 0.52 0.43 0.39
Spherical roller bearings	$For\ bores:$ $25 - 35\ mm$ $40 - 45$ $50 - 100$ $100 - 200$	1	 2.1 2.5 2.9 2.6	0.67	 3.1 3.7 4.4 3.9	 0.32 0.27 0.23 0.26
Taper roller bearings	**For bores** $\mathbf{30 - 40\ mm}$ $\mathbf{45 - 110}$ $\mathbf{120 - 150}$	1	0	0.4	1.60 1.45 1.35	0.37 0.44 0.41

Source :(Khurmi & Gupta, 2005)

Table 1.9. Values of service factor

S.No.	**Type of service**	**Service factor (KS) for radial ball bearings**
1	Uniform and steady load	1.0
1.	Light shock load	1.5
2.	Moderate shock load	2.0
3.	Heavy shock load	2.5
4.	Extreme shock load	3.0

Source :Khurmi and Gupta, 2005

Table 1.10. Life of bearings for various types of machines.

S. No.	***Application of bearing***	***Life of bearing, in hours***
1.	Instruments and apparatus that are rarely used	
	(a) Demonstration apparatus, the mechanism for operating sliding doors	500
	(b) Aircraft engines	1000–2000
2.	Machines used for short periods or intermittently and whose breakdown would not have serious consequences *e.g.* hand tools, lifting tackle in workshops, and operated machines, agricultural machines, cranes in erecting shops, domestic machines	4000–8000

3.	Machines working intermittently whose breakdown would have serious consequences *e.g.* auxiliary machinery in power stations, conveyor plant for flow production, lifts, cranes for piece goods, machine tools used frequently.	8000-12000
4.	Machines working 8 hours per day and not always fully utilized *e.g.* stationary electric motors, general-purpose gear units.	12000-20,000
5.	Machines working 8 hours per day and fully utilized *e.g.* machines for the engineering industry, cranes for bulk goods, ventilating fans, countershafts.	20,000-30,00
6.	Machines working 24 hours per day *e.g.* separators, compressors, pumps, mine hoists, naval vessels.	40,000-60,000
7.	Machines required to work with a high degree of reliability 24 hours per day *e.g.* pulp and paper making machinery, public power plants, mine-pumps, waterworks.	100 000 – 200 000

Source : Bsen-ds, (2018) and Meissner, (1997)

Table 1.11. Basic static and dynamic capacities of various types of a radial ball

Bearing No.	**Basic capacities in kN**							
	Single row deep groove ball bearing		Single row angular contact ball bearing		Double row angular contact ball bearing		Self-aligning ball bearing	
(1)	Static (C0) (2)	Dynamic (C0) (3)	Static (C0) (4)	Dynamic (C) (5)	Static (C0) (6)	Dynamic (C) (7)	Static (C) (8)	Dynamic (C0) (9)
200	2.24	46.3	—	—	4.55	7.35	1.80	5.70
300	3.60		—	—	—	—	—	—
201	3	5.4	—	—	5.6	8.3	2.0	5.85
301	4.3	7.65	—	—	—	—	3.0	9.15
202	3.55	6.10	3.75	6.30	5.6	8.3	2.16	6
302	5.20	8.80	—	—	9.3	14	3.35	9.3
203	4.4	7.5	4.75	7.8	8.15	11.6	2.8	7.65
303	6.3	10.6	7.2	11.6	12.9	19.3	4.15	11.2
403	11	18	—	—	—	—	—	—
204	6.55	10	6.55	10.4	11	16	3.9	9.8
304	7.65	12.5	8.3	13.7	14	19.3	5.5	14
404	15.6	24	—	—	—	—	—	—
205	7.1	11	7.8	11.6	13.7	17.3	4.25	9.8
305	10.4	16.6	12.5	19.3	20	26.5	7.65	19
405	19	28	—	—	—	—	—	—
206	10	15.3	11.2	16	20.4	25	5.6	12
306	14.6	22	17	24.5	27.5	35.5	10.2	24.5
406	23.2	33.5	—	—	—	—	—	—
207	13.7	20	15.3	21.2	28	34	8	17
307	17.6	26	20.4	28.5	36	45	13.2	30.5
407	30.5	43	—	—	—	—	—	—
208	16	22.8	19	25	32.5	39	9.15	17.6
308	22	32	25.5	35.5	45.5	55	16	35.5
408	37.5	50	—	—	—	—	—	—
209	18.3	25.5	21.6	28	37.5	41.5	10.2	18
309	30	41.5	34	45.5	56	67	19.6	42.5
409	44	60	—	—	—	—	—	—
210	21.2	27.5	23.6	29	43	47.5	10.8	18
310	35.5	48	40.5	53	73.5	81.5	24	50
410	50	68	—	—	—	—	—	—

Table 1.12..Materials cost estimation

Parts	**Material Used**	**Specification**	**Quantity**	**Cost (birr)**

Drum	Waterproof wood	3000×200×2mm	1	600
Pump Cylinder	Aluminum	Un molded chips	5kg	500
Connecting Rod	Flat iron	400 ×40 ×2mm	1	100
Pump Rods	Aluminum	Un molded chips	3kg	300
Crank	Flat Iron	200×20×2mm	1	100
Pedal	Cast Iron	320mm dia	1	150
Shafts	Round mild steel	20mm dia × 6000mm	1	1500
Handle	Galvanized pipe	18mm Dia ×1000mm length	1	50
Bicycle sprocket	Cast Iron	85mm dia with 20 teeth	1	80
Water pipe	Plastic	1inch dia×3000mm	1	40
Chain	Cast iron	06B with 1300 mm length	1	100
Standard finished items	Roller Chain, Sprockets, Bolts and Nuts, and Washers			200
Pulleys	Cast Iron	75mm, 140mm, 75mm, and	4	1200
Belts	Rubber	28 and 29 code numbers	2	200
Electrode	2.5	300	1 packet	300.00
Paint		150	2 liter	300.00
Total				**5720**

Table 1.13. Machine and labor cost.

Machine	Working hour	Machine Cost/hr	Cost (birr)	Labor Cost/hr	Working hour	Cost (birr)	Total price(birr)
Welding	15	6	90	6.25	15	93.75	183.75
Grander	20	1	20	6.25	20	125	145
Lathe	46	15	690	6.25	46	287.5	977.5
Drilling	10	3	30	6.25	10	62.5	92.5
Molding	20	10	200	6.25	20	125	325
Powerhack saw	4	16	64	6.25	4	100	164
Total	**107**	**35**	**1094**	**6.25**	**107**	**793.75**	**1887.75**

Table 1.14. Cost summery.

No	Cost variables	Cost (EB)
A	Raw materials cost	5720
B	Materials wastage (2.5%)	143
C	Machine cost	1094
D	Labor cost	793.75
E	Overhead cost 5% (C+D)	94.39
F	Profit 10% (A+B+C+D+E)	673.24
G	Selling tax 15% (A+B+C+D+E)	7845.14
Selling price (EB)		16363.52

COST ANALYSIS

Assumption Made

Some assumption was made based on the quality and durability of the materials used for the development of the carrot washing and grading machine.

- The expected life of the machine = 5years
- Expected operational hour per day (daily use) = 8 hr.

- Expected operational hour per year (annual use) = 2400 hr.
- Salvage value =10% of total cost of the machine=1636.35
- Interest rate 15%

Operational cost of machine

Fixed cost of peeler

a) Depreciation (Dp),

$$\mathrm{Dp} = \frac{\text{SP-SV}}{\text{EL}} = \frac{\text{16363.52ETB-1636.35ETB}}{5\times 2400\text{hr}} = 1.23 ETB/hr$$

Where: Dp = depreciation cost, SP = selling price of peeler, SV = salvage values (10% of selling price), and EL=Economic life

b) Interest on capital (IC),

$$\mathrm{IC} = \frac{\text{SP+SV}}{2\text{U}} \times I\% = \frac{\text{16363.52ETB+1636.35ETB}}{2\times 2400\text{hr}} \times \frac{15}{100} = 0.56\text{EB}/hr$$

Where: IC = interest on capital, SP = selling price, SV = salvage value, U = annual use, I = interest

The total cost is the sum of depreciation cost and interest on capital

Total fixed cost = Dp + IC= 1.23ETB/hr + 0.56EB/hr Total fixed cost = 1.79EB/hr

Variable cost

Labor cost or wages

Labor wages of 150ETB per day, 8hr in a day.

$$\text{LW}=\frac{\text{DLW}}{\text{DWH}}, \text{LW}= \frac{\text{150EB}}{\text{8hr}} = 18.75\frac{ETB}{hr}$$

$Where: LW = labor\ wages, DLW = daily\ labor\ wages,$ and $DWH = daily\ working\ hour$

Total cost = Total fixed cost + Total variable cost

Total cost = 1.79EB/hr+18.75 ETB/hr= 20.54ETB/hr

Taking the through output capacity of 120Kghr^{-1}, the cost to wash and grade 1kg can be calculated as

$$\text{Cost of washing and grading 1kg of carrot tubers by machine} = \frac{\text{total cost per hour}}{\text{machine capacity o per hour}} EB$$

Cost of washing and grading1 kg of carrot tubers = $\frac{\text{20.54ETB/hr}}{\text{108kg/hr}}$

Cost of washing and 1 kg of carrot tubers =0.19ETB/kg

Benefit from one kg carrot tuber.

The average manual washing and grading capacity was also determined for two people; the average manual washing and grading capacity of carrot tubers was recorded to be 40Kghr^{-1}. Taking for a single person performing at a maximum washing and grading capacity, the average throughput of labor was 20kghr^{-1} and the labor cost of 150EB in the day (8 hours) per person was 18.75EBhr^{-1}

- Cost of manual washing and grading = 0.94ETB/Kg

Cost of Washing by machine = 0.19ETB/kg

washing and grading 1kg of carrot tuber in the traditional method is estimated to be 0.94ETB/Kg while an estimated cost to wash and grade the same amount of tuber was 0.19ETB/kg using a developed g machine.

Cost-Benefit Ratio = 1:4.95

ANNEXES

Pictures showing some procedures for determination of Physical propertiesof carrot tuber

Pictures showing some procedures for construction of machine

Pictures showing some procedures for performance evaluation