DESIGN OF A RICE REAPER

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PROJECT TOPIC

DESIGN OF RICE REAPER

THI IS SUBMITTED IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF POSTGRADUATE DIPLOMA (PGD) IN AGRICULTURAL ENGINEERING.

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CERTIFICATE OF APPROVAL

This is to certify that this project work DESIGN OF A RICE REAPER is an original work undertaken by MOHAMMED ALIYU TSADZA PGD/AGRIC/97/98/027 under the guidance and supervision of Dr. M. G. Yisa.

The project is prepared in accordance with the regulations governing the prepa ration and presentation of project in the school of Engineering and Engineering Technology, Federal University of Technology, Minna.

.... 19/12/2000 ....

DR M. G. YISA (SUPERVISOR) DATE

.... 19/12/2000 ....

DR. M.G. YISA (H.O.D) DATE

DECLARATION

I Mohammmed Aliyu Tsadza hereby declared that this project work "DESIGN OF A RICE REAPER" was undertaken solely by me under the supervision and guidance of DR M.G. YISA. I have neither copied someones work nor have someone else done it for me. Writers whose work have been referred to in this project have been acknowledged.

....

M.A. TSADZA

PGD/AGRIC/97/98/027

Technical Specification

Engine type (prime mover) 4 -cycle air cooled 3.5hp

Displacement - 148cc

Fuel - Regular gasoline fuel

Tank capacity - 1 litre

Tyres - Mower type

Speed of operation 3600 rpm

Cutting device - Reciprocating knife bar

Cutting height - adjustable (100 -200mm)

Cutting angle - 60^{o} (reshetove, 1978)

Frame design

The machine frame is design to support all the basic components of the machine to ensure proper location with respect to one another and resist all prinpal forces that will act on the machine. The machine will be design to supports both the axial and lateral loads.

The frame of the machine is made from angle bar iron of size 50 x 50 mm with thickness of 3mm.

The engine seat is constructed from flat bar and welded to the frame.

The handle

The handle was made from the galvanised hollow pipe made from mild steel. The handle provides the point of articulation for pushing the machine. The pipe has a diameter of 21.60mm, 1200mm long, 41.5mm wide and 88mm high. The handle is also welded to the frame of the reaper.

Track wheel

The reaper has four track wheels to allow for easy movement, it has a diameter of 19cm the wheel is bolted to the frame

Bolts and nuts.

Bolts and nuts of various sizes 8m, 12mm and 13mm are used in joining various components together. This is to allow for adjustment, replacement and easy dismently of the machine.

The cutting unit.

The cutting unit consist of main blades the stationary and the moveable blocks. The stationary blades has five cutter bars with 375 mm long and the moveable blades has four cutter bars and its 300mm long.

The stationary blades is jointed to the frame while moveable is fixed in a groove through a cam follower.

CHAPTER FOUR

4.0 CONCLUSION

In Nigeria, the only methods of harvesting rice is by means of using the traditional methods and fairly by combine harvester. The combine is a sophisticated machine and requires a skill technicians to handles it. It is also a very costly machine above the means of the rural farmers. Therefore, in view of these, a mechanical rice reaper is designed for harvesting rice which is affordable to the rural farmers and also to serve as a substitute for the traditional methods of harvesting.

4.1 RECOMMENDATION

The machine designed can only cut at this stage. Therefore a means of windrowing should be provided in the on-ward re-designed and the machine should be fabricated.

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DEDICATION

This project work is dedicated to my family and the rural farmers.

ACKNOWLEDGEMENT

In the name of Allah the beneficient, the Merciful. May peace and Blessing of Allah be upon our Holy Prophet Mohammed (S.A.W.).

Thanks be to the Almighty (S.W.A) for giving me the opportunity to undertook this project work. At this point I want to express my hearty gratitude and appreciation to my project supervisor Dr. M.G. Yisa for his close supervision and assistance offered me during my project work. My profound and hearty appreciation also goes to Engr Suleiman J.G. Mohammed for his moral support during my period of stay at Minna.

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To those who have not been mentioned but has contributed in one way or the other to make this work a success I say thank you and may Allah bless and reward you all abundantly.

Alhamdulillah

ABSTRACT

A mechanical rice reaper is designed for harvesting of rice using the cutter bar machine. The machine is to be powered by a 3.5hp, 4 – cycle air cooled petrol engine mechnism. The power from the engine is transmitted to the cam with the help of a V-belt and a drive shaft connected to the curvilinear cam. The engine is mounted on the base set upon the body of the machine and held tightly by bolt and nut. It has two-cutter bars one stationery and the other moveable m connected to the frame of the machine. The rated power of the mac hine is 2.61kw with 0.54kw as the power required for cutting.

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CHAPTER ONE

INTRODUCTION.

Rice (Oryza Sativa) is an important and a leading Cereal crop in many countries. It belongs to the Graminae family. It is grown on the continents.

The species Oryza Sativa was introduced into Africa some centuries ago from Asia. It is the most widely species, however, there is another species oryza glabenima which originates from Africa (Michael & Brigitte, 1987).

There are two main varieties grown in Nigeria, the upland rice and the lowland rice. The upland rice is grown on the farm and requires less water while the lowland is grown on the fadama area and it requires much water.

Of the worlds cereal crop, rice is a leading food crop cultivated for its edible starchy grain. Its usefulness is indeed universal because man benefits not only from its starchy grains but also from other part of the plants and the by-products of its processing. It is the stable food for almost half of the world population and an essential sources of human subsistance in south and south East Asia where the average annual consumption of rice per capita is about 100kg.

Growing rice facilitates the improvements of saline and alkaline soils which after 2 or 3 years in rice, can successfully be used for the growing other commercial crops to the benefits of the rice grower. (konokhola, 1985).

Because of its high digestibility and high nutritive value, white rice has become indispensable for use in baby and breakfast foods and in diets for the sick. Rice is also used for the production of starch, alcoholic, beverages and soft drinks in manufacturing industries. Rice hulls and polish that includes bran and the germ are used in the pharmaceutical industries for the production of drugs. Rice straw is also an importants raw material for manufacturing of high quality paper, hand bags, sacks etc.

Inspite of the enumerated important of rice to human diets, its harvesting in Nigeria had been and has remain a serious problem to the farmers (Michael etal, 1987).

The technique for harvesting of rice are still traditional by using mainly sickle which is 60 - 80 man-h/ha, of which 60-100 man-h are used in cutting and laying the crop (Gajendra single etal 1988). The harvesting period is very short and crop losses increases rapidly with delay in harvesting. Delayed harvesting of mature crops also exposes its to many hazards like rains, windstorms and fire. At the peak harvest, scarcity and unavailability of labour increases labour cost with the result that the farmers now pays at exorbitant amount of money per hectare. Due to rapid urbanization and migration of farm labour to cities a big vacuum has been created in the supply and demand ration of farm labour. This scarcity of labour has forced farmers to go for mechanization. At the peak harvest, scarcity and unavailability of labour increases labour cost, since the farmers has to pay more money per hectare.

AIMS AND OBJECTIVE.

In view of the problems enumerated above, it is therefore necessary to design a simple rice reaper using a cutter bar mechanism.

JUSTIFICATION

The machine if fabricated would serve as an alternative to the traditional methods of harvesting thereby increasing rice production, at a lesser cost.

CHAPTER TWO

LITERATURE REVIEW.

Harvesting of rice is one of the most labour intensive operation in the production of rice, and this must be done fairly quickly to obtain a good grade of rice, and to reduce grain damage and losses.

In the early days, harvesti;ng of crops was limited to the use of simple and relative inefficient tools, such as hoes, cutlasses, sickles etc. These tools made agricultural operation very tedious using the simple tools, a limited area of land could be managed. Consequently, the volume of production was low. However, with the advent of science and technology, sets of machines that can imporve production and increase efficiency were developed.

Throughout the tropics, harvesting of rice is done by hand labour. The rice is cut in clumps by the use of sickles and knives.

Much work has also been done on mechanical reapers, such as the use of cutter bars. The cutter bar is mounted on a two - wheel tractor which harvest in swathes, with tractive power of 3. hp (Michel etal 1983).

In Nigeria, harvesting of rice is still mainly by traditional method using sickle and knife. These tools are labour intensive and time consuming, and the imported machines are complex, costly and mostly requires skills for their operation.

In Nigeria, less research has been done on rice reaper because of their dependability on the imported ones. Much research has been done in developed and some developing countries. Therefore mechanical harvesting of rice has come along way in some countries.

2.1 PREVIOUS WORK.

In the developing counties of South and Soth East Asia nearly all paddy fields of small farmers are harvested or reaped mannually by groups of labourers using mainly sickles and knives in harvesting (Stickney etal, 1985).

The labour requirement for harvesting of rice with these tools is between 80 - 160 man-h/ha, of which 60-100 man-h are used in cutting and laying the crop (Gajendra singh etal 1985).

Reaping requires a high labour input because it is the most strenuous activity of rice production. The labour wage rate in (cash or kind) is often substantially higher than for other activities. Consequently reaping is a major expensive in production of rice.

A Vertical - conveyor reaper was developed in China in the early 1960s (Stickney etal 1985). It is mounted on a two-wheel or four wheel tractor. Its cuts the crop and lay or windrow it on the right hand side.

The advantage of this design compared with the conventional horizontal or inclined designs is that its light in weights, simple in construction, ease of front mounting on small two-wheel or four-wheel tractors and also imporves the efficiency of the machine in small fields.

In the late 70s, the International Rice Research Institute (IRRI) in collaboration with the Chinese Academy of Mechanization Science (CAAMS) designed a CAAMS - IRRI 1.6M reaper. This reaper is suited to the harvesting of rice, for small farms in south and east Asia.

In 1980, three Chinese engineers worked with IRRI engineers in Phillipines to develop a simplified reaper which was lighter, less expensive, and easier to fabricate with simple tools, materials and components available to small scale manufactures in Phillipines and other developing countries.

IRRI developed a rice stripper harvester in 1960. Stripping was carried out on a belt fitted wire-loop teeth moving parallel to the direction of travel of the machine. The crop is fed on to the belt by plant gathering bars. Development work on the machine was stopped due to high shatter losses and poor performance particularly in different crops (Prince 1989).

The first known stripper harvester was described by the Roman historian pliny around 70 A.D. the equipment known as Hallic vallus (Quick 1978) was a simple wooden container with a forward projecting Comb mounted on wheels and pushed into the crop by donkey. Stripped grains were raked into the container by an attendant walking along the side of the machine. Development on this equipment was stopped as a result of time consuming, shalter losses and poor performance.

2.2 CURRENT WORK

A modified version of chinese vertical reaper was developed by the engineers from farm machinery institute (FMI) in 1981. (Rohman 1981). These Institutes fabricated two types of prototype chinese vertical reaper. The reaper is mounted on the front of a 4-wheel tractor and drive from tractor pto driven 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/ha and field efficiency of 59.4%.

2.3 AMRI REAPER.

The Agricultural Mechanization Research Institute designed and developed a reaper windrower at mutun (Amjad, 1988). The AMRI Reaper windrower is a tractor mounted pto operated by pto shaft power through a pair of belt pulleys, a propeller shaft and a cam. An over head reel is used to support and gather the crops being cut and to lay the same on the conveyor, while windrowing is done through a set of deflectors. The AMRI reaper windrower has an average travel speed of 3.01km/hr, working width of cut of 2.04m, field capacity of 0.31ha/hr, with field efficiency of 59.5% and fuel consumption of 3.2 lit/ha.

The Ittefaq Reaper

The ittefaq reaper is a tractor front mounted machine. It is a redesigned version of the Augostimin cutter - binder without a crop binding mechanism. In this type of cutter binder the power from the tractor pto 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 gear box of the machine. Automotive power from the engine is transmitted to the machine gears bringing the cutting blades into a to and fro motion through belts and pulleys. The collecting hooks comes into operation through chain and sprockets. These hooks after collecting the wheat 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 tractor and later picked up by a labourer an average travel speed of 2.63km/hr, working width of cut 2.06m, field capacity of 0.35ha/h with field efficiency of 64.% and fuel consumption rate of 3.31 L/Hr.

2.5 SIDE STAKING REAPER.

Another type of reaper is a side staking reaper developed in China. This reaper can be powered with an average of 11-18kw conventional tractor or with 6-9kw walking tractor. It has crop lifters, cutter bar and a raking/staking device. The working width is 1.6m with a staking distance range of 2m, 2.5m and 3.2m with a total weight of 137kg.

In 1992, the International Rice Research Institute in collaboration with the Chinese Academy of Agricultural Mechanization (CAAMS) developed a motorized reaper (I an and Rodriguez 1992). It comprises of a reaper unit built unto a power tiller with a 2.2kw petrol engine and a cage wheel, but it is adaptable to other walking type tractor unit. It is of all steel construction, except for metallic star wheels. It has a walking width of 1m with adjustable cutting height from 70mm, forward speed of 2.5 - 4-5km/hr, work rate 2 ha/day and fuel consumption of 1 lit/hr and total weight of 135kg.

The motorized two - row reaper binder is manufactured in Japan and Korea. It is a two wheeled machine designed primarily for row-crops of rice, barley and wheat. It cuts the crops into bundles and dropped them 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 subsequent gathering operations. The machine is carried on two wheels with low pressure pneumatic tyres and manually steered. Its powered by a 2.6-3.7kw, 4-stroke petrol engine and has six forward gears. Plus two-reserve gears. It has a working width of 550 and a total weight of 165kg.

2.7 MOTORIZED REAPER BINDER.

This motorized reaper binder were manufactured in Italy. It is a two wheel, manually steered machine and is fitted with linked plate type wheels for operation in rice fields. This motorized binder is powered by a 7.5-10kw petrol or diesel engine. It has a working width of 1.3m and weighs 480kg.

2.8 REAPER BINDER

Reaper binder is also produced in staly. It has a ride - on unit with a single or double wheel trailed bulky - type driving seat. It is powered by 7.5 – 10kw petrol or diesel engine and has four forward gears plus reverse gear. The crop is cut by a reciprocating cutter bar collected into sheares and tied with string. The harvesting machine can have a manually or hydraulically list. It has a working width of 1.4m.

2.9 MADHO WHEAT HARVESTER.

This reaper is fits on 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 70cm. The reaper can be tilted back on its mounting frame into a transport position. It has a working width of 2.2m. This machine is produced from madho mechanical works, India.

2.10 STRIPPER HARVESTER.

This is an harvesting techniques that is used in standing crop of uniform height. The principles of stripping is that it takes the seeds from the standing plants without harvesting the straw (Tado etal 1998.)

This presents a bright prospects in Mechanical harvesting technology since the amount of straw handles by the machine is considerably reduced. The main advantge of the stripper harvester is that there is a possibility of increasing the harvesting capacity at a reduced power requirement and more combine hours at harvest. The reduced straw intake also offers potential for a reduction in size and weight for a machine of given capacity.

The most promising stripper system at present is the stripper header (Tado etal'98) developed at silsoe research institute U.K. The silsoe stripper uses the tansverse rotor arranged transversely to the direction of travel. The upward rotation of the rotor with respect to the crop enables it to pick up lodged crop.

Recent development in stripper harvesting technology points out the feasibility of stripper. Self propeller track is another type of stripper harvester which essentially composes of a pick - Up stripper thresher, pneumatic conveyor system etc. In this machine, the plants are defflected by the fingers on chain of the pick up and gently depressed further by feeding belt. Under the action of teeth on thresshing drum and air suction from pneumatic conveying system the panicle are fed into the thresher (Jiang 1991).

THE COMBINE:

The combine is a sophisticated machine that carries out many operation in harvesting. It s intended for use on larger farms.

Basically a combine consist of four units, the cutting unit, the threshing unit, cleaning and bagging unit. The cutting unit consist of the cutter bar, the crop conveying unit which cuts the crop and then conveys it to the threshing unit for threshing.

A combine has the advantage of being able to carry out cutting, threshing, cleaning and bagging in one operation.

The size of a combine is given by the width of the cutter bar, thus a 2.5m combine will have a cutter bar that is 2.5 long. The combine harvester is not acceptable to most farmers as they need chopped straw for animal feeds.

The combine is a sophisticated, very costly machine and requires large fields, highly skilled operations and technicians to use and maintains it, therefore not affordable by the rural farmers.

CHAPTER THREE

DESIGN METHODOLOGY

3.1 Design consideration.

Before the design of any machine it is necessary to consider the engineering properties of the materials and the stresses to which the material will be subjected to and the strength to withstand these stresses. Reaper will be design to achieve high efficiency in its performance and economy. The following parameters are considered in the design;

1. The mechanical properties e.g. strength, ductility rigidity, machinability and welderbility.
2. The chemical properties i.e resistance to corrosion.
3. Analysis of the forces required for cutting and the power available for cutting
4. The maximum area of cut per unit time.
5. The working principle; to make sure that the chosen working principle produce the desired effects and advantage.
6. Safety:- Factors affecting safety of the components, its operation and the operator will be considered.
7. Ergonomic:- That is man-machine relationship will also be put into consideration.

3.2 Design approach.

The principles of scissors - like cutting action will be used in this design. The machine will have scissors-like cutter (stationary and moveable cutter) that will be coupled to the frame of machine and supported by four wheels. The clipping shears of the cutter will be powered by a 3.5 hp engine which transmit power (motion) to the cam through a v-belt through a shaft connected to the cam. The cam will have a groove that can accommodates a pinion attached to a follower. As the cam rotates the cam follower connected to the moveable cutter together oscillates to and fro on the stationary cutter. This will then cause a shearing action (see fig 1).

3.3 Design parameters/Calculation.

The basic design parameters considered are the torque transmitted by the engine, the power rating the varried speed of cut, the unit area of cut per unit time and the efficiency of the machine.

3.3.1 Design Calculation.

Torque transmitted; The torque transmitted by the engine on the shaft can be calculated using the power and the speed of the engine. Assume the nominal engine power of 3.5h.p with a speed of 3,600 rpm.

Nominal engine power = 3.5hp

but 1hp = 0.745 kw

$\therefore$ 3.5x0.745 = 2.61kw

Therefore torque applied $= \frac{P}{w}$ --- (1)

Where p = power (kw)
w = Angular speed. (radlsec)

Angular speed $= \frac{2\pi n}{60}$ --- (2)

where n = rotational speed of the shaft

=> Torque applied (iii) $= \frac{9550 \times kw}{n}$

$$= \frac{9550 \times 2.61kw}{3600} = 6.92 \text{ NM}$$

Therefore the torque transmitted is 6.92 NM.

The angular speed (w) $= \frac{2 \pi \times 3600}{60} = 376.99$ rad/sec

<u>Force at the engine pulley;</u>

The force at the engine pulley (F) $= \frac{Px\ 60}{\pi\ d_{ini}}$ --- (3)

The transmission ratio equation $= \frac{d_1}{d_2}$ --- (4)

Where d_1 = engine pulley diameter = 2.5 cm
d_2 = cam pulley diameter = 12.5cm

But the speed ratio $= \frac{n_1}{n_2} = \frac{12.5}{2.5} = 5$ --- (5)

$n_2 = \frac{3600}{5} = 720$ rpm

The force at the engine pulley will therefore be;

From $F_1 = \frac{p \times 60}{d_{ini}} = \frac{2.61 \times 10^3 \times 60}{\pi \times 2.5 \times 10 \times 3600}$

$$= 553.86N$$

Force at the driven pulley F2 $= \frac{P(kw) \times 60}{d_{2n2}}$

$$= \frac{2.61 \times 103 \times 60}{12.5 \times 10 \times 3600}$$

$$= \underline{110.77N}$$

Therefore the force available at the driven pulley is 110.77N.

To calculate the power available for cutting using the formular:

$$P = \frac{F_2 \times d_{2n2}}{60}$$

Where F_2 = force at the cam pulley (N)
d_2 = diameter of driven pulley (mm)

n_2 = speed (rpm)

But $F_2 = 110.77N_1$ $d_2 = 12.5x10^2$ mm, $n_2 = 720$ rpm

$$\therefore P = \frac{110.77 \times \pi \times 12.5 \times 10^{-2} \times 720}{60} = 521W$$

$$= 0.521KW$$

This is the power available for cutting.

Linear Velocity of the moveable cutter

The linear velocity of the moveable cutter will be determined using the formular V = Wr.

Where V= linear volecity (mls)
W= angular speed (radls)
r= distance from pivot of cam follower to the point of contact with cutter bar (mm) = 75×10^{-2} mm

$$\text{but from } W = \frac{2\pi \times 720}{60} = 75.41 \text{ radls}$$

but r = 0.075m

ii V = 75.41 x 0.075 = 5.67mls

Therefore 5.67mls is the linear velocity of the moveable cutter.

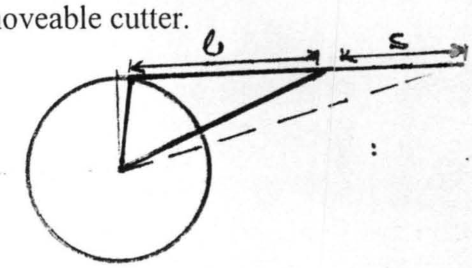


Fig 1

Cutter bar and Cam design

Fig 1. Cam arrangement for cutter bar movement

Where l = length of the cam follower (mm)
s = stroke length, mm
r = cam radius.

The knife has to be registered, that is the reciprocating knife (moveable cutter) has to move from the centre of the stationary knife to the other end. The cutter Bar used in the conventional reciprocating type cutter bar will be driven by the cam mechanism directly from the petrol engine of 3.5hp. using the recommended speed ratio of average cutter bar, a speed to forward speed of machine is 1.3 - 1.4 mls (Devani and pandry, 1985).

$$=> \frac{Vc}{Vm} = 1.4 \text{ mls} \quad \text{--- (7)}$$

Where Vc = cutter bar speed
Vm = machine speed

Therefore the cutter bar speed can be calculated from the relationship.

$$Vc = \frac{S \times Nc}{30} \quad \text{--- (8)}$$

Where Nc = rev/min of the cam
S = stroke length, mm

but the speed of the cam through the designed petrol will be calculated from

$$Nc = \frac{\text{Engine speed}}{\text{speed ratio}}$$

Assume belt slippage of 10%

$$=> Nc = \frac{Vc \times 30}{S} = \frac{1.4 \times 30}{0.075} = 560 \text{ rpm}$$

The speed of the cam through the designed petrol engine will be

$$Nc = 3600 \times \frac{1}{5} = 720 \text{ rpm}$$

Assume belt slippage of 10%

$$=> Nc = Nc \times 0.9$$
$$= 720 \times 0.9 = 648 \text{ rpm}$$

∴ The absolute velocity of the cutter bar will be;

$$V_{max} = rw$$

Where r = cam length
w = angular velocity of Cam

$$\therefore V_{max} = 0.018 \times 75.41 = 1.36 \text{ mls.}$$

Therefore 1.36mls is the absolute velocity of the cutter bar.

The maximum acceleration and intertia force of knife cutter bar is determined from the expression

$$max = \frac{SW}{2} \left(\frac{1 + \pi}{L/r} \right) \quad \text{--- (9)}$$

Where L cam follower length (m)

m = mass of cutter bar knife, kg

max = maximum acceleration f knife bar (mls^2)

$$\therefore \quad \max = 0.075 \times 75.41 \left(0.018 + \frac{\pi}{0.18/0.0625}\right)$$

$$= 31.29 \text{ mls}^2$$

but the mass of the cutler bar knife is assumed to be 1.680 kg + 1.34kg = 3.024 kg.

Hence the maximum Intertia force F = Ma --- (10)

$$= 3.024 \times 31.29$$
$$= 94.62 \text{ kg mls}^2$$

The reaper power requirement will be the total power consumed in cutting which is 0.521kw. Assume 90% power transmission efficiency

$$\therefore \text{ The efficiency} = \frac{0.521\text{kw}}{0.9} = 0.58\text{kw}$$

BELT DRIVE DESIGN

Belt selection - V - belt (based on usual load of 2 - 15kw
Engine speed = 3600 rpm
" power = 3.5 hp . (2.61 kw)

Recommended pulley pitch diameter = 63 mm

Normal top width bo = 10
Nominal thickness h = 6, ho = 6
sheave groove angle = 0.40^0
weight per meter kgf = 0.189 kgl
Density of rubber belt = 1250 $kglm^3$

Fig Belt dimension

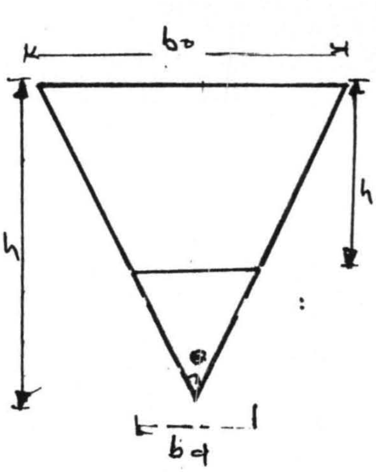


fig II

To find belt speed $S = \frac{\pi dn}{60}$

Where d = pulley diameter (mm)
n = shaft speed (rpm)

$$= \frac{\pi \times 63\times10^{-3} \times 3600}{60} = 11.88\text{mls}$$

$S_1 = 11.88$ mls

Angle of contact

$B = (180 - 0)/_2$

$= (180 - 40)/_2 = 70^o$

$B = 70^0$

Belt cross - sectional area A.

$A = (bo + bd)hr/2$

$A = (10+8.5)\ 8.^1/_2$

$= 74.925x10^{-6}m$

Mass per unit length will be

From $M = \rho A$

Where ρ = density of the rubber kglm^3
A = cross-sectional area of the belt.

$M = 1250x74.925x106$
$= 93656.25x10^{-6}$
$= 0.0936$

Speed ratio $\frac{n_1}{n_2} = 5$

$= n2 = \frac{n_1}{5} = \frac{3600}{5} = 720$ rpm

$\therefore\ d_2 = \frac{n_1 d_1}{n_2} = \frac{3600x63}{720} = 315mm$

$\therefore\ d_2 = 315$ m (diameter of pulley 2)

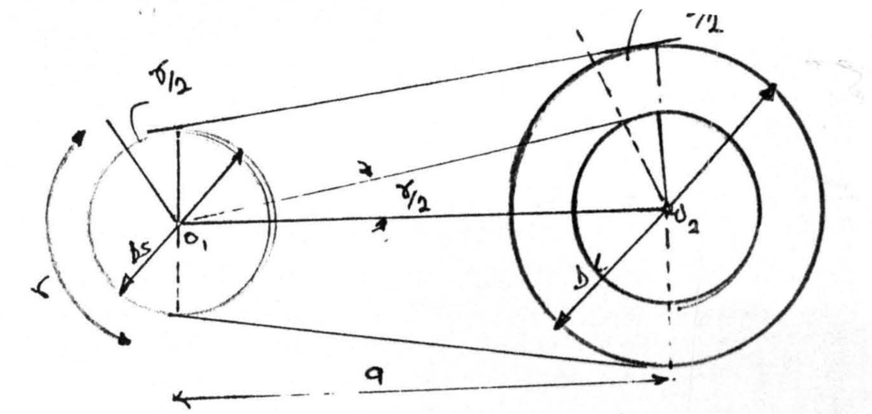
Owing to inevitable creep, the peripherial speed on the driven pulley S_2 is less than the speed on the driving pulley S_1

$\therefore\ S_2 = (1 - \xi)S_1$ where $\xi = 0.01$ (creep factor) (Reshetorve 1979)

Hence true speed ratio (u) $\frac{n_1}{n_2} = \frac{d_2}{d_1(1-\xi)}$

Therefore $u = \frac{0.315}{0.063(1-0.01)} = 5.05$

=



Principal geometric relationship in belt drives. Fig III

In designing belt drives the following geometrical parameters are to be calculated. The angle (γ) between the sides of the belt, are of contact α , with the small pulley, belt length(L) and centre-to-centre distance a (where an endless belt is tobe used).

The angle between sides of the belt is determined from the auxillary triangle O_1AO_2 (fig above).

$$\sin \gamma/2 = \frac{\Delta}{a} \text{ - but } \Delta = \frac{\Delta L - Ds}{2}$$

Hence the angle between the sides of the belt is

$$r = 2 \sin^{-1} \frac{\Delta}{a} \approx 2\frac{\Delta}{a}$$

The arc of contact on the small pulley is

$$\alpha = 180^\circ - \gamma \quad 180 - \frac{2\Delta}{a} \times 57^\circ \text{ (from Table)}$$

though V-belt drives can operate with efficient reliability with an area of contact equal to $90^\circ{}_1$ it is not advisable to use an are of contact of less than 120° for a V-belt drive (Reshetor 1978)

$$\Delta = \frac{DL - DS}{2} = \frac{0.315-0.063}{2} = 0.126$$

The length of a belt (not taking into account the sag and the initial stretch) is equal to the sum of the lengths of the arc_2 of contact on the pulleys plus the length of the stretch portions of the belt thus.

$$L = \pi\left(\frac{DL - DS}{2}\right) + \gamma\left(\frac{DL - DS}{2}\right) + 2a\cos\frac{\gamma}{2}$$

Transforming the above formula using the approximate relationship $\cos \alpha/2 = 1 - \frac{1}{2}\left(\frac{\gamma}{2}\right)^2$

Substituting the value of γ and replacing $\frac{D_L + D_S}{2}$ by D_m

it becomes

$$L = \pi D_m + \left[2 + \frac{(4)2}{2} \right] a$$

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

$$\frac{0.315+0.063}{2} = 0.189$$

The chosen belt length L = 1500 mm = 1.5 (from standard table).

$$\therefore \quad = \quad \frac{1.5 - \pi(0.189)}{4} + \frac{1}{4}\sqrt{1.5 - \pi(0.189)^2 - 8(0.126)}$$

$$= 0.22656 + 0.20831$$

$$= 0.435\text{m}$$

From $\gamma = 25\text{m}^{-1}$

$$= 2\sin^{-1}\frac{0.126}{0.435} = 33.68$$

$\therefore \gamma = 33.68$

Therefore the arc of contact of the small pulley will be,

$$180 - \alpha = 180 - 33.68$$
$$= 146.32$$

Forces and stresses in V - belt drive

The peripheral force acting on the pulley or useful load of a belt Kv is calculated from the relationship $F = \frac{2KT}{D} = \frac{102KP}{V}$

Where T = Torque (KNm) on a pulley of diameter (Dm)
P = power transmitted (Kw)

$K \quad = \quad$ dynamic load and service factor.

Though rubber and V-belts are non-homegeneous, they have various structural components. Fabric belts are not continous in cross-section. Nevertheless, all belts can be arbitrarily be considered as being continous and homogeneous in calculations basing the latter on the nominal stress.

The stress from the peripheral force F is

$K \quad = \quad F/A$ where A = cross-sectional area of the belt in m^2

The initial tension So of the belt is chosen from the condition that the belt retains its tension for a sufficiently long time without excessive stretch, and has a satisfactorily service life. The initial stress in V - belts for standard and series belts may be $= 12 - 16\ Kn/cm^2$.

The nature of the dynamic load of the reaper is considered as steady and thus have a dynamic load and service factor of 1 (table 12.5)

Therefore the peripheral force acting on the pulley is

$$F = \frac{102KP}{V}$$

but

$$V = 11.88mls$$
$$K = 1.$$

$\therefore$

$$P = 3.5hp = 3.5x0.746 = 2.61kw$$
$$F = \frac{102(1)2.61\ (Kw)}{11.88mls} = 22.43KN$$

Also, from the table; the area of the belt is given as $0.47cm^2$.

$\therefore$ The Stress from the peripheral force F is

$$K = F/A = \frac{22.43KN}{0.47cm^2} = 47.72\ KN\ Cm^2$$

<u>Bearing Selection</u>

Ball bearing will be use for this design based on the type of loading life requirement and since all bearing is frequently under conditions of variable speeds.

The analysis for the selection of the type of bearing is done by determing the radial and axial load.

Ball bearing is made in several types such as a single row radial, angular contact, thrust and slf aligning. Based on these analysis, self aligning single row radialball bearing with the following parameters is selected.

ISI NO (SKF) = 20TA12

Bearing No = 6313Z

Internal diameter = 20 mm

External diameter = 40mm

width B = 14

Ball bearing was selected for this design because of the following reasons.

1. It can operate in dusty condition
2. It does not needs any lubrication
3. It reduces friction between moving elements
4. It absorbs vibration

Description of the Machine

The main features of the reaper includes the transmission unit, the cutting unit the conveying unit, the frame and the operating handle.

The reaper is mannually pushed, light in weight. It has a 30cm long cutter bar and cuts a single row of crop with 25 - 30cm row spacing.

The following is the techical and physical specification of the machine.

Physical Specification

Machine length	=	1320mm
width	=	800mm
Height	=	1020mm

11. Reshetov D.N. (1^78)
Machine Design Mis Publishers - Moscow.

12. Rahman Z.U. (1981)
Reaper development at AMD. Proceedings of a one Day Workshop on Farm Mechanical in University of Agric. Failsalabad, Pakistan.

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Critical Design Parameters and Development of CAAMS - IRR Reaper. Proceedings of 32nd Annual Convention of the Philipine Society of Agric. Engineers Manila, Philipines.

14. Rhyder, G. H. (1982).
Strength of Materials 2nd Edition
PSG College of Technology Design Data.

15. Brian B. (1993).
50 years of Farm Machinery.

16. PSG Design Data (1982
Design data compiled by Faculty of Mechanical Engineering, PSG College of Technology, Combatore 64 1004 India.

TABLE 12.8. Allowable Useful Load on a V Belt at $\alpha = 180°$, $v = 10$ m/s and a Steady Load

Section	Minimum and pitch diameters of small pulley, mm	Initial stress σ_0, kgf/cm² 12	14	16	Section	Minimum and pitch diameters of small pulley, mm	Initial stress σ_0, kgf/cm² 12	14	16
		Allowable useful load F_0, kgf (per belt)					Allowable useful load F_0, kgf (per belt)		
O	63	6.3	7.0	(7.8)	D	315	76.2	84.9	90.5
	71	7.1	7.8	8.5		355	84.9	93.5	101.3
	80	7.5	8.4	8.9		400	(56.8)	97.2	106.6
	90 and over	8.0	8.9	9.4		450 and over	(92)	(104.7)	112.0
A	90	10.9	12.1	(13.5)	E	500	117	131	138
	100	12.2	13.5	14.6		530	123	135	147
	112	13.0	14.3	15.4		560	(126)	141	155
	125 and over	14.4	15.9	17.2		630 and over	(133)	(152)	162
B	125	18.6	20.7	(23.0)	F	800	198	221	234
	140	20.8	23.0	24.8		900	(213)	239	262
	160	23.4	26.1	27.6		1000 and over	(224)	(258)	275
	180 and over	(25.2)	28.2	30.9					
C	200	34.7	38.4	41.4					
	225	38.9	43.5	46.0					
	250	(41.9)	47.0	51.5					
	280 and over	(44.1)	(50.6)	54.0					

Owing to the unequal lengths and nonuniform load exerted on the V belts of a set, it is not advisable to have more than 8 to 12. In exceptional cases, when the belts are accurately matched in length and the shafts are sufficiently rigid, up to 16 or 18 belts may be used in a set.

The design of drives with the *narrow series of V belts* is based on integrated calculations of service and pulling capacity according to ISO recommendations (see page 244). Narrow V belts have from 1.5 to 2 times the load capacity of belts of the same width in the standard series.

The belts are selected from tables listing the load that can be transmitted by each section, depending on the speed ratio, rotational speed and the diameter of the small pulley. Table 12.9 lists certain

TABLE 12.9. Power P_c Transmitted by a Narrow Series V Belt of Conditional Length L_c at an Arc of Contact $\alpha = 180°$ with the Pulley and a Steady Load

Diameter of small pulley	Transmitted power (kW) (per belt) at n, rpm 950	1450	2800
	Section NO at $L_0 = 1.6$ m		
63	0.78-[illegible]	1.08-1.23	1.74-2.02
71	0.9[illegible]	1.40-1.55	2.29-2.58
80	1.2[illegible]	1.74-1.90	2.90-3.19
90	1.[illegible]	2.13-2.28	3.55-3.84
	Section NA at $L_0 = 2.5$ m		
90	1.7[illegible]	2.35-2.69	3.64-4.29
100	2.1[illegible]	2.94-3.28	4.64-5.28
112	2.60-2.8[illegible]	3.65-3.98	5.79-6.44
125	3.12-3.34	4.39-4.73	6.99-7.65
	Section NB at $L_0 = 3.55$ m		
140	4.29-4.75	5.90-6.61	8.54-9.86
160	5.47-5.93	7.58-8.24	10.9-12.3
180	6.62-7.09	9.20-9.86	13.0-14.4
200	7.80-8.24	10.7-11.4	14.8-16.1
	Section NC at $L_0 = 5.6$ m		
224	11.3-12.4	14.9-16.7	15.2-18.6
250	13.6-14.8	18.0-19.7	16.3-20.3
280	16.3-17.4	21.2-22.9	17.4-20.8
315	19.1-20.3	24.6-26.3	—

The smaller values of P_c [illegible] $u = 1.2$, the larger to $u \geq 3$.

data from such tables for the principal speeds of induction motors and the more frequently used standard diameters of pulleys.

The power P which can be transmitted by a belt under actual working conditions is determined by multiplying the tabular value P_c by the correction factors. Thus

$$P = \frac{P_c C_\alpha C_L C_M}{K}$$

where K = dynamic load and service factor (see Table 12.5)

C_α = angle (arc) of contact factor (see Table 12.6)

C_L = factor taking the belt length into account: $C_L = \sqrt[m]{\frac{L_0}{L}}$ (L = belt length, L_0 = conditional length of the belt, and m = power from the belt fatigue curve equation; m may be taken equal to 6)

under the same [illegible]
12 kgf/cm². For synthetic belts with a polyamide film at $\sigma_0 = 75$ kgf/cm², $\frac{\delta}{D} = \frac{1}{100}$, $\alpha = 180°$ and $v = 10$ m/s, the allowable stress $[k]_0 = 60$ kgf/cm². These are among the most typical conditions for belt drives.

Allowable stress $[k]_0$ is related to φ_c by the equation: $[k]_0 = 2\varphi_c\sigma_0$.

The coefficient of friction between the belt and pulley is reduced with an increase in the normal pressure which, all other conditions being equal, [illegible] upon the pulley diameter and the belt thickness. Therefore [illegible] allowable useful stress also depends upon these factors, wh[illegible] has been experimentally investigated.

The influence of [illegible] service conditions, arc of contact, velocity and other factors [illegible] allowable useful stress $[k]$, has also been established experimentally.

In operation under damp or dusty conditions, the allowable useful stress is to be reduced by 10 to 30%.

If the rims of the pulleys are made of laminate fabric base or other plastics or wood, the coefficient of friction increases and the allowable useful stress can be increased by 20%.

The influence of the principal parameters of the drive and the service conditions is taken into account by correction factors by means of which the design useful stress is found for the actual working conditions of the drive. Thus,

$$[k] = [k]_0 C$$

where $C = C_0 C_\delta C_\alpha C_v$

C_0 = factor taking into account the tensioning conditions and the arrangement of the drive.

If a belt drive is horizontal or nearly so, the weight of the belt, if the loose side is on top, improves its grip on the pulleys. But in vertical drives or ones nearly vertical, this weight impairs belt grip on the lower pulley. Factor $C_0 = 1$ for drives with automatic belt tensioning by means of a weight or spring. In drives in which the belt is periodically tightened, $C_0 = 1$ if the line of centres is inclined through an angle from 0 to 60° to the horizontal; $C_0 = 0.9$ for an angle from 60° to 80° and $C_0 = 0.8$ for an angle from 80° to 90°. Factor C_0 is reduced by 10% more for crossed-belt and 20% for crossed-axes drives.

Factor C_δ takes into consideration the influence of the ratio $\frac{\delta}{D}$; $C_\delta = 1.2 - 5\frac{\delta}{D}$ for rubber belts; $1.6 - 15\frac{\delta}{D}$ for leather belts; $1.4 - 10\frac{\delta}{D}$ for canvas and woolen belts; and $1.6 - 60\frac{\delta}{D}$ for synthetic belts.

capacity of the drive decreases with the angle of contact α (Table 12.6). Factor $C_\alpha = 1 - c_\alpha(180 - \alpha)$. For flat belts $c_\alpha = 0.003$; for V belts at $\alpha = 150°$ to $180°$, $c_\alpha = 0.0025$.

TABLE 12.6. Angle of Contact Factor C_α

Belt	Values of factor C_α at an angle of contact of							
	70°	80°	90°	100°	110°	120°	[illegible]	[illegible]
Flat	—	—	—	—	—	[illegible]	[illegible]	[illegible]
V	0.56	0.62	0.68	0.71	0.78	[illegible]	[illegible]	[illegible]
Belt	Values of factor C_α at an angle of contact of							
	150°	160°	170°	180°	190°	200°	210°	220°
Flat	0.91	0.94	0.97	1.00	1.05	1.10	1.12	1.15
V	0.92	0.94	0.98	1.00	—	—	—	—

Velocity factor C_v, introduced for drives without automatic belt tensioning by means of a weight or spring, takes into account the loosening of the grip of the belt on the pulleys due to the centrifugal force (Table 12.7). It is equal to

$$C_v = 1 - c_v(0.01v^2 - 1)$$

TABLE 12.7. Velocity Factor C_v

Belt	Values of factor C_v at a belt velocity [illegible]						
	1	5	10	15	20	25	30
Ordinary flat	1.04	1.03	1.00	0.95	0.88	0.79	0.68
High-speed synthetic flat	—	—	1.00	0.99	0.97	0.95	0.92
V	1.05	1.04	1.00	0.94	0.85	0.74	0.60

For medium-speed flat belts of conventional materials, $c_v = 0.04$; for high-speed rubber belts it is 0.03; for high-speed canvas belts, 0.02; for high-speed synthetic belts, 0.01; and for V belts, 0.05.

The final values of the useful force F (kgf) and power P (kW) transmitted by a belt are

$$F = \frac{A[k]_0 C}{K} \quad \text{and} \quad P = \frac{vA[k]_0 C}{102K}$$

16*