6

Independent Suspension

A Kinematic Overview

(It's just not this simple!)

Before beginning the kinematic chapters I would highly recommend that anyone who is not thoroughly familiar with all aspects of racecar vehicle dynamics take time to build a series of models such as the spindle/upright model shown in Figures (6-1 & 6-2). They don't have to be pieces of artwork as long as they are closely representative of what you are studying and can have their various co-ordinates quickly changed. An accurately working model is worth its weight in gold. The hardest thing about building both representational and to-scale models is not creating unwanted moments - (Book Two).

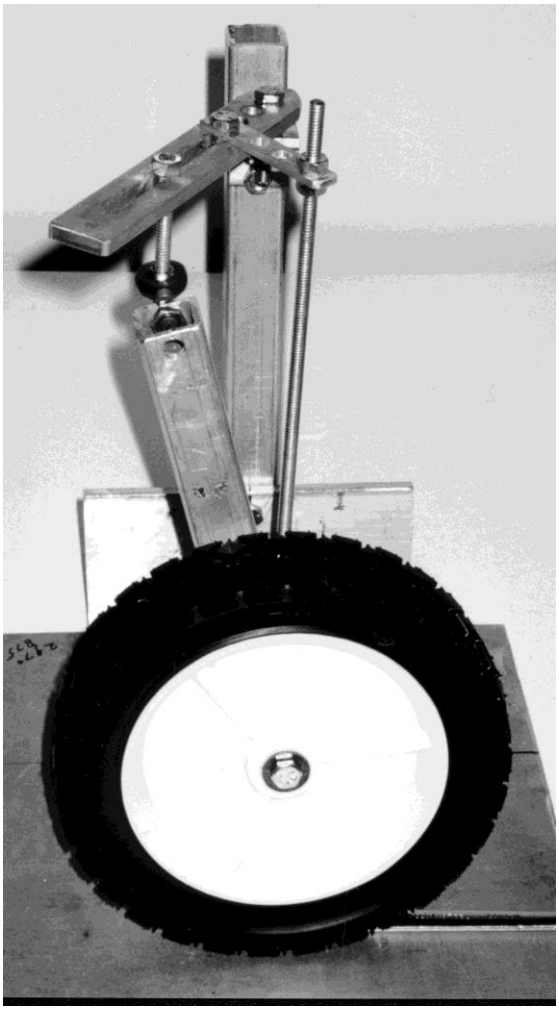


Figure 6-1 – Jig used to understand front suspension

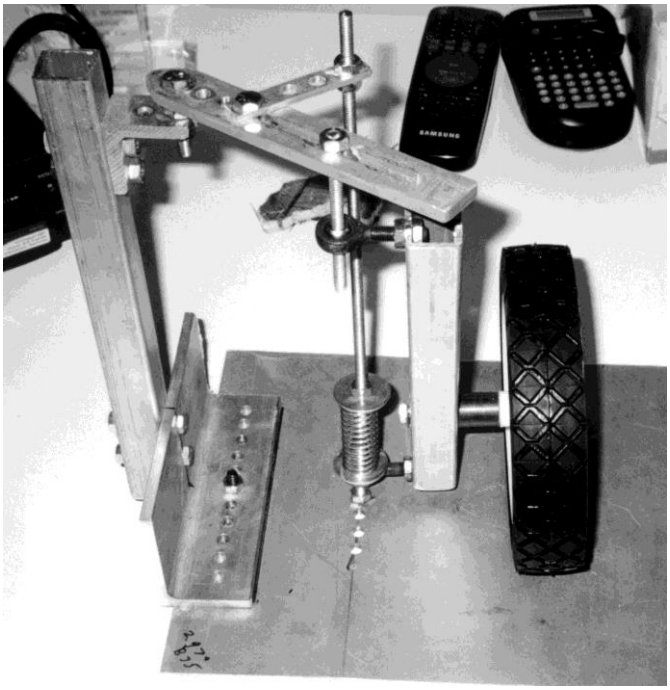


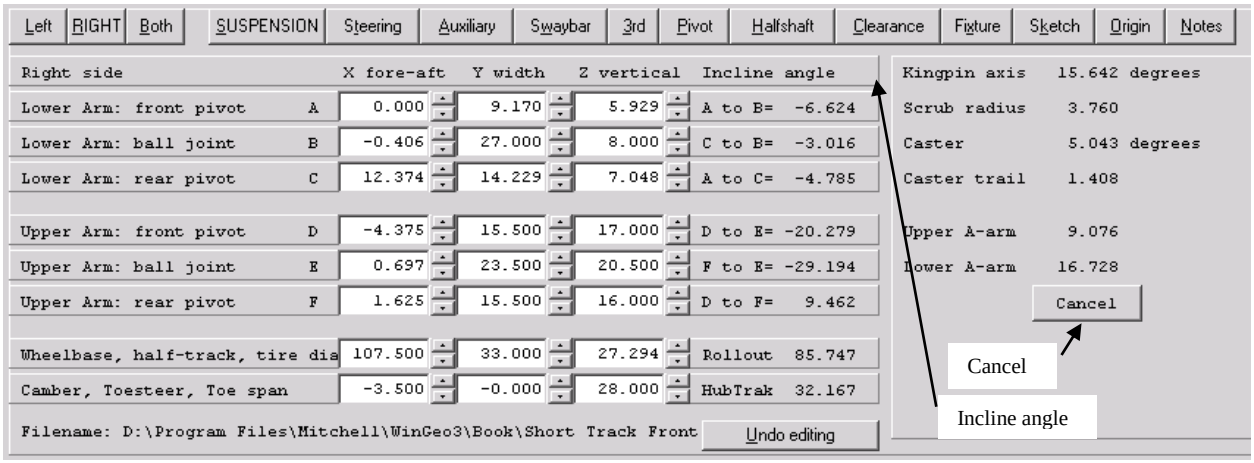
Figure 6-2

Important Note: Many of the examples in this chapter, for the sake of clarity, are shown in single plane format which may not always represent the realities of three dimensional co-ordinate analysis. Points that intersect on a single plane do not **necessarily** intersect in three dimensional space. That is why the use of a three dimensional kinematics program such as WinGeo3 is so important.

A Quick WinGeo3 Review

- The **Edit menu** is where you enter measured data from an existing chassis as shown in Win 6-1. Until changes made in any of the WinGeo3 menus have been saved to a file, clicking on **cancel** will return the co-ordinates to their **original settings.**

- The user in this menu has the option of seeing either the incline angles or the distances between the various control arm co-ordinates. Win 6-1 shows the angle between the various points. Go to *Utility / Edit option* and click on *Check sums* to toggle between distance and *Incline angle*.



Win 6-1

- The **Design menu** is where the user can look at various design options i.e. create a suspension which meets specific design objectives and/or a component with specific characteristics (co-ordinates). The menu can also be used to design a new suspension. It can not be used to modify an existing suspension. The menu does **not allow** the user to start with a **predetermined part** such as a **spindle**.

- The **Build Menu** allows the user to create a suspension from existing parts.

- The **Compute Menu** allows the user to specify the problem to be solved.

Even if the user is involved in only one type of racing, he/she should use, at a minimum, both the short track and open wheel WinGeo3 examples when exploring each area of chassis kinematics. These chassis examples represent the two ends of the design spectrum. **Many** of the kinematic trends are not always immediately apparent when looking at one chassis type alone.

The three link short track suspension system, as emphasized in the live axle dynamics section of the book, is featured rather than the truck arm suspension because the live axle system is also commonly found on many GT road course chassis types, including the Trans Am series. If the user is involved in live axle truck arm suspension, they should **also** spend time looking at the WinGeo3 truck arm examples.

Remember that this first book is only able to scratch the kinematic surface. Obviously, the serious race engineer will take the opportunity to work past the surface on his/her own.

Note: Crews involved in different types of racing often use different terms to describe the same basic item. For example, open wheel and sports car crews will refer to an upright while stock cars and GT crews refer to the same type of thing as the spindle. It is my hope that this book will be user friendly for all types of racers so what to call each part has been rather problematic. I first tried combining names such as upright/spindle, toe-link/tie-rod and wishbone/A-Arm but nobody seemed to like that. I next tried using specific engineering terms for specific functions such as point of axial rotation to replace ball joint/rod end/ spherical bearing/ mono-ball and that didn't go over well either. In the end, if I am speaking in generic terms, I have decided to use just one or two names for each component type, provide a list of alternative names, and hope no one racing type gets too bent out of shape with my choices. If I am describing a specific type of car, I will use the most common name for that car type.

1. Spindle: Upright
2. Tie Rod: Toe link
3. Spherical bearing: Mono-ball
4. A-Arm: Wishbone - Control arm
5. Ball Joint: Rod end - Spherical bearing
6. Compression: Bump
7. Extension: Rebound
8. Understeer: Push
9. Oversteer: Loose

Track

- The front view distance from the center of the right side hub mounting surface to the center of the left side hub mounting surface is the most common method of measuring track - Figure (6-3 & 6-4). A trammel gauge is often used for this type of measurement.

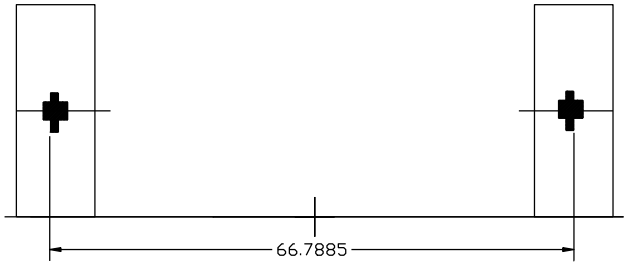


Figure 6-3

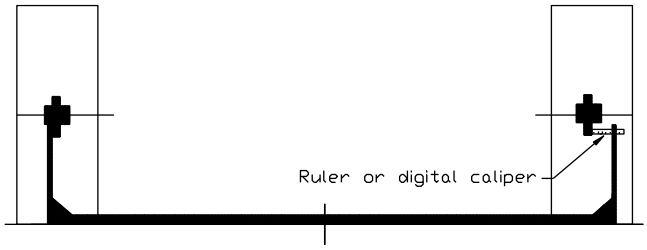


Figure 6-4

- Many race sanctioning bodies measure track from outer sidewall to outer sidewall at hub center height and at a specified psi. If the track is measured at the road surface, rim back spacing (offset), kingpin angle and camber angle need to be taken into consideration. This latter type of track measurement is used when working out the distance between contact patches.

Begin WinGeo3 Example*

Utility: ***Track Option***

Track can be entered as *Full-track* or *Half-track*. *Full-track* is normally used for symmetric cars while *Half-track* is used for asymmetric cars. The Half-track measurement is from the centerline of the car (however that is determined) to the center of the tire contact patch.

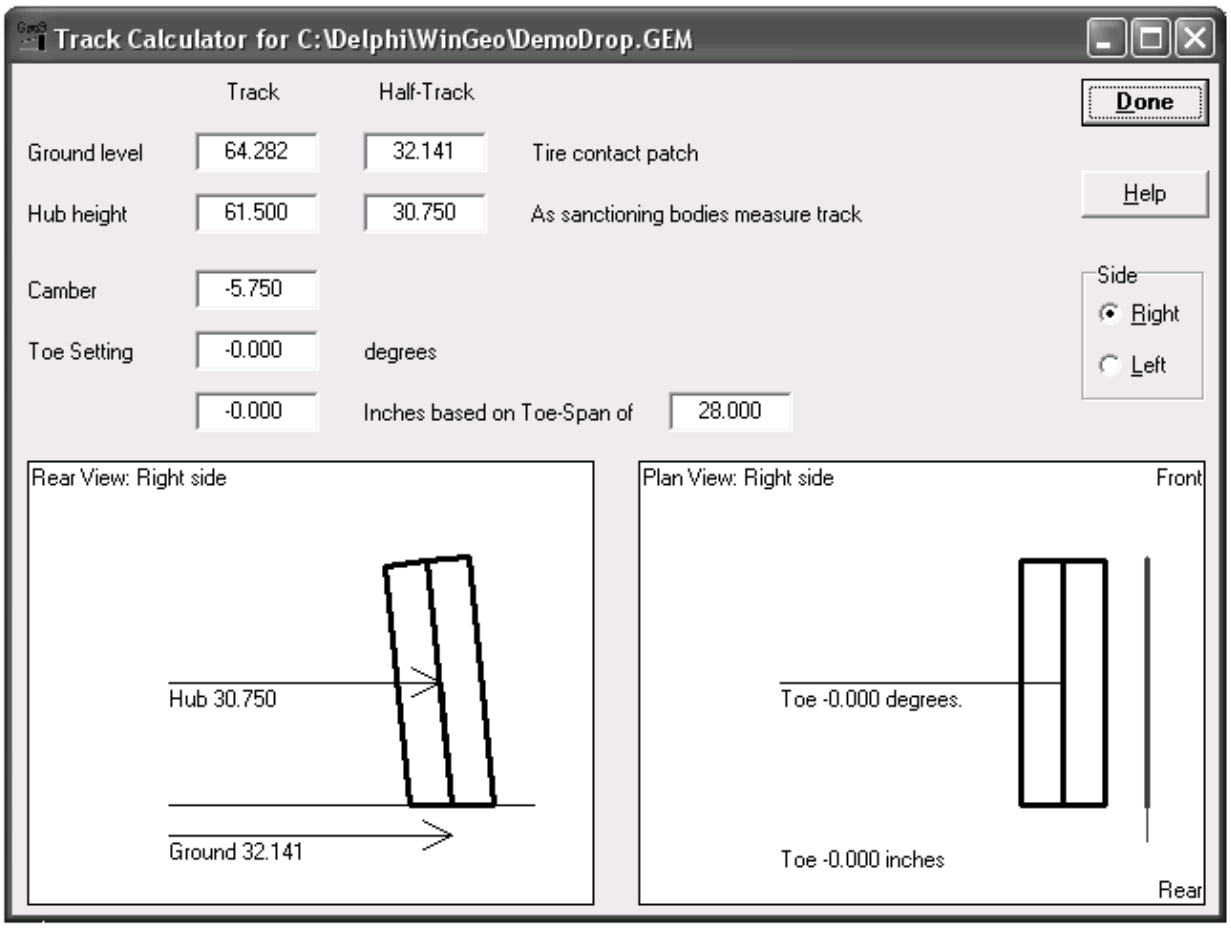
Edit: ***Track Calculator***

Track is entered under *Edit / Suspension*.

The Advanced WinGeo3 program measures track at ground level at the center of the tire contact patch. Most sanctioning bodies measure track at hub height by measuring from the outside of one tire to the outside of the opposite tire. These measurements can be quite different when either or both of the tires have static camber. The Track Calculator allows the user to toggle between the contact patch track and the track at the hub.

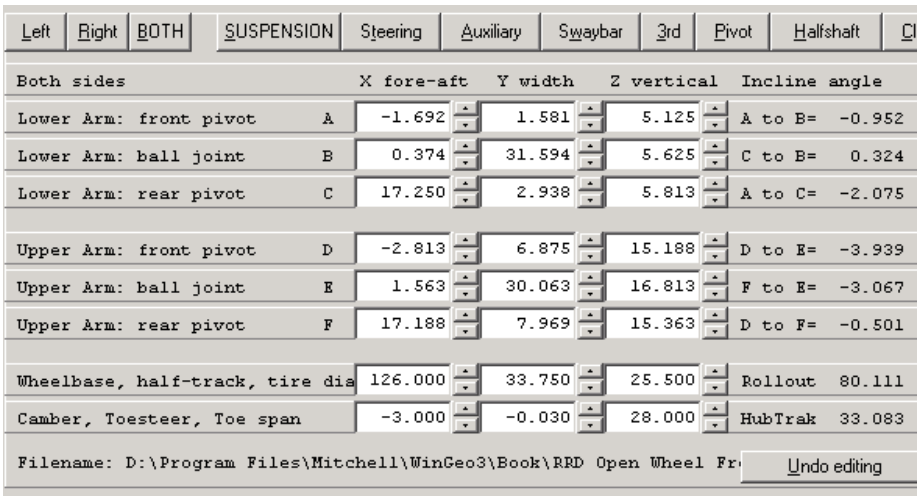
In the example from DemoDrop.gem file, the right side camber is -5.750 degrees. The half-track at ground level is 32.141 inches. At hub height the value is 30.750.

The Track Calculator, as shown in Win 6-2, allows you to specify a hub-height value and a static camber value and see the corresponding ground level value. It will do both right and left sides. However, when using the Advanced WinGeo3 version, the results of these calculations are not automatically applied to the suspension. You must manually enter the desired track value.

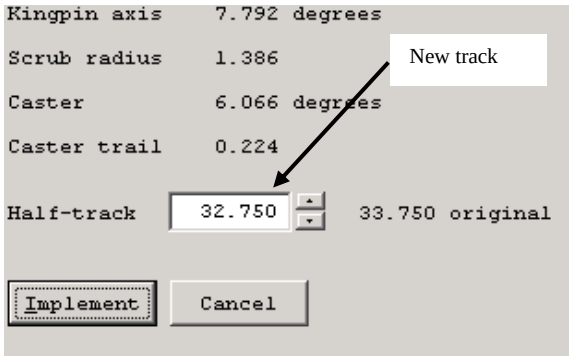


Win 6-2

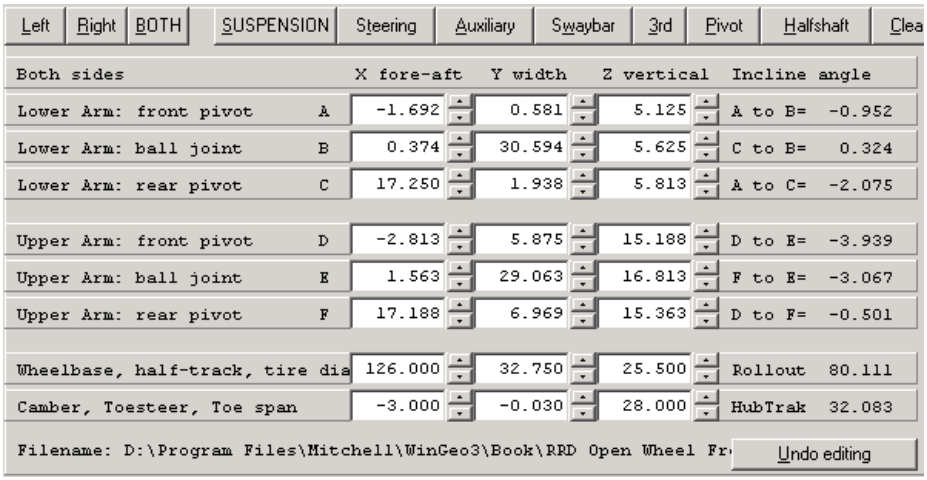
Build: Track Width - Changing the track width moves the ball joints and changes the track. The track is changed by altering the A-arms and preserving the hub/upright and the kingpin parameters. Only the Y-axis (lateral) value is changed. Win 6-3 shows the upper and lower A-arm co-ordinates before the track is changed. Win 6-4 shows both the new track and original track. Once the track has been altered in the track box the *Implement* box must be clicked in order to implement the change. Win 6-5 shows the resulting change in the A-arm **y** co-ordinates. When using *Build Track width* both the upper and lower control arms' y components are changed by the same amount.



Win 6-3



Win 6-4



Win 6-5

*****End WinGeo3 Example*****

Wheelbase - Wheelbase is measured in the longitudinal or side view plane from the center of the front wheel to the center of the rear wheel - Figure (6-5). Using WinGeo3 the wheelbase is entered under *Edit / Suspension*.

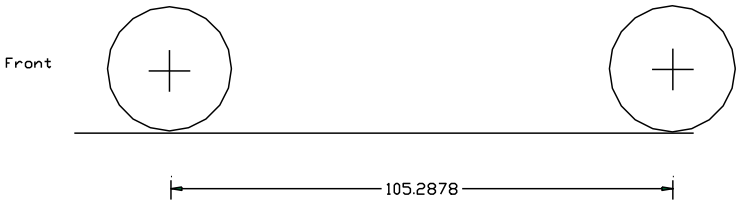


Figure 6-5

Backspacing - Backspacing is the distance from the back face of the center section (the hub mounting surface) to the inboard surface of the rim (the rim surface nearest the chassis centerline) minus the bead width - Figure (6-6).

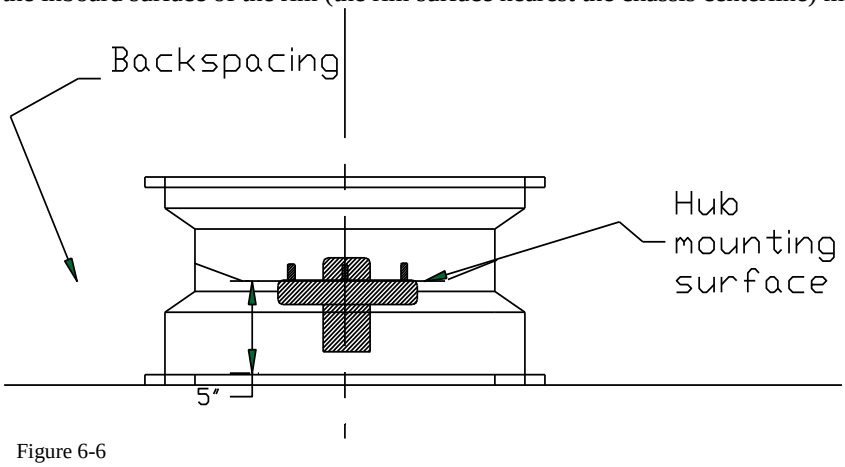


Figure 6-6

Offset - Offset is the distance the hub mounting surface is from the center of the rim - Figure (6-7). If the back face of an 8" rim is in the center of the rim, you have 0 offset and 4" backspacing. If on an 8" rim the backspacing is 3", the offset is 1" out/positive. Rim width is always measured from inside of bead to inside of bead.

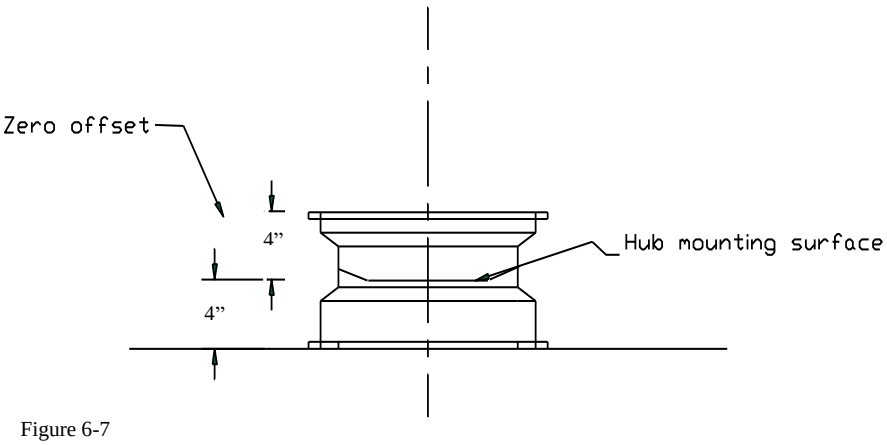


Figure 6-7

****WinGeo3**** – Refer to *Build / Fixture / Enter fixture/spindle point* to work with rim offset.

Uprights and Spindles

The spindle/upright is the heart of the independent suspension system. All suspension design characteristics begin with the spindle co-ordinates. When designing an independent suspension system, if the spindle design is free, the instant centers (as discussed shortly) are most likely the first points the designer will locate. If the spindle co-ordinates are predetermined by the sanctioning body, previous design restraints and/or financial restraints, the instant center locations will, to some extent, be restricted by the current upright/spindle design.

Kingpin Axis and Inclination Angle - The angular difference between true vertical and a line drawn through the center of the upper and lower ball joints as viewed from the front is referred to as the kingpin angle ("Kingpin axis" is a line not an angle)- Figure (6-8). Positive kingpin **angle** has the top ball joint moved towards the chassis and the bottom ball joint moved towards the tire. Kingpin inclination angle is a compromise between scrub radius and weight jacking as will be discussed later in this and subsequent chapters. Adding kingpin angle is used to reduce the kingpin offset moment. Changing the kingpin inclination can also change the steered camber and caster angles, which, in turn, can raise the spindle and lower the chassis or lower the spindle and raise the chassis. This, in turn, changes the amount the springs, bars and dampers are compressed, and, ultimately, the loads at each contact patch. Changing anything changes everything else to some degree or other.

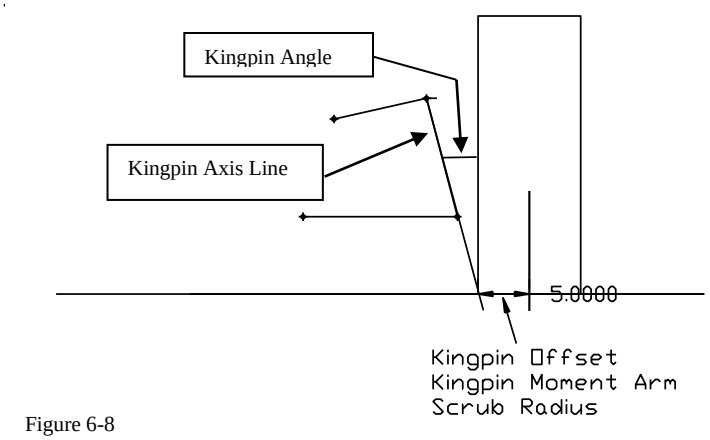
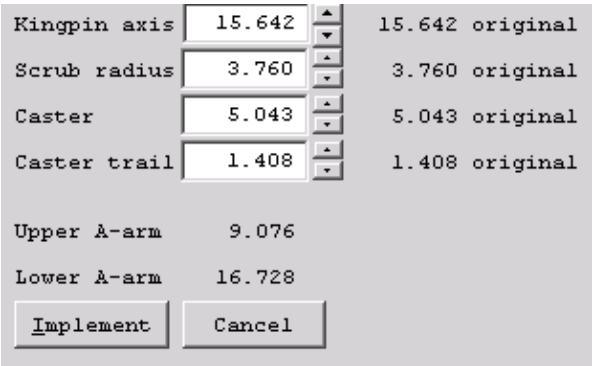


Figure 6-8

Begin WinGeo3 Example

Design: **Kingpin, Scrub and Caster**

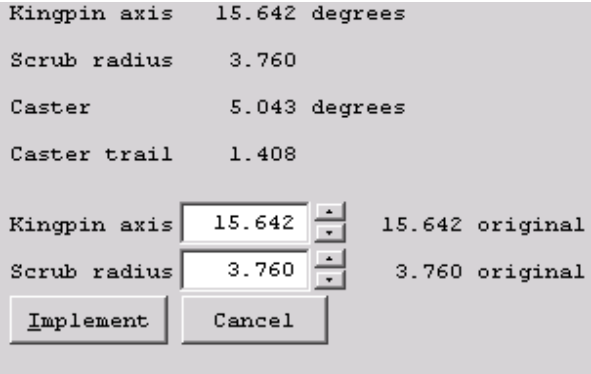
Using this menu, the spindle kingpin axis, scrub radius, caster and caster trail can be modified by modifying the numbers in the upper right hand corner of the page as seen in Win 6-6. Because it **maintains** the original **camber** angle both the spindle and control arm co-ordinates are modified. Going to *Build* / *Fixture* is the best place to view the changes in suspension and upright/spindle co-ordinates. **Changes** made in *Design / Kingpin, Scrub and Caster* **change** the suspension co-ordinates in Edit / Suspension.



Win 6-6

Build: **Caster Angle and Trail**

In this menu the control arms are altered to move the kingpin axis while **preserving** the spindle/upright – Win 6-7. The ball joints are moved **and** the static **camber** is **changed**. Changes made in *Build / Caster angle and trail* changes the control arm co-ordinates in *Files / Edit*.



WinGeo3 6-7

*****End WinGeo3 Example*****

Most designers and race engineers understand the importance of the kingpin offset moment arm. Not as many realize how the very small changes in camber, caster and steering as the result of changing the kingpin angle can dramatically change the loads seen at the contact patches. As we will see later in the book, a change of even 10 lb. of normal load at a contact patch can change the balance of the chassis. A 20 lb. load change can make a considerable change in a chassis' handling. The following tables are a quick first look at how dramatic small changes can be. Table (6-1) has a kingpin angle of 7 degrees and is used as the base setting. Table (6-2) has a larger kingpin angle of 12 degrees and Table (6-3) has a smaller kingpin angle of 2 degrees. Though the resulting camber, caster and steering changes are quite small, the resultant load changes are quite considerable. Once again, this type of exploration is impossible without computers and appropriate software. Later in the book, we will spend time examining actual on track data suspension acquisition.

Table 6-1

Effect of Kingpin Angle Change. **Base Setting** (RF Spring = 900 lb. - LF Spring = 800 lb. - Bar = 268 lb.)						
Kingpin Angle 7 Degrees (Base Setting)	Zero Suspension Motion	-1" of Ride Motion (Dive)	1^0 of Roll Motion (Left Turn)	-1^0 of Left Steering Motion (8.395 Dg)	Ride, Roll & Steer Combined	Change From Base Setting
Camber RF	-3.500	-4.465	-3.027	-4.052	-4.740	
Caster RF	5.007	-4.465	5.390	4.921	5.894	
Steering RF	-0.125	-0.348	-0.270	8.395	7.993	
Load RF	-0.000	545.931	1516.434	-325.077	1511.902	
Load LF	-0.000	429.091	-1535.76 *	319.194	-611.511	
Spring RF	0.000	-0.844	-0.489	0.101	-1.199	
Spring LF	0.000	-0.835	0.860	-0.009	-0.476	
Bar Twist	0.000	0.111	5.667	-1.213	4.227	

(* Actual force depends on spring preload)

Table 6-2

Effect of Kingpin Angle Change. **12 Degrees** (RF Spring = 900 lb. - LF Spring = 800 lb. - Bar = 268 lb.)						
Kingpin Angle 12 Degrees	Zero Suspension Motion	-1" of Ride Motion (Dive)	1 Degree of Roll Motion (Left Turn)	-1 of Left Steering Motion (8.395 Dg)	Ride, Roll & Steer Combined	Change From Combined Base Setting
Camber RF	-3.500	-4.778	-3.203	-3.962	-5.122	0.382 (more)
Caster RF	5.007	5.589	5.338	4.931	5.780	0.114 (less)
Steering RF	-0.125	-0.331	-0.261	8.391	7.978	0.015 (less)
Load RF	-0.000	490.832	1351.981	-321.964	1281.862	230.04 (less)
Load LF	-0.000	342.621	-1382.69	316.097	-496.480	-115.03 (less)
Spring RF	0.000	-0.812	-0.473	0.097	-1.152	0.047 (less)
Spring LF	0.000	-0.791	0.461	-0.116	-0.474	0.002 (less)
Bar Twist	0.000	0.184	5.459	-1.289	3.949	0.278 (less)

Table 6-3

Effect of Kingpin Angle Change. **2 Degrees** (RF Spring = 900 lb. - LF Spring = 800 lb. - Bar = 268 lb.)						
Kingpin Angle 2 Degrees	Zero Suspension Motion	-1" of Ride Motion (Dive)	1 Degree of Roll Motion (Left Turn)	-1 of Left Steering Motion (8.395 Dg)	Ride, Roll & Steer Combined	Change From Base Setting
Camber RF	-3.500	-4.235	-2.899	-4.134	-4.475	0.265 (less)
Caster RF	5.007	5.721	5.431	4.915	5.984	0.009 (more)
Steering RF	-0.125	-0.336	-0.264	8.403	8.072	0.079 (more)
Load RF	-0.000	561.399	1562.667	-302.661	1627.851	116.0 (more)
Load LF	-0.000	472.839	-1563.580	291.511	-680.215	- 69.0 (more)
Spring RF	0.000	-0.852	-0.492	0.101	-1.215	0.016 (more)
Spring LF	0.000	-0.851	0.484	-0.082	-0.463	0.013 (less)
Bar Twist	0.000	0.065	5.713	-1.103	4.402	0.175 (less)

All spindles and uprights are designed to have a prescribed kingpin angle built in - Figures (6-9/6-11).

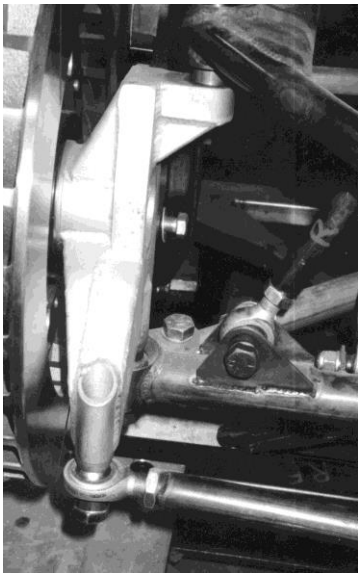


Figure 6-9 – GT Chassis

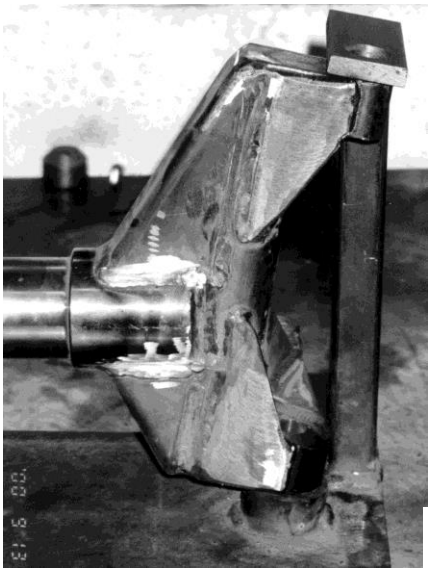


Figure 6-10 – 5x5 Stock Car Spindle

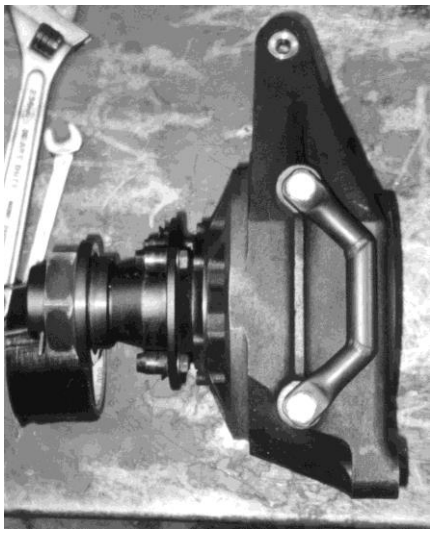


Figure 6-11 – Formula Upright

Kingpin Offset - Kingpin offset is determined by first projecting a line through the upper and lower ball joints to the ground and then measuring the distance from where this line intersects the ground to the center of the tire contact patch (front view) - Figures (6-12). The kingpin offset creates a front view moment arm extending from the center of the tire contact patch to the intersection of the spindle/upright axis with the road surface.

The angle of the kingpin axis will determine the type of force seen at the center of the hub. If the spindle/upright has zero kingpin inclination, the forces at the hub center line will be purely vertical. If the upright/spindle kingpin axis is inclined at some angle other than vertical, the hub will see a resultant force. *Key tuning tool*.

The center of the tire contact patch will not necessarily be at the center of the tire depending on rim offset, suspension geometry and /or lateral forces. The center of the tire contact patch will also move with changes in the tire's rolling radius, shear angle and shear percentage, which, in turn, changes the length of the kingpin offset moment arm.

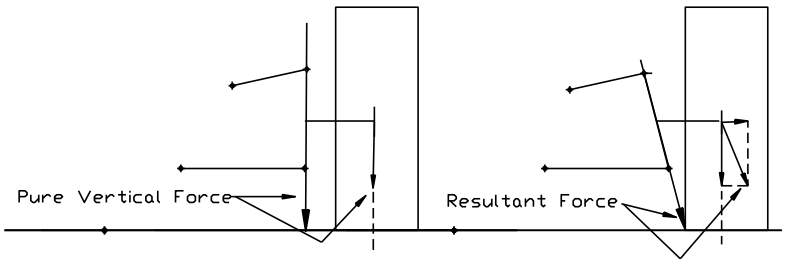


Figure 6-12

- The contact patch turning axis is directly below a point at the vertical centerline of the hub and the rim mounting surface.
- The kingpin offset moment arm and scrub radius are one and the same.

WinGeo3 Example - Use the same WinGeo3 examples as in Kingpin Axis.

Stock Car Spindles - Spindles used on many modern stock cars can be purchased in a semi-finished form from companies such as **Speedway**. All spindle shaft machining is completed but both the lower and upper portions of the spindle are extended and meatier so the team can place the upper ball joint and steering arm/lower ball joint mount co-ordinate anywhere within a prescribed range. Both the upper height and kingpin angle can be changed within a prescribed area with the excess material machined off. Similarly, the lower ball joint and steering arm can be moved up or down a prescribed amount with the excess material machined off - Figure 6-13.

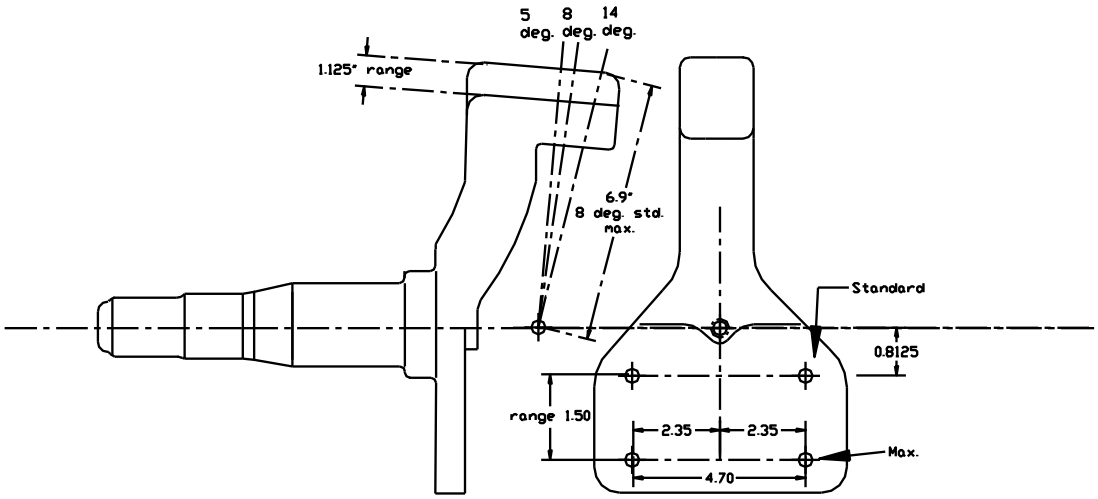


Figure 6-13

The steering arms can also be purchased in a number of different configurations including standard and dropped. Figures 6-14 & 6-15 show a spindle that began life as semi-finished. The steering arm is a dropped design.

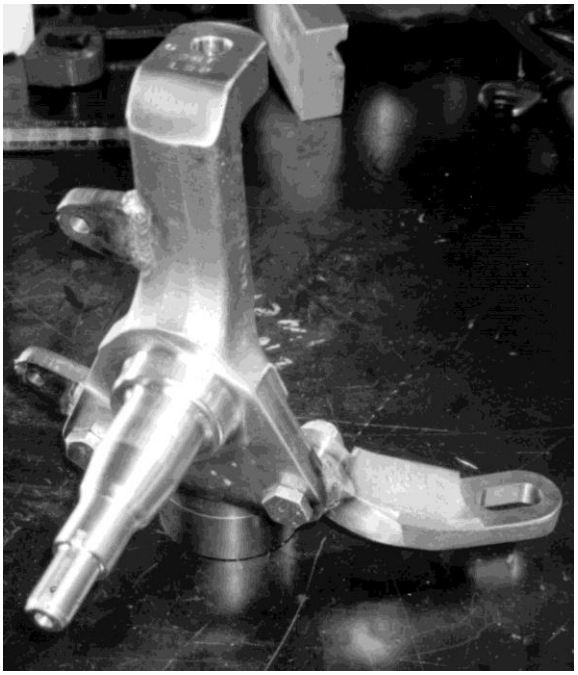


Figure 6-14



Figure 6-15

Camber

Camber - The inward or outward tilt of the wheel relative to a vertical line projected through the center of the contact patch is referred to as camber. Negative camber has the top of the tire pointed inwards towards the chassis - Figure 6-16.

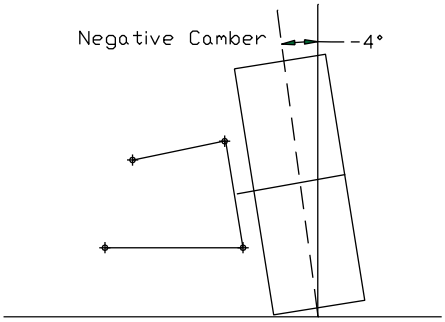


Figure 6-16

Camber Gain (Change in Either Direction) - Camber gain is the amount camber changes in relation to chassis bump and roll. A flat track requires more camber gain (positive at the inside wheel, negative at the outside wheel) than a banked track primarily because on a banked track, you have less body roll i.e. stiffer roll rates and lower overturning moments. When a wheel is moved up in relation to the chassis (bump), it normally results in negative camber gain i.e. both the inside and outside wheels see an increase in negative camber. However, body roll normally results in positive rather than negative outside wheel camber gain and negative gain at the inside wheel. Even when bump and roll are combined, the result may well be a positive and not negative gain at the outside wheel. A computer is needed to run through all the required iterations.

Bump Camber Gain - The direction and rate of camber change during bump is determined by the suspension co-ordinates. The two primary factors which determine camber gain in bump are A-arm/wishbone length and angle.

Bump Camber Gain - A-Arm/Wishbone Length - A shorter upper A-Arm length will result in the arc tightening (horizontal distance between the outer and inner upper control arm mounts decreasing), which, in turn, increases the camber gain rate. Whether the gain is positive or negative depends on the placement of the arc - Figures 6-17 & 6-18.

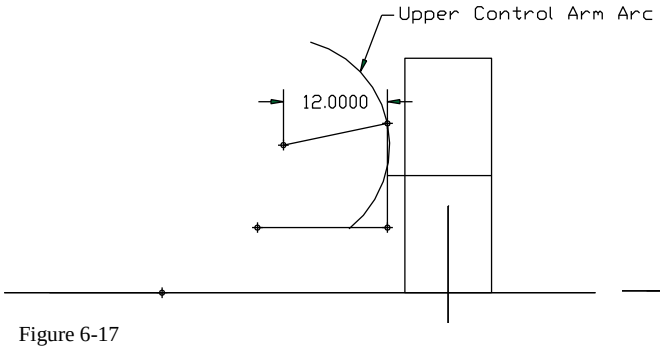


Figure 6-17

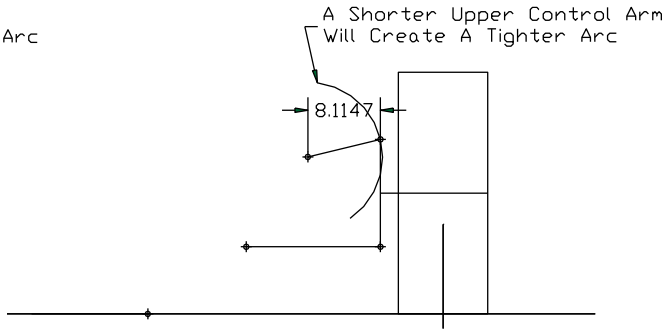


Figure 6-18

Bump Camber Gain - A-Arm/Wishbone Angle - The angle of the upper and/or lower A-Arm, in turn, determines the instant center height and length. Instant centers as discussed later in this chapter are a point of intersection between lines drawn through the upper outer and inner suspension points and the lower outer and inner suspension points (two-dimensional front view) – Figures 6-19 & 6-20.

- As the instant center length is increased, camber gain due to bump slows down i.e. the swing arm (distance between the instant center and the spindle's point of axial rotation) radius increases. The shorter the swing arm radius, the tighter the arc.

- Changing the height of the instant center has somewhat less effect on bump camber change than changing the instant center length.

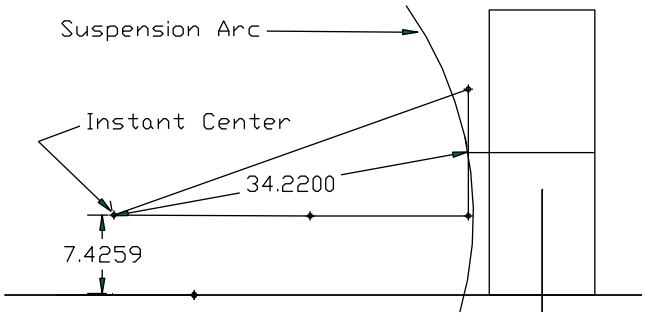


Figure 6-19

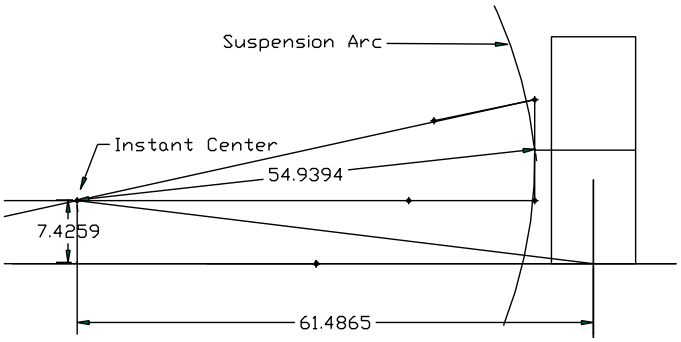


Figure 6-20

Figures 6-21 & 6-22 show portable jigs used to build control arms of different lengths and offsets. Portable jigs are often used during a test session when the team is away from home base and the race engineer wants to try a new control arm length.

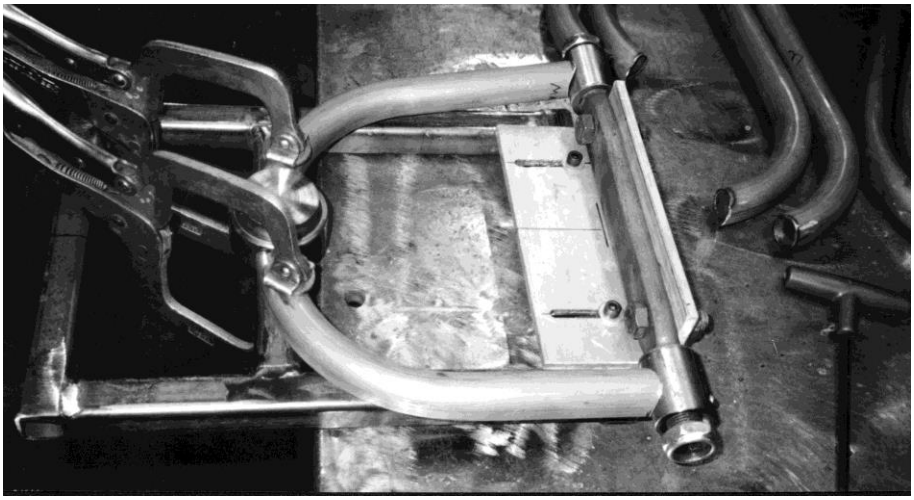

Figure 6-21 – Built in twenty minutes just before a five day test session, this jig was used to build eighteen different stock car upper suspension A-Arms.



Figure 6-22 – This portable jig was set up to build all the upper and lower A-Arms for a formula car. One side was designed for the left side suspension while the other side was used for the right side suspension.

Roll Camber Gain - Body roll at the chassis centerline affects camber gain or loss in the opposite direction to suspension bump - Figures 6-23. In this instance, body roll produces positive camber gain on the outside and negative camber gain on the inside. Outside bump would produce negative camber gain while outside droop would produce positive camber gain.

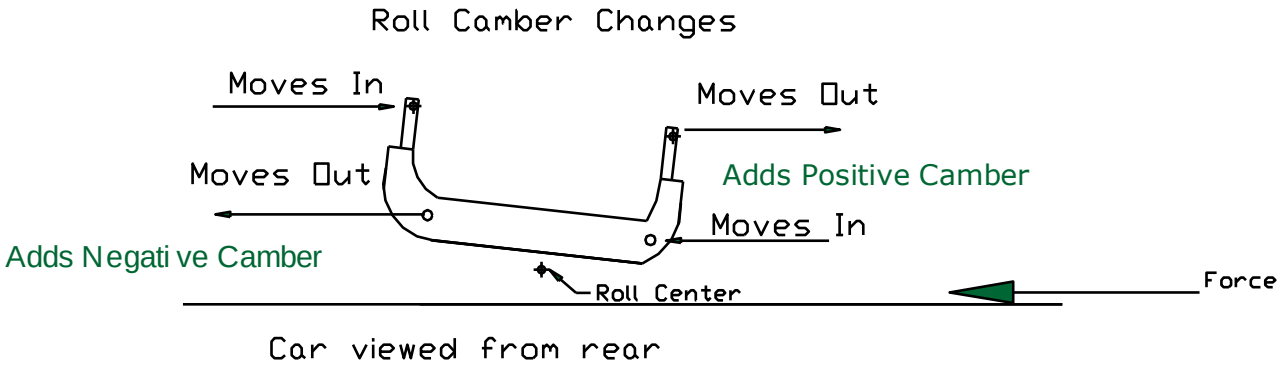


Figure 6-23

Instant Center Location Effects on Camber during Roll:

- Body Roll Camber Change - Instant Center Beyond the Center Line of the Chassis
- Camber angle decreases as the result of body roll when the instant center point is beyond the centerline of the chassis.
- Body Roll Camber Change - Instant Center at the Center Line of the Chassis
    - Camber angle does not change appreciably as the result of body roll when the instant center point is at the centerline of the chassis.
- Body Roll Camber Change - Instant Center Before the Center Line of the Chassis
    - Camber angle increases as the result of body roll when the instant center point is before the centerline of the chassis.

Steering Iterations and Camber Change - Steering angle iterations in combination with bump and/or roll introduces an additional set of camber change iterations.

The amount of camber a car requires depends on the track surface, banking, wheel rates, scrub or lateral displacement, tire width to rim width, tire sidewall height and stiffness (more negative camber is needed with a softer and/or taller sidewall), air pressure, tire compound, car weight, driver style, track temperature etc. To determine the right amount of camber needed, look at tire temperatures, tire surface patterns and tire wear. In spite of what many people think, this is not that easy to do. The only accurate way to observe tire temperature during cornering is with real time infrared data. The bottom line for determining the required camber is the stopwatch versus tire degradation rate. Optimum tire operating temperature is far more important than whether you have a textbook camber temperature profile.

Camber is most often added or subtracted by adjusting rod ends - Figure 6-24, by using a series of camber shims - Figures 6-25 & 6-26 or by employing a slotted steering link - Figure 6-27. Normally, changing camber by adjusting a rod end will also change the static toe settings while using a shim or slotted system does not.

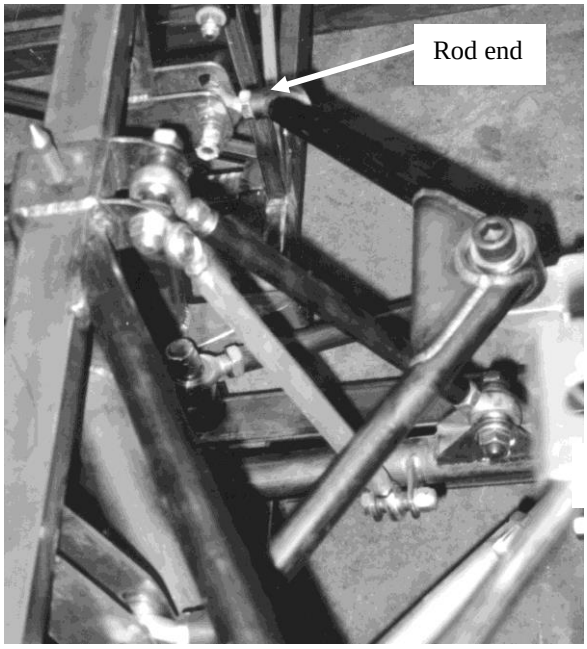


Figure 6-24

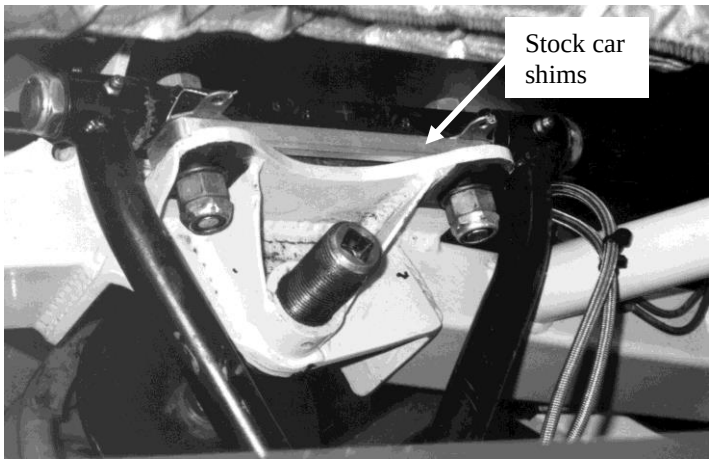
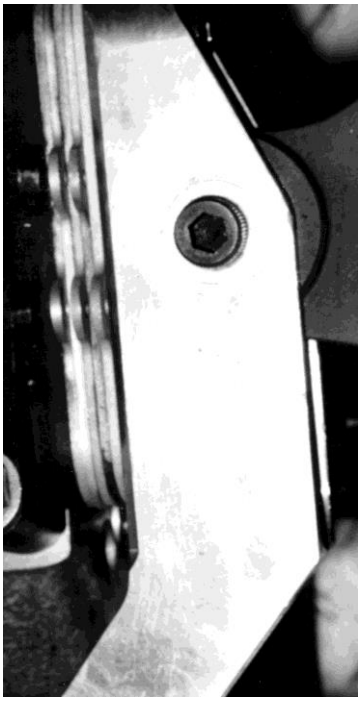


Figure 6-25


Figure 6-26

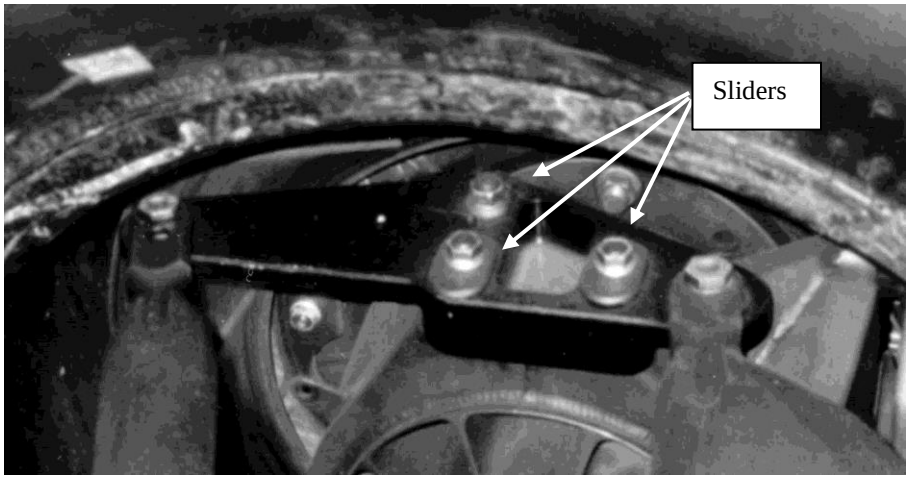
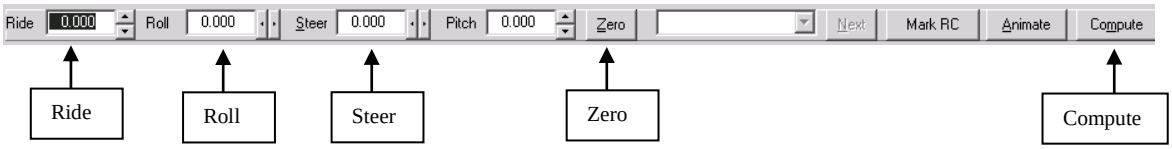


Figure 6-27

Begin WinGeo3 Example*

Looking at Camber Change

Single Ride Calculations – From the main WinGeo3 graphics screen, enter a ride calculation and click on compute. A negative ride number moves the wheels in bump i.e. the wheels move up/chassis moves down. Entering a number in the ride box moves both wheels the same amount. Click on the zero box to return the ride to zero. Spend considerable time studying the kinematic changes. Pay particular attention to whether the right and left side camber is increasing or decreasing.



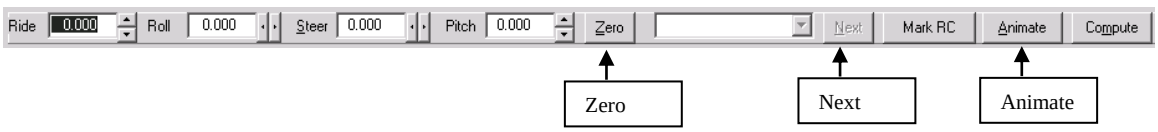
Single Roll Calculations – From the main WinGeo3 graphics screen, enter a roll calculation and click on compute. A negative roll calculation rolls the chassis to the left while a positve roll calculation rolls the chassis to the right. Click on the zero box to return the roll to zero degrees. Spend considerable time studying the kinematic changes while paying particular attention to whether the right and left side camber is increasing or decreasing.

Single Steer Calculations – From the main WinGeo3 graphics screen, enter a steer calculation and click on compute. A negative steer calculation steers the wheels to the right while a positve steer calculation rolls the chassis to the left. Click on the zero box to return the steer to zero. Spend considerable time studying the kinematic changes. Pay particular attention to whether the right and left side camber is increasing or decreasing. Check to see that steer is set at *Angle – Uitility / Toesteer measurment / degrees*. The various steer calculations will be covered in Chapter 7 including the steering ratios and derivatives.

Combined Single Calculations - From the main WinGeo3 graphics screen enter a variety of different ride, roll and steer combinations. Pay particular attention to whether the right and left side camber is increasing or decreasing.

Path Files – Go to *Reports / Load Path file* and load the appropriate path file. Begin with one of the ten phase cornering files. Be aware that you can not run the short track file with the formula car graphics because the formula car suspenion can not handle that amount of motion. Each path file will run 21 lines of data.

- Click on *Reports*
- Click on *Load Path file*
- Click on an appropriate path file
- To scroll through each iteration one at a time, click on the *Next* box
- Click on the Animate box in order to watch the iterations scroll through automatically
- The rate of animation can be changed by going to *Utility / Anitmation delay* and then clicking on the appropriate rate.



- To stop the animation reclick on *Animate*.
- Click on zero to return the ride, roll and steer to zero.

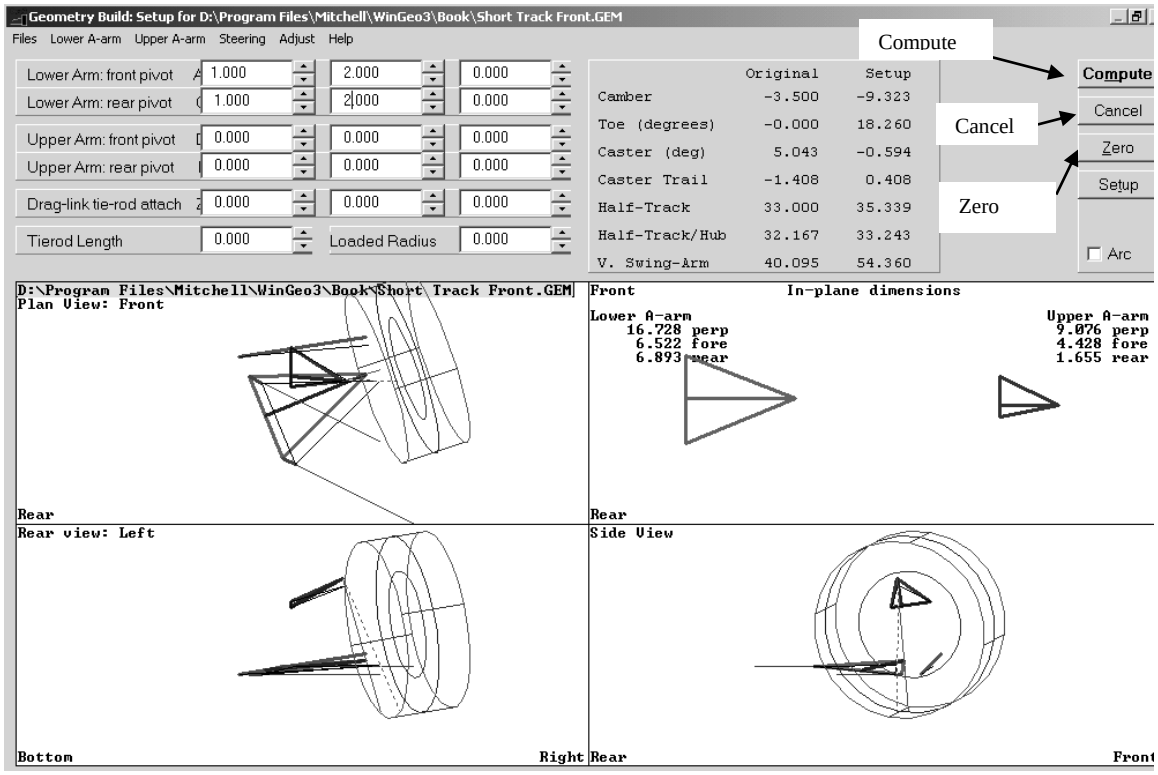
Changing Camber Angle

Don't change the camber angle in *Edit / Suspension*. This file is strictly for entering measured data as the chassis sits on the set up pad or from design data. Changing the camber angle from this file will distort the upright/splindle co-ordinates.

Design: **Extend Upper A-arm** – Changing the control arm perpedicular lengths in this menu does not change the camber angle but rather changes the x, y and z inner pickup points. The user must click on the immplement button before the program makes the change.

Build: **Camber Angle** - Changing the camber angle in this menu changes both the upper and lower control arm ball joint co-ordinates. The camber change is made by rotating the upright/spindle about the hub center. This is the least used camber change menu.

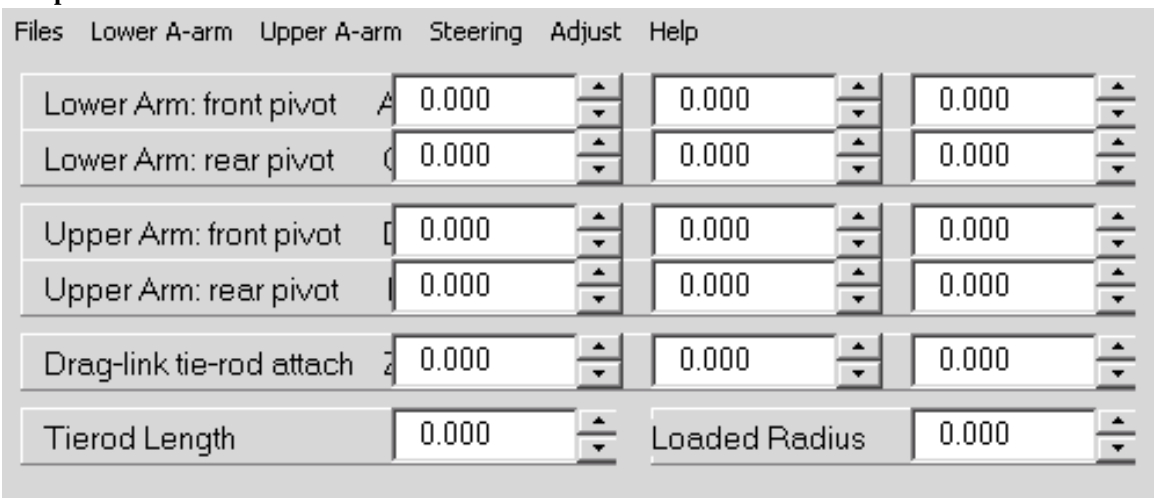
Build: **Set up permanent changes** - This form allows the user to manipulate the inner pickup points of the suspension. In the Win 6-8 example, both lower A-arm inner pickup points are moved 2.0 inches wider and 1.0 inch towards the rear. The movement is shown in the software graphics as a thick red line. The extensions of the movement lines exaggerate the direction of the movement. This form allows the user to make a wide variety of set up changes. Clicking on compute makes the changes in the *Edit / Suspension file*. Clicking on zero returns all of the values to zero while clicking on cancel aborts the changes and returns the program to the main graphics screen.



Win 6-8

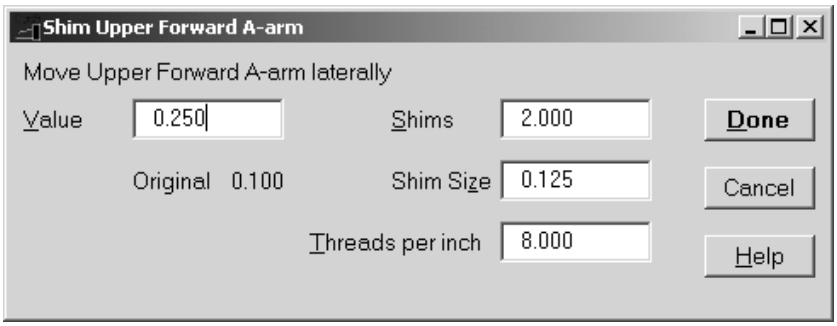
Changes made in Build / Setup are automatically saved in the Edit / Suspension file. Notice that even if a change was made at an upper control arm co-ordinate only, both the x, y, and z upper and lower ball joint co-ordinates end up being moved. This is the result of the upright/spindle being pivoted about the tire contact patch.

Setup Menus – WinGeo3 6-8



Win 6-9

Lower A-arm – The suspenion co-ordinate changes can either be made by using the provided boxes located on the *Build / Setup* screen or by clicking on the upper or lower A-arm menu as seen in Win 6-9. Clicking, for example, on the upper A-arm menu allows the user to choose from 18 different set up scenarios as will be outlined shortly. Once a set up scenario has been picked, a form as shown in Win 6-10 will come up. The user can then enter a real value in either absolute terms or a number of physical



Win 6-10

shims. For instance, if you are using 1/8-inch shims (0.125”), then two shims represent a 0.250” change. In the Win 6-10 example, two shims produce a movement of 0.250”. You can also turn a rod end in or out a prescribed number of turns. Screw size is the reciprocal of shim size. A shim of 0.125” corresponds to eight threads per inch. Two turns of a bolt that has 8-threads-per-inch would represent a movement of 0.250”. Use the Done button to accept the value and leave the form. Use the Cancel button to leave the form without using the value.

You may also enter a negative number of shims, which would correspond to reducing the number of shims or adding shims to the inside of the inner pickup points (stock car).

Setup Form: Files menu

Specify whether the setup changes apply to the Feature file, all files except the Feature file, or all files. See Setup: Files: Select files. This menu also contains a command to print the graphics and an Exit command to leave the form without performing a calculation.

Set up form: Lower A-arm menu - Make a variety of set up changes to the lower A-arm.

- Shim lower width - Move both lower inner pickup points A and C laterally. The primary change is to camber. A **positive shim** moves the inner pickup points outwards and increases static camber. A **negative shim** (fewer shims outboard or more shims inboard) moves the inner pickup points inward and decreases static camber.

- Shim lower fore-aft - Move both lower inner pickup points A and C fore-aft. The primary change is to caster.

- Eccentric caster change - Move the lower rearward pickup point perpendicular to the A-C axis. This is a common stock car adjustment. Primary changes are to camber and caster.

- Raise height - Raise both lower inner pickup points A and C vertically. The primary change is to the instant center.

- Lower height - Lower both lower inner pickup points A and C vertically. The primary change is to the instant center.

- Shim Forward width - Move the forward lower inner pickup point A laterally. The primary change is to camber and SVSA (Side View Swing Arm (IC)).

- Shim Rearward width - Move the rearward lower inner pickup point C laterally. The primary change is to camber and SVSA.

- Raise Forward height - Move the forward lower inner pickup point A vertically. The primary change is to the instant center and SVSA.

- Raise Rearward height - Move the rearward lower inner pickup point C vertically. The primary change is to the instant center and SVSA.

- Lengthen forward - Lengthen the forward link A-B. The primary change is to caster.

- Lengthen rearward - Lengthen the rearward link C-B. The primary change is to caster.

- Slide A-arm along base - Move points A and C along their axis.

- Shim A-arm along perpendicular - Move points A and C perpendicular to their axis.

- Shim A-arm behind perpendicular - Move points A and C perpendicular to their axis.

- Change Lower A-arm for camber - Enter a desired camber value and move A and C.

- Change Lower A-arm for caster - Enter a desired caster value and move A and C.

- New length forward - Enter a length for the front link A-B and move the point A along the axis A-B to achieve the desired length.

- New length rearward - Enter a length for the rear link C-B and move the point C along the axis C-B to achieve the desired length.

- Design Lower A-arm - Enter the dimensions of the lower A-arm and move the points A and C along the axis A-B and C-B.

Steering menu – This menu will be covered in Chapter 7.

Adjust menu – See *Compute / Solve for Setup changes (temporary)* for an explanation of the *Adjust menu*.

Compute: **/ Solve for Set up changes (temporary)** – This option works much the same as the *Build / Setup permanent* changes except the changes are not automatically added to the *Edit / Suspension* menu.

When the control arm co-ordinates are changed, this will also result in a slight dimension change to the control arm itself. For example, when an inner rearward rod end is turned out (lenghtened) the distance between the two inner points will increase which means the inner points will need to be compressed in slightly in order for the control arm to fit back into the inner mounts. Doing so will result in a small co-ordinate change at the upper and lower ball/rod end joints. While using *Compute / Solve for Setup changes (temporary)* while making a 0.500" change at the y co-ordinate of the Upper Arm front pivot, the program will indicate the D-F distance has changed from 6.083" to6.103" (red box). The user can correct for that change by going to the *Adjust* menu and selecting either the *Upper A-arm forward D* or *Upper A-arm rearward F* menu. If, for example, *Upper A-arm forward D* was opened the user can then choose whether the change will be made in the x, y or z direction. In Win 6-11, the adjustemt was made at the x – for-aft co-ordinate i.e. the upper front inner mount was moved longitudinal 0.021" in order to fit back into the original front inner mount.

Lower Arm: front pivot	A	0.000	0.000	0.000	A-C	13.415
Lower Arm: rear pivot		0.000	0.000	0.000	A-C	13.415
Upper Arm: front pivot	D	0.021	0.500	0.000	D-F	6.083
Upper Arm: rear pivot		0.000	0.000	0.000	D-F	6.083

WinGeo3 6-11

Build: **Fixture** – The solve for fixture menu, as described in Chapter 5, changes the camber angle, amongst other things by changing the dimensions of the upper and/or lower control arms. Changes made in the *Build / Fixture* menu automatically adjusts the *Edit / Suspension* co-ordinates.

*****End WinGeo3 Example*****

Caster temporary

Caster or Side View Kingpin Offset - Caster is viewed from the side view or longitudinal plane and is the angular difference in steering axis inclination from true vertical in the longitudinal plane. Caster is the side view of steering axis inclination - Figure 6-28.

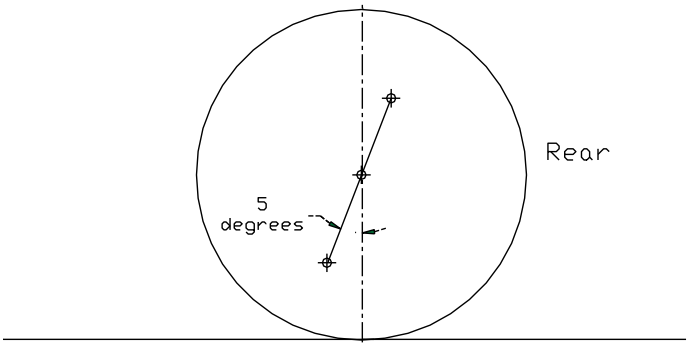


Figure 6-28

Positive caster has the upper ball joint moved towards the rear of the car or the lower ball joint moved towards the front of the car in the side view. Caster on most modern racecars is adjusted by moving the upper and/or lower ball joints/axial pivot points along their x co-ordinates (longitudinally). These points on open wheel and sports cars are normally moved by changing the length of the inner rod ends on the upper and/or lower A-Arms i.e. lengthening the upper forward rod end and shortening the upper rearward rod end would increase positive caster by moving the upper axial mounting point towards the rear of the chassis (this adjustment would also have some effect on the camber angle, static toe etc.). The adjustments needed to change the caster a prescribed amount are one of many set up procedures that need to be worked out and documented in the shop and not at the track or even during final set up at the shop.

Changing the caster angle also **changes** the **wheelbase** i.e. when caster angle is changed on one side only, the side to side wheelbase becomes asymmetrical. Adding positive caster angle shortens the wheelbase – Figures 6-29- 6-32.

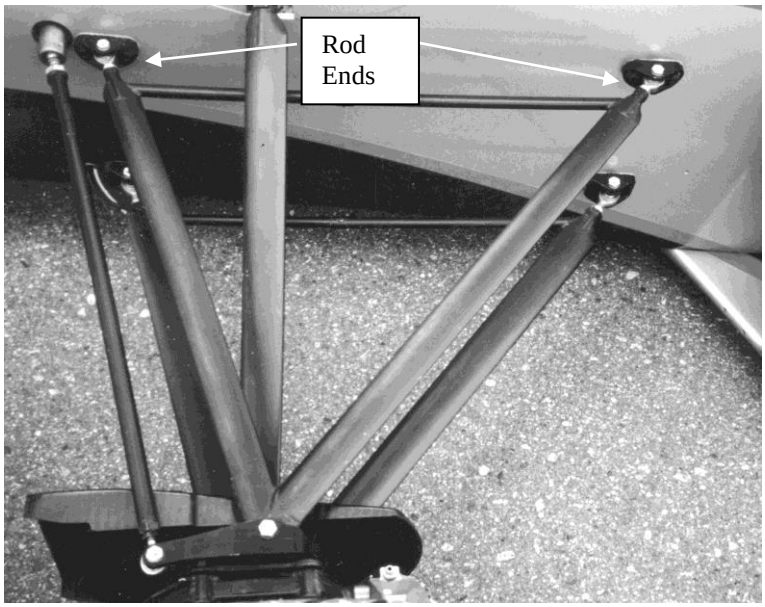


Figure 6-29 – Shortening the upper rearward rod end on a front suspension system adds positive caster and shortens the wheelbase on that side



Figure 6-30 – Shortening the upper rearward rod end on a rear suspension system decreases the caster angle and lengthens the wheelbase on that side of the chassis. Shortening the lower rod end increases the caster angle while also lengthening the wheelbase.

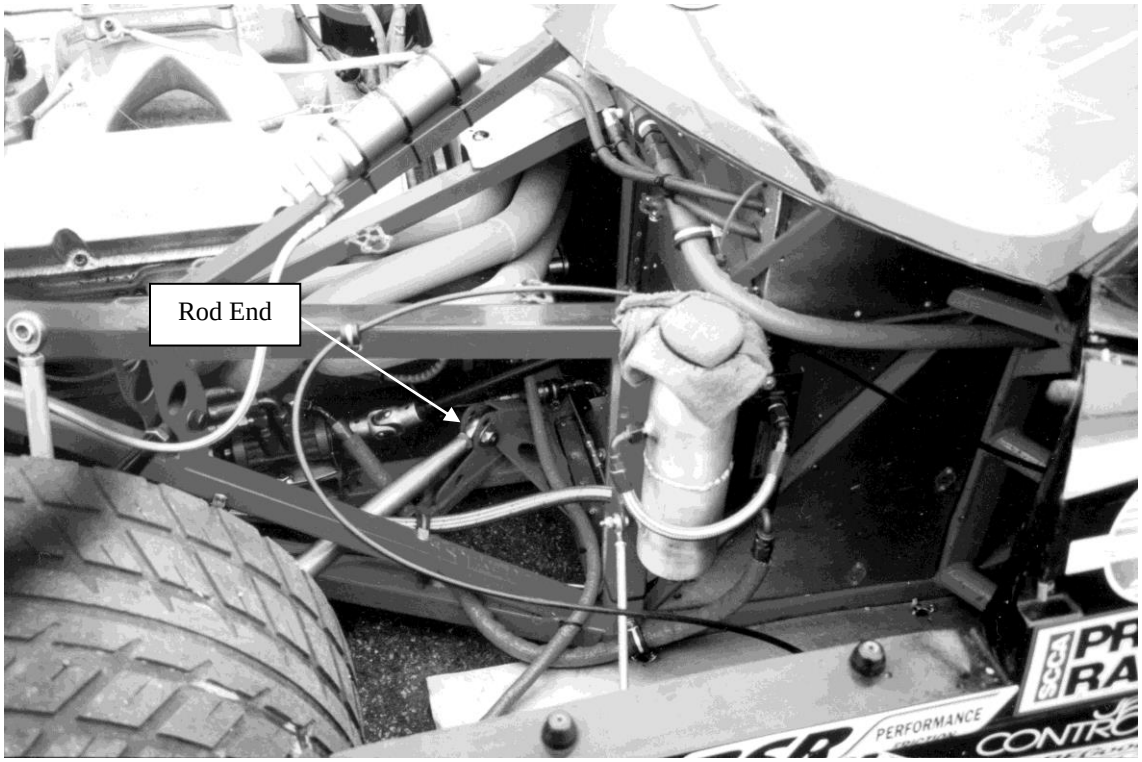


Figure 6-31 – Shortening the lower rearward rod end on this Trans Am front suspension decreases the caster angle and shortens the wheelbase.

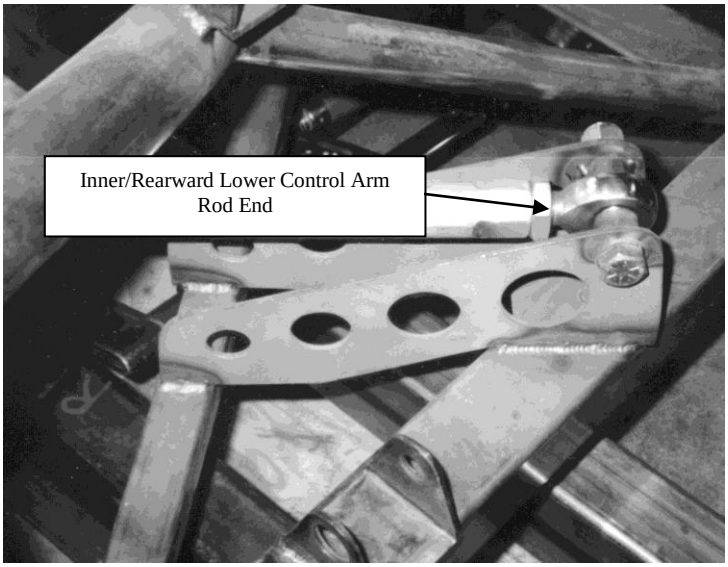


Figure 6-32

Caster Adjustment - Stock Cars - Caster adjustment on stock car types of suspension can be accomplished in a number of ways:

- Shimming of the inner mounting points of the upper control arm different amounts at the front and rear i.e. shimming the rearward mounting point to move inward towards the center line of the chassis and/or shimming the forward mount to move outward away from the center line of the chassis adds positive caster - Figure 6-33.

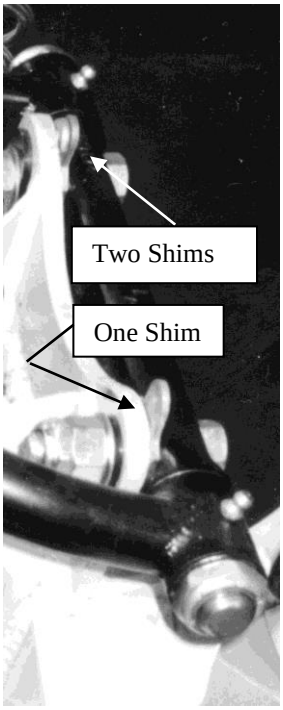
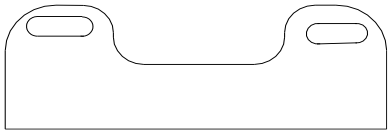


Figure 6-33

- Using slotted cross shafts (inner mounting points) so that the cross shaft can be slid longitudinally (x co-ordinates) - this allows for caster adjustment without changing the camber angle - Figure 6-34. Plugs are fitted into the cross shaft with various hole iterations.


Figure 6-34

- Slotting the upper A-Arm mounting bracket longitudinally also allows caster to be adjusted without changing camber angle - Figure 6-35. Plugs are fitted into the mounting brackets with various hole iterations.


Figure 6-35

Caster Design - Caster is designed into or out of a suspension system for four primary reasons:

- The shopping cart effect
- Aligning torque
- Weight (load) jacking
- Camber gain

The Shopping Cart Effect - Positive caster, because its steering axis is ahead of the centerline of the tire, has the effect of straightening the wheel as the car moves forward. The wheel is dragged behind the steering axis. The universal example used to show the effect of caster is the shopping cart - Figure 6-36. The right side of this same drawing shows a spindle with offset upper and lower ball joints which produce an offset steering axis. The centerline of the wheel follows along behind the steering axis.

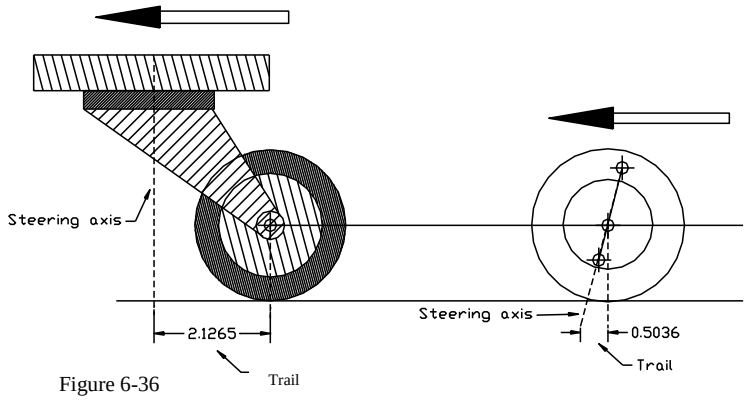


Figure 6-36

Aligning Torque - Aligning torque as the result of positive caster settings produces a self- centering effect that wants to return the steered wheels back to their neutral position. The higher the positive caster angle, the harder it will be for the driver to turn into the corner. Conversely, the wheels will return to their straight ahead position with less effort. The greater the caster offset moment arm, the greater the effort needed to turn the wheels into the corner and out of the corner (no caster split) i.e. higher percentage of feedback to the driver. A smaller caster offset moment arm produces less road feel for the driver. The various moment arms involved in the realignment torque (like all moment combinations) are quite complex to model mathematically (Book Three). In its most basic form, the aligning torques are developed by the various spring and bar forces (the vertical loads) reacting against the moment arms created between the caster inclination angle and the longitudinal center line of the tire - Figures 6-37 & 6-38.

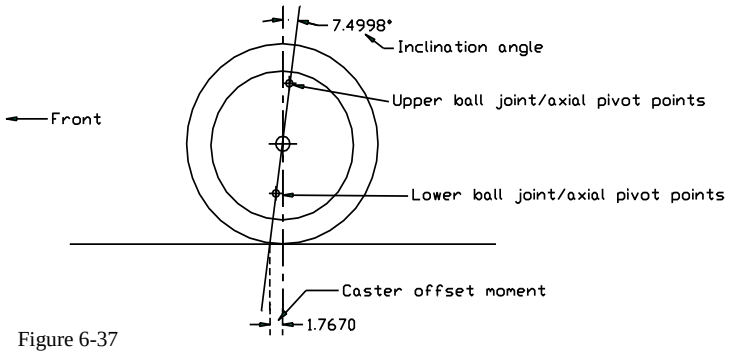


Figure 6-37

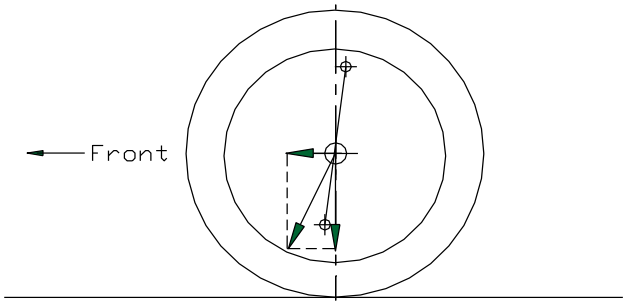


Figure 6-38

Camber Gain - An increase in caster angle results in an increase in negative camber at the outside wheel and an increase in positive camber at the inside wheel when the wheels are steered - Figure 6-39. The tire gains camber as the spindle is turned because of the angled steering axis. If the spindle was rotated 90 degrees from vertical, only camber gain would result from the steering link being moved. Changing the steering axis inclination changes both the steering ratio and camber ratio. Actually, it changes many more things than just those two but that is all we are looking at for the moment.

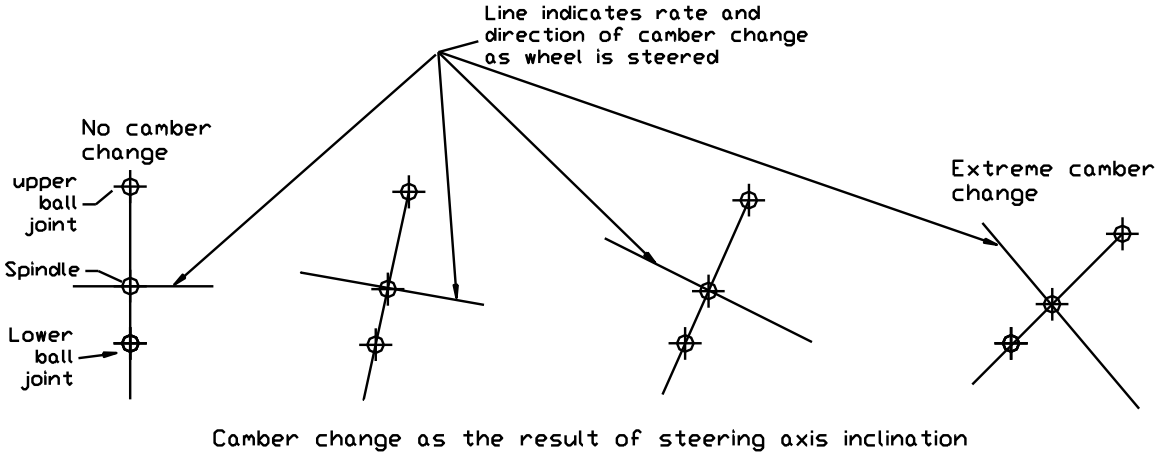


Figure 6-39

When the upper mount of the upright/spindle is angled rearward along the longitudinal plane, the forces seen at the spindle/upright mounting points are not purely vertical but are resultant forces which also contain longitudinal (x plane co-ordinate) forces. These longitudinal forces tend to torque the spindle, hub and tire. If the spindle/upright has positive caster (top inclined towards the rear of the chassis), the wheel will turn in towards the center line of the chassis. If the wheel had no steering linkage restraints, a resultant force applied at the bottom and/or top ball joint would torque the spindle, hub and tire in an inward manner past its straight ahead position until the tire is restrained from turning further (most likely by a frame member) - Figures 6-40 / 6-42. The point to remember here is that positive caster resultant forces want to turn the wheel inwards. If both the right wheel and left wheel were not restrained by steering linkage under resultant forces, they would both have a tendency to turn in towards the center line of the chassis.

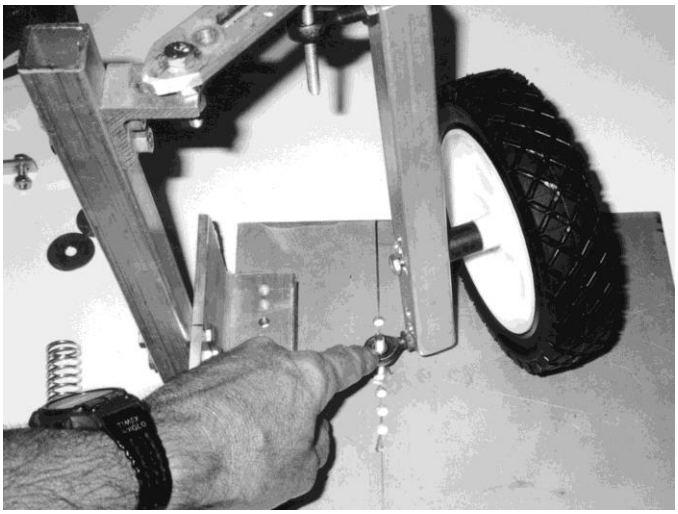


Figure 6-40 – Before vertical load is added to the lower control arm rod end, the wheel is turned out away from the chassis centerline.

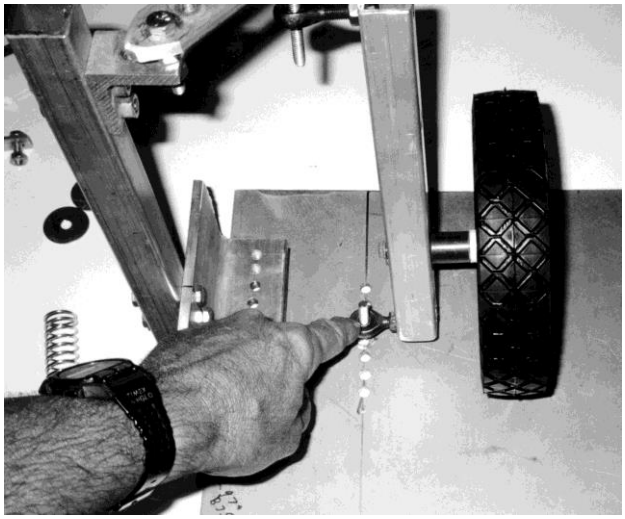


Figure 6-41 – As vertical load is applied to the lower control arm rod end, the wheel begins to turn in towards the centerline of the chassis.

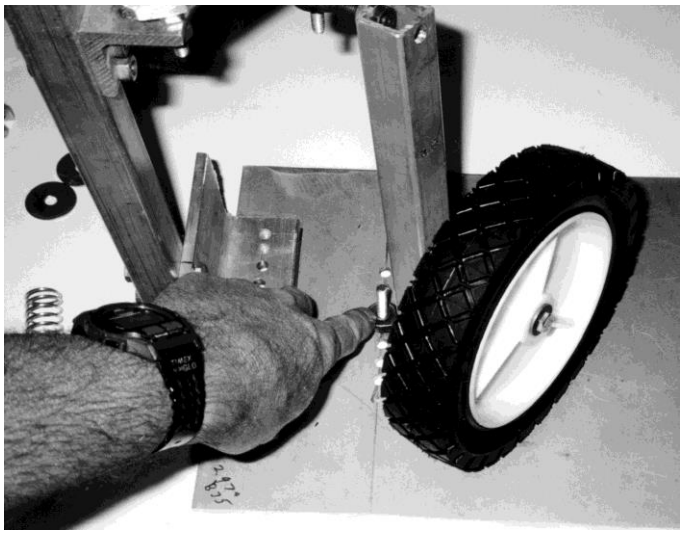


Figure 6-42 – Continued vertical load at the lower control arm rod end sees the wheel turn in towards the chassis centerline.

Because both the right and left front wheels are restrained from freely rotating by various steering linkages, the two aligning torques tend to cancel each other when the tires are at their straight ahead position. If the caster setting and resultant moment on one wheel is greater than the other wheel, the aligning torques will tend to balance at some steering position other than straight ahead. *Key tuning tool*.

The resultant forces seen at the spindle/upright mounting points are the result of spring and bar forces. It does not matter, for example, whether these forces are applied at the bottom of the spring (from the road through the spring to the chassis) or the top of the spring (from the chassis through the spring to the road) or in some combination of both. As long as there is a resultant force between the spring and/or bar and the upright/spindle, an aligning torque will develop at the tire's center of rotation - Figures 6-43 / 6-45. If both corners have an equal amount of caster and camber but one corner (spring and bar) is under a higher load, that corner will exhibit a greater aligning torque.

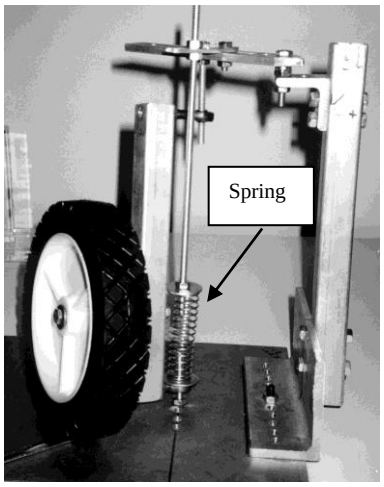


Figure 6-43 – In this example, the front suspension spring is applying vertical load to the lower control arm.

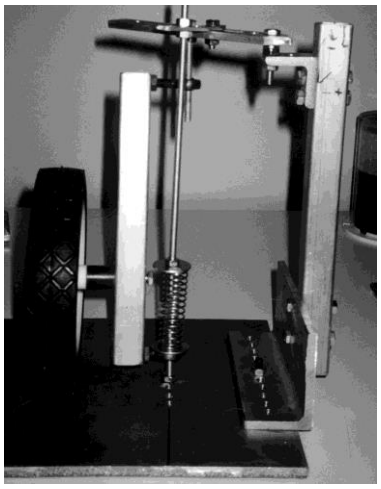


Figure 6-44

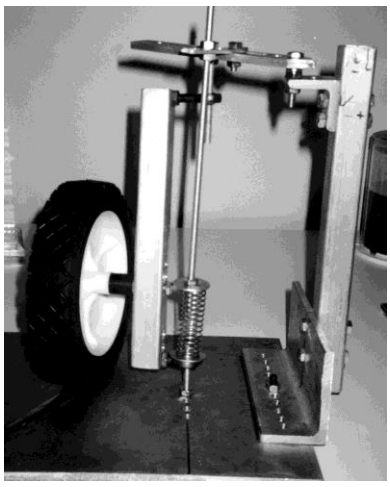


Figure 6-45

Caster Jacking - Caster Load Transfer - When the wheels are steered left in combination with positive caster, the left front spindle/upright (along with the hub and ball joints) will move in an upward direction, thereby compressing the spring and/or bar adding additional load to that corner and the right rear while the right front spindle will move down, taking weight off that corner and the left rear Figure 6-46. "When steering left with positive caster, the left spindle and tire move **down** relative to the control arm, but because they are constrained by the ground, the arm is forced **up** instead, compressing the spring and raising the chassis." William L. Ball

Figure 6-47 shows the straight ahead position of a left side upper ball joint. During a left turn, the left side spindle along with the upper and lower ball joints are raised as shown in Figure 6-48. This, in turn, compresses the springs, bars and dampers. *Key tuning tool*.

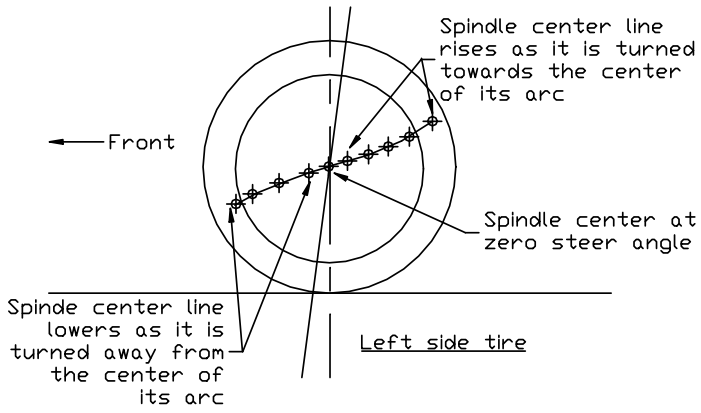


Figure 6-46

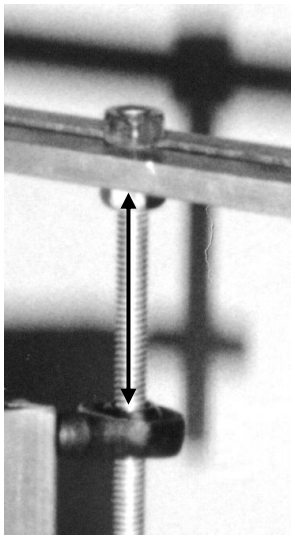


Figure 6-47

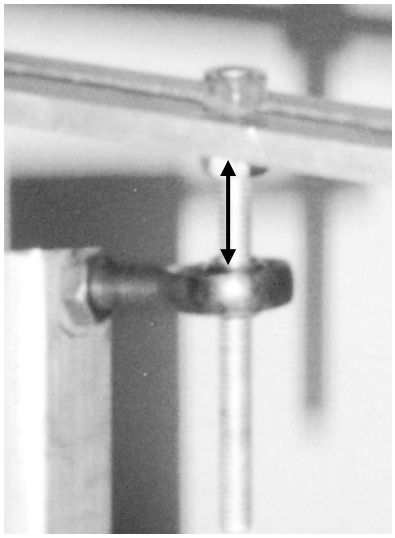


Figure 6-48

Caster Split - Caster split is the difference between the amount of right side and left side caster. Oval track cars normally run with one to three degrees more caster on the right side. In this case, the torque inducing forces on the right side are greater than on the left side, which, in turn, will result in a chassis that wants to steer left. *Well, it's not that simple:*

During a left turn, if both corners had equal amounts of caster and carried **equal loads**, then, as the springs and bars were compressed (as the result of caster gain) on the inside corner (spindle moves up) and extended on the outside corner (spindle moves down), the chassis would want to steer to the outside corner - higher inside wheel torque.

- During a left turn, if both corners carried **equal loads** but the inside tire had less caster angle than the outside wheel, the increased moment on the outside wheel may make up for the lower load on the outside spring. Depending on the size of the larger moment (larger caster angle) on the outside and the larger load on the inside, the aligning torque could even out or even make the chassis want to turn to the inside.

- It would be highly unusual to have a situation as described in the first two examples where the outside and inside loads would be equal during lateral acceleration; however, even if the outside load is greater those caster effects will still play a role in the resultant steering effects.

- **Actual On-Track Condition** - Even though the outside wheel spring and bar are extended by the spindle movement during a turn, they will also exhibit higher loading because of load transfer, which, in most cases, will more than compensate for the spindle controlled extension i.e. the outside spring and bar will still end up compressing more than the inside spring but to a lesser extent than if there was no caster inclination. *Key tuning tool*. We will come back to this point when discussing load (weight) transfer in subsequent chapters.

Caster Jacking Combined With Roll and/or Ride - Adding roll and/or ride when looking at the effects of caster jacking provides the race engineer with a whole new set of parameters. You will need a three dimensional computer program such as WinGeo3 to look at the endless number of iterations i.e. loads seen by the contact patches.

Wheel Rates, Aligning Torque and Weight (Load) Jacking - When working out the various aligning torques and corner load changes, one needs to calculate the changes in terms of wheel rates rather than spring and bar rates (Book Three). Soft wheel rates will produce much lower caster jacking forces than stiff wheel rates. This type of set up is common for oval track racing.

Track banking also affects the resultant caster forces.

Note: Unlike suspension jacking forces which pass through the suspension components, caster jacking forces are fed through the springing components (springs, bar and dampers).

****WinGeo3 Examples**** - The WinGeo3 caster menus operate in the same manner as those menus introduce thus far in Chapters 5 and 6. The user needs to spend considerable time working through the various caster change menus, as listed below, while running various kinematic iterations from the main graphics menu and the *Reports / Path files*.

- *Design / Kingpin, Scrub and Caster*
- *Build / Kingpin Angle and Scrub*
- *Build / Fixture*
- *Build / Set up Permanent Changes*
- *Compute / Solve for Set up Changes (temporary)*

Camber and Caster Effects - If the suspension corner has no caster, no kingpin angle and no camber, the spindle will remain parallel to the track surface as it turns. However, if the suspension corner is set with zero caster, adding camber and/or kingpin angle will still produce a certain amount of caster effect i.e. the spindle will move in an arc rather than level as it turns. Increasing camber angle increases the jacking effect the aligning torque caster produces when the wheels are turned. The following tables are examples from a specific chassis. Different chassis may well produce different results depending on the component positioning in relation to the arc centers.

Table 6-4

Caster Effects - Positive Caster Chassis 3A327A # 3 - 43 - 8 Date_____		
	Left Wheel	Right Wheel
Left Turn	spindle raises - spring is compressed, increasing load at the contact patch	spindle lowers - spring is extended, decreasing vertical load at the contact patch
Right turn	spindle lowers - spring is extended, decreasing vertical load at the contact patch	spindle raises -spring is compressed, increasing load at the contact patch
Important Note: The resulting effects of positive caster are in direct proportion to the amount of caster inclination, action and reaction loads at the spindle and wheel rate.		

Caster Gain - If the upper and/or lower A-arms are mounted at an angle rather than parallel to the horizontal plane of the chassis and/or each other, the bump and rebound movements of the lower and/or upper ball joint/axial pivot point will move in an arc rather than straight up and down. This will cause caster gain or loss through bump, rebound and body roll - Figure 6-49. It will also result in a change in wheelbase at that side of the chassis as well as changing the mean wheelbase.

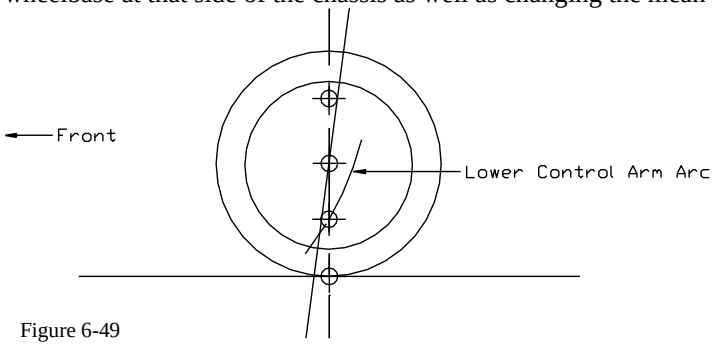


Figure 6-49

For the chassis shown in Figure 6-50, the upper ball joint will move up and down without moving longitudinally (it runs parallel with the lateral center line of the chassis) while the lower ball joint will move in an arc that moves the joint towards the rear of the chassis as it moves above and below the midpoint of its arc, thereby decreasing the angle of caster inclination. Changing the mounting co-ordinates and/or the design of the wishbones/A-Arms can have a dramatic effect on the overall spindle kinematics. *Key tuning tool*.

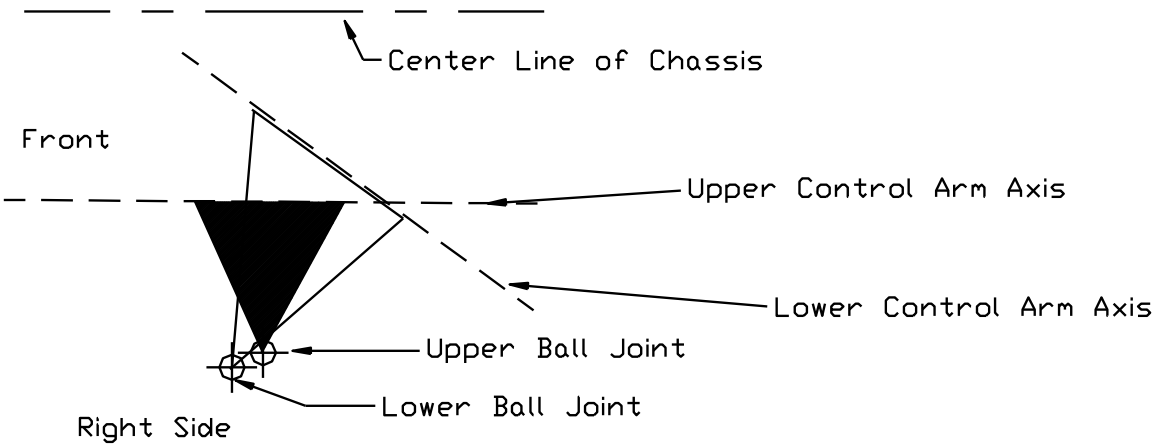


Figure 6-50

Begin WinGeo3 Example

Once the user has thoroughly familiarised him/herself with the various menus used to change the basic suspension kinematics, the next step is to develop iterative reports geared towards specific changes.

Table: **Show** – Going to the *Table* menu and then clicking on *show* brings up a data table at the bottom of the main graphics screen. The user can choose the type of file to be utilized by clicking on any of the built in tables or using the non demo version of WinGeo3. He/she can either redesign one of the existing files and/or create an entirely new table. Each table is capable of holding over one thousand entry lines.

- To clear the file – *Table /Clear table*

- With the non demo Advanced version, the best way to look at the file is to save it in a CSV (ASCII comma separated) format and then reopen it in Excel. Once in the Excel format, the data can be used to further enhance many of the current simulation programs (The demo version does not allow the user to save the file). Win 6-12 shows a right front suspension file using the data from the *Reports / Load Path file – (Short Track Ten Phase front*) iteration. The WinGeo3 Premier version allows the race engineer to down load track data from the acquisition system. This, in turn, not only allows the team to look at on track kinematics in minute detail but also allows the team to use the resulting Excel files to power simulation programs as introduced in the Dynamics chapters. The fact that WinGeo3 data can be converted to spread sheet files provides the race engineer with a whole new set of tools. *I love this program*.

Part 1 **Short track Ten Phase Clearance front**

Table 17-A	Ride	Roll	Steer	Camber R	Steer R	NetSteer	Ackerman	Scrub R
B=1.00g T=0.00g 1	0.073	0.031	0	-3.389	-0.015	0.012	-0.025	-0.006
B=1.25g T=0.50g 2	0.099	0.431	0	-3.288	0.039	0.14	0.021	0.016
B=0.60g T=1.00g 3	0.073	0.834	0	-3.271	0.102	0.262	0.048	0.044
B=0.15g T=1.25g 4	0.055	1.053	0	-3.272	0.134	0.327	0.062	0.06
B=0.05g T=1.53g 5	0.073	1.287	0	-3.22	0.16	0.397	0.075	0.073
A=0.00g T=1.37g 6	0.053	1.159	0	-3.263	0.148	0.358	0.068	0.067
A=0.30g T=1.00g 7	0.082	0.866	0	-3.254	0.104	0.272	0.05	0.045
A=0.40g T=0.70g 8	0.086	0.611	0	-3.281	0.068	0.195	0.034	0.028
A=0.65g T=0.50g 9	0.127	0.44	0	-3.244	0.033	0.144	0.019	0.013
A=0.85g T=0.00g 10	0.165	-0.012	0	-3.266	-0.047	0	4.796	-0.018

Part 2 **Short track Ten Phase Clearance front**

NetScrub	IC lng R	RC Height	RC Width	RC moment	Spring R	Motion R	Clear1 R
-0.012	40.358	2.686	7.019	15.386	0.025	0.446	4.568
-0.005	39.4	2.584	8.775	15.492	-0.067	0.445	4.398
0.012	38.251	2.452	10.524	15.575	-0.181	0.443	4.175
0.022	37.637	2.375	11.495	15.621	-0.245	0.443	4.05
0.028	37.149	2.302	12.548	15.698	-0.297	0.442	3.953
0.026	37.372	2.339	11.973	15.65	-0.273	0.442	3.996
0.011	38.209	2.445	10.669	15.588	-0.185	0.443	4.168
0.002	38.873	2.526	9.549	15.527	-0.118	0.444	4.297
-0.01	39.506	2.591	8.841	15.513	-0.056	0.445	4.422
-0.029	40.923	2.729	6.966	15.437	0.077	0.446	4.681

Win 6-12

- **Modifying the *Reports / Path files*** – The WinGeo3 demo version allows the user to modify and save *Path files*.

    - *Reports / Edit*: - To modify the *Short Track Ten Phase front* Path file. Once that file is loaded click on *Edit*. Selecting *Edit* will open the path file so it can be edited – Win 6-13. Make sure the screen is maximized.

Ride	Roll	Steer	Pitch	Left	Right	Option	Name
0.000	0.000	0.000	0.000	4.350	4.568	14 Clear	B=1.00g T=0.00g 1
0.000	0.000	0.000	0.000	4.571	4.398	14 Clear	B=1.25g T=0.50g 2
0.000	0.000	0.000	0.000	4.742	4.175	14 Clear	B=0.60g T=1.00g 3
0.000	0.000	0.000	0.000	4.831	4.050	14 Clear	B=0.15g T=1.25g 4
0.000	0.000	0.000	0.000	4.963	3.953	14 Clear	B=0.05g T=1.53g 5
0.000	0.000	0.000	0.000	4.881	3.996	14 Clear	A=0.00g T=1.37g 6
0.000	0.000	0.000	0.000	4.766	4.168	14 Clear	A=0.30g T=1.00g 7
0.000	0.000	0.000	0.000	4.646	4.297	14 Clear	A=0.40g T=0.70g 8
0.000	0.000	0.000	0.000	4.604	4.422	14 Clear	A=0.65g T=0.50g 9
0.000	0.000	0.000	0.000	4.421	4.681	14 Clear	A=0.85g T=0.00g 10
0.000	0.000	0.000	0.000	4.262	4.510	14 Clear	B=0.00g T=0.00g 0

4 Clearance

Win 6-13

- Data can be entered into the *Path* file in one of seven ways. Select the type of data to be entered through the box located at the bottom left of the screen. The chosen option will then be Option column. With the current *Path* file, the chassis kinematic motion is controlled by clearance/ride change. The far right hand column is used to name each particular calculation.

Path file Options

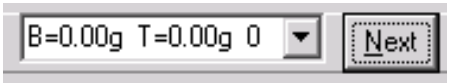
- Ride and Roll
- Spring motion
- Shock motion
- Clearance
- Data potentiometer motion
- Load
- Wheel travel

- The user can now make the required changes. In this case, as shown in Win 6-14, various left turn steer iterations were added. Once the changes have been made the user goes *to Save As*, changes to name of the file to reflect the changes and then saves the file. Check to see that steer is set at *Angle* rather than *Toe Span* (*Utility / Toe Steer Measurement / Degrees*).

Ride	Roll	Steer	Pitch	Left	Right	Option	Name
0.000	0.000	0.000	0.000	4.350	4.568	14 Clear	B=1.00g T=0.00g 1
0.000	0.000	0.500	0.000	4.571	4.398	14 Clear	B=1.25g T=0.50g 2
0.000	0.000	1.000	0.000	4.742	4.175	14 Clear	B=0.60g T=1.00g 3
0.000	0.000	2.000	0.000	4.831	4.050	14 Clear	B=0.15g T=1.25g 4
0.000	0.000	3.000	0.000	4.963	3.953	14 Clear	B=0.05g T=1.53g 5
0.000	0.000	2.300	0.000	4.881	3.996	14 Clear	A=0.00g T=1.37g 6
0.000	0.000	1.000	0.000	4.766	4.168	14 Clear	A=0.30g T=1.00g 7
0.000	0.000	0.700	0.000	4.646	4.297	14 Clear	A=0.40g T=0.70g 8
0.000	0.000	0.500	0.000	4.604	4.422	14 Clear	A=0.65g T=0.50g 9
0.000	0.000	0.000	0.000	4.421	4.681	14 Clear	A=0.85g T=0.00g 10
0.000	0.000	0.000	0.000	4.262	4.510	14 Clear	B=0.00g T=0.00g 0

Win 6-14

- Once the new report is loaded, click on *Next* until the zero iteration appears in the iteration box to the left of the *Next* button – Win 6-15.



Win 6-15

- **Table**: Show – Reopen and/or clear the *Table*.
- Using the *Next* button, scroll through the ten cornering phases.
- Save the new *Table* to the appropriate CSV file and then reopen the file in Excel – Win 6-16.

Part 1 **Short track Ten Phase Clearance front/Steer**

Table #1	Ride	Roll	Steer	Camber R	Steer R	NetSteer	Ackerman	Scrub R
B=1.00g T=0.00g 1	0.073	0.031	0	-3.389	-0.015	0.012	-0.025	-0.006
B=1.25g T=0.50g 2	0.099	0.431	-0.05	-3.321	0.526	0.134	-1.351	0.027
B=0.60g T=1.00g 3	0.073	0.834	-0.1	-3.339	1.072	0.245	-3.139	0.065
B=0.15g T=1.25g 4	0.055	1.053	-0.25	-3.436	2.545	0.261	-18.8	0.108
B=0.05g T=1.53g 5	0.073	1.287	-0.35	-3.445	3.52	0.282	-33.989	0.135
A=0.00g T=1.37g 6	0.053	1.159	-0.25	-3.428	2.557	0.29	-16.91	0.115
A=0.30g T=1.00g 7	0.082	0.866	-0.16	-3.36	1.653	0.24	-8.29	0.078
A=0.40g T=0.70g 8	0.086	0.611	-0.07	-3.328	0.749	0.185	-1.989	0.044
A=0.65g T=0.50g 9	0.127	0.44	-0.05	-3.278	0.52	0.138	-1.271	0.024
A=0.85g T=0.00g 10	0.165	-0.012	0	-3.266	-0.047	0	4.796	-0.018

Part 2 **Short track Ten Phase Clearance front/Steer**

NetScrub	IC lng R	RC Height	RC Width	RC moment	Spring R	Motion R	Clear1 R
-0.012	40.358	2.686	7.019	15.386	0.025	0.446	4.568
0.003	39.439	2.584	8.769	15.492	-0.064	0.444	4.398
0.028	38.326	2.452	10.513	15.575	-0.176	0.443	4.175
0.052	37.812	2.375	11.465	15.622	-0.232	0.441	4.05
0.062	37.384	2.302	12.509	15.699	-0.279	0.44	3.953
0.057	37.547	2.338	11.944	15.65	-0.26	0.441	3.996
0.034	38.326	2.445	10.651	15.589	-0.177	0.442	4.168
0.014	38.926	2.526	9.541	15.527	-0.115	0.444	4.297
-0.002	39.545	2.591	8.835	15.513	-0.054	0.444	4.422
-0.029	40.923	2.729	6.966	15.437	0.077	0.446	4.681

Win 6-16

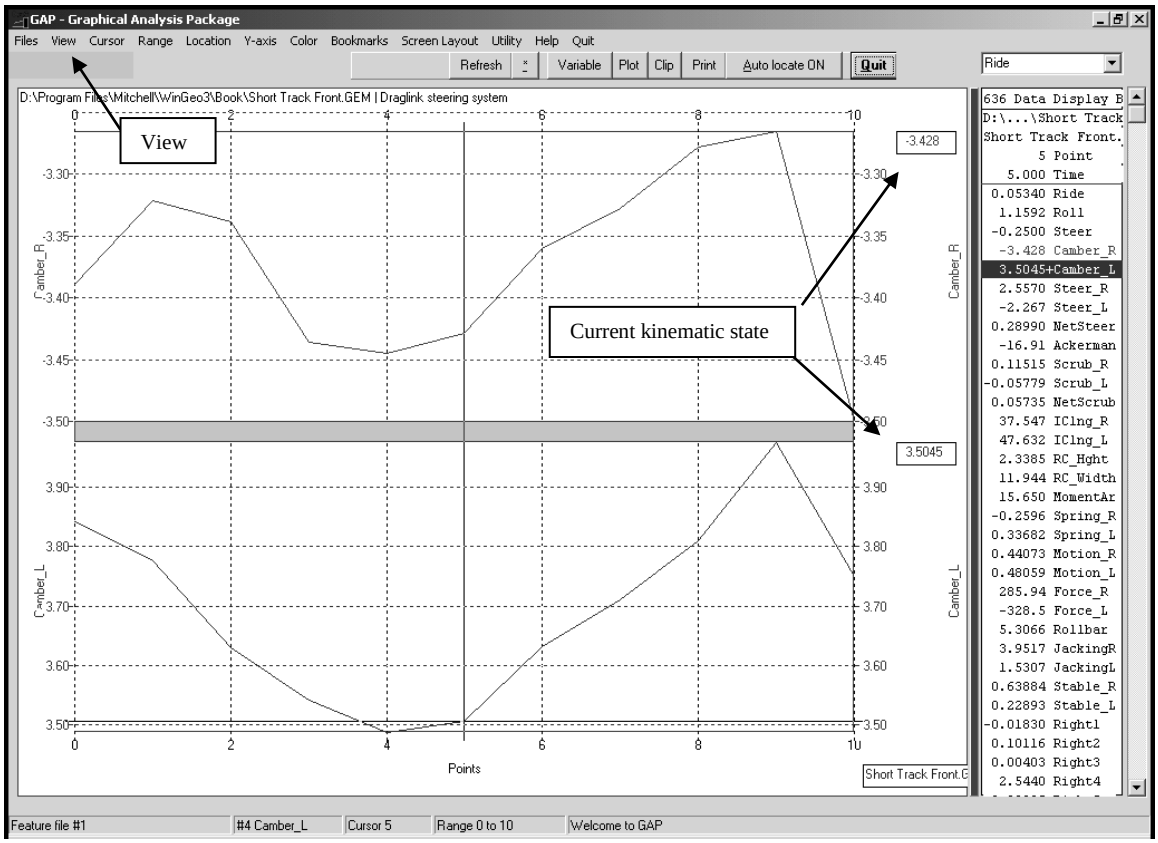
- The user can now compare the various components of the two files (without steer/with steer) such as camber change – Win 6-17.

Phase	No Steer	With Steer
	Camber R	Camber R
1	-3.389	-3.389
2	-3.288	-3.321
3	-3.271	-3.339
4	-3.272	-3.436
5	-3.22	-3.445
6	-3.263	-3.428
7	-3.254	-3.36
8	-3.281	-3.328
9	-3.244	-3.278
10	-3.266	-3.266

Win 6-17

Reports: GAP Analysis – The *Gap Analysis* (graphic analysis package) allows the user to look at various kinematic components over a series of iterations.

- **Reports**: Clear GAP array
- **Reports**: Load an appropriate *Path file*.
- **Using** the *Next* button on the main graphics screen, scroll through the *Path File* iterations.
- **Reports**: GAP Analysis – Clicking on the *GAP Analysis* menu opens the graphic analysis package (maximum screen size) – Win 6-18. Open the *View* menu and click *on Graph by Points* (even if the *Graph by Points* option is already selected).
- The user can now select the components at which he/she wants to use from the *Data Display* menu located to the left of the graphics screen. Clicking on *Camber R* will show the right side camber motion as a result of the *Path file* iterations. Win 6-18 has the right and left side camber selected.
- The user can scroll through the various iterations by using the left mouse button, using the arrow keys or by going to the *GAP Utility* menu and clicking on *Animation*. The animation delay can also be adjusted in the *GAP utility* menu.



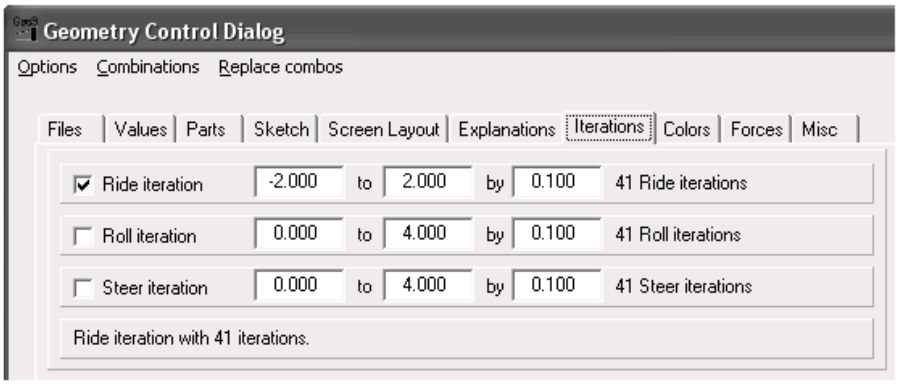
Win 6-18

- The *Data Display* area, far left hand side, shows the current kinematic state of the 69 kinematic components. Win 6-19 illustrates a small example of these *Data Display* kinematic components. The box(es) just to the right of the graphic screens, as illustrated in Win 6-18, also show the current kinematic state of the component entered on the graphics screen.

```
 0.05340 Ride
  1.1592 Roll
-0.2500 Steer
  -3.428 Camber_R
  3.5045 Camber_L
 2.5570+Steer_R
  -2.267 Steer_L
 0.28990 NetSteer
  -16.91 Ackerman
 0.11515 Scrub_R
-0.05779 Scrub_L
 0.05735 NetScrub
  37.547 IClng_R
  47.632 IClng_L
  2.3385 RC_Hght
  11.944 RC_Width
```

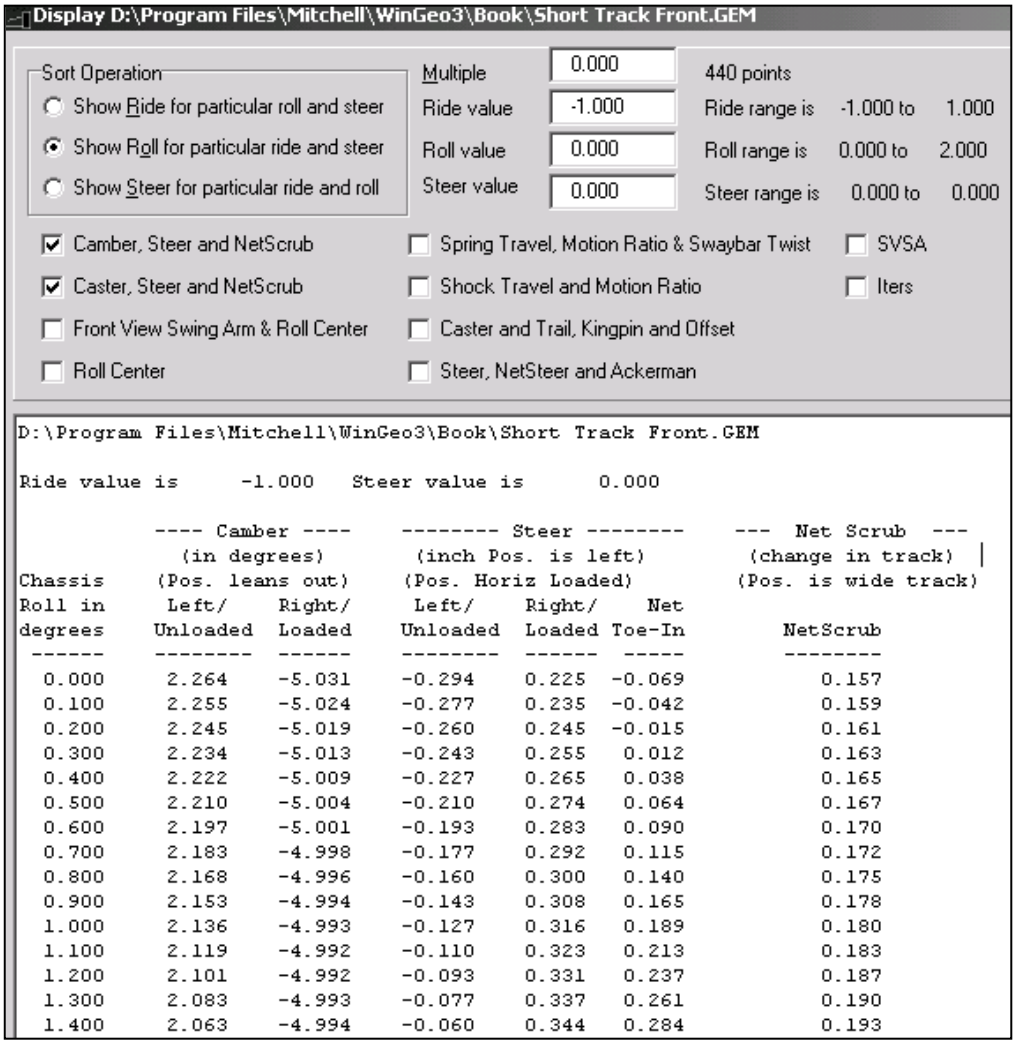
Win 6-19

Compute: **Iteration** – This element of the program allows the software to run through a series of user defined iterations such as *Ride iteration, Roll iteration* etc. When the user selects an iteration option, a Geometry Control Dialogue menu comes up which asks the user to select the number and increment of the iteration(s) – Win 6-20. Single calculation types such as *Ride, Roll* and *Steer* can be analyzed using *GAP* while all of the iteration types, including the combined iterations, can be illustrated in *Report* format.



Win 6-20

- Once the iterations have been run, a series of prompts will be shown.
  - *Display Carpet Plots* (This option appears only when combined iterations have been run such *as Ride and Roll Iteration*s) – The carpet plot option is not used in Book One.
  - *Produce* Reports – This option allows the user to look at a myriad of different iteration combinations as shown in Win 6-21. The report data is displayed only after the user clicks the *Report* button found on the upper left side of the *Reports* screen. The reports can be printed but not saved. Depending on the options selected, these reports can become quite lengthy so the user needs to choose the options prudently.
  - *GAP Analysis – GAP Analysis* only works for a single iteration run such as *Ride, Roll* or *Steer*.



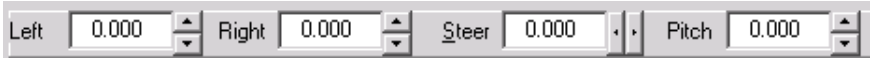
Win 6-21

Compute: Solve For – The user can manipulate the suspension kinematic motion *through Reports / Path files* or by working with the **main graphics screen** in any of six different ways.

- Ride, Roll and Steer – This is the programs base preset. As previously discussed, each time the program is loaded the main graphics screen is configured to allow the user to change the *Ride* (equal motion at both wheels), *Roll*, *Steer* and/or *Pitch*.



- Compute: Wheel Travel – This option allows the user to enter left and right *Wheel Travel*, *Steer* and/or *Pitch.*



- Compute: Solve for Spring – This option is the same as *Wheel travel* except it controls the L & R *Spring* motion.

- Compute: Solve for Shock - This option is the same as *Wheel travel* except it controls the L & R *Shock* motion.

- Compute: Solve for Data Pot - This option is the same as *Wheel travel* except it controls the L & R *Data Pot*entiometer motion.

- Compute: Solve for Clearance - This option is the same as *Wheel travel* except it controls the L & R *Clearance*/ride motion.

- To **return** the main graphics screen to the Base Ride, Roll, Steer and Pitch option, click on the **Zero** button. This will return the internal workings of the program to the base setting but it does not always change the *Left* and *Right* designation back to *Ride* and *Roll*. Clicking on *Next* once and then *Zero* will also change the designation.

*****End WinGeo3 Example*****

Scrub - On the surface, scrub is a very simple phenomenon. In basic terms, it is the scrubbing of the tire laterally due to suspension's geometric restraints during ride and roll iterations**.**

When looked at using three dimensional kinematic software, scrub is shown as lateral motion. For example, when a wheel moves in bump past the center of its arc, the tire will be moved in towards the chassis. The amount of scrub can vary anywhere from only a few thousandths to the extreme of over an inch (I have seen it). At first glance, if the amount the tire is scrubbed is only in the range of 0.040, it hardly seems like anything worth worrying about until you review how much a tread particle is able to cantilever from its at rest position. Combine that with the fact that the tire is being scrubbed back and forth several hundred times a second and one begins to understand how much scrub can **steer** the chassis. The following table shows data from a right front tire measured every twentieth of a second. It shows on track data (lateral displacement from chassis centerline) from a competitive stock car running in a major series - Table 6-6.

Table 6-6

Each iteration represents one twentieth of a second. The car was traveling at approximately 126 mph								
-0.206	-0.221	-0.234	-0.269	-0.307	-0.324	-0.368	-0.352	-0.337

Bump Scrub - The horizontal scrubbing of the tire contact patch as the result of kinematic suspension movement in bump or droop is referred to as bump scrub - Figure 6-51. The lateral movement of the tires scrubs off speed, uses up engine power to overcome the related drag, which, in turn, tends to heat the tires. Most of all, tire scrub tends to steer the tire (chassis).

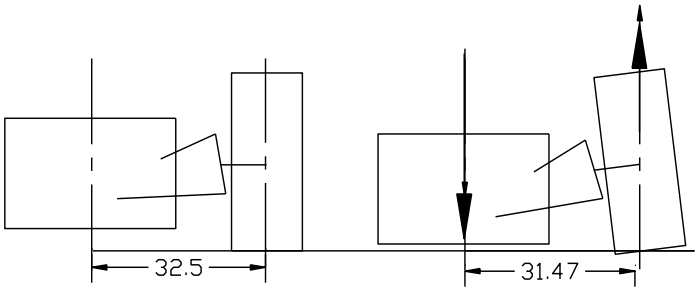


Figure 6-51

Lateral outside front tire displacement which widens the outside wheelbase in relation to the chassis center line tends to loosen (free) the car up (oversteer) while lateral tire displacement which narrows the outside wheelbase tightens up the car (understeer). *It's not that simple.*

"Scrub must be defined relative to the chassis' rotation point. WinGeo3 (as a default) rotates the chassis about ground level. The choice of rotation point impacts scrub because it moves the entire chassis laterally and scrub is measured as an absolute location. Raise the rotation point and the chassis moves less under roll and the calculated scrub will change.

Thinking of scrub as dragging the tire across the pavement brings to mind terrible forces. It would be easy to degrade a tire by dragging it laterally across rough pavement. But the tire is rotating at high speed as it moves a fraction of an inch in scrub. For a moving race car, scrub really represents a steering input, not a shearing force. A car traveling 60 mph goes 88 feet per second, or 1056 inches per second. If the tire scrubs 1 inch in a second, that represents a lateral movement of 1 inch with a forward movement of 1056 inches. This is a small angle." Wm. C. Mitchell

Roll Scrub - Roll scrub is the same as bump scrub except it is the result of body roll. It is measured as lateral movement of the tires versus angle of body roll - Figures 6-52 & 6-53.

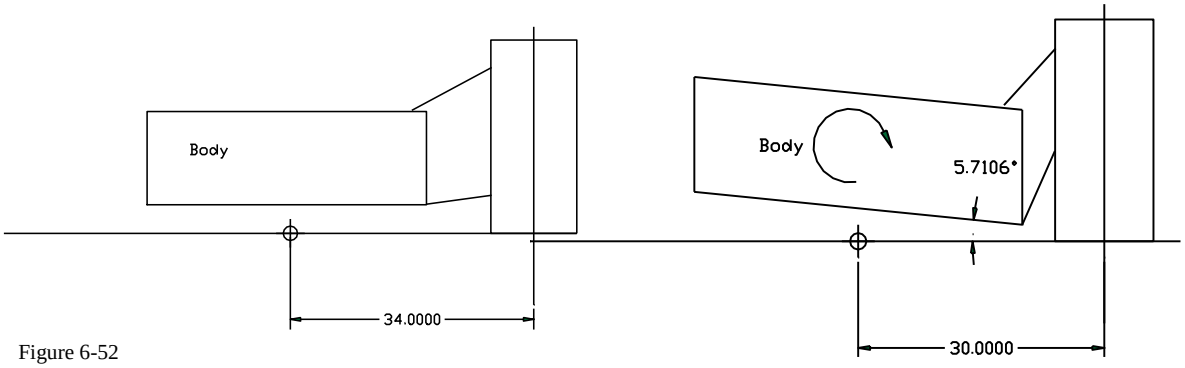


Figure 6-52

Figure 6-53

Net Scrub - The total change in track when considering both the left and right side tires is called net scrub and is normally a combination of both bump and roll scrub. The chassis as viewed in Figure 6-54 shows both the left and right tires scrubbing in towards the center of the chassis with the right side tire moving more than the left side tire.

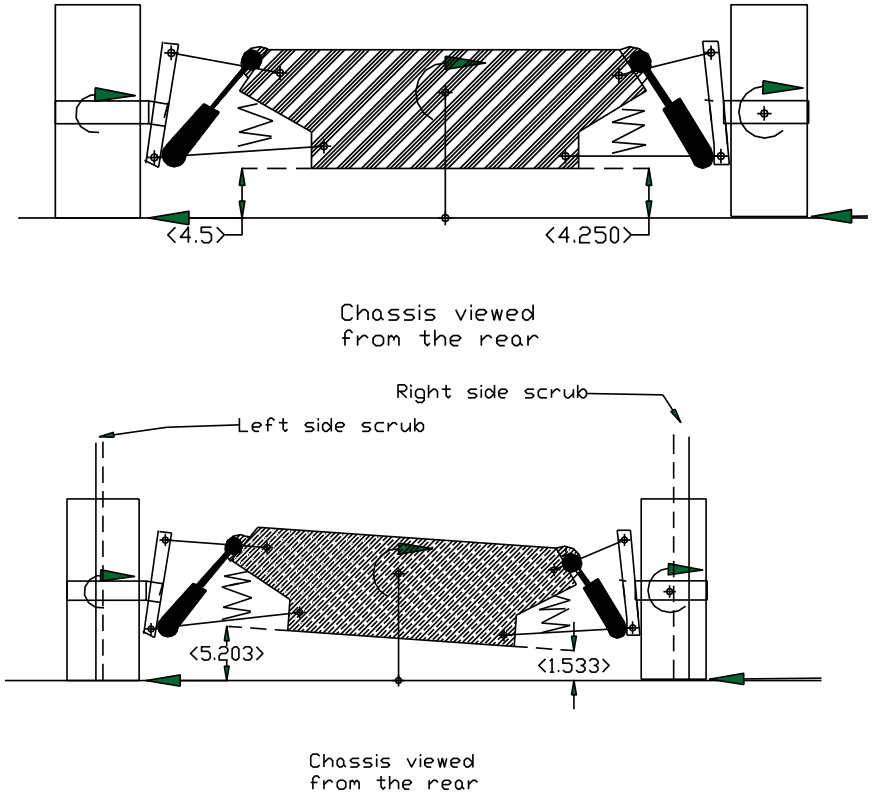


Figure 6-54

Relative Lateral Movement Versus Scrub -The term relative lateral movement is used because it is not necessarily only the tires that are moved as the result of scrub geometry. In most cases, it will be a combination of both tire(s) and chassis movement i.e. if the right side tire contact patch is generating sufficient lateral load to maintain its relative position, either the chassis and/or the inside tire will be moved (scrubbed) towards the outside tire, or, if the inside tire is also able to maintain its relative position the chassis alone (sprung mass) will move vertically, horizontally and/or change its roll angle. At this point, we should reiterate that scrub only takes place as the result of either suspension motion or sprung mass motion. So when we state that the chassis position is changed due to scrub, we are describing the combined chassis motion that produced the scrub (first order) and the secondary chassis motion resulting from scrub (second order). It is a closed loop. Everything is interdependent.

The Bottom Lines of Scrub

- When a tire is scrubbed it tends to **steer** the chassis.

- Changes in the sprung mass co-ordinates will change the chassis performance characteristics i.e. changing the jacking versus springing load transfer and changing the roll moment arm co-ordinates.

- The percentage of tire versus body motion holds considerable ramifications for the chassis' ultimate handling capabilities and will be looked at in some detail throughout the book.

Further Examples of Scrub

- During a turn, and, as the result of body roll, the outside tire may be moved laterally very little relative to its arc (the chassis moves laterally, not the tire) while the inside tire may be moved in towards the center of the chassis to a much greater extent (tire moves laterally rather than the chassis). This often leads to hunting which is the result of the tires and chassis trying to reach a state of equilibrium.
- During right side **one** wheel bump, if the chassis was held stationary, the right wheel would move in towards the chassis considerably more than the left side wheel. However, if the right wheel was more heavily laden than the left, it would tend to follow its projected path while pulling both the chassis and left wheel in towards itself. *Key tuning tool*.

- Using new sticky tires at the beginning of the run, both the inside and outside tires may scrub very little which results in the sprung mass reacting to the scrub geometry. As the tires begin to lose some of their lateral force capacity, the inside tire may begin to scrub in addition to a proportion of sprung mass scrub induced motion. Near the end of the run, as the tires begin to lose their lateral force capacity, both inside and outside tires will begin to scrub more, and, as a result, the sprung mass will show less scrub induced motion. *You know it's not that simple*.

Even though there are an endless number of possibilities, the good news is that the race engineer does have some control over what happens and when it happens. However, before the engineer can think about controlling these sequences, he or she must first understand what is happening and be able to recognize why. To that end, building and studying a model will help a great deal. As always, the model does not have to be a work of art. It just has to be functionally representative. The pictures in Figures 6-55 /6-58 show a very simple model that took me less than half an hour to build. The "tires" could be locked in place with locking pliers or restrained in extension or compression using a variety of springs locked in place over large washers welded to the tire (the "tire" was represented by a door hinge which allowed it to move up and down).

****WinGeo3 Examples**** - WinGeo3 shows the amount of resultant scrub at each wheel due to kinematic motion. Using all of the processes as previously discussed in Chapters 5 and 6, the race engineer is able to ascertain how kinematic design and motion affects the amounts of corner and combined scrub.

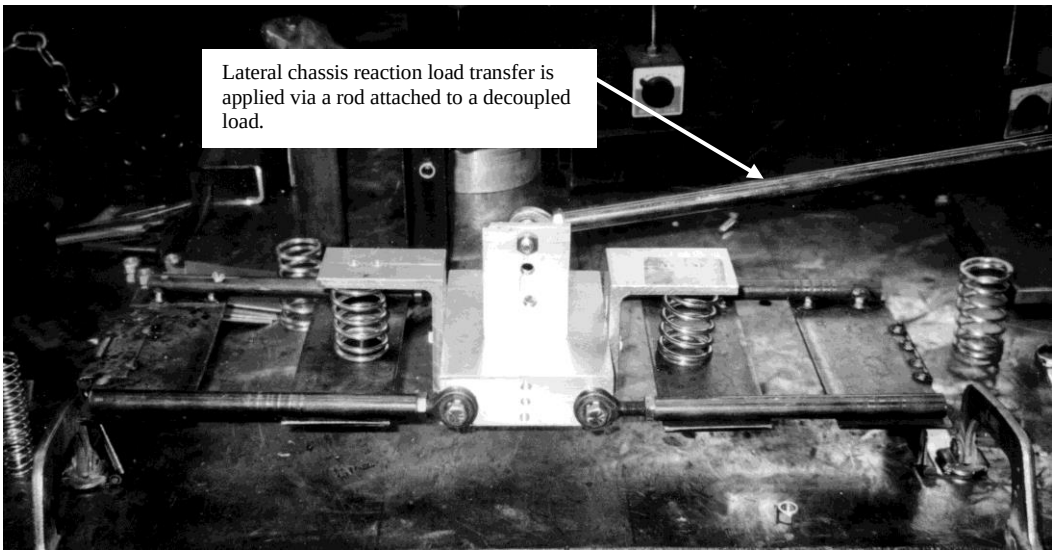


Figure 6-55

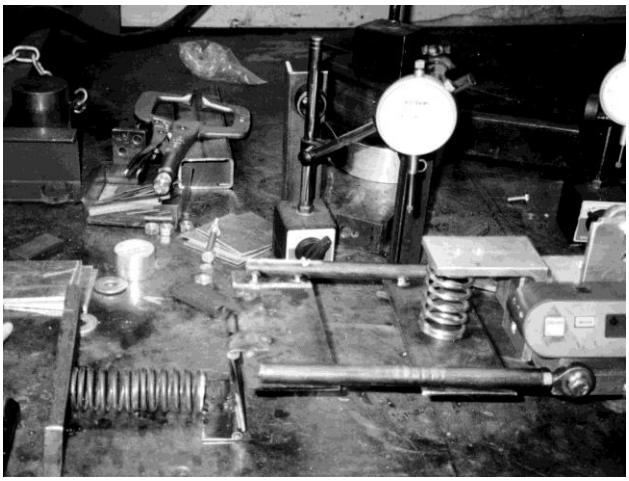


Figure 6-56

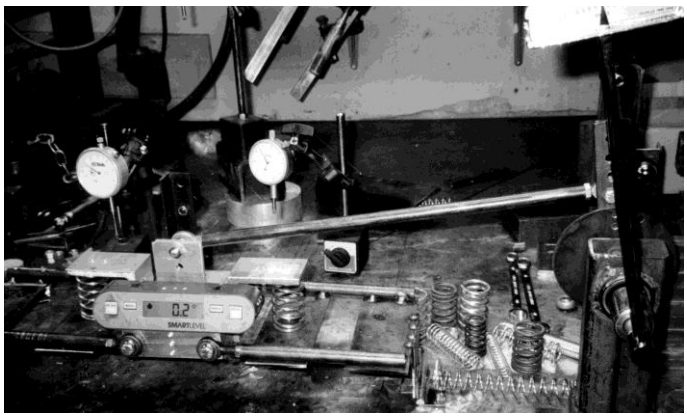


Figure 6-57

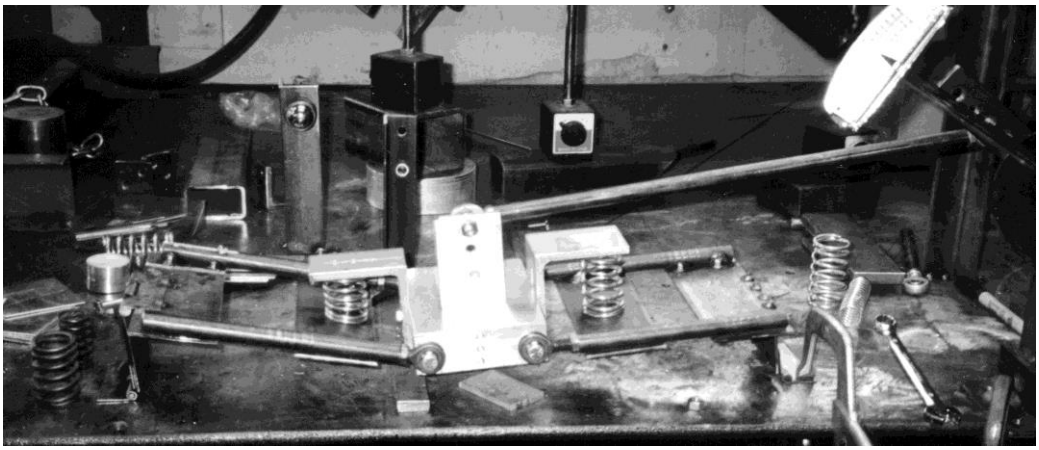


Figure 6-58

Instant Centers

Most of the basic concepts in this book are first discussed in two dimensional terms for the purpose of clarity. But, as always, never lose sight of the fact that all aspects of vehicle dynamics work in three dimensions. Any attempt to analyze kinematic motion in two dimensions, other than for conceptual purposes, would be wrong, period. Also remember that in a dynamic mode, each part of the suspension system is in a constant state of change, including the roll centers and instant centers.

Front View Instant Center - The front view instant center is an intersection point of two lines drawn from the upper A-Arm ball joint through its mount extended into space and a line drawn from the lower A-Arm ball joint through its inner mount extended into space. Where these two lines intersect is referred to as the instant center - Figures 6-59 & 6-60. It is called the **instant** center because, during the dynamic state, it is only at a particular set of co-ordinates for any given instant. Each time the suspension moves, the instant center position moves. Depending on the chassis design parameters, the left and right instant centers may not be symmetrical.

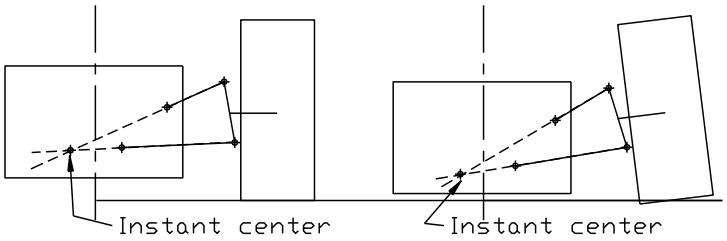


Figure 6-59

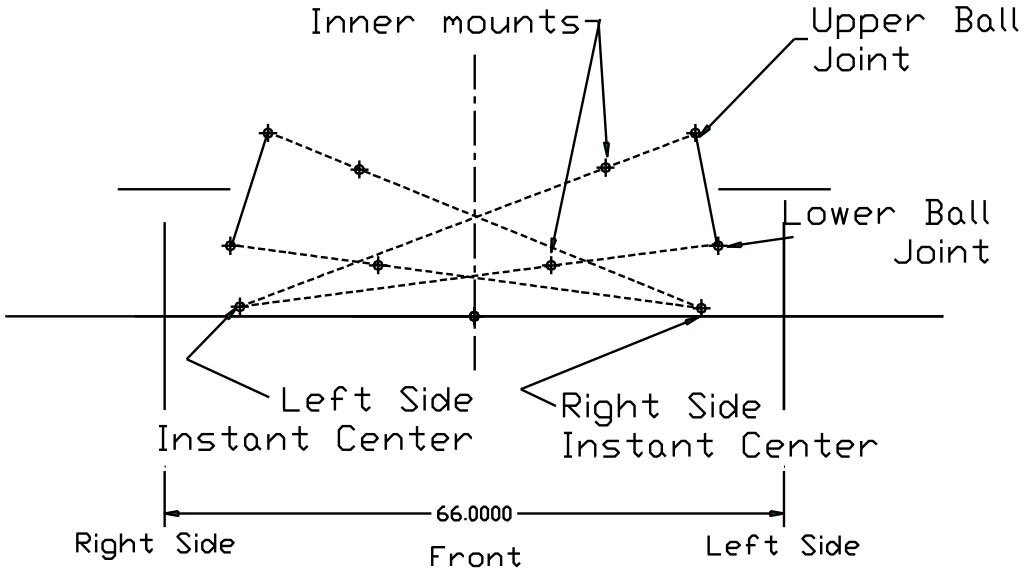


Figure 6-60

Important Note: **Front View Instant Center** – "The lines are not really drawn through the ball joints. They are drawn through the plane of the centerline of the tire contact patch. This makes a small difference because the ball joints may be forward or behind the centerline of the tire to produce caster. Simply looking at the inner mounting points of A arms can be misleading when the A arms are angled and the inner points are at different lateral points, i.e. stock car lower A arm. The **side view instant center** is found by projecting lines through the upper and lower A arms parallel to the centerline." Wm. C. Mitchell

Side View Instant Center - The side view instant center is found by projecting a line through the upper and lower inner suspension mounting points in the side view. Where these lines intersect is the side view instant center. If the upper and lower A-Arms' mounting points are parallel with each other, the lines would intersect at infinity, thus the IC is infinitely long. The side view instant center length and height indicates the percent of anti dive, anti lift, anti squat (rear), caster change rate (arc) and the wheel's fore/aft path present in the suspension geometry - Figure 6-61.

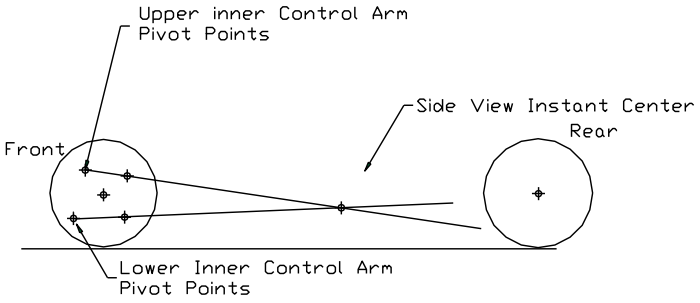


Figure 6-61

Again, remember the front and side view **instant centers** are a projected imaginary point, which, in effect, are the **pivot points** of the suspension linkage at that instant. A proper suspension design takes into account not only the static instant centers but also the movement of the dynamic instant centers and how fast and how much they move. The front view instant centers on many open wheel chassis may only move a few thousandths over the whole range of suspension motion. With the short A-Arms used on a stock car and the relatively large suspension motions, it is much more difficult to control the instant centers compared to the longer and less angled A-Arms used with formula car suspensions.

Instant center length and height determines the kinematic roll center movements during body roll. A longer instant center length results in the kinematic roll center moving up and down less radically during, for example, left turn body roll but also provides less negative camber gain at the right front. The up and down movement of the geometric (jacking) instant center during body roll can play a small to major role in the loading and unloading of the inside and outside chassis' corners during the various bits of the turn. As always, without good three dimensional kinematics software and a dynamic simulation program, it would be very difficult to find the optimum instant centers. As previously stated, the instant center location points play a major role in the various geometric jacking effects, which, in turn, have a considerable influence over how the lateral and longitudinal acceleration loads are transferred.

Note: The dynamics section of Book One covers the jacking and anti force calculations and simulation in considerable detail.

Swing Arms - If the suspension links were replaced with one imaginary link running from the instant center to the center of the spindle's axis of rotation, you have what is called a **swing arm**. It is simply another way of looking at how the instant center affects kinematic tire motion. Just as there are two instant centers, front view and side view, there are also two arms. Though not visible, the swing arm forces are very real. During a snapshot in time, the swing arm and its pivot can be used to replace the suspension system; however, as soon as the suspension kinematics are changed in the dynamic mode, the swing arm is also changed. The forces acting and reacting through the suspension system **pivot** at the instant center and follow an **arc** about the spindle center line. - Figure 6-62 & 6-63.

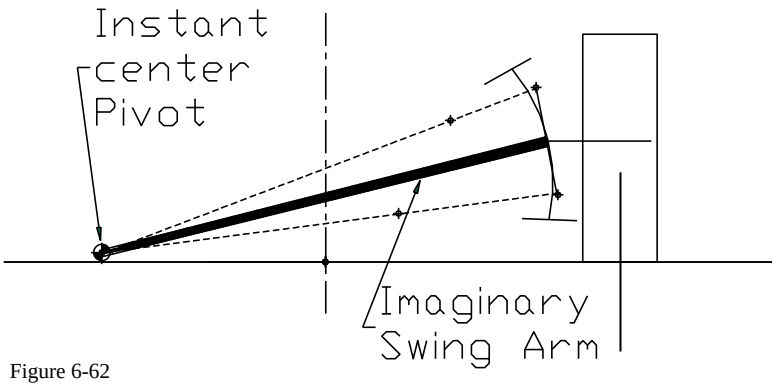


Figure 6-62

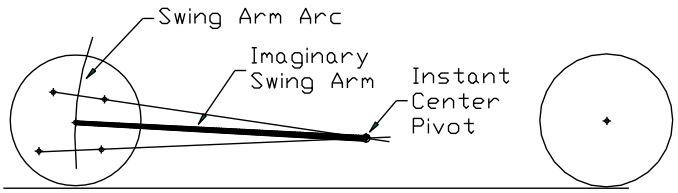


Figure 6-63

Instant Axis - When the instant centers are viewed in three dimensional space, the instant centers are replaced with an instant axis. This axis, for simpler equations, is found by projecting a line from the front view instant center back to the side view instant center - Figures 6-64 & 6-65. This instant axis is what defines the motion of the spindle/upright relative to the chassis. Because the spindle always operates in a three dimensional mode, three dimensional geometric analysis procedures must be used when determining what is actually happening within the chassis and suspension. **The importance of the instant axis cannot be overemphasized.** Everything moves about this instant axis. As seen later, the effect of the instant axis on the steering geometry as the chassis rolls can be quite dramatic. This is why traditional methods of measuring bump steer only tell half the story. Like every part of a suspension system, steering geometry is controlled by both bump and roll.

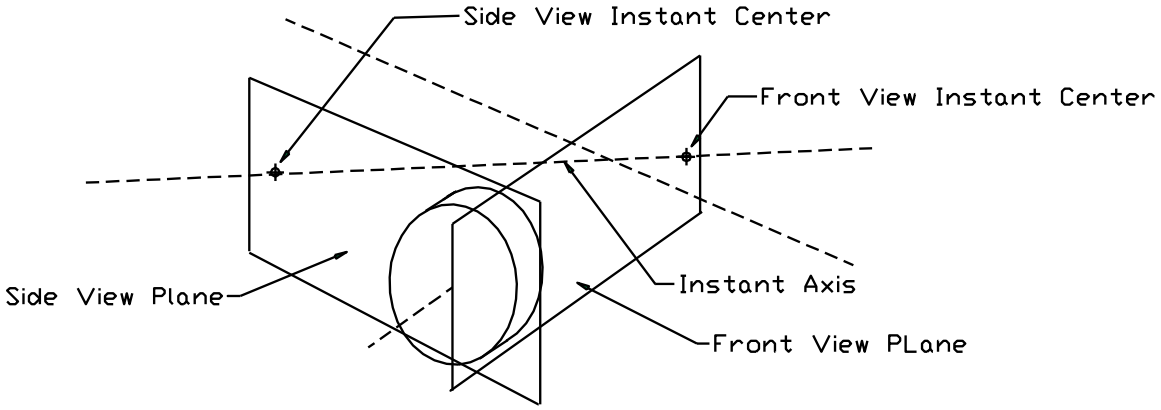


Figure 6-64

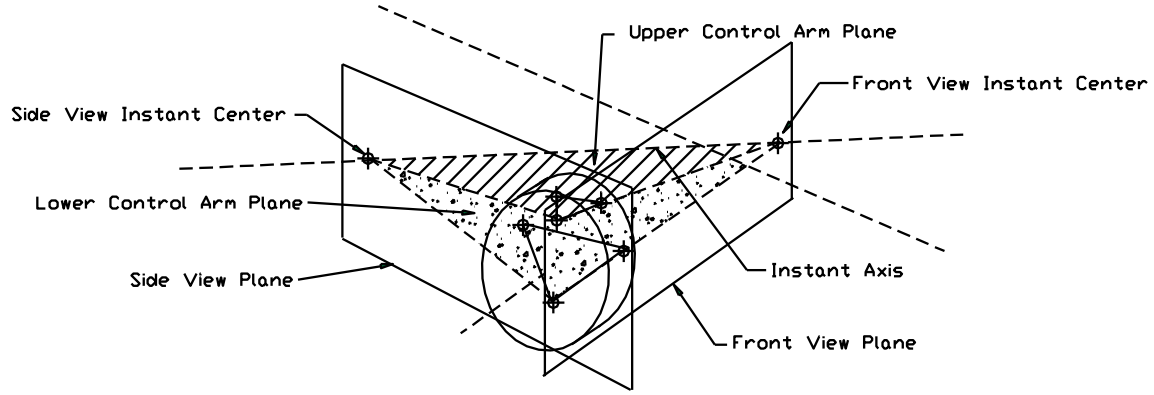


Figure 6-65

For a more in-depth analysis of three dimensional independent suspension, the planes will need to run in relation to the control arm arcs. For example, if the inner pivots of the lower control arm are mounted at an angle to the center line of the chassis, the side view plane will be angled accordingly i.e. the instant axis points will not be located on the tire's side and front view planes - Figure 6-66. As William C. Mitchell is fond of saying, "It's just triangles - a whole bunch of triangles."

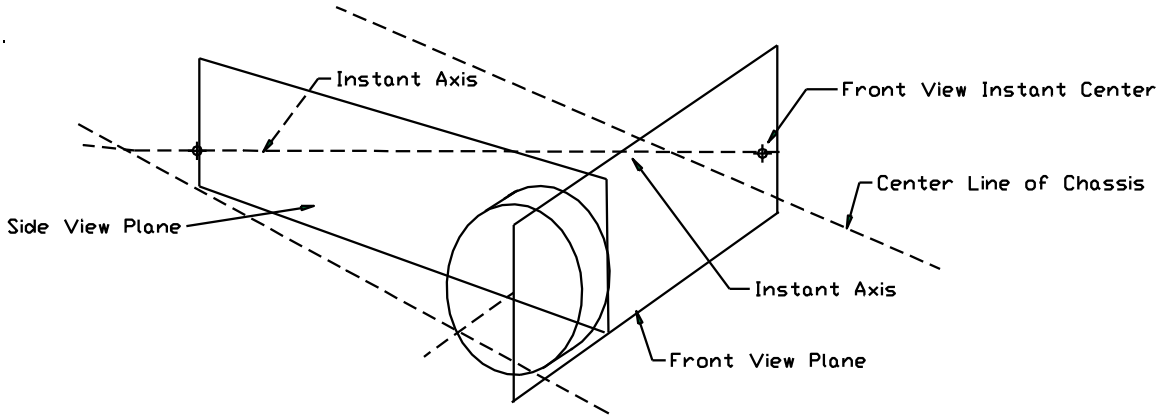
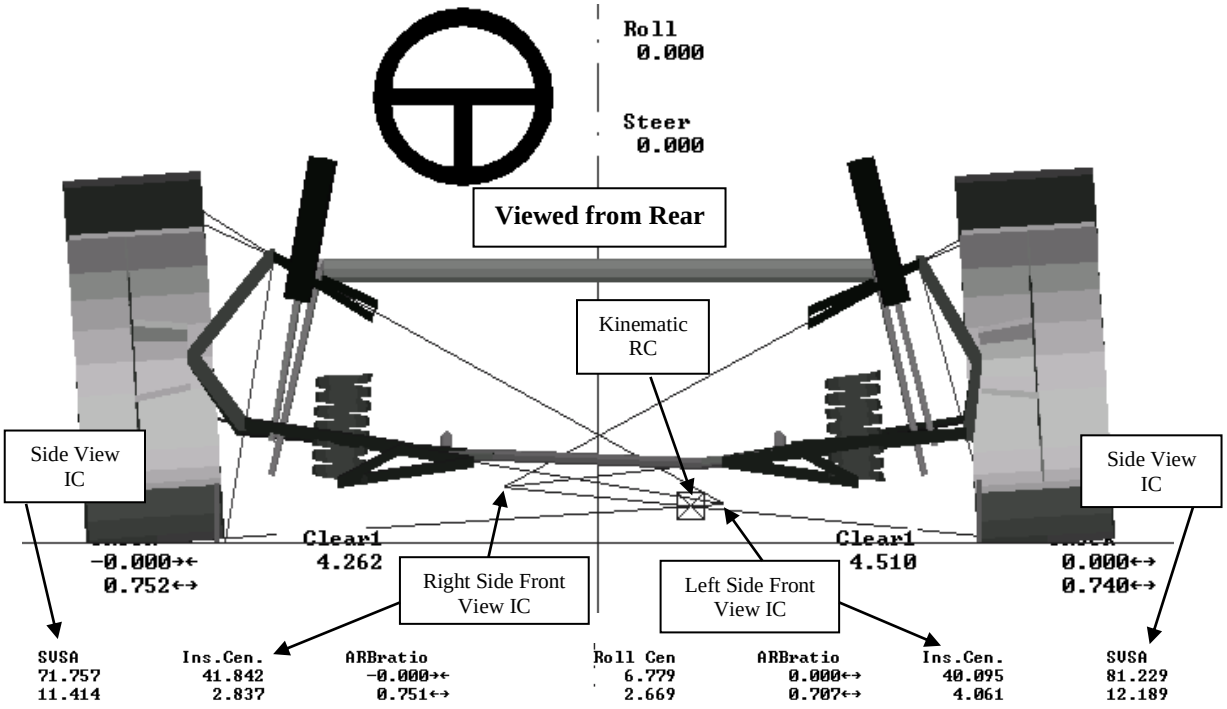


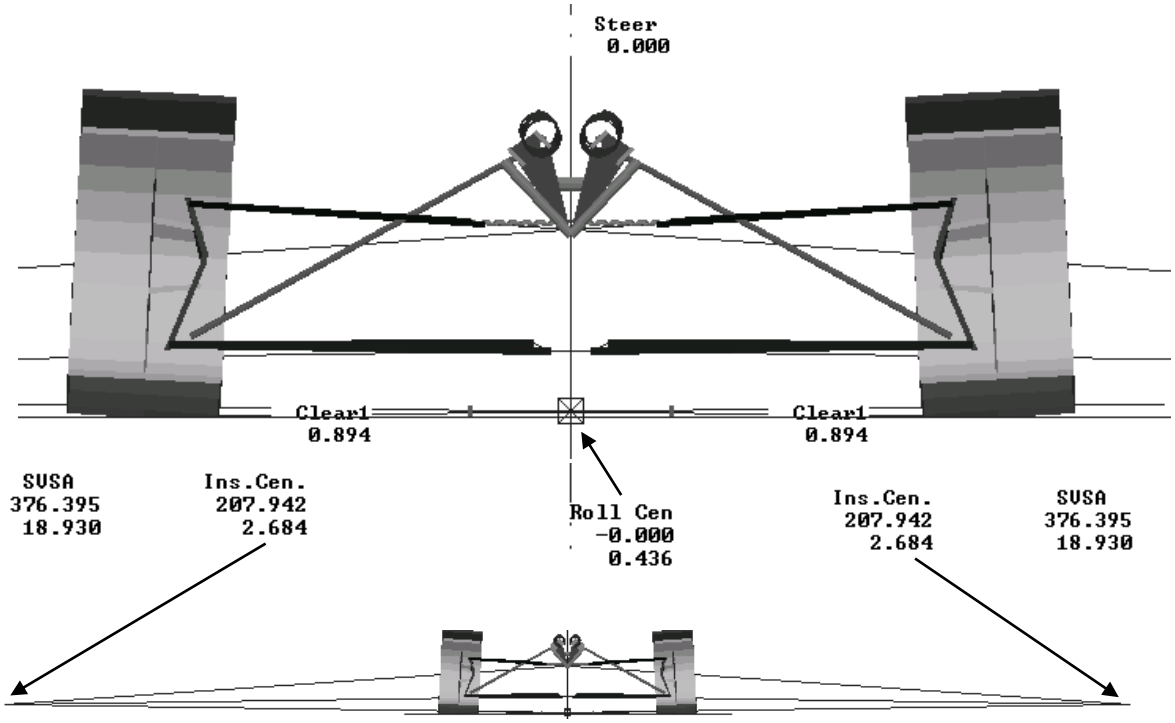
Figure 6-66

** Begin WinGeo3 Example**

Win 6-22 / 6-25 illustrate the differences between the kinematic roll centers, front view instant centers and side view centers of a stock car and an open wheel car. The front view and side view instant center co-ordinates are shown with the IC height above the IC length. The Dynamics chapters use the IC co-ordinates to calculate jacking and anti forces.



Win 6-22 Stock Car - Viewed from the rear



Win 6-23 Open Wheel – Viewed from the rear

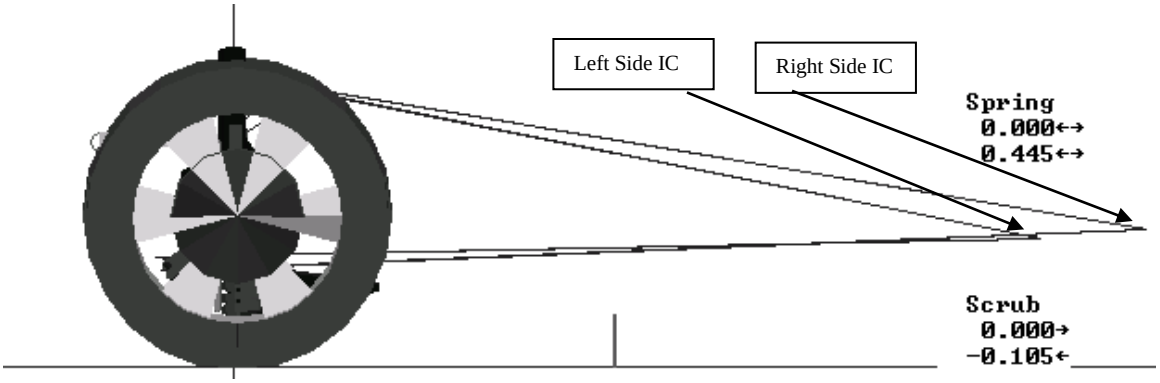
Notice the extreme differences between the front view IC positioning in Win 6-22 & 6-23 and the side view IC positioning in Win 6-24 & 6-25.

Kinematic Motion

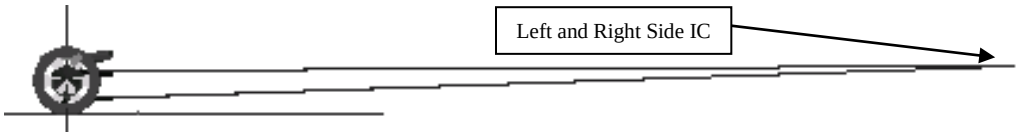
- The greater the IC length, the less it will move under equal amounts of chassis and/or suspension motion.
- The longer the suspension links, the less the IC will move for any given amount of chassis and/or suspension motion.
- The less angled the suspension links, at static ride height, the less the IC will move for any given amount of chassis and/or suspension motion.
- The greater the chassis and suspension motion, the more the IC co-ordinates will move.

Geometric (jacking/anti) Forces

- The shorter the IC length, for a given IC height, the greater the geometric (jacking/anti) force will be.
- The higher the IC, the greater the geometric (jacking/anti) force will be.



Win 6-24 Stock Car – Viewed from the rear outside



Win 6-25 Open Wheel Viewed from the side

Using all of the exploratory methods previously discussed in Chapters 5 and 6, the race engineer can look at any number of design or redesign options in order to try and control the amount of IC horizontal and vertical motion during a window of chassis and/or suspenion dynamic motions. To develop an overall view of IC motion throughout the dynamic kinematic window, the race engineer can develop *Path files* that will run the chassis and suspension through a wide variety of dynamic kinematic options. In the animation mode, the race engineer can watch the IC motion. Watching the extent and direction of kinematic roll center motion while running *Path files* will give a quick overview of how quiet or noisy the IC motions will be. It is important to run combinations of combined and single component iterative motion.

Because a stock car, compared to an open wheel chassis, generally sees much greater dynamic motion and has considerably shorter and more highly angle suspension links, it is virtually imposible to design a stock car suspension system that does not see a great deal of IC motion throughout its dynamic window. For this reason, stock car designers and race engineers spend considerable time trying to control the geometric (jacking/anti) forces to help chassis performance rather than trying to create a design that will mostly eliminate those forces.

Every chassis design and set up will have its own unique IC signature, and, as a result, teams use kinematic programs such as WinGeo3 to run thousands of small design and/or set up changes through the chassis' various dynamic windows. Being able to run on track data through the kinematic program is extremely helpful. It will be left to the reader to spend **inordinate** amounts of time working through a **wide variety** of **small** design and set up changes using the provided WinGeo3 examples. I cannot stress how **important** this process is in the development of the skills and knowledge needed by a race engineer.

** End WinGeo3 Example**

The Centers (Kinematic, Spring and Geometric)

How and why a chassis rolls, pitches and heaves is the fundamental basis of vehicle dynamics and though the various vehicle dynamics theories are discussed in subsequent chapters, we need to introduce several of the terms at this point.

Front and Rear View Kinematic Roll Center (KRC) - This is the point about which the sprung mass would roll if unaffected by outside forces. During dynamic conditions, this is primarily a theoretical point as there will always be forces involved in any form of lateral acceleration and these forces will have an impact on the KRC. The front view instant centers are generally considered far more important than the kinematic roll center. A symmetrical chassis while at rest will have its horizontal kinematic center co-ordinate at the front view chassis center line. However, as soon as the suspension components are moved in ride and/or roll, the horizontal co-ordinate will most often be offset to the left or right of the center line. For a well designed chassis with long control arms, the amount of offset may be very small while for a chassis using short control arms this horizontal offset may be considerable.

- **Front and Rear Springing (Elastic) Roll Center (SRC)** – This is the point about which the sprung mass would roll if only the springing forces (springs, bars and dampers) were involved. Springing forces are a combination of springs, anti roll bar and damper forces times their respective motion ratios squared (Wheel Rate). These forces are called elastic forces and result in a theoretical elastic roll center. Though the forces involved and how those forces affect the actual/datum roll center are quantifiable, I don't think it is possible to actually separate the dynamic roll center from the mix of other forces involved in lateral acceleration. Nor do I think it is necessary to do so. How these forces affect the actual point about which the chassis rolls is what the engineer needs to understand. **The elastic forces are one of the two primary components which control the ride and roll rates**.

- **Front View Geometric (Jacking/Anti) Center (FAP – Force application Point)** – This is the point about which the sprung mass is jacked as a result of the geometric jacking forces only. Jacking/Anti forces are the result of forces passing through the suspension geometry and are called **Anti Roll Forces**. They can apply either a positive or negative vertical force. The combination of all jacking forces result in a geometric **Force Application Point** or center (FAP). As will be seen later**, instant center placement** plays a considerable role in the resulting dynamic jacking/anti forces. Through anti roll forces are normally thought of as having only a vertical component they can and most often do have combined X, Y, and Z force vector elements. **The anti forces are the other primary component which control ride and roll rates. "**There is a **Force Application Point** for each tire. It is where the CP-IC line passes under the center of gravity." Wm. C. Mitchell

- **Datum Roll Center** – This is the point about which the chassis rolls when the springing (elastic) and geometric (jacking/anti) forces are combined. The datum roll center is the actual point about which the chassis rolls. The previous three centers, to a greater or lesser degree, affect the position of the datum roll center but at no time does the chassis actually roll about a pure kinematic, dynamic or geometric roll center. It is important to understand that though the chassis rolls about the datum roll center, it is, in fact, the springing (elastic), geometric (jacking/anti) forces and suspension kinematics that determine the chassis' load transfer dynamics.

Note: When discussing roll centers and roll axis throughout much of this book, it is assumed that the chassis is infinitely rigid and there is no compliance in the suspension parts. Of course this is seldom true but because of the complexities involved, chassis and suspension compliance factors will not be thoroughly covered until Book Three.

Looking at the Kinematic Roll Center (KRC)

Kinematic Roll Center - The kinematic roll center is an imaginary point in the front view plane around which the sprung mass would roll if unaffected by the elastic and geometric roll forces. It is most often shown as laying on the same plane as the front view instant centers. As with everything involved in the study of vehicle dynamics, the kinematic roll center operates in **three dimensional** space. However, it is still well worth the reader's time to draw out several two dimensional front view suspension systems in order to develop an understanding of the instant centers and the kinematic roll center.

The following example is a two dimensional, front view, independent suspension system. The instant centers are positioned at designer determined lateral and vertical co-ordinates. The spindle co-ordinates are also predetermined.

- Using spindle, ball joints, hub, rim and tire co-ordinates, locate the x, y, z pivot center co-ordinates of the lower and upper ball joints/axial attachment points measured from the center line of the tire's contact patch.

- Draw the transverse ground plane, vertical centerline of the chassis and the contact patch center line of the two tires.

- From the y and z pivot center co-ordinates determined in Point One, mark the initial placements for the lower and upper ball joints/axial attachment points as per your predetermined track - Figure 6-67.

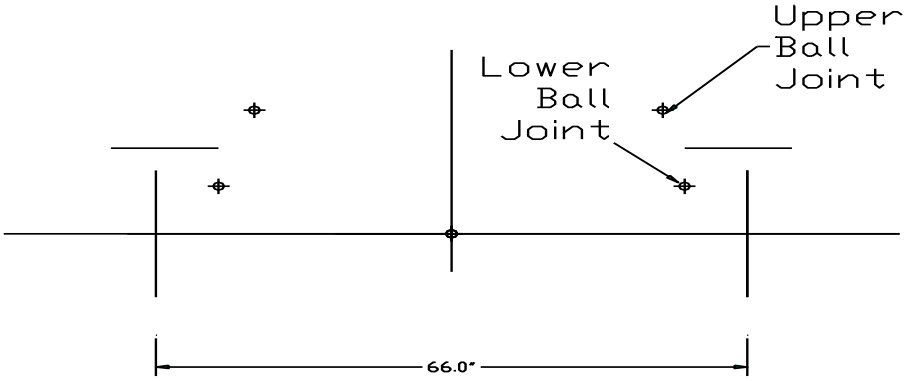


Figure 6-67

- **Instant Center Predetermined Placement** - Locate the predetermined instant center positions. For this example, the instant center lateral location is placed just inside the opposite side ball joint/axial attachment point i.e. the left side instant center lateral line is drawn just inside the right side ball joint/axial attachment point - Figure 6-68. Remember the instant center placements play a key role in the chassis' dynamic performance. Try a number of different instant center iterations.

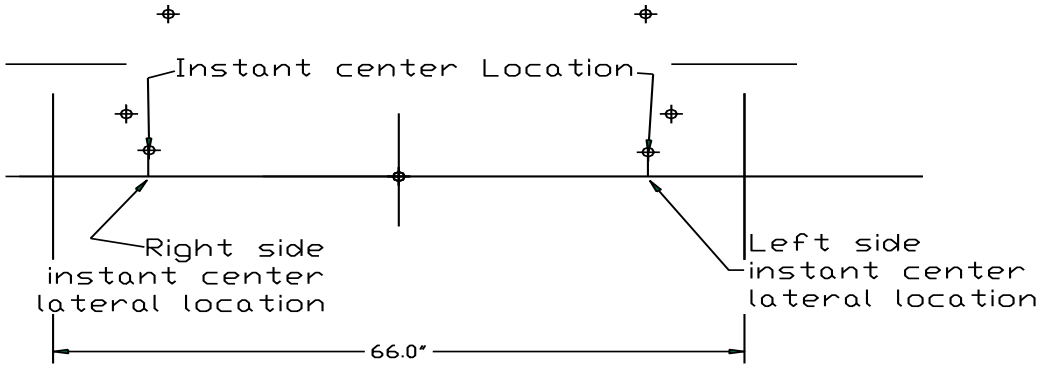


Figure 6-68

- Draw a line from the left front instant center through to the lower ball joint/axial attachment point. Do the same for the right side. Draw a line from the instant centers through to the upper ball joints - Figure 6-69.

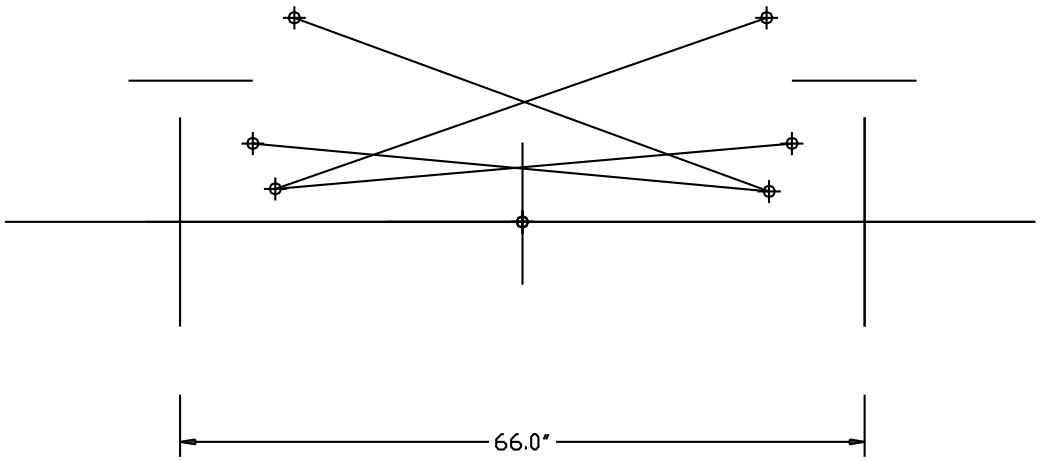


Figure 6-69

- The lines drawn from the front instant centers and the ball joints determine the planes on which the inner pivot points need to be placed. The inner pivot points and the A-Arm lengths are determined by the camber curve desired. Place the inner mounting points for the upper and lower wishbone/A-Arms along these lines. Shorter wishbones/A-Arms result in a faster camber curve – Figure 6-70.

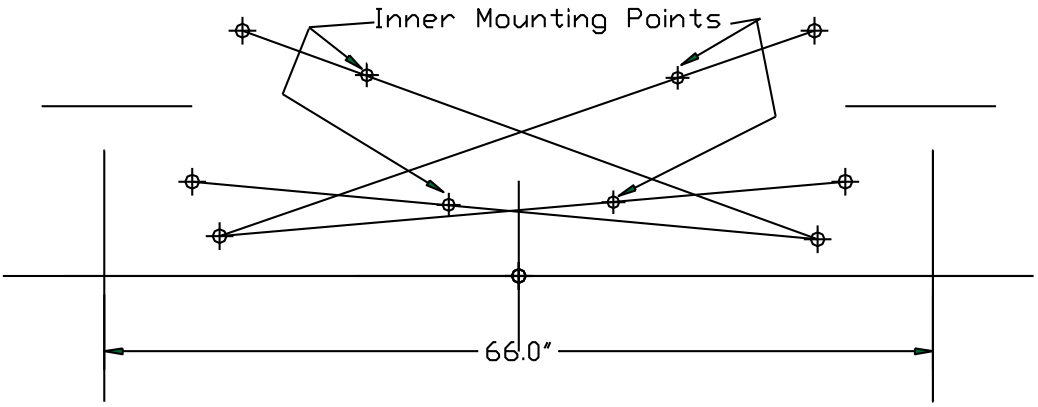


Figure 6-70

- Finding The Kinematic Roll Center - Draw a line from the left and right instant centers to their respective contact patch center points. Where these two lines intersect are the kinematic roll center co-ordinates - Figure 6-71.

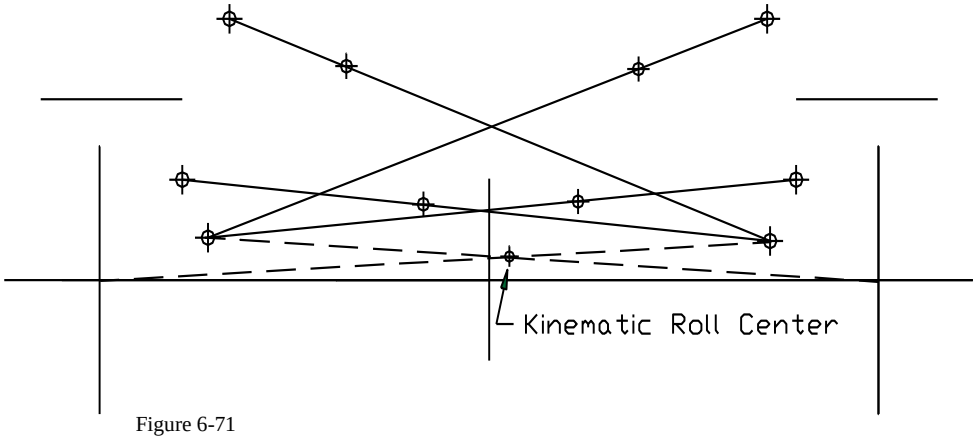


Figure 6-71

- Repeat the exercise several times using different instant center co-ordinates and/or different spindle/upright co-ordinates.

- Having completed this exercise, note that what you have accomplished to this point is to illustrate the basic theory of how a single plane, two dimensional kinematic roll center and related instant centers are found. Computers and accurate three dimensional suspension programs are of utmost importance for an accurate three dimensional design.

The Datum Roll Center - (Actual Roll Center)

As previously discussed, the Datum or actual roll center is simply an instantaneous point in space about which the chassis is rotated and/or pitched as the result of combined acceleration forces reacting against the combination of springing forces (spring, bar and damper forces) and geometric forces (anti and jacking forces) in consort with the various motion ratios i.e. the points about which the front and rear of the chassis rotate are constantly changing in relation to the strength of the chassis' overturning forces reacting against the springing and geometric set up.

The chassis' kinematic design and set up, in turn, determines the individual wheel rates. The wheel springing rate is a combination of the spring, bar and damper rates in consort with their individual motion ratios while the wheel geometric rate is a combination of its anti (side view) and jacking (front view) forces. Both the springing forces through the instant motion ratios, and geometric forces through IC placement, are controlled through the chassis' kinematic motion. A change in suspension motion will result in a change in the suspensions motion ratios and IC co-ordinates.

Everything within a suspension system is interconnected. For example, changing a corner's spring rate will affect the amount of motion seen at that corner, which, in turn, will affect the IC placement, which, in turn, affects the amount of anti or jacking force seen at that corner. This is a **key** set up consideration with chassis using considerable amounts of anti and/or jacking geometry, particularly with those chassis whose design restraints dictate sizeable amounts of suspension and IC motion. For a chassis of this type, the resultant **geometric** forces can play a **significan**t role in the combined final wheel rate.

The Datum roll center is a *result* not an *effect*. The positioning of the Datum roll or pitch center is the result of the springing and geometric set ups. It does not control how the chassis rotates, but, instead, simply shows how the chassis has rotated. The various Datum roll centers are merely a point of reference of how the chassis is rolling and/or pitching. The stiffer a particular corner's wheel rate, the less that corner will compress for any given load. Therefore, if the race engineer increases the outside wheel rates, the chassis may well roll the same amount for any given sum of overturning force, but, as a result, the outside corners will compress less while the inside corners will extend more. In this case, the Datum or actual roll center has moved laterally towards the outside corner. If the outside wheel rates are softened, the opposite will happen. The chassis may well roll the same amount but the right outside will move down more and the inside will move up less.

The chassis would have rolled the same amount in the previous two examples if only spring rates were considered but because the suspension kinematics will be quite different between the two examples, the change in IC co-ordinate positioning will result in different amounts of geometric force at each corner. This difference in the combined wheel rates at each corner will no doubt also **change** the actual amount of chassis roll and pitch for the same amount of lateral acceleration.

As Wm. C. Mitchell points out, the point of rotation does little to affect the springing forces. However, because the geometric forces are affected by the kinematic placement of the ICs, the point about which the chassis rotates certainly does affect the geometric forces. How the chassis moves in roll and pitch also affects the kinematic changes in camber, caster, toe steer, etc.

The points about which the chassis platform moves also affect the velocity of damper motion, which, in turn, affects the corner wheel rates. If the Datum roll center is positioned towards the outside of the chassis the outside damper will have a lower velocity compared to the inside corner i.e. for a given amount of time the stiffer outside corner compresses less than the inside corner extends.

There is both a front and rear instant lateral Datum roll center and a left side and right side instant longitudinal Datum pitch center. If the chassis is infinitely stiff, in theory, a single **instant** Datum rotation point can be defined. This theoretical single point of Datum rotation can only be found using mathematical models as actual, on track data will always be skewed by a certain amount of chassis/suspension flex, twist and compliance. The individual front and rear Datum roll centers and side to side pitch centers can be mathematically calculated using on track data as introduced in Book Two. Comparing the on track Datum

roll centers with the modeled roll centers will give the race engineer some idea of the amount of twist, flex and compliance the chassis and suspension is seeing. From this data, mathematical algorithms can be developed to **help** pinpoint where the chassis is flexing, twisting and/or compliant.

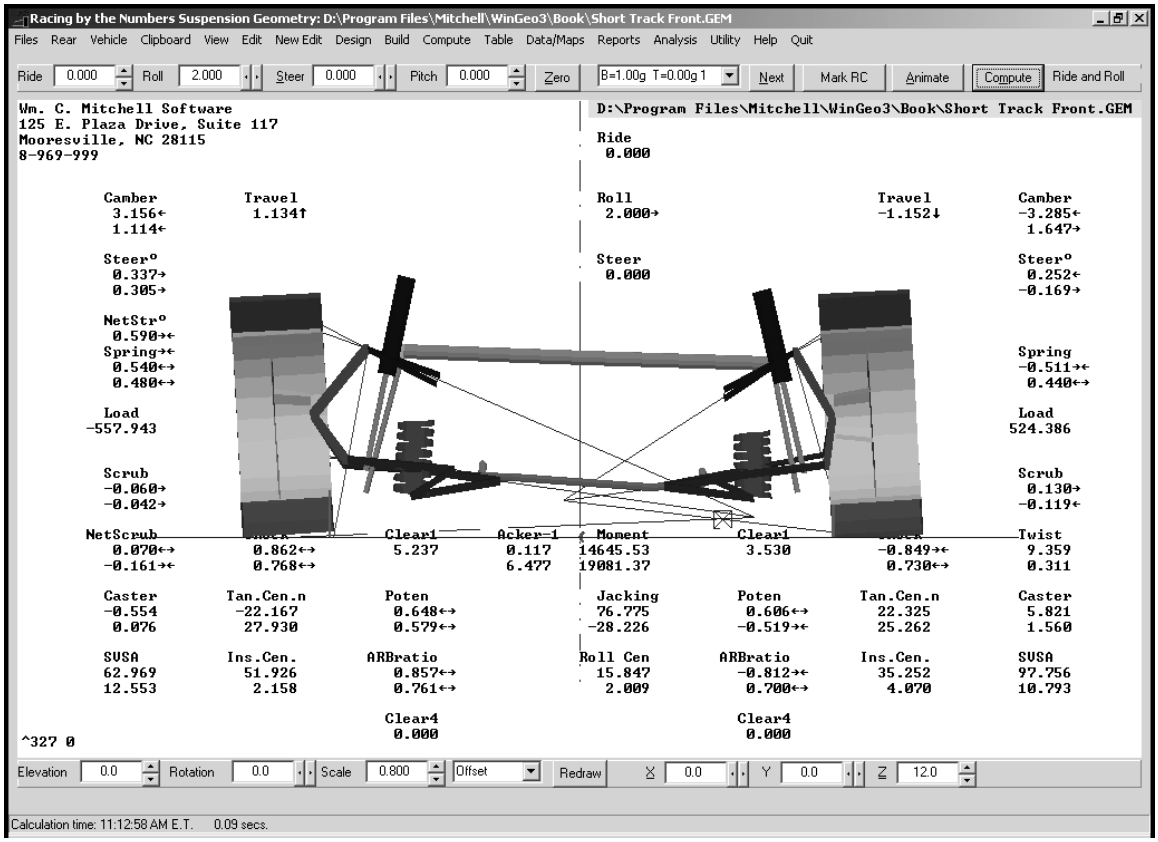
As previously discussed, the datum roll centers are simply tools to help the race engineer understand how a rate change at one or more corners will affect the chassis' overall balance and performance level. The **same** type of **information** can be found by looking at the changes in **suspension** and/or **ride height** motion. Knowing at what points the chassis is rolling and pitching (changes in the chassis platform attitude) can give the race engineer a quick overview of how making a change geared towards improved mechanical grip can affect the aero loads and aero pressure center.

Datum Roll center Note: We had originally intended to include several datum roll center calculators with Book One that could be used with on track data and/or the simulation models included in the dynamics section of this book. Unfortunately, we ran out of time trying to fit the model to the Excel format (it requires a considerable number of closed loop calculators).

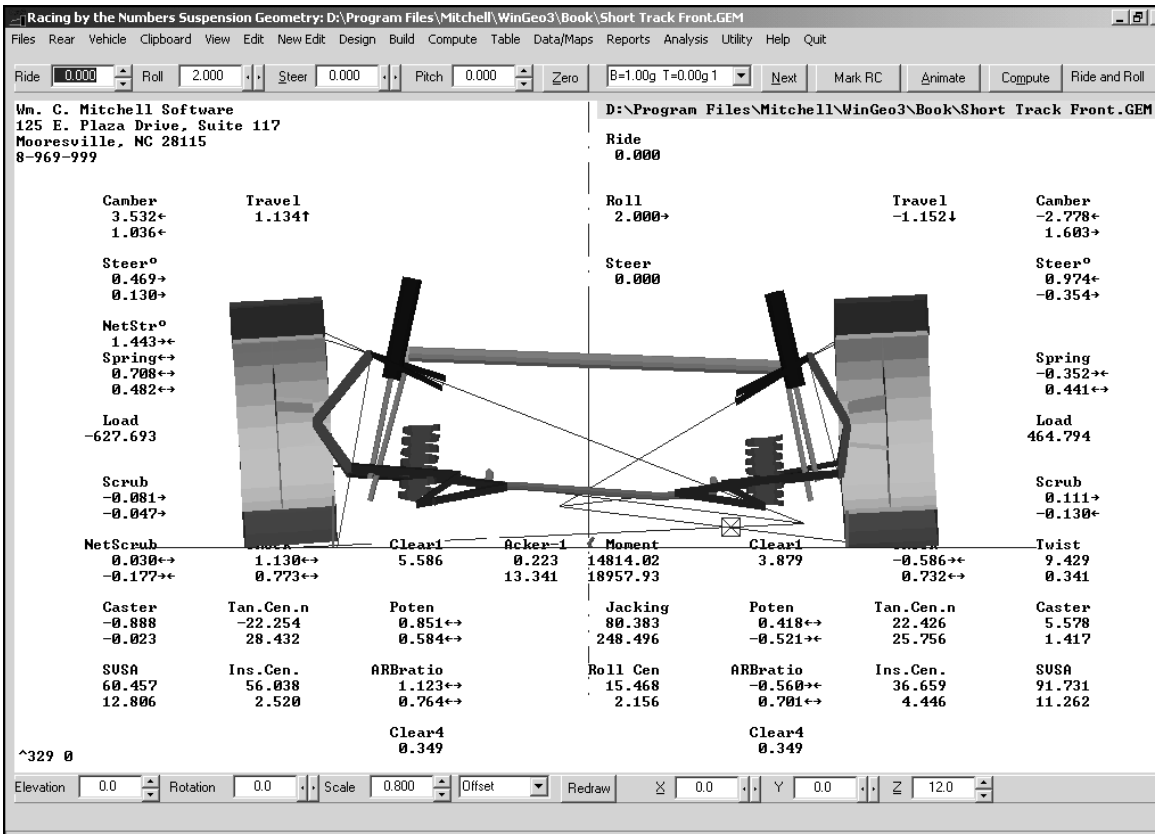
The datum roll center calculators, when ready, will probably be sold separately.

** Begin WinGeo3 Example**

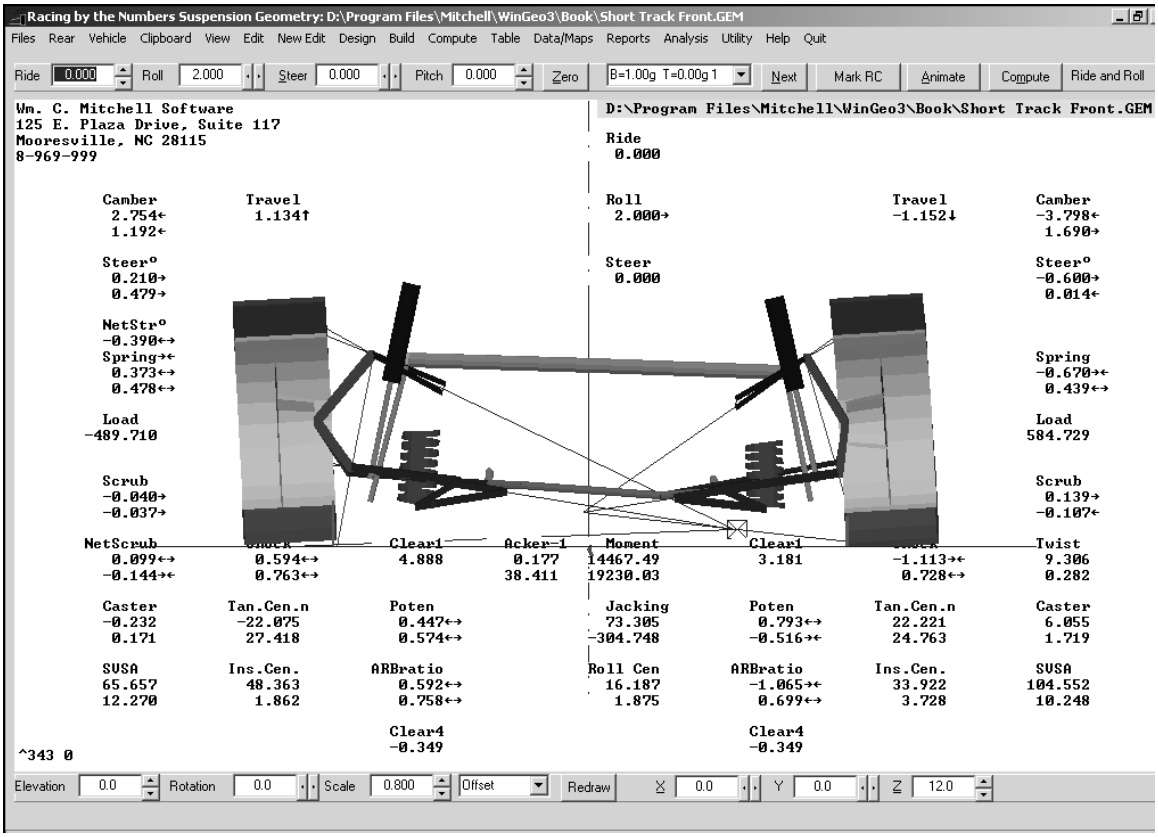
The base roll co-ordinates used by WinGeo3 is at y = 0 and z = 0. The actual point of rotation, as previously stated, is the result of the suspension kinematics in combination with the elastic and anti/jacking forces. To determine the actual point of rotation requires dynamics simulation software or on track data. When working with ***Roll,*** the user can change the point of chassis rotation (datum roll center) by going to the *Edit* menu. Click on *Rotation Point* and change the co-ordinates. Spend considerable time comparing the various kinematic components between the five datum roll center combinations. **However**, changes due to the positioning of the datum roll center are normally inputted into WinGeo through changes in suspenion or ride height co-ordinates.



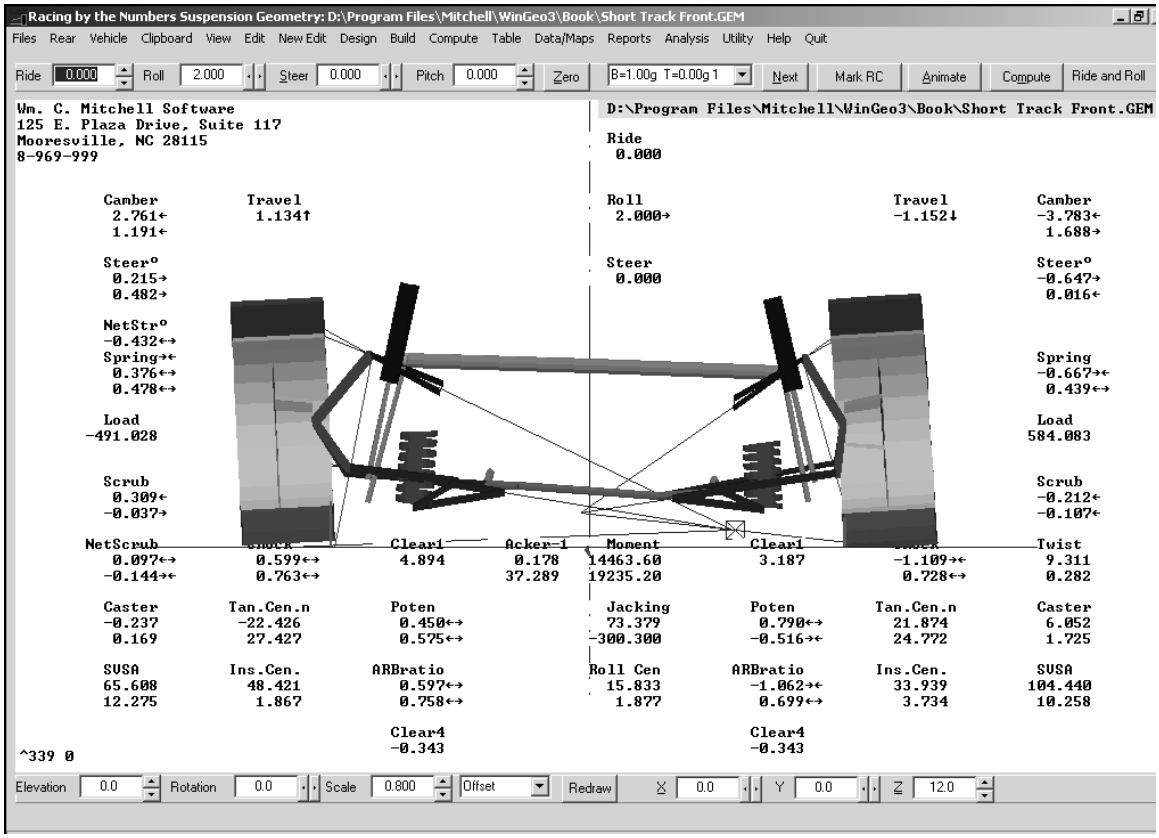
Win 6-26 y = 0, z = 0



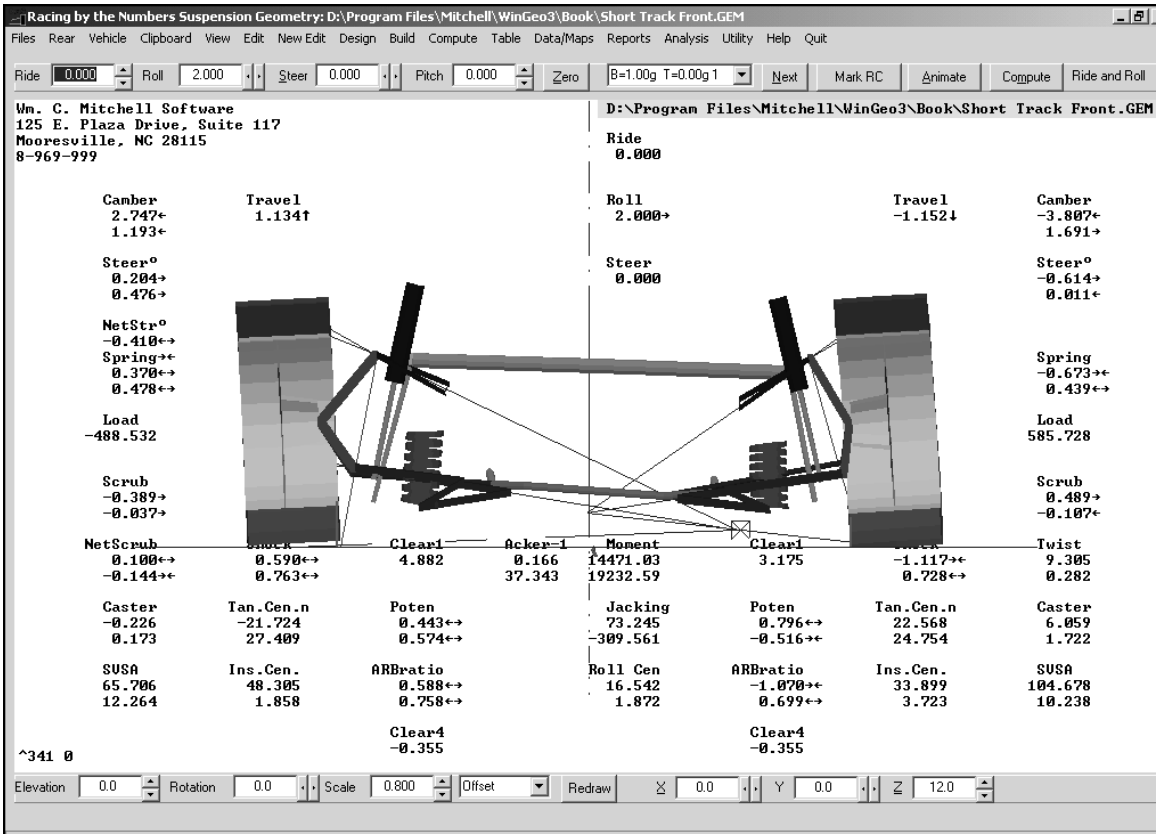
Win 6-27 y = 10, z = 0



Win 6-28 y = -10, z = 0



Win 6-29 y = -10, z = 10



Win 6-30 y =-10, z = -10

*****End WinGeo3 Example*****

The Two Primary Longitudinal Roll Axes - The first longitudinal roll axis is a straight line drawn from the rear force application point center (FAP_R) to the front force application center (FAP_F)- Figure 6-72. As previously stated, it is this axis that determines the sprung mass overturning moment. The second axis is drawn between the front and rear datum roll centers. Though the datum roll centers are the centers the chassis actually rolls about, they only marginally affect the amount of load transfer and how the load is transferred.

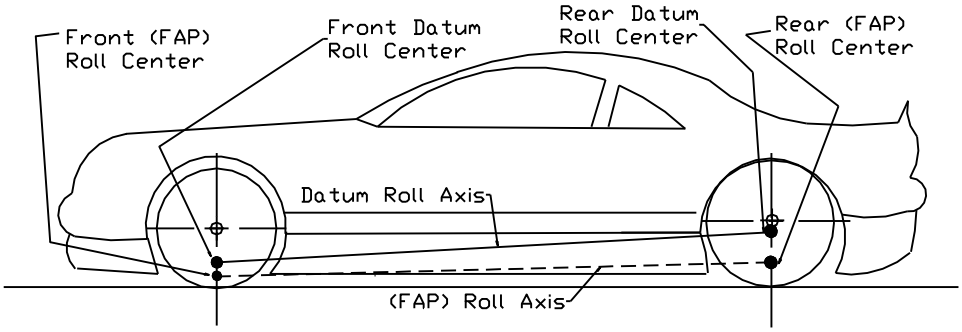


Figure 6-72

For the model shown in Figures 6-73 & 6-74, the bottom plane represents the datum roll axis in relation to the chassis and the track while the top plane represents the y (lateral offset) and z (vertical rake) placement of the chassis in relation to the roll axis and the track. As previously discussed, the placement of the chassis in relation to the datum roll axis is primarily concerned with the physical motion of the chassis i.e. component to track clearance and aerodynamic considerations. The datum roll axis is a result of the elastic and anti roll forces. It does not in itself control how dynamic loads are transferred other than the result of changes in aero loads and the aero center of gravity co-ordinates.

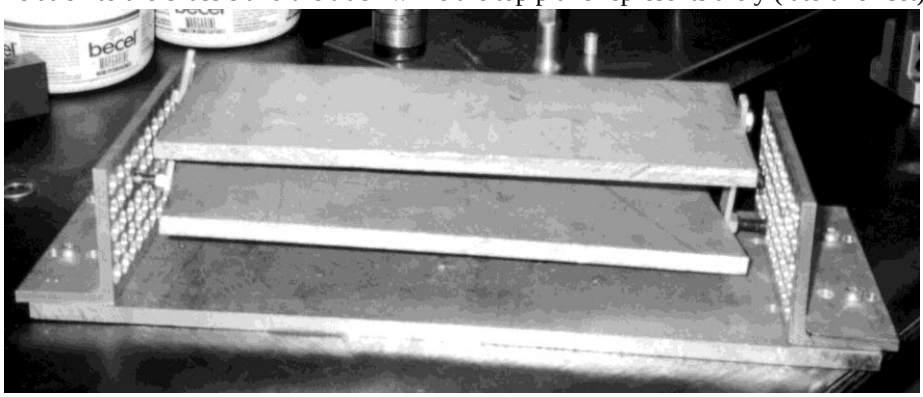


Figure 6-73

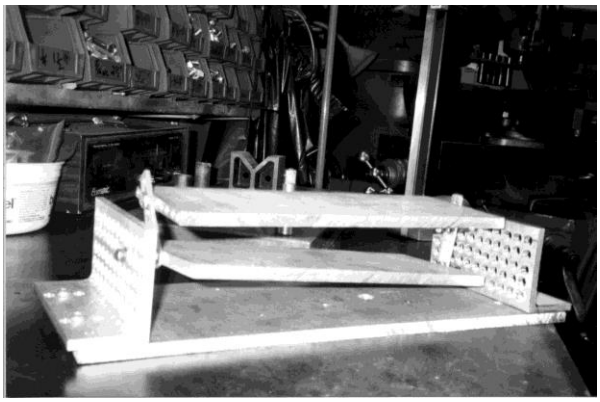


Figure 6-74

Center of Gravity - CG - The mean/total center of gravity is a three dimensional point at the junction of the x, y, z, axis while the body is at equilibrium. If the body was suspended from this point, it would be balanced and would not rotate on any axis - Figure 6-75. To quote Carroll Smith, "All accelerative forces acting on a body can be considered to act through the center of gravity of that body". As can be seen in the dynamics chapters, the race car's center needs to be as low as possible i.e. less load transfer. A higher center of gravity means more load will transfer during centrifugal (lateral) and/or inline (longitudinal) acceleration. The best example I can think of to describe this phenomenon is the considerable loss in chassis performance when the British 1600 series changed from a Kent Formula Ford engine to the Zetec Formula Ford engine with its considerably higher center of gravity.

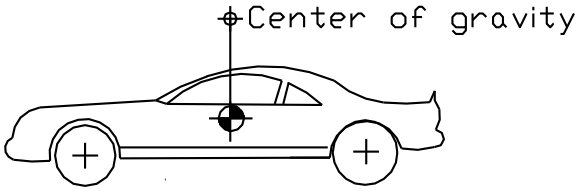


Figure 6-75

The Other Centers of Gravity - Besides the mean center of gravity, the race engineer is also concerned with the sprung mass CG, the mean unsprung CG and the four individual corner unsprung CG's. Each of these centers of gravity have their own individual X, Y, and Z co-ordinates. The sprung mass center of gravity is particularly important as it is the one used to calculate the moment arm length, angle and offset used in calculating the roll moment.

Mass Centroid - If the car is viewed as a series of sections like a loaf of bread, each section will have its own center of gravity or centroid.

Mass Centroid Axis - If each of these centroid lines are joined from center to center, it creates the mass centroid axis. The mass centroid axis is far from a straight line in either the side or plan view - Figures 6-76 & 6-77.

If the chassis is not sufficiently stiff, the relationship between the roll center, roll axis and the mass centroid axis has a major effect on the car's balance and handling. Most, **but certainly not all** modern chassis are not affected to any great extent by the mass centroid axis because they are sufficiently stiff to be considered rigid in relation to angular twist of the chassis and particularly in regards to the rigidity of the suspension system.

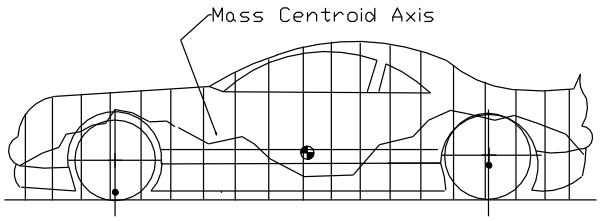


Figure 6-76

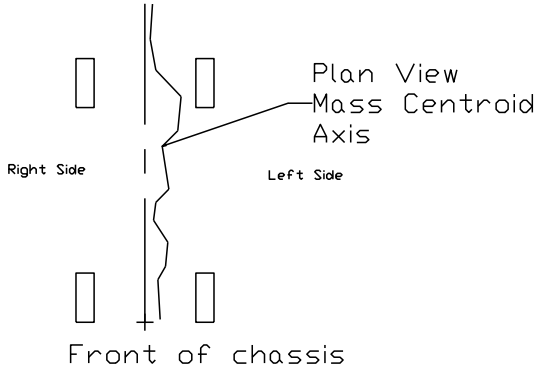


Figure 6-77

Polar Moment Of Inertia - The mass distribution of the chassis' various components about its center of rotation is referred to as its rotating inertia. "We have seen that a body with a low moment of inertia is one with a low resistance to rotational acceleration. A vehicle with a low polar moment of inertia is one which displays fast steering and cornering response - i.e. a maneuverable vehicle. We achieve this desirable feature by concentrating the mass of the vehicle within the wheelbase and as close to the longitudinal location of the CG as possible".[1]

As seen in Figures 6-78 & 6-79, the concentration of mass for the open wheel car is much closer to the longitudinal center of gravity than for the stock car.

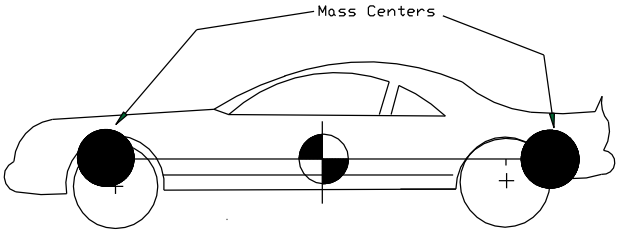


Figure 6-78

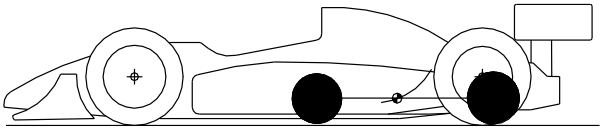


Figure 6-79

Front View Jacking/Anti Roll Centers

Important Note: The following section about geometric jacking/anti forces is simply a kinematics overview of these forces. ***It is absolutely not this simple***. The Dynamics chapters contain both the analytical mathematics and the simulation models used to work with the various geometric forces.

Geometric Jacking/Anti Force Kinematics - If an independent suspension system has its outside instant center above ground level, the line of action from the outside tire and its resultant side forces will be angled up towards the outside instant center- Figure 6-80. This lifting action tends to raise or hold up the chassis mass at that end of the car. If the outside instant center is below ground, the chassis will tend to jack down. The outside tire jacking force contributes to the combined anti roll rate (outside plus inside tires).

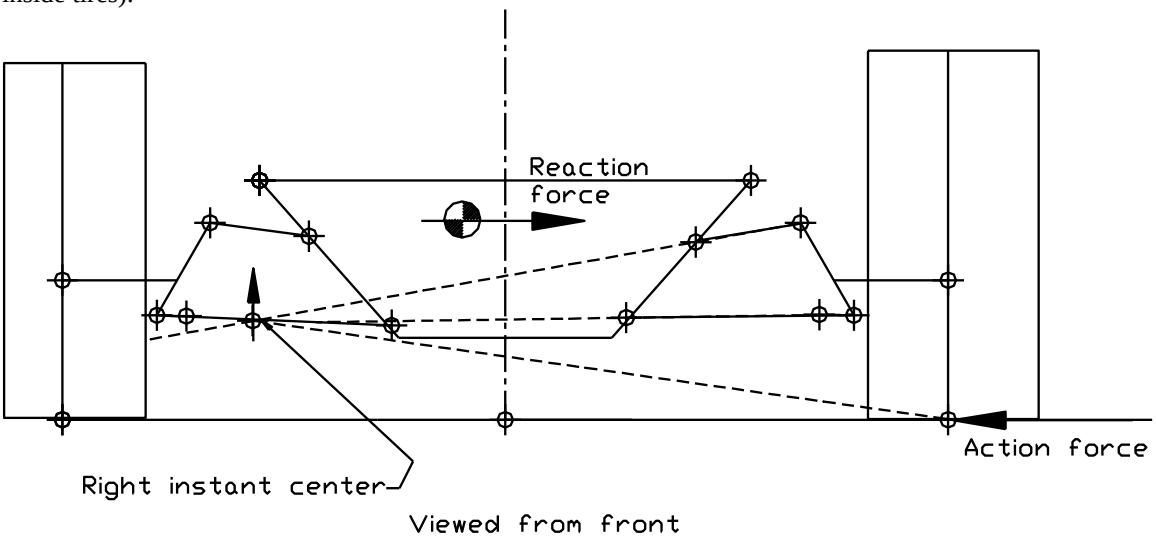


Figure 6-80

[1] Carroll Smith. "Tune To Win" USA. Aero Publishers, Inc. 1978. Pg. 31

Though the inside tire produces a smaller amount of lateral force at the contact patch (less load on the tire due to lateral load transfer), it also produces a jacking force which is reacted at the inside instant center. If the inside instant center is above ground, the jacking force will tend to jack the chassis down - Figure 6-81. This inside negative jacking force tends to pull the sprung mass down. If the inside instant center is below ground, the jacking force will tend to jack the chassis up. The inside jacking force contributes to the combined anti roll rate.

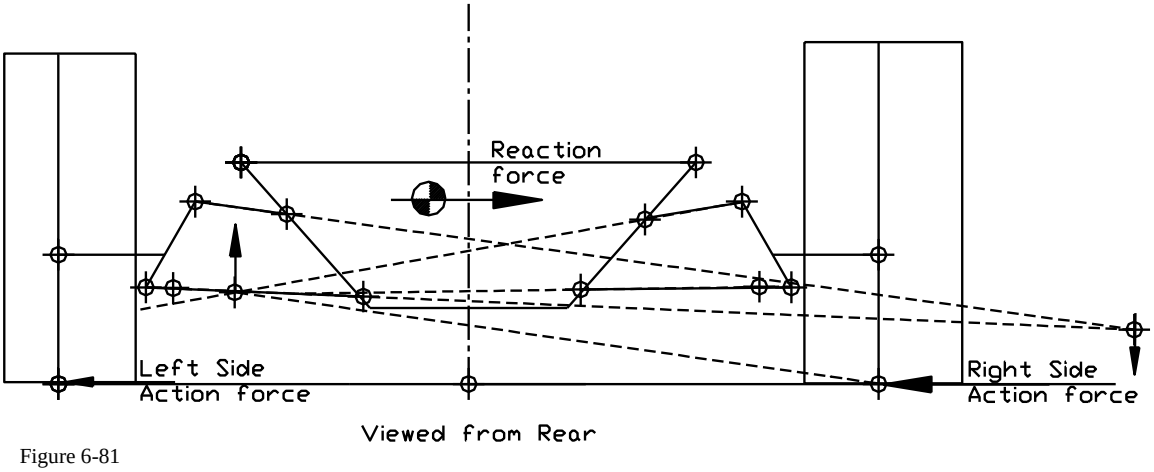


Figure 6-81

Side View Longitudinal Jacking Centers

Anti dive – Using Outboard Brakes - Anti dive is often included in the front suspension design to reduce the amount the car dives or pitches at the front due to longitudinal load transfer during braking. It can be achieved, for example, by keeping the lower A-Arm inner pivot points level while slightly sloping the upper A-Arm inner rear pivot point down towards the back of the car, sloping the lower inner rear mounting point up towards the rear of the car, or, some combination of both. Anti dive geometry does not reduce the amount of load transfer under braking (well maybe a little bit through compliance and energy absorption but it's very difficult to calculate). Anti dive geometry works by using brake torque reaction through the suspension links. Because anti dive suspension geometry moves in an arc rather than straight up and down when brake torque is applied, it tends to wrap or jack up the sprung mass of the chassis. In other words, it is using the upward force of brake torque to counteract the downward force of the load transfer. As the brake pads which are affixed to the caliper, which, in turn, are affixed to the upright grasp the counter clockwise rotating rotor, the result is a counter clockwise force being produced at the upright. This, in turn, tends to load the suspension components. The upward torque of the angled suspension links then tend to counteract the load transfer diving/pitching torque.

If the mounting axis of the suspension links are not angled i.e. they run parallel to the ground, and, the inner front and rearward mounts are parallel with the longitudinal center line of the chassis, there will be no resulting moment between the suspension links and the chassis. This is because the force input is parallel to the resistance of the mounting axis at the tangent point. On the other hand, if a line drawn through the upper A-Arm inner mounts slopes down towards the rear of the car and/or a line drawn through the lower A-Arm inner mounts slopes up towards the rear of the chassis, the upright torque motion will produce a vertical force component which counteracts the pitching moment - Figure 6-82 The vertical force component is seen at the resulting side view instant center (at the point where the two projected lines intersect) - Figure 6-83. If the chassis is symmetrical and the right and left sides have the same anti dive geometry, then the mean side view jacking center would be horizontally situated along the longitudinal center line of the chassis. Most oval track cars are asymmetrical and run different amounts of anti dive on each side. In this case, the side view jacking center would tend to be offset as shown in Figure 6-84.

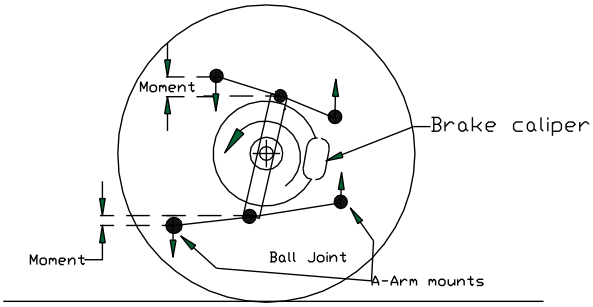


Figure 6-82

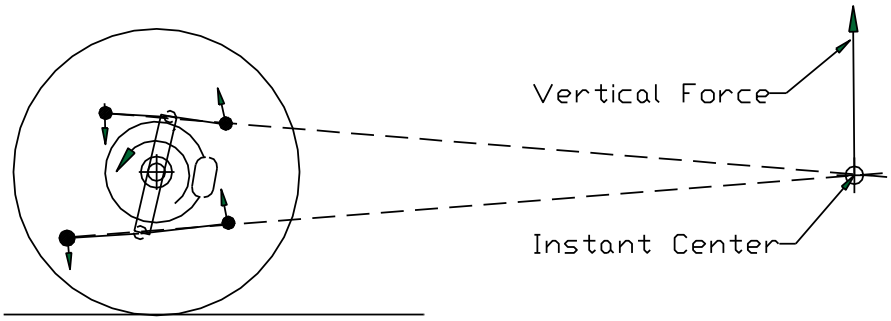


Figure 6-83

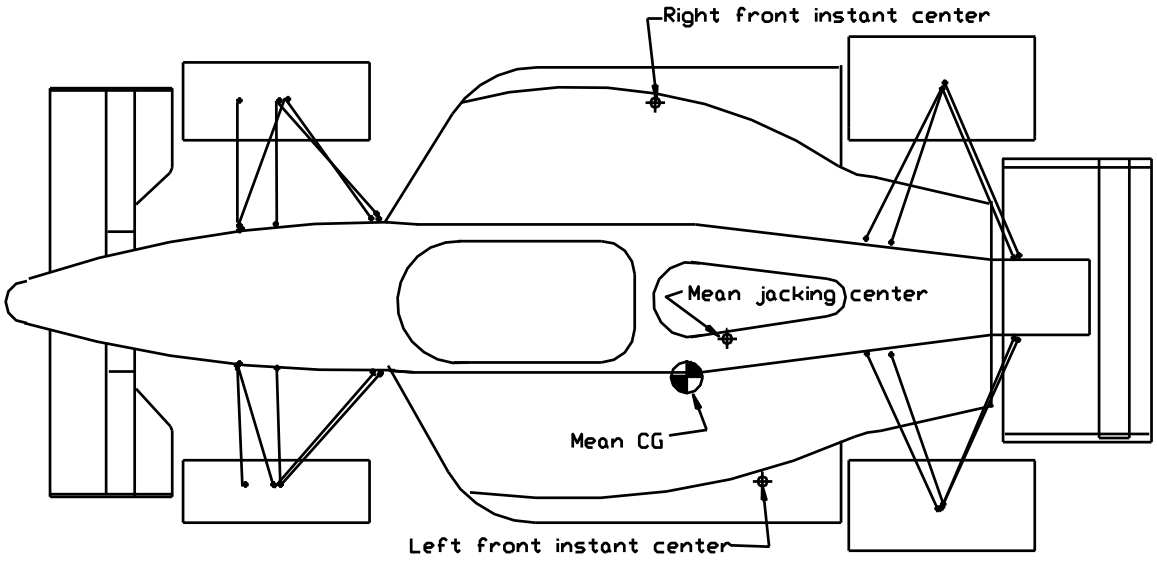
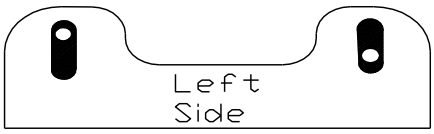


Figure 6-84

Anti dive geometry can result in a considerable number of side effects such as longitudinal wheel movement and large amounts of caster gain. Stock car types of suspension with their tight arcs and relatively large amounts of suspension motion also have a tendency to bind and even lock up under heavy load transfer during braking.

Despite the side effects, anti dive geometry is routinely designed into most types of race car suspensions. For example, considerable adjustment is normally built into a stock car chassis where the upper control arm mounts are slotted, allowing tabs with holes drilled at various heights to be inserted into the slot – Figures 6-85 /6-87. The lower control arms can also be slotted and adjusted by moving an eccentric. It can be designed to move vertically (anti dive adjustment) or laterally (caster and wheelbase adjustment).



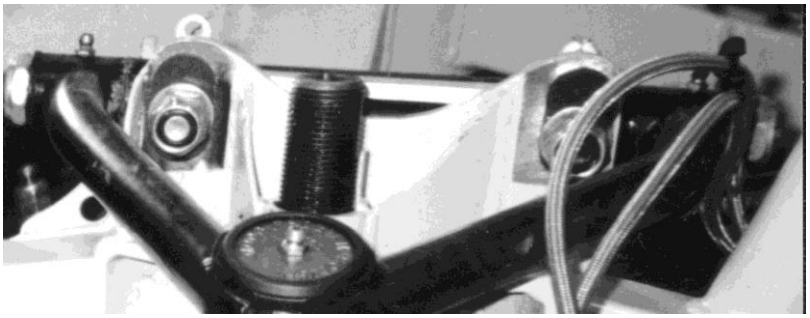


Figure 6-85 - Left front suspension

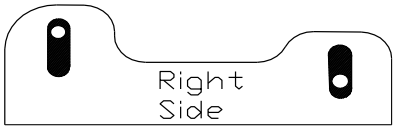




Figure 6-86 - Right front suspension

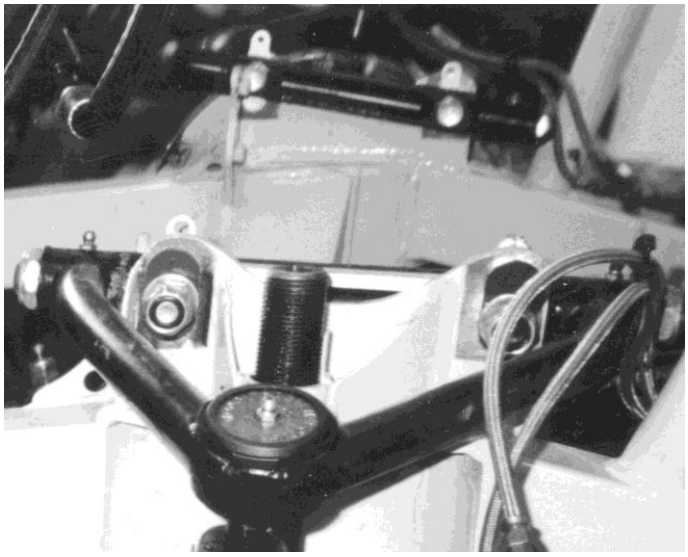


Figure 6-87 – Notice the difference between the left and right side.

A-Arm Co-ordinates Versus Anti dive Forces - The placement of the wishbone/A-Arm co-ordinates play a considerable role in determining the amount of anti dive force. Experimenting with representative and/or scale models is an excellent way to understand the various torques and forces as mathematically modeled in subsequent chapters - Figures 6-88 & 6-89.

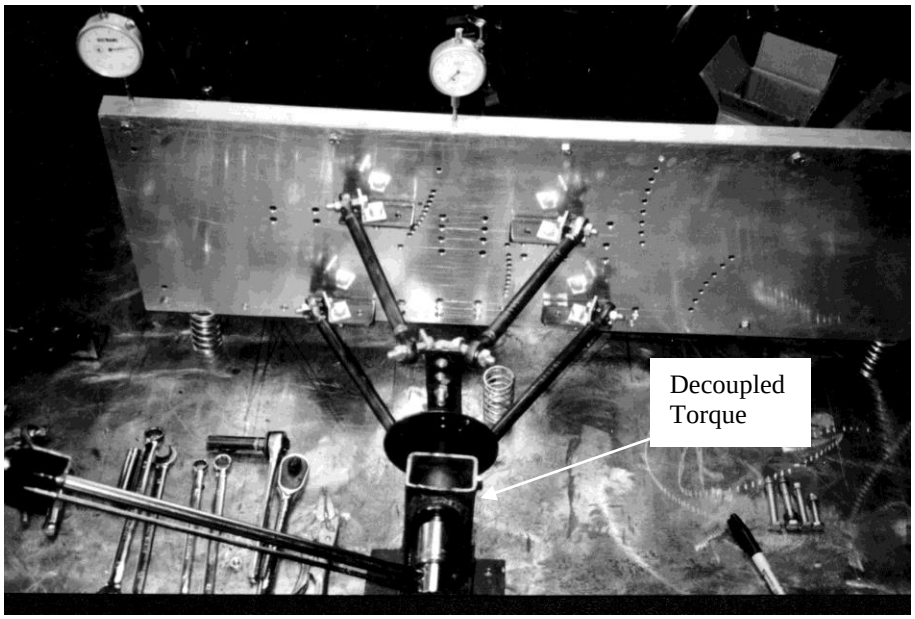


Figure 6-88

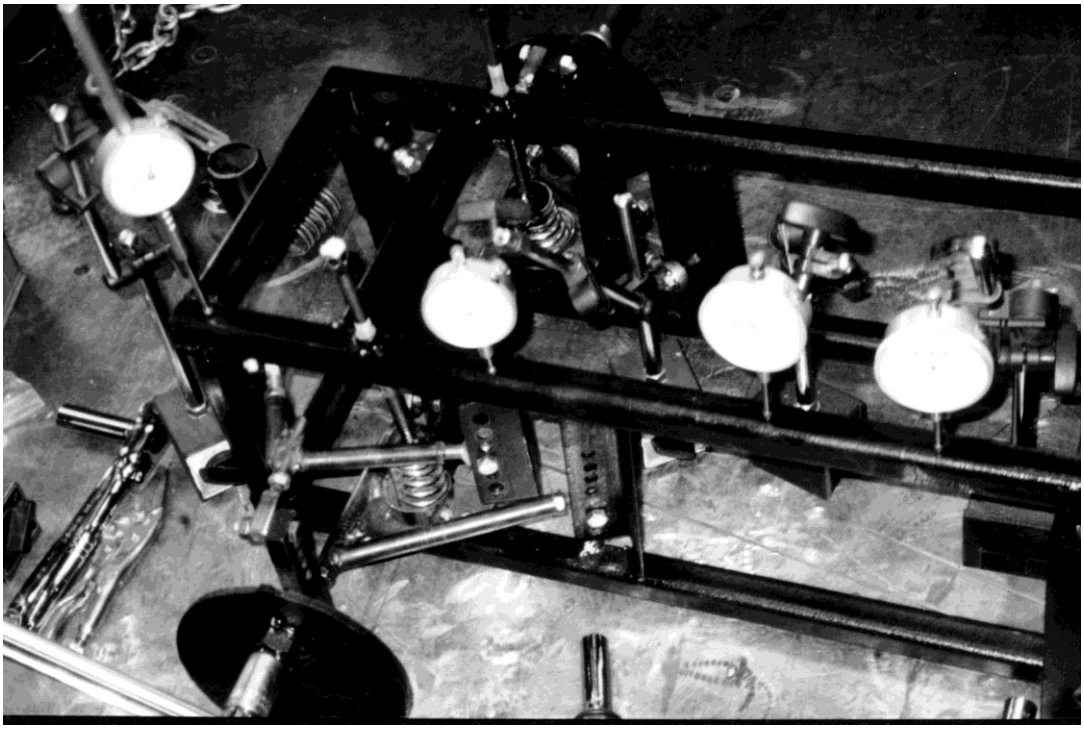


Figure 6-89