Design and Development of Intercrop Grass Removing Machine

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Abstract— the aim of paper is design and development of self-powered and self-propelled agricultural device which can use to remove grass from crops having intercrop distance so less that tractor and such machines not useful over there because of bigger size. Reduce size and cost of machine by exploring the idea of narrowing down the utility and making soil specific machine.

Key words: Intercrop Grass Removing Machine, Intercrop

I. INTRODUCTION

At present, most of the power tiller manufactured in the country are in the range of 8-10 hp and weigh about 400 kg. The power tillers are not potentially used in hilly areas due to the lack of its manoeuvrability on slopes. This is primarily due to its heavy weight, which needs to be optimized further. There was a long felt need to develop a lightweight, portable and propelled two-wheeled walking type tractor for use in hilly areas, orchards and small farms. Such power tiller should also be able to be used for inter row tillage and other inter-culture utilities. It should be light enough for two persons to easily lift it manually for shifting from one field to another. This feature is particularly useful when operating in terrace fields and fields with high bounds. Therefore it is felt necessary to develop a lightweight power tiller fitted with 2-4hp engines.

II. PROBLEM DEFINITION

As per the discussion with Climber Engineering it is found that they are facing many problems regarding design and development of special purpose power tiller, which are listed below:

1) Design and develop a lightweight power tiller for operation in farms with minimum inters row distance of 1.2m which also leads to reduction of cost.
2) Make easy maintenance by standard replaceable spares

III. OBJECTIVE OF THE STUDY

1) To reduce the size and cost of the machine.
2) To make it easy to operate and simple in construction.

IV. ANALYTICAL DESIGN CALCULATION FOR POWER TILLER

We explored the idea of narrowing down the utility and making a soil specific machine so as to reduce size and cost of machine.

We choose a petrol/ kerosene engine of 3.5 hp and 3000rpm. Instead of gear drive, we decided to use a chain drive for speed reduction to reduce cost and make the machine compact.

A. *Designs for Tilling Speed*

Required Speed for Tilling: 150 TO 200 rpm [agriculture dept. R&D]
Required Power: 3.5Hp

Engine Specifications:

– 3.5Hp=2.61 KW
– Petrol Start, Kerosene Operated GREEVES Engine
– Cylinder Capacity= 100 cc
– Speed of engine: $N=N_0=3000$ rpm
– Engine Pulley A Diameter= 75 mm

D = diameter of sprocket
N = speed of rotation of shaft

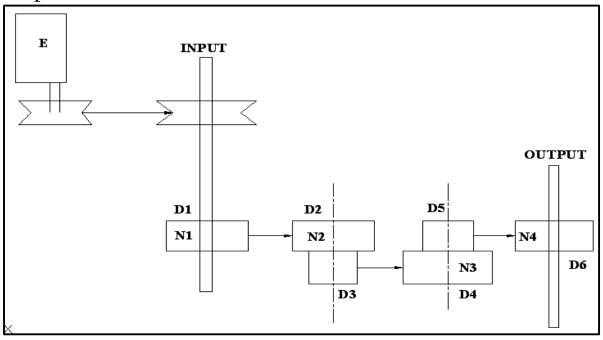

Fig. 1: Speed Reduction Mechanism

Input Shaft Pulley B Dia. = 5’’= 125 mm [1]

$$\frac{N0}{N1}=\frac{DB}{DA}\ \frac{3000}{N1}=\frac{125}{75}$$

∴N1=1800 rpm

By trial and error method and by using previous machine data, we assume sprocket diameters as follows:

1) D1=55.5mm
2) D2=174.3mm
3) D3= 60.4mm
4) D4=174.3mm
5) D5=60.4mm
6) D6=70.2mm

$$\frac{N1}{N2}=\frac{D2}{D1}$$

$$N2=\frac{N1D1}{D2}=\frac{1800\times 55.5}{174.3}$$

∴N2=573.149rpm

$$\frac{N2}{N3}=\frac{D4}{D3}$$

$$N3=\frac{N2D3}{D4}=\frac{573.149\times 60.4}{174.3}$$

∴N3=198.613 rpm

$$\frac{N3}{N4}=\frac{D6}{D5}$$

$$N4=\frac{N3D5}{D6}=\frac{198.613\times 60.4}{70.2}$$

∴N4=170.88 rpm

B. *Torque Calculations*

1) *Input Torque (Tp)*

$$P=\frac{2\pi NT}{60000}$$

$$T=\frac{60\times 1000\times P}{2\pi N}$$

$$Tp = \frac{60\times1000\times2.61}{2\pi N1}$$

$$Tp = \frac{60\times1000\times2.61}{2\pi\times1800} = 13.8465 \text{ Nm}$$

Tp =13846.5 N mm

2) *Intermediate Shaft 1 Torque(Tq)*

$$Tq = \frac{60\times1000\times2.61}{2\pi\times N2} = \frac{60\times1000\times2.61}{2\pi\times573.149}$$

Tq = 43.485 Nm
Tq = 43485 N mm

3) *Intermediate Shaft 2 Torques (Tr)*

$$Tr = \frac{60\times1000\times2.61}{2\pi\times N3} = \frac{60\times1000\times2.61}{2\pi\times198.613}$$

=125.4885 Nm
Tr =125488.5 N mm

4) *Output Shaft (Ts)*

$$Ts = \frac{60\times1000\times2.61}{2\pi\times N4} = \frac{60\times1000\times2.61}{2\pi\times170.88}$$

Ts =145.8548Nm

5) *Torque Produced By Engine (Te)*

$$Te = \frac{60000\times2.61}{2\pi\times3000} = 8.3079\text{Nm}$$

Te = 8307.9 N mm

C. *Thrust Calculations :(For Chain Selection)*

Maximum Torque Is Given By Output Shaft, So Maximum Force Is On Output Shaft.

$$F_{MAX} = F_S = \frac{TS}{RS} = \frac{TS}{D6/2} = \frac{145.8548}{35.1\times0.001}$$

F_{MAX}=4155.4074 N
F_{MAX} =424 KGF
This Is Maximum Force In The Chain.

1) *Chain Selection*

From Design Data Book, We Select Chain 10B Simple Type.
Pitch=15.87mm=P
Roller Diameter =Dr=10.16mm
Breaking Load of This Chain=2270KGF

Such A Higher Grade Of Chain Is Used Since The Load On The Machine Will Be Impact Loading And Of Unknown Magnitude At Best Of Times.

D. *Sprocket Specifications*

$$D = \frac{P}{SIN\left(\frac{180}{Z}\right)}, \quad Z = \text{No. of Teeth}$$

Sr No.	Sprocket	Diameter	Teeth
1	D1	55.5 mm	11
2	D2	174.3 mm	35
3	D3	60.4 mm	12
4	D4	174.3 mm	35
5	D5	60.4 mm	12
6	D6	70.2 mm	14

Table 1: Sprocket Specifications

E. *Design of No of Chain Links*

1) *For Shaft P to Q*

p=Pitch
a=centre distance (30-50)*p

For compact design and space limitations, we take a=155mm.

For (P-Q) the compensation is done by using an idler to provide required Chain lap angle.

$$Ln_{(P-Q)} = \left(2\times\frac{a}{p}\right) + \left(\frac{z1+z2}{2}\right) + \left[\left(\frac{z2-z1}{2\pi}\right)^2 \times \frac{p}{a}\right]$$

$$= \left(2\times\frac{155}{15.875}\right) + \left(\frac{11+35}{2}\right) + \left[\left(\frac{35-11}{2}\right)^2 \times \frac{15.875}{155}\right]$$

= 44.022

$\boxed{Ln_{(P-Q)}}$=45 links

2) *For Shaft Q to R*

a=169mm

$$Ln_{(Q-R)} = \left(2\times\frac{169}{15.875}\right) + \left(\frac{12+35}{2}\right) + \left(\frac{35-12}{2\pi}\right)^2 \times \left(\frac{15.875}{169}\right)$$

=46.05

$\boxed{Ln_{(Q-R)}}$=47 links

3) *For Shaft R to S*

a=135mm

$$Ln_{(R-S)} = \left(2\times\frac{135}{15.875}\right) + \left(\frac{12+14}{2}\right) + \left[\left(\frac{14-12}{2\pi}\right)^2 \times \left(\frac{15.875}{135}\right)\right]$$

=30.019

$\boxed{Ln_{(R-S)}}$=31 links

For odd no. of chain links we add one more link called offset link.

So we have,

Shaft	Center Distance	No. Of Links
P-Q	155mm	46
Q-R	169mm	48
R-S	135mm	32

Table 2: No of Chain Links

The corrected centre distance between sprockets is given by

$$a = \frac{p}{4}\left\{\left[Ln - \left(\frac{z1+z2}{2}\right)\right] + \sqrt{\left[Ln - \left(\frac{z1+z2}{2}\right)\right]^2 - 8\left[\frac{z2-z1}{2\pi}\right]^2}\right\}$$

$$a_{(P-Q)} = \frac{15.875}{4}\left\{\left[46 - \left(\frac{46}{2}\right)\right] + \sqrt{[46-23]^2 - 8\left[\frac{12}{\pi}\right]^2}\right\}$$

$\boxed{a_{(P-Q)}}$=171.865 mm

$$a_{(Q-R)} = \frac{15.875}{4}\left\{[48-23.5] + \sqrt{[48-23.5]^2 + 8\left[\frac{23}{2\pi}\right]^2}\right\}$$

$\boxed{a_{(Q-R)}}$=202.7947 mm

$$a_{(R-S)} = \frac{15.875}{4}\left\{[32-13] + \sqrt{[32-13]^2 + 8\left[\frac{2}{2\pi}\right]^2}\right\}$$

$\boxed{a_{(R-S)}}$=150.897 mm

∴Corrected center distances are as follows:

Shaft	Sprocket Center Distance	No. of Chain Links	Sprockets Involved	Z1	Z2
P-Q	171.865mm	46	D1 & D2	11	35
Q-R	202.7947mm	48	D3 &D4	12	35
R-S	150.897mm	32	D5 & D6	12	14

Table 3: Corrected Centre Distances

The corrected centre distances are not exactly used in practical machine due to space limitations.

The compensation is made using idlers.

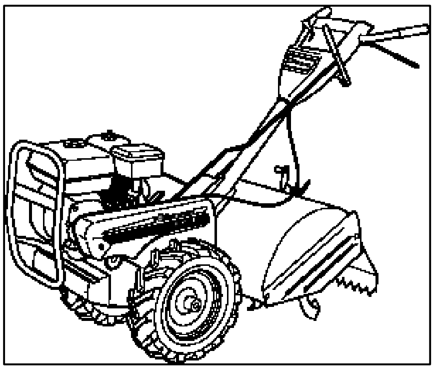

Fig. 1: idlers

V. Proposed Flow of Work & Methodology

1) Study of Present power tiller available in market.
2) Take practical input from industry.
3) Problem Identification
4) Failure Analysis of existing power tiller & mountings on power tiller.
5) Conceptual redesign of power tiller by using input & Modeling.
6) Analyze model and select best from available.
7) Material Optimization.
8) Shape Optimization.
9) Analysis for different materials.
10) Results

VI. Conclusion

From the above theoretical study we can conclude that design and development of light weight, low cost power tiller with less maintenance is critical part of power tiller. We are tried to reduce the excessive weight in power tiller and also attaining for optimization of machine cost.

Acknowledgement

I would like to express my gratitude to many peoples who have assisted me during this project work special thanks must go to my guide Prof. S. J. Madki for their continued support and guidance. Also I would like to give special thanks to our ME Coordinator, HOD, and principal.

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