University of Southern Queensland
Faculty of Engineering & Surveying

Analysis of a no-till toolbar for a Two Wheeled Tractor

A dissertation submitted by

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towards the degree of

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Abstract

Currently there is a research project being undertaken by Mr R.J Esdaile of Tamworth, New South Wales, in conjunction with the Australian Centre for International Agricultural Research on the development of a No Tillage Seed Drill for uses in third world countries. This project involves the development of low-cost no tillage seed drills to be made available to farmers in South East Asia.

A prototype seed drill has been designed and manufactured. However there was no rigorous analysis of the structural integrity of the steel frame (toolbar) and tine design. The aim of this research project is therefore to analyse the current design and to make recommendations that may improve the product as a whole.

In order to determine the forces that the toolbar may encounter, tractive force predictions were calculated, along with a sensitivity study of the impact of extra wheel weights. It was found that when compared to the fundamental Bekker model, a more reasonable result is obtained by the empirical Gee Clough tractive force prediction model. The predicted tractive force is approximately 1.5 kN. It was also found that the rake angle of the tine should be reduced from 55 to 30 degrees in order to minimise the draft force being produced.

Using the predicted tractive force, analysis of key components of the toolbar was also undertaken using the Finite Element software COSMOSWorks. It was found that the strength of original design is more than adequate for the designed purpose. It is therefore suggested to decrease the current wall thickness of the square hollow section from 4mm to 3mm.

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I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Mark John Fraser

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Signature

Date

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Chapter 1

Introduction

This research and design project was made available through Dr. Guangan Chen of The University of Southern Queensland but is sponsored by Mr R.J. Esdaile of Tamworth NSW. A current research project being conducted by Mr Esdaile of ACIAR on the design and manufacture of no-till seed drills for power tillers or 2 wheeled tractors for use in third world countries of southern Asia. The aim of this research and design project is to investigate the structural integrity of the frame of the seed drill that has been constructed.

1.1 Development of the Project Topic

Whilst on a recent trip to Bangladesh and Cambodia, the sponsor, Mr. Jeff Esdaile, noticed a lack of conservation farming practices in place. This can be attributed to the lack of information available to the farmers in such countries. Whilst there is International Maize and Wheat Improvement Center (CIMMYT) and the Wheat Research Centre in NW Bangladesh, the flow of information between researchers and farmers is limited.

This led Mr. Esdaile to approach the Australian Centre for International Agricultural Research (ACIAR) for a grant to conduct a research project into the development of conservation farming implements to be used by two wheeled tractors (power tillers) in Bangladesh, Cambodia and Laos. This project involves the local manufacture and testing of prototype seed drills. Once tested, the seed drills are then transferred to key ACIAR projects in South Asia. As a part of the ACIAR project, this specific research project is responsible for investigating the structural integrity of the universal style toolbar for the power tiller seed drill.

1.1.1 Farming Practices in Southern Asia

As most countries in southern Asia are third world countries, most people only eat the staple foods. This is because they can not afford to import produce from other countries. Consequentially, the majority of food for the country is grown by the people.

As money is hard to come by and some comodities can be quite expensive, waste is very scarce. This means that if a farmer leaves the stubble from a crop in the paddock, someone else will come along and take it themselves. This is because the crop stubble can be used as a fuel for a stove or as a bed or feed for an animal. Also, if the farmer doesn't take it and use it as fuel for the family stove for example, he must then go out and purchase Kerosene for it. Consequentially, only minimal stubble is left in field.

The International Maize and Wheat Improvement Center (CIMMYT) and Wheat Research Center have done work on cropping in countries such as Cambodia, Bangladesh and Laos. This includes work on many different plants, in different soil types. It also involved the correct application rates for both seed and fertilizer. Farmers are aware of this and it is in practise in many places.

1.1.2 Project Background

There was a project done by the International Maize and Wheat Improvement Centre in conjunction with the Wheat Research Centre (WRC) on a no till seed drill. This research was completed soley in Bangladesh. However, the main aim of the project was to investigate the benefits of no till or conservation farming techniques, not the structural integrity of a universal no till seed drill.

The machine that was used for the trials is a very simple looking machine as can be seen in Figure 1.1. It incorporates a set of press wheels that follow the tines to cover the seed and aid in depth control during sowing and transport. However, there was no mention of the structural integrity of it or the

The conclusion from this research project was that planting costs were significantly less than conventional methods but this came at a cost of a slightly reduced yield. The literature released upon completion of this research encouraged farmers to take on a zero or no till farming system in order to facilitate timely planting of crops.

Figure 1.1: A modified version of the no till seed drill made by CIMMTY.

1.1.3 Conventional Seed Drills

A seed drill is a mechanical device that is used for planting seed into the soil surface, usually between 40 and 100mm from the soil surface. A tine or disc is pulled along by a frame or toolbar through the soil, usually with a tube behind it which is used to deliver the seed and/or fertilizer. Figure 1.2 shows the set up of a basic tine assembly used for seeding. There have been numerous modifications made to this basic design, the most noted being the inclusion of a spring release mechanism which is otherwise known as a stump jump tine. This is to allow the tine to move backwards if it hits an anchor such as a rock or a tree root instead of causing the whole machine coming to a dead stop. Whilst this does make it more expensive to manufacture, it decreases wear on the entire machine and thus it will last longer. See Figure 1.3 for the basic design of a spring release or stump jump tine.

Disc openers have only started to be used in seeding relatively recently. However, the cost of the disc opener setup is much greater because of the increased number of moving parts and the engineering involved in the manufacture of them. The advantages with using a disc opener setup is that it is able to cope with very large amounts of residue on the soil surface and it requires less power to pull it through the soil due to the decreased draft force. See Figure 1.4 for a diagram of the workings of a disc opener arrangement.

As the tine passes through the soil, a metering wheel turns a series of gears which turn rollers. The rollers control the rate at which the seed and fertilizer are dropped down the delivery tubes and placed into the soil via the tine boot which is attached to the sowing tine. Some larger machines used forced air flow to push the seed through the delivery tubes as not all delivery tubes are vertical.

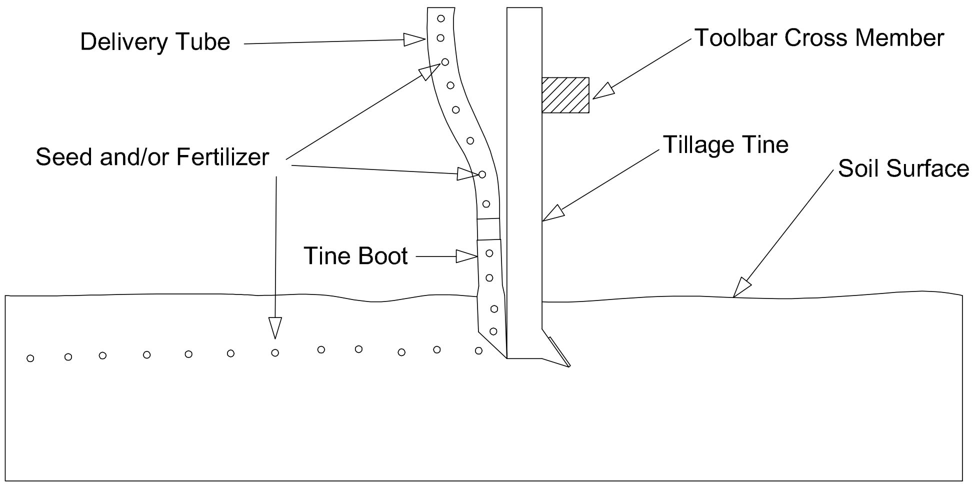

Figure 1.2: A typical seeding tine setup showing the delivery of seed and/or fertilizer into the soil.

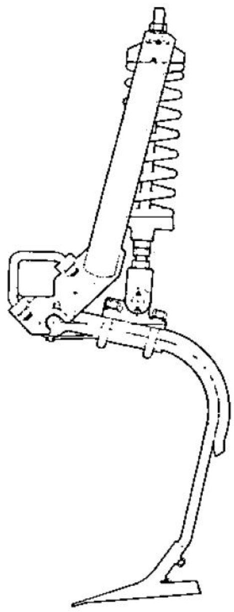

Figure 1.3: A tillage tine with a spring release mechanism (Source: G. Quick 1982).

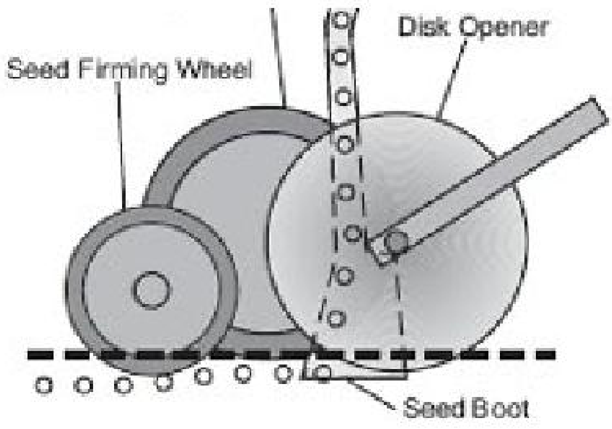

Figure 1.4: A disc opener seeding arrangement (Source: Grisso et al 2007).

1.1.4 Strip Tillage Seed Drills

Strip tillage planting systems are not as common as conventional seed drill systems but are somewhat similar. Instead of large volumes of soil being disturbed, only the small area directly in front of the sowing tine is tilled. This is achieved using a rotary hoe or rotovator attachment operating in front of the sowing tine. This is done to break down larger clods of soil to create a better seed bed for the soil. As well as doing this, it reduces the draft on the sowing tine because there is less resistance as the soil clods have already been broken up.

A rotary hoe or rotovator is an implement that is mainly used in relatively small agricultural operations. It consists of a horizontal shaft to which a number of angled blades are connected. The horizontal shaft is then connected to the power output of the tractor via a gearbox of some kind. See Figure 1.5 for a diagram of a rotary hoe designed for a small 4 wheeled tractor with three point linkage. Figure 1.6 shows a power tiller with a rotary hoe attached.

Figure 1.5: A typical rotary hoe designed for a small tractor.

Figure 1.6: A power tiller with a rotary hoe attached.

Figure 1.7: A trial of a strip tillage seed drill in Bangladesh.

One of the drawbacks of a rotary hoe is that some of the soil tilled by the rotovator does not fall back into the slot that has been cut. This results in lines being left across the field which can be seen clearly in Figures 1.7 and 1.8. Also, if the soil is too moist and press wheels are being used to ensure the seed has good contact with the soil, it can make a very big mess in the field as seen in Figure 1.8. The main disadvantages with using strip tillage is the cost associated with the rotovator or rotary hoe attachment and the large power requirement of the rotovator/rotary hoe.

Figure 1.8: The result of using a strip tillage seed drill in soil that has a high moisture content.

1.1.5 Operating Issues

The completed machine is being designed for use in third world countries. After looking at their farming practices from photos, it is fair to assume that safety guards designed and put on the power tiller and/or seed drill will be removed immediately after delivery. Due to project funding issues, it was decided that safety guards wouldn't be included as this would increase the final cost and they wouldn't be used when the implement is being used.

Due to the nature of the machine having a diesel engine, an exhaust is a necessary component. However, it does pose a number of issues for the operator. The main issue with an exhaust system is noise prevention or lack thereof. It has been proven that prolonged exposure to very loud noises does lead to deafness. Along with this, there is the issue of the exhaust gases being blown back onto the operator who has to walk behind the power tiller. Prolonged exposure to the exhaust gasses produced by the diesel engine may lead to respiratory problems in later life.

With the simple design of the power tiller, apart from the natural give in the tyres, there is no extra suspension to remove vibrations caused by the travel of tyres over the ground. As well as this, there are vibrations caused by the single cylinder diesel engine. Due to the simplicity of the machine which is required for use in third world countries, there is limiting dampening between the engine and the frame of the power tiller.

The issue of exposed belts and pulleys poses an enormous issue to the operator. This is because clothing and limbs of the operator can become caught or entangled. This can ultimately result in the loss of limbs and possible fatalities.

For a graphical illustration of the above problems, see Figure 1.9. The specific problems related to exposed chains and belts are enclosed by bright green boxes.

Figure 1.9: A photo highlighting some of the safety issues related to the project.

1.2 Project Objectives

1.2.1 Aim

This project aims to research current no-till systems for a power tiller and analyse a current supplied design.

1.2.2 Objectives

1. Identify requirements of target user market of the product.
2. Research current products for power tiller systems incorporating no till systems.
3. Complete a Finite Element Analysis of the current design as supplied from the sponsor.
4. Suggest improvements that could be made to the design of the frame and the power transmission system, and discuss the feasibility and practicality issues of such improvements.
5. Derive the maximum number of tines that can be used by the power tiller in a no till system.

1.2.3 Optional Objectives as Time Permits

6. Investigate the possibility of increasing the maximum tractive force exerted by the power tiller and implications on the other components of the machine and;
7. Modify the toolbar and tine design accordingly.

1.3 Methodology

In order to be able to arrive at a firm conclusion, a number of steps and processes will be used to analyse the frame of the no-till seed drill for failure points. The processes used involve the theoretical testing of the frame using Finite Element Analysis (FEA) by computer software packages. A literature require of the applications of the Finite Element Method is found in chapter 2. Whilst results from the FEA for this project can be found in chapter 5 and parts of chapter 6.

Although in the power tiller operators manual there is a value supplied for the maximum tractive force that a power tiller can exert, the weight of the new seed drill will alter that. The Bekker and Gee Clough models will be used for calculating the maximum tractive force. Along with this, a theoretical scenario of adding a 20 kilogram weight to both of the drive wheels will be investigated to see the impact on the maximum tractive force. FEA will be used to calculate if the frame will be able to withstand all the forces and torques. Once the original design has been tested using FEA, reductions in section sizes, mainly wall and solid section thicknesses will be attempted and put through the same process.

There is also the issue of the 4 bolts that are used to connect the mounting bracket of the frame to the rear of the transmission casing being able to withstand the forces. If the bolts will no be able to withstand the maximum force exerted by power tiller then a lower force will have to be adopted for the maximum draft force that the seed drill will be able to exert. If the bolts can not withstand the maximum force of the power tiller then when the seed drill exerts the draft force, it may end up shearing the bolts and leaving the seed drill behind.

Chapter 2

Literature Review

With ever increasing costs involved in agricultural production, there is a lot of work being done in the field of conservation tillage or zero tillage. This is in an effort to decrease the cost of growing a crop. A lot of working being done is concentrating on reduced or no tillage cropping systems. This is because results can be seen relatively quickly making it easy to class as it either a success or failure.

The process of predicting parameters such as the draft force on a tine or the rolling resistance of a tyre involves the use of both theoretical and empirical models. As with any model though, there are a number of assumptions made. The biggest issue is finding a model that produces results that conform with results collected from practical experiments. When working with soil, it must be remembered that it is an anisotropic substance. Because it varies so much, this causes inaccuracies in results collected from experiments. Whilst there are a few common soil parameters used in different models, not all are the same and there are also different methods of determining the same parameter. Whilst it is possible to get a very realistic answer, it should only be used as a representation and not as solid fact as there will always be slight inconsistencies caused by soil inconsistencies.

2.1 Farming Techniques

2.1.1 Conventional Tillage

Conventional tillage cropping systems use tillage events as a method of weed control and preparing a good seed bed. During the process of tilling the soil, bigger clods which may contain weed roots are broken up into small pieces and the surface soil is often inverted. This not only kills the weeds but it also destroys soil structure. This causes water infiltration and nutrient uptake rates to be significantly reduced. The task of pest and weed control is achieved through soil inversion but comes at the cost of a decrease in soil structure and stability.

2.1.2 Conservation and Zero Tillage

Conservation tillage involves applying different management techniques compared to conventional tillage in terms of field preparation in order to grow a successful crop. In this situation, conservation and zero tillage are very similar and the words are constantly interchanged. The main aim is to minimize soil disturbance prior to sowing and to increase soil cover in the form of crop residue. Once the crop has been harvested, the crop residue or stubble is left on the field to decompose on or into the soil. There are a number of benefits that come with using a conservation or zero tillage cropping system. Whilst there are physical factors such as increased crop yields and decreased erosion caused the increase in ground cover, there are other benefits that can not be seen by the naked eye.

By reducing the number of tillage events before planting, the operator is achieving a number of small objectives which make up a conservation tillage farming system. The reduced traffic cuts the fuel requirement per unit area to have the crop planted. By reducing the field traffic, compaction is reduced which can be responsible for a breakdown in soil structure and decrease in water infiltration capacity. Consequentially, soil stability and porosity is increased due to the lack of disturbance which provides an environment which will promote plant growth.

According to Charman and Murphy, soil fertility is defined as the soil's ability to support plant growth whilst being able to maintain a stable soil structure. The most important factor related to maintaining soil fertility is the organic matter content of the soil. Organic matter in the soil is responsible for a lot of small things that add up to have a large impact on the growth of plants. Everything from nutrients to aeration is affected by the organic matter content. See Figure 2.1 for how the organic matter content of a soil affects the yield of a wheat crop.

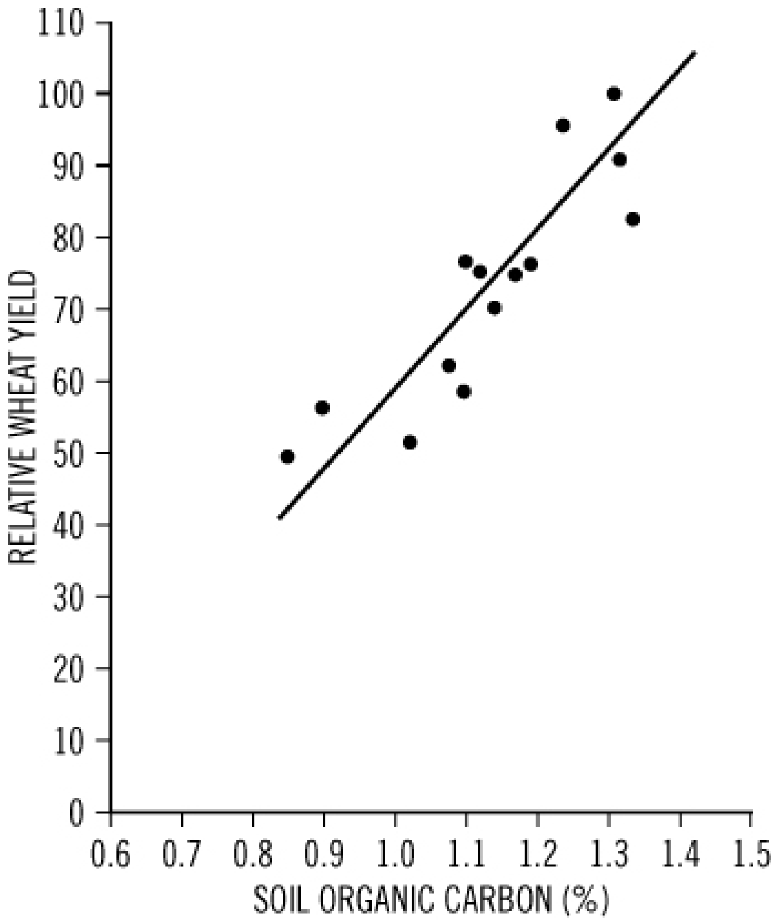

Figure 2.1: A graph showing the trend between organic carbon content and yield of a wheat crop (Source: Charman and Murphy 2007).

2.2 Power Tillers

A Power Tiller or Two wheeled Tractor is a very common piece of machinery found predominantly in third world countries. Being relatively cheap to purchase, they have facilitated the mechanization of farming practices in many third world countries. They are able to do be used for many different applications on farms in third world countries which make them a very good investment. These activities include soil tillage, seeding, pumping of water for irrigation via the use of a bolt on pump and haulage of goods to market via the connection of a trailer.

They are powered by a 12 to 15 horsepower diesel engine mounted at the front of the machine. Power from the engine is transferred to the gearbox via a "v" belt drive. The gearbox is very simple and can range anywhere from a single forward and reverse configuration to a five or six forward gears and one or even two reverse gears. The engine is started manually using a crank handle and a decompression valve with an electronic cutoff switch. See Figure 2.2 for a picture of a power tiller.

Figure 2.2: A Power Tiller.

2.3 Other Available Seed Drills

Due to the nature of the implement being investigated, other suitable alternatives are very hard to find. This is primarily due to the fact that power tillers are mainly used in third world countries. The majority of the seed drills made for the power tiller are made by local fabricators and are not mass produced. There have been a number of research projects funded by the WRC, ACIAR and CIMMYT which have had seed drills built for no till farming systems but nothing else has resulted in terms of making more to be sold to the farmers themselves.

During researching for other available products, there was another commercially made seed drill for no tillage systems but there was no information or specifications available on it. See Figure 2.3 for a photo showing the lack of residue clearance between the frame and the tine. Also, it is clear from the photo that it is not built very strong and may well not be able to stand up to a no tillage situation.

Figure 2.3: A cheap Indian made seed drill.

Figure 2.4: A Chinese made seed and fertilizer drill.

2.3.1 Commercial Products

When researching other available commercial products, there are very few that are advertised. Due to relatively low labour costs and cheap parts, they are mainly manufactured in China. Again, there is a lack of available information on these seed drills. From what can be found, they are very simple and do not look very strong and may not be able to cope with the larger forces involved in no till situations.

Both the seed and fertilizer meter are driven from a steel wheel which follows behind the sowing tine. This is a problem because if there is not adequate weight on it, it slips across the soil surface. This will result in no seed or fertilizer put down the tube behind the tines. See Figure 2.4 for a picture of the Chinese seed drill. From Figure 2.4, it is easy to see that the clearance between the tine and the seed and fertilizer box is very small. If being pulled through large amounts of crop residue, there is a high probability that this will end up being blocked up. Ultimately, it is not a very good option if a sound, reliable system is wanted for sowing seed.

2.4 Finite Element Analysis

The Finite Element Method (FEM) is a numerical tool used to predict the behaviour of a continuous physical system. A finite number of algebraic equations which describe a mathematical model of the system are used in order to create a solution. A mesh or grid consisting of a finite number of elements is created of the system. The solution to each element is found by simultaneously solving the equations which are used to model the system.

There have been a number of studies undertaken to validate the use of the FEM in the area of agricultural implement force analysis. In particular, the FEM has been validated as an appropriate tool for tillage tool design. Laboratory tests done by Mouazen et al (1999) were used to validate a FEM model of a subsoiler. Tests were completed for four combinations of different tine and shank rake angle. The draft force and the soil failure dimensions were recorded for each test. Whilst the FEM did predict a larger force, the error was only in the range of 11 to 20%. It was found that the FEM did make a good approximation of the soil failure.

The Finite Element Method was also used by Formato et al (2005) for simulating the soil interaction with a Mouldboard plough. The process involved simulating the plough operation by creating a three dimensional model of the Mouldboard and a material that had similar characteristics to soil. Once the theoretical analysis had been completed, field tests were performed. In order to make a comparison against the FEM results, piezo-resistive transducers which measured normal stresses. A grid of 14 points on the working surface of the Mouldboard was used and through interpolation, a comparison could be done with the theoretical analysis. It was found that the maximum error calculated between the two sets of results was found to be less than 10%.

These papers validate the application of the FEM to modelling agricultural systems. With the different tests yielding a maximum error of 20% between the theoretical analysis results and the results from field testing, this is a very acceptable result. This is due to the anisotropic nature of soil, predicting any force relating to it is inherently inaccurate.

Chapter 3

Methodology

3.1 Power Tiller and Seed Drill Specifications

The power tiller of interest for this project is a Dong Feng brand DF-12. It is made in China with a cost of approximately $2000 USD. The specifications listed below in Table 3.1 were sourced from the owners manual which was supplied with the power tiller. The information regarding the weight of the traction machine is not clear to whether it includes the weight of the rotovator or not. It was decided that the dry weight would include the weight of the rotovator. The standard tyre for a power tiller is a pneumatic tyre with a diameter of 0.6m and a width of 0.15m. This can be seen in Figure 3.3. Figures 3.1 and 3.2 show the approximate main dimensions of the seed drill and power tiller.

Power Tiller Specifications	
Engine Power	8.95 kW
Maximum Torque	42 N.m
Rated Speed	2000 RPM
Gearbox Arrangement	6 Forward, 2 Reverse
Operating Speed Range	1.4 - 15 km/hr
Tractive Force	2126N
Tyre Diameter	0.6m
Tyre Width	0.15m
Tyre Wall Height	0.15m
Wheel Track Width	580 - 800 mm (adjustable)
Fuel Consumption	257g/kW.h
Seed Drill Specifications	
Dry Weight	150 kg
Total Commodity Box Capacity	65Litres
Seed Capacity	25kg
Fertiliser Capacity	35kg
Weight when loaded	210kg
Implement Width	1.1m
Implement Height	1.5m
Tine Width	12mm
Press Wheel Diameter	250mm
Overall Specifications	
Total Weight (dry)	500kg
Total Loaded Weight	560kg
Maximum Width	1m
Maximum Height	1.5m
Total Length	2.7m

Table 3.1: Specifications of the Power Tiller and Seed Drill.

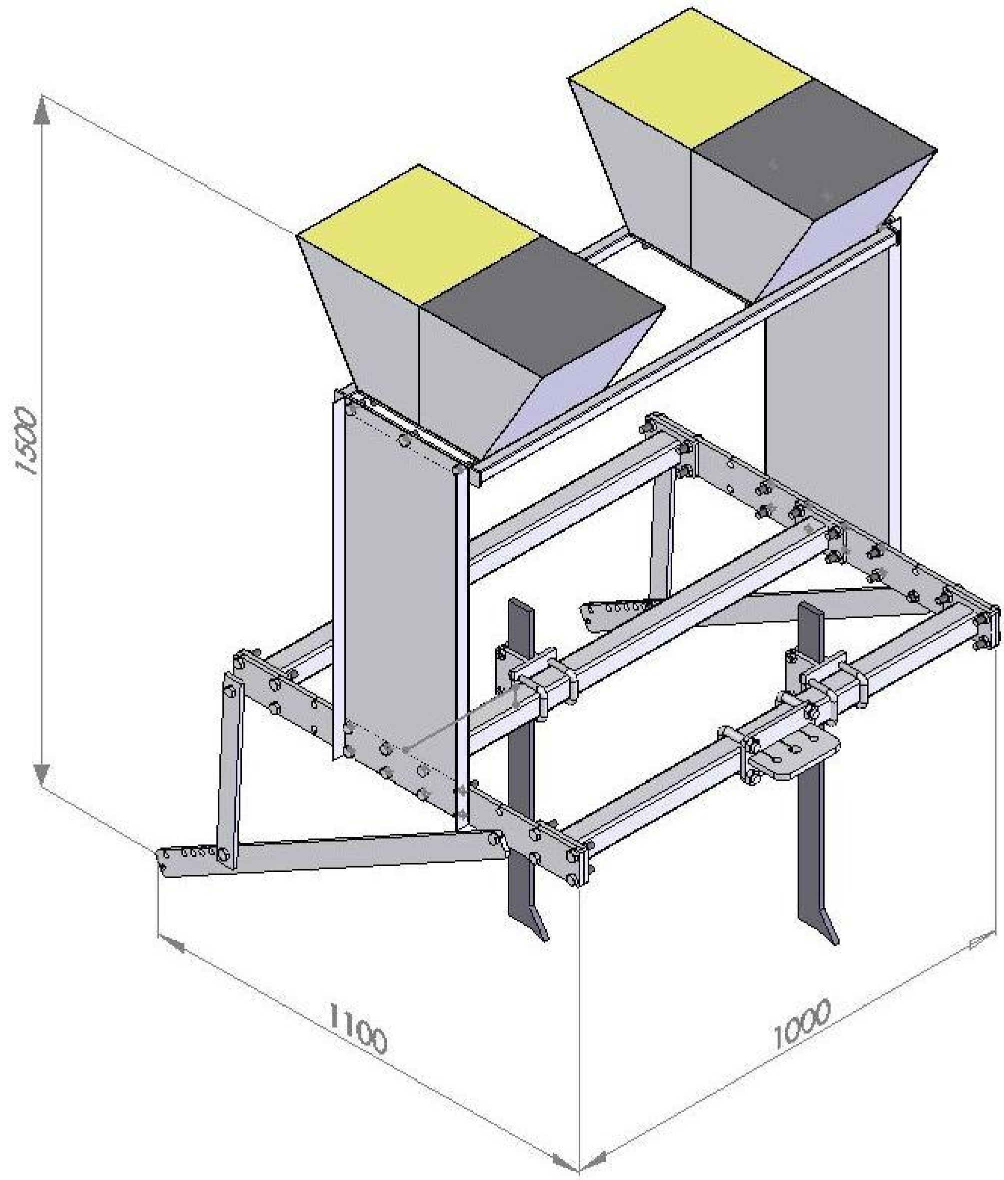

Figure 3.1: An illustration showing approximate dimensions of the seed drill.

Figure 3.2: An illustration showing approximate dimensions of the power tiller and seed drill.

Figure 3.3: An illustration showing the standard pneumatic tyre for a power tiller.

3.2 Soil Conditions

Due to project limitations, it is not possible to travel to varying locations in southern Asia to collect data relating to local soil conditions. In order to simplify the analysis of the force prediction, a number of assumptions were made. From a pdf document posted on the Cranfield University website written by Godwin et al, there is a list of typical soil parameters that are recommended where data can not be obtained. Along with data from the Cranfield University website, soil stiffness coefficients were found from Appendix 4 from McKeys (1985). For all of the analysis, see the table below for a list of soil parameters used.

Bulk Unit Weight	14 kN/m^3
Cohesion	30 kN/m^2
Internal Friction Angle	10^o
Angle of Soil-Metal Friction	6^o
Adhesion	0 kN/m^2
k_c	6 kPa/m^{n-1}
k_ϕ	400 kPa/m^n
n	0.16
K	0.06m
Cone Index	600 kPa

Table 3.2: Table of soil parameters used in tine draft force and tractive force predictions.

3.3 Tine Draft Force Prediction Model

Over the years, there has been a lot of work done on the interactions that occur during tillage events. The main interactions that are of interest occur between soil particles, the metal of the tillage tool and other soil particles. There have been a number of formulae derived for calculating the forces involved with tillage tools of varying sizes and shapes. It is a very complex topic as it is a dynamic system and is complicated by the anisotropic nature of soil composition.

The main method of failure in soil is through shear. The shear strength of a soil can be measured in a laboratory or in the field. In a laboratory, there are two main tests that can be performed, the direct or triaxial shear test. In the field, it can be done using a shear box, shear vane or a penetrometer. From a shear test of a soil, you can collect important data such as Cohesion and Internal Friction Angle. When put together, you can create a graph that shows the relationship between normal and shear stresses. Typical values of cohesion range between 10 and 40 kPa and internal friction angle ranges between 25 and 40 degrees. Cohesion is influence mainly by the organic material content of the soil whilst internal friction angle is influenced by clay content and the degree of compaction. See Figure 3.4 for a graphical representation of this relationship.

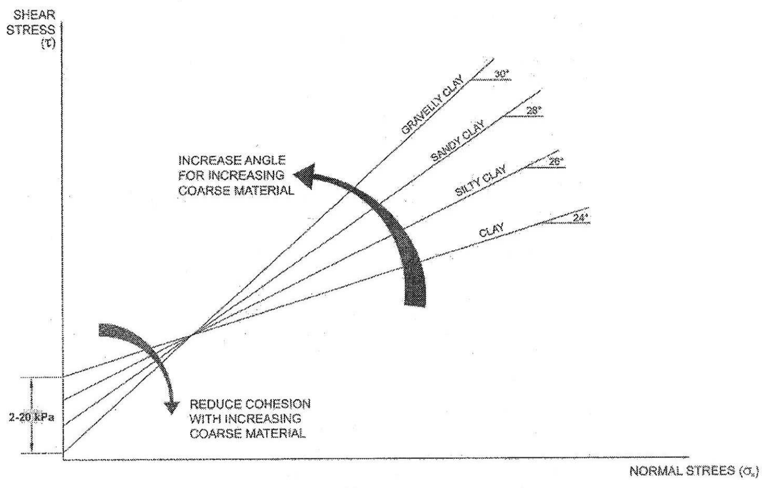

Figure 3.4: Indicative variation of clay strength with changing granular content.

3.3.1 Soil Failure

During the process of soil tillage, a blade is pulled through the soil to cause failure. This results in the clods of soil being broken apart. Depending on the depth at which a cutting blade operates, failure can occur by either tensile or compressive stresses as well as shear stresses. When looking at the specific case of a blade being pulled through the soil, there are a number of methods of analysis. The simplest yet the most commonly used model of three dimensional soil cutting is three dimensional wedges model by Mckeys and Ali (1977). See Figure 3.5 for a graphical display of the three dimensional wedges model. Whilst being very similar to the model put forward by Godwin and Spoor (1977), there are slight differences which makes analysis simplified but with the same degree of accuracy.

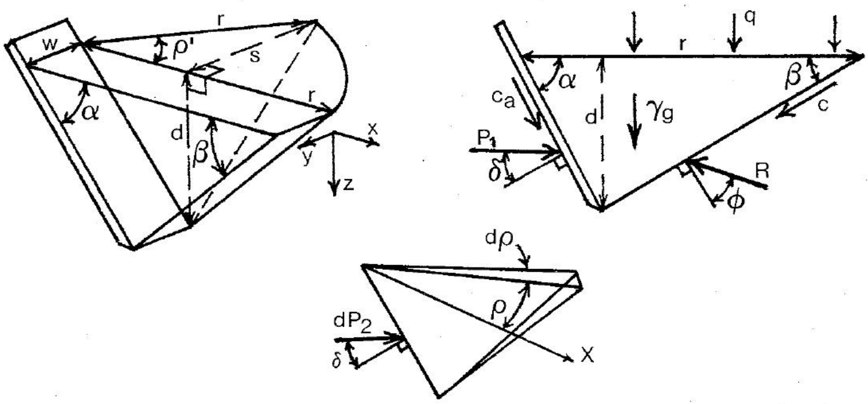

Figure 3.5: The three dimensional cutting model of Mckeys and Ali (1977), showing the forces and pressures on the centre zone, and an elemental segment of included and dρ in the side crescent. Source: Mckyes (1985)

3.3.2 Tillage Force Prediction Models

There have been a number of people that have developed equations that can be used to predict the force encountered by different types of tillage tool. R.J. Godwin and E.Mckeys are the authors of many papers that have been written on the subject but equations developed by numerous other people are also used. However, because soil is an anisotropic substance, all models that are used to predict the total force encountered by a tine during a tillage event will have a degree of error.

The first parameter that must be defined is the type of tool being used. According to Godwin et al, tines are divided into three categories depending on the depth to width ratio. See Figure 5.1 for illustration of the modes of failure for each tine category.

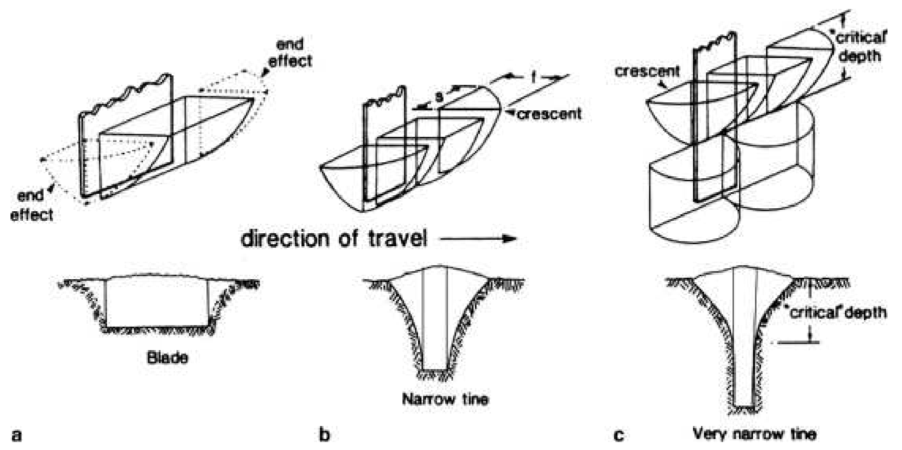

Figure 3.6: The effect of depth/width (d/w) ratio of the tine on the mechanics of soil failure.

1. Wide/blade Tine - depth/width ratio <0.5
2. Narrow/Chisel Tine - 1 <depth/width ratio <6
3. Very Narrow/Knife Tine - depth/width ratio >6

For analysis of wide blades, a two dimensional approach can be taken. This is because the relatively small contribution made by the sideways movement of soil, i.e. in the y direction as seen in Figure 5.1. However with a constant blade width, as depth increases so to does the contribution made by sideways movement of soil.

3.3.3 Governing Formulae

Regardless the tine being investigated, according to A.R. Reece the total tool force can be calculated from formula 3.1 found in Soil Cutting and Tillage by.

$$P = (\gamma g d^2 N_\gamma + c d N_C + c_a d N_{ca} + q d N_q) w \tag{3.1}$$

This formula is known as the Universal Earthmoving Equation. Where:

- P is the total tool force;
- γ is the total soil density;
- g is the acceleration due to gravity;
- d is the tool working depth below the soil surface;
- c is the soil cohesion strength;
- c_a is the soil adhesion strength;
- q is the surcharge pressure vertically acting on the soil surface;
- w is the tool width and
- N_γ, N_C, $N_c a$ and N_q are dimensionless factors which are dependant on the tool and soil properties.

N_γ, N_C, $N_c a$ and N_q are dependant on the tool depth to width ratio, rake angle of the tine, angle of surface friction and the angle of internal friction of the soil. Calculation of the total force P can be achieved by the use of the charts in Appendix 3 of Soil Cutting and Tillage by Mckeys (1985) to obtain values for N_γ, N_C, $N_c a$ and N_q.

Once the total force P, has been calculated, it is easily resolved into horizontal and vertical components using the following formulae as found in Soil Cutting and Tillage by McKeys (1985):

$$H = P\sin\alpha + \delta + c_a dw\cot\alpha \tag{3.2}$$

$$V = P\cos\alpha + \delta - c_a dw \tag{3.3}$$

However, some assumptions have been made. These being that unless there has been prior tillage, the surcharge loading will be zero and unless soil adhesion is known, it is assumed to be zero. R.J. Godwin and M.J. O'Dogherty state that the soil-metal adhesion had a very small effect on the cutting force P, it can be deemed to be negligible. This simplifies it down to the following equations found in Soil Cutting and Tillage by McKeys (1985) to calculate the horizontal and vertical forces:

$$H = P\sin\alpha + \delta \tag{3.4}$$

$$V = P\cos\alpha + \delta \tag{3.5}$$

3.3.4 Critical Depth

When investigating the different cases as shown in Figure 5.1, the different modes of soil failure must be noted. When dealing with either wide or narrow cutting blades, the mode of failure is only crescent. This involves the soil being moved forwards, upwards and sideways. However when dealing with very narrow cutting blades, if a tine is operating below a certain depth known as the "critical depth" then the failure mode changes. Below the critical depth, there is lateral failure in which the soil is pushed only sideways and forwards. There is no upwards movement of any soil below the critical depth. Lateral failure involves the soil failing by exceeding the compressive strength of the soil. Soil can tolerate much larger compressive stresses than tensile stresses. This results in a much larger force if a tine operates below the critical depth.

The location of the critical depth varies from soil to soil and is also dependant on the tine parameters. The location of the critical depth is found by using an iterative procedure and finding the depth at which a tine encounters a minimum resultant total draft. This is found by adding H and Q found from equations 3.4 and 3.9 from McKeys (1985). In order to calculate the critical depth, the following parameters must be calculated.

$$K_o = 1 - \sin\phi \tag{3.6}$$

$$N'_q = \left[\frac{1+\sin\phi}{1-\sin\phi}\right] e^{2(\pi/2+\phi)\tan\phi} \tag{3.7}$$

$$N'_c = \cot\phi \left\{ \left[\frac{1+\sin\phi}{1-\sin\phi}\right] e^{2(\pi/2+\phi)\tan\phi} - 1 \right\} \tag{3.8}$$

$$Q = \left[cN'_c(d-d_c) + \frac{1}{2}\gamma g K_o N'_q (d^2 - d_c^2) \right] w \tag{3.9}$$

To calculate the total force required to pull a tine through soil, it is necessary to calculate both H and Q. This involves using a number of varying trial critical depths. By using a spreadsheet layout, a very versatile solution can be created and can be easily made to fit different situations i.e. different tine dimensions or soil conditions.

3.4 Traction Force Prediction Models

Traction is the ability of a tyre of a vehicle to exert a force on the surface over which it is travelling. If a motorized vehicle is pulling a tillage implement, the net tractive force produced by the vehicle must be larger than the force required to pull the implement through the soil if the vehicle is to travel forward. However during the transfer of engine power to the ground surface, there will always be losses of some kind, whether it be through wheel slip or transmission inefficiencies. As the force that a tyre may exert on the soil surface is dependant on the mass distribution, a free body diagram of all forces being exerted on a tractor is a suitable starting point for the analysis of traction forces.

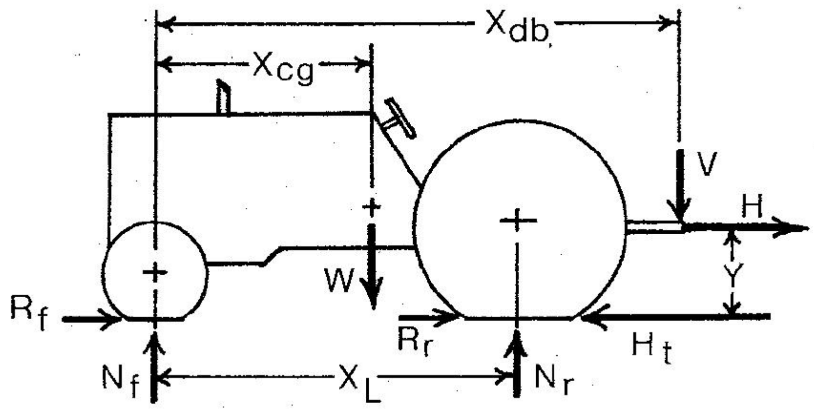

Figure 3.7: The free body diagram of a traction machine (Source: McKeys 1985).

Where:

- N_r is the total upward force from the ground to the rear wheels,
- N_f is the total upward force from the ground to the front wheels,
- w is the total machine weight, excluding the weight of the implement,
- X_{cg} is the distance from the front axle to the machines center of gravity,
- V is the vertical force transferred downward from the implement attached,
- X_{db} is the distance from the front axle to the implement attachment point,

- H is the total horizontal force required to pull the tillage implement,
- H_t is the total traction force of all driven wheels on the ground,
- Y is the height of the implement attachment point,
- X_L is the machine wheelbase,
- R_f is the rolling resistance force on the front tyres due to soil and tyre deformation,
- R_r is the rolling resistance force on the rear tyres due to soil and tyre deformation,

When doing the force balance equations, it comes out as follows:

Horizontal Axis:

$$H_t = H + R_f + R_r \tag{3.10}$$

Vertical Axis:

$$N_f + N_r = V + W \tag{3.11}$$

$$N_f = V + W - N_r \tag{3.12}$$

$$N_r = (WX_{cg} + VX_{db} + HY)/X_L \tag{3.13}$$

There are two main approaches that can be taken when approaching a problem involving traction of a machine. There is an empirical approach using the Gee-Clough equations for cultivated fields and fields with stubble still standing. The Gee-Clough equations were derived from experimental data using the line of best fit method for a number of different cases. The other approach being the theoretical model, the Bekker model which can be found in the book Soil Cutting and Tillage by E.McKeys (1985).

3.4.1 Coefficient of Rolling Resistance

As a tyre travels over the soil surface, the downward pressure on the soil will cause sinkage. If the traction machine is moving forward, it must overcome the horizontal resistance of the soil and "climb" out of the dent created by the sinkage of the soil. As the soil surface hardens, the rolling resistance will decrease as there will be decreased sinkage.

The most commonly used model for modelling rolling resistance is the Bekker model. However, when using this model a number of assumptions are made. The length of the contact area the tyre has with the ground is approximated as half the diameter of the tyre and the value tyre deflection for agricultural purposes is taken as 0.04.

Depending on the drive configuration of traction machines, the coefficients of rolling resistance either change or are assumed to be equal. This causes slight changes the process of analysis for the traction system. For example, in a traction machine with both the front and rear wheels providing traction, analysis of the system begins with calculating the coefficient of rolling resistance of the front wheels from the first 5 steps of the Bekker model. It is then assumed that the coefficient of rolling resistance of the rear wheels is only 60% of the coefficient of rolling resistance of the front wheels.

However, when dealing with a traction machine that is only driven by the rear wheels there is a different process. Analysis of the system starts with the rear wheels. Once again, using the first 5 steps of the Bekker model, calculate the coefficient of rolling resistance of the rear wheels. It is then assumed that the front wheels have the same coefficient of rolling resistance.

3.4.2 Wheel Slip

As soil is not a rigid material, when put under stress, deformation will occur. In the case of a tyre rolling across the surface trying to exert a horizontal traction force, shear stresses will cause horizontal deformation. See Figure 3.8 for a diagram of horizontal deformation.

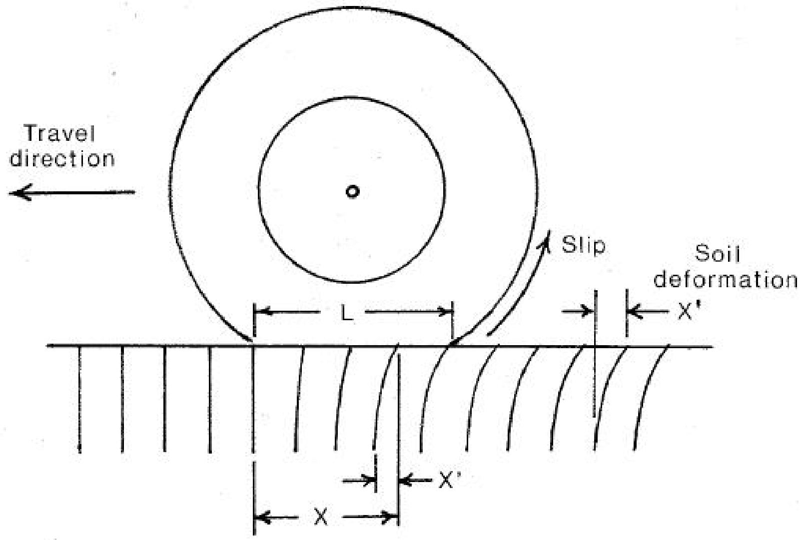

Figure 3.8: Horizontal deformation of soil as a flexible tyre rolls and slips as it applies a tractive force (Source: McKeys 1985).

In a controlled testing environment over a pre-measured distance, wheel slip is easily calculated using equation 3.14 and it is usually represented as a percentage.

$$S = \left(\frac{N_1 - N_0}{N_0}\right) \times 100 \tag{3.14}$$

Where:

- S is the wheel slip as a percentage.
- N_1 is the number of wheel revolutions under a loaded condition.
- N_0 is the number of wheel revolutions under an unloaded condition.

Alternatively, in the field, wheel slip can be calculated by comparing the velocity of the wheel to the velocity of the traction machine itself. See equation 3.20. As it is a comparison between the vehicle and wheel velocities, if they are the same then no wheel slip is occurring. However, in reality this is rarely ever the case.

$$S = \left(\frac{v_w - v}{v_w}\right) \times 100 \tag{3.15}$$

- S is the wheel slip as a percentage.
- v_w is the velocity of the wheel with respect to the vehicle.
- v is the vehicle speed with respect to the ground.

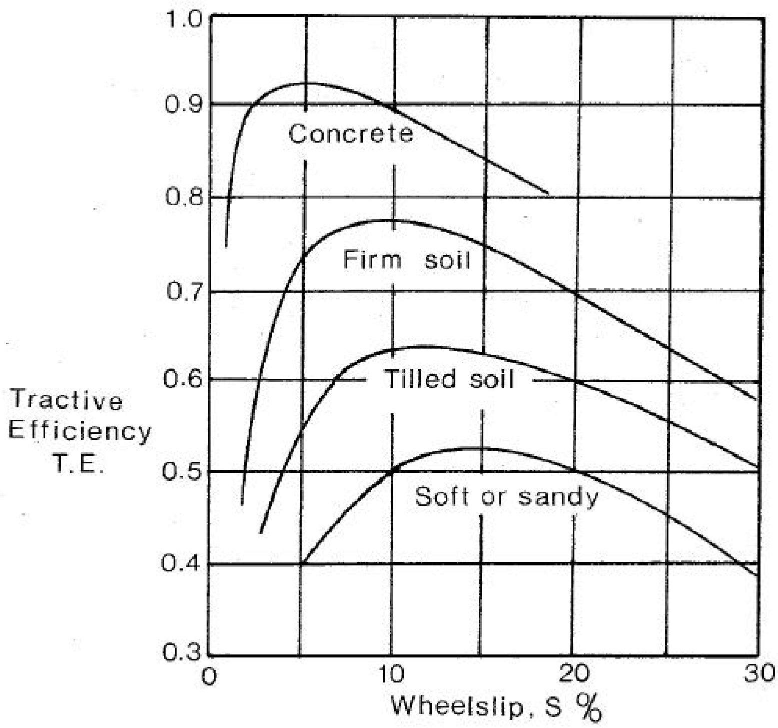

Figure 3.9: Typical curves of tractive efficiency plotted against wheelslip for two wheeled tractors with lugged agricultural tyres on different surfaces (Source: McKeys 1985).

When looking at the peak tractive efficiencies from Figure 3.9, it is important to realize that it is possible to achieve a higher efficiency by changing either the tyre or weight distribution but this does give a good starting point for the design of a traction system.

3.4.3 Process of System Analysis - Empirical Model

In order to be able to use the Gee-Clough equations to predict the tractive force, the following steps must be followed:

1. Calculate the tyre mobility number for the specific soil and tyre.

$$MN = \frac{CI \times b \times d}{W} \times \sqrt{\frac{\delta}{h}} \times \frac{1}{1+b/(2 \times d)} \tag{3.16}$$

   It is assumed that at the manufacturer's recommended load and pressure, for agricultural purposes δ/h will be equal to 0.2.

2. Calculate the Coefficient of Rolling Resistance.

$$CRR = 0.049 + 0.287/MN \tag{3.17}$$

3. Calculate the Maximum Coefficient of Traction.

$$CTmax = 0.796 - 0.92/MN \tag{3.18}$$

4. Calculate the constant k from the following equation.

$$kCTmax = 4.838 + 0.061MN \tag{3.19}$$

5. Calculate the Coefficient of Traction.

$$CT = CTmax \times \left[1 - e^{-kS}\right] \tag{3.20}$$

6. Calculate the Tractive Efficiency.

$$\eta = \frac{CT \times (1-S)}{CT + CRR} \tag{3.21}$$

Where:

- d is the tyre diameter.
- b is the width of the contact area.
- L is the length of the contact area.
- k is a soil constant.
- δ_t is the tyre deflection.
- S is the wheelslip.
- CRR is the Coefficient of Rolling Resistance.
- CT is the Coefficient of Traction.
- CTmax is the Maximum Coefficient of Traction.
- H is the total horizontal force at the drawbar.
- H_t is the total tractive force.
- R is the rolling resistance force.
- η is the tractive efficiency.

From here, it is possible to calculate the tractive forces exerted by the machine as well as the rolling resistance of the tyres in order to be able to calculate a net drawbar force. This is achieved by multiplying the coefficients of traction and rolling resistance by the verticle load on them. Formula for calculating the net drawbar force is as seen in equation 3.22.

$$H = H_t - R \tag{3.22}$$

3.4.4 Process of System Analysis - Theoretical Model

There a number of steps that must be taken when analysing a traction system. See below for a step by step guide of using the Bekker model for solving a simple traction situation.

1. Calculate the contact area of the tyre and the soil surface. On soft surfaces, it is accepted that the length of contact area is half of the tyre diameter.

$$A = bL \tag{3.23}$$

2. Calculate the pressure of the tyre on the soil surface.

$$P_{wheel} = \frac{N_{tyre}}{A_{tyre}} \tag{3.24}$$

3. Calculate the soil stiffness constant for sinkage.

$$k = \frac{k_c}{b} + K_\phi \tag{3.25}$$

4. Calculate the sinkage of the soil caused by the tyre.

$$z = \left(\frac{P}{k}\right)^{\frac{1}{n}} \tag{3.26}$$

5. Calculate the Coefficient of Rolling Resistance.

$$C_R = \frac{(z + \delta_t)}{d} \tag{3.27}$$

6. Calculate the rolling resistance force.

$$R = C_R N \tag{3.28}$$

7. Calculate the maximum tractive force available.

$$H_m = cA + N \tan\phi \tag{3.29}$$

8. Calculate the total traction force.

$$H_t = H_m \left[1 - \left(\frac{K}{SL}\right)(1 - e^{\frac{-SL}{K}})\right] \tag{3.30}$$

9. Calculate the total force at the drawbar.

$$H = H_t - R \tag{3.31}$$

10. Calculate the tractive efficiency of the system.

$$\eta = \frac{(H_t - R)(1 - S)}{H_t} \tag{3.32}$$

Where:

- d is the tyre diameter.
- b is the width of the contact area.
- L is the length of the contact area.
- A is the contact area of the tyre/surface interface.
- P is the pressure on the contact area of the tyre/surface interface.
- k is a soil stiffness constant for sinkage.
- c is the soil cohesion.
- ϕ is the internal friction angle of the soil.
- k_c and K_ϕ are soil stiffness coefficients.

- z is the tyre sinkage distance into the soil surface.
- n is a soil constant.
- δ_t is the tyre deflection, assumed to be 0.04 for agricultural purposes.
- C_R is the coefficient of rolling resistance.
- R is the rolling resistance force.
- S is the wheelslip.
- K is a soil stiffness value.
- H_m is the maximum tractive force.
- H_t is the total tractive force
- H is the total horizontal force at the drawbar.
- η is the tractive efficiency.

For a number of reasons the value obtained from the Bekker model will have a degree of error in it. These being that soil is an anisotropic substance and soil factors such as k_c, K_ϕ, K, n, c and ϕ can be found using a number of different apparatus and the amount of care taken collecting such data will reflect in the accuracy of the result. Even though there is a value given for the maximum tractive force that a power tiller is able to exert, analysis of the system will be undertaken as the maximum tractive force varies with soil conditions.

Chapter 4

Force Analysis

4.1 Power Tiller Weight Distribution

Before calculating the tractive force of the power tiller, it is necessary to calculate the weight distribution and if need be, add a counterweight to the front of the power tiller. This can be necessary as the seed drill will add a lot of weight to the rear of the power tiller and the operator needs to be able to lift it out of the ground at the end of the row and turn the implement around to begin the next row. The known weights are the engine, the seed drill, the seed and fertiliser and the force needed to lift the machine when stationary. These being 150kg, 150kg, 60kg and 15kg respectively. This gives the fully loaded seed drill a total weight of 210kg.

The process listed below was followed to calculate the various pieces of required information. All masses and forces are expressed in kilograms and distances are in millimeters. The process revolves around taking bending moments about the drive axle because it cancels the effect of the mass over the axle itself as there is no distance to multiply the force by.

1. Whilst stationary with no seed or fertilizer, approximately 15kg of effort was required to lift the power tiller. With the mass and center of mass of the seed drill known, it was possible to calculate the center of mass of the engine.

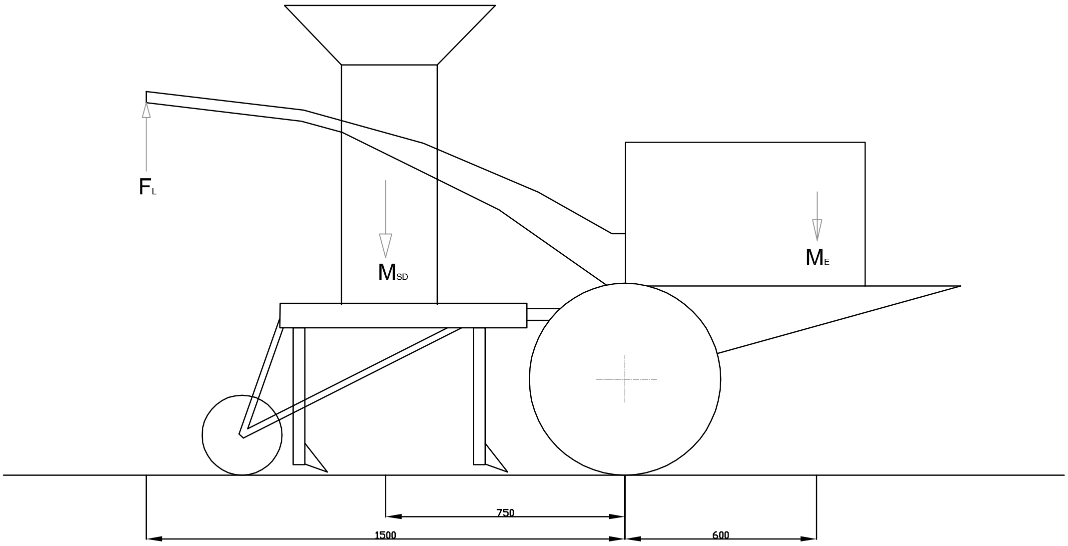

Figure 4.1: Stationary Power Tiller with Seed Drill to calculate the center of mass of the engine.

2. When the seed drill is fully loaded, the force required to lift the power tiller was calculated.

3. In order to reduce the force required to lift the fully loaded power tiller, the effect of a counterweight added to the front of the power tiller was investigated. See Figure 4.2 below for a graphical representation of the relationship between the mass required to lift the power tiller and the mass of the counterweight.

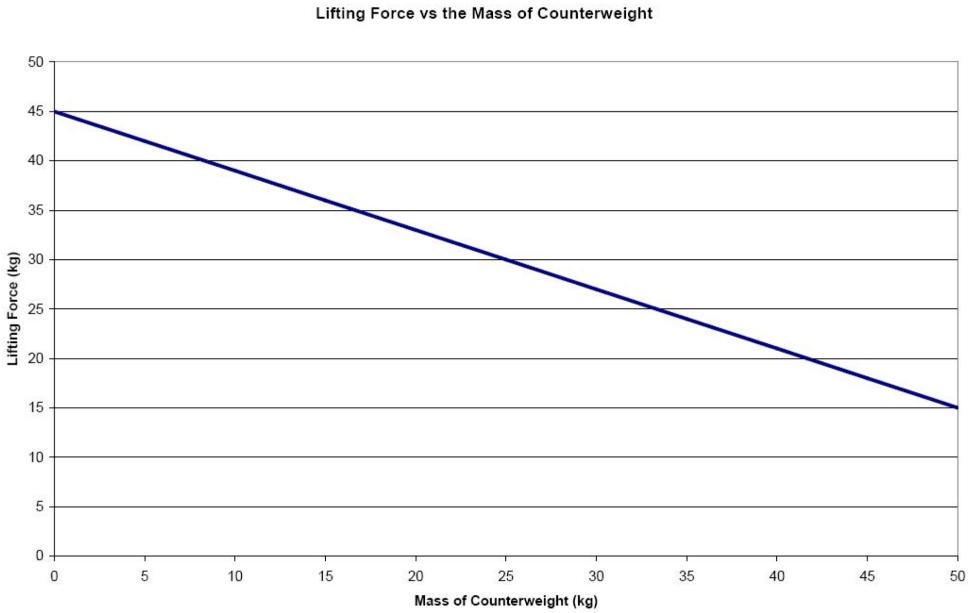

Figure 4.2: The relationship between the mass of the counterweight and the weight required to lift the fully loaded power tiller.

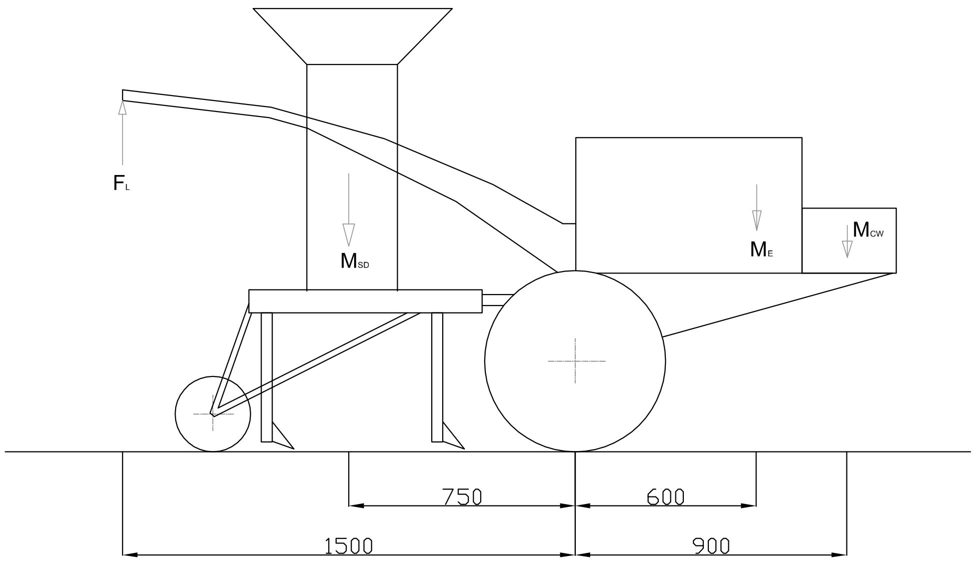

Figure 4.3: A diagram showing how the mass of the counterweight was determined.

4. Once the mass of the counter weight had been decided on, it was necessary to check that this still allowed for a reasonable weight to remain on the press wheels following behind the sowing tines. This is because some weight on the press wheels is required to ensure that the soil is pressed to ensure good seed-soil contact. It has been found that some extra weight is required to maintain good seed-soil contact. It is known that the weight of a person standing on the axle the press wheels are mounted on is too much and can cause excessive wheel slip to occur.

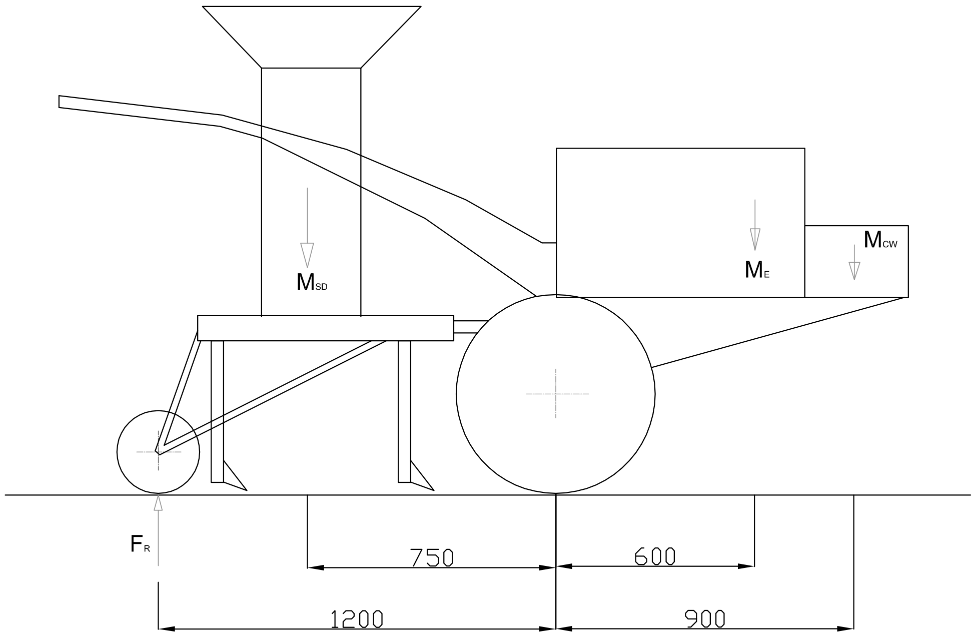

Figure 4.4: A diagram showing how the weight on the press wheels was calculated.

5. As the full weight of the power tiller and seed drill is known as well the weight on the press wheels, the mass over the drive axle can be calculated. This is the mass that will be used in any tractive force predictions done.

The center of mass of the engine was found to be 600mm from the center of the axle. When there was no counter weight, it required 45kg to lift the seed drill out of the ground. As this is quite considerable, it was decided a 25kg counter weight would be added as this decreased the lifting force to 30kg. With the 30kg, the weight over the rear press wheels is 37.5kg. This leaves 547.5kg over the drive wheels when the seed drill is fully loaded with seed and fertilizer.

4.1.1 Assumptions

With the process used in calculating the weight distribution of the power tiller, there are a number of key assumptions that have been made in order to simplify the problem. These are that apart from engine, the remainder of the weight of the power tiller is centered over the drive axle. This is a major assumption but was the best option as the mass and center of mass of every component was not known. Whereas the mass of the engine is specified in the operators manual as 150kg. It was not clearly stated if the weight listed in the owners manual included the weight of the rotovator or not. For this reason, a dry weight of 500 kg was used.

It is also assumed for any calculations involving the addition of wheel weights, the weight is added exactly over the center of the axle and none of the extra weight is distributed to another axle.

It has also been assumed that when the seed drill is operating with the tines in the soil, the vertical component of the total force experienced by the tine has been considered to be negligible. In reality, this force would exist but would be very small and may contribute less than 10% of the total force experienced by the tool.

4.2 Traction Force Predictions

Even though a value for tractive force is given in the power tiller operators manual, both the empirical and theoretical models will be analysed to calculate the maximum tractive force on a typical clay soil as well as a very basic prediction method that does not take into account soil conditions, tyre parameters or wheel slip. In the article by Narang and Varshney (2005), they tested the tractive ability of a power tiller on tilled land. Testing was done in India using a 8.95kW power tiller and when using pneumatic tyres similar to the one being investigated. They found that in a normal situation with pneumatic tyres, a tractive force of approximately 800N was achieved.

As optimum slip for agricultural purposes is approximately 15%, this will be used in all calculations as a constant value. As there is the need to take into consideration the addition of the mass of the counterweight and the seed drill, the total mass of the power tiller, seed drill, seed and fertiliser and counterweight is being taken as 585kg. Of this, 547.5kg is distributed over the drive axle and 37.5kg is distributed over the press wheels. There is the option of adding 2 20kg wheel weights to each of the drive rims. This increases the total weight to 625kg and is added directly to the drive axle. This makes the weight over the drive axle 587.5kg.

Even though there is no literature which mentions the relationship between the weight of a traction machine and the tractive force it is able to produce, there is one that was recommended. That is the tractive forces is approximately 40 percent of the weight of the traction machine. As a power tiller is only driven by 2 wheels, only the mass over the drive wheels will be used. Traction force predictions will be done with and without the additional wheel weights to investigate if it is worth adding them.

Symbol	Name of Parameter	Value for analysis
CI	Cone Index of soil	600 kPa
d	Tyre Diameter	0.6m
b	Tyre Width	0.15m
δ/h	tyre deflection	0.2
S	wheel slip	15%
L	length of the contact area	(Calculated)
k	soil constant	(Calculated)
CRR	Coefficient of Rolling Resistance	(Calculated)
CT	Coefficient of Traction	(Calculated)
CT_{max}	Maximum Coefficient of Traction	(Calculated)
H	Total horizontal force at the drawbar	(Calculated)
H_t	Total tractive force	(Calculated)
R	Rolling resistance force	(Calculated)
η	Tractive Efficiency	(Calculated)

Table 4.1: Values of tyre and soil conditions for analytical traction force prediction.

4.2.1 Analysis Parameters

Table 4.1 lists the different parameters that are required in order to be able to use the theoretical tractive force prediction model

Symbol	Name of Parameter	Value for analysis
d	tyre diameter	0.6m
b	width of the contact area	0.15m
N_{tyre}	Vertical Weight on Tyres	5.37kN
δ_t	Tyre Deflection	0.04
S	Wheel Slip	15%
n	Soil Constant	0.16
K	Soil Stiffness Value	0.06m
c	Soil Cohesion	30kPa
ϕ	Soil Internal Friction Angle	10 degrees
k_c	Soil Stiffness Coefficient	6 kPa/m^{n-1}
K_ϕ	Soil Stiffness Coefficient	400 kPa/m^n
L	length of the contact area	(Calculated)
P	Pressure on the tyre/surface contact area	(Calculated)
C_R	Coefficient of Rolling Resistance	(Calculated)
R	Rolling Resistance of Traction	(Calculated)
CT_{max}	Maximum Coefficient of Traction	(Calculated)
H	Total horizontal force at the drawbar	(Calculated)
H_m	Maximum Tractive Force	(Calculated)
H_t	Total tractive force	(Calculated)
R	Rolling resistance force	(Calculated)
η	Tractive Efficiency	(Calculated)

Table 4.2: Parameters of tyre and soil conditions for theoretical traction force prediction.

4.2.2 Empirical Traction Force Prediction

$$\begin{aligned} MN &= \frac{CI \times b \times d}{W} \times \sqrt{\frac{\delta}{h}} \times \frac{1}{1 + b/(2 \times d)} \\ &= \frac{600 \times 0.15 \times 0.6}{5.37} \times \sqrt{0.2} \times \frac{1}{1 + 0.15/(2 \times 0.6)} \\ &= 3.997 \end{aligned}$$

$$\begin{aligned} CRR &= 0.049 + \frac{0.287}{MN} \\ &= 0.049 + \frac{0.287}{3.997} \\ &= 0.121 \end{aligned}$$

$$\begin{aligned} CTmax &= 0.796 - \frac{0.92}{MN} \\ &= 0.796 - \frac{0.92}{3.997} \\ &= 0.566 \end{aligned}$$

$$\begin{aligned} k &= \frac{4.838 + 0.061 \times MN}{CTmax} \\ &= \frac{4.838 + 0.061 \times 3.997}{0.566} \\ &= 8.981 \end{aligned}$$

$$\begin{aligned} CT &= CTmax\left(1 - e^{-kS}\right) \\ &= 0.566\left(1 - e^{-8.981 \times 0.15}\right) \\ &= 0.419 \end{aligned}$$

By multiplying the Coefficients of Rolling Resistance (CRR) and Traction (CT) by the normal weight applied to the tyres, we can get values for the Rolling Resistance and Tractive forces respectively. To find the Total Tractive force of the machine, the Rolling Resistance is subtracted from the Tractive force.

$$\begin{aligned} Ht &= 2.249kN \\ RR &= 0.649kN \\ H &= Ht - RR \\ H &= 1.6kN \end{aligned}$$

$$\begin{aligned} \eta &= \frac{CT \times (1 - S)}{CT + CRR} \\ &= \frac{0.419 \times (1 - 0.15)}{0.419 + 0.121} \\ &= 66\% \end{aligned}$$

4.2.3 Tractive Force Results

As the calculation process for the Theoretical Tractive Force model is a lot more complicated, a spreadsheet was constructed in Microsoft Excel. The results of the spreadsheet are shown below as well as the results from the Empirical Gee-Clough Model. These results do not include the additional wheel weights.

	Theoretical Model	Empirical Model
RR	0.22kN	0.649kN
Ht	3.60kN	2.249kN
H	3.37kN	1.60kN
η	79.7%	65.97%

Table 4.3: Table of results for tractive force using both Empirical and Theoretical Models.

Using the recommended relationship of traction force is 40% of the weight of the traction machine, theoretically a traction force of 2.1kN is possible. However, this does not take into account soil conditions, tyre parameters or wheel slip .

4.2.4 Effect of Additional Wheel Weights

In an effort to increase the tractive force of a power tiller, there is the option of using additional 20kg wheel weights which bolt onto the wheel rim. This adds a total of 40kg to the power tiller directly over the wheel. With the addition weight, see the tables below for a list of results.

	Theoretical Model	Empirical Model
RR	0.24kN	0.727kN
Ht	3.67kN	2.371kN
H	3.43kN	1.645kN
η	79.4%	76.5%

Table 4.4: The table of results for tractive force predictions with additional wheel weights.

4.2.5 Discussion of Traction Force Predictions

When comparing the results of the various tractive force prediction models against the results as given in the article by Narang and Varshney (2005), there is a very wide range of values obtained for the traction force. With pneumatic tyres, Narang and Varshney (2005) achieve a maximum tractive force of 800N without wheel weights which increased to 900N with the 2 additional 20kg wheel weights. If the basic relationship of traction force is equal to 40% of the weight of the traction machine, a traction force of 2100N is possible. When compared to the results achieved in tables 4.3 and 4.4, there is a very large range of results for the traction force. Without the use of wheel weights, the traction force ranges from 800N to 3370N. This is 400% of what has been measured in practical experiments. With the use of wheel weights, the traction force ranges from 900N to 3430N.

There are a number of difficulties in predicting the tractive force for any traction machine. The biggest one being the soil. Because the soil is not an isotropic substance, it does not always fail at the same point and strength can change with moisture content as well as composition (sand, silt and clay content). Soils can also vary a lot spatially. Over a 100 hectare area, there can be an unlimited number of soil conditions. For example, on a low lying area there could be a very black silty soil then there could be a red clay on the top of a hill then over the back of the hill there could be a sandy bank. Because of the changes in composition, testing the strength of soil can produce very varied results. It is these test values that are used in traction force predictions.

The biggest flaw with using the model that suggests a direct relationship between the weight and the traction force it is able to produce is the lack of any terms that incorporate wheel slip and any soil conditions. Whilst this may seem good in a situation where you done know the soil conditions and terrain, the application of this model does not fit well against the data collected by Narang and Vareshney (2005). It predicts a force almost 3 times of that which was recorded in the article.

Whilst the Empirical Gee Clough Model still predicts a tractive force twice as large as the recorded results, it is the most believable as it is not an unreasonable prediction. The addition of wheel weights causes only a small increase (45N) in the net tractive force but increases the tractive efficiency (η) by over 10%. Whilst the tractive force is double of what Narang and Vareshney (2005) recorded, it is not completely unreasonable due to soil conditions etc.

The theoretical model should be able to produce the most accurate prediction of the traction force but the complicated way of calculating everything may also be its downfall. By trying to take into account so many different variables that can not always be estimated accurately, it could create an error that may grow exponentially throughout the large number of calculations. The theoretical model predicts a tractive force of over 4 times the size of the force recorded by Narang and Vareshney (2005). This is quite unbelieveable and should be considered invalid.

4.3 Bolting System

The seed drill attaches to the power tiller through a casing that bolts to the rear of the transmission housing. There are four 12mm studs that hold the casing onto the back of the transmission casing. It was bought up as an area of concern and it was wanted to know if there was any possibility of the bolts breaking off. If a maximum tractive force of 3500N is some how achieved, this force is then transferred through the 4 12mm diameter bolts that hold the casing onto the back of the transmission. If it assumed that each bolt takes an even share of the load this results in a force of 875N in each bolt. If the force is acting perfectly normal the end of the bolt, this creates a normal stress given by the following formula. Where F is the normal force and A is the cross sectional area

$$
\begin{aligned}
\sigma &= \frac{F}{A} \\
&= \frac{4 \times F}{\pi \times D^2} \\
&= \frac{4 \times 875}{\pi \times 12^2} \\
&= \frac{3500}{\pi \times 144} \\
&= 7.7367 \frac{N}{mm^2} \\
&= 7.7367 MPa
\end{aligned}
$$

According the Australian Standard AS 4291.12000, the lowest Property Class Rating (3.6) specifies a Minimum Tensile Stress of 330 MPa but can significantly increase with the Property Class Rating. The stress occurring in each of the bolts of the back of the power tiller is less than 2.5% of the minimum stress any bolt must be able to withstand according to Australian Standards. According to this analysis, the bolts should not break through tensile stresses created by the attachment of the seed drill.

4.4 Tine Draft Force Predictions

4.4.1 Tine Dimensions

From the following illustration, tine dimensions are easily defined. The original job card for the manufacturing of the tines specified Bisalloy 360 grade flat material with a thickness of 12mm. Figure 4.5 is a cross section of the tine illustrating the rake angle of the original tine.

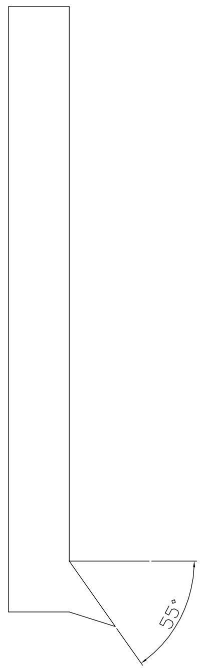

Figure 4.5: A diagram illustrating the rake angle of the current tine design.

4.4.2 Tine Draft Force Prediction Results

Using the single tine force model based on the research done by Professor Dick Godwin, data was collected by using constant soil conditions then varying the rake angle of the tool over a range of depths. The depth range over which the tine was tested ranged between 0.01m and 0.1m in increments of 0.01m. By using this range of depth values, it can be seen if there is any dramatic change in the draft force if the critical depth is reached. Velocity is taken as 1 m/s as this represents the upper range of the speeds that could be used for planting due to the gearing of the transmission on the power tiller.

The depth range was also chosen because of research done into the depth at which different crops are sown at. Values for sowing depth were found at the DPI website as well as in the article written by M. Enamul Haque. Depths for sowing Wheat seed range between 50 and 70mm from the surface and in the tests on no tillage farming, M. Enamul Haque sowed the seed between 30 and 40mm from the surface. According the DPI, Maize seed is sown between 50 and 70mm as well but M. Enamul Haque sowed the Maize seed between 40 and 50mm from the surface.

As illustrated in figure 4.6, the relationship between the draft force experienced by a tine and the depth at which it is operating is almost linear in this case. Table 4.5 shows a list of values that were used to construct figure 4.6 that were calculated using the spreadsheet made by McKeys et al.

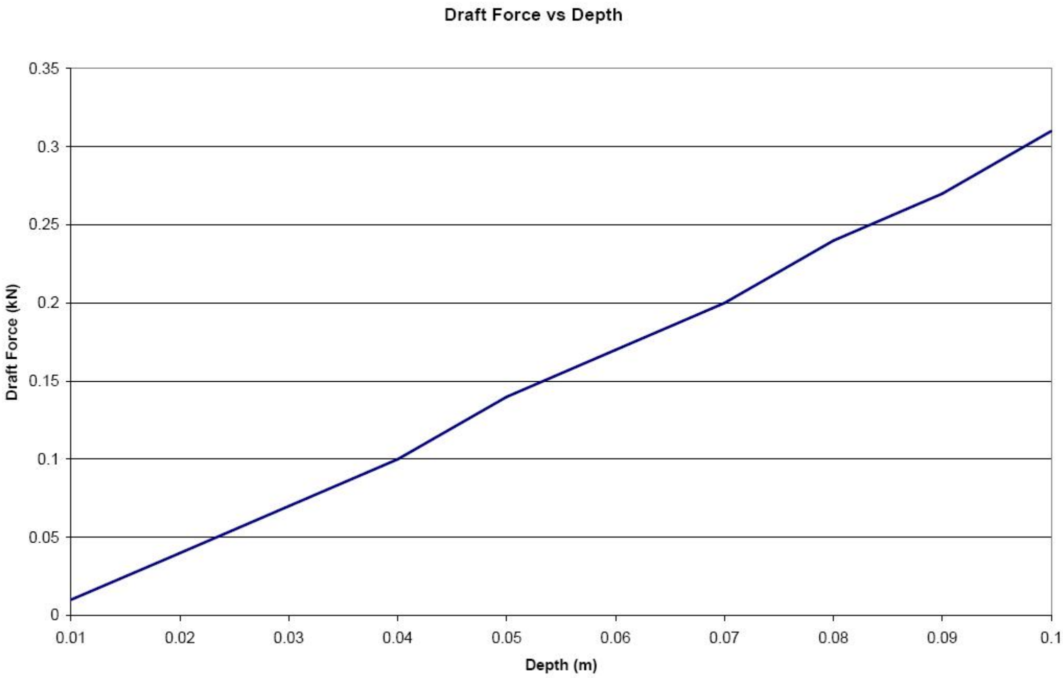

Figure 4.6: A graph showing the relationship between draft force and depth for the given tine design and soil parameters.

Tool depth (m)	Draft Force (kN)
0.01	0.01
0.02	0.04
0.03	0.07
0.04	0.10
0.05	0.14
0.06	0.17
0.07	0.20
0.08	0.24
0.09	0.27
0.10	0.31

Table 4.5: Values of draft at varying depths for the original tine design.

4.4.3 Discussion of Tine Prediction Results

Tine draft force predictions were straight forward as there is only one model that has been deemed valid. From the graph in figure 4.6, for the given tine and soil characteristics there is almost a linear relationship between operating depth and draft force. If the implement was going to be used at a depth greater than 0.01m then to calculate the draft force it would only involve a basic linear interpolation to get a fair prediction. Accord to Godwin et al, due to the anisotropic nature of soil, any prediction relating to the draft force experienced by a tine will only be accurate to the range of $\pm$ 10%. This means the maximum according to this prediction is 0.341kN. The results calculated from this prediction does not seem unreasonable and can be considered to be adequate.

Chapter 5

Design Evaluation

5.1 Finite Element Method Theory

The Finite Element Method (FEM) is a commonly used mathematical tool for predicting the behaviour of a physical system. It is a mathematical method which solves extremely large numbers of equations which are used to form a model of a system. The real system being analsyed is represented in the form of a mesh or grid of elements.

As the name suggests, in any system there is a finite number of elements. When each element is properly defined, the unknown variables can be found via algebraic equations that are used to describe the system. A complete solution is then pieced together using the values calculated from each element.

As unknown parameters are calculated for each element, they are used in the calculations for the next element. Depending on the size of the system be analysed, this can result in an extremely large number of equations to be solved. Whilst it is possible to complete an analysis of a system by hand, there can be a large number of calculations that must be solved. For this reason, it is more common to make use of a software package to solve the system.

5.2 Finite Element Analysis Software

The chosen software to be used for the Finite Element Analysis component of this project is COSMOSWorks. This is a program which is coupled with the 3D modelling package SolidWorks. It is sold by Solidworks Corporation and is commonly coupled with the 3D modelling software. It is extensively used in industry because of its ability to reduce prototyping costs and reduce the time to develop a product.

It is able to analyze individual components as well as component assemblies. It is able to analyse a large number of common situations that can be faced every day in a real life design analysis. These can include standard stress analysis, fluid mechanics and electromagnetic systems.

Finite Element Analysis is performed by specifying the component material properties then applying loads and restraints. Before the chosen analysis can be performed, either a shell or solid mesh of the component is created by COSMOSWorks. COSMOSWorks has an automatic meshing tool which generates a standard mesh but element size can be modified to create either a finer or coarser mesh depending on what the user requires. A finer mesh creates a larger number of elements which will take longer to compute.

Solidworks and COSMOSWorks both have a library of various common materials. Properties such as density, Modulus of Elasticity, Tensile Strength and Yield Strength are just some of those given and are used in the solving the many simultaneous equations required to perform the Finite Element Analysis. There is the option to create a new material and specify the various properties. COSMOSWorks also has the ability to simulate plastic materials.

Once material properties, loads and restraints have been defined and an analysis has been performed, COSMOSWorks can calculate if failure will occur. The most commonly used failure criterion is Von Mises Stress Failure Criterion. It states that failure will occur if the stress occurring in the component exceeds the Yield Strength of the material. This can be visually represented using a plot created in the reporting process and the scale of the plot can be magnified to show weak areas in the design.

Once an analysis has been conducted, COSMOSWorks can prepare a report compiled of various visual representations. It can also produce animations of the displacement which will occur if required. The user can select what types of plots to create and it is possible to change the scale and the orientation of the model. Plots can be exported individually or a compiled report can be exported with plots stored as simple jpeg images.

5.3 Analysis of Current Frame Design using Finite Element Analysis

In order to be able to analyse the design of the current frame, there were a number of steps that had to be followed. Three dimensional models of each component had to be created in Solidworks. This was done by copying the side profile of each component from a supplied AutoCAD drawing into Solidworks then extruding it to create a three dimensional solid model. Components were then assembled together to create assemblies of various sections of the seed drill. Each assembly was then imported into COSMOSWorks for analysis.

Chris Holland is the designer and current manufacturer and of the current seed drill design and drew the 2 dimensional drawings of each component in AutoCAD. Chris is the owner of Spring Ridge Engineering, who also make Rogro Machinery, specifically no till planters for broadacre applications.

5.3.1 Construction of 3D Assembly Models

Once each part had been modeled in Solidworks and using these parts assemblies were made by creating relationships (mates) between the components and relative geometries. This process involved the addition of nuts and bolts that were sourced from the Solidworks Toolbox. The Solidworks Toolbox is a library of common parts such as various different types of fasteners (nuts, bolts, screws etc). These assemblies were made to simulate the actual setup of the seed drill in real life. Because of the design of the machine, there are an infinite number of combinations of ways that the machine can be arranged. As it is not possible to test every combination, some research was done on row spacings that different crops were sown at.

Wheat and Maize are the crops that this seed drill has been primarily designed to sow. According to the DPI website, Wheat should be sown at row spacings ranging between 15 and 30cm. Where as it is recommend that Maize be sown at row spacings of 75 to 91cm but should be matched with the harvester width. In the paper written by M. Enamul Haque, they did testing of no tillage farming systems in Bangladesh. Testing was done on Wheat and Maize and used row spacings of 20cm and 70cm respectively. With this information on row spacings, different tine configurations were determined. If a maximum of 4 tines were able to be used, when sowing wheat, 4 tines were used with a spacing of 19.6cm between the center of each tine. However when sowing Maize because of the wide row spacings, only 2 tins could be used as it has a row spacing of 70cm.

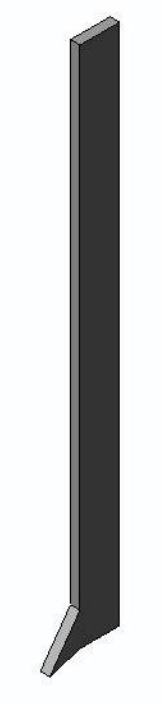

Figure 5.1: A 3D model of the current tine design.

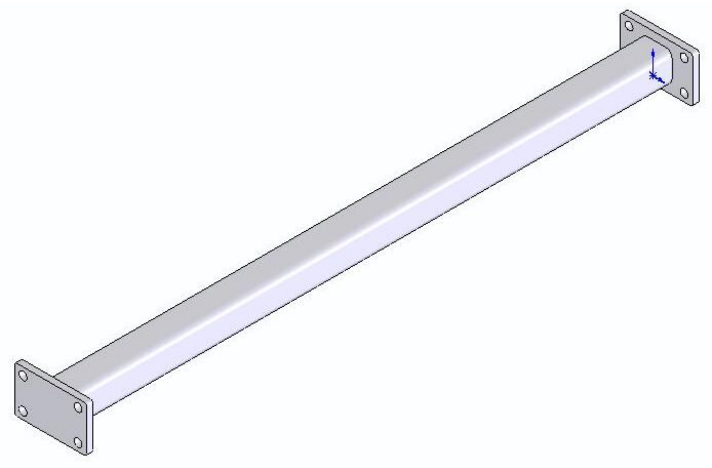

Figure 5.2: A 3D model of a cross beam.

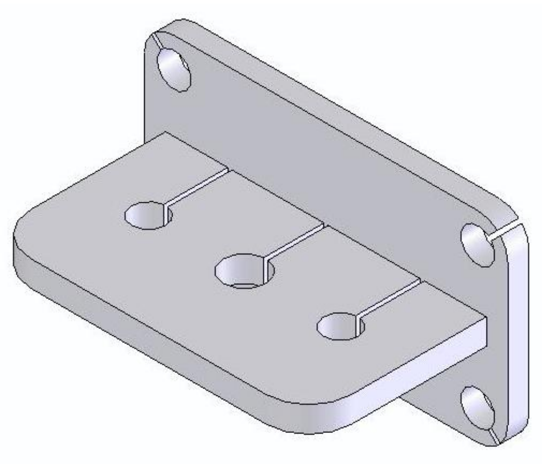

Figure 5.3: A 3D model of the mounting bracket used for connecting the seed drill to the power tiller.

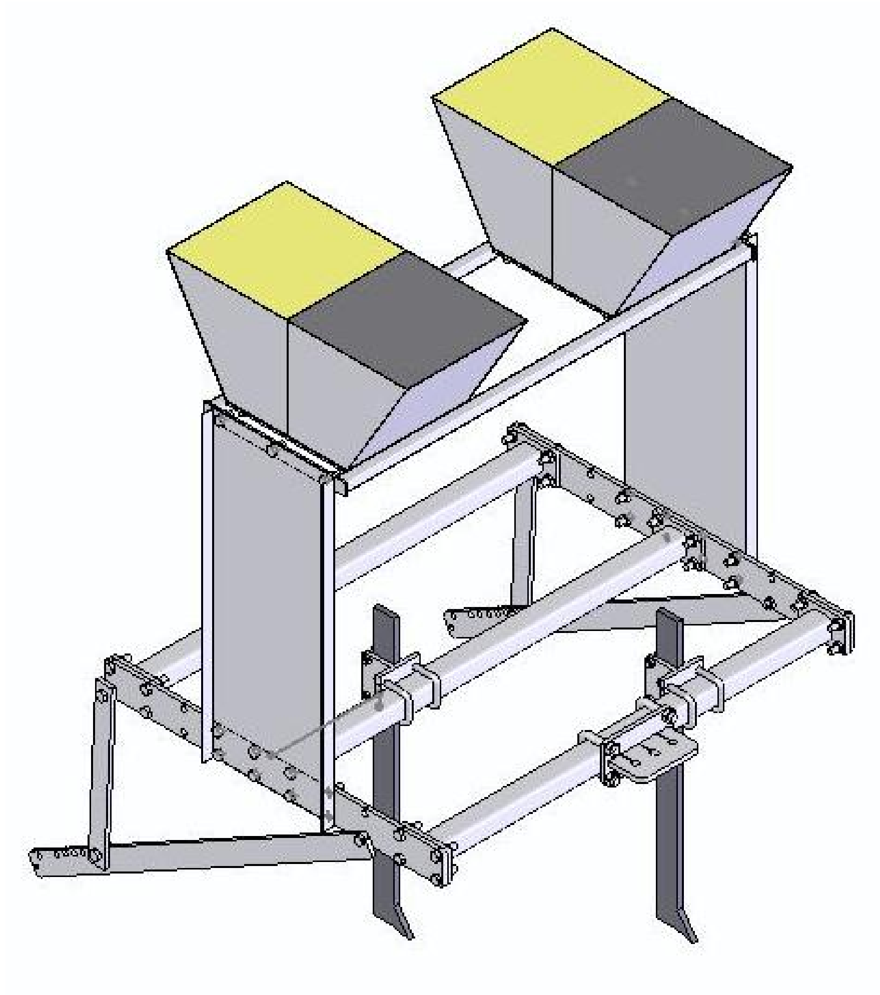

Figure 5.4: A 3D assembly of the full seed drill except for press wheels.

5.3.2 Analysis Process

Due to limitations on COSMOSWorks, complete models of the whole seed drill (Figure 5.4) caused the software to crash. Because of this limitation, it was decided that the next best alternative would be attaching tines to the cross beams and testing them individually with a constant load. As the draft force of the tine had been estimated, to give a safety of factor in the design the assembly was subjected to a 1kN static load. This was done to simplify analysis as the tine only moves at a speed of 1 meter per second.

The positioning of the tines on the cross beam (see Figure 5.2 for a 3d model of a cross beam) was calculated by investigating various tine arrangements on the seed drill. The positions for the tines was chosen to allow trash or stubble to pass easily between the tines and to minimize any residual stresses that might be created by the tine as it passes through the soil.

The testing method involved testing a number of different arrangements. These involve a single tine attached to the cross beam then two tines attached to the cross beam but have 2 models with two tines with varying spacings. This was done to see if the extra tine would create enough extra stress to cause failure.

To simulate having the beam being directly attached to the power tiller, the mounting bracket was attached to some of the assemblies to investigate if it would have any effect on residual stresses and stress concentrations in the crossbeam.

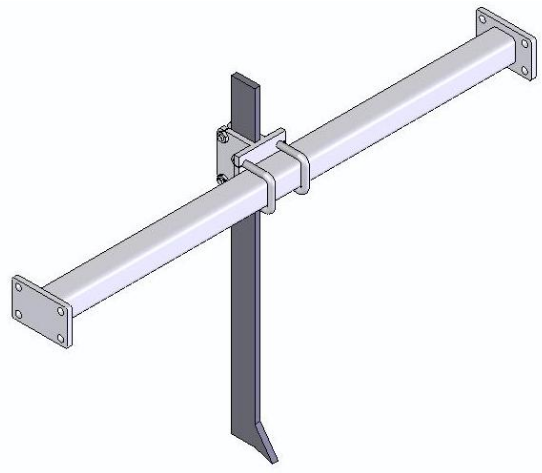

Figure 5.5: A 3d model of a crossbeam with 1 tine attached in the middle.

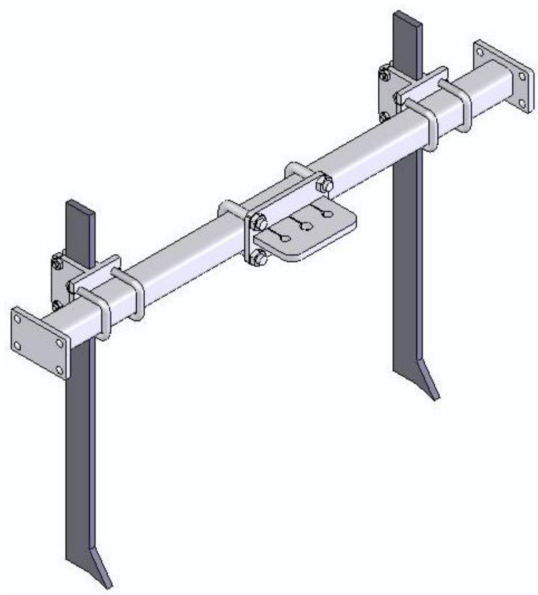

Figure 5.6: A 3d model of a crossbeam with 2 tines and the mounting bracket attached with the correct spacing for sowing Maize.

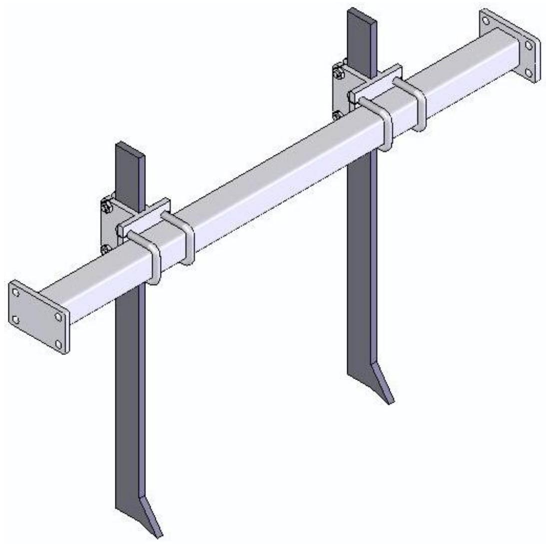

Figure 5.7: A 3d model of a crossbeam with 2 tines attached with the correct spacing for sowing Wheat.

Once the assembly was loaded into COSMOSWorks and the loads, restraints and materials were defined. All other parts were defined as plane carbon steel as the material used by the manufacturer is not galvanized. Each tine is subjected to a 1kN force acting perpendicular to the point of the tine. If two tines are attached then the total force is 2kN. The failure criterion being used for the analysis is Von Mises Stress Criterion. This is defined as the stress at which the material will begin to yield. See tables 5.1 and 5.2 for values for the properties of the two different types of steel used in analysis of the seed drill. As the tines are manufactured using Bisalloy 360 grade high strength steel, the closest match to this was AISI 1020 steel as specified in Solidworks.

Property Name	Value
Elastic Modulus	2.1×10^{11} N/m^2
Poisson's Ratio	0.28
Shear Modulus	7.9×10^{10} N/m^2
Mass Density	7800 kg/m^3
Tensile Strength	3.9983×10^8 N/m^2
Yield Strength	2.2059×10^8 N/m^2
Thermal Expansion Coefficient	1.3×10^{-5} /K
Thermal Conductivity	43 W/m.K
Specific Heat	440 J/kg.K

Table 5.1: Properties of plain carbon steel used for FEA.

Property Name	Value
Elastic Modulus	2.0×10^{11} N/m^2
Poisson's Ratio	0.29
Shear Modulus	7.7×10^{10} N/m^2
Mass Density	7900 kg/m^3
Tensile Strength	4.2051×10^{8} N/m^2
Yield Strength	3.5157×10^{8} N/m^2
Thermal Expansion Coefficient	1.5×10^{-5} /K
Thermal Conductivity	47 W/m.K
Specific Heat	420 J/kg.K

Table 5.2: Properties of the high strength steel used for FEA.

5.3.3 Software Setup

Steps of COSMOSWorks being setup can be seen in Figures 5.8 to 5.11. This only shows major steps of choosing the type of study, applying restraints and loads and creating the solid mesh. Figure 5.11 shows the model with the mesh that is created by COSMOSWorks as well as the loads and restraints. The green arrows represent the three dimensional restraints and the magenta arrows represent the force loads being applied to the system. The same process is used for the analysis of all the models. With the exception being those with the mounting bracket, which includes extra restraints on the three vertical holes.

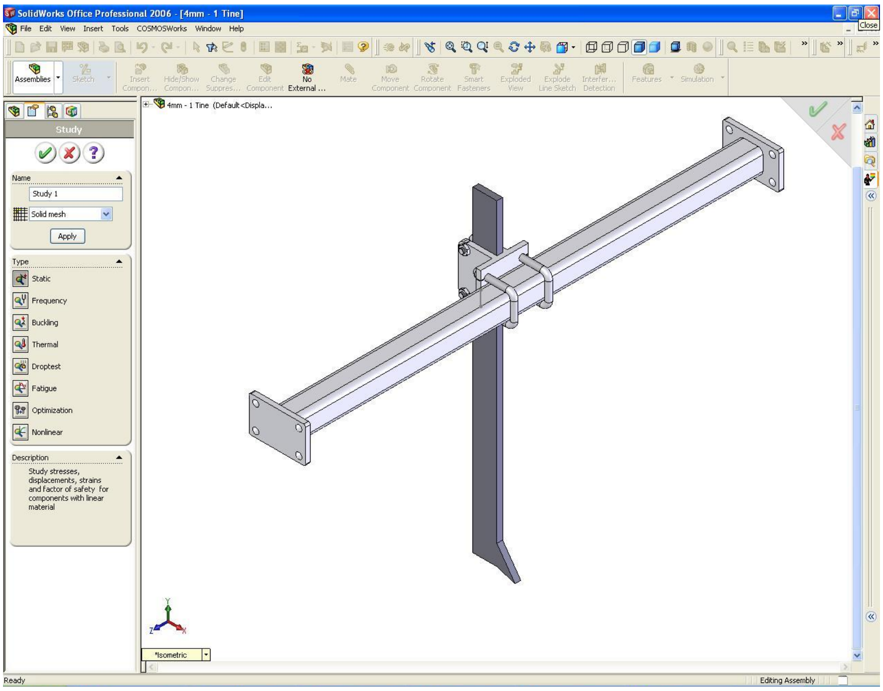

Figure 5.8: A screen shot showing the step of choosing the type of study in COSMOSWorks.

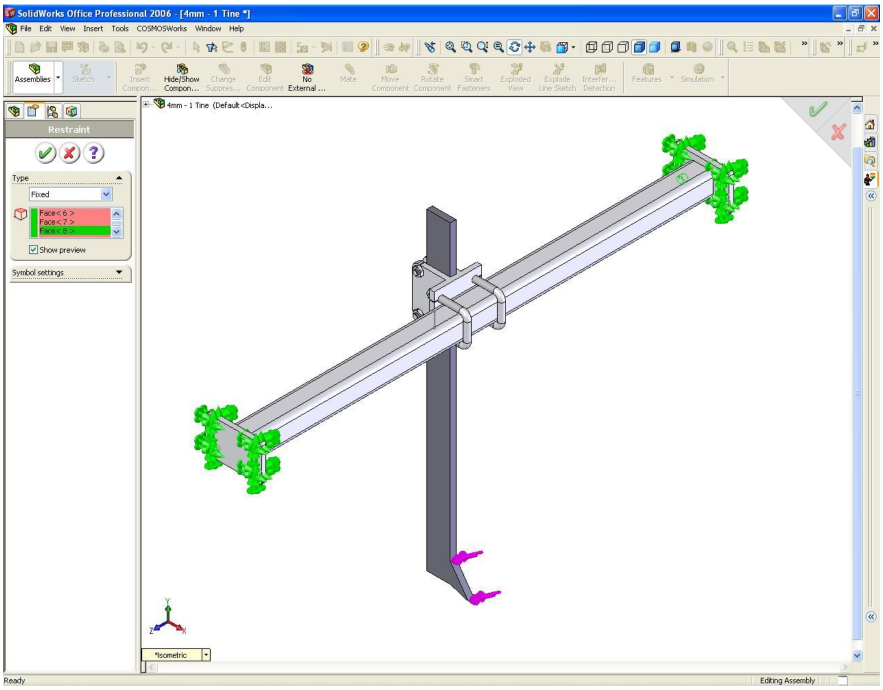

Figure 5.9: A screen shot showing the process of applying the restraints to the system.

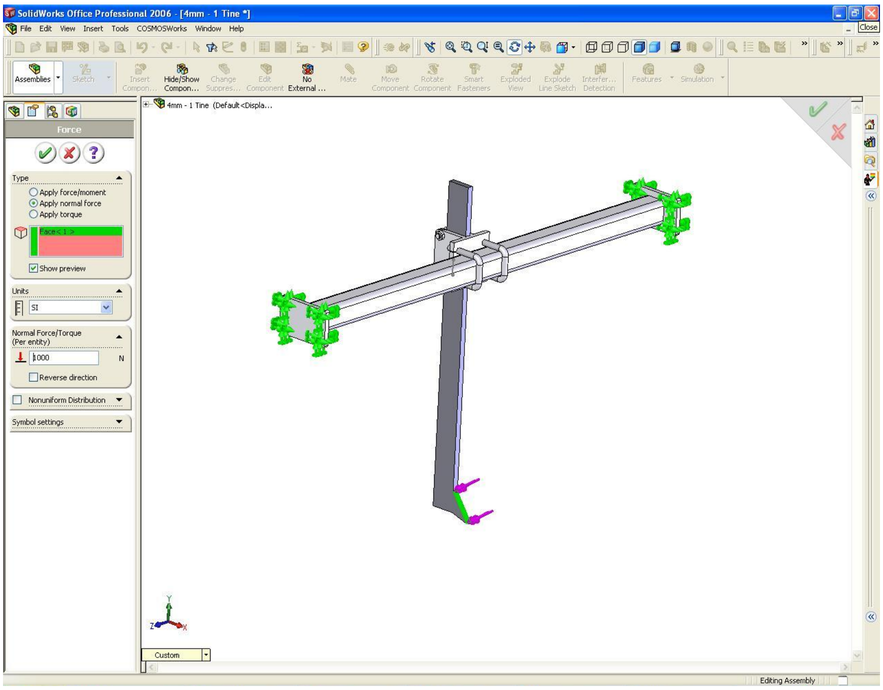

Figure 5.10: A screen shot showing the process of applying the load to the system.

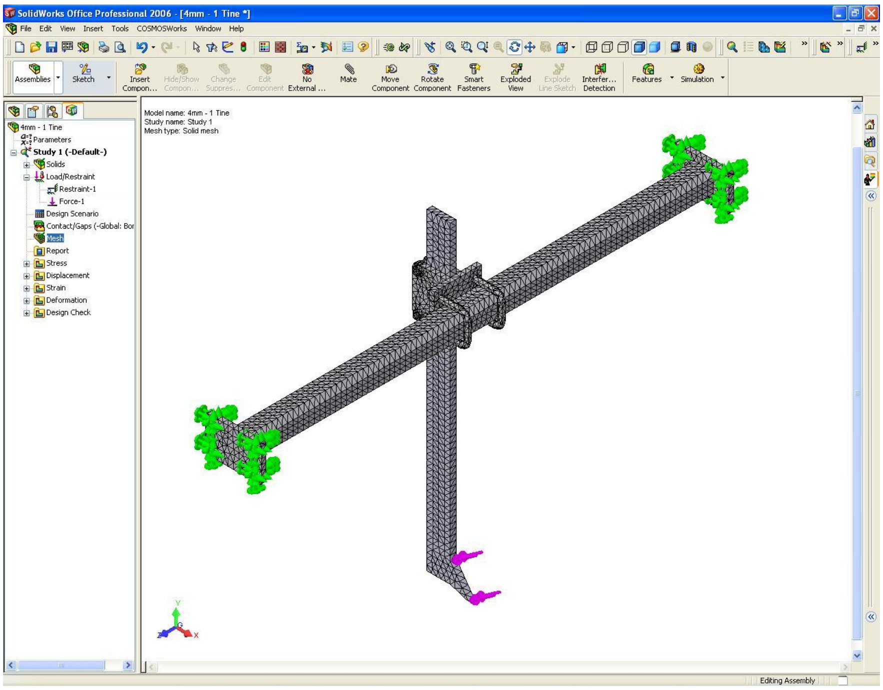

Figure 5.11: The mesh created for the analysis of a single tine crossbeam arrangement.

5.3.4 Finite Element Analysis Results

After running each of the 5 different arrangements in COSMOSWorks, a generic report was created listing all the parts in the assembly, numerical results as well as pictures illustrating the different changes in stress, strain and displacement. Another illustration created by the COSMOSWorks reporting utility is the plot of design check. This is a plot comparing calculated stress concentrations to the yield strength of the material to determine if the design will fail. This failure criterion is known as Von Mises Stress Failure Criterion. The different colour regions in the design check illustration represent different Factors of Safety against failure according to Von Mises stress failure criterion. It is important when analyzing any of the graphs produced by COSMOSWorks to look at the scale used.

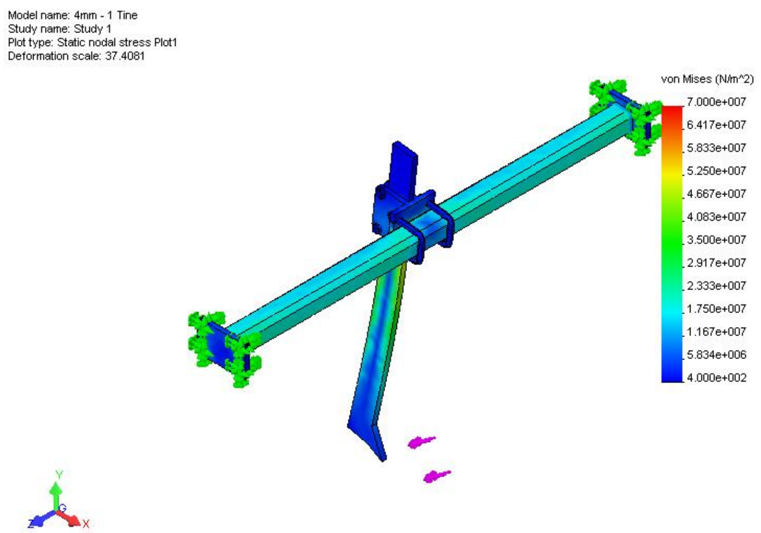

Figure 5.12: A plot of stress concentrations for a crossbeam with a single tine.

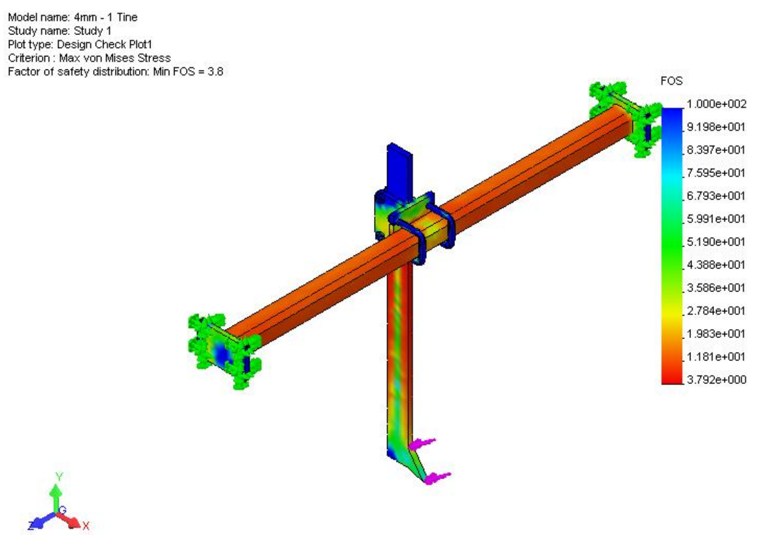

Figure 5.13: The Design Check for a crossbeam with a single tine.

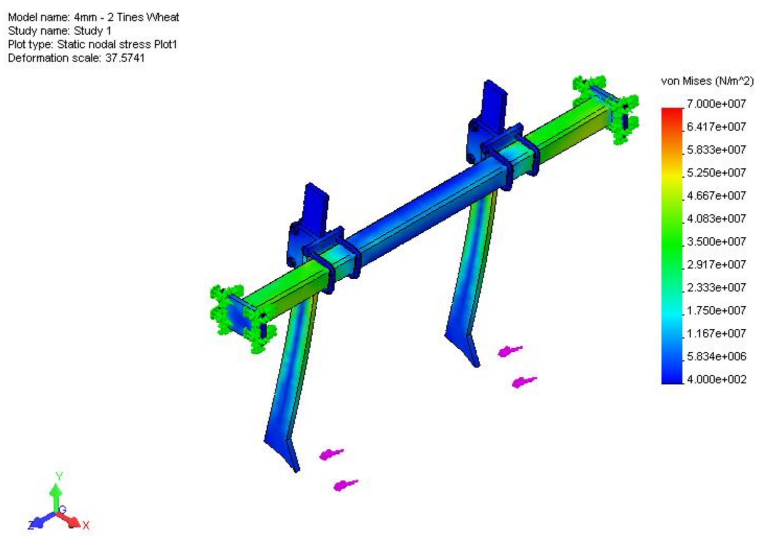

Figure 5.14: A plot of stress concentrations for a crossbeam with two tines set up for Wheat.

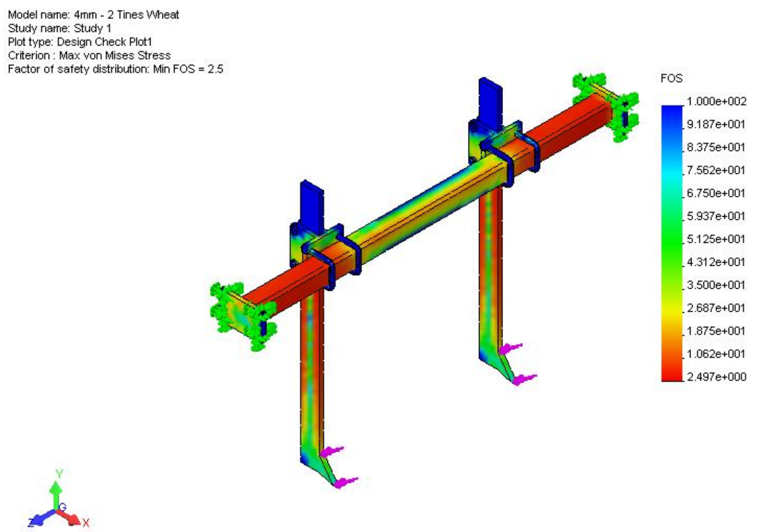

Figure 5.15: The Design Check for a crossbeam with two tines set up for Wheat.

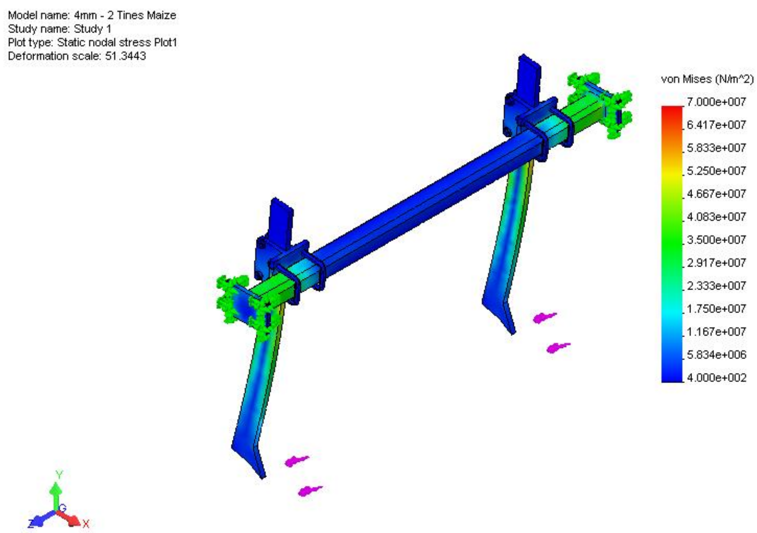

Figure 5.16: A plot of stress concentrations for a crossbeam with two tines set up for Maize.

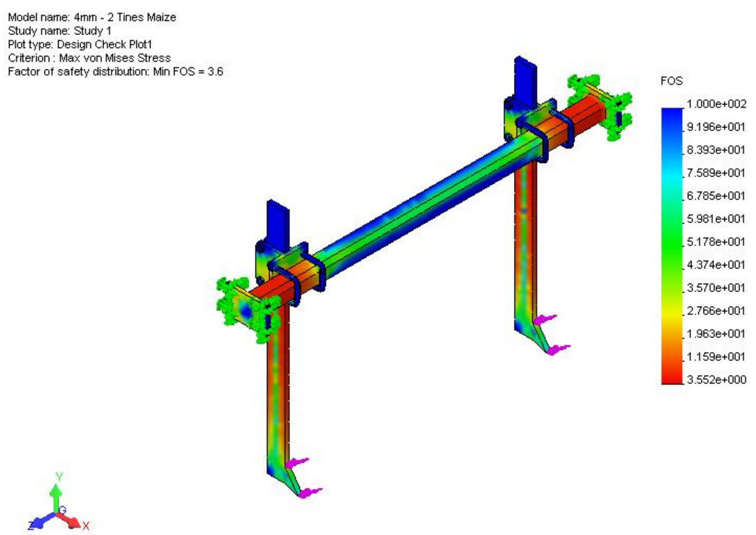

Figure 5.17: The Design Check for a crossbeam with two tines set up for Maize.

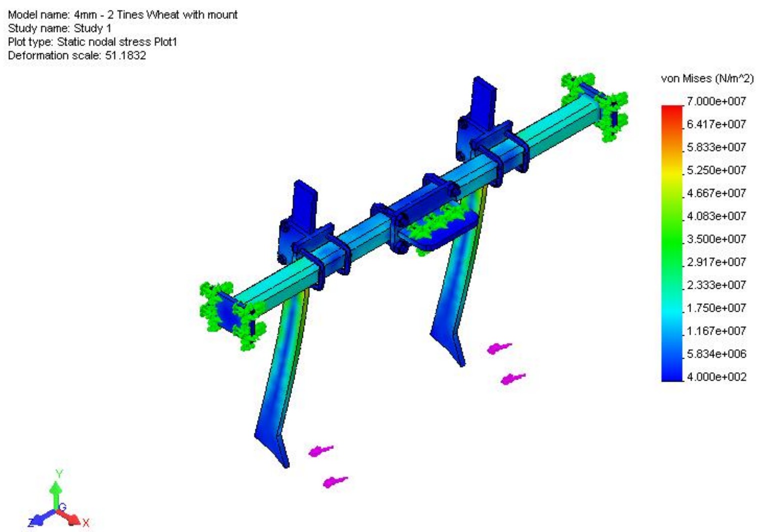

Figure 5.18: A plot of stress concentrations for a crossbeam with two tines set up for Wheat and the mounting bracket.

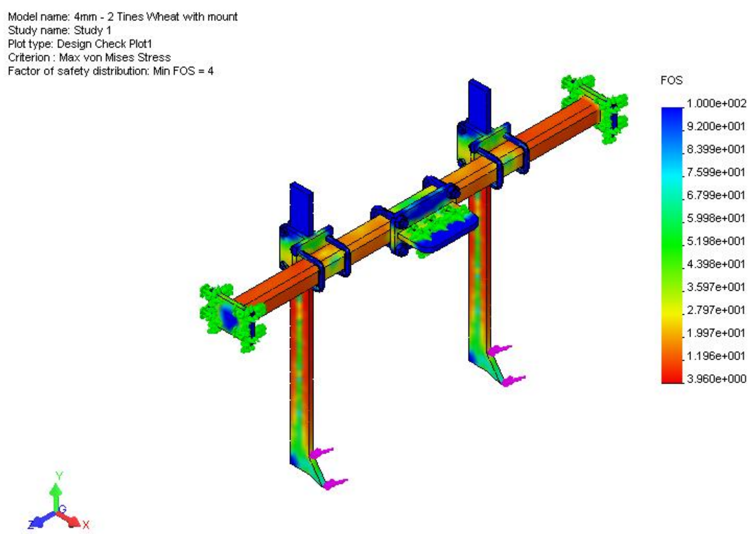

Figure 5.19: The Design Check for a crossbeam with two tines set up for Wheat with the mounting bracket.

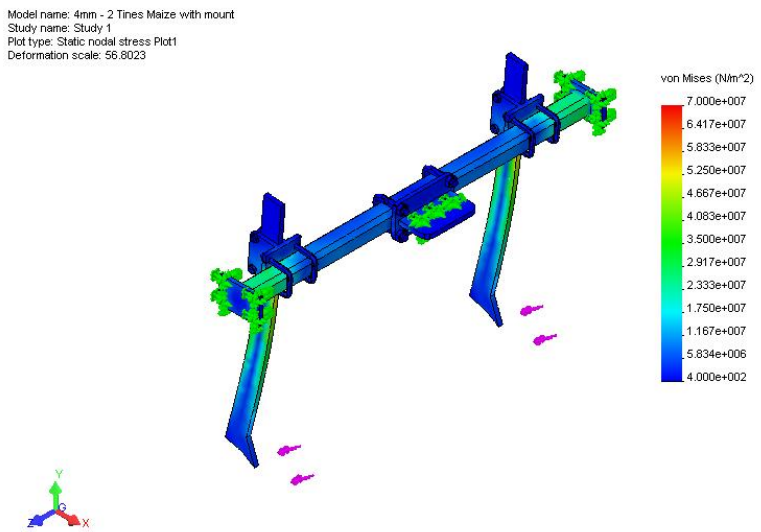

Figure 5.20: A plot of stress concentrations for a crossbeam with two tines set up for Maize and the mounting bracket.

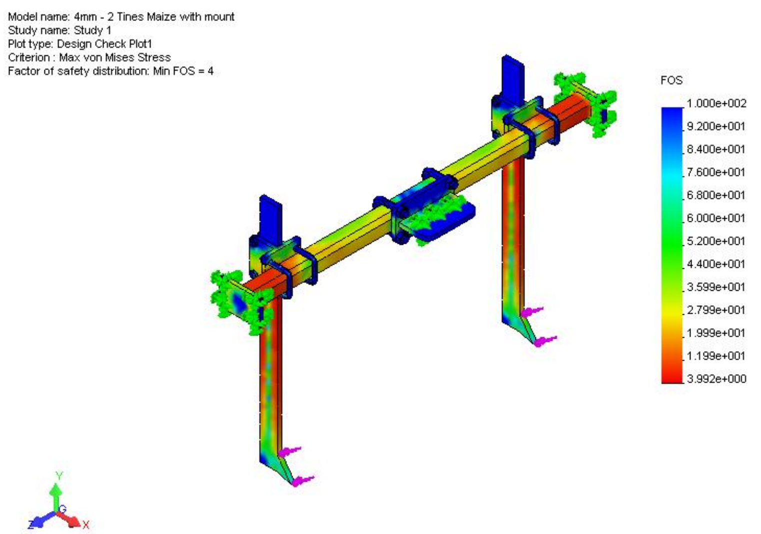

Figure 5.21: The Design Check for a crossbeam with two tines set up for Maize and the mounting bracket.

5.3.5 Discussion of FEA Results

From the above pictures produced by COSMOSWorks after restraining the bolt holes at the end of each crossbeam and where applicable restraining the bolt holes on the mounting bracket, it is clear that the current design of the crossbeam and tine is very satisfactory. The minimum Factor of Safety against failure according to Von Mises Stress Failure Criterion is 2.5. What must be taken into consideration is the fact that a force of approximately 3 times the predicted is being applied to each of the tines. This translates to a total force of up to 6 times the predicted force that the assembly will encounter. Ultimately, a Factor of Safety of 2.5 is more than ample to ensure failure will not occur in any of the sections.

However there is something that can not be modelled or taken into account when doing the Finite Element Analysis. These are the welds that bind some components together. When individual components were modelled in Solidworks, where a new piece of material was to be welded to an existing piece, a new sketch was created on the existing face and the section was extruded and was forced to merge with the existing solid body. This is a flaw in the modelling process but the welding process should not weaken the material that much to cause failure in the current design.

Considering that the current design is currently in use, this analysis confirms that the design of the crossbeams, tines and all mounting brackets have been built using material that will be strong enough to withstand any forces that it will encounter if used with a 8.95kW (12hp) power tiller.

Chapter 6

Design Modification and Evaluation

6.1 Design Modification Considerations

As with the modification of the design of any product, there are a number of different criteria that must be taken into account. Each different criteria must be prioritized and rated before a final decision can be made regarding any change to the current design. As with the modification of any product there is an required design life, cost, the user requirements and the ability to be able to retrofit the modification to older equipment. These different factors will be investigated and will be used to determine what changes may be able to made to the current design of the seed drill.

6.1.1 Required Design Life

When a customer purchases a product of any kind, they are expecting it to last a certain time. With agricultural machinery, as the initial outlay cost for the equipment is normally very large, it is expected to last a very long time if properly maintained. If the piece of equipment breaks 2 years after the date of purchase, this could be seen as very poor foresight on the designers behalf.

There a number of factors that can either increase or decrease the anticipated design life of a product. These are the working conditions it is subjected to and the degree of care that is given to it (level of maintenance). The owner of an implement can not expect it to last anywhere near as long it should if he or she is using it for a purpose other than what it was originally designed for.

If a piece of equipment is cared for and maintained regularly, the life of the equipment can be increased significantly. This is because if there is a small problem evident i.e. a small stress crack appearing, it is dealt with immediately and not left to grow and cause a much larger problem. The small defect may ultimately lead to the complete failure of a section and may result in the piece of equipment being rendered useless. Any modifications made to the frame of the seed drill must not decrease the design life of the drill. The testing that must be done on any modifications must be more rugged than what it may actually encounter when in use.If the testing of any new component does not meet the required standard, then it is best not to make any changes.

6.1.2 User Requirement

The user requirement can also be described as the purpose of the machine. The purpose of the machine must be known in order for a suitable testing procedure to be designed and implemented on the proposed changes to the machine. Depending on the nature of the product, it may be necessary to consider some other uses of the product other than the main purpose. From here, different types of testing can be devised and implemented on any of the proposed design modifications.

6.1.3 Cost

When designing any product to be sold to the general public, it is important to make the product available to all by keeping the cost to a reasonable limit. If the price is too high on a product, the majority of the population who may want to use it will not be able to as they can not justify purchasing it. By reducing the cost of the product, it will be able to gain a much larger portion of the market share because a greater percentage of the target users will be able to afford it.

When modifying the design of a product, by reducing the thickness of a section or the size of a component can reduce the cost be reducing the amount of material required to manufacture a component. Also, simplifying a design to make manufacturing process more efficient can reduce the cost that a manufacturer must charge in order to maintain a profit on the product.

6.1.4 Ability to be Retrofitted

If a modification is made to a component of a previous model of a product then it is important that the new component can be still be used even with older products. The ability to be able to retrofit new components into an older product makes it much more attractive when prospective buyers are looking at investing in new equipment. This is because if there is a problem with the equipment and a particular component has been redesigned, the ability to be able to fit the new part to the older models of the product make it much more appealing. It also makes obtaining replacement parts much easier.

6.2 Implications of Design Considerations on the Current Design

6.2.1 Required Design Life

As the seed drill is being made for use in third world countries, the ability to make repairs is limited. Therefore it is much more important that the equipment is going to have a very good design life even if it is subjected to very hard working conditions and a lack of maintenance. In order to know just how long the seed drill will need to last in order to be able to pay itself off, the cost of the drill must be compared to the average profit made by using the seed drill to plant crops.

There are a number of a factors that will influence the time required to make enough profit to pay off the initial investment of a the seed drill. Because of the fluctuations that occur in the Wheat and Maize markets, it is hard to predict the price per unit weight long periods of time in advance. Crop yield is very hard to predict as there are so much varying input factors, of which some can be controlled by the farmer and some that can't be controlled i.e. the weather.

Another variable that will change seasonally is the area being sown with the seed drill and the yield that is achieved over the area. It is common practice in countries such as Bangladesh for one man to own a power tiller and to do "contract cropping" for many of the other land owners in the close district. In a "contract cropping" scenario, the owner of the power tiller and seed drill is paid for the use of the equipment either by time that the equipment is used for if an operator is not required. If an operator is required then it can be charged per unit area and the wage of the operator is incorporated into the fee for the service. In third world countries where money can be hard to come by, payment can be made in the form of produce i.e. Wheat or Maize seed or in the form of a return service such as assistance in harvesting the crop at the end of the growing season.

Because of the numerous unpredictable factors that are involved in growing crops, it makes the task of predicting the required design life of a seed drill extremely challenging. This is because it is so hard to predict:

- The area that the seed drill will be used to sow each year, including the amount of contract work that will be done;
- the seasonal weather factors such as rainfall and temperature during crucial stages of plant growth;
- the average yield of the crop being grown; and
- the average price per unit weight of the produce being payed to the growers.

Considering money is not always easy to come by, to be sure that the seed drill will be attractive to those who own power tillers, the required design life of the seed drill was decided to be between 5 and 10 years. To guarantee this design life, one must take into consideration the normal "wear and tear" on components and the possible lack of regular maintenance and ability to do regular repairs. Taking all of this into account, the seed drill must be made to withstand a very harsh testing procedure as well as unforeseeable circumstances. The original frame was exerted to a total force of up to 6 times greater than the predicted maximum in the the form of Finite Element Analysis. For the given reasons, any modifications will be put through the same testing procedure.

6.2.2 User Requirement

The seed drill has been specifically designed for use behind primarily 12 horsepower power tillers but there are 15 horsepower models on the market but are not as common. The seed drill must be able to withstand the forces that a power tiller will be able to exert. From the previous chapter, various models were used to predict the tractive force that a traction machine is able to produce. Results from these predictions varied a lot and even more so with the collected data as tabulated in the article by Narang and Vareshney (2005). The collected data gave a traction force ranges between 800 and 900N whilst the smallest prediction gave a force of 1600N.

Due to the difficulty that may occur in developing countries if complicated components need to be repaired, simplicity is a key to the marketability of the seed drill. Use of standard length and diameter bolts is also a very good idea as if a nut and/or bolt are lost, it is not a big issue to find a replacement that will fit. If non-standard or high tensile bolts were required, these may be very hard to come by or extremely expensive in a developing country.

6.2.3 Cost

The target users of the seed drill are the owners of power tillers. Due to the Occupational Health and Safety (OH&S) issues surrounding power tillers, they are not used in developed countries. Rather, they are extremely common in third world or developing countries such as Bangladesh, Cambodia, Laos, India, Nepal and Malaysia. This is because they are cheap to manufacture and they use very primitive technology which makes them easy to repair. As money in third world countries is not always easy to come by, cost needs to kept to a bare minimum. This means where possible, avoid the use of non-standard and/or high tensile nuts and bolts, as well as flat plate or hollow tubing. This is advantageous in the situation where a bolt is lost or if a crossbeam is bent or broken and needs to be replaced.

6.2.4 Ability to be Retrofitted

If a new model of the seed drill is released and parts for the old drill are no longer being manufactured, the ability to be able to buy a replacement part and bolt it straight onto an old model makes it much more marketable. For example, if the drill was widened and longer crossbeams were used, the ability to bolt them onto the existing side bars with the existing nuts and bolts would make it a much more attractive implement. Where possible, by maintaining small details such as bolt diameters and bolt patterns, modifications can be compatible with any model of seed drill that was produced.

6.3 Design Modification Options

6.3.1 Tine Design

As the tractive force that a power tiller is able to produce is restricted by the machine itself, in order to be able to use more tines as one time the only option is to try to reduce the draft force on the tines. With the tine itself, there are some restrictions on what can and can not be changed.

As the main purpose of the tine is to cut a small slot into the soil for the seed and/or fertilizer to be placed in, there must be plenty of width to allow the seed to fall down the seed boot and land in the slot in the soil. To prevent the seed boot from being crushed by any sideways movement that occur whilst using the seed drill, the boot must be made of material which is strong enough to withstand these forces. Because of this, thin material is not a good choice and if used could bend and cause a blockage. If the boot is not to create a force resisting the direction of motion then the width of the boot is restricted to the width of the tine. As Maize seed is not the smallest seed and must be able to fall freely down the boot into the slot in the soil, the width of the tine must remain constant.

The dimension on the tine which is easiest to modify to be able to reduce the draft force is the rake or attack angle. This is illustrated in Figure 6.1 where the rake angle is shown by the angle α. If starting with a high rake angle, the major component is the horizontal force. By reducing the rake angle, it decreases the net force experienced by the tine to a certain point. It then begins to increase again because the more dominant component of the force becomes the vertical component.

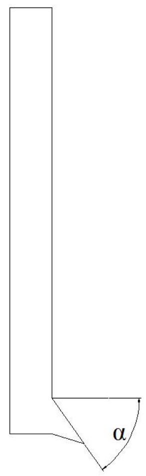

Figure 6.1: An illustration showing the rake or attack angle of a tine shown by α.

What also must be considered when modifying the design of the tine is the degree of loosening and particle pulverization. When sowing a crop, a good seed be is required but the aim of using a no tillage farming system is to minimize soil disturbance to maintain soil structure and the soil moisture content. By decreasing the rake angle α whilst maintaining the same ratio of operating depth to tine width, soil pulverization and tine draft force decreases but soil cutting efficiency increases. With an increased rake angle, more of the soil pulverization and draft force increase creating a decrease in the soil cutting efficiency. For a no tillage farming system, soil pulverization and loosening should be kept to a minimum. By doing so, the draft force experienced by the tine should also be kept comparatively low. See figure 6.2 for a graphical representation of the relationship between rake angle, slenderness ration, draft force and degree of soil loosening and pulverization.

Using the same spreadsheet as used for the original tine draft force predictions, changes in rake angle were made and the draft force was recorded. These results were then tabulated and a graph was produced showing how the draft force varied over the same range of depths using varying rake angles between 30 and 60 degrees. See Figure 6.3 for a graphical illustration of the results which can be found in table 6.1.

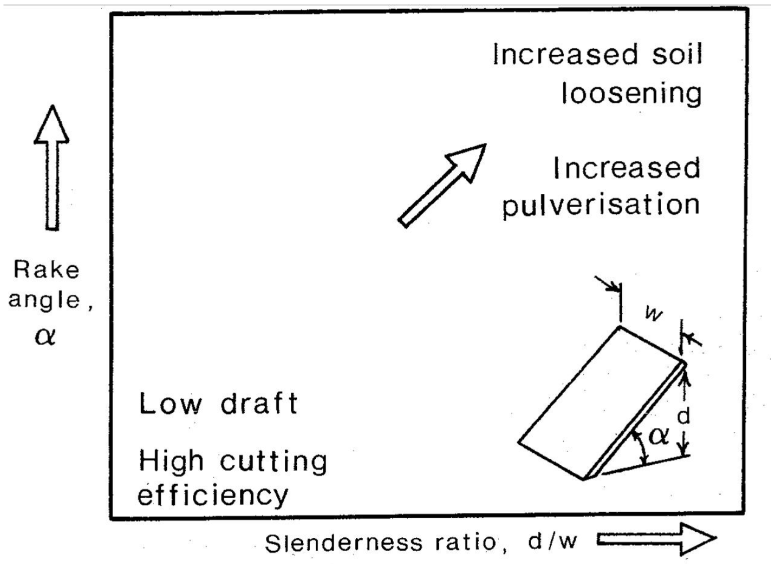

Figure 6.2: An illustration showing rake angle and ratio of d/w effects soil loosening Source: Mckeys (1985).

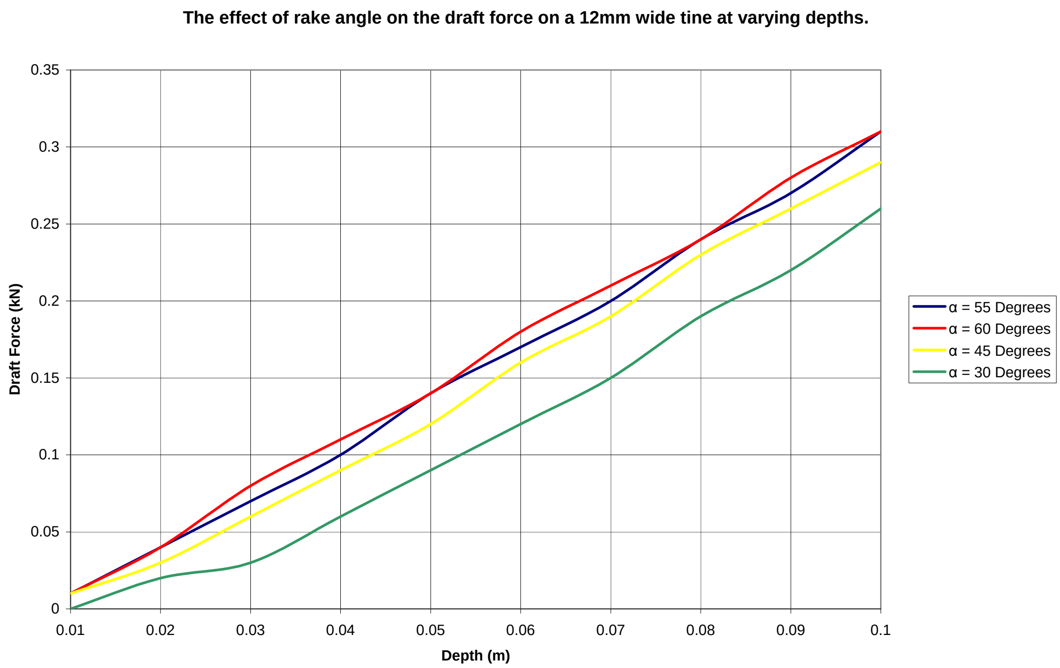

Figure 6.3: A graph showing how the draft force varies with rake angle and depth for the 12mm wide tine.

	Draft Force (kN)			
Tool depth (m)	$\alpha = 30^o$	$\alpha = 45^o$	$\alpha = 55^o$	$\alpha = 60^o$
0.01	0.00	0.01	0.01	0.01
0.02	0.02	0.03	0.04	0.04
0.03	0.03	0.06	0.08	0.08
0.04	0.06	0.09	0.11	0.11
0.05	0.09	0.12	0.14	0.14
0.06	0.12	0.16	0.18	0.18
0.07	0.15	0.19	0.21	0.21
0.08	0.19	0.23	0.24	0.24
0.09	0.22	0.26	0.28	0.28
0.10	0.26	0.29	0.31	0.31

Table 6.1: Values of draft at varying depths a range of rake angles.

6.3.2 Analysis and Discussion of Results of Tine Modifications

If the design of the tine is to be modified, specifically the rake angle, the draft force can be reduced by up to an average of over 35% over the range of depths specified in table 6.1. This would involve changing the rake angle from 55^o to 30^o. In terms of manufacturing techniques, nothing will be different as they are currently manufactured by Spring Ridge Engineering, Spring Ridge, NSW, using a profile cutter. With the limited tractive force being produced by the power tiller, the reduction in the draft force produced by the tine would make it possible to use an extra tine if required. The maximum vertical component produced by the various rake angles on the tine is constant. This will ensure that the tine won't exert an excessive vertical force and cause the seed drill to "bury" itself and cause excessive wheel slip.

6.3.3 Crossbeam Modifications

In an effort to reduce the overall cost of the complete seed drill, a reduction in the the wall thickness of the square hollow tubing in the crossbeams was suggested. The original design specified a wall thickness of 4mm in 50mm square hollow section for the crossbeams. From 4mm wall thickness, the next thinnest is 3mm, followed by 2.5mm. It was decided that both models using 3mm and 2.5mm wall thickness 50mm square hollow sections would be tested.

It was decided that the testing procedure was to remain the same as what original frame was subjected to. This enables the analysis of the 4mm 50mm square hollow section to be used as a benchmark. The analysis involved a 1kN force acting perpendicular to the face of the point of each tine, restraints on the 4 bolt holes on the pieces of 10mm flat steel on each end of the crossbeam and where applicable the 3 bolt holes on the mounting bracket were also restrained.

A static finite element analysis was selected for the analysis of the modification to the crossbeams as the tine is only moving at 1m/s and a dynamic test would have taken a lot longer. A larger force than what the tine may experience was used to incorporate a Factor of Safety.

6.3.4 Results of Analysis of 3mm Wall Square Hollow Section

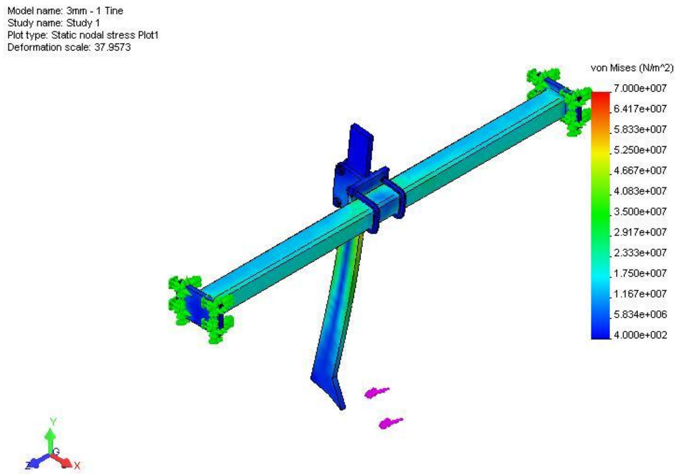

Figure 6.4: A plot of stress concentrations for a crossbeam with a single tine.

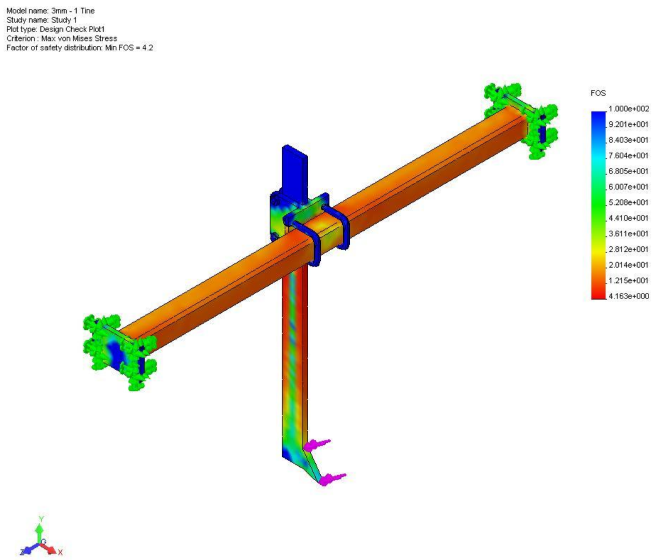

Figure 6.5: The Design Check for a crossbeam with a single tine.

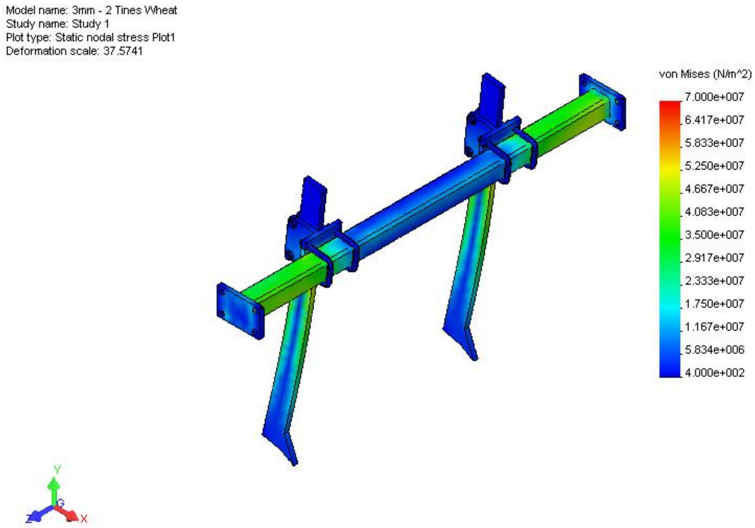

Figure 6.6: A plot of stress concentrations for a crossbeam with two tines set up for Wheat.

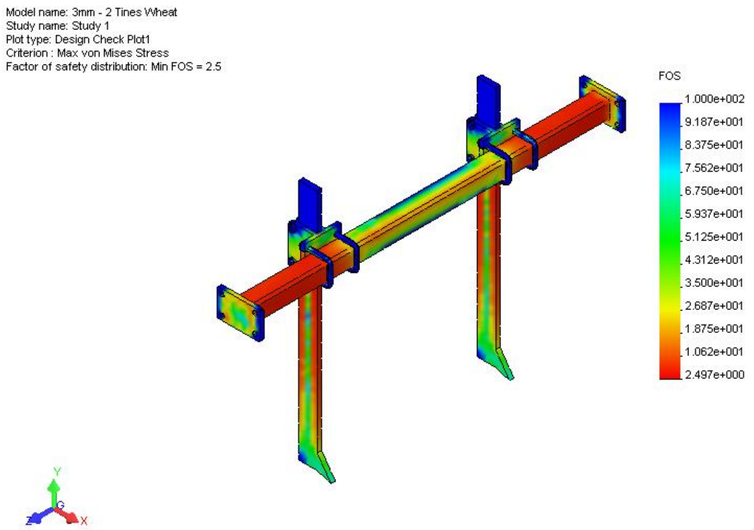

Figure 6.7: The Design Check for a crossbeam with two tines set up for Wheat.

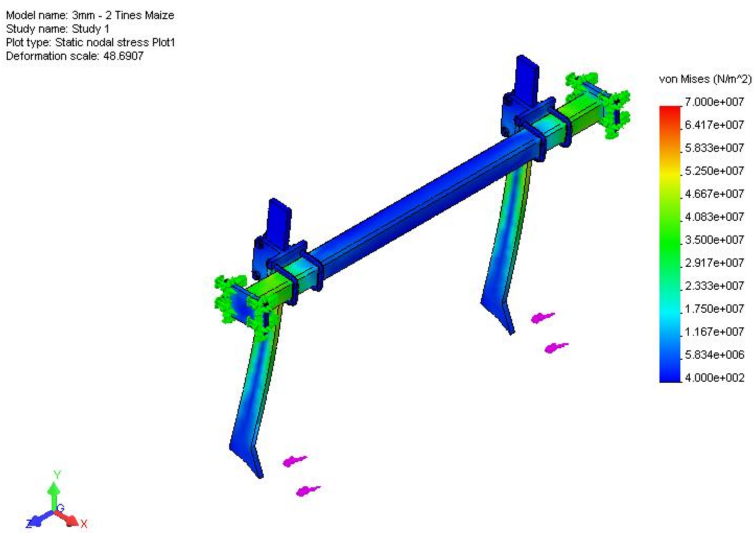

Figure 6.8: A plot of stress concentrations for a crossbeam with two tines set up for Maize.

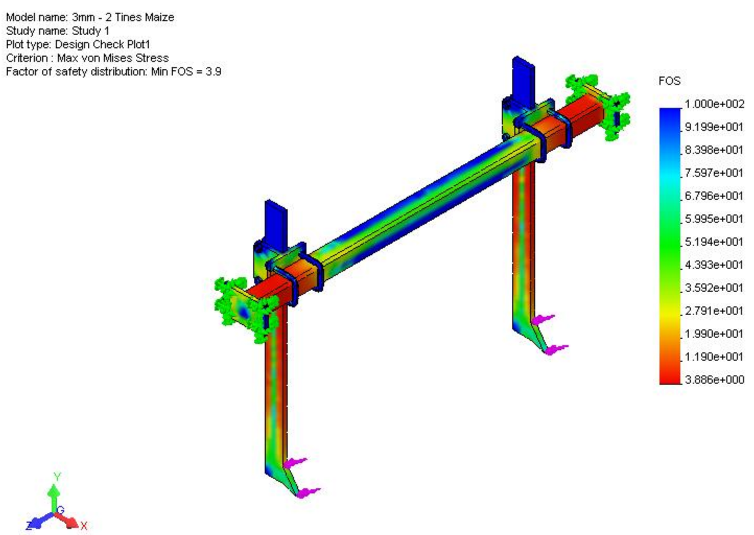

Figure 6.9: The Design Check for a crossbeam with two tines set up for Wheat.

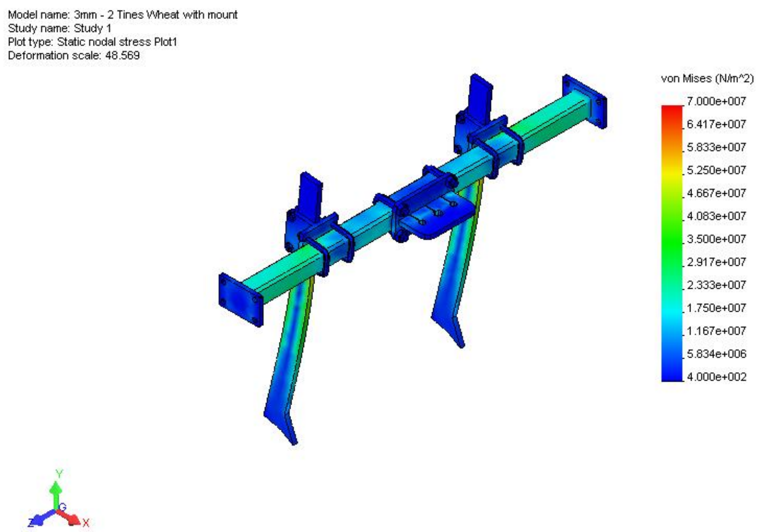

Figure 6.10: A plot of stress concentrations for a crossbeam with two tines set up for Maize and the mounting bracket.

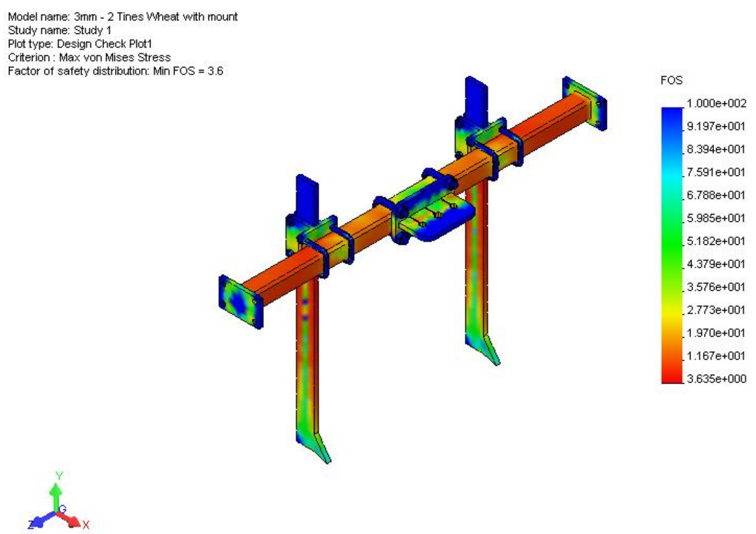

Figure 6.11: The Design Check for a crossbeam with two tines set up for Wheat with the mounting bracket.

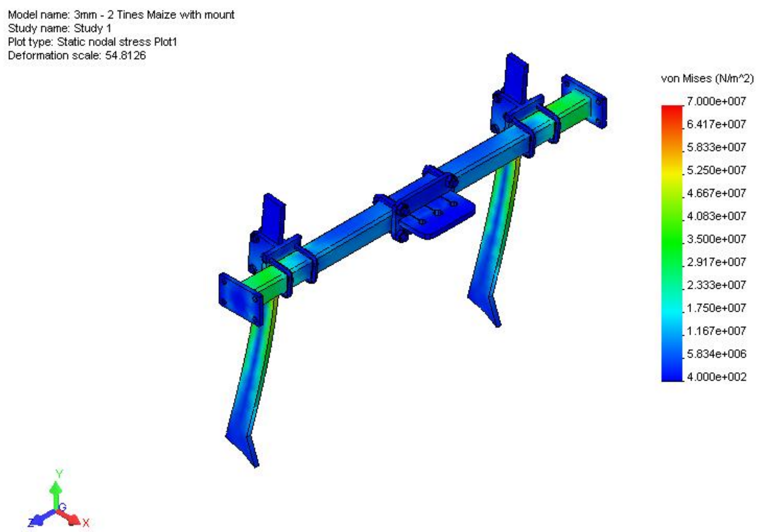

Figure 6.12: A plot of stress concentrations for a crossbeam with two tines set up for Maize and the mounting bracket.

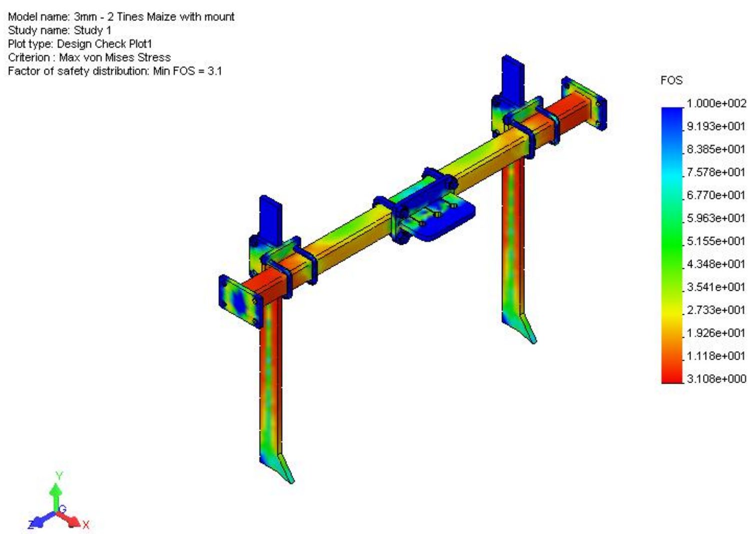

Figure 6.13: The Design Check for a crossbeam with two tines set up for Maize and the mounting bracket.

6.3.5 Results of Analysis of 2.5mm Wall Square Hollow Section

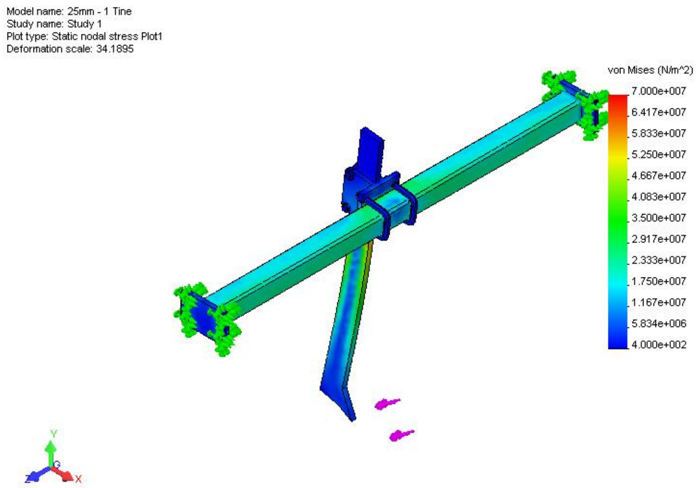

Figure 6.14: A plot of stress concentrations for a crossbeam with a single tine.

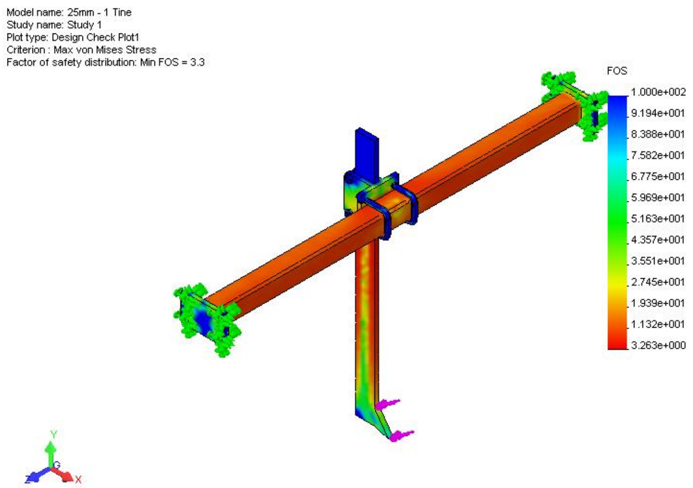

Figure 6.15: The Design Check for a crossbeam with a single tine.

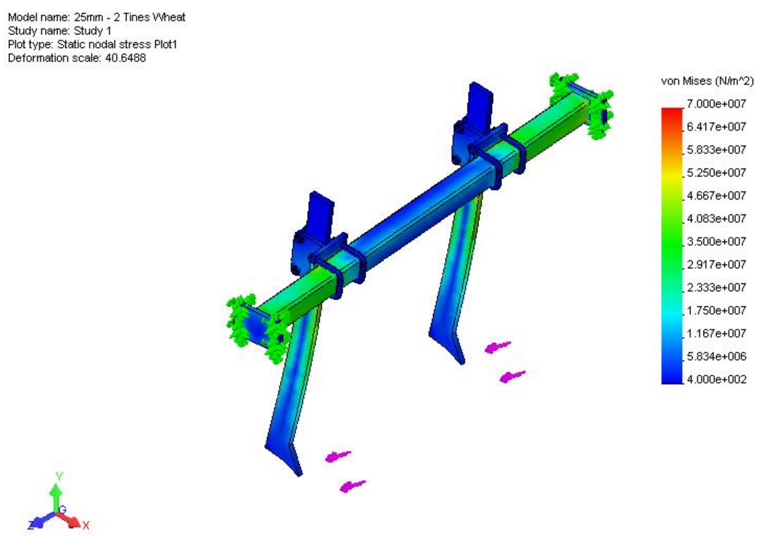

Figure 6.16: A plot of stress concentrations for a crossbeam with two tines set up for Wheat.

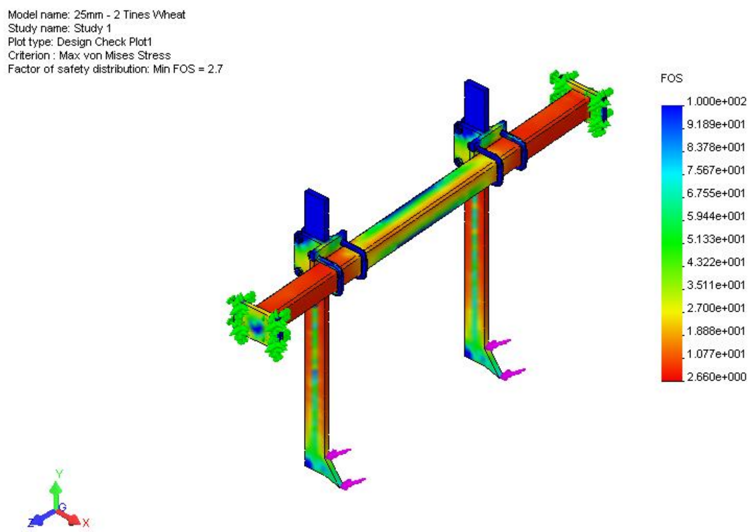

Figure 6.17: The Design Check for a crossbeam with two tines set up for Wheat.

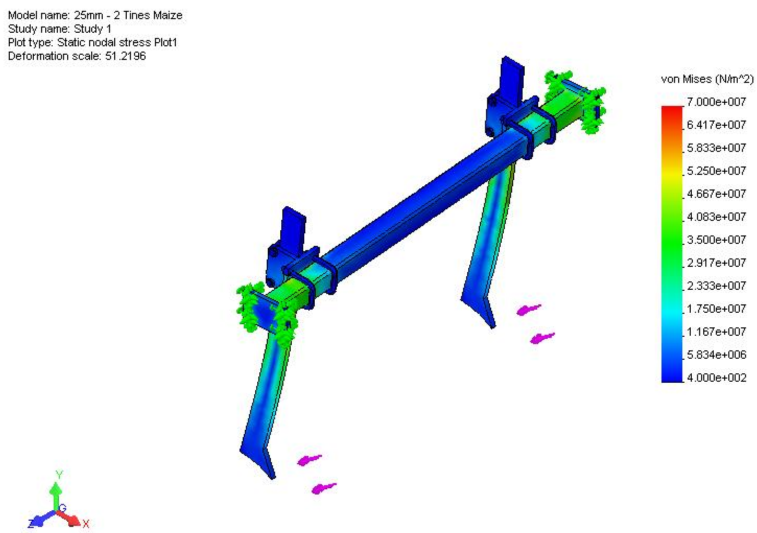

Figure 6.18: A plot of stress concentrations for a crossbeam with two tines set up for Maize.

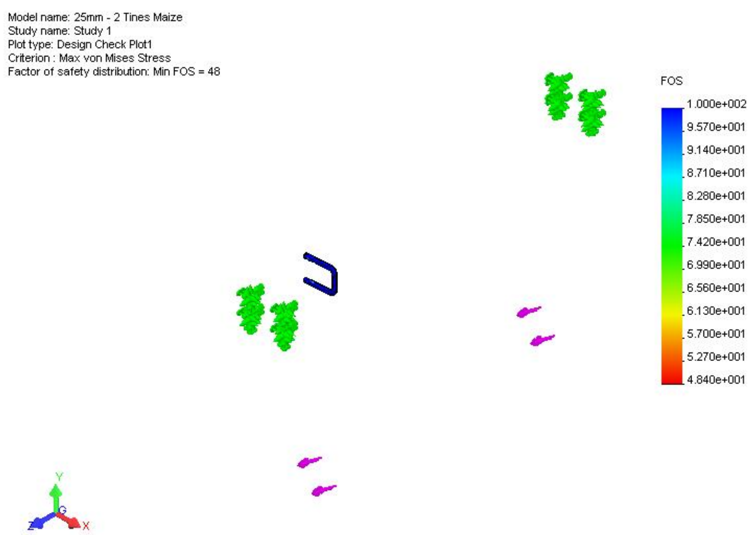

Figure 6.19: The Design Check for a crossbeam with two tines set up for Maize.

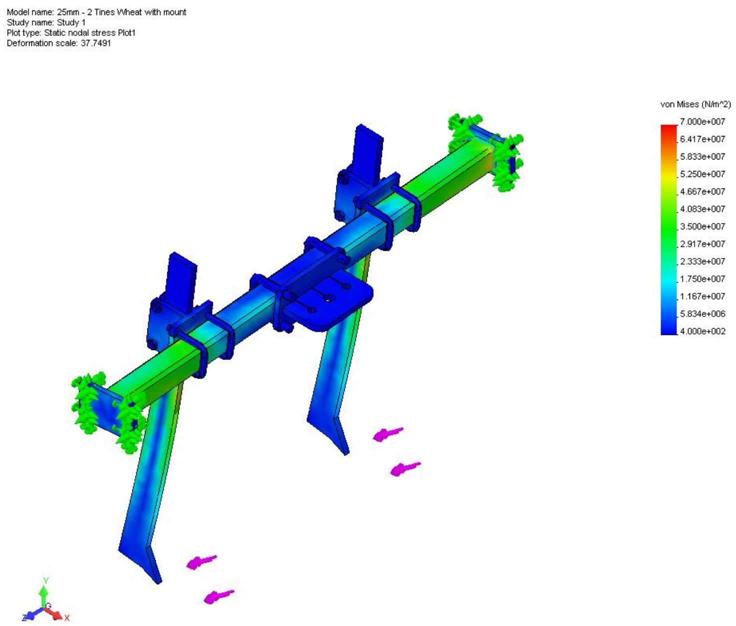

Figure 6.20: A plot of stress concentrations for a crossbeam with two tines set up for Wheat and the mounting bracket.

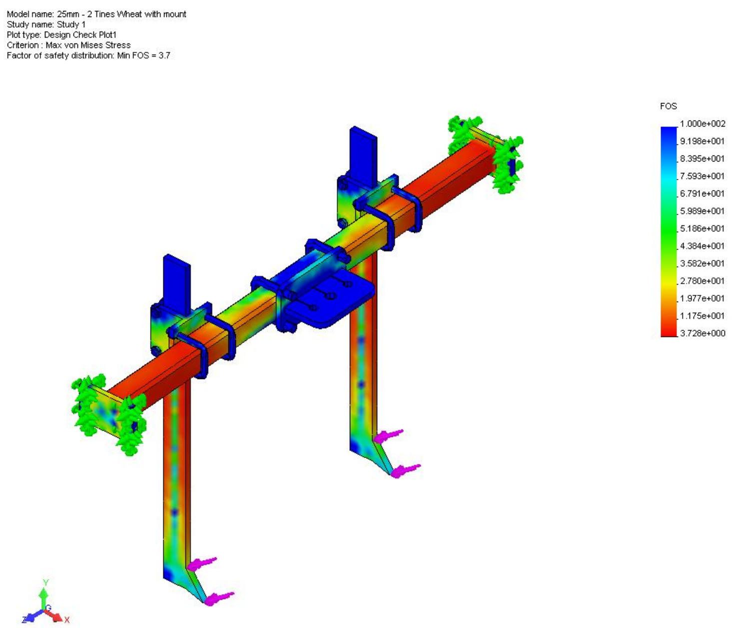

Figure 6.21: The Design Check for a crossbeam with two tines set up for Wheat with the mounting bracket.

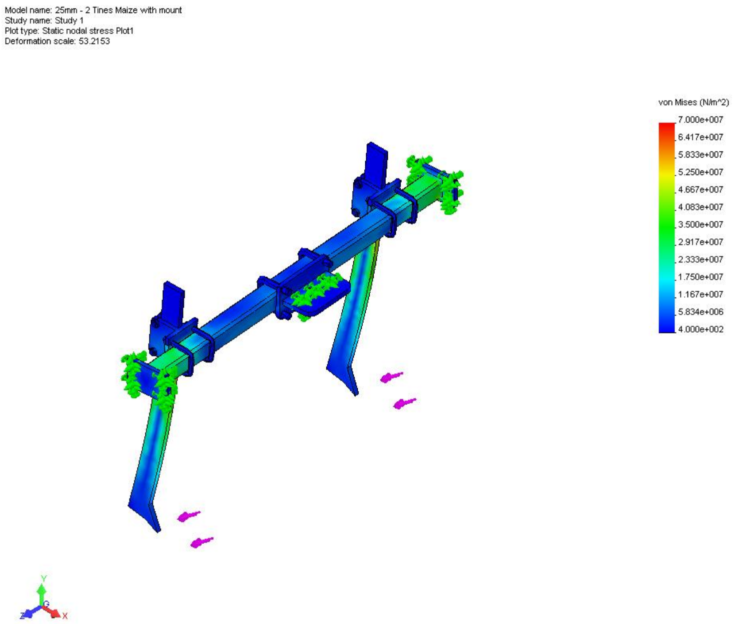

Figure 6.22: A plot of stress concentrations for a crossbeam with two tines set up for Maize and the mounting bracket.

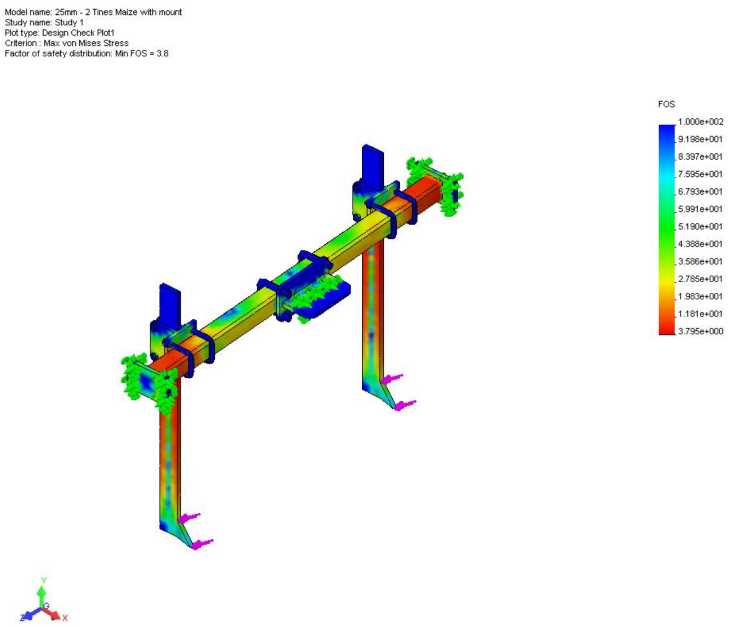

Figure 6.23: The Design Check for a crossbeam with two tines set up for Maize and the mounting bracket.

6.3.6 Analysis and Discussion of Results of Crossbeam Modifications

Rather than looking at individual results, the results from the FEA of the modified designs will be compared against the results from the analysis of the original frame. This is because the original design specifications of the frame have been proven to work in the field and also stood up to the standard FEA procedure.

Compared against the crossbeams made with square hollow section with 4mm wall thickness, the 3mm wall thickness material has comparable results. The size of areas with higher stress concentrations do not increase dramatically and the values of the maximum stresses do not increase. The Factor of Safety (FOS) according to the same Von Mises Stress Failure criterion as used in the analysis of the original frame does decrease with the thinner wall thickness by up to 20% but this results in a new FOS of 3.1. When looking at the testing procedure that is used to achieve these results, a FOS of 3.1 is still very acceptable as the forces being exerted on the assembly are very large in comparison to the predicted values.

When comparing the results between the 2.5mm wall thickness against those of the original 4mm wall thickness of the 50mm square hollow tubing there is more of a noticeable difference. This is most likely due to the 37.5% reduction in the thickness of the material. Stress concentrations in both the original and modified models are extremely similar except the minimum FOS is actually increased in one case. This is the opposite of what would have been expected in this situation. The minimum FOS is approximately 2.5 which can be considered acceptable in this situation. There can be other implications for using a thinner walled material but in terms of the predicted forces the assembly will encounter, it is adequate.

For unknown reasons, the design check image would not work correctly for the Maize tine arrangement on the 2.5mm 50mm square hollow section. This was most likely caused by a bug in the software. This problem occurred in a number of the models but was rectified relatively easily. It is not known why this particular model would not display the required information.

Chapter 7

Discussion of Results

7.1 Tractive Force Prediction

With the use of various models, both theoretical and empirical, predictions were made for the maximum tractive force that a power tiller is able to produce with a seed drill attached. Along with these predictions, results of a practical experiment have been published in a research journal article. There was quite a large discrepancy between the predictions and the measured results with all the predictions being close to or greater than twice as large as the recorded results.

It is difficult to be able to definitively conclude that a power tiller is only capable of producing a maximum of 1kN when using pneumatic tyres because of the variation in soil conditions that can be encountered. Not to mention the effects of dynamic weight transfer which occurs with excessive wheel slip. Also changing the diameter and width of a tyre will also change the traction and rolling resistance forces as well. The addition of wheel weights in the prediction of tractive forces did not significantly increase the net tractive force but did improve the tractive efficiency.

However, in a no tillage or conservation farming system heavy machines cause a lot of soil compaction. In an effort to keep in line with a conservation farming system, the weight of the traction machine and implement should be kept to a minimum. As the average contact area of a power tiller tyre is relatively small, the impact of additional wheel weights could potentially be very large. For this reason, a modification to reduce the draft force on a tine is preferable to maintaining an inefficient tine design and using extra weight to gain more traction.

7.2 Attachment of the Seed Drill

According to calculations done on the 4 studs responsible for attaching the transfer casing which holds mounting bracket of seed drill onto the back of the transmission housing, a stress of 7.367 MPa was calculated. According to Australian Standards, the Minimum Tensile Stress any bolt, screw or stud should be able to withstand is 330 MPa. Considering that the stress in the studs is less than 2.5% of this value, there should not be any fear of them breaking off. However, there is the problem of the threads of the studs being pulled off or pulling the thread out of the bolt holes in the transmission housing. This was unable to be analysed as the type and pitch of the thread on the bolts is unknown, not to mention the Property Class Rating of the bolts and the material that transmission housing is made of.

7.3 Tine Draft Force Predictions

Tine draft force predictions were made over a range of depths. The range of depths was decided upon by investigating the range of depths at which Wheat and Maize seed would normally be planted. The maximum operating depth was then increased by 30mm to allow for an inconsistent surface profile. A set of standard soil conditions were used for the analysis of both the draft forces and the tractive forces to maintain the ability to be able to compare the results of each testing procedure.

According to Godwin and O'Dogherty (2006), the results given by the spreadsheet are the mean values of draft that a tine will produce. The actual force experienced is cyclical with an amplitude of 20% about the mean, especially with low rake angles and narrow tines.

As soil is an anisotropic substance, it is extremely hard to be able to predict how it will fail. There are so many parameters that modify the tensile strength of a soil which can change with distance in all 3 directions. The presence and size of clods in the soil will also change how the soil fails as well because of the stronger bonds between the soil particles.

The spreadsheets written by McKeys et al made the task of predicting the draft force a lot easier. This is because it is a spreadsheet designed purely for predicting such a force and it was able to calculate a lot of dimensionless factors that otherwise would have had to have been read from charts found in Appendix 3 of Soil Cutting and Tillage by Mckeys (1985). The only parameters needed to be entered were soil conditions and tine specifications. The results produced are the mean results and can vary by up to 20% but are still considered to be the most accurate predictions available.

7.4 Current Design

From the Finite Element Analysis performed on the frame design supplied, the material chosen for key components has been correct. According to the analysis performed, there is a minimum Factor of Safety against failure according to Von Mises Stress failure criterion of 2.5. It must be noted that it is hard to simulate a weld where two separate pieces of material are joined. This because when the parts were modeled, the two pieces were forced to merge into one.

The analysis involved restraining the ends of the crossbeams and applying a force to the point of each tine. As the applied force creates a bending moment about the end plate, the areas of high stress which occur around where the end plate is welded to the square hollow section. The bending moment is caused because the end plates of the crossbeams are fixed and the applied force cause the square hollow section to bend. This results in one side of the material to be in compression and the other to be in tension.

However, as the seed drill has already been put through in-field testing, this is purely a confirmation of the design. Should it be used behind a traction machine able to produce a larger maximum tractive force, the design should be reanalyzed. Stress concentrations on the original design are not very high and are not considered to be near the value that would induce failure but are still considered to be realistic.

7.5 Implications of Design Modifications

Before any modifications were made to the current design, there were certain criteria that had to be met. The modification had to be able to stand up to the same testing procedure that the original design was subjected to. It had to be able to be fitted with the original frame using pre-existing bolt holes or use the same sized "U" Bolts. The change in design should not increase the cost of the seed drill itself as it is being marketed at owners of power tillers which are used predominantly in third world countries.

Chapter 8

Conclusion

From this research project, theoretically a 8.95 kW should be able to pull 5 tines through soil that would be common in countries such as Bangladesh, Cambodia and Laos. Due to anisotropic nature of soil, tractive force and tine draft forces are extremely hard to accurately predict. Both tractive force predictions overestimated the maximum tractive force by a minimum of 150%. The most reasonable tractive force prediction was given by an empirical model. This is most likely due to the fact it is derived from data collected in the field. From the data collected and analysed in the article by Narang and Vareshney (2005), a power tiller would be capable of pulling only 3 tines at once. According to reports from the sponsor who has used the seed drill, this is the reality.

The current design of the frame of the seed drill has been verified as being satisfactory for use behind a 8.95kW power tiller. From here, there are modifications that can be made to the design of various components of the seed drill. These changes will impact either cost or the draft force on the tine. The key criteria for the changes was reliability, if a change in design could not withstand the testing process that the original design was subjected to then it was considered to be an acceptable modification.

8.1 Recommended Design Modifications

8.1.1 Tine Design

Whilst the current design of the tine is satisfactory, there is an modification that could improve the project as a whole. As the tractive force of the power tiller is restricted, in order to be able to pull more tines through the soil, a change had to be made to the design of the tine. Due to the size of seed which has be placed in the slot in the soil cut by the tine, the width of the tine must remain the same. As is the case with the depth that the tine operates at. This resulted in a decision to modify the rake angle of the tine. In order to minimise the draft force, a number of a different rake angles were analysed.

As the analysis was being performed over a range of depths, rake angles were varied by 15 degrees at a time. This gave new draft values for rake angles between 30 and 60 degrees. With the original rake angle being 55 degrees, using a rake angle of 30 degrees will decrease the average draft force by 40% over the given range of depths. This decrease in draft force will make the tine much easier to move through the soil as well as decrease the amount of disturbance occurring which is a priority of the no tillage farming system which the product is being designed for.

8.1.2 Crossbeam Dimensions

With the seed drill being aimed at users predominantly in third world countries, cost is a major issue when trying marketing the product. In order to cut the overall cost, the possibility of reducing the thickness of some sections was investigated. In particular the thickness of the wall of the square hollow section. This is because it is a major component of the seed drill. The original frame was made using a 4mm wall thickness on a 50mm square hollow section. In commonly used section thicknesses, the next thinnest are 3mm and 2.5mm. As it was not very hard to change the wall thickness of the section in Solidworks and repeat the analysis procedure, both new wall thicknesses were analysed using the same process that the original frame design was subjected to.

The original frame using the 4mm wall square hollow section was used as a benchmark. This is because of the fact that it has been put through some field trials and has been able to stand up to the forces a power tiller can exert on it. As well as the field trials, FEA has verified its integrity. From the results of FEA done on thinner wall thicknesses, stress concentrations are marginally increased and the Factor of Safety does decrease in some cases. The results do suggest that a 2.5mm wall thickness should be able to withstand the forces that a power tiller should be able to exert but there are other implications of using a thinner wall thickness.

Thinner material is much harder to weld together and can result in an increased number of weak points. Where the 50mm square hollow section is welded to the end plate which bolts onto the side bars would be the weakest point in the design as there is not as much material to weld together. Even though the Finite Element Analysis does suggest that material with a 2.5mm wall thickness will be able to cope with the forces, the testing done does not take into account the change in strength cause by welding activities. For this reason, a 2.5mm wall thickness may not be suitable.

If the seed drill is subjected to extreme working conditions such as Rice paddies or an environment where there is a lot of moisture, rust and corrosion may lead to a weakening of the frame. If a 2.5mm wall thickness material was used for the crossbeams, the rust and corrosion may cause a reduction in the strength of material and catastrophic failure may occur. This would most likely occur where the square hollow section is welded to the flat plate with 4 bolt holes. In order to reduce the cost but maintain a good design life, a 3mm wall thickness 50mm square hollow section is recommended. The thicker material will assist in achieving a strong weld which should be able to withstand any forces that will be exerted on it. The effects of rust and corrosion should not cause it to weaken to the point where it will fail.

8.2 Future Work

8.2.1 Hitching Method

There has also been a suggestion to modify the mounting mechanism to allow the seed drill to be used behind small 4 wheeled tractors in such applications as vegetable growing. This would involve the construction of an "A Frame" to attach to the toolbar to couple onto three point linkage system on the back of the tractor. Whilst this would make the seed drill much more marketable and be much more versatile in its applications, a small 4 wheeled tractor is most likely capable of producing a larger traction force than a power tiller. This will mean more tines will be able to be used on the toolbar at any time. This will increase stress concentrations within the toolbar and may result in failure.

Figure 8.1: A basic design of a Three Point Linkage hitching system for the seed drill.

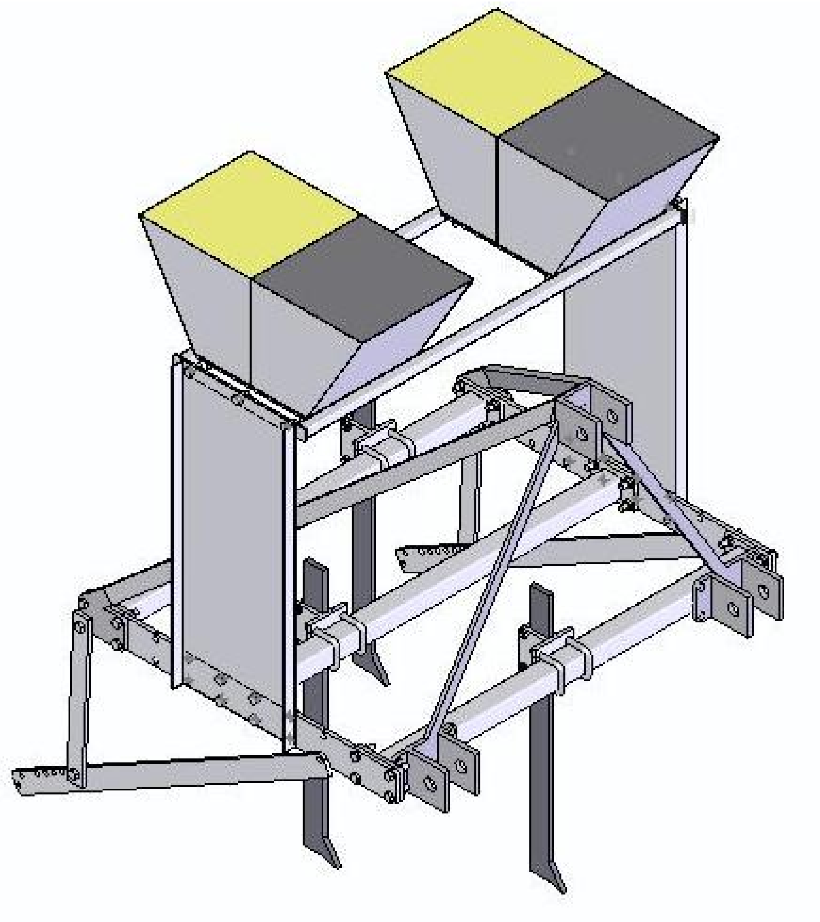

Figure 8.2: An illustration of the possible design of an "A Frame" type setup for the seed drill.

Figure 8.1 is a 3D model of a basic "A Frame" hitch which can be bolted on to the seed drill frame without having to make any modifications. If the seed drill was going to be used behind a tractor able to pull more tines through the soil, larger crossbeams and more and or larger seed and fertilizer bins may also be extra options for modifications. Due to the modular nature of the drill, there would still be numerous common components to all of the seed drills which is advantageous for manufacturing and sourcing replacement parts when needed.

If a Three Point Linkage system was made for the seed drill, it could be utilized by a wider range of users. There would be a percentage of farmers, most likely in the high value crops such as vegetables, in Australia that would use the seed drill. The advantage is that it is not very large and does not require large amounts of horsepower to pull it and it has been designed specifically for no tillage systems. It would make the seed drill much more marketable as the range of possible users would increase.

A future project topic may involve looking into applications for the seed drill in the domestic market, specifically the growers of high value crops. The growing of high value crops will most likely occur on raised seed beds and seeding will occur on top of these beds. A wider seed drill could be manufactured that would be able to be used for the planting of the crops. The project may look into the effects of widening the toolbar and a modified hitch to utilise a three point hitch. Another part of the project may involve looking at modifications to seed and fertiliser boxes and the feasibility of such modifications.

8.2.2 Limitations

The design shown in Figures 8.1 and 8.2 has not been subjected to any testing at all and should not be considered to be suitable for immediate implementation. There are a number of different hitching methods and designs of hitching devices on the market at the moment and there needs to be research done into the main type of traction machine that the implement will be used behind and category or type of hitch that is installed on it. There may also be a much more efficient way of ensuring that the hitch will be attached properly.

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Appendix A

Project Specification

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 Research Project
PROJECT SPECIFICATION

STUDENT NAME:	Mark Fraser
TOPIC:	Development of no-till toolbar for a two wheeled power tiller
SUPERVISOR:	Dr Guangnan Chen
SPONSORSHIP:	Jeff Esdaile
PROJECT AIM:	This project aims to research current no-till systems for a power tiller and analyse a current supplied design.

PROGRAMME: **Issue A, 4th March 2008**

1. Identify requirements of target user market of the product.
2. Research current products for power tiller systems incorporating no till systems.
3. Complete a Finite Element Analysis of the current design as supplied from the sponsor.
4. Suggest improvements that could be made to the design of the frame and the power transmission system, and discuss the feasibility and practicality issues of such improvements.
5. Derive the maximum number of tines that can be used by the power tiller in a no till system.

As time permits,

6. Investigate the possibility of increasing the maximum tractive force exerted by the power tiller and implications on the other components of the machine and;
7. Modify the toolbar and tine design accordingly.

AGREED

_________________ (Student) ___________________, _________________(Supervisors)

___/___/___ ___/___/___ ___/___/___

Appendix B

AutoCAD Job Card Drawings

JOB CARD

CAD FILE. Esdaile
PAGE

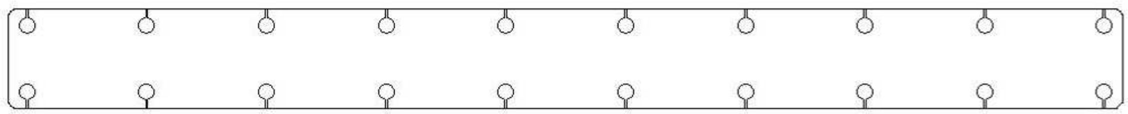
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD01

Description :



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 10mm plate DRW No. ESD01	6	
		Clean and pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

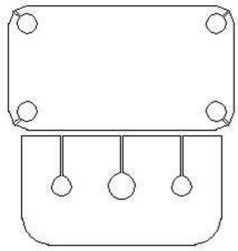
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD02

Description :



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 12mm plate DRW No. ESD02	3	
		Clean and pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile

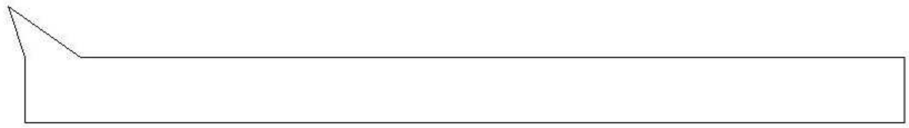
Job : Jeff Esdaile

Description : tine & point

Part No. ESD03

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			24	Profile cut 12mm Bissaloy 360 grade		
				Clean & pack away		

55

TOTAL

JOB CARD

CAD FILE. Esdaile

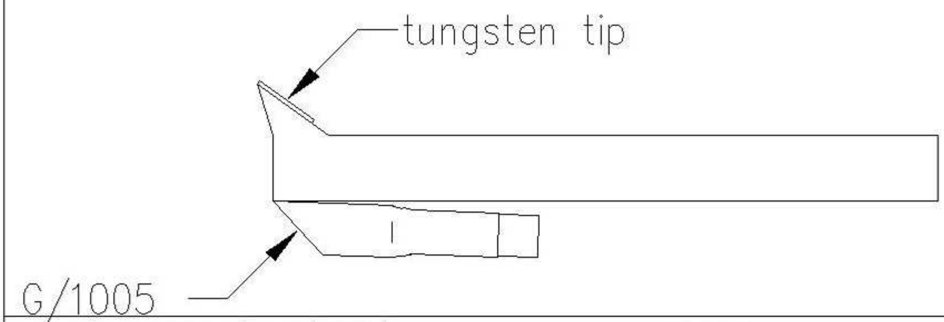
Job : Jeff Esdaile

Description : tine point & boot

Part No. ESD-03

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			24	* Silver solder tungston to point		
			24	* weld on G/1005 boot		
			24	* Paint Black		

TOTAL

JOB CARD

CAD FILE. Esdaile

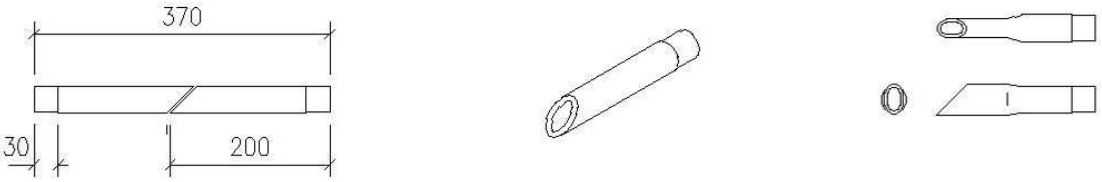
Job : Jeff Esdaile

Description : second boot

Part No. G/1005

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			12	* cut 25NB x 4.0 (red end) pipe x 370 long ends cut square		
				* machine ends down to ø32mm to fit bar flow hose		
				* cut in half @ 45° in cold saw		
			24	* flatten in jig		
				* debur and true up leading edges		
				* pack away		
				TOTAL		

JOB CARD

CAD FILE. Esdaile

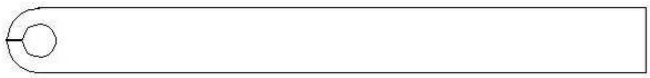
Job : Geoff Esdaile

Description : Coulter shank

Part No. ESD10

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			3	Profile cut 12mm Bissaloy 360 grade		
				Clean & pack away		
					TOTAL	

55

JOB CARD

CAD FILE. Esdaile
PAGE

DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD15

Description : Clutch fork

DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 6mm plate	6	
		Bend in jig		
		Paint Black		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

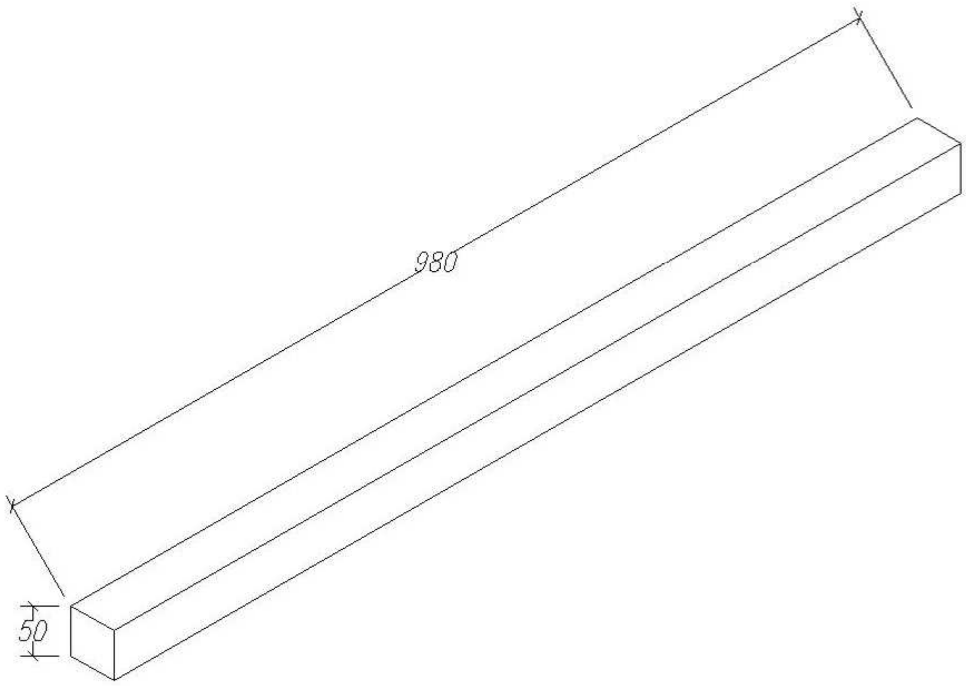
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD04

Description :



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Cut 50x50x4mm RHS 980mm O.A	9	
		Debur ends		
		Clean and pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile

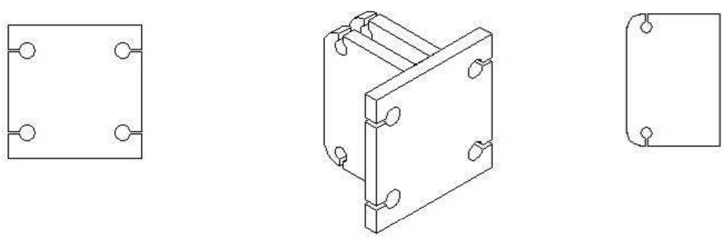
Job : Jeff Esdaile

Description : Tine U/Bolt Bracket & clamp

Part No. ESD-01

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			24	Profile cut 10mm DRW No. ESD05		
			48	Profile cut 10mm DRW No. ESD06		
				*Weld up parts in jig		
				* Clean Paint & pack away		
					TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

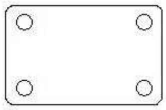
DATE

Bin Location

Job :

Job No : ESD04

Description : U/Bolt 12mm



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 10mm Plate	16	
		Clean & pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

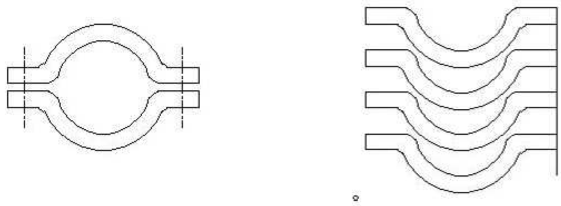
DATE

Bin Location

Job :

Job No : ESD07

Description : Strip tillage tool bar clamp



0.007 ea

DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 50mm Plate	8	
		Punch 12.5mm hole 12.5mm from end		
		Clean & pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

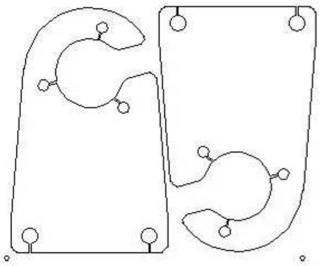
DATE

Bin Location

Job :

Job No : ESD08

Description : Bearing carrier



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 6mm Plate	8	
		Clean & pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

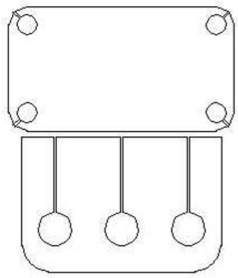
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD02A

Description :



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 12mm plate DRW No. ESD02A	2	
		Clean and pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

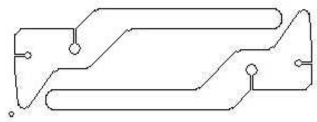
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD14

Description : Clutch lever



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 6mm plate	6	
		Bend in jig		
		Paint black		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

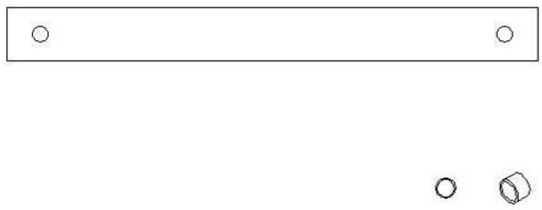
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD13

Description : Press wheel stay arm



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Cut 40x10mm flat 400mm O.A.	12	
		Punch 12.5mm holes 25mm from each end		
		Cut 5/8" Hyd tube 10mm long & debur	12	
		Debur ends & paint		
			TOTAL	

JOB CARD

CAD FILE. Esdaile

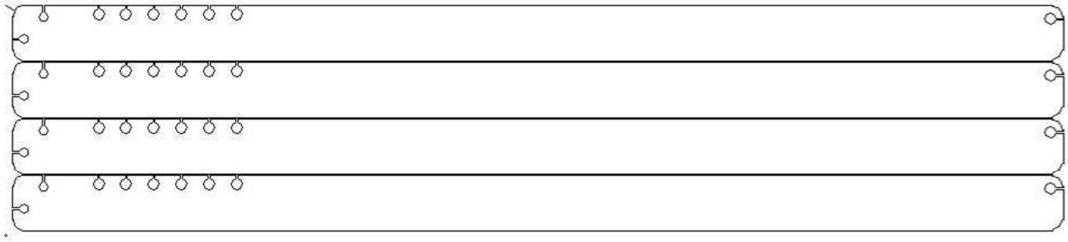
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD12

Description : Press wheel adjustment arm



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 10mm Plate	8	
		Clean & pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

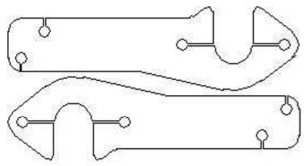
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD11

Description : Press wheel bearing carrier



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 10mm Plate	12	
		Clean & pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile
PAGE

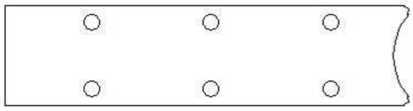
DATE

Bin Location

Job : Jeff Esdaile

Job No : ESD09

Description : Strip tillage tool bar clamp plate



DATE	EMPLOYEE	LABOUR & MATERIALS	QTY	HOURS
		Profile cut 10mm plate DRW No. ESD09	4	
		Clean and pack away		
			TOTAL	

JOB CARD

CAD FILE. Esdaile

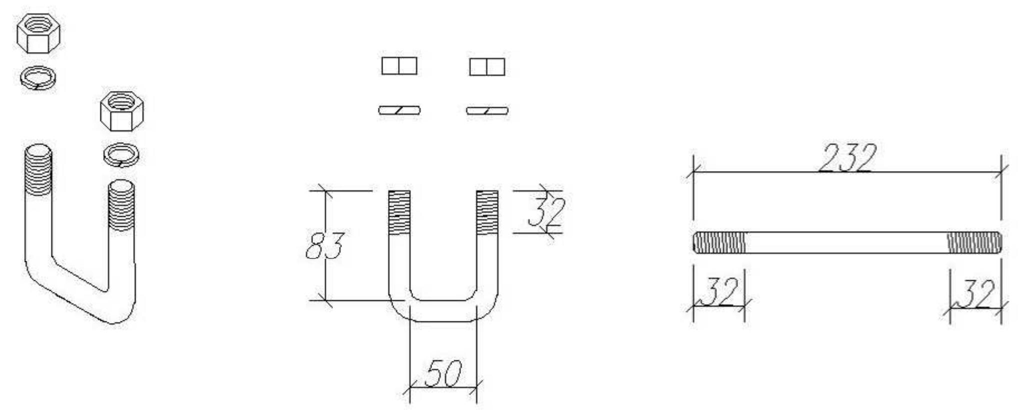
Job : Jeff Esdaile

Description : Tine U/Bolt Bracket & clamp

Part No. UB165083

updated 30.5.07

LABOUR & MATERIALS



DATE	EMPLOYEE	ITEM	QTY	MATERIAL	✓	HOURS
			9	Cut 16mm round bar 232mm O.A		
				* Debur ends 2mm		
				Thread 32mm each end *check with nut every 10 rods*		
				Bend in jig on press		
				Wash and paint black		

55

TOTAL