

**DESIGN, CONSTRUCTION AND PERFORMANCE
EVALUATION OF A PORTABLE MOTORISED SEED
AND FERTILIZER BROADCASTER**

BY

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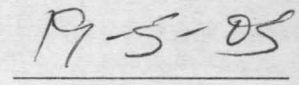
MAY, 2005

CERTIFICATION

In meeting one of the requirements for the award of Post Graduate Diploma (PGD) of the Department of Agricultural Engineering, Federal University of Technology (FUT) Minna, this project work have been certified done solely by the student and with proper supervision and approval



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DEDICATION

To my Wife, Rhoda

ACKNOWLEDGEMENT

My first gratitude goes to the almighty God for the grace of life and his infinite mercy especially as I traversed far and wide in the course of writing the project work.

Most importantly, special appreciation to my Supervisor Engr. Dr. D. Adgidzi who incidentally doubles as my Head of Department for his encouragement, effective supervision through constructive criticism, proper direction and above all his effort to make me a true scholar, made this project report what it is as you read through.

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May the almighty God reward all. Amen.

ABSTRACT

In an attempt to ease one of the production constraints (i.e broadcasting) of our peasant farmers, a portable motorized spinning disc type broadcaster was designed, constructed and evaluated to ameliorate the broadcast seeding and fertilizer application. Particle motion is a function of its shape and frictional coefficient with the disc and vanes.

The machine was constructed with a net weight of 7.5kg and gross weight of 23.5kg. Performance test was done using four materials, rice, soyabean, Beniseed and Urea fertilizer. The test results gave the spreading with as 2.3m for rice, 2.8m for soyabean, and 2.5m for Benniseed. One person can successfully broadcast any material in a field of 1 hectare within 1 hour. The design and construction proves that all materials are obtainable locally and the construction can be done by any local fabricator in the country.

TABLE OF CONTENTS

Title Page	i
Certification	ii
Dedication	iii
Acknowledge	iv
Abstract	vi
Table of Contents	vii
List of Tables	viii
List of Figures	x
Key of symbols	x
CHAPTER ONE	
1.0 INTRODUCTION	1
1.1 Aims and Objectives	3
1.2 Justification for the Project	4
CHAPTER TWO	
2.0 LITERATURE REVIEW	5
2.1 Brush-feed grass Seeders	6
2.2 Fertilizer Distributors	6
2.3 Theory of Centrifugal Distribution	11
CHAPTER THREE	
3.0 DESIGN OF THE MACHINE	17
3.1 Design calculation	17
3.1.1 Pulley Size	17
3.1.2 Power required by the Machine	18
3.1.3 Length of belt required	23
3.1.4 Shaft Diameter	23
3.2 Torsional Rigidity of Shaft	28

3.3	Rivetted and Pin Joints	29
3.3.1	Rivetted Joint	29
3.3.2	Failure Condition	32
3.3.3	Pin Joint Design	33
3.4	Hopper Design	34
3.4.1	Hopper' Shape	34
3.4.2	Hopper Capacity	34
3.5	Discharge Aperture	37
3.6	Application rate	41
3.6.1	Theoretical flow rate	41
CHAPTER FOUR		
4.0	CONSTRUCTION AND PRINCIPLE OF OPERATION	43
4.1	Main Features of the Machine	43
4.2	Operating Principles	46
4.3	Test Performance of the Machine	46
CHAPTER FIVE		
5.0	RESULTS AND DISCUSSION	50
5.1	Test Result (i) and (ii)	50
5.2	Test Result (iii) – Field Capacities	35
5.3	Discussion	53
CHAPTER SIX		
6.0	CONCLUSION and RECOMMENDATIONS	55
6.1	Conclusion	55
6.2	Recommendations	56
	References	57
	Appendix 1	60
	Appendix 2	61

Appendix	3	62
Appendix	4	63
Appendix	5	64
Appendix	6	65

List of Tables

1.	Table (1): Flow rate of various materials	42
2.	Table (2): Cost estimate for the production	49
3.	Tables (3): Field test result	50
4.	Table (4): Field capacity	53

List of Figures

1.	Figure (1): Important variables in centrifugal distribution analysis	11
2.	Figure (2): Variation of δ with different values of ϵ	13
3.	Fig. (3): Bending moment diagram	24
4.	Shear Force and Bending Moment Diagram	25
5.	Fig. (5): Spinning disc showing pin and riveted joints	29
6.	Fig. (6): Hopper Shape	34
7.	Fig. (7): Aperture dimension	41

Key to Symbols

a	=	Distance particle is fed off-centre
b	=	Distance of particle from centre on second and subsequent impact
m	=	Mass of the particle

r	=	Distance of the particle from centre of disc
R	=	Disc radius
U	=	Radial velocity of particle at disc radius
V	=	Absolute speed of particle.
γ	=	Angle turned between successive impact of the vane and particle
ε	=	Velocity coefficient of restitution between particle and vane
θ	=	Angle of dispatch, measured between radius at point of exit and part of the particle.
λ	=	Angle turned while particle slide on the vane
\varnothing	=	Angle between radii at point of feed-on and of exit
ω	=	Angular velocity of disc
η_{\min}	=	Minimum frequency of rotation of disk, rpm
f	=	Coefficient of friction of material with disc
g	=	Acceleration due to gravity m/s^2
ψ_0	=	Initial angle between the vane surface and the radius to the centre of the disc r_0
T	=	Time measured from time of feed-on of the particle on the disc, (sec).
δ	=	Angle between direction of particles leaving the disc and the plane of the disc, deg.

CHAPTER ONE

1.0 INTRODUCTION

Seed and fertilizer placement operation is done either on a straight row, hip or sown at a depth or scattered on the surface of the soil otherwise termed broadcasting.

Broadcasting operation, either seed or fertilizer has been an old method carried out by peasant farmers especially with small feeds and later manure and fertilizer. Because the depth to which seeds should be sown is proportional to its size, small seeds such as beniseed and rice can be broadcasted as they can easily fall between soil practical and be covered on already prepared land while bigger or larger seeds such as soya beans must be covered with a harrowing operation after the broadcast operation.

The ultimate goal of every farmer is to obtain optimal yield when he is sowing his seeds, though most crops can tolerate moderate variation in the uniformity of plant spacing without any serious effect on yield. However, this operation must be done having average population density within the optimum range. There are of course, factors other than yield that are of consideration when trying to establish best population density for any crop and these include lodging or breaking due to over population for standing or upright crops which is undesirable during harvesting operation.

In considering the significance of these class of broadcast seeds, great attention should be drawn to the economic importance they have attained especially of recent in Nigeria, while that of fertilizer as a yield increasing input cannot be overemphasized. It is known that Nigeria has embarked on the importation of rice as local production do not meet

demand. While on the other hand benniseed and soyabeans have varying degree of nutritional and industrial importance being in high demand for export.

Broadcasting is done either by hand or mechanical means. The former method is the one widely used by peasant farmers due to their subsistence level of farming. However, with the growing demand of these industrial crops, majority of the peasant farmers in the middle belt region of Nigeria where most of these crops are grown, are increasing there hectarage making hand broadcast tedious, hence unreliable.

The broadcast operation is usually done after the land has been prepared. The seed is usually carried in an open container, calabash or even polythene bags and scattered with the throw of hand. Uniformity of broadcast is usually not achieved especially when the operation is carried out after land preparation. And this non-uniformity results in the seed germinating in cluster or overcrowded thereby affecting yield. Likewise, non-uniformity of fertilizer broadcast affects yield as some plants will get the nutrients more than others or scotching of plants where they are hipped.

Also, when the farmer is desirous of optimum yield, thinning and or supplying are done for plants such as soyabeans and rice respectively. These exercises are of course laborious and time consuming resulting into time loss, and hence increase in production cost.

Mechanical broadcasting, on the other hand is done by a variety of machines. The most common among them is the spinning disc type broadcaster. It is also known as centrifugal type broadcaster. This broadcaster have been in use for the purpose of broadcasting both seed and fertilizer in some advanced countries such as Britain and the United

states of America for over a century (Smith, 1965). It is often used for broadcasting granular fertilizer such as superphosphate and small seeds such as clover or alfafa (Aji, 1977).

The spinning disc is usually of circular type with radial vanes mounted perpendicular to the disc surface. The disc is mounted to a vertical shaft and rotates horizontally with the driven shaft. Seed or fertilizer falling on the rotating disc from the hopper are accelerated radially outward by the centrifugal force generated by the rotating disc. At the edge of the spinning disc the seed or fertilizer are projected out and spread on the soil surface. The advantage of the spinning disc type broadcaster is its ease of operation and simplicity of manufacture and maintenance. It is also easily used on irregular fields and those having surface obstructions.

Due to the inherent advantages mentioned above associated with the spinning disc type broadcaster, it is hereby being modeled along the line of knapsack sprayer for effective seeding and fertilizer broadcasting by the individual peasant farmer in order to enhance production.

AIMS AND OBJECTIVES

This project is carried out with the sole aim of providing a simple, appropriate, durable, lightweight and affordable machine that will not only be attractive in solving one of the farmers numerous production constraints but enhance qualitative production process for increased economic return. The specific objectives are therefore:

1. To design the machine.
2. To construct one using locally available materials.
3. To carry out performance test of the machine.

JUSTIFICATION OF THE PROJECT

The success of this project work will greatly enhance the following:

- i. Reduce the drudgery associated with the traditional method of hand broadcasting
- ii. Saves time and money for other production work.
- iii. Optimize yield for good economic returns
- iv. It eases harvesting operation
- v. It makes the broadcast operation attractive.

CHAPTER TWO

2.0 LITERATURE REVIEW

Today there are as many types and models of seed and fertilizer placement equipment in the market that meets varying requirement of the farmer. Whether the farmers requirement is drilling, hill dropping, parallel row or broadcast placement, there is an equal model designed to achieve these purposes. All the farmer needs is to select the type and model that meets his requirement and available fund.

Though, there are several methods of seed and fertilizer placements, the broadcast method is the oldest and still the most widely used today especially for smaller seeds such as grain seeds and other cereals. Smith, (1965) reported that broadcasting seeds over broken soils, covering them with some type of harrow was the common method of planting (in the United States) until 1840 when William T. Pennock of East Marlboro, Pennsylvania, started manufacturing grain drills. Therefore, broadcasting can be said to be the oldest and simplest method of sowing seeds. And broadcasting by machine is more accurate and rapid than by hand.

In the 18th and early 19th century there were quite a lot of development in the United States and Britain in the manufacture of some types of broadcast seeders. For instance, Smith, (1965) reported some broadcast machines produced in the United States such as the knapsack, endgate, two wheel horse driven weeder-mulcher and the use of air planes.

For example the knapsack type broadcast seeder consist of a canvas sack fastened to a seeding mechanism and the whole equipment is hanged over the shoulder. A crank turned by hand revolve a wheel

having several different radial ribs for scattering the seeds. This device was said to be used commonly for broadcasting clover seeds and small grass seeds

Culpin (1978) also reported the use of a broad casting device in Britain known as the "fiddle" in the early 19th century. It was adopted commonly for use especially at Hillsides where maneuvering with tractor mounted equipment seems to be difficult. The device consists of a hopper and a mechanical distributor and is strapped to a man's shoulder. As the sower walks along, he draws to and fro a 'bow' which cause a small ribbed platform to rotate and flings the seeds out to the sides.

Today, there are so many types and models of broadcast equipment in the market some of which are briefly described below.

2.1 BRUSH-FEED GRASS SEEDERS

Feed mechanisms is usually of the revolving brushes inside the hopper that picks up seeds and flings them out through an aperture under the hopper. The land wheels usually drive the mechanism.

2.2 FERTILIZER DISTRIBUTORS

Distributors are further classified by their features which are characterized by the type of feed mechanisms employed. Some distributors widely in use today are:

2.2.1 Reciprocating plate distributor:

With this type of distributors, fertilizer discharge is achieved by the action of two slotted plates, positioned one above and the other below a fixed aperture plate at the bottom of the hopper. The slotted plates

slides with a reciprocating action along the upper and lower faces of the aperture plate and filter the fertilizer through. Application rate is regulated by either varying the throw of the reciprocating plate or by using plates with different sized slots.

2.2.2 Rotating plate and flicker distributor:

With this model, a concave disc rotates continuously receiving a constant flow of fertilizer from the hopper. The fertilizer is then flicked over the edge of the disc by small fingers rotating in the vertical plane just above the surface of the disc.

2.2.3 Impeller and pendulum spout distributor

This model looks similar to the more popular spinning disc type in construction but uses an impeller and swinging pendulum spout to control distribution more accurately.

2.2.4 Radial spout distributor

This is also similar in construction to the pendulum spout distributor just described above except that instead of the swinging pendulum, the fertilizer are distributed through radially fixed spouts.

2.2.5 Moving flow distributor

This is another spinning disc type distributor conceived for large capacity work. It consists of a trailer constructed with a Vee-shaped trough, at the bottom of which is an auger or slate-type conveyor to carry material to the rear. At the rear, the content are dropped on one or two rotating spinning discs for distribution. The spinning discs are either

driven by the tractor PTO or tractor hydraulic system through a hydraulic motor. Application rate is regulated either by altering the speed of the conveyor or auger and an adjustable slides at the rear. Two to three tons capacities are common.

2.2.6 Spinning disc distributor

The spinning disc type distributor is the one that influences the overall design of other distributors and is still doing so today. It consist of a cone-shaped hopper which incorporates an agitator to break up large lumps and keep the material flowing. The spinning disc is either driven by the PTO, land wheel or by means of a hydraulic motor powered by the tractor hydraulic system. Fertilizer delivery is regulated by an adjustable shutter. Depending on the speed of the rotating disc and adjustment of the discharge shutter between the hopper and the disc, operation width varies from 5-16meters? Uniformity of distribution is a function of condition and type of the fertilizer. Two valuable features of this distribution type are (a) narrow transport width and (b) Wide operational span.

As earlier mentioned, the spinning disc type distributor is the most versatile of all the distributors or broadcasters. Due to the numerous advantages it possesses over other types, farmers in the United states and Britain realised it and since over a century have converted it for the broadcast of seeds even though it was manufactured solely for fertilizer application. Whereas in Nigeria, even though this equipment have been in use over the last 50 years, no attempt have been made to utilize this simple machine in whatever form for the broadcast of seeds.

However, mentioned must be made to the few attempt made by the students of Ahmadu Bello University, Zaria and college of agriculture, Samaru-Zaria. Aji, (1977) in his unpublished B. Sc Engineering work tried to study the possibility of using the spinning disc type broadcaster for the broadcast seeding of benniseed. The work involved the use of three different diameters spinning disc of 15cm, 20cm and 30cm at different, speed respectively. In conclusion, the 30cm diameter size gave the best result in terms of distribution pattern and population density per unit area. Also, it was found that the optimal speed for best result was between 400-600rpm and any speed lower than 300rpm is unsuitable for benniseed broadcast.

And from Samaru college of Agriculture, Garba, (1998) fabricated a shoulder strapped hand fertilizer broadcaster. It consists of a hopper, with a spinning disc directly mounted below the hopper centrally fed and driven by a crank lever through two bevel gears in mesh. The whole construction weighs about 5.1kg. Not much was given about its test evaluation.

Again, Garba (2002) tried the construction of the ground driven model of his earlier work. Virtually the same parameters were used except for the little change in the method of drive. Here, instead of the crank lever being rotated by hand, the gear drive are rotated by the land wheel. In this work, a gear ratio was arbitrary chosen at 2:1 and no other design consideration was given in determining any parameter or arriving at the ways the various components of the machine were fabricated. Compound NPK and Super Phosphate SSP brand of fertilizers were used to test the machine but without adequate information on the test.

In Samaru College of Agriculture, there is the imported version of seed broadcaster from "the cyclone seeder company, Urbans, Indiana USA" simply named "cyclone". The machine was brought in for the purpose of teaching and research, it consist of a hopper that can take about 5kg of material made from a thin iron sheet of about 1mm, seated on a wooden plank. The discharge opening made through the wooden plank is regulated with a shutter at the side by hand through a lever. The shutter seats on a crank that continuously rocks the shutter up and down thereby making the shutter behave as an agitator. The gear drives are bevel gears (as main drive for the spinning disc) and multiple spur gears that drive the agitating shutter. The spinning disc is mounted off-centre to receive and broadcast materials. Due to the advanced technology of the country of manufacture, the whole equipment was compactly designed with net weight of about 3.5kg.

The fundamental principle on which the spinning disc distributor uses in broadcasting material is centrifugation. Therefore, for the actual design and eventual construction of machine the basic theory of the centrifugal distribution must be applied

2.3 THEORY OF CENTRIFUGAL DISTRIBUTION

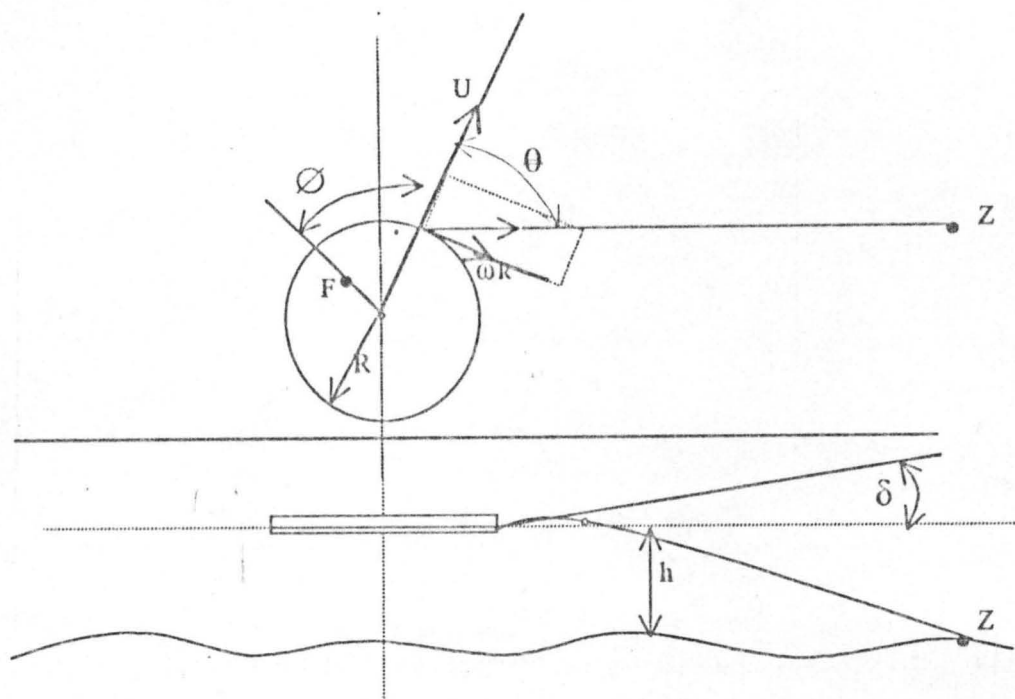


Fig. (1) important variables in the centrifugal distribution analysis (Paterson and Reece, 1962).

Consider a particle dropping on to a disc at a point F, fig (1) above, neglecting the effect due to interference by adjacent particles, motion of individual particles can be examined and projected. The acceleration of the particle on the disc can be determined and so also the angle which depends upon the time that it remains upon the disc and hence the point at which it leaves the disc. The trajectory can be calculated and hence the location of the point Z at which the particle reaches the ground surface. If the angle θ which describes the direction of flight, the magnitude V of the velocity of projection, the angle δ and height h , and the particle size and shape which controls the air resistance be obtained accurately, it would be possible to work backwards from the desire distribution point

such as Z to a feed-on pattern of points such as F which will give a uniform distribution on the ground.

The principal factors that control the motion of a particle are its coefficient of friction relative to the vanes and shape which decides whether or not it has any possibility of rolling along the vane instead of merely sliding. Therefore, depending on the shape of the particles, in the first instance, the particle may slide along the vane and disc, in which case the frictional force between the particle and the vane and the disc will be at its limiting value. On the second instance, the particle might be of the shape that rolling will occur on one or both surfaces in contact with the particle. The frictional force developed may not reach its limiting value since a small force may be sufficient to provide the necessary torque for rotational motion of the particle. The above theory describes the typical near centre-feed as described by Patterson & Reece (1962).

However, Inns & Reece (1962) acknowledged the fact that in practice, materials are fed to the spinning disc off-centre. For this reason, the effect of impact between the particles and the vanes and disc must be considered. This is because the usual method of feeding material s on to the disc-involves dropping them vertically downwards on the disc at some distance from its center. They therefore, enter the zone of the vane with a small vertical velocity, but otherwise stationary in space.

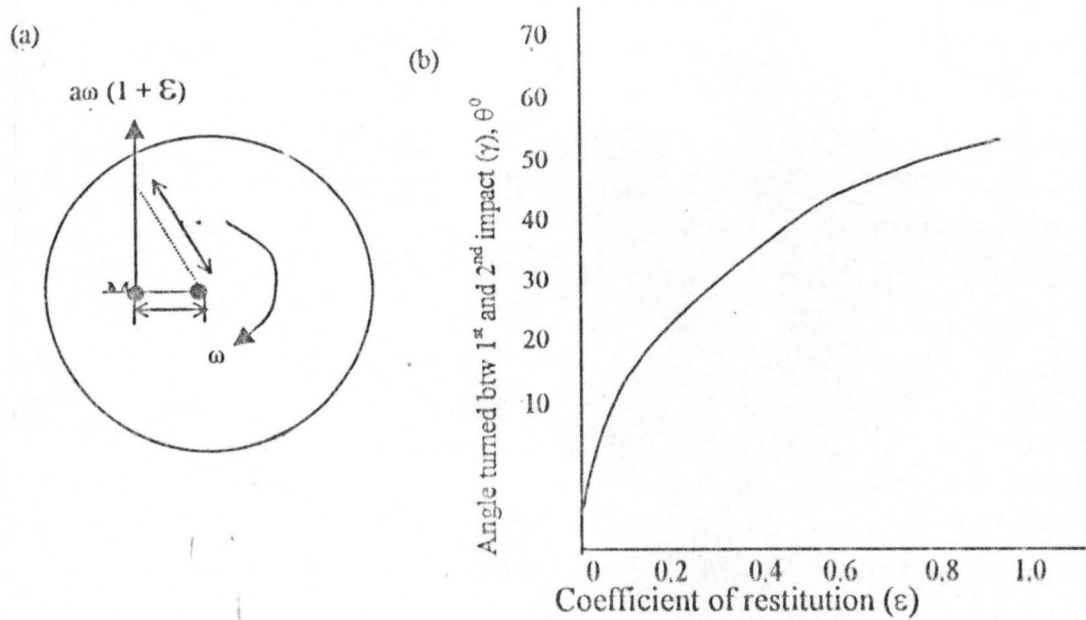


Fig.(2) Variation of δ with different values of ϵ (Reece & Inn 1962)

From fig. 2(a) above at the moment of feeding-on, the particle is stationary in space at a distance 'a' from the centre of the disc, while the point of the vane which is about to make contact with the particle is approaching at a velocity ' $a\omega$ ' at right angle to the vane. At impact therefore, the velocity of the particle relative to the vane will be reversed and reduced by a ratio depending on the velocity coefficient of restitution and between the particle and vane. After impact, the particle will therefore have a velocity relative to the vane of $\epsilon a\omega$. Since the vane itself has a velocity in space of $a\omega$ at that point, the total velocity of the particle will therefore be $a\omega + \epsilon a\omega = a\omega (1 + \epsilon)$ (Inns and Reece, 1962).

The particle will then be projected from the vane in a straight line at right angle to the face of the vane. If the point of impact is near the periphery of the disc, the particle will escape with this speed and direction. With the point of impact nearer the centre of the disc, the

particle will not escape from the disc before it makes a second contact with the vane. This second impact will occur at a distance b_i , from the centre where $b_i > a$. The disc would have turned an angle γ_i in a time T_i and this can be determined.

From fig 2a above.

$$\begin{aligned}\tan \gamma_i &= \frac{a\omega(1+\epsilon)t_i}{A} \\ &= \omega T_i(1+\epsilon)\end{aligned}$$

$$\text{But } \omega T_i = \gamma_i$$

$$\text{Thus, } \tan \gamma_i = \gamma_i(1+\epsilon) \quad \text{----- 20}$$

It should however be noted that angle γ_i is independent of a and ω . Fig 2b above shows how γ_i varies with different values of ϵ .

However, because relative motions of the particle at the point of impact with the vane is not at right angle to the face of the vane, the motion therefore is more complicated thus allowance must be made for effect of friction. For practical calculations, the minimum frequency of rotation required for the material to overcome frictional force must be determined, the angular displacement of the particle in absolute motion from the feeding moment to the moment it leaves the disc, hence the time and the final velocity with which the particle leaves the disc.

Considering fig 2a, if materials are delivered at arbitrary point 'M' with an initial velocity equal to zero, in this case the forces applied on the particle are the frictional force $F = fmg$ and the transient motion of centrifugal force $I_c = ma\omega^2$. The point M on the disc has a

circumferential velocity $V_c = a\omega$. Therefore, materials reaching the disc at that point acquires the same velocity. The equilibrium condition can be expressed as:

$$ma\omega^2 - fmg = 0$$

Thus, angular velocity

$$\omega = fg/a$$

Hence the minimum frequency of rotation of the disc will be

$$\eta_{\min} = 30/\pi \sqrt{fg/a} \text{ ----- 2.1}$$

If $fmg < ma\omega^2$, the particle will move in its relative motion on the disc surface.

To determine the angular displacement of the particle in absolute motion from the feeding moment to the moment it leaves the disc the final value of the disc radius 'r' is substituted in the following equation (Reece & Inn, 1962)

$$r = \sqrt{\left\{ \left[\frac{fg}{\omega^2} - a \frac{\cos(\psi_0 \pm \phi)}{\cos \phi} \right] \left[\frac{1}{\lambda_2 - \lambda_1} (\lambda_2 e^{\lambda_2 T} - \lambda_1 e^{\lambda_1 T}) - 1 \right] + a \cos^2 \psi_0 \right\}} \rightarrow$$

$$\rightarrow + a^2 \sin^2 \psi_0 \text{ 2.2}$$

However, where the vanes are mounted on the disc, radially, the above equation becomes

$$r = \sqrt{\frac{fg}{\omega^2} \left\{ \frac{1}{\lambda_2 - \lambda_1} \left[(\lambda_2 e^{\lambda_1 T} - \lambda_1 e^{\lambda_2 T}) - 1 \right] \right\}} \text{ 2.3}$$

The time t during which the particle remains on the disc is then found, during which the disc rotates through an angle

$$\theta = \omega t_{pr} \text{ --- --- --- 2.4}$$

and the final velocity of particle

$$V = \frac{f_g}{\omega^2} \left\{ \frac{1}{\lambda_2 - \lambda_1} \left[(\lambda_2 e^{\lambda_1 T} - \lambda_1 e^{\lambda_2 T}) \right] - 1 \right\} \text{ 2.5}$$

Bosoi et al (1988) stated that for fertilizer, the disc works with a frequency of rotation of 400-600 rpm and that vanes usually rotate at an angle of 12° - 18° (the most uniform distribution are observed at this angle while uniformity of distribution increases with increased frequency of rotation of the disc and decrease with the increase of the diameter.

CHAPTER THREE

3.0

DESIGN OF THE MACHINE

3.1 DESIGN CALCULATION

3.1.1 PULLEYS SIZE

The machine will be powered by a D.C motor having a speed of 4500 RPM. The drive pulley is chosen to be 0.03m diameter so as to reduce considerably the driven speed. The driven pulley can then be calculated from the relationship

$$\underline{N_1 D_1} = N_2 D_2 \text{ ----- 3.01}$$

Where,

N_1 = Speed of motor; 4500RPM

N_2 = Speed of spinning disc (1000RPM max.)

D_1 = Diameter of drive pulley (0.03 chosen)

D_2 = Diameter of driven pulley

Hence, from equ. (3.01) above

$$\underline{D_2 \frac{N_1 D_1}{N_2}} \text{ 3.02}$$

$$= \frac{450 \times 0.03}{1000} = 0.135\text{m}$$

Therefore $D_2 = \underline{0.135\text{m or } 135\text{mm}}$

Initial parameter are therefore:

Speed of motor = 4500RPM

Speed of machine = 1000RPM (chosen – max.)

Diameter of drive pulley = 0.03m (chosen)

Diameter of driven pulley = 0.135m (calculated)

Thickness of pulley = 0.12m each

3:1.2 POWER REQUIRED BY THE MACHINE

The power required by the machine is a function of the total load, both axial and radial and the speed of the rotating belt.

Hence,

$$\text{Power, } P = F_1 V \text{ ----- 3.03}$$

Where,

P = Power required, watts

F_1 = Total load (forces) to be overcome by belt power, N

$$= [(M_s + M_d + M_p)g + CF_d + CF_p + T_b] \text{ N}$$

V = Speed of the machine, m/s

$$\text{Thus, } P = \{[M_s + M_d + M_p] g + CF_d + CF_p + T_b\} \times V, \text{ watts ----- 3.04}$$

- M_s = Mass of spinning of shaft, kg
- M_d = Mass of spinning disc, kg
- M_p = Mass of pulley, kg
- g = Gravitational constant
- F_b = Force to overcome friction by belt, N
- T_b = Belt tension, N
- V = speed of machine, m/s

3:1.2(a) MASS OF SPINNING DISC

The mass of the spinning disc is the totality of the mass of the rotating disc and the four vanes riveted on it.

Hence,

$$M_d = \delta \{[\pi D^2 t_1]/4 + A_v t_2\} \text{ ----- 3.05}$$

Where,

δ = density of material = 2698 kg/m^3 for Alluminium (Kempis Engineers year book 1989)

D = diameter of disc = 0.25 m

T_1 = thickness of disc = 0.0015 m

A_v = Area of Vane, $(0.035 \times 0.01) = 0.0035 \text{ m}^2$

T_2 = thickness of vane = 0.0015 m

Substituting values in to equ. (3.05) above

$$\begin{aligned} M_d &= 2698 \left\{ \frac{[\pi \times 0.25^2 \times 0.0015]}{4} + [0.0003 \times 0.0015]4 \right\} \\ &= 2698 [7.36 \times 10^{-5} + 2.1 \times 10^{-6}] \\ &= 2698 \times (7.57 \times 10^{-5}) \\ &= 0.2 \text{ kg} \end{aligned}$$

3:1.2(b) MASS OF DRIVEN PULLEY

The mass of the driven pulley is calculated from the equation

$$M_p = \delta V = \frac{\delta (\pi D^2 t)}{4} \text{ ----- 3.06}$$

Where,

δ = density of pulley material = 7860 kg/m^3

V = Volume of pulley, M^3

D = diameter of pulley = 0.135 m

T = thickness of pulley = 0.012 m

substituting values on equation (3.06) above

$$M_p = 7860 \left[\frac{\pi \times 0.1352 \times 0.012}{4} \right]$$

$$= 7860 \times 172 \times 10^{-4}$$

$$\text{Therefore } M_p = \underline{1.35\text{kg}}$$

3:1.2(c) BELT FRICTION FORCE

One of the forces to be overcome by the belt power is the friction force between the belt and the pulley and this can be calculated from the equation below as given by Haul et al (1980)

$$F_b = fdN \text{ ----- 3.07}$$

Where,

f = Coefficient of friction between belt and pulley = 0.3 for rubber belt on steel

d = diameter of pulley = 0.135m

N = speed of pulley = 1000RPM

Substituting values in equation (3.07) above

$$F_b = 0.3 \times 0.135 \times \frac{1000\pi}{60}$$

$$=$$

$$2.12\text{N}$$

3:1.2(d) BELT TENSION

Belt tension can be calculated from the following equation as given by Haul et al (1980).

$$\frac{T_1 - mV^2}{T_2 - mV^2} = e^{(f\alpha/\sin\theta/2)} \dots\dots\dots 3.08$$

Where,

T_1 = Belt tension on tight side, N

T_2 = Belt tension on tight side, N

M = Mass of 1m of belt, 0.42kg from the selected belt, width = 10mm, thickness = 6mm (machinery handbook, by OBERG et al, 24 Ed. 1992) and density of rubber belt = 1400kg/m³ (Sharma and Aggarwal 1998)

V = Belt velocity, m/s

f = Coefficient of friction – belt on pulley = 0.3

α = angle of wrap on smaller pulley, 0°

θ = Groove angle of pulley = 36°

But,

T_1 = belt width x thickness x shear stress where shear stress was taken as 1.2mpa x 10⁶ N/m² (Haul et al, 1980).

Therefore $T_1 = 0.01 \times 0.06 \times (1.2 \times 10^6)$

$$= 72N$$

$$V = \frac{\pi DN}{60} = \frac{\pi \times 0.135 \times 1000}{60} = 7.07m/s$$

Wrap angle, $\alpha = 180 - 2\sin^{-1} \frac{(D_2 - D_1)}{2C}$ Sharma and Aggarwal
(1998) 3.09

but centre distance, $C = \frac{1}{2}(D_2 + D_1) + D_2$ ----- 3.10

$$= \frac{1}{2}(0.135 + 0.030) + 0.135$$

$$= \frac{0.165 + 0.135}{2}$$

$$C = 0.218\text{m}$$

Putting back value of 'C' into equ (3.09)

$$= 180 - 2\sin^{-1} \frac{(0.135 - 0.30)}{2(0.218)}$$

$$= 180 - 2\sin^{-1} \frac{0.105}{0.436}$$

$$= 180 - 27.87$$

Therefore $\alpha = 152.13 \frac{(152.13 \times \pi)}{180}$ 2.66 rad

substituting values above into equation (3.08)

becomes

$$\frac{72 - 0.42(7.07)^2}{T_2 - 0.42(7.07)^2} = e^{(0.3 \times 2.66) / \sin 18}$$

$$\frac{72 - 20.99}{T_2 - 20.99} = e^{0.798 / 0.3090}$$

$$\frac{51.01}{T_2 - 20.99} = 13.23$$

$$T_2 = \frac{13.23}{51.01} + 20.99$$

therefore $T_2 = 21.25\text{N}$

Total belt tension, $T_1 + T_2 = 72 + 21.25 = 93.25\text{N}$

3.1.3 LENGTH OF BELT REQUIRES

Total belt length required was given by the formular proposed by Sharma and Aggarwal (1998) as follows:

$$L = 2C + 1.57 (D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C}, \text{ m} \quad \text{-----} \quad (3.11)$$

Where C = Centre distance between pulleys, $= 0.218\text{m}$

D_2 = diameter of larger pulley $= 0.135\text{m}$

D_1 = diameter of smaller pulley $= 0.03\text{m}$

Substituting,

$$L = 2(0.218) + 1.57(0.135 + 0.03) + \frac{(0.135 - 0.03)^2}{4(0.218)}$$

$$= 0.436 + 0.259 + 0.013$$

$$= \underline{0.708\text{m}} \quad \text{or} \quad \underline{708\text{mm}}$$

3.1.4 SHAFT DIAMETER

The critical loads acting on the shaft are:

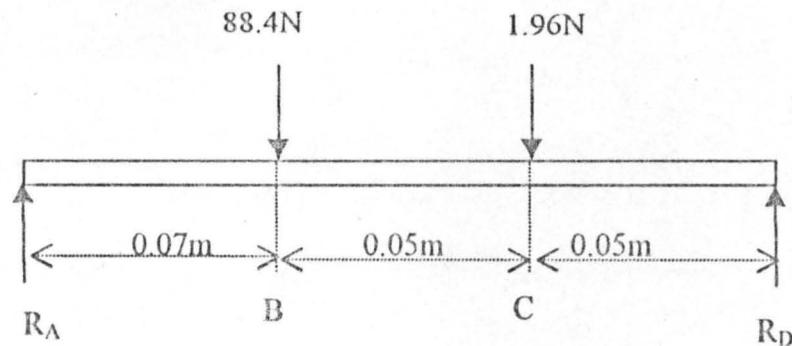
a) (i) weight of pulley $= 1.35\text{kg} \times 9.81 = 13.24\text{N}$

(ii) Total belt tension, $T_1 + T_2 = 72 + 21.25 = 93.25\text{N}$

Total force acting through the pulley $= 106.5\text{ N}$

b) (i) weight of spinning disc = $0.2\text{kg} \times 9.81 = 1.96\text{N}$

To arrive at the diameter of the shaft, the maximum bending moment and torsional moment acting on the shaft must be determined.



c) Fig. (3) Bending moment diagram (loads acting on the shaft)

Taking moment about R_D

$$R_A \times 0.17 = 1.96 \times 0.05 + 88.4 \times 0.1$$

$$= 0.098 + 8.84$$

$$R_A = \frac{8.938}{0.17}$$

$$= \underline{52.58\text{N}}$$

$$R_A = 88.4 + 1.96 - R_A$$

$$= 90.36 - 32.58$$

$$\text{Therefore } R_D = 37.78$$

Shear force, SF

$$\text{At pt A, } S_f = R_A = 52.58$$

$$\text{At pt B, } S_f = R_A - 88.4 = 35.82$$

$$\text{At pt C, } S_f = R_A - 88.4 - 1.96 = -37.78$$

Bending Moment, BM

$$\text{At pt A, BM} = R_A \times 0 = 0$$

$$\text{At pt B, BM} = R_A \times 0.07 - 88.4 \times 0$$

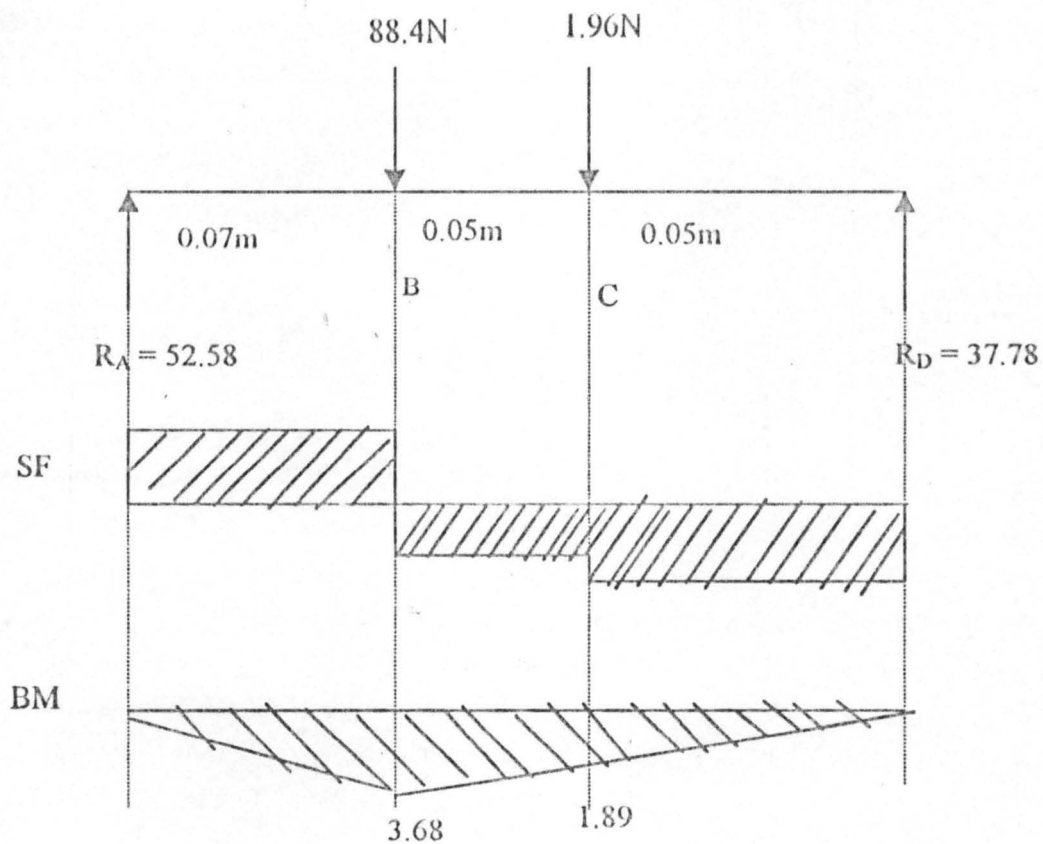
$$52.58 \times 0.07 = 3.68$$

$$\text{At pt. C, BM} = R_A \times 0.12 - 88.4 \times 0.05 - 1.96 \times 0$$

$$52.58 \times 0.12 - 88.4 \times 0.05 = 0$$

$$6.31 - 4.42 = \underline{1.89}$$

$$\text{Check: } R_B \times 0.05 = 37.78 \times 0.05 = 1.89$$



F) Fig (4) Shear force and bending moment diagram

Note:

The maximum bending moment is at 0.07m from A where the shear force = 0

$$\text{Hence, } M_{b\max} = \underline{3.68 \text{ Nm}}$$

g) Torsional moment, $M_t = (T_1 - T_2)R$, Nm -----
 ---- (3.12)

$$= (72 - 21.25) 0.0675$$

$$= 50.75 \times 0.0675$$

$$\text{therefore } M_t = \underline{3.43 \text{ Nm}}$$

(h) Determination of Shaft diameter

Shaft diameter can be calculated from the following empirical equation as given by Haul et al (1980)

$$\text{i.e. } d^3 = \frac{16}{\pi S_s} \left[[K_b \times M_b]^2 + [K_t \times M_t]^2 \right] \text{----- (3.13)}$$

where

S_s = allowable stress = 40mPa for shaft with key ways

K_b = Combine shock and fatigue factor applied to bending moment = 1

K_t = Combine shock and fatigue factor applied to torsional moment = 1.5

M_b = Maximum bending moment = 3.68 Nm

M_t = torsional moment = 4.65 Nm

Substituting values in equation (3.15) above

$$d^3 = \frac{16}{\pi(40 \times 10^6)} \sqrt{[1 \times 3.68]^2 + [1.5 \times 4.65]^2}$$

$$= \frac{16}{1.3 \times 10^6 \sqrt{13.54 + 48.65}}$$

$$\text{Therefore, } d = 3 \square 9.7 \times 10^{-2} = 0.0099\text{m or } 10\text{mm}$$

Hence

$$\text{Mass of shaft, } M_s = \rho \pi (r^2 l)$$

Where,

$$\rho = \text{density of steel} = 7860 \text{ kg/m}^3$$

$$r = \text{radius of shaft} = 0.005\text{m}$$

$$l = \text{length of shaft} = 0.22\text{m}$$

Therefore,

$$M_s = 7860 \{ \pi [0.005^2] \times 0.22 \}$$

$$= 0.1358 \text{ kg or } \underline{0.14\text{kg}}$$

h) Power required by the machine

$$P = F_T V$$

From equ. (3.04)

$$P = \{ [0.14 + 0.2 + 1.35] 9.81 + [2.12 + 93.25] \} \times 7.07$$

$$= 108.95 \times 7 \times 07$$

Therefore, $P = 770.28 \text{ watts}$

Rated power for machine = 770.28 watts

Design power = rated power x service

And from Sharma and Aggarwal, (1998), the service factor was chosen as 1.2

Hence, Design Power = $770.28 \times 1.2 = \underline{924 \text{ watts}}$

3.2 TORSIONAL RIGIDITY OF SHAFT

Haul et al (1980) gave an imperical equation for the calculation of torsional rigidity of shafts as

$$\theta = \frac{584M_t L}{Gd^4} \text{-----} (3.14)$$

Where,

θ = angle of twist, deg

L = Length of shaft = 22cm (0.22m)

M_t = torsional moment = 4.65 Nm (calculated)

G = torsional modules of elasticity = $80 \times 10^9 \text{ N/m}^2$

d = shaft diameter = 0.01m (calculated)

substituting values in equation (3.17) above

$$\frac{584 \times 4.65 \times 0.22}{80 \times 10^9 \times 0.01^4} \\ \frac{597.432}{8000} = \underline{0.75^\circ}$$

The radial deflection is within limits permissible (ie 0.3 – 3 deg/m (Haul and Aggarwal, 1980).

3.3 RIVETTED AND PIN JOINTS

3.3.1 Rivettted Joint

The alluminium spinning disc is designed to be riveted to a 2mm steel flange which in turn is secured to the drive shaft by a cylindrical pin.

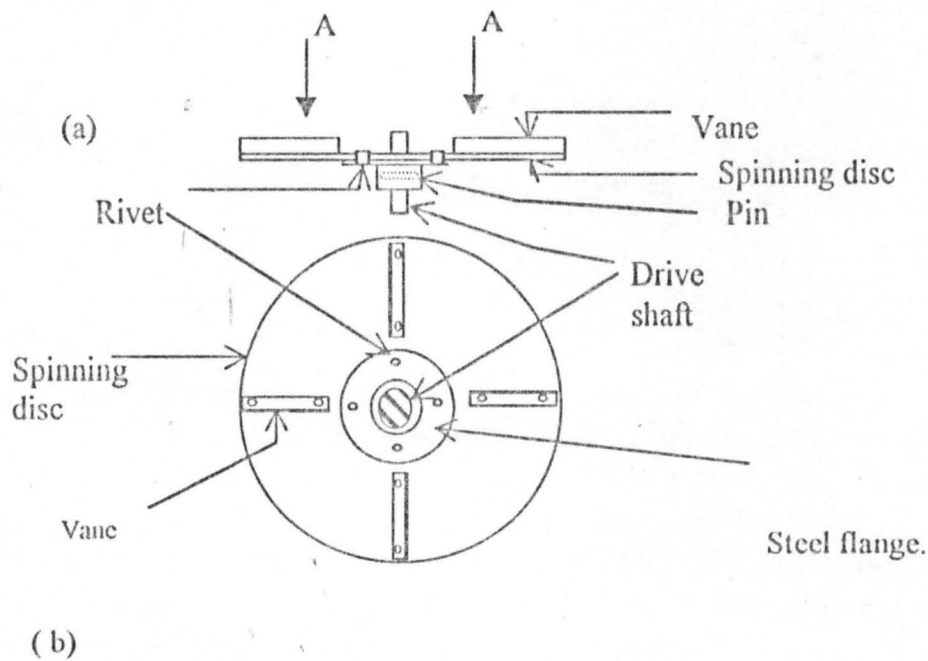


Fig (5): Spinning disc showing (a) Pin Joint and (b) Rivet Joint

The steel plate flange to be riveted to the spinning disc is of thickness 2mm. Hence the diameter of the rivet hole can be calculated as given by Sharma and Aggarwal (1998) thus

$$D = 6.05 \sqrt{t} \text{ ----- (3.15)}$$

Where

D = diameter of the rivet hole, mm

t = plate thickness, mm

therefore

$$\begin{aligned}
 D &= 6.05 \sqrt{2} \\
 &= 6.05 \times 1.4142 \\
 &= 8.5559 \text{ or } \underline{8.6\text{mm}}
 \end{aligned}$$

However, for a 2mm sheet plate, the standard hole and rivet diameter are taken from appendix (5) as given by sharma and Aggarwal 1998 (Table 7.9) as 8.4mm and 8mm respectively, which confirms with the calculation above.

The number of rivets required to properly secure the disc to the flange is given by

$$Z = \frac{G}{y_s} \text{ ----- 3.16}$$

where,

Z = number of rivets

G = total load on the rivet, N

y_s = shear stress of rivet = 70 N/mm² (Sharma and Aggarwal 1998)

But G , which is the load on the rivet is the force of rotation of the spinning disc. This force is the product of the mass of the disc and its angular velocity.

$$\text{Hence } G = m\omega^2 r \text{ ----- 3.17}$$

Where,

G = rotating force of disc, N

M = mass of disc = 0.2kg (calculated)

ω = angular velocity = $\frac{2 \pi N}{60} = \frac{2 \pi \cdot 1000}{60} = 104.7 \text{ rad/sec}$

r = radius of spinning disc = 0.125m

$$\begin{aligned} \text{therefore } G &= 0.2 \times 107.7^2 \times 0.125 \\ &= 274\text{N} \end{aligned}$$

Substituting values back into equ(3.18)

$$Z = \frac{274}{70} = 3.914$$

The number of rivets required = 4

The Pitch or rivet distance therefore

$$= \frac{360^\circ}{4} = 90^\circ \text{ (apart)}$$

The shaft diameter = 10mm; $r = 5\text{mm}$.

The lapping margin is given as $m = 12d$

Therefore, the riveting is done at $12d + 5$

(where d = rivet hole diameter)

$$\text{Thus } 12d + 5 = (2 \times 8.4) + 5 = 21.8\text{mm}$$

The four rivet shall be spaced of 90° and at a radius of 22mm from the centre of the disc.

3.3.2 FAILURE CONDITION

The maximum load that the rivet joint can withstand is investigated through the following

(a) Load to shear one rivet, $G = \gamma_r \times 2\pi \times \frac{d^2}{4}$

Where γ_r – shear stress of rivets = 70 N/mm (Sharma and Aggarwal, 1998)

Hence, $G = 70 \times 2 \times \pi \times \frac{8.4^2}{4} = 7758.48\text{N}$

Therefore load to shear at the rivets

$$= 7758.48 \times 4 = 31033.92\text{N}$$

b) Load to crush one rivet, $G = \delta_c dt$

Where δ_c = bearing pressure = 185 N/mm² (Sharma and Aggarwal 1998)

Hence, $G = 185 \times 8.4 \times 2 = 3108\text{N}$

Therefore, load to crush all the rivet = $3108 \times 4 = \underline{12432\text{N}}$

c) load to tear the plate along the pitch circle is given as $G = \delta_T (p-d)t$

where δ_T = tensile stress = 135 N/mm (Kempes Engr. Year book 1989)

$$G = \text{pitch} \times \frac{2\pi r}{4} = \frac{2 \times \pi \times 11}{4} = \frac{69.12\text{mm}}{4} = 17.28\text{mm}$$

Substituting values

$$G = 135 (17.28 - 8.4) \times 2$$

$$= 135 \times 8.88 \times 2$$

$$= \underline{2397.6\text{N}}$$

d) Load to tear the plate towards end of margin $G = \gamma_r 2m.t \dots 3.19$

where γ_r = shear stress of plate = 28 N/mm^2

m = lapping margin = 16.8 mm

Substituting:

$$\begin{aligned} G &= 28 \times 2 \times 16.8 \times 2 \\ &= \underline{1881.6 \text{ N}} \end{aligned}$$

c) Efficiency of the riveted joint is given as $\eta = \frac{G-d}{G} \times 100 \dots (3.20)$

where, G = pitch = 16.8 mm

d = hole diameter = 8.4 mm

Substituting values

$$\begin{aligned} \eta &= \frac{16.8 - 8.4}{16.8} \times 100 \\ &= \frac{8.4}{16.8} \times 100 = 50\% \end{aligned}$$

3.3.3 PIN JOINT DESIGN

The spinning disc is secured to the drive shaft by means of a cylindrical pin (fig. 3) Sharma et al, (1998) gave the mean diameters of this pins to be $0.2d - 0.25d$ (where d = shaft diameter). Therefore, given our shaft diameter to be 10 mm

Pin diameter, $d = 0.25 \times 10 = 2.5 \text{ mm}$

However, from the standard ISO – 2340 table given in Krutz et al (1984) the least pin diameter = 3mm while the hole = 3.10mm

3.4 HOPPER DESIGN

3.4.1 HOPPER SHAPE

The Hopper shape is cylindrical with a conical bottom half to allow for greater flow of materials

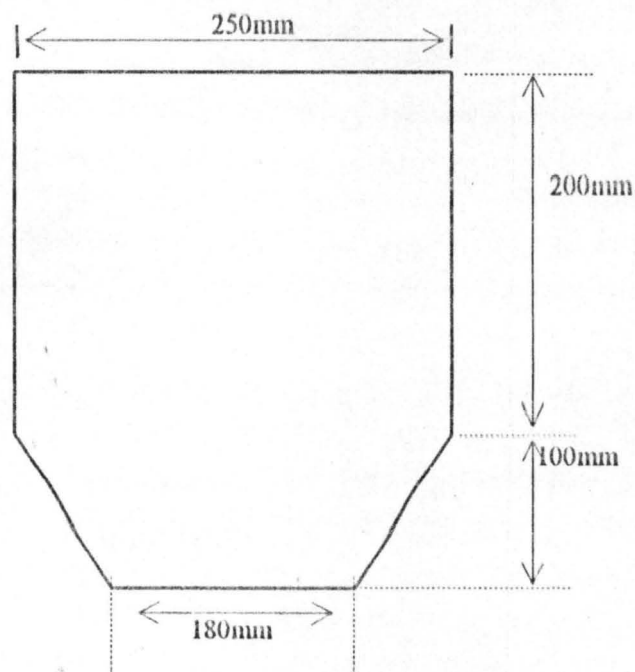


Fig (6) Hopper shape

3.4.2 HOPPER CAPACITY

The continuous load carrying capacity for an average weight male of 60kg is 20kg (Hiraga, 1988). In designing for the 20kg gross weight, an allowance is made for the net weight of the machine.

Bosai et al (1988) stated that it is a standard practice to make the neck width of hoppers between 250-300mm. For this reason, the neck width of the hopper is also chosen to be 250mm (see fig. 6 above)

The hopper is planned to have an overall height of 30cm. The cylindrical part is 20cm high and the conical second half is 10cm.

From the volume equation

$$V = \pi r^2 h \text{ -----3.21}$$

Where,

r = radius of hopper,

h = height

$$\begin{aligned} \text{Therefore } V &= \pi (12.5)^2 \times 20 \\ &= \underline{9817.5\text{cm}^3} \end{aligned}$$

And for the conical second half, the diameter of the base is 18cm

$$\text{Hence } V = 1/3\pi h (R^2 + Rr + r^2) \text{ -----3.22}$$

Where

h = vertical height = 10cm

R = upper radius = 12.5cm

r = lower radius = 9cm

substituting

$$V = 1/3\pi \times 10 [12.5^2 + (12.5 \times 9) + 9^2]$$

$$= 10.47 (156.25 + 112.5 + 81)$$

$$= 10.47 \times 349.75$$

$$V = \underline{3661.88\text{cm}^3}$$

$$\text{Total volume of the hopper therefore} = 9817.5 + 3661.88 = 13,479.38\text{cm}^3$$

And from density equation, the total mass of material the hopper can hold is

given by,

$$M = \rho V \text{ ----- } 3.23$$

$$\text{Where, } V = \text{volume} = 13,479.38\text{cm}^3$$

$\rho = \text{density} = 0.83\text{g/cm}^3$ (average for rice, soyabean, phosphate, Urea fertilizer (extracted from appendix 2))

Therefore,

$$M = 0.83 \times 13479.38$$

$$= 11,187.89 \text{ g} \quad \text{or}$$

$$\underline{11.2\text{kg}} - \text{average}$$

For individual materials whose densities are stated in the table the mass of the materials are:

11.9kg for Rice

10.12kg for Soyabean

16.85kg for phosphate rock

5.9kg for Urea crystal

3.5 DISCHARGE APERTURE

Mohsenin, (1970) stated that "to determine the critical dimensions of the hopper opening (aperture), failure condition must be established for two basic obstructions; namely "ARCHING" where no flow may take place and "PIPING", where flow may be reduced or limited". These obstructions come as a result of insufficient static pressure at the bottom of the hopper especially when the aperture created are small.

Hence, to obtain optimum aperture diameter, Jansen's equation (in Mohsenin 1970) was used in predicting pressure at varying depth in a slope sided hopper. The equation is as follows

$$L_p = \frac{W}{R} [1 - e^{-K_f s^{1/R}}] \text{ -----3.24}$$

where

L_p = lateral pressure, g/min

R = Hydraulic radius or cross-section area to
circumference m^3/m

W = Weight density of materials, kg/m^2

F_s = Static coefficient of friction of material against wall

h = depth of material, m

The experiment was conducted using two hoppers, one with wall inclination of 0° while the second with wall inclination of 20° . For coarse materials it was discovered that if the diameter of the aperture $B > 8d$ (where d = largest particle size diameter), a base pressure of 49 – 98 kg/m^2 will provide satisfactory flow. Using the equation for finding minimum dimension which states that:

$$B = \frac{P \sin 2\alpha}{W_2} \quad (\text{Mohsenin, 1970}) \quad \text{-----} \quad 3.25$$

Where,

B = aperture min. diameter, mm

P = base pressure, kg/m^2

W = bulk density of material, kg/m^3

α = Arching angle of slot opening (Monsoon, pp 601 –
602) with horizontal (assumed 30°)

Taking minimum base pressure of 49kg/m^2 and the various material bulk densities and substituting values in equ (3.25), the minimum aperture opening will be as follows

3.5.1 Calculating for the rice;

$$B = \frac{49\text{kg/m}^2 \times \sin(2 \times 30^\circ)}{0.88\text{g/cm}^3}$$

where,

$$\begin{aligned} 0.88\text{g/cm}^3 &= 0.88 \times 10^{-3} \times 10^6 \text{ kg/m}^3 \\ &= 880\text{kg/m}^3 \end{aligned}$$

Hence,

$$\begin{aligned} B &= \frac{49}{880} \times \sin 60^\circ \\ &= 0.055689 \times 0.8660 \\ &= 0.048\text{m or } 4.8\text{cm} \end{aligned}$$

$$B = \text{Approx. } 5\text{cm}$$

3.5.2 For Soyabean,

$$B = \frac{49\text{kg/m}^2 \times \sin 60^\circ}{0.75\text{g/cm}^3}$$

$$\text{And } 0.75\text{g/cm}^3 = 0.75 \times 10^{-3} \times 10^6 \text{ kg/m}^3 = 750\text{kg/m}^3$$

Hence,

$$\begin{aligned} B &= \frac{49 \times 0.8660}{750} \\ &= 0.0653 \times 0.8660 = 0.0565\text{m or } 5.65\text{cm} \end{aligned}$$

$$B = \text{Approx. } 6\text{cm.}$$

3.5.3 For Phosphate rock,

$$B = \frac{49 \text{ kg/m}^2}{1.25 \text{ g/cm}^3} \times \sin 60^\circ$$

where,

$$\begin{aligned} &= 1.25 \text{ g/cm}^3 \times 10^{-3} \times 10^6 \text{ kg/m}^3 \\ &= 1250 \text{ kg/m}^3 \end{aligned}$$

Hence,

$$\begin{aligned} B &= \frac{49}{1250} \times 0.8660 \\ &= 0.0392 \times 0.8660 = 0.0339 \text{ m or } 3.39 \text{ cm} \\ B &= \text{Approx. } 3 \text{ cm} \end{aligned}$$

3.5.4 For Urea fertilizer,

$$B = \frac{49 \text{ kg/m}^2}{0.44 \text{ g/cm}^3} \times 0.8660$$

where,

$$\begin{aligned} 0.44 \text{ g/cm}^3 &= 0.44 \times 10^{-3} \times 10^6 \text{ kg/m}^3 \\ &= 440 \text{ kg/m}^3 \end{aligned}$$

Hence,

$$\begin{aligned} B &= \frac{49}{440} \times 0.8660 \\ 0.1114 \times 0.8660 &= 0.0965 \text{ m or } 9.65 \text{ cm} \\ B &= \text{Approx. } 10 \text{ cm.} \end{aligned}$$

The minimum average dimension of the aperture = 6cm

For a satisfactory flow therefore $B > 8d$. If soyabean has the largest particle size of 6.6mm

It implies that $6 > 8 \times 0.66$

Therefore, $6 > 5.28$

This implies a satisfactory aperture dimension

Here, slotted aperture is chosen, hence extreme dimension are used as can be seen in figure 7 below (i.e. 50cm x 7cm)

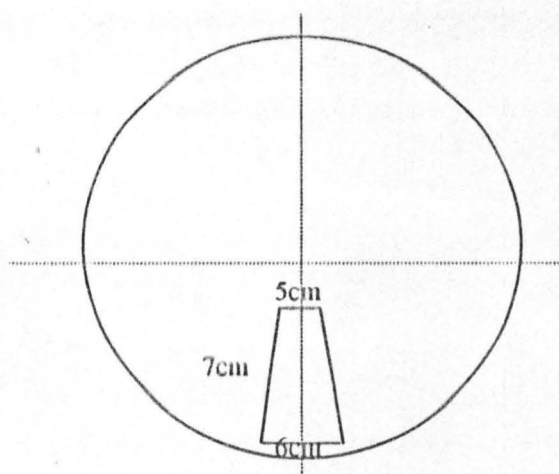


Fig (7): Aperture Dimension

3.6 APPLICATION RATE

3.6.1 THEORETICAL FLOW RATE

Mohsenin, (1970) quoted the equation for the calculation of flow of granular materials through orifice as proposed by Beverloo et al (1961) as follows.

$$Q = 35W \sqrt{g (B-1.4d)^{2.5}} \text{ ----- 3.26}$$

where

Q	=	flow rate, g/s
W	=	Bulk density, g/cm ³
g	=	gravitational acceleration, cm/s ²
B	=	aperture diameter, cm
d	=	average diameter of the particles, cm

Using W and d parameters for the four materials used for this design work and substituting into equation (3.26) above the flow rates are shown in table below

	Rice	Soyabean	Phosphate rock	Urea Crystal
Bulk density, W	0.88	0.75	1.25	0.44
Particle diameter, d	0.32	0.66	0.043	0.066
Flow rate, Q	68g/s	51g/s	106g/s	37g/s

Table (1): Flow rate of various materials through the orifice: (Extract from Appendix 6).

CHAPTER FOUR

4.0 CONSTRUCTION AND PRINCIPLE OF OPERATION

4.1 MAIN FEATURES OF THE MACHINE

Fig. (8), appendix 1 shows the Isometric view of the machine while the details as explained in section 4.1.1 to 4.1.11 below are shown in Figures (9) and (10) appendix 2 and 3 respectively.

4.1.1 HOPPER

The hopper is constructed from 1.5mm thick aluminium sheet. It is an open cylinder with a conical bottom. The cylindrical portion is 20cm in height which the conical bottom is 10cm high. The total height of the hopper is 30cm.

4.1.2 HOPPERS BOTTOM

The hopper's bottom is made from 18mm thick plywood. The discharge aperture is provided here. The size of the aperture is also tied opening made large enough 5cm x 7cm to allow free flow of materials it is same as in (Fig. 7).

4.1.3 APERTURE ADJUSTING A PLATE

In order that the discharge materials can be regulated, an adjusting plate with a handle is placed under the hoppers bottom to open or close the discharge aperture. It is made of steel sheet same size of hopper bottom and having the same aperture shot on it.

4.1.4 SPINNING DISC

In order to reduce load on the electric motor, the spinning disc and the vanes are made from 1.5mm thick aluminium sheet. The disc diameter is 25cm with four vanes riveted at equal distance (See Fig 5).

4.1.5 DRIVE SHAFT

The driver shaft is expected to encounter both torsional and bending loads as a result of the rotating action of the driven pulley, spinning disc and also belt tension. In considering these load factors and also availability and marketability, steel is chosen. The shaft is 10mm in diameter and 220mm in length.

4.1.6 BEARINGS

The shaft rotates continually in operation. It therefore creates friction both inertia and in operation. Hence, wear and high initial force of rotation must be overcome. Therefore, two appropriate ball roller bearings are used to guide the shaft in operation.

4.1.7 ELECTRIC MOTOR (DC)

The machine is powered by a direct current (DC) motor. Since the machine is designed to broadcast various materials, variable speed is therefore a critical factor. Therefore the motor was selected to accommodate these varying speed required through the use of speed regulator.

4.1.8 BELT AND PULLEYS

There are two pulleys fabricated from steel material. The drive pulley is 30mm in diameter while the driven pulley is 135mm in diameter. On the other hand the belt is carefully selected from the general agricultural rubber belts measuring (9.5mm & 700mm).

4.1.9 SUPPORTING FRAME

Due to weight consideration, the machine was supported by a simple frame structure using a 2.5mm flat steel bar with upper and lower stationary steel cover which have roller bearings and aluminium sheet cover.

4.1.10 BATTERY

The electric motor is a 12 volt induction motor, hence the choice of a 12 volt – 4AH motorcycle battery. The choice of a motorcycle battery was made due to the fact that most farmers own motorcycles and therefore can easily be accessible both for use and recharging.

4.1.11 Dynamo

A 12V bicycle dynamo is provided for battery recharging for a longer operating time to be sustained

4.2 DESCRIPTION OF THE MACHINE

The machine was design with a simple frame structure that made it lock very compact with overall height standing at 450mm and width 380mm. The main drive shaft is held in place by the two ball roller bearing fixed to the upper and lower stationary plate of the frame and is extended upward into the hopper to hold the two agitators.

The spinning disc which is made of allumniuim sheet and riveted to a steed flange is also keyed to the main drive shaft to receive dive. The spinning disc is placed immediately on top of the upper stationary plate and directly below the hopper to receive materials to be broadcasted. The hopper is mounted on the frame work directly above the spinning disc. The fixed hopper bottom and the aperture adjusting (opening and closing) disc are directly mounted to the hopper bottom to receive and discharge materials accordingly.

At the protruded side of the frame structure is mounted the D.C motor to provide rotational drive of the spinning disc and agitators via pulleys and belt drive. A 12V-moto cycle battery is provided to induce power to the motor while a regulator is also provided to regulate the speed of the motor to suit the type of material to be broadcasted (Fig. 11).

A dynamo was provided for continuous recharging of the battery while in operation. It is mounted to one of the frame vertical support and receives drive from the drive belt.

4.3 OPERATING PRINCIPLES

With the discharge aperture closed, materials to be broadcasted were poured into the hopper. The machine was lifted up against the chest and carried on the shoulder by the use of the two belts straps. Operation commences by first switching on the motor. As the disc rotates, materials are let loose on to the rotating disc by opening of the discharge aperture through the feed regulating handle. Forward movement was carried out steadily with constant pace to achieve uniformity of spread and rate of application.

In operation, the spinning disc rotating action was achieved when the switch was switched-on, power from the battery was directed to the electric motor through the regulator. The regulator allows for speed variation. Through the multiplier effect of the field winding, the high current developed by the field windings induces rotation of the armature. The rotating action of the armature shaft is then transferred to the spinning disc shaft via the belt.

4.4 TEST PERFORMANCE OF THE MACHINE

After the construction and complete assembly, the machine was tested to ascertain certain parameters.

1. Free and efficient rotation of component parts unloaded and loaded with materials
2. Field capacities for various materials

4.4.1 Procedure used in carrying out tests

Test (i)

The battery was fully charged and properly connected. The hopper cleared making sure it is free from any material or obstruction either to the agitators or spinning disc. The machine was then switched on for the motor to run all component parts.

Test (ii)

The Test (i) above was then repeated with materials (Rice) loaded into the hopper in order to test the power of the motor pushing the agitators through the materials.

Test (iii)

Field capacities for four (4) materials, namely, Rice, Soyabeans, urea fertilizer and Beniseed were evaluated through the following methodology. All the materials are local varieties purchased from the local market.

1. Two (2) measures (mudu) each of the materials were purchased properly cleaned and weighed with the following figures obtained. Rice – 2.2kg, Soyabeans – 2.8kg, Beniseed – 2.3kg. and urea fertilizer – 2.3kg.
2. A length of 40 metres was marked out from a field for the purpose of the test.
3. The person carrying the machine for the test was asked to lift and carry the machine on his shoulder as he will normally do during field operation. The spreading height was measured from the surface of the ground to the level of the spinning disc and 1 metre was obtained.

4. The material (seed) was then emptied onto the hopper-ready to commence test.
5. The machine was switched-on, with a constant pace, the operator began his movement about five metres before the marked line. As he approaches the line, the discharge aperture was opened through the means of an operating lever.

Note: All test were carried out with the aperture operating lever half opened. The operator goes through the 40 metres marked and shuts the operating lever as he crosses the last marked line.

6. Time taken to go through the 40 metre was noted by means of a timing watch.
7. The width of spread was measured.
8. Quantity used in spreading through the 40 metres length was calculated by weighing the balance of material in the hopper and subtracting it from the initial quantity before operation commenced.
9. Application rate was calculated (See next Chapter)

Note: For each material, the above test were repeated two times and the average taken.

The results of the tests are given in chapter five.

Fig. (8): Picture of the machine in action

Table (2) COST ESTIMATE FOR THE PRODUCTION OF THE BROADCASTER

S/N	Component	Material	Specification	Qty	U/Cost (N)	T/Cost
1.	Hopper	Mild steel sheet	0.24m ²	1	204.00	204.00
2.	Spinning Disc	Alluminium sheet	Ø 0.25m	1	38.00	38.00
3.	Aperture Adj. Plate	Mild steel sheet	Ø 0.18m	1	21.00	21.00
4.	Stationery aperture disc	Plywood	Ø 0.18m	1	150.00	150.00
5.	Drive shaft	Steel	Ø 0.01mx0.022m	1	150.00	150.00
6.	Agitator	Mild steel	Ø 0.005mx0.06m	2	25.00	50.00
7.	Drive Pulley	Mild steel	Ø 0.03m	1	150.00	150.00
8.	Driven Pulley	Mild steel	Ø 0.135m	1	450.00	450.00
9.	Drive Belt	Rubber	Length 0.71m	1	250.00	250.00
10.	Electric Motor	-	D.C	1	2000.00	2000.00
11.	Regulator	-	-	1	250.00	250.00
12.	Switch	-	-	1	650.00	650.00
13.	Dynamo	-	12V	1	850.00	850.00
14.	Main Frame	Mild steel		1	200.00	200.00
15.	Upper & lower fixed plates	Mild steel sheet	Ø 0.3m	2	60.0	120.00
16.	Side cover plates	Alluminium sheet	0.022mx0.1m	9	16.00	144.00
17.	Shoulder strap	Nylon woven	0.25mx1m	2	60.00	120.00
18.	Welding & fabricating work	-		-	-	<u>2000.00</u>
						<u>7,797.00</u>

CHAPTER FIVE

5.0 RESULTS AND DISCUSSION

5.1 RESULTS (i) & (ii)

The first and second tests carried for the sole purpose of testing the harmonious and efficient working of all component parts of the machine was done. The first test was done with an empty hopper with materials. With this first test, all component parts rotate smoothly without any hindrance.

However, when the machine was loaded with 2.2kg of rice and switched on there was a 2 seconds hick-up before the spinning disc began to rotate. This shows the resistance the motor have to overcome to be able to push the agitators through the material (rice). However, the machine ran smoothly thereafter with drastic reduction in speed from 500rpm to 350rpm. Note – A mechanical tachometer was used to measure the speed of the main drive shaft before and when machine was under operation.

5.1.1 Test result (iii) – Field Capacities

The result of field test carried out for different (agricultural) materials namely:- Rice, Soyabean, Urea Fertilizer and Beniseed are given in table (3).

Table (3) Field Test Result

S/N	Material	Qty used (kg)	Time taken (sec)	Width of spread (m)
1.	Rice	1.1	36	2.3
2.	Soyabeans	1.4	34	2.8
3.	Urea Fertilizer	1.15	35	2.6
4.	Beniseed	0.95	35	2.5

The above field test result is used to further obtain additional information needed such as area covered by the various materials through the 40 metric strip earmarked for the test, total time it will require to cover 1 hectare of land and the rate of application of each material per hectare of land.

- a) Area covered through the 40 metre length is given as: $\text{Area} = L \times B$
 where, L = Length of field used in testing = 40m
 B = width of spread of material, (m)

Therefore, for rice,

$$\begin{aligned}\text{Area} &= L \times B \\ &= 40 \times 2.3 = 92\text{m}^2\end{aligned}$$

For soyabeans,

$$\begin{aligned}\text{Area} &= L \times B \\ &= 40 \times 2.8 = 112\text{m}^2\end{aligned}$$

For Urea fertilizer,

$$\begin{aligned}\text{Area} &= L \times B \\ &= 40 \times 2.6 = 104\text{m}^2\end{aligned}$$

For Benniseed ,

$$\begin{aligned}\text{Area} &= L \times B \\ &= 40 \times 2.5 = 100\text{m}^2\end{aligned}$$

- b) Total time taken to cover 1 hectare is given as:

$$T_t = \frac{\text{Area of 1ha (m}^2\text{)} \times \text{time taken for 40m length, (sec)}}{\text{Area of 40m length (m}^2\text{)}}$$

Hence, for rice,

$$\begin{aligned}T_t &= 10,000 / 92 \times 36 \\&= 3913 \text{ sec or } 65 \text{ min}\end{aligned}$$

For soyabeans,

$$\begin{aligned}T_t &= 10,000 / 112 \times 34 \\&= 3035.7 \text{ sec or } 51 \text{ min}\end{aligned}$$

For Urea fertilizer,

$$\begin{aligned}T_t &= 10,000 / 104 \times 35 \\&= 3365.4 \text{ sec or } 56 \text{ min}\end{aligned}$$

For banniseed,

$$\begin{aligned}T_t &= 10,000 / 100 \times 35 \\&= 3500 \text{ Sec or } 58 \text{ Min}\end{aligned}$$

c) Application rate (kg/ha) of individual material was calculated as

$$\text{Rate} = \frac{\text{Area of 1ha} \times \text{Qty of material used in 40m length}}{\text{Area for 40m length}}$$

Hence, for rice,

$$\begin{aligned}\text{Application rate} &= 10,000 / 92 \times 1.1 \\&= 108.69 \times 1.1 \\&= 119.56 \text{ or about } 120 \text{ kg/ha}\end{aligned}$$

For soyabeans,

$$\begin{aligned}
 \text{Application rate} &= 10,000 / 112 \times 1.4 \\
 &= 89.29 \times 1.4 \\
 &= 125 \text{ kg/ha}
 \end{aligned}$$

For Urea fertilizer

$$\begin{aligned}
 \text{Application rate} &= 10,000 / 104 \times 1.15 \\
 &= 96.15 \times 1.15 \\
 &= 110.57 \text{ or about } 111 \text{ kg/ha}
 \end{aligned}$$

For Bennissed

$$\begin{aligned}
 \text{Application rate} &= 10,000 / 100 \times 0.95 \\
 &= 100 \times 0.95 = 95 \text{ kg/ha}
 \end{aligned}$$

S/N	Material	Area covered for 40m (m ²)	Time cover 1ha (min)	Application rate (kg/ha)
1.	Rice	92	65	120
2.	Soyabeans	112	51	125
3.	Urea Fertilizer	104	56	111
4.	Beniseed	100	56	95

Table 4: Field Capacities as calculated using table 3.

5.2 DISCUSSION

The initial hick-up when loaded with materials was as a result of the use of inappropriate motor capacity. The designed capacity DC motor that is portable and light weight could not be obtained in the market. The alternative used is about half the designed capacity.

Table (3) above shows that the pacing was fairly uniform in all the test performed. While the pattern of width of spread (swath) obtained conforms with the factors that controls the motion of particles as reported by Patterson & Reece, (1962) in chapter two. The swath obtained for Rice therefore could be adjudged from its frictional resistance even though, it has a higher density than that of urea fertilizer. Soyabean which has a smooth surface and circular in shape was able to attain a higher spreading width because it was able to overcome frictional resistance of both the disc and vane and also air as it slides and rolls on the disc and vane and so also through the wind resistance due to its weight. The result obtained on Beniseed confirms with Aji, (1977)

In Table (4), the results obtained through calculations from the data in Table (3) shows that:

- (1) The machine can adequately be used to cover 1 hectare of land within 1 hour of operation for all the materials.
- (2) Application rates calculated for all materials are within recommended agronomic specifications. With proper aperture adjustments to suit each type of material, specific application rates can be achieved. For instance, recommended application rate for the broadcasting of rice is 80kg/ha – 100kg/ha. While that of Soyabean is 100kg – 130kg and Urea fertilizer is 100kg/ha – 150kg/ha.

CHAPTER SIX

6.0 CONCLUSION AND RECOMMENDATIONS

6.1 CONCLUSION

This portable motorized seed and fertilizer broadcaster was designed, constructed and evaluated with the following conclusion reached.

1. It has a net weight of 7.5kg and a gross weight of 23.5kg which is slightly higher than the recommended continuous load carrying capacity for an average weight male of 60kg
2. The design and construction is simple and adaptable.
3. The material used in construction are easily obtainable and affordable.
4. The overall performance of the machine is excellent except for DC motor where the designed capacity could not be obtained.
5. All broadcast operations could be achieved within the range of one hour per hectare and this time can further be reduced with the right capacity motor. This could further improve on the width of spread when the right speed of rotation is attained.
6. With further work, the machine has very high prospect of being adopted by farmers with interest generated by local farmers around the field where the performance test was done.
7. The overall cost of the machine is about ₦7,800. 00
8. A swath width of 2.3 – 2.8 metres was attained for the various materials.

6.2 RECOMMENDATIONS

From the results of the evaluation done, the machine has very high potential if the following could be addressed.

1. Special design of 1kw light and compact DC motor for the machine or
2. Redesigning of the agitators in such a way that may not have a direct load bearing from the material in the hopper (i.e. agitator outside the hopper).
3. Overall size/weight through the use of strong but light materials.

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Appendix 1

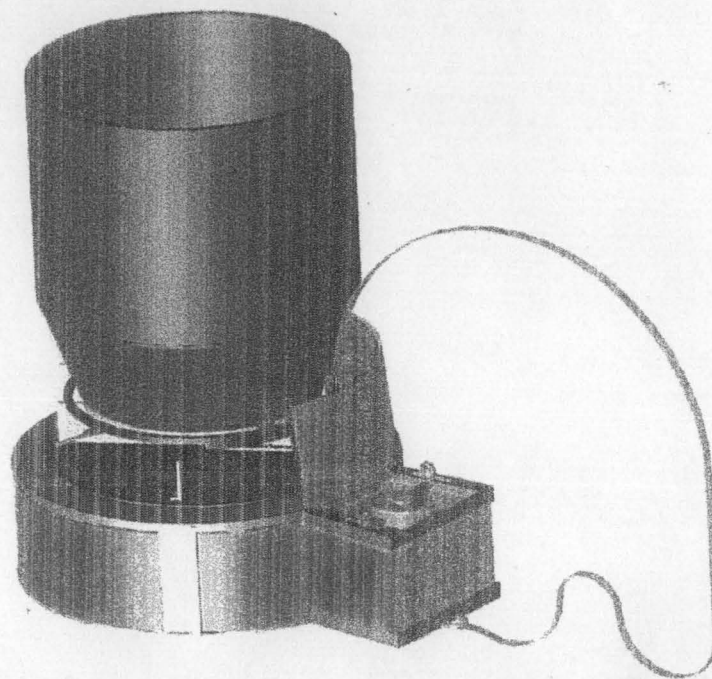
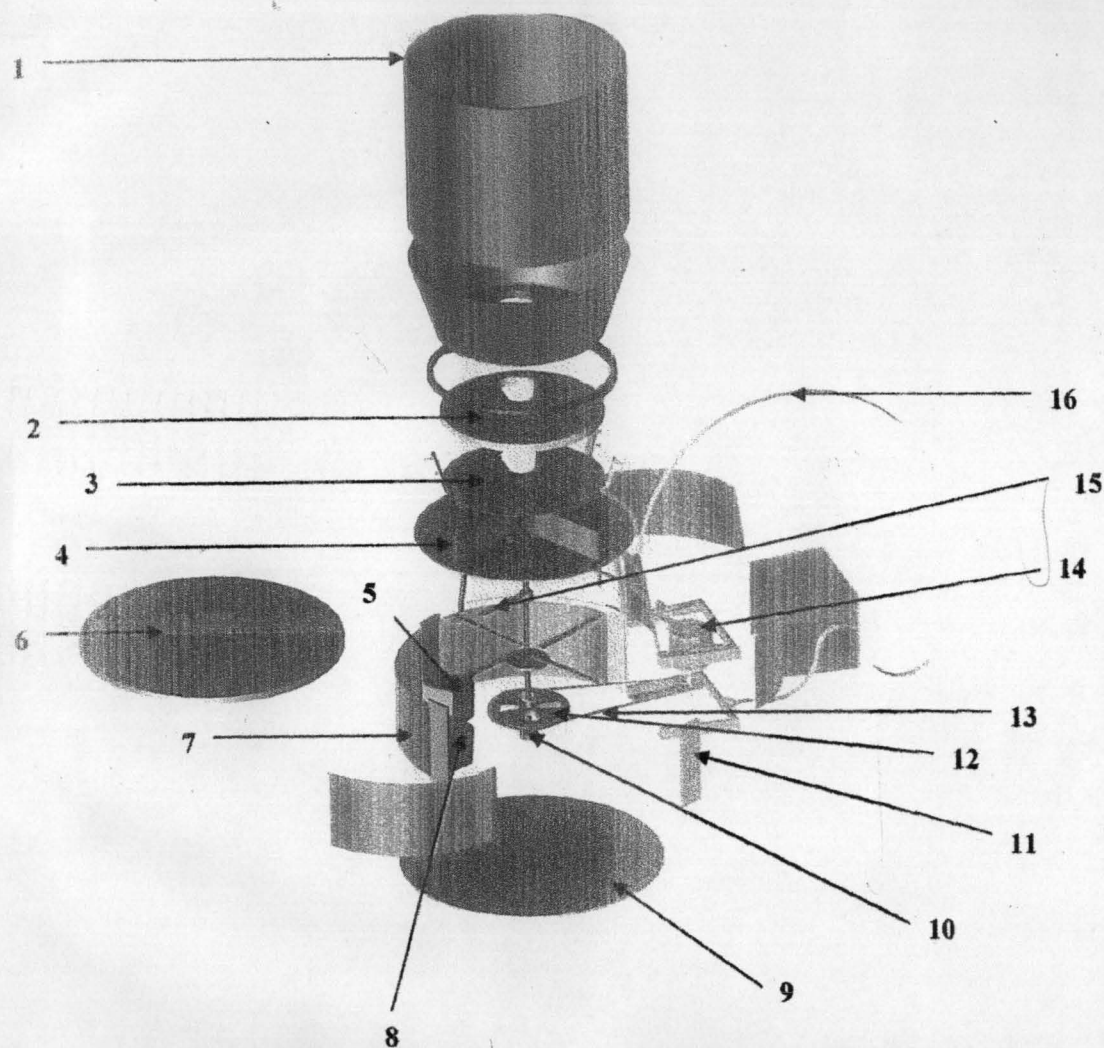


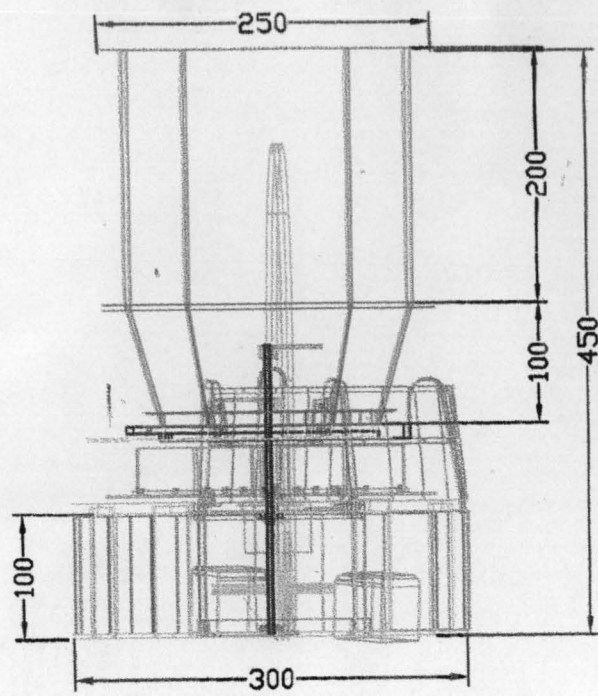
FIG (g) ISOMETRIC VIEW OF PORTABLE MOTORIZED SEED AND FERTILIZER BROADCASTER
ALL DIMENSIONS IN (MM)
SCALE 1: 6

Appendix 2

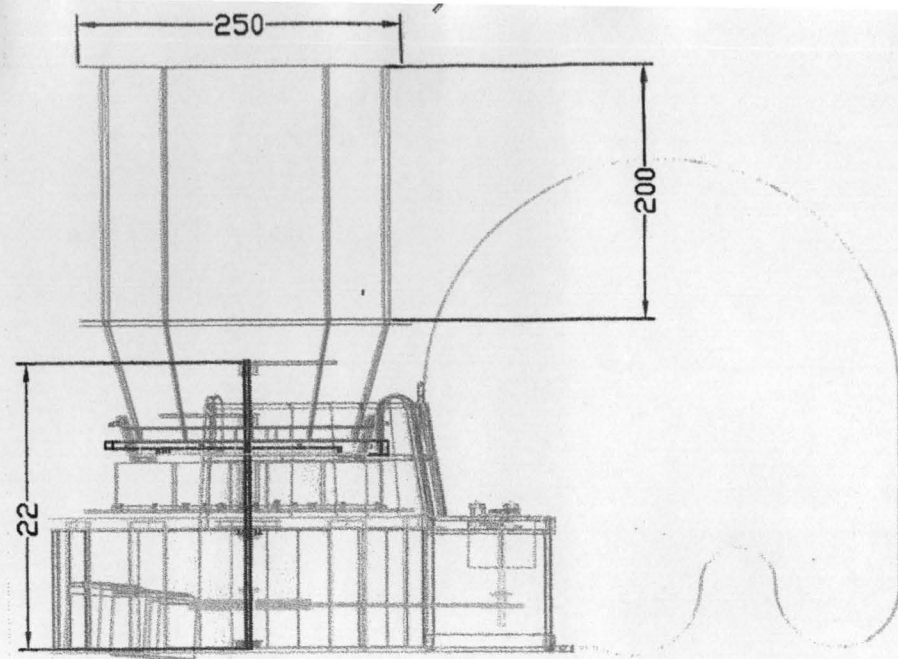


S/NO	LEGEND
1	Hopper
2	Stationary Aperture Plate
3	Aperture Adjusting Plate
4	Spinning Disc
5	Battery (12V)
6	Upper Fixed Plate
7	Cover Sheet
8	Regulator
9	lower fixed plate
10	Drive Shaft
11	Frame Support
12	Pulley (Ø 135 mm)
13	Driving Belt
14	D.C Motor
15	Agitator
16	Shoulder Stripe

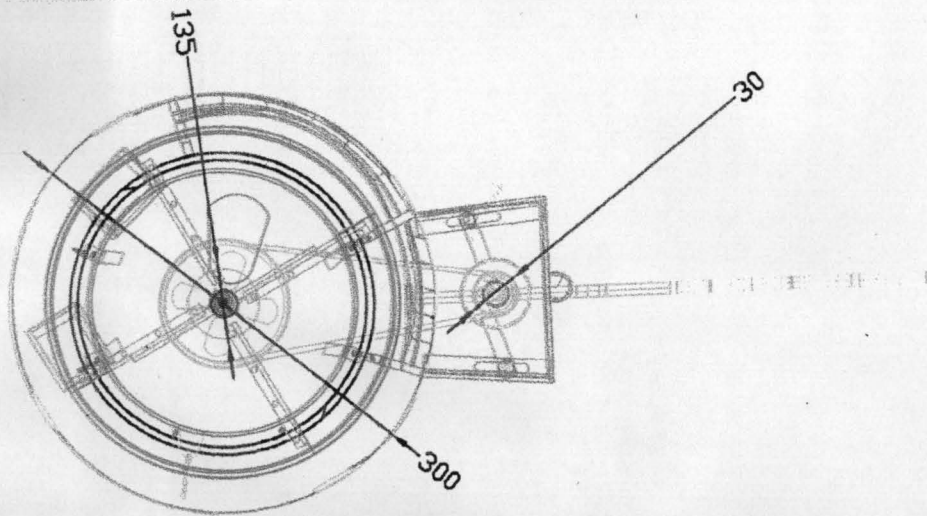
Fig (7) Exploded View of Potable Motorized Seed and Fertilizer Broadcaster Scale 1: 8
All Dimensions in (mm)



FRONT VIEW



SIDE VIEW

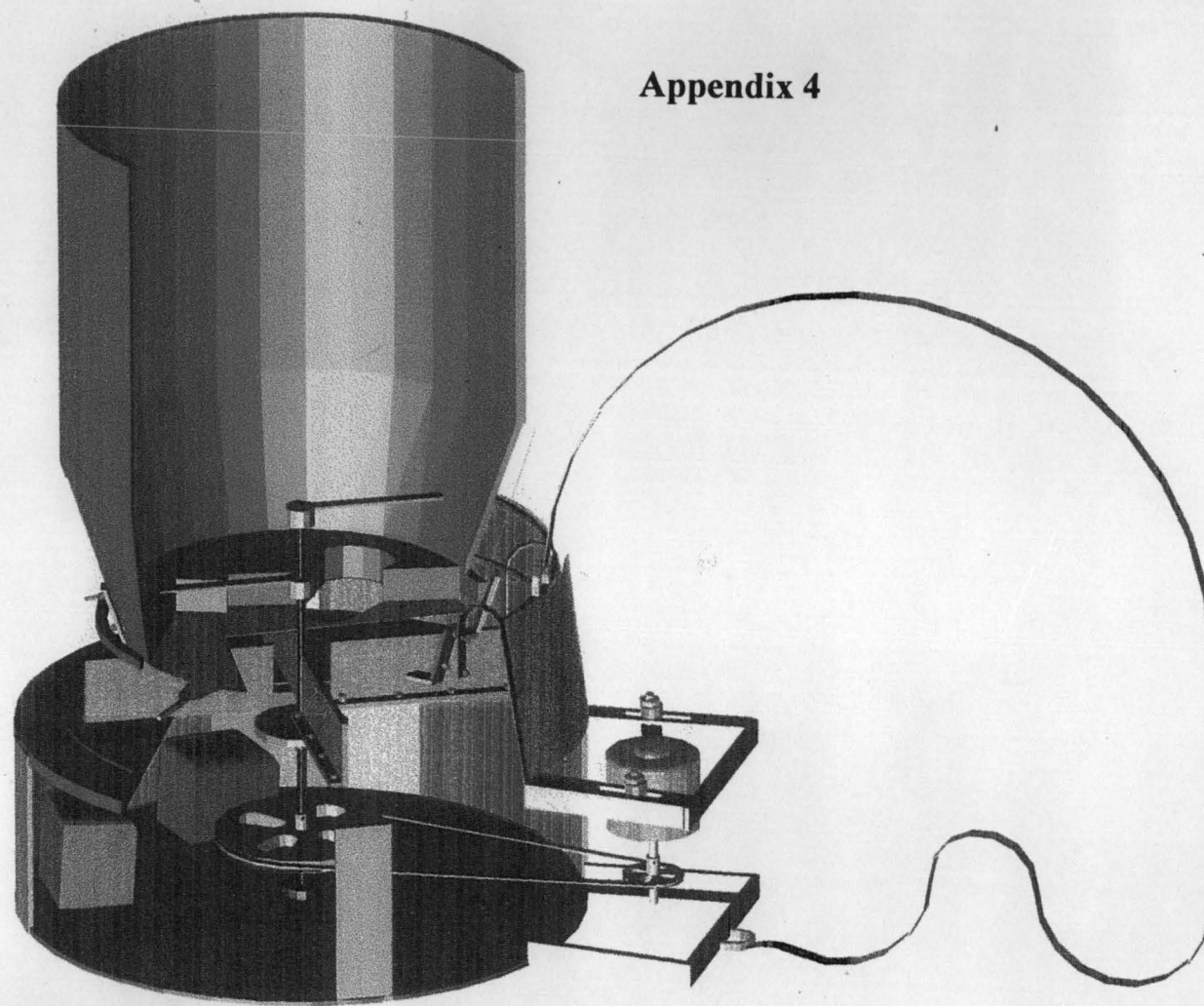


PLAN

FIG (10) ORTHOGRAPHIC
PROJECTION OF MOTORIZED SEED
AND FERTILIZER BROADCASTER
ALL DIMENSIONS IN (MM)

SCALE 1:6

Appendix 4



**FIG (4) SECTIONAL VIEW OF PORTABLE MOTORIZED SEED AND FERTERLIZER BROADCASTER
SCALE 1 :5**

APPENDIX 5

Value of rivets, holes and pitch of rivets with different plate thickness.

Thickness of plate mm	Diameter of rivet hole mm	Diameter of rivet mm	Pitch of rivets $p =$ $3d + 5 \text{ mm}$	Marginal pitch m in mm
2	8.4	8	29	16
3	9.5	9	32	17
4	11	10	35	17
5-6	13	12	41	18
6-8	15	14	47	21
8-12	17	16	53	25
11-15	21	20	65	30

Source: Sharma and Aggarwal (1998) Table 7.9

Appendix 6
Summary of observed and calculated flow rates for Varian agricultural materials
(taken table 10.4 in mohseun).

Material	Orifice diam. B Mm	Particle diam. D mm	Bulk density w mm	Solid density w ₂ g/cc	Repose p deg	Wall angle from vertical o (deg)	Pressure in g/min obsd.	Calcd.	Predicting eqn.
Lead shot #10	10.1	1.78	6.55		19.5	15	7200	7350	10.33
Lead shot #12	10.70	1.25		10.8	20.2	30	5560	5520	10.33
						0	4990	5450	10.34
Glass beads	10.1	3.54	1.57		27.4	15	1295	1315	
Glass beads	10.7	0.788		2.77	25	3000	961	961	10.33
Marbles	75	13.5	1.32		27	0	271	220	10.33
Crushed rock	34.3	4.01		3.3	37	30	129000	137000	10.34
Sand	50.7	0.92	16			0	17400	18600	10.34
Phosphate rock	5	0.43	1.25		34.6	0	80,000	87,500	10.36
							147	154	10.33
Urea crystals	10.1	0.66	0.44		45	30	333	335	10.33
Puffed Rice	33.7	5.1		0.12	31.5	0	133	135	10.33
Sugar	25	0.9	0.83			0	263	313	10.34
Vetch seed	15	3.4	0.82		22.5	45	750	690	10.37
Kale seed	10.1	1.7	0.69		22.5	15	8050	8415	10.33
Mustard seed	10.1	2.2	0.75		25.1	30	1400	1450	10.33
Green peas	40	7.0	0.84			0	675	690	10.37
Soyabeans	40	6.6	0.75			15	621	699	10.37
Lupin	40	6.5	0.70			15	14340	12590	10.37
Wheat	25.1	5.0	0.8			0	3830	3820	10.37
Wheat	25.2	4.1	0.86			0	4440	5980	10.36
Rice	25.3	3.2	0.88			0	5640	6920	10.36