DESIGN AND FABRICATION OF A RICE REAPER

By

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DEDICATION

This project is dedicated to Almighty Allah, my parents Alhaji Shehu Musa and Hajiya Hauwa S. Musa, my wives Mrs. Amina A. Shehu and Mrs. Ramatu A. Shehu, my children, brothers and sisters for their support and encouragement.

APPROVAL PAGE

I declare that this work was carried out by Abubakar Alhaji Shehu in the Department of Agricultural Engineering. Federal University of Technology, Minna.

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ABSTRACT

Rice Harvesting had been and still remains a serious problem to the peasant farmers in Nigeria. The techniques for harvesting rice are still traditional based on using mainly hand tools. This method of rice harvesting is laborious, time wasting and uneconomical. Also, commonly available rice reapers are mostly imported and thus not affordable to majority of Nigerian farmers. Combining basic engineering design considerations with other requirements, a rice reaper was designed and fabricated. It was fabricated from locally available materials. The reaper consists mainly of cutting unit, the transmission unit, the frame and handle. The reaper cuts and conveys rice in vertical position to its right hand side. Test results show that the reaper has a field efficiency of 79.15% and a field capacity of 0.0338ha/hr. The reaper showed a remarkable improvement over manual method of rice harvesting.

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CHAPTER ONE

1.1 INTRODUCTION

Rice *(Oryza sativa L)* is an important cereal crop, which belongs to the gramineae family. Rice is a leading cereal-crop in many countries. It is grown on all the continents. The species oryza sativa was introduced into African countries long ago from Asia. It is the most widely grown species. However, there is also another species oryza glaberrima, which originated in Africa (Michael and Brigitte, 1987). According to Michael and Brigitte (1987), upland varieties include, moroberekan, IRAT13, OS6 and lowland varieties include Im16, and Gambiaka.

Of the world's rice crop, about 26.1 million tonnes (6.3%) are produced in the developed countries (FAO, 1982) the usefulness of rice is indeed universal because it is the stable food for almost half of the world's population. Man benefits not only from its starchy grain for food but also from other parts of the plant, and the by-products of its processing.

The grain with the hull, brain and germ removed by milling is used in dry cereals. The dry matter of milled rice contains 88% starch (amylose and amylopectin), 6 to 8% protein, 0.5% fats and 0.5% sugars. Because of its high digestibility (98%) and high nutritive value, white rice has become indispensable for use in baby and breakfast foods; and indeed for the sick. Rice is also used for the production of starch, alcoholic, beverages and soft drinks; Rice flour may be used as a blending material in baking white bread and in biscuits.

Rice hulls and polish that include bran, aleurone layer and the germ are used in the pharmaceutical industries for the production of phytin and vitamin B (thiamine, riboflarin, niacin). Rice bran and polish are fed to farm animals. It is also used for the production of high quality, rice oil used in medicine and in corrosive-resistant coatings. Chemical processing of rice chaff gives furfural, which is a basic raw material for the manufacture of plastics, (Sheruddin *et al*, 1991).

Rice straw is an important raw material for manufacture of high-quality paper; other products include cardboards, ropes, packing material, handbags, rugs, hats and sandals, sacks and baskets, brooms etc.

Growing rice facilitates the improvement of saline and alkaline soils which, after two or three years in rice, can successfully be used for growing other commercial crops to the benefit of the rice grower.

In spite of the enumerated importance of rice to human diet, its harvesting in Nigeria had been and remains a serious problem to the farmers. (Michael and Brigitte, 1987). The techniques for harvesting are still traditional based on using mainly sickle. The total labour requirement for traditional manual harvesting with sickle is 80-160 man-h/ha of which 60-100 man-h are used in cutting and laying the crop. The harvesting period is very short and crop losses increase rapidly with delay in harvesting. Delayed harvesting of mature crops also exposes it to many hazards like rains, windstorms and fires (Gajendra *et al*, 1988). Also due to rapid urbanization and migration of farm labour to cities a big vacuum has been created in the supply and demand ratio of farm labour. This paucity of labour force has been forcing farmers to go for mechanization (Sheruddin *et al*, 1991).

Based on the above reasons it is therefore necessary to design and fabricate a rice reaping machine which is less expensive but simpler in design to the imported ones.

1.2 OBJECTIVE

The objective of this project is to design and fabricate, from locally available materials, a rice-reaping machine that is affordable by the local farmers. It is to be of low cost without compromising functionality and durability.

The objective above is directed towards the following aims: -

(I) To encourage the peasant farmers to embark upon increased production of rice, and

(ii) To improve the production efficiency, time usage and output, while cost and drudgery are reduced.

1.3 JUSTIFICATION OF THE RESEARCH

(i) This machine will serve the rural farmers as an alternative to the traditional method of harvesting which is labour intensive and time consuming.

(ii) It will also reduce grain damage, seed losses, as well as improve the quality of harvested rice. This will encourage the peasant farmers to embark on large-scale production of rice.

1.4 LIMITATION OF THE WORK

This research work is limited to the design, fabrication and testing of a rice reaping machine that can harvest (cut and convey) rice grain to the right hand side of the machine.

CHAPTER TWO

2.0 LITERATURE REVIEW

Reaping is a very old harvesting concept that continues to challenge designers through the centuries. In Nigeria, less research has been done on rice reapers because of the over-dependence on the imported ones. Much research has been done in developed and some developing countries on mechanical harvesting of rice (Stickney *et al*, 1985).

2.1 PREVIOUS WORK

In the developing countries of South and South East Asia, nearly all paddy fields of small farmers are reaped manually by groups of labourers using knives or sickles (Stickney et al, 1985). Knives are used for cutting plant stalks or grain heads of crops like millet, sorghum and rice (Ian and Rodriguez,, 1992). It is efficient and cost effective, but labour requirements are very high.

A wide range of sickles is used to harvest the majority of cereals and pulses in developing countries. Basically they consist of a metal blade, usually curved attached to a wooden handle. The degree of curvature and length of blade, the angle of attachment, and the shape of the handle all vary from area to area. Sickles are cheap and efficient. Its disadvantages are that the labour requirement is high, time wasting and depend on the operators' ability.

Scythe is a curved blade, usually 700-1000mm long, connected to a long shaft in various ways, some allowing adjustment of the angle between the two for different crop conditions. The greater the angle, the more material is cut at each stroke, with more labour. Scythes are efficient harvesting tools, but require considerable skill to use properly.

For cereal harvesting, a cradle attachment collects the cut crop and allows it to be deposited at the end of the stroke. The most common arrangement is a group of four or five wooden fingers parallel to the blade. Paddy is not normally scythed because rice straw is softer and tougher than wheat straw and more prone to lodging.

Reaping, manually (cutting only, not hauling or threshing) requires a high labour input. Reaping hook is a compromise between a sickle and a scythe. It is short handled and has to be used in crouching position, but the sharp blade will cut the crop without having to hold it. Reaping hooks are frequently used with hooked sticks to gather the crop as it is cut.

Development of a vertical - conveyor reaper in China began in the early 1960's (Stickney et al, 1985). The advantage of this design compared with the conventional horizontal or inclined conveyor designs are lightweight, simple construction and ease of front mounting on small two wheel and four-wheel tractors to improve maneuverability in small fields. Improvements were made during the 1960's and 1970's. In the late 1970s, International Rice Research Institute (IRRI) engineers observe the Chinese reaper and are enthusiastic about its potential applications in the developing countries. This led to the engagement between CAAMS and IRRI to collaborate in modifying the design for small rice farms in South and South East Asia. In 1980, three Chinese engineers worked with IRRI engineers in the Philippines to develop a simplified reaper, which was lighter, less expensive and easier to fabricate with shop tools, materials and components in the Philippines and other developing countries [Stickney et al, 1985].

Efforts to develop a suitable wheat harvester in Pakistan started in the late 1970s. Adaptation work on the reaper in Pakistan did not start until after engineers from the Farm Machinery Institute (FMI) Islamabad, visited IRRI in the middle of 1981 and observed the 1-meter width machine in operation. This led to the fabrication of two prototypes of the Chinese vertical reaper in 1981. One was a power tiller mounted 1-m width machine much like the one adapted at IRRI. The other a 2m tractor front-mounted machine, was a redesigned version powered from tractor's power take off (P.T.O) Shaft. Both operate similarly. As the tractor moves forward and the reaper power is engaged, the inclined star wheels (driven by lugs on a flat conveyor) are vertical to the side. The tractors hydraulic system operates the lifting mechanism. The 2m widths were later increased to 2.2m (now the standard width for most tractor mounted reapers made in Pakistan) (Gajendra *et al* 1988).

Animal powered harvesters are relatively rare in developing countries. Ox-drawn reapers, based upon designs of machines used with horses, have been tried in India but are not used widely. The limited draught available from Ox-pairs, the problems of harvesting lodged crops and the cost of the machines compared to a tractor drawn harvesters have limited the commercial development of animal drawn harvesters.

2.2 CURRENT DEVELOPMENTS

The FMI reaper is a modified version of the Chinese vertical reaper (Gajendra *et al*, 1988). The reaper is mounted on the front of a 4-wheel tractor and driven from the tractor PTO by a long drive shaft. The crop is cut, transported vertically to one side and laid down to form a windrow. It has an average travel speed of 3.28km/h, working width of cut 2.01m, field capacity of 0.42ha/h, and field efficiency of 59.4%.

The AMRI reaper windrower was designed and developed by Agricultural Mechanization Research Institute (AMRI) at Multan. The AMRI reaper windrower is a tractor-mounted tractor's power takeoff (P.T.O) operated and hydraulically controlled machine. Its cutter bar is operated by P.T.O. shaft power through a pair of belt pulleys, a propeller shaft and a cam. An overhead reel is used to support and gather the crop being cut and to lay the same on the conveyor. While windrowing is done through a set of deflector wire.With an average travel speed of 3.01km/h and a working width of cut of 2.04m,it develops a field capacity of 0.37ha/h, field efficiency of 59.5% and fuel consumption rate of 3.21l/hr.

The Ittefeq reaper is a tractor front-mounted machine (Gajendra *et al*, 1988). It is a redesigned version of the Augostini cutter - binder without a crop binding mechanism. The power from the tractor's power takeoff (P.T.O) is transmitted to the front with the help of a V-belt and a drive shaft. The front end of the shaft is properly secured with the gearbox of the machine. Automotive power from the engine is transmitted to machine gear bringing the cutting blades into a to-and-fro motion through the belts and pulleys. The collecting hooks come into operation through chain and sprockets drive. These hooks, after collecting the

crop stalk, pass it on to the conveying channel, which piles up the stalk in a uniform layer or windrow on the left side of the tractors. It has an average travel speed of 2.63km/ha, width of cut of 2.06m; field capacity of 0.35ha/h, field efficiency of 64.6% and fuel consumption rate of 3.31l/hr.

Two-wheel reaper is a two wheel, pedestrian controlled reaper developed in Union Tractor Workshop, India. It has a cutter bar moving unit and a side stacker. It is powered by a 3.7kW air-cooled diesel engine and has clutch steering. It is suitable for row crops including rice, wheat and barley. The assembly can be altered for use as an animal-drawn light power tiller. Its cutting width is 1m, working width 1.25m, and weight 300kg.

The International Rice Research Institute in collaboration with the Chinese Academy of Agricultural Mechanization and Sciences (CAAMS) developed a motorized reaper (Ian and Rodriguez, 1992) comprises a reaper unit built onto a power tiller with a 2.2kW petrol engine and cage wheels but it is adaptable to the walking - type tractor units. It is of all-steel construction, except for non-metallic star wheel. It has walking width is 1m, it has an adjustable cutting height with forward speed of 2.5 to 4.5km/hr, work rate 2.4ha/day, fuel consumption 1litre/hour and total weight of 135kg.

Another reaper, side - stacking reaper was developed in China. This reaper can be used with an 11 to 18kW conventional tractor or with a 6 to 7kW walking tractor. It has crop lifters, a cutter bar and a raising/stacking device. It has an operator's seat. The working width is 1.6m with a stacking distance range of 2m, 2.5m and 3.2m. It has a total weight of 137kg.

Madho wheat-harvester is a machine produced from Madho - Mechanical Works, India (Ian and Rodriguez; 1992). This reaper fits onto the front of a tractor and leaves the cut crop-turned and laid to the right hand side. Its cutting height is adjustable, from minimum of 70mm. The reaper can be tilled back on its mounting frame into a transport position. It also has a working width of 2.2m.

Agrimec paddy reaper is a machine produced in Srilanka. It is similar to the side-stacking reaper, with a windrower to turn the crop to the right - hand side. The original design was an attachment for a 3kW walking tractor, but a self-propelled version is also available (Ian and Rodriguez, 1992). It has a working width of 1.2m and has total weight of 80kg.

Motorized reaper binders, are motorized reaper binder manufactured in Italy. It is a two-wheeled manually steered machine and is fitted with linked plate type wheels for operation in rice fields. It is powered by a 7.5kW - 10kW petrol or diesel engine. It has a working width of 1.3m and weight of 480kg.

Reaper binder is also produced in Italy. This is a ride-on unit with a single or double wheel trailed sulky-type driving seat. It has a 7.5 - 10kW, petrol or diesel engine and four forward gears plus reverse. The crop is cut by a reciprocating cutter bar, collected into sheaves and tied with string. The harvesting mechanism can have a manual or hydraulic list. Its working width is 1.4m.

Motorized two row reaper binders are also available. These machines were manufactured in Japan and Korea. It is a two-wheeled machine designed primarily for row-crops of rice, barley and wheat. It cuts two rows per pass, binding the cut crop into bundles and dropping then to the right - hand side of the machine. Jute or polypropylene string can be used for binding. An optional attachment accumulates the bundles and drops four or five at a time, to ease the subsequent gathering operation. The machine is carried on two wheels with low-pressure pneumatic types and is manually steered. It is powered by a 2.6 - 3.7kW, 4-stroke petrol engine and has six forward gears plus two - reverse. It has a working width of 550mm and a total weight of 165kg.

Four-row paddy reaper, is a manually - steered reaper which cuts four rows of rice simultaneously turning the cut crop to the side of the machine. This is fitted with a Kubota GS2JN, four-stroke 2.8kW petrol engine and has a working width of 1.2m. This reaper was developed in Indonesia at the Center for Development of Appropriate Agricultural Technology.

Strippers' harvesters of a different design are available (Price, 1989). They include stripper harvester for wheat, rice stripper harvester, the Chinese stripper, the Silsor stripper and others. The stripping techniques of most strippers are based on a longitudinal rotor principle. There stripping mechanism is composed of a divider, gathering system mounted on the front of the machine and a threshing chamber with a drum studded with wire loops. Most of these strippers are efficient, safe time and durable. However, they are equally very costly and not affordable to peasant or local farmers, high shatter losses and require skills for its operation.

The stripper system generally composes of pickup system for harvesting lodged crop, a draw type thresher to thresh the standing rice, a pneumatic conveyor system to provide air suction from reducing grain losses. Also most strippers gave high shattering losses and many plants cut by the beaters rendering the stripping action ineffective.

Mowers are primarily designed for cutting grass but may be adapted for use in harvesting Cereals like wheat (Kaul and Egbo, 1985). Two types of mowers are generally used: They are the cutter bar mowers and rotary mowers.

The cutter bar mower has two plates, one of which is stationery. The other plate moves backwards and forward against the stationery plate as the mower passes through the crop, thereby cutting the crop by shear action, rather like a pair of scissors. The two units that are stationery and movable plates are mounted on a plate called cutter bar. Again, as with a pair of scissors, for normal cutting action a particular clearance must be provided between the two plates. A moderate or good clearance is necessary to ensure good cutting. To ensure correct clearance between the two plates, the knife section is held against the stationery plate by knife clip. The sickle section is driven from the tractor power take-off by a gear system. The rotary motion from the power take off is converted to reciprocating motion for sickle by means of a crank and pitman attachment.

A rotary mower has one or more rotating blades mounted horizontally, and is generally used for cutting down weeds, stalks and brush. As rotary mowers are low in height they are commonly used to clear weeds under trees in orchards. These mowers are mostly imported; they are costly and thus not affordable to the local farmers. They also require skill for there operation.

The above machines are mostly sophisticated and very costly. Also the operator must have "the technical knowledge to be able to operate and maintain them and are not likely to be within the financial capabilities of the average farmers in Nigeria. Furthermore, the reapers are generally too large for waterlogged fields.

CHAPTER THREE.

3.0 METHODOLOGY

3.1 DESIGN CONSIDERATION

The reaper is a simple design compared to the imported ones. It is made up of the following major parts; the frame, handle, four wheel, belts and pulleys, cutting knives and conveyor chains see fig. 1 in appendix.

A rectangular frame was assumed. It forms the mounting support for all other units of the reaper. It is made up of mild steel flat bar of 50mm x 50mm x 6mm. The track wheels are adjustable and are bolted to the frame. The handle is made of galvanised hollow pipe. The cutting knives are made of harden steel, this is to prevent bending and edge dullness. This enhances the strength of the knives. The conveyor unit consists of two chain and sprockets; two shafts and these are mounted on the frame.

Determination of some properties of rice

The following properties of rice pertinent to the design of the reaper were determined; height of rice plant, height of rice grain and height of stalk of rice crop.

Table 1: Some properties of rice plant

S/N	Height of rice plant(cm)	Length of stalk of rice crop (cm)	Length of rice stem (cm)
1	110	90	20
2	114	92	22
3	102	80	22
4	95	70	15
5	107	89	18
6	98	78	20
7	116	93	23
8	86	56	20
9	65	50	15
10	111	91	21
Average	100.4	78.9	19.6

The result of preliminary studies on the rice plant pertinent to the design process is presented in Table 1 above. The above parameters were necessary and hence used in the rice reaper design process.

The machine was also designed to ensure that the overall layout, components, shapes, materials and dimensions provide adequate durability (strength), permissible deformation (stiffness), stability, acceptable corrosion and wear with stipulated service life and loads.

3.2 DESIGN APROACH

The principle of scissors like cutting is used in this design. The machine has scissors like cutter (stationery bar and moving bar). The two cutters are then coupled to the frame of the machine and supported by a slider crank mechanism through a connecting bar. The moving bar and the conveyor are powered by a 2.0hp engine. The engine help to transmit power (motion) to the slider crank mechanism and conveyor through a belt and through a shaft connected to the slider crank mechanism. The connecting bar attached to the movable cutter together oscillates to and fro on the stationary cutter. As the moving bar oscillates the two-conveyor chains also rotates and there by convey rice to the right hand side. The to and fro motion of the cutter cause a shearing action. See Fig. 1 in appendix.

3.3 Design Parameters and Analysis

3.31 Cutter bar and Slider crank mechanism Design

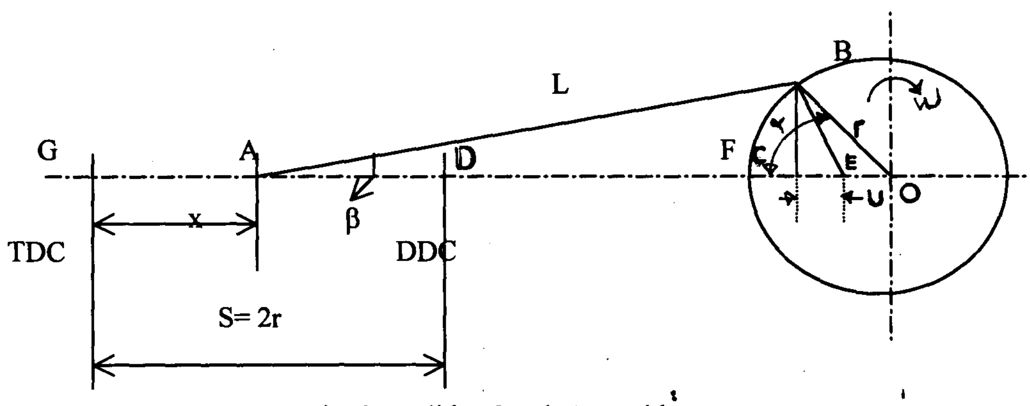

Fig. 3.1: Slider Crank Assembly

This is a mechanism that converts rotary motion to reciprocating motion.

L = Length of the connecting bar

r = Radius of the crank.

S = Stroke Length = 2r

$$h = r/L \quad \text{or} \sin \alpha \ r/L$$

W = 2π.n rad/s

α = Angle turned by the crank measured as from when it is at TDC.

TDC = Outer Dead Centre

DDC = Inner Dead Centre

β = Angle between the axis of the connecting bar and that of the stationary blade.

3.32 Kinematics of the Slider Crank Mechanism

1. Distance Covered by the Moving Blade.

If after time t seconds, the crank OB is turned α° then the connecting bar will be making an angle of β° with axis of the stationary blade. The moving blade will move a distance x to the right from outer dead center.

$$X = GO - AO = (GF + FO) - (AC + CO)$$

$$GF= L, \ FO= r, \ AC=L\cos\beta, \ CO=r\cos\alpha$$

$$\Rightarrow X = (r+L) - (r\cos\beta + L\cos\alpha) \quad \text{-------------} \quad (1)$$

To express x as a function of only one parameter α, length BC is obtained from triangle ABC and triangle BCO.

$$BC = r \sin\alpha = L \sin\beta$$

$$\sin \beta = r/L \sin\alpha = \lambda\sin\alpha \quad \text{-------------} \quad (2)$$

$$\text{Since } \sin^2\beta + \cos^2\beta = 1$$

$$\cos\beta = (1 - \sin^2\beta)^{1/2}$$

$$= (1 - \lambda^2 \sin^2\alpha)^{1/2} \quad ----- (3)$$

Expanding eqn(3) in Maclaurian series and considering the addition of the first two components, then

$$\cos\beta = 1 - \tfrac{1}{2}\lambda^2 \sin^2\alpha \quad ------------------ (4)$$

Substituting eqn(4) in eqn(1)

$$X = r + L - (r\cos\alpha - L(1- \tfrac{1}{2}\lambda^2 \sin^2\alpha)$$

To ease calculations substitute

$$\sin^2\alpha = (1 - \cos 2\alpha)/2$$

$$\Rightarrow X = r[(1 - \cos\alpha) + \lambda/4(1 - \cos 2\alpha)] \quad -------- (5)$$

$$\lambda = r/l = 75/100 = 0.25$$

$$\Rightarrow \sin\alpha = 0.25$$

$$\Rightarrow \alpha = \sin^{-1}0.25$$

$$= 14.5^\circ$$

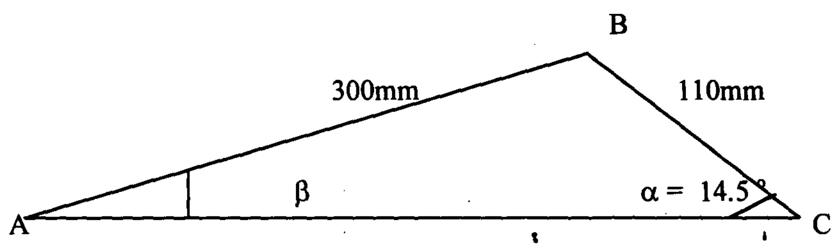

Fig. 3.2 : Slider Crank Analytical Nomenclature

From eqn (5)

$$X = 75/2[(1 - \cos 14.5) + 0.0625(1-\cos 2x14.5)]$$

$$= 37.5(0.0267) = 1.002m$$

2 Speed of the Moving Blade

Turning angle of the crank α is given by

$$\alpha = \omega t \quad \text{----------------} \quad (6)$$

$$X = r[(1 - \cos\omega t) + \lambda/4(1-\cos 2\omega t)]$$

differentiating with respect to time

$$V = dx/dt = r\omega\sin\omega t + r\omega\lambda/4\sin 2\omega t$$

$$V = r\omega(\sin\omega t + \lambda/2\sin 2\omega t$$

$$V = r\omega(\sin\alpha + \lambda/2\sin 2\alpha) \quad \text{------------------------} \quad (7)$$

$$= 75/2x2\pi n/60\ (\sin 14.5 + 0.25/2\sin 14.5)$$

n = crank speed taken to be 1000rpm (Kepner et al, 1978)

$$V = 75/2 \times 2\pi 1000/60(\sin 14.5 + 0.25/2\sin\ 2x14.5)$$

$$= 3.926 \times 0.3125$$

$$= 1.227m/s$$

Moving blade speed is zero at turning point and the maximum speed is not exactly at the center between the turning points but moved towards outer dead center. The lower is λ the closer the maximum speed is to points 90° and 270°.

3 Acceleration of the Moving Blade

Differentiating speed with respect to time t to obtain acceleration.

$$a = dv/dt = r\omega^2(\cos\omega t + \lambda\cos 2\omega t)$$

$$= r\omega^2 (\cos\alpha + \lambda\cos 2\alpha) \quad ------- \quad (8)$$

Where ω = angular speed

$= 2\pi n/60 = \pi n/30$

$a = 75/2(2\pi 1000/60)^2[\cos 14.5 + 0.25 x \cos 2x14.5]$

$= 411.23(1.4520)$

$= 597 m/s^2$

The maximum accelerations are obtained at outer dead center (TDC) and inner dead center(DDC). The absolute are different and the signs also differs.

When $\alpha = 0^0$, that is TDC, $a=r\omega^2(1+\lambda)$

When $\alpha = 180^0$, that DDC, $a= -r\omega^2(1-\lambda)$

4 Forces Acting on the Crank Mechanism

Forces acting on the crank mechanism is determined from

$$F = ma \quad ------------------ \quad (9)$$

where F= forces acting on the crank mechanism

m= mass of the moving bar = 1.34kg

a= maximum acceleration, m/s^2

$F = 1.34 \times 597$

$= 799.98N$

5 Power Required for Cutting

Power required for cutting is determined from

$$P = FxV \quad ----------------- \quad (10)$$

where P = power required for cutting, Kw

$P = 799.98 \times 1.227 = 981.58w = 0.982Kw$

with 90% power transmission efficiency,

the average efficiency = 0.982x0.9 = 0.884Kw

6 Power required for the Conveyor Chain

Power required for the conveyor chain for moving crop is assessed at 50% of cutting power (Devani and Pandey, 1985)

power required for conveyor chain is determined from

0.884 x 50% of the cutting power -------- (11)

= 0.884 x 0.5 = 0.442Kw

7 Total power required for cutting and conveying

the total power required is determined to be

cutting power + conveyor power

= 0.884 + 0.442 = 1.326Kw

From 1hp = 0.745Kw.

1.326Kw = (1.326 x 1)/0.745

≐ 1.78hp

⇒ A 2.0hp prime mover with 3600rpm is chosen for this design.

3.4 Belt Drive Design

Engine pulley diameter = 11cm

Slider crank (driven pulley diameter) = 14.5cm

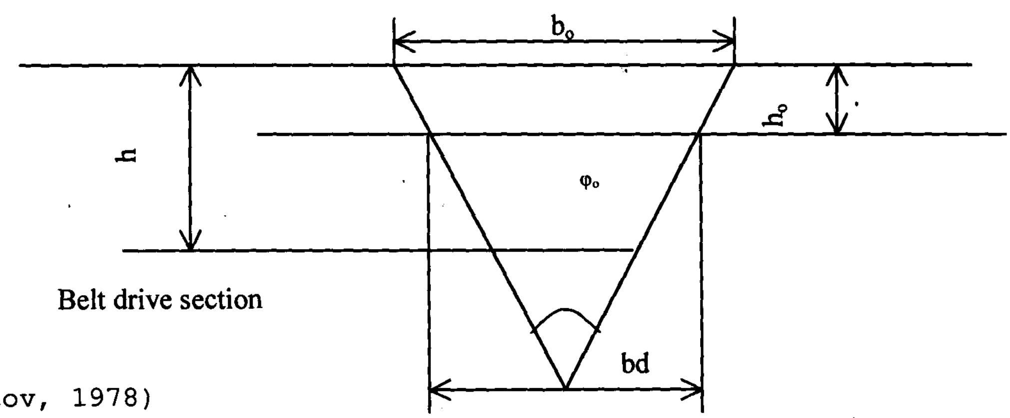

Fig. 3.3: Belt drive section

(Reshetov, 1978)

The peripheral velocities (m/s) on the pulleys (Fig.3.4 below).

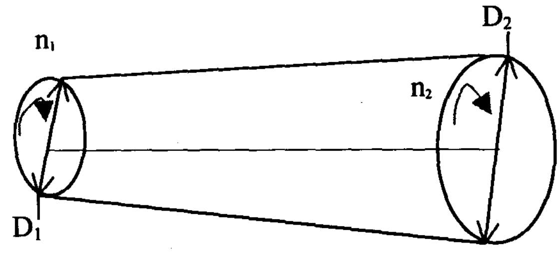

Fig. 3.4 The Peripheral velocities on the pulleys

From

$$V_1 = \frac{\pi D_1 n_1}{60} \quad \text{-------------------------------} (12)$$

And $$V_2 = \frac{\pi D_2 n_2}{60} \quad \text{-------------------------------} (13)$$

where

D_1 and D_2 are diameters of driving and driven pulleys (m)

n_1 and n_2 are the speeds of driving and driven pulleys (rpm).

From

$$V_1 = \frac{D_2}{D_1} = \frac{n_1}{n_2} \quad \text{-------------------------------} (14)$$

From

$$\frac{n_1}{n_2} = 1.32 \Rightarrow n_2 = \frac{n_1}{1.32} = \frac{3600}{1.32}$$

$$= 2727\text{rpm}$$

$$\therefore \quad V_2 = \frac{\pi D_2 n_2}{60} = \frac{\pi \times 0.145 \times 2727}{60} = 20.71\text{m/s}$$

Owing to inevitable creep, the peripheral velocity V on the driven pulley is less than the velocity V_1 on the driving pulley, thus;

$$V_2 = (1 - \xi)V_1 \quad \text{-----------------} (15)$$

Hence the true speed ratio is

$$U = \frac{n_1}{n_2} = \frac{D_2}{D_1(1-\xi)} \quad \text{-------------------} (16)$$

Where

U = speed ratio

ξ = creep factor

For the chosen cord core V-belts, the recommended creep factor is 0.01. (Reshetov, 1978).

Hence,

$$u = \frac{0.45}{0.11(1-0.01)}$$

$$= 1.33$$

Principal geometric relationship in belt drive.

The geometrical parameters to be considered are the angle (γ) between the sides of the belt, arc of contact (α) with the small pulley, belt length, L, and center to center distance a (since an endless belt is to be used). See fig. 3.4 below.

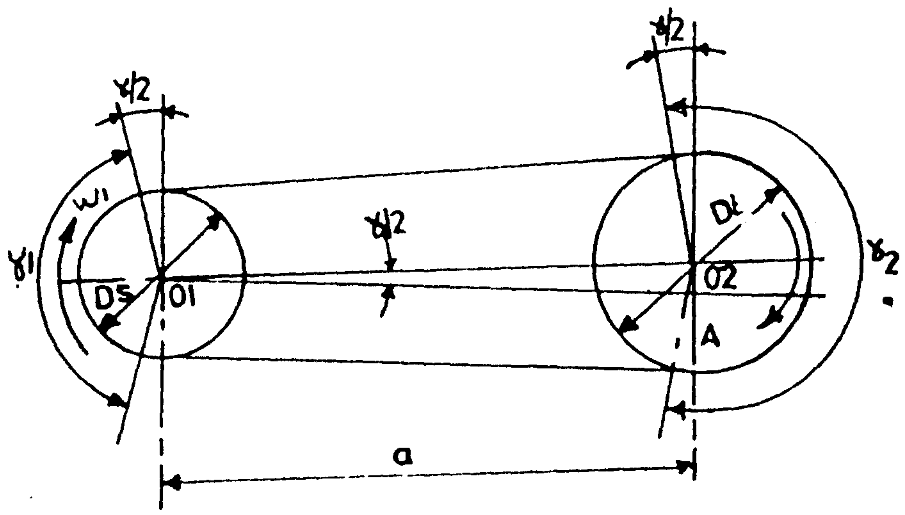

FIG 3.5 : Basic belt drive Geometry

The angle between the sides of the belt is determined from the auxiliary triangle OIAO2 (Fig. 3.4 above).

From $$\sin\frac{\gamma}{2} = \frac{\Delta}{a} \qquad (17)$$

But $\Delta = \frac{D_i - D_s}{2}$ -------------------------------- (18)

Hence, the angle between the sides of the belt.

$\gamma = 2\sin^{-1}(\Delta/a) \simeq 2\Delta/a$ ---------------------------- (19)

The arc of contact on the small pulley is

$$\alpha = 180° - \gamma° \simeq 180 - 2\Delta/a \times 57° \text{ -------(20)}$$

From eqn (20)

$$\Delta = \frac{D_i - D_s}{2} = \frac{0.145 - 0.11}{2} = 0.0175$$

The length of the stretch portions of the belt, thus can be calculated from

$$L = \pi\frac{(D_i - D_s)}{2} + \gamma\frac{(D_i - D_s)}{2} 2a \, \text{Cos}\gamma/2 \text{ ---------} \quad (21)$$

Transforming the above formular by using the approximate relationship

$$\text{Cos}\frac{\alpha}{2} \simeq 1 \tfrac{1}{2}(\gamma/2)^2 \text{ ------------------------} (22)$$

Substituting the value of γ and replacing by Dm, it becomes

$$L \simeq \pi D_m + \left\{2 + \left(\frac{\Delta}{a}\right)^2\right\} a \text{ -------------------------------} (23)$$

The distance between the axis of the pulleys for a selected stock length of belt is

$$a = \frac{L - \pi D_m}{4} + \tfrac{1}{4}\sqrt{(L - \pi D_m)^2 - 8\Delta^2}$$

But

$$D_m = \frac{D_L + D_S}{2} = \frac{0.145 + 0.11}{2} = 0.128$$

$$\therefore \quad a = \frac{1.5 - \pi(0.128)}{4} + \tfrac{1}{4}\sqrt{1.5 - \pi(0.128)^2 - 8(0.0175)^2}$$

$a = 0.2745+0.3001 = 0.575$

From $\gamma = 2\text{Sin}^{-1}0.0175/0.575 = 3.5°$

The arc of contact of the small pulley α is then

$180° - \alpha = 180 - 3.5 = 176.5°$

Forces and Stresses in V-belt drives

The peripheral force acting on the pulleys or useful load of a belt (KN) is

$$F = \frac{2KT}{D} = \frac{102KP}{V} \quad \text{------------------(2.4)}$$

(Reshetov, 1978)

where,
T = torque, kNm on a pulley of diameter D(m)
P = power transmitted in kW
K = dynamic load and service factor,

Considering the belt as continuous and homogeneous in calculations basing the later on the nominal stress.

The stress from the peripheral force F is

$$K = F/A \quad \text{------------------------------(2.5)}$$

where

A = x-sectional area of the belt in m^2

The dynamic load of the reaper is assume to be steady and thus have a dynamic load and service factor of 1

From $F = \frac{102KP}{V} = \frac{102 \times 1 \times 3.5 \times 0.746}{20.73} = 12.85kN$

Also, from the standard table; the area of the belt is equal to $0.47cm^2$

The force from the peripheral force F is

$$K = F/A = 12.85/0.47 = 27.33kN/\ cm^2$$

The effective pull was determined from

$$P = \frac{(T_1 - T_2)V}{1000} \Rightarrow T_1 - T_2 = \frac{1000 \times P}{V} \text{ ------------------(26)}$$

$$T_1 - T_2 = \frac{1000 \times 2.61}{20.73} = 12.59N$$

The torque acting on the driver sheave was determined from

$T_R = (T_1 - T_2)D_R/2$ --------------------------------------(27)

And on the driven sheave, the torque acting is

$T_N = (T_1 - T_2)D_N/2$ --------------------------------------(28)

Where T_1 = tight side tension, N

T_2 = slack side tension, N

$T_1 - T_2$ = effective pull, N

T_R = Driver sheave torque, Nmm

D_R = Driver sheave diameter, mm

D_N = Driving sheave diameter, mm

But

$D_N = 14.5mm$

$D_R = 11mm$

From eqn. (29)

$T_R = 12.59 \times 11/2 = 69.25Nmm$

From eqn. (30)

$T_N = 12.59 \times 14.5/2 = 91.28Nmm$

The allowable tension ratio relates the tight side tension to the slack side tension of a drive;

$$R_A = \frac{T_1}{T_2}$$

The allowable tension also for V-belts in V-sheaves is;

$$R_A = e\,x\,\rho\left(\frac{(0.5123)\theta\pi}{180}\right) \quad \text{-- (29)}$$

(Gary et al 1984)

R_A = allowable tension ratio
θ = arc of contact, degree
π = 3.14159 rad

$$\Rightarrow R_A = e\,x\,\rho\left(\frac{(0.5123)(176.5 \times 3.14159)}{180}\right)$$

$$= e\,x\,\rho\left(1.5 + 81\right) = 4.8$$

$$\Rightarrow R_A = 4.8 \simeq 5.0$$

3.5 CHAIN DRIVE DESIGN

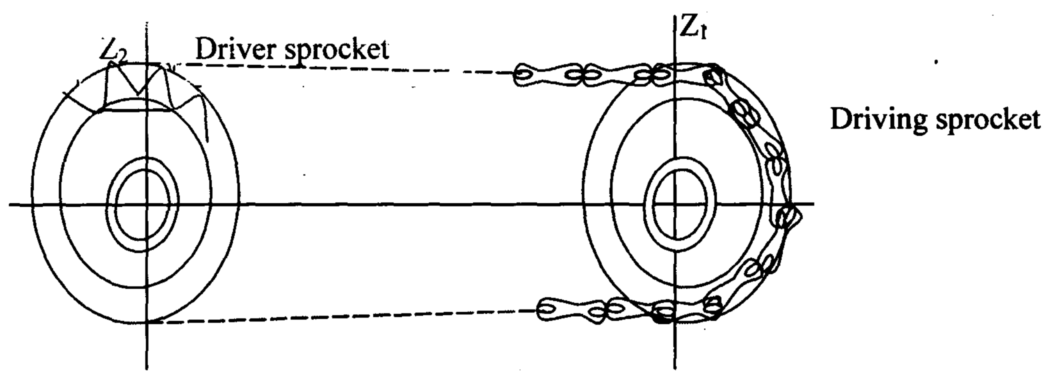

Fig. 3.5 Chain drive

The chain drive above consists of the driving and driven chain sprockets and a power chain which runs over the sprockets and meshes with their teeth (Figure3.5). It consists of links connected by joints, which provide for articulation or "flexibility" of the chain. The two set of chains and sprockets were designed to convey rice crop in a vertical position to the right hand side.

3.5.1 Velocity and speed of the chain

The velocity of chains and the speed of rotation of sprockets are limited by chain wear and noise made by the drive. The two aspects were put into consideration.

The velocity(V) was determined from

$$v = \frac{PN \times rpm}{376} \text{ -----------------------------------(30)}$$

(Gary *et al*, 1984)

where, P = chain pitch

N = number of teeth on driven sprocket

rpm = sprocket speed rpm (speed of relation of the sprocket).

$$rpm = \frac{v \times 376}{0.075 \times 32}$$

Assume chain velocity = 1.2m/sec (Reshetov, 1978)

$$rpm = \frac{1.2 \times 376}{0.075 \times 32} = 451 rpm$$

(Reshetov, 1978.)

Distance between the sprocket axes and the chain length.

The minimum center - to - center distance.

a_{min} is determined from $$a_{min} = \frac{D_1 + D_2}{2} + 50 \text{-----------------} (31)$$

where, D_1 + D_2 are the outside diameters of the sprockets, mm.

$$D_1 = D_2 = 155mm$$

$$\therefore\ a_{min} = \frac{155 + 155}{2} + 50 = 155 + 50 = 205mm$$

The optimum center-to-center distance is

$$a\ (30 \text{ to } 50) \text{-----------------------------------} (32)$$

(Reshetov, 1978)

$$a_{miax} = 80p \text{------------------------------------} (33)$$

$$\therefore\ a = 50 \times 7.5 = 375mm$$

$$a_{max} = 80 \times 7.5 = 600mm$$

The required number of links, W, of the chain is determined from the tentatively selected center-to-center distance (a), pitch(P) and the number of teeth N_1 and N_2 of the sprockets. $$W = \frac{N_1 + N_2}{2} + \frac{2_a}{P}\left(\frac{N_2 - N_1}{2\pi}\right)^2 \frac{P}{a} \text{-------------------} (34)$$

a = 375mm

but $N_1 = N_2 = 32$ and sides of the chains are parallel

$$W = \frac{32+32}{2} + \frac{2x375}{7.5} = 132$$

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

$$a = \frac{P}{4}[W - \frac{Z_1 + Z_2}{2} + \sqrt{W - \frac{Z_1 + Z_2}{2} - 8(\frac{Z_2 + Z_1}{2\Delta})^2}] \text{.......................}(35)$$

(Gary et al,1985)

$$= \frac{7.5}{4}\left(132 - 32 + \sqrt{132 - 32 - 0}\right)$$

$$= \frac{7.5}{4}\left(100 + 10\right)$$

$$= 206.65\text{mm}$$

The pitch diameter of the sprocket was calculated using the formula

$$P_D = \frac{P}{\text{Sin}[\frac{180}{N}]} \text{...}(36)$$

where, P_D = pitch diameter of the sprocket

P = chain pitch (mm)

N = number of teeth on the sprocket

$$P_D = \frac{7.5}{\text{Sin}\left(\frac{180}{32}\right)} = 76.5\text{mm}$$

The chain pull or working load was calculated from

$$Cp = \frac{100P}{V} \text{ ---------------------------------------(37)}$$

where

P = power, kW

V = chain velocity, m/s

Cp = chain pull or working load

$$\therefore \; Cp = \frac{100 \times 2.61}{12}$$

$$= 21.75N$$

3.6 Slider Crank Shaft design

Shaft design consist primarily of the determination of the correct shaft diameter to ensure satisfactory strength and rigidity when the shaft is transmitting power under various operating conditions.

Design of Slider crankshaft

It is a rotating member carrying the cam pulley and bearings. In its design bending and torsional failure were guarded against. The design is governed by the maximum shear theory.

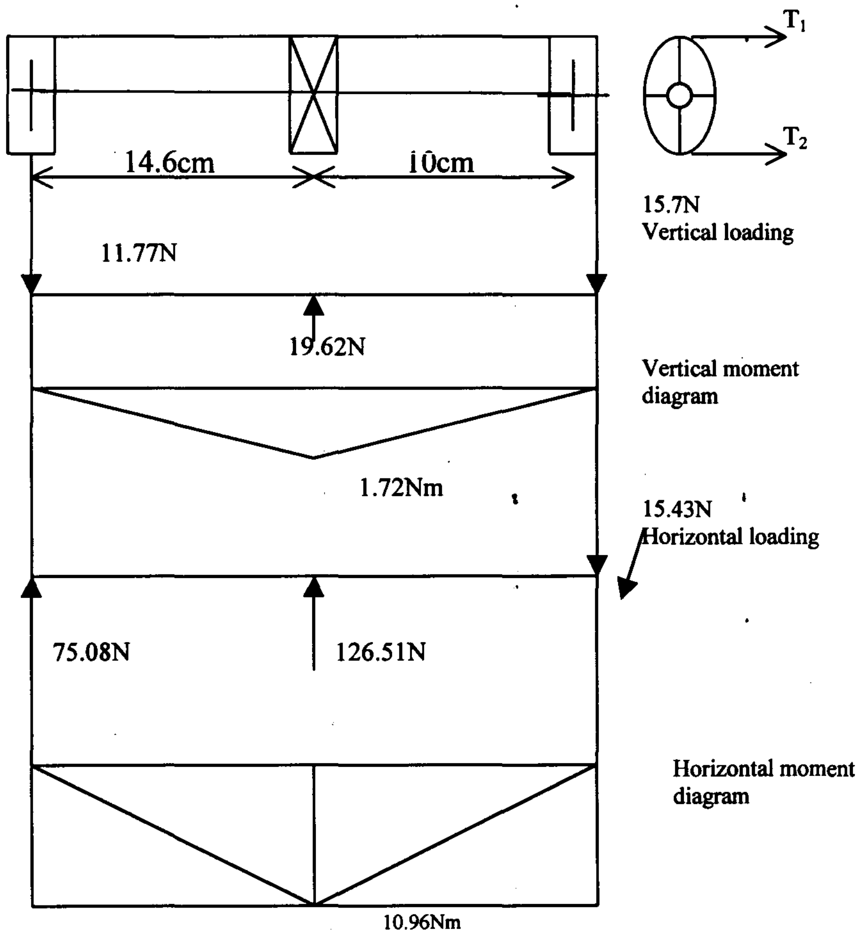

Fig.3.6 Loading on the Slider crankshaft

Bending moment (Vertical) = 11.17 x 0.146

= 1.430

Bending moment (horizontal) = 75.08 x 0.146 = 10.96Nm

Resultant bending moments

$$Mb(max) = \sqrt{Mb^2V + Mb^2h} \qquad \text{------------------(38)}$$

where Mb^2V = Vertical bending moment

Mb^2h = horizontal bending moment

$$\Rightarrow Mb(max) = \sqrt{1.63^2 + 10.96^2}$$

= 11.08Nm

Torsional loading

From $$T = \frac{P}{\omega} \text{ -----------------------------------(39)}$$

where

T = torque applied

P = Power transmitted

ω = Angular velocity

But

$$\omega = \frac{2\pi n}{60} \text{ --------------------------------(40)}$$

where n = revolution per minute

$$T = \frac{P}{2\pi n} \times 60 = \frac{2.61 \times 10^3 \times 60}{2\pi \times 3600}$$

$$= 6.92\text{Nm}$$

The diameter of the shaft was determined from the relationship

$$d^3 = \frac{16}{\pi S_S}\sqrt{(k_b m_b)^2 + (k_t m_t)^2} \text{ -----------------------------(41)}$$

(Hall et al, 1988)

where

d = shaft diameter
S_S = allowable stress for shaft with key way
k_b = Combined shock and fatigue factor applied to bending moment
k_t = Combined shock and fatigue factor applied to torsional moment
m_t = maximum torsional moment
m_b = maximum bending moment

Service Condition

In operation the machine could experience light shock because of the type of job it does, Assume,

Light shock factor for bending, m_b = 1.5 and Light shock factor for torsion, m_b = 1.5 for this design. *(Hall et al, 1988)*

Assumed design bending moment = $m_{b(max)}$ x 2

= 11.08 x 2

= 22.16Nm

Design torque $m_{t(max)}$ = $m_{t(max)}$ x 2

= 6.92 x 2

= 13.84Nm

S_S(allowable) = 55 x $10^6 N/m^2$ per ASME Code

From eqn (36)

$$\Rightarrow d^3 = \frac{16}{\pi x 55 x 10^6}\sqrt{(1.5 \times 22.16)^2 + (1.5 \times 13.84)^2}$$

$$d^3 = \frac{16}{\pi x 55 x 10^6}\sqrt{1535.99}$$

$$d = \sqrt[3]{0.000003629}$$

$$d = 0.0154m$$

$$\Rightarrow d = 15.4mm$$

The diameter was chosen to be 18mm

3.7 Conveyor shaft design

The conveyor shaft consists of two chains and sprockets, bearing and pulley. The design is o governed by the maximum shear theory.

Below is the loading on the conveyorshaft

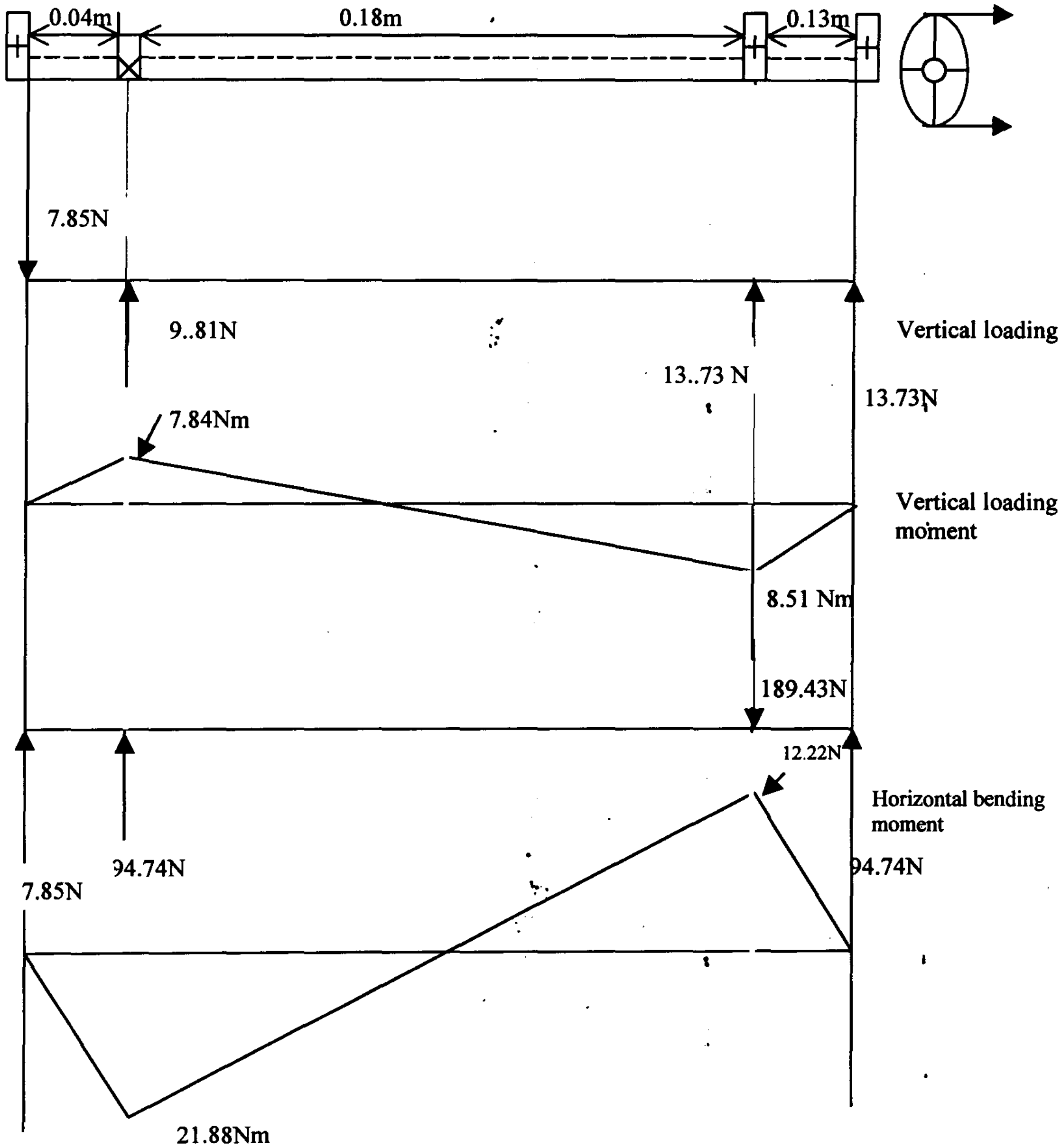

Fig.3.7 Loading on the conveyor shaft

Vertical bending moment = $m_b(V)$ = 27.46 x 0.31

= 8.51Nm

Horizontal bending moment ($m_b h$) = 94.71 x 0.13

= 12.22Nm

= 189.43 x 0.18

= 34.0974 - 12.22

= 21.88Nm

$$m_b(max) = \sqrt{mb^2V + mb^2h}$$

$$= \sqrt{8.51^2 + 21.88^2}$$

= 23.48Nm

From - eqn (34)

$$T = \frac{P}{\omega}$$

where,

$$\omega = \frac{2\pi n}{60}$$

$$T = \frac{P \times 60}{2\pi n} = \frac{2.61 \times 60 \times 1000}{2\pi \times 1880}$$

= 13.26Nm

Also from,

$$m_t = (T_1 - T_2)R \text{ -- }(42)$$

where

T_1 = tight side of chain on the sprocket

T_2 = loose side of chain on sprocket, N

R = radius of sprocket

where R = 0.14m

$\Rightarrow$ $13.26 = (T_1 - T_2)\, 0.14$

$$(T_1 - T_2) = \frac{13.26}{0.14} = 94.71N$$

$T_1 - T_2$ for the two chain drive will be

$$2(T_1 - T_2) = 2 \times 94.71$$

$$= 189.43N$$

Assumed design bending moment = $m_b(max) \times 2$

$$= 21.95 \times 2$$

$$= 43.9Nm$$

Design torsional moment

$$= m_t(max) \times 2$$

$$= 13.26 \times 2$$

$$= 26.52Nm$$

S_s(allowable) = $55 \times 10^6 N/m^2$ per ASME code

From eqn (36) was used to determined the diameter of the shaft

$$d^3 = \frac{16}{\pi S_S}\left[(k_b m_b)^2 + (k_t m_t)^2\right]^{1/2}$$

The terms are as defined in eqn (36)

$$d^3 = \frac{16}{\pi x\ 55\ x10^6} \quad (1.5 \times 43.9)^2 + (1.5 \times 26.52)^2$$

Assumed $\mathbf{k}_b$ = 1.5 and $\mathbf{k}_t$ = 1.5

m_t = 26.52Nm m_b = 43.9Nm

$$d^3 = 0.0926 \times 10^6 \quad (5\,918.66$$

$$\Rightarrow d = 3\sqrt{0.000007123}$$

$$= 0.0192m$$
$$= 19.2mm$$

Based on the above 18mm diameter was chosen

3.8 Bearing selection and design for the shaft.

Bearings are machine members or components, which permit, connected members to either rotate or translate relative to one another. Ball bearing was selected for this design, based on the type of loading, life requirement and the speed. Determining the radial and axial load did the analyses for the selection of the type of bearing.

From the formular

$$L^{10}h = \frac{10^6}{60(n)} \left(\frac{C}{P}\right)^6 \text{-----------------------------------(43)}$$

(*Reshetov, 1978*)

Where:

$L^{10}h$ = basic rating life in operating hours

Assumed 2000hrs   n = number of revs per min

= 3103rpm

C = basic dynamic load

P = VFr --(44)

where;   Fr = radial load

V = rotation factor = 1.0 for inner ring.

b = exponent for life bearing (for this design b = 3 for ball bearing)

$$\therefore\ C = \sqrt[b]{\frac{L^{10}h \times 60 \times n}{10^6}}$$

$$= \sqrt[3]{\frac{2000 \times 60 \times 3103}{10^6}} = 7.19N$$

From;

$$Fra = \sqrt{Ra^2V + Ra^2H} \quad \text{-------------(45)}$$

where;

RaV = Vertical load carry by the bearing
RaH = Loading in the horizontal direction

$$\therefore\ Fra = 14.715^2 + 9.81^2$$
$$= 17.69N$$

From

$$C = 7.19 \times V \times Fra \quad \text{---(46)}$$
$$= 7.19 \times 1 \times 17.69$$
$$= 127.19N$$

Since the loading is at two points, C = 254.74N

Based on the analyses, self-aligning ball bearing with the following parameters were selected

bearing with cylinder bore 6204Z

bore diameter(d) = 18mm

outer diameter (D) = 47mm

width(B) = 14mm

basic static capacity (C_O) = 655N

basic dynamic capacity (C) = 1000N

maximum permissible speed of 1600rpm

.9 Frame design

Frame supports the total weight of the machine components.

For axially and laterally loaded frame

$$\frac{F_c}{P_c} + \frac{F_{bc}}{P_{bc}} < 1 \quad \text{--(47)}$$

where

F_c = actual direct axial stress
F_{bc} = actual direct bending stress
P_{bc} = allowable bending stress
P_c = allowable axial stress.

But

$$F_c = \frac{F}{A} \quad \text{--(48)}$$

where

F = axial load
A = Cross – sectional area of the section

$$F_{bc} = \frac{m}{Z} \quad \text{--(49)}$$

where

M = moment
Z = Sectional modulus

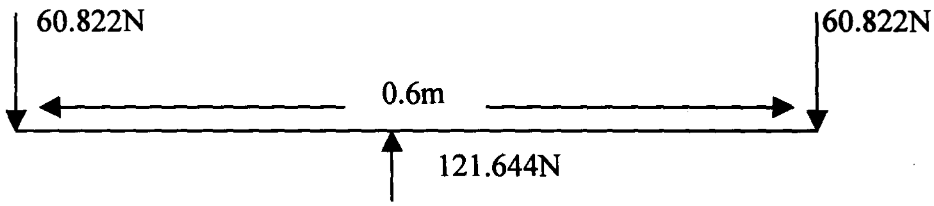

Fig. 3.9 loading on the frame

From

$$Fbc = \frac{m}{Z}$$

Assumed frame to be rectangular

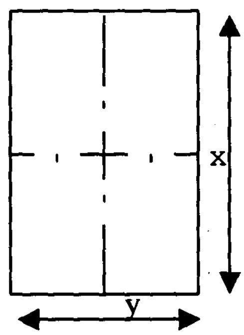

Fig. 3.8 Frame arrangement

y = 450mm
x = 600mm

M_{xx} = 121.64 x 0.3 = 36.49Nm
M_{yy} = 121.64 x 0.225 = 27.37Nm

$$Z_{xx} = \frac{bd^2}{6} \quad \text{---(50)}$$

$$Z_{yy} = \frac{db^2}{6} \quad \text{---(51)}$$

where;
b = width, d = depth.

From eqn (45)

$$Z_{xx} = \frac{0.45 \times 0.6^2}{6} = 0.027m^3$$

$$Z_{yy} = \frac{0.6 \times 0.45^2}{6} = 0.0203m^3$$

$$Fbc = \frac{36.49}{0.027}$$
$$= 1351.48N/m^2$$

From the above analysis

$$\Rightarrow \frac{Fc}{Pc} + \frac{Fbc}{Pbc} < 1$$

And since the sum above is less than 1, it implies that the frame is safe to carry the design loads.

CHAPTER FOUR

4.0 DESCRIPTION AND OPERATION OF THE RICE REAPER

The assembly drawing of the rice-reaping machine is given in Figure 2. The reaper was design and constructed to faithfully simulate the traditional manual rice harvesting technique. The rice reaper has the following major components; transmission unit, conveyor unit, the structural frame and the handle.

4.1 Description of the machine

The main features of the reaper are; the transmission unit, the cutting unit, the conveyor unit, supporting frame and operating handle. The assembly drawing and the overall dimensions of the reaper are shown in Fig 2 over leaf. The reaper cuts the crop and convey the crop in vertical position to its right hand side which can then be easily packed by the farm labourers.

4.1.1 Transmission unit

The transmission unit is the drive mechanism, which transmits power from the engine to the cutting blade. The unit includes; pulleys, V-belt, shaft, bearing, slider crank mechanism and connecting bar. Three different pulleys were used in the design of this reaper. The driven pulley is a double groove pulley with diameter of 14.5cm and the driver pulley of 11cm in diameter. One pulley is attached to the engine shaft, one double groove pulley attached to the slider crank mechanism and the third one attached to the conveyor shaft. The two V-belt was properly fixed to connect the driver from the engine to the slider crank mechanism and to the conveyor chain. The slider crankshaft was made of mild steel and of diameter 18mm and the shaft length of 246mm. Two bearings of series 6204z were mounted on the shaft at the lower end of the pulley to allow for free movement of the cam. Four more bearings of series 6204z were also mounted on the two-conveyor shafts to allow for sprocket and chain movement.

4.1.2 Conveyor Unit

Two vertical shafts of diameter 20mm were placed between the cutting units 460mm apart, one shaft is moveable at both ends and the other one is fixed at both ends. Each shaft has two sprockets (32t) and two complete chains (420t). Four cut metal plates of 5cm were welded to the two chains at four different positions, to enhance conveying of rice plant.

4.1.3 Cutting unit

The cutting unit consists of 5 gang stationary blades and 4 - gang moving blades. The width of each blade is 75mm. The blades are triangular in shape and made of mild steel.

The movable blades has four-gang, this is to allow for it's over lapping over the stationary blade when in operation. The stationary and the moving blades were bolted to a flat bar form a single gang. The stationary blades were welded to the structural frame in front of it, while the moving blades are bolted to the cam follower.

4.1.4 The Structural Frame

The structural frame forms the mounting support for all other units of the reaper. It is rectangular in shape and made of mild steel angle iron bar. Four main pieces of this frame of 600mm x 405mm was cut to form the rectangular frame. Welding using electric arc joined the cut pieces. The sizes of electrode used are sizes 10 and 14.

4.1.5 Handle

The handle was made from the galvanized hollow pipe. The handle provides the point of articulation for pushing the reaper from one place to another. The pipe diameter is 21.6mm; it is 120mm long, 41.5mm wide and 88mm high. The handle has two major parts; they are the short part and the longer part. The short part was welded to the frame and the longer part was bolted to the shorter part of the handle.

The Cover (Guide)

The guide serves as the housing, which guides the slider crank mechanism, crankshaft and the V-belt and also covers other components of the machine. The guide is made from galvanized iron sheet of 18mm gauge and bolted to the frame of the machine.

Engine Seat

The engine seat was constructed to withstand all the forces from the engine. The engine seat was constructed using flat bar 50mm x 3mm and welded to the frame.

Track wheel

The reaper constructed consists of four track wheels. This is to allow easy movement of the reaper in the farm. The track wheels are adjustable and they are bolted to the frame. The diameter of the track wheels is 190mm.

Bolts and Nuts

Series of bolts and nuts were used among which are 12mm, 8mm, 13mm diameter bolts and nuts. They are used in joining various components and it is to allow for adjustment replacement and easy dismantling of the machine as the case may be.

4.1.6 Machine Assembly

The rectangular frame was cut and welded together. The stationary blades gang was welded to the frame at the two ends. The moveable or reciprocating blades gang was placed on top of the stationary one and then bolted to the connecting bar. A guide was then provided to hold both the stationary and moveable blades. This guide was welded to the frame. Clearance was also provided to allow easy movement of the moveable blade gang or moveable cutter bar.

The eccentric turning slider crank and crankshaft was provided with two bearings to bring about easy movement of the slider crank. A double pulley was placed on the crankshaft and key to the shaft. The slider crank mechanism and crankshaft were bolted to the frame. The slider crank and the connecting bar were then connected to the pinion with the help of a washer, bolt and nut. This was done to serve as a guide to the connecting bar. The engine seat was then cut and welded to the frame. It was built in such away that the engine pulley is aligned with the crankshaft pulley. The V-belt of size A-38 was then placed between the engine shaft and the crankshaft pulley to transmit the power required from the engine to the slider crank. Two vertical shaft of length 460mm each were bolted to the frame at the two ends. Bearings, sprockets and two chains were then fixed to the shaft. Another V-belt was then placed between the crank shaft and one of the conveyor shaft, this helps in transmitting power to the chains and sprocket and thus aid the conveying of crop to the right hand side of the machine.

The four wheels were bolted to the frame this was to allow for adjustment. There are two wheels in front and two at the rear.

The handle was partially welded to the frame. As only the down part of the handle was welded to the frame and the upper part was bolted to the lower part so as to give room for adjustment. The cover (guide) was measured, cut and bolted to the frame Fig. 4.

4.2 Operation of the reaper

The reaper is a portable machine. The reaper operates on the principle of scissors - like cutting action. The machine has stationary and moveable cutter, a crankshaft through a connecting bar drives the moveable cutter. A petrol engine powers the clipping shears of the cutter. When the engine is started, the engine transmits power to the slider crank through a V-belt through the shaft connected to the crank. As the crank rotates the connecting bar, which is attached to the moveable cutter together oscillates to - and - fro on the stationary (fixed) cutter. This rotation of the crank now transmits power to one of the conveyor shaft through another V-belt. Thus, as the crank rotates and the blades reciprocates, the two conveyor chains and sprockets also moves in horizontally, this in turn convey the cut crop in a vertical position to the right hand side of the machine and the crop is finally collected manually.

CHAPTER FIVE

5.0 COST ESTIMATION

5.1 MATERIALS COST

The table 2 below shows the cost of materials for building the rice reaper at the time the project was carried out. The cost is as a result of a market survey carried out as at the time the project work was in progress.

S/N	Material specification	Qty	Unit cost (₦)	Total cost (₦)
1	Galvanised iron sheet G18	1	2,000	2,000
2	M.S. Shaft, 20mm ϕ	1	1,000	1,000
3	Two sets of chains and sprockets	2	600	1,200
4	M.S. Shaft, 80mm ϕ 200mm long	1	500	500
5	Ball bearings (6204z)	6	250	1,500
6	M.S. Flat bar 50 x 3mm	1	2,000	2,000
7	Angle iron (50 x 50 x 6)	1	800	800
8	G.I Pipe	1	1,200	1,200
9	Fasteners, bolts and nuts	1	1,000	1,000
10	Electrodes (gauge 10 & 14)	1 pkt	1000	1,000
11	Wheel tyres	4	500	2,000
12	Prime mover (petrol engine 3.5hp)	1	15,000	15,000
13	Miscellaneous (transport and others)			1,500
	Sub-total			30,700

Add 10% inflation for possible increase in price sub-total = ₦30, 700 + 3070 = ₦33770

The following assumption were made for adequate labour costing, with respect to labour situation in the country as at the time of construction of this machine.

(i) Engineer's fee - ₦500 per day ⇒₦62.5 per hour
(ii) Machinist fee - ₦250 per day ⇒₦31.25 per hour
(iii) Technicians/Welders charge.-₦200 per day ⇒ (₦)25 per hour
Assumed number of working hours per day is 8hrs

Table 3 Labour cost:

S/N	Unit/ Operation	Time (min)	No of Personal	Rate (₦)/hr	Total cost (₦)
1	Cutting unit				
	Cutter bar				
	blade				
	Marking out	30	1 Technician	25	12.5
	Cutting	60	"	25	25
	Welding	60	1 welder	25	25
2	slider crank				
	Eccentric turning	60	1 machinist	31.25	31.25
3	connecting bar			31.25	23.44
	Turning/drilling	45	1 machinist	31.25	10.42
	Marking out	20	"		
4	Handle				
	Marking out	20	1 Technician	25	8.33
	Cutting	20	"	25	8.33
	Welding	30	1 welder	25	12.5
5	Bearing installation	60	1 machinist	31.25	31.25
6	Installation				
	Machine	60	1 Engineer	62.5	62.5
	assembly	120	1 welder	25	50.0
7	Conveyor unit				
		120	1 machinist	31.25	62.5
	Painting	60	1 painter	25	25
	TOTAL				₦388.02

Production cost = material cost + labour cost

= 33770 + 388.02 = ₦34158.02

Add 10% contingency allowance to production cost.

Therefore;

Total production cost = ₦37573.82

Note;

The student under the supervisor of the project, advisers and the University workshop staff, supplied all the labour.

The labour cost shown is therefore opportunity cost.

CHAPTER SIX

6.0 PERFORMANCE EVALUATION

6.1 Test Procedure

After the rice-reaper was designed and fabricated, a number of tests were conducted in the field to ascertain its field capacity and field efficiency. It was tested at National Cereal Research Institute (NCRI), Badeggi; near Bida - Niger state. During testing, the machineries and equipments used include rice reaper, $1m^2$ frame, polythene bags, weighing balance and camera. The tested area covers three plots (Figs. 7,8 and 9) of farm land, each $10m^2$. There are three varieties of rice crop in that farmland. The varieties include Faro 35, Faro 9 and CP. The reaper was test run for ten minutes before the test commences. It was then operated at uniform speed. Adjusting both the two front two rear wheels controlled the height of cut. The height of cut was in the ranges of 10cm and 20cm. Nevertheless, during the test, a height of 20cm was used.

The moisture content of rice grain was determined before harvesting. Three samples of grain were taken for moisture content analysis. The samples were placed on a container and weighed. After weighing, the samples were oven dried at temperature of 130°C for 15hrs (1994). The moisture content on wet basis was then determined from the expression;

$$Mw_b = \frac{\text{Weight of wet sample} - \text{Weight of dry sample}}{\text{Weight of wet sample}} \times 100\%$$

The length of rice crop and length of grain portion was measured using measuring tape (see Table 1)

The test performance data was analyzed to determine the field capacity, effective field capacity and field efficiency.

6.1.1 Field Capacity

The field capacity was determined by taking the total time the operator used to make to and fro journey between each row in a plot and then measured the total area covered. This was then expressed in ha/hr.

$$\text{Field Capacity} = \frac{\text{total area covered}}{\text{Total time taken}} \text{ ha/hr}$$

6.1.2 Field efficiency

The rice density was measured for each plot by counting the rice crops that fells within 1m x 1m frame for each plot at three different locations and then the average found for each plot. The field efficiency was determined by finding the difference before and after harvesting and expressing the result as a ratio of density before harvesting.

6.2 Results and Discussions

6.2.1 Field Capacity

The test results related to field capacities are presented in Table 5. From this table the maximum field capacity of 0.0353ha/hr occurred at depth of 20cm. The minimum capacity was 0.0327ha/hr and at the depth of 10cm. This gave an overall average field capacity of 0.0338ha/hr. The low capacity might be as a result of the nature of the land where the test was conducted. The machine might be operated below optimum speed due to unfavourable field conditions.

6.2.2 Field efficiency

The average value of field efficiency is presented in table 5. The maximum and minimum values are 83.7% and 73.3% respectively and at the depth of 20cm and 10cm depth. This gave an overall average of 79.15%. From table 6.2 also, the maximum efficiency was at the depth of 20cm, this might be as a result of occasional blockage of the belt at the lower depth operation and less belt drive obstruction at the higher depth of operation.

Table 4: Length of stalk of rice crop and stem portion

S/N	Length of Stalk of rice crop (cm)	Length of rice stem (cm)
1	90	20
2	92	22
3	80	22
4	70	15
5	89	18
6	78	20
7	93	23
8	56	20
9	50	15
10	91	21
	Average $\overline{X}$ 78.9	19.6

Table 5: Average field capacity (ha/hr) and field efficiency (%) of the reaper for the two stages of harvesting

S/N		D_1	D_2
1	Fc(ha/hr)	0.0327	0.0353
2		0.0332	0.0343
3		0.0329	0.0346
1	Fe(%)	73.3	82.9
2		75.7	83.7
3		77.9	81.1

D_1 and D_2 are height of operation
D_1 = 10cm D_2 = 20cm

Table 6: Average capacity (ha/hr) and field efficiency (%)of the reaper for each plot with their corresponding rice density.

N/S	Rice density/50m^2	F.c(ha/hr)	Fe(%)	M.C. %
1	275	0.0327	73.3	14
2	284	0.0332	75.7	14
3	292	0.0329	77.9	14
4	311	0.0353	82.9	14
5	314	0.0343	83.7	14
6	304	0.0346	81.1	14
Average 296.7		0.0338	79.15	

Where; M.C. = Moisture Content.

F.C. = Field Capacity

F.e. = Field efficiency

CHAPTER SEVEN

CONCLUSION

7.1 CONCLUSION

A rice reaper to harvest rice and conveyed rice on flat land has been designed and fabricated. The rice reaper cuts (harvest) and conveyed rice in vertical position to right hand side of the reaper. Field test were conducted and the results showed that a considerable quantity of rice plants was harvested and conveyed with an average field capacity of 0.0338ha/hr, field capacity was found in the range of 0.0329ha/hr to 0.0352ha/hr. the field efficiency achieved was in the range of 73.6% to 83.7% and with an average field efficiency of 79.15%.

The results also show that, the labour requirements for harvesting and conveying are reduced to 30.5 man-hr/ha, that is one third of the labour requirement compared to the manual harvesting method using hand tools. Thus, the use of this rice reaper can reduce or eliminate peak demand of labour during harvesting season.

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FIG15:Complete rice reaper

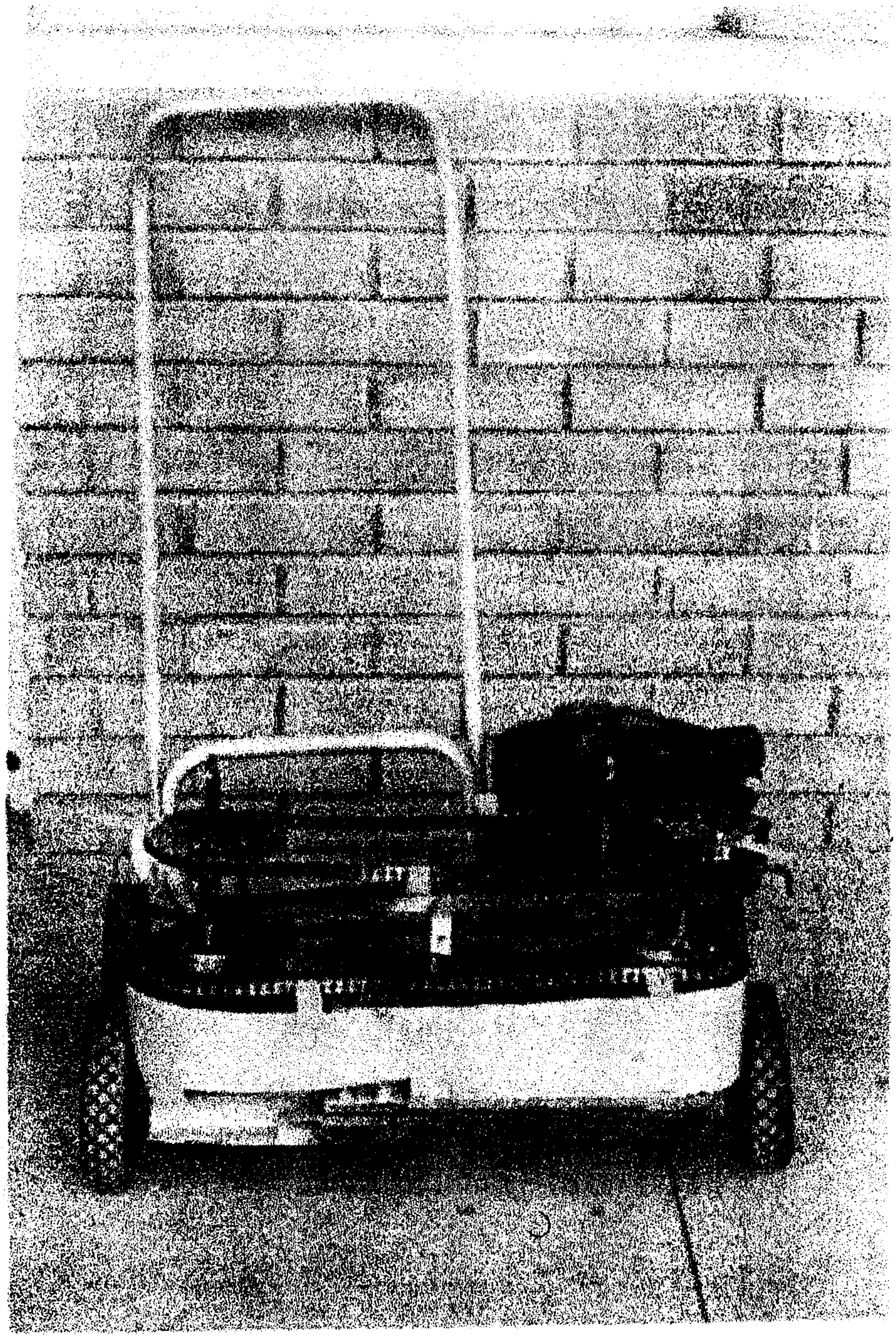

FIG. 6 : Rice reaper showing the conveying mechanism on the farm

FIG 7 : Rice reaper on the farm during testing (Plot 1)

FIG. 8: Rice reaper on the farm during testing (Plot 2)

FIG. 9 : Rice reaper on the farm during testing (Plot 3)