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Design and development of power operated walking type weeder cum furrow maker for vegetable crops

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Abstract
The developed power operated weeder cum furrow maker machine consists of discarded scooter engine, gear reduction unit, shafts, chain sprocket mechanism, blade wheel, furrow maker plough. It is durable to bear the loads while operating the machine. Weeding working width can be adjusted from 30 cm to 70 cm with two rows working at a time. Furrow maker easily covers the row spacing of 25 cm. Engine rotates at 215 rpm which in firstly reduced by gear reduction at 16:38 and then reduction in wheel takes place with chain sprocket mechanism of 14:38.

Keywords: Weeding, furrow making, weeder cum furrow maker

Introduction
Horticulture is an integral part of agriculture. It shares 4.5 per cent in agriculture gross domestic product (GDP) which is 13.7 per cent in India. The total horticulture production in Chhattisgarh was 10.22 MT out of 0.86 million hectares in which the total vegetable production was 6.89 MT in the year 2018-19.

Weeds are the undesirable plants which absorbs plants nutrient and affect the growth. Weeding is an important intercultural operation in agriculture. 30-40% yield loss occur due to weeds. Weeding operation is very difficult practice with human labor thus a mechanical weeding technique becoming useful and is being used partially. Mechanical weeding has been considered economical compared to manual weeding. In mechanical weeding use of implements like cultivators, weeders, and harrows driven by animals or engine are common practice.

There are many mechanical weeders, but weeder cum furrower availability is very less. Hence a plan was made to develop weeder for wide row crops with a facility of furrow making. In this case a discarded petrol-based scooter engine was supposed to be used as a power source. A machine is likely to be developed by using scooter engine which acts as weeder as well as furrow making for vegetable crops. The developed machine may be useful for weeding and furrow making work. The machine can be used for weeding operation for two rows at a time with an adjustable row spacing of 30 cm to 70 cm.

Materials and Methods
Design procedure of components of weeder cum furrow maker
Power requirement of engine
The power requirement for the developed power weeder cum furrow maker was calculated by using equation 1 (Oyaole *et al.*, 2003). The fig: 1 shows the selected engine for development of power operated weeder cum furrow maker.

$$P = \frac{SR \times d \times w \times v}{75} \text{ hp} \qquad \text{(Eq.1)}$$

P = 0.94 hp = 0.70 kW

Where,
SR = soil resistance, N/mm^2
d = depth of cut, cm
w = effective width of cut, cm

$P_t = \frac{P}{\eta}$ = 0.854 kW = 1.14 h (Eq. 2)

Where,
P_t = Total power required
P = Power required for digging of soil, kW
η = Transmission efficiency, % (82 %)
v = speed of operation, ms^{-1}

Torque of the engine
The formula used for the calculation of the torque transmitted through the shaft. (Khurmi, 2012) [5]

$$T = \frac{P \times 60 \times 10^3}{2 \times \pi \times N} = 37.93 \text{ N-m}$$

Where,
P = power, kW
T = torque transmitted by the shaft, N-m
N = revolutions per minute

Fig 1: Engine

Design of shafts
Maximum tangential force

$$K_s = \frac{R_s \times 75 \times N_c \times \eta_c \times \eta_z}{u}$$ (Bernacki *et al.*, 1972) [1]

Where,
K_s = Maximum tangential force, kg
R_s = Reliability factor (1.5 for non-rocky soils and 2 for rocky soils)
N_c = Power of engine, hp
η_c = Traction efficiency for the forward rotation of rotor shaft as 0.9
η_z = Coefficient of reservation of engine power (0.7- 0.8)
u = Minimum tangential speed of blades

Allowable stress in shaft

$$\tau_{all} = \frac{0.577 \times k \times \sigma_y}{f}$$ (Mott, 1985) [6]

Where,
τ_{all} = allowable stress on main-shaft and blade-shaft, kg-cm^{-2}
k = coefficient of stress concentration (0.75)
f = factor of safety (1.5)
σ_y = yield stress, 380 MPa (Yield stress of carbon steel (40C8)

Diameter of shaft

$$D = \sqrt[3]{\frac{16 \times M_s}{\tau_{all} \times \pi}}$$

By using that calculation of the diameter of shaft which is selected as 30 mm in diameter which is suitable for the machine. Fig. 2 shows the shafts of the developed machine.

Fig 2: Rotor Shaft

Design of chain
Length of chain

$$L = L_n \times p$$

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right)$$
(Bhandari, 2010)

Where, a = centre to centre distance axes of main shaft sprocket to axes of blade-shaft (mm)
z_1 = no. of teeth on the smaller sprocket of main shaft
z_2 = no. of teeth on the larger sprocket on blade-shaft

By using that formula, we have selected length of chain from main shaft to blade shaft is 90 cm and from main shaft to wheel shaft is 178 cm. Fig. 3 shows the chain-sprocket mechanism.

Fig 3: Chain Sprocket mechanism

Design of blade
The L-shaped blade, which is ideal for weeding work, is used to increase performance. The force acting on the blade K_e was calculated by the following equation
(Shigley *et al*., 2004):

$$K_e = \frac{K_s \times C_p}{i \times Z_e \times n_e} = 27.063 \text{ kg}$$

Where,
K_e = Force acting on the blade, kg

K_s = Maximum tangential force, kg
C_p = Coefficient of tangential force
i = Number of flanges, 4
Z_e = No. of blades on each side of the flanges is 4
n_e = No. of blades which act jointly on the soil by the total no. of the blades
Considering the shape of the blades, the bending stress (σ_{zg}), shear stress (τ_{skt}), and equivalent stress (σ_{zt}) can be calculated by the following equations (Bernacki *et.al.*, 1972)

$$\sigma_{zg} = \frac{6K_s \times S}{b_e \times h_e^2} = 177.85 \text{ MPa}$$

$$\tau_{skt} = \frac{3K_s \times S_1}{\left(\frac{h_e}{b_e} - 0.63\right) \times b_e^3} = 292.57 \text{ MPa}$$

$$\sigma_{zt} = \sqrt{\sigma_{zg}^2 + 4\tau_{skt}^2} = 611.57 \text{ MPa}$$

The bending stress, shear stress, and equivalent stress were determined as 177.85 MPa, 292.57 MPa, and 611.57 MPa respectively. Fig. 4 shows the L-shaped blade which is used for weeding operation.

Fig 4: Blade

Design of furrow maker plough
Design of the furrow maker plough has to be based on various parameters of the plough which are given as:
L = length of tyne from the tip of the shovel to the frame, mm
L_b = length of tyne/shank from the frame to the top of the end of the breast, mm
h = height of tyne/shank from its tip to the bent portion, mm
R = radius of curvature of bent portion of tine (generally = 120 mm)

Also, $R = \frac{h - l\sin\alpha}{\cos\alpha}$

d = maximum operating depth of furrow opener, mm
l = breast length of the shovel, mm
α = load angle, degrees
b × t = cross-section of tyne, mm^2
b = width of tyne, mm
t = thickness of tyne, mm

Calculate the draft load on furrow opener tyne/shank (D_f): force exerted on the ridger type furrow opener for vegetable crops

$D_f = k \times w \times d = 151.78$ kg-f

Where,
D_f = draft of furrow opener, kg or N
k = specific soil resistance, kg cm^{-2} (Taken as 0.7 kg cm^{-2})
w = width of furrow opener, cm
d = depth of sowing, cm
Now, for carbon steel factor of safety will be 2.
So, design draft of furrow opener will be = $D_f \times 2 = 303.56$ kg-f

Determine the bending moment in the furrow opener According to (Dubey 1985)

$$R = \frac{h - l\sin\alpha}{\cos\alpha} = 62.60 \text{ mm}$$

Now considering the furrow opener as cantilever beam of 510 mm size fixed to the frame at the end (Krutz *et al.* 1984), maximum bending moment (M) in the tyne is given as
M = Design draft × beam span = 15481.56 kg-f-cm
Now section modulus of tyne Z is calculated as

$$\sigma_b = \frac{MC}{I} = \frac{M}{Z} \left(\text{as } Z = \frac{I}{C}\right) \quad (1)$$

Where,
σ_b = bending stress in tyne, kg cm^{-2}. We can take bending stress for mild steel as 1000 kg cm^{-2} (Sengar, 2002)
M = bending moment in tyne, kg-cm
C = distance from neutral axis to the point at which stress is calculated, cm
I = moment of inertia of rectangular section, cm^4
Z = section modulus of the tyne

Also,
For rectangular sections,

$$Z = t \times \frac{b^2}{6} \quad (2)$$

The ratio of thickness and width (t: b) = 1: 4

So, b = 4t

Therefore, from equation 2, we get

$$Z = \frac{16t^3}{6} = \frac{M}{\sigma_b} = 15.48 \text{ cm}^3$$

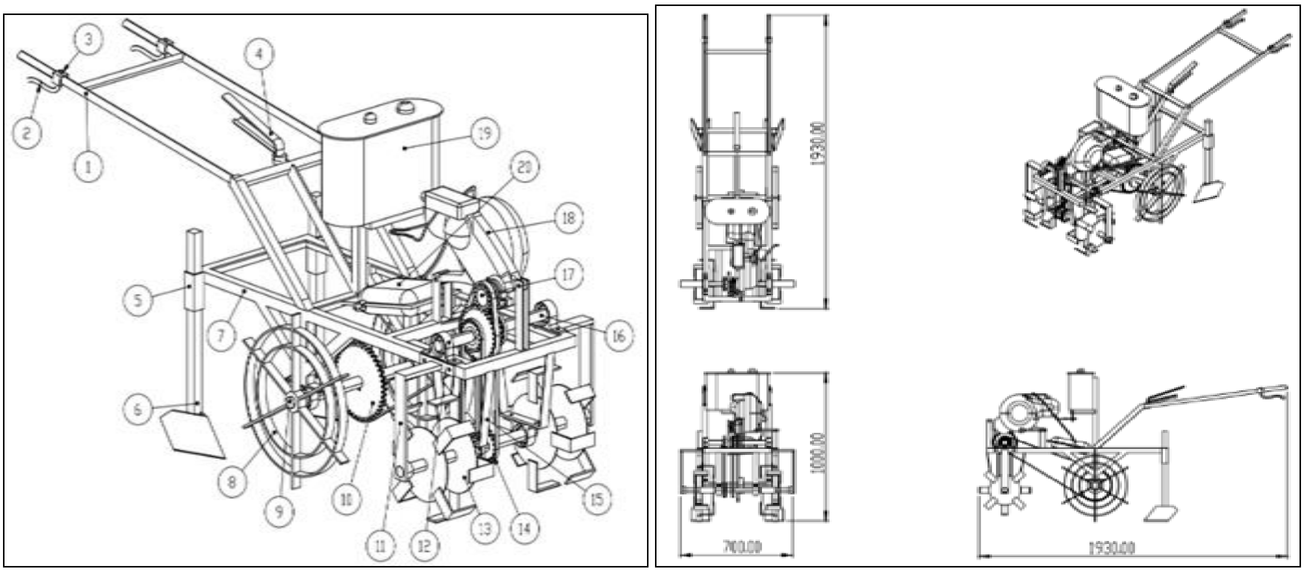
t = 17.9 mm
b = 71.6 mm

Therefore, cross-section of the tyne is = 17.9 × 71.6 mm. Fig. 5 shows the furrow maker.

Fig 5: Furrow maker

Results

The power operated weeder cum furrow maker was designed and developed for vegetables crops as per methodology and calculations mentioned in materials and methods. The prototype of power operated weeder cum furrow maker was first designed on computer aided software solid works and then fabricated by assembling all the designed and selected parts of the machine. The required power for developed prototype was 3 hp and a cubic capacity of 147.55 cc. It works on the principle of conversion of thermal to mechanical energy by using the petrol as a fuel. The engine produces the energy which gives that energy to the output shaft from that can transferred to gear reduction unit at that speed, blades of machine revolve by using rotor shaft. Further driving of machine, the drive is transferred to transport wheel through chain and sprocket mechanism. The complete specification of developed prototype was mentioned in the table:1. The developed prototype was fabricated and tested under different laboratory and field conditions. This gives its best per its soil pulverization, depth of furrow maker and width of cut. In Fig. 4.10 various components of the machine have shown.



(A) (B)

Fig 6: Isometric view of developed weeder cum furrow maker

In Table 1 different componets of the developed machine is mentioned wbich is shown in Fig. 6 (A).

Table 1: Components of the developed weeder cum furrow maker

S. No.	Components	S. No.	Components
1	Handle	11	Supporting frame for wheel shaft
2	Accelerator lever	12	Bearing block
3	Power switch	13	Flange
4	Gear changing lever	14	Small Sprocket
5	Hitch point of furrow maker	15	Blade shaft chassis
6	Furrow maker	16	Main shaft
7	Furrow maker frame	17	Gear reduction unit
8	Traction wheel	18	Engine
9	Wheel shaft	19	Fuel tank
10	Large sprocket	20	Silencer

Fig 7: Line diagram of the developed machine

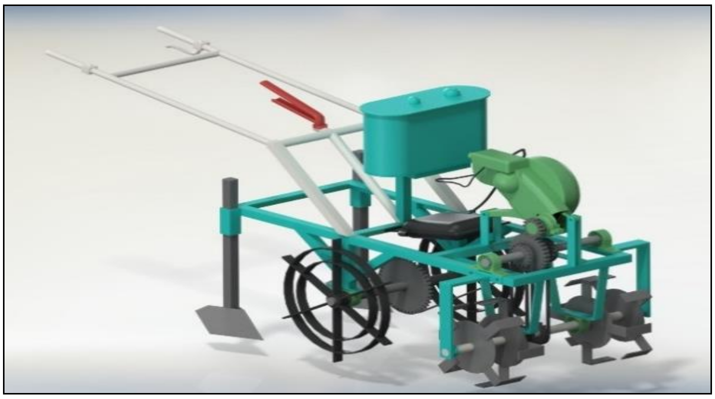


Fig 8: Final prototype of the machine

In Table 2 specification of the developed weeder cum furrow maker.

Table 2: Specification of developed weeder cum furrow maker

S. No.	Name	Particulars	Specifications
1	Main Frame	Length, mm	1140 (GI Angle)
		Width, mm	425 (GI Angle)
2	Material used frame	GI Angle, mm	38.1
		GI Square Box, mm^2	38.1×38.1
3	Wheel unit	Type	Lugs
		Width, mm	38.1
		Diameter, mm	406.4
4	**Power transmission unit**		
	Gear reduction unit	No. of teeth on driver	16
		No. of teeth on driven	38
	Sprockets	No. of teeth on driver	14
		No. of teeth on driven	38
		Pitch, mm	2
	Chain	Length for the blade, mm	900
		Length for wheel, mm	1789.2
		Material	Steel
5	Taper roller bearing	Inside diameter, mm	30
		Outside diameter, mm	62
		Width, mm	16
		Material	Cast iron
6	Flanges	No. of flanges	4
		No. of blades per flange	4
		Outer diameter, mm	200
		Inner diameter, mm	30
7	Blade	Length, mm	63.6
		Width, mm	3.81
		Thickness, mm	8
		Material of fabrication	Carbon steel
8	Furrow maker	Length of the frame, mm	590
		Width of share, mm	260
9	Total weight of weeder cum furrow maker, kg		175

Conclusions

The power operated weeder cum furrow maker for vegetable crops was designed and fabricated by assembling different parts such as engine, gear reduction unit, chain and sprocket mechanism, shaft (main shaft, blade shaft and wheel shaft), rotary cutting blades and transport wheel. The engine provide power to the machine in which gives power to the gear reduction unit and then to the chain and sprocket mechanism through that movement of blades and wheels takes place with the help of shafts containing in it.

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