Original Research Article  

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

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Abstract  

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.  

Keywords  

Paddy harvester, Rotary cutter for paddy harvester, Paddy cutter and sprayer powered paddy harvester, Field capacity of paddy cutter.  

Article Info  

Accepted: 17 September 2017  
Available Online: 10 November 2017  

Introduction  

Manual harvesting of paddy is more time consuming, and has 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 evaluate 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 by 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

\[ \frac{V_m}{V_e} \leq \frac{nh}{2\pi R} \]

Where, \( V_m \) = Machine forward speed, m/s; \( V_e \) = 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,

\[ V_m = 2.73 \text{kmph} = 0.76 \text{m/s} \] (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,

\[ V_e = \frac{\pi DN}{60} \leq 60 \text{m/s} \]

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, 
\[ V_c = \frac{\pi \times 0.3 \times 3000}{60} = 47.12 \text{ m/s} \]

**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 \( \times (2\pi) \)

**Torque at the cutter**

Power produced by the engine at 4800 RPM
\[ = \frac{\text{RPM of the Engine}}{\text{Maximum RPM}} \times \text{Rated Power} \]
\[ = \frac{4800}{5600} \times 0.82 = 0.71 \text{ kW} \]

Power available at the cutter = Power Produced by the engine \times Efficiency of the transmission system
\[ = 0.70 \times 0.75 = 0.52 \text{ kW} = 520 \text{ W} \]

Torque at the cutter = \[ \frac{\text{Power}}{\text{Angular Velocity}} \]

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

Torque at the cutter = \[ \frac{520}{314.15} = 1.66 \text{ N-m} = 0.17 \text{ kg} - \text{m} \]

One revolution of the cutter gives the energy = \[ 0.17 \times 2\pi^2 = 1.07 \text{ Kg-m} \]

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

= \[ \frac{60}{3000} = 0.02 \text{ s} \]

in 0.02 s, the machine moves a forward distance

= (Machine forward speed) \times \text{time}

= 0.76 \times 0.02 = 0.015 \text{ m} \]

**Area covered in 0.02s**

\[ = 0.02 \times (\text{diameter of the cutter}) = 0.02 \times 0.3 = 0.006 \text{ m}^2 \]

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

Consider a least spacing of 0.1 m \times 0.1 m

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

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 \times 12 = 0.61 \text{ Kg-m}

Hence it has been observed that the cutter has \[ \frac{1.07 - 0.61}{1.07} \times 100 = 50\% \text{ of excess energy} \]

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
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.

\[ \frac{V_m}{V_c} \leq \frac{nh}{2\pi R} \]

\[ = \frac{0.76}{47.12} \leq \frac{4 \times 0.025}{2 \times \pi \times 0.15} = 0.016 \leq 0.106 \]

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

\[ \frac{V_m}{V_c} \leq \frac{nh}{2\pi R} = \frac{0.76}{47.12} \leq \frac{1 \times 0.025}{2 \times \pi \times 0.15} = 0.016 \leq 0.026 \]

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

\[ T = \frac{Fs}{J} \]

Where,

\[ T = \text{Torque acting on the shaft, N-m} \]

\[ J = \text{Polar moment of inertia of the shaft about the axis of rotation, m}^4 \]

\[ F_s = \text{Maximum Allowable shear stress in the material, N/m}^2 \]

\[ R = \text{Distance from the neutral axis to the outermost fibre = } \frac{d}{2}, \text{ m} \]

\[ D = \text{Diameter of the shaft} \]

By substituting the, for \[ J = \frac{\pi d^4}{32} \]

Simplifying,

\[ \text{Torque acting on the shaft,} \]

\[ T = \frac{\pi x f_s x d^3}{16} \] \[ \text{--------- (1)} \]

\[ \text{Torque, T can be found from the equation of power} \]
P \text{ = } \frac{2\pi NT}{60} \quad \text{(2)}

Where, P = \text{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 \text{ RPM; P = 0.82Kw = 820 W}

Substituting in equation (2)

820 = \frac{2 \times 3.142 \times 4800 \times T}{60}

T = 1.63 \text{ N-m} = 1630 \text{ N-m}

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

Therefore the Equivalent torque, Te = Kt \times T = 2 \times 1630 = 3260 \text{ 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 \text{ N/mm2},

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

d = 6.67 \text{ 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.

\[ \text{Sped transmission efficiency} = \frac{\text{Output Speed}}{\text{Input Speed}} \times 100 \]

\[ = \frac{N2}{N1} \times 100 \]

Where,

N1 = \text{RPM of the Engine}
N2 = \text{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, N₁ RPM | Engine Speed, N₂ RPM | η% = \( \frac{N₂}{N₁} \times 100 \) |
|------|----------------------|----------------------|-----------------|
|      | 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, N₁ RPM | Engine Speed, N₂ RPM | η% = \( \frac{N₂}{N₁} \times 100 \) |
|------|----------------------|----------------------|-----------------|
|      | 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

| Specification                        | Details |
|--------------------------------------|---------|
| 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

![Graph showing transmission efficiency vs engine input RPM]

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

![Graph showing transmission efficiency vs engine input RPM]
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)

\[
= \frac{2.73 \times 1000}{3600} = 0.76 \text{ m/s}
\]

Width of cut = 0.3 m

Distance to be moved by the machine to cover 1 hectare

\[
= 1 \times 10000 / 0.3 = 33333.3 \text{ m}
\]

Time to be taken by the machine to move 33333.3 m

\[
= 33333.3 / 0.76 = 43859.6 \text{ s} = 12.18 \text{ hours}
\]

Field capacity = \[
1 / 12.18 = 0.082 \text{ hectare / hour}
\]

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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How to cite this article:
Ravindra Raju, R., P. Shankar Prasad and Rayan Rao, V. 2017. Design, Development and Performance Evaluation of the Manually Controlled Rotary Type Paddy Cutter with the Knapsack Sprayer Power Utilized. Int.J.Curr.Microbiol.App.Sci. 6(11): 2187-2195.
doi: https://doi.org/10.20546/ijemas.2017.611.258