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

BY

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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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Mentioned must be made of my dear friends Mr. B. O. Angba of the Works Department, Kaduna Polytechnic for his hospitality and encouragement, Mr. P.V. Kwaya of the Department of Agricultural Engineering, Federal Polytechnic, Nasarawa also his hospitality and guidance during the construction. I will like to extend special gratitude to my old teacher and a dear friend, Mr. E.C. Nwalor of the Department of Agricultural Engineering, Kaduna Polytechnic as usual for his untiring words of encouragement and guidance. He directed me to all the appropriate places at ABU Zaria for most of the literature I got. Also, mention must be made of Mr. P. Gwajacks of the Department of Agricultural Engineering Workshop, ABU Zaria who painstakingly guided me through both the school and Departmental library. Special thanks goes to Mr. Paul Gabriel of

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

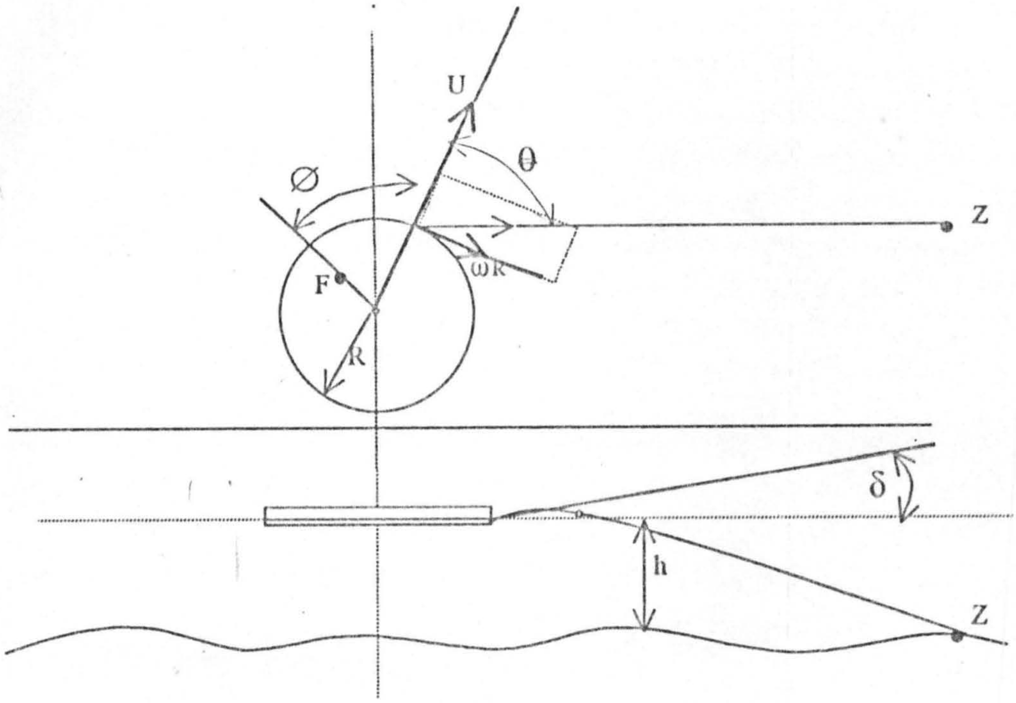


Fig. (1) important variables in the centrifugal distribution analysis (Paterson and Reece, 1962).

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.

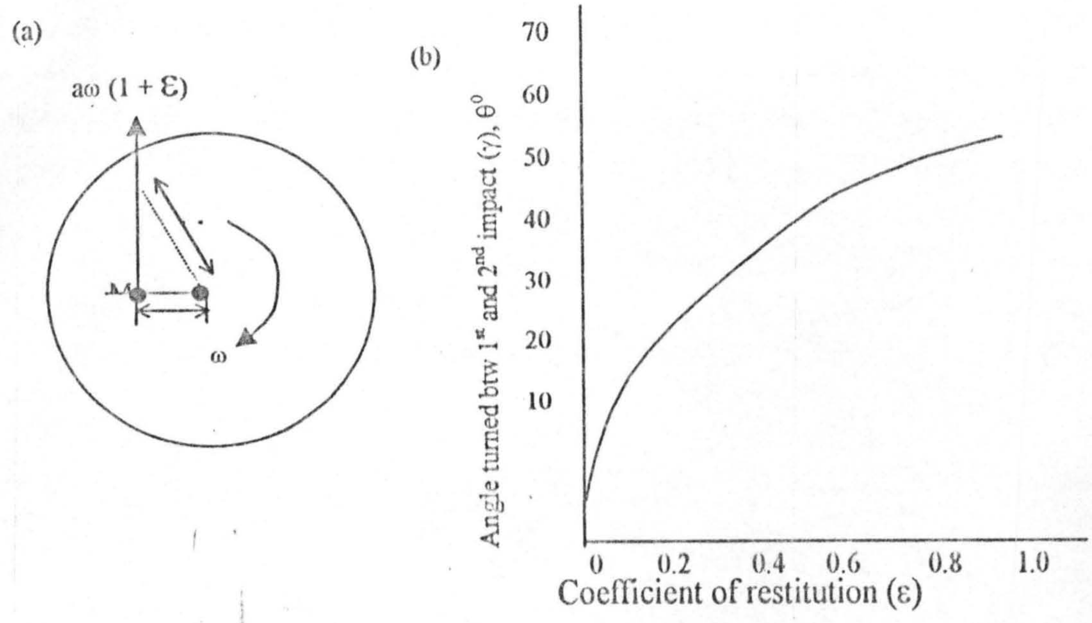


Fig.(2) Variation of δ with different values of ε (Reeece & Inn 1962)

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 'aω' 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 εaω. Since the vane itself has a velocity in space of aw at that point, the total velocity of the particle will therefore be aω + εaω = aω (1 + ε) (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.

$$\tan \gamma_1 = \frac{a\omega (1 + \varepsilon) t_1}{A}$$

$$= \omega T_1 (1 + \varepsilon)$$

But $\omega T_1, = \gamma_1$

Thus, $\tan \gamma_1 = \gamma_1 (1 + \varepsilon)$ -------------------------------- 20

It should however be noted that angle γ_1 is independent of a and ω. Fig 2b above shows how γ_1 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 Ic = $ma\omega^2$. 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)

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

$$\rightarrow \quad + a^2 \sin^2 \psi_o \text{ .... } 2.2$$

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

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

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

$$\varnothing = \omega t_{pr} \quad \text{--------- 2.4}$$

and the final velocity of particle

$$V = \frac{fg}{\omega^2}\left\{\frac{1}{\lambda_2-\lambda_1}\left[(\lambda_2 \varnothing^{\lambda 1T} - \lambda_1 \varnothing^{\lambda 2T})\right] - 1\right\} \quad \text{.........2.5}$$

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 12^0-18^0 (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

$$\underline{N_1D_1} = N_2D_2 \quad \text{-------- ----------------} \quad 3.01$$

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

$$D_2 \frac{N_1D_1}{N_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.02$$

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

Therefore D_2 = <u>0.135m or 135mm</u>

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,

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

Where,

P = Power required, watts

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

$= [(M_s + M_d + M_p)g + CF_d + CF_p + T_b]$ N

V = Speed of the machine, m/s

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

- 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,

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

Where,

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

D = diameter of disc = 0.25m

T_1 = thickness of disc = 0.0015m

A_v = Area of Vane, (0.035 x 0.01) = 0.0035m^2

T_2 = thickness of vane = 0.0015m

Substituting values in to equ. (3.05) above

$$M_J = 2698 \{[\frac{\pi \times 0.25^2 \times 0.0015}{4}] + [0.0003 \times 0.0015]4\}$$

$$= 2698 [7.36 \times 10^{-5} + 2.1 \times 10^{-6}]$$

$$= 2698 \times (7.57 \times 10^{-5})$$

$$= \quad 0.2kg$$

3:1.2(b) MASS OF DRIVEN PULLEY

The mass of the driven pulley is calculated from the equation

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

Where,

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

V = Volume of pulley, M^3

D = diameter of pulley = 0.135m

T = thickness of pulley = 0.012m

substituting values on equation (3.06) above

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

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

Therefore $M_p =$ **1.35kg**

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

$$F_b = 0.3 \times 0.135 \times \left[\frac{1000\pi}{60}\right]$$

$$= 2.12N$$

3:1.2(d) BELT TENSION

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

$$\frac{T_1 - mV^2}{T_2 - mV^2} = \mathcal{C}^{(f\alpha/\mathrm{Sin}0/2)} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.08$$

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/m^3 (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 °

0 = Groove angle of pulley = 36°

But,

T_1 = belt width x thickness x shear stress where shear stress was taken as 1.2mpa x 10^6 N/m^2 (Haul et al, 1980).

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

$= 72N$

$$V = \underline{\pi 0 DN} = \frac{\times\ 0.135 \times 1000}{60} = 7.07\mathrm{m/s}$$

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

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

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

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

$C = 0.218m$

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

$$= 180\text{-}2Sin \frac{(0.135\text{-}0.30)}{2(0.218)}$$

$$= 180 - 2Sin \frac{0.105}{0.436}$$

$$= 180 - 27.87$$

Therefore $\alpha = 152.13 \quad \frac{(152.13xPi)}{180} \quad 2.66$ 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 $T_2 = 21.25N$

Total belt tension, $T_1 + T_2 = 72 + 21.25 = 93.25N$

3:1.3 LENGTH OF BELT REQUIRES

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

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

Where C = Centre distance between pulleys, = 0.218m

D_2 = diameter of larger pulley = 0.135m

D_1 = diameter of smaller pulley = 0.03m

Substituting,

$$L = 2(0.218) + 157(0.135+0.03) + \frac{(0.135-0.03)^2}{4(0.218)}$$

$$= \quad 0.436 + 0.259 + 0.013$$

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

3:1.4 SHAFT DIAMETER

The critical loads acting on the shaft are:

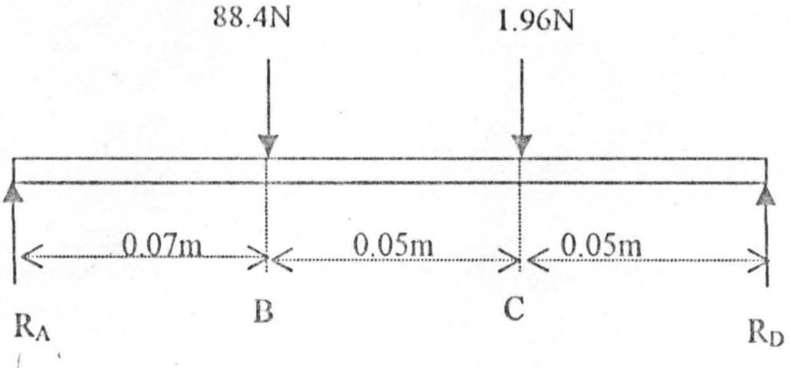
a) (i) weight of pulley = 1.35kg x 9.81 = 13.24N

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

Total force acting through the pulley = 106.5 N

b) (i) weight of spinning disc = 0.2kg x 9.81 = 1.96N

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

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

$$= 0.098 + 8.84$$

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

$$= \underline{52.58N}$$

$$R_A = 88.4 + 1.96 - R_A$$

$$= 90.36 - 32.58$$

Therefore $R_D = 37.78$

Shear force, SF

At pt A, Sf = Ra = 52.58

At pt B, SF = R_A- 88.4 = 35.82

At pt C, SF = R_A – 88.4 – 1..96 = -37.78

Bending Moment, BM

At pt A, BM = $R_A \times 0 = 0$

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

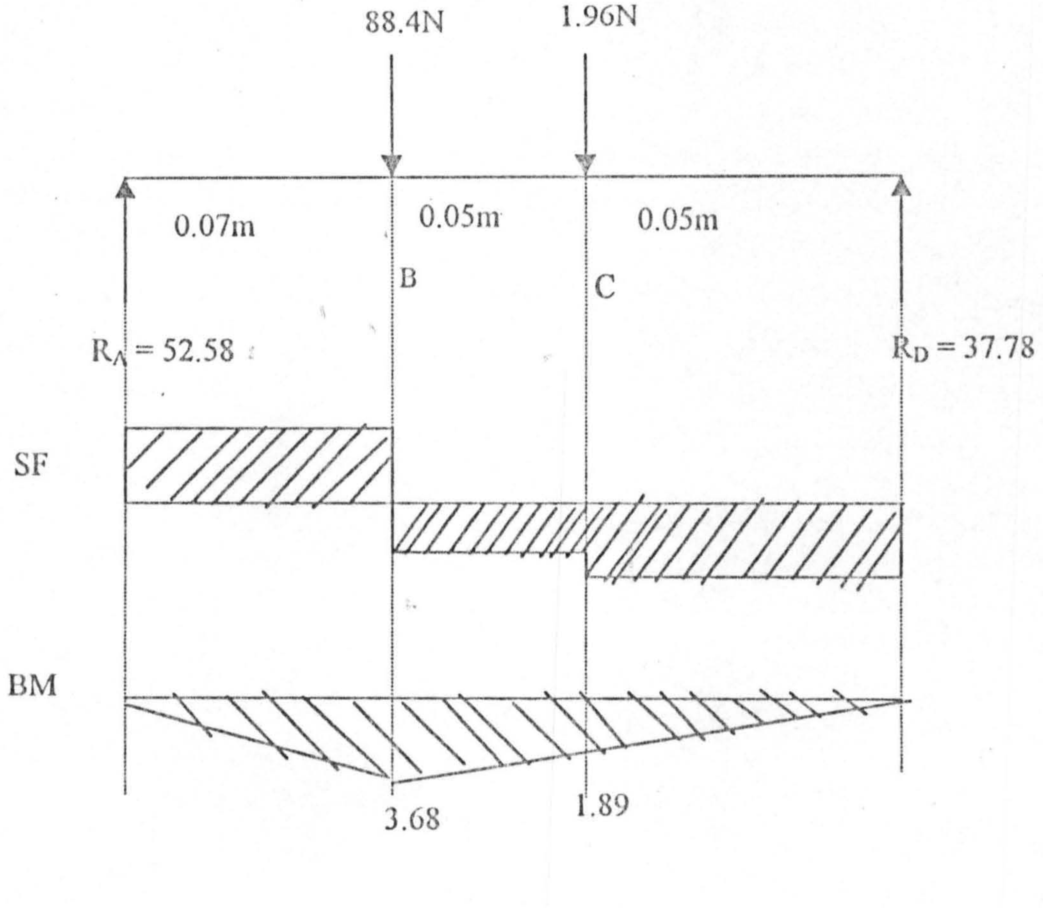
$52.58 \times 0.07 = 3.68$

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

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

$6.31 - 4.42 = \underline{1.89}$

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

Hence, M_{bmax} = **3.68 Nm**

g) Torsional moment, Mt = $(T_1\text{-}T_2)R$, Nm ---------------------------- (3.12)

= (72 – 21.25) 0.0675

= 50.75 x 0.0675

therefore M_t = **3.43 Nm**

(h) Determination of Shaft diameter

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

i.e. $d3 = \frac{16}{\pi S_s}$ □ $[K_b \times M_b]^2 + [K_t \times M_t]^2$ ------------------ (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

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

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

Therefore, $d = 3\square\ 9.7 \times 10^{-2} = 0{,}0099\text{m}$ or 10mm

Hence

Mass of shaft, $Ms = \rho\pi\ (r^2 l)$

Where,

ρ = density of steel = 7860 kg/m^3

r = radius of shaft = 0.005m

l = length of shaft = 0.22m

Therefore,

$Ms = 7860\ \{\pi\ [0.005^2) \times 0.22\}$

= 0.1358 kg or **0.14kg**

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 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 x 1.2 = 924watts

3.2 TORSIONAL RIGIDITY OF SHAFT

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

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

Where,

θ = angle of twist, deg

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

M_t = torsional moment = 4.65 Nm (calculated)

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

d = shaft diameter = 0.01m (calculated)

substituting values in equation (3.17) above

$$\frac{584 \times 4.65 \times 0.22}{80 \times 10^9 \times 0.01^4}$$

$$\frac{597.432}{8000} = 0.75^\circ$$

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 Rivetted 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.

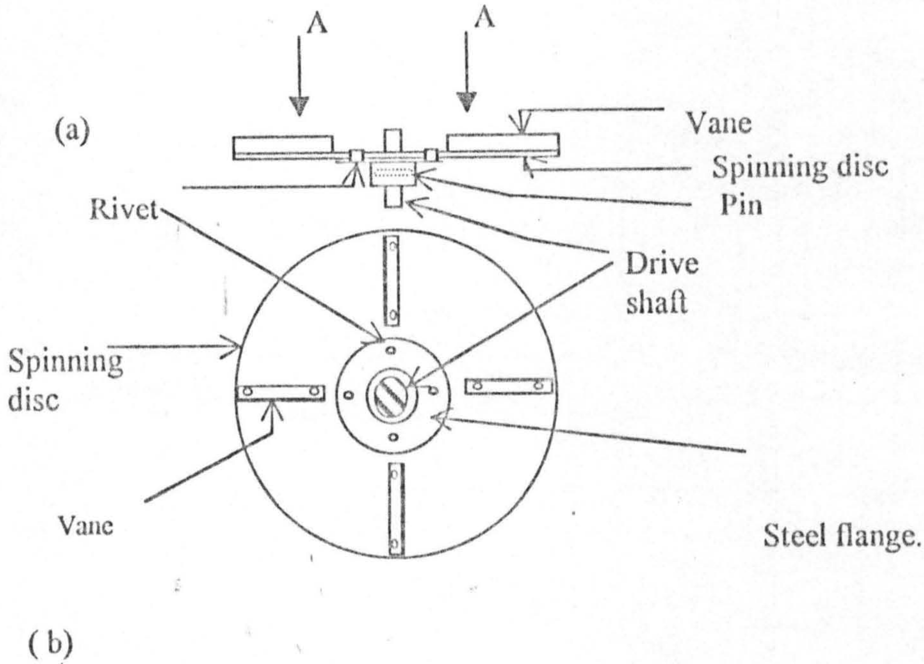


Fig (5): Spinning disc showing (a) Pin Joint and (b) Rivet Joint

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

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

Where

D = diameter of the rivet hole, mm

t = plate thickness, mm

therefore

$$D = 6.05\sqrt{2}$$

$$= 6.05 \times 1.4142$$

$$= 8.5559 \text{ or } \underline{8.6mm}$$

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

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

where,

Z = number of rivets

G = total load on the rivet, N

y_s = shear stress of rivet = 70 N/mm^2 (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.

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

Where,

G = rotating force of disc, N

M = mass of disc = 0.2kg (calculated)

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

r = radius of spinning disc = 0.125m

therefore G = 0.2 x 107.7^2 x 0.125

= 274N

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

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

The shaft diameter = 10mm; r = 5mm.

The lapping margin is given as m = 12d

Therefore, the riveting is done at 12d + 5

(where d = rivet hole diameter)

Thus 12d + 5 = (2 x 8.4) + 5 = 21.8mm

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

Therefore load to shear at the rivets

$= 7758.48 \times 4 = 31033.92N$

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

Where δc = bearing pressure = 185 N/mm^2 (Sharma and Aggarwal 1998)

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

Therefore, load to crush all the rivet = 3108 x 4 = <u>12432N</u>

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)

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

Substituting values

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

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

$$= \underline{2397.6N}$$

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

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

m = lapping margin = 16.8mm

Substituting:

$G = 28 \times 2 \times 16.8 \times 2$

$= \underline{1881.6\ N}$

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

where, G = pitch = 16.8mm

d = hole diameter = 8.4mm

Substituting values

$$\eta = \frac{16.8 - 8.4}{16.8} \times 100$$

$$= \frac{8.4}{16.8} \times 100 = 50\%$$

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 10mm

Pin diameter, d = 0.25 x 10 = 2.5mm

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

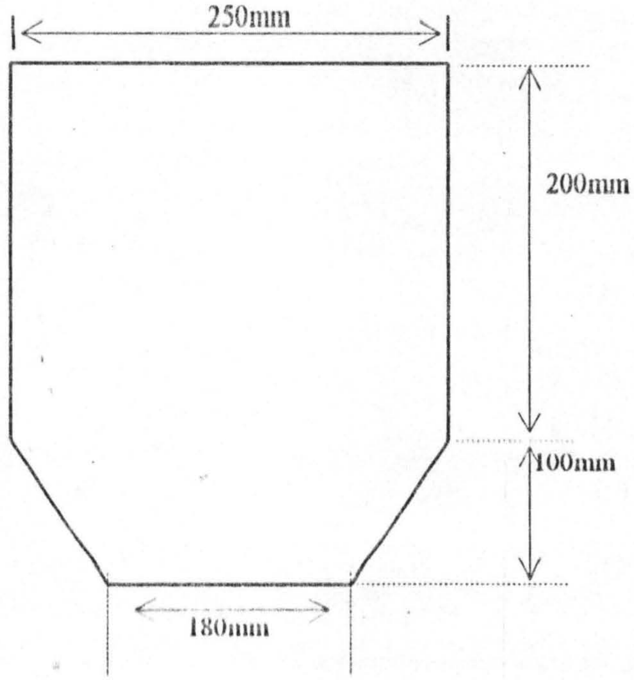


Fig (6) Hopper shape

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

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

Where,

r = radius of hopper,

h = height

Therefore $V = \pi (12.5)^2 \times 20$

$= \underline{\mathbf{9817.5cm^3}}$

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

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

Where

h = vertical height = 10cm

R = upper radius = 12.5cm

r = lower radius = 9cm

substituting

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

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

$$= 10.47 \times 349.75$$

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

Total volume of the hopper therefore = 9817.5 + 3661.88 = 13,479.38cm^3

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

given by,

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

Where, V = volume = 13,479.38cm3

ρ = density = 0.83g/cm^3 (average for rice, soyabean, phosphate, Urea fertilizer (extracted from appendix 2)

Therefore,

$$M = 0.83 \times 13479.38$$

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

11.2kg – 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

$$L_p = \underline{\gamma WR}\ [\,1-\varnothing^{-Kf}s^{h/R}\,] \quad \text{------------------------} \; 3.24$$

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^o while the second with wall inclination of 20^o. 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/m2 will provide satisfactory flow. Using the equation for finding minimum dimension which states that:

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

Where,

B = aperture min. diameter, mm

P = base pressure, kg/m^2

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

$\propto$ = Arching angle of slot opening (Monsoon, pp 601 – 602) with horizontal (assumed 30^0)

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;

$$B = \frac{49kg/m^2 \times \sin(2 \times 30^0)}{0.88g/cm^3}$$

where,

$$0.88g/cm^3 = 0.88 \times 10^{-3} \times 10^6 \; kg/m^3$$

$$= 880kg/m^3$$

Hence,

$$B = \frac{49}{880} \times \sin 60^0$$

$$= 0.055689 \times 0.8660$$

$$= 0.048m \text{ or } 4.8cm$$

B = Approx. 5cm

3.5.2 For Soyabean,

$$B = \frac{49kg/m^2 \times \sin 60^0}{0.75g/cm^3}$$

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

Hence,

$$B = \frac{49}{750} \times 0.8660$$

$$= 0.0653 \times 0.8660 = 0.0565m \text{ or } 5.65cm$$

B = Approx. 6cm.

3.5.3 For Phosphate rock,

$$B = \frac{49kg/m^2}{1.25g/cm^3} \times \text{Sin } 60^0$$

where,

$$= 1.25g/cm^3 \times 10^{-3} \times 10^6 \, kg/m^3$$

$$= 1250kg/m^3$$

Hence,

$$B = \frac{49}{1250} \times 0.8660$$

$$= 0.0392 \times 0.8660 = 0.0339m \text{ or } 3.39cm$$

B = Approx. 3cm

3.5.4 For Urea fertilizer,

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

where,

$$0.44g/cm^3 = 0.44 \times 10^{-3} \times 10^6 \, kg/m^3$$

$$= 440 \, kg/m^3$$

Hence,

$$B = \frac{49}{440} \times 0.8660$$

$$0.1114 \times 0.8660 = 0.0965m \text{ or } 9.65cm$$

B = Approx. 10cm.

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)

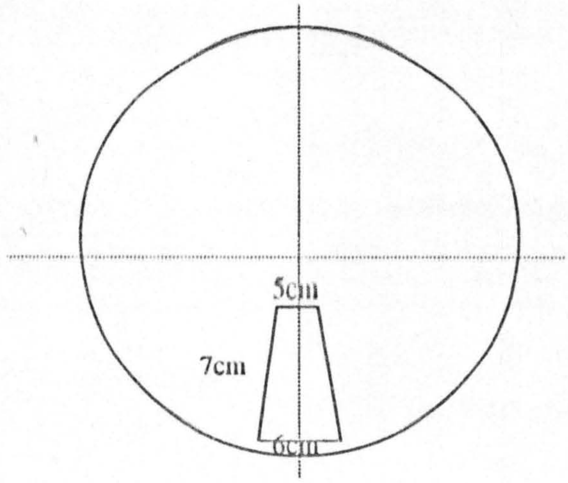


Fig (7): Aperture Dimension

3.6 APPLICATION RATE

3:6.1 THEORETICAL FLOW RATE

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

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

where

Q	=	flow rate, g/s
W	=	Bulk density, g/cm^2
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 alluminium 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 steed 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 alluminium sheet. The disc diameter is 25cm with four vanes riverted 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 steed covr which have roller bearings and alluminium 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 ascertained 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 ratewas 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) **<u>COST ESTIMATE FOR THE PRODUCTION OF THE BROADCASTER</u>**

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 Pully	Mild steel	Ø 0.03m	1	150.00	150.00
8.	Driven Pully	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	2000.0
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	-		-	-	2000.0
						7,797.

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 mean 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: Area = LxB

where, L = Length of field used in testing = 40m

B = width of spread of material, (m)

Therefore, for rice,

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

For soyabeans,

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

For Urea fertilizer,

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

For Benniseed ,

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

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

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

Hence, for rice,

T_t = 10,000 / 92 x 36

= 3913 sec or 65min

For soyabeans,

T_t = 10,000 / 112 x 34

= 3035.7 sec or 51min

For Urea fertilizer,

T_t = 10,000 / 104 x 35

= 3365.4 sec or 56min

For banniseed,

T_t = 10,000 / 100 x 35

= 3500 Sec or 58 Min

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

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

Hence, for rice,

Application rate = 10,000 / 92 x 1.1

= 108.69 x 1.1

= 119.56 or about 120kg/ha

For soyabeans,

Application rate = 10,000 /112 x 1.4

= 89.29 x 1.4

= 125 kg/ha

For Urea fertilizer

Application rate = 10,000 /104 x 1.15

= 96.15 x 1.15

= 110.57 or about 111kg/ha

For Bennissed

Application rate = 10,000 / 100 x 0.95

= 100 x 0.95 = 95kg/ha

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 soot 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.

REFERENCES

1. Aji, U.K. (1977), Investigation of the application of the spinning plate type broadcaster for Beniseed. Unpublished project thesis submitted to the Department of Agricultural Engineering, ABU Zaria for the award of (B.Eng) Degree.
2. Bosoi, E:S, Verniaev, O:V and Smirmov, I I (1988), Theory, construction and calculations of Agricultural machines. Russian translations series 66. Vol. I
3. Culpin, C (1980), Farm Machinery. The English language book Society and Crosby Lockwood Staples, London 9th Edition.
4. Garba, I H (1998), Construction and Testing of Hand operated fertilizer broadcaster, An OND Project presented to the Department of Agric. Engineering, College of Agric. Samaru – Zaria
5. Garba, I H (2002), Construction and Testing of ground wheeled driven fertilizer broadcaster. HND project submitted to the Department of Agric. Engineering, College of Agric. Samaru – Zaria
6. Hall A S, Holowenko, A.R, Laughlin HGs (1980), Theory and Problems of Machine design. Schaum's outline series. McGraw-Hill Book Company. SI (metric) Edition.
7. Hiraga, M. (1988), Physical Strains in carrying of Heavy packages by human labour. From the proceedings of the tenth congress of the Int. Ergonomics Association Titled "Ergonomics International - 88" Edited by Adams, A.S et al. Sidney – Australia.

8. Inns & Reece, (1962) Theory of centrifugal distribution II : Motion on the disc, off – centre feed. Journal of Agric Engineering Research, Vol. 7 (pp 345 – 353).

9. Kwaya, P.V. (2000), Design and Construction of soyabean grinder/Milk extractor. An Unpublished Post Graduate Diploma Project submitted to the Department of Agric. Engineering FUT-Minna for the award of PGD Agric. Engineering.

10. Krutz, G, Thompson, L and Claar, p (1984), Design of Agricultural Machinery, John Willey & Sons – New York

11. Mohsenin, N.N. (1970), Physical properties of plant and animal materials. Gordon and Breach Science Publishers Vol. I

12. Oberg E, Jones FD, Horton HL and Ryffel. H H (1992), Machinery Hand book. Industrial press Inc. New York, 24th Edition.

13. Patterson & Reece, (1962), Theory of centrifugal distribution I : Motion on the disc,

near –centre feed. Journal Agric Engineering Research. Vol. 7 (pp 232 – 240).

14. Ryder, G.H. (1969) Strength of Materials. Macmillan. 3rd Edition (SI Units).

15. Smith, H.P. (1965), Farm Machinery and Equipment McGraw-Hill Book Company, Fifth Edition (pp 167 – 168)

16. Sharpe, C. (1989), Kemps Engineers year book. Morgan-Grampian book publishing Co. Ltd, London. Vol.I, 94th Edition, pp A6/2 – A6/5

17. Sharma, P.C. and Aggarwal, D.K. (1998), A textbook of machine design. S.K. Kataria & Sons Publishers and book sellers, Naisarak, Delhi. (8^{th} Revised and enlarged Edition).

Appendix 1

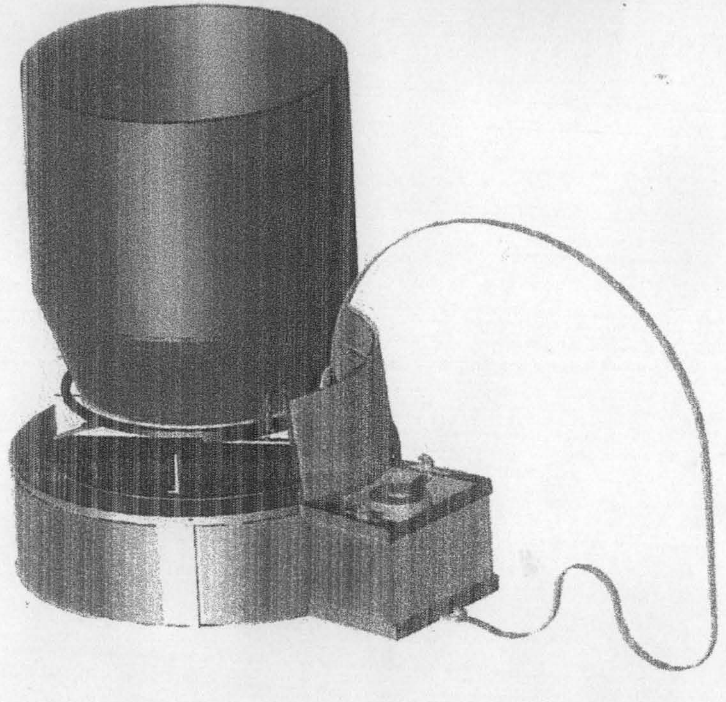
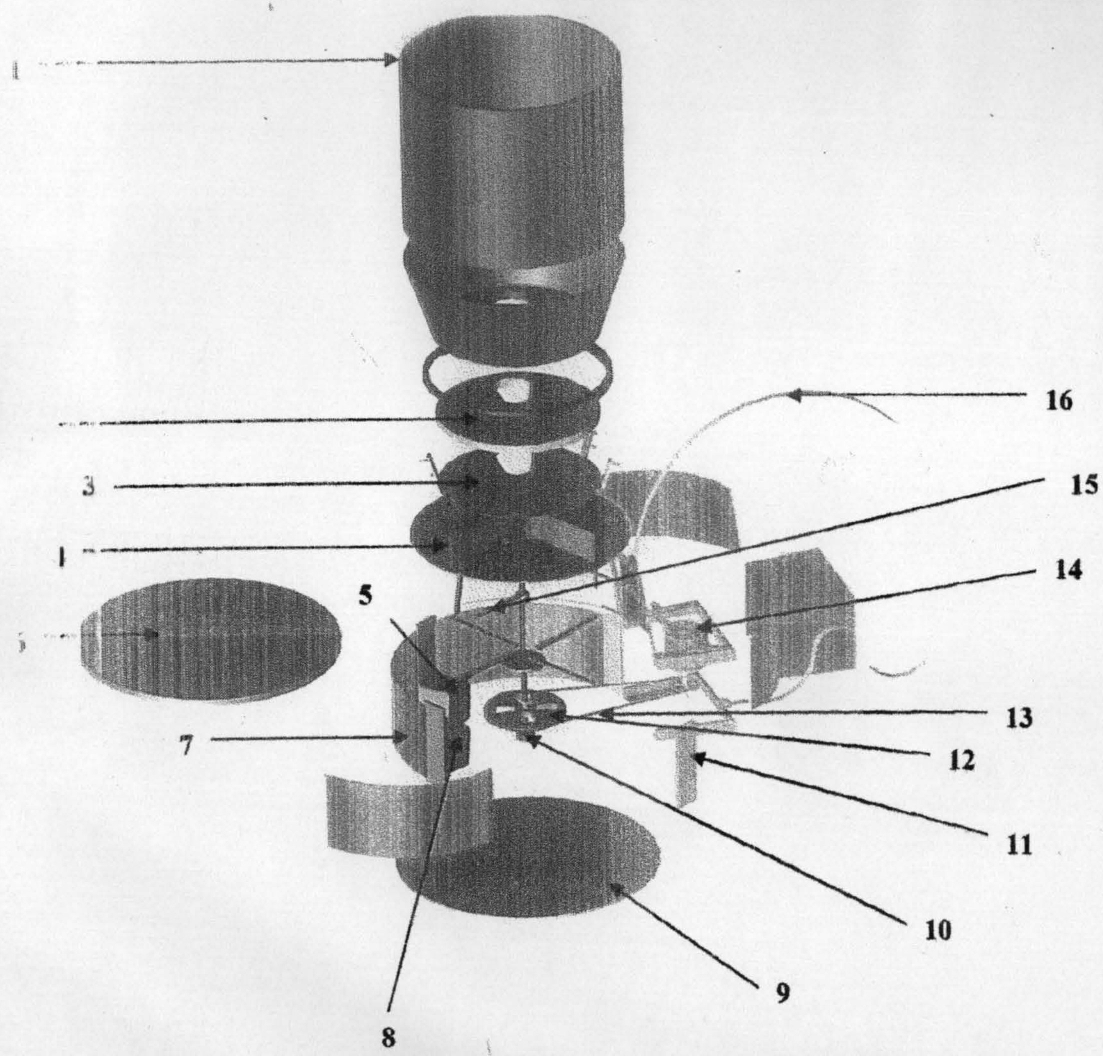


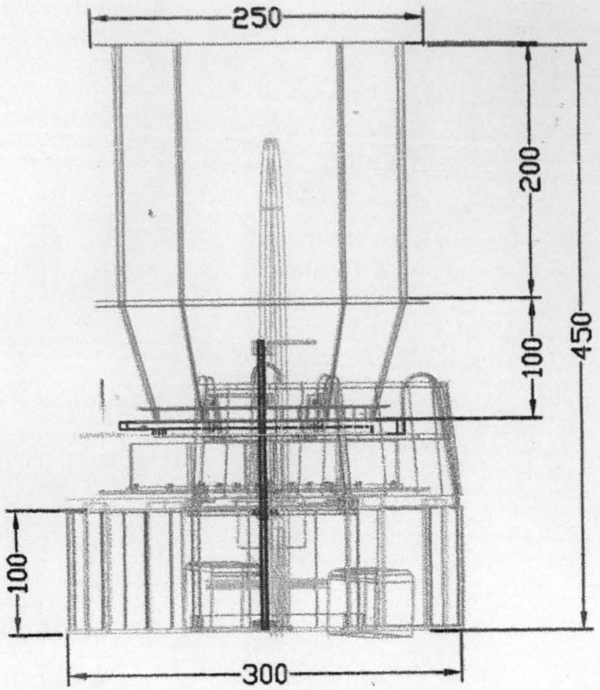
FIG (8) ISOMETRIC VIEW OF PORTABLE MOTORIZED SEED AND FERTILIZER BROADCASTER
ALL DIMENSIONS IN (MM)
SCALE 1: 6

Appendix 2

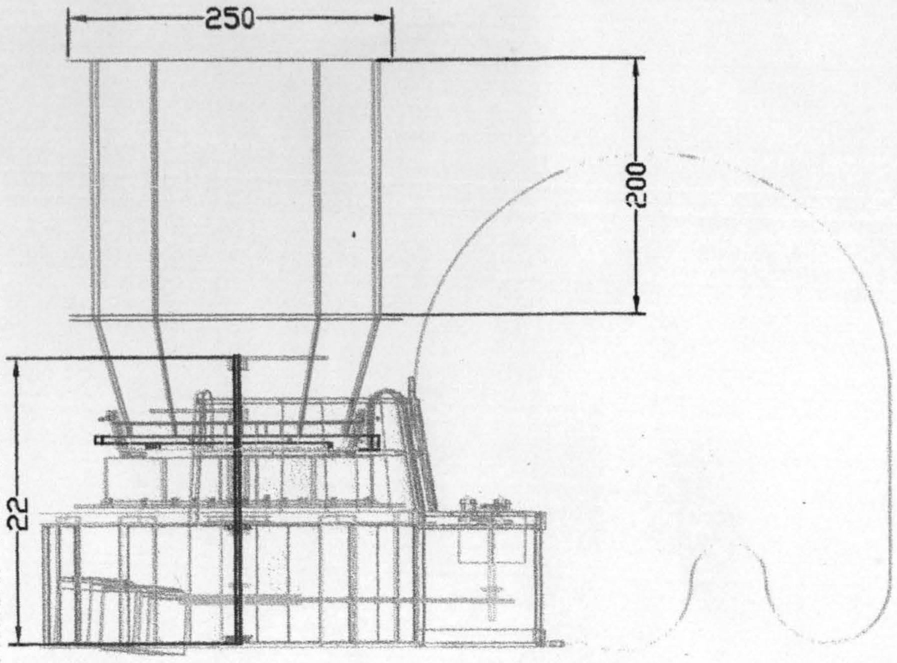


S/NO	LEGEND
1	Hopper
2	Stationary Aperture Plate
3	Aperture Adjusting Plate
4	Spinning Disc
5	Battery (12V)
6	Upper Fixed Plate
7	Cover Sheet
8	Regulator
9	lower fixed plate
10	Drive Shaft
11	Frame Support
12	Pulley (Θ 135 mm)
13	Driving Belt
14	D.C Motor
15	Agitator
16	Shoulder Stripe

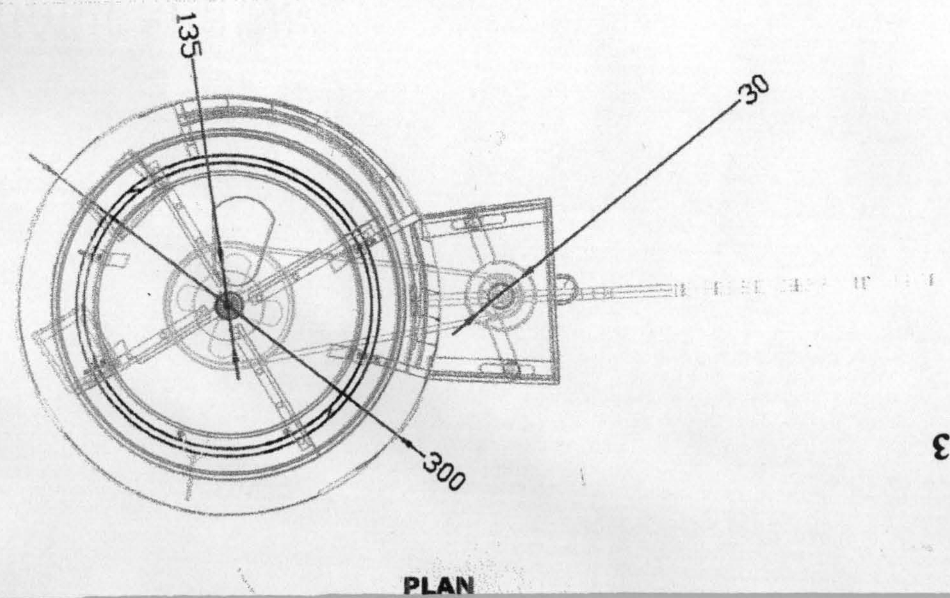
Fig. 1. Exploded View of Potable Motorized Seed and Fertilizer Broadcaster Scale 1: 8
All Dimensions in (mm)



FRONT VIEW



SIDE VIEW



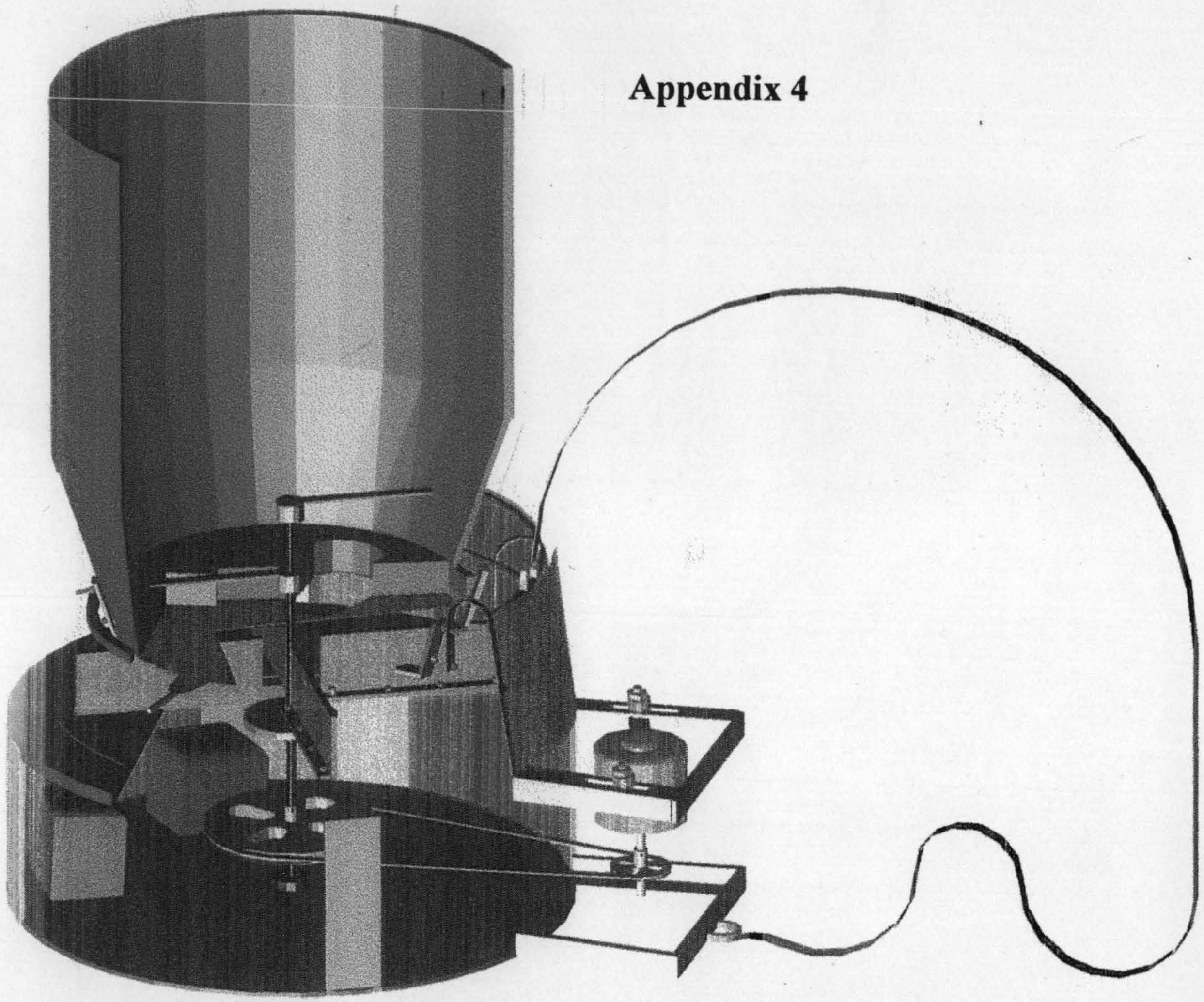
PLAN

FIG (10)ORTHOGRAPHIC PROJECTION OF MOTORIZE SEED AND FERTERLIZER BROADCASTER ALL DIMENSIONS IN (MM)

SCALE 1:6

Appendix 3

Appendix 4



(11)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 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 Aggarnal (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	11.70	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	73	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	23.7	5.1		0.12	31.5	0	133	135	10.33
Sugar	25	0.9	0.83			0	263	313	10.34
Vetch seed	16	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