THESIS

THE OPTIMUM DESIGN PARAMETERS FOR A ROTARY BLADE POWER TILLER UNDER UNSATURATED SANDY CLAY LOAM CONDITION

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This work is devoted to my beloved country, Ethiopia, and its great people, and as their representative for small and medium scale holders.

Finally, I give all glory to my Lord and Savoir, Jesus Christ. “Without God’s blessing, I look for no success; and for every success, my prayer is that all glory will be given unto Him, to whom it is properly due.

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One of the major concerns of agricultural machinery engineers is to develop high performance tillage equipment with low energy requirements for desired soil manipulation capability. Optimization theory has been applied widely in agricultural, mechanical, electrical and civil engineering fields, Particularly, it is used in engineering science for the development of new electronics machines, computers, and robots to reduce labour power. Moreover, it has been also applied for the development of agricultural machinery.

The study focused on optimizing design parameters in terms of total specific energy requirements of various rotary blades of a power tiller, namely, a “Pick”, a “C”, an “I” an “L” and a “J”- shaped rotary blade under unsaturated sandy clay loam condition. Using an optimization solver, optimum design parameters of the rotary blades have been obtained that have influenced the total specific energy requirement model. The simulated total specific energy requirement (E_{TSP}) was predicted to be 231.61, 160.72, 196.87, 168.56 and 167.56 kJ/m^3 for the “Pick”, the “C”, the “I”, the “L” and the “J”- shaped rotary blade, respectively. The highest specific energy requirement was exhibited by the “Pick”- shaped and lowest by the “C”- shaped. Optimum design parameters of the” Pick”- shaped, namely, rotational velocity, rotor radius, depth of tillage and cutting width of 150 rpm, 170, 100 and 10 mm were obtained, respectively. The “Pick”- shaped blade was predicted to be optimum in the preset study. Accordingly, the higher specific energy requirement the lower volume of soil tilled and the most effective and optimum soil tillage operational cost were determined. Compared to another study in the same soil condition the specific energy requirement per volume of soil tilled by the “Pick”- shaped was exhibited 1900 kJ/m^3 which was higher by 87.81 % than the “Pick”- shaped blade of the study. Consequently, the specific fuel consumption requirement was predicted to be 1800 kJ/L.

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LIST OF SYMBOLS

Symbol	Description
α	Angle of direction, degree
φ_1	Angle of peripheral, degree
R	Rotor radius, *m*
L_B	Bite length, *m*
V_P	Blade peripheral velocity, *m/sec*
η_Z	Coefficient including a reverse of power tiller power mounting, percent
B_W	Cutting width, *m*
L_d	Depth of tillage, *m*
μ_K	Kinetic coefficient of soil –metal friction
F_P	Peripheral force, *N*
η_C	Power tiller efficiency mounting
R_T	Soil reaction force, *N*
E_E	Specific work, kJ/m^3
F_T	Torque requirement, *Nm*
V_{ST}	Volume of soil tilled, m^3
P_{Cut}	Cutting power requirement, *W*
S_{Pw}	Dry soil bulk density, kg/m^3
V_{Cut}	Soil slice cutting velocity, *m/s*
K_{SP}	Specific soil resistance, kg/m^2
F_C	Fuel consumption, *L/ kWh*

R_{RX}	Horizontal component of reaction force, *N*
P_{fircn}	Overcoming soil –metal friction power requirement, *W*
P_{push}	Pushing power requirement, *W*
F_{SPC}	Specific fuel consumption, *L/ kW*
P_{TSP}	Total specific power requirement, kJ / m^3
P_{Total}	Total power requirement, *W*
P_{Throw}	Throwing the cut soil slice power requirement, *W*
V_f	Machine forward velocity, *m/s*
P	Draw bar power, *kW*

LIST OF ABBREVIATIONS

%	=	Percent
<	=	Less than
>	=	Greater than
Agric.	=	Agricultural
Appl.	=	Applied
ASAE	=	American Society of Agricultural Engineers
CAC	=	Central Agricultural Census
Eng	=	Engineering
Engng.	=	Engineering
Engrs.	=	Engineers
Eqn	=	Equation
g	=	Gram
GAMS	=	General Algebraic Modelling System
GO	=	Global optimization
h	=	Hour
ha	=	Hectare
hp	=	Horse power
Inst.	=	Institute
J.	=	Journal
k	=	Kilo
kg	=	Kilogram
kg	=	Kilo gram
km	=	Kilometre
kW	=	Kilo watt
KWh	=	Kilo watt hour
m	=	Metre
Mach.	=	Machinery
Mat.	=	Mathematics
Mech.	=	Mechanization
Min.	=	Minute

MJ	=	Mega Joule
mm	=	Millimetre
N	=	Newton
N.m	=	Newton metre
Rad.	=	Radian
Res.	=	Research
s	=	Second
Sci.	=	Science
Soc.	=	Society
Terramech.	=	Terramechanics
Trans.	=	Transaction

THE OPTIMUM DESIGN PARAMETERS FOR A ROTARY BLADE POWER TILLER UNDER UNSATURATED SANDY CLAY LOAM CONDITION

INTRODUCTION

Agricultural Mechanization Status of Ethiopia

Ethiopia is a mountainous country located in the tropics between 4^0 and 14^0 North and 33^0 to 48^0 East with rugged terrains consisting of high mountains, steep hills, and consists of a central high plateau bisected by the segment of the great rift valley into northern and southern highlands and surrounded by lowlands, more extensive on the east and southeast than on the south and west. The plateau varies from 1500 to 3000 meters above sea level and features mountainous uplands separated by deep gorges and river valleys, in the east, the Denakil Depression, part of the rift valley, is 115 meters below at sea level and is one of the hottest places on earth. Principal rivers originate in the highlands and drain into the surrounding lowlands. The Abay (Blue Nile) is the largest river, the Tekezé, and the Baro flow west into the Nile River in Sudan, the Blue Nile contributes some two-thirds of the Nile's volume below Khartoum. The Awash flows east through the northern Rift Valley and disappears into saline lakes in the Denakil Depression. In the south, the Genale and Shebele flow southeastward into Somalia, the Omo drains southwest and empties into Lake Turkana on the border with Kenya. Rainfall and temperature patterns vary widely because of Ethiopia's location in the tropics and its diverse topography. The highlands above 1500 meters enjoy a pleasant, temperate climate, with daytime temperatures between 16° C and 30° C and cool nights. In areas below 1500 meters, such as large river valleys, the Denakil Depression, the Ogden in the southeast, and parts of the southern and western borderlands, daytime temperatures range from very warm (30° C) to torrid (upwards of 50° C), sometimes accompanied by high humidity. Precipitation is determined by differences in elevation and by seasonal shifts in monsoon winds. The highlands receive by far the most rainfall, most of it between mid-June and mid-September, whereas lower elevations receive much less.

Relative humidity and rainfall decrease from south to north and vary from scant to negligible in the eastern and south-eastern lowlands. Annual precipitation ranges from 800 to 2200 mm in the highlands (>1500 meters) and varies from less than 200 to 800 mm in the lowlands (<1500 meters).

It is endowed with a wide range of natural resources such as land, irrigation potential and agro-ecological diversities favourable for the growing various crops, such as cereals, fibre crops, oil seeds, coffee, tea, fruits and vegetables are grown in the country. Population is approximately 77.1 million with a growth rate of 2.77 %. The total land area is 1127127 km^2, land 1119683 km^2 and water 7 444 km^2. Land use covered by rural and urban areas is categorized as temporary 8.19, permanent crops 0.67, grazing 0.96, fallow 0.84, wood 0.087 and other lands 0.30 million hectares, respectively. The agricultural holders are categorized as "crop only", "livestock only" and "crop and livestock ". All types of the holdings are classified in to seven – area size of holdings , namely, under 0.10, 0.10 to 0.50 ,0.51 to 1.00 , 1.01 to 2.0, 2.01 to 5.00 , 5.01 to 10.00 and above 10 hectares, respectively (CAC, 2003). About 11.05 million hectares are under rural and urban areas, out of which about 10.89 in rural and 0.16 million hectares in urban areas. There were 54.5 million people in 10.6 million agricultural households. The average size of land holding in the rural areas of the country was estimated to be 1.03 hectares per holder, while 0.79 hectare in urban areas (CAC, 2003). Besides, about 90 % of foreign currency comes from the agricultural sector.

Agricultural production in Ethiopia is mainly undertaken under rain fed condition, with very limited areas currently under irrigation. Of the 4.3 million hectares of the potential of irrigated agriculture only 5% is currently utilized. It depends mainly on traditional methods of farming.

Food sufficiency is important to develop a country. Agricultural mechanization can enhance food production by increasing cropping intensity. Increasing mechanization in agriculture can result in increased employment and income as well as improving the economic and social well-being of people. It helps to increase production and to preserve food and fiber crops and improve efficiency. Mechanization includes a use of large machines, improved hand tools, animal-drawn implements, and rotary walking tractors.

Mechanization plays a tremendous role in the agricultural sector as pre-harvest and post harvest technology. Its main benefits are the reduction of drudgery of labor and a reduction of loss of crop and food products. The general trend in agricultural mechanization throughout small holders of the country ranges from manual use to animal drawn agricultural implements.

Sahay (1992) stated that farm mechanization is the application of engineering and technology in agricultural operations to enable jobs to be done in a better way to improve productivity. It includes the development, application and management of all mechanical aides for field production, water control, material handling, storage and processing. Mechanical aides include hand tools, animal drawn equipment, power tillers; tractors oil engines, electric motors, processing and hauling. A rotary walking tractor is a prime mover where the operator walks behind it, holding the two handles to perform the direction of travel and its control for field operation. A rotary walking tractor consists of an engine; transmission gears, clutch brakes and rotary unit for field operation for the soil cutting and pulverization purpose. A rotary unit is a centre drive type, with transmission at the centre and a side drive type has transmission at one side.

In Ethiopia, the system of mechanization is a mixed mechanization system, and in the farm sector operations, mainly cultivation and harvesting, are carried out partial by manual and by animal – drawn implements such as traditional designs of a wide variety of hand tools such as shovels, spades, forks, digging hooks, hoes, ploughs, and animal drawn implements.

Traditional farm implements have been used by small holders under Ethiopian farm mechanization. However, some medium scale holders and commercial farms are more mechanised. Despite the fact of that capital intensive, mechanized and market oriented farming systems, with increased use of modern management practices and agricultural inputs such as, use of high of tech-farm machinery and implements, irrigation scheme, use of chemical fertilizers, pesticides and improved seeds, medium scale holders and commercial farms are not widely spread, as a result, contribution to the country's gross total agricultural output was at a stagnated level of 4 to 6 %. However, 94 to 96 % of the annual gross total agricultural output of the country was generated by small holding farmers in high land area approximately 54 million of the total population of the country (Figure 1).

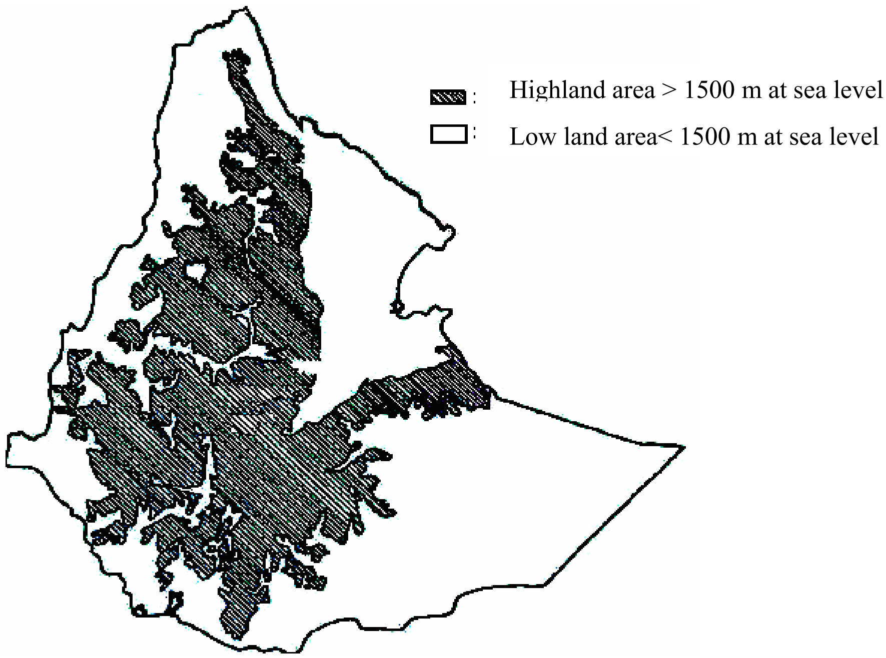


Figure 1 The Ethiopian highlands

Animal traction in Ethiopia is believed to have been introduced between 1000 and 4000 BC by Semitic speaking invaders from south Arabia (Goe, 1987). The animal drawn mouldboard plough was for the first time introduced to Ethiopia by Italians in 1939 (Goe, 1987). However, farmers rejected the plough because of its heavy weight, high draft power requirement and complicated adjustment and attachment systems. The Italians concluded that the Ethiopian farmers were conservative and did not want to take on new technologies. A number of researches conducted several trials on small farm implements in different institutions and national agricultural mechanization research programs. However, the implements have not been significantly accepted by farmers.

The only implement used for land preparation and planting is the traditional plough or 'maresha' which is a pointed, steel-tipped, tine attached to a draught pole at an adjustable shallow angle. Narrow wooden wings attached on each side of the tine push soil to either side but do not invert it.

It has certain advantages. Apart from the metal point and the hook it is entirely home made. It is light, not exceeding 25 kg (Goe, 1987), and can easily be carried to and from fields. The power requirement can be adjusted by the depth control and does not normally exceed the power developed by a pair of local Zebu oxen. About 2 to 5 passes have been made by the implement before the land is ready for sowing. Each pass is made perpendicular to the previous one. Average Time required for land preparation is 120 h/ha depending on the soil type. After being broadcast, seeds are unevenly covered by a final pass with the implement and germination is poor. It has been overcoming this problem that farmers generally have used higher seed rates (Astatke and Matthews 1983).

Statement of the problem

Unbalanced crop production and population growth, erratic rain fall, frequent droughts, migration, deforestation and soil degradation, inadequate power equipment, and soil compaction are interrelated problems of the country. One of the Ethiopian agricultural research organization strategies is to facilitate training program to strengthen capacity building, adapt imported designs, develop and to improve small mechanical farm power and tillage implements.

Agricultural mechanization in Ethiopia lags behind other countries in Africa. Agricultural mechanization has not progressed well. Agricultural development in Ethiopia is still limited and will require better and improved seeds, fertilizers, and pest and weed control methods, and especially land preparation should be done on time. Farmers could adapt the new technology using low power equipment, such as walking type power tillers, by producing agricultural equipment, and importing appropriate agricultural machinery from other counties to help increase their agricultural production, and the trend in the transition of simple designs to small tractorization. However, at present, farmers lack appropriate draft power for land preparation and water for irrigation. This presents major problems, so farmers cannot grow crops at all without timely land preparation and water supply. The major contribution of agricultural mechanization is to help ensure optimum conditions for growing a crop at the right time.

However, the majority of small holders are living in the highlands of the country where big tractors are not affordable and, therefore, they are engaged to use unimproved animal drawn implements. Almost all of the field operations are carried out using hand - tools and local tillage implements with human and animal power. Similarly, farmstead operations in crop production and animal husbandry, and forestry works by and large are performed with bare hands or with very rudimentary tools and implements. As a result labour productivity is very low. Several studies have shown that there is a great imbalance between the labour input and the output obtained in all agricultural production work. This has led to a chronic shortage of food in the farming population. In many cases farmers have been unable to feed their own family all year round and are unlikely to have a surplus product for marketing and earning cash.

Land and labour productivity are low in Ethiopia because of the low level of technology utilization. In the cereal growing areas, land preparation is carried out with the help of a locally constructed traditional ploughs ('maresha') drawn by a pair of oxen. The plough doesn't turn the soil but merely scratches the surface. The land is ploughed several times before planting. Hand cultivation is the dominant practice in the perennial growing areas. And the whole process of land preparation takes a lot of time, up to six months, and is exhaustive for the farmer and is ineffective against weeds.

Harvesting is done entirely manually by using a sickle. A farmer cultivating two hectares of cereals may require more than one and half months to complete the harvesting operation. Threshing is carried out by driving cattle over the straw. The chaff is separated from the grain by winnowing the mixture by hand in the wind. The daily output of such technique is low and involves a significant loss of output due to forage by animals and damage by rain and quality deterioration due to animal and soil contamination (USAID, 1995).

Thus, in order to enhance the soil moisture retention, crop establishment, water and nutrient use efficiency, farm implements play a great role. The rate of growth in the population of Ethiopia is continuously increasing and with it grows the necessity to meet people's need for food and other materials of vegetable and animal origin. The ability to increase crop yields in present cultivated areas at a pace far ahead of population growth in the county must be the basis for scientific planning, sound economic development, and conservation of the environment inhabited by man. To feed this growing population, agriculture must be intensified by increasing the yield of the crops or by growing more than one crop per year.

Similarly, in drought areas where erratic rain fall farmers rearing animas and sub-cropping. There has been shortage of forage for rearing animals. So dwellers have migrated for pasture in Afar region in the North Eastern of the country. However, the soil type in this region has been transported from the high land of the country during the rain seasons is very fine for land preparation is similar to in this study. One of the Ethiopian agricultural development policies is rearing animals and crop production in the shortest time in this area. Land preparation must be also done in the shortest possible time for crop production. The time available for land preparation is generally very short. Small power driven rotary tillers can be used for land preparation in the shortest possible time, as the required soil tilth is obtained in a fewer number of passes under firm soil condition. The rotary tiller with" Pick"-shaped blade has been available under unsaturated soil condition because of the design of this rotary blade is different from other rotary blades, namely, a "C", an "I", a "L" and a "J"- shaped. The advantage of the rotary blade is cutting and throws the soil slices, the seed bed by the rotary blades has been uniformly tilled for optimum moisture holding capacity for the development of plant roots and uniformly plant population growth to increase crop productivity.

OBJECTIVES

General Objectives

To apply optimization theory for the development of farm mechanization under Ethiopian condition.

Specific Objectives

To optimize design parameters in terms of the total specific energy requirements of the rotary blades.

Scope of the study

To optimize design parameters in terms of the total specific energy requirements of various rotary blades, namely, for a “pick”- shaped, a “C”- shaped, an “I”- shaped a “L”- shaped and a “J”- shaped rotary blade under unsaturated sandy clay loam condition.

LITERATURE REVIEW

1. Historical background of rotary tiller

The history of the rotary cultivator has begun with the introduction of steam tractors into agriculture. When tractors with internal combustion engines had been introduced into agriculture, new conceptions as to the use of rotary machines emerged. In Germany in 1911, Siemens-Schukert Werke started to produce rotary cultivators according to the patent of Myenburg. Meyenburg's rotary cultivator provided with spring teeth became a prototype of small engine garden cultivators used until now (Figure 2). All other rotary machines designed by Siemens-Schukert, Lanz and later on by Comfrasch fell out of use some years after World War I. They were heavy, self-propelling rotary machines intended exclusively for deep tillage. The cultivating machine designed by Siemens-Schukert was driven by an electric engine, by Lanz- by internal combustion engine on wheel truck, by Comfrash-on a caterpillar truck. These self-propellering machines were not successful resulting –first of all –from their complicated and heavy structure and, besides, they were very expensive in comparison with traditional ploughs. Servicing these machines was very inconvenient since their working elements, operating at high speed at large depth, quickly become worn down and, encountering stones, deflected or cracked (Bernacki, 1972). After World war II, attention had been drawn to the rotary machines. Their task, as put before designers, was not to improve the plough but to design an efficient agricultural machine for the preparation of the soil for sowing, more intensive crushing and pulverizing soil which is better than the passive implements. After the successful introduction of rotary cultivators into agricultural practice, the attention of designers has been drawn to old patents of rotary machines designed for deep ploughing. Simultaneously new inventions were patented. A tendency, persisting till now, of reducing specific weight and increasing tractor power was decisive of all attempts to design rotary machines. Relatively lighter tractors yielded no adequate draft, and better utilization of power of these tractors could be affected only by direct power input from the power take-off (PTO) PTO shaft. Power is transmitted from the engine to the rotary blades through the PTO shaft is shown in Figure 2.

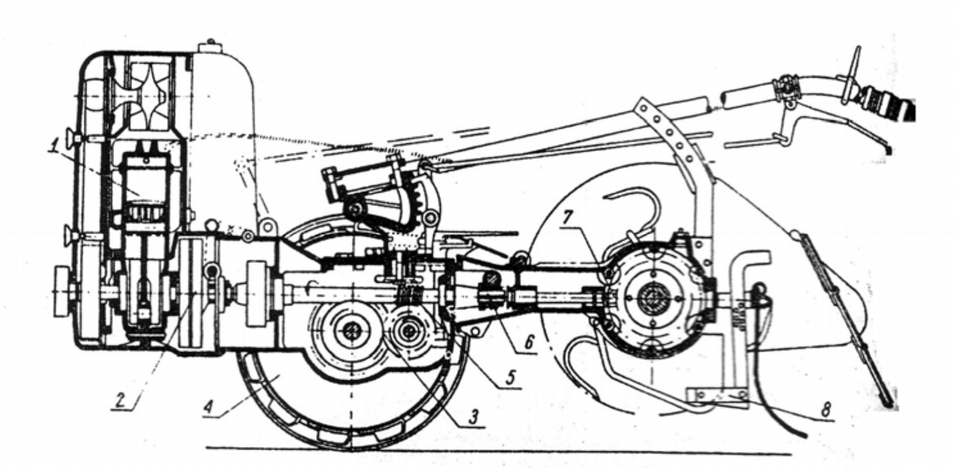


Figure 2 Meyenberg's rotary cultivator

Source: Bernacki (1972)

1- Single cylinder engine: 2- Coupling: 3 - Gear transmission: 4 - Wheels of rotary cultivator: 5 - Worm gear: 6 - Coupling of the shaft driving the drum of rotary cultivator: 7 - Bevel gear: 8 - Shaft for controlling cultivation depth.

Introduction of rotary tillers in Japan began with the import of walking type garden tractors (Utilita from U.S. and Simar from Switzerland). These machines were redesigned and tested thoroughly to meet local conditions. Locally made hand tractors began to appear in Japan since 1953. These models were developed after World War II, and introduced for light soil (Sakai, 1973a).

A rotary tiller uses a power driven rotor to replace the conventional tillage implements (Hedrick and Gill, 1971b). It can achieve advantages in terms of its lower draft requirement, better soil break-up and more efficient inversion and trash- mixing. However, the rotary tiller may be used for certain specialized operations and in certain types of soil. The high specific energy requirement of rotary tiller is one of the draw backs of the current design. Developing blades which reduce soil pulverization and power requirements are the objectives of current designers.

Rotary tillers can be used for the full preparation of soil without applying other implements. They obtain part of their energy in more than one manner. These tillage implements obtain part of their energy from a rotary source, usually the tractor PTO (Power take-off). These machines offer a great advantage in manipulating the soil by virtue of their reduced draft requirements and greater versatility. When the draft requirements are low, soil compaction is reduced as tractors can be made with less mass since at least a part of the tractor's output is utilized through non tractive means. The task to be fulfilled by rotary cultivators is shallow tillage and preparation of soil for seedling. The drum of the rotary cultivator is protected by casing from above and from the back in order to prevent the cut off soil slices from being throwing too far backward. An adjustable iron sheet screen is hinged to an iron sheet casing. Drums are placed on bearings on both ends of the shaft and are driven by means of spur gears reducing the revolutions of PTO shafts (Bernacki *et al.*, 1972).

The rotational speed of the rotor axel is usually in the range of 130 to 400 rpm with 12 to 18 rotary tines, The maximum depth of tillage was usually 13 to 15 cm in Japan. The width of the tillage is about 55 to 75 cm. Although the performance of the machine as "Rotary' is changeable under many conditions, normal one with common width and depth is 1 to 15 h / ha with a water cooled diesel engine and hours power of about 8 to 14 ps (Sakai,1973).

The power is transmitted to the rotating tillage mechanism by means of a power-take off shaft, gears, chains, shafts, and so forth. Rotary tillage width is least equal or wider than the outside width of both drive wheels. The forward speed recommended is the range of 25 to 70 cm/s for seeding and fertilizering, 40 to 50 cm/s for normal tillage condition and 50 cm/s to 1.2 m/s for paddling and harrowing (Sakai,1973). The rotary blades of a rotary tiller are equivalent to the shear and mouldboard plough because the movement of the mouldboard plough can be considered as the movement of a rotary blade with a large rotor radius.

The use of rotary cultivators in primary tillage in fieldwork is somewhat limited mainly because of the high power requirements and excessive soil pulverization. Rotary tiller is a power intensive process. Results indicate that the power consumed by a rotary tiller is greater than that by ploughs. However, the rotary soil tillers are better for loosening soil clods and weed infested soils so that power consumption during subsequent soil working is reduced (Godwin *et al*., 1984).

Rotary tillers, which may be used as the powered rotating mechanism in Asia and Europe, are popular for rice cultivation. The rotary tiller, which produces a finely harrowed seedbed, was better than the use of ploughs or disk harrows. After seedbed preparation, the field has to be peddled with sufficient water. The soil conditions have to be kept flat having uniform soft tilled soil. Tillage by rotary tiller, therefore, helps to accomplish many operations with one or two passes (Godwin *et al*., 1984).

One rotary tillage operation may be equivalent to several conventional tillage operations as far as the quality of the seedbed is concerned. Rotary tillers can replace the plough, disk and harrow. A negative draft produced by the rotary action results in a lower power requirement, and less soil compaction. The total power requirement for rotary tillers is generally higher than for conventional ploughs. In maize and spring barley production systems in combination with a chisel plough, rotary tillers have been found to have high energy requirements, but rotary tilling is more effective in saving labour compared to conventional tillage systems (Kosutic and *et al.,* 1994). To increase crop intensity, rotary tillers are more suitable for tillage operations close to the main crops. For a complete mixing of residues into the soil and faster decomposition.

2. Types of Rotary Tillers

Rotary tiller is a common name for hand tractor, two-wheeled tractor guided and support by hand. It is widely available in small holder areas where saturated and unsaturated soil condition are prevalent. Hand tractors are equipped with a rotary tiller, and hence, the name" power tiller". The usual power of this tractor ranges 2 to 12 kW (Liljedahal, 1979).

George (1983) investigated and showed the utilization of walking tractor with many mounted attachment implements and equipment including mouldboard ploughs, disc ploughs, row crop cultivators, disc harrows, spike tooth harrows, planters, drills, seeds, fertilizer, sprayers, threshers, fanning mills, generators, air compressors, rotary mowers, and other harvesting equipment.

According to the design of the drive of transmission case the rotary tillers has been classified into the following groups (Kepner *et al.*, 1965).

1. Chain drive (most modern cultivators)
2. V-belt drive (cultivators - Cutts)
3. Gear driven (tillers of modern design)
4. Worm gear drive (small cultivator cutters)
5. Hydraulic drive and electric drive (prototype small garden type cutters)

Hand tractor (walker tractor) self-propelled machines have a single axle designed primarily to pull and propel trailed or mounted agricultural implements and machinery.

Some of rotary tillers axle (chain drive) and PTO driven are shown in Figures 3 and 4. Transmission system of the PTO driven rotary tiller also is shown in Figures 5.

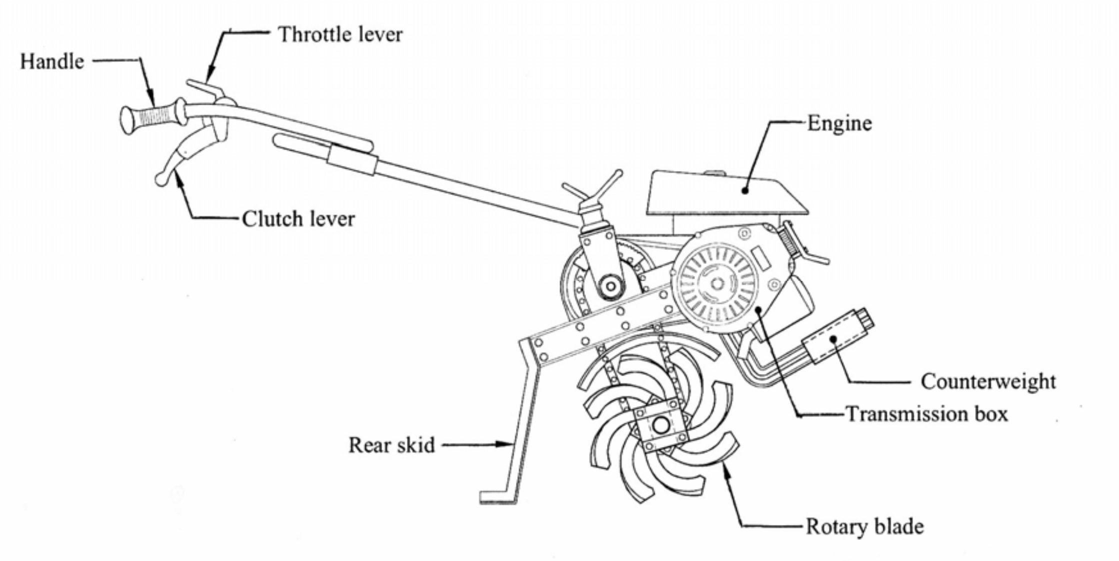


Figure 3 Axle –driven rotary tiller (chain drive).

Source: Sakai (1973 a)

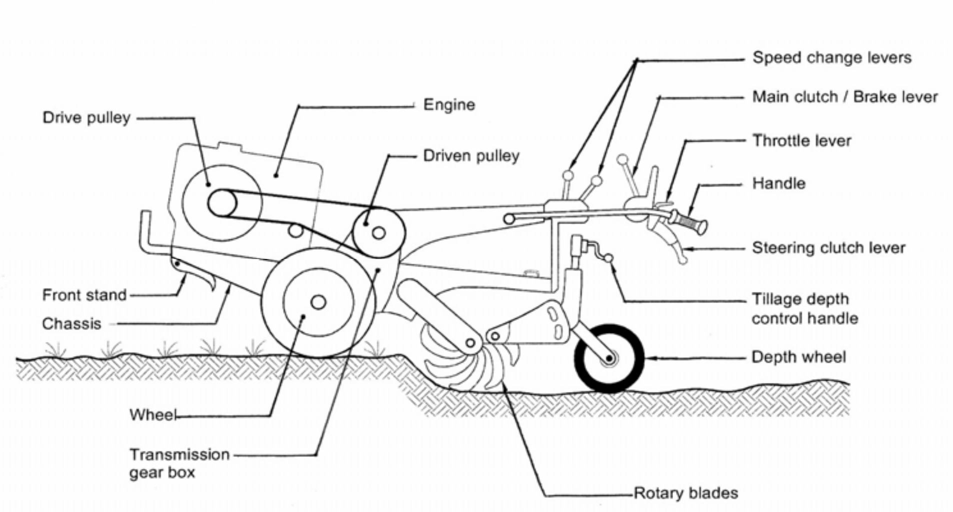


Figure 4 Typical PTO - driven rotary tiller.

Source: Sakai (1973 a)

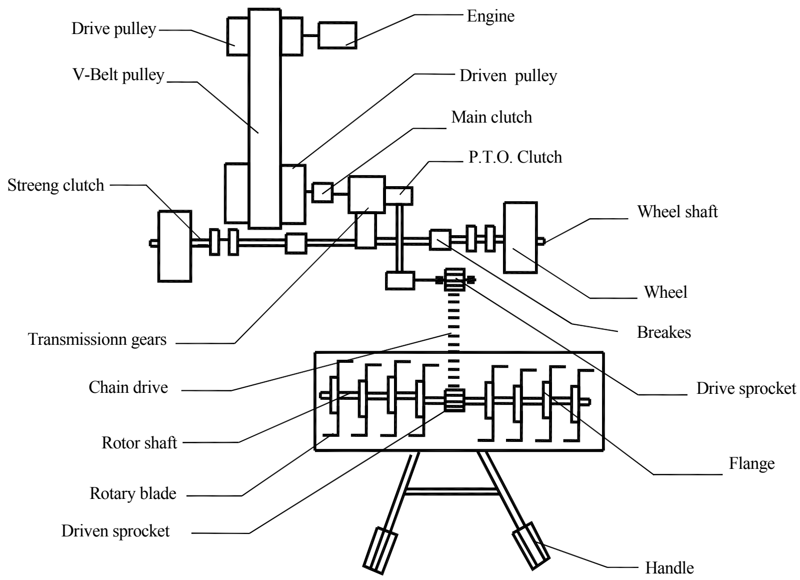


Figure 5 Transmission system of power tiller.
Source: Modified from Sakai (1973a).

According to the location of the transmission case in relation to the direction of advance, these machines are divided into the following groups:

1) With horizontal axis of rotation of the rotor drum perpendicular to the direction of advance of the tiller.
2) With a vertical axis of the rotation of the cutter drum.
3) With a cutter drum placed at an angle to the direction of advance of the machine.

There are two types of transmission systems used in the rotary tiller; one is “Centre Drive Type” and the other is “Side Drive Type” (Figure 6). Centre drive type is more convenient to adjust the tilling width than side type, but this type is appropriate to leaving some untilled portion under the centre-reduction case, especially in the case of wet and sticky fields. In such cases, the rotary tiller is equipped with additional tools like a coulter or special rotary knives to cut off under the transmission case.

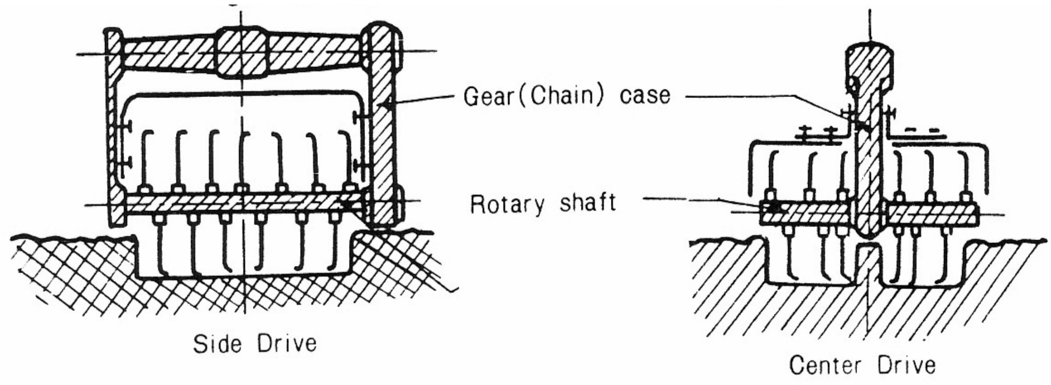


Figure 6 Types of rotary drives.
Source: Sakai (1973)

The rotary blades are divided in to two groups: European L-shaped (Figure 7) and the Japanese C-shaped rotary blades (Figure 8). The Japanese blade is generally made of spring steel (SUP6) or carbon tool steel (SK5) (Sakai and Hai, 1980) and it has good spring structure, good hardening property, and resistance to abrasive wear after heat treatment. In particular, , the spring steel has a tension strength of $125kg\ /\ mm^2$, however, it contains many angles of elements and has a difficult design.

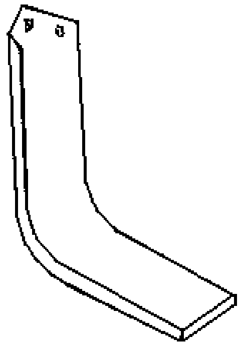


Figure 7 European L-shaped.

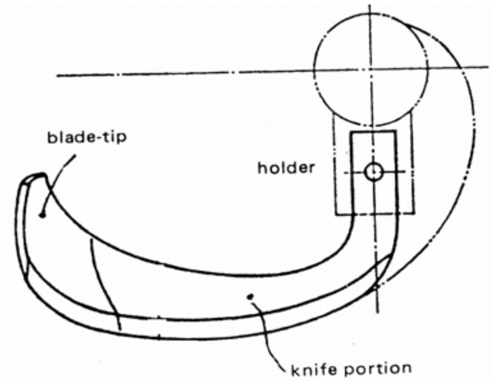
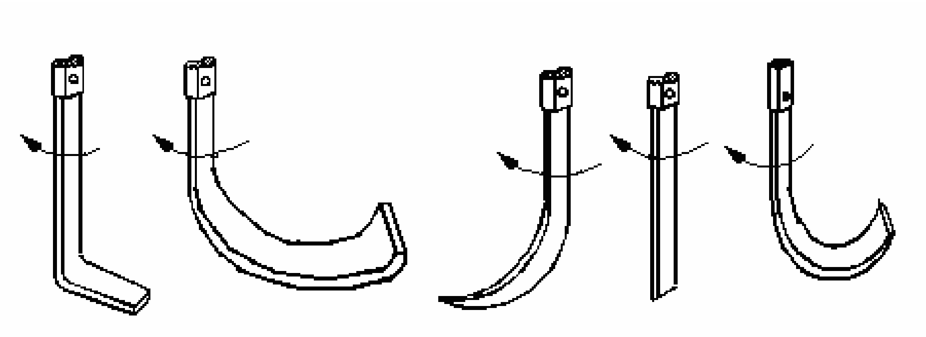


Figure 8 Japanese C-shaped rotary blade.

2. 1 Types of Rotary blades

Types of rotary blades which have been available for small tractors are shown in Figure 9.



1 2 3 4 5

Figure 9 Types of rotary blades.

1- “L”-shaped tyne: 2 - Back ward curved tip twist (“C”-shaped) blade
3- Hock (“Pick”-type) blade: 4 - Straight (“I”-shaped) blade : 5 - “J”-shaped (Scoop)

Source:- Beeny and Khoo (1970) and Kataok and Shibusawa (2002).

2. 1. 1 “L”- Shaped (Rotary hoe)

The “L”-shaped blade has a bent and curved knife-structure and the blade is not free from hooking. The cutting portion of the rotary blade has pressed the uncut soil surface because of the higher the curvature between the depth of cut and cutting width, and the lower the clearance angle, the sweep clearance angle and the radius of curvature than the “C”- shaped (Figure 11). Consequently, the power requirement of the rotary blade and the amount of soil pulverising, throwing, and mixing considerable increased.

Sakai (2000) conducted an experiment to evaluate the efficiency of rotary blades and concluded that the “L”– shaped blades were difficult to use for paddy farming because weeds and plant residues were difficult to cut on soft paddy soil, though they would be cut easily on upland hard soil. Moreover, rice stalks have much greater tensile strength than wheat stalks, so the tiller blades easily hook rice straw and plant residues, and they make the rotor turn into a drum of straw.

Sakai (1975) conducted an experiment to study the performance efficiency of rotary blades and the results of the experiment showed that “L”- shaped blades were better than the “Pick”- shaped in trashy conditions. They were also more effective in killing weeds, but they do not pulverize the soil much. 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.

2. 1. 2 “C”- Shaped (Slicer type tyne)

The “C”-shaped blade has a bent and curved knife-structure of changing thickness from the base to the tip. The blade is free from hooking problems with plant residues. It provides sliding and cutting of the fibrous soil with minimum resistance and has self - cleaning characteristics when working in damp soils. These blades also have sufficient resistance to breakage and abrasive wear. From power consumption point of view these types of tools are as good as hoe tynes. However, they are not as desirable as with regard to the pulverization and their ability to chop surface organic matter, particularly when cultivating at shallow depths.

Salokhe *et al.* (1993) observed that the C- shaped tines consumed less power than the L- shaped and the C- L shaped (combination) tines at any forward speed and pass. During their study, it was found that the L- shaped tines consumed more power than the C-shaped blades by 33, 24, and 14% at 1.0, 1.5, and 2.0 km/h forward speeds, respectively.

2. 1. 3 "Pick"- shaped (Hook - type)

Sakai (1975) investigated that the "Pick"-shaped tine which has been used for deep soil preparation in clean ground, wheat stubble and sod ground with cover crop removed, and should never be used in tall tough cover crops. This blade type is most effective for cultivating upland fields or unsaturated soils that have little grass. The point of this type enters the ground at the angle and causes it to dig continually throughout its entire contact with the untilled soil on the blade circle, much like the point of the plough. Thus the action of this blade is to dig itself in and pull the entire tilling units down. The depth control is, therefore, of greatest importance. Without it, the tendency is to go deeper and deeper and beyond the power available. This blade while excelling in this subsoil characteristic, is not as desirable from a power standpoint, and has poor chopping and winding characteristics as far as surface organic matter is concerned.

2. 1. 4 "I" – shaped (Straight type tine)

These are applicable for loosening meadow sods, and for working new lands with moderate quantities of grassy vegetation and roots to a depth of 120 to 150 mm where ordinary ploughing may cause undesirable turning of the soil layers. Soil cutting with straight knives, small zones of deformation, comparatively low power consumption for cutting, and less chance of clogging with plant residue. These tynes with small angles of attack are used to get good turning of the separated layer of soil and sufficient burial of plant residue.

2. 1. 5 "J"- shaped (Scoop type tyne)

The scoop type of rotary blade ("J"- shaped) has been used for reverse rotational rotary tiller for reducing the power requirement in deep tillage. The maximum cutting depth is 350 mm and the blade rotation is 250 mm. The rotational direction of the blade is opposite to the rotation of the machine tires and the position of the rotary axle is the ground level (Shibusawa, 1993). The blade consists of two components, one vertical and the other horizontal. The horizontal portion consists of the out side cutting edge and scoop surface. The scoop surface has an arc with a 60 mm of curvature spanning 50^0. The horizontal portion occupies the main role of soil cutting and throwing, while the vertical portion supports the horizontal portion. The rotary blade has a high backward throwing ability: provide pure soil clods with a single cut without re-tilling.

Configuration of various rotary blades are shown in Figure 10:

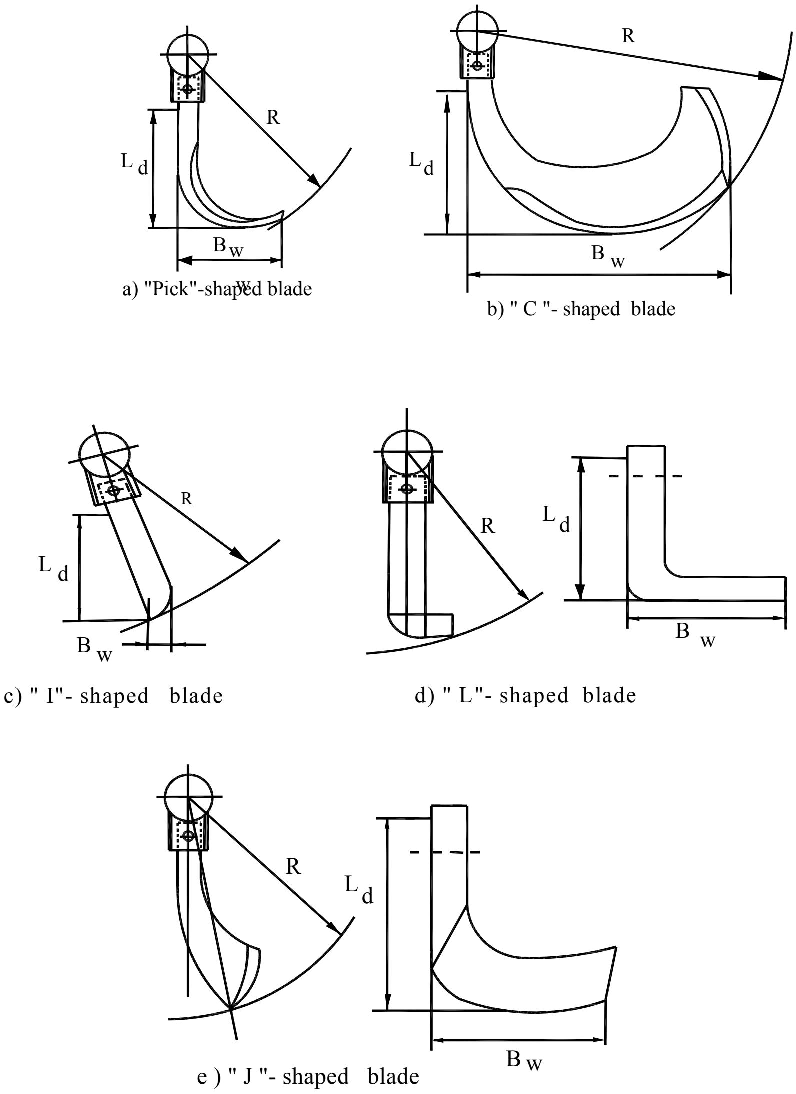


Figure 10 Configuration of rotary blades.

R, rotor radius: L_d, depth of tillage: B_W, Cutting width.

Source: - 1- “Pick”- shaped (Sarasswat, 1987).

2- “I”, “C” and “L”-shaped (Yatsuk *et al.,* 1971, Beeny and Khoo, 1970 and Sakai, 2000).

3- “J”-shaped (Kataoka and Shibusawa, 2002).

3. Rotary blade design and development

Most rotary blades have been designed and developed on the basis of experience and inventiveness.

3. 1 Rotary tiller development

Numerous researchers conducted experiment for the development of rotary tiller in different periods and soil conditions as:

Adams and Furlong (1959) evaluated the performance characteristics of hoe type, slicer type, and pick type rotary tiller tynes. The width was varied from 300 to 600 mm and depth of cut from 50 to 150 mm. The tynes were tested at varying rotor speeds from 121 to 302 rev /min and shaped rake angles from 10 to 300^{0}. The data for a typical hoe shaped tyne indicated that the rotor tip speed had a marked influence on the power requirements even though a constant volume of soil was tilled. The results therefore, suggest that with due consideration for usage such as crop row width, available horse power etc., the rotor assembly should be widened for the best utilization of power.

Dalin and Pavlov (1950) evaluated from the pick type test the reverse rotation required 12 to 16 % less energy for rotation, the total power requirement was 14 to 28 % less for forward rotation because of the horizontal component. For the testing using a C – shaped blade, the general results were the same, with a reduction in power for reverse rotation, but the total power requirement for reverse rotation was 0.5 to 11 % greater than forward rotation. Reversing the rotation of the C- shaped blade did not reduce the rotary energy as much as the pick shaped blade because the C- shaped blade does more cutting than the pick shaped blade and the advantage of reverse rotation is in increasing failure by tension.

Furlong (1956) studied the rotary and draft power requirements and soil pulverization and other factors as a function of the direction of rotor rotation, the width of the rotor, peripheral velocity, tillage pitch and the depth of tillage. In all cases except the Pick tyne, the revere rotation required more rotor power. On average, reverse rotation required 70% more power than forward rotation when operating under the conditions of tests.

Tsuchiya and Honami (1963) studied the cutting characteristics of specific tyne shapes and arrangement of tynes on the rotor using a model soil bin and half–scale tiller tynes. It was found that reducing the rotor speed reduced the power requirement, but increased the clod size. The power required dropped with a decrease in the number of tynes and with an increase in blade width (to a width of 100 mm). A relationship for the order in which tynes contact the soil was presented to equalize the load among the tynes. Wider tynes from the power point reduce torque variation two or three tynes were recommended to cut at the same time. And this resulted in a short vertical portion and smooth horizontal portion to be adopted for the construction of the blade.

Matsuo (1963) developed theoretical and experimental data comparing forward, down cut, and reverse, up-cut tiller operation using a Japanese style blade having rotor radius of 220 mm. The experiments were conducted over a depth range of 20 to 150 mm and a range of rotational velocity of 50 to 350 rev / min. It was found that the power requirements were less for reverse rotation, given the same pitch of cut, and that the reduction in power become larger as the soil strength became smaller.

Bok (1965) reported that, based on theoretical analysis, the main disadvantages of reverse rotation were the throwing of soil forward and the reversing of horizontal components on the drawbar from pushing to pulling.

Matyashine (1968) reported that at shallow tilling depths (cutting depth less than rotor radius), forward rotation required 10 to 15% less energy than reverse rotation. When tilling depth (cutting depth greater than rotor depth), reverse rotation reduced the energy requirement by 20 to 30 %.

Grinchuk and Matyashine (1968) summarized the results of a few Russian researchers and reported that in general the reverse rotation decreased the force of cutting by 1.5 times, gave better depth stability, reduced breakdowns of tools in stony soils, and made possible a wide range of peripheral to forward velocities. Among the disadvantages they associated with reverse rotation were a greater energy requirement at a depth of operation of cutting depth less than rotor radius, and the need to increase the rotary velocity to prevent throwing soil ahead of the tiller.

Butterworth (1972) designed “C” and “L” – shaped blades for deep and shallow tillage for tillers with a rotor speed of about of 180 rev/min, forward speed of 2.01 km /h and a bit length of 100 mm appeared optimum for power and cultural optimization.

Burema and Perdok (1973) studied an effort to relate the torque input to rotor shape with its cutting width, bite length, and rotor speed in sand or clay soil at different depth and moisture contents. For a hoe type shape, cutting width of 25 to 200 mm, bite length of 25 to 100 mm and rotor speed of 34 to 560 rev /min was recommended.

Mashchenski (1973) conducted several laboratory tests on saturated peat and loam soils with four narrow shaped designs at peripheral speed of 12.5 m/s. The optimum approach angle was found to be 55 to 650 and the clearance angle 15 to 200. Increasing the width of the blade from 30 to 70 mm increased cutting resistance at a faster rate than from 70 to 100 mm. Increasing width of cut reduced specific resistance up to about 80 mm, and then remained relatively constant. Straight and curved cutting edges were more efficient than cupped or toothed edges. The cutting force increased rapidly as bite length increased to 8 mm, then remained relatively constant. Specific energy had an inverse relationship with bite length.

Shibusawa and Kawamura (1985) studied the effects of scoop surface configuration and tilling conditions on torque and reaction force on the blade. The circular scoop surface is defined by the radius of curvature of the arc, the central angle phi subtending the arc and the angle phi 1 between the lines through the blade tip, the centre of rotation and the centre of curvature of 6 the arc. In the case of up-cut rotary tilling, tilling torque and tilling reaction force increased in direct relation to rho and phi , and showed minimum values at phi 1 = 40 deg. Up-cut rotary tilling produced less torque and reaction force than down-cut rotary tilling. Improved backward throwing of soil clods and reduced tilling resistance were simultaneously achieved in up-cut rotary tilling by using a blade having a scoop surface with smaller rho for phi = 50 deg and phi 1 = 40 deg and/or by increasing rotational speed and reducing tilling pitch.

Shibusawa and Kawamura (1985) studied rotary blades in up-cut tilling for deep tillage. Results of their trials showed that the test blades in up-cut rotary tillage threw more soil clods backwards at various tillage depths and achieved a considerable reduction in the clods re-tilled in the rotating zone of the blades. The trial blades reduced tillage power requirement by 40 - 60% in deep tillage compared to curved blades for both up-cut rotary tillage and down-cut rotary tillage. The smaller curvature radius of the scoop surface of the trial blades resulted in improved backward throwing of clods and less power requirement for deep tillage.

Walton and Warboys (1986) conducted experiments to determine the factors affecting the power requirements of the way double digger including the variation of the rotor speed between 94 and 242 rev/min and forward speed between 1.8 and 5.1 km/h. There have been used 16 and 24 straight –shank tynes and 12 pick type tynes. The experiments showed that there was an increase in take off power with an increase in either rotor or forward speed. Draft power increased with forward speed, the total power requirements increased with both an increase in forward and rotor speed. However, the total specific energy requirements increased with an increase in rotor speed but fell with an increase in forward speed. The straight –shank tynes used up to 50% more power tynes.

Chakkaphak (1988) showed that, in practice, the power tiller is the most commonly used for land preparation and for transportation of agricultural inputs or outputs in Thailand. These locally made units are steadily gaining popularity, mainly due to their low initial cost, low maintenance cost, efficient utilization of energy, the availability of spare parts and materials, the ease of operation and manoeuvrability.

Thai farmers generally use mouldboard ploughs, disc ploughs, and rakes with power tillerd. Disc ploughs with power tiller are being used by farmers for dry land preparation in Thailand and mouldboard ploughs, rake, and peddler are used for wet rice cultivation in low land areas. In practice, the locally manufactured power tillers are simpler versions of the imported ones, but do not have PTO shat for driving the rotary cultivators. A locally assembled imported diesel engine is being fitted on the locally manufactured power tiller farm. They have steering clutches and gears for power train. The operating speed can be adjusted by positioning the acceleration lever and shifting the gears lever. Engagement or displacement of the power to be transmitted is done with the help of an idler pulley. They are steadily gaining popularity mainly due to low initial cost, efficient utilization of energy, the availability of spare party and materials, the easy of operation, and manoeuvrability.

Sakai et al. (1990) designed analytical equations of the performance of a rotary cultivator which depend on the optimum design of the rotary shaft. Evaluation theories of the working characteristics of the rotary shaft are analyzed theoretically and empirically, and the optimum design theory for rotary blade arrangement is established by using computer aided design (CAD). Analytical equations for the edge-curve of a rotary blade are derived and the relative kinematics relationship between rotary blades and soil surface is investigated. The relationship between velocity rate, cultivation depth and cutting angle of the blade is described analytically, since these parameters relate to unstable cultivation by a rotary cultivator.

Similarly, in Thailand the development of rotary tillers has started since 1910, as has the import of riding tractors as well as farm implements from advanced countries (Surin, 1992).

Salokhe *et al.* (1993) investigated the effect of blade type on the power requirements of a tractor drawn rotary cultivator during puddling in Bangkok clay-soil. The “L”-shaped blade attachment required 33%, 24% and 14% while the “L” and “C” combination blade attachments required 14%, 12% and 4% higher power than the “C”-shaped blade attachment during pass one at 1 km/h, 1.5 km/h and 2.0 km/h forward speed, respectively. The power consumed decreased at higher passes for all the blades.

The Japanese rotary tiller designs evolved from rotary tillers used for garden work in the western world during the 1920s. The Japanese adapted them for rice mechanization. The two wheel tractors or walking tractors and their attachments for upland and low land paddy field have been developed since 1960. At the beginning, the walking tractor called “iron buffalo” had a very simple power transmission system with a chain and sprocket and no clutches (Roy, 1994). All of its parts are being produced by domestic manufacturers. Most agricultural machinery manufacturers are private enterprises having factories, garage, and machine shops (Roy, 1994).

]

Niyamapa *et al.* (1994) determined the optimum parameters for the design of a rotary cultivator in a laboratory study in a soil bin with clay soil at a moisture content of 23.26% (dry basis) and a dry bulk density of 1.29 g / cm ^{2}. Experiments were con- ducted at working depths of 12 and 18 cm, rotor speeds of 140, 160, 180, 200, and 220 rpm and forward speeds of 0.16, 0.23, 0.35, 0.64, and 1.25 m/s. The width of cut was 55 cm. From the speed and rotor torque data, power consumption was calculated. The power requirement for cutting and throwing the soil increased with an increase in rotor speed, forward speed, and tilling depth at a given width of tiller. These three parameters also affected the soil breakage. Larger clod sizes were found when tillage depth and forward speed were high and the rotor speed was low. Smaller clod sizes were found when tillage depth and forward speed were low and the rotor speed was high. The optimum parameters for the design of a rotary cultivator were found to be a tillage depth of 18 cm at a forward speed of 0.35 m/s and rotor speed range of 165 - 220 rpm. At these operating conditions power consumption was 2.70 - 3.50 kW.

Niyamapa and Rangdag (1998) conducted an experiment to develop 2 models of rotary tillers for locally made walking tractors in the field conditions. The first model consisted of 14 C– Shaped blades. The second one had 18 C-shaped blades. The designed rotary tiller was operated in the field to evaluate the field performance in terms of field capacity and field efficiency. The developed rotary tillers gave satisfactory results. Based on walking test data, the rotary tiller model of 14 blades was selected to be constructed to be attached to the locally made walking tractors. the two models were tested in the field to evaluate the field capacity and field efficiency. The field tests of the first model for tilling at a width of 0.51 m, tilling depth of 6.8 cm and at a for ward speed of 0.52 m/s gave an average field capacity of 0.093 ha / h. The theoretical field capacity was 0.119 ha / h and thus the field efficiency was 78.7%. The test result of the second model showed that at a width of 0.63m, tilling depth of 7.03 cm and at a forward speed of 0.55 m/s gave an average field capacity of 0.11 ha / h and the field efficiency of 82.1%. Finally, the rotary tiller model of 14 blades was selected to be developed.

Salokhe and Ramlingam (2003) conducted experiments to evaluate the performance of a rotary tiller equipped with reverse or conventional blades. The conventional rotary tiller was equipped with reverse or conventional blades The conventional rotary tiller was equipped with C-type blades whereas the reverse-rotary tiller had new types of blades. Tests were conducted on wet land as well as in dry land. Tests were conducted at tractor forward speeds of 1.0, 1.5 and 2.0 km/h. A power-take-off (PTO) power consumption was calculated from the PTO torque and speed. The results indicated that the PTO power consumption was less for the reverse-rotary tiller compared to the conventional tiller for all passes and forward speeds. For both rotary tillers, power consumption decreased as the number of passes increased, whereas power consumption increased when the forward speed was increased. At all forward speeds, the power consumption was the highest during the first pass and lowest during the third pass. The maximum difference of PTO power requirement was after the first pass at 1.0 km/h forward speed. Therefore the reverse-rotary tiller consumed about 34% less PTO power under this condition.

3. 2 Rotary blade design

The rotary cultivator has been considered to be the most important tool for land preparation. The advantage of the rotary tiller fine degree of pulverization and intimate mixing of soil and water in unsaturated and saturated soil condition. Basic rotary blade design has been given by Beeny and khoo (1970). Figure 11 shows basic rotary blade design parameters includes, depth of soil cut, width of soil cut, blade width, radius of curvature, the thickness and sharpness of the rotary blade, the sweep back angle and the clearance angle.

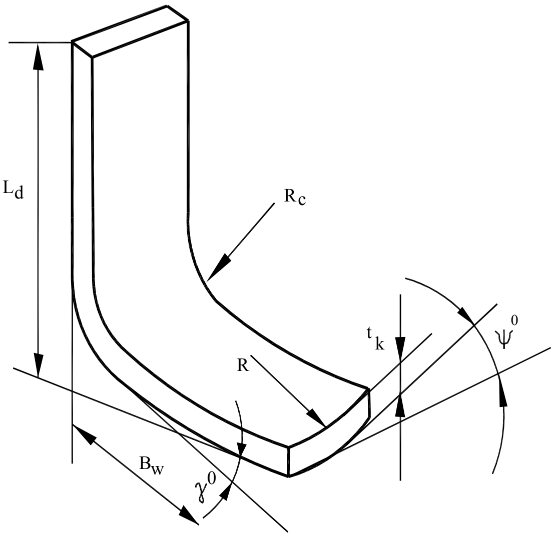


Figure 11 Diagram for design parameters of the rotary blade

L_d , depth of soil cut, B_W , width of soil cut, W , blade width, R_C , radius of curvature,
t_K , the thickness and sharpness of the rotary blade, γ^0 , the sweep back angle, ψ^0 , the clearance angle

3. 2. 1 Motion characteristics and design principles of rotary blades and rotary shafts

Design principles and equation of rotary blades and rotary shafts has been formulated their relative motions and actual operations between soil and these components. The equations of motion have been applied to study rotary blade. The trajectory of motion of the rotary blade has determined in order to select the rotor radius of the rotary blade, the number of blades on a disc Z, and the angular and translatory velocities ω and V_f respectively of the drum. Diagram for the determining the trajectory of motion of rotary tiller blades has been given by Kawamura (1999) in Figure 12 and Sakai (2000) for "L' and "C"-shaped blades in Figure 13.

3. 2. 1. 1 Motion equations of a rotary blade

The locus curve parametric equations of trochod motion for any point on a rotating radius R is given as (Figure 12):

:

$$X = V_f t - R\sin \omega t \qquad (1)$$

$$Y = \left(\frac{V_f}{\omega}\right) - R\cos \omega t \qquad (2)$$

Where

X = horizontal component force, N,

Y = vertical component force, N,

V_f = travel speed of the machine, m/s,

t = time , s,

ω = the angular velocity of the rotary blade, rad/s,

R = rotation radius of the point, m,

The pitch of the rotary blade has been given as:

$$L_B = \frac{60V_f}{NZ} \qquad (3)$$

Where

L_B = the pitch of the rotary blade, m,

N = rotational speed of the tillage axle, rpm,

V_f = travel speed of the machine, m/s,

Z = number of blades in one rotational plane,

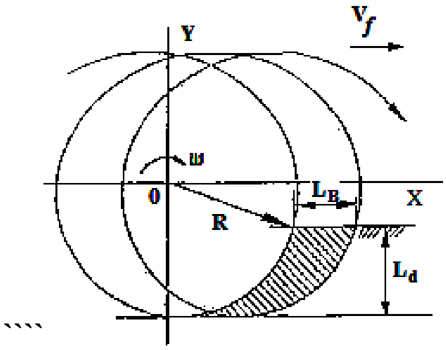


Figure 12 General trochoid motions of a rotary blade.

Source: - Kawamura (1999).

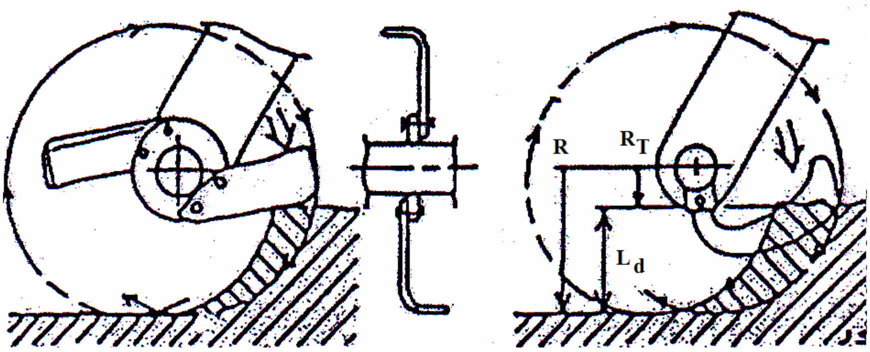


Figure 13 Trochiod motion by L -shaped and C-shaped rotary blade.

Source:- Sakai (2000).

The maximum depth of tillage by the rotary blade is given by (Figure 13)

$$L_d = R - R_T \tag{4}$$

3. 2. 1. 1. 1 Power requirement by rotary blade

Operational power requirement during the process soil disturbance by a rotary tiller is given by

$$P_T = \frac{\left(2\pi \times \mathrm{N} \times \mathrm{T_{SP}} \times B_W \times L_d\right)}{60} \qquad (5)$$

Where

P_T = operational power requirement, kW,

N = rotational speed of the tillage axle, rpm,

T_{SP} = specific torque ,N m,

B_W = cutting width of the rotary blade, m,

L_d = cutting depth of the rotary blade, m,

3. 2. 1. 2 Analysis of action and reaction forces of rotary tiller

Two kinds of external force acting from the soil act on the rotary tiller attached to the two –wheel tractor includes, rotary tillage resistance force acting on many tiller blades and reaction forces of dynamic loads acting on the rear wheel. Rotary tillage resistance force acting on many tiller blades expressed as:

1. External forces acting on the Blade tip are total radial suction force (centrifugal force) and the total torque resistance force.
2. External forces acting on longitudinal portion are centripetal force and torque resistance force.

3. 2. 1. 2. 1 Total tillage resistance force

Total tillage resistance force has been occurred during the process of rotary tillage operations. Figure 14, 15 and show he radial suction force on the scoop surface, and the centripetal force on the straight blade, Right side view of forces expected to be resultant forces and total tillage resistance force T_3 (Figure 17) its two component forces R_3 and P_3 in vertical and horizontal directions. Total tillage resistance force consists of four kinds of external forces acting on the rotary tiller blade. The radial suction force and the torque resistance force, acting every moment on the scoop surface, centripetal force and the torque resistance force on the straight blade in the soil which have been given by Eqns (6), (7), (8), and (9).

a) Total radial suction force

$$\vec{S} = \overrightarrow{\sum \Delta S} \tag{6}$$

b) Total torque resistance force

$$\vec{\tau} = \overrightarrow{\sum \Delta T_1} + \overrightarrow{\sum \Delta T_2} = \vec{T_1} + \vec{T_2} \tag{7}$$

c) Total centripetal force.

$$\vec{E} = \overrightarrow{\sum \Delta e} \tag{8}$$

c) Total tillage resistance force.

$$\vec{T_3} = \vec{S} + \vec{E} + \vec{\tau} \tag{9}$$

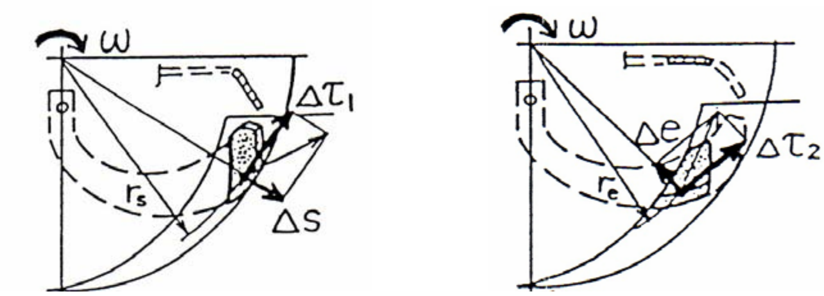
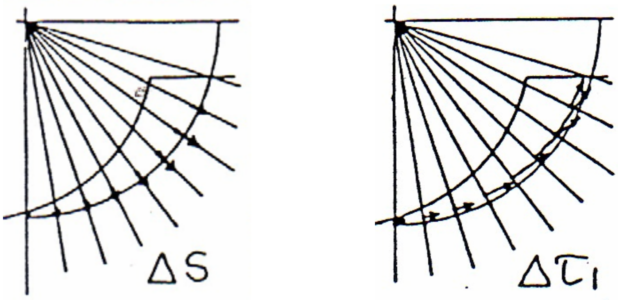
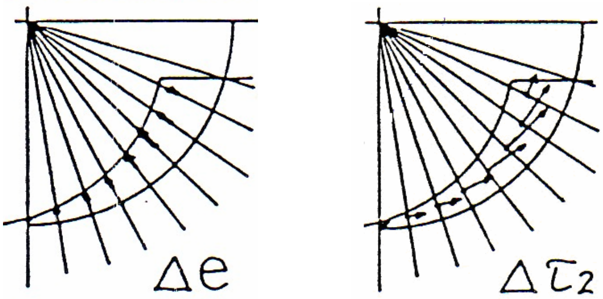


Figure 14 The radial suction force on the scoop surface, and the centripetal force on the straight blade.

Source: - Sakai (2000).



a) External Forces acting on blade –tip



b) External Forces acting on longitudinal portion

Figure 15 Right side view of forces expected to be resultant forces.

Source: - Sakai (2000).

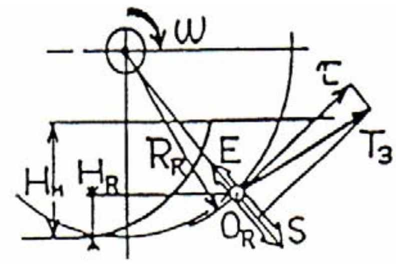


Figure 16 The total tillage resultant force T_3 which consists of S,E and τ .
Source: - Sakai (2000).

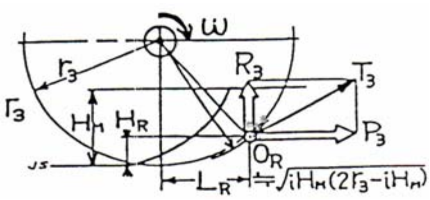


Figure 17 T_3 and two component forces R_3 and P_3 in vertical and horizontal directions
Source: - Sakai (2000).

Dynamic principles of the rotary rower tiller for stable Tillage

In order to carry out stable tillage, external forces acting on the rotary power tiller satisfy three balancing conditions as the following:

1. Vertical balance of the machine is required to be:

$\sum$ (Downward external forces) = $\sum$ (upward external forces) (10)

2. Horizontal balance of the machine is required to be:

$\sum$ Forward external forces) = $\sum$ Backward external forces (11)

3. Moment balance about the point' O' under the wheel axle centre is required to be:

$$\sum \text{(Counter clockwise moments)} \quad = \quad \sum \text{(clockwise moments)} \qquad (12)$$

External forces acting on the power tiller are:

Forward external force (P_1) at the rear wheel to the direction of the machine and (P_2) tillage and press forward force at the rotary tillage blades, reaction vertical forces R_1 and R_2 and the total weight of the power tiller is shown in Figure 18.

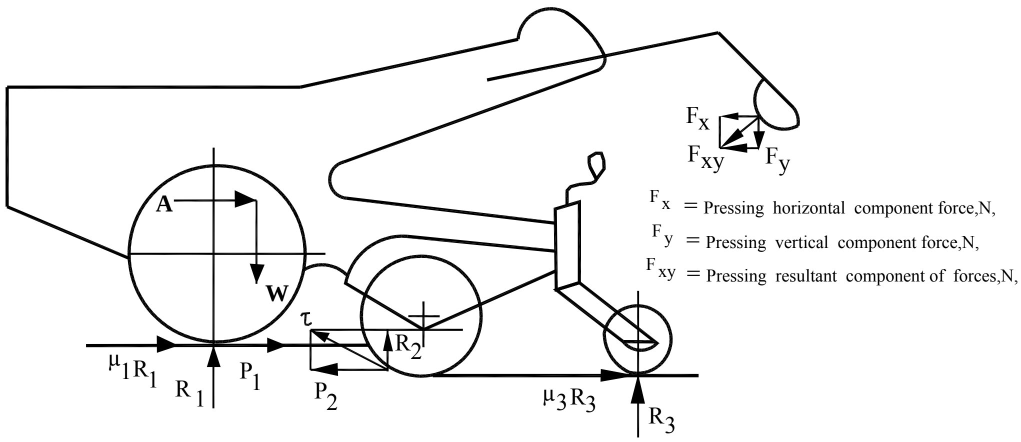


Figure 18 Free body diagram of force analysis of a rotary tiller.
Source: - Sakai (1983).

Where:

W = total weight of the whole machine at work, *kg,*

A = Acceleration resistance, *N,*

$= m(1+e_t)dv/dt$

m = total machine weight, *kg,*

e_t = a correction coefficient for the rotating mass in the machine to evaluate the equivalent mass, weight of the machine,

R_1 = upward dynamic load reaction force to the dynamic load 'W' in the down ward direction, *N*,

R_2 = tillage lifting resistance force, *N*,

R_3 = Net dynamic load reaction that is the dynamic load reaction acting on a wheel with out the lift resistance (N) (the reaction force of a dynamic load in ASAE terminology)

P_2 = tillage and press Forward force, *N*,

τ = total tillage resistance force, *N*,

P_1 = forward external force, N,

μ_1 = Coefficient of friction between wheel of the tractor and the tilling soil(motion resistance ration).

μ_3 = Coefficient of friction between wheel of depth control and the tilling soil.

$\mu_1 R_1$ = backward external force acting the rear wheel.

$\mu_3 R_3$ = backward external force acting the adjustable r wheel.

Action of Acceleration Resistance Force (A):

When the machine is starting, the position action of the acceleration resistance force due to horizontal direction and moment at the point "o" is given as (Figure 19):

$$A = Wa / g \qquad (13)$$

Where:

A = Acceleration resistance , *(kgf)* .

W = the total weight of the machine, *(kgf)*.

a = acceleration due to horizontal direction *m/sc.*^{2}

g = gravitational acceleration *m/sc.*^{2}

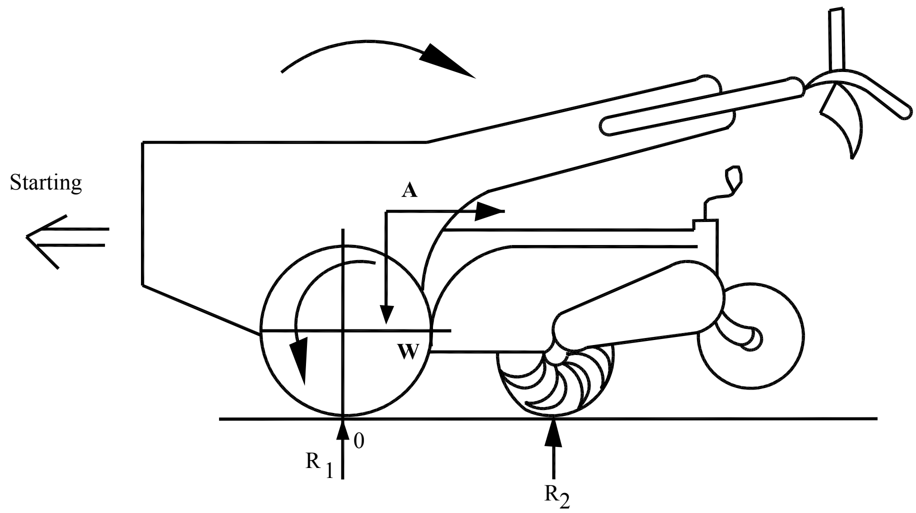


Figure 19 Acceleration resistance force at the starting condition of rotary tillage work.

Source: - Sakai (1983).

As shown in the diagram of Figure 19, the moment of the machine around point"0" : (Anticlockwise direction) at R_1 has more magnitude than moment at R_3 so that makes the rotary blade rotated.

In the free body diagram as shown in Figure 20. Balance forces are:

1. Acceleration resistance force.
2. Upward reaction force.
3. Moment about point "0" anticlock –wise direction.

Horizontal action and reaction forces are shown as followed:

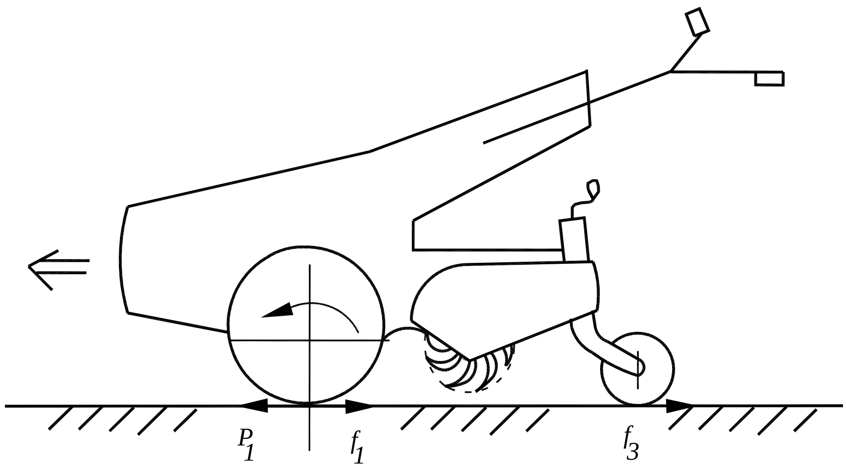


Figure 20 Dynamic reaction and action forces due to horizontal direction.
Source:- Sakai (1983).

Horizontal balance of the machine is required to be:

$\sum$ (Forward external forces) = $\sum$ (Backward external forces)

$$P_1 = f_1 + f_3 \tag{14}$$

Where

P_1 = forward horizontal force, *N*,

f_1 = friction force acting between the soil and the rear wheel, *N*,

f_3 = friction force acting between the soil and adjustable wheel, *N*.

Figure 21 shows when the rotary tiller is under minimum soil tillage condition

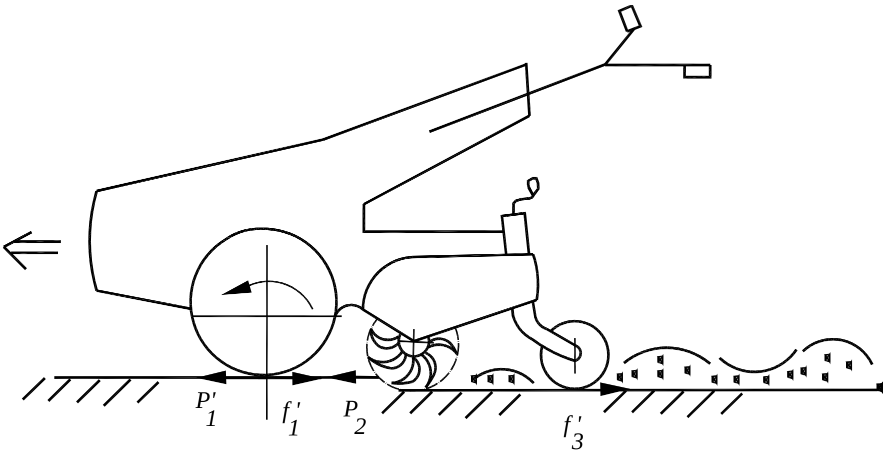


Figure 21 Dynamic reaction forces due to horizontal position udder minimum soil tillage condition.

Source: - Sakai: (1983).

Horizontal balance of the machine is required to be:

$$\sum \text{(Forward external forces)} \quad = \quad \sum \text{(Backward external forces)}$$

Where

f_1' = friction force acting between the soil and the rear wheel, N,

f_3' = friction force acting between the soil and adjustable wheel, N,

P_1' = reaction force at acting on the front wheel of the machine, N,

P_2 = tillage and press forward force, N.

Figure 22 shows that the power tiller is in static position, static vertical forces (action and reaction forces).

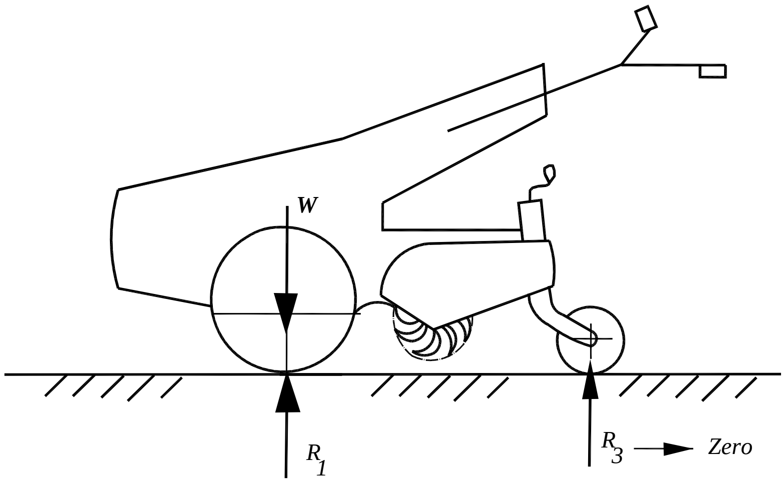


Figure 22 Diagram showing when the power tiller is in static position.
Source: - Sakai: (1983).

Vertical balance of the machine is required to be:

$$\sum \text{(downward external forces)} = \sum \text{(upward external forces)}$$

$$W = R_1 + R_3 \tag{16}$$

Where:

R_1 = static upward reaction force acting on the wheel axle, *N*,

R_3 = reaction force acting on the adjustable wheel, *N*,

W = total weight of the machine, *N*.

Reaction force R_3 becomes near to zero, then $W = R_1$

Figure 23 shows that dynamic upward reaction forces under minimum soil tillage.

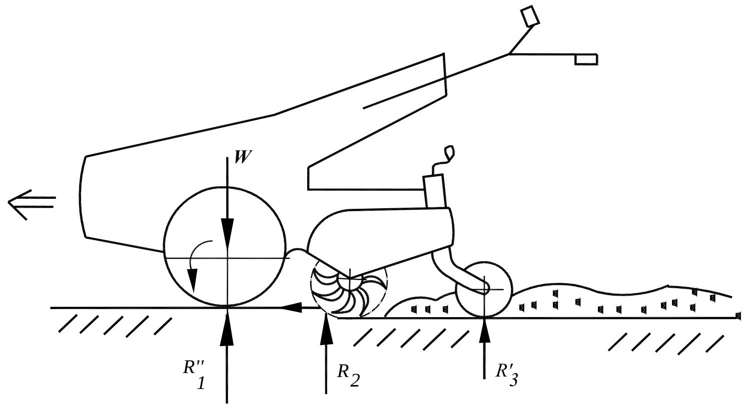


Figure 23 Dynamic upward reaction forces under minimum soil tillage condition.
Source: - Sakai: (1983).

Vertical balance of the machine is required to be:

$\sum$ (downward external forces) = $\sum$ (upward external forces)

$$W = R_2'' + R_2 + R'_3 \quad (17)$$

Where

W = total the weight of the machine, kg,

R_2'' = dynamic upward reaction force acting on the real wheel, N,

R_2 = lifting force (dynamic upward reaction force acting on the rotary blade, N,

R'_3 = dynamic upward reaction force acting on adjustable wheel, N.

Figure 24 shows when the rotary tiller is under maximum tillage condition

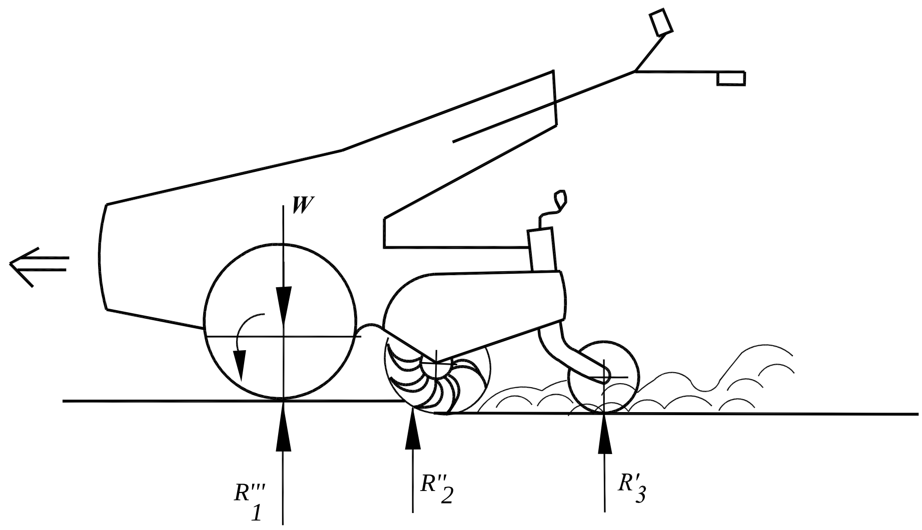


Figure 24 Dynamic upward reaction forces under maximum soil tillage condition.
Source: - Sakai: (1983).

Vertical balance of the machine is required to be:

$\sum$ (downward external forces) = $\sum$ (upward external forces)

$$W = R_2''' + R''_2 \tag{18}$$

Where

W = total the weight of the machine, *kg,*

R_2''' = dynamic upward reaction force acting on the real wheel, *N,*

R''_2 = dynamic upward reaction force acting on the rotary blade, *N.*

Figure 25 shows when the power tiller is under balancing conditions

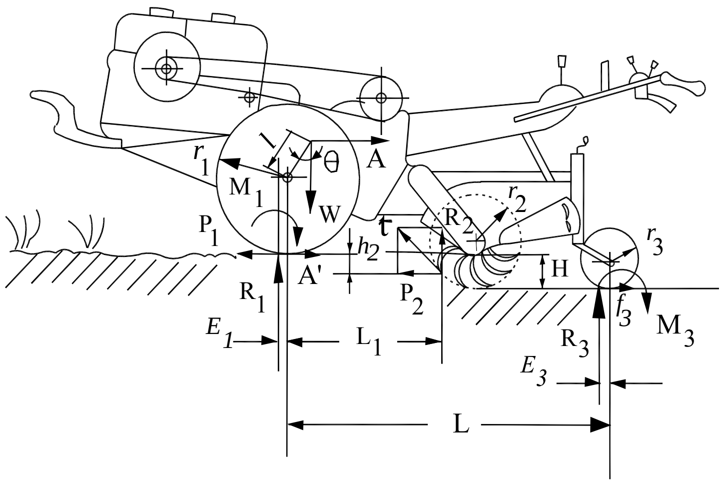


Figure 25 External forces acting on a rotary power tiller on a dry hard field
Source: - Sakai: (1983).

As shown in Figure 26 all external forces acting on the rotary power tiller satisfying the following balancing condition s as a free body on the earth, in order to achieve stable tillage.

1. Vertical balance of the machine is required to be:

$\sum$ (downward external forces) = $\sum$ (upward external forces)

$$W = R_1 + R_2 + R_3 \tag{19}$$

2. Horizontal balance of the machine is required to be:

$\sum$ (Forward external forces) = $\sum$ (Backward external forces)

$$P_2 = P_1 + A + f_3 \quad (20)$$

3. Moment balance about the point '0' under the wheel axle centre is required to be:

$\sum$ (counter-clockwise moments) = $\sum$ (clockwise moments)

$$R_2L_1 + R_3\ (L - E_3) + f_3H = WL\sin\theta + Ah_w + R_1E_1 + P_2h_2 \quad (21)$$

Where

W = total weight of the whole machine at work, *N,*

A = acceleration resistance, *kg,*

R_1 = upward dynamic load reaction force to the dynamic load 'W' in the down ward direction, *N,*

R_2 = tillage lifting resistance, *N,*

R_3 = lift resistance, *N,*

P_2 = tillage trust force, *kgf,*

f_3 and P_1 = motion resistance, kg,

τ = total tillage resistance force, *N,*

A = frictional reaction of the lug wheels on the soil, *kg,*

L_1 = distance between the center of gravity and the wheel center, *cm,*

h_w = height of the center of gravity of the from the field surface (cm)

$h_w = r_1 + l\cos\theta$.

θ = location angle of the center of gravity from the vertical line on the wheel center, degree,

$\theta = \tan^{-1} H / (r_2 + L_1)$

H = depth of tillage, cm,

L_1 = horizontal distance between the wheel center and the tillage lifting resistance,cm,

L = horizontal distance between the wheel center rand wheel depth control center, cm,

r_1 = effective radius of the drive wheels, cm,

r_2 = radius of the rotary blade, cm,

r_3 = radius of wheel of depth control, cm,

M_1 = Moment of force of wheel of tractor , $M_1 = f_1 r_1$

h_2 = the height from tilling and press Forward Force to the ground level

$h_2 = (2/3)$ H

$$R_2 = \frac{WL\sin\theta + Ah_w + M_1 + M_3 + P_2h_2 + LR_3 + f_3H}{L_1} \qquad (22)$$

3. 2. 1. 2. 1. 1 Downward pressing force

Downward pressing forces are one of rotary blade design criteria which has been occurred during the process of rotary tillage operations. The tillage lifting force has been expressed in terms of the total pressing downward forces given by Eqns (23) to (27).

a) Pressing downward force

The down ward force has been occurring due to the centre of gravity of the rotating machine is given by:

$$R_a = \frac{WL\sin\theta}{L_1} \qquad (23)$$

b) Pressing downward force

The down ward force has been occurring due to acceleration resistance is given by:

$$R_b = \frac{Ah_w}{L_1} \tag{24}$$

c) Pressing downward force

The down ward force has been occurring due to moment force of the wheel of the tractor and the wheel of the depth control is given by:

$$R_c = \frac{M_1 + M_2}{L_1} \tag{25}$$

d) Pressing downward force

The down ward force has been occurring due to tillage trust force is given by:

$$R_d = \frac{P_2 h_2}{L_1} \tag{26}$$

d) Pressing downward force

The down ward force has been occurring due to lifting resistance force is given by:

$$R_e = \frac{-R_3 L + f_3 H}{L_1} \tag{27}$$

Tillage lifting force has been also given as:

$$R_2 = \frac{71620 \times N_e \times \eta_R \times C_L}{\left(C_R \times R \times N\right)} \qquad (28)$$

Where

R = tillage lifting resistance force, N,

N_e = maximum engine horse power, hp,

η_R = power transmission efficiency, η_R = 0.05,

C_L = coefficient of tillage lifting force, C_L = 1.0,

C_R = radius of coefficient, C_R = 1.0,

R = radius of the rotary blade, m,

N = revolution speed of the rotary blade in design, rpm.

3. 2. 2 Total tillage tool force requirement

During a tillage operation, force required by the tillage tool was formulated by Reece (1965), a mathematical equation of the total force model is exhibited as:

$$P = \left(\gamma g d^2 N_r + cdN_c + c_a dN_a + qdN_q\right) w \qquad (29)$$

Where

P = Total tillage tool force requirement, N

γ = Total soil density, N m^{-3}

g = Acceleration due to gravity, 9.81 m s^{-2}

d = Total working depth below the soil surface, m

c = Soil cohesion, N m^{-2}

q = Surcharge pressure vertically acting on the soil surface, N m^{-2}

a c = Soil-tool adhesion, N m^{-2}

$w =$ Tool width, m

$N, N_c, N_q =$ Factors depend on soil frictional strength, soil geometry, and tool to soil Strength properties

3. 2. 2. 1 Soil resistance force

Soil resistance force (R_T) encountered by the rotating working part when cutting the soil slice and acting on the knife of a rotary blade at angle of (ψ) (Figure 26) and is exhibited by Bernacki *et al.* (1972) as:

$$R_T = \frac{F_T}{R} \tag{30}$$

Where:

R_T = soil resistance force, N,

F_T = torque on rotor shaft, N.m,

R = arm of the force R_T.

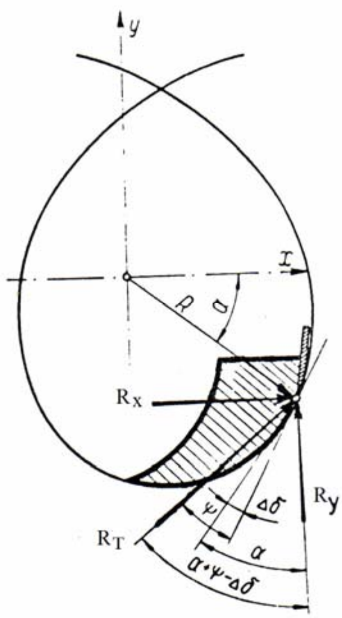


Figure 26 Reaction force components acting on a rotary blade.
Source: - Bernacki *et. al.* (1972).

Components of the soil resistance force amounts to,

$$R_X = R_T \sin(\alpha + \psi - \Delta\delta) \tag{31}$$

$$R_Y = R_T \cos(\alpha + \psi - \Delta\delta) \tag{32}$$

Where:

$\Delta\delta$ = increment angle.

α = angle of rotation of the shaft.

ψ = angle of application of soil cutting resistance.

$\psi - \Delta\delta$ = angle that the force R_T makes with the tangent to the path of blade tip (Trochoidal).

Using equation (32) , R_T is given as:

$$R_T = \frac{F_T}{(\alpha + \psi - \Delta\delta)} \tag{33}$$

The cutting resistance R_T and the peripheral force F_P are acting on the constant arm R. Subsequently, the peripheral force, necessary, for cutting off a soil slices at a peripheral speed equal to zero and is computed as the following:

$$F_P = \frac{F_T}{R} \tag{34}$$

Where:

F_P = Peripheral force. N,

R = rotor radius.

F_T = torque on the rotor shaft, N.m.

3. 2. 3 Equation of rotary tiller power requirement

Hendrick and Gill (1971) presented a general equation of a rotary tiller power requirement model (originally by Dalin and Pavlov,1950) as:

$$P_{Total} = P_{Cut} + P_{Throw} + P_{Pull} + P_{Trans} + P_{Roll} \tag{35}$$

Where

P_{Total} = Total power requirement,

P_{cut} = Cutting power requirement, W,

P_{Throw} = Throwing power requirement, W,

P_{Pll} = Pulling power exerted by the operator, W,

$P_{Roll} =$ Rolling power requirement, W.

$P_{Tract} =$ Tractive power for overcoming the resistance of soil reaction force, W.

The rotational power requirement $\left(P_R\right)$ has been defied as the torque and rotational velocity of the rotary machine. The rotary power is given by Barger *et al.* (1963) as:

$$P_R = 2\pi N F_T / 60000 \tag{36}$$

Where:

$P_R =$ Rotary power, kW,

$N =$ Rotational velocity, rev/min,

$F_T =$ Torque requirement, Nm.

Despite the negative draft, better soil break up and more efficient inversion and trash mixing, rotary tillers (the attachments of the power tiller) are the most power and energy intensive among other tillage implements. These is because of their design stricture, characteristics of cut the untilled soil and pulverize the soil clods to give optimum soil pores for better soil moisture and nutrient holding capacity for the desire of plant roots. The power processors includes the pushing power exerted by the operator behind the power tiller, the cutting and loosing the soil slice, the overcoming soil-metal friction between the soil and the knife of the rotary blade and the throwing the cut soil slice by the centrifugal action of the rotary blade. The measurement of the total power required has been expressed in terms of total specific energy required per unit volume of soil tilled. Each sub- power requirement considerably is influencing fuel consumption requirements of the power tiller.

3. 2. 3. 1 Pushing power requirement

The pushing power of a power tillers has been exerted by the operator behind the machine. In the process of pushing the machine, pushing power is required which is given by Sineokov (1977):

$$P_{Push} = R_X V_f \tag{37}$$

Where:

P_{Push} = Pushing power exerted by the operator, W,

R_X = Horizontal component of forces , m,

V_f = Forward velocity of the machine, m/s.

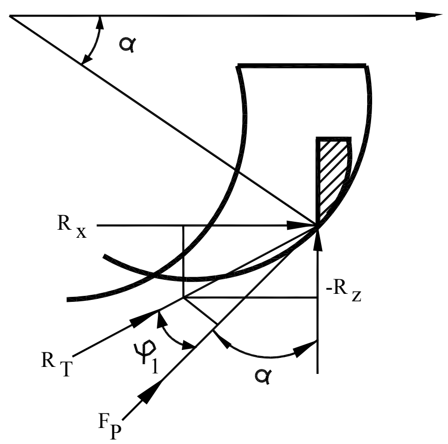


Figure 27 Diagram for determining the magnitude of the components R_x and $R_{Z.}$
Source: - Sineokov (1977)

$$R_X = R_T \sin(\alpha + \psi_1) \tag{38}$$

$$-R_z = R_T \cos(\alpha + \psi_1) \tag{39}$$

Where:

R_X = the horizontal components of forces, N,

R_z = the vertical component of force, N,

R_T = soil reaction force, N,

R_P = peripheral force, N,

α = angle of direction , $\left(^0\right)$,

ψ_1 = angle of periphery, $\left(^0\right)$.

3. 2. 3. 2 Power requirement to cut and loosing the soil slice

In the process of cutting and loosing the soil slice of a rotary blade required cutting and loosening the slice power which is by Guptya and Visvantan (1993) and computed as:

$$P_{cut} = \sigma_{imst} \mathrm{A}_{\mathrm{ds}} \left(R - L_d / 3\right) \qquad (40)$$

Where:

P_{cut} = Cutting the surface of the soil slice power requirement, W,

σ_{imst} = Impact shear stress to cut the soil slices, N/m^2, $\sigma_{imst} = \dfrac{F_{cut}}{A_{ds}}$,

Total area of cutting surface of the soil slice of rotary blade, $\mathrm{A}_{\mathrm{ds}} = L_d L_B$.

The force to cut the soil slice $\left(\mathrm{F}_{\mathrm{cut}}\right)$ and the total area of cutting surface of the soil slice of rotary blade $\left(\mathrm{A}_{\mathrm{ds}}\right)$ are indicated in Figure 28.

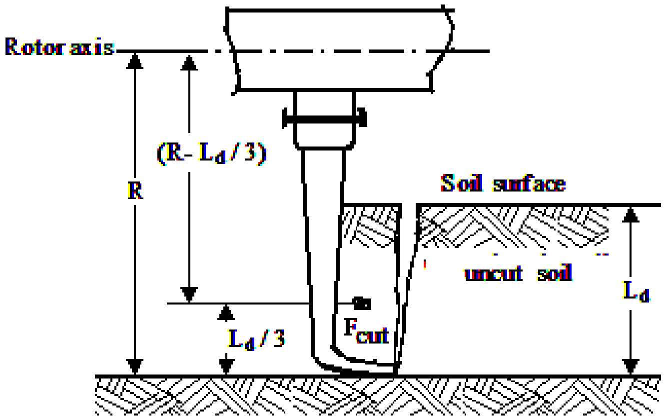


Figure 28 Force due to cutting the soil.

Source: - Gpta and Visvanathan (1993).

3. 2. 3. 3 Power requirement to over come soil -metal friction

In the process of overcoming soil -metal friction rotary blades require soil -metal friction power which is by Guptya and Visvantan (1993) and computed as:

$$P_{fircn} = L_B B_W S_{PW} \mu_k R \tag{41}$$

Where:

P_{fircn} = Overcoming soil –metal friction power requirement, W,

L_d = Bite length, m,

B_W = Cutting width, m,

S_{PW} = Dry soil bulk density,

μ_k = Kinetic coefficient of soil –metal friction ,

R = Rotor radius, m.

3. 2. 3. 4 Power requirement to throw the cut surface of the soil slice

The throwing power is accomplished by the action of the centrifugal force on the rotary blade (Figure 29) which is computed by Guptya and Visvantan (1993) as:

$$P_{Throw} = \frac{MV_s^2}{R}\left(R - L_d/3\right) \qquad (42)$$

Where

P_{Throw} = Throwing the cut soil slices power requirement, W,

M = Mass of the soil slices, kg, $\mathrm{M} = \mathrm{L_d L_B B_W S_{PW}}$ /G , G is acceleration due to gravity.

$\mathrm{A_{ds}}$ = Total area of cutting surface of the soil slice of rotary blade, $\mathrm{A_{ds}} = L_d L_B$.

The force to cut the soil slice $(\mathrm{F_{cut}})$ and the total area of cutting surface of the soil slice of rotary blade $(\mathrm{A_{ds}})$ is indicated in Figures 28 and 29.

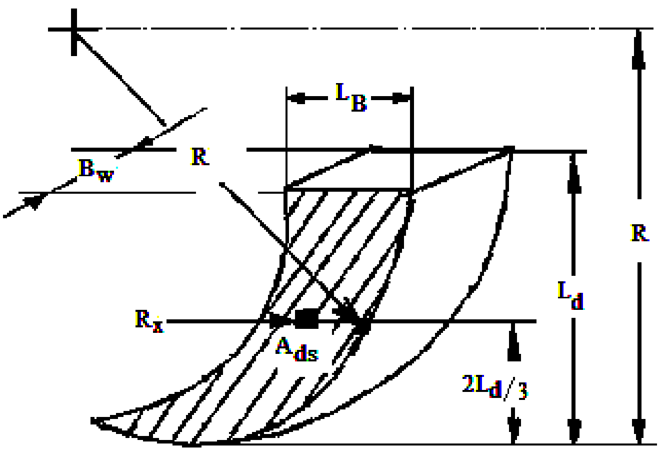


Figure 29 Force due to soil –soil share (Gpta and Visvanathan, 1993)

Source: - Gpta and Visvanathan (1993).

Table 1 Average minimum bulk densities that restrict root penetration in soils of various textures.

Texture	Bulk Density g/cc
Coarse, medium ,and fine sand	1.80
Loamy and sandy clay loam	1.75
Loam and sandy clay loam	1.70
Clay loam	1.65
Sandy clay	1.60
Silt and silt loam	1.55
Slity clay loam	1.50
Cay	1.40

Source: - James (1972), For the present study the bulk density is 1.70 g/cc was selected.

3. 3 Energy requirement for rotary tillage operation

Energy has been defined as the ability to do work. Manual, animal and mechanical energy have been available in agricultural production. Soil tillage operation from the initial to the final condition operation needs considerable application of energy. An engine of an agricultural tractor converts the diesel fuel energy into useful mechanical energy to the drawbar and rotational power of the application.

It is the fact that tillage operation requires a considerably higher amount of energy demand in tillage and traction systems. The energy can be expressed in terms of energy per unit area or per volume of disturbed soil (Panwar and Siemens, 1972). Since energy requirement per unit volume has the same unit as draft per unit area, it can be shown as the draft requirement to cut a furrow in an exact cross section as expressed by McKyes (1985) and Chancellor (1994). Energy can also be expressed as the rate of energy per depth of operation (Darmora and Pandey, 1995).

The most important factors in the determination of energy requirement of a tillage tools are draft and the amount of disturbed soil. In other words, if those two values are available, energy can easily be calculated as the product of draft and the length of disturbed soil. For a tillage operation, the report of energy requirement should include the depth of operation as well. However, in rotary cultivators not only the depth of tillage but also the cutting width, forward and rotational velocities for energy requirement are relevant.

Kosutic et al. (1996) studied the energy requirements of rotary cultivators with two blade arrangements (flat and steep spiral) and two blade shapes ("L"-shaped and "I"-shaped). It was concluded that the blade shape had greater influence on the energy requirement than the arrangement of blades. The "I"-shaped blade required 21.2% to 25.7% less energy than the L-blade. Forward speed and depth of tillage also influenced the energy requirement. Increasing the velocity from 0.68 m/s to 1.40 m/s decreased the energy requirement per unit volume of tilled soil by 23%. While increasing the depth of tillage by 40% decreased the energy requirement per unit volume of tilled soil by 19.3%.

3. 3. 1 Rotary tiller specific energy requirement

The efficiency of a rotary cultivator has been measured in terms of specific energy requirement. The specific energy for the rotary cultivator for a particular operation is exhibited by a number of researchers. Beeny and Khoo (1970) defined the complex three dimensional shapes of a rotary cultivator blade. They studied and compared the performance of three blade shapes, namely: "L", "I" and "C –shaped rotary blades having different radii of curvature. The specific work was defined as ratio of work done per bite to the volume worked per bite as:

$$\text{Specific work} = \frac{\text{Work done per bite}}{\text{Volume worked per bite}}$$

The specific work requirement of the “L” blade was found comparatively higher than of the other two types over a similar range of operating conditions .The horse power requirement of the blade I was considerably less than other two even though this blade experiences a high degree of slippage between the blade and the soil at higher speeds. They did not find much difference in the area /volume ratios between the blade types. The “L” shape blade gave the greatest forward thrust to the vehicle on which it was fitted. They found a deference of 0.022 N thrust per bank between “L” and “C” blades which is not worth considering in view of the 30 percent power reduction in using “C” blades.

Sohne and Eggenmuller (1959) set out to compare the power needs of rotary cultivators and other cultivating tool as the following expression:

$$E_W = \frac{E_E}{V_{ST}} \tag{43}$$

Where:

$E_W =$ Specific work, kJ/m^3,

$V_{ST} =$ Volume of soil tilled per second, m^3, The volume of soil tilled is given by Hendrick and Gill (1971), (Figure 30) as:

$$V_{ST} = L_d L_B B_W \tag{44}$$

Where:

V_{ST} = Volume of soil tilled per second, m^3,

L_d = Depth of tillage, m,

B_W = Cutting width, m.

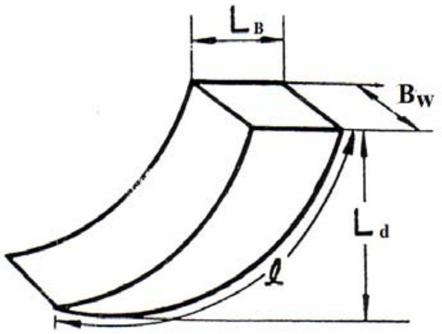
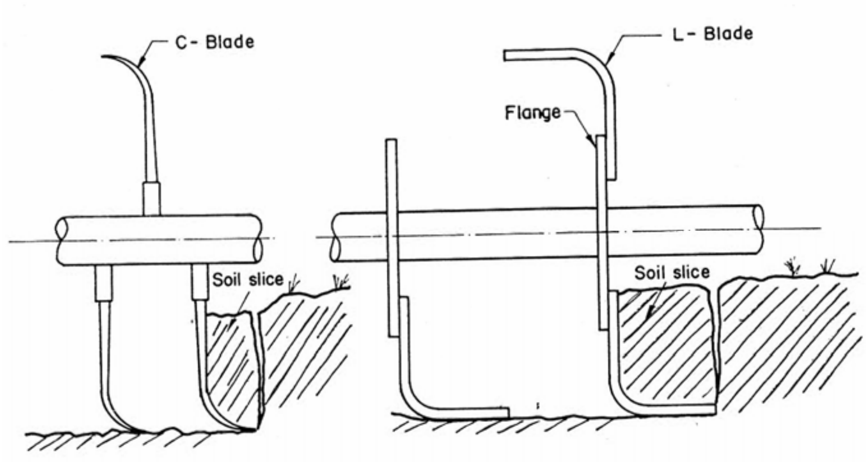


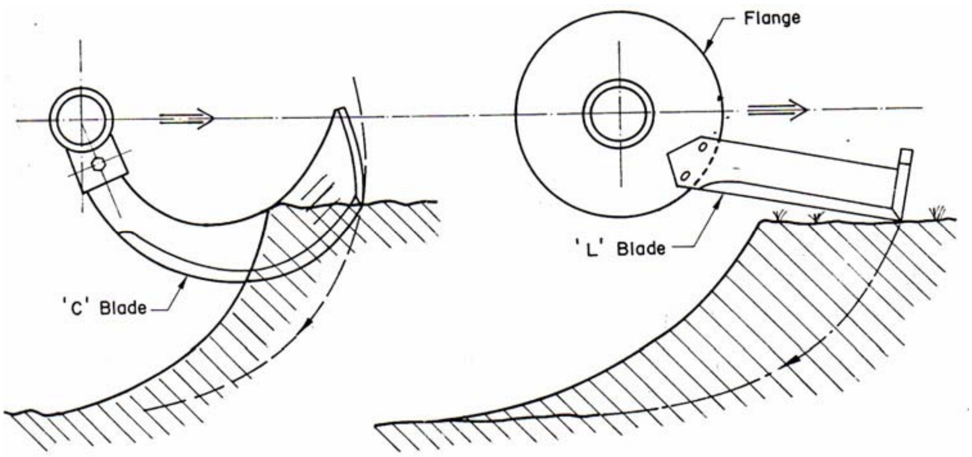
Figure 30 Dimensions of the soil slice.

Source: Hendrick and Gill (1971).

Some of the geometry of soil slices depends on the configuration of the rotary blades are shown in Figure 31.



(a) Side-view



(b) Front - view

Figure 31 Geometry of soil slice cut by “C” and “L”- shaped rotary blades.

Sourec: Salokhe *et al.* (1993)

Bernacki *et al.* (1972) expressed the specific work in terms of the torque requirement of the rotor of the rotary tiller per volume of soil tilled as :

$$E_W = \frac{2\pi F_T}{Z L_d B_W L_B} \tag{45}$$

Where:

E_W = Specific work, kJ/m^3,

L_d = Depth of tillage, m,

F_T = torque on rotor shaft, N.m,

B_W = Cutting width, m.

Z = Number of working elements operating in one plane,

L_B = Bite length, m.

Shinners *et al*. (1990) specific energy is computed as:

$$E_{Sp} = \left(\frac{P_{To}/\mu_{PTO} + P_{dr}/\eta_{dr}}{V_f \times B_W} \right) \times 10 \tag{46}$$

Where:

E_{Sp} = Specific energy, kWh/ha,

P_{PTO} = PTO power, kW,

μ_{PTO} = PTO power transmission efficiency,

P_{dr} = Drawbar power, kW,

μ_{dr} = Draw bar power tractive and transmission efficiency,

V_f = The specific fuel consumption, $m\ L\ kWh^{-1}$,

B_W = The hours of use of power source, h,

3. 4 Rotary blade design parameters

The design parameters of the rotary blades includes, soil condition, the shape, the cutting depth, the cutting width, the peripheral and forward velocities, the velocity of ratio, the rotor radius, the radius of the curvature angle of elements and the direction of rotation. These parameters have a marked influence on all phases of tiller operations, from the power required to the final soil condition. Cutting process by the rotary cultivator has been accomplished by trajectory path. The rotational and the forward velocities are occurred simultaneously. The power requirement has been influenced by the parameters of the rotary tiller from the initial to the finical soil condition.

Numerous studies have been reported on the performance and design parameters of various rotary tillers. Niyamapa *et al*. (1994) determined the design parameters of rotary cultivators, namely, the depth of tillage, the rotor and forward speed influence on the power requirement to cut and throw the soil clods under soil bin laboratory study and recommended clod size development for soybean cultivation.

Soehne (1957), under soil bin operation, studied the influence of rotary blades with various geometric factors (cutting width, cutting angle, radius of curvature and operational speed) influencing specific work requirements. Adams and Furlong (1959) studied the performance of the "pick", the "L", and the "C"- shaped blades on rotor horse power and soil reaction forces. Beeny and Khoo (1970) conducted some experiments to study "I", "L" and "C"- shaped rotary blades power requirements and forward thrusts.

3. 4. 1 Depth of Tillage

Depth of tillage is an important factor in the dissemblance, the mixing of soil and cutting of roots. The depth of operation has considerable effect on the power requirement and performance of rotary tillers because the actual depth of tillage for each blade during its operation varies throughout the rotational path of the blade (Hedrick and Gill, 1971b, and Gill and Hedrick, 1976). The average depth of tillage is controlled by peripheral and forward velocities, lateral spacing of the rotary blades, ratio of rotor radius and depth and in towed tillers the weight of the machine. Depth of tillage has been increased, other constant conditions, increase the power requirement of the rotary blade per unit volume of soil tilled. Power requirement increased as depth of tillage increased for direction of rotation, peripheral velocity and bit length (Hedrick and Gill, 1971b). Dalin and Pavlov (1950) evaluated the performance efficiency of three types of rotary blades, namely, an “L”, an “I”, and “Pick” –shapes at the same exerting pushing power requirements of the rotary blades, the “I”, and a “Pick” –shape (Figure 32).

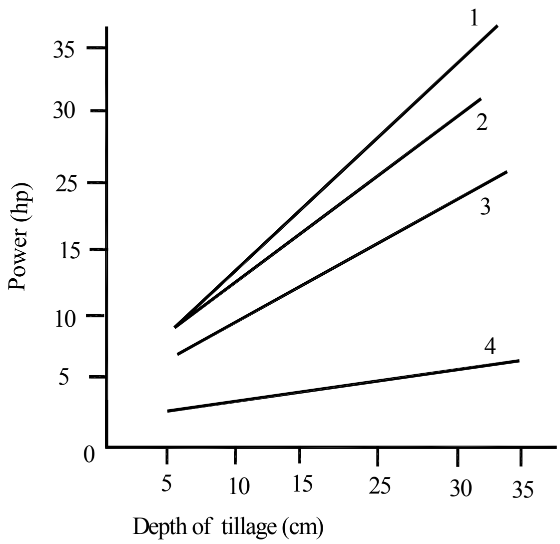


Figure 32 Influence of depth of tillage on the power requirements of rotary blades

Source: - Dalin and Pavlov (1950)

1. “L” ” –shaped
2. “I” ” –shaped
3. “Pick” –shaped

At a constant rotational velocity, the input power requirement of rotary blade has been increased with an increase in forward velocity at deeper than at shallow depth of operation (Singh, 1979).

Depth of tillage, as similar to other tillage methods, has considerable influence on the power requirements and performance of rotary blades. The average depth of tillage is governed by the depth of effective tillage, the bit length, the peripheral and forward velocities, and the lateral spacing of the blades. Dalin and Pavlov (1950) presented input power versus depth for “L”, “I” and “Pick”-shaped rotary blades. The pushing power exerted by each type of rotary blade was the same. Specific power to till remained constant regardless of depth. Furlong (1956) investigated the influence of the depth of operation on rotor power when operating with ‘L’ and ’C’ shaped blades with forward and reverse rotation. The study was carried out at two peripheral velocities, i.e. 121 and 213 m/min and three tilling pitches 50,100 and 150 mm. Observations were taken while power input increased with depth, the specific power requirement decreased for both directions of rotation and peripheral velocities including all three tilling pitches. Tsuchiya (1965) investigated the rotary power requirement increase with increasing tillage depth. It was concluded that increasing the tillage depth increases the total power requirement, but decreases the specific power requirement. With an increase in the cutting depth from 50 to 75 mm, the specific energy requirements of the “Pick”-shaped rotary blades decreased, at the rotation speeds of 150 to 300 rev/ min (Saraswat, 1987).

3. 4. 2 Cutting width

Cutting width is one of the determining factors affecting specific energy requirements of rotary blades. According to Saraswat (1987) specific energy requirement continuously decreased from 5072 to 3156 kJ/m^3 when the cutting width was increased from 10 to 22 mm at 50 mm depth of tillage. However at 75 mm depth of tillage, the mean value of specific energy requirement was first found to increase from 3388 to 3785

kJ/m^3 with increase in cutting width from 10 to 16 mm and then it dropped to 2887 kJ/m^3 when the cutting width was increased to 22 mm. At both depths, the minimum value of specific energy was observed at 22 mm cutting width and it was found to be significantly lower than the values obtained at the other two widths. The trend showing a decrease in specific energy with an increase in cutting width is also supported the findings of Soehne (1957), Adams and Furlong (1959), Mashchenski (1973) and Tsuchiya and Honami (1963). This may have been due to the fact that the volume of soil tilled increased in cutting width resulting in the increased specific energy requirement.

3. 4. 3 Peripheral and forward velocities

The performance of the rotary tillage tools depends on a velocity ratio λ which is given by Hedrick and Gill (1971) as:

$$\lambda = \frac{\text{Rotor peripheral velocity}}{\text{Machine forward velocity}} \qquad (47)$$

$$= \frac{R\omega}{V_f}$$

Where R is the rotor radius, m, ω is the angular velocity, rads^{-1}, and V_f is the travel velocity, km h^{-1}. Peripheral velocity $(V_P) = \omega R$. Changing the rotor radius, rotary velocity and machine forward velocity can vary velocity ratio.

Forward velocity varied, while holding the rotational velocity of the rotor constant to obtain different values for bit length. Having used “C”- shaped blades in which both forward and reverse rotations were used at a rotor speed of 192 rev/min, rotor diameter of 660 mm, rotor width of 1 m and depth of tillage of 150 mm, the range of forward speed was from 0.154 to 0.81m/s, which varied velocity ratio between 42 and 8.1. The tilling power requirement increased with increasing forward velocity while the specific energy decreased (Dalin and Pavlov, 1950).

According to Furlong (1956), varied velocity ratio by increasing forward velocity while maintaining a constant value of ω. Three rotary velocities were used in reverse as well as in forward directions. It was observed that as tilling pitch increased, draft power also increased for each set of rotor speeds.

El' Gurt (1965) conducted a laboratory study on the power requirements of a forward and reverse rotating tined tiller. The power inputs to the tiller were measured at different forward speeds and the specific energy in each case was determined. It was observed that in forward rotation, a greater bit length reduced the specific energy. However, in reverse rotation, increased bit length had relatively little effect on specific energy.

Kisu *et al.* (1966) reported that the typical power requirement reduced with increases in bit length when forward velocity was held constant as rotational velocity was reduced. This was due to a decreased cutting path per unit slice volume as the bit length was increased. The power requirement began to rise again as ω reached some point.

Alekseenko (1972) conducted studies on the operation of rotary tillers for the surface tillage of solonetz soils. Experimental depth rotary tillers were operated on medium solonetz soils at 100 mm deep to determine the effects of peripheral speed and rotor diameter on soil pulverization and power. The ratio of peripheral speed to forward speed remained constant.

Similar to the result of the above studies Hedrick (1971) concluded as the following:

1. Decreasing the velocity ratio (λ) by increasing forward speed resulted in an increase in the power requirement, but a reduction in specific power.
2. Decreasing the velocity ratio λ by decreasing rotor velocity decreased the power requirement and specific power.
3. Increasing the velocity ratio λ resulted in greater value of the ratio between cutting area and volume per soil slice cut

3. 4. 4 Radius of curvature

Radius of curvature influences the torque requirement of the rotary shaft. Soehne (1957) and other researchers have investigated the effect of radius of curvature on the L-shaped rotary blade. The radius of curvature and cutting width of "pick"-shaped blades were observed to influence the specific energy requirement at the cutting depth of 50 and 75mm. At 40, 55, 70, and 85 mm of radii of curvature tested of the "pick" – shaped rotary blade, the minimum mean value of specific energy requirement was found to be 3021 kJ/m3 at the cutting depth of 50 with 55 mm curvature, which was significantly lower than the minimum values observed at other curvatures (Saraswat, 1987).

3. 4. 5 Mode of rotation (Direction of rotation)

During the process of soil cultivating, rotary blades execute rotational and translational motion. The trajectory motion of each point on the rotary blade depends upon the circumferential and translatory of velocity. For rotary blades with a horizontal axis of rotation, the trajectory of motion of the blade is an extended cycled (trochoid). The dimensions and form of the cut soil slice are determined by the trajectories of motion of two successively working the rotary blades. In cutting the soil slice from the above downward (Figure 33a), its cross section decreases fro maximum to zero, while in reverse rotation (Figure 33b), the soil slice cross section increases from zero to maximum.

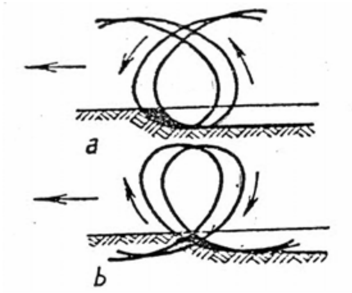


Figure 33 Diagram showing the formation of the soil slice for forward and reverse rotation.

Source: - Bosoi *et al*. (1988)

The direction of the rotation of the rotor is a basic rotary tiller design parameter. As a machine moves to the right, it is considered to have " forward rotation" if the axis is turning clockwise, and " reverse rotation" if the rotary axis rotation is counter - clockwise(Figures 34 and 35). When forward rotation was used, each rotary blade cut increment of undisturbed soil while entering from the surface and with the reverse rotation the soil increment was cut from the bottom upward. Reversing the direction of rotation changes the geometry of the soil tool system, even where the ratio of rotor peripheral speed to forward machine speed is constant. The shape of soil slice is one of the more obvious differences, even though the volume of the slice is essentially equal in both cases. A point on the periphery of an individual blade describes a trochoidal path during operation (Hendrick and Gill, 1971).

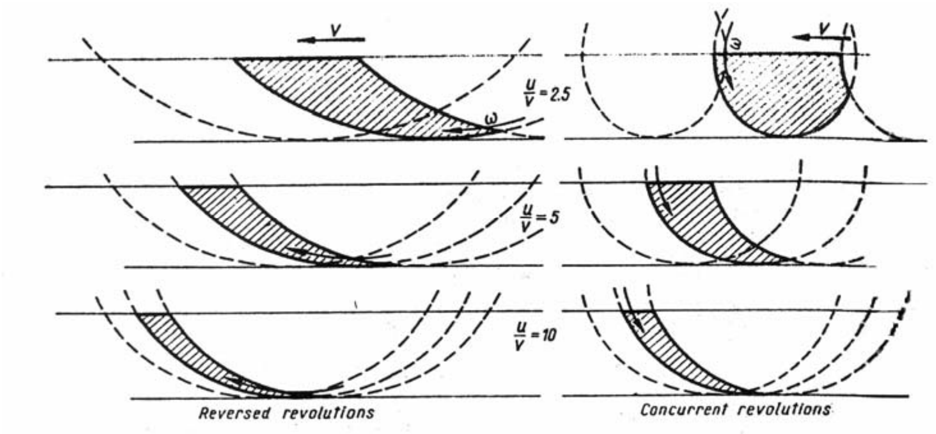


Figure 34 Types of soil slices for forward and reverse rotation as a function of the ratio of peripheral and forward speeds: $\frac{R\omega}{V_f} = \frac{60V_P}{V_f} = \lambda$

Source: -Bernancki (1962).

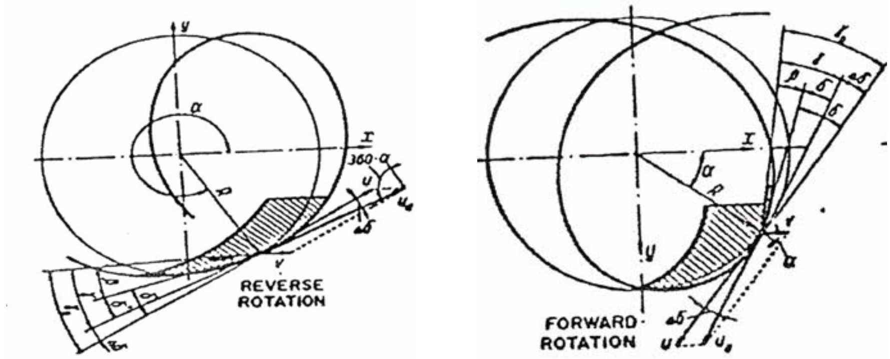


Figure 35 Angles and speed of cutting with forward and reverse rotation.
Source: -Bernancki (1962).

The path intersection angle, $\Delta\partial$ is the angle between the tangents to the rotor circumference and the trochoid at the point of intersection. The magnitude of the $\Delta\partial$ depends up on the value of $V_f / V_P = \lambda$ (the ratio of the forward and peripheral velocity of the rotor), the angle of the rotation and the direction of rotation. The smaller value of λ, the greater the change in $\Delta\partial$ during one cut, and thus, the greater must be the angle δ (apparent clearance angle –the angle between the back surface of shaped edge and tangent to the rotor circumference) to prevent compaction of uncut soil. The largest value of $\Delta\partial$ occurs near the beginning of the cutting forward rotation, and at the end of the cutting for reverse rotation. For a given situation, the variation of $\Delta\partial$ in one cutting stoke is less for reverse rotation than for forward rotation .Hence, the rotary blade can operate at nearly an optimum angle in reverse rotation through the cutting reprocess.

A few researchers reported the effect of the direction of rotation of various rotary blades. Many experiments were conducted by using “pick” and “C”-shaped blades operating in forward and reverse directions. In each test, the rotary power was greater for forward than for reverse rotation. The horizontal component to move the tiller during reverse rotation was found negative.

Dalin and Pavlov (1950) evaluated from the pick type test the revise rotation required 12 to 16 % less energy for rotation, the total power requirement was 14 to 28 % less for forward rotation because of the horizontal component. For the testing using“C“ pick”-shaped blades the general results were the same, with a reduction in power for reverse rotation, but the total power requirement for reverse rotation was 0.5 to 11 % greater than forward rotation. Reversing the rotation of the “C”-shaped blade did not reduce the rotary energy as much as the “pick”-shaped blade because the “C”-shaped blade does more cutting than the “pick”-shaped blade and the advantage of reverse rotation is in increasing failure by tension.

Furlong (1956) studied the rotary and draft power requirements and soil pulverization and other factors as a function of the direction of the rotor rotation, the width of rotor, the peripheral velocity, the tillage pitch and the depth of tillage .In all cases except "pick"-shaped blades, the reverse rotation required more rotor power. On average, reverse rotation required 70% more total power than forward rotation when operating under the conditions of tests.

Matsuo (1963) developed theoretical and experimental data comparing forward, down cut, and reverse, up-cut tiller operation using a Japanese style blade having a rotor radius of 220 mm. The experiments were conducted over a depth range of 20 to 150 mm and a range of rotational velocities of 50 to 350 rev / min .It was found that the power requirements were less for reverse rotation, given the same pitch of cut, and that the reduction in power becomes larger as the soil strength becomes smaller.

Bok (1965) reported that, based on theoretical analysis, the main disadvantages of reverse rotation were the throwing of soil forward and the reversing of horizontal components on the drawbar from pushing to pulling.

Matyashine (1968) reported that at shallow tilling depths (cutting depth less than rotor radius), forward rotation required 10 to 15% less energy than reverse rotation. When tilling depth (cutting depth greater than rotor depth), reverse rotation reduced the energy requirement by 20 to 30 %).

Grinchuk and Matyashine (1969) summarized the results of a few Russian researchers and reported that in general the reverse rotation decreased the force of cutting by 1.5 times, gave better depth stability, reduced breakdowns of tools in stony soils, and made possible a wide range of peripheral to forward velocities. Among the disadvantages they associated with reverse rotation were a greater energy requirement at a depth of operation of cutting depth less than rotor radius, and the need to increase the rotary velocity to prevent throwing soil ahead of the tiller.

3. 5 Rotary blade design equation

Rotary blade design equation is one of the most important design criteria of rotary blade. The procedure to trace the logarithmic curvature for the rotary blade test the following equation has been used (Sakai, 2000).

$$R = R_0 \, e^{-(Cot\alpha)\theta} \tag{48}$$

Where

R = radius vector, mm,

R_0 = value of R when θ is equal to zero,

e = base of natural logarithms,

θ = angular displacement, radian,

α = angle between a radius vector and its corresponding tangent of the spiral.

Differentiate equation (48) for an ideal edge – curve of rotary tiller blades has been given by Sakai (2000) as:

$$R = R_0 \sin^{(1/k)} \alpha_0 \times \sin^{(-1/k)} (\alpha_0 \times k\theta) \qquad (49)$$

Where R and θ: radius (mm) and angle $(^0)$ in polar coordinates

R_0 and α_0 : radius and edge – curve angle at the tip of the edge curve in polar coordinates, when $\theta = 0^0$.

3. 5. 1 Rotary blade design test criteria

Before fabricating the rotary blade, it should be designed from a strength point view. The strength of the rotary blade is calculated in terms of maximum peripheral force using Bernacki *et al.* (1972) as:

$$F_P = \left(\frac{750 P_e}{V_P} \right) \eta_C \eta_Z \qquad (50)$$

Where

F_P = peripheral force, N;
P_e = engine power of the power tiller, hp;
V_P = peripheral speed of the rotary blade, m/s;
η_C = power tiller efficiency mounting to about 0.9 for machines with concurrent revolutions;
η_Z = coefficient including a reverse of power tiller power mounting to 0.7 to 0.8.

3. 5. 2 Peak peripheral force

The peak peripheral force of an individual element which is applied to the tip of the rotary blade at a certain distance from the hole of the blade has been determined by the peak peripheral force of the entire machine divided by the number of working elements. All individual working elements are not uniformly loaded when the overloaded factor is introduced. Therefore, the peak peripheral force of an individual element is given by (Bernacki *et al.*, 1972):

$$K_e = \frac{K_P C_P}{iZn_e} \tag{51}$$

Where

K_e = peak peripheral force of an individual element, N;

K_P = peak peripheral force, N

C_P = overloaded factor of an individual working element,

i = number of sets in the machine;

Z = number of working elements in the set,

n_e = fraction determining how many working elements operate simultaneously.

The peak peripheral force, $K_P = F_P C_S$, where F_P is peripheral force, N, C_S = over load factor. The peak moment on the rotor shaft of the rotary blade is given by:

$$M_S = K_P R = 2RF_P \tag{52}$$

Where R is rotor radius.

3. 5. 3 Determination of the diameter of rotor shaft

The diameter of the rotor shaft has been determined as:

$$d = \sqrt[3]{\frac{16M_S}{\pi\tau}} \qquad (53)$$

Where

d = diameter of the rotor shaft, mm,

M_S = peak moment on the rotor shaft of the rotary blade, Nm,

τ = torsion stress, N/m^2.

3. 5. 3. 1 Bending stress

The standard of a rotary blade is determined by bending and direct stress. The bending stress is given by:

$$\sigma = \frac{M_B}{\mathrm{I}} d_S \qquad (54)$$

Where

σ = bending stress, N/m^2; $\sigma = K_e\, d_e$

M_B = bending moment at the given section, Nm,

I = moment of inertia of the cross-section about the axis, m^4,

d_S = distance from the neutral surface to the extreme fibre, mm,

d_e = distance from the tip of the rotary blade to the centre of hole of the rotary blade; neutral surface to the extreme fibre, mm.

3. 5. 3. 2 Moment inertia

Moment inertia of the cross - section about the neutral axis is determined as:

$$\mathrm{I} = \frac{S_d\left(B_W\right)^3}{12} - \frac{S_d^{'}\left(B_W^{'}\right)^3}{12} \qquad (55)$$

Where

$\frac{S_d\left(B_W\right)^3}{12}$ = moment inertia of inertia of standard, m^4,

$\frac{S_d^{'}\left(B_W^{'}\right)^3}{12}$ = moment inertia of a square hole in the standard, m^4,

I = moment of inertia of the cross-section about the axis, m^4,

S_d = standard thickness, mm,

$S_d^{'}$ = thickness, mm,

B_W = standard width, mm,

$B_W^{'}$ = width, mm.

3. 5. 3. 3 Shear stress

The shear stress is given by

$$\tau = \frac{K_e}{A} \qquad (56)$$

Where

τ = shear stress, N/m^2;

K_e = peak peripheral force of an individual element, N,

A = area of cross-section of the standard, $A = \left(B_W S_d \text{ - } \left(S_d\right)^2\right) \times 10^{-6} m^2$.

Thererefore, checking the safe design of the rotary blade is determined by maximum shear stress theory (Kiatiwat, 2005) as:

$$\left(\frac{\sigma}{\sigma_Y}\right)^2 + \left(\frac{\tau}{\tau_Y}\right)^2 \le 1 \quad (57)$$

Where

σ_Y = yield strength, N/m^2,

τ_Y = Shearing yield strength, N/m^2.

3. 6 Rotary blade design used material

The most commonly used materials are spring steel and carbon steel having the following compositions (Table2)

Table 2 Types of rotary blade design used materials.

Material	Chemical composition (%)				
	Carbon(C)	Silicon(Si)	Manganese (Mn)	Prosperous(P)	Sulphur(S)
Spring Steel(SUP 6)	0.55 to 0.65	1.5 to1.8	0.7 to 1.0	0.30	0.03
Carbon Tool Steel(SK 5)	0.0 to 0.90	0.35	0.50	0.30	0.03

Source: -Sakai and Hai (1980)

3. 7 Technological process manufacturing rotary blade

1. Material selection

Selection of the proper material for a particular application on the rotary blades is important from the stand point of cost, durability, availability, and machine performance.

Rotary blades should generally be designed to utilize the lowest cost material that will satisfactorily and give adequate life. Spring steel and carbon tool steel have a good spring structure, harding property and resistance to abrasive wear after heat treatment. Types of rotary blade design used materials are shown in Table 2.

The Technological process for manufacturing rotary blades has been considered as the following

Preparation

1. Cutting the steel strips and heat treatment so that the cutting knife is strengthened to overcome soil metal friction:

The steel strips are cut in to pieces a 30- ton hydraulic press. The bits are then heated to a temperature of 1200 ^{0} C to make them elastic. They are then pressed into shape in hydraulic presses or shaped in roller presses. The technological procedure is shown in Figure 36 and 37.

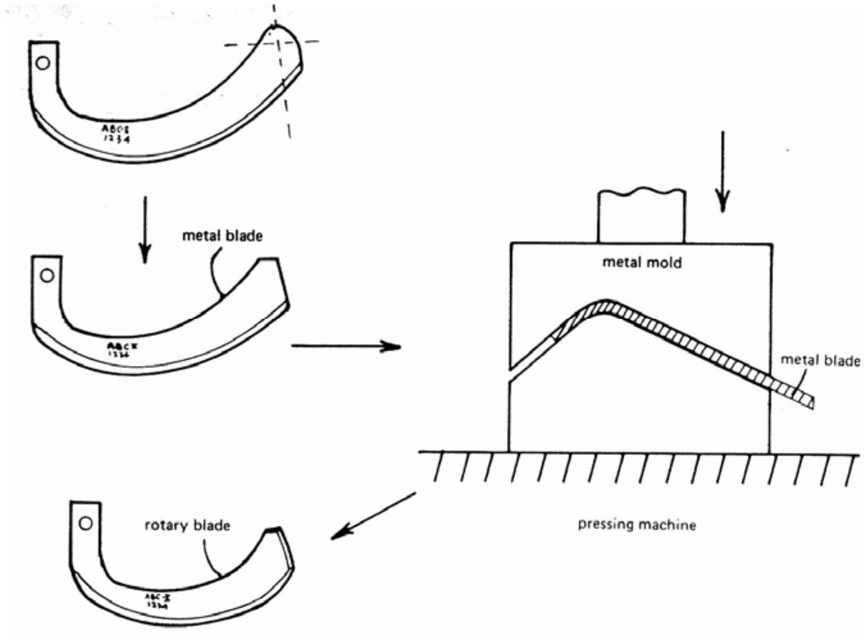


Figure 36 Bending of sidelong blade in hydraulic press

Sourec: -Sakai and Hai (1980).

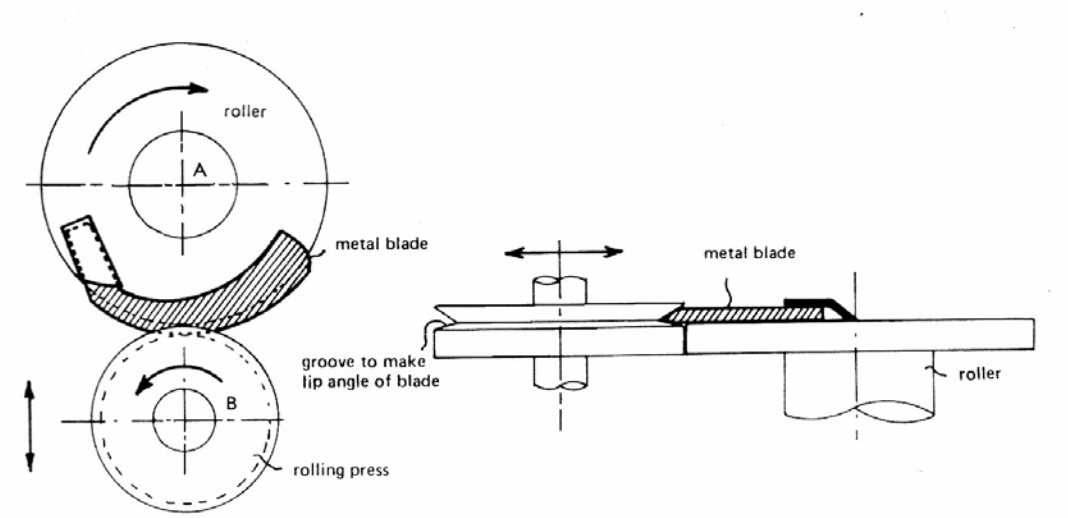


Figure 37 Bending rollers for making of edge curve.

Sourec: - Sakai and Hai (1980).

2. Forming press blade
3. Forming of holding portion
4. Bending of knife portion
5. Hole making
6. Punching mark and cutting tip knife portion
7. Forming blade –tip
8. Cooling and inspection

4. Fuel energy calculation

Fuel energy consumption required of a power tiller for a particular operation was calculated by Mittal *et al*. (1985) from the following formula:

$$F_C = \frac{L_{CF} \times R_{PI} \times F_{SPC}}{100 \times H_{OU}} \tag{58}$$

Where:

$F_C =$ The fuel consumed, $L\ ha^{-1}$,

$L_{CF} =$ The load coefficient for the operation,

$R_{PI} =$ The rated power of the source, kW,

$F_{SPC} =$ The specific fuel consumption, $L\ kWh^{-1}$,

$H_{OU} =$ The hours of use of power source, h,

Suresh and Varshney (2005) calculated the specific fuel consumption of power tiller engine by dividing the fuel consumption (L/hr) with drawbar power (kW) and formulated the general form of an empirical equation to relate fuel consumption and drawbar power on tilled land as follows:

$$F_{SPC} = a + bP + cP^2 + eP^2 \qquad (59)$$

$$F_C = a + bP \qquad (60)$$

Where F_{SPC} is specific fuel consumption, L / kWh, F_C is fuel consumption, L / hr ; P is Drawbar power, kW, a, b, c and e are parameters.

4. 1 Fuel consumption model for a specific operation

A fuel consumption model for a specific operation is determined by total tractor power for that operation and the equivalent PTO power is divided by the rated maximum to get a percent load for the engine. The fuel consumption at that load is obtained from ASAE D497, clause 3 (ASAE, 2000):

$$F_{EC} = F_{Spc} P_{Total} \qquad (61)$$

Where:

F_{EC} = Estimated fuel consumption for a particular operation *L/h*;

P_{Total} = Total tractor power (PTO equivalent) for the particular operation kW (1 hp = 0.74 kW),

F_{Spc} = Specific fuel consumption for the given tractor, determined from ASAE D497, clause 3, *L/kW h.*

As cited by the American Society of Agricultural Engineers standards (ASAE EP496, 2001) average fuel consumption of a tractor operating under a range of load conditions was determined as:

$$F_C = 2.64X + 3.91 - 0.203\sqrt{738X + 173} \quad (62)$$

Where F_C is fuel consumption in *L/ kWhr*, X is ratio of equivalent PTO power required by operation divided by the maximum available PTO power.

Grisso *et al.* (2004) developed a generalized fuel consumption model and is computed as the following equation:

$$F_{SPC} = (0.22X + 0.096)(1 - (0.0045XN_{\mathrm{Re}d} + 0.00877N_{\mathrm{Re}d}))PTO \quad (63)$$

Where F_{SPC} is diesel fuel consumption at partial load and full/reduced throttle, L/hr; X is ratio of equivalent PTO power to rated PTO power, decimal; $N_{\mathrm{Re}d}$ is percentage of reduced engine speed for a partial load from full throttle, % and PTO is rated PTO power, *kW*.

Kheiralla *et al*. (2004) formulated a fuel consumption model of rotary tillers in Serdang sandy clay loam soil as:

$$F_C = 0.2156PTO + 6.2347 \quad (64)$$

Where F_C fuel consumption is rate *L/ hr*, PTO is power take –off (PTO), *kW*.

Power-takeoff power is power required by the implement from the PTO shaft of the tractor or engine. A typical PTO power requirement has be determined using rotary power requirement parameters given by ASAE D497, clause 4. Implement power take-off power has been also calculated as (ASAE, 2000):

$$P_{PTO} = a + bw + cF \qquad (65)$$

Where:

$P_{PTO} =$ Power-takeoff power required by the implement kW,

a, b and c are machine specific parameters (ASAE D497).

$F =$ Material feed rate, ton / h (wet basis).

5. Optimization

Historical Background

The term "optimization" refers to the action of finding the best solution and has been determined by a particular function either maximizing or minimizing the objective function with relative constraint parameters. The word "modelling" comes from the Latin word modellus and refers to an abstract representation of a real-world object. i.e., a typical human way of coping with the reality which has been in use since the stone age, a fact backed up by cavemen paintings. The real breakthrough of modelling came with the cultures of the Ancient Near East and with the Ancient Greek. The first recognizable models were numbers; counting and "writing" numbers (e.g., as marks on bones) is documented since about 3000 BC. Astronomy and Architecture were the next areas where models played a role, already about 4000 BC. It is well known that by 2000 BC at least three cultures (Babylon, Egypt, India) had a decent knowledge of mathematics and used mathematical models to improve their every-day life. Most mathematics was used in an algorithmic way, designed for solving specific problems. The development of philosophy in the Hellenic Age and its connection to mathematics lead to the deductive method, which gave rise to the first pieces of mathematical theory. Starting with Thales of Miletus at about 600BC, geometry became a useful tool in analyzing reality, and analyzing geometry itself sparked the development of mathematics independently of its application (Schichl, 2000).

The goal of optimization is to create attention for policy makers, planners, and advisors, and the trend is economic development in the global world. It has been widely applied under agricultural, mechanical, electrical and civil engineering fields, particularly in engineering science for the development of new electronics machines, computers, and robots to reduce labour power. However, it has not been applied for the development of rotary tillers. Therefore, an optimization technique has been undertaken under this study to optimize the design parameters of various rotary blades.

5. 1 Mathematical model

A mathematical model is a symbolic representation of the essential aspects of an existing system in usable form and gives updated information for agricultural machinery managers, designers and investigators. Optimization modelling is described as a branch of mathematical modelling concerned with finding the best solution to a problem and finding maximum or minimum values of a function in a given system (Rao, 1985). The process of a model and optimization is summarized in Figure 38.

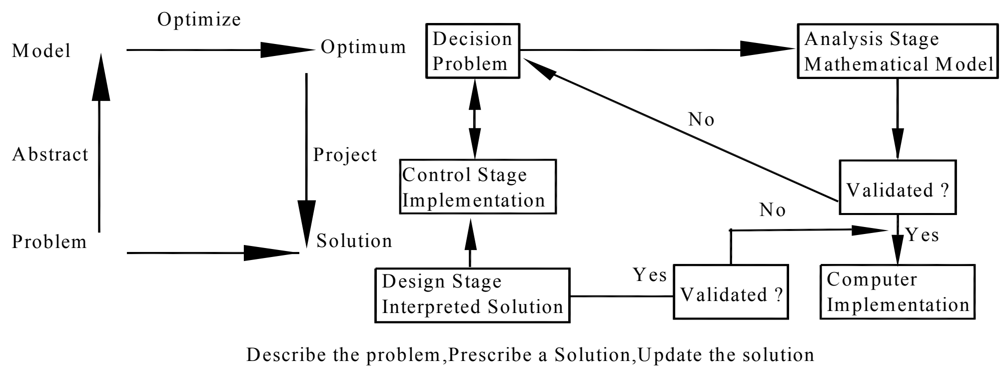


Figure 38 Procedure of optimization (Modified from Eudoxus Systems of Mathematical Programming 1995 – 2003)

The advantage of mathematical models is that they can be analyzed in a precise way by means of mathematical theory and algorithms. However, the sheer amount of computational work needed for solving the models restricts their use to qualitative analysis and to very small and simple instances. The development of algorithms like the Runge - Kutta method or the Fast Fourier Transform made complex models accessible to computers. In the beginning of the twentieth century human workers were used as "computers", and the problem size was still very limited. It was possible for the first time to use mathematical modelling for solving practical problems of significant size. The improvements in computer technology in the years since and the enormous gain in storage capacity and speed have made mathematical modelling increasingly attractive for the military and industry, and a special class of problems, optimization problems, have become very important. The success in solving real world problems increased the demand for better and more complex models, and the modelling process itself was investigated (Schichl, 2000).

The structure of the models has changed throughout history, as the mathematical community gained increasing insight into the foundations of mathematics and formal logic. The introduction of variables, function spaces, and of all the mathematical structural theory has made mathematical models increasingly formal. To date a mathematical model consists of concepts like variables: These represent unknown or changing parts of the model, e.g., whether to take a decision or not (decision variable), how much of a given product is being produced, the thickness of a beam in the design of a ceiling, an unknown function in a partial differential equation, an unknown operator in some equation in infinite dimensional spaces as they are used in the formulation of quantum field theory (Schichl, 2000). Model processing is depicted in Figure 39.

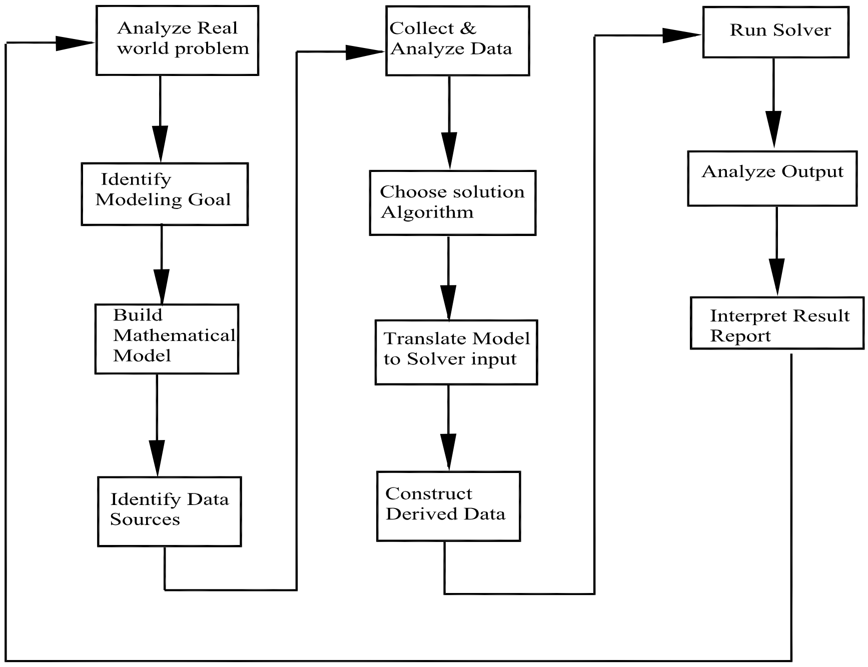


Figure 39 Modelling cycle.

Sourec: - Schichl (2000).

5. 2 Nonlinear Optimization

Nonlinearity is a key characteristic of a vast range of objects, formations and processes in nature and in society. Consequently, nonlinear descriptive models are relevant in many areas of the sciences and engineering. Managing nonlinear systems leads to nonlinear optimization – a subject that has been of great practical interest, at least since the beginnings of mathematical programming. Algorithmic advances and progress in computer technology have enabled the development of sophisticated nonlinear optimization software implementations. A detailed review of local and global optimization algorithms is provided by Bliek *et al.* (2001). Without prior information that provides a suitable starting point, even the best local search methods encounter difficulties in solving general nonlinear models. Such methods will typically find only a local solution (when better solutions may exist), or they may return a locally infeasible result in (globally) feasible models. Clearly, if one i) does not have sufficient insight to guarantee an essentially convex model structure, and ii) does not have access to a good starting point that will lead to the best possible solution, then the application of a global scope search strategy becomes desirable. High-quality local optimization software has been in use for decades with considerable success. A global scope search, however, can bring tangible benefits to both model development (by enabling more general and thereby perhaps more realistic formulations) and solution (by making possible global search when it is not guaranteed that local search will suffice). The field of global optimization (GO) has been gaining increasing attention in the past few decades, and in recent years it has reached a certain level of maturity. The key theoretical developments have been followed by solution algorithms and their software implementations (Pinter, 2003c).

5. 3 Global optimization

A branch of mathematics which encompasses many diverse areas of minimization and optimization, Optimization theory is the more modern term for operations research. Optimization theory includes the calculus of variations, control theory, complex optimization theory, decision theory, game theory, linear programming, Markove chains, network analysis, optimization theory, queuing systems (Dill *et al*.,1997).

The objective of global optimization is to find the globally best solution of (possibly nonlinear) models, in the (possible or known) presence of multiple local optima. Formally, global optimization seeks global solutions of a constrained optimization model. Nonlinear models are everywhere in many applications, e.g., in advanced engineering design, biotechnology, data analysis, environmental management, financial planning, process control, risk management, scientific modelling, and others. Their solution often requires a global search approach (Dill et al., 1997).

MATERIALS AND METHODS

The objective of the study was to optimize the design parameters in terms of the total specific energy requirements of the rotary blades. It has been expressed as the total power requirement per unit volume of soil tilled. The pushing power exerted by the operator of the walking tractor, the cutting and loosing of the soil slice, the overcoming of soil - metal friction and the throwing of the cut soil slice by the centrifugal action of the rotary blades are total power requirements. Therefore the model was formulated in terms of integrated constraint parameters of the rotary blades.

1. Mathematical model selection

The model selection technique was determined with relative to the objective of the study and on the facts of the rotary tiller power requirements studied in different periods, soil types and conditions as follows:

Dalin and Pavlov (1950) presented a general theoretical equation of rotary tiller to predict total power requirement as:

$$P_{Total} = P_{Cut} + P_{Throw} + P_{Loss} + P_{mf} + P_{Push} \tag{67}$$

Where

P_{Total} = total power requirement, kW,

P_{Cut} = cutting power requirement, kW,

P_{Throw} = throwing power the cut soil slice power requirement, kW,

P_{Loss} = Power loss in the power train, kW,

P_{mf} = overcoming soil-metal friction power requirement, kW,

P_{Push} = pushing power requirement, kW.

This model is defined but was not validated in terms of total specific energy requirement. Sohne and Eggenmuller (1959) used a general specific energy model for evaluating the efficiency of rotary cultivator and other cultivating tools is given by:

$$E_W = \frac{E_E}{V_{ST}} \tag{68}$$

Where

E_W = specific work, $N.m/m^3$,

E_E = energy expended $N.m/m^2$,

V_{ST} = volume of soil tilled, m^3.

Bernacki *et al.* (1972) formulated specific work under the description of torque per unit volume of soil tilled as:

$$E_{SW} = \frac{2pF_T}{ZL_d B_W L_B} + \frac{0.1R_X}{B_W} \tag{69}$$

Where

E_{SW} = specific work, $N.m/m^3$,

E_E = torque on the shaft of the rotary blade, $N.m$,

V_{ST} = volume of soil tilled, m^3,

R_X = pushing force of the machine, component of the cutting resistance parallel to the direction of machine travel, *N*,

Z = number of working elements operating in one plane,

B_W = cutting width, *m*,

L_d = cutting depth, *m*,

L_B = tillage pitch, *m*.

As maintained above both models, however, are undefined in terms of total specific energy requirement. Similarly, Sineokov (1977) formulated total power requirement of a rotary cultivator as:

$$P_{Total} = P_{Cutl} + P_{KE} + (1-\eta)(P_{Cutl} + P_{KE}) + \frac{V_f}{75}(\mu Q_z - R_X) \qquad (70)$$

Where

P_{Total} = total power requirement, *kW*,

P_{Cut} = cutting power requirement, *kW*,

P_{KE} = power for imparting kinetic energy to the soil, *kW*,

Q_z = vertical load on the supporting wheels of the rotary cultivator, *N*,

V_f = machine forward velocity, *m/s*,

η = efficiency of the drive, percent,

μ = rolling coefficient of the support wheels of the rotary cultivator, percent.

As mentioned in the above model, the pushing, and the over coming soil-metal friction power requirement and the volume of soil tilled were excluded. Saraswat (1987) under lateritic unsaturated sandy clay loam in soil bin laboratory evaluated the performance efficiency of "Pick"-shaped rotary blade in terms of specific energy requirement, pushing and rotary power requirement per unit of volume of soil tilled as:

$$E_{TSP} = \frac{P_{Push} + P_{Rotary}}{V_{ST}} \qquad (71)$$

Where

E_{SP} = specific energy requirement, kJ/m^3,

P_{Push} = pushing power requirement by an operator of the walking tractor, *kW*,

P_{Rotary} = rotary power requirement, *kW*,

V_{ST} = volume of soil tilled, m^3.

However, as mentioned in the above model, the cutting and the soil-metal friction were not described significantly. Gupta and Visvanathan (1993) did experiment under saturated soil condition to predict power requirement of "L"-shaped rotary blade. The power requirement were, cutting the soil $\left(P_{Cut}\right)$, throwing the cut soil slices by centrifugal action of the blade $\left(P_{Throw}\right)$, overcoming soil-metal friction $\left(P_{mf}\right)$, soil-soil sliding friction $\left(P_{Sf}\right)$ as:

$$P_{Total} = P_{Cut} + P_{Throw} + P_{mf} + +P_{Sf} + \text{idle power} \tag{72}$$

Where

P_{Total} = total power requirement, kW,

P_{Cut} = cutting power requirement, kW,

P_{Throw} = throwing power the cut soil slice power requirement, kW,

P_{mf} = overcoming soil-metal friction power requirement, kW,

P_{Sf} = soil-soil sliding friction power requirement, kW,

In the above mentioned model the pushing power $\left(P_{Push}\right)$ and volume of soil tilled $\left(V_{ST}\right)$ were excluded. Therefore, the model has not been governed in terms of total specific energy requirement. Dalin and Pavlov's (1950) theoretical model is more describable among the above mentioned models which includes power requirements for cutting $\left(P_{Cut}\right)$, throwing the cut soil slice $\left(P_{Throw}\right)$, power loss $\left(P_{Loss}\right)$, overcoming soil-metal friction $\left(P_{mf}\right)$ and pushing power $\left(P_{Push}\right)$. Therefore, this model is utilised in this study.

2. Mathematical model modification

The total specific energy requirement (power per unit volume of soil tilled) modification model has been defined as a function of pushing $\left(P_{Push}\right)$, cutting and loosening the soil slice $\left(P_{Cut}\right)$, over coming soil- metal-friction $\left(P_{mf}\right)$ and throwing the cut soil slice $\left(P_{Throw}\right)$ power requirement and volume of soil tilled $\left(V_{ST}\right)$. The modified total specific energy requirement for the various rotary blades is exhibited as

$$E_{TSP} = \frac{P_{Push} + P_{Cut} + P_{mf} + P_{Throw}}{V_{ST}} \tag{73}$$

Hence, P_{Push} is pushing power requirement, kW,

$$P_{Push} = \frac{7161.96\left(V_f P_e\right)\eta_C \eta_Z}{RNcos(\varphi_1)}\left[sin(\alpha)cos(\varphi_1) + cos(\alpha)sin(\varphi_1)\right] \tag{74}$$

(Modified from Saraswat, 1957)

Where V_f is the machine forward velocity $\left(V_f = 0.20 \text{ - } 30 \text{ m /s, Sakai, 2000}\right)$, P_e is the engine horse power of power tiller (Mamasari, 1998) , P_e = 5.0 -12.16 hp, η_C is the power tiller efficiency and η_Z is coefficient including a reverse of power tiller power (Bernacki *et al.,* 1972), η_C = 0.9 , η_Z = 0.75,; R is the rotor radius (m), N is the rotational velocity (rpm), α is the angle of direction , α = 42^0 and φ_1 is the angle of periphery, φ_1 = 15^0 (Sineokov,1977).

$$P_{Cut} = K_{SP} B_W L_d V_f \tag{75}$$

(Modified from Dalin and Pavlov, 1957)

Where P_{Cut} is the cutting power requirement, K_{SP} is specific soil resistance

resistance$\left(K_{SP}\right.$ = 7000 kg /m^2 for firm soil ; Bernacki, *et al*., 1972 $\left.\right)$, B_W width of soil cut(m), and L_d is depth of soil cut(m).

$$P_{mf} = L_d R V_f B_W S_{PW} \mu_k \tag{76}$$

(Modified from Gupta and Visvanathan, 1993)

Where P_{mf} is the over coming soil –metal friction torque requirement(kW), S_{PW} is the dry soil bulk density$\left(S_{PW} = 1700 kg/m^3\right)$, μ_K is the kinetic coefficient of soil –metal friction, $\mu_K = \dfrac{1.09}{\sqrt{0.105RN}}$ (Gupta and Visvanathan, 1993). The kinetic coefficient of soil –metal friction occurs between the uncut soil and the metal surface under any soil condition by soil amendment implements. Besides, the value of the kinetic coefficient of soil –metal friction has been calculated using the following equation as:

$$P_{Throw} = \frac{0.219RNL_d V_f B_W S_{PW}\left(3R-\right)L_d}{GZ} \tag{77}$$

(Modified from Gupta and Visvanathan, 1993)

Where P_{Throw} is the throwing the cut soil slice torque requirement(kW), Z is the number of blades on the drum, and G is the acceleration due to gravity. The total specific energy requirement $\left(E_{TSP}\right)$ used for the various rotary blades was modified from Eqn (73) and it's derivation, Eqn (74) to Eqn (77), is written as the following:

$$E_{TSP} = \left\{\left(\frac{4834.323\left(V_f P_e\right)sin\left(\alpha+\varphi_1\right)}{RNcos\left(\varphi_1\right)}\right) + B_W L_d S_{PW}\left(\frac{K_{SP}}{S_{PW}} + \frac{1.09R}{\sqrt{0.0105RN}} + \frac{0.219RN\left(3R-L_d\right)}{G}\right)\right\}/V_{ST} \tag{78}$$

Where V_{ST} is the volume of soil tilled per second, 10^{-6}m^3. V_{ST} was calculated on equation cited by Bernacki et al. (1972); $V_{ST} = B_W L_d V_f$.

3. Mathematical model optimization

The mathematical model optimization has been defined as a function of design parameters and other rotary tiller working parameters as a function (f) of soil condition, rotational velocity, forward velocity, rotor radius, depth of tillage, cutting width of the rotary blade, dry bulk density, specific soil resistance, engine horse power of power tiller, angle of direction and angle of peripheral (Eqn. 78)

$$f\left(N, V_f, V_P, R, L_d, B_W, S_{PW}, K_{SP}, P_e, \alpha, \varphi_1\right) \qquad (79)$$

Where

N = rotational velocity, rpm,

V_f = machine forward velocity, m/sec,

R = rotor radius, m,

L_d = depth of tillage, m,

B_W = cutting width, m,

S_{PW} = dry bulk density, kg/m^3,

K_{SP} = specific soil resistance,

α = angle of direction,

φ_1 = angle of peripheral,

G = acceleration due to gravity, m/s^2 ,

P_e = maximum engine horse power of power tiller, hp.

4. Formulating the optimization model

To obtain the optimum solutions of system and design parameters for the determination of specific energy and fuel consumption requirements of various rotary blades of power tiller, non-linear programming type of problems was fitted. Non-linear programming type of problems was exhibited by Intaravichai (1994) as the following:

$$\text{Minize } f(X) \tag{80}$$

$$\text{Subject to} \quad \mathrm{L_i} \le g_i \le U_i \quad \mathrm{i} = 1 \ldots \mathrm{m}$$

$$\mathrm{L_{m+i}} \le x_i \le U_{m+i} \quad \mathrm{i} = 1 \ldots \mathrm{n}$$

$$\mathrm{L_i} \le U_i$$

Where:

L : Lower bound of the function i or variable i

U : Upper bound of the function i or variable i

m : Number of constraints

n : Number of variables.

Total specific energy model has been under taken based on Eqn (78) and computed as:

Objective function

$$Min\, E_{TSP} = \left\{ \left(\frac{4834.323\left(V_f P_e\right) sin\left(\alpha + \varphi_1\right)}{RNcos\left(\varphi_1\right)} \right) + B_W L_d S_{PW} \left(\frac{K_{SP}}{S_{PW}} + \frac{1.09R}{\sqrt{0.0105RN}} + \frac{0.219RN\left(3R - L_d\right)}{G} \right) \right\} / V_{ST}$$

Subject to:

$0.20 \le V_f \le 0.30$, $5.0 \le P_e \le 12.16$, $35^0 \le \alpha \le 42^0$, $10^0 \le \varphi_1 \le 15^0$, $150 \le N \le 300$, $S_{PW} = 1700$, $5000 \le K_{SP} \le 7000$, $G = 9.81 m/s^2$, For the "Pick"-shaped rotary blade

$0.17 \le R \le 0.220$, $0.01 \le B_W \le 0.022$, $0.050 \le L_d \le 0.100$,

For the "C"-shaped rotary blade

$0.245 \le R \le 0.250$, $0.035 \le B_W \le 0.045$, $0.050 \le L_d \le 0.150$,

For the "I"-shaped rotary blade

$0.200 \le R \le 0.225$, $0.035 \le B_W \le 0.065$, $0.050 \le L_d \le 0.150$,

For the "L"-shaped rotary blade

$0.235 \le R \le 0.260$, $0.045 \le B_W \le 0.080$, $0.050 \le L_d \le 0.150$,

For the "J"-shaped rotary blade

$0.235 \le R \le 0.250$; $0.040 \le B_W \le 0.060$; $0.050 \le L_d \le 0.240$,

2 Fuel consumption of rotary blade

2. 1 Fuel consumption models

There are many rotary tiller fuel consumption models reported by a number of researchers and investigators. A fuel consumption model for a particular operation was exhibited by Grisso *et al.* (2001) as:

$$F_C = F_{SPC} \times P_{Total} \tag{81}$$

Where F_C is estimated fuel consumption for a particular operation, l/hr ; F_{SPC} is specific volumetric fuel consumption for the given tractor, l/kW and P_{Total} is total tractor power (PTO equivalent) for the particular operation, kW .

As cited by the American Society of Agricultural Engineers standards (ASAE, 2001) the average fuel consumption of a tractor operating under a range of load conditions was determined as:

$$F_C = 2.64X + 3.91 - 0.203\sqrt{738X + 173} \tag{82}$$

Where F_C is fuel consumption in l/kWh; X is ratio of equivalent PTO power required by operation divided by the maximum available PTO.

Grisso *et al.* (2004) developed a generalized fuel consumption model and it is computed as the following equation:

$$F_C = (0.22X + 0.096)(1 - (0.0045XN_{\text{Red}} + 0.00877N_{\text{Red}}))PTO \tag{83}$$

Where F_C is diesel fuel consumption at partial load and full/reduced throttle, l/hr; X is ratio of equivalent PTO power to rated PTO power, decimal; N_{Red} is percentage of reduced engine speed for a partial load from full throttle, % and PTO is rated PTO power, *kW*.

Kheiralla *et al.* (2004) formulated a fuel consumption model of rotary tiller in Serdang sandy clay loam soil as:

$$F_C = 0.2156PTO + 6.2347 \tag{84}$$

Suresh and Varshney (2005) calculated the specific fuel consumption of power tiller engine by dividing the fuel consumption (l/hr) with drawbar power (*kW*) and formulated the general form of an empirical equation to relate fuel consumption and drawbar power on tilled land as follows:

$$F_{SPC} = a + bP + cP^2 + eP^2 \quad (85)$$

$$F_C = a + bP \quad (86)$$

Where F_{SPC} is specific fuel consumption, l/kWh, F_C is fuel consumption, l/hr; P is Drawbar power, kW, a, b, c and e are parameters.

As in the above mentioned models, the one developed by Kheiralla *et al*. (2004) that described only one configuration of rotary blades was employed as it was used for sandy clay loam soil which was similar to this study.

2. 1. 2. Fuel consumption model selection

1. As cited in the literature review, the soil type of the study was similar to the Kheiralla *et al.* (2004) rotary tiller fuel consumption model which was tested under unsaturated sandy clay loam.

2. Power take-off (PTO) in the Kheiralla *et al.* (2004) fuel consumption model was converted into system parameters with corresponding fuel consumption requirements following the description by Berger *et al.* (1963), Bernacki *et al.* (1972) has used for various rotary blades in the present study.

Based on the system parameters with corresponding fuel consumption requirement the performance of each rotary blade was evaluated in terms of fuel consumption requirement. Detailed analysis is exhibited as the following

From Kheiralla *et al.* (2004) the empirical model constrained by PTO power was best expressed by the following regression equation.

$$F_C = 0.2156PTO + 6.2347 \quad (87)$$

Where F_C is fuel consumption rate l/hr, PTO is Power take–off, kW. Further to the investigation of the model, Barger, *et al*. (1963) PTO was defined by the torque and rotational velocity of the rotary machine. The rotary power is exhibited by the following equation:

$$PTO = 2\pi NF_T / 60000 \quad (88)$$

Where PTO is power –take off supplied to the rotor of rotary tiller, kW, N is rotational velocity, rev/min and F_T is rotor torque, Nm. Consequently, the rotor torque was defined by peak peripheral force $K_P = F_P C_V$ (Bernacki, et al., 1972), C_V is factor = 1.5 for stone less soil, C_V = 2 for stony soil. Therefore C_V = 2 for the preset study. The peak moment on the rotor equals to the torque, $F_T = K_P R = 2\text{R} F_P$. Peripheral force F_P was exhibited by Bernacki et al. (1972) in the following equation:

$$F_P = \left(\frac{750P_e}{V_P}\right)\eta_c\eta_Z \quad (89)$$

Where F_P is the peripheral force, N, P_e is the engine power of the power tiller, P_e = 6.45 hp; V_P is the peripheral speed of the rotary blade, m/s; η_C is the power tiller efficiency mounting to about 0.9 for machines with concurrent revolutions, η_Z is the coefficient including a reverse of power tiller power mounting to 0.7 to 0.8,

The volume of soil tilled and velocity ratio were given (Hendrick and Gill, 1971) as

$$V_{ST} = B_W L_d V_f \quad (90)$$

Where V_{ST} is volume of soil tilled per second, $10^{-6}\mathrm{m}^3$, B_W is cutting width, m, L_d is depth of tillage, m; V_f is machine forward velocity, m/s,

Velocity ratio, $\lambda = \dfrac{V_P}{V_f}$

$\lambda V_f = V_P$, putting $V_f = \dfrac{V_{ST}}{B_W L_d}$ in λV_f; V_P has become $\dfrac{\lambda V_{ST}}{B_W L_d}$;

Substituting $\dfrac{\lambda V_{ST}}{B_W L_d}$ into Eqn (89) the peripheral force is obtained as $\left(\dfrac{750 P_e B_W L_d}{\lambda V_{ST}}\right)\eta_c \eta_Z$.

Then,

$$F_T = 2\mathrm{R}F_P = \left(\frac{1500 P_e B_W L_d R}{10^{-6}\lambda V_{ST}}\right)\eta_c \eta_Z \tag{91}$$

Substituting $\left(\dfrac{1500 P_e B_W L_d R}{10^{-6}\lambda V_{ST}}\right)\eta_c \eta_Z$ in place of F_T in Eqn (88), the PTO become as

$$PTO = \frac{539691.42 N B_W L_d R}{\lambda V_{ST}} \tag{92}$$

Substituting Eqn (92) in to Eqn (87), the modified fuel consumption model has the formation:

$$F_C = 116357.47(N * R * Ld * Bw / \lambda * Vst) + 6.2347 \tag{93}$$

Table 3 Working parameters of various rotary blades

Types of rotary blade	Rotational velocity (N) rpm	Rotor radius (R) m	Depth o tillage (L_d) m	Cutting width (B_W) m
"Pick"-shaped	150 to 300	0.17 to 0.220	0.050 to 0.100	0.01 to 0.022
"C"-shaped	150 to 300	0.245 t0 0.250	0.050 to 0.150	0.035 to 0.045
"I"-shaped	150 to 300	0.200 to 0.225	0.050 to 0.150	0.035 to 0.065
"L"-shaped	150 to 300	0.235 to 0.260	0.050 to 0.150	0.045 to 0.080
"J"-shaped	150 to 300	0.235 to 0.250	0.050 to 0.240	0.040 to 0.060

3 . Solving for optimization problems

Non-linear optimization theory has been applied under this study. Data were analyzed using the LINDO Systems (2005). LINDO Solver Suite, version 10.0 to optimize integrated system and design parameters of various rotary blades.

RESULTS AND DISCUSSION

This chapter deals with the optimization results of total specific energy requirement and fuel consumption requirements of the “pick”- shaped rotary blade. As per proposed objectives of the study, the results are presented in the following as:

1. Optimal solutions of design parameters and simulated specific energy requirements of various rotary blades

The optimal solutions of design parameters and corresponding specific energy requirements of individual rotary blades are presented in Table 4. From the results, at the same forward and rotational velocity, the angle of rotation and angle of periphery, the specific soil resistance and dry soil bulk density, the specific energy requirement was predicted to be 231.61, 160.72, 196.87, 168.56 and 167.560 kJ/m^3 for the “pick”, the “C”, the “I”, the “L” and the “J”- shaped rotary blade, respectively. The highest specific energy requirement was exhibited by the “Pick”- shaped blade and lowest by the “C”- shaped blade. The higher the specific energy requirement, the lower volume of soil tilled and the most effective and optimum soil tillage operational cost is achieved. Compared to another study in the same soil condition, the specific energy requirement per volume of soil tilled by the “pick”- shaped blade was exhibited at 1900 kJ/m^3 which was higher by 87.81 % than the “pick”- shaped blade of the present study. Therefore, the study could be used to develop rotary tiller under local conditions..

On the other hand Beeny and Greing (1965) reported that the lower values of the specific work requirement with a small range of volume of soil worked per rotor blade cut and these volumes were dependant only on the combination of depth and bite length of the blade over the range of values measured.

Table 4 Optimal solutions of design parameters with corresponding total specific energy requirement of individual rotary blade.

Type of rotary blade	V_f m/s	P_e hp	R m	B_W m	L_d m	N rpm	α^0	$\varphi_1{}^0$	S_{PW} Kg/m^3	K_{SP} Kg/m^2	V_{ST} $10^{-6}\,m$	E_{TSp} kJ/m^3
"Pick" shaped	0.20	5.00	0.17	0.010	0.100	150	37.65	10.05	1700	5000	200	231.61
"C"-shaped	0.20	5.00	0.245	0.035	0.150	150	37.65	10.05	1700	5000	1050	160.72
"I"-shaped	0.20	5.00	0.200	0.035	0.150	150	37.65	10.05	1700	5000	1050	196.87
"L"-shaped	0.20	5.00	0.235	0.045	0.150	150	37.65	10.05	1700	5000	1300	168.56
"J"-shaped	0.20	5.00	0.235	0.040	0.240	150	37.65	10.05	1700	5000	1900	167.56

2. The fuel consumption requirement of the "Pick"-shaped rotary blade

Optimum design parameters have influenced fuel consumption requirement of the "Pick"-shaped rotary blade. The optimum design parameters are shown in Table 4. The values of optimum parameters are listed as:

$$N^* = 150 \text{ rpm}$$

$$R^* = 0.17\text{m}$$

$$L_d{}^* = 0.100 \text{ m}$$

$$B_W{}^* = 0.010 \text{ m}$$

$$\lambda^* = 13.35$$

$$V_{ST}{}^* = 200 \ m^3$$

Using Equation(93), the fuel consumption of the "Pick"-shaped rotary blade is calculated as:

$$F_C = 116357.47(150 * 0.170 * 0.100 * 0.010/(13.35 * 200) + 6.2347$$

$$= 7.35 \ L/h$$

The specific fuel consumption requirement has been calculated using conversation equation (81) as:

$$F_{spc} = \frac{F_C}{P_{Total}} = \frac{7.35}{3.7} = 2.0 \ L/KWh$$

$$KWh = 3600kJ$$

$$\therefore \ F_{spc} = 1800kJ/L$$

The constracutiral details of the optimum configuration of the "pick"- shaped rotary blade has been validated in term of logarithmic spiral and the radius vectors for different values of θ were calculated using equation(48). The values of logarithm curvature are shown in Table 5. With the help of the values, the logarithmic curvature was constructed as shown in Figure 40.

Table 5 Radius of vectors at different magnitude of angle of polar coordinate for "Pick"- shaped rotary blade.

θ	0^0	10^0	20^0	30^0	40^0
R (mm)	170	160	150	140	130

Finally, the optimum configuration of the rotary blade under this study can be shown as followed:

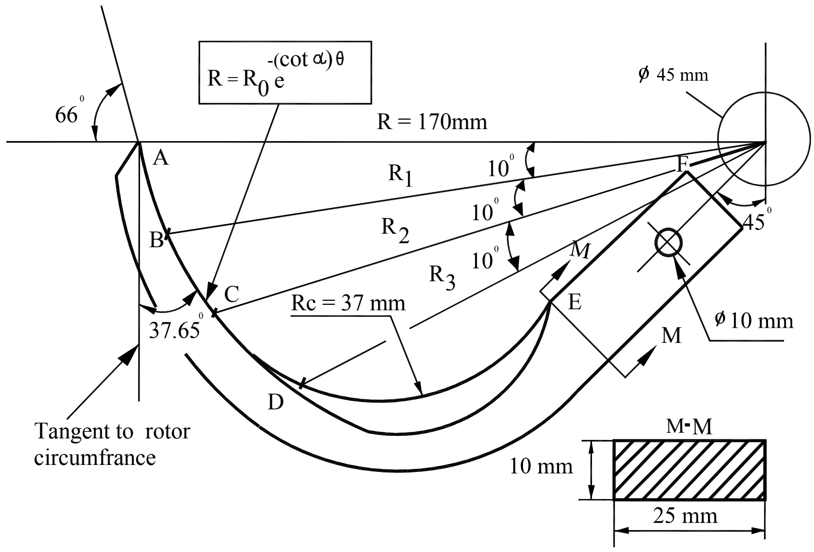


Figure 40 The obtained optimum configuration of the "pick"- shaped rotary blade

CONCLUSION AND RECOMMENDATION

In spite of negative draft rotary tiller are more power consumers other passive implements. Moreover, they are designed complex. The rotary blades of the rotary machines execute a complex motion comprising of relative –rotary motion around the axis of the rotor shaft with rotational velocity and translatory forward motion. Subsequently, they have commonly basic design parameters which have influenced power requirements from the initial to the final soil condition during the process of soil tillage operations. It is also remarkably, difficult to select the power requirement to achieve soil tillage operational cost. Therefore the study has focused on optimum design parameters in terms of total specific energy requirements. Specific energy requirement is also one of the performance measurement criteria of tillage implements. The specific energy consumption requirements of various rotary blades, namely, for a “Pick”, a “C”, an “I” an “L” and a “J”- shaped rotary blade under the same working parameters and soil condition was suggested as the following:

1. The highest specific energy requirement was exhibited by “Pick”- shaped blade and the lowest by the “C”- shaped blade. The higher the specific energy requirement, the lower the volume of soil tilled and the most effective and optimum soil tillage operational cost was observed. Compared to another study in the same soil condition, the specific energy requirement per volume of soil tilled by the “Pick”- shaped blade was exhibited $1900\,kJ/m^3$ which was higher by 87.81 % than the “pick”- shaped blade of the present study. Consequently, the fuel consumption requirement of the “pick”- shaped blade was found to be $1800kJ/L$.
2. The optimum engine horse power , forward velocity and rotational velocity was predicted to be 5 hp, 0.20 m/s and 150 rpm, respectively for a “Pick”, a “C”, an “I” an “L” and a “J”- shaped rotary blade

3. The angle of direction, the angle of periphery, the dry bulk density, the specific soil resistance influencing total specific energy requirement of the various rotary blade was predicted to be 37.65^0, 10.05^0, 1700 kg/m^3 and 5000 kg/m^2, respectively.
4. Obtained optimum design parameters interims of total specific energy requirements of the rotary blades.
5. Predicted fuel consumption requirement based on integrated optimum design parameters of the "Pick – shaped" blade.
6. Predicted optimum configuration of the "Pick – shaped" blade with related optimum rotor radius
7. A walking tractor is a simple, multi-purpose hand tractor designed and is easily made under regional rural technology promotion centres of Ethiopia. It will be available for small holders for land preparation. The potential of the walking tractor will make available hilly areas and narrow bench terraces that are difficult to negotiate, where farmers cannot afford to buy tractors. It creates the least disturbance of the soil structure, can cross over the bunds separating the small holdings, does not sink in wet paddy fields, can climb steep hill slopes and conveniently operates in bench terraces and can manoeuvre the narrow pathways of rural areas.

Therefore, the study could be used to develop a rotary tiller under Ethiopian local condition.

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