Design, Fabrication and Performance Evaluation of a Two Wheel Tractor Operated Onion Harvester

Getu Tesfaye Degabas

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 in Agricultural Machinery Engineering

Office of Graduate Studies
Adama Science and Technology University

August, 2023
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APPROVAL SHEET

I, Dr. Amana Wako entitled "Design, Fabrication and Performance Evaluation of a Two Wheel Tractor Operated Onion Harvester" developed by Getu Tesfaye Degabas, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

Advisor: Dr. Amana Wako Signature_______________ Date _________

We, the undersigned, members of the Board of Examiners of the thesis by Getu Tesfaye Degabas have read and evaluated the thesis entitled "Design, Fabrication and Performance Evaluation of a Two Wheel Tractor Operated Onion Harvester" and examined the candidate during the open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Agricultural Machinery Engineering.

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DECLARATION

I hereby declare that this Master Thesis entitled by "Design, Fabrication and Performance Evaluation of a Two Wheel Tractor operated Onion Harvester" 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.

Getu Tesfaye Degabas   Signature ________________________ Date ________________

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ACRONOMYS AND ABREVIATIONS

AAERC	Asela Agricultural Engineering Research Center
ANOVA	Analysis of variance
CSA	Central Statistical Agency
CV	Coefficient of variance
Dam %	Damage percentage
Eqn.	Equation
FAO	Food and Agricultural Organization
FAOSTAT	Food and Agricultural Organization Statistics
Fc	Fuel consumption
F_{ca}	Field capacity
KATC	Kulumsa Agricultural Training Center
M.a.s.l	Meter above sea level
SE	Standard Error
USDA	United States Department of Agriculture
2WT	A two wheel tractor
η_{dig}	Digging Efficiency
$\%_{miss}$	Missing percentage

ABSTRACT

Onion` is a root crop that had been harvested manually in Ethiopia which was tedious, consumes time, labor-intensive, and costly operation. A two-wheel tractor operated onion harvester has not designed specifically in Ethiopia for effective harvesting operation and the harvester designed by foreigners were applied on raised beds only, and scatters onion bulbs on the field because of the designed machines have not cutting disc and collector. Therefore, this study was focused on the design, development, and performance evaluation of a two-wheel tractor-operated onion harvester. The physical properties of onion and soil relevant to the design of the digger were studied and each component of the machine was designed accordingly. The experimental design used was laid in a split-plot design. The design of the harvester was based on the use of locally available materials to reduce costs. A prototype of the harvester was constructed and tested in the field to evaluate its performance. The performance of the harvester was evaluated based on its efficiency, productivity, and cost-effectiveness. The maximum and minimum digging efficiency of the machine was 97.03% at 1.39km/hr forward speed and 10cm depth of operation and 84.60% at 4.15km/hr forward speed and 5cm depth of operation respectively. The maximum mean damage percentage, missing percentage, drawbar pull, fuel consumption, wheel slip, field efficiency, and field capacity were 4.56%, 11.36%, 2935.4N, 15.51 l/ha, 20.82%, 90.84% and 0.157 ha/h respectively. Based on the results, the machine was found to be acceptable for use by small-scale farmers.

Keywords: *Design, development, evaluation, forward speed, depth of operation, onion tuber*

CHAPTER ONE

INTRODUCTION

1.1. Background and Justification

Onion (Allium Cepa) is a type of vegetable that was believed to have originated in southwestern Asia and it has since been spread around the world (Brewster, 2008 and Kitila et al., 2022). China, India, USA, Turkey, Japan, Spain, Brazil, Poland, and Egypt were the most active producers of onions (FAO, 2018). Ethiopia was one of the top three countries that produce the onions in Africa after Egypt and South Africa contributing 2.7% of the world's production between 2000 and 2011 (FAOSTAT, 2019). Onion is widely used for its flavor and nutritional value and is eaten daily in many dishes regardless of religion, ethnicity, or culture (Bagali et al., 2012). It is also believed to have some medicinal properties.

Onion is an important and profitable crop that was grown by both small-scale's farmers and larger businesses in Ethiopia for both local consumption and exportation (Aklilu, 1994 and Muluneh et al., 2019). It was the second-most productive of all vegetable crops next to tomatoes, which have been grown mostly in the upper Awash and lake Ziway areas of the country's central rift valley (Bossie, 2009, Muluneh et al., 2019). In Ethiopia, 28,185 hectares of land were used to produce onions with 262,478 tons of annual onion bulb production with a yield of 9.1 tons per hectare (CSA, 2019). In 2017- 2018, the Oromia region alone produced 13,669.5 tons of onions per hectare, resulting in 1,033,485.45 tons of bulbs with an average yield of 7.56 tons per hectare (Demis Fikre, 2021). In the East Shoa Zone, onion production was particularly high in the towns of Meki and Ziway with 135,600 tons and 34,766 tons respectively each year (Meki and Ziway agricultural office, 2014 and Habtamu, 2017).

Harvesting is an important operation in onion cultivation. It is the process of cutting and gathering the mature onion crop from a field. Root crop harvesting requires manually digging up the edible part of the crop beneath the soil's surface. Manual harvesting is the most common practice in developing countries, where the onions are pulled out of the ground by a hand shovel and can take a lot of time and manpower. One of the main reasons for low productivity in agricultural farming was a lack of available power and a low level of farm mechanization. This means that the equipment used on farms is not modern or advanced, and they need to be done by hand without any help from machines. For example, harvesting onions typically requires manual labor like using hand shovels, which consumes time and is often costly. Manual harvesting is not only slow, but it also causes very high labor requirements and losses. Studies have also shown that manual labor can take 403 man-hours per hectare as opposed to 390 man-hours for partially mechanized harvesting operations (N.P. Laryushin, A.M. Laryushin, 2009). In Ethiopia, farmers still use traditional methods of harvesting such as using hand hoes or pulling the onions from the ground by hand. This is labor intensive and requires a large workforce. Traditional harvesting methods, such as those used to pick onions, have significant drawbacks, including high labor costs (Anonymous, 2018a).

Mechanical harvesting is an option used by farmers in more economically developed countries, especially on large-scale farms. Machines are typically used to harvest onions quickly, efficiently, and with less labor. Machines also help to get the harvest done on time, reducing losses and extending the storage life of the onions. An onion harvester is used as an automated process that begins with loosening the soil and ends with removing the onion together with the leaves above the ground (Gujar et al., 2019). Mechanical harvesting is beneficial to the farmer because it is much faster and requires less labor, leading to greater profit. Additionally, mechanical harvesting is frequently used because it allows the farmer to harvest the onions in a short period, thus reducing the risk of losses due to deterioration during storage and giving the onion a longer shelf life (Srivastava et al., 2001). Although similar machinery has been developed for potatoes, the biometric properties of onions are very different from potatoes.

Smallholder farmer’s farmland in Ethiopia is often small, hilly and split up into fragments. A two-wheel tractor is a type of tractor that is specifically designed to be used on small and medium-sized farms and referred to as a power tiller, walking tractor, hand tractor, or garden tractor which was mainly used for rotary tilling and other tasks on small and medium farm (Laike and Bisrat, 2017). These are multipurpose machines designed for tasks on the farm and they use less fuel, need less labor and require less maintenance which makes them ideal for small farms. They are smaller than other types of tractors, so they can be an affordable option, especially for small farms. They are also more productive than traditional farming tools pulled by animals, and they require less time to maintain compared to other types of tractors. This makes it easier for individual farmer to use, as they have more control over their work and can use the latest farming technology (Bill, 1999). There are organizations known as service providers and farmer unions that offer two-wheel tractor services to their members and non-members, allowing them to make money from the machines in Ethiopia. Similarly, corporate unions now have expanded their businesses and started offering 2WT-based services to their members and non-members. A 2WT is important because in Ethiopia about 60% of households which was 14.7 million households (CSA, 2012), operate on less than one hectare of land and only 1% on more than four hectares. Furthermore, the shortage of farm power and its inefficient use of it lead to low crop yields due to untimely planting, weeding, and harvesting (Sims and Kienzle, 2006). Therefore, onions are a very important vegetable crop grown on small scales in Ethiopia (Aklilu et al, 2015), and are easier to harvest using two-wheel tractors.

A 2WT-operated onion harvester which was designed and developed has the following main components such as cutting disc, digging blade, conveyor and separator unit, depth control wheel, collector, and power transmission (gear, pulley, chain, and belt). The onion harvester is attached at the back end of a 2WT from where power is supplied to it through a pulley and belt. The machine works its task starting from digging the onion bulbs with soil by using a digging blade and in case depth of operation is controlled by a depth control wheel, then the soil-onion tuber mixture goes to the conveyor and separation unit and the soil is removed, then the cleaned onion tubers were drawn on the collector. Finally, as a 2WT goes forward the collector puts the onion tuber on narrow strips which is easy for the collection of tubers. This research was unique because previously designed two-wheel-tractor operated onion harvesters did not have a cutting disc, collector, or specifically designed two-wheel-tractor operated onion harvester in Ethiopia. Therefore, this study was aimed to design, develop, and evaluate the performance of a 2WT-operated onion harvester that is affordable for Ethiopia's small-scale farmers and the designed harvester would save time and labor as well as reduce physical effort for farmers during harvesting.

1.2. Statement of the Problem

Harvesting of a root crop entails digging up the tuber and retrieving the edible portion that is buried beneath the soil. Onion is one of the root crops. Onion tubers can be harvested using one of two techniques. The first is manual harvesting which is done by hand hoe, while the second is mechanical harvesting which is done using equipment like machine. Manual harvesting requires a lot of manpower, tiresome and takes more time. Additionally, the posture of laborers is mostly leaning forward while harvesting, which renders them ineffective and adds drudgery to their tasks. According to Vatsa et al., (1996) found that labor requirements for picking tubers digging manually with a spade or hand hoe were 403 man-h/ha. This implies that it is tedious, consumes time, increases drudgery, and requires more manpower which increases the cost of operation. Onion is a perishable crop and time-dependent which causes losses of productivity if not harvested at recommended maturity stage. There is usually a lack of manpower available because many people move to cities and towns in our country Ethiopia. That is why a two-wheel tractor-operated onion harvester would be designed to reduce the aforementioned problems. Although there was a two-wheel tractor-operated onion harvester designed by foreigners, the design of the cutting disc, collector and transmission system was not ideal and could not be applied for flat beds and windrowing onion bulbs in narrow strips. Therefore, a new design of onion harvester was needed to solve these problems and which is affordable for small-scale farmers of Ethiopia.

1.3. Objective of the Study

1.3.1. General Objective

The general objective of the study was to design, fabricate and evaluate the performance of a two-wheel tractor-operated onion harvester.

1.3.2. Specific Objective

- To design components of a two wheel tractor operated onion harvester
- To construct prototype of a two wheel tractor operated onion harvester
- To evaluate the performance of a two wheel tractor operated onion harvester

1.4. Significance of the Study

Onion is a very important vegetable that is used as a medicine and a spice in Ethiopia. Harvesting of onion should have to be done carefully to keep the quality of the onion and yield. Manual harvesting is the typical method used, but it is a very laborious and time-consuming operation. A better option is mechanical harvesting which is faster, requires less effort, and reduces the number of laborers needed. The issue was that these machines were often expensive and inaccessible to small-scale farmers. Therefore, this study was needed to design and evaluate a new, low-cost machine that was small and more suitable to these farmers that would reduce the amount of labor needed and make harvesting more efficient.

1.5. Scope of the Study

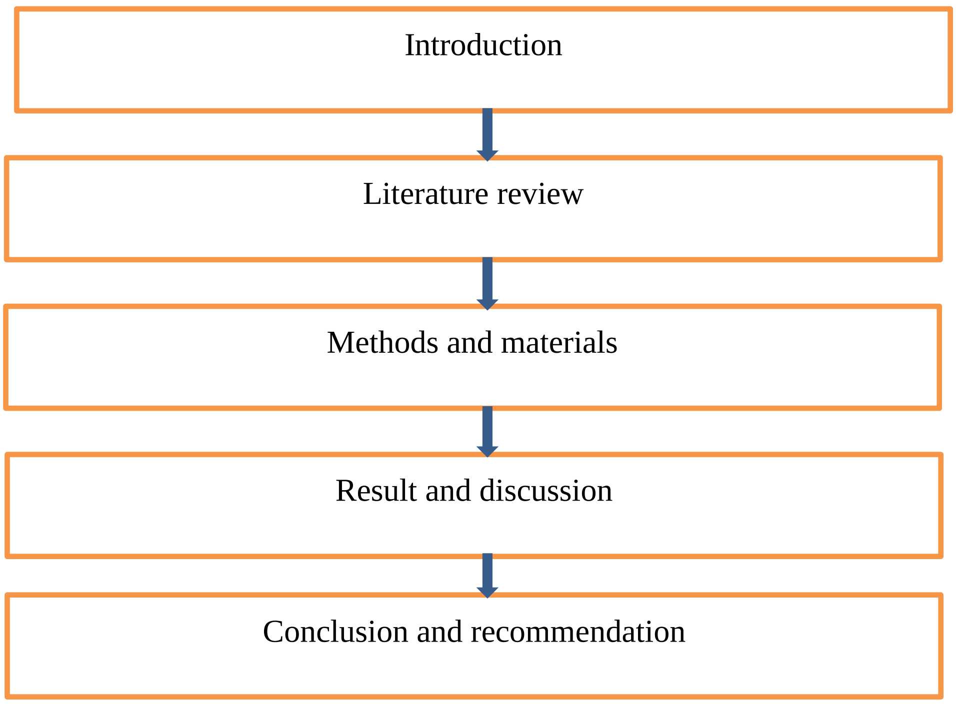
The design, development, and performance evaluation of a 2WT-operated onion harvester was the only focus of this study. During designing, the developed onion harvester was used available local materials and only operated using a two-wheel-tractor. The soil parameters including soil moisture content (% d.b.), and bulk density (kg/m^3) would be taken into consideration when developing onion harvesters. A basic performance test of the onion harvester would be conducted in the field after the prototype's construction was complete. The performance evaluation was tested using parameters such as, percentage of damage, missing percentage, digging efficiency, fuel consumption (l/hr), wheel slippage,(%), drawbar pull (N), field capacity (ha/hr), field efficiency (%).

1.6. Limitation of the Study

The following was mentioned regarding the study's limitations. These include sample size restrictions, which imply that the study was focused on a single region and a single variety of onion. Additionally, the designed machine could only be drawn by a 2WT, not a four-wheel tractor. The other drawback was that the onion was planted in the rain fed season, which excluded irrigation farming. Particularly for female small-scale farmers, the ergonomic presumptions have not been taken into account.

1.7. Organization of the Study

The study has the following organizations.



CHAPTER TWO
LITERATURE REVIEW

Different researchers have previously studied how onions can be harvested in various locations around the world. These reviews include timeliness of onion harvesting, physical properties of onion, soil parameters needed during designing of the machine, machine parameters, development of onion harvester, and performance evaluation of designed onion harvester literatures.

2.1. Timeliness of Onion Harvesting

Harvesting onion crops was a very important part of the onion production process. When at least half of the tops have fallen over and begun to dry, onions are ready to be picked (Jauron, 2009). Mature onions should be harvested before heavy rain or prolonged periods of moisture. Rain on mature onions promotes the growth of rot-causing fungus and bacteria and makes human and automated harvesting procedures more difficult. For these reasons, harvesting should be done as quickly as possible in dry climatic conditions to avoid poor weather and to prevent damage to onions (Mansour, 1985). Depending on variety, harvesting can be done 90 – 150 days after transplanting. Early harvesting of onions reduces the yield, whereas if harvested late, this can cause diseases caused by root-dwelling organisms (Maw and Mullinix, 2005). Before harvesting onion, it must complete its physiological maturity and hence farmers and extension agents should strictly obey it. Accordingly, harvesting usually takes place when 80% of the onion plants have soft necks or dry and the foliage is starting to collapse or fallen over.

Bulb size and firmness are required at harvest, along with full, closed necks and scales. According to Wright and Grant (1997), the features of an onion's quality and storage life can be significantly impacted by the onion's harvesting and curing processes. According to S.H. Nile et al. (2016), the digging time of the onion bulbs from the field was usually determined by the percentage of fallen leaves. According to Maw et al. (1998), a key element in determining the storage life of onion bulbs is harvesting crops of onions at the right stage of maturity. The onion crop must be harvested at the ideal maturity stage within a predetermined time frame (Khura et al., 2010). A successful harvesting needs to be completed within a specific window to maximize the onion bulbs’ quality and storage life.

2.2. Physical Properties of Onion

Physical characteristics including size, shape, surface area, volume, density, porosity, color, and appearance play a vital role in the design of certain machinery. Mohsenin (1986) conducted research which suggested that physical properties when designing systems for harvesting, lifting, sorting, transporting, transforming, and packaging agricultural product. According to Bamgboye and Adebayo (2012), shape, size, mass, bulk density, true density, porosity, and static friction coefficient are the primary moisture-dependent physical properties of biological materials that are susceptible to change in a short amount of time. The engineering properties of agricultural materials and food products are important in many grains/seeds, vegetables, and food materials handling and processing operations. Rapid and accurate determinations of physical properties are needed in processing agricultural materials (Krishnakumar, 2021).

The design parameters of the digger are influenced by the crop's physical and biometric characteristics (Khura et al., 2010). Ashok P. (2003) evaluated the distinctive biometric characteristics, which comprised mean values for the horizontal and vertical diameters of the bulb and neck thickness. He discovered that the average horizontal diameter, average vertical diameter of the bulb, and average neck thickness ranged from 61.80 to 74.80 mm, 44.20 to 64.00 mm, and 9.60 to 16.20 mm, respectively, at the time of harvest. The significance of these factors is related to the analysis of the product's behavior during a process or the design of a specific machine (Singhal and Samuel, 2003).

According to Ashok P. (2003), the distinctive biometric characteristics for ten unique types of onion bulbs were evaluated and the result revealed that, the mean values for the horizontal and vertical diameters of the bulb and neck thickness ranged from 61.80 to 74.80 mm, 44.20 to 64.00 mm, and 9.60 to 16.20 mm, respectively, at the time of harvest. The significance of these factors is related to the analysis of the product's behavior during a process or the design of a specific machine (Singhal and Samuel, 2003).

2.3. Soil Parameters

Both bulk density and soil moisture content have a significant impact on a tool's ability to operate with soil. The bulk density of the soil fully determined the amount of force needed to draw a tool that interacted with the soil. According to Godzo and Adama (2003), the cone index of a soil was typically expressed in kpa and depended on the bulk density and water content of the soil. USDA (1990), states that soil moisture content affects penetration resistance, with more water in the soil resulting in greater resistance. As a result, when measuring cone index, the soil's water content should be taken into consideration. Ramachandran and Jesudas (2017) calculated the soil's cone index on wet terrain using a digital cone penetrometer. At the beginning of the pudding operation by tractor, soil penetration resistance was evaluated at four different locations beneath wet land. With an increase in soil moisture content, the data of cone index values fell. The strength of the soil was measured using a cone penetrometer with a cone base area of 7.80 cm^2.

2.4. Machine Parameters

2.4.1. Blade Geometry

The size and shape of the digging blade are part of its geometry. The geometry of the blade has a significant impact on the root crop harvester's effectiveness. As a result, choosing and fine-tuning the blade shape is crucial for maximizing root crop harvester efficiency.

Table 2.1. Research works on root crop harvesters with different digging blades

S. No.	Digging blade		Power source	Harvesting efficiency, Per cent	Crop	Authors
	Shape	Size, mm				
1.	Crescent	457	Bullock	80.00	Groundnut	Narayana Rao, 1974
2.	V-shaped	200	Tractor	98.80	Sugar beet	Srivastava and Yadhav, 1978
3.	V-shaped	900	Tractor	97.00	Cassava	Odigboh and Ahmed, 1982
4.	Trapezoidal	250	Power Tiller	-	Potato	Misener, 1982
5.	Share type	600	Bullock	86.24	Groundnut	Awadhawal et al., 1995
6.	V-shaped	700	Power Tiller	93.20	Onion	Jadhav et al., 1995
7.	Straight	825	Tractor	95.00	Turmeric	Murugesan and Tajuddin,1995
8.	Straight	1800	Tractor	99.89	Groundnut	Duraisamy, 1997
9.	V-shaped	900	Tractor	96.00	Onion	Sandeep and Sudhama, 1998
10.	Straight	600	Tractor	94.00	Turmeric	Anon., 2002.
11.	V-shaped	500	Tractor	93.00	Potato and peanut	Ibrahim et al., 2008
12.	V- shaped	440	Bullock Drawn	94.00	Turmeric	Munde et al., 2009
13.	V- shaped	1000	Tractor	93.64	Sweet potato	**A**khir et al., 2014
14.	Angular tynes	600	Mini-tiller	87.00	Coleus	Younus and Jayan, 2016
15.	Sweep type	450	Bullock Drawn	83.00 &85.00	Turmeric and ginger	Zate et al., 2018
16.	Inverted V-type	1000	Tractor	81.80	Ginger	
17.	V-shaped	600	Mini-Tractor	89.00	Coleus	Md Favazil et al., 2019

(Source: Basavaraj, 2020)

2.4.2. Rake Angle of the Blade

The rake angle of the blade (θ) is the angle between the digging direction and a line normal to the blade edge. Since the blade rake angle impacts the energy consumption when cutting and digging the soil, it was suggested that the best values of rake angle for root crops be between 10 and 30^{o} (Yumnam and Pratap 1991). A greater angle of 30^{o} may require more traction force for tuber crops while a smaller angle of 10^{o} may not penetrate sufficiently in medium soils to dig out the tuber. Hence, the present investigation was conducted at three levels of rake angles of 15, 20 and 25^{o}. According to Gujar et al., (2020), blade angle used was 17^{o}. Khurana et al., (2013) designed a root crop harvester for different crops such as onion, garlic, carrot, and potato. The harvester had a 1.14 m wide by 16 mm thick digging blade that was mounted on a frame at a 20^{o} angle to the horizontal.

2.4.3. Operating Speed

According to Mehta and Yadav (2015), operating speed should be range between 2.0 and 4.0 km/h. High operating speed will increase the field's capacity, but they should also consider bulb damage. The accelerating component of the soil particles causes the operating speed to have an impact on the tillage tool's draft. As a result, the study's operating speeds of 2.5, 3.25, and 4.0 km/h were chosen (Naik et al., 2022). According to Sungha Hong et al. (2014), the harvester's performance was evaluated using operating speeds of 5.0 cm/s, 11.4 cm/s, and 15.8 cm/s. Omar et al. (2018) designed and evaluated an onion harvester's performance under four different forward speeds of 0.720, 0.837, 0.947, and 1.125 km/h. Ibrahim et al., (2008) has designed a multipurpose digger and its performance was evaluated under three forward speeds of 1.4, 1.8, and 2.3 km/h for peanuts, and 1.8, 2 and 2.6 km/h for potatoes.

2.4.4. Depth of Operation

According to Omar et al., (2018) the performance of onion harvester developed was evaluated under four digging depths (4, 6, 8 and 10 cm). According to Budhale et al., (2019), the depth of operation was depending on the depth of the onion in the soil bed. The depth of harvesting is in the range of 7–10 cm when taking into account the average biometric characteristics of onions.

2.4.5. Slope of Conveyor

According to research findings, the elevator's inclination angle shouldn't exceed 15^{o} (Azizi et al., 2014; Younus and Jayan, 2015). According to Kheiry et al. (2018), an increase in conveyor inclination from 15-20^{o} for chain conveyor type potato and onion digger resulted in a significant and consistent rise in the percentage of tubers lifted. According to Hevko et al. (2016), the semi-trailing root crop conveyor harvester has a running speed of 1.2 m/s and a slope angle of 15^{o}.

2.5. Development of Onion Harvesters

According to Ghule et al., (2018), a manually operated machine was developed to harvest onions that was designed to be smaller, cheaper, and easier to use than the existing machines, which are usually very large, difficult to operate, and too expensive for most farmers. The machine was attached to a hand tractor and reduced the labor needed for harvesting onions from 122 to 69 man-hours per hectare.

Khurana et al., (2013) created a root-crop harvester that could be used to harvest different types of crops, such as onions, garlic, carrots, and potatoes. The harvester had a 1.14m wide digging blade, which was 16mm thick. It was mounted on a frame, which was angled at 20° from the horizontal surface. Above the digging blade was an elevator conveyor, which had parallel rods spaced 20mm apart. Finally, two oval agitators were added to the conveyor system to help separate the soil particles from the tubers as shown in Figure 2.1.

Figure 2.1. Root crop harvester

Joshi et al., (2020) designed and developed an onion harvester that runs on a vehicle power supply and didn't need any additional external power source. It was designed for better performance in terms of productivity, fuel economy and comfort for the operator and it was meant to reduce damage to crops, reduce human fatigue, and save time. This harvester has two different parts, the digging system used to loosen the soil and get the onion bulbs out without damaging and conveying system that conveys the onion bulbs.

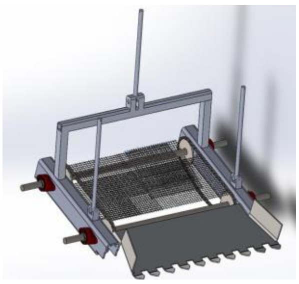


Figure 2.2. Onion harvester assembly

Parab et al., (2019) conducted a study on onion harvester that could be used with tractors of small horsepower. They looked at the farming habits and traditions of people in certain localities, and designed a manually operated harvester for use in India. Gavin et al. (2018) conducted study on power tiller-driven onion harvester and aimed to provide an onion harvester for raised beds and focused on the size of the digger blades when designing the harvester. A simple low-cost self-propelled onion digger windrower was designed and evaluated specifically for the purpose of harvesting onions (Jadhav et al., 1992). The windrower was powered by a 5 hp diesel engine, which was mounted on a frame along with the main gearbox.

Sose et al., (2019) conducted a research on onion harvesting with machinery. They examined customer satisfaction by interviewing farmers who had used the harvester. Additionally, they conducted a conjoint analysis, which involved asking the farmers to rate different ‘simulated concepts’ to see what kind of harvester design the farmers preferred. The study focused on making the harvester better and the results were calculated using experiments conducted on laboratory soil bin.

Figure 2.3. Self-propelled onion harvester

Khura et al., (2011) designed and developed a machine that was pulled by a tractor and was used to harvest onions. The machine had three main parts: one for digging, one for conveying (moving) the onions, and one for separating the onions from the soil. They tested six different shapes of blades for the digging part to see which worked best.

Root crop harvester was designed using materials that were available in local areas (Younus and Jayan, 2015). It was designed to be simple to use and seemed to work well in testing (Sharma et al, 2019). However, when the machine was actually used in the fields, there were problems with it causing damage to the crop, delays in harvesting, and it even broke down at times.

2.6. Performance Evaluation of Onion Harvester

Omar et al., (2018) conducted an experiment to measure the performance of an onion harvester in different soil conditions by adjusting the digging depth and forward speed. It was found that the highest field efficiency and lifting efficiency were 73.9 and 99.2% respectively, when the speed was 0.720 km/h and the digging depth was 4 cm. The lowest total losses, power and energy consumed were 1.9%, 10.11 kW, and 59.5 $kW.h.fed^{-1}$ when the same conditions were used. Lastly, the total cost was 674.33 $LE.fed^{-1}$ at the same forward speed with a digging depth of 10 cm. The cost was lower compared to the manual method which was 2400 $LE.fed^{-1}$.

Mahesh (2014) developed a digging machine specifically for harvesting onion bulbs. It was tested at a speed of 4 km/hr, and it was found to have minimal losses and a high field capacity of 1.10 fed/h. The optimal depth at which to dig was 7.62 cm, and it was done in a way that prevented nearly any damage to the onion bulbs. The mean efficiency of the digging machine was 89.8%, the lifting efficiency was 94.9%, and the damage rate of the onion bulbs was 5.1%.

Khurana et al., (2012) conducted research on a prototype harvester to determine how much of a root crop like onion, garlic, or carrot it could dig. They tested the harvester at different speeds and measured how much of the root crop was able to harvest and how much of it got damaged or exposed in the process. The results showed that the harvester was able to do the job with less labor and cost compared to manual harvesting. Therefore, the harvester could be the solution to reduce labor, time, and total cost when harvesting onion crops.

The field capacity was a measurement of how quickly the machine can produce its desired outcome and the rate at which a machine can work the hectares per hour (Hanna, 2002). This measure was used to plan for field operations and labor, as well as estimating the costs of operating the machine. The field capacity and field efficiency of an onion harvester attached to a hand tractor were reported as 0.086 hectares per hour and 80.52% respectively. This means that the onion harvester was able to complete 0.086 hectares of work per hour and 80.52% of the work was done successfully (Gavin et al, 2018).

A power tiller operated onion harvester was a machine that has been designed and developed to pick onions from the ground and was harvest an area of 0.08 ha/hr and has been tested and found to have 97.4% harvesting efficiency, 86.9% soil conveying efficiency, and 84% soil separation efficiency (Nisha and Shridar (2018).

Khura et al., (2011) conducted a test to determine the best design for a tractor-drawn onion harvester. They tested six different shapes of digging blades and found that the inverted V-shape digging blade had the lowest draft force, which was 625.6N. After testing, the researchers established the length, speed and slope of the elevator that achieved the best performance; these elements were set at 1.2m, 1.25:1 and 15° respectively.

Singh et al., (2014) conducted the study to test the effectiveness of a new digger designed for harvesting onions against manual labor. During the study, the digger was operated across a range of different speeds, ranging from 3.76-4.83 km/h, with the best results at 4 km/h in the first gear, and with a field capacity of 0.46 ha/h. The study found that the digger had an average operating depth of 7.62 cm, resulting in the digger having an efficiency of 89.8%, and it produced 58% savings in labor and 49% savings in cost.

Welsh onion harvester developed an experiment that was done for evaluating harvesting performance in an actual Welsh onion farm (Hong et al., 2014). They tested the harvester at different speeds - 5.0 cm/s, 11.4 cm/s, and 15.8 cm/s and tracked how efficient it was, how much it could harvest at each speed, and how many onions were damaged. The aim of this experiment was to improve the labor productivity and cultivation environment of the farm, as the manual harvesting process requires a lot of hours of labor.

Khura et al., (2011) was testing a prototype onion digger, and they evaluated its efficiency, separation index, damage to bulbs, fuel consumption, and draft to determine how well the prototype worked. The result was that the prototype had a 97.7% efficiency, 79.1% separation index, 3.5% damage to bulbs, 12.81 liters/ha fuel consumption, and 10.78kN draft. They compared this result to a manual method and found that the maximum field capacity of the prototype was 0.180 fed/h at a speed of 1.125 km/h and its maximum field efficiency was 73.9 % at a speed of 0.720km/h. The manual method had a maximum field capacity of 0.125 fed/h and a field efficiency of 84.26%. Finally, the prototype and the manual method had a maximum lifting efficiency of 99.2% and 98.1 %, respectively, with the prototype having a minimum total loss of 1.9% and the manual method having a minimum total loss of 2.5%.

A tractor-mounted harvester-cum-elevator for digging root crops was developed (Anon, 2008a). The field capacity of the machine includes three elements - the activity (or work) level (ha/h), the speed (km/h), and the amount of damage (as a percentage) done to the crop. To make the harvester work properly, a blade is attached to the machine's head. This blade was used to dig out and take the onion bulbs out of the soil. The blade angle and the harvesting depth have a big effect on how efficiently the harvester works and how much damage is done to the onions. In this study, the head of the onion harvester was designed, and then tested in a farm with clay-loam soil (soil type). A four-bar mechanism was used to change the blade angle. The experiments were run at speeds of 1.8, 2.4, and 3 km/h and blade angles of 12, 15, and 20 degrees. In the end, the harvester worked best at 1.8 km/h with a blade angle of 20 degrees, and did less damage than manual harvesting.

Mohamed Ibrahim (2011) claims that the onion harvester's field capacity increased when the forward speed was increased from 1.4 to 2.3 km/h but declined when the speed was further increased to 4.9 km/h owing to machine slide. The capacity of the field also decreased as the digging depth was increased because doing so required more force, which made the machine slip. The maximum slip value, 27.4%, was recorded at a forward speed of 4.9 km/h and a digging depth of 22 cm, while the lowest slip value, 9.5%, was recorded at a forward speed of 1.4 km/h and a digging depth of 12 cm.

Mohamed Ibrahim (2011) found that as harvesting speed increased from 1.4 km/h to 2.3 km/h, the percentage of missing tubers dropped. However, the percentage of missed tubers owing to slip rose as the speed was raised from 2.3 km/h to 4.9 km/h. At 1.4 km/h and a digging depth of 12 cm, the proportion of missing tubers was 27.5%, while at 2.3 km/h and a digging depth of 22 cm, the percentage was 7.6%. Additionally, the proportion of missed tubers decreased when the speed was increased from 1.4 km/h to 2.3 km/h, but the percentage of damaged tubers rose as a result of friction between the blade and the tuber. The energy directly depends on the field capacity and forward speed. Energy use decreased by 1.4 to 2.3 km/h as a result of the corresponding increase in field capacity. Then, as field capacity decreased, speed changed from 2.3 to 4.9 km/h, causing energy to increase. On the other hand, energy increased as the depth of the hole was dug since a greater amount of power was required to overcome the soil's greater resistance.

2.7. The Research Gap

Generally from literatures discussed above, the following gaps were obtained. A two wheel tractor operated onion harvester has to be applied for both flat and raised bed. To make two wheel operated onion harvester suitable for these both beds, it needs to design cutting disc. The soil destruction device or cutting disc reduces the uplift load on the onions and reduces the traction resistance in the soils, such as volcanic and sandy soils, and it has a low withdrawal force on the Welsh onions (Sungha et al, 2014). In addition, Gavino et al, (2018) recommend that the length of the digger can be made shorter in order to avoid piling of soil and onions that causes clogging during the operation. According to Nisha and Shridar, (2018) and Gavino et al, 2018) recommend that cover must be installed in the sprocket and chain assembly of the conveyor to avoid clogging the sprocket wheels. Onion harvesters designed for two wheels tractors have not a collector which places the onion bulbs in narrow bands which is simple for hand picking after digging and separating operations. Additionally, even if potato harvester is designed in Ethiopia in different research centers and academic institutions (Dessye Belay, 2021), biometric properties of onion are entirely different from potato and it does not effectively and efficiently harvests onion and it have not cutting discs. The components that are going to be designed are cutting disc, cover for chain and sprocket, collector.

CHAPTER THREE
MATERIALS AND METHODS

3.1. Description of the Study Area

The study was carried out at AAERC, which is geographically located in Ethiopia and 175 km far from Addis Ababa. It is located at an altitude of 2,430 meters above mean sea level, and its latitude and longitude are 7°55'57.91" N and 39°7'53.43" E. This area generally experiences a dry moist, sub-humid with extremely variable annual rainfall ranging between 1300 and 1350 mm.

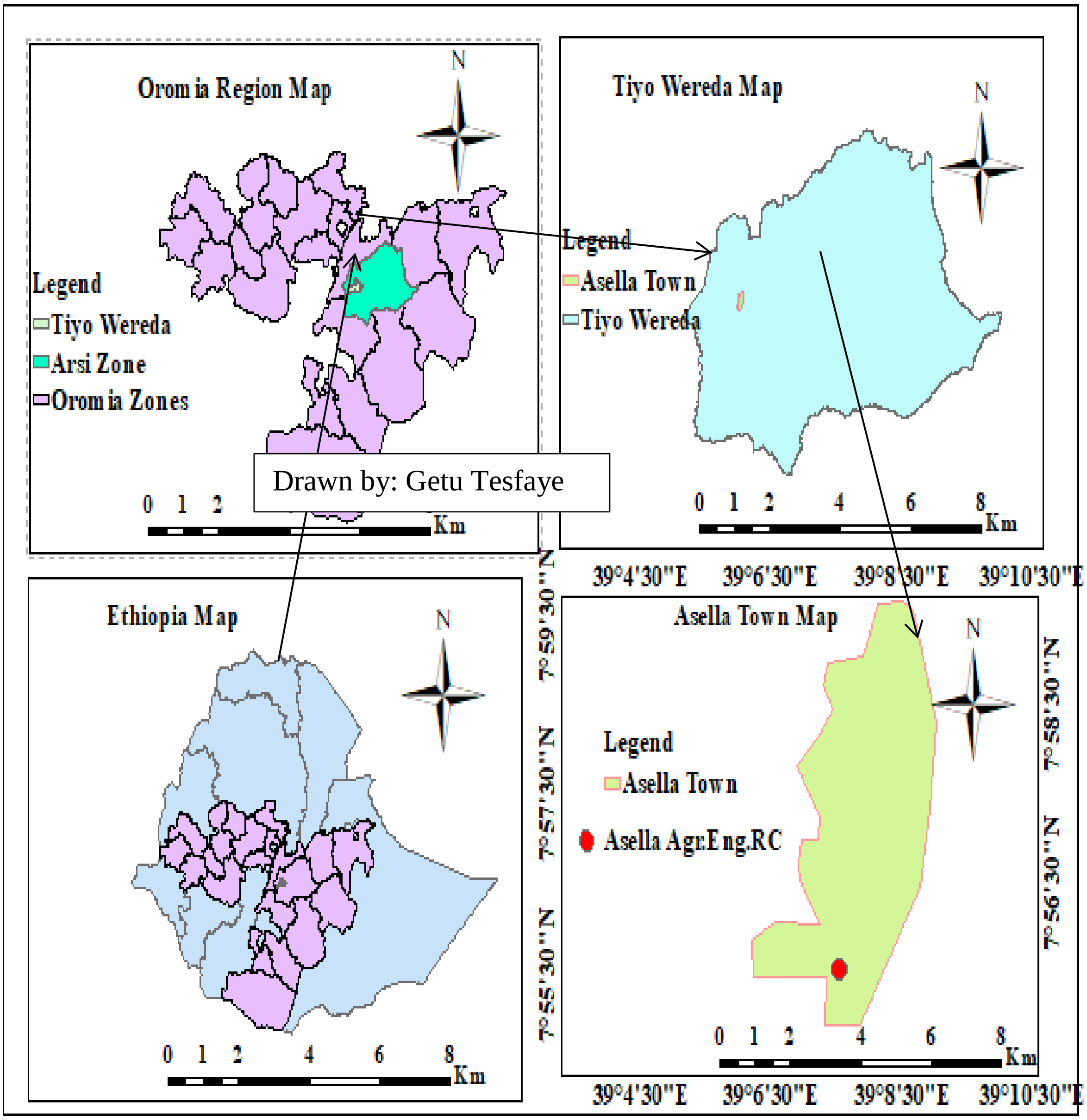


Figure 3.1. Map of the study area

3.2. Materials

3.2.1. Machines and Tools Used for Fabrication

i). 2WT: Used as a power source for operation

Brand: Sifang power tiller

Engine model: S1100N

Maximum power: 15 HP (11.03 kW)

Origin: China

Table 3.1. Machines and tools

S.No	Machine/tool name	Purpose
1	Drilling machine	Hole/cell making
2	Lathe machine	Threading/cutting/finishing/cutting/shaping/machining
3	Grinding machine	Grinding/cutting tool
4	Power shear machine	Cutting more than 3mm thickness sheet metals
5	Welding machine	Welding for joining metal
6	Hand grinder	Grinding metal
7	Vernier calipers	Measurement of outer diameter and inner diameter
8	Hammer	Used to strike an object
9	Snips	Cutting sheet metal
10	Vice	Clamping or holding
11	Spanner	Tightening nut and bolt
12	Flat file	Smooth rough edges
13	Steel tape	Measurement of linear distance

3.2.2. Component Parts, Specifications and Materials

Table 3.2. Different components parts, specifications and materials

S.No	Components	Specification	Material
1	Blade	720mm* 360mm* 6 mm	Flat mild steel
2	Frame	850 mm *40mm	MS square pipe
3	Shaft	30 mm and 25mm diam with length of 900	Mild steel
4	Depth control	200mm diameter, 4mm thickness	Sheet metal
5	Drawbar hitch	320mm* 40mm	Hollow pipe angle iron
6	Bearing	Diam 30mm and 25mm	High carbon steel
7	Side plate	850mm* 560mm*4mm	Sheet metal
8	Cutting disc	Diameter 340mm and 6mm thickness	Sheet metal
9	Collector	900mm*30mm*20mm	Sheet metal
10	Conveyor	680mm length and 12 diam	Iron steel round bar
11	Chain	4200mm length	Alloy steel
12	Belt	720mm length	Rubber
13	Pulley	255mm diam	Steel
14	Sprocket	24mm diam, 17 and 14 teeth	Stainless steel

3.3. Methods

3.3.1. Conceptual Design

Identify needs of farmers

Problem identification

Concept generation

Detailed design and analysis

Material selection

Manufacturing

Prototype and testing

Multiplication

a) Identifying needs of farmers

The design process typically begins with the awareness of a need, a problem, or a prospective market for a given product, device, or procedure. Because the need is frequently not obvious and can be ambiguous, it calls for a lot of imagination and ingenuity to identify it. In order to make the lives of the stakeholders easier and more appealing while reducing drudgery, the demands of farmers were evaluated by interviewing small-scale farmers.

b) Problem identification

The problem definition is more detailed and includes all the requirements for the designed object, including the input, or resources to be used, expected output, cost constraints, quantity needed, expected life and reliability and expected variations in the variables, dimensional and weight constraints, manufacturing constraints, etc. A constraint is anything that restricts the designer's ability to make choices, and all constraints must be specified in the problem definition. Even while developing something new, it can be quite beneficial to research similar items, technology, patents, and technical publications that already exist.

c) Concept generation

The goal of the concept generating process known as synthesis is to generate as many ideas as you can to provide viable solutions to the issue that was previously identified. The initial focus is on identifying the primary demand, after which all practical design options are generated. In this stage, different suggested alternatives shouldn't be contrasted with one another; rather, they should each be individually assessed using predetermined standards.

d) Detailed design and analysis

The actual designs of each component must then be developed after the initial designs and specifications of each system have been established. During this phase of the project, detailed engineering design drafting and analysis will occur, necessitating a large amount of documentation. The information and documentation from this step will have a direct impact on the manufacturing process, so care must be taken to ensure that no engineering information about the design of any system is overlooked and that it is clearly documented. Solid works were created during the conceptual phase, but because they are so high-level in nature, they mostly help to make the process of choosing an overarching design concept easier.

e) Material selection

Material selection is a step in the process of designing any physical object. In the context of product design, the main goal of material selection is to minimize cost while meeting product performance goals. Systematic selection of the best material for a given application begins with properties and costs of candidate materials. Material selection is often benefited by the use of material index or performance index relevant to the desired material properties. Some of the important characteristics of materials are: strength, durability, flexibility, weight, resistance to heat and corrosion, ability to cast, welded or hardened, machinability.

f) Manufacturing

Once an engineering design has been completed, the crucial next step is to properly translate these blueprints into actual, repeatable production processes. The manufacturing tolerances on all 2D drawings of assemblies and parts, as was indicated in the step before, should be properly specified by the designer. To ensure that a certain component can be made correctly, manufacturing tolerances alone are not enough to guarantee that they are specified. As a result, in order to guarantee accurate and effective manufacturing standards, a number of crucial document kinds must be produced.

f) Prototype and testing

This phase often involves creating and testing a prototype model of the design as the last step in determining whether it meets the original requirements. Design was also assessed for its dependability, competitiveness, economic viability, maintenance needs, profitability, and other factors.

g) Multiplication

After the machine has tested and its performance has accepted according to scholars recommendations, the machine should have to multiplied and distributed for farmers.

3.3.2. Working Principles and Description of the Machine

The power from the 2WT engine clutch transmitted into two components of a machine, one is to conveyor and drawbar hitch of a machine using pin joint hitching that used for draft pull. When the 2WT moved forward, the digging blade (9) and cutting discs (10 and 12) breaks and split the ridges up to the desired depth and lifts the onion tubers and soil clods with the rest of the plant body. The machine was elevating the tuber-soil mix directly to conveyor (7) from the digging blade. The conveyor separates the soil and then transports the onion bulbs to the rear end through rods and collected on collector (6) which puts the onion bulbs in narrow strips and then picked manually by hand.

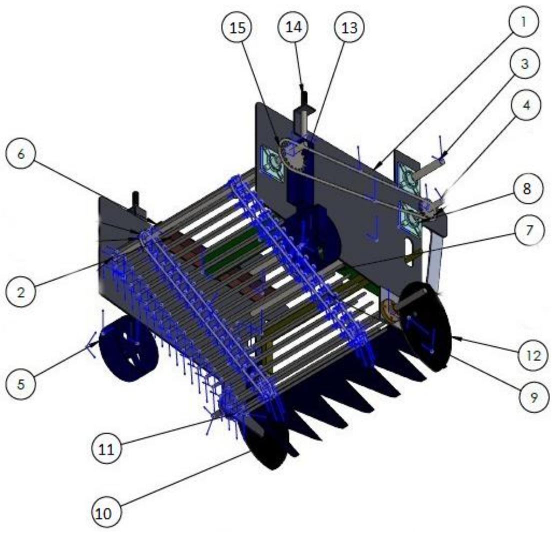


Figure 3.2. Components of the machine

1: Side plate, 2: Chain conveyor, 3: Rear conveyor shaft, 4: Front conveyor shaft, 5: Depth control wheel, 6: Collector, 7: Conveyor, 8: Power transmitting chain, 9: Digging blade, 10: Left cutting disc, 11: Conveyor shaft bearing, 12: Right cutting disc, 13: Conveyor sprocket, 14: Adjustable tread.

3.3.3. Design Considerations and Assumptions Made

During the designing of the onion harvester, the numerous design considerations were carefully analyzed based literature reviews. Some of these considerations are the following:

- The machine must be adaptable for both flat and raised-bed planting methods of onions;
- The cost of the machine should be considered for small scale farmers;
- Locally available materials would be used for the fabrication of the machine;
- The machine should provide a significant reduction of the labor force in onion harvesting;
- The digging depth must be adjustable based on the onion depth;
- The angle of digging blade was fixed to 20^o;
- The conveyor slope was fixed to 20^o;
- Damage to tuber during harvesting operation should be as low as possible and;
- It should be operated by a 2WT of 15 hp, which was available on the hands of farmers at study area.

3.3.4. Determination of Physical Properties of Onion Tuber

The onion digger's design required knowledge of the physical characteristics of onion tubers. Size, shape index, moisture content, and coefficient of static friction were among the parameters that would be assessed.

3.3.4.1. Size

The two different sizes of onion bulb diameter are polar diameter and equatorial diameter. The distance between the onion's crown and its point of root attachment is measured by the polar diameter. Equatorial diameter is the largest width of the onion in a plane perpendicular to the polar diameter. Each of the 100 bulbs' thickness, polar diameter, and equatorial diameter were determined using a caliper with a resolution of 0.01 mm. The Arithmetic mean was calculated using the equation described by Mohsenin (1986)

$$\text{Arithmetic mean diameter} = \frac{(D_e + D_p + T)}{3}, \text{ cm} \qquad \text{3.1}$$

Where:

D_e = Equatorial diameter, cm

D_p = Polar diameter, cm

T = Thickness, cm

3.3.4.2. Shape Index

The shape index, which is used to assess the shape of onion bulbs, was computed using the following eqn. 3.2.

$$\text{Shape Index} = \frac{D_e}{\sqrt{D_p + T}} \qquad \text{3.2}$$

The onion bulb is regarded as an oval if the shape index >1.5, and a sphere if the shape index is <1.5.

3.3.4.3. Moisture Content of Onion

An essential factor that directly affects harvesting and tuber quality is the moisture level of the tuber. By using a gravimetric approach, the moisture content of onions was determined. A 50g sample was weighed, placed in an empty moisture box, and its combined weight was recorded. For 24 hours, the moisture box was placed in oven dry at 105°C ± 2 °C. The weight of the moisture box containing the sample was determined after 24 hours. The formula below can be used to calculate the onion's moisture content.

$$Mc = \frac{M2 - M3}{M2 - M1} \times 100 \qquad \text{3.3}$$

Where:

M_w=Moisture content, % (W.b)

M_1=Weight of moisture box, g

M_2= Weight of moisture box + tuber before drying, g

M_3= Weight of moisture box + onion tuber after drying, g

3.3.4.4. Coefficient of Static Friction of Onion

The galvanized sheet, mild steel, aluminum sheet, and plywood were used to determine the onion's coefficient of static friction (μ). The slope was gradually raised while the onion was kept on a horizontally level surface. The angle (α) at the moment of impending slip was noted. The experiment was repeated three times, and the mean inclination angle was calculated. The coefficient of static friction was:

$$\mu = \tan \alpha \qquad \text{3.4}$$

3.3.5. Determination of Physical Properties of Soil

3.3.5.1. Moisture Content of Soil

Soil samples were collected up to a depth of 10 cm in order to assess the soil moisture content of the area. Three random locations on each test plot were used to gather the samples. These samples were weighed and baked for 24 hours at 105°C. It was the mass's difference in percentage between before and after oven drying (dry weight basis). The formula below can be used to calculate the moisture content on a dry weight basis (Khura, 2008).

$$Mc = \frac{W_1 - W_2}{W_2} \times 100 \qquad \text{3.5}$$

Where:

Mc = Percent moisture content, %

W_1 = Weight of the wet soil sample, g

W_2 = Weight of dry soil sample, g

3.3.5.2. Bulk Density of Soil

A core sampler was used for taking the soil sample. Samples were weighed and kept in the oven at 105 ± 5^0C for 24 hours. Bulk density of soil can be calculated using the equation given below (Khura, 2008). Initially volume of a cylinder was determined by measuring the internal diameter (10 cm) and height of core cutter (12.5 cm) and empty core cutter was weighed. A cylindrical core cutter was pressed into the soil mass using the rammer with dolley placed over the top of the core cutter. Surrounding soil of core cutter was removed and it was taken out. Top and bottom surface of the core cutter was carefully trimmed using a straight edge. Core cutter filled with soil was removed and weighed.

$$\rho = \frac{M}{V} \qquad \text{……3.6}$$

Where:

ρ = Bulk density, g/cm^3

M = mass of soil sample, g

V = volume of core sampler, cm^3

3.3.5.3. Penetration Resistance

Soil cone penetrometer was used to measure the penetration resistance of the soil. A cone penetrometer with proving ring having dial gauge was used to measure the soil resistance to penetration before harvesting. The cone penetrometer was positioned in the field and slightly pressed on the handle. Cone index provides an indication of soil resistance and it is expressed as force per square centimeter required for a cone of standard base area to penetrate into soil to different depths. Cone index for the same soil varies with the cone apex angle, area of cone base and depth of penetration. A uniform force was placed on the handle and deflection of dial gauge was noted for 5 cm in depth. The solid stem penetrated into the soil and force was measured from the deflection of the needle of proving corresponding to the insertion of 30 degree cone. The cone index was measured for 5cm, 10cm, and 15cm depth and recorded manually. The same procedure was repeated to measure cone index at various location of the study area (Venkatareddy, 2018).

$$Pz = \frac{F_a}{A_b} \qquad \text{……3.7}$$

Where:

P_z = penetration resistance, MPa

F_a = Force applied, N

A_b = Cone base area, mm^2

3.3.5.4. Shear Strength

Shear strength of a soil was the maximum resistance offered by the soil to shearing stresses (Venkatareddy, 2018). The shear strength of the soil was observed using a shear vane tester in 10cm depth of soil. To measure the un-drained shear strength (S_{uv}) in the soil before harvesting field inspection vane tester with heights to diameter ratio of 2 was used.

According to ASTM (2002), soil shear strength is contributed by cohesion and angle of internal friction.

$$\tau=\frac{6}{7}\left(\frac{T}{\pi D^3}\right) \ldots\ldots 3.8$$

Where:

T= Torque (Nm)

D= Diameter (m)

Bore holes were excavated at depths of 10, 15, and 20 cm. Since the casing was extended to these depths, the entire unit had to be fixed in place during the test. Using spikes, the torque applicator was fastened to the support. A 37.5 mm diameter vane was chosen, and it was attached to a vane rod with the same female thread. To the required depths, the vane was lowered. After inserting the vane, it was gently pushed down until it reached a depth of 50 mm below the bottom of the bore hole. After that, it was let to continue moving for 5 minutes. The gear handle was twisted to rotate the vane at a rate of 0.1 degrees per second while the dial gauge's initial reading was set to zero. This helped to achieve a consistent pace of 12 rotations per minute. To disrupt the soil, the vane was fully turned ten times. At 30-second intervals, the reading on the torque indication dial gauge was recorded, and the vane continued to rotate until it began to deviate noticeably from its maximum value.

3.3.5.5. Angle of Soil - Metal Friction

According to Khura (2008), the inclined plane method is used to estimate the angle of soil-metal friction on a soil block. The soil block kept on horizontal plane and slope was gradually increased. The coefficient of static friction computed using the following formula:

$$\mu = \tan\delta \ldots\ldots 3.9$$

Where:

μ = Coefficient of friction

δ = Angle of soil- metal friction (^{o})

3.4. Design and Fabrication of Onion Harvester

The machine was constructed based on the established size for each component specified in the design specification. Solid Work software was used to present the design of the onion harvester machine's component parts. The major components of this machine were made of mild steel (AISI 1020), which contained 0.16–0.30% carbon. Mild steel is readily accessible, inexpensive, malleable, and employed in applications where ductility is crucial while having a relatively low tensile strength (Gupta, 2009). The fundamental factors to be taken into account included the determination of forward speed, depth of operation, input, output, and conveyor shaft, digging blade, conveyor with separation unit, collector, cutting disc, depth controller, and selection of (pulley, chain, belt, spur gears, bearings) were decided based upon different research outputs.

The power source was a 2WT which has 15hp.
Brake horse power (BHP) = 15 hp
Soil type = Clay loam
Drawbar horse power (DBHP): 60 % of BHP of 2WT (Sharma and Mukesh, 2010)
Therefore, DBHP = 15hp × 0.6 = 9hp
The tractor drawbar horse power (DBHP), Draught available (Df) was calculated by eqn. 3.10 and 3.11 (Sharma and Mukesh, 2010).

$$DBHP = \frac{D_{f} \times V}{270} \quad \ldots 3.10$$

Where:

Df = Draft force, (kg)

V = Speed, (km/hr)

Let the forward speed of the 2WT be 3km/hr.
Putting the values in the equation (3.10)

$$\text{Draught available} = \frac{9hp \times 270}{3.5km/hr} \quad \ldots 3.11$$

3.4.1. Power Required For Soil Cutting

Digger blade would execute initial digging of onion tubers. The width of digger blade was an important factor, as it would cover all plant rows in a bed without damaging standing crop. Therefore, it was decided on the basis of the width of the bed that the root crop was grown in rows. The thicknessof the blade was designed on the basis of load acting on it. This could be theoretically determined by analysing various different forces acting on the blade. Onion digger having maximum depth and width of 10cm and 36cm respectively that was depth with width ratio of 0.208 which is less than 0.5 called as wide blade (Khura, 2008). The soil cutting force of the blade was calculated using eqn. (3.12).

$$Fs = (\gamma d^2 N_\gamma + qdN_q + cdN_c + c_a dN_{ca}) \times w \quad \ldots 3.12$$

Where:

γ = unit weight of soil, kg/ m^3

C = apparent cohesion, kN/m²

$_{Ca}$ = soil-interface adhesion, kN/m²

d = depth of operation, m

w = width of operation, m

q = surcharge pressure on soil, kN/m²

F_s = soil resistance force (KN)

N_γ = soil friction cutting coefficient

N_q = soil overburden cutting coefficient

N_c = soil cohesion cutting coefficient

N_{ca} = soil adhesion cutting coefficient

Nγ, Nc, Nca, Nq are dimensionless Reece factors, which states the shape of soil-failure surface.

The following assumptions, according to Sose et al. (2019), were made:

a) Soil is isotropic and homogeneous.

b) The bulk density of soil is assumed to be 1.45g/mm^3 on average.

c) The soil has moisture content within the friable range, cohesiveness of 0.07 kg/mm^2, an internal friction angle of 25°, and a soil metal friction angle of 20°.

d) The soil's adhesion can be assumed to be zero, with the assumption that soil-metal, m = 0.

e) Zero friction caused by soil scouring the blade.

f) The surcharge over the soil failure zone in front of the soil is insignificant, or q = 0.

g) Numerous researchers have reported that the typical differences in the rake angle of the digging share range from 15° to 20°. A rake angle of 20 gave minimum draft and maximum upward force for a simple Tyne as reported by Payne (1956) and Khurana (2013).

Based on suggestions and presumptions made by various works of literature during the design and evaluation of tillage implements and root crop harvesters, the estimated soil cutting force (Fs) of the onion harvester was developed. Surcharge (q) and machine adhesion (Ca) of soil utilized by various researchers can be regarded as insignificant. Equation (3.12) can therefore be simplified to equation (3.13):

$Fs=(\gamma d^2 N_\gamma + cdN_c) \times w$…………………………………………………………..…....…3.13

The relationship between the N factors and the rake angle at different angle of internal friction for a blades with perfectly smooth (δ=0) and perfectly rough (δ = Φ) interface was take from graph (Hettiarachi, 1966) which was putted on appendix graph I. The values of N factors for intermediate degree of roughness of the interface can be interpolated using the following equation (Mckyes,1989):

$N = No \left(\frac{N\emptyset}{No}\right)^{\frac{\delta}{\emptyset}}$ ……………………………………………………..…………………....3.14

Where:

N_δ = the required value of the appropriate N factors like as N_γ or N_c

$N_\delta = 0$ and $N_\delta = \Phi$ are the corresponding value of the N-factor at $\delta = 0$ and $\delta = \Phi$, respectively obtained from appropriate chart.

Based on the assumptions made above, values of optimum parameters were listed below for the determination of soil passive resistance (Budhale et al, 2019 and Sose et al, 2019).

$\gamma = 1450\ kg/m^3$, $C= 30.18 KN/m^2$, $\Phi = \delta = 20^o$, $\alpha = 25^o$ and $d = 0.1$ m

Value of N-factor was selected as follows from (appendix graph I).

$N_\gamma = 1.65$ when $\delta = 0$

$N_\gamma = 1.83$ when $\delta= \Phi$

$N_c = 1.72$ when $\delta = 0$

$N_c = 1.68$ when $\delta = 20$

Now substituting the values of N_γ and N_c determined, the passive resistance (F_s) is given as:

$F_s = (1450\times9.81\times(0.1m)^2\times1.83 + 30.18\ N/m^3\times1000\times0.1m\times1.68)) \times 0.36m$

$= (260.308\ Nm + 5070.24\ N/m)\times0.36m = 1919\ N$

The passive resistance force (Fs), according to Khura (2008), is acting at an angle of friction (δ) with normal and perpendicular to the contact of a blade. Hence, the component parallel to the blade face (F_h) was:

$F_h= F_s \sin\delta$

$F_h = 1919\ N \sin 20^o = 656.34\ N$

Component perpendicular to the blade face (F_v) was given as:

$F_v = Fs \cos\delta$

$F_v = 1919N \cos 20^o = 1803.27N$

Power required for soil cutting (P_{sc}) was:

$P_{sc} = Fs \times V$

Where:

F_s = soil resistance force, (N)

V = operating speed (m/s), let's take V=3.5Km/hr = 0.972m/s

$P_{sc} = 1919N \times 0.972m/s = 1865.27w$

3.4.2. Design and Construction of Soil Cutting Blade

The front end of the conveyor has a trapezoidal-shaped blade with a significant width. In the process of functioning, it moves into the ridge beneath the onion zone and separates the ridge slice from the soil mass and designed using high carbon steel. After being sharpened, the blade's cutting edge tapered at a 16^{o} (Khura, 2008). With screws for easy replacement, the blade was fixed to the shank. While Fv is the vertical component which would result in a bending moment and Fh is the horizontal component which would place direct stress on the blade. According to Figure 3.3, the force would be applied at the blade's center of resistance. It was assumed that average soil resistance of the blade acts at a distance of 0.2z1, measured from the cutting edge (Bernacki, 1972). The blade is supported on the shank at a distance 200mm from the cutting edge. Therefore, the distance between the center of resistance and point of support (Dr) could be determined by:

D_r =100mm – 0.2×100 mm = 80mm

Length of blade from blade support to center of resistance (y):

$y = \frac{80mm}{\sin 25^{o}} = 189.3mm$

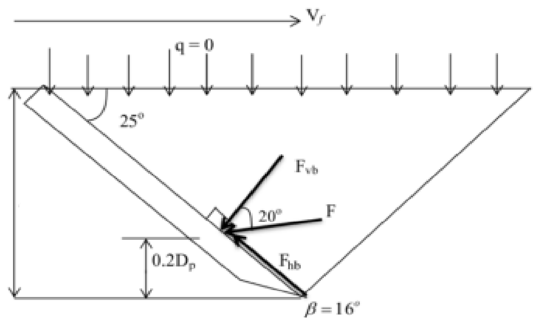


Figure 3.3. Soil reactions acting on a blade (Khura, 2008)

Therefore, the bending moment (M) due to Fv would be:

$M = F_v \times y$……………………………………………………………..…………...…3.15

$= 1803.27N \times 189.3mm = 341{,}359Nmm$

Bending stress on the blade (σ_b) is:

$\sigma_b = \frac{M}{\frac{1}{6}bt^2}$……………………………………………………………………………………3.16

Where:

M = Bending moment, Nmm

t = Thickness of the blade, mm

b =Width of blade at the point of mounting, mm

y = distance from cutting edge to center of the resistance, mm

$$\sigma_b = \frac{341{,}359\text{Nmm}}{\frac{1}{6}\times 720\text{mm}\times t^2} = \frac{2844.66\text{N}}{t^2}$$

Direct stress (σ_d) due to F_h was:

$$\sigma_d = \frac{F_h}{b\times t} \quad \text{3.17}$$

$$= \frac{656.33\text{N}}{720\text{mm}\times t} = \frac{0.91\text{N/mm}}{t}$$

Total stress (σ) could be:

$$\sigma_t = \sigma_b + \sigma_d \quad \text{3.18}$$

$$= \frac{2844.66\text{N}}{t^2} + \frac{0.91\text{N/mm}}{t}$$

By taking factor of safety 3 for agricultural machines (Sharma and Mukesha, 2010) and yield stress of 294.74MPa of AISI 1020 mild/low carbon and by equating the total stress, the thickness of blade (t) was determined as:

$$\sigma_y = \left(\frac{2844.66\text{N}}{t^2} + \frac{0.91\text{N/mm}}{t}\right)\times \text{fos}$$

$$294.74\text{Mpa} = \left(\frac{2844.66\text{N}}{t^2} + \frac{0.91\text{N/mm}}{t}\right)\times 3$$

$t = 5.90\text{mm} \approx 6\text{mm}$

The width of blade was calculated as effective zone of triangular V-shaped blade was (Sharma and Mukesha, 2010):

$$Zf = w + 2d\tan\emptyset \quad \text{3.19}$$

Where:

Z_f = Effective zone of blade, cm (50 cm, Jadhav et al., 1995)

W = Cutting width of blade, cm

d = Depth of operation (10cm, on the basis of tuber depth)

ϕ = Angle of internal friction which depend upon soil type, ($\phi = 35^O$)

Therefore, width of onion digger blade was computed as:

$50\text{cm} = w + 2\times 10\text{cm} \tan 35^o$

$w = 35.99\text{cm} \approx 36\text{cm}$

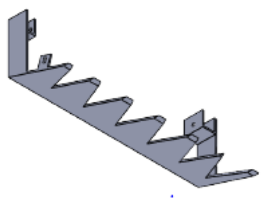


Figure 3.4. Digging blade

The solid work force analysis of the model is carried out to optimize the design by analysing the reaction of blade under the loading conditions. As per the design calculations, the thickness of the blade is 6 mm and it is tested under the draft forces acting on the blade. The boundary condition is considered as the zero displacements of the side end of the blade at top corners, which will be joined to the conveying system. The forces applied on each tip of the blade and the resulting total deformation and equivalent stress has been verified and ascertained to be safe. The figure 3.5 below shows that the plot type being analyzed was static displacement which likely refers to the displacement of an object or structure under a static load. The deformation scale is given as 7.01157, which could refer to the amount of deformation or strain being analyzed in the study. "ESTRN" is likely an abbreviation for "strain", which is a measure of the deformation of a material relative to its original size. The values in the table represent the strain at different points in the material. The values in the table are presented in scientific notation, which is a way of expressing very large or very small numbers using powers of 10. For example, 8.622e-04 means 8.622 times 10 to the power of -4, or 0.0008622. The values in the table are presented in a column format, with each row representing a different point in the material. The values in the table represent the strain at each point, with positive values indicating stretching and negative values indicating compression. The value in the table ranges from -0.00054876 to 0.02406004 with the majority of value being is very small (less than 0.001). The last value in the table is 0.000, which likely represents a point in the material where there is no strain

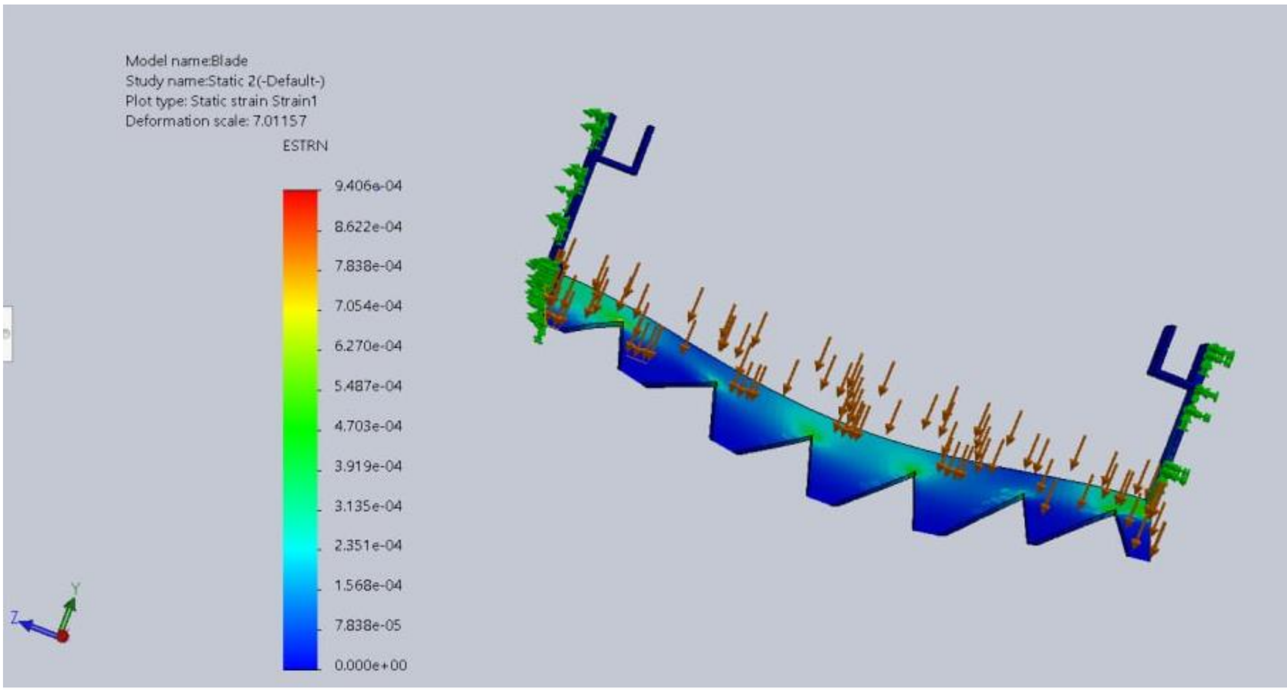


Figure 3.5. Solid work force analysis on digging blade

3.4.3. Design of Shank

The draw bar hitch and the blade were connected by MS straight rectangular section shanks. The total of the horizontal force (F_{sh}) and the vertical force (F_v) on the blade was the force needed to move the machine.

Horizontal forces (F_{sh}) can be found using in eqn. (3.20).

$F_{sh} = F_s \sin(\alpha + \delta) + cadw\cot\alpha$……………………………………………………………..3.20

$= 1919N \sin(25^o + 20^o) + (0)\times 0.1m\times 0.36m\times \cot 25^o = 1356.94N$

The bending moment (M) on the shank due to F_{sh} was:

$M_{shank} = Dd \times Ls$…………………………………………………………..……..…..….3.21

Where:

Dd = Designed draft, N or F_{shank}

Ls = length of shank M_{sh}

$M_{sh} = 1356.94N \times 450 = 610{,}623Nmm$

Bending stress on blade (σ_b) was:

$$\sigma_b = \frac{610{,}632\ Nmm}{\frac{1}{6}\times 50mm\times ts^2} = \frac{73{,}275.84\ N}{ts^2}$$

Direct stress (σ_d) caused by vertical force of soil was:

$F_v = F_s \cos\ (\alpha+\delta) + cadw$ ……………………………………………..………..…..….3.22

Where:

F_v = vertical force

δ = Angle of friction

α = rake angle

$F_v = 1919N \cos(25^o + 20^o) + 0\times 0.36m\times 0.1m$

$= 1356.94N$

$$\sigma_d = \frac{Fvs}{b\times t} = \frac{1919N}{50mm\times t} = \frac{38.38N/mm}{t}$$

Total stress (σ) computed as:

$\sigma = \sigma_b + \sigma_d$

$$\sigma = \frac{73{,}275.84N}{t^2} + \frac{38.38N/mm}{t}$$

$$294.74N/mm^2 = \left(\frac{73{,}275.84N}{t^2} + \frac{38.38N/mm}{t}\right)\times 3$$

$t = 27.5mm \approx 30mm$

3.4.4. Analysis of Drawbar Power

The tractor's mechanism both the transmission and the wheels converts the engine's rotating action to the drawbar's linear motion. Eqn. (3.23) states that a tractor's drawbar power is a function of the draw bar force and forward speed (Macmillan, 2002).

$P_d = P \times V$……………………………………………………………………………......3.23

Where:

P_d = drawbar power, w

P = Drawbar pull, N

V = Travel speed, m/s

The draft is the horizontal component (F_{sh}) of soil cutting force (F_s) (Mckyes, 1989):

$Pd = F_{sh} \times V$…………………………………………………………..………........3.24

$= 1356.94N \times 0.972m/s = 1319$ w

For equilibrium of the external horizontal forces acting on the tractor (Macmillan, 2002),

$H = F_{sh} + R$……………………………………………………………………..…...3.25

Where:

H = Soil reaction force/Soil thrust, N

R = rolling resistance force, N

Coefficient of rolling resistance of agricultural tires of different diameter on various surfaces was graphically explained (Macmillan, 2002). Coefficient rolling resistance of 1000 mm diameter tractor wheel on stubble surface was about 0.124. Then, rolling resistance (R) was computed as:

$R = \mu W$……………………………………………………………………..….…3.26

Where:

μ=Rolling coefficient

W=Weight

m = mm + ms…………………………………………………………….…..….…3.27

Where:

mm = mass of the machine, ms = mass of soil

$m = 135.14kg + 58.37kg = 193.51kg$

$W = mg = 193.51kg \times 9.81m/s^2 = 1898.33N$

$R = 0.124 \times 1898.33N = 235.4N$

$H = 1356.94N + 235.4N = 1592.34N$

Comparative slip-pull performance wheel slip of two-wheel drive tractor on stubble, uncultivated and loamy surface soil at 3137.3 N drawbar pull is about 1-5% was graphically explained Macmillan (2002)

$$i = \frac{Vo - V}{Vo} \qquad \text{...3.28}$$

Where:

i = wheel slip (%)

V_o = linear velocity of ground wheel of tractor, m/s

$$0.05 = \frac{Vo - 0.972 \text{ m/s}}{0.972 \text{ m/s}} = Vo = 1.02 \text{ m/s}$$

Wheel rpm of 2WT can be determined as:

$$N_w = \frac{60 \times Vo}{\pi D} \qquad \text{...3.29}$$

Where:

D = drive wheel diameter, (1m)

N_w = wheel speed, rpm

Power loss (P_l) of 2WT due to wheel slip (*i*) and rolling friction was (Macmillan, 2002):

$$P_l = P_w - P_d \qquad \text{...3.30}$$

Where:

P_w = Wheel power, w

P_d = Drawbar power, w

$$P_l = HVo - PV \qquad \text{...3.31}$$

$$= 1592.34\text{N} \times 1.02\text{m/s} - 1356.94\text{N} \times 0.972\text{m/s}$$

$$= 305.24 \text{ W}$$

Wheel power (P_w) was:

$$Pw = P_{sc} + P_l \qquad \text{...3.32}$$

Where:

P_{sc} = Power required for soil cutting, ($P_{sc} = 1865.27W$ as calculated above)

P_w = 1865.27w + 305.24w = 2170.51w

Power losses in mechanical transmission of two-wheel tractors are usually less than 10% i.e. 90% efficiency (Macmillan, 2002). Hence, power at mechanical system of the tractor (P_t):

$$P_t = \frac{P_w}{\eta} \qquad \text{...3.33}$$

= 2411.68W

Power at a variable speed V-belt drive pulley (clutch) of the 2WT tractor (Pi) divided to conveyor separator system and drawbar transmission system of the machine.

3.4.5. Design of Conveyor and Separation Unit

The soil crop material dug by a digging unit is the soil with the crop that has been dug out of the ground by a machine. The separation unit was placed directly behind the blade to separate the soil from the crop. The separation unit used stainless steel plates with a round shape that were arranged along the line of travel of the onion digger. The conveyor is mounted at the rear end of the digger blade and consists of two shafts with ball bearings fixed apart at the center distance of 680mm. Four sprockets of 9.5 cm diameter with two chains combination were mounted on the conveyor shafts.

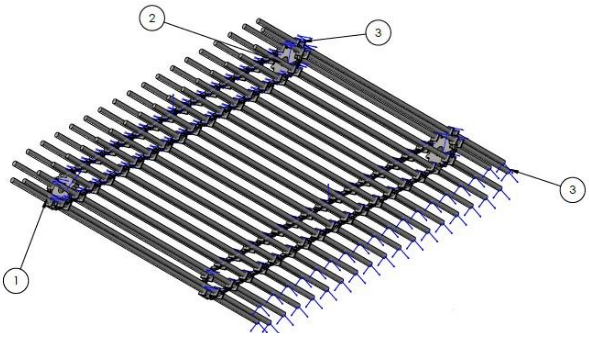


Figure 3.5. Conveyor and separation unit

1: Conveyor chain, 2: Conveyor sprocket and 3: Rod bars.

3.4.5.1. Power Required for Conveyor and Separation Unit

According to Primo (2009), a point A located on the end of the blade and the beginning of the conveyor has a particular angle to the ground; this helps to calculate and analyze the movement of the soil-tuber mass on the conveyor to separate the crop from the soil.

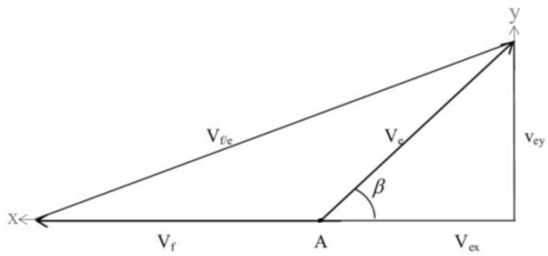


Figure 3.6. Soil-tuber velocity analysis on conveyor

Where:

A = a point of onion tuber moving to chain elevator.

V_f = machine forward velocity, m/s

V_e = resultant velocity of elevator chain, m/s

V_{ey} and V_{ex} = vertical and horizontal component velocity of elevator chain, m/s

$V_{f/e}$ = relative velocity of 2WT forward speed and elevator speed, m/s

β = the angle between the movement direction and the chain conveyor speed, (^{o})

The relative speed of chain and tractor was the speed of soil tuber mass on a conveyor:

Vf /e =Vf + Ve………………………………………………………………..……..…….3.34

Where:

V_f = travel speed of a machine, 0.972m/s

V_m= speed of elevator, 1.5m/s

$V_{f/e}$ = relative speed /speed of soil tuber mass, m/s.

V_{ex}, and V_{ey} is X and Y component of V_e (velocity of elevator), m/s

Based on trigonometric relation, V_{ex} and V_{ey} was:

$V_{ex} = V_e \cos25^o = 1.5\cos25^o = 1.36m/s$

$V_{ey} = V_e \sin25^o = 1.5\sin25^o = 0.634m/s$

The relative speed of soil tuber mass in x direction:

V_x =Vex +Vf………………………………………………..………………..……...3.35

$V_x = 1.36\ m/s - 0.972\ m/s = 0.388\ m/s$

The relative speed of soil tuber mass in Y direction,

$V_y = V_{ey} = 0.634m/s$

The relative speed of soil tuber mass $V_{f/e}$, was:

$Vf/e^2 = V_x^2 + V_y^2$……………………………………….………………….……...3.36

$$Vf/e^2 = \sqrt{V_x^2+V_y^2}$$

$$= \sqrt{(0.388m/s)^2+(0.634m/s)^2} = 0.743m/s$$

3.4.5.2. Selection of Conveyor Chain and Sprocket

Dessye Belay (2021) studied a conveyor chain drive with two chain and sprocket wheel combinations on opposite sides of the conveyor web. The chain drive was operating at 300 rpm, which was necessary in order to reach the desired speed of 1.5 m/s and have a power output of 983.97W. In order to calculate the overall service factor a few different factors were taken into account. The load factor takes into consideration the amount of variable load, with a medium shock factor, which was set to K_1=1.25. There was also a lubrication factor for periodic lubrication, set to K_2= 0.85, and a rating factor for 8-hours of operation per day, set to K_3=1

$K_s = K_1 \times K_2 \times K_3$……………………………………………………….……………..…...3.37

$= 1.25 \times 0.85 \times 1 = 1.875$

$P_d = P_c \times K_s$…………………………………..……..……………………………..………3.38

$= 528.5w \times 1.875 = 990.94w$

Chain number 10B had a capacity to transmit power of 3150W per strand but it wasn't strong enough to undertake a welding job with angle iron on a chain plate for the conveyor rod. Therefore, it was necessary to use chain number 12B instead. This chain had a power transmitting capacity of 5430W and had the following specifications: Pitch (p), distance between the center of one roller link to the center of the next one (19.05mm); Roller Diameter (d) the diameter of the roller around which the chain link rotates (12.07mm); Required width of roller (w), the width of the roller around which the chain link rotates (22.30mm); Breaking Load (WB) the maximum load a strand can withstand before it breaks (28.9kN).

The conveyor speed is 1.2-1.5 of forward speed (Schweers et al., 2015). Hence,

$V_c = V_f \times 1.5 = 1.5 \times 0.975 m/s \approx 1.5$ m/s

The pitch circle diameter of the conveyor sprocket was:

$V = \frac{\pi DN}{60}$ …………………….……………………………………..…...……….………...…..3.39

1.5 m/s$= \frac{\pi \times D \times 300 \text{ rpm}}{60} = D = 9.5$ cm

Load on the chain (W) could be (Kurmi and Gupta, 2005):

$W = \frac{P_d}{V_c}$……………………………………………………………….……………..……3.40

$= \frac{990.94 \text{ w}}{1.5 \text{ m/s}} = 660.62$ N

Factor of safety (fos) was calculated using:

$fos = \frac{W_b}{W}$ ……………………………………………………………..……………..3.41

$= \frac{28900N}{1229.96N} = 23.5$

Center distance of chain and sprocket is center to center distance of conveyor was 68m. Average velocity of the chain was given by:

$T = \frac{60 \times Vc}{P \times N}$ …………………………………………………..……..…………..……3.42

Where:

D = Pitch diameter of the sprocket, m

p = Pitch of the chain in meters, m

T = Number of teeth

$$T = \frac{60 \times 1.5\text{m/s}}{0.01905\text{mm} \times 300\text{rpm}} = 15$$

The required number of teeth on the large sprocket for power transmission was:

$$V.R = \frac{N_1}{N_2} = \frac{T_2}{T_1} \qquad \text{3.43}$$

$$T2 = 13 \times \frac{360.1\text{rpm}}{300\text{rpm}} = 15.6 \approx 16$$

Service factor (K_S) of conveyor chain was 1.875. Then designed power of transmission chain (P_{dt}) is:

$$P_{dt} = P_c \times K \qquad \text{3.44}$$

$$= 1.875 \times 983.97\text{w} = 1844.95\text{w}$$

Corresponding to a pinion speed of 360.1 rpm, the power transmitted for chain no. 10B is 4191.6W per strand (appendices table D_3). Pitch, roller diameter, width of roller and breaking load of chain number 10B was 15.875 mm, 10.16 mm, 9.65 mm and 22.2 KN respectively (appendices table D_1). Pitch diameters of driver sprocket could be:

$$D_1 = \frac{P}{\sin\left(\frac{180^o}{T1}\right)} \qquad \text{3.45}$$

$$= \frac{15.875\text{mm}}{\sin\left(\frac{180^o}{13}\right)} = 66\text{mm}$$

Pitch diameters of the driven sprocket also:

$$D_2 = \frac{P}{\sin\left(\frac{180^o}{T2}\right)} \qquad \text{3.46}$$

$$= \frac{15.875\text{mm}}{\sin\left(\frac{180^o}{16}\right)} = 81.4\text{mm}$$

Pitch line velocity of driver sprocket (V_1) was:

$$V_1 = \frac{\pi D_1 N_1}{60} = \frac{\pi \times 0.066\text{m} \times 360.1\text{rpm}}{60} = 1.246 \text{ m/s}$$

Load on chain (W) would be:

$$W = \frac{P_{dt}}{V_1} \qquad \text{3.47}$$

$$= \frac{1844.95\text{w}}{1.246\text{m/s}} = 1480.7\text{N}$$

And factor of safety was calculated using the following equation:

$$fos = \frac{W_B}{W} \qquad \text{3.48}$$

$= \frac{22200N}{1480.7N} = 14.99 \approx 15$

The factor of safety is a way of measuring the stress load of the sprockets, which should be higher than 8.55-9.35. The center distance between the smaller and larger sprocket should be at least 30 to 50 times the size of the sprocket pitch. In this example, the average of these two numbers was taken 40 which mean the center distance was 680mm. Finally, the number of links and the length of the chain (L) could be calculated using an equation below.

$K = \frac{T_1 + T_2}{2} + \frac{2x}{p} + (\frac{T_2 - T_1}{2\pi})^2$ ……………………………………………….…………..…..…..3.49

$= \frac{13 + 16}{2} + \frac{2\times40}{15.875mm} + (\frac{16-13}{2\pi})^2 = 95$

$L = K \times p$………………………………………………………………………………..3.50

$= 95\times 15.875mm = 1492.25mm$

The total load on driving side of chain is the sum of tangential driving force (F_t), centrifugal tension on chain (F_c) and the tension due to sagging (F_s) (Kurmi and Gupta, 2005). Hence,

$F_t = \frac{P}{V}$……………………………………………….……….……..……………..….3.51

$= \frac{983.97w}{1.246m/s} = 789.7N$

$F_c = mv^2$………………………………………………………………………...…..…..3.52

$= 1.2kg/m \times (1.246m/s)^2 = 1.863N$

$F_s = Kmgx = 4 \times 1.2kg/m \times 9.81m/s^2 \times 0.680m = 29.67N$

The total tension load on driving of chain (T):

$T = F_c + F_t + F_s$…………………………………..………………………….…….…..3.53

$= 1.863N + 789.7N + 29.67N = 821.23N$

Power required for driving (P_r) was:

$P_r = T\times V$……………………………………………………………….…………..….3.54

$= 821.23N \times 1.246m/s = 1038.04w$

The conveyor power (Pc) transmitted by thus chain and sprocket was 983.97W at input speed of 360.1 rpm and output speed (conveyor rpm) of 300 rpm. A roller chain, minimum number of teeth on the driver sprocket (T_1) for a velocity ratio of 1.2 should be 13. Speed (rpm) and pitch diameter of conveyor sprocket (D) is 300rpm and 81.4mm respectively. The driver sprocket was fixed at output shaft with rpm (N_o) and pitch diameter (D_o) is 360.1rpm and 66.1mm respectively. The Transmission ratio between input and output shaft could be:

$\frac{Nm}{No} = \frac{D_o}{D_i}$………………………………………………………………...………..….3.55

Where:

D_o = pitch diameter of driven gear at output shaft, cm

D_i = pitch diameter of driver gear at input shaft, cm

$$\frac{945\text{ rpm}}{360.1\text{rpm}} = \frac{Do}{Di} = 2.62$$

Distance from the center of the driving gear to the center of the driven gear (C_d) is the sum of the pitch radii of two gears in meshing (Mott, 2004):

$$C_d = \frac{(D_o + D_i)}{2} \quad \ldots\ldots 3.56$$

Space available between output and input shaft (C_d) was 17.25cm. Hence,

$$17.25\text{cm} = \frac{(D_i + 2.3D_i)}{2},\ D_i = 10.5\text{cm},\ D_o = 24\text{cm}$$

The pitch diameter and number of teeth of the driving and driven gear were 10.5 cm, 24 cm, 30, and 70, respectively. This means that the driving gear had 10.5 cm in diameter and 30 teeth, and the driven gear had 24 cm in diameter and 70 teeth. The material used for the gears is called through-hardened steels (AISI 1020), which is a type of steel known for its durability, availability, and cost-effectiveness. The transmitted force, or F_t, and normal force, or F_n, on the gears has determined as below.

$$F_t = \frac{2T_o}{D} \quad \ldots\ldots 3.58$$

$$= \frac{2\times 132.75\text{Nm}}{0.24\text{m}} = 1106.27\text{N}$$

$$F_n = F_t\tan\theta \quad \ldots\ldots 3.59$$

$$= 1106.27\text{N}\times\tan 20^o = 402.65\text{N}$$

The power was transmitted between the shafts using chain and sprocket mechanism that inclined to 20^o to horizontal.

3.4.5.3. Design of Conveyor rod

The length of chain conveyor L_c

$$K = \frac{T_1+T_2}{2} + \frac{2x}{p} + \left(\frac{T_1-T_2}{2\pi}\right)^2 \quad \ldots\ldots 3.60$$

$$K = \frac{16+16}{2} + \frac{2\times 68\text{cm}}{1.905\text{cm}} + \left(\frac{16-16}{2\pi}\right)^2 = 103.39$$

Lc = 97.1×p = 103.39×1.905cm = 196.95cm

Load applied on each conveyor rod (W_r) at a time was:

$$Wr = \frac{2(Q_s)g}{nr} \quad \ldots\ldots 3.61$$

$$= \frac{2\times 88.26\text{kg}\times 9.81\text{m/s}^2}{53} = 32.67\text{N}$$

Maximum bending Moment (M_{br}) at each conveyor rod is:

$$Mbr=\frac{W_r \times L}{2} \qquad \text{3.62}$$

Where:

L= length of conveyor rod, mm

W_r = load acting on each conveyor rode, N

$$M_{br} = \frac{32.67N \times 750mm}{2} = 12,251.25mm$$

Maximum bending stress (σ_b) was:

$$\sigma_b = \frac{32M_{br}}{\pi d^3} \times fos \qquad \text{3.63}$$

$$294.74N/mm^2 = \frac{32 \times 12,251.25Nmm}{\pi d^3} \times 3,\ d = 10.83mm \approx 12mm$$

Diameter of conveyor rod and minimum tuber diameter (D_{min}) was taken 12mm and 25mm respectively.

Spacing (S) between rods was using eqn. (3.64):

$$S=\frac{L_{cs}}{n_r} - Dr \leq Dmin \qquad \text{3.64}$$

$$=\frac{197cm}{n_r} - 1.2cm \leq 2.5cm,\ nr \geq 53rods$$

Where:

S = space between conveyor rods, cm

l_c = total length of conveyor chain, cm

n_r = number of conveyor rode

D_r = diameter f rod, cm

D_{min} = minimum onion tuber diameter (2.5cm),

3.4.5.4. The Materials Conveyed

The mass of soil tuber material flow (Q_t) on conveyor was calculated as:

$$W_s = V_s \times \rho_s \qquad \text{3.65}$$

$$Vs = A_b \times d \qquad \text{3.66}$$

Where:

V_s = volume of soil flow, m^3/s

d = depth of operation, 0.1m

A_b = base area of trapezoidal blade, m^3

ρ_s = density of soil, kg/m^3

$V_s = \frac{0.72m + 0.36m}{2} \times 0.42m \times 0.1m = 0.023m^3$

Ideal bulk density at optimum plant growth was 1.4 g /cm^3 (USDA, 2000):

$Ms = 0.023\ m^3 \times 1400\ kg/m^3 = 32.2kg$

According to Tarek, (2016), the mass of conveyor parts are very small and can be neglecting, the volume of the material flow on the conveyor was:

$Vm = d \times w \times Vf/e$ ……………………………………………………………..…....3.67

Where:

w = Effective width of operation/minor width of blade, 0.36 m

$V_{f/e}$ = relative speed of soil tuber mass and tractor, m/s.

d = depth of operation

Optimum operation depth of onion harvesting is 10cm. Hence,

$Vm = 0.1m \times 0.36m \times 0.743m/s = 0.02675\ m^3/s$

$Ms = Vm \times \rho s$…………………………………………………….………...….…3.68

$= 0.02675m^3/s \times 1400kg/m^3 = 37.45kg/s$

Weight of tuber handled on a conveyor using (Khura, 2008):

$Wt = Wc \times Vf/e \times Y$…………………………………………………….…....……..…..3.69

Where:

W_t = Mass of tuber, kg

w_c = width of conveyor, 0.72m

y = Average yield of tuber, kg/m^2

However, the production and productivity of the crop are far below (10.02t/ha) the world average (19.7t/ha) since, 1 ton is around 1016.05kg.$_f$

$M_t = 0.72m \times 0.743m/s \times \frac{1016.05kg}{1ton} \times \frac{19.7ton}{10000m^2} = 1.071kg/s$

$Q_f = 1.071kg/s + 37.45kg/s = 38.52kg/s$

By equating the volume of material flow with volume of the conveyor, assume the material was spread uniformly on the conveyor with a thick layer (t), using eqn. below:

$Q_f = \gamma \times L_c \times t \times S_c$………………………………………………………………...3.70

$38.52kg/s = 1400kg/m^3 \times Lc \times 0.04m \times 0.743m/s = Lc = 92.6cm \approx 95cm$

The time taken (t) a material remaining on conveyor:

$t = \frac{L_c}{S_c}$………………………………………………………………………………..3.71

$= \frac{0.95m}{0.743m/s} = 1.28s$

Total mass of soil tuber (M_m) carried by conveyor:

$M_m = Q_f \times t$……………………………………………………………………….……….…3.72

$=38.52 kg/s \times 1.28 s = 49.30$ kg

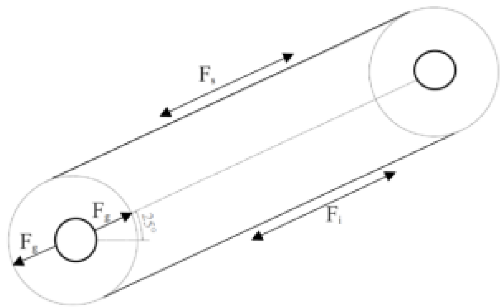


Figure 3.7. Free body diagram of conveyor

The upper chain tension of elevator was using equation below:

$F_s = (Q_t + Q_r + Q_c)\ gf$……………………………………..……………………….....3.73

$= (38.52kg/s + 18.63kg/s + 1.22kg/s)\times 1.5m/s\times(1 + 0.3) = 113.82N$

The chain tension (F_i) could be:

$F_i = \frac{(Qr + Qc)Ve\ (1 + f1)}{2}$…………………………………….………..….……………….…….3.74

$= \frac{(18.63kg/s + 1.22kg/s)\times 1.5m/s\ (1 + 0.125)}{2} = 16.75N$

Considering inclined chain, the lifting chain tension required to lifting tuber soil mass at a constant speed was computed using the following equations:

$F_g = \frac{P_g}{V_e}$…………………….………………………………………………….….……..3.75

$Pg = (Qc + Qr + Qt)gh + \mu(Qc + Qr + Qt)glcos\theta$………………………..…………….….3.76

Where:

P_g = power to lift the material, W

h = height of elevator, m

l= center length of elevator, m

θ = conveyor inclination, °

f_1 = Down chain friction factor (0.10 -0.15)

Q_t = Weight of conveyed material, kg/s

Q_c = Total chain mass, kg/s

Q_r = Total elevator rod mass, kg/s

f = Upper chain friction factor (0.25- 0.35)

μ = friction coefficient between elevator rod and soil-tuber mass

$P_g = (1.22kg/s + 18.63kg/s + 38.52kg/s)9.81m/s^2 \times 0.4m + 0.3(1.22kg/s +$

$18.63kg/s + 38.52kg/s) \times 9.81m/s^2 \times 0.72\cos 25^0$

$= 339.84W$, hence,

$F_g = \frac{339.84w}{1.5m/s} = 226.56N$

Power required to drive conveyor (P_c) was:

$P_c = T \times V_c$……………………………………………………………………………3.77

Where:

T = total tension, N

V_c = velocity of conveyor

$T = F_s + F_i + F_g$……………………………………………………………………………3.78

Where:

T = Total chain tension, N

F_s = Upper chain tension, N

F_i = Down chain tension, N

F_g = Lift chain tension, N

$T = 113.82N + 16.75N + 226.56N = 357.13N$

$P_c = 357.13N \times 1.5m/s = 535.69w$

Generally, conveying and separation unit could be concluded as follows:

The soil crop material dug by a digging unit would be directly forwarded to a conveying and separation unit. The soil separation unit was placed just behind the blade to receive the dugout crop and soil mass. To separate the soil from the tuber, stainless steel of round shape of 12mm diameter were arranged and welded on angle iron which was attached on the chain. The conveyor is mounted at the rear end of the digging blade and consist two shafts with ball bearings fixed apart at the centre distance of 680 mm. Four sprockets of 9.5cm diameter with two chains combination were mounted on the conveyor shafts and the slope of the conveyor was fixed at 20^o. When 2WT moves forward, the digger blade slices the soil and automatically lifts with onion tuber to the conveyor assembly. The soil and tuber passes through the conveyor, the soil drops between the conveyor rods but the onion tuber discharges on the collector which puts the tuber in narrow strips.

3.4.6. Design and Analysis of Output Shaft

The output shaft is a uniform-diameter metal piece with a circular cross-section and small indentations called keyways which was used to transfer power from the input shaft to another component, like a spur gear. This component is then able to drive things like conveyor systems which might be exposed to a combination of both torsional (twisting) and bending forces. Figure 3.8 shows the design of the output shaft, which includes a driven gear, chain, driver sprocket, and bearings. The power (Kw) required to drive the output shaft is calculated by multiplying the torque developed by the speed of the shaft. In addition, the output shaft is subject to horizontal and vertical loads from the belt and chain tensions. These horizontal forces can also be calculated.

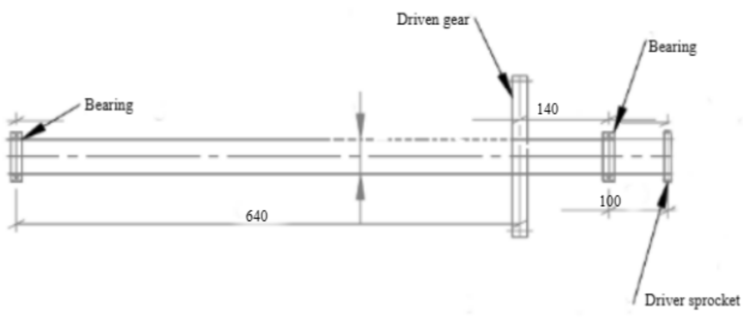


Figure 3.8. Output shaft with a driven gear, chain, driver sprocket and bearings

The power driven by output shaft (P_o) was:

$P_o = P_c = 807.13W$

Hence, torque developed by output shaft (T_o) at N_o=360.1rpm was:

$$T_o = \frac{P_o \times 60}{2\pi \times No} \quad \ldots\ldots 3.79$$

Where:

P = power require to drive, kW

N_o = Speed of the shaft, rpm

F = force required to drive the machine, N

r = radius of shaft, m

$$= \frac{807.13w \times 30}{2\pi \times 0.015 \times 360.1rpm} = 1427.64 \text{ Nm}$$

Bending and twisting moment occur on shaft as a result of applied loads, belt and chain tensions. Out shaft was subjected to both horizontal and vertical load acting on the shaft.

Horizontal forces acting on output shaft was:

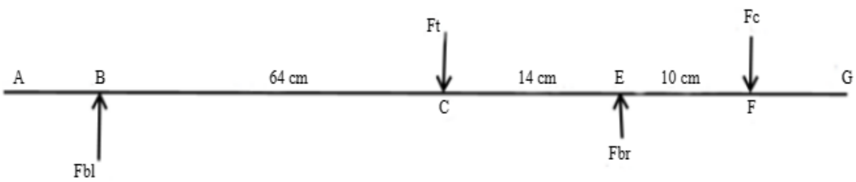


Figure 3.9. Free body diagram of forces on output shaft at horizontal plane

Using trigonometric relations,

F_{cx} = Fccosγ, F_{brx}=Fbrcos∅, F_{blx}=F_{bl}cos∅ and F_t=$F_g$cosε

In order to evaluate horizontal component of bearing reaction forces F_{bl} and F_{br}, was considered bending moment at F:

$\sum M_A=0$

$-F_g\cos\varepsilon \times 0.64m + F_{br}\cos\emptyset \times 0.78m - F_c\cos\gamma \times 0.88m$

$F_g\cos\varepsilon = F_t = 1106.27N$ (Gear force was identified)

$F_c = T = 821.23N$ (chain tension was determined)

The chain drive was inclined to 20^o to horizontal. Hence,

$F_c\cos\gamma=821.23N\times\cos20^o=771.7N$, then

$\sum M_F = 0$

$-1106.27N \times 0.64m + Fbr\cos\emptyset \times 0.78 - 771.7N\times0.88 = 0$

$F_{br}\cos\emptyset = 1778.33N$

Similarly, the horizontal bearing reaction forces F_{bl}, was calculated using total load at E:

$\sum M_E = 0=F_g\cos\varepsilon - F_{bl}\cos\emptyset + F_c\cos\gamma - F_{br}\cos\emptyset$

$1106.27N - F_{bl}\cos\emptyset +771.7N - 1778.33N$

Hence, $F_{bl}\cos\emptyset = 99.64N$

The horizontal bending moments on the output shaft were:

$\sum M_F = F_g\cos\varepsilon \times 0.24m - F_{br}\cos\emptyset \times 0.1m - F_{bl}\cos\emptyset \times 0.88m$

$1106.27N \times 0.24m - 1778.33N\times0.1m - 99.64N \times 0.88 = 0$

$\sum M_E = - F_c\cos\gamma \times 0.08m= - 771.7N\times0.1m= - 77.17Nm$

$\sum M_c = F_{br}\cos\emptyset \times 0.14m - F_c\cos\emptyset \times 0.24m = 1778.33N \times 0.14m - 771.7N \times 0.24m$

$= 63.75Nm$

$\sum M_F = - 1106.27N \times 0.64m +Fbr\cos\emptyset \times 0.78m - Fc\cos\gamma \times 0.88m$

$= -1106.27N \times 0.64m +1778.33N \times 0.78m - 771.7N \times 0.88m = 0$

Based on the magnitude and location of all forces acting on the output shaft, shear force and bending moment on the horizontal plans containing the spur gear and conveyor chain were computed as in figure below.

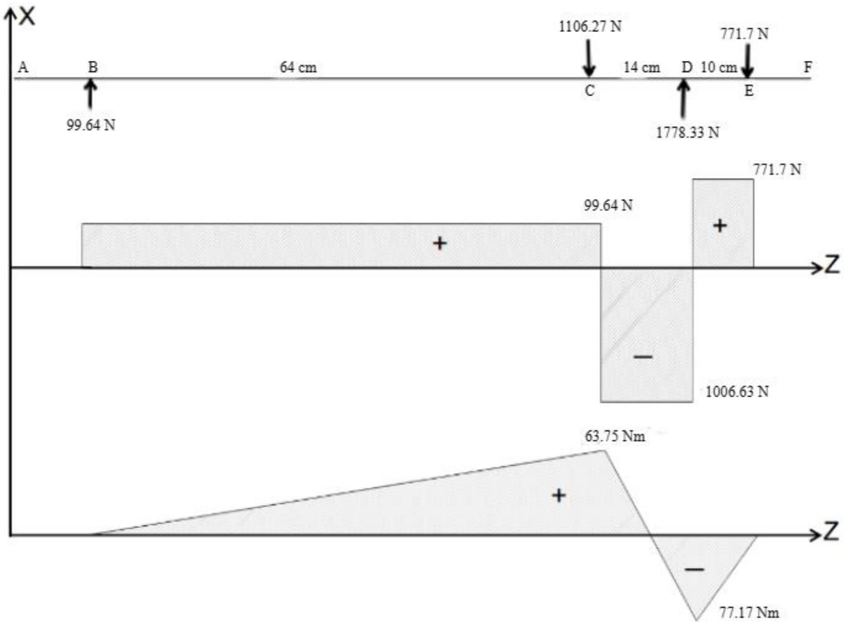


Figure 3.10. Shear force and bending moment diagrams on vertical directions of the shaft

The diameters of the output shaft were determined using maximum shear stress theory using equation. From the above bending moment diagram the maximum bending moment and maximum torque transmitted by output shaft, were 27.54 Nm and 63.75Nm at point G, respectively, Assume K_b = 1.5 and K_t = 1, and allowable Stress, 40MPa (for steel shaft with keyway) was used.

$$d_o^3 = \frac{16}{\pi\tau}\sqrt{(KbMb)^2+(KtMt)^2} \qquad \text{3.80}$$

$$= \frac{16}{\pi\times 40\text{Mpa}}\sqrt{(1.5\times 63750\text{Nmm})^2+(2\times 2750\text{Nmm})^2} = 21.51\text{mm} \approx 25\text{mm}$$

Von Mises is a measure of the stress in a material, given in units of Newton's per square meter. In figure shown below, von Mises stress values for different points in a material. The values are given in scientific notation, where "e" represents "times ten to the power of". The values were ranging from $6.538e^{+05}$ to $8.006e^{+04}$ N/m^2. These values can be used to analyze the behavior of the material under stress and determine if it is likely to fail or deform

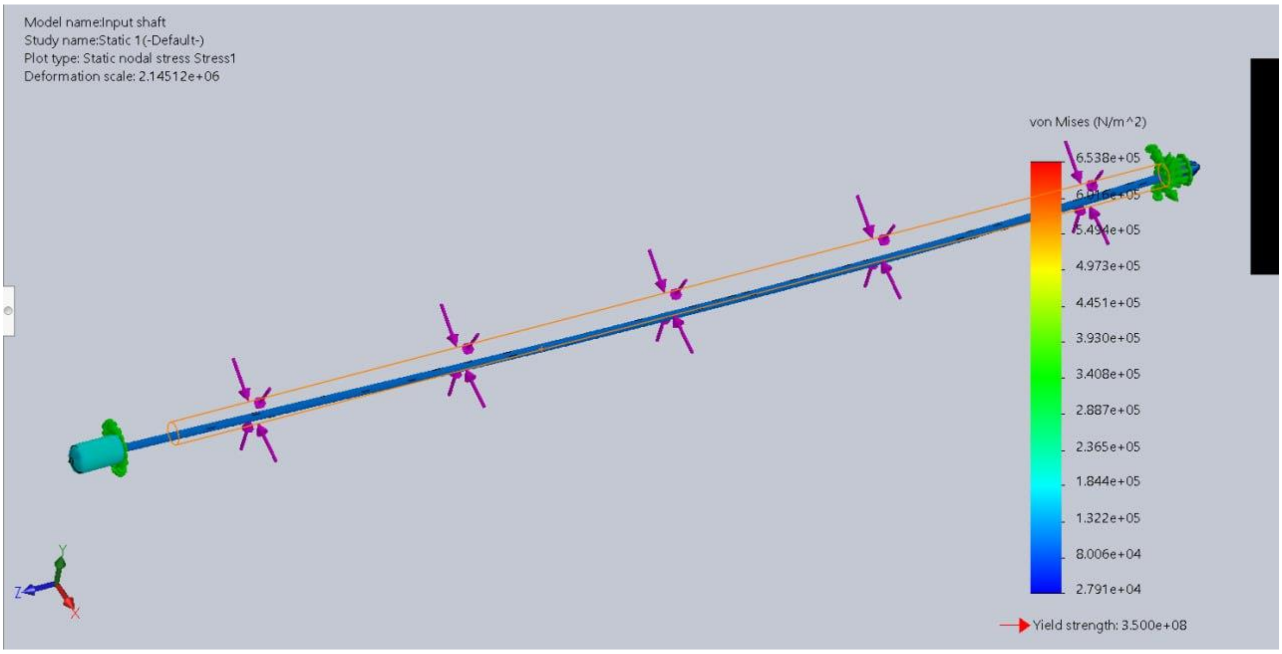


Figure 3.10. Solid work force analysis on output shaft

3.4.7. Design and Analysis of Input Shaft

The input shaft is a cylindrical piece of metal with grooves or "keyways" cut into it that was used to transfer power from the 2WT clutch to other parts in the system. To drive the input shaft, pair of V-belts and pulleys was used to transfer the power from the 2WT clutch. The shaft was then held in place by bearings, with a pulley and driver gear attached that was subject to both twisting and bending forces (as shown in Figure 3.11).

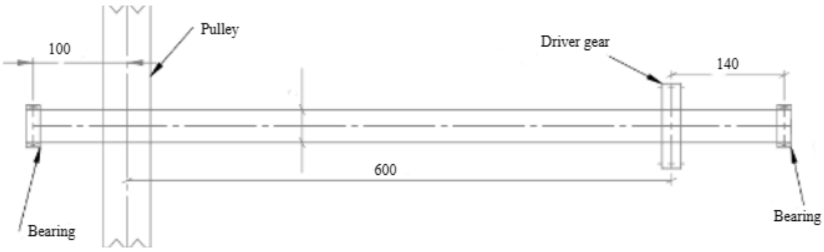


Figure 3.11. Input shaft profile

Load on input shaft are applied vertical and horizontal directions in xyz plane as shown in figure 3.12:

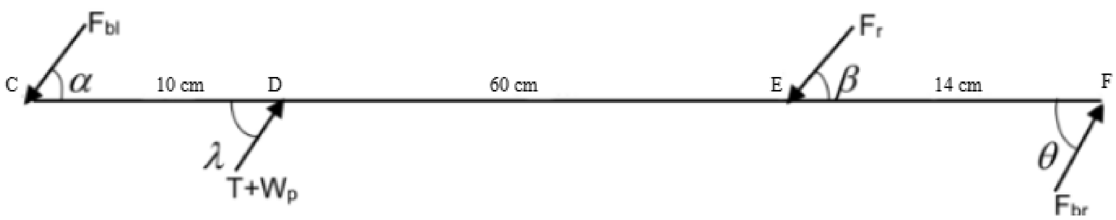


Figure 3.12. The forces acting on input shaft

Shear force and bending moment diagram on horizontal plane was:

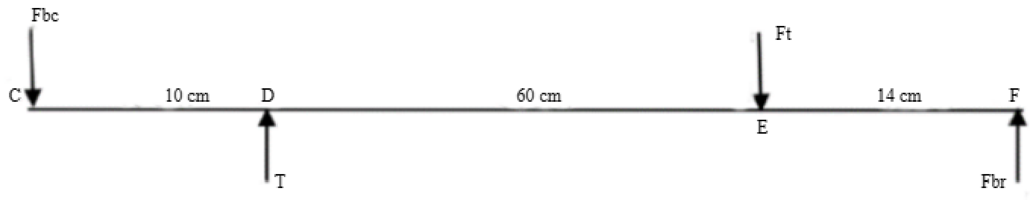


Figure 3.13. Free body diagram of the input shaft on horizontal plane

Using trigonometric relation:

$F_{bc} = F_b\cos\alpha$, and $F_{bc} = F_{br}\cos\theta$…………………..……………………………….……..…..3.81

Gear forces and belt tension was:

F_t=1106.27N, F_n = 402.65N and $T = T_1+T_2$ = 668.95N

Determine reaction forces F_{brx} and F_{blx}, through moment at L and O was:

$\sum M_c = (T_1+T_2) \times 0.1m - F_t \times 0.70m + F_{br}\cos\theta \times 0.84m = 0$

$= (668.95N) \times 0.1m - 1106.27N \times 0.70m + Fbr\cos\theta \times 0.84m$

$F_{br}\cos\theta = 842.25N$

$\sum Fx = - T + F_t + F_{bc}\cos\alpha - F_{br}\cos\theta$

$=-668.95N + 1106.27N + Fbc\cos\alpha - 842.25N$

$F_{bc}\cos\theta = 404.93N$

Then, the horizontal moment of input shaft would be:

$\sum M_E = F_{br}\cos\theta \times 0.14m = 842.25N \times 0.14m = 117.91Nm$

$\sum M_D = - F_t \times 0.60m + F_{br}\cos\theta \times 0.74m$

$= -1106.27N \times 0.60m + 842.25N \times 0.74m = - 40.5Nm$

$\sum M_c = (T_1+T_2) \times 0.1m + F_{br}\cos\theta \times 0.84m - F_t \times 0.70m$

$=668.95N \times 0.1m + 842.25N \times 0.84m - 1106.27N \times 0.70m = 0$

Based on the magnitude and location of forces acting on the input shaft, shear force and bending moment on the horizontal plane of the shaft were computed and plotted as follows:

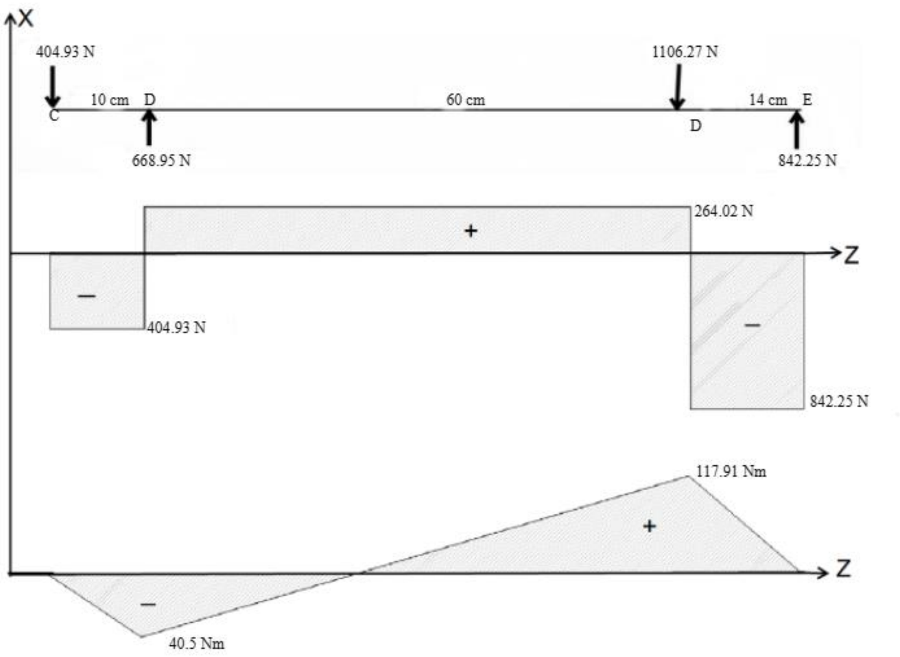


Figure 3.14. Horizontal plane shear and bending moment diagrams on the shaft

Shear force and bending moment diagram on vertical plane was:

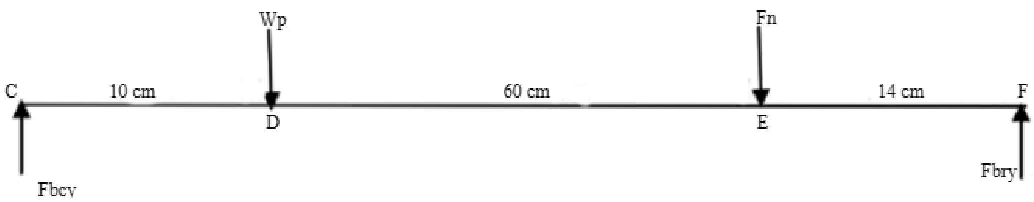


Figure 3.15. Free body diagram of the input shaft on vertical plane

Using trigonometric relations:

$F_{bc}\sin\alpha = F_{bcy}$, and $F_{br}\sin\theta = F_{bry}$……………………………………………..…..…..….3.82

Assume no slip of gear and belt, the vertical component of bearing reaction force were computed through moments at C and F. respectively:

$\sum M_c = 0 = W_p \times 0.1m - F_n \times 0.70m + F_{br}\sin\theta \times 0.84m$

$\quad = 22.84N \times 0.1m - 402.65N \times 0.70m + Fbr\sin\theta \times 0.84m$

$F_{br}\sin\theta = 332.82N$

Equilibrium of vertical forces (F_y):

$\sum F_y = - F_{bcl}\sin\theta + W_p + F_n - F_{br}\sin\theta = 0$

$-F_{bcl}\sin\theta + 22.84N + 402.65N - 332.82N$, then $F_{bcl}\sin\theta = 92.67N$

Total bending moments of input shaft on vertical plane

$\sum M_E = F_{br}\sin\theta \times 0.14m = 332.82N \times 0.14m = 46.59Nm$

$\sum M_D = - F_n \times 0.60m + F_{br}\sin\theta \times 0.74m$

$= -402.65N \times 0.60m + 332.82N \times 0.74m = 4.7Nm$

$\sum Mc = W_p \times 0.1m - F_{br}\sin\theta \times 0.84m + F_n \times 0.70m$

$= 22.84N \times 0.1m + 402.65N \times 0.70m - 332.82N \times 0.84m = 4.57Nm$

Shear force and bending moment diagram on vertical plane

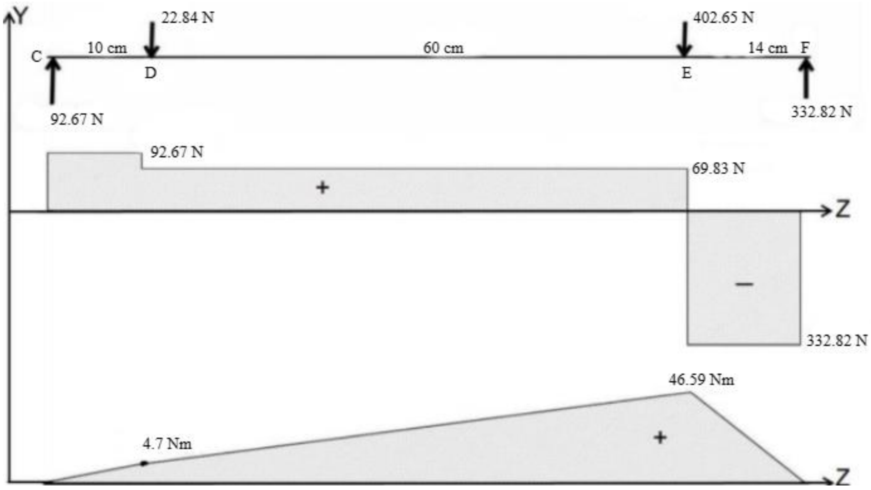


Figure 3.16. Vertical plane shear and bending moment diagrams on input shaft

The maximum torque on the input shaft (M_t) developed through machine pulley was: T_t =530.45N and T_s=138.5Nm),

$$M_t = (T_t - T_s)D_p = (530.45N - 138.5N) \times 0.255m = 99.95Nm$$

Using eqn. (3.60), the diameters of the input shaft were determined as follows:

$$= \frac{16}{\pi \times 40Mpa}\sqrt{(1.5\times117910\ Nmm)^2+(99950\ Nmm)^2} = 28.17mm \approx 30mm$$

A model is a representation of a system or process used to study its behavior or performance. "Input shaft" is the name given to the model being studied. "Static 1" refers to a type of study where the system is analyzed under static conditions, i.e., when it is not in motion. "Static displacement Displacement1" refers to the type of plot being generated, which shows the displacement of the system under static conditions. Deformation scale is a measure of the amount of deformation or distortion in the system being studied. In this case, the deformation scale used is 2:14512e^{+06}.

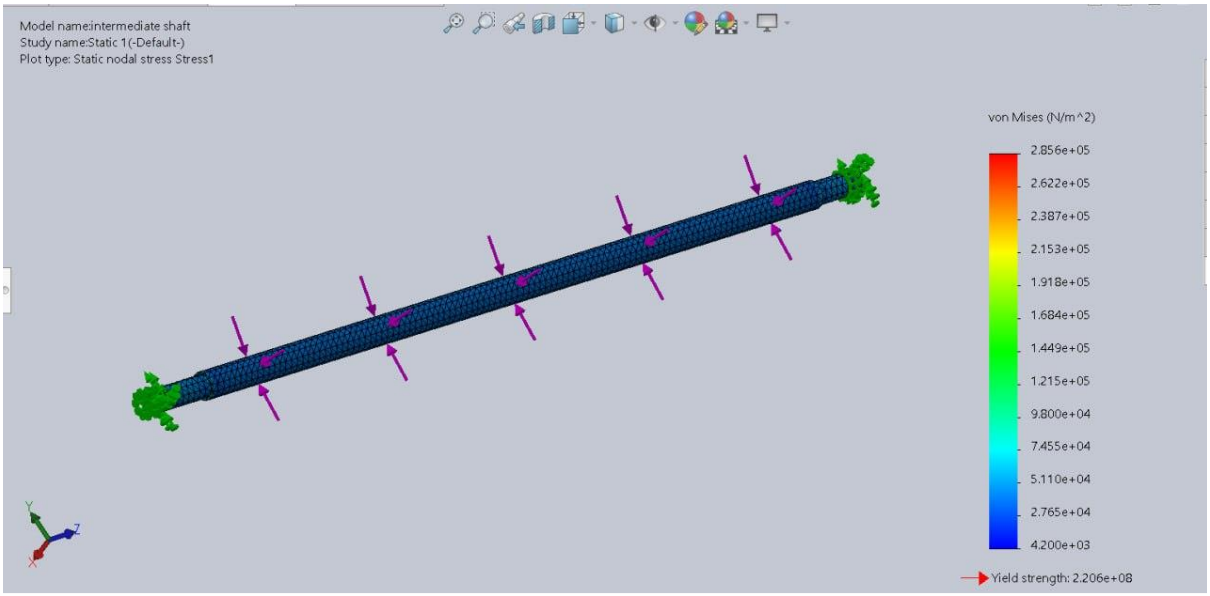


Figure 3.17. Solid work force analysis on input shaft

3.4.8. Selection of V-belt and Pulley

3.4.8.1. Selection of Pulley

The driver pulley and driven pulley are components of a two-wheel tractor's engine and a machine, respectively. The diameter of the clutch, which connects the driver and driven pulley was 21.2mm. The rpm of the driver pulley/clutch of the tractor and the driven/machine pulley were 1162.8 rpm and 961.5 rpm, respectively. The diameter of the driven pulley was calculated using equation (3.63) based on the necessary rpm.

$N_c \times D_c = N_m \times D_m$……………………………………………………………………….3.83

$$D_m = \frac{N_c \times D_c}{N_m} = \frac{1162.8\text{rpm} \times 212\text{mm}}{961.5\text{rpm}} = 255\text{mm}$$

Aluminum pulley with 255mm diameter was purchased from market

3.4.8.2. Selection of V-belt

The V-belt drive transmits power from a motor attached to a clutch to a machine such as a pulley. In this particular case, the net power being transmitted was 11.03 kW at a speed of 1170.73 rpm. The machine was running 8 hours a day, and a correction factor was applied that was equal to 1.2. Hence, design power (P_{ds}) was (Bhandari, 2010):

$P_{ds} = P_{tr} \times f_a$……………………………………………………………………………..3.84

$= 10.3 \times 1.2 = 12.36$ Kw

The selected belt used to drive 12.36kW design power and 1170.73rpm was B-section V-belt (appendix graph F2). Therefore, diameter of driver (d) (clutch) and driven (D) pulley (machine pulley) were 212mm and 255mm respectively. Pitch length of belt (Lb) using equation below.

$$L_b = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C} \quad \ldots\ldots 3.85$$

$$= 2450mm + \frac{\pi(255mm + 212mm)}{2} + \frac{(255mm - 212mm)^2}{4C} = 1634.22mm$$

The required pitch length for B section belt was around 1700mm (appendix table E3). Using quadratic equation theorem, the corrected center distance was:

$$1700mm = 2 \times C + \frac{\pi(255mm + 212mm}{2} + \frac{(255mm - 212mm)^2}{4C}$$

$$2C^2 - 966.81C + 462.25 = 0,\ C = 483mm$$

Determination of the number of belts required was estimated as follows:

The speed ratio (S_r) was:

$$S_r = \frac{1170.73rpm}{800rpm} = 1.46$$

3.4.9. Selection of Bearings

The bearing that is best for the machine's nature, the area inside the machine, the spindle specification, the lubrication system, and the drive system must take into account the bearing's design life, precision, rigidity, critical speed, and other properties. A square flange and four drill holes for attaching bolts with shaft sizes ranging from 15 to 100 mm are present in the housings. For metric shafts, there are two sets of bearing units depending on the housing design (ASTM, 2014).

3.4.10. Depth Control Systems

According to Dessye Belay, (2021), the mechanism for depth control system was constructed to set up variation of harvesting depth. The sliding arm with whole machine body slides by rotation of screw to change depth of operation during transport as well as operations. The power screw rotates manually by a handle, which fixed on both side of operator site on side flanges. Acme threaded screw with 28 mm; 13.5mm and 4mm outer diameter, inner diameter and pitch diameter respectively were selected from screw selection table (appendices table F9). Angle of acme threads (2β), where, $\beta = 14.5°$ (Bansal, 2009). The screw with single start acme threads of 28mm nominal diameter and 5mm pitch is suitable for depth control system of the machine (Dessye Belay, 2021). The depth of the lift arm varied within a range of zero to 150mm vertically as shown figure 3.13.

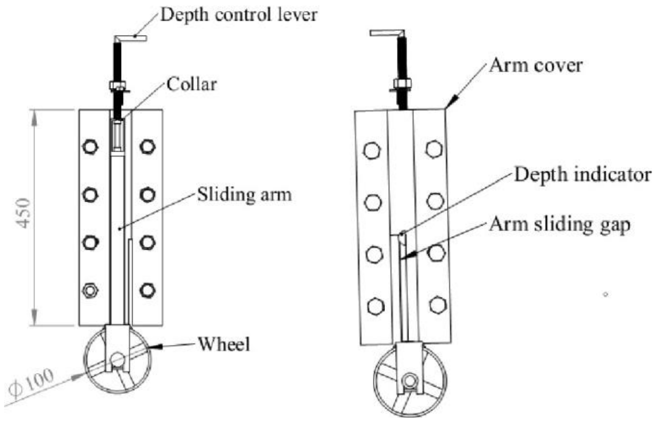


Figure 3.17: Depth control system

3.4.11. Cutting disc

Cutting disc has the diameter of 333mm and 6mm thickness was sharpened using grinder for easily slicing the soil.

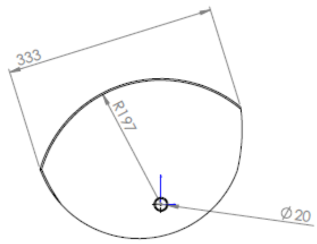


Figure 3.18. Cutting disc drawing

3.4.12. Collector

The collector of 900 mm lengths, 30mm width and 2mm thickness has used for collecting the onion tuber from conveyor and attached to the side plates of the machine with angle of 30^o.

3.4.13. Determination of Total Transmission Ratio

The field speed of onion harvester was 0.833m/s. The ratio of conveyor speed of onion digger was similar to potato digger which was 1.2 -1.5 (Schweers *et al.*, 2015). Maximum conveyor speed should be 1.52m/s at 1-1.5 of forward speed. Finally, to meet 1.5m/s conveyor speed, simultaneous speed reduction was applied.

From 15hp of manufacturers' catalogue N_e, D_e and D_c of 2WT was 2000 rpm, 12cm, and 20.5cm respectively. The power transmission from 2WT engine flywheel to clutch is through variable V-belt. Using rpm of clutch was:

$$N_c = \frac{Ne \times De}{Dc} \quad \text{……3.86}$$

Where:

D_e =diameter of engine pulley

D_c= diameter of clutch

N_e=rated engine rpm

$$= \frac{2000\text{rpm} \times 12\text{cm}}{20.5\text{cm}} = 1170.73\text{rpm}$$

The engine power at a clutch of a 2WT was transmitted to input shaft through V-belt and pulley and also it drives output shaft through a combination of spur gear, then to conveyor systems. Machine pulley was fixed at input shaft with diameter of 30cm and rpm of machine pulley (N_m) was:

$$N_m = \frac{Nc \times Dc}{Dm} \qquad \text{3.87}$$

$$= \frac{1170.73\text{rpm} \times 20.5\text{cm}}{30\text{cm}} = 800\text{rpm}$$

3.4.14. Power Transmission System

The power transmission system was used to help the 2WT perform its functions. It had two stages - first was the use of a V-belt, which connected the 2WT pulley to a machine-driven pulley of 255mm diameters. This changed the direction of rotation from counterclockwise to clockwise. The second stage involved a combination of spur gears which were used to reduce the speed. The conveyor system was then used to carry and drop onion tubers on the ground in narrow strips as the 2WT was moved forward.

3.4.15. Determination of Total Power

The power at the transmission input (P_i) is 91.08% of the engine brake power of the 2WD tractor (Pb). The brake power of the 2WD Sifang G121-15 tractor was 11.03 Kw.

$$P_b = \frac{P_i}{\eta_w} \qquad \text{3.88}$$

Pi = 10.046kw

Total power (P_{dt}) required driving the machine could be:

$$P_{dt} = P_{is} + P_t \qquad \text{3.89}$$

Power at input shaft (P_{is}) was equal to power at output shaft (p_o) by considering no friction as well as churning and windage losses of gear. Power at mechanical system of the tractor (P_t) and power at input shaft (P_{is}) were 3126.61W and 5124.57W as analyzed respectively. Therefore, the total power required to drive the machine was:

P_{dt} = 3126.61w + 5124.57w = 8251.18w

3.5. Performance Evaluation of the Onion Harvester Machine

Performance evaluation of the machine was conducted in KARC research site because of AAERC have not their own research practice site. Onion was planted on both raised and flat beds in KARC. Planting of onion was done with a manual row planting. The onion crop was ready for harvesting after 120 days of planting. Harvesting was carried in the month of January 2022 by onion digger mounted on 15hp Sifang power tiller. Field tests were conducted in the month of January 2022, and average digging depth was 10cm.

3.6. Data Collection and Variables

3.6.1. Independent variables

a) Forward speed

During performance evaluation, the onion digger prototype machine was attached to 2WT drawbar hitch. 1.39 km/hr, 2.47km/hr, 4.15km/hr speed was used during performance evaluation of the prototype.

b) Depth of operation

Depth of operation used during performance evaluation of onion digger was 5cm, 7cm and 10 cm.

3.6.2. Dependent Variables

a) Percentage of damage

Damage occurs during digging, loading, and transporting operations. The damage resulting of impact on tubers during harvesting operations alone may cause losses in excess of 20% (Storey and Davies, 1992). Damage to bulb within permissible limit was less than 5 per cent (Basaravaj, 2020). Percentage of damage was determined on 2m length of row randomly marked and collected and counted all the visible tubers through collector of onion digger and surface. The damaged tubers were separated and counted and undamaged potato tubers after harvesting and recorded as percentage of damage. The damage percentage could be calculated using equation below (Ibrahim *et al*, 2008).

$$\eta_d = \frac{Md}{Mnd + Md} \times 100 \quad \text{……3.90}$$

Where:

M_d = Mass of seriously damaged or cut root crop, kg

M_{nd} = Mass of root crop exposed and not damaged, kg.

b) Fuel consumption

Before and after the digging operation, the fuel tank of the 2WT was filled to the neck with fuel. The fuel consumption for digging operations was assessed after the test and was expressed as L/hr. The fuel used to refill the vehicle was measured in milliliters. According to Macmillan (2002) fuel consumption was:

$F_c = \frac{C_r}{A} \times C_1$……………………………………………………..………..…………3.91

Where:

F_c = fuel consumption rate, l/ ha

C_r = reading of cylinder, l.

A = area, m^2

C_1 = conversion factor

d) Wheel slip ($S_{\%}$)

The wheel slip of the machine with load and without load was measured at 15m distance. Wheel slip can be calculating by counting the number of revolutions in a given distance (In terms of distances for a given number of wheel revolutions) and recorded as $S_{\%}$ in %Wheel slip can be calculating by counting the number of revolution in a given distance and expressed as (Macmillan, 2002):

$S = \frac{m_o - m}{m_o} \times 100$…………………………………………………………..……...….3.92

$m_o = 2\pi r N w$…………………………………………………………………………..….3.93

Where:

N_w = number of wheel revolution, rpm

r = radius of wheel, m

m = travel distance measured (length of test truck), m

e) Field capacity

Field capacity was an actual field capacity computed by the actual average time consumed during digging operation (lost time + productive time) and recorded in ha/h Effective field capacity of the machine is calculating as given below (Moayad et al., 2014).

$Efc = \frac{A}{T_p + T_1}$ …………………………………………………………………….……..…3.94

Where:

Efc = Effective field capacity, ha/ h

A = Area covered, ha

T_p = Productive time, h

T_1 = Non-productive time, h

f) Field efficiency

Field efficiency can estimated as (Moayad et al., 2014)

$Fe = \frac{Efc}{C_t} \times 100$……………………………………………………………..…..…….3.95

Where:

Fe = Field efficiency, %

Efc = Effective field capacity, ha/hr

C_t = Theoretical field capacity, ha/ hr

S = Speed of operation, km/hr

g) Drawbar pulls

The draft requirement of a given machine can be measured using drawbar dynamometer attached to the machine and the power sources or using two tractors and a dynamometer between them. The first tractor was power sources while the second one, which was towed and served as machine/digger carrier. Drawbar pull requirement (Df), a machine was measured using drawbar dynamometer using two tractors and a dynamometer between them; the first tractor was power sources while the second one was 2WT towed served as machine/digger attached to it and recorded as D_f in kN.

h) Missing percentage

The missed onion tuber is harvested by hand and its percentage was calculated according to the data obtained from experimental field.

I) Digging efficiency

Digging efficiency was calculated using the following formula:

$$\text{Digging efficiency} = \frac{A}{A+B} \times 100 \qquad \ldots 3.96$$

Where:

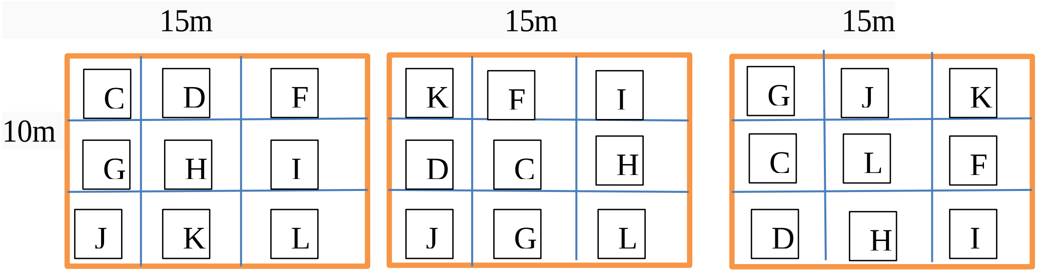
A = mass of tuber dugout by harvester in unit area, kg

B = mass of tuber left in soil after harvesting in unit area, kg

After each test run a sample area of 3m x 3m was demarcated at three places randomly. The sampling area was thoroughly cleaned and the weight of tuber, both exposed and covered with soil was recorded. The result was presented on percentage basis. A higher percentage of tuber harvested indicate better performance of harvester (Wajire et al., 2018).

3.7. Experimental Design

Performance evaluation of the onion digger was through three levels of forward speed: S_1 = 1.39 km/he, S_2 = 2.47km/hr and S_3 = 4.15km/hr as subplot factor and three levels of depth of operation: D_1 = 5cm, D_2 = 7cm and D_3 = 10cm as main plot factor were used. The experimental design was split-plot design with 3^2 factorial experiments and three replicates as block giving 27 total experimental units. The test runs were carried out on a 57 x 36 m field with a 10 x 15 m plot size.



Where:

C= F1*D1 D=F2*D1 F=F3*D1
G=F1*D2 H=F2*D2 I=F3*D2
J=F1*D3 K=F2*D3 L=F3*D3
F=Forward speed and D= depth of operation

3.8. Data Analysis

Using Microsoft Excel (2010), Statistix 8.0, and Rstudio, a for-profit software program created by the USDA, all the data gathered during the laboratory and field performance evaluations were examined. The data were subjected to an analysis of variances using a method suitable for the experiment's design (Gomez, 1984).

3.9. Cost of production

Table 3.4. Cost of materials

No.	Material	Quantity	Unit cost(birr)	Total cost
1	70x50x800mm square pipe	1	850.00	850.00
2	450x750x4mm mild steel sheet metal	2	700.00	1400.00
3	900x30x2mm mild steel sheet metal	2	530.00	1060.00
4	500x30x30mm angle iron	2	150.00	300.00
5	Φ12x680mm round bar	50	22.50	1125.00
6	Φ255x30mm Aluminum pulley	1	800.00	800.00
7	V–belt, B-72	2	120.00	240.00
8	Bearing with housing UCP 206	8	120.00	960.00
9	Φ30xmm mild steel shaft	2	250.00	500.00
10	Chain, B-10	1	650.00	650.00
11	Chain, B-12	2	750.00	1500.00
12	Φ80x5mm Sprocket	6	140.00	840.00
13	Φ3.2mm Electrode	2 Pac	160.00	320.00
Total material cost				**10,815.00**

Table 3.5. Fabrication and labor cost

No.	Activities	Working time(hr.)	Labor cost (birr/hr.)	Cost (birr)
1	Rolling	25	45	1125.00
2	Cutting	30	48	1440.00
3	Welding	20	22	440.00
4	Grinding	26	30	780.00
Total fabrication cost for labor				**3785.00**

Table 3.6. Machine cost

No.	Equipment/machine	Working time (hr.)	Machine cost/hour	Cost (birr)
1	Grinding machine	42	16	672.00
2	Shearing machine	16	125	2000.00
3	Drilling machine	35	28	980.00
4	Rolling machine	14	18	252.00
5	Arc welding machine	38	16	608.00
6	Power ax saw	48	13	624.00
7	Lathe machine	24	30	720.00
Machine cost				**5856.00**

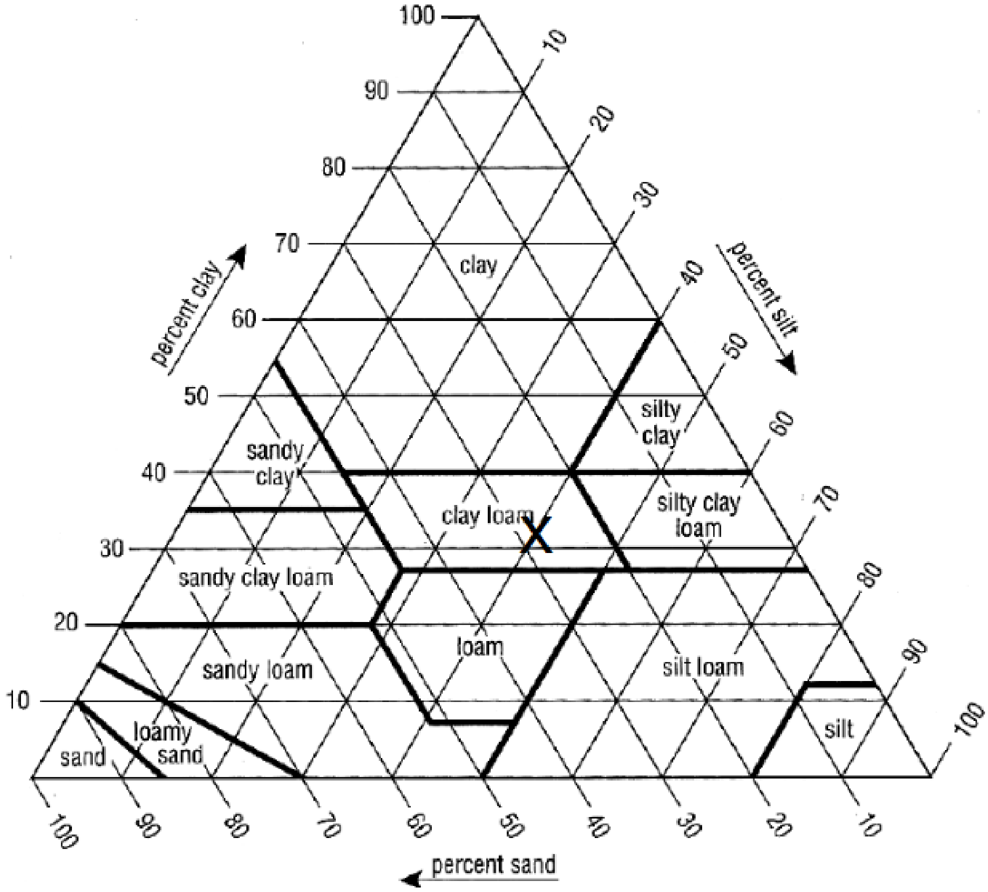
Thus, the production cost 20,456.00 Birr was calculated.

CHAPTER FOUR

RESULT AND DISCUSSION

4.1. Physical Properties of Soil

During harvesting, soil samples were collected from the field for study of the soil properties. Using samples taken from the top 10cm of the soil, the soil texture was assessed using the hydrometer method, and the results were shown in Appendix A. The predominant soil type in the area was a clay loam, which had 28% sand, 40% silt, and 32% clay. Soil moisture and bulk density were determined by using oven drying methods of samples at 105^{o} for 24hr. The average moisture content and the average bulk density of soil were found to be 11.20 % and 1.141g/cm^3 respectively at harvesting time (Appendix B). The shear strength and penetration resistance of soil at depth of 10cm of the experimental field was found to be 0.168kN/cm^2 and 0.391kN/cm^2.



Source: The USDA soil textural triangle (Soil Survey Division Staff 1993)

Figure 4.1. Soil classification

4.2. Physical Properties of Onion

A 2WT operated onion harvester was designed and fabricated at AAERC based on physical and mechanical properties of the onion tuber and soil. The measurements undertaken were revealed that, moisture content, polar diameter, equatorial diameter, thickness, arithmetic mean, Sphericity, volume, density were 68.23%, 52.4mm, 38.65mm, 25.97mm, 57.40mm, 72.76%, 85.32cm^3, 2.85g/cm^3 respectively (Appendix table C1). The static coefficient of friction on galvanized steel sheet, mild steel sheet, Aluminium sheet and plywood surfaces were 0.41, 0.38, 0.43, 0.27 respectively (Appendix table D).

4.3. Performance Evaluation Onion Harvester Machine

4.3.1. Damage Percentage

Table 4.1 shows that at a forward speed of 1.39km/hr, the tuber damage was highest at a depth of operation 5cm (4.15%), followed by 7cm (2.48%) and 10cm (1.13%). At a forward speed of 2.47km/hr, the tuber damage was highest at a depth of operation of 5cm (4.46%), followed by 7cm (2.57%) and 10cm (1.25%). At a forward speed of 4.15km/hr, the tuber damage was highest at a depth of operation of 5cm (4.56%), followed by 7cm (2.64%) and 10cm (1.54%). This implies that as depth of operation increases mean onion tuber damage decrease while as forward speed increases; mean onion tuber damage percentage also increases which was the same as the result of Mohamed Ibrahim (2011). Generally, maximum value of damage percentage of onion tuber was 4.56 % at forward speed of 4.15km/h and digging depth of 5cm, while the minimum value was 1.13% at forward speed of 1.39 km/h and digging depth of 10 cm. However, the interaction effect of (Fs) and D on damage percentage of tuber was not significantly affected (P>0.05) at which grand mean, CV(Rep*Depth) and CV (Rep*Depth*Fs) was 2.75, 15.16 and 5.06 respectively as shown in ANOVA (Appendix table E1).

Table 4.1. Means with SE of damage percentage at various Fs and

Treatments	Depth of operation (cm)		
F_s(km/hr)	5	7	10
1.39	$4.15^a \pm 0.52$	$2.48^b \pm 0.17$	$1.13^c \pm 0.10$
2.47	$4.46^a \pm 0.38$	$2.57^b \pm 0.22$	$1.25^c \pm 0.18$
4.15	$4.56^a \pm 0.42$	$2.64^b \pm 0.23$	$1.54^c \pm 0.36$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and depth of operation of the machine. From regression coefficients (Appendix table F1), their relationship could be expressed by the following equation.

$$\text{Dam }\% = 0.163F_s - 0.601D + 6.833,\ R^2 = 0.91$$

Based on the regression equation above, the percentage of damage is the dependent variable and the depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is -0.601, which means that for every unit increase in the depth of operation, the percentage of damages decreases by 0.601%. The coefficient of the forward speed (F_s) is 0.163, which means that for every unit increase in the forward speed, the percentage of damage increases by 0.163%. The coefficient of determination (R^2) is 0.91, which indicates that 91% of the variation in the percentage of damage can be explained by linear relationship between the depth of operation and forward speed. In simple terms, the equation shows that the percentage of damage to the onion bulbs during harvesting is influenced more by the depth of operation than the forward speed of the machine.

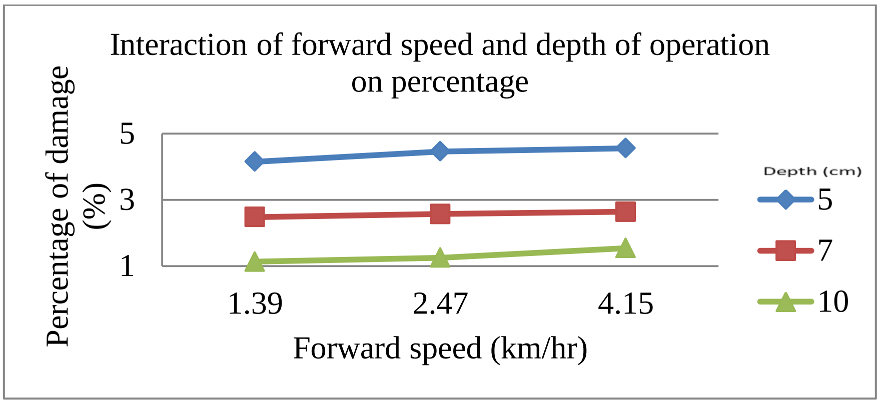


Figure 4.2. Effect of forward speed and depth of operation on damage percentage

4.3.2. Digging Efficiency

The digging efficiency is related with the raised damaged and undamaged onion tubers. Table 4.2 shows that the digging efficiency of an onion harvester at different forward speeds and depths of operation. The ANOVA indicated that, digging efficiency of the machine was significantly affected by forward speed and depth of operation at ($P < 0.05$). The results show that the digging efficiency of the harvester varies with the forward speed and depth of operation. At a forward speed of 1.39km/hr and a depth of operation of 10cm, the harvester had the highest digging efficiency of 97.03%, while at a forward speed of 4.15km/hr and a depth of operation of 5cm, the harvester had the lowest digging efficiency of 84.60%.

Generally, digging efficiency was decreased by increasing the speed from 1.39 – 4.15km/h, but the digging efficiency was increased with increasing the digging depth of operation from 5cm-10cm which was the same as the result of Mohamed Ibrahim (2011). However, the interaction effect of forward speed (Fs) and depth of operation on digging efficiency of harvester was not significantly affected ($P > 0.05$) at which grand mean, CV (Rep*Depth) and CV (Rep*Depth*Fs) was 92.41, 0.98 and 0.71 respectively as shown in ANOVA (appendix table E2).

Table 4.2. Means with SE of digging efficiency at various forward speed and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$87.04^{e} \pm 0.62$	$95.52^{bc} \pm 1.03$	$97.03^{a} \pm 1.06$
2.47	$86.17^{e} \pm 0.37$	$95.40^{c} \pm 0.62$	$96.63^{ab} \pm 0.91$
4.15	$84.60^{f} \pm 0.61$	$93.67^{d} \pm 0.77$	$95.64^{bc} \pm 0.63$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and depth of operation of the onion digger machine. From regression coefficients (appendix table F2), their relationship could be expressed by the following equation.

$\eta_{dig} = 1.975D - 0.945F_s + 79.82$, $R^2 = 0.76$

The equation is in the form of a linear regression model, where the percentage of digging efficiency is the dependent variable and the depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is 1.975, which means that for every unit increase in the depth of operation, the percentage of digging efficiency increases by 1.975%. The coefficient of the forward speed (F_s) is -0.945, which means that for every unit increase in the forward speed, the percentage of digging efficiency decreases by 0.945%. The coefficient of determination (R^2) is 0.76, which indicates that 76% of the variation in the percentage of digging efficiency can be explained by the depth of operation and forward speed. In simple terms, the equation shows that the digging efficiency is influenced more by the depth of operation than the forward speed of the machine.

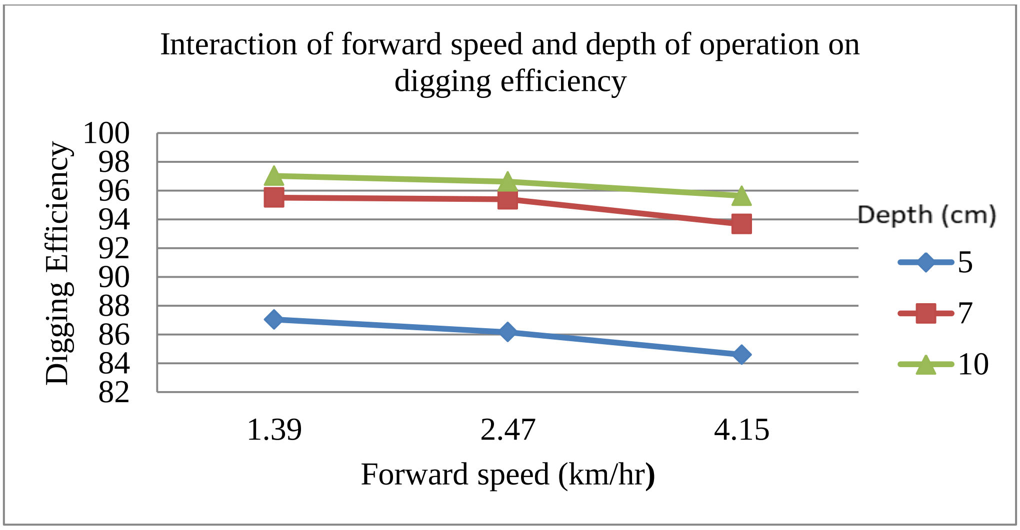


Figure 4.3. Effect of forward speed and depth of operation on digging efficiency

4.3.3. Missing Percentage

Table 4.3 shows that the means and standard errors (SE) of missing percentage of onion digger at various forward speeds and depths of operation. The percentage of missing tubers decreased with increasing the speed from 1.39 km/h to 2.47 km/h. After increasing speed from 2.47 to 4.15 km/h, the missing tubers percent increased due to increasing the slip. It was noticed that with increasing the digging depth, the missing tubers decreased due to the distance between the top of the line and the lowest point of the tubers range from 5 to 10 cm. The maximum value of missing tubers was 11.36 % at forward speed of 1.39 km/h and digging depth of 5 cm, while the minimum value was 3.47 % at forward speed of 2.47 km/h and digging depth of 10 cm. However, the interaction effect of forward speed (Fs) and depth of operation on percentage of missed onion tuber was not significantly affected (P>0.05) at which grand mean, CV(Rep*Depth) and CV(Rep*Depth*Fs) was 8.135, 15.44 and 21.78 respectively as shown in ANOVA (appendix table E3).

Table 4.3. Means with SE of missing percentage at various forward speed and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$11.36^{a} \pm 0.17$	$9.707^{ab} \pm 0.29$	$7.157^{b} \pm 0.31$
2.47	$7.257^{b} \pm 0.35$	$7.11^{b} \pm 0.24$	$3.470^{c} \pm 0.14$
4.15	$10.76^{a} \pm 0.12$	$9.047^{ab} \pm 0.04$	$7.347^{b} \pm 0.05$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and depth of operation. From regression coefficients (Appendix table F3), their relationship could be expressed by the following equation.

$\%_{miss} = -0.762D - 0.178F_s + 14.08$, $R^2 = 0.35$

The equation is expressed as a linear regression model, where the missing percentage is the dependent variable and the depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is -0.762, which means that for every unit increase in the depth of operation, the missing percentage decreases by 0.762%. The coefficient of the forward speed (F_s) is -0.178, which means that for every unit increase in the forward speed, the missing percentage decreases by 0.178%. The coefficient of determination (R^2) is 0.35, which indicates that 35% of the variation in the missing percentage can be explained by the depth of operation and forward speed. Based on the regression equation above, depth was found to be greatest negative impact for the contribution of missing percentage.

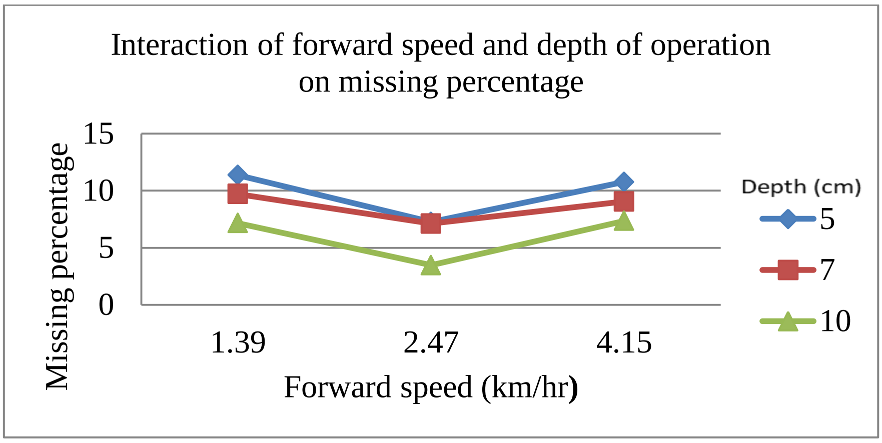


Figure 4.4. Effect of forward speed and depth of operation on missing percentage

4.3.4. Drawbar Pull

Table 4.4 shows that the means with standard errors (SE) of the onion digger at various forward speeds and depths of operation. The table also shows that the highest mean value of the onion digger was observed at 4.15 km/hr forward speed and 10 cm depth of operation, which was 2935.4 N. The lowest mean value was observed at 1.39 km/hr forward speed and 5 cm depth of operation, which was 2772.4 N. The results imply that draft increased with increase of depth of operation and forward speed which was the same as the result of Mohamed Ibrahim (2011). However, the interaction effect of forward speed (Fs) and depth of operation on drawbar pull was not significantly affected ($P > 0.05$) at which grand mean, CV

(Rep*Depth) and CV (Rep*Depth*Fs) was 2845.1, 1.64 and 0.21 respectively (Appendix table E5).

Table 4.4. Means with SE of DF (N) at various forward speeds and depths of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$2772.4^{g} \pm 22.45$	$2824.7^{d} \pm 40.09$	$2904.5^{a} \pm 88.57$
2.47	$2779.9^{h} \pm 23.16$	$2835.0^{e} \pm 32.51$	$2921.1^{b} \pm 81.08$
4.15	$2786.9^{i} \pm 22.57$	$2846.2^{f} \pm 35.16$	$2935.4^{c} \pm 78.18$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and depth of operation of the machine. From regression coefficients (Appendix table F4), their relationship could be expressed by the following equation.

Drawbar pull= $28.14D+11.16F_s+2616.48$, $R^2=0.64$

The coefficient of the depth of operation (D) is 28.14, which means that for every unit increase in the depth of operation, the drawbar pull increases by 28.14%. The coefficient of the forward speed (F_s) is 11.16, which means that for every unit increase in the forward speed, the drawbar pull increases by 11.16%. The coefficient of determination (R^2) is 0.64, which indicates that 64% of the variation in the drawbar pull can be explained by the depth of operation and forward speed. Based on the regression coefficients, depth of operation was found to make dominantly affected the drawbar force as compared to forward speed.

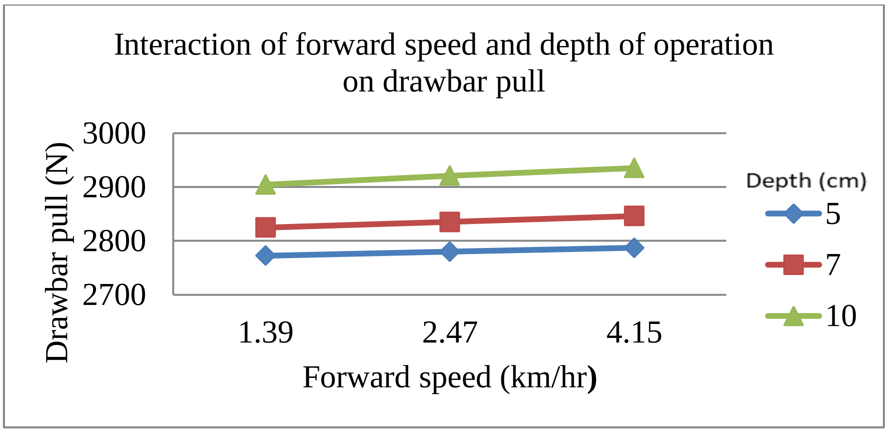


Figure 4.5. Effect of forward speed and depth of operation on drawbar pull

4.3.5. Fuel Consumption

Table 4.5 shows that the mean values of fuel consumption (Fc) for the onion digger at different forward speeds and depths of operation. The values in the table are means with standard errors (SE) of fuel consumption. The ANOVA, on fuel consumption of the onion harvester, revealed that forward speed and depth of operation had highly significant at (P<0.05). Maximum fuel consumption was 15.51L/ha at 4.15km/hr forward speed and 10cm depth of operation and minimum fuel consumption was 12.36 L/ha at 1.39 km/hr and 5cm depth of operation. The grand mean, CV (Rep*Depth) and CV (Rep*Depth*Fs)) of fuel consumption was also 13.71L/ha, 1.16 l/ha and 2.45 l/ha respectively (Appendix table E6). For summarizing, fuel consumption of the machine increased with in increasing both forward speed and depth of operation which was the same as the result of Mohamed Ibrahim (2011).

Table 4.5. Means with SE of fuel consumption (Fc) at various forward speed and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$12.36^{e} \pm 0.13$	$12.56^{de} \pm 0.31$	$12.73^{cde} \pm 0.33$
2.47	$13.05^{cde} \pm 0.60$	$13.65^{bcd} \pm 0.78$	$13.85^{bc} \pm 0.16$
4.15	$14.54^{ab} \pm 0.40$	$15.12^{a} \pm 0.36$	$15.51^{a} \pm 0.50$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and depth of operation of the machine on fuel consumption. From regression coefficients (Appendix table F5), their relationship could be expressed by the following equation.

$Fc= 1.25F_s+0.14D+10.18$, $R^2=0.87$

The equation is in the form of a linear regression model, where the fuel consumption is the dependent variable and the depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is 0.14, which means that for every unit increase in the depth of operation, the fuel consumption increases by 0.14%. The coefficient of the forward speed (F_s) is 1.25, which means that for every unit increase in the forward speed, the fuel consumption increases by 1.25%. The coefficient of determination (R^2) is 0.87, which indicates that 87% of the variation in the fuel consumption can be explained by the depth of operation and forward speed. The coefficients of the equation indicate that the forward speed has a greater effect on the drawbar pull than the depth of operation. A higher drawbar pull indicates that the machine is working harder and may result in higher fuel consumption and lower field efficiency.

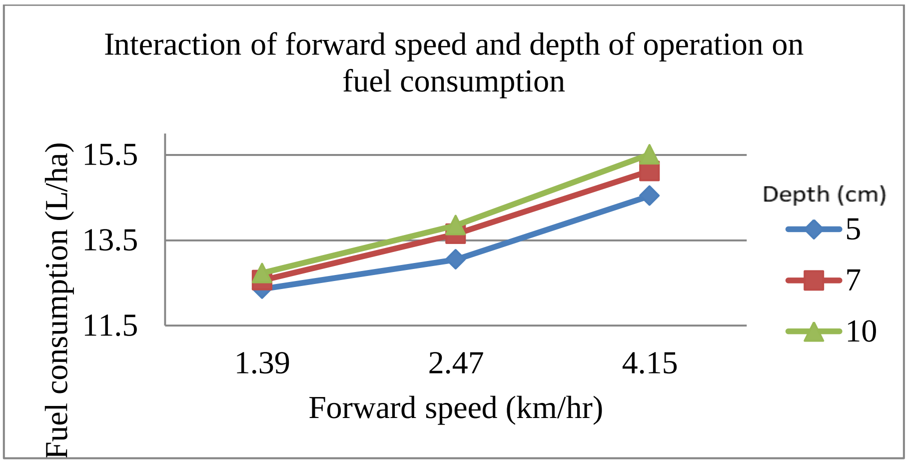


Figure 4.6. Effect of forward speed and depth of operation on fuel consumption

4.3.6. Field Capacity

Field capacity is a measure of the machine's ability to harvest onions in a given area of land within a specific time frame. Table 4.6 shows that the means with standard errors (SE) of field capacity (Fca) of an onion digger at various forward speeds (Fs) and depths of operation. The table shows that at a forward speed of 1.39 km/hr, the field capacity of the onion digger was 0.147, 0.140, and 0.135ha/hr at depths of operation of 5cm, 7cm, and 10cm, respectively. Similarly, at a forward speed of 2.47 km/hr, the field capacity was 0.155, 0.146, and 0.139ha/hr at depths of operation of 5cm, 7cm, and 10cm, respectively. Finally, at a forward speed of 4.15 km/hr, the field capacity was 0.157, 0.151, and 0.143ha/hr at depths of operation of 5cm, 7cm, and 10cm, respectively.

The field capacity was affected directly by forward speed and depth of operation. The analysis of variance, on field capacity of the onion harvester, revealed that forward speed and depth of operation had significant ($P < 0.05$) effects on the field capacity of the machine (appendix table E6).The digger field capacity was increased by increasing the forward speed from 1.39 to 4.15 km/h and decreased by increasing the digging depth from 5cm to 10cm due to increased force with increasing the depth which was the same as the result of Mohamed Ibrahim (2011). The maximum value of the digger field capacity was 0.157 ha/h at forward speed of 4.15 km/h and digging depth of 5 cm, while the minimum value was 0.135 ha/h at forward speed of 1.39 km/h and digging depth of 10 cm. The high speed with low tillage depth gave the highest rate of field capacity and vice versa. The grand mean, CV (Rep*Depth) and CV (Rep*Depth*Fs)) of fuel consumption was also 0.146ha/hr, 3.73ha/hr and 1.87ha/hr respectively (Appendix table 7).

Table 4.6. Means with SE of field capacity (Fca) at various forward speed and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$0.147^{a} \pm 0.0068$	$0.140^{a} \pm 0.0025$	$0.135^{a} \pm 0.0021$
2.47	$0.155^{a} \pm 0.0032$	$0.146^{a} \pm 0.0035$	$0.139^{a} \pm 0.0055$
4.15	$0.157^{a} \pm 0.0038$	$0.151^{a} \pm 0.0011$	$0.143^{a} \pm 0.0064$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and digging depth of the machine. From regression coefficients (Appendix table F6), their relationship could be expressed by the following equation.

$F_{ca}= 0.0047F_s-0.0028D+0.157$, $R^2=0.77$

The equation is in the form of a linear regression model, where the field capacity is the dependent variable and the depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is -0.0028, which means that for every unit increase in the depth of operation, the field capacity decreases by 0.0028%. The coefficient of the forward speed (F_S) is 0.0047, which means that for every unit increase in the forward speed, the field capacity increases by 0.0047%. The coefficient of determination (R^2) is 0.77, which indicates that 77% of the variation in the field capacity can be explained by the depth of operation and forward speed.

Finally, based on equation, the forward speed was found to be the highest impact for the contribution of field capacity as compared to depth of operation.

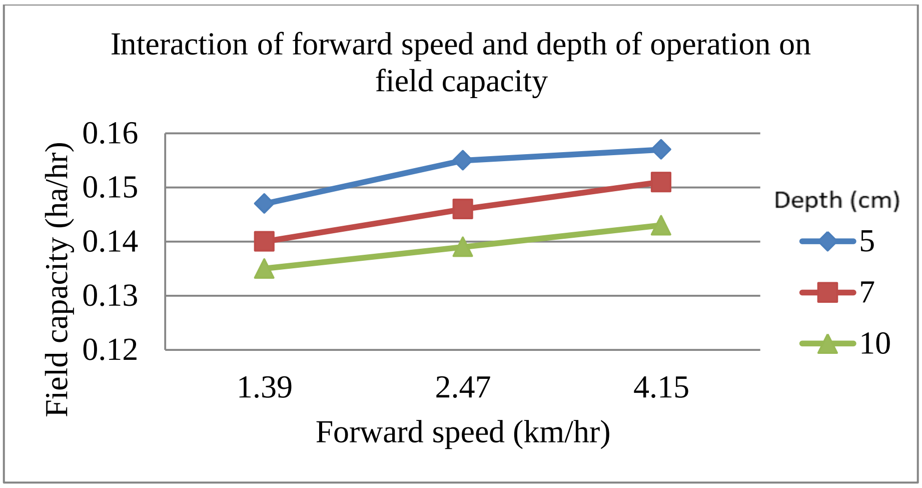


Figure 4.7. Effect of forward speed and depth of operation on field capacity

4.3.7. Field Efficiency

Field efficiency is a measure of how well the machine performs in the field. Table 4.7 shows that the means with standard errors (SE) of field efficiency of an onion digger at various forward speeds and depths of operation. The table shows that the field efficiency of the onion digger decreases as the forward speed and depth of operation increase. The standard errors (SE) indicate the variability of the means, with smaller SE indicating less variability and greater precision.

Increasing forward speed from 1.39 to 4.15 km/h decreased field efficiency and increasing the depth of harvest from 5 to 10 cm decreases field efficiency which was the same as the result of Mohamed Ibrahim (2011). The analysis of variance, on field efficiency of the onion harvester, revealed that forward speed and depth of operation had significant ($P<0.05$) effects on the field capacity of the prototype machine (Appendix table E7). Mean maximum field efficiency was 90.84 at minimum forward speed of 1.39km/hr and at minimum digging depth of 5cm. Mean minimum field efficiency was 73.06 at maximum forward speed of 4.15km/hr and at maximum digging depth of 10cm. In general, field efficiency of the machine decreased with in increasing both forward speed and digging depth. The grand mean, CV (Rep*Depth) and CV (Rep*Depth*Fs)) of field efficiency was also 80.5%, 3.98% and 1.05% ha/hr respectively (Appendix table 8). The major reason for the reduction in field efficiency by increasing forward speed may be due to the increasing maintenance requirement by increasing depth and increasing the quantity of soil on the elevator device.

Table 4.7. Means with SE of Field efficiency of onion digger at various forward speeds and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$90.84^{a} \pm 0.17$	$82.18^{c} \pm 0.29$	$78.37^{d} \pm 0.31$
2.47	$85.77^{b} \pm 0.35$	$78.81^{d} \pm 0.24$	$76.26^{e} \pm 0.14$
4.15	$82.33^{c} \pm 0.12$	$76.75^{e} \pm 0.04$	$73.06^{f} \pm 0.05$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and digging depth of the machine. From regression coefficients (Appendix table F7), their relationship could be expressed by the following equation.

Field efficiency= $-2.01D-3.21F_s+101.63$, $R^2=0.84$

The equation is in the form of a linear regression model, where the field efficiency is the dependent variable, depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is -2.01, which means that for every unit increase in the depth of operation, the field efficiency decreases by 2.01%. The coefficient of the forward speed (F_s) is -3.21, which means that for every unit increase in the forward speed, the field efficiency decreases by 3.21%. The coefficient of determination (R^2) is 0.84, which indicates that 84% of the variation in the field efficiency can be explained by the depth of operation and forward speed. Based on the regression equation above, forward speed was found to be the greatest negative impact for the contribution of field efficiency as compared to depth of operation of the machine.

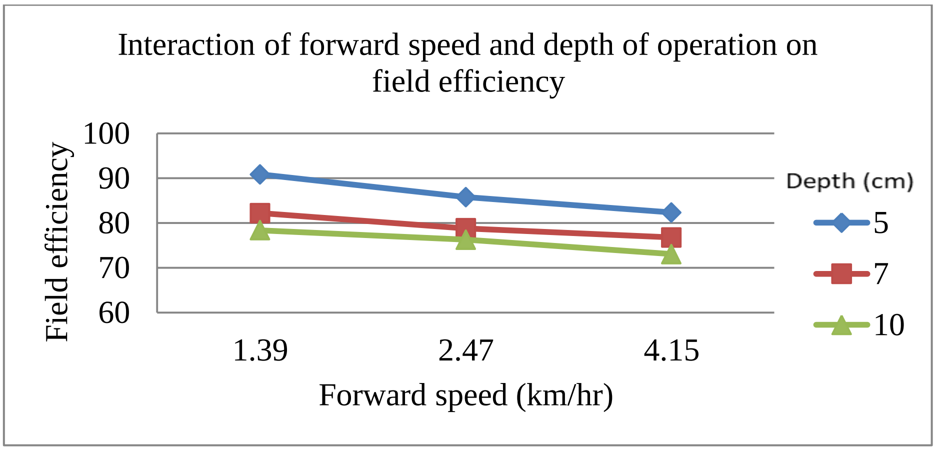


Figure 4.8. Effect of forward speed and depth of operation on field efficiency

4.3.8. Wheel Slip

Table 4.8 shows that the means with standard errors (SE) of wheel slip of an onion digger at various forward speeds and depths of operation. The wheel slip can affect the performance of the machine, as it can cause damage to the onion bulbs and reduce the digging efficiency. The analysis of variance, on wheel slip of the onion digger, revealed that forward speed and depth of operation had highly significant ($P < 0.05$) effects on the wheel slip of the prototype machine (Appendix table E8). Mean maximum wheel slip was 20.82% at maximum forward speed of 4.15km/hr and maximum depth of operation at 10cm. While, mean minimum wheel slip was 4.47% at minimum forward speed of 1.39km/hr and minimum depth of operation at 5cm. In general, wheel slip of the machine increased with increasing both forward speed and digging depth which was the same as the result of Mohamed Ibrahim (2011). The grand mean, CV (Rep*Depth) and CV (Rep*Depth*Fs)) of field efficiency was also 11.5%, 10.71% and 5.87% ha/hr respectively (Appendix table 9).

Table 4.8. Means with SE of wheel slip of onion digger at various forward speeds and depth of operation

Treatments	Depth of operation (cm)		
Fs (km/hr)	5	7	10
1.39	$4.47^{f} \pm 0.78$	$7.33^{e} \pm 0.85$	$9.53^{d} \pm 0.33$
2.47	$8.01^{e} \pm 0.62$	$12.31^{c} \pm 0.43$	$15.22^{b} \pm 0.59$
4.15	$10.5^{d} \pm 0.63$	$15.28^{b} \pm 0.70$	$20.82^{a} \pm 1.72$

Means followed by the same letters superscript are not significantly different at 5% level of probability; SE = Standard Errors.

The regression analysis was made to establish relationship between forward speed and digging depth of the machine. From regression coefficients (Appendix table F8), their relationship could be expressed by the following equation.

Wheel slip= $1.48D+4.21F_s-7.78$, $R^2=0.93$

The equation is in the form of a linear regression model, where the wheel slip is the dependent variable, depth of operation and forward speed are the independent variables. The coefficient of the depth of operation (D) is 1.48, which means that for every unit increase in the depth of operation, the wheel slip increases by 1.48%. The coefficient of the forward speed (F_s) is 4.21, which means that for every unit increase in the forward speed, the wheel slip increases by 4.21%. The coefficient of determination (R^2) is 0.93, which indicates that

93% of the variation in the wheel slip can be explained by the depth of operation and forward speed. Based on the regression equation above, forward speed was found to be the greatest positive impact for the contribution of wheel slip as compared to digging depth of machine.

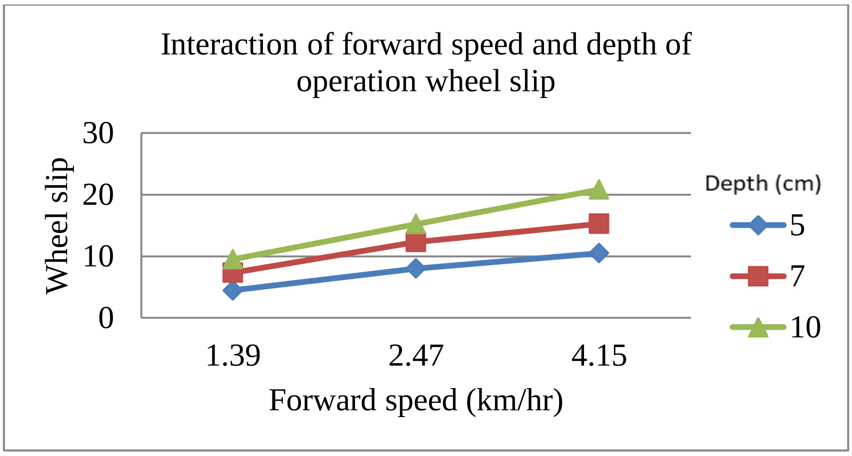


Figure 4.9. Effect of forward speed and depth of operation on wheel slip

4.4. Cost Estimation of the Machine

The cost and benefit that can be obtained by using the onion harvester machine had been evaluated in terms of raw material, production (machine and labor) and operation costs. The cost of operation associated with existing practice (manual hand hoe digging) of onion was compared with the cost of operation of developed onion harvester. The estimated cost of the machine was around birr 20,000.00 with the cost of operation of 1858.92birr/ha (Annex II). The cost of operation of manual hand hoe digging computed was 8006.25 Birr.ha-1. The saving in cost per hectare with the use of the onion harvester was found 6147.33 Birr/ha. It saved 88% labor (percentage of labor involved relative to manual hand hoe harvesting) and 76.78 % cost of harvesting in comparison to the manual hand hoe harvesting practice. The breakeven point (BEP) for the onion harvester was 104.25 h.yr-1 i.e. annual coverage (Ac) or annual utility of 35.75 ha/yr. The percentage of BEP was 37.9%. BEP is achieved at about 37.9% of the annual utility or area coverage in 275 h/yr. of use of the operation of a machine. Payback period (PBP) also 1.59 years and the benefit-cost ratio (BCR) was 4.33; when more than unity shows the machine could be worthwhile, it can provide additional benefit returns over the costs of a machine. Details of economic analysis of a 2WT operated onion harvester have described in Annex II.

5. CONCLUSIONS AND RECOMMENDATION

5.1. Conclusion

The harvesting of onion tubers is a crucial process in the production of onions, and it demands rapid attention to design the right mechanical harvesting technology. But in Ethiopia, onion crop harvesting is still done using outdated, labor-intensive techniques that require digging up the onion from the soil's subsurface. The design of a suitable onion harvester is required because of the shortage of workers, the expensive wages sought by them to harvest the crop, as well as the increased field losses and damages caused by hand harvesting. It was decided to use 2WT when constructing the onion harvester since onions are produced in small-scale locations and owning high horsepower tractors is expensive. A 2WT operated onion harvester was then designed, built, and its performance assessed in order to reduce harvesting loss, drudgery, labor need, production expense, and waste of time and energy. Through the use of experimental plots, it was possible to study the physical characteristics of the soil at the time of harvest, including soil type, moisture content, bulk density, cone index, shear strength, and various onion physical characteristics that affect the design of the onion harvester. The developed 2WT operated onion harvester was evaluated with several treatment combinations of forward speed at different levels of 1.39, 2.47, and 4.15 km/hr and at various levels of operating depth of 5, 7 and 10 cm. The 2WT operated onion harvester's performance was assessed in terms of damage %, missing percentage, digging efficiency, drawbar pull, digging efficiency, fuel consumption, field capacity, field efficiency, and wheel slip. Rstudio, Statistix 8, and Microsoft Excel 2010 were the softwares used throughout the analysis of performance evaluation. The maximum value of damage percentage of onion tuber was 4.56 % at forward speed of 4.15 km/h and digging depth of 5 cm, while the minimum value was 1.13 % at forward speed of 1.39 km/h and digging depth of 10 cm. As depth of operation increases onion tuber damage decreases, but as forward speed increases, onion tuber damage percentage also increases. The maximum value of digging efficiency of harvester was 97.03 % at forward speed of 1.39 km/h and digging depth of 10 cm, while the minimum value was 84.60 % at forward speed of 4.15 km/h and digging depth of 5 cm. Digging efficiency was decreased by increasing the speed, but increased with increasing the digging depth of operation. The maximum value of missing tubers was 11.36 % at forward speed of 1.39 km/h and digging depth of 5 cm, while the minimum value was 3.47 % at forward speed of 2.47 km/h and digging depth of 10 cm. The maximum value of drawbar pull was 2935.4N at forward speed of 4.15 km/h and digging depth of 10 cm, while the minimum value was 2772.4N at forward speed of 1.39 km/h and digging depth of 5 cm. Maximum fuel consumption was 15.51L/ha at 4.15km/hr forward speed and 10cm depth of operation and minimum fuel consumption was 12.36 L/ha at 1.39 km/hr and 5cm depth of operation. The maximum value of the digger field capacity was 0.157 ha/h at forward speed of 4.15 km/h and digging depth of 5 cm, while the minimum value was 0.135 ha/h at forward speed of 1.39 km/h and digging depth of 10 cm. Maximum field efficiency was 90.84 at forward speed of 1.39km/hr and at digging depth of 5cm and minimum field efficiency was 73.06 at maximum forward speed of 4.15km/hr and at digging depth of 10cm. Maximum wheel slip was 20.82% at forward speed of 4.15km/hr and depth of operation at 10cm. While, mean minimum wheel slip was 4.47% at minimum forward speed of 1.39km/hr and minimum depth of operation at 5cm. The developed prototype onion harvester was required 7.69 hours for digging a hectare of land. The saving in the cost of digging of onion tuber per hectare by using onion harvester was found 6147.33 birr/ha. It saved 88% labor and 76% cost of harvesting compared to the manual hand hoe harvesting. Performance evaluation concluded that the performance of the onion harvester prototype machine has done the task effectively. The performance of the machine was significantly affected mainly by the depth of operation than forward speed. The study indicated the optimum combination of forward speed and depth of operation to be 4.15km/hr and 10cm, respectively.

5.2. Recommendations

The following recommendations were made from these studies:

1. The onion digger is needed to be tested and demonstrated in the farmer's field.
2. Multiplication of the developed onion digger technology needs to be made.
3. Systematic, coordinated and relentless efforts are made to get the onion digger adopted, effectively and efficiently used by the small-scale farmers
4. The onion digger must be modified for large horse power of a four wheel tractors.

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APPENDICES

Appendix A: Soil classification

Calculations of hydrometer soil analysis Hydrometer correction

(T-20) x 0.36 + H = corrected hydrometer

Sand determination (%)

100 – ((corrected hydrometer at 40sec/ dry weight of soil sample) *100) = % sand in sample

Clay determination (%)

((corrected hydrometer at 6hr 40 sec / dry weight of soil sample)*100) = % clay in sample

Silt determination (%)

100 – (%sand +%clay) = % silt in sample

Texture

% sand +%silt +%clay = 100

Hydrometer calibration temperature =20^{o}c

Suspension temperature at 40 sec = 22^{o}c Suspension temperature at 5 hours = 24

Hydrometer reading at 40 sec = 12, Corrected hydrometer reading = 36 + (1*0.36) = 36.36 Hydrometer reading at 5 hours 40 sec = 12, Corrected hydrometer reading = 33 + (3*0.36) =34.08

% Sand = 100 – ((Corrected hydrometer at 40sec/ dry weight of soil sample) *100)

= 100 – ((36.36 / 50)*100) = 27.28~28 %

% Clay = ((corrected hydrometer at 6hr 40 sec / dry weight of soil sample)*100)

= ((34.08/50)*100) = 31.84~32%

% Silt = 100 – (% sand + +%clay)

= 100 – (28 + 32) = 40 %

Where:

T = temperature

H =hydrometer

Appendix B

Table B1. Moisture content and Bulk density of soil of study area

Observation	Weight of soil sample (gm)	Weight of soil after oven-dried(gm)	Soil moisture content (d.b) %	Volume of sampler (cm^3)	Bulk density (g/cm^3)
1	624.16	585..42	10.73	512	1.143
2	612.52	589.66	11.62	512	1.152
3	675.84	578.14	11.26	512	1.129
Average	637.50	584.40	11.20	512	1.141

Appendix C: Physical Property of Onion Tuber

Table C1. Means with SE of physical properties of onion tuber

Properties	Nafis onion variety
Moisture content (w.b.),%	68.23 ± 3.08
Polar diameter, mm	52.4 ± 1.68
Equatorial diameter, mm	38.65 ± 1.30
Thickness, mm	25.97 ± 1.08
Arithmetic mean dia., mm	55.40 ± 0.22
Sphericity, %	72.76 ± 1.67
Volume, cm^3	85.32 ± 3.22
Density, g/cm^3	2.85 ± 0.14

Appendix D

Table D1. Mean with SE of static coefficient of friction for onion tuber and surfaces

Surface	Coefficient of friction
Galvanized steel sheet	0.41 ± 0.084
Mild steel sheet	0.38 ± 0.031
Aluminium sheet	0.43 ± 0.003
Plywood	0.27 ± 0.007

Appendix E

Table E1. ANOVA of damage percentage on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	0.863	0.432		
Depth	2	43.203	21.602	123.99*	0.0003
Error Rep*Depth	4	0.697	0.174		
Fs	2	0.478	0.239	12.30*	0.0012
Depth*Fs	4	0.098	0.025	1.27^{ns}	0.3356
Error Rep*Depth*Fs	12	0.233	0.019		
Total	26	45.5728			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df, degrees of freedom.

Measuring parameters	Values
Grand Mean	2.75
CV(Rep*Depth)	15.16%
CV(Rep*Depth*Fs)	5.06%

CV = coefficient of variance, Fs = Forward speed, Rep = Replication

Table E2. ANOVA of digging efficiency in % on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	2.037	1.018		
Depth	2	576.616	288.308	349.58*	0.0000
Error Rep*Depth	4	3.299	0.825		
Fs	2	17.479	8.740	20.07*	0.0001
Depth*Fs	4	1.137	0.284	0.65^{ns}	0.6365
Error Rep*Depth*Fs	12	5.226	0.435		
Total	26	605.794			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df, degrees of freedom.

Measuring parameters	Values
Grand Mean	92.41
CV(Rep*Depth)	0.98%
CV(Rep*Depth*Fs)	0.71%

Table E3. ANOVA of missing percentage on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	10.541	5.271		
Depth	2	67.287	33.644	21.33*	0.0073
Error Rep*Depth	4	6.308	1.577		
Fs	2	65.251	32.625	10.39*	0.0024
Depth*Fs	4	4.760	1.190	0.38^{ns}	0.8194
Error Rep*Depth*Fs	12	37.690	3.141		
Total	26	191.837			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	8.135
CV(Rep*Depth)	15.44%
CV(Rep*Depth*Fs)	21.78%

Table E4. ANOVA of Drawbar pulls on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	39297	1.04		
Depth	2	90264	1.18	14.76*	0.0142
Error Rep*Depth	4	12230	0.025		
Fs	2	2241	14.42	30.75*	0.0000
Depth*Fs	4	204	0.08	1.40^{ns}	0.2916
Error Rep*Depth*Fs	12	437	0.11		
Total	26	144674			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	2845.1
CV(Rep*Depth)	1.94
CV(Rep*Depth*Fs)	0.21

Table E5. ANOVA of fuel consumption percentage on performance of onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	2.08	1.04		
Depth	2	2.37	1.18	47.13*	0.0017
Error Rep*Depth	4	0.10	0.025		
Fs	2	28.84	14.42	127.96*	0.0000
Depth*Fs	4	0.32	0.08	0.72^{ns}	0.5948
Error Rep*Depth*Fs	12	1.35	0.11		
Total	26	35.07			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	13.71
CV(Rep*Depth)	1.16
CV(Rep*Depth*Fs)	2.45

Table E6. ANOVA of field capacity on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	0.00013	6.300E-05		
Depth	2	0.00090	4.488E-04	15.16*	0.0136
Error Rep*Depth	4	0.00012	2.961E-05		
Fs	2	0.00041	2.048E-04	27.44*	0.0000
Depth*Fs	4	0.00001	2.389E-06	0.32^{ns}	0.8591
Error Rep*Depth*Fs	12	0.00009	7.463E-06		
Total	26	0.00165			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	0.146
CV(Rep*Depth)	3.73
CV(Rep*Depth*Fs)	1.87

Table E7. ANOVA of field efficiency on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	12.19	6.096		
Depth	2	509.16	254.580	24.83*	0.0136
Error Rep*Depth	4	41.02	10.254		
Fs	2	185.91	92.957	129.86*	0.0000
Depth*Fs	4	12.11	3.027	4.23*	0.0231
Error Rep*Depth*Fs	12	8.59	0.716		
Total	26	768.98			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	80.5
CV(Rep*Depth)	3.98
CV(Rep*Depth*Fs)	1.05

Table E8. ANOVA of wheel slip on performance of prototype onion digger

Source of variations	DF	SS	MS	F-value	P-value
Rep	2	0.913	0.456		
Depth	2	255.360	127.680	84.21*	0.0005
Error Rep*Depth	4	6.065	1.516		
Fs	2	321.109	160.554	352.03*	0.0000
Depth*Fs	4	22.079	5.520	12.10*	0.0004
Error Rep*Depth*Fs	12	5.473	0.456		
Total	26	610.999			

** Significant at 1% level;* significant at 5% level; ns, non-significant; df,

Measuring parameters	Values
Grand Mean	11.5
CV(Rep*Depth)	10.71
CV(Rep*Depth*Fs)	5.87

Appendix F

Table F1. Regression coefficient relating the forward speed and depth of operation against mechanical damage of the machine

Mechanical damage (R^2 = 0.91)

	Coefficients	Std. Error	F-value	P-value
Constant	6.833	0.348	19.66	0.0000
Depth	-0.601	0.096	-15.77	0.0000
Fs	0.163	0.038	1.70	0.1023

Table F2. Regression coefficient relating the forward speed and depth of operation against digging efficiency of the machine

Digging efficiency (R^2 = 0.76)

	Coefficients	Std. Error	F-value	P-value
Constant	79.82	2.102	37.97	0.0000
Depth	1.975	0.230	8.58	0.0000
Fs	-0.945	0.579	-1.63	0.1161

Table F3. Regression coefficient relating the forward speed and depth of operation against missing percentage of the machine

Missing percentage (R^2 = 0.35)

	Coefficients	Std. Error	F-value	P-value
Constant	14.08	1.952	7.21	0.0000
Depth	-0.762	0.214	-3.56	0.0016
Fs	-0.178	0.538	-0.33	0.7440

Table F4. Regression coefficient relating the forward speed and depth of operation against drawbar pull of the machine

Drawbar pull (R^2 = 0.64)

	Coefficients	Std. Error	T-value	P-value
Constant	2616.48	39.86	65.64	0.0000
Depth	28.14	4.37	6.44	0.0000
Fs	11.16	10.99	1.02	0.3201

Table F5. Regression coefficient relating the forward speed and depth of operation against fuel consumption of the machine

Fuel consumption (R^2 = 0.87)

	Coefficients	Std. Error	F-value	P-value
Constant	10.18	0.37	27.40	0.0000
Depth	0.14	0.04	3.40	0.0024
Fs	1.25	0.10	12.25	0.0000

Table F6. Regression coefficient relating the forward speed and depth of operation against field capacity of the machine

Field capacity (R^2 = 0.77)

	Coefficients	Std. Error	T-value	P-value
Constant	0.15677	0.00337	46.49	0.0000
Depth	-0.00277	3.694E-04	-7.50	0.0000
Fs	0.00472	9.296E-04	5.08	0.0000

Table F7. Regression coefficient relating the forward speed and depth of operation against field efficiency of the machine

Field efficiency (R^2 = 0.84)

	Coefficients	Std. Error	T-value	P-value
Constant	101.626	1.94498	52.25	0.0000
Depth	-2.00751	0.21306	-9.42	0.0000
Fs	-3.20889	0.53619	-5.98	0.0000

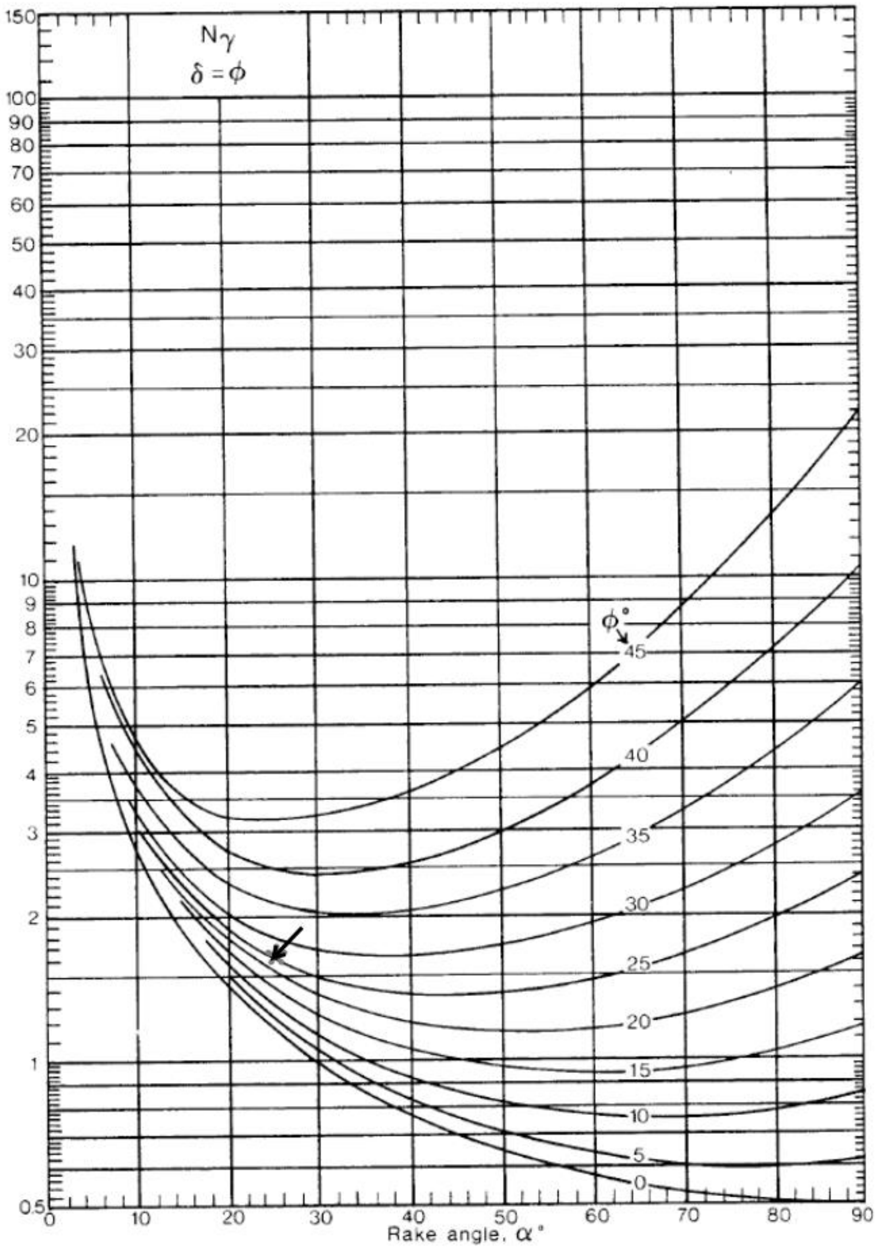
Table F8. Regression coefficient relating the forward speed and depth of operation against wheel slip of the machine

Wheel slip (R^2 = 0.93)

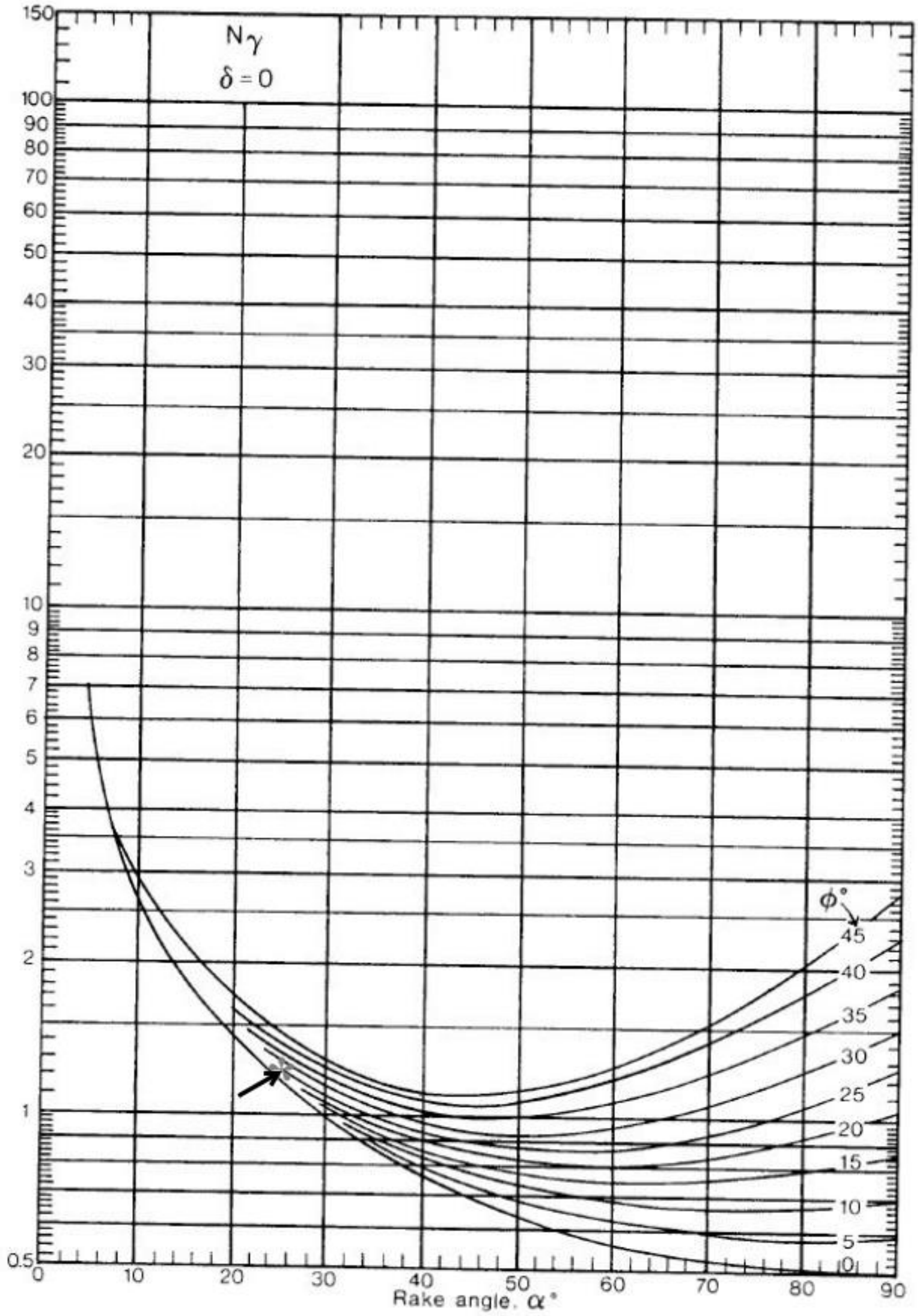
	Coefficients	Std. Error	T-value	P-value
Constant	-7.78193	1.12766	-6.90	0.0000
Depth	1.48026	0.12353	11.98	0.0000
Fs	4.21278	0.31087	13.55	0.0000

Appendix I: Graph of Soil cuting force N-factors

Graph I1. The value of N_γ *at* $\delta = \phi$, on Rake angle (α) and internal soil friction angle (ϕ) (Source: Mckyes, 1989)

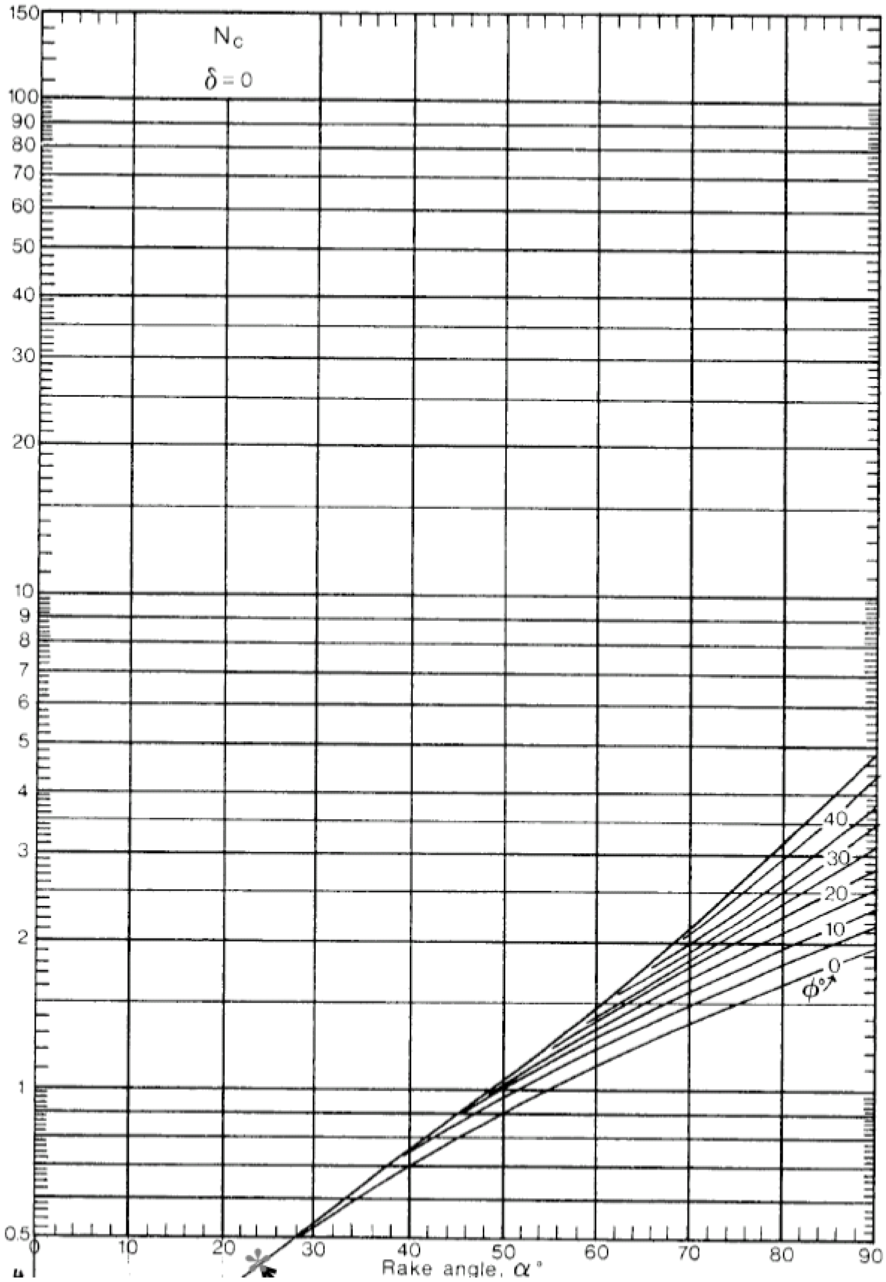


Graph I2. The value of N_γ $at\ \delta = 0$, on rake angle (α) and internal soil friction angle (ϕ)
(Source: Mckyes, 1989)

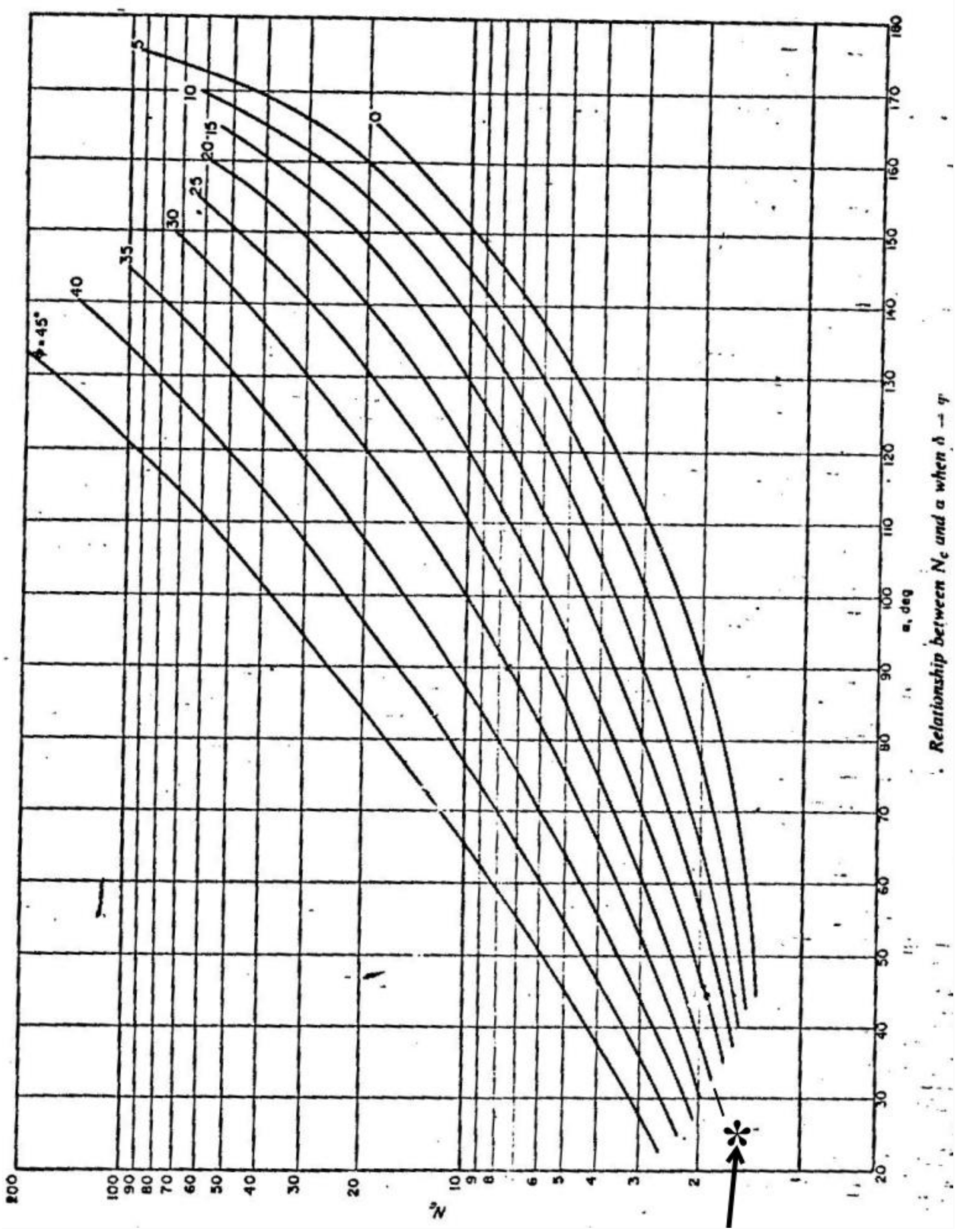


Graph I3. The value of N_c $at\,\delta = 0$, on rake angle (α) and internal soil friction angle (ϕ)

(Source: Mckyes, 1989)

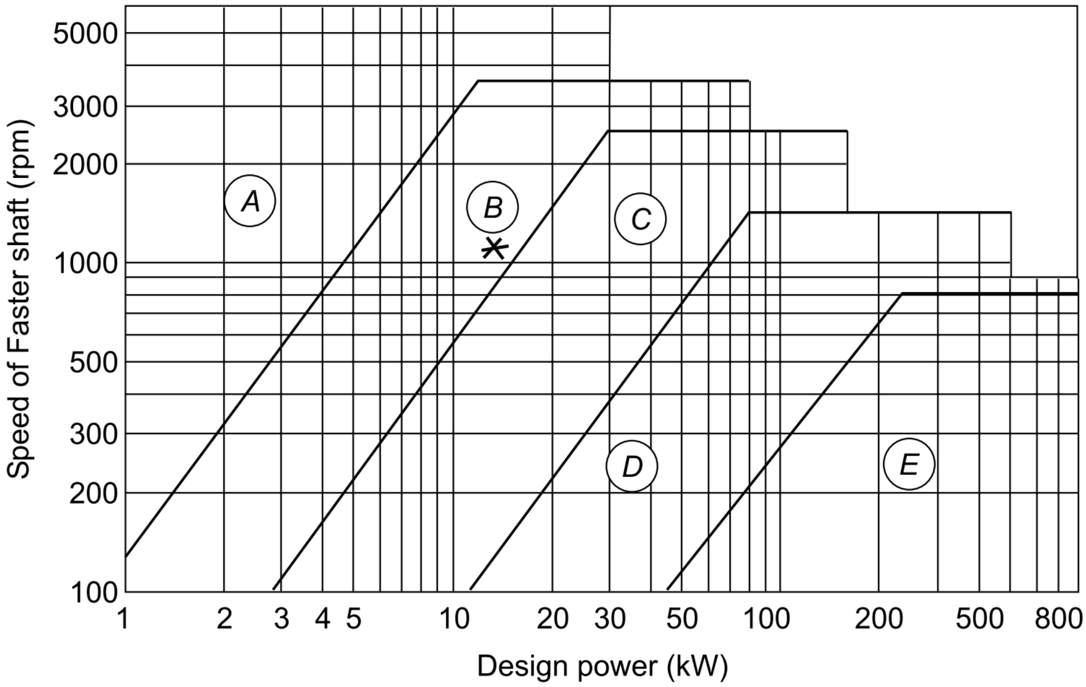


Graph I4. The value of N_c $at\,\delta = \phi$, on rake angle (α) and internal soil friction angle (ϕ) (Source: Hettiaratchi, Witney and Reece, 1966, sited in Khura, 2008)



Appendix J

Graph of Selection graph of V-belt cross-section (Source: Bhandari, 2010)



Appendix k: The raw data obtained from the field

S.N	Rep	Fs	D	Dam %	Dig %	Miss (%)	D_f (N)	Fc (L/ha)	S (%)	F_{ca} (ha/h)	F_{ef} (%)
1	1	1.5	5	4.61	87.75	9.68	2775.53	12.37	4.50	0.142	82.62
2	1	1.7	5	4.86	86.24	10.82	2768.22	13.49	7.34	0.156	85.22
3	1	2	5	5.00	84.6	12.38	2763.06	14.95	10.3	0.159	88.50
4	1	1.5	7	2.64	96.33	7.15	2812.35	12.86	7.80	0.140	78.36
5	1	1.7	7	2.80	94.7	9.45	2810.54	14.46	12.46	0.149	81.57
6	1	2	7	2.89	92.82	10.34	2804.44	15.42	15.72	0.152	84.34
7	1	1.5	10	1.22	98.25	4.48	2882.08	13.10	9.22	0.137	74.66
8	1	1.7	10	1.46	97.46	6.73	2875.82	13.92	15.83	0.146	77.90
9	1	2	10	1.50	96.0	8.66	2868.55	16.03	22.66	0.150	79.52
10	2	1.5	5	4.25	86.64	8.24	2772.45	12.22	5.23	0.145	80.95
11	2	1.7	5	4.40	85.77	9.50	2764.88	13.29	8.56	0.151	84.15
12	2	2	5	4.53	85.22	11.68	2756.03	14.54	11.22	0.153	90.38
13	2	1.5	7	2.51	94.36	6.42	2835.92	12.24	6.35	0.138	76.14
14	2	1.7	7	2.56	95.60	8.94	2822.66	13.59	11.83	0.142	77.68
15	2	2	7	2.59	93.85	9.46	2804.31	15.23	14.47	0.150	81.45
16	2	1.5	10	1.02	96.30	2.68	2886.43	12.47	9.50	0.133	72.30
17	2	1.7	10	1.14	95.66	3.42	2872.86	13.97	14.65	0.136	76.55
18	2	2	10	1.2	94.92	6.34	2850.80	15.49	20.55	0.138	78.88
19	3	1.5	5	3.59	86.73	10.02	2812.79	12.49	3.67	0.155	83.42
20	3	1.7	5	4.11	86.51	10.73	2806.57	12.36	8.14	0.157	87.94
21	3	2	5	4.15	84.00	11.97	2798.16	14.14	10.00	0.160	93.65
22	3	1.5	7	2.30	95.86	8.20	2890.50	12.58	7.85	0.143	75.76
23	3	1.7	7	2.36	95.89	8.75	2871.92	12.91	12.65	0.146	77.19
24	3	2	7	2.43	94.33	9.32	2865.28	14.72	15.66	0.150	80.75
25	3	1.5	10	1.14	96.54	3.25	3037.62	12.62	9.88	0.134	72.21
26	3	1.7	10	1.17	96.78	4.58	3014.76	13.67	15.18	0.137	74.33
27	3	2	10	1.92	96.01	6.47	2994.21	15.02	19.25	0.140	76.71

ANNEXES



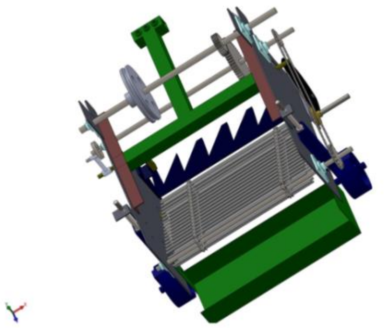
a) Preparing the beds b) Planting the onion c) Onion after plantation


d) Onion after removal of weeds e) Matured onion

f) Manufacturing of the components


g) Machine in the field h) Collection of the onion

i) The fabricated machine



j) Drawing of designed machine

II. Economic analysis of a 2WT Operated onion harvester

1. Cost of operation of 2WT tractor

Assumptions!

- Initial cost (P_t), birr = 40,000.00 with 11% salvage value (S_t), birr = 4400.00
- Average annual use (H_t), h = 800.00 and service life (L_t), yr. =10
- Fuel cost (F_c), Birr / liters =60.00
- Interest rate (I_t), is 12% of P, Birr/yr. = 4400.00
- Operator wages (O_w), a man-birr/day = 250.00
- Daily worker wage rate of birr 150.00 per man per day of 8 hours

1.1 Fixed cost of 2WT

The initial cost of a 15 hp single axle tractor in Birr was 40,000.00.

- Depreciation (Df), in (Birr/h) was (Pavani, 2017):

$$Df=\frac{C-S}{L\times H}=\frac{40000.00-(\frac{11}{100}\times 40000.00)}{10\times 800}=4.45\text{birr/hr}$$

- Interest (I_t) in (Birr/h) was (Naresh, 2015):

$$It=\frac{C+S}{2}\times\frac{L}{H}=\frac{40000+4400}{2}\times\frac{11}{10\times 800}=5.77\text{birr/hr}$$

- Housing, taxes and insurance cost at 3 % of the initial cost, birr/h = 0.15 (Khura,2008)
- Repair and maintenance cost at 10 % of initial cost, birr/h = 0.5 (Naresh, 2015)
  - ➢ Total fixed cost of 2-wheel tractor was:

Total fixed cost= $D_t + I_t + T_{rm} + C_{hti} = 4.45\text{birr/h} + 5.77\text{birr/h} + 0.15\text{birr/h} + 0.5\text{birr/h}$ = 10.87birr/h

1.2.Variable cost of 2WT tractor

- Fuel consumed (F_c) by the machine in, l/h = 1.95
- Fuel cost (F_{cl}), Birr/l = 60
- Fuel cost (F_{ch}), Birr/h:

$$F_{ch} = F_{cl} \times F_c = 60\text{birr/h} \times 1.95\text{l/h} = 117\text{birr/h}$$

- Lubricants cost (L_{bc}) was at 30% of fuel cost, Birr/h, (Khura, 2008):

$$Lbc=Fc\times\frac{30}{100}=\frac{117\text{birr}}{h}\times\frac{30}{100}=35.1\text{birr/h}$$

- Wage of tractor operator (O_w), in birr/h, at 250 birr per day of 8

normaldaily working hours of a man in Birr/h was 31.25 (Khura, 2008).

❖ Total variable costs for tractor (T_{vc}) were :

$$T_{vc} = O_w + L_{bc} + F_{ch} = 117\text{birr/ h} + 35.1\text{birr/ h} + 31.25\text{birr/ h} = 183.35\text{birr/h}$$

❖ Cost of tractor operation (C_{ot}), Birr/h was (Khura, 2008):

Cot=Tvc+Tf=183.35+10.87=194.22

2. Cost of operation of potato digger

Assumptions!

- Initial cost of onion harvester in birr = 20,000.00
- Average annual use (H_m)= 250 h and life of potato digger (L_m) = 10 years
- Salvage value (S_m) = 10 per cent of initial cost (Khura, 2008).
- Person-hour required for helm removing (N_{orm}. man-h) per ha = 20 (Bouman, 1993).
- Number of man-hours required for picking dug tubers per ha (N_{odpm}.mh/ha) = 11

2.1. Fixed cost of potato digger

Cost of potato digger with all accessories = 20,000.00

- Depreciation (D_m) was:

$$Dm=\frac{C\text{-}S}{L\times H}=\frac{20000.00\text{-}(\frac{12}{100}\times 20000.00)}{10\times 275}=6.4\text{birr/hr}$$

- Interest (I_m) was:

$$Im=\frac{P+S}{2\times H}\times\frac{i}{100}=\frac{20000.00+2200}{2}\times\frac{11}{100\times 800}=3.05\text{birr/hr}$$

- Taxes, insurance and shelter charges (C_{tis}), Birr /h, (at 3% of the initial cost per annum) it was 1.96. In addition, repair and maintenance cost (C_r); (at 10% of the initial cost per year) was also 6.54 birr/yr. (Naresh, 2015).

❖ The total fixed cost of the machine (*Tft*) in Birr/h was (Pavani, 2017):

Tft=Cr+Ctis+Dm+Im=6.4+3.05+1.96+6.54=17.95

2.1. Variable cost (C_{vm}) of onion harvester, in Birr/h, was (Pavani, 2017):

- Lubrication cost (L_{cm}), of 5% of fuel cost in Birr/h, was 0.935
- Repair and maintenance cost (C_{rm}) at 10 % of P_m in birr/h was 6.54
- The total cost of for helm removing (CN_{orm} man-h/ha) in Birr/ha was:

$$CNorm\ man\text{-}\frac{h}{ha}=\frac{Norm.man\text{-}\frac{h}{ha}\times Lwg}{\frac{8hr}{day}}=\frac{20man\text{-}\frac{h}{ha}\times\frac{100birr}{day}}{\frac{8hr}{day}}=250birr/ha$$

The total cost of for helm removing (C_hN_{orm} man-h/ha) in Birr/ha was also:

ChNorm man-h/ha=AFC (ha/h) ×CNorm man-h/ha=0.13ha/h×250 birr/ha=32.5birr/h

- The total cost of manual digging and picking (CN_{odpm}.mh/ha) was:

$$CNodpm.mh=\frac{CNodpm.mh\times Lwg}{8h/day}=\frac{10man\text{-}h/ha\times 150birr/day}{8h/day}=187.5birr/ha$$

$$C_h\ N_{odpm}man-h/ha = AFC,(ha/h)\times CN_{odpm}man-h/ha$$

$$= 0.13ha/h\times 187.5birr/ha = 24.37birr/h$$

❖ The total variable cost (C_{vm}) of potato digger was:

$$Cvm = L_{cm} + C_{rm} + C_h\ N_{orm}man-h/ha + C_h\ N_{odpm}man-h/ha$$

$$= 0.935+6.54+32.2+24.37= 31.84\ birr/h$$

❖ Total operating cost of onion harvester(O_m):

$$O_m = C_{vm} + C_{fm} = 31.84 +15.6= 47.44birr/h$$

The cost involved in mechanical onion digger (O_{ct}) (Pavani, 2017):

$$O_{ct} = O_t + O_m = 47.44\ birr/h + 194.22\ birr/h= 241.66birr/h$$

O_{ct} in ha/h can be computed as follows:

$$Oct=\frac{Oct}{AFC}=\frac{241.66birr}{h}\times\frac{1h}{0.13ha}=1858.92birr/ha$$

The cost of manual digging of onion

Assumptions for the calculation of manual digging:

- Number of hand hoes required (N_h) is 5 with the cost of a hand hoe (C_h) in Birr is40 (Dagninet *et al.*,2015)
- The total cost of hand hoe (C_{th}) in Birr was: $C_{th} = N_h \times C_h$ =200.00
- Man-hour required in dig and pick (N_{odp}.mh/ha) = 403man-h/ha
- The total cost of manual digging and picking (CN_{odp}.mh/ha) was:

$$CNodp\ man\text{-}h/ha=\frac{CNodp\ man\text{-}h/ha\times Lwg}{8h/day}=\frac{403man\text{-}h / ha\times 150birr/day}{8h/day}=7556.25Birr/ha$$

- Number of man-hour (N_{or}. man-h) required in helm removing per ha = 20 (Bouman,1993)
- The total cost of for helm removing per ha (CN_{or} man-h/ha) was:

$$CN\ or\ man\text{-}h/ha=\frac{N\ or\ mh/ha\times Lwg}{8h/day}=\frac{20man\text{-}h / ha\times 100birr/day}{8h/day}=250birr/ha$$

- The total cost of manual digging ($C_t N_{or} man\text{-}h/ha$) was:

$$C_t N_{or} man - h/ha = CN_{odp} man - h/ \quad + CN_{or} man - h/ha$$

$$= 250 + 7556.25 + 200 = 8006.25 birr/ha$$

4.1. Calculation of saving cost, break-even point and a payback period of potato digger: Cost of mechanical onion digging (O_{ct}), Birr/ha = 1858.92birr/ha

Cost of manual digging of onion ($C_t N_{or} man - h/ha$), Birr/ha = 8006.25birr/ha

- Saving cost (Sc), in Birr/ha was:

Sc=Ct Nor man-h/ha-oct=8006.25birr/ha-1858.92birr/ha=6147.33birr/ha

Saving cost, percent ($S_{c\ \%}$), hence,

$$Sc\% = \frac{Sc}{\text{Ct Nor man-h/ha}} \times 100\% = \frac{6147.33 birr/ha}{8006.25.00 birr/ha} \times 100\% = 76.78\%$$

- Saving in manpower, percent ($S_{p\ \%}$), was,

Total manpower used in mechanical onion harvesting =6

Total manpower used in manual onion harvesting (Mm) = 50

$$Sp\% = \frac{Mm\text{-}Md}{Mm} \times 100\% = \frac{50-6}{50} \times 100\% = 88\%$$

4.1.1. Break-even point, (BEP) (Khura, 2008):

$$BEP = \frac{\text{Annual fixed cost,birr / h}}{\text{custum fee,}\frac{birr}{h} - \text{operatingcost,birr/h}}$$

- Annual fixed cost (A_{fc}) was: $A_{fc} = H_m \times T_{ft} = 275h \times 15.604 Birr/h =$ 4291.1 Birr/yr.
- Custom fee (CF) in birr/h was also:

$$CF = 1.25 \times O_m + 0.25 \times 1.25 \times O_m = 1.25 \times 73.16 + 0.25 \times 1.25 \times 73.16 = 114.32 birr$$

- The break-even point of the machine was computed using (Naresh, 2015).

$$BEP, h/yr. = \frac{\text{Afc , birr/yr}}{\text{CF ,}\frac{birr}{h}\text{- C, birr/h}} = \frac{4291.1 birr / yr}{114.32 birr/h - 73.16 birr / h} = 104.5 h/yr$$

- BEP, ha/year also can be computed as,

$$BEP_{ha/yr} = BEP_{h/yr}.\times AFC_{ha/h} = 104.h/yr. \times 0.13 ha/h = 13.55 ha/yr.$$

- Annual coverage (Ac) (annual utility) of onion digger in ha/yr.:

$Ac_{ha} = AFC_{ha/h} \times H_m = 0.13 ha/yr \times 275h = 35.75 ha/yr.$

$$BEP\,(\%) = \frac{BEP\ ha/yr}{Acha} \times 100 = \frac{13.55 ha/yr}{37.75\ ha/yr} \times 100\% = 37.9\%$$

Therefore, BEP is achieved at about 37.9% of the annual utility or area coverage in 275 hours of use and operation of potato digger,

Payback period (PBP) (Naresh, 2015):

$$PBP = \frac{P_m}{AAP}$$

Average net annual profit (AAp) was:

$$AAP = CF - O_m = (114.32 Birr/h - 73.16 Birr/h) \times 275h/yr. = 11,319.00 birr/yr.$$

$$PBP = \frac{18000.00 birr}{11319.00 birr/year} = 1.59\ year$$

Benefit-cost ratio (BCR) (Naresh, 2015):

$$BCR = \frac{\text{Gross return, birr / ha}}{\text{Cost of operation, birr / ha}}$$

$$= \left(\frac{AAP, \frac{birr}{yr}}{Hm, \frac{h}{yr}}\right) \times \frac{AFC, \frac{h}{ha}}{Om, \frac{birr}{ha}} = \left(\frac{11,319.00}{275}\right) \times \frac{7.69}{73.16} = 4.33$$

The B.C.R. is more than unity shows that project investment to be considered was worthwhile, it can provide some additional return over the costs for clear decision.