Rotary Tiller Design Proportional to a Power Tiller using Specific Work Method (SWM)

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Abstract: The present research has dealt with rotary tiller design for the power tiller model Mitsubishi VST SHAKTI 130DI that is made for using in primary and secondary tillage. In this design, the rotary tiller work width determined 70 cm appropriate to the power tiller, by estimating the total specific work, which is equal to the sum of static and dynamic specific work of tiller. It was revealed that the tiller model Mitsubishi VST SHAKTI 130DI is only able to pull the rotary tiller with 70 cm width and 15 cm depth, at gear I. For selecting work width of tiller, maximum benefit power of the power tiller was considered, which it could be decreased by increasing work speed. In designing the rotary tiller shaft, it was revealed that in addition to the torsional moment, the flexural moment was also effective on the system safely design. It was also recognized that in designing a rotary tiller, blades are most subjected to be fractured by incoming stresses. The optimal value of rotor's diameter considering the values of maximum tangent force was determined about 3.94 cm. This paper presents a theoretical method for rotary tillers design. The results of this study should be verified by further tests on rotary tillers according to the results offered in this paper. [Nature and Science 2010;8(9):39-45]. (ISSN: 1545-0740).

Key words: Rotary tiller, Garden tractor, Design, specific work

1. Introduction

A rotary tiller is a type of motorized cultivating equipment that breaks or works the soil with the aid of rotating blades. Rotary tillers are available with advanced technologies and innovative designs which provide great performance. The rotary tiller can be self-propelled and driven forward on wheels. Featuring a gearbox, the rotary tiller enables one to increase the rotation speed of the blades more than the forward speed of the equipment. Rotary tillers have become world famous for preparation of seedbed in fields. These equipments are often used for breaking or working the soil in lawns, gardens, etc (Hendrick *et al.*, 1971). Garden rotary tillers are used for the plantation of seeds in your garden or backyard. Electric rotary tillers are commonly used in gardens and fields. Nowadays, utilization of rotary tillers has been increased in agricultural applications because of simple structure and high efficiency for this type of tillage implements. By taking advantage of rotary tillers, the primary and secondary tillage applications could be conjugated in one stage (Topakci *et al.*, 2008). Therefore, the agricultural soil could be prepared with only one cross of this type of tillage implements from the land. This results in a decrease in the number of machinery passes on the land and subsequently, causes a decrease in the soil compaction which could be obtained due to the excessive equipments crosses from the land. Despite of their high energy consumption, since rotary tillers have the ability of making several types of tillage applications in one stage, the total power needed for these equipments is low (Culpin, 1981). Because rotary tillers power is directly transmitted to the tillage blades, the power transmission efficiency in rotary tillers is high. Moreover, the negative traction existence in rotary tillers causes the required tractive force to be decreased and consequently, smaller tractors could be used with this type of tillage implements for land preparation. Rotary tillers are classified as active implements. In these machines, power is transferred to the tiller from the tractor via the power-take-off drive. A shaft containing blades is located at 90°to the line of travel and rotates in the same direction as the forward travel of the tractor. Since the shaft turns at a rate that is considerably faster than the corresponding tractor speed, soil pulverization is accomplished. Power to operate the rotary tiller is restricted by available tractor power (Yatsuk *et al.*, 1981; Srivastava *et al.*, 2006).

The objective of this study was to determine the optimal design of a rotary tiller proportionate to the power tiller model of Mitsubishi VST SHAKTI 130DI that is the most commonly power tiller used on the north of Iran farms. This paper presents a new theoretical approach to design main tillage components of rotary tillers.

Materials and Methods

Technical characteristics of the power tiller

For designing the rotary tiller, the power tiller model of Mitsubishi VST SHAKTI 130DI was considered as the rotary tiller power supply. The technical characteristics of the power tiller model of Mitsubishi VST SHAKTI 130DI are presented in Table 1.

Table 1- Technical specifications of power tiller model of Mitsubishi VST SHAKTI 130DI

Specification	Value
Number of cylinder	1
Engine maximum power at 2400 rpm	13 hp
Engine maximum torque at 1900 rpm	4.2 kh-m
Rotational speed PTO shaft	540 rpm
Total weight	120 kg

Specific work method (SWM)

In order to design a rotary tiller, the special work of the tiller and also the performable work of the tractor should be determined. The specific work of rotary tiller is defined as the work carried on by rotary tiller at each rotation of tillage blades per the volume of broken soil, which could be calculated by the following equation (Bernacki *et al*, 1972):

$$A = A_0 + A_B \qquad \left(\frac{kg-m}{dm^3}\right) \tag{1}$$

Where: A_o and A_B are the static specific work and dynamic specific work of rotary tiller (Kg-m/dm^3), respectively, which can be calculated throw the following equations (Bernacki *et al*, 1972):

$$A_0 = 0.1 C_0 K_0 \qquad \left(\frac{kg-m}{dm^3}\right) \tag{2}$$

$$A_B = 0.001 a_u u^2 \qquad \left(\frac{kg-m}{dm^3}\right) \tag{3}$$

$$A_B = 0.001 a_v v^2 \qquad \left(\frac{kg-m}{dm^3}\right) \tag{4}$$

Where: C_o is the coefficient relative to the soil type, K_o is the specific strength of soil (kg/dm^3), u is the tangential speed of the blades (m/s), v is the forward speed (m/s), a_u and a_v are dynamical coefficients that are relative together throw the following equation:

$$a_v = a_u \lambda^2 \qquad \left(\frac{kg-s^2}{m^4}\right) \tag{5}$$

Where:

$$\lambda = \frac{u}{v} \tag{6}$$

The performable work of the tractor (A_c) could be calculated by the following equation (Bernacki *et al*, 1972):

$$A_C = \frac{7.5 N_C \eta_C \eta_Z}{vab} \qquad \left(\frac{kg-m}{dm^3}\right) \tag{7}$$

Where: N_c is the power of tractor (hp), v is the forward speed (m/s), η_c is the traction efficiency that its value for the forward rotation of the rotary tiller shaft is 0.9; whilst the value for the reverse rotation of the rotary tiller is considered between 0.8-0.9, η_z is the coefficient of reservation of tractor power which is between 0.7-0.8, a is the rotary tiller work depth (dm) and b is the tiller work width (dm).

Matyashin (1968) reported that at the forward rotation of the rotary tiller shaft, the tillage power consumption is decreased 10-15 %, in comparison with the shaft reverse rotation. Hence, in this design the forward rotation was considered for the rotary tiller shaft to reduce the tiller power consumption and also utilization of the rotary tiller thrust force at the forward rotation. In designing the rotary tiller, the hard condition of the soil was considered. The values of C_o, K_o and a_u in very heavy soils are 2.5, 50 (kg/dm^3), and 400 (kg.s^2/m^4), respectively (Bernacki *et al*, 1972). Therefore, the static special work of the rotary tiller could be calculated by replacing the values in the equation (2):

$$A_0 = 0.1 \times 2.25 \times 50 = 11.25 \qquad \left(\frac{kg-m}{dm^3}\right)$$

Since the values of b, v, λ and a_v are not given in the equations (4), (5) and (7), a proper domain should be defined for the values at first, with respect to the technical specifications of the selected power tiller for this design. Then, the optimum condition for the rotary tiller design could be selected from the domain. The recommended work width for the power tiller model of Mitsubishi VST SHAKTI 130DI is 60 cm. The distance between rotary tiller flanges in this design was considered equal to 20 cm. Therefore, the work width domain of the rotary tiller that is the multiple of the distance between the flanges is in the range of 50, 60, 70 and 80 cm. This range was selected with respect to the power tiller work width. The forward speed of the power tiller model of Mitsubishi VST SHAKTI 130DI at different transmission gears are presented in Table 2.

Table 2- The forward speed of the power tiller model of Mitsubishi VST SHAKTI 130DI at different transmission gears.

Transmission gear	Forward speed (km/h)	Forward speed (m/s)
1 - Forward -Heavy	1.80	0.50
2 - Forward -Heavy	2.64	0.73
3 - Forward-Heavy	4.20	1.17
1 - Forward - Low	6.42	1.78
2 - Forward - Low	9.50	2.64
3 - Forward - Low	15.00	4.17
1 - Reverse	1.55	0.43
2 - Reverse	5.60	1.56

Because at the high levels of power tiller forward speeds, the penetration ability of the rotary tiller blades in the soil reduces, in this design only the heavy transmission gears were considered as the domain of forward speed. Hendrick and Gill (1971) suggested the minimum value of 2.5 for λ. Hence, a domain from 2 to 22 and from 0.2 to 2 was considered for λ and forward speed, respectively, to provide a large section range for the rotary tiller design.

According to the explanations offered above and by equations (1), (4) and (7), the values of *A* and A_c could be calculated:

$$A = A_o + A_B = 11.25 + 0.001 a_v v^2 \tag{8}$$

$$A_c = \frac{43.87}{vb} \tag{9}$$

By using equations (8) and (9), the values of *A* and A_c with respect to the defined domain for λ and v are obtained in Table 3 (Mohammadi Alasti *et al.*, 2008).

Table 3- The values of specific work of rotary tiller and maximum work of power tiller at different values of V and λ

		Gear Number				1 High		2 High		3 High				
λ=u/v	a_v	v (m/s)	0.2	0.3	0.4	0.5	0.6	0.73	0.8	1.17	1.2	1.4	1.6	2
2	1600	A ($kg.m/dm^3$)	11.31	11.39	11.50	11.65	11.83	12.10	12.27	13.44	13.55	14.38	15.35	17.65
3	3600		11.39	11.57	11.83	12.15	12.55	13.17	13.55	16.18	16.43	18.31	20.47	25.65
4	6400		11.51	11.83	12.27	12.85	13.55	14.66	15.35	20.01	20.47	23.79	27.63	36.85
6	14400		11.83	12.55	13.55	14.85	16.43	18.92	20.47	30.96	31.98	39.47	48.11	68.85
8	25600		12.27	13.55	15.35	17.65	20.47	24.89	27.63	46.29	48.11	61.43	76.78	113.65
10	40000		12.85	14.85	17.65	21.25	25.65	32.56	36.85	66.01	68.85	89.65	113.65	171.25
12	57600		13.55	16.43	20.47	25.65	31.98	41.94	48.11	90.09	94.19	124.15	158.71	241.65
14	78400		14.38	18.30	23.79	30.85	39.47	53.02	61.43	118.57	124.15	164.91	211.95	324.85
16	102400		15.35	20.46	27.63	36.85	48.11	65.82	76.78	151.42	158.71	211.95	273.39	420.85
18	129600		16.43	22.91	31.98	43.65	57.91	80.31	94.19	188.65	197.87	265.27	343.03	529.65
20	160000		17.65	25.65	36.85	51.25	68.85	96.51	113.65	230.27	241.65	324.85	420.85	651.25
22	193600		18.99	28.67	42.23	59.65	80.95	114.42	135.15	276.27	290.03	390.71	506.87	785.65
b (dm)	5	A_c ($kg.m/dm^3$)	43.87	29.24	21.93	17.55	14.62	12.02	10.97	7.49	7.31	6.27	5.48	4.38
	6		36.56	24.37	18.28	14.62	12.18	10.01	9.14	6.25	6.09	5.22	4.57	3.65
	7		31.33	20.89	15.67	12.53	10.44	8.58	7.83	5.36	5.22	4.47	3.92	3.13
	8		27.42	18.28	13.71	10.96	9.14	7.51	6.85	4.68	4.57	3.92	3.43	2.74

The proper selection of the forward speed is dependent to the tangential speed of the blades (that is a function of rotational speed of rotor) and the length of sliced soil. The tangential speed of the blades (*u*), the rotational speed of the rotor (*n*), and the length of sliced soil (*L*) could be obtained by the following equations:

$$u = \frac{2\pi nR}{60000} \tag{10}$$

$$n = \frac{6000\lambda v}{2\pi R} \tag{11}$$

$$L = \frac{2\pi R}{\lambda Z} \tag{12}$$

In the equations, *R* is the rotor radius (cm), *v* is the forward speed (m/s) and *Z* is the number of blades on each side of the rotor flanges. In this design, two blades were considered on each side of the flanges (*Z*=2). The working depth selected for the rotary tiller in this design was 15 cm. 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 (Matyashin, 1968). Considering these explanations, a 50 cm diameter was diagnosed to be appropriate for the rotary tiller rotor. By replacing the selected value for the rotor diameter in the equations (11) and (12), we will have:

$$n = \frac{6000\lambda v}{2\pi \times 25} = 38.21\lambda v \qquad (13)$$

$$L = \frac{2\pi \times 25}{\lambda \times 2} = \frac{25\pi}{\lambda} \qquad (14)$$

The total possible selections for the rotary tiller working width (b), forward speed (v) and rotational speed of rotor (n) are presented in Table 4. Table 4 is obtained through equations (13) and (14). Firstly, for each of the selected working widths in Table 3, the closest value of the rotary tiller special work to the performable work of the power tiller was determined at each of the forward speeds. Then, the corresponding values of λ for each forward speed were determined to calculate the rotor speed and the length of sliced soil (Table 3). By selecting the rotary tiller special work and the performable work of the power tiller close together at each of the forward speeds, an appropriate conformity will be continued between the rotary tiller and power tiller.

Kepner (1972) reported that the power needed of the tractor PTO for supplying a rotary tiller should be approximately 1 hp for each centimeter of working width. Considering the suitable domain obtained for the rotor speed, the length of sliced soil and the forward speed, at the working width of 70 cm, this width was selected as a proper working width for the power tiller. Moreover, at the selected working width there was a little difference between the rotary tiller special work and the performable work of the power tiller (Table 4).

Table 4- Total possible selections for the rotary tiller working width, forward speed and rotational speed of rotor

Selection NO.	Working width (cm)	Forward speed (m/s)	(λ)	Rotor speed (rpm)	Length of sliced soil (cm)	Difference between rotary tiller specific work and tractor maximum work (kg.m/dm^3)
1	50 cm	0.2	22	168.08	3.57	24.88
2		0.3	22	252.12	3.57	0.57
3		0.4	12	183.36	6.54	1.46
4		0.5	6	114.60	13.09	2.70
5		0.6	4	91.68	19.63	1.07
6	60 cm	0.2	22	168.08	3.57	17.57
7		0.3	18	206.28	4.36	1.46
8		0.4	10	152.80	7.85	0.63
9		0.5	4	76.40	19.63	1.77
10		0.6	2	45.84	39.27	0.35
11	70 cm	0.2	22	168.08	3.57	12.34
12		0.3	16	183.36	4.91	0.43
13		0.4	8	122.24	9.82	0.32
14		0.5	3	57.30	26.18	0.38
15	80 cm	0.2	22	168.80	3.57	8.43
16		0.3	12	137.52	6.54	1.8
17		0.4	6	91.68	13.09	0.16

Considering the results presented in Tables 1 and 2, it becomes evident that the selected power tiller for this design only at the gear one can supply a rotary tiller with the working width of 70 cm and working depth of 15 cm.

After specifying the appropriate working width for the power tiller, the length of sliced soil, the rotational speed of the rotor and the tangential speed of the blades should be calculated at the selected gear (the forward speed of 0.50 m/s). Before performing the mentioned calculations, the appropriate value of λ proportional to the selected forward speed for the power tiller should be obtained. For this purpose, the special work of the rotary tiller and the performable work of the power tiller should be equal together. Therefore, we will have:

$$GearNo.:1 \rightarrow v = 0.5\ m/s \xrightarrow{A=A_C} \lambda = 3.58$$

By representing the obtained value for λ at the equations (10), (11) and (12) we will have:

L=21.92 cm,

n=68.37 rpm,

u=1.81 m/s

Design of the power transmission system

After calculating the rotor speed, a transmission system should be designed for the rotary tiller. The coefficient of power transmission (i_c) for the rotary tiller could be calculated by the following equation:

$$i_c = \frac{n_d}{n_a} \quad (15)$$

In the equation, n_a and n_d are the rotational speed of the power tiller PTO shaft (rpm) and the rotational speed of the rotary tiller rotor (rpm), respectively. The rotational speed of the power tiller selected for this design is 540 rpm (Table 1). The rotational speed of the rotary tiller rotor ws also calculated at the previous section. By representing the values of na and nd at the equation (15), the coefficient of power transmission will be obtained:

$i_c = 0.127$

This means that a power transmission system should be selected for this design to give the calculated coefficient of speed alteration equal to 0.127.

Design of rotor shaft

For designing the rotor shaft, the maximum tangential force which can be endured by the rotor should be considered. The maximum tangential force occurs at the minimum of blades tangential speed is calculated by the following (Berbacki *et al*., 1972):

$$K_s = C_s \frac{75 N_c \eta_c \eta_z}{u_{\min}} \quad (16)$$

C_s is the reliability factor that is equal to 1.5 for non-rocky soils and 2 for rocky soils (Berbacki *et al.*, 1972). From equations (6) and (12), it becomes evident that u_{min} is obtained at λ_{min}; and λ_{min} is obtained at L_{max}. So, we will have:

$$\lambda_{\min} = \frac{2\pi R}{Z L_{\max}} = \frac{2\pi \times 25}{2 \times 21.92} = 3.58$$

$$u_{\min} = v\lambda_{\min} = 0.5 \times 3.58 = 1.79 \quad m/s$$

By representing the values of λ_{min} and u_{min} at the equation (16), the maximum tangential force on the rotary tiller shaft will be obtained:

$$K_s = 1.5 \times \frac{30 \times 13 \times 0.9 \times 0.75}{1.79} = 551.50 \quad kg$$

The maximum moment on the rotor shaft (M_s) is calculated through the following:

$$M_s = K_s R = 551.50 \times 25 = 13787.62 \quad kg - cm$$

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

Considering the results obtained above, the rotor should be made from roll steel (AISI 302) having yield stress of 520 MPa. The allowable stress on the rotor (τ_{all}) is calculated by the following equation (Mott, 1985):

$$\tau_{all} = \frac{0.577 k \sigma_y}{f} \xrightarrow{\sigma_y = 500 MPa} \tau_{all} = \frac{300k}{f} \quad (17)$$

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 (17):

$$\tau_{all} = \frac{300}{2} \times 0.75 = 112.5\ MPa = 1147.18 \quad kg/cm^2$$

The torsional moment is the most important factor that significantly affects the rotor shaft design (Yatsuk *et al*., 1981). Considering the equation for calculating the torsional moment on rotating shafts, the proper diameter for the rotary tiller shaft could be obtained:

$$d = \sqrt[3]{\frac{16 M_s}{\tau \pi}} = \sqrt[3]{\frac{16 \times 13787.62}{1147.18 \times \pi}} = 3.94\ cm$$

Rotary tiller blades design

The design of rotary tiller blades depends on the type and number of the blades and also the working condition of rotary tiller. In this design, the L type blades were considered for the rotary tiller according to the working condition presented. In rotary tillers, one fourth of the blades action jointly on the soil. The total power of the machine is distributed between the blades. The number of rotary tiller flanges (*i*) can be calculated by the following equation:

$$i = \frac{b}{b_i} = \frac{70}{20} \approx 3$$

Which: *b* is the working width and b_i is the distance between the flanges on the rotor. Four blades are considered on each of the flanges (Z_e = 4). Therefore, the total number of the blades is obtained:

$$N = i \times Z_e = 3 \times 4 = 12$$

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

$$K_e = \frac{K_s C_p}{i Z_e n_e} \quad (18)$$

Which: K_s is the maximum tangential force (kg), C_p is the coefficient of tangential force, i is the number of flanges, Z_e is the number of blades on each side of the flanges, and n_e is obtained through division the number of blades which action jointly on the soil into the total number of blades. Considering these definitions, K_e can be calculated:

$$K_e = \frac{551.5 \times 2}{3 \times 4 \times \frac{1}{4}} = 367.67 \quad kg$$

The dimensions of the blades are defined as the form presented in Figure 1. The values of be, he, Ss and S1 are considered equal to 1, 4, 9 and 6 mm, respectively.

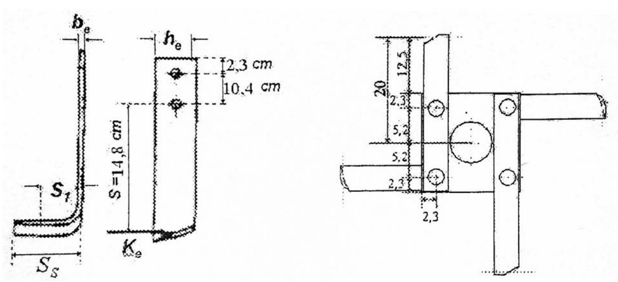

Figure 1. Dimensions of flange and blade

Considering the shape of the blades, the bending stress (σ_{zg}), the shear stress (τ_{skt}), and the equivalent stress (σ_{zt}) can be calculated by the following equations (Bernacki *et al.*, 1972):

$$\sigma_{zg} = 6\frac{K_e S}{b_e h_e^2} = 6 \times \frac{367.67 \times 14.8}{1 \times 4^2} = 2040.57 \quad kg/cm^2$$

$$\tau_{skt} = \frac{3 K_e S_1}{\left(\frac{h_e}{b_e} - 0.63\right) b_e^3} = 1963.82 \quad kg/cm^2$$

$$\sigma_{zt} = \sqrt{\sigma_{zg}^2 + 4\tau_{skt}^2} = 4426.08\, kg/cm^2 = 434.05 \quad MPa$$

Conclusion

Optimal working width and optimal diameter of rotary tiller proportionate to the power tiller model of Mitsubishi VST SHAKTI 130DI were determined in order to achieve to the maximum field efficiency for the rotary tiller and to minimize the consumed materials in the building of this machine. The rotary tiller was designed with the working width of 70 cm having 3 flanges on the rotor shaft and four blades on each flange. It was also concluded that the power tiller selected for supporting the rotary tiller, could only pull the rotary tiller at first heavy gear. A rotational speed of 69 rpm was selected for the rotor. The optimal value of rotor's diameter considering the values of maximum tangent force was determined about 3.94 cm. This paper presents a theoretical method for rotary tillers design. The results of this study should be verified by further tests on rotary tillers according to the results offered in this paper.

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