University of Southern Queensland

Faculty of Engineering & Surveying

The Design of a Three Point Linkage Implement Height Control System

A dissertation submitted by

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ABSTRACT

The purpose of this project was to design a height control system that could be applied to a Three Point Linkage mounted implement. Several methods of achieving this goal were investigated. The two methods that were investigated in detail were an electrical over hydraulic system and a mechanical over hydraulic system.

The unique operation of the three point linkage implements when lift assist is used presents problems when accurate frame height is required, such as when seeding operations are being performed. The system has been designed so that it is constantly sensing the height of the frame at both ends of the implement. The output from this sensing is used to control a hydraulic valve mechanically or electrically. If a height discrepancy between the front and rear of the implement is detected, this valve will be activated to allow a flow of hydraulic oil to the lift assist system, levelling the implement.

The investigation of the electrical control system involved constructing a prototype circuit that was based on a PICAXE microcontroller to provide computing power. This chip was programmed to read the output from two variable resistors and detect any height discrepancies. When a discrepancy was detected, the microcontroller activated a servomechanism that used mechanical motion to correct the position of one of the resistor to match the mechanical position of the two resistors, thus proving the concept of mechanical height control.

The mechanical system was designed so that the output from two height sensing wheels was mechanically input into the hydraulic valve to actuate it when the wheels sensed that there was a height discrepancy. This system was modelled using the solid modelling package SolidWorks and basic engineering analysis was performed on the design.

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ENG4111 Research Project Part 1 &
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Limitations of Use

CERTIFICATION

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.

Student Number:

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Signature

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Date

ACKNOWLEDGMENTS

The execution of this project was supervised by Professor David Ross. I would like to thank him for his support and guidance without which I could not have produced this project.

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GLOSSARY

Implement: Any device drawn behind a tractor to perform an agricultural operation. In this context, the word implement refers to a device that either mechanically prepares the soil or places seeds and other material such as fertiliser into the soil

Tractor: A vehicle used to provide tractive effort and height control to the implement

Tine: a ground engaging device that creates a slot in the ground for the seed to enter

Air Seeder: A device that uses a flow of air to deliver the seed and fertiliser from the storage hopper to the implement

Distributor head: A device that separates the seed and fertiliser from the airflow from the Air Seeder and distributes the solid material to the individual tines

Three Point Linkage: A system used to connect an implement to a tractor using three points of attachment

Lift Assist: A system of wheels that attaches to the rear of a Three Point Linkage implement that supports the weight of the rear of the implement

Hydraulic Ram: A device consisting of a cylinder and piston that converts a flow of hydraulic fluid oil into linear motion

Servo: An abbreviation of servomechanism, which is an geared electric motor used for positional control

CHAPTER 1 – INTRODUCTION

1.1 Introduction

1.1.1 Background Information

Modern farming has become a global enterprise where farmers compete on an international playing field for the ultimate goal of making the largest profit possible. Australian farmers must overcome many obstacles including often poor commodity prices, competition with increasingly subsidised European and American farmers (Reynolds, 1999), an unpredictable climate and the increasing prevalence of drought (The Australian, 2006) and the rising cost of raw materials such as fuel and fertiliser. The local response to these problems in the past has been the rapid increase in the efficiency of the farming process and timely innovation (Howard, 1992).

The adoption of the zero till planting method is a prime example of this increase in efficiency, as economically and environmentally undesirable soil erosion has been reduced under this system (Plaster, *c*. 2002). More fuel efficient tractors, cheaper herbicides and the adoption of GPS (Global positioning system) to guide tractors have also slightly decreased the cost of crop production. As efficiency is the Australian farmer's main weapons in the battle for profit, it is important to continue to come up with better ways to farm so the advantages that the Australian farmers now have are not lost.

There has been a recent movement toward the simplification of agricultural equipment, in particular, planting machines. As the price of steel rises, so does the cost of large planting machines. The employment of larger machines makes economic sense, as wider equipment means a greater area can be planted in one day by a single operator and tractor. This arrangement obviously increases productivity and means the seed is planted in a smaller time window. This is critical as planting time can significantly affect yield (Woodruff &

Tonks, 1981). Another factor to consider is that humans do not like to spent weeks on end working long days, so larger equipment is appreciated by the operator.

Larger Three Point Linkage machines combine simplicity with size and are a natural progression of the move towards more efficient farming methods. These machines can require less material in their construction due to the absence of a large hitch and a reduction in the number of weight bearing wheels. Figure 1.1 shows a Three Point Linkage machine that displays these attributes to some extent, i.e. there is no long hitch and only two weight bearing wheels. These types of systems have several design flaws that are made more acute by an increase in size. These issues and the rectification of these issues are the subject of this project.

Figure 1.1: Image of a three point linkage planter fitted to a tractor

Source:

http://www.gyral.com.au/images/Gyral%20T821%20PARALLELOGRAM%201.jpg

1.1.2 Recent Design Improvements

Large equipment requires a large amount of steel, and engineering and manufacturing costs are higher. It makes sense that planting equipment be simplified so certain parts are made redundant. Three point linkage planters are one example of this trend. Figure 1.1 shows a large, modern trailing seeding implement. Several features are evident in this image (the tractor is obviously not shown). The large hitch and multiple support wheels are several items visible in the image. A trailing air seeder is visible at the rear of the machine. The distributor heads are visible above the frame at the left and right of the planter. A casual observation of the frame of the planter makes it clear that a large quantity of steel has been used in its construction.

Figure 1.2 shows another option that is available to farmers. This system takes all of the parts of the system previously described and incorporates it into one compact unit. This system uses the three point linkage device of the tractor to provide draft and height control. Three Point Linkage is a system that connects the machine to the tractor with three arms. This transfers the weight to the tractor and allows the tractor to raise and lower the machine. This system has many benefits. The machine can be quite light due to the absence of hydraulic systems and a long hitch. The machine sits quite close to the tractor, making the machine easier to turn. The overall simplicity of the machine makes it lighter and cheaper than a towed machine.

Figure 1.2 shows an image of a typical three- point linkage system at the rear of a Case 8950 tractor. This tractor features a quick hitch system which is a device that allows rapid connection of the implement. The three control arms and three points of connection that give the system its name can clearly been seen in this image. The introduction of larger tractors that have three point linkage systems has been one reason behind an increased need for Three Point Linkage planters. One notable feature of the planter in figure 1.4 is the two large wheels situated behind the planter. This is a recent innovation called lift assist. This is simply the placement of wheels behind the planter frame. These wheels are raised and lowered by hydraulic rams operated with the tractor's auxiliary hydraulic system. These wheels take some of the weight of the planter and are necessary to prevent excessive

Figure 1.2: Photograph of a three point linkage system

weight transfer to the rear wheels of the tractor. This system also allows the air seeders to be mounted directly to the rear of the planter, as has happened in figure 1.4. Almost all large three point linkage planters utilise this system.

Figure 1.3 Gyral zero-till planting system

Source: http://www.gyral.com.au/images/Penetrator.JPG (2006)

It can be seen from comparison between figures 1.3 and 1.4 that a conventional planting system can be greatly simplified with the application of sound engineering principles. It is the improvement of the three point linkage system that is the subject of this project.

Lift assist in its current application creates problems when it comes time to turn the tractor. When the end of the run is reached, the driver needs to lift the front of the machine with the three point linkage control switch and lift the back of the implement with the auxiliary hydraulic lever. These actions lift the ground engaging parts (eg. tines) clear of the ground and transfer all of the weight to the lift assist wheels and the tractor. This is necessary to allow the machine to be turned on the smallest possible radius.

Figure 1.4 Homan three point linkage zero-till planting system

Source: http://www.homan.com.au/Airseeders/Airseeders.asp (2006)

These actions become difficult for the driver when the tractor must be steered at the same time as the implement is lifted, as three actions are needed. This means that the driver must lift the implement well before the end of the run is reached and then start to turn. These actions must be repeated after the machine has been turned, although obviously the implement is lowered instead of raised. The two independent height inputs mean that it can be difficult to level the machine from front to back, and the machine must be adjusted prior to use by placing travel limiters on the shafts of the lift assist hydraulic rams. It can be difficult and time consuming to get this adjustment correct.

To simplify the process of lifting and lowering the implement as well as initially setting up the implement for a job, there is a need for a system that synchronises the lift assist height setting and the Three Point Linkage height setting. This system will automatically level the implement regardless of the height setting of the three point linkage setting and will require no input from the operator once it has been initially calibrated.

1.1.3 Objectives

The aims of this project are to develop an automatic height control system that will sense when the implement is not level. When this occurs, a hydraulic control valve will be actuated, and oil will flow into the appropriate side of the lift assist hydraulic system which will in turn level the implement. The overall effect of this system will be to keep the rear of the implement at the same height as the three point linkage height setting.

Two types of height control will be investigated. The first is electrical over hydraulic, in which sensors are used to measure the height of the front and rear of the implement and a computer processes the information from the sensors. If a height discrepancy is detected, the computer sends a signal to a solenoid operated hydraulic valve to provide oil flow to correct the height of the lift assist system.

The second type of system that will be investigated is a fully mechanical sensing system that uses linkages to actuate a hydraulic valve to restore the rear of the implement to the correct height. If it is decided that this system is the most appropriate device, it will be designed using a solid modelling package.

This system will be constructed to be as robust as possible to be able to cope with all expected operating conditions. The operation of the system must display a high level of accuracy and repeatability even in the harshest conditions and it must be extremely reliable. It must not interfere with the basic operation of the implement and shall not require any modifications to the frame of the implement. It must be adaptable to suit most machines. The device must be able to be built quickly and cheaply with the most basic tools. Finally, a farmer with minimum mechanical skill should be able to mount it to their machine in a short period of time.

1.2 Assessment of Consequential Effects

The device that is being designed during the completion of this project will be mechanically simple, however if any part of the implement is used to physically support any part of a planting implant, this presents a safety risk in the case of failure. For example, if the implement was in the raised position and a mechanic was underneath the frame servicing it, there needs to be no chance of sudden failure. It is, therefore my duty to design any load bearing device to national standards using best practice engineering knowledge. If the control system utilises electronic components in any form, it is important to ensure that they do not pose an electrical hazard. It is very unlikely that this would occur, however, as the system will be designed to run on a very safe 12 volts. This will be necessary as almost all tractors have 12 volt electrical systems.

As this device is to be mounted to an agricultural machine that physically affects the soil, this technology could cause an environmental impact. It should be noted that the author is not designing the implement or the ground engaging devices, only a height control system. Therefore as the function of the implement is unchanged, any adverse environmental impacts would have been directly a result of either irresponsible farming practices or a poorly designed planting machine. Modern farming practices are as environmentally friendly as possible and any responsible farmer is always conscious of potential damage to the environment, so this is unlikely to be an issue.

1.3 Project Methodology

A project of this magnitude needs to be conducted in a predetermined manner to ensure that all aspects of the research and development process are satisfactorily completed in a timely manner. The first task is to research the type of machines that are in the marketplace and identify the features that may be relevant to this project. This background information has been found in books, on the internet and in farming journals. The next step is to examine the current designs of three point linkage implements and investigate the type of three point linkage machines that would benefit from height control. This process will involve research into specifications such as current machine sizes and the lifting weights of tractors.

The next logical step is to attempt to find if anyone has designed a similar system and try to find the performance of their device/s to guide this design in the right direction. After several systems have been investigated, it will be possible to use this information to come up with several unique solutions to the problem of height control. After this has occurred, research will be conducted into the operation of a three point linkage planting machine and features will be identified that will need to be exhibited for successful operation of the seeding machine.

The next step will be to design the system in a way that would be most suitable for the expected operating conditions. An electro/mechanical system will be designed to start with. A prototype circuit will be constructed and simulated operating conditions will be used to test whether it will work effectively in the field. If this is not the case, a wholly mechanical control system will be designed. All mechanical parts will be designed in a solid modelling program. The features and analysis of the final design will then be reported on as a conclusion to the project.

1.4 Risk Assessment

There are only plans for one prototype system to be made, and this is the electronic control system for the depth control unit. As the construction of this device is the only planned physical work planned, the risk assessment will be based on this only. This device is a simple circuit, consisting of several simple components. All of these components will be rated to twelve volts to safely handle the planned input voltage. There are two feasible hazards involved in the construction and testing of this circuit. The first is the risk of electric shock. As the voltage is so low, the risk of this occurring is extremely slight. The nature of the testing means that the level of exposure to this hazard will be almost continuous. The consequences of a problem like this occurring will be very slight, possibly a slight tingling sensation. The benign nature of this risk means that no controls will be implemented to prevent its occurrence.

The other possible risk is that a component such as a capacitor could be installed incorrectly. The risk of this situation arising is significant, as the circuit calls for a very specific layout of many components. The level of exposure will be rarely, as it should only take one or two tries to get the circuit assembled properly. The consequences of this occurring range from nothing to minor component damage, and due to this, the only control that will be implemented will be ensuring that the circuit is double checked for faults before testing proceeds.

1.5 Resource planning

In order to build and test a circuit, several components will be needed. The specifications of the components are not known as yet as they will be subject to availability. The circuit will be either wholly analogue or analogue with a digital processor. The electrical engineering department is able to provide any component that may feasibly be required, although the circuit will have to be designed around the particular versions that are available. There will be no direct costs involved as the equipment will only be used once and then returned. If in the unlikely case that a particular component is unavailable, another version of the circuit will be constructed.

The only other resource that will be required is access to the solid modelling software in the engineering computer laboratories. Access to this resource is assured whenever they may be needed. Even if the computers were experiencing technical difficulties, they would be rapidly repaired in a matter of days. Thus it is highly unlikely that this project would be compromised due to a lack of resources.

I LITERATURE REVIEW

CHAPTER 2 – SENSTEK DCM-2 DEPTH CONTROL SYSTEM

2.1 Review of Depth Control Systems

There have been several attempts in the past to design a mechanism to control the depth of seed drills. Two systems in particular have been extensively tested by the Alberta Farm Machinery Research Centre. A series of reports detailing the results of these tests have been released by the Prairie Agricultural Machinery Institute (or PAMI). These tests are slightly outdated as they were carried out in the early to mid 1980's. Only one product, however, could be said to be out of date, as it utilises ultrasonic transducers. These devices, like all electronic devices, have become much more accurate and can cope with a wider range of conditions than the ones utilised in this product. The results gained from the testing of this product are still relevant, however, as the basic operating principles remain.

The other product uses only very basic electronic components, which are still very much the same now as they ever were, if not slightly smaller. The distinction between the aim of these systems and the aim of this project is that the products that exist are designed as fully automatic systems to control the height of drawbar hitched trailing implements for precision seed placement. This project aims to extend this research and develop a system that uses the hydraulic system on the tractor to keep a three point linkage drawn implement frame and wings level.

2.2 Ultrasonic Sensing System

The first system to be reviewed by PAMI was the Senstek DCM-2 depth control system. A photograph of the system is shown in figure 2.1.

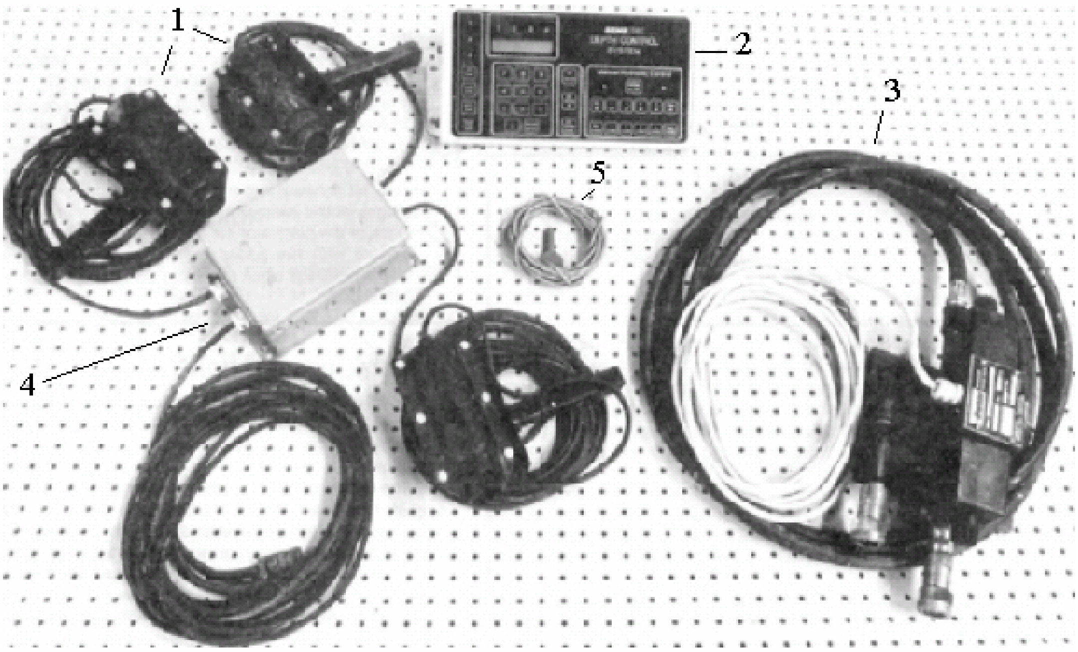


Figure 2.1. Parts making up the Senstek DCM-2 depth control system

(1) Sensors (2) Control console (3) Electro-hydraulic control (4) Implement module

(5) Temperature sensor

Source: http://www1.agric.gov.ab.ca/$department/deptdocs.nsf/all/

eng3075/$FILE/349.pdf (1984)

The Senstek system works by mounting the three depth sensors on the frame of the implement. The sensors work by releasing a pulse of sound and timing how long it takes to bounce off the ground and return to the sensor. They are positioned so they are evenly spaced from the front to back and from left to right of the implement eg. front left, the centre and rear right. This arrangement is designed to give the best possible indication of the average working height of the implement, as the outputs from these sensors are averaged. This averaged value coupled with a variable gain set by the operator and the output from the temperature sensor is directed to the solenoid valve. This valve is mounted to the rear of the tractor, and controls the height of the implement by controlling the oil flow to the height control rams.

The temperature sensor is made necessary because of the reliance on ultrasonic sound as the sensing system. Temperature has a marked effect on the speed of sound in air, thus there is a need for a temperature gain to be multiplied by the sensor readings. Only then is the distance reading from the sensors valid. Other features include the inclusion of a gain that could be adjusted by the user in the tractor that increases or decreases the response time of the hydraulic control system. This works by damping the signal to the electro-hydraulic valve. This means the implement can be towed over rough ground without the undesirable effect of over-compensation.

2.3 Testing

Several issues were identified during the testing of this system. The overall performance of the system was stated to be "good" (*Prairie Agricultural Machinery Institute* 1984, p. 2), as the readings from the sensors were translated to height adjustment with high levels of accuracy. The issues identified all stemmed from the surface of the field that was being worked. When heavy trash or very soft soil was present, the system would "overcompensate" resulting in "erratic depth control" (*Prairie Agricultural Machinery Institute* 1984, p. 2). It was suggested that increasing the amount of signal dampening available could solve this problem.

When the implement was drawn parallel to rills from a previous tillage, the height readings from each sensor would often vary considerably. The effect of this was to cause the cause "the depth control system to alternately raise and lower the implement beyond acceptable limits" (*Prairie Agricultural Machinery Institute* 1984, p. 4). Other minor problems that were encountered involved the failure of two integrated circuits. The cause of these failures was not discovered, but it is possible that vibration was the culprit. These failures indicate the need for electronic components that are both simple and extremely durable, if electronics are to be used at all.

2.4 Conclusions drawn from testing

The Senstek system in its current form was found to be a satisfactory solution to the issue of depth control in certain situations. The ideal conditions for the optimum performance of the system was ground that was uniform in hardness and the absence of stubble or trash, although light trash did not cause any issues. It is also noted that the implement should not be operated in the same direction as previous passes. From these results, it is clear that using ultrasonic sensors in a ground sensing role would not be appropriate under Australian farming conditions. Typical seed beds in Australia where minimum or zero tillage is used consist of a medium to heavy stubble cover and a tramline system which ensures that all passes are made in the same direction. It is clear that the Senstek system is suitable for bare fallow farming in soils of uniform quality.

CHAPTER 3 – INVENTRONICS DEPTH MASTER DEPTH CONTROL SYSTEM

3.1 Height Sensing Wheel Height Control System

A different system also reviewed by PAMI is the Depth Master automatic depth control system made by Inventronics Saskatchewan Ltd. A photograph of the system is shown in figure 1.4. The Depth Master system is built around three control wheels that are mounted along the length of the planter frame. Each wheel is mounted on a trailing arm which pivots near where it attaches to the frame. There is a spring and damper unit mounted to the trailing arm. The spring keeps the wheel firmly pressed to the ground and the damper helps to smooth out small variations in the soil surface such as stones and bumps. The movement of the arms actuates potentiometers that send signals to the control consol situated in the cab.

The control console has a computer that averages the signal across the three control wheels and damps the signal to remove spikes and dips in the signal. The average implement height is then calculated from this corrected signal, and compared to the desired implement height set by the driver. If a discrepancy is detected, an electro - hydraulic valve at the rear of the tractor is activated which sends hydraulic fluid to the hydraulic system on the implement to correct the height. As in the Senstek DCM-2 system, the operator could adjust the level of signal dampening performed by the computer, resulting in a faster or slower response time.

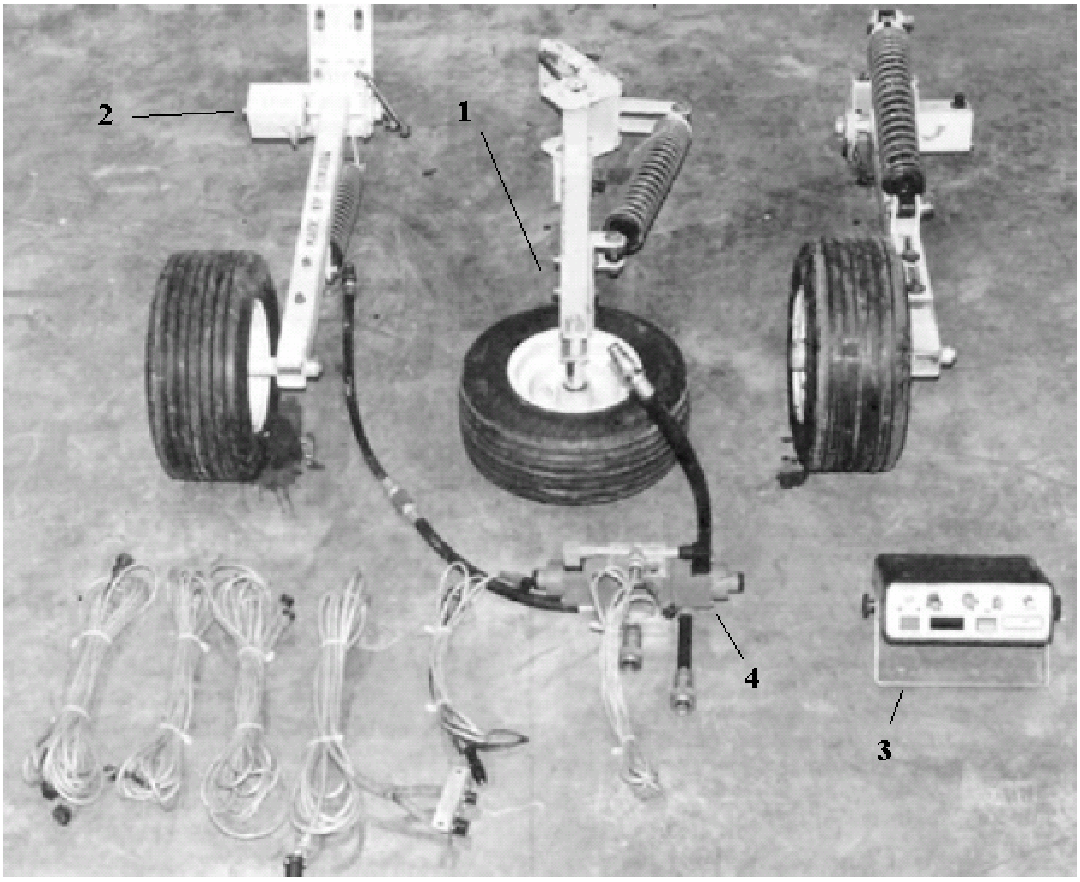


Figure 2.2. Parts making up the Inventronics Depth Master depth control system

(1) Control wheels and arms (2) Variable potentiometer (3) Control console (4) Electro-hydraulic control

Source: http://www1.agric.gov.ab.ca/$department/deptdocs.nsf/all/eng3075/$FILE/470.pdf (1986)

3.2 Testing

The Depth master's performance was said to be "good" (*Prairie Agricultural Machinery Institute* 1986, p. 2). It is noted that the seed placement was quite accurate, even with soils that varied in hardness. The main problem that was highlighted was that when the machine was working in "moist" conditions due to the build up of soil on the gauge wheels (*Prairie Agricultural Machinery Institute* 1986, p. 4). The manufacturer stated in the report that if the correct tyre pressures are maintained and conditions are not extreme, the action of the tyre treads flexing would tend to remove mud from the control wheels. As with the Senstek

DCM-2 system, if the implement was towed parallel to surface ridges from previous passes, uneven height control was experienced. Response time was said to be “adequate” in all conditions (*Prairie Agricultural Machinery Institute* 1986, p. 4).

3.3 Conclusions drawn from testing

The Depth Master system in its current form was found to be a satisfactory solution to the depth control problem in almost all situations. It was noted that moist conditions presented mud build up problems; however working wet soil is an uncommon practice in Australia. It is common practice to guide implements parallel to previous passes, so the issues associated with this mode of operation will have to be solved before control wheels are considered as a viable system in Australian conditions. Speculating, it may be beneficial to provide wider tyres with a flat tread profile. This will allow all of the wheels to keep clear of the hollows most of the time.

CHAPTER 4 – SENSING TECHNOLOGY

4.1 Sensor Application

The height control package will consist of a system that detects the height of the three point linkage system and compares this value to the height of the back of the main implement frame and if necessary, the height of the wings with respect to the ground contours. The system will then either send an electronic message to a hydraulic control system mounted to the implement or mechanically activate a hydraulic valve. The height of the back of the implement will then be adjusted appropriately. This proposed system will require sensors or wheels to detect the height of the frame, devices to interface the sensors with the ground, a device to interpret the signals from the sensors (if electrical) and a system to activate the height control system.

4.2 Variable resistors

Variable resistors are a simple electronic device that has an adjustable resistance. The variable resistor is designed to take an input of physical movement which changes the internal resistance of the device to make the resistance proportional to the movement input. This can be used to create a variable voltage which is proportional to the movement. (*Variable Resistors*, c.2006) Used in this situation, these variable resistors will be best employed connected to height sensing wheels. These wheels are connected to a frame that allows the wheel to move vertically with respect to the implement frame, therefore allowing it to float over the ground countors.

The variable resistor would be mounted to the frame of these wheels so that as the wheel moves vertically, the resistor shaft is twisted and the voltage output changes. The voltage outputs from two or more of these sensors are fed into a computing device that determines whether a height difference exists, and if so, activates a mechanism to allow oil flow to travel to the lift assist mechanism to correct the height discrepancy.

4.3. Ultrasonic transducers

Small ultrasonic transducers usually utilise a phenomena known as piezoelectrics. When certain substances are exposed to a potential difference, they deform slightly. This phenomena is well documented in quartz crystals, however is known to occur in several other substances. Ultrasonic transducers utilise this effect by exposing a specially shaped and heat treated quartz crystal to alternating current at a particular frequency. This alternating voltage causes the crystal to contract and expand at the same frequency as the current. This causes the air to vibrate, resulting in sound. The frequency of this sound is far above the range of human hearing, hence the name ultrasonic**.** (*Piezoelectric Transducers* c. 2006)

The burst of sound is detected by a receiver, which is similar in construction to the transmitter. The receiver crystal works in the reverse of the transmitter, in that mechanical deformation of it by the sound waves causes an alternating electrical current. The sound is picked up by the receiver either at the target where the sound travels in one direction or built into the transducer where the sound is reflected off the target. Built into the transducer is circuitry that measures the time taken for the pulse to travel from the transmitter to the transducer. A time measurement that is increasing in length indicates that the distance between the transducer and the target is increasing. This change in distance creates a voltage that varies proportionally to the height of the implement. (*SRF04 - Ultra-Sonic Ranger* c.2006)

Ultrasonic transducers are a common tool used in many industries as a method of measuring distance. An example of this is the use of depth gauges in vessels containing liquids or granulated substances. The transducer is situated at the top of the vessel and faces the surface of the liquid. Pulses of sound are directed at the substance, and the reflection time is measured. (Määttä & Kostamovaara, c. 2000)

It is conceivable that this technology could be adapted to measure the height of the planter frame in relation to the ground. There are transducers on the market today that are sufficiently rugged to be used on a planter frame, where it would experience vibration, dust and small impacts with field debris (Bullock, B. K. 2006, Pers comm. 17 August 2006). If used within its design limits, the transducer could make an effective depth control system, as modern transducers are accurate to around 1cm – 10m (*SRF04 - Ultra-Sonic Ranger* c.2006). To reduce the need for complex circuitry, the transducers would be situated so the outputs from the transducers on the extremities of the frame are compared to the output from the master transducer. The master transducer in this case is the one that detects the height of the three point linkage. The slave transducers would be located at the critical points of the planter frame. These critical points are the ones that give the best indication of the angle of the main planter frame and the wings with respect to the ground contours. The most important aspect of such a design is how to use the transducers to allow for accurate and reliable readings regardless of field conditions.

The capabilities of the transducers are best utilised in a situation where they point directly at the soil surface and project sound waves at the ground and measure the response. This system was used in the Senstek DCM-2 depth control system discussed in chapter 2. When the results from the testing of this system were reviewed, it is clear that the unreliability of the sensors when soft soil and heavy trash are encountered preclude its use as a direct ground sensing device in a system that must be made as reliable as possible.

The transducers can be used to sense the height of a height sensing wheel using a steel plate that moves closer to and away from the transducer as the frame moves vertically. As this system requires the use of a height sensing wheel, the benefits of using a transducer as opposed to a variable resistor mounted to a height sensing wheel are now eliminated. Due to these reasons, the use of ultrasonic range finding will not be investigated further.

4.4 Direct mechanical control

One other system will be investigated. This system will use two height sensing wheels, one at the front and one at the rear of the implement. These wheels will be identical, and they will operate linkages that in turn operate a valve that controls the flow of oil to the lift assist mechanism. This system is advantageous as it does not require any electronic device, therefore increasing reliability. If the system is properly designed, it has the potential to also offer a high level of accuracy.

II CONTROL MECHANISM DESIGN

CHAPTER 5 – ELECTRONIC CONTROL

5.1 Design guidelines for an electronic system

The system that will be investigated here will be based around two frame height sensing wheels, one at the front to sense the height of the Three Point Linkage system and one at the rear to sense the height of the rear of the implement. The frames that support these wheels will be mechanically linked to variable resistors in a way that translates vertical movement of the wheel into rotation of the input shaft of the resistor. The two resistors will then generate two voltage readings which will be input into a microcontroller. This microcontroller will compare the two voltage readings and find differences that indicate that the frame is not level. The microcontroller will then activate a hydraulic valve using a solenoid to allow oil to flow into the lift assist mechanism which will correct the level of the implement.

A test circuit was constructed to evaluate the feasibility of using a microcontroller to compare two variable voltages and output a suitable command that would turn one of the resistor shafts to correct any discrepancy between the two resistors. In this case, two small variable resistors were connected to a PICAXE 18A microcontroller, the operation of which will be described in another section. This microcontroller detects any differences in voltage that indicate a positional difference between the two resistors. The microcontroller then outputs a signal to a servomechanism that uses an electric motor to rotate the shaft of one of the resistors, correcting the positional difference. It is thought that constructing this system, programming the chip and evaluating the operation of the system will determine the suitability of using this system to control the height of an implement.

5.2 Description of the PICAXE microcontroller

PICKAXE chips are similar in size and shape to operational amplifiers and other similar electronic devices. Figure 5.1 depicts an image of the 18 and 28 pin units which are two of the four chip designs. There are four different chip sizes to allow for different numbers of input/output pins. There are up to three versions of each chip size, each with a different level of functionality. The different versions of each chip size are distinguished from each other with the use of different letters after the chip number. For example, the base model 18 leg chip is named the PICAXE – 18, the more advanced version the PICAXE – 18A, and the most advanced version the 18X. All chips roughly follow this naming convention, although the 40X chip is the only chip with 40 pins and the 08M chip is primarily an educational tool designed to play music. Schematics of the eight chip types with pin labels can be found in the appendix.

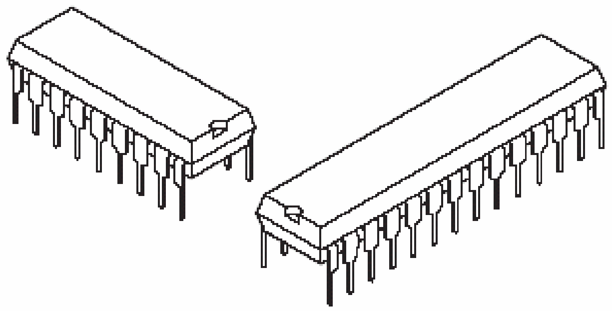


Figure 5.1 - Images of the 18 pin (left) and 28 pin devices

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

5.3 Operation of the PICAXE system

The PICAXE microcontrollers can be described as a small computer on a chip. The chips are fitted with enough memory to hold a medium sized program and a processor. The memory is of the same type found in small USB (universal serial bus) pen drives and some MP3 players, also known as flash memory. (*PICAXE Manual Section 1* c. 2005, p. 27)

The processor runs at a nominal speed of 4MHz, but this can be increased to 8 or 16MHz with the overclocking function (*PICAXE Manual Section 1* c. 2005, p. 52).This overclocking function will not be used in this project as it is not necessary, and the Picaxe operating manual does not recommend the use of this function.

The basic specifications of each chip are listed in table 5.1. The IC size in this table indicates the total number of inputs and outputs built into the integrated circuit. The programmable memory is measured in lines of code instead of the number of bytes, possibly because this is more useful to the programmer when designing the program.
The more advanced controllers use two blocks of memory, one for the code and one for data. All of the chips support an ADC (analogue to digital converter), which can translate a variable voltage input to a digital reading. This means the chip can be used for positional control when a device that outputs a variable voltage is used. These devices include ultrasonic range finders and variable resistors, each of which can be used to measure frame height. The more advanced chips have a higher resolution during this operation than the smaller cheaper chips.

PICAXE Type	IC Size	Memory (lines)	I/O Pins	Outputs	Inputs	ADC (L =low)	Data Memory
PICAXE-08	8	40	5	1-4	1-4	1L	128 - prog
PICAXE-08M	8	80	5	1-4	1-4	3	256 - prog
PICAXE-18	18	40	13	8	5	3L	128 - prog
PICAXE-18A	18	80	13	8	5	3	256
PICAXE-18X	18	600	14	9	5	3	256 + i2c
PICAXE-28A	28	80	20	8	8	4	64 + 256
PICAXE-28X	28	600	21	9-17	0-12	0-4	128 + i2c
PICAXE-40X	40	600	32	9-17	8-20	3-7	128 + i2c

Table 5.1 - Table showing specifications of the various PICAXE chips

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

The PICAXE 18 chip uses an internal comparator to perform a low resolution ADC conversion and can generate 16 discrete values. The 18A chip which is the chip model used in the model uses an 8 bit converter, so can generate 256 values. The 18X chip which is the most advanced version of the 18 pin chips uses a 10 bit converter giving it the ability to read 1024 discrete voltage increments (*PICAXE Manual Section 1* c. 2005, p. 70).

5.4 Construction of the model system

The PICAXE chip used in this model was of the type 18A. The reason this chip was used was because the author was advised that it would be satisfactory for the task. The circuit board used for the testing was a K and H breadboard model RH – 21B. This board was selected as it appeared to be of sufficient size for the task. The chip was mounted to the board as shown in figure 5.2. The chip is the black rectangle near the centre of the image. The white cable leaving the image at the top is the cable used to upload data from the serial port on a computer.

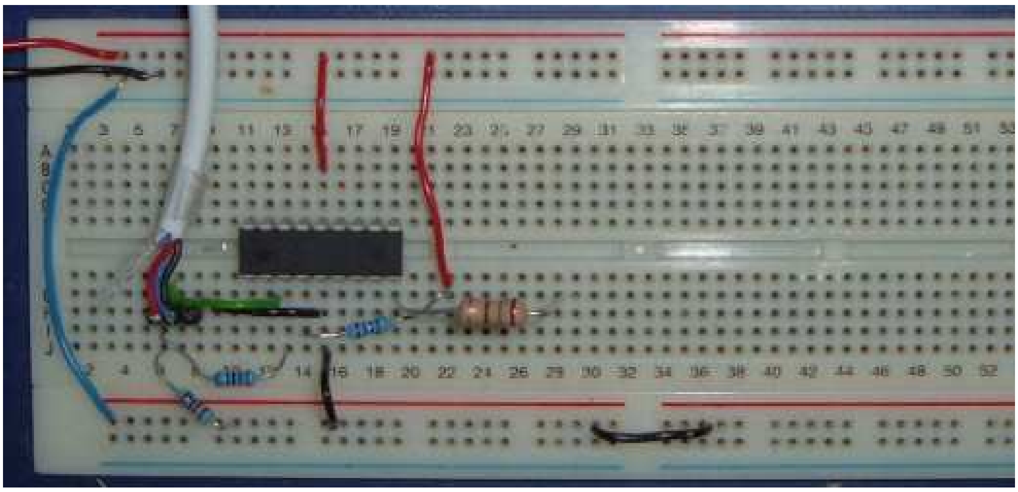
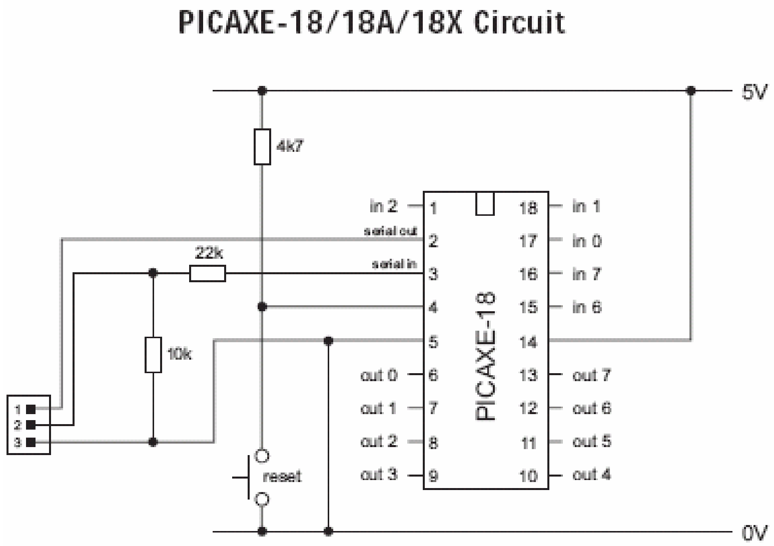


Figure 5.2: Image of the PICAXE chip installed onto the breadboard

The minimum operating circuit shown in figure 5.3 was then added to the board. This simple circuit consists of a 4k7 Ω (4700 ohm), a 22kΩ (22000 ohm), and a 10kΩ resistor. These resistors and several lengths of wire were used to tie the appropriate pins to the positive rail and the ground rail, as well as connecting the serial in and serial out pins to the serial connector. The chip was powered by a rechargeable 4.8 volt battery pack, with the positive and negative wires connected to the positive and earth rails respectively.



<u>Figure 5.3 PICAXE-18/18A/18X minimum operating circuit</u>

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

The serial connector was used along with a serial cord to provide an interface between the PICAXE chip and the serial port on a personal computer. This enabled the software on the PC to download programs into the memory of the chip, hence enabling quick program changes. All of the processes needed to set up the circuit, connect the chip to the PC and download a program were simple and very user friendly due to the simplicity of the chip and the software and the extensive documentation that can be found in the users manual. Once the circuit was set up and connected to the power source and the PC, several light emitting diodes were connected to the output pins and a simple program was written to make them flash. This confirmed that the circuit was constructed correctly and all systems were operational.

The next step was to add two variable resistors. The potentiometers that were used were basic 10 kΩ units with 23mm long input shafts and a range of movement of approximately 300 degrees. A high resistance was required so that during normal operation there was only a very small current passing through the resistor, and therefore the drain on the battery would be minimal. The resistors were connected so that the positive (red) and negative (black) wires were connected to the positive and earth rails that also powered the chip. The signal wire (white) was connected to one of the three pins that support analogue to digital conversion. In this case, pins 0 and 1 were used (see figure 5.3).

When the resistor has a potential difference applied across the positive and ground wires, the potential difference between the signal wire and ground is proportional to the physical position of the shaft around the arc of movement. This is due to another resistor (non variable) built into the body of the potentiometer and wired in series with the variable resistor. It is the varying voltage that results from the interaction between these two resistors that is read into the analogue to digital converter to provide a digital positional reading. In the case of an 8 bit converter, the range of motion of the potentiometer is translated into 256 discrete segments.

Once the two potentiometers were built into the model, a simple program was written to test the operation of the resistors and the chip. Once everything worked correctly, the servomechanism was added to the system. The 18A chip has a system that allows it to drive servomechanisms that have been designed for radio controlled models, which are also known as ‘servos’. It was decided that to simulate the electrically - actuated hydraulic height control system that is recommended for this design, the best device was a servo mounted beside one of the potentiometers and a linkage mechanism connecting the output shaft on the servo to the input shaft on the potentiometer.

Figure 5.4 shows a photograph of the test system in its final incarnation. The PICAXE chip (A) is powered by battery (A). The position signals are sent in to the chip from the two resistors (B) and (D). the output signal from the chip is sent to the servo (C) via a current limiting resistor (G). The servo is powered by battery pack (E) and this rotates the shaft of the upper resistor (B).

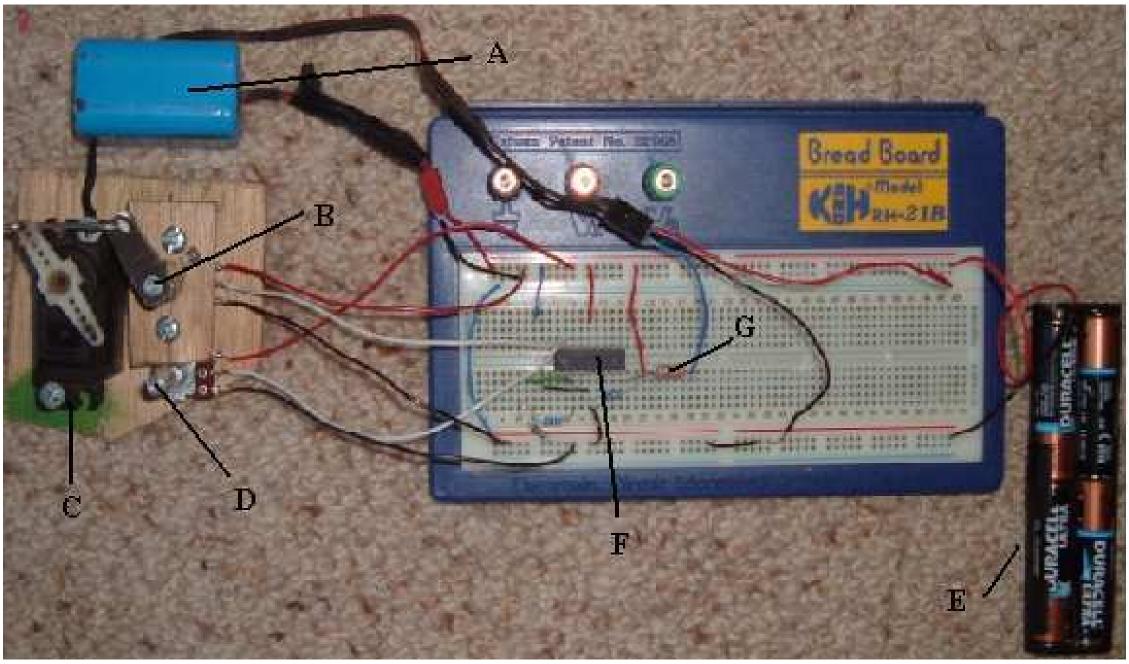


Figure 5.4 Image of the system used to test the operation of the PICAXE chip

When everything is connected correctly, the microcontroller will read in a positional value for each potentiometer, compare the two values, and send a signal to the servomechanism that will turn one of the variable resistors so the physical position of the two resistors match each other to a predetermined level of accuracy, thus achieving the goal of positional control.

5.5 Programming the model

Once the system was ready, the act of programming the chip could start. All programming was done using the customised program editor that came with the PICAXE chip. The chip was programmed using a special version of BASIC (*PICAXE Manual Section 2* c. 2005), so learning the programming language was a matter of reading through the examples provided in the program documentation. A description of the main commands that were used is included next.

5.6 Description of Program_1

The first approach to programming the PICAXE controller was an attempt to minimise problems caused by the oscillations of the height sensing wheels (both programs can be found in the appendicis). The main function of Label_30, which is repeated every cycle of program operation, is to read the positions of each of the variable resistors into one of the available general purpose byte variables available. The data is read into the RAM using the readadc command. This is repeated with an appropriate time interval until 12 of the 14 byte variables contain height data relating to either the front or the back of the implement frame. In the test program which is designed for the model, two variables are left free in order to store data relating to the position of the servo. Once six height readings from each variable resistor have been gathered, the program calculates an average value of the height of each end of the implement frame. This average value is more useful than a single height value as it minimises the effect of sudden deviations from normal caused by field irregularities.

The time interval between the height measurements can be adjusted to suit the required sensitivity and response time of the system. As the height of the implement is only corrected once all of the measurements have been collected, a smaller interval between data collection results in an increase in sensitivity and a decrease in response time to genuine lifting or lowering actions. A field that is relatively flat, such as a stone free irrigation paddock, will allow for a shorter interval between measurements. A very rough or stony paddock will require a longer time interval to average out the effects of deviations, and hence increase the response time to lifting and lowering actions.

After the average value has been calculated, the program then calculates which end of the implement is higher by calculating which resistor has the lowest resistance. This action is performed at the end of Label_30 and uses the if- then command as well as the greater than symbol. The syntax used in this program is as follows:
(x and y are height variables, z is the difference in height)
if x > y then z = x – y
if y > x then z = y – x

This ensures a positive z value at all times.

The reason positive values are required is because the byte variables available in the RAM can only store positive numbers between 0 and 255.

Now that the height difference is known, it is necessary to determine if this difference in height is large enough to warrant a correction. This is done by calculating if the height difference falls outside of a tolerance band. The tolerance band is necessary because otherwise constant corrections would take place even if the implement frame is nearly level. The size and even the existence of a tolerance band in the model system is not critical, as the input variable resistor is adjusted by hand to simulate the vertical movement of the three point linkage system. This method of testing will not be able to accurately test the reaction of the system to small random movements that would be encountered in a rough paddock; it will only test the reaction to large, slow movements that would occur during raising and lowering of the implement. The tolerance will be included in the model to make the results from testing the model as useful as possible.

The tolerance band is specified in label_B5, and the code uses three simple let commands to specify the limits of the tolerance band. In this case, it was determined that the height difference value that did not warrant a height adjustment was ± 2 units of height. This corresponds to 2.67 % of the stroke of the servomechanism, as calculated below.

Stroke of servo = 150 units

Positional tolerance = ± 2 units = 4 units

Tolerance percentage = (4 / 150) * 100 = 2.67 %

This value was decided upon by what seemed to work, although a wide range of tolerances were tried during testing that did not adversely affect the program. In real life operation, the farmer would adjust this tolerance band to arrive at a balance between the number of adjustments in a certain period of time that are performed and how level the implement is kept.

If the two height values are not excessively different, the program performs no height adjustment and skips back to label_30 start the collection of data again. If the height difference falls out of the tolerance range, the program runs either Label_EB to increase the height value of label_F2 to decrease the height value. The amount of adjustment given changes in proportion to the amount of change needed. For example, in this case, the change in height is defined by the variable B3, so as the difference in height increases, B3 increases. This variable is used in Label_EB and label_F2 where it is added or subtracted from the current servo position to reposition the servo. The servomechanism used in the testing rig requires a pulse ranging in length from 0.75 to 2.25 ms every 20 ms. If the length of any of the input pulses is smaller than 0.75 ms or larger than 2.25 ms, the servo will malfunction. To prevent this from occurring, a min and max command were added to the appropriate labels.

It was found with experimentation that an increase or decrease of servo position of the height difference B3 resulted in too much movement. This is because the stroke of the variable resistors is divided into 256 evenly spaced points, but the stroke of the servo consists of only 150 points. Therefore it is not possible to directly compare the height readings from the variable resistors to the position of the servo. To fix this problem, B3 was divided by a constant (C) greater than one in order to enable direct translation of variable B3 to the required positional change of the servomechanism. The optimum value of this constant can be found by dividing the variable resistor travel segments by the servo stroke segments, eg. C = 256/150 = 1.7. It is not possible to use this as a value in the PICAXE program, as the chip can only process integers, so the number 2 will be used for C. When used with the actual system controlling a solenoid valve, this constant would be determined using trial and error by the farmer to suit each set of field conditions, much like the value of the tolerance band. The label_EB and label_F2 code segments use the servo command to drive the servo using the method described in the previous paragraphs.

5.7 Testing of Program_1 and Results

The program was loaded onto the PICAXE microcontroller in a variety of configurations. Once a satisfactory base program was written, it was a simple process to experiment with the program constants. As outlined above, the critical constants were the interval between the height data readings and the constant that the height difference is divided by (called c in this case). The interval between data readings was a major problem in the running of this program. Initially, it was thought that the appropriate interval over which the six height readings for each resistor be gathered should be in the order of one or two seconds. A collection time of two seconds results in a time delay between readings of approximately 330 milliseconds, as can be seen in the delays in the code in Label_30. This would eliminate the effects of most of the small, sharp bumps that would be expected in typical operating conditions. It was found that as the servo position is adjusted every time the program cycles, so if the data was collected over a seemingly reasonable two seconds, there would be at least a two second delay between height corrections. This resulted in a very slow and jerky response to any input. This response is far from the smooth and accurate response required from any real system using this technology.

Figure 5.5 is a graphical representation of the system response to an idealised input. The input and outputs were graphed using Microsoft Excel. The response was calculated using a smooth, constant input comparable to the input created when the implement frame is lifted.

The constants used to calculate the response are:
Time step = 330 milliseconds
$C = 256 / 150 = 1.7 \approx \underline{2}$

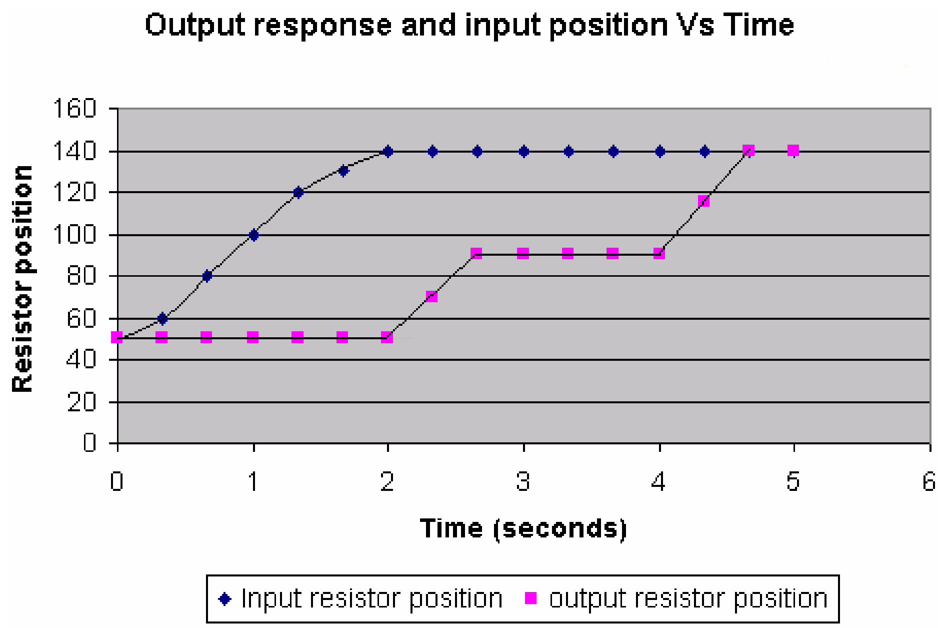


Figure 5.5: Graphical indication of the system response from program_1

Even though figure 5.5 shows an idealised case with no overshoot because the numbers add up to require only two corrections to perfectly reposition the output resistor, a case that would almost never occur in real life, it can be seen that such a program is clearly unsatisfactory. Even if the speed of the servo was increased, it would still take a minimum of 4.5 seconds to correct for the input. This is clearly too long to wait to level the implement after it is lifted or lowered. Based on observations of farmers turning on the end of a pass, where the lifting is done as quickly as possible to allow the turning action to occur, the maximum acceptable time delay between a lifting input to the front of the implement and the rear of the implement correcting to match the new height would be approximately 0.5 seconds.

The only feasible way to make this program run smoothly is to reduce the time interval between height measurements to an order of tens of milliseconds. As the main strength of this program is its ability to ‘average out’ the bumps over a period of time in the order of several seconds, there is no real benefit gained from taking a rolling average if the program is looped every tenth of a second, as the average will not vary significantly from any of the values that the average is calculated from. Based on this conclusion, it was decided to use a single height value from each of the variable resistors instead of the average of six height values. The program was modified using this method, and is described in the next section.

5.8 Description of Program_2

The second approach to programming the controller utilised the processing of only one height variable per resistor every time the program cycles. It was decided that removing the excess data gathering would greatly increase the cyclic speed of the program, resulting in a smoother, faster and more accurate response. Although various configurations were tried, it was found that the most successful way to write the program was to simply replace the repeated data gathering from Program_1 with a single data collection. As there are now no pauses coded into Label_30, there is no need for the farmer to specify the length of such a pause.

It was found that a pause was needed after each adjustment to allow the servo enough time to reposition. The time needed for repositioning to occur is dependant on the distance needed for the servo to travel to reposition the servo. This distance is dictated by the height difference initially measured that triggered that change. It was decided for this reason that the length of the pause would be based on B3, the variable that determined the initial time response. Thus in Label_F2 and Label_EB a pause was included with a specified length (in milliseconds) of B3. It was found that variable B3 was too small a value to trigger a sufficient pause. To solve this problem, a multiplier was used before the pause to increase the value of B3. The variable D is used in the appropriate syntax as a multiplier. The appropriate value of D used in the testing was determined using trial and error.

It was discovered during testing that the position control system would be unstable if the length of the pause after the reposition command was less than approximately 20ms. This is because the servomechanism is positioned with a pulse train where each pulse occurs every 20 ms. If reposition commands are received more often than this, the servo will malfunction. To prevent this problem occurring, a constant E was added to the variable b3 to ensure the length of the pause never fell below E milliseconds.

The length of the pause in the model system will be much shorter than the length of the pause required in a real system, as the response speed of a radio controlled servomechanism is much faster than the response of a large implement. As stated previously, the model varies significantly from the real system, and is only used for proving a concept and testing the limitations of the PICAXE system. As stated previously, the value of each constant will have to be found using trial and error to suit each planting system. Apart from the changes described here, the code of Program_1 is almost identical to the Program_2 code.

The removal of the rolling average means that there is now no way for the program can damp the signal. It is proposed that instead of using software to remove spikes and dips in the signal, all damping should occur before input into the variable resistors. This will require the use of a suitable damper on the frame of the sensing wheel and special attention to the selection of the tyres mounted to the height sensing wheels. The selection of the appropriate damper and tyre is fully explored in the chapter describing the design of the height sensing mechanism.

5.9 Testing of Program_2 and Results

The program was initially uploaded into the PICAXE chip without the variable pause and with a C value of 2. Theoretically this should have provided an almost immediate response with high accuracy and repeatability. It was discovered that this resulted in uncontrollable overshoot and a highly unstable system. A long period of time (anywhere between 2 and 10 seconds) was needed to reposition the output resistor to the correct value, and once the servo did finally centre, it would twitch back and forth uncontrollably.

Further investigation into this problem led to the conclusion that the servo did not have enough time to reposition itself in response to a change in input before the program took the next measurement. Another related source of instability was not allowing enough time between servo reposition commands. The addition of variable pauses with a minimum time of E milliseconds in the method described previously seemed to solve the problems described above.

The values of constants C, D and E determined the response of the test system. The desired response was a smooth, accurate and immediate response with a high level of repeatability. Various values were experimented with to obtain the desired response, and as the PICAXE microprocessor can only process integers, some problems were encountered with finding appropriate values for C and D. However, it was found that when C = 2, D = 3 and E = 30ms, a satisfactory response with the desirable characteristics listed above was delivered by the model.

It is not necessary to provide a graph of input versus output for this case, as when the input resistor is moved at a slow rate of rotation, the servo moves the output resistor with a high degree of accuracy and the response appears to be immediate. Thus a graph of input and output position against time would display very little difference between the two lines. Changing the positional tolerance band has no affect on the running of the program, but as expected, the smaller the tolerance band, the smaller the final difference in height, and the smaller the input needed to provoke a response.

5.10 Relevance of the model

The model created to test the PICAXE system was useful for determining the capabilities and the limitations of the PIC microcontroller; however it was not possible to test the response to every operating condition that may occur in real life. There were several fundamental differences between the model system and the proposed system. The most obvious difference was the use of a radio controlled servomechanism to provide physical motion in the place of a hydraulic pressure system and an electrically actuated solenoid valve.

When this difference was investigated, it was realised that the main limitation of the PICAXE system is that there is no proportional output available. This means that if the output of the microcontroller was used to operate a solenoid actuated valve, there would be no oil flow to the lift assist system while the program determined that the height difference was within a predetermined tolerance band. When the height difference crept above this threshold, the output pin on the microprocessor would be set from ‘low’ to ‘high’. This would throw the solenoid valve to the fully open position, which would direct the maximum possible flow of hydraulic oil to the lift assist system.

This immediate flow of fluid would correct the height of the implement; however the speed of the lifting or lowering action may be so great that overshoot could occur before the microcontroller can return the valve to the centred position and stop the flow of oil. A system that displays instability in this manner is undesirable, and it is not possible to predict this instability with the model that was constructed.

Another unpredictable element in this design is how the electronic components will be able to withstand the extreme operating conditions that are present on a planting implement. Elements such as vibration, heat, sunlight, dust and even an unreliable electricity source can all cause damage to or affect the performance of electronic equipment. These conditions are always present in varying degrees on farm machinery, and it would require a large investment in technology to isolate the electronics from them. For these reasons, it is decided that even though electronics may work flawlessly in a controlled environment, the reliability of the system may be poor if they were mounted to an implement. Therefore it is suggested that a fully mechanical system be used.

CHAPTER 6 - DESIGN OF A HEIGHT SENSING WHEEL

6.1 Design Outline

It is possible to eliminate the electrical system as a method of translating mechanical motion to a solenoid if the electrical system is replaced with a mechanical system that will directly actuate a hydraulic spool valve. It is proposed that this be done to eliminate the many problems that the electrical system presents. The mechanical system will be made in the following way. A height sensing wheel mechanism will be placed at the front and rear of the implement frame. A linkage mechanism will be connected to each wheel mechanism which will translate mechanical movement relating to the vertical movement of the wheel to a common area on the implement.

A device will be designed that will accommodate a hydraulic spool valve and provide the correct motion to both the valve body and the actuation lever. This motion must be such that while the height of the rear sensing wheel is within a specified deviation band of the front sensing wheel, no valve actuation is allowed. When the height of the rear wheel deviates from this band, such as when a lifting action is takes place, the action of the linkages should actuate the valve which will allow oil to flow into the hydraulic rams of the lift assist system. This change of oil volume will correct the height of the rear of the implement back to within the specified height tolerance, thus levelling the implement. To make the device more useful, the levelling action will occur right throughout the height range of the implement.

6.2 Design constraints

There are many issues that need to be addressed if this device is to work effectively. For correct height control, it would be possible to use a different height sensing wheel on each end of the implement, however this would complicate the design of the linkage mechanism to the point where it would be unfeasible. It is proposed that the same sensing mechanism be used on the front and back. This would help with the design of the linkages and will also halve the number of different parts that need to be made to produce the control wheel mechanisms if they are exactly the same.

The entire mechanism will be designed to clamp onto the frame rails of the implement with the use of U-bolts. This means that the different parts of the device can be positioned to suit the layout of the frame and also to avoid clashing with other bolt-on devices such as tines and seed distribution equipment such as distributor heads. To help with possible space constraints, the entire device will be made to be compact. It is not necessary to fully pursue the reduction in size, as the layout of most bolt-on implement devices, especially tines, can be changed to suit the height control device.

This system will be designed to be flexible enough to fit as many implement frames as possible. As everything will be attached with U-bolts, the layout is highly flexible and could be fitted to a wide variety of frames as long as the manufacturer provides linkage arms that are customised to suit the customer’s implement frame geometry. This is because longer implements will require longer linkages than shorter implements. It may be possible to design linkages that are telescopic, so that one device can be adapted to fit any implement.

6.3 Height control wheels

6.3.1 Wheel positioning

It has been determined that the same height sensing mechanism will be used on both the front and the rear of the implement. It is desirable to have the rear wheel trailing behind the implement for simplicity and to keep it out from underneath the frame where it may clash with the tines which sometimes swing backwards to clear obstacles. The front wheel presents a different problem due to the frame geometry. If the front height control wheel trails behind the front frame rail, it will operate underneath the frame. Even though this is the most desirable layout due to simplicity, there is a chance that it will clash with tines and even the frame itself.

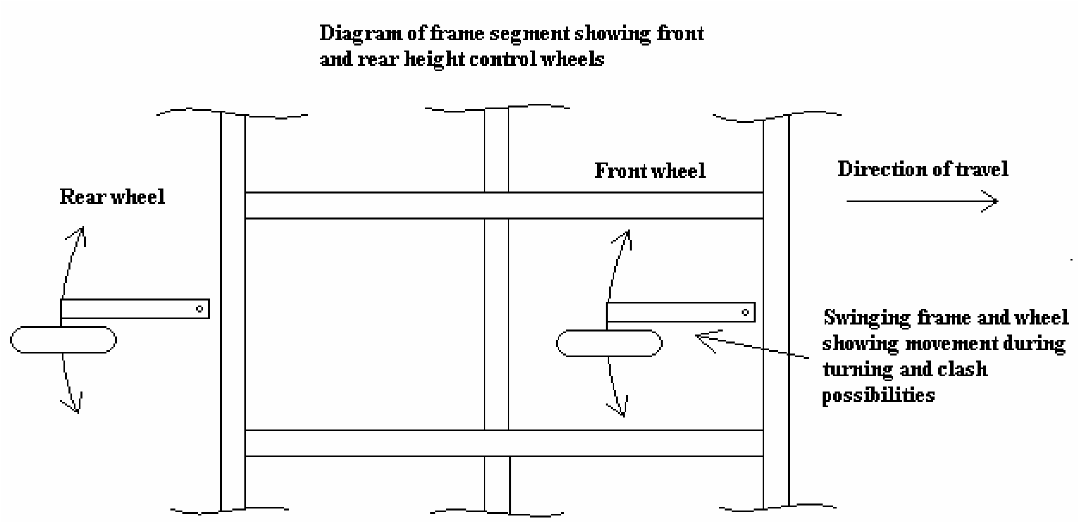


Figure 6.1: Image depicting the disadvantage of a trailing front wheel

This problem is compounded by the fact that the trailing arm will have to swing through an arc to allow the wheel to turn, thus increasing the chance that a clash may occur. Figure 6.1 illustrates this possibility with a diagram of such a system. Note that almost the entire area in the square section in which the front control wheel is situated now becomes almost unusable due to the possible clashes between the wheel and the tine. Another issue to address is lateral positioning of the system. To reduce clearance problems with the three point linkage and the lift assist, it is proposed that the wheels be mounted to one side of tractor as close to the rear wheel as possible. This will reduce the error resulting from tractor and implement roll.

To eliminate problems relating to the positioning of the wheel under the frame, it is proposed that the rear wheel will trail, as seen in Figure 6.1, and the front wheel will protrude from the front of the implement, facing the opposite direction to the rear wheel frame. This situation is illustrated in Figure 6.2.

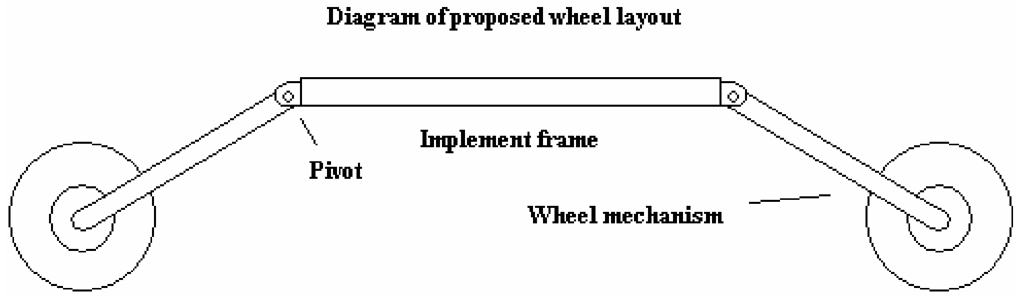


Figure 6.2: Image depicting desired wheel layout

It is immediately clear that this layout will prevent the front wheel from having the ability to castor. This ability is necessary because if the wheels could not castor, they would skid sideways in a turning manoeuvre, stressing the frame and bearings and wearing out the tyre. For a wheel to castor properly, the pivot must be situated ahead of the axle so the wheel will trail behind this pivot and continually face the correct direction. Therefore the frame will be designed so that a linkage mechanism extends a pivot in front of the implement that a separate frame that the wheel is attached to can use as a castor pivot. Figure 6.3 illustrates this scenario.

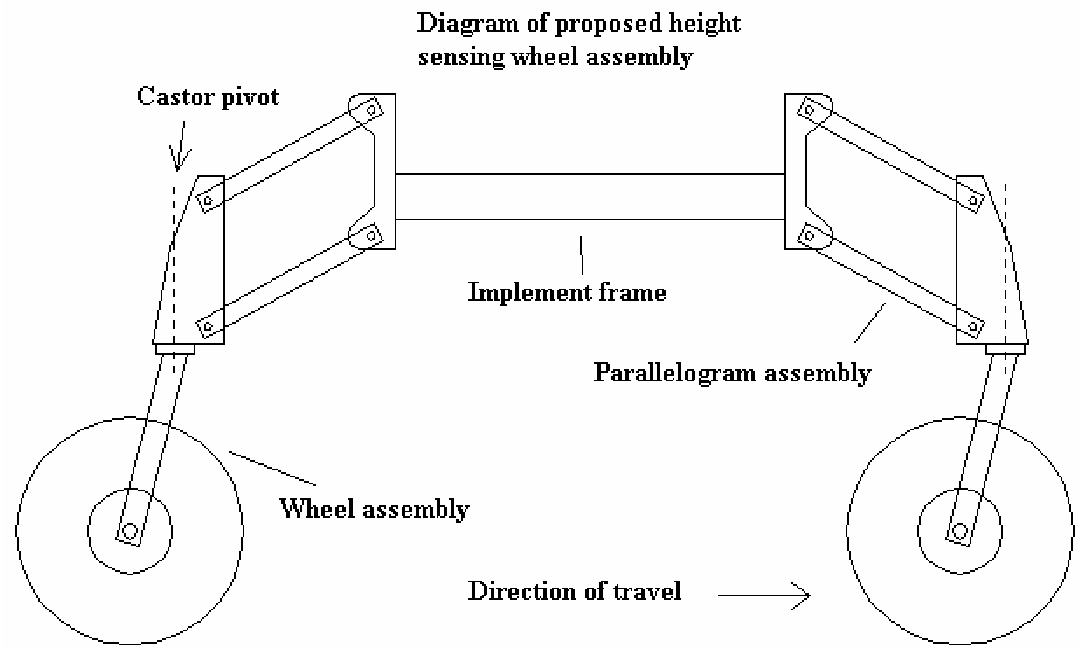


Figure 6.3: Image depicting the final design concept of the wheel frames

This diagram indicates how the vertical movement of the wheels is allowed to occur through the movement of the parallelogram assembly, and the angled axle brace allows the wheels to trail behind the castor pivot regardless of the direction of travel and the positioning of the mechanism. Note also that this arrangement allows the same components to be used twice, as physically there is no difference between the front and rear control wheels. This helps to keep the cost of production to a minimum and inherent symmetry simplifies the design of the control linkages.

6.4 Design of height sensing wheel

6.4.1 Materials selection

Three main requirements of the material used in this device are high strength, high fatigue resistance and low cost. Other more minor requirements include ease of manufacture and high availability. High strength is required to deal with the often extreme loads that are placed upon agricultural equipment. Fatigue resistance is needed to deal with the cyclic loads and vibration that result from travelling across rough ground at moderate speeds. Cost is a major requirement as agricultural machines must be heavily overbuilt to cope with extreme conditions, meaning that much more material is used in the manufacture of the equipment than is needed during normal operation. (Bullock, B. K. 2006, Pers comm. 17 August 2006)

Ease of manufacture is important because complex shapes must be able to be made from the material with a minimum of manpower and machines to reduce costs. High availability is important as much material will be needed to manufacture a large number of these devices, and this will need to be sourced rapidly and effectively. These combined requirements preclude the use of all but one material, which is steel. Steel is the metallic element iron (Fe) that has had the alloying element carbon added during production. Steel is suitable as it is strong, has a high fatigue resistance and can be easily cut to shape with a variety of methods. These shapes can be joined together to form strong parts using inexpensive welding methods.

Due to the abundance of iron in the form of minable iron ore and carbon in the form of coking coal, steel is one of the cheapest and readily available manufacturing materials available, therefore the parts can be overbuilt without incurring a significant added cost. Almost all agricultural machinery is constructed almost totally from steel due to these reasons (Bullock, B. K. 2006, Pers comm. 17 August 2006). Steel can be purchased in a variety of different alloy types ranging in yield strength from 284 MPa (AISI 1015) to 688 MPa (AISI 8650) (Juvinall & Marsheck 1999, page 846). As a general rule, the higher the AISI number, the more carbon is present in the alloy, the higher the strength and the costlier the steel. The most suitable steel for the proposed frame components is AISI 1020, which has a yield strength of approximately 350MPa (Juvinall and Marsheck 1999, page 846). The designation 350 is used when specifying this type of steel in sheet form. This steel type will be used for all of the structural components.

6.4.2 Manufacturing method

The wheel mechanism will be designed to fulfil all of the requirements of a height sensing wheel while using a minimum amount of material, while still being easy to manufacture and using sections of readily obtained steel. To make complex shapes, appropriate shapes will be cut from steel plate using either hot gas (usually oxygen/fuel mixture) or laser cutting methods, depending on availability and cost. These shapes will then be welded together, again using the most readily available welding method. Holes that will accommodate shafts will be initially cut with the gas/ laser method and then drilled or reamed out to size to obtain a smooth surface finish and sufficient dimensional accuracy.

Extra support will be given to shafts by welding concentric collars to the plate surface which will increase the surface area of the loaded internal surface, reducing any stress concentrations. These collars will also be used to secure the shafts by drilling small holes through the centre of them at right angles to the shaft axis and corresponding holes through the shafts. Small bolts can then be inserted once the shafts are fitted which after they are fitted with nuts and tightened will secure the shafts in place.

The shafts will be turned from AISI 1020 steel rod of an initial diameter slightly larger than the final diameter. The collars will be turned from the same grade of steel tubing of the correct initial diameter. It is necessary to turn the inside of the tubing to ensure the correct internal diameter and to add chamfers to the sharp edges. Wherever a shaft is being used as a pivot, it will be secured to the steel plates that surround it with a bolt. The bearing surface will be a piece of steel tubing that is welded to the piece that is rotating with respect to the pin. This tubing will have an internal diameter that is slightly larger than the pin diameter and a large enough wall thickness to allow for significant wear to occur before replacement is necessary and also to allow a grease nipple to be threaded into the surface to provide lubrication.

The bearing tube will be of sufficient length to provide a large surface are so bearing pressures are not excessive, keeping wear to a minimum. The proposed pivot system is illustrated in figure 6.4.

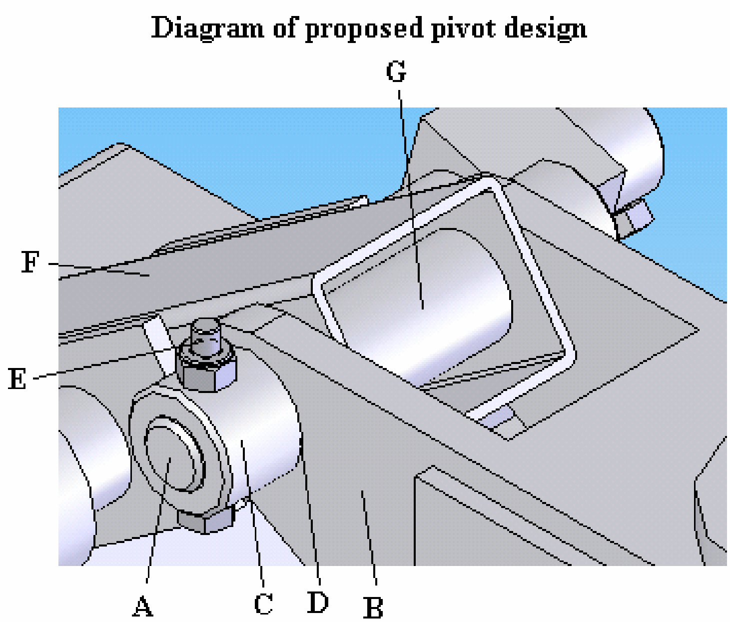


Figure 6.4: Diagram of proposed pivot design

In this diagram, it can be seen that the pivot pin (A) is secured to collar (C) with bolt (E) which is welded to part (B) with a weld bead around join (D). Part (F) is welded to bearing tube (G) which rotates relative to the pin. This is a simple way of making a long lasting pivot that is easy to assemble and disassemble. Note that a grease nipple would be fitted to the centre of the bearing tube but this has been omitted to simplify the model. Five of these types of pivots are needed on each control wheel mechanism, four small ones for the parallelogram mechanism and one large one for the wheel castor pivot.

6.5 Parallelogram Mechanism

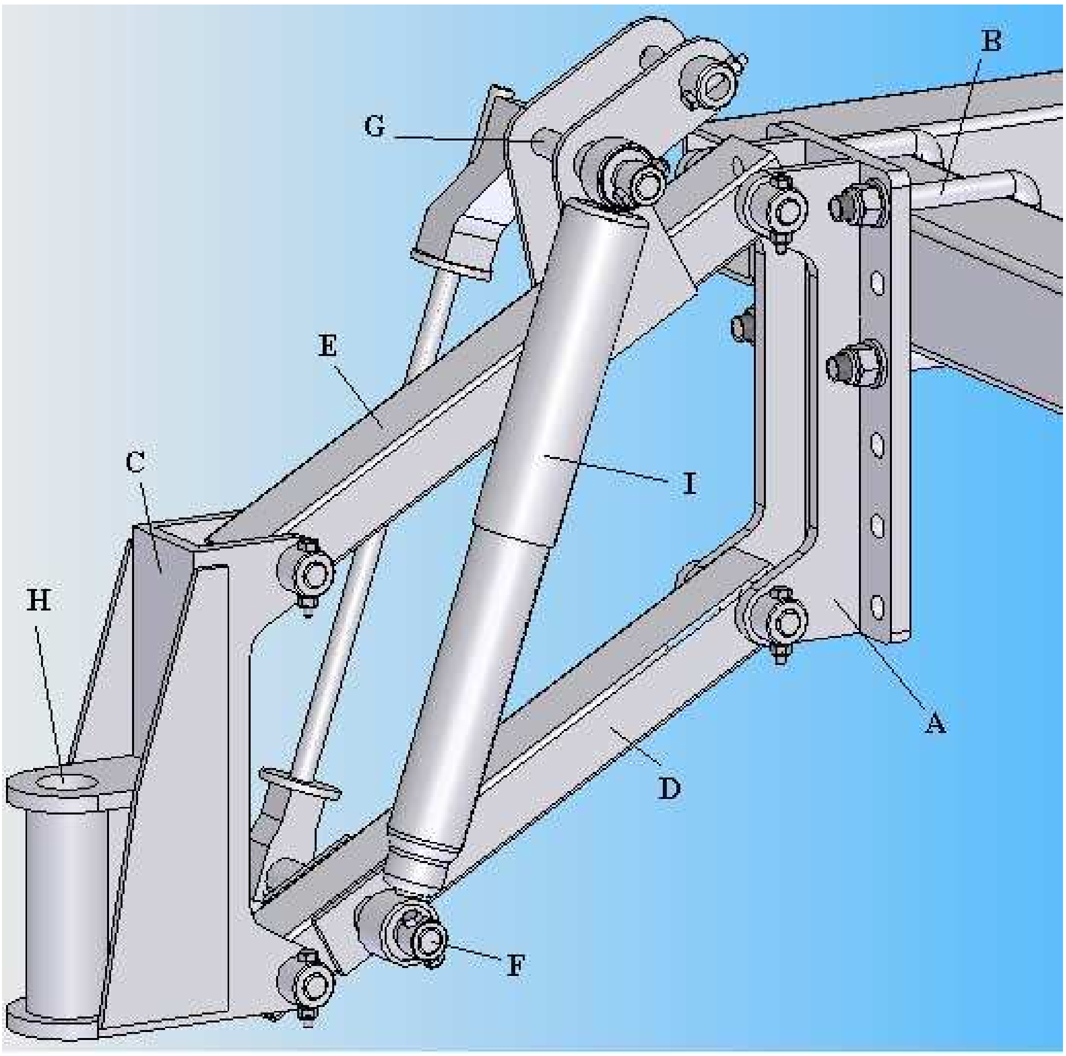


Figure 6.5: Image showing the final parallelogram design

The parallelogram mechanism was built to combine many functions in the same unit. Figure 6.5 shows an image of the solid model of the mechanism. Backing plate (A) acts as an interface between the mechanism and the implement frame. This is secured to the frame with the U-bolts (B). Note that there are four different height settings that are available from the twelve holes in the backing plate. This allows the height sensing mechanism to be adapted to suit the height of most implements. The frame in the image is made of simple 4” square rolled hollow section (RHS), although adapter blocks will be supplied with the device to fit these U-bolts to a variety of different frame sizes. The height setting shown shows the correct setting to suit a (lowered) frame height of 800mm, as may be found on large trailing implements. The large height travel range of the mechanism coupled with the four height settings ensures that this device can be fitted to most three point linkage and trailing implements if this is necessary.

The lower link (D) and upper link (E) are made of 50x63mm RHS with a 3mm wall thickness. Note that these dimensions correspond to 2 x 2½", which is a common RHS size. Both are fitted at each end with greasable bearing tubes as described previously that are welded onto the walls of the section. Four of these make up the pivoting points for the mechanism. The upper and lower links are both fitted with 20mm shafts (F and G) that protrude from the links and provide mounting points for the shock absorber (I) and spring mechanism. Item C forms the fourth segment of the parallelogram, and is the part that holds the castor frame which in turn holds the wheel. The castor pivot (H) is made of another greasable bearing tube that is sized to fit a 30mm shaft. Full detail of all of these parts arcan be found in the appendices.

The shock absorber is fitted to the frame to damp excess motion resulting from rough terrain. If this was not done, there would be almost constant activation of the spool valve in even the smoothest terrain, resulting in erratic height control and excess wear of the spool valve and linkages. The shock absorber specified for this application is a unit from the rear of a Nissan Patrol Four wheel drive, part number IA30. This unit was designed for a heavy vehicle, so offers a high amount of energy absorption capacity, as well as a long life and reliability. The stroke of this unit is measured at 230mm, which if the geometry is set correctly is enough for this application.

The geometry of the final design of the mechanism means that the greatest damping force occurs at the working height where it is most needed, and almost no damping force is present when the frame is fully extended and damping is not as critical. This method of design means that the damping force can be adjusted by changing the height setting of the U-bolts that secure the backing plate to the frame. Using the upper bolt holes will increase the damping force by raising the wheel, whereas using the lower bolt holes will reduce the damping force by effectively lowering the wheel.

A view of the other side of the model is shown in figure 6.6. This shows the mechanism that mounts a spring to the mechanism. This spring is necessary to provide force to push the wheel down to provide an accurate contour following ability. The other job that the spring must perform is to force the wheel to travel downwards fast enough so that even in a rapid lifting phase, the wheel remains in contact with the ground. This means that the spring must have enough force to extend the shock absorber at the correct speed to keep the wheel touching the ground.

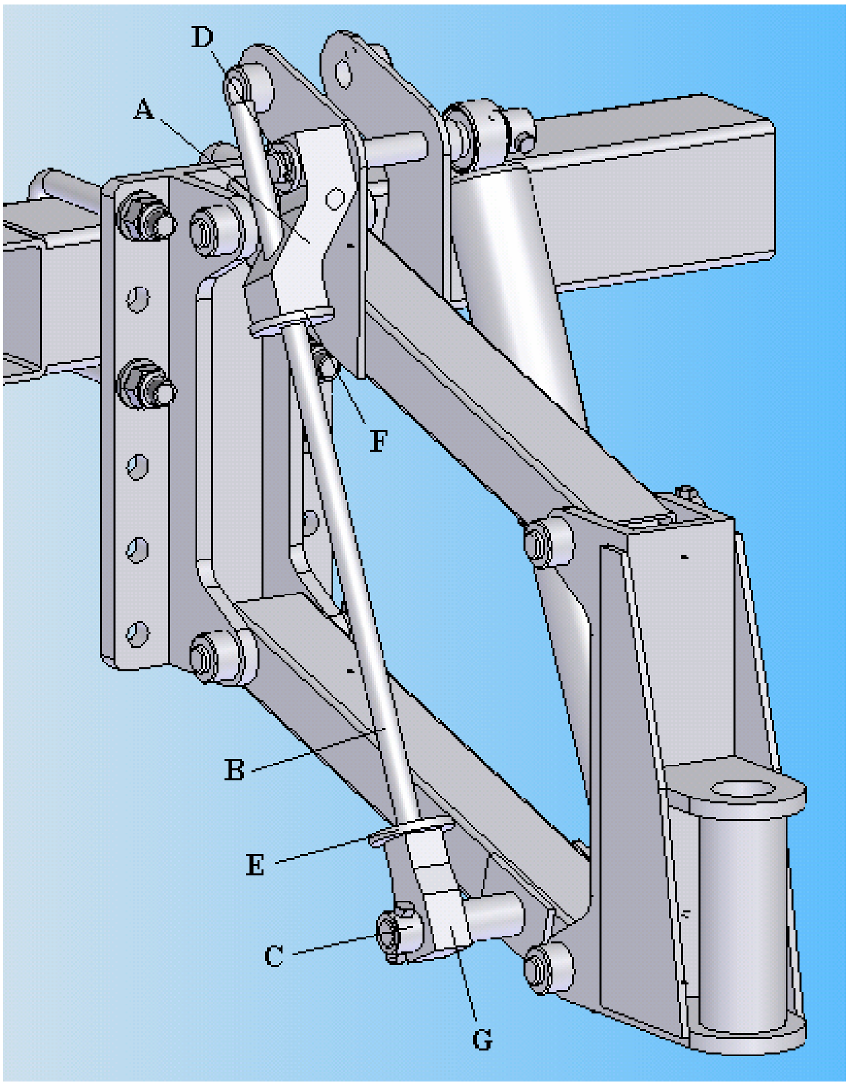


Figure 6.6: Secondary image of the final parallelogram design

The spring holder works by providing two plates (F and E) that press on either end of the spring and contain it. These plates move relative to each other as the frame moves vertically, keeping the spring compressed and providing downward force to the wheel. The plates are aligned with a shaft B that is connected to boss assembly G and slides relative to alignment boss A. The sliding shaft is also located coaxially inside the coils of the spring, therefore containing it between the two plates. The alignment boss is held on the upper 20mm shaft by a split pin and a washer (D) instead of a collar and bolt (C). This is because the sliding shaft is too close to the pivot shaft to allow a collar and bolt to be used.

6.6 Spring Design

As the spring holder and the shock absorber share the same mounting points top and bottom, the extension force on the shock absorber during lifting of the implement is equal to the compression force from the spring minus the force that the spring is exerting on the ground through the tyre. The faster the lifting action, the more of the spring's force is going into extending the shock absorber. Therefore the spring should be able to exert enough force to extend the shock absorber at a fast enough rate to keep up with a rapid lift and provide enough additional force to keep the tyre in contact with the ground at the same time.

The shock absorber unit that was obtained for modelling purposes was worn out, so the compression damping force that was measured was not representative of a new unit. The extension damping force felt a lot stronger however, and was enough to provide a rough guide to the expansion damping factor of a new unit. This is the critical damping factor that would govern the design of the spring. To get a feel for the damping factor, an extension force of an estimated 20kg was applied to the shock absorber by hand and the extension rate was estimated. The testing procedure was only an approximation because as mentioned previously the test unit was worn out and could only provide a rough guide anyway.

This test indicated that a force of 20kg would result in an extension speed of approximately 20mm/second. It is estimated by visual observation that the maximum expected lift rate of a three point linkage implement is approximately 100mm/second. Note that this device is not being customised to fit one tractor; it is designed to be universally used on a wide variety of tractor/implement combinations. Thus the aim of this design process is to come up with a maximum feasible lift rate figure that could be encountered and design a spring that will be able to extend the shock absorber at the correct rate and at the same time supply sufficient force to press the tyre into the ground. Thus based on the observations of the operation of three point linkage systems, it is decided to design the spring to lower the wheel at a rate of 200mm/s. This speed is far greater than any lifting rate that may be encountered, so will be sufficient to keep the wheel pressed into the ground at all times.

At working height which is defined as the height of the implement that forms the parallelogram into a rectangle (the links are horizontal) it is necessary to find the ratio of change in spring extension to the change in wheel height. This will indicate the speed ratio between the rate of spring extension to the rate at which the wheel changes height. This information is necessary to find how fast the damper should be extended to provide a vertical tyre movement of 200mm/s.

Using the measuring tool in Solidworks, it was found that at working height, a vertical tyre movement of 35.5mm resulted in a spring/damper extension of 13mm. this gives a speed ratio of ≈0.366. This indicates that a vertical tyre speed of 200mm/s requires a spring extension speed of approximately 70mm/s. If 10kg of force gives a velocity of 10mm/s, then this indicates a required spring force of 70kg. To allow for error and a new shock absorber, the spring will be designed to have a working compression force of 100 kg. When the mechanism is fully extended, the spring will be expected to develop a force of less than half of this to continue to apply some force to the tyre. An estimate of the required force is 30kg. This value means the free length of the spring will be slightly longer than the longest working length of the spring, simplifying the design.

The spring will be designed so the solid length will be met before the shock absorber reaches the end of its stroke to protect the shock absorber from crushing. The figure shown at the left of Figure 6.7 shows an image of the solid model of the parallelogram mechanism positioned at this setting. From measurements taken from the model, this indicates a solid height of 204mm.

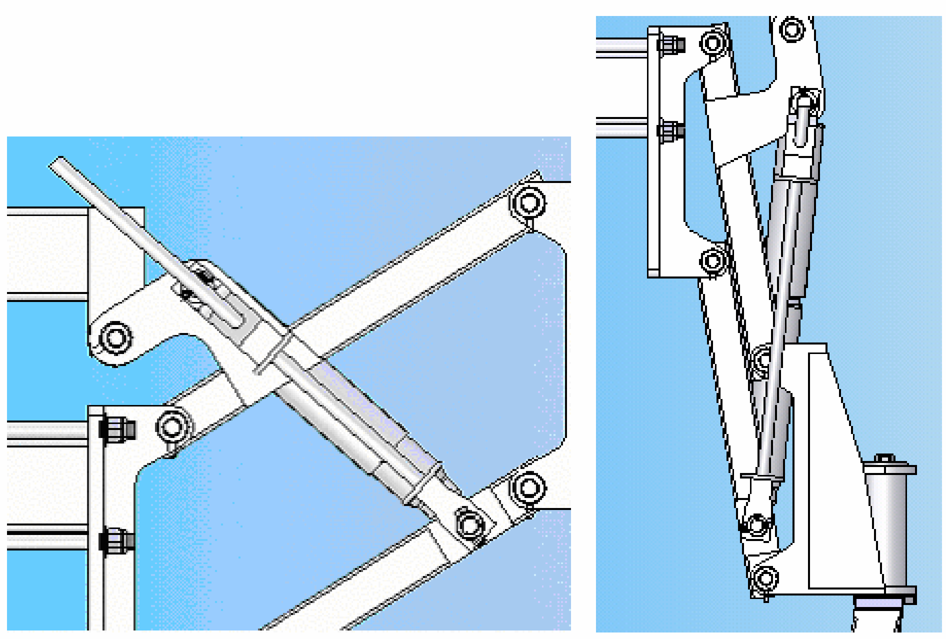


Figure 6.7: Image of the parallelogram mechanism at two height extremes

Similar measurements show a fully extended spring height of 413mm. The figure at the right of Figure 6.7 shows the same mechanism in the fully extended state. Thus the spring parameters are now known so an appropriate helical compression spring can be designed.. The spring design process found in Juvinall and Marsheck (1999) will be followed

Spring length (minimum load) = 413mm

Solid length = 204mm

Spring length (maximum load) = 10% longer than solid length

$$= 204 + (204 \times 0.1)$$

$$= 224.4 \text{ mm}$$

Minimum load = 294 N

Maximum load = 981 N

Spring rate = change in load / change in distance

= (981-294)/(413-224.4)

= 3.64 N/mm or 3642.6 N/m

Free length = (minimum load / k) + Spring length (minimum load)

= (294 / 3.64) + 413

= 493mm

Force required to fully compress the spring (F_s) =

(Free length – spring length (solid)) × k

= (493 - 204) × 3.64

= 1052 N

The diameter of the spring as measured at the spring wire axis is the variable that will be found first. This will be done by starting with the known variables F_s, spring rate k, number of turns N and shear modulus G. Values for spring diameter D and wire diameter d will be estimated from the start.

$G = E / (2(1 + v))$ (Juvinall and Marsheck 1999, p. 94)

Where $E = 207\times10^9$ Pa

And v = 0.3 for steel (The spring will be made of cold drawn steel wire)

$G = 207\times10^9 / (2 \times (1 + 0.3))$

$= 7.961\times10^{10}$

$k = (d^4G) / (8D^3N)$ (Juvinall and Marsheck 1999, p. 495)

The wire diameter d has been found to be 0.0064m. This value has already been calculated using iteration of the following method.

The value of N (the number of turns of the wire) will be calculated from this data using Juvinall and Marsheck (1999), p. 500 which states that for a helical compression spring with squared and ground ends (suitable for nesting against flat surfaces)

Solid length $L_s = N.d$

$N = 204 / 6.4$

$N = 31.875$ turns

Now the spring diameter D can be found from equation 12.8.

$k = (d^4G) / (8D^3N)$

$3642 = (0.0064^4 \times 7.961 \times 10^{10}) / (8 \times D^3 \times 31.875)$

$D = 0.0518$m

or 51.8mm

Now the necessary ultimate tensile strength of this wire will be found to provide final verification of the spring design. The shear stress experienced when the spring is compressed solid (τ_{solid}) should be limited to less than $0.45S_u$ (for ferrous spring material without presetting), where S_u is the ultimate tensile stress of the spring wire (Juvinall and Marsheck 1999, p. 499). This is necessary to limit long term set to 2%, which is desirable to preserve the performance of the spring. This value is obtained from Juvinall and Marsheck (1999), p. 494.

$\tau_{solid} = ((8 \times F_s \times D) / (\pi \times d^3)) / K_s$

K_s is found using Juvinall and Marsheck (1999), p. 494

where $C = D/d = 51.8 / 6.4 \approx 8$ which is in the preferred range for springs with ground ends. This gives a K_s value of ≈ 1.05.

$\tau_{solid} = ((8 \times 1052 \times 51.8) / (\pi \times 6.4^3)) \times 1.05$

$= 555$ MPa

$S_u = \tau_{solid}/0.45$

$= 1235$ MPa

This is approximately 1.5% below the ultimate tensile strength of 6.4mm spring wire as given by Juvinall and Marsheck (1999), p. 497.

Thus the spring will be made with the following parameters:

$D = 51.8$ mm

$d = 6.4$ mm

$N = 31.875$

$L_{free} = 493mm$

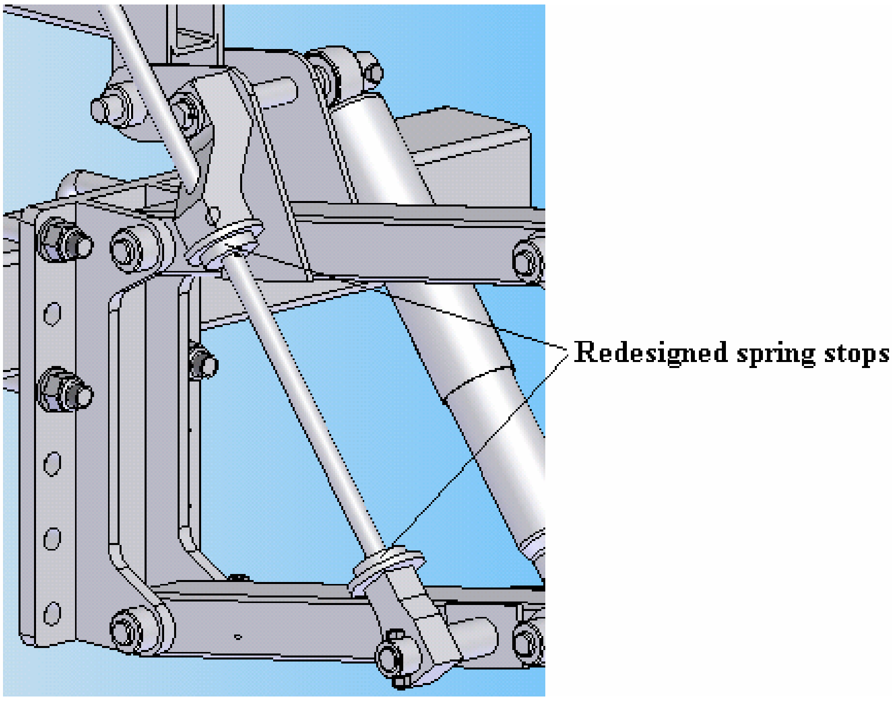


Figure 6.8: Image showing the redesigned spring stops

The spring stops must be designed to support this type of spring. **Figure 6.8** illustrates the changes that were made within the model. Note that the end faces of the spring sit on the face of the larger diameter plate and the inner circumference of the spring is supported by the smaller turned boss.

6.7 Design of castor frame

The castor frame is the device that attaches to the parallelogram assembly through a shaft (A) that allows it to pivot on a vertical axis. Two images of the solid model are shown in figure 6.9. Attached to the lower part of the frame is the shaft (D) that supports the wheel hub. This shaft is horizontal and is shaped to allow easy installation of the hub and bearings. To enable the height sensing mechanism to be used on both the front and the back of the implement, the castor frame must be able to castor through 360 degrees. This is achieved by bending the upper support (F) to provide clearance and carefully designing the pivot mechanism. The castor frame is secured with a bolt through threaded hole (G) which will be explained later.

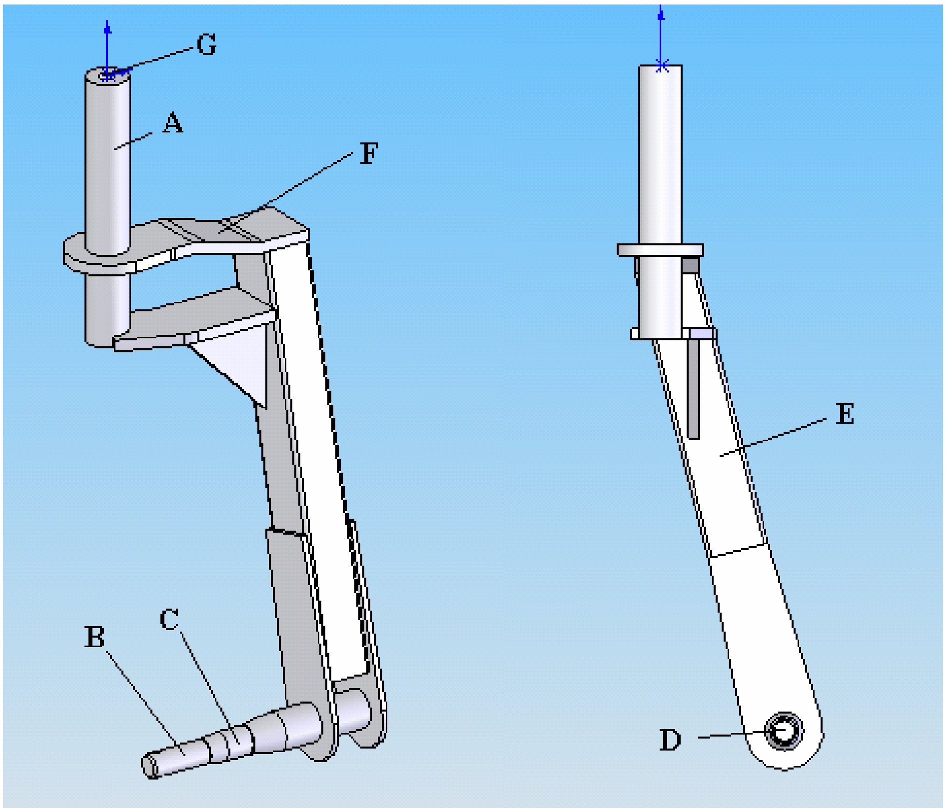


Figure 6.9: Two images of the castor frame

The wheel will always trail behind the pivot by angling the main spar (E) backwards so the axis of the axle is behind the axis of the pivot. When the hub is installed onto the shaft, the inner bearing seats onto the surface (C) and the outer bearing seats onto surface (B). The shaft is shaped like this so that the inner bearing can be installed without first having to be hammered over surface (B). Juvinall and Marsheck (1999), p. 601 indicates a variety of bearing sizes and bearing numbers. A bearing with basic bearing number 206 was selected for the inner shaft diameter. This bearing has a bore of 30mm, an outer diameter of 62mm and a width of 16mm. A 205 bearing was chosen for the outer shaft diameter as it has a bore of 25mm, an outer diameter of 52mm and a width of 15mm. the shaft and hub were designed accordingly. Juvinall and Marsheck (1999), p. 604 indicates that for a 90×10^6

revolution life with 90% reliability, the 25mm light duty bearing must be exposed to a constant load of less than 3650N. The 30mm bearing can be exposed to up to 5400N to obtain the same life. These load figures are not expected to ever be reached in operation due to the small loads that will be applied by the spring. Each bearing may be expected to have upper load spikes of no more than 500N in even the most extreme conditions (note that the wheel load is shared between the two bearings). No further analysis will be conducted upon these bearings.

6.8 Hub design and wheel selection

The most appropriate tyre for this application is an ATV tyre. These tyres have a large rolling diameter, a wide contact pattern and a very thick profile. The large contact area and low loads they experience mean that they are only inflated to very low pressures. They are designed to be used in an off-road situation and are therefore inherently tough, durable and puncture resistant, making them suitable for this application. The reason a thick profile and low inflation pressures are desirable is because these traits give the tyre the ability to deform heavily as they travel at moderate speeds over rough, stony ground. Where a high pressure tyre such as a car tyre would deflect upward against the damper whenever a bump was encountered, the soft ATV tyre would simply deform and absorb the bump with the absolute minimum of wheel deflection. This is important as any undesirable vertical movement of the parallelogram could trip the spool valve and result in poor height control and unnecessary wear of the mechanism.

The tyre that was selected for this application was a MAXXIS 25×10-12 NHS tyre. This Tyre has a 25” external diameter, is 10” wide and fits a 12” diameter ATV rim. An image of this tyre fitted to a rim is seen in figure 6.10. There is no specified rim for this application as there is an abundance of makes and models of 12” ATV rims and the manufacturer of this device will make a selection based on price and availability at the time of production. This tyre was selected as it has a particularly large external diameter, therefore will ride over rough ground quite smoothly.

Figure 6.10: Image of the tyre that will be used

The hub is to be machined on a lathe from steel. The stud pattern will depend on the exact rim selected, so no dimensions are given here. The internal surfaces must be machined to provide a press fit for the outer bearing races. Two images of the solid model of the hub are shown in figure 6.11.

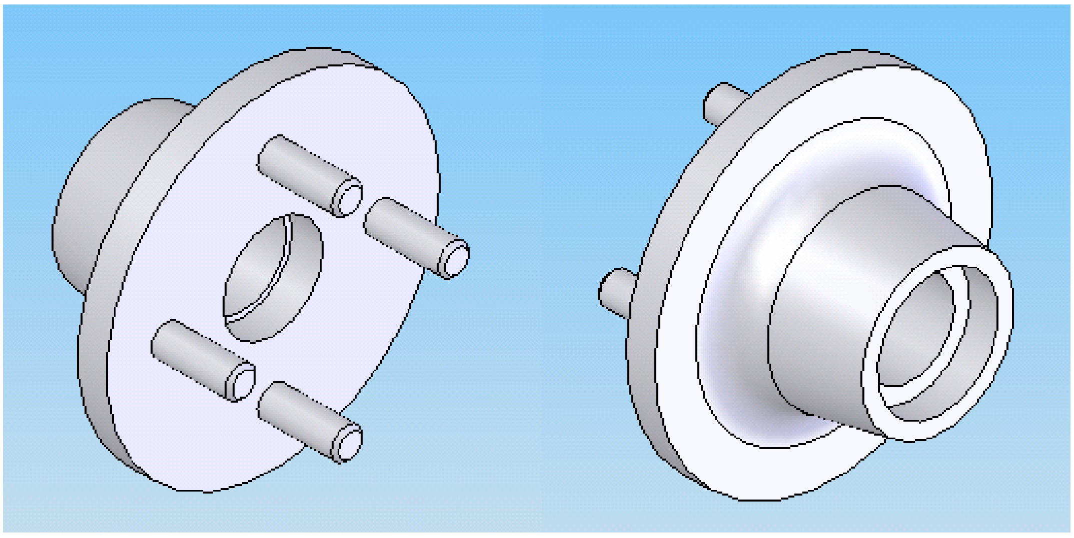


Figure 6.11: Image of the hub model

The image on the left shows the face that the wheel will seat on and a typical four stud pattern. These studs are threaded into holes that have been tapped into the hub. The smaller bearing fits into the machined internal surface and is inserted until it touches the ledge seen inside the hole. The image on the right shows the other side of the hub and the machined circumference that the larger bearing must fit into. The fit of the bearings into their respective holes must be of the correct tolerance to be useful. Juvinall and Marsheck (1999), p. 880 shows a table of various fits that are used in manufacturing. The most appropriate fit for a bearing given the nature of the application is a tight fit.

The force that is required to fit the bearing will depend on the exact dimension of each individual bearing housing. The most appropriate nature of the bearing fit would be one which requires light hammering to insert it. Juvinall and Marsheck (1999), p. 880 shows that for a tight fit, the hole diameter tolerance = $0.0052\times(\sqrt[3]{d})$ where d is the nominal hole size. It will be assumed that the tolerances of the outer bearing races are negligible. The nominal hole diameter for the smaller bearing is 25mm. This means that the size of the hole can be calculated.

Smaller hole diameter tolerance = $0.0052\times(\sqrt[3]{25})$

$= 0.00152$

This means that the hole should be $25\pm(0.00152/2)$

or 25 ± 0.0076mm

Larger hole diameter tolerance = $0.0052\times(\sqrt[3]{30})$

$= 30\pm(0.00161/2)$

or 30 ± 0.0081mm

An image of the entire wheel assembly is shown in figure 6.12. This image shows the device mounted to a sample 4” implement frame. The castor assembly is positioned the way it would be if the device was mounted to the front of the implement. The height setting of the frame is typical of the operating height of the frame assembly when the implement has been lowered. The wheel bearings have not been modelled as they are off the shelf items. Note that the shock absorber and spring holder mechanism are sitting about midway through the stroke in this operating position which is a desirable situation as this gives the wheel the ability to deflect up and over obstacles if this is necessary.

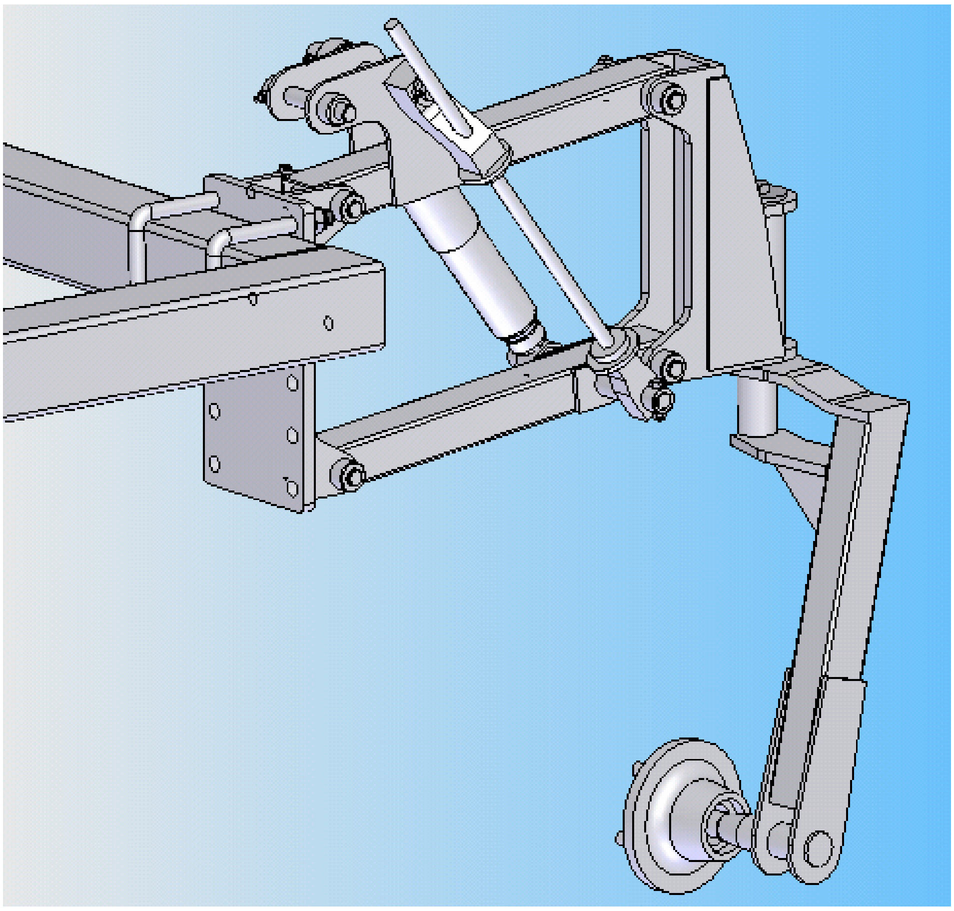


Figure 6.12: Image of the final mechanism model

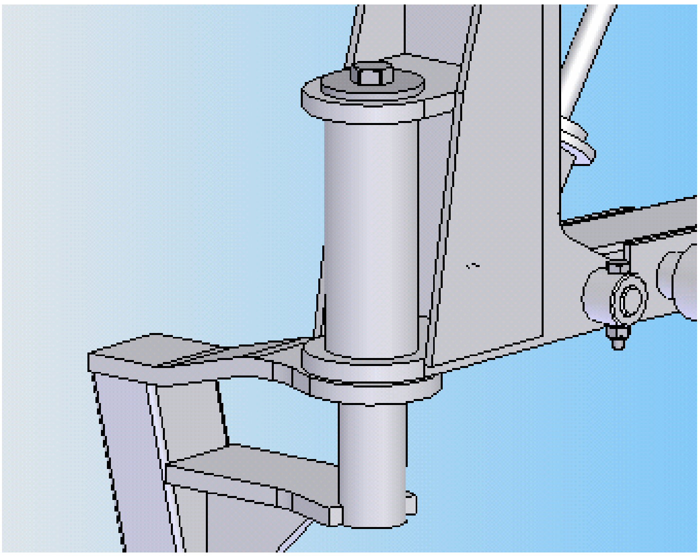


Figure 6.13: Close up image of the castor frame pivot

Figure 6.13 has been included to show how the castor frame is secured to the parallelogram assembly with one M16 × 50mm bolt and a Ø65mm, 5mm thick washer.

6.9 Spool valve control

The proposed mechanical spool valve activation will require careful design of the linkage system to ensure that the frame remains level at all height settings. The spool valve that will be used will be a heavy duty, single spool and lever operated “open” valve. Open spool valves are different to closed valves in that when they are in the neutral position, oil is allowed to pass freely through them and be recirculated through the exhaust lines and back to the oil reservoir. This means that pressure port on the lift assist valve can be fed a constant supply of oil from one of the hydraulic remotes from the tractor. When no height correcting oil flow to the lift assist is needed, the bypass circuit build into the open valve lets oil cycle through the system at low pressure and return to the oil reservoir.

Figure 6.14: Image of a hydraulic ram

Figure 6.14 shows a photograph of a hydraulic ram used in an agricultural application and indicates how lines connect to each end of the ram. Each of these lines connects to a spool valve which controls the flow of oil to and from each of these lines. When oil flows into one end of the ram, it pushes the piston away from this end and oil flows out of the other. It is necessary for the valve to allow this exhaust oil to flow away from the ram as oil is pumped into it for the ram to function. Commercial spool valves have this capability engineered into them, and an off-the-shelf item will be used in this application.

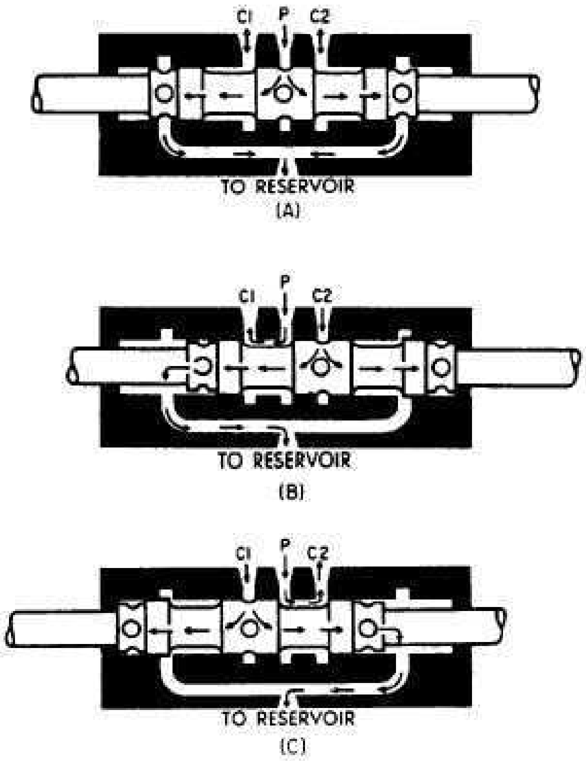


Figure 6.15: Diagram of the operation of an open spool valve

http://www.tpub.com/content/engine/14105/css/14105_108.htm

Figure 6.15 shows a schematic of an open spool valve. Port P represents the pressure input from the tractor to the valve. Points C1 and C2 illustrate the ports that connect the two hydraulic ram lines to the valve. Image (A) shows the neutral position where no oil flow to the rams are required. Oil always flows in from the tractor through P, regardless of valve position. In this case, no oil flows to or from the ram through C1 and C2. The oil simply flows into and then through the centre of the control rod, through the ports at the ends of the valve and collects back at the exhaust port to go to the reservoir. There is no change in ram position.

Image (B) shows a schematic of the valve when the control rod has been moved to the right. In this case, oil flows into the pressure port and is diverted to C1 through the oil passage created by the movement of the control rod, hence the piston moves away from the side of the ram that is supplied by port C1. Oil is allowed to return from the ram to C2 and is routed through the centre of the control rod and then to the exhaust port to be returned to the reservoir. Image (C) shows the same situation where the control rod has been moved to the left. In this case, oil flows to the ram through port C2, and oil is allowed to return to the exhaust port from C1. The effect of this reversed flow is to move the ram in the opposite direction to that which would have occurred if the control rod was moved to the right as in image (B).

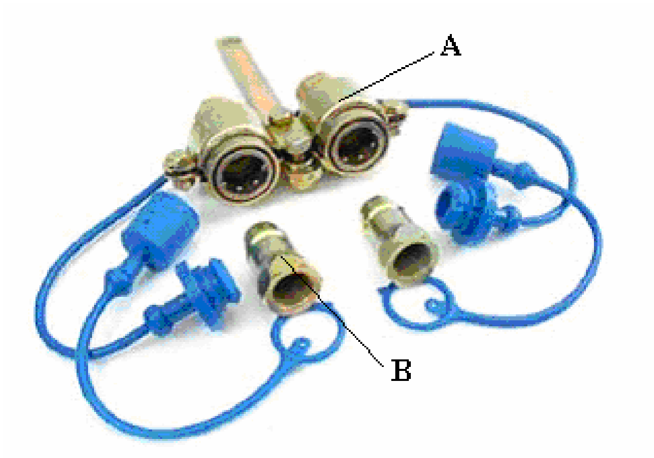


Figure 6.16: Image of a quick hitch coupling

http://www.bareco.com.au/files/hydraul2001/hy9.htm

If a closed valve was used, there would be no oil flow through the system unless a height correction was needed. This absence of oil flow means that the pressure relief valve on the tractor hydraulic pump would be constantly activated, causing rapid wear and failure of the pump and valve system. The oil flow to the valve will be controlled by the lever in the cab that activates one of the tractor remotes.

An image of one of these quick-hitch remotes is shown in Figure 6.16. In this case, the body of the mechanism (A) is bolted onto the back of the tractor. Each of the two connectors that make up this body can be used as a pressure or exhaust line, depending on the position of the lever that controls the spool valve that these lines are connected to. The other part of the quick-hitch mechanism (B) can be connected a hose and quickly joined to the connectors to form a high pressure connection.

Most modern tractors have a feature that allows one of the remote levers to be locked forward to supply a constant flow of oil to the implement, usually to run hydraulic motors and devices similar to that being designed here. These devices mean that modern tractors have the ability to supply this device with a constant flow of oil.

6.10 Spool valve selection

Selected for this application is a Hydreco V37D114A single spool open valve that is manufactured by David Brown Hydraulics. An image of a similar valve is shown in figure 6.17. This valve is a heavy duty valve designed for oil flows of up to 266 l/min and built to safely handle oil pressures of 2500 lbs/in^2, or 172 Bar. (*Hydreco mobile control valves* c. 2005)

Figure 6.17: Image of a spool valve similar to the one being specified here

Source: http://www.hydreco.com/New/downloads/V37_V38_MVT150.pdf

The weight of the valve is given as approximately 15kg. The documentation supplied by Hydreco gives technical drawings indicating the shape of the valve and all of the dimensions required to produce a valve mount. This information is sufficient to design a bed to mount the valve to and a lever system to move the control rod as well as giving a visual indication of the position of the oil inlet and outlet passages. From this, the system can be designed.

Chapter 7 - Control system design

7.1 Design outline

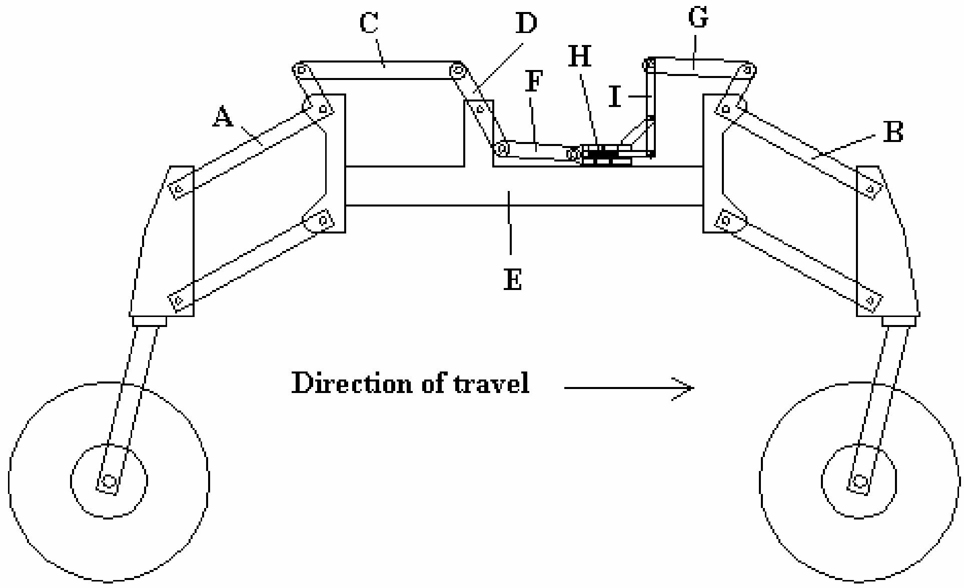


Figure 7.1: Conceptual image of the valve control mechanism design

The control system will use a system of linkages and pivots to control the position of a sliding bed that the spool valve will be bolted to. Figure 7.1 shows a conceptual schematic of the proposal, where the frame is represented by (E). The upper linkage arm of the rear parallelogram is fitted with a bar that juts out at right angles from the body. The rearward end of linkage (C) connects to the pivot that is fitted to the end of this bar. The forward end of linkage (C) pivots on the end of crank (D). At the centre of this crank there is another pivot and the crank rotates about this pivot. The lower end of the crank is connected to the spool valve via another link (F). The spool valve can slide forward and backwards via a sliding deck. The nature of this motion is that the position of the spool valve is directly related to the vertical height of the rear wheel.

The front parallelogram mechanism has a similar pivot that juts out from the top of the upper linkage. The forward end of link (G) is fitted to this pivot, and the other end is fitted to the pivot at the top of the spool valve lever (I). This lever is connected to the control rod of the spool valve, so any ‘rocking’ motion of this lever will slide the control rod of the spool valve relative to the body of the valve, activating the valve and directing oil flow to the lift assist mechanism.

If the device has been set up properly, the geometric arrangement of the linkages will ensure that the spool valve lever will remain vertical and hence no flow of oil will occur if the wheels are at the same height. If the three point linkage system is activated to lift the front of the implement, there will be a height position discrepancy between the front and rear mechanism, i.e. the front wheel will be lower (with respect to the frame) than the rear.

This motion will tilt the valve lever toward the front, initiating a flow of oil to the lift assist system. As the overall height of the frame with respect to the ground increases, the vertical motion of the rear wheel will slide the spool valve body toward the front of the implement. This will have the effect of tilting the valve lever backwards. When the three point linkage system ceases to raise the front of the implement, the rear of the implement will rise until the control rod reaches a position where no more oil flows to the lift assist system. This situation will occur when the implement is nearly level with respect to the ground. The implement cannot be held perfectly level as there will be a small movement of the valves control rod before any oil flow occurs.

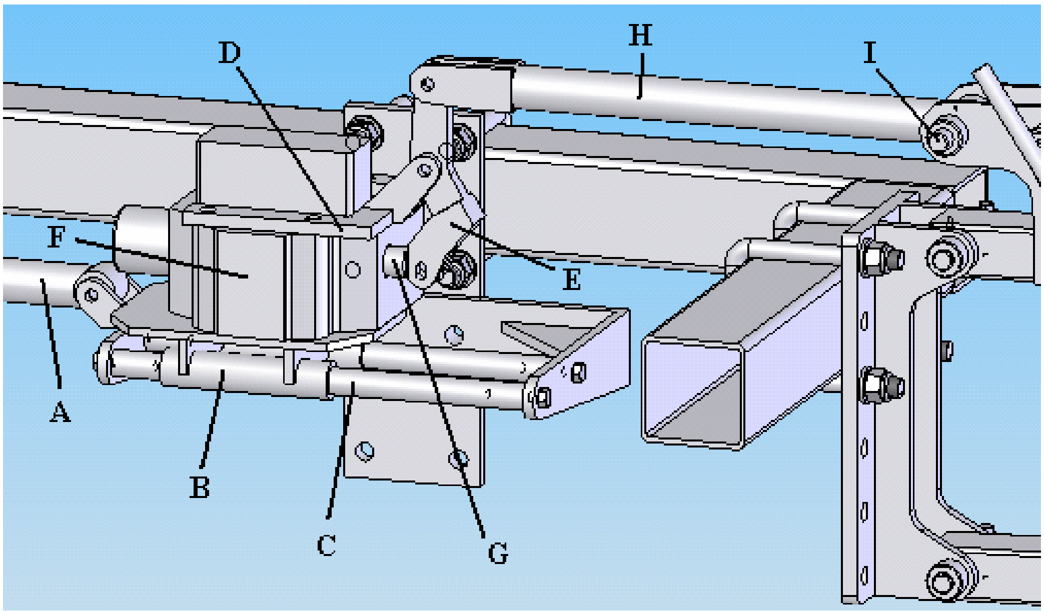


Figure 7.2: Close up view of the valve and valve slide assembly

A small section of implement frame was modelled to provide a visualisation of how the different parts would connect to the frame. The technical drawings supplied by Hydreco provided sufficient information to model a basic representation of the spool valve to provide clarification of the design and fit. Figure 7.2 shows a view of this spool valve (F) sitting on the sliding deck (B). This deck has holes that match the hole pattern of the spool valve to provide a method of bolting it onto the deck.

The sliding deck can move relative to the slider frame rods (C) by a sliding action between the sliding tubes and the rods. The back of the sliding deck is connected to linkage arm (A) which is connected to the rest of the system (not shown). Bolted to the upper surface of the valve body is a frame (D) that provides a mounting point for the valve lever (E). Movement of this lever moves the valve control rod (G) by pivoting about this frame. The lever is operated by linkage arm (H) which connects to the top arm of the front parallelogram assembly (I).

Figure 7.3 shows a similar image of the rear part of the mechanism. The top link of the rear parallelogram (A) moves connecting link (B) as the wheel moves vertically. This link is connected to the crank (C) which pivots on rod (D). The other end of this crank is connected to link (E) which is connected to the sliding deck.

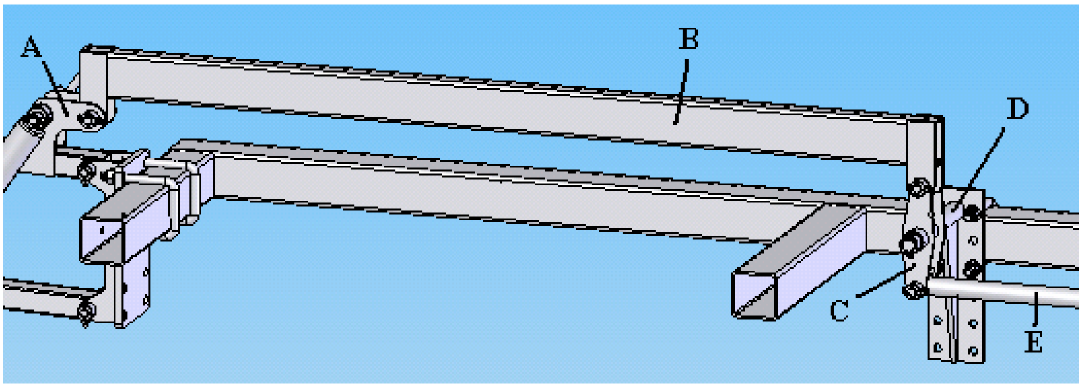


Figure 7.3: Image of the rear linkage mechanism design

7.2 Design criteria

The linkage system is designed with the same greased bearing design as was featured in the height control parallelogram system. The two bolt on components (the slider frame and the crank pivot) are provided with the same system of connection through the 12 holes in the backing plate and U-bolts as the height control wheels. These holes are spaced to provide the correct height for the two devices as long as the same holes are used on all of the four attachments.

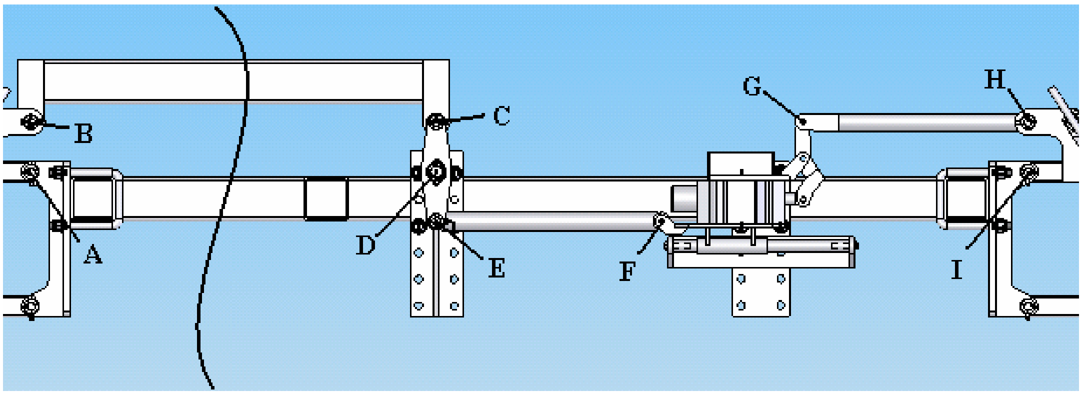


Figure 7.4: Image depicting all of the relevant linkage pivots

For the height control system to provide an acceptable level of accuracy, the geometry of the linkages must be made to a particular set of rules. Figure 7.4 shows a shortened version of the test frame model with all of the relevant pivots illustrated alphabetically. The distances between the links are constrained to the following rules. Note that GH, for example, is the distance between links G and H and GF (vertical) for example is the vertical height separation between links F and G.

AB = CD = DE = IH

CE = GF (vertical) = (0.5 AB)

GH = EF

BC (vertical) = EF (vertical) = GH (vertical) = 0

7.3 Design review

Several design features shown have not been explained as yet. It will be noted that there is a crank located almost half way between the two wheel assemblies. The rear linkage is connected to the top pivot of this crank. There is another link connected to the bottom of this crank, which then connects to the pivot on the valve slider. This crank is essential as it reverses the motion of the rear link so that when the rear link is moving backwards when the frame is rising for example, the middle link is forcing the valve slider forward. This is essential for the correct operation of the valve.

Another design feature is the location of the spool valve. This has been placed as far forward in the frame as practical. This allows the hydraulic lines from the tractor to be shorter than they would be if the valve was near the back of the implement. The hydraulic circuit design is an important aspect of the design; however it is a simple system that does not require a great deal of explanation. The pressure and exhaust ports on the spool valve need to be connected to the remotes on the tractor through flexible hydraulic hose and quick hitch couplings.

The other two ports on the valve are to be connected to the lift assist hydraulic system through the same type of hydraulic hose. The correct polarity of the lines should be observed, so oil flows in the correct direction to achieve the desired height correction. The type of hoses used will be decided by the company that produces the implement, and will depend on factors such as pressure, flow rate and cost.

The device has been designed to be as compact as possible; however it may still not fit all implements. To remedy this problem, the design should be made highly flexible with many different configurations available from the manufacturer and the ability to custom fit certain parts to fit different machines. As mentioned in chapter 6, one modification that could greatly increase the adaptability of the design is the use of telescopic linkages that can be locked to any length.

The four separate devices are fitted to the implement frame with the use of eight U-bolts bent from Ø16mm steel rod and threaded at each end with a M16 x 2mm thread. These bolts are designed to be fitted to 100mm square RHS, however they could be used to secure the device to other frames with the use of the appropriate cast iron adaptor blocks.

It is desired that the ground following wheels remain in contact with the soil surface at all times. The range of heights that the parallelogram must be able to move the wheel through to allow this to happen is dependant on the height range that the implement frame experiences between working height and lifting height. If the current design is used, there is approximately 700mm of vertical wheel travel available. If the device is to be fitted to a implement that has a larger vertical travel range than this, longer parallelogram linkage arms should be designed and used.

7.4 Construction

The pins that are used to secure the linkages at B, C, E and H (Figure 7.4) are the same as the ones used to secure the parallelogram assembly. These are secured with the same size bolt and nut. The crank pivot is secured with a collar and a bolt. Links F and G and the two other pivots that the valve lever pivot on are secured by the appropriate size bolt and nut, as the light loads these parts experience do not make a pin necessary.

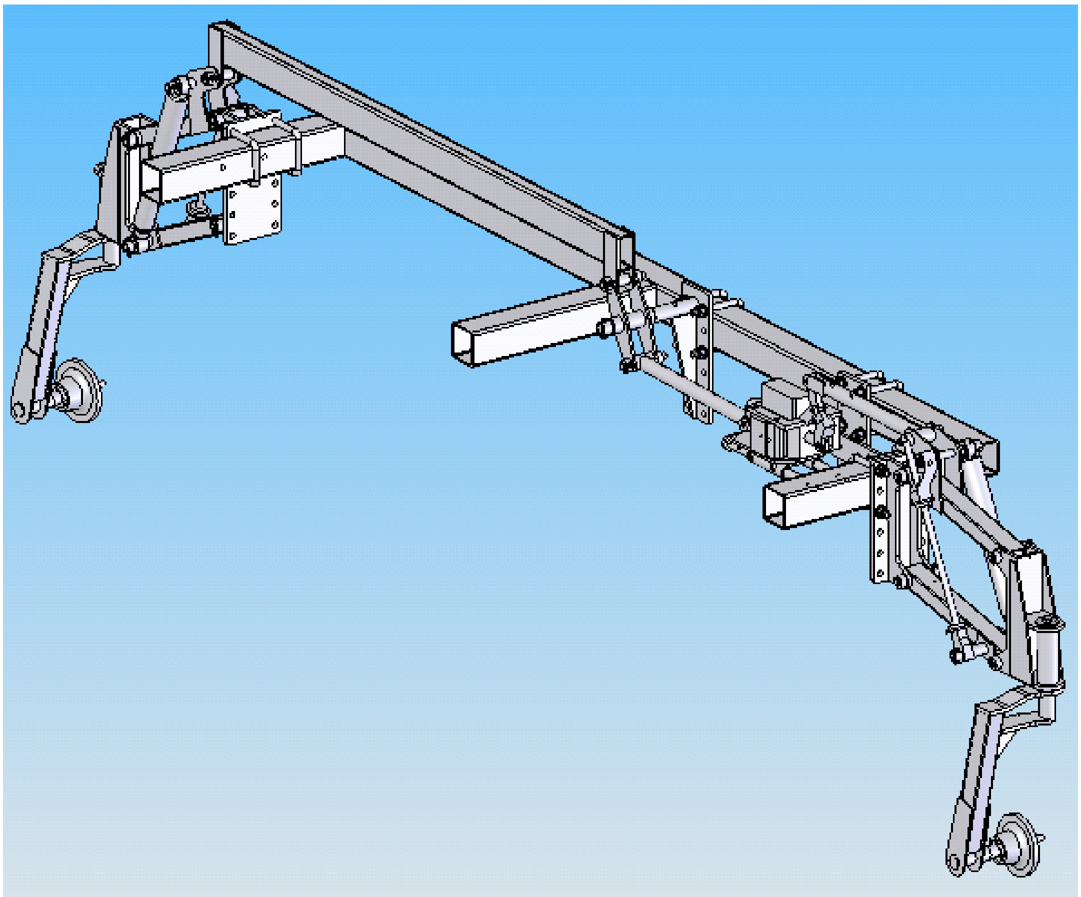


Figure 7.5: Image showing the final mechanism assembly

The technical drawings for all of the parts and the assembly drawings are shown in the appendix. These drawings provide sufficient information for a tradesman to weld the different metal pieces together to make that particular part. The dimensions that are needed to make each piece are not shown; these parts will automatically be created with the cutting device from drawing files loaded into the computer. The raw materials needed to make the pieces that make up each part are tabulated in the drawing files. A list of fasteners required for the assembly are detailed in table 7.1.

Fastener	Use/s	Number required
M16 × 50mm bolt	Securing castor frame	2
M8 × 50mm bolt	Securing pivot pins/ securing lower valve lever pivot	14
M8 Nyloc nut	Securing pivot pins and shock absorber/ spring mounts/ lower valve lever pivot	20
M8 × 40mm bolt	Securing shock absorber/ spring mounts	6
4mm split pin	Securing top of spring mount	2
M16mm Nyloc nut	Securing U-bolts	16
M10 × 40mm bolt	Securing slider bars	4
M12 × 50mm bolt	Securing upper two pivots on valve lever	2
M12 × 80mm bolt	Securing pivot F **(figure 7.4)**	1
M12 Nyloc nut	Securing upper valve pivots/ pivot F	3
M14 × 130mm bolt	Securing valve body	4
M14 Nyloc nut	Securing valve body	4

Table 7.1: Table of required fasteners

Other important information that is needed to assemble the device is details of the collars needed. Table 7.2 detail the type and number of collars needed. Figure 7.6 indicates the critical dimensions. R = bolt hole radius, ID = internal diameter, OD = outside diameter and L= length. The bolt hole is placed in the centre of the length.

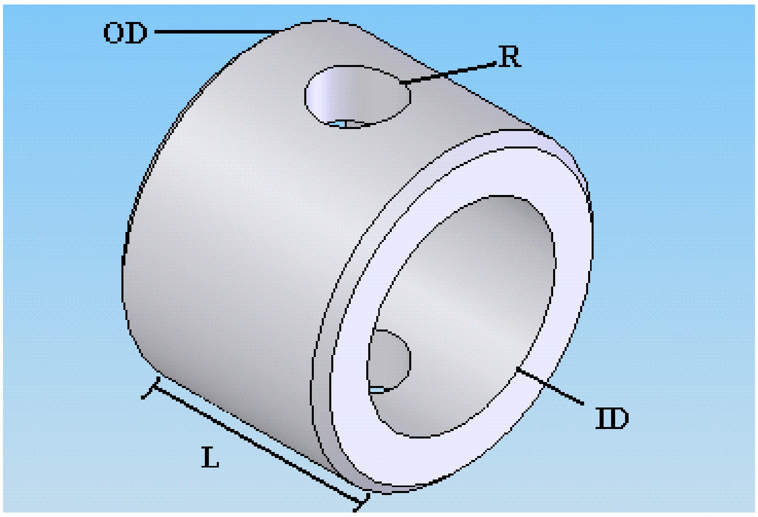


Figure 7.6: Image showing the critical collar dimensions for table 7.2

Collar type				
OD	ID	L	R	Number required
30mm	20mm	22mm	4.25mm	6
40mm	30mm	28mm	4.25	1

Table 7.2: Table detailing the required collars

The level of design detail that was discussed in this report could have been far greater, for example the thread depths and bolt grades were not discussed, as were the washer sizes that were needed. This level of detail would have unnecessarily increased the length of the report to describe things that the tradesmen and farmers that assemble the device would already know from experience, and these details are not critical anyway. The aim of this report was to develop and design a system to control the height of a planting implement, and this has been done and thoroughly reported about. Due to this, no more design detail will be discussed here.

CHAPTER 8 – DESIGN VERIFICATION

8.1 Analysis of the main control linkage

8.1.1 Vibration analysis

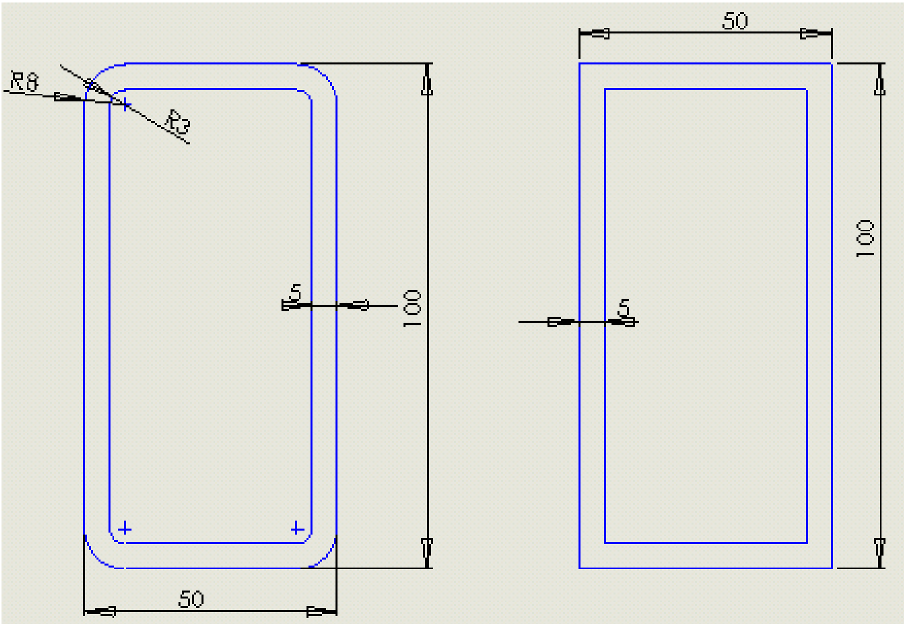


Figure 8.1: Images of the real and simplified main linkage cross section

Shown on the left side of figure 8.1 is a dimensioned cross section of the longer control arm that connects the rear height sensing parallelogram to the pivot near the front of the implement. As this beam is pivoted at both extremities and will be the longest unsupported control arm used, it is necessary to determine the natural frequency of the beam.

As the implement has no real suspension, even when connected to the tractor through the three point linkage, the implement frame undergoes many modes of vibration due to the rough surface of the field. Therefore it is important to know whether in rough conditions any damaging modes of vibration will be experienced in the beam. A small amount of vibration will cause the spool valve to trip unnecessarily due to the changing geometry of the beam, resulting in poor height control. Large amounts of vibration will result in this coupled with the added risk that the beam will fail through fatigue.

The implement frame will only experience serious vibration on the vertical axis due to rough ground. This assumption is based on personal observations while operating a large planting machine. Thus the calculations will only be done in relation to the vertical axis. In this case, the height of the beam is 0.1m and the width is 0.05m. The image show above on the left shows the actual cross section of the beam. An idealised cross section of the beam without any curving geometry is shown on the right. The second moment of inertia (I) will be calculated manually from this cross section, as it is quite easy to do this manually and it will be a good approximation to the actual value of the cross section. Note that the wall thickness of the beam = 0.005m.

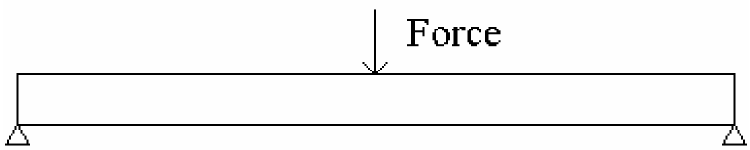
Iyy = Iyy (outer rectangle) – Iyy (inner rectangle)
Iyy = $bh^3/12$ (From Juvinall and Marsheck, page 834)
Iyy = $(0.05\times0.1^3)/12 - (0.04\times0.09^3)/12$
Iyy = 1.736×10^{-6} m^4

Now the spring constant of the beam can be found. Balachandran and Magrad (2004), p. 31 shows a table of spring constants for common elastic elements. Item 5 shows a case of a beam supported at both ends with hinged pins with a mass located at a point along the beam. The restraint scenario is correct in this case, however the load on the beam is not a point load applied at a particular point, it is an evenly distributed load caused by the weight of the beam. Therefore this distributed load needs to be converted into a point load in the middle of the beam.

The easiest and most accurate way to do this is to assume that one third of the weight of the beam is resting on a point in the middle of the beam that has no mass. This is accurate because the weight of the outer two thirds of the beam is close to the supporting pins and will have little effect on the centre of the beam where weight is the most important in this case. This loading scenario will give a result accurate enough for the purpose of this report, which is to evaluate the viability of this system.

To test this scenario, the mass of the beam must be found. This mass is dependant on the length of the beam, and this length is dependant on the length of the frame that the system is being mounted on. For simplicity, the length of the beam in the model will be tested, and if necessary, thicker beams cross section will be recommended for larger machines.

Using the mass properties tool in Solidworks, it was found that the volume of a beam of this cross section and length of 1900mm was 0.00283605 cubic meters. The density of typical carbon steel as given in Juvinall and Marsheck (1999), p. 840 as approximately 7700 kg/m^3 and the Modulus of Elasticity is stated as 207GPa (this is needed in later calculations). Thus the weight of the beam is calculated at 21.83 kg and the point force placed in the middle of this beam is calculated at 7.3 kg.



The loading scenario is shown in the above diagram. The spring constant equation relating to this scenario is given in Balachandran and Magrad (2004), p. 31, and is defined as

$k = (3\times E\times I\times L)/(0.5L)^4$

Where E = Modulus of elasticity (207×10^9 Pa)

I = Second moment of inertia (1.736×10^{-6} m^4)

L = length of beam (1.9m)

$k = (3\times207\times10^9\times1.736\times10^{-6}\times1.9)/(0.5\times1.9)^4$

$k = 2514782.9$

Natural frequency is give on Balachandran and Magrad (2004), p. 74 as

$\omega_n = (k / m)^{0.5}$

$= \sqrt{(2514782.9 / 7.3)}$

$\omega_n = 586.9$ radians/second

To convert to cycles/second, divide ω_n by 2π to get 95 Hz.

This frequency is high enough to be relatively unaffected by vibration, as it is assumed that the larger vibration modes of an implement would occur at 60Hz, or one every second. If this device is ever fitted to a machine that requires a beam that is longer than this, a thicker cross section such as a 127mm×76.2mm hollow beam should be used to reduce the risk of destructive vibration.

8.1.2 Strength analysis

Another important characteristic of this beam is the ability to support the weight of a man. When an implement is being serviced, it is common for a mechanic to stand on the frame of the implement. It is therefore reasonable to assume that this beam will be stepped on, and it is important that it will be able to handle the weight. The second moment of inertia has already been calculated. The loading scenario in this case is simplified by halving the length of the beam and modelling a cantilever. Note that the load in this case also needs to be halved (see figure 8.2 for diagram).

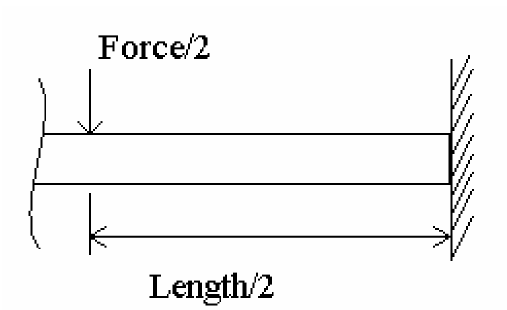


Figure 8.2: Beam loading scenario

The upper weight of a mechanic will be estimated at 120kg, as it is the upper weight that needs to be tested here, not the average. In this case, the weight is halved to 60kg and this is converted to newtons to get a force of 588.6N. The length of the cantilever beam is halved to 950mm, or 0.95m. The universal expression for the stress in a cantilever beam is stated as

$\sigma = (M \times y)/I$ (Durack 2000, p. 17.3)

Where σ = stress (MPa)

M = moment (N/m)

I = second moment of inertia (m^4)

y = distance from centreline to where the stress is being calculated (m)

M = Length×force

= 0.95 × 588.6

= 559.17 N/m

The value of y in this case is half the height of the beam as it is the maximum stress that needs to be found. In this case, this value is 0.05m. I has already been calculated at 1.736×10^{-6} m^4.

$\sigma = (M\times y)/I$

$\sigma = (559.17\times0.05)/\ 1.736\times10^{-6}$

σ = 16.1MPa

This stress is well within the yield stress of the steel. Again, if a longer beam is required, this should be rechecked to ensure stress levels are not too high.

8.2 Stress analysis of castor frame

There are a wide variety of potential static and dynamic stresses that could occur in this system in any one type of environment. When this factor is added to the almost infinite types of operating environments that this system may encounter, it would be foolish to think that the system could be designed to operate under any condition or designed to handle all possible load conditions. Due to this formidable design envelope, each part was designed to be as strong as possible without containing excessive amounts of steel that would increase the build price without adding to the sale point value of the device. As the device has been designed to handle extreme loads, the magnitude of which can only be estimated, a detailed stress analysis of each part is not warranted.

One part which could occasionally be under large stresses is the frame that castors about a shaft and supports the wheel. If the travel speed was high and the wheel encountered a large object such as a rock or stump, it is conceivable that the axle could break under the stress caused by being thrust upwards against the resistance of the damper. It is estimated that an extreme event like this could develop a vertical force of 200kg on the wheel.

It was decided to model this scenario using COSMOS, which is a simplified Finite Element Analysis tool built into Solidworks. This tool is extremely easy to use as once the model has been restrained and had force applied by the user, the program automatically meshes the model in an appropriate manner and solves it. There are not many restraint types and loading scenarios available in this software, but there are enough to provide a rough estimation of the loads.

In a real operating scenario, the force would be applied to the axle as a pressure through the inner bearing races. This would have been difficult or impossible to mimic in the model, so it was decided to apply the whole force at the very end of the axle for simplicity. Even though this represents an extreme loading scenario, it still provided verification for the final design, as if the stresses that occur during this situation are too high, it is unlikely that a slight change in the load position would bring the stresses down to an acceptable level. The restraint was applied to the surface of the castor axle. Figure 8.3 shows how this restraint was applied (left image) and how the force was applied (right image). Note that the direction of the force is directed vertically (the arm is angled back to provide a castor effect).

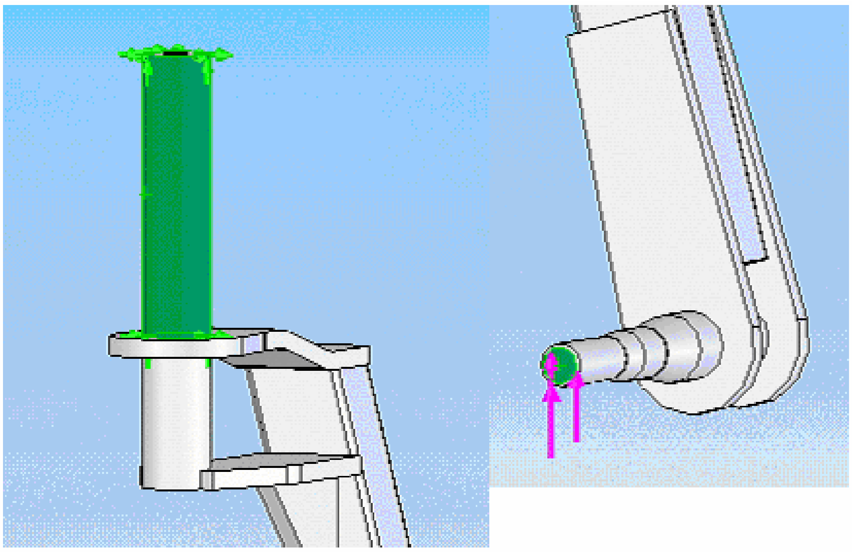


Figure 8.3: A diagram depicting the restraints and loads used in the stress analysis

The force applied to the axle was given a magnitude of 1962 N as per the scenario. The material selected was AISI 1020 steel, and all of the constants relating to this material were stored within the program. The program settings indicated that the elements were sized at approximately 11mm. Solving took approximately 10 seconds which was short due to the coarse mesh. The validity of this mesh size will be verified using manual calculations later.

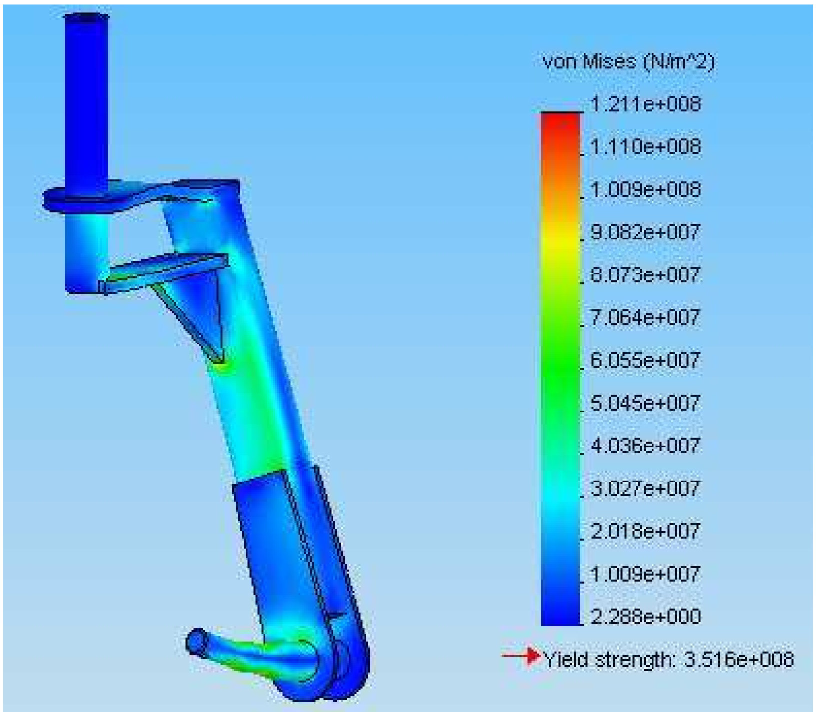


Figure 8.4: An image of the stress distribution after finite element analysis

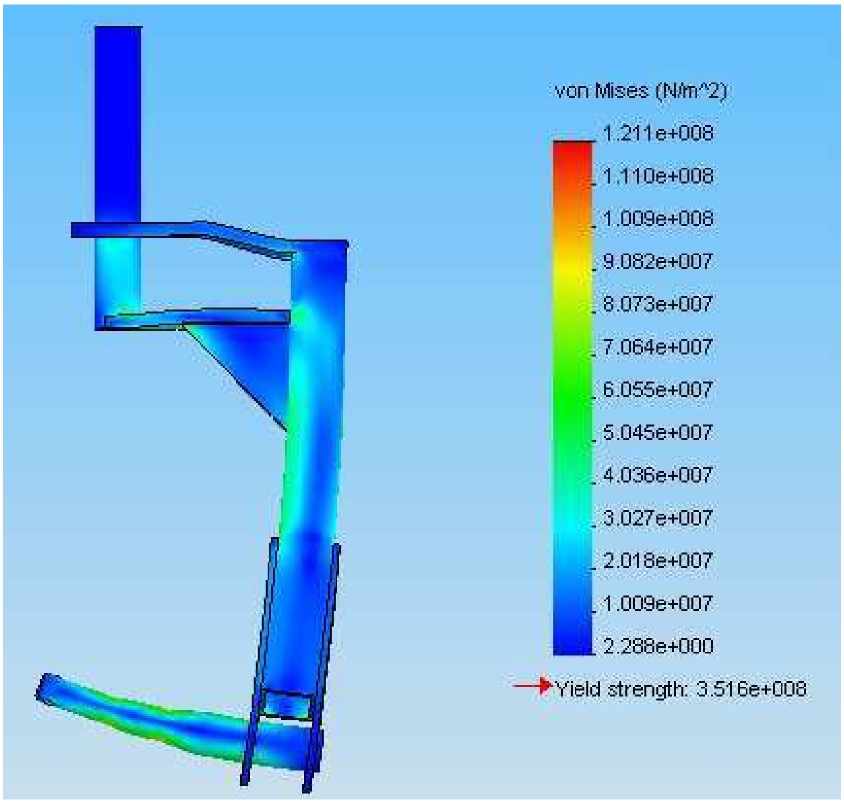


Figure 8.5: Another image showing the stress distribution in the model

After solving, the solid model was depicted in a deformed shape where the extent of the deformation was exaggerated to provide a better visualisation of the results. Colours ranging from blue to red indicated low and high stresses respectively. Two images of the same part are shown in images 8.4 and 8.5. These images show that although the wheel axle does show significant distortion and stress, these stresses still give a factor of safety of approximately 4, which means that the axle would be able to handle even this extreme load scenario.

The rest of the frame shows an expected stress distribution with most stresses rising no higher than 70MPa. The highest stress (≈120MPa) occurs where the bottom corner of the vertical triangular bracing web intersects with the angled spar. Even this stress concentration has a factor of safety of above 2.9, which again given the extreme loading scenario does not indicate any problems.

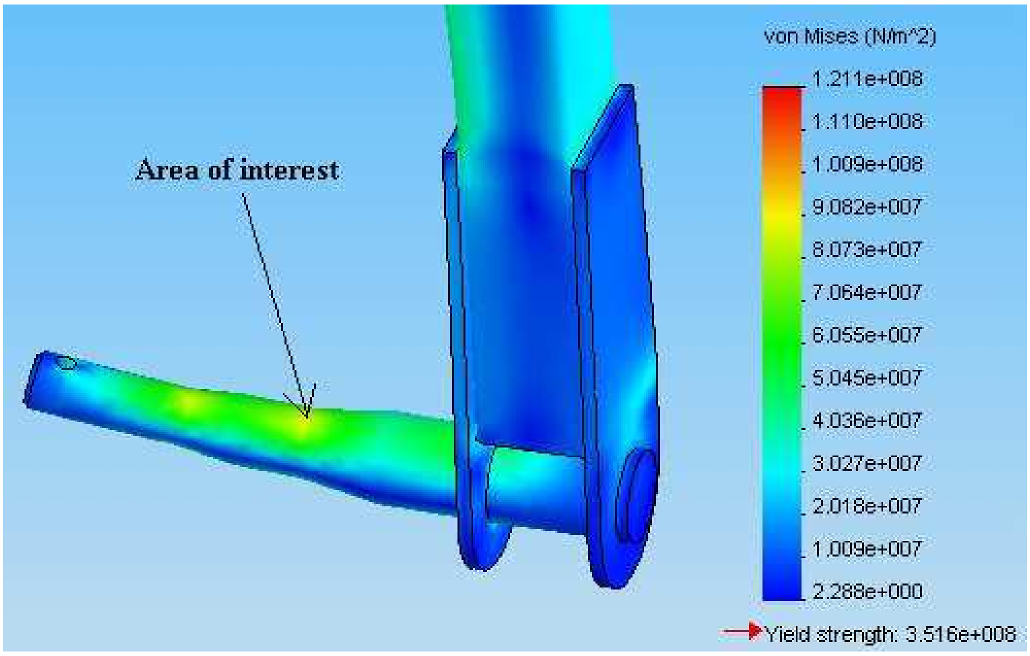
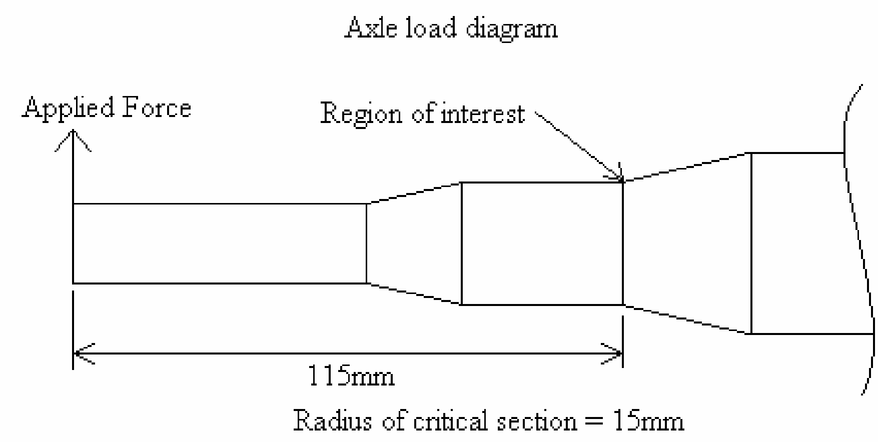


Figure 8.6: Close up image showing axle stress

To verify the accuracy of the estimated stress distribution, manual calculations will be done to provide a means of comparison. **Figure 8.6** shows an image of the stress distribution in the axle. The indicated yellow area on the top of the axle indicates a compression stress of approximately 8.9×10^7 Pa. The manual calculations will be done to verify this result.



As stated before: $\sigma = (M\times y)/I$

Where $I = \pi \times d^4/64$ for a cylindrical cross section **Juvinall and Marsheck, 1999, p. 834**

$I = \pi \times 0.03^4/64$

$= 3.9 \times 10^{-8}$

M = force × distance

$= 1962 \times 0.115$ (where the force = 200kg)

$= 225.63$ N/m

y = 0.015m

$\sigma = (M \times y)/I$

$\sigma = (225.63 \times 0.015)/3.9 \times 10^{-8}$

$\sigma = 8.7 \times 10^7$ Pa

The stress in the same point of the model was estimated at around 8.9×10^7 Pa, and given that the manual calculation neglected shear stress, this is a credible validation of the accuracy of the model.

CHAPTER 9 – SUMMARY AND CONCLUSIONS

9.1 Summary

After investigating the need for a height control system, two main avenues of approach were investigated; the mechanical / electrical / hydraulic system and the mechanical / hydraulic system. Both proposed systems were designed to use height sensing wheel frames as a method of sensing the height of the implement frame after deciding that ultrasonic sensing technology would not be suitable for this application in their current format due to possible interference problems with stubble cover and soft soil. It was discovered after testing of the PICAXE microcontroller that it was possible to use this chip to process height data, however the lack of a fully proportional, variable voltage output capability reduced the potential for this device to actuate a solenoid activated spool valve.

This left the mechanical / hydraulic system as a viable option. The ground following wheel mechanisms were designed first. After the action of these mechanisms were analysed, a way of using them to actuate a spool valve was devised, and a design concept was formulated. This concept was then turned into solid models of the parts that would make up the mechanism. An assembly was made from these solid models and the operation of the device was examined by moving the virtual components. It was decided that the operation of the device was refined to the point where the production of a prototype for testing could nearly go ahead. The technical drawings of all of the major parts were produces for reference to the tradesman.

Critical parts of the designed were analysed with basic engineering stress and vibration analysis where applicable. The results of this analysis verified these parts of the design. One aspect of the design that was not explored was how the device would fit on various implements. This was partly because there are so many different designs of implements available, and it is difficult to see how a ‘one size fits all’ device could be built. The device was simply made as compact and as unobtrusive as possible, and remedies were suggested to fix possible clash problems.

9.2 Further Work

There is much more work that could be done to advance the design of the mechanism. The mechanical / hydraulic design has room for further refinement that has not been investigated here. The system could be simplified and made more accurate if the front parallelogram mechanism was turned around and made to trail. This would remove the need to reverse the motion of the rear linkage using a crank mechanism, meaning there would only be a need for two linkages and no crank or valve lever. This would make the device more compact and simple as it would make several parts redundant. To enable the front wheel mechanism to trail without impinging on other items such as tines, the implement would most likely have to be modified with the addition of another section of frame jutting from the front of the implement to act as an attachment point.

It is most important that before any prototype is built, a system is designed where if one wheel becomes too high or low with respect to the other wheel for any reason, a device releases and allows the linkages to move as far as they need to without destroying the spool valve as it reaches the end of its stroke. This problem must be solved before a prototype is constructed.

The electrical system that was theorised in this project showed promise, however the limitations that exist at the present moment prevent it from being recommended as a solution for this application. There is one foreseeable method for obtaining a proportional output from the PICAXE chip. The PICAXE – 28A (see relevant figure in appendix) has seven output pins. It may be possible to create a seven step proportional output from these pins if they are ‘ramped up’, or activated in a way that the greater the height difference, the more pins are set to ‘high’. This would mean that an auxiliary circuit would need to be created that could convert this output from the chip into a seven step variable voltage. This voltage could then be used to activate a relay that would control a flow of current to a solenoid operated valve.

If this system was coupled with field - proven ultrasonic sensors and the whole system was made durable enough to handle the harshest field conditions, it will arguably represent the cheapest, best, most compact and most accurate solution to the problem of height control. The final aspect of this design that needs to be stated is the potential for it to be applied to conventional trailing implements that have passive floating hitches. Passive floating hitches simply allow the implement to follow the ground contours more effectively by allowing the implement frame to rock back and forth. This technology could conceivably be adapted to provide an active floating hitch which uses hydraulic rams to alter the attitude of the frame as it moves over uneven ground.

9.3 Conclusions

From the completion of the aim of this project, which was the design of a height control system to suit a three point linkage implement; the conclusions are as follows:

- From the literature review (I), it can be seen that there have been several attempts at the solution of height control for agricultural machines. The varying degrees of success that have been achieved by these devices and the fact that these systems are not widely in use has indicated that no system has been completely successful.

- PICAXE microcontrollers are the most compact and the cheapest solution to this problem, however better ultrasonic transducers or another cheap method of height sensing as well as better programming methods should be found before the full capability of these devices can be exploited.

- It is possible to control a solenoid valve using height control wheels and a fully mechanical system as has been shown, however this system is bulky and due to the amount of steel involved in its construction, is most likely more expensive than the electronic option.

- The height control system could be adapted to hydraulically control the floating hitches on trailing implements

- The actual operational characteristics of any control system cannot be evaluated without the construction of a prototype

- The height sensing wheels and frames can be used whenever a ground following, height sensing wheel is needed in any agricultural situation.

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III APPENDICIES

APPENDIX A: PROJECT SPECIFICATION

University of Southern Queensland
FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 Research Project

PROJECT SPECIFICATION

FOR:	**DANIEL JOHN CLARKE**
TOPIC:	DESIGN OF AN AUTOMATIC HEIGHT CONTROL SYSTEM FOR A THREE-POINT LINKAGE PLANTER
SUPERVISER:	Prof. David Ross
PROJECT AIM:	This project aims to develop an effective method to control the height of a three point linkage planter frame fitted with folding wings and a lift-assist mechanism at the rear.

PROGRAMME: **Issue A, 24th March 2006**

1. Research the need for effective depth control on the type of planter described above and investigate other attempts at depth control systems.

2. Research and describe potential systems that could be used for depth control purposes, and use critical evaluation to select the most appropriate system.

3. Design a control system that will sense the height of the planter frame as close to the tractor as possible, and use this information to control the height of both the lift assist mechanism and the wheels on the wings. Feedback in the form of height information may be provided to the driver through a form of a simple user interface.

4. Design the mechanical systems that will support and operate the sensing equipment.

5. Analyse the theoretical performance of the control system in terms of in field performance, for example the response time and system stability.

6. Research other potential applications of this control system, such as the adaptation to suit conventional trailing planting machines.

As time permits

7.Construct the basic control system and a test rig to evaluate the actual performance of the control system.

AGREED:

_________________________(Student) ____________________________(Supervisor)

____/____/____

PICAXE PROGRAMS

Program_1

```
main:
                let b12= 150            'Set initial servo position to middle of stroke

label_30:       readadc  0,b0           'load byte variable b0 with the digital position
                                        'of resistor 1
readadc  1,b1                           'load byte variable b1 with the digital position
                                        'of  resistor 2
pause 330                               'pause for 330 milliseconds
                readadc  0,b2
                readadc  1,b3
                pause 330               'Repeat process with b2 and b3
                readadc  0,b4
                readadc  1,b5
                pause 330               'Repeat process with b4 and b5
                readadc  0,b6
                readadc  1,b7
                pause 330               'Repeat process with b6 and b7
                readadc  0,b8
                readadc  1,b9
                pause 330               'Repeat process with b8 and b8
                readadc  0,b10
                readadc  1,b11          'Repeat process with b10 and b11 but without a
                                        'pause
                b10 = b0 + b2 + b4 + b6 + b8 + b10 / 6
                                        'Average the height values for resistor 1
                b11 = b1 + b3 + b5 + b7 + b9 + b11 / 6
```

```
                                    'Average the height values for resistor 2
                if b10>b11 then label_A6
                                    'Ensure a positive difference is found
                let b3=b11-b10      'Find difference between average height values

label_B5:       let b0=b10+ 3
                let b1=b11+ 3
                let b2=b11- 3       'specify tolerance values
                if b11>b0 then label_EB
                                    'If out of tolerance range than reposition servo
                if b10>b1 then label_F2
                                    'If out of tolerance range than reposition servo
goto label_30                       'If in tolerance range then do nothing and repeat
                                    'process

label_A6:       let b3=b10-b11      'Find difference between average height values
                goto label_B5       'Return to start of label B5

label_EB:       b3 = b3 / C         'Reduce position difference value b3 by a factor
                                    'of C
                b12 = b12 + b3 max 225
                                    'Increase servo position variable b12 by variable
                                    'b3 with a maximum value of 225
                servo 3, b12        'Reposition servo to the correct position
                goto label_30       'Restart process

label_F2:       b3 = b3 / C         'Reduce position difference value b3 by a factor
                                    'of C
                b12 = b12 - b3 min 75
                                    'Decrease servo position variable b12 by
                                    'variable b3 with a minimum value of 75
```

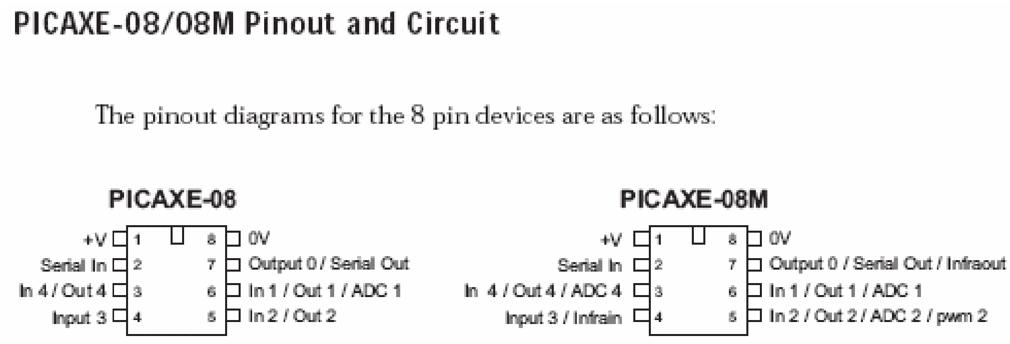
```
servo 3, b12        'Reposition servo to the correct position
goto label_30       'Restart process
```

Program_2

```
main:
                let b12= 150            ‘Set initial servo position to middle of stroke
label_30:       readadc  0,b10          ‘load byte variable b10 with the digital position
                                        ‘of resistor 1
        readadc  1,b11                  ‘load byte variable b11 with the digital position
                                        ‘of resistor 2
                if b10>b11 then label_A6
                                        ‘Ensure a positive difference is found
                let b3=b11-b10          ‘Find difference between height values

label_B5:       let b0=b10+ 3
                let b1=b11+ 3
                let b2=b11- 3           ‘specify tolerance values
                if b11>b0 then label_EB
                                        ‘If out of tolerance range than reposition servo
                if b10>b1 then label_F2
                                        ‘If out of tolerance range than reposition servo
                pause 20                ‘pause for 20 milliseconds
goto label_30                           ‘If in tolerance range then do nothing and repeat
                                        ‘process

label_A6:       let b3=b10-b11          ‘Find difference between height values
                goto label_B5           ‘Return to start of label B5

label_EB:       b3 = b3 / C             ‘Reduce position difference value b3 by a factor
                                        ‘of C
                b12 = b12 + b3 max 225
                                        ‘Increase servo position variable b12 by variable
                                        ‘b3 with a maximum value of 225
```

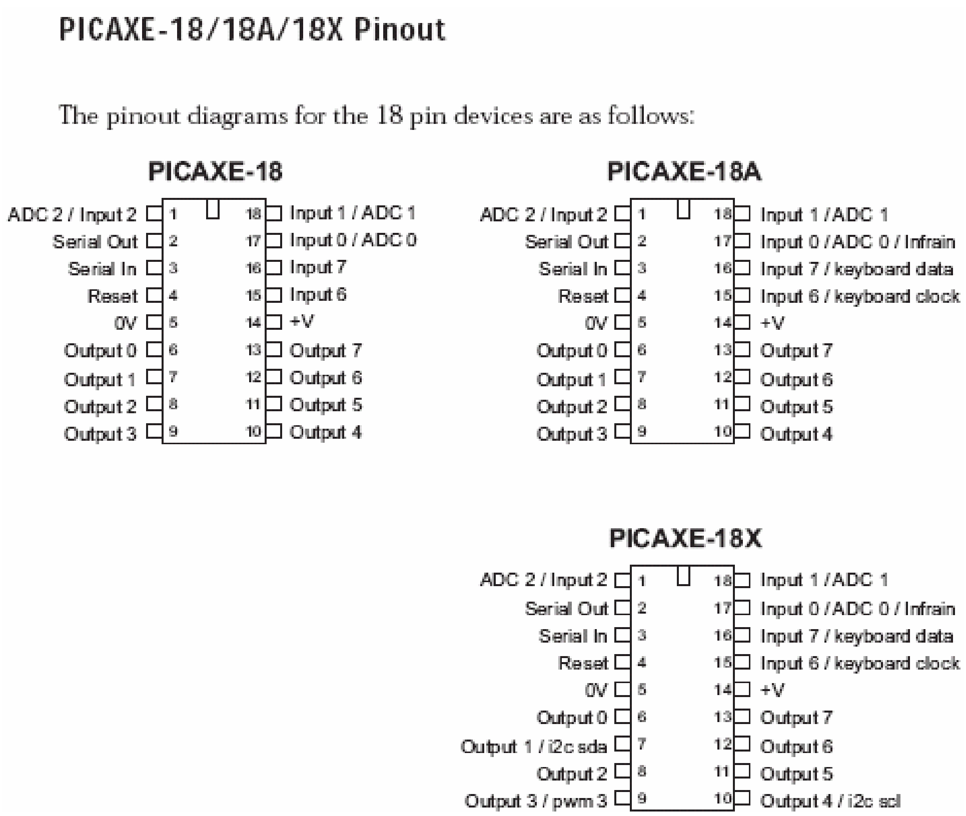
```
                servo 3, b12            'Reposition servo to the correct position
        b3 = b3 * D                     'increase height difference variable b3 by a
                                        'factor of D
b3 = b3 + E                             'Add constant E to ensure the variable pause is
                                        'at least E milliseconds
                pause b3                'pause for b3 milliseconds
                goto label_30           'loop program

label_F2:       b3 = b3 / C             'Reduce position difference value b3 by a factor
'of C
                b12 = b12 - b3 min 75
                                        'Decrease servo position variable b12 by
                                        'Variable b3 with a minimum value of 75
                servo 3, b12            'Reposition servo to the correct position
        b3 = b3 * D                     'increase height difference variable b3 by a
                                        'factor of D
b3 = b3 + E                             'Add constant E to ensure the variable pause is
                                        'at least E milliseconds
                pause b3                'pause for b3 milliseconds
                goto label_30           'Loop program
```

Schematics of the eight chip types



08/08M Pin Diagram

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

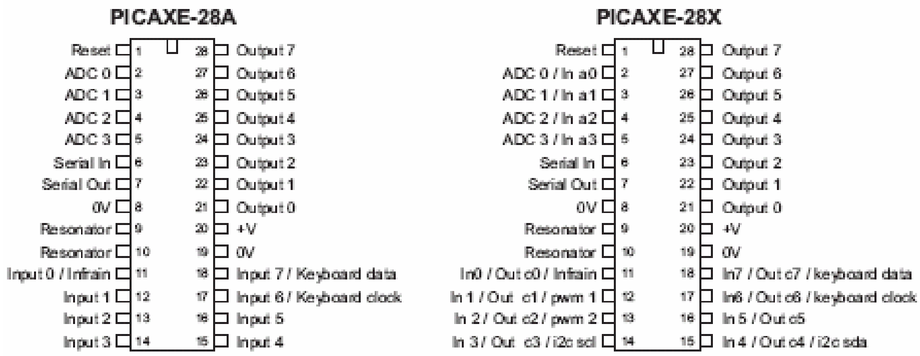


18/18A/18X Pin Diagram

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

PICAXE-28A/28X Pinout

The pinout diagrams for the 28 pin devices are as follows:

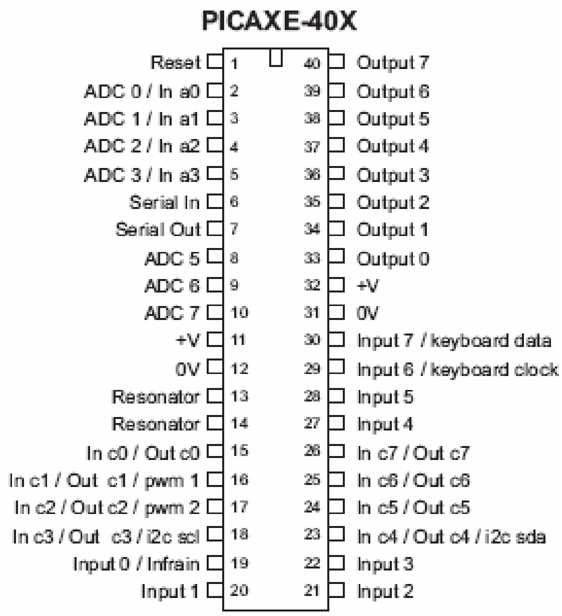


28A/28X Pin Diagram

Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

PICAXE-40X Pinout

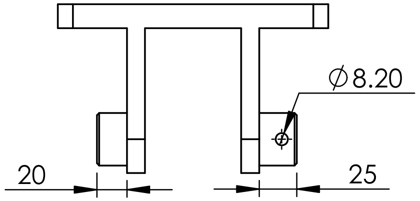
The pinout diagram for the 40 pin device is as follows:



40X Pin Diagram

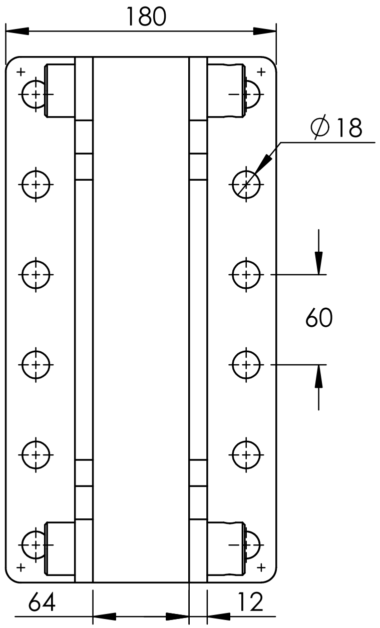
Source: http://www.rev-ed.co.uk/docs/picaxe_manual1.pdf

ITEM	DESCRIPTION	SIZE	GRADE
1	16MM BACKING PLATE	180 X 350MM	350
2	12MM WEB PLATE	101 X 350MM	350
3	Ø35 OD X Ø20 ID BEARING TUBE	20MM	1020
4	Ø35 OD X Ø20 ID BEARING TUBE	25MM	1020

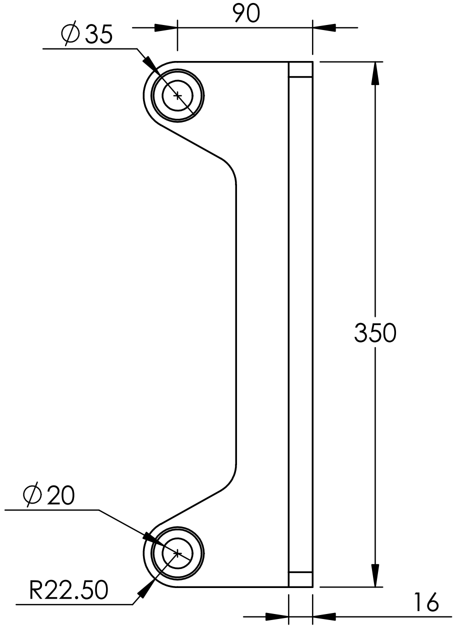


TOP VIEW

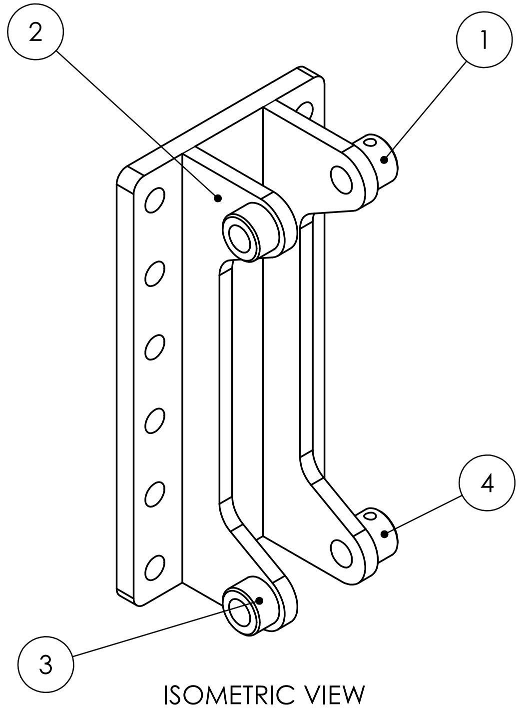
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



FRONT VIEW



SIDE VIEW

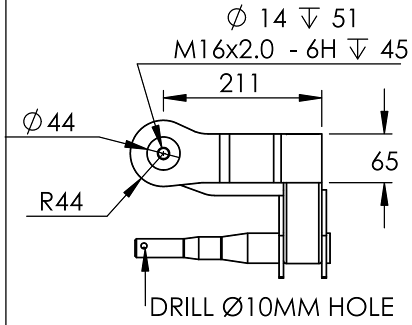


ISOMETRIC VIEW

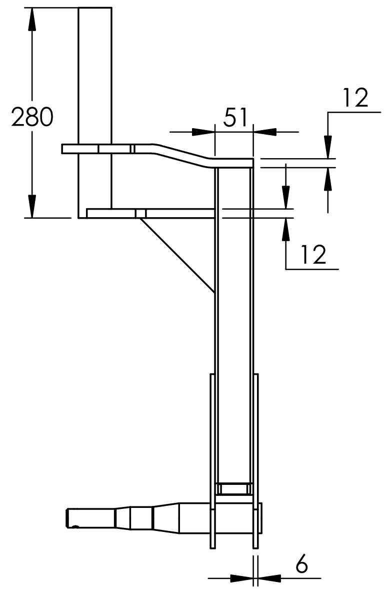
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
DRAWN	NAME	SIGNATURE	DATE	TITLE:	INNER FASTENER PLATE
	DANIEL CLARKE		10/10/06		
Q.A			MATERIAL: STEEL	DWG NO. 1	A4
			WEIGHT:	SCALE:1:5	SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	Ø44MM STEEL SHAFT	280MM	1020
2	12MM BENT UPPER LINK	88MM X 255MM	350
3	12MM LOWER LINK	90MM X 175MM	350
4	12MM WEB	100 X 100 X 141MM TRIANGLE	350
5	RECTANGULAR RHS LEG	51MM X 63MM RHS	1020
6	12MM AXLE SUPPORT	80MM X 230MM	350
7	Ø40MM TURNED AXLE	260MM	1020

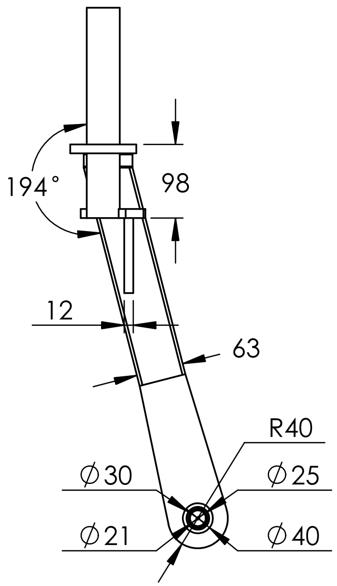
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



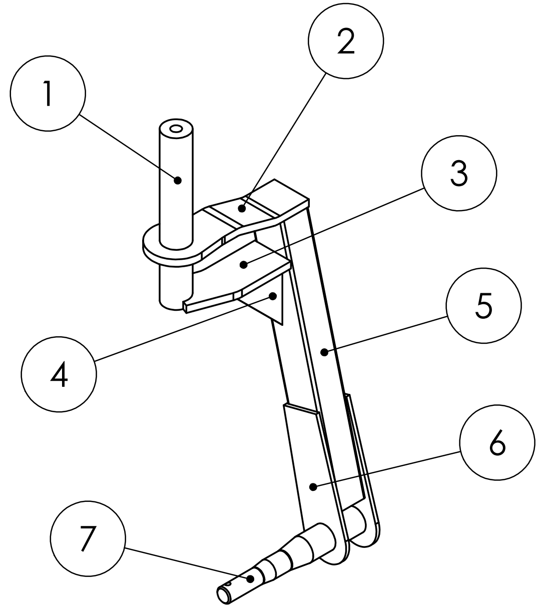
TOP VIEW



FRONT VIEW



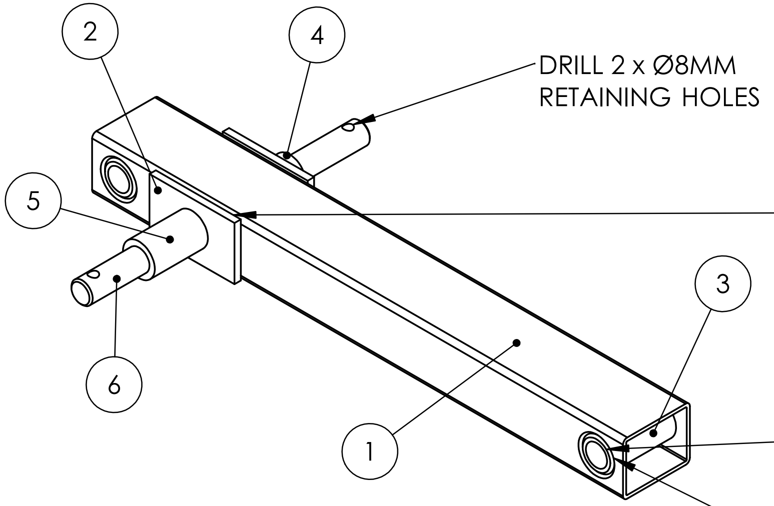
SIDE VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
DRAWN	NAME	SIGNATURE	DATE	TITL E: CASTOR FRAME
	DANIEL CLARKE		10/10/06	
Q.A			MATERIAL: STEEL	DWG NO. 2 — A4
			WEIGHT:	SCALE:1:10 — SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	63 X 50MM RHS BEAM	500MM	1020
2	6MM STRESS PLATE	45X85MM	350
3	Ø30MM OD X Ø20MM ID BEARING TUBE	63MM	1020
4	Ø30MM OD X Ø20MM ID COLLAR	17MM	1020
5	Ø30MM OD X Ø20MM ID COLLAR	50	1020
6	Ø20MM SHAFT	245	1020



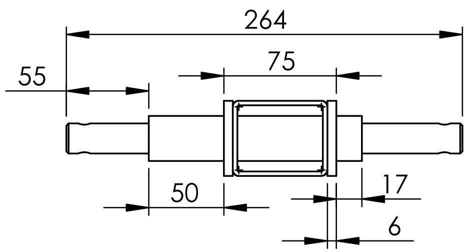
DRILL 2 x Ø8MM RETAINING HOLES

WELD TOP AND BOTTOM IN THIS GROOVE BEFORE BORING HOLE TO ENSURE CORRECT ALIGNMENT. AFTER DRILLING, INSERT LONG ROD INTO THE HOLE AND THEN SLIDE COLLARS ONTO THE ROD. APPLY A 6MM WELD BEAD TO WHERE THE COLLARS TOUCH THE STEEL PLATE.

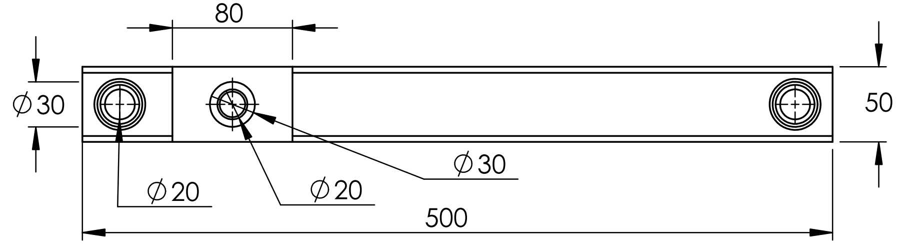
ADD 2MM BEVELLED EDGE TO OUTSIDE OF BEARING TUBE BEFORE INSERTION. APPLY WELD INTO RESULTING VEE AND GRIND WELD FLUSH.

BORE 30MM HOLE AND COUNTERSINK TO 2MM DEPTH AT 45 DEGREES.

ISOMETRIC VIEW



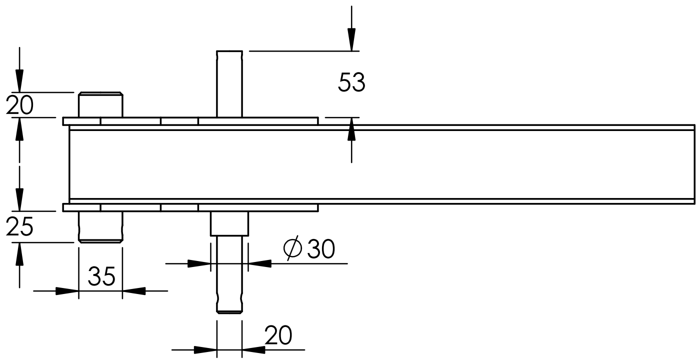
FRONT VIEW



SIDE VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING	REVISION
DRAWN	NAME	SIGNATURE	DATE	TITLE:	LOWER LINK
	DANIEL CLARKE		10/10/06		
Q.A			MATERIAL: STEEL	DWG NO. 3	A4
			WEIGHT:	SCALE:1:5	SHEET 1 OF 1

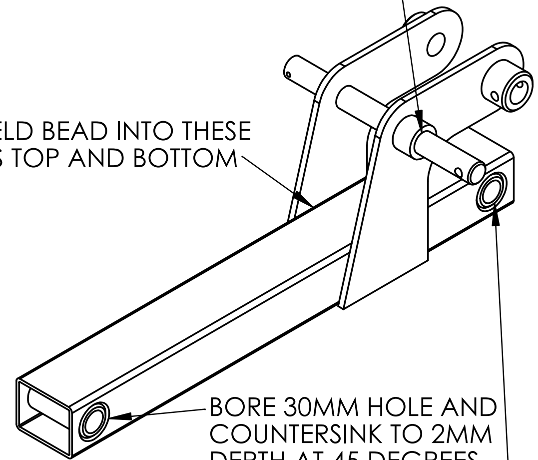
ITEM	DESCRIPTION	SIZE	GRADE
1	63 X 50MM RHS BEAM	500MM	1020
2	Ø30MM OD X Ø20MM ID BEARING TUBE	63MM	1020
3	6MM LEVER PLATE	170 X 205MM	350
4	Ø20MM SHAFT	205.5MM	1020
5	Ø30MM OD X Ø20MM ID COLLAR	19.5MM	1020
6	Ø35MM OD X Ø20MM ID PIN COLLAR	20MM	1020
7	Ø35MM OD X Ø20MM ID PIN COLLAR	25MM	1020



TOP VIEW

INSERT LONG ROD INTO THE HOLES AND THEN SLIDE THE COLLAR ONTO THE ROD. APPLY A 6MM WELD BEAD TO WHERE THE COLLAR TOUCHES THE STEEL PLATE. WHEREVER POSSIBLE, APPLY 6MM WELD BEAD TO THE ROD TO SECURE IT TO THE STEEL PLATES.

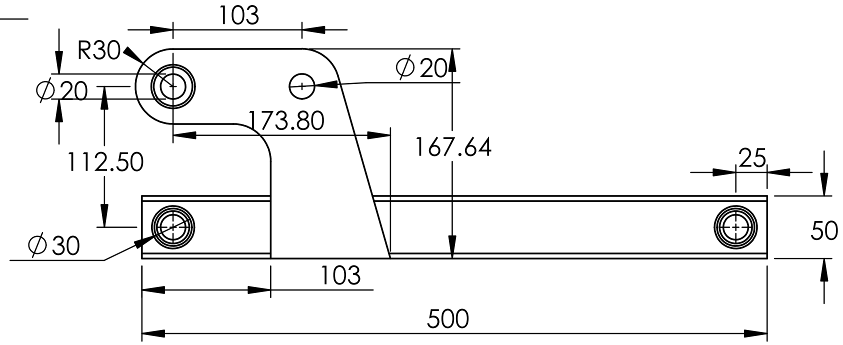
APPLY WELD BEAD INTO THESE HOLLOWS TOP AND BOTTOM



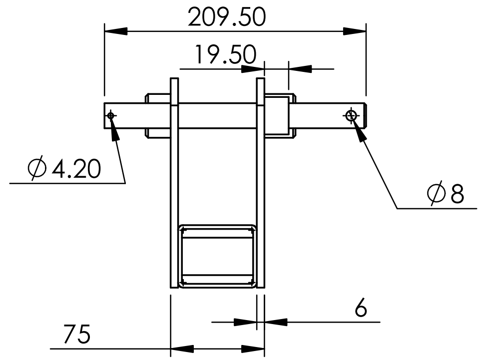
BORE 30MM HOLE AND COUNTERSINK TO 2MM DEPTH AT 45 DEGREES.

ADD 2MM BEVELLED EDGE TO OUTSIDE OF BEARING TUBE BEFORE INSERTION. APPLY WELD INTO RESULTING VEE AND GRIND WELD FLUSH.

ISOMETRIC VIEW



SIDE VIEW



FRONT VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		10/10/06
Q.A			

TITLE: UPPER LINK

MATERIAL: STEEL

WEIGHT:

DWG NO. 4

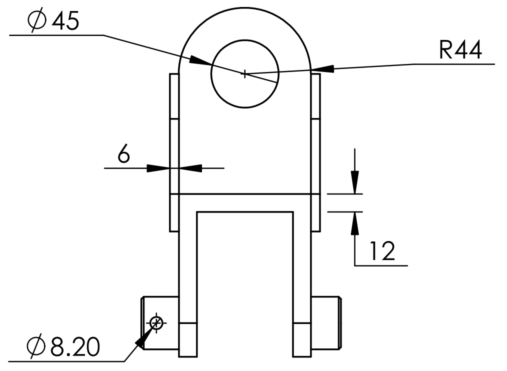
A4

SCALE: 1:6

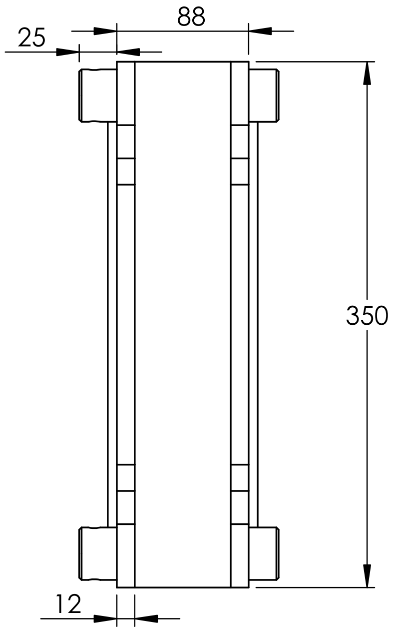
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	12MM WEB PLATE	113MM X 350MM	350
2	12MM WEB CONNECTOR PLATE	64MM X 350MM	350
3	6MM BRACING PLATE	105MM X 330MM	350
4	12MM BEARING TUBE HOLDER	88MM X 149MM	350
5	Ø60 OD X Ø45 ID BEARING TUBE	180MM	1020
6	Ø35 OD X Ø20 ID PIN SUPPORT TUBE	20MM	1020
7	Ø35 OD X Ø20 ID PIN SUPPORT TUBE	25MM	1020

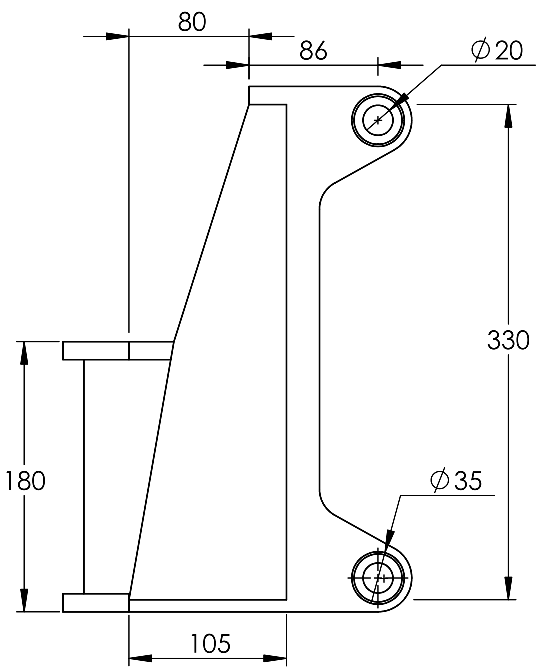
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



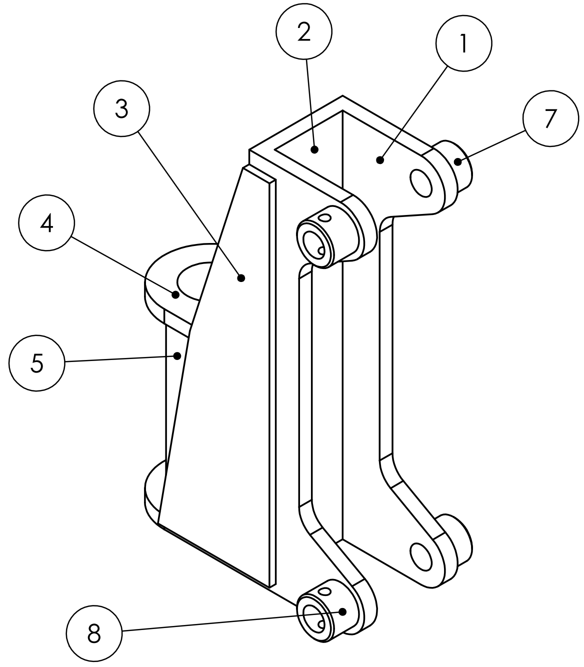
TOP VIEW



FRONT VIEW



SIDE VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
DRAWN	NAME	SIGNATURE	DATE	
	DANIEL CLARKE		10/10/06	
Q.A				

TITLE:

REAR PIVOT BODY

MATERIAL: STEEL

DWG NO. 5

A4

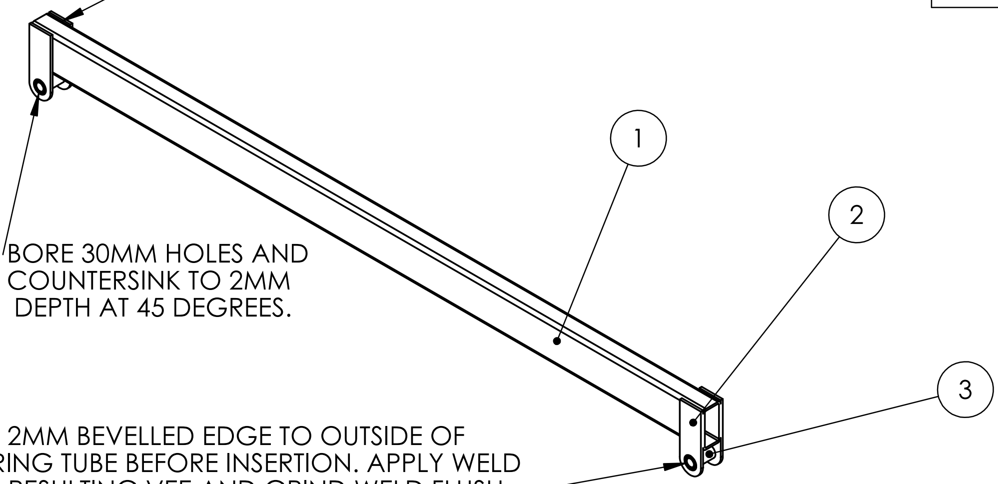
WEIGHT:

SCALE:1:5

SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	50 X 100MM RHS MAIN BEAM	1900MM	1020
2	6MM END PLATE	60 X 179MM	350
3	Ø30MM OD Ø20 ID BEARING TUBE	62MM	1020

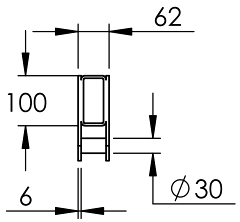
WELD TOP AND BOTTOM IN THIS GROOVE BEFORE BORING HOLES TO ENSURE CORRECT ALIGNMENT.



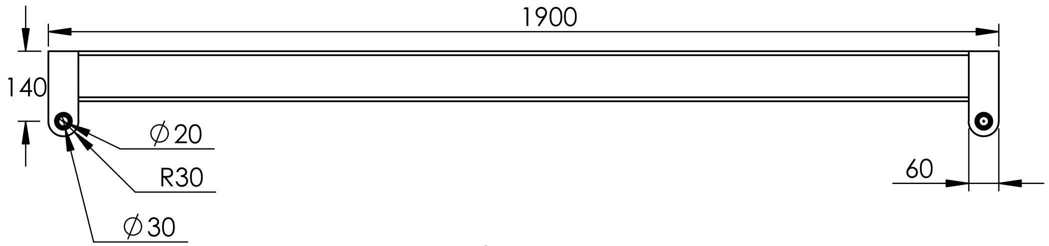
BORE 30MM HOLES AND COUNTERSINK TO 2MM DEPTH AT 45 DEGREES.

ADD 2MM BEVELLED EDGE TO OUTSIDE OF BEARING TUBE BEFORE INSERTION. APPLY WELD INTO RESULTING VEE AND GRIND WELD FLUSH.

ISOMETRIC VIEW



END VIEW

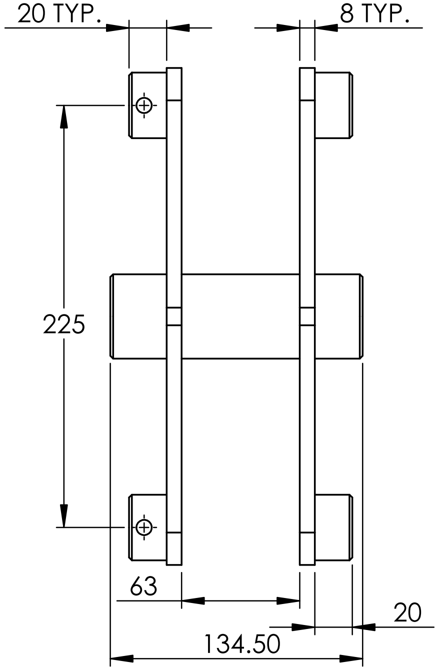


SIDE VIEW

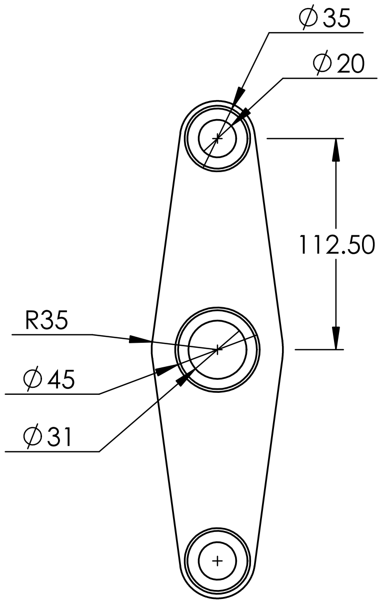
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:				DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING / REVISION
DRAWN	NAME	SIGNATURE	DATE		TITLE: MAIN CONTROL LINK
	DANIEL CLARKE		10/10/06		
Q.A				MATERIAL: STEEL	DWG NO. 6 / A4
				WEIGHT:	SCALE:1:20 / SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	8mm Thick STeel PLate	71x265MM	350
2	Ø45 OD x Ø31 ID STEEL TUBE	135MM	1020
3	Ø28 OD x Ø20 ID STEEL TUBE	20MM	1020
4	Ø28 OD x Ø20 ID STEEL TUBE	20MM	1020

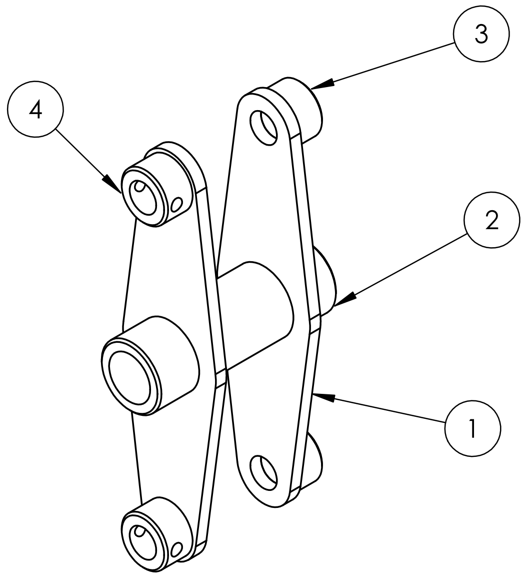
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



FRONT VIEW



SIDE VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		10/10/06
Q.A			

TITLE: PIVOT CRANK

MATERIAL: STEEL

WEIGHT:

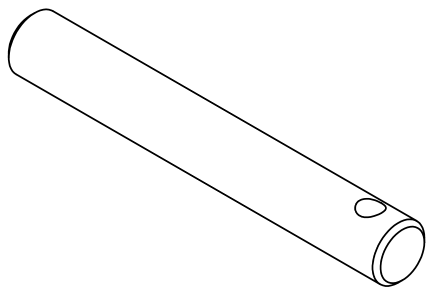
DWG NO. 7

A4

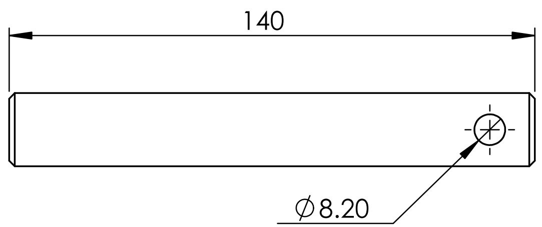
SCALE:1:4

SHEET 1 OF 1

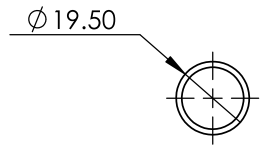
ITEM	DESCRIPTION	SIZE	GRADE
1	Ø19.5MM TURNED SHAFT	140MM	1020



ISOMETRIC VIEW



FRONT VIEW



END VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		16/10/06
Q.A			

TITLE:

UNIVERSAL PIN

MATERIAL: STEEL

WEIGHT:

DWG NO. 8

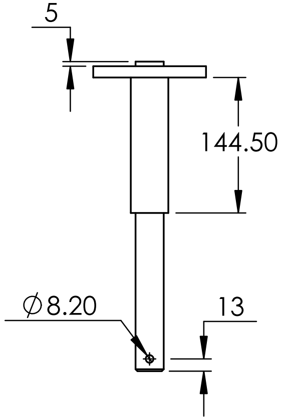
A4

SCALE:1:2

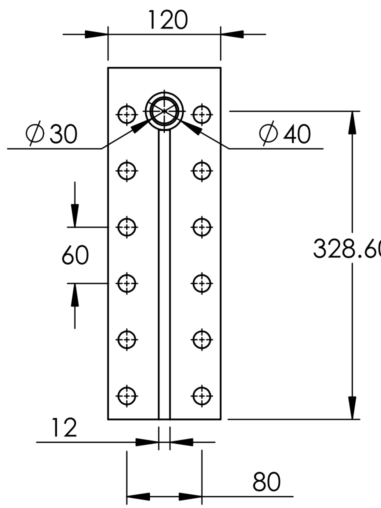
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	12MM BACKING PLATE	375 X 120 MM	350
2	Ø30MM SHAFT	330MM	1020
3	Ø40MM OD X Ø30MM ID SHAFT COLLAR	144.5MM	1020
4	12MM SUPPORT RIB	50 X 308.5 MM	350

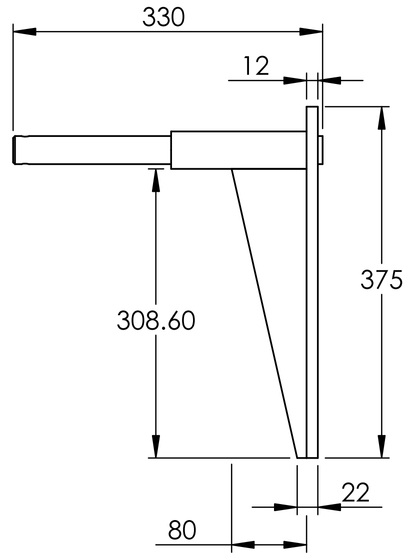
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



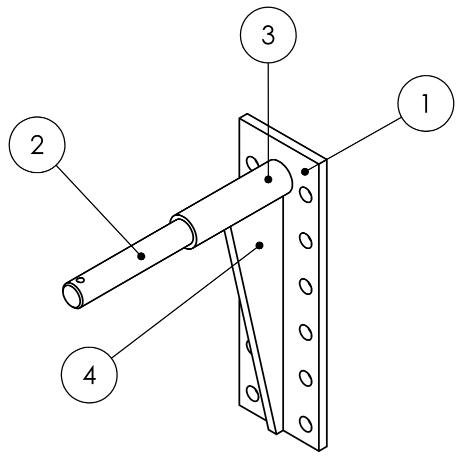
TOP VIEW



FRONT VIEW



SIDE VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE
DRAWN	DANIEL CLARKE		12/10/06
Q.A			

TITLE:

CRANK PIVOT

MATERIAL: STEEL

WEIGHT:

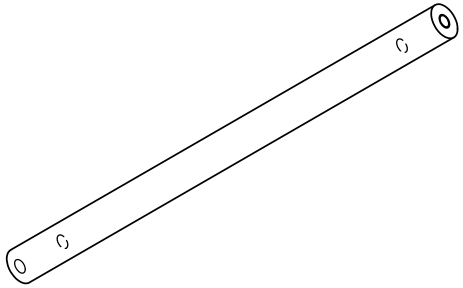
DWG NO. 9

A4

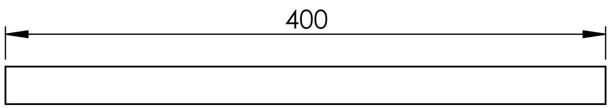
SCALE:1:10

SHEET 1 OF 1

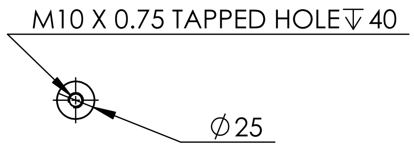
ITEM	DESCRIPTION	SIZE	GRADE
1	Ø25MM SLIDER BAR	400MM	1020



ISOMETRIC VIEW



FRONT VIEW



SIDE VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		12/10/06
Q.A			

TITLE:

SLIDER BAR

MATERIAL: STEEL

DWG NO. 10

A4

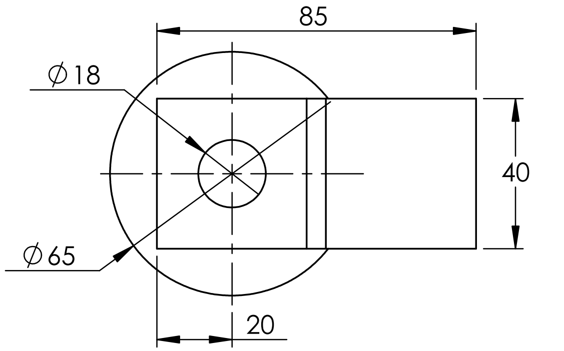
WEIGHT:

SCALE:1:5

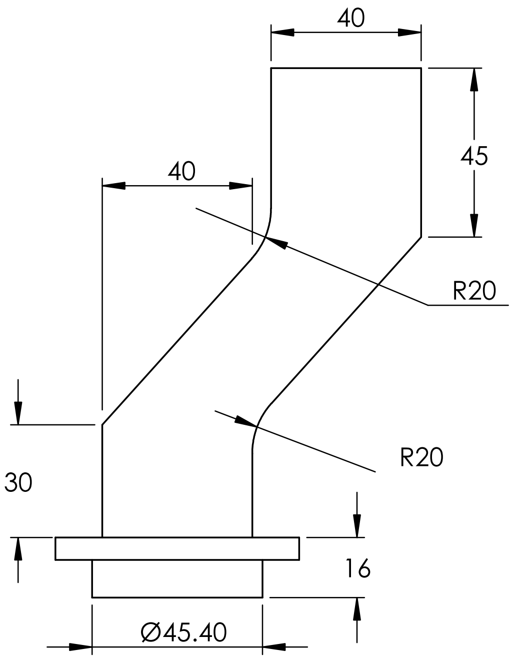
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	40MM SPRING PIVOT	85MM X 125MM	350
2	Ø65 MM TURNED SPRING STOP	16MM	350

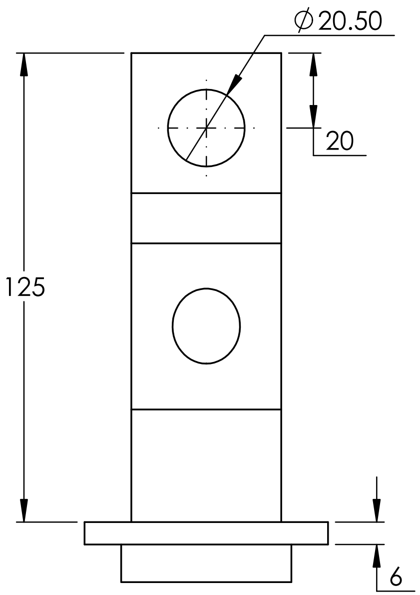
THE SPRING PIVOT IS NOT WELDED TO THE SPRING STOP



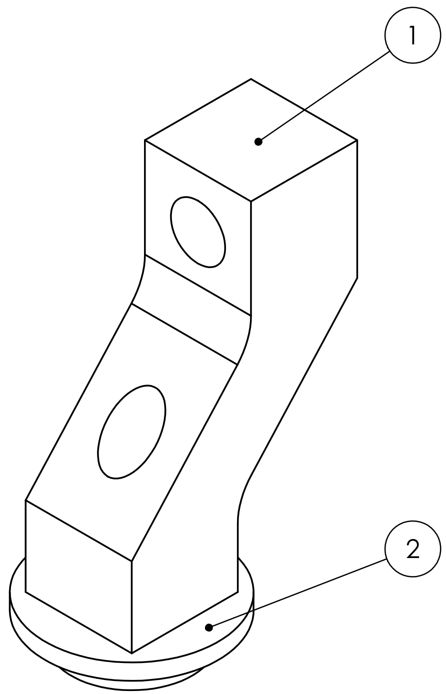
TOP VIEW



FRONT VIEW



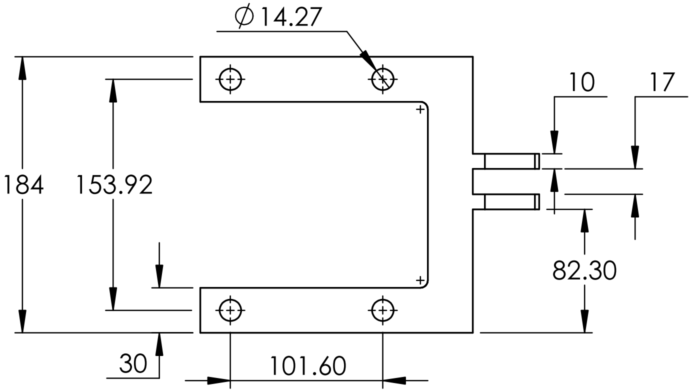
SIDE VIEW



ISOMETRIC VIEW

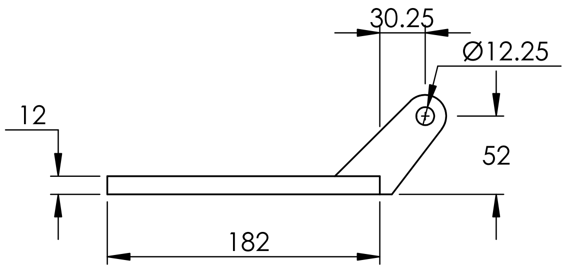
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
DRAWN	NAME	SIGNATURE	DATE	TITLE: UPPER SPRING STOP
	DANIEL CLARKE		12/10/06	
Q.A			MATERIAL: STEEL	DWG NO. 11 — A4
			WEIGHT:	SCALE:1:2 — SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	12MM BEDDING PLATE	184 X 182MM	350
2	10MM ARM PIVOT	85 X 40MM	350

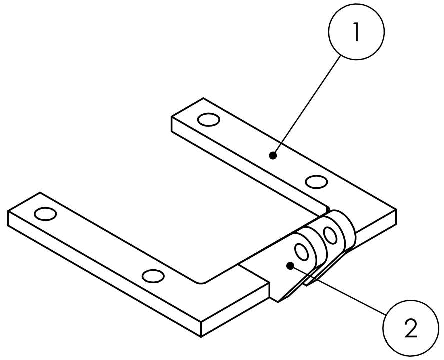


TOP VIEW

APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



FRONT VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
		REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		12/10/06
Q.A			

TITLE: VALVE LEVER BRACE

MATERIAL: STEEL

DWG NO. 12

A4

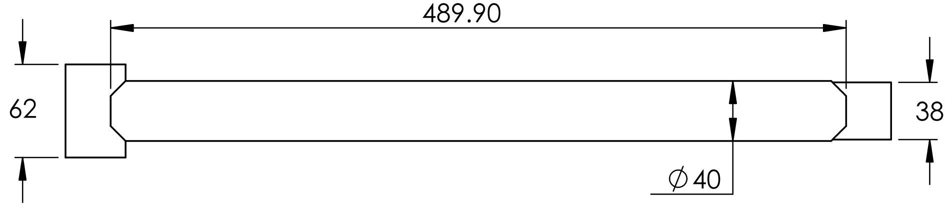
WEIGHT:

SCALE:1:5

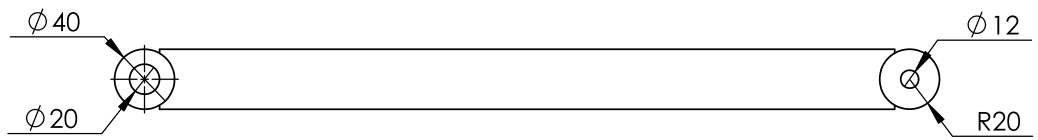
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	Ø40MM OD X Ø30MM ID MAIN TUBE	531MM	1020
2	Ø40MM OD X Ø20MM ID BEARING TUBE	62MM	1020
3	Ø40MM OD X Ø12MM ID BEARING TUBE	38MM	1020

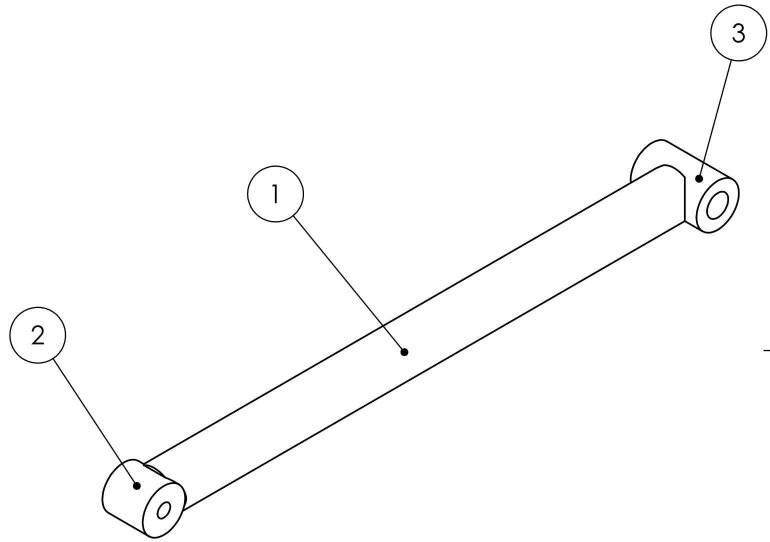
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



TOP VIEW



FRONT VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	DANIEL CLARKE		12/10/06
Q.A			

TITLE:

REAR SHORT LINK

MATERIAL: STEEL

WEIGHT:

DWG NO. 13

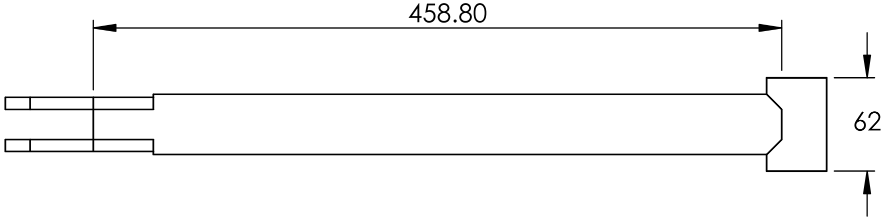
A4

SCALE:1:5

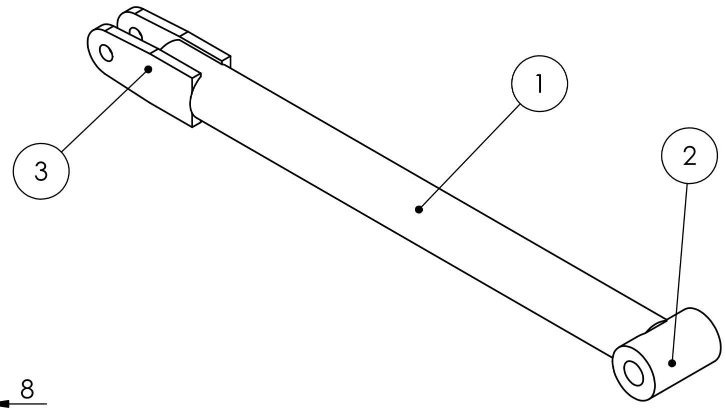
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	Ø40MM OD X Ø30MM ID MAIN TUBE	500MM	1020
2	Ø40MM OD X Ø18MM ID BEARING TUBE	62MM	1020
3	8MM END PLATE	81MM X 40MM	350

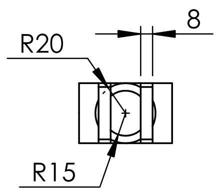
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



TOP VIEW



FRONT VIEW



END VIEW

ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		13/10/06
Q.A			

TITLE:

LEADING LINK

MATERIAL: STEEL

WEIGHT:

DWG NO. 14

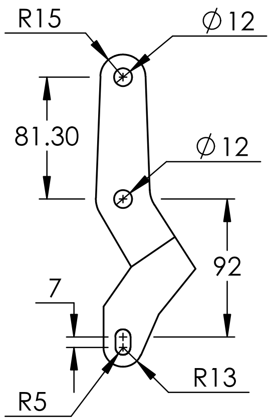
A4

SCALE:1:5

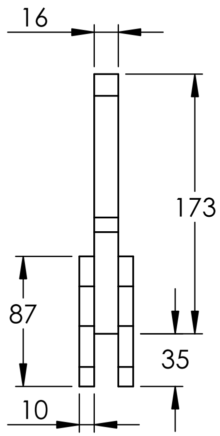
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	16MM UPPER LEVER	173 X 67MM	350
2	10MM LOWER WEB	62 X 87	350

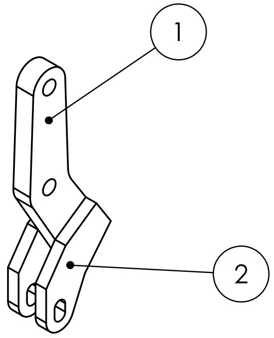
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



SIDE VIEW



FRONT VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:	DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
		REVISION

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		13/10/06
Q.A			

TITLE: VALVE LEVER

MATERIAL: STEEL

WEIGHT:

DWG NO. 15

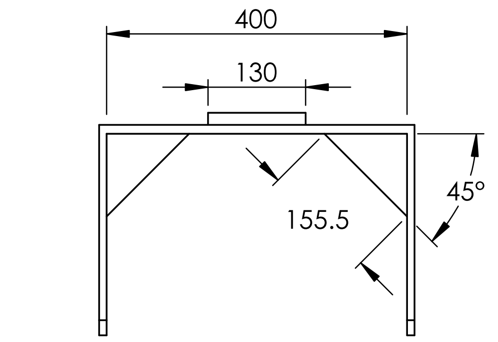
A4

SCALE:1:5

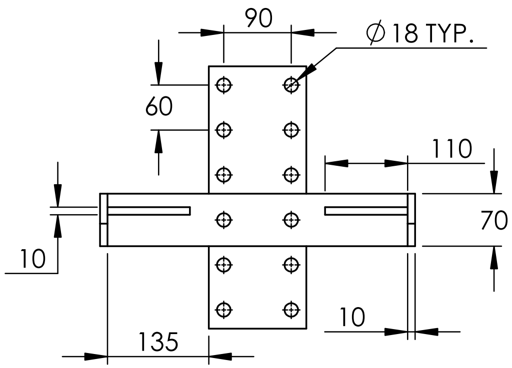
SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	16MM BACKING PLATE	130MM X 350MM	350
2	12MM CROSS PLATE	70MM X 400MM	350
3	10MM BRACING WEB	70MM X 190MM	350
4	10MM TRIANGLE BRACE	110MM X 110MM	350

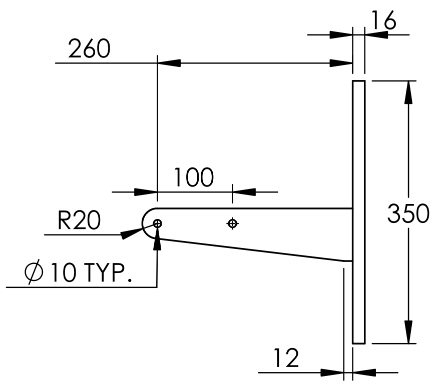
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



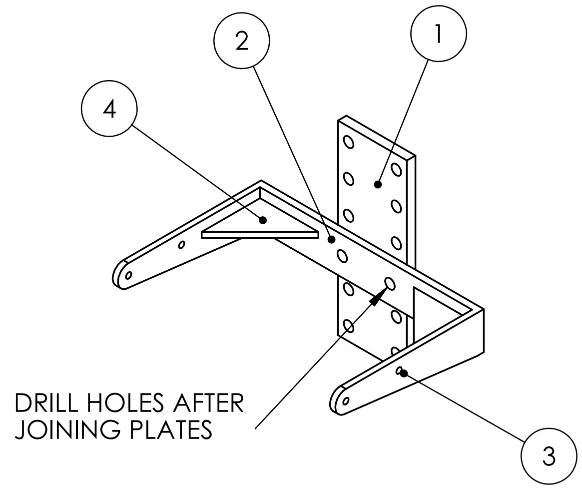
TOP VIEW



FRONT VIEW



SIDE VIEW

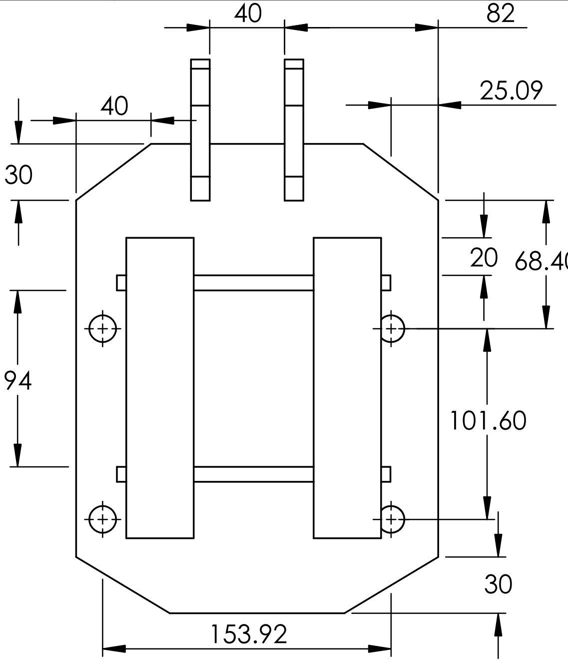


ISOMETRIC VIEW

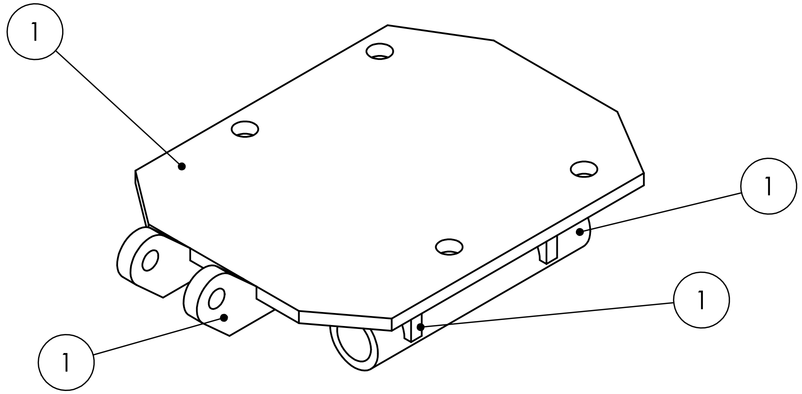
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:			DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING	REVISION
DRAWN	NAME	SIGNATURE	DATE		TITLE: VALVE SLIDE BODY	
	DANIEL CLARKE		12/10/06			
Q.A				MATERIAL: STEEL	DWG NO. 16	A4
				WEIGHT:	SCALE:1:10	SHEET 1 OF 1

ITEM	DESCRIPTION	SIZE	GRADE
1	8MM BED PLATE	193 X 220MM	350
2	8MM CONNECTING PLATE	146 X 40MM	350
3	Ø36 OD X Ø25.5 ID SLIDING TUBE	160MM	1020
4	8MM PIN PLATE	75 X 44MM	350

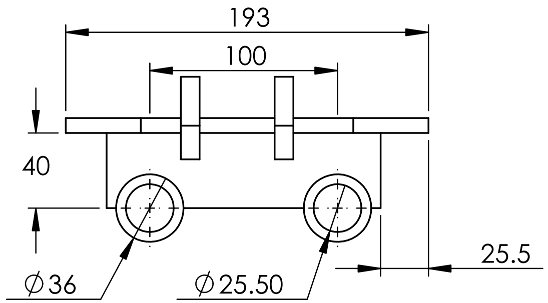
APPLY 8MM WELD BEAD TO ALL ACCESSABLE JOINS



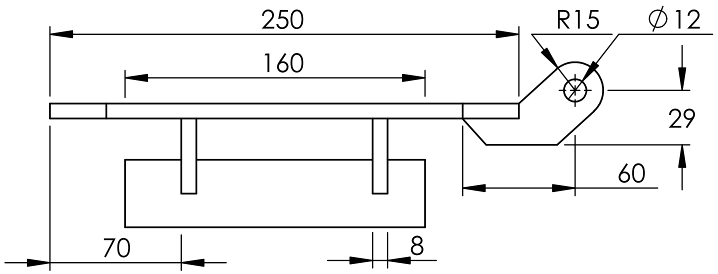
BOTTOM VIEW



ISOMETRIC VIEW



FRONT VIEW



SIDE VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:		DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
DRAWN	NAME	SIGNATURE	DATE	
	DANIEL CLARKE		17/10/06	
Q.A				

TITLE: VALVE SLIDE PLATE

MATERIAL: STEEL

WEIGHT:

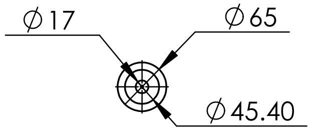
DWG NO. 17

A4

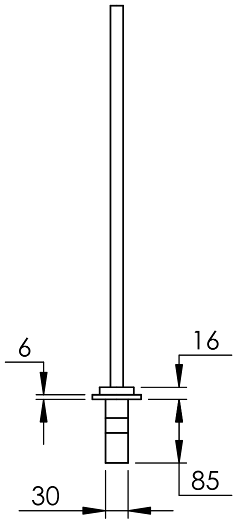
SCALE:1:2

SHEET 1 OF 1

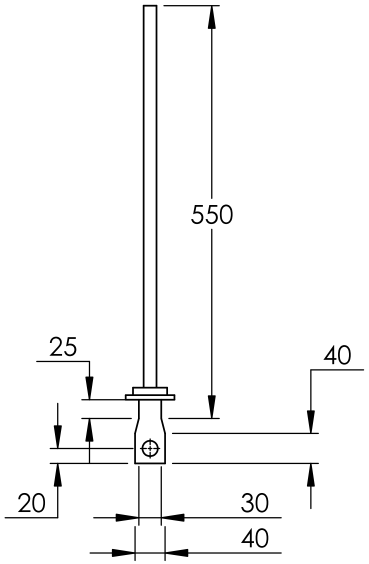
ITEM	DESCRIPTION	SIZE	GRADE
1	30MM HINGE BLOCK	40MM X 85MM	350
2	Ø65MM TURNED SPRING STOP	16MM	350
3	Ø 17MM SLIDING SHAFT	550MM	1050



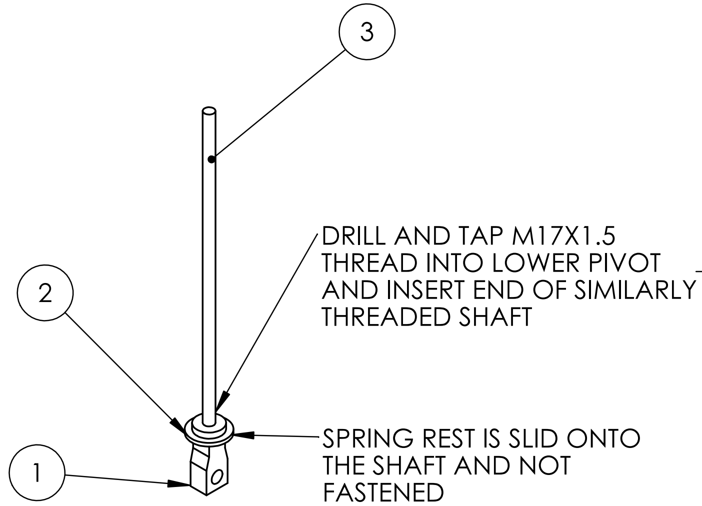
TOP VIEW



SIDE VIEW



FRONT VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES:
LINEAR:
ANGULAR:

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

TITLE:

LOWER SPRING REST

DRAWN	NAME	SIGNATURE	DATE
	DANIEL CLARKE		14/10/06
Q.A			

MATERIAL: STEEL

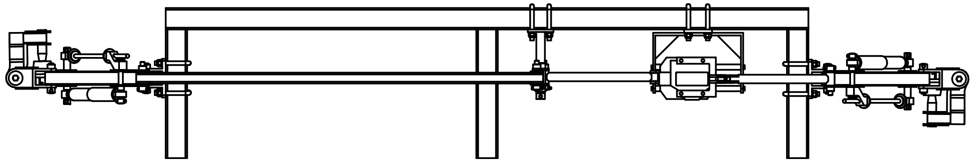
WEIGHT:

DWG NO. 18

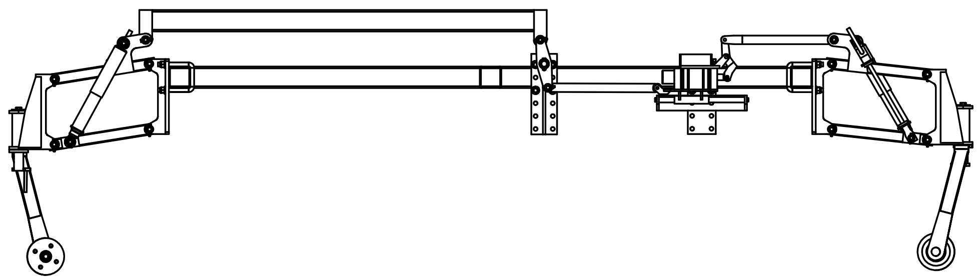
A4

SCALE:1:10

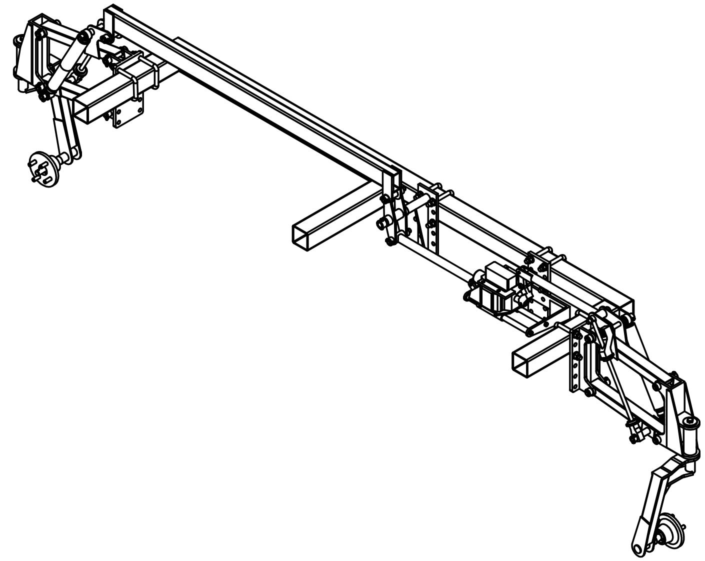
SHEET 1 OF 1



TOP VIEW



FRONT VIEW



ISOMETRIC VIEW

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS TOLERANCES: LINEAR: ANGULAR:		DEBUR AND BREAK SHARP EDGES		DO NOT SCALE DRAWING	REVISION
DRAWN	NAME	SIGNATURE	DATE	TITLE:	
	DANIEL CLARKE		24/10/06	ASSEMBLY VIEW	
Q.A			MATERIAL: STEEL	DWG NO. 19	A4
			WEIGHT:	SCALE:1:50	SHEET 1 OF 1