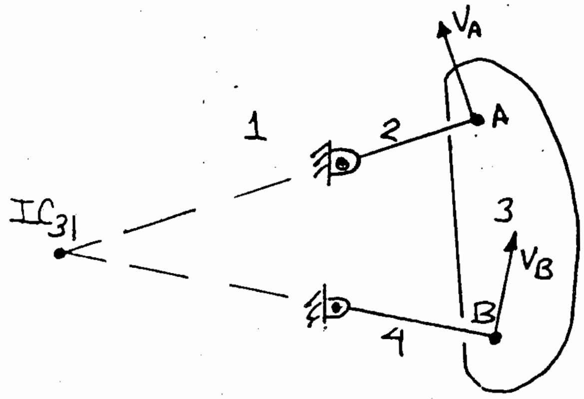
Handout 3

# Roll Centers & Suspension Geometry

When a vehicle with suspension is in a turn, it usually "leans", or in dynamics terms, the sprung mass rotates about some front-rear axis. That axis is called the "roll-axis" and is not usually a fixed line in space, but actually moves about as the vehicle rolls. The roll axis connects two points in the planes of the front and rear suspension "axles". Those points are called "roll centers", and are the "instant centers" (in kinematics terms) of the sprung mass with respect to the ground at the lateral planes through the front and rear axle lines.

Roll centers are important since they:
(1) affect the <u>scrub</u> or lateral (side-to-side) motion of the tire contact patch
(2) affect the <u>camber change</u> as the wheels move up and down,
(3) affect the <u>lean angle</u> that the sprung mass attains in a turn, and
(4) affect the <u>manner</u> in which <u>weight is transferred</u> laterally ("from inside to outside" during a turning manouver,

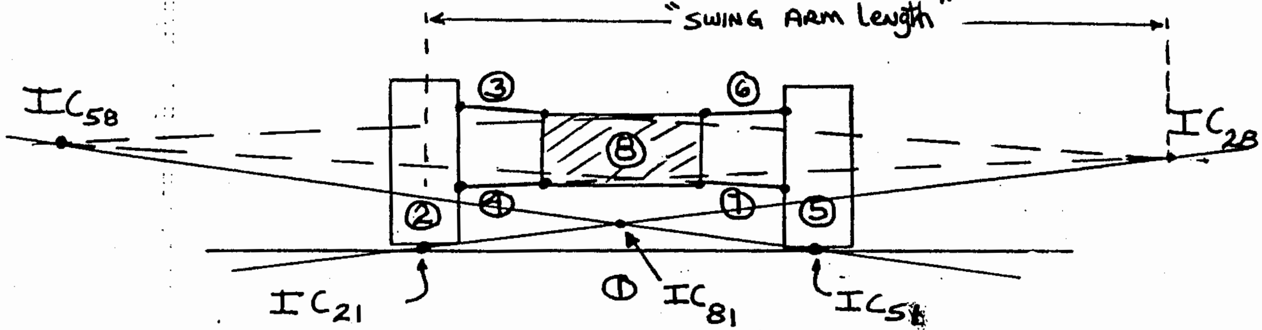
The following describes how to find the roll center of a double A-arm suspension using ideas from ME 3-222, where you will examine movement of a planar 4-BAR LINKAGE:




Links 2,3,4 are rigid bodies and fastened together with friction-less pins. Links 2 and 4 are also pinned to ground (Link 1)

If the linkage is moving, the direction of the velocities at A and B must be perpendicular to links 2 & 4 respectively as shown. Since points A & B are also on Link 3, we could extend the lines of links 2 & 4 as shown and they intersect at point $IC_{31}$. This is the "instant center" of Body (Link) 3 with respect to ground. At this position, link 3 appears to be rotating around $IC_{31}$ when seen from the ground.

How about a DOUBLE A-ARM suspension ON A VEHICLE? This system has 8 links.




WE WANT TO KNOW how the SPRUNG MASS, LINK 8, ROTATES with RESPECT to GROUND, OR $IC_{81}$.

WE'll use KENEDY'S Theorem —

FIRST WE must say how the TIRES "ROTATE" with Respect to ground in this view. WE know we are doing small motions, so we'll assume that the tire ROTATES back & forth about the CENTERS of their contact patches, OR $IC_{21}$ and $IC_{51}$.

Then the Instant Center of EACH TIRE w.r.t. the SPRUNG MASS can be identified as $IC_{58}$ and $IC_{28}$, using the 4-bar linkage procedure.

KENEDY'S Theorem states that $IC_{81}$ will be on the lines $IC_{58} \leftrightarrow IC_{51}$ and $IC_{28} \leftrightarrow IC_{21}$.

(COMMON LINKS "CANCELL")

$IC_{81}$ is the ROLL CENTER of the SPRUNG MASS w.r.t. ground

The "swing arm length" is shown above for Tire ②. It is a simple way of characterizing the

KINEMATIC PROPERTIES of a suspension.

The following pages are taken from:
TUNE to WIN, Carroll Smith, Chpt 4
Competition Car Suspension, ALAN STANIFORTH

They are written for an audience that is interested in RACE CARS — not solar cars. The main intent for RACE CARS is to try to have the loaded wheels (the "outside" wheels in a turn, the rear wheels when accelerating and the front wheels when braking) nearly perpendicular to the ground when viewed from the front, in order to maximize CORNERING power and traction. I am including them here in order to provide a discussion of wheel motion with a double-a-arm suspension, and the various conflicts that are present in suspension design.

The design choices are the lengths of the A-ARMS and the locations of their pivots. These authors describe various options and summarize some "Guidelines" for Design of Race cars. A separate handout will discuss the needs of a solarcar front suspension.

Smith also describes a "paper doll" exercise and provides templates to build a cardboard suspension model to study its properties, and Staniforth refers to a similar exercise with his "string computer". These may seem out of date with CAD capabilities, but they provide a good hands-on feel for the motions of the Double A-arm suspension.

The "swing arm length", shown on the figure two pages earlier, is also referred to in these references —

Smith does not define it, but on his "page 54" uses the terms:

"instantaneous swing arm length" and
"effective swing arm length"

Staniforth uses the term

"swing axle length"

and categorizes suspension properties according to its ranges of values.

36

From Tune to Win,

Carroll Smith

## THE OBJECTIVE OF THE SUSPENSION SYSTEM

So much for history. Now let's see just what we want the wheel suspension system to accomplish. First of all, we must have four-wheel independence, so that as far as possible, upsets will be confined to the wheel and tire which experiences the upset. We are certain of this much and, within reasonable limits, any independent system will give it to us. Second, although we must provide enough vertical wheel movement so that the wheels and tires can absorb road surface bumps and vertical accelerations of the sprung mass, we want there to be no change in toe-in—or at least adjustable change in toe-in—while the wheels are moving. With attention to detail, this is not a problem. Third, we want no compliance within the suspension system or its attachment to the sprung mass. This is a question of the stiffness (rather than the strength) of the links and the rigidity of the pivots, axles, hub carriers, and attachment points as well as the direction in which the loads are fed into the chassis and the base over which the loads are spread. The four bar link system lends itself admirably to this goal—more so than any other arrangement. Attention to detail design is required and many designers are deficient in this respect, but the system itself is not. All links can be arranged so that they are loaded in straight tension or compression with no bending moments imposed and link stiffness is merely a question of calculating compression loads. Feeding the loads into the chassis properly requires a bit more thought, but it is not that difficult.

Next we require minimum weight—and again the system is ideally configured to achieve it. Further, the wide base over which we can feed the loads into the chassis obviates the necessity for massive and heavy attach structure.

This much is easy. Next we want to control change of wheel camber angle and change of track dimension with wheel and/or sprung mass movement. There are two separate problems here. In order to achieve the maximum footprint area and an even pressure pattern so that we can realize maximum tire tractive effort under braking and acceleration, we wish the wheel to remain upright when the suspension is subjected to the vertical movement of the sprung mass caused by longitudinal load transfer. We also want it to remain upright when the wheel itself is displaced vertically by a bump or a dip—although this is a more transient condition and less important in the overall scheme of things. At the same time, and for the same reasons, we want both the inboard and outboard wheels to remain vertical *to the track surface* as the sprung mass rolls due to centrifugal acceleration. We also do not want the track dimension *at the contact patch* to change under any of these conditions as that would cause the tire to be scrubbed sideways across the race track when it is already at or near its limit of adhesion and would upset things in the traction department. While all of this is going on it would be nice if the roll centers at each end of the car were to remain a constant distance away from their respective centers of mass so that we could retain our linear rate of roll generation and lateral load transfer.

Sounds simple enough—but it is just not possible to achieve. While we have infinite permutations available with combinations of link lengths and inclinations, none of the combinations will achieve all of the above.

## THE NATURE OF WHEEL MOVEMENT

Let's look at what actually happens with wheel or chassis movement. There are two separate types of movement—vertical movement of either the wheels or the chassis and the movement of the chassis in roll. First we'll look at Figure (19) while I explain what we will be looking at in the suspension diagrams from now on. The right side of Figure (19) shows what the rear suspension of a typical Formula 5000 or Can Am Car might look like when viewed from the rear. The left side shows how we are going to represent the linkages of that system in our discussions. This representation has the double advantage of making the pertinent points easier to see and the drawings easier to make.

We'll consider vertical movement first. It doesn't matter, from the geometric point of view, whether the wheel moves because of a bump or a dip in the road or whether the chassis moves in response to a load transfer or to a change in aerodynamic downforce. If the wheel moves, it takes the outboard pivot points of the suspension links with it which forces the links to describe arcs about their inboard pivots. The wheel must then change its angular position relative to both the road surface and to the chassis as a function of those arcs. If the chassis moves, the inboard pivot points move with it and the same thing happens. The geometric results will be the same. Since movement of the chassis in response to load transfers is of more interest to us than transient wheel movement in response to bumps, all of the illustrations will show this case. Figure (20) shows the effect of bump and droop movement on wheel camber, spring axis length, drive shaft length and roll center location. There are no surprises here except for the fact that, due to camber angle, track change at the center of the footprint is not equal to the change in length of the half shaft.

When the sprung mass rolls, however, as in Figure (21) the whole picture changes. In this case the inboard link pivots move with the chassis which must roll about the instantaneous roll center of the suspension. This means that, on the laden side (side away from the center of the turn) of the chassis, both the upper and lower pivot points will move downward and out from the chassis centerline. The upper

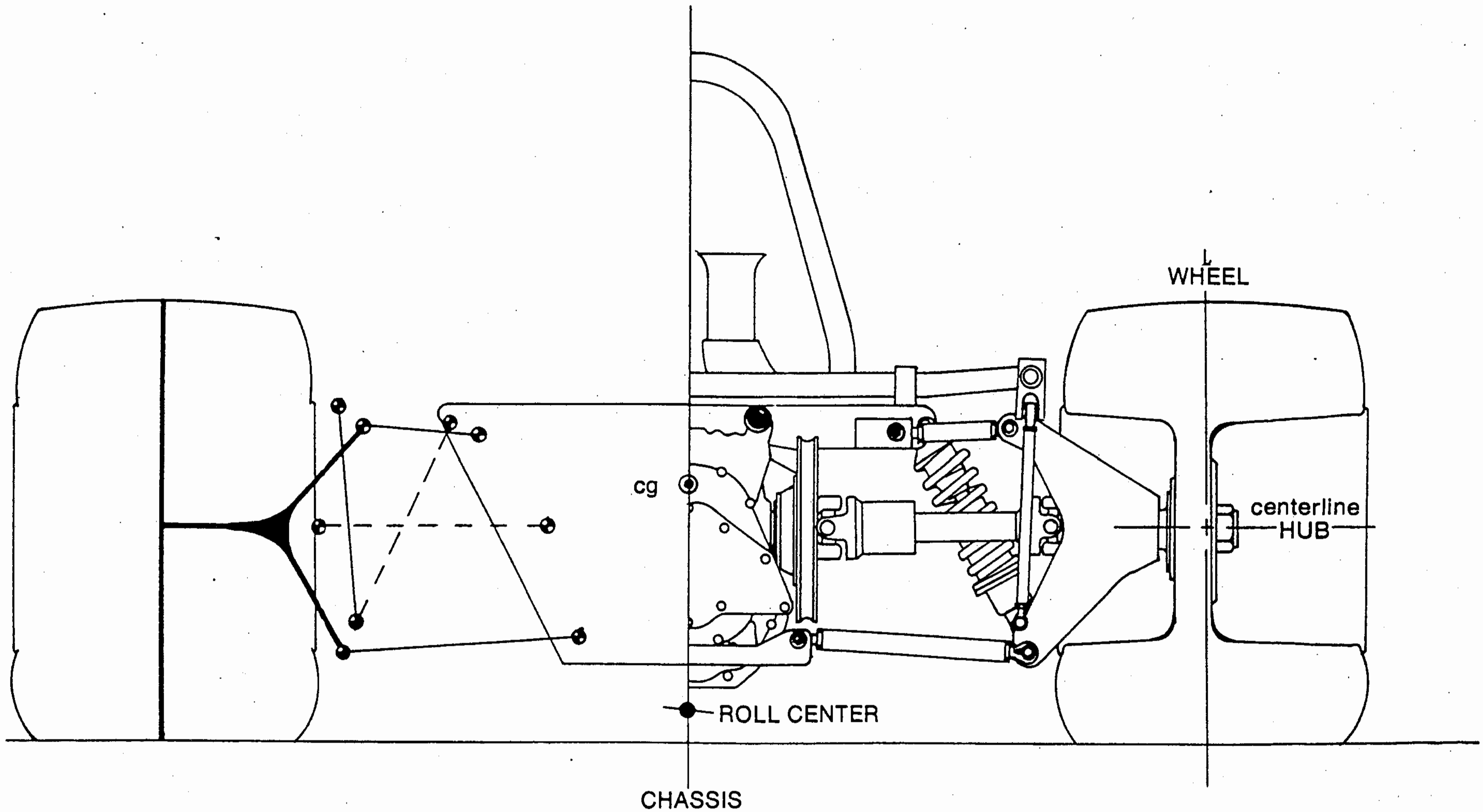



*Figure (19): Explanation of diagrams to be used to illustrate suspension geometry—right side shows rear suspension in end view—left side shows links, link pivots centerlines and instantaneous centers only:*



37

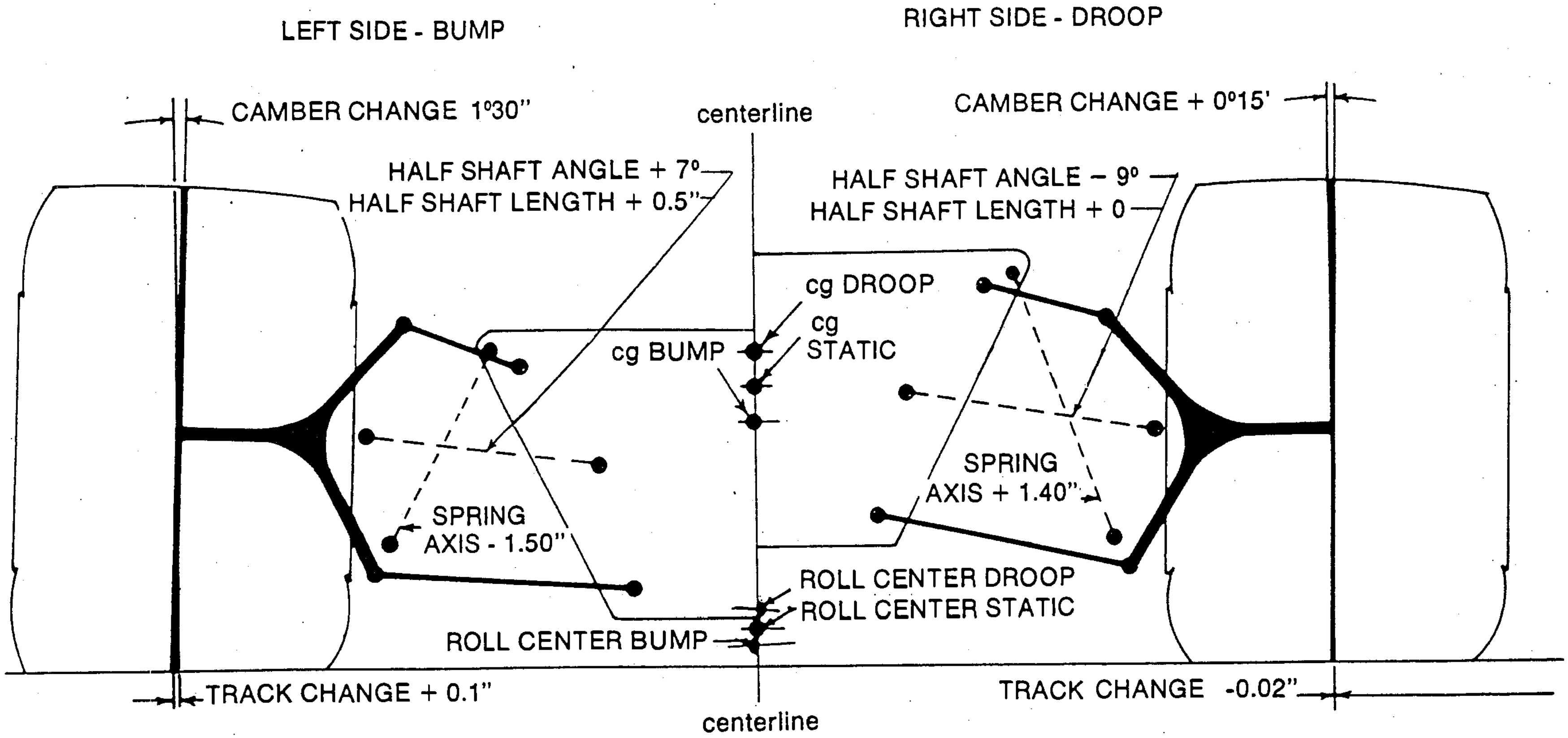



*Figure (20): Effect of vertical chassis movement on wheel camber, spring axis length, drive shaft length and roll center location.*



(38)

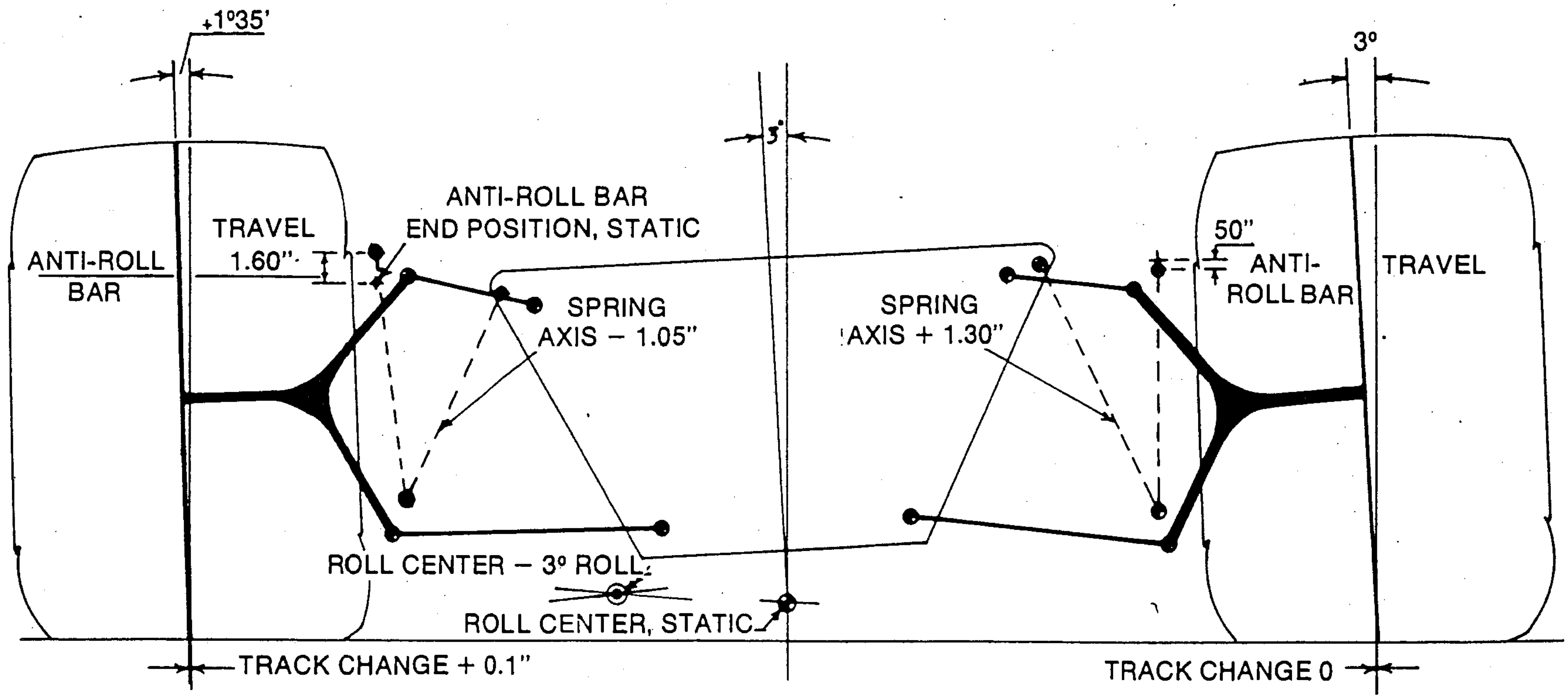



Figure (21): Effects of chassis roll.



39

pivot point, being on a longer radius from the roll center, will, however, move further than the lower. Since the suspension links are of fixed length, this difference in pivot point movement will force the laden wheel to assume a positive camber angle (out at the top) relative to the surface of the race track. The opposite set of conditions exist on the inboard or unladen side so that tire will be pulled to a negative camber angle. This is not what most of the books tell us for the simple reason that most of the books reference wheel camber to the chassis. Unfortunately, no one tells the tire about any camber relationship except that which exists between the tire and the road surface. We could care less about the angular relationship between the wheel and the chassis.

The next shock is what happens to the location of the roll center when the chassis rolls—it moves—not only downward but also sideways. Again most books tell us that the roll center, and therefore the roll axis, remains on the vehicle centerline. It doesn't—not when the vehicle rolls. The roll center of a vehicle in a roll conditon is the intersection of the line drawn between the instantaneous center of the laden wheel and the center of its contact patch with the similar line drawn between the instantaneous center and the contact patch of the unladen wheel. It is very unlikely that this intersection will ever be located on the centerline of the chassis. This is not shown in Figure (21) because I ran out of room on the paper. It is shown in Figure (22) which illustrates the effects of a combination of chassis roll and bump travel—conditions which exist at the front of the car on corner entry and at the rear on corner exit.

## PAPER DOLLS

About now, we are faced with two basic choices on how to attack the rest of the chapter. I can write and draw until I am blue in the face—and still not put a dent in the possible combinations of link lengths and angles—or we can construct a two dimensional model of the four bar link suspension system and you can play games with it. Since the choice is mine, we will construct the ¼" scale model shown in Figure (23). Somewhere in the back of the book—if I don't forget to put it in—you will find a tear-out page on which the pieces for the model are printed. Glue them onto an old manila file, cut them out, get a cheap protractor and lay out the background shown in Figure (23A). With a box of thumbtacks, a couple of straight edges and some string, you are now equipped to spend hours at a card table driving yourself nuts—and convincing anyone who happens to wander in that you have already succeeded. By punching suitably placed holes in the chassis, suspension upright and link portions of paper doll and inserting thumbtacks for pivot points you can construct a scale model of any independent suspension system that you like. Hold the tire centerpoints against a straight edge on the ground line and move the chassis up and down to observe the effects of bump and droop movement—wheel camber and track change read directly on the background. Find the roll center by extending the link pivot axes with either a straight edge or string, stick a thumbtack through the roll center, roll the chassis one degree and watch the wheels. Find the new roll center and repeat the exercise. Then combine roll and vertical chassis movement. The comparison will not be exact because we are ignoring a few factors, but it is plenty close enough to be educational—and it is going to save me writing several thousand hard to follow words. You will learn more playing with the model.

## BASIC LAYOUTS

Although there are endless possible combinations of link lengths and inclinations, we can break them down into three basic layouts, equal length and parallel links, unequal length and parallel links and unequal length, non parallel links. We will briefly examine the characteristics of each in turn.

## EQUAL LENGTH AND PARALLEL LINKS

Figure (24) shows an equal length and parallel link system with short link lengths. Because the links form a parallelogram, there will be no camber change with vertical movement. There is, however, considerable change in track width—which is not good. When the chasis rolls, the wheels and tires change camber by the exact amount of chassis roll—with the outside wheel cambering in the positive direction. This is not good under any condition and, the wider the tire involved, the less good it is. Since the links remain parallel under all conditions, the location of the instantaneous center—the intersection of the extended linkage axes—is located at an infinite distance from the chassis centerline. We assume the roll center to be at ground level and to pretty much stay there. Not on Drawing

We can reduce the amount of track change for a given amount of vertical motion by the simple expedient of lengthening the suspension links, as in Figure (25). With this change, a given amount of vertical wheel or chassis movement results in less angular displacement of the wheel and therefore in less change in the track dimension. Alas, the linkage remains a parallelogram and the roll camber situation remains basically as before, although the amount of camber change is slightly reduced because the inboard pivots are closer to the vehicle centerline and so are displaced less for a given amount of roll. Also, while we can reduce the track change by lengthening the links, we cannot eliminate it, or even get it down to reasonable dimensions—and we will not have room for infinitely long links.

## UNEQUAL AND PARALLEL LINKS

If we make the upper link relatively shorter than the lower, as in Figure (26), we achieve some significant changes in the wheel paths. Now, in vertical travel, the upper link has a shorter radius than the lower which results in the wheel assuming a negative camber angle in both bump and either negative or positive camber droop. The amount of camber change is dependent upon the relative lengths of the upper and lower links—the shorter the upper link becomes, the steeper the camber change curve. The assumption of negative camber reduces the change in track dimension considerably and, with care, it can become insignificant.

When the sprung mass rolls, the wheels are still forced into camber angles in the same direction as the chassis roll, but the positive camber assumed by the all important laden wheel is considerably reduced. Unfortunately, the negative camber of the unladen wheel is increased.

Although the links are parallel to each other at ride height, the fact that they are unequal in length means that they will not remain parallel with vertical wheel movement (they

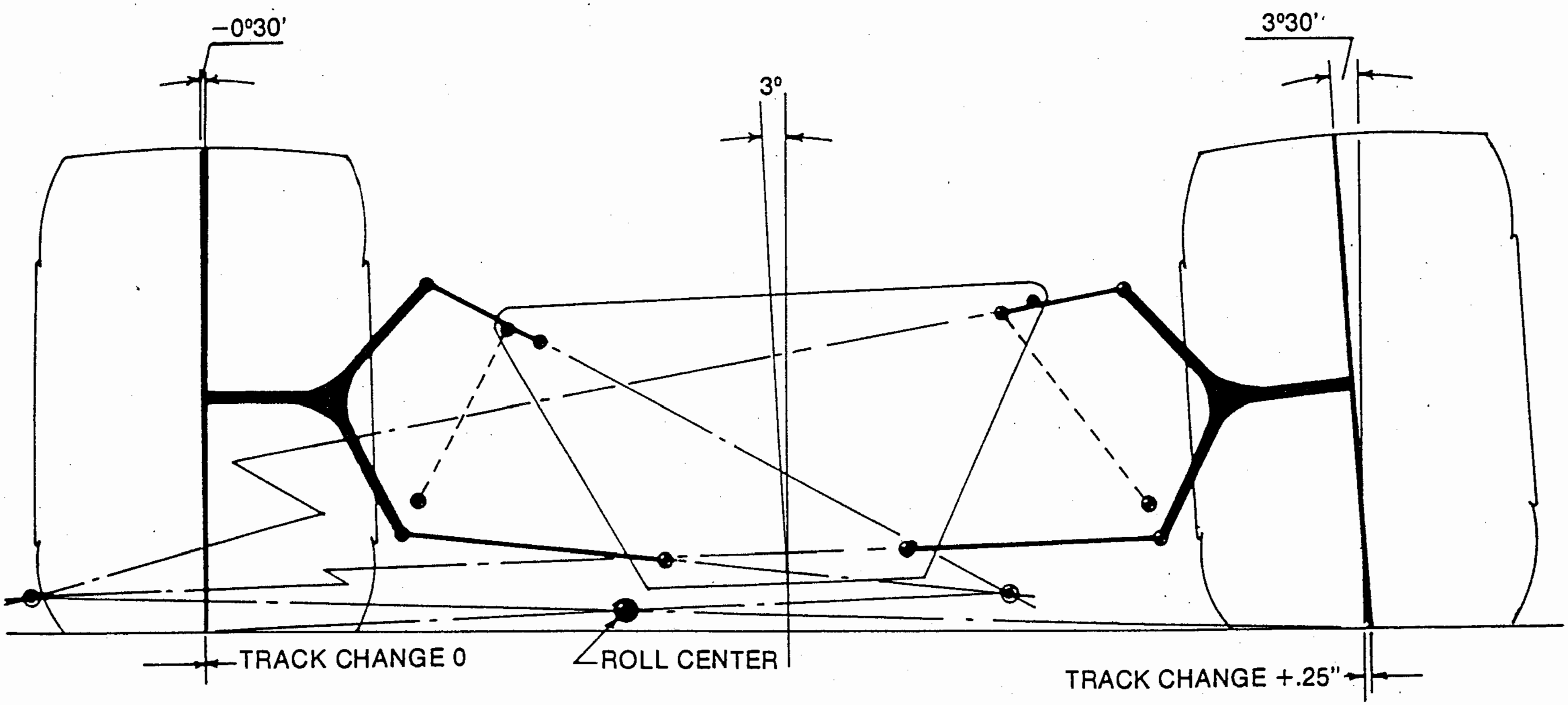



*Figure (22): Effects of combination of chassis roll and bump movement.*



41

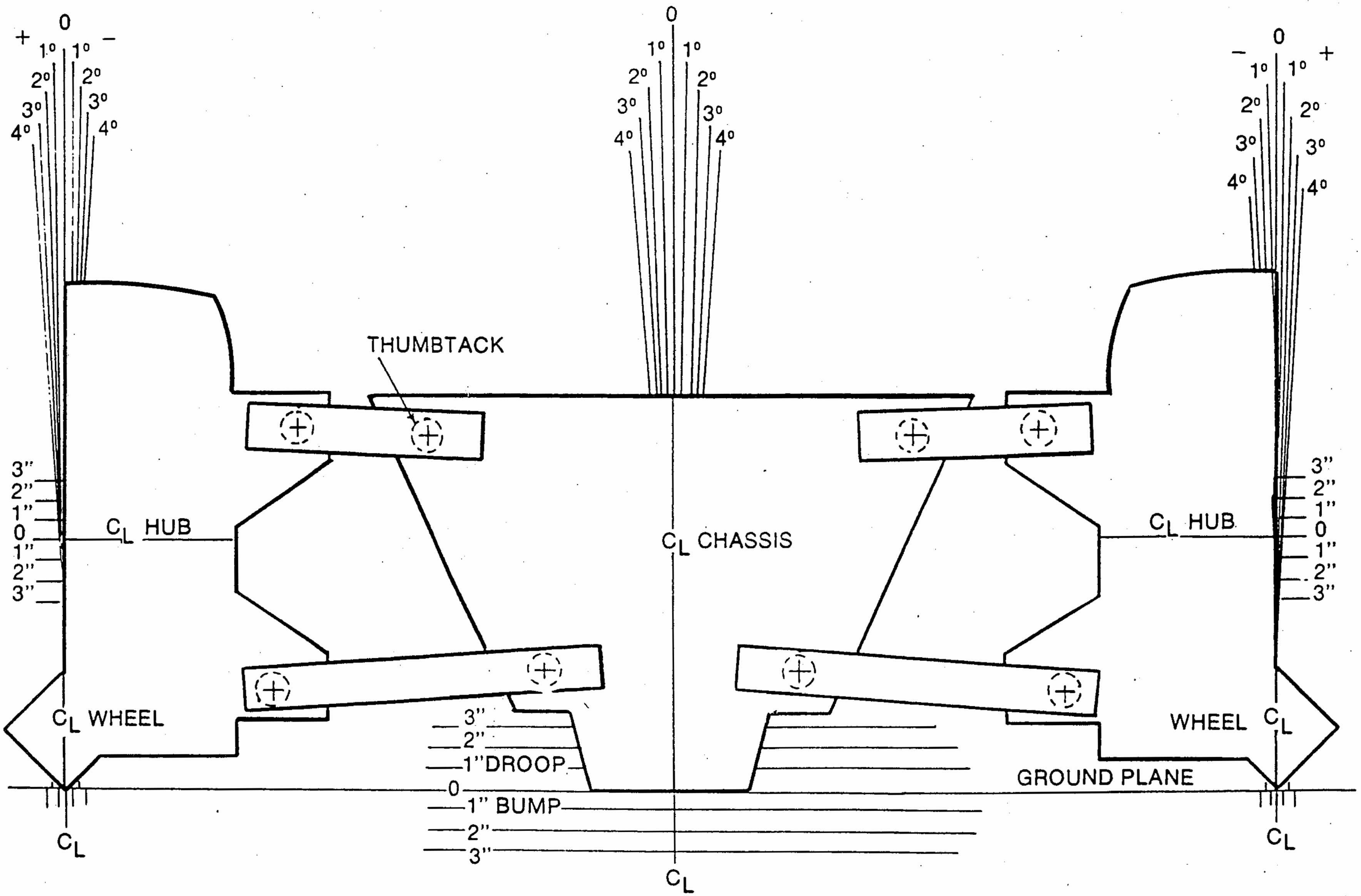



Figure (23): Scale of model of suspension linkage geometry.

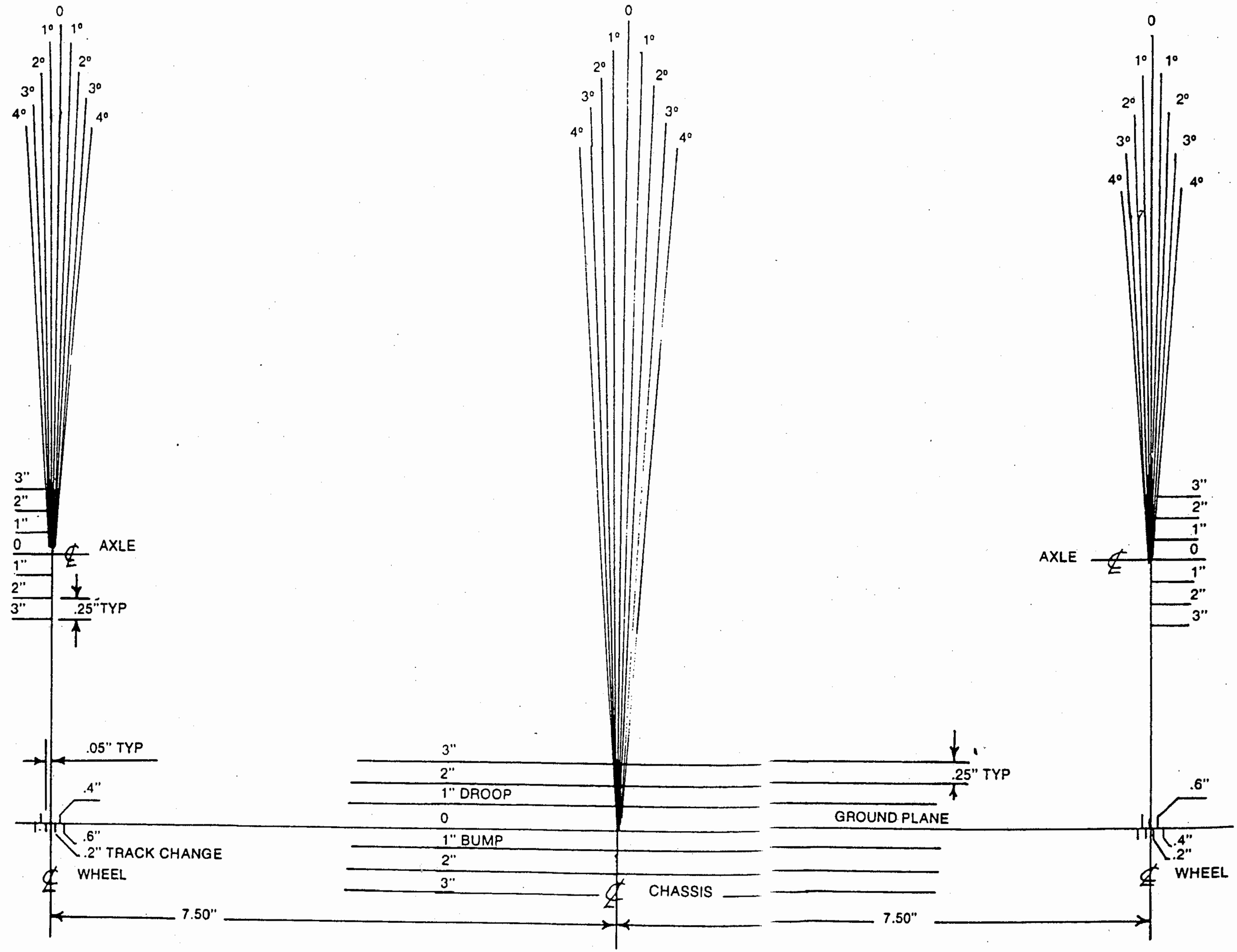



Figure (23a): Background for ¼ scale suspension geometry model

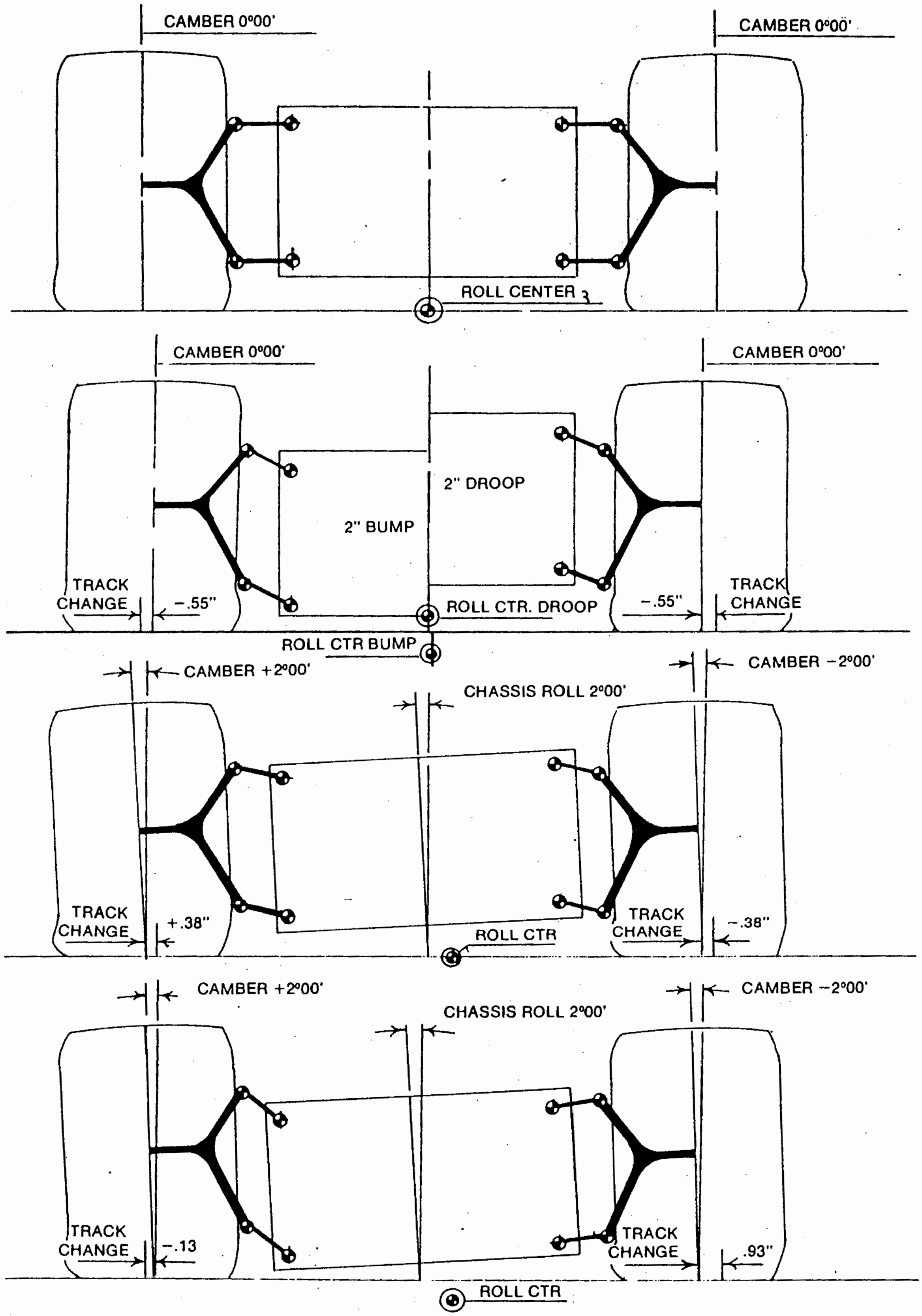



*Figure (24): Equal length and parallel link system with short links.*

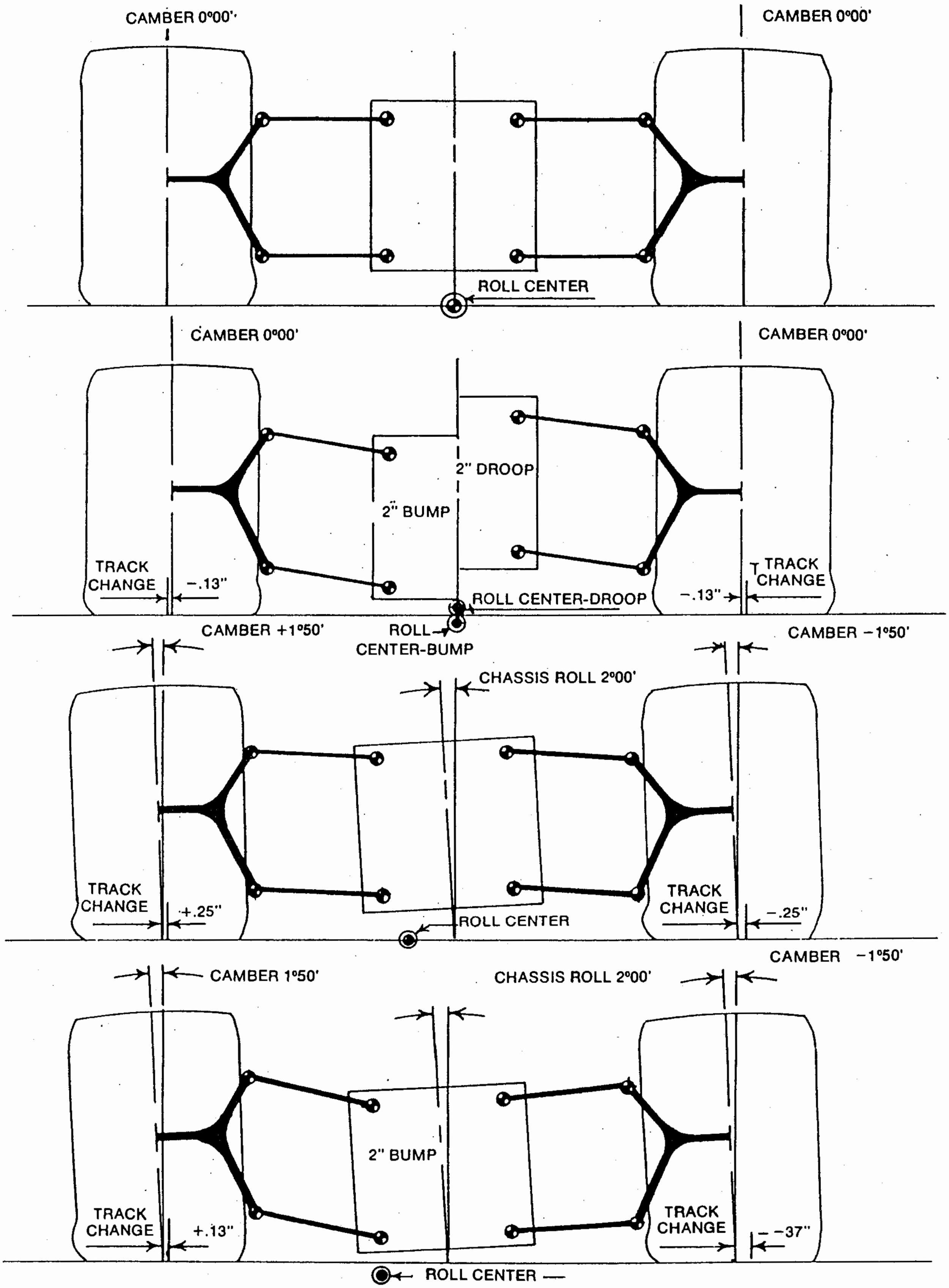



*Figure (25): Equal length and parallel link system with relatively long links.*

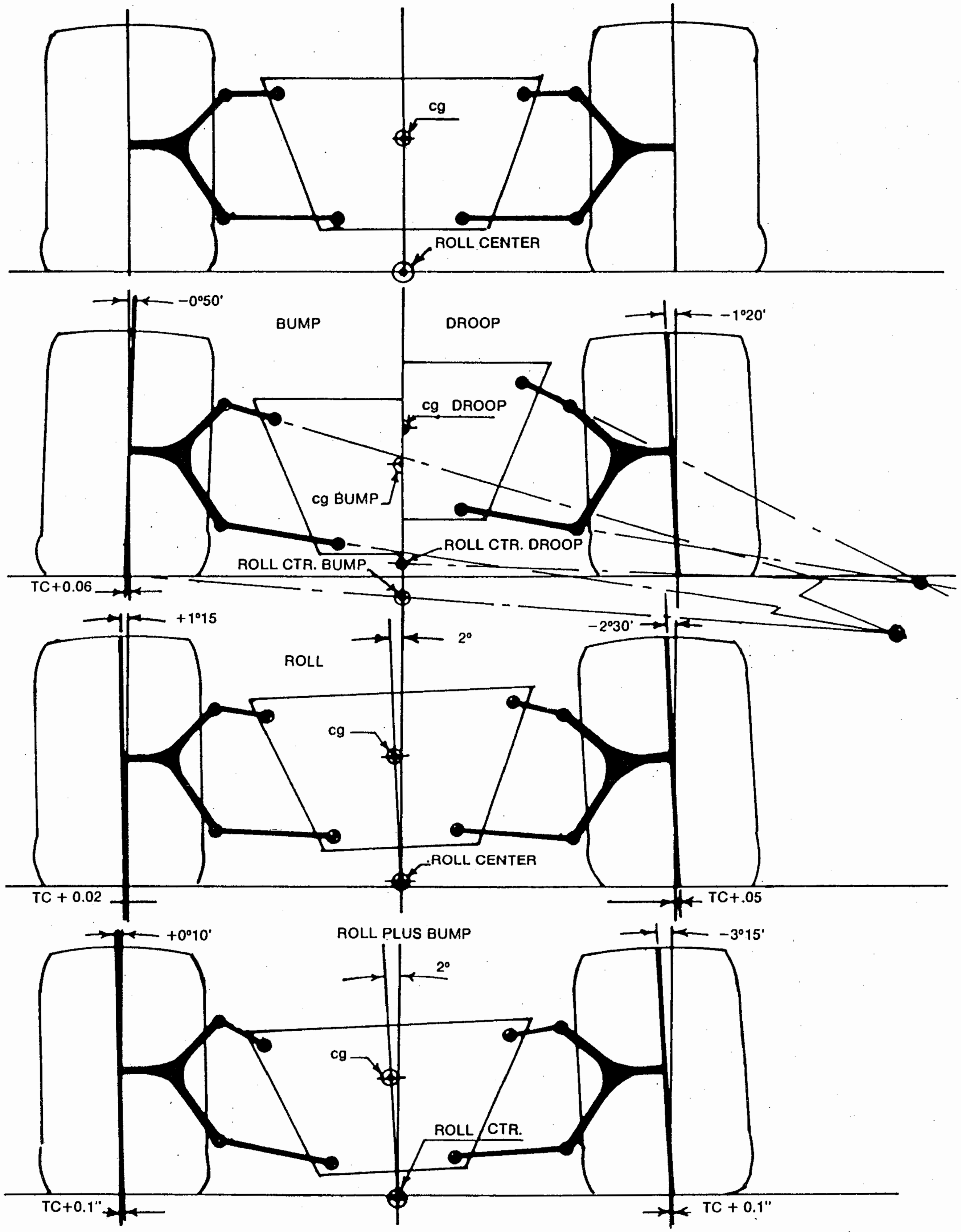



*Figure (26): Unequal length parallel links.*

(47)

important But NOT DEFINED

almost do in roll), so the instantaneous swing arm length varies quite a bit. This means that, if the wheels are allowed to travel very much, the camber curves will become very steep indeed. If great gobs of wheel travel are required—as in off-road racing—it is necessary to make the links closer to each other in length—try it on the model. At any rate, the roll center with unequal but parallel links stays pretty constant in relationship to the center of mass. Therefore the roll moment remains more or less constant, which is a good thing.

NOTE RECALL HO9,10

Naturally, there is no law that states that unequal and parallel links must be parallel to the ground at ride height—but a little experimentation with the model will explain why they normally are. About now I should mention that static ride height may well be different from the operating ride height if wings or effective spoilers are employed to generate downforce in meaningful quantities. Further, the operating ride height will then vary with road speed. Just one more little complication that we really don't need.

## UNEQUAL AND NON PARALLEL LINKS

While the unequal and parallel link set up reduces the positive camber of the laden wheel in roll, it does not reduce it enough for some tires to get really happy—and it produces really low roll centers. By inclining the link pivot axes with respect to each other we can place the roll centers wherever we please—at least in the static position—and we can further reduce the positive camber of the laden wheel in roll. Figure (27) illustrates. Admittedly things are a bit extreme in this diagram, but I wanted to illustrate what can happen when we go too far in any given direction. In this case, inclining the upper link downward toward the centerline of the vehicle has indeed notably reduced the positive camber of the laden wheel when the chassis rolls. But it has also rooted everything else. What has happened is that the inclination of the upper link is too steep, resulting in a very short instantaneous swing arm with the attendant very steep camber curves. By raising the inboard points of both the upper and the lower links we would achieve far better camber curves while maintaining the roll center in the same static location—of course then the roll center would move around more . . . As I said, I could go on forever but this is what the model is for.

## BASIC TRUTHS

After you have played with the model long enough, some general truths will begin to become evident:

(1) While it is possible to control wheel camber either during vertical movement or during chassis roll, it is not possible to achieve very good camber control under the combined conditions—we have an "either—or" situation.

(2) The longer we make the suspension links, the less angular and linear wheel displacement will result from a given amount of chassis or wheel movement.

(3) In vertical movement, the roll center moves with the center of gravity, tending to keep the roll moment constant.

(4) Increasing the effective swing arm length decreases the amount of camber change due to vertical wheel movement, decreases the amount of vertical roll center movement relative to the c.g. and increases the amount of lateral roll center movement.

(5) Except in the case of equal length and parallel links, long effective swing arms don't stay long when the wheel moves into the bump position or, for the laden wheel, when the chassis rolls.

(6) Increasing the inclination of the upper link (or shortening its relative length) results in more negative camber in bump, less positive camber on the laden wheel in roll and a decrease in the amount of wheel or chassis movement before we lose camber control.

## COMPROMISE

Given the fact that we cannot achieve Utopia in the geometry department, it becomes necessary to compromise. Everyone in this business has his own ideas as to which aspects of wheel path and roll center location control are more important and so we are very liable to see, in the same class of racing cars, lots of geometric variation. Despite this variation, most racing cars work very well. This is due to three factors:

(1) The present generation of racing tires is relatively insensitive, within reasonable limits, to camber change.

(2) Load transfer characteristics are more important to tire performance and vehicle balance than camber curves are.

(3) Different design philosophies tend to even out in terms of lap time—the car whose geometry tends to limit its absolute cornering power may well put the power down better—what you gain on the straights you lose in the corners and so on.

A few basic guidelines do exist to aid us in the selection of our geometric compromises:

(1) The front camber curve should keep the laden wheel more upright in roll than the rear. As the vehicle is turned (or pitched) into the corner, the combination of load transfers is going to compress the outboard front spring a whole bunch and we will need all of the camber compensation we can stand to keep from washing out the front end. In addition, due to its lower section height, the front tire is liable to be less tolerant of camber than the rear. For the same reason the front tire will offer more directional stability than the rear in order that the vehicle's steering response will be predictable and precise. A third factor is that, since the major portion of total vehicle lateral load transfer will take place at the front, the rear will roll less anyway. ??

(2) The front roll center will always be lower than the rear. If it is too much lower, we will have a car that does not enter corners well and which exits corner on three wheels. The big trick here is to keep the front and rear roll center movements approximately equal to each other—and in the same direction—as the car does its various things while negotiating a corner.

(3) We can control wheel camber within narrow limits of chassis roll and rather more broad limits of vertical movement. At some point in the generation of roll or vertical movement, the geometry will go to hell and the wheel paths will start to change very rapidly. The longer that we make the suspension links, the more movement can take place before we lose camber control—and the less wheel displacement we will suffer per unit of chassis movement.

48

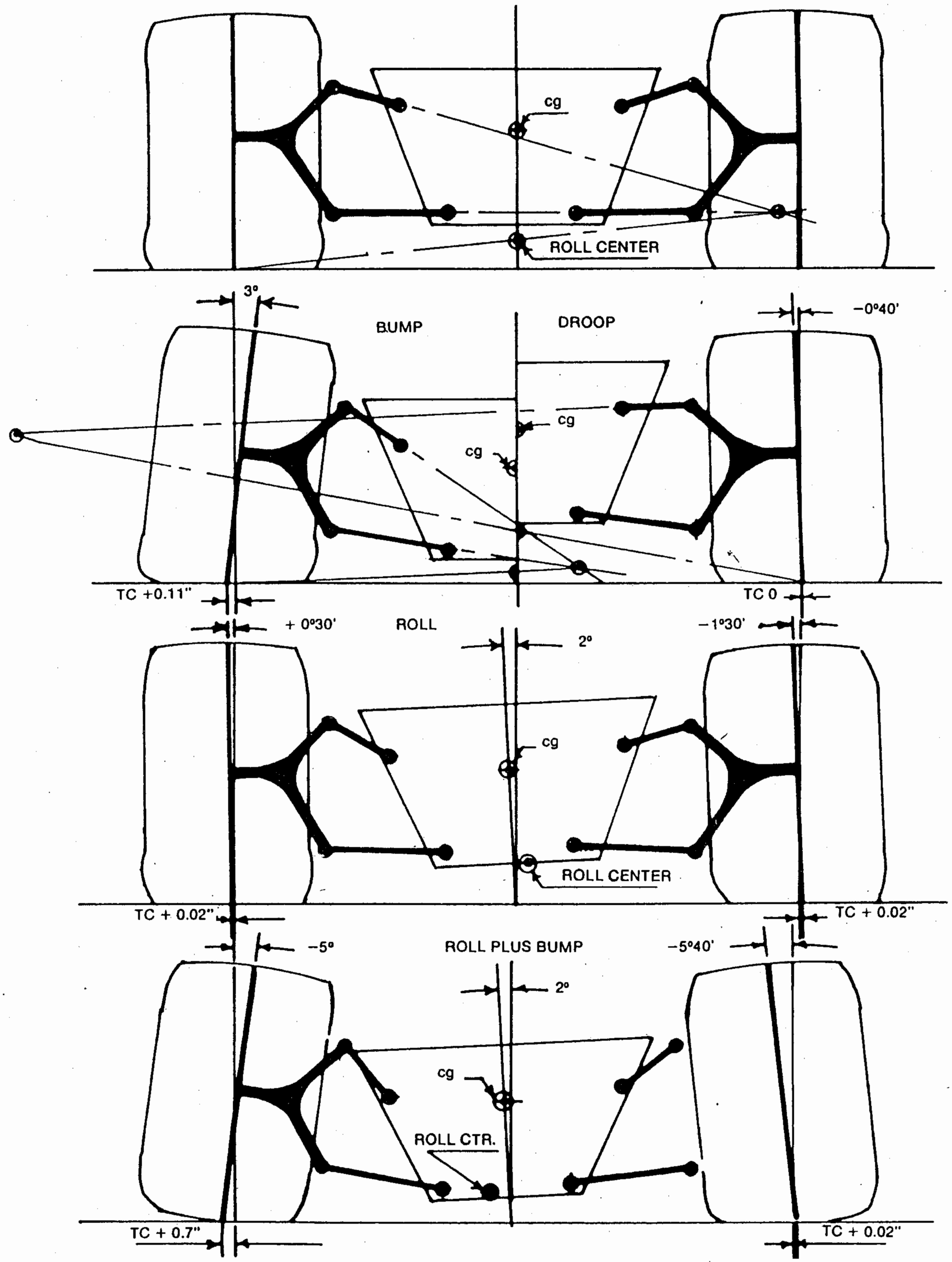



*Figure (27): Unequal and non-parallel links.*

49

My own pet ideas on suspension geometry and camber control stem from my firm belief that vehicle balance or driveability is more important in terms of lap time and winning races than ultimate cornering power. If I were a racing tire, I would resent any tendency on the part of my suspension links to abruptly change my camber, or to suddenly scrub me across the race track as I tried to smoothly change my operating mode from braking to cornering to acceleration in my efforts to follow the rim of traction circle. I would respond to such attempts by breaking traction momentarily. I would do the same if the lateral load transfer at one end of the car suddenly became a lot more than that at the other end because the roll moment at that end suddenly increased. I would bite and grip again after things had settled down—if they did—but I would momentarily lose traction due to the upset. It is not very likely that the driver would appreciate these antics.

So, I feel that we should design the geometry of our suspensions to minimize rapid changes of camber and relative front to rear roll center movement as the car goes through its transitions from braking to cornering acceleration.

The geometric possibilities are limited here and we are going to find it necessary to restrict the amount of chassis movement that takes place in response to centrifugal and to longitudinal acceleration. On most race tracks, we can strongly restrict chassis roll with only minor adverse side effects. We cannot, however, usually restrict vertical wheel movement without running into reduced tire compliance which will inevitably produce severe side effects—like slow lap times.

We have four methods available to us to restrict chassis roll—or reduce its effects:

(1) We can use high roll centers which result in low roll moments. We do not want to follow this approach because we will then have poor camber curves and high jacking forces.

(2) We can use anti-roll bars at each end of the car stiff enough to restrict roll to our desired maximum.

(3) We can use the suspension springs to restrict roll—either by making them stiffer, which is a bad idea, or by optimizing their placement so that we get maximum linear spring travel per degree of roll generated.

(4) We can use longer suspension links to reduce the amount of camber change generated per degree of roll.

We will go into these options in more depth in Chapter Six.

## TRACK AND WHEELBASE DIMENSIONS

The last geometrical considerations which we will consider are the length of the wheelbase and the widths of the track dimensions.

The advantages of a relatively long wheelbase are increased straight line stability, reduced longitudinal load transfer and pitching moments, somewhat easier reduction of the polar moment of inertia and more room to put things in.

The advantages of a relatively short wheelbase are reduced overall weight and increased maneuverability.

The advantages of wide track widths are reduced lateral load transfer for a given amount of centrifugal acceleration and room for longer suspension links. The major disadvantage is increased frontal area. When we get into aerodynamics, we will see that, at least on open wheeled cars, the importance of frontal area is overrated.

NO "theory

Very basically, the racing car with a long wheelbase and relatively narrow track widths will be very stable in a straight line at the expense of cornering power and maneuverability. The vehicle with a shorter wheelbase and wide tracks will be less stable, more maneuverable and will develop more cornering power. It will also be more difficult to drive to its limits. In general I favor moderately long wheelbases and wide tracks. I will point out, however, that if all of the corners are very fast, the disadvantages of narrow tracks can be overcome with aerodynamic downforce and, for USAC type racing the idea of a narrow tracked car with long suspension links and reduced frontal area is very attractive.

The situation becomes more complex when we consider the relative width of the front and rear track dimensions. I believe that the front track should be considerably wider than the rear track. More heresy! My reasons have to do with turning the car into corners and jumping on the power coming out. The wider the front track, the more resistance there is going to be to diagonal load transfer and the lesser will be the tendency for the car to "trip over itself" on corner entry and/or to push into the wall from the effect of the drive on the inside rear wheel when the power is applied. I believe that most of our present road racing cars, with roughly equal front and rear tracks, would benefit from an increase in front track width. The slower the corners to be negotiated, the more important this relative track width becomes.

GOOD!

## DIFFERENT STROKES FOR DIFFERENT FOLKS

The compromises in suspension geometry will vary with the type of vehicle and the nature of the race track upon which the car will do its thing. Factors to be considered include:

(1) Power to weight ratio

(2) Aerodynamic downforce to be generated and range of vehicle speeds

(3) Tire width and characteristics

(4) Track characteristics—smoothness, corner speed, degree of banking present and the amount of braking that will take place.

Let's now briefly consider the specific case of some different types of race cars and see how the operational conditions and factors affect the design of the geometry.

The ubiquitous Formula Ford features low engine power, low gross weight, narrow tires, virtually no down force generation, and crazy drivers. They do not accelerate very hard because they don't have much torque. Since they are not allowed to run wings, the operating ride height does not change much with road speed. The narrow tires will tolerate a fair amount of camber. What Formula Fords need from the suspension geometry is maximum braking power and maximum cornering power. They need the braking power, because one of the few places for a Formula Ford to get by another one is in the braking area. They need the cornering power, because they cannot afford to slow down any more

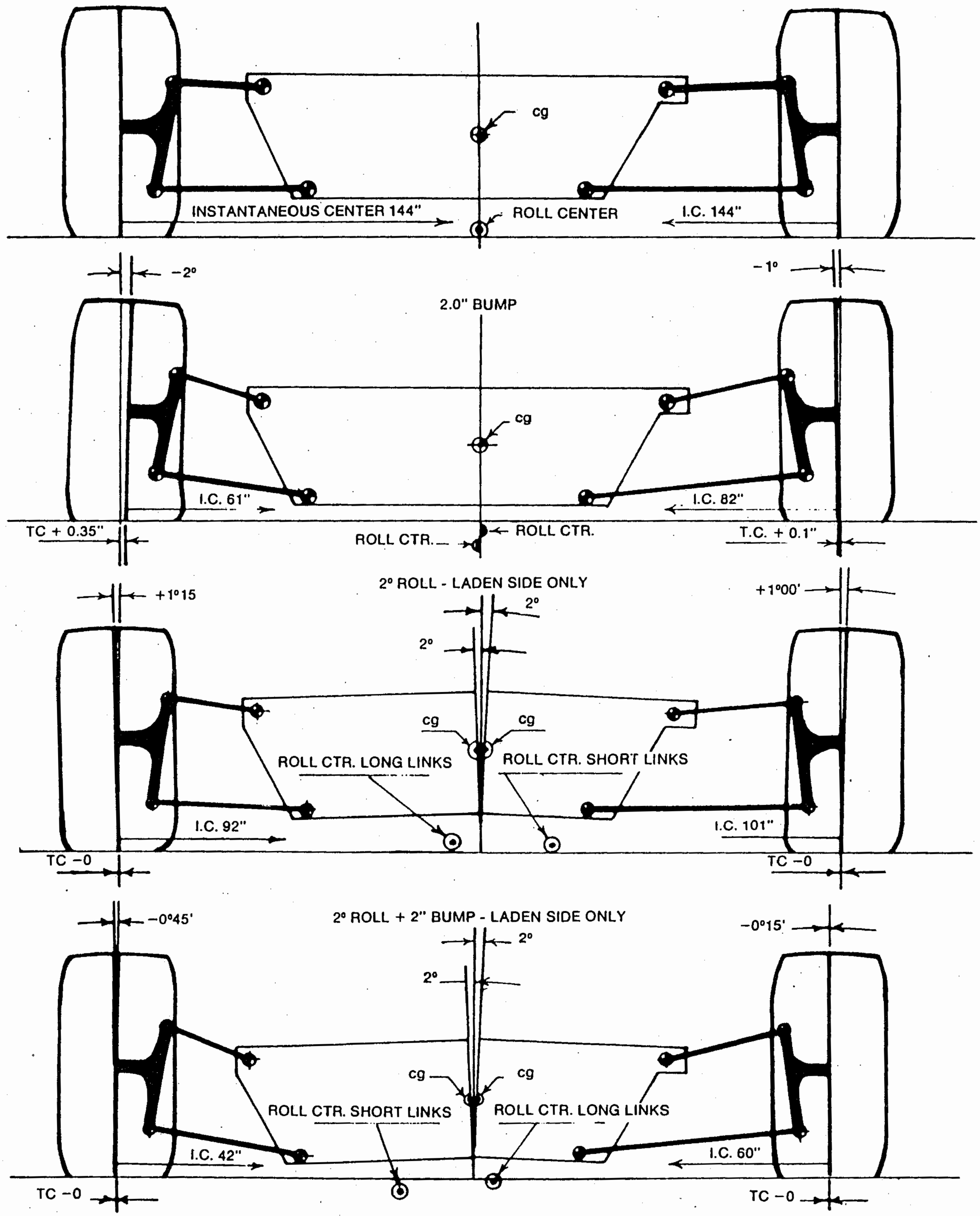



*Figure (28): Long links vs short links.*

than is absolutely necessary—with their low available torque, it takes forever to regain the lost speed. We get the braking power by keeping the front wheels as upright as possible in bump and not allowing the rear wheels to go into positive camber in droop. This means long links with not much inclination at the front—take a look at an ADF or an Eaglet. At the rear, we don't need to worry a lot about the effects of squat since we won't have enough torque to cause much of it. We do, however, have to worry about the camber of the laden wheel. As no limited slip diffs are allowed, we also have to avoid inside rear wheelspin which means lots of droop travel, avoidance of extreme negative camber generation on the inside wheel and minimum lateral load transfer at the rear.

Formula One, Can Am and the late lamented Formula 5000 cars offer a more complex set of operating conditions. Their road speed on a given track can vary from about forty mph to over one hundred ninety. The wings generate gobs of downforce which causes large differences in operating ride height from high speed to low speed. The tires are very wide and camber sensitive, and there is a lot of torque available to squat the chassis out of low and medium speed corners. The key to lap time in these vehicles lies in acceleration out of the corners. We have to ensure that the camber doesn't vary much with the changing ride height and that the rear camber doesn't get all upset as the chassis squats. To achieve this we sacrifice keeping the tires upright in roll and accept a somewhat lesser ultimate cornering power at the rear. This is compensated for by the simple fact that the rear tires are enormously larger than the fronts to accept the engine torque and that they will tolerate more camber than the fronts will anyway.

If we can tolerate some camber change at the rear, we cannot at the front. The low section tires just don't like it at all. The Chevy-engined brigade doesn't seem to have caught on to the advantages of very long front suspension links, but the Formula One group surely has. Figure (28) illustrates the effect of lengthening the links of a front suspension setup while maintaining the relative link lengths, track width and static roll center location the same. It gives one pause for thought.

Indy Cars on 2½ mile ovals operate in a relatively narrow, if very high speed, range—say 180 mph at corner apex to 220 at the end of the straights. While the torque available to squat the chassis is, even at those speeds, considerable—it is the same for each corner exit. Ride height change due to downforce is not super critical so long as it is realized that the operating ride height has little to do with the static ride height. When laying out the geometry and while aligning the car, the change in ride height from the shop floor to rolling into a slightly banked corner at 200 mph must be taken into account. Nose dive under the brakes is not a factor—except on the mile tracks or the road circuits—so negative camber due to forward load transfer can be pretty much ignored. Since the tracks are relatively smooth and the road speeds are very high indeed, relatively stiff springs and bars can be employed and chassis roll can be—and is—severely restricted. The compromise is weighted toward reduction of bump camber and track change.

Front engined sedans, with their high cg's and forward weight biases require that the outside front tire be kept as upright as possible—even at the cost of heavy bump camber change which can be reduced by anti-dive suspension.

In the world of Off Road Racing a number of things that the rest of us just barely realize the existence of become critical—like pitching moments. Roll shrinks to relative unimportance, and it becomes a matter of vast amounts of suspension travel and very effective damping. The big thing would seem to be to keep the wheels—particularly the driving wheels—on the ground for traction. Track change is not likely to be critical on offroad courses, but bump and droop camber probably are. I doubt that enough centripetal force can be generated on the surfaces involved to make roll camber very important, but the release of the energy stored in the rear springs when the vehicle hits one of those mini-cliffs that they call bumps can—and does—cause some spectacular endos. Why they still use swing axles is beyond me. My own opinion, totally unsupported by any experience, is that there is a lot of performance to be gained in this field in the geometry, cg height and polar moment areas.

## THE RELATIVE PLACE OF LINKAGE GEOMETRY IN THE OVERALL PICTURE

I believe that it is a hell of a lot more important to get the roll center locations and movements happy with each other and with the mass centroid axis than it is to get the camber curves perfect—which we can't do anyway. When we change the suspension pivot points —either inboard or outboard— and register a gain it is almost always because we have changed the roll center location rather than because we have modified the camber curve. I must also admit that we usually improve the balance of the typical English Kit Car by raising the front roll center—even at the cost of shortening the effective swing arm length. Mainly it is a question of getting the rate of generation of the front and rear lateral load transfers happy with each other.

## MODIFYING THE GEOMETRY

Once we have decided that our particular race car might benefit from a modification to its suspension geometry, we are faced with some decisions about how best to accomplish the desired end. Here we have to bear several factors in mind—structural soundness, cost—in both time and dollars—ease of returning to where we started (in case it doesn't work) and the feasibility of doing a valid back to back test to find out whether it works or not.

Changing link length or track dimensions is going to require the fabrication of new suspension links which, depending on the skills, time and equipment available, may or may not be a big deal. If you decide to make the links longer, take a really good look at the structural factors involved—they will necessarily have to be stiffer, particularly at the front, due to the brake torque loads being reacted over a longer distance.

Raising or lowering pivot points, at the front, is simply a case of making spacers for the ball joints, or of reducing the height of the uprights. It is always easier to do it outboard than inboard—except on production cars. The opposite condition exists at the rear where the outboard pivots are pretty well fixed in the hub carrier design but the inboards are bolt on structures or cross members which can be pretty easily replaced or modified.

So do what you think that you have to do. Align and bump steer the car with the alternate setups, write down how many turns you have to move what to achieve alignment and bump steer after you change setups; go to the race track and find out if it works. If it does, you may pat yourself on the back and feel good—but try to figure out WHY it worked while you are congratulating yourself. If it doesn't work, do not commit suicide—most bright ideas do not work. Make sure that you have not overlooked a contributing factor—like not readjusting the wheel alignment or bumpsteer when you changed the setup—and try to reason out why it didn't work. We normally learn at least as much from our mistakes as we do from our successes. The best development driver/engineer I ever knew once told me that he reckoned that about 20% of his bright ideas worked.

FROM Competition Car Suspension,
By Alan Staniforth

Even should you not be planning to go further than reading the foregoing, it will, perhaps, give a flavour of the questions and decisions in this area alone that face a designer up against the need to produce a winner - or find a new job. He has also to choose between a multiplicity of targets, many in total opposition to each other. These may be summed up in bare outline as:

"targets"

1 Roll Centre.

More than a little disagreement about how vital is its accurate location but a personal view with nothing convincing to alter it over quite a few years is "absolutely vital". It is common ground that a car in roll under a cornering G force must be rotating about some point in space. The dissent is usually about where it might be, if it has moved from a theoretical position and why. These arguments are absolutely no reason not to make strenuous attempts to locate and pin it down. Ignorance of at least an idea of the location of the front and rear roll centres, both statically and dynamically will prevent any attempts at logical thought or modifications when you have problems.

1st target

RIGHT

Varying roll centres at each end of the car must result in varying loads and weight transfer front, back and diagonally under differing circumstances. A moment's thought about the possible result for the driver in a fast left-right flick would suggest he will be in next lap complaining bitterly and calling for immediate improvement.

Current thinking on roll centre heights is of the order of 2 ins. above ground level to 1 in. below at the lighter/lower end of the car, with 4 ins. above down to ground level at the other end. These figures normally give an inclined roll axis along the car tending to even out the various forces involved if nothing else, but are by no means a fixed approach. It is a free world.

So we will aim at a dynamically steady roll centre so far as it may be possible, or at the very least similar movement at each end of the car.

"1st Target"

2. Outer wheel vertical in roll.

With the two outer tyres doing the major part of the work in accepting cornering forces, this gives the tyre contact patches their best chance in life. Many cars, from a Mini to a Formula One, do not do this at all well geometrically and need the wheels set with a negative camber so that roll pushes them out towards vertical.

"2nd Target" (poorly stated)

3. Wheel angle in bump/droop.

In direct opposition to (2), this has become ever more important in heavy braking and acceleration with wide flat slicks. In both situations there is a tendency under pitch forces for wheels to go into bump at one end and droop at the other, doing grip no good at all. A number of experiments over the years saw tyre companies (Dunlop for the Mini, Pirelli for Toleman and Ferrari) producing assymetric tyres with radiussed inner shoulders to try and alleviate this.

"3rd Target"

4. Track variation (scrub).

This means that individual wheels will, under certain circumstances, follow a private wavy path rather than a straight line. Most important when running straight, but

4th Target

may be a contribution to momentary under or oversteer lurch on turn-in to a corner.

For the intrepid souls who are going ahead with the String (or any other) Computer, a few guidelines.

From scratch, work from a bottom link parallel to the ground and as long as practicable, combined with a top link two-thirds of the bottom running downhill from wheel to chassis at 15 degrees or so from horizontal. Note every experiment in detail as you carry it out. Trying to remember results and then compare them will only lead to a hopeless muddle.

starting point with or without SOFTWARE

Operations and effects:

1. Alteration of wishbone lengths - little major effect.
2. Lengthening both at once - often poorer.
3. Vertical movements of chassis mounts - major effects.
4. Scrub - rarely any problem in roll, can be serious in bump/droop.

54

5. "Skewed roll" - this may, for instance, combine 2 degrees of roll with 1 in. of bump on a front or rear wheel and can often give excellent results in terms of outer wheel angle.

"fifth" Target

All the variations we have considered here will, of course, be reduced by restricting the actual movement of the vehicle, a basic approach explored to the full both in Indy Cars and Formula One, giving minimal roll (controlled by extremely strong bars) and almost fixed ride heights (very hard suspension and high wheel frequencies). Having said this, one has to admit that several newest generation Formula One cars have only a modest front bar and no rear bar at all, the inference being that wheel frequencies are rising substantially and coils are taking back much of the task.

Such a simplistic method transfers much of the suspension's task onto the tyre carcase - and if you thought geometry was a little tricky it pales beside the mysteries of what a tyre may be doing in ten thousand situations, all different.

Here are some general pointers (irrespective of the method used).

1. Very short swing axle lengths (20 - 40 inches):
a) Roll centre (RC) generally high.
b) RC location generally good.
c) RC sideways movements at a minimum.
d) Very good wheel angle control in roll.
e) Camber variations in bump/droop very bad (almost linear with roll in some cases).
f) Bad performance in scrub.

2. Long swing axle lengths (70 - 180 inches):
a) RC low.
b) RC location reasonable, subject to (c).
c) RC can move considerable distances sideways due to the shallow angles involved.
d) Mediocre wheel angle control in roll (worst on inner wheel).
e) Camber alterations in bump/droop good.
f) Good performance in scrub.

3. Medium swing axle lengths (40 - 70 inches):
As might be expected, the results lie between short and long.

4. Ultra-long swing axle lengths (effectively out to infinity, with certain near-parallel designs):
a) RCs very low, at or below ground level.
b) RC location very good in vertical terms.
c) RC sideways movement can be very great, with a reversal from inside to outside of the corner possible due to the very narrow, near parallel angles.
d) Wheel angle control poor - may be near direct equivalency to car's roll angle.
e) Camber alterations in bump/droop very small.
f) Scrub - good.

Summary of findings of fooling around with software or the "String computer" - Expressed as SWING ARM LENGTHS