# PERFORMANCE TESTING AND MODIFICATION

# OF A MULTIPURPOSE WINNOWER

BY

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MINNA, NIGER STATE - NIGERIA

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PROJECT REPORT

SUBMITTED IN PARTIAL FULFILMENT OF THE

REQUIREMENTS FOR THE AWARD OF BACHELOR OF

ENGINEERING

DEGREE (B.ENG) IN THE DEPARTMENT OF AGRICULTURAL

ENGINEERING OF THE FEDERAL UNIVERSITY OF TECHNOLOGY,

MINNA, NIGERIA

MARCH. '98

**APPROVAL SHEET**

This project report has been read and approved as having met the standard required for the award of Bachelor of Engineering degree in the Agricultural Engineering Department of the Federal University of Technology Minna.

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## DEDICATION

This project is dedicated to God for his guidance and protection throughout the duration of the course in spite of all odds.

Also to my parents Mr Roland Okorhi, my Mum Mrs Maria Okorhi and brothers and sister.

## ACKNOWLEDGEMENT

With deep gratitude, I acknowledge the fatherly advice ever coming from Mr and Mrs Ogwuagwu O.V of the department of mechanical Engineering who collectively have taken me as their own child and also for offering me great assistance in my project work and my parents Mr and Mrs R.E Okorhi for their love and care throughout my academic pursuit.

Acknowledgement is here by made to all my brothers and Sisters Juliet, Tony, Henry, Samuel, Voke and others. All lecturers of Agricultural Engineering Department who taught me during my B.Eng. Degree programme and to my Supervisor, Mrs. Z.D. Osunde who took pains to read through this write up and offering useful criticisms and corrections.

Also not forgetting my very good friends like Mr Agbebaku Lucky, Dr. patrick Omu, Mr Sam Omu, Mr Kelly Orilolo, Mr Ikue Joe of Nigerian Part Authority, Warri, Mr Avwaruroro Henry Mr Charles Oparike and also not forgetting Mr Sam Igbudu, Glory Obanovwe, Gordon Atugbe, Mr Doni, Mr Banseka H.S.T, Mr Stephen Tolufase, Mr John J. Musa, Seyi Ajala, Aluta Emmanuel, Mr Durosinmi Wasiu, Mr Efe Egudu, Mr Felix O.E, Mr Abundance Yono and a host of others whose names are not mentioned but contributed to this project in one way or the other.

To you all I say, a big thank you and God bless.

## ABSTRACT

In this project report, the design and modification of a multipurpose winnower is presented. For the design, the physical characteristics (groundnut, milled rice, millet and soyabean) such as particle density, angle of repose, angle of inter-granular friction, sphericity, geometric mean diameter of particle and aerodynamic characteristics were taken into consideration in this work.

The method of throttling the air inlet into the fan is adopted for this modification work.

The machine was evaluated for its performance in terms of cleaning efficiency and percentage grain loss. The result of the test indicated that the cleaning efficiencies of the modified multipurpose winnower are 96% for rice, 80% for millet, and 90% for groundnut at a feed rate of 72.97kg/hr, 38.57kg/hr and 50kg/hr respectively.

The capacity of the machine is 29.19kg/hr for rice, 21.43kg/hr for millet, and 32kg/hr for groundnut at the indicated feed rates above. The power required for operating the winnower at the speed of 1084 rpm is 1007.48 watts (1.35 hp).

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# LIST OF SYMBOLS

| | |
|---|---|
| $A$ | area |
| $A_p$ | Projected area |
| $a_1, a_2$ | Upper and lower dimensions of hopper |
| $a^1$ | hydraulic radius of cross - section |
| $B$ | Diameter of hopper discharge outlet |
| $b$ | Width of key and keyway |
| $b_3, b_4$ | Impeller blade width |
| $c$ | Overall dray coefficient |
| $c_f$ | frictional dray coefficient |
| $c_D$ | drag coefficient |
| $c_k$ | centre distance |
| $c_o$ | Basic load rating (static) |
| $c_r$ | Dynamic load rating |
| $D, d$ | diameters |
| $d_e$ | diameters of equivalent sphere |
| $d_h$ | hub diameter |
| $d_{hl}$ | hub length |
| $d_{sh}$ | shaft diameter |
| $f$ | friction factor |
| $F_L$ | lift force |
| $G$ | modulus of rigidity |
| $g$ | acceleration due to gravity |
| $H$ | power |
| $h$ | pressure head |
| $h_f$ | head loss through cleaning chamber |
| $h_L$ | head loss through nozzle head |
| $H$ | Height of hopper |

$h_s$ height of stored materials in hopper

$i$ transmission ratio

$J_w$ polar sectional modulus

$K_b$ combined shock and fatigue factor applied to bending moment

$K_t$ combined shock and fatigue factor applied to torsional

$L$ length of belt

$L_k$ length of keyway .

$L_{it}$ Rated life of ball beaning

$m_b$ Bending moment

$m_f$ mass of fluid

$m_p$ mass of particle

$N,n$ speed in rpm

$N_R$ Reynolds number

$p_a$ power transmitted by belt

$p_e$ Equivalent load

$p_s$ fan static pressure

$p_t$ fan total pressure

$p_v$ fan velocity pressure

$Q$ flow rate

$R$ radius of pulley (large)

$R_1,R_2$ reaction forces

$r_p$ radius of pulley (small) radius of hopper discharge outlet

Sut ultimate tensile strength of material

$S_s$ allowable shear stress of material

$S_y$ Yield strength of material

$t$ thickness of the hoppor.

$T_1$, $T_2$ belt tensions

$V$ fluid velocity

| | |
|---|---|
| $V_o$ | impelled inlet velocity |
| $V_p$ | particle velocity |
| $V_s$ | setting velocity |
| $V_t$ | terminal velocity |
| $W$ | weight |
| $W_p$ | weight of particle |
| $W_s$ | weld leg size |
| $Z$ | sectional modulus of weld |
| $Z_n$ | number of impeller blades |
| $Z_w$ | Sectional modulus of weld |
| $\theta$ | angle of twist in degree |
| p | PI, 3.142 or 22/7 |
| $\mu$ | coefficient of friction |
| $\mu$ | viscosity |
| $\rho_f$ | fluid mass density |
| $\rho_p$ | particle mass density |
| $\omega$ | angular velocity |
| $\phi$ | volume coefficient |
| $\beta$ | blade angle |
| $\eta$ | efficiency |

<u>NB</u> - Symbols may be defined in the text for a specific purpose.

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# CHAPTER ONE

## 1.0 INTRODUCTION

The purpose of seed cleaning is to remove undesirable materials such as inert matter, weed seeds, other crops seed broken seeds, shrivelled seeds e.t.c.

Cleaning can be used to achieve these goals, cleaning can be done basically by two means; the wet and dry method the wet method uses liquid (water) for removing unwanted materials from agricultural produces while dry method uses pneumatic and other physical means for separating the seeds from their chaff and contaminants.

Cleaning using air current (winnowing) can be logically traced back to pre-historic times when the early man realised the need to store food.

Winnowing according to the oxford advanced learner's dictionary of English is defined as using a stream of air to separate dry outer coverings from grain. i.e blow husk, chaff away from grains/legumes.

Cereals and legumes e.g maize, millet, rice, soyabean, groundnut, cowpea etc thrives well in the tropics.

In Nigeria they are grown in virtually all the Northern states Soya bean is only rarely grown in the Southern part of the country compared to maize, groundnut and rice.

Although government programmes recently through the Agricultural Development project have lunched campaigns on the need to grow Soyabean.

Cereals and legumes are seasonal crops and as such handling aid processing of these crops require great care so as to guide against scarcity which may arise from drought and poor production.Winnowing is a unit operation which is used for removing chaff, light soil particles, weeds, broken grains e.t.c. from whole grains based on their aerodynamic characteristics.

Some of the objectives of grain cleaning include:

(a) Improve product quality i.e market value.

(b) Increase their storability i.e reduce susceptibility to spoilage and insect infestation

(c) Cleaning and effective dust control procedure, significantly reduce explosion risk in elevators.

(d) Improve eating quality of grains/agricultural produces.

## 1.1 TRADITIONAL WINNOWING TECHNIQUES.

Traditionally winnowing is usually carried out by women. The simplest method is to place some newly threshed grains on a flat or curved split Bamboo tray and elevating it above the head to loose grain up into the air. The grain and pieces of straw and chaff will be blown to one side by wind while the heavier whole grain fall as pile at their feet. This process may need to be repeated several times in order to have a satisfactory clean grain farmers encounter a lot of problems during the process of seed/grain cleaning which can be summarised into the following;

(a) The process is very tedious i.e a lot of human energy is expended.

(b) The process is time consuming.

(c) The process is weather dependent

(d) The process is not very efficient as stones and other contaminants gets into the grains/seed during the process.

Following the problems associated with the traditional method of grain cleaning more improved methods have been developed to assist our local farmers.

## 1.2 IMPROVED GRAIN CLEANING TECHNIQUES

Commercially various winnower/cleaners/separators are available, these include air-screen cleaners, indent disk and Cylinder Separators, winnower, Specific gravity Separator e.t.c. These machine are used in many parts of the world to clean seeds, grains, and oil seeds on farms, food industries and government seed processing unit. Most of these machines, however, also have their own limitations.

## 1.3 OBJECTIVES

The objectives of this project are listed below:

(i) To test the existing machine in the department note its short coming and based on which modify the machine to correct its defect

(ii) Increase the number of grain for which the original machine was designed for.

(iii) Incorporate the advantages of

(a) reduction in human labour wastage and power requirement.

(b) reduction in time wastage

(c) low cost

(d) portability

(e) durability

(f) ease of manoeuvrability

( lV) increase cleaning efficiency

(v) Trouble free maintenance.

Such a winnower would find its way into many Nigerian farms/homes.

## 1.4 JUSTIFICATION

Various commercial winnower/cleaners are available in the market today, they vary from simple machine designs which uses small horsepower elective motors to very complex mechanical designs which have high feed rate, throughout aid fitted with auxiliary components such as conveyors/loaders, reciprocating serves e.t.c. Considering the Nigeria situation where most of our farmers are poor and line in the rural areas, if a machine which can to a large extent reduce the drudgery involved by the local or traditional method of winnowing portability low cost, it will go along way to

(a) Increase the income of the farmers

(b) reduce time spent on winnowing operation

(c) and also reduce fatigue during cleaning operation

# CHAPTER TWO

## 2.0 LITERATURE REVIEW

Traditional winnowing techniques are initially employed for grain cleaning before the advert of cleaning machines.

Hand/manual winnowing machines were later developed as an improvement on the traditional winnowing method. Draught is created in this machine by a centrifugal fan as the grain mixture fall twin a hopper through the air, the light material are blown away while the heavier ones fall into a container.

As a result of improvement in technology, as applied to cleaning of grains, the following are some machines employed for cleaning grains:

## 2.1 REVIEW OF SOME MECHANICAL GRAIN CLEANERS

Air screen cleaners: are used for cleaning and sorting grains it does this based on three basic physical characteristics viz: size, shape and density. Various make and model of air screen cleaners are available, they ranges twin small one fan, single screen machine to large multi-fan, six or eight screen cleaners.

Indent disk and cylinder separator: employs length as a basis for separation the cylinder separator consist of a revolving drum with indentation or pockets which are carried to the top of the cylinder and dropped into a screen conveyor for removal.

Rotary screen cleaners:- The most common type of farm and small elevator cleaner is the rotary screen cleaner, which uses one or more screens attached to an include revolving drums. Grain tumbles inside the screens by the rotating action for greater efficiency in separating broken fines and trash from the grain.

Stationary or Vibrating Gravity Flow Screen Cleaner:- are included wire mesh screen or perforated metal bottom through that sift out part of the fines and small particles. Multiple level screen units with large mesh oven smaller mesh or perforated metal sheet cause trash and large foreign objects to "slide" across the top screen while the grain drops through the top screen on to the lower panel where smaller openings sift out dust and small grain traction. Handling and cleaning capacity to determined by width, length, slope, perforation on wire mesh size, and vibration frequency.

Aspirating Vibratory or Shaker Cleaners:- These commercial grade or industrial quality cleaners have adjustable lower type screen with opening adjustment, inclination adjustment, oscillation speed and air volume control, the aspiration system removes light chaffy particles, light trash and some smaller grain fractions and dust.Vibrating screens separates small dense particles. Capacity of this machine is based on width and length of screen units aspiration airflow rates, screen openings, oscillation rates and inclination.

Aerodynamic Gravity Cleaner: These cleaner, typically used in flour mills or food industries, are designed for dockage fraction removal using aerodynamic grain shape and density characteristics for precise separation at design grain flow rates. These units are expensive, but are highly reliable and have long wear life.

Spiral separator:- This machine separates seeds an the basics of shape, density and ability to roll.

Electrostatic/magnetic separators: They separate see according to texture of grains, since rough and smooth.

Seed react differently to electric and magnetic fields.

Velvet roll seed separator: This machine separates seeds on the basics of surface texture (roughness) and shape.

## 2.2 REVIEW OF GRAIN SEPARATION USING AIR

In the development of winnowing/cleaning system using pneumatic power; various studies have been carried out on the aerodynamic characteristics of grains, Bilanski and Lal (1964) have reported on the aerodynamic behaviour of threshed materials in a vertical wind tunnel. The due found out the aerodynamic drag coefficient, the resistance coefficient and also reported on the terminal velocity of wheat straw pieces without a node. The duo showed that the terminal velocity of the straw without a node followed a pattern similar to terminal velocity of circular cylinders reported by Rice (1960).

The data on terminal velocity of wheat straw in this work does not agree with similar data reported by Uhl and lamp (1966).
In fact the plotted data on straw length versus terminal velocity gave trend opposite to those by Bilanski and Lal.

Bilanski and Lal (1964) have pointed out this lack of agreement and claimed that it was due to the fact that uhl and lamp did not consider the node position in the straw tested. Also the drag coefficient for various grains reported by Bilanski and Lal are lower than those reported by Uhl and lamp for similar grain the primary reason for this could be the different in the assumption made for the frontal area used in the calculation of the drag coefficient.

Bilanski and Lal, as previously indicated, based their calculation of the frontal area on the two largest dimensions while Uhl and lamp considered the least cross-sectional area as the frontal area. The values of terminal velocity of the various grains reported by one group, however lie within the range of terminal velocity reported for that grain by the other group.

Uhl and lamp (1966) also found the percent separation of grain, straw and chaff at various air velocity. Some of the physical characteristics of various grain and the range of air velocity required to air borne these grains.

Bilanski and Manzies (1967) conducted aerodynamic test on alfalfa particles and reported terminal velocities in terms of particle weight so that the effect of moisture intent can be taking into consideration, Having terminal velocity from experimental data, result of this work according to Bilanski aid manzies stated that the behaviours of alfalfa particles is an air stream depends only on length, nodal position and length to diameter ratio.

Tiwari (1962) employed the aerodynamic principle to investigate possibilities of pneumatic separation of good beans from samples of threshed dry edible bean crop containing undesirable materials such as damaged beans, stones, leaves, stems, roots, etc. He found out that the terminal velocity of four variety considered range from 40 to 65ft/see (12.192 to 19.812m/s) and that within this range about 80% of damaged beans could be separated without significant loss of whole beans light materials such as leaves, stems and roots were completely separated at terminal velocity of about 21ft/s (6.4m/s).

Keck and Goss (1965) and Garrette and Brooker (1965) also determined time displacement curve of grain and seeds in free fall. They noted that using the diameter of equivalent spheres gives a lower terminal velocity which indicates that the geometric mean diameter may vary considerably for irregular objects such as seeds, for this reason, keck and Goss suggested the determination of a shape factor.

The technique used by Garrette and Brooker (1965) was unique in that the actual time displacement curves of grain falling in still air was recorded photographically velocity time curves were plotted and consequently aerodynamic drag of corn, oats and what falling in still air were calculated taking the buoyant effect of the air into consideration.

The possibility of using cyclone, for separating grain from chaff in combine harvester was investigated by

Hassebrauck (1962). In his work he pointed out that the actual range of velocity ratio in a equilibrium of the forces. In other words, the centrifugal forces considerably exceeds the force of air resistant so that only separation of the grain and chaff particles from the air can be expected and not separation of one solid particles from another.

Based on the aerodynamic characteristics studies carried out, various winnower/cleaners have been developed.

Klein (1968) described grain cleaning by winnowing in United states. He said that grains are discharged from an elevated discharge put into a strong air flow, either wind or from a large fan, to flow light chaff, leaves, and low density or shrivelled kernels out of and beyond the grain stream. He explained that cleaning can only be achieved if the discharged grain falls in this strong and stream. He noted that this method of cleaning is slow.

Nurul Islam et al (1980) designed and constructed a manually operated seed cleaning and grading machine using wooden frame.

Kebede Desta and T.N Mishra (1990) in the development and performance evaluation of a sorghum thresher reported a mean cleaning efficiency of about 92.7% at a feed rate of 6kg/min (360kg/hr).

Aloba G.A (1995); In the department of Agricultural Engineering designed and fabricated a multipurpose winnower, based on which this project (Performance Testing and modification of multipurpose winnower) is to be carried out.

## CHAPTER THREE

### 3.0 TESTING OF THE MULTIPURPOSE WINNOWER IN THE DEPARTMENT.

A proper test could not be carried out on the winnower to assertion the cleaning efficiency as the pulleys ($\phi$150,$\phi$250 and $\phi$180mm) meant for the test were not available in the department.

Rather a test to see how the machine would function using a $\phi$160mm pulley available in the department was performed. The test was carried out using milled rice and chaff mixture. The mixture was placed in the hopper and allowed to fall into the air stream generated by the fan which was connected to a 3.7kw electric motor via a v - belt.

The following were the observations made during the testing.

(i) During operations the machine showed tendency to lift off the ground.

(ii) The grain and chaff falling from the hopper do not fall straight into the air stream, rather they are scattered.

Also the modifications presented in this report include some recommendation of Mr. Aloba G.A and other general observations which will tend to increase the overall performance of the machine. Based on the observation high lighted above, the cleaning efficiency of the machine could not be estimated. Hence this has led to the modifications presented on this project report.

The modifications include

(i) Design of the feed hopper

(ii) Design of a cleaning chamber for this machine

(iii) Design of a fan that will not require varying of its pulley for the various grains/legumes to be handled.

(iv) Stability of the machine in operation is taking into consideration.

(v) Design of a drive guard plate for the machine.

## 3.1 DESIGN FEATURES

The main features of the winnower include

(i) feed hopper

(ii) flow control slide plate

(iii)power transmission (V-belt drive)

(iv) Centrifugal fan

(v) Collection tray (optional)

(vi) drive guard plate

## 3.2 DESCRIPTION OF MACHINE/MODE OF OPERATION

The winnower was developed specifically to meet the need for efficient cleaning of grains and legumes such as Soya bean, millet Groundnut and Rice. Cleaning of grains/legumes is an important unit operation aimed at increase their shelf live.

The winnower consist of the power transmission system, a centrifugal fan, a feed hopper fitted with a flow control plate, and an optional collection tray. The power transmission system consist of a V-belt and pulley drive, the pulley is keyed to the fan shaft and supported on bearings.

See DRG NO EPL-001-98 FP.

The machine has a total length of 825mm, a width of 500mm and a height of 840mm. The fan shaft is 215mm from the basement of the winnower. See DRG.No EPL - 002 - 98 FP.

Power is supplied to the machine from an electric motor or a fossil fuel engine by means of v-belt drive set the fan impeller in motion. The fan delivers 0.064m$^3$/s or 0.048m$^3$/s of air (depending on the grain which is to be cleaned) on the falling grain from the feed hopper, hence cleaning action is initiated. The quantity of grain falling from the hopper may be regulated with the flow control plate fitted on it.

## 3.3 THEORETICAL ANALYSIS

### 3.3.1 DESIGN CONSIDERATION

#### (A) DESIGN ANALYSIS

Since the machine is to be motorised (either electric motor powered or fossil fuel powered) One of the greatest problem faced is that of determining the design power requirement for driving the fan and the power transmission system.

The machine is designed to deliver $0.064m^3/s$ of air at 4.0m/s and $0.048m^3/s$ at 3.0m/m on the falling grains/legumes depending on which is to be cleaned at a speed of 544.44 evolutions per minute.

#### (B) COMPACTNESS

In carrying out this design work, much effort was directed towards obtaining a system that would give the desired compactness. the dimensions of the various components were chosen so as to minimize size, weight and cost of the machine and at the same time not compromising the efficient operation of this components.

#### (C) SERVICEABILITY

The various materials used were chosen to optimise cost while at the same time ensuring serviceability and an acceptable strength cost ratio.

The fan shaft, pulley materials were chosen based on service conditions and cost of low section steel. The structural members (L-sectioned steel bar) where chosen to ensure a reasonable strength to cost ratio.

#### (D) MAINTAINABILITY

Maintainability factor were not left out in the design work the system is designed to ensure that low level management of maintenance does not disrupt operation.

The machine has been designed to facilitates the detection and rectification of component failure during maintenance or repair.

(E) HUMAN PHYSIOLOGICAL CONSIDERATION

The human physiological consideration were not left out in this design work. since the machine is to be motorised, an input speed at the fan shaft of 544.44/pm is desired for this design. Also the height of the machine has been made in such away that the Operator may not bend down too frequently so as to reduces fatigue.

It is recommended that the various design speeds should be maintained during operation to ensure optimum capacity utilization and minimum energy wastage.

(F) COST OF PRODUCTION

The various components of the machine were chosen such that the cost of production will be relatively cheap.

The choice of L - sectioned steel bar for the machine framing is to ensure proper stability, weight reduction and at the same time reducing cost of production. Also the choice of mild steel sheet for the fan blades and casing mild steel rod for shaft material were based on the production ease, cost as well as service conditions.

### 3.3.2 AERODYNAMIC CHARACTERISTICS CONSIDERATION

A particle in air current can be considered as a sphere of known diameter and density falling through a fluid of known density and viscosity .

The forces acting on the falling object are those of Drag, Buoyancy and Gravitational effect due to the weight of the particle. When fluid flow occur about immersed particle, the particle according to stokes (1932) is acted upon by a resultant force Free which may be resolved into components.

$F_D$, the drag and $F_L$, the lift forces.see figure 3.2

Figure 3.2 Forces acting on a particle in air current

The equations for calculating drag and lift forces have been derived by stokes (1932) using dimensional analysis and assuming a smooth object having projected area A, moving through a fluid of mass density ;$\rho_f$ ; viscosity;ʃ; and modulus of elasticity;E, with a velocity, V. The components $F_D$ and $F_L$ can be expressed as a function shown below.

$F_D = f_1 (A_p, \rho_f, ʃ, E, V)$

$F_L = f_2 (A_p, \rho_f, ʃ, E, V)$

Employing the method of dimensional analysis, the following equations have been established([8]) for drag and lift forces.

$$f_d = \frac{C_D A_P P_F v^2}{2} \qquad (3.1)$$

$$F_L = \frac{C_L A_p \rho_f V^2}{2} \qquad (3.2)$$

Where $C_D$ and $C_L$ are the (dimensionless) drag coefficient and lift coefficient of the object respectively.

In most practical cases where objects are free to assume its own random orientation, the net resistance force Fr can be gives in terms of an overall drag coefficient; c as follows

$$F_r = \frac{1}{2} C_{Ap} \rho_f V^2 \qquad (3.3)$$

However, in certain cases it is desirable to resolve the resultant force into the components of frictional drag due to tangential forces an the body surface and profile drag due to pressure distribution around the body. In laminar or low velocity flow where variation in fluid density is small viscous action governs the flow, the profile or pressure drag is negligible.

In turbulent or high velocity flow, where fluid compression and not viscous action governs the flow, the frictional drag is negligible.

The following have been given for frictional drag coefficient $C_f$ (Vernard, 1961, Prandtl, 1932).For flat plate with a laminar boundary layer.

$$C_f = \frac{1.328}{(N_R)^{0.5}} \qquad (3.4)$$

For flat plate with turbulent boundary layer

$$C_f = \frac{0.455}{(\log N_R)^{2.58}} \qquad (3.5)$$

In these equations $N_R$ is the Reynold number defined by

$$N_R = \frac{V d \rho_f}{\eta} \qquad (3.6)$$

Where d is the effective dimension of the object such as length of plate or diameter of a sphere and $\sigma$ is the absolute viscosity of the fluid.

The equation for the transition region, where laminar flow changes to turbulent flows is given by prandtl (1932) as follows

$$C_f = \frac{0.455}{(\log N_R)^{2.58}} - \frac{1700}{N_R} \qquad (3.7)$$

Having the coefficient $\rho_f$, the drag force can be calculated using equation 3.1 above.

When a blunt object, such as a sphere is placed in a fluid flow, the frictional drag usually but not always, can be neglected because of the small surface area on which frictional effect can act. The exception is the case of flow at very low Reynold numbers where stokes law is applicable. Stokes has proven that a Reynold number less than unity, where the inertia forces may be neglected and those of viscosity alone considered, the flow close behind a sphere like object and the profile drag is composed primarily of friction drag. For a sphere of diameter $d_p$, moving at a velocity V through a fluid of viscosity, $\mu$ stokes

$$F_D = 3\pi\eta V d_p \qquad (3.8)$$

Equating the above to 3.1 and taking $A_p$ to be the frontal area and equal to, the profile drag coefficient is found to be

$$C_D = \frac{24}{N_R} \qquad (3.9)$$

As Reynold number exceeds unity, the stokes law is no longer applicable because flows opens up behind the blunt object and the drag coefficient is a combination of frictional drag as well as pressure drag in a range up to $N_R = 10^3$.

Above this figure the frictional effect may be neglected.

In free fall, the object will attain a constant terminal velocity, $V_t$ at which the net gravitational acceleration force, $F_g$, equal the resisting upward drag force, $F_r$. Under the steady state condition, where terminal velocity has been achieved, if the particle density is greater than the fluid density, the particle motion will be downward.

If the particle density is less than the fluid density, the particle will rise.

For practical application such as pneumatic conveying and separation from foreign materials, an expression for terminal velocity is given

$$F_g = F_r \qquad when \qquad V = V_t$$

Substituting for $F_g$ and $F_r$, the expression for terminal velocity will be

$$m_p g[(\frac{\rho_p - \rho_f}{\rho_p})] = \frac{1}{2} C A_p \rho_f V_t^2 \qquad (3.10)$$

Note that $F_g$ is corrected for buoyancy effect. From equation (2.10) we obtain

$$V_t = [2W \frac{(\rho_p - \rho_f)}{\rho_p \rho_f A_p C}]^{\frac{1}{2}}$$

and

$$C = \frac{2W(\rho_p - \rho_f)}{V_t^2 A_p \rho_p \rho_f} \qquad (3.11)$$

Where g = acceleration due to gravity.

Mp = mass of particle.

$\rho_p$ = mass density of particle.

$\rho_f$ = mass density of fluid.

$A_p$ = project area of the particle normal to the motion.

W = weight of particle.

The drag coefficient is given by C which is considered to be overall drag coefficient such that $C = C_f + C_D$, where flows is laminar, $C_f$ is generally negligible for turbulent flow, $C_f$ is usually negligible except for streamlined bodies. The expression for terminal velocity of object of various shapes are given below.

Spherical bodies.

For a sphere of diameter $d_p$, substituting for

$$A_p = \frac{\pi d_p^2}{4}$$

and

$$W = \frac{\pi}{6} \rho_p g d_p^3$$

yield the following expression for terminal velocity.

$$V_t = [\frac{4gd_p(\rho_p - \rho_f)}{3\rho_f C}]^{\frac{1}{2}} \qquad (3.12)$$

For condition of laminar flow, the values of drag coefficient, C are calculated from (3.9) for Reynold number less than 1.0. Substitution of C and $W_R$ in (3.12), gives the equation of terminal velocity also called (settling velocity, $V_s$) according to stokes law.

$$V_t = \frac{gd_p^2(\rho_p - \rho_f)}{18\eta} \qquad (3.13)$$

Even for Reynold number of 2, equation (3.9) gives a good approximation of drag coefficient (Lapple,1956) for Reynolds number greater than 2, the values of drag coefficient can be found from figure 2.3.

For conditions of turbulent flow in the region where

$10^3 < N_R < 2x10^5$ and C is equal to approximately 0.44, the following equation has been for terminal velocity (Lapple, 1956).

$$V_t = 1.74[\frac{gd_p(\rho_p - \rho_f)}{\rho_f}]^{\frac{1}{2}} \qquad (3.14)$$

Finally for an intermediate region $2 < N_R < 10^3$, the drag coefficient is given by

$$C = \frac{18.5}{(N_R)^{0.6}} \qquad (3.15)$$

and terminal velocity is given by

$$V_t = \frac{0.153\, g^{0.714} d_p^{0.142} (\rho_p - \rho_f)^{0.714}}{\rho_f^{0.286}\, \eta^{0.428}} \qquad (3.16)$$

NON-Spherical bodies

In tables[8], comparative equations for sphere and other objects with regular geometric shapes are given. Very little work has been done on irregular shapes which are particularly complicated by their random orientation and the variety of method of expressing size and dimensions to be used in calculations of Reynold number and frontal area. Schiller (1932) has presented data on drag coefficient in terms of Reynold number for some irregular particles. His graphical presentation[8] is based on the work of several investigations using such materials as Sand, grains, gravel, coal, quartz, dust particles with shapes not well defined geometrically as well as such geometric shapes as spheres and cubes. Drag coefficients and Reynold numbers were obtained by determining the suspension velocity of the experimental particles under free fall in air, liquid paraffin and water.

Since both drag coefficient and Reynold number equations include a velocity term, calculations of terminal velocity from Reynold number and drag coefficient relationship require a trial- and - error solution. The term $CN_R^2$ or $C/N_R$ are first calculated and plotted against $N_R$. Since these two terms do not include velocity, $V_t$, and particle - diameter, $d_p$ respectively depending on whether $V_t$ or $d_p$ is unknown, the value of $CN_R^2$ or $C/N_R$ can be calculated from the given information and by reference to the appropriate plot, the corresponding value of $N_R$ can be obtained.

For spherical particles the terms.

$$CN_R^2 = \frac{4g\rho_f d_p^3(\rho_p - \rho_f)}{3\eta^2} \qquad (3.17)$$

and

$$\frac{C}{N_R} = \frac{4g\eta(\rho_p - \rho_f)}{3\rho_f^2 V_t^2} \qquad (3.18)$$

are obtained by combining (3.6) and (3.12) f (3.11) and (3.6) are combined, the term $CN_R^2$ will not include dp but will require a knowledge of weight and mass density of the object, $\rho$p, as seen from the following resulting equations.

$$CN_R^2 = \frac{8W\rho_f(\rho_p - \rho_f)}{\pi\eta^2\rho_p} \qquad (3.19)$$

Values of $CN_R^2$ are calculated[8] from which Reynolds number are read. A modifications of equation 3.14 have being developed by Condolios and Chapus (1963). The equation is given as

$$V_t = [(\frac{4}{3})gd_e\frac{(\rho_p - \rho_f)}{\rho_f}(\frac{\psi}{C})]^{\frac{1}{2}} \qquad (3.20)$$

Where

$d_e$ = diameter of equivalent sphere (m)

$\rho_p$ = density of particle (kg/m$^3$)

$\rho_f$ = density of fluid (kg/m$^3$)

C = overall drag coefficient which is taken as 0.44 for spheres (stokes 1932).

$\psi$ = The sphericity of the particle.

This modification takes into account the sphericity of the grain/particles

### 3.3.3 FANS

(i) OVER VIEW.

It is probable that the majority of all fans are the centrifugal or radial flow type although other group such as the propeller or axial flow fans exist. Centrifugal fan consist of an impeller running in a casing having a spirally shaped contour.

The air enters the impeller in an axial direction and is discharge at the periphery, the impeller rotation being towards the casing outlet. The amount of work done on the air, evident in the pressure development of the fan, depends primarily on the angle of the fan blades with respect to the direction of rotation at the periphery of the impeller. Three main forms of blade are common viz-:

(a) Backward curved blade:- tips are inclined away from the direction of rotation, and the blade angle $\beta_2$ is said to be less than 90°.

(b) Radial bladed:- Where the blade tips (or vein the whole blade in the case of paddle blade fan) are radial, that is $\beta_2 = 90°$.

(c) Forward cured:- Where the blade tips incline towards the direction of rotation, and $\beta_2$ is greater than 90°.

For the purpose of this project, a forward curved straight blade centrifugal fan will be adopted. Forward curved centrifugal fan have the advantage of "higher volume flow and pressure head". The following definition are related to fans.

(a) Fan total pressure ($P_t$) is the difference between the total pressure at the fan outlet and inlet.

(b) Fan static pressure ($P_s$) is the fan total pressure minus the fan velocity pressure; and

(c) Fan velocity pressure is the velocity pressure corresponding to the average velocity at the fan outlet (found by dividing the volume flow of air by the area of the fan discharge orifice).

(d) Fan efficiency which is the ratio of output power to mechanical power and is usually expressed as a percentage [13]. Fan total efficiency;

$$\eta_t = \frac{air \ power \ (total)}{measured \ fan \ input \ power} \ x \ 100\% \qquad (3.21)$$

Fan static efficiency;

$$\eta_s = \frac{air \ power \ (static)}{measured \ fan \ input \ power} \ x \ 100\% \qquad (3.22)$$

$$Air \ power \ (p) = Pressure \ x \ discharge \ (Q) \qquad (3.23)$$

Pressure may be fan total pressure resulting from air power (total) or pressure may be fan static pressure resulting in air power (static).

Also it is interesting to note that the fan total pressure is the difference between the total pressure at the fan outlet ($P_{t2}$) and the total pressure of the fan at inlet ($P_{t1}$)

ie

$$P_t = P_{t1} - P_{t2} \qquad (3.24b)$$

When fan draws air directly from the atmosphere; $P_{t1} = 0$

When fan discharges air directly to the atmosphere;

$P_{t2} = P_v$. where $P_v$ is the velocity pressure.

(ii) **FAN LAWS**

The performance of a fan in terms of pressure, volume flow and power absorbed depends on a number of factors, the most obvious of which are:

(a) the design and type of fan

(b) the point of operation on the volume/pressure characteristics

(c) the size of fan

(d) the speed of rotation of the impeller

(e) the condition of the air or gas passing through the fan.

The mathematical relationships relating fan laws are given below for similar fans, operating at a given point on their efficiency curve and at some speed performance.

(i) Capacity varies directly as cube of diameter, $Q \propto D^3$

(ii) Pressure varies directly as square of diameter;

$P \propto D^2$

(iii) Power varies directly as fifth power of diameter;

$H \propto D^5$

The expressions above can be combined to relate the performance of a family of similar fans in terms of dimensionless coefficient as in equation below[13].

Capacity coefficient; $= Q/ND^3$

Pressure coefficient; $= gP/N^2D^2$

Power coefficient; $= gH/wN^3D^5$

where $Q$ = capacity, $m^3/s$

$w$ = weight density, $kg/m^3$

$g$ = gravitational acceleration, $m/s^2$

$N$ = revolution per minute of fan impeller.

## (iii) CENTRIFUGAL FAN DESIGN CONSIDERATION

Consider a velocity impeller to which air approaches at a radius $r_i$ with some absolute velocity $V_1$, and enters with a relative velocity $\omega_1$ and at an angle $\beta_1$ to the impeller periphery. A vane is fixed to the impeller at an angle $\beta_1$ at the point of entry allowing the air pass over the surface at this point without change of direction. The air passes along the vane, leaving it with a relative velocity $\omega_2$ at the impeller outer radius $r_2$, where the angle made by the vane to the periphery is $\beta_2$. The absolute leaving velocity of the air (relative to some fixed point outside the impeller), $V_2$ will depend on $\omega_2$ and the impeller periphery velocity $U_2 = \omega r_2$.

Considering the velocity to be coplanar, velocity triangle may be constructed to show vectorially the relative velocity at inlet and outlet of the impeller as shown in Figure 3.4. The tangential components of absolute velocities $V_1$ and $V_2$ are $V_{u1}$ and $V_{u2}$, whilst the radial (or meridional) components are $V_{m1}$ and $V_{m2}$. The work done on the air by the impeller will be the difference between the energy in the air leaving and the energy in the air entering the impeller in the direction of rotation[13].

Energy in air leaving the impeller

$$= mV_{u2} \times r_2 \times \omega_2$$

for a mass flow of air of m.

Similarly, energy in air entering the impeller

$$= mV_{u1} \times r_1 \times \omega_1$$

Thus, work done by the impeller

$$= m(V_{u2}U_2 - V_{u1}U_1) \qquad (3.25)$$

The load (h) developed by the impeller is given by

$$h = \frac{V_{u2}U_2}{g} - \frac{V_{u1}U_1}{g} \qquad (3.26)$$

Equation (3.26) is sometimes referred to as Euler's equation for an impeller, and the head h as Euler's head.

Also, in terms of head and pressure, equation (3.26) can be re-written as

In terms of velocity, equation (2.25) can also be expressed as

$$h = \frac{U_2^2 - U_1^2}{2g} + \frac{V_2^2 - V_1^2}{2g} + \frac{\omega_1^2 - \omega_2^2}{2g} \qquad (3.28)$$

Since the vast majority of centrifugal fans do not have inlet guide vanes, air will tend to enter in a radial direction
as uniform pressure distribution might be expected at both inlet and outlet peripheries. In such cases, $V_{u1} = 0$ and the total pressure developed,

$$P = \rho_f U_2 V_{u2} \qquad (3.29)$$

Variable guide vanes are sometimes used for regulation of volume flow. These reduce the amount of work done on the air by the impeller by impacting rotation of the air in the direction of the impeller rotation, thus giving finite values to $V_{u1}$.

For normal centrifugal fans, it is seen from equation (3.29) that the work done by an impeller increases from. From Figure 3.4 it can be seen that as the blade angle $\beta_2$ increases the ratio $V_{u2}/U_2$ also increases for the type of impeller shown.

(i) backward curved; $\beta_2 = 90°$, $V_{u2} < U_2$

(ii) radial tipped; $\beta_2 = 90°$, $V_{u2} = U_2$

(iii) forward curved; $\beta_2 > 90°$, $V_{u2} > U_2$

The volume flow through the impeller will be the product of velocity and the area normal to the direction of flow.

Take the radial velocity $V_m$: Volume flow,

$$Q = \pi d_2 b_2 V_{m2} = \pi d_1 b_1 V_{m1}$$

were $b_2$ and $b_1$ are the blade widths (axially) at the diameters $d_2$ and $d_1$ respectively. In practice, $V_{m1}$ and $V_{m2}$ are very nearly the same in value, and of the order of $0.2U_2$ (Osborne, 1978). Ideally, the velocity through the impeller inlet (or `eye' as it is sometimes called), $V_o$, should not be more than $V_{m1}$. However, this cannot often be achieved since the inlet diameter becomes too large, and $V_o$ is generally of the order of twice $V_{m1}$ or $0.4U_2$.

Since volume flow is a function of $V_{m2}$, theoretical characteristics of impeller total pressure and power requirements may be derived, assuming radial entry:

(a) $\beta_2 < 90°$: Figure 3.5 shows that

$$U_2 - V_{u2} = V_{m2}\cot\beta_2$$

$$\therefore V_{u2} = U_2 - V_{m2}\cot\beta_2$$

Impeller total pressure,

$$P_t = \rho U_2 V_{u2} - \rho U_2 V_{m2}\cot\beta_2$$

Volume flow

$$Q = \pi d_2 b_2 V_{m2}$$

Thus,

$$P_t = \rho_f U_2^2 - \rho_f U_2 \frac{Q}{\pi d_2 b_2}\cot\beta_2 \qquad (3.30(a))$$

This may also be written in terms of non-dimensional performance coefficients

$$\psi = \frac{P_t}{\frac{1}{2}\rho_f U_2^2}; \qquad \phi = \frac{4Q}{\pi d_2^2 U_2} \qquad (3.30(b))$$

and substituting these values in equation (3.30(a)) gives

$$\psi = 2(1 - \frac{\phi d_2}{4b_2}\cot\beta_2) \qquad (3.31)$$

Power coefficient,

$$\lambda = \phi\psi = 2\phi(1 - \frac{\phi d_2}{4b_2}\cot\beta_2) \qquad (3.32)$$

(b) $\beta_2 = 90°$: In this case

$$U_2 = V_{u2}$$

$$P_t = \rho_f U_2 V_{u2} = \rho_f U_2^2 \qquad (3.33)$$

and

$$\psi = 2; \quad \lambda = 2\phi \qquad (3.34)$$

(c) $\beta_2 > 90°$: Here,

$$V_{u2} - U_2 = V_{m2}\cot(180 - \beta_2)$$

from which

$$P_t = \rho_f U_2 V_{u2} = \rho_f U_2 V_{m2}\cot(180 - \beta_2)$$

$$\therefore P_t = \rho_f U_2^2 + \frac{\rho_f U_2 \phi}{\pi d_2 b_2}\cot(180 - \beta_2) \qquad (3.35)$$

and

$$\psi = 2[1 + \frac{\phi d_2}{4 b_2} \cot(180 - \beta_2)] \qquad (3.36)$$

$$\lambda = 2\phi[1 + \frac{\phi d_2}{4 b_2} \cot(180 - \beta_2)] \qquad (3.37)$$

These theoretical characteristics are plotted in graphs and are seen to differ in shape as from those of actual fans.

## (iv) PRESSURE LOSS

The head loss $h_f$, or energy loss per unit weight, due to friction can be conveniently expressed in terms of the velocity head of the fluid by using the Darcy formula[5]

$$h_f = \frac{fL U^2}{2gm} \qquad (3.38)$$

where m = A/P

Equation (3.38) becomes

$$h_f = \frac{fL U^2}{A\ 2g} \qquad (3.39)$$

For a rectangular section with sides *a* and *b*

A = ( a x b)

P = 2(a + b)

Therefore, head loss is given by

$$h_f = \frac{2fL(D + W)}{DW}\frac{U^2}{2g} = \frac{fL(D + W)U^2}{DWg} \qquad (3.40(a))$$

where f is a `friction factor' with some dependence on Reynold number and for galvanized sheet is in the order of 0.005[13].

Also, the head loss through the nozzle head[13] is given by

$$h_L = \frac{V_1^2 k}{2g} \qquad (3.40(b))$$

where $V_1$ = air velocity at entrance

k = a loss factor available from tables[13] with dependence on nozzle geometry.

g = acceleration due to gravity.

### 3.3.4 POWER TRANSMISSION SYSTEMS

Since the winnower is to be motorised, a V-belt has been adopted for the design work. V-belt drives provide a positive means of transmitting power between parallel shaft at a constant angular velocity ratio. The value of the velocity ratio is the same as would be obtained by two imaginary cylinders pressed together and rotating without slippage at their line of contact.

Some of the advantages of belt drives over other forms of power transmitting drives include:

(a) Installation is relatively easy.

(b) Belt can be used for shaft spacing too great for gears.

Some of the disadvantages of using belt drives as a means of transmitting power include

(a) Slippage between belt and pulley, which is very undesirable if constant speed ratio is to be maintained.

## BELT DRIVE DESIGN DETAILS

Consider the drive shown in Figure 3.6; power is transferred from the smaller pulley *A* to another pulley *B*. Pulley *A* is attached to the electric motor and has diameter, $b_1$ and rotates at speed, $N_1$ (rpm), Pulley *B* is keyed to the fan shaft, it has diameter $d_2$ and rotates at $n_2$ (rpm).

The expression relating the power transmitted by the drive to the tension and velocity of the belt[4] is given by

$$H_E = (T_1 - T_2)V \qquad (3.41)$$

where $T_1$ = belt tension on tight side, N

$T_2$ = belt tension on the slack side or loose side, N

V = belt velocity (m/s)

The angle of wrap _ on pulley is related to $T_1$ and $T_2$ as

$$\sin\beta = \frac{R - r}{C_k} \qquad (3.42)$$

and

$$\alpha_1 = 180° - 2\beta = 180° - 2\sin^{-1}(\frac{R - r}{C_k}) \qquad 3.43$$

$$\alpha_2 = 180° + 2\beta = 180° + 2\sin^{-1}(\frac{R - r}{C_k}) \qquad (3.44)$$

The maximum tension in the tight side of the belt depend on the allowable stress of the belt material. Leather and cotton duck impregnated with rubber built up in piles are generally used[4]. The allowable tensile stress for leather belting is usually 2 to 3.45MPa and the allowable stress for rubber belting will run from 1 to 1.7MPa, depending on the quality of the material[4].The density of leather is about 970kg/m$^3$. Rubber belting has density of 1250kg/m$^3$.

The following formula for determining the value of $T_2$ for both flat and V-belts applies when the width and thickness of the belt are known[4].

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\left(\frac{f\alpha}{\sin 1Over2\theta}\right)} \qquad (3.45)$$

$$m = bt\rho$$

where m = the mass of 1m of belt

v = belt velocity, m/s

b = belt width, m

t = belt thickness, m

ρ = belt density, kg/m$^3$.

The quantity $mv^2$ is due to centrifugal tension which tends to cause the belt to leave the pulley and reduces the power that may be transmitted.

Load carrying capacity of a pair of pulley is determined by $e^{(f\_/\sin ½\Theta)}$. It is determined by the pulley that has the smaller value.

### 3.3.5 DESIGN OF HOPPER

#### A. THEORETICAL ANALYSIS

In designing a hopper, it is recommended that the angle of inclination of the sides of the hopper to the horizontal must be greater than the angle of internal friction between the hopper wall and the stored material. Rickey et all (1961) recommended that the angle of inclination be 10° higher than the natural angle of repose of the stored material. Pieper (1977) explained emptying of materials from silo by identifying three zones within the cell as illustrated in Figure 3.6.

In zone 3 the stored material is likely to move as a mass, although with different relative velocities across the section. In zone 2, funnel flow (piping) develops, a case where there is flow but it is limited. In zone 1, a case referred to as `arching' where the material appear to be standstill. The relative height of the zones may be quite different and any of the three may vanish.

The largest lateral pressure occur in region A, Figure 3.6, because of the abrupt change in material across-section. In region B, development of domes may inhibit material flow.

These domes are short-lived. Following the collapse of a dome the entire material above falls rapidly and creates a shock. Periodic repetition of this phenomenon creates a hammering effect. In region C the firm walls of the funnel offer the necessary support for development of a large domes which may halt material flow.

Mester (1980) gave the following two conditions to be satisfied to ensure that arching and funnelling do not occur.

$$\frac{r}{H_o} \geq \mu \tan \phi_i \qquad (3.46)$$

$$\frac{r}{H_o} \geq \mu \tan(\theta + \beta_k) - \tan \beta \qquad (3.37)$$

The approximate dimensioning of hoppers can be achieved by using the desired flow rate, Q. In the expression of the form due to Mohsenin, 1978

$$Q = kB^n \qquad (3.48)$$

From equation (3.48) above, the flow rate is an experimental function. k and n are two constants which are found earlier by substituting experimental data from two set of test and solving the two equations simultaneously or by determining directly from the shape and intercepts of the straight line plot of logQ versus logB on a graph sheet.

The procedure followed is the same as that used by

Chukwu, O (1987) in determining k and n values. See Table 3.1 for values used in the determination of k and n and also Figure 4.10 for the graph of logQ against logB.

TABLE 3.1 Values used in determining k and n

| Q | 5 | 10 | 15 | 20 | 25 | 30 |
|---|---|---|---|---|---|---|
| Y = logQ | 0.69 | 1.00 | 1.17 | 1.30 | 1.39 | 1.47 |
| B | 10 | 20 | 30 | 40 | 50 | 60 |
| X = logB | 1.0 | 1.30 | 1.47 | 1.60 | 1.69 | 1.77 |

Source: CHUKWU, O (1987): DESIGN OF A HEAT GENERATOR FURNACE.

**B FORCE ANALYSIS ON HOPPER**

For the purposes of this project, Remibert (1961) approach has been adopted.

Remibert in 1961 quoted the following expression for the forces acting on the straight wall silo but could also be adopted for this hopper design.

The equations are as given below:

The vertical pressure, q acting on the wall is given by

$$q = \gamma [Y(\frac{Y}{C} + 1)^{-1} + \frac{h_s}{3}] \qquad (3.49)$$

The lateral pressure, p on the wall surface is given by

$$p = P_{max} [1 - (\frac{Y}{C} + 1)^{-2}] \qquad (3.50)$$

The vertical frictional force per unit width of wall, V is given by

$$V = (\gamma Y - q)R \qquad (3.51)$$

The height of slopping top surface of stored material, $h_s$ is given by

$$h_s = \frac{D}{2}\tan\phi_r \qquad (3.52)$$

The hydraulic radius[2] of cross-section a' is given by

$$a' = \frac{2a_2b_2}{a_2 + b_2} \qquad (3.53)$$

(Rectangular cross-section along the longest side).

*C* is calculated as

$$C = \frac{a'}{\pi\mu'k} - \frac{h_s}{3} \qquad (3.54)$$

and $P_{max}$ from the relation

$$P_{max} = \frac{\gamma a'}{4\mu'} \qquad (3.55)$$

## C DESIGN FOR CRITICAL DIMENSIONS OF HOPPER ORIFICE

Critical dimension of the hopper opening is determined for failure condition which must be established for two basic obstructions; namely `arching' where no flow may take place, and `piping' where flow may be reduced or limited.

Figure 3.7 shows the free body diagram of a granular material with bulk density, W, forming an arch with a uniform thickness, T. Let *B* denote the diameter of a circular hole and $B_o$ the width of a slot with length, L.

For small arcs, the equilibrium of forces, resulting from the weight of the mass acting downward and the vertical component of force due to compressive pressure, P in the arch acting upward, yields (see Figure 3.7):

$$WB_o LT = 2PLT \cos\alpha \sin\alpha$$

$$\therefore\ B_o = \frac{P}{w} \sin 2\alpha \qquad (3.56)$$

for slot, and

$$\omega \frac{\pi}{4} B^2 T = P\pi\, B_o T \cos\alpha \sin\alpha$$

$$\therefore\ B_o = (\frac{2P}{\omega}) \sin 2\alpha \qquad (3.57)$$

(for circle)

Given the above analysis, Johnson and Colijn (1964) suggested that in order for failure to occur, the major compressive pressure, P should be equal to the unconfined yield strength, $\sigma_c$. Substituting $\sigma_c$ for P in the above expression, and assuming sin $2\alpha$= 1, which considers the strongest possible arch that may from, the critical opening dimension $B_o$ becomes

$$B_o \geq \frac{\sigma_c}{\omega} \qquad (3.58)$$

for slot opening and

$$B_o \geq \frac{2\sigma_c}{\omega} \qquad (3.59)$$

for circular opening.

The critical opening dimension to prevent `piping' has been derived from the stability analysis of vertical pipes and is given in terms of $\sigma_c$, $\omega$ and angle of internal friction $\phi_i$ (Jenike, 1961).

$\sigma_c$ can be calculated from the Rankine equation for shallow bins as

$$\sigma_c = \gamma Y \tan^2(45° - \frac{\phi_r}{2}) \qquad (3.60)$$

# CHAPTER FOUR

## 4.0 DESIGN CALCULATIONS

### 4.0.1 TERMINAL VELOCITY DETERMINATION

Design data from table 4.1.

For Soya bean:

Mass of single particles ($m_p$) = $2.059 \times 10^{-4}$ kg

Weight of particle ($w_p$) = $m_p \times g = 2.059 \times 10^{-4} \times 9.81$

$= 2.0203 \times 10^{-3}$ N

Geometric mean diameter of particle = $6.763 \times 10^{-3}$m.

Absolute velocity of air $\eta = 1.7907 \times 10^{-5}$kg/ms

Particle Density, $\rho_p$: This is computed for soya bean assuming it to be spherical in shape. The volume of the particle is obtained from the relation

$$v_p = \frac{\pi}{4} x \pi x r^3 = \frac{4}{3}\pi(\frac{6.763 x 10^{-3}}{2}) = 1.61 x 10^{-7} m^3$$

$$\rho_p = \frac{m_p}{v_p} = \frac{2.059 \times 10^{-4}}{1.61 \times 10^{-7}} = 1271.275 \, kg/m^3.$$

Density of air, $\rho_f$ =1.293kg/m$^3$ (Rogers, G.F.C et al, 1967). From equation (3.20), the terminal velocity assuming turbulent flow condition, can be estimated as

$$V_t = [(\frac{4}{3})\frac{gd_p(\rho_p - \rho_f)}{\rho_f}(\frac{\psi}{C})]^{\frac{1}{2}}$$

Sphericity of particle $\psi = 0.756$

C = 0.44 for turbulent flow conditions for spheres and streamlined bodies[5].

Also,

$$V_t = 1.74[\frac{9.81 \times 6.763 \times 10^{-3} \times (1271.275 - 1.293) \times 0.756}{1.293}]^{\frac{1}{2}}$$

$$\therefore V_t = 12.21\, m/s$$

Values of terminal velocities for rice, groundnut, and millet are computed the same way and are shown in table 4.1.

TABLE 4.1

| S/N | Grain | A | B | C | D | E | F |
|---|---|---|---|---|---|---|---|
| 1 | RICE | 50 | 0.02 | 3.2 | 1165.68 | 0.378 | 7.09 |
| | CHAFF | - | - | - | - | - | 1.10-2.00 |
| 2 | SOYABEAN | 50 | 0.2059 | 6.763 | 1271.275 | 0.756 | 12.0 |
| | CHAFF | - | - | - | - | - | - |
| 3 | GROUND-NUT KERNELS | 82 | 0.390 | 9.76 | 801.0 | 0.642 | 10.73 |
| | SHELL | - | - | - | - | - | 0.33-3.30 |
| 4 | MILLET | 82 | .00831 | 2.61 | 892.0 | 0.720 | 6.20 |
| | CHAFF | - | - | - | - | - | - |

KEY: A - No of grains tested

B - Mass of single grain

C - Equivalent diameters

D - Particle density (Calculated) $kg/m^3$

E - Sphericity ($\psi$)

F - Terminal velocity (m/s)

From table 4.1 and 4.2 the desired air velocities chosen for cleaning are:

(i) Rice and chaff is 3.0m/s

(ii) Millet and chaff is 3.0m/s

(iii) Groundnut and shell is 4.0m/s

(iv) Soyabean and chaff is 4.0m/s

These velocities were carefully considered after studying experimental and calculated values of terminal velocities of grains above in an aspiration column as observed by Grover, P.C. and Kashya,P (1980), table 4.2 and values calculated fro the respective grains and legumes in table 4.1.

## 4.2 FAN DESIGN CALCULATIONS

The requirement for discharge through a blower can be estimated on the basis of velocity of air required for separation, V; depth of air stream, D; and the width over which the air stream is required to have effect W.

For the purpose of this design The value of D and W are chosen to be

D = 200mm = 0.2m and W = 80mm = 0.08m

Design will be based on the highest velocity of 4.0m/s

Actual discharge of fan

$$Q = VDW = 4.0 \times 0.2 \times 0.08 = 0.064 \; m^3/s$$

From equation (3.30(b)) we had

$$Q = \frac{4Q}{\pi d_2^2 U_2}$$

Also, the value of $\phi$ for forward curved centrifugal fans with outlet angle of 140° as quoted by Osborne (1977) is in the order of 0.18 or 18%.

The impeller peripheral speed, $U_2$ is given as

$$U_2 = \frac{\pi d_2 n}{60}$$

where $d_2$ = impeller diameter, m

n = speed of impeller shaft in rpm.

For the purpose of this design, the impeller shaft speed is taken to 544.44 rpm. Therefore peripheral speed of impeller is calculated as

$$U_2 = \frac{\pi d_2 \times 544.44}{60} = 9.074\pi d_2 m/s$$

Substituting this value into equation (3.30(b)), we obtain

$$\phi = \frac{4Q}{9.074\pi^2 d_2^3}$$

From which

$$d_2 = [\frac{4Q}{\phi \pi^2 \ x \ 9.074}]^{\frac{1}{3}}$$

But $\phi = 0.18$; $Q = 0.064 m^3/s$

$$\therefore \ d_2 = [\frac{4 \ x \ 0.064}{0.18 \ x \ \pi^2 \ x \ 9.074}]^{\frac{1}{3}} = 0.2514 \ mm = 251.4 \ mm$$

For the purpose of this design an impeller diameter $d_2 = 260mm$ is adopted.

The mean outlet velocity of fan, $V_3$ is given as 22% of peripheral velocity (Osborne, 1977) ie

$V_3 = 0.22U_2$

but

$$U_2 = 9.074\pi d_2 = 9.074 \ x \ \pi \ x \ 0.260 = 7.412 \ m/s$$

$$\therefore \ V_3 = 0.22 \ x \ 7.412 = 1.63 \ m/s$$

The air inlet velocity[13], $V_o$ at the impeller eye is given by

$$V_o = 0.4U_2 = 0.4 \ x \ 7.412 = 2.9648 \ m/s$$

$$V_{m1} = V_{m2} = 0.5V_o = 0.5 \ x \ 2.9648 = 1.4824 \ m/s$$

The casing inlet diameter[13], $d_1$ is given by

$$d_1 = \sqrt{\frac{4Q}{\pi V_o}} = \sqrt{\frac{4 \times 0.064}{\pi \times 2.9648}}$$

$$\therefore\ d_1 = 0.1658\ mm = 165.8\ mm$$

For the purpose of this design, a casing inlet diameter

$d_1$ =166mm is adopted.

Blade widths[13] can be estimated from the expressions given by

$$\therefore\ b_3 = \frac{Q}{\pi d_1 V_{m1}} = \frac{0.064}{\pi \times 0.166 \times 1.4824} = 0.08276\ m$$

$$\therefore\ b_3 = 82.76\ mm$$

Rounding off to the nearest metric standard gives $b_3$ = 85 mm.

$$\therefore\ b_4 = d_1 \frac{b_3}{d_2} = 0.166 \times \frac{0.085}{0.260} = 0.05443\ m$$

$$\therefore\ b_4 = 54.43\ mm$$

For the purpose of this design, a blade width $b_4$ = 56 mm will be used.

The casing width is in the order of[13] 2.5$b_4$ for backward curved blade fans. Hence,

$C_w$ = 2.5 x 56 = 140 mm.

The casing outlet area

$$A_c = \frac{Q}{V_3} = \frac{0.064}{1.63} = 0.03964\ m^2$$

The casing outlet height

$$H_c = \frac{A_c}{C_w} = \frac{0.039264}{0.14} = 0.2805\ m = 280.5\ mm$$

For the purpose of this design, a casing outlet height of

280 mm is adopted.

Blade inlet angle[13] is given by

$$\beta_1 = \tan^{-1}\left(\frac{V_{m1}}{U_1}\right) = \tan^{-1}\left(\frac{V_{m1} d_2}{U_2 d_1}\right)$$

$$\therefore\ \beta_1 = \tan^{-1}\left(\frac{1.482 \times 0.260}{7.412 \times 0.166}\right) = 17.39°$$

Power required[13] to drive the fan, H is given by

$$H = \frac{Q \times P_s}{\eta}$$

Now, $Q = 0.064\ m^3/s$; $\eta_t$ 75%[13]. $P_s$ can be found from the relation[13].

$$\frac{P_s + \frac{1}{2}\rho_f (0.22 U_2)^2}{\eta_t} = \rho_f U_2^2 + \frac{\rho U_2 Q}{\pi d_2 b_4} \cot(180° - \beta_2)$$

with $\rho_f = 1.293\ kg/m^3$; $0.22U_2 = V_3 = 1.63\ m/s$; $\beta_2 = 140°$;

$\cot(180° - \beta_2) = \cot 40° = 1.192$; $\eta_t = 75\% = 0.75$

$U_2 = 7.412$ m/s; $d_2 = 0.26$ m; $b_4 = 0.056$ m; $Q = 0.064$ $m^3/s$

Substituting these values into the above equation we obtain

$$\frac{P_s + 0.5(1.293)(1.63)^2}{0.75} = 1.293 \text{ x } (7.41)^2 + \frac{1.293 \text{ x } 0.064}{\pi \text{ x } 0.026 \text{ x } 0.056} \cot(180° - 140°)$$

$$P_s + 1.718 = 73.152 \text{ x } 0.75$$

$$\therefore \; P_s = 54.864 - 1.718 = 53.146 \; N/m^2$$

The fan velocity pressure[13] is computed from the expression given by

$$P_v = \frac{1}{2} \rho_f V_3^2$$

$$= \frac{1}{2} \text{ x } 1.293 \text{ x } (1.63)^2$$

$$\therefore \; P_s = 1.718 \; N/m^2$$

The fan total pressure

$$P_t = P_s + P_v = 53.146 + 1.718 = 54.86 \; N/m^2$$

Hence, the required fan design power

$$H = \frac{0.064 \text{ x } 53.146}{0.75} = 4.535 \; watts$$

But because of friction, the power required is far higher than this[15].

Euler head or pressure head developed by fan is computed from thew relation

$$h = \frac{U_2 V_{u2}}{g}$$

But for forward curved blades,

$$V_{u2} = U_2 + V_{m2} \cot(180° - \beta_2)$$

Hence with $\beta_2 = 140°$; $U_2 = 7.412$ m/s; $g = 9.81$ m/s$^2$ and

$V_{m1} = V_{m2} = V_m = 1.4824$ m/s, then, we obtain

$$V_2 = 7.412 + 1.4824 \cot(180° - 140°) = 9.147 m/s$$

Hence,

$$h = \frac{7.412 \times 9.176}{9.81} = 6.93\ m$$

Since this represent the height to which air must be raised to give the same energy per unit weight as the work done by the impeller, the equivalent head[13] of water is easily computed from the relation

$$\rho_w g H_w = \rho_f g h$$

Taking $\rho_w = 1000$kg/m$^3$ we obtain

$$H_w = \frac{\rho_f h}{\rho_w} = \frac{1.293 \times 6.93}{1000} = 8.96 \times 10^{-3}\ m\ \ of\ \ water$$

DETERMINATION OF NOZZLE HEAD DIMENSIONS AT FAN OUTLET

Applying continuity equation we obtain

$$Q_1 = Q_2$$

$$A_1V_1 = A_2V_2$$

$$\therefore V_2 = \frac{A_1V_1}{A_2} = \frac{0.0392 \times 1.63}{0.016} = 3.9935\ m/s$$

Which is approximately 4.0 m/s.

Figure 4.1(a) can be projected to meet at point A as shown in figure 4.1(c). From triangle ABC we obtain

$$\tan 30° = \frac{BC}{AG}$$

$$AC = \frac{BC}{\tan 30°} = \frac{40}{\text{Tan } 30°} = 69.28\ mm \approx 69\ mm$$

$$\therefore CF = 180 - 69 = 111\ mm$$

Since the fan is to be used to clean more than one grain/legume, the method of throttling the air inlet diameter is adopted. In order to obtain the desired inlet diameter required to clean rice and chaff and millet and chaff at 3.0 m/s while keeping other factors like speed of fan shaft, casing outlet area, impeller diameter and blade width constant, $d_1$ can be calculated from

$$d_1 = [\frac{4Q}{\quad}]^{\frac{1}{2}}$$

The required discharge, Q corresponding to the inlet diameter can be estimated as follows:

From the continuity equation

$$A_1V_1 = A_2V_2$$

$$V_1 = \frac{A_2V_2}{A_1} = \frac{0.016 \times 3.0}{0.0392} = 1.224\ m$$

Hence, the desired discharge

$$Q = 1.224 \times 0.0392 \approx 0.048\ m^3/s$$

$$\therefore\ d_1 = [\frac{4 \times 0.048}{\pi \times 2.7644}]^{\frac{1}{2}} = 0.14868\ m = 148.68\ mm$$

For the purpose of this design, $d_1$ = 149 mm will be used.

## PRESSURE LOSS ESTIMATION

The head loss, $h_f$ or energy per unit weight loss due to friction through the length of the cleaning diameter is estimated from Darcy's equation[5] given by

$$h_f = \frac{fLP}{A}\frac{V^2}{2g} = \frac{fL(D + W)V^2}{DWg}$$

For V = 4.0 m/s; L = 265 mm = 0.265 m; g = 9.81m/s$^2$ and

f = 0.005 for galvanized iron sheet[13], we obtain

$$h_f = \frac{0.005 \times 0.265 \times (0.2 + 0.08) \times (4.0)^2}{0.2 \times 0.08 \times 9.81} = 3.782 \times 10^{-2}\ m\ \ of\ \ air$$

For V = 3.0 m/s; L = 0.265 m; g = 9.81 m/s$^2$ and

f = 0.005 we obtain

$$h_f = \frac{0.005 \times 0.265 \times (0.2 + 0.08) \times (3.0)^2}{0.2 \times 0.08 \times 9.81} = 2.127 \times 10^{-2} \; m \;\; of \;\; air$$

Head loss through the nozzle head is computed from equation (3.40(b)) as

$$h_L = [\frac{1}{2}\rho_f V_1^2 k]\frac{1}{\rho_f g} = \frac{V_1^2 k}{2g}$$

where k = a loss factor; for tapered changes of section without much change in area, where the included angle (Fig. 4.1(c)) is greater than or equal to 60°, a loss factor of 0.15 has been suggested[13].

Hence, at $V_1 = V_3$ = 1.63 m/s and g = 9.81 m/s$^2$ we obtain

$$h_L = \frac{(1.63)^2 \times 0.15}{2 \times 9.81} = 0.0203 \; m \;\; of \;\; air$$

Also, at $V_1 = V_3$ = 1.224 m/s and g = 9.81 m/s$^2$ we obtain

$$h_L = \frac{(1.224)^2 \times 0.15}{2 \times 9.81} = 0.012 \; m \;\; of \;\; air$$

Hence, total pressure loss at air flow velocity of 4.0 m/s is compute as

$$h_t = h_f + h_L = (3.782 \times 10^{-2} + 0.0203)m = 0.05812 \; m \;\; of \;\; air$$

Similarly, total pressure loss at air flow velocity of 3.0 m/s is computed as

$$h_l = (2.127 \times 10^{-2} + 0.012)m = 0.03327\,m \text{ of air}$$

The value of head loss obtained for air flow velocities of 4.0 m/s and 3.0 m/s are far less than that developed by the fan (6.93 m).

### 4.3.0 POWER TRANSMISSION SYSTEM DESIGN ANALYSIS

(A) Determination Of Fan Shaft Diameter, $D_s$.

(i) Estimation Of Impeller Weight

Impeller mass = mass of backplate + mass of hub + mass of blades

Considering the blade profile as shown in Figure 4.2. Length of blade = $D_1$ - $D_2$ where $D_1$ and $D_2$ are the impeller root diameters and impeller eye diameters respectively. Hence,

Length of blade = 260 - 166 = 94 mm.

Area of blade

$$A_b = \frac{1}{2}(0.085 + 0.056) \times 0.094 = 0.006627\ m^2$$

Assuming the fan is to be constructed from a gauge 16 mild steel sheet metal of thickness 1.6 mm then,

Volume of blade = area of blade x thickness of blade

= 0.006627 x 1.6 x $10^{-3}$ = 1.0603 x $10^{-5}m^3$.

Mass of blade = volume of blade x density of blade material

Now, density of steel = 7850 kg/$m^3$

Hence, mass of blade = 1.0603 x $10^{-5}$ x 7850 = 0.0832 kg.

Total mass of blades = 0.0832 x 12 = 0.9988 kg.

Mass of backplate

$$m_{bp} = \frac{\pi}{4} \times (0.260)^2 \times 1.6 \times 10^{-3} \times 7850 = 0.6668\ kg$$

Assuming the hub mass is estimated as 20% of the mass of backplate and total mass of blade, then, the mass of hub is

$$m_h = \frac{20}{100}(0.9988 + 0.6668) = 0.333\ kg$$

Mass of impeller

$$m_{imp} = (0.9988 + 6668 + 0.333) = 1.9987\ kg$$

and weight of impeller, $w_{imp}$ = 1.9987 x 9.81 = 19.60 N

Assuming that the machine is to be powered from an electric motor rotated at 1400 rpm and pulley diameter 70 mm, then the required pulley diameter at the fan shaft is computed from the relation

$$D_p = \frac{N_m D_m}{N_p} = \frac{1400 \times 70}{544.44} = 180\ mm$$

where $N_m$ = motor speed = 1400 rpm

$D_m$ = motor pulley diameter = 70 mm

$N_p$ = desired fan shaft speed = 544.44 rpm.

For the purpose of this design a fan shaft pulley diameter $D_p$ = 180 mm is adopted.

(ii) Computation Of Belt Tensions:

From equation (3.45) we had

$$\frac{T_1 - mv^2}{T_2 + mv^2} = e^{\frac{f\alpha}{\sin\frac{1}{2}\theta}}$$

Adopting an A-section v-belt with the following data[3]:

Maximum kilowatt = 0.75 - 5 kW

Belt top width = 13.0 mm

Belt thickness = 8.0 mm

Density of belt material = 1250 kg/$m^3$[4].

Then mass of belt

$$m = 1250 \times 13 \times 10^{-3} \times 8 \times 10^{-3} = 0.18 \text{ kg/m.}$$

Adopting 1.7 MPa as allowable stress for rubber belt[4] and assuming a friction coefficient between belt and pulley of 0.3[3], then the belt tensions can be evaluated as follows.

The angle of wrap on the smaller pulley $_1$ is given by eqn. 3.43 45

$$\alpha_1 = 180° - 2\sin^{-1}\left(\frac{R - r}{C_k}\right)$$

For the purpose of this design a centre distance,

$C_k$ = 400 mm is adopted. Hence,

$$R = \frac{D_p}{2} = \frac{180}{2} = 90\,mm$$ (Radius of fan pulley).

$$r = \frac{D_{mp}}{2} = \frac{70}{2} = 35\,mm$$ (Radius of motor pulley).

$$\alpha_1 = 180° - 2\sin^{-1}\left(\frac{90 - 35}{400}\right) = 164.19°$$

$$\alpha_2 = 180° + 2\sin^{-1}\left(\frac{90 - 35}{400}\right) = 195.80°$$

Substituting these values into equation (3.45) we obtain

$$e^{\frac{0.3 \times \left(\frac{164.19 \times \pi}{180}\right)}{\sin\frac{1}{2}(40°)}} = 2.514 \quad for \quad \alpha = 164.190°$$

$$e^{\frac{0.3 \times \left(\frac{195.80 \times \pi}{180}\right)}{\sin\frac{1}{2}(40°)}} = 2.9975 \quad for \quad \alpha = 195.80°$$

Since $e^{f\_/\sin½Θ}$ for _ = 164.19° is lower, it guides the design.

Given that N = 544.44 rpm and d = 180 x $10^{-3}$m, then the belt speed, V in meters per second is given by[4]

$$V = \frac{\pi d N}{60} = \frac{\pi \times 180 \times 10^{-3} \times 544.44}{60} = 5.13\,m/s$$

The maximum permissible tension, $T_1$ on the tight side of the pulley is computed as

$T_1 = 1.7 \times 10^6 \times 13 \times 10^{-3} \times 8 \times 10^{-3} = 176.8$ N

Hence

$$\frac{176.8 - 0.13 \times (5.13)^2}{T_2 - 0.13 \times (5.13)^2} = 2.415$$

$$\frac{173.3788}{T_2 - 3.42} = 2.514$$

$$T_2 - 3.42 = \frac{173.3788}{2.514}$$

$$\therefore T_2 = 67.784 + 3.42 = 72.385\,N$$

The total force acting shaft is computed as

$T_1 + T_2 = 176.8 + 72.385 = 249.185$ N

The shaft can be considered as a simply supported beam carrying a weight of 280N at its left end and impeller weight of 19.60N at the right end as shown in figure 4.3(a).

Assuming the belt to be inclined at 45° to the horizontal, the components of the belt tension $(T_1 + T_2)$ are

$(T_1 + T_2)\sin 45°$ - Horizontal component

$(T_1 + T_2)\sin 45°$ - Vertical component

Considering the horizontal component, the forces acting on the shaft are shown in Figure 4.3(b)

(iii) Determination Of Shaft Reactions:

$$\Sigma F_v \uparrow = R_1 + R_2 = 176.2 + 19.60 = 195.80\,N$$

Taking moment about R, we obtain

$$\Sigma M_{R1} = 0 = -176.2 \times 0.1 + 19.60 \times 0.25 = R_2 \times 0.2$$

$$\therefore \; R_2 = \frac{-17.62 + 4.9}{0.2} = -63.60\,N$$

which implies that $R_2$ acts in the opposite direction.

$$\therefore \; R_1 = 195.8 - R_2 = 195.8 - (-63.6) = 259.4\,N$$

(iv) Determination Of Shaft Shearing Force:

For $0 \leq x_1 \leq 0.1\,m$

$$V_x = 196.20\,N$$

For $0.1 \leq x_2 \leq 0.3\,m$

$$V_x = -176.2 + 259.4 = 83.2\,N$$

For $0.3 \leq x_3 \leq 0.35\,m$

$$V_x = -176.2 + 259.4 - 63.60 = 19.6\,N$$

(v) Determination Of Bending Moment:

$$For \quad 0 \leq x_1 \leq 0.1\, m$$

$$M_x = -176.2\, x_1$$

$$At \quad x_1 = 0, \quad M_x = 0$$

$$At \quad x_1 = 0.1$$

$$M_x = -176.2 \times 0.1 = -17.62\, N.m$$

For $0.1 \leq x_2 \leq 0.3\ m$

$$M_x = -176.2x_2 + 259.4(x_2 - 0.1)$$

At $X_2 = 0.1$,

$$M_x = -176.2 \text{ x } 0.1 = -17.62\ Nm$$

At $x_2 = 0.3$,

$$M_x = -176.2 \text{ x } 0.3 + 259.4(0.3 + 0.1) = -0.98\ N.m$$

$$For \quad 0.3 \leq x_3 \leq 0.35\, m$$

$$M_x = -176.2x_3 + 259(x_3 - 0.1) - 63.6(x_3 - 0.3)$$

$$At \quad x_3 = 0.3,$$

$$M_x = -176.2 \times 0.3 + 259.4(0.3 - 0.1) = -0.98\, N.m$$

$$At \quad x_3 = 0.35,$$

$$M_x = -176.2 \times 0.35 + 259.4(0.35 - 0.1) - 63.6(0.35 - 0.3) = 0$$

The vertical component will give the same values since the values of their components are the same, hence the values of the resultant bending moment and shearing forces will remain unchanged.

The shearing force and bending moment diagrams are shown in Figure 4.4.

The maximum bending moment for the shaft from Figure 4.5 is

$$M_{bmax} = \sqrt{(17.62)^2 + (17.62)^2} = 24.918 N.m$$

The torque, $M_t$ transmitted by the shaft is computed from the relation [4]

$$M_t = (T_1 - T_2)r_p = (176.8 - 72.385) \times \frac{180 \times 10^{-3}}{2} = 9.39735\, N.m$$

Adopting carbon steel 0.3C 080M30 with the following material properties: $S_y$ =385MPa; $S_{ut}$ = 550MPa and using the ASME code for transmission shafting for steel purchased under definite specification we obtain

$$S_s = 0.35 \text{ x } S_y = 0.35 \text{ x } 385 = 134MPa$$

$$= 0.18 \text{ x } S_{ut} = 0.18 \text{ x } 550 = 99MPa$$

Since $S_s$ = 99MPa is lower, it is adopted for the design. Also, since there is a keyway, the code further states that $S_s$ should be reduced by 25%.

$$\therefore\ S_s = 99 \text{ x } 0.75 = 74MPa$$

The shaft diameter is given by[4]

$$d_{sh}^3 = \frac{16}{\pi S_s}[(k_b M_b)^2 + (k_t M_t)^2]^{\frac{1}{2}}$$

where $k_b$ = combined shock and fatigue factor applied to bending moment = 1.5 for suddenly applied loads[4].

$k_t$ = combined shock and fatigue factor applied to torsional moment = 1.5 for suddenly applied loads.

Hence

$$d_{sh} = [\frac{16}{\pi \text{ x } 74 \text{ x } 10^6}[(1.5 \text{ x } 24.918)^2 + (1.5 \text{ x } 9.3973)^2]^{\frac{1}{2}}]^{\frac{1}{3}}$$

$$\therefore\ d_{sh} = 0.01401\ m = 14.01\ mm$$

From tables[3] of standard shaft diameters, we select a

$\phi$20 mm shaft for the purpose of this design.

(v) Design Of Shaft For Torsional Rigidity:

The torsional rigidity of a shaft is given by [4]

$$\theta = \frac{584\, M_t\, L}{G d^4}$$

where $M_t$ = 9.39735 N.m; L = 0.35 m; G = 80 x $10^9$ N/m$^2$. Hence,

$$\therefore = \frac{584 \times 9.39735 \times 0.350}{80 \times 10^9 \times (20 \times 10^{-3})^4} = 0.15^\circ \text{ twist.}$$

or

0.428 °/m twist.

This is far less than the allowable quoted for line shafting i.e. 3.0°/m twist[4]. It follows that the diameter is safe.

(vi) Determination Of Power Transmitted By The Belt:

The power transmitted by the belt at the design speed of 5.13 m/s is computed from equation (3.41) as

$$H_E = (T_1 - T_2)V$$

where V = 5.13 m/s

$T_1$ = 178.80 N

$T_2$ = 72.2385 N

Hence the power transmitted is

$$H_E = (178.80 - 72.2385) \times 5.13 = 0.547 \text{ kW}$$

Since this is greater than the power requirements of the fan, design is safe.

(vii) Determination Of Required Belt Length:

The required belt pitch length is given by[3]

$$L = 2C_k + \frac{\pi(D + d)}{2} + \frac{(D - d)^2}{4C_k}$$

$$\therefore L = 2 \text{ x } 400 + \frac{\pi(180 + 70)}{2} + \frac{180 - 70)^2}{4 \text{ x } 400} = 1200.26 \ mm$$

From tables[3], the nearest standard belt length = 1204 mm.

Consequently, the exact centre distance corresponding to the standard pitch length of 1204 mm (nominal inside length;

1168 mm) is found thus:

$$C = (A + \sqrt{A^2 + B}; \quad A = \frac{L}{4} - \frac{\pi(D + d)}{8}; \quad B = \frac{(D - d)^2}{8}$$

$$A = \frac{1204}{4} - \frac{\pi(180 + 70)}{8} = 202.8252296$$

$$B = (180 - 70)^2/8 = 1512.5$$

$$C = 202.8252296 + \sqrt{(202.8252296)^2 + 1512.5} = 409.35\ mm$$

Since the source of power is not to be attached to the machine frame, the value of C = 409.35 will not pose any problem to the design. Hence a V-belt designated as A1168/46, IS: 2494 is recommended for this design.

## 4.4 DESIGN/SELECTION OF SHAFT KEY

For the purpose of this design, a Gib-head key shall be used.

Allowing for shaft shoulder, the pulley is to be keyed to a 16.50 mm diameter portion of the shaft. The hub diameter of the pulley = 29 mm; hub length = 25 mm. From tables[3], the diameter (16.50 mm) falls between 12.00 - 17.00, its key and keyway details are as shown in Figure 4.11.

## 4.5.0 BEARING SELECTION/ANALYSIS

For the purpose of this project, single row deep groove ball bearings are adopted.

The life of a bearing depends on such factors as the load, which it carries, number of revolutions per minute which it is operated among other factors.

The rating life ($L_H$) for ball bearings[4] is given by

$$L_{10} = (\frac{C_r}{P})^k$$

where k = 3 for ball bearings.

$C_r$ = the dynamic load rating or bearing capacity.

P = the equivalent or radial load.

In bearing selection, machines for series of short duration or intermittent operation, where service interruptions are of minor importance, hand tools,
hand-driven machines in general, farm machinery e.t.c. Desired life in hours is taken as between 400 - 8000 hours. (Kent's Mechanical Engineer's Handbook). Adopting 8000 hours for the purpose of this design, therefore, desired life in revolution for 90% of the bearing to show its first element of fatigue ($L_{10}$) is computed as

$$L_{10} = 8000 \times 60 \times n$$

$$\therefore \; L_{10} = 8000 \times 60 \times 544.44 = 261.33 \times 10^6 \quad revolutions$$

Also, from the life equation above, we obtain

$$C_r = L_{10}^{\frac{1}{3}} \times P$$

$$\therefore \; C_r = (261.33)^{\frac{1}{3}} \times 259.4 = 1.658\ kN$$

where P = 259.4 N is the maximum shaft reaction.

On proper investigation of the SKF bearing tables[3], reveals the details given below as satisfactory for the loading conditions.

TABLE 4.3 BEARING SELECTION TABLE

| No. | Bore (mm) | C (kN) | $C_o$ (kN) | B (mm) | O.D (mm) | Q (min) |
|---|---|---|---|---|---|---|
| 6203 | 170 | 7.35 | 4.50 | 12 | 40 | 17 |

KEY: C: Basic load rating under dynamic condition

$C_o$: Basic load rating under static condition

B: Bearing width

OD: Bearing outside diameter

Q: Minimum shaft shoulder allowed.[14]

Since the dynamic and static load; C and $C_o$ are greater than the calculated value of $C_r$ (1.685kN) for the shaft. Therefore the selected bearing chosen is satisfactory.

## 4.60 HOPPER DESIGN CALCULATIONS

### A. HOPPER SIZE DETERMINATION

From graph (Figure 4.10); the slope n = 1.01 and the intercept = -0.31, consequently

$\log_{10} = -.031$, k = 0.489.

Therefore, the law relating the flow with the opening diameter is

$$Q = 0.489\, B^{1.01}$$

For the purpose of this design, we adopt a hopper discharge capacity, Q of 5 kg/min (300 kg/hr).

From the above equation we obtain

$$\frac{Q}{0.489} = B^{1.01}$$

Taking the Naperian log of both sides we obtain

$$\ln \frac{Q}{0.489} = \ln B^{1.01}$$

Substituting values we obtain

$$1.01 \ln B = \ln \frac{5}{0.489}$$

$$\ln B = \frac{2.32}{1.01} = 2.30$$

$$\therefore B = e^{2.30} = 9.99\ cm = 9.99 \times 10^{-2}\ m$$

where B is the diameter os a circular orifice. Therefore, since we are designing for a pyramidal frustum, the equivalent dimensions of the hopper base are determined from the relation

$$\frac{\pi B^2}{4} = a_2 \times b_2$$

where $a_2$ and $b_2$ are dimensions indicated as shown in

Figure 4.8. Substituting values we obtain

$$\frac{\pi \times (9.99)^2}{4} = a_2 \times b_2$$

$$a_2 \times b_2 = 78.4\ cm^2$$

Adopting $b_2 = 6.00$ cm, we obtain

$$a = \frac{78.4}{6} = 13.00\ cm$$

The elevation of the pyramidal frustum is shown in

Figure 4.8.

The angle of repose, $\phi_r$, quoted for rice, soyabean, millet and groundnut falls within the range of 30-45°

(German Gurfinkel, 1979). For the purpose of this project we adopt $\phi_r$ = 44°. Thus, it follows that the inclination of the hopper to the horizontal

_ = (10° + 44°) = 54°. (Rickey et al, 1961)

From Figure 4.8 we obtain

$$\tan 54° = \frac{H_o}{FA}$$

$$But\ FA = \frac{30 - 13}{2} = 8.45\ cm$$

$$\therefore\ H_o = BC \tan 54° = 8.5 \tan 54° = 11.699\ cm$$

For the purpose of this design we adopt $H_o$ = 14.00 cm.

Also,

$$AB = \frac{8.5}{\cos} 54° = 14.46\ cm$$

For the purpose of this design, AB = 15 cm will be used.

To check for arcing and funnelling conditions, equations (3.64) and (3.65) are used.

Hence

$$\gamma = \frac{B}{2} = \frac{9.99}{2} = 4.995\ cm$$

$$H_o = 14\ cm; \quad Q_r = 44°; \quad \beta_k = 90° - 54° = 36°$$

Angle of internal friction (German Gurfinkel, 1979)

$$\phi_i = 0.8\phi_r = 0.8 \times 44° = 35.20°$$

Also, for grain against steel surface (German Gurfinkel, 1978)

$$\mu' = 0.26 - 0.42$$

Adopting $\mu' = 0.34$ for the design we obtain

$$\frac{\gamma}{H_o} = \frac{4.995}{14} = 0.35678$$

$$\mu' \tan\phi_r = 0.34 \tan 44° = 0.328$$

Hence, the condition

$$\frac{\gamma}{H_o} > \mu \tan\phi_r$$

is satisfied.

Similarly,

$$\mu' \tan(\phi_i + \beta_k) - \tan\beta_k$$

$$= 0.34\tan(35.2° + 36°) - \tan 36°$$

$$= 0.99 - 0.7265 = 0.263$$

Also the condition

$$\frac{Y}{H_o} > \mu' \tan(\phi i + \beta_k) - \tan\beta_k$$

is satisfied. It follows that the condition of funnelling and arcing have been satisfied.

B. HOPPER FORCE ANALYSIS

For ease of analysis, Y is taken as acting $\_H_o$ from the top of hopper (Mohsenin, 1978).

$$\therefore Y = [\frac{2}{3} x\ 14] = 9.33\ cm$$

The height, $h_s$ is computed from equation (3.70) as

$$h_s = \frac{D}{2}\tan\phi_r$$

where D is the equivalent diameter of a circular opening corresponding to inlet $a_1$ and $b_1$ which is calculated as

$$D = \left[\frac{4 \times a_1 \times b_1}{\pi}\right]^{\frac{1}{2}}$$

$$taking \quad a_1 = 30.00\ cm; \quad b_1 = 26.34\ cm$$

$$D = \left[\frac{4 \times 30 \times 26.34}{\pi}\right]^{\frac{1}{2}} = 31.72\ cm$$

$$\therefore\ h_s = \frac{31.72}{2}\tan 44° = 15.32\ cm$$

The bulk density of grain such as wheat, corn, soya beans, barley, peas, beans, oats, rice, rye e.t.c. is between 704.537-993.147 kg/m$^3$ (German Gurfinkel, 1979). Adopting a bulk density of $\gamma_d$ =993.147 kg/m$^3$ for the purpose of this design we then find the maximum pressure due to grain as follows.

The hydraulic radius is obtained from equation (3.35) with $a_2$ = 13 cm and $b_2$ = 6 cm as

$$a' = R = \frac{2 \times 13 \times 6}{13 + 6} = 8.21\ cm = 8.21 \times 10^{-2} m$$

Remibert's characteristic abscissa, c is calculated from equation (3.54) as

$$c = \frac{a'}{\pi\mu' k} - \frac{h_s}{3}$$

$$where \ \mu' = 0.34; \quad h_s = 15.32 \ cm$$

$$k = \frac{1 - \sin\phi_r}{1 + \sin\phi_r}; \quad \phi_r = 44°$$

$$\therefore k = \frac{1 - \sin 44°}{1 + \sin 44°} = 0.18$$

Hence,

$$c = \frac{8.21 \times 10^{-2}}{\pi \times 0.34 \times 0.18} - \frac{15.32 \times 10^{-2}}{3} = 37.6 \times 10^{-2} \ m$$

From equation (3.55) we obtain

$$P_{max} = \frac{\gamma_d a'}{4\mu'} = \frac{993.147 \times 8.21 \times 10^{-2}}{4 \times 0.34} = 59.95 kg / m^2$$

The lateral load is calculated from equation (3.50) as

$$P = P_{max}[1 - (\frac{Y}{c} + 1)^{-2}]$$

$$= 59.95[1 - (\frac{9.33 \times 10^{-2}}{37.6 \times 10^{-2}} + 1)^{-2}]$$

$$\therefore P = 21.47 kg/m^2$$

The vertical pressure, q is calculated from equation (3.47) as

$$q = \gamma_d[\frac{Y}{(\frac{Y}{c} + 1)} + \frac{h_s}{3}]$$

With $\gamma_d$ = 993.147kg/m$^3$ ; Y = 9.33 x 10$^{-2}$m; c = 37.6 x 10$^{-2}$m and $h_s$ = 15.32 x 10$^{-2}$m we obtain

$$q = 993.147[\frac{9.33 \times 10^{-2}}{(\frac{9.33 \times 10^{-2}}{37.6 \times 10^{-2}} + 1)} + \frac{15.32 \times 10^{-2}}{3}]$$

$$= 124.89 kg/m^2$$

The vertical frictional force per unit width of wall, $V_f$ is calculated from equation (3.51) as

$$V_f = (\gamma_d Y - q)R$$

With $\gamma_d$ = 993.147 kg/m$^3$; Y = 9.33 x 10$^{-2}$m; q = 124.89 kg/m$^2$ and R = 8.21 x 10$^{-2}$m we obtain

$$V_f = (993.147 \times 9.33 \times 10^{-2} - 124.89) \times 8.21 \times 10^{-2}$$

$$= -2.646 kg/m \ \ of \ \ hopper \ \ width$$

This implies that the friction load acts against the direction of flow during emptying.

The lateral and vertical pressure are acting on an inclined surface as shown in Figure 4.9(a) and (b).

The resultant load, R is obtained considering the sum of P and q vertically as

$$R = [(q')^2 + (p')^2]^{\frac{1}{2}}$$

$$with \ \ q' = q \sin\alpha = 124.895 \sin 54° = 101.038 kg/m^2$$

$$p' = p\cos\alpha = 21.47 \cos 54° = 12.62 kg/m^2$$

$$\therefore R = [(101.038)^2 + (12.62)^2]^{\frac{1}{2}} = 101.82 kg/m^2 = 998.88 N/m^2$$

The hopper material thickness can be estimated from the equation of the wall cylinder subjected to internal pressure, R and diameter, D[12] as

$$t = \frac{DR}{2 S_{all}}$$

where $s_{all}$ = allowable tensile stress of material. For the purpose of this project, we use the allowable tensile stress of mild steel as 78.75MPa.[12]

D = B = 9.99 cm is the equivalent diameter of the opening.

R = 998.88 N/m$^2$ is the internal pressure of stored grain.

Hence,

$$t = \frac{9.99 \times 10^{-2} \times 998.88}{2 \times 78.75 \times 10^{6}} = 6.34 \times 10^{-7} m = 6.34 \times 10^{-4} mm$$

For the purpose of this project we adopt a gauge 18 (1.2 mm) thick mild steel to allow for buckling of hopper which may occur due to the frictional effect of the grain on the hopper wall and possible expansion of the material (mild steel) at its base.

C. DESIGN FOR CRITICAL ORIFICE OPENING

From equation (3.58), the critical diameter of opening (discharge) is determined from the relation

$$B_o \geq \frac{\sigma_c}{\gamma}$$

Where $\sigma_c$ is calculated from equation (3.60) as

$$\sigma_c = \gamma_d Y \tan \phi_r$$

$$= 993.147 \times 9.33 \times 10^{-2} \tan^2 (45° - \frac{44}{2}°)$$

$$\therefore \sigma_c = 16.695 kg/m^2$$

where $\gamma_d$ = 993.147 kg/m$^3$; Y = 9.33 x 10$^{-2}$m; $\phi_r$ = 44°

$$\therefore \frac{\sigma_c}{\gamma_d} = \frac{16.695}{993.147} = 0.0168 = 1.68\ cm$$

$$Sice \quad B_o > \frac{\sigma_c}{\gamma} \quad i.e \quad 60\ cm > 1.68\ cm$$

the critical flow condition is satisfied.

## D. ESTIMATION OF HOPPER CAPACITY

The volume of the pyramidal frustum, V can be calculated from the expression[13] as

$$V = \frac{H_o}{4}(a_1 + a_2)(b_1 + b_2)$$

With $H_o$ = 14 cm = 14 x $10^{-2}$ m; $a_1$ = 30 cm = 30 x $10^{-2}$ m;

$a_2$ = 13 cm = 13 x $10^{-2}$ m; $b_1$ 26.34 cm = 26.34 x $10^{-2}$ m; and

$b_2$ =6 cm = 6 x $10^{-2}$ m, we obtain

$$V = \frac{14\ x\ 10^{-2}}{4}[(30\ x\ 10^{-2} + 13\ x\ 10^{-2})(26.34\ x\ 10^{-2} + 6\ x\ 10^{-2})]$$

$$= 4.867\ x\ 10^{-3}\ m^3$$

# CHAPTER FIVE

## 5.0 TESTING OF THE MODIFIED WINNOWER

### 5.1.0 TESTING

The testing of the machine was done in order to determine the effect of air blast from the fan on the falling grains/chaff mixture. The machine was tested for its ability to separate chaff/husk from grain/seed.

The cleaning efficiency of the machine is given by

$$E_a = \frac{W_T - W_o}{W_c} x\ 100\% \qquad (5.1)$$

where $W_T$ = total weight of chaff/grain mixture (kg)

$W_o$ = weight of clean grain collected (kg)

$W_c$ = weight of chaff introduced (kg)

### 5.1.1 TESTING PROCEDURE

Two major tests were carried out on the machine, viz

(a) the no-load test

(b) the load test

(a) The No-Load Test:

Under this test, the machine was connected to an electric motor with power rating of 3.7 kW and maximum speed of

1440 rpm via a V-belt drive and was allowed to run for about 15 minutes. The no-load test was conducted to asses the performance of the whole machine.

(b) The Load Test:

The test was carried out using rice, millet and groundnut as test materials. In each case, test mixture of known weight of clean whole/broken grains and chaff were prepared and fed into the hopper with the control plate closed.

The machine was set into operation with the aid of the electric motor which drive the fan through a V-belt drive. Grain-chaff mixture are then allowed to fall into the air stream produced by the fan for separation. There are two collection points of the test materials on the machine.

(i) the clean grain outlet chute provided and

(ii) the chaff outlet opposite the fan discharge duct.

### 5.1.2 RESULTS AND CALCULATIONS

The no-load test demonstrated that the machine operates smoothly with all its rotating parts working as expected. No adverse effect was observed with continuous operation of the machine.

The results of the test performed under the load test are tabulated and the cleaning efficiencies computed using equation (5.1) as shown in the table below.

TABLE 5.1. TEST RESULTS

| S/N | Crop | $W_c$ (kg) | $W_g$ (kg) | $W_T$ (kg) | $W_o$ (kg) | A | B | C $(W_T/T)$ | D $(W_o/T)$ | E | F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Millet | 0.25 | 0.20 | 0.45 | 0.25 | 1:1.25 | 42 | 38.57 | 21.43 | 5 (0.01) | 80 |
| 2 | Rice | 0.50 | 0.25 | 0.75 | 0.27 | 1:2.00 | 37 | 72.97 | 21.19 | 6 (0.015) | 96 |
| 3 | G/nut | 0.10 | 0.15 | 0.25 | 0.16 | 1:1.50 | 18 | 50.00 | 32.00 | 6.60 (0.01) | 90 |

KEY: A - Grain to chaff ratio
B - Time, T (sec.)
C - Feed rate (kg/hr)
D - Output, (kg/hr)
E - Percentage grain loss (kg)
F - Cleaning efficiency, (%)
$W_c$ - Weight of chaff introduced (kg)
$W_g$ - Weight of grain introduced (kg)
$W_T$ - Total weight of chaff/grain mixture (kg)
$W_o$ - Weight of clean grain collected from the grain outlet chute (kg)

CHAPTER SIX

# 6.0 MATERIAL SELECTION AND COSTING

## 6.1 MATERIAL SELECTION

While selecting suitable materials, the requirements of the relevant parts relating to their function, stress conditions, and service life have been taken into consideration in this design work. Also taken into consideration is the method of welding, finishing of the parts, the cost of production and also the availability and procurement of the materials.

The power transmission shaft and pulley are to be made from carbon steel. Other members like the machine frame is to be made from an L-section angle iron to reduce the overall weight of the machine. The fan is to be made from mild steel sheet so that the effect of friction in the fan casing and the blades would be significantly reduced.

The need to maintain low level of friction in the power transmission drive prompted the choice of deep groove ball bearings for the fan shaft support.

All standard components such as keys, bolts and nuts, belts, bearings e.t.c. are to be purchased locally from the market.

## 6.2 MATERIAL COSTING

The tables below gives a summary of material costings.

TABLE 6.1 MATERIAL COSTING

| S/N | ITEM DESCRIPTION | QTY. | UNIT COST (₦) | TOTAL COST (₦) |
|---|---|---|---|---|
| 1 | Angle Iron (1½" x 1½") | 3 | 540.00 | 1620.00 |
| 2 | Mild steel sheet gauge 16 | ¾ | 2400.00 | 1800.00 |
| 3 | Bearings (6203) | 2 | 150.00 | 300.00 |

| | | | | |
|---|---|---|---|---|
| 4 | Electrodes G12 | 100 | 5.00 | 500.00 |
| 5 | Bolts & Nuts<br>(a) M16x1 Foundation bolts and nuts<br>(b) M13x1 Bearing housing retaining bolts and nuts<br>(c) M4x1 Guard plate suspension bolts and nuts | 8<br>4<br>4 | 20.00<br>15.00<br>10.00 | 160.00<br>60.00<br>40.00 |
| 6 | V-belt: A46 | 1 | 300.00 | 300.00 |
| 7 | $\phi$20mm shaft (500mm length) | 1 | 500.00 | 500.00 |
| 8 | Base plate (100x100x3mm) | 4 | 200.00 | 800.00 |
| 9 | Pulley ($\phi$170mm) | 1 | 300.00 | 300.00 |
| 10 | Oil Paint | 450ml | 200.00 | 200.00 |
| 11 | Thinner | 150ml | 100.00 | 100.00 |
| | Sub Total | | | 6680.00 |

Labour cost[15] is given as 25% of the total cost of materials:

Labour cost = 0.25 x 6680.00 =₦ 1670.00

Also, over-head cost [15] is given as 60% of labour cost:

Over-head cost = 0.60 x 1670.00 =₦ 1002.00

Hence.

Total cost of machine = (6680 + 1670 + 1002) =₦ 9,352.00

It is pertinent to mention here that the prices quoted in the table above are valid as at the time their costing were made and are subject to changes depending on market trend and inflationary rate.

# CHAPTER SEVEN

## 7.0 CONCLUSION AND RECOMMENDATION

### 7.1 CONCLUSION

The modification of an already existing multipurpose winnower in the Department of Agricultural Engineering is presented in this report. It is re-designed for cleaning soya bean, millet, groundnut and rice.

The method of throttling the air intake into the fan has been adopted for this modification work considering the initial cost of purchasing drive pulleys of different sizes and installation time.

The machine cleaning efficiencies was evaluated as 96%, 80%, and 90% for rice, millet, and groundnut respectively at a speed of 1084 rpm of the fan.

To ensure optimum machine utilization, users are advised to maintain an input speed of $\leq$ 1084 rpm while maintaining the direction of rotation of the fan impeller i.e. clockwise.

Also, to ensure a prolonged life of the threaded screws, users should limit the unscrewing of the casing assembly screws to lubrication and inspection periods only.

### 7.2 RECOMMENDATIONS

The following recommendations are given for further works:

(i) The possibility of including reciprocating sieves should be considered to further increase the capacity and cleaning efficiency of the winnower.

(ii) The flow control slide plate should be re-adjusted such that it closes length wise rather than across.

The designer welcomes any useful criticism that would lead to improvement in the cleaning efficiency, through-put as well as cost reduction of this design work.

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## APPENDIX A

DETERMINATION OF IMPELLER FILLET WELD THICKNESS

IMPELLER HUB:

Shaft bending moment, $M_b$ = 24.918 N.m

Shaft torsional moment, T = 9.39735 N.m

Adopted factor of safety, n = 2.

(i) Computation of Sectional Modulus Of Weld Treated as a Line: This is obtained as[4]

$$Z_w = \frac{1}{4}\pi d_h^2 = \frac{\pi x (29 x 10^{-3})^2}{4} = 6.605 x 10^{-4} m^2$$

where $d_h$ = 29 mm is the hub diameter.

The highest loading occurs at the top and bottom of shaft where

$$S_b = \frac{nM_b}{Z_w} = \frac{2 x 24.918}{6.605 x 10^{-4}} = 0.075452 MN / m$$

(ii) Section Polar Moment of Inertia of Weld Treated as a Line: This is obtained as[18]

$$J_w = \frac{\pi d^3}{4} = \frac{\pi x (29 x 10^{-3})^2}{4} = 1.916 x 10^{-5} m^3$$

The highest loading occurs at the top and bottom of the shaft where

$$S_s = \frac{nTc}{J_w}$$

$$\text{where } c = \frac{d}{2} = \frac{29 \text{ x } 10^{-3}}{2} = 14.5 \text{ x } 10^{-3} m$$

$$\therefore S_s = \frac{2 \text{ x } 9.39735 \text{ x } 14.5 \text{ x } 10^{-3}}{1.916 \text{ x } 10^{-5}} = 0.0142 MN/m$$

(iii) Determination of Resultant Force on Weld: The resultant for per unit length on weld, S is obtained as

$$S = [(0.075452)^2 + (0.0142)^2]^{\frac{1}{2}} = 0.076777 MN/m$$

(iv) Determination of Weld Size, $w_s$: The weld size can be computed from the relation given by[4] as

$$w_s = \frac{S}{0.707 S_{all}}$$

$$\text{Also } S_{all} = 94 \text{ MPa}^{[4]}$$

$$\therefore w_s = \frac{0.076777 \text{ x } 10^6}{0.707 \text{ x } 94 \text{ x } 10^6} = 1.1553 \text{ x } 10^{-3} m \approx 1.2 \text{ mm}$$

The minimum weld size required is 1.20 mm. For the purpose of this design, a weld size $w_s$ = 4.00 mm will be used.

## APPENDIX B

FAN BLADES

The total length of fillet weld

$l_w = 2 \times 12 \times 47 \times 10^{-3} = 1.128$ m

Since the blades are to be welded on both sides and a length of 47 mm, then required weld length per blade

$$l_{wb} = \frac{1.128}{12} = 0.094\ m$$

Weld throat area, A is computed as[4]

$$A = 0.707\, w_s d$$

$$where \quad d = 0.094\ m$$

$$\therefore A = 0.707 \times w_s = 0.132956\, w_s \ m^2$$

Also, mass of blade

$$m_b = 0.0832\ kg \quad (section \quad 4.3.0)$$

Assuming that the centrifugal force acts at the mean blade radius, then

$$\frac{1}{4}(d_2 - d_1) = \frac{1}{4}(260 - 166) = 23.5 \times 10^{-3}\ m$$

Hence, the centrifugal force acting on the blade is computed from the relation[3,4,13]

$$C_f = m\omega^2 r$$

$$= 0.0832 \text{ x } [\frac{2\pi \text{ x } 544.44}{60}]^2 \text{ x } 23.5 \text{ x } 10^{-3}$$

$$\therefore \; C_f = 6.35548 \, N$$

Average weld stress,

$$S_{av} = \frac{6.35548}{0.132956 \, w_s} = \frac{47.8014}{w_s} \; N/m^2$$

Weld size, $w_s$ can be computed from the relation[4]

$$w_s = \frac{S}{0.707 \, S_{all}} = \frac{47.8014}{w_s \text{ x } 0.707 \text{ x } 94 \text{ x } 10^6}$$

$$w_s^2 = \frac{47.8014}{0.707 \text{ x } 94 \text{ x } 10^6}$$

$$\therefore \; w_s = 8.48 \text{ x } 10^{-4} = 0.85 \, mm$$

For the purpose of this design, a weld size $w_s$ = 4.00 mm will be used.

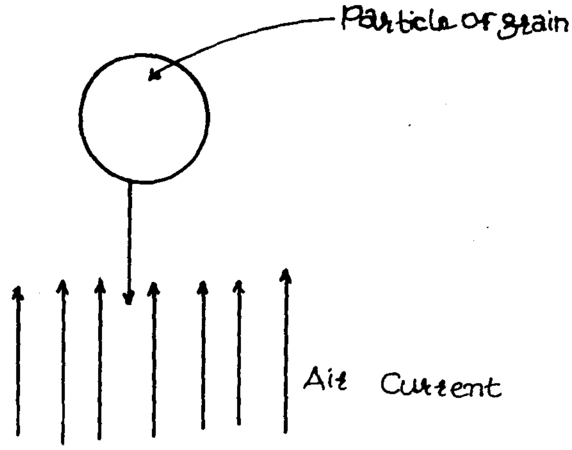



FIG. 3.1 Grain falling through air current

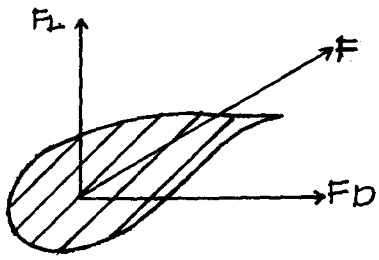
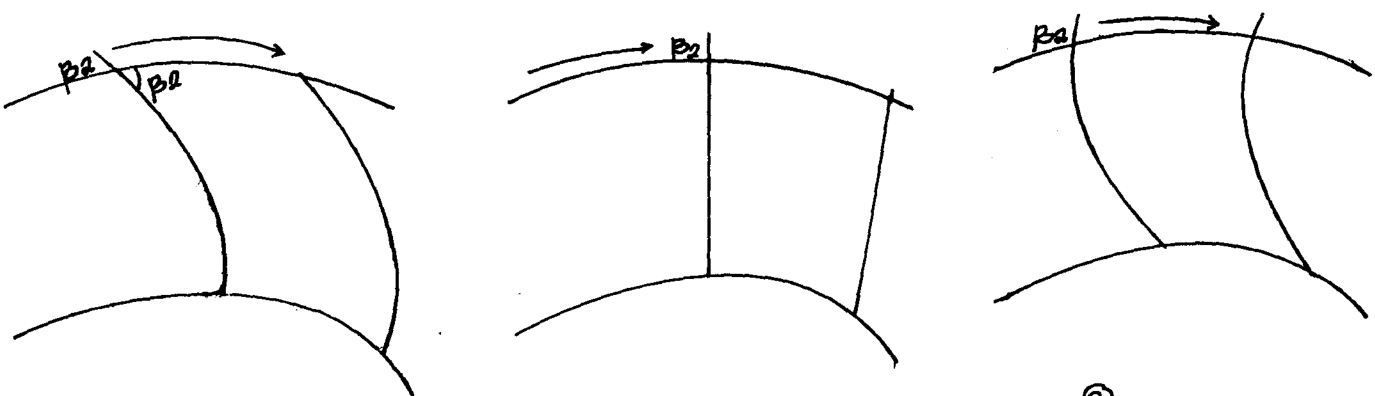



FIG. 3.2 Forces Acting on a particle under air current.




(a) $\beta_2 < 90°$

(b) $\beta_2 = 90°$
RADIAL BLADE FORM

(c) $\beta_2 > 90°$
FORWARD BLADE FORM

FIG. 3.3 CENTRIFUGAL FAN BLADE SHAPE

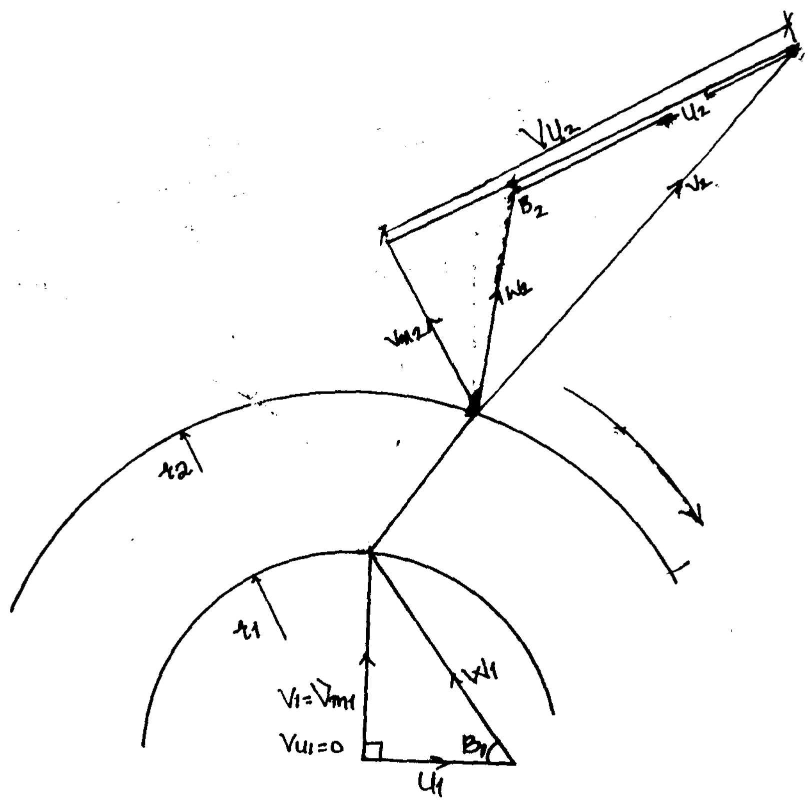



FIG. 3.4 Forward CURVED BLADE VELOCITY DIAGRAM AT INLET AND OUTLET

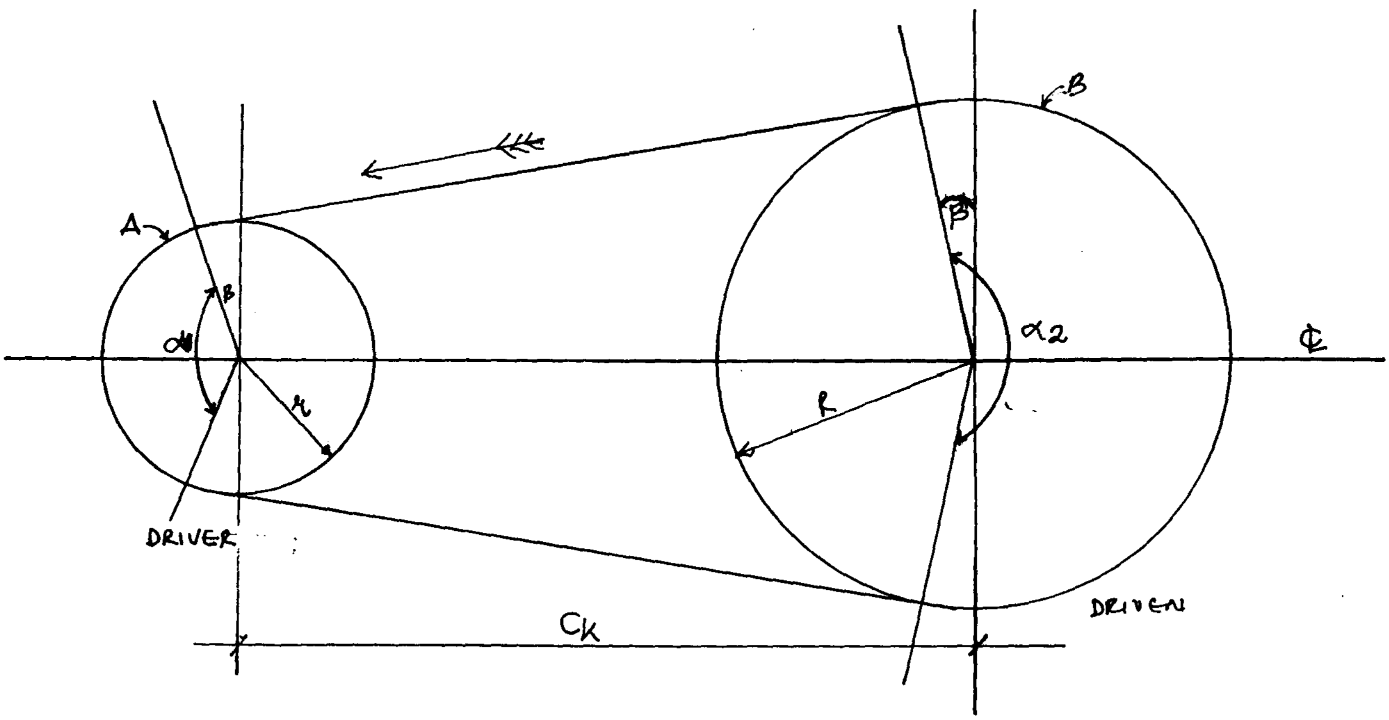



FIG: 3.5 Belt drive geometry

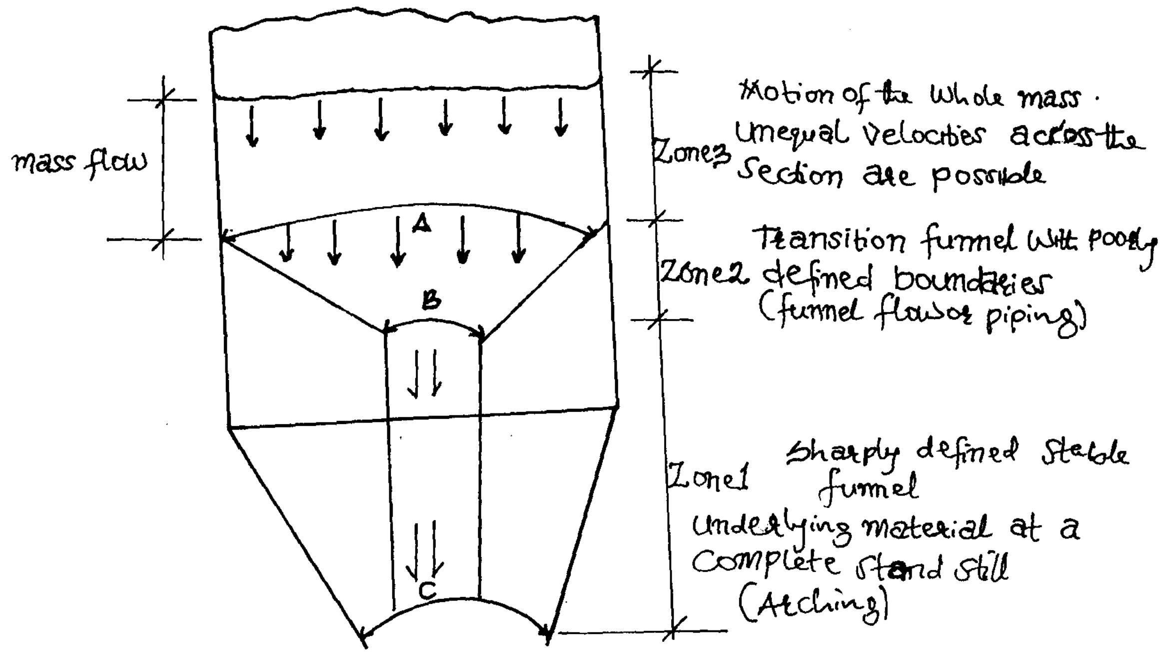



FIG 3.6 Flow pattern during empting of hopper/silo

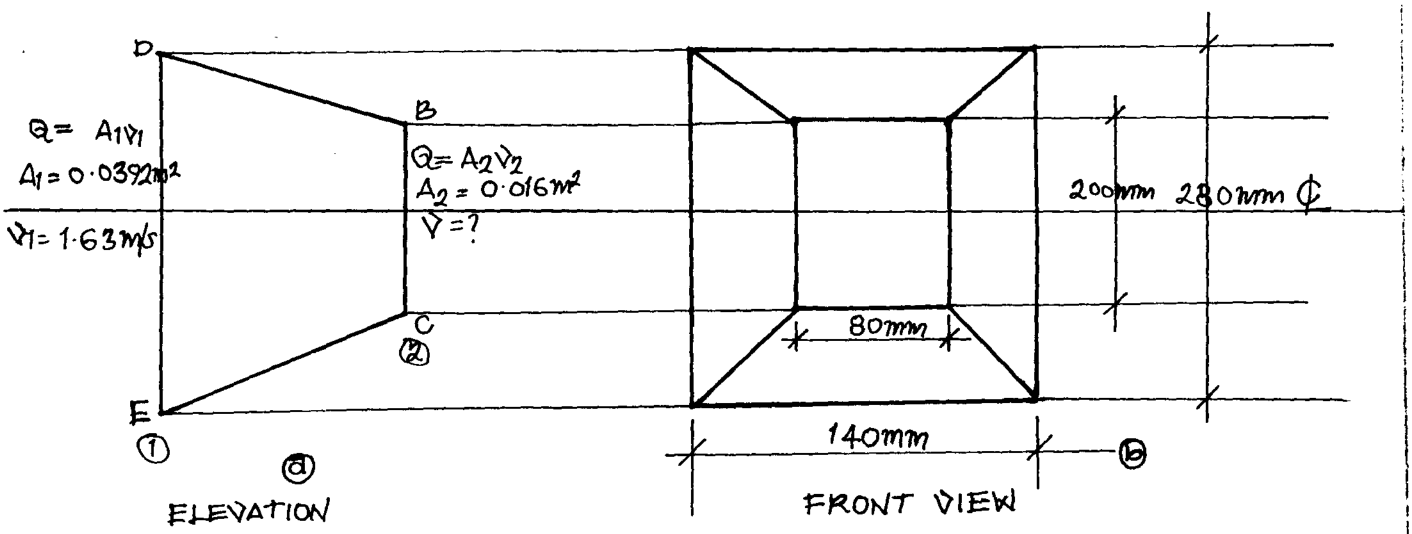



FIG. 4.1 FAN NOZZLE HEAD

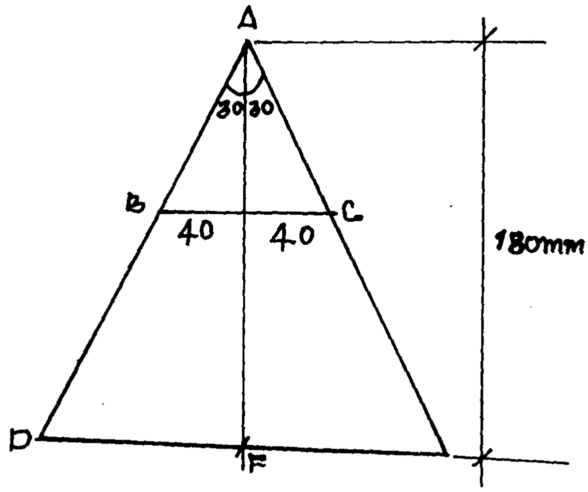



FIG 4.1 ©

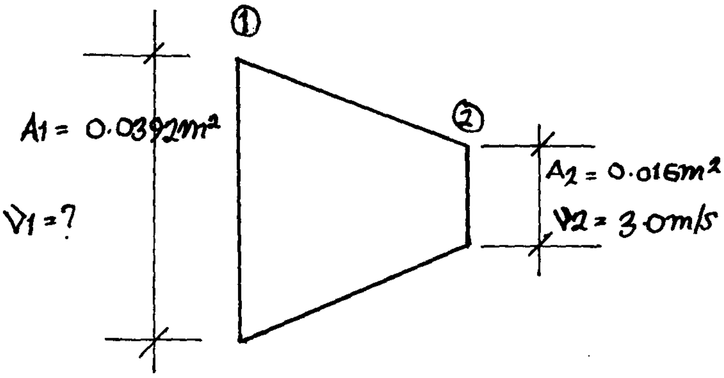



FIG. 4.1 (d)

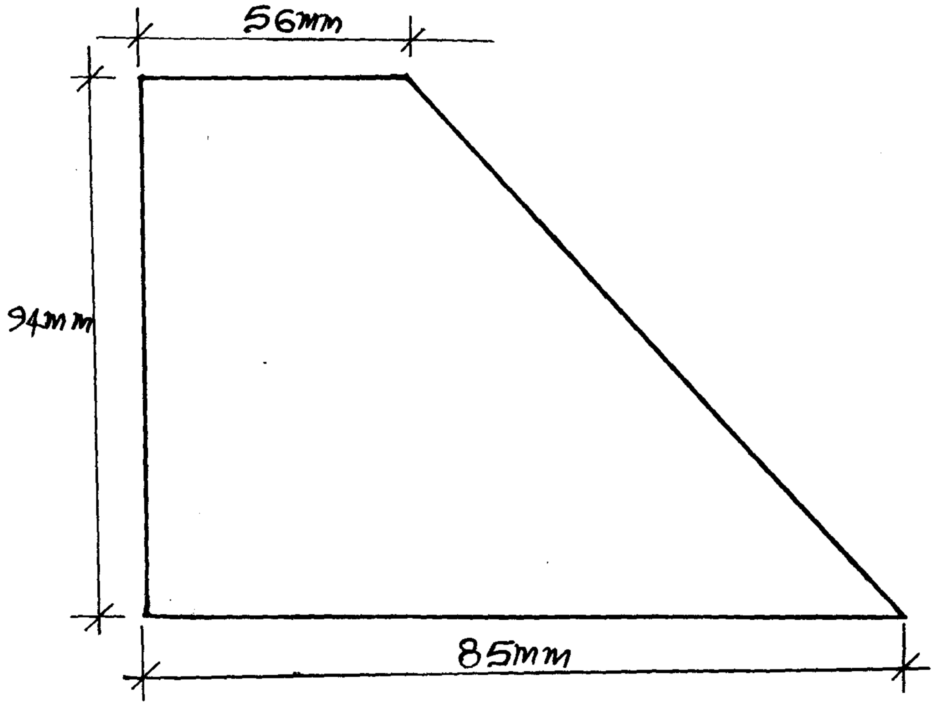



FIG: 4.2 BLADE PROFILE

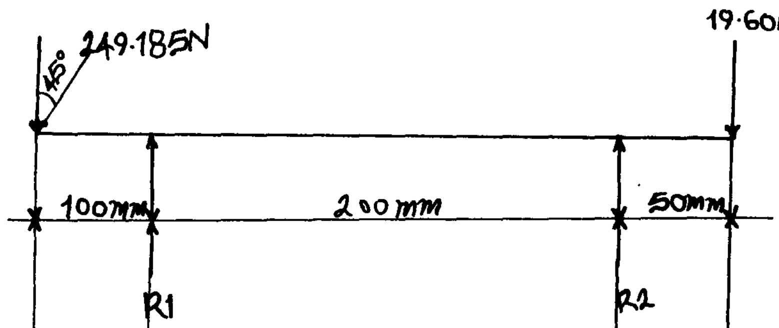



FIG. 4·3ⓐ Forces acting on fan Shaft

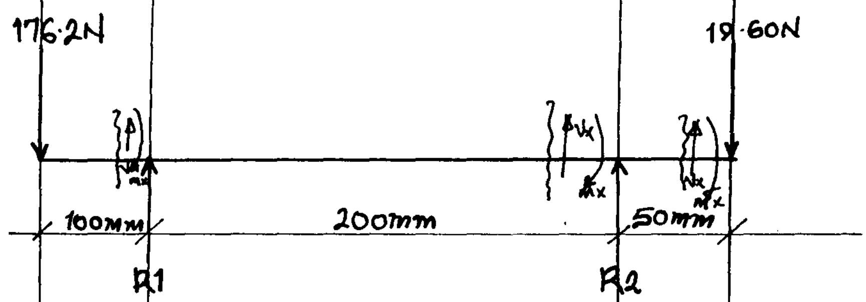



FIG. 4·3ⓑ Forces acting on shaft: Considering horizontal component of $(T_1+T_2)$

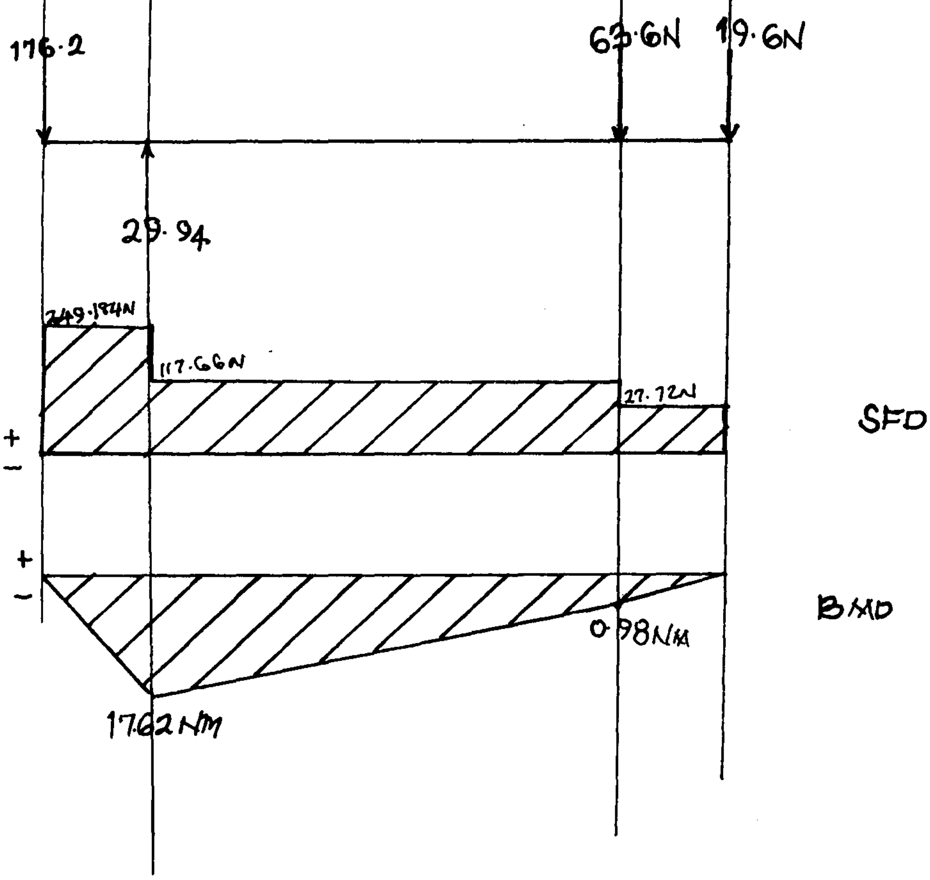



FIG. 4·4: Shearing forces and Bending moment diagrams Considering the Horizontal Component of $(T_1+T_2)$

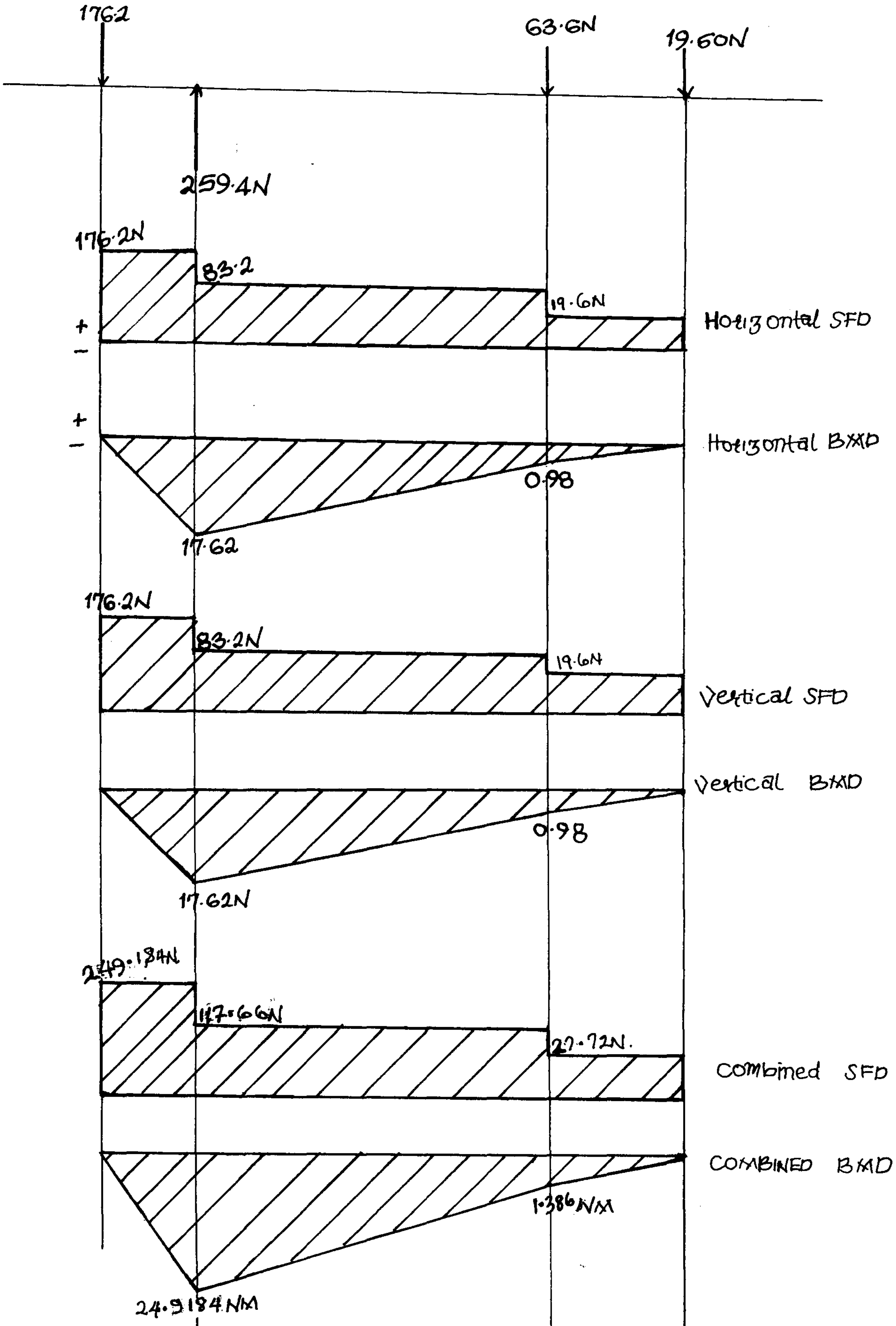



Figure 4·5:
COMBINED SHEARING AND BENDING MOMENT DIAGRAM FOR FAN SHAFT.

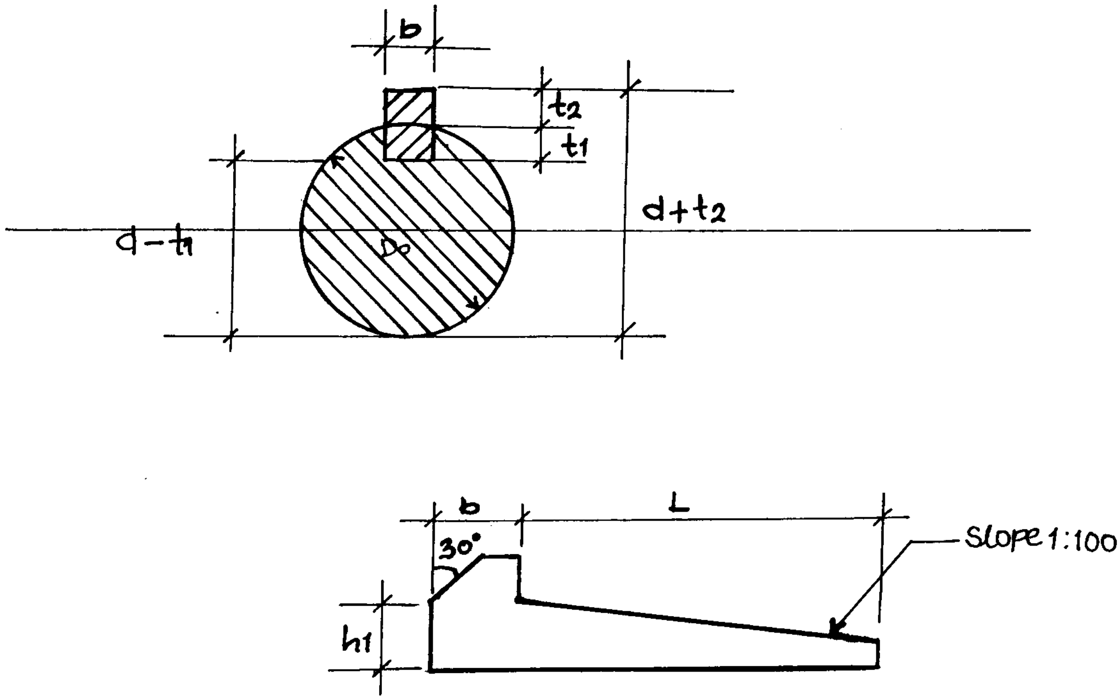



FIG: 4·6 DETAILS OF KEY AND KEYWAY

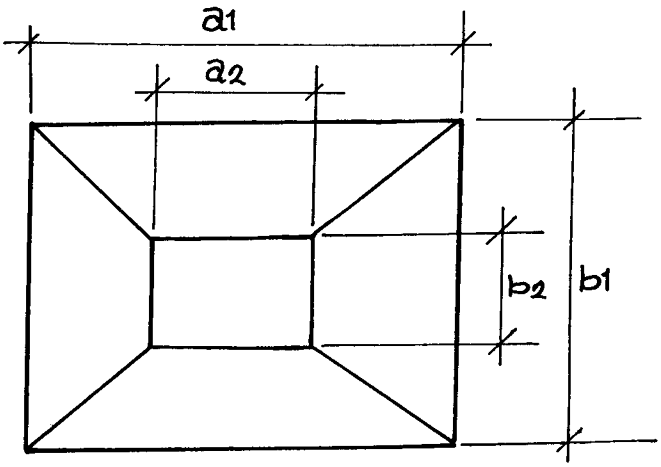
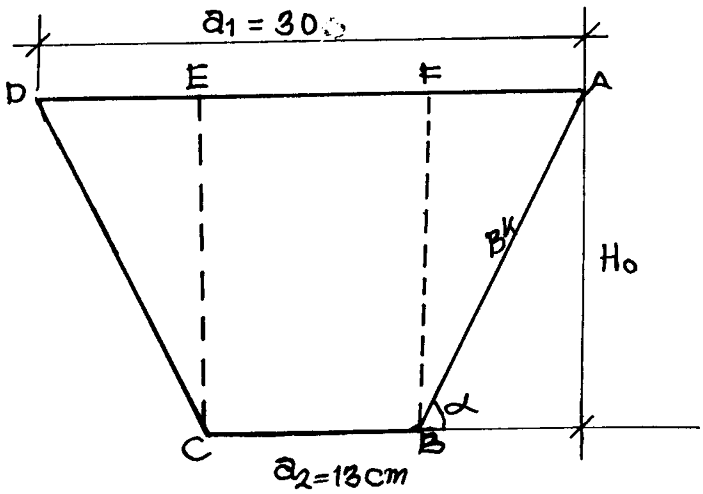



FIG. 4·7 PLAN VIEW OF HOPPER.




Adopting $a_1$ to be 300mm (30cm)

FIG. 4·8 ELEVATION OF HOPPER

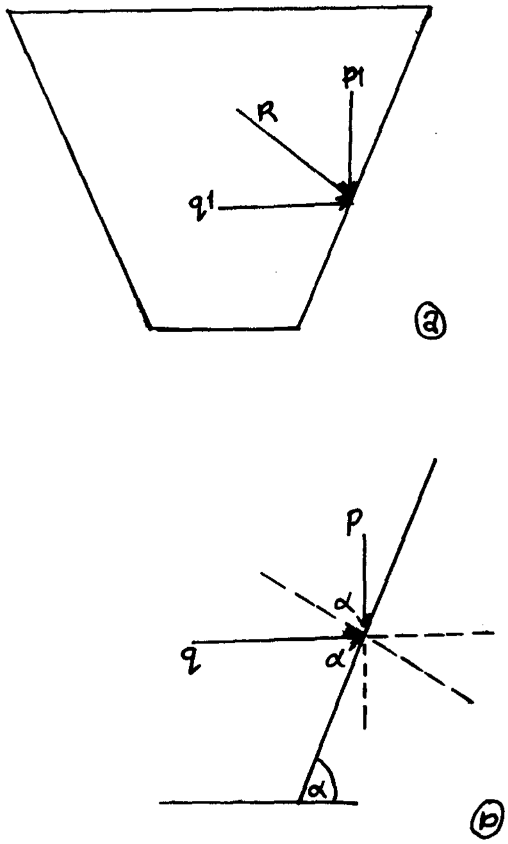



FIG. 4.9 FORCES ACTING ON HOPPER SIDES

TABLE: 3.1 VALUES USED IN DETERMINATION OF K AND n

| Q | 5 | 10 | 15 | 20 | 25 | 30 |
|---|---|---|---|---|---|---|
| Y = Log Q | 0.69 | 1.00 | 1.17 | 1.30 | 1.39 | 1.47 |
| B | 10 | 20 | 30 | 40 | 50 | 60 |
| X = Log B | 1.00 | 1.30 | 1.47 | 1.60 | 1.69 | 1.77 |

* SOURCE : CHUKWU.O (1987): DESIGN OF A HEAT GENERATOR FURNANCE

SEE FIG: 4.10 FOR THE GRAPH OF LOGQ AGAINST LOGB

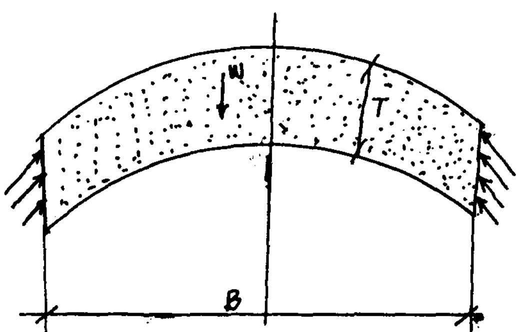



FIG : 3.7 FREE-BODY DIAGRAM OF A MASS OF GRANULAR MATERIAL FORMING AN ARCH

$$\text{Slope} = \frac{BC}{AC}$$

① $\text{Slope} = \dfrac{1.198 - 0.3}{1.5 - 0.61}$

$= 1.01$

② $\text{Log } k = -0.31$

$k = 0.489$

∴ The Law connecting the flow rate with the hopper opening is

$Q = 0.489B^{1.01}$

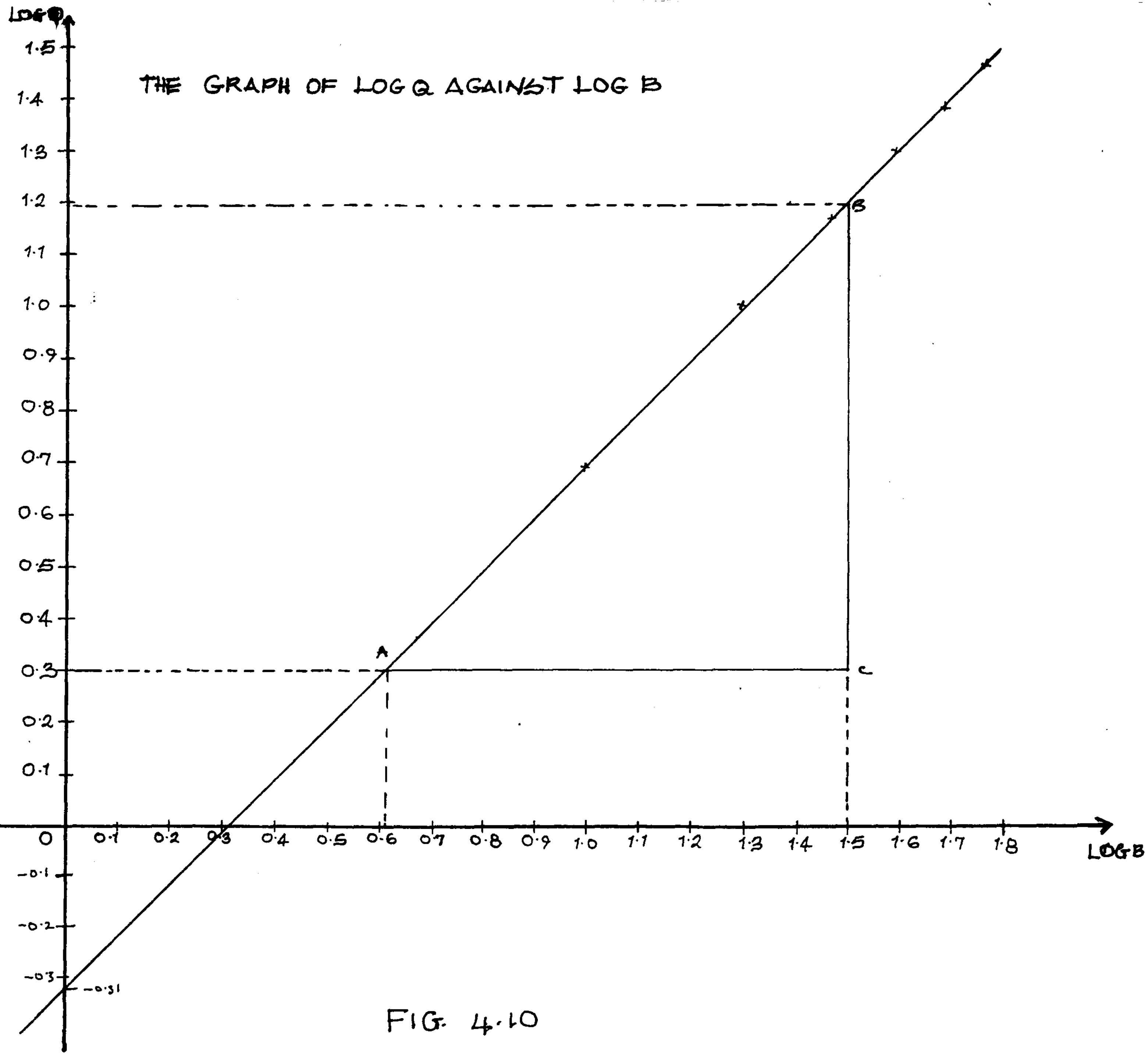



FIG. 4.10

TABLE 4.2 Comparison of theoretical and observed Terminal Velocities

| Sr. No. | Crop | Particle density (g/cc) | Dp range (cm) | Sphericity psi | Method I: Cd | Method I: NRe | Method I: Vt (m/sec) | Method II: ($Cd \times NRe^2$) | Method II: NRe | Method II: Vt | Experimental terminal velocity range m/sec |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Paddy (*Basmati*) | 1,158 | 0,30-0,33 | 0,323 | $1,92 \times 10^6$ to $2,55 \times 10^6$ | 650 to 800 | 3,25-3,63 | $6,14 \times 10^6$ to $8,17 \times 10^5$ | 1300 to 1600 | 6,50-7,27 | 5,50-9,00 |
| 2 | Paddy (IR-8) | 1,367 | 0,35-0,38 | 0,420 | $2,66\ 10^6$ $3,40 \times 10^6$ | 950 to 1500 | 4,07-5,92 | $1,12 \times 10^6$ to $1,36 \times 10^6$ | 2400 to 3000 | 10,28-11,84 | 6,00-10,00 |
| 3 | Head rice (*Basmati*) | 1,408 | 0,25-0,30 | 0,350 | $9,97 \times 10^5$ to $1,72 \times 10^6$ | 450 to 700 | 2,70-3,50 | $3,52 \times 10^5$ to $6,02 \times 10^5$ | 900 to 1300 | 5,40-6,50 | 6,00-10,00 |
| 4 | Head rice (IR-8) | 1,408 | 0,30-0,34 | 0,40 | $1,72 \times 10^6$ to $2,50 \times 10^6$ | 750 to 960 | 3,75-4,23 | $6,68 \times 10^5$ to $1,00 \times 10^6$ | 1300 to 1800 | 6,50-7,94 | 6,50-10,00 |
| 5 | Groundnut pods (M-145) | 0,644 | 1,16-1,74 | 0,525 | $4,05 \times 10^7$ to $8,56 s 10^7$ | 2500 to 4000 | 3,17-5,17 | $2,30 \times 10^7$ $3,75 \times 10^7$ | 7800 to 11000 | 8,39-11,63 | 6,60-13,20 |
| 6 | Groundnut kernels | 1,041 | 0,65-1,05 | 0,63 | $1,29 \times 10^7$ to $5,46 \times 10^7$ | 2000 to 5600 | 4,60-8,00 | $8,03 \times 10^6$ to $3,44 \times 10^7$ | 4700 to 9200 | 10,85-13,14 | 7,70-13,20* |
| 7 | Rice brokens (IR-8) | 1,388 | — | — | — | — | — | — | — | — | [illegible],10-8,80 |
| 8 | Paddy husk | 1,250 | — | — | — | — | — | — | — | — | 0,[illegible]-2,00 |
| 9 | Broken shells of groundnut | 0,566 | — | — | — | — | — | — | | — | 0,33-3,30 |

*Groundnut/kernels could not be subjected to velocities higher than 13,20 m/sec, being the upper limit of the velocity available from the aspirator column used