DEVELOPMENT AND PERFORMANCE EVALUATION OF A MANUALLY-OPERATED MULTI-CROP PLANTING MACHINE

BY

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A PROJECT THESIS SUBMITTED TO THE POST - GRADUATE SCHOOL IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE AWARD OF MASTERS OF ENGINEERING DEGREE IN AGRICULTURAL ENGINEERING, (FARM POWER AND MACHINERY OPTION) OF THE FEDERAL UNIVERSITY OF TECHNOLOGY, MINNA, NIGER STATE, NIGERIA.

APRIL, 2005.

CERTIFICATION

This thesis "Development and Performance Evaluation of Manually Operated Multi-crop Planting Machine" by Idris Rabi'u Daniya (M.ENG/SEET/2001/789) meets the regulation governing the award of the degree of masters in Engineering (M.Eng) of the Federal University of Technology, Minna, and is approved for its contribution in scientific knowledge and literacy presentation.

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DEDICATION

This study is dedicated to my self and the members of my family.

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Pls, use 12 fonts!

ABSTRACT

A manually operated multi-crop planting machine was designed, constructed and tested. The results showed that it has high working capacity (1.82ha/hr), efficiency (95.07%) and can be operated at speed range of 0.6 - 0.75m/s with low rate of seed damage (0.9% to 2.8%) even for large size seed. It planted maize seed at the rate of 67kg/ha with a germination rate of 96.88%, seed spacing of 0.987 and seed distribution pattern of 98.7%. It maintained constant working depth depending on the setting of the furrow opener. The effective and theoretical field capacities obtained are 0.131ha/hr and 0.154ha/hr respectively and a field efficiency of 84.44%. The designed power of the machine is 146W while the operating power obtained is 136W which indicates that it can be conveniently operated by an average person. When compared with manual sowing the machine is six times faster. The wheel skid of the machine was low (0.2%) and the weight is light (18.5kg). It can be drawn by animal by means of a hook provided. It requires less maintenance (periodic greasing of chain and tightening of nuts and bolts) and is easy to operate by farmers. It can be used to sow the most common types of cereals cultivated in the upland areas of North Western Nigeria where the soil is predominantly sandy (75% to 92% sand). It can be used to substitute the traditional planting method. The values shown above indicated higher performance when compared to Dighi's machine which produced 23% efficiency and seed rate of 17.62kg/ha and Nwigwe's modification which produced 64.4% efficiency and seed rate of 73.89kg/ha.

Recast, Pls.

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DESIGN NOTATIONS

Dsr = Distance between center of holes on the circumference of feed mechanism. [mm]

Ps = Required plant spacing. [mm]

Sr = Distance moved by the machine in one rotation of feed mechanism. [mm]

L' = Diameter of hole on the surface of feed mechanism. [mm]

Si = Distance moved by the machine in one rotation of the ground wheel. [mm]

L = Approximate length of chain. [mm]

C = Exact distance between center of sprockets. [mm]

N = Number of teeth on sprocket.

V = Number of rotations made by sprocket.

P = Static pressure in the orifice produced by seed mass. [N/mm^2]

F = Cross sectional area of the orifice. [mm^2]

F' = Cross sectional area of seed stream. [mm^2]

G' = Quantity of seeds pouring out per unit time.

W = Critical speed. [m/sec]

Dk = Diameter of driving wheel. [mm]

ι = Total drive transmission ratio.

γ = Bulk density of seed. [g/mm^3]

D = Outer-most diameter of feed roll. [mm]

d = Inner most diameter of feed roll. [mm]

L" = Thickness of feed roll. [mm]

Z = Number of holes on the feed roll.

Qha = Quantity of seeds in kg/ha.

do = Diameter of the orifice on the bottom plate of the hopper where the grains pour out. [mm]

d^1o = Diameter of the orifice as the pains pour out. [mm]

G = Quantity of seeds poured out in a unit time. [g]

$\theta =$ Angle of inclination of the pouring planes near the orifice. [°]

$\delta_i =$ Angle of inclination of front and rear walls of the grain box. [°]

$\delta =$ Index of flow.

$ę=$ Angle of internal friction of grain. [°]

$d^1 =$ Contraction of the diameter and is always smaller than the orifice's diameter. [mm]

$g =$ Gravitational acceleration.[m/s^2]

$a =$ Distance between center of furrow openers, where applicable. [mm]

$b =$ Distance between one side of box's wall and wheel. [mm]

$z =$ Number of furrow openers.

Pls, use 12 fonts and double spacing.

CHAPTER ONE

1.0 INTRODUCTION

In order to obtain the highest possible yield of crop in a farm culture it is essential to ensure appropriate spacing of seeds in the soil. Optimum distances between individual seeds or seed groups (hills) and optimum depth of seeds to be placed down under soil surface is a desirable factor and a pre-requisite of good crop yield. Apart from proper land preparation, adequate fertilizer application and conducive environmental conditions, plant population and intra-row spacing play a vital role in the overall yield per unit area. For this reason, the use of planters becomes very important. In view of this mechanical placing and spacing of seeds by means of suitable sowing machine has been explored in this study (Bernacki et al, 1978; Yisa and 1dah, 1999; Corrie, 1995).

Sowing is the process of placing grains in rows at uniform rate, depth and interval and covers them with soil layer and application of appropriate pressure on the soil layer.

Planting of seeds of various crops in Nigeria are usually done manually owing to the small size of farms (small-scale farming) in the country (James, 2002). Majority of crops grown in the country are from peasant farmers whose land holdings are between 0.1 to 1 hectare except in some few mechanized farms where tractors and planters are used (James, 2002). Most of the peasant farmers use the traditional method of planting which is associated with fatigue and drudgery. The manual method of planting is uneconomical and in accurate in placement of seeds. It is laborious, time consuming and makes mechanization difficult.

There exist several types of seed sowing machines (drills and planters) in the open market for example KUHN seed drills, ALMACO seed drills, M.F seed drills, John Deere seed drills, Bamlet and Fionner seed drills, but their designs in most cases do not fit into local conditions (Yisa and Idah, 1999, Obioha, 1998; James, 2002).

Many problems are associated with the indiscriminate introduction into developing countries of the technology evolved in Europe and North America, since the technical, economic and social conditions are entirely different. For the same reasons, the assumption that an intermediate level of technology is required, leading eventually to the application of the same higher technology, could be regarded as less than ideal. It is felt that a more promising approach is the development of an "appropriate" technology, in which the most suitable aspects of all available expertise are logically applied to produce a satisfactory solution to the problems which are unique to developing countries (Kilgour and Crosslay 1978).

Every country and or locality has its own crops, level of technology and needs. Therefore solution to design problems lies in designing and constructing a machine that satisfy local requirement ((Yisa and Idah, 1999). Sometimes it is necessary to re-design or modify imported equipment in order to adapt them to local conditions. The rapid development of engine powered machinery sometimes raise questions as to whether or not powered farm machine is necessary especially where the use of such equipment do not pay for itself. The use of farm machinery can only be profitable where the output of the farming system justifies the cost for its purchase, operation, repairs and depreciation (Yisa and Idah, 1999).

In most cases, the parts of the entire machines are imported, a practice which hinders maintenance of these machines. A part from the large mechanized farms which use these imported machines almost all the crops that are planted in this country are being carried out manually (Yisa, and Idah, 1999; James, 2002). This calls for improvement since the operation of planting involves a lot of drudgery.

As a result of high demand for grains, a lot of research works in the area of design and fabrication have been done on grain planting machines but these are mostly powered by engines (Obioha, 1998; Yisa and Idah, 1999).

Some locally existing planters that were examined are.

i. Anurages single row multi-purpose planter
ii. Hassan's single row planter
iii. Nwigwe's Manually operated seed Drill
iv. Joshua's Jab planter (James, 2002).
v. Dighi's single row rice drilling machine
vi. Yisa and Idah's' single row seed drilling machine (Yisa and Idah, 1999)
vii. IITA Two Row No Till Planter
viii. Rolling injector planter

The above listed planters were locally developed. However, the problems with them range from lack of versatility in grain sowing operation, low qualities and efficiencies as a result of inappropriate construction materials, high power demand from the operator, unadaptability to varied local condition, lack of acceptability by the farmers, low durability and high costs.

1.1 Methods of Grain Planting in Nigeria

In this country planting of crops is done on ridges or flat ground but ridge planting is preferable for most grain crops (Akinsanmi, 1981; James, 2002). There are two methods used in row planting by Nigerian farmers. They are:

1.1.1 Traditional Planting Method:-

This involves the use of simple tools such as hoes, cutlasses and sticks. These tools are used for opening and covering the soil after the seeds have been poured into the dug holes. But the most common method is the use of foot to cover the grains with a layer of soil some farmers use hand broadcasting (scattering the seed on the land surface by the use of bare hands without covering them, for example rice grains) (James, 2002).

1.1.2 Mechanical Method of Planting

This involves the use of machines (engine powered or manually operated types) to plant grains at required spacing and quantities. These machines are used to:

i. Open soil to the proper depth for the seeds

ii. Metering of seeds from the grain box.

iii. Placement of seed in the open furrow in an acceptable pattern.

iv. Covering of the seeds with soil layer.

v. Firming the soil around the seeds to the proper degree for the particular crop involved. The planters perform these operations in such a manner that all the factors affecting germination and emergence of seeds will be as favorable as possible. For effective operation, a planter should perform these functions accurately at fairly high rates of speed (James, 2002)

1.2 Factors Affecting yield of cultivated crops

Yield of cultivated crops is influenced by the following factors such as plant population per unit area, spacing, types of crop, soil and its level of fertility, available moisture, methods of weed control, cultivation and harvesting and protection against diseases and pests (James, 2002).

1.3 Statement of the Problem

Most of the farmers in Nigeria are small scale holders of small fragmented land areas which make mechanization difficult, and the big complex and costly imported machines unsuitable since the farmers lack the skills and wealth to own, operate and maintain them. In this study, solutions to the above problems were sought by designing and constructing a grains sowing machine that can be used to sow different crops at the required spacing and quantities

1.4 Aims and Objectives

The general objective of this study is to design and construct a grains sowing machine that can be used to sow the most common types of crops (maize, millet, guinea corn, cowpea, upland rice and groundnuts) grown in Nigeria. It can be used to substitute the use of traditional planting method where local tools and hand broad casting have been in use for centuries.

The specific objectives are:

i. To design and construct a manually operated grains sowing machine that can work on ridges and flat ground.

ii. To construct different seed metering devices depending on the type of crop required to be sown,

iii. To evaluate the performance of the machine using maize, millet, guinea corn, cowpea upland rice and groundnuts.

1.5 Justification

Mechanization of agriculture is a costly and capital intensive operation. This is more so with present value of the Naira. Thus, improvising farm implements that are acceptable, affordable, appropriate and durable is a universal wish of all modern societies. There is human desire to ease the process of sowing crops grown in Nigeria by using simple locally made planting implement to avoid the traditional planting method of broadcasting and the use of hand tools.

Planting, as a seasonal operation has only a short period of time available in which to perform the task. In view of this, there is need to help the farmers in making best use of the available time to accomplish this operation. This is the reason why this sowing machine was designed with high components of locally source materials in order to reduce cost, human labour and drudgery associated with traditional planting system in

Nigeria. The machine can sow different grains at the required spacing and quantities depending on the type of crop being sown. The purpose is to assist farmers to meet different planting requirement of crops, save time and cost, and increase production, and to replace the use of tools (hoes, cutlass, and sticks) and foot in planting operation.

1.6 Limitation

1. This machine is limited to use in upland areas of North Western Nigeria and similar conditions where the soil is predominantly sandy (75% to 92%, 1% t0 11% silt, and 4% to 20% clay) (Graham, 2004).
2. the machine should not be used in mud and water logged conditions because it will be hard to push and wet clay can stick to the working components (wheels, furrow opener and covering device) which results in poor performance and low efficiency (Bernack, 1978).
3. The machine should not be used in areas where large stones, stumps and roots are prevalent because this will result in unsteady movement of the machine and the hooking and or damaging of the furrow opener and covering devices.
4. It is limited to planting of millet, guinea corn, cowpea, maize, upland rice and groundnuts.

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1 History of Planter Development

The idea to sow with the aid of machine dates as far back as the times of Greeks and Romans. Persians and Hindus, according to old chronicles, are said to have avail themselves of seeding machines (Bernacki et al, 1972). However, history of grain drills can be traced back to primitive Assyrian designs as early as 680BC. (Brunt, 2001). In Europe, however, sowing was done by hand (broadcasting), up to the end of the 17th century and even later in all European countries and up to today in many developing countries like Nigeria.

The first European drill was developed in 1639 by Joseph Localtelli of Corinth. The machine was named "sembradore" by the designer (Bernacki et al, 1972). It had a cylindrical wooden tank, inside which a shaft with spoons rotated throwing seeds through holes into sagging tubes falling short of soil surface. The "sembrandore" could not deposit seeds in the ground but laid them only on rows on the surface. Nonetheless, in comparison with manual sowing, seed location was less confused (Bernacki et al, 1972).

At the end of the 17th century, Locatellis drill was improved by Jentro Tull, an English man who considered the advantages offered by mechanical sowing on a more carefully prepared soil. In 1785 James Cook designed a seeding machine, the principal idea of which has survived to our times. The machine was extensively used in Great Britain (Bernacki et-al 1972).

In 1804. Ducket, a German designed a seeding machine consisting of two separate parts independent of each other. One part, a six row marker drawn by a horse forming furrows into which the seeds are sown by proper three rows manually pushed drill (Bernacki et-al 1972). Seeding was done by three grooved shafts placed at the bottom of the seed box and mounted on ground wheel axle. Compared with Cook's drill, the double compartment machine of Ducket was primitive. It was not until this drill was improved upon by Alban of Austria, that it found broader uses (Bernacki et al, 1972).

In 1844 Garret built a seeding machine with a distributing appliance consisting of a series of discs mounted on common shaft. Fixed to each disc on both sides were spoons on arms to remove seeds from the box and throw them into tubes connected with furrow opener

s. This arrangement has the advantage that the drill was highly sensitive to shock and inclination when worked on slopes (Bernacki et-al 1972).

In the United States of America, the first cotton and corn planters were invented in 1825 and 1940 respectively (James, 2002). The first automatic check row sower was tried in 1875 and the single kernel cumulative drop planter in 1890. In 1900, fertilizer attachment were introduced, and in 1923 tractor mounted planters were developed (James 2002). The use of tool bar made it possible to adjust planters to any row width. The automatic check sower was invented in 1975 and the Dowlaw cotton planter in 1970 (Obioha, 1989; James, 2002).

2.2 Characteristics of Seeding Machines

Seeding machines are mainly distinguished by the mechanical devices the drills are opened, seed fed, and the drills reclosed upon the seed. Of these the feeding device is the most essential feature, and this usually involves either means for varying the quantity of seed fed by varying the escape-openings, or by positive mechanical movements variable in speed. The principal requirement are capability of distributing seed with a continuous and regular discharge from each distributor or grown tube; accuracy in quantity of seed discharged; efficiency in regulating the same under all circumstances on inclined, level, or irregular land; changeability of feed apparatus to suit coarse or fine seed, and facility of adjustment (James, 1999).

Grain drills give more uniform seeding rate, and bury the seed at relatively constant depth allowing use of less seed. Many drills are also set up to apply fertilizer in the same pass over the field. Drill technology has not changed a great deal since the original horse-drawn units appeared in 1800s (Bokklen 2003).

Agronomist define seed rate needed by the number of plants per unit area and the field emergence, thus by the number of seeds per unit area. Instead of this with grain drills the farmer try to adjust the seed mass per unit area in kg/ha based on the crop being sown. In doing this, they (farmers) attempt to attain the number of seeds per unit area needed. Yet they have to cope with deficiencies, which arise from the varying wheel slip due to firm-or soft soil as well as dry-or moist soil. Deficiencies can be due to the varying bulk density of the seeds, since grain drills are designed for bulk or volume

CHAPTER THREE

3.0 MATERIALS AND METHODS

3.1 Design Consideration

Basic engineering design considerations such as power requirement, machine capacity, suitability, adaptability and safety were combined with functional requirement and cost in order to achieve the desired objective of the study.

3.2 Determination of Plant Spacing in Relation to the Rotation of Feed Mechanism and the Travel Distance of the Machine

The table below shows the various grains considered in this study with their respective spacing of intra-row and inter-row (Akinsanmi, 1975)

Table 1: Types of grain crops with their respective quantities per hole and spacings (Akinsanmi, 1975)

S/No.	Type of crop	Number of seeds per hole	Intra-row spacing (cm)	Inter-row spacings (cm)
1.	Millet	5	50	75
2.	Guinea corn	5	50	75
3.	Cow peas	2	50	75
4.	Ground nuts	1	25	75
5.	Maize	2	25	75
6.	Upland rice	5	30	50

~~Source: Akinsanmi, (1975)~~.

Calculation Procedure

Let:

S = distance moved by the machine in one rotation of the driving wheel. Taking into consideration number of seeds per hole and the spacing shown in table 1 above as well as the size of the machine in view (suitability and compactness of the machine) take S = 50cm.

D = diameter of the feed mechanism. For suitability and compactness in conjunction with the size of the machine in view, assume D = 7.0cm

t = Thickness of the feed mechanism = 2.5cm

ι = Total drive transmission ratio.

To satisfy the above conditions, a driving sprocket of 13 teeth can be used to rotate another (driven) sprocket of 39 teeth on whose shaft is attached the seed metering mechanism. This will give a drive ratio of 3:1.

DK = diameter of the driving wheel= circumference of the driving wheel divided by pie. That is

$$Dk = Cir/\pi = S/\pi \quad \text{...(3.1)}$$

$$= 50/\pi$$

$$= 15.91549431 \text{ cm}$$

$$= 20\text{cm}$$

Hence from the relationships stated above the driving wheel on whose shaft is attached the driving sprocket, will rotate three times for the driven sprocket and seed metering device to rotate once. At this instance the machine must have moved a distance of 50 x 3 = 150cm in one rotation of the driven sprocket and feed mechanism. This is because the drive ratio (ι) is 3:1, and the circumference of the driving wheel is 50cm.

Ns = number of holes to be drilled on the surface of the feed mechanism.

Dsr = distance between center of holes on the surface of the feed mechanism.

Since the diameter of the roll is 7.0cm (D = 7.0cm) the circumference will be

$$Cir = \pi D$$

$$= \pi \times 7.0$$

$$= 21.99$$

$$\cong 22 \text{ cm}$$

Using the data above, the following procedure can be used in designing and calculating plant spacing in relation to the rotation of the seed metering mechanism and the travel distance of the machine.

I - Millet crops (Setaria italica): Ex Borno

It is required to plant millet grains at the rate of 5 seeds per hole at intervals of 50cm between holes in a row.

Thus

NS (no. of holes to be drilled on the surface of the feed mechanism) = Distance moved by the machine in one rotation of the feed mechanism divided by required plant spacing.

between holes in a row:

$$NS = 150/50$$
$$= 3$$
$$= 3 \text{ holes}$$

Dsr = 22/3

= 7.3cm

Size of hole = 13mm (MASDAR 1978)

Depth of hole = 7mm (MASDAR 1978)

3. Cowpea (Vigna unguculata)

To plant erect bean at 2 seeds per hole at intervals of 30cm between holes in a row.

NS = 150/30

= 5/holes

Dsr= 22/5

= 4.4cm

Size of hole = 13mm (MASDAR 1978)

Depth of hole = 7mm (MASDAR 1978)

4 Groundnuts (Arachis hypogaea): runnerT37

To plant groundnuts at 1 seed per hole at internals of 25cm between holes in a row:

Ns = 150/25

= 6.0

= 6 holes

Dsr = 22/6

= 3.70cm

size of hole = 16 mm (MASDAR 1978)

Depth of hole = 8mm (MASDAR 1978)

5. Rice (Oryza sativa):

To plant upland rice at 5 seeds per hole at intervals of 30cm between holes in a row:

Nsr = 150/30

= 5/holes

Dsr= 22/5

= 4.4cm

Size of hole = 3mm (MASDAR 1978)

Depth of hole = 2mm (MASDAR 1978)

6. Maize (Zea mays): Mexico5

a – To plant maize at 2 seeds per hole at interval of 25cm between holes in a row

Ns = 150/25

= 6 holes

Dsr = 22/6

= 3.667

≅ 3.7 mm

Size of hole = 15mm (MASDAR 1978)

Depth of hole = 8mm (MASDAR 1978)

3.3 Design of chain drive for the machine

The chain drive for the machine in view consists of driving and driven sprockets and a power chain which runs over the sprockets and meshes with their teeth. This arrangement inco-operates a tensioning device for the chain.

The velocity of the chain is calculated by using the following expression.

V = ZnP/ 60x1000 (Reshetov, 1978) ....(3.3)

Where

Z = number of teeth on the sprocket

n = speed of rotation of the sprocket (r.pm)

p = chain pitch (mm)

P = V x 60 x 1000/Zn, from 3.3

V = 0.75m/s (Obioha, 1998)

Let D_K = diameter of driving wheel

= 50cm (0.5m)

V = Speed of rotation

= 0.75m/s

Then in one minute, the wheel will rotate 0.75 x 60 = 45r.p.m.

Assume: Z_2 = 39teeth

Z_1 = 13teeth

Therefore, the chain pitch is calculated as follows:

$P = 0.75 \times 60 \times 1000/39 \times 45$

$= 12.70mm$

or $P = 0.75 \times 60 \times 1000/13 \times 135$

$= 25.64mm$

The speed ratio of the drive is calculated from the condition of equality of the average velocities of the chain on the sprockets and is given by the following expression.

$Z_1\ n_1p = Z_2\ n_z\ p$ (Reshetov, 1978)..............................(3.4)

i.e (13) (135) (25.64) = (39) (45) (25.64)

44998.2 = 44998.2

3.3.1 Determination of Speed ratio of the drive

The speed ratio (ratio of speed of rotation of the high to the low speed sprocket) is given by the following expression

$U = n_1/n_2 = Z_2/Z_1$ (Reshetov, 1978)..............................(3.5)

i.e. 135/45 = 39/13

= 3

This implies that the speed ratio of the driving and driven sprockets is 3:1, which means that the driving (small) sprocket will rotate three times for the driven sprocket to rotate once.

The speed ratio (U) is limited by the allowable overall size of the drive, angle of contact area of meshing of the chain on the smaller sprocket and the number of teeth. The condition $U \leq 10$ is satisfied for a low speed drive with sufficient space.

3.3.2 Calculation of angle of links turn

The angle through which the links turn as they run onto and off the sprocket is given by the following expression.

$\theta_1 = 360/Z_1 = 360/13 = 27.69^0$ (Reshetov, 1978)..................(3.6)

$\theta_2 = 360/Z_2 = 360/39 = 9.32^0$ (Reshetov, 1978)...................(3.7)

The minimum allowable number of teeth on sprockets in power drive with roller chain is Z_1min = 13 to 15 for low speeds (Reshetov, 1978). This machine operates at low speed of 0.75m/s average and therefore a small sprocket of 13 teeth was selected. The reason behind this is that the un-even number of teeth of the small sprocket selected, which is in combination with the even number of chain links facilitates more uniform wear

3.3.3 Calculation of Distance between the Sprockets' Axes and Chain Length

The minimum center-to-center distance (a min {mm}) is determined from the condition that the angle of contact of the chain with small sprocket should be at least 120^0 (Reshetov, 1978). Thus at $U \leq 3$,

$$a\ min = D_1 + D2/2 + (30\ to\ 50) \qquad (3.8)$$

Where D_1 = Outside diameter of small sprocket

D_2 = outside diameter of large sprocket

This gives $a\ min = P\ (30\ to\ 50)$ ....(3.9)

$= P \times 30$

$= 25.64 \times 30$

$= 769.2\ mm$

The required number of links (W) of the chain is determined from the tentatively selected center to-center distance (a), pitch (P) and the number of teeth Z_1 and Z_2 of the sprockets.

$$W = \frac{Z_1 + Z_2}{2} + \frac{2a}{P} + \frac{(Z_2 - Z_1)^2}{2\pi} (P/a) \text{ (Reshetov, 1978)} \qquad (3.10)$$

Substituting,

$W = (13+39/2) + (2 \times 769.2/25.64) + (39-13/2\pi)^2\ (25.64/769.2)$

$= 26 + 60 + (17.12328004)\ (0.0333333)$

$= 26 + 60 + 0.57077600$

$\cong 90$ Links

The distance between the sprockets axes is finally determined (without taking sag of the chain into account) by the following formula.

$a = P/4\ \{W - Z_1 + Z_2 + \sqrt{(W-Z_1+Z_2)^2 - 8(Z_2-Z_1/2\pi)^2}\}$

(Reshetov, 1978)..(3.11)

Substituting the values we obtained

$a = 25.64/4\ \{87 - (13+39)/2 + \sqrt{87-(13+39)/2) - 8\ (39-13)/2\pi)^2}\}$

$= 25.64/4\ \{61 + \sqrt{37210 - 136.9862403})$

$= 25.64/4 \{61 + 59.8666311)$

$= 25.64/4 (120.8666331)$

$= (25.64/4) (120.8666331)$

$= 774.76mm$

In order to provide a small sag allowing the chain links to take the best position on the sprocket teeth, the center to-center distance is reduced by (0.003)a (Reshetov, 1978)

i.e

$a = (774.76 - (0.003 \times 774.76)$

$= 774.76 - 2.32420$

$\cong 772.44mm$

But for minimum allowable load for a given pitch, the following condition must be satisfied:

$a/80 \leq P \leq a/25$ (Reshetov, 1978)..(3.12)

i.e $\quad 9.656 \leq 25.64 \leq 31$

The condition is satisfied. Note that 772.44 is the minimum center to-center distance of the sprockets.

3.3.4 **Determination of Load Carrying Capacity of the Chain Drive**

The principal criterion of the chain drive performance (wear resistance of the chain joints) is that the pressure in the joint must not exceed the allowable value.

The power function between the pressure (δ) and the path (S) of friction is given by the following formula:

δ^m S. = C (Reshetove, 1978)...(3.13)

Where C = Constant – which depends on pressure and friction of drive.

Hence $\delta = c / \sqrt[m]{s}$

Where:

m = an exponent which depends on the type of friction. m = 1.5 for medium lubrication. But the allowable useful force that can be transmitted by a chain with sliding friction is

$$F = \frac{(\delta)o\,A}{Ks} \quad \text{...(3.14)}$$

Where:

(δ)o = allowable pressure in the joints for medium service condition = 3.15 kgf. (from table 15.4 of page 424) (Reshetove, 1978)

A = area of projection of the pin joints, for medium service condition (mm^2).

A = B X d

Where B = the roller link width = 2.4 mm

d = the pin diameter = 3.6 mm

(From table 15.1 of page 414 (Reshetov 1978)

Ks = Service factor, which is the product of certain partial factors.

Thus Ks = Kil x Ka x Kin x Kad x Klu x Kd

(Reshetov, 1978)...(3.15)

Where:

Kil = impact load factor = 1 for steady load.

Ka = chain length factor = 1 for a = (30 to 50)P:

Kin = drive inclination factor = 1 for inclined angle of up to 60^0 to the horizontal. The inclined angle of 45^0 was used in this design.

Kad = adjustment facilities factor = 1 for drives with adjustment of the axis of one of the sprocket as in the case of this machine.

Klu: = Lubrication factor = 1.5 for intermittent lubrication

Kd = duty factor = 1.25 for two shift operation

therefore Ks = 1 x 1 x 1 x 1.5 x 1.25=1.875. This value is less than 3 which implies that the design will work.

Substituting in equation 3.14 gives

$$F = \frac{3.15 \times 2.4 \times 3.6}{1.875}$$

$$= 14.52\text{kgf}$$

The power that can be transmitted by this drive is calculated by using the following expression (Reshetov, 1978):

$$P_1 = (\delta)_0 \, A.V/102(Ks)(kW) \quad \text{.............................(3.16)}$$

Substituting

$$P_1 = \frac{(3.15)\,(2.4x3.6)\,(0.75)}{(102)\,(1.875)}$$

$= 0.106729411$ kW

$= 106.73x\ 10^{-3}$ kW

3.3.5 **Determination of Sprocket Sizes**

The sprockets were selected in accordance with standard (Reshetov, 1978) which covers wear resistance profiles without seating curve displacement and with displacement for non-reversible drives.

The center of the joint of the chain links meshing the sprocket are located on its pitch diameter (dp), and is given by the following expression (Reshetov, 1978):

$$dp = P/\sin(180/Z) \quad \text{...................................(3.17)}$$

Where: dp = pitch diameter of the sprocket (mm).

P = chain pitch (mm)

Z = number of teeth of the sprocket

For the driving sprocket (Z_1), the pitch diameter is calculated as follows:

$dp = 12.7/\sin(180/13)$

$= 53.06$ mm

For the driven sprocket (Z_2), its pitch diameter is calculated as follows:

$$dp = \frac{12.7}{\text{Sin}\ (180/39)}$$

$= 157.83$ mm

The out side diameter of a sprocket is given by the following expression:

$Do = (P\ \ 0.5 + \text{Cot}\ 180/z)$ (Reshetov, 1978) ....(3.18)

Where: Do = outside diameter of the sprocket (mm)

P = chain pitch (mm)

Z = number of teeth of the sprocket

: - The diameter of the small sprocket is given by:

$D_1 = 12.7\ (0.5 + \text{Cot}\ 180/13)$

$= 59.42$ mm

While the diameter of the large sprocket is

$D^1 = 12.7\ (0.5 + \text{Cot}\ 180/39)$

$= 164.18$mm

In view of the above results a driving sprocket of 60mm outside diameter with a pitch of 12.7mm and 13 teeth could be used for the machine.

Also, a driven sprocket of 160mm outside diameter with a pitch of 12.7mm and 39 teeth could be used for the machine. These two sizes of sprockets are readily available in the market.

3.3.6 Determination of Chain Length

The approximate length of chain in pitches is given by the following formula.

$L = 2a + (N2 + N1)/2 + (N2 - N1)/4\pi^2 a$ (Krutz et al, 1984) ....(3.19)

Where

L = Length of chain in pitches

a = distance between center of sprockets = 400mm

N_1 = number of teeth of small (driving) sprocket

= 13

N_2 = number of teeth of large (driven) sprocket

= 39

Substituting in the above equation, we have

$$L = 2\ (400) + \left[\frac{39+13}{2}\right] + \frac{39-13}{4\pi^2(400)}$$

= 800 + 26 + 0.002

= 826.002 mm

≅ 826 mm

3.3.7 Determination of Fatigue Life of the Chain

The chain operates at low speed and the horse power capacity is determined by the fatigue life of the link plates which is calculated as follows (Obioha 1998)

$HP = 0.004\ N_1^{1.08}\ n_1^{0.9}\ P^{3.0} - 0.007P$........................(3.20)

Where:

N_1 = number of teeth of small sprocket = 13

n_1 = speed of small sprocket = 273 rpm

P = chain pitch = 12.7mm = 1.27cm

Substituting,

$HP = (0.004)(13)^{1.08}(273)^{0.9}(1.27)^{3.0} - 0.007(1.27)$

$= (0.06384\ 3812)(319.1225877) - 0.00889$

$= 20.365kW$

3.3.8. Determination of Design Power

The design power is given by the following equation

$Pd = Pm \times SF1\ SF_2$ (Obioha, 1998)....................................(3.21)

Where:

Pd = design power

P_1 = power transmitted = 0.106729411 kW

SF_1= service factor 1 =1.4 for moderate

Shock

SF_2 = Service factor 2 = 1.4 for atmospheric condition.

One tenth of a horse power is the power that can be developed by an average man working at a speed of 0.6 to 0.9m/s

:- average speed = (0.6+0.9)/2 = 0.75m/s.

Power transmitted = 1/10 x 746 = 74.6W (Obioha 1998)

Therefore Pd = 74.6x 1.4x1.4 = 146.216W.

3.3.9 Determination of Chain Torque

From the first principle, torque (T) is given by

$T = Pd/\omega$; (Obioha, 1998) ....(3.22)

Where T is the torque in Nm, Pd is the design power in kW and ω is the angular velocity in rad/sec.

$\omega = 2\pi n_1/60$ = angular speed of small sprocket

$= 2\pi(273)/60 \cong 28.6$ rad/sec

The torque (T) on small sprocket is

T = 146.216/29

= 5.042 kNm.

3.4.0 Determination of Sizes of Machine Components

3.4.1 Determination of the size of the frame.

An angled bar of equal angles (90^0) is selected for the frame. The properties of structural shape of an angled bar is given as follows (Shigley, 1978).

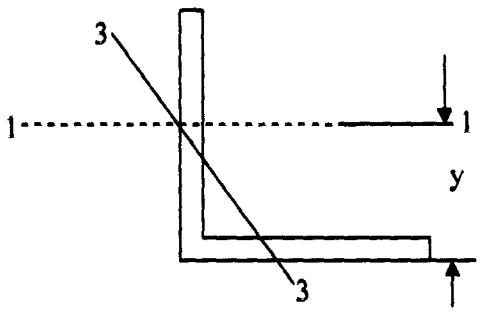

Figure 1. Structural shape of an equal angles mild steel bar.

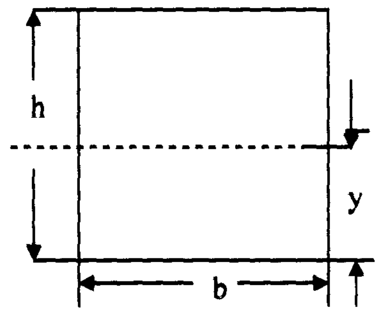

Figure 2. Rectangular section.

The above bar can be taken as two rectangles of equal sizes joined together to form an angle of 90^0. For a rectangular section shown on figure ii above, the following notations are used for the structural properties of the frame.

A = Area, cm^2

$= h \times b$

I = moment of inertia, cm^4

$= bh^2/12$

Z = Section modulus, cm^3

$= bh^2/6$

$\dot{y}$ = Centroidal distance, cm

$= h/2$

K = radius of gyration, cm

$= 0.289h$

Assume: $h = 80cm = 0.8m$

$b = 4cm = 0.4m$

Area $A = 0.8 \times 0.04$

$= 0.032m^2 \times 2 = 0.064m^2$ (for two rectangles of equal sizes)

$I = (0.04 \times 0.83)/12$

$= 0.001706666m^4 \times 2 = 0.003413332m^4$

$Z = bh^3/6$

$= (0.04 \times 0.8^3)/6$

$= 0.003413333m^3 \times 2 = 0.4624m$

$\bar{y} = h/z$

$= 0.8/2$

$= 0.4m.$

The long member of the frame is as shown in figure 1a. For the width of the frame (figure 1b) the following values were obtained:

Assume: $h = 40cm = 0.4m$

$b = 4cm = 0.04m$

$A = b \times h$

$= (0.4 \times 0.04) \times 2$

$= 0.032m^2$

$I = [(0.04 \times 0.43)/12] \times 2$

$= 0.000426666m^4$

$Z = bh^3/6$

$= [(0.04 \times 0.4^3)/6] \times 2$

$= 0.000853333m^3$

$K = 0.289h$

$= 0.289 \times 0.04$

$= 0.01156m$

$\bar{y} = h/2$

$= 0.4/2$

$= 0.2m$

3.4.2 Determination of Hopper Capacity

The grain box (hopper) is made up of steel (1010) sheet of 1mm thickness. The shape of the box is a frustum. The angle of inclination of its walls is 45^0 so that seed materials can slide down wards even if moistened by insecticide chemicals (Herman et al, 1978). The Capacity of the grain box should not be more than 15kg so that too much force will not be required from the operator during work.

The hopper consists of the following parts:

1. The bucket (figure iii e)
2. Bottom plate (figure. iii a)
3. Circular ring (figure.iii b)
4. Cover (lid). (figure. iii d)
5. Hopper guide (figure iii c)

Assume:

h = Vertical height of the bucket = 286mm = 0.286m

D = diameter of top opening = 260mm = 0.26m

d = diameter of bottom opening 119mm = 0.119m

L = slant height = 288mm = 0.288m

a = slanting angle of bucket walls = 45°

3.4.2.1 **Volume of the Box**

The volume of the frustrum box is given by the following formula:

$V = (1/3)\pi h\ (R^2 + Rr + r^2)\ (m^3)$ (Greer and Hancox, 1978).........(3.23)

Where

V = volume (m^3)

h = vertical height of the box = 0.286

R = large radius = $D/2 = 0.2b/2 = 0.13$m

r = small radius = $d/2 = 0.11/2\ 0.0595$m substituting.

$V = (1/3)\pi(0.286)\ \{(0.13)^2 + (0.13)\ (0.0595) + (0.0595)^2\}$

$= 0.2995\ (0.0281750)$

$\cong 0.00844m^3$

3.4.2.2 **The Curved Surface Area of the Box**

The curved surface area is calculated as follows:

$Ac = \pi(R + r)$(Greer and Hancox, 1978)..........................(3.24)

$= \pi(0.13 + 0.0595)\ (0.288)$

$= 0.1715m^2$

3.4.2.3 **Total Surface Area**

Total surface area given is by the following formula:

$At = \pi(R + r)\ L + \pi R^2 + \pi r^2$ (Greer and Hancox, 1978)...............(3.25)

$= 0.1715 + \pi(0.13)^2 + \pi(0.0595)^2$

$= 0.2357m^2$

3.4.2.4 The Bottom Plate

The bottom of the hopper is covered with a circular plate (figure 3a). For a circular section, the following notations were used to obtain the structural shape (Shigley, 1978).

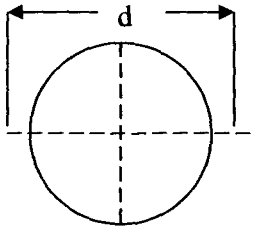

Figure 3. Circular Section

$A = \pi d^2/4$, m^2

Where d = diameter of the plate

$I = \pi d^2/64$, m^4

$Z = \pi d^3/32$, m^3

$J = \pi d^4/32$, m^4

Where J = Polar moment of inertia

$K = d/4$

$\bar{y} = d/2$

Take d = 160mm = 0.16m

The area $A = \pi\ (0.16)^2/4$

$= 0.02m^2$

$I = \pi\ (0.16)^4/64$

$= 3.2169 \times 10.5m^4$

The bottom plate is as shown on figure iii (a) in appendix A

3.4.2.5 Circular Ring

The bottom plate is fixed to a circular ring (hollow circle). And for a hollow circular section, the following notations are used to obtain its structural shape and size (shigley, 1978)

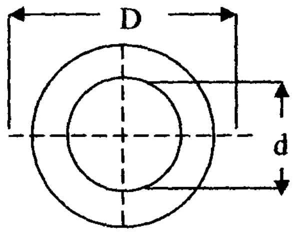

Figure 4: Circular ring

The area $A = \pi/4\ (D^2 - d^2)$

Assume: $D = 160mm = 0.16m$

$d = 120mm = 0.12m$

Therefore, $A = \pi/4\ (0.16^2 - 0.12^2)$

$= 0.0088m^2$

The moment of inertia (I)

$I = \pi/64\ (D^4 - d^4)$

$= 0.00055m^4$

The section modulus (Z)

$Z = \pi/32d\ (D^4 - d^4)$

$= 0.818123086 \times 0.0112$

$= 0.0092m^3$

The polar moment of inertia (J)

$J = \pi/32\ (D^4 - d^4)$

$= 0.0011^4$

Radius of the gyration (K)

$K = \sqrt{D^2 + d^2/16}$

$= \sqrt{0.16^2 + 0.12^2/16}$

$= 0.05m$

The centroidal distance (y)

$\dot{y} = D/2$

$= 0.16/2$

$= 0.08m$

The circular ring is as shown on figure iii b in appendix A.

The hopper is also shown on figure. xiii No. 3 in appendix B

3.4.6 **Handles**

The handles are to be gripped by hand and use for controlling the machine. For a portion to be gripped by hand, the maximum diameter (grasp) should not exceed 4cm (Smith et-al, 1994). The handles are inform of hollow cylinders whose mass (m) and moment of inertia can be calculated as follows (Shigley, 1978).

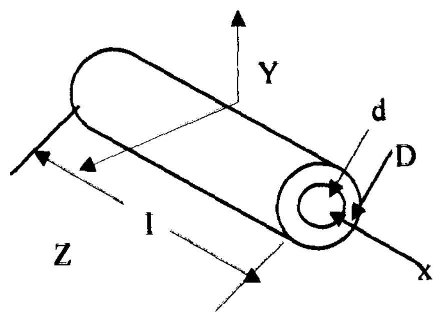

Figure5: Hollow Cylinder

The mass of inertia m $=\pi 6p/4g\ (D^2 - d^2)$

Assume:

L = Length of the pipe = 1050mm = 1.050m.

ρ = density of material (steel 1010)= 7830kg/m^3

d = Outside diameter = 32mm = 0.03m

d = Inside diameter = 28mm = 0.03m

g = Acceleration due to gravity = 9.81m/s^2

$m = \pi\ (1.050)/4(9.81)\ (0.03^2 - 0.028^2)$

0.000009751kg

$I_x = m/8\ (D^2 - d^2)$

$= 1.413895 \times 10^{-10} m^4$

$I_Y = I_z = m/48\ (3D^2 + 3d^3 + 4l^2)$

$= 8.53 \times 10^{-7}$

The sketch of the handles are as shown on figure ii a of appendix A, and figure xii No. 6 in appendix B shows the finished part.

3.4.6.1 **Cross Bar**

The purpose of the cross bar (figure 2b) is to keep the two handles rigidly apart at a distance of not more than 60cm as shown on figure 2c. From ergonomic point of view hand held handles of agricultural and transportation machinery should not be more than 60cm apart (Smith et-al, 1994). The bar can be in the form of rectangular prism whose structural shape, size, mass and mass moment of inertia can be obtained as follows:

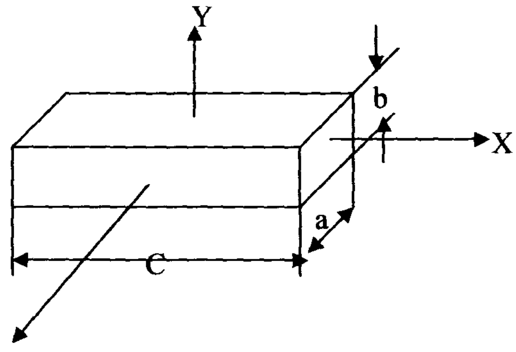

Figure 6: Rectangular prism

The mass (m) is given by the following expression

$M = abc\rho/g$ (Shigley, 1978)

Where a = width of the prism [m]

b = thickness of the prism [m]

c = length of the prism [m]

ρ = density of material [kg/m^3]

g = gravitational acceleration [ms^{-2}]

Assume: a = 40mm = 0.4m

b = 2mm = 0.02m

c = 550mm = 0.55m

$\rho = 7830kg/m^3$ (steel)

$g = 9.81m/s^2$

Therefore, The mass m = (0.4)(0.002)(0.55)(7830)/9.81

$= 3.4452m^2/9.81$

= 0.3512kg

Moment of inertia in x-direction is given by the following formula:

$I_x = m/12(a^2 + b^2)$

$= 0.0293(0.4^2 + 0.002^2)$

$= 0.004683m^4$

$I_y = m/12(a^2 + c^2)$

$= 0.0293(0.4^2 + 0.55^2)$

$= 0.01355m^4$

$I_z = m/12(b^2 + c^2)$

$= 0.0293(0.002^2 + 0.55^2)$

$= 0.0088634m^4$

The cross bar is shown in figure 13 No. 7

3.4.6.2 **Height Adjustment Device**

The handles are equipped with height adjuster so that variable heights of the operating posture can be attained to suit the height of the operator (in ergonomics point of view (Smith et-al, 1994)). The devices can be in form of rectangular prisms whose mass and mass moments of inertia can be obtained as follows:

The mass (m) is given by the following expression:

$m = abc\rho$ (Shilgey, 1978)……………………………………………(3.26)

Where a = Width

b = thickness

c = Length

ρ = density

Assume: a = 10mm = 0.01m

b = 2mm = 0.002m

c = 600mm = 0.0m

$\rho = 77830kg/m^3$

Therefore, m = (0.01)(0.002)(0.60)(7830)

= 0.09396kg.

The moment of inertia in x direction is given by the following expression:

$I_x = m/12\ (a^2 + b^2)$ (Shigley, 1978)…………………………………(3.27)

$= (0.09396/12)(0.01^2 + 0.002^2)$

$= 8.14 \times 10^{-7}m^4$

$I_y = m/12\ (a^2 + c^2)$

$= 281.96 \times 10^{-5} m^4$

$I_z = m/12(b^2 + c^2)$ (Shigley, 1978))..(3.28)

$= 781.06 \times 10^{-7} m^4$

The sketches of the devices are shown in figure 2d of appendix A. The finished members are shown in figure xii No. 8 of appendix B.

3.4.7 **Front Wheels Assembly**

The above assembly consists of the following parts.

1. Circular bars
2. Spokes
3. Strakes
4. Hubs
5. Shaft
6. Bearings

3.4.7.1 **Circular Bars**

These are two in number made up of flat steel of length 1600mm, 45mm wide and 4mm thick for each piece. The diameters of the wheels were selected based on the height of an average size ridge of 150mm. This is to enable the radius of the wheel to accommodate the ridges height and allow some clearance for the furrow opening and covering devices.

The bars (figure iva appendix A) are formed into circles of diameters 510mm each as shown on figure. 4e. This makes the wheels to be inform of hollow cylinders whose masses and mass moment of inertia can be determined as follows:-

For a hollow cylinder, the mass of inertia is given by the following expression:

$m = \pi L\rho/4g\ (D^2 - d^2)$ (Shigley, 1978)....................................(3.29)

Where all the parameter are as defined earlier.

Assume: $L = 45mm = 0.045m$

$D = 510mm = 0.51m$

$d = 510mm - (2 \times 45)mm$

$= 420mm = 0.42m$

$\rho = 7830 kg/m^3$

The mass of inertia is

$m = \pi\ (0.045)(7830)/(4)(9.81)$

$\cong 28.21kg$

$I_x = m/8\ (D^2 + d^2)$

$\cong 28.21/8\ (0.51^2 + 0.42^2)$

$1.5393m^4$

$I_y = I_z = m/48\ (3D^2 + 3d^2)$

$= 28.21/48$

$\cong 0.7696m^4$

3.4.7.2 **Spokes**

The purpose of the spokes is to join the hub to the inner circumference of wheel. This circumference is given by the formula

$$Cir = \pi dm^2$$

Where d = inside diameter of the wheel.

= 0.42m (as above).

Therefore, $Cir = \pi(0.42)$

$= 1.319468915m$

If there or six (6) spokes for each wheel in order to balance the weight property then the distance between each spoke on a wheel is:

Distance between spokes = 1.350884841/6

$\cong 0.225m$

The spokes can be inform of rod whose mass and mass moment of inertia are expressed by the following equations

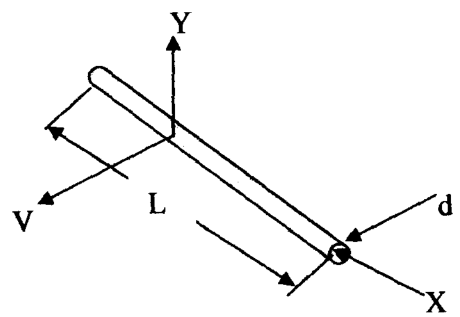

Figure 7. Rod

The mass (m) is given by the following equation:

$m = \pi d^2 lp/4g$ (Shigley, 1978; Krutz et al, 1984)

Where all the terms are as defined previously.

Assume d= 10mm = 0.01m

$L = 220mm = 0.22m$

$\rho = 7830kg/m^3$

$g = 9.81m/s^2$

Substituting

$m = [\pi(0.01)^2(0.22)(7830)]/4x9.81$

$\cong 0.01379kg.$

The mass moment of inertia of a rod is given by the formula

$I_y = I_z$

$= ml^2/12$ (Shigley, 1978; Krutz et al, 1984)

Substituting, $I_y = [(0.01379)(0.22)^2]/12$

$= 55.619 \times 10^{-6}m^4$

3.4.7.3 **Strakes**

The purpose of the strake is to provide a good grip between the wheels and the soil surface so as to reduce wheel slip. Strakes can be positioned on the outer circumference of the wheel at distances of between 50m to 120mm apart {Yisa and Idah, 1999]. The circumference of the front wheel is 1602.21mm. To fix the strakes at distances of 11.44cm apart, the number of strakes required per one wheel is given by:

No. of strakes required = Circumference of wheel/strakes interval

=π x diameter of wheel/strakes interval

= π x 0.51m/0.1144

= 14 strakes per wheel.

This implies that the number of strakes required is 14 per one wheel and they are to be placed at intervals of 11.44cm apart along the outer circumference of the front wheel.

The strakes can be inform of rectangular prism whose mass and mass moment of inertia can be obtained by using the following expressions.

The mass (m) is given by the equation:

$m = abc\rho/g$ (Shigley, 1978; Krutz et al, 1984)

Where all the terms are as previously defined.

Assume a = 3mm = 0.003m (Felix, 1998)

b = 15mm = 0.015m

$c = 45mm = 0.045m$

Substituting

$m = (0.003)(0.015)(0.045)(7830)/9.81$

$= 161.63 \times 10^{-5}kg$

$I_x = m/12\ (a^2 + b^2)$ (Shigley, 1978; Krutz et al, 1984)

Substituting

$I_x = 3.78 \times 10m^4$

$I_y = 32.87 \times 10^{-7}m$

$I_z = m/12\ (b^2 + c^2)$(Shigley, 1978; Krutz et al, 1984)

Substituting,

$I_z = m/12(b^2 + c^2)$ (Shigley, 1978; Krutz et al, 1984)

Substituting,

$I_z = 36.36 \times 10^{-7}m^4$

3.4.7.4 The Hubs Should not be more 3 figures

These couplings are capable of transmitting the torque capacity of the shafts and accommodate any misalignment between them. Rigid couplings are used because they are suitable for low speeds and accurately aligned shafts. In view of this flung coupling was selected and used as follows.

Let:

D_{BC} = diameter of the bolt cycle, mm

D_H = diameter of the hub, mm

D_S = diameter of the shaft, mm

M_t = torque capacity, N.m

S_S = allowable shear stress, pa (N/m^2)

S_B = allowable bearing pressure for the bolt or web, pa(N/m^2)

t = thickness of the web, mm.

n = number of effective bolts.

The shear of the web (torque capacity based on the shear of the minimum area at the junction of the hub and web) is given by the following formula:

$M_t = S_S\ (\pi D_H t)\ D_H/2$ (Krutz et al, 1984).

The thickness t is given by the formula

$t = 2M_t/\pi S_S D_H^2$

For bearing load caused by bolt and web, the torque capacity is given by the formula

$M_t = S_B(dt)\, D_{BC}/2\ (n)$ (Krutz et al, 1984)................................(3.30)

$t = 2M_t/S_B d D_{BC} n$ (Krutz et al, 1984)

Take: bore = 50mm

D_{BC} = 125mm

Ultimate strength (steel 1030) = 551.6Mpa

Yield stress intension = 344.7Mpa

Shock and fatigue factor = 1

Stress concentration for a key way = 0.75

Allowable shear stress = 18% of ultimate stress or 30% of the yield point.

Thus $S_S = 0.18(551.6) = 99.3 \times 10^6$pa

$S_S = 0.3(344.7) = 103.4 \times 10^6$pa

The shaft capacity is calculated as follows:

$M_t = \pi\, S_S\, D_S^3/16$ (Krutz et al, 1984)

= 243.72N.m

The bolt diameter is

$M_t = S_S\,(1/4\ \pi d^2)\,(1/2\ D_{BC})n$ (Krutz et al, 1984)

d = 8.0 mm

The front wheel shaft is made of steel rod 25mm diameter. It is 700mm long and machined to 20mm diameter for a length of 110mm at each of its ends. Threads of 2.5 pitch were made at each of the 20mm diameter ends for a length of 70mm while the remaining 40mm is for bearing seats. Two holes of 8mm diameter each were drilled so that the shaft can be bolted to the main frame.

3.4.8 Driven Sprocket

This was selected based on the design calculations of the drive ratio. It has a major diameter of 160mm, minor diameter of 150mm and 39 teeth. This corresponds to sprocket B 428.

3.4.9 Seed Metering Mechanism

This is inform of circular disc with holes drilled on its circumference. It rotates below and partly inside the hopper (grain box) in order to serve as agitator. The number of holes on the feed mechanism determines the spacing and quantity per hole for the particular crop being sown. The discs are made of steel 1030 to resist environmental hazards, bending and torsion due to loading and rotation. The mass of the disc is calculated ass torsion due to loading and rotation. The mass of the disc is calculated as follows

$m = \pi d^2 t \rho / 4g$ (Shigley, 1978; Krutz et al, 1984)

Where m = mass of the disc, kg

d = diameter of the disc, mm

t = thickness of the disc, mm

g = gravitational acceleration m/s^2

ρ=density

The moment of inertia in the x-directions is given by the following equation

$I_x = md^2/8$ (Shigley, 1978; Krutz et al, 1984)

The moment of inertia in y – direction is

$I_y = I_z = md^2/16$ (Shigley, 1978; Krutz et al, 1984)

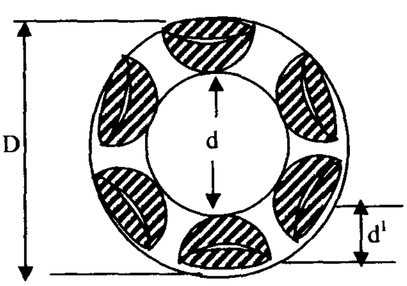

Figure 8: Sketch of seed metering device showing dimensions interest.

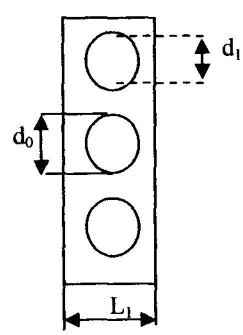

Figure 9: Sketch of seed metering device showing dimensions of interest.

Where:

D=Outside diameter of disc (mm)

d=Inside diameter of disc (mm)

d_0=Outside diameter of hole (mm)

d_1=Inside diameter of hole (mm)

$d^1 = \frac{D-d}{2}$ = depth of hole (mm)

L_1=Thickness of disc (mm)

3.4.9.1 Determination of the Diameter of the Seed Metering Mechanism

The diameter of the feed roll is given by the following expression (Bernacki et al 1978)

$$D=k(\sqrt{Zxax\ Qha/2800\gamma\pi})(1-a)\beta 1n..........(3.31)$$

Where: D=diameter of the feed roll(cm)

K=Co-efficient which takes into account

The width of the slot = 0.9

Z=number of holes on the roll = 6

Qha = seed rate (quantity) = 65kg/hα (maize)

n = number of revolutions of feed roll per one

turn of the driving mechanism = 3

ι = length (thickness) of roll =Assume L=2.5cm for light weight and convenience of the feed roll in relation to the maximum and minimum number and sizes of holes to be made on its surface with regards to the physical dimension of the type of crops to be used for the machine

β = Co – efficient of material feed reduction = 0.7 for maize[Bernakietal,1978]

a = constant co-efficient = 0.35 (for maize) [Bernaki et al,1978]

γ = bulk density of maize seeds = 0.75 kg/cm^3[Bernaki et al,1978]

Substituting

$$D = 90\sqrt{(6)(0.35)(65)/(2800)(0.75)\pi}(1-0.35)(0.7)(0.25)(3)$$

$$= 7.008cm$$

$$\cong 7cm$$

This diameter (7cm) of the roll for all the other crops is the same, but the number and sizes of holes on the circumference of the device are different based on the type of crop, its spacing and quantity per hole. This is necessary, so that each device can fit in the same position below the grain box.

Figures 8 and 9 above show the diagram of the feed mechanism for maize seeds.

Let:

L_l = thickness of the feed roll = 2.5cm

D = outer diameter of the roll = 7.0cm

d = Inner diameter of the roll = 5.4cm

d = D-d = depth of hole = 8mm for maize

L = size of hole = 15mm for maize grains

Z = number of holes on the feed roll = 6 for maize planted 2 seeds at 25 cm interval

S = distance of seed from the perimeter of the feed roll = 5mm for maize due to large dimension of seeds

Vt = volume of seeds fed by one hole

V_T = total volume of seeds delivered by one turn of driving shaft

V_{Ti} = total volume of seeds delivered by one turn of the driven shaft

Dk = diameter of driving wheel

Qha = as above

M = mass of roll

F = Force applied by the roll

F1 = Total area of the roll

$= \{\pi D^2/4 - \pi do^2/4\} = \{\pi(0.07)^2/4 - \pi(0.0165)^2/4\}$

$= 0.003634626m^2$

Vt [volume of seeds feed by one hole]

$= \pi(0.015)^2/4 \times (0.008)$

$= 1.413 \times 10^{-6}m^3$

This is taken as volume of 2 grains of maize.

V_T= total volume of seeds delivered by one turn of the driving shaft $= 1.413\times10^{-6} \times 2$

$= 2.827 \times 10^{-6}m^3$

V_{Ti} = total volume of seeds delivered by one turn of the driven shaft = $1.413 \times 6\times10^{-6}$

$VT\iota = 6.858 \times 10^{-6}\ m^3$

This is equivalent to the volume of 2 x 6 = 12 grains of maize.

DK = diameter of driving wheel = 0.159m.

ι = total drive transmission ratio

γ = bulk density of maize seeds = 0.75kg/m^3

Qha = quantity in kg/ha = 65

M = Mass of roll = (volume of roll – volume of 6 holes) x density of the material.

$= \{0.00364 - (6 \times 1.413 \times 10^{-6})8\}\ (7830)$

2.8kg

Force applied by the roll

F = Mass x acceleration due to gravity

$= 2.8 \times 9.81$

$\cong 27.5N.$

A piece of steel pipe of the same size as that shown in figure. 6b of appendix A which is also shown on figure. 6e is joined to the center hole of the feed roll so that the driven shaft can be inserted in to the hole and then keyed to the shaft as shown on figure 6f. The arrangement of the driven shaft is now completed as shown on figure. 6f, and figure 13 No. 15. The feed mechanisms are shown on figure xiii No. 30 of appendix B

3.4.10 Design of Driving Shaft

The stresses at the surface of a solid round shaft subjected to combined loading of bending and torsion are:

$\delta_x = 32m/\pi d^3$ (Shigley 1978) ....(3.32)

$\tau_{xy} = 16T/\pi d^3$..(3.33)

Where δx = bending stress

τ_{xy} = torsional stress

d = shaft diameter

M = bending moment at critical section

T = torsional moment at critical section

The maximum shear stress is

τ maximum $= \sqrt{(\delta_x/2)^2 + \tau^2_{xy}}$ (Shigley, 1978) ....(3.34)

$= 16/\pi\, d^3 \sqrt{(M^2+T^2)}$

of allowed.

i.e

NS = S/Ps ....(3.2)

= 150/50= 3 holes.

Dsr (distance between center of holes on surface of the feed mechanism) = circumference of the feed mechanism divided by number of holes on the surface of the feed mechanism.

i.e

$$Dsr = \frac{\text{Cir. of the feed mechanism}}{\text{No of holes on the feed mechanism}}$$

= 22/3

= 7.3 cm

The sizes and depths of holes used in this section are from the 1978 publication of the Management of Agricultural Services, Development and Research (MASDAR). This is because the values are found to be in agreement with the ones obtained in this study in relation to the measurement of physical dimensions of the crops used for the machine.

Size of hole = 5mm (Management of Agricultural Services,
Development and Research (MASDAR))

Depth of hole = 3mm (MASDAR, 1978)

2. Guinea corn (Sorghum bicolor): White, YG5670

To plant guinea corn at 5 seeds per hole at intervals of 50cm between holes in a row

NS = S/PS

= 150/50

= 3 holes

Dsr = 22/3

= 7.33 cm

Size of hole = 6mm (MASDAR 1978)

Depth of hole = 7mm (MASDAR 1978)

3. Cowpea (Vigna sinensis): Kano white

To plant beans (local type) at 2 seeds per hole at intervals of 50cm

The maximum shear stress of static failure is

$S_{xy}\ S_y/2$. ....(3.35)

where S_y = ultimate tensile stress

S_{xy} = maximum shear stress

By employing a factor of safety n, the equation becomes

$S_y/2n = 16/\pi d^3 \sqrt{M^2 + T^2}$

Giving

$d = [(32n/\pi S_y)(M^2 + T^2)^{1/2}]$ (Shigley 1978)

From the distortion energy theory,

$d = [32n/\pi S_y (M^2 + 3T^2/4)^{1/2}]^{1/3}$

For reversed bending and steady torsion,

$\delta_a = 32m/3d^3$; τ maximum $= 16T/\pi d^3$

Also, $\delta_a = S_e/n$

Where:

S_e = endurance limit

n = safety factor

Giving $d = (32Mn/\pi S_e)^{1/3}$.This shaft is required to resist tortional and bending stresses under the toughest working condition. In view of this, the material (Steel No: 1010) was selected for use as the shaft materials.

The shaft is forced through the two pieces of pipes attached to the sides of the driving wheel and by means of the circular plates as shown on figure. 5ι.of appendix A. The driving sprocket is fixed to the driving shaft as shown. This implies that the revolution-per-minute (rpm) will transmit the same torque on the shaft. If Ts is the torque on the shaft, then

Angle turned by the shaft in one revolution =

2π radians.

Angle turned by the shaft in n revolutions = $2\pi n$

Where n = number of revolutions/minute

Therefore the work done by the shaft per minute will be:

Work done per minute = Ts x $2\pi n$ kg m.

The power transmitted by the shaft is then:

Pm = Work done per minute/4500

But 0.1kN = 102 kg m/sec.

= 102 x 60 kg m/min

:- $T_s = P_m \times 102 \times 60/2\pi n$

$= P_d \times 102 \times 60/2\pi n$

$= 146.216 \times 10^{-3} kw \times 102 \times 60/2\pi 239$

$\cong 0.6000$ Nm

Vertical loading

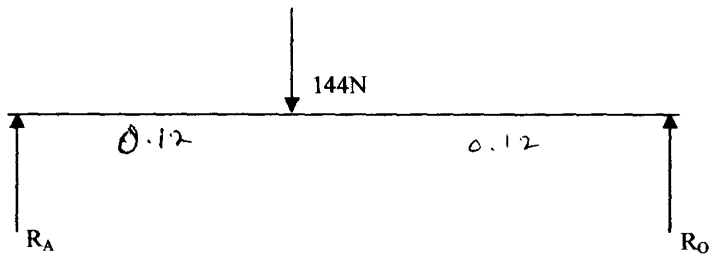

Figure 10:

$\Sigma F_x = 0$

$\Sigma F_x = 0$

$\Sigma M_A = 0$

Taking moment about R_A

$144 \times 0.12 = RB \times 0.24$

$17.28 = 0.24RB$

$RB = 17.28/0.24 = 72N$

For $0 < x < 0.12$

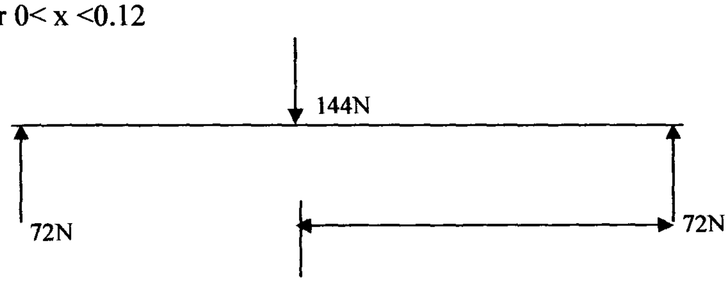

Figure 11. Vertical Loading

$\Sigma F_y = R_A(0.12)$

$\Sigma M_A = R_A(0.12)$

$= 72X0.12$

$= 8.64Nm$

$= 0.12 < x < 0.24$

$\Sigma\ Fy = 72\text{-}144$

$= 72N$

$\Sigma\ M_A = (72x0.24) - (144x0.12)$

$= 17.28 - 17.28$

$= 0$

Horizontal Loading

The horizontal force acting on the shaft is in the x – direction component of the workload and is calculated as follows:

$F = 1036 \text{ Cos } \theta$

$= 1036 \text{ Cos } 36$

$= 838.\ 14N$

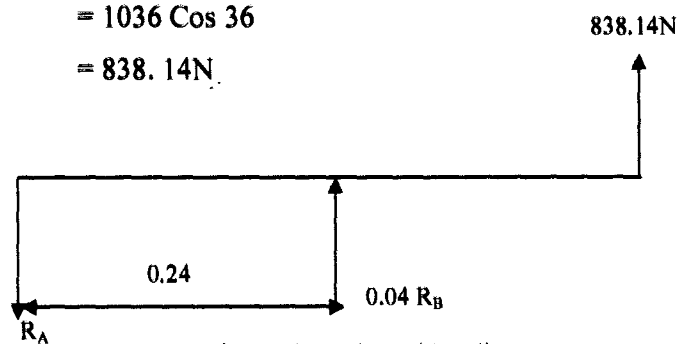

Figure 12: Horizontal Loading

$\Sigma\ Fx = R_A + R_B$

$= 838.14N$

$\Sigma\ m = (R_B x\ 0.24) - (838.14x0.28) = 0$

$0.24R_B - 234.68 = 0$

Therefore $R_B = 234.68/0.24$

$= 977.83N$

But $R_A + R_B = 838.14N$

:- $R_A = 838.14 - R_B$

$= 838.14\ 977.83$

$= -\ 139.69N$

$0<x<0.24$

$\Sigma\ M_B(0.24) = -\ (R_A)\ (0.24) + (0.04)\ (R_B)$

$= -\ (139.69x0.24) + (0.04x977.83)$

$= 9$ (rpm of driven sprocket).

3.4.10.3 **Determination of Force of Inertia**

The power of a rotation shaft is given by the following formula:

$P = F\omega r$ ....(3.37)

Where

P = power (kW)

F = Force (N)

ω= angular velocity (rad/sec)

r = radius of sprocket

$P = 146.216W$

$r = 0.03m$

$\omega = 25$ rad/s

:- $F = P/\omega r$

$= 146.216/(25)(0.03)$

$= 195$ N

3.4.10.4 **Determination of Tortional Moment**

The equation of torsion is given by:

$$Mt = \frac{9550 \times P\ (kW)}{N_2}$$

Where:-

Mt = Tortional moment

P = power = 0.146216 kW

N_2= rpm of large sprocket = 90

Therefore $Mt = (9550\ x0.146216)/90$

$= 15.35$ Nm

3.4.10.5 **Maximum Bending Moment Determination**

$Mb(\text{Maximum}) = \Sigma M_A^2 + \Sigma M_B^2$

$= 8.64^2 + (-33.53)^2$

$\cong 12NM$

3.4.10.6 **Determination of Driving Shaft Diameter**

Diameter of the driving shaft is given by the following expression (Yisa and Idah 1999)

$d^3 = 16/\pi Ss \sqrt{(K_b x M_b)^2 + (Kt \times Mt)^2}$ ....(3.38)

Where:-

d = diameter of the shaft.

K_b = combined shock and fatigue factor applied to bending moment

Kt = Combined shock and fatigue factor applied to Tortional moment

Mt = Tortional moment

M_b = maximum bending moment

Ss = allowable stress for shaft without key way = 55×10^6 N/m2

$K_b = 1.5$ (constant)

$K_t = 1.0$ constant

Substituting,

$d^3 = 16/\pi (55 \times 10^6) \times \sqrt{(1.5 \times 12.0)^2 + (1.0 \times 15.35)^2}$

$= 9.2 \times 10^{-8} \sqrt{563.0116}$

$= (9.2 \times 10^{-8}) (23.72786548)$

$= 0.2182 \times 10^{-5}$

Therefore d $= 3\sqrt{0.2182 \times 10^{-5}}$

$= 0.01297$m

= 13mm.

However, because of some physical constraints, a diameter of 16mm was used for the driving shaft of the machine as shown in figure. 5F of appendix A

3.4.10.7 **Bearing Design and Selection**

From the American society of Mechanical Engineers' (ASME) code bearing catalogue for Agricultural Machines, bearings of 300 to 3000 operating hours is appropriate for this machine. (Obioha, 1998)

Assume:

Fr = radial load on the bearing

P = Power transmitted

Fa = axial load

n = number revolution per minute (rpm)

The radial load is calculated as follows:

$Fr = P/2\pi\ n/60$

$= 1678.5/2\pi\ (273/60)$ ....(3.39)

$= 58.46KN$

The axial load = 146.5N

$Fa = 146.5N$

$= 0.1465KN$

Therefore the equivalent dynamic load acting on the bearing is calculated by the following expression:

$P = (x\ Fr + YFa)$

Where:

$x = 0.56$; $Y = 1.8$ (obioha, 1998)

Substituting,

$P = \{(0.56)\ (58.46) + (1.8)\ (146.5)\}$

$= 296.44N.$

Substituting,

$P = \{(0.56)\ (58.46) + (1.8)\ (146.5)\}$

3.4.10.8 **Bearing Life**

The required life of a bearing is given the following formula.

$Lh=16666/n(c/p)^{\rho}$ .... (3.40) [Felix1985]

Where:

Lh = required bearing life

C = dynamic load

n = speed in r.p.m. = 273

$\rho = 10/3 = 3.33$

$c/p = 2.12$

$Lh = 16666/273\ (2.12)^{3.333}$

$\cong 745.36$ hr

But the required life of the bearing in million revolution per second is

L = rpm x operating life of bearing

= 270 x 745.36

= 203484.00 (rev/sec)

3.4.10.9 **Dynamic Capacity of Bearing**

The dynamic capacity of the bearing is calculated by using the following expression (Obioha,1998)

$L = (C/Fr)^n$ ....(3.41)

Where n = 3 for ball bearing

Fr = radial load = 58710N

C = dynamic capacity of bearing

Transforming the above formula and making C the subject, we will have

$C = L^{1/n}\ Fr\ C$ (Obioha , 1998)

Substituting,

$C = (203484.00)^{1/3} \times 58710 \text{x} 3.33$

= 11499.16N.

The chosen bearing from the table that corresponds to the dynamic load of 11499.2 is CBA 6301Z sealed bearing.

3.4.11 **Design of Shoe Opener**

The shoe opener is designed based on the following notation (Bernacki et al 1978):

Let:

R = Soil reaction

T = Frictional force of soil

μ = co-efficient of friction of soil

G = Force applied

F = Cross sectional area of furrow

K = Specific resistance of soil

V_m = Working speed (m/s)

P_U = draft

δ = stress of internal friction of soil

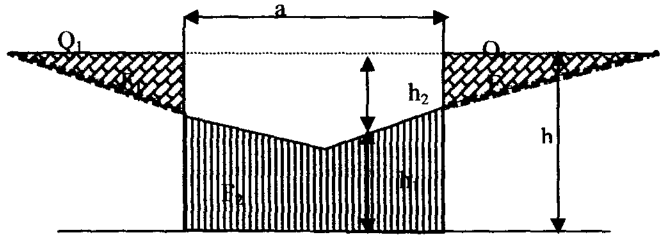

Figure 14: Cross section of the soil after passing of the shoe opener wings.

From Coulomb's equation of soil shearing given by

$$T = C + \delta\mu \quad \text{......(3.42)}$$

C = 0.225 for sandy loam {kg/cm^2)

$\mu = 26$

$\delta = 18 kg/cm^2$

t = shear stress of soil

The shearing force of sandy loam soil is $t = C + \delta\mu$

$= 0.225 + 18 \times 26$

$= 468\ kg/cm^2$

Frictional force (T)

$$T = \mu R$$

Where

$\mu = 0.73$ for steel

$R = 468\ kg/cm^2$

$T = 468 \times 0.73 \times 10^{-2} \times 9.89$

$\cong 36.4N$

The draft (P_u)

$$P_u = W \cos\theta \quad \text{......(3.43)}$$

Where $W = G = 3.35\ kg$

$\theta = 36^0$

$P_u = 3.35 \times \text{Cos } 36^0$

$= 2.71kg$

$= 26.59$ N/cm

From figure. 7 above, the wings are positioned parallel

Therefore $a = (h - h_1)^2/h_1$

where:

a = width of wings

h = 6cm

h_1 = 3cm

: - $a = (6\text{-}3)^2/3$

= 9/3

= 3cm

=30mm

Sliding length S' = W/Sin θ (Bernacki et al, 1978)

Where W = 75; $\theta = 36^0$ (nose angle)

$S' = W/\sin 36^0$

= 75/Sin 36

= 128mm (approx)

= 12.8cm

The calculations above indicates that the wings of the opener can be made 30mm apart (as shown in figure. 8g) and parallel. The sliding length (S') of 12.8cm is appropriate, while a bevel angle of 15^0 is sufficient to provide the necessary sharpness of the cast iron shear and a nose angle of 36^0 was provided for the opener. The sketch of the shear (shoe) opener is shown on figures. Xiii h and figure xiii I of appendix A. Also, the furrow opener assembly is as shown on figure xiii No.1 of appendix B

3.4.12: **Furrow Covering Device**

3.4.12.1 **Determination of Disc Diameter**

The Discs are two in number made of cast steel and hardened 50mm from the edges. Diameter of the disc is calculated as follows (Bernacki et al 1978):

$D = K (a/\cos\theta)$ ....(3.44)

Where:

D = diameter of the disc

A = maximum depth of cut = 50mm

β = Inclination angle of disc = 38^0

K = dimensionless constant = 3

Substituting

D = 3 x 5.0/Cos 38^0

= 19cm

The apex angle is calculated as follows:

$\varphi = \theta_0 - \iota - \varepsilon$ ....(3.45)

where:

θ_0 = setting angle = 15^0 [Yisa and Idah, 1999]

i = angle of sharpness = 11^0 [Yisa and Idah, 1999]

ε_0 = -1^0 (relief angle) [Yisa and Idah, 1999]

Substituting

= 15 – 11-(-1)

= 5^0

The cutting angle is calculated as follows

$y = \iota + \varepsilon_0$.. ....(3.46)

= 11 + (-1)

= 10^0

The minimum thickness of the disk is calculated as follows

δ = 0.008 D + 1mm

where δ = thickness of the disc (minimum)

D = diameter of the disc

Substituting

δ = 0.008 (19) + 1

= 1.15mm

However, 2 mm was selected as the thickness of the disc for wear resistance

The distance (spacing) between the two discs is calculated as follows:

$t_t = \{2\sqrt{[(c/\cos\beta)\ (D - C/\ \cos\beta) + e]}\} \tan\theta_0$ [Bernacki et al, 1978]....................(3.47)

Where:

t_t = spacing of the discs

e = transposition value which can make

t_t greater than 2a, therefore e =30cm

C = 0.5a where a = 5cm maximum depth of cut.

C = 0.5x5=2.5cm

D = Disc diameter = 19cm

θ_0 = setting angle of disc = 15^0

Substituting.

$t_t = \{2\sqrt{(2.5/Cos38^0)}\,(19-2.5/cos38) + 30\} \times 0.27$

$= \{2\sqrt{3.2})\,(19-3.2) + 30\} \times 0.27$

$\cong$ 20cm.

From the calculations above, the diameters of the discs is 19cm each spaced 20cm apart to avoid soil clogging between the discs during operation. Four holes of 8mm diameter each were drilled (as shown on figure ix a) for bolting the discs to the hub.

3.4.12.2 **Welded Joints**

The stress ($\delta\omega$) of a butt welded joint subjected to a force p is given by the following expression.

$\delta\omega$ = P/A

= P/Lt … …………………………………………………………………(3.48)

(Shigley, 1978; Krutz et al 1984)

Where A = Cross sectional area

$\delta\omega$ = allowable weld stress

P = Load

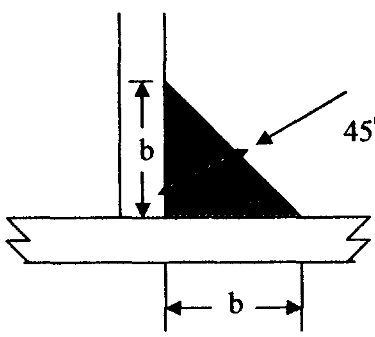

Figure 15: fillet weld

t = b sin 45^0 (Shigley, 1978; Krutz et al 1984)

$= 0.707 \times b$

Where

t = throat

b = leg size

The shear stress of weld is given by the formula

$\gamma\omega = P/0.707tL$

For a single fillet weld subjected to loading, the bending stress is given by the expression

$\delta_b = 3P/hL$ [Shiglay, 1978]

Where h = metal thickness

L = length of weld.

3.4.12.3 **Bolted Joints**

When bolting two plates together, the preload (W_1) which results in bolt tension is given by the following expression.

$W_1 = K_b\delta_b = Kj\delta_j$ (Shigley, 1978) ....(3.49)

Where

j = joint

b = bolt

δ = deflection, cm

K = Stiffness N/cm

= force/deflection

= AE/L

Kj = joint stiffness for 1 member of joint

K_b = stiffness for 1 bolt

A = area affected (bolt or plate)

E = modulus of elasticity

L = length of bolt or plate not to exceed distance between head and nut.

For size 10 bolt,

Root area = $5.80mm^2$

Torque = 19.49N.m (Shigley, 1978)

Proof load = 13.1KN

For size 13 bolt,

Root area = 8.43mm^2

Torque = 33.99N.m (Shigley, 1978)

Proof load = 19.0K.N.

For size 24 bolt,

Root area = 353mm^2

Torque = 284.66n.m (Shigley, 1978)

Proof load = 79.4K.N

For size 36 bolt,

Root area = 817.0mm^2

Torque = 988.24N.m (Shigley, 1978)

Proof load = 184K.N

Hexagonal nuts coarse thread

Size 10:

Roof load = 33.1KN

Torque = 35ft 1b = 4.84N.m

Size 13:

Proof load = 48.1KN

Torque = 80ft 1b $\cong$ 11.06N.m

Size 24:

Proof load = 201KN

Torque = 110ft 1b $\cong$ 15.2N.m

Size 36:

Roof load = 446KN

Torque = 300ft 1b $\cong$ 41.78N.m

3.5 **Construction Procedure:** The construction contains a lot of cutting, drillings and welding all of which were carried out in Waziri Umaru Polytechnic Mechanical Engineering workshop. The cuttings were done by using hacksaw, the welding by arc welding machine and the drillings by radial drilling machine.

3.5.1 **The Frame (Figure xii No. 1)**

The angled steel was cut into four pieces by using hacksaw. Two pieces are 800mm long each while the other two pieces are 400mm long each. Twenty two holes of

8mm diameter each were drilled by using electrically powered radial drilling machine as shown on figure. ic.

3.5.2 The Hopper (Figure xii No 2)

Steel sheet was cut into dimensions as shown on (figure iii a) to g. The parts of the hopper consist of the bottom plate (figure iii a), and the bucket (figure iii e).

3.5.3 The Handles (figure xii No. 6)

Steel pipe was cut by using hacksaw into two pieces 1050mm long each. The two gripping points were bent at angles 45^0 each by using an iron bending machine in WUP workshop, as shown on figures ii a and iib

3.5.4 Crossbar (figure xii No. 7)

Flat steel bar was cut 600mm long by using hacksaw. Two holes of 8mm diameter were drilled as shown on figure. ii b.

3.5.5. Handle's Height Adjustment Devices (figure xii No. 8)

The steel bar was cut with a hacksaw, into the dimension shown in figure. ii d. Holes of 8mm diameter are drilled as shown (figure ii d).

3.5.6 Front Wheels. (figure. xii No.4)

Two pieces of flat steel bar of length 1600mm each were cut with a saw and formed into circles.

3.5.7 Driven Shaft. (figure. xii No. 14)

Steel rod 16.5 mm diameter was cut with a hacksaw to a length of 400mm. It was then machined to 9.5 mm for a length of 20mm at each of its two ends to make seats for the selected bearings. This was carried out in WUP Mechanical Engineering workshop

3.5.8 The Seed Metering Devices (Figure xii.No. 30).

An iron rod of 100mm diameter was machined to 70mm diameter by using lath machine. Five pieces of 25mm length each were cut to make the seed metering devices for 6 different crops. A hole of 16.5 diameter was drilled on the center of each of the devices so that the driven shaft can pass through each one of them. Figures vi d and vi e respectively. The holes were drilled on the circumference of each device based on the crop to be sown as explained in section 3.2 above.

3.5.9 Chain Adjustment Devices (Figure. xii No. 17)

Two pieces of angled iron were cut with a saw into dimensions as shown on figure vii d. The slots for chain adjustment were cut as shown in figure vii d where each slot is 60mm long.

3.5.10 **The Driving Wheel (Figure. xii No. 14)**

The wheel was constructed by cutting the steel with a hacksaw to the dimensions indicated above. The other parts of the wheel which include the two circular plates, the strakes and the pieces of pipe were cut with a hacksaw to dimensions as explained in figures 5a to 5e respectively.

3.5.11 **Driving Shaft**

This shaft was made of steel rod as explained in section above. The detailed procedure was as shown on figure v f.

3.5.12 **Coupling Accessories (Figure. xii Nos. 20, 21, 22).**

These accessories were cut with a saw to dimensions as explained above, and as shown on figures vii a, vii b, and vii c respectively. The same gauge of flat steel bar was used for the three components as shown in figure vii. The accessories consist of frame's rear supports and hopper supports.

3.5.13 Furrow **Opener Assembly (Figure. xiii No. 1)**

This assembly consists of the brackets, furrow depth adjuster, and shoe opener. Each of this part was constructed as explained above, and as shown in figures 8a to 8h respectively.

3.5.14. **Furrow Covering Device (Figure. xiii No. 2)**

This device consists of two plain discs, two rings, two bearing housings and shafts, two shanks, the depth adjuster and the bracket. Each of these parts were cut with hacksaw to dimensions as shown in figures ix a to ix g respectively. The bracket was constructed as shown in figure ix h.

3.6 **Assembling**

3.6.1 **The Frame**

The four pieces of cut-angled iron were joined together by means of the holes shown, with nuts and bolts. The assembled frame is shown in figure i c. and figure. xii No.1

3.6.2 The Hopper

The circular ring (figure. iii b) was joined to the base of the bucket (figure. iii e) by welding. The bottom plate was then bolted to the circular ring as shown in figure iii f. The lid (figure. iii d) was placed on the top of the bucket thus completing the hopper as shown in figure iii g, and figure. xii No. 3

3.6.3. The Handles

Each of the handles was bolted to either side of the frame. The cross bar was bolted on the handles to provide the necessary width and to make the handles rigid as well. The handles height adjusters were then bolted to the handles and the lower sections of the coupling accessories. The handles are as shown on figure. xii No. 6.

3.6.4 Front Wheels

The spokes were welded by using arc welding machine, to the circular rings and the hubs were joined to the spokes as shown in figure. iv e. The strakes were then welded on the circumference of the ring at intervals of 260mm as shown in figure ix e. The bearings were then forced into the hubs as shown on figure iv f thus completing the wheel assembly.

The wheels shaft was then passed through the hubs and bolted with nuts and then the frame was bolted to the wheel shaft. The front wheels are as shown on figure.xiiNo. 4

3.6.5 Driven Shaft Assembly

The driven sprocket was keyed to the driven shaft and seed meter is forced through the shaft and keyed by screwing the bolt provided as shown in figure vif. The two bearings were then forced into their positions on the shaft and the whole arrangement was bolted to the frame by means of the bearing housings. The assembly is as shown in figure. xii No. 15

3.9.1 **Materials Costing:**

Table2: The table below gives a summary of materials, and their costs.

S/NO	MATERIAL	SPECIFICATION	QUANTITY	RATE(₦)	COST(₦)
1	Iron rod (mid steel)	¾" (19mm) Dia	1 length	2500.0	2500.0
2	Iron rod (mid steel)	¼" diameter.	1 length	900.00	900.00
3	Iron rod (mid steel)	3.5" diameter	20cm	1000.00	1000.00
4	Angled iron (90°)	40mmx40mmx2mm	1 length	2500.00	2500.00
5	Steel sheet	1mm thickness	½ sheet	1900.00	1900.00
6	Steel sheet	2mm thickness	¼ sheet	4000.00	1000.00
7	Cost Steel sheet	3mm thickness	¼ sheet	6000.00	1500.00
8	Steel pipe	32mm diameter	1 length	1200	1200.00
9	Steel pipe	60mm diameter	¼ length	3500.00	875.00
10	Steel pipe	17mm diameter	½ length	1000.00	250.00
11	Flat Steel bar	40mm x 2mm	1 length	1600.00	1600.00
12	Flat Steel bar	10mm x 2mm	1 length	500.00	500.00
13	Flat Steel bar	45mm x 4mm	1 length	1800.00	1800.00
14	Flat Steel bar	50mm x 4mm	½ length	350.00	1750.00
15	bearings	CBA 6304z	4Nos.	350.00	1400.00
16	bearings	CBA 6302z	4Nos.	120.00	480.00
17	Sprocket	39teeth, N482	1 mNo.	400.00	400.00
18	Sprocket	13teeth, N4820	1 No.	100.00	100.00
19	Roller chain	CN 428	1 No.	350.00	350.00
20	Bolt,Nuts and Washers	M13	100	15.00	1500.00
21	Nuts and Washers	M22, M36	15	50.00	750.00
22	Welding electrode	Guage 12	1 packet	1000.00	1000.00
23	Paints	Oil paint: dark red colour	3 tins	120.00	360.00
				TOTAL	24345.00

3.9.2 **Cost Analysis**

The total cost of materials is estimated as ₦24345.00.

Cost analysis is classified into three (James, 2002).

i. Cost of materials (mc)

ii. Cost of labour or workmanship (LC)

iii. Over head cost (OC).

- Materials cost is the total cost of materials estimated as ₦24,345.00
- Labour cost (cost of workmanship) is taken as 30% (Sahabi, 2002) of material cost.

Therefore, labour cost = 30/100 x 24065

= ₦7, 219.50

- Over head cost is taken as 60% of labour cost, (Sahabi, 2002)

Therefore, Overhead cost = 60/100 x 7219.50

= ₦ 4331.7

The total cost of fabrication is the sum of the above three types of costs = Material cost + labour cost + overhead cost.

Therefore, total cost = 24065 + 7,219.50 + 4331.7

= ₦35616.2

The profit is considered as 10% of the total cost of fabrication (James, 2002).

Therefore profit = 10/100 x 35616.2

= ₦3561.62

The price of the machine is taken to be total cost of fabrication + profit

Therefore, price of the machine = 356162.0 + 3561.62

= ₦39,177.82

3.6.6 **Driving Wheel Assembly**

The two circular plates were welded [arc welding] to either side of the steel pipe used to make the wheel. The two pieces of pipes were then welded to the plates, one on each side, while the strakes were welded [arc welding] to the circumference of the wheel as shown on figures v e at intervals of 60mm. The driving shaft (figure. v f) was then slided in to the wheel hubs and keyed by the bolt provided so that they rotate as one. The driving sprocket is then bolted to the driving shaft by means of a nut. The two bearings were then forced into their position on the shaft and this whole arrangement was bolted to the chain adjustment devices bolted to the frame. The chain was them placed on its position on the two sprockets and then tensioned by means of the adjusters. This arrangement is shown in figure vi.

The hopper was then bolted to the frame on its position above the seed metering device. The seed dispenser unit was then bolted below the hopper. This assembly is as shown on figure. xii No. 14

3.6.7 **Furrow Opener Assembly**

The depth adjuster (figure viii f) was joined to the bracket (figure. viii e) by arc welding. This bracket was then bolted the other bracket shown in figure viii g. The shoe opener was then welded to the bracket shown in figures viii g. The complete assembly is as shown on figure viii i. The furrow opener bracket (figure viii d was then bolted to the frame and the assembled furrow opener was attached to the bracket thus completing the assembly. This assembly is as shown on figure xiii No. 1 and figure xii No. 24 of appendix B.

3.6.8 Furrow Covering Device

The circular ring (figure ix b) was welded to the hub (figure. ix c) the bearing was forced into the hub while the shaft (figure ix d) was forced into the bearing and hub. The shank (figure ix e) was then bolted to the shaft. The disc (figure ix a) was then bolted to the circular ring. The same procedure was repeated for the other disc. The bracket shown in figure ix h was then bolted on to its position on the frame. The two discs were then bolted to the depth adjuster shown in figure ix F. This arrangement was then bolted to the bracket (figure. ix h) thus completing the furrow coverer assembly; as shown on figure ix I, and figure. xiii No. 2

3.6.9 The Hopper Housing

The hopper housing (figure. xi) was bolted to the hopper guide (figure. iii c) by means of bolts and nuts, through the holes provided, thus housing the hopper. This now makes the machine completely assembled and ready for testing. This housing is as shown on figure. xii No. 2. Please see figures xiv, and xvi for complete assembly of the machine.

3.7 Description of the Machine

The grains sowing machine designed, constructed and assembled consists of the following major parts.

i. The frame assembly
ii. Sprocket and chain assembly
ii. Driving wheel assembly
iii. Front wheels assembly
iv. Furrow opener assembly
v. Furrow coverer assembly
vi. Grain box assembly
vii. Seed dispenser unit
viii. Seed metering mechanisms
ix. The handles

The planter consists of the main frame supporting a hopper (grain box), which incorporates a seed metering mechanism on its underside. This mechanism is driven by the driving wheel through sprockets and chain. The adjustable furrow opener (shoe type) is supported by the frame at the front of the machine, and is responsible for digging a furrow (hole) for seeds. The seed dispenser unit which directs the seeds into the dug hole is supported by the frame and is at the rear and in-line with the furrow opener. The furrow covering device which is also adjustable –is a double discs coulter type and is supported by the frame and it is positioned at the rear and in-line with seed dispenser unit. The handles are bolted to the rear side of the frame, they can be raised up or lowered as desired by the operator. The assembled machine is as shown on figure. xvi.

3.8 Principles of Operation of the Machine

Given a push manually or pull by an animal, the machine moves forward. In this instance, the rotation of the driving (rear) wheel is used to rotate the seed metering mechanism through the driving and driven sprockets and shafts, by means of a power chain. Suitable holes on this mechanism meter out correct quantity of seed from the orifice at the bottom of the grain box (hopper) and discharges them at intervals-according to the required spacing and quantity per hole of the particular crop being sown into the conveyor tube of the seed dispensing unit, and hence into the furrow dug by the furrow opener. The furrow covering device at the rear of the seed tube then covers the grains with soil layer appropriately. The seed meter is replaceable so as to meet the planting requirements of different crops. The furrow opening and covering devices are adjusted by slanting to the required depth. The handles of the machine can be raised or lowered depending on the height of the operator. Chain slackness can be tensioned by adjusting the driving wheel rearwards.

3.9 Materials Selection

The materials selected and used in this design depend on the requirements of the relevant parts relating to their functions, stress conditions and service life. The methods of welding, bolting and finishing of parts, were considered. Availability and procurement of material and cost of production were also considered.

Mild steel rod was used for power transmission shafts. Angle iron was used for the frame in order to reduce the overall weight of the machine.

For the operating handles mild steel pipe was selected and used because of its light weight and strength. For the wheels flat bar of size 50mm x 3mm was used for its ease of manipulation. The need to maintain low level of friction in power transmission systems prompted the choice of deep groove ball bearings for rotating shafts.

All standard components such as chain, sprockets, bearings, bolts, nuts and washers were purchased locally from the market.

3.9.3 Testing and Evaluation

Test Measurements

3.9.4 Determination of Theoretical Working Width

Theoretical working width of the machine is obtained by measuring the actual width of the machine by means of a measuring tape (Smith et al, 1994). The width was measured from the front of the machine by measuring the distance from the left end of the left wheel to the right end of the right wheel.

3.9.5 Determination of Effective Working Width

A plot of 20m width and 40m length was measured and cleared. The plot was ploughed and harrowed. The moisture content of the soil was determined. The machine was set to plant maize grains at 2seeds per hill at intervals of 25cm between hills. The effective working width of the machine was obtained as follows.

Effective width = width of the plot (20m)/number of passes necessary (made) to complete (cover) the plot by the machine in operation (the rows are spaced 58cm apart) (Smith et al, 1994).

3.9.6 **Determination of Wheel Skid**

Will skid will occur in normal work because the machine is land wheel driven even though strakes were provided for the wheels.

Procedures:-

i. A mark was made on the ground. The driving wheel was set on the mark and the machine was pushed to cover a distance of 5 revolution of the driving wheel and the distance was recorded. This was done with machine out of work.

ii. The same method was used with machine in operation and the distance for 5 revolution of the driving wheel was measured and recorded.

The percentage wheel skid is then calculated from following formula

Wheel skid (%) =[A – B]/B x 100 (Smith et al, 1994)..........(3.50)

Where:

A = Distance covered in 5 revolutions of the driving wheel with machine out of work

B = Distance covered in 5 revolutions of the driving wheel with the machine at work.

3.9.7 **Determination of Seed Spacing Evenness**

A distance of 10m was measured on a leveled field. The surface was covered with dry sand to prevent the grains from bouncing.

Grains of maize were put inside the hopper and the machine was pushed to cover the above distance. The distances between seed hills on the surface of field were measure as deposited. The same procedure was repeated using grains of millet, guinea corn, cowpea and groundnuts. The seed spacing evenness of each of the crops above was calculated as follows.

Evenness of spacing = seed spacing (average) – deviation of seed spacing (average) (Smith et al, 1994).

Where:

Deviation of spacing = measured spacing (average)- intended spacing (average.)

3.9.8 **Determination of Working Depth**

The working depth may depend on the particular crop being sown. Small grains like millet and guinea corn may have shallow sowing depth, while large seeds like maize, cowpeas or groundnuts may have deep sowing depth. However, the designed total working depth of the machine is 6cm.

After the field test, the planted maize seed was allowed to germinate. Some plants (5 hills) were carefully dug at one side of the soil surface so that the plants were exposed from the soil surface to the root zone without allowing the plants to fall down, and the depth of deposition of each plant was measured by means of a rule and the average planting depth was calculated as follows:

Average planting Depth = Sum of depths of deposition of all the plants/number of plants (Smith et al, 1994).

3.9.9 Determination of Seed Weight

Ten samples of 100 grains each of maize, millet, Guinea corn, groundnuts and cowpeas at 18.5% moisture content were measured on a weighing scale. And from this figure the weight of one grain for each of the above crops was estimated as follows:

Weight of seed = Total weight of 10 samples/100 (Smith et al, 1994).

3.9.10 Determination of Seed Size

The length, width and thickness of 100 seed, each of maize, millet, Guinea corn, cowpeas and ground nuts were measured with Vanier calipers and the average sizes were estimated for each of the crops. Size = (Length x Width x Thickness) x 1/3 (James, 2002). ....(3.51)

3.9.11 Test of Seed Metering Mechanism

The objective here was to examine the performance of seed metering mechanism so that seed range can be determined. The seeds used did not contain any damaged grains seed to enable the estimation of any damage caused by the machine.

Procedure:

i. The machine was raised up at the rear and the driving wheel was rotated 30 times at constant speed of 0.75m/s with the hopper full. The time taken to complete the rotation was recorded, (Smith et al, 1994).
ii. The test was repeated with hopper half-full and the time taken to complete 30 rotations was recorded, (Smith et al, 1994).
iii. Again, the test was repeated with the hopper 1/3

full and the time taken to complete 30 rotations were recorded, (Smith et al, 1994).

The same procedure was repeated for crops of millet, Guinea corn, cowpeas and groundnuts. The result of the test for each crop was tabulated.

3.9.12 **Determination of Seed Distribution Pattern**

The machine was operated on level track 10m long at speed of 0.75m/sec. The surface beneath the discharge mechanism was covered with clean sand to prevent the seeds from bouncing. The test was repeated 3 times using maize grains (Smith et al, 1994).

1st run:

distance moved = 10m (1000cm)

No of rotations of driving wheel

= 20 rotations

No. of rotations of driven wheel

$\cong$ 7 rotations)

No. of seeds discharged = 80 seed

No. of drops (hills) – 40 drops

No. of seed per hill = 2 seed.

Average distance measured between hills

= 25cm.

2nd run:

distance run = 10m (1000cm)

No. of rotations of driving wheel = 20

No. of rotations of driven wheel $\cong$ 7

No. of Seed discharged = 80

No of drops (hills) = 40

No of seed per hill = 2

Average distance between hills 26cm

3rd run:

distance moved = 10m (1000cm)

No. of rotations of driving wheel = 20

No. of rotations of driven wheel $\cong 7$

No. of seeds discharged = 80

No of drops (hills) = 40

Average distance between hills = 25cm.

3.9.13 **Field Test**

a. Area of the field = 20m wide, 40m long. The shape of the field is rectangular. The area is 40m x 20m = $800m^2$

b. The soil is sandy loam. Topography is flat (level) land.

c. Type of cultivation made is ploughing followed by harrowing.

d. Moisture content of soil = 26%

Test procedure

1. The hopper was filled with maize grains weighing 8kg, ($0.0084m^3$).
2. Care was taken to ensure that the seed metering mechanism is working correctly as set by rotating the driving wheel several times and watching the movement of the feed mechanism below the grain box, and there is no blockage at the seed outlet.

The following measurements were made, (Smith et al, 1994).

a. Number of passes made to complete planting on the plot = 35

b. Number of rows per pass = 1

c. Row spacing = 58cm

d. Depth of setting of furrow opener = 3cm

e. Forward speed = 0.75m/sec.

f. Time spent on turning = 330 seconds

g. Total operating time = 18667 sec.

= 31minutes, 7sec.

Observations were also made on the following

i. Ease of handling = adequate

ii. Ease of adjustment = adequate

iii. Maintenance of depth = constant

iv. Blockage of working parts = low

h. A farmer was asked to plant the same area field under the same conditions so that comparison can be made between the use of the machine and manual method by planting 2 seeds of maize at 25cm intra-row and 58cm inter-row spacing.

3.9.14 **Germination Rate Determination**

This is the percentage of seeds from the sample which have germinated. The procedure below was used to determine the rate of germination of the planted grains of maize (Smith et al, 1994).

Length of plot = 40m (4000cm)

Width of plot = 20m. (2000cm)

Spacing between rows = 0.58m (58cm)

No of passes made along the width of the row = 20m/0.58m

$\cong$ 35 passes

Seed spacing = 2 seeds per hill at 25 cm intervals.

No of hills (drops) along the length (40m) of the plot = 40m/0.25m

= 160 hills (drops)

No of seeds per pass = 160 x 2 seeds

= 320 seeds

No. of seeds sown on the plot = 320 x 35

= 11200 seeds

Weight of seeds filled in the hopper (6 measures)

= 14759 x 6mudus = 8850g); use 8000g

Weight of 1 grain = 0.198g.

No of grains in the hopper = 8000/0.198

$\cong$404040. grains

CHAPTER FOUR

4.0 RESULTS AND DISCUSSION

4.1.1 Theoretical Working Width

The theoretical working width of the machine was measured as 58cm (0.58m).

4.1.2 Effective Working Width

The effective working width of the machine was obtained as follows:

Width of the plot = 20m (2000cm)

Number of passes = 35

Effective working width = 2000/35

= 57cm (0.58m)

4.1.3 Wheel Skid

The wheel skid was calculated as follows:

A = 250cm

B = 249.5cm

:- Wheel skid (%) = [(250-249.5)/250]x100

=0.2%

4.1.4 Seed Spacing Evenness

i. Maize seeds.

Required spacing = 25cm between hills.

Measured spacing = 26.1cm (average of ten readings)

Difference = 26.1 – 25 = 1.1cm

Evenness of spacing = (26.1 – 1.1)/26.1

$\cong$ 0.96.

ii. Millet seed:

Required spacing = 50cm

Measured spacing = 51cm

Difference = 51-50 = 1cm

Seed spacing evenness = 51-1/51 = 0.99

iii. Guinea corn seed:

Required spacing = 50cm

Measured spacing = 50.8cm

Difference = 50.8-50 =0.8cm

Seed spacing evenness = $\frac{508 - 0.8}{50.8} = 0.98$

iv. Cowpea seed:

Required spacing = 50cm

Measured spacing = 51cm

Difference = 51-50 = 1cm

Seed spacing evenness = 51-1/51 = 0.99

v. Groundnuts seed

Required spacing = 25cm

Measured spacing = 26.1cm

Difference = 26.1 – 25 = 1.1cm

Seed spacing evenness = 26.1-1.1/26.1

$\cong 0.96$

4.1.5 Working Depth

No. of hills dug after germination = 5

No of plants per hill = 2 plant

Total number of plants dug = 5 x 2

= 10 plants

Table 3: Measurement of depth of deposition of maize seed

Plant No.	Depth measured from surface of land to root zone (cm)
1.	3.3
2.	2.9
3.	3.0
4.	3.3
5.	3.0
6.	3.0
7.	3.4
8.	3.0
9.	2.9
10.	3.0
Total	30.8

Average depth = 30.8/10

= 3.08cm

$\cong$ 3.1cm

This result was obtained due to the fact that the furrow opener was set to dig at a depth of 3cm, and the plot was ploughed and harrowed.

4.1.6 **Seed Weight**

i. Maize seed (Mexico5)

Ten samples of 100 grains each, of maize were weight in the laboratory. The result is shown in table below. The average moisture content is 18.5%

Table 4: Weight of 1000 grains of maize seed (Mexico 5)

Sample No.	No. of grains	Weight (g)
1	100	21
2	100	21
3	100	21
4	100	20
5	100	16
6	100	19
7	100	21
8	100	20
9	100	21
10	100	18
Total	1000	198
Average weight of grain		0.198

Groundnuts (runner T37)

Table 5: Weight of 1000 seed of Groundnut (creeping variety) at 18.5% moisture content. The moisture content was obtained by the use or electrically operated moisture meter.

Sample No.	No. of grains	Weight (g)
1	100	34
2	100	32
3	100	34
4	100	34
5	100	31
6	100	32
7	100	33
8	100	31
9	100	28
10	100	29
Total	1000	318
Average weight of grain		0.318

i. Groundnuts (Samaru 38)

Table 6: Weight of 1000 grains of Groundnuts (erect type) at 18.5% moisture content. Percentage moisture obtained by moisture meter.

Sample No.	No. of grains	Weight (g)
1	100	50
2	100	41
3	100	51
4	100	50
5	100	37
6	100	48
7	100	47
8	100	37
9	100	41
10	100	47
Total	1000	0.449
Average weight of grain		0.318

iv. Cowpea – local variety (Kano white)

Table:7 Weight of 1000 seeds of cowpea at 18.5% moisture content obtained by the use of moisture meter.

Sample No.	No. of grains	Weight (g)
1	100	18
2	100	21
3	100	18
4	100	18
5	100	18
6	100	20
7	100	21
8	100	21
9	100	19
10	100	20
Total	1000	318
Average weight of grain		0.318

v. Millet seed (Ex Borno)

Table 8: Weight of 1000 grains of millet at 18.5% moisture content obtained by the use of moisture meter.

Sample No.	No. of grains	Weight (g)
1	100	1
2	100	1
3	100	1
4	100	1
5	100	1
6	100	1
7	100	1
8	100	1
9	100	1
10	100	1
Total	1000	10
Average weight of grain		0.01

vi. Guinea corn seed. (White,YG5760)

Table 9: Weight of 1000 seed of Guinea corn at 18.5% moisture content obtained by the use of moisture meter.

Sample No.	No. of grains	Weight (g)
1	100	4
2	100	3
3	100	3
4	100	3
5	100	3
6	100	3
7	100	3
8	100	3
9	100	3
10	100	3
Total	1000	25
Average weight of grain		0.025

The weight of I measured at 18.5% percent moisture content was obtained, by weighing on a scale, for each of the crops above

i. Maize = 1475 g

ii. G. Nuts creeping type = 2247 g

iii. G. Nuts erect type = 1112 g

iv. Cowpea (creeping type) = 1239 g

v. Millet = 1368 g

vi. Guinea corn = 1473 g

4.17 **Seed Size**

The length, width, and thickness, of the crops of maize, cowpeas, groundnuts millet and guinea corn were estimated and the results are tabulated as shown in the following tables.

Table 10: Determination of the size of maize seed (Mexico5)

Sample No.	No. of grains	Average length (x)(mm)	Average width (mm)	Average thickness (h)(mm)
			9.01	4.5
1	10	11.01	9.02	4.01
2	10	10.02	10.0	4.5
3	10	10.5	8.01	4.01
4	10	10.01	10.01	5.0
5	10	10.01	8.02	4.01
6	10	8.5	9.0	4.01
7	10	11.0	7.01	4.0
8	10	9.01	8.5	4.5
9	10	9.0	8.5	5.01
10	10	8.5	6.01	4.0
11	10	7.01	5.01	4.5
12	10	8.0	10.0	4.5
13	10	11.01	6.5	3.02
14	10	7.5	7.5	3.01
15	10	9.0	8.5	3.5
16	10	9.01		
Average		9.3	8.2	4.1

Table 11: Determination of the size of cowpeas (kano white)

Sample No	No of grain	Average length (x)(mm)	Average width (y)(mm)	Average thickness (h)(mm)
1	10	10.0	8.01	6.5
2	10	10.5	8.5	7.01
3	10	10.01	8.5	7.5
4	10	8.02	7.01	5.5
5	10	9.5	7.01	4.5
6	10	7.5	6.5	5.5
7	10	9.5	8.5	7.0
8	10	9.5	7.5	6.5
9	10	10.01	7.5	6.0
10	10	10.01	7.02	6.5
11	10	10.01	8.5	7.0
12	10	10.01	7.5	6.5
13	10	10.01	8.5	6.5
14	10	12.0	8.5	7.0
15	10	10.02	8.02	6.5
16	10	7.5	7.01	5.0
17	10	10.5	7.5	6.0
18	10	8.5	7.5	5.5
19	10	8.01	6.5	5.0
20	10	9.5	7.02	5.5
Total	200.0	190.5	152.5	123
Average		9.53	7.63	6.15

Table 12: Determination of size of Ground nuts (runner T37).

Sample No.	No. of grains	Average length (x)(mm)	Average width (y)(mm)	Average thickness (h)(mm)
1	10	13.01	9.01	8.5
2	10	11.5	8.5	8.01
3	10	11.01	8.02	7.5
4	10	11.02	7.5	7.01
5	10	10.5	7.01	6.5
6	10	10.5	8.0	7.01
7	10	11.01	8.01	6.5
8	10	9.01	6.5	6.01
9	10	15.0	8.02	7.5
10	10	10.01	6.5	6.01
11	10	11.02	6.5	6.01
12	10	12.01	8.5	7.5
13	10	9.5	6.5	6.02
14	10	11.5	8.01	7.5
15	10	12.5	8.5	8.01
Total	150.0	168.5	115	112
Average	100	11.23	7.67	7.47

Table 13: Determination of the size of groundnut seed (Sumaru 38)

Sample No.	No. of grains	Average length (x)(mm)	Average width (y)(mm)	Average thickness (h)(mm)
1	10	21.0	9.5	9.01
2	10	14.5	9.02	7.5
3	10	7.5	7.01	7.01
4	10	19.5	8.5	8.01
5	10	18.01	9.5	9.01
6	10	13.01	8.0	7.5
7	10	16.5	11.01	8.5
8	10	13.5	8.5	8.01
9	10	15.01	9.01	8.5
10	10	16.01	9.01	8.5
11	10	14.01	8.02	7.5
12	10	16.5	9.01	8.5
13	10	15.02	9.01	8.5
14	10	13.5	10.0	9.5
15	10	16.5	9.5	9.01
Total	150.0	229.5	134.5	124.5
Average		15.3	8.97	8.3

Table 14: Determination of the size of millet seed (Ex Borno)

Sample No.	No. of grains	Average length (x)(mm)	Average width (y)(mm)	Average thickness (h)(mm)
1	10	4.01	3.02	2.5
2	10	3.5	2.5	2.01
3	10	3.5	3.01	2.5
4	10	4.01	3.01	2.5
5	10	3.5	3.01	3.01
6	10	4.01	3.01	2.5
7	10	3.5	3.5	2.01
8	10	4.02	3.01	3.02
9	10	3.01	2.5	2.00
10	10	3.5	2.5	2.01
11	10	4.01	2.5	2.5
12	10	4.01	3.02	2.5
13	10	3.5	3.01	2.5
14	10	3.01	2.03	2.01
15	10	3.5	2.5	2.02
Total	150.0	54.5	41	35.5
Average	100	3.6	2.7	2.4

Table 15: Determination of the size of Guinea corn.(White, YG5760)

Sample No	No of Grain	Average length (x)(mm)	Average width (x)(mm)	Average thickness (h)(mm)
1	10	4.5	4.0	2.5
2	10	5.0	4.5	3.0
3	10	5.0	4.5	3.0
4	10	4.5	4.0	2.5
5	10	5.0	4.5	3.0
6	10	5.5	4.5	3.0
7	10	5.0	4.0	2.5
8	10	5.5	4.5	2.5
9	10	4.5	3.0	2.0
10	10	5.0	4.0	3.0
Total	100	49.5	40.5	26
Average		4.95	4.05	2.6

4.1.8 **Seed Metering Mechanism**

I Maize Seed:

Weight of 1000 grains = 198g

Average length = 9.3mm

Average width = 8.2mm

Average thickness = 4.1mm

Bulk density = $0.70 kg/cm^3$

Angle of repose = 31^0

Average moisture content = 18.5%

Seed Range and Damage Determination

Table 16: Seed Range and Damage for Maize Crop.

Sample No.	No. of rev. of driving wheel	Time taken (sec)	Quantity dropped	No. of damaged seeds
1	30	20	120	2
2	30	25	120	2
3	30	22	120	2
Total	90	67	360	6

At 30 revolutions of the drive wheel the machine is expected to cover a distance of 15 meters. The speed ranges are calculated as follows:

Test No. 1: Speed = 15m/20sec. = 0.75m/sec.

Test No. 2: Speed = 15m/25sec = 0.6m/ sec.

Test No. 3: Speed = 15/22 = 0.68m/sec.

Seed damage = (6/360) x 100

$\cong$ 1.67%.

ii. Millet seed

Weight of 1000grains = 10g

Average length = 3.6mm

Average width = 2.7mm

Average thickness = 2.4mm

Average moisture contents 18.5%

Table 17: seed range and damage for millet seed.

Sample No.	No. of rev. of driving wheel	Time taken (sec)	Quantity dropped	No. of damaged seeds
1	30	21	149	2
2	30	20	148	1
3	30	23	150	1
Total	90	64	447	4

At 30 revolution of the driving wheel, the machine is expected to cover a distance of 15 meters. The speed ranges are calculated as follows.

Test No 1: Speed = 15/21 = 0.7m/sec.

Test No. 2: Speed = 15/20=0.75m/sec

Test No. 3: speed = 15/23 = 0.65 m/sec.

Seed damage = (4/447) x 100

$\cong$ 1/%

iii. – Guinea corn seed

Weight of 1000 grains = 25g

Average length = 5.0mm

Average width = 4mm

Average thickness = 2.6mm

Moisture content = 18.5%

Table 18: seed range and damage for Guinea corn seed

Sample No.	No. of rev. of driving wheel	Time taken (sec)	Quantity dropped	No. of damaged seeds
1	30	25	150	2
2	30	20	149	2
3	30	22	150	0
Total	90	67	449	4

At 30 revolutions of the driving wheel, the machine will cover a distance of 15 meters. The speed ranges are

Test No. 1: Speed = 15/25 = 0.6m/Sec

Test No.2: Speed = 15/20 = 0.75m/sec

Test No. 3 Speed = 15/22 = 0.68m/sec.

Seed damage = (4/449) x 100

$\cong$ 0.9%

iv. Cowpea seed

Weight of 1000 grain = 318g

Average length = 9.53mm

Average width = 7.63mm

Average thickness = 6.15mm

Moisture content = 18.5%

Table 19: seed Range and damage for cow peas seed.

Sample No.	No. of rev. of driving wheel	Time taken (sec)	Quantity dropped	No. of damaged seeds
1	30	26	60	1
2	30	23	60	2
3	30	20	60	1
Total	90	69	180	4

At 30 revolution of the driving wheel, the machine covers a distance of 15 meters. Therefore the speed ranges in this test is ad follows

Test No. 1: Speed = 15/26

$\cong$ 0.58m/sec

Test No. 2: Speed = 15/23

$\cong$ 0.65 m/sec.

Test No. 3: Speed = 15/20

$\cong$ 0.75 m/sec

Rate of seed damage = (4/180) x 100

$\cong$ 2.22%

i. Groundnuts (creeping type)

Weight of 1000 grains = 318g

Average length $\cong$ 11.2mm

Average width $\cong$ 7.7mm

Average thickness $\cong$ 7.5mm

Moisture content = 18.5%

Table 20: Seed Range and Damage for Groundnuts Seed.

Sample No.	No. of rev. of driving wheel	Time taken (sec)	Quantity dropped	No. of damaged seeds
1	30	25	60	1
2	30	23	60	2
3	30	20	60	2
Total	90	68	180	5

At 30 rev. of the driving wheel, the machine will cover a distance of 15 meters. The speed ranges in this test are given as follows:

Test No; 1: Speed = 15/25 = 0.60m/sec.

Test No. 2: Speed = 15/23 = 0.65 m/sec

Test No. 3: Speed = 15/20 = 0.75 m/sec

Rate of seed damage = (5/180) x 100 = 2.8%

4.1.9 Seed Distribution Pattern

From the experiment conducted in section 3.9.12. The result in run number 2 differs from those in runs 1 and 3 with respect to distance between hills.

Average spacing for maize = (25+26+25)/3

= 76/3

= 25.33cm.

Evenness of seed spacing

$$S = \frac{25.33-25}{25.33}$$

Percentage seed distribution pattern = 100-1.3

= 98.7%

4.1.10 Germination Rate

From the experiment carried out in section 3.9.1.4, the number of seeds which germinated in a row obtained by head count = 310

Weight of seed sample having passed through the mechanism =

0.198 x 11200 seed

= 2217.6g

= 2.2176kg.

Expected No. of seed damage within the sample of maize seed at 1.67% damage rate = 1.67/100 x 11200

= 187 seed.

Weight of damaged seed = 187x0.198g

=37.03g.

Total No. of seed that germinated = 310x35

= 10850 seed.

Total No. of seeds which did not germinate

= 11200 – 10850

= 350 seed

Percentage germination = (11200-350)/11200 x 100

≅ 96.88%

Test setting (kg/ha) = A + A (100-G/100) (Smith et-al, 1994).

Where:

A = recommended rate maximum = 65kg/ha

G = germination rate = 96.88%

:- Test setting = 65+65 (100-96.88/100)

= 67 kg/ha (approx)

% of undamaged seed = $\frac{22176-37.03}{22176}$

= 0.9983x100

= 99.83%

Table 21: Seed Efficiency calculation table for maize

Table 21: Seed Efficiency calculation table for maize

	Value			
Laboratory and filed	Unit	Symbol	Lab	Field
Seed spacing (average)	cm	Ss	-	-
Seed spacing deviation	cm	SSD	-	-
Seed spacing evenness (ss-ssD)/ss	-	Eu	-	-
Seeding depth	cm	d^1	3	2.9
Seeding depth standard deviation	cm	d_d^1	0	0.1
Seeding depth evenness $(d^1 - d_d^1)/d^1$	-	Ed	1	0.97
Hill spacing (average)	cm	HS	25	26.1
Hill spacing standard deviation	cm	HSD	0	1.1
Hill spacing evenness-(HS-HSD)/HS	-	E_h	1	0.96
Number of seeds per hill	-	H	2	2
Number seeds standard deviation	-	HsD	0	0
seeds per hill evenness	-	E_h	1	1
From the laboratory				
1000 grains weight	g	1000gw	198	-
Seed dimensions:				
L=length,	mm	L	L=9.3	-
W= width	mm	w	W=8.2	-
t = Thickness	mm	t	t=4.1	-
Germination rate				
Weight of seed sample having passed through the metering mechanism	g	W^1	22176	22176
Weight of broken seeds in this sample	g	b^1	37.3	37.03
Seed breakage efficiency = $\{(W^1-b^1)/W^1\}$	-	E_b	0.9983	0.9983

4.11 Determination of Field Efficiency of Machine

The field efficiency of an agricultural farm machine is given by the following expression

$$\text{Field efficiency} = \frac{\text{Effective field capacity}}{\text{Theoretical field capacity}}$$

Where:-

Theoretical field capacity = Total area covered. in doing no work

Effective field capacity = total area covered/time spent in doing useful work or productive time.

Length of plot = 40m

Width of plot = 20m

Area of plot = 20x40

$= 800m^2 = 0.08ha$

Time taken to complete a pass =

$\cong 53.3$ sec

Time taken to complete 35 passes = 53.33x35

$\cong 0.52$hrs

Total non-productive time = planting time + total turning time + time for adjustment = 1867

+ (7x35) + 90 = 1.867 + 330

Total time = 1867 + 330 = 2197 sec

$\cong 0.13$/ha/hr.

theoretical field capacity = 0.08/0.62

$\cong 0.154$ ha/hr

Field Efficiency = (0.13/0.154) x 100

= 84.44%

4.12 Comparison with Manual Sowing.

The time taken by a farmer to plant a row (on the same plot size under the same condition in which the machine was tested is six minutes. The total time taken to plant 2 seeds of maize at 25cm intra-row and 58cm inter row spacing on a plot 20m x 40m ($800m^2$) is obtained as follows.

Time taken to complete a row = 6 minutes

Time taken to complete 35 rows = 6 x 35

= 210 minutes

= 3.5hrs.

Time taken by the machine to complete the plot

= 31 minutes, 42 sec.

$\cong 32$ minutes

$210/32 = 6.5655$

This implies that the machine is at least 6 times faster when compared to manual sowing.

Discussions

i. The machine is 58cm (0.58m) wide with a wheel skid of 0.2%.

ii. The seed spacing evenness of the crop used for the test of seed metering device is approximately constant with an average value of 0.98.

iii. The furrow opener provided a constant working depth depending on the setting. A depth of approximately 3cm was obtained at the same setting during field test.

Iv The weight of a given seed depends on many factors. The most important of which are moisture content and quality of the seed used for testing. The higher the moisture content and quality of seed, the higher the weight. The average weight of 1000 grains of each of the crops used for testing are summerized in table 22 below. The moisture content is 18.5%

Table 22: weight of 1000 grains of selected crops

S/NO	CROP	WEIGHT OF 1000 GRAIN (g)
1.	Maize	198
2.	G. Nuts (Creeping)	318
3.	G. Nuts (erect)	449
4.	Cowpeas (creeping)	194
5.	Millet	10
6.	Guinea corn	25

5. The sizes of 100 grains of each of the selected crops are summarized as shown in table 23 below.

Table 23: Summary of sizes of grains of selected crops at 18.5% moisture content.

MEASURED PARAMETER / TYPE OF GRAIN	Length (x) (mm)			Width (y) (mm)			Thickness (h) (mm)			Size 1/3 (xyh) (mm3)			Sphericity (size/x) {mm^2}			Weight of 1000 grains (g)
	MAX	MIN.	AV.	MAX.	MIN	AV	MAX.	MIN.	AV.	MAX.	MIN.	AV.	MAX.	MIN.	AV.	
Maize	11.0	7.5	9.3	10.0	6.5	8.2	5.0	3.0	4.1	166.7	48.8	108.0	16.7	6.5	11.3	198
Cowpeas (creeping Type)	12.0	7.5	9.5	8.5	6.5	7.6	7.0	4.5	6.2	238.0	86.7	147.7	21.3	10.8	15.8	194
Millet	4.0	3.0	3.6	3.5	2.5	2.7	3.0	2.0	2.4	10.5	4.0	8.2	3.0	1.1	2.3	10.0
Groundnuts (Erect type)	21.0	7.5	15.3	10.0	7.0	9.0	9.5	7.0	8.3	598.5	122.5	389.5	31.7	16.3	25.0	449
Ground nuts (creeping type)	13.0	9.0	11.2	9.0	6.5	7.7	8.5	6.0	7.5	331.5	123.5	208	25.5	13.0	18.2	318
Guineacorn	5.5	4.5	5.0	4.5	3.0	4.0	3.0	2.0	2.6	22.5	9.0	18.8	4.5	2.0	3.8	25.0

The lengths (x), widths (y) and thickness (h) were obtained by measurements, while the sizes and sphericity were obtained by calculations.

6. The machine capacity was obtained as 1.82ha/hr.

Width of implement = P x 3.6/v x draft/m

Draft/meter = P x 3.6/v x width of implement

= (123) (3.6)/ 0.75

≅ 0.59 kW

Total draft = draft per meter x implement draft

= 1.04 x 0.54

= 0.614 kN

= 614N

The force transmitted by man is 1036N which is greater than the total draft against the implement. Hence, soil resistance against the coulter was overcome by the workload in the forward motion.

4.1.3 **Determination of Power Consumed**

The power consumed during an operation is given by the expression:

Power = force x working speed

Where:

Force = Mass x gravitational acceleration

M = Mass of the machine = 18.5kg.

Working speed = 0.75m/s.

Therefore P = (18.5x 9.81) x (0.75)

= 136 W

Design power = 146W.

Since the effective power (136W) is less than the design power (146W), it is evident that less power could be required to operate the machine.

7. The rate of seed damage obtained for the crops considered are.

a. Maize = 1.67%

b. Millet = 1.0%

c. Guinea corn = 0.9%

d. Cow peas = 2.22%

e. Groundnuts = 2.8%

8. The seed spacing evenness of the machine is 0.987 and the evenness of seed spacing is 0.013.

9. The germination rate for maize crop, as planted by the machine, is 96.88%
10. The seed breakage efficiency of the machine is 99.83%
11. The effective field capacity of the machine was obtained as 0.131ha/hr, the theoretical field capacity was obtained as 0.54ha/hr and the field efficiency of 85.07% was obtained.
12. The implement draft was obtained as 0.59 kW/hr. and the total draft as 614N.
13. The designed power is 146W and the power consumed when operating the machine is 136W.

The results obtained from the tests carried out indicate that machine deposits seed at approximate constant spacing based on the type of crop being sown. The furrow opener provides a constant working depth as required depending on the setting of the working depth.

The rate of seed damage by the machine is low even though rate of damage depends on seed size. The larger the seed, the higher the rate of damage. The rate of damage for seed of millet and Guinea corn is lower (1% proximately) than those of maize, cowpeas and Groundnuts (2-3% approximately). The germination rate is high (96.88%) because of the quality of seed material used and low rate of damage to seed by the machine.

The machine has high efficiency (96.07%), low draft (0.6 kWh approx) and low operating power (146W). This indicates that the machine is working correctly as the design intends.

The working capacity of the machine is 1.82 ha/hr. This indicates a high working capacity. The machine is easy to handled and adjust.

The grain box contains the following weights and number of grains when completely filled with grains:

1. Maize = 8850g. = 8.85 kg. This is an equivalent of 44,697 grains
2. Groundnuts (erect type) = 6672g = 6.672kg. This is an equivalent of 14,860 grains.
3. Groundnuts (creeping type) = 13482g = 13.482kg. This is an equivalent of 42,396 grains

4. Millet = 8208kg = 8.208kg. This is an equivalent of 820, 800 grains

5. Cowpeas (creeping type) = 7434g = 7.434kg. This is an equivalent of 38, 320 grains

6. Guinea corn = 8838g = 8.838kg. This is an equivalent of 58, 920 grains.

When using the machine on the field, the following spacing and quantity per hole for the following crops were obtained.

Millet – 5 seed per hole at intervals of 50cm.

Guinea corn – 5 seed per hole at intervals of 50cm

Ground nuts – 1 seed per hole at intervals of 25cm

Maize - 2 seed per hole at intervals of 25cm.

These results highly conform to those published by Akinsonmi in 1981

CHAPTER FIVE

5.0 CONCLUSION AND RECOMMENDATION

5.1 Conclusion

A multi-crop sowing machine was designed, constructed and tested. It has high working capacity of 1.82 ha/hr. and an efficiency of 95.07%. It can plant maize at the rate of 67kg/ha with low seed breakage (0.17%) and high germination rate (96.88%).

The seed spacing evenness of the machine is 0.987.The furrow opener can dig up to a depth of 6cm. The effective and theoretical field capacities of the machine are 0.131ha/hr. and 0.154ha/hr. respectively, with an implement draft of 0.59 kW/hr. It has a designed power of 146W and an operating power of 136W.

The machine can be used on different soils of North Western Nigeria and similar conditions where the soil types are cambisols, fluvisols, Nitrosols and Acrisols. Their textures are predominant sand, sandy loam, loamy sand sometimes with gravelly sub-soils. They are deep well drained soils with 75% to 92% sand, 1% to 11% silt, and 4% to 20% clay (Graham, 2004). The soil of Northern Nigeria is shown on figure 14. The machine can also be used in Eastern Nigeria and any where the soil characteristics are the same as those stated above. It can be used to plant most crops cultivated in the upland region of these areas such as Millet, Guinea Corn, Maize, Cowpeas and Ground nuts. The capacity of the grain box is approximately 8kg ($0.00844m^3$) for most of the crops mentioned above.

The speed range of the machine is 0.6m/s to 0.75m/s with a seed damage rate of 1.67% for maize seed, 0.65 to 0.75m/s with a seed damage rate of 1% for Millet, 0.65-0.75m/sec with a damage rate of 0.9%, 0.58-075m/sec with a damage rate of 2.22% for cowpeas, 0.6-075m/sec with a damage rate of 2.8% for Groundnuts. This means that the machine can be operated at speed range of 0.58 to 0.75m/sec with very low rates of seed damage. The machine is six times faster when compared with manual sowing.

Finally, it can be concluded that this machine can be used to substitute the use of local tools, hands and feet used in traditional method of planting.

5.2 Recommendation

1. The machine is recommended for use by small scale farmers in Northern Nigeria, Western and similar conditions since it can be used to plant most crops grown in the upland region

2. The machine should be cleaned and well stored at the end of the planting season.
3. The chain should be lubricated from time to time and the tension should be adjusted adequately to prevent the chain from jumping off-the sprocket during work.
4. All slack nuts and bolts should be re-tightened at the end of the day's work
5. The machine should not be used in water logged area for example in the swampy areas.
6. A chain cover is recommended for the machine
7. Damaged bearings and lost nuts and bolts should be replaced immediately.
8. The machine is an off-road (farm machine) type and should not be pashed or towed on concrete surfaces or tarred road.
9. Investigation on fertilizer application-at the time of planting by the machine is desirable.

Rewrite!

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APPENDIX A
SKETCHES OF THE COMPONENTS OF THE MACHINE

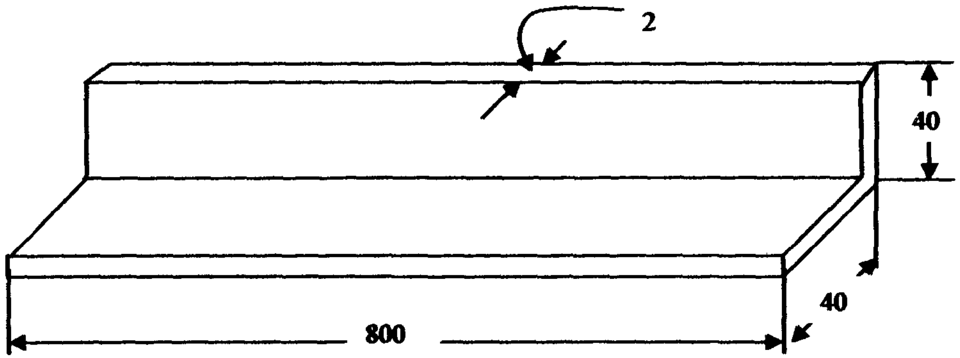


Figure i a. Left and right sides of the frame

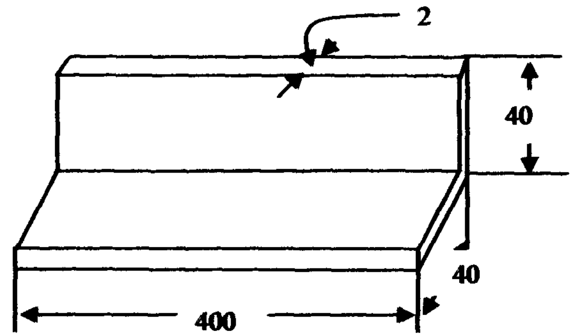


Figure i b . Front and rear sides of the frame

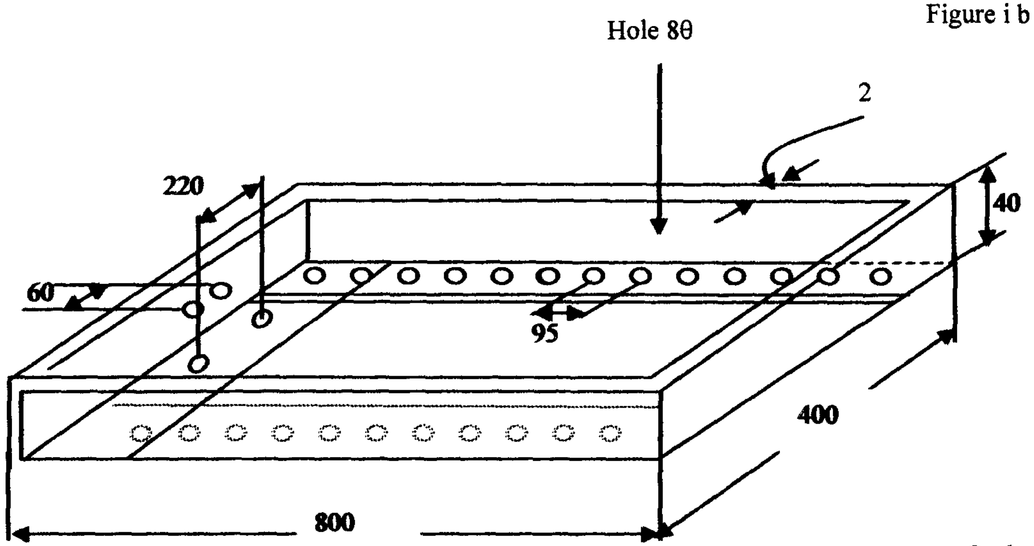


Figure i c: Frame Assembly

Scale: 1:8
Dimensions in mm.

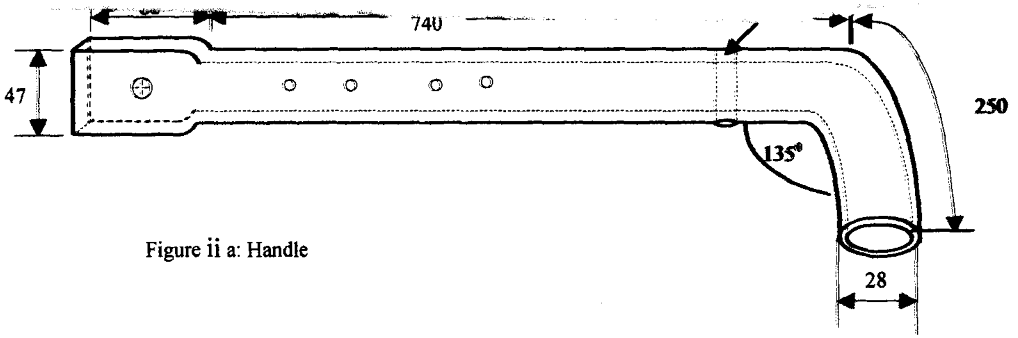


Figure ii a: Handle

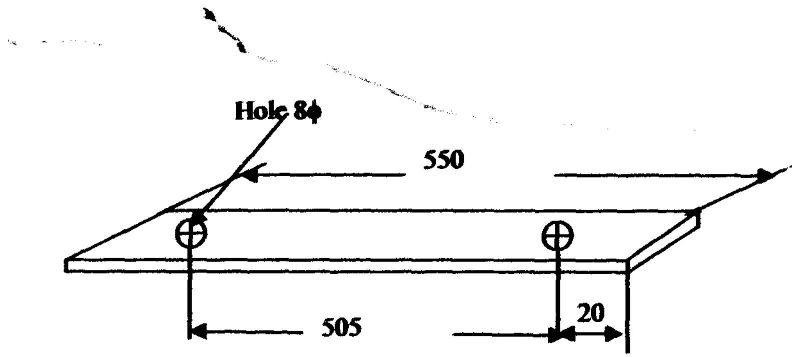


Figure ii b Cross bar

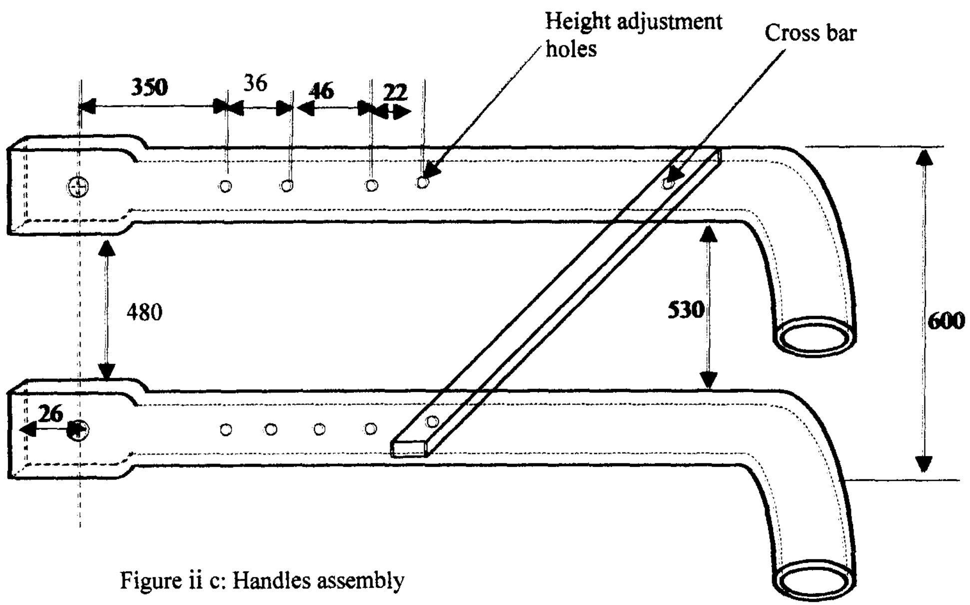


Figure ii c: Handles assembly

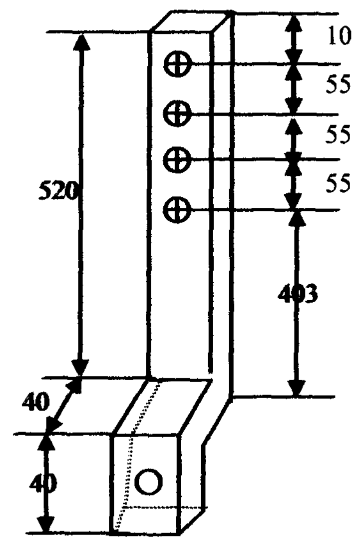


Figure ii d: Handles height adjustment device.

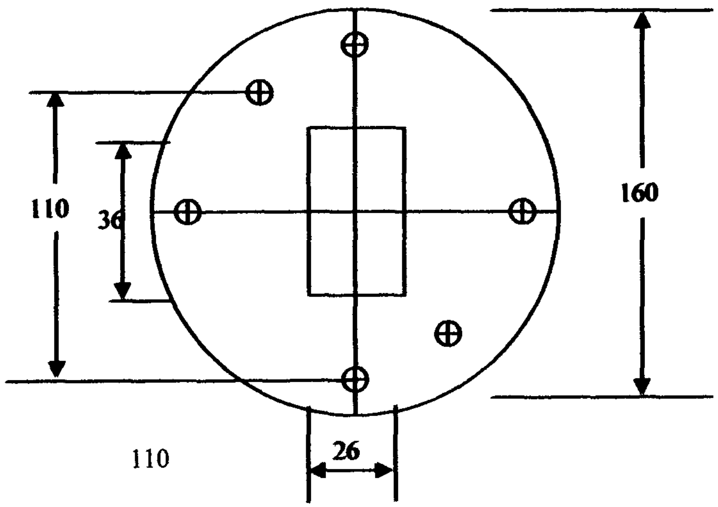


Figure iii a: Bottom Plate.

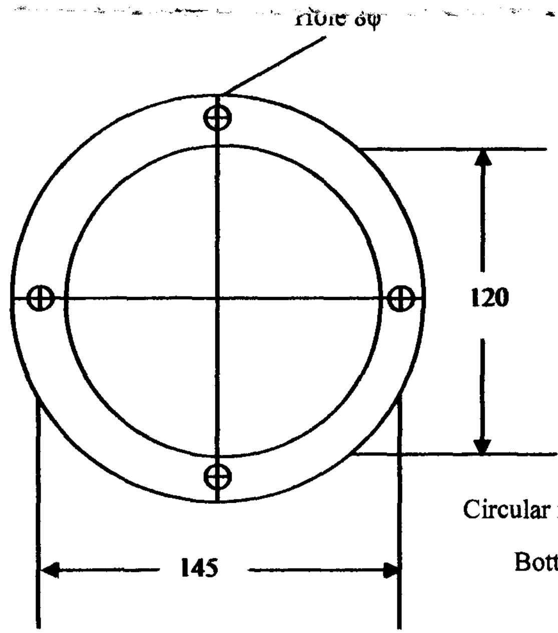


Figure iii b: Circular ring

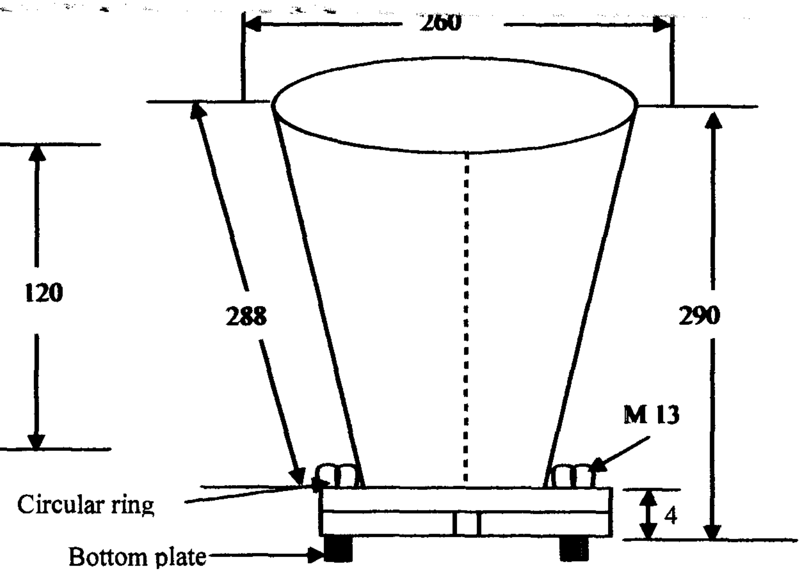


Figure iii f. Hopper assembly

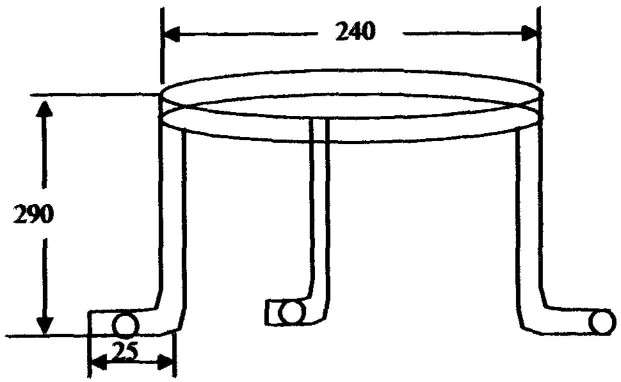


Figure iii c: Hopper guide

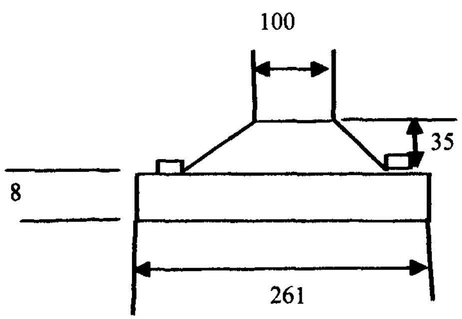


Figure iii Hopper lid

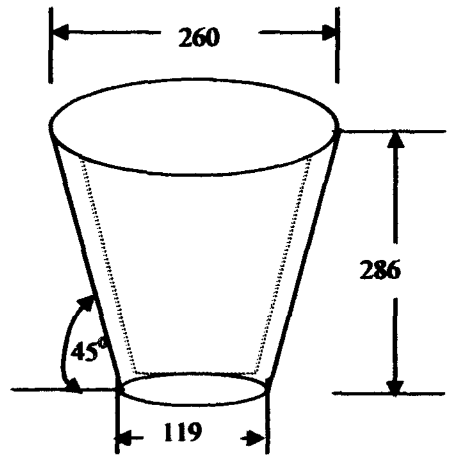


Figure iii e: Bucket

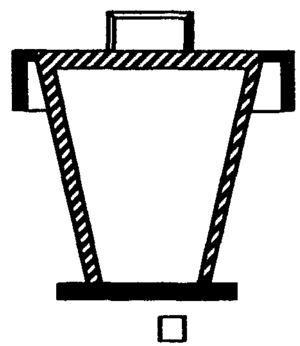


Figure iii g. Sectional View

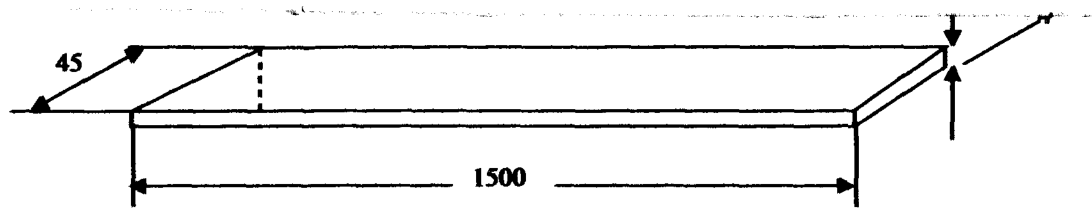


Figure iv a: Iron bar to be formed into a circular ring.

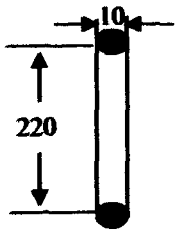


Figure iv b: Spoke.

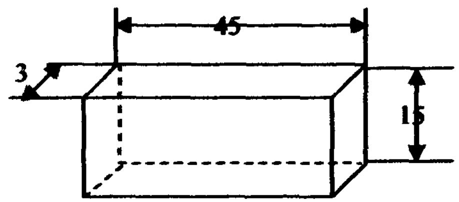


Figure iv c: Strake

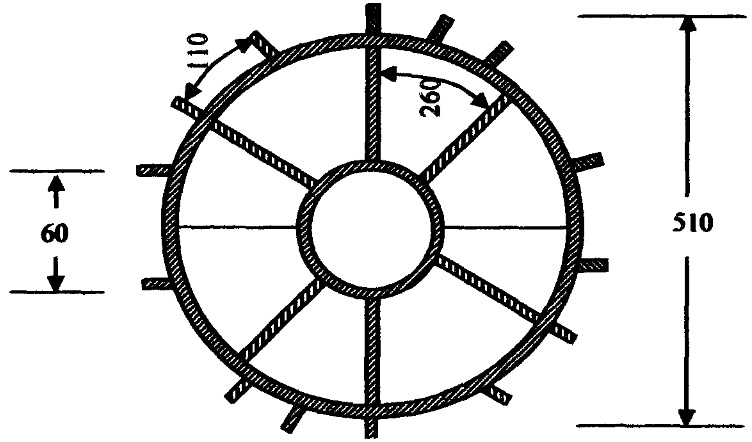


Figure iv e: Cross section of wheel.

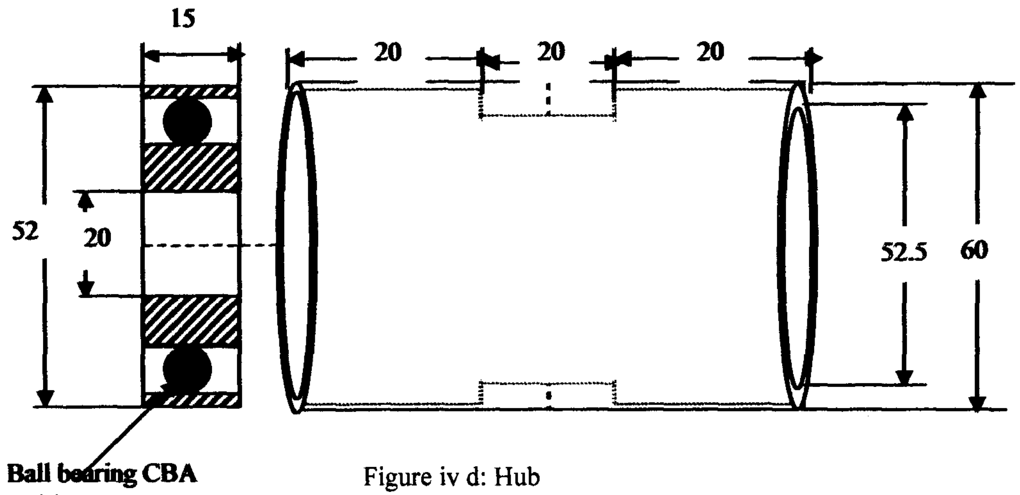


Figure iv d: Hub

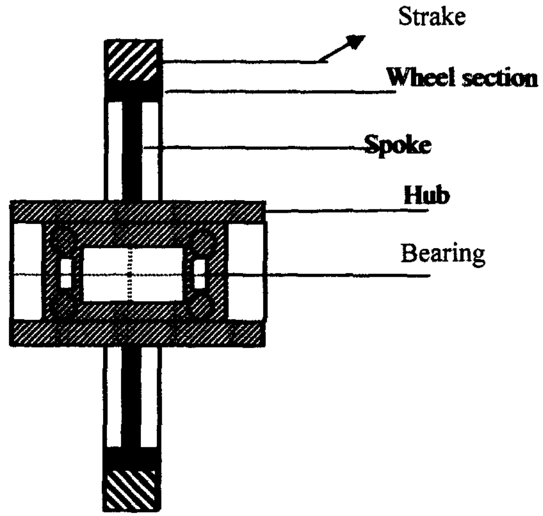


Figure iv f. Cross section of assembled wheel.

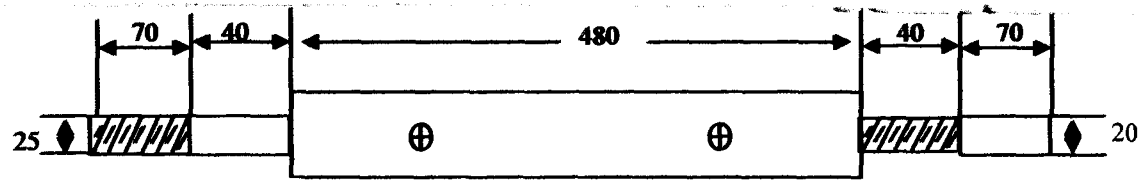


Figure iv g (i):- front wheel shaft

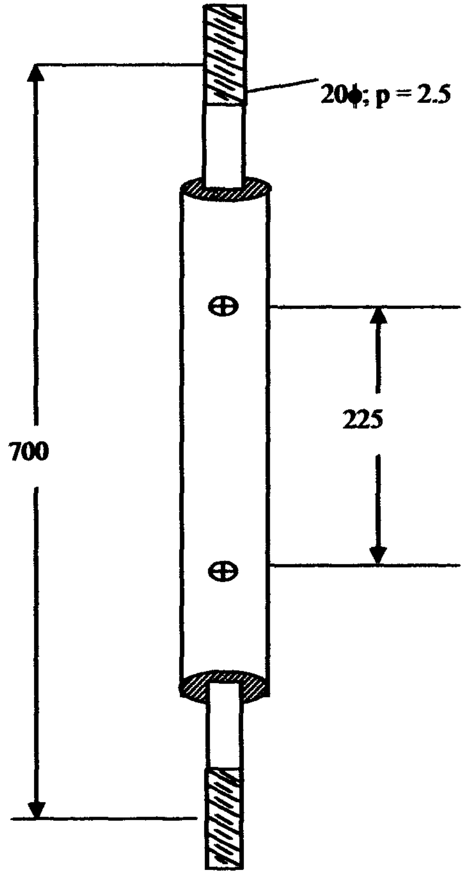
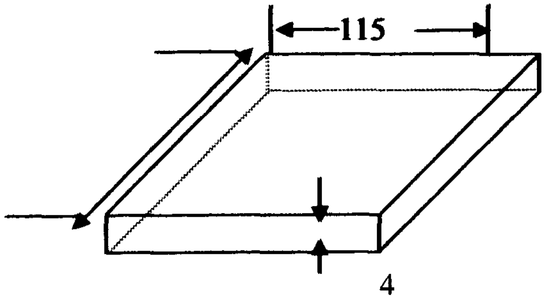


Figure iv g(ii) Front Wheel Shaft



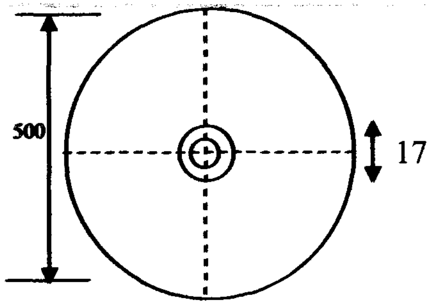


Figure v b: circular plate

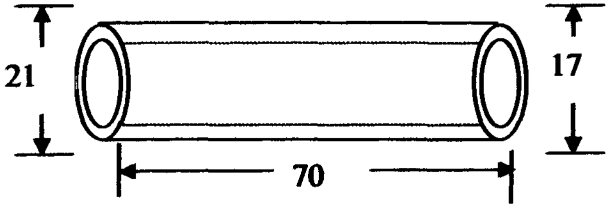


Figure v c: Steel pipe

Figure v a: Rectangular steel plate

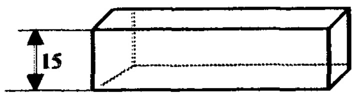


Figure v d: Strake 20Nos.

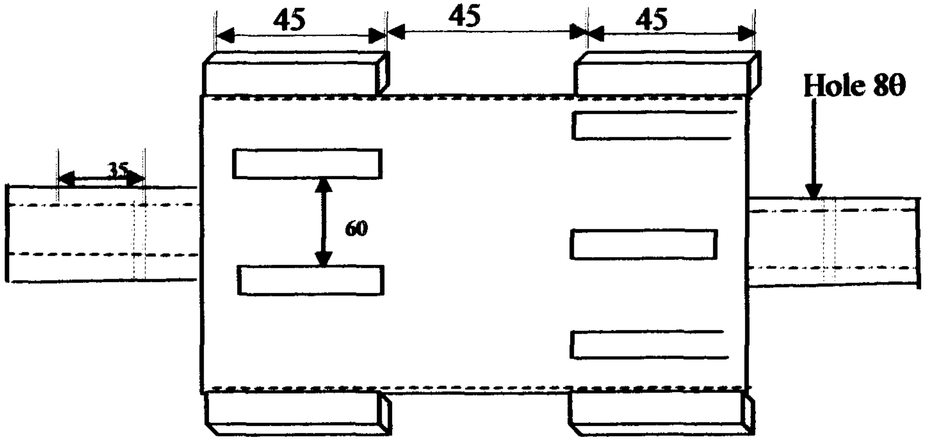


Figure v e: Driving wheel

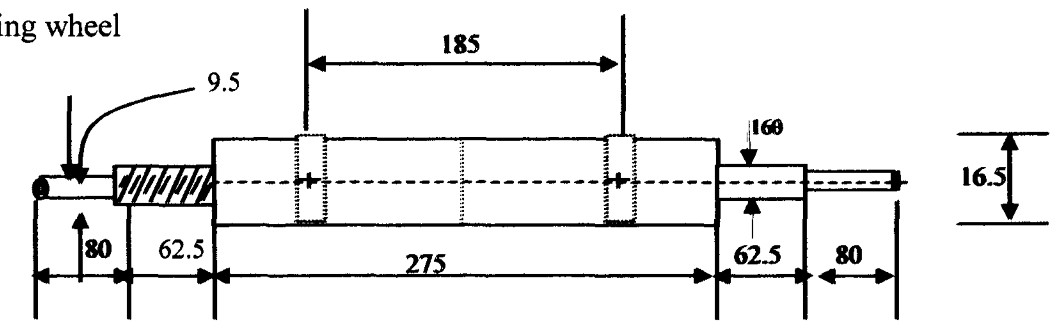


Figure v f: Driving shaft

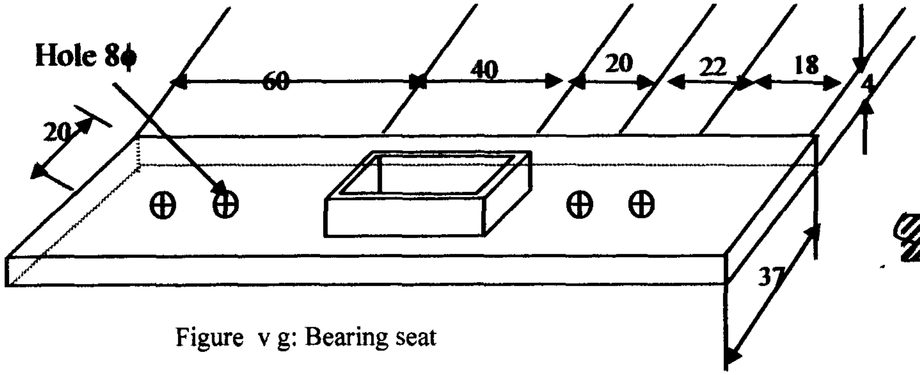


Figure v g: Bearing seat

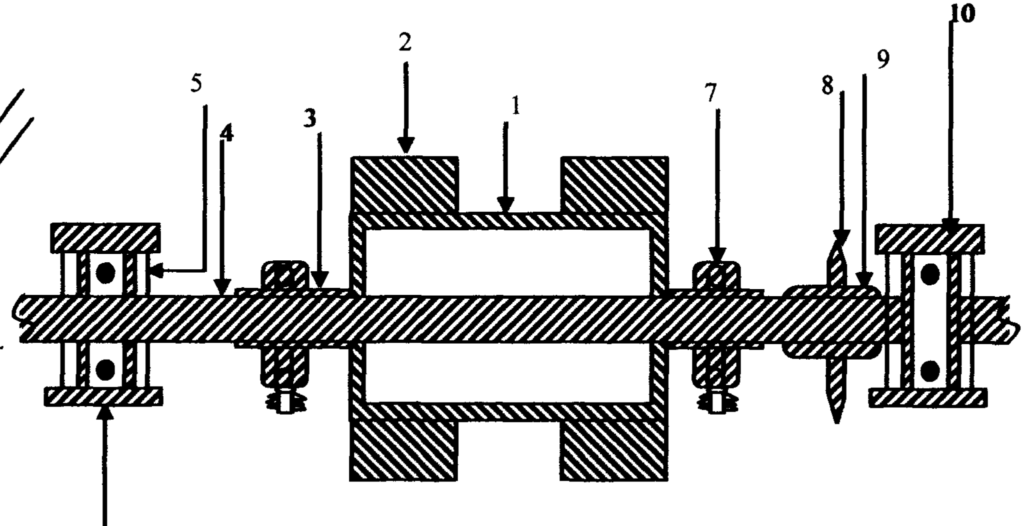


Figure v i: Sectional view of the driving wheel assembly

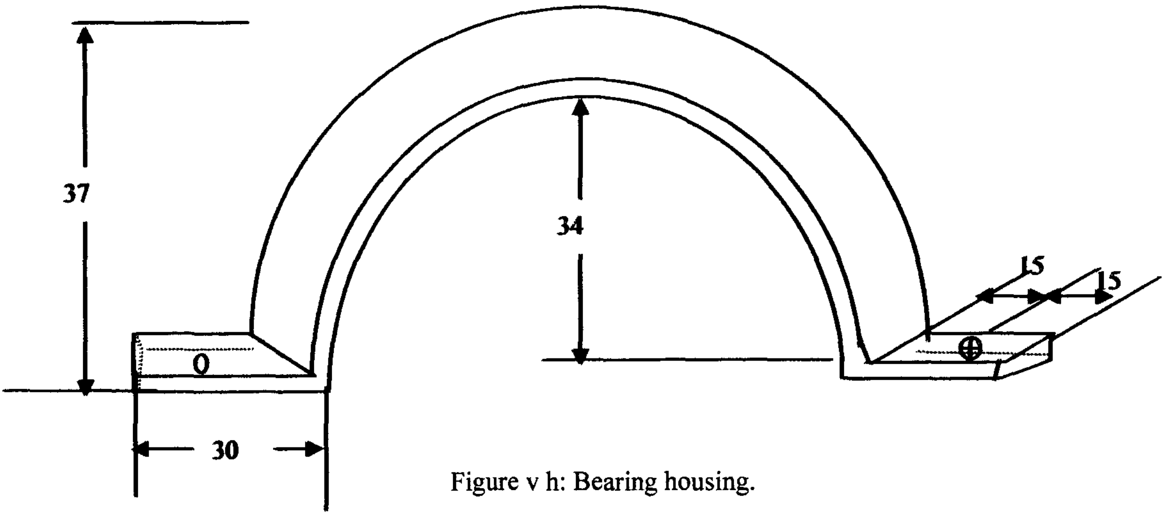


Figure v h: Bearing housing.

KEY

1- Driving wheel
2- Strake
3- Hollow pipe
4- Driving shaft
5- Bearing
6- Bearing seat
7- Nut and bolt
8- Driving sprocket
9- Locking nut
10- Bearing housing

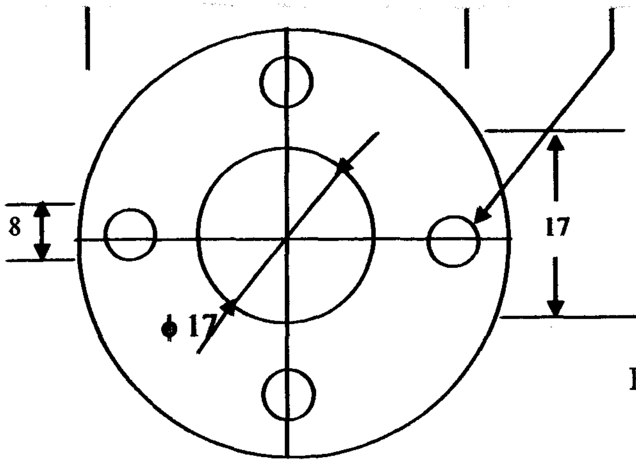


Figure vi a: Circular plate

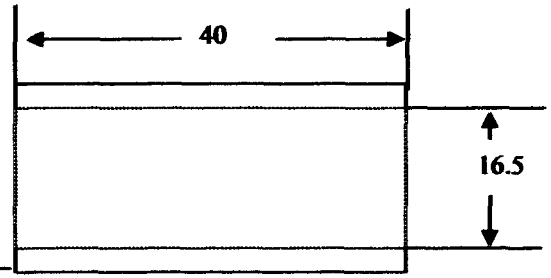


Figure vi b: Hollow pipe

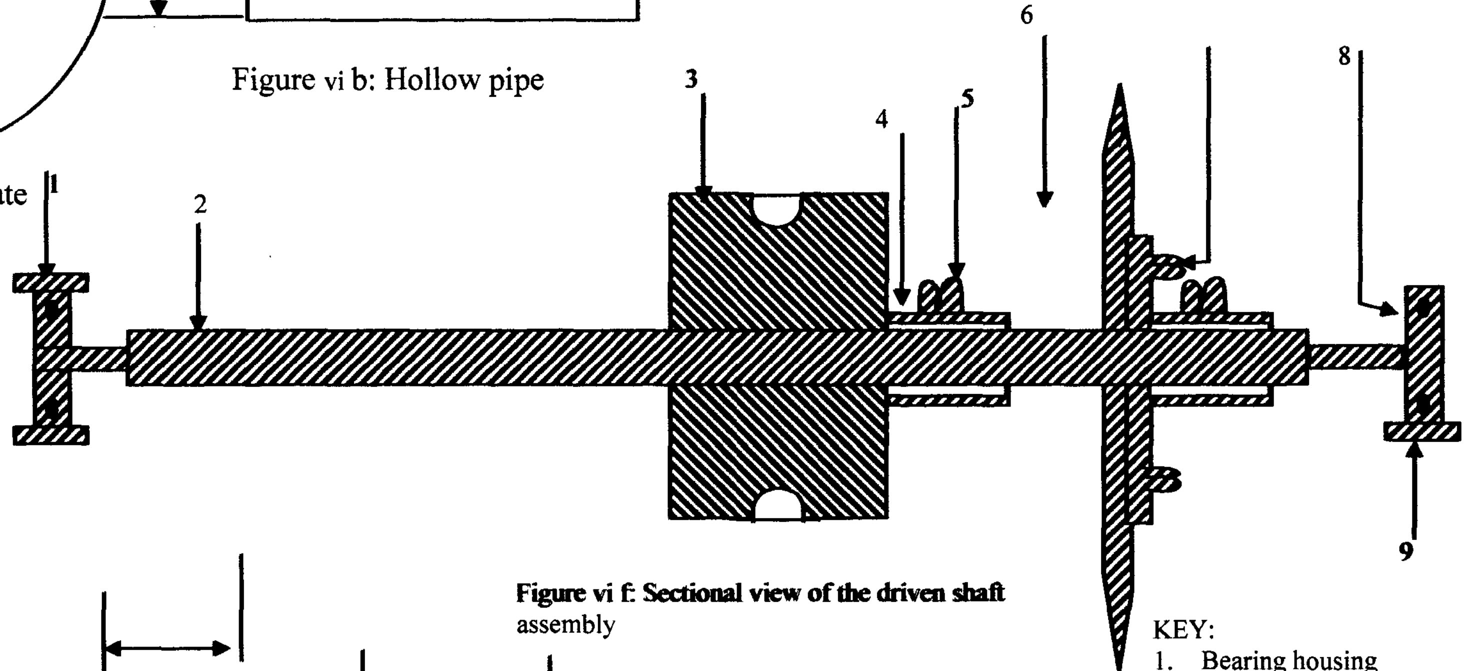


Figure vi f: Sectional view of the driven shaft assembly

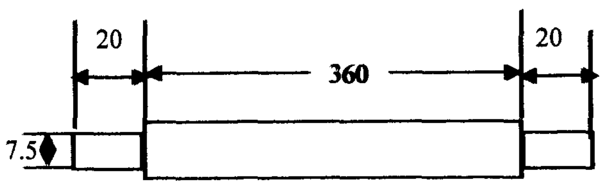


Figure vi c: Driven shaft

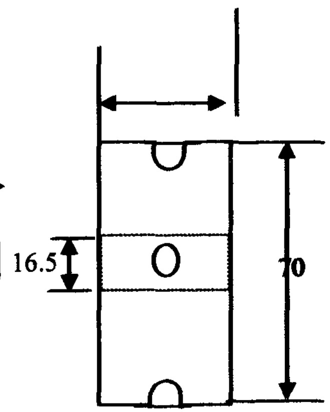


Figure vi d : Seed metering device

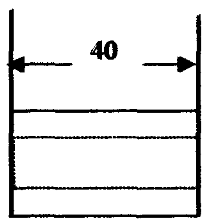


Figure 6e: Pipe

KEY:
1. Bearing housing
2. Shaft
3. Seed meter
4. Pipe
5. Key
6. Sprocket
7. Circular plate
8. Bearing
9. Bearing seat

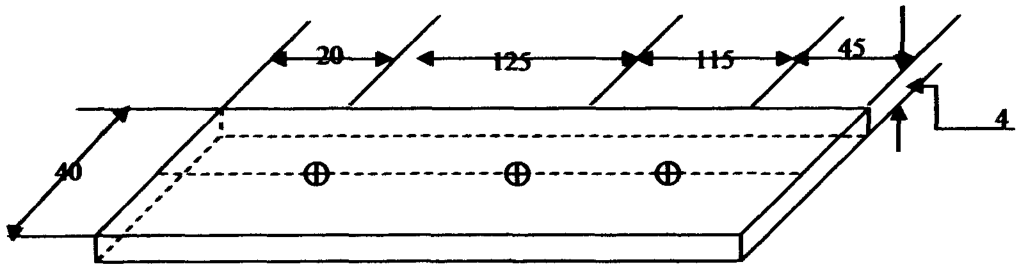


Figure viii a: Frame's rear support.

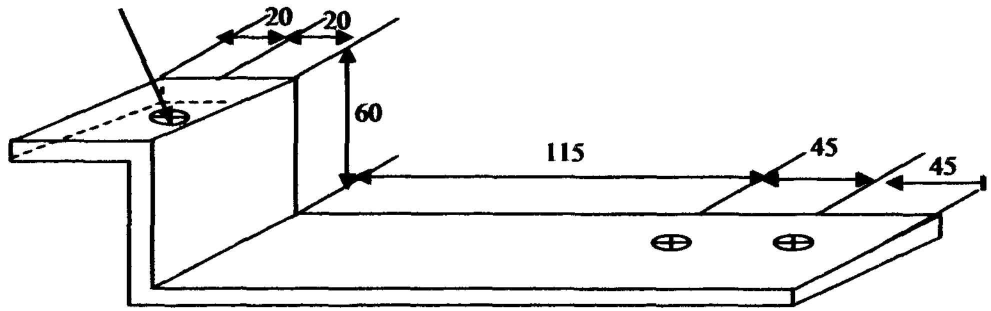


Figure viii c: Hopper support.

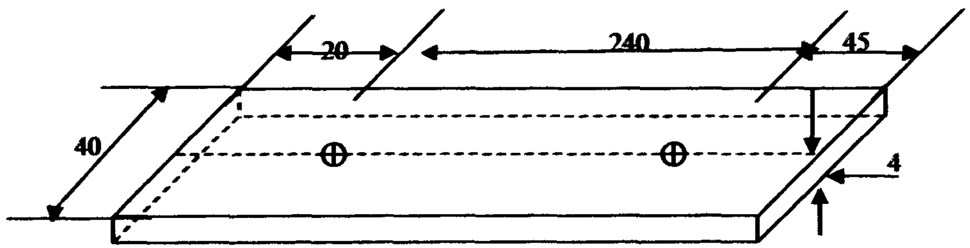


Figure viii b: Frames rear support.

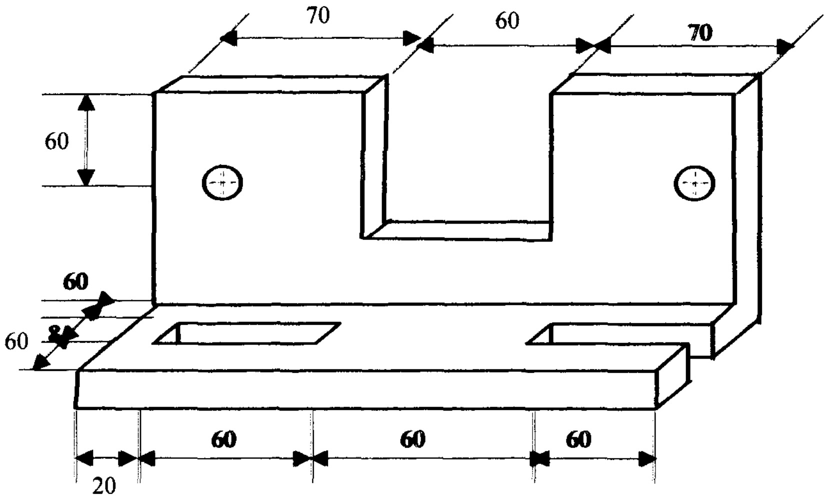


Figure viii d : Chain adjustment device/driving shaft support

COUPLING ACCESSORIES

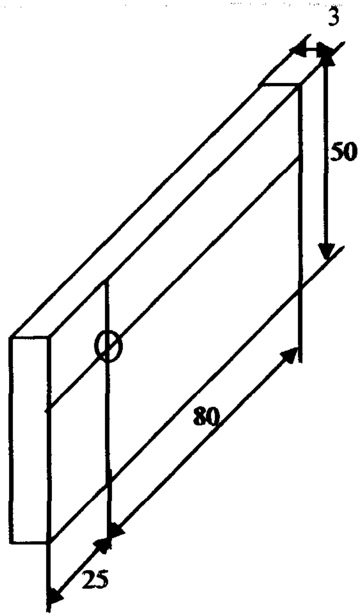


Figure. viii a: Rectangular bar

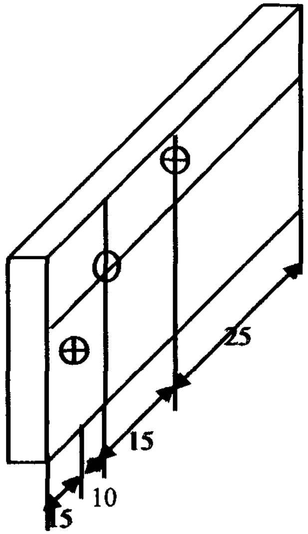


Figure viii b: Rectangular bar

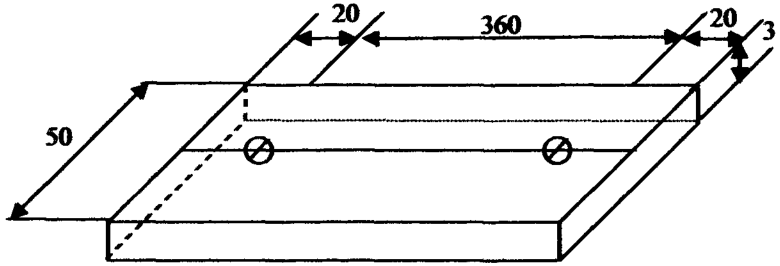


Figure. viii c: Rectangular bar.

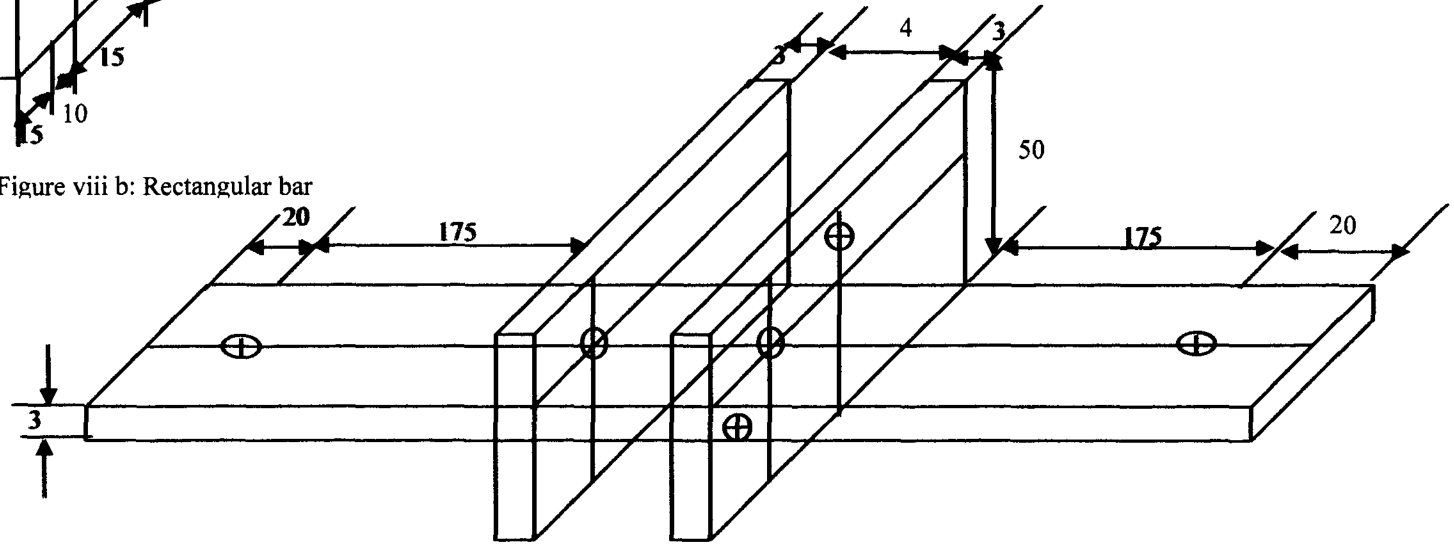


Figure viii d: Furrow opener bracket/depth adjustment device.

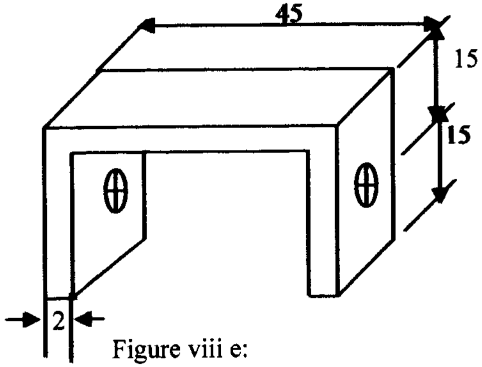


Figure viii e: Bracket

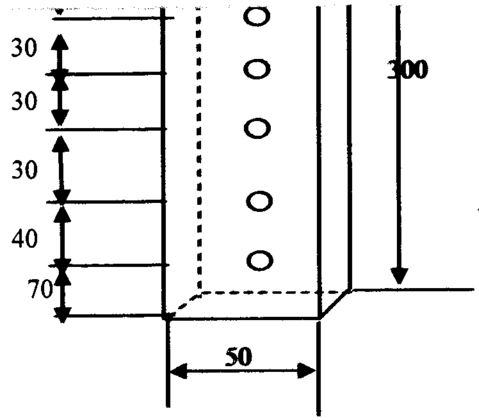


Figure viii f: Depth adjuster

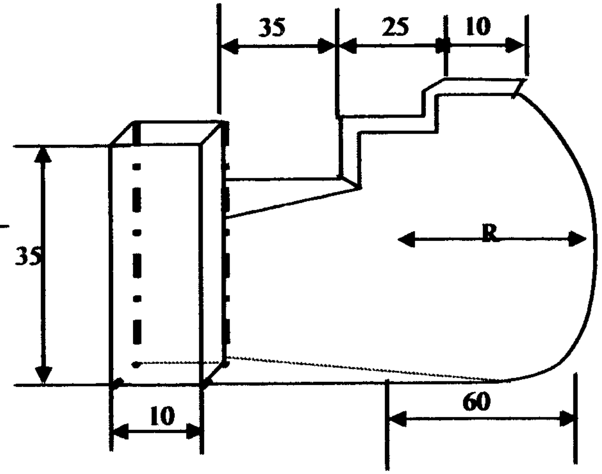


Figure viii h: Shear opener

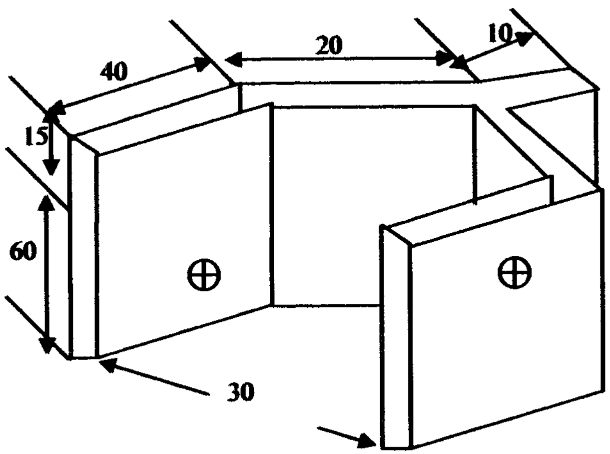
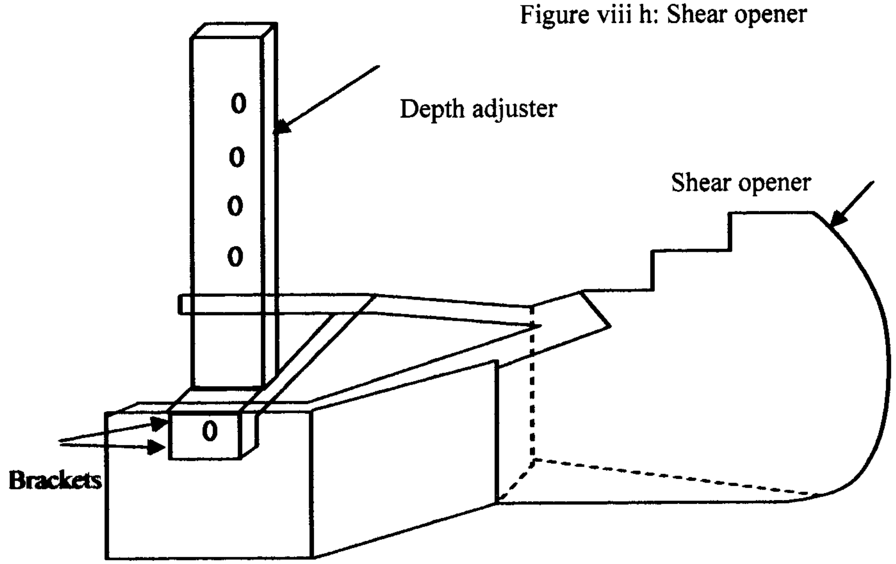


Figure viii g: Bracket



Figureviii i: Furrow opener assembly.

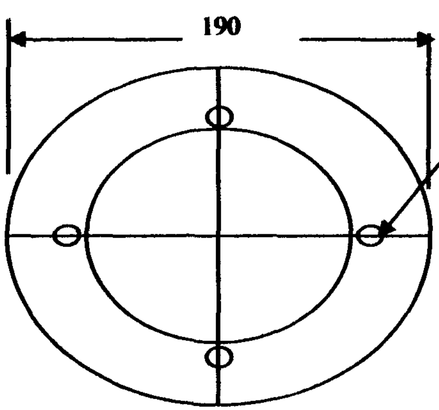


Figure ix a: Plain disc.

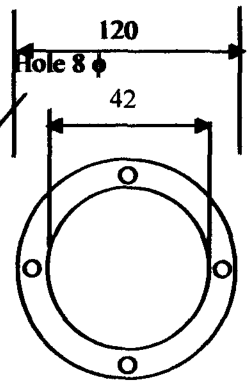


Figure ix b: Ring

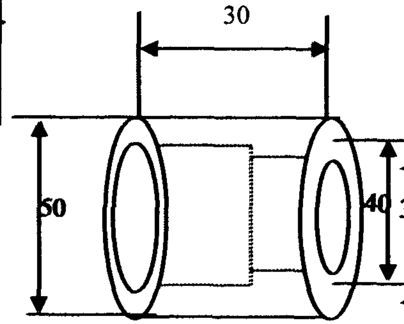


Figure ix c: Bearing housing.

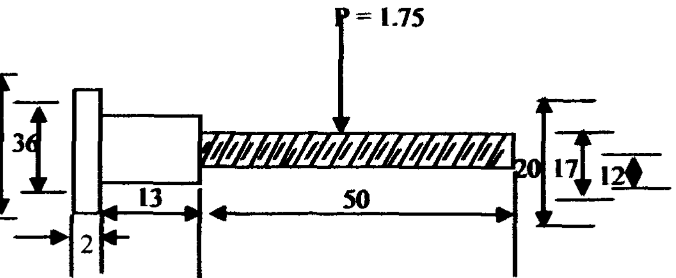


Figure ix d: shaft.

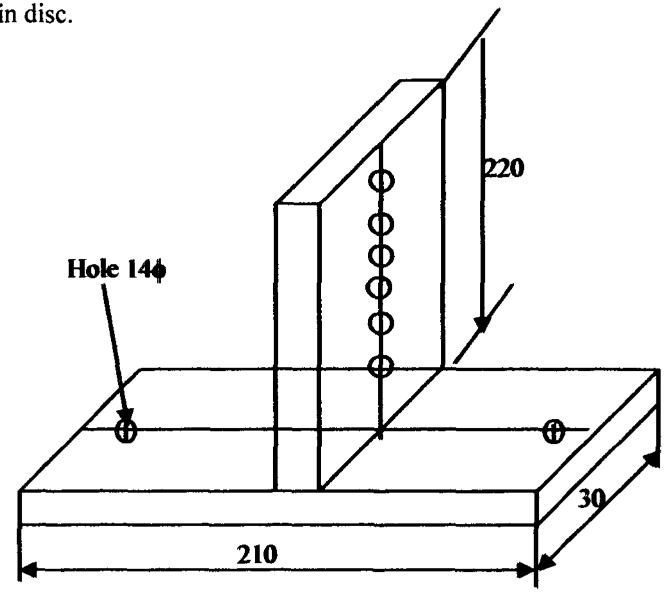


Figure ix f: Depth adjuster

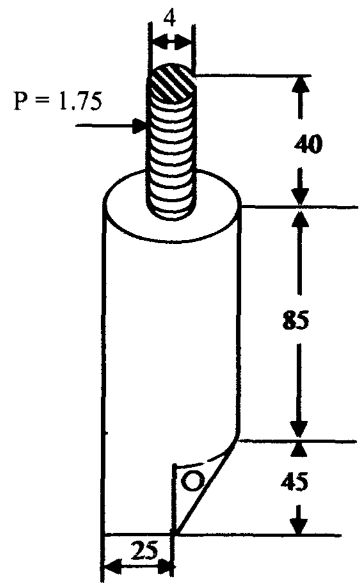


Figure ix e: Shank

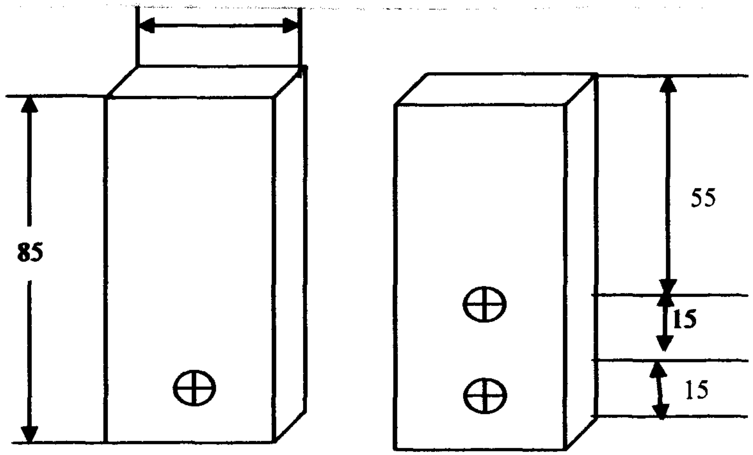


Figure 9g: Rectangular bars

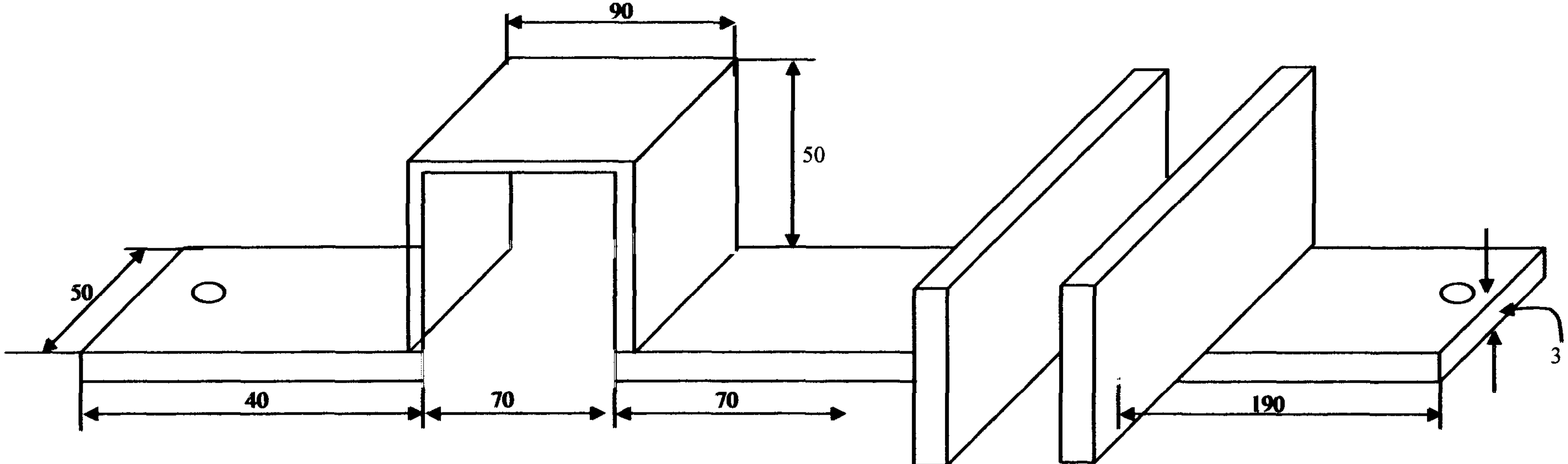


Figure ix h: Bracket

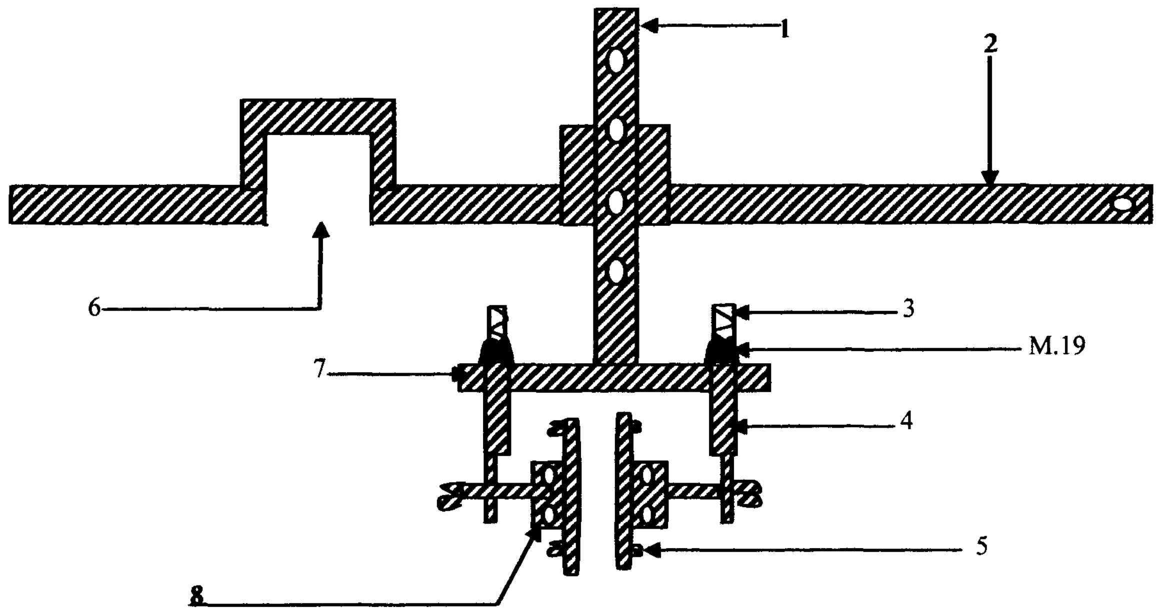


Figure ix i: Sectional view of furrow covering device assembly

KEY
1. Depth adjuster
2. Cross bar
3. Disc angle adjuster
5. Disc
6. Chain space
7. Supporting bar
8. Bearing housing

FURROW COVERED ASSEMBLY

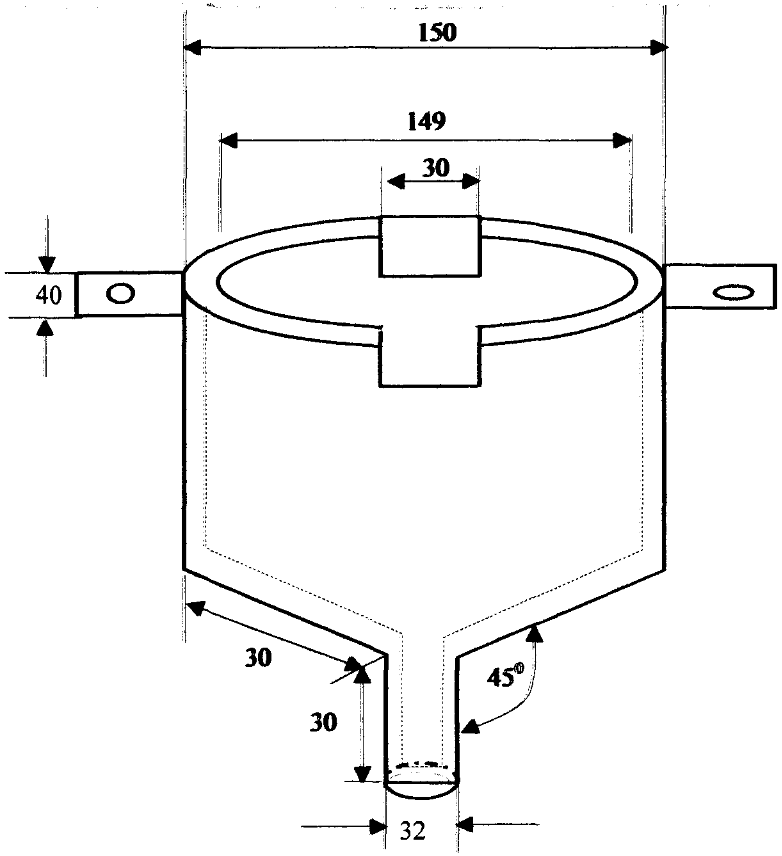


Figure x: Seed dispenser unit

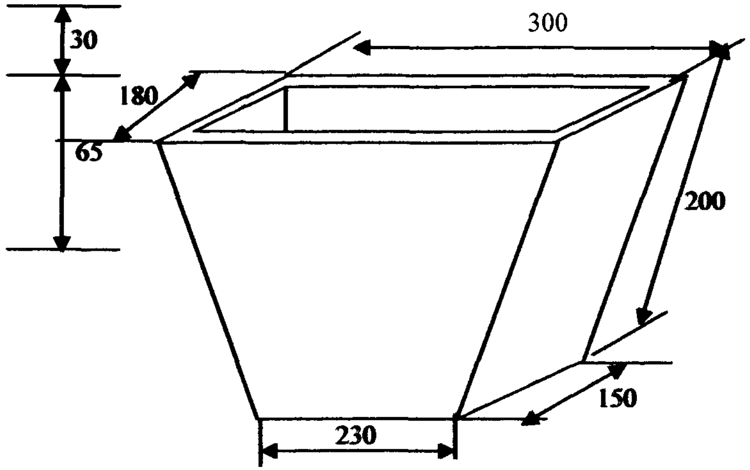


Figure xi: Housing for hopper

APPENDIX B

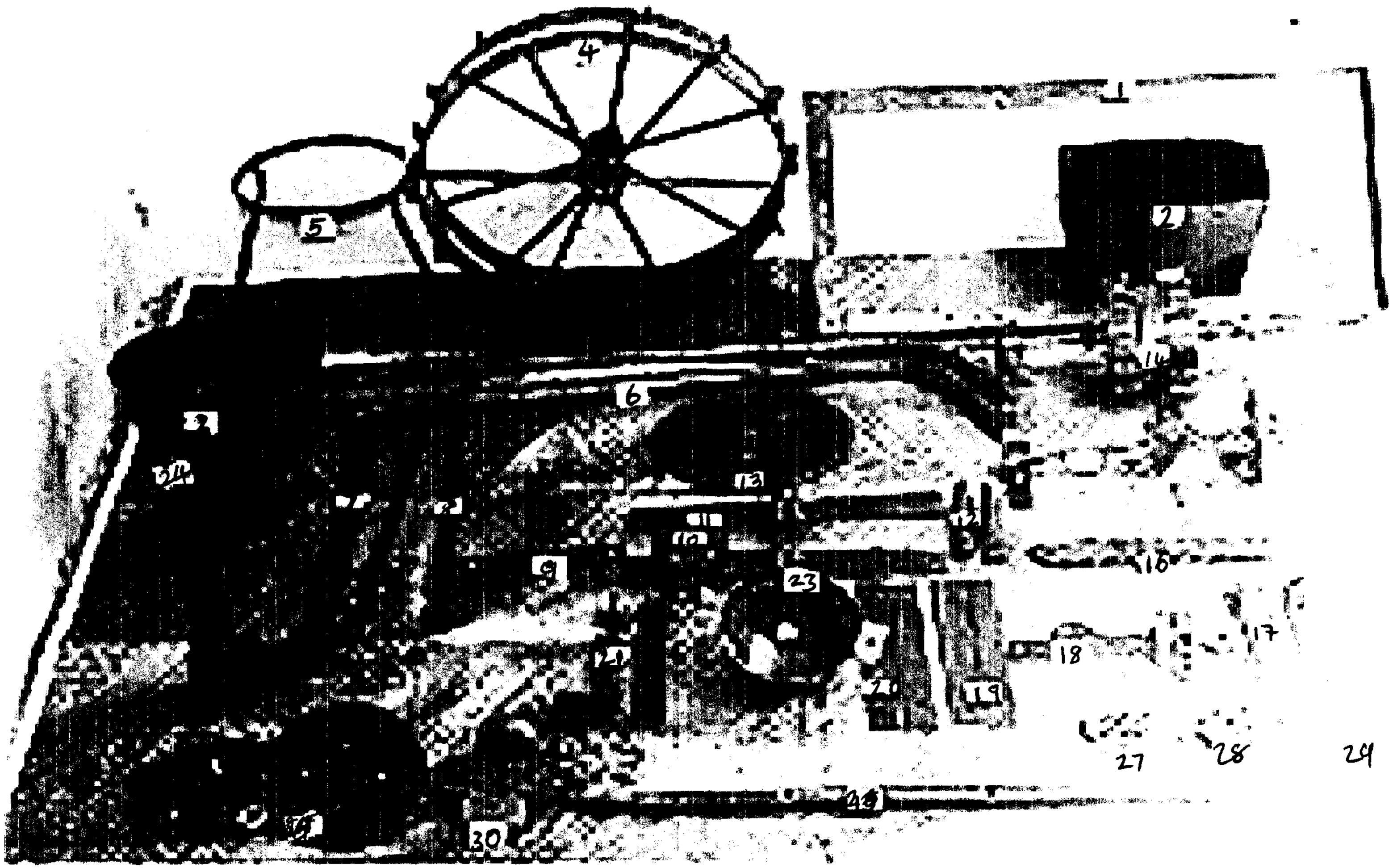


Figure xii Components of the Machine.

KEY

1. Frame
2. Grain box housing
3. Grain box
4. Front wheels
5. Hopper guide
6. Handles
7. Cross bar
8. Handle adjustment devices
9. Disc coupling
10. Furrow opener support bracket
11. Furrow coverer support bracket
12. Shanks
13. Hopper lid (cover)
14. Driving wheel assembly
15. Driven shaft assembly
16. Roller chain
17. Chain adjustment devices
18. Bearing seat
19. Bearing housing
20. Coupling accessories for rear side of driving wheel
21. Coupling accessories for front side of the driving
22. Hopper support bracket
23. Seed dispenser

24 Furrow opener

25 Plain discs
26 Front wheel shaft
27-29 Set of nuts bolts and washers
30 Seed metering devices

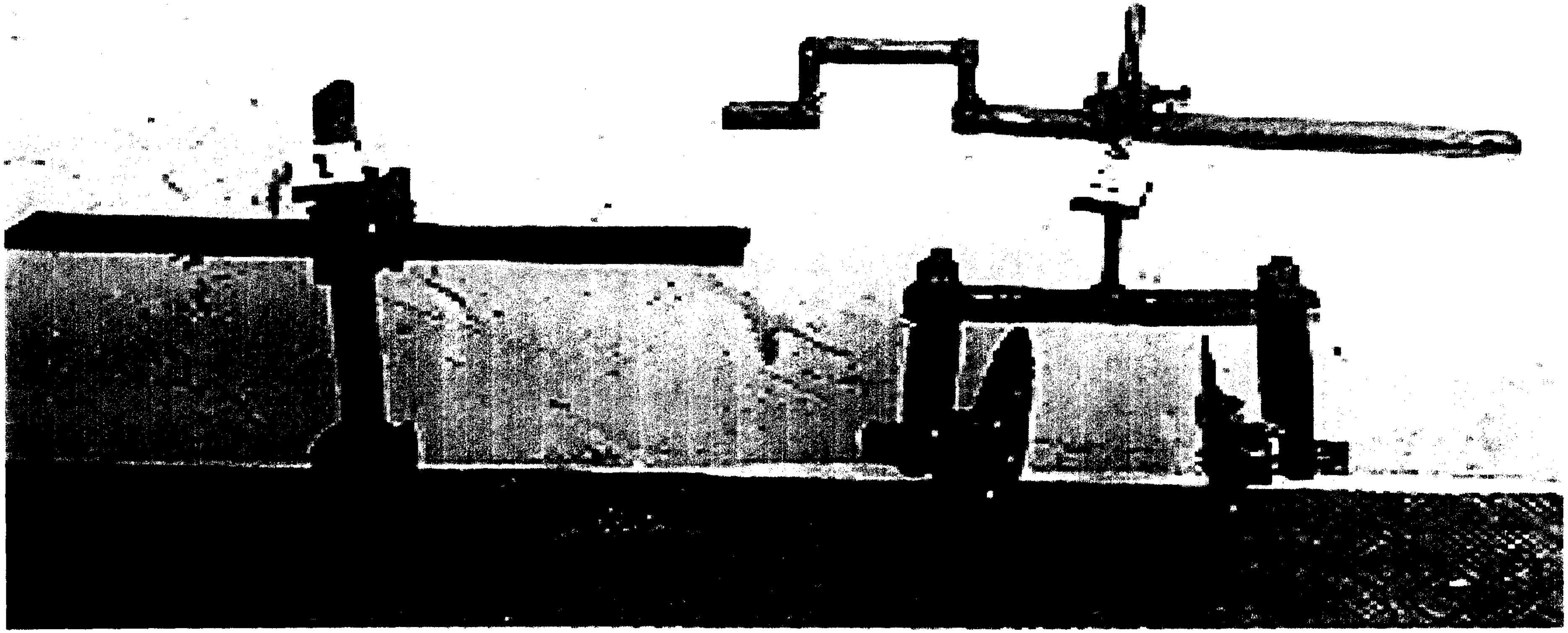


Figure xiii: Soil engaging components
1. Furrow opener assembly
2. Furrow coverer assembly

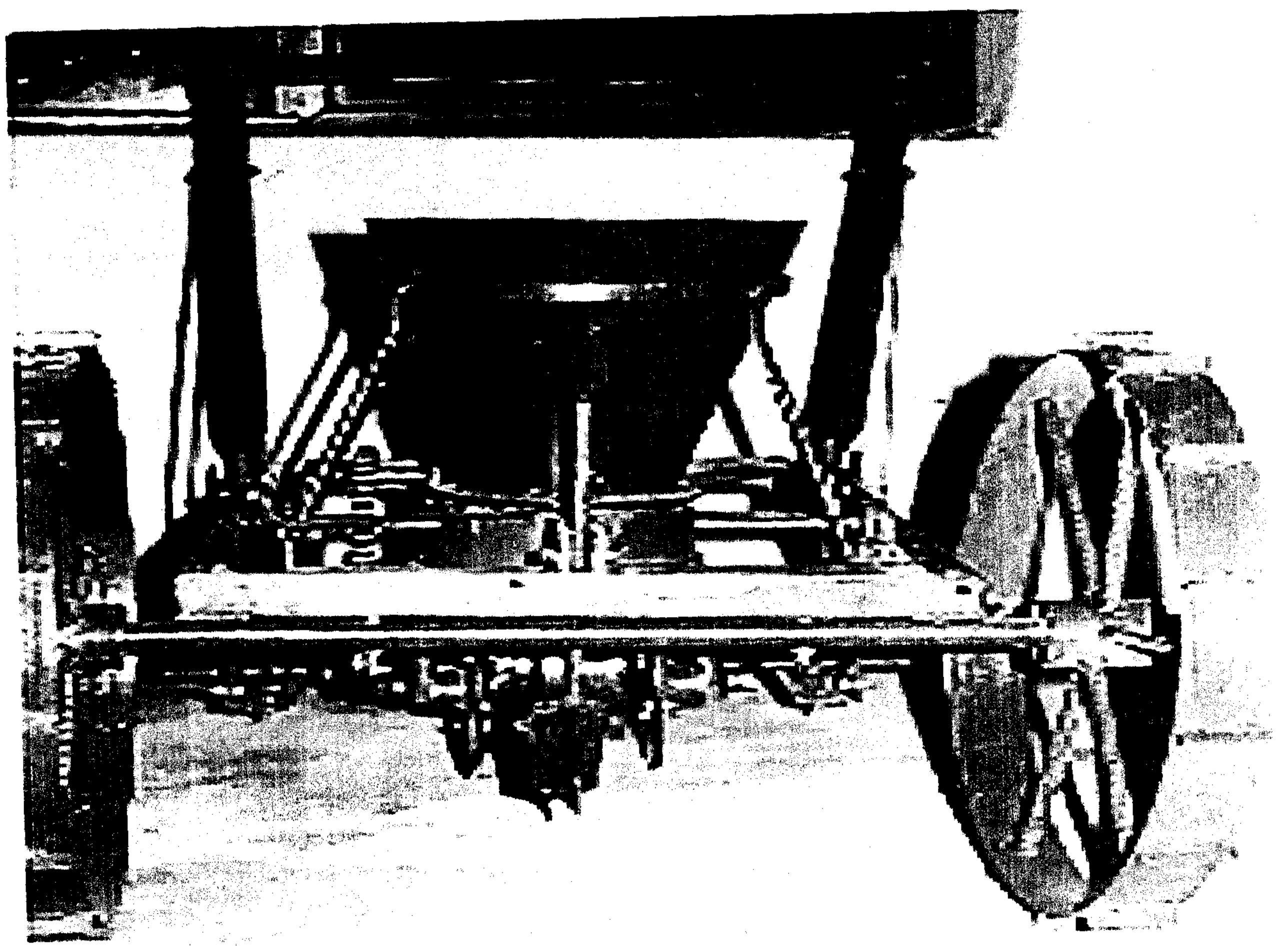


Figure xiv: Front view of the assembled machine

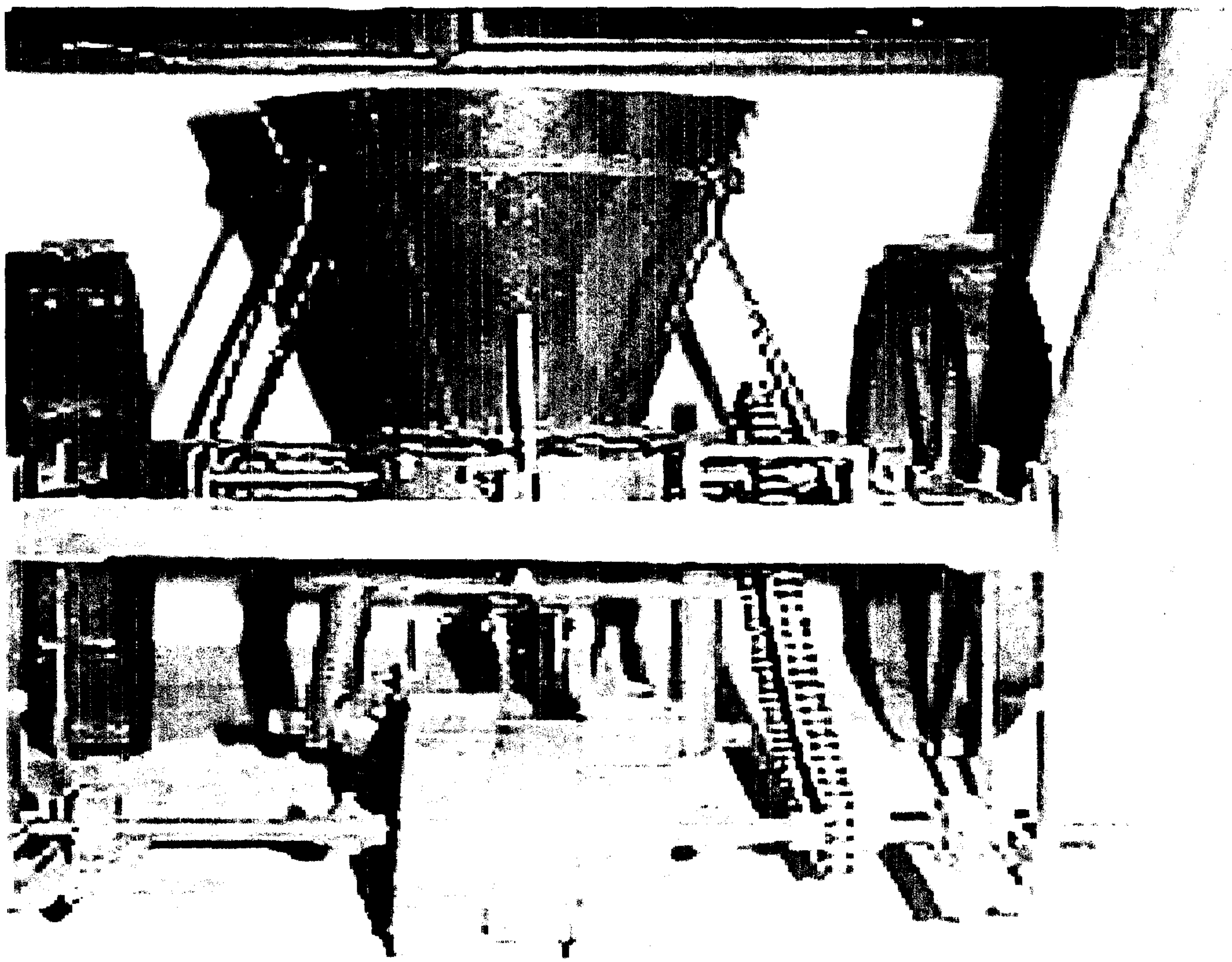


Figure xv: Rear view of the machine

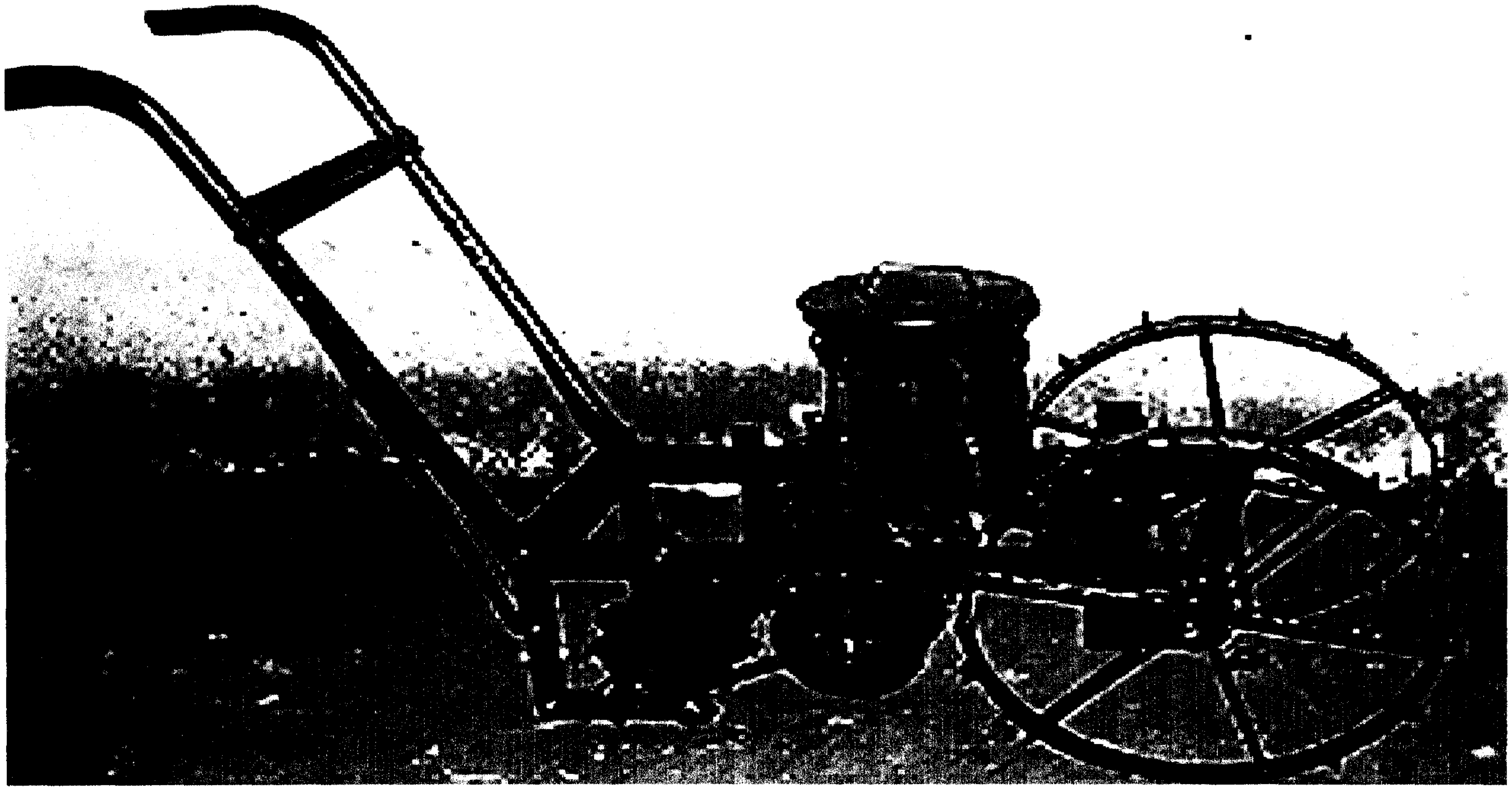
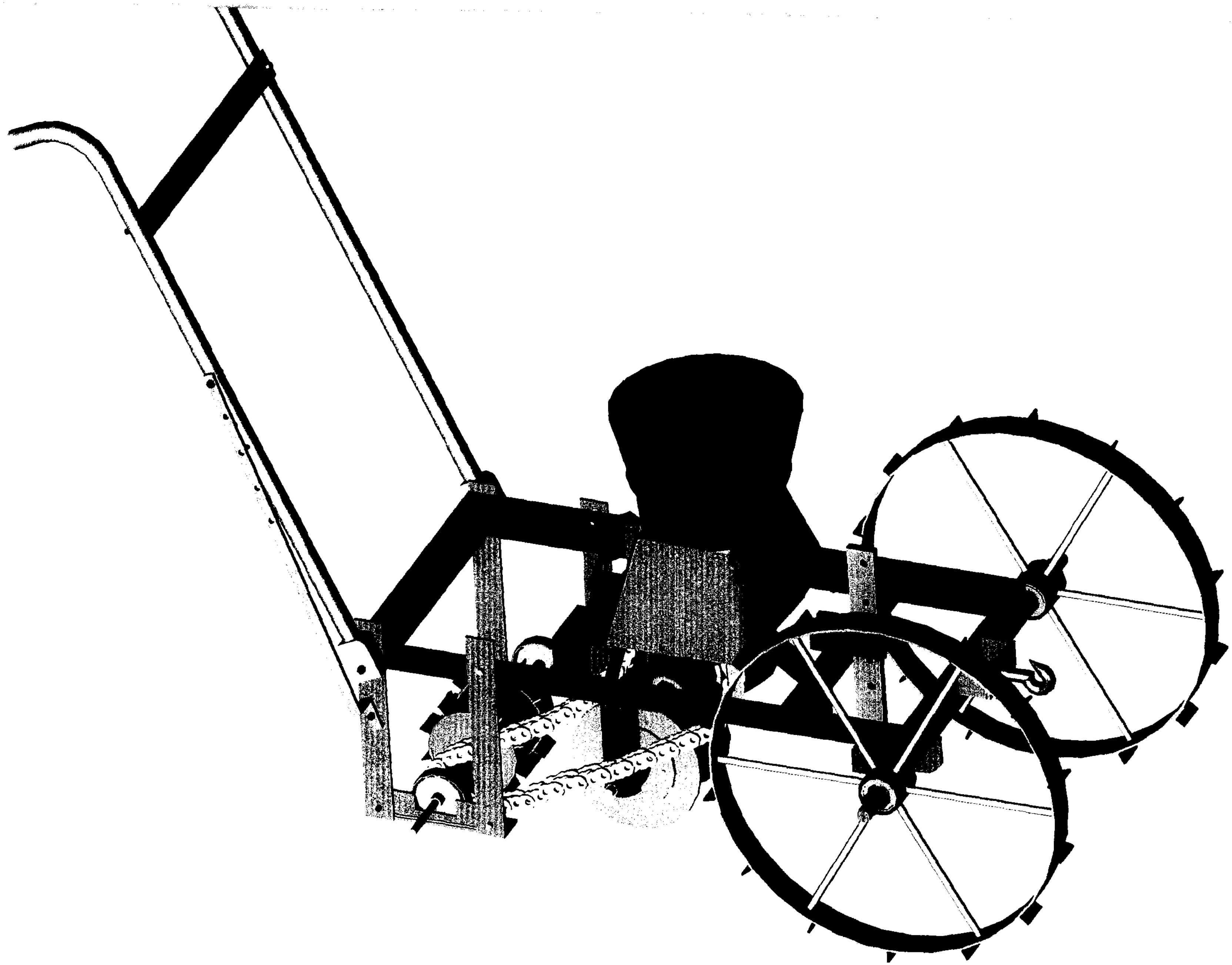
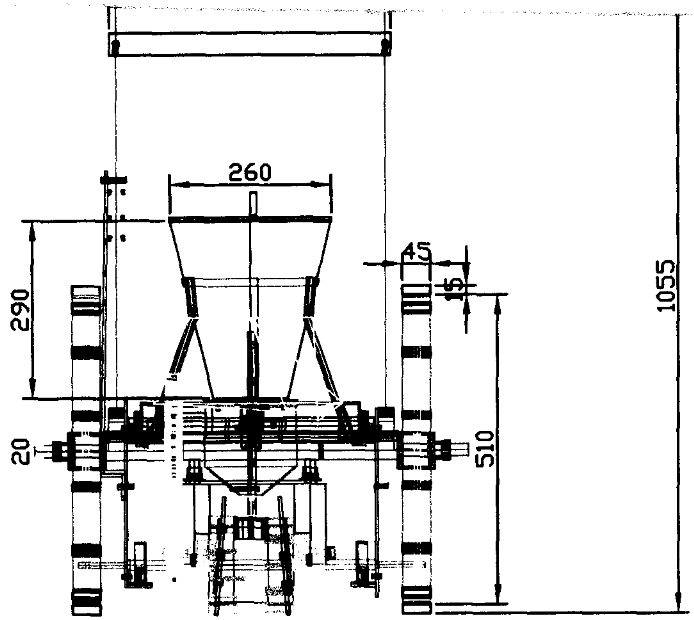


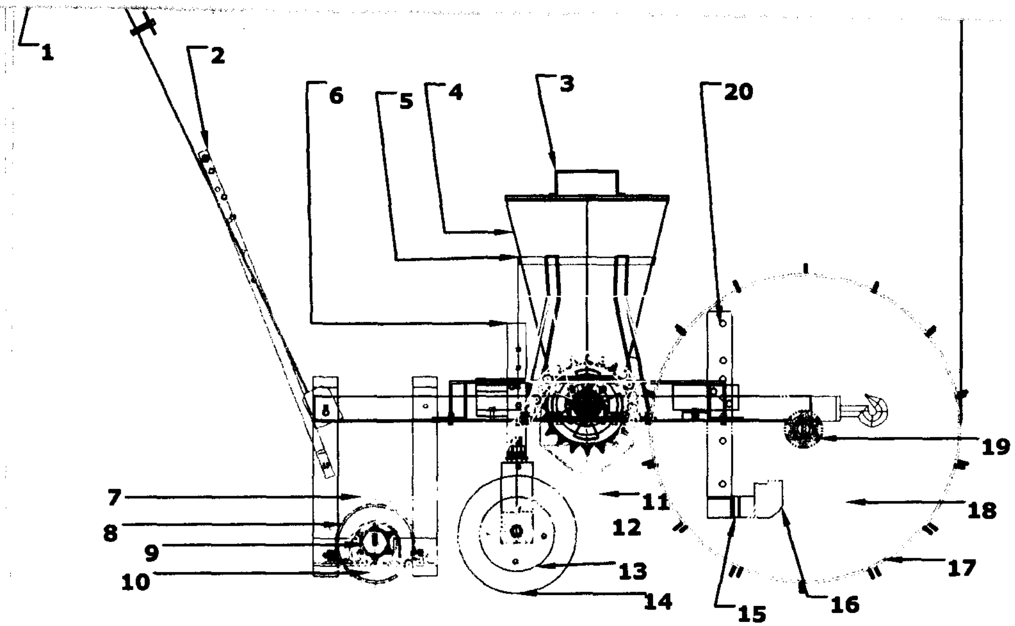
Figure xvi: Assembled Manually Operated Multi-crop planting Machine



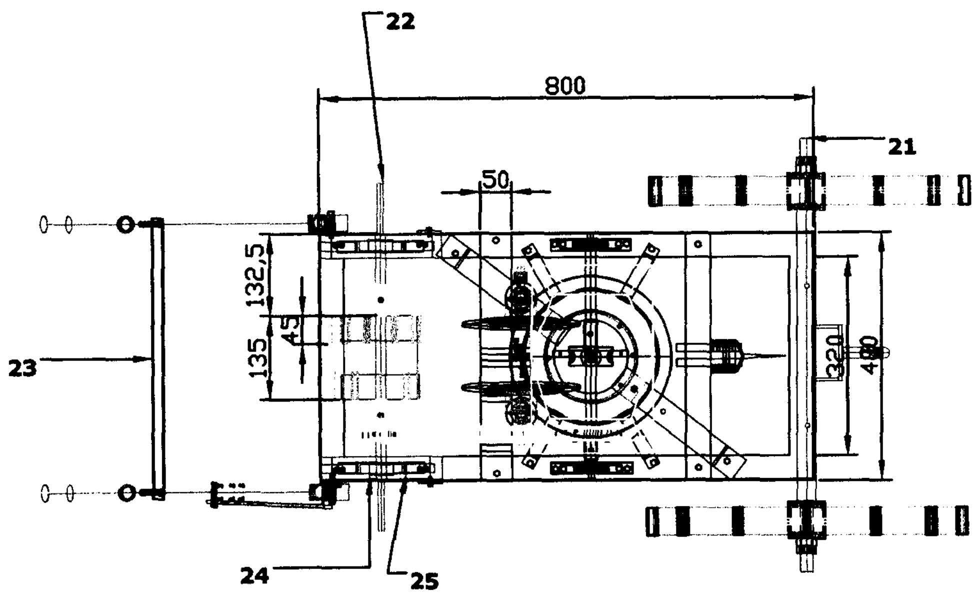
ISOMETRIC VIEW OF MANUALLY OPERATED MULT-CROP PLANTING MACHINE



FRONT ELEVATION



END ELEVATION



PLAN

OTHOGRAPHIC VIEW OF MANUALLY OPERATED MULTI-CROP PLANTING MACHINE

25	CHAIN ADJUSTER D/DIVING SHAFT SUPPORT
24	DRIVING SHAFT'S BEARING
23	CROSS BAR
22	DRIVING WHEEL SHAFT
21	DRIVEN WHEEL SHAFT
20	FURROW OPENER DEPTH ADJUSTER
19	HUB
18	SPOKE
17	DRIVEN WHEEL
16	FURROW OPENER
15	FURROW OPENER'S BRACKET
14	PLAIN DISC
13	RING
12	SHANK
11	SEED DISPENSER UNIT
10	CHAIN
9	SPROCKET
8	DRIVING WHEEL
7	STRAKE
6	DEPTH ADJUSTER
5	HOPPER'S GUIDE
4	HOPPER
3	HOPPER'S LID
2	HANDLE'S HEIGHT ADJUSTER
1	HANDLE
PART NO	PART NAME