DESIGN AND CONSTRUCTION OF A MANUALLY OPERATED ACHA HARVESTER

A PROJECT REPORT SUBMITTED TO THE DEPARTMENT OF AGRICULTURAL ENGINEERING IN PARTIAL FULFILMENT FOR THE AWARD OF THE POST GRADUATE DIPLOMA (P.G.D.) IN AGRICULTURAL ENGINEERING (FARM POWER AND MACHINERY OPTION).

BY

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APPROVAL PAGE

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Engr. (DR) D. ADGIDZI
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Engr. (DR) M.G. YISA
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EXTERNAL EXAMINER.

DECLARATION

I hereby declare that this project was conducted by me under the strict supervision and guidance of ENGR. (DR) D. ADGIDZI of the Department of Agricultural Engineering, Federal University of Technology Minna, Niger State, during the 1998/99 Academic Session.

Signature: ....

Registration No: PGD/AGRIC/98/99/61

Date: 27 - 7 - 2000

CERTIFICATION

This is to certify that this project "Design and construction of a manually operated Acha Harvester" is the original work of Aminu Derwam carried out by him under the supervision and guidance of Engr. (Dr) D. Adgidzi and duly submitted to the Department of Agricultural Engineering, Federal University of Technology, Minna.

....

Engr. (DR) D. ADGIDZI
SUPERVISOR.

DEDICATION

This project report is dedicated to God Almighty, to my entire family and the Peasant Farmers out there in Plateau, Nassarawa and Southern Kaduna States who have been struggling "TOOTH & NAIL" to make Acha production a success.

ACKNOWLEDGEMENT

First and foremost, I thank the Almighty God for his infinite mercy and glory for giving me the strength and understanding for completing this project.

I wish to express my profound gratitude and doff my hat to my supervisor Engr. (DR) D. Adgidzi for his untiring efforts in giving full attention to me during the session and inspite of his tight academic schedules.

My sincere appreciation goes to my Head of Department Engr. (Dr) M.G. Yisa for his concern and full involvement in the general affairs concerning our programme (P.G.D.)

I therefore wish to express my profound gratitude to my former Director Mr. K.N. Kakiyes (retired) for his unique leadership quality and encouragement during the pursuance of this course and project. May God bless him abundantly in his future endeavour.

My appreciation goes to Mr. Emmanuel Olawoyi a 100 level student of Mechanical Engineering who assisted a great deal in every aspect.

The contributions made by my colleagues of Defunct NALDA Plateau and Niger States cannot be over-emphasized.

My appreciation also goes to Miss Patience Okenini who devoted her precious time in typing the manuscript, may God Bless her Amen.

I am indebted to Mr & Mrs Jigam a lecturer and student in the University for their moral and financial support.

My special thanks goes to my friends and course mates for their co-operation as the case may be and the entire staff of Minesfield Engineering Company Limited in Bukuru, where the fabrication was carried out.

Finally, my sincere thanks goes to my wife & step son for their understanding and support during my absence while at school and the handling of this project.

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ABSTRACT

The challenges which faces Engineers are profound. Engineers have a vital role in the rapidly changing technological society and traditional sector of the economy of this Country.

Thus the mechanization of Acha harvesting remains one of such challenges inspite of world-wide research efforts made on minimizing post-harvest losses, it still pose problem which needs careful study. Hence the development of this machine.

This project presents a review of similar harvesting machines. The objectives of mechanizing machine are discussed. A detailed description is given of the various features including their fabrication.

The machine will harvest cereals, pulses, soyabean. It requires a draft of 2.5km/hour. The special feature of this machine is transformation of rotary motion from the bevel gear into reciprocating motion through the crank.

The following are the machine's requirements:

a.	Power output	=	0.189 watts
b.	Efficiency	=	80%
c.	Capacity	=	1 hec/hr

Mechanization of Acha is gradually paving way for successful implementation of large scale production of Acha grains for both small scale and large factories.

CHAPTER ONE

1.0. INTRODUCTION

ACHA (Digitaria Exilis), is a stable food crop grown mostly in the uplands of Plateau, Nassarawa and Southern Kaduna States and total area is probably not more than 20,000 hectares. (Enwezer, W.O. 1990).

Acha has over the years continued to gain a lot of global acceptance because of its over increasing economic importance. A further boost to the production is the recent discovery that the crop is the only safe cereal crop that is found to be scientifically consumable by diabetic patients.

Harvesting of Acha is done with hand sickle or the local blacksmith knife despite all it's deficiencies, in the absence of improved harvesting machines. Manual harvesting is tedius and requires a lot of labout to harvest an acha crop. Due to increased shortage of agricultureal labour during the peak season, farmers have to face great difficulties in timely harvesting of this crop.

Heart worming as the development is, production has tended to decline due to the cumbersome nature of harvesting. In most production areas, manual labour is still being only option available to the small-scale producer who form over 90% of the farmers engaged in Acha production.

Little work has been done with little success in the fabrication of machinery for harvesting of Acha. The situation can be worsensed when the available technology is un-accessible to the farmer due to cost implication.

1.1. STATEMENT OF PROBLEMS

The following problems have been noticed during harvesting. They include the following:

1. Time consumption.
2. It is tedious.
3. Considerable loss of grains.

1.2. JUSTIFICATION

With high demand for increased food production, many Countries tried to device ways of simplifying post Harvest Operations. This leads to the design of different harvesting machines capable of handling all grass-like crops from all sources all the year round despite variability in physical properties.

Most of the harvesting done today are not done with the combination of optimum harvesting speed, and reduction in loss of grains. The principal reason being the lack of appropriate technology.

The harvesting done by hand slows rate of production. Accidents occur during harvesting whereby the labour force employed cut their lands with the sharp knives, also the number of knives needed by the labour force is considerable.

All these result into a lengthy and expensive method, the search for a satisfactory way to ease harvesting and increase food production is what this project work aims to achieve.

1.3. OBJECTIVES

The objectives of this project is to fabricate a machine that will suit the agro-ecological and socio-economic conditions of the rural dwellers. Research centers and small-scale/medium industries to facilitate harvesting of Acha and to some extent rice, soya bean and Mowing grass on the farm stead for livestock use.

The specific objectives are:

1. To design and construct an Acha harvester.
2. Cost Analysis of Acha harvester.

CHAPTER TWO

2.0. LITERATURE REVIEW

The grain harvesting machine is one of the most important agricultural machine.

It is often called the "combine" because it carries out several operations using a single machine. Present day harvesters are as a result of a combination of the ideas, efforts and energies of men of many centuries.

2.1. CONVENTIONAL CUTTER BAR

Quick and Buchele (1974) studied the characteristics of the conventional header, asserting that 80% of the loss occurred as a cutter bar loss. They observed that the standard 76mm knife section appeared over crowded at speeds 5.1. kM/h, leading to evaluation of 38mm narrow patch sections with the same stroke.

Grow et al. (1958) determined leader loss of grain barley in conventional cutters and the effect of reel adjustments. The results show that high reel speed index (red peripheral speed divided by forward speed) can cause excessive seed losses, particularly in upright crop, with either the fixed-bat reel, speed index of 1.25 to 1.5 gave consistently satisfactory performance without excessive shatter losses.

Increasing the ratio to 2.8 nearly doubled the header loss. They suggested the optimum location of the reel relative to the tips of the knife section where a fixed bat reel ordinarily should be 15cm to 25cm forward from the tips of the knife sections, and at a height such that the lowest position of the bats is a little below the lowest heads. A pick-up reel, when used in lodged crops, should be lower and a little further forward. Also to simplify the problem of maintaining constraint reel speed index under all operating conditions, they indicated that the reel be driven from ground wheel or from some shaft whose speed is proportional to forward speed.

2.1.1. FLOATING CUTTER BAR.

Nave et al. (1976) used an airjet system as an attachment to convectional floating cutter bar. This system when tested on soya bean, reduced the losses approximately from 45 to 4%. A concept reported by Boddiford and Richey (1975) to reduce the height of cut, was based on the punching of soyabeans. This device which was used on row crop, had rubber tyres to pull the stalks with or without cutting of the roof system. Loss reduction was reported to range from 47 to 72%.

Schertz et al. (1976) using rotary cutters or belts on soyabeans, reported loss reduction of41 to 67 percent over the conventional header. In order to reduce the shatter losses, pans were suspended ahead of the floating cutter bar between the row to catch shattered pods due to cutting of the stalk. Shrock et al (1974) reported loss reduction of 72% and 28% per 18cm and 76cm beans respectively over the conventional header.

Other attempts reported by Aichel et al (1976) have been made in the reduction of header losses.

2.1.2. ANIMAL - DRAWN HARVESTER

The first reaper was developed by Cyrus McCormick as early as 1933 (Hopper et al, 1953 and Sernani, 1980). It was patented in the year 1945. This machine was found working satisfactory for cereal harvesting. The machine had many components such as conventional cutter-bar and reel. A pair of horses were used to pull the machine. There has been approaches for developing a manually operated harvester (Saran and Ofha, 1967) but power for the operation was a great limitation.

The proto- types of Mc Cormick reapers were exported to different countries of the world for evaluation and testing.

In India work on such machines was started in 1961 - 62 at the Allahabab Agricultural Institute and in 1964 - 65 at Punjab Agricultural University Lithuania Verma (1970), Verma and Bhatnagar (1970) made considerable efforts to develop Bullock-drawn reapers suitable for Indian situations.

In 1968, the Indian Agricultural Research Institute (I.A.R.I.), New Delhi started the research work on animal-drawn reaper. This research effort was similar to the work done at P.A.U. (Punjab Agricultural University), Lud Mana, except the introduction of horizontally operated belt conveyor.

Windrower behind the cutter bar, Khanna (1970). This additional provision was provided to windrow the harvested crops on one side so that immediate removal and cleaning of swath for the subsequent sum of the machine is done. However, the problem of high draft for a pair of bullock remained unsolved.

It was reported, that the machine was operated by an auxiliary engine and pulled by a pair of bullocks. Some problems were experienced regarding the goading of bullocks. Due to its design features, this machine also could not get acceptance of the farmers and manufacturers.

The research work on the prototype machine was further continued by introducing an auxiliary engine in the design singh and singh (1987), Bansal and Singh (1988) incorporating a vertical conveyor windrower on the system. The prototype was fabricated and testing is in progress. Modifications are on.

Similar work was also carried out on the design, development and field evaluation of animal drawn reaper at J.N.K.R.V., Jabal in the 1970's. In the opinion of the authors, these designs have some limitations such as.

1. Problem of goading the animals due to noise and vibrations of the auxiliary engine.
2. The economic feasibility of this machine has to be examined in line with recently developed self propelled engine operated, manually guided machines and with other power tiller-operated harvesters even with the limitations of the operating speeds of animals and.
3. High initial cost.

2.1.3. IMPROVEMENTS ON MANUAL HARVESTING TOOLS:

Literature on improved manual harvesting tools is scarce. There has been a direct pump from hand tools to machines driven by animals and later by engines. Not much attention has been paid to the efficient use of manually operated tools. Some of the basic mechanism used on engines and animal powered equipment could be used efficiently with man powered equipment as well (Saran and Ofha, 1967; Siddiqur et al; 1980).

There have been cases of improvement in hand tools and manually operated harvesters. Nwuba (1981) in studying the energy demands for selected agricultural hand tools observed that a reduction in energy requirement for a matched was achieved by increasing its length from 63cm to 86cm. The corresponding decrease in energy demand was 5.00KJ/mm to 4.19KJ/mm.

The chopping hoe (Magirbi) was also modified by increasing the length of its handle from 57cm to 150cm thereby eliminating the bending posture.

Pandey and Devnani (1981) tested and compared the field performance of four cereal harvesting sickles with the incorporation of certain features into earlier models of sickles, an improved one was developed which proved to have an advantage of saving 5 - 7 man - hour/ha on paddly fields.

Hand pullers of cotton stalks which work on principles of levers were developed in Gexira-Sudan for clearing cotton fields in preparation for the next year's planting. This tool proved to be of much advantage over direct pulling of stalks by hand (Damian, 1979).

At two different points in time, two similar hand operated grain harvesters were developed by Savan and Ofha (1967) and Sidiqur et al. (1980).

The harvester was a push type machine which, during forward motion, power from traction wheels was transmitted through a system of chain and sprockets to a cam wheel. The cam in return, actuated a lever arm which carries and actuates a lutter-bar knife. A cutting speed of 200 rpm to 225rpm was used in the design and operator walking speed of 1.6 km/l was assumed. This machine could cover an area of 0.049 ha/h. Which is about six times the area covered by sickle and twice the area covered by scythe.

A single-row sun flower harvester was developed by Shrivastava and Dyck (1978). It was also a push type machine.

Vander Sar (1979) developed a hand operated cassava harvester which also works on principle of levers. With a mechanical advantage of three and a downward effort of about 50kg (490N) exerted by the operator weight, an upward lifting force of about 150kg (147ON) could be obtained at the other side of the fulcrum which is used for lifting cassava roots out of the soil.

He concluded that this harvester had a saving in time and labour requirement from 100 man-h/ha.

The Japanese cecoco hand rice reaper cutter is an imported hand tool for harvesting rice, wheat and similar crops. This unit is normally pushed against the crop plant while cutting is effected by two serrated fixed edges. This harvester was tested and observed to be of advantage over the traditional sickle (Kalkat and Kanl, 1978). A saving in labour requirement from 67man ha/ha to 52 man - h/ha over the sickle was achieved on paddy fields.

In addition some efforts on development of Oxen-drawn reapers were made in Austria, China and many other countries. The Chinese design is a single-wheel and single animal machine. This machine has conventional cutter bar. The design of animal-drawn reaper looks simple and light-weight machine but not enough information is available. This machine has cutter bar for cutting plants and reel for gathering the harvested crop on the board.

The overall picture of development of animal-drawn reapers indicates that considerable efforts have been made on the design and development. Due to one or other limitations not even a single design could become popular in any country. Thus seeing the pressing needs of animal drawn reapers in the country for cereals, pulses and oilseeds, the design and development work was initiated at the central institute of Agric..-Engineering (CIAE), BLOPAL, YADOW (1984, 1986), YADAV AND YADAV (1985, 1987).

2.1.4. THE VERTICAL CONVEYOR REAPER (V.C.R.)

Paddy reaper windrowers have been successfully modified and field tested in various research stations. This is an ideal harvester for Indian conditions. By this machine the crop can be harvested and windrowed in a row which can be collected and taken to the thresher. A modified 5h.p self propelled machine is basically an IRRI design, but when used as such in the wet lands' of Icerala (India), many problems like, difficulty in turning in the small plots, slipping of the twisted V-belt, excessive vibration at the handle, improper balancing and consequent dozing of the cutter bar were noticed. The following modifications were therefore incorporated to make the machine acceptable to the **KERALA** farmers:-

1. Left and right side clutches were provided and thus improved the maneuvrability to a large extent.
2. The mounting structure for the 5 h.p diesel engine was modified so that the belt adjustment was easy and the position of the engine could be reversed for mounting a rotovator, hence the machine can be used as a tiller for this also avoided the additional balance weight.
3. The crop dividers were modified.
4. The simple tube handle was replaced with a double tube reducing the vibration at hold-grips considerable.
5. A rotovator attachment was attempted for multi-purpose use of the machine.

2.1.5. MOWERS

The reciprocating knife cutter bar can be traced back for at least 150 years. Many farm machinery manufactured made trailed powers in the 1950's and by the end of the decade most of them were making semi-mounted and mounted mowers.

Agricultural Engineers were always searching for something better than the reciprocating knife cutter bar. The principle of the rotary mower is as old as the reciprocating knife but it made little impact on the farm scene.

2.1.6. WINDROWERS

This has a 6-bat pickup reel and vertical cutter bar for separation of crops such as oil seed rape. It canvases, deliver the crop centrally to a deep and wide passage within the wheel tracts developed by SHELBOURNE REYNOLDS (CULPIN, C, 1981, Farm Machinery).

CLAAS manufactured a harvester which has forks mounted on cranks above the shakers to lift and spread the straw. The dust proof ventilated cab provides a good view of vital processes and a healthy environment for the operator, all other components are hydraulically operated.

Massey Ferguson developed the two drum mower with top link driven by shaft and bevel gears, weight of the rotor is supported by the draught - bar to which are attached skids for cutting height control.

2.1.7. BINDERS

These are the second generation harvesters, most were trailed but some companies made semi-mounted and mounted binders.

Massey-Harris Ferguson developed the P.T.O. and land wheel drive 6ft cut sunshine 6B binders in the 1950's. They cut the crop with a smooth section knife which was guided onto an endless canvas cross conveyor by the reel. The conveyor moved the crop side ways to the elevators which lifted it to the knotter deck.

However, after the thorough evaluation of the existing machines, the results obtained on their short comings as relates to the Acha harvester are:-

1. Inability to effectively handle different crops especially (Tropical ores).
2. High initial cost
3. Adaptability/suitability on tropical soils.

It is due to these inadequancies that no satisfactory harvester has been successfully introduced into the world market to cater for grass like crop grains.

This inspite of the indicated strong world-wide research thrust the development of a technically or economically acceptable harvester confirms to pose a challenging problem.

2.1.8. DESIGN CONSIDERATION OF HARVESTER

After conducting a survey and careful study on the various available designs and their limitations, a manually operated harvester was considered based on the following design criteria:

1. The machine should be simple, light weight and sturdy in design.
2. The draft requirements of the machine should be well within the draftability of an average built human which is 0.8kw (Kaul, R.N. (1984).
3. It should be functionally flawless for harvesting the crop(s) for which it had been designed for.
4. To meet the requirements of small and medium size land holdings, the unit should be affordable in price.
5. It should be a multipurpose machine without compromising quality.
6. Harvesting losses by the machine should be within the acceptable limits.
7. Easy access to all parts.
8. The machine should be easily to operate and maintain.

2.1.9. TECHNICAL CONSIDERATION

Many machine designs consider the reduction in labour with increased productivity especially with increasing demand for food production.

The crucial problem that must be overcome in order to realise the full potential of mechanizing Acha harvesting is the development of reliable and effective ways to:-

i. Monitor the reciprocating speed/strokes of the moveable blades.
ii. Reducing the high operating labour cost.
iii. Uniformity of harvesting.

2.0. ACHA AGRONOMIC PRACTICES

(a) Fertilizer Sources and Rates:-

No recommendations are presently available, but in order to maintain the fertility of the soil a token application of nitrogen (N) and potassium (P), is suggested as follows: 13kg N/ha (i.e 50 kg/ha of calcium ammonium Nitráte (CAN) or 30kg/ha of urea + 9kg/ha $P_2 0_5$ (i.e 50kg/ha of single super phosphate (S.S.P).

(b) **Fertilizer Application**:

Both N and P should be broadcast and lightly worked into the soil a day or two before seeding or sowing.

(d) **Varieties:**

There are no improved varieties of Acha and therefore none is recommended. It is suggested that the best local varieties should always be used.

(e) **Cultural Practices**:

No formal recommendations are available but Acha is known to perform well in areas with annual rain fall of 1000 - 1,300mm. Soils should be light sandy and not necessarily very fertile. It is suggested that planting should be done in May - June as it allows the crop to be harvested under cool and dry weather. Seed should be broadcast immediately following a shower at the rate of 45 - 55kg/ha on the flat or slightly raised ridges.

(f) **Yield Expectancy:**

Farmers are known to have obtained yield which vary between 300 and 1000kg/ha of an threshed grain.

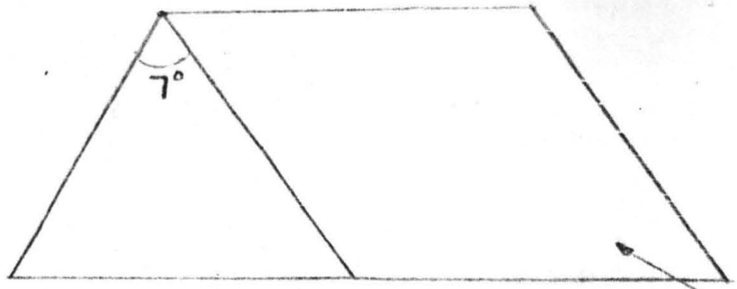
Source: (Enwezor, W.O. et al (1989), fertilizer use and management practices for crops in Nigeria. Series No. 2. Federal Ministry of Agriculture, Water Resources and Rural Development.

CHAPTER THREE

3.0. METHODOLOGY

The method adopted in this project is to study existing harvesting machines with a view to finding out their short comings and to model the new machines to eliminate such problems. The new machine thus conceived is as described.

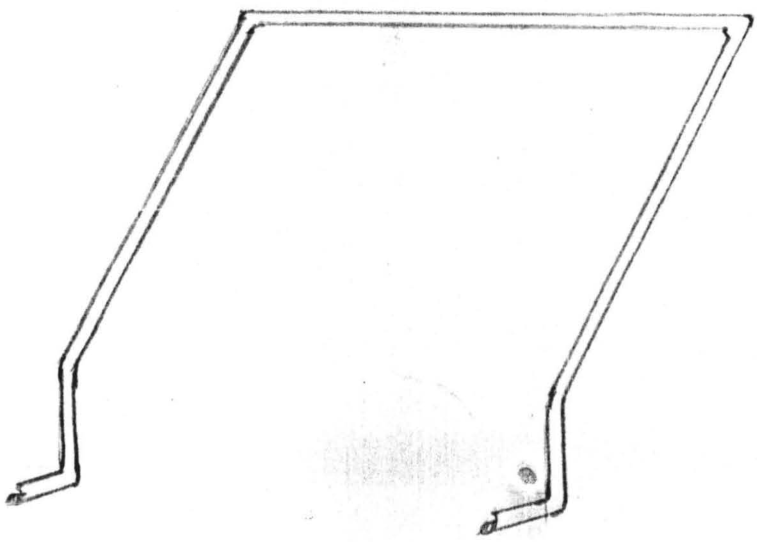
The sketch of the general view of the harvesting machine is shown in fig. 1 and it is composed of the Beverl Gear System, Drive handle, flywheel, Pitman or connecting rod, land wheel, spring-load clamp, fixed and reciprocating blade, rotary driveshaft, Hollow pipe frame, roller bearing, wear plat and the main frame.



Surface on which the cut crop will fall and roll to the ground in a windrow.

HOLLOW PIPE FRAME

The frame is the component to which the header, rotary shaft, bevel gear system, main frame rotating handle are welded to.



MAIN FRAME

The main frame forms almost the entire unit of the harvester. The cutting unit reciprocating unit, land wheel and the hollow pipe frame are attached to this unit. It is fabricated with a tin sheet.

ROLLER BEARING

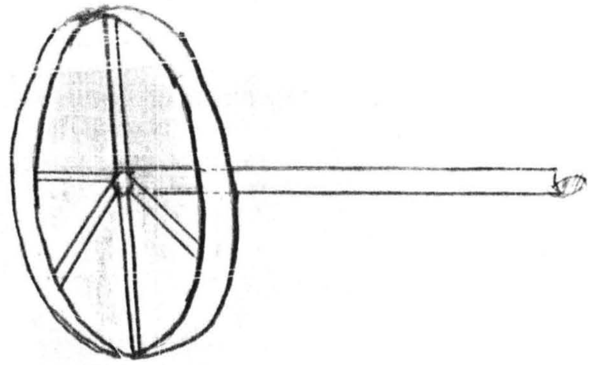
The drive handle, rotary shaft and the land wheel shafts fit snugly and rotate freely with the help of the roller bearing which are constantly lubricated for a smooth operation.

LAND WHEEL

The forward and reverse movement of the operator/machine is achieved through the land wheel as it rolls on the ground. It's function is two-fold,

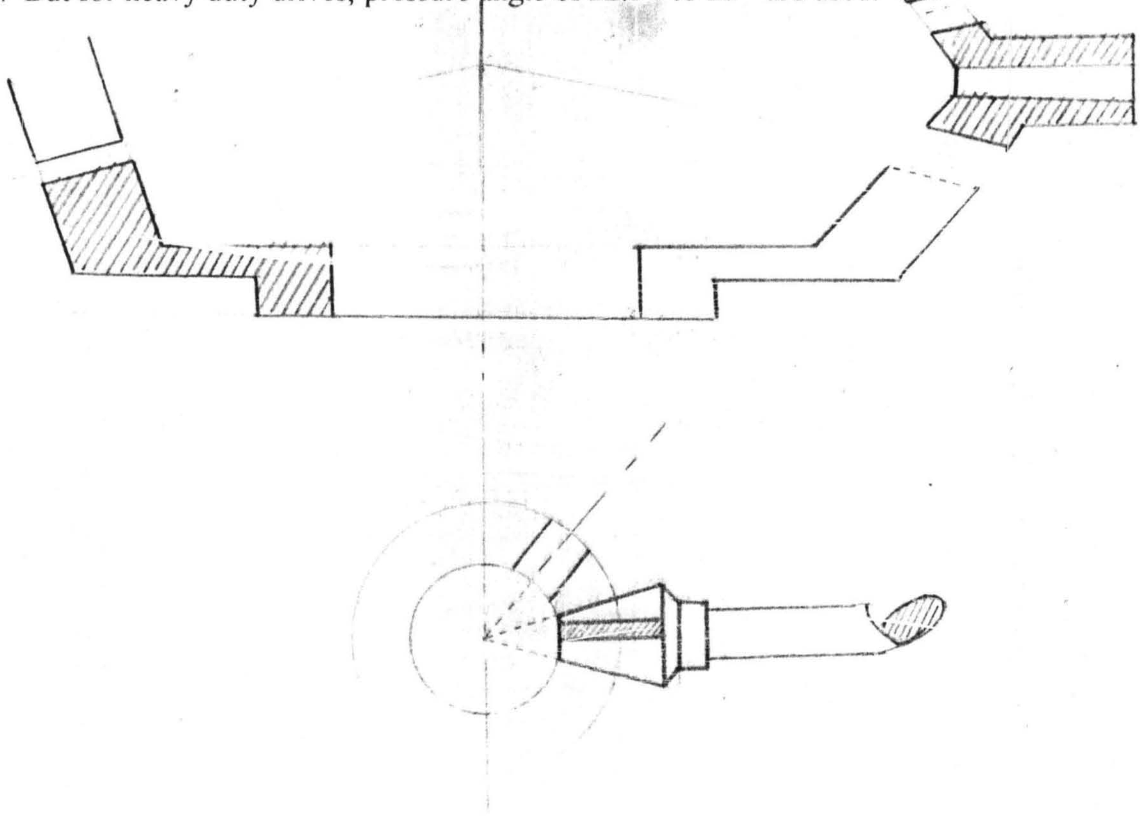
i. It serves as a medium for enhancing the forward and reverse movements of the operator.

ii. It tend to counterbalance the dead load of the entire machine.



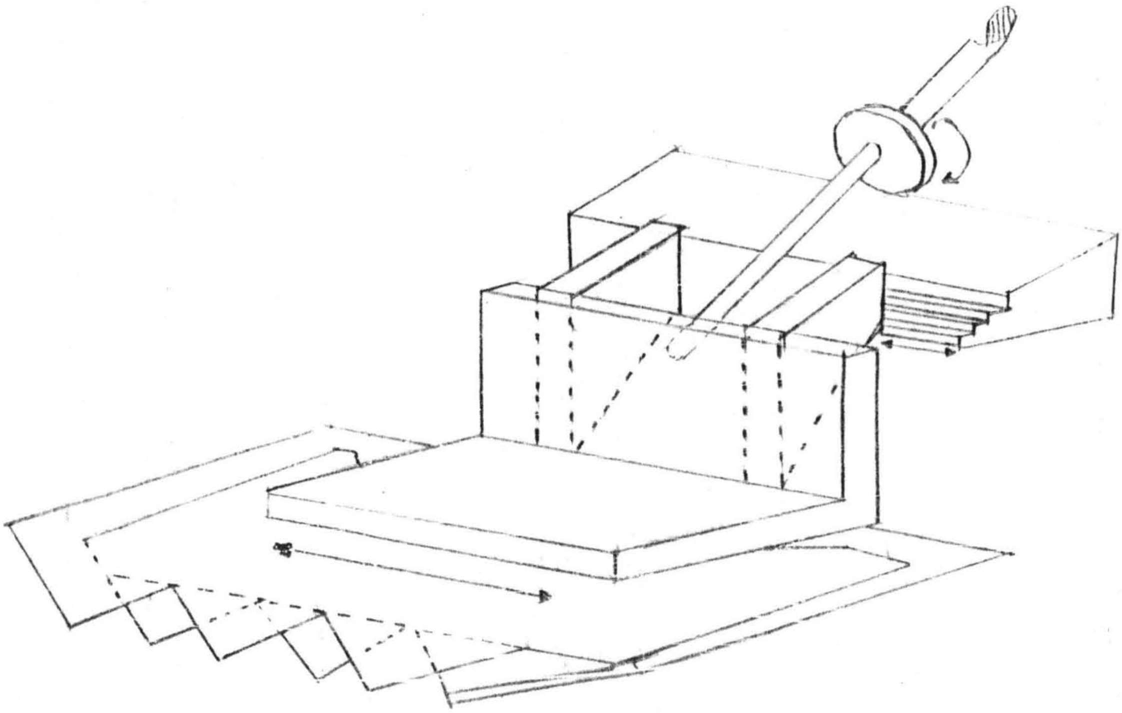
BEVEL GEAR SYSTEM

This is the prime mover where the drive is transmitted to the entire cutting unit. The type of gear on this machine is the straight bevel gear, with a pressure angle of 20° as it is for light operation. But for heavy duty drives, pressure angle of 22.5° to 25° are used.



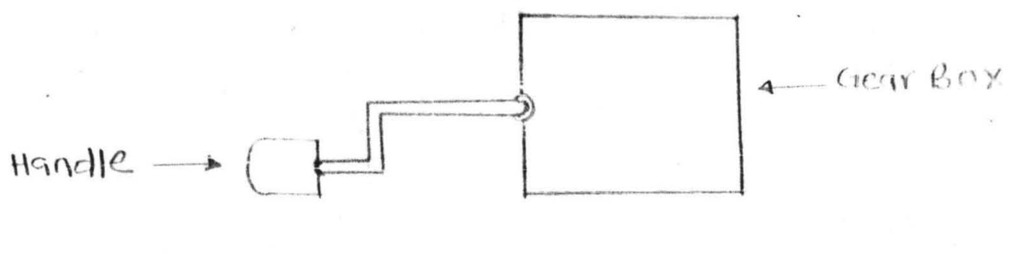
MOVEABLE FLAP & BLADE

This is the section that the pitman strikes during its (pitman) straight - in -line or reciprocating movement. Also bolted or welded to this unit is moveable blade.



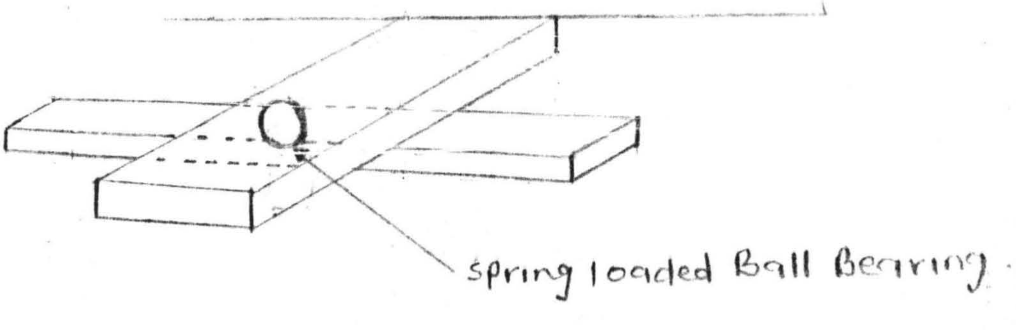
DRIVE HANDLE

This is a crank-shaft that is keyed to the bevel crown wheel system where it serves as the prime mover through the rotary effort applied by the operator.



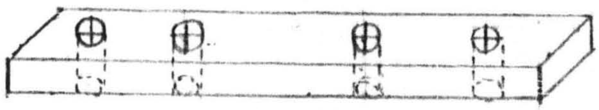
SPRING LOADED CLAMP

They are to keep the fixed and moveable knives in position or alignment most especially when the knives strikes an obstruction or when too much material is being cut by the knife thereby clogging it then the knife springs up or down to clear the obstruction.



WEAR PLATE

This is a replaceable spring plate designed to protect the reciprocating knife from wear by receiving the backward thrust of the knife when in work. It is adjustable through the bolts and nuts to compensate for wear on the knife back and to accommodate the fullness of a new knife when fitted.



3.1.3 DESIGN CALCULATIONS

3.1.4. SHAFTS

Shafts and axles are the machine members mostly cylindrical in cross-section, which support the revolving part and may either be fixed relative to the supports or rotate together with the part mounted on them. Hence by the axle as a bending load and it is subjected to bending stress.

On the other hand, not only supports a revolving part but also transmits torsional moment. As a result, the shafts are subjected to bending as well as torsional stresses.

Shafts can be classified according to their use:

i. Prime mover shafts, e.g. turbine shafts, motor shaft engine shafts etc.

ii. Machine shafts

iii. Power transmission shafts, e.g. line shafts, countershaft, jack shaft etc.

3.6.2. DESIGN FOR STRENGTH

a. Pure torsional load:- when the shaft is subjected to pure torsional load, the principal stress induced in the shaft will be the shear stress. The maximum shear stress induced in the shafts is given by the relation:

$$fs = \frac{16Mt}{\pi d^3} \text{ N/m}^3$$

where Mt = Torsional moment, Nm

d = Diameter of shaft.

The above relation is for a solid shaft.

3.1.5. GENERAL SHAFT DESIGN CALCULATION

Maximum shearing stress of a solid round bar subjected to steady combine static bending and torsion is given as:

$$\tau_{max} = \sqrt{(\sigma_x)^2 + \tau^2} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

(Dentschman et al., 1975)

Where $\sigma x = \dfrac{32m}{\pi D^3}$

$\tau = \dfrac{16T}{\pi D^3}$

replacing X and in equation (1)

$$\tau_{max} = \frac{16T}{\pi D^3} \sqrt{M^2 + T^2} \text{ .... (2)}$$

considering shock and fatigue factor, equation (2) becomes

$$\tau \text{ max} = \frac{16T}{\pi D^3} \sqrt{(K_m M)^2 + (K_E T^2)} \text{ .... (3)}$$

On the design for shaft diameter, the maximum shearing stress, τ max is replaced allowable shearing stress, τa. That is

$$\tau a = \frac{16}{\pi D^3} \sqrt{(K_m M)^2 + (K_E T^2)}\,]^{1/3}$$

The direct stress due to an axial load P is

$$S = \frac{4P}{\pi d^2}$$

These stresses may be combined by use of the combined stress equation as given below:

$$S(max) = \frac{16}{\pi d^3} \sqrt{(T^2 + \frac{(PD)}{8})^2} \text{ .... (4) (Black, 1955)}$$

By substituting the allowable shear stress for S(max), the above equation may be solved by trial and error.

3.6.4. DESIGN OF BEVEL GEAR

The Harvester is required to use a pair of bevel gears to transmit power. They are the most efficient means of transmitting rotation between angularly disposed shafts. Power requirements may be in Thousands of horse power.

The power a gear set can transmit depends on it's ratio, speed, size, hardness of material, and quality (chironis, 1967). There are three basic types of bevel gears - straight, spiral and zerol. In this work, straight bevel is employed.

The gear is to transmit a power of 0.08KW Human factor in Engineering. A transmission ratio of 3.1. is to be maintained between the gear and the pinion. Other advantages of using such a transmission system ie the gear and pinion are as follows:

i. Durability.

ii. Economic of manufacture.

iii. Adequate strength of the gear.

3.6.5. MATERIALS USED FOR GEAR DESIGN

There are different types of materials used for manufacturing gears. The materials available are: gray and cast iron, brass, alloy steels, plastics and carbon steel.

DESIGN CONSIDERATION

The gears can be designed for surface durability and strength, using the following two equations.

Surface durability is the resistance to potting and involves the stress at the point of contact. Strength is the resistance to tooth breakage and refers to the calculation of bending stress in the root of the tooth.

Expressions for surface durability and strength are given below:

Surface Durability:

$$T = \frac{FJK_V}{2K_m} \frac{(S_C D)^2}{Cp} \quad \ldots\ldots (5)$$

Strength:

$$T = \frac{FJK_V}{2K_s K_m} \frac{(std)}{Pd} \quad \ldots\ldots (6)$$

Where:

T	=	Maximum allowable torque (Kg/cm)
Sc	=	Allowable contact stress
S_t	=	allowable bending stress.
d	=	pinion pitch diameter.
P_d	=	diametral pitch.
F	=	Face width.
C_p	=	elastic coefficient
I	=	geometry factor (durability)
J	=	geometry factor (strength)
K_m	=	load distribution factor
K_v	=	dynamic factor
Ks	=	Size factor.

3.6.6. DESIGN OF PINION

The number of teeth taken is to be 18 and the module is 3 and pressure angle of 20^0. Therefore the pitch circle diameter for pinion Pd or D.

P_D or D.

$P_D = M \times Np$

where M = Module = 3

Np or Ng = Number of teeth on pinion.

$P_D = 18 \times 3 = 54$mm

Circular pitch Pc $= \frac{\pi P_D}{Np} = \frac{\pi \times 54}{18}$

Pc = 9.426mm

Diametrial pitch (Ppe) $= \frac{Np}{P_D} = \frac{18}{54} = 0.33$mm

Addendum (A) $= 1/Ppe = \frac{1}{0.33} = 33$m

Clearance (cp)	=	0.157 x m	=	0.157 x 3 = 0.471mm		
Dedendum	=	1.157 x 3 = 0.471mm				
Outside Diameter	=	pitch circle diameter x 2 addendum			=	54 x 2 (3)
					=	60mm
Whole depth (wp)	=	2.157 x m				
	=	2.157 x 6.471mm.				
Root depth (RP)	=	outside diameter - 2 whole depth				
	=	60 - 2 (6.471)				
	=	47.058mm				
Tooth Thickness (EP)	=	$\frac{1.5708}{Ppe}$	=	$\frac{1.5708}{0.33}$	=	4.76mm
Working depth (We)	=	whole depth - clearance				
	=	6.471 - 0.471				
	=	6mm.				
Radius of Base circle	=	Pitch radius x cos 20°				
	=	27Cos 20°				
	=	25.37mm				
Face width (FP)	=	6 x m	=	6 x 3		
	=	18mm				
Hub length (Hp)	=	1.5 Ds				
Ds	=	Diameter of shaft				
	=	1.5 x 29	=	43.5mm		

Hub Diameter DH = 1.6 Ds

= 1.6. x 29 = 46.4mm

Web Thickness = 3 x 3 = 9mm

Chordal Thickness = 0.5 x 9.426

= 4.713.

Thickness of rim = 2.5 x m = 2.5 x 3

= 7.5mm

3.1.7. DESIGN OF GEAR (DRIVEN)

The speed ratio is given as

$$\frac{Ng}{Pp} = \frac{Vp}{Vg}$$

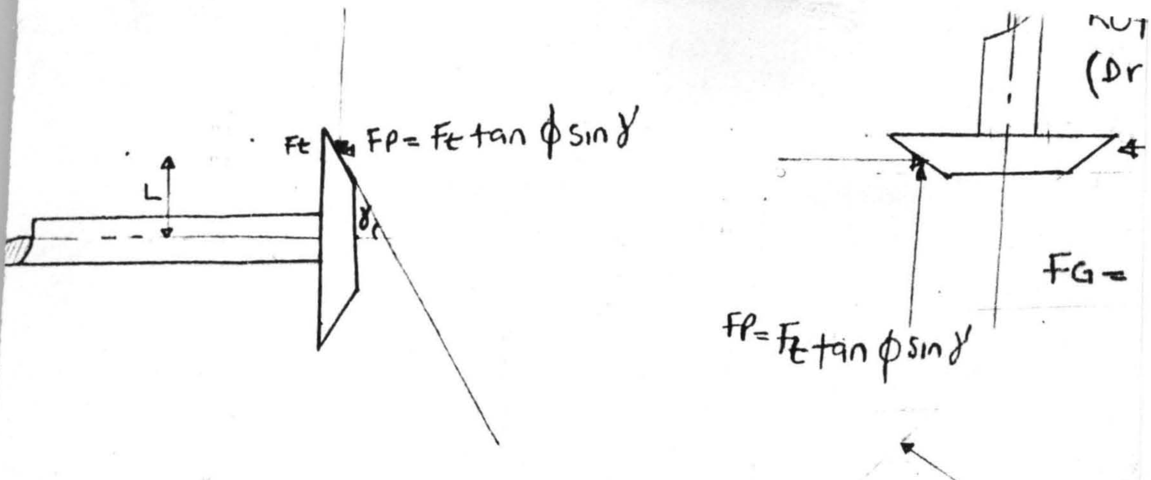
Where Ng and Np are the number of teeth on the gear and pinion respectively while Vg and Vp are gear and pinion speeds respectively.

The speed ratio is 3:1 while the number of teeth on pinion is 18m,

Hence, $\frac{Ng}{18} = {}^3/_1$

Ng = 18 x 3

Ng = 54 teeth



Source: **Hall e t al. (1961)**

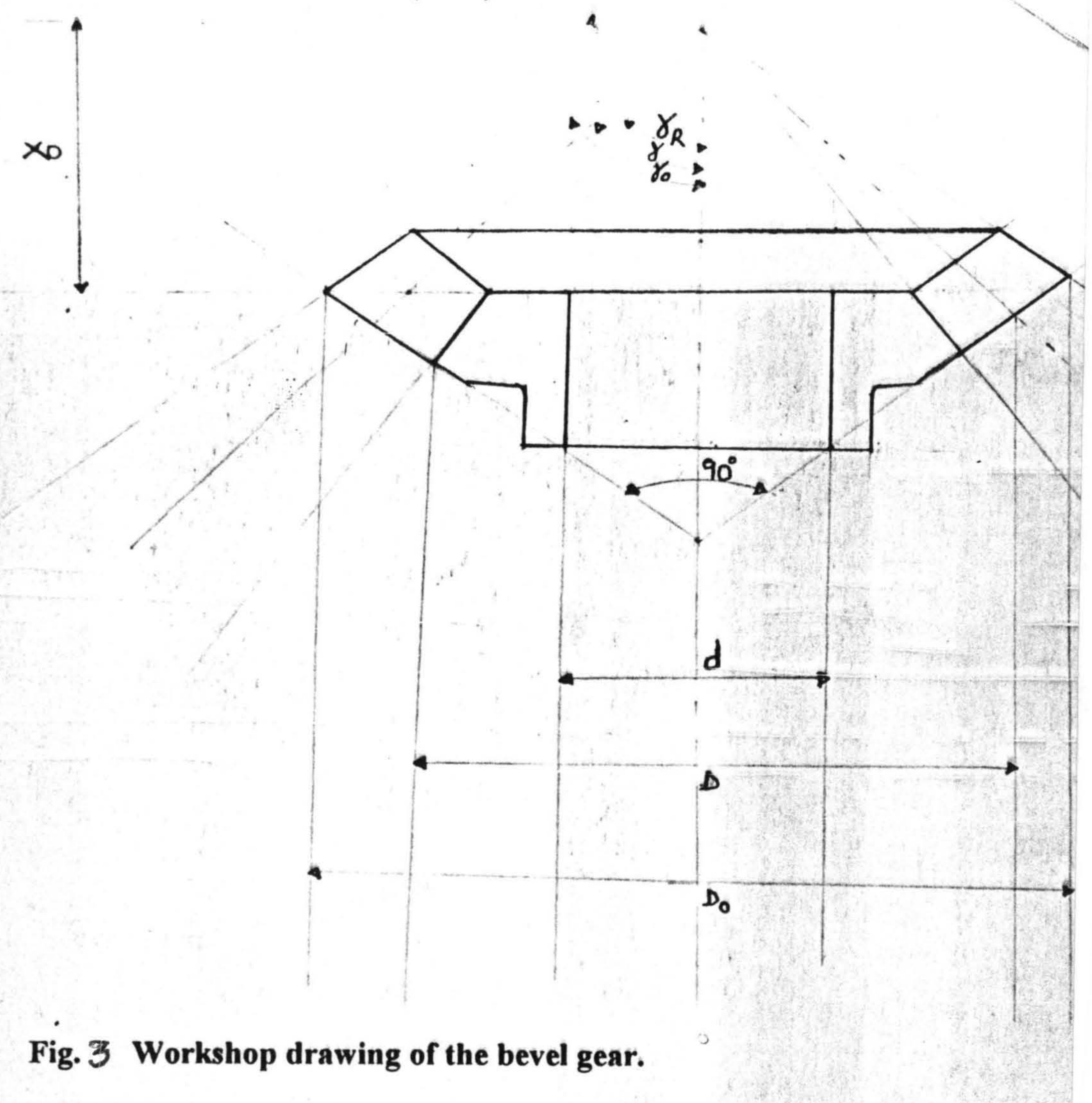


Fig. 3 Workshop drawing of the bevel gear.

Where:

Np	=	Number of pinion teeth	F	=	face width
NG	=	Number of gear teeth	ϕ	=	Pressure ang
Pd	=	Diameter pitch	hk	=	Working dep

ht	=	whole depth
Σ	=	shaft angle
d	=	pitch diameter
$\gamma = r$	=	pitch angle
Ao,	=	cone diameter
p	=	circular pitch
$\gamma r = \sqrt{R}$	=	Root angle (base angle)
do = Do	=	Outside diameter
Xo = xo	=	Pitch apex to crown
t = T	=	Circular Thickness

To determine the pitch circle diameter of the gear (Pg) using the relationship ie.

$$\frac{Ng}{Ng} = \frac{Pg}{P_D}$$

Ng and Np as defined above while Pg and P_D is the pitch circle diameter on gear and pinion respectively.

Therefore,

$$\frac{54}{18} = \frac{Pg}{54}$$

$$Pg = \frac{54 \times 54}{18} = 162 \text{ mm}$$

$$\text{Diametrical pitch (Pge)} = \frac{Ng}{Pg} = \frac{54}{162} = 0.33\text{mm}$$

For the two gears to mesh properly with each other, they have the same module.

Thus, mg = mp

mg = 3mm

Addendum	=	$\frac{1}{Pge}$ - $\frac{1}{0.33}$	=	3mm
Circular pitch (P_gc)	=	$\frac{\pi Pg}{Ng}$	=	$\frac{\pi(162)}{54}$
	=	9.426mm		
Dedendum	=	1.157 x 3		
	=	1.157 x 3		
	=	3.471mm		
Clearance (Cg)	=	0. 157 (m)		
	=	0.157 x 3		
	=	3.471.		
Outside Diameter	=	pitch circle diameter + 2(addendum)		
	=	162 + 2(3) = 168mm		
Whole depth (Wg)	=	2.157m		
	=	2.157 x 3		
	=	6. 471mm		
Root depth (Rg)	=	Outside diameter - 2 whole depth		
	=	168 - 2 (6.471)		
	=	155 . 06.		
Tooth Thickness (tg)	=	$\frac{1.5709}{Pg}$	=	$\frac{1.5708}{0.33}$
	=	4.76mm		

Working depth (Wtg) = whole depth - clearance

= 6.471 - 0.47

= 6mm

Radius of base circle = Pitch radius x Cos20°

= 76.12mm

Diameter of the base circle for gear = 76.12 x 2

= 152.24mm

Face width (Fg)	=	6 x m	=	18mm
Hub length Hg	=	1.5 Ds		
Hg	=	1.5 x 29	=	43.9mm
Web thickness	=	3 x 3	=	9mm
Rim thickness	=	2.5 x m	=	7.5mm
Chordal thickness	=	0.5 x circular pitch		
	=	0.5 x 9.426		
	=	4.713mm		

Hub diameter Hdg = 1.6Ds

= 1.6 x 29

= 46.4mm

Cone distance L $= 0.5m \times \sqrt{Ng^2 + Np^2}$

$= 0.5 \times 3 \sqrt{54^2 + 18^2}$

= 85.38mm

$$\tan\lambda = \frac{2}{\sqrt{Ng^2 + Np^2}}$$

$$= \frac{2}{\sqrt{54^2 + 18^2}}$$

$\tan\lambda \quad = \quad 2.01^{o}$

$$\tan \gamma = \frac{2.4}{\sqrt{Ng^2 + Np^2}}$$

$$= \frac{2.4}{\sqrt{54^2 + 18^2}}$$

$$= 2.41^{o}$$

$$\tan \delta^2 = \frac{Z_2}{Z_1} \div \frac{54}{18}$$

$$= 71.56^{o}$$

Root angle on gear $\delta tz \quad = \quad \delta - \gamma$

Face angle of gear $\delta a_2 \quad = \quad \delta + \gamma$

The back angle $\delta b \quad = \quad 90^{o} - \delta$

Pitch angle on the pinion $\delta 1 \quad = \quad \delta b$

To avoid interference, the addendum of the gear will be

$$\leq \sqrt{(\text{base circle radius})^2 + \frac{(\text{Center distance})^2 \sin^2 \phi}{2}}$$

where ϕ is pressure angle

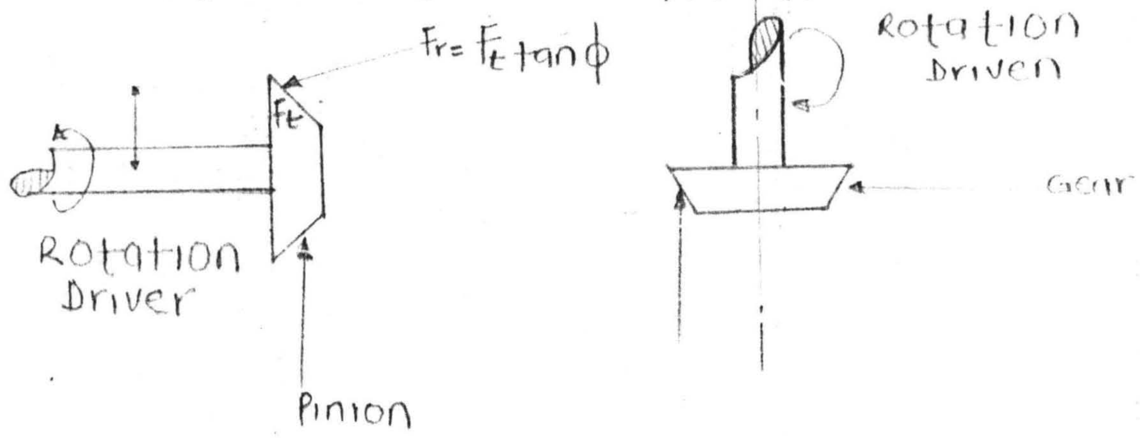
therefore: $\sqrt{(76.12)^2 + _{½}(54 + 162)^2 \times \sin^2 20^{o}}$

$= 84.60\text{mm}$

3.1.7. FORCE COMPONENTS OF STRAIGHT TOOTH BEVEL GEAR

1. Tangential force $F_t \quad = \quad M_{t/r}$, this force is considered acting at the mean pitch radius r. M_t is the pinion torque.

2. Separating force F_r = $F_t \tan \phi$ where ϕ is the pressure angle. The separating force can be resolved into two components, the force component along the shaft axis of the pinion is called the pinion thrust force F_p, and the force component along the shaft axis of the gear is called the gear thrust force F_p (fig a & b).



Source: (Hall e t al (1961)

Fig.4 ***Resolution of the Separating Force in to two Components.***

The three mutually perpendicular components are shown in fig (3). These are:-

a. The tangential force F_t = $M_{t/r}$

b. The pinion thrust force $F_p = F_t \tan \phi \sin \gamma$ where γ is the pitch cone angle of the pinion.

c. The gear thrust force F_g = $F_t \tan \phi \cos \gamma$

3.1.8. WEIGHT OF PINION

To determine the weight of the pinion, therefore, use the value obtained from the calculation of the design of pinion.

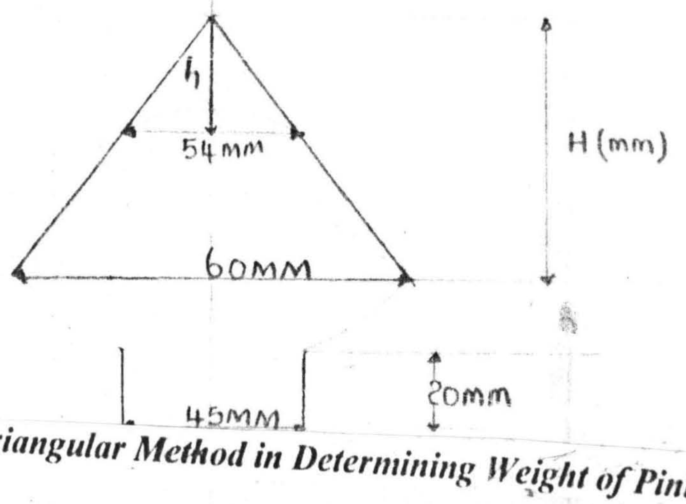


Fig.5 ***Triangular Method in Determining Weight of Pinion.***

Using similar triangle method.

$$= \frac{h + 27.06}{h} = \frac{30}{27}$$

$$27(h + 27.06) = 30h$$

$$h = \frac{730.62}{3}$$

$$h = 243.54mm$$

for smaller cone volume

$V = {}^1/_3\ \pi\ r^2\ h$ .... (7)

$= {}^1/_3 \times \pi \times (27)^2 \times 243.04$

$= 185944.25mm^3$

for bigger cone volume.

$V = {}^1/_3\ \pi\ r^2\ h = {}^1/_3 \times \pi \times (30)^2 \times 2 + 0.6$

$= 255067\ .\ 56mm^3$

Volume of the frustrum = volume of the bigger cone + volume of the smaller cone.

V_f = $(255067.56 - 185944.25)mm^3$

V_f = 69123.31mm

Total volume of the pinion = volume of frustrum + Volume of the cylindrical part.

V_p = $(69123.31 + 31812.75)mm^3$

V_p = $1000936.06mm^3$

V_p = $0.00010096m^3$

To obtain its density the case hardened steel is chosen to be used in cutting the pinion and it's density is $7870kg/m^3$.

Therefore, using the relationship:

$$\text{Density} = \frac{\text{mass (m)}}{\text{volume (v)}}$$

$$P = {}^{m}/_{v} = M = v$$

mc 7870 x 0.000100936

= 0.7943kg.

weight of pinion is

$$F = mg = 0.7943 \times 9.81$$

$$= 7.79N$$

3.7.0 BEARING SELECTION

To select the bearing needed, some parameters have to be determined as:-

(1) Radial load Fr [Ref]: [PSG TECH]

$$Fr = \frac{P}{2\pi nr} \quad \text{.......................................} (8)$$

Where P = Power transmitted = 0.8kw = 80W

n = Speed of rotation per minute = 30 r.p.m .

r = Radius of the shaft = 0.0145m

$$Fr = P = \frac{80}{2\pi\ 30 \times 0.145} = 551.24N$$

ii. The equivalent load acting on the bearing.

$$P = (X\,Fr + Yfa)s \quad \text{..............................} (9)$$

Where P = Equivalent load

X = radial factor = 1

Y = thrust factor = 0

S = service factor = 1.5

Fa and Fr as defined above

$P = (1 \times 5517.24 + 0 \times 160.5.34)\ 1.5$

$P = 82 + 5.80N$

iii. The life expectancy in hour Lh

$Lh = \frac{16666}{n} \times \left(\frac{C}{P}\right)^T$ [Design Data Psg Tech] .... (10)

where Lh = Required life of bearing

n = speed of rotation in rpm = 30rpm

c = Dynamic load capacity

p = equivalent bearing load

T = 10/3 for line contact.

Therefore, to obtain the value of (c/p) from standard table i.e revolution per minute (rpm) against life in hours L.

For this type of Agricultural machine i.e harvester the life in hours is between 4000 - 8000hrs.

Then an average is taken which is 6000hrs.

Therefore, c = 2.212

Substituting the value into the equation:

$Lh = \frac{16666}{30} [2.12]^{3.33}$ .... (11)

$Lh = 6782.97$ hour

v. The required line of bearing in million revolutions (L).

L = No. of rev/sec of shaft x operating life of bearing

L = 0.5 rev/sec x 6782 . 79 hour

L = 3391 . 39 mv

vi. Dynamic capacity of bearing

$$L = \frac{(C)^n}{(Fv)} \quad \text{.......................................} \quad (12)$$

Where n = 3

L = dynamic capacity

Thus $C = \frac{L1}{n} \quad Fr$

$C = (3391\,.\,39)^{1/3} \times 5517.24\ N$

$C = 80676\,.\,45N$

But in Kgf the dynamic capacity is $\frac{80676\,.\,45}{9.18} = 8223.89$ Kgf

So, the bearing corresponding to the value obtained above ISI No 30 BC 03 of number Skf 6306.

(Design Data PSG Tech).

3.1.9. KEY DESIGN

The shear stress acting on the key is given as:

$$Ss = \frac{F}{BL} = \frac{Fr}{bLr} = \frac{1}{bLr}$$ [machine design S.I. metric edition].

Where	F	=	force acting on the shaft
	b	=	the width of the key
	L	=	the length of the key
	r	=	the radius of the shaft
	T	=	the torque acting on the shaft.

The width of the flat and square key is normally one-fourth the diameter of the shaft i.e.

$^1/_4 \times d = b$

Where d = diameter of the shaft

Thus, from the above equation, substituting:

$$b = {^1/_4} \times 29mm = 7.25mm$$

for a squate key width (b) = height = 6.75mm

To obtain the length of the key

$$Ss = \frac{T}{bLr} \quad \ldots\ldots (1.3)$$

$$T = Ss\ bLr$$

For the torque that a shaft of diameter d and transmit, the values are to be reduced by 25% due to stress.

Therefore, $T = \frac{0.75\ \pi d^3\ Ss}{22} = 15$

Equating equation (14) and (15) it becomes: $0.75\ \pi d^3\ Ss = 16Ss\ Lbr$

$$L = \frac{0.75\ \pi d^3}{16br} - (23)$$

Substituting the values:

$$L = 0.75\ \pi\ (0.029)^3$$

$$= 16 \times 0.00725 \times 0.0145$$

$$L = 0.03416m$$

$$L = 34.16mm$$

To obtain shearing stress acting on the shaft from equation (19) i.e.

$$Ss = 25.47$$

$$0.00725 \times 0.03416 \times 0.0145$$

$$Ss = 7092593.57N/m^2$$

To calculate the compressive stress:

$$Ss = \frac{F}{(h/2)L} = \frac{F}{(h/2)Lr} = \frac{T}{(h/2)Lr} \quad -(16)$$

As stated above, height of square key = bradth (b)

$$\text{Then} \quad Sc \quad \frac{T}{(h/2)} = \frac{25.47}{\frac{(0.00725)}{2} \quad (0.03416 \times 0.0145)}$$

$$Ss = 14185187.14 \text{ N/m}^2$$

From equation (16), the torque capacity of the key in shear:

$$T = \frac{0.75\ \pi D^3\ Ss}{16}$$

$$= \frac{0.75ay\ (0.029)^3 \ \times \ 709253.572}{16}$$

$$T = 25.47\text{Nm}$$

The torque capacity of the key in compression is:

$$T = Sc \ \frac{(t)}{(2)} \quad Lr$$

where t = thickness of the key

$$T = 141815187.14 \ \frac{(0.00752)}{2} \ \times \ 0.03416 \times 0.0145$$

$$T = 25.469\text{Nm}$$

4.0. ANGLE OF TWIST (θ) OF THE SHAFT

Design of shaft for torsional rigidity is based on permissible angle of twist. The amount of twist permissible depends on particular application, and varies from 0.3 deg/m for machine tool shafts to about 3 deg/m for line shafting.

Angle of twist $\theta = \frac{584 M_t L}{G d^4}$ for solid circular shaft.

Where L = Length of shaft = 880mm

Mt = Torsional moment = 25.47N

G = Torsional modules of elasticity

= $80 \times 10^9 N/m^2$ (80GPa)

d = diameter of shaft = 0.029m

$$\theta = \frac{584 \times 25.47 \times 0.65}{80 \times 109 \times (0.029)^4}$$

$$= \frac{9668.412}{56582.48}$$

$$\theta = 0.170^\circ \text{ cm} = 0.1708 \text{ deg/m}$$

Therefore the angle of twist of the torsional rigidity is within the allowable ranges, since permissible varies about 0.3 deg/m for machine tool shaft to about 3 deg/m for line shafting.

41.0. KINEMATICS OF CUTTING BLADES

DESIGN CONSIDERATION:

The knife of the harvester performs a motion consisting of a translation motion together with the machine and reciprocating motion relative to the machine. The reciprocating motion of the knife, with the crank rotating uniformly, is directed perpendicular to the line of travel of machine and presented by a simple harmonic motion which is defined by second order differential equation as:

$$\frac{d^2x}{dt^2} = -\omega^2 x \quad \text{(14)}$$

where: $\frac{d^2x}{dt^2}$ = acceleration of the knife moving along the X - axis .

X: = position coordinates

ω: = angular velocity of the crank.

It follows that the relative movement of the knife can be taken to movement of the crankpin projection corresponding to the circumferential travel of crankpin. Thus knife stroke can be considered as:-

$$S = 2r \quad \text{.......} \quad (15)$$

Where S = Knife stroke

r = Crank radius.

The knife speed is variable therefore was varied from zero to maximum within one revolution of the crank.

Average volume of the knife speed is express as:

$$Var = {}^{5}/_{60} = \frac{4r\,n}{60} = \frac{r\,n}{15} \quad \text{.......} \quad (16)$$

where n = crank speed in (rpm).

(Singh, 1978)

Kinematics of Blade During Cutting:

Studies on forage harvesters are mainly confirmed to time out put and performance efficiencies. The actual cutting energy requirement for forage crops, although so important, has been neglected in most studies. Without the optimum energy requirement of forage crops, it is hardly possible to design and efficient forage harvesting machine (yumnam, J et al 1991).

In general, the energy consumed in shearing stems is normally less than 3% of total energy where as chopping process utilises about 35% and crop acceleration and conveyance normally consume more than 50%. Thus energy consumption can be greatly reduced if crop acceleration and conveyance energy requirement can be reduced. The cutting energy requirements of forage crops is mainly affected by two factors, namely, physical and mechanical properties of plant stem and the cutter head parameters.

To minimise the cutting enery requirement, a detailed study of cutting mechanies is essential. Various works done by different research workers are reviewed and attempts are made to correlate the research findings for a better understanding of energies of forage chopping.

The kinematic relations obtained during cutting by the interaction of blade and plant stem is illustrated in fig. 6.

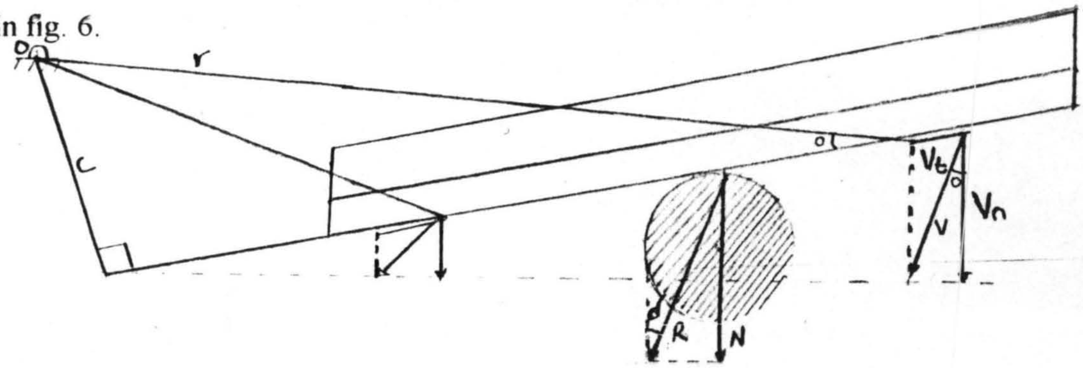


Fig. 6 Kinematic relations in cutting.

A given point on the cutting edge of blade rotates around point 'O' with a velocity 'V' which has tangential and radial components as V_t and V_n, respectively.

The angle of slide, 'Θ' is given by:

$$\tan \Theta = V_t/V_n = \frac{C}{\sqrt{(r^2 - C^2)}}$$

where:

tan Θ is called sliding coefficient. The peripheral force, 'P' is given by

$$P = P_1 + P_2 = N \cos \Theta + T_{\sin} \Theta \quad \text{.................................. (1)}$$

N = Normal force acting to the cutting edge.

Also,

N = P.1 and, T = μ.N

Where:

P = Specific cutting resistance per cm

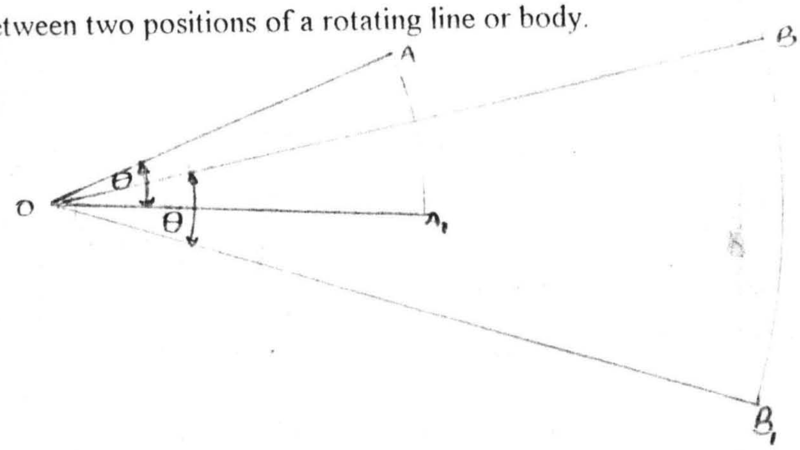
μ = Frictional coefficient

Substituting these values in equation (1), we have peripheral force, 'P' as,

$$P = P.1 (\cos \Theta + \sin \Theta)$$

4.1.1. ANGULAR MOTION

The relationship between linear and angular. Angular displacement (θ), is the angle between two positions of a rotating line or body.



OAB rotates about O to the position

O A, B, OA turns through angle AOA, = θ.

OB turns through angle BOB, = θ.

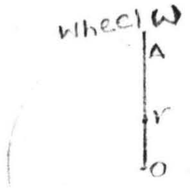
All lines of a body turn equal angle.

Radian

A radian is an angle subtended by an arc of a circle equal to in length to the radius of circle. Since the circumference of a circle is equal to $2\pi r$, there will be 2π radian in 360°

360° = 2πrad = 2.3142 = 6.28rad.

Relationship between linear and angular displacement.



Unit	Deg	Rad	Grad
1 rev	360°	2 π	400
½ rev	180°	π	200
1/4 rev	90°	π/2	100

Any point on the wheel ω such as A, remains at a constant distance 'r' from the centre O and travels a circular path as the wheel (flywheel) turns, when 'W' makes a complete revolution the angular displacement of line OA will be 360° or 2 πrad.

The point A will travel a distance equal to the complete circumference of radius r. i.e. 2 πr.

The path of A will always be proportional to the angular displacement of OA.

Therefore, $\dfrac{\text{path of A}}{\theta} = \dfrac{2\pi r}{2\pi}$

OR distance OA $= R_{cv}$

which also means that path of point A = r θ = A'

Distance travelled by point A = radius X angular displacement.

When wheel ω turns clockwise with an angular velocity ω, the line OA has the same angular velocity as ω and point A travels a circular path of radius r = OA.

A has no motion away or towards center O (fixed axis) and therefore has no velocity along OA. The velocity of A is tangential to the circular path.

At any given instant $V_A = \frac{\Delta s}{\Delta t}$

and angular velocity $\omega = \frac{\Delta \theta}{\Delta t}$

as delta $\Delta\theta$ and Δs approaches zero,

$\Delta s = r\Delta\theta$, $V_A = \frac{r\Delta\theta}{T}$

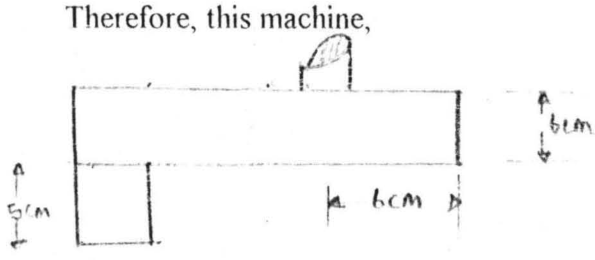
So $V_A = r\omega$

$$\boxed{V = r\omega}$$

To determine the angular velocity of a body when the linear velocity of a point and distance from the axis are known:

$$\boxed{\omega = {}^{v}/r}$$

Therefore, this machine,



Speed of rotation	=	30rpm	=	0.5 rad/S
Con rod	=	5cm		
radius	=	6cm	=	0.06

$$\omega = 2\pi N \text{ rad/S}$$
$$= 2\pi\, 0.5 \text{ rad/S}$$
$$= 3.142 \text{ rad/S}$$

$$\omega = \frac{v}{r}$$

$$\therefore V = \omega r$$
$$= 3.142 \times 0.06$$
$$= 0.18852 \text{ m/s}$$
$$= 0.2\text{m/s}$$

4.1.1 DESCRIPTION/PRINCIPLES OF OPERATION

The machine is pushed forward by the operator at a forward average speed of about 2.5. km/hr while rotating the drive handle at 30r.p.m.

The rotary motion received from the bevel gear set is converted into reciprocating motion or straight - in - line motion as a result of the connecting rod or pitman which is keyed eccentrically to the flywheel.

The flywheel further smoothens the variation in the rotary drive shaft and the inertia effect of the reciprocating blades.

The movable blade cuts the standing crop through a shearing force against the fixed blade. As the cut is made, the crop falls in a windrow by the sides of the machine as there is an attachment (two sided cover) mounted above the blades whose sides slopes at 7°. The cut crop falls on the sides and roll to the ground orderly. Instead of having it (crop) scattered all over the place with the crop lined up in a windrow or loose swath, it is subsequently collected and tied up in desirable bale sizes.

The height of cut is determined by the operator by tilting up or down depending upon the variety of crop and where there are undulating land.

The factors affecting the effectiveness of this harvester are:

1. Moisture content of crop
2. Speed of rotation of drive handle

3. Forward speed of operator.

4. Topograph of land

By its design features, the machine should harvest all grass-like crops and soyabeans.

4.1.2. POWER REQUIREMENTS

To determine the power used for or transmitted by a machine, three things must be known: the force, the distance through which the force acts, and the time required. It's unit is the Newton meter per sec (watts).

Power is defined as the rate of doing work. When work is accomplished, a force must act through a distance. The words "work" and "power" are often confused or inter changed. The term power adds the idea of time to work done. Power means the speed, or time - tate, of doing work.

Work = force x distance or W = F x D

$$\text{Power} = \frac{\text{work}}{\text{time}}$$

$$\text{power} = \frac{\text{force x distance}}{\text{time}} \text{ or } P = \frac{F \times D}{T}$$

$$\text{since} = \frac{\text{Distance}}{\text{Time}} = \text{Velocity}$$

power = force x velocity

therefore for this Acha harvester, the power requirement is:-

$$\text{power} = \frac{F \times D}{T}$$

given:

Force = 0.08kw (Human factor in Engineering)

Distance (a) 28cm from drive handle to bevel gear wheel = 0. -28m

(b) 114cm from pinion gear to pitman = 0.114

$$\text{Therefore: } P = F \times D = 0.08 \times \frac{(0.028 \times 0.114)}{360}$$

$$= \mathbf{0.189\ watts}$$

4.1.3. MACHINE CAPACITIES

Measures of agricultural machine performance are the rate and quality at which the operations are accomplished (Donell, H. 1973). Rate is an important measure because few industries require such timely operations as agriculture with it's sensivity to season and bad weather.

The following areas of machine performance should be borne in mind during any field operation(s).

a. **Capacity**: This is expressed only as area per time, is usually not a sufficient indicator of a machine's time performances, particularly with harvesting machines.

Differences in crop yields and crop conditions can mean that one machine have a low area per hour capacity but a high mass per hour capacity when compared with an identical machine in a different field. In this case a valid comparative capacity would be mass per hour.

The concepts of mass and weight must be understood for confidence in expressing machine capacities and crop yields.

Mass is to be thought of as the substance of a body that resists acceleration and is attracted to the mass of the earth. A body will accelerate rapidly towards the center of the earth unless restrained. This restraining force is equal to the body's weight.

The relationship between mass and weight is:

$$F = ma$$

where: F = Force acting on the body

M = Mass of the body

A = Resulting acceleration in units of distance / S^2

When acceleration is caused by the earth's gravitation attraction, the term, a, is labeleed g and the force, f, is called weight.

Therefore, the formula to calculate machine capacity is:

$$M = \frac{Swey}{C}$$

where: M = Material capacity units per hour.

y = Yields, unit/area.

e = field efficiency as a decimal

w = rated width of implement in (m)

c = constant, [10].

A more precise mathematical statement of field capacity for normally operated machines (without breakdowns or unexpected stoppages) can be developed without reference to such as all-inclusive and general term as full efficiency.

$$C = \frac{S_w \; L \; E_w}{(c1)\,L + D\,S_w\,L\,E_w + (c2)\,St}$$

where: C, S, W have the same units as in field efficiency.

E_w	=	effective swath coverage, decimal of rated width.
D	=	unproductive time, (hr/ha)
L	=	length of field, (m)
t	=	turning time, (s/turn)
c1	=	constant, (10)
c2	=	constant, (2.7778)

This formula applies when only turning time swath overlap, and such area related times as filling seed boxes, unloading grain tanks, or unhitching yeild - collecting wagons detract from machine performance. A rectangular field with head land is assumed.

b. **Field Capacities**:

i. Theoritical field capacity: This is the rate of work that would be achieved if a machine were performing it's function at full rated forward speed for 100% of the time.

ii. Effective field capacity: This is the actual rate of work, usually expressed in hectares per hour ie.

$$C = \frac{Swe}{C}$$

where

C	=	effective capacity, (ha/hr)
S	=	speed (km/hr)
w	=	rated width of implement(m)
e	=	field efficiency as a decimal
c	=	constant (10)

Therefore for this harvester, the field capacity is given as:

s	=	2.5. km/hr
w	=	50cm = 0.05m
e	=	?
c	=	$\frac{Swe}{c}$

$$c = \frac{2.5 \times 0.5e}{10}$$

$$10c = 2.5 \times 0.05c$$

$$e = \frac{10}{2.5 \times 0.05}$$

$$e = 80\%$$

$$c = \frac{Swe}{c}$$

$$= \frac{2.5 \times 0.05 \times 80}{10} = 1 \text{ hec/hr}$$

Therefore

Machine capacity will be 1hec/hr.

The factors that tend to affect the field efficiency of any agricultural machine are:-

1. Theoritical capacity of the machine
2. Machine manuevrability
3. Field shape
4. Field pattern
5. Field size
6. Yield (if a harvesting operation)
7. Soil and crop condition
8. System limitation.

4.1.4. MAINTENANCE

Maintenance is the process of up keeping a tool or a machine to a standard or acceptable value of performance, by effecting repairs, lubrications and replacement of parts where necessary or appropriate.

For this machine, the parts to be maintained are:-

a. Bearings
b. Moveable flap
c. Fixed and reciprocating knives
d. Wear plate
e. Spring loaded clamp
f. Bevel gear system.

A. **BEARING**

The bearings should be smeared with grease every three days for smooth operation or rotation of the drive shaft and to reduce undue wear, rust and corrosion of the members. Defective bearings should be replaced.

B. **MOVEABLE FLAP**

The moveable should be lubricated on daily basis whilst in operation especially along the line of movement.

C. **FIXED AND RECIPROCATING KNIFE**:

The cutting edge of both fixed and moveable knives should be sharpened when blunt and be checked frequently during operation for bending when it strikes on obstruction. Grinding is the most effective way of sharpening the fingers; for accurate work, they must be removed from the clamp and flap.

D. **WEAR PLATE:**

Check for wear on the surface and the forward and backward adjustment for even wear on the surface.

E. **SPRING LOADED CLAMP:**

The tension of spring on the clamp should be maintained and lubrication of the ball bearing in desirable.

F. **GEAR SYSTEM:**

To obtain the proper service and maximum life, the gears must be generously lubricated with a proper lubricant.

Check for abrasion and wear, tooth failure, pitting. If these aspects are serious, replace the entire system.

Give the entire machine on end - of - season attention by cleaning, lubricating, replacement of defective components and store under cover against the next cropping season

3.7.1. **ASSEMBLING OF MACHINE**

The assembling is done with reference to the exploded view shown in fig. 2.

STEPS

1. Weld frame containing the bevel gear system to the main frame centrally.
2. Weld wear plate to moveable flap.
3. Fix the fixed blade to the base of the main frame platform.
4. Fix the reciprocating blade over the fixed blade to be held in position by the screws on the wear plate and spring loaded clamp.
5. Fix the wheel shaft under the main frame.
6. Fix the land wheel snuggle through the shaft.

CHAPTER FOUR

5.0. COST ANALYSIS

5.1.0. MANUFACTURING COST

Generally, cost for manufacture are broadly classified by behaviour, element and functions cost by element is the material, labour and over head, while cost by behaviour is asking whether the cost is fixed or variable and cost by function requires the purpose of spending an amount.

The cost of manufacturing the machine will be divided into three major cost thus:

i. Material cost

ii. Labour cost

iii, Overhead cost

The material cost is a function of its own weight, material type and the shape has on supply. In other words, the cost of material is proportional to these elements. This fabrication involves the use of Bevel gear, iron rods, flat mild steel sheets, pipe, screw nuts and bolts, bearings/ bushings.

The table below shows the cost of the materials used for fabrication. This can be stated that the price listed were valid as at the time of costing and fabrication but are subject to changes

Table 1 Details of Components, Materials and Specification used for Fabrication.

S/No.	Component	Materials	Specification	Qty	Price N : K
1	Shaft	Mild steel	29 x 650 and 29 x 330	1 pair	700
2.	Bevel gear and pinion	case-hardened steel	3:1	"	2,500
3.	Flap	mild steel	511.86 x 558.8	"	900
4.	Bolts/Nuts	mild steel	M8	6	200
5.	Electrodes		E12	20	300
6.	Frame	I.W.N sheet	G20 ISI No. 30 Bco3	1 pair	500
7.	Bearing		SKI 6306	5	750
8.	Handle	mild steel	10 x 420	1	100
9.	Pipe	Galvanised pipe	30 x 220	1	400
					6,350.00

LABOUR COST

This is considered as 20% of the total material cost which is :-

$$\text{Labour cost} = \frac{20}{100} \times \text{material cost}$$

$$= \frac{20}{100} \times 6350 = ₦1,270$$

Labour cost = ₦1,270

OVER HEAD COST (C.E)

The over head cost is 10% of the material cost for the fabrication expressed as:

$$0C = \frac{10}{100} \times \text{material cost}$$

$$= \frac{10}{100} \times 6350 = ₦635$$

0C = ₦635

TOTAL COST OF FABRICATION (Tc)

T.C. = Material cost + Labour cost + overhead cost.

T.C. = 1,270 + 635 + 6350

= ₦8,255.

5.1. COST EFFECTIVENESS

The purpose of mechanizing Acha harvesting is to exploit all resources as much as possible and perform the harvest operation at a minimum cost and time.

The cost per day in harvesting especially by fully mechanized machines e.g. combine etc can be extremely high. The machine is so fabricated that all members are seen during working operations.

This is cost effective in minimizing maintenance down time and ensuring that the machine do not deteriorate to a condition where emergency action is required.

Machines vibrate and according to their elements, those responsible for their up keep should pay allention to those reports and need their complaints.

The cost of harvesting labour (compensation per man/ hour) relative to manufacturing productivity (out perman/hour) has increased rapidly. Labour cost has been rising faster than manufacturing productivity. This could cost inflation of finished products. Therefore, with development and implementation of Acha harvesting machine, the number of manual labour are greatly reduced which in decrease the cost of labour.

CHAPTER FIVE

6.0. ADVANTAGES, CONCLUSION AND RECOMMENDATION

6.1.0 ADVANTAGES

The advantages of mechanizing Acha harvesting are as follows:

i. To increase production rate

ii. Reduce man power requirement/ drudgery

iii. Reduce grain losses

v. To save time.

6.1.1. CONCLUSION

i. With the emerging Technologies and needs for more food production, mechanization of Acha harvesting are of prime importance to flour processing industries, Hospitals and Hotels.

ii. The mechanization of harvesting does not merely refer to harvesting by machine but include effective harvesting without loss of grains.

The machine possess the following characteristics:

i. Simplicity in operation

ii. Safe to operate

iii. Easy access to all parts.

6.1.2. RECOMMENDATIONS

1. Further research and development should be intensified in the area of harvesting of tropical crops.

2. An alternative design of a power assisted harvester will also be another outstanding innovation in this area of research.

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