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Design, Development and Performance Evaluation of the Manually Controlled Rotary Type Paddy Cutter with the Knapsack Sprayer Power Utilized

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Manual harvesting of paddy is more time consuming and has become a costly operation. Hence there is a need for mechanization of harvesting, at present the available machinery for harvesting are large in size and expensive and are not reaching the small scale farmers. The rotary type cutting mechanism has been developed with the proper transmission system by using the knapsack power sprayer engine as a prime source. This entire equipment can be utilized as a multipurpose for both operations separately as a sprayer and as a paddy harvester. In this study the engine of 1.64 kW with rated RPM of 5600 has been selected and calculated the flexible shaft diameter as a 16mm and to reach the required rpm of the cutter shaft. The transmission system with bevel gear of reduction of 1:2 has been designed. The trapezoidal shape of blades with serrations have been designed with the height of cut was 25 mm. the rotary cutter diameter has been designed 300 mm with the peripheral speed of 60 m/s for cereal crops. The overall transmission efficiency of the machine with input speed of 4800 with output speed at cutter shaft is achieved 75%. The theoretical field capacity of the paddy harvester has been achieved as 0.082 ha/h.

Introduction

Manual harvesting of paddy is more time consuming, and has been become a costly operation. Manual harvesting of paddy with local sickle takes 170 – 180 man-hours per hectare. Hence, there is a need for the mechanization of harvesting. At present the available machinery for harvesting are large in size and expensive, and are not within the reach of marginal farmers. The present day harvesters have two types of cutting mechanisms, one is of shear type reciprocating mechanism and other is of impact type rotary cutter.

The reciprocating cutter bar is commonly used for harvesting cereal crops and pulse crops. Rotary type disc cutter has more scope and potential in harvesting because they are simple and sturdy in construction. Due to the fewer moving parts in the rotary cutting system the design becomes more reliable and convenient to handle. Hence, there is a need to develop a rotary harvester, there by project on Design, Development and Performance Evaluation of the Manually Controlled Rotary type paddy cutter with the knapsack sprayer power utilized" has been taken up with the following objectives.

To design the rotary cutter for the paddy harvester.

To design the Transmission system for the Cutting system.

To test the overall efficiency of the machine.

To evaluates the field capacity of the machine.

Materials and Methods

The details of design of Manually Controlled Rotary type paddy cutter with the knapsack sprayer power utilized and the procedure for testing the speed transmission efficiency are below.

Design the rotary cutter for the paddy harvester consists of Details of power source.

Design of Cutting system and Design of Transmission system

Details of power source

The existed Honda engine has been taken as power source for operating the power operated and manually controlled paddy harvester.

The specifications of the power source have been shown in table 1.

One end of the flexible shaft has been connected to the crankshaft b threading.

Design of cutter assembly

Majumdar *et al.*, (1983) presented the kinematics of a single rotary blade harvesting machine.

The optimized equation has been presented as

$$Vm / Ve \leq nh / 2 \pi R$$

Where, Vm = Machine forward speed, m/s; Ve = Cutter peripheral velocity, m/s; n = No. of teeth on the periphery of the disc; h = Height of the teeth over the disc of the cutter, m; R = Radius of the cutter, m

Machine forward speed which is the operator's average walking speed,

Vm = 2.73kmph = 0.76 m/s (Srinivas *et al.*, 1996).

The peripheral velocity should be between 20 m/s and 60 m/s for cutting the cereal crops) Akritiids, 1974).

The speed of the cutter shaft = 3000 RPM

We know, peripheral velocity of the cutter, $Ve = \frac{\pi DN}{60} \leq 60$ m/s

Where, D = Diameter of the Cutter; N = Speed of the cutter, 3000 RPM

To have the velocity of the cutter,

$$20 \leq \frac{\pi DN}{60} \leq 60 = 0.13 \leq D \leq 0.38$$

Therefore diameter of the cutter bar should be between 0.13 m and 0.38 m.

So an intermediate diameter of the cutter of 0.3 m has been taken.

Now, the peripheral velocity of the cutter,
Vc = $\frac{\pi \times 0.3 \times 3000}{60}$ = 47.12 m/s

Height of the teeth on the cutter (h)

Height of the teeth has been taken as 2.5 cm, to accommodate to cut the crop at least a single stubble at a time.

No. of teeth on the periphery (n)

This depends on the forward speed velocity and energy of cutting.

One revolution of cutter gives the energy = Torque at the cutter x (2 π^c)

Torque at the cutter

Power produced by the engine at 4800 RPM

= $\frac{\text{RPM of the Engine}}{\textit{Maximum RPM}}$ X Rated Power

= $\frac{4800}{5600}$ x 0.82 = 0.71 kW

Power available at the cutter = Power Produced by the engine x Efficiency of the transmission system

= 0.70 x 0.75 = 0.52 kW = 520 W

Torque at the cutter = $\frac{\textit{Power}}{\text{Angular Velocity}}$

Angular velocity of the cutter shaft = $\frac{\pi DN}{60}$ = $\frac{2\,\pi \times 3000}{60}$ = 314.15 rad/s

Torque at the cutter = $\frac{520}{314.15}$ = 1.66 N-m = 0.17 kg – m

One revolution of the cutter gives the energy = 0.17 x 2 π^c = 1.07 Kg-m

For one revolution of the cutter the time taken in seconds= $\frac{60}{\text{RPM of the cutter}}$

= $\frac{60}{3000}$ = 0.02 s, in 0.02 s, the machine moves a forward distance

= (Machine forward speed) x time

= 0.76 x 0.02 = 0.015 m nearly 0.02m

Area covered in 0.02s

= 0.02 m x (diameter of the cutter) = 0.02 x 0.3 = 0.006 m^2

No. of hills in 0.006 m^2 area to be cut by the cutter

Consider a least spacing of 0.1 m x 0.1 m

Therefore No. of hills in 0.006 m^2 = $\frac{0.006}{0.1 \times 0.1}$ = 0.6 = nearly 1 hill

Maximum transverse impact rupturing work of 1 single stem has been taken as 0.0508 Kg-m

Considering a hill contains 12 stems

The energy required to rupture 1 hill = 0.0508 x 12 = 0.61 Kg-m

Hence it has been observed that the cutter has = $\frac{1.07-0.61}{1.07}$ x 100 = 50 % of excess energy

It has been considered that the 50% of energy can be utilized to accelerate the cutter and energy losses in bending while impact. Thus the cutter energy is sufficient.

So, no. of teeth. n required on the periphery can be found by the following equation

$$= \frac{\textit{Forward movement of the machine in 1 s}}{\text{Height of the teeth on the cutter}} = \frac{0.02}{0.025}$$

= 0.8 = nearly 1 teeth

But to accommodate uniform energy utilization under closed cropping it has been selected 4 no. of teeth on the periphery.

Therefore No. of teeth on the periphery, n = 4

For verification of the design parameters of the cutter, kinematics equation of a cutter has been used.

$$\frac{Vm}{\text{Vc}} \leq \frac{nh}{2\pi\text{R}}$$

$$= \frac{0.76}{47.12} \leq \frac{4\ X\ 0.025}{2\ \text{X}\ \pi\ \text{X}\ 0.15} = 0.016 \leq 0.106$$

The above relation says that underutilization of energy, but it is not true because it has been taken no. of teeth, n = 4 instead of n = 1, By substituting n = 1

Which gives

$$\frac{Vm}{\text{Vc}} \leq \frac{nh}{2\pi\text{R}} = \frac{0.76}{47.12} \leq \frac{1\ X\ 0.025}{2\ \text{X}\ \pi\ \text{X}\ 0.15} = 0.016 \leq 0.026$$

which has been satisfied the equation of the kinematics.

Cutter shape

An oblique angle of 25° has been provided. A sharpening angel of 20° has been provided based on the data given on the research data.

Serration are been given for efficient penetration on 3mm hardened tool steel.

The cutter teeth have been fixed on a 3 mm MS sheet of 25 cm diameter.

The MS sheet has been fixed with bolts and nuts below a 1.5 kg. MS disc of 12.5 cm which is attached to the cutter shaft with the help of welding.

Transmission system

Transmission consists of flexible Shaft

Calculation of diameter of flexible shaft

The shaft encounters only power operated and manually controlled paddy harvester twisting moments with suddenly applied load with a major shock.

Diameter of a flexible shaft has been found torsion equation by considering the solid mild steel shaft.

Torsion equation

$$\frac{T}{J} = \frac{Fs}{r}$$

Where,

T = Torque acting on the shaft, N-m

J = Polar moment of inertia of the shaft about the axis of rotation, m4

Fs = Maximum Allowable shear stress in the material, N/m2

R = Distance from the neutral axis to the outermost fibre = d/2, m

D = Diameter of the shaft

By substituting the, for $J = \frac{\pi d4}{32}$ and Simplifying,

Torque acting on the shaft,

$$\text{T} = \frac{\pi \text{ x fs x } d3}{16} \text{ ----------- (1)}$$

Totque, T can be found from the equation of power

$P = \frac{2\pi NT}{60}$ --------- (2)

Where, P = Rated power of the power source, W; N = RPM of the power source, RPM; T = Torque supplied by the power source, N-m

By taking the speed of the power source,

N = 4800 RPM; P = 0.82Kw = 820 W

Substituting in equation (2)

$820 = \frac{2 \text{ X } 3.142 \text{ x} 4800 \text{ x } T}{60}$

T = 1.63 N-m = 1630 N-m

By considering the factor for suddenly applied load with major shock, Kt = 2.0

Therefore the Equivalent torque, Te = Kt × T = 2 × 1630 = 3260 N-mm

Substituting the Equivalent torque in place of torque acting on the shaft in equation (1) and maximum allowable shear stress for the mild steel.

Fs = 56 N/mm2,

$3260 = \pi \times 56 \times d^3 / 16$

d = 6.67 mm

Therefore the diameter of the shaft has been taken with the higher available diameter of 16mm. Other end of the flexible shaft is connected to the pinion shaft with a suitable coupling.

Gear box

Gear box has been provided for two purposes

To reduce rpm by 1.2 times. To change the axis of transmission horizontal to vertical.

One end of the pinion shaft has been attached with flexible shaft and other end has been fixed with pinion.

Gear shaft has been fixed vertically to the ground. One end of the gear shaft has been fixed with the gear to match with the pinion and other end has been for mounting the cutter.

Testing of the overall transmission efficiency

The RPM at the engine has been fixed and the flexible shaft of the assembled power operated and manually controlled paddy harvester has been attached to the crank shaft.

The RPM at the end of the gear box shaft and cutter shaft has been measured by different trials. The transmission efficiency has been calculated using the following equation.

Sped transmission efficiency = $\frac{\textit{Output Speed}}{\text{Input Speed}}$ x 100

$= \frac{N2}{N1} \text{ X } 100$

Where,

N1 = RPM of the Engine
N2 = RPM at the end of the cutter shaft

Results and Discussion

Gear box has been tested for speed transmission efficiency and the reading has been noted in table 3. Speed transmission efficiency at different speeds has been calculated. A graph between input speed and transmission efficiency have been shown in above, it has been observed that the speed transmission efficiency varies between 90.5 to 91% for input speed variation from 3600 – 5600 (Fig. 1).

Table.1 The specifications of the power source

Rated power	1.64 kW
RPM	3600 – 5600
Weight	5 kg

Table.2 Specifications of the gear box

Type of Gears	Bevel Gears
No.of Teeth	Pinion = 12, Gear = 14
Shafts	Pinion shaft Diameter = 14 mm Gear shaft Diameter = 16 mm
Casing	Cast iron
Lubrication	Grease
Bearings	Ball Bearings

Table.3 Measurement of the RPM at the end of the gear shaft

S.no	**Engine Speed, N_1 RPM**				**Engine Speed, N_2 RPM**				**$\eta\% = \frac{N2}{N1}$ X 100**
	Trial 1	**Trial 2**	**Trial 3**	**Mean**	**Trial 1**	**Trial 2**	**Trial 3**	**Mean**	
1	5420	5510	5505	5478	4900	4936	5020	4984	91.2
2	5290	5320	5390	5333	4850	4920	4925	4847	90.9
3	4890	4789	4796	4825	4150	4270	4500	4381	90.8
4	4510	4210	4110	4276	3950	4020	4095	3878	90.7
5	4120	3900	4020	4013	3730	3850	3945	3635	90.6
6	3620	3611	3609	3613	3330	3420	3485	3269	90.5

Table.4 Measurement of the RPM at the end of the cutter shaft

S.no	**Engine Speed, N_1 RPM**				**Engine Speed, N_2 RPM**				**$\eta\% = \frac{N2}{N1}$ X 100**
	Trial 1	**Trial 2**	**Trial 3**	**Mean**	**Trial 1**	**Trial 2**	**Trial 3**	**Mean**	
1	5560	5540	5550	5550	4200	4202	4184	4195	74.9
2	5220	5210	5210	5210	3920	3900	3880	3900	74.8
3	4810	4610	4870	4830	3600	5550	3480	3543	73.5
4	4470	4380	4820	4436	3240	3220	3180	3200	72.1
5	4230	3920	4040	4070	3020	2900	2830	2915	71.8
6	3600	3650	3640	3626	Over load	Over load	Over load	Over load	----------

Specifications of the power operated manually controlled paddy harvester

Power source	Knapsack mist blower engine, 0.82kW Two stroke petrol engine 3600 – 5600 rpm
Transmission System	Flexible shaft 16mm diameter and 2 m length. And gear box with bevel gears.
Height of cut	10cm
Diameter of cutter	30 cm
Design RPM of cutter	3000 RPM
Blade shape	Trapezoidal serrated
Design forward speed of the machine	2.73 kmph

Fig.1 Variations of speed and overall speed transmission efficiency of gear shaft of the machine assembly with respect to engine speed

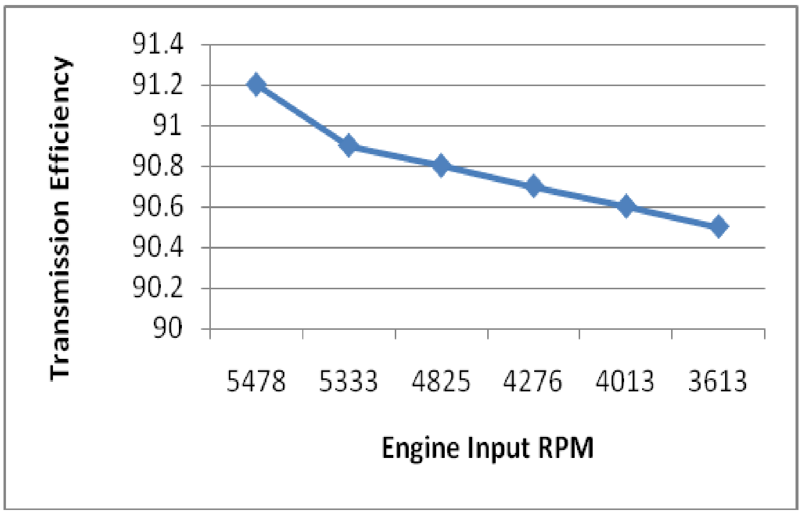
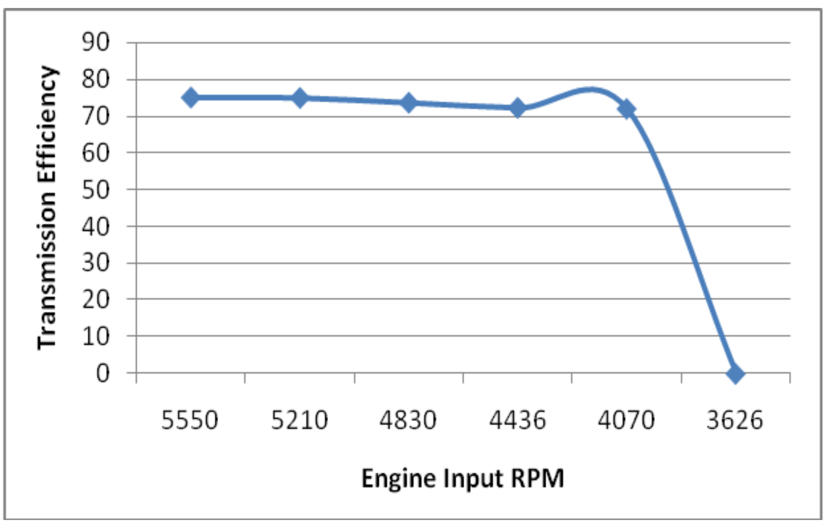


Fig.2 Variations of speed and overall speed transmission efficiency of cutter shaft of the machine assembly with respect to Engine speed



Power operated and manually controlled paddy harvester has been tested for the speed transmission efficiency and readings have been noted in table 4. Speed transmission efficiency at different speeds has been calculated.

A graph between input speed and transmission efficiency have been shown in figure 2, it has been observed that the speed transmission efficiency varies between 71.8 – 74.9 for the input speed variations from 4000 – 5600 rpm.

The machine has been automatically stopped due to the load coming on to the engine.

Field capacity of the machine

$$\text{Field capacity} = \frac{\text{Area Covered ,ha}}{\text{Time taken to cover the area}}$$

Take area covered = 1 hectare

Forward speed of the machine = 2.73 kmph (Design speed)

= 2.73 x 1000 / 3600 = 0.76 m/s

Width of cut = 0.3 m

Distance to be moved by the machine to cover 1 hectare

= 1 x 10000 / 0.3 = 33333.3m

Time to be taken by the machine to move 33333.3 m

= 33333.3 / 0.76 = 43859.6 s = 12.18 hours

Field capacity = 1 / 12.18 = 0.082 hectare / hour.

This paper deals with the Design, Development and Performance Evaluation of the Manually Controlled Rotary type paddy cutter with the knapsack sprayer power utilization with the following objectives

To design the rotary cutter for the paddy harvester.

To design the Transmission system for the Cutting system.

To test the overall efficiency of the machine.

To evaluates the field capacity of the machine.

The power operated and manually controlled paddy harvester which is given by knapsack mist blower engine and can be controlled manually has been developed. The power source had been kept at 4800 RPM to give 3000 RPM at the cutter with the overall speed transmission efficiency of 75%.

The transmission system has been designed to give a speed reduction of 1.2 by providing bevel gear system. From the designed calculations and dimensions of the machine, it has been calculated the theoretical field capacity as 0.08 ha/h with a width of cut of 30 cm.

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