DESIGN, FABRICATION AND PERFORMANCE EVALUATION OF MOTORIZED HOLE DIGGER FOR PERENNIAL CROP TRANSPLANTATION

MSc THESIS

FIKRE YIGEZU

JUNE 2025

HARAMAYA UNIVERSITY, HARAMAYA

DESIGN, FABRICATION AND PERFORMANCE EVALUATION OF MOTORIZED HOLE DIGGER FOR PERENNIAL CROP TRANSPLANTATION

A Thesis Submitted To the Department of Agricultural Engineering

Post Graduate Program Directorate

HARAMAYA UNIVERSITY

In Partial Fulfillment of the Requirement for Degree of

MASTER OF SCIENCE IN AGRICULTURAL MACHINERY ENGINEERING

Fikre Yigezu Abebe

June 2025

Haramaya University, Haramaya

HARAMAYA UNIVERSITY

POST GRADUATE PROGRAMS DIRECTORATE

I hereby certify that I have read and evaluated this Thesis entitled **Design, Fabrication and Performance Evaluation of Motorized Hole Digger for Perennial Crop Transplantation** prepared under my guidance by Fikre Yigezu. I recommend that it be submitted as fulfilling the thesis requirement.

Solomon Abera (Dr. Ing)	____________________	_________________
Major advisor	Signature	Date

As a member of the board of examiners of the MSc Thesis Open Defense Examination, We certify that we have read and evaluated the thesis prepared by Fikre Yigezu and examined the candidate. We recommended that the thesis be accepted as fulfilling the thesis requirement for the degree of Master of Science in Agricultural Machinery Engineering.

_____________________________	_________________	____________
Chairperson	Signature	Date
_____________________________	_________________	____________
Internal examiner	Signature	Date
_____________________________	_________________	____________
External examiner	Signature	Date

Final approval and acceptance of the Thesis is contingent upon the submission of its final copy to Council of Post Graduate Programs (CPGP) through the candidate’s department graduate committee (DGC).

DEDICATION

I dedicate this thesis to my parents, brothers, sisters and to my partner who have been my beacon of light during this journey with their unfailing love, support, and encouragement. I will always be thankful for their presence in my life since their sacrifices and faith in me have made all the difference.

STATEMENT OF THE AUTHOR

By my signature below, I declare and affirm that this Thesis is my own work. I have followed all ethical and technical principles of scholarship in the preparation, data collection, data analysis and compilation of this Thesis. Any scholarly matter that is included in the Thesis has been given recognition through citation.

This Thesis is submitted in partial fulfillment of the requirement for a MSc degree in Agricultural Machinery Engineering at Haramaya University. The Thesis is deposited in the Haramaya University Library and is made available to borrower under the rules of the Library. I solemnly declare that this Thesis has not been submitted to any other institution anywhere for the award of any academic degree, diploma or certificate.

Brief quatations from this Thesis may be made without special permission provided that accurate and complete acknowledgment of the source is made. Requests for permission for extended quotations from reproduction of this Thesis in whole or in part may be granted by the head of the department when in his or her judgment the proposed use of the material is in the interest of scholarship. In all other instance, however, permission must be obtained from the author of the Thesis.

Name: Fikre Yigezu Abebe Signature: ___________________

Date: __________________________________

Department: _____________________

BIOGRAPHICAL SKETCH OF THE AUTHOR

The author of this Thesis, Fikre Yigezu was born on January 16, 1996, in Oromia region, Arsi Zone, Aseko Woreda, Haro Ale Kebele, from his father Yigezu Abebe and his mother Buzunesh Kidanie. He attended his Elementary and Primary School and high school education in Aseko Elementary School from 2006 to 2011 and Aseko High School from 2012 to 2013. He also attended his preparatory education in Abomsa Senior Secondary School from 2014 to 2015. After that, he joined Haramaya University in 2015 and upon the completion of his studies in 2021, he obtained his Bachelor of Science Degree in Agricultural Engineering. The author was employed by Haramaya University as assistant lecturer shortly after graduation. After serving for one and half year, he started postgraduate study in the same University in 2022 to pursue further study for a Master of Science Degree in Agricultural Machinery Engineering.

.

ACKNOWLEDGMENTS

I would like to express my sincere gratitude to my advisor Solomon Abera (Dr. Ing), for his tireless help, invaluable advice and knowledgeable guidance during this project. His enlightening criticism, support and commitment have greatly influenced this work.

I would like to sincerely thank Abebe Fanta (PhD.), a specialist in the design of agricultural machinery, for his tireless assistance and expert guidance during design phase of this thesis project. His insight and guidance have not only enhanced my academic experience but also motivated me to pursue excellence, for which I am really thankful.

My deepest gratitude and acknowledgement go to Haramaya University and Haramaya Institute of Technology for giving me the chance to pursue my study for MSc in Agricultural Machinery Engineering and for providing financial support for research expenses.

I would like to extend my heartfelt gratitude to the Department of Agricultural Engineering staffs for their unwavering support. I am especially thankful to Mr. Ashebir Shimelis, the Head of the Department, who have given me a lot of strength and drive via his kindness and support.

I want to express my profound gratitude to my coworkers, especially Mr. Endrias Sibihat, Mr. Hailemariam Biratu, Mr. Tesfaye Gelan and Mr. Fami Abdi who kindly offered their time, knowledge, and assistance throughout the prototype's development and testing for this thesis project.

At last I would like to extend my heartfelt appreciation to Mr.Firomsa Dadi, Head Department of Mechanical Engineering and Mrs. Amsale Abebe (Store Keeper) for their support in providing necessary tools and machines during manufacturing the prototype of the hole digger. I am appreciative of their commitment and assistance, as it was crucial to the successful completion of this project.

ACRONYMS AND ABBREVIATIONS

AISI	American Iron and Steel Institute
ANOVA	Analysis of Variance
BS	British Standard
ESS	Ethiopian Statistics Service
DC	Direct Current
DF	Degree of freedom
FAO	Food and Agriculture Organization
GDP	Growth Domestic Product
IS	Indian Standard
MS	Mean sum of square
SS	Sum of square

TABLE OF CONTENTS

Continues..

Continues..

LIST OF TABLES

LIST OF FIGURES

LIST OF FIGURES IN THE APPENDIX

Appendix Figure **Page**

DESIGN, FABRICATION AND PERFORMANCE EVALUATION OF MOTORIZED HOLE DIGGER FOR PERENNIAL CROP TRANSPLANTATION

ABSTRACT

Ethiopian smallholder farmers are confronting a shortage of farm machinery for growing perennial crops. They are still employing old conventional farm hand equipment and human force to make holes for planting perennial crop seedlings. However *this kind of farming is exceedingly tedious, time-consuming and labor-intensive and resulted in extremely low production and productivity. Therefore, this study was conducted to improve production and productivity those farmer's and make attractive perennial crop farming environment by developing a hole digger for perennial crop production. In this study a motorized hole digger was designed, fabricated and evaluated for its performance. To evaluate the digging performance of the digger, an experiment with full factorial design at three replications was conducted on sandy and clay soil at different soil moisture content and working speed. The digger was tested on clay and sandy soils separately at soil moisture content of (15.00, 22.00 and 29.00%) and working speed of (100.00, 150.00 and 180.00 rpm). The result of the test revealed that the maximum digging capacity and efficiency of the digger were 45.00, 43.33 hole/hr and 85.00%, 84.67% observed at working speed of 180.00 rpm and soil moisture content of 29.00 % for hole dug on sandy and clay soil respectively at depth of 40.00cm and 35.00 cm diameter. The highest fuel consumption of 0.68 L/hr was observed when the soil moisture content was 15.0% and at working speed of180.00 rpm on clay soil. Generally, the study was successful in designing, fabricating, and testing a motorized hole digger prototype. The performance evaluation result showed that the digger's digging capacity and efficiency were greatly increased by raising the working speed from 100.00 to 180.00 rpm and the soil moisture content from 15.00 to 29.00%.*

Key words*: Perennial crop, hole digger, performance, digger capacity, digger efficiency*

1. INTRODUCTION

1.1. Background

Agriculture is the backbone of Ethiopia's economy, accounting for nearly 32.1% of the nation's gross domestic product (GDP), 70.00% of national export values and employing 72.70% of the labor force (Zerssa et al., 2021, Hailu, 2024, Mohammed and Beyene, 2024). According to Welteji (2018), smallholder mixed farming producers dominates the country’s agricultural sector. Those small-scale farmers are practicing rain-fed farming by using traditional technology and adopting low-input, low-output production systems.

In addition to grain and horticultural crops, Ethiopian farmers grows perennial crops such as fruits, stimulants, sugarcane, enset etc (ESS, 2021). According to ESS (2021), perennial crops covered about 1.95 million hectares. Perennial crops such as fruits (mango, avocado, orange, papaya, banana, guava, lemons and pineapple) cover about 161,471.00 hectares, stimulants (chat, coffee and hops) cover about 1,229,499.00 hectares, and the remaining 532,751.00 hectares and 29,520.00 hectares of land are covered by enset and sugar cane, respectively.

Land preparation for perennial crop production is critical for achieving maximum establishment and yield. This procedure includes a variety of techniques that improve soil health and enable the healthy establishment of perennial crops. Perennial crop growing operations begin with the multiplication of seedlings and the careful preparation of transplanting holes. Effective land preparation for transplanting seedlings involves selecting the right soil, digging appropriate holes, applying fertilizers, erecting protective structures, and preparing a suitable planting medium. These steps are essential for ensuring high survival rates and successful growth of the seedlings (Guan, 2016, Megersa, 2022).

Before planting these crops, a hole should be staked out and drilled that is at least twice the diameter of the root system even though hole size for planting perennial crops in the field is dependent on agro-ecological parameters. Wider and deeper holes drilled early in the dry season are ideal for a higher rate of survival and better field establishment, especially in locations with low moisture. Topsoil should be placed in one pile, while subsoil in another. The hole should then be filled with a mixture of 50% topsoil and 50% well-rotten manure, compost, or other decomposed organic material (Netsere, 2015, De, 2010).

Even though those Ethiopian smallholder farmers are doing their level best in doing these farm activities for better production of perennial crops, their production and productivity is extremely low. Lack of farm power and technologies are considered as one of the root causes of the low production and productivity of the farmers (Stellmacher and Kelboro, 2019). Human beings and draught animals powered the entire farm operations which clear indicates the drudgery and an inability to perform operations in a timely manner (Ayele, 2021).

Farmers use traditional farm tools such as spade, hand hoe and pick axe to prepare transplanting holes to plant perennial crops seedlings. This style of farming is very tedious, time consuming, labour-intensive, less productive. Hence, replacing hand tool farming with mechanical power is mandatory to increase farmers' productivity and to make attractive farming of perennial crops. Therefore this thesis research was undertaken to solve lack of hole digger, through developing a motorized hole digger for perennial crop production.

1.2. Statement of the Problem

Ethiopian smallholder farmers are facing lack of modern farm machinery to grow perennial crops. They are still using old traditional farm hand tools such as spade, hand hoe and pick axe and human power to dig hole for plant perennial crops seedlings. This makes their farming activity very tedious and enervating. Furthermore, farmers are not using their land effectively since traditional farming is very slow, laborious and time consuming. In addition to these, it threatens the productivity of farmers and lead them to deficiency of income. Therefore, the transition of traditional hand tool farming to modern mechanical power farming is very vital solution to solve those problems facing the farmer's. Cognizant of the problems faced by perennial crop growers, this thesis research was initiated to address the problem of lack of mechanical hole digger, through design, fabrication and performance testing a motorized hole digger. The digger can be used to make holes in the farm field to plant seedlings of mango, avocado, orange, papaya, banana, guava, lemons, chat, coffee, hops, seed cane and enset.

1.3. Scope of the Study

This study was all about design, fabrication, performance evaluation and production of complete report on a prototype motorized hole digger.

1.4. Significance of the Study

The hole digger will play a significant role in solving lack of hole making machine to plant perennial crops. It can reduce cost, time and labour requirement and can make perennial crops farming attractive and productive. In addition to these, the developed digger can also be used to dig holes for other purposes up to the depth of 60.00 cm. Furthermore the study generated information on the design, material requirement and manufacturing techniques of a motorized hole-digger making it available for large scale production by investors for distribution to farmers.

1.5. Objective

1.5.1. General objective

The general objective of this study was to design, fabricate and evaluate the performance of a motorized hole digger.

1.5.2. Specific objectives

- To determine physio mechanical properties of the soil
- To estimate the power requirement of the digger
- To design and fabricate different functional parts of the digger and assemble them
- To test and evaluate the performance of the prototype digger

2. LITERATURE REVIEW

2.1. Field Establishment of Perennial crops

2.1.1. Fruit crops

Fruit plants are most productive if carefully matched with the proper climate and planting site. Planting distances of fruit crop is another factor, which determines productivity of the crops (Usha et al., 2015). According to Usha et al. (2015), the recommended plant spacing are 6 m x 6 m, 5 m x 5 m, 3 m x 3 m, 1.25 m x 1.25 m, 6 m x 6 m, 5 m x 5 m and 0.3 m x 0.6 m for banana, guava, mango, papaya, orange, lemon and pineapple, respectively. After site selection and layout of planting is decided, holes that is large and deep enough to hold all the root system of the fruit seedling dug in the soil. Approximately the hole should be twice the diameter of the fruit seedling root system and 0.61 m deep to accommodate the entire root system (Usha et al., 2015).

2.1.2. Stimulant crops

Coffee production systems in Ethiopia are classified into forest coffee, semi-managed forest coffee, garden coffee, and plantation coffee. Nowadays, due to strong competition in the world coffee market and effects of climate change, the coffee production system is shifted from traditional to plantation system. The first and fundamental practice in the plantation system is the preparation of coffee seedlings, plantation site selection and preparation and transplanting the seedlings (Megersa, 2022). After selecting the site and clearing, holes are dug for planting with recommended hole spacing. The optimal spacing for Arabica coffee seedling is 2.40 m x 2.40 m and for Robusta coffee seedling is 3.00 m x 3.00 m. The recommended diameter and depth of hole are dug at 60.00 and 60.00 cm, respectively (Megersa, 2022). In another way the hole should be dug before one month of planting at 60.00 cm, 60.00 cm and 60.00 cm in length, width and depth respectively (Edward et al., 2005).

2.1.3. Enset crop

Enset is a staple food for over 20.00 million people in Ethiopia via its starch-rich corm and pseudo-stem (Borrell et al., 2019). The majority of small-scale farmers practices transplanting enset suckers across the enset-growing areas (Blomme et al., 2018). Enset transplanting may entail the removal of all plants or the selective thinning of some plants only. Plants might be transplanted to a uniform stand of only removed plants or incorporated into a field with plants of similar size, but of different age. In view of limited land availability for enset cultivation a 3.00 m × 1.50 m plant spacing is recommended (Borrell et al., 2019). Enset suckers are typically planted at depth of 10.00-15.00 cm to facilitate proper root establishment and minimizes the risk of exposure to pests. The width of planting hole should be between 30.00 to 50.00 cm to accommodate root development and ensure adequate soil aeration (Blomme et al., 2018).

2.2. Review on Post Hole Diggers

Preparing the planting hole for seedling transplantation is the most time consuming operation. Approximately 30.05% of the overall time spent on mechanical seedling transplantation is spent on preparing the planting hole (El-Halim et al., 2009). Post hole diggers come in a variety of designs, each adapted to specific uses and user requirements. These tools are necessary for efficiently digging holes for posts, plants, and drainage systems. Based on their design and power source, post hole diggers can be grouped in to Manual post hole diggers, tractor mounted post hole diggers, power tiller post hole digger and self-driven post hole diggers.

2.2.1. Manual post hole diggers

Manual post hole diggers are tools designed to dig shallow holes for posts or deeper holes for drainage. They typically consist of a steel shaft, cutting mechanisms, and a turning handle, allowing users to manually excavate soil efficiently (Bugeja, 2009). The traditional post hole diggers consist of two handles and a pair of blades that pivot to loosen and remove soil. They are simple and effective for small jobs (Freeman et al., 2007 and Domian, 2009). The modified designs of these models have offset blades feature to enhance soil removal and improve user control during operation (Domian, 2009).

2.2.2. Tractor mounted post Hole diggers

These machines are powered by a tractor's PTO (power take-off) and are designed for larger projects, allowing for deeper and wider holes. They utilize various blade sizes for different hole dimensions (Samuel et al., 2014).

2.2.3. Power tiller driven post hole diggers

These are post hole diggers that attached to power tillers and dig hole by using the power from the power tiller. Minaei and Arizdeh (2000) designed and developed auger drill for attachment to two wheels tractors for tree-seedlings planting.The machine excavates a hole that is 180 mm in diameter and 400 mm deep. Field tests revealed that two crucial elements influencing bit penetration rate are soil resistance and moisture content. As soil moisture content rose and soil penetration resistance decreased, the drilling rate rose as well. In sandy clay soil with a 25% moisture content, the average drilling rate was 1.4 m/min and the digging capacity of the machine was 100 holes/hour.

Chaaban et al. (2007), designed, constructed and tested a hole digger attached to power tiller (Figure 1). The result revelled that, a maximum digging capacity with cleaning was 56 hole/hr and obtained with auger speed of 200 rpm, hole diameter 15 cm, hole depth 20 cm, and sandy soil. The minimum hole digging capacity, with cleaning of 14 hole/h was obtained by auger speed of 75 rpm, hole diameter 30 cm, hole depth 40 cm on loamy soil. The power requirements increased in loamy soil than sandy loamy and sandy soil by 14.8 and 28.4 % respectively at different parameters. The minimum fuel consumption of 0.37 L/h was obtained with auger speed of 75 rpm, auger diameter 15 cm, hole depth 20 cm on sandy soil. The minimum power requirements of 1.18 kW was obtained with auger speed of 75 rpm, hole diameter 15 cm, hole depth 20 cm on sandy soil.

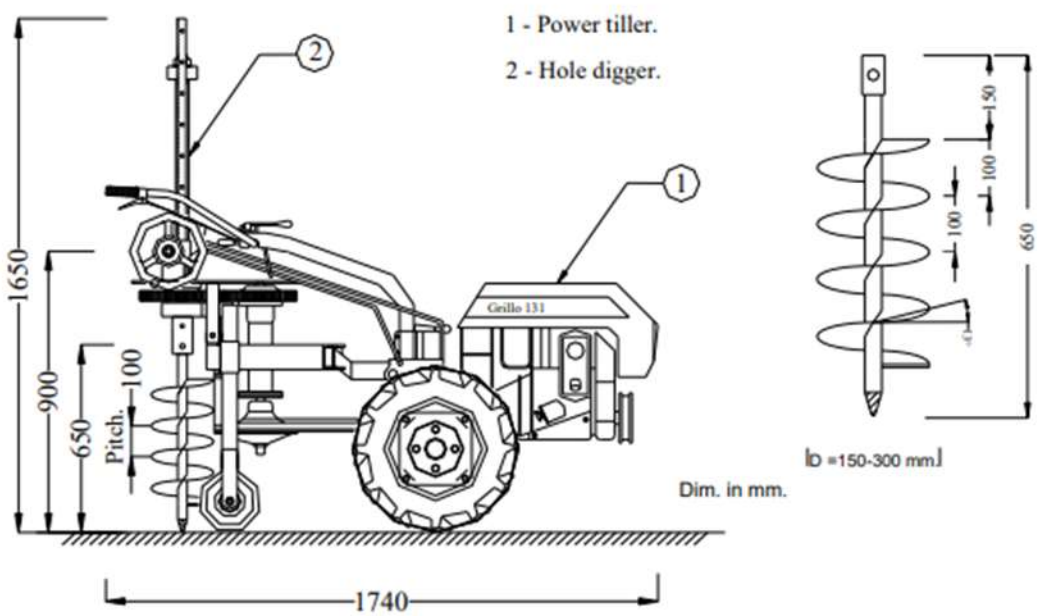


Figure 1. Hole digger attached to power tiller

Yehia et al. (2009), studied the factors affecting development of a hole diggers. An experiment was conducted on three auger diameter (15, 20, 25 cm) and operated at different auger speed (75, 100 and 150 rpm) to assess the performance of a developed hole digger by Chaaban et al.

(2007). The result of the experiment showed that, the maximum hole productivity of 324 hole/h was obtained with auger speed of 150 rpm, hole diameter 15 cm, hole depth 20 cm, auger pitch 20 cm at moisture content 26%. The maximum fuel consumption was 0.69 L/h was obtained with auger speed of 150 rpm, hole diameter 25 cm, hole depth 40 cm, and auger pitch 10 cm. The maximum power requirements was 2.72 kW was obtained with auger speed of 150 rpm, hole diameter 25 cm, hole depth 40 cm, auger pitch 10 cm and moisture content 18 %. The operation costs by using a hole digger attached to a power tiller decreased by about 500 - 950 % compared with manual digging.

2.2.4. Self-driven post hole diggers

The self-driven post hole diggers are those which have engine or motor build on the as a power source

Misr J. (2012), investigated some of the variables influencing the performance of the hole digger in sandy soil. Different auger speeds (75, 100, and 150 rpm), auger pitches (10, 15, and 20 cm), hole depths (20, 30, and 40 cm), hole spacing (5 m), and soil types (sandy) with moisture contents (15, 22, and 32%) were used to evaluate the designed hole-digger having 250mm auger diameters. The results obtained showed, the highest hole productivity rate for sandy soil was 335 holes per hour with an auger speed of 150 rpm, hole depth of 20 cm and an auger pitch diameter of 20 cm at a moisture content of 32%. The penetration resistance of sandy soil at 15% moisture content rose by 30 and 40.6% in comparison to sandy soil at moisture contents of 22% and 32% respectively.

With an auger speed of 150 rpm, hole depth of 40 cm, and auger pitch of 10 cm, the maximum fuel usage was 0.59 L/h. With an auger speed of 150 rpm, hole depth of 40 cm, auger pitch of 10 cm, and moisture content of 15%, the maximum power requirements of 1.91 kW were achieved. At an auger pitch of 15 cm and a hole depth of 20 cm, the minimal operating cost was 0.04 L.E/hole. However, at an auger pitch of 10 cm and a hole depth of 40 cm, the maximum operating cost was 0.14 L.E./hole.

Zong et al. (2016), designed an engine powered hole digger (Figure 2), for orchard trees establishment. The prototype of the digger was tested for power consumption, driller speed and feed rate of the driller into the soil. The result of the test showed that, the power consumption of the machine increased as the depth of the hole drilled increased. In another way, the feed rate and rotational speed of the soil driller decreased as depth of hole increased.

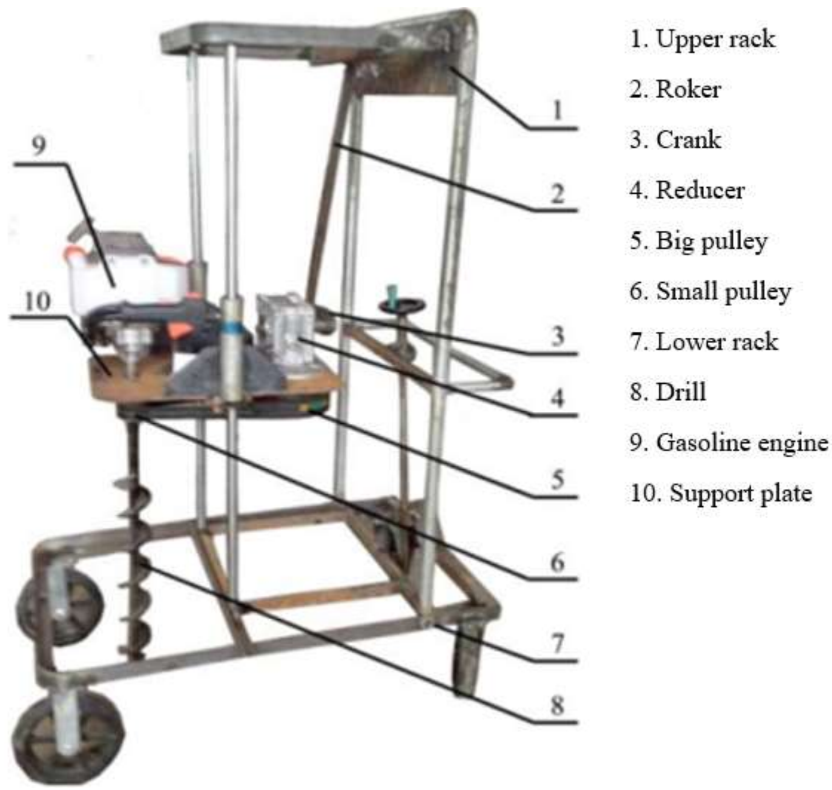


Figure 2. Engine powered hole digger

A motorized hydraulic hole-digger (Figure 3) was designed and manufactured by Mallapour et al. (2018). The machine was designed with working diameter and depth of 20.00 cm and 20.00 cm and operating speed of 100.00 to 160.00 rpm. The digger was evaluated for performance on two citrus gardens with silty-clay and sandy loam textured soils. Soil moisture content and depth of boring were considered as independent variables. Specific fuel consumption, machine effectiveness, auger torque, auger power and consumed energy showed that in both soil textures, the effects of soil depth and moisture content on the measured parameters were significant. The working capacity of the hole-digger at a depth of 30.00 cm at low and high soil moisture were 90.00 and 88.00 pits per hour in silty clay and 101.00 and 95.00 pits per hour in sandy loam, respectively. Additionally, the maximum power of 2.548 kW was required in silty-clay soil at a depth of 30.00 cm and low soil moisture.

Figure 3. Motorized hydraulic hole digger

Varghese et al. (2020), designed a portable soil digger (Figure 4) for plantation of seedlings of trees such as rubber, coconut, banana, apple and pines. The machine was designed by adopting principle of auger drilling machines used in pile foundation in construction work. It was intended to be operated by dc motor or engine as an alternative. It was mainly consisted of dc motor or engine as power source, auger drill bit, manual drill bit feed mechanism, wire mesh, frame and frame supports. The size of the auger drill bit designed was 30.00 cm in diameter and 90.00 cm in length. The prototype was tested in red and sandy loam soil. The test result showed that the machine can dig a hole with diameter of 340.00 mm to a depth of 450.00 mm in 140.00 seconds.

Figure 4. Portable hole digger

2.3. **Factors Affecting Performance of Post Hole Digger**

There are number of factors that influence the performance post hole digger. According to Zong et al (2016) and Misr J. (2012), working speed, working depth, working diameter, soil type, soil moisture content, soil penetration resistance and feed rate of the digger into the soil affect the performance of the digger.

2.4. **Literature Gap**

Even though the previously designed hole diggers were good in many ways, they have some limitations. The first limitation was that most of the hole diggers digs only a fixed diameter of hole similar to the driller diameter. The second limitation was lack of capacity to dig adequate hole depth for the crops seedling need. The digger developed by Mallapour et al. (2018) digs only to 20.00 cm diameter and 20.00 cm depth. The other hole diggers which evaluated by Misr J. (2012) and Chaaban et al. (2007) digs a hole 10.00 to 20.00 cm in diameter and 20.00 to 40.00 cm. The digger developed by Varghese et al. (2020), can dig a hole up to 34.00 cm diameter and 45.00 cm depth but still it is not enough hole size for crops that need 60.00 by 60.00 cm diameter and depth of hole such as coffee and some fruits (Edward et al., 2005; Habtamu, 2022; Usha et al., 2015). Most of the auger type diggers use interchangeable drilling tool to alter the size of hole to be dug which leads to increased power requirement per hole.

On other way, all of these machines are not being manufactured considering the soil condition in Ethiopia. Since all are imported from abroad, there is additional cost of transportation and tax in addition to its price which makes the machine costly and not affordable by small scale farmers to own. Therefore, these machines have limitation in terms of functional flexibility to dig appropriate depth and diameter of hole and not being produced considering the local soil characteristic.

Therefore, the aim of this study was to solve these limitation of hole diggers through designing and fabricating motorized hole digger that excavates soil from hole by bucket elevator system. The bucket soil elevating system makes the motorized hole digger different and functionally flexible to dig any required width of hole. The digger can also dig a hole up to the maximum depth of 60.00 cm which is recommended depth of planting hole for perennial crops according to Edward et al. (2005, Usha et al. (2015) and Megersa (2022). Again since the digger was developed in the country considering the local soil type, through taking feedback the digger can be improved according to the need of the farmers.

3. MATERIALS AND METHODS

3.1. Description of the Study Area

The fabrication of prototype of motorized hole digger was done in Agricultural Engineering Labratory at Haramaya University. Soil data collection and performance evaluation of the prototype was conducted on research farm in the university. The university is located in Oromia regional state in east Hararge zone at about 510 km distance from Addis Ababa. Haramaya University is located at 42° 2' 7" E longitude and 9°25' 15" N latitude as shown on Figure 5 (Turyasingura et al., 2023) at average elevation of 2043 meters above sea level. The area has a bimodal rainfall distribution with mean annual rainfall of 790 mm and experiences mean lowest temperatures of 12.6^{o}C and mean high temperature of 28.5^{o}C (Abebe et al., 2015).

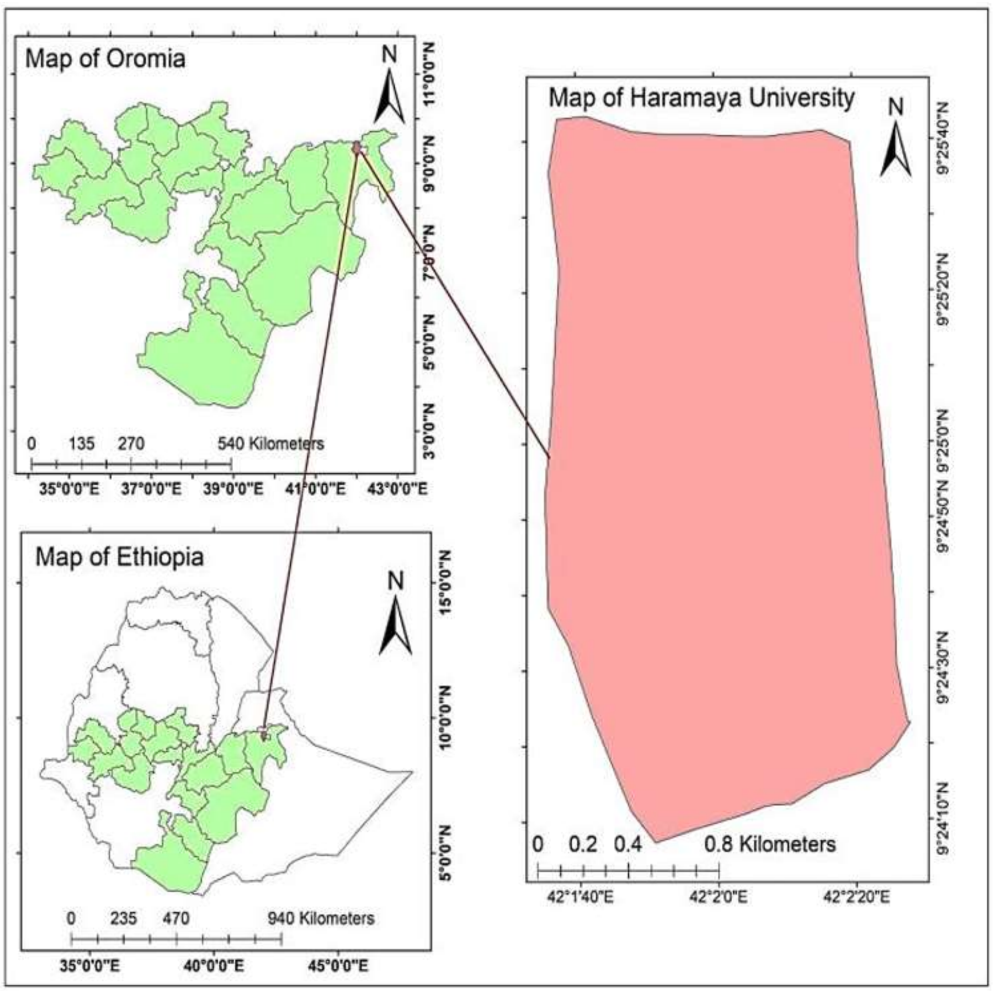


Figure 5. Map of the study area

3.2. Materials and Instruments

To fabricate the prototype of the digger, mild steel angle irons, sheet metals, round bars, square hallow bars, pipes, high-carbon steel flat bars, bolts and nuts, bearings, belts, pulleys, bevel gears, chain and sprockets were used and it was driven by petrol engine. These materials were selected due to the reasons that these materials are locally available, low cost and has good durability. In order to collect soil parameter data, different instruments was used. Table 1 shows the instruments that were used during soil testing and experiments that was done for performance evaluation the digger.

Table 1. Instruments used

No	**Type of instruments**	**Model**	**Accuracy**
1	Pilcon Hand Vane Tester	OR4548	1 kpa
2	Vernier calliper	Mitutoyo 530-122	0.02 mm.
3	Balance	WT-A	0.01 g
4	Measuring tape	RH-9034	1mm
5	Drying oven	STH-4A	±1°C
6	Tachometer	DT-2230	0.10 rpm

3.3. Determination of Physio-mechanical Properties of Soil

The soil factors that affect the power requirement and functionality of the digger were soil shear strength, soil bulk density, soil moisture content, angle of repose of soil and soil condition and in this study the ideal stone-free soil condition was considered. These factors were determined as follows.

3.3.1. Shear strength

The handheld vane shear test is a reliable method for determining undrained shear strength in situ, like clay soil, where traditional sampling methods may fail (Chung et al., 2007). The test can be performed in both horizontal and vertical planes, providing a thorough understanding of soil behaviour under diverse loading conditions. (Grabowski & Mary, 2014).

Clay soils generally exhibit higher shear strength compared to sandy soils due to their cohesive nature and lower porosity. Clay particles, being smaller and more fine-grained, have greater surface area and more electrostatic attraction, leading to stronger particle-to-particle bonds and thus higher shear strength (Yokoi, 1968). Due to these factors, clay soil was selected for the experiment and the shear strength of the soil was measured on field using Pilcon Hand Vane Tester and an undrained shear strength on both horizontal and vertical surface were determined. The 19mm diameter vane was used to measure the shear strength of the soil on vertical plane and 33mm vane was used to measure the shear strength of the soil on horizontal plane. The shear strength test was conducted on five plots in the field.

3.3.2. Moisture content

According to Smith et al. (1994), the soil moisture content on dry weight basis can be determined using equation (1) by weighing the soil sample before and after drying it in oven. In this study, five soil samples were taken from five different sites in the fields at 15.00 cm depth using soil sampling kit. The samples were transferred to a container of known weight ($\mathrm{m_1}$) and weighed immediately ($\mathrm{m_2}$) and their weight recorded. The laboratory samples were oven dried at 105°C for 8 hours, cooled and then re-weighed ($\mathrm{m_3}$). Then, the moisture content of the soil was determined using equation (1) and the results are presented in Table 5.

$$\mathrm{w_o} = \frac{\mathrm{m_2 - m_3}}{\mathrm{m_3 - m_1}} \times 100 \qquad (1)$$

Where:

$\mathrm{w_o}$ = soil moisture content (%)

$\mathrm{m_1}$ =mass of container (g)

m_2 = the initial mass of the soil sample (g)

m_3 =the final dry mass of the soil sample (g)

3.3.3. Bulk density

To determine wet soil bulk density, soil samples are usually taken with hand sampler or a tube sampler, which reduces disturbance and compaction. The soil sample's wet weight is recorded shortly after collection to prevent moisture loss (Doran and Mielke, 1984). The volume of the soil sample is calculated depending on the size of the sampling tube used, assuring accurate estimates (Shaw, 1917). Soil samples were taken from five different sites in the fields using sampler tube and weighed in laboratory. The volume of the sample was determined by measuring diameter and height of the soil sample inside the sampler tube. Then, the wet soil bulk density was calculated using equation (2) (Gautam A et al., 2019).

$$\rho_s = \frac{m_s}{v_s} \tag{2}$$

Where:

ρ_s = density of the soil (g/cm^3)

m_s = mass of the soil sample (g)

v_s =volume of the soil sample (cm^3)

3.3.4. Angle of repose

A fixed funnel method of determining angle of repose was used to determine the repose angle of the soil. A funnel was used to pour the soil through to create a cone. Then the angle of repose was determined using equation (3) after measuring the height and width of the created soil cone.

$$\theta_r = tan^{-1}\left(\frac{2h}{d}\right) \tag{3}$$

Where:

θ_r = angle of repose (°)

h = height of the soil cone (m)

d = diameter of the soil cone (m)

3.4. Description of Working Principle of the Digger

The machine which designed and fabricated in this study was transplanting hole digger, which its overall structure is shown in Figure 6. The main purpose of the digger was to dig hole on agricultural land and hills for planting seedlings of perennial crops and trees. The machine mainly has two frames (main frame and digger frame) which one slides up and down on the other. The digger frame (Figure 13) slides up and down on the main frame (Figure 14). All the functional components of the digger such as soil cutting components (soil cutting tool and rotor shaft), soil elevator components (bucket elevators and soil discharging chute), power source and transmission systems (petrol engine, bevel gears, shafts, belts and pulleys) and up and down moving pinion and rack system are assembled on digger frame as shown on Figure 6 and the handle and ground wheels are assembled on main frame.

The digger digs a hole on the basis of soil cutting and elevating system. Soil cutting tool cuts the soil and soil elevating system lift up the cut soil to the ground surface and consequentially creates a hole. The cutting component are rotor shaft and soil cutting tool. The soil cutting tool is attached on the bottom end of the rotor shaft. The top end of rotor shaft is engaged to power source via power transmission system. Then when the power source start up, the rotor shaft with soil cutting tool revolve and cut the soil. The soil elevating component have two bucket elevators installed on digger frame side by side along the length of the rotor shaft. The digger can move up and down on main frame by means of rack and pinion feeding system. As shown on Figure 15, the top end of rotor shaft and elevators drive shaft are inter connected by bevel gear drive system. Again elevator drive shaft is engaged to elevators shaft by v-belt drive system on its ends. The elevator shafts drive the elevator belts which buckets are installed on. When the engine stats to run the soil elevator system also run and lifts up the soil under it. Therefore, the soil cutting tool cuts the soil and soil elevator lift up the soil simultaneously. The feeding system of the digger in to the ground is by manual means. Manual feeding handle is installed on digger frame to push down the soil cutter and elevator into the hole and to pull up it after completion of digging operation to the desired depth. Feeding the digger continuously into the ground, makes continues soil cut and excavation which creates a hole to the desired level.

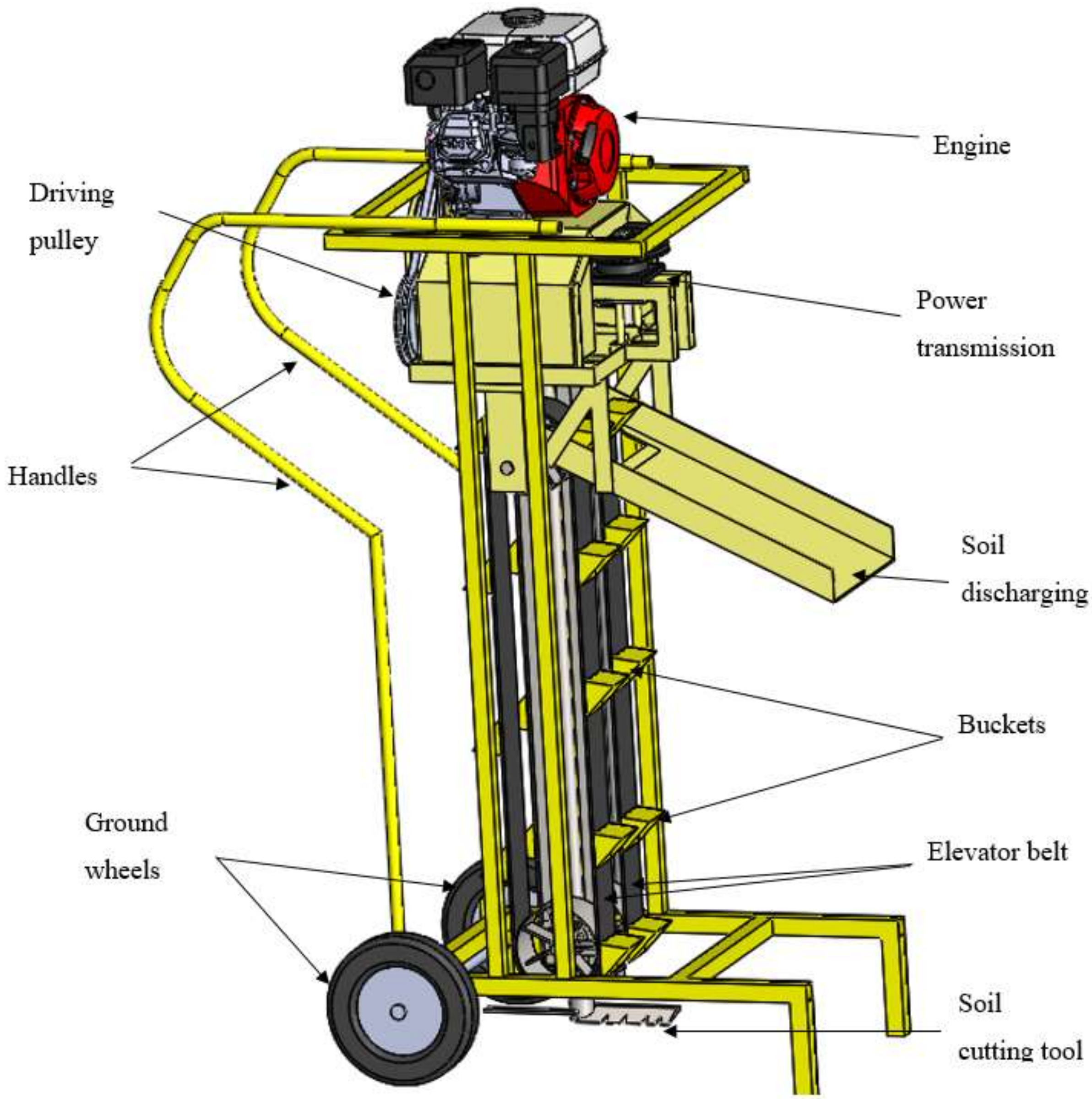


Figure 6. Schematic drawing of the motorized hole digger

3.5. Design Parameters and Requirements

The parameters and requirements considered in designing the digger were depth of hole, diameter of hole, working speed, soil shearing strength, soil lifting rate and safety of operator.

3.6. Design Specification

In designing the proposed hole digger, parameters (maximum working depth, soil cutting diameter, cutting depth, bucket capacity, cutter feed rate and speed of cutter) were specified as a starting point based on different reasons. According to Usha et al. (2015) recommendation, the hole size for planting fruit crops should be 61.00 cm deep and to accommodate the entire root system, the hole width should be approximately twice of the diameter of the seedling root system. In other hand, according to Megersa (2022), the recommended diameter and depth of hole for coffee planting is 60.00 and 60.00 cm, respectively. According to Edward et al. (2005), for planting coffee, the hole should be dug with 60.00 cm in length, width and depth each.

Applying the above recommendations, the maximum depth of hole dug by the digger was specified to be 600.00cm. However, to reduce energy requirement by the digger, the radius of hole and depth of cutting per revolution were fixed to be 160.00 and 12.00 mm respectively. To increase the digging capacity and to shorten digging time, the speed of the soil cutter was decided to be 180 rpm as shown in Table 2.

Table 2. Design specifications

Parameters	**Specifications**
Maximum depth of hole	600.00 mm
Diameter of hole per one drill	320.00 mm
Depth of cutting per revolution	12 mm
Bucket theoretical capacity	100.01 cm^3
Cutter downward feeding rate	2.00 cm/sec
Working speed of soil cutter	180.00 rpm

3.7. Design Analysis of Soil Cutting System

The soil cutting system of the digger is the part of the digger that cuts a layer of soil. The system is comprised of soil cutting tool and rotor shaft.

3.7.1. Soil cutting tool

The soil cutting tool was made up of AISI1095 high carbon steel plate welded to a tip of hollow shaft. The schematic drawing of the tool is shown on Figure 7.

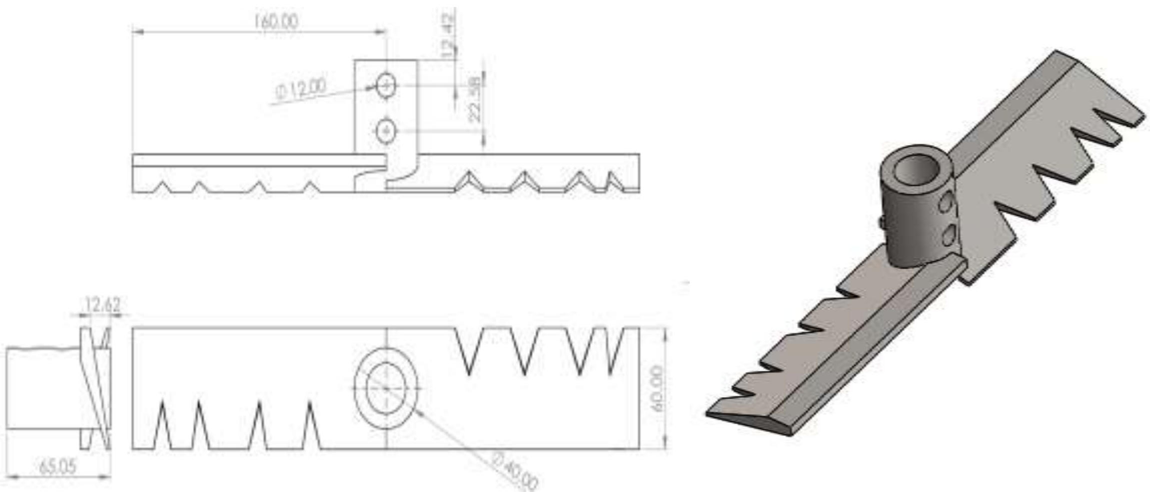


Figure 7. Schematic drawing of soil cutting tool

Based on principle of design for the maximum, the maximum values of soil shear strength on horizontal and vertical plane (3.70 and 39.00 MPa) taken from soil shear test were used to determine soil shearing torque required by the soil cutting tool of the digger to overcome soil shear resistance. Therefore, applying these soil shear strength values, 0.32 m digging diameter and 0.012 m soil cutting depth, the shearing torque required to shear the soil on vertical and horizontal plane and the total soil shearing torque were found to be 75.24, 63.45 and 138.69 Nm respectively determined using equation (4), (5) and (6) given by Mir (2021). The size of the tool was determined based on simulation study done on it against these soil resistance torques using solid work 2022 design software as shown on Appendix Figure 1.

$$T_V = C_{uv}(\pi dh)\frac{d}{2} \tag{4}$$

$$T_h = C_{uh}\left(\frac{\pi d^2}{4}\right)\left(\frac{2d}{3}\right) \tag{5}$$

$$T_{max} = C_{uv}(\pi dh)\frac{d}{2} + C_{uh}\left(\frac{\pi d^2}{4}\right)\left(\frac{2d}{3}\right) \tag{6}$$

Where:

d =soil cutting diameter of the digger (m)

h =soil cutting depth of the digger (m)

C_{uv} = undrained shear strength of the soil on vertical plane (kPa)

C_{uh} = undrained shear strength of the soil on horizontal plane (kPa)

3.7.2. Rotor shaft

The rotor shaft (Figure 8) is solid round bar that transmits rotational power from transmission system of the digger to soil cutting tool at speed of 180 rpm. According to Khurmi and Gupta (2005), for designing transmission shafts based on rigidity, deflection angle from 2.5 to 3 degree can be used as limiting value of shaft deflection per meter length. The shaft transmits 158.39 Nm torque to soil cutting tool as calculated by equation (7). Therefore based on torsional rigidity theory, the diameter of 1065.00 mm long mild steel shaft at limiting deflection of 2.5 degree was determined using equation (8) and found to be 25.00 mm.

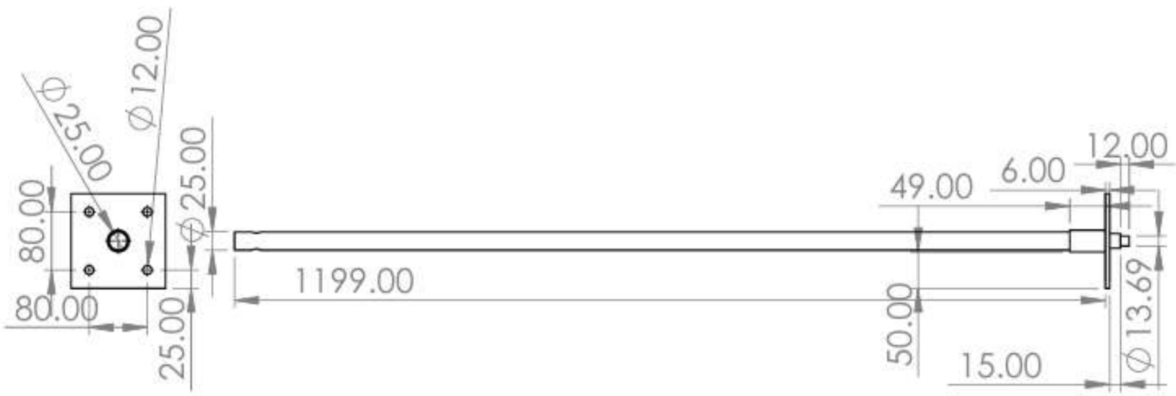


Figure 8. Schematic drawing of rotor shaft

$$T = \frac{P \times 60}{2\pi \times N_r} \quad (7)$$

$$D^4 = \frac{32T \times L}{\pi\theta \times G} \quad (8)$$

Where:

θ= torsional deflection (rad)

T= torque on shaft (Nm)

L = length of the shaft (m)

D= diameter of the shaft (mm)

G= modulus of rigidity of shaft material (MPa)

P = power of engine (hp)

N_r = speed of the shaft (rpm)

3.8. Design Analysis of Soil Excavating System

Soil excavating system is the part of the machine that comprises two bucket elevators arranged side by side on the pulley guide frame as shown in Figure 9.

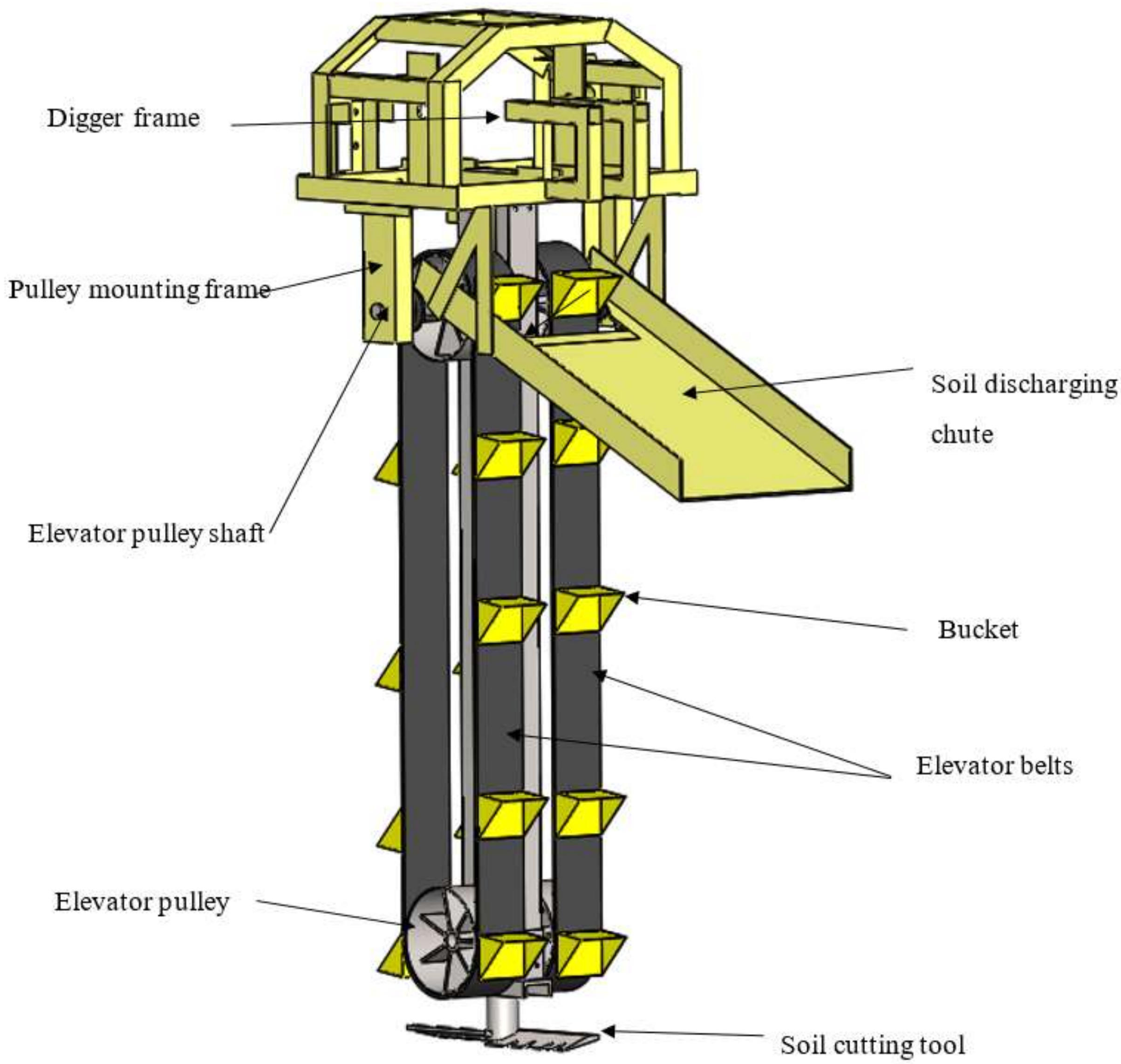


Figure 9. Schematic drawing of soil elevator system

3.8.1. Capacity of the bucket elevator

According to Siddhartha (2008), elevators are divided in to centrifugal discharge elevators, positive discharge elevators and Continuous discharge elevators based on bucket spacing and mode of material discharge. In this study, a centrifugal discharge elevator was selected, since the buckets were spaced at a regular intervals to avoid interference in loading and discharging.

Since the digger had operating diameter of 32.00 cm, depth of 60.00 cm and soil cutter feed rate of 1cm/s, the expected soil elevating capacity of the elevator was $803.84\,cm^3/s$ as determined by equation (9). Since the digger had two elevators, the soil elevated per one elevator was $401.92\,cm^3/s$.

$$c_e = \frac{\pi D^2}{4} \times r \qquad (9)$$

Where:

c_e = soil elevating capacity of the elevators (cm^3/s)

D = operating diameter of the digger (cm)

r = feed rate of the cutter (cm/s)

3.8.2. Bucket

Mild steel sheet metal was used to fabricate the buckets and each of the buckets were designed to hold $100.00 cm^3$ volume of soil. The number of buckets required per elevator were 4 buckets/s as determined by equation (10). The schematic drawing of the bucket is shown on Figure 10.

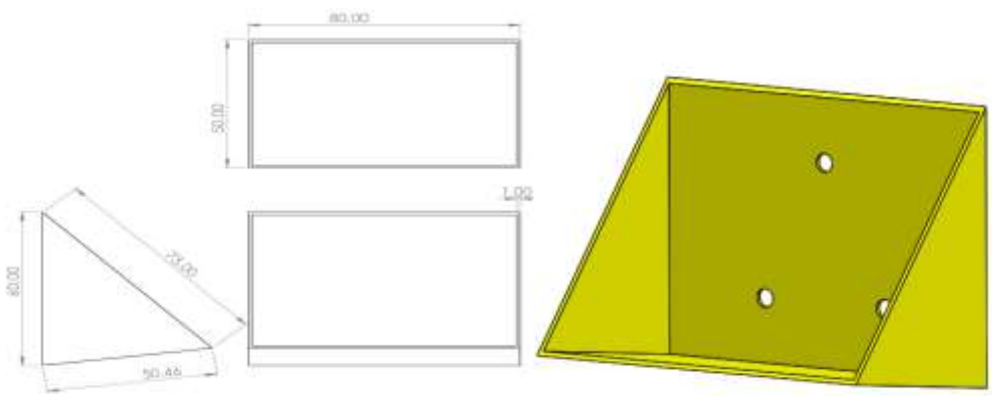


Figure 10. Schematic drawing of bucket

$$N_b = \frac{C_{Pe}}{C_b} \qquad (10)$$

Where:

N_b=number of buckets required per elevator

C_{pe}= soil elevating capacity per elevator (cm^3/s)

C_b= bucket capacity (cm^3)

3.8.3. Bucket drive system

Even though both belt and chain drive systems are possible for driving the bucket elevators, belt drive system was selected for this study due to its easiness in installation, operation as well as for maintenance.

i. Determination of diameter of elevators pulley

The diameter of the elevator pulley(Figure 11) was determined by equating centrifugal force created during belt rotation and gravitational force on the soil equation (11)given by Mg et al. . (2019). Deciding the rpm of the pulley to be 112.50 rpm and using trial and error method, gravitational and centrifugal force becomes equivalent, when radius of the pulley equals to 0.0708m. Accordingly, the diameter of pulley was found to be 141.60 cm and it's had 80.00 mm face width. Then, applying equation (12), the velocity of the pulley was found to be 0.83 m/s.

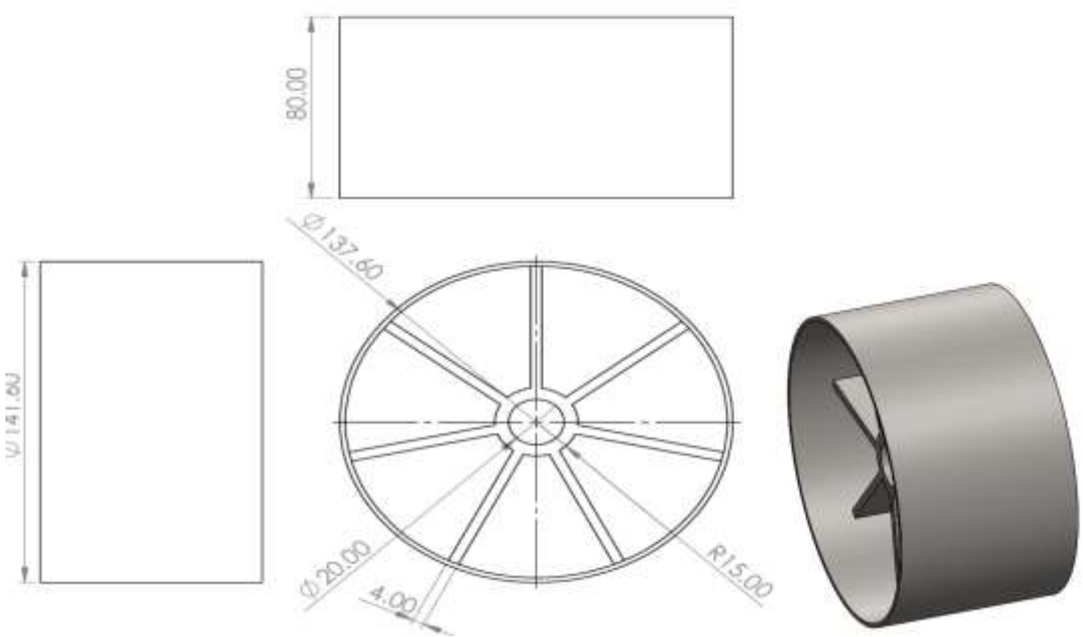


Figure 11. Schematic drawing of elevator pulley

$$mg = \frac{mv^2}{r} \quad (11)$$

$$v = \frac{2\pi r \times N_r}{60} \quad (12)$$

Where:

V = belt speed (m/s)

r = radius of pulley (m)

m = mass of soil (kg)

g = gravitational acceleration (m/s^2)

ii. Bucket spacing

The bucket spacing on the belt was determined using equation (13) and was found to be 20.75 cm/bucket.

$$P_S = \frac{V}{P_n} \quad (13)$$

Where:

V = belt speed (cm/s)

P_n =number of buckets required per second

P_s = bucket spacing (cm/bucket)

iii. Required length of the belt

Taking 83.47cm centre distance between pulleys, the required length of the belt was found 211.00 cm as determined using equation (14) (Khurmi & Gupta, 2005).

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \quad (14)$$

Where:

L=length of belt (cm)

r_1=radius of first pulley (m)

r_2=radius of second pulley (m)

x = centre distance between elevator shafts (m)

iv. Required number of buckets

The total required number of buckets per elevator were determined by applying equation (15) and were found to be 10.00 buckets.

$$b_t = \frac{L}{P_n} \quad (15)$$

Where:

b_t= total required number of buckets per elevator

L=length of belt (cm)

P_n =number of buckets required per second

v. Selection of elevators belt size

The estimated weights per meter of soil in each bucket, the bucket itself and the belt were 9.19 N/m, 8.24 N/m and 8.57N/m respectively. The tension of the bucket elevator belt in tight and slack side, centrifugal tension due to centrifugal force and maximum tension in the belt were found 21.70 N, 14.55 N, 2.55 N and 38.80 N respectively as determined using Eqn. (16), (17), (18) and (19) which are given by Raymond and Kulwiec (1985).

$$T_1 = (W_c + W_b + L_l) \times X \quad (16)$$

$$T_2 = (W_c + W_b) \times X \quad (17)$$

$$T_c = (\frac{T_1 + T_2}{g}) v^2 \quad (18)$$

$$T = T_1 + T_2 + T_c \quad (19)$$

Where:

T =maximum tension in the belt (N)

T_1 = tension in the tight side of belt (N)

T_c = tension due to centrifugal force (N)

T_2 =tension in the slack side of belt (N)

W_c =weight of belt (N/m)

W_b =weight of bucket(N/m)

L_b = live load (N/m)

x = centre distance between elevator shafts (m)

The selected rubber belt has 80.00mm width, 7.00mm thickness and allowable stress of 64 MPa. As tested using Eqn. (20), the selected belt can withstand the total tension applied on it.

$$b \times t = \frac{T}{\sigma} \quad (20)$$

Where:

T =maximum tension in the belt (N)

σ = allowable stress (N/m^2)

b = width of belt (m)

t = thickness of belt (m)

3.8.4. Soil discharging chute

A sloping metallic channel was designed for receiving the soil from the revolving buckets and conveying it to the ground surface. The chute was made adjustable to make the discharging slop, greater than soil-meta friction and soil repose angles as required. Mild steel sheet metal having 1.00 mm thickness was used to fabricate it. The overall dimension of the chute was shown on Figure 12.

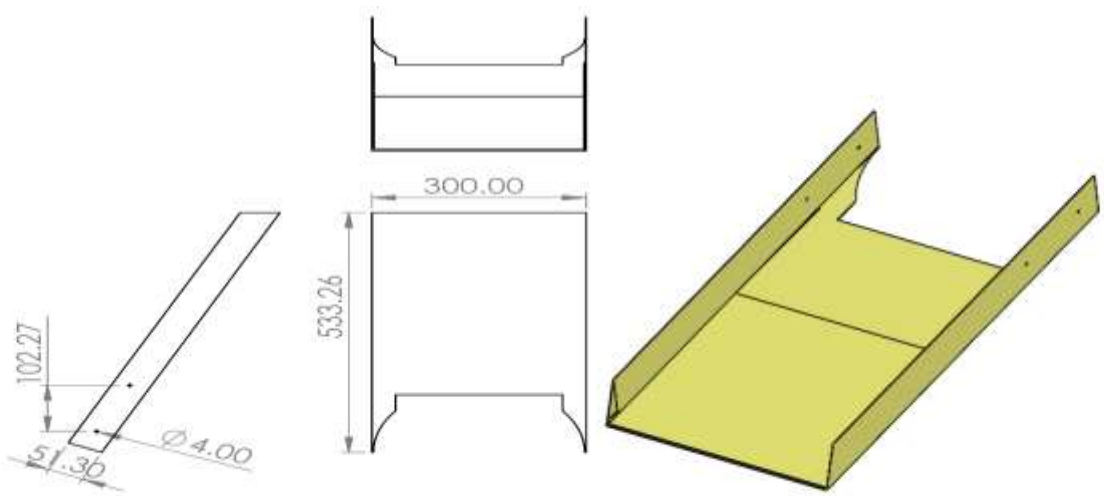


Figure 12. Schematic drawing of soil discharging chute

3.9. Determination of Required Engine Power

The power required by the digger is the summation of power required by soil cutting tool and soil elevator system. Therefore, the power required by soil elevator and soil cutting tool were calculated using Eqn. (21) and (22) given by Khurmi and Gupta (2005) and found to be 0.09 and 3.52 hp respectively. Then, the total engine power required was 3.61 hp as determined by equation (23). Therefore, approximately 4 hp gasoline engine was selected as a power source for the digger.

$$P_e = 2T_{tmax} \times v_e \tag{21}$$

$$P_r = \frac{T_{max} \times 2\pi N}{60} \tag{22}$$

$$P_d = P_e + P_r \tag{23}$$

Where:

T_{max} = maximum torque due to soil resistance (Nm)

P_e = power required by the elevators (w)

P_r = power required by the soil cutting tool (w)

P_d = power required by the digger (w)

N= speed of rotor shaft (rpm)

T_{tmax}= maximum tension load on the elevators (N)

v_e = velocity of elevator belt (m/s)

3.10. Design of Machine Frame

3.10.1. Digger frame

The digger frame (Figure 13) is the part of the machine which, engine, soil elevators and power transmission systems were assembled on it. It was fabricated from angle iron having cross-sectional dimension of 30.00 mm×30.00 mm×1.50 mm and 25.00 mm×25.00 mm×1.50 mm and square hallow bar of 25.00 mm×25.00 mm×1.50 mm dimension.

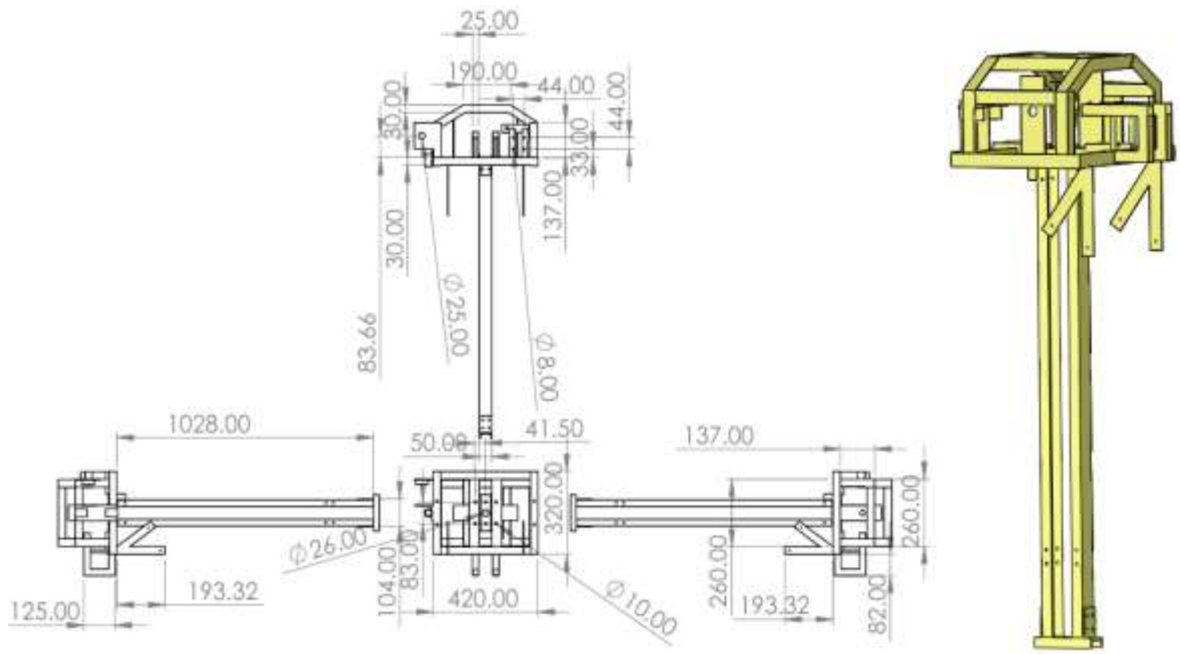


Figure 13. Schematic drawing of digger frame

3.10.2. Main frame

The main frame (Figure 14) is the back bone of the digger which all the components were assembled up on it and made from mild steel hallow square bars. It supports the loads that come from weight of digger, engine and power transmission systems. It was fabricated from $30.00\ mm \times 30.00\ mm \times 1.50\ mm$ and $25.00\ mm \times 25.00\ mm \times 1.50\ mm$ mild steel square hallow bars and it has 725.00 mm and 1469.00 mm overall width and height respectively.

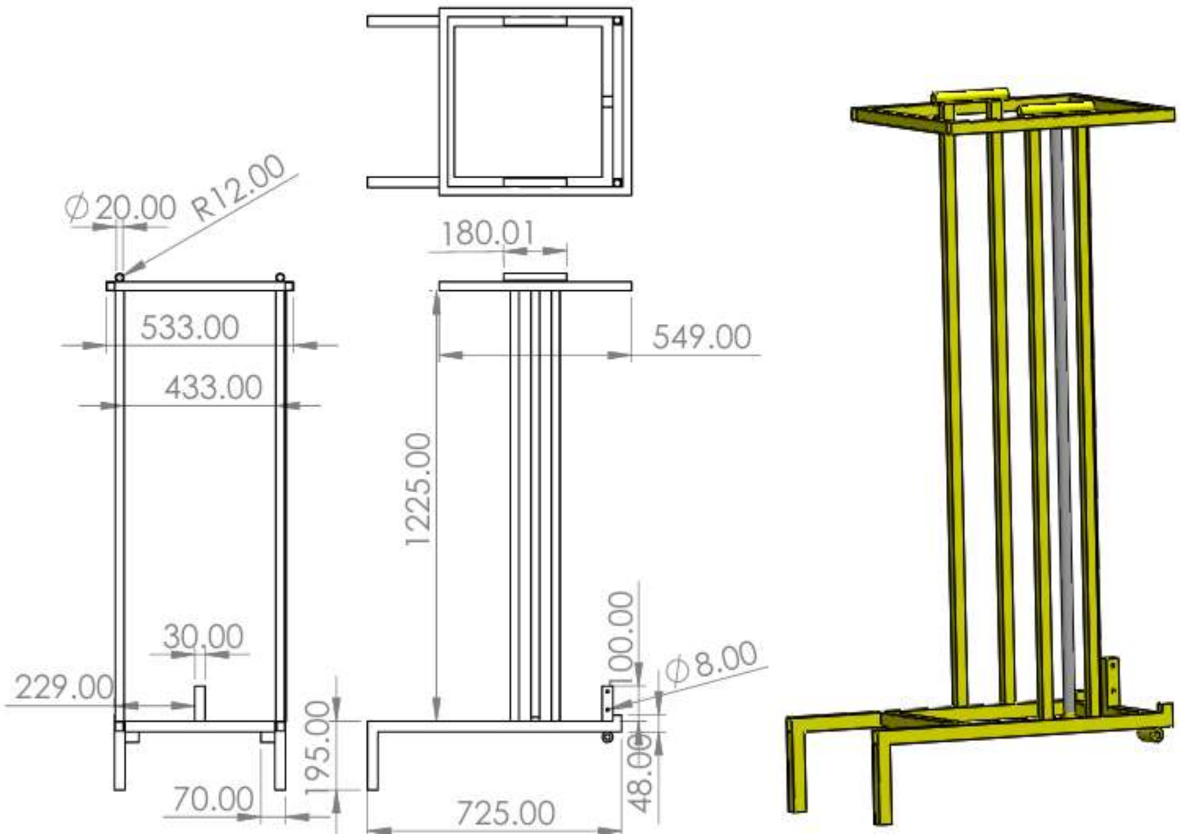


Figure 14. Schematic drawing of main frame

3.11. Power Transmission Components

The power transmission components (Figure15) of the machine has two bevelled gear mechanisms, one chain drive and four v-belt drive systems.

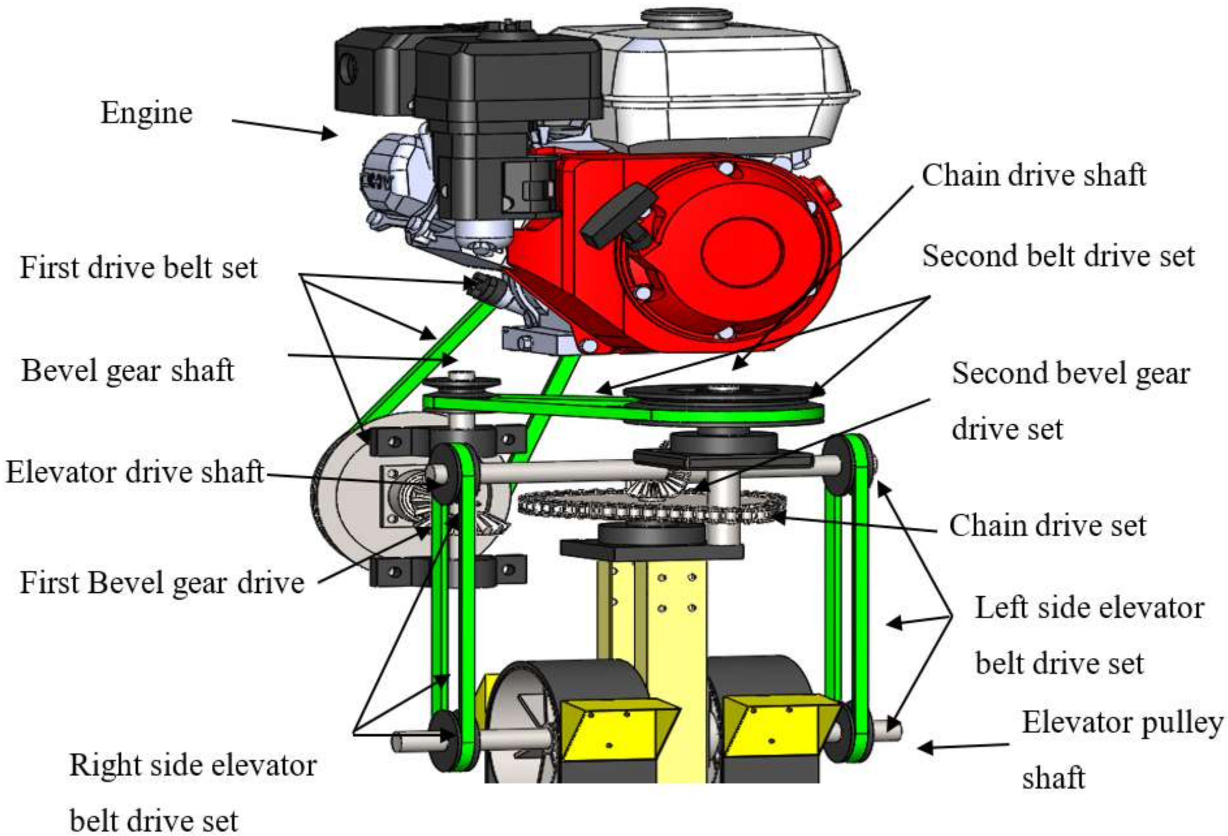


Figure 15. Schematic drawing of power transmission system

3.11.1. V-belt, chain and gear drive component

To reduce 3600 rpm of engine speed in to 180 rpm speed of soil cutting tool, two v-belt drive and one chain drive system were used. The first pulley set was intended to reduce 3600 rpm to 1663.19 rpm which have 75.00 mm and 162.34 mm smaller and larger pitch diameter of pulleys driven by A-type v-belt were selected based on Indian standard 2494-1974 for belt and pulley selection and the rotation of the drive is on vertical plane. Then, to convert the vertical rotation on to horizontal rotation, bevel-gear mechanism having 13 smaller gear teeth and 18 larger gear teeth were selected and the rpm was reduced to 1201.20 rpm. Again a second set of v-belt drive having 75.00 mm and 143.00 mm pitch diameter of smaller and larger pulley which driven by A-type v-belt was selected and installed to reduce 1201.20 rpm to 630.00

rpm. Then after a chain drive having 14 and 49 tooth smaller and larger sprockets tooth respectively was selected to covert 630 rpm to 180 rpm.

The maximum and centrifugal tension on the belt of A-type were found to be146.69 and 21.96N as determined by Eqn. (24) and (25) given by Khurmi and Gupta (2005). The velocity of the belt was 14.13m/s determined by Eqn. (26).

$$T = \sigma \times A \tag{24}$$

$$T_c = m \times v^2 \tag{25}$$

$$v = \frac{\pi d_1 N_1}{60} \tag{26}$$

Where:

T = maximum tension in the belt (N)

T_c = centrifugal tension (N)

σ = 1.75MPa, allowable stress of belt material

A = cross-sectional area of the belt (m^2)

V = velocity of belt (m/s)

N_1 = speed of belt (rpm)

m=0.11kg/m, mass of the belt

N_1= diameter of smaller pulley (m)

The contact angle between the belt and pulley was calculated by equation (27) and (28) which given by Khurmi and Gupta (2005).

$$\theta_s = 180^{\circ} - 2sin^{-1}\left(\frac{r_2 - r_1}{x}\right) \tag{27}$$

$$\theta_l = 180^{\circ} + 2sin^{-1}\left(\frac{r_2 - r_1}{x}\right) \tag{28}$$

Where:

θ_s=contact angle on smaller pulley (°)

θ_s= contact angle on larger pulley (°)

r_1= radius of smaller pulley (m)

r_2= radius of larger pulley (m)

The maximum tension in tight and slack side of the belt were 124.73 and 10.91N respectively as determined using Eqn. (29) and (230) given by Khurmi and Gupta (2005). Here, 0.3

coefficient of friction between belt and pulley, 2.372rad contact angle and 17degree value of β were used.

$$T_1 = T - T_c \tag{29}$$

$$2.3log\left(\frac{T_1}{T_2}\right) = \mu\theta \times cosec\beta \tag{30}$$

Where:

T_1 = tension in the tight side of the belt (N)

T_2 = tension in the slack side of the belt (N)

T = maximum tension in the belt (N)

T_c = centrifugal tension (N)

θ_s= contact angle between belt and pulley (°)

μ= coefficient of friction

β= half of groove angle (°)

The power transmitted by the belt and the number of belts required was 1,608.23w and 1 respectively, as calculated by Equation (31) and (32) given by Khurmi and Gupta (2005).

$$P_b = (T_1 - T_2)v \tag{31}$$

$$N_b = \frac{P_t}{P_b} \tag{32}$$

Where:

P_b = power transmitted per belt (W)

P_t = total power to be transmitted (W)

N_b = number of belts required

V = velocity of belt (m/s)

T_1 = tension in the tight side of the belt (N)

T_2 = tension in the slack side of the belt (N)

Using the same procedure from equation (28) to (34), the value of T,T_1,T_2,T_c, v, P_b and N_b for second pulley drive set were found to be 146.69 N, 144.25 N, 7.68N, 2.44 N,4.71 m/s, 643.25 w and 2 respectively.

3.11.2. Bearings

The selection of the bearings was based on load carrying capacity and the geometry of the bearing that, insure to be installed conveniently on the machine. Ball type bearings was selected for all shafts. Square flange bearing with bearing number 206, pillow block bearing with bearing number 204, 205 and others with bearing number 6204 and 6205 were used.

3.11.3. Pinion shaft

The effective loads on the pinion shaft are tangential load from bevel gear drive and the tensioning force transmuted by the belt (135.64 N).The torque transmitted by the shaft and the tangential force acting the shaft were 17.14 Nm and 612.14 N, which determined by eqn. (33) and (34) given by Khurmi and Gupta (2005).

$$T_p = \frac{P \times 60}{2\pi \times N_p} \quad (33)$$

$$W_T = \frac{T_p}{R_m} \quad (34)$$

Where:

T_p= torque on pinion shaft (Nm)

P= power of engine (W)

W_T = tangential load (N)

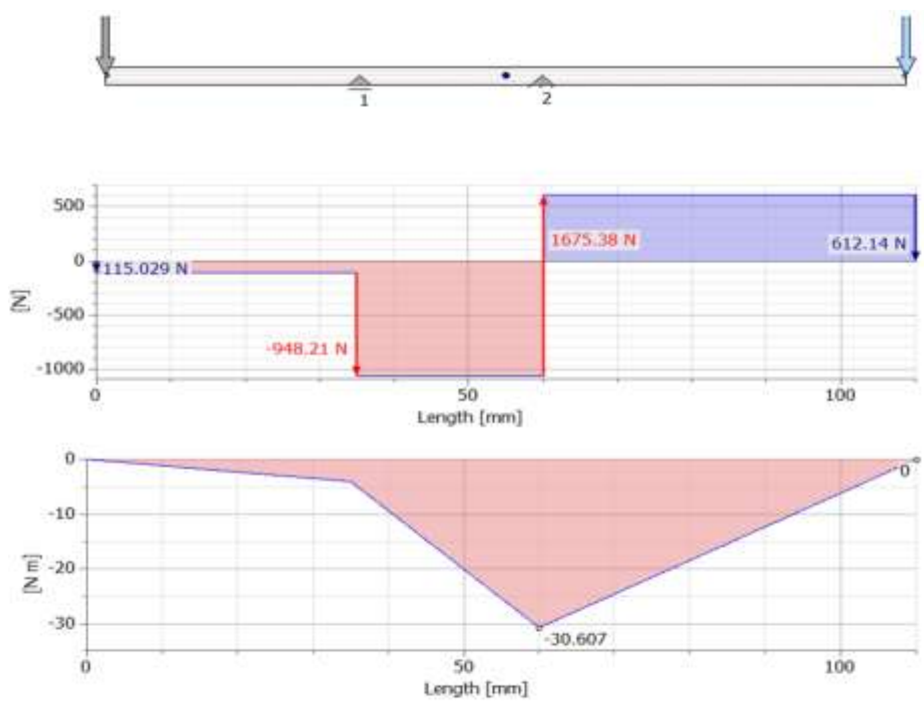


Figure 16.Shear force (b) and bending moment (c) diagrams of pinion shaft

Taking the maximum bending moment (30.61 Nm) from Figure 16, the diameter of the shaft was computed using equation (35) given by Khurmi and Gupta (2005) and found to be 20.00 mm.

$$d_p{}^3 = \frac{16}{\pi\tau}\sqrt{\left(k_m \times M_p\right)^2 + (k_t \times T_p)^2} \quad (35)$$

Where:

T_p= torque on pinion shaft (Nm)

R_m = 0.028m, mean radius of pinion gear

d_p=diameter of the pinion shaft (mm)

τ = allowable shear stress for the material of the pinion shaft (42MPa)

k_m =2, combined shock and fatigue factor for bending

k_t =2, combined shock and fatigue factor for torsion

3.11.4. Gear shaft

The effective forces acting on the gear shaft are tangential load and tensional load (151.93 N) from second pulley drive set. The torque transmitted by gear shaft and the tangential load acting on the shaft were computed using eqn. (36) and (37) given by Khurmi and Gupta (2005) and found to be 23.73 Nm and 697.94 N respectively.

$$T_g = \frac{P \times 60}{2\pi \times N_g} \quad (36)$$

$$W_T = \frac{T_g}{R_m} \quad (37)$$

Where:

T_g= torque on gear shaft (Nm)

R_{mg} = 0.034m, mean radius of gear

d_g=diameter of the gear shaft (mm)

N_g= speed of gear (rpm)

P= power of engine (W)

Taking the maximum bending moment (11.44 Nm) from Figure 17, the diameter of the gear shaft was determined to be 20.00mm, using Eqn.(38) given by Khurmi and Gupta (2005).

$$d_g{}^3 = \frac{16}{\pi\tau}\sqrt{\left(k_m \times M_g\right)^2 + (k_t \times T_g)^2} \quad (38)$$

Where:

T_g= torque on gear shaft (Nm)

M_g= maximum bending moment on gear shaft (Nm)

d_g=diameter of the gear shaft (mm)

τ = 42MPa, allowable shear stress for the material of shaft

k_m = 2, combined shock and fatigue factor for bending

k_t =2, combined shock and fatigue factor for torsion

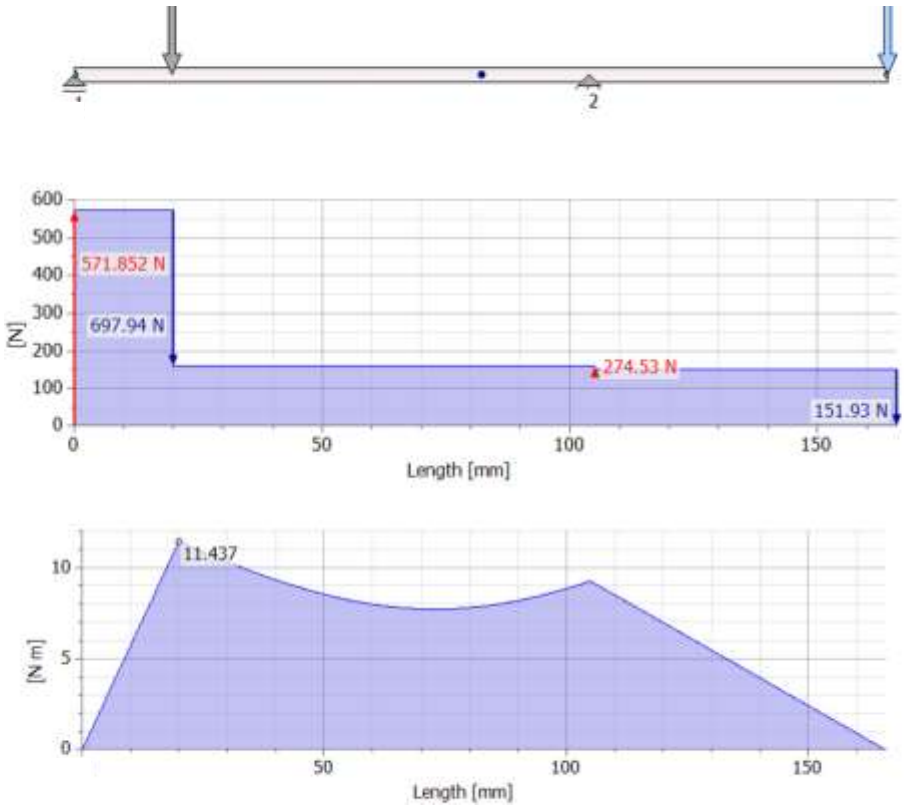


Figure 17. Shear force (b) and bending moment (c) diagrams of gear shaft

3.11.5. Chain drive shaft

The loads acting on the chain drive shaft are the tension force due to belt and chain which is 151.93 N for each and the maximum bending moment due to this forces was 10.33 Nm as show on Figure 18. The torque delivered by the shaft and the diameter of the shaft were calculated using Eqn. (39) and (40) given by Khurmi and Gupta (2005) and found to be 45.25Nm and 25.00 mm respectively.

$$T_{cs} = \frac{P\times 60}{2\pi\times N_{cs}} \tag{39}$$

$$d_{cs}{}^{3} = \frac{16}{\pi\tau}\sqrt{(k_m \times M_{ce})^2 + (k_t \times T_{cs})^2} \quad (40)$$

Where:

M_{cs}=maximum bending moment chain drive shaft (Nm)

T_{cs}= torque on chain drive shaft (Nm)

d_{cs}=diameter of the chain drive shaft (mm)

N_{cs}= speed of chain drive shaft (rpm)

P= power of engine (W)

τ = allowable shear stress for the shaft material (42MPa)

k_m = 2, combined shock and fatigue factor for bending

k_t = 2, combined shock and fatigue factor for torsion

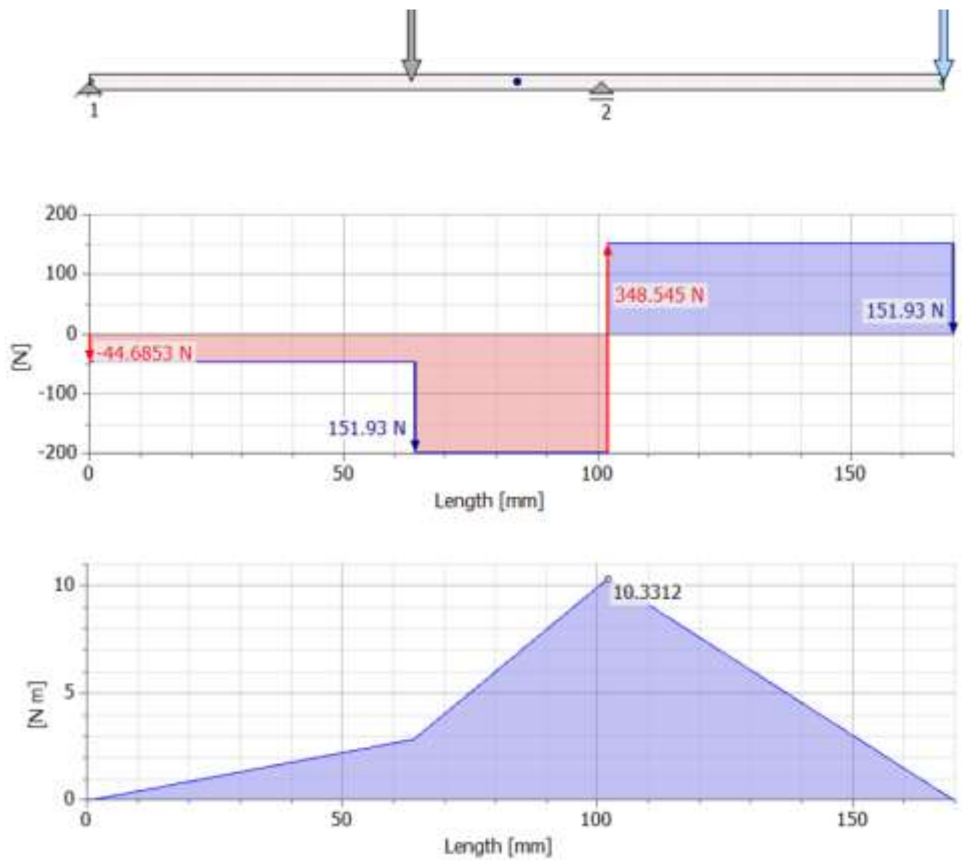


Figure 18. Shear force (b) and bending moment (c) diagrams of chain drive shaft

3.11.6. Elevators driving shaft

The effective loads on the shaft are tension (96 N) from elevator belts and tangential load from the bevel gear drive system. Using Eqn. (41) and (42) given by Khurmi and Gupta (2005), the torque transmitted by elevator shaft and tangential load due to bevel gear drive was computed and found to be 6.34 Nm and 186.34 N respectively.

$$T_{es} = \frac{P \times 60}{2\pi \times N_{es}} \quad (41)$$

$$W_{Te} = \frac{T_{es}}{R_m} \quad (42)$$

Where:

W_{Te}= tangential load on elevator drive shaft (N)

T_{es}= torque on elevator drive shaft (Nm)

N_{es}= speed of elevator drive shaft (rpm)

R_{mg} = mean radius of gear (0.034m)

P= power delivered through elevator shaft (W)

Then taking the maximum bending moment (17.61 Nm) from Figure19 and inserting in to Eqn. (43) given by Khurmi and Gupta (2005), the diameter of the shaft was determined to be 20.00 mm.

$$d_{es}{}^{3} = \frac{16}{\pi\tau}\sqrt{(k_m \times M_{es})^2 + (k_t \times T_{es})^2} \quad (43)$$

Where:

M_{es} = maximum bending moments on elevator drive shaft (Nm)

T_{es}= torque on elevator drive shaft (Nm)

d_{es}=diameter of the elevator drive shaft (mm)

τ =42MPa, allowable shear stress for the material of the shaft

k_m = 2, combined shock and fatigue factor for bending

k_t = 2, combined shock and fatigue factor for torsion

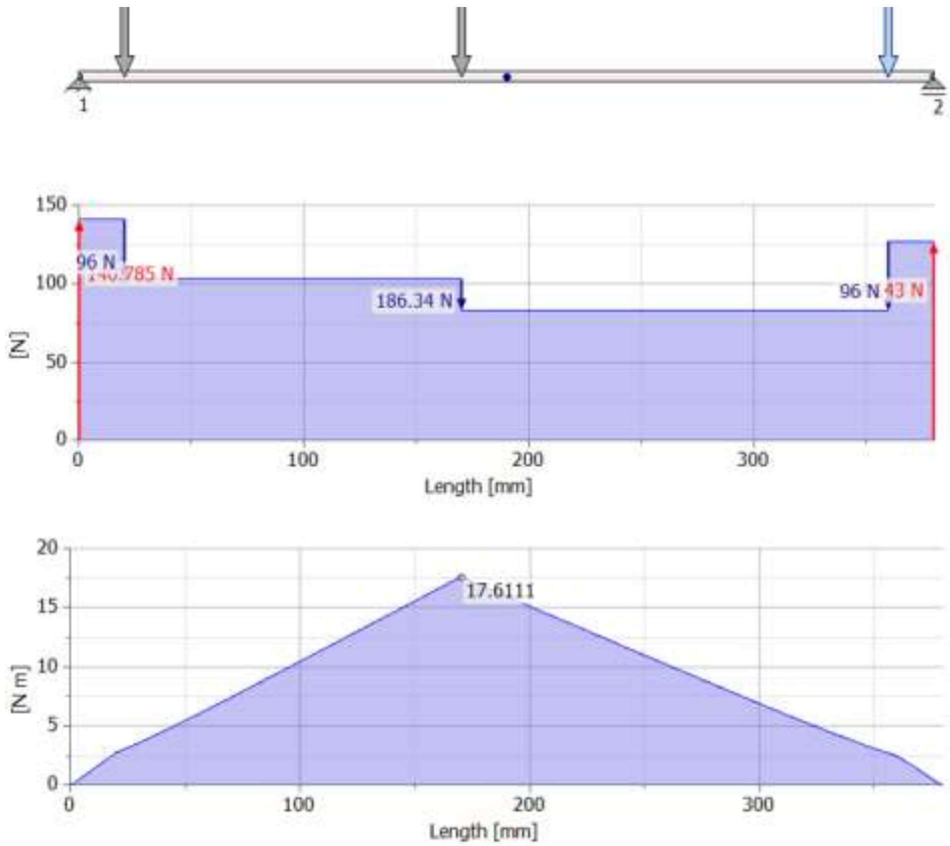


Figure 19. Shear force (b) and bending moment (c) diagrams of elevator drive shaft

3.11.7. Elevators shaft

The torque transmitted by elevator shaft and the total tension force (F_1) and (F_2) on the shaft were 2.85 Nm, 96.00 N and 40.27 N respectively as determined by Eqn. (44), (45) and (45) given by Khurmi and Gupta (2005).

$$T_e = \frac{P \times 60}{2\pi \times N_e} \quad (44)$$

$$F_1 = \frac{P \times 60}{2\pi r_1 \times N_e} \quad (45)$$

$$F_2 = \frac{P \times 60}{2\pi r_2 \times N_e} \quad (46)$$

Where:

F_1= tension force on drive pulley (N)

F_1= tension force on elevator pulley (N)

T_e= torque on elevator shaft (Nm)

P= power of transmitted by elevator shaft (hp)

r_1= 29.7 mm radius of elevator driving pulley

r_2= 70.8 mm, radius of elevator pulley

N_e=112.5 rpm, speed of elevator shaft

As shown on Figure 20, the maximum bending moment on the shaft was 3.20 Nm. Using Eqn. (47) (Khurmi and Gupta, 2005), the diameter of the shaft was determined and found to be 20.00 mm.

$$d_e{}^3 = \frac{16}{\pi\tau}\sqrt{(k_m \times M_{ed})^2 + (k_t \times T_{ed})^2} \qquad (47)$$

Where:

d_e= diameter of elevator shaft (m)

T_e= torque on elevator shaft (Nm)

M_e= maximum bending moment on elevator shaft (Nm)

τ = allowable shear stress for the material of the shaft (42MPa)

k_m =2, combined shock and fatigue factor for bending

k_t = 2, combined shock and fatigue factor for torsion

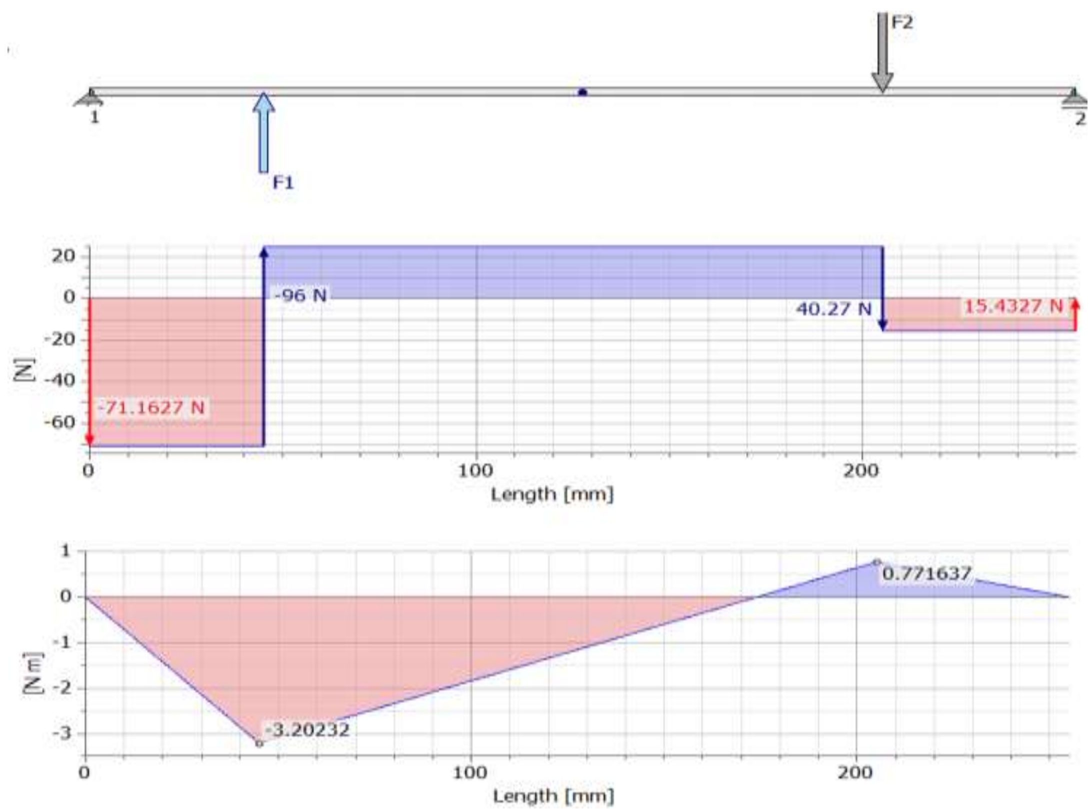


Figure 20. Shear force (b) and bending moment (c) diagrams of elevator shaft

3.12. Digger Up and Down Moving System

In order to move the digger into and out of the hole, a rack and pinion type driving system using chain and sprocket was installed on the digger. The chain was fixed vertically on main frame while small sprocket was attached on a rotating handle shaft through two bearings fixed on digger frame and the chain and the sprockets are made to mate. As the handle shaft rotated, the sprocket moves on the fixed chain which result in movement of the digger up and down as required.

3.13. Machine Mobility Component

The transporting component of the digger has two rubber ground wheels which are fixed at both bottom back sides of the digger on axil shaft. The wheels have 230mm diameter and 50mm width. The wheel shaft of digger (Figure 21) is subjected to bending moment due to weight of the digger. The digger has 1007.6 N weight and the maximum bending moment due to this weight was computed to be 17.63 Nm. Therefore the diameter of wheel shaft that withstand this load was determined using Eqn. (48) (Khurmi & Gupta, 2005) and found to be 15.00 mm as

$$d = \sqrt[3]{\frac{16M}{\pi\tau}} \tag{48}$$

Where:

d = diameter of wheel shaft (mm)

M = maximum bending moment on wheel shaft (Nm)

T = twisting moment (Nm)

τ_{all} = 42 MPa, allowable shear stress of the shaft material

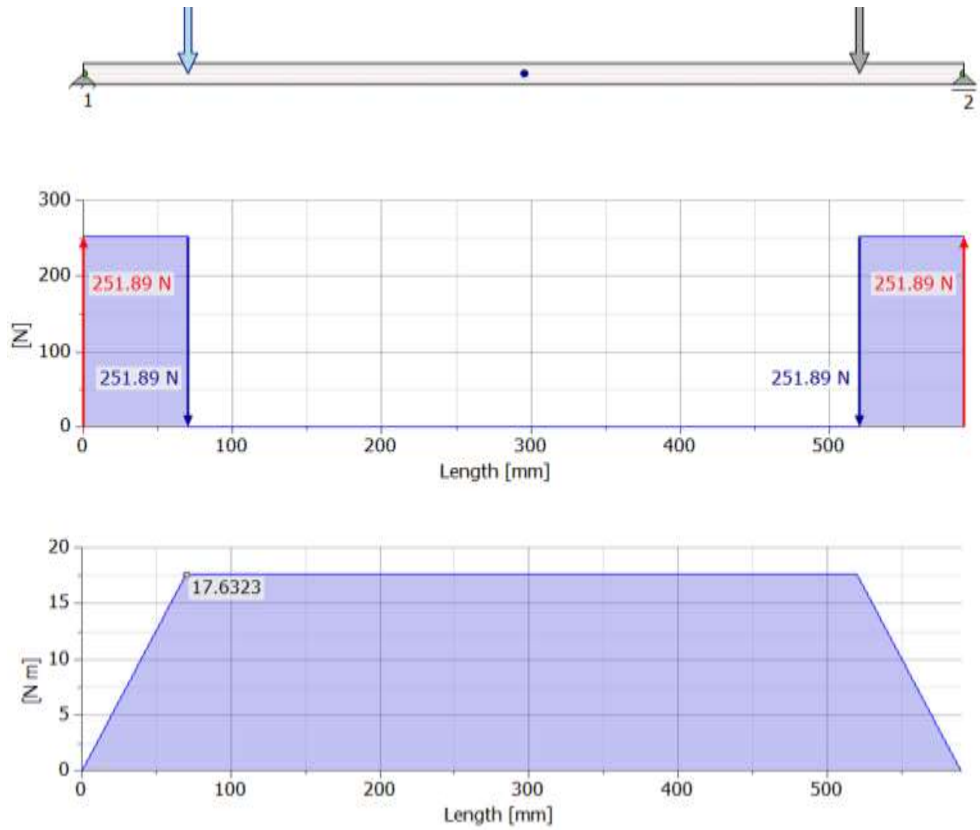


Figure 21. Shear force (b) and bending moment (c) diagrams of axel shaft

3.14. Performance Evaluation of the Digger

The performance of the digger was evaluated in terms of digging capacity, digging efficiency and fuel consumption.

3.14.1. Digging capacity

The digging capacity is the number of holes dug per time by the digger and it was determined using Equation (49).

$$C_d = \frac{H_n}{t} \qquad (49)$$

Where:

C_d = digging capacity (holes/hr)

H_n = number of holes

t = time it takes (hr)

3.14.2. Digging efficiency

The digging efficiency was calculated by using equation (50) given by Yehia et al. (2009).

$$\eta_d = \frac{V_{out}}{V_{tot}} \times 100 \tag{50}$$

Where:

η_d= Efficiency of digging (%)

V_{out}= Volume of the soil excavated from the hole (m^3)

V_{tot} = Volume of total soil inside and outside the hole (m^3)

3.14.3. Fuel consumption

The fuel consumption of the digger was measured using the refill method. Prior to and after the test run, the fuel tank was filled to its full capacity. A graduated cylinder was used to measure the fuel amount added to the tank. Fuel consumption was calculated based on the fuel amount difference between before and after the test run. The fuel needed for digging per time was estimated using Equation (51).

$$F_C = \frac{V_f}{T_t} \tag{51}$$

Where:

F_C = rate of fuel consumption (L/hr)

V_f= volume of consumed fuel (L)

T_t = time taken (hr)

3.15. Experimental Design

To know the effect of soil moisture content and working speed on the digging performance of the digger, an experiment with full factorial design at three replications was conducted. The factors of the experiment were soil moisture content and soil cutting tool working speed at three levels (15.00, 22.00 and 29.00 %) and (100.00, 140.00 and 180.00 rpm) respectively. To perform the performance evaluation a total of 27 test runs was conducted on sandy and clay soils separately.

3.16. Statistical Analysis

After conducting experiment, the obtained data was subjected to Minitab statistical software and Statistix 8.0 software for analysis of variance (ANOVA) to determine the significance of the factors and interaction of them on the digger’s performance. Level of significance of the main factors and their interaction was determined using p-test based on analysis of variance and conclusion was drawn.

4. RESULTS AND DISCUSSIONS

4.1. Physio Mechanical Properties of the Soil

The physio-mechanical properties of clay soil such as shear strength, wet bulk density, moisture content and repose angle were investigated on the study area and the results are summarised in table 3

Table 3. Physio-mechanical properties of the soil

Test site number	Shear strength on horizontal plane (C_{uh}) in kPa	Shear strength on vertical plane (C_{uv}) kPa	Bulk density (kg/m^3)	Repose angle (°)	Moisture content (%)
1	3.70	39.00	1962.95	30	22.32
2	3.0	24.50	1996.10	34	29.11
3	3.50	35.50	1964.90	31	24.27
4	3.20	28.50	1995.60	33	26.49
5	3.40	35.00	1966.30	32	25.30

4.2. Prototype Motorized Hole Digger

The designed, fabricated and assembled prototype of the intended motorized hole digger is presented on Figure 22. Its component parts are soil cutting components which consists of soil cutting tool and rotor shaft; soil elevator components such as buckets, bucket elevator belts, soil elevator pulleys, soil elevator shafts and soil discharging chute; power source and transmission systems such as petrol engine, bevel gears, pinion shaft, gear shaft, v-belts and pulleys, chain and sprocket, elevator drive shaft, chain drive shaft; body and transporting components like main frame, digger frame, handle, digger up and down sliding mechanism and handle, ground wheels and bearings.

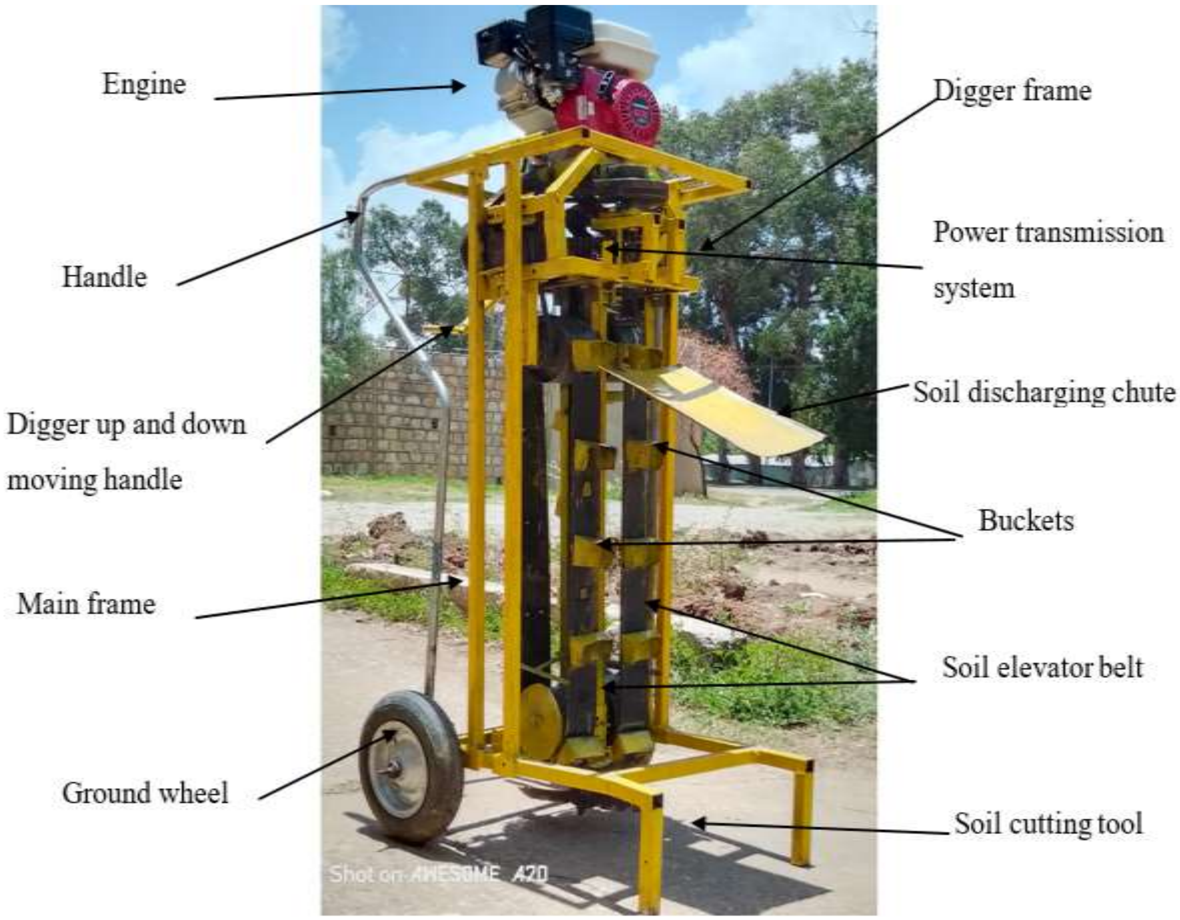


Figure 22. Prototype motorized hole digger

4.3. Effect of Working Speed and Soil Moisture on Digger Performance on Sandy Soil

4.3.1. Digger capacity

The mean main effect and interaction effects of working speed and soil moisture content on the digging capacity of the prototype of the digger on sandy soil are presented in Table 4. As illustrated on Table 5, analysis of variance revealed that working speed and soil moisture content had a significant effect on the digger capacity on sandy soil ($p < 0.05$). The interaction between working speeds and soil moisture content also had a significant impacted on the digger capacity ($p < 0.05$). Figure 23 indicates the patterns of the main effects of the factors and the interaction between them on digger capacity. At soil moisture content 15.00 %, 22.00% and 29.00%, the mean digging capacity of 26.44, 31.78 and 34.56 hole/hr was observed respectively. On the other hand, at working speed of 100.00, 140.00 and 180.00 rpm, mean digging capacity of 19.33, 32.56 and 40.89 hole/hr was recorded respectively. The increment of mean digging capacity from 15.00% to 29.00% soil moisture content was lesser than the mean digging capacity increment observed in between 100.00rpm and 180.00rpm working speed. This situation indicates that the digging capacity of the digger more depended on working speed rather than soil moisture content in the range of 15.00 to 29.00% and 100.00 to 180.00rpm working speed.

The maximum digging capacity of 45.00 hole/hr was observed at working speed of 180.00 rpm and soil moisture content of 29.00% whereas the minimum digging capacity of 16.67 hole/hr was recorded at working speed of 100.00 rpm and soil moisture content of 15.00 % for holes dug at 40.00 cm depth and cm 35.00 cm diameter on sandy soil as presented on Table 4.

Table 4. Average effect of working speed and moisture content on digger capacity on sandy soil (hole/hr)

Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	16.67^{I}	19.33^{H}	22.00^{G}	**19.33**
140.00	27.33^{F}	33.67^{E}	36.67^{C}	**32.56**
180.00	35.33^{D}	42.33^{B}	45.00^{A}	**40.89**
Mean	**26.44**	**31.78**	**34.56**	
LSD (%)	0.57			
CV (%)	2.24			

Mean values, the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation

Table 5. Analysis of variance for digger capacity on sandy soil

Source	DF	SS	MS	F-Value	P-Value
Working speed	2	2126.74	1063.37	2208.54	0.000
Soil moisture	2	305.85	152.93	317.62	0.000
Working speed*Soil moisture	4	22.59	5.65	11.73	0.000
Error	18	8.67	0.48		
Total	26	2463.85			

(a)

(a)

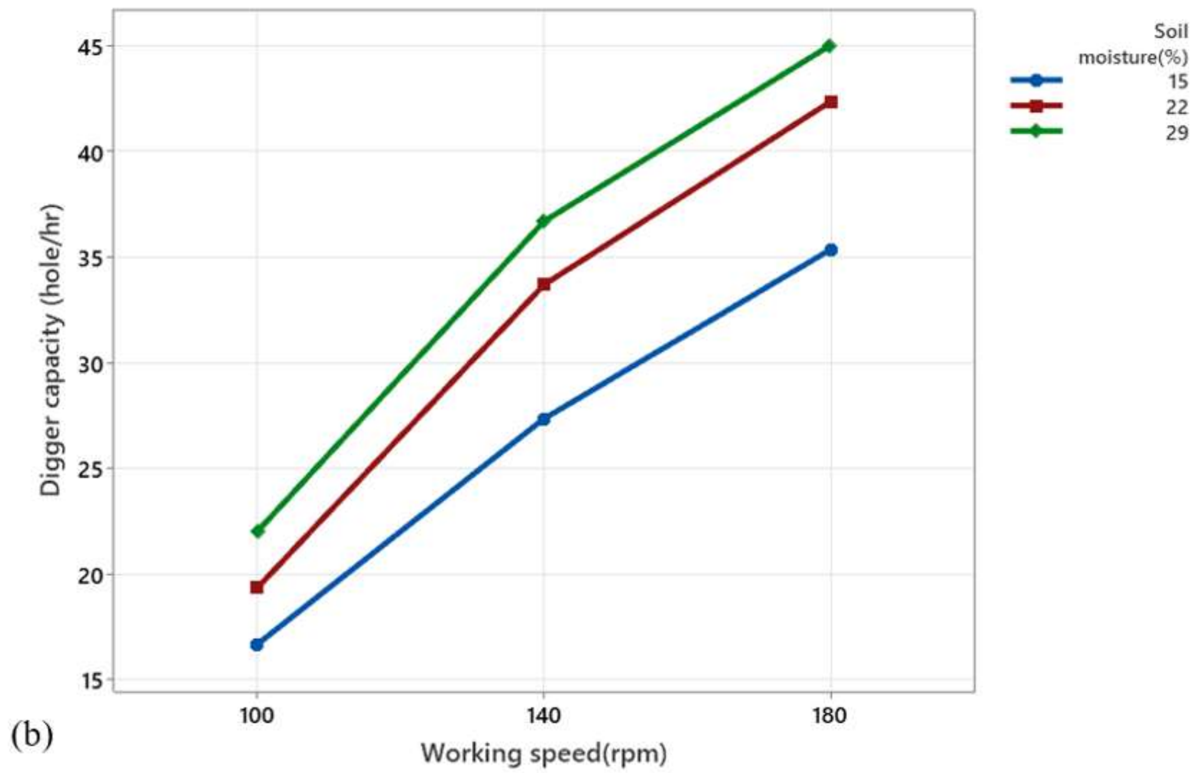


Figure 23. Main (a) and interaction (b) effect plot for digger capacity on sandy soil

4.3.2. Digger efficiency

Table 6 shows the impact of soil moisture content and working speed on digging efficiency of digger prototype on sandy soil. According to the result of analysis of variance (Table 7), digger efficiency was significantly impacted by soil moisture content and working speeds ($p < 0.05$). A p-value of greater than 0.05 indicates that there was no significant impact on the digger's efficiency from the interaction between working speed and oil moisture content on sandy soil. When the soil moisture content was 29% and the working speed 180 rpm, the highest digging efficiency of 85% was recorded. At soil moisture content 15% and working speed of 100 rpm, the minimum digging efficiency of 66.67% was observed as shown on Table 6. The patterns of effects of the factors (Figure 24) illustrates that, as the working speed increased from 100.00 to 180.00 rpm, the mean efficiency of the digger on each working speed rose rapidly from 68.67 to 83.67hole/hr and resulted in highest digging efficiency at 180.00rpm. On the other side as soil moisture content increased from 15.00% to 29.00%, the mean efficiency of the digger on each moisture content rose progressively from 75.00 to 78.55 %. This trend indicates that working speed of the digger have more impact on digging efficiency of the digger than the soil moisture content. Furthermore, since the soil elevator system depends on centrifugal force to throw the soil to outside of the hole, the higher efficiency at higher working speed is the expected result.

Table 6. Average effect of working speed and soil moisture on digging efficiency on sandy soil (%)

Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	66.67^{F}	69.00^{EF}	70.33^{DE}	**68.67**
140.00	76.67^{DE}	79.67^{D}	80.33^{C}	**78.89**
180.00	81.67^{B}	84.33^{B}	85.00^{A}	**83.67**
Mean	**75.00**	**77.67**	**78.55**	
LSD (%)	0.82			
CV (%)	1.36			

Mean values, the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation

Table 7. Analysis of variance for digger efficiency on sandy soil

Source	**DF**	**SS**	**MS**	**F-Value**	**P-Value**
Working speed	2	1043.85	521.926	327.72	0.000
Soil moisture	2	65.85	32.926	20.67	0.000
Soil moisture *Working speed	4	0.59	0.148	0.09	0.983
Error	18	28.67	1.593		
Total	26	1138.96			

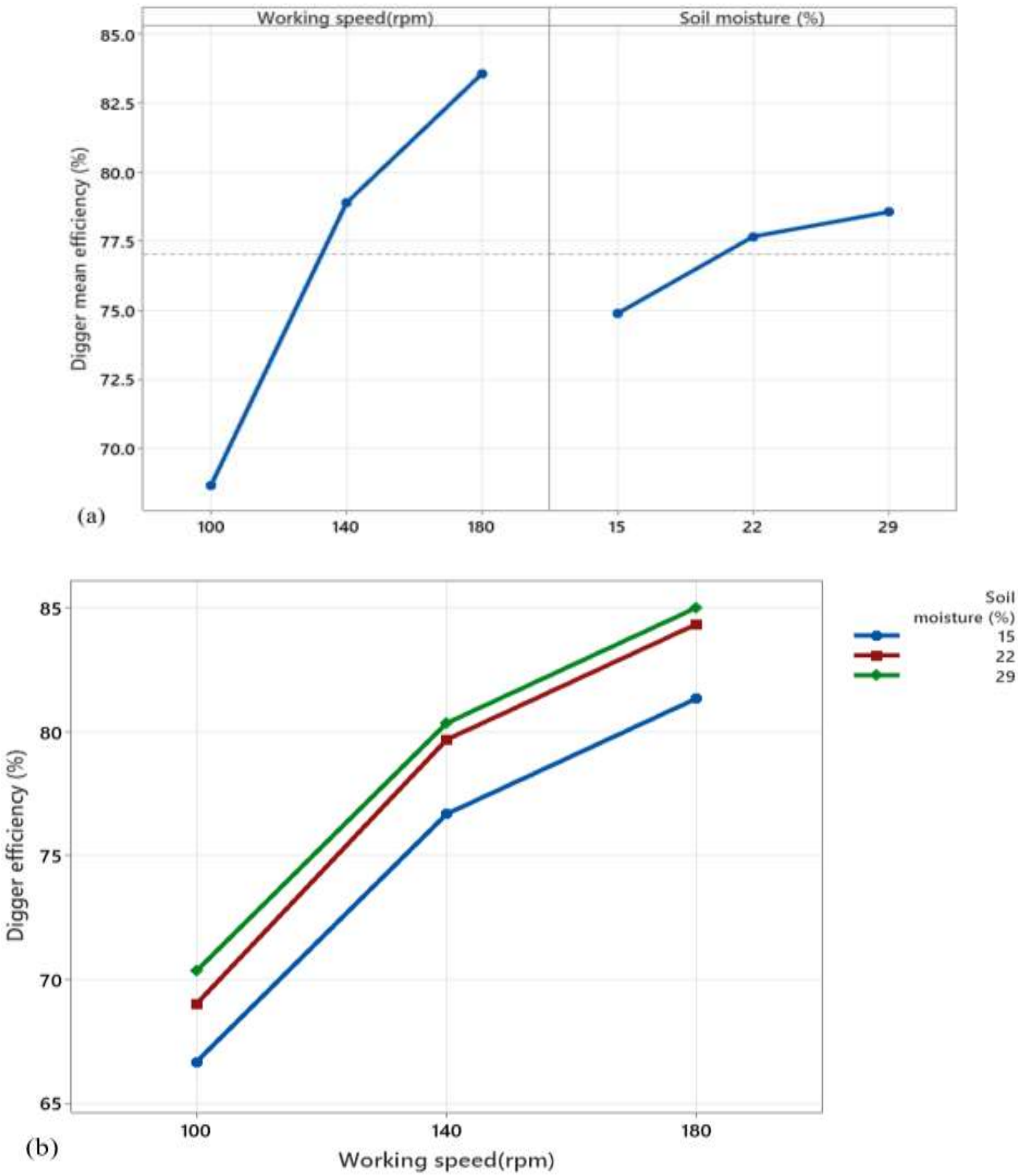


Figure 24. Main (a) and interaction (b) effect plot for digger efficiency on sandy soil

4.3.3. Fuel consumption

The effects of working speed and soil moisture content on the fuel consumption of prototype digger on sandy soil is presented in Table 8. Analysis of variance (Table 9) reveled that the fuel consumption of the digger on sandy soil was significantly impacted by soil moisture content and working speed ($p<0.05$). The interaction effect of working speed and soil moisture content did not significantly affected the digger's fuel consumption ($p>0.05$).

As presented on Figure 25, the pattern of effect of the factors showed that, as soil moisture content decreased, the fuel consumption of the digger increased gradually. On the other hand while the working speed increased from 100.00 to 140.00 and 180.00 rpm, there was a fast rise of fuel consumption for each soil moisture content. This indicated that, working speed and fuel consumption of the digger have direct relationship while soil moisture content and fuel consumption of the digger have indirect relationship.

The maximum fuel consumption of 0.67L/hr was recorded at working speed of 180.00 rpm and soil moisture content of 15.00% whereas the minimum fuel consumption of 0.32L/hr was observed at working speed of 100.00 rpm and soil moisture content of 29.00 % on sandy soil as presented on Table 8.

Table 8. Average effect of working speed and soil moisture on fuel consumption of digger on sandy soil (L/hr)

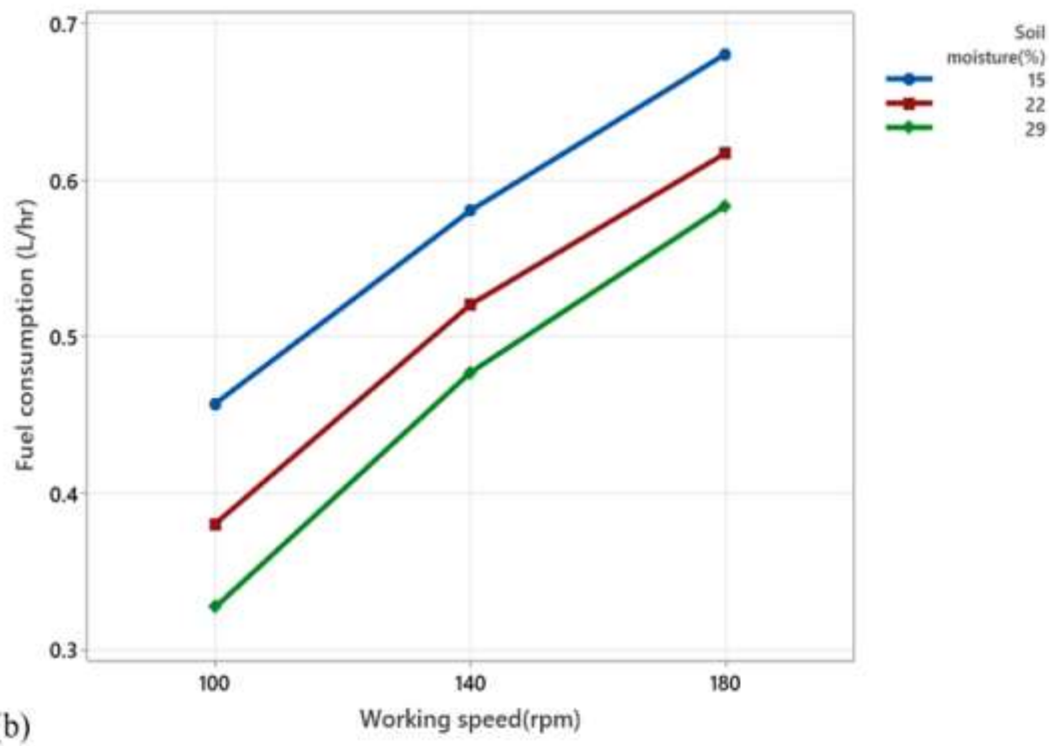
Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	0.46^{C}	0.38^{D}	0.32^{E}	**0.39**
140.00	0.59^{B}	0.52^{C}	0.47^{C}	**0.53**
180.00	0.67^{A}	0.62^{B}	0.58^{B}	**0.62**
Mean	**0.57**	**0.51**	**0.46**	
LSD (%)	0.02			
CV (%)	2.17			

Mean value, the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation,

Table 9. Analysis of variance for fuel consumption of digger on sandy soil

Source	DF	SS	MS	F-Value	P-Value
Working speed	2	0.258822	0.129411	1204.86	0.000
Soil moisture	2	0.055267	0.027633	257.28	0.000
Working speed*Soil moisture	4	0.000978	0.000244	2.28	0.101
Error	18	0.001933	0.000107		
Total	26	0.317000			

(a)



(b)

Figure 25. Main (a) and interaction (b) effect plot for fuel consumption on sandy soil

4.4. Effect of Working Speed and Soil Moisture on Digger Performance on Clay Soil

4.4.1. Digger capacity

Table 10 shows how the digging capacity of the digger prototype was affected by working speed and soil moisture content of clay soil. As indicated on Table 11, analysis of variance reveled that, digger capacity was significantly affected by both main effect and interaction effect of soil moisture content and working speed ($p < 0.05$) on clay soil. As can be understand from the pattern of the effects on Figure 26, there was a progressive increase in the digger's digging capacity at each working speed as soil moisture content rose. Nevertheless, the mean digging capacity rapidly rose as the working speed increased from 100.00 to 140.00 and 180.00 rpm. The incremental value of main effect of soil moisture from 15.00% to 29.00% was less than the incremental value of main effects of working speed from 100.00 to 180.00 rpm on digger's digging capacity. This indicates that working speed is the influential factor on digger capacity. As shown on Table 10, the lowest digging capacity of 14.67 holes/hr was noted at working speed of 100.00 rpm and soil moisture content of 15.00%, while the greatest digging capacity of 43.33 holes per hour was noted at working speed of 180.00 rpm and soil moisture content of 29.00% for holes dug at 40.00 cm depth and 35.00 cm diameter on clay soil.

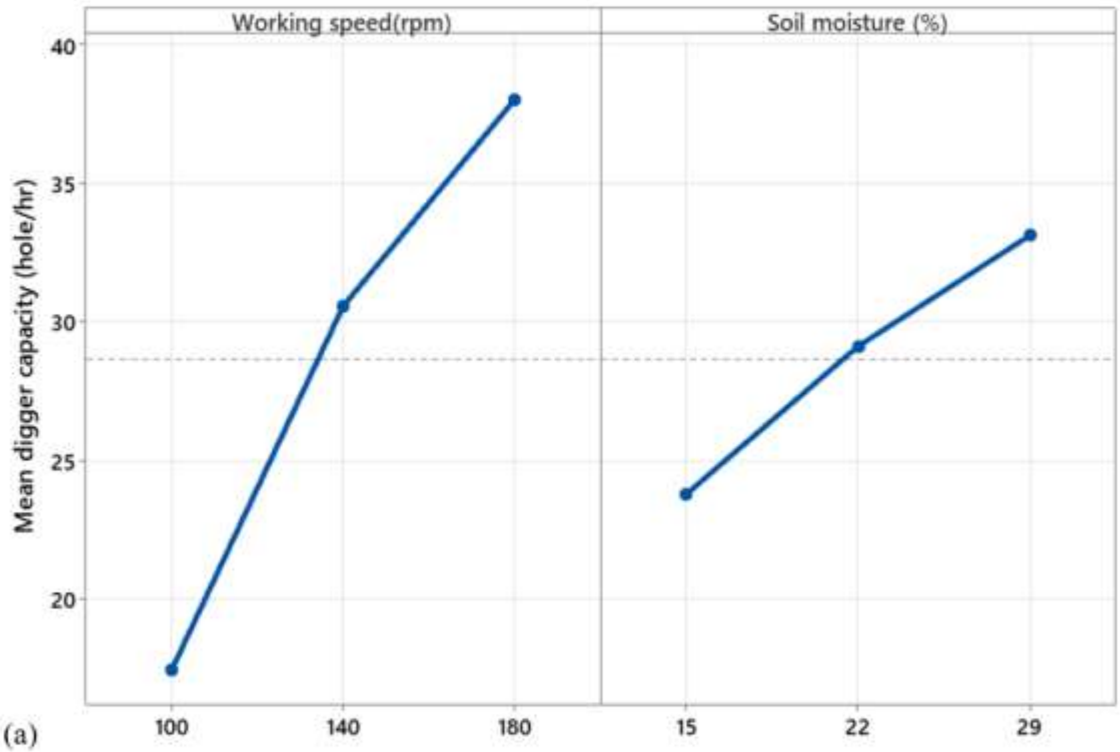
Table 10. Average effect of working speed and soil moisture content on digger capacity on clay soil (hole/hr)

Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	14.67^{H}	16.67^{G}	21.00^{F}	**17.45**
140.00	24.67^{E}	32.00^{D}	35.00^{C}	**30.56**
180.00	32.00^{D}	38.67^{B}	43.33^{A}	**38.00**
Mean	**23.78**	**29.11**	**33.33**	
LSD (%)	0.52			
CV (%)	2.23			

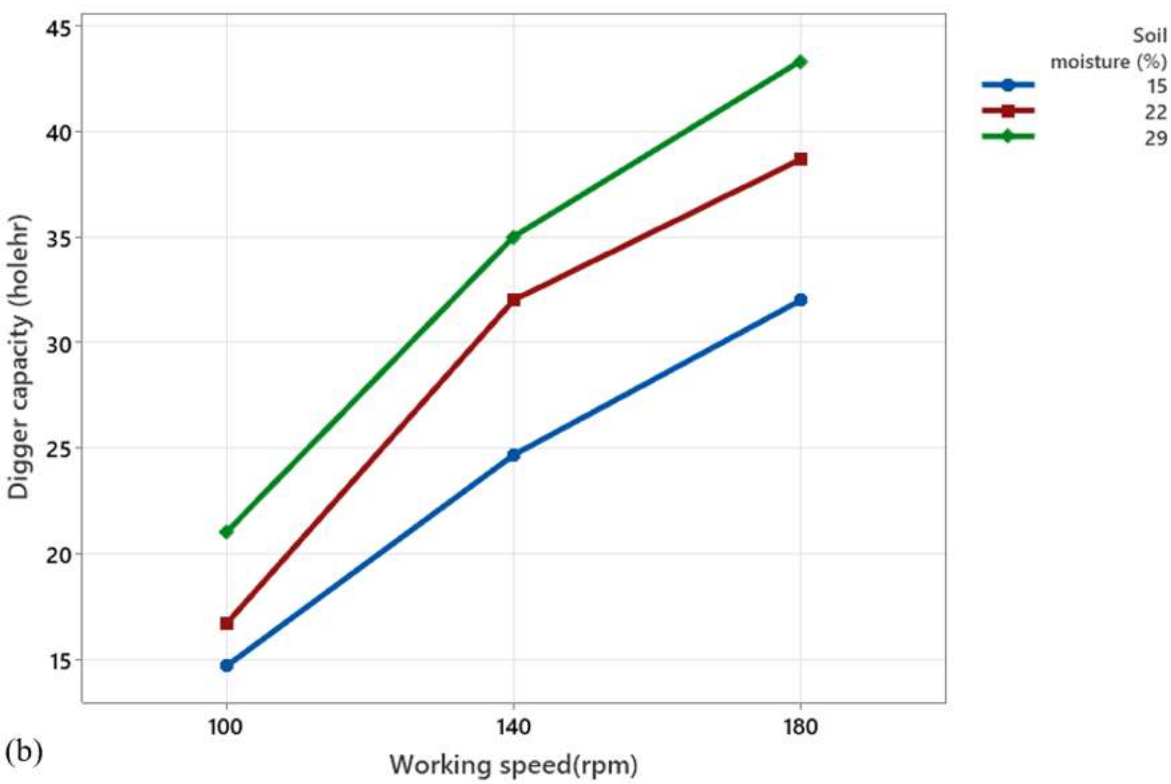
Mean values, the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation

Table 11. Analysis of variance for digger capacity on clay soil

Source	**DF**	**SS**	**MS**	**F-Value**	**P-Value**
Working speed	2	1949.56	974.778	2392.64	0.000
Soil moisture	2	394.67	197.333	484.36	0.000
Working speed*Soil moisture	4	32.44	8.111	19.91	0.000
Error	18	7.33	0.407		
Total	26	2384.00			



(a)



(b)

Figure 26. Main (a) and interaction (b) effect plot for digger capacity on clay soil

4.4.2. Digger efficiency

Table 12 shows the effects of working speed and soil moisture content on digging efficiency of the digger prototype on clay soil. According to an analysis of variance result (Table13), the effect of working speeds and soil moisture content significantly impacted digger efficiency on clay soil ($p < 0.05$). The interaction impact of working speed and soil moisture content had no significant effect on digger efficiency ($p > 0.05$). As illustrated on Figure 27, the higher digger efficiency was more from the increment of the working speed than the increment of soil moisture content. When working speed increased from 100.00 to 180.00 rpm, mean digging efficiency of the digger was seen rising rapidly from 67.67 to 82.67%. But as soil moisture rose from 15.00% to 29.00%, the mean digging efficiency of the digger rose slowly from 74.00 to 77.55%. As demonstrated on Table 12, the highest digging efficiency of 84.00% was seen at working speed of 180.00 rpm and a soil moisture content of 29.00%, while the lowest digging efficiency of 67.00% was found at a working speed of 100.00 rpm and a soil moisture level of 15.00%

Table 12. Average effect of working speed and soil moisture on digging efficiency on clay soil (%)

Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	65.67^{F}	68.00^{E}	69.33^{E}	**67.67**
140.00	75.67^{E}	78.67^{E}	79.33^{D}	**77.89**
180.00	80.67^{C}	83.33^{B}	84.00^{A}	**82.67**
Mean	**74.00**	**76.67**	**77.55**	
LSD (%)	0.77			
CV (%)	1.28			

Mean values, the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation

Table 13. Analysis of variance for digger efficiency on clay soil

Source	DF	SS	MS	F-Value	P-Value
Working speed	2	1043.85	521.926	327.72	0.000
Soil moisture	2	65.85	32.926	20.67	0.000
Soil moisture *Working speed	4	0.59	0.148	0.09	0.983
Error	18	28.67	1.593		
Total	26	1138.96			

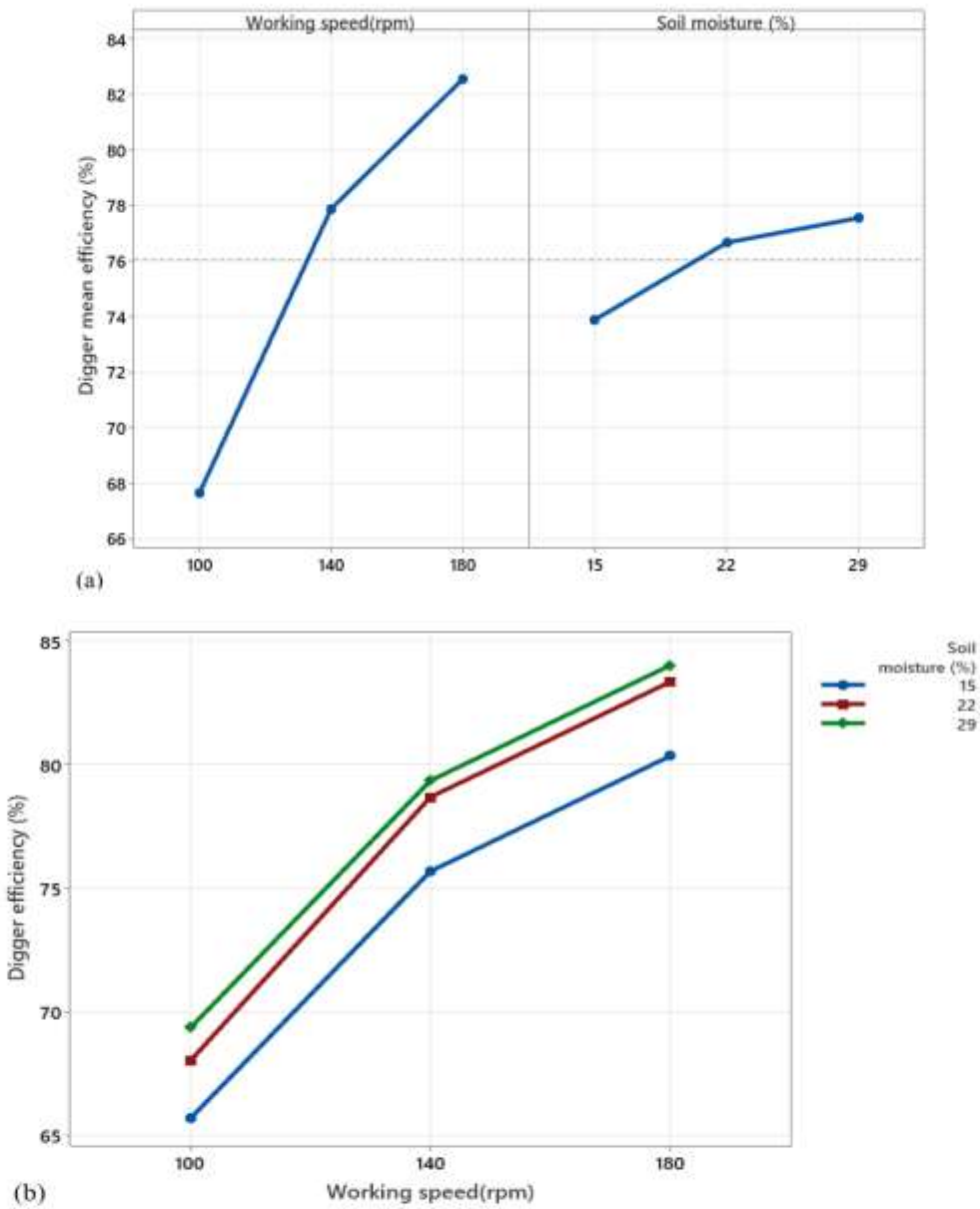


Figure 27. Main (a) and interaction (b) effect plot for digger efficiency on clay soil

4.4.3. Fuel consumption

Table 14 shows the average effect of working speed and soil moisture content on the digger's fuel consumption on clay soil. The analysis of variance (Table 15) indicated that the working speed and soil moisture content significantly affected the fuel consumption on clay soil ($p < 0.05$). The interaction effect of working speed and soil moisture content had also significant effected on fuel consumption ($p<0.05$). As noted on Figure 28, the fuel consumption of the digger increased as soil moisture content decreased. On the other hand while the working speed increased from 100.00 to 140.00 and 180.00 rpm, there was a fast rise of fuel consumption for each soil moisture content. This indicated that, the fuel consumption of the digger more depended on working speed rather than soil moisture content in the range of 15.00 to 29.00%. As illustrated on Table 14, maximum fuel consumption of 0.68L/hr was recorded at working speed of 180.00 rpm and soil moisture content 15.00%, whereas minimum fuel consumption of 0.33L/hr was observed at working speed of 100.00 rpm and soil moisture content of 29.00 %.

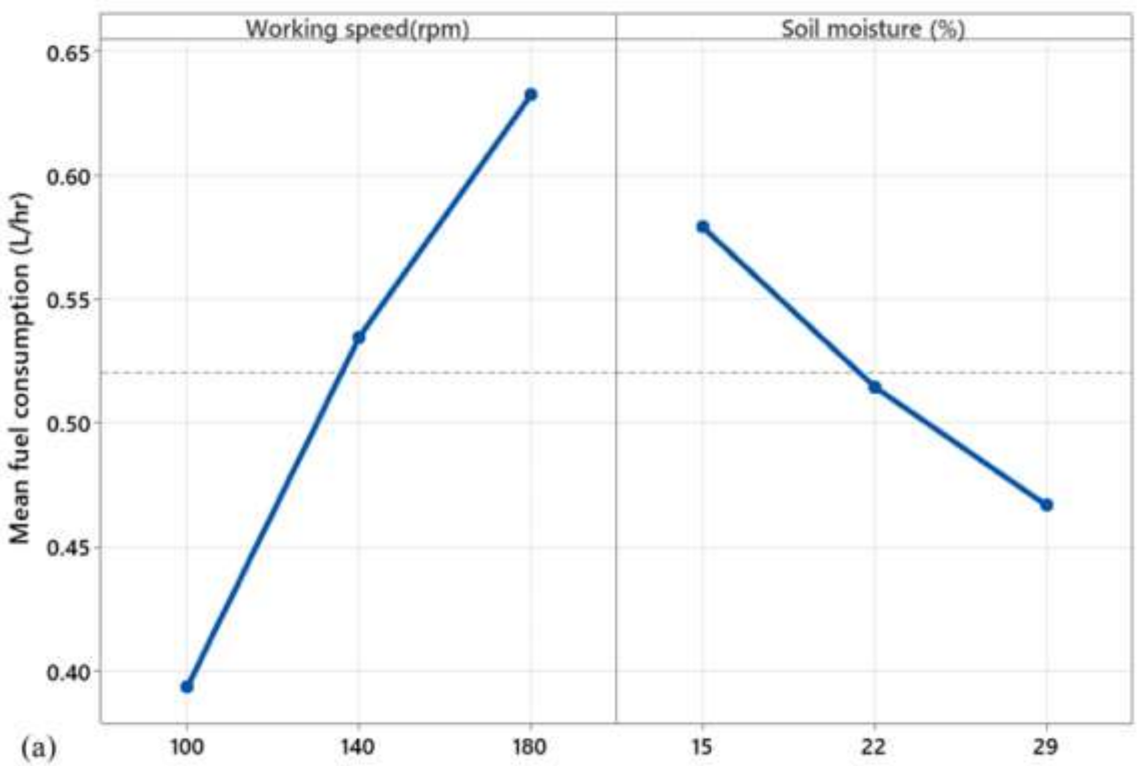
Table 14. Average effect of working speed and soil moisture on fuel consumption of the digger on clay soil (L/hr)

Working speed(rpm)	**Soil moisture content (%)**			
	15.00	22.00	29.00	**Mean**
100.00	0.46^{C}	0.39^{D}	0.33^{E}	**0.39**
140.00	0.59^{B}	0.53^{C}	0.48^{C}	**0.52**
180.00	0.68^{A}	0.62^{B}	0.59^{B}	**0.63**
Mean	**0.58**	**0.51**	**0.47**	
LSD (%)	0.02			
CV (%)	5.92			

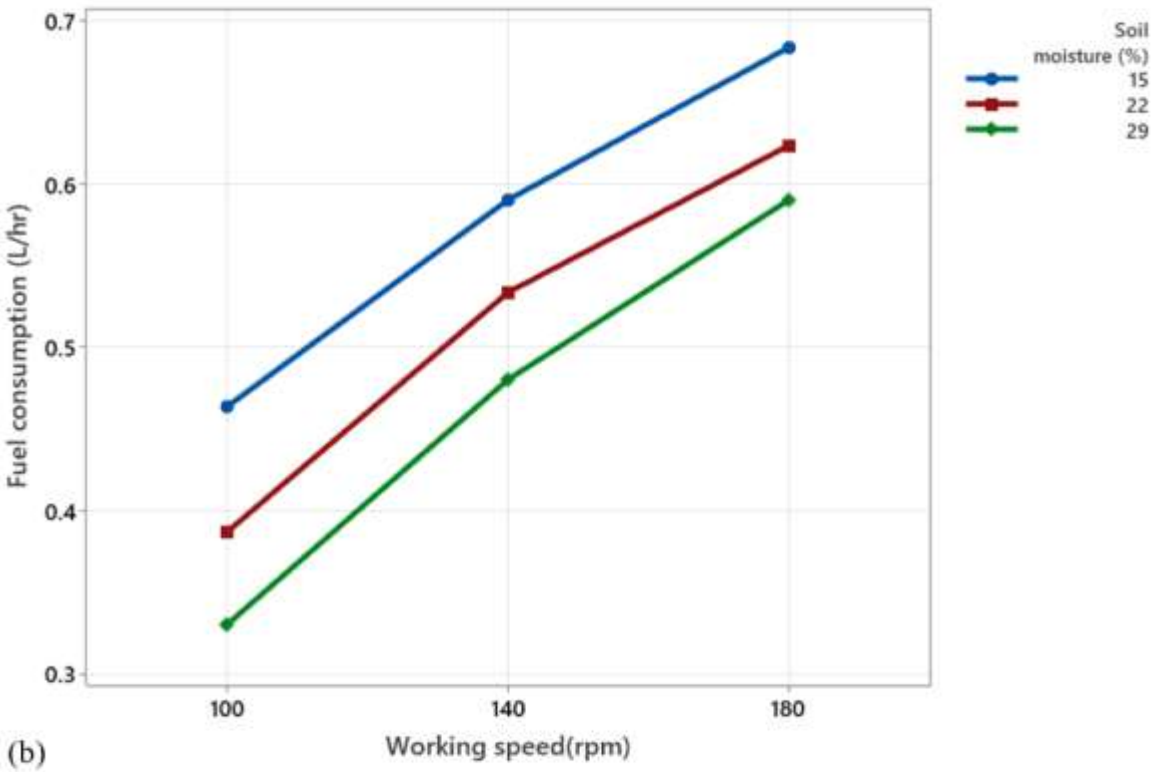
Mean values, *the means assigned with same letters are not significantly different (P>0.05) from one another, LSD = least significance difference, CV = coefficient of variation*

Table 15. Analysis of variance of fuel consumption of the digger on clay soil

Source	**DF**	**SS**	**MS**	**F-Value**	**P-Value**
Working speed(rpm)	2	0.259622	0.129811	1348.04	0.000
Soil moisture (%)	2	0.057089	0.028544	296.42	0.000
Soil moisture (%)*Working speed(rpm)	4	0.001356	0.000339	3.52	0.027
Error	18	0.001733	0.000096		
Total	26	0.319800			



(a)



(b)

Figure 28. Main (a) and interaction (b) effect plot for fuel consumption on clay soil

5. SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

5.1. Summary and Conclusions

The study was all about design, fabrication and performance evaluation of a prototype of motorized hole digger which its purpose is to dig a hole for transplantation of perennial crop. In designing the machine the soil physio mechanical properties that can affect the power requirement and functionality of the digger such as soil shear strength, soil bulk density, soil moisture content and angle of repose of clay soil were investigated. From the investigation result, the maximum values of soil shear strength on horizontal and vertical plane (3.70 and 39.00 kPa) at $1962.95 kg/m^2$ wet soil bulk density, 22.32% soil moisture content and 34 degree soil repose angle were used in order to design the digger. Based on these physio mechanical properties of the soil, the power requirement of the digger was estimate to be 4.00 hp. Then, the detailed design and fabrication of the components of the digger such as soil cutting system of the digger (soil cutting tool and rotor shaft), Soil excavating system (bucket elevators), power transmission systems, frames and transporting parts were done. Finally, the fabricated prototype was tested and evaluated for its performance. The performance test was conducted on sandy soil and clay soil separately to assess the effect of soil moisture content and working speed on the digger capacity, efficiency and fuel consumption. The factors of the experiment were soil moisture content at levels of ($15.00, 22.00\ and\ 29.00$ %) and working speed at (100.00, 140.00 and 180.00 rpm). Analysis of variance revealed that working speed and soil moisture content had a significant effect on digger performance (capacity, efficiency and fuel consumption) on both soil types ($p < 0.05$). The interaction effect between working speed and oil moisture content had significant effect on digger capacity on both soil types and on fuel consumption on clay soil ($p < 0.05$).

The maximum digging capacity and efficiency of the digger were 45.00, 43.33 hole/hr and 85.00%, 84.67% which were observed at working speed of 180.00 rpm and soil moisture content of 29.00 % for hole dug on sandy and clay soil respectively at depth of 40.00cm and 35.00 cm diameter. The minimum digging capacity and efficiency of the digger were 16.67,14.67 hole/hr and 65.67%, 66.67%, recorded at working speed of 100.00 rpm and soil moisture content of 15.00 % for hole dug on sandy and clay soil respectively at 35.00 cm diameter 40.00cm depth.

The greatest fuel consumption of 0.68 L/hr was observed when the soil moisture content was 15.0% and working speed of180.00 rpm on clay soil. Whereas the lowest fuel consumption of 0.32L/hr was recorded at 29.00% soil moisture content and at working speed of 100.00 rpm on sandy soil.

Generally, the study successfully designed, fabricated and evaluated prototype of motorized hole digger. The major findings indicated that, increasing working speed from 100.00 to 180.00 rpm and soil moisture content from 15.00 to 29.00% significantly enhanced digging capacity and efficiency of the digger.

5.2. Recommendations

Based upon the findings from testing and performance evaluations of the digger, the following recommendations are proposed to be improved to enhance the performance the digger and its usability by the farmers.

- The design of the digger should be more compact to reduce power consumption and to address problem related to transportation during operation.
- To reduce weight and design complexity the power transmission system from power source to rotor shaft should be replaced by worm gear drive mechanism.
- The soil elevator part of the digger should be changed to chain type soil elevator to avoid slippage and increase soil elevating efficiency of the digger.
- The digger should be self-propelled to reduce human drudgery.

6. REFERENCE

Abebe, W., Amente, G., and Goro, G. 2015. Estimation of lobal solar radiation using solar pv and its comparison with sunshine duration using quadratic and gaussian fits. *East African Journal of Sciences*, *9*(2), 97–104.

Ayele, S. 2021. The resurgence of agricultural mechanisation in Ethiopia: rhetoric or real commitment? *The Journal of Peasant Studies*, *49*(1), 137–157

Blomme, G., Jacobsen, K., Tawle, K., and Yemataw, Z. 2018. Agronomic practices with a special focus on transplanting methods for optimum growth and yield of enset [Ensete ventricosum (Welw.) Cheesman] in Ethiopia. *International Journal of Tropical and Subtropical Horticulture*, *73*(6), 349–355.

Borrell, J. S., Goodwin Mark, Blomme Guy, Jacobsen Kim, Wendawek Abebe M., Gashu Dawd, Lulekal Ermias, Asfaw Zemede, Demissew Sebsebe, and Wilkin Paul. 2019. Enset-based agricultural systems in Ethiopia: A systematic review of production trends, agronomy, processing and the wider food security applications of a neglected banana relative. *Plants People Planet*, *2*(3), 212–228.

Chaaban, M. A., El Awady, M. N., Yehia, I., and Khalil, S. K. 2007. Factors affecting the design of a hole digger attached to a power tiller. *Journal of Environmental Science*, *2*.

CSA (Central Statistical Agency). 2021. *Report on Farm Management Practice (Private Peasant Holdings, Meher Season)*.

Edward, W., Jacques Op de Leak, Tony Marsh, Herbert Lempke, and Keith Chapman. 2005. *Arabica Coffee Manual for Lao-PDR*. FAO Regional Office for Asia and the Pacific.

El-Halim, S., Morghany, H., and Aboukarima, A. 2009. Evaluating performance of a post hole digger. In *J. Agric. Sci. Mansoura Univ* (Vol. 34, Issue 5).

Etissa, E., Dagnew, A., Ayele, L., Assefa, W., Firde, K., Kiflu, E. and Getachew, B. 2016. Fruit crops research in Ethiopia: achievements, current status and future prospects. *Agricultural Research for Ethiopian Renaissance*.

De Bac, G., 2010. Technical guidelines on tropical fruit tree management in Ethiopia.

Gautam A, Sinha Yomesh, Patel Basant, and Kumar Kawer Jayant. 2019. Development and Modification of Conventional Seed-Cum-Fertilizer drill into broad bed furrow Machine. *International Journal of Chemical Studies*, *7*(5), 1734–1738.

Gebissa, Y. W. 2021. The challenges and prospects of Ethiopian agriculture. *Cogent Food and Agriculture*, *7*(1), 1–2.

Guan, T. 2016. Transplanting method for passion fruit seedlings.

Habtamu, G. M. 2022. Coffee (Coffea arabica L.) field establishment and management practices in Ethiopia. *American Journal of Engineering and Technology Management*, *7*(3), 48–58.

Hailu, D. 2024. Development of the Agricultural Sector and Its Contribution to the Growth of the Ethiopian Economy. *International Journal of Agricultural Economics*

Khurmi, and Gupta, J. K. 2005. Textbook of Machine Design. In Khurmi & J.K. Gupta (Eds.), *Engg. Services* (14th ed.). Eurasia Publishing House (PVT.) LTD.

Mallapour, s, D.Kalantari, and M.Rajabi, V. 2018. Design, development and evaluation of a new motorized hydraulic hole digger for spot treatment. *Journal of Agricultural .Machinery*, *8*(2), 235–248

Megersa, H. G. 2022. Coffee (Coffea arabica L.) Field Establishment and Management Practices in Ethiopia. *American Journal of Engineering and Technology Management*, *7*(3), 48–58.

Mg, T. Z. O., Ma Myat Win Khaing, and Ma Yi Yi Khin. 2019. Analysis of belt bucket elevator. *International Journal of Trend in Scientific Research and Development*, *3*(5), 760–763.

Mir, B. A. 2021. Vane Shear Test for Cohesive Soils. In *Manual of Geotechnical Laboratory Soil Testing* (pp. 267–279). CRC Press.

Mohammed, M. K., and Beyene, A. K. 2024. Situation Analysis of Agricultural Development in Ethiopia. *Science*, *5*(1), 19-30.

Netsere, A., Kufa, T., & Shimber, T. 2015. Review of Arabica coffee management research in Ethiopia. *Journal of Biology, Agriculture and Healthcare*, *5*(13), 235-258

Raymond, A. K. 1985. *Materials Handling Handbook, Second Edition* (Raymolad A. & I. Kulwiec., Eds.; 2nd ed.). Wlley-Inter-science Publication.

Siddhartha, R. 2008. Introduction to Material Handling. New Age International (P) Ltd, publishers.

Smith, D., Sims, B., and O.Neill, D. 1994. *Testing and Evaluation of Agricultural Machinery and Equipment: Principles and Practices*. FAO.

Stellmacher, T., and Kelboro, G. 2019. Family farms, agricultural productivity, and the terrain of food insecurity in Ethiopia. *Sustainability*, *11*(18), 1–10.

Turyasingura, B., chavula, P., Mohammed, Y., Abdi Ali, E., Girma, T., Sadeso, K., Shentema, S., Abebe, A., Katel, S., and Timsina, S. 2023.Laboratory analysis of soil physicochemical properties based on agricultural fertilizer input requirement application: a case of Haramaya University, Ethiopia. *Plant Physiology and Soil Chemistry*, *3*(1), 09–16.

Usha, K. , Madhubala, T., Amit, K., Goswami, K., and Deepak, N. 2015. *Fundamental of Fruit Production*. Division of fruits and horticultural technology, Indian agricultural institute.

Varghese, A. K., Joseph, J. G., Abraham, J. C., Regi, J., and Jayan, J. T. 2020a. *Agro auger a portable soil digging machine* (Vol. 7).

Varghese, A. K., Joseph, J. G., Abraham, J. C., Regi, J., and Jayan, J. T. 2020b. Agro-auger: a portable soil digging machine. *Journal of Emerging Technologies and Innovative Research, 7*(4), 1105–1112.

Welteji, D. 2018. A critical review of rural development policy of Ethiopia: Access, utilization and coverage. *Agriculture and Food Security*, *7*(55), 1–6.

Yehia, I., Kabeel, M. H., and Khalil, S. K. 2009. Affecting factors on performance of a developed hole digger. *Journal of Soil Sciences and Agricultural Engineering*, *34*(5), 5575–5591.

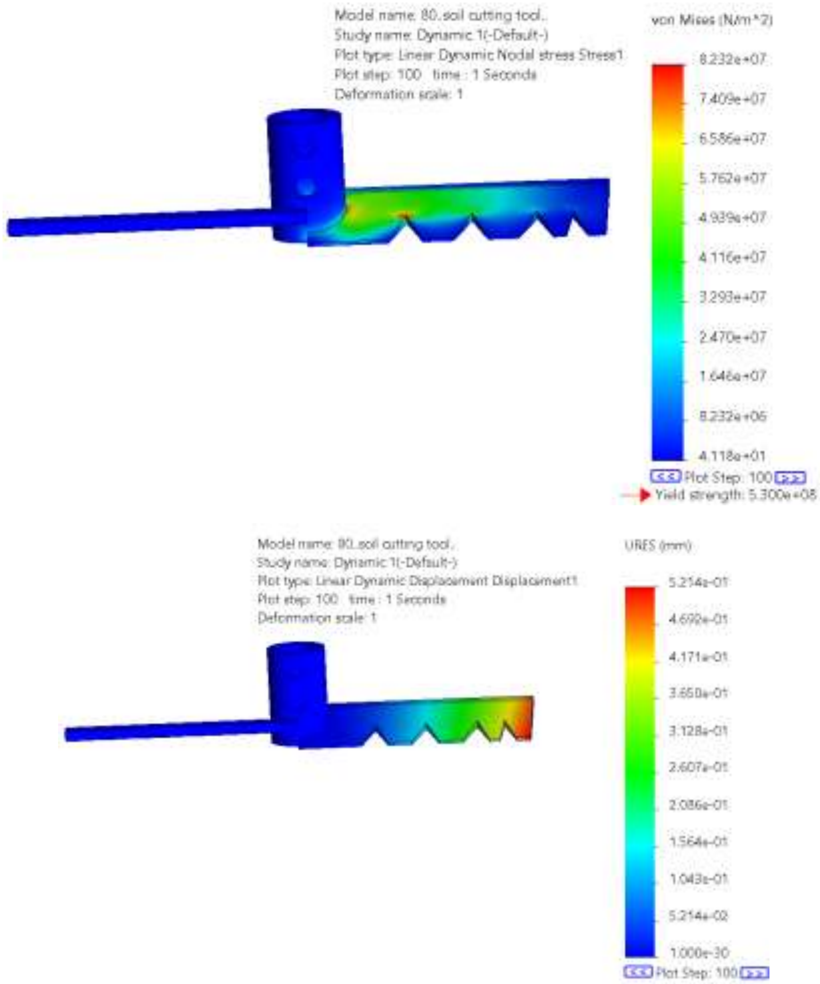
Zerssa, G. W., Feyssa, D. H., Kim, D.-G., & Eichler-Löbermann, B. (2021). Challenges of Smallholder Farming in Ethiopia and Opportunities by Adopting Climate-Smart Agriculture. *Agriculture, 11*(3), 192

Yokoi, H. 1968. Relationship between soil cohesion and shear strength. Soil Science and Plant Nutrition, 14(3), 89–93.

Zong, W. Y., Wang, J. L., Huang, X. M., Yu, D., Zhao, Y. B., and Graham, S. 2016. Development of a mobile powered hole digger for orchard tree cultivation using a slider-crank feed mechanism. *International Journal of Agricultural and Biological Engineering*, *9*(3), 48–56.

7. APPENDIX

7.1 Appendix Figures



Appendix Figure1. Stress and deformation analysis of soil cutting tool

Appendix Figure2. Prototype of the motorized hole digger during testing