7

Steering Geometry
Kinematics and Definitions
(It's just not this simple!)

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.

The Five Toes - There are five types of toe:

- Static toe - as the car sits on the set up pad - (Static)
- Bump steer toe - as the result of **suspension** motion in **bump**- (Dynamic)
- Roll steer toe - as the result of **chassis** motion in **roll** - (Dynamic)
- Ackerman steer toe - as the result of steering input and steering kinematics - (Dynamic)
- Net/resultant toe

Static Toe Out - Static toe out is the amount the tires are angled out at the front as viewed along their longitudinal axis in the plan view - Figure 7-1 Toe in is simply the opposite of toe out. Toe can be measured as:

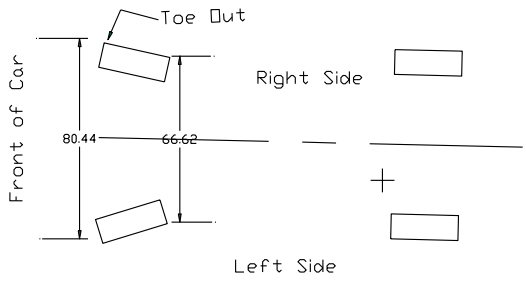


Figure 7-1

- The difference in distance between the front and rear of the tires normally measured at spindle height with the steering box or rack and pinion centered and the chassis at ride height.
- The angular difference between the tires i.e. 2 degrees out at the right tire and 1 degree out at the left tire equals 3 degrees toe out - Figure 7-2.

Static toe measurement and adjustment is one of those things that everyone believes they know how to do. Wrong! It sounds easy but there are so many ways to make mistakes. Book Two spends considerable time looking at the many ways static toe can be measured and adjusted.

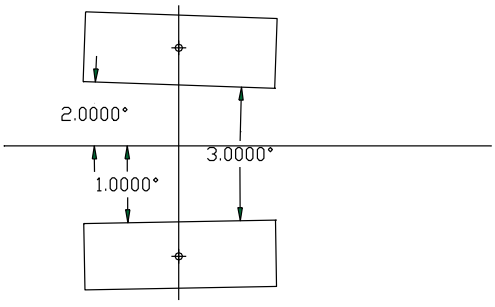


Figure 7-2

Toe Span - The difference between the front and rear of the tire or toe plate is called toe span - Figure 7-3. This measurement is a relative number in relation to the diameter of the tire or the toe plate being used. If a 20" toe plate was used rather than a 24" toe plate, the measured difference between the front and rear of the plate would obviously change with the shorter toe plate yielding a smaller number.

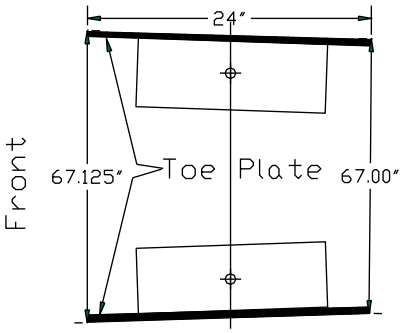
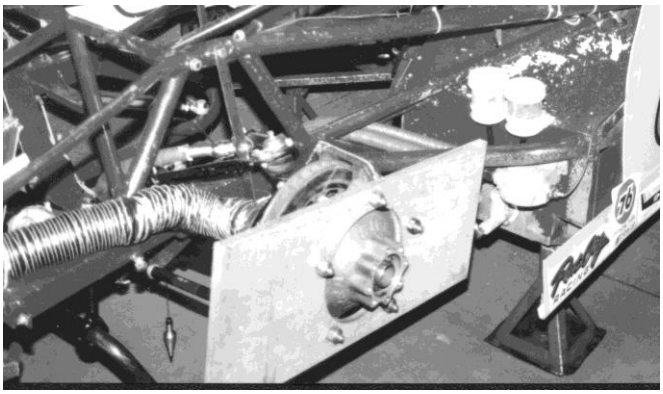
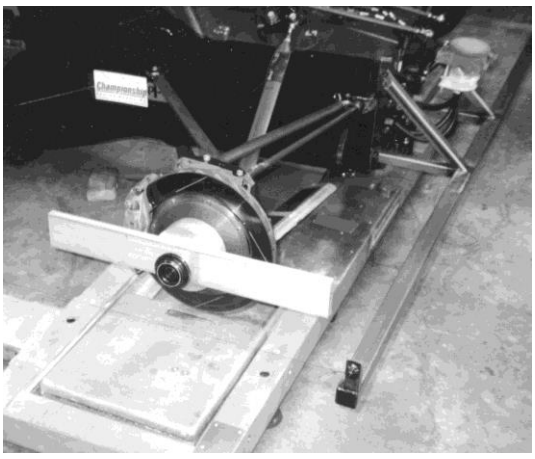


Figure 6-3

Figure 7-4 shows a large toe plate with a toe span of 32". This type of plate is used for precise shop set up. Plumb bobs are dropped from small holes in the center of the outer edges down to the surface plate where the tip of the plumb bob is very carefully poked into a piece of masking tape. The small puncture hole is marked with a pencil dot so it can be used to measure the distance between left and right sides. **Precision is of utmost importance.** To prevent the string from contacting the side of the angled toe plate if the hub has negative camber, the plumb bobs are strung from the inside and if the hub has positive camber, they are hung from the outside. If the plumb bobs are strung from the outside, as in the case of positive camber, the thickness of the toe plate needs to be subtracted when calculating the track.


Figure 7-4

Formula cars often use toe plates in combination with toe bars (stringing arms, etc.) which are carefully and very precisely fixed to the front and rear of the chassis. Strings are then run between the front and rear on both sides of the car which allow the set up person to measure the distance between the string and the front and rear of the toe plate. If the suspension is not set with zero droop, adjustable rods need to be inserted in place of the dampers.


Figure 7-5

This procedure is not nearly as simple as it sounds and will be addressed in considerable detail in Book Two. Never attach the toe bar to something that can easily get out of alignment (nose box etc.).

Machining the toe bars from solid 6061 T6 aluminum looks cool; however, even thin wall steel tube works well if properly designed and carefully fabricated. Great care needs to be taken that nothing warps when welding bits onto the main tube - Figure 7-5.

Heavily Laden Wheel - Toe and steering are normally controlled by the most heavily laden tire. As the car turns left, more load is transferred to the right front than the left front, and, as a result, the right front is controlling a greater percentage of the angular acceleration. The toe out, while controlled by the right side tire, is actually seen on the more lightly laden left tire, thus, in theory, helping the left tire negotiate the tighter inside radius.

Static toe causes the tires to constantly scrub relative to the chassis straight ahead direction which increases tire drag and heat generation. If the front has net toe-in, the tires will try to roll along paths that tend to intersect each other thereby providing a certain amount of stability in the straights (the wheels are not scribing a turn). Even though a small amount of front toe-in tends to stabilize the chassis during the straight bits, many race cars still run a small amount of static toe-out to help small radius, slow speed turn in. A rear drive car is virtually never set up with rear toe-out at the rear axle as it produces power on instability. Even live axle rear end housings are set with a small amount of toe-in because they tend to toe-out under hard acceleration.

Steering Ratio - Steering ratio can be seen as either the angular motion of the steering wheel versus angular movement of the hub (8 degrees at the steering wheel input = one degree of steer angle at the tire) or as the steering box/rack ratio (8:1, 10 :1, 12:1, 16:1 etc). Because there are always a number of variables beyond the rack or box rate, the steering ratio is normally seen as the ratio between the angle the steering wheel is turned versus the resulting angle of the hub. If the tires are placed on turn plates (set at zero) and the steering wheel turned one half turn (180 degrees) and if the resulting tire (hub) angle shown on the turn plate is 15 degrees, the steering ratio is 180/15 = 12:1. If the resulting tire angle is 20 degrees, the steering ratio would be 180/20 = 9:1 – Figure 7-6.

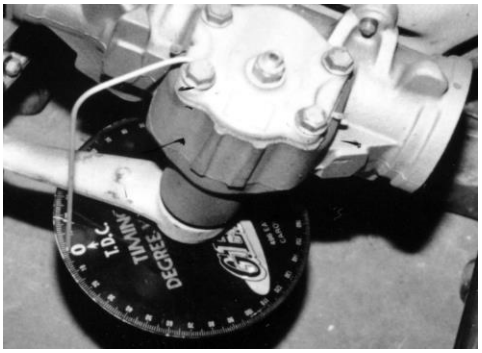


Figure 7-6

Steering Speed - Drag Link System - Steering speed is primarily controlled by the steering box ratio (8:1, 10 :1, 12:1, 16:1 etc), pitman and idler arm length and spindle steering arm length for drag link systems. The box ratio is worked out using a degree wheel i.e. if the input shaft is turned 360 degrees and the output shaft moves 37 degrees, the box ratio is 360/37 = 9.7:1. A degree wheel is also used to determine whether a steering box's ratio is a constant or variable ratio. A constant ratio steering box or rack is normally used for racing applications.

Steering Speed - Rack and Pinion - For rack and pinion systems, steering speed is controlled primarily by the rack ratio and steering arm length. Rack ratios are not universally the same. A 8:1 ratio for one manufacturer may represent 3" of rack or rod end travel with 3.4" for another manufacturer. Rack and pinion movement is best described as rod end/spherical bearing travel per turn or half turn of the pinion (steering wheel) e.g. one turn equals 2.88" of rack or rod end travel.

Net/Resultant (Dynamic) Toe - Dynamic toe is the toe change during kinematic suspension and/or chassis motion. Dynamic toe can be the result of bump steer, roll steer and/or Ackerman steer. Dynamic toe steer is most often a combination of all three.

Bump Steer - The toeing in or out of the front tires as the wheel travels in bump or rebound is referred to as bump steer. Bump steer is the result of the steering kinematics differing from the suspension kinematics in bump (they have different arcs). Bump steer can be measured either empirically using a bump steer gauge - Figure 7-7 - or by a three dimensional computer program. In actual fact, both should always be used and the results compared. Bump steer geometry is discussed later in this chapter.

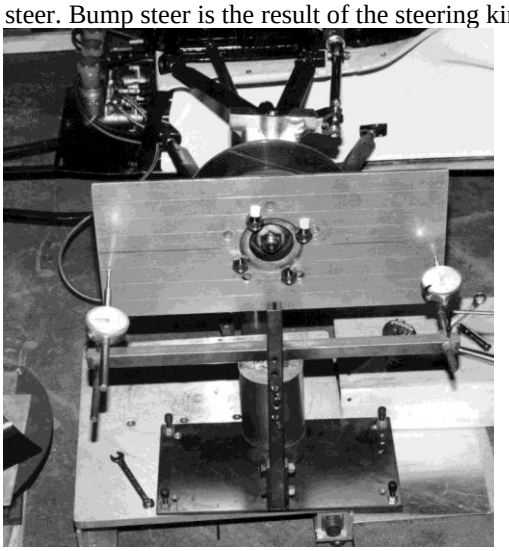


Figure 7-7

Roll Steer - Roll steer is the toeing in or out of the front tires as the steering kinematics change in relation to the suspension kinematics as the chassis rolls. Roll steer is the result of the three dimensional steering arcs not running in consort with the three dimensional suspension arcs during chassis roll. Roll steer can be calculated by:

- Using a three dimensional computer program - this method is very accurate if you are using a good **three** dimensional program and the measured co-ordinates are **accurate**.

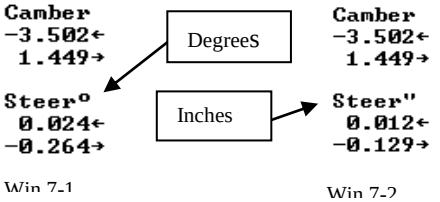
- With springs, bars and dampers removed, place the chassis on wheel stands (the stock car term is elephant stands), hold the chassis at ride height using either three or four jacks (there are two methods of doing this). The chassis can then be rolled through various iterations by moving the jacks up and down. The changes can be measured using any number of methods from string, plumb bobs, masking tape and extruded aluminum square tubing to using a digital measuring arm - Book Two. This method will give you a good indication of trends but is difficult to get extremely accurate data. You always want to make sure your front and rear stands are the same height - (*Sorry, I just had to get that little dig in*).

- Using a rather complex roll test jig as described in Book Two.

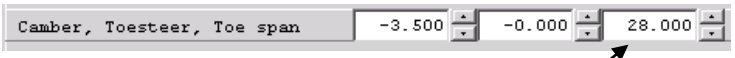
Remember, a bump steer gauge can **only** measure bump steer which is often less that half the equation. Three dimensional roll steer must be calculated in consort with three dimensional bump steer. Bump and roll steer cannot simply be added together. For example, if a wheel sees 0.060 toe out with 1" of bump and 0.030 toe out with 1^0 of roll, it does not follow that the two effects together will result in 0.090 of toe out. In most cases, combined bump and roll steer will result in an even higher toe number such as 0.110 toe out.

Begin WinGeo3 Example

Utility: Toe Steer Measurement – The *Toe Steer measurement* menu allows the user to switch between having the amount of steer and/or static toe shown in either degrees or linear distance (inches or mm's). Win 7-1 shows steer/toe in degrees while Win 7-2 shows steer/toe in inches. The amount of steer and/or toe when displayed in inches is dependent on the length of the *Toe Span*/toe plate. The *Toe Span* length is entered through the *Edit* menu (*Edit / Suspension / Toe Span)* – Win 7-3. If the user chooses to see steer/toe as linear distance, the *Toe Span* must be set accordingly.



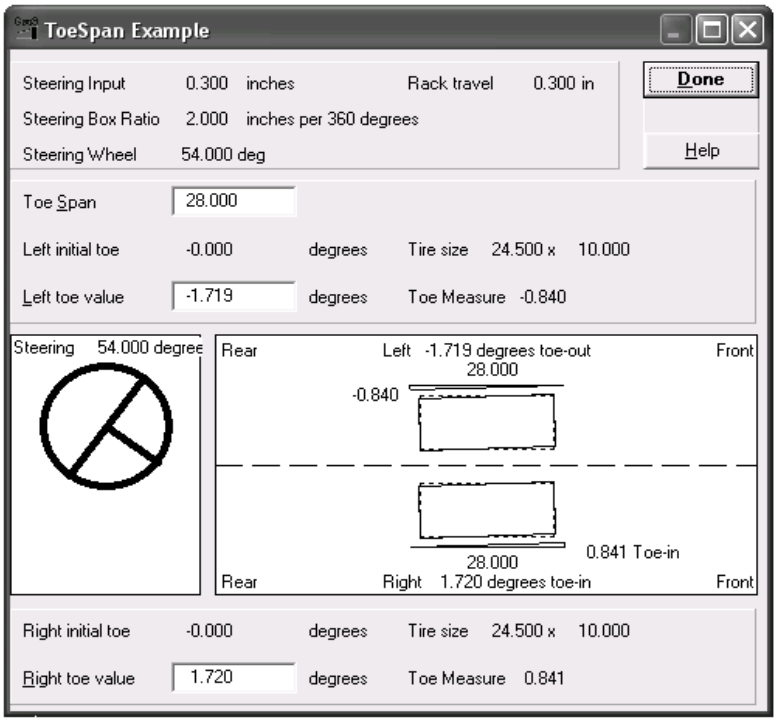
Win 7-1 Win 7-2



Win 7-3

Toe Span

The *Utility* menu also includes an example of *Toe Span* as shown in – Win 7-*4* (*Utility / Toe Steer measurement / Example of Toe Span*). The example shows how –1.719 degrees of toe on the left side translates to –0.840". On the right side, 1.720 degrees of toe equates to 0.841". The *Toe Span Example* reflects the amount of steering input indicated on the main graphics screen. This is a **very useful menu** to help the user understand all steer/toe options and calculations.



Win 7-4

Steering Input: Utility / Steering Box Ratio – Steering can be entered in one of two ways when using the advanced version of WinGeo3. Go to the *Utility* menu, click on *Steering Box Ratio* and choose either *Steering Wheel Degrees* or *Rack Travel*. Choosing *Steering Wheel Degrees* allows the user to input the amount the steering wheel is turned in degrees while choosing *Rack Travel* inputs steering as rack travel in linear distance. With the *Steering Wheel degrees* option, the user enters the degrees of rotation of the steering wheel and the program calculates rack travel.

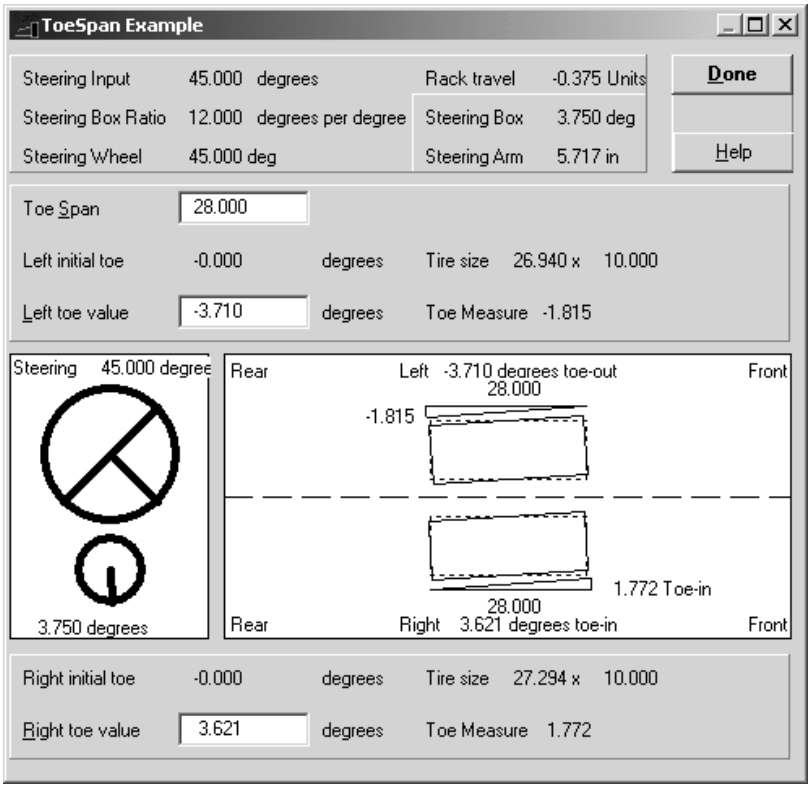
Steering Box Ratio: Edit / Steering – If the user chooses the *Steering Wheel Degrees* option, he/she must also indicate what the steering box or rack ratio is. Using the *Edit* menu click on *Steering* and then enter the appropriate *Steering Box*/Rack ratio – Win 7-5.



Win 7-5

Steering Input: Rack and Pinion - Steering input can be specified as rack travel or steering wheel degrees. Rack travel is entered as inches (standard units) or millimeters (metric units). If you specify the use of the steering box ratio, then you enter the steering wheel degrees and the rack travel is calculated as Input Degrees * Box Ratio / 360.0. The units of the Box Ratio value are inches per 360 degrees (or millimeters per 360 degrees). See *Edit: Steering.*

Steering Input: Drag link - For drag link steering, systems the box ratio is degrees per degree. A 12:1 box will reduce 180 degrees of steering wheel movement to 15.0 degrees of rotation of the steering box. This value is then translated to lateral movement at the end of the drag link arm. In the example below, 45 degrees of steering wheel movement represents 3.75 degrees of rotation of the steering arm. Since the arm is 5.717 inches long, the effective rack travel is 0.375inches.



Win 7-6

Steering Input: Angle - The steering input can also be specified as degrees of the right front tire. This works if you calibrate your steering value in degrees. For instance, you can specify a steering angle of -1.2 degrees and the program will determine the steering input that produces that value. This option can be done with *Compute*: *Solve for Steering Angle* or with Data Input.

Steering Input: Data - When you take the steering input from a data file, there is one additional option. You can rescale the steering value with a multiple and additive factor. See Data: Compute map / animate

End WinGeo3 Example

Ride and Roll Steer - Table 7-1 illustrates the differences in ride steer and roll steer for a stock car. Ride steer is the same as equal left and right side bump steer. The steering input is zero. Table 7-2 illustrates the same ride and roll iterations but with 9.134 degrees of left turn steering input.

Table 7-1

Ride and Roll Steer			Stock Car - Aiello	
Zero steering angle				
Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	0.031 out	0.125 out	0.156 out
-0.5	0	0.501 out	0.254 out	0.755 out
-1	0	0.929 out	0.348 out	1.277 out
-1.5	0	1.316 out	0.409 out	1.725 out
0	0.5	0.251 in	0.204 out	0.048 in
0	1	0.548 in	0.270 out	0.277 in
0	1.5	0.858 in	0.326 out	0.532 in
-1	0.5	0.694 out	0.387 out	1.081 out
-1	1	0.446 out	0.416 out	0.862 out
-1	1.5	0.184 out	0.435 out	0.619 out

Table 7-2

Ride and Roll Steer			Stock Car - Aiello	
Steering angle = 9.134 degrees				
Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	8.589 out	9.259 out	0.670 out
-0.5	0	9.530 out	8.223 in	1.306 out
-1	0	9.939 out	8.095 in	1.844 out
-1.5	0	10.296 out	7.992 in	2.304 out
0	0.5	8.791 out	8.287 in	0.504 out
0	1	8.496 out	8.202 in	0.294 out
0	1.5	8 .185 out	8.127 in	0.058 out
-1	0.5	9.718 out	8.032 in	1.686 out
-1	1	9.481	7.978 in	1.502 out
-1	1.5	9.227 out	7.932 in	1.295 out

Tables 7-3 use the same ride and roll iterations as in the previous examples but Table 7-1 has the right side camber increased by minus one degree while Table 7-4 also has the right side steering arm shortened 1".

Table 7-3

Ride and Roll Steer Stock Car - Aiello

Steering angle = 9.134 degrees

LF camber =3.5 **RF camber = -4.5**

LF caster = 3.88 RF caster = 5.007

Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	9.070 out	8.389 in	0.680 out
-0.5	0	9.530 out	8.226 in	1.304 out
-1	0	9.939	8.091 in	1.847 out
-1.5	0	10.296 out	7.984 in	2.312 out
0	0.5	8.791 out	8.291 in	0.500 out
0	1	8.496 out	8.203 in	0.293 out
0	1.5	8.185 out	8.125 in	0.060 out
-1	0.5	9.718 out	8.026	1.692 out
-1	1	9.481 out	7.969 in	1.511 out
-1	1.5	9.227 out	7.921 in	1.307 out

Table 7-4

Ride and Roll Steer Stock Car - Aiello

Steering angle = 9.134 degrees

LF camber =3.5 **RF camber = -4.5**

LF caster = 3.88 RF caster = 5.007

Left steering arm shortened 1"

Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	9.070 out	9.630 in	0.560 in
-0.5	0	9.530 out	9.443 in	0.096 out
-1	0	9.939 out	9.272	0.667 out
-1.5	0	10.296 out	9.141 in	1.154 out
0	0.5	8.791 out	9.512 in	0.721 in
0	1	8.496 out	9.407 in	0.910 in
0	1.5	8.185 out	9.312 in	1.127 in
-1	0.5	9.481 out	9.193 in	0.525 out
-1	1	9.481 out	9.124 in	0.356 out
-1	1.5	9.227	9.066	0.162 out

Bump Steer Geometry - Front View - When designed in the traditional two dimensional front view plane, the arc the wheel scribes as it moves up and down must closely match the arc of the outer tie rod end as it also moves up and down - Figure 7-8. If the tangent of the arcs made by the wheel and tie rod end are not parallel, the result will be bump steer. If the race engineer is looking for zero bump steer in the front view plane, the tie rod path must originate from the front view instant center i.e. the center of the arc followed by the steering arm ball joint must be coincident with the suspension instant center.

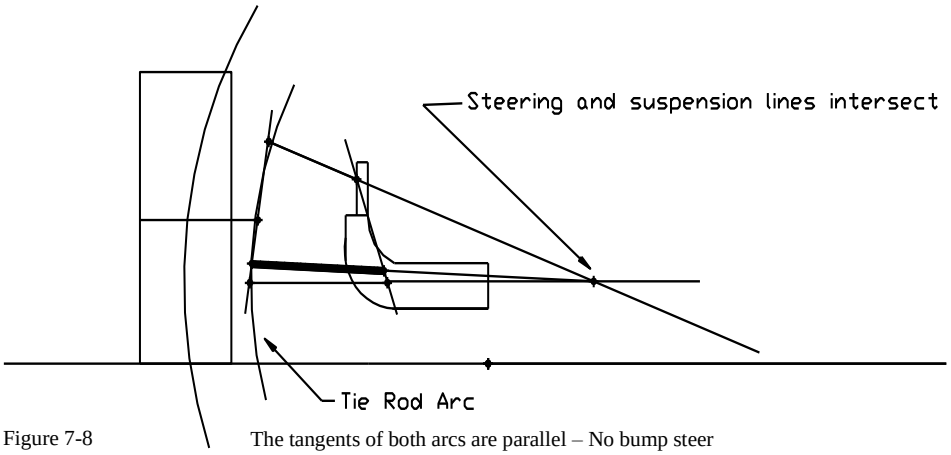


Figure 7-8 The tangents of both arcs are parallel – No bump steer

The length of the tie rod is determined by projecting a line drawn vertically through the upper and lower ball joints and a second vertical line projected down through the front upper and lower inner A arm pivots. The required tie rod end points are where these lines intersect the steering swing arm - Figure 7-9. **Unfortunately**, this type of design only works in two dimensions with zero anti dive and both wheels pointed straight ahead. However, it is a very important conceptual exercise to practice.

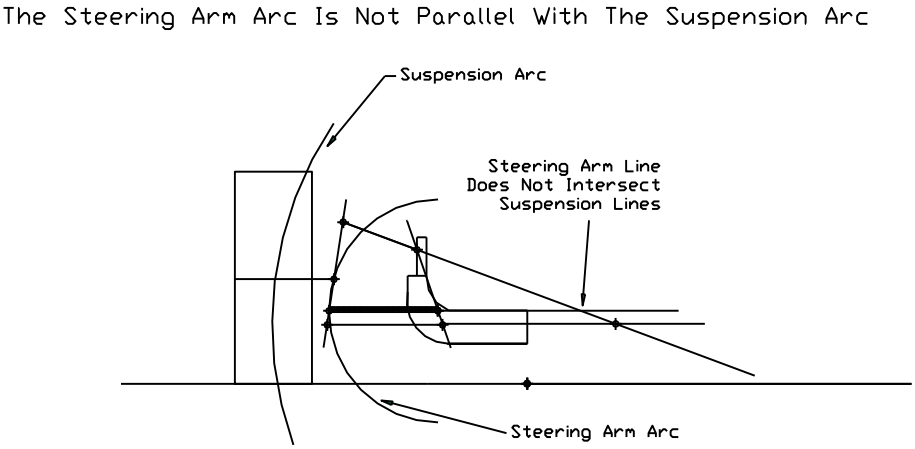


Figure 7-9

Bump Steer Geometry - Side View - The side view instant center, steering axis, caster, caster gain, scrub and particularly the suspension's instant axis will also control bump steer. As stated earlier, this is far too important to ignore but a good three dimensional computer program is needed to work through all the necessary iterations.

Combined Bump and Roll Steer Geometry - There is only one way to accurately determine the amount of combined bump and roll steer a chassis has, or, conversely, to design a steering system with a known amount of combined bump and roll steer, and that is using a three dimensional computer program, period. **As always, garbage in equals even more garbage out. The fabrication and measured accuracy of the suspension and steering co-ordinates are of upmost importance**. A chassis can be accurately measured using string, plumb bobs and very straight beams if enough time and care is taken, and, conversely, I have observed teams with high tech digital measuring devices totally mess it up. Fabricating a chassis precisely and measuring chassis components precisely is the focus of Book Two. *It is far from simple and that is about as large an understatement as you can get*.

Bump and Roll Steer Adjustment - Bump and roll steer are adjusted by moving the inner and/or outer toe link/tie rod x, y, z co-ordinates.

Vertical Co-ordinate Change - Vertical change is accomplished by shimming the toe links/tie rods up or down with spacers, or, in the case of a rack, the rack can also be shimmed up or down - Figures 7-10 & 7-11.

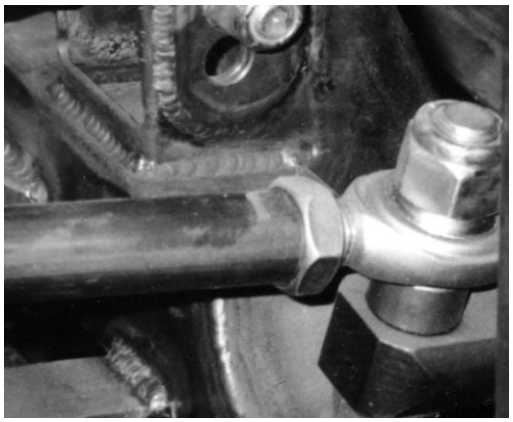


Figure 7-10

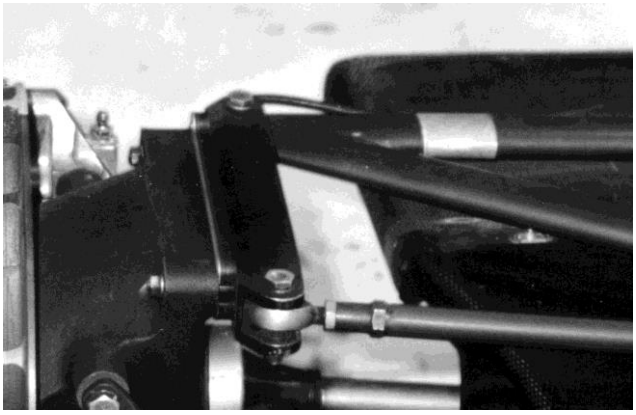


Figure 7-11

The tie rod mounts on a drag link system can be moved vertically by either a spacer mechanism at the steering link, idler arm and/or pitman arm - Figures 7-12 & 7-13, or, in the case of the inner tie rod mounts, by slotted inserts in the drag link – Figure 7-14.

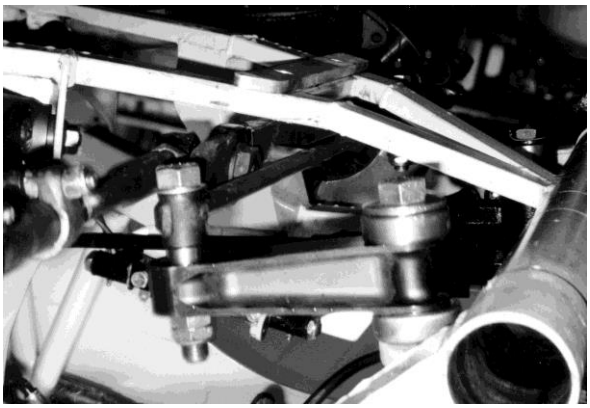


Figure 7-12

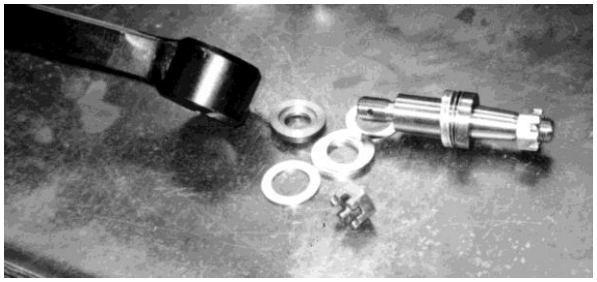


Figure 7-13



Figure 7-14

Lateral Co-ordinate Change - For the most part, changing the lateral co-ordinates simply changes the tie rod length, thereby changing the radius of the arc. Shortening the arm tightens the arc while lengthening the arm increases the arc radius. Outer lateral co-ordinate changes can be accomplished by redesigning the steering link location further or closer to the wheel centerline. As we will see shortly, changing the steering link co-ordinates also changes the amount of Ackerman steer.

Changing the inner lateral mounting points either requires a rack redesign, or, in the case of a drag link system, moving the mounting points on the drag link.

Fore/Aft Co-ordinate Changes - With a rack system, the inner mounting co-ordinates can be changed by shifting the rack fore or aft while on a drag/center link system the box and idler arm mounts can be relocated or the pitman arm and idler arm can be lengthened or shortened. Changing the pitman and idler length also changes the steering rate while changing one arm to a different length than the other changes Ackerman steer as well as the steering rate (ratio). With either system, changing the fore/aft co-ordinates at the steering arm also changes the steering rate.

Developing even basic steering maps as shown in Tables 7-5 / 7-6 need to be worked out with a three dimensional computer program for each individual chassis. This is not a one day job and data is seldom, if ever, transferable from one chassis to another. Though not realistic as an actual chassis set up, for this dynamic toe steer example all arcs are considered to be at their mid point.

Table 7-5 Steering Map

Steering Map Chassis Grace #2021 Date_____ Type: Bump&Roll			
Dynamic Toe Change In Bump and/or Roll Due To Tie Rod/Toe link Co-ordinate Change(s) **All co-ordinates at mid arc**			
Inner - Raised	Toe out	Outer - Raised	Toe in
Inner - Lowered	Toe in	Outer - Lowered	Toe out
Inner - Inward Towards center of chassis - Longer toe link	Toe out	Outer - Inward Towards center of chassis - Shorter arm	Toe in
Inner - outward Away from center of chassis - Shorter toe link	Toe in	Outer - outward Away from center of chassis - Longer arm	Toe out
Inner - Forward	Toe out	Outer - Forward	Toe in
Inner - Rearward	Toe in	Outer - Rearward	Toe out

Table 7-6

Steering Map Chassis Grace #2021 Date_____ Type: Bump & Roll			
Dynamic Toe Change In Bump and/or Roll Due To Tie Rod Co-ordinate Change(s)			
	-1" Bump	1 Degree Roll	Notes
Datum	-0.348 out	-0.270 out	
Inner Forward 0.5"	-0.357 out	-0.278 out	
Inner Rearward 0.5"	-0.338 out	-0.262 out	
Inner Up 0.5"	-1.617 out	-1.007 out	
Inner Down 0.5"	+0.834 in	+0.418 in	
Inner Out 0.5"	-0.150 out	-0.167 out	
Inner In 0.5"	-0.496 out	-0.348 out	

Each car will have a unique steering profile depending on where the mounting points lie on their related arcs. There are literally tens of thousands of combinations. Beyond the simple outlines as shown in Tables 7-5 & 7-6 the team will need to develop maps to which they can refer when making a change. For example, if the engineer wants to change the bump steer on the outside wheel by x amount in combination with 0.855 degrees of roll and 9 degrees of steer, where do the changes need to be made and by how much? To make things more complex, there are always a number of combinations which will accomplish a particular change, but each one will carry varying ramifications on short run versus long run handling characteristics - testing, testing, testing.

It seems that everything to do with vehicle dynamics is interconnected, and, of course, each aspect of the steering system is no different. For example, as the spindle rises or falls in steer due to the varying combinations of caster, camber and kingpin angle, this, in turn, affects the steered angle i.e. The steering arm mounting point is both raised, and, to a lesser degree, rotated because of the various spindle arcs, changing the distance between the inner and outer tie rod mounting points. If the inner points are not allowed to move, (driver not changing the steering wheel position) then the spindle will be forced to rotate as a consequence of the distance change between the inner and outer ball joints, unless, of course, the tie rod bends.

To help visualize this, first think of the spindle with its center of axial rotation as shown in Figure 7-15. Broken down into its simplest form, the spindle can be seen as two points. The inner point is the front view instant center and the outer point is the spindle's point of axial rotation as seen in Figure 7-16. The tie rod mounting points are also simplified down to two points - Figure 7-17. As the spindle is raised, the distance between the IC and the spindle axial rotation point stays the same (if the IC doesn't move), but, because the steering system follows a different arc, its point to point distance changes. Therefore, if, the tie rod inner mount is held at a given co-ordinate, the spindle will be forced to rotate to compensate for the change in inner to outer steering arm distance - Figure 7-18.

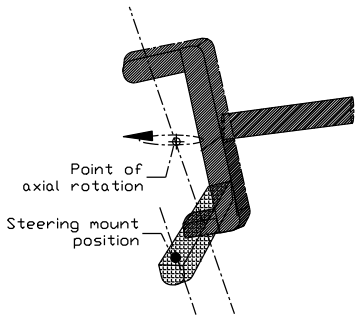


Figure 7-15

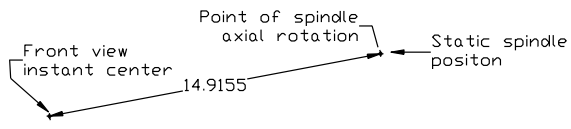


Figure 7-16

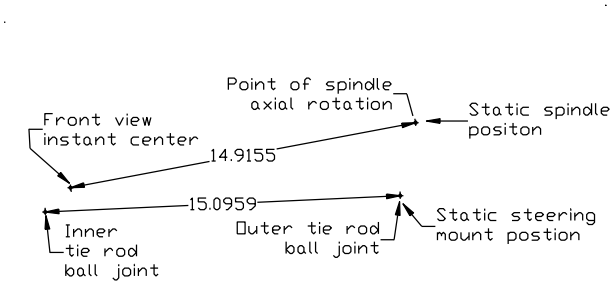


Figure 7-17

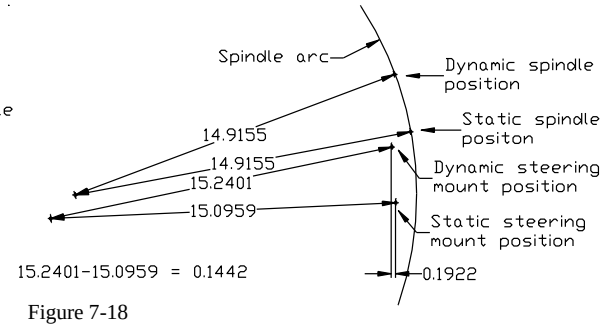


Figure 7-18

Once again, playing with a model helps conceptualize the various steering changes as the result of different suspension and steering mounting points - Figure 7-19. However, to accurately calculate the changes, one needs to use a kinematic computer program.

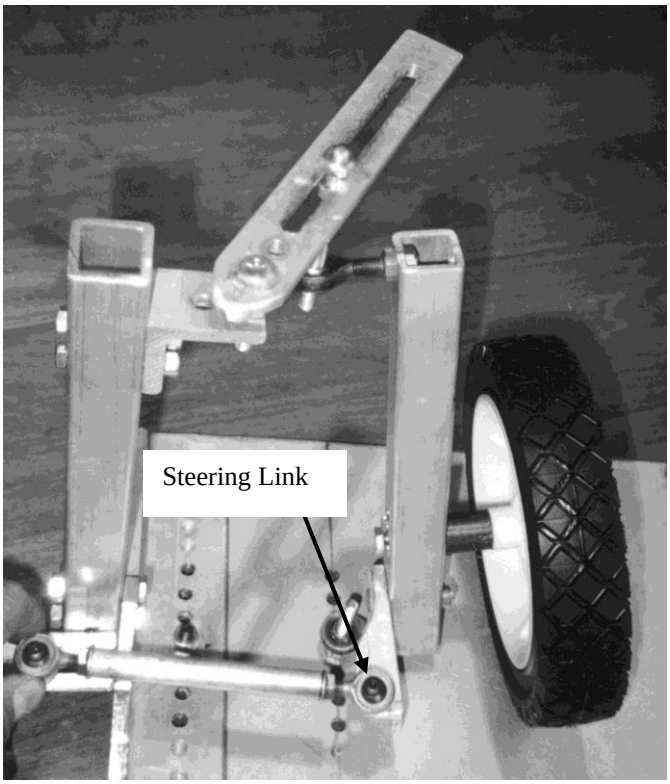


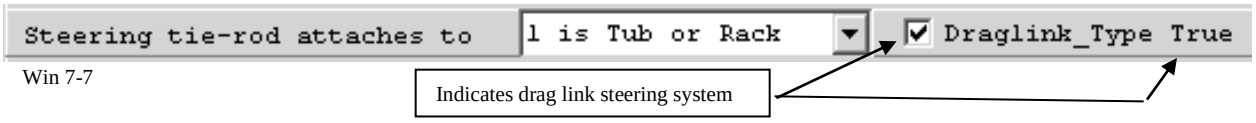
Figure 7-19

Begin WinGeo3 Example

Edit: Steering – Steering component co-ordinates as measured on an existing chassis or from chassis design data is entered using the *Edit / Steering* menu. Design and/or set up changes **cannot** be made using the *Edit / Steering* menu.

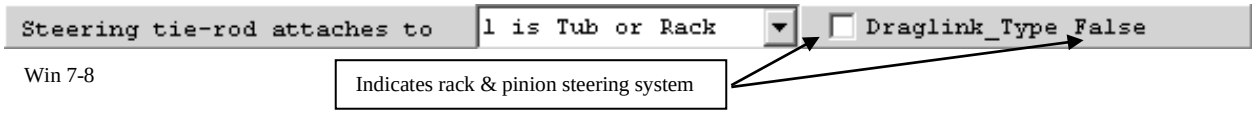
The steering components can be entered either as a rack and pinion or drag link system. Within the *Edit / Steering* menu the user clicks on the *Drag Link Type* button to indicate whether the steering is a rack or drag link system. If there is a check mark in the *Drag Link Type* box, the program will show "*True*". If the box does not show a check

mark, the program will show "*False*", indicating the steering system is a rack and pinion type. Win 7-7 indicates the steering system is a drag link type while Win 7-8 indicates a rack and pinion steering system.



Win 7-7

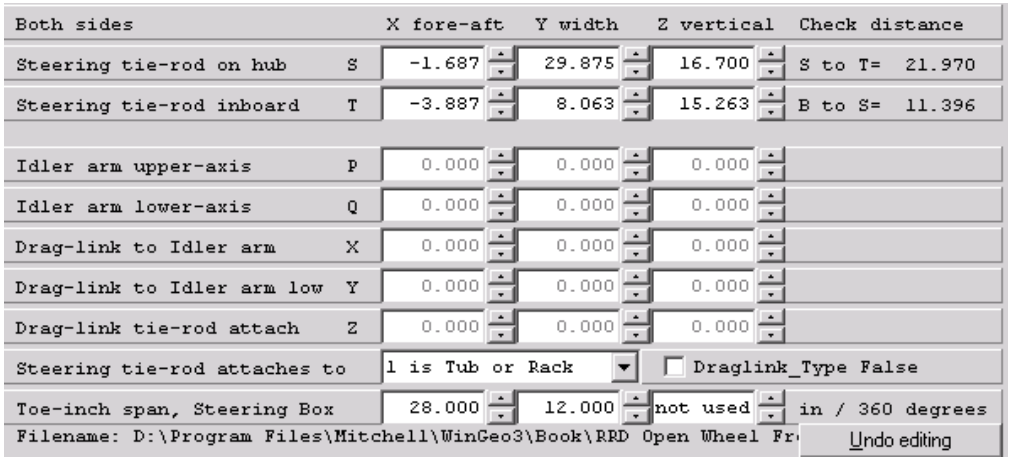
Indicates drag link steering system



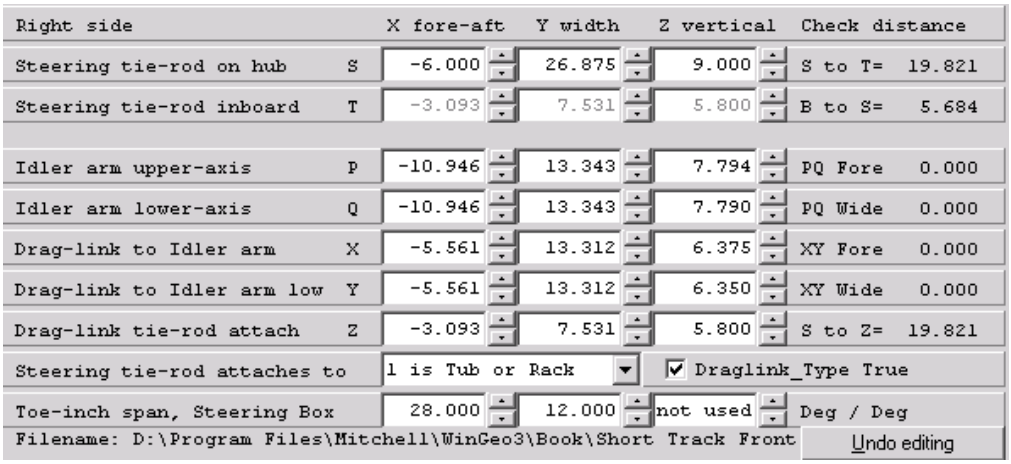
Win 7-8

Indicates rack & pinion steering system

Once the type of steering system has been indicated the software automatically activates and deactivates the appropriate steering system co-ordinate components. The pinion steering system as shown in Win 7-9 only allows the user to enter the tie rod/ toe link co-ordinates S and T. The drag link system as shown in Win 7-10 requires co-ordinate points to be entered in S, P, Q, X, Y & Z while no co-ordinate points are entered into T.



Win 7-9



Win 7-10

Notice that WinGeo3 requires an upper and lower axis point for both the idler arm (right side suspension) and the pitman arm (left side suspension). The *Axis Calculator* found in the *Edit* menu allows the user to define any point along the idler and/or pitman arm axis.

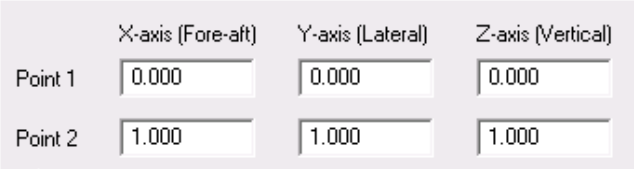
Edit: Axis Calculator - This determines points on a line or axis. If you know two points on a line or axis, this form will calculate other points on the same line. For instance, if you know the Upper A arm axis points where the A arm attaches to the chassis, you may want to calculate other points on that axis. You may have measured the points where the bolts attach the A arm to the chassis. These points would be used in the *Set Up* to calculate the effect of shimming the points. But you might also want to calculate the points where the A arms attach to calculate forces on the A arm without changing the kinematics.

The Drag link steering system and Bell-Crank Pivots use rotation axis and require the calculation of points on an axis. The Swaybar also uses several points on an axis.

Using the Axis Calculator – The axis calculator determines points on a line or axis. If you know two points on a line or axis, this form will calculate other points on the same line. For instance, if you know the Upper A arm axis points where the A arm attaches to the chassis, you may want to calculate other points on that axis. You may have measured the points where the bolts attach the A arm to the chassis. These points would be used in the *Set Up* to calculate the effect of shimming the points. But you might also want to calculate the points where the A arms attach to calculate forces on the A arm without changing the kinematics.

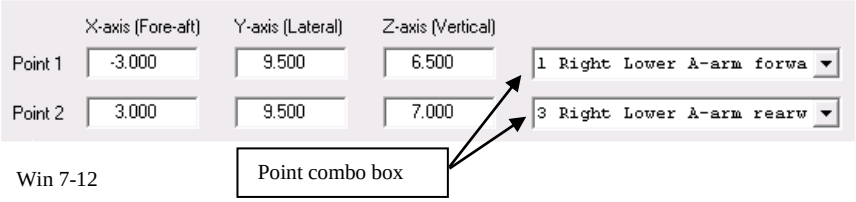
The first step is to specify the axis by declaring two points. This can be done by:

- Directly entering the two points in the appropriate editing fields, as shown in Win 7-11.



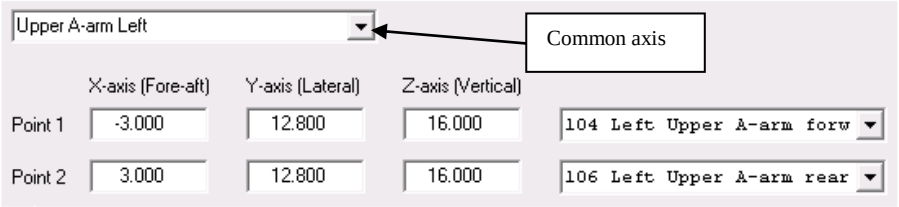
Win 7-11

- Selecting a point from the featured geometry file. This is done with the point combo boxes, as shown below. In this example, Point 1 is the Lower A arm forward point on the right side and Point 2 is the Lower A arm rearward point on the right side.



Win 7-12

- Selecting a common axis or pairs of points, such as the Left Lower A arm as shown below. Note that left side points are shown with Y positive as in the editing forms.



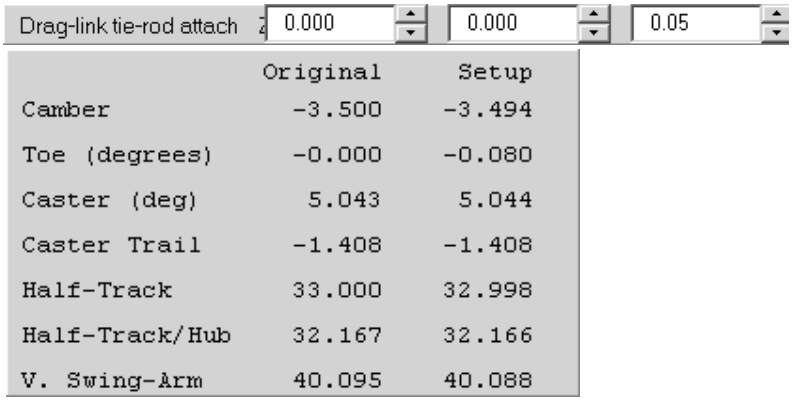
Win 7-13

Changing the Static Toe Settings

Build: Set Up Permanent Changes – As discussed in Chapter 6, the *Build / Set Up* menu allows the user to make a number of tie rod/toe link changes at the upright/spindle.

- Move upright/spindle link attachment point out (away from chassis)
- Move upright/spindle link attachment point in (towards chassis)
- Move upright/spindle link attachment point forward
- Move upright/spindle link attachment point rearward
- Raise upright/spindle link attachment point
- Lower upright/spindle link attachment point

As always, "**Changing One Thing Changes Everything Else".** As shown in Win 7-14, raising the upright/spindle mounting point by 0.05" while not changing the tie rod/toe link length, changes all of the upright/spindle parameters. As shown in Win 7-15, after the upright/spindle mounting point has been altered to accomplish a dynamic change, the static settings were returned to their base setting by shortening the tie rod/toe link by -0.008".



	Original	Setup
Camber	-3.500	-3.494
Toe (degrees)	-0.000	-0.080
Caster (deg)	5.043	5.044
Caster Trail	-1.408	-1.408
Half-Track	33.000	32.998
Half-Track/Hub	32.167	32.166
V. Swing-Arm	40.095	40.088

Win 7-14



	Original	Setup
Camber	-3.500	-3.500
Toe (degrees)	-0.000	-0.000
Caster (deg)	5.043	5.043
Caster Trail	-1.408	-1.408
Half-Track	33.000	33.000
Half-Track/Hub	32.167	32.167
V. Swing-Arm	40.095	40.095

Win 7-15

Build: Set Up / Tie Rod length – This option allows the user to experiment with changing the static toe by lengthening or shortening the tie rod/toe link.

Build: Set Up / Solve for Toe – This option allows the user to determine what the change in tie rod/toe link co-ordinates would be for a given amount of static toe change. Win 7-16 shows a change of 0.125" toe out.



Figure 1

Build: Set Up / Solve for Bump Steer – This option allow the user to enter an amount of ride height change and see the resulting changes at that corner. Win 7-17 shows a change in ride height of –1.0" (1" of bump) and the resulting changes at that corner.



	Original	Setup
Camber	-3.500	-5.028
Toe (degrees)	-0.000	0.224
Caster (deg)	5.043	5.774
Caster Trail	-1.408	-1.624
Half-Track	33.000	33.096
Half-Track/Hub	32.167	31.900
V. Swing-Arm	40.095	35.945

Win 7-17

Build: Fixture / Rotate about Lower Ball Joint – The Fixture menu can be used to enter the desired static toe change and see the resulting change in the tie rod/toe link co-ordinates. Notice that there is also a small change to the upper and lower ball joint/rod end co-ordinates as a result of the upright/spindle rotating about the center of the tire's contact patch.

Note: Notice that not only does changing the static and dynamic toe setting result in all other aspects of the suspension co-ordinates changing, but changing any other suspension co-ordinate also **results** in a toe change. The user needs to spend considerable time working through a series of suspension co-ordinate changes, including toe changes, running various *Path files* and then studying the resulting *Tables*. Comparing many different set up files is an extremely useful tool for determining the best set up compromise.

End WinGeo3 Example

Ackerman Steer - Ackerman enables the inside front tire to turn in a tighter arc than the outside tire. It is really another form of dynamic toe used to increase or decrease the amount of tire scrub on the inside tire during cornering. The tighter the turn radius the car is negotiating, the greater the amount of Ackerman needed to have both the inside and outside wheels follow their respective arcs. If the front track is widened, a higher Ackerman percentage is also needed. Ackerman, like toe out, may help the car at turn entry through the apex, mid turn and/or even turn exit.

As will be discussed in more detail shortly, the amount of Ackerman steer a chassis has is often measured by comparing the difference in steering angle between the inside and outside tire. If the inside turns 11 degrees and the outside tire turns 10 degrees, the result will be 1 degree of Ackerman steer using 10 degrees of steer at the outside wheel as a reference angle. Though there is no set and fast rule, Ackerman is most often measured at 10 degrees of steer at the outside wheel. This type of measurement can be accomplished by placing the chassis on degree wheels and physically measuring the difference between the inside and outside steer angle or by programming the chassis co-ordinates/design co-ordinates in a three dimensional kinematics program. Ackerman also needs to be calculated with various amounts of ride and roll.

As discussed later in this book, the race engineer will not always want the inside wheel to follow the inside arc**.** He/she, instead, may want the inside wheel to run at a lesser or greater angle, depending on the race track, driving style, tire shear angles, construction, compound, the various steering and suspension axes, chassis set up, chassis weight, etc. There are also certain instances where the engineer will run anti-Ackerman, in which case, the inside wheel turns less than the outside wheel.

The amount of Ackerman steer required for a low speed turn is very different than for a higher speed turn where the tires are not running on the same path as the steered angle of the tires. The correct amount of Ackerman for one turn radius will not be correct for any other turn radius. Ackerman set up, particularly on a road course, will always be a series of compromises.

The correct inner and outer steer angles for both low speed and higher speed turns are also dependent on the wheelbase, track, turn radius, tire design, car type and the talent of the driver.

The classic (slow speed turn) formula for determining the required amount of Ackerman is: Arctan(wheelbase/radius of turn at outside wheel) - if the wheel base is 107" the turn radius 1200" and the track 68", then the outside wheel needs to be steered at Arctan(110/1200) = 5.237 degrees and the inside wheel needs to steer at Arctan(110/1132) = 5.55 degrees. In this classic case, the inside wheel will require a steer angle of 0.313 degrees greater than the outside wheel. This is referred to as 100% Ackerman i.e. both tires follow the proper arc. *It's not that simple.* A chassis is said to have 100% Ackerman if lines projected perpendicularly from the front tires intersect at the same point along a line projected perpendicularly through the rear tires. The engineer must remember that the steered path a chassis follows when running at low speeds is different than at higher speeds and that net/dynamic toe (combination of static, bump, roll and Ackerman toe) needs to be set with racing speed, radius, set up and driver style in mind.

Note: The term "slow speed turn" is a bit of a misnomer. The larger the turn radius the faster the car can travel and still follow the **traditional** Ackerman path. In addition, the wider and stickier the tires the faster the car can travel and still follow the **traditional** Ackerman path. Everything needs to be looked at in relative terms.

Using a good quality scale model to study various chassis path combinations is an excellent way to visualize the effect of Ackerman on slow and higher speed turns. To accomplish this, liberally spray paint a sheet of Bristol Board™ with black paint and run the model back and forth over it until the tires are completely covered with paint. Next carefully roll the car on a new sheet of Bristol Board™. If you put an equal load on each tire by placing a finger on suspension bits near the tires while also supplying a longitudinal force, the chassis will steer in a balanced slow speed manner. Because the paint makes the tires somewhat slick, you can also simulate various combinations of understeer and oversteer by applying different loads to the four tires. The less load you place on any particular tire, the more that tire will tend to follow a path having a greater radius than if the loads were balanced.

Figures 7-20 & 7-21 shows the slow speed path of a chassis having 100% Ackerman geometry. * *All photos of this type have the car placed back at the start of the pattern.*

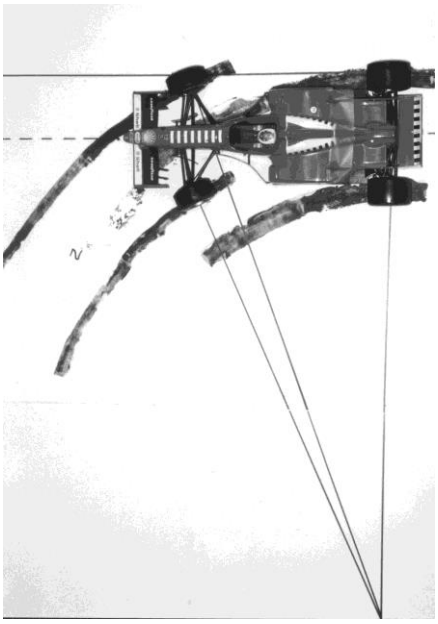


Figure 7-20

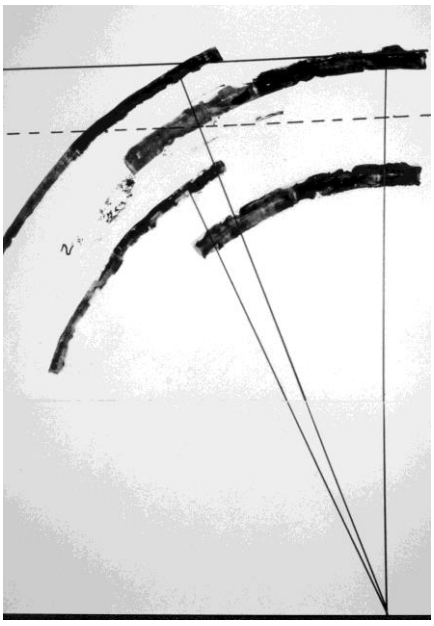


Figure 7-21

Figures 7-22 & 7-23 show the same car (same steering input and Ackerman) as in the previous example but with different loads on the tires. Notice that in this case the car does not follow as tight a path (steered path and actual path are different). The car in this second example also shows a different angle of yaw (dotted lines). Figure 7-24 shows the two examples side by side with the slow speed (chassis following the steered path) example above.

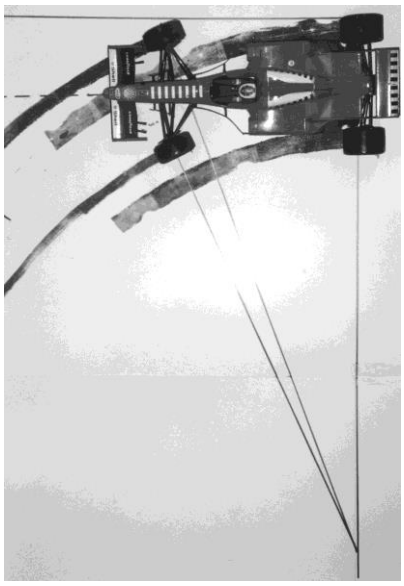


Figure 7-22

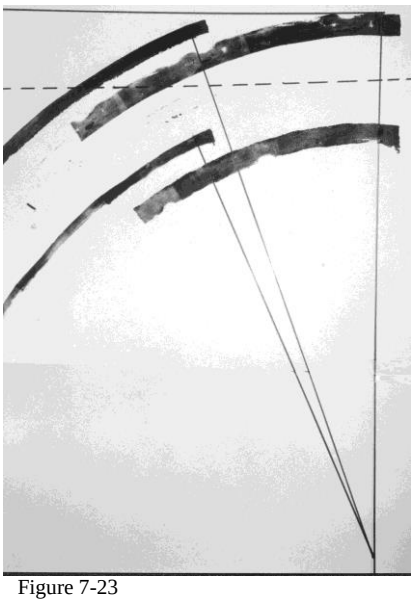


Figure 7-23

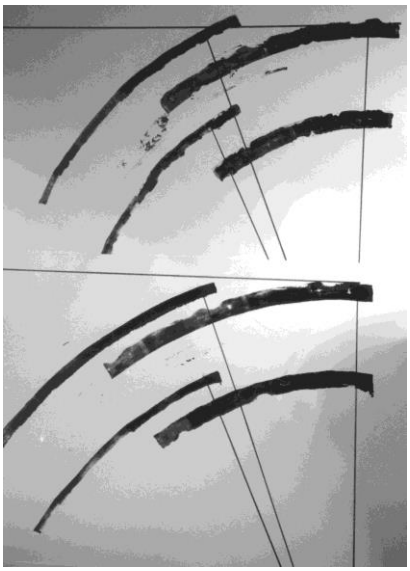


Figure 7-24

A chassis will often come from the manufacturer with a variety of steering links, which, in turn, provide numerous steering variations ranging through positive Ackerman to anti Ackerman. Running a variety of anti Ackerman set ups is far more common than might be expected, particularly when running the latest generations of radial tires through tight sharp corners. Well heeled teams may well have an arsenal of twenty or more steering links in their inventory.

Ackerman variations in consort with static toe combinations provide the race engineer with a way to tune for tracks which have a combination of open and tight corners. Though inexpensive and easy to manufacture, having access to a large number of steering links can also result in mass confusion if the team doesn't have a well-designed and documented test program. As previously stated, finding the right Ackerman combinations is almost always a mater of testing rather than through mathematical analysis.

The only way to design changes in Ackerman in consort with the combined steering system is through the use of a good **three dimensional** software program such as WinGeo3. Trying to determine how the movement(s) of each suspension component is affected with a change in the steering system is truly a mind boggling exercise. Trying to design a suspension system in two dimensions on the garage floor is simply an exercise in futility. A simple steering component change, even if only in one axis, will require the running and studying of up to a hundred different dynamic iterations. It is so, so easy to actually compromise the suspension/steering system when you think you are dialing in the new demand tweak. Remember, a change in steering angle at one corner is almost always accompanied with a load change at all four corners. It is not unreasonable to spend several days of analysis for each design change.

Determining Amount of Ackerman Steer – Aside from the traditional method of comparing the difference between the outside and inside steer angle, race engineers can also look at the intersection point co-ordinates of lines drawn through the steering arm mounting center and the center of the spindle/upright rotational axis. These co-ordinates are used as reference points which help to show the engineer the rate of steer change as the result of Ackerman - Figure 7-25. Many designers consider a point of intersection at the center of the rear axle as 100% Ackerman.

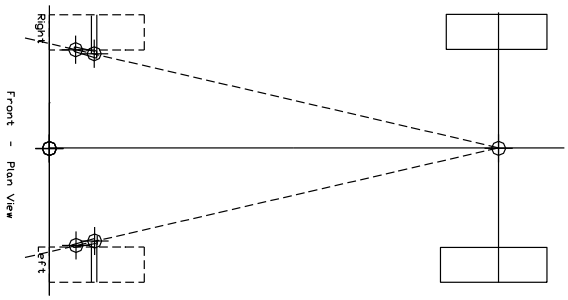


Figure 7-25

Ackerman is a favorite subject of bench racers and academics who may have few practical and/or theoretical skills. What you hear and read from many of these sources often just doesn't work on the track, especially when the person is working in the world of two dimensions (just my opinion mind you). Perhaps as we learn more about the science of tire performance, we will be able to more accurately predict the amount of Ackerman a particular steering system needs, but, until then, the best way to determine the amount of Ackerman steer the chassis wants is through carefully designed and executed testing. A change in Ackerman for one type of chassis, turn radius and/or speed may dramatically change the handling, while, for a different combination of chassis etc., there may be little on no performance change. Additionally, one driving style may benefit a great deal with a change in Ackerman while a different driving style may not benefit from the change at all. Ackerman always involves a series of compromises between individual corner performance, tire heat generation and degradation rate.

Why Ackerman - Ackerman is a more controlled form of toe-out than bump and/or roll steer toe change because it can be made to increase or decrease in direct relation to the amount the steering wheel is turned. Determining the percentage of static toe-out, bump toe-out/in, roll toe-out/in and Ackerman toe-out/in that is required is both an art and a science - testing, testing, testing.

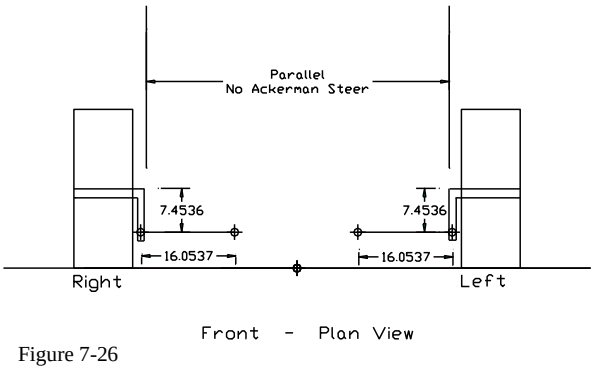


Figure 7-26

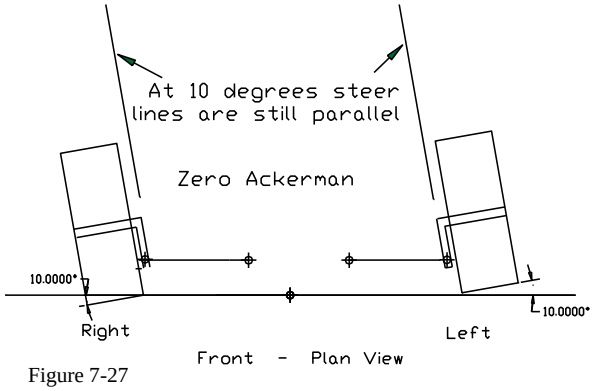


Figure 7-27

As shown in Figures 7-26 7 7-27, zero Ackerman will result if the instant center lines (lines drawn from the steering arm tie rod mounting point through the spindle center of rotation out into space) run parallel to each other at both zero steer and ten degrees of steer.

Changing Suspension Co-ordinates - The engineer must always be aware that changing virtually any suspension co-ordinate will also result in the amount of steered angle at the tire versus steering wheel input changing. For example, if you change the camber angle on the right front and then readjust the toe so it is the same as before the camber change given the same steering wheel input, the angle of steer at the tire will now be different. Making a change, any type of change, never just affects one thing.

If the steering arms on a front steer car are moved inboard of the spindle rotational axis point, the inner wheel will turn at a higher rate than the outside wheel i.e. Ackerman steer. When Ackerman is present, the instant centerlines will not run parallel with each other, but, instead, will intersect at some point aft of the spindle centerlines. Anti-Ackerman will result if the instant centerlines intersect in front of the spindle centerlines. All other things being equal, (symmetrical chassis and with the tires pointed straight ahead and zero toe), if both steering arms are at the same angle and co-ordinate positioning, their instant center lines will intersect along the chassis' longitudinal center line.

If the chassis is asymmetrical or the steering arms are at different angles, the instant center point will be offset from the chassis longitudinal centerline. In either case, once a steered angle has been introduced, the steering instant center point will be offset from the chassis centerline.

As indicated in Figure 7-28, lines extended forward along the inner surfaces of the tires diverge as the result of Ackerman steer i.e. the inner tire is angled more than the outer tire.

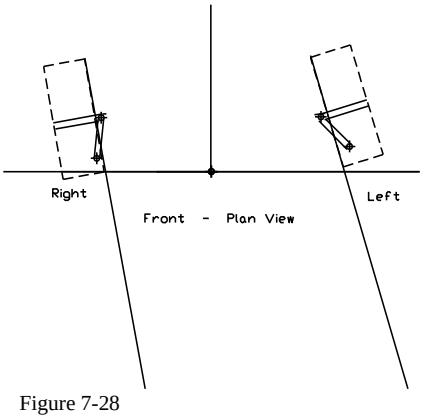


Figure 7-28

Figures 7-30 / 7-34 show examples of open wheel steering links. Notice that camber shims can be added or removed without changing the toe setting.

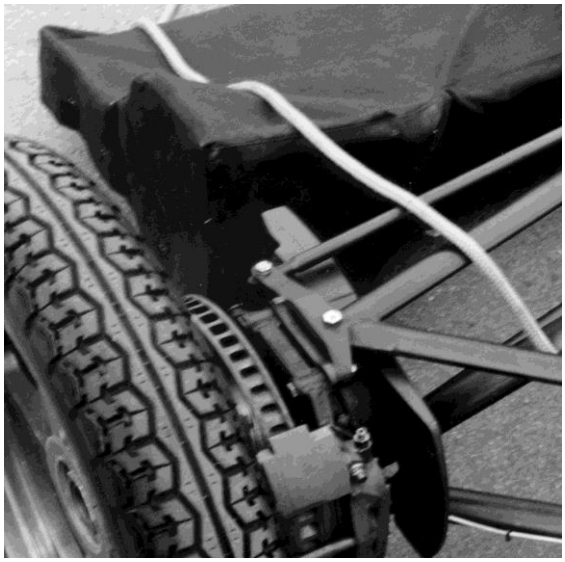


Figure 7-30

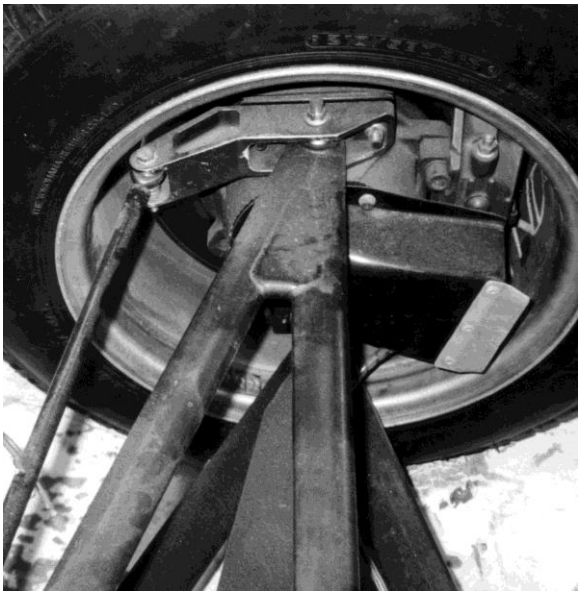


Figure 7-31

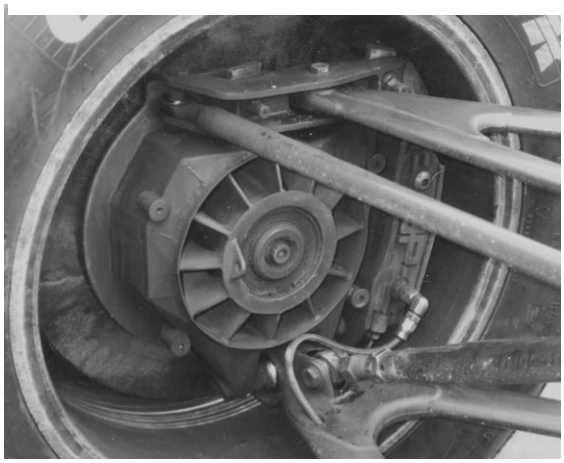


Figure 7-32



Figure 7-33

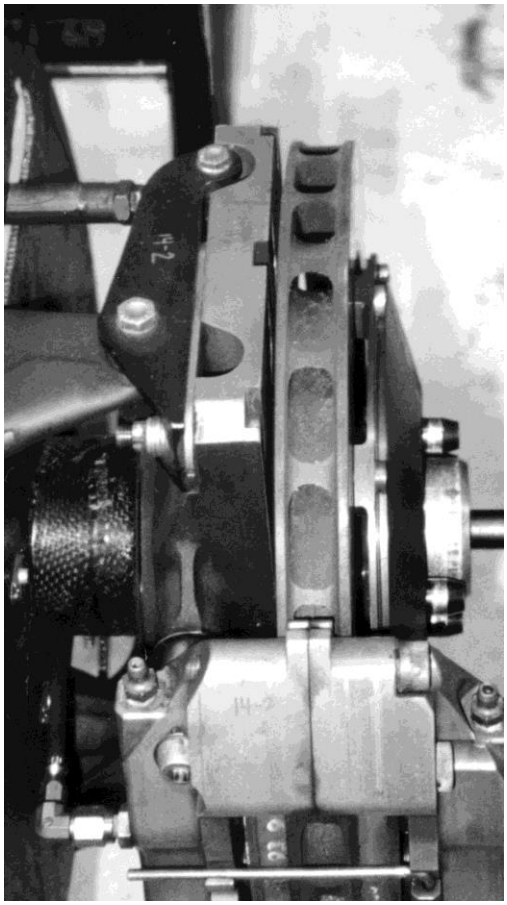


Figure 7-34

Figure 7-35 shows a front steer stock car steering arm that has anti-Ackerman i.e. the steering arm's tie rod attachment point is inboard of the spindle axis center point. This type of design is more often the result of packaging restraints than it is about the intended design.

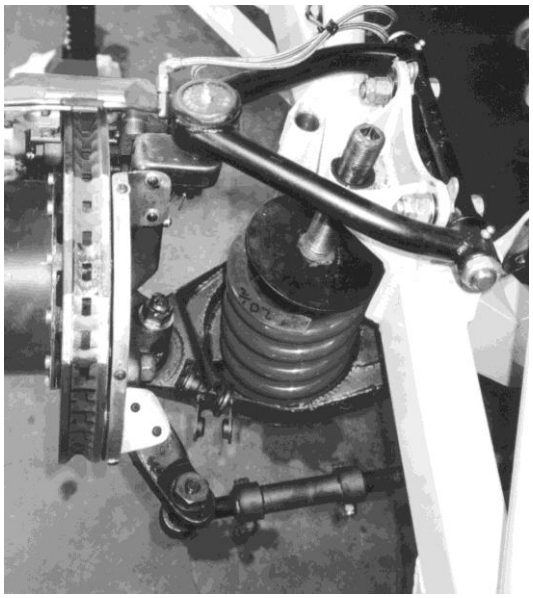


Figure 7-35

For oval track cars, Ackerman can also be increased by shortening only the left side steering arm and/or lengthening the right side steering arm, thus speeding up the steering ratio on the left side. However, shortening the left side steering arm may present some problems when making a quick right. As the car turns right, the left becomes the heavily laden tire and because it turns faster than the right tire, the result is snap toe in.

Stock cars normally run with slotted steering arms that allow plugs to be inserted in the slots. The plugs are fabricated with a variety of offset holes - Figures 7-36 & 7-37.

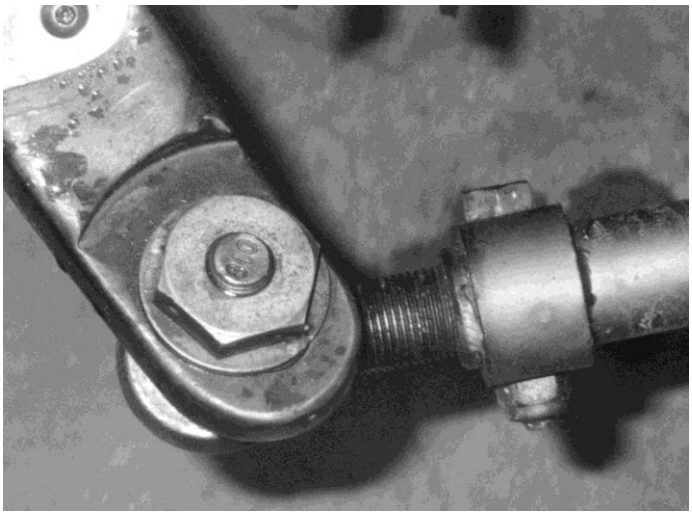


Figure 7-36

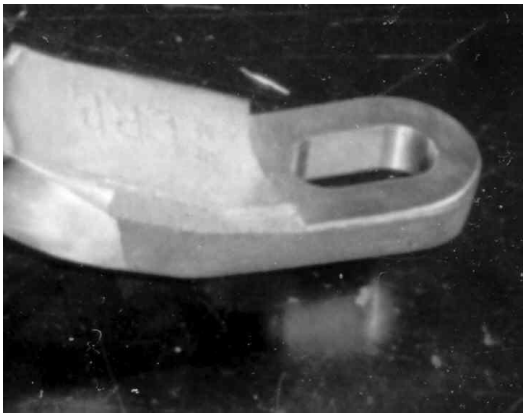


Figure 7-37

When using a drag link system, the pitman and idler arms are normally the same length running parallel with each other with the distance from the inner mounting point of the pitman arm to the outer mounting point of the idler arm the same as the distance from the outer mounting point of the pitman arm to the inner mounting point of the idler arm - Figure 7-38. If not, the steering system will exhibit some form of Ackerman steer.

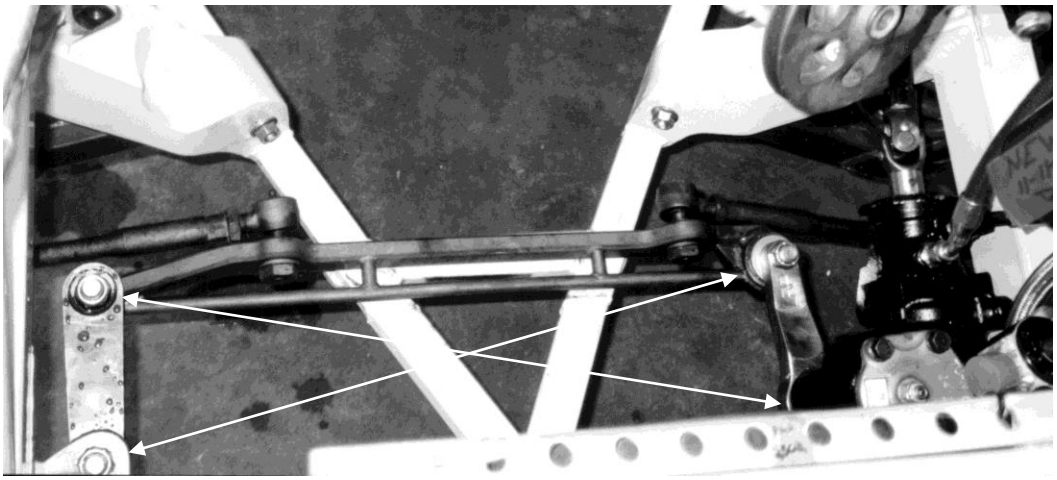


Figure 7-38

While a car is running on a banked track, the left side tires will be carrying a higher percentage of the load than they would on a flat track (due to gravitational forces). Therefore, the left side tires will produce a higher coefficient of friction (lateral forces) than possible on a flat track, which, in turn, may well change the Ackerman percentage design parameters.

When using a drag link steering system, changing the lateral position of the steering box and/or the idler arm affects the steering arcs, and, as a consequence, the amount and type of Ackerman steer. As long as the race engineer understands the resultant kinematic changes, this type of component movement offers a number of set up possibilities - Figures 7-39 / 7-41.

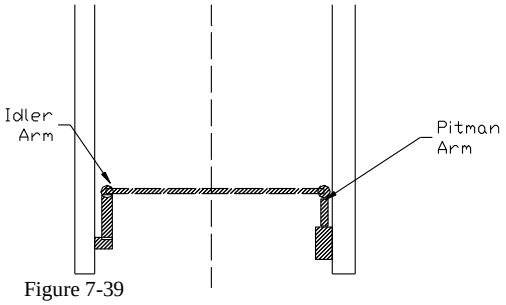


Figure 7-39

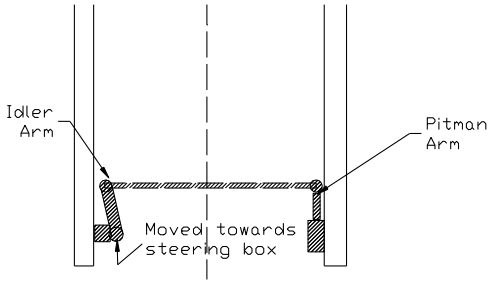


Figure 7-40

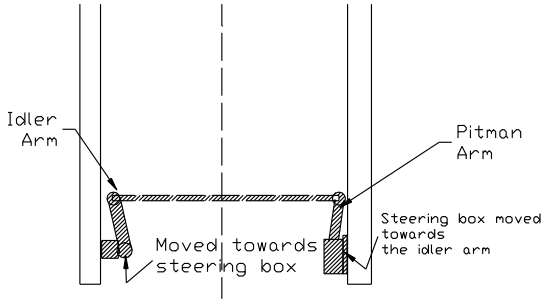


Figure 7-41

Table 7-7 is the base set up while Table 7-8 has both the left and right steering arms raised 1". Notice the difference in net steer between the two set ups.

Table 7-7

Ride and Roll Steer				Stock Car - Aiello
Steering angle = 9.134 degrees				
LF camber =3.5		RF camber = -3.5		
LF caster = 3.88		RF caster = 5.007		
Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	8.589 out	9.259 out	0.670 out
-0.5	0	9.530 out	8.223 in	1.306 out
-1	0	9.939 out	8.095 in	1.844 out
-1.5	0	10.296 out	7.992 in	2.304 out
0	0.5	8.791 out	8.287 in	0.504 out
0	1	8.496 out	8.202 in	0.294 out
0	1.5	8 .185 out	8.127 in	0.058 out
-1	0.5	9.718 out	8.032 in	1.686 out
-1	1	9.481	7.978 in	1.502 out
-1	1.5	9.227 out	7.932 in	1.295 out

Table 7-8

Ride and Roll Steer				Stock Car - Aiello
Steering angle = 9.134 degrees				
LF camber =3.5		RF camber = -3.5		
LF caster = 3.88		RF caster = 5.007		
Both left and right steering arms are raised 1"				
Ride	Roll	Left Wheel Steer	Right Wheel Steer	Net Steer
static	static	8.868 out	8.168 in	0.699 out
-0.5	0	9.105 out	8.226 in	0.879 out
-1	0	9.306 out	8.298 in	1.008 out
-1.5	0	9.472 out	8.382 in	1.090 out
0	0.5	8.720 out	8.199 in	0.521 out
0	1	8.561 out	8.236 in	0.326 out
0	1.5	8.392 out	8.277 in	0.115 out
-1	0.5	9.198 out	8.345 in	0.854 out
-1	1	9.080 out	8.395 in	0.684 out
-1	1.5	8.985 out	8.449 in	0.500 out

Important Note:

- **Steered Path, Shear Path and Actual Path** - It is true, as discussed in Chapter 3, that a tire's path, other than at low speeds, is seldom the same as the tire's steered path or even the same as the tire's contact patch shear angle; however, in order to make things a little bit simpler for this next section, the tire's path and the tire's shear angle will be considered to be one and the same. We will speak in terms of the angle the tire is turned and the actual path the tire is taking. The angle between the steered tire and the actual path of the tire is the shear angle.

The Turn Radius - When a car, with an unlocked rear end, is moving slowly (low speed turning), for example when running in the pits, insufficient force is produced at the contact patches to develop significant shear forces. In this case, the car rotates around an axis which has its instantaneous turn center point at an intersection of two lines, one drawn perpendicular to the rear tire and another drawn through the front tire at the angle the tire is turned. As can be seen in Figure 7-42, if both front tires are rotated the same amount, they produce different turn radii i.e. they both follow different slow speed paths.

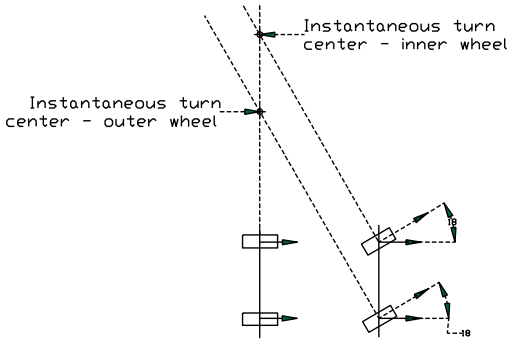


Figure 7-42

This is one reason Ackerman steer is used so that both the left and right side rotate around the same instant turn center - Figure 7-43 (slow speed turn). In order for the car to rotate around this turn center as the turn speed is increased the differential needs to be unlocked so the inside tire can rotate at a proportionately slower speed than the outside tire (no rear stagger). If the differential is locked the inside rear tire must have a proportionately smaller diameter than the outside rear tire (stagger) which, in turn, allows the inside tire to rotate at a slower rate than the outside tire.

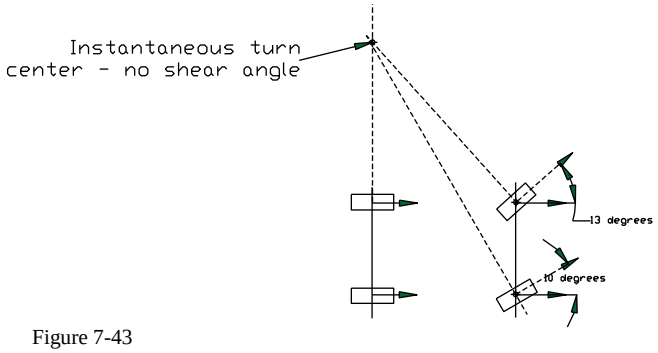


Figure 7-43

Higher Speed Turn - As discussed in Chapter 3, a shear angle is produced as the result of contact patch lateral loading**. The resultant shear angles, in turn, play a primary role in determining the car's actual path of rotation**. The shear angle of rotation is normally less than the angle of tire rotation as seen in Figure 7-44. The intersecting lines that determine the instantaneous turn centers are drawn perpendicular to the shear angles.

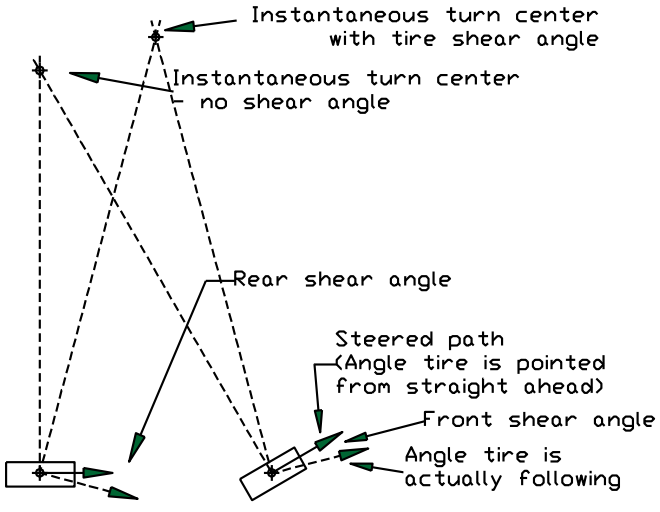


Figure 7-44

As illustrated in Figure 7-45, if both front and rear tires are operating at the same shear angle and with all other things being equal, the car will operate in a neutral mode (in equilibrium). The car will not yaw tail out or nose out (its yaw center will be close to the center of the chassis). The neutral steer arc is the arc the car would follow at slow speed or in a balanced condition at higher speeds.

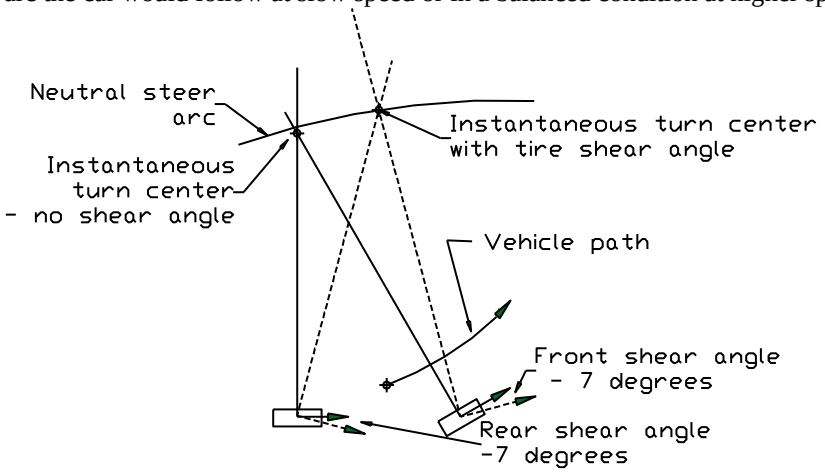


Figure 7-45

Because the outside wheels are the most heavily laden, and, therefore, hold the highest percentage of control over the car's actual path, the inside wheels are not included in the next few illustrations for the sake of simplifying the concepts. Of course when developing performance scenarios, the race engineer would need to include all four tires in his or her calculations.

As long as the front and rear tires are operating at the same shear angles, the instantaneous turn centers will intersect the neutral steer axis - Figures 7-46 & 7-47.

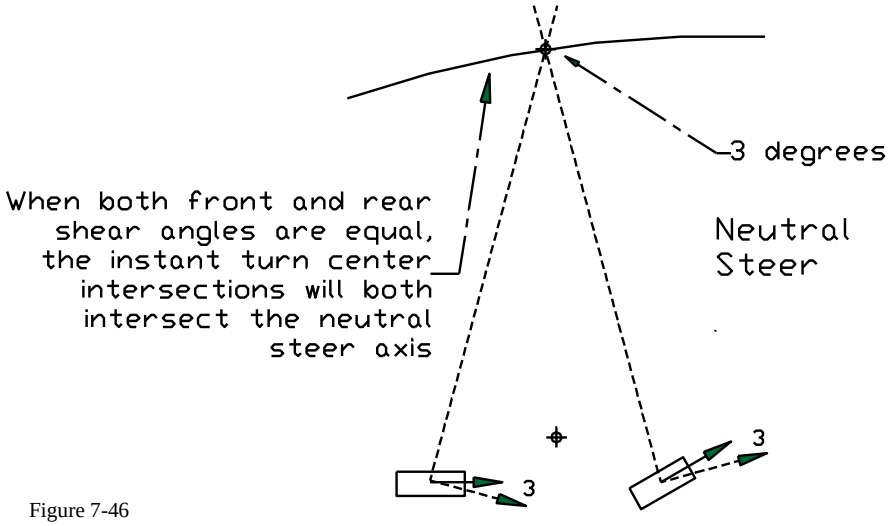


Figure 7-46

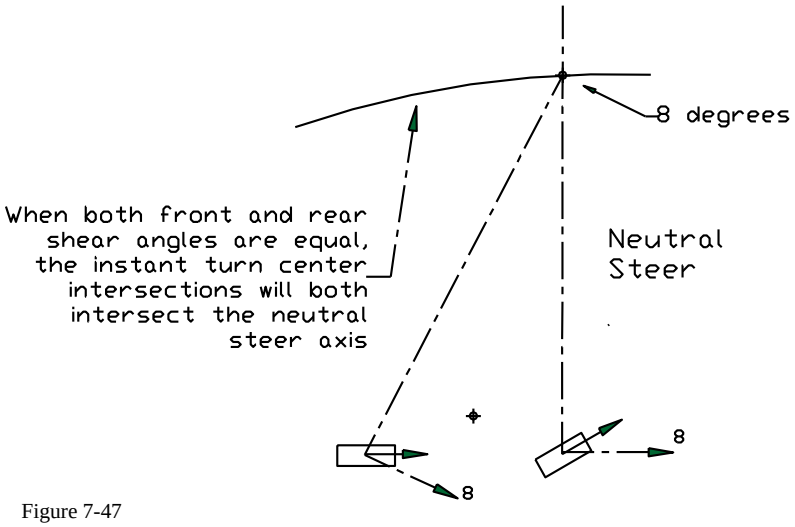


Figure 7-47

Begin WinGeo3 Example

Ackerman Analysis – One of the best ways to study the results of changing the amount of Ackerman steer is to run a series of *Path File* steer iterations and save to a *Table*.

Geometry Control Dialogue: Screen Layout – The WinGeo3 premier version has a *Special Display* option within the *Screen Layout* menu that allows the user to open three simultaneous graphic analysis screens alongside the main graphics screen. Looking at an Ackerman graphics analysis is one of the ten different options.

End WinGeo3 Example

Yaw (Rotation) at Higher Speeds - Yaw is defined as the rotational movement of the chassis about a vertical axis. This vertical axis is often described as being through the car's plan view center of gravity; at the edge of the tire performance envelope this is seldom, if ever, the case. The reaction forces may well torque about the center of gravity and the center of gravity certainly influences the extent and placement of yaw rotation but the car seldom rotates about the exact center of gravity. If the car is in an extreme oversteer (loose) condition, the yaw center will be located at some point on the center line of the front axles while moving rearward for less severe oversteer (loose) conditions. The more the car yaws tail out, the further ahead the yaw center will be. This is graphically illustrated when watching a stock car running on a one mile oval get "Bad loose".

The more efficiently a car is able to turn at speed, the further rearward its yaw center will be. For heavier cars such as stock cars and GT cars, the peak center of yaw at higher speeds will be near the longitudinal center of the chassis. At the other extreme, are the front running Formula 1 cars using high down force set ups while running through tight fast turns. They will have their yaw centers much nearer the rear axle. Watching the fastest Formula 1 cars compared to the back markers is quite revealing. The faster a car can turn in a controlled manner, the further rearward its balanced center of yaw (rotation) will be. Again, slow speed turns and higher speed turns are two very different things.

Action and Reaction - A Fundamental Point

Primary Action and Reaction Forces and Secondary Action and Reaction Forces - It seems there are as many opinions as there are race engineers as to whether a force at any particular place, at any particular time is an action force or a reaction force. Under static conditions, if an action force pushes against an object the body reacts with an equal and opposite reaction force. However, if there are two objects, one touching the other, the situation changes slightly. As seen in Figure 7-48, Box A and Box B are touching. If a primary action force is applied to Box A, it, in turn, reacts with a secondary action force on Box B. Box B pushes back against box A with an equal and opposite secondary reaction force, and, in turn, Box A reacts against the primary action forces with a primary reaction force. *Of course, it is not that simple.*

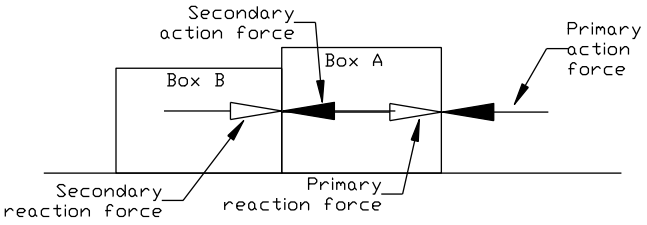


Figure 7-48

This same situation exists under dynamic chassis conditions. In this case, the **primary action forces** move through the steered front tires into the sprung mass. If the **primary reaction forces** at the front tires are less than the **primary action forces,** the front tires will begin to accelerate towards the center of their arc. At about the same time, the sprung mass also reacts against the rear tires (**secondary action force**) and the rear tires, in turn, react against this force (**secondary reaction force**). If the rear tires were on ball bearings, no secondary reaction force would develop and as a result, the chassis would tend to yaw about an instant yaw center somewhere along the front axle centerline.

The sprung mass forces are seen as **primary reaction forces** in front of the instant yaw center. Rearward of this instant yaw center, the sprung mass forces are seen as **secondary action forces** acting on the rear tires.

- Front primary action forces try to turn the car away from its original path and into the steered tires' intended path
- Front primary reaction forces try to prevent the front of the car from turning (changing its direction of travel i.e. no front lateral (centripetal) acceleration)
- Rear secondary action forces try to push the rear tires' tail out
- Rear secondary reaction forces try to keep the rear tires from moving tail out i.e. prevent rear lateral (centrifugal) acceleration.

Figure 4-49 illustrates a chassis whose front primary action and reaction forces are balanced and whose rear secondary action and reaction forces are balanced. The chassis will neither laterally accelerate or yaw. The front outside action and reaction forces don't have to balance as long as the combined front action versus reaction forces balance. The same is true for the rear forces.

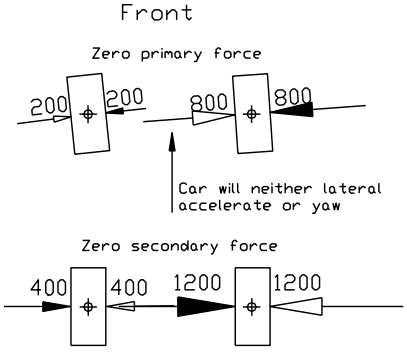


Figure 7-49

If either the front and/or rear action forces are greater than the front and/or rear reaction forces, the chassis will have a tendency to yaw (rotate). The instant yaw center position is determined by the amount of imbalance between the action and reaction forces at each end of the chassis. In the extreme, if the rear tires develop no reaction forces, (the ball bearing example) the instant yaw center will be at a point along the front axle center line. If, on the other hand, the secondary action and reaction forces at the rear tires are balanced while the front tires have a much greater net action force than reaction force, the chassis will tend to yaw about a yaw instant center much nearer the rear axle center line. If the action to reaction imbalance is equal front and rear, the chassis will tend to rotate about an instant yaw center close to the center of the chassis.

Both the lateral and longitudinal positioning of the instant yaw center are constantly changing in relation to the action/reaction forces at each contact patch. Abrupt force changes will also result in abrupt yaw center changes. *Key tuning tool*.

The two primary points to remember here are that without an imbalance between action and reaction forces, the chassis will not yaw, rotate or turn and that the action force at the front of the chassis are directed towards the turn center while the action forces at the rear are directed away from the center of the turn. *Key tuning tools*. Figures 7-50 & 7-51 show chassis with varying action and reaction imbalances at the four tires. Note the change in the instant yaw centers. Remember, action forces at the front point into the turn while action forces at the rear point away from the turn.

In order for a race car to successfully negotiate a curve at speed, the understeer/push and oversteer/loose conditions will need to be only momentary in nature. If either condition continues unchecked, the car will not be able to follow the required arc. If, for example, rear imbalance causes a decrease in rear lateral force and a subsequent increase in yaw angle with the yaw center moving forwards, to re-establish equilibrium, the higher secondary reaction force needs to match the secondary action force. High performance driving and chassis set up is all about running that fine line between understeer and oversteer. At peak performance levels, even a so called balanced chassis will move back and forth between oversteer and understeer many, many times during each corner. The ability to stay on that knife edge (not allow the chassis to move too far into understeer or oversteer) while maintaining maximum speed is what separates the good drivers from the great drivers.

In Figure 7-50 the front tires have a net turning force of 200 while the rear tires produce zero turning force. If this chassis was using a locked rear differential, the rear tires would have a greater tendency to keep the car moving in a straight line. If the differential was open, the car would rotate much the same as in the low speed models. Using an open differential, this car would turn in well. The driver would need to adjust the steer angle to keep the car on the required arc. If the front tires were set at too great a steer angle, the chassis would scribe too tight an arc but the chassis would not be considered loose. If both the outer and inner tires produced a higher primary action force than primary reaction force, the car would turn in even faster.

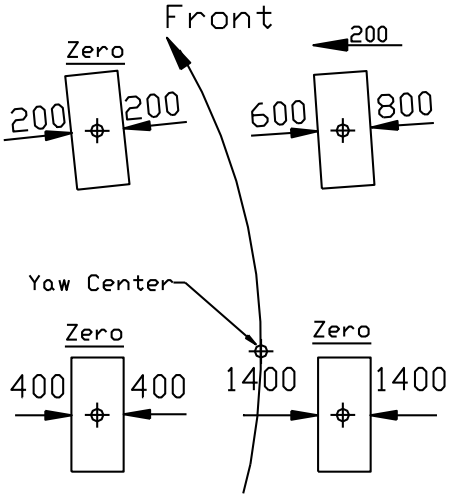


Figure 7-50

Figure 7-51 illustrates a chassis in a slightly loose condition because the tail out forces are greater than the front in forces.

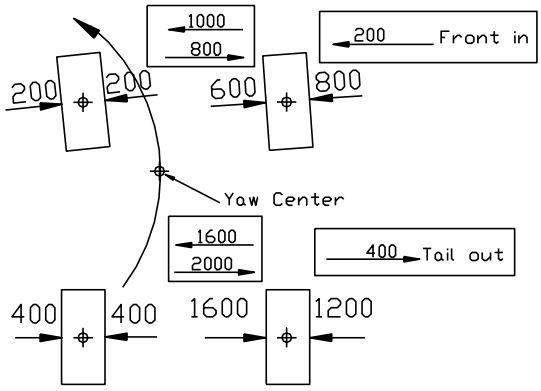


Figure 7-51

Yaw Observation - The difference in the yaw position between a balanced car and one with a slight amount of understeer or oversteer is very subtle, but, with practice, one is able to observe a car's rotation points at various speeds and levels of balance. Running a video tape of a car cornering in a known condition, back and forth, especially if the car is taped from the top of the stands, on a hill, or, best of all from a helicopter, is a very good way to acquire the skills to observe yaw and yaw placement. Determining the reasons the car is rotating at a given point is many times more difficult.

One of the best methods of studying understeer and oversteer conditions is using a model car while placing Scotch Tape™ over various combinations of the tires. Make sure the model's tires are soft enough to provide the needed lateral forces. If you want to look at a loose condition, simply tape one or both of the rear wheels. Propelling the car at a variety of forces while maintaining a constant steer angle changes the amount the chassis understeers or oversteers in addition to changing the center of yaw.

Cars running asymmetric suspension and weight percentages will have their yaw centers laterally offset from the longitudinal center line of the chassis.

Tracking - Oversteer/Loose Condition at Speed - Figures 7-52 & 7-53 show a mild oversteer condition. Notice that the rear wheels don't move beyond the line representing the straight ahead condition but the rear wheel paths are quite a bit closer to the front wheel paths than in the classic slow speed turn shown in Figure 7-20 & 21.

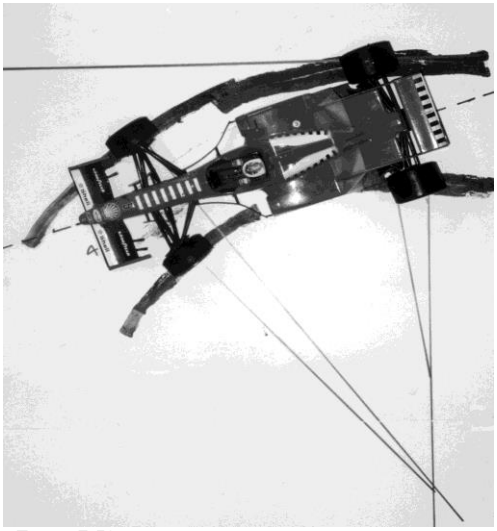


Figure 7-52

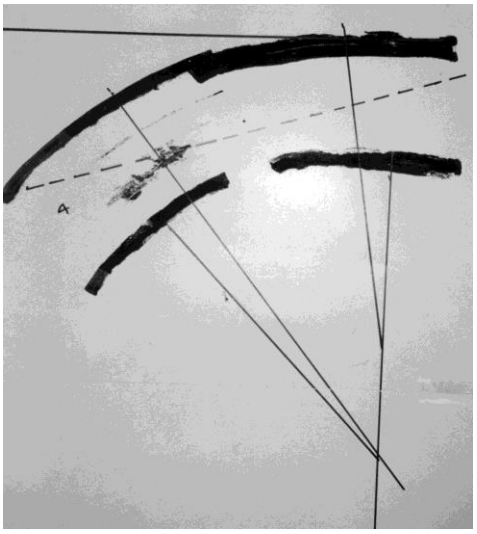


Figure 7-53

Figures 7-54 & 7-55 illustrate a more severe oversteer condition. Notice that the rear wheels move beyond the line representing the straight ahead condition and the rear wheels run in an arc outside the front tires. Also notice that at this point, the tangent of the rear tire arcs are no longer running parallel to each other. We are now into accident reconstruction.

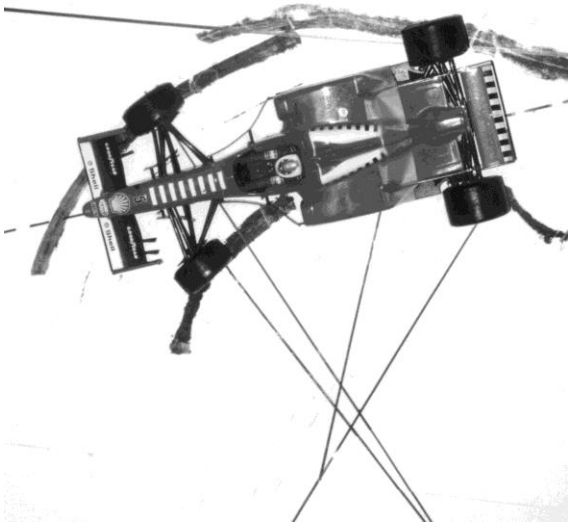


Figure 7-54

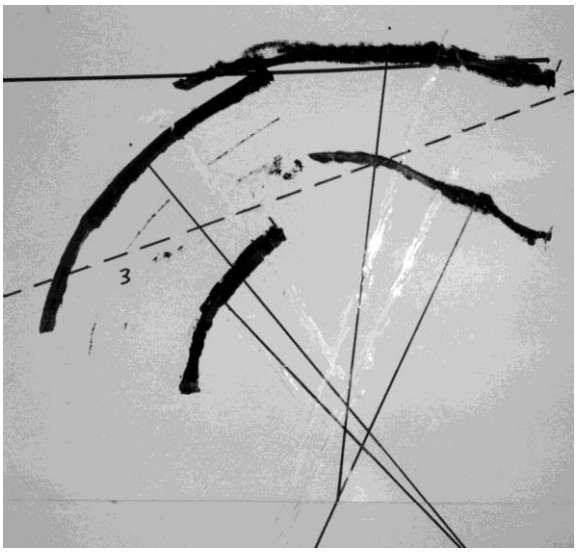


Figure 7-55

Figures 7-56 & 7-57 illustrate an understeer/push condition.

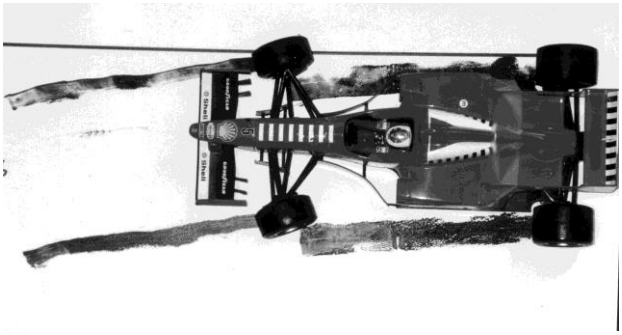


Figure 7-56

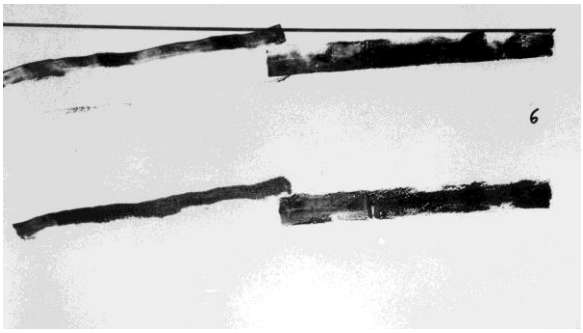


Figure 7-57

Figures 7-58 & 59 show a typical condition where the chassis is both understeering/pushing and at the same time, indicates a slightly loose/oversteering condition at the rear. This type of pattern can be the result of a number of different conditions, which, when looking at the acquisition data can be quite difficult to analyze. It may be the result of the car drifting, the front and rear tire sliding either at the same time or alternating back and forth, or, with a talented driver, it is most likely a case of the driver balancing the car between understeer and oversteer during corner exit if the car was exhibiting a tendency to understeer at the corner apex. This is the opposite of a less than talented driver who continues to apply more steering angle to the car at the apex to try and overcome a push condition only to find the car in a snap loose (oversteer) condition as soon as the power is applied. Nine times out of ten, the less talented driver will only report that, “The car goes snap loose when I try to put the power down”, which means they have not figured out that it only goes loose because of the power being applied. It goes loose because the wheels are steered too much, and, as a result, as weight is transferred to the rear, the front wheel loads begin to balance out and the rear tires lose some of their grip, (because of inline acceleration) the front wheels suddenly are able to produce a higher coefficient of friction (they stick), and, voila, the back end wants to come around.

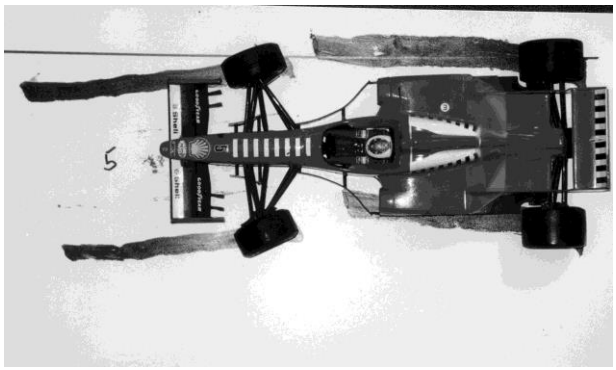


Figure 7-58

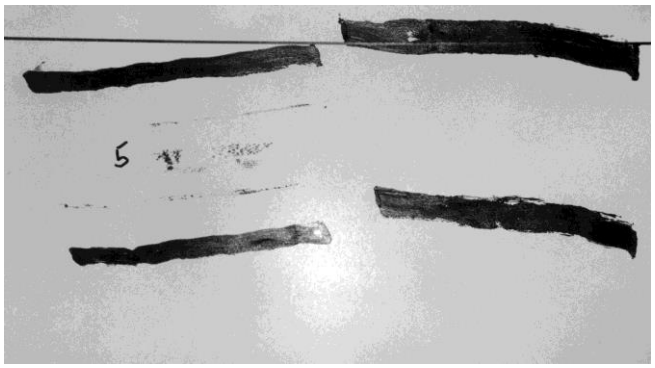


Figure 7-59

Modifying the model so you can change the front and rear track as well as changing the amount of Ackerman steer is extremely informative but it does nothing for the model. Figure 7-60 illustrates the outside tire paths with the original rear wheel base, while Figure 7-61 illustrates the tire paths of a wider rear wheel base. Over the years, I have made hundreds of these tire path models and every single one of them has taught me something. What we think a particular path will look like and what it actually looks like is often quite different.

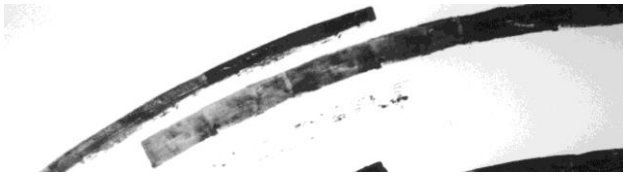


Figure 7-60

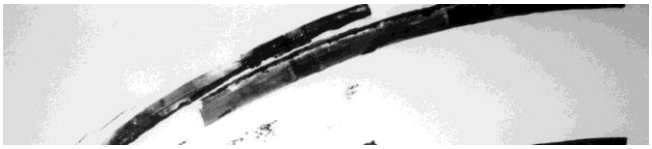


Figure 7-61

Yaw Center Illustrations - When learning how to observe a car's handling characteristics, I have found it useful to place a black dot representing the yaw center on a model car and move it back and forth along either a straight line or a representative line of the various turn radius scenarios - Figures 7-62 / 7-66. Try doing this while watching videos of a race or test sessions. Watching test session videos are particularly useful because you often know from driver debriefing and data acquisition what the car was doing.

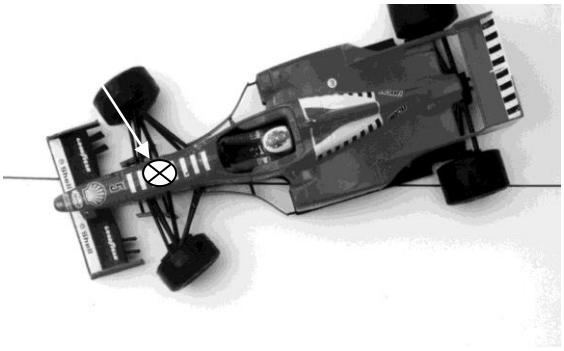


Figure 7-62

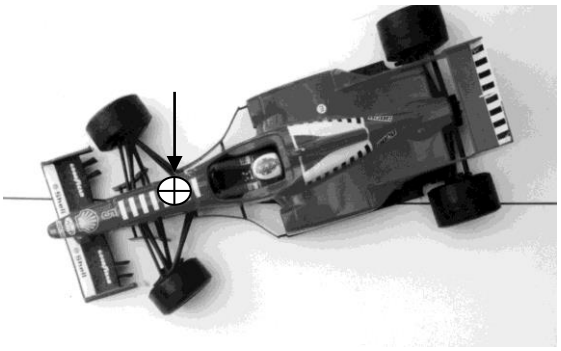


Figure 7-63

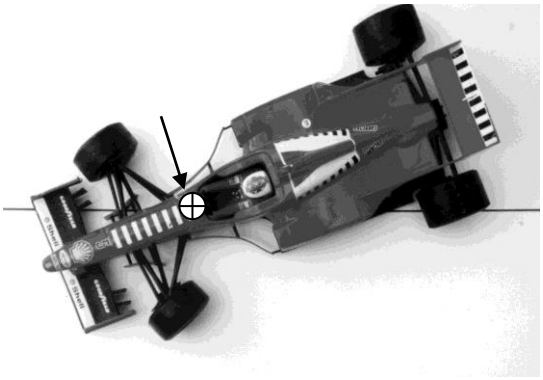


Figure 7-64

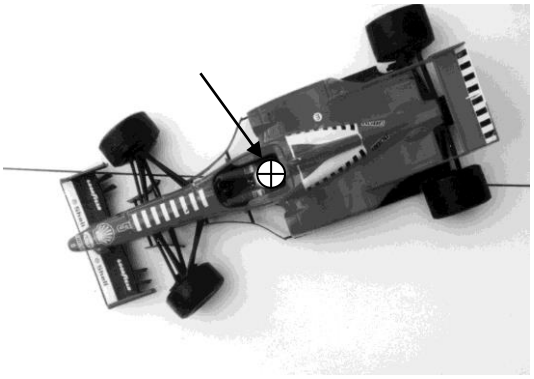


Figure 7-65

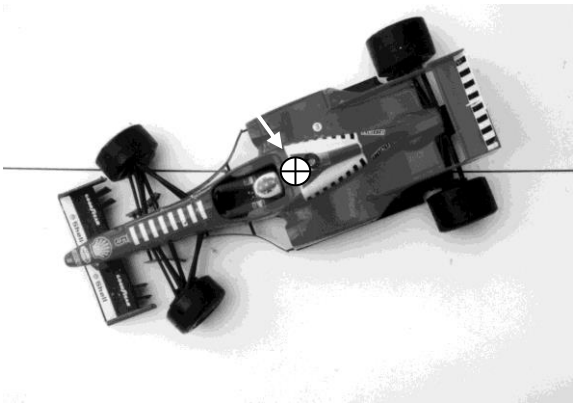


Figure 7-65

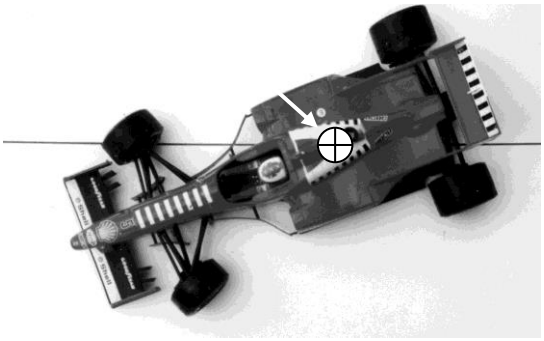


Figure 7-66

Figures 7-67 & 7-68 illustrate yaw centers that are offset from the chassis' lateral centerline.

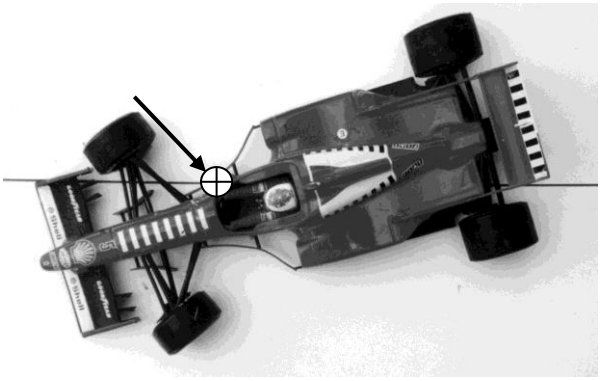


Figure 7-67

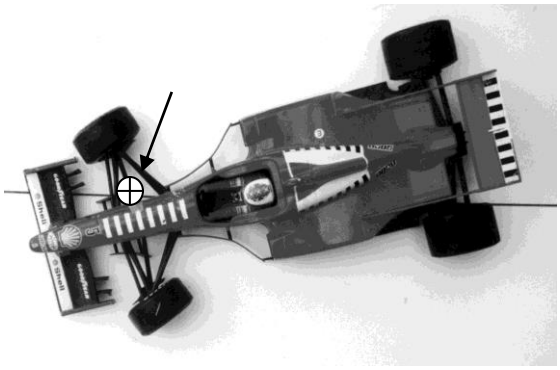


Figure 7-68

Yaw Centers - Figure 7-69 illustrates a model car running in circles on a large flat surface propelled by a string. If the string is placed at the mass center point between the front and rear wheels, the car is balanced. During a steady state turn, the chassis maintains a constant yaw angle i.e. there is no yaw acceleration. If the string is ahead of the mass center point, the car runs with a tail out (oversteer) attitude.

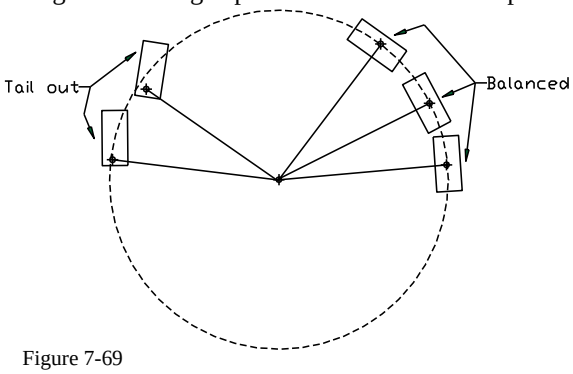


Figure 7-69

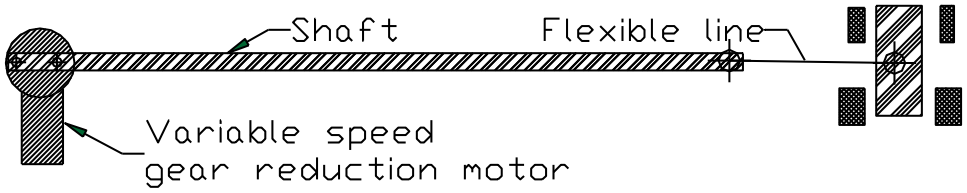


Figure 7-70

An even better way to study yaw angles when using a model is with a variable speed gear reduction motor as shown in Figure 7-70. By affixing the line in various places, you can stand back and watch any number of yaw iterations. Using a good quality model, such as the F1 model as shown in previous pages, you are also able to change the track and put Scotch Tape™ over various tire combinations - very informative.

Developing Yaw - As previously stated, in order for yaw to take place, a lateral force must be developed, and, in order for this to take place, (other than for wind gusts or other sideway forces such as large moving objects) one or more of the tire contact patches must run at a shear angle.

Anything that generates yaw can be defined as having a steering effect. Steering, therefore, can be defined as the control of yaw moments acting about the vertical yaw axis. A yaw moment is not necessarily generated with a change in steering angle (rotating the steering wheel). However, if the chassis exhibits yaw acceleration, it will almost certainly be the result of shear angle at one or more of the tires, unless, of course, all four tires are sliding with virtually no coefficient of friction i.e. the car is spinning out of control.

Tires - Drag and Shear - Even when a car is running in a straight line and the wheels are pointed straight ahead, there will still be a certain amount of drag (friction) produced at the tire contact patches. This can be the result of component rotational friction from the engine, drive train, hubs, etc., aerodynamic drag, in addition to the tire's coefficient of rolling friction which may not be balanced side to side. Complicating matters further is the fact that a race car virtually never runs in a straight line for more than a few milliseconds at a time due to the fact that it is constantly being bombarded by internal and external forces such as suspension compliance, brake drag, track irregularities, wind gusts, driver input etc. Each time the driver makes a lateral or longitudinal correction, the tires will begin to develop a shear angle and shear angles, by their nature, produce scrub. Scrub means that some portion of the contact patch is running - pushing - at a angle different than the angle of tire rotation. Scrub, as discussed in Chapter Three, produces a type of friction referred to as drag. A higher percentage of total left or right side drag in itself will produce chassis yaw.

Yaw Equilibrium - Yaw equilibrium (constant yaw velocity) is achieved in three ways:

- Straight line travel - no yaw (no angular rotation of the chassis)
- Steady state cornering - constant speed and constant radius
- Increasing or decreasing speed and turn radius at the same rate

Yaw - If the chassis is not in one of these three modes, it will be exhibiting yaw angle acceleration.

If the action and reaction forces are equal front to rear, the chassis is in a steady state condition - no yaw acceleration - Figure 7-71. However, during a normal cornering attitude, the chassis will almost always have the tail out, be it ever so slightly, even in an understeer condition. This is a very important point.

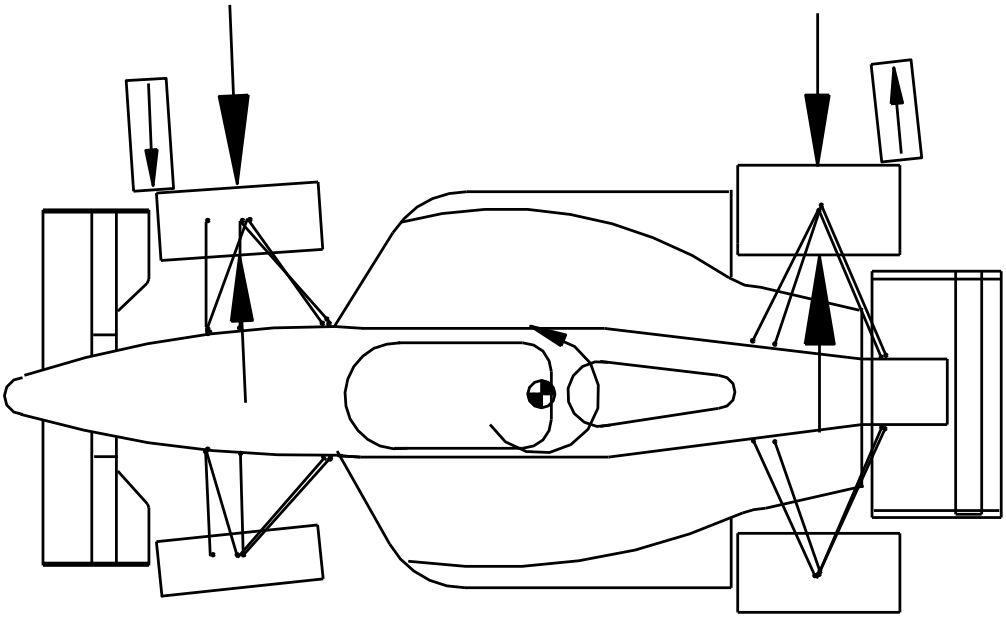


Figure 7-71

Oversteer - When the rear of the chassis wants to come around further than the front of the chassis wants to turn in, we say the chassis is oversteering or loose - Figure 7-72.

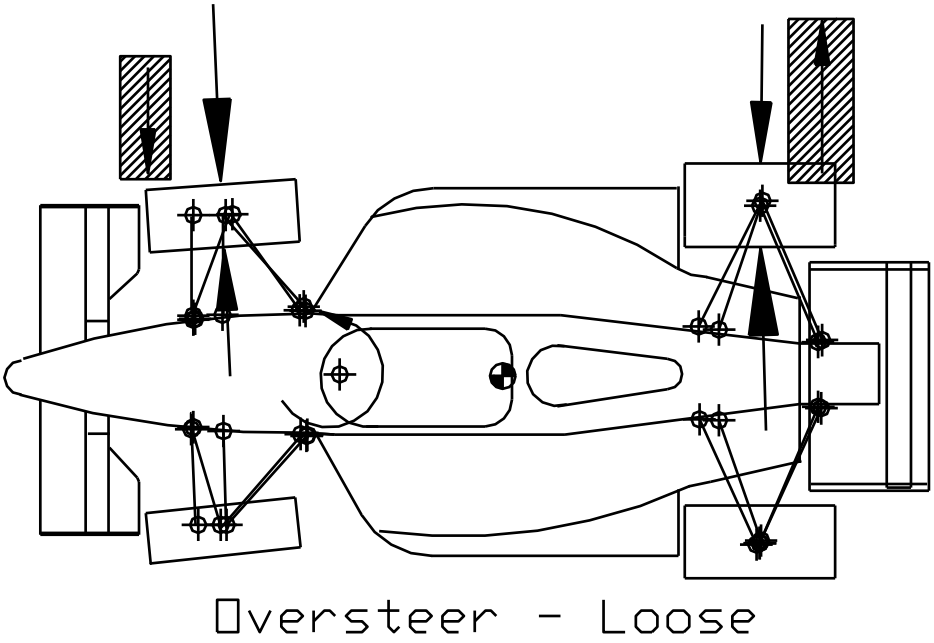


Figure 7-72

The theoretical turn radius in the oversteer mode is smaller than the neutral turn radius - Figure 7-73.

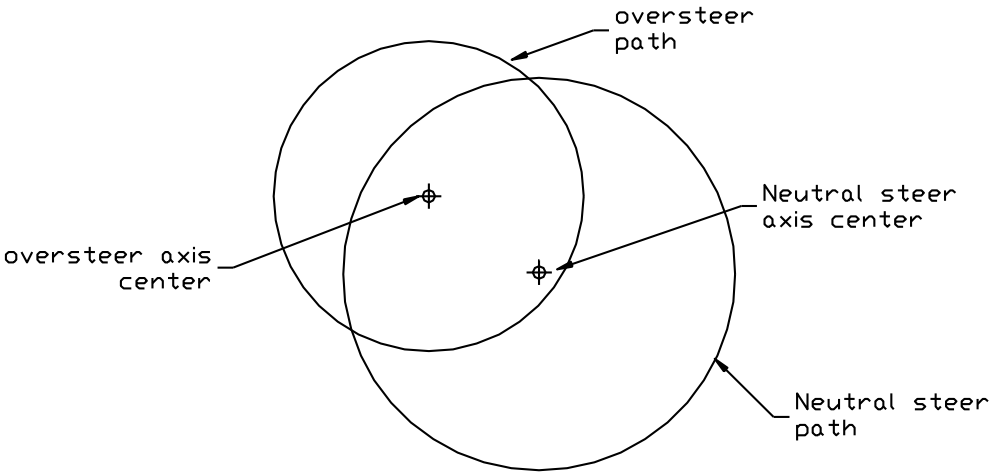


Figure 7-73

Understeer - The understeer instant axis is on the far side of the neutral turn axis - Figure 7-74. The greater the understeer, the further the yaw (rotation) center moves away from the front axle line and back towards the center of gravity. In this example, even though the action and reaction forces at the rear tire are much greater than at the front tire, both the front and rear action and reaction forces are balanced i.e. the car does not want to turn.

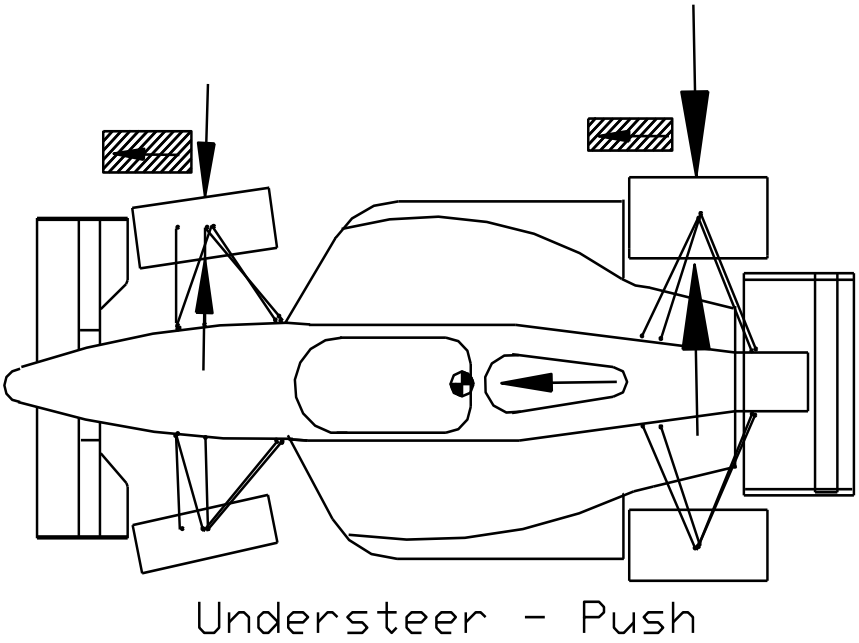


Figure 7-74

The understeer axis is larger than the neutral steer axis - Figure 7-75.

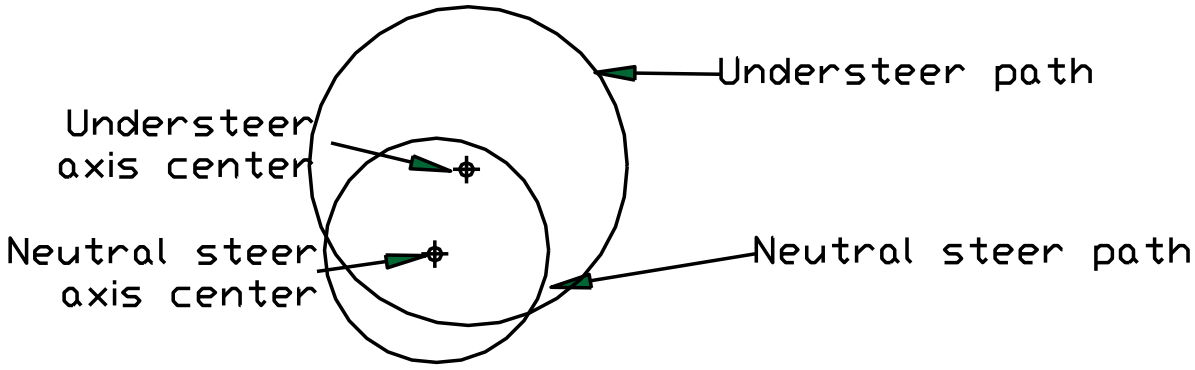


Figure 7-75

There are tens of thousands of combinations that will determine whether a chassis is balanced, understeering or oversteering. Getting the right set up is not just about whether the car is balanced or not but also how stable it is for any particular condition. For example, a car that is slightly on the oversteer side may well end up being a faster race car if the amount of oversteer stays consistent rather than a car that flips back and forth between balanced and unbalanced with only a few pounds load difference on any particular wheel. *Key tuning tool*.