University of Southern Queensland

Faculty of Engineering and Surveying

Design of a rear suspension configuration for a live axle race car to achieve optimum handling characteristics.

A dissertation submitted by

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in the fulfilment requirements of

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towards the degree of

Bachelor of Engineering (Mechanical)

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Abstract

Looking at various rear suspension configurations, design a rear suspension configuration to replace the factory Ford Escort Hotchkiss style rear suspension with a design that will allow for ideal suspension geometry to be kept in check throughout variances in ride and roll as well as allowing for adjustability to allow the suspension to be tuned for certain track conditions as well as being capable of working in with different vehicle settings as a whole.

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Limitations of Use

Certification

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

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

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Signature

Date

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Glossary

Before getting in depth with the principles of suspension design some of the terminology relating to this topic must first be established.

Spring

The role of the spring is to absorb changes in road surface by compressing. The stiffness of the spring determines the harshness of the vehicle's ride.

Damper

The function of the damper or shock absorber is to suppress oscillation and control the motion of the suspension system.

Bump / droop

When the suspension is at its highest possible point it is said to be in full bump and at its lowest possible point it is said to be in full droop.

Camber

The angle in which the wheel points inward or outward from the vertical, viewed from the front or rear of the vehicle. It is basically a compromise setting to maximise tyre contact under all conditions.

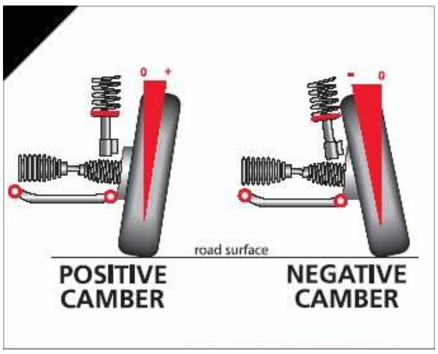

Figure 1.1: Camber
http://www.redranger.com.au/images/Faqs/Camber%20angle.JPG

Castor

The product of the inclination of the wheel fore and aft of the vertical, this directly influences the feel of the road through the steering wheel.

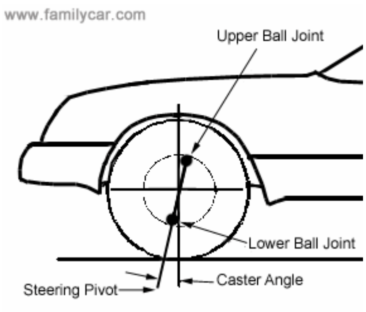

Figure 1.2: Caster angle
http://www.familycar.com/classroom/Images/Align_Caster.gif

Toe In /Out

The small angle in which the wheels point inward or outward when viewed from above. The toe ensures wheel stability under all conditions, braking, cornering, etc. The toe mainly relates to the front wheels.

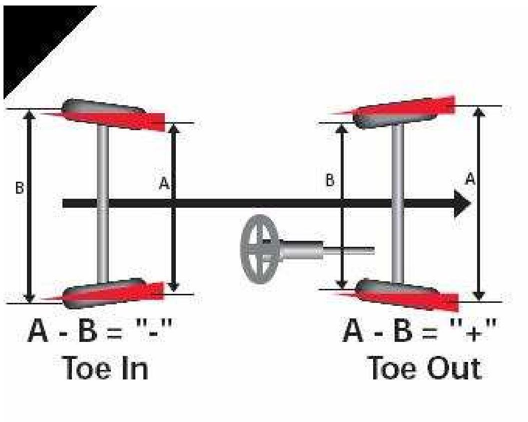

Figure 1.3: Toe in / toe out

Zero Steer

This is a term relating to four link trailing arm set-ups, but is rarely entirely factual. Consider a leaf sprung unlinked suspension, if one rear wheel is deflected in an upward direction the whole rear suspension would be inclined to twist. Four link arrangements minimise this effect by securing the axle from twist.

Roll

Rotation of the vehicle sprung mass about a fore aft axis with respect to a transverse line joining a pair of wheel centres. It is positive for clockwise rotation and is viewed from the rear (Dixon 1996).

Roll Centre

The instantaneous point about which the chassis will roll on its springs about a particular centre. Dramatic changes in a car's optimum cornering potential can be affected by altering the vehicle's roll centres.

Anti-squat and anti-dive

To withstand front end dive under hard braking or rear end squat under hard acceleration, suspension pivot points (such as four link bars) are usually angled to provide upward reactions in response to high wheel torque inputs. (Barstow, Whitehead,1993)

Pinion angle

Pinion angle simply refers to the angle of the differential's pinion in relation to the driveshaft. The goal is to create a straight line from the back of the crankshaft through the transmission, driveshaft, and the pinion of the differential. Although due to the tendency of the pinion to rise under load due to the suspension squatting as a result of a weight transfer to the rear of the chassis, some angle must be present at rest to account for this (Carcraft, n.d.).

Instant centre

Instant centre is the theoretical point through which tyre motion is transferred into vehicle motion. Setting up an ideal instant centre for a particular chassis is a critical factor in the design of any suspension system, particularly in that of a race car. Adjustments to the instant centre affect the anti-squat and anti-dive characteristics of the vehicle.

With a vehicle utilizing a parallel four link arrangement this point can be found by extending two theoretical lines out from each link bar out co-linearly and the point where the two lines intersect in the Instant Centre of the vehicle (Wikipedia, 2007).

Wind up

Axle wind up is the tendency for the axle to twist or wind up due to the torque inputs from the wheels causing the axle to turn on its own centreline.

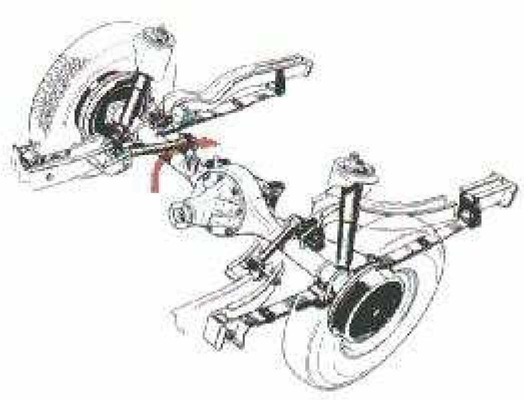

Figure 1.4: Axle wind up

Roll steer / axle skew

Roll steer or Axle skew is the unwanted twisting or turning of the rear axle in roll. This phenomenon can also occur due to torque from the drivetrain

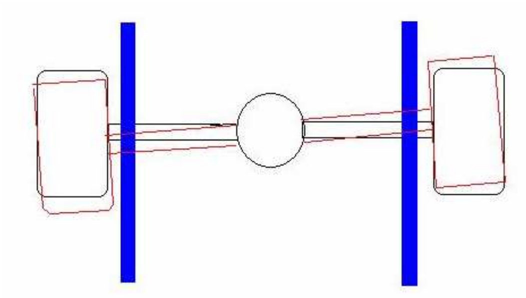

Figure 1.5: Axle skew or Roll steer

Tramp

Tramp occurs when the axle skews or winds up under hard acceleration, it is where the driven wheels hop up and down violently causing the whole car to shake and generally a loss of traction.

Sprung and unsprung weight

Sprung weight is the weight that is supported by the cars springs. Sprung weight includes the drive train, chassis and anything else that is supported by the weight of the springs. Components that are not supported by the springs are referred to as the unsprung weight of the vehicle. Unsprung components generally consist of items such as the wheels and tyres, brakes, hubs, axle assembly and differential.

Reducing the mass of the unsprung weight effectively increases vehicle control. The tyre and wheel assemblies react to road irregularities on the road surface, whereas the sprung components are relatively isolated from those occurrences. Essentially, the lower the weight of the sprung mass, the less the work that the spring and damper have to do (Gilles, 2005).

Suspesion geometry

The broad subject of how the unsprung mass relates to the sprung mass that it is connected to. These connections not only dictate the path of relative motion, they also control the forces that are transmitted between them (Millekin, Millekin, 1995)

Hotchkiss rear suspension

The standard rear suspension configuration found on the 1970's Ford Escort. Two leaf springs are used to attach the rear axle to the chassis providing both longitudinal and lateral location as well as providing the springing for the rear suspension.

Figure 1.6: Hotchkiss rear axle

Chapter 1

Introduction

A motor vehicle’s suspension system is the link between the wheels and the chassis, transmitting the weight of the vehicle on to the wheels. A suspension system must keep the wheels in proper camber and steer attitudes toward the road surface, resist chassis roll to an extent and keep the tyres in contact with the road surface with minimal load variations for the vehicle to handle in a desirable manner. (Gillespie, 1992)

Whether it be a family sedan with a comfortable and smooth ride for passengers on their daily duties or an all out race car where ride quality is considerably compromised in the quest for all out performance and road holding, there are many factors that need to be analysed in the suspension design of a motor vehicle. Some of the major factors include, camber, castor, toe in / toe out and roll centre. These will be discussed later in more detail.

Choosing the right suspension configuration for a particular application is crucial and usually compromises need to be made, whether it be in the geometry to allow for more space inside of the car or in the stiffness of the ride to achieve the desired handling characteristics planning a suspension system is a significant task.

The aim of this project is to look into improving the standard rear suspension setup on a 1970’s Ford Escort from an unlinked leaf sprung arrangement to a coil sprung linked set up that will be constrained both longitudinally and laterally to minimise axle skew and wind up and excessive chassis roll, allowing only two degrees of freedom for the configuration which in turn will allow the dampers and springs to work effectively in a straight up and down plane unaffected by any unwanted movement of the axle. The suspension will allow for adjustments to help achieve optimum roll centre and toe steer geometries for the circuit and anti squat geometry for the drag strip.

Chapter 2

An introduction to racing and vehicle dynamics

The main aim in motor vehicle racing is to achieve an overall vehicle configuration, which abides by the rules of a given class, which can cover a given race track in a minimal time, when piloted by a driver utilizing techniques within their capabilities. (Milliken and Milliken, 1995). There are many factors to be considered when designing and building a race vehicle, but for this case we will only be looking into the rear suspension setup.

The primary forces by which a high speed motor vehicle is controlled is developed in four small patches, this is where the four tyres make contact with the road surface. A knowledge of the forces and moments generated by the tyres at the ground is essential to understanding vehicle dynamics (Gillespie, 1992).

To maximize the tyre contact with the road at all times it is important to set up the suspension geometry correctly and select the correct springs and shock absorbers for the task at hand as evidently the more contact the tyres have with the road surface, the more grip or traction the vehicle will have which in turn will allow the car to travel at higher speeds more safely.

What upsets the dynamics of a vehicle whilst in motion? There are a number of factors which will now be discussed. The first and most obvious is the quality of road surface, as bumps and dips most certainly upset the handling of the vehicle. The second factor is the tire/ wheel assembly. Ideally the assembly is soft and accommodating to absorb small vibrations and bumps to some extent as well as running true, which essentially means it isn’t adding any vibrations to the vehicle in the form of unbalance in the wheel assembly, dimensional variations and stiffness variations (Gillespie 1992).

The next factor to be considered, which is a major factor that is being considered in this project in particular is driveline excitation. This is where the torque transmitted from the engine through the gearbox on to the driveshaft and into the differential will try to twist the rear axle assembly along its centre line which is referred to as axle wind up, this causes the axle to skew on its springs forcing the whole rear of the car to hop or tramp, this also tends to force the car sideways due to the skewing of the axle. The redesigned rear suspension will strive to keep this ‘drive train excitation’ to a minimum (Gillespie 1992).

When setting up a suspension system for a race car it is necessary to make the distinction between changes that will always improve the performance of the car and changes that will tune or balance the car. The first category consists of things like adding more power or getting better tyres, where as the second category consists of things such as making changes to spring rates and roll centre adjustments (Milliken and Milliken 1995).

Racing is all about driving a car at its limits. Finding a vehicle configuration which can be driven to produce this desired performance is referred to as setting up or chassis tuning. It is a difficult task requiring many compromises to fit each circuit and the driver’s capabilities and preferences. Often it will not be possible to achieve the optimum setup in the available time for testing prior to a race event and the driver will have to make do with any deficiencies that remain in the vehicle (Milliken and Milliken 1995).

The principal objectives in setting up the car are achieving a good cornering balance (neutral steer), finding a compromise between cornering and drag on high speed circuits and eliminating specific control and stability problems at any points on the race circuit as reported by the driver (Milliken and Milliken 1995).

There are many factors to be considered when designing the suspension set up of a race car, as each component and adjustment will have an effect on the way the vehicle handles and will also have an effect on the manner in which the other related components will operate. The effects of weight shifts, brake bias, roll stiffness, roll centre, anti dive, camber are all factors that need to be carefully looked into to achieve a desirable set up (Milliken and Milliken 1995).

Two of the major objectives of this project are to achieve optimum anti squat and roll centre geometrys, both of these geometrys will be adjustable to allow for minor tweaks to optimize the use of the rear suspension system in different circumstances.

Roll centre heights partially determine the way the roll moment on the car from lateral force is distributed. Lowering the roll centre on the rear of the vehicle will lower the roll moment resisted by that end; the wheels on that end will be more evenly loaded in cornering compared to the front of the car (Milliken and Milliken 1995).

To set up anti squat, the effective point pivots or suspension travel paths of the tyre to road contact need to be known to calculate the optimum angles for this geometry to be achieved. (Bastow, Whitehead. 1993)

The goal in setting up anti squat is to transfer just enough force to the tyres to keep them from loosing traction and then allow the rest of the force to push the car forward. If an imaginary line is drawn through the lower link bar in a forward direction toward the front of the car and another imaginary line is then drawn through the upper link bar forward until it intersects with the lower line, the Instant Centre (IC) is found. (Bastow, Whitehead. 1993)

Now imagine that the Centre of Gravity (CG) of the vehicle is concentrated at the gearstick. Where the IC is located when compared with the CG is what determines how the force applied to the rear suspension from the drive train acts on the vehicle to get it moving. If the IC is too high then there will be too much energy wasted pushing the rear of the car skyward. If the IC is set too low then there will not be enough force applied to the rear tires and as a result there will be excessive wheel spin. There are also variables if the IC is in front of the CG or behind the CG. Somewhere there is going to be a position that will apply just enough force to the tires to keep them from spinning and the rest of the force will push the car forward, this can be determined using anti squat calculations, filling in the variables for the particular car in question. (Bastow, Whitehead. 1993)

Once the roll centre and anti squat geometry is optimised is it then time to look at spring rates. Ride spring rates affect the lateral load transfer distribution. Lowering the spring rate on one end of the vehicle will even out the loads on that particular end and this will raise the lateral force available from that end. The fitment of stiffer springs than what is found on a production car will generally limit the amount of body motion in pitch and roll, this is desirable for a race car. Race cars generally wind up with the un-driven end of the car very stiffly sprung relative to the driven end, this keeps drive wheels more evenly loaded for traction (Milliken and Milliken 1995).

2.1 Control, stability and handling

The rest of this chapter will endeavour to introduce some other factors that will directly affect the performance of the new rear suspension.

The manipulation of the controls by the driver intending to influence the motion and or direction of the car is referred to as control, as the driver is attempting to control the path and speeds in which the vehicle is moving. The reluctance of a car to allow a change in path is referred to as stability, in moderation this is generally a desirable attribute. The ability of a car to go around corners effectively, the study of how this take places as well as how the driver perceives the vehicle’s behaviour whilst cornering is known as handling. Considered as a whole, the vehicle can only be influenced by forces exerted on it by the road, the atmosphere and by gravity. Due to this, the cornering characteristics of the tyres and the forces imposed on them are essential to consider in the design of any car, the aerodynamic properties of the body are also important to consider (Dixon, 1996).

The tyre has the very important role of being the only component on the vehicle which is in contact with the road surface. The ability for the tyres to grip to the road surface and hence the forces that are exerted on the tyres by the road will depend on many factors. Some of these factors include, the structure and materials in which the tyres are designed and manufactured, their width and profile and the angles in which they are presented to the road, which is where the suspension design becomes of importance (Dixon, 1996).

It is important to consider the tyre itself in detail, although without a suspension system that doesn't keep the tyres evenly loaded and at proper camber and steer angles toward the road, the car will not work properly as a package. The study of vehicle dynamics involves looking both lateral and longitudinal motions of the vehicle. Lateral motions, which consider how lateral accelerations (in cornering) affect the balance of the chassis is the main area of interest (Dixon, 1996).

The ability for a car to perform a specified manoeuvre assuming a flawless driver is referred to as roadholding. Handling is essentially an assessment of how a car will consistently react to driver inputs. The handling of a particular car will then rely on both the driver and the environment in which the car is being tested (Dixon, 1996).

Due to the fact that the aim of this project is to design a rear suspension configuration that will allow for optimum handling characteristics, achieving a suspension that is capable of high levels of roadholding and handling will be considered over that of achieving one with good ride quality, which is the capability for a suspension to dampen out harshness inside the cabin due road roughness.

2.3 Body stiffness

When considering the design of a suspension for any car it is firstly important to consider the stiffness of the car's chassis / body, as it is worthless going to all the effort of designing a suspension system to constrain the rear axle from unwanted movement if the chassis is too weak to resist the forces imposed on it from the suspension, as the chassis will flex, effectively making the implementation of the new suspension a waste.

Therefore from a handling perspective it is desirable for a car's chassis to have a high stiffness, thus allowing the suspension and tyres to work as they were designed without any negative influencing factors such as chassis flex. The principal loads transmitted from the suspension on to chassis are the vertical ones. These loads are transferred on to the chassis of the vehicle through the suspension links, springs and dampers and are fundamentally the same as the vertical forces in which the tyres are subjected to.

In the case of roll skew in cornering, the sums of the vertical forces on diagonals of the chassis are not equal and the chassis is in torsion. It is common in suspension design for the chassis to set up such an effect so that the normal forces imposed on the tyres affect the lateral forces in a desirable respect (Dixon, 1996).

So to a large extent the torsional rigidity of a chassis / body can affect the handling performance of a vehicle, especially in case the where the chassis is lacking in stiffness as this will reduce the overall performance of the suspension as it will not be functioning in the way it that was designed. A chassis that flexes will also be prone to fatigue and further weakening due to stress reversals during use, which may eventually lead to breakages (Milliken, Milliken, 1995).

In most cases for a car to race in a particular class it is necessary for the car to be fitted with a six point steel roll cage as per the CAMS specifications for the particular class in which the car is being raced. A full roll cage as shown below will obviously strengthen the chassis up a large amount. If allowed in the rules picking up suspension points with the cage is also a great thing, as it will greatly improve chassis rigidity as the cage is further strengthening the points in which the forces from the road are directly transmitted to the sprung mass.

Figure 2.1: 6 point roll cage
http://www.autopowerindustries.com/Images/lg/DSC_2840.jpg

Seam welding the chassis rails and other structural members to the body shell will also add to the rigidity of the chassis and is a worthwhile venture if allowed by the rules for the particular class in which the car is to be raced.

2.3 Differentials

The differential (commonly referred to as the diff) basically transmits the torque transmitted from the driveshaft on to the wheels. There are three main types of differential to be discussed here, the open differential, the locked type (known as a spool) and the Limited slip differential (LSD). The type of diff chosen will greatly affect the amount of traction a car will have.

Most factory road going cars are fitted with open differentials. An open differential works in the following manner. Refer to the figure below and while reading the following. A pinion gear which is rotated by the drive shaft engages with the much larger ring gear or crown wheel. The ring gear drives a small set of three planetary gears which are designed so that the two side gears can turn in opposite directions to one another. These two side gears turn the axle half shafts, which in turn drive the wheels (Wikipedia, 2007).

The reason for the two outer planetary gears being able to rotate against one another is so that the right and left wheels can rotate at different speeds when the car is negotiating a corner, although the same torque is supplied to each of the axle shafts under all conditions.

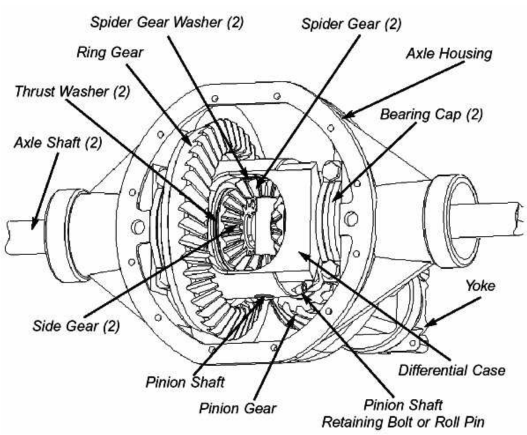

Figure 2.1 Open differential
http://www.offroaders.com/info/tech-corner/project-cj7/images/locker/open-differential-parts-id.jpg

The torque on each wheel is a result of the engine and transmission applying torsion against the resistance of the traction at that wheel. Unless the load is exceptionally high, the engine and transmission can usually supply as much torque as necessary, so the limiting factor is usually the traction under each wheel. It is therefore convenient to define traction as the amount of torque that can be generated between the tyre and the ground before the wheel starts to slip. If the total traction under all the driven wheels exceeds the torque required to push the car forward at any particular instant, the vehicle will be driven forward; if not, then one or more wheels will simply spin (Wikipedia, 2007).

The limitation of the open differential is due to the fact that it will always allow the wheel of lesser resistance to spin faster relative to the other wheel.

For example in a left hand turn the weight of the unsprung mass will tend to shift to the right hand side of the car in roll therefore applying a greater force into the road to the right wheel and effectively a lesser force into the road on the left wheel, this will generally to cause the left driven wheel to spin or loose traction. In this process the planetary gears inside the diff will start to rotate against each other as the loaded wheel (right) has a greater resistance from spinning than the left wheel. The open differential will allow the left wheel to spin faster, with the right wheel maintaining its current speed.

Note that the open differential supplies the same amount of torque to both wheels, so both wheels are striving to push the car forward the same amount, although due to the force required for the left wheel to break traction being much less than the right wheel it will effectively spin much faster relative to the right wheel (Wikipedia, 2007).

This has led many to believe that the open differential is only one wheel drive, although this type of differential always provides equal torque to both driven wheels. Due to varying levels of traction at each of the driven wheels as a result of variances in road surface, weight shifts and twisting torques from the engine and driveline etc the open differential is not desirable for racing conditions.

Enter the Limited slip differential. The most commonly used form of LSD is the clutch type LSD. This differential uses clutch plates which are used to limit the speed difference between the side gears and thus the two wheels. By restricting the speed difference between the two driven wheels, useful torque can be transmitted to both wheels as long as there is some friction available on at least one of the wheels. The LSD is the most commonly used form of differential in circuit racing cars (Wikipedia, 2007).

The locked differential or spool does not contain planetary gears and the speed of the two wheels will always be the same. The major limitation of the spool is that due to not allowing the wheels to rotate at different speeds, the car is harder to turn into corners as front end understeer is heavily promoted due to the fact that the two driven wheels will always be trying to push the car in a straight line. Spools are most commonly used in drag racing and low budget circuit race cars as they are much less expensive than LSD's.

Chapter 3

Discussion of different rear suspension configurations.

3.1 IRS and Live axle configurations

In this project we are only considering the rear suspension setup of a rear wheel drive car. The two main types of rear suspension used in motor cars are either the Live (solid) axle arrangement or the Independent rear suspension (IRS) configuration.

Figure: 3.1 Hotchkiss live axle configuration

(This is very similar to that used in a Ford Escort)

Figure 3.2: Independent rear suspension

Most modern passenger vehicles today use IRS, due to the capability of each wheel travelling independently of one another a smoother ride can be achieved. Although most load carrying vehicles still retain live axle arrangements due to their better weight bearing capabilities.

Many race cars still utilize a solid rear axle configuration. Even though in theory an IRS suspension setup is superior, not many race teams have built and implemented an IRS car that has proven to be successful on the race tracks against a well set up live axle car. This is mainly due to the fact that the IRS setup is much more complicated and thus much harder to set up when compared to the much simpler live axle setup.

The simplicity of the live axle means that in many ways it can’t be beaten, the major drawback is in the un-sprung weight of the setup, it will always be quite heavy and there is not much that can be done to reduce it. On independent systems the centre section (consisting of the crown wheel, pinion, differential unit and case) is fixed to the chassis of the vehicle thus meaning that the unsprung items will only be half of every linkage going to the wheel hubs and then the hubs and wheels themselves. So this allows the IRS setup to achieve less unsprung weight than a live axle setup.

The problem with most IRS setups fitted to cars is that passengers need to be seated in places where engineers want to position inboard pickup points, so the geometry is and always will be a compromise and as a result of this the camber changes can be horrific from a traction point of view. The drawing below shows the drawback, the tyres footprint is not kept flat due to the wishbones on the IRS setup moving in an arc type fashion. The live axle setup is shown on the left and the IRS setup is shown on the right. As you can see the camber changes due to the IRS suspension travelling in arcs causes camber changes in both roll and in squat. Lifting the tyre from the ground means that there will be less traction available, so in many cases the driver of the car will not be able to use the power available from the engine.

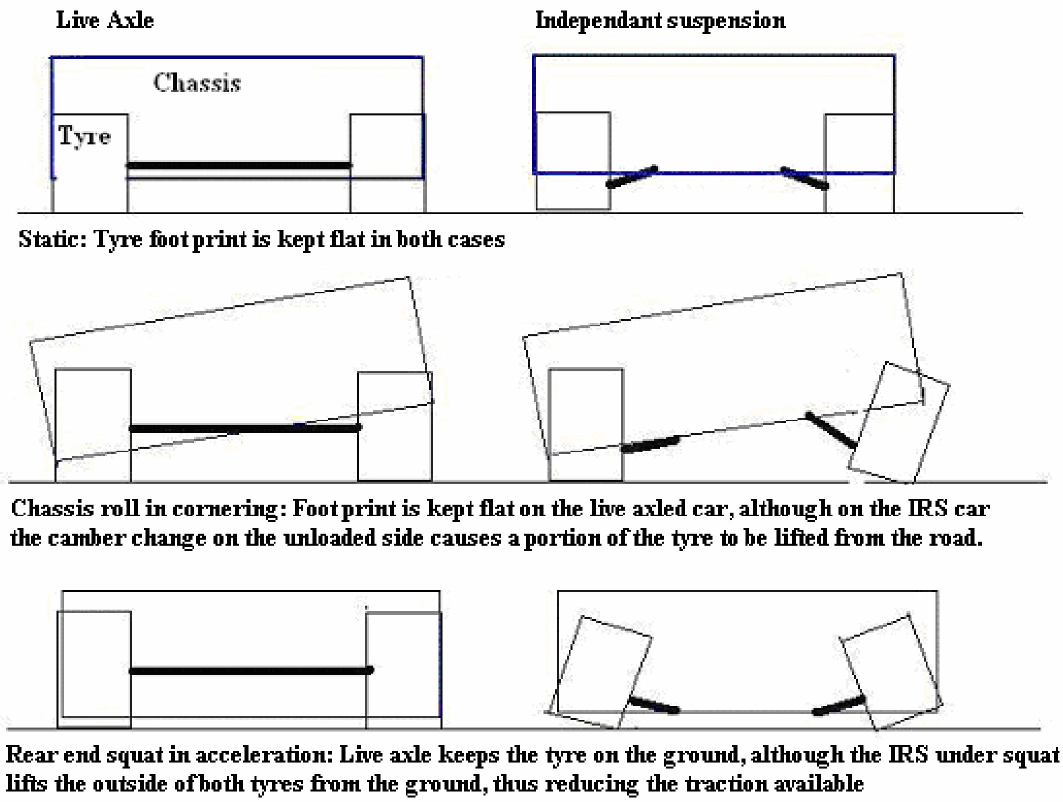

Figure 3.3: Live axle Vs IRS

There are of course ways to minimize the camber changes in an IRS setup, but due to the simplicity of the live axle setup and the fact that the Ford Escort is already fitted with a live setup this project will be based around improving a live axle setup only from here on in.

3.2 Commonly used rear suspension components

<u>Leaf springs</u>

Figure 3.4: Leaf springs
http://img.alibaba.com/photo/50595287/Leaf_Springs.jpg
http://www.4wdworld.com.au/products/4x4suspensions/suspension_02.jpg

Leaf springs are possibly the simplest and least expensive of all suspension setups. They locate the axle in both longitudinal and lateral directions, attach it to the chassis of the vehicle, as well as acting as a spring for the suspension system. While compliant in the vertical direction, the leaf spring is fairly rigid in the lateral and longitudinal directions, thus reacting the various forces between the sprung and unsprung masses.

Leaf springs were used on most passenger cars into the 1970's and are still used on most small trucks and utilities today. The major downfall for leaf springs is that they are quite stiff, which leads to a harsh ride. When passenger vehicle manufacturers tried to soften up spring rates there were notable losses in side stability of the springs. With softer springs, the vehicles would keel over in cornering causing axle skew and the axle would windup or twist under braking and accelerating conditions in more powerful vehicles. (Gillespie, 1992)

Coil Springs

Figure 3.5: Coil springs
http://www.dow.com/amplify/images/springs_72dpi.jpg

The coil spring is the most common type of spring used in passenger vehicles of today. This spring is simply a steel bar that has been bent into a flexible coil. The spring functions by compressing in and recoiling back to its original height to absorb shock forces. Unlike leaf springs, coil springs are not used to locate the axle, they solely are there to act as a spring. In the case of a coil sprung suspension system the axle must be located in the lateral and longitudinal directions by some other means. Coil springs can be located in between wish bones, chassis and control arms or fitted to most strut assemblies.

Radius arms

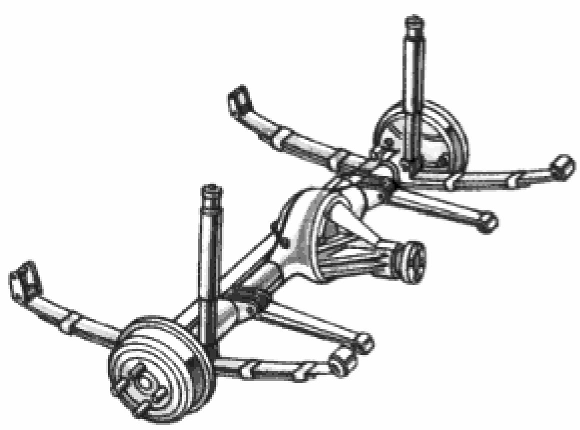

Figure 3.6: Radius arms
http://www.pixelmatic.com.au/cortina/mk1pix/rearends.gif

Generally used as an upgrade on leaf spring cars to prevent axle wind up and tramp, radius arms constrain the axle from winding up under torque and skewing in body roll. They are an improvement over an unlinked leaf sprung suspension setup, although they will still allow the axle some freedoms in suspension travel that are not necessarily wanted.

Four link

Figure 3.7: Converging four link
http://www.popularhotrodding.com/tech/0604_rear_suspension_guide/photo_07.html

There are two basic types of four link, the first is the converging type. The main purpose of the bottom two links is to locate the axle in a longitudinal sense, while the upper two links ensure that the pinion angle and anti squat geometry is fixed throughout suspension travel as well as locating the axle in a lateral sense.

The major drawback with this system is that when pushed to the limit the setup suffers from significant bind due to the angled upper arms, and this can force the setup to be unpredictable (Lateral dynamics, n.d.).

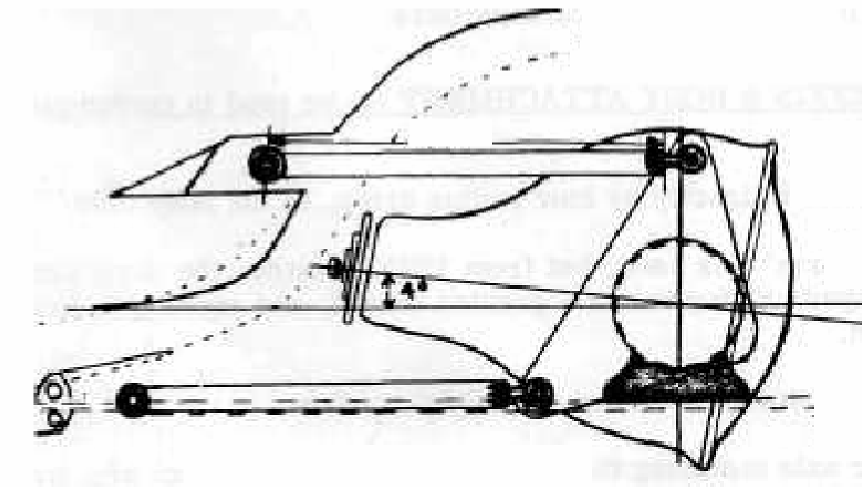

Figure 3.8: Parallel Four link

The parallel four link system consists of four forward facing bars which are parallel to the each other and protrude forward at 90 degrees to the axle housing. The link bars are free to pivot at each end and attach the axle housing to the chassis of the vehicle, therefore locating the axle in the longitudinal direction and effectively determining the castor of the rear suspension as well as the squat/ anti squat geometry.

Most factory four link arrangements are compromise setups, as for optimum suspension geometry to be set up and retained the upper link bars really need to be as long as the lower link bars and for this to occur the upper link bars generally need to penetrate the rear floor where the rear seat would usually be fitted and for this reason short upper links or converging upper links are used.

When a parallel four link setup is used the axle will also need to be located in a lateral sense, with either a panhard rod or a watts linkage.

Three Link

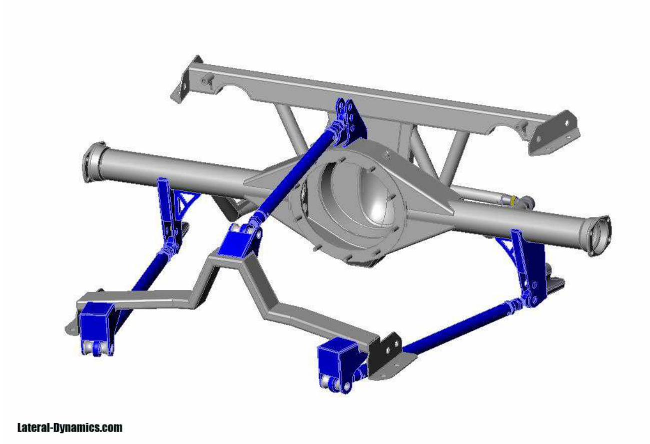

Figure 3.9: The Lateral Dynamics Three link

A three link rear suspension setup is a design which uses three forward facing control arms to attach a solid rear axle to the chassis of a vehicle. The two bottom links are fitted in the typical location as used in the four link arrangement, although there is only one single top link instead of the two used in a four link setup. The top link is attached to the top of the differential housing in the centre. It is said that this setup is 'kinematically free' of bind in roll.

Much like the parallel four link setup, the axle will also need to be located in a lateral sense, with either a panhard rod or a watts linkage.

Torque arm

A torque arm setup is similar to a three link setup in that it uses two lower forwards facing control arms as well as having a third link that extends from the centre of the differential housing out towards the front of the car. Although a torque arm generally extends all the way to the gearbox cross-member. This setup also requires to be laterally located (Crankshaft coalition, n.d.).

Below is a pic of the RRS bolt on rear end for Falcons and Mustangs, it features a long torque arm as shown.

Figure 3.10: RRS torque arm setup

Panhard rod

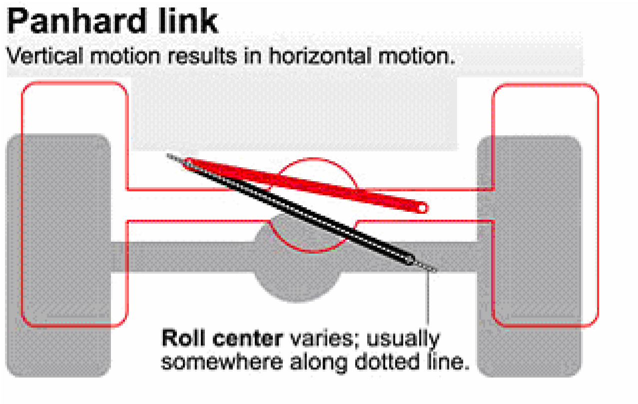

Figure 3.11: Panhard rod
http://www.lateral-dynamics.com/products/source/_PanhardWattsWeb.jpg

A panhard rod locates the axle in a lateral sense, constraining the axle from side to side movement. The Panhard rod is very simple in design, although it is disadvantaged by the fact that the axle must move in an arc in relation to the vehicle's chassis. The panhard rod consists of a single bar the pivots from the axle housing and the chassis.

Watts linkage

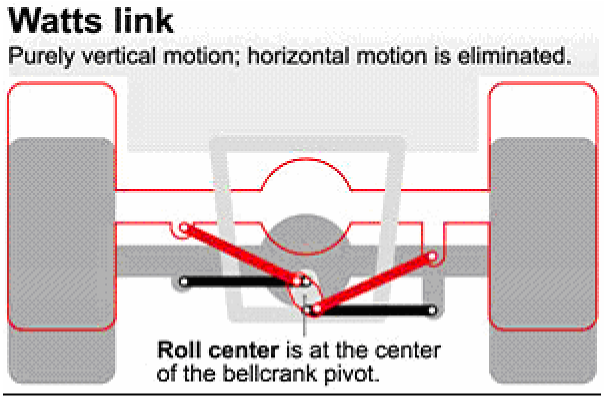

Figure 3.12: Watt's linkage
http://www.lateral-dynamics.com/products/source/_PanhardWattsWeb.jpg

The watts linkage is an improved method of laterally locating a live axle over the panhard rod arrangement, ensuring absolute vertical motion of the axle, rather than moving in an arc type fashion as with the panhard rod.

Chapter 4

Selecting a rear suspension configuration

4.1 Overview and limitations of the factory suspension

As discussed previously, the focus of this project is to improve on the standard rear suspension setup on the Ford Escort from an unlinked leaf sprung arrangement to a coil sprung arrangement that will constrain the axle in both longitudinal and lateral directions, thus permitting only two degrees of freedom and allowing for minor adjustments in areas such as roll centre and anti squat / anti lift. The chosen suspension design will also make certain that the desired axle positioning is kept in check throughout suspension travel.

If the new system is designed correctly and implemented the car will be much more stable on the road especially under hard acceleration and in both entering and exiting corners, allowing the car to be much quicker on a race circuit.

Figure 4.1: Leaf sprung Hotchkiss axle

The standard rear suspension setup on an Escort, as pictured above is one of the most simple rear suspension configurations. It may have worked well for a standard road going passenger vehicle of that era, but when more power is extracted from the engine and other modifications are carried out to make the car faster on a race circuit, it quickly becomes obvious that the rear suspension is in need of attention. The car will tend to get out of control easily due to the back axle tending to wind up or twist under hard acceleration, causing rear steer forcing the car in any direction bar the desired one.

A quick note on how a leaf spring operates, referring to the figure below. A leaf spring is located at two points on the chassis. The rear of the leaf is affixed to what is referred to a spring hanger. The spring hanger is attached to the chassis and is able to pivot when the spring moves, as due to its semi elliptical design when the spring is loaded and compresses it effectively gets longer, moving the spring hanger rearward and vice versa for when the spring decompresses.

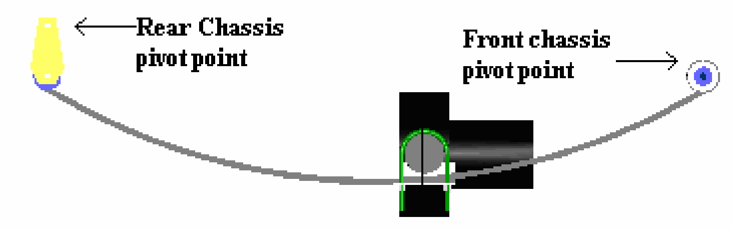

Figure 4.2: Escort rear suspension

Axle wind up is more extreme in Hotchkiss style suspension configurations such as that found on the Ford Escort due to the fact that the axle is relying on the leaf springs to not only control the springing, but to locate the axle as well and thus resisting torque inputs imposed them from torque being delivered through the drive train and then on to the wheels. Due to the locating device for the axle being a spring it will deform elastically as a result of the wheel torques allowing for the axle to wind up more easily than that of a configuration in which the axle is constrained by rigid links.

Another major issue that arises when using an unlinked leaf sprung rear end is that when the car keels whilst cornering, for example if the car is corning hard right, the car will try and resist the change in direction and thus there will be a significant weight shift on to the left side of the car. Now baring this in mind and referring to the diagram below, the leaf on the loaded (left) side flattens, while the leaf on the unloaded (right) side curves elliptically in an upward direction, as displayed in the diagram below left. The leaf on the loaded side effectively becomes longer and the leaf on the unloaded side becomes shorter, this forces the axle to skew (As shown below right) forcing the car into oversteer undesirably.

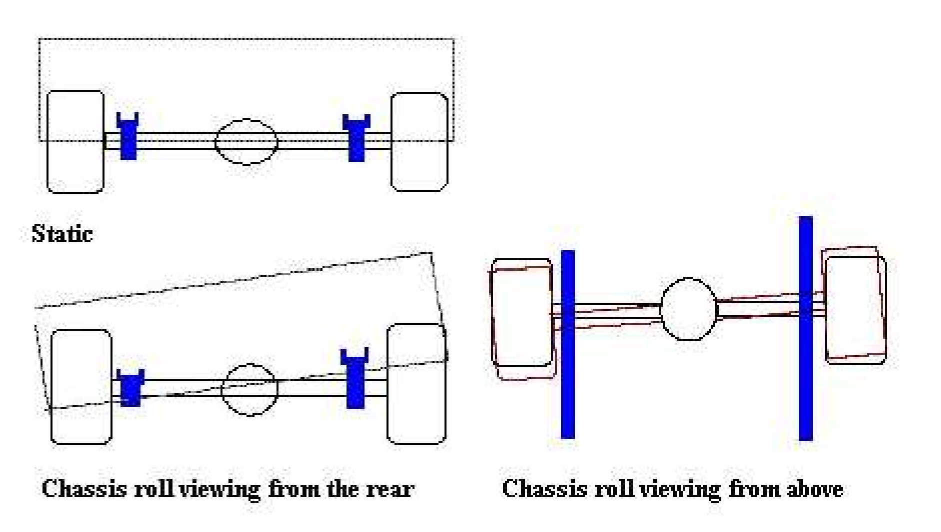

Figure 4.3: Limitations of leaf springs in roll

At this point it is necessary to look at ways to improve on these matters. The first natural thought would be to stiffen up the springs and or to add a rear anti roll bar to prevent the car from keeling or rolling on its springs whilst cornering. This is all well and good in theory, as if the cars keel in cornering is kept to a minimum then the axle skew or roll steer will be kept to a minimum and this is correct, although this then introduces another issue. If the rear end resists keel and the body doesn't roll in cornering the rear end will become very sensitive to driver inputs and unpredictable with generally large amounts of uncontrollable oversteer, which is a less desirable characteristic than that of roll steer.

If nothing can be done with the springs to improve the situation significantly it is now time to look into ways of linking the axle to constrain the it from skewing under load and in roll.

There are many small improvements that can be made to improve the leaf sprung rear end arrangement, drag racers use devices such as caltracs or slapper bars to help reduce wheel tramp, which is high frequency up and down movement of the rear axle under hard acceleration and sometimes deceleration due to axle wind up and there are of course items such as radius arms, four links, three links, panhard rods etc can be used in conjunction with leaf springs to improve the handling of the vehicle, although in this case it has been decided to change to a coilover setup in the rear, thus doing away with the leaf springs all together.

4.2 Coil springs and rear spring / shock absorber location

Coil springs are a far superior design of spring than that of a leaf spring, the sole purpose of the coil spring is to act as exactly that, a spring. Therefore coil springs do not locate the axle and this allows the use of much softer spring rates possible if needed without having to worry about axle wind up and axle skew in body roll. Coil spring arrangements also allow for easier interchange-ability and adjustable ride height in cases where a coil over strut assembly is utilized.

There are a few different ways to mount up coil springs, although in this case a coilover shock type arrangement will be used as it is simple, readily available and allows for easy ride height adjustment by simply lowering or raising the threaded spring platforms.

Figure 4.4: Coilover shock absorber assembly

Most cars of today have a vertical coil over damper setup from the factory, although the Escort did not. The factory dampers were angled inward, possibly to avoid interfering with the rear seating area. A damper will work more effectively mounted in a near vertical position rather than at a steep angle as a dampers performance will reduce the more inclined it is. For instance if a damper is at a 45degree incline and the axle moves 50mm, the damper will only move 25mm. If the damper is mounted vertically when the axle moves the damper will move the same amount, which will allow the damper to perform its job more effectively.

Therefore to mount the coilovers vertically on a Ford Escort it is necessary to house the coilover struts in a fabricated housing or turret which will be mounted to the inner section of the factory wheel tubs, allowing for the coilovers to be mounted vertically as opposed to the angled setup used from the factory. Also by doing this the shock absorber top mount will be mounted much higher than the factory Escort setup which will allow for more suspension travel than the factory setup which can become very minute when the ride height is lowered dramatically.

Common sense will denote that the closer the coilover struts are mounted to the wheel hubs, the smaller the leverage effect and therefore force moment on the springs and dampers will be generated in suspension travel and this inturn will allow the car's roll stability to be increased. The coilover turret is generally fitted to the inside of the wheel tub as this is as far outboard as the struts can be mounted without affecting the width of wheel that can be fitted under the wheel arch, as of course the tyres are the sole components that all the moments and forces from the road surface are initially transmitted through, so it is fair to say that the size of the tyre contact patch on the road is a factor that cannot be compromised anymore than it already is.

Figure 4.5: Turrets (left) Turrets fitted to the inside of the wheel tub (right)

Now that the leaf springs have been made redundant, there now is no way of locating the axle in both the longitudinal and lateral directions, so now it is necessary to adopt a linkage set up to constrain the axle.

4.3 Longitudinal location

Firstly considering locating the axle in the longitudinally, the following discussions will lead to a decision on the configuration in which will be used to locate the axle longitudinally and resist wheel torque inputs.

4.3.1 Radius arm and torque arm configuration

The first setup to consider would be the simple radius arms. These are something that can readily be purchased and are a fairly simple kit to fit.

Figure 4.6: Escort radius arm kit

Attaching the top of the differential housing to the chassis rail this setup serves a purpose, helping to reduce axle wind up. Although, the bars are too short to allow for optimal geometry and are designed to be used in conjunction with the factory leaf springs. When using coilovers there needs to be a method of resisting the torque inputs from the drivetrain which want to force the axle to wind up along its centreline and therefore at least one more link is required to constrain the axle from the underside.

As shown earlier below an example of a rear end configuration designed by RRS for Falcons and Mustangs which they refer to as their 3 link rear suspension setup.

Figure 4.7: Torque arm

This setup uses two lower radius arms similar to those which have just been discussed to position the axle in a longitudinal sense and then a torque arm / tube is used to control torque inputs from the drive train along with the pinion angle and the anti squat of the axle. The torque arm is rigidly attached to the top and bottom of the centre of the differential housing and extends forward just beneath the drive shaft to its own cross member where it is free to pivot up and down. The torque arm crossmember affixes in between the chassis rails just behind where the gear shifter would be found. The reason for using such a long arm is so that the arm acts very close to the centre of gravity of the chassis, which will allow for very good anti squat geometry to be set up. Another advantage of this type setup is that the floor does not need to be cut up for protruding link bars etc.

Torque arm setups can be very harsh due to torque inputs from the rear axle due to being rigidly attached to the axle housing. This type of setup is more of a drag racer oriented type arrangement due to the fact that there is too much anti squat geometry dialled in for circuit duties. When a car has anti squat geometry the rear of the vehicle will always be very stable and firm on the road, although the front suspension will feel loose and cause the car to understeer when attempting to turn in on corners. Torque arm setups generally do not allow for good ground clearance either as they have to hang below the drive shaft generally making the torque arm the lowest point of the vehicle.

4.3.2 Truck arm configuration

Nascars use what is referred to as a truck arm setup. This setup was originally found under a number of GM trucks in the 1960s and are now used by the majority of Nascar leagues. The truck arm configuration consists of two long steel bars which are rigidly attached to the axle housing and protrude forward mounting to the chassis where they are free to pivot.

Figure 4.8: Truck arm configuration
http://www.psrstreetrods.com/rear_suspension.htm

The rigid attachment of the arms to the axle housing forces the suspension setup to resist chassis roll completely, thus leaving the axle only one degree of freedom. The only time that there would be any chassis roll would be from flex in the front bushings (which would cause the bushes to wear rapidly) or flex in the actual arms themselves.

4.3.3 Three link arrangement

Another method of locating the rear axle to consider is the three link arrangement, which consists of two lengthy lower link bars which extend forward and one upper link bar of similar length which also extends forward as pictured below.

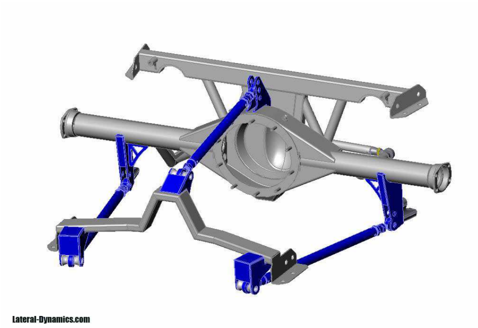

Figure 4.9: 3-Link by Lateral Dynamics

This arrangement is far superior to the above configurations which have just been discussed, although the first major drawback is that the top link bar needs to protrude into the floor pan which affects the rear seating area. For this reason this type of configuration is generally ruled out for many street cars, due to the primary purpose of a street car being to carry passengers, although in racing rear seating capacity is not needed and is generally impossible due to the fitment of a roll cage as per race regulations.

In the search for optimum handling characteristics many sacrifices will need to be made and link bars protruding into the rear floor area is just going to be one of them.

The 3 link arrangement pictured above is manufactured by a company known as Lateral Dynamics and their system uses two lower control arms to locate the rear axle longitudinally and a third top link which is mounted in the centre of the differential housing which constrains the axle from wind up. The best thing about this arrangement is that the axle is said to be “kinematically free” which means that there is no binding of the arms resisting each others in suspension travel and particularly in roll.

This system uses very long control arms relative to what is found on most other similar configurations. Most other configurations are compromise setups which use shorter arms to prevent having to protrude into the rear seating area, although the use of shorter arms will limit the control that the link bars will have over the axle.

The link bars need to be made as long as practically possible, as when the suspension moves through bump and droop (travel) the arms that are constraining the axle are moving in an arc, thus the longer the arms the less dramatic the changes in the caster positioning of the axle assembly will be. This in turn allows the chassis to be more stable. Effectively the longer the arms are the less axle skew / roll steer there will be throughout the suspensions travel.

This consequently allows the handling of the vehicle to be far more predictable in its handling characteristics.

Configurations such as the 3 link offered by Lateral Dynamics allow for adjustability, which is needed on a race car as small changes in the cars setup and different track conditions call for different suspension geometries to be used. A small change such as ride height can put the geometry of the 3 link out of its optimum range.

4.3.4 Parallel four link

The next method of longitudinally locating a live axle assembly is the four link arrangement. As previously discussed the parallel four link is the better of the two main types of four link arrangements as it does not suffer from bind in bump and droop in the same fashion that the converging type does.

Figure 4.10: Parallel four link arrangement

This method of locating the axle longitudinally is very similar in theory to the three link arrangement which was just discussed, although it features two upper link bars mounted vertically above the two lower link bars rather than just one located over the centre of the differential housing. The four link allows for more adjustability than the three link.

This setup much like the three link which was previously discussed will also need to protrude the rear floor pan area to achieve the desired geometry and link bars should be made as long as practically possible to minimize axle skew in suspension travel.

It is desirable to mount the link bars on either side as far outboard as possible to reduce leverage forces that are imposed on the link bars from wheel torque inputs.

The link bars can be made adjustable for anti squat and to iron out any other undesirable tendencies relating to the longitudinal locating of the axle by having several different mounting points available for selection. As previously mentioned this adjustability is important for a race car as adjustments in ride height will lead to changes in the four link geometry and different tracks and different wheel / tyre combinations can also require minor adjustments to be made for the best to be had out of the setup.

4.3.5 Decoupled configuration

The final configuration to which will be discussed in this category is the decoupled rear suspension configuration. This design was created to disassociate some features from others(Millekin, Millekin, 1995).

The setup consists of one upper and lower link bar on each side similar to that of the four link discussed above and the arms can either be arranged parallel to each other both directed toward the front of the car or one bar can go forward and the other rearward. The arms are attached to the axle in a similar fashion to that of a four link, although the mounting is free to rotate around the axle tube, whereas in other cases it would be fixed(Millekin, Millekin, 1995).

Due to the fact that the four link bars do not react any torques due to their mounting not being fixed to the axle tube another method of resisting the toque inputs from the axle must be implemented. In this case a torque arm similar to that of which was discussed earlier is used, although it uses a coilover shock system to connect to the chassis which somewhat smoothes out the torque reactions that travel up the tube from the axle (Millekin, Millekin, 1995).

This setup will require a lateral restraint system such as the panhard rod or watts link setup much the same as the above cases.

The advantage of this complex setup over the others is that it is possible to change one of the primary kinematic features without affecting any of the others. Although due to its complexity this arrangement is not widely used and there are many simpler and more cost effective methods of locating an axle that will yield excellent results with less complications (Millekin, Millekin, 1995).

4.3.6 Chosen configuration

The parallel four link arrangement is the favourable method of longitudinally locating the axle in this scenario, as it allows for excellent control over the axle provided that the link bars are long and it can be set up to allow for more adjustability and therefore control over the axle without being too complicated in design. (Millekin, Millekin, 1995)

4.4 Lateral location

Now that a decision has been reached with regards to the longitudinal location of the axle, it is now time to look into constraining the axle laterally.

As previously discussed, the most effective way of doing this is to set up a watts linkage as it ensures absolute vertical motion of the axle, rather than moving in an arc type fashion like the more commonly known panhard rod.

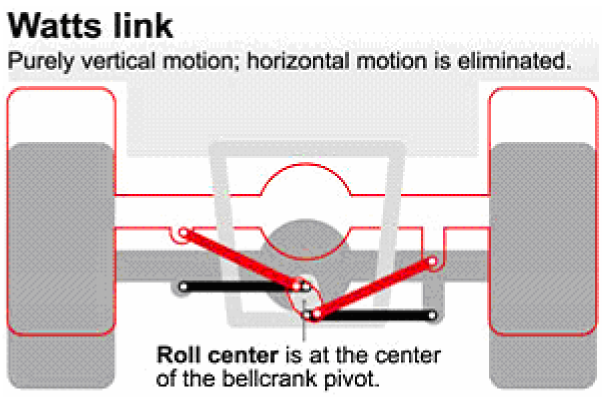

Figure 4.11: Watts Linkage
http://www.lateral-dynamics.com/products/source/_PanhardWattsWeb.jpg

Shown above is the most common form of watt's linkage, and is found on many road cars including Ford Falcons up to the release of the BA. The two lateral link bars are attached to the differential housing via a bell crank and then extend out parallel to the axle housing affixing to the chassis.

Changes in the height of the bell crank on the linkage alters the rear roll centre of the vehicle. The roll centre is the instantaneous point about which the chassis will roll on its springs about a particular centre. Altering the roll centre will determine how much the vehicle will keel over whilst cornering, rear roll centre adjustments also affect how much a car will tend to oversteer whilst cornering.

The major limitation of the watts linkage design pictured above is that it has a fixed roll centre. However, there is another form of watts linkage which is not used so much on factory cars, but is used more on race cars due to the roll centre adjustability it allows for. Instead of the bell crank being affixed to the differential housing and the two link bars being attached to the chassis it is the other way around, the two link bars are attached to the differential housing and bell crank is housed in a bracket that is affixed to the chassis which can allow for height adjustment of the bell crank, which allows for the roll centre to be moved.

Figure 4.12: Adjustable watts linkage

This type of arrangement is what will be used in this case as it allows for ease of adjustment which is most important for a race car.

Chapter 5

Design considerations

Now that a decision has been made as to the type of rear suspension configuration that will be designed, it is now time to look into some of the kinematic fundamentals of a suspension system.

It is important to note that even though the chosen 4 link, watt's link and coil over rear end is a far superior concept than that of a Hotchkiss suspension configuration, without the correct geometry set up this configuration could well operate far worse in practice than the factory configuration. A particular geometry must be designed to meet the needs of the particular vehicle in which it is being applied. There is no single best geometry that will meet the needs of every application (Millekin, Millekin, 1995).

There are five fundamental kinematic properties that need to be constrained by the suspension links. The anti lift, anti squat and the path of the wheel are defined by the instant centre. The height of the roll centre and roll steer are defined by the roll axis. The foundation for gaining an understanding of a live axle rear suspension is to understand how the instant centre and roll axis are determined (Millekin, Millekin, 1995).

5.1 Degrees of freedom

In a solid or live axle the two wheels are tied together and hence the motion of one wheel directly affects the other. A live axle should only have two motions relative to the chassis; The first is where the two wheels can travel up and down in unison and the second is where the wheels can move in opposing directions (chassis roll). In kinematic terms a properly constrained live axle has two degrees of freedom of motion relative to the chassis (Millekin, Millekin, 1995).

5.2 Instant centre

The word instant relates to the particular position of the linkage and the term centre refers to the projected imaginary point that is effectively the pivot point of the linkages (Millekin, Millekin, 1995).

The instant centre is the theoretical point through which tyre motion is transferred into vehicle motion. Setting up an ideal instant centre for a particular chassis is a critical factor in the design of any suspension system, particularly in that of a drag racing car. Adjustments to the instant centre affect the anti-squat and anti-dive characteristics of the chassis.

Instant centres came from the study of kinematics in two dimensions. The instant centre is excellent graphic aid in understanding relationships of motion between the sprung and unsprung masses. In suspension design and analysis it is common practice to simplify the three dimensional problem into two dimensional problems for simplicity. The two views that present the most useful information are the rear and side views. The top view does not convey much useful information about the path of the wheel and therefore isn’t very useful when considering the instant centre of a suspension.

When lines are projected forward from the upper and lower link bars of the four link, the two lines will intersect at some imaginary point. This point is the instant centre of the suspension. The main information that can be yielded from a rear view will be the scrub motion in suspension travel and some information needed to determine the steer characteristics. When looking at the side view, the instant centre will outline the fore and aft path of the wheel, anti lift and anti squat can be calculated and caster change rate can also be determined (Millekin, Millekin, 1995).

5.3 Instant axis

When the instant centre is found in the two views the two centres can be joined together to form what is known as the instant axis. Solid axles have two instant centre axes, one for parallel bump motion and the other for roll, these axes will move with changes in ride height (Millekin, Millekin, 1995).

5.4 Bump and heave

Bump is upward displacement of a wheel relative to the vehicle's body. The opposite, a lowering wheel, is called droop. Bump received its name due to occurring when a single wheel passes over a bump in the road. When a pair of wheels rise together in unison it is referred to as double bump, this is vertical motion of the wheels without the unsprung mass rolling. Heave is a vertical upward motion of the unsprung mass without roll, the opposite of double bump.

Bump and heave influence axle steer angles relative to the body and road, they also influence the spring and damper forces and hence tyre forces (Dixon, 1996).

5.4 Roll

Rotation of the vehicle sprung mass about a fore aft axis with respect to a transverse line joining a pair of wheel centres. It is positive for clockwise rotation and is viewed from the rear (Dixon, 1996).

Roll is the geometrical equivalent to one wheel going into bump and the other going into droop, relative to the chassis. When the vehicle is in roll there is changes in suspension geometry which generally leads to wheel (toe) steer angles of the axle relative to the chassis (Dixon, 1996).

5.5 Roll axis

In the case of a live axle the motion of the whole axle is to be considered in roll. The roll axis is generally drawn on the side view of the suspension model so that it appears as a line. This line controls roll steer and roll centre height. Depending on the suspension configuration being used there are many different methods of determining the roll axis.

For the case of a parallel four link complemented with a watts link the roll centre is defined by the height of the bell crank centre bolt and the roll axis is parallel to the lower link bar

The figure below shows how the instant centre and roll axis can be defined for this particular suspension configuration.

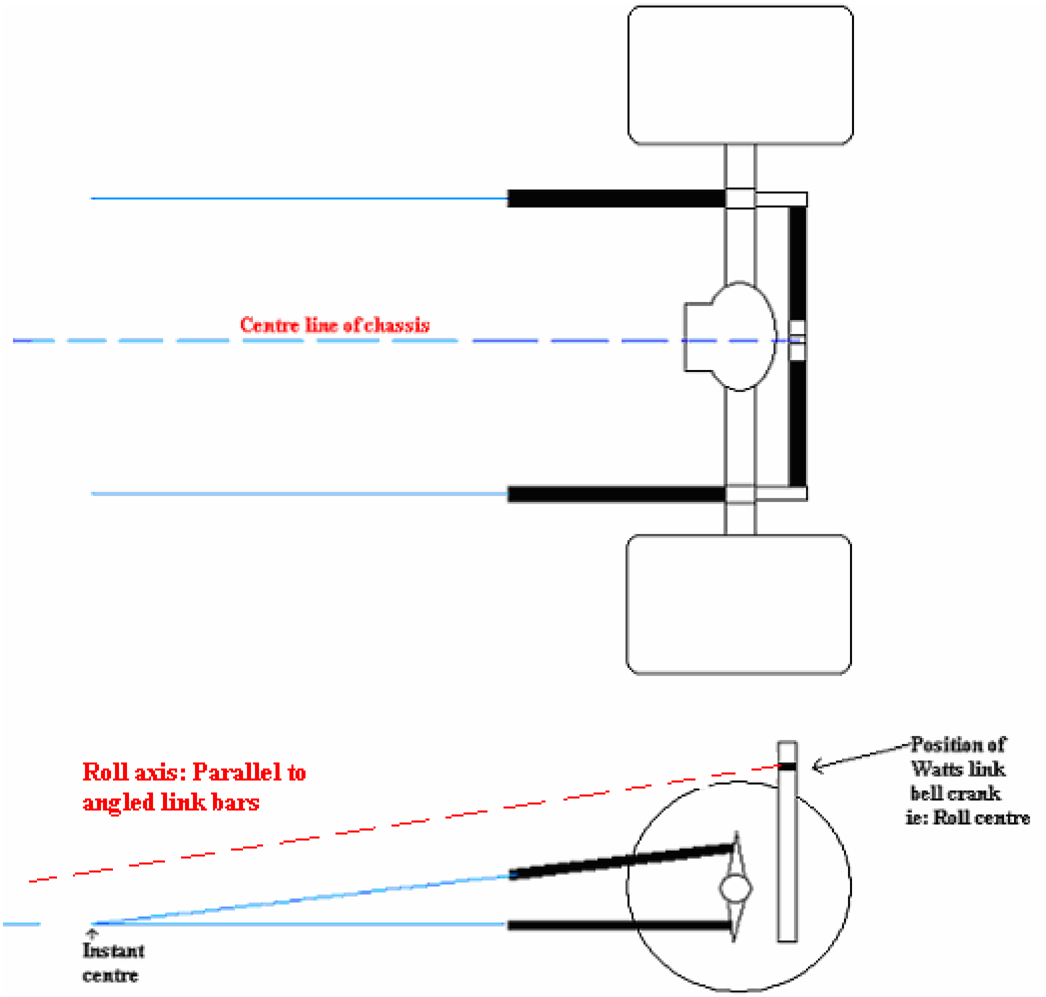

Figure 5.1 Roll axis and Instant centre defined

5.6 Roll steer

The angle of the slope of the roll axis multiplied by 100 represents the percent roll steer value. If the rear roll axis tilts down to the front of the vehicle when viewed from the side then the suspension has roll understeer. If the roll axis tilts upwards towards the front of the chassis then it has roll oversteer geometry.

Essentially the axle roll is relative to that of the chassis, the wheels travel in a path that is right angled to the roll axis. With roll understeer the wheel going into bump moves forward and the wheel drooping moves rearward, effectively when the vehicle rolls left it steers left and roll over steer is the opposite of this. Due to the wheels being conjoined by what is in essence a rigid beam, the forward motion in bump and rearward motion in droop means that the whole axle is steering relative to the centreline of the chassis (Millekin, Millekin, 1995).

5.7 Roll centre

The instantaneous point about which the chassis will roll on its springs about a particular centre. Although sometimes good to use as a graphic aid to gain an understanding of the concept, it is important to recognize that the roll centre is a force centre rather than a centre of motion.

The roll centre sets up the forcing coupling point between the sprung and unsprung masses. When a car corners, the centrifugal force at the centre of gravity is reacted by the tyres. The lateral force at the centre of gravity can be translated to the roll centre if the appropriate forces and moments about the roll centre are shown.

With regards to roll centre heights, the higher the positioning of roll centre is, the less the rolling moment about the roll centre which must be resisted by the springs will be. On the other hand lowering the roll centre increases the rolling moment. With a higher roll centre the lateral force acting at the centre will be higher from the ground. This lateral force multiplied by the vertical distance to the ground can be referred to as the non-rolling overturning moment. Bascially roll centre height is a trade off of the relative effects of the rolling and non rolling moments (Millekin, Millekin, 1995).

5.7 Roll couple

If the vertical distance between the roll axis and the vehicle's centre of gravity is not zero, a rolling moment between the centre of gravity and the roll axis will be exerted on the sprung mass, causing the body to lean towards the outside of the turn. This force is called the roll couple (Answers, 2007).

5.8 Roll skew

Considering the whole car and its two roll centres, the front roll centre and the rear roll centre and joining a line between them as shown below. This is the roll axis of the entire suspension, which determines how the entire chassis will behave in roll, rather than just one end.

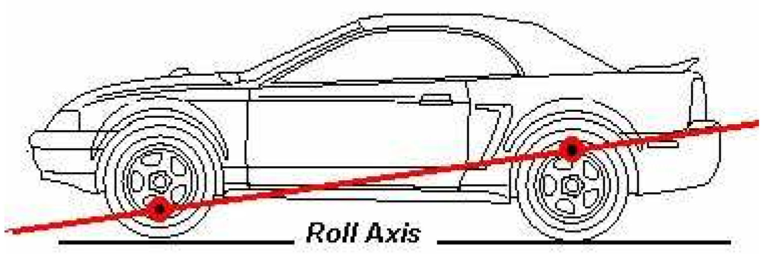

Figure 5.2: Roll skew
http://www.miracerros.com/mustang/t_rollaxis.jpg

The rear roll centre is always higher than the front roll centre and keeping in mind that the basic concept of the roll centre is that it is the imaginary point about which the chassis will roll on its springs, this axis will show the behaviour of the entire chassis in roll.

When the chassis is in roll and is rolling along this roll axis for example if the car is cornering left it is rolling right, the rear inside wheel (rear left) will become unloaded and all of the weight is shifted onto the rear right wheel causing the left front wheel to lift its weight from the ground when powering out of a corner and the reverse happens when entering a corner under braking, this phenomenon is roll skew and it can be minimised by levelling out the difference in heights between roll centres.

If the difference between front and rear roll centre heights is levelled out, the tendency for the car to lift the outside rear wheel when braking into corners and to lift the inside front wheel when powering out of corners is reduced, therefore lowering the amount of weight shift to the loaded side of the car, which inturn results in a flatter cornering car with more evenly loaded wheels.

It is important to note that in suspension travel the roll centre heights are constantly changing and it is important to make sure that in any point of suspension movement that the rear roll centre does not cross to a lower height than that of the front roll centre, as this will cause the chassis to change from the roll understeer geometry, consequential of having the roll axis tilted downward to roll oversteer geometry in an instant which will upset handling dramatically and can cause the car to oversteer excessively in that instant.

Chapter 6

Four link design

6.1 Four link design

The first thing that was considered in the design of the four link was the length of the link bars, they needed to be made as long as practically possible but keeping behind the centre of gravity of the chassis. The longer the bars are the smaller the caster changes will be in parallel bump and droop as the axle effectively moves in an arc about the points where it is hinged. It is also important to have the link bars of equal length as caster and steer values are compromised as the top and bottom of the axle will be moving at differing radiuses, this will also cause variances in the differential's pinion angle.

It was decided to locate the front of the bars at the position of the original rear seat cross member, which runs across just rearward of the B pillars of the car. This position means that the drivers seating position is not compromised and the link boxes which the link bars mount to at the front can be strengthened by cross bracing to the B pillars on either side. This mounting position allowed for the length of the link bars to be 600mm from centre to centre, which in simulated conditions has shown very minute values of caster change and roll steer as will be discussed later.

The chosen material for the link bars is 26.7mm OD seamless cold drawn carbon steel pipe. At the front of the bars Ford escort lower control arm bushes are used to mount the link bars to the chassis mounted front link boxes to try and dampen the shocks from drive train torques, and to allow for any minute tendency for the link bars to bind against each other in roll which may cause the link bars to try and resist roll which is not desirable as that is not the task of the four link.

Meanwhile at the rear of the link bars on the axle housing end, 3/8 spherical bearing rod ends have been selected to allow for minor adjustments in the length of the link bars and to provide a solid location of the link bars at the axle end. As shown below a steel plug is used to fit the rod end to the end of the link bar. This plug (which is supplied with the rod end along with a locking nut) is welded into the link bar and is threaded in the centre to allow for the rod end to be screwed in.

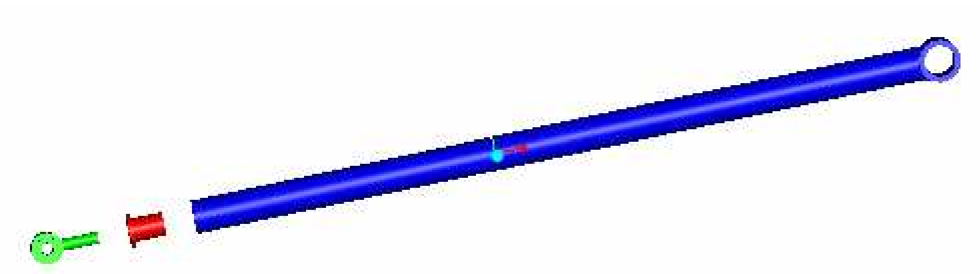

Figure 6.1: One of four Link bars

There are many available mounting points for the link bars in this particular design. These differing mounting positions allow for the suspension to be tuned for a particular set of track conditions and or vehicle conditions.

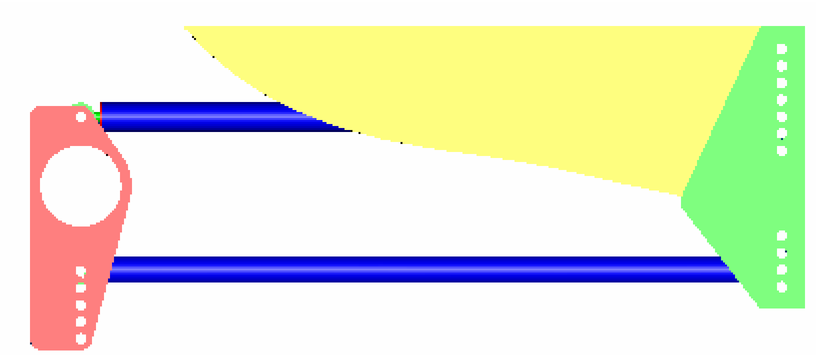

Figure 6.2: Four link side view showing the range of adjustments allowed for.

Initial calculations have been made for the link set up to be pre set to for circuit conditions or drag racing conditions. For circuit racing the link bars are to be set up horizontally and parallel to each other to allow for the best suspension geometry to be kept in roll and in suspension travel. For drag racing the instant centre of the link bars is set up to converge at the estimated centre of gravity of the vehicle.

Due to the fact that many factors such as ride height, weight and weight distribution for the car in which this suspension configuration is to be fitted have not been finalized an estimated centre of gravity was assumed to be at a point around the gearstick hole of the car, this is a common assumption made for that of a front engine / rear wheel drive chassis.

An accurate calculation of the centre of gravity is not critical in the design for a suspension system such as this as over the course of a race car's development many changes tend to take place such as changing rolling diameters, ride heights and weights / weight distributions which therefore means that the centre of gravity will always be changing every time any minor change is made. Trialling of different coil spring rates and shock absorber damping rates sometimes also call for adjustments to be made to the four link geometry. It is therefore necessary to take this in to account and allow for suitable adjustments through out a foreseeable range to permit for the desired suspension geometry to be dialled in with the simple change of the four link mounting points to work in with the constant development of the rest of the car's set up.

A small change in ride height either upward or downward will change the angle of the bars and put the bars out of their optimum geometry range. Consider the case below. The solid lines represent the suspension at the current calculated ride height of the vehicle and the link bars are set up in the horizontal position. If the car is lowered down 50mm (effectively the wheel is raised 50mm into the chassis), represented by the dashed lines in this case the link bars lift up with the rear wheel and axle assembly thus angling them downward into the ground. This change in geometry will promote roll understeer geometry and squatting of the rear of the vehicle under acceleration which is generally not desirable, although setting the car up for a minor amount of roll understeer geometry is sometimes a desirable trait, this will be discussed later. If the car is raised 50mm on the other hand the bars will tend to push the chassis in an upward direction.

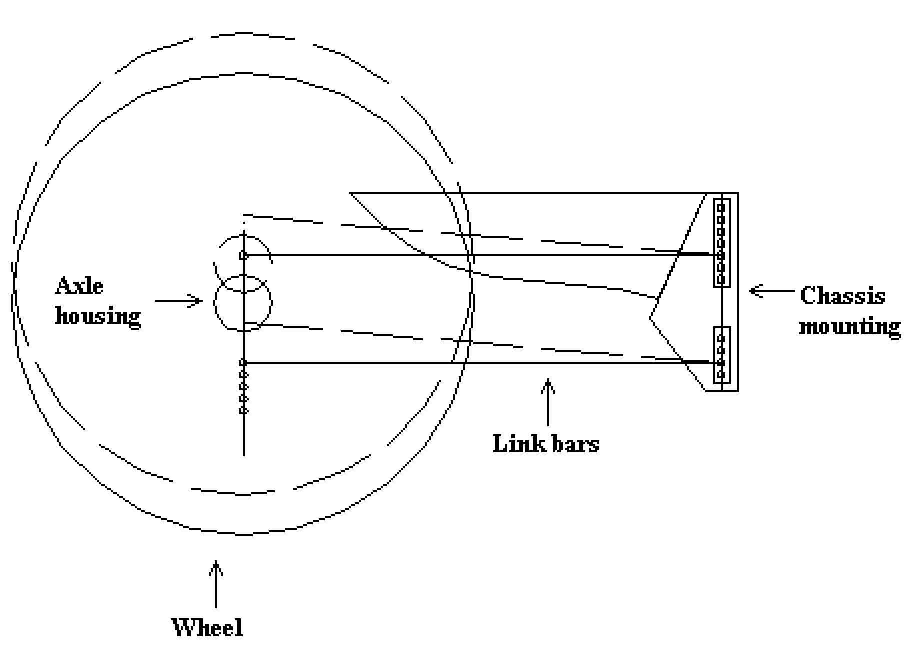

Figure 6.3: Four link schematic 1

Changes in ride height to a foreseeable extent have been taken in to account in this design and therefore if the ride height were to be lowered 50mm as in the case above all that will be required to set the geometry up to parallel once more is a change in the mounting positions of the link bars.

The angles of the bars represent the line of force in which the wheel and axle assembly is pushing the car forward. In the case of horizontal and parallel bars the power transmits to the chassis horizontally and thus no energy is wasted in attempting to push the chassis upward or downward.

The front link boxes (shown below) which are to be seam welded to the chassis are made out of 1.6mm folded sheet steel and are double skinned on the mounting points for the link bars for extra strength. As discussed the several available mounting positions for the upper and lower link bars allow for adjustments in the four link geometry to be made. The rear floor needs to be cut to locate the front link box in either side of the chassis. The top half protrudes the floor on the inside of the cabin and the lower half hangs below the floor on the underside of the vehicle.

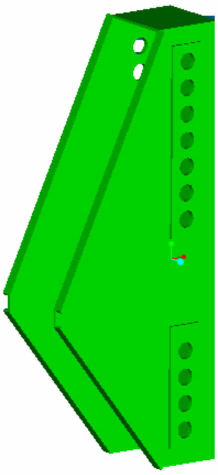

Figure 6.4: Front link box

The rear mounting brackets for the links (birdcages) are made from 3mm folded sheet steel as pictured below. These mounting brackets are seam welded to the axle housing.

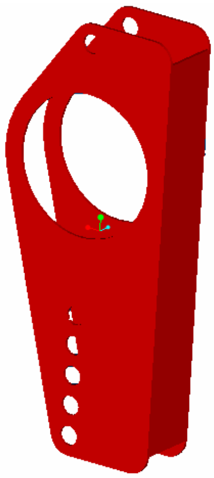

Figure 6.5: Four link Birdcage

Careful consideration has to be made for the angles in which the birdcages are welded to the differential housing. The angle of the differential pinion in relation to the driveshaft and gearbox output shaft need to be aligned to avoid extra stresses on the universal joints on the driveshaft. If the universal joints are constantly at extreme angles they will be prone to excessive wear and or premature failure.

The angle in which the birdcages are set at effectively determines the pinion angle of the differential housing. The pinion angle is the difference between the driveshaft angle and the pinion angle on the differential. Unfortunately due to the birdcages needing to be welded to the axle housing for this style of suspension to function, once it is set it is fixed and cannot be adjusted as the birdcages are welded to the axle housing.

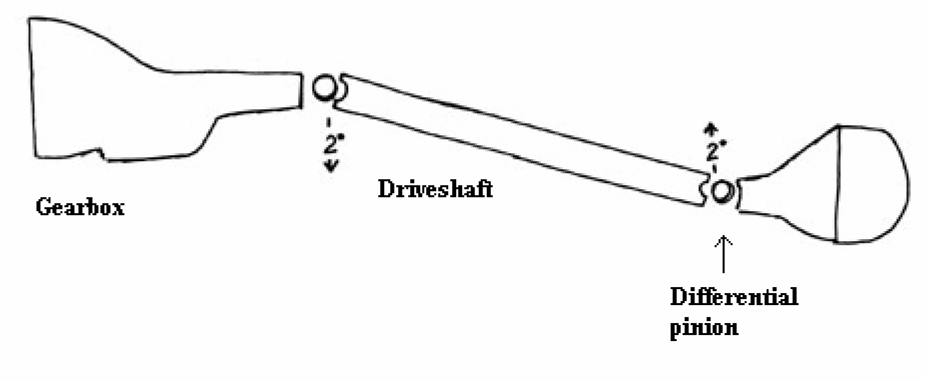

Figure 6.6: Relationship between gearbox output shaft, driveshaft and pinion

Under general conditions it is fair to assume that the rear of the chassis will squat slightly when under load and thus to achieve the desired alignment, the pinion needs to be pointing slightly down from the horizontal when the vehicle is static or not under any loads.

Due to the four link system's excellent geometry throughout suspension travel and being capable of nearly eliminating axle wind all together up a preset pinion angle of generally only 0.5 to 1.5 degrees needs to be built in, where as in a typical leaf sprung arrangement a pinion angle of 5 – 8 degrees needs to be set due to the excessive axle wind up that tends to occur under load. A positive pinion angle is something that should never be set (Watson, n.d.)

One method of determining the pinion angle is by mocking up the whole rear suspension beneath the car and following this simple guide provided by Wolfe Race Craft at the following URL:
http://www.wolferacecraft.com/pinionangle.aspx

The pinion angle is found by using an adjustable protractor to measure the difference between the pinion flange and the drive shaft directly. These gauges are available for under $25 from hardware stores.

Refer to the diagram below while reading the following instructions.

- Place the edge of the gauge vertically against the front of the pinion flange, beside the driveshaft.
- Extend the measuring arm forward parallel to the bottom of the driveshaft.
- Extend a straight edge under the driveshaft to the measuring arm of the angle gauge.
- Hold the straight edge flat against the bottom of the driveshaft and adjust the measuring arm to read the angle.
- Depending on the gauge you use, you may have to subtract 90° from your reading to get the correct number.
- The desired pinion angle of between 0.5 and 1.5 degrees can now be determined.

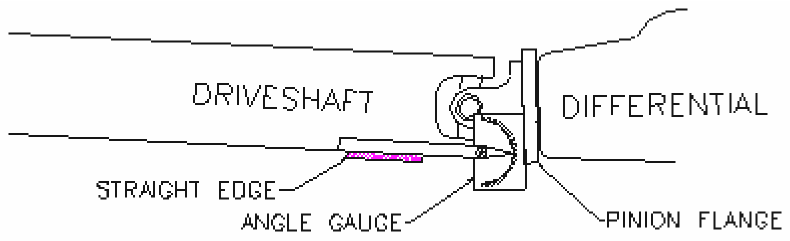

Figure 6.7: Setting pinion angle

Now that the pinion angle can be set, the bird cages can be welded to the axle housing.

Due to the upper link bars protruding through into the rear of the floor inside of the cabin, some basic covers have been designed (pictured below) for mainly cosmetic reasons as well as to seal the cabin. They are 1.4mm folded sheet steel and are to be tack welded on to the top of the front link box and on to the floor of the vehicle.

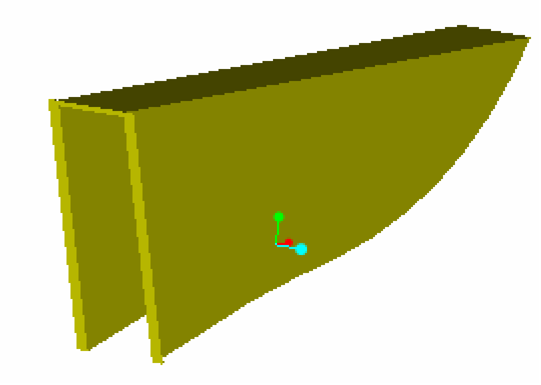

Figure 6.8: Upper link bar covers

6.2 Setting up four link geometry

The parallel positioning of the link bars is good for a circuit race car as the car spends most of its time on the track moving and cornering and in the parallel position the axle is most neutral in its movements relative to the chassis. In circuit racing getting away from the line at a standing start is not the most important consideration in suspension design, although in the case of drag racing getting away from the line as fast as possible is the most important part of the race, roll steer is not a major consideration as the car is not cornering, it is only running in a straight line.

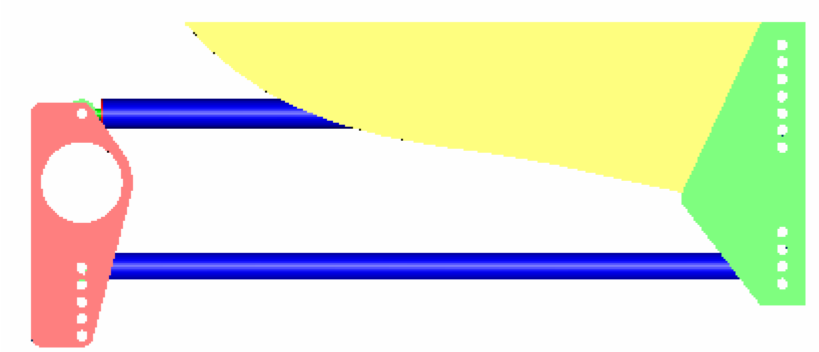

Figure 6.9: Parallel orientation of link bars

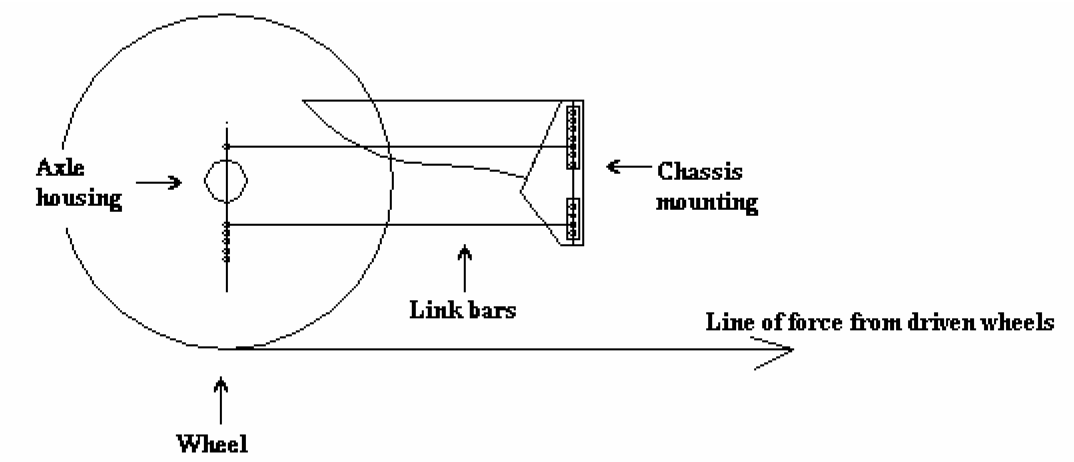

Figure 6.10: Four link schematic 2 (Parallel four link bars)

6.2.1 Four link set up for Drag racing

If the instant centre of the link bars are set up to converge at the height of the centre of gravity of the chassis some of the energy transmitted from the engine through the drivetrain and then on to the wheels is used to push the chassis upwards, thus effectively pushing the wheels down into the ground and effectively increasing the available traction. At this point 100% anti squat is achieved. This is the point where the suspension will neither lift nor drop, just enough force is applied to push the wheels down enough to achieve maximum traction, although no extra energy is wasted in the force from the driven wheels trying to push the chassis upward or downward (RaceGLIDES. n.d.)

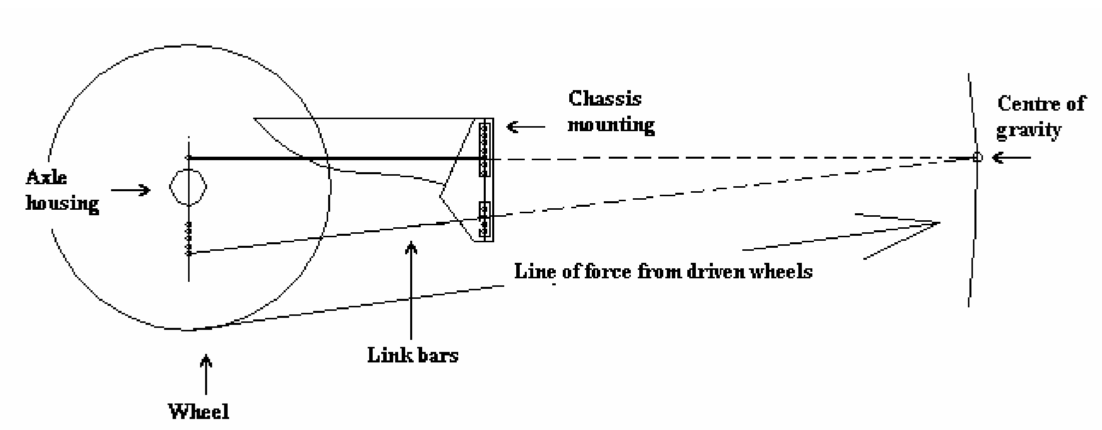

Figure 6.11: Four link schematic 3
Four link set up for 100% anti squat geometry

If the instant centre is below this line the rear of the vehicle will squat under acceleration and if the instant is above this line the rear of the vehicle will rise respectively (RaceGLIDES. n.d.).

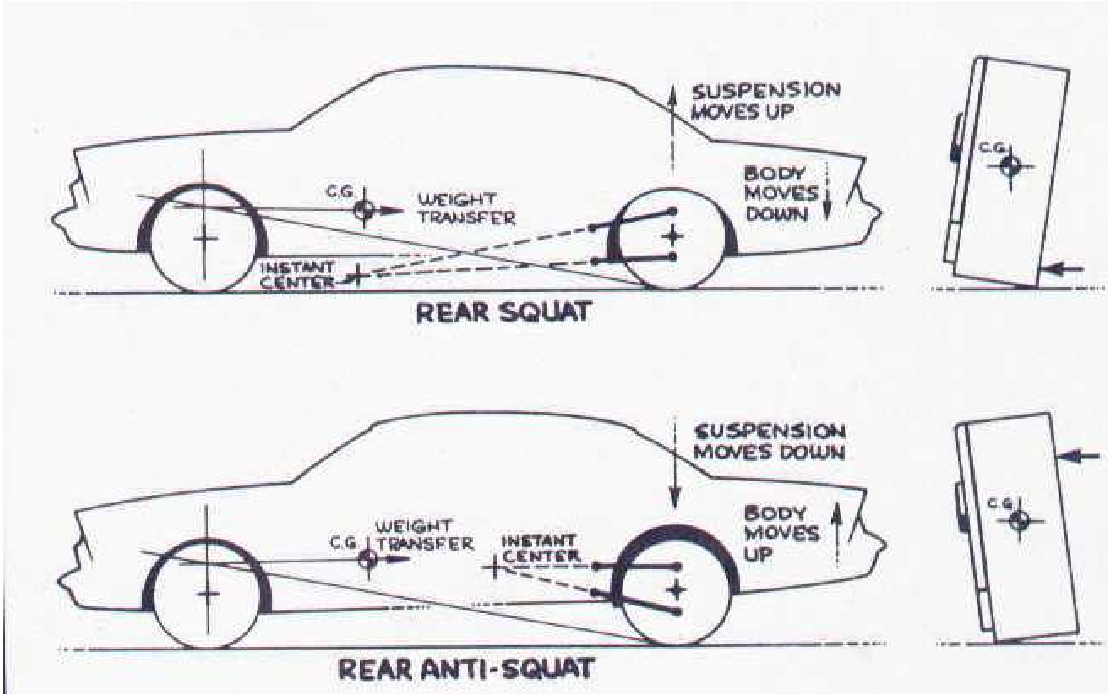

Figure 6.12: Rear squat Vs Anti squat

Again due to the fact that the centre of gravity has only been estimated in this case, the geometry can only be set up to achieve a base line anti squat and using the car on the track will soon show if any adjustments will need to be made to the geometry.

If the anti squat is configured correctly the rear tyres should not spin when launching and the car should start moving forward as quickly as possible. The rear of the car should not be lifting or squatting and the front of the car shouldn’t be wheel standing. If the rear of the car tends to squat, the instant centre can be raised to compensate for this and if the rear tends to lift too much the instant centre can be lowered. If the car tends to try and pick the front wheels off of the gournd, the link bars can be set to converge further rearward on the chassis and in the case of if the tyres are being shocked too violently when powering off of the line the reverse can be done . (RaceGLIDES. N.d.)

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This is of course considering that rear shock absorber and spring adjustments along with the front suspension are all set up correctly. Once the car is set up for optimum anti squat geometry only minor adjustments will need to be made for varying track conditions or other changes to the car’s set up . (RaceGLIDES. N.d.)

6.2.2 Four link geometry set up for a circuit race car

As previously discussed the base line adjustment for circuit conditions is to have the four link bars parallel and horizontal to the ground as in this position the axle is kept most neutral in its movements relative to the chassis, as roll steer is minimized as the roll axis is kept horizontal and caster change is minimized.

On a circuit racer car there is no need for large amounts of anti squat geometry to be set up as for the most part of the race the car is moving and once the car is moving the chances of having loss of traction in a straight line is much less than that of starting from a standing start and therefore it is generally much better off having the bars set near parallel to the ground using all of the power transmitted to the rear wheels to push the car forward and keeping the caster change and roll steer to a minimum.

Although having the four link bars setup parallel minimizes caster change and roll steer in some conditions it might be desirable to induce small amounts of roll steer by angling the bars to help the car steer into corners better or in the case of a slippery track it may be desirable to dial in a small amount of anti squat geometry to increase traction in a straight line. Although of course angling the bars too much will sacrifice traction in cornering as excessive roll steer will upset the balance of the car.

6.2.3 Adjustments in four link geometry for circuit conditions

By simply angling the link bars upward slightly under load the link bars will act to push the chassis upward which effectively is pushing the tyres into the ground, see the exaggerated figure below. The angling of the bars of course increases the roll steer and caster changes in suspension travel, although on a circuit with more straights than corners it may be advantageous to compromise cornering ability to achieve more traction accelerating on the straights, this would only be in a case where traction is an issue of course.

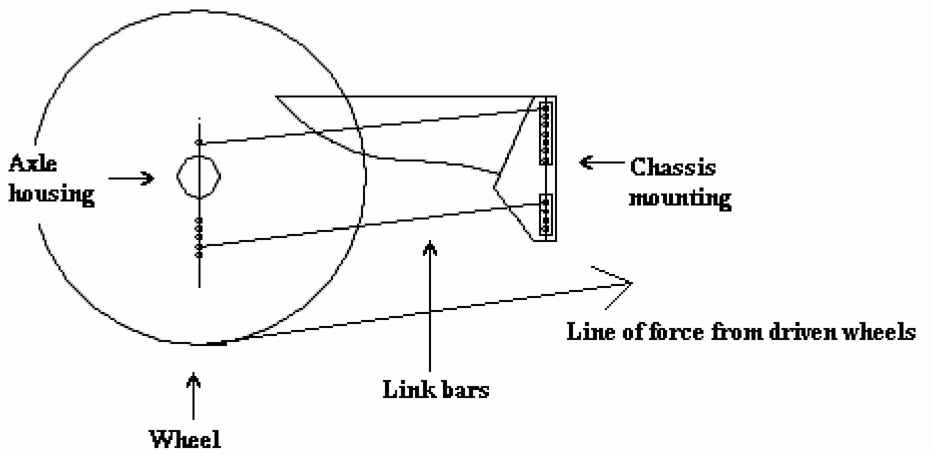

Figure 6.13: Four link schematic 4
Upwardly angled link bars for increased straight line grip

Suspension geometry will always be a compromise. Another draw back of having link bars angled upwards is that in roll the link angles change and if the link angles become angled more upward on the left side than on the right, the left wheel will be pushed into the road surface harder than the right which generally induces oversteer as the left wheel will be driving the car forward more than the right wheel. In cases such as oval track racing where the car is only turning in one direction (left) a foreseeable solution to this problem would be to angle the right link bars slightly more upward when the car is at rest so that the bars will be flatter in roll (AFCO Racing. N.d.). Changes in the rear roll centre height relative to the roll centre height of the front suspension will decrease the intensity of this problem as well, this will be discussed later.

Another issue that may arise when the link bars are angled upward is that the force pushing the chassis upward or pushing the wheels down (whichever way you want to look at it) is that the force applied from the link bars can hold the chassis up under acceleration which reduces the effectiveness of the springs and shock absorbers. This is not desirable, especially on a rough surface, as the car will tend to become unstable and skip all over the track due to the bumps in the track not being absorbed aptly (AFCO Racing. N.d.).

Adjustments to the lower links can help reduce the excessive roll steer caused by the upwardly angled upper link bars or vice versa. Adjustments to the angles of the links changes the paths travelled by the respective link bars and any change to the path travelled by any link bar will directly affect roll steer. In the illustration above (Figure 6.13) where both of the link bars are angled upward when the car corners left in roll the right wheel becomes more loaded and moves upward effectively getting longer and the left wheel moves downward effectively becoming shorter (remembering that the suspension moves in an arc and that the axle is a beam, thus the movement of one wheel directly affects that of the other). This will tend to cause excessive roll steer.

If the lower links were to be lowered back to their initial horizontal position (or the other way around, the upper links are made horizontal) and the upper links are left in their current position as shown below some of this roll steer can be minimized as this will change the paths in which the respective link bars will travel. When considering the case of the car cornering left again, the right side of the suspension moves upward and both links become shorter although the lower link’s change in path is less due to being horizontally located to begin with and thus the forward movement is minimized and for the left side the backward movement of the axle is also reduced for the same reasons, thus roll steer is minimized.

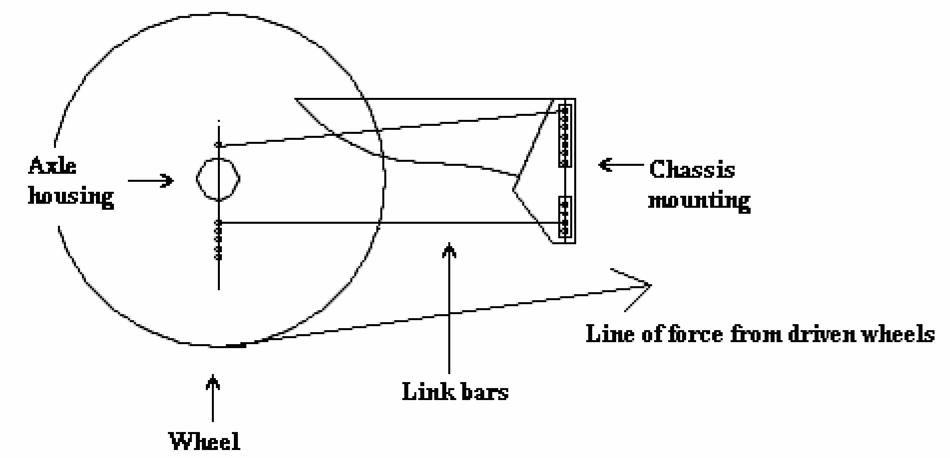

Figure 6.14: Four link schematic 5
Lower link bars put into the horizontal position

In a general sense the idea here is to set up the four link to provide to some force to push the wheels down into the track if traction is a problem, yet minimising the amount of roll steer generated by changing the angles of the link bars.

6.3 Four link analysis using Wingeo3

The suspension modelling software package Wingeo3 version 4.00 by Wm. C. Mitchell software Racing by the Numbers has been used to obtain caster and steer values for changes in ride and roll angles for various four link geometry changes.

The program has a preset four link solid axle template in which the dimensions and properties of the four link and vehicle can be inputted to simulate geometry changes in variances for ride and roll.

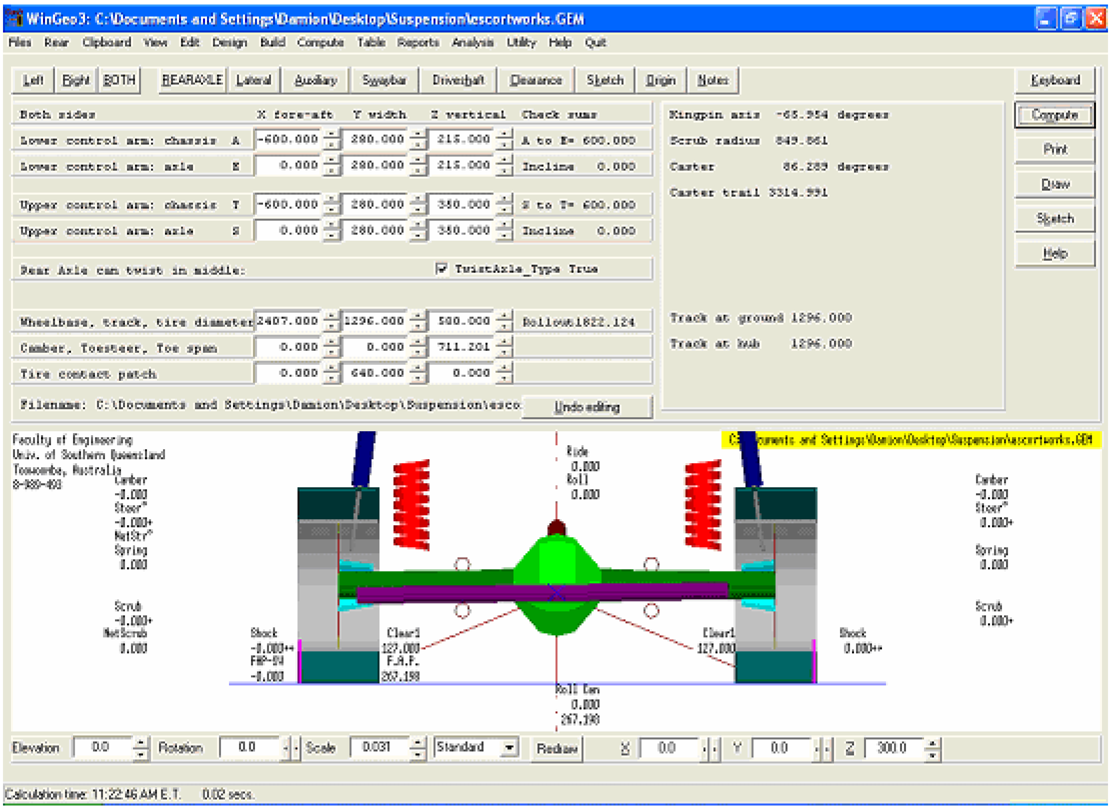

Figure 6.15: Wingeo3 Inputting dimensions to analyse four link geometry

Once all of the data relating to the rear suspension is defined, data can be tabulated for the suspension for differing ride and roll values. The main values of interest in this case were the variances in caster and toe steer. All of the data calculated in Wingeo3 has been transferred over into Microsoft Excel for ease of organizing and plotting the data generated by the software package.

Bare in mind that negative values for caster indicate that the axle is moving forwards toward the front of the car.

6.3.1 Caster change in varying vertical ride

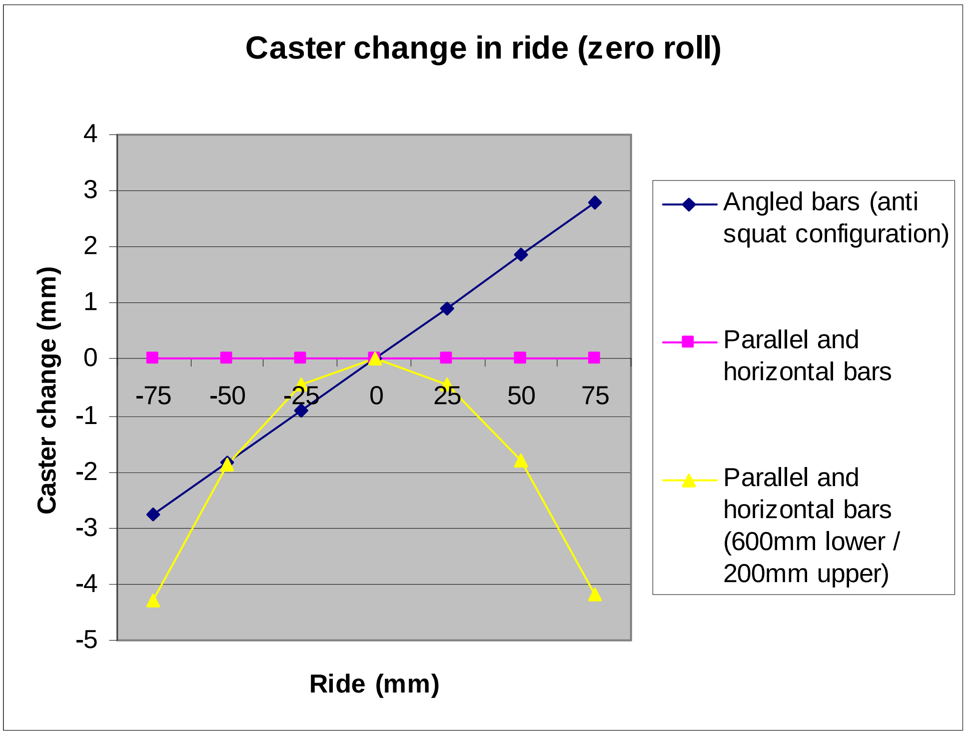

Figure 6.16: Caster change with changing ride heights

The above figure shows caster change for changes in ride height from -75mm to 75mm. The horizontal line represents the chosen 600mm link bars set up parallel to each other and horizontal. As predicted in vertical suspension travel the caster change is negligible. Wingeo has indicated that as long as the link bars are kept parallel to each other the caster change will be minimal.

In the case of the anti squat configuration, where the upper bars are horizontal and the lower bars are angled upwards it can be seen that the caster change in travel is quite dramatic as predicted.

The final case in the plot above is a case where the lower link bars are 600mm in length and the upper bars are 200mm in length, the bars are parallel to each other and horizontal. The caster change is dramatic and this is why it was important to choose link bars of the same length. Shorter upper link bars are commonly used in four link's to allow for rear seating capacity, this is obviously a major compromise for geometry, although it all comes down to what is considered to be more important when the car is being used for its purpose. For a passenger car interior space takes precedence, although on a race car it is more important for the geometry to be consistent throughout variances in ride and roll values.

6.3.2 Toe steer in roll

The following figure shows toe steer with respect to the right wheel in changing values of roll. A positive steer value denotes that the wheel is toeing in or the axle is steering left or rotating counter clockwise when looking from above (plan view).

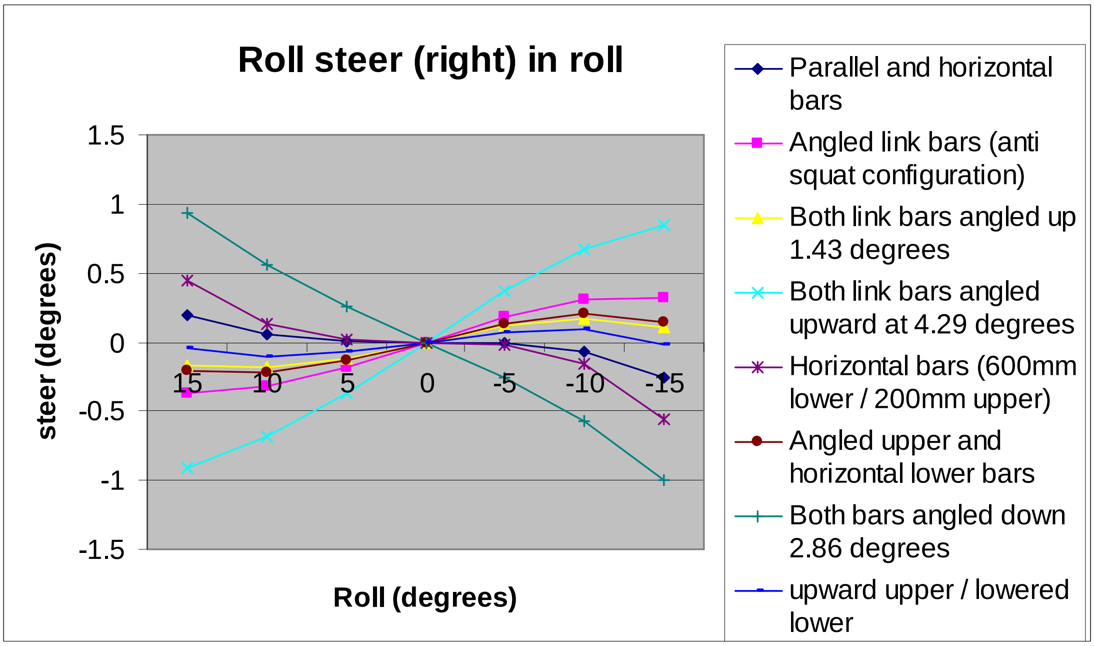

Figure 6.17: Toe steer in roll

As can be seen, the toe steer is kept to a minimum when the link bars are positioned parallel to each other and horizontal to the ground. In this case toe steer doesn't seem to be dramatically affected by whether the link bars are parallel or not, rather it is the angles of the link bars (when the car is static) that affects the toe steer. As previously discussed two link bars angled upwards will yield larger roll/toe steer values than that of one bar angled upward and the other being horizontal to the ground, this can be noted when examining the figure above. Obviously the greater the angles of the bars at static ride height, the more dramatic the steer will be in roll.

When the link bars are angled upward and hence the roll axis is angled upward in positive roll (corning left) the link bars on the loaded side (right side) flatten out when causes the right side of the axle to move back slightly and on the unloaded side (left side) the upward angles of the link bars increase causing the axle to move forward slightly, this in turn causes the axle to toe out when cornering which is not a desirable trait. This is roll oversteer geometry.

As can be noted when looking at figure 6.17, the exact opposite will occur when the link bars are angled down slightly and as a result of this the axle toes in whilst cornering which assists the car to turn into corners better. This is roll understeer geometry and this reduces the tendency for the rear of the car to oversteer whilst powering out of a corner.

When tuning the four link geometry for the circuit it is desirable to set the axle up to toe in whilst cornering as the car's tendency to oversteer from the rear is kept down and this will increase the stability and predictability of the car.

6.3.3 Caster change in roll

The following figures show caster change with respect to varying values of roll. A negative caster value denotes that the wheel is moving toward the front of the chassis and a positive caster value denotes that the axle is moving toward the rear of the chassis.

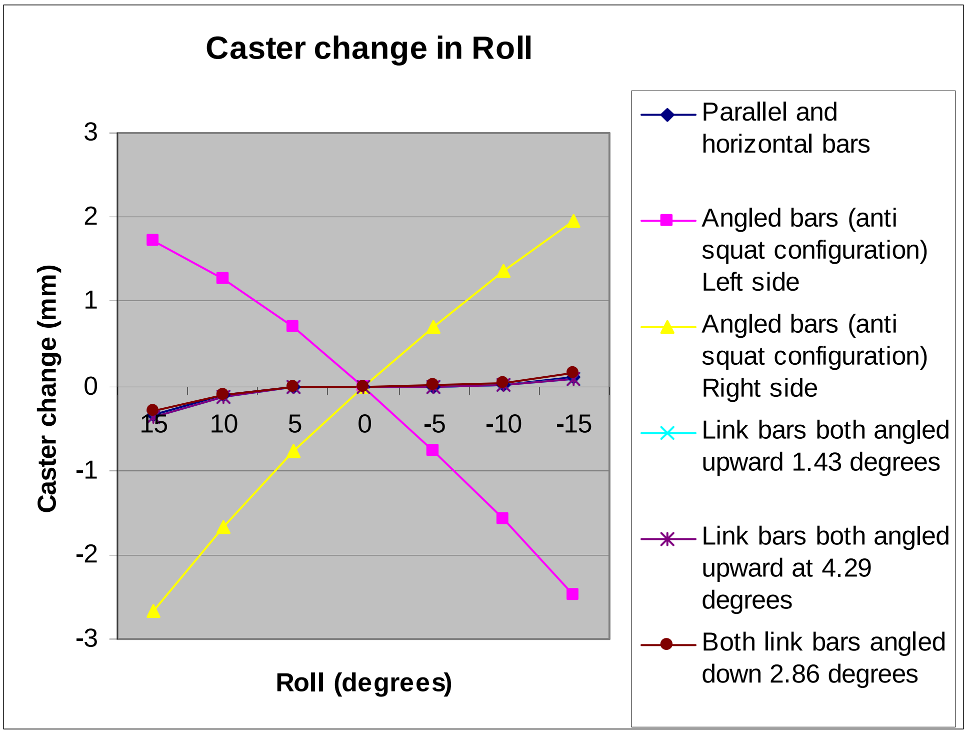

Figure 6.18: Caster change in roll 1

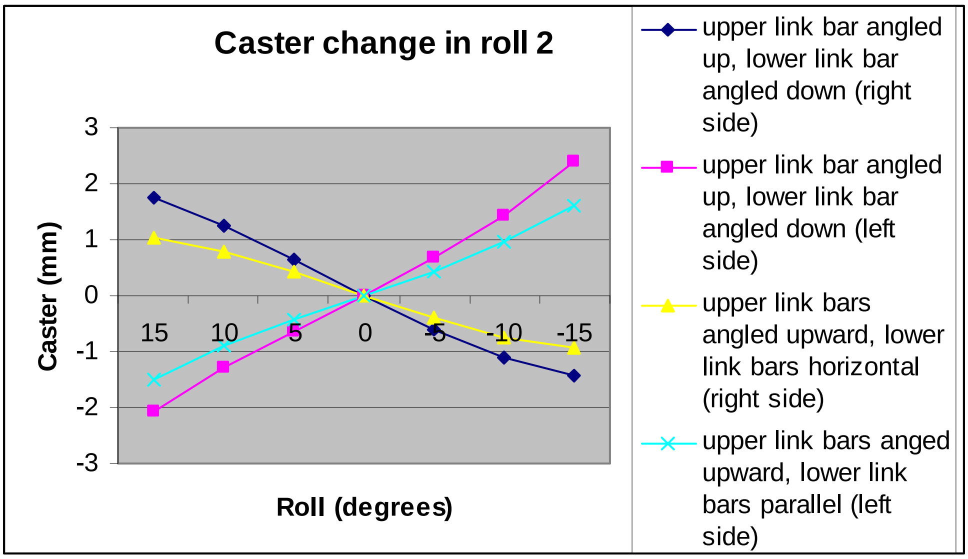

Figure 6.19: Caster change in roll 2

The above plots show that regardless of the angles of the link bars as long as they are kept parallel the caster change in roll is kept to a minimum and the caster change very similar for both the left and right wheels. Although when the bars are not parallel the caster change is very close to opposites for the left and right wheels.

6.4 Final points on four link design and geometry

The 4 link is a relatively complex rear suspension that is very sensitive to adjustments. A change in link bar length of 25mm or a small change in the angle of the link can make a noticeable change to the way the vehicle handles.

When tuning the four link it is important to understand the relationships between varying link angles and how they affect roll steer and tyre loadings in corning (AFCO racing. N.d.).

For drag racing roll steer is not an issue as the car is not cornering, the four just needs to be configured for the car to anti squat properly. A base line anti squat geometry has been calculated for anti squat where the upper link bars are parallel and the lower link bars are angled upwards with the lowest hole on the axle being selected and the third hole from the bottom at the front as shown in the diagram below. The only way to perfect the geometry is to see how the car performs in practice and adjustments can be made from this baseline, improvements can quickly be pin pointed by quicker 60 foot and ¼ mile times and vice versa if the adjustments worsen the situation.

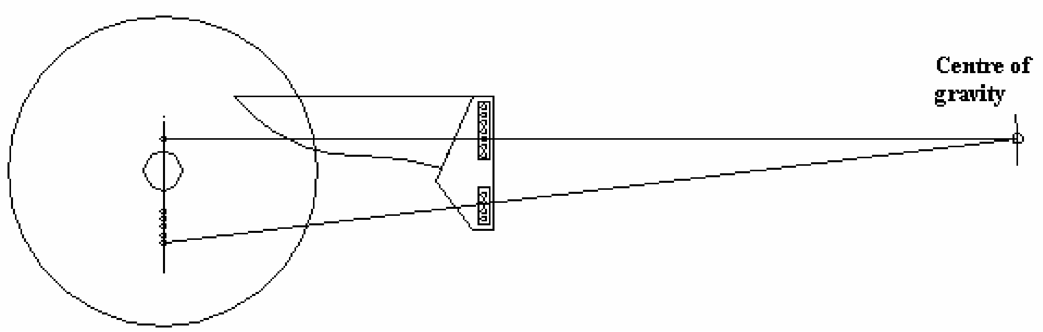

Figure 6.20: Base line anti squat geometry

The computer program '4 Link calculator v2.0 A.014' by Performance trends, inc allows the user to input the four link dimensions for the car in question and it then can calculate the amount of anti squat a particular geometry possesses.

This confirmed that the base line geometry is very close to achieving 100% anti squat geometry with a calculated value of 99.3%. If the lower link bars are raised up one more hole (the top hole) on the front link boxes the anti squat geometry is found to be 119.1% indicating that the geometry will push the sprung mass upward during a launch. In the case where the lower link bars are set one hole lower than the base line setting on the front link boxes (second hole from the bottom) the calculated anti squat is found to be 79.2%

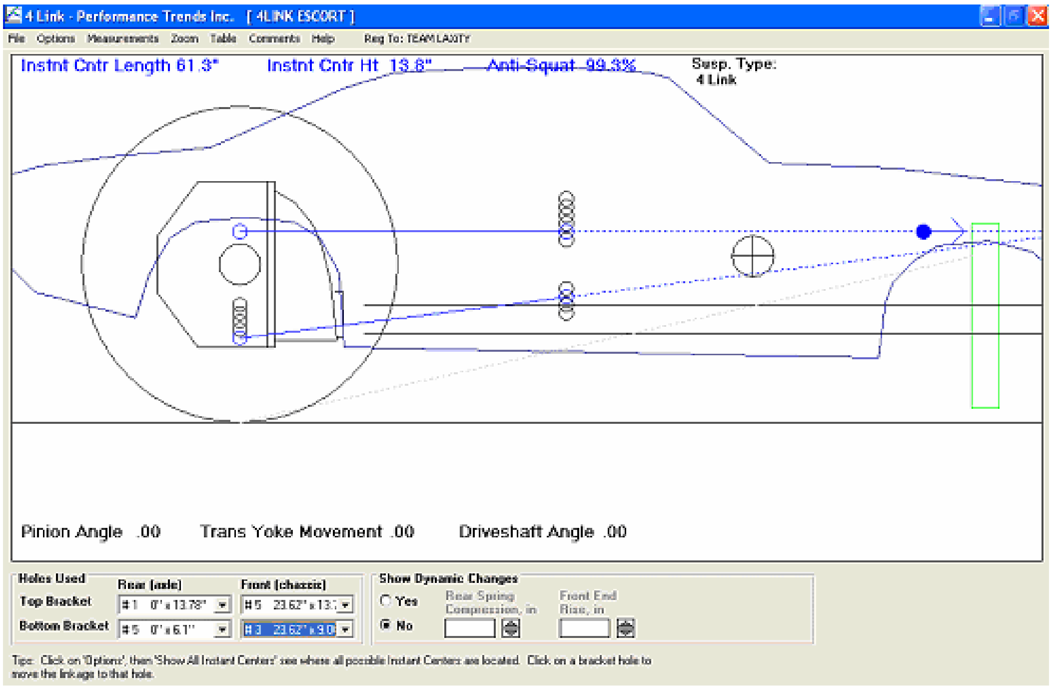

Figure 6.1: Anti squat calculation using Performance Trends 4 link calculator

For circuit racing the geometry is to be set up parallel for a base line as the least caster change and roll steer occurs in this orientation. If the car needs help turning into corners better (oversteer), the link bars can be angled down slightly to achieve axle toe in roll, if the car suffers from grip substantially the link bars can be angled up slightly to add some extra force pushing the wheels down into the track surface and in the case of an oval track the angles of the link bars on each respective side can be altered to allow for more even tyre loading when exiting corners. Shown below is a diagram of how the four link base line geometry is to be set up.

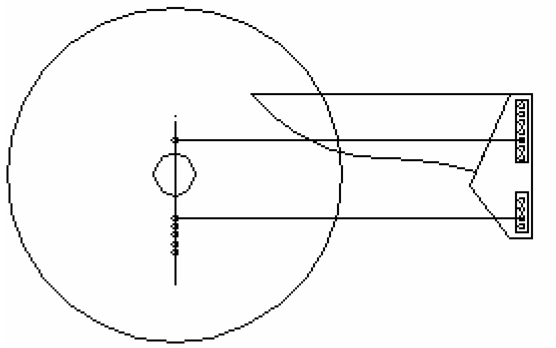

Figure 6.22: Base line circuit geometry

Suspension set up will also vary from driver to driver as many drivers have different driving styles which require the suspension to be tuned differently.

6.5 Four link fitment

First starting with fitting the bird cages to the axle or differential housing, strip the axle back to a bare casing and grind off all of the factory leaf spring brackets and anti roll bar brackets. Mark the centre points of the bird cages which are to be 280mm either side of the centre line of the housing. Now set the pinion angle using the process that has been previously discussed. Weld the bird cages to the axle casing with a continuous weld on each side of both of the cages.

Next is fitment of the front link boxes, first find the centre line of the car and from there measure out 280 either side and mark out the area to be removed for fitment of the link boxes, cut the areas out and dummy fit the link boxes in to make certain they fit tight up against the rear seat cross member. The top of the link boxes are to be 150mm above the floor at the rear seat cross member. Check that everything is in place and tack the link boxes in and check that all measurements are still correct and then seam weld both link boxes to the chassis.

The last piece to fit is the link box covers, they are to be placed on top of the link boxes and against the floor and then tacked in place.

Chapter 7

Watts linkage design

7.1 Determination of roll centres

When considering the design of the watts linkage the first thing to consider was where to mount the watts linkage housing. There is a flat section of floor pan just to the rear of the differential housing and it was decided to mount the watts housing to this section. The minor disadvantage of this is that the watts linkage is mounted slightly further back than ideal and this increases the length that the mounting points for watts link bars protrude from the birdcages, which in turn effectively weakens them.

Figure 7.1 Escort underside

The next thing to consider when designing a watts linkage is the roll centre height. A good starting point is to consider the roll centre height of the standard suspension, as the height of the rear roll centre in relation to the front roll centre is a critical factor in determining how the vehicle will handle. This can be obtained as follows.

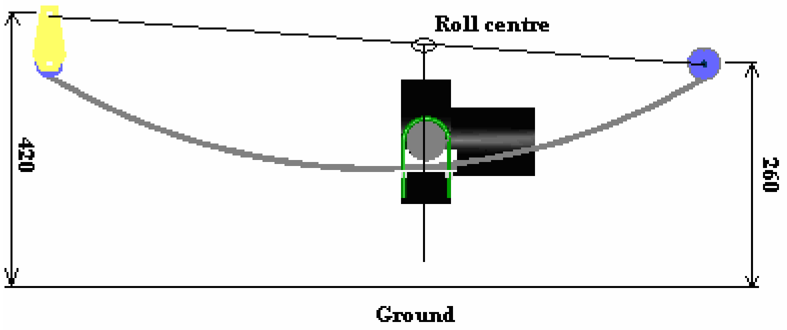

Figure 7.2: Roll centre location on a Hotchkiss axle.

The roll centre is determined by projecting a line from the top of the rear spring hanger to the front locating point of the leaf spring and then the point at which this line crosses the centre line of the axle housing is the roll centre.

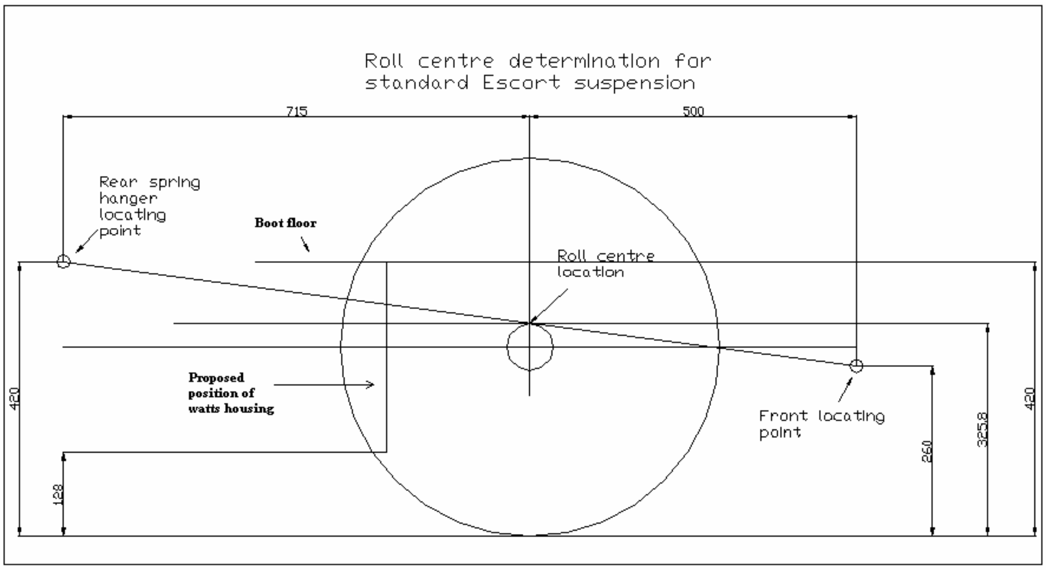

Figure 7.3: Roll centre location for standard suspension.

As can be seen the roll centre location is 325.8mm from the ground when the vehicle is at rest. This provides a good base line for the roll centre location to be set up initially for the new watts linkage.

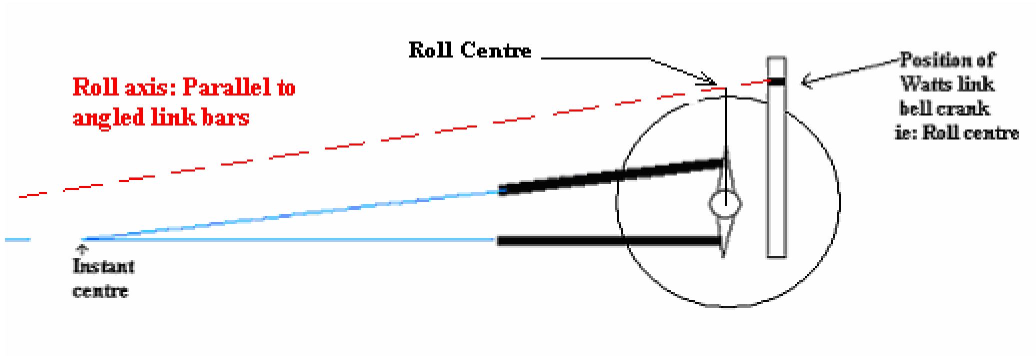

Figure 7.4: Roll centre location determination for new suspension design

With a four link and watts link suspension the roll centre is determined from the height of the watts link's bell crank centre bolt which is then projected along the roll axis to the centre of the differential housing. As previously mentioned the angle of the roll axis is defined by the angle of the link bars. For the case where the link bars and parallel and horizontal to the ground, the roll axis is parallel to the link bars and therefore parallel to the ground.

7.2 The role of the Watt's link

The first task of the watt's link is to locate the axle laterally and hence resist the lateral forces imposed on the tyres and then on to the watt's linkages whilst the vehicle is in motion, it will serve this purpose well as proven on many other road cars.

The second task of the watt's link is the determination of the roll centre location. As previously stated the height of the bell crank in the centre of the watt's housing helps to determine the roll centre location. The roll centre location is critical to the way in which the vehicle handles and must be considered in conjunction with the front roll centre of the suspension be set up correctly.

Effectively adjusting the height of the roll centre upwards puts more of the lateral loads through the linkages, whereas lowering the roll centre reduces the load transfer through the watt's linkage and therefore increases the loads absorbed by the springs (Dixon, 1996).

In practice lowering the rear roll centre will tighten corner handling, allowing the car to turn in to the corner harder and put the power down exiting the corner harder without excessive oversteer. Although, lowering the roll centre too much can cause extreme chassis roll depending on the springs being used and this causes issues with suspension geometry, namely excessive toe steer.

Excessive chassis roll is also renowned for delaying acceleration exiting corners due to it taking longer for the car to be able to put the power down properly as the weight across the car will not be being transferred equally thus allowing the wheel on the loaded side to have more traction than the wheel on the unloaded side, thus the car will become unstable (AFCO Racing. N.d.).

Raising the rear roll centre height will tend to reduce chassis roll, this tends to loosen corner handling allowing the car to oversteer from the rear.

Essentially, if the roll centre is adjusted too low the vehicle will experience excessive chassis roll and will be sluggish and slow to respond to driver inputs. A roll centre adjusted too high will tend to resist chassis roll and thus induce an oversteering rear end. This will make the rear of the car very sensitive and unforgiving to driver inputs (AFCO Racing. N.d.).

7.3 Adjustments to minimise roll skew

When setting up the rear roll centre assuming that the front suspension is already well set up, it is essential to consider the front roll centre of the chassis in relation to the rear roll centre as the difference between the two will determine the amount of roll skew the vehicle will have.

Firstly the front roll centre needs to be determined. The method of doing so is vastly different for the Macpherson strut type assembly used in the front of the Ford Escort than discussed for the rear suspensions above. Shown below is a basic representation of a Macpherson strut front suspension.

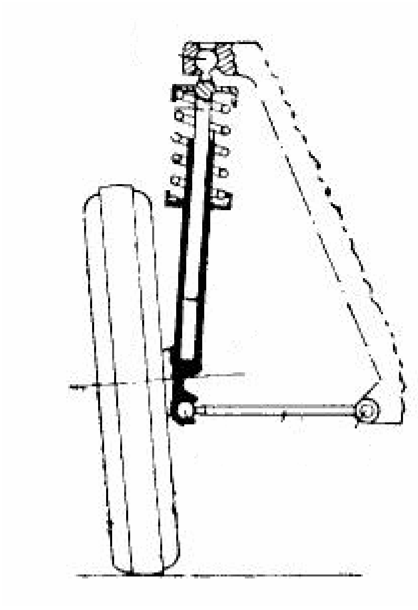

Figure 7.5: Macpherson strut front suspension assembly
http://www.f.kth.se/~f95-lsa/msc_thesis_work-filer/image010.jpg

Being an independent suspension, each side of the suspension has its own instant centre and as the suspension is symmetrical about the centreline of the chassis only one side of the suspension needs to be analysed to determine the roll centre of the suspension. The following diagram shows how the roll centre of the suspension was basically determined only looking at one side of the car.

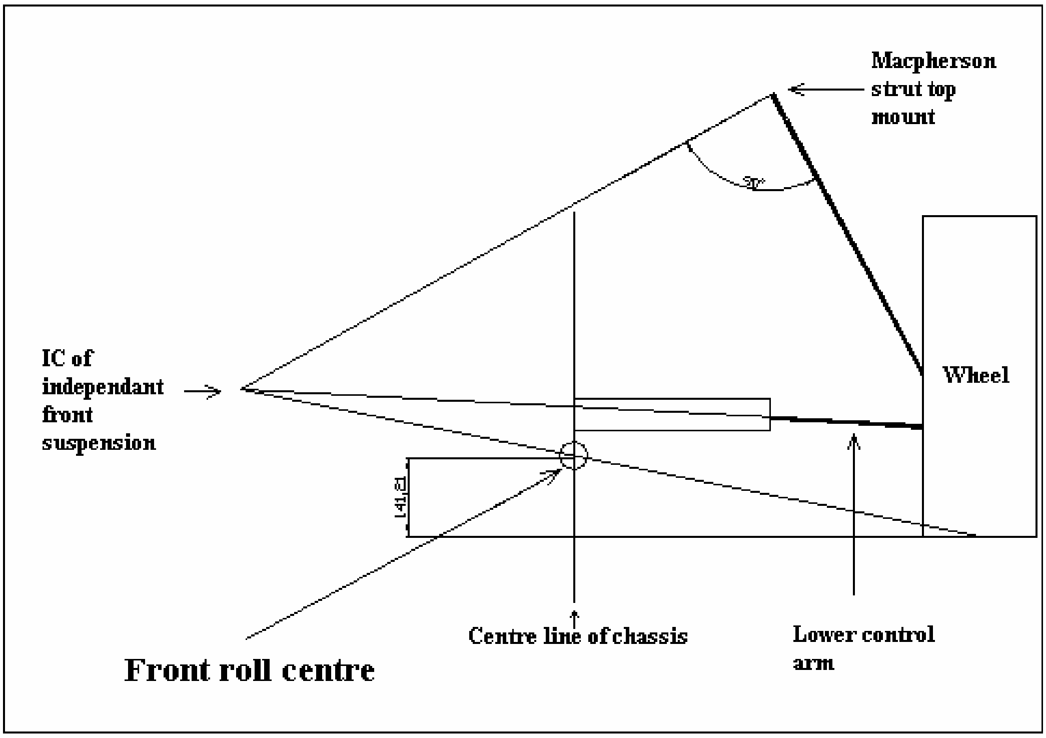

Figure 7.6: Front roll centre determination

As can be noted from the diagram above, the front roll centre is 141.21mm from the ground, which is substantially lower than that of the rear roll centre, which is 325.8mm from the ground.

As previously discussed it is desirable to minimise the difference in heights between the front and rear roll centres to minimise roll skew, although it is important to make sure that the dynamic roll centres to not cross each other whilst the car is in motion, as a sudden change from roll understeer geometry to roll oversteer geometry will cause erratic handling.

Watt's link design 7.4

The first piece of the watts link to design was the watts link housing which is to be affixed to the chassis by four 12mm bolts bolting through the floor pan to a 6mm steel plate on the inside of the boot floor. The roll centre height of the factory configuration was used as the baseline for this arrangement and as the baseline four link geometry is set parallel and horizontal the height of the hole in which bell crank is to pivot is set at the same 325mm off the ground as the roll axis will be horizontal, this in turn allows for the same roll centre to be achieved when the vehicle is static.

Taking into account that the boot floor in which the watts housing is to be mounted is 420mm from the ground and that the desirable roll centre height is 325mm from the ground, the design came together as shown below. The design also had to allow for suitable space for the bell crank to pivot in the housing.

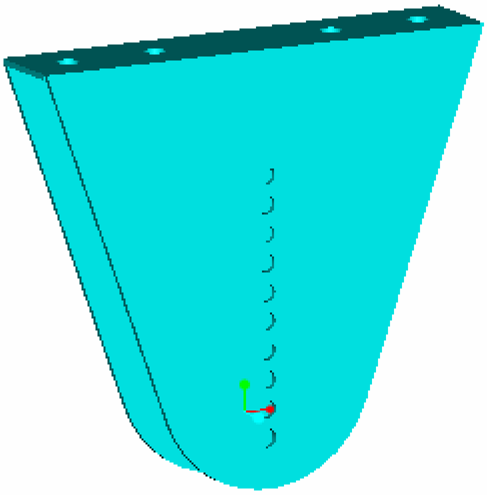

Figure 7.7: Watt's housing

This design is to be constructed from 3mm folded sheet metal and allows for adjustments to raise or lower the roll centre from its baseline position. As with the four link changes in the vehicle's ride height will always change the roll centre of the suspension and this configuration will allow for changes to be made to adjust for this.

A 6mm plate as shown below is mounted on the inside of the boot floor for the watts link housing to be bolted through to. The plate is to be stitch welded to the floor.

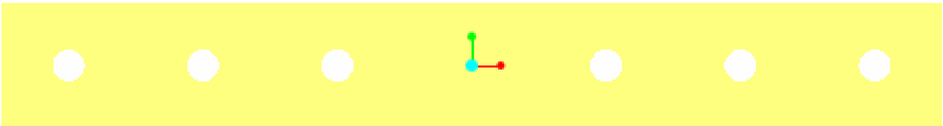

Figure 7.8: Watt's housing mounting bracket

The next piece to design for the watt's linkage is the bell crank. The bell crank attaches the two watt's link bars to the centre of the chassis on the watt's housing, allowing the bars to pivot in chassis roll yet resisting lateral motion of the axle in all circumstances.

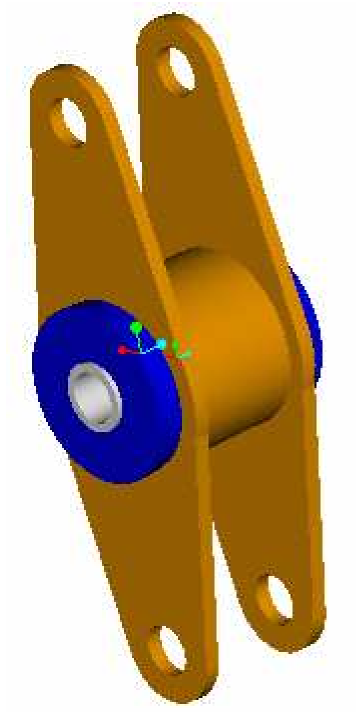

Figure 7.9: watt's bell crank

The two outer surfaces of the bell crank are cut out from 3mm sheet steel and a 33.4mm OD piece of seamless carbon steel pipe is welded into the centre of the two pieces of sheet steel allowing for a urethane Ford Escort lower front control arm bush to be used to dampen out some of the shock loadings imposed on the pivoting point in the centre of the bell crank.

Next is the watt's link bars. The watt's link bars are obviously to be located on the mounting points of the bell crank and from there they protrude out to their respective axle mounting points, which are to be conveniently protruded rearward from the four link axle brackets. Due to the four link brackets being mounted 280mm from the centre of the track on either side (560mm apart) the watts link bar lengths can be determined.

It has been decided to use the same seamless 26.7mm OD carbon steel pipe for the watts link bars as what was used for the four link bars. It has been chosen to use 3/8th spherical bearing rod ends on either end of the watt's link bars as it allows for reliable and solid mounting of the bars and as with the four link the plugs supplied with the rod ends are to be welded into the ends of the link bars to allow for the spherical rod ends to be screwed into the ends of the link bars.

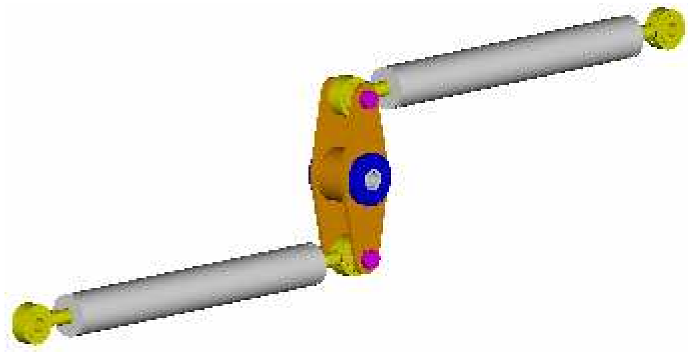

Figure 7.10: Watts link assembly 1

Shown above is the Watt's link assembly not including the watt's housing. The final piece of the design is the axle mounting brackets and as previously discussed the axle mounting brackets are to be built off of the existing four link birdcages.

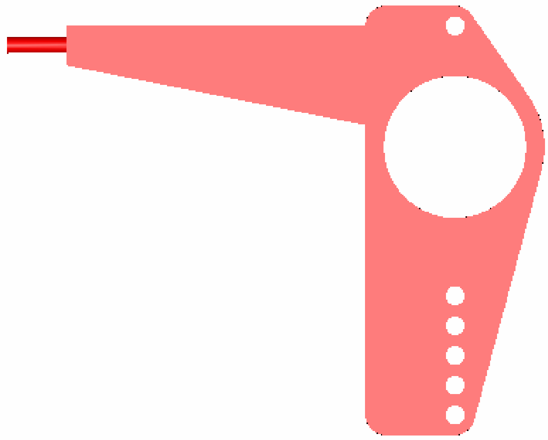

Figure 7.11: Watts link axle bracket and four link birdcage (Right side)

These mounting brackets are 3mm folded sheet steel once again and are to be seam welded to the four link birdcages. There is a 30mm 3/8 bolt welded to the bracket to allow for the spherical rod ends to be attached.

7.5 Watt’s link tuning

As formerly mentioned it has been decided to use the roll centre height of 325mm from the ground calculated from the factory Hotchkiss suspension configuration as the baseline roll centre setting as this is a setting that is known to relatively stable and from this baseline adjustments can be made to fine tune this setting.

Roll centre tuning starting at the baseline setting it is obvious that to level out the differences in heights between the front and rear roll centres the rear roll centre needs to be lowered.

The process for tuning the roll centre height is to take the car out on the track for a few laps and then come in and lower the roll centre height one notch at a time taking notice of the reduced tendency for the rear to oversteer and the increased tendency for rear body roll, at the setting when the car begins to roll too much and feel unresponsive to driver inputs when exiting corners adjust the belt crank up to the previous setting.

Generally where setting up the roll centre height is concerned, if the car has a tendency to roll too much whilst cornering this is indicating the roll centre to be too low or for the reverse where the car has a tendency to snap oversteer, this generally would indicate the roll centre to be too high.

The goal with setting up the roll centre is to achieve a car that corners flat, thus sharing equal loads on each wheel and achieving more traction.

Chapter 8

Coil-over mounting and selection

8.1 Turret selection and fitment

Before deciding on the spring and damper configuration, it was first necessary to look into the housing or turret in which the coil-overs are to be mounted. Initially it was thought that this was something that would need to be designed for this project, although it was later discovered that the Ford Escort rally cars of the 1970's used vertically mounted shock absorbers to allow for more suspension travel needed to cope with the harsh off road conditions encountered in rallying.

It was found that a company referred to as Palmside Motorsport Parts located in New Zealand sell two different types of turret to suit the escort as shown below.

Figure 8.1: Round and Square turrets

For purely aesthetic reasons it has been chosen to use the round turrets, although in reality it wouldn't matter which type are used. These turrets can be landed in Australia for around $340.00 including freight. The turrets are supplied with axle mounting brackets (Shown below) that are to be welded to the top of the axle housing. These brackets simply allow for the base of the coilover to be bolted through on to the axle housing.

Figure 8.2: Axle bracket for coil over

8.1.1 Fitment of turrets to the vehicle

When measuring up to fit the turrets it is important to have the rear axle fitted up to the car with the four link and watts link attached at their base line settings and for the axle to be raised up into the chassis (which will be raised from the ground on a hoist or stands) to approximately the height where it is desired for the axle to sit relative to the chassis when the car on the ground with the wheels fitted. From here the centreline of the axle / differential housing should to be determined.

With the centreline of the axle found, project and scribe a vertical line up the inside of the of the wheel arch. Given that the width of the turret is 100mm, now mark two lines 50mm either side of the centreline which has just been scribed along the inner section of the wheel arch. Now sitting the turret on the inside of the wheel arch, cut out the small section of floor until the turret fits down against the chassis rail. Now sitting the turret in the position where it is to be fixed, scribe a line around the top of the turret and then remove the turret and scribe a line 10mm inside of it the whole way round, now cut out the unwanted section of floor and fold up a 10mm lip, clean the weld areas and stitch weld the turret in position.

Once the coil overs have been purchased they can be set in place and attached to the turret at the top and the axle brackets purchased with the turrets can then be a fixed to the bottom of the coilover units and the axle offered up on to the brackets and check for clearance all around, i.e. the coil overs are centred in the turrets and that the wheels are tyres are going to clear the coilovers. If everything check's out, tack weld the bracket to the axle and remove the coil overs from the brackets and lower the axle down and then proceed to fully weld the brackets.

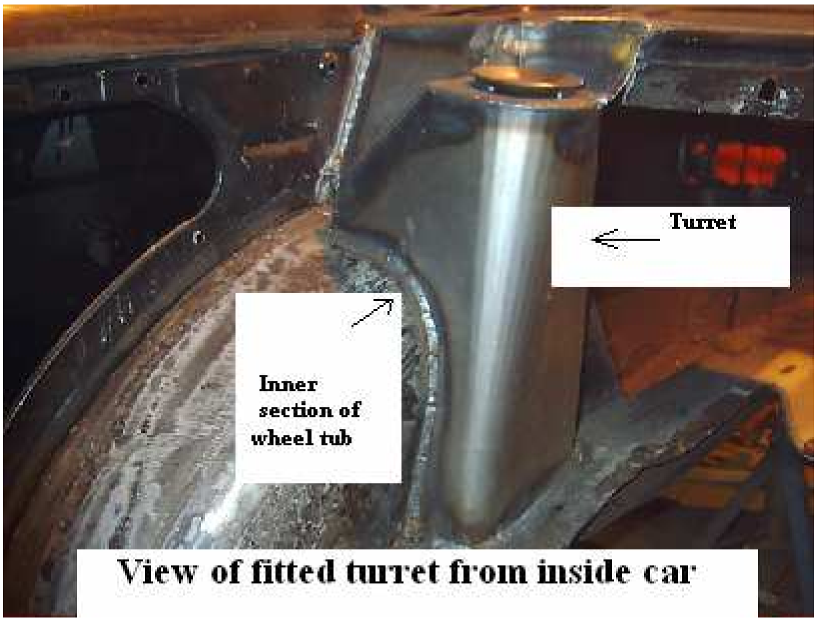

Figure 8.3: Turret fitted by Retromotorsport

8.2 Coil-over selection

Figure 8.4: Gaz Coilovers

There are many different spring and damper / shock absorber manufacturers that all make high quality products, generally the chosen make comes down to price and availability to the car builder. In this case it has been chosen to use the Gaz brand of coilovers.

The process of selecting the length of damper needed is as follows. After the coilover turrets have been fitted and the axle is in situ with the four link attached, raise the axle up into the chassis until it hit the chassis rails on either side. Now placing the axle mounts for the turrets on top of the axle, measure from the centre (ie where the bolt goes through) of the axle bracket up to the top of the inside of the turret, this length measured will be the closed length of the shock absorber.

Initial measurements have indicated that the closed length of the damper/ shock absorber needs to be around 300mm, although it would pay to remeasure this distance once the turrets and four linked axle are fitted up to the car. When referring to page 15 of the Gaz catalogue (appendix C) it can be noted that all the measurements are taken in inches, in that case 300mm is equal to 11.8” . The chosen coilovers are the 190/120R12-2’s which have a closed length of 12” (305mm) and an open length of 19” (483mm) 2” bodies (51mm) and 12mm rods. The 12mm rod is threaded at the top of the coilover to allow for it to be attached to the turret, whereas at the other end spherical bearings are used to attach the base of the coilovers to the axle. The spherical bearings have been chosen over the bonded bushes for added reliability as the two coilovers now will have the responsibility of carrying the weight of the rear of the car and therefore the bearings will be less prone to wear than the bushes.

As specified in the Gaz catalogue, the selected coilovers are multi adjustable. The units are adjustable for bump and rebound by simply twisting a control knob on the side of the unit. These coilovers suit 2.25”ID coil springs which are used in most coilover applications, therefore a wide host of spring lengths and spring rates are available for these units.

8.3 Spring selection

The coil springs to be used with the chosen coil overs above are 2.25" ID units. Gaz are able to supply springs to suit for a whole range of differing spring rates. Now would be a good time to define what is exactly is meant by spring rate.

The spring rate is determined simply by the amount of mass needed to compress the spring one inch and is usually measured in pounds (lbs) per inch. There are three factors that are used to determine the spring rate of a particular coil spring.

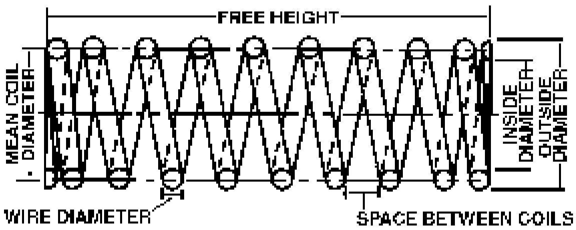

Figure 8.5: Determining the spring rate

The first is the wire diameter of the spring, obviously the thicker the wire, the higher the spring rate. Next is the mean diameter of the spring, when the mean diameter increases, so does the spring rate. Lastly is the number of active coils. The number of active coils includes the total number of coils on the spring in question minus one for springs with one end closed and the other open. The higher the number of active coils is, the lesser the spring rate will be (AFCO Racing, n.d).

There are generally two different types of coil spring used in motor vehicles, the first being the linear rate coil spring and the second being the progressive rate coil spring.

A linear spring has a constant spring rate throughout its range, therefore its rate is not affected by changing load put onto the spring. The spring rate of progressive rate springs increase progressively as they are compressed. Generally when the spring is compressed the coils at the bottom of the spring (which have a greater number of active coils per given unit of measurement as compared to the top of the spring) come into contact with each other and therefore cease springing, leaving the portion of spring with the lesser number of active coils and thus higher spring rate to control the springing, effectively increasing the spring rate. There are also cases where springs consist of different diameter wire or different diameter coils which also yield the same result (AFCO Racing, n.d.).

Most race cars use linear spring rates as they are more predictable and allow for the rest of the suspension to be tuned more easily. Varying loads on different sides of the car will cause the progressive springs to resist motion at different rates to each other bringing a whole other complexity into the suspension tuning as the car well essentially be prone to roll more on one side as compared to the other.

The chosen spring will be a Gaz supplied 2.25” ID linear spring. The chosen rear spring rate will have a lot to do with the front spring rate in which is being used on the car as on most race cars the un-driven end of the car, which is the front end in this case is usually very stiffly sprung relative to the driven end (rear end) as this keeps drive wheels more evenly loaded for traction and in the case of the Ford Escort the fact that there is more weight over the front wheels than the rear wheels must be taken into account.

Essentially spring rates affect the lateral load transfer distribution. Lowering the spring rate on one end of the vehicle will even out the loads on that particular end and this will raise the lateral force available from that end (Milliken, Milliken. 1995).

It has been recommended to use 250lbs front springs and 190lbs rear springs as a starting point and then from there, trialling different springs with different spring rates will soon determine what is suitable. The aim is to choose a relatively stiff spring that will allow for the suspension to be rigid and avoid excessive roll and squatting under load, but be soft enough to allow for the suspension to work as intended.

8.4 Basic workings of a damper

Before going into the basics of tuning the dampers for the track, it is first important to gain an understanding of the role of the damper or shock absorber and how it operates. The damper is most commonly known as the shock absorber, although the thought that shocks are absorbed is misleading somewhat.

The role of the damper is to dissipate any energy in the vertical motion of the body and springs mainly caused from control inputs or road surface variations.

As a whole, a vehicle can be considered a vibrating system in between its sprung and unsprung masses in which is in need of dampers to optimize control behaviour by preventing response overshoots and to minimise some unavoidable resonances. The mathematical theory of vibrating systems fundamentally uses the notion of the linear damper (where force is proportional to extension speed) mainly due to the fact that this model yields equations for which the solutions are well documented and understood. The typical hydraulic damper functions approximately in this manner (Dixon, 1996).

The telescopic damper is the most common form of damper fitted to most road and race cars of today. There are two main types of telescopic damper, the single tube type and the double tube type.

Figure 8.5: Single tube and double tube dampers

They are most easily understood in reverse order of invention. The single tube (mono tube) damper is purely a piston and rod in a fixed volume cylinder containing a mixture of oil and nitrogen gas. The piston contains two valves, one which controls the pressure differential across the piston when the suspension is going into bump and the other valve controls the pressure differential across the piston during rebound (Dixon, 1996).

When the suspension is going into bump the damper compresses and the piston rod is forced into the chamber, thus forcing a pressure rise in the chamber which the oil/gas mixture in the chamber try to resist, therefore dampening the shock imposed on the body. The single tube damper shown in the previous figure is referred to as an anti-emulsion single tube damper as it has the gas separated at the bottom of the chamber by a floating piston, although it works similarly in operation to this emulsified type damper just discussed. This design of damper has little problems with reduced efficiency due to the oil in the damper overheating in continually harsh conditions and for this reason, this type of damper is popular in rallying and other types of off road racing (Dixon, 1996).

Rearranging the reservoir as an annulus around the main working cylinder yields the double tube type telescopic damper, this design is more prone to overheating than the single tube type, although there are means to minimize this. The disadvantage of the double tube design is that the reservoir must be mounted to the unsprung side which is forced to deal with severe oscillations when the vehicle is in motion. The single tube damper does not suffer from this, although is more susceptible to damage due to the active cylinder being exposed (Dixon, 1996).

8.5 Damper tuning

The chosen Gaz coilovers are adjustable for both bump and rebound, although what exactly does these mean?

Firstly, adjusting the bump damping will be discussed. The function of a damper in bump is to control how the damper reacts when it is compressed. If the bump damping is too soft the suspension will oscillate too much reducing the stability of the car causing the car to bounce and walk all over the track effectively reducing the contact the wheels have with the road surface. If the dampers are set too hard, the ride quality will be harsh and the car will tend to become unstable and sensitive to drive inputs with regards to understeer and oversteer (Thorney Motorsport, n.d.).

One method of damper tuning is to set the dampers for soft for both bump and rebound and then drive the car for a few laps trying to ignore chassis roll and mainly concentrating on how the car feels whilst entering corners especially on rough surfaces. Increase the bump stiffness a few clicks and take the car back out and the track, continue to increase the bump stiffness until the car starts to oversteer on entry to corners signifying that the bump stiffness has been set too high. When this occurs back the bump stiffness off slightly and the dampers are now tuned, Obviously this damper tuning will coincide with tuning of the front dampers on the car as well (Thorney Motorsport, n.d.).

Now to look at setting up the rebound damping, this controls how the car reacts to dips in the road surface (the opposite of a bump) as well is how quickly the suspension reaches its roll limit whilst performing a turn. Remembering that the rebound damping has already been set to the soft setting, take the car out on the track once more this time taking notice of how quickly the car rolls into corners. Increase the damping stiffness and continue to do so until a point is reached where the car does not have any aggressive pitch changes when coming into corners and the amount of chassis roll is minimised and then soften up the rebound slightly (Thorney Motorsport, n.d.).

Changes to rebound damping does not affect the amount of chassis roll, although it affects how it is attained. As previously discussed roll is controlled by spring rates and roll centre heights.

Chapter 9

Conclusions and final points

It is important to realise that a race car is to be considered as a whole system and that each individual part of the car has a specific purpose and that changes to one sole component can directly influence the performance of the rest of the car. The rear suspension is just one small portion of the car to consider, it should be designed to work in with the rest of the car, and it will never be capable of functioning at its full capacity if the rest of the car isn't set up to suit.

Some of the other major sections that are subject to modification when a road car is converted into a race car are of course the wheels and tyres, the front suspension, the brakes, the engine and driveline and the chassis and or body shell itself, each and every part has to work together for a desirable result to be achieved in the way in which the car functions. Most of the time there will be at least one part of the car that will be letting down the rest. The design and construction of a race car is one of constant revisions and refinements. Improving on one part of the car will lead to the weak point of another and so forth, thus it is important to be able to tune the whole car to run at the best of it capabilities for a particular scenario.

9.1 Final comparisons

The new rear suspension design is vastly different than the factory configuration. The figures below show the two configurations excluding dampers (and coils for the new configuration).

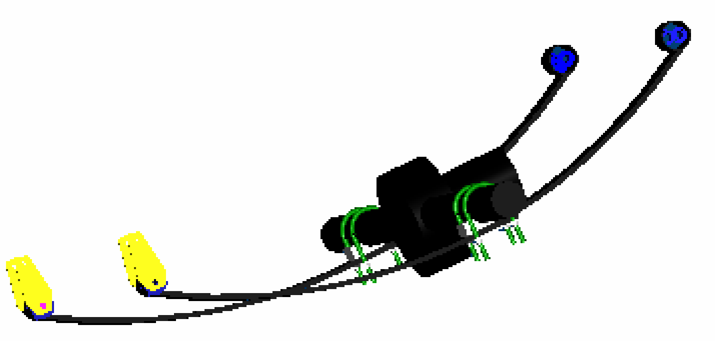

Figure 9.1 Standard rear suspension

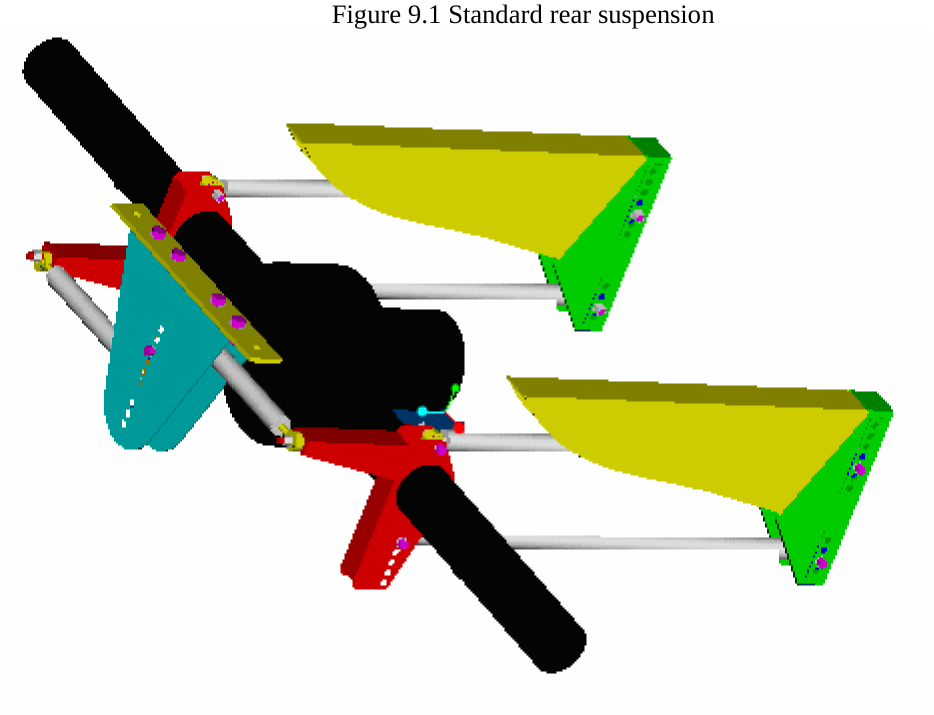

Figure 9.2a: New rear suspension

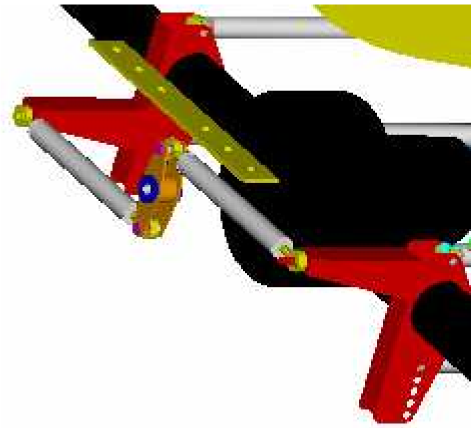

Figure 9.2b New rear suspension (hidden watt's housing)

The factory configuration has simplicity in its favour over the new design, although bar that one aspect, the new design is far superior.

With regards to the longitudinal location of the axle, the factory leaf springs will always allow for excessive caster changes in variances of ride and roll due to the springs lengthening and moving the axle rearward when loaded and the reverse occurring when unloaded and there is little than can be done to avoid this. The leaf springs will also always have roll oversteer geometry which is not usually desirable. Due to being a deformable spring the leafs are also subjective to excessive axle wind up.

The four link design allows for caster change and roll steer to be minimised and controlled with the available adjustability. Simple changes to the mounting points of the four link bars will effectively change the geometry of the rear suspension and this allows for the suspension to be tuned to suit a particular set of track conditions. Due to the axle being located at the top and bottom on both sides by four essentially rigid links axle wind up is negligible.

With regards to the lateral location of the axle, the leaf springs are fairly rigid when subjected to lateral loads, although they are still known to be compliant to an extent. With the watt's linkage arrangement, again it is essentially a rigidly mounted linkage arrangement and this allows for the axle to be constrained from any lateral motion relative to the chassis of the vehicle.

The only way to vary the roll centre on a Hotchkiss suspension is to change the ride height of the suspension, which along with changing the roll centre will effectively shift the centre of gravity of the car which can prove to be an issue in itself, where as the watt's link design in this case allows for roll centre height to be changed by simply moving the bell crank pivot point to one of the other available mounting positions.

Another disadvantage of the Hotchkiss suspension is that to change the spring rate which involves changing the springs themselves (in either case) it is a fairly involved task, whereas when using a coilover arrangement, changing the springs will be a much simpler task due to being only attached at two points. The same goes for changes in ride height, coilovers allow for height adjustment by simply raising or lowering the spring platforms, whereas with leaf springs lowering blocks need to be fitted or preferably the springs need to be removed from the car and reset to the desired height by a spring specialist.

Besides the issues with variances in geometry and axle wind up another major pitfall with the Hotchkiss rear suspension configuration for racing is that it is impossible to change one aspect of the suspension without affecting the rest of the suspension and to make a change usually requires replacing the spring altogether or sending it away to a spring specialist which takes time. For example choosing a softer spring rate will lead to the axle having a lesser resistance to axle wind up.

The new suspension configuration allows for quick and easy adjustments to be made to many different aspects of the suspension without affecting the geometry or function of other aspects, and this allows for the suspension to be tuned much more easily.

The adjustability of the new suspension configuration allows it to be a much less compromised configuration as it can be fine tuned for each set of conditions with ease in between practice sessions and races, rather than the factory Hotchkiss configuration which allows for no adjustability (other than damping, if adjustable units are fitted), so the suspension will usually be set up for the best performance it can offer across all conditions in which will be encountered as no adjustments in geometry can be made.

9.2 Further work

Before the construction of this suspension design begins it would be advisable to conduct FEA testing on the whole model for simulated wheel loads and applied drive train torques to gain a knowledge of the forces acting on each component. This will add some reassurance for the ability of the chosen materials to withstand the forces in which they will be subjected to.

It was initially planned to include some FEA testing in this dissertation, although due to the unforeseen circumstance of the USQ Engineering faculty's Ansys license being outdated for most of the second semester it did not become a reality.

More research into selecting spring rates using predicted wheel loads and ride and roll rates may prove beneficial over the guess and check method used in this case.

Finally to build, implement and test the design and compare it with the factory suspension. Once the new suspension is tuned its superiority should be quickly noted by the ultimate track performance indicator, the stopwatch!

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Appendix A

Project specification

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG4111/4112 Research Project
PROJECT SPECIFICATION

FOR: **Damion Parkin** *(0050025797)*

TOPIC: Design of a rear suspension configuration for a live axle race car to achieve optimum handling characteristics.

SUPERVISOR: Dr. Jayantha Ananda Epaarachchi

SPONSORHSIP: USQ

PROJECT AIM: Redesign the standard rear suspension setup of a Ford Escort from a leaf sprung unlinked configuration to a coil sprung, four link and watts link arrangement allowing for adjustment in roll centre, anti squat and ride height and use a computer model and simulation to indicate the improved performance of the system.

PROGRAMME: **Issue A, 26 March 2007**

1. Introduce the function of suspension in a motor vehicle, identify the two main types of rear suspension. Define all of the key elements in a rear suspension system along with some terms relating to suspension geometry.

2. Establish the advantages of a coil sprung multilinked suspension configuration over the standard Ford Escort arrangement.

3. Research handling factors including steady state cornering, anti squat geometry and roll centre analysis.

4. Research and design the coil over turrets, four link boxes and arms and watts linkage setup for ideal geometry, allowing for minor anti squat and roll centre adjustments. Investigate and select suitable coil overs for the above setup.

5. Model current and proposed rear suspension setups and carry out FEA testing on wheel loads and applied drive train torques.

6. Carry out material selection for the new suspension configuration now that the stresses on each of the individual components is known due to the FEA tests.

7. Use a suspension geometry package to model the variances in toe, camber, caster, roll steer, etc, for variances in ride and roll values and make adjustments to optimise the geometry as required.

AGREED (student)
(supervisor)________________________________ ________________________________

Date: / / 2007 Date: /
/ 2007

Co-examiner:______________________________

Appendix B

Tabulated Wingeo results

The following tables contain the data generated by Wingeo3 for the four link for changes in caster and steer values for varying ride and roll values.

Steer and roll are in degrees and caster and ride are in mm. A positive roll value indicates the car is rolling right (left hand turn) and a positive caster value indicates that the axle is moving rearward of its initial position.

Caster change with respect to change in vertical ride height for 600mm link bars set up parallel and horizontal					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
-25	0	0	0	0	0
-50	0	0	0	0	0
-75	0	0	0	0	0
0	0	0	0	0	0
25	0	0	0	0	0
50	0	0	0	0	0
75	0	0	0	0	0

Table B.1

Caster change with respect to change in vertical ride height for 600mm link bars set up for 100% anti squat					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
-75	0	-2.753	-2.753	0	0
-50	0	-1.834	-1.834	0	0
-25	0	-0.917	-0.917	0	0
0	0	0	0	0	0
25	0	0.92	0.92	0	0
50	0	1.848	1.848	0	0
75	0	2.785	2.785	0	0

Table B.2

Caster / steer changes with respect to change in roll values for 600mm link bars set up parallel and horizontal					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-0.345	-0.346	-0.198	0.198
0	10	-0.116	-0.116	-0.062	0.062
0	5	-0.02	-0.02	-0.008	0.008
0	0	0	0	0	0
0	-5	-0.003	-0.003	0.009	-0.009
0	-10	0.02	0.02	0.074	-0.074
0	-15	0.109	0.109	0.258	-0.258

Table B.3

Caster / steer changes with respect to change in roll values for 600mm link bars set up for 100% anti squat					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	1.708	-2.677	0.37	-0.37
0	10	1.275	-1.662	0.321	-0.321
0	5	0.693	-0.779	0.184	-0.184
0	0	0	0	0	0
0	-5	-0.77	0.703	-0.183	0.183
0	-10	-1.584	1.353	-0.31	0.31
0	-15	-2.48	1.967	-0.314	0.314

Table B.3

Caster / steer changes with respect to change in roll values for 600mm both link bars angled up 1.43 degrees from horizontal					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-0.368	-0.367	0.171	-0.171
0	10	-0.126	-0.126	0.187	-0.187
0	5	-0.022	-0.022	0.117	-0.117
0	0	0	0	0	0
0	-5	-0.005	-0.005	-0.116	0.116
0	-10	0.01	0.01	-0.175	0.175
0	-15	0.087	0.087	-0.111	0.111

Table B.4

Caster / steer changes with respect to change in roll values for 600mm both link bars angled up 4.29 degrees from horizontal					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-0.365	-0.362	0.909	-0.909
0	10	-0.124	-0.124	0.683	-0.683
0	5	-0.022	-0.022	0.366	-0.366
0	0	0	0	0	0
0	-5	-0.005	-0.004	-0.365	0.365
0	-10	0.013	0.014	-0.671	0.671
0	-15	0.092	0.094	-0.849	0.849

Table B.5

Caster / steer changes with respect to change in roll values for 600mm upper link bar angled up 2.86 degrees and lower link bar horizontal					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-1.513	1.031	0.213	-0.213
0	10	-0.91	0.79	0.214	-0.214
0	5	-0.429	0.422	0.13	-0.13
0	0	0	0	0	0
0	-5	0.444	-0.407	-0.13	0.13
0	-10	0.963	-0.736	-0.201	0.201
0	-15	1.608	-0.935	-0.15	0.15

Table B.6

Caster / steer changes with respect to change in roll values for 600mm both link bars angled down 2.86 degrees					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-0.3	-0.305	-0.937	0.937
0	10	-0.096	-0.097	-0.559	0.559
0	5	-0.015	-0.015	-0.258	0.258
0	0	0	0	0	0
0	-5	0.002	0.002	0.258	-0.258
0	-10	0.039	0.039	0.57	-0.57
0	-15	0.153	0.152	0.997	-0.997

Table B.7

Caster / steer changes with respect to change in roll values for 600mm upper link bar angled up 2.86 degrees and lower link bar down 1.43 degrees					
Ride	Roll	Caster Left	Caster Right	Steer Left	Steer Right
0	15	-2.079	1.74	0.05	-0.05
0	10	-1.298	1.253	0.104	-0.104
0	5	-0.631	0.646	0.075	-0.075
0	0	0	0	0	0
0	-5	0.67	-0.607	-0.074	0.074
0	-10	1.445	-1.106	-0.091	0.091
0	-15	2.381	-1.438	0.014	-0.014

Table B.8

Appendix C

Materials, parts needed and parts to be manufactured

The following is a list of all of the parts and materials required for this design to be constructed including parts that need to be fabricated and others that need to be purchased. Information regarding the chosen seamless pipe, spherical rod ends and coilovers used is included at the end of this appendix

C.1 Complete parts list

The following table contains all of the components required to construct this design followed by illustrations of each component.

Components to be fabricated			
Item no.	**QTY**	**Description**	**Materials used**
1	2	Front link box	1.6mm Sheet steel (double skinned over bolt holes)
2	2	Front link box covers	1.4mm Sheet steel
3	4	Four link bar	26.7mm OD & 33.4mm OD end carbon steel seamless line pipe
4	2	Watt's link bar	26.7mm OD carbon steel seamless line pipe
5	1	Bell crank	3mm sheet steel & 33.4mm OD Carbon steel seamless line pipe
6	1	Watt's top plate	6mm steel plate
7	1	Watt's housing	3mm sheet sheel
8	1	Left bird cage and watt's mount	3mm sheet steel
9	1	Right bird cage and watt's mount	3mm sheet steel
Components to be purchased			
Item no.	**QTY**	**Description**	**Material / Grade**
10	3	Escort lower control arm bushes	Poly Urethane
11	7	3/8" x 60mm bolt	Grade 8.8 High tensile steel
12	24	3/8" Flat Washer	Zinc plated steel
13	4	3/8" x 45mm bolt	Grade 8.8 High tensile steel
14	2	3/8" x 30mm bolt	Grade 8.8 High tensile steel
15	4	M12 x 35mm bolt	Grade 8.8 High tensile steel
16	20	3/8" castle nut	Zinc plated steel
17	6	3/8" Nut	Zinc plated steel
18	4	M12 Nut	Zinc plated steel
19	4	3/8" Sperical bearing rods end	Surface hardened low carbon steel body
20	2	3/8" Sperical bearing rods end (left hand thread)	Surface hardened low carbon steel body
21	2	Round Shock absorber turrets	
22	2	Gaz coil overs	

Table C.1: Complete parts list

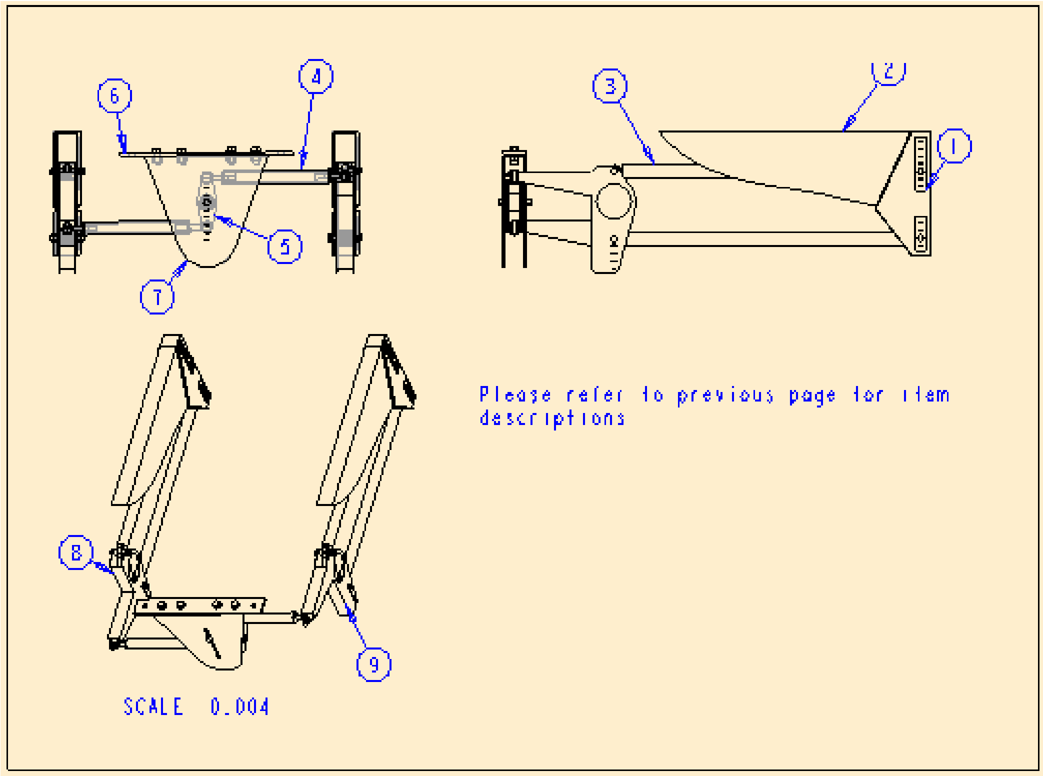

Figure C.1: Assembly drawing indicating major components

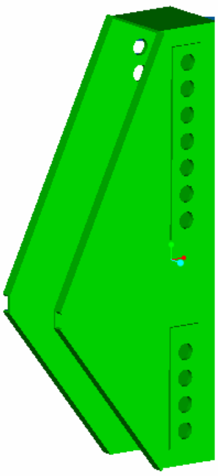

Figure C.2: Item 1; Front link box

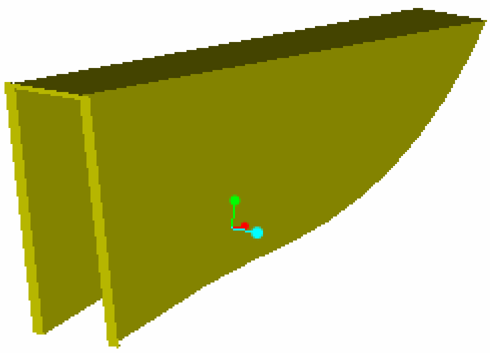

Figure C.3: Item 2; Link box cover

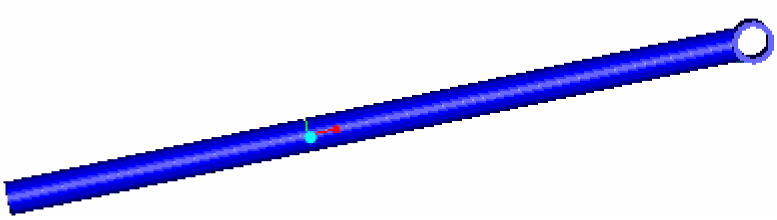

Figure C.4: Item 3; Four link bar

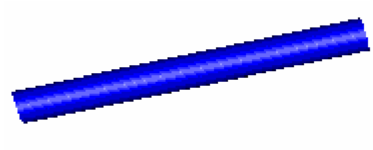

Figure C.5: Item 4; Watt’s link bar

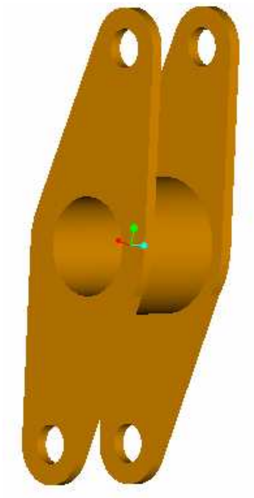

Figure C.6: Item 5; Bell crank

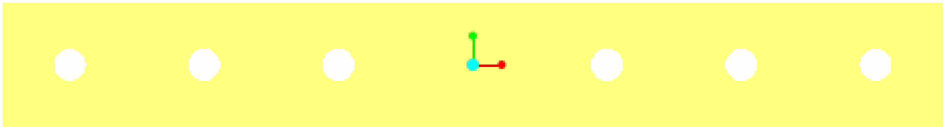

Figure C.7: Item 6; Watt’s top plate / mounting bracket

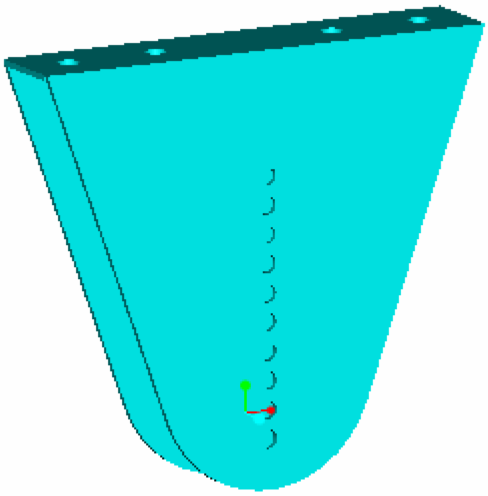

Figure C.8: Item 7; Watt's housing

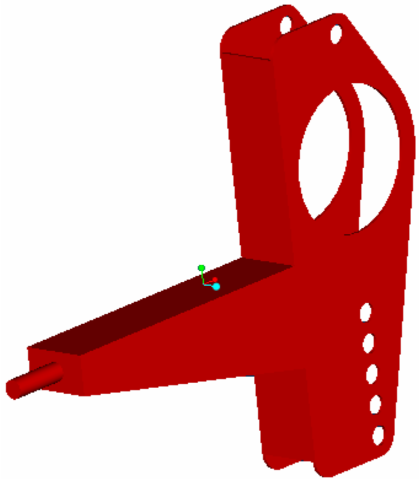

Figure C.9: Item 8; Left bird cage and watt's mount

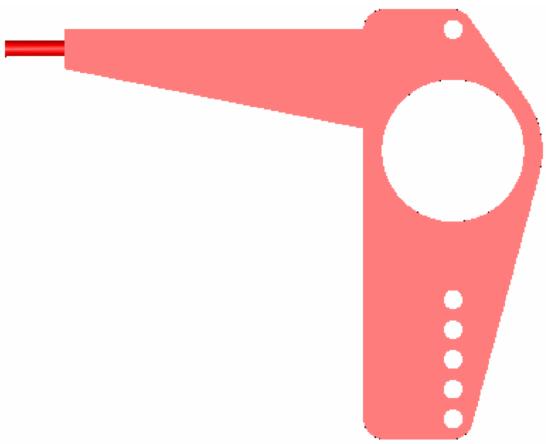

Figure C.10: Item 9; Right bird cage and watt's mount

Figure C.11: Item 10; Escort lower control arm bushes

Figure C.12: Items 11 through 18; Various Nuts, bolts and washers

Figure C.13: Items 19 & 20; 3/8 Male Rod ends
supplied with end plugs to weld into link bars and lock nuts

Figure C.14: Item 21; Round shock absorber turret

Figure C.15 Item 22; Gaz coilover

C.2 Material / component specifications

The following includes information about the seamless pipe used, the spherical rod ends used and the Gaz coilovers chosen for this application.

The chosen pipe for the link bars is a grade A carbon steel that has an OD of 26.7mm and an thickness of 7.82m.

The pipe that is used on the ends of the four link bars and in the centre of the bell crank has an OD of 33.4mm and a thickness of 9.09mm and is also a grade A carbon steel.

The spherical bearing rod ends used are part numbers EM6 (four of) and EML6 (two of). The difference is the EML6 has a left hand thread which is used on one end of each of the watt's link bars. It allows for the length of the bars to be changed more easily.

The chosen coilovers are 190/120R12-2 as listed on page 15 of the Gaz catalogue.

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SEAMLESS LINEPIPE

OUR STANDARD STOCK PIPE IS MANUFACTURED TO ASTM A106B

SPECIFICATION – ASTM A106

Black seamless (welded not permitted) fully killed Carbon steel pipe for high-temperature, high-pressure service in three grades of seamless pipe of varying strength.

Although the physical and chemical properties for Grades A and B are comparable to those for A53 pipe, and the types of testing required for both specifications are similar, the tests prescribed for A106 are more stringent and are applied to smaller lots of pipe. Therefore, A106 is preferred for exacting services.

Grades A and B are obtainable in most sizes and schedule numbers, Grade B permits higher carbon and manganese contents than Grade A. The A106 Grade B supplied by Smorgon Steel Pipeline Supplies has a maximum carbon content of 0.23%.

PIPE & PIPING COMPONENTS SPECIFICATION SUMMARY

CARBON STEEL

GRADE A			GRADE B			GRADE C			LOW TEMPERATURE			HIGH YIELD		
PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES
A53-A A106-A A135-A A139-A A155-C50,C55 API-5L-A	A234-WPA	A105	A53-B A106-B A135-B A139-B A155-KC65,KC70	A234-WPB	A105	A106-C	A235-WPC	A105	A333-6	A420-WPL6	A350-LF2	A381-35 API5LX-X42 API5LX-X46 API5LX-X52	WPY35** WPY42** WPY52** WPY52**	AI05 A182-F1

CHROME MOLLY STEEL

1/2Cr-1/2Mo			1Cr-1/2Mo			11/4Cr-1/2Mo			21/4Cr-1Mo			5Cr-1/2Mo		
PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES
A155-1/2CR A335-P2 A369-FP2	GRADE WP2	A182-F2	A155-1CR A335-P12 A369-FP12	A234-WP12	A182-F12	A155-11/4CR A335-P11 A369-FP11	A234-WP11	A182-F11	A155-21/4CR A335-P22 A369-FP22	A234-WP22	A182-F22	A155-5CR A335-P5 A369-FP5	A234-WPF5	A182-F5

LOW TEMP FERRITIC STEEL

31/2N			Cu-Ni LOW ALLOY STEEL		
PIPE	WELDING FITTINGS*	FLANGES	PIPE	WELDING FITTINGS*	FLANGES
A333-3	A420-WPLC	A350-LF3	A333-9	A420-WPL9	A350-LF9

CARBON MOLLY STEEL

1/2 Mo		
PIPE	WELDING FITTINGS*	FLANGES
A155-CM70 A335-P1 A369-FP1	A234-WP1	A182-F1

NOTE * WHEN FITTINGS ARE MANUFACTURED USING WELDED CONSTRUCTION THE GRADE SYMBOL WILL INCLUDE A 'W'

** THERE IS NO ASTM SPECIFICATION THEREFORE THE STANDARD NOMINATED IS MMS SP-25

Pipe Ends

Unless otherwise specified pipe ends are normally supplied as below:

- **a.** Up to and including 48.3mm O.D size are supplied with plain ends cut square;
- **b.** Above 48.3mm O.D sizes (except for Double Extra-Strong pipe) are supplied with plain ends bevelled;
- **c.** All Double Extra-Strong pipe is supplied with plain ends cut square.

End Preparation

- **a.** Bevelled ends for API steel linepipe are normally to API specification ie: Angle 30° +5° -0°
- **b.** Bevelled ends for steel pipe to ASTM specifications are normally to ANSI B16.25 ie: Angle 30° +5° -0°

Dimensional Standards

The Dimensional standards for pipe to ASME B36.10 includes products covered by the following material specifications.
ASTM – A53, A106, A134, A135, A139, A333, A335, A369, A376, A381, A405, A409M, A426, A523, A524 and A530
API – 5L, 5LX and 5LS

SEAMLESS LINEPIPE

Issued January 2004

SMORGONSTEEL

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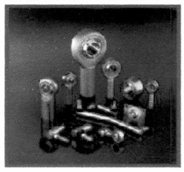
SEAMLESS LINEPIPE

DIMENSIONS & WEIGHTS

		SCHEDULE													
NOMINAL PIPE SIZE mm	OD mm	10	20	30	STD	40	60	XS	80	100	120	140	160	XXS	DETAILS
6	10.3				1.73			2.41							WALL THICK mm
					0.37			0.47							WEIGHT kg/m (PLAIN END)
8	13.7				2.24			3.02							WALL THICK mm
					0.63			0.80							WEIGHT kg/m (PLAIN END)
10	17.1				2.31			3.20							WALL THICK mm
					0.84			1.10							WEIGHT kg/m (PLAIN END)
15	21.3				2.77			3.73					4.78	7.47	WALL THICK mm
					1.27			1.62					1.95	2.55	WEIGHT kg/m (PLAIN END)
20	26.7				2.78			3.91					5.56	7.82	WALL THICK mm
					1.69			2.20					2.90	3.64	WEIGHT kg/m (PLAIN END)
25	33.4				3.38			4.55					6.35	9.09	WALL THICK mm
					2.50			3.24					4.24	5.45	WEIGHT kg/m (PLAIN END)
32	42.2				3.56			4.85					6.35	9.70	WALL THICK mm
					3.39			4.47					5.61	7.77	WEIGHT kg/m (PLAIN END)
40	48.3				3.68			5.08					7.14	10.16	WALL THICK mm
					4.05			5.41					7.25	9.56	WEIGHT kg/m (PLAIN END)
50	60.3				3.91			5.54					8.74	11.07	WALL THICK mm
					5.44			7.48					11.11	13.44	WEIGHT kg/m (PLAIN END)
65	73.0				5.16			7.01					9.52	14.02	WALL THICK mm
					8.63			11.41					14.90	20.39	WEIGHT kg/m (PLAIN END)
80	88.9				5.49			7.62					11.13	15.24	WALL THICK mm
					11.29			15.27					21.35	27.68	WEIGHT kg/m (PLAIN END)
90	101.6				5.74			8.08							WALL THICK mm
					13.57			18.63							WEIGHT kg/m (PLAIN END)
100	114.3				6.02			8.56			11.13		13.49	17.12	WALL THICK mm
					16.07			22.32			28.32		33.54	41.03	WEIGHT kg/m (PLAIN END)
125	141.3				6.55			9.52			12.70		15.88	19.05	WALL THICK mm
					21.77			30.94			40.28		49.11	57.43	WEIGHT kg/m (PLAIN END)
150	168.3				7.11			10.97			14.27		18.26	21.95	WALL THICK mm
					28.26			42.56			54.20		67.56	79.22	WEIGHT kg/m (PLAIN END)
200	219.1		6.35	7.04	8.11		10.31	12.70		15.09	18.26	20.62	23.01	22.22	WALL THICK mm
			33.31	36.31	42.55		53.08	64.40		75.92	90.44	100.92	111.27	107.88	WEIGHT kg/m (PLAIN END)
250	273.1		6.35	7.80	9.27		XS	12.70	15.09	18.26	21.44	XXS	28.57	25.40	WALL THICK mm
			41.75	51.01	60.29		XS	81.52	95.97	114.70	133.00	XXS	172.21	155.09	WEIGHT kg/m (PLAIN END)
300	323.9		6.35	8.38	9.53	10.31	14.27	12.70	17.48	21.44	XXS	28.57	33.32	25.40	WALL THICK mm
			49.71	65.18	73.78	79.70	108.92	97.43	132.04	159.86	XXS	208.00	238.68	186.90	WEIGHT kg/m (PLAIN END)
350	355.6	6.35	7.92	STD	9.53	11.13	15.09	12.70	19.05	23.83	27.97	31.75	35.71		WALL THICK mm
		54.69	67.90	STD	81.25	94.55	126.71	107.39	158.10	194.96	224.65	253.56	281.70		WEIGHT kg/m (PLAIN END)
400	406.4	6.35	7.92	STD	9.53	XS	16.66	12.70	21.44	26.19	30.96	36.53	40.49		WALL THICK mm
		62.64	77.83	STD	93.17	XS	160.12	123.30	203.53	245.56	286.64	333.19	365.35		WEIGHT kg/m (PLAIN END)
450	457.2	6.35	7.92	11.13	9.53	14.27	19.05	12.70	23.83	29.36	34.92	39.67	45.24		WALL THICK mm
		70.60	87.75	122.43	105.10	155.87	205.83	139.20	254.67	309.76	363.64	408.45	459.59		WEIGHT kg/m (PLAIN END)
500	508.0	6.35	STD	XS	9.53	15.09	20.62	12.70	26.19	32.54	38.10	44.45	50.01		WALL THICK mm
		78.55	STD	XS	117.02	183.42	247.83	155.12	311.17	381.53	441.49	508.11	564.81		WEIGHT kg/m (PLAIN END)
550	558.8	6.35	STD	XS	9.53		22.23	12.70	28.58	34.93	41.28	47.63	53.98		WALL THICK mm
		86.54	STD	XS	129.13		294.25	171.09	373.83	451.42	527.02	600.63	672.76		WEIGHT kg/m (PLAIN END)
600	609.6	6.35	STD	14.27	9.53	17.48	24.61	12.70	30.96	38.89	46.02	52.37	59.54		WALL THICK mm
		94.46	STD	209.50	140.88	255.24	355.02	186.94	441.78	547.33	639.58	719.63	807.63		WEIGHT kg/m (PLAIN END)
650	660.4	7.92	XS		9.53			12.70							WALL THICK mm
		127.43	XS		152.80			202.85							WEIGHT kg/m (PLAIN END)
700	711.2	7.92	XS	15.88	9.53			12.70							WALL THICK mm
		137.32	XS	271.21	164.85			218.69							WEIGHT kg/m (PLAIN END)
750	762.0	7.92	XS	15.88	9.53			12.70							WALL THICK mm
		147.28	XS	292.18	176.84			234.67							WEIGHT kg/m (PLAIN END)
800	812.8	7.92	XS	15.88	9.53	17.48		12.70							WALL THICK mm
		157.24	XS	312.15	188.82	342.91		250.64							WEIGHT kg/m (PLAIN END)
850	863.6	7.92	XS	15.88	9.53	17.48		12.70							WALL THICK mm
		167.20	XS	332.12	200.31	364.90		266.61							WEIGHT kg/m (PLAIN END)
900	914.4	7.92	XS	15.88	9.53	19.05		12.70							WALL THICK mm
		176.96	XS	351.70	212.56	420.42		282.27							WEIGHT kg/m (PLAIN END)
950	965.2				9.53			12.70							WALL THICK mm
					224.54			298.24							WEIGHT kg/m (PLAIN END)
1000	1016.0				9.53			12.70							WALL THICK mm
					236.53			314.22							WEIGHT kg/m (PLAIN END)
1050	1066.8				9.53			12.70							WALL THICK mm
					248.52			330.19							WEIGHT kg/m (PLAIN END)
1100	1117.6				9.53			12.70							WALL THICK mm
					260.50			346.16							WEIGHT kg/m (PLAIN END)
1150	1168.4				9.53			12.70							WALL THICK mm
					272.25			351.82							WEIGHT kg/m (PLAIN END)
1200	1219.2				9.53			12.70							WALL THICK mm
					284.24			377.79							WEIGHT kg/m (PLAIN END)

SEAMLESS LINEPIPE



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Home Linkage Products Spherical Rod Ends Metric Rod Ends Ball Joints HVAC Damper Control Swivels
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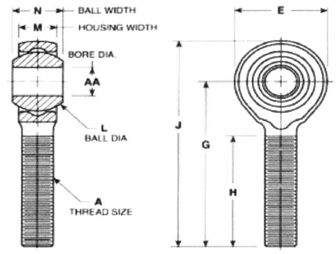
Heavy Duty Double Ended Rim Studs and Nuts Fasteners for Brake Drum Assemblies
Locking Hex Head Six Slot Place Bolts Grade 8 GM Type AA

SPHERICAL BEARING ROD ENDS

MALE SERIES SPM, EM, EM-T, INDUSTRIAL / COMMERCIAL, 2 PIECE



- THE MALE SERIES SPM, EM, EM-T SPHERICAL ROD ENDS are constructed to SAE J1120 specifications..
- THE MALE SERIES SPM, EM, EM-T SPHERICAL ROD ENDS spherical ball is precision made from low carbon steel, surface hardened for wear resistance and plated.
- THE MALE SERIES SPM, EM, EM-T SPHERICAL ROD ENDS body is made of low carbon steel, zinc and dichromate treated.
- Left hand threads are available in all sizes. and are marked for easy user identification. (Add "L" to part number. i.e. MLJ10M).

MALE SERIES SPM, SELF LUBRICATING, NYLON RACE, SPHERICAL ROD ENDS

THE MALE SPM SPHERICAL ROD END features high strength injection molded, self lubricating nylon raceways that are locked in place by a annular groove trunion design. The raceways will withstand temperatures from -30° to 220° F under normal load conditions and will not conduct electricity. They feature high wear resistance and a low coefficient of friction.

AETNA PART NO.	A Thread Size	AA	E	G	H	J	L	M	N	Ultimate Radial Load Capacity (Pounds)
SPM3	10-32	.189/.191	.625	1.250	.750	1.562	.438	.250	.312	1210
SPM4	1/4-28	.250/.252	.750	1.562	1.000	1.937	.516	.281	.375	2510
SPM5	5/16-24	.3125/.3145	.875	1.875	1.250	2.312	.625	.344	.437	3430
SPM6	3/8-24	.375/.377	1.000	1.937	1.250	2.437	.719	.406	.500	5520
SPM7	7/16-20	.4375/.4395	1.1256	2.125	1.312	2.687	.812	.437	.562	5350
SPM8	1/2-20	.500/.502	1.312	2.437	1.500	3.094	.938	.500	.625	8690
SPM10	5/8-18	.625/.627	1.500	2.625	1.625	3.375	1.125	.562	.750	10300
SPM12	3/4-16	.750/.752	1.750	2.875	1.750	3.750	1.312	.687	.875	10900

MALE SERIES EM, METAL TO METAL RACE, SPHERICAL ROD ENDS MALE

THE MALE SERIES EM SPHERICAL ROD ENDS have similar features of the SPF but with a metal raceway construction.

AETNA PART NO.	A Thread Size	AA	E	G	H	J	L	M	N	Ultimate Radial Load Capacity (Pounds)
EM4	1/4-28	.250/.252	.750	1.562	1.000	1.937	.516	.281	.375	2510
EM5	5/16-24	.3125/.3145	.875	1.875	1.250	2.312	.625	.344	.437	3430
EM6	3/8-24	.375/.377	1.000	1.937	1.250	2.437	.719	.406	.500	5520
EM8	1/2-20	.500/.502	1.312	2.437	1.500	3.094	.938	.500	.625	8690
EM10	5/8-18	.625/.627	1.500	2.625	1.625	3.375	1.125	.562	.750	10300

SERIES EM-T, ALL METAL, TEFLON* LINED RACE, SPHERICAL ROD ENDS

GAZ COILOVER SHOCK ABSORBERS

www.gazshocks.com

All open and closed dimensions are metal to metal and units are fitted with a 1" bumpstop. This therefore increases the closed length and reduces the stroke length by 1". All units are fully dyno tested before despatch.

VEHICLE MODEL	OPEN LENGTH	CLOSED LENGTH	STROKE	PRICE
1 3/4" DIAMETER BODIES WITH 12mm RODS AND BONDED BUSHES				
090/070B12	9"	7"	2"	£60.00
100/075B12	10"	71/2"	21/2"	£60.00
110/080B12	11"	8"	3"	£60.00
120/085B12	12"	81/2"	31/2"	£60.00
130/090B12	13"	9"	4"	£60.00
140/095B12	14"	91/2"	41/2"	£60.00
150/100B12	15"	10"	5"	£60.00
160/105B12	16"	101/2"	51/2"	£60.00
170/100B12	17"	11"	6"	£60.00
180/115B12	18"	111/2"	61/2"	£60.00
190/120B12	19"	12"	7"	£60.00
200/125B12	20"	121/2"	71/2"	£60.00
1 3/4" DIAMETER BODIES WITH 12mm RODS AND SPHERICAL BEARINGS				
090/070R12	9"	7"	2"	£72.50
100/075R12	10"	71/2"	21/2"	£72.50
110/080R12	11"	8"	3"	£72.50
120/085R12	12"	81/2"	31/2"	£72.50
130/090R12	13"	9"	4"	£72.50
140/095R12	14"	91/2"	41/2"	£72.50
150/100R12	15"	10"	5"	£72.50
160/105R12	16"	101/2"	51/2"	£72.50
170/110R12	17"	11"	6"	£72.50
180/115R12	18"	111/2"	61/2"	£72.50
190/120R12	19"	12"	7"	£72.50
200/125R12	20"	121/2"	71/2"	£72.50
2" DIAMETER BODIES WITH 12mm RODS AND BONDED BUSHES				
090/070B12-2	9"	7"	2"	£70.00
100/075B12-2	10"	71/2"	21/2"	£70.00
110/080B12-2	11"	8"	3"	£70.00
120/085B12-2	12"	81/2"	31/2"	£70.00
130/090B12-2	13"	9"	4"	£70.00
140/095B12-2	14"	91/2"	41/2"	£70.00
150/100B12-2	15"	10"	5"	£70.00
160/105B12-2	16"	101/2"	51/2"	£70.00
170/110B12-2	17"	11"	6"	£70.00
180/115B12-2	18"	111/2"	61/2"	£70.00
190/120B12-2	19"	12"	7"	£70.00
200/125B12-2	20"	121/2"	71/2"	£70.00
2" DIAMETER BODIES WITH 12mm RODS AND SPHERICAL BEARINGS				
090/070R12-2	9"	7"	2"	£87.50
100/075R12-2	10"	71/2"	21/2"	£87.50
110/080R12-2	11"	8"	3"	£87.50
120/085R12-2	12"	81/2"	31/2"	£87.50
130/090R12-2	13"	9"	4"	£87.50
140/095R12-2	14"	91/2"	41/2"	£87.50
150/100R12-2	15"	10"	5"	£87.50
160/105R12-2	16"	101/2"	51/2"	£87.50
170/110R12-2	17"	11"	6"	£87.50
180/115R12-2	18"	111/2"	61/2"	£87.50
190/120R12-2	19"	12"	7"	£87.50
200/125R12-2	20"	121/2"	71/2"	£87.50

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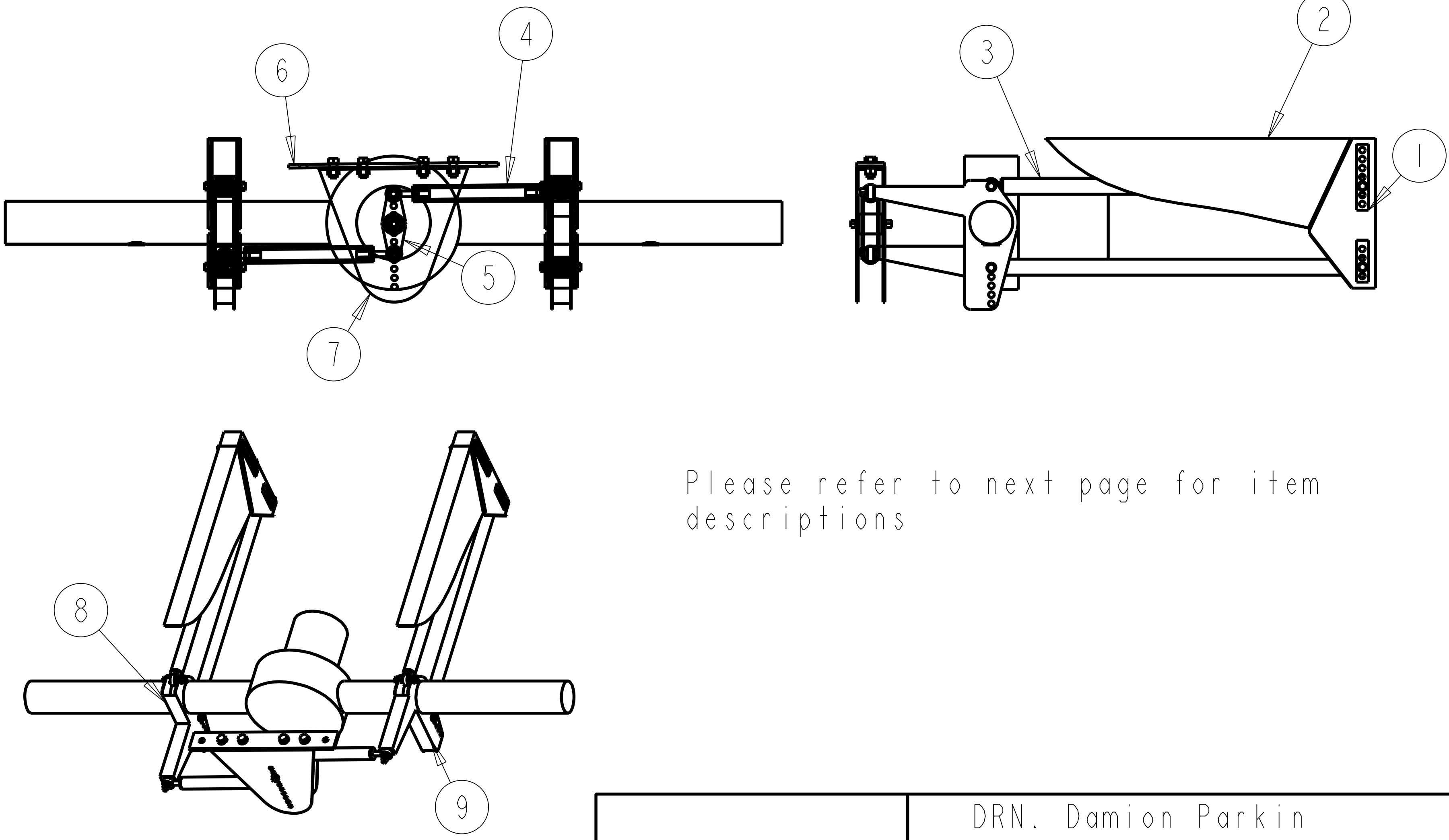
Appendix D

Detail drawings of rear suspension components

The following pages include detail drawings of all of the components that need to be fabricated to construct the four link and the watt's link designs.



SCALE 0.003

Please refer to next page for item descriptions

	DRN. Damion Parkin
	Assembly drawing and indication of major components
Sheet 1 of 11	21/10/07 A4

ITEM NO.	QTY	DESCRIPTION
1	2	Front link box
2	2	Front link box cover
3	4	Four link bar
4	2	watt's link bar
5	1	bell crank
6	1	watt's top plate
7	1	watt's housing
8	8	Left bird cage
9	9	Right birdcage

DRN. Damion Parkin

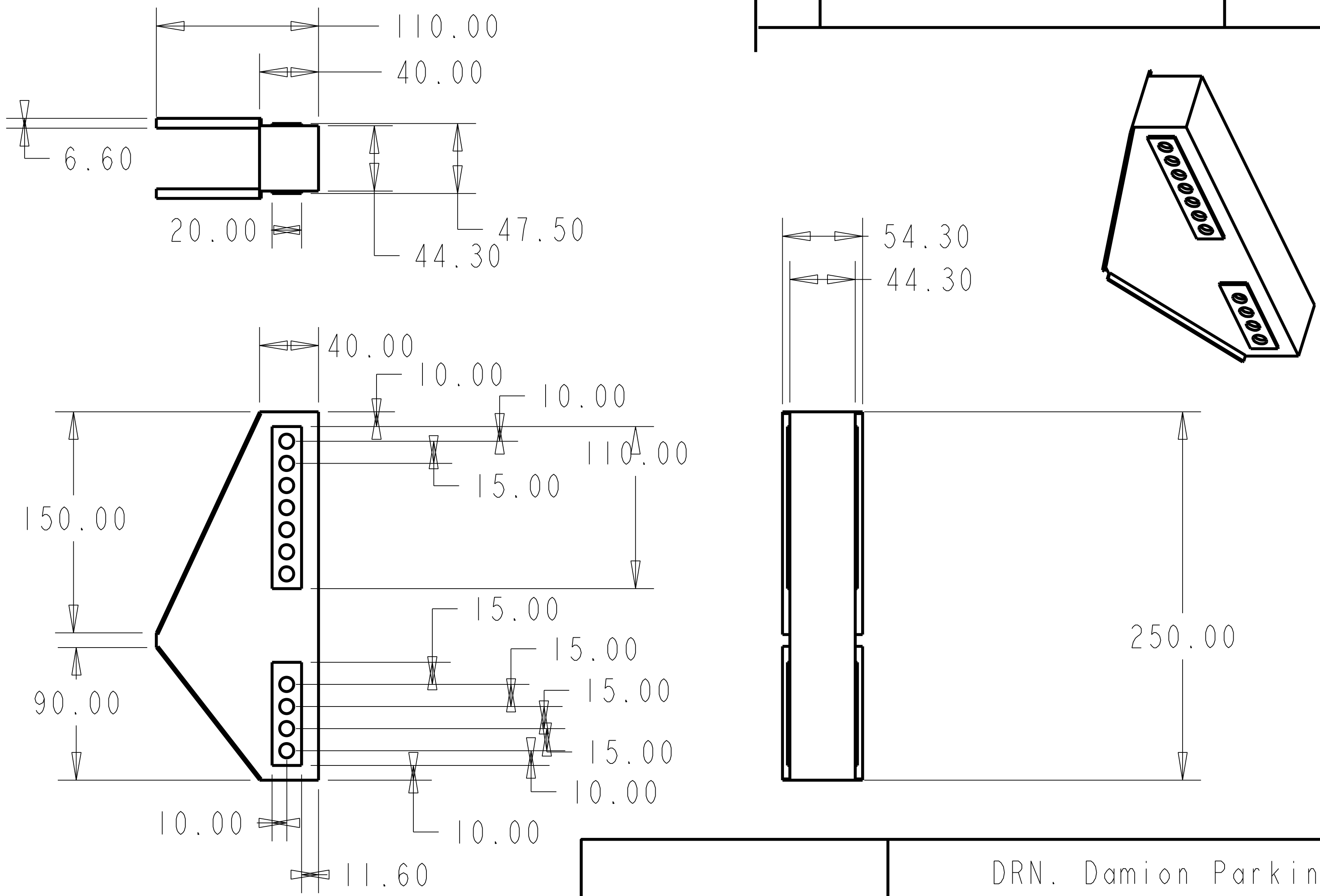
Parts list for assembly drawing on previous page

Sheet 2 of 11

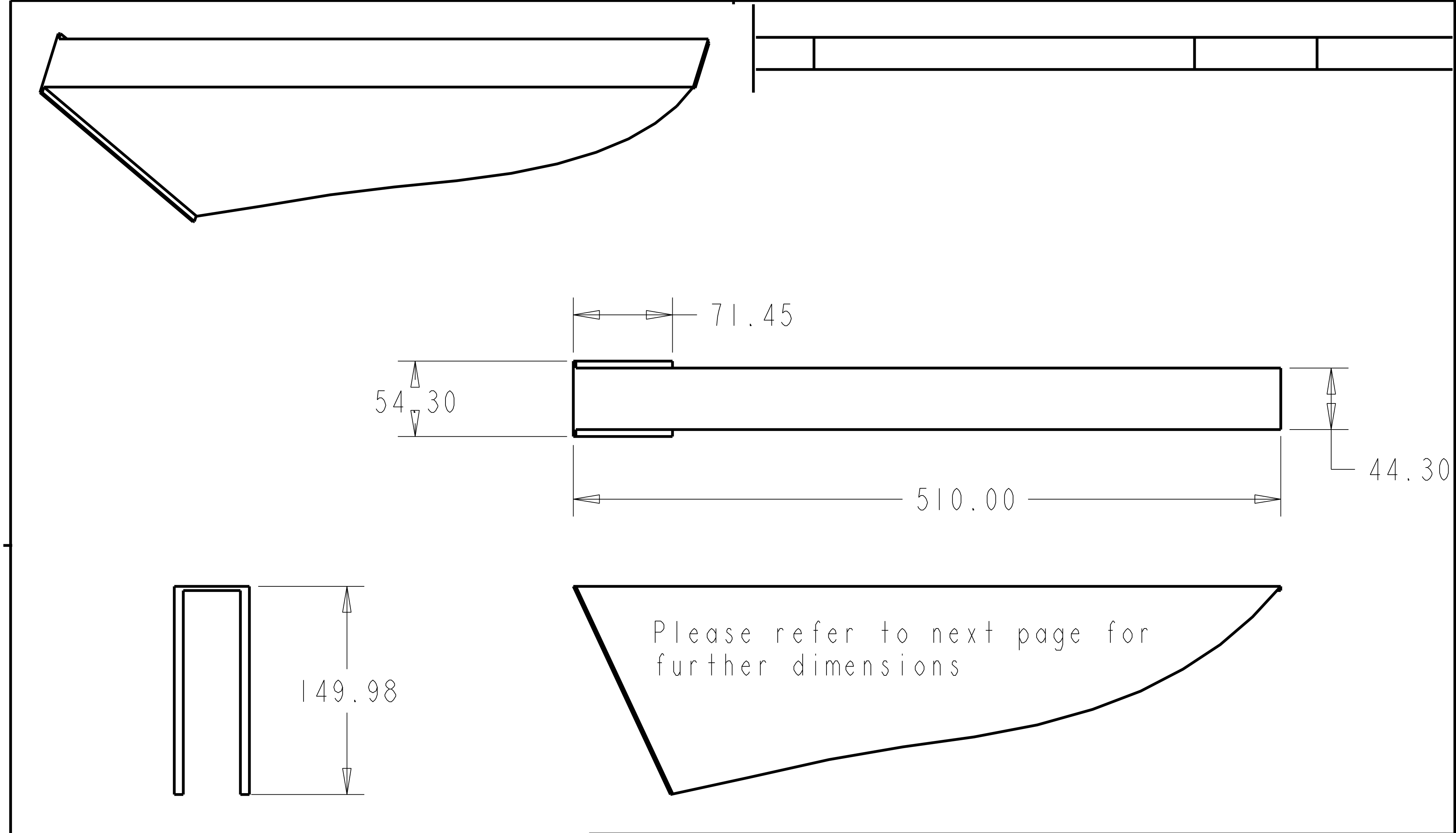
21/10/07

NOT TO SCALE

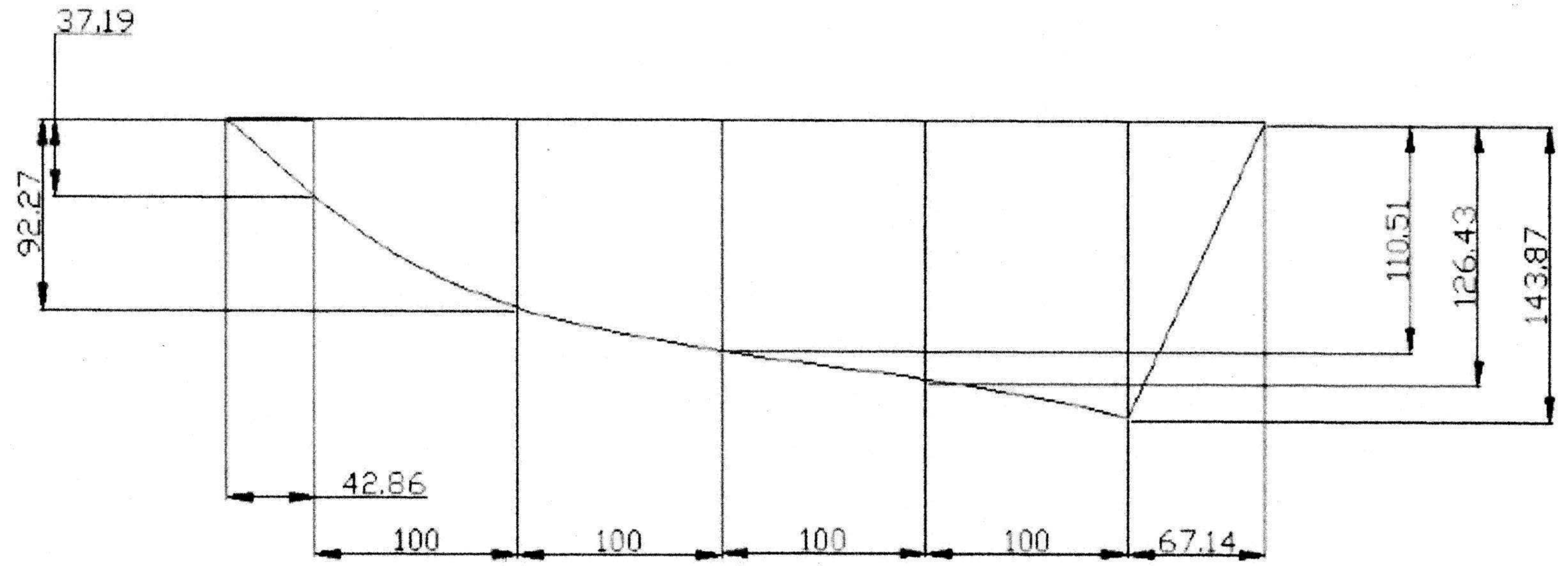
A4



1.6mm sheet steel	DRN. Damion Parkin
	Front link box
Sheet 3 of 11	21/10/07 Scale: 0.017 A4

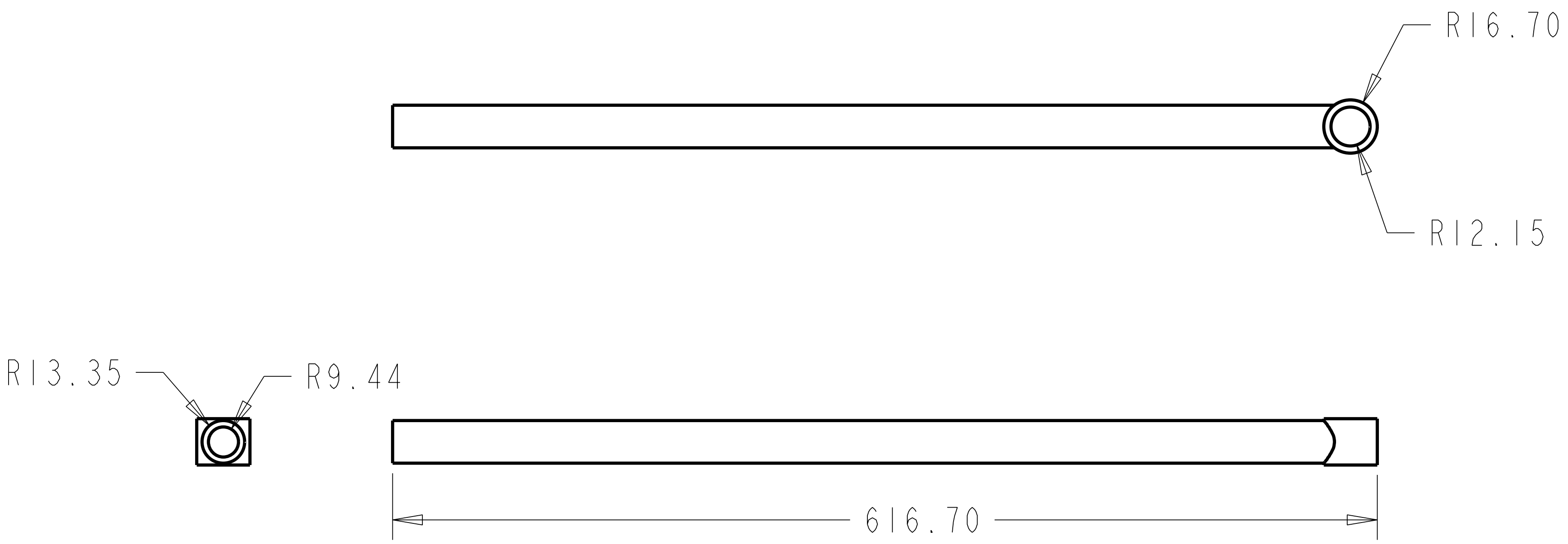


1.4mm sheet steel	DRN. Damion Parkin
	Link box cover
Sheet 4 of 11	21/1/07
	A4

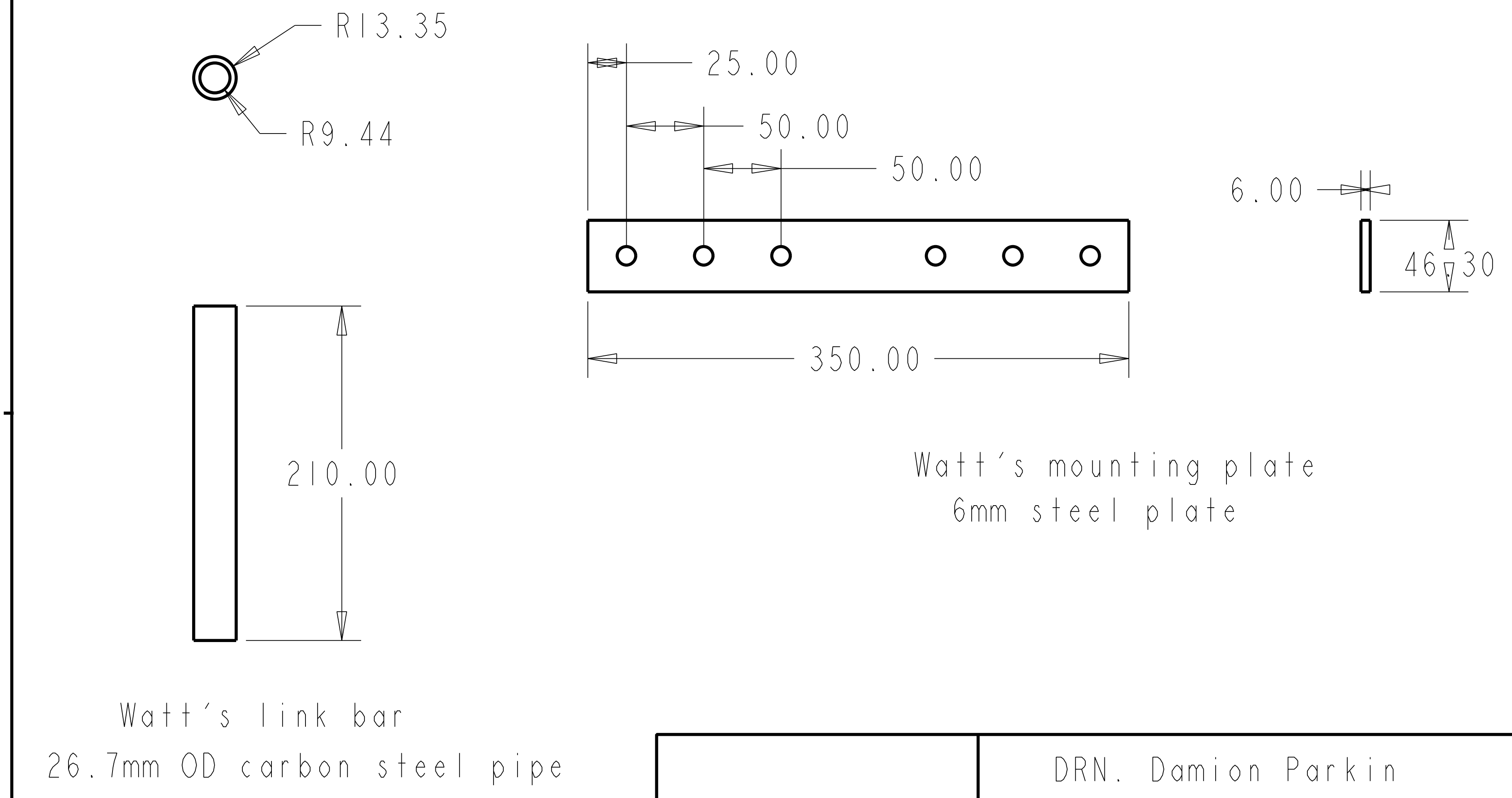


SIDE VIEW

1.4mm Sheet steel	DRN. Damion Parkin
	Link Cover dimensions
Sheet 5 of 11	21/10/07 | NOT TO SCALE | A4



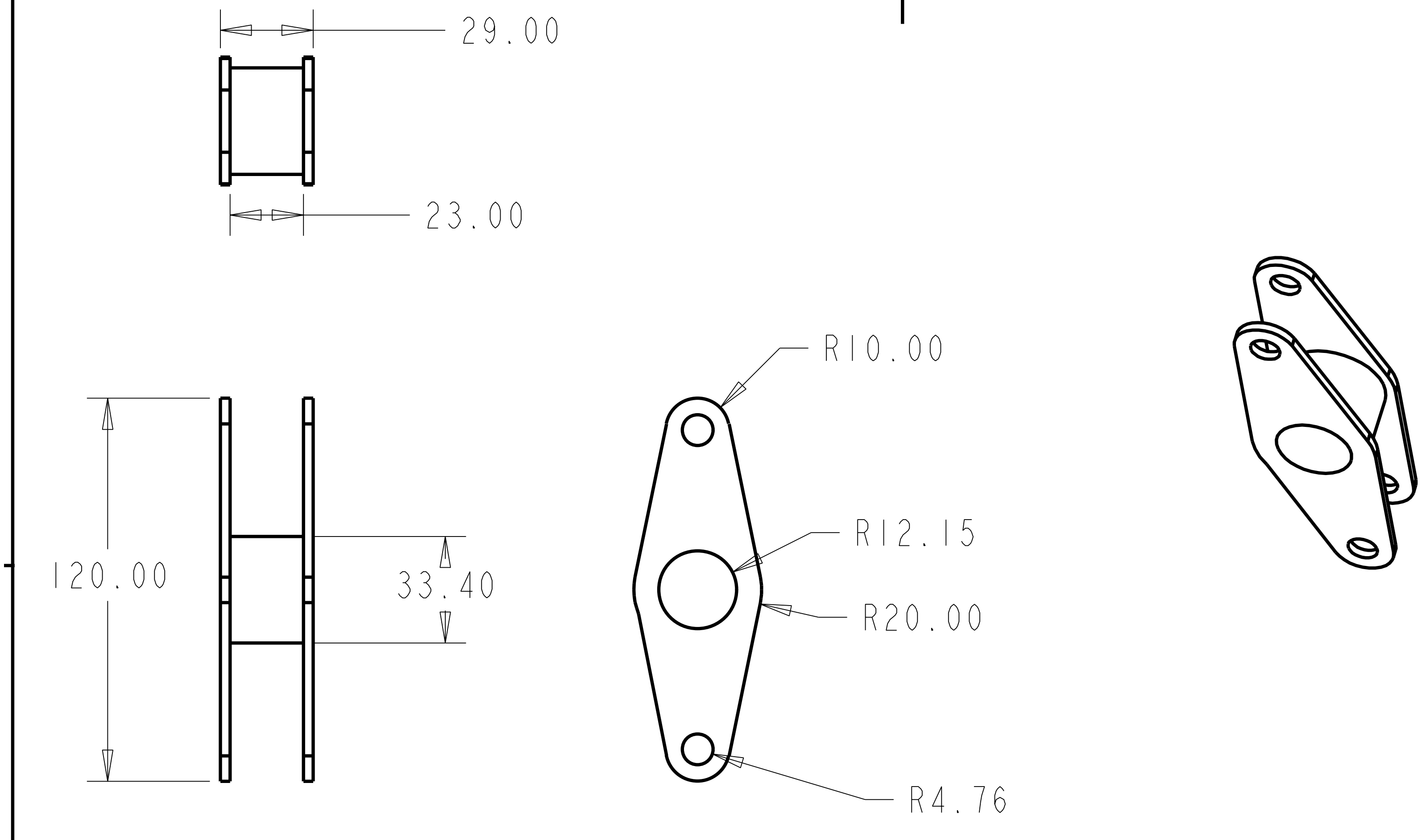
26.7mm OD Carbon Steel pipe	DRN. Damion Parkin
	Four link bar
Sheet 6 of 11	21/10/07 / SCALE 0.010 / A4



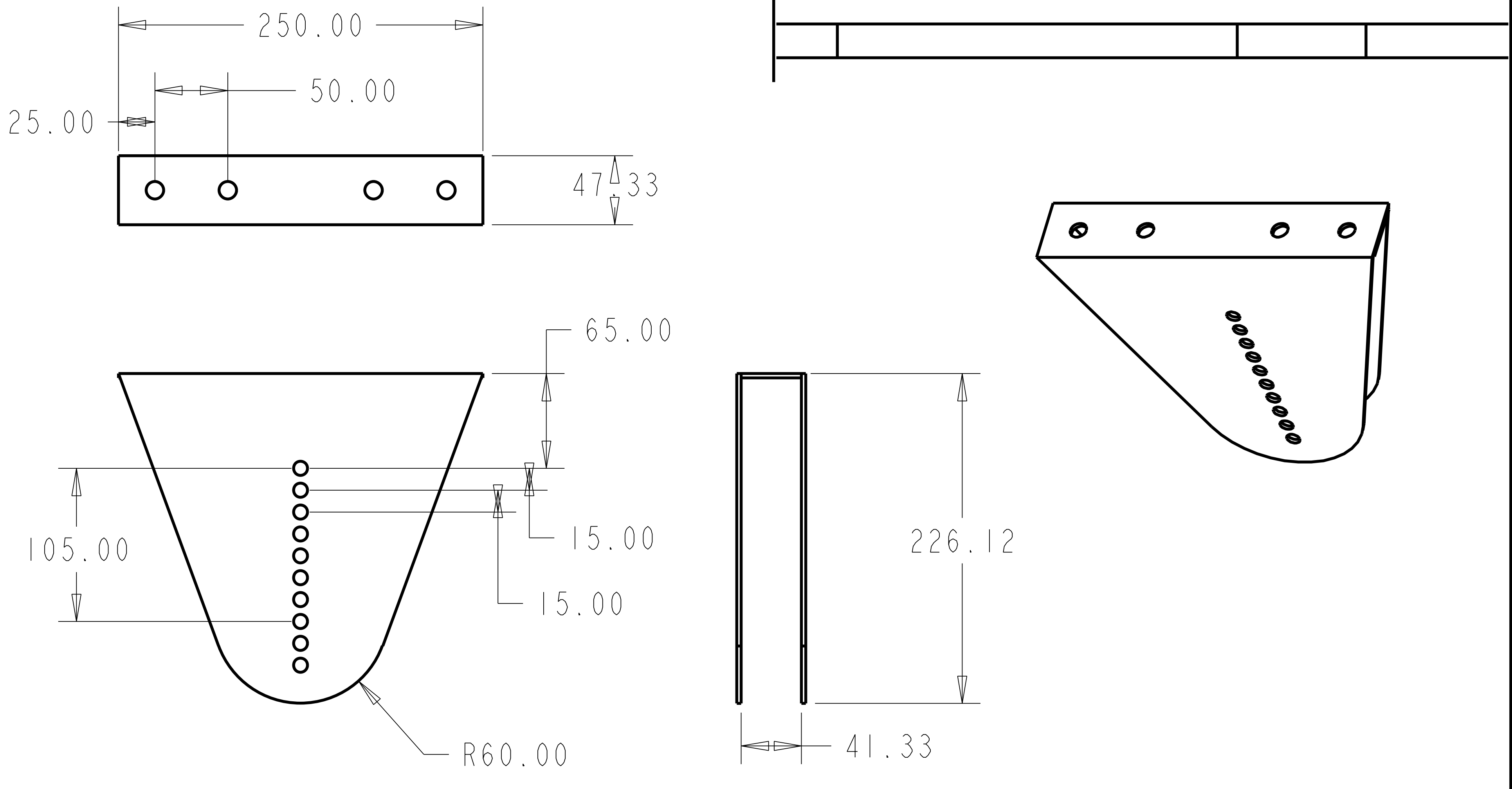
DRN. Damion Parkin

Watt's link bar &
Watt's housing mounting plate

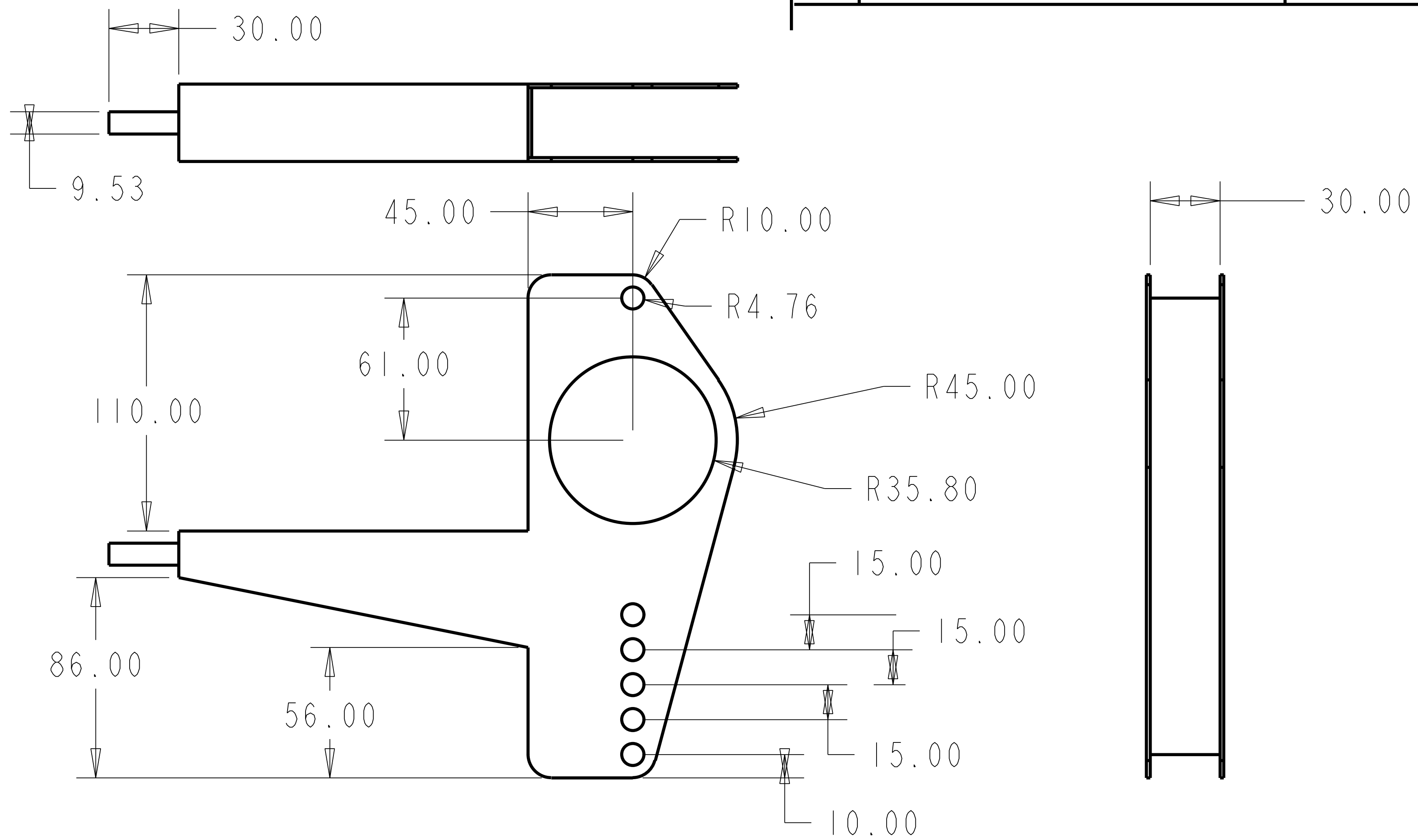
Sheet 7 of 11	21/10/07	Scale: 0.017	A4



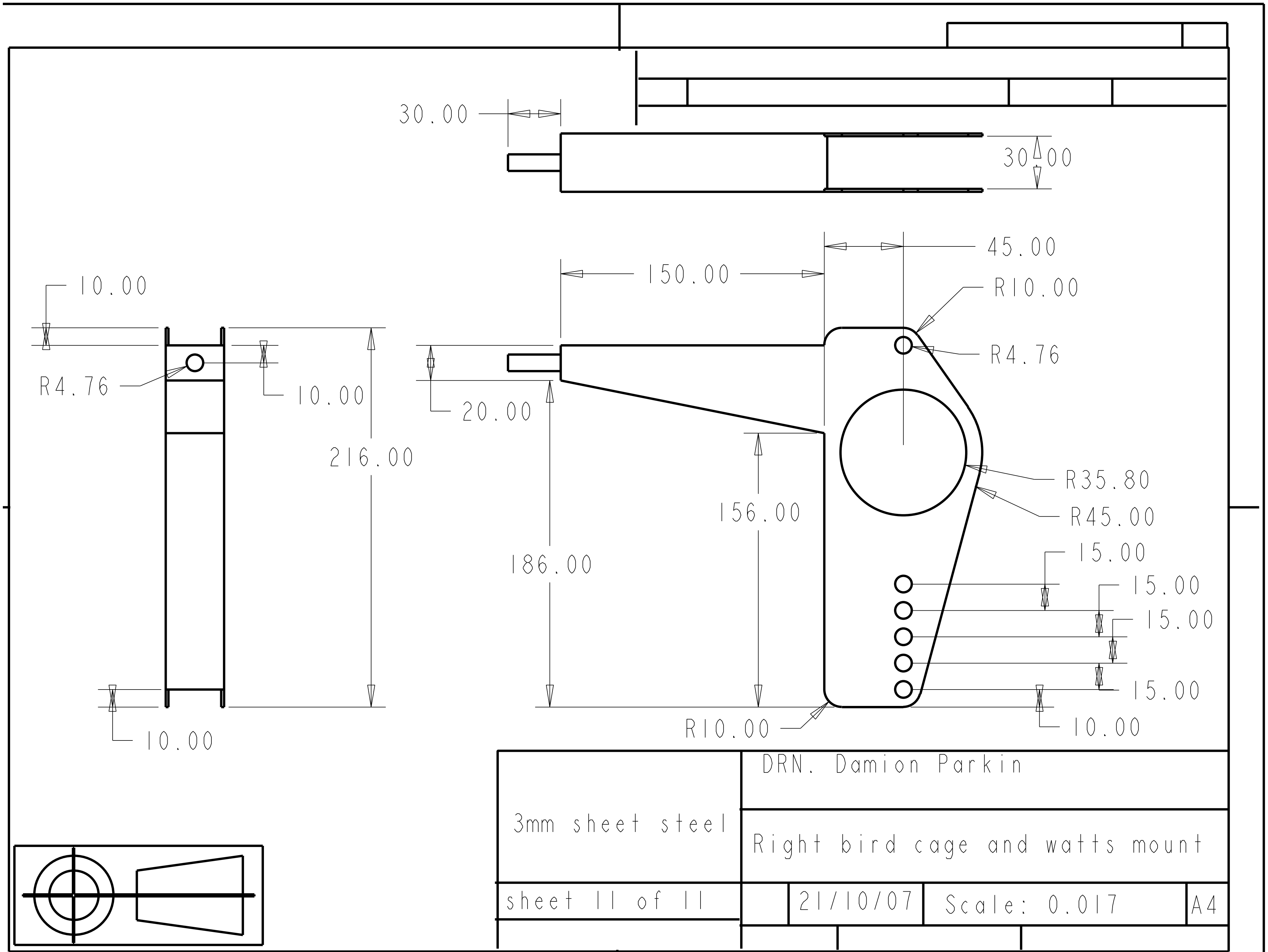
3mm Sheet steel & 33.4mm OD Carbon steet Pipe	DRN. Damion Parkin		
	Watt's bell crank		
Sheet 8 of 11	21/10/07	Scale: 0.017	A4



3mm sheet steel	DRN. Damion Parkin		
	Watt's housing		
Sheet 9 of 11	21/10/07	Scale: 0.017	A4



DRN. Damion Parkin			
left bird cage and watts mount			
Sheet 10 of 11	21/10/07	Scale: 0.017	A4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186.00
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15.00
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10.00
R10.00
10.00
DRN. Damion Parkin
3mm sheet steel
Right bird cage and watts mount
sheet 11 of 11
21/10/07
Scale: 0.017
A4