PROJECT TITLE: Design of a Modified Hand Operated Maize Sheller

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Declaration

I declare that this project is my work and has not been submitted for award of a degree in any University.

Signature: ........................................ Date: ........................................

Nyongesa Fredrick Wanjala

This report has been submitted for examination with my approval as a University Engineering Design Project supervisor.

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Eng. Joackim M. Mutua
Dedication

I dedicate this Engineering Design project to my beloved Parents, my siblings and friends for their kindness and support throughout my undergraduate study.
Acknowledgement

I sincerely thank the almighty God for seeing me through the five years in campus and having given me good health, mental and physical strength throughout my stay as an undergraduate at the University of Nairobi.

Special thanks also go to Eng. Joackim M. Mutua for his guidance and great intellectual support. My gratitude also goes to the able EBE technical staff especially Mr. Wamutitu Wilfred Mushogo for the proficient guidance he continuously offered me throughout this project.

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List of Acronyms and Abbreviations

PTO  - Power Take off

ADC  - Agricultural Development Corporation

FAO  - Food and Agriculture Organization

KSC  - Kenya Seed Company

SSA  - Sub-Saharan Africa

BEME - Bill of Engineering Measurement and Evaluation

RPM  - revolutions per minute (rpm)
Abstract

Maize shelling or simply maize threshing is the most important aspect of post-harvest operation of maize. It involves detaching of the maize grain from its cobs. The threshing technique applied is also important in other grain post harvest and handling processes such as sorting for quality as well as storage.

The traditional method of grain threshing is by use of hand held simple tools or hitting with stick which is laborious, time consuming, causes loss of grains and has a very low output. For instance, the traditional hand maize shelling technique can only achieve up to 25kg/hour of shelled maize. The performance of an improved pedal maize threshing machine which consists of a single spiked disc in the threshing unit and whose results show that the machine can shell about 80 kg/hour of maize is not very efficient since the machine is turned manually and requires very high energy inputs in cycling.

Mechanized shelling techniques such as the motor operated maize sheller as well as the tractor PTO operated sheller are the most efficient but post a challenge for adoption by the rural farmer because of the high costs involved in either hiring their services or purchasing the machines.

The design modification presented in this paper is aimed at complementing the performance of the cob master threshing machine by incorporating mechanisms such as gear system, flywheel and a chain drive system to achieve desirable threshing Torque as well as minimize on the reciprocating force and energy requirements in operating the machine. The design also furnishes an alternative to the costly mechanized maize shelling machinery thereby providing an appropriate mechanism to meet the shelling demand of rural maize farmers who basically have a priority in subsistence farming. In addition the design can be implemented in the fabrication of shellers for such farmers who practice backyard chicken keeping and grow, dry and shell their own maize as a cost-effective alternative or supplement to commercial chicken feed or feed for other small livestock.

Keywords: Cost, Energy, mechanisms, machine, maize, threshing/shelling
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Chapter One

1.0 Introduction

1.1 History of the Maize Sheller

The invention of the modern crank-operated maize sheller is widely attributed to Mr. Lester E. Denison from Middlesex County, Connecticut. Denison was issued a patent on August 12 1839, for a freestanding, hand-operated machine that removed individual kernels of maize by pulling the cob through a series of metal-toothed cylinders which stripped the kernels off the cob.

During that same century, dozens of American patents were filed for maize shellers made of wood, iron or a combination of the two, including one in 1845 by Joseph Briggs of Saratoga County, New York. His sheller produced similar results to that of the Denison sheller but was a compact unit, designed to be supported on a bench or chair.

In the early 1900's, a number of engine-powered maize shellers were developed which provided the foundation for modern commercial and agricultural shellers. These large stream-powered machines have now been mostly replaced with the use of the modern combine harvester that strips the kernels from the maize cob while the maize is being harvested in the field.

Since the introduction of the modern maize sheller in the 1800's, the basic design and function of this machine has remained the same with most modern-day maize shellers bearing a strong resemblance to the original models designed by inventors like Denison and Briggs.
1.2 Background

Maize is one of the most important staple crops in the world. In Kenya, for example, 45% of the population considers maize meal (Ugali) to be their survival food, making it the most consumed food of the country. According to the D-Lab Corn Sheller writing at the Massachusetts Institute of Technology (Accessed on Oct 4th 2013), maize accounts for 43% of the Latin American diet. In Asia, maize production is over 200 billion kilograms a year and it is expected that the total maize production in developing countries will eventually overtake production in industrialized countries.

Maize is the most important cereal grain in the world, after wheat and rice, providing nutrients for humans and animals and serving as a basic raw material for the production of starch, oil and protein, alcoholic beverages, food sweeteners and, more recently, fuel. It is because of the important place of maize that its handling, processing and preservation within the optimum conditions must be analyzed.

The major steps involved in the processing of maize are harvesting, drying, de-husking, shelling, storing, and milling. All these processes are costly and for the rural farmers to maximize profits on their produce, appropriate technology that suites their needs must be used.

Maize processing not only prolongs its useful life but also increases the net profit farmers make from mechanization technologies. It is in this line that one of the most important processing operations done to bring out the quality of maize is shelling or threshing of maize. It is basically the removal of the maize kernels from the cob. This separation, done by hand or machine, is obtained by threshing, by friction or by shaking the products; the difficulty of the process depends on the varieties grown, and on the moisture content as well as the degree of maturity of the grain.
1.3 Problem Statement and Problem Analysis

1.3.1 Problem Statement

For a long time now, shelling maize to remove the grain from the cob has been a time consuming, tedious and a mind cracking process especially to the many small scale farmers in the country who basically practice subsistence maize farming. However, traditional shelling methods do not support large-scale shelling of maize, especially for commercial purposes. Hand shelling takes a lot of time, even with some hand operated simple tools. In this paper’s study area, most mechanized shellers designed for maize threshing or shelling are tractor PTO shaft operated and cause great damage to the maize seeds likewise breaking the cob to pieces. Such shellers are equipped with rotating threshing drum with beaters or teeth, which cause damages to the seed. Besides, the cost of purchasing such shellers are high for the rural farmer and therefore call for the need of a relatively low cost maize shelling mechanism that will be affordable to such farmers not only to meet their shelling requirement but also to improve the threshing efficiency and reduce damage to the seed.

1.3.2 Problem Analysis

Many small scale maize farmers opt to shell their maize produce by use of hand, something that is time consuming and tiresome. Shelling the annual maize harvest by hand typically takes weeks with children sometimes kept out of school to help with the work of shelling the maize to meet their daily food requirements. This is because processing food for survival takes priority over education in subsistence farming households since the staple food in the country is maize meal (Ugali). In addition, the hardened, dry maize can also be painful to shell and lead to hand injuries. For this reason, other such farmers choose to use simple hand held tools which are strenuous as well as slow.

For the large scale maize farmers, those who tend over 10 acres of maize crop for commercial purposes, shelling their produce has not really been a big problem majorly because they have sufficient capital to hire combined harvesters from well established companies and organizations including amongst others KSC and ADC. Alternatively, quite a number of such extensive maize farmers own tractors or they have the capacity to hire tractors which operate sheller machines.
It is in this regard that this paper presents the design of modified hand operated maize sheller which is typically a thresher for the small scale farmers who tend to maize farms less than two acres. For these farmers, the produce is approximately twenty sacks of maize in cobs or less per acre of cultivated farm.

1.4 Site Analysis and inventory

The study area in this paper’s design considerations is Kapkoi, which is in Kwanza district in Trans-Nzoia County. Due to increasing levels of poverty as well as poor cultivation techniques that result to low yields during harvest, large scale maize farming in this area is carried out by the rich including local politicians and middle class farmers. However a bigger portion of maize cultivation is carried out on small scale and mostly by women and is done on farms ranging from between one to three acres per farmer. The expected yields are approximately 20 bags per acre, of unshelled maize (maize in cobs) on the higher side when the weather conditions are favourable right from the planting to the harvesting periods. On the lower side, the yields can be as little as 10 bags per acre, of cob maize especially when there is insufficient rainfall at maize flowering periods.

Existing alternatives to shelling maize by hand are often unaffordable or difficult to obtain for subsistence farmers. An estimated 550 million small-holder farmers in the world lack access to mechanized agricultural technology. Industrial tractor PTO operated maize shellers are prohibitively expensive, with a cost range of Ksh 96000-170000; motorized shellers available in the markets cost up to Ksh 40000 depending on the horse power capacities of the motors. Small-scale mechanical hand-cranked or pedal-powered maize shellers cost upto ksh. 10000, but the technicality of their operation limits their use.

While industrial shellers are highly productive, their energy infrastructure requirements can render them unusable in rural villages. Furthermore, mechanized equipment and stationary pedal-powered devices are difficult to transport to the users. As a consequence, farmers may be required to travel long distances to process their crops or the technology may not be able to reach the communities who need it most.
1.5 Justification

Kapkoi area requires a conventional maize shelling technique that would significantly cater for the farmers harvest capacity and which many households can afford. This is with due consideration to the following reasons:

- Most of the maize grown by such rural farmers is for food rather than for commercial purpose.
- Industrial maize shellers are too expensive to be purchased by such rural farmers.
- For most of the farmers, the cost of hiring the service of industrial shellers is high with respect to the amount of grain output at the end of the farming season.
- Rotary and pedal-powered maize shellers require too much energy inputs which limits their adoption by most of the farmers since they become cumbersome to use and result to too much fatigue.
1.6 Objectives

1.6.1 Broad objective

The broad objective of this project is to design a modified hand operated maize sheller.

1.6.2 Specific objectives

The specific objectives include the following:

1. To review the existing maize shelling techniques in Trans-Nzoia county.
2. To design a hand operated mechanical maize sheller.
3. To determine energy requirement for operating the machine.

1.7 Statement of the scope

This modified sheller is to be a manually operated equipment. Its work output will depend on the operator(s) as well as on the machine itself. The operator is to perform the maize shelling operation by rotating a crank handle and therefore, proper crankshaft height and crank length would be necessary for efficient operation of the machine. Improper crankshaft height and shaft length will result in discomfort to the operator and difficulties in the smooth operation of the equipment, thus resulting in lower work efficiency.

In view of the above this paper focuses on energy considerations which arise from among other factors, the physiological and psychophysical responses of the rural farmer during operation of the maize sheller at different shaft handle heights and shaft lengths and to carry out design modification in work system so as to have higher man-machine system efficiency. It is in this regard that speed of the toothed gearing in the design system will be analyzed to facilitate the determination of forces in the primary shaft as well as the power transmitted to the threshing unit.
Chapter Two

2.0 Literature Review

Maize shelling is an important step towards the processing of maize to its various finished products like flour. Threshing or shelling operations of maize follow the harvest and whatever pre-drying of the crop is undertaken. This operation can be carried out in the field or at the storage environment. The different methods of maize shelling can be categorized based on various mechanization technology used. These includes: hand-tool-technology, animal technology, and engine power technology. (FAO Corporate Document Repository on Agricultural engineering in development - Post-harvest operations and management of food grains)

2.1 Maize Shelling Techniques

Depending on the influence of agronomic, economic and social factors, threshing or shelling is done in different ways:

- threshing or shelling by hand, with simple tools;
- mechanical threshing or shelling, with simple machines operated manually;
- mechanical threshing or shelling, with motorized equipment.

2.1.1 Hand shelling

The easiest traditional system for shelling maize is to press the thumbs on the grains in order to detach them from the ears. Another simple and common shelling method is to rub two ears of maize against each other. These methods however require a lot of labour.

It is calculated that a worker can hand-shell only a few kilograms an hour. Shelling of maize, as well as of sunflowers, can be more efficiently accomplished by striking a bag full of ears or heads with a stick. Maize and sunflowers can also be shelled by rubbing the ears or heads on a rough surface.

Small tools, often made by local artisans, are sometimes used to hand-shell maize. With these tools, a worker can shell 8 to 15 kg of maize an hour.
2.1.2 Maize-shelling with Rotary Equipment

Manual shellers, which are relatively common and sometimes made by local artisans, permit easier and faster shelling of ears of maize. These come in several models, some of them equipped to take a motor; they are generally driven by a handle or a pedal. Use of manual shellers generally requires only one worker. A good example is the Antique maize shellers.

The major setbacks with these shellers are that their threshing capacities are low and most of them require to be fixed on benches before operation. Also their method of operation is too cumbersome from the fact that the crank handle is directly connected to the threshing chamber and therefore the effect of friction is too vigorous during the threshing process.
2.1.3 Mechanized threshing or shelling with motorized equipment

Nowadays many small maize shellers, equipped with a rotating cylinder of the peg or bar type, are available on the market. Their output ranges between 500 and 2000kg per hour, and they may be driven from a tractor power take off or have their own engine; power requirements vary between 5 and 15hp according to the equipment involved. For instance the French Bourgoin "Bamba" model seems well-suited to rural areas in developing countries because of its simple design, easy handling and versatility (maize, millet sorghum, etc.).

Figure 2: "Bamba" motorized maize sheller
It is also important to consider the fact that the operations of harvesting and threshing or shelling can be carried out simultaneously, by combine-harvesters or picker-shellers. Whatever the system used, it is very important that threshing or shelling be done with care. Otherwise, these operations can cause breakage of the grains or protective husks thus reducing the product’s quality and fostering subsequent losses from the action of insects and moulds. Transport of the product from the field to the threshing or shelling place must also be handled with special care, since it can bring about severe losses. Maize grain losses contribute to food insecurity and low farm incomes not only in Kenya but also in other SSA countries (Compton, 1992; Azu, 2002; Republic of Kenya, 2004). Therefore, efficient post harvest handling, storage and marketing can tremendously contribute to social economic aspects of rural communities in Kenya as stipulated in Vision 2030 (Republic of Kenya, 2007). The losses are directly measurable in economic, quantitative, qualititative, (nutritional) terms. Economic loss is the reduction in monetary value of maize grain as a result of physical loss. Quantitative maize loss involves reduction in weight and therefore can be defined and valued. Qualitative loss although difficult to assess because it is frequently based on subjective judgments (like damage), can often be described by comparison with locally accepted quality standards (Magan and Aldred, 2007). Such losses lead to lower levels of food security, hunger and low on farm incomes (Republic of Kenya, 2004).

2.2 The link in the Maize shelling techniques

Although there exist a number of maize shelling techniques as earlier discussed, each of the technique has its own shortcomings which in general call for the design of this modified maize sheller. These drawbacks include the tediousness involved in manually operating hand held maize shelling devices, the cost of buying motorized maize shelling equipment is unaffordable to the small scale maize farmers, existing hand operated rotary shellers especially those constructed by the local artisans require the farmer to use too much energy to shell very little maize. Most farmers find this as a waste of time and other valuable resources. The recent developments in maize shelling techniques as well as the design presented in this paper would provide alternative options that can be adopted to meet the sheller needs of such rural farmers.
2.3 Theoretical Framework

The design of this modified maize sheller aids its possible construction from locally available materials to shell maize and separate the cob from the grains and its cost is projected to be low and affordable. Its threshing efficiency is projected to be above 90% and breakage is estimated to be very insignificant, as well as losses.

Moisture content seriously affects the threshability of maize (Agricultural Innovations for Sustainable Development—Contributions from the Finalists of the 2009/2010 Africa-wide Women and Young Professionals in Science Competitions Volume 3 Issue 2). An average moisture content of 15% to 18% is sufficient for maize threshing.

Another factor that affects the threshability of maize in a mechanized system is the size of the maize cob. The mechanical shellers need to be adjusted to the various sizes of cobs ranging from 50mm to 85mm depending on the variety.

There are also engineering design factors that affect the design of mechanical shellers. These factors are the design of the power transmission shaft, key, selection of the prime mover, type of pulley, appropriate chain drive design and selection of appropriate bearings support.

The power delivered by a shaft is given by:

\[ P = F \times V \]  \hspace{1cm} (1)

Where \( P \) = power (Nms\(^{-1}\)), \( F \) = Force of threshing (N), and \( V \) = linear velocity (m/s).

Force required to thresh the maize is given by

\[ F = mw^2r \]  \hspace{1cm} (2)

Where \( F \) is force required to thresh maize, \( m \) is mass of the prime movers, \( w \) is the angular velocity of shaft.
The angular velocity \( w \) is determined by the equation \( 2 \pi N/60 \), where \( N \) is the speed of threshing which is in revolutions per minute.

The power delivered by the shaft is \( Fwr \).

The total threshing power is determined by an appropriate threshing speed that will give very low mechanical damage, but high threshing output within the range of 40 - 100 revolutions per minute.

The relationship in the speed of the driver and driven gears of the spur gearing is given by

\[ N1 \cdot T1 = N2 \cdot T2 \] (3)

where \( N1 \) and \( N2 \) are the speeds of the driver and driven gears whereas \( T1 \) and \( T2 \) are the number of teeth of the respective gears.

The relation can also be stated as;

\[ N1 \cdot D1 = N2 \cdot D2 \] (4)

where \( D1 \) and \( D2 \) are the overall diameters of the driver and driven gears.

The chain drive system in this design is composed of the larger sprocket and smaller sprocket. The exact length of the chain may be determined by the relation \( L = pK \) where \( p \) is the pitch of the chain and \( K \) a multiplying factor determined from the number of teeth on the larger sprocket (\( T1 \)) and the number of teeth on the smaller sprocket (\( T2 \)).

The momentum of the flywheel that is mounted at the far end of the primary shaft determines the torque and contributes to the total power that the chain drive transmits to the threshing chamber.

\[ L = I \cdot \omega \] (5)

where \( L \) is the angular momentum, \( I \) the mass moment of inertia and \( \omega \) the angular velocity of the flywheel. \( I = m.k^2 \), \( k \) being the radius of gyration of the flywheel.
The shelling machine to be designed will be recommended for testing to determine its effective use with respect to the work to be done. Its performance is to be evaluated based on the throughput capacity, effective throughput capacity and its mechanical efficiency.

The throughput capacity (Tp) is given as:

\[ Tp = \frac{W_t}{t_t} \text{ in kg/hr}, \]  

(6)

where \( W_t \) is the total weight of material handled, which includes threshed and unthreshed, and \( t_t \) is the total time taken in handling the materials.

The effective throughput capacity is the ratio of actual weight of grains handled that was not damaged to the effective time of operation.

Also, \( T_{pe} = \frac{W_a}{t_e} \)

where \( T_{pe} \) is the effective throughput capacity, \( W_a \) is the actual weight of grain handled in kg, and \( t_e \) is the effective operating time in hour.

The efficiency in \( \% \), \( \eta \) is the percentage of the ratio of the total weight of grain actually handled (output), \( W_a \text{(kg)} \) to the total weight of grain to be handled (input), \( W_t \text{(kg)} \)

\[ \eta = \frac{W_a}{W_t} \]  

(7)
Chapter Three

3.0 Design Methodology

The methods used in this design are in three phases; the first phase involves the collection of rural farmer sheller needs. The second stage is the design of an appropriate system to meet their needs, and finally to communicate results to the farmers and determine whether their problem will be solved.

3.1 Collection of the rural farmer sheller needs

The farmer sheller needs are vital in identifying the performance of the shelling machines currently available in the market. Majority of the farmers in the site under study lamented on the high costs charged by local business men and women who offer maize shelling services normally for hire. Such shellers are usually powered by tractor PTO and require more than eight people for its effective operation. The farmers are therefore forced to dig deep into their pockets to facilitate payment of such casual workers alongside the payment of machine service. Considering the low harvest capacity of individual farmers from the sizes of their farms as stated earlier in this paper, such a move would cost them a lot and cut a great deal on their returns.

On the other hand the farmers termed such techniques as those of shelling by hand and use of sticks to beat maize in sacks so as to shell as stone age shelling techniques that are a waste of time and energy and which would only be adopted as a fall back plan if all means to shell their produce fail. The use of small rotary shellers such as the antique sheller presented a further challenge since they had to set up a working bench on which they could mount the machine. Additionally, the farmers also complained of the slow rate at which such equipment remove kernels from the cobs and most of them were reluctant to use them frequently.

They suggested that if a cheaper solution would be availed to them to solve their maize shelling problem, with minimal breakage of the kernel and cob then such a technique would be warmly welcome and embraced by individual farmers or a group of such rural farmers.
On the field determination of farmer shelling capacity is vital. Comparison was made on the time taken to shell the quantity of maize harvested per farmer and the time taken before deterioration sets in. It is important to consider the fact that appropriate technology for storage of maize produce is not readily available including pesticides to handle weevil attack. Pesticides are purchased from local agro-vets and do not last long hence the need to design a threshing mechanism that would eliminate breakage of kernels as much as possible to reduce chances of easy attack by the pests.

Essentially, rural farmers’ problems in relation to maize shelling had to do with energy inputs in shelling of their produce, cost of the shelling technology used and mechanical advantage of using the equipment as well as self reliance of the equipment in terms of rigidity, support and portability.

3.2 Design of the hand operated Mechanical Maize sheller

3.2.1 Design Considerations

The parameters considered in the design of this modified hand operated maize sheller include the following:

➢ Dried cobs with maize kernel moisture content of 15% to 18% to ease the removal of the seeds from the cob.
➢ Overall height of the machine to facilitate ease of operation by a rural farmer of average height.
➢ Overall width and breadth of the machine for purposes of storage space in the rural farmers granaries.
➢ Weight of the equipment for portability

3.2.2 Working Principle of the Machine Design

The uniqueness of this design is that it works on a different principle of threshing. As compared to other designs which work on the principle of impact force, this design works on the principle of abrasion; an application of force tangentially on a surface.
The machine is to be operated by applying force to rotate the crank handle. Motion of the handle provides an angular velocity that is translated to the toothed gearing system. The driven gear is fixed on the primary shaft hence the resultant velocity of the driven gear generates power on this shaft. As the shaft rotates, the flywheel mounted on its far end provided an angular momentum which adds more power to that generated along the length of the shaft.

This total power is transmitted to the secondary shaft via the chain drive hence providing rotary motion of the beater discs which pull and shell the maize cobs by friction and shearing action against the spiked cast iron projections on either side of the thresher bar. The empty cobs will pass out through the cobs outlet opening and are thrown out by the force of rotation of the shelling discs, and then grain will spread through the grain outlet (collector and port).

3.2.3 Design specifications

The design of this maize sheller is based on consideration of design specifications whose choice is based a number of factors that include the availability of construction materials needed for a further fabrication of the work presented in this paper, cost of such materials, desired size of the machine for Ergonomics of using it, machinability factor which includes, installation, simplication, and durability as well as the prolonged life of using the machine. These design specifications include the following;

Overall dimension 1000mm x 690mm x 1500mm
Shaft of atleast 700mm in length and 40mm in diameter
Hopper of Overall Height 520mm inlet allowance of 550mm by 550mm
Crank handle of length 170mm and height 300mm
Spur gears of gear ratio above 3, with the driven preferably being a freewheel cog gear.
Small/large sprockets (4-13 STD/8-23STD) respectively
Flywheel of atleast 14kg in weight
Thresher discs of atleast 20kg total weight
Centre to centre shaft distance not less than 500mm
Angle steel bars of $1\frac{3}{2}$ by $1\frac{1}{2}$ and 2mm thickness
3.2.4 Design Diagram

Figure 3: Design diagram
3.2.5 Components of the Design

Figure 4: components of the design
3.3 Design Analysis

3.3.1 Handle-gear Mechanism

![Gear Mechanism Diagram]

Fig 5: Gear Mechanism

The loads on the shaft caused by gear contact forces are determined by:

N1, N2; the teeth numbers of the gears mounted on shafts 1 (connected to the handle) and shaft 2 (the primary shaft).

R1, R2; the gear radii of the driver and driven components

ω1, ω2 and v1, v2; the velocities of the driver and driven gears (linear and angular)

Gear ratio = \frac{N1}{N2}

Torque T1 = \frac{T2}{\lambda}
3.3.2 Toothed Gearing

The driver and driven toothed gearing system in this design project is meant to minimize the effect of slipping which would otherwise reduce the velocity ratio of the system.

The motion and power transmitted by gears is kinematically equivalent to that transmitted by friction wheels or discs as described by the following figure;
For two toothed gears mounted on shafts, having sufficient rough surfaces and pressing against each other, a little consideration will show that when wheel A is rotated by a rotating shaft, it will rotate the wheel B in the opposite direction as shown in the figure above. The teeth minimize slipping and for as long as the tangential force, P exerted by the wheel A does not exceed the maximum frictional resistance between the two wheels, the wheel B will be rotated by wheel A.

The relationship in the speed of the driver and driven gears of the spur gearing system is given by

\[ N_1 \cdot T_1 = N_2 \cdot T_2 \]

Where \( N_1 \) and \( N_2 \) are the speeds of the driver and follower gears where as \( T_1 \) and \( T_2 \) are their respective number of teeth. This design incorporates a gear system where the driver and the driven gears have a teeth ratio \( A:B \) with the speed of the driven being \( A/B \) times that of the driver wheel.

For this design, the toothed gearing is the aspect of the drive train that determines the relation between the cadence, the rate at which the crank handle and the rate at which the driven gear turn. A spur gear system with the driver having 42 teeth and the driven freewheel cog gear
having 13 teeth is recommended. This combination would result to a gear ratio of 3.23. However, any other gear combination that would allow a gear ratio above 3 is adoptable.

The driven freewheel cog gear is chosen on the basis of having a freewheel mechanism which basically is a design consideration to allow for coasting. The gear works using internal planetary or epicyclic gearing which alters the speed of the hub casing and the wheel itself, relative to the speed of the driving gear.

**Advantages of the gear drive**

1. It transmits exact velocity ratio
2. It has a high efficiency
3. It has reliable service.
4. It has compact layout
5. It may be used to transmit large power.

**Disadvantages**

1. The manufacture of gears requires special tools and equipment.
2. The error in cutting teeth may cause vibrations and noise during operation.

The metallic gears with cut teeth are commercially obtainable in cast iron, steel and bronze. For the purpose of this design, the choice of cast iron as gear material for the system is preferable compared to the steel one majorly because of cost effectiveness.
3.3.3 Hopper Design

The hopper is designed to be fed in a vertical position only. The material to be used for the construction is mild steel sheet metal, which is readily available in the market at affordable costs. The hopper has the shape of a frustum of a pyramid truncated at the top, with top and bottom having rectangular forms. This is illustrated by the following diagram.

![Figure 8: Hopper Construction](image-url)
From the principle of similar triangles, for triangles $PMG$ and $POC$ with M and O being the centres of EFGH and ABCD respectively:

$$\frac{PM}{MG} = \frac{PO}{OC}, \text{ or } PM = PO \times MG/OC.$$  

Then the volume of the hopper is given by:

$$V_{\text{hopper}} = \frac{(\text{Area of Base}) \times \text{height}}{3} = \frac{(AB \times BC) \times h - (EH \times HG) \times x}{3}, \quad (8)$$

where,

h - overall height

x – height of the truncated top

3.3.4 The Main Frame

The main frame supports the entire weight of the machine. The total weights carried by the main frame are:

- weight of the hopper and housing;
- weight of the threshing chamber;
- the collector and pot; and
- the bearings, gears and Chains.

The two design factors considered in determining the material required for the frame are weight and strength. In this design work, angle steel bar of $\frac{1}{2}$" by $\frac{1}{2}$" and 2mm thickness is to used to give the required rigidity.
3.3.5 Shaft Design

A shaft is a rotating or stationary member, usually of circular cross-section having such elements as gears, pulleys, flywheels, cranks, sprockets and other power transmission elements mounted on it Shigley (1986).

The design presented in this paper comprises of two shafts; the primary and the secondary shaft. The primary shaft is to be fitted to the spur gear system on one end and to a flywheel on the other end. The secondary shaft is designed to have threshing discs attached to it (by welding) and parallel to the primary one with a chain drive mounted to connect the two. Both shafts are solid shafts of ductile material and circular cross-section designed to be supported on bearings.

Shaft design consists primarily of the determination of the correct shaft diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating and loading conditions.

For this design a shaft of diameter 40mm and length 730mm shall best serve the interests of achieving the overall machine width and also lower the costs of materials that would be required if the machine is to be constructed. Preferably, such a shaft should have a key way and with the specified dimensions it will serve best in both cases of the shaft requirement since this design incorporates two shafts. The key way is a design consideration meant to firmly hold in place the flywheel to be mounted as well as the driven gear.
3.3.5.1 Proportion of a Key

For a good result, the width of a key is made one-quarter the diameter of the shaft. The thickness of a key for equal strength of the key in failure by shearing of the key, and compression on the key may be determined by the corresponding allowable stresses in shear and compressions.

The length of the key can be calculated as \( L = \frac{d}{2} = 1.57d \).

The forces on the top and bottom of the key resist tipping of the key, and the force, \( F \), between the side of the key and the key way in the hub is due to the resisting torque, \( T' \):

\[
T' = \frac{Fd}{2} = \frac{FL}{\pi},
\]

where:

\( T' \) = resisting torque;

\( F \) = resisting force;

\( d \) = diameter of shaft;

\( L \) = length of key.
3.3.5.2 Calculation of Reactions RB and RD on the primary shaft

From the above figure;

$XT$ - total length of shaft;

$X$ - distance along the shaft

$W_p$ – weight of the chain drive

$T_1$ and $T_2$ – Tensions in the chain

Taking moment about $RB$:

Sum of clockwise moments equal = sum of anticlockwise moments, i.e.,

$$(W_p + T_1 + T_2)XT + Hx6 + Gx5 + Ex4 + Dx3 + Cx2 = RDx7,$$

$$RD = [(W_p + T_1 + T_2)XT + Hx6 + Gx5 + Ex4 + Dx3 + Cx2]/x7.$$
But sum of upward forces = sum of downward forces:

\[ RD + RB = WP + T1 + T2 + C + D + E + G + H, \]

\[ RB = (WP + T1 + T2 + C + D + E + G + H) - RD. \]

3.3.6 Bearing Selection

Bearing must be selected based on its load carrying capacity, life expectancy and reliability (PSG Tech 1989). The relationship between the basic rating life, the basic dynamic rating and the bearing load is:

\[ C = \left[ \frac{L}{L10} \right]^{1/KP}, \text{ or } C/P = \left[ \frac{L}{L10} \right]^{1/K}, \text{ that is; (10)} \]

\[ [C/P]K = L/L10, \text{ or } L10 = [C/P]K/L. \]

But \( L = 60n/106 \) million revolutions, therefore, \( L10 = (106/60n) \times [C/P]K \),

where:

\( L10 = \) life of bearing for 90% survival at one million revolutions;

\( L = \) required life of bearing in million revolutions (mr);
\( n = \) rotational speed (rev/min);
\( C = \) basic dynamic load rating (N);
\( P = \) equivalent dynamic bearing load (N);
\( K = \) exponent for life equation with:
\( K = 3 \) for ball bearing;
\( K = 10/3 \) for roller bearing.

Also, \( P = radial \ load + axial \ load, \)
\( P = (XFr + YFa), \)
where:
\( X = \) radial load factor for the bearing;
\( Y = \) axial load factor for the bearing;
\( Fr = \) actual radial bearing load (N);
\( Fa = \) actual axial bearing load (N).
Table 1 below shows the recommended life value in operation. It is assumed that this machine will be designed to operate for 8 hours per day intermittently and whose breakdown will have serious consequences.

The bearing life in operating hours is chosen to be 8,000 as illustrated by the table below;

<table>
<thead>
<tr>
<th>Type of operation</th>
<th>Life in operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Infrequently operated</td>
<td>500</td>
</tr>
<tr>
<td>Brief operation only</td>
<td>4,000-8,000</td>
</tr>
<tr>
<td>Intermittent operation</td>
<td>8,000-15,000</td>
</tr>
<tr>
<td>One shift operation</td>
<td>15,000-30,000</td>
</tr>
<tr>
<td>Continuous operation</td>
<td>30,000-60,000</td>
</tr>
<tr>
<td>Continuous operation with high production capacity</td>
<td>100,000</td>
</tr>
</tbody>
</table>

Table 1: Recommended life value of bearings

3.3.7 Other design Considerations

3.3.7.1 Calculation of the Shearing Force and Bending Moment of the Shaft at Different Sections of the Shaft

Here:

S.F. = upward forces – downward forces;

B.M = forces x perpendicular distances.
3.3.7.2 Force in the beater discs required for threshing

The beater discs whose surfaces are designed with alternating rasp bars and grooves, and which are attached to the secondary shaft extending to the threshing chamber, rotate with the shaft, giving rise to centripetal force:

\[ F = m \omega^2 r, \]  
(11)

where:

\( F \) = centripetal force;

\( m \) = mass of discs;

\( \omega \) = angular velocity;

\( r \) = max disc radius.

3.3.7.3 Determination of Angular Velocity, \( \omega \)

The angular velocity, \( \omega \), is given by:

\[ \omega = \frac{2\pi N}{60}, \]  
(12)

where:

\( N \) = speed of the shaft in r.p.m.

3.3.7.3 The Radius, \( r \), of the Threshing Arm

The radius, \( r \), of the threshing arm increases along the length of the shaft and also decreases towards the other end of the shaft, where:

\( r \) = radius of threshing arm and it is given as

\( r_{max} = 0.045m \) (assumed),

\( r_{min} = 0.035m \) (assumed), so that

centripetal force at \( r_{max} \) (\( F \)) = \( m r_{max}^2 \),

centripetal force at \( r_{min} \) (\( F \)) = \( m r_{min}^2 \).
3.3.7.4 Determination of Threshing Torque

The torque, $T$, is given by:

$$ T = F \times r, \quad (13) $$

where:

$F =$ available centripetal force;

$r =$ threshing radius

3.3.7.5 Determination of the Power Delivered By primary Shaft

Power, $P =$ work done per second:

$$ P = \frac{(\text{work done})}{\text{time}} = \frac{(\text{force} \times \text{distance})}{\text{time}} = \text{force} \times \text{velocity}. $$

velocity $= r,$

where:

$\omega =$ angular velocity;

$r =$ radius.

Therefore, $\text{power} = F \times r$

3.3.7.6 Determination of Torsional Moment, $MT$

The torsional moment, $MT$, is given by

$$ MT = 9,550 \times \frac{\text{KW}}{N}, $$

where:

KW = power delivered;

$N =$ revolutions per minute.
3.3.7.7 Bending Stress

According to Hall et al. (1988), for bending load, bending stress (tension or compression) is:

\[ S_b = \frac{M_b r}{I} \]  

(14)

Hence,

\[ S_b = \frac{(32M_b)}{(d^3)}, \]

where:

\( S_b \) = bending stress;
\( M_b \) = bending moment;
\( d \) = shaft diameter;
\( I \) = moment of inertia

Also, \( I = \frac{d^4}{64} \), for a cylindrical shaft (PSG Tech 1989).

3.3.7.8 Torsional Stress

According to Black and Adams (1968), torsional stress is determined using

\[ \tau_{xy} = \frac{M_T r}{J} \]  

(15)

But \( J = \frac{d^4}{32} \), hence, \( \tau_{xy} = \frac{(16M_T)}{(d^3)} \),

where:

\( \tau_{xy} \) = Torsional stress N/m²;
\( M_T \) = torsional moment;
\( r \) = radius of shaft;
\( J \) = Polar moment of area;
\( d \) = diameter of shaft.
3.3.7.9 Torsional Rigidity

Torsional rigidity of a shaft is based on permissible angle of twist. The amount of twist permissible depends upon the particular application and varies from 0.3 degree/m for a machine tools shaft to about 3 degree/m for line shafting as given by:

\[ \theta = \frac{(584 \, MTL)}{(Gd^4)}, \]  

where:

\[ \theta = \text{angle of twist (degree)}; \]
\[ L = \text{length of shaft (m)}; \]
\[ MT = \text{torsional moment (Nm)}; \]
\[ G = \text{Torsional modulus of rigidity (N/m}^2); \]
\[ d = \text{diameter of shaft (m)}. \]

3.3.7.10 Lateral Rigidity

The lateral rigidity of a shaft is based upon the permissible lateral deflection for proper bearing operation, accurate machine tool performance, shaft alignment etc. Amount of deflection can also be calculated by two successive integrals of the formula:

\[ \frac{d^2y}{dx^2} = \frac{(Mb)}{(EI)}, \]  

where:

\[ Mb = \text{bending moment N/m}^2; \]
\[ E = \text{Modulus of elasticity (N/m}^2; \]
\[ I = \text{Moment of inertia (m}^4). \]
3.3.7.11 Shear Stress

The shear stress on the shaft is determined by the formula below (Black and Adams 1968):

\[ \tau = \frac{16T}{\pi d^3}, \]  

(18)

where:

\[ \tau = \text{Shear stress N/m}^2; \]

3.3.8 Flywheel Fitting

The flywheel to be fitted in this design is meant to retain the momentum established by the gear system as the primary shaft rotates and in turn rotating the secondary shaft via the chain drive such that even when the operator stops the cranking cycles, the grooved-rasped discs at the threshing unit are still in motion and maize threshing progresses. In general, the flywheel disc is fit to provide continuous energy when the energy source is discontinuous. In such cases, the flywheel stores energy when torque is applied by the energy source and it releases stored energy when the energy source is not applying torque to it. The energy is stored in the rotor as kinetic energy, or more specifically, rotational energy given by the following equation:

\[ E_k = \frac{1}{2} I \omega^2 \]  

(19)

Where:

- \( \omega \) is the angular velocity, and
- \( I \) is the moment of inertia of the mass about the center of rotation.

\[ I = \frac{d^4}{64}, \text{ for a cylindrical shaft (PSG Tech 1989)}. \]
The flywheel to be used in this design is one made of mild steel and which conforms to the following specification; mass-14kg, external diameter D-360mm, internal diameter d-41mm. This is illustrated by the following figure;

![Flywheel Image]

Fig 10: flywheel

3.3.9 Chain Drive

Since the vertical distance between the primary and secondary shafts is short, steel power transmitting chains which have provision for efficient lubrication are used in this design for two major reasons:

1. To transmit motion and power from one shaft to another.
2. In order to avoid slipping usually encountered in rope and belt drives.

The chains are made up of rigid links which are hinged together in order to provide the necessary flexibility for warping around the driving and driven wheels. The wheels have projecting teeth and fit into the corresponding recesses, in the links of the chain. The wheels and the chain are thus constrained to move together without slipping and ensure perfect velocity ratio. The toothed wheels are known as sprocket wheels or simply sprockets.

These wheels resemble spur gears.
Mechanisms consisting of two sprockets connected by a chain are designed to provide a specific mechanical advantage in a power transmission system.
The velocity $v$ of the chain is the same when in contact with the two sprockets:

$$v = r_A \omega_A = r_B \omega_B$$

where the input sprocket $A$ meshes with the chain along the pitch radius $r_A$ and the output sprocket $B$ meshes with this chain or along the pitch radius $r_B$, therefore;

$$\frac{\omega_A}{\omega_B} = \frac{r_B}{r_A} = \frac{N_B}{N_A}$$

where $N_A$ is the number of teeth on the input sprocket and $N_B$ is the number of teeth on the output sprocket.

The mechanical advantage of a pair of a chain drive or toothed belt drive with an input sprocket with $N_A$ teeth and the output sprocket has $N_B$ teeth is given by;

$$MA = \frac{T_B}{T_A} = \frac{N_B}{N_A}$$

Chain drives are smaller in size, lose less power in friction, and are much cheaper in maintenance than belt drives. Also chain being inherently stronger than belts, they can handle heavier loads than their belt counterparts. Chain drive are especially effective where not more than 100kw of power is to be transmitted at a peripheral speed of up to $15\text{ms}^{-1}$ and a speed ratio of up to 8.

Chain drives may be used as both step-down and step-up transmission (the latter been employed in this design). Used by man from time immemorial, chains have nevertheless lost none of their importance. Today, chain drives can be found practically everywhere from a modest household to a major industrial plant.
3.3.9.1 Advantages and Disadvantages of Chain Drive over Belt or Rope Drive

The following are the advantages and disadvantages of chain drive over belt or rope drive:

**Advantages**

1. As no slip takes place during chain drive, hence perfect velocity ratio is obtained.

2. Since the chains are made of metal, therefore they occupy less space in width than a belt or rope drive.

3. The chain drives may be used when the distance between the shafts is less.

4. The chain drive gives a high transmission efficiency (upto 98 per cent).

5. The chain drive gives less load on the shafts.

6. The chain drive has the ability of transmitting motion to several shafts by one chain only.

**Disadvantages**

1. The production cost of chains is relatively high.

2. The chain drive needs accurate mounting and careful maintenance.

3. The chain drive has velocity fluctuations especially when unduly stretched.
3.3.9.2 Relation between Chain Speed and Angular Velocity of Sprocket

Since the links of the chain are rigid, therefore they will have different positions on the sprocket at different instants. The relation between the chain speed \( v \) and angular velocity of the sprocket \( \omega \) also varies with the angular position of the sprocket. The extreme positions are shown in Figures 11 \((a)\) and \((b)\) below.

\[
\text{For the angular position of the sprocket as shown in fig (a): } v = \omega \times OA
\]

\[
\text{For the angular position of the sprocket as shown in fig (b): } v = \omega \times OX = \omega \times OC \cos (\theta/2) = \omega \times OA \cos (\theta/2)
\]

and therefore \( OC = OA \).

3.3.9.3 Length of Chain

For an open chain drive system connecting the two sprockets in this paper’s design, the exact length of the chain may be determined as discussed below:

Let \( T1 = \text{Number of teeth on the larger sprocket} \),

\( T2 = \text{Number of teeth on the smaller sprocket} \), and
\[ p = \text{Pitch of the chain.} \]

Pitch of the chain is the distance between the hinge centre of a link and the corresponding hinge centre of the adjacent link as shown in the above figure. It is usually denoted by \( p \).

On the other hand, Pitch circle diameter of the chain sprocket is the diameter of the circle on which the hinge centres of the chain lie, when the chain is wrapped round a sprocket. From the above figure, the points \( A, B, C, \) and \( D \) are the hinge centres of the chain and the circle drawn through these centres is called pitch circle and its diameter \( (d) \) is known as pitch circle diameter.

The diameter of the pitch circle is \( d = p \cdot \csc \left( \frac{180^\circ}{T} \right) \) or \( r = \frac{p}{2} \cdot \csc \left( \frac{180^\circ}{T} \right) \)

For larger sprocket \( r_1 = \frac{p}{2} \cdot \csc \left( \frac{180^\circ}{T_1} \right) \)

For smaller sprocket \( r_2 = \frac{p}{2} \cdot \csc \left( \frac{180^\circ}{T_2} \right) \)

Since the term \( (r_1 + r_2) \) is equal to half the sum of the circumferences of the pitch circles, therefore the length of the chain corresponds to;
\[ (r_1 + r_2) = \frac{p}{2} (T_1 + T_2) \]

but;

\[ L = \pi (r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \]  \hspace{1cm} (20)

Substituting the values of \( r_1, r_2 \) and \( r_1 + r_2 \) in the equation 20 above, the length of chain is given by:

\[ L = \frac{p}{2} (T_1 + T_2) + 2x + \frac{p}{2} \csc \left( \frac{180^\circ}{T_1} \right) - \frac{p}{2} \csc \left( \frac{180^\circ}{T_2} \right) \]
If \( x = m \varphi \), then

\[
L = p \left[ \frac{(T_1 + T_2)}{2} + 2m + \left( \frac{\csc(180^\circ)}{T_1} - \frac{\csc(180^\circ)}{T_2} \right)^2 \right] = p.k
\]  

(21)

Where \( k \) is the multiplying factor.

### 3.3.10 Threshing Unit

The threshing unit for this design is composed of two metallic discs mechanically attached to the secondary shaft and rotating against a rigid spiked metallic surface. The discs are connected facing each other and rotate as block and their surfaces is to be of grooves and rasp bars so as to provide a rough contact on the maize cob against the spiked rigid metallic surface. Hence this forced friction results to the threshing of maize kernels from the cob.

Below the discs, a collector and pot metallic member is to be fitted and is designed also to incline towards its central longitudinal axis where an opening is allowed for grain collection as the cobs pass over the member surface towards the exit where they are collected. The grain is to be collected below the machine.
Fig 16: The Threshing Unit
Chapter Four

4.0 Design Calculations, Results and Discussion

4.1 Design Calculations

4.1.1 Length of Chain

We know that the pitch of the chain, \( p \) is given by:

\[
p = \frac{d \sin\left(\frac{360^\circ}{2T}\right)}{d \sin\left(\frac{180^\circ}{T}\right)}
\]

Using standard sprocket dimensions from the table A1 in the Appendix Section of this paper, the desired sprocket sizes are determined by the pitch circle diameters \( d_1 = 7.910" \) and \( d_2 = 4.350" \) for the large (23 teeth) and small (13 teeth) sprockets respectively.

Let \( p = \) pitch of the chain and since the pitch circle diameter of the sprocket to be driven is 4.350" = 11.049cm. Hence;

\[
p = 11.049 \times \sin\left(\frac{180^\circ}{13}\right) = 2.644cm
\]

The design centre to centre distance between the two sprockets or rather the distance between the central axes of the primary and secondary shafts is considered to be 500mm. Therefore;

\[
m = \frac{x}{p} = \frac{50}{2.644} = 18.911
\]

\[
k = \left[ \frac{T_1 + T_2}{2} + 2m + \cfrac{\csc\left(\frac{180^\circ}{13}\right) - \csc\left(\frac{180^\circ}{23}\right)}{4m} \right]^2
\]

\[
= \frac{(23 + 13)}{2} + 2 \times 18.911 + \frac{\csc\left(\frac{180^\circ}{23}\right) - \csc\left(\frac{180^\circ}{13}\right)}{4 \times 18.911}^2
\]

= 55.954
And therefore;

\[ L = p.k = (55.954 \times 2.644) \text{ cm} \]

\[ = 147.94 \text{ cm} \]

4.1.2 Force and power required for threshing

For this design the average crank speed for high and efficient threshing is estimated at 50rpm. This speed is used as the basis for the preceding calculations.

The driver gear rotates with a velocity given by;

\[ \omega_1 = \frac{2\pi N}{60} = \frac{2\pi \times 50}{60} = 5.236 \text{ rad s}^{-1} \]

\[ v_1 = \omega r = 5.236 \times 0.3 = 1.571 \text{ m s}^{-1} \]

For the toothed gearing system, consider the relationship \( N_1V_1 = N_2V_2 \)

\[ v_2 = \frac{N_1V_1}{N_2} = \lambda v_1 = \frac{42 \times 1.571}{13} = 5.076 \text{ m s}^{-1} \]

The above calculation yields \( v_2 = 5.076 \text{ m s}^{-1} \) which is the speed of the driven gear hence the rotating linear speed of the primary shaft.

To aid the rotary motion of the beater/thresher discs in the threshing chamber, the force to be applied to move the crank handle tangentially is calculated while putting into consideration the mass of the flywheel (\( m_f \)), mass of the two discs (\( m_d \)), the mass of the primary and secondary shafts(\( m_s \)) and the mass of the gears, sprockets and the chain (\( m_o \)). For this design, these components are estimated as 14kg, 20kg, 8kg and 8kg.
Hence, the total effective mass for calculations of the force of threshing is given as;

\[ M = m_f + m_d + m_s + m_o = 14 + 20 + (2 \times 4) + 8 = 50 \text{ kg} \]

Therefore the threshing force to be applied at the handle can be given by the relation;

\[ F = m \omega^2 r, \]

\[ = 50 \times 5.236^2 \times 0.3 = 411.23 \text{ N} \]

This is approximately considered as 412 N

Hence the power needed to drive the crank handle at 50rpm is calculated from the relation;

\[ P = F \times v = (412 \times 1.571) \text{ Nms}^{-1} = 647.25 \text{ Watts} \approx 650 \text{ W} \]

**4.1.3 Torque developed on the primary shaft**

The Torque on the primary shaft is a function of the Torque on the shaft of the handle, connecting it to the driver gear. Therefore, the Torque on the handle’s shaft \( T_1 \) is determined by the relation;

\[ T_1 = F \times r = 412 \text{N} \times 0.03 \text{m} = 12.36 \text{ Nm} \]

And the torque developed on the primary shaft can be determined by the relation;

\[ T_2 = T_1 / \lambda = \frac{12.36}{3.23} = 3.83 \text{ Nm} \]

**4.1.4 Power Developed on the shafts**

The power delivered by the primary shaft of diameter 40mm is determined by the relation;

\[ P = F \times v = 412 \times 5.076 = 2091.31 \text{ W} \approx 2092 \text{ W} \]

The flywheel fitting is designed to store kinetic energy or rotational energy when the torque is applied and release this energy when the crank cycles are discontinued. This energy is determined as follows;
\[ E_k = \frac{1}{2} I \omega^2 = \frac{1}{2} (m r^2) \omega^2 = \frac{1}{2} \left( \frac{\pi d^4}{64} \right) \omega^2 \]

but \( \omega_2 = \frac{v}{r} = 5.076/0.18 = 28.2 \text{ rads}^{-1} \)

Hence \( E_k = 0.5 \times 14 \times 0.18^2 \times 28.2^2 = 180.36 \text{ J} \)

The angular momentum of the mounted flywheel determines the torque and contributes to the total power that the chain drive transmits to the threshing chamber.

\[ L = I \times \omega = m r^2 \omega = (14 \times 0.18^2 \times 28.2) = 12.79 \text{ kgm}^2/\text{s} \]

The power generated per second of the flywheel revolutions can therefore be given as

\[ \frac{180.36 \text{ J}}{1\text{s}} = 180.36 \text{W} \approx 181 \text{ W} \]

And the total power transmitted to the threshing chamber via the drive chain is therefore

\[ P_{\text{total}} = (2092 + 181)\text{W} = 2273 \text{ W} \]

Considering a 2% loss in power transmission in the drive chain, the power delivered to the secondary shaft can be determined as follows;

\[ P_s = \frac{98}{100} \times 2273 = 2227.54 \text{ W} \approx 2228 \text{ W} \]

For a chain drive system;

\[ \frac{\omega_A}{\omega_B} = \frac{r_B}{r_A} = \frac{N_B}{N_A} \]

But \( \omega_2 = \omega_A = 126.9 \text{ rad/s} \) and since the design specifications require that \( N_A \) be 23 teeth and \( N_B \) of 13 teeth as per table A1 appended, then;

\[ \omega_B = \frac{\omega_A}{13} = \frac{28.2}{0.565} = 49.91 \text{ rad/s} \]

And therefore;
\[ v_B = \omega \cdot r = 49.91 \times 0.04 = 1.9964 \approx 2 \text{ m/s}^{-1} \]

\[ MA = \frac{T_B}{T_A} = \frac{N_B}{N_A} = \frac{13}{23} = 0.565 \]

Since this mechanical advantage is less than 1, this means that the secondary shaft will rotate at a faster speed compared to the cranking speed. From the above calculation, it is evident that \( V_1 = 1.571 \text{ m/s} \) and \( V_B = 2 \text{ m/s} \).

The centripetal force gained by the thresher discs is therefore the total force in the threshing unit and is given by:

\[ F_{\text{threshing unit}} = m\omega^2 r = 20 \times 49.91^2 \times (0.34/2) = 8.5 \text{kN} \]

If the average crank speed is considered to be below or above 50rpm and all other design parameters kept constant, the table 2 below presents the expected Crank force and Torque from iterations at speeds of 40rpm, 50rpm and 60rpm.

<table>
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<tr>
<th>Crank speed (rpm)</th>
<th>( \omega_1 = \frac{2\pi N}{60} ) (rads(^{-1}))</th>
<th>( F = m\omega^2 R ) (N)</th>
<th>( T = F \times r ) (Nm)</th>
<th>( P = F \times v )</th>
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<td>5.24</td>
<td>412</td>
<td>12.36</td>
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</tr>
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<td>17.74</td>
<td>1115</td>
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Table 2: Force and Torque at three different crank speeds

<table>
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<th>Crank speed</th>
<th>Modified Design (Calculated results)</th>
<th>Cob master Maize sheller (Performance Results)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( T ) (Nm)</td>
<td>( F_{\text{crank}} ) (N)</td>
</tr>
<tr>
<td>50rpm</td>
<td>12.36</td>
<td>412</td>
</tr>
</tbody>
</table>

Table 3: Force and Torque comparisons
4.2 Discussion

4.2.1 Discussion of the Results

Since this modified hand operated maize sheller is to be a manually operated machine requiring at most two subjects, its work output will depend on the machine as well as on the operators. One operator will perform the maize shelling operation by rotating a crank handle while the other feeds the cobs and therefore, proper crank height and crank length is necessary for efficient operation of the machine. Improper cranks height and length will result in discomfort to the operator doing the cranking and difficulties in the smooth operation of the equipment, which in turn would result in lower work efficiency. It is for this reason that this design is based on the specified parameters of 170mm and 300mm for the crank handle length and height respectively.

An average crank handle speed of 50 rpm generates an initial torque of 12.36 Nm as obtained from the calculated results which is sufficient to initiate rotation of the threshing discs in the shelling chamber at a speed of 2 m/s. This generates a potential of 8.5 kN of the centripetal force required to initiate shelling in the threshing chamber. Since the thresher discs gain more rotational momentum from the flywheel, the variation in feed rates might largely depend on the subjects operating the machine; the arm power, speed of operation and torque applied to thresh the maize.

A work cycle of 1 h for subjects could be planned by shifting the subject from operating to feeding. After that a rest of 15 min could be provided to both the subjects. This would be necessary that arm work should be adjusted with adequate rest pause in accordance to Aminoff et al. (1998).

The length of the chain to be fit on the sprockets was found to be 147.94 cm but to allow for effective sag on either sides of the chain drive system, an allowance of about 3cm can be made so as to adjust this value to approximately 1.5m. This way, the chain will drive the secondary shaft more efficiently with less noise as a result of excessive friction which would otherwise have occurred if the chain is tightly fit.

Chain drives are found to be effective when the power to be transmitted does not exceed 100kW at a peripheral speed of up to 15ms⁻¹ and a speed ratio of up to 8 (R.S KHURMI and J.K.
GUPTA, 2005). The total power developed on the primary shaft in this design is just 2.228 kW hence transmission with an allowance of 2% loss is reasonable with the results presenting a peripheral speed observed in the secondary shaft of 2 ms\(^{-1}\) and the speed ratio of 1.77 values that are way below the theoretical limits.

Ergonomical studies on a hand operated rotary maize sheller to enhance human-machine compatibility done by M.M. Deshmukh1 and H.S. Kharade from the Department of Farm Power & Machinery in collaboration with P.S. Tiwari and L.P. Gite from the Agricultural Mechanization Division, Central Institute of Agricultural Engineering (India) indicated from a set of trials that a torque of 18 N-m is the torque required to operate a hand operated rotary maize sheller at speed of 50 rpm. From table 2, the torque that would be required to operate a prototype of the design presented in this paper was found to be 12.36 Nm. This represents a 31.33% reduction in energy input at the same crank speed an indication that the prototype from this design can be more efficient than other mechanical rotary maize shellers. It is also important to note that the energy input in operating such a prototype would largely depend on the speed of motion of the crank handle since different speeds present different energy requirements as observed in the results presented in the Table 2.

### 4.2.2 A Review of the existing Maize shelling Techniques (Cost Evaluation)

Maize shellers in Trans-Nzoia County are available in simple handheld or rotary and pedal operated designs as well as large commercial and agricultural units, with prices for new shellers ranging from as low as Kshs. 1500 to over Ksh. 850,000. While every corn sheller will produce similar results, the speed, materials and source of power varies widely between the different types and styles of corn shellers. The various types of threshing equipment cater to differing scales of production and conditions such as domestic use, co-ownership by small farmers, use by independent farmer on a daily hire bases and ownership by custom or merchant mills.

**Handheld Shellers**

The simplest type of maize sheller is a circular handheld device made of cast iron or cast aluminum, with new models retailing for under Ksh. 1500. To use this type of sheller, the operator holds and rotates the sheller in one hand while pushing the cob of corn through the teeth of the machine. This type of sheller is best suited for occasional use by maize growers requiring
the seed or kernel samples for moisture and disease testing. These simple shellers can also be used by home gardeners who want to shell a few dried maize cobs for use as chicken feed. Since using a handheld sheller is slow and labor intensive, this model is not suitable for processing multiple cobs of corn at once.

**Rotary Maize Shellers**

The most common type of mechanical maize sheller in the market today is a rotary sheller which can shell up to six cobs of corn per minute translating to approximately 50kg/hr. A new such sheller made of cast-iron can be purchased for less than Ksh. 8500 at farm and garden supply stores. However, Antique maize shellers (grinder pitter tool - "never fail" plymouth root heath corn shellers) are often available in the market and are the preferred type of rotary maize shellers each at a sale price of Ksh. 10700.

Most mechanical corn shellers that are produced and sold today are constructed of cast iron with a hand crank. These machines are usually designed to be either mounted on a workbench or suspended between two supports over a bucket where the kernels are collected. This on its own presents a setback for their use as most farmers do not have much time and energy to mount and unmount the equipment every time they need to shell their produce.

Furthermore, the output capacity of this sheller is relatively low for the rural farmers harvesting more than 20 sacks of maize and this would require them to spend quite some time in extends of hours or days to complete the entire threshing process.

**Powered Maize Shellers**

Well of Farmers, grain mills and others entities who require large quantities of shelled maize often chose corn shellers that are powered directly by an onboard engine or indirectly through a tractor-driven PTO. When purchased new, these shellers can cost upwards of Ksh.850,000 each especially the combined maize harvesters and are designed for heavy-duty continuous use in a commercial or agricultural environment such as a livestock feedlot, grain mill or bio-fuel distribution company, seed processing and distribution companies as well as parastatals such as the ADC. The high rates of throughput require the use of cob loading elevators and bagging equipment.
Most of these threshers utilize a pegged drum, mounted on a horizontal shaft, which rotates at about 700,000 rpm. A concave metal screen, with holes approximating to the size of the grain is located around the drum. It contains the cobs while shelling takes place. A baffle plate restricts the flow of the cobs, and maintains the required shelling pressure. A strong fan discharges the stripped cobs centres and other large debris. A second, smaller fan is often used at the grain discharge point for the removal of the remaining dust and finer particles. Available information indicates that an average thresher grain output of 900kg/hr/ installed kw/h may be obtained from these threshers.

### 4.2.3 Effectiveness of the Project’s Machine Design

With the recent surge in the cost of fuel combined with the renewed interest in home gardening and eco-friendly living, growing numbers of the rural maize farmers are exploring the use of shellers that are less cumbersome, technically feasible and cost friendly. Hence the need for the design of maize shellers that will suit the needs of such farmers as presented by the one described in this paper. This designed sheller utilizes grooved discs to pres out the grain while the cob is held by an adjustable guide plate. Depending on the number of operators employed, the capacity of this machine if constructed can be up to four times larger than that of the smaller rotary threshers. This may be explained by the use of low – friction bearings and of simple gearing which result in steady and operating speeds. The threshing principles are similar to those of the smaller rotary threshers.

### 4.2.3.1 Bill of Engineering Measurement and Evaluation

For cost evaluation purposes, the following table presents a BEME for the cost list of material of this project modification and possible construction or fabrication of this design’s hand operated maize threshing machine.
Table 4: Bill of Engineering Measurement and Evaluation

Although the cost for fabrication of this proposed design (standing at Ksh.28100) is higher than the cost of purchase of both the rotary maize shellers described earlier, the ease of operation of the machine at an average torque of only 12.36 Nm presents an added advantage in terms of the cumbersomeness and fatigue that come along with the above named shellers. The force input capacity to initiate the threshing process is also very low at an average of 412N such that both male and female rural farmers can operate the machine.
The above total cost can also be lowered if the prototype is well fabricated and commercialized such that the design is on high demand and its production costs would reduce. Alternatively, two or so farmers can merge and purchase the prototype at manageable costs instead of spending costs not less than Ksh. 6000 per acre of maize produce only on hiring tractor powered maize shellers as well as labour expenses during every end of maize cultivation season. This way, they can designate a common shelling point where they would collect all their harvests and do the shelling depending on their energy inputs.

Also the design would lead to construction of equipment that is less costly compared to the tractor powered maize shellers which are expensive to hire or purchase for such small capacity threshing requirements of the rural farmers. Furthermore the design does not require fitting of an engine or motor hence such high costs of fuel purchases are completely eliminated.

**4.2.4 Performance analysis**

The follow up to the performance of this design can be conducted after its fabrication or construction and may proceed as follows. The throughput capacity, the actual throughput capacity and the mechanical efficiency are to be determined using the equation (6) and (7) mentioned earlier.

If 200kg of un-threshed maize is to be measured using a weighing scale and a local farmer to load the cobs three at a time into the hopper, the time taken to load and finish threshing the 200kg is to be determined. The total weight of thresh grains is also a vital parameter to be determined. The total weight of the broken or damaged grain is also to be determined, and the weight of the cob also taken. The percentage mechanical damage is to be considered and a comparison is to be established between human performance index and the machine performance index.
Chapter Five

5.0 Conclusion and Recommendations

5.1 Conclusion

Self-reliance is the major drive of development and vibrant economy. This machine has been
designed to be fabricated with the use of locally available materials. The machine is simple, less
bulky and the ergonomic considerations in the design would allow for its comfortable use in a
standing/sitting posture for it can easily be operated by either male or female subjects with either
left or right hands. This is justified by an overall height of 1.25 m of the machine design as well
as the low energy requirement at an average of 412N and an initial average torque of 12.36 Nm..
The thresher can help to substantially reduce the human labour involved in threshing at an
affordable cost and also reduces the time used for threshing operation on small farms considering
the deduced fact that energy requirements solely depend on the crank speed of operating the
machine. There is no doubt that the machine will ease the long term problem of maize shelling
especially for the rural farmers.
5.2 Recommendations

- Fabrication and construction of the machine from the design presented in this paper.
- Testing of the machine for its performance and efficiency at 50rpm handle cranking to compare the expected results and the achievable outcome.
- Incorporation of cleaning and separation device for the removal of unwanted material.
6.0 References


Appendices

Appendix A

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<th>SPROCKET DESIGNATION</th>
<th>NOMINAL DIAMETER</th>
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Table A1: Sprocket specifications
Appendix B

Shucking and shelling maize in rural Kenya

Shelling maize by hand

Sheet metal maize sheller and shelled maize

Shelling maize with a sheet metal sheller

Varieties of hand held devices for maize shelling made from different materials and methods
Appendix C

Sample freewheel cog gear

Side view of the flywheel

Cob master maize sheller

Motor Powered Maize sheller

Tractor PTO operated maize sheller
Appendix D

Work programme (Ghant Chart)

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