CHAPTER 1

INTRODUCTION

1.1 Background

The rotavator (derived from *rota*ry culti*vator*) or rotary tiller is a tillage tool primarily comprising L-shaped blades mounted on flanges, which are attached to a shaft that is driven by the tractor power-take-off (PTO) shaft. It is an active tillage tool that processes the soil at a speed that is different from the forward travel speed of the tractor. With respect to depth of tillage, the rotavator is unique in that during its operation, the actual depth of tillage for each blade changes throughout the rotational path of the cutting blade.

Historical records have traced the cultivation of land, in Europe, with active tillers back almost 170 years when steam engine tractors were introduced in agriculture (Bernacki, Haman & Kanafojski, 1972). Since their original design in the 1840s, along with the application of steam power to agriculture (Spence as cited by Hendrick & Gill, 1971a), they have been devices that inspired both enthusiasm and controversy. The controversy still exists but a number of inherent advantages of tillage tools that transmit power directly to the soil require that they be considered as an alternative to tools that are drawn through the soil (Hendrick & Gill, 1971a).

The entry of the rotavator in agriculture is relatively recent in comparison to hand tools and animal-drawn tillage tools. The delayed entry of rotavators is attributed to the lack of suitable sources of power, prior to the development of the steam powered tractor. The original rotavators were intended for deep land preparation as an alternative to the drawn or passive tillage tools. They were heavy machines that expended excessive energy per unit mass of processed soil at the intended depths of operation. These rotavators suffered frequent mechanical breakdowns during tillage operations (Bernacki *et al.*, 1972). The combination of the excessive power demand and frequent breakdowns led to their rejection as an alternative to the passive tools for primary land preparation.

In tillage, rotavator blades may be rotated on vertical-axis or horizontal-axis rotor shafts (Figure1.1). The resulting texture of rotavator-tilled soil is a function of the soil condition, blade kinematics and soil flow dynamics (Kinzel, Holmes & Huber, 1981). The soil flow dynamics depends on whether the rotor axis of the cultivator is horizontal or vertical; and on the direction of rotation of the blades. For the horizontal rotor axis, the blades may be rotated in the down-cut or up-cut direction; while for the vertical axis, the direction of rotation may be clockwise or anti-clockwise. In general, rotavators work on the horizontal axis rotor; and seldom on a vertical or series of vertical axis rotors (Bukhari, Bukhari, Leghari & Memon, 1996).

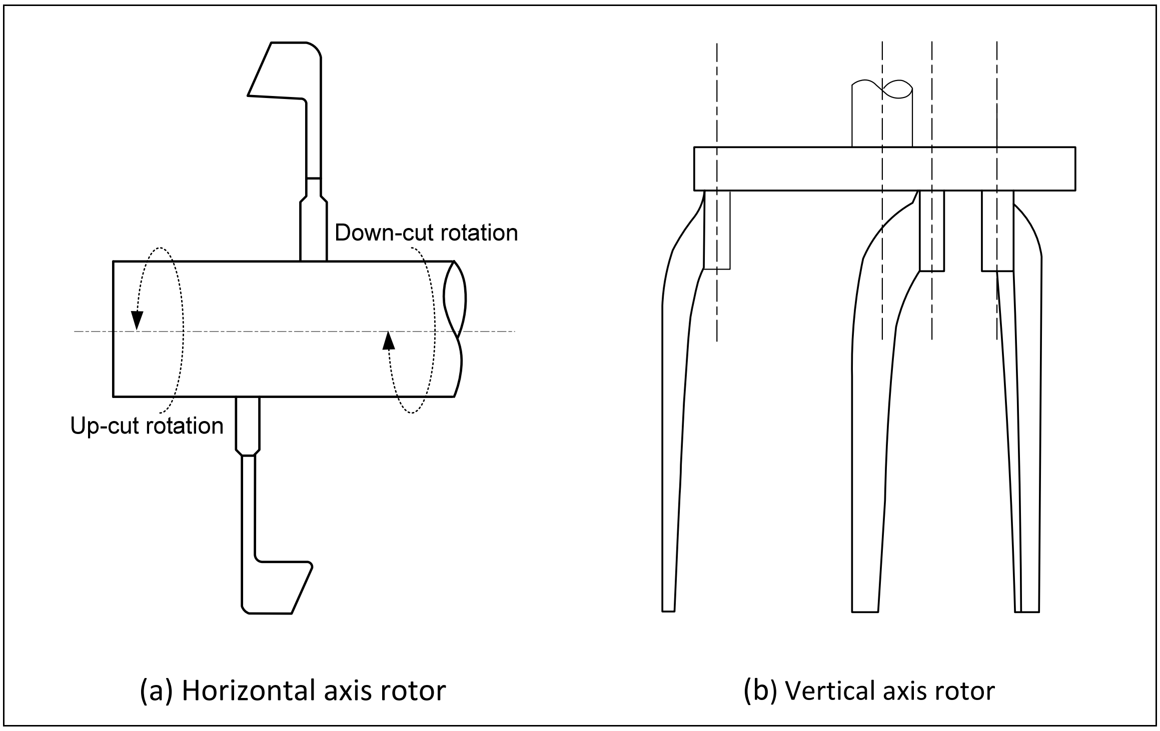

Figure 1.1: Illustrations of horizontal and vertical axis rotavators (Bernacki *et al.*, 1972)

Down-cut rotavators may exert a forward force/thrust to the tractor driveline while the up-cut ones pull the tractor rearwards (Bernacki *et al.,* 1972). The forward thrust, generated by the down-cut rotavator, though possibly detrimental to the tractor driveline, offers the following advantages in tillage (Manian & Kathirvel, 2001; Shinners, Wilkes & England, 1993):

- The thrust force generated by the rotavator blades can be used as traction aid and contributes significantly to the reduction of the rolling resistance of the tractor.
- Reduced draught of rotavators results in less wheel slip at the tractor tyre - soil interface, thus improving field productivity and efficiency.
- Reduced draught of rotavators allows the use of lighter tractors, thus reducing the soil compaction levels and the purchase price of tractors required to operate the tillers.
- Reduced draught of rotavators allows tillage operations to be performed in more difficult traction conditions.

In addition to the aforementioned advantages, the rotavator also mixes and pulverizes the tilled soil well; resulting in a good clod size distribution. The number of tillage passes required to achieve an acceptable tilth quality, using rotavators, is also significantly reduced (Destan & Houmy, 1990) in comparison to the series of operations that would result in the same tilth quality with the use of passive tools. .

In situations with limited traction conditions and/or where thorough mixing of the tilled soil layer is a requisite, the down-cut rotavator is the preferred tillage implement. This preference is due to its ability to generate a forward thrust that aid in traction under such conditions, and the attainment of a well-mixed tilled soil layer. Consequently, the down-cut rotavator is applied widely in the preparation of paddy rice fields in the Asian subcontinent. Paddy rice fields are characterized by limited traction conditions, and the resultant tilled soil needs to be well-mixed.

In spite of these advantages, the down-cutting rotavator is not the implement of choice for land preparation due its perceived excessive specific energy requirements. To date, the application of the rotavator in crop production systems, that require the manipulation of the soil at deeper depths, has been hindered by its excessive energy demands. Because of this, all the commercially available rotavators are presently designed for shallow tillage. This view has been controversial and a number of studies have been conducted with the object of comparing the energy demands for the realization of an acceptable seedbed. According to Manian and Kathirvel (2001), the view of the excessive energy consumption by rotavators only holds when the energy consumption of the rotavator is compared to the energy consumed by individual passive tillage tools, without considering the quality of the resultant tilth and the number of tillage operations required under conventional tillage. When the total energy demand for producing an acceptable tilth is considered, the series of passive tillage operations needed to realise the same tilth consumes more specific energy than the rotavators (Manian & Kathirvel, 2001; Prasad, 1996). The series of tillage operations for passive tools also require more time and additional resources, thus increasing the production costs of crops.

With such desirable characteristics, there is need to find ways of utilizing the rotavator in the preparation of land for other crops, besides rice and vegetables. In particular, the forward thrust generated could be useful in other land preparation conditions where high traction is required, such as the deep tillage for the rehabilitation of compressed arable land layers and the incorporation of soil amendment materials due to it its superior mixing ability. In Africa, crops that can greatly benefit from the use of rotavators include horticultural crops, orchards, and perennial tree crops such as coffee that require periodical rehabilitation of soils they stand on. The need for rehabilitation arises from the compaction of the root-zone layer due to the numerous surface operations inherent in perennial crop production systems. Also there may be a need to retrieve plant nutrients that might be leached beyond the reach of root zone.

According to Hendrick and Gill (1971a, b, c), power requirements of rotavators might be reduced by considering the relationships between the tiller design and its operational parameters. These parameters include the direction of rotation of the blades, depth of tillage, ratio of peripheral to the forward speed (λ) and the soil condition.

1.2 General hypothesis and model

From the profitable farming perspective, the rotavator hold immense potential for reducing the cost of production of crops especially if methods for reducing its perceived excessive power requirements in deep tillage can be found. One way of doing this is to carry out studies that would establish the effects of the rotavator design parameters and soil conditions on its performance. Practically, this can be accomplished by designing and fabricating a deep-tilling rotavator with requisite instrumentation to obtain information on the soil-tool interaction system that can be used to study its performance.

In tillage, the performance of tools is determined by their specific draft and energy requirements, and the quality of work. Whereas there is no precise definition of the quality of work, it is generally evaluated by the clod size, the evenness of the operative depth and percentage of plant residue covered after a tillage operation (Srivastava, Goering & Rohrbach, 1993). Though numerous attempts have been made to quantify the performance of rotavators, little is known about how the rotavator design parameters and soil conditions influence the energy requirements and the quality of work due to the empirical nature of the studies on performance (Marenya, du Plessis & Musonda, 2003).

In order to understand the influence of tool design parameters and the influence of soil conditions on performance, Gill and Vanden Berg (1967) emphasized that mathematical description of a tillage process can be accomplished only when all the elements of the tillage process are expressed in a quantitative sense. They gave a hypothetical model to illustrate the factors involved in influencing the desired quality of operation and resulting forces from the design point of view. The hypothetical model they presented considered the initial soil conditions, shape and the manner of movement of the tool as input factors; and tool forces and the final soil condition as output parameters.

In this study, a similar model, in the form of a block diagram (Figure 1.2), is proposed as a basis for gauging the performance of an experimental deep tilling rotavator, fitted with L-shaped blades. The ASAE Standards (2000) defines deep tillage as any tillage operation undertaken at depths in excess of 300 mm. The block diagram includes additional factors, such as the direction of rotation of the blade, bite length and operational depth.

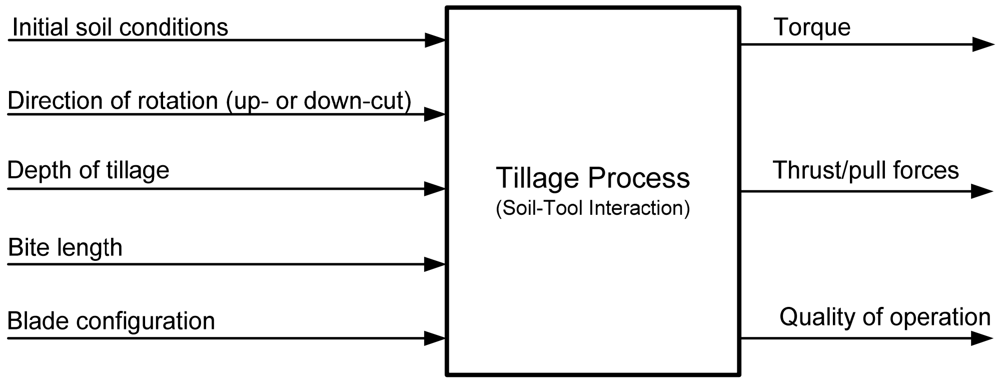

Figure 1.2: Proposed block diagram for the rotavator tillage process model

CHAPTER 2

LITERATURE REVIEW

During rotavator tillage operations various factors affect its energy requirements. These factors (Figure 1.2) can be divided into three categories, *viz*. the soil condition, operational conditions and rotavator configuration. One of the ways for evaluating the performance of such a tillage tool is by determining the tool's total energy requirements. An estimation of the total energy requirements for a tillage tool, such as a rotavator, can be obtained by determining the energy requirements resulting from each group of these factors and their interaction with the tool.

In attempts to quantify the energy requirement of tillage tools, a number of models have been developed that relate the input and output parameters depicted in Figure 1.2 Such models predict the forces acting on soil tillage tools in relation to tool geometry, soil physical properties and the nature of soil disturbance ahead of the tool (Godwin & O'Dogherty, 2006). Therefore, the purpose of this chapter was to review the existing literature on soil parameters, tool geometry and soil-tool interaction models in order to develop a basis for the quantification and prediction of rotavator tillage energy requirements.

2.1 Soil parameters

The soil physical, mechanical and soil dynamic properties, have significant influence on the energy requirements of tillage tools. This is because these properties affect the soil strength, which has to be overcome by a tillage tool during a soil tillage operation (Gill & Vanden Berg, 1967). In this section, the pertinent soil physical and soil dynamic properties, which affect energy requirements of tillage tools, are discussed.

2.1.1 Soil physical properties

The physical condition of a soil greatly influences the energy/power requirement and performance of tillage tools (Ros, Smith, Marley & Erbach, 1995). Soil physical properties include the soil water content (or moisture content), bulk density, texture, temperature, colour, and pore (void/porosity) space. The soil water content, bulk density and soil texture affect mechanical behaviour and strength of a soil (Gill & Vanden Berg, 1967).

The soil water content of agricultural soils is commonly expressed as the ratio of mass of water contained in the mass of dry material (dry-basis soil water content). There are many methods for determining the soil water content (Erbach, 1987; Gardner, 1986), but the usual way involves the placing of a weighed soil sample in a ventilated oven at a temperature of between 100^{o}C and 110^{o}C until the sample mass becomes constant (Gardner, 1986). Camp and Gill (1969) and Smith (1964) reported that shear strength parameters of fine grained soils decreased with increasing soil water content. However, in the same study, it was also observed that the soil bulk density was concurrently decreasing as the soil water content was increased. Thus, soil water content is a vital parameter in tillage since it influences many parameters that affect the energy requirements of tillage tools.

Soils at same mechanical and environmental conditions, but different textures, behave differently (Gill & Vanden Berg, 1967). Presence of water in void space of soils, for example, can have major impacts on the mechanical behaviour of the soils, owing to its influence on soil strength defining parameters such as the soil shear and soil metal friction parameter (Chancellor, 1994; Spoor & Godwin, 1977). Therefore, in a tillage study, it is important to know not only how much water is present in a soil, but also how this presence affects its engineering or mechanical behaviour. The soil water content is correlated to the soil engineering properties such as strength, compressibility, swelling potential, clay mineralogy, and stress-strain history (Gitau, Gumbe & Biamah, 2006).

Bulk density of a soil is a function of soil water content at any given amount of compactive effort (Marenya & Nyakiti, 2000). As the soil wetness increases, the soil moisture weakens the inter-particle bonds, causing swelling and reducing internal friction making the soil more workable and compactable (Hillel 1980). However, as the soil wetness nears saturation, the fractional volume of expellable air is reduced and the soil can no longer be compacted to the same degree as before, with the same compactive effort. The optimum moisture content is the point at which the soil wetness is just enough to expel all the air from the soil, and the corresponding density is the maximum dry density. Mouazen and Ramon (2002) reported that draft force of a subsoiler was increased with wet and dry bulk densities where it decreased with soil moisture content. Draft force changed linearly with moisture content where it was a quadratic function of wet bulk density and a cubic function of dry bulk density respectively.

Mechanical behaviour of a soil is directly influenced by changes that occur in its bulk density (Gitau, *et al*., 2006). Soil water content is known to affect the soil bulk density. Changes in the levels of the soil water content within different depth ranges in a soil influenced soil bulk densities within such depth ranges (Chancellor, 1994). Ayers (1987) studied the effect of soil moisture content and density effect on soil shear strength parameters of three coarse grained soils during tillage operations. The author concluded that the soil cohesion and friction angle of the three coarse-grained soils increased with increasing soil density. Therefore, in any tillage study, particularly one aimed at the quantification of energy requirement by tillage tools, it is imperative that the soil bulk density is maintained constant throughout the depth of the test layer.

2.1.2 Soil shear strength and soil failure

The strength of a soil is its ability or capacity at a particular condition, to resist or endure an applied load (Gill & Vanden Berg, 1967). The strength of soil to be tilled maybe described by evaluating the parameters involved in the soil's yield conditions on application of a load. In tillage, soil yielding or failure is achieved through a combination of tillage and traction forces. The combined force system can cause the soil to fail or yield by shear, compression, tension and/or plastic flow (Johnson, Grisso, Nichols & Bailey, 1987; Gill & Vanden Berg, 1967; Schafer, Bochop & Lovely, 1963).

Yield or failure conditions in agricultural soils are much more complex than in many engineering materials. This is due to the fact that the conditions of these soils may vary from a near liquid state (like in rotavator-puddled clay soil) to a brittle state. Shear failure and fracture for brittle materials have a clear meaning. Fracture by shear is also observed in agricultural soils (Gill & Vanden Berg, 1967). In some cases, however, fracture is not apparent in soils which may exhibit plastic flow and permanent deformation. The stress state that causes soil fracture or plastic flow is a measure of the soil's shear strength. Shear failure is, therefore, some function of the stress state at which failure occurs (Johnson *et al*., 1987).

Yielding of agricultural soils can be approximated using the Mohr-Coulomb failure criterion (Johnson *et al*, 1987; Gill & Vanden Berg, 1967). The criterion postulates that failure occurs when the maximum shear stress of a material on any plane reaches some critical value that is equal to the shear strength of the given material. Gill and Vanden Berg (1967) traced the history of the original Coulomb theory from its postulation to straight line failure envelope of the Mohr failure theory. This failure envelope is expressed as:

$$\tau_{\max} = C_c + \sigma_n \tan\phi \qquad \text{... (2.1)}$$

where:

τ_{max} = maximum shearing stress at failure (kPa)

C_c = soil cohesion (kPa)

σ_n = normal stress (kPa)

ϕ = angle of internal friction of soil (deg.)

In classical soil mechanics, Equation (2.1) is referred to as the Mohr-Coulomb equation. This equation represents shear failure at a point within a soil mass. Therefore, the criterion of shear failure using this expression must be carefully distinguished from the distribution of shear point failures i.e., the shear plane or surface (Gill & Vanden Berg, 1967). The application of this equation is limited to effective stress cases and does not apply when the total stresses are considered (Chancellor, 1994).

Gill and Vanden Berg (1967) referring to Equation (2.1) stated that: "*while the straight line envelope of Mohr theory does not rigorously represent shear yield in all soil conditions, the theory has been close enough so the extent that this equation has, almost universally, been accepted as a law. One confusing factor is the fact that C_c and ϕ are so* *firmly entrenched that they are often referred to as real physical properties of the soil. In reality they are only parameters of the assumed yield equation; and their logical existence can be explained only by an interpretation of the equation, and not from the physical nature of the soil itself."*

From Equation (2.1), it is evident that the magnitude of the soil shear strength depends on the soil-soil friction and the soil cohesion. The soil cohesive parameter (C_c) represents the maximum shear stress value of the soil when the normal stress is equal to zero. The cohesive parameter depends only on the strength of the *in situ* bonds between soil particles. Therefore, C_c is constant, irrespective of the magnitude of the normal stress applied to a body of soil. The shear stress associated with soil–soil friction, on the other hand, results from the sliding of soil over soil, and is therefore directly influenced by the magnitude of the normal load. Stafford and Tanner (1983) reported a linear relationship between the maximum shear strength and the normal stress applied.

Both the cohesion (C_c) and the coefficient of internal friction ($\tan\phi$) are influenced by soil water content, porosity and grain size distribution of the soil material (Spoor & Godwin, 1977; Chancellor, 1994; Gitau *et al*., 2006). In general, C_c and $\tan\phi$ increase with decrease in porosity, whereas cohesion at any given density increases with soil water content reaching a maximum at some intermediate level and decreasing thereafter as the soil water content increases (Chancellor & Vomicil, 1971; Stafford & Tanner, 1983a). Although a similar observation was made by Wells and Treesuwan, (1978), a study by Kuipers and Kroesbergen (1966) indicated a decrease of the cohesion value with increasing soil water content. This disagreement in the behaviour of soil cohesion could be due to the differences in the ranges of the soil water content levels studied by respective researchers. In addition, it is probable that the textures of the soil studied by these researchers were different resulting in the reported differences in the variation of soil cohesion with soil water content. In general, soil cohesion increases with soil water content, reaching a peak value and thereafter, decrease with further increase in soil water content (Gitau *et al*. 2006; Ayers, 1987). The magnitude of this variation is affected by the soil texture and the soil bulk density (Ayers, 1987).

2.1.3 Soil-metal friction

With any given tillage tool, soil failure does not only occur in the shear mode along the internal rupture planes during tillage but, also at the boundary between the tool surface and the soil. The force to be overcome at the boundary between the tool surface and the soil is called friction force. For agricultural soils sliding against common materials used for making tillage tools, the relationships found by numerous investigators (Koolen & Kuipers, 1983; Hendrick & Bailey, 1982; Gill & Vanden Berg, 1967;) appear to be neither simple nor consistent across soil type and conditions.

The relationship for determining frictional stress considered capable of accounting for the complexities and inconsistencies associated with the agricultural soils (Chancellor, 1994; Stafford & Tanner, 1977) takes the form that is similar to Equation (2.1), and is expressed as:

$$\tau_f = C_a + \sigma_n \tan\delta \qquad \text{... (2.2)}$$

where:

C_a = adhesion stress between the soil and the tillage tool (kPa)

δ = angle of soil-metal friction (degrees)

σ_n = normal stress (kPa)

τ_f = maximum frictional stress at the soil-tool interface (kPa)

Like in the case of shear strength, Equation (2.2) indicates that the maximum frictional shear stress at the soil-tool interface is determined by adhesion and soil-metal friction. There are many factors that influence the magnitude of the resultant frictional shear stress at the soil-tool interface. Chancellor (1994) listed some 12 factors that affect the magnitude of the maximum frictional stress at the soil-tool interface. In tillage, for a given soil, the pertinent factors influencing the magnitude of τ_f include level of the normal stress, soil water content, soil porosity or density, sliding velocity, the maximum value of the normal stress during the course of the test history, and sudden changes (increase or decrease) of normal stress.

From Equation (2.2), the normal stress significantly affects the value of the maximum frictional shear stress. A study by Stafford & Tanner (1983b) reported that for most soils, frictional stress tends to change linearly with normal stress level. However, cases in which small increments in the coefficient of friction μ, ($\tan\delta$ in Equation 2.2) occurred with the increase of normal stress for a given soil/moisture condition were reported in the same study. Cases of small decrease in μ values have also occurred with constant normal stress for different soil/moisture condition (Soehne as cited by Chancellor, 1994). Therefore, the application of this Equation (2.2) must be accompanied by explicitly quantified soil condition, in particular the level of the soil water content.

The level of the soil water content, like in the case of soil shear strength, affects the maximum frictional stress at the soil-tool interface. For many soils (Stafford & Tanner, 1977; Stafford & Tanner, 1983b) the coefficient of friction rise to a maximum value at some intermediate soil water content, with lower values of the coefficient at lower and higher soil water content levels. This effect is more pronounced with fine grained soils than with coarse-grained soils (Robinson, Schafer, & Johnson, 1988).

Soil water content also has a significant effect on adhesion, with adhesion increasing as the soil water content increases from low to intermediate values (Stafford & Tanner, 1977). During tillage, the soil-to-tool bonding depends on the soil water tension. It is, therefore, expected that more fine-grained soils would exhibit greater adhesion at certain soil water contents than would be the case for coarse-grained soils. It has been observed that at very high soil water contents that would produce positive pore water pressure under load, tension would decrease thus reducing adhesion (Neal, 1966), and may even increase to effect lubrication (Chancellor, 1994; Srivastava, *et al.,* 1993).

Soil density or porosity has an insignificant effect on the coefficient of friction, but significantly affects the adhesion component of the frictional shear stress (Stafford & Tanner, 1977). The adhesion shear stress tends to increase as the porosity is reduced, i.e., as the soil density increases. In a study by Butterfield and Andrawes (1972) using pure sand at 36 % and 44 % porosity, the coefficient of friction on six different surfaces, at higher porosity, were 66 % that at the lower porosity.

2.1.4 Dynamic soil strength components

Gill and Vanden Berg (1967) defines the soil dynamic properties as those properties of the soil that become manifest through soil movement. From this definition, if a block of soil starts moving on a flat surface, the resultant friction angle is a dynamic property of soil as it does not appear unless soil starts moving over the flat surface. Also, when loose soil is compacted, its bulk density and shear strength increases; therefore, soil shear strength is another dynamic property of soil.

The soil dynamic properties are affected by the both the soil water content and the soil texture (McKyes, 1985). Based on the soil texture alone, soils can be classified as cohesive on non-cohesive, or as frictional and non-frictional. Measurements have shown that for the purely frictional soils (sandy), the shear strength of the soil does not vary a great deal with shear rate (Stafford & Tanner, 1983a). For such a soil, the inertial force involved in accelerating the soil is the important factor as the operating speed of the tillage tool increases. On the other hand, purely cohesive soils (clay soils) exhibit marked changes in shear strength. Increasing the shear rate at sufficiently high tools speed resulted in the shear strength forces nearly outweighing the inertial forces for clayey soils (McKyes, 1985).

The possible causes for the increases in soil strength with increasing operating tillage tool speed has been studied by many researchers (El-Domiaty & Chancellor, 1970; Aref, Chancellor & Nielson, 1975; Fleniken, Henfer & Weber, 1977, Koolen & Kuipers, 1983; Stafford & Tanner,1983a, 1983b; Glancey, Upadhyaya, Chancellor & Rumsey, 1996; Zhang & Kushwaha, 1999). Zhang and Kushwaha (1999) reported that three mechanisms accounting for the draft increase with increase in operating tool speed were; the soil inertial effect, the soil strength rate effect, and the wave propagation effect. The last effect has been reported on extensively by many Russian researchers (Azyamova 1963; Katsygin, 1964; Vetrov & Stanevski, 1972), who reported inverse decrease in draft requirements with increasing the operating tillage tool speed beyond some limits. This observation was attributed to the fact that tool speed increased faster than the increase in the wave of soil stress propagation. Theoretically, the plastic zone of soil in front of the tool decreased or even disappeared, thus the soil cutting resistance decreased. The significance of these finding is the existence of an optimum operating speed for set soil conditions and blade configuration at which the draft requirement, and therefore energy requirements of passive tillage tools was optimal.

Equations (2.1) and (2.2) describe the stresses interacting on a failure plane within the soil body and on the soil-tool interface, respectively. These two failure criteria are static in nature, and in tillage, they do not account for the effect of speed of the tool. There are a number of studies (El-Domiaty & Chancellor, 1970; Aref *et al.*, 1975; Fleniken, Henfer & Weber, 1977; Koolen & Kuipers, 1983; Stafford & Tanner, 1983a; Glancey *et al.*, 1996) reported in literature indicating that the strength of a soil is a function of the deformation rate. Therefore, the overall soil strength overcome by a tillage tool consists of the static (from Mohr-Coulomb failure criterion) as well as dynamic components. By taking the dynamic component of the soil strength into account, Glancey *et al*. (1996) formulated a general expression for the maximum shear strength of a soil under the influence of a tillage tool as:

$$\tau_{\max} = \tau_o + \tau_i V_f \qquad \text{... (2.3)}$$

where:

$\tau_{\max}$ = maximum shear stress at failure (kPa)

τ_o = soil property related to static component of the static shear strength (kPa)

τ_i = soil property related to the dynamic component of soil shear strength, proportional to the operating speed (kPa $(\text{ms}^{-1})^{-1}$)

V_f = forward travel speed of the tool (ms^{-1})

In a study of the draft force requirements, Stafford (1979) developed the expression:

$$F_d = F_s + f(v) \qquad \text{... (2.4)}$$

where:

F_d = draft force under dynamic conditions (kN)

F_s = static draft force component (kN) *f(v)* = function containing the soil inertial term (kN)

In 1983, Stafford and Tanner (1983a, 1983b) conducted studies on the effect of rate of change on soil shear strength and the soil-metal friction on a clay and sandy clay loam soil. The results of these studies indicated that the soil-metal friction angle increased logarithmically with speed over a wide range of tool velocity and soil water content; and could be fitted by the following logarithmic expression.

$$\delta = a + b\log V_s \qquad \text{... (2.5)}$$

where:

$\delta =$ soil-metal friction angle (deg.)

V_s = sliding velocity of the blade (ms^{-1})

a, b = regressions coefficients, which were all significant at 1 % level

For the shear strength, Stafford and Tanner (1983a) found a logarithmic relationship between deformation rate and the cohesion component for the two soils over a wide range of soil water content for tool velocities in the 0.0015 – 5 m/s range. The fitted logarithmic expression for this case was of the form:

$$C_c = k_1 + k_2 \log(1 + k_3 v) \qquad \text{... (2.6)}$$

where:

k_1, k_2, k_3 = constants for both soils at all soil water contents.

2.2 Tillage tool parameters

2.2.1 Blade configuration

Many types and shapes of blades have been developed for rotavators (Kepner, Bainer & Barger, 1978). Currently, some of the commercially available blade configurations include the L-shaped, the C-shaped, the C-L hybrid, the hook-shaped or pick-type blades, and the hoe-type (Salokhe, Hanif & Hoki., 1993). Because these blades have different soil-tool interaction systems, they affect both the power performance and the quality of work of rotavators. The development of rotavator blades is an on-going process and new blades, particularly in the Asian subcontinent and Japan, where the rotavator is widely used in the preparation of paddy rice fields, has been reported in the recent past (Shibusawa, 1993; Salokhe *et al.,* 1993).

The blade configuration influences the performance of rotavators. Studies of the effects of the blade configuration on performance of rotavators have been investigated by many researchers (Beeny & Khoo, 1970; Shibusawa, 1993; Salokhe *et al.*, 1993; Lee, Park, Park & Lee 2002). Salokhe, *et al.* (1993) investigated the performance characteristics of three types of blades, *viz*, 'C', 'C-L' and 'L' shaped blades in terms of power requirements and the puddling[1] quality of a tractor-driven rotavator in a wet clay soil. They conducted field tests at different speeds at an average tillage depth of 100 mm in a saturated soil, with the wetness maintained at a standing water level of 40 mm. Results of their experiments indicated that the torque requirements for the L-shaped blade was significantly higher than for the other two types of blades, and that the C-shaped blade required the least torque. On the quality of work, which they quantified in terms of the percent reduction in the bulk density of the tilled soil and the puddling index, the C-shaped blade gave better results.

Baloch, Bukhari, Kilgour and Mughal (1986) evaluated the performance of new slashers, old (used) slashers and 'L' shaped blades in terms of the work rate, fuel consumption and specific energy consumption in a soft and a hard soil. The results indicated that the old slasher blades gave the highest field capacity, consumed less fuel and demanded less power in both soft and hard soils at different speeds. Comparison of the L-shaped and new slasher blades, showed that the L-shaped blades performed better in both soft and hard soils at low and high rotor speeds, using the stated performance parameters.

Beeny and Khoo (1970) investigated the performance of different shaped blades for rotary tillage in a wet rice soil. The blades studied were of the 'L-', C- and I-shapes. The results obtained indicated a relatively high specific work requirement for the L-shaped

[1] the process of churning soil and water in a flooded field so as to form a homogenous mixture such that the soil particle remain in suspension during the time of planting (Gupta & Visvanathan, 1993b)

blade, compared with the other two types over the same range of operating regimes. The L-shaped blade, on average, required about 30 % more energy for the specific work than the I- and C-shaped blades. The study also indicated that the L-shaped blades gave the greatest forward thrust/push to the tractor, followed by the C-shape and the I-shape respectively. This thrust may be worth paying for in difficult traction conditions such as those prevalent in the paddy rice fields.

From the findings of Beeny and Khoo (1970) it appeared that a 'hybrid' blade from the C- and L-shaped parent shapes, might give better performance results. Consequently, Beeny (1973) developed two modified blades and conducted a study under typical swampy rice field conditions to compare the performance of these specially designed blades. He referred to these blades as the 'power' blade, which was essentially a L-shaped blade with an extended blade length and 'speed' blade, which was a 'hybrid' between the 'power' blade and C-shaped blade. The study found that the fitting of 'speed' blades effected a saving of about 25 % in brake horse power (bhp) of the tractor over those fitted with the 'power' blade. However, the 'power' blade gave greater forward thrusts compared to the 'speed' blade. The author hypothesised that the power saved could be utilized by increasing the working width of a rotavator fitted with 'speed' blades by 25 % and that this would increase the forward thrust of the 'speed' blade fitted rotavator by 25 %. This hypothesis was not verified.

A remarkable reduction of specific power demanded by an up-cut rotavator as a function of blade configuration was reported by Shibusawa (1993). In this study, he observed the dynamics of the rotavator tilled-soil and the blade kinematics and noted tremendous amounts of re-tillage occurring as the depth of tillage increased. He surmised such re-tillage to be the main source of high energy expenditures associated with deep rotary tillage. From the observations, Shibusawa (1993) hypothesised that significant reduction in energy and power requirement for deep tilling rotavators can be realised if re-tillage could be avoided.

In a study that investigated the minimum tillage characteristics by rotavator blades, Lee *et al.* (2002) used three types of blades, *viz,* two general-purpose rotary blades, used with tractors and power tillers; and a levelling rotary blade for tractor use in wet paddy fields. Of the three types of rotary blades used in this study, the rotary blade for the power tiller was considered most satisfactory for strip tillage on the basis of its low torque requirement and the highest ratio of soil breaking.

2.2.2 Direction of rotation

The direction of rotation of the rotor is a basic rotavator design parameter that has a significant influence on the power performance and the quality of work of rotavators (Hendrick & Gill, 1971a). For the horizontal axis rotavators, the direction of rotation influences the power demand; and determines whether a thrust force or pull force will be exerted on the tractor. In general, for the same design dimensions and operational conditions, the energy required for only processing the soil in up-cut rotation is less than the requirement for down-cut rotation (Lee *et al*., 2003; Salokhe & Ramalingam, 2001; Shibusawa, 1993; Hendrick & Gill, 1971a). However, this may not be true for the total energy required, which includes the energy required for the forward propulsion of both the tractor and the rotavator in addition to the energy required to till the soil.

The difference in specific energy demand for tilling the soil between the up- and down-cut rotavator has been attributed to the resultant soil-blade dynamics and the difference in the soil failure process, between the down- and the up-cut rotary tillage (Hendrick & Gill, 1971a; Kataoka & Shibusawa, 2002). When down-cut rotation is used, each blade cuts an increment of undisturbed soil while entering from the surface. With the up-cut rotation; the soil increment is cut from the bottom upward. For the up-cut rotation, after the initial entry of the blade into the soil, the blade for subsequent cuts operates from an area that had been tilled by the preceding blade.

The direction of rotation also affects the manner in which soil failure occurs during the rotary tillage operation. Up-cut rotation causes the blades to operate towards an unconfined area (the soil surface) with more of the soil failing in tension and shearing than in the case of down-cut process (Kataoka & Shibusawa, 2002). Reversing the direction of rotation also changes the geometry of the soil-tool system (Thakur & Godwin, 1990); even where the other operational parameters are constant. The significant differences observed in the energy requirements for tilling the soil between the down- and up- cutting directions can therefore be attributed to the different soil-tool systems and the different soil failure processes taking place during up-or down-cutting tillage.

The segment in which the majority of energy is expended during a complete rotation of a rotavator is different for the up-and down-cut directions of rotation. When the down-cut direction is used, the majority of the energy is expended in the first quadrant (between 0 and 90 degrees). For the up-cut direction, the cutting is mainly in the fourth quadrant and occurs between 270 and 360 degrees (Hendrick & Gill, 1971a). These differences in the segments where the majority of energy is expended during rotary tillage for the up- and down-cut directions may possibly influence the energy requirements.

Lee *et al*., (2003) measured the tilling torque variation through 180^{o} for a rotavator fitted with three different types of blades for the down- and up-cut directions and reported a 20-30 % higher torque requirement during the down-cut for all blade shapes. Salokhe and Ramalingam (2003) working in Bangkok clay, evaluated the draft and power requirements of rotavators equipped with up-cut and down-cut blades. The study reported less PTO power consumption for the up-cut direction than for the down-cut direction of rotation for all passes and forward speeds.

Whereas the magnitudes of the thrust or pull forces exerted on the tractor by a down-cut or up-cut rotavator have significant influence on the total energy, there is almost lack information on the impact of the direction of rotation, the soil condition and other rotavator design factors on the magnitude of forward thrust or pull forces generated by rotavators during the tilling process. What is acknowledged in literature is the fact that the use of the down-cutting rotavator under difficult traction conditions, such as in the paddy rice fields, aid in traction. This is attributed to the forward thrust generated while processing the soil in the down-cut (concurrent) direction of rotation (Shinners *et al.,* 1993; Maniam & Kathirvel, 2001).

2.2.3 Depth of tillage

As with any other tillage tool, the depth of operation has significant influence on the power requirement and performance of a rotavator (Gosh, 1967; Hendrick & Gill, 1971b; Shibusawa, 1993). Hendrick and Gill (1971b) reported that increasing the depth of operation, while holding other rotavator design parameters and soil conditions constant, resulted in increased energy requirement for both directions of rotation. From the previous research work reviewed by Hendrick & Gill (1971b), there was no consensus on the relationship between depth and energy requirements.

In most cases, specific energy requirement increases at an increasing rate with the depth of tillage (Shibusawa, 1993) although some studies (Beeny & Khoo, 1970) have shown a linear relationship between increments in these two factors for same soil conditions for a given rotavator. This increasing rate of the specific energy requirement with depth has been the main reason for restricting the rotavator to be operated at shallow depths of tillage. A combination of deep depth of tillage, with increasing blade rotational velocity, results a rapid increase in specific energy (Hendrick & Gill, 1971b).

While the energy demand increased with depth, the specific energy requirements, i.e., energy per unit volume of tilled soil, decreases. The degree of the decrease in the specific energy requirements has been shown to be greatly influenced by the direction of rotation of the rotor (Shibusawa, 1993). Furlong (as cited by Hendrick & Gill, 1971b) reported a ratio of 3:2 for the increase in soil-volume-tilled to the increase in the specific energy requirements for a down-cut rotavator. Hendrick & Gill, 1971b citing the findings of Furlong stated that this ratio is unaffected by the speed of the rotor.

Until the study by Shibusawa (1993) the reason behind the rapid increase in specific energy of rotavators with increase in working depth had remained unclear. The author reported that a tremendous amount of re-tillage occurred during rotavator tillage for a rotavator fitted with commercial blades, and that as the depth increased, the amount of soil being re-tilled increased significantly. This resulted in rapid increase of the total specific energy requirements of the rotavator. Using special blades, that avoided re-tillage, the author reported a 50 percent reduction in specific energy requirements for both up-and down-cut rotavator tillage operations at depths of 300 mm or greater. From this study it is evident that avoiding re-tilling is a plausible means of reducing the specific energy requirement of rotavators.

Specific energy requirements of rotavators have also been reported to be influenced by the ratio of the tillage depth to the radius of the rotor (Shibusawa, 1993; Hendrick & Gill, 1971b). However, relationship between the rotavator radii with the energy requirement is unclear (Hendrick & Gill, 1971b). Shibusawa (1993) reported that for a given direction of rotation, a smaller radius rotor required less energy when this ratio is less than unity. Hendrick and Gill (1971b), after reviewing a number of past researches, concluded that the optimum rotor diameter-to-depth ratio appeared to be in the range of 1.1 to 1.4 since this is the range in which the minimum specific energy requirement occurs.

In general, when the depth of tillage is greater than the rotor radius, the up-cut rotavators require 20 to 30 percent less specific power (Salokhe & Ramalingam, 2001). When the tillage depth is less than the rotor radius, the rotavator tends to throw more soil forward, resulting in significant re-tillage (Hendrick & Gill, 1971b; Shibusawa, 1993; Lee *et al*., 2003) and hence the higher energy requirements for down-cutting rotavators.

2.2.4 The rotavator kinematic parameter, λ

The kinematic parameter, λ is the ratio of the blade peripheral velocity to the forward travel velocity of the tractor. It is perhaps the most important rotavator operational parameter for quantifying this tool's tillage performance because it influences both the energy requirements and the resultant tilth quality of a rotavator (Hendrick & Gill, 1971c). Being a ratio of the blade peripheral velocity to the forward travel velocity, λ can be varied in the following ways: (1) changing the rotor radius, (2) changing the rotor velocity, and (3) changing the machine travel velocity. In practice the rotor radius for a rotavator of given configurations remains unchanged and the manufacturer only provides means of varying the ratio between the tractor PTO and the rotor velocity as a means of changing the λ. This provision is usually by means of suitable gearing mechanism (Manian & Kathirvel, 2001).

Changing of λ has a significant influence on the performance of a rotavator since this changes the bite length or tilling pitch, which affects the volume of the soil cut per revolution by the rotavator blades. From the findings of Beeny and Greig (1965), larger values of λ mean more cutting by the blade per unit volume, which increases the specific work of the rotavator. This effect holds whether λ is increased by increasing rotor radius, *R*; the rotational speed, ω; or by decreasing the forward travel speed V_f while holding the other operational parameters constant. However, decreasing λ does not necessarily decrease the specific power requirement, except within certain limits. Beeny and Greig (1965) showed that increasing the forward travel velocity beyond the point of providing adequate clearance angle, caused the specific power to increase drastically.

Interesting observations have been reported on the influence of λ on the power or specific energy requirement. Depending on the parameter(s) that is being changed to change λ, the accompanying changes in the specific energy or power requirements are significantly different (Hendrick & Gill, 1971c). Kisu *et al.* (as cited in Hendrick & Gill, 1971c) reported that when λ is changed by holding V_f constant and decreasing the rotational rotor velocity ω, the power required decreases to a minimum value and then increases as the clearance angle approaches zero. In the same study, when λ was changed by holding ω constant and increasing V_f, the specific power requirement decreased up to a point and thereafter remained constant.

From a review of independent past studies by Hendrick and Gill (1971a, 1971b, 1971c), Hendrick (1980), Beeny and Khoo (1970) on the effect of λ on the specific energy requirements of rotavators, there appear to be an optimal value of λ at which the rotavator of a given rotor radius should be operated. When *R* and ω are held constant and V_f is increased (i.e., decreasing λ), the specific energy or power required initially decreases to a minimum value and then increases. Similar observations were reported for the case of changing λ by holding *R* and V_f, and then varying the rotational velocity, ω, of the rotor. The optimal λ value appears to be influenced by the direction of rotation (Dalin & Pavlov as cited by Hendrick & Gill, 1971c). Hendrick (1980) suggested a practical lower limit of the ratio of rotor tip speed to forward speed, i.e. velocity ratio, to be 2.5. Lisunov as cited in Hendrick & Gill (1971c) found the optimum value of λ to be 2.4. At low velocity ratios, the backside of the blades would contact uncut soil resulting in drastic increase in the specific energy requirements (Hendrick, 1980).

After comprehensively reviewing research literature of past studies on the effect of λ on the specific energy or power requirements, Hendrick and Gill (1971c) made the following general conclusions:

- Decreasing λ by increasing forward travel speed, results in an increase in the power requirement, but a reduction in the specific power (provided the geometry of the soil-tool system is not significantly varied).
- Decreasing λ by decreasing angular velocity of the rotor, decreases the power requirement and the specific power.
- Increasing λ results in a greater value of the ratio between cutting area and the volume of the soil slice cut.

2.3 Modeling energy requirements of tillage tools

In Figure 1.2 (see §1.2), it is the tillage process or soil-tool interaction that interfaces the input parameters and output performance indicators used for assessing the performance of a tillage tool. Therefore, any model that aims to predict the energy requirements of tillage tools must be capable of developing relationships for the interaction process between a soil in a given state or condition, and a given tillage tool during tillage. Over the years, a number of analytical, empirical and numerical models have been developed for predicting the energy requirements of different tillage tools (Kushwaha, Chi & Shen, 1993; Marenya, du Plessis & Musonda, 2003). These models have been developed using the empirical, analytical, and numerical modeling approaches for both the passive and active tillage tools.

Analytical models have been extensively used, for the past six or so decades, for predicting the draft and energy requirements of narrow passive tillage tools (Payne, 1956; O'Callaghan & Farrelly, 1964; Hettiaratchi & Reece, 1967; Godwin & Spoor, 1977; McKyes & Ali, 1977; Perumpral, Grisso & Desai, 1983; Swick & Perumpral, 1988; Zeng & Yao, 1992). On the other hand, a number of empirical models have been developed for the active tillage tool, particularly the rotavator. The absence of the application of analytical methods for the prediction of the energy requirements of active tillage tools, including the rotavator was attribute to the complex manner of movement of these tillage tools(Marenya *et al.*, 2003).

The application of the numerical methods such as the Artificial Neural Network (ANN), Discrete Elements Methods (DEM), Finite Element Methods (FEM), and Computational Fluid Dynamics (CFDs) for the modeling of tillage tools energy requirements only became feasible with the advent of powerful computers (Kushwaha *et al.*, 1993) and appearance of improved computing software. Among these approaches, FEM was applied earlier than the other numerical methods in the analysis of the cutting process of simple passive tillage tools (Yong & Hanna, 1977). The results of the FEM soil cutting process available in literature (Yong & Hanna, 1977; Chi & Kushwaha, 1989, 1990, 1991a, Kushwaha & Shen, 1995; Fielke, 1999; Mouazen & Nemenyi, 1999) indicate that this method produced acceptable results. In addition, FEM has showed more flexibility to simulate tillage operations under different tools shapes.

In an extensive review of literature on analytical and numerical models Kushwaha *et al*., (1993) stated that FEM not only calculated the soil forces but also provided a progressive failure zone, soil acceleration field, and stress distribution on the tool surface. This means that FEM models can be used to better understand the soil tool interaction during tillage operations. However, this method has a serious limitation because all FEM models require a constitutive relation relationship of a material, which for agricultural soils, is not yet fully understood. This shortcoming of FEM eliminated the need for further analysis and discussion of the FEM models in this study.

The application of CFDs technique in the analysis of the soil-tool interaction for the prediction of the tillage tool energy requirements is a recent development. This technique, like the FEM, requires the use of powerful computers and specialized software for their successful implementation. Karmakar and Kushwaha (2005a) applied this technique to simulate the flow of soil around a simple tool using a vertical blade in rectangular flow domain. The soil was treated as a Bingham viscoplastic material in respect to its non-Newtonian rheology. The free-surface simulation of an open channel visco-plastic soil flow indicated soil deformation patterns, and the effect of speed on the failure front propagation. Soil deformations, as the flow of a visco-plastic material with yield stress, were observed to possess "plastic flow" and "plug flow" patterns at different tool travel speeds. In a related study, Karmakar and Kushwaha (2005b) reported that soil pressure on the tool surface increased with the tool operating speed, with the pressure being greatest at the tool tip.

From the literature review presented in this section, there was no documentation in literature, on the application of the ANN and DEM for modeling techniques for the predicting the energy requirements of both passive and active tillage tools. Therefore, the remainder of this section is devoted to the analysis of some of the past analytical models because they may be useful in the development of an analytical model for the prediction of rotavator torque requirements.

2.4 Analytical soil failure models

There are many analytical prediction models available, that could be used for predicting the forces on the tillage tool (Kushwaha, *et al.*, 1993). These models have been developed for simple rectangular blades passively drawn through the soil (Marenya *et al*., 2003). Depending on the depth-to-width ratio (*d/w*) for such tools, three categories of these blades have been distinguished, viz., wide blades ($d/w < 0.5$); narrow tine or chisel ($1 < d/w < 6$); and very narrow (knife) tines ($d/w > 6$). Depending on the blade category, both 2 – and 3 – D models have been developed using the analytical approach (Godwin & O'Doherty, 2006). All the 2-D models reviewed are of the static type while the 3-D models comprise both static and dynamic analytical models. In static models, the effect of the travel speed of the tool is not considered. In the subsequent subsections, detailed analysis of the existing analytical models is presented.

2.4.1 Two-dimensional models

The critical soil cutting models were first developed, in the early 1960's, for 2-D soil cutting caused by a wide blade based on Terzaghi's passive earth pressure theory. Based on this theory, a failure zone was assumed to exist ahead of a cutting blade and the soil in the failure zone was assumed of be in the critical failure state. The slope-line theory was then applied to predict the soil forces. The calculations of the shape of the slope-line and the failure areas were cumbersome, at that time, as the proposed solution used the finite difference method (Kushwaha, *et al.*, 1993), which is highly iterative.

According to Kushwaha *et al*., 1993, to obtain a simpler solution, a semi-empirical failure zone, was earlier suggested by Terzaghi (1943) for 2-D soil failure that consisted of a Rankine passive zone and a complex shear zone, bounded by part of a logarithmic spiral curve, was developed (Figure 2.1). The resulting force, P on the blade was calculated by assuming static equilibrium along the boundary and determining the position of O to provide a minimum value (Hettiaratchi *et al.*, 1966).

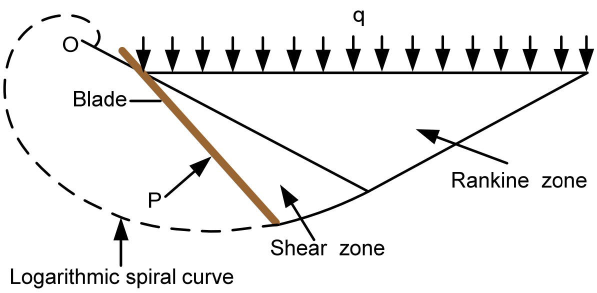

Figure 2.1: Logarithmic spiral failure zone

2.4.1.1 Reece model

Reece (1965) hypothesized that the mechanics of earthmoving by a blade is similar, in many respects, to the soil bearing capacity of shallow foundations as described by Terzaghi (1959). Using this theory and the mathematical force expression developed by Osman (1964), he proposed the following expression, popularly known as Reece's earthmoving equation (Equation 2.7), for describing the force required to cut the soil by a tool.

$$P = \left(\gamma g d^2 N_\gamma + C_c d N_c + C_a d N_a + q d N_q\right) w \qquad \text{... (2.7)}$$

Equation (2.7) is widely accepted by many researchers (Hettiaratchi & Reece, 1967; Godwin & Spoor, 1977; McKyes & Ali, 1977; Perumpral *et al*., 1983) for predicting the draft force of wide blades. This equation takes into account the effect of the soil bulk density, soil cohesion, soil frictional strength, geometry of cut soil, tool to soil adhesion and friction angle, tool operating depth, and tool width; all of which contribute to total soil resistance that must be overcome in tillage operations by a blade. However, it does not include the speed of the blade, which is known to have effects on the draft force (Perumpral, Grisso & Desai, 1983; McKyes, 1989).

Based on the logarithmic spiral failure zone (Figure 2.1), Hettiaratchi and Reece (1974) developed charts for determining the different N-factors contained in Equation (2.7). These N-factors were functions of the blade rake angle, soil internal friction and the soil-metal friction. To compensate for this shortcoming of Reece's equation, McKyes (1985) proposed another model (Equation 2.8) that basically was the same as the Reece (1965) model, with an additional term that accounted for the effect of tool speed on the draft force requirements.

$$\left.\begin{aligned} P &= \left(\gamma g d^2 N_\gamma + C_c d N_c + C_a d N_{ca} + q d N_q + \gamma V_f^2 N_a\right) w \\ N_a &= \frac{\tan\rho + \cot(\rho+\phi)}{\left[\cos(\beta+\delta) + \sin(\rho+\delta)\cot(\rho+\phi)\right]\left[1+\tan\rho\cot\beta\right]} \end{aligned}\right\} \quad \ldots (2.8)$$

2.4.2 Static three-dimensional models

2.4.2.1 Payne model

From literature, it is evident that the first analytical 3-D model for predicting the force requirements of a simple inclined tillage tool was developed a Payne (1956). The model adapted quasi-static Mohr-Coulomb soil mechanics using both the passive retaining wall and the bearing capacity theories by studying the soil failure patterns in a series of field and laboratory tests. By observing the upward displacement of soil ahead of the tillage tool, Payne (1956) assumed a failure zone for tines with a width-to-depth ratio of less than 1:1.

Further experiments done by Payne and Tanner (1959) in a follow-up study, showed that the shape of the failure zone changed with the geometry of the tool such as rake angle, depth and width. In this latter work by Payne and Tanner (1959), no equations were developed for evaluating the draft force. In another related study, Osman (1964) improved the model proposed by Payne (1959) by introducing factors such as soil properties, tool rake angle and tool-surface roughness, to the force equation, using the dimensional analysis techniques.

2.4.2.2 O'Callaghan and Farrely model

Based on the work of Payne (1956), O'Callaghan and Farrely (1964) undertook extensive field experiments for three soil conditions. Based on the observation made from the extensive field measurement undertaken, a soil failure model shown in Figure 2.2 was proposed.

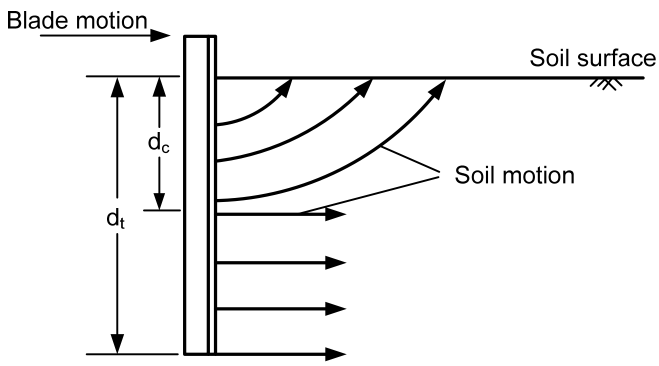

Figure 2.2: Schematic illustration of the crescent and lateral failure zones observed by O'Callaghan and Farrely (1964)

Based on Figure 2.2 and Terzaghi's passive earth pressure theory, O'Callaghan and Farrely (1964), proposed a model consisting of a crescent failure above the critical depth, d_c and lateral failure below the critical depth. The d_c was assumed to be equal to the width of the tool for a smooth blade and half the tool width for a free surface with normal restraint. Depending on the depth at which the tool was operated at, they proposed two equations for predicting its (blades') draft force by applying Terzaghi's method (Terzaghi, 1959). For shallow blades, operated above the d_c, the draft prediction equation, which only accounts for the crescent failure, was given by Equation (2.9).

$$H_s = w\left(C_c d N_c + \gamma d^2 N_\gamma\right) \quad \text{... (2.9)}$$

When the blade was operated at a depth greater than d_c, the total force required for both the crescent and lateral failure was predicted by Equation (2.10).

$$H_{dc} = \frac{C_c w(d - kw)}{\tan\phi}\left[\tan^2\left(\frac{\pi}{4} + \frac{\phi}{2}\right)e^{\pi\tan\phi} - 1\right] + H_s \quad \text{... (2.10)}$$

In this model, the effect of two side crescents for the failure above the d_c plus the forces due to adhesion and soil interface friction were not included in the equations developed for the prediction of the blade draft force. In addition, the effect of the gravity term, in Equation (2.7), was considered negligible owing to the small masses of soil involved. Comparison between the model and test data, were generally good for a vertical blade. However for hard soil, the model under-predicted the draft force. The slight disparity between the test data and the predicted draft force for hard soils was attributed to the lack of incorporating all the identified soil resistance elements in the model.

2.4.2.3 Hettiaratchi and Reece model

After developing the two dimensional model discussed above (§2.4.1.1), Hettiaratchi and Reece (1967) developed a three-dimensional (3-D) soil failure model. This model was, in some respects, similar to the earlier model developed O'Callaghan and Farrely (1964) since the soil failure configuration was divided into forward failure, ahead of the soil-tool interface; and traverse failure, i.e., the horizontal transverse movement of the soil away from interface (Figure 2.3).

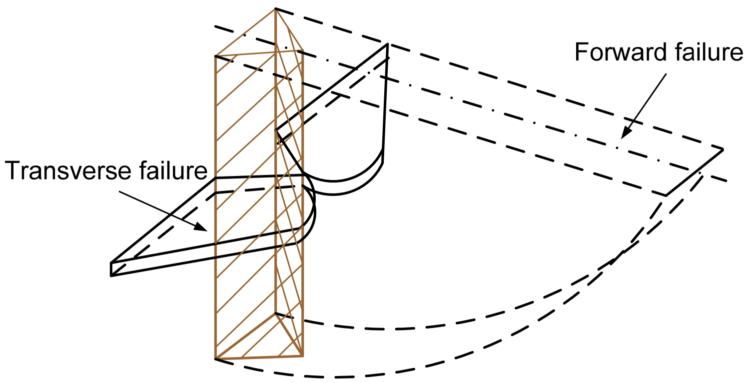

Figure 2.3: Failure zone of the Hettiaratchi and Reece model (Hettiaratchi & Reece, 1967)

The total force comprised was calculated as the sum of the force for forward failure and the transverse failure. The force due to the forward failure was found using the 2-D equation (Equation (2.7), §2.4.2.1) while the force due to the transverse failure was calculated similar to that derived by O'Callaghan and Farrelly (1964) with an additional component to account for the gravitational component. The resultant model was as follows:

$$P_f = \left(\gamma d N_\gamma + C_c d N_c + q d N_q\right) \quad \text{... (2.11a)}$$

$$P_{trans} = \left[\gamma\left(d_e + \frac{q}{\gamma}\right)^2 w N_{s\gamma} + C_c w d_e N_{sc}\right] K_i \quad \text{... (2.11b)}$$

$$H = P_f \bullet \sin(\alpha + \beta) + P_{trans} \bullet \sin\alpha + C_a d \cos\alpha \quad \text{... (2.11c)}$$

$$V = P_f \bullet \cos(\alpha + \delta) + P_{trans} \bullet \cos\alpha + C_a d \quad \text{... (2.11d)}$$

This model included the soil and soil-metal frictional properties as well as the tool geometry, including the rake angle, depth and width. The model has been found to over-predict the draft force (Grisso & Perumpral, 1985).

2.4.2.4 Godwin-Spoor model

Godwin and Spoor (1977) studied the soil failure patterns with narrow tillage tines; and proposed a 3-D crescent failure above the critical depth and a 2-D horizontal failure pattern below the critical depth. For the 3-D crescent failure, a failure model was proposed as a parallel centre wedge flanked with two side crescents (Figure 2.4). The lateral failure below the critical depth was similar to earlier horizontal failure models proposed by O'Callgahan and Farrelly (1964) and Hettriaratch and Reece (1967). The model included an addition parameter, r, the rupture radius. They defined r as the total forward distance of soil failure on the surface from the tool face.

The total force was calculated as the sum of the forces due to the three sections. The centre wedge force was calculated using Equation (2.7). This 2-D expression was also used to calculate the force for small elements cut from the side crescents (Figure 2.3) as:

$$dP_q = \left(\gamma d_c^2 N_\gamma + C_c d N_c + q d_c N_c\right) \bullet \frac{r d\eta}{2} \quad \text{... (2.12a)}$$

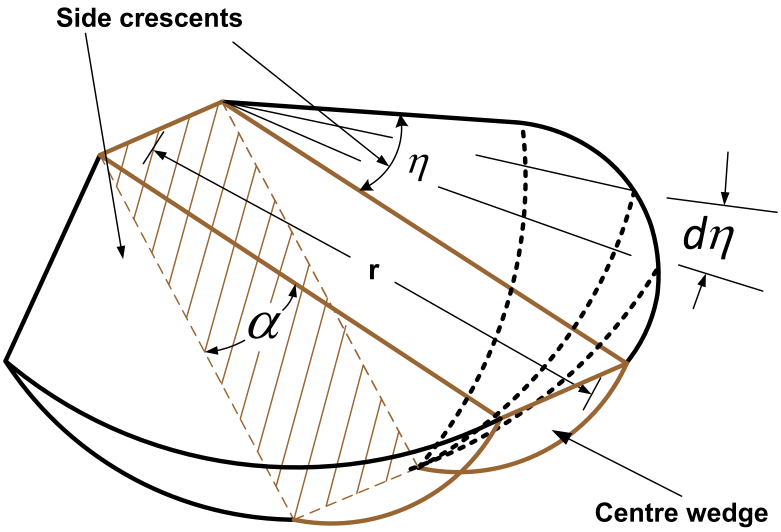

Figure 2.4: Failure zone of the Godwin and Spoor model (Godwin & Spoor, 1977)

The total applied force due to the side crescents was obtained by integration. The integration was simplified by assuming that the failure boundary on the top surface was circular. The draft (H_t) and vertical (V_t) forces for the 3-D crescent failure, above the critical depth was then given as:

$$H_t = \left[\gamma d_c^2 N_\gamma + C_c d_c N_c + q d_c N_q\right] \bullet \left[w + r\sin\eta\right]\sin(\alpha+\beta) + C_a w d_c\left[N_a \sin(a+\delta)+\cos\alpha\right]$$

... (2.12b)

$$V_t = -\left[\gamma d_c^2 N_\gamma + C_c d_c N_c + q d_c N_q\right] \bullet \left[w + r\sin\eta\right]\cos(\alpha+\delta) - C_a w d_c\left[N_a \cos(\alpha+\delta)+\sin\alpha\right]$$

... (2.12c)

The extended angle, η in Equation (2.12b & 2.12c) was calculated as:

$$\eta = \cos^{-1}\left(\frac{d_c \cos\alpha}{r}\right) \qquad \text{... (2.12d)}$$

The lateral draft soil failure force was given as:

$$H_l = w\left[C_c N_c^{'}(d-2d_c) + 0.5(1-\sin\phi)\gamma w N_\gamma^{'}\left(d^2 - d_c^2\right)\right] \qquad \text{... (2.12e)}$$

The total draft force for deep blade was given as:

$$H = 2H_t + H_l \qquad \text{... (2.12f)}$$

The application of this model required prior knowledge of *r*. However, *r* was affected by the blade rake angle and depth. Therefore, in order to test and verify this model, Godwin and Spoor (1977) developed a graph using the data from Payne (1956) and Payne and Tanner (1959); and Reece (1967) to determine the distance ratio (rupture distance over depth, i.e., *d/w)* and the tool rake angle. According to Shen and Kushwaha (1998), the determination of *r* was difficult.

2.4.2.5 McKyes and Ali model

McKyes and Ali (1977) developed an analytical model for narrow blades. Like was the case in the Godwin and Spoor (1977) model, a failure wedge was proposed ahead of the cutting blade (Figure 2.5). The model was similar to that of Godwin and Spoor (1977) as it consisted of a centre wedge and two side crescents. However, the bottom surfaces of both the centre wedge and the side crescents of this model were different from that of Godwin and Spoor (1977) (see Figures 2.3 & 2.4). In this model, the failure shape of the centre wedge's bottom was assumed to be a plane, while the two side crescents were assumed to be a straight line.

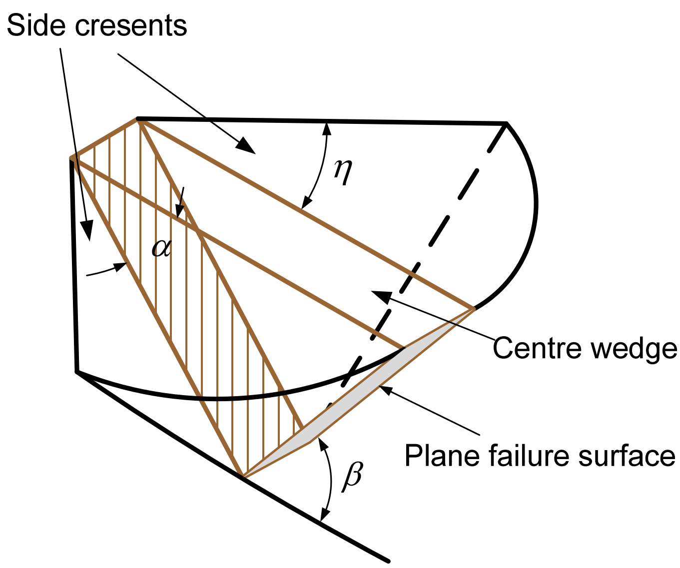

Figure 2.5: Single wedge failure zone of the McKyes and Ali model (McKyes & Ali, 1977)

Unlike the foregoing models, that used the 2-D slip-line theory to evaluate the forces, this McKyes and Ali (1977) model proposed a new approach to overcome the shortcoming of the cumbersome slip-line theory. In this model, the forces on each section were determined by applying the mechanics of equilibrium directly rather than using the equation and *N*-factors of the 2-D soil failure (Kushwaha *et al.*, 1993). The assumed flat bottom plane of the centre wedge and the straight line at the bottom of the side crescents enabled the defining of the direction of the reaction forces at the bottom of the failure zone.

The total draft force expression developed (Eqn. 2.13a) for the failure wedge shown in Figure 2.5 consisted of the force contributions from the centre wedge and the two side crescent wedges (McKyes, 1989, pp 198), and was given as:

$$P = \left[\gamma d^2 N_{\gamma H} + C_c d N_{cH} + q d N_{qH} + C_a N_{ca}\right] w \qquad \text{... (2.13a)}$$

Equation (2.13a) is similar to the Reece (1965) equation but with N-factors re-evaluated for the 3-D soil failure. The expressions for the N-factors in this case were as follows:

$$N_{\gamma H} = \frac{\dfrac{r}{2d}\left[1 + \dfrac{2r}{3w}\sin\eta\right]}{\cot(\alpha+\delta) + \cot(\beta+\phi)} \qquad \text{... (2.13b)}$$

$$N_{cH} = \frac{\left[1 + \cot\beta\cos(\beta+\phi)\right]\left[1 + \dfrac{r}{w}\sin\eta\right]}{\cot(\alpha+\delta) + \cot(\beta+\phi)} \qquad \text{... (2.13c)}$$

$$N_{qH} = \frac{\dfrac{r}{d}\left[1 + \dfrac{r}{w}\sin\eta\right]}{\cot(\alpha+\delta) + \cot(\beta+\phi)} \qquad \text{... (2.13d)}$$

The rupture distance r, used in Equations (2.13b) to Equation (2.13d) was calculated as:

$$r = d(\cot\alpha + \cot\beta) \qquad \text{... (2.13e)}$$

The N-factors in Equation (2.13b) to Equation (2.13d) are all a functions of angle β. The angle β was calculated by minimizing the factor $N_{\gamma H}$, which was the factor for the gravity term in the draft equation. The angle β, obtained by the minimization, of $N_{\gamma H}$ was thereafter used to calculate the remaining N-factors. McKyes (1989) developed sets of charts for determining the N-factors in the above expression. However, since the advent of good computing power and software, it is now easier to write simple computer programs to determine the N-factors. Kushwaha *et al.* (1993) stated that this model was easier to use than the Godwin and Spoor (1977) model, and that it does not require prior knowledge of the rupture distance, r.

2.4.2.6 Perumpral, Grisso and Desai model

Perumpral *et al.* (1983) developed another 3-D for narrow tillage tools. In this model, the weight of the soil due to the two side failure crescents of the Godwin and Spoor (1977) and McKyes and Ali (1977) were replaced by two reaction forces (R_{cr}) acting on the center wedge as shown in Figure 2.6. The model assumed that soil heaved ahead of the blade and that the failure wedge's slip bottom was a straight line, similar to that of the McKyes and Ali (1977) model.

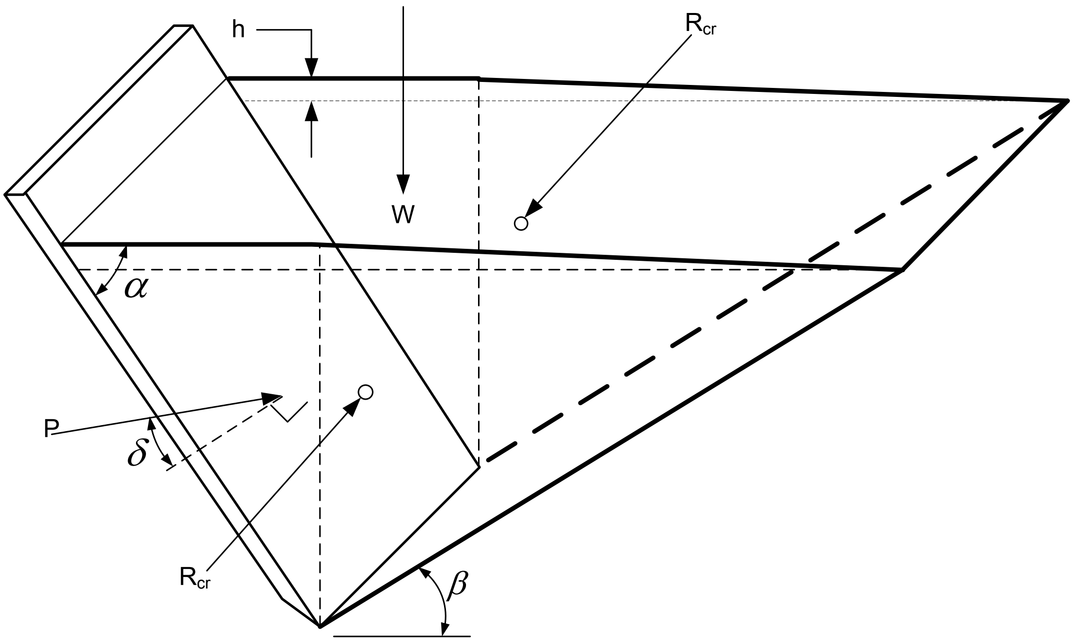

Figure 2.6: Failure zone of Perumpral *et al* model (Perumpral *et al.*, 1983)

The total draft force was calculated by considering the equilibrium of all the forces acting on centre wedge. The final force equation was given as:

$$P = w\left[\gamma d^2 N_\gamma + C_c d N_c + C_a d N_a\right] \quad \text{... (2.14a)}$$

The N-factors in Equation (2.14a) were given by the following expressions

$$N_{\gamma} = \frac{\frac{A}{wd^2}\left[2K_o d_z \sin\phi + w\sin(\phi+\beta)\right]}{\sin(\alpha+\beta+\phi+\delta)} \qquad \text{... (2.14b)}$$

$$N_c = \frac{\cos\phi\left[\frac{2A}{wd} + \frac{1}{\sin\beta}\right]}{\sin(\alpha+\beta+\delta+\phi)} \qquad \text{... (2.14c)}$$

$$N_a = \frac{-\left[1+\frac{h}{d}\right]\cos(\alpha+\beta+\phi)}{\sin(\alpha+\beta+\delta+\phi)} \qquad \text{... (2.14d)}$$

The coefficient of earth pressure, K_o and the average depth to the centroid of the failure wedge, d_z from the soil surface were given by the following equations:

$$K_o = 1 - \sin\phi \qquad \text{... (2.14e)}$$

$$d_z = \frac{1}{3}(d+h) \qquad \text{... (2.14f)}$$

The angle β was determined by minimizing the total force P in Equation (2.14a). A computer program was developed that minimized the total force and determined the failure plane angle, β. The average depth at which the centroid of the failure wedge was located from the soil surface was found to be a function of failure plane angle, β.

2.4.3 Dynamic three dimensional models

2.4.3.1 Swick and Perumpral model

Swick and Perumpral (1988) modified the Perumpral *et al* (1983) model and proposed a dynamic 3-D soil cutting model that incorporated the tool dynamic effects. The model formulation included soil shear rate effects, soil inertial forces and soil-tool interaction parameters. The soil failure zone for this model was similar to that of McKyes and Ali (1977) model; and consisted of a centre wedge and two side crescents, with a straight rupture plane at the bottom (Figure 2.7).

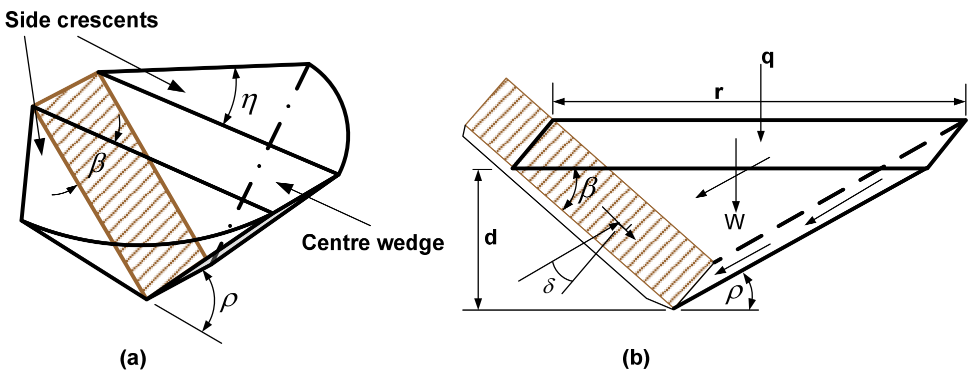

Figure 2.7: Failure zone of the Swick and Perumpral model (Swick & Perumpral, 1988)

In this model, the centre and the side sections were considered separately. Forces acting on the centre wedge are shown in Figure 2.7(b). Swick and Perumpral (1988) observed that the approach of Godwin and Spoor (1977) and McKyes and Ali (1977), in which the extreme outer points of the crescent sides were assumed to lie in a vertical plane passing through the forward tip of the tool, over-predicted the size of the side crescents. Therefore, based on the observations from the soil bin tests, Perumpral and Swick (1988) proposed the extended angle η as a function of the rupture angle distance, *r* and the rake angle β ; and was expressed as (Kushwaha *et al.*, 1992):

$$\eta = \frac{\sin^{-1}\left(-6.03 + 0.46r + 0.0904\sin\beta\right)}{r} \quad \text{... (2.15a)}$$

The force equation, for this model, was derived in the same manner as the McKyes and Ali (1977) model but, with an acceleration force added, to account for travel speed effect. Based on the equilibrium equations for the centre wedge, the force on the centre wedge was given as:

$$P_1 = \frac{\dfrac{C_a d\cos\left(\beta+\phi+\rho\right)}{\sin\beta} + \left(\dfrac{\gamma dr}{2} + qr\right)\sin\left(\phi+\rho\right) + \left(\dfrac{C_c d}{\sin\phi} + F_{acc}\right)\cos\phi}{\sin\left(\beta+\phi+\rho+\delta\right)} \quad \text{... (2.15b)}$$

The expression for the acceleration force, F_{acc}, was given as:

$$F_{acc} = \frac{\gamma d V^2 \sin\beta}{g\sin\left(\beta+\rho\right)} \quad \text{... (2.15c)}$$

The forces on the side crescents was derived by considering the equilibrium of a small slice crescent and then integrating this force for the entire side crescent (Figure 2.8). The force on one side of the crescent was given as:

$$P_2 = \frac{\left\{\left[\frac{\left(\gamma dr^2 + qr^2\right)\sin\left(\phi+\rho\right)}{\sin\left(\beta+\phi+\delta+\rho\right)}\right] + F_{acc2}\cos\phi\left(\frac{\eta}{2}+\frac{\sin 2\eta}{4}\right) + C_c dr\cos\phi\sin\eta\right\}w}{\sin\left(\beta+\phi+\rho+\delta\right)} \quad (2.14d)$$

The side crescent acceleration force, in this case, was given as:

$$F_{acc2} = \frac{\gamma dr V^2 \sin\beta}{2g\sin\left(\beta+\rho\right)} \quad \text{... (2.15e)}$$

The total force, for the centre wedge and the two side crescents was expressed as:

$$P = P_1 + 2P_2 \quad \text{... (2.15f)}$$

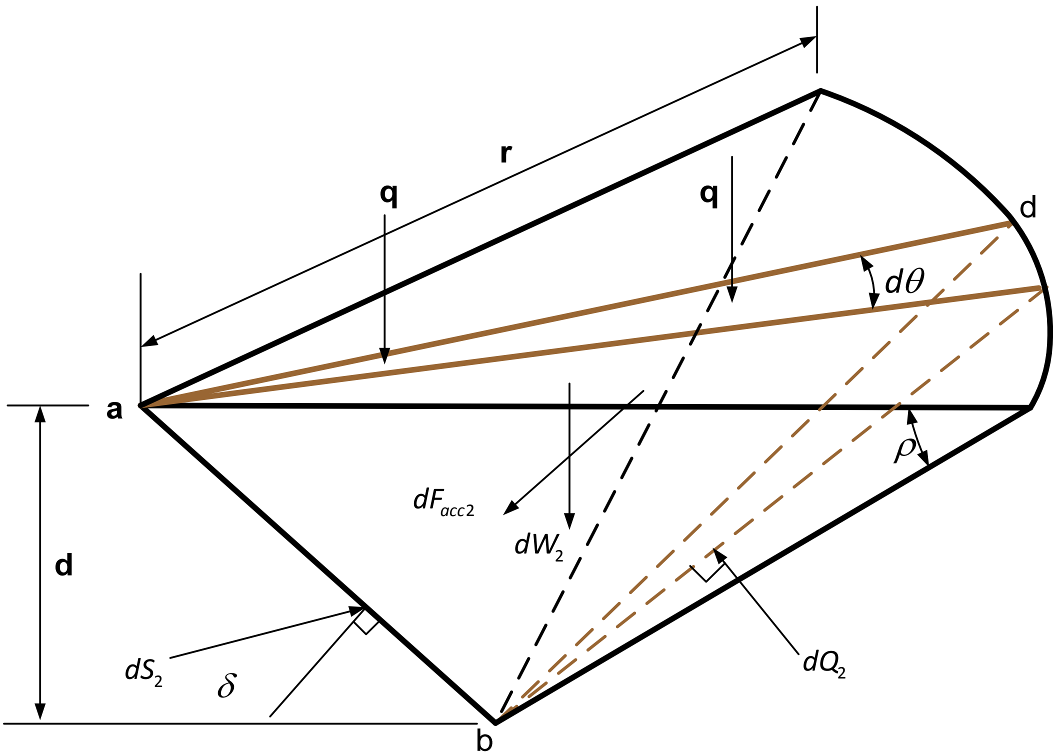

Figure 2.8: A side portion of the idealized failure wedge considered for the model and failure forces (Swick & Perumpral, 1988)

The force, P is a function of rupture angle ρ (Figures 2.7 & 2.8). Since the tool movement creates a passive condition, Swick and Perumpral (1988) used the Passive Earth Pressure Theory to determine the rupture angle. According to this theory, passive failure occurs when the resistance to the soil wedge is minimal. The wedge creating the minimum resistance was determined, in this case, by minimizing the total force (Equation (2.15f)) with respect to the rupture angle, using a numerical procedure. This model gave reasonable draft force predictions for the narrow tools tested.

2.4.3.2 Zeng and Yao model

Zeng and Yao (1992) proposed a soil cutting dynamic model to predict forces on wide and narrow blades. The model incorporated shear rate effects on soil shear, soil-metal friction and inertial effects. The failure zone of was similar to that of McKyes and Ali (1977). The soil failure zones of this model were as shown in Figure 2.9.

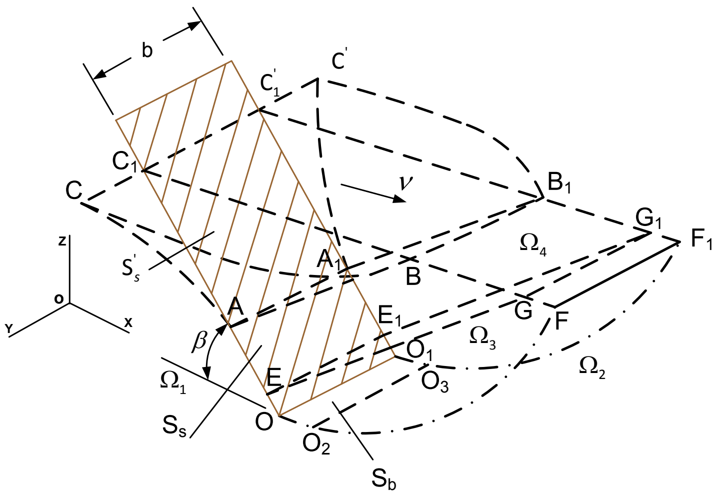

Figure 2.9: Failure zone of the Zeng and Yao model (Zeng & Yao, 1992)

There were two main differences between this model and the one proposed by McKyes and Ali (1997). The first difference was that this model required prior knowledge of the failure shear strain for the determination of the position of shear failure boundary shown in Figure 2.9. Secondly, the total draft force, P_X, for this model comprised five components, namely: the compressive force of the soil, P_G; inertia force of the soil in acceleration, P_A; bottom-wedge cutting force, P_C; frictional force along the cutting board surface, P_F; and soil shear strength force, P_{SH}. The total draft force was expressed as follows:

$$P_x = P_G \sin\beta + \left(P_{SH} + P_A\right)\cos\beta_1 + P_F \cos\beta + P_C \quad \text{... (2.16)}$$

Comparison between the measured and predicted draft force produced acceptable results (Zeng & Yao, 1992). Some significant features of this model were that: soil-metal friction and shear strength were not influenced by shear rate; the model was not applicable for soils that flow plastically.

2.5 Rotavator performance prediction models

2.5.1 Empirical models

Extensive and comprehensive literature reviews on rotavators or rotary tillers have been undertaken by a number of researchers (Marenya, *et al.*, 2003; Hendrick & Gill, 1971a, 1971b, 1971c; Thakur & Godwin, 1989). The findings from these reviews indicated that no fundamental research capable of producing soil-tool mechanics has been undertaken for active tillage tools (Marenya, *et al.*, 2003). This is because all the literature reviewed by the above researchers indicate, that all the torque prediction models developed in the past, were empirically-based. The use of the empirical models for the prediction of rotavator torque requirement was probably due to the perceived complexity in the manner of the movement of a rotavator blade during tillage operations. However, empirical models lack the wide applicability, afforded by analytical models; and are therefore of limited use to designers of tillage tools.

2.5.2 Analytical models

There are very few analytical models that have been proposed for predicting the performance of rotavators (Marenya *et al.,* 2003). The few studies found in literature only modelled some aspects of energy requiring components; and did not use any of the known blades types in the model development. Details of some of the models found in literature that model some different aspects of performance of rotavators are discussed in the subsequent paragraphs.

Thakur and Godwin (1989) have presented the conceptual mechanism of soil failure by a wire under two different situations, i.e., when working through a semi-infinite homogeneous soil mass, and working close to an open wall. They surmised that the soil failure mechanism of a wire, cutting a varying thickness of the soil slice as in rotary tilling can be considered asymmetrical. In a follow-up study, Thakur and Godwin (1990) observed that, in general, soil shear failure planes developed towards the free curved face at an angle of $(45 - \phi/2)^{o}$ with the direction of the major principal stress, while cutting a full soil slice. They developed an analytical model by considering the soil resistance force to failure and reaction due to local shear failure. The model was applied to predict various force components for various bite lengths and depths of tillage. The relation between observed and predicted sets of data returned a high correlation coefficient of 0.94.

Shibusawa (1993) developed a model based on the trochoidal motion of the blade and the sliding motion of the soil over the blade that was capable of avoiding re-tillage in the up-cut direction. It was observed during this study, that significant amount of re-tillage occurred with down-cut rotavators; particularly for set tillage depths greater than 350 mm. Based on this model, a new blade, that enabled maximum backward throwing of the soil, sufficient to avoid re-tillage, was designed, fabricated and tested. On testing the new blade at tilling depths greater than 300 mm, it was found that the energy required in both directions of rotation was about half that demanded by rotavators fitted with conventional blades.

Gupta and Visvanathan (1993b) developed an analytical model for predicting the power requirement of a rotavator in saturated soil. The model was based on the predicted behaviour of saturated soil under impact and pure shear loading. In this model it was assumed that the power required by a rotavator is used for cutting the soil, throwing the cut soil slices by centrifugal action of the blades, overcoming soil-metal friction and overcoming parasitic forces. In this model, all the energy requirements were expressed in terms of torque requirements. The application of this model showed that for saturated clay under rotavator tillage, the power requirement consisted of 0.34 to 0.59 % for cutting, 30.5 to 72.4 % for throwing out cut slices by centrifugal action of rotary blades, 0.96 to 2.45 % for overcoming soil-metal friction, 0.62 to 0.99 % for soil-soil sliding friction, and 23.1 to 64.6 % for idle power. This model was however, not based on any of the standard rotavator configurations, which makes its suitability for wider application questionable.

2.6 Summary and conclusion from the reviewed literature

The power performance and quality of work of rotavators is affected by many factors including the blade configuration, direction of rotation, depth of tillage, ratio of the peripheral speed to the forward travel speed (λ), and the soil condition. Irrespective of the direction of rotation, when other rotavator operational factors and the soil condition are held constant, increasing the depth of operation increases the specific energy requirements. In most cases the specific energy requirements increases at an increasing rate with depth of tillage but some studies reported a linear relationship between increments in these two factors for the same soil conditions for a given rotavator. Consequently, there is no consensus on the specific energy requirements of rotavators with the increase in depth of tillage.

The direction of rotation of a rotavator has significant influence on the power performance and the quality of work. For the horizontal axis rotavators, the direction of rotation influences the power demand and determines whether a thrust force or a pull force will be exerted on the tractor. In most studies cited in reviewed literature, the magnitude of the push/pull forces were not measured and were therefore not included in the calculations of the rotavator's power requirements. There are two reasons why the magnitude of the push/pull forces should be recorded. First, it would ensure that the calculations of the specific energy requirements takes into account the influence of the push/pull forces. The secondly reason is that it would allow for the quantification of the magnitude of the push forces that may be detrimental to the tractor's operation. The measurement and determination of the resultant horizontal force would enable the correct determination of total power requirement for a given rotavator operated under specified operational and soil conditions.

The soil physical condition has a marked influence on the power performance and the quality of work of rotavators. The soil condition is affected by the soil water content, which has significant influence on other soil strength properties. Pertinent soil parameters that must be monitored for the quantification of tillage tool's performance include the soil water content, soil bulk density, soil shear strength and the soil-metal friction parameters.

There was no documentation in literature, on the application of the ANN and DEM for modelling techniques for the predicting the energy requirements of both passive and active tillage tools. This is attributed to the difficulty of modelling soil-tool interaction and soil failure by these methods, during dynamic processes such as tillage operations. On the other hand the application of the analytical models for the prediction of energy requirements for passive tillage tools has been extensive, and has produced acceptable results for predicting the energy requirements of passive tillage tools. Furthermore, since the behaviour of the rotavator blade, at an instantaneous time moment resembles that of a wide passive blade, the analytical models that have been used to predict the energy requirements of passive tillage tools can, form a basis for the development of an energy requirements prediction model, for rotavators.

There are many models that have been developed for the prediction of the draft force for a passive blade. However, for the rotavator, only empirical models have been proposed, for the different cutting blades in different soils. Empirical models, by nature are limited in applicability, and can therefore not be used to design blades that may result in improved performance of rotavators. From the review of literature presented in this chapter, the development of analytical models for prediction of torque requirement of rotavators is necessary, if better rotavator blades are to be developed.

2.7 Justification

The literature reviewed revealed that numerous studies have been undertaken on the passive tillage tools. The literature reviewed also indicated that only a limited number of studies have been undertaken on active tools. Furthermore, the studies undertaken for predicting the rotavator torque requirements were empirically-based, thus having limited applicability. Therefore, there is need for the development of analytical models based on the actual rotavator blade configurations for the accurate prediction performance of rotavators.

As a tillage tool, the rotavator has a number of advantages over the passive tillage tools. However, its adoption as an alternative to the passive tools has been hampered by its perceived excessive power requirement, particularly in deep tillage. Owing to its capability for significant reduction in the number of tillage operations necessary for achieving the required tillage tilth, this tool offers great potential for the reduction of the land preparation costs. Land preparation costs using the conventional tillage system is becoming prohibitively expensive owing to the escalating cost of fossil-based fuels and oils. Therefore, there is need to characterize the performance of the rotavator over a wider range of tillage depths. In addition, there is need to quantify the behaviour of the resultant thrust force generated when the rotavator is operated in the concurrent and counter-current directions of rotation, at different tillage depths.

Therefore, a study on performance characteristics of a deep tilling rotavator was proposed.

2.8 Hypotheses

The hypotheses of this study were as follows:

- The existing analytical models for passive blades can be adopted and modified to form a basis for formulating a model that can be used to model the performance indicators of a rotavator fitted with a standard commercially available rotavator blade

- The performance of a rotavator can be quantified through the measurement of its torque requirements and the resultant horizontal thrust force generate during its operation.

- There exists an optimum depth at which the resultant horizontal thrust force generated is greatest for a down-cutting rotavator of given design dimensions and operational conditions.

2.9 Objectives

The objectives of this study were to:

1 Design, fabricate and test an instrumented experimental deep-tilling setup to measure rotavator performance, including the thrust forces generated and the corresponding instantaneous rotavator operational parameters including, set tillage depth, instantaneous depth, rotational and forward travel speed and forward distance travelled.

2 Design, fabricate and test an instrumented soil strength characterisation apparatus capable of measuring the soil strength properties at different depths.

3 Quantify the field performance of an experimental deep-tiling rotavator over a wide range of set tillage depths based on the measured torque requirements and the resultant horizontal thrust forces generated.

4 Develop an analytical model that uses the soil dynamic parameters to predict the torque requirements of a rotavator fitted with commercially available rotavator blades.

5 Compare the predicted and the measured torque requirements of a rotavator fitted with a commercially available rotavator blade type.