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Design and development of mini tractor operated wavy disc type PTO powered tillage implement

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Abstract
The present investigation was carried out to study design and development of mini tractor operated Wavy disc PTO powered tillage implement in the year 2014-15.A PTO powered wavy disc type tillage implement with four numbers of wavy discs of 43 cm was designed and developed for tillage operation and tested in field condition to evaluate its performance. A wavy disc type tillage implement consisting of wavy discs, frame, depth control wheels and power transmission system was developed. The performance of the developed implement was assessed by varying three different parameters - gang angle (20°, 25°, 30°), forward speeds (0.35, 0.77, 1.19 m/s) and depth of operation (4, 8, 12 cm) and performance was observed in terms of mean mass soil diameter, draft, fuel consumption, wheel slip and field capacity. Performance analysis of the implement was carried out with Response Surface Methodology. Statistical significance of the terms was examined by analysis of variance (ANOVA) for each response using face centered central composite design. The average values of mean mass diameter, draft, fuel consumption, wheel slip and EFC obtained during the experiments were 12.28 mm, 106.12 KGF, 11.65 l/ha, 7.27 % and 0.178 ha/h, respectively. The overall best results for the developed tillage implement were obtained at 25° gang angle with forward speed of 0.77 m/s and depth of operation as 8 cm.

Keywords: design, development, mini tractor operated wavy disc PTO

1. Introduction
Tillage is considered to be one of the most energy consuming farm operation. Mechanization of agricultural operations by improved tools/implements is one of the ways to increase the productivity of the crop. It has been estimated that about 16-25 percent of the total energy available for rural sector is used for agricultural production (Singh, 1997), of which, Tillage is the most labour consuming and difficult operation, compared to all subsequent agricultural operation for crop production in the field about 20 percent energy is consumed only in seedbed preparation (Anonymous, 1984). The conservation tillage system involves a reduction in the number of passes over the field with tillage implements including the elimination of ploughing (Kumar *et al.,* 2012).
Further, many studies (Hoki *et al.,* 1988) [5] reported major reduction in a draft requirement (up to 10%), specific draft requirement and sometimes penetration resistance also, when disc implement was changed from freely rotating to the driven type. Perdok and Kouwenhoven (1994) mentioned that powered disc rotary harrow seems to offer the best potential for many tillage tasks to be carried out on the farm all year round.
The disc blades with wavy edges rotate better and brake soil under more energetic action. A desired effect of the waves in the blades is that the discs get a gearing coupling with the soil in order to have a better rotation than the discs with cutting edges having no waves.Considering the merits of wavy edges, a wavy disc type coulter was selected for the study.
After a limited research work and considering the merits, the work on the design and development of a wavy disc type PTO powered tillage implement was resumed in the department of Farm machinery and Power Engineering, College of Agricultural Engineering and Technology, An and Agricutural University, Godhra.

2. Material and Method
2.1 Design considerations
2.1.1 Disc Diameter
Disc diameter should be minimum permissible as with an increase in diameter the load necessary for penetration sharply increases. The disc diameter (D) is related to the tilling depth

(Td) by the following equation:

$$D = K_d \times T_d \qquad (1)$$

Where, K_d value for disc ranges from 3-5

Since, the total power requirement should not exceed the power available from the tractor selected. In the present study, as the implement is to be operated by mini tractor, thus, the tilling depth may vary from 10-15 cm.Here, a maximum tilling depth of 14 cm and value of K_d was taken as 3. Thus, nearest size available in the market of discs 43cm were selected.

2.1.2 Disc Spacing

Disc spacing should be adequate to prevent clogging and also produce a reasonable furrow profile. For disc tiller ridge height is given by:

$$C = \frac{T_d}{2} \qquad (2)$$

Where, T_d = Tilling depth (assume 14 cm)

Therefore, with the help of Eq.2,

$$C = \frac{14}{2} = 7 \text{ cm}$$

$$\text{Disc spacing, } S = 2\sqrt{C\,(D - C)\tan\alpha} \qquad (3)$$

Where,

C = 7 cm

D =dia. of disc, 43 cm

α= angle that total soil resistance made with the line of motion in horizontal plane and varies between 15° to 20° for all soil conditions (Bernacki*et al*., 1972) adopted 15° for design.

Therefore, from Eqn. 3,

S = 16.44 ≈ 16.5 cm

Therefore disc spacing of 16.5 cm was taken. The desired flange made of cast iron was used to fix the discs on the shaft.

2.1.3 Design of disc shafts

As the shaft is rotating element so it is subjected to torsion and due to vertical soil support and the weight of implement it is also subjected to bending

Length of shaft = (Number of disc on shaft - 1) × Disc Spacing (4)

= (5-1) × 16.5

(Number of disc were taken as 5 considering the width of implement as 82.5 cm with spacing of 16.5 cm so that available tractor power may operate the implement)

= 66 cm

However, for fitting nuts and sprocket additional clearance of 16.5 cm was taken. Thus, the total length of the shaft was equal to 66 + 16.5 = 82.5 cm

The tillage implement was assumed to be under the two force systems. In the vertical plane, the upward force on the disc during tillage and the weight of the disc gang assembly acting on the shaft. In the horizontal plane, soil resistance on the discs acting on the shaft.

Surface area of one disc under soil was calculated by using equation of Pythagoras theorem (fig 1)

$$r^2 = (r-d)^2 + x^2 \qquad (5)$$

Where,

r = radius of disc, 21.5 cm

d = depth of cut, assume 14 cm

x = BC

With the help of Eq. 5,

x = 20.14 cm = BC

AC = 2 × 20.14 = 40.28 cm

Now,

$$\text{Area of triangle AOC} = \frac{1}{2} \times 7.5 \times 40.28 = 151.05 \text{ cm}^2$$

$$\text{Area of OADC} = \frac{\pi r^2 \times \theta}{360} \qquad (6)$$

$$\sin^{-1}\emptyset = \frac{BC}{OC} = \frac{20.14}{21.5} = 0.93$$

Therefore, ∅ = 69.51°

Therefore, θ = 2 × 69.51° = 139.02°

With the help of Eq.6,

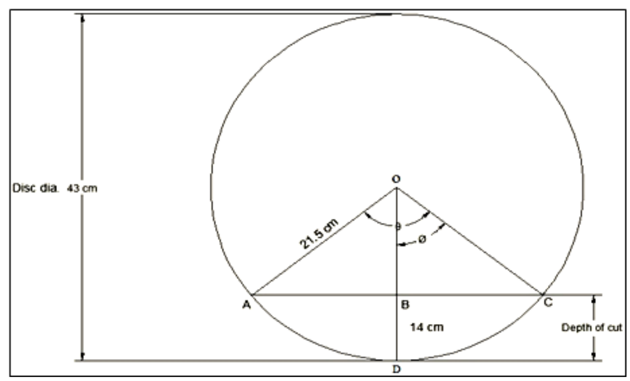


Fig 1: Schematic of surface area of disc under soil

Area of triangle OADC = 560.5 cm^2

Therefore,

Area of ADC = area of OADC – area of triangle OAC

= 409.45 cm^2

Soil force = unit draft of soil x surface area of implement in contact with soil

Soil force = (0.6 × 409.45) kg (Considering soil as medium soil, the value of unit draft is taken as 0.6)

= (245.67 × 9.81) = 2410.02N

Now

Horizontal force on each disc, P_x = 2410.02 N

Considering $\frac{P_x}{P_y} = 1$,

Vertical force (P_y) on each disc was taken as 2410.02 N

Weight of the disc gang assembly (W = 100 kg),

W = 981 N

By simple force analysis,

Reactions at both bearings (R_a and R_b) were found to be equal to,

$$= \frac{5 \times 2410.02\ N - 981\ N}{2} = 5534.5\ N$$

The free-body diagrams of disc gang shaft with different forces and moment arms are shown in Fig. 2
Moment due to vertical force:
Maximum bending moment,
M = (2410.02 × 165 + 2410.02 × 330 + 2410.02 × 495 + 2410.02 × 660) – (5534.5× 412.5)
= 1693551.75 N mm

Moment due to horizontal force:
Torque applied,
T = horizontal force × (radius – $\frac{depth}{3}$) × number of disc
= 2028393.33 N mm

Assuming the disc shaft had to experience heavy shock loads, the load and fatigue factors were considered as:

Load factor, $K_m = 3$ and fatigue factor, $K_t = 2.5$

Ultimate bending stress for 20 Mn Cr5 steel,
$\sigma_{b\ (max)} = 350\ N/mm^2$

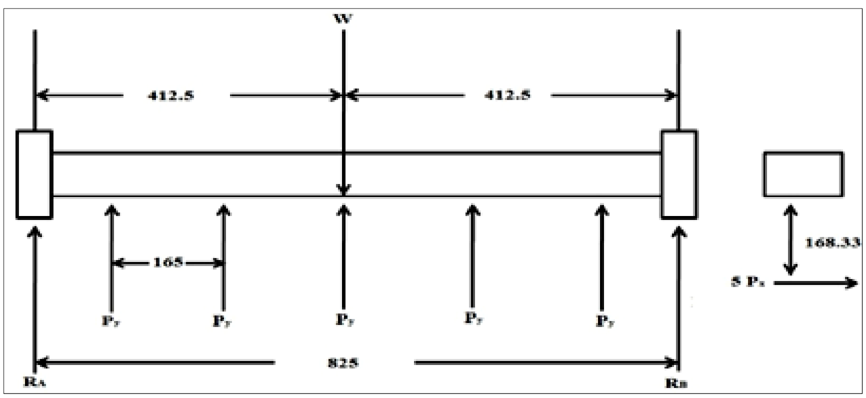
According to maximum normal stress theory, the maximum normal stress in the shaft

$$\frac{\pi}{16} \times \sigma_{b\ (max)} \times d^3 = [(K_m\ M) + \sqrt{(K_m\ M)^2 + (K_t\ T)^2}] \quad (7)$$

Where,
$\sigma_{b\ (max)}$= maximum bending stress, N/mm^2
d = diameter of shaft, mm
M =maximum bending moment, N mm
T = maximum torsional moment, N mm
K_m= combined shock and fatigue factor for bending
K_t= combined shock and fatigue factor for torsion
From Eq. (7) we got,

d = 56.30 mm
As per nearest size available in market diameter of shaft was selected size of 65 mm.

Therefore, a hollow circular section shaft of 65 mm dia was chosen,as this was the standard size for mounting the slot of wavy discs.



(a) Vertical forces Note: all dimensions are in mm, (b) Horizontal force
Fig 2: Moment arms for both vertical and horizontal forces acting on the disc gang shaft.

2.1.4 Power Transmission System
To transfer the power from PTO to disc shaft, suitable power transmission components for mini tractor were selected as per availability in the market. The power transmission system of vertical tillage implement is shown in Fig.3.

2.1.4.1 Gear Box
The gear box made of SG Iron grade material with available size of pinion gear having 9 teeth with dia of 46 mm and also crown gear having 15 teeth with dia of 125 mm were selected. The angle between driving shaft and drive shaft was selected as 65° to transmit the power from PTO to disc shaft. The driven shaft of 20 Mn Cr5 steel of 30 mm dia was used to transmit the power from gear box to disc shaft.

2.1.4.2 Bearings
Bearings on agricultural implements are often required to operate under extremely dusty or dirty conditions. The bearings of the disc which are mounted on a shaft and the bearings which mounted on a gear box are subjected to fairly heavy and radial thrust loads at low speeds and the disc shaft bearings operate in contact with dirt in most of the times. Therefore, the ball bearings were selected for disc shaft and tapper roller bearing was selected for the gear box shaft.

2.1.4.3 Chain and sprocket
A readily available heavy chain having pitch of 31.75 mm, roller dia of 19.05 mm and length of 1143 mm made of carbon steel was selected. Two sprockets made of 20 Mn Cr5 grade materials for transmitting the power from gear box shaft to disc shaft were selected. One sprocket of 11 teeth having dia 125 mm, thickness of 18.5 mm and pitch of 35.7 mm was mounted on gear box shaft and other 16 teeth sprocket having dia 175 mm, thickness of 18.5 mm and pitch of 34.34 mm was mounted on disc shaft.

2.1.4.4 Universal Joint
A readily available universal joint of maximum length of 630 mm and minimum length of 490 mm was selected for transmitting power from tractor PTO to pinion gear.

2.1.5 Design of Frame
Magnitude of various forces acting on disc blade were determined as under for designing the frame as shown in Fig. 4
The width of cut of disc (W) was determined as shown in Fig. 4.
$W = D_c \sin\theta$ (8)

Where,
W = width of cut, cm
D_c = effective disc dia. at maximum depth of cut, 14 cm
θ = gang angle of vertical discs, 25^0

Referring to Fig. 3.4,

$D_c = 2$ AC

By geometry,

$$AC = \sqrt{AB^2 - BC^2}$$
= 20.14 cm
$D_c = 2 \times 20.14 = 40.28$ cm

Therefore,

W = 40.28 Sin 25°
=17.4 cm

Theoretical draft of the implement was determined as,
Theoretical draft = n × W × d × s (9)

Where,

n = number of discs, 5
W = width of cut, 17.4 cm
d = depth of cut, 14 cm

s = specific soil resistance for medium soil, 0.6 kg/cm^2

Substituting the values in Eq. 9,
Theoretical draft = 730.8 KGF (7169.1 N)

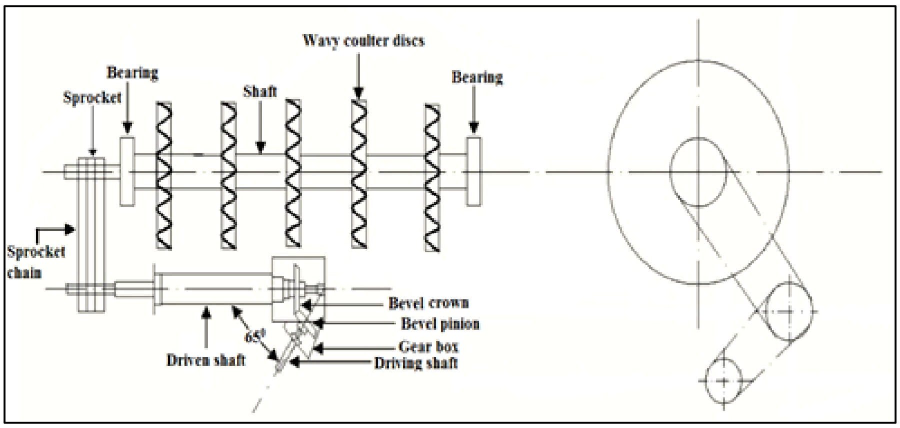


Fig 3: Power transmission system of vertical tillage implement

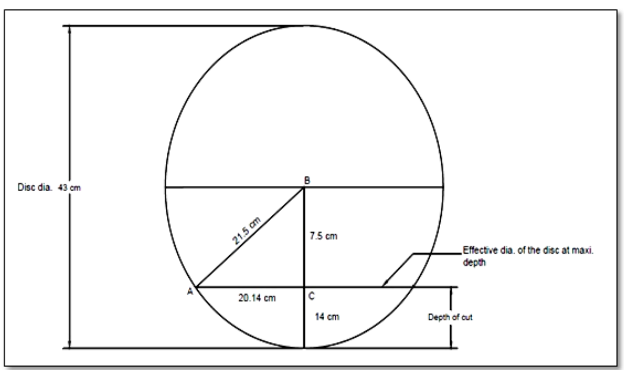


Fig 4: Determination of cutting width of disc by geometrical method

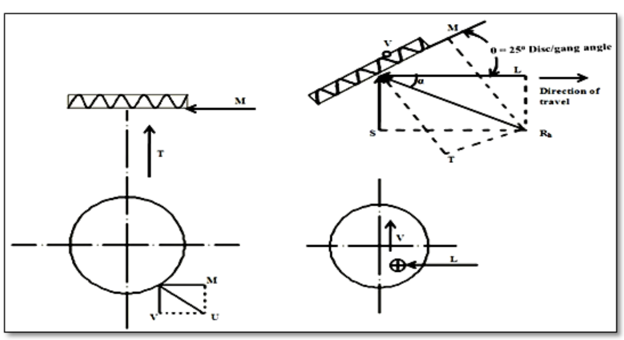


Fig 5: Force acting on wavy coulter disc

Referring to Fig. 5,
Calculating the force R_h (resultant of L and S) was found as,

$$R_h = \frac{L}{\cos \alpha} \quad (10)$$
= 7708.7 N

Where,
L = horizontal component which lies in the direction of travel, i.e. draft of the implement.
α = angle that total soil resistance made with line of motion in horizontal plane and varies between 15° to 20° for all soil condition (Bernacki*et al*., 1972), adopted 20° for design.
The side force (S) acting perpendicular to draft (L) was obtained as,

$$S = R_h \sin \alpha \quad (11)$$
= 2636.5 N

The radial force (M) was calculated as,
$M = R_h \cos(\alpha + \theta)$ (12)
= 5450.8 N
The thrust force (T) acting parallel to axis plough gang bolt was obtained as,

$$T = R_h \sin(\alpha + \theta) \quad (13)$$
= 5450.4 N

The resultant of all useful forces (R) acting on the disc was,

$$R = \frac{R_h}{\cos \theta_0} = 8029.8 \text{ N} \quad (14)$$

Where,
θ_0 = angle that makes with horizontal in vertical plane and varies between 15° to 20° (Bernacki*et al*., 1972).

Adopting 15°for design, the vertical force (V) acting on disc blade was,

$$V = R \sin \theta_0 = 2078.2 N \quad (15)$$

The resultant force of M and V i.e. radial force (U) acting on the disc was obtained as,

$$U = \sqrt{M^2+V^2} = 5833.5 \text{ N} \quad (16)$$

The frame consisted of four members AC, BD, AB and CD (shown in Fig. 6). The members BD, AC are forebar and rearbar of the frame respectively, whereas members AB, CD are the two cross bars (side bars). Member EF is the arbour bolt which is connected to the cross bars through M.S. angle iron with nuts and bolts.

Forces acting on member EF

To take into account the possible impact due to sudden pull by the tractor an impact factor of 1.0 for design purpose as considered, and therefore, the thrust force acting on 5 discs was found as,

T = 5450.4× 2 = 10900.8 N

Bending moment,

$$B.M. = \frac{\text{thrust force} \times \text{spacing between the discs}}{\text{number of disc}} = 359726.4 \text{ N-mm}$$

Sum of horizontal reactions i.e. ΣH = 0 at A or B = 10900.8 N

Vertical reaction at A' or B' (taking moment about A' or B'), i.e. ΣM = 0

$$= \frac{5 \times BM}{\text{span}} = 1284.7 \text{ N}$$

Resultant of horizontal and vertical reactions (F) was obtained as,

F = 10976.2 N

The magnitude of forces M and V after considering impact factor of 1.0 would be,

M = 10901.6 N

Similarly, for vertical force (V) would be,

V = 4156.4 N

The resultant P of these two forces would be,

$$P = \sqrt{M^2+V^2} = 11667 \text{ N}$$

Hence, reaction at A and B = 2P = 23334 N

Combined resultant force (C_{rf}) due to F and P at the centre of member AB would be,

$$C_{rf} = \sqrt{F^2+2P^2} = 25786.6 \text{ N}$$

Design of member AB or CD (cross bar)

The bending moment induced in the member due to C_{rf} was found as,

$$BM = \frac{C_{rf} \times \text{span}}{5} = 2475513.6 \text{ N-mm}$$

Sectional modulus (Z) for member AB or CD would be,

$$Z = \frac{M}{F_b} = \frac{2475513.6}{161.8} = 15300 \text{ mm}^3$$

Where, F_b = permissible bending stress considering 161.8N/mm^2 for mild steel.

Therefore, a M.S. flat section of 125 mm × 8 mm was selected with Z_{yy} as 22800 mm^{3}as per availability in the market.

Design for member AC or BD (forebar)

Total tension in the member AC or BD = reactions' at A or B

$$= \frac{C_{rf}}{2} = 12893.3 \text{ N}$$

Therefore, the area ofthe member to bechosen would be,

$$= \frac{\text{reaction}}{F_t} = 87.6 \text{ mm}^2$$

Where, F_t = permissible tensile stress, as 147.15N/mm^2 for mild steel.

Also, the member BD (forebar) would experience a pull force (draft) exerted by the tractor. Let the maximum pull or draft be 730.8 KGF (as calculated earlier).

$$BM = \frac{\text{Pull} \times \text{span}}{5} = 2007348 \text{ N-mm}$$

Sectional modulus (Z) for member AC or BD would be,

$$Z = \frac{M}{F_b} = \frac{2007348}{161.8} = 12406.3 \text{ mm}^3$$

Where, F_b = permissible bending stress considering 161.8 N/mm^2 for mild steel.

Therefore, an M.S. square pipe of 50 mm × 50 mm × 5 mm thickness was chosen as per the availability in the market was selected for fabrication.

Based on the design criterion discussed above, a mini tractor operated disc type vertical tillage implement was fabricated.

All the parameters of the tractor-implement performance were measured and recorded in line with the recommendations of RNAM (Regional Network for Agricultural Machinery) test codes and procedures for farm machinery technical series (1983).Mahindra Yuvraj tractor of 15 hp was used for conducting the field experiments.

2.1.6 Design of the experiments

Response surface methodology (RSM) was employed for experiment and the main advantage of RSM is to reduce number of experimental runs needed to provide sufficient information for statistically acceptable results.The independent variables selected for the experiments were Gang angle, *x1,* 20°, 25° and 30°, Forward speed, *x2,* 0.35, 0.77 and 1.19 m/s,Depth, *x3,* 4, 8, and 12cm.Face centered central composite design in coded and un-coded levels of three variables and three levels was employed for experiments of vertical tillage implement and for analysis of data, A complete second order quadratic model employed to correlate the independent process variables. Experimental data were fitted to the selected models and regression coefficients obtained. Statistical significance of the terms in the regression equation was examined by analysis of variance (ANOVA) for each response.

2.1.7 Experiment details
The experiments were conducted on 20 plots of net size 1m x 30m.The effect of various parameters like gang angle, gang speed, depth and speed of operations were studied on various soil parameters like soil texture, soil moisture content, bulk density, soil pulverization and machine parameters like field capacity, field efficiency, slippage, fuel consumption, draft and economics of operation etc.

2.1.8 Performance Evaluation
The performance of developed vertical tillage implement was evaluated in field having sandy loam soil with average moisture content of 15.10 %. The soil moisture revealed that field had moisture in friable range to conduct the field experiments.

3. Result and Discussions
The average bulk density was 1.71 g/cc before tillage operation. Similarly, the average bulk density after operation of developed implement was 1.31 g/cc. The developed vertical tillage implement was evaluated in the field by varying the forward speed, gang angle and depth of operation. And the effect was observed on operating speed, wheel slip, draft, fuel consumption, field capacity, soil pulverization as well as soil properties. The experiments were conducted as per the design using response surface methodology.

Performance was assessed in terms of draft, wheel slip, fuel consumption, effective field capacity and MMD. To conduct the experiments face centered central composite design was adopted with three factor three variables. The data obtained as shown in table 1 from the 20 runs of experiments were analyzed using Face Centered Central Composite Design. The optimization with different independent (Factors) and dependent parameters (responses) was carried out using manual regression quadratic model with the help of Design Expert 8.0 software.

The optimized values for the best performance of developed implement was assessed at different working depth and it was found out that as at different depth of operation ranging from 4 to 14 cm, the optimized value of gang angle was found between 21.55° to 22.71°as shown in table 4.1. Also, values of different responses under study at optimized values at different depths are given in the above table.

The optimized values for the best performance of developed implement was assessed at different working depth and it was found out that as at different depth of operation ranging from 4 to 14 cm, the optimized value of gang angle was found between 21.55° to 22.71°as shown in table 2.

Since, in the developed machine, it was very difficult to vary the gang angle, therefore, to found out the best results among different gang angles under study, gang wise optimum values were calculated and given in table 3. It is clear from the data that at 25° gang angle will give superior desirability as compared to 20° and 30° gang angle at forward speed of 0.77 and depth of 8 cm.

Table 1: Analyzed data for developed vertical tillage implement using face centered central composite design

Run	Independent variables			Dependent variables				
	Factor-1 X_1 Gang angle (degree)	Factor-2 X_2 Forward speed (m/s)	Factor-3 X_3 Depth (cm)	Response-1 Y_1 Draft (KGF)	Response-2 Y_2 Fuel consumption (l/ha)	Response-3 Y_3 Wheel slip (%)	Response-4 Y_4 EFC (ha/h)	Response-5 Y_5 MMD (mm)
1	30	1.19	4	112.68	10.00	6.99	0.26	11.50
2	30	0.77	8	112.08	14.37	7.42	0.17	12.52
3	25	0.77	8	102.60	10.00	7.15	0.18	12.04
4	25	0.77	8	105.36	10.31	7.03	0.18	12.07
5	25	0.77	8	102.76	10.00	6.92	0.17	11.98
6	25	0.77	4	95.76	12.50	7.10	0.17	11.09
7	25	0.77	12	112.00	10.00	6.86	0.17	12.49
8	20	1.19	12	116.68	8.12	6.72	0.26	12.49
9	30	1.19	12	118.16	10.00	6.74	0.26	13.08
10	25	1.19	8	116.56	8.75	7.14	0.26	11.49
11	20	0.35	4	91.60	12.50	7.66	0.09	12.50
12	20	0.35	12	100.76	18.75	7.77	0.08	14.08
13	25	0.77	8	104.16	10.25	6.91	0.18	12.03
14	30	0.35	4	107.28	12.50	7.67	0.08	12.51
15	25	0.77	8	102.72	10.00	6.90	0.18	12.00
16	25	0.77	8	101.12	10.50	6.99	0.18	12.01
17	25	0.35	8	94.24	18.75	7.13	0.08	12.47
18	30	0.35	12	114.60	18.75	7.15	0.09	14.51
19	20	1.19	4	110.56	7.50	7.19	0.26	10.80
20	20	0.77	8	100.64	9.37	9.92	0.18	11.77

Table 2: Depth wise optimized data of the developed implement at different depth

Depth (cm)	Gang angle (degree)	Forward speed (m/s)	Fuel Consumption (l/ha)	Draft (KGF)	Wheel slip (%)	EFC (ha/h)	MMD (mm)	Desirability
4	21.55	1.19	8.65	107.35	7.25	0.26	10.74	0.69
5	21.66	1.19	8.56	108.34	7.50	0.26	10.94	0.68
6	21.77	1.19	8.48	109.33	7.66	0.26	11.14	0.68
7	21.91	1.19	8.40	110.31	7.72	0.26	11.33	0.67

8	22.03	1.19	8.32	111.31	7.70	0.27	11.53	0.66
9	22.15	1.19	8.23	112.31	7.58	0.27	11.73	0.65
10	22.25	1.19	8.15	113.33	7.38	0.27	11.93	0.64
11	22.36	1.19	8.06	114.36	7.09	0.27	12.13	0.63
12	22.49	1.19	7.98	115.37	6.70	0.27	12.33	0.61
13	22.60	1.19	7.90	116.40	6.22	0.27	12.54	0.60
14	22.71	1.19	7.81	117.44	5.65	0.27	12.74	0.59

Table 3: Overall best results at optimized value for the developed implement

Gang angle (degree)	Forward speed (m/s)	Depth (cm)	Fuel consumption (l/ha)	Draft (KGF)	Wheel slip (%)	EFC (ha/h)	MMD (mm)	Desirability
20	0.77	8.28	10.09	102.16	8.65	0.18	12.13	0.71
25	0.77	8	9.49	106.84	6.99	0.18	11.64	0.78
30	0.77	4	10.43	107.24	7.61	0.17	11.77	0.61

4. Conclusions

Mean mass diameter of the vertical tillage implement ranged from 10.80 mm to 14.51 mm with an average value of 12.28 mm. Draft of the vertical tillage implement ranged from 91.60 KGF to 118.16 KGF with an average value of 106.12 KGF. It increased with increase in gang angle, forward speed and depth of operation.Fuel consumption increased with the increase in gang angle, forward speed and depth of operation with a maximum and minimum value of 18.75 l/ha and 7.50 l/ha, respectively.

Wheel slip of the vertical tillage implement ranged from 6.72 % to 9.93 % with an average value of 7.27 %.Effective field capacity of the vertical tillage implement ranged from 0.089 ha/h to 0.268 ha/h with an average value of 0.178 ha/h.

The optimization with regards to working depth 4 to 14 cm for developed implement showed that gang angle between 21.55° to 22.71° is required for the optimized performance of the developed implement. Therefore, a common value of gang angle 22.50° may be chosen considering the constraints in fabrication and operation for the development of the implement and assuming implement to be operated about 10 cm depth at different depth.

While overall best results for the developed tillage implement were obtained at 25° gang angle with forward speed of 0.77 m/s and depth of operation as 8cm.Overall, the performance of developed machine was satisfactory for tillage operation.

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