11

Anti Roll Bars and Motion Ratios

(*It's just not this simple!)*

Anti Roll Bars

Anti Roll Bars/Sway Bars – Anti roll bars are used to control the amount of body roll (chassis roll angle per amount of lateral force) and to change the roll rate/weight transfer between the front and rear of the car (cornering balance of the car). Anti roll bars primarily affect the wheel roll rate but if the chassis is asymmetric, the bar can also affect the wheel rate in ride.

Right Wheel Bump - If the car sees right wheel bump due to lateral acceleration, the chassis will tend to roll to the right (no left side droop). As the body attempts to roll, the right side anti roll bar arm is twisted up. If the left side arm had no load/resistance to motion, the left arm would move up the same amount as the right side and the bar/torque tube would not twist. However, in most situations, the left side arm is resisted by the weight of the left side unsprung components and the force of the left side spring compressed load.

If the anti roll bar was infinitely stiff and the left and right motion ratios were the same, the left side suspension components, including the wheel, would be lifted the same amount as the right side anti roll bar arm was moved up. As load is added to the outside wheel, an equal amount of load will be removed from the inside wheel.

If, on the other hand, the torsional stiffness of the bar is less than the combined inside unsprung weight and the spring load, as the outside suspension moves into bump and the outside anti roll bar arm is forced up, rather than simply lifting the inside arm, the bar will twist in relation to the applied load and the bar's angular spring rate. A soft bar will twist more than a stiffer bar for a given amount of applied load just as a softer spring will compress more than a stiffer spring for a given load.

The most important concepts to remember about anti roll bars are:

- That as the result of the bar being twisted/loaded, the outside wheel will gain load and the inside wheel will lose load.
- That no matter how high the bar's wheel rate is, the load seen by the torque tube (twisted portion of the anti roll bar) can not be greater than the inside unsprung weight and the spring's compressed load.

Right/Left Movement - In virtually all dynamic situations, both the left and right side suspensions will see movement either in opposite directions or the same direction but with different amounts of motion. The amount of anti roll bar angular twist is the result of the difference between the left and right side motions.

Angular Rate Versus Linear Rate - Angular rate is the force necessary to twist the torque tube/torsion bar a given amount. Linear rate is the distance the pickup point of the anti roll arm is moved up or down when measured at right angles to the axis of the bar.

Shear Modulus (G) Versus Tensile Modulus (E) - When calculating the angular rate of a bar, be sure you are using the shear modulus (G) which is several times lower than the tensile modulus (E). The shear

modulus for E-4130 (chromemoly) ranges between $11.4*10^6$ and $11.7*10^6$ while the tensile modulus for E-4130 is close to $30*10^6$.
Note: The following angular and linear bar rate calculations are approximate at best.
To precisely rate an anti roll bar, the bar must be tested in an accurate test apparatus.

Angular Rate – Tube - Angular rate of a tube in radians is found by the following formula:

$K = G*(OD^4 - ID^4)/l$
K = bar spring rate in force per radian
G = shear modulus G = modulus* Pi/32
OD = outside diameter of the tube
ID = inside diameter of the tube
l = length of tube/bar - End to end of bar versus distance between the mounting bushings

Angular Rate - Solid Bar - To calculate the angular rate of a solid bar use $K = G*OD^4/l$

Because the OD and ID are taken to the fourth power it is critical that they are measured accurately. Additionally, due to the 4^{th} power of the diameter in the equation, it can be deduced that a hollow bar will be stiffer for the same weight.

Angular Rate Calculation:
OD =1"
ID = 0.75"
$G = 11.5*10^6*Pi/32 = 1129010$
l = 46"

Tube - $K = 1129010 *(1^4-0.75^4)/46 = 16777.92$ lb. force per radian twist or 16777.92 / 57.29 = 292.86 lb. per degree of twist

Bar - $K = 1129010*(1^4)/46 = 24543.7$ lb. force per radian twist or 24543.7 / 57.29 = 428.41 lb. per degree of twist

Linear Rate: Linear rate of a bar per inch of arm deflection is found by using the following formula $K = G*(OD^4-ID^4)/l*(a^2)$ where a = arm length.

Linear Rate calculation:
OD =1"
ID = 0.75"
$G = 11.5*10^6*Pi/32 = 1129010$
l = 46"
a = 10.05"

Tube - $K = 1129010 *(1^4-0.75^4)/(46*10.05^2) = 166.11$ lb. force per inch of arm deflection

Bar - $K = 1129010*(1^4)/(46*10.05^2) = 243.00$ lb. force per inch of arm deflection

Solid Versus Hollow Torque Tubes/Torsion Bars - During linear/strain distortion, "the displacement of the fibers increases with the distance that the individual fiber is located from the center of the bar. In the linear sense, the outer fibers are stretched a much greater distance than the inner ones. Since stress and strain don't know about angular displacement, linear is all that counts, and, in any torsionally loaded shaft, the inner fibers are along for the ride."[1] As shown in Figure 11-1, the solid shaft on the left has an unequal distribution of stress during torsional loading while the tube on the right has a more uniform distribution of stress. Using a hollow bar rather than a solid bar considerably reduces the weight of the bar. Spend time working with the Excel anti roll bar calculators to understand the differences in angular rate between a solid and hollow tube.

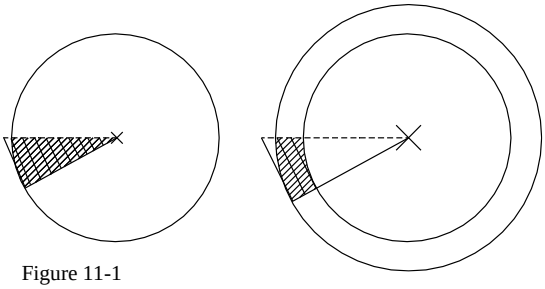


Figure 11-1

During outside wheel bump, the inside anti roll bar arm does not move appreciably until the anti roll bar forces are greater than the inside unsprung load/weight.

Actual Anti Roll Bar Rates - Rating an anti roll bar should be more straightforward than rating a helical spring, but it isn't. In fact, the often quoted line, " A disagreeable fact of racing is that we need to interact with the world as it is, not as we have calculated it to be,"[2] comes to mind when working with anti roll bars.

Anti roll bars are most often made from a steel alloy containing a carbon content anywhere between 0.28 to more than 0.43 percent. The most common steels used in torque tubes/torsion bars are 4130, 4140 and 4340. Though all of these steels have a fairly similar shear/torsion modulus (G), they will have very different yield points/elastic limits depending on how they have been heat treated and tempered. A torsion bar at Rockwell C22 will have a much lower elastic limit and yield point than the same bar at C38 - Figure 11-2. The yield point is higher and the amount of deformation (twist) within the elastic limit is greater for higher Rockwell numbers. Torque tubes/torsion bars must be designed for the proper yield and elastic range. *It's not that simple*.

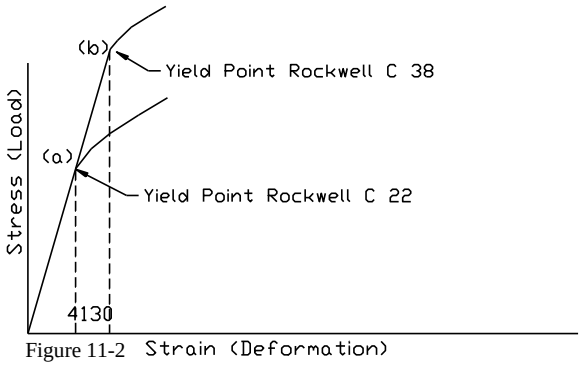


Figure 11-2

One of the many problems in accurately calculating the angular and/or linear rate of an anti roll bar is the variance in composition percentages for even AMS 2301 rated E-4130 chromemoly. For example, the carbon content can be between 0.28 and 0.33, manganese content between 0.40 and 0.60, chromium content between 0.80 and 1.10 and molybdenum content between 0.15 and 0.25, etc. These composition differences affect both the shear modulus (G) to a small extent, and, most importantly, the resultant effects of heat treating and tempering. Remember, there is a relationship between hardness and yield strength.

[1] Carroll Smith, Engineer to Win, Aero Publishers, Inc., 329 West Aviation Road, Fallbrook, CA 92028

[2] Author Unknown

The anti roll bar arms are seldom loaded in pure bending which means they can also be subjected to various axial loads and forces, which, in turn, affect the final linear rate.

The method by which the bar is affixed to the chassis is also a source of considerable rate change due to binding and misalignment forces. Stiction and/or friction between the bar and the mounting bushing can result in abrupt changes in bar rate seen by the outside and/or inside wheel. Stiction is defined as extra or increased friction force at zero or low velocities.

If blades are used in place of the arms, the blade spring rates in bending also need to be determined in one degree iterations from their soft to hard settings. In this case, the anti roll bar is a combination of two springs acting in series. The first spring is the blade(s) which transfer(s) the motion of the suspension to the torsional rotation of the roll bar. The second spring is the torsion bar/torque tube which produces resistance per degree of twist/rotation.

The Bottom Line - Anti roll bars in conjunction with their arms/blades and mounting systems need to be tested using accurate test apparatus. Like the spring tester previously described, anti roll bar testers use massive plates for all mechanical grounds. Book Two shows the construction of three different anti roll bar testers. Testing bar rates using the arms and mounting fixtures is not a simple matter, but, as in the case of helical springs, a team needs to know the actual rates, not the calculated rates.

The U Anti Roll Bar - Figure 11-3 illustrates the basic U anti roll bar. This type of bar consists of a torque tube/bar with arms attached perpendicularly to the tube/bar at both ends. The bar is affixed to the chassis at both ends of the torque tube/bar with some type of bearing or phenolic bushing.

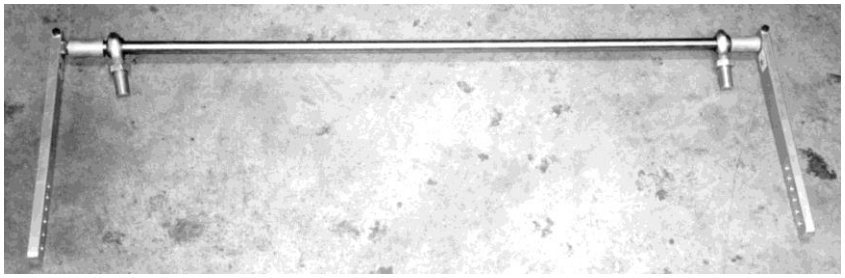


Figure 11-3

Figure 11-4 illustrates a splined front torque tube with an anti roll bar arm attached. The bar is attached to the chassis via an aluminium pillow block. Normally, a phenolic bushing is inserted into the pillow block so the torque tube rotates on the bushing rather than directly on the aluminum. This is done for three reasons: various bushing diameters can be machined to fit different bar sizes, aluminum tends to wear quite quickly in which case the entire pillow block will need to be replaced, and, most importantly, aluminum tends to gall and cause all sorts of unwanted friction forces.

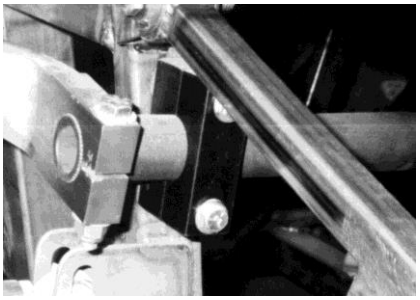


Figure 11-4

Figure 11-5 shows a very long anti roll bar arm with three bends in it. An arm of this design often introduces twisting loads into the arm, which, in turn, affects the bar and wheel rate. It is the last bend near the suspension attachment point that will cause many of the problems.

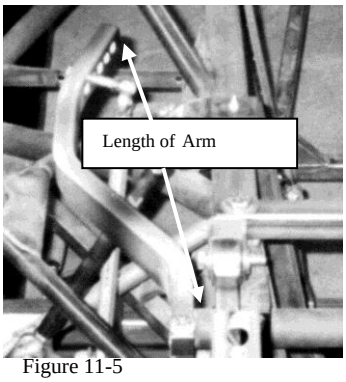
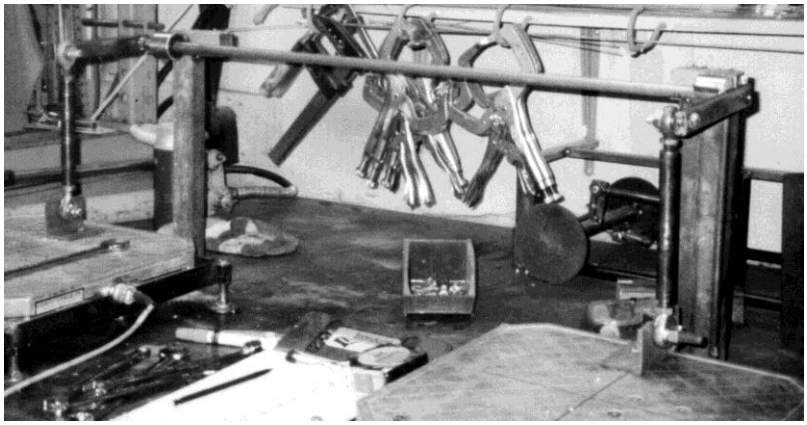


Figure 11-5

Figure 11-6 illustrates a very basic anti roll bar model used to help the engineer understand how an anti roll bar works. This type of jig is **not even close** to being stiff enough to be used to test a bar's angular and linear rate.


Figure 11-6

Figures 11-7 & 11-8 show a stock car type of anti roll bar. Spherical bearings are mounted at the right and left side frame mounts through which a splined torque tube runs. Teams used to use ball or needle bearings until they began to realize the effects of chassis flex on the bar rates.

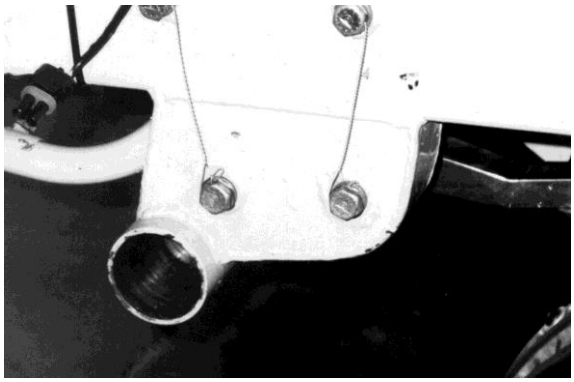

Figure 11-7


Figure 11-8

Different torque tube angular rates can be achieved by machining the OD to different diameters except for the portion of the torque tube that runs in the spherical bearing which is left at the bearing ID diameter - Figure 11-9.

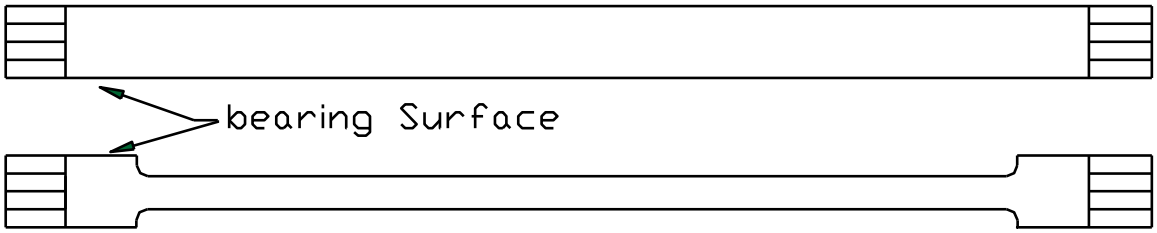


Figure 11-9

Figure 11-10 shows 4 different torque tubes. Notice that the top tube has a larger OD than the bearing OD while the 3 bottom tubes all have smaller OD's than the bearing surface.

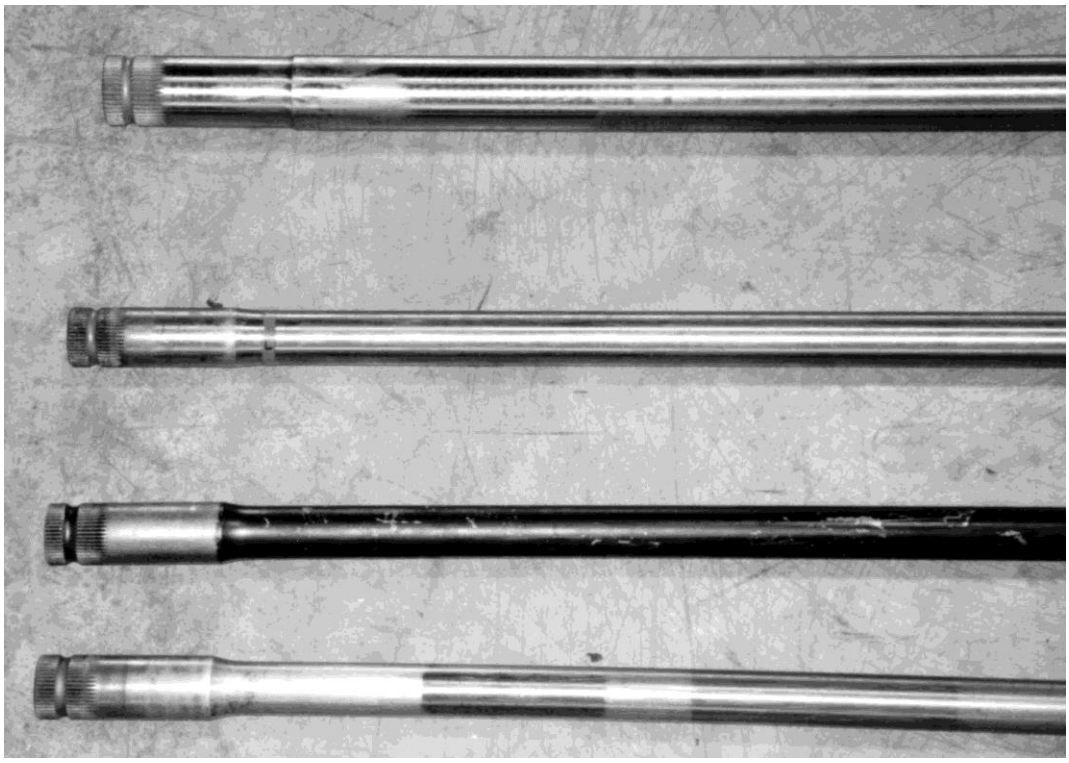


Figure 11-10

Figures 11-11 & 11-12 show a U bar with blade arms. Figure 11-11 shows both arms at the softest setting while Figure 11-12 shows one arm at the soft setting and the other arm at its hard settings. With a blade system, one or both blades are normally adjustable from the cockpit. In this example, the right blade with the small attached lever arm can be adjusted by the driver while the left blade needs to be adjusted by the crew. The bar rotates on bearings located at each end of the bar. The bearings are attached to the tube with pillow blocks.

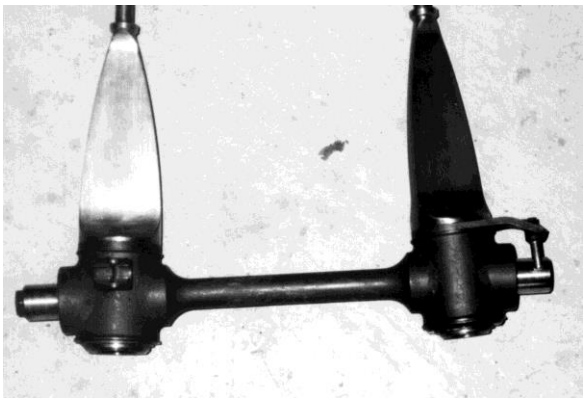


Figure 11-11

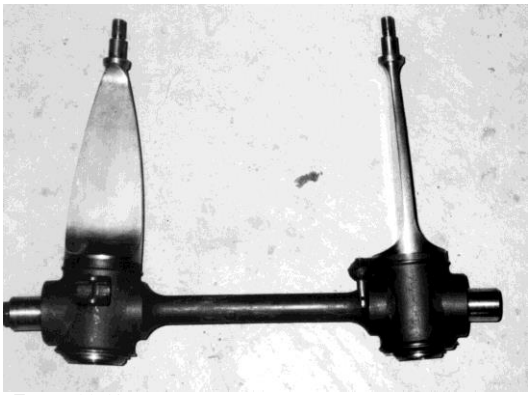


Figure 11-12

Note: Take time to calculate a few hundred different linear and angular rates using the Excel calculators

Figures 11-13 & 11-14 illustrate two different U bars used on formula cars. In these examples, both blades are turned in unison with each other as the driver makes a blade angle change from the cockpit.

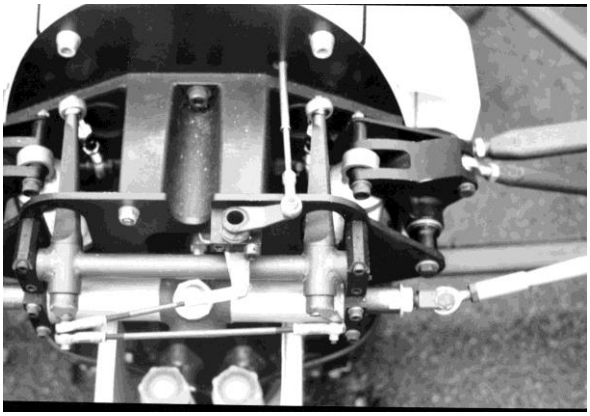


Figure 11-13

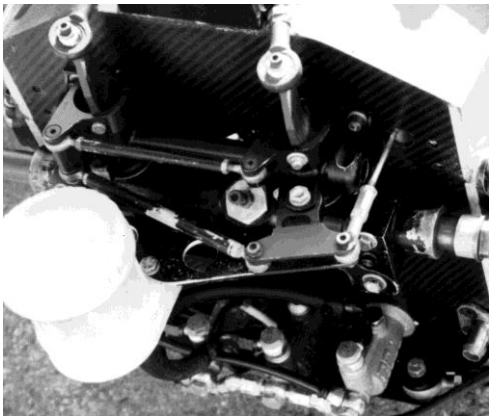


Figure 11-14

Most Formula cars allow for the driver to adjust both the front and rear blade angles from the cockpit. Figure 11-15 shows the cockpit blade adjuster mechanism. The inside lever adjusts the front blades while the outside lever is used to adjust the rear blades.

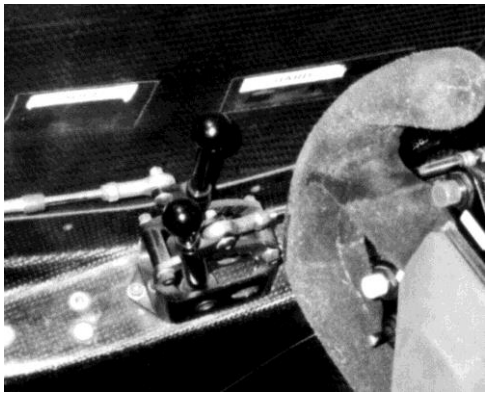


Figure 11-15

Table 11-1 and accompanying graph shows an example of five different bars used with a Formula car. The working blade length is 3.31" long with a working thickness of 0.20".

Table 11-1
Anti roll Bar Linear Rates Date
Bar # 045-3402 - 05, 045-3402 - 0625, 045-3402 - 0812, 045-3402 - 1, 045-3402 - 125, 045-3402 - 146
Blade Length = 3.31" Blade Thickness = 0.200"

Blade	Bar OD	Bar OD	Bar OD	Bar OD	Bar OD	Bar OD
Angle	0.500"	0.625"	0.812"	1.000"	1.250"	1.460"
0.0	508	668	835	890	931	960
22.5	528	728	913	1020	1040	1060
45.0	636	940	1270	1493	1536	1584
67.5	796	1340	2140	2860	2987	3220
90.0	913	1700	3180	4400	5720	6720

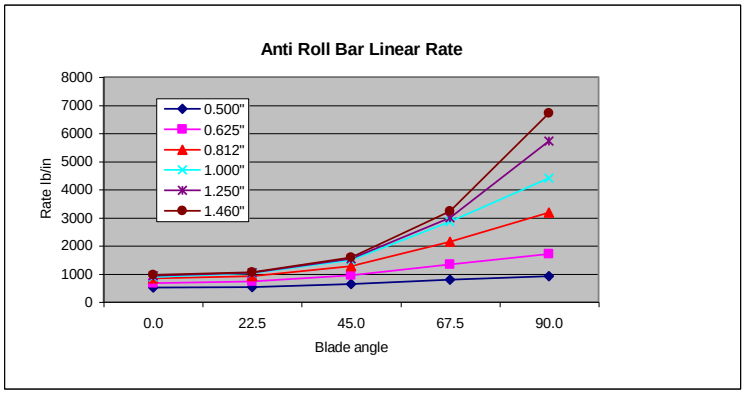
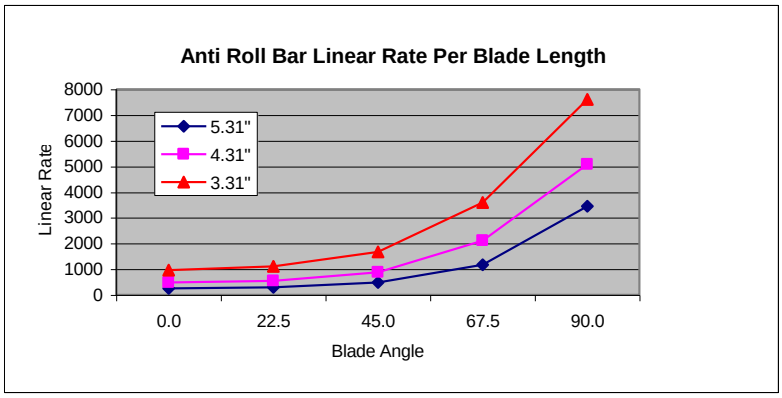


Figure 11-2 and accompanying graph illustrates rate changes with three different blade lengths using a 1.5" torque tube.

Table 11-2
Anti roll Bar Linear Rates
Bar OD = 1.500"

Blade	Blade	Blade	Blade
Angle	Length	Length	Length
	5.31"	4.31"	3.31"
0.0	260	470	970
22.5	300	550	1107
45.0	480	880	1680
67.5	1180	2120	3600
90.0	3440	5080	7600



GT Anti Roll Bars - Figure 11-16 shows a front anti roll bar used on a TransAm car. This car uses a single front adjustable blade. Notice the phenolic insert in the aluminum pillow block. Figure 11-17 illustrates a TransAm rear anti roll bar. It also uses a single adjustable blade. The rear bars used with a live axle rear housing traditionally have a much lower angular rate than the front bars; however, because of differences in the various motion ratios, that does not necessarily mean the rear linear rate is that much less than the front linear rate.

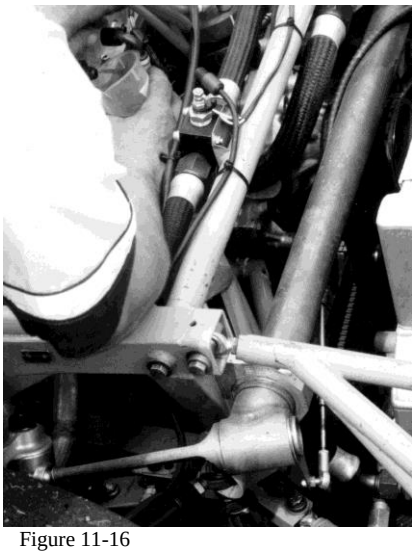


Figure 11-16

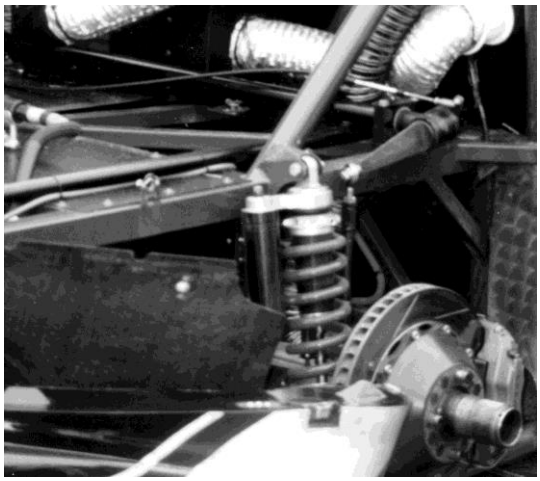


Figure 11-17

Figures 11-18/11-20 illustrate another front TransAm anti roll bar system. Rather than the torque tube being mounted via phenolic bushing and pillow blocks, metal inserts are welded into the tube itself which allows rod ends to be used to affix the tube to the chassis - Figure 11-20. This type of mounting system drastically reduces the amount of friction and stiction seen when pillow blocks are used.

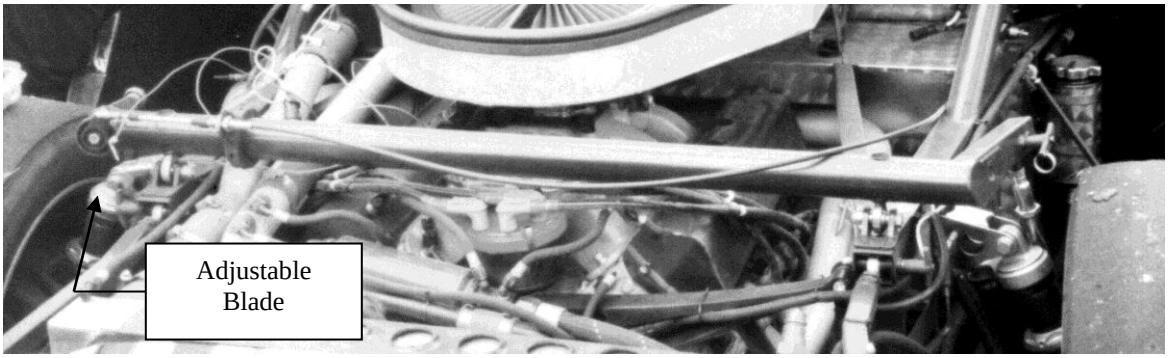


Figure 11-18

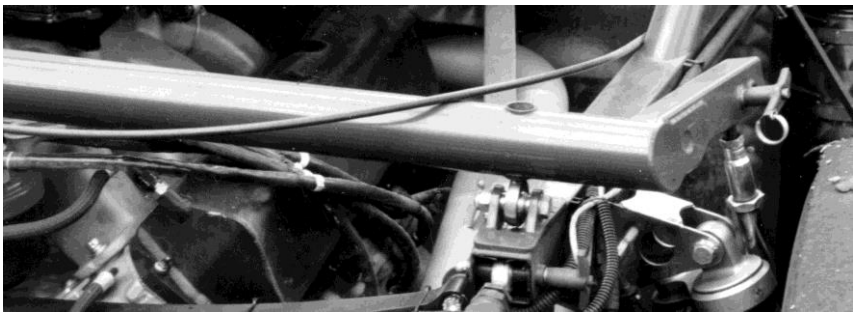


Figure 11-19

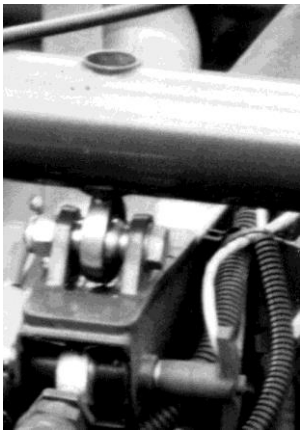


Figure 11-20

Table 11-3 presents a number of front tubular GT type anti roll bar scenarios using a 2" OD torque tube with rigid arms.

Table 11-3
Front Tube Bar

OD	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00
Wall	0.065	0.065	0.065	0.035	0.035	0.035	0.095	0.095	0.095	0.095
ID	1.87	1.87	1.87	1.93	1.93	1.93	1.81	1.81	1.81	1.81
Tube Length	40.00	46.00	46.00	46.00	40.00	40.00	40.00	40.00	46.00	46.00
Arm Length	10.50	10.50	12.00	12.00	12.00	10.50	10.50	9.00	10.50	12.00
Linear Rate	965.60	839.65	642.86	362.21	416.54	544.06	1348.45	1835.40	1172.57	897.75

Table 11-4 presents a number of front tubular GT type anti roll bar scenarios using a 1.75" OD torque tube with rigid arms.

Table 11-4
Front Tube Bar

OD	1.75	1.75	1.75	1.75	1.75	1.75	1.75	1.75	1.75	1.75
Wall	0.065	0.065	0.065	0.035	0.035	0.035	0.095	0.095	0.095	0.095
ID	1.62	1.62	1.62	1.68	1.68	1.68	1.56	1.56	1.56	1.56
Tube Length	40.00	46.00	46.00	46.00	40.00	40.00	40.00	40.00	46.00	46.00
Arm Length	10.50	10.50	12.00	12.00	12.00	10.50	10.50	9.00	10.50	12.00
Linear Rate	637.84	554.64	424.65	240.83	276.95	361.74	884.90	1204.45	769.48	589.13

Tables 11-5 & 11-6 illustrate the changes in wheel rate and resultant load at the tire for various spring only and spring plus bar iterations. Load at the tire is wheel rate plus preload. Table 11-5 is an asymmetric stock car while Table 11-6 is a symmetric Formula car.

Table 11-5

Spring Versus Spring and Bar Wheel Rates — **Stock Car**
Left and Right Side Springs = 1000 lb. — Bar Linear Rate = 392.667 lb.

	Ride	Roll	Left Rate	L Preload	Load @ Tire	Right Rate	R Preload	Load @ Tire
Spring Only	0	1	-299.238	1000	700.762	297.357	700	997.357
Spring & Bar	0	1	-651.650	1000	348.350	643.036	700	1343.036
Spring Only	0	2	-616.178	1000	383.822	571.591	700	1271.591
Spring & Bar	0	2	-1334.070	1000	-334.070	1247.253	700	1947.253
Spring Only	-1	0	483.828	1000	1483.828	502.347	700	1202.347
Spring & Bar	-1	0	477.304	1000	1477.304	509.020	700	1209.020
Spring Only	-1	1	220.697	1000	1220.697	758.416	700	1458.416
Spring & Bar	-1	1	-114.632	1000	885.368	1084.570	700	1784.570
Spring Only	-1	2	-64.119	1000	935.881	988.003	700	1688.003
Spring & Bar	-1	2	-742.254	1000	257.746	1615.049	700	2315.049

Table 11-6

Spring Versus Spring and Bar Wheel Rates — **Formula Car**
Left and Right Side Springs = 1000 lb. — Bar Linear Rate = 392.667 lb.

	Ride	Roll	Left Rate	L Preload	Load @ Tire	Right Rate	R Preload	Load @ Tire
Spring Only	0	0.5	-198.111	300	101.889	221.881	300	521.881
Spring & Bar	0	0.5	-282.977	300	17.023	317.067	300	617.067
Spring Only	0	1	-375.592	300	-75.592	471.492	300	771.492
Spring & Bar	0	1	-535.962	300	-235.962	673.665	300	973.665
Spring Only	-0.5	0	402.401	300	702.401	402.401	300	702.401
Spring & Bar	-0.5	0	402.401	300	702.401	402.401	300	702.401
Spring Only	-0.5	0.5	160.592	300	460.592	675.556	300	975.556
Spring & Bar	-0.5	0.5	56.800	300	356.800	792.139	300	1092.139
Spring Only	1	1	335.686	300	635.686	1642.505	300	1942.505
Spring & Bar	1	1	93.443	300	393.443	1958.033	300	2258.033

T Bars - Except for the design configuration, a T bar as typically used on formula cars has the same function as a U bar. They can have solid arms or use a blade set up. Figure 11-21 shows a number of bars made for a Formula 2000 test – Figure 11-21. Because of the higher spring rate accuracy inherent in T bars using arms as opposed to using a blade set up, Formula car tests are often first run without the blade bars in order to find the upper and lower bar spring rate limits needed. Once the upper and lower limits have been established, a blade type T bar can be designed for the application. Notice that the T bar on the far left is made to accommodate a blade set up. Figure 11-22 shows a typical Formula car T bar blade. Figure 11-23 shows a rear F 2000 T bar which uses blades that are considerably longer than in the previous example. The longer blades would reduce the rate seen by the wheel.

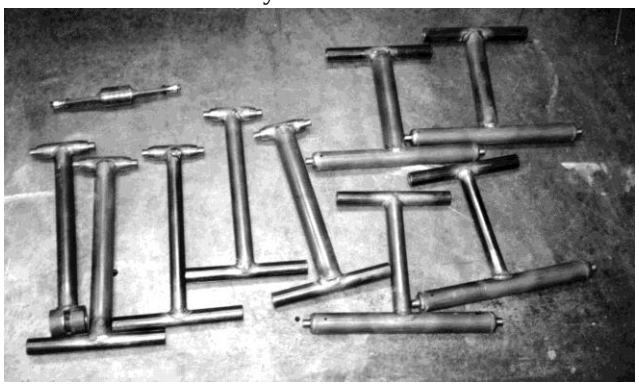


Figure 11-21



Figure 11-22 Can be made from steel or Titanium.

The T bar rates are calculated the same as for a U bar except the length of the torque tube is doubled to accommodate the fact that both the arms are affixed to one end of the torque tube (i.e. they are both active).

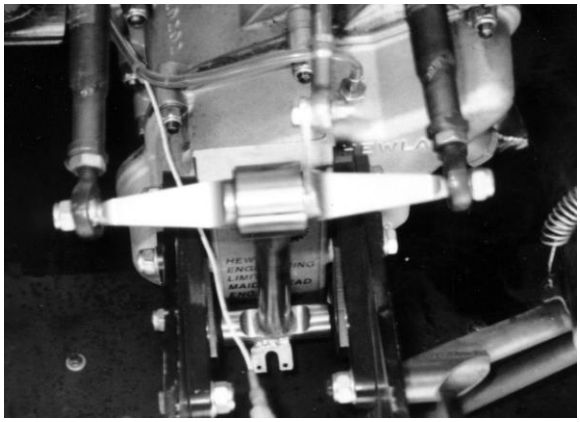


Figure 11-23

Figure 11-24 illustrates an Indy Car front T bar. Notice that the blade is at the bottom of the torque tube and is mounted to the tub via roller bearings and pillow blocks. The blade can be adjusted from the cockpit by a cable affixed to a small lever arm threaded into the center of the blade mechanism. The links which connect the T bar to the bell cranks can be mounted to the T bar at four different places, which, in turn, will change the linear bar rate. The links can also be mounted in an asymmetric manner. The post at the top of the bar is used to connect a third spring set up. The tub post used to mount the third spring can be seen at the center of Figure 11-25. Figure 11-26 shows a T bar and third spring set up used on an ALMS sports car.

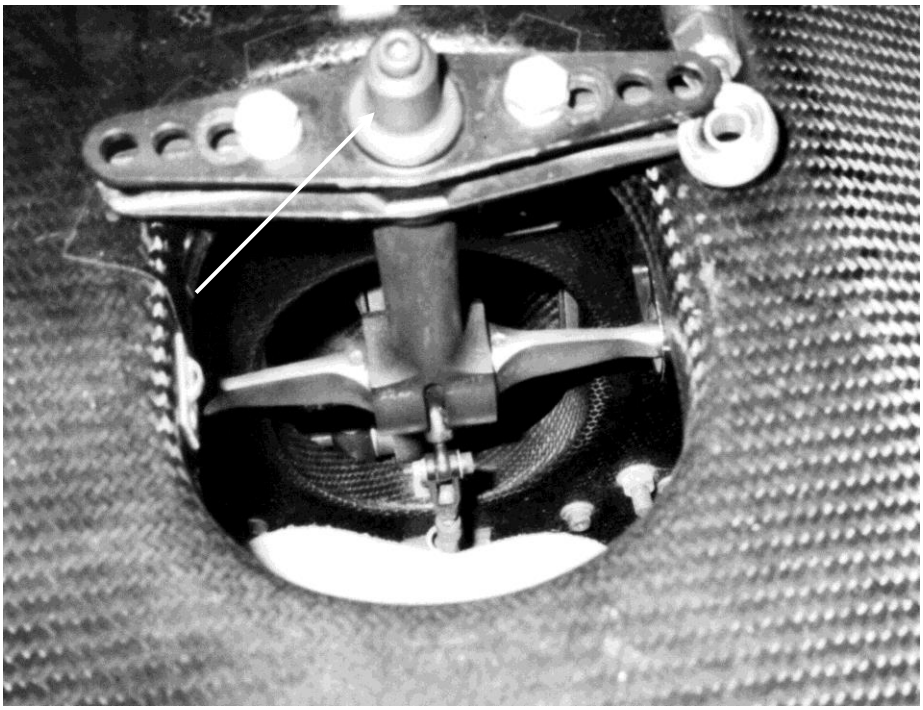


Figure 11-24

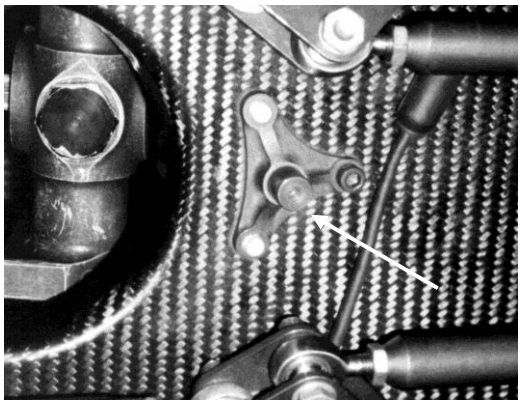


Figure 11-25

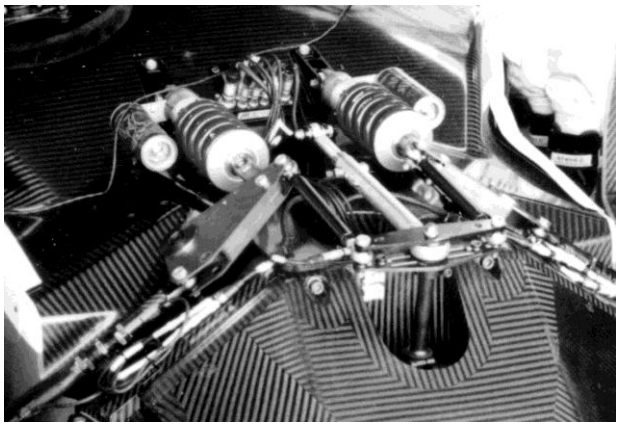


Figure 11-26

Third Spring - A third spring set up is used to increase the wheel rate in ride without increasing the wheel rate in roll. It is used primarily to offset the effects of aero loading while running down the high speed straight bits. The third spring is normally attached at the center top of the T bar while running back to a center mounted post on the tub. In pure roll, the T bar is twisted but does not move in the fore aft direction. However, during front ride (equal side to side bump), the bar will move aft, thereby compressing the front third spring, which, in effect, increases the ride wheel rate. The rear third spring works in a similar manner but moves in the opposite direction - Figure 11-27. The third spring can be designed as a conventional helical coil spring, some type of a damper (often interconnected with other dampers), a series of bump rubbers, a combination of a coil spring and damper, an air spring and/or a magnetic spring etc. Using a central third spring will allow the race engineer to set up the car with a more compliant suspension setting for the low speed corners while maintaining aero driven ride heights and pitch angles during the faster bits. This sounds great but as practice has shown, it doesn't always work. Obviously, if a third spring is used

without an accompanying damper, the two outside dampers will need to be changed to match the new ride rates. Third springs can be run at both the front and rear or just at one end depending on the track conditions.

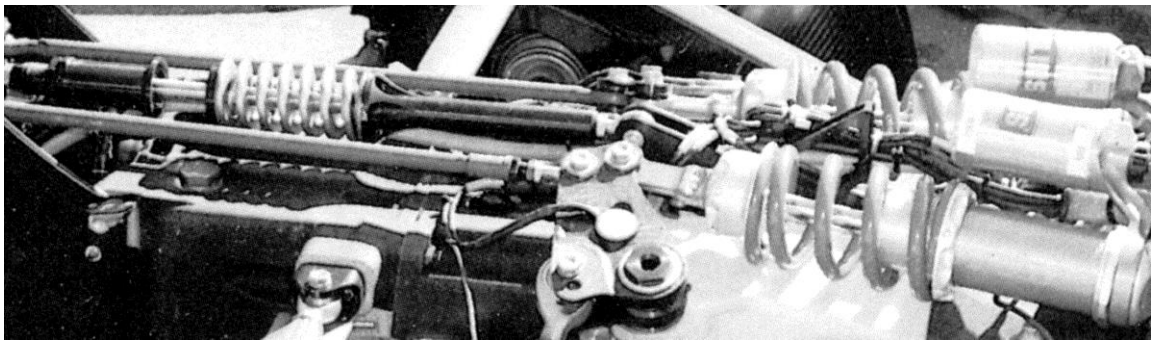


Figure 11-27

Monoshock - A typical monoshock system as shown in Figures 11-28 & 11-29 uses a single rocker/bell crank to which both the left and right push rods are attached i.e. the two wheels are linked to a single spring and damper. In ride (both wheels moving in bump the same amount) both push rod loads would be fed through the bell crank into the single damper and spring. Because the loads from both wheels are being transferred to a single spring and damper in ride, both the spring rate and damper settings normally need to be much higher than with a conventional bell crank push rod system.

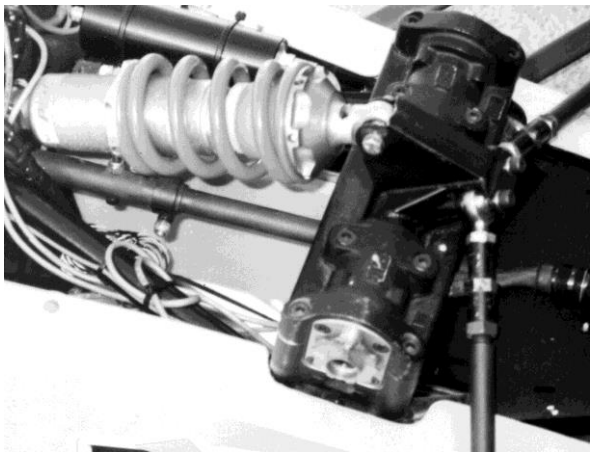


Figure 11-28

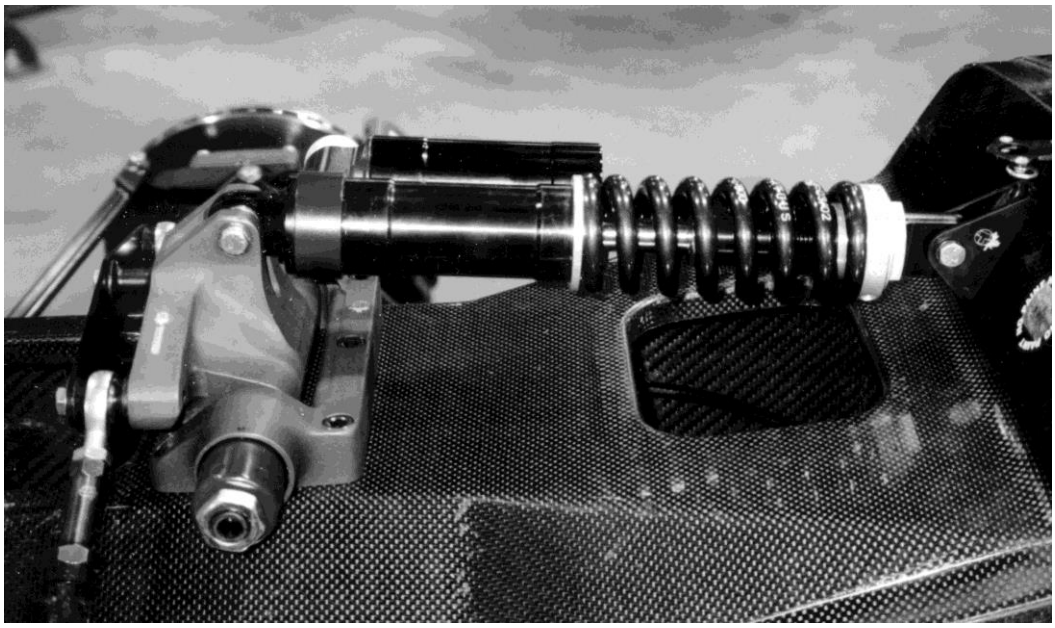


Figure 11-29

If the rocker/shuttle can not be moved back and forth horizontally, then in roll loads from the outside wheel will be transferred into the inside wheel and no roll will take place except as the result of suspension and tire compliance. In order for roll to take place, the monoshock system allows the bell crank/shuttle to move horizontally along a shaft - Figure 11-30. The amount of horizontal movement is controlled by a series of springs or Bellville washers or dished washers. If a higher roll rate is required, more washers can be added, or, in the case of the dished washers, the direction of the dish can also be reconfigured. Preload can be added by turning in a nut on the end of the washer cap. The amount of left and right roll rate and preload can be set to different rates side to side.

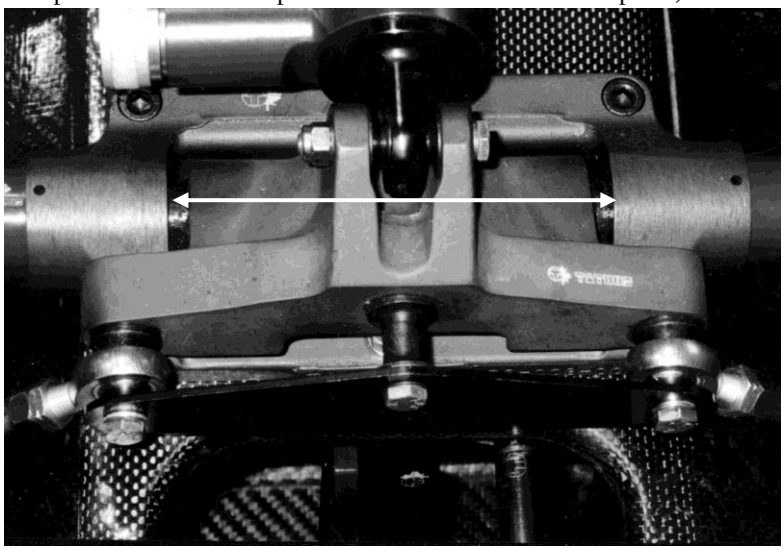
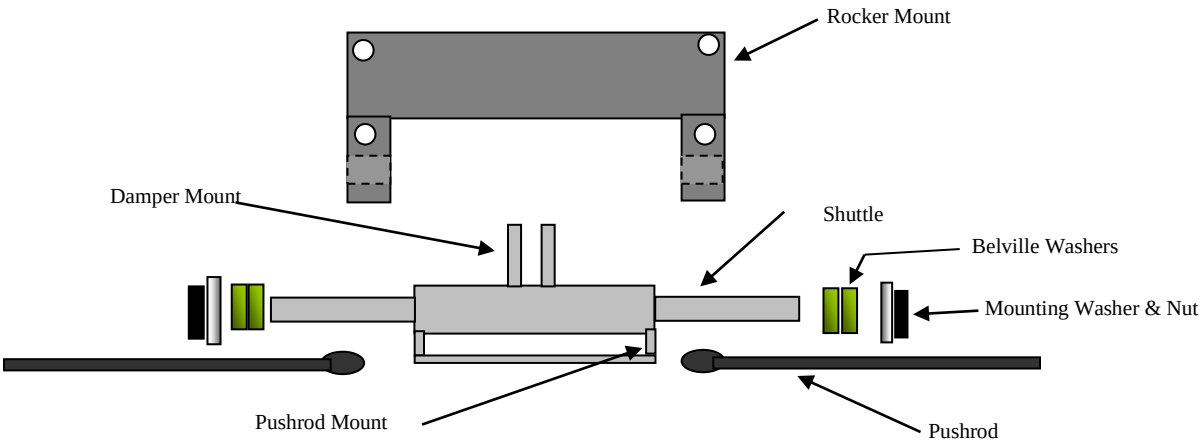
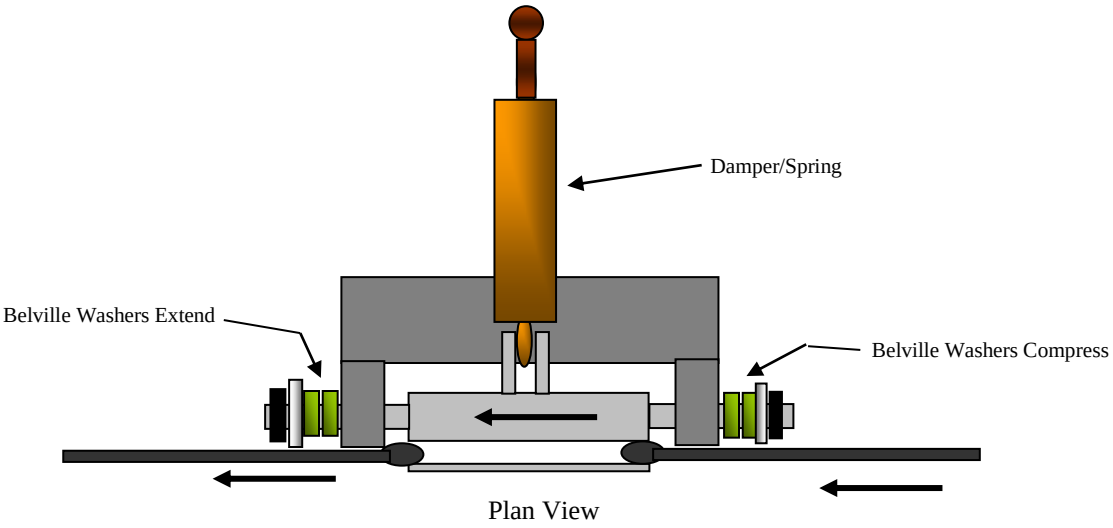


Figure 11-30

In theory, the monoshock system is quite simple, but, in practice, it is not. Working mostly in North America, I have had very little experience working with this type of system except for a number of research projects for open wheel teams running on both oval tracks & road courses. In fact, I have never been able to find **anyone** who thoroughly understands how this system works – although I have run into lots of people who think they do. In theory, one of the reasons for using a monoshock system is to decouple bump from roll, something some mono systems do fairly well while others do not. Again, in theory, it is fairly easy to mathematically model a mono system in pure bump or pure roll but very difficult to mathematically model both events together. Other reasons stated for running a front and/or rear mono system are less sprung weight (not the case with the complex Dallara rear mono system), various aero platform advantages (this can be an advantage under certain conditions) and because, in theory, it can work like a Z bar rather than a U bar roll system i.e. moves load from the outside wheel to the inside wheel – the optimum words here are "in theory". As an aside, a number of IRL teams have working with both front and/or rear mono system.

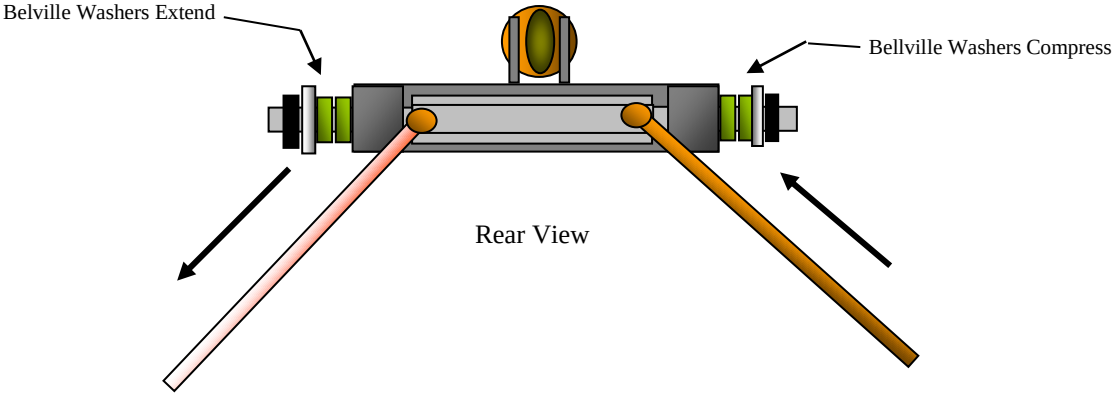
The following figures in Word Draw show the basic monoshock components.





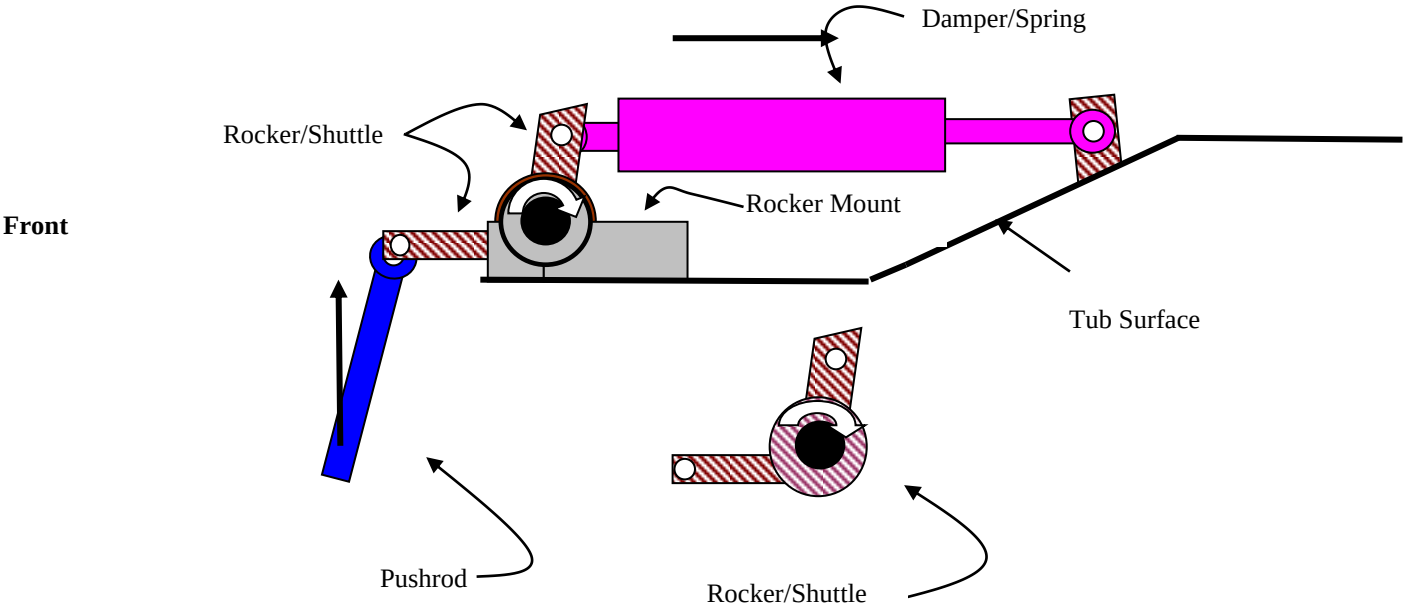
Plan View

Figure 11-31



Rear View

Figure 11-32



Side View

Figure 11-33

I would not suggest, as a first step, trying to calculate the various motion ratios through mathematical modeling but rather work it out through very careful in-shop testing and measurement. Even here you will find a multitude of measurement pitfalls. Under load, many of these systems are plagued with a considerable amount of stiction, deflection, compliance and varying spring/Bellville problems. The motion ratios are more often than not **quite non-linear**.

The bottom line is most of these systems do not work exactly as you would expect them to work – that is a rather large **understatement**. When you add in the fact that there are quite a number of non-linearities built into most of the systems, it's easy to end up rather confused. After running tests using three different systems plus two R & D built systems, I ended up with a large number of opposing answers to the same problem. Like taking psychology classes, I should have stopped at one. None of the systems I tested, when under load, worked as described by the manufacturer. Remember, lifting an outside wheel to see what happens to the loads on the inside wheel, under static conditions, is very different than under dynamic conditions!

I have a considerable amount of test material and mathematical modeling on the monoshock system, but, unfortunately, it has all been misplaced until 2010. Figures 11-34 through 11-36 show a simple test jig built to look at some of the monoshock's idiosyncrasies.

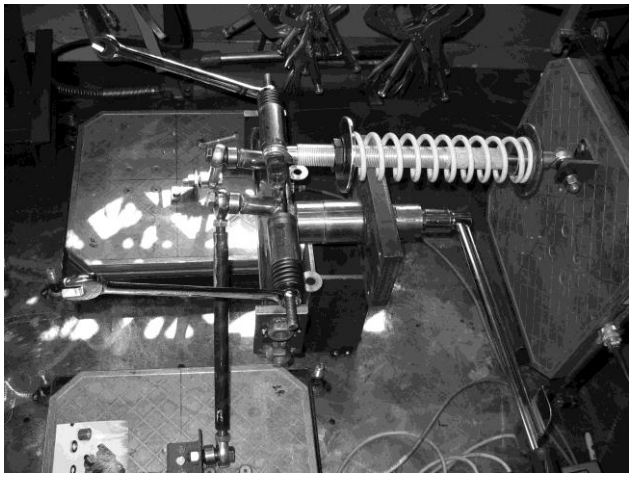


Figure 11-34

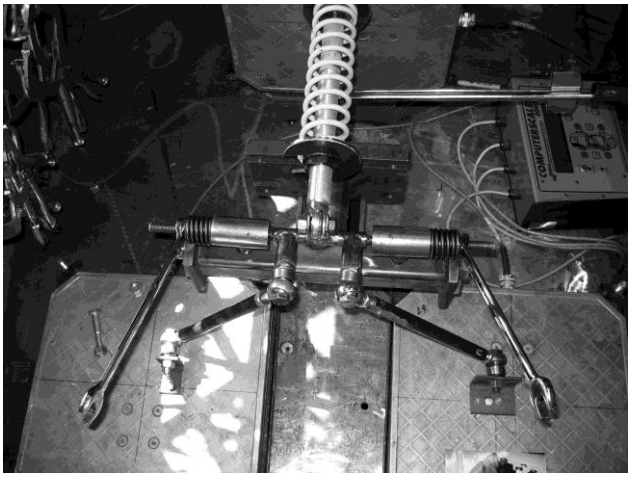


Figure 11-35

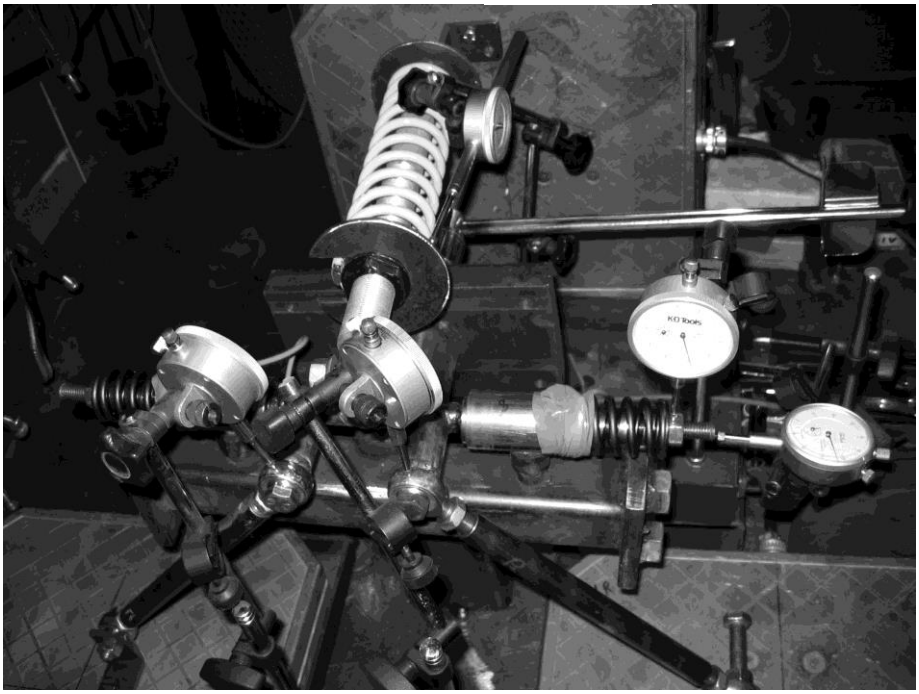


Figure 11-36

The monoshock system used with the following BMW system uses a T anti roll bar rather than a shuttle. It has an **extremely** high bar rate i.e. the roll rate is primarily controlled from the front tires and/or suspension compliance in conjunction with the rear suspension system – Figure 11-37/11-38.

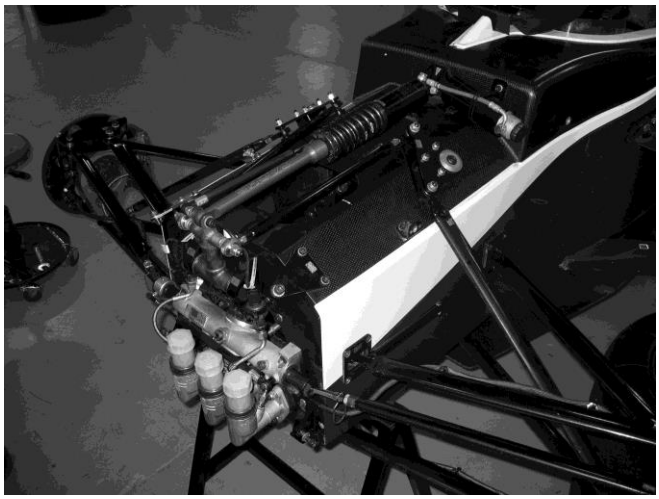


Figure 11-37

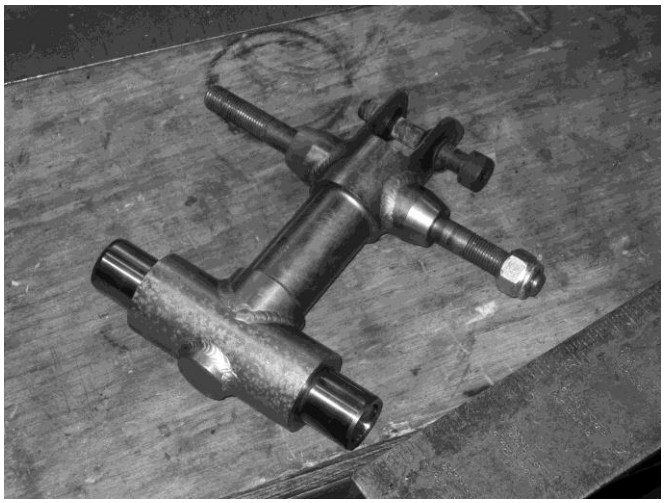


Figure 11-38

Howe Bar - The Howe bar is used extensively on short track oval cars. It may look a bit crude, but, in fact, works very well. It is a system that can be adjusted to couple in left turn roll (chassis rolls to the right) and decouple everywhere else - Figure 11-39. The bar itself is a one piece hollow tube with each end bent to form the arms. The bar is suspended from the chassis by a u-bolt on each side of the frame which can be used to adjust either end of the bar up or down - Figure 11-40.

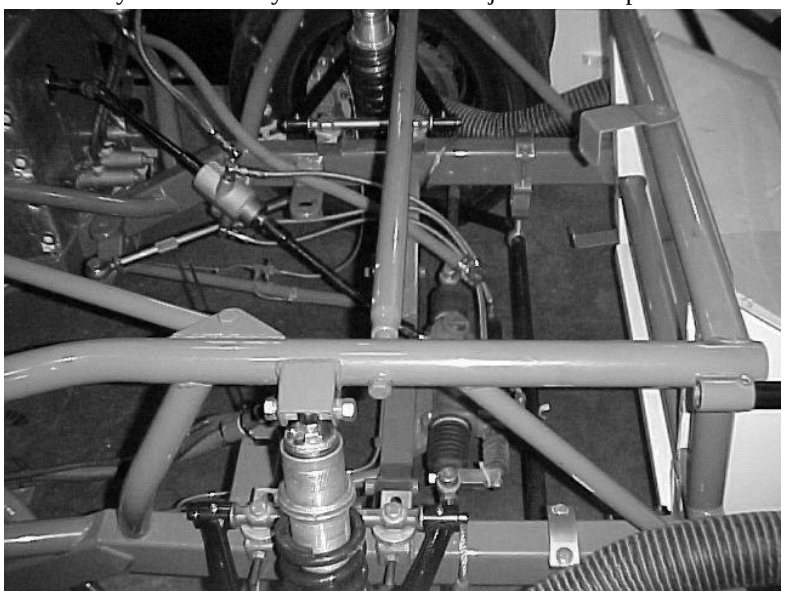


Figure 11-39



Figure 11-40

The **bottom** of the right side arm rests on a rigid pad suspended from the right side lower control arm shown in Figure 11-33 while the top of the left side arm just touches a pad on the bottom of the left side lower control arm (no preload) – Figure 11-34. The u-bolts can be adjusted up and/or down so the left side arm sees any amount of preload for which the engineer calls or it can have negative preload, in which case the left side will hang loose. If the bar is set up with negative preload, the chassis will need to roll to the right (left turn) a certain amount before the slack is taken up and the bar becomes active. The use of negative preload allows the chassis to have a different balance at corner entry than later in the corner.

Highly asymmetric cars under hard braking can upset the handling at corner entry if the anti roll bar is coupled. This is one reason short track teams sometimes run the Howe bar slightly loose at ride height.

The most common bar sizes range from 1" to 1 1/4". **Approximate** rates for 1" of linear motion are 1" = 150 lb., 1 1/16" = 175 lb., 1 1/8" = 200 lb., and 1 1/4" = 300 lb.

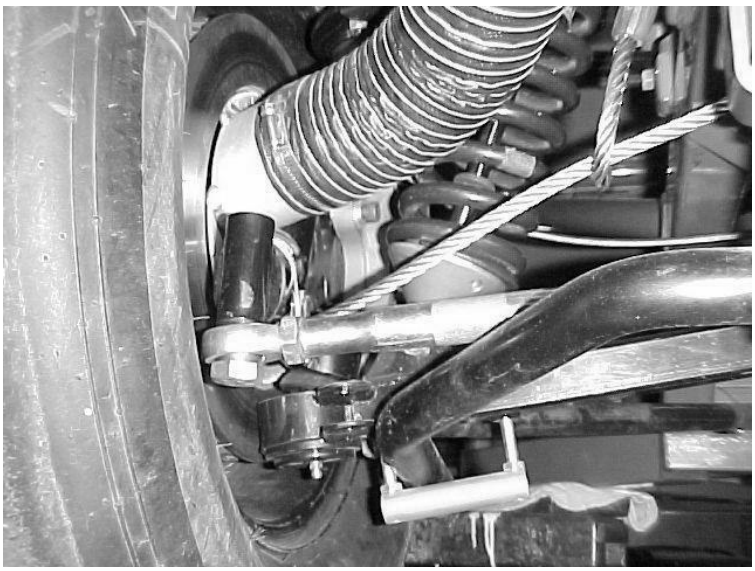


Figure 11-33

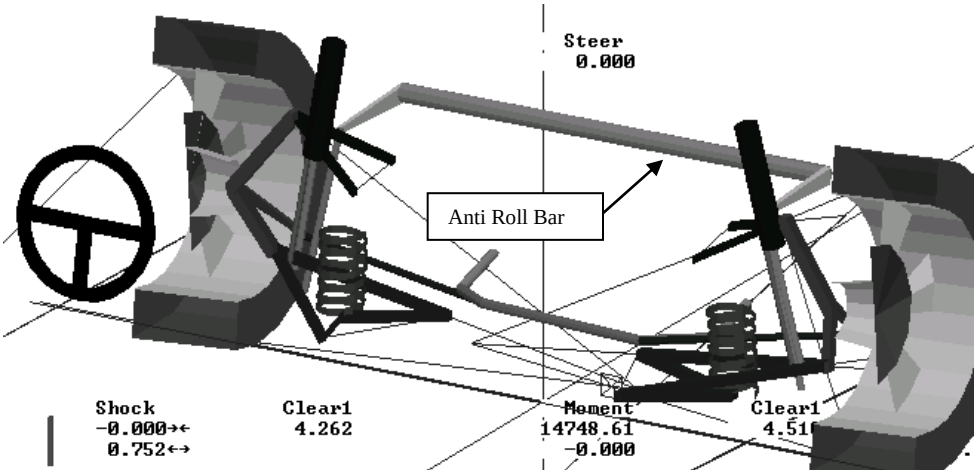


Figure 11-34

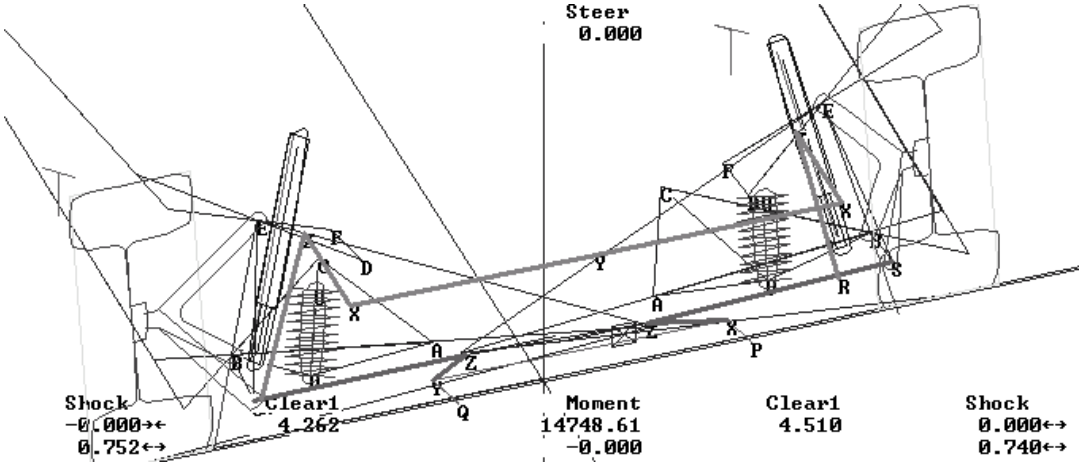
****Begin WinGeo3 Example(s)****

U Bar

Win 11-1 & Win 11-2 show an anti roll bar mounted above the frame. Though stock car front anti roll bars are normally mounted below the frame, the WinGeo3 demo has the front bar mounted higher for the sake of clarity.



Win 11-1



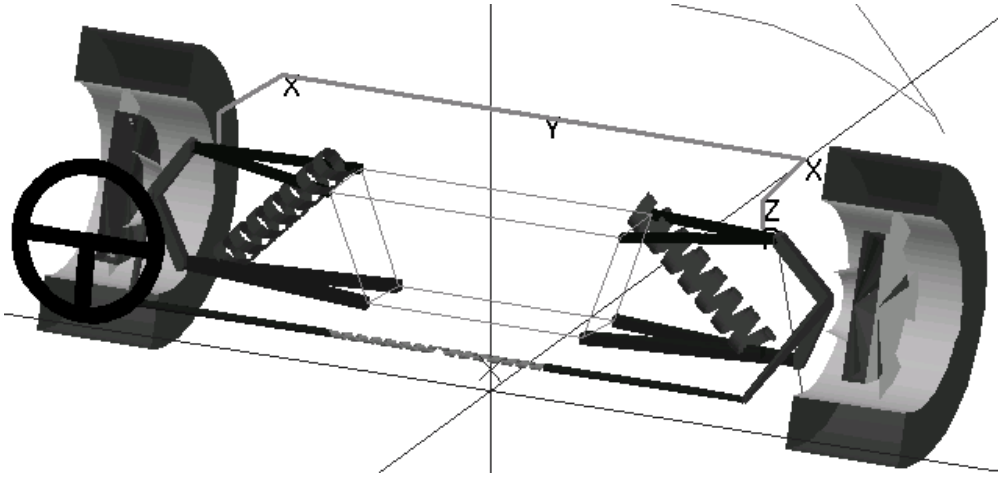
Win 11-2

Edit: Swaybar - Edit the anti roll bar or swaybar points. This can be either a U-bar type (usually found on sedans, stock cars and production cars) or the T-bar type (often found on formula cars with push rods or pull rods). T-bar types often have a third spring though a U-bar may also have a third spring.

Both sides		X fore-aft	Y width	Z vertical	Check sums
Anti-roll bar axis	X	-10.000	23.000	20.000	X to Z= 10.050
Anti-roll bar mid-point	Y	-10.000	0.000	20.000	Length= 46.000
Anti-roll bar blade	Z	0.000	24.000	20.000	Z to R= 2.500
Anti-roll bar attach	R	0.000	24.000	17.500	
Attach_Swaybar is:		2 is Upper A-arm			☑ Want_Swaybar True
Swaybar is a standard U-bar					☑ Want_Ubar True
Swaybar is a T-bar					☐ Want_Tbar False

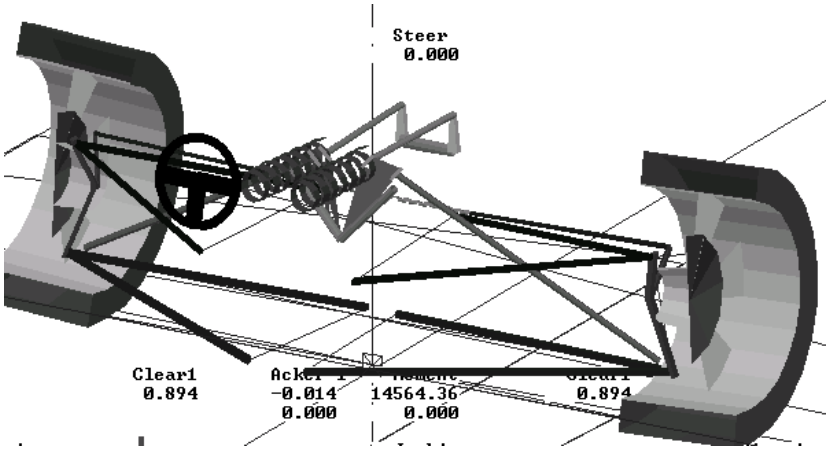
Win 11-3

The U-bar type swaybar bar shown in Win 11-3 & 11-4 is located on the chassis at points X and Y. There is an extension of the bar (or a blade) running from point X to point R. Then there is a link from point R down to the upper A-arm. Be careful that the *Swaybar* is attached to the proper point. The Want Swaybar box must be checked so that *Want Swaybar* is *true*.



Win 11-4

Win 11-5 shows a U bar used on an open wheel chassis.

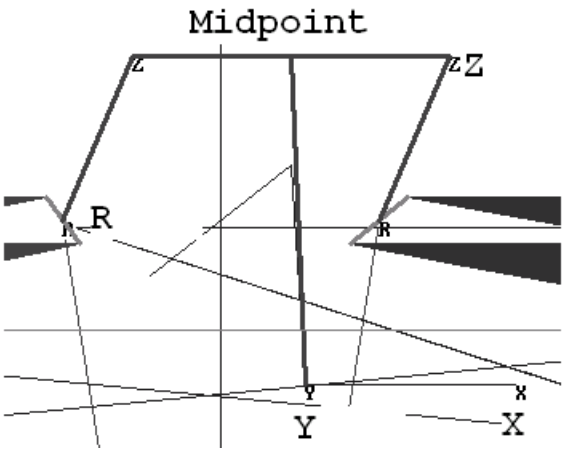


Win 11-5

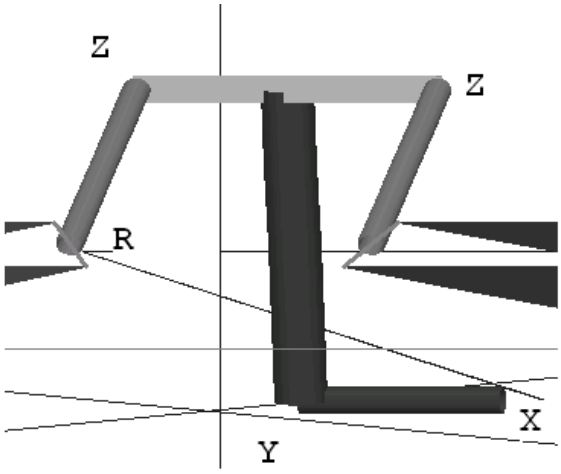
T Bar

T-bar Type Swaybar - These devices have the base of the T (Point Y) attached to the chassis. The top of the T (Midpoint) can rotate as well as move fore/aft. The tips of the T (Point Z) produce a rotation about the axis Y-Midpoint when the car rolls. Ride changes cause both tips of the T to move fore/aft without rotating. This can displace the third spring which is attached to the Y-Midpoint axis. The X-Y axis at the bottom of the T describes how the T-bar rotates fore/aft in relation to ride height changes.

Win 11-6 & 11-7 show a *T-bar* anti roll bar. The links Z-R activate the T-bar. These links are attached to the bell crank.



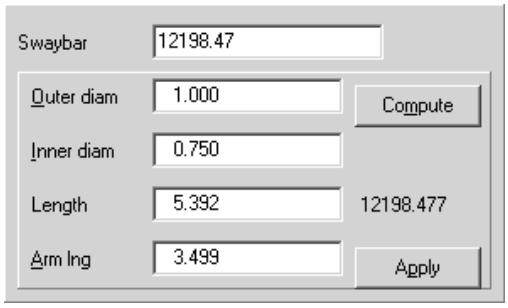
Win 11-6



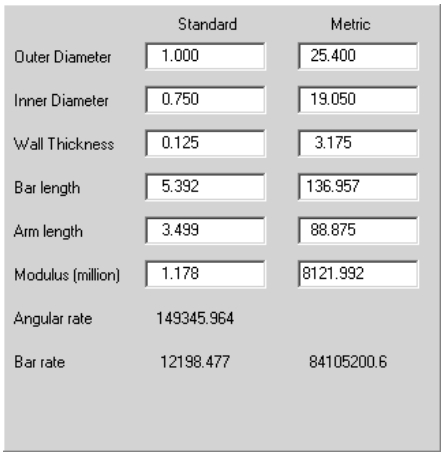
Win 11-7

Anti Roll Bar Rates

Edit: Forces (Springs and Bars) – If the user knows the bar rate, it can be entered directly into the *Swaybar* box – Win 11-8. To calculate a bar rate, click on the *Compute* button and enter the appropriate dimensions into the *Compute* menu – Win 11-9. Clicking on *Done* returns the user to the Win 11-8 *Forces* menu. Clicking on the *Apply* button enters the bar rate into the *Swaybar* box. Applying a bar rate in the *Swaybar* box includes the bar rate in the *load* calculations.

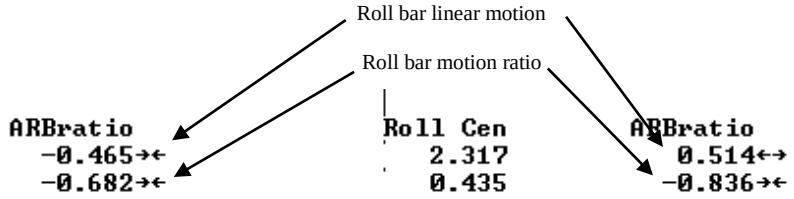


Win 11-8



Win 11-9

The amount of anti roll bar linear motion is shown on the main graphics screen as *ARBratio.*



Win 11-12

Win 11-13 shows roll bar twist in degrees and linear motion in inches for a series of open wheel anti roll bar iterations.

Roll Bar Linear Motion and Roll Bar Twist Open Wheel

	Ride	Roll	Rollbar Twist	ARBdis_L	ARBdis_R
B=2.00g T=0.00g 1	-0.136	0	0	-0.103	0.766
B=2.00g T=1.00g 2	-0.149	0.133	-1.984	-0.174	0.775
B=2.50g T=2.00g 3	-0.13	0.266	-3.956	-0.22	0.781
B=1.00g T=3.00g 4	-0.091	0.395	-5.845	-0.249	0.785
B=0.00g T=3.50g 5	-0.055	0.464	-6.818	-0.253	0.786
A=0.00g T=4.00g 6	-0.05	0.525	-7.712	-0.277	0.789
A=0.60g T=2.85g 7	-0.035	0.374	-5.477	-0.195	0.778
A=1.00g T=2.00g 8	-0.023	0.262	-3.836	-0.135	0.77
A=1.45g T=1.50g 9	-0.006	0.198	-2.882	-0.093	0.765
A=1.68g T=0.00g 10	0	0	0	0	0.753

Win 11-13

End WinGeo3 Example(s)



Trygve Rangen – You continuously impress me with your dogged determination to learn. It's great to see you working in F1.