



# **A Theoretical Method for Efficient Design of Power Tiller Rotavator Satisfying Multiple Objectives**

**Jyotirmay Mahapatra<sup>1\*</sup>, Vinita Kashyap<sup>2</sup> and Ajay Kumar Sharma<sup>3</sup>**

<sup>1</sup>Central Institute of Agricultural Engineering, Bhopal, 462038, Madhya Pradesh, India.

<sup>2</sup>Krishi Vigyan Kendra, Begusarai, Dr. Rajendra Prasad Central Agricultural University, Pusa, Samastipur, 848125, Bihar, India.

<sup>3</sup>College of Technology and Engineering, Udaipur-313001, Rajasthan, India.

## **Authors' contributions**

*This work was carried out in collaboration among all authors. Author JM designed the study, performed the analysis, wrote the protocol and wrote the first draft of the manuscript. Author VK managed the literature searches. Author AKS guided the research work. All authors read and approved the final manuscript.*

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## **ABSTRACT**

Several advantages of rotavator including negative draft generation make it the most suitable attachment to power tiller. Many factors affect the economy and performance of power tiller which includes specific energy consumption, resourceful use of available power, produced soil conditions and blade durability. Hence a systematic design method was developed for power tiller rotavator while considering all these factors simultaneously. Specific work done by rotavator was equalized with performable work of power tiller for different combination of design and kinematics parameter. The value of design and kinematic parameters included in the previous step were limited by the extremities. These limitations were imposed so as to get desired soil conditions and blade durability. The best combination of parameters that produces the desired soil condition with minimum specific energy consumption while utilizing the available power resourcefully was selected. Other design parameters were calculated using parameters from the selected combination. Rotavator design and kinematic parameters satisfying the above objectives were calculated and found to be velocity ratio of 5.12, forward velocity of 0.41 m/s, rpm of 80.19, operating width of 7.5dm and 30 number of

\*Corresponding author: E-mail: [jyotirmayfmpe@gmail.com](mailto:jyotirmayfmpe@gmail.com);

blades mounted in 5 number of flanges. This method of optimisation can be adopted for improved power tiller rotavator design and efficient operation. But as this method is purely theoretical; practical verification of the design is essential before adopting for large scale industrial production.

*Keywords: Rotavator; blade durability; soil conditions; specific work; design parameters; kinematic parameters.*

## 1. INTRODUCTION

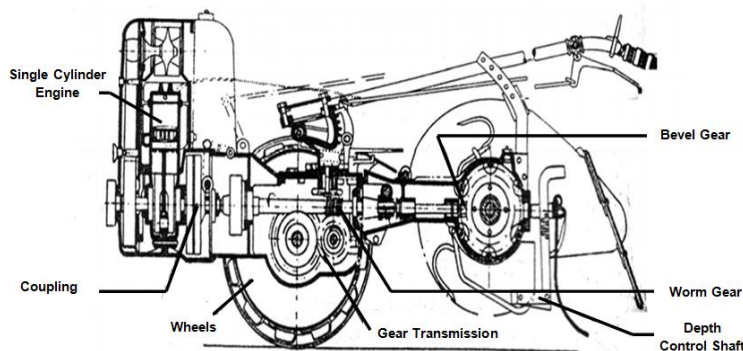
Rotavator is a special tillage implement which uses rotational power to till the soil. An excellent seedbed can be produced by using rotavator in minimum operation time. Many superior qualities of rotavator over other tillage implements have made it world famous. By taking advantage of rotavator, the primary and secondary tillage applications can be combined in one stage [1]. As far as the quality of the seedbed is concerned, one pass of the rotavator is equivalent to several conventional tillage operations [2,3,4]. Therefore, the agricultural soil can be prepared with only one pass of this type of tillage implements on the land.

Tillage is the most important operation in farming which ultimately affects productivity and rotavator is intensively used for the purpose. As it generates negative draft, It is the most suitable attachment to small power source like power tiller. Power tiller plays an important role in agricultural operation in small farms and hilly areas. It has other benefits of better manoeuvrability and higher weeding efficiency. Therefore it is essential to optimise the power tiller rotavator design and kinematic parameters so as to achieve the desired soil tilth while minimising losses.

Many researcher optimised rotavator for different results like minimum energy consumption, best seedbed preparation, maximum utilisation of

available power, increasing the durability of blades etc. Sakai [5] from his performance testing resulted that L shaped blade were better for trashy land operation and had better weeding efficiency. The “L”- shaped are overall the best when operated at lower tip speeds and intermediate increments of cut for the best utilization of power, cutting and dispersing surface organic matter and pulverizing the soil. Baloch et al. [6] compared the L-shaped and new slasher blades in terms of work rate, fuel consumption and specific energy consumption and observed that the L-shaped blades performed better in both soft and hard soils at low and high rotor speeds. Adams and Furlong [7] studied the performance of the “pick”, the “L”, and the “C”- shaped blades and suggested that considering the usage such as crop row width, available horse power etc., the rotor assembly should be widened for the best utilization of power.

Hedrick and Gill [8] and Gill and Hedrick [9] reported that depth of tillage is an important factor in the dissemblance, the mixing of soil and cutting of roots. The depth of operation had considerable effect on the power requirement and performance of rotary tillers. They suggested that increasing the depth increases total power requirement but decreases the specific power requirements in general. In deeper tillage more blades pass through the area occupied by a previously cut soil segment, and the probability of soil being re-circulated is increased.



**Fig. 1. Meyenberg's rotary cultivator [26]**

Soehne [10] studied effect of some of the tool parameters on specific energy consumption by dividing the average torque by volume of disturbed soil. He examined the above in different cutting conditions like shearing and cutting and concluded that minimum cutting width should be 75mm, the minimum radius of curvature should be 30mm and the clearance angle should be maintained at about 20 degree in order to get low specific energy consumption. Similar results were obtained from experiments carried out by Mashchenski [11]. The specific energy consumption reduces by increasing cutting width [7,12-14].

Hendrick [15] concluded that decreasing the velocity ratio ( $\lambda$ ) by increasing forward speed resulted in an increase in the power requirement, but a reduction in specific power. Decreasing the velocity ratio  $\lambda$  by decreasing rotor velocity decreased the power requirement and specific power. Similar results were also obtained by Grinchuk [16], Walton and Warboys [17], Niyamapa et al. [18], Asl and Singh [19], Pal et al. [20].

On the other hand decreasing velocity ratio increases length of bite and irregularities of bottom layer [21,22]. These are not desired in a seedbed. In analysis conducted by Celik A and

Altikat S [22] the ridge height was observed to be decreasing from 160mm to 20mm as it reaches velocity ratio of 5 and was observed very less at a bite length of 100mm. Bite length of 100 mm was found to be optimum for power and cultural optimization [23]. Pulverizing effect of soil is maximum for velocity ratio in the range of 4 to 10 [24,25]. Again lower velocity ratio; achieved by decreasing the peripheral velocity increases the magnitude of maximum tangential force on the blade [26]. It ultimately affects the durability of blade.

At deep tillage (cutting depth more than rotor radius), reverse rotation reduces energy by 20 to 30% [27]. Whereas at shallow tilling depths (cutting depth less than rotor radius), forward rotation required 10 to 15% less energy than reverse rotation. Dalin and Pavlov [28], Furlong [29], Bok [30], Blaauw [31] also pointed out the advantages of forward rotation over reverse rotation.

Study on durability of blade by FEA analysis of equivalent von mises stress and total deformation were done [32-34] which shows that with increase in blade cutting width both the parameters (equivalent von mises stress and total deformation) increases.

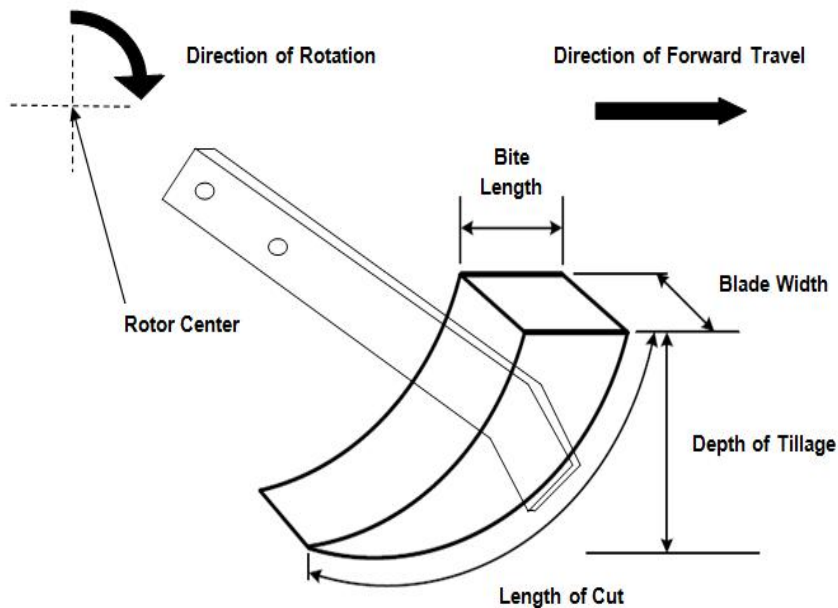


Fig. 2. The soil-tool interaction for a forward rotating rotavator

## 2. METHODOLOGY

### 2.1 Specific Work Method

This is an efficient method applied for optimum use of generated power of power tiller to carry out the maximum volume of soil tillage [35,36]. For this optimum utilization of power the depth of operation, width of operation of power tiller, bite length, velocity ratio, forward speed and RPM can be decided. Specific work method equalizes the specific work and the performable work of the power tiller which can be proceeded in the following way.

The specific work of power tiller( $A$ ) can be calculated by the Equation (1) [26]:

$$A=A_0+A_B \text{ (kg.m/dm}^3\text{)} \quad (1)$$

Where,  $A_0$  and  $A_B$  are the static specific work and dynamic specific work of power tiller rotavator(kg-m/dm<sup>3</sup>), respectively, which can be calculated by the Equation (2) and (3) [26]:

$$A_0 = 0.1C_0K_0 \text{ (kg.m/dm}^3\text{)} \text{ and} \quad (2)$$

$$A_B = 0.001a_u v^2 \text{ (kg.m/dm}^3\text{)} \quad (3)$$

Where,  $C_0$  = Coefficient relative to the soil type;  $K_0$  = Specific strength of soil(kg/dm<sup>3</sup>);  $v$  = forward speed(m/s);  $a_u, a_v$  = dynamical coefficients and  $a_v = a_u \lambda^2 \text{ (kg.s}^2\text{/m}^4\text{)}$

$$\text{Where, } \lambda = u/v \text{ and} \quad (4)$$

$u$  = peripheral velocity(m/s).

The performable work of the power tiller( $A_c$ ) can be calculated by the Equation (5) [26]:

$$A_c = \frac{75N_c \eta_c \eta_z}{vab} \text{ (kg.m/dm}^3\text{)} \quad (5)$$

Where,  $N_c$  = power of power tiller (hp);  $v$  = forward speed (m/s);  $\eta_c$  = power tiller efficiency amounting to about 0.9 for machines with concurrent revolutions;  $\eta_z$  = coefficient of

reservation of power tiller power amounting to 0.7 to 0.8;  $a$  = tiller work depth (dm) and  $b$  = tiller work width (dm).

The soil condition considered for design is very heavy soil. For very heavy soil coefficient relative to the soil type ( $C_0$ ) = 2.5 kg/dm<sup>3</sup>, Specific strength of soil ( $K_0$ ) = 50 kg/dm<sup>2</sup>, dynamical coefficient ( $a_u$ ) = 400 kg.s<sup>2</sup>/m<sup>4</sup> [26]. Since value of  $b, v, \lambda, a_v$  are interdependent a domain is defined for the values at first, with respect to the technical specification of the selected power tiller. The optimum condition for the rotavator design was selected from the domain.

Here the cutting width of blade was considered 7.5 cm, as decreasing below this value increases the specific energy consumption at a higher rate [10, 11] and increasing working width increases value of equivalent stress and total deformation [34]. The distance between flanges of rotavator in this design was considered 15 cm. Therefore working width included in the domain are 60, 75, 90 cm. The forward speed of the selected power tiller in different transmission gears are presented in Table 1 [37].

In this design only the first two heavy transmission gears were considered for building the domain, as at high level of power tiller forward speeds the penetration ability of the rotavator blade reduces. While following specific work method for optimisation prime importance was given to obtain the desired soil condition and hence the velocity ratio considered for the design is from 4 to 10 [38,39,25].

Firstly, the value of rotavator specific work which is closest to the performable work of the power tiller was determined for each of the selected working widths at each of the forward speeds. Then the corresponding values of velocity ratio for each forward speed were determined to calculate the rotor speed and bite length using Equation (6) and (7).

**Table 1. The speed of the selected power tiller in different transmission gear**

Transmission gear	Forward speed (m/s)
1-Forward-Heavy	0.41
2-Forward-Heavy	0.60
3-Forward-Heavy	1.16
1-Forward-low	1.46
2-Forward-low	2.14
3-Forward-low	3.38
1-Reverse	0.35
2-Reverse	1.26

$$n = \frac{6000\lambda v}{2\pi R} \quad (6)$$

$$L = \frac{2\pi R}{\lambda Z} \quad (7)$$

Where,  $n$  = RPM of the rotor,  $\lambda$  = Velocity ratio,  $R$  = Rotor radius(cm),  $L$  = Bite length(cm),  $v$  = Forward speed(m/s),  $Z_e$  = Number of blades on each sides of the rotor flange,  $u$  = Peripheral speed of rotor(m/s).

In this design, three blades were considered on each side of the flanges ( $Z_e = 3$ ). As Soil profile density reduces with increase of blades per flange and power consumption do not increases significantly by increasing the number of blades per flange from 4 to 6 [19]. The working depth selected was 15 cm which satisfies the requirement of field operations by power tiller and is commonly observed [40]. The conventional diameter for rotary tillers rotor is variable from 30 to 50 cm. Moreover, the radius of rotor for rotary tillers should be selected greater than the working depth [27]. Considering these explanations, a 50 cm diameter was selected for the purpose.

To have maximum utilization of available power (minimum difference between performable work and specific work), a suitable combination of maximum working width and forward speed (relates to maximum field capacity) was selected.

The value of velocity ratio at which the difference between the specific work required by rotavator and performable work of power tiller becomes zero was calculated. At this velocity ratio for the selected working width and selected forward velocity the RPM, bite length, peripheral velocity were calculated using Equation (6) and (7).

## 2.2 Design of Power Transmission System

The desired rpm for operation ( $n_d$ ) is calculated in the above process and the rpm supplied from the source ( $n_a$ ) is known to us. So coefficient of power transmission ( $i$ ) could be obtained by the Equation (8).

$$i = \frac{n_d}{n_a} \quad (8)$$

## 2.3 Power Tiller Rotavator Shaft Design

For designing the rotor shaft, the maximum peripheral force which can be endured by the

rotor is considered. Maximum peripheral force acting on the rotor assembly is calculated by Equation (9):

$$K_s = \frac{C_s 75 N_c \eta_c \eta_z}{u_{min}} \quad (9)$$

Where,  $K_s$ = maximum peripheral force on rotor assembly, kg;  $C_s$ = reliability factor that is equal to 1.5 for non-rocky soils and 2 for rocky soils;  $N_c$ = engine power of the power tiller, hp;  $u_{min}$ = minimum peripheral speed of the rotavator blade, m/s(obtained from SWM);  $\eta_c$  = power tiller efficiency mounting to about 0.9 for machines with concurrent revolutions and  $\eta_z$  = coefficient including a reverse of power tiller power mounting to 0.7 to 0.8 (Considered 0.75 as average value for present calculation).

The value of maximum moment on the rotor shaft ( $M_s$ ) was calculated through the following:

$$M_s = K_s \times R \quad (10)$$

In the above equation,  $R$  is the rotor radius (cm).

Considering the results obtained above, a suitable material with appropriate yield stress should be selected for manufacturing of rotor. The allowable stress on the rotor ( $\tau_{all}$ ) is calculated by the following equation [41]:

$$\tau_{all} = \frac{0.577k\sigma_y}{f} \quad (11)$$

In the equation,  $k$  is the coefficient of stress concentration equal to 0.75 and  $f$  is the coefficient of safety, which is equal to 2.

By replacing the values of  $k$  and  $f$  in the equation allowable stress on the rotor was found. The torsional moment is the most important factor that significantly affects the rotor shaft design [42]. Considering the equation for calculating the torsional moment on rotating shafts, the proper diameter for the rotary tiller shaft could be obtained:

$$d = \left( \frac{16M}{\tau_{all}} \right)^{1/3} \quad (12)$$

## 2.4 Power Tiller Rotavator Blade Design

The design of rotary tiller blade depends on type of blade, number of blade and arrangement of blade. In a rotary tiller 25% of blades act jointly on the soil [26]. The number of flanges for the rotary tiller is calculated by dividing the total

working width of power tiller by distance between two consecutive flanges.

The soil force acting on each of the blades ( $K_e$ ) is calculated by the following equation:

$$K_e = \frac{K_s C_p}{i Z_e n_e} \quad (13)$$

Where,  $K_s$  = maximum peripheral force, kg;  $C_p$  = overload factor of an individual working element, accepted as 1.5 for stone less soil and 2 for stony soil;  $i$  = number of flanges;  $Z_e$  = number of blades on each side of the flanges and  $n_e$  = the number of blades which action jointly on the soil / total number of blades of the rotavator.

### 3. RESULTS AND DISCUSSION

The calculated value of specific work and performable work is presented in Table 2. The required specific work increases with increase in forward velocity and velocity ratio as the dynamic specific work is directly proportional to square of forward velocity and velocity ratio. The performable work, that the power tiller is able to supply decreases with increase in forward velocity and width of operation.

In the below table all possible combinations of working width, forward speed and velocity ratio having specific work values less than performable work values were analyzed. Out of these, combinations having least difference

between performable work and specific work were selected for further analysis (Table 3). Rotor speed and bite length of selected combination were calculated using Equation (6) and (7) respectively and are updated in Table 3.

Achieving a higher forward velocity and width of operation is important to have a higher field capacity whereas a velocity ratio that is sufficient to create desired soil condition but low enough to have lower specific energy consumption is desired. Selection number 8 from the above table was selected as the best combination which uses available power resourcefully and also provides better field capacity. Theoretical field capacity of 0.1107 ha/h can be achieved with this forward velocity of 0.41 m/s and width of operation of 0.75 m in very heavy soil while tilling a depth of 15 cm. This performance is much better than earlier research conducted on the same power tiller model when considered with depth of operation and soil conditions [43-46]. Considering combination at selection number 8 and velocity ratio as 5; the value of velocity ratio at which the difference between the specific work required by rotavator and performable work of power tiller becomes zero was calculated, which was found to be 5.12. At this velocity ratio for working width 75 cm and forward velocity 0.41 m/s the RPM, bite length, peripheral velocity were calculated using Equation 6 and 7 and found to be 80.19, 10.22 cm, 2.09 m/s.

**Table 2. Calculated value of required specific work (for different speeds and velocity ratio) and performable work (for different speeds and width of operation)**

		<b>A (kg.m/dm<sup>3</sup>)</b>				
		<b>Gear Number</b>				
		-	-	1 heavy	-	2 heavy
		<b>v (m/s)</b>				
<b>λ = u/v</b>	<b>a<sub>v</sub></b>	<b>0.2</b>	<b>0.3</b>	<b>0.41</b>	<b>0.5</b>	<b>0.60</b>
4	6400	12.75	13.07	13.57	14.1	14.8
5	10000	12.9	13.4	14.18	15	16.1
6	14400	13.08	13.8	14.92	16.1	17.68
7	19600	13.28	14.26	15.79	17.4	19.55
8	25600	13.52	14.8	16.8	18.9	21.72
9	32400	13.79	15.41	17.94	20.6	24.16
10	40000	14.1	16.1	19.22	22.5	26.94
<b>b (dm)</b>		<b>A<sub>c</sub> (kg.m/dm<sup>3</sup>)</b>				
		<b>v (m/s)</b>				
		<b>0.2</b>	<b>0.3</b>	<b>0.41</b>	<b>0.5</b>	<b>0.60</b>
	6	36.56	24.37	17.83	14.62	12.18
	7.5	29.25	19.5	14.26	11.7	9.75
	9	24.37	16.25	11.89	9.75	8.12

**Table 3. Possible combinations of working parameters found**

Selection number	Working width (cm)	Forward speed (m/s)	$\lambda$	Rotor speed (rpm)	Bite length (cm)	Difference between specific work and performable work
1	60	0.2	10	76.4	5.23	23.71
2		0.3	10	114.6	5.23	9.52
3		0.41	8	125.29	6.54	1.08
4		0.5	4	76.4	13.08	0.52
6	75	0.2	10	76.4	5.23	15.15
7		0.3	10	114.6	5.23	8.27
8		0.41	5	78.31	10.47	0.08
10	90	0.2	10	76.4	5.23	10.27
11		0.3	10	114.6	5.23	0.15

The coefficient of power transmission in this study was found to be 0.148. Maximum peripheral force on rotor assembly was found to be 472.33 kg and the value of maximum moment on the rotor shaft ( $M_s$ ) was found as 11808.25 kg.cm. Considering the results obtained above, the rotor should be made from roll steel (AISI 302) having yield stress ( $\sigma_y$ ) of 520 MPa. Allowable stress on the rotor was found to be 1147.18 kg/cm<sup>2</sup>. The diameter of the rotor shaft calculated using the above equation and was found to be 3.74cm. Here number of flanges required is 5. Hence total 30 numbers of blades are required for the power tiller (considering  $Z_e = 3$ ). Here the soil force acting on each blade was found as 125.95 kg. The blade can be designed to bear this calculated value of soil force.

#### 4. CONCLUSION

The power tiller rotavator design was optimised for best utilisation of available power and to obtain desired soil condition while minimising specific energy consumption and enhancing the blade life. For this 13hp power tiller the optimum kinematic and design parameters were found as velocity ratio of 5.12, bite length of 10.22 cm, forward velocity of 0.41 m/s, rpm of 80.19, coefficient of power transmission 0.148, rotor shaft diameter of 3.74 cm, 30 number of blades mounted on 5 number of flanges.

This method of optimisation can be adopted for improved power tiller rotavator design and efficient operation. But as this method is purely theoretical; practical verification of the design is essential before adopting for large scale production.

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#### COMPETING INTERESTS

Authors have declared that no competing interests exist.

#### REFERENCES

1. Topakci M, Celik H K, Yilmaz D, Akinci I. Stress analysis on transmission gears of a rotary tiller using finite element method. *Akdeniz Universitesi Ziraat Fakültesi Dergisi*. 2008;21(2):155-166.
2. Godwin RJ, Spoor G, Soomro MS. The effect of tine arrangement on soil forces and disturbance. *Journal of Agricultural Engineering Research*. 1984;30:47-56.
3. Destain MF, Houmy K. Effects of design and kinematic parameters of rotary cultivators on soil structure. *Soil and Tillage Research*. 1990;17(3-4):291-301.
4. Mandang T, Hayashi N, Watanabe K, Tojo S. Studies on the overturning properties of soil by rotary blade using CT image analyzer. *ASAE Paper No. 93-3035*. St. Joseph, Michigan: American Society of Agricultural Engineers; 1993.
5. Sakai J. Theoretical approach to the hand tractor of rotary tillage. *Japan Agricultural Research Quarterly*. 1975;9(1):40-47.
6. Baloch JM, Bukhari SB, Kilgour J and Mughal AQA. Performance of power tiller blades. *Agricultural Mechanization in Asia, Africa and Latin America*. 1986;17:22-26.
7. Adams WJ, Furlong DB. Rotary Tiller in soil preparation. *Agr. Eng.* 1959;40:600-607.
8. Henderick JG, Gill WR. Rotary tiller design parameters, Part II, Depth of operation. *Transactions of the ASAE*. 1971b;14:675-678.

9. Gill WR, Hendrick JG. The irregularity of soil disturbance depth by circular and rotating tillage tools. *Transactions of the ASAE*. 1976;19(2):230-233.
10. Soehne W. Influence of the shape and arrangement of the tool upon the driving torque in rotary cultivators. *Grundlagen der Landtechnik*. 1957;9:69-87.
11. Mashchenski AA. Special process of high speed cutting of saturated soils with high speed rotary working tools. Springfield National Technical Information Service Publication; 1974.
12. Tsuchiya M, Honami N. Studies on the power reduction of rotary type power tillers. *Journal of the Japanese Society of Agricultural Machinery*. 1963;24(4):207-214.
13. Godwin RJ, Spoor G. Soil failure with narrow tines. *Agric. Eng. Res.* 1977;22(4):213.
14. Ahmad D. A Width of cut analysis on the performance of a rotary strip tiller. *PERTANIKA Journal of Science and Technology*. 1986;9:7-15.
15. Hendrick JG, Gill WR. Rotary tiller design parameters, Part I, Direction of rotation. *Transactions of the ASAE*. 1971;14:669-674.
16. Grinchuk IM, Matyashin YI. Systems of operation of soil rotary tillers. *MESK*. 1968;26(6):7-9. (NIAE trans. GILL/107).
17. Walton PG, Warboys IB. An investigation in to the power requirements of an experimental Wye double digger. *Journal of Agricultural Engineering*. 1986;35:213-225.
18. Niyamapa T, Thampatpong S, Rangdang C, Salokhe VM, Singh G. Laboratory investigation on design parameters of rotary tiller. In *Proceedings of the International Agricultural Engineering Conference and Exhibition, Bangkok, Thailand*. 1994;206-214.
19. Asl JH, Singh S. Effect of different machine and operating parameters on performance of rotary blades for strip tillage. *J Res Punjab agric Univ*. 2007;44(4):342-48.
20. Pal R, Bhimwal JP, Choudhary S. Effect of  $\lambda$  – ratio and depth of cut on draft, fuel consumption, power consumption and field efficiency of an offset rotavator under different type of orchards. *Journal of Advances in Biology & Biotechnology*. 2015;3(2):77-83.
21. Baker CJ, Saxton KE, Ritchie WR, Chamen WCT, Reicosky DC, Ribeiro F et al. *No-tillage seeding in conservation agriculture*. 2<sup>nd</sup> ed. Rome: FAO, CABI; 2007.
22. Celik A, Altikat S. Geometrical analysis of the effects of rotary tiller blade path on the distribution of soil slice size. *Applied Engineering in Agriculture*. 2008;24(4):409-413.
23. Butterworth B. *Rotary cultivators*. Power Farming. 1972;10(1):91-92.
24. Kosutic S, Filipovic D, Gospodaric Z. Agro technical and energetic characteristics of a rotary cultivator with spike tines in seedbed preparation. *Agric. Eng. J.* 1997;6:137-144.
25. Guo ZJ, Zhou ZL, Zhang Y. Bionic optimization design of soil cultivation components. *Science in China Series E: Technological Sciences*. 2009;39(4):720-728.
26. Bernacki H, Haman J, Kanfojki CZ. *Agricultural Machines: Theory and Construction*. Warsaw, Poland: The Scientific Publications Foreign Cooperative, Centre of Central Institute for Scientific Technical and Information. 1972;1.
27. Matyashin Yu. Means of decreasing energy requirements of rotary tillers. *Vestnik Selskokhozyais Vennoi Nuaki*. 1968;9:131-133.
28. Dalin AD, Pavlov PV. *Rotary soil working and earth moving machines*. In: Thakur TC, Godwin RJ, editors. *The present state of force prediction models for rotary powered tillage tools*. Pergamon Press Plc; 1950.
29. Furlong DB. *Rotary tiller performance tests on existing tines*. Tech. Report No-1049. Saint Jose, California, USA: FMC Corporation; 1956.
30. Bok NB. Determination of basic parameters of soil rotary tillers. *Tractory i Sel' khoz*. 1965;5(7):30-32.
31. Blaauw A. Development of the case hydraulic tiller. ASAE Paper No. 68-615. St. Joseph, Michigan: American Society of Agricultural Engineers; 1968.
32. Mandal SK, Bhattacharyya B, Mukherjee S, Chattopadhyay P. Design & Development of Rotavator blade: Interrogation of CAD Method. *International Journal of Scientific Research in Knowledge*. 2013;1(10):439.



33. Srinivasan K, Viswanath RP. Design and optimisation of blades for rotavators. *International Journal of Innovative Research in Science, Engineering and Technology*. 2015;4(4):128-137.
34. Mahapatra J, Sharma AK, Mehta AK, Mathur SM. Response surface design optimisation of power tiller rotavator blade using FEA Technique. *Green Farming*. 2018;9(6):1072-1075.
35. Mollazade K, Jafari A, Ebrahim E. Application of dynamical analysis to choose best subsoiler's shape using ANSYS. *New York Science Journal*. 2010; 3(3):93-100.
36. Mandal S, Bhattacharyya B, Mukherjee S. Design of rotary tiller's blade using specific work method (SWM). *Journal of Applied Mechanical Engineering*. 2015;4(3).
37. Anonymous. Speeds of VST Shakti 130 DI-Power Tiller; 2019. (Accessed 10 October 2019) Available: <https://www.vstractors.com/products/power-tiller/vst-shakti-130-di-power-tiller>
38. Kosutic S, Filipovic D, Gospodaric Z. Agro technical and energetic characteristics of a rotary cultivator with spike tines in seedbed preparation. *Agric. Eng. J*. 1997; 6:137-144.
39. Celik A, Altikat S. Geometrical analysis of the effects of rotary tiller blade path on the distribution of soil slice size. *Applied Engineering in agriculture*. 2008;24(4):409-413.
40. Sakai J. History of the development and classification of Japanese power tillers and hand tractors of multipurpose performance. *Agricultural Mechanisation in Asia, Africa and Latin America*. 1973;4(1): 89-94.
41. Mott R. *Machine elements in mechanical design*. Columbus, Ohio: Merrill publishing company; 1985.
42. Yatsuk EP. *Rotary soil working machines: construction, calculation and design*. New Delhi, India: Amerind Publication; 1981.
43. Adamu FA, Jahun BG, Babangida B. Performance evaluation of power tiller in Bauchi State Nigeria. *Journal of Biology, Agriculture and Healthcare*. 2014;4(9):10-14.
44. Ahmed B, Bekele A, Hailu G. On farm evaluation and demonstration of sifang power tiller for tillage and water lifting in arsi zone. *Journal of Biology, Agriculture and Healthcare*. 2018;8(7):28-32.
45. Rangapara DK, Dabhi KL and Makwana AD. Comparative performance of tractor drawn Implements tillage system with rotavator tillage system. *International Journal of Agriculture Sciences*. 2017;9(5): 3743-3748.
46. Ademiluyi SY, Oladele OI. Field performance of VST Shakti power tiller on sawah rice plots in Nigeria and Ghana. *Bulgarian Journal of Agricultural Science*. 2008;14(5):517-522.

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