10

Springs and Motion Ratios

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

The Spring

The Coil Spring - A helical compression spring is really a torsion spring. As the spring is compressed or extended, the wire twists. The wire's resistance to this twisting motion is what supplies the spring force known as spring rate. A compression spring works the same as a torsion spring or anti roll bar i.e. it twists. Every helical compression spring has a designed resistance to compression which is controlled primarily by the number of active coils, the mean diameter of the coil, the OD of the wire and the spring material.

The Spring's Jobs - The primary jobs of a spring are:

- To keep the contact patch loaded at the track surface over bumps etc. A tire needs to be kept in contact with the track surface in as constant a manner as possible. If the tire is bouncing or floating over the track, it will be of little help in maintaining steady state or acceleration forces. In this regard, the rebound (extension) or unloaded side of the spring is just as important as the bump or compression side

- To provide ride height control during ride and bump

- To resist overturning moments during lateral and longitudinal load transfer i.e. roll and pitch resistance

- To provide a method of changing the front versus rear balance of the car

- To absorb and store ride, bump, roll and pitch loads. Dampers react against and absorb road forces but they don't store them; the springs do that. A damper's primary job in regards to the spring is to control the energy input and release rate of the spring

- Provide adjustable ride characteristics

Types of Springs - There are basically two types of springs used in racing: springs closed at both ends (flattened to provide a full circle smooth contact surface) and springs closed at one end while open at the other end. This latter spring is only used at the front of a NASCAR type stock car.

Spring Quality – Two types of spring that give good results are Hyperco and Eibach. Hyperco springs are epoxied in blue while Eibach are epoxied in a brownish red color. I am less than happy when I have to work with springs having a quality inferior to these two.

If the spring has a lifetime guarantee but it doesn't state what it is that's guaranteed, don't use it. **If a spring is only guaranteed for free height, don't use it.** High quality springs seldom cost much more than a midrange spring but in time and performance they are worth their weight in gold. **Don't ever use a spring that does not guarantee spring rate i.e. a prescribed force under a prescribed amount of deflection**. High quality springs are one of those unfair advantages about which racers are always talking. The bottom line is that springs are incredibly important to the performance of the car so only use top quality springs or be content with not running at the front. I spend inordinate amounts of time testing and retesting all the springs I use providing I have access to a quality spring tester. If not, I don't test at all because all that does is lead to more confusion.

All springs you use should be guaranteed:

- Have both ends sit square on a flat surface. I use a surface plate and feeler gauges to check the squareness of the ends. I get very concerned with any measurement over 0.006".

- Sit perpendicular on the surface plate with parallel sides to within +/- 1% of total length. For a ten inch spring, that would be 0.9 degrees or 0.157" ... that is still too much.

- Have square axial compression. The spring must compress squarely. If it doesn't, it may be the result of a design and/or manufacturing problem or a suspension kinematic's problem which is most commonly found on stock cars - - more on this later.

- Have their free height within +/- 1 %. For a ten inch spring that would be 0.100". For virtually any type of race car, that is still way too much and for a formula type car, it's extreme. Figure 10-1 shows two springs whose free heights are 9.930" and 10.050. Both are within the +/- 1% limit, but, as we will discover shortly, that can result in some **very** nasty asymmetric loads.

- Have the indicated spring rate to within +/- 1% for springs using coil diameters of 0.625" or less and to within +/- 2% for springs using wire diameter over 0.625. Hyperco guarantees both of these rates. For a 250 lb./in. spring that could result in a variance of +/- 2.5 lb. per 1" of deflection while for a 3000 lb./in. spring, the variance could be upwards of +/- 60 lb. per 1" of deflection.

- Have an accurate and endlessly repeatable load height. This means that for a given deflection, the spring will always exhibit the same force.

- Have a linear rate of +/- 3% over 95% of its deflection. No spring will be linear over its full range, particularly during the first portion of deflection from free height and as the spring begins to coil bind. The ground coil(s) at the end of the spring subtract slightly from its linear quality.

- Be matched in pairs to within +/- 1% when used for symmetrical chassis.



Figure 10-1

All steels used in springs and anti roll bars have an elastic and plastic range in both their bending and twisting mode. If a force is placed on a test strip, as shown in Figure 10-2, the strip will bend under load. Up to a point, the strip will snap back to or close to its original position when the load is removed i.e. the strip was not bent past its elastic limit. We used the term, "or close to," because all steels, if not properly tempered and stress relieved, will take a small amount of set when first stressed i.e. spring sag and bar set. This is why it is so important that spring and bar manufacturers properly temper and stress relieve their products after the heat treating process. If the test strip is forced past its elastic limit, it becomes plastic and will not snap back to its original position i.e. the elastic limit has been exceeded.

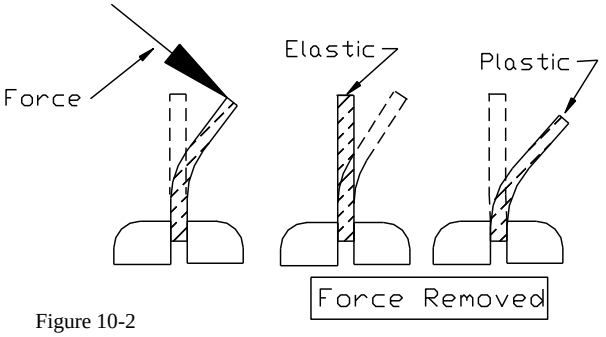


Figure 10-2

If a spring or bar is forced past its elastic limit into its plastic range, it will not return to its original setting. Figure 10-3 illustrates the various parts of a stress versus strain graph. Alloy steel heat treated to a higher Rockwell hardness will have a higher yield strength, and hence, can be bent more than a softer or annealed specimen before taking a permanent set. *It's just not this simple*.

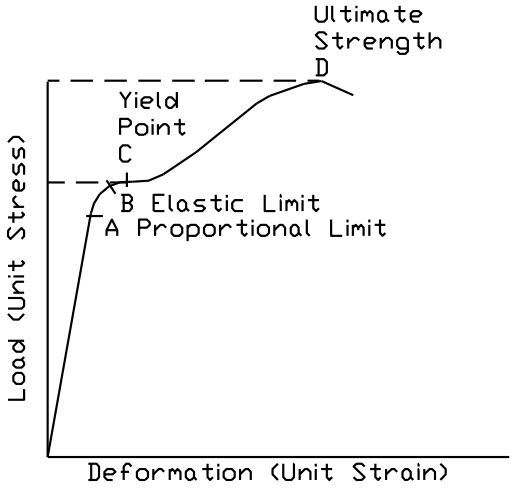


Figure 10-3

Basic Spring Rate Formula - The following simplified formula is used to help the reader understand how a spring is designed. It is not intended to be used to work out the rate of an existing spring. *It's just not that simple*.

Rate (F/S) = $Gd^4/8ND^3$
Where:
F = spring rate
S =spring deflection
G = shear modulus. Depending on the various alloy contents and percentages, the torsional modulus of a spring will be somewhere between $11.4*10^6$ and $11.7 * 10^6$. *Actually, it is not nearly that cut and dried*.
d = wire diameter
D = mean average coil diameter. Distance from center of the coil to center of coil.
N = number of active coils

- Because the wire diameter is raised to the fourth power, a small change in wire diameter will dramatically change the spring rate. For example, in a spring design having a rate of 423 lb. with a wire diameter of 0.5", if the diameter is changed to 0.6, the spring rate changes to 878 lb.

- As the number of coils and/or the mean average diameter increase, the spring rate will go down. If the 878 lb. spring from the previous example had the number of its coils increased from 8 to 10, the spring rate would change to 702 lb. If the mean coil diameter was also increased from 3.0" to 5.0", the rate would change to 152 lb.

Testing the Spring Rate - The good news is it can be done. The bad news is I know of only a few relatively inexpensive testers that are able to test spring rate with any consistency. I have worked with most, if not all of the commercially available, well-advertised and inexpensive portable spring testers and have found them to be wanting. The type of tester as shown in Figure 10-4 is time consuming but the higher level of repeatability is well worth the time.

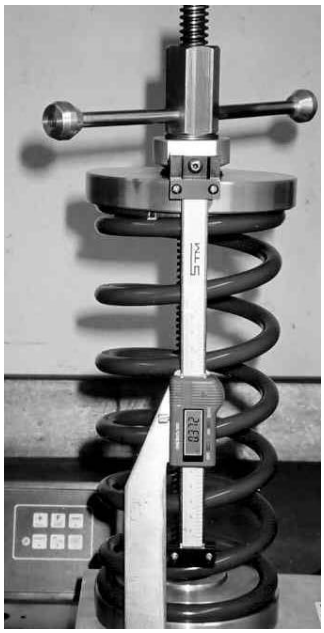


Figure 10-4

Using an inconsistent tester is much worse than using no tester at all.

- The problems with the majority of small portable testers are:
    - The quality of the load cell
    - The design of the load cell mounting point(s)
    - The tester's inability to compress the spring squarely/linearly
    - The overall lack of rigidity
    - The inaccuracy of motion measurement

If you do not have access to a high quality spring tester, it is better to rely on test sheets specific to your spring provided by your spring manufacturer, for a small cost of course. If your spring manufacturer can't supply this information, you have to ask yourself why. The cost of a high quality NASCAR spring is a fair bit more than its counterpart but it is still very much worth the money. Don't leave home without it. The current crop of aero driven specialty springs being manufactured for Winston Cup cars can cost over $1000.00 per spring. These ultra soft springs are definitely not designed to aid mechanical grip, but, instead, to help improve the aero drag coefficients.

Roehrig Engineering Inc. offers the best of the upper end portable spring testers. This tester uses the company's shock dynamometer platform combined with a hand actuated hydraulic ram. This unit is used in conjunction with an excellent software package - SPRING5.1®. NASCAR uses this unit for their pass/fail spring tests. Though not to the standard of a four post tester, Roehrig is constantly developing the rig in conjunction with such companies as Hyperco.

Many teams are now designing and building their own spring testers using die technology as shown in Figure 10-5. The surfaced plates can often be purchased used from companies which use large dies. If you don't have considerable knowledge and experience designing load cells, leave that bit to the people who know how to do that sort of thing. Picking out a load cell from your load cell dealer, even a top quality load cell, and then simply bolting it on to your spring tester seldom has a happy ending. Many of these test apparatus use multiple load cells, both for a higher level of accuracy and to allow for the analysis of axial loads. This type of tester normally has plates at least 1" thick and between 15" and 24" square combined with four good-sized ground shafts and linear bearings. The plates need to be surface finished. Motion of the driven plate needs to be controlled using two to four linear potentiometers. A good software package needs to be developed to make the whole thing useful. Most testers of this type have the hydraulic ram controlled by software.

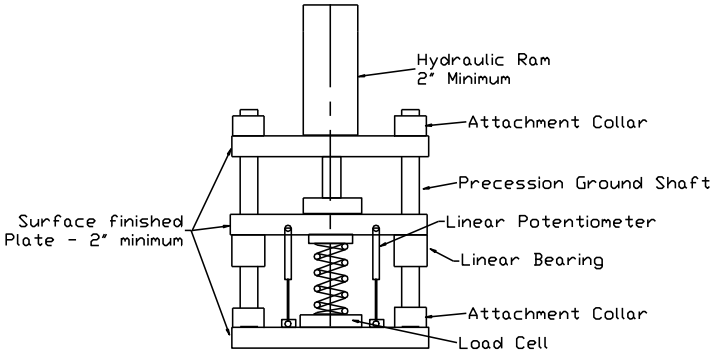
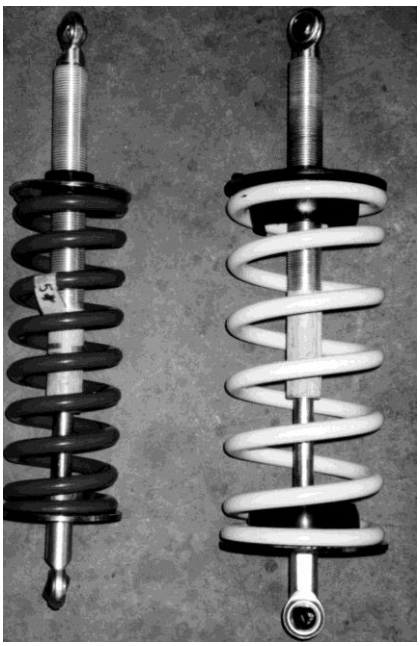


Figure 10-5

If you have to use a portable tester for rating your **lower** rate springs, try using a set up such as shown in Figures 10-6 & 10-7. This helps compress the springs in a linear manner and applies the load directly into the center of the load cell. Using a ball-bushing-bearing at the end of the dummy damper shaft rather than a Teflon™ bushing also helps.


Figure 10-6


Figure 10-7

If you mount your coil-over springs in the car using thrust washers between the spring and the damper adjuster nut/spring perch, as you should, make sure you also test the spring with "Torrington" thrust washers. Thrust washers are used for two reasons:

- To prevent stiction
- To allow the adjuster nut/spring perch to be turned more easily

Most coil over applications now use an adjuster nut that allows the adjuster nut load bearing surface to tilt back and forth to keep the spring surface flat on the nut at all times. This considerably reduces the bending loads on the damper and aids in the linear loading of the spring.

All springs need to be tested for their ability to compress squarely, **particularly** the open ended stock car type of spring. Most shops use a spherical bearing in the upper and lower spring mounting plate - Figure 10-8. In order to make sure the mounting plates stay perfectly aligned with each other, they need to be threaded into place. The plates are designed so the spring sits down over a small machined ridge. Figure 10-9 shows a spring used by a big budget stock car team that clearly does not compress perpendicularly to the surface of the tester - look at the angle between the two plates. Figure 10-11 shows that this same spring does not sit perpendicular to a flat surface. Figure 10-10 shows a spring that has compressed in a perpendicular manner. This type of apparatus only takes a few minutes to fabricate but can save you lots of grief at the track.



Figure 10-8

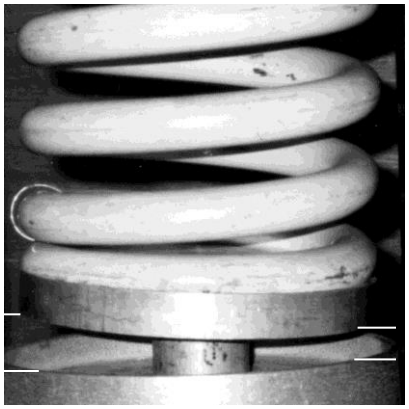


Figure 10-9

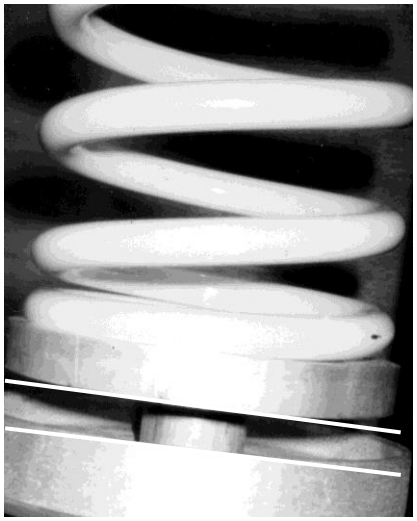


Figure 10-10

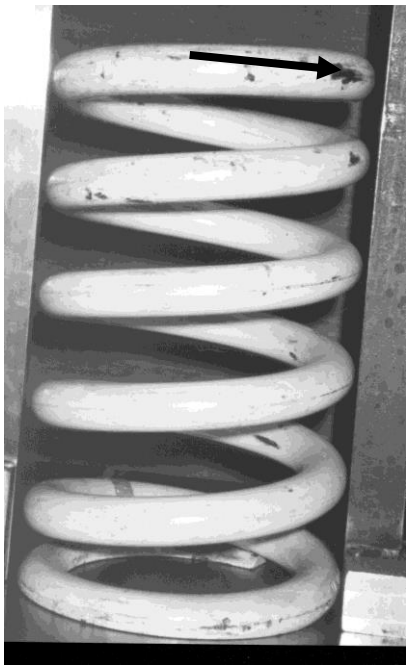


Figure 10-11

Sag, Free Height and Loaded Height - Sag is the result of a change in spring free height. If you have ever raced in a showroom stock series where you are required to use stock springs, you will be intimately acquainted with spring sag (the change in free height after the spring has been run). Before running springs of this type, most showroom stock teams will have the springs stress relieved. For those last minute emergencies, teams will coil bind their springs in a press and then stress relieve them using an air chisel with an end ground to fit over the coil. The chisel is slowly run down the coils from top to bottom using a fairly low psi setting. Each spring takes about 30 minutes.

Coil Bind - Considering the trend towards running much softer spring rates in many of the professional racing series from NASCAR to Champ cars, spring compression to coil bind has become much more of a problem in recent years. Even with the development of new spring material and manufacturing procedures plus the race for higher helical coil angles and fewer coils which allow the spring to compress further before becoming coil bound, overload in coil bind can drastically shorten a spring's usable life. Loading a spring past coil bind by 25 lb. is one thing but loading a spring, 500 lb. and more past coil bind is quite another i.e. impact loading a coil bound spring can and does affect the springs molecular structure, which, in turn, can change its free and loaded height. With the new mega soft aero driven stock car front and rear springs, even the best springs can only stand so much overload before the free height and loaded height will change. A spring will not lose its rate but it can and does lose free height, and, subsequently, loaded height. This can lead to handling changes during a race and tech problems after the race.

Coil bind can also be a problem with the quest to lighten everything, including the springs used on a formula car. As the helical angle increases and the spring length decreases, coil bind overload becomes more of a problem even when packers and bump stops are used.

Rather than using a tape measure and the top of the work bench, always use a good quality calliper and surface plate to test free height - Figure 10-12. Once the free height has been established, assign a number to the spring, scribe that number into the bottom of the spring and then recheck the free height after each use. Never scribe any place other than the bottom of the spring. When measuring springs with open ends for free height, measure in from the end of the wire 0.0625". This is done because the end of this type of spring often has the epoxy chipped off after a few uses.



Figure 10-12

Table 10-1 shows a small portion of an Excel program used to compare spring deflection to spring load (assuming the spring is perfectly linear in loading). The user can quickly change the deflection increment, spring rate and/or spring free height. The race engineer will normally run multiple copies with different rates etc. in order to determine which spring to use for a particular set of motion ratios and acceleration loads. This type of table is used for rough calculations only. Under actual conditions, no spring is linear which means if a team has an accurate rate tester, they will need to determine the spring's load at various heights and then develop a best fit formula to fill in the blanks. Developing this type of table for each spring is time consuming but should be skipped only if you are not interested in the highest level of performance.

Table 10-1

Spring Rate Versus Deflection Height				
Column #1		Spring #	2336	
Deflection	Spring	Spring	Spring	Spring
Increment	Rate	Height	Deflection	Load
	1000	10.000		
0.001	1000	9.999	0.001	1.00
0.001	1000	9.998	0.002	2.00
0.001	1000	9.997	0.003	3.00
0.001	1000	9.996	0.004	4.00
0.001	1000	9.995	0.005	5.00
0.001	1000	9.994	0.006	6.00
0.001	1000	9.993	0.007	7.00
0.001	1000	9.992	0.008	8.00
0.001	1000	9.991	0.009	9.00
0.001	1000	9.990	0.010	10.00

		Column #2		
Deflection	Spring	Spring	Spring	Spring
Increment	Rate	Height	Deflection	Load
0.001	1000	9.953	0.047	47.00
0.001	1000	9.952	0.048	48.00
0.001	1000	9.951	0.049	49.00
0.001	1000	9.950	0.050	50.00
0.001	1000	9.949	0.051	51.00
0.001	1000	9.948	0.052	52.00
0.001	1000	9.947	0.053	53.00
0.001	1000	9.946	0.054	54.00
0.001	1000	9.945	0.055	55.00
0.001	1000	9.944	0.056	56.00

Compliance and Torsional Rigidity - Using only high quality springs that have been accurately tested for rate consistency helps us to eliminate the spring as a problem when looking at the effects of suspension and chassis compliance, bind and torsional rigidity. So why is it that after we have gone through all the trouble to make sure we have springs which have consistent load versus deflection rates when testing the chassis, we still don't get consistent load cell readings for a given wheel deflection? Unfortunately, it is not as simple as saying it is the result of chassis and suspension compliance, angular deflection, bind and torsional rigidity but that is a **big part** of it. Testing for compliance, bind and torsional rigidity is not the domain of Book One but I thought we should at least comment about it so the race engineer can be thinking about potential problems when designing and working with the car. Book Two looks at a plethora of compliance and torsional rigidity tests run on chassis ranging from stock cars, GT cars, open wheel tube frame cars through to open wheel and sports cars using composite tubs. Teams are now finding that how and in how many ways they run their tests yield very different bits of information i.e. not just applying the test loads at the suspension mounts in pure compression or extension. They are also finding that finite analysis is not yet an exact science even for composite tubs let alone tube chassis with all their inherent weld inconsistencies. **Yes**, poorly designed and/or fabricated composite tubs do flex, many of them a great deal, particularly well-used ones.

We have all heard the old adage that if making large changes in spring and bar rates doesn't seem to affect the handling, most likely the chassis is flexing. In recent years, teams have now learned that even the smallest amount of flex, bind or compliance anywhere affects handling and that every component of the car needs to be looked at on its own and in combination with the whole system right down to the rims. Don't ever take your rims for granted as many, even well known brands, have a considerable amount of flex and resultant harmonic motion as well as undesirable dynamic camber changes.

Four and seven post shakers are very useful tools for what they do, and they can do quite a bit, but because they don't also apply longitudinal and lateral forces in combination with bump and aero forces, they often don't reveal compliance, bind and torsional rigidity problems, which, in turn, affect the effective spring and wheel rates. At the time of writing Book One, I have only been involved in low frequency longitudinal and lateral load inputs in combination with both low and higher frequency bump input testing, but even these results have been rather revealing. Designing suspension systems and chassis structure is still a developing science.

How teams install their strain gauges and load cells can ultimately change the spring and wheel rates i.e. putting a large hydraulic load cell between the load bolt and the upper spring perch. The engineer needs to spend time making sure that what they think they are load testing is actually what they are testing and that their test doesn't change the performance characteristics of the chassis. Pi in conjunction with Applied Vehicle Science makes a good load-sensing spring mount for use with stock cars.

Spring rate versus wheel rate considerations are an extremely important performance factor in design and set up. The smallest inconsistencies in this area can change the performance envelope. Remember that springs produce force in **both** compression and extension.

Searching After the Elusive Rate Changes

Rate Overlap - When one spring rate, because of manufacturing inconsistencies, overlaps a higher or lower rated spring, they are said to have overlapping rates. If a 900 lb. spring is out +/- 3%, its rate could be as low as 900 * 97% = 873 lb. and as high as 900 * 103% = 927 lb. If a 875 spring was out +/- 3%, it could have a rate as low as 875 * 97% = 848.75 lb. and as high a rate as 875 * 103% = 901.25 lb. If we replace the 900 lb. spring which actually had a rate of 873 lb. with a 875 lb. spring which actually has a rate of 901 lb., we will have increased the spring rate by 28 lb., not decreased it by 25 lb. As the manufacturer's specified spring rate goes up, the chances of having rate overlap increases. I have seen springs used by Cup teams out as much as 7% (2000 * 93% = 1860 lb.).

Free Height Inconsistences - If we have two 900 lb. springs with a free height difference of 0.103" (A = 10.027" - B = 9.924") and we preload both springs to a perch to perch height of 8.5" (not allowing for linear load % changes), Spring A will see a load of 1374.3 lb. while Spring B will see a load of 1281.6 lb. or a difference of 92.7 lb. The nonlinear loads seen for the first portion of the spring motion often tends to increase the difference between the two springs. This difference in free height length is within the range of the best quality springs - not a good thing.

Compliance and Torsional Rigidity - If, because of suspension and/or chassis compliance, a spring sees a variance in motion of +/- 0.244", a 900 pound spring would see a difference of 220 lb. over 1" of deflection. (1-0.244)*900 = 680 / (1+0.244)*900 = 1119.6.

Combined Rate Changes - It doesn't take a rocket scientist to figure out that the combined effects of rate overlap, free height differences and compliance differences can change the load seen at the spring and wheel quite dramatically.

Kinematic Deflection Problems - Suspension and chassis component deflection and compliance can have a considerable effect on spring deflection motion and rate. Though not nearly as much of a problem when used in a coil over configuration, spring set ups as seen on the front and rear of a stock car certainly can be affected. If the front spring is compressed in a nonlinear manner due to misalignment of the upper load bolt perch or the lower control arm pocket and/or the lower control arm arc is too extreme (as if we have a choice), the sides of the spring will be deflected away from the center line of the spring. This does nothing for the consistency of the spring rate.

The rear springs used on a live axle stock car can also become misaligned due to kinematic problems and/or a series of harmonic misalignments due to rapid scrub motions between the axle housing and the chassis. Remember, if a spring is not compressed squarely, its rate will not be linear.

We spend inordinate amounts of time measuring and setting ride height and corner loads but if we don't have everything right with the springs, we are simply going through an exercise in futility. If the handling goes away, it may not be due to track conditions, tire conditions, the wrong air pressure, damper problems, etc., it instead could simply be a difference in loaded rate or free height.

Preload Length Versus Load Rate - All springs need to be rated according to preload length versus load rate as shown in Table 10-2. In this way, the mechanics can accurately and quickly put in any amount of spring preload required.

Table 10-2
Spring Preload Car# Date

Spring no.	Spring Rate lb./in	Free Length Inches	Preload Length 75	100	150	185	200	250	Number of Turns 75	100	150	185	200	250
51	500	5.030	4.880	4.830	4.730	4.660	4.630	4.530	1.91	2.54	3.81	4.70	5.08	6.35
52	500	5.021	4.871	4.821	4.721	4.651	4.621	4.521	1.91	2.54	3.81	4.70	5.08	6.35
61	600	5.060	4.935	4.893	4.810	4.752	4.727	4.643	1.59	2.12	3.18	3.92	4.24	5.29
62	600	5.060	4.935	4.893	4.810	4.752	4.727	4.643	1.59	2.12	3.18	3.92	4.24	5.29
91	900	5.090	5.007	4.979	4.923	4.884	4.868	4.812	1.06	1.41	2.12	2.61	2.82	3.53
92	900	5.135	5.052	5.024	4.968	4.929	4.913	4.857	1.06	1.41	2.12	2.61	2.82	3.53

Note: Spring # marked at end of closed coil

Coilover Nut Advance Rate
1 Full Turn (6 holes) = 2mm = 0.0787 1 Hole = 0.33 mm = 0.0131"

Decimal to Hole Rotational Conversion Chart

Hole	1	2	3	4	5
Decimal	0.17	0.33	0.5	0.67	0.83

Motion Ratios

Motion Ratio -Motion ratio is the difference in motion between two interconnected components due to some type of mechanical advantage. In Figure 10-13, the mechanical advantage (motion ratio) is 6/12 = 0.5. Therefore, an upward motion on the right side of 0.8371" will result in a downward motion on the left side of 0.8372 * 0.5 = 0.4185". The inverse would be 12/6 = 2. In this case, if we knew the left side motion, we would be able to determine the right side motion - 0.4185 * 2 = 0.8371. Figure 10-14 has the pivot point moved further to the right so that the motion ratio is now 10.3724/7.6276 = 1.35987. Therefore, if the right side moves up 0.5321, the left side moves down 0.5321* 1.35987 = 0.723.

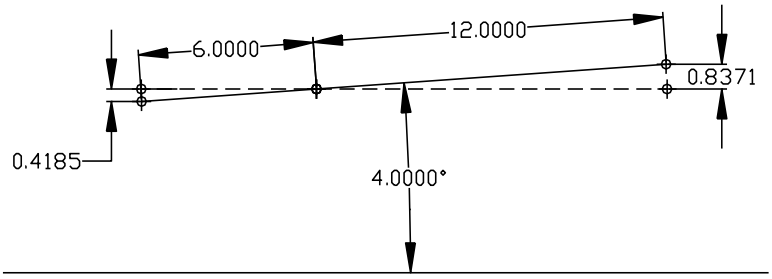


Figure 10-13

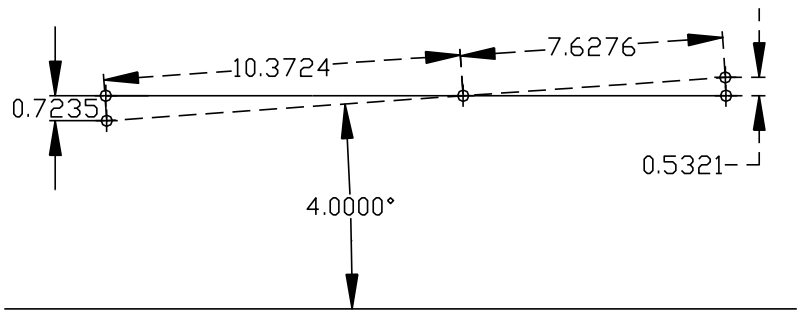


Figure 10-14

Wheel Rate - Figure 10-15 shows a test jig used to demonstrate mechanical advantage. In the example shown, the mechanical advantage (motion ratio) is 6/12 = 0.5. Therefore, if the adjustable rod (wheel travel) is lengthened by 1", the spring will be depressed only 0.5". If the spring rate is1000 lb., the scale under the spring will see a load of 1000* 0.5 = 500 lb. Now we have a spring with a force of 500 lb. pushing on the scale. To find the load at the scale under the adjuster rod (wheel), we need to once again use the mechanical factor. The load at the scale with the adjuster rod is 500 * 0.5 = 250 lb.

- To find the mechanical advantage of the pivot rod and pivot point, we divide the distance to the left of the pivot point by the distance from the right of the pivot point ... (6"/12" = 0.5)
- To find the motion at the spring, we multiply the adjuster (wheel) travel by the mechanical advantage ... (1"* 0.5 = 0.5")
- To find the spring force, we multiply the spring travel by the spring rate ... (0.5" * 1000 = 500 lb.
- To find the force at the adjuster (wheel) scale as the result of the adjuster (wheel) rod being extended 1", we multiply the calculated deflected spring rate by the motion ratio ... (500 lb. * 0.5 = 250 lb.)

From the above examples, we see that to find the resultant load at the adjuster rod (wheel) scale for 1" of travel, we multiply the spring rate by the motion ratio (mechanical advantage) twice (1000 * 0.5 * 0.5) or spring rate * motion ratio squared ... (1000* 0.5^2 = 250). As we will see shortly, when working with a three dimensional suspension, it's not just a matter of finding a simple motion ratio and then squaring it to find the spring rate or wheel rate.

This system operates as a first class lever (rocker arm) as was popular on many of the older formula cars.

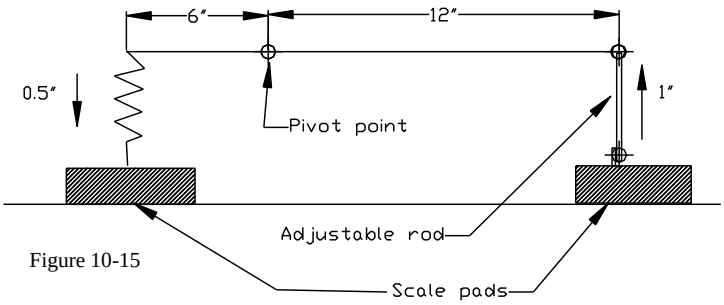


Figure 10-15

Motion Ratios - The Simple Answer - This next section on motion ratios is for the explanation of motion ratio principles and can not be used to find the actual hub versus spring travel and the motion ratio that will determine the load seen by the spring and wheel. The only way to accurately determine the above is through the use of three dimensional software such as WinGeo3 - to be covered shortly.

* All motion ratio and rate calculations in this section of the book assume the tires are infinitely stiff. Chapter 12 introduces tire deflection/squish.

Primary Motion Ratio - Unlike the first class lever as shown in Figure 10-15, the suspension system as illustrated in Figure 10-16 is a second class lever like that used on the front suspension of a stock car. "A" is the distance between the center of the spring perch and the inner lower control arm mounting point. "B" is the distance from the outer lower control arm mount and its inner mount. The motion ratio is A/B (7.4408/15.0 = 0.496). With 1" of wheel movement, the spring would deflect 1 * 0.496 = 0.496". If the rate of the spring is 1000 lb. the spring would now see a force of 1000 * 0.4963 = 496 lb. and the wheel rate (force seen at the contact patch) would be 496 lb. * 0.496 = 246 lb.

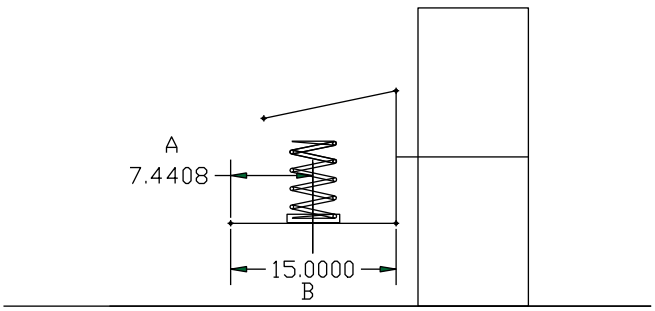


Figure 10-16

Instant Center Motion Ratio - Because the type of suspension system in these examples uses upper and lower control arms of different lengths, the kinematic motion of the two arms, in turn, affect the kinematics of the spring i.e. the amount the spring is deflected. The instant center motion ratio is determined by dividing the distance from the outer lower control arm mount to the instant center (C) by the distance from the center of the hub to the **instant center** (D) C/D. The motion ratio in Figure 10-17 is 30.37/37.58 = **0.808**. Notice this calculation is much greater than the previous 0.496 calculation while maintaining the same geometry. As in the previous example, this is not the actual or three dimensional ratio. The further the instant center is from the center of the hub, the less effect the instant center motion ratio has as seen in Figure 10-18 ... 73.27/80.48 = **0.910**. A short instant center can have a considerable effect on the spring rate as seen in the two previous examples.

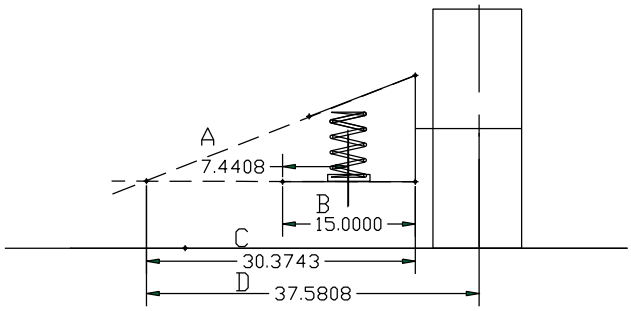


Figure 10-17

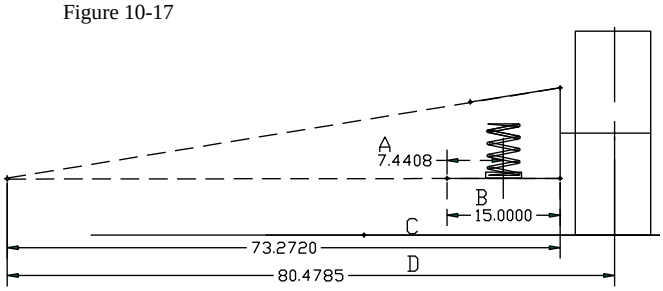


Figure 10-18

Combined Primary and Instant Center Motion Ratios

The resulting motion ratio when the primary and IC ratios are combined except when the IC is at infinity will be less than the primary ratio on its own $(A/B)^2*(C/D)^2$.

Spring Angle Motion Ratio - The angle at which the spring is set also creates a motion ratio. The spring angle motion ratio is the cosine of the angle, or, in our example, cosine $20^0 = 0.940$. As in the previous examples, to accurately calculate the spring angle motion ratio, three dimensional geometry must be used. If a push rod and bell crank system is used, the angles of the push rod and pivot points plus the ratio of the bell crank itself also need to be calculated.

The resulting motion ratio when the spring angle factor is included will, except when the spring is at 90^o, always be less than the combined motion ratio $((A/B)^2*(C/D)^2)*(Cos\theta)^2$.

Determining Wheel Travel Versus Spring Deflection - As can be seen from the previous examples, determining the actual amount of spring deflection versus wheel travel is not that straightforward, and, in fact, for a three dimension suspension system, it can be quite difficult. There are two ways to work out spring deflection versus wheel movement:

1. Physically measuring spring deflection (perch to perch) versus wheel travel (travel at the center of the hub at the rim mounting surface) in small increments ... normally 0.020" or less per iteration - - Book Two. To confirm accuracy, this procedure is done twice - - once moving the hub and measuring the change at the spring and a second time moving the spring and measuring the change at the hub. When measuring hub to spring deflection, a rod with fitted left and right rod ends is affixed to the hub center and the surface plate, (normally requires the fabrication of a special hub), while a dummy damper and linear potentiometer are affixed between the spring/damper mounts. The spring to hub measurements are taken via a rod with fitted left and right rod ends between the spring/damper mounting points while a linear potentiometer is affixed between the hub and the surface plate. If different, as seen on a stock car, the spring and damper motion ratios need to be measured separately. Finally, the anti roll bar and any related apparatus (third spring etc.) needs to be measured.

2. Carefully measuring all suspension co-ordinates and then entering those co-ordinates into a three dimensional software program such as WinGeo3 - - Book Two. When you use WinGeo3 to calculate motion ratio, you will find that the motion ratio is slightly different than the ratio you would get by dividing the spring deflection by the wheel travel. The greater the wheel travel, the greater the difference between the calculated motion ratio and that found by simply dividing the total spring defection by the total wheel travel. When you use the original position and the final position to determine motion ratio, this is an "average" of the various motion ratios. It is not a simple number but is weighted in an indeterminate way. Motion ratio, when using WinGeo3, is calculated using 0.01" iterations in ride height no matter how large the actual ride change. This allows for a much more accurate motion ratio calculation. Ah, the wonders of calculus! For example, for a stock car front suspension with 1" of ride, the instantaneous motion ratio is 0.701" while the motion ratio calculated by dividing the total spring deflection by the total wheel travel is 0.717. When calculating load seen at the contact patch for a 1000 lb. spring, we end up with a 36 lb. difference - $1000 * 0.717^2 = 514$ lb. compared to $1000 * 0.717 * 0.701 =$ 478.25 lb. Over-calculating the load seen at the tire by 36 lb. is quite a lot, even for a stock car suspension system.

Most teams use both methods to check one against the other. Once it has been determined that both methods match i.e. both the measurement of the wheel and spring deflection and the measuring of the suspension co-ordinates and the entering of the co-ordinates into the software program have both been executed with a high degree of accuracy, the computer program can be used for all future calculations. As discussed in Chapter Two, there are many software programs out there that do not accurately reflect suspension kinematic motion.

All motion ratio tests need to be **run in bump and droop** to an amount slightly more than will occur during dynamic conditions.

That Other Team - "Team what's their name is running that whatever pound spring. Don't you think we should be running that spring?" The fact is, unless you know that "the other car" and your car have exactly the same motion ratio, putting in a spring with the same rate is pretty dumb. Duplicating the same wheel rate is less dumb, but not much. Many open wheel teams running in a one make series think that putting in the same spring as their competition will automatically give them the same wheel rate. But that is not necessarily so if both teams are not running the same bell crank angle at set up height or the same spring preload. Irrespective of the spring rate, lateral forces generated by the two chassis can be considerably different, depending on the tire air pressure, camber and caster settings, static and dynamic toe, etc. The following calculators can be found as part of the Kinematics Calculator which is included on the accompanying CD.

Motion Ratios versus Velocity Ratios

Wheel Rate	**(1) LF**	**(2) RF**	**(3) LR**	**(4) RR**
Spring Rate	850	900	225	200
Motion Ratio	0.473	0.455	1	1
Wheel Rate	190	186	225	200

Velocity Ratio				
Motion Ratio	0.473	0.455	1.000	1.000
Velocity Ratio	2.114	2.198	1.000	1.000

Motion Ratio				
Velocity Ratio	2.114	2.198	1.000	1.000
Motion Ratio	0.473	0.455	1.000	1.000

Spring Rate				
Wheel Rate	190	186	225	200
Velocity Ratio	2.114	2.198	1.000	1.000
Spring Rate	850	900	225	200

Roll Rates	**(1) LF**	**(2) RF**	**(3) LR**	**(4) RR**
Spring Rate	850	900	225	200
Motion Ratio	0.473	0.455	1	1
Spring Roll Rate	Front	437	Rear	106
Wheel Roll Rate	Front	94	Rear	106

$MR = 1/VR$

$VR = 1/MR$

$\text{Wheel Rate} = \text{Spring Rate} * MR^2$

$\text{Spring Rate} = \text{Wheel Rate} * VR^2$

Extremely Important Concept: The following memo written by William C. Mitchell is one of the **most important concepts** presented in this book. As Bill did such an elegant job of describing this phenomenon in his memo, I thought I would simply include it verbatim.

Roll Rate

Wm. C. Mitchell

When a car is subjected to a lateral force, there is a resulting torque about the center of gravity. This torque is resisted by the springs (and swaybars, which we are going to ignore for now). This torque (let's call it overturning torque) is resisted by the torques generated by the two springs. These torques are the moment arm (distance from CG to an extension of the spring axis) multiplied by the spring force, which is proportional to the displacement of the spring.

The chassis does not care whether these resisting torques are generated by the outside spring in compression or the inside spring in extension. The effect on the chassis is the same whether it is being pushed or pulled.

Let us assume the two springs have the same moment arm. The combined effective spring rate is K1*K2/(K1+K2). The following pairs of springs have the same roll rate:

Left:	Right:	Roll Rate:
200	400	133.33
300	239	133.33
267	267	133.33
239	300	133.33
400	200	133.33
500	181	133.33
600	171	133.33
800	160	133.33
1000	153	133.33
infinite	133	133.33

All of these spring combinations produce the same roll resistance. They will produce the same amount of roll and the same load transfer. It matters not one bit which pair of springs you select.

But the spring choice does affect the resulting ride height. If you put the stiffer spring on the outside, then the chassis will be higher than if you put the stiffer spring on the inside. If you are concerned about keeping the right front corner of the car from hitting the track while on a high-banked oval (turning left), then placing the stiffer spring on the right side will help to support the right side of the car and keep it from hitting the track. Placing the stiffer spring on the inside or left side will produce the same roll angle but the chassis will be lower.

If you are not concerned about hitting the track but want to minimize ground clearance for aerodynamic effects, then put the stiffer spring on the inside and the chassis will ride lower.

Spring Center:

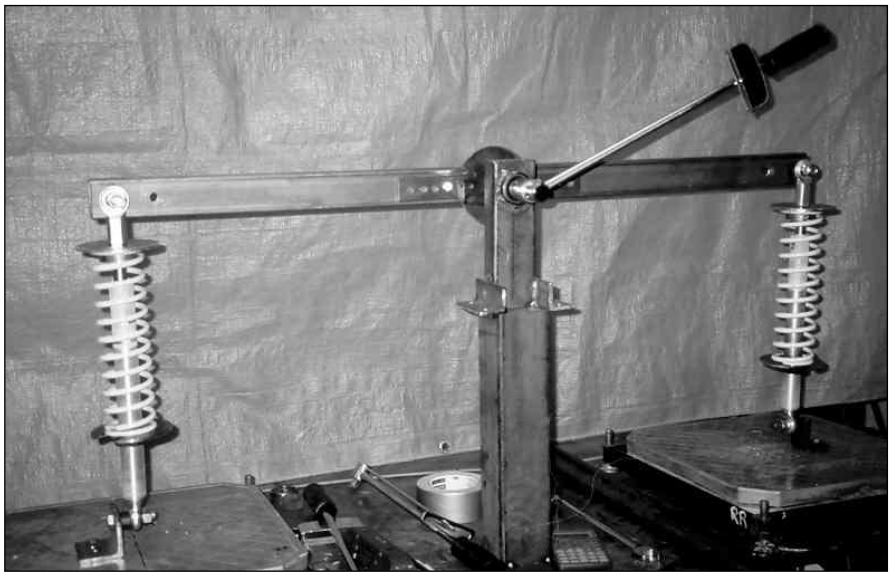
The process just described is called a spring center. If two ends of a beam are supported by springs at either end and we apply a torque to the beam, the spring center is the point that will remain stationary as the beam rotates. If the beam is 100 inches long and we measure the spring center from the left end:

Left:	Right:	Spring Center	R side Drop	
133	infinite	100	0.000	Beam rotates about right end
200	400	66.7	0.333	
301	239	44.3	0.557	
267	267	50	0.500	
239	300	55.7	0.443	
400	200	33.3	0.667	
500	181	26.6	0.734	
600	171	22.2	0.777	
800	60	16.7	0.833	
1000	153	13.3	0.867	
infinite	133	0	1.000	Beam rotates about left end

The spring center formula is SC = 100 * R / (L + R)

Saved as C:\WCM\RollRate.doc October 31, 2002
The End

If you have trouble digesting the above phenomenon, you might want to build a test jig as shown below. However, as in the case of all mechanical test apparatus, one needs to be very careful that they don't bias the outcome. This concept will be much easier to understand once we begin using the Wan31 software beginning in Chapter 14.



Bell Crank and Push Rod Independent Suspension - One of the most common suspension systems found on Formula cars and LMP types of sports cars uses a push rod and bell crank (pivot) to actuate the springs, bars and dampers. The following pictures illustrate a number of different bell crank design ideas in addition to various push rod, spring and bar mounting co-ordinates. Figure 10-19 is used on a Reynard CART Indy car. The top left bell crank hole is the damper/spring mount while the two holes on the right side are the anti roll bar rod mounts. The two anti roll bar mounting positions allow the team to quickly change the anti roll bar motion ratio. Also notice the adjusters at the end of the push rod and toe/steering links that allow the team to adjust the rod length without having to turn the rod itself. The wire coming out of the shrink rap on the rod end is from a strain gauge used to measure loads seen at the spherical bearing. Figure 10-20 shows the rear suspension system of the same car as in the previous example. Notice the difference in geometry between the front and rear bell cranks. Rather than running parallel with the center line of the chassis as in the case of the front dampers, the rear dampers run at a considerably different angle necessitated by various installation considerations.

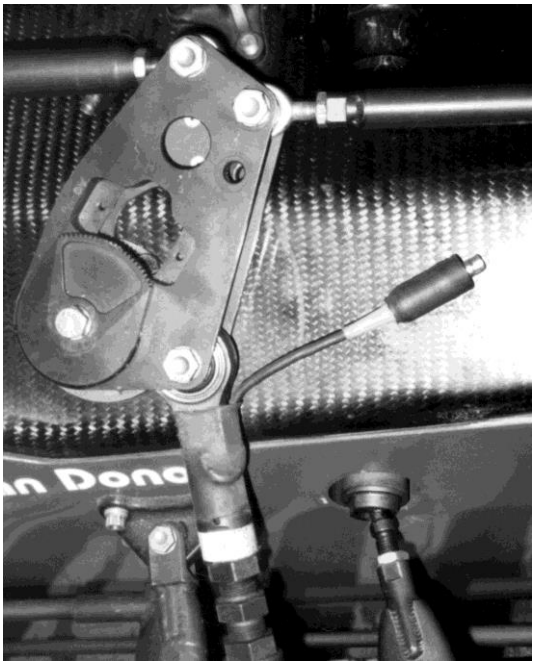


Figure 10-19

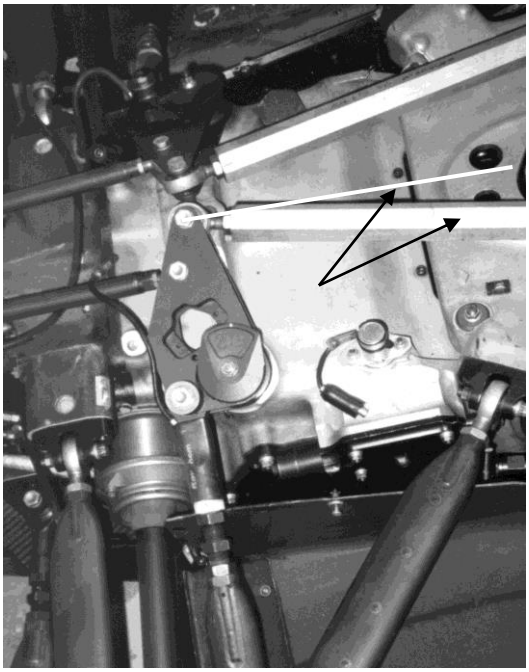


Figure 10-20

Figure 10-21 illustrates the rear push rod installation angle of a Lola CART Indy car while Figure 10-22 shows the lower push rod mounting position at the front of a Reynard Indy car.

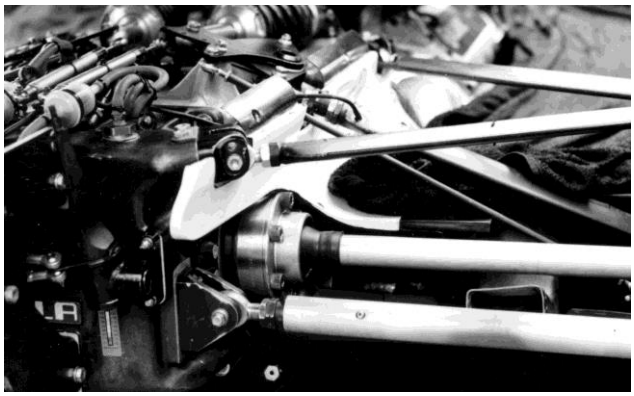


Figure 10-21



Figure 10-22

Figures 10-23 & 10-24 are from a Swift Toyota Atlantic car.

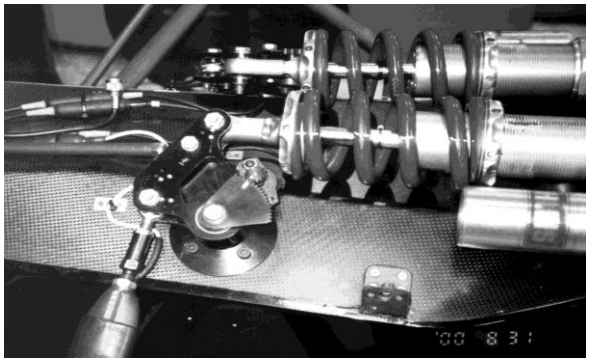


Figure 10-23

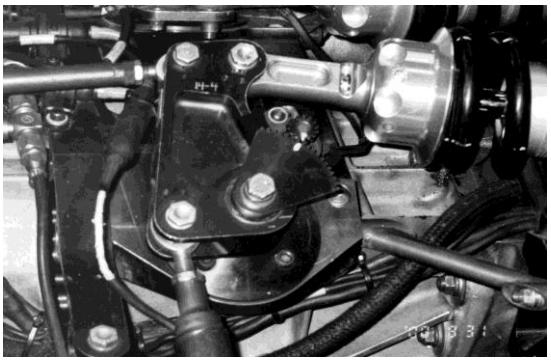


Figure 10-24

Figure 10-25 illustrates a push rod system with two push rod mounting holes while the anti roll bar extension rod mounts are part of the damper extensions. Figure 10-26 shows a bell crank and damper set up from the rear of a FRAM AM 2L car. Notice that in the last two examples, the push rod was mounted above/beyond the bell crank pivot point whereas the previous examples had the push rod mounted just above to slightly below the pivot point.

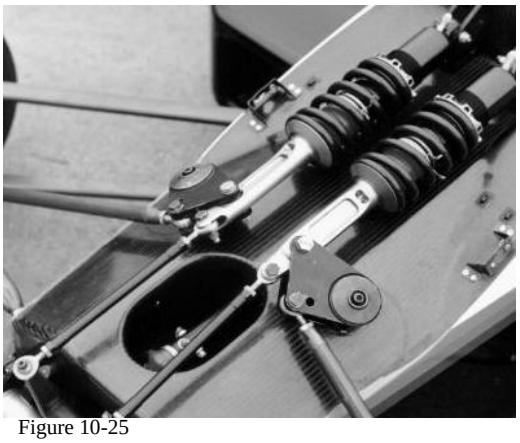


Figure 10-25



Figure 10-26

Figure 10-27 illustrates a system which has the spring mounted vertically while Figure 10-28 shows a bell crank from a Ralt RT41 with the bar mount below the pivot point and the damper mount above the pivot point.

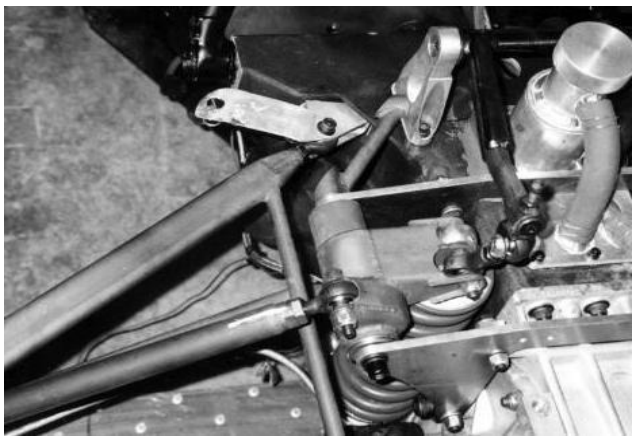


Figure 10-27



Figure 10-28

Figure 10-29 shows dial gauges being used to determine bell crank geometric ratios. Existing and new bell crank ratios should be tested in at least two different ways either using trigonometry, a CAD program, measuring on car motions and/or using three dimensional kinematic software. Spring versus wheel motion almost always needs to be measured at least three times in at least two different ways. It is so easy to make a mistake in this area and the result is that everything from that point on is compromised. The basic bell crank motion ratio is found in the same way as the basic calculations illustrated at the beginning of the motion ratio section (motion divided by motion). Figures 10-30 & 10-31 illustrate a simple bell crank design. In this case, if the push rod end has moved 0.3615" from its static position and the damper end has moved 0.5651", the motion ratio is 0.5651/0.3615 = 1.56.... (2.5985-2.2370 = 0.3615) / (2.8952-2.3301 = 0.3615). But as we have already discovered, the ratio we just calculated is a weighted motion ratio. In order to find the actual motion ratios, we would need to have the cad program run through a number of small motion iterations throughout the intended bump and droop range while calculating the ratios at each iteration. It is quite clear in the second drawing how much the change in both the push rod and damper angle is going to affect the rate of change.

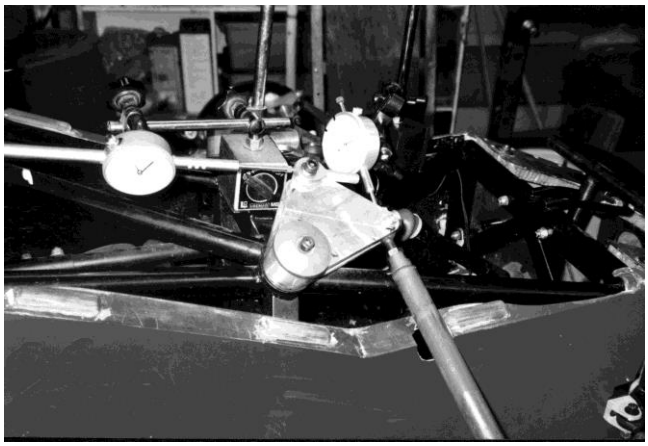


Figure 10-29

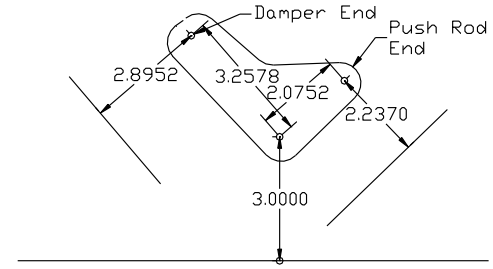


Figure 10-30

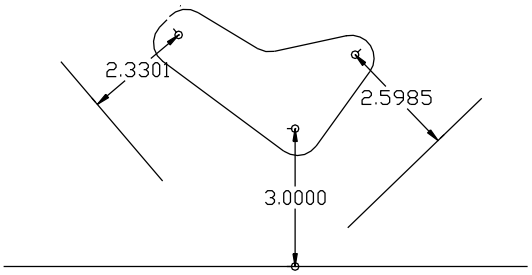


Figure 10-31

The only way to run reams of set up and/or design iterations for any suspension system is to use a program like WinGeo3. Even with a good software program, a race engineer or designer will spend days trying different kinematic possibilities. The following examples are for changes in **ride** and only measure the resultant loads at the wheel contact patch as the result of spring deflection. Combined anti roll bar and spring forces will be introduced in Chapter11 while damper forces will be discussed in the dynamics section of the book. Tables 10-3/10-10 use exactly the same suspension system and spring rate. The only thing that is changed is the bell crank damper lateral mounting position. 2 B - 2 is the standard bell crank, 2 C - 1 has the damper mounting hole moved in towards the center line of the chassis 1" while 2 - A - 3 has the damper mounting hole moved out from the center line of the chassis 2" from the standard unit. Nothing else in the three suspension systems has been changed.

Tables 10-3 & 10-4 use the base bell crank 2 B - 2. Table 10-3 shows 1 1/2" of bump motion while Table 10-4 shows 1" of droop. This is a rising rate system as indicated by the constant increase in motion ratio in bump. If at 1.427" of damper travel the motion ratio was still at the zero damper position of 0.853, we would only see a load at the wheel of 1000 * 1.427 * 0.853 = 1217 lb. rather than 1531 lb. as indicated in the table. Notice that at 1" of bump, the spring load has increased to 914 lb. and the wheel rate has increased to 900 lb. while at 1" of droop the spring load has decreased by only 801 lb. and the wheel load has only decreased by 604 lb. This is the result of the decreasing ratio in droop (rising rate system). This has considerable ramifications when calculating the amount of preload needed to keep load on the spring in droop (providing the system is set up for droop). The bell crank push rod system is very flexible in design which allows the engineer to tailor the rates to match his/her needs. For example, the system can be designed to move from rising to falling rate as the chassis moves above or below the designated ride height and/or to precisely control the droop rates. Rates can also be controlled by the angle of the push rod and/or the dampers.

The various tables in this chapter use a ride change of 0.100" which is many times larger than would be used by the race engineer. This is done to try and keep the tables somewhat manageable for the book.

Table 10-3

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 B - 2

Track - Mudilville Test # 244 - 2 B

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.853	0	0.00	0.00
13.4	0.1	0.1	12.324	0.086	0.086	0.864	0.011	86.00	74.30
13.3	0.1	0.2	12.237	0.087	0.173	0.876	0.012	173.00	151.55
13.2	0.1	0.3	12.149	0.088	0.261	0.888	0.012	261.00	231.77
13.1	0.1	0.4	12.06	0.089	0.350	0.900	0.012	350.00	315.00
13	0.1	0.5	11.969	0.091	0.441	0.913	0.013	441.00	402.63
12.9	0.1	0.6	11.877	0.092	0.533	0.926	0.013	533.00	493.56
12.8	0.1	0.7	11.784	0.093	0.626	0.940	0.014	626.00	588.44
12.7	0.1	0.8	11.689	0.095	0.721	0.954	0.014	721.00	687.83
12.6	0.1	0.9	11.593	0.096	0.817	0.969	0.015	817.00	791.67
12.5	0.1	1	11.496	0.097	0.914	0.985	0.016	914.00	900.29
12.4	0.1	1.1	11.397	0.099	1.013	1.001	0.016	1013.00	1014.01
12.3	0.1	1.2	11.296	0.101	1.114	1.018	0.017	1114.00	1134.05
12.2	0.1	1.3	11.193	0.103	1.217	1.035	0.017	1217.00	1259.60
12.1	0.1	1.4	11.089	0.104	1.321	1.054	0.019	1321.00	1392.33
12	0.1	1.5	10.983	0.106	1.427	1.073	0.019	1427.00	1531.17
					1" Sum	10.068	0.132		

Table 10-4

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 B - 2

Track - Mudilville Test # 244 - 2 B

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.853	0	0.00	0.00
13.6	-0.1	-0.1	12.495	-0.085	-0.085	0.842	-0.011	-85.00	-71.57
13.7	-0.1	-0.2	12.578	-0.083	-0.168	0.831	-0.011	-168.00	-139.61
13.8	-0.1	-0.3	12.661	-0.083	-0.251	0.821	-0.010	-251.00	-206.07
13.9	-0.1	-0.4	12.742	-0.081	-0.332	0.811	-0.010	-332.00	-269.25
14	-0.1	-0.5	12.823	-0.081	-0.413	0.801	-0.010	-413.00	-330.81
14.1	-0.1	-0.6	12.902	-0.079	-0.492	0.791	-0.010	-492.00	-389.17
14.2	-0.1	-0.7	12.981	-0.079	-0.571	0.782	-0.009	-571.00	-446.52
14.3	-0.1	-0.8	13.059	-0.078	-0.649	0.773	-0.009	-649.00	-501.68
14.4	-0.1	-0.9	13.135	-0.076	-0.725	0.764	-0.009	-725.00	-553.90
14.5	-0.1	-1	13.211	-0.076	-0.801	0.755	-0.009	-801.00	-604.76

Figures 10-5 & 10-6 use bell crank 2 C - 1 which has the damper mount moved 1" closer to the center line of the chassis. This is also a rising rate system but the motion ratio at static ride height is much higher at 1.041 as opposed to 0.853. The wheel loads for all corresponding ride changes are considerably higher.

Table 10-5

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 C - 1

Track - Mudilville Test # 245 - 2C

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	1.041	0	0.00	0.00
13.4	0.1	0.1	12.305	0.105	0.105	1.056	0.015	105.00	110.88
13.3	0.1	0.2	12.199	0.106	0.211	1.071	0.015	211.00	225.98
13.2	0.1	0.3	12.091	0.108	0.319	1.087	0.016	319.00	346.75
13.1	0.1	0.4	11.982	0.109	0.428	1.103	0.016	428.00	472.08
13	0.1	0.5	11.871	0.111	0.539	1.120	0.017	539.00	603.68
12.9	0.1	0.6	11.758	0.113	0.652	1.137	0.017	652.00	741.32
12.8	0.1	0.7	11.643	0.115	0.767	1.155	0.018	767.00	885.89
12.7	0.1	0.8	11.527	0.116	0.883	1.174	0.019	883.00	1036.64
12.6	0.1	0.9	11.409	0.118	1.001	1.194	0.020	1001.00	1195.19
12.5	0.1	1	11.289	0.12	1.121	1.214	0.020	1121.00	1360.89
12.4	0.1	1.1	11.166	0.123	1.244	1.236	0.022	1244.00	1537.58
12.3	0.1	1.2	11.042	0.124	1.368	1.258	0.022	1368.00	1720.94
12.2	0.1	1.3	10.915	0.127	1.495	1.282	0.024	1495.00	1916.59
12.1	0.1	1.4	10.786	0.129	1.624	1.306	0.024	1624.00	2120.94
12	0.1	1.5	10.654	0.132	1.756	1.332	0.026	1756.00	2338.99
					1" Sum	12.352	0.173		

Table 10-6

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 C - 1

Track - Mudilville Test # 245 - 2C

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	1.041	0	0.00	0.00
13.6	-0.1	-0.1	12.513	-0.103	-0.103	1.027	-0.014	-103.00	-105.78
13.7	-0.1	-0.2	12.615	-0.102	-0.205	1.013	-0.014	-205.00	-207.67
13.8	-0.1	-0.3	12.716	-0.101	-0.306	1.000	-0.013	-306.00	-306.00
13.9	-0.1	-0.4	12.815	-0.099	-0.405	0.987	-0.013	-405.00	-399.74
14	-0.1	-0.5	12.913	-0.098	-0.503	0.974	-0.013	-503.00	-489.92
14.1	-0.1	-0.6	13.01	-0.097	-0.600	0.962	-0.012	-600.00	-577.20
14.2	-0.1	-0.7	13.105	-0.095	-0.695	0.950	-0.012	-695.00	-660.25
14.3	-0.1	-0.8	13.2	-0.095	-0.790	0.938	-0.012	-790.00	-741.02
14.4	-0.1	-0.9	13.293	-0.093	-0.883	0.927	-0.011	-883.00	-818.54
14.5	-0.1	-1	13.385	-0.092	-0.975	0.916	-0.011	-975.00	-893.1

Tables 10-7 & 10-8 are again from the same suspension system but use bell crank 2 -A- 3 which has its damper mount moved 1" away from the center line of the car. The motion ratio has a rising ratio but starts at a much lower ratio of 0.658 compared to the base 0.853 ratio. The resulting wheel loads per ride change is considerably reduced.

Table 10-7

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 - A - 3

Track - Mudilville Test # 238 - 2A

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.658	0	0.00	0.00
13.4	0.1	0.1	12.344	0.066	0.066	0.666	0.008	66.00	43.96
13.3	0.1	0.2	12.277	0.067	0.133	0.675	0.009	133.00	89.78
13.2	0.1	0.3	12.209	0.068	0.201	0.684	0.009	201.00	137.48
13.1	0.1	0.4	12.14	0.069	0.270	0.693	0.009	270.00	187.11
13	0.1	0.5	12.07	0.07	0.340	0.702	0.009	340.00	238.68
12.9	0.1	0.6	12	0.07	0.410	0.712	0.010	410.00	291.92
12.8	0.1	0.7	11.928	0.072	0.482	0.722	0.010	482.00	348.00
12.7	0.1	0.8	11.855	0.073	0.555	0.733	0.011	555.00	406.82
12.6	0.1	0.9	11.782	0.073	0.628	0.744	0.011	628.00	467.23
12.5	0.1	1	11.707	0.075	0.703	0.755	0.011	703.00	530.77
12.4	0.1	1.1	11.631	0.076	0.779	0.767	0.012	779.00	597.49
12.3	0.1	1.2	11.554	0.077	0.856	0.779	0.012	856.00	666.82
12.2	0.1	1.3	11.475	0.079	0.935	0.792	0.013	935.00	740.52
12.1	0.1	1.4	11.395	0.080	1.015	0.806	0.014	1015.00	818.09
12	0.1	1.5	11.315	0.080	1.095	0.820	0.014	1095.00	897.90
					1" Sum	7.744	0.097		

Table 10-8

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod Car # - 8823 Bell crank 2 - A - 3

Track - Mudilville Test # 238 - 2A

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.658	0	0.00	0.00
13.6	-0.1	-0.1	12.475	-0.065	-0.065	0.650	-0.008	-65.00	-42.25
13.7	-0.1	-0.2	12.54	-0.065	-0.13	0.642	-0.008	-130.00	-83.46
13.8	-0.1	-0.3	12.604	-0.064	-0.194	0.634	-0.008	-194.00	-123.00
13.9	-0.1	-0.4	12.667	-0.063	-0.257	0.626	-0.008	-257.00	-160.88
14	-0.1	-0.5	12.729	-0.062	-0.319	0.619	-0.007	-319.00	-197.46
14.1	-0.1	-0.6	12.79	-0.061	-0.380	0.612	-0.007	-380.00	-232.56
14.2	-0.1	-0.7	12.851	-0.061	-0.441	0.605	-0.007	-441.00	-266.81
14.3	-0.1	-0.8	12.911	-0.06	-0.501	0.598	-0.007	-501.00	-299.60
14.4	-0.1	-0.9	12.971	-0.060	-0.561	0.591	-0.007	-561.00	-331.55
14.5	-0.1	-1	13.029	-0.058	-0.619	0.584	-0.007	-619.00	-361.5

Table 10-9 shows the resulting change in motion ratios, spring loads and wheel loads as a result of the damper chassis mount being moved away from the chassis center line by 3". This table shows a wheel load of 845 lb. at 1" of ride change compared to 900 lb. for the same ride change seen in the base model.

Table 10-9

Motion Ratio Measurement - Wheel Versus Spring Motion — **Bump**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod — Car # - 8823 — Bell crank 2 B - 2

Track - Mudilville — Test # 244 - 2 B — Spring angle #22-19 — Out from c/l at chassis mount

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.797	0	0.00	0.00
13.4	0.1	0.1	12.33	0.080	0.08	0.811	0.014	80.00	64.88
13.3	0.1	0.2	12.248	0.082	0.162	0.826	0.015	162.00	133.81
13.2	0.1	0.3	12.165	0.083	0.245	0.841	0.015	245.00	206.05
13.1	0.1	0.4	12.08	0.085	0.330	0.856	0.015	330.00	282.48
13	0.1	0.5	11.994	0.086	0.416	0.873	0.017	416.00	363.17
12.9	0.1	0.6	11.905	0.089	0.505	0.890	0.017	505.00	449.45
12.8	0.1	0.7	11.816	0.089	0.594	0.908	0.018	594.00	539.35
12.7	0.1	0.8	11.724	0.092	0.686	0.926	0.018	686.00	635.24
12.6	0.1	0.9	11.631	0.093	0.779	0.946	0.020	779.00	736.93
12.5	0.1	1	11.535	0.096	0.875	0.966	0.020	875.00	845.25
					1" Sum	9.640	0.169		

Table 10-10 represents the base bell crank but with the lower/suspension mount end of the push rod moved in and down 1". Notice that at 1" of bump the wheel load is 855 lb. compared to 845 lb. in the previous example despite the fact that the motion ratio is lower in this last example. The motion ratio is lower but the amount the spring is compressed has increased to 0.892 compared to 0.875 in the previous example. Like everything else in racing, motion ratios are always a compromise. The team with the best set of compromises wins.

Table 10-10

Motion Ratio Measurement - Wheel Versus Spring Motion — **Bump**

Spring Rate Versus Wheel Rate

Open Wheel - Push Rod — Car # - 8823 — Bell crank 2 B - 2

Track - Mudilville — Test # 249 - 3 B — Push Rod angle increased - lower mount down 1" and in 1"

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	12.41	0.000	0.000	0.834	0	0.00	0.00
13.4	0.1	0.1	12.326	0.084	0.084	0.845	0.011	84.00	70.98
13.3	0.1	0.2	12.241	0.085	0.169	0.856	0.011	169.00	144.66
13.2	0.1	0.3	12.155	0.086	0.255	0.867	0.011	255.00	221.09
13.1	0.1	0.4	12.068	0.087	0.342	0.879	0.012	342.00	300.62
13	0.1	0.5	11.979	0.089	0.431	0.891	0.012	431.00	384.02
12.9	0.1	0.6	11.89	0.089	0.520	0.903	0.012	520.00	469.56
12.8	0.1	0.7	11.799	0.091	0.611	0.916	0.013	611.00	559.68
12.7	0.1	0.8	11.707	0.092	0.703	0.930	0.014	703.00	653.79
12.6	0.1	0.9	11.613	0.094	0.797	0.944	0.014	797.00	752.37
12.5	0.1	1	11.518	0.095	0.892	0.958	0.014	892.00	854.54
					1" Sum	9.823	0.124		

Coil Over Independent Suspension - (Live axle coil over systems will be discussed later in this chapter) Suspension systems known as coil over suspensions have their spring/damper set ups mounted outside the frame as shown in Figures 10-32/10-35. This type of system is used on both independent and live axle suspensions. Figure 10-32 shows a front independent set up used on a TransAm car while Figure 10-33 illustrates a rear live axle set up on the same car. Figure 10-34 is from the front suspension of a Howe™ short track oval car. Figure 10-35 is from the same oval track car but showing the rear live axle coil over suspension.

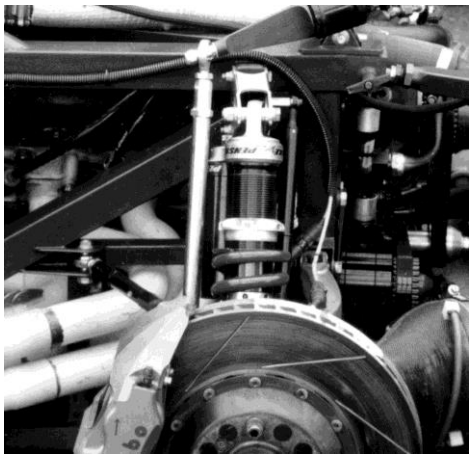


Figure 10-32

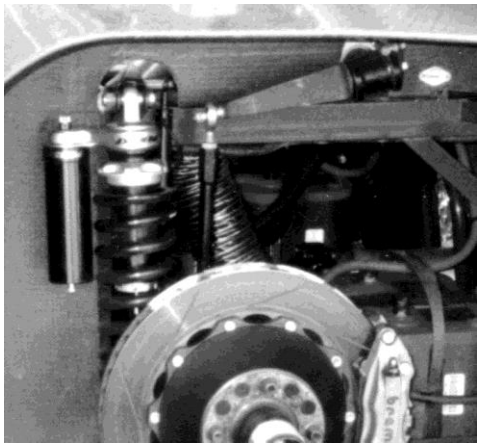


Figure 10-33

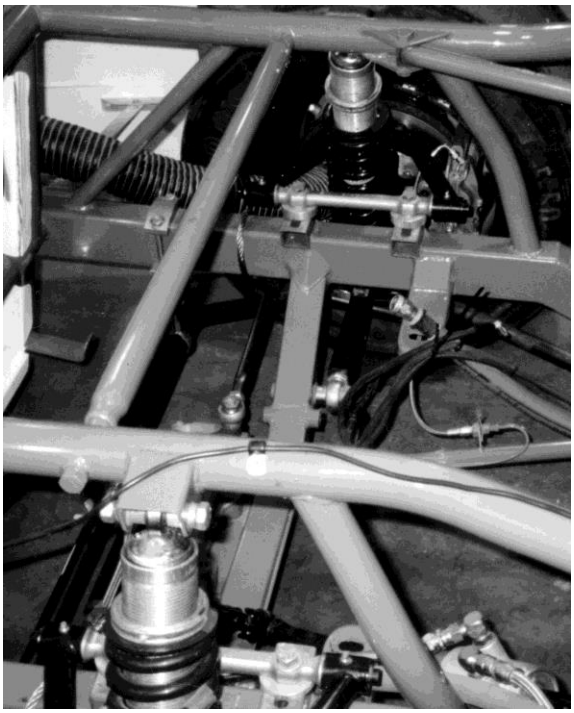


Figure 10-34

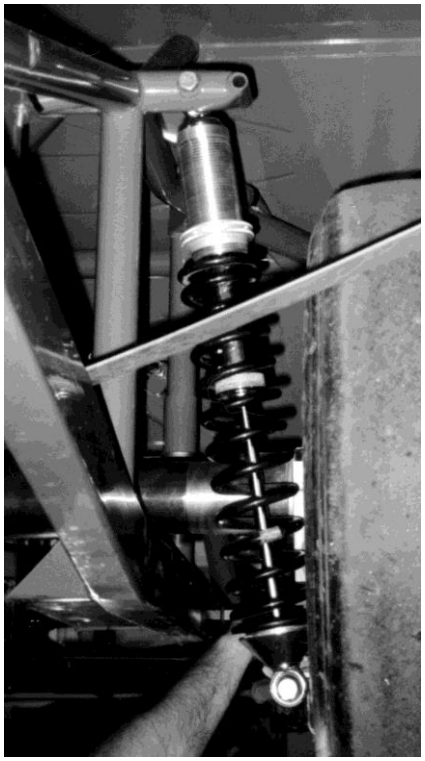


Figure 10-35

Coil Over Independent Systems - The following tables illustrate the motion ratios, spring rates and wheel rates from a typical GT car. Notice that in all of these examples the motion ratio has a falling rate. At 1" of droop the spring has lost 1045 lb. of load while at 1" of bump the spring has gained 1012 lb. of load. Though the motion ratio is falling, it is much less pronounced than in the bell crank/push rod systems previously outlined. The bell crank systems routinely saw an increase in motion ratio of 0.011 to 0.019 per 0.100" of compression while the coil over examples only range between -0.003 and -0.004 per 1" of ride change.

Table 10-11

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Coil Over GT # 483b-61 - Right Front

Track - Mudilville Test # 0806 -q

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	18.439	0.000	0.000	1.029	0	0.00	0.00
13.4	0.1	0.1	18.336	0.103	0.103	1.026	-0.003	103.00	105.68
13.3	0.1	0.2	18.234	0.102	0.205	1.022	-0.004	205.00	209.51
13.2	0.1	0.3	18.132	0.102	0.307	1.019	-0.003	307.00	312.83
13.1	0.1	0.4	18.03	0.102	0.409	1.015	-0.004	409.00	415.14
13	0.1	0.5	17.929	0.101	0.510	1.012	-0.003	510.00	516.12
12.9	0.1	0.6	17.828	0.101	0.611	1.008	-0.004	611.00	615.89
12.8	0.1	0.7	17.727	0.101	0.712	1.005	-0.003	712.00	715.56
12.7	0.1	0.8	17.627	0.100	0.812	1.001	-0.004	812.00	812.81
12.6	0.1	0.9	17.527	0.100	0.912	0.997	-0.004	912.00	909.26
12.5	0.1	1	17.427	0.100	1.012	0.994	-0.003	1012.00	1005.93
12.4	0.1	1.1	17.328	0.099	1.111	0.990	-0.004	1111.00	1099.89
12.3	0.1	1.2	17.229	0.099	1.210	0.986	-0.004	1210.00	1193.06
12.2	0.1	1.3	17.131	0.098	1.308	0.982	-0.004	1308.00	1284.46
12.1	0.1	1.4	17.033	0.098	1.406	0.978	-0.004	1406.00	1375.07
12	0.1	1.5	16.935	0.098	1.504	0.974	-0.004	1504.00	1464.90
					1" Sum	11.128	-0.035		

Table 10-12

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Coil Over GT # 483b-61 - Right Front

Track - Mudilville Test # 0806 -q

Software - WinGeo3 Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	18.439	0.000	0.000	1.029	0	0.00	0.00
13.6	-0.1	-0.1	18.542	-0.103	-0.103	1.032	0.003	-103.00	-106.30
13.7	-0.1	-0.2	18.645	-0.103	-0.206	1.035	0.003	-206.00	-213.21
13.8	-0.1	-0.3	18.749	-0.104	-0.31	1.038	0.003	-310.00	-321.78
13.9	-0.1	-0.4	18.853	-0.104	-0.414	1.042	0.004	-414.00	-431.39
14	-0.1	-0.5	18.958	-0.105	-0.519	1.045	0.003	-519.00	-542.36
14.1	-0.1	-0.6	19.062	-0.104	-0.623	1.048	0.003	-623.00	-652.90
14.2	-0.1	-0.7	19.167	-0.105	-0.728	1.051	0.003	-728.00	-765.13
14.3	-0.1	-0.8	19.272	-0.105	-0.833	1.054	0.003	-833.00	-877.98
14.4	-0.1	-0.9	19.378	-0.106	-0.939	1.057	0.003	-939.00	-992.52
14.5	-0.1	-1	19.484	-0.106	-1.045	1.060	0.003	-1045.00	-1107.70

Tables 10-13 & 10-14 are the same as in the previous examples except the angle of the spring is increased by moving the top of the spring in towards the chassis center line. Increasing the coil over angle decreases the motion ratio, spring load and wheel load but it also slightly decreases the loss in motion ratio per 1" of ride change.

Table 10-13

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Coil Over GT # 483b-61 - Right Front — Coil Over angle increased (2824p)

Track - Mudilville — Test # 0809 -qt

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	18.439	0.000	0.000	0.931	0	0.00	0.00
13.4	0.1	0.1	18.346	0.093	0.093	0.929	-0.002	93.00	86.40
13.3	0.1	0.2	18.253	0.093	0.186	0.927	-0.002	186.00	172.42
13.2	0.1	0.3	18.16	0.093	0.279	0.926	-0.001	279.00	258.35
13.1	0.1	0.4	18.068	0.092	0.371	0.924	-0.002	371.00	342.80
13	0.1	0.5	17.976	0.092	0.463	0.922	-0.002	463.00	426.89
12.9	0.1	0.6	17.884	0.092	0.555	0.920	-0.002	555.00	510.60
12.8	0.1	0.7	17.792	0.092	0.647	0.918	-0.002	647.00	593.95
12.7	0.1	0.8	17.7	0.092	0.739	0.915	-0.003	739.00	676.19
12.6	0.1	0.9	17.609	0.091	0.830	0.913	-0.002	830.00	757.79
12.5	0.1	1	17.517	0.092	0.922	0.911	-0.002	922.00	839.94
12.4	0.1	1.1	17.426	0.091	1.013	0.909	-0.002	1013.00	920.82
12.3	0.1	1.2	17.336	0.090	1.103	0.906	-0.003	1103.00	999.32
12.2	0.1	1.3	17.245	0.091	1.194	0.904	-0.002	1194.00	1079.38
12.1	0.1	1.4	17.155	0.090	1.284	0.902	-0.002	1284.00	1158.17
12	0.1	1.5	17.065	0.090	1.374	0.899	-0.003	1374.00	1235.23
					1" Sum	10.136	-0.02		

Table 10-14

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Coil Over GT # 483b-61 - Right Front — Coil Over angle increased (2824p)

Track - Mudilville — Test # 0809 -qt

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	18.439	0.000	0.000	0.931	0	0.00	0.00
13.6	-0.1	-0.1	18.532	-0.093	-0.093	0.933	0.002	-93.00	-86.77
13.7	-0.1	-0.2	18.626	-0.094	-0.187	0.935	0.002	-187.00	-174.85
13.8	-0.1	-0.3	18.719	-0.093	-0.28	0.936	0.001	-280.00	-262.08
13.9	-0.1	-0.4	18.813	-0.094	-0.374	0.938	0.002	-374.00	-350.81
14	-0.1	-0.5	18.907	-0.094	-0.468	0.939	0.001	-468.00	-439.45
14.1	-0.1	-0.6	19.001	-0.094	-0.562	0.941	0.002	-562.00	-528.84
14.2	-0.1	-0.7	19.095	-0.094	-0.656	0.943	0.002	-656.00	-618.61
14.3	-0.1	-0.8	19.189	-0.094	-0.750	0.944	0.001	-750.00	-708.00
14.4	-0.1	-0.9	19.284	-0.095	-0.845	0.946	0.002	-845.00	-799.37
14.5	-0.1	-1	19.378	-0.094	-0.939	0.947	0.001	-939.00	-889.23

As can be seen in Table 10-15, substantially increasing the coil over angle results in the ride height motion ratio beginning at a considerably lower ratio (requiring a stiffer spring to obtain the same wheel load/rate). The loss of motion ratio throughout the suspension travel range is very small, averaging only 0.001 per 0.100" of ride change i.e. the falling motion rate is reduced only **slightly** per ride motion iteration.

Table 10-15

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Coil Over GT # 483b-61 - Right Front — Top of Coil Over moved in 11" from base suspension (2992-q)

Track - Mudilville — Test # 0809 -qt

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Load	Wheel Load
13.5	0	0	18.439	0.000	0.000	0.625	0	0.00	0.00
13.4	0.1	0.1	18.376	0.063	0.063	0.626	0.001	63.00	39.44
13.3	0.1	0.2	18.314	0.062	0.125	0.627	0.001	125.00	78.38
13.2	0.1	0.3	18.251	0.063	0.188	0.628	0.001	188.00	118.06
13.1	0.1	0.4	18.188	0.063	0.251	0.628	0.000	251.00	157.63
13	0.1	0.5	18.125	0.063	0.314	0.629	0.001	314.00	197.51
12.9	0.1	0.6	18.062	0.063	0.377	0.630	0.001	377.00	237.51
12.8	0.1	0.7	17.999	0.063	0.440	0.631	0.001	440.00	277.64
12.7	0.1	0.8	17.936	0.063	0.503	0.631	0.000	503.00	317.39
12.6	0.1	0.9	17.873	0.063	0.566	0.632	0.001	566.00	357.71
12.5	0.1	1	17.81	0.063	0.629	0.633	0.001	629.00	398.16
					1" Sum	6.920	0.008		

Figure 10-36 shows a GT suspension with the damper/spring set at a considerable angle (the damper/spring is replaced by a set up rod in this picture). This type of suspension is often used on GT types of chassis that use engines with a small horizontal (y axis) cross section. This allows the designer to use a much narrower frame.

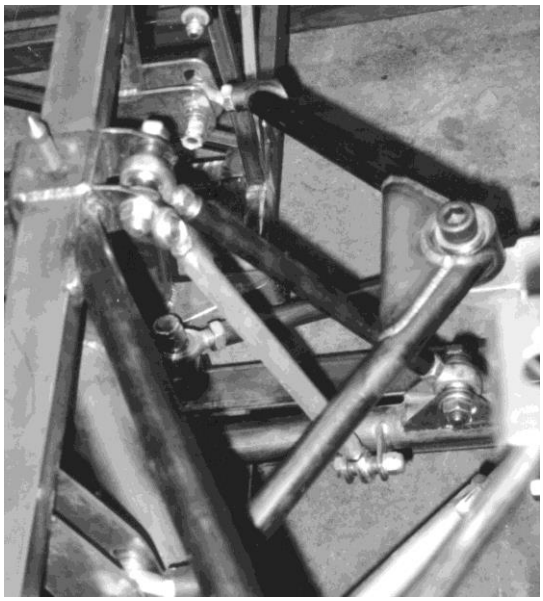


Figure 10-36

Stock Car Independent Suspension - A stock car front suspension with its short arcs and tightly controlled rules is one of the most difficult suspension systems to get right in terms of motion ratios, spring loads/rates and wheel loads/rates. Even using the best springs and having an accurate spring rate tester to constantly check them, the wheel loads seen by the chassis compared to the theoretical loads generated from software can be quite different. This paradoxical problem is partly the result of the short arcs generated by the upper and lower control arms, compliance problems just about everywhere and how the spring fits the upper and lower spring perches.

Because of the short suspension arms and large amounts of suspension motion and despite the fact that teams spend a great deal of time trying to prevent it, the springs are often not compressed in a linear manner. High speed photography reveals that the springs are often warped, sometimes resulting in more that one frequency node. If this situation is duplicated in an accurate spring tester, the changes in spring load per degree of deflection can be considerable. Additionally, the inside spring can become sufficiently uncoupled in droop to allow the spring to slide and/or rotate ever so slightly on the upper and lower spring perch. It only takes a repositioning of a few thousandths to change the spring load. If a hole is simply drilled in the upper spring perch for the end of the load bolt to fit through, this often results in considerable binding, which, in turn, can cock the spring at an angle with the bottom perch. This can happen at both the inside and outside suspension corners. Most teams now use a spherical bearing to prevent load bolt to perch binding. The spherical bearing allows the spring perch to operate at an angle which helps keep it parallel with the mounting surface of the spring.

When we consider that two of the main problems with a portable spring tester are:

- it is not rigid enough even though it is normally many times more rigid than the suspension system and the load bolt mounts.

- it doesn't compress the spring in a perfectly linear manner even though it normally does a much better job than how the suspension system compresses the spring.

It's easy to see why we may have a problem producing constant and repeatable spring and wheel loads for a given iteration of ride change.

One reason it is so important that we know the precise rate and percentage of linear repeatability of our spring is so we can eliminate the spring as a possible cause for any particular handling anomaly. Oh yes, finding ways to accurately mount load cells and strain gauges on this type of suspension system is also **next** to impossible. (Nothing is impossible).

As previously discussed, the front springs used with this type of suspension have the last coil on the bottom of the spring open. Figures 10-37 & 10-38 show a spring pocket built into the lower control arm. In a world where lightness is everything, making this mount rigid is a constant challenge.

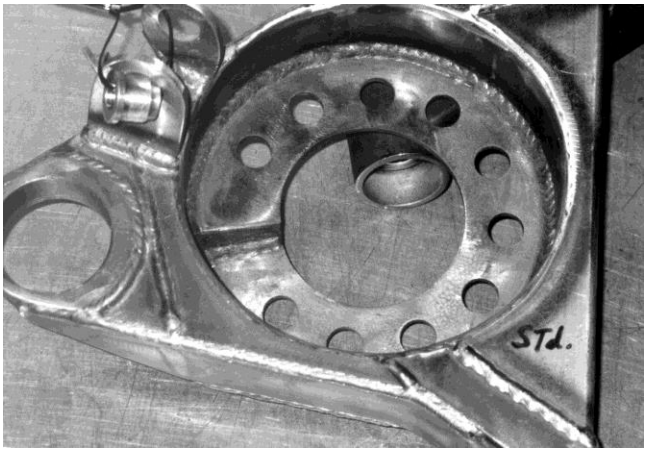
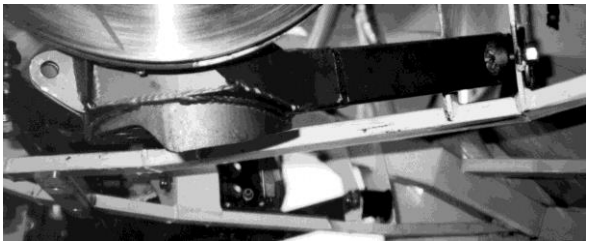


Figure 10-37


Figure 10-38

With this type of suspension system, the distance between the spring perches is adjusted by turning a load bolt in or out as shown in Figures 10-39 & 10-40 - changing the spring preload. Notice that the load bolt and top spring perch are angled to help match the change in spring angle as the suspension moves into bump. Compliance between the load bolt and load bolt nut, plus compliance between the end of the load bolt and the spring perch is often a significant problem.

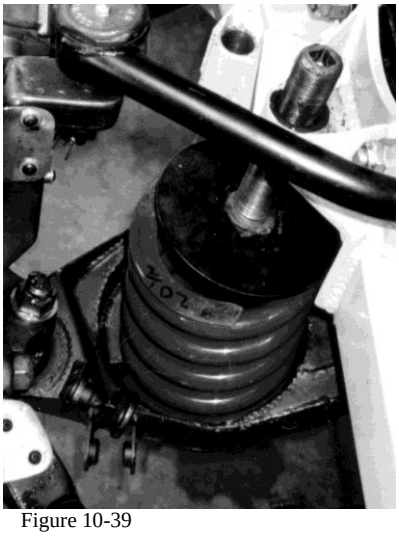


Figure 10-39

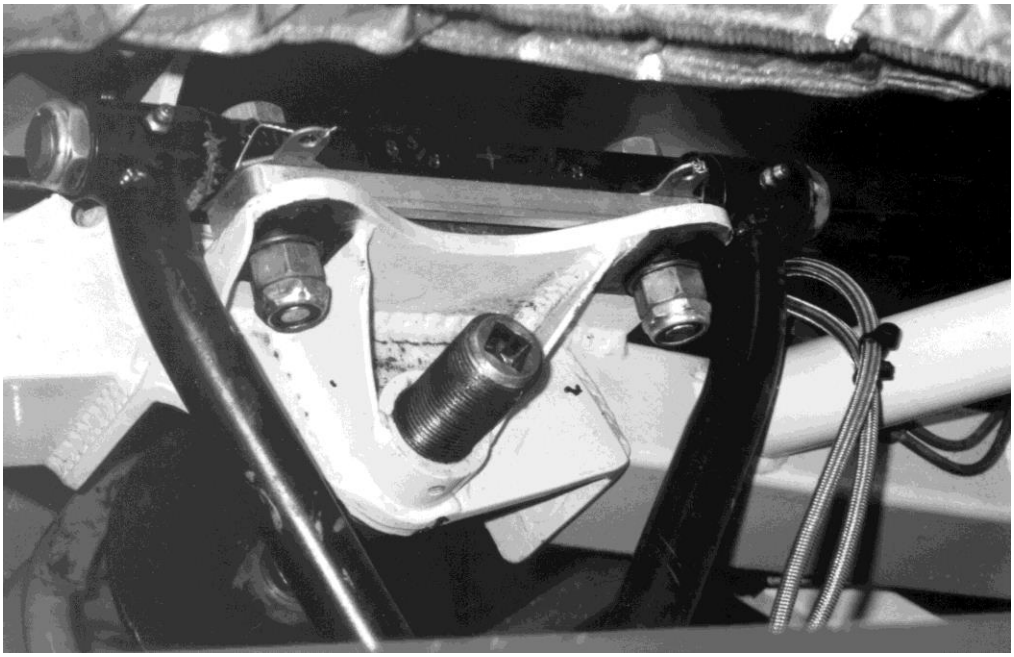


Figure 10-40

Tables 10-16 & 10-17 illustrate the motion ratios found on a typical stock car front suspension. This type of suspension always has a low initial motion ratio and a fairly severe falling rate because of the distance the spring centerline is from the lower ball joint. Notice that for this 1000 lb. spring rate suspension set up, the 1" ride travel wheel load is 503 lb. while the base coil over suspension example had a 1" ride travel wheel load of 1006 lb. The wheel load/rate is half the spring rate of 1000 lb./in.

Table 10-16

Stock Car # 23b-441 - Right Front — **Bump**

Track - Mudilville — Test # 1812 -b

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.732	0	0.00	0.00
13.4	0.1	0.1	7.989	0.073	0.073	0.729	-0.003	73.00	53.22
13.3	0.1	0.2	7.916	0.073	0.146	0.726	-0.003	146.00	106.00
13.2	0.1	0.3	7.844	0.072	0.218	0.723	-0.003	218.00	157.61
13.1	0.1	0.4	7.772	0.072	0.290	0.720	-0.003	290.00	208.80
13	0.1	0.5	7.7	0.072	0.362	0.717	-0.003	362.00	259.55
12.9	0.1	0.6	7.628	0.072	0.434	0.714	-0.003	434.00	309.88
12.8	0.1	0.7	7.557	0.071	0.505	0.711	-0.003	505.00	359.06
12.7	0.1	0.8	7.486	0.071	0.576	0.707	-0.004	576.00	407.23
12.6	0.1	0.9	7.416	0.070	0.646	0.704	-0.003	646.00	454.78
12.5	0.1	1	7.345	0.071	0.717	0.701	-0.003	717.00	502.62
12.4	0.1	1.1	7.275	0.070	0.787	0.698	-0.003	787.00	549.33
12.3	0.1	1.2	7.206	0.069	0.856	0.694	-0.004	856.00	594.06
12.2	0.1	1.3	7.136	0.070	0.926	0.691	-0.003	926.00	639.87
12.1	0.1	1.4	7.068	0.068	0.994	0.687	-0.004	994.00	682.88
12	0.1	1.5	6.999	0.069	1.063	0.684	-0.003	727.09	727.09
					1" Sum	7.884	-0.031		

Table 10-17

Motion Ratio Measurement - Wheel Versus Spring Motion — **Droop**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441 - Right Front

Track - Mudilville — Test # 1812 -b

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.732	0	0.00	0.00
13.6	-0.1	-0.1	8.135	0.073	-0.073	0.735	0.003	-73.00	-53.66
13.7	-0.1	-0.2	8.209	-0.074	-0.147	0.737	0.002	-147.00	-108.34
13.8	-0.1	-0.3	8.283	-0.074	-0.221	0.740	0.003	-221.00	-163.54
13.9	-0.1	-0.4	8.357	-0.074	-0.295	0.743	0.003	-295.00	-219.19
14	-0.1	-0.5	8.431	-0.074	-0.369	0.746	0.003	-369.00	-275.27
14.1	-0.1	-0.6	8.506	-0.075	-0.444	0.748	0.002	-444.00	-332.11
14.2	-0.1	-0.7	8.581	-0.075	-0.519	0.751	0.003	-519.00	-389.77
14.3	-0.1	-0.8	8.656	-0.075	-0.594	0.754	0.003	-594.00	-447.88
14.4	-0.1	-0.9	8.732	-0.076	-0.670	0.756	0.002	-670.00	-506.52
14.5	-0.1	-1	8.808	-0.076	-0.746	0.759	0.003	-746.00	-566.21

Tables 10-18 & 10-19 use the same suspension as in the previous example except the spring is moved out 1" towards the lower ball joint. This increases the wheel load/rate at 1" of ride travel by 88 lb.

Table 10-18

Motion Ratio Measurement - Wheel Versus Spring Motion **Bump**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441 - Right Front — Spring co-ordinate change 1' towards wheel - #24-28q

Track - Mudilville — Test # 1816 -dd

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.793	0	0.00	0.00
13.4	0.1	0.1	7.983	0.079	0.079	0.790	-0.003	79.00	62.41
13.3	0.1	0.2	7.904	0.079	0.158	0.786	-0.004	158.00	124.19
13.2	0.1	0.3	7.826	0.078	0.236	0.783	-0.003	236.00	184.79
13.1	0.1	0.4	7.747	0.079	0.315	0.780	-0.003	315.00	245.70
13	0.1	0.5	7.67	0.077	0.392	0.777	-0.003	392.00	304.58
12.9	0.1	0.6	7.592	0.078	0.470	0.774	-0.003	470.00	363.78
12.8	0.1	0.7	7.515	0.077	0.547	0.770	-0.004	547.00	421.19
12.7	0.1	0.8	7.438	0.077	0.624	0.767	-0.003	624.00	478.61
12.6	0.1	0.9	7.361	0.077	0.701	0.763	-0.004	701.00	534.86
12.5	0.1	1	7.285	0.076	0.777	0.760	-0.003	777.00	590.52
12.4	0.1	1.1	7.209	0.076	0.853	0.756	-0.004	853.00	644.87
12.3	0.1	1.2	7.134	0.075	0.928	0.753	-0.003	928.00	698.78
12.2	0.1	1.3	7.059	0.075	1.003	0.749	-0.004	1003.00	751.25
12.1	0.1	1.4	6.984	0.075	1.078	0.745	-0.004	1078.00	803.11
12	0.1	1.5	6.910	0.074	1.152	0.742	-0.003	1152.00	854.78
					1" Sum	8.543	-0.033		

Table 10-19

Motion Ratio Measurement - Wheel Versus Spring Motion **Droop**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441 - Right Front — Spring co-ordinate change 1" towards wheel - #24-28q

Track - Mudilville — Test # 1816 -dd

Software - WinGeo3 — Spring Rate 1000

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.793	0	0.00	0.00
13.6	-0.1	-0.1	8.141	-0.079	-0.079	0.796	0.003	-79.00	-62.88
13.7	-0.1	-0.2	8.221	-0.08	-0.159	0.799	0.003	-159.00	-127.04
13.8	-0.1	-0.3	8.301	-0.08	-0.239	0.802	0.003	-239.00	-191.68
13.9	-0.1	-0.4	8.382	-0.081	-0.320	0.805	0.003	-320.00	-257.60
14	-0.1	-0.5	8.462	-0.08	-0.400	0.808	0.003	-400.00	-323.20
14.1	-0.1	-0.6	8.543	-0.081	-0.481	0.810	0.002	-481.00	-389.61
14.2	-0.1	-0.7	8.624	-0.081	-0.562	0.813	0.003	-562.00	-456.91
14.3	-0.1	-0.8	8.706	-0.082	-0.644	0.816	0.003	-644.00	-525.50
14.4	-0.1	-0.9	8.788	-0.082	-0.726	0.819	0.003	-726.00	-594.59
14.5	-0.1	-1	8.87	-0.082	-0.808	0.822	0.003	-808.00	-664.18

Live Axle Motion Ratios

Motion Ratios in Ride – Side View

Live Axle - Springs on Top of Axle Housing - In ride (equal two wheel bump) the springs are compressed the same amount and the motion ratio will be 1. Over one wheel bump, both springs will see some deflection with the spring closest to the wheel that is in bumped seeing the greater motion. It is the combined motion of the two springs that determines the motion ratio acting on the bumped wheel. If two equally rated springs are mounted an equal distance from the center line of the tire, then the motion ratio at the bumped wheel will still be very close to 1 because the combined motion of the two springs will almost equal the total amount of motion seen at the bumped wheel. The end of the axle that is not bumped will pivot about the tire. Figure 10-41 shows a live axle with a wheelbase of 66", the springs are 15" from the center of the tires and 18" center to center from each other. Figure 10-42 shows the same live axle having one wheel bump of 1.7427" with the combined motion of the two springs equalling 1.7322" - (7.3782 - 6.0368 = 1.3414), (7.3782 - 6.9874 = 0.3908), (1.3414 + 0.3908 = 1.7322").

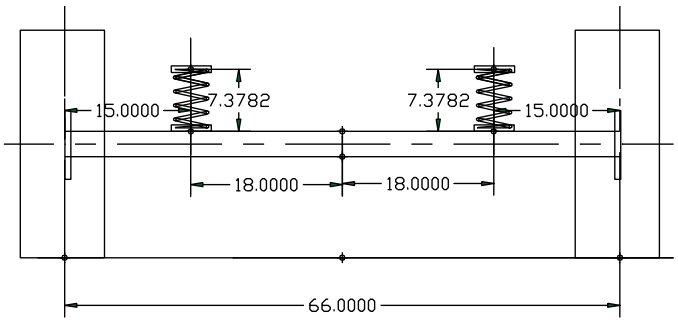


Figure 10-41

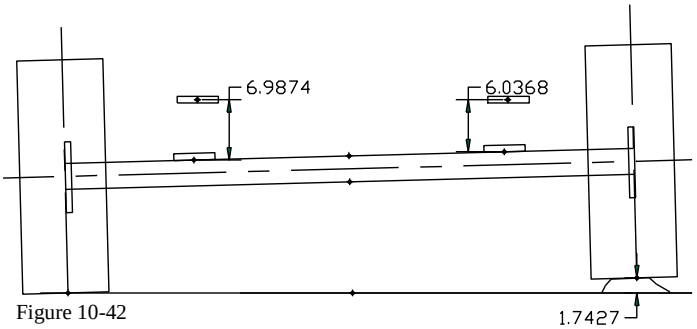


Figure 10-42

Maintaining a motion ratio of 1 in ride is the primary reason why a designer would mount the springs on top of the axle housing.

Figure 10-43 illustrates the misalignment between the upper and lower spring perches as the result of 1.74" of one wheel bump. If the angle of the track (panhard) bar results in live axle scrub, this can either increase or decrease the misalignment depending on which way the axle is scrubbed.

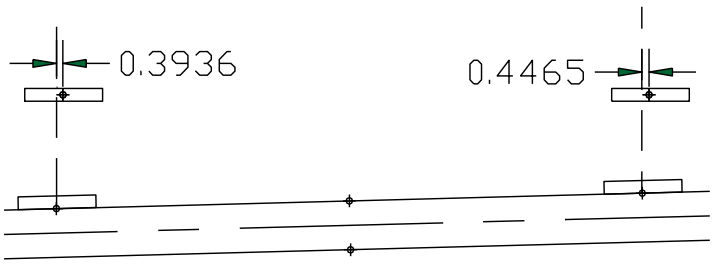


Figure 10-43

Live Axle Motion Ratios in Roll - The amount each spring is compressed or extended during body roll requires a rather more complex set of calculations. These equations, developed by Wm. C. Mitchell, can be found on pages 10 -14 and 11-15 and in Chapter 15. The live axle, unlike an independent suspension system, requires two motion ratio calculations: the side view as described below and the rear view as described in Chapter 15.

Live Axle - Springs Mounted Ahead or Behind the Axle Housing - If the springs are mounted in front of or behind the axle housing and are equal distances from the center line of the spring to the center of the tire and from center line to center line of each other on the y axis, the motion ratio is worked out the same as an independent suspension except from the side view rather than the front view.

If the spring is in front of the axle housing, the motion ratio will be less than 1 because A is smaller than

B - Figure 10-44. If the spring is behind the axle housing, the motion ratio will be greater than 1 because A is greater than B - Figure 10-45. As in the case of the independent suspension system, there are many more factors involved than simply dividing A by B to find the actual motion ratio, and, as always, the motion ratio will change slightly for each motion iteration. From the rear view, if the springs are not an equal distance from their center lines to the center of the tires and/or not an equal distance from each other, many more asymmetric factors are introduced in order to calculate an accurate motion ratio.

As in the case of the independent suspension system, there are only two ways to find the correct motion ratio per motion iteration and those are by either measuring the spring versus wheel travel through a series of ride iterations and/or use of a software program such as WinGeo3. In fact, both methods should be used to verify accuracy.

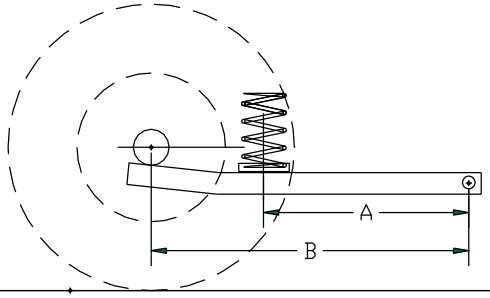


Figure 10-44

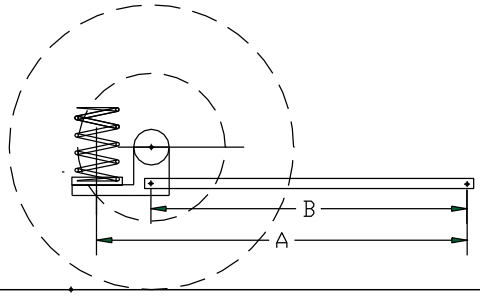


Figure 10-45

Table 10-20 illustrates motion ratios as the result of a 1" ride change for a typical three link live axle system. Because the springs are placed behind the centerline of the axle, the motion ratio is greater than 1 and therefore the wheel rate is greater than the spring rate. Also notice that in this example, the motion ratio is constant throughout the 1" of ride.

Table 10-20

Motion Ratio Measurement - Wheel Versus Spring Motion **Ride**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441- R Live Axle Three Link Spring behind axle housing 5"

Track - Mudilville Test # 1827 -dd Top of spring in 1" towards chassis center line

Software - WinGeo3 Spring Rate 300

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	1.079	0	0.00	0.00
13.4	0.1	0.1	7.954	0.108	0.108	1.079	0.000	32.40	34.96
13.3	0.1	0.2	7.846	0.108	0.216	1.079	0.000	64.80	69.92
13.2	0.1	0.3	7.738	0.108	0.324	1.079	0.000	97.20	104.88
13.1	0.1	0.4	7.63	0.108	0.432	1.079	0.000	129.60	139.84
13	0.1	0.5	7.522	0.108	0.540	1.079	0.000	162.00	174.80
12.9	0.1	0.6	7.415	0.107	0.647	1.079	0.000	194.10	209.43
12.8	0.1	0.7	7.307	0.108	0.755	1.079	0.000	226.50	244.39
12.7	0.1	0.8	7.199	0.108	0.863	1.079	0.000	258.90	279.35
12.6	0.1	0.9	7.091	0.108	0.971	1.079	0.000	291.30	314.31
12.5	0.1	1	6.983	0.108	1.079	1.079	0.000	323.70	349.27
					1" Sum	11.869	0		

Table 10-21 shows the same suspension system but with the springs mounted ahead of the axle housing. In this case the motion ratio is less than 1, and, as a result, the wheel rate is less than the spring rate. Also notice this is a falling rate system i.e. the motion ratio decreases as the ride motion increases.

Table 10-21

Motion Ratio Measurement - Wheel Versus Spring Motion

Spring Rate Versus Wheel Rate

Stock Car # 23b-441- R — Live Axle — Three Link — Spring ahead of axle housing 5"
Track - Mudilville — Test # 1828 ee — Top of spring in 1" towards chassis center line

Software - WinGeo3 — Spring Rate 300

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate
13.5	0	0	8.062	0.000	0.000	0.913	0	0.00
13.4	0.1	0.1	7.971	0.091	0.091	0.912	-0.001	27.30
13.3	0.1	0.2	7.88	0.091	0.182	0.911	-0.001	54.60
13.2	0.1	0.3	7.788	0.092	0.274	0.910	-0.001	82.20
13.1	0.1	0.4	7.697	0.091	0.365	0.910	0.000	109.50
13	0.1	0.5	7.607	0.09	0.455	0.909	-0.001	136.50
12.9	0.1	0.6	7.516	0.091	0.546	0.908	-0.001	163.80
12.8	0.1	0.7	7.425	0.091	0.637	0.907	-0.001	191.10
12.7	0.1	0.8	7.334	0.091	0.728	0.906	-0.001	218.40
12.6	0.1	0.9	7.244	0.090	0.818	0.905	-0.001	245.40
12.5	0.1	1	7.153	0.091	0.909	0.904	-0.001	272.70
					1" Sum	9.995	-0.009	

Table 10-22 is the same as Table 10-21 but the top of the spring on a coil over system is moved in towards the centerline of the chassis 4" (5" from vertical). The motion ratio is now less than 1 even though the springs are mounted 5" behind the centerline of the axle. In addition, this is a falling rate system as opposed to the constant system seen in Table 10-20.

Table 10-22

Motion Ratio Measurement - Wheel Versus Spring Motion — **Ride**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441- R — Live Axle — Three Link — Spring behind axle housing 5"
Track - Mudilville — Test # 1827 -dd — Top of spring in 5" towards chassis center line

Software - WinGeo3 — Spring Rate 300

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.986	0	0.00	0.00
13.4	0.1	0.1	7.964	0.098	0.098	0.983	-0.003	29.40	28.90
13.3	0.1	0.2	7.865	0.099	0.197	0.981	-0.002	59.10	57.98
13.2	0.1	0.3	7.767	0.098	0.295	0.978	-0.003	88.50	86.55
13.1	0.1	0.4	7.67	0.097	0.392	0.975	-0.003	117.60	114.66
13	0.1	0.5	7.572	0.098	0.490	0.972	-0.003	147.00	142.88
12.9	0.1	0.6	7.475	0.097	0.587	0.969	-0.003	176.10	170.64
12.8	0.1	0.7	7.378	0.097	0.684	0.966	-0.003	205.20	198.22
12.7	0.1	0.8	7.282	0.096	0.780	0.962	-0.004	234.00	225.11
12.6	0.1	0.9	7.186	0.096	0.876	0.959	-0.003	262.80	252.03
12.5	0.1	1	7.09	0.096	0.972	0.955	-0.004	291.60	278.48
					1" Sum	10.686	-0.031		

Figure 10-46 illustrates the spring placement used with a truck arm system. Notice the various misalignments between the upper and lower spring perches. Figure 10-47 shows the upper spring perch and load bolt used on the rear truck arm suspension.



Figure 10-46



Figure 10-47

Table 10-23 shows a typical truck arm live axle rear suspension system. Because the spring is mounted much further forward (12") from the center line of the axle than in the previous examples, the motion ratio is quite small, which, in turn, results in the wheel rate being much lower than the spring rate. For all intents and purposes, this is a constant ratio system. For a chassis of comparable mass, a truck arm suspension will require a higher spring rate to achieve equivalent wheel rates when compared to the three link live axle system.

Table 10-23

Motion Ratio Measurement - Wheel Versus Spring Motion **Ride**

Spring Rate Versus Wheel Rate

Stock Car # 23b-441- R — Live Axle Truck Arm — Spring center line is 12" from axle center line

Track - Mudilville — Test # 1827 -dd

Software - WinGeo3 — Spring Rate 300

RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.815	0	0.00	0.00
13.4	0.1	0.1	7.98	0.082	0.082	0.815	0.000	24.60	20.05
13.3	0.1	0.2	7.899	0.081	0.163	0.816	0.001	48.90	39.90
13.2	0.1	0.3	7.817	0.082	0.245	0.816	0.000	73.50	59.98
13.1	0.1	0.4	7.736	0.081	0.326	0.816	0.000	97.80	79.80
13	0.1	0.5	7.654	0.082	0.408	0.816	0.000	122.40	99.88
12.9	0.1	0.6	7.573	0.081	0.489	0.816	0.000	146.70	119.71
12.8	0.1	0.7	7.491	0.082	0.571	0.816	0.000	171.30	139.78
12.7	0.1	0.8	7.409	0.082	0.653	0.816	0.000	195.90	159.85
12.6	0.1	0.9	7.328	0.081	0.734	0.816	0.000	220.20	179.68
12.5	0.1	1	7.246	0.082	0.816	0.816	0.000	244.80	199.76
					1" Sum	8.974	0.001		

Table 10-24 illustrates the same system except the springs are moved 1” further forward.

Table 10-24

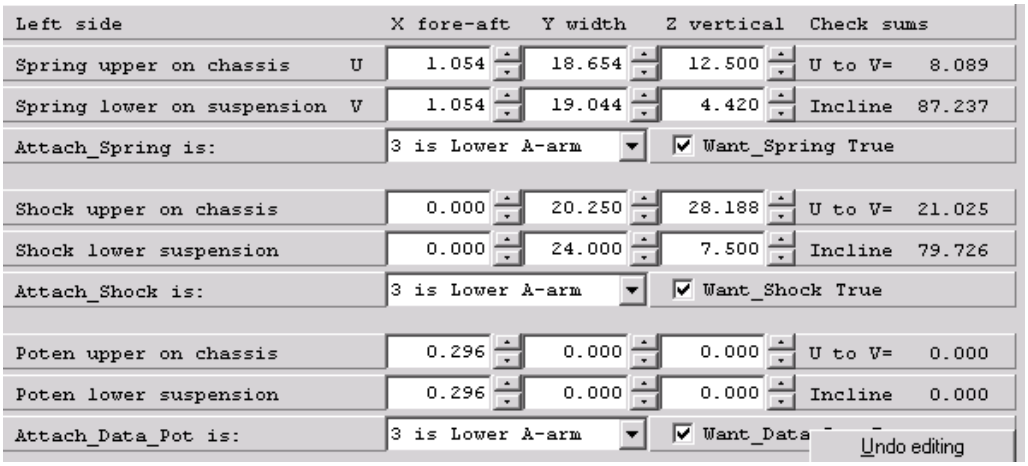
Motion Ratio Measurement - Wheel Versus Spring Motion									**Ride**
Spring Rate Versus Wheel Rate									
Stock Car # 23b-441- R		Live Axle Truck Arm			Spring center line is 13" from axle center line				
Track - Mudilville		Test # 1827 -dd							
Software - WinGeo3				Spring Rate 300					
RF Hub Height	RF Ride Change	Total ride Change	RF Damper Length	RF Damper Change	Total Damper Change	RF Motion Ratio	RF - MR Change	Spring Rate	Wheel Rate
13.5	0	0	8.062	0.000	0.000	0.800	0	0.00	0.00
13.4	0.1	0.1	7.982	0.080	0.08	0.800	0.000	24.00	19.20
13.3	0.1	0.2	7.902	0.08	0.16	0.800	0.000	48.00	38.40
13.2	0.1	0.3	7.822	0.08	0.24	0.800	0.000	72.00	57.60
13.1	0.1	0.4	7.742	0.08	0.320	0.800	0.000	96.00	76.80
13	0.1	0.5	7.662	0.08	0.400	0.800	0.000	120.00	96.00
12.9	0.1	0.6	7.582	0.08	0.480	0.800	0.000	144.00	115.20
12.8	0.1	0.7	7.502	0.08	0.560	0.801	0.001	168.00	134.57
12.7	0.1	0.8	7.422	0.08	0.640	0.801	0.000	192.00	153.79
12.6	0.1	0.9	7.342	0.080	0.720	0.801	0.000	216.00	173.02
12.5	0.1	1	7.262	0.08	0.800	0.801	0.000	240.00	192.24
					1" Sum	8.804	0.001		

** Begin WinGeo3 Example(s)**

Asymmetrical Chassis

Edit: Auxiliary (Stock Car Springs and Shocks) – As in all other cases, the initial measured or design data is entered using the *Edit* menu. The *Edit* menu is **not** used to modify the original suspension data.

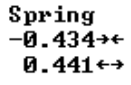
Win 10-1 shows the *Edit /Auxiliary* co-ordinates for the *Short Track* stock car springs and shocks. Notice that the bottom of the spring, shock and potentiometer are all attached to the lower control arm and that all three *Want* boxes show a check mark. Because this chassis is asymmetric, the left and right sides need to be entered separately.



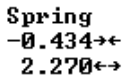
Win 10-1

Utility: Motion Ratio - The spring and shock motion ratio can be displayed as either *Motion Ratio* or *Velocity Ratio*. The *Motion Ratio* is spring displacement / wheel travel and is usually less than 1.0. *Velocity Ratio* is wheel travel / spring displacement and is usually greater than 1.0. Motion ratio is the demo version preset. The shock ratios are calculated in the same manner. To **view** the *Motion/Velocity* ratios, make sure that the derivative box found in the *Geometry Control Dialog / Values* menu has a check mark in it.

Win 10-2 shows the amount of spring compression (indicated by the two inward pointing arrows) and the spring motion ratio. Win 10-3 shows the same amount of spring motion but with the velocity ratio shown as the derivative.

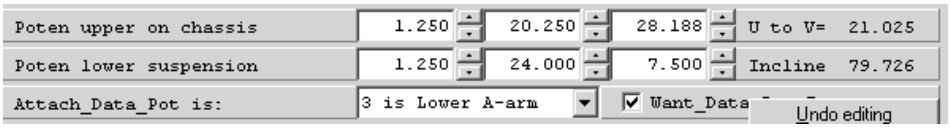


Win 10-2

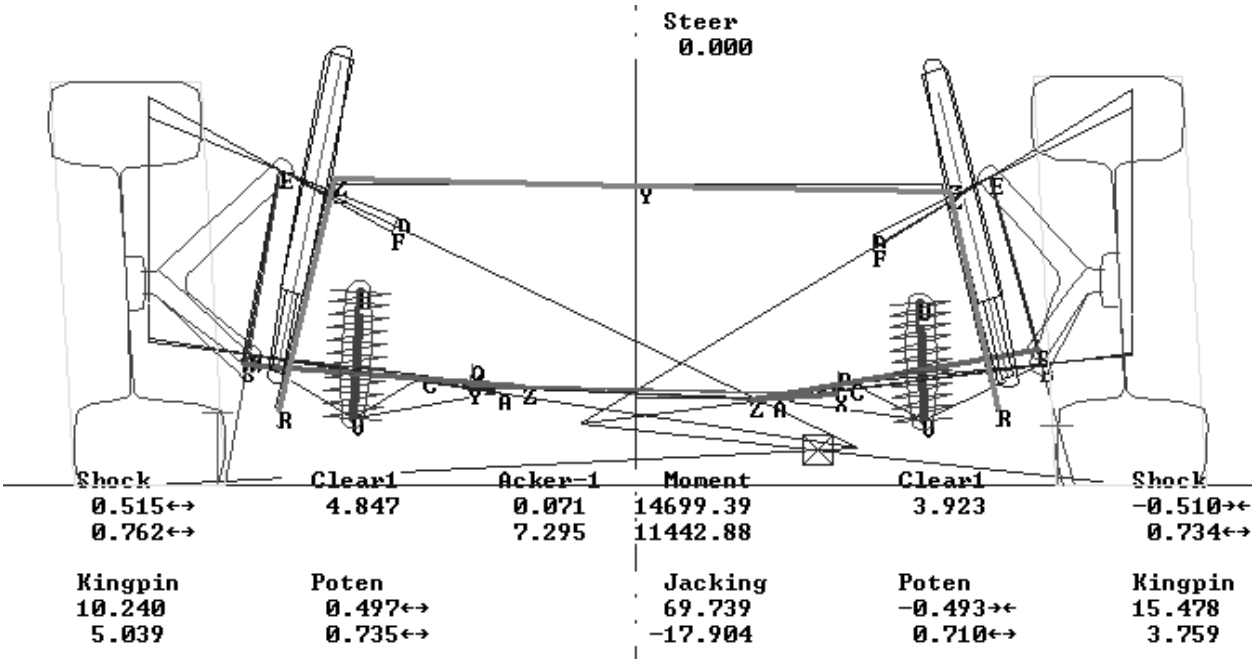


Win 10-3

Data Pot: - A linear potentiometer is attached to the WinGeo3 model just as it would be on an actual chassis – Win 10-4. The user will need to enter appropriate *Edit / Auxiliary* co-ordinates.



Win 10-4

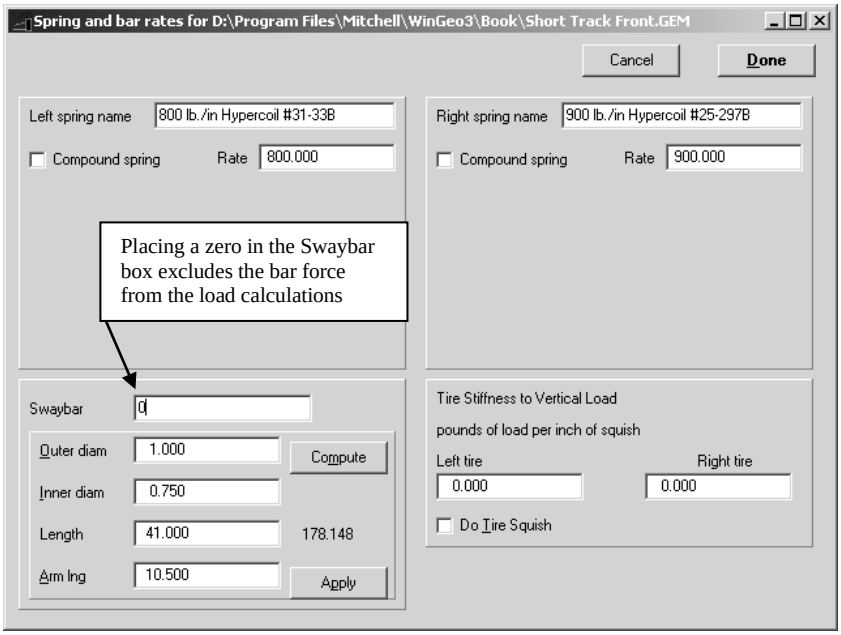


Win 10-5

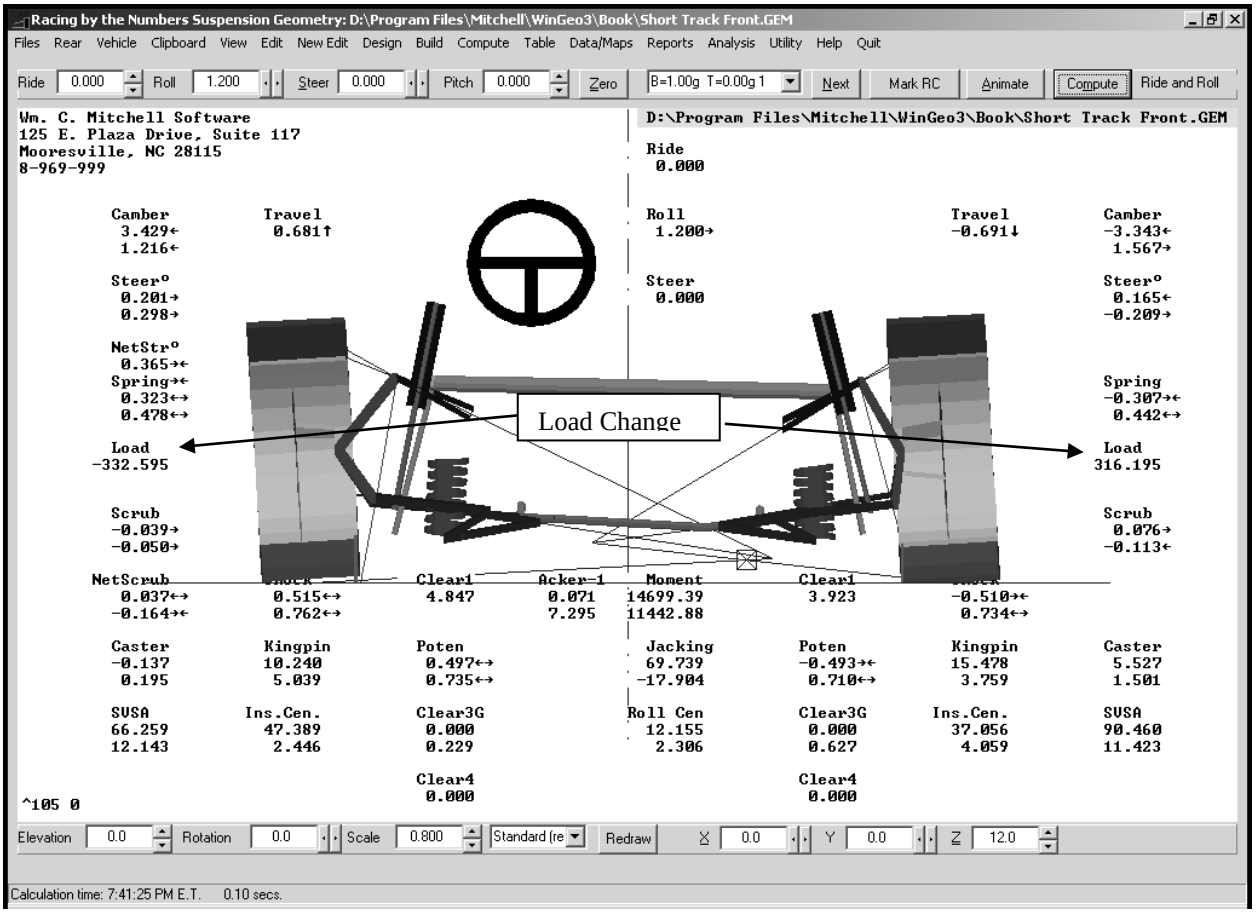
Edit: Forces (Springs and Bars) - This menu allows the user to enter spring and swaybar rates – Win 10-6. These spring and bar rates are used to calculate load **changes** at the tire contact patch. Currently, the springs are linear. In the future, compound-rate springs may be added. The user places a zero in the Swaybar box if he/she does not want the anti roll bar force to be included in the calculations.

In order to view the load changes, the *Loads* box found in the *Geometry Dialog / Values* menu must have a check mark in it. The load changes **do not** reflect shock or anti/jacking forces – Win 10-7.

In the Premier version of the program, you can also enter a spring rate for the tires and then compute suspension travel taking into account tire squish. This can have a significant effect on the results.



Win 10-6



Win 10-7

Symmetrical Chassis

Edit: Auxiliary (Springs/Shocks) – On a coilover suspension system, the springs and shocks are entered together in the spring co-ordinate boxes. Because this chassis is symmetrical, *Both Sides* are entered at the same time – Win 10-8. The *Want Shock* box has no check mark. In the case of a bell crank/Pivot suspension system, the *Spring Lower on Suspension* attachment is at the bell crank – see *Attach Spring is (4 is Pivot Bell-Crank*).

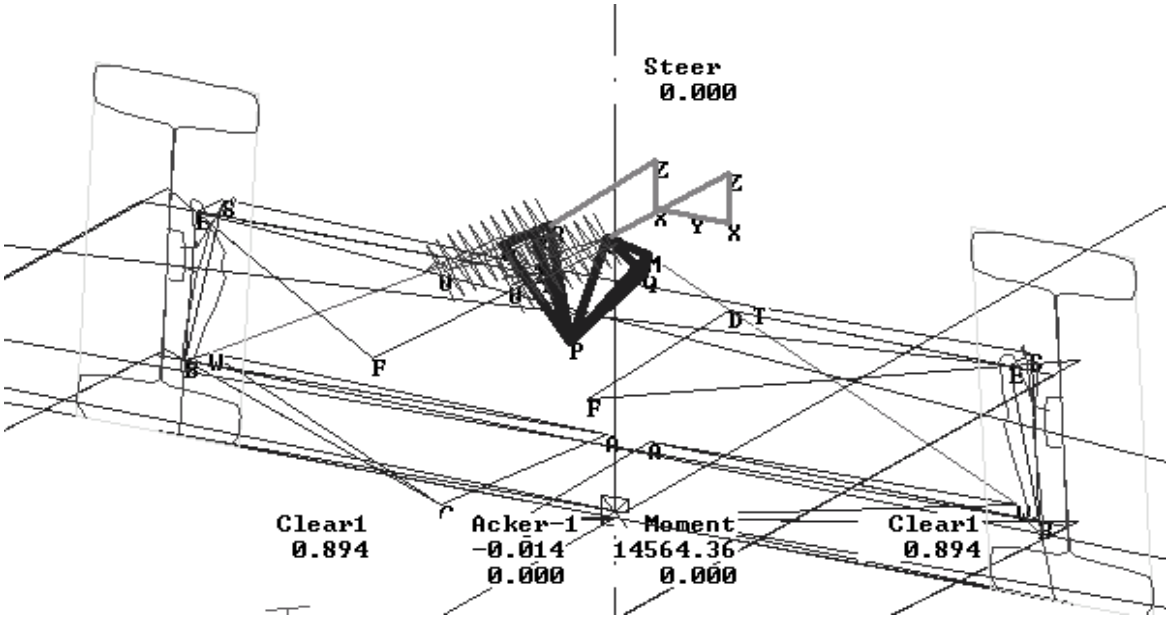
Both sides		X fore-aft	Y width	Z vertical	Check sums
Spring upper on chassis	U	17.894	2.625	22.618	U to V= 13.209
Spring lower on suspension	V	4.714	2.196	21.864	Incline 119.638
Attach_Spring is:		4 is Pivot Bell-cra		☑ Want_Spring True	
Shock upper on chassis		0.000	0.000	0.000	U to V= 0.000
Shock lower suspension		0.000	0.000	0.000	Incline 0.000
Attach_Shock is:		4 is Pivot Bell-cra		☐ Want_Shock False	

Win 10-8

Edit: Pivot – This menu is used to enter the upper and lower *Push/Pull Rod Attachments* and the bell crank pivot axis – Win 10-9. **Accurately** establishing the upper and lower points of the pivot axis is extremely important. Many designers and race engineers extend the bell crank/pivot axis down to a point where the left and right side axis intersect as shown in Win 10-10. Refer to *Edit / Axis Calculator* to help extend the bell crank/pivot axis. The Premier version of WinGeo3 has a bell crank analysis menu under *Analysis / Bell Crank* that is very useful when designing or redesigning a bell crank.

Both sides		X fore-aft	Y width	Z vertical	Check sums
Pivot axis one	P	5.622	0.000	14.328	P to Q= 7.788
Pivot axis two	Q	5.317	5.288	20.038	Incline -47.150
Push/Pull rod pivot	M	3.110	4.285	20.531	M to W= 29.434
Push/Pull rod attach	W	0.066	29.877	6.313	Incline 28.884
Attach_Push_Pull is:		3 is Lower A-arm		☑ Want_Shock_Pivot True	

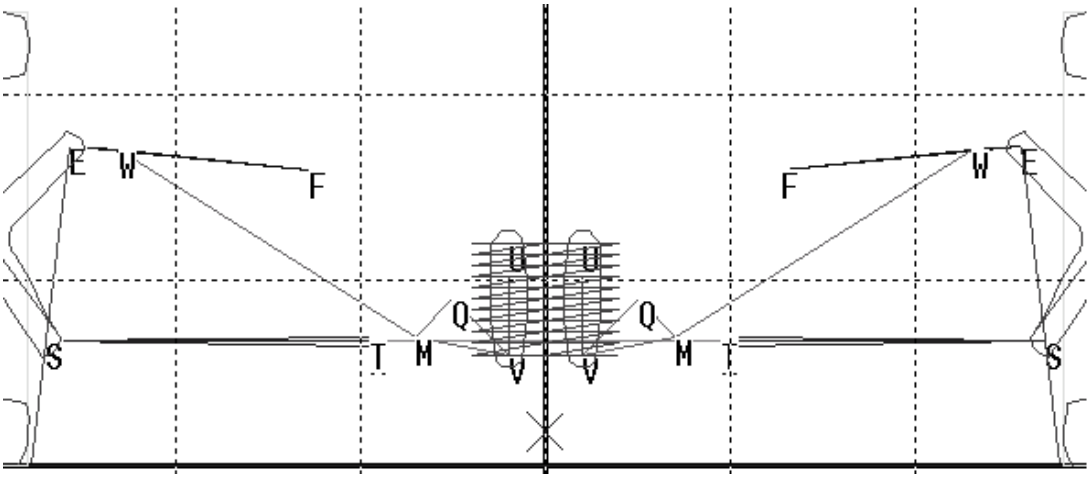
Win 10-9



Win 10-10

Edit: Pivot – WinGeo3 Help Files

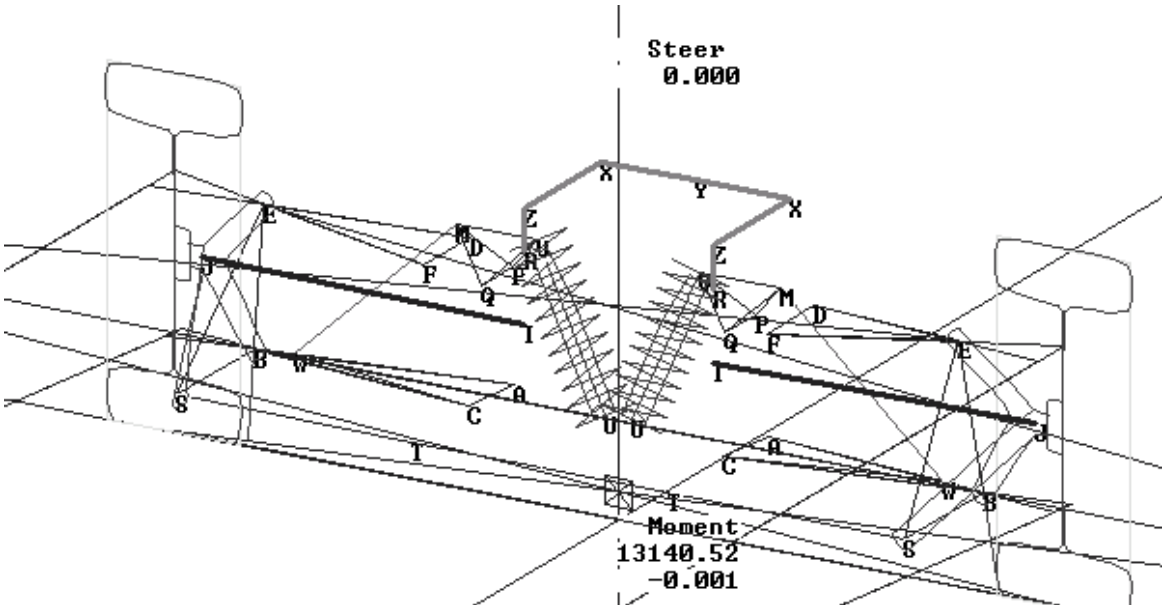
The bell crank pivots about the axis P-Q. The pull rod attaches to the bell crank at point M. The other end of the pull rod, point W is attached to the upper a-arm. The spring attaches to the bell crank at point V. Point P is behind point Q in the graphic below.



Win 10-11

As the lower A-arm moves the bottom of the push rod (point W) activates the push rod. The push rod is attached to the *Bell Crank Pivot* at point M. The *Bell Crank* rotates about the P-Q axis and moves the swaybar (point R) and the spring (point V). Point V is not clear but it is the third corner of the pivot. The spring compresses or extends along the axis U-V.

Care should be taken that the *Bell Crank* pivot rotates freely about the P-Q axis. The *Push Rod* W-M and the spring U-V should both be perpendicular to the P-Q axis.



Win10-12

** End WinGeo3 Example(s)**