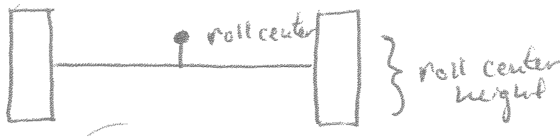
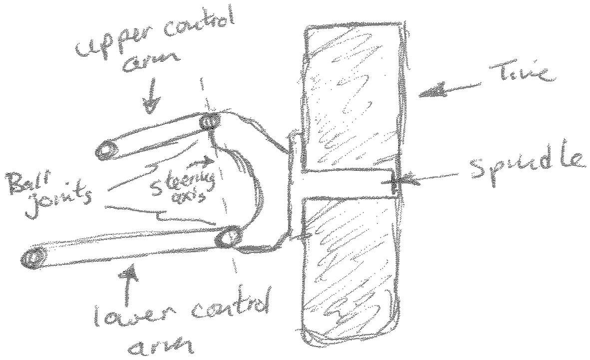
Suspension Design + Roll Centers

We introduced the simple model below to model a generic suspension type as far as load transfer is concerned:

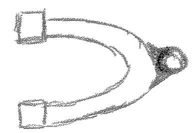


This definitely bears little resemblance to the physical characteristics of independent suspensions so where does it come from? This requires a look at the kinematics and force transfer of a suspension starting with...

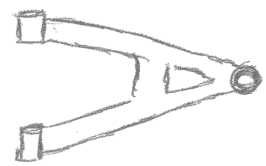
The Double Wishbone (aka double A-arm, SLA or "short/long arm")



Each control arm is a wishbone...

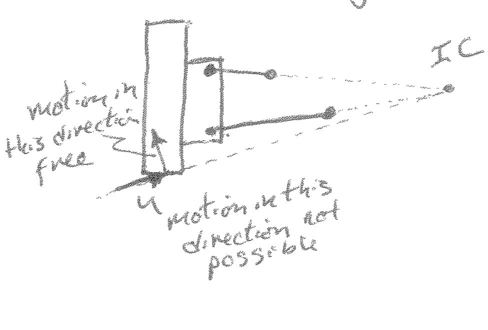


or an A-arm

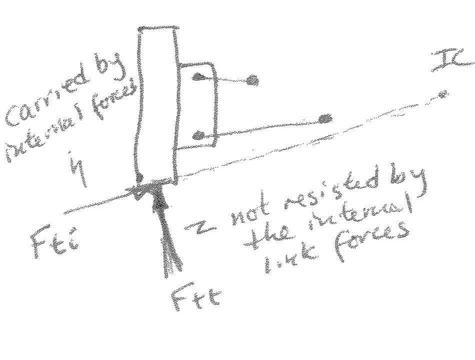


This suspension gives a lot more flexibility to the designer than other choices though tends to be more costly.

The key to analyzing the roll properties of this suspension is to see that from the front it is nothing more than a four-bar linkage!



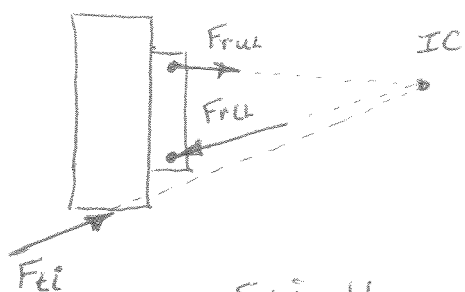
The car is link 1, the wheel body is 3 and the links form 2 and 4. As a result there is an instantaneous center of wheel rotation relative to the vehicle body. Ignoring any compliance in the bushings, the center of the contact patch can only move perpendicularly to the line connecting it to the I.C.



We can also divide the forces at the contact patch into the component lying along the line to the I.C. and the component perpendicular to this line. The former is reacted by forces in the links; the latter produces suspension motion.

For the results that follow, assume that the spring force is applied directly to the wheel in the same direction as the component that can deflect the suspension. This is not strictly the case and a similar development can be made for springs mounted, for instance, on the lower control arm. This just makes life more complicated without really adding intuition (it should be clear from a virtual work argument that the springs must in any case react the forces perpendicular to the line to the I.C.).

So after removing these forces, the wheel looks like:



Frul - (r)eaction force in (u)pper control arm on (l)eft side of car

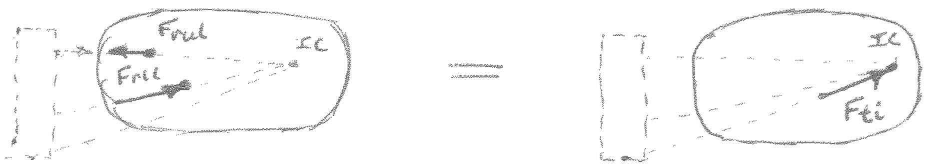
Fti - (t)ire force along line to (i)c.

Since these must be in balance, taking components

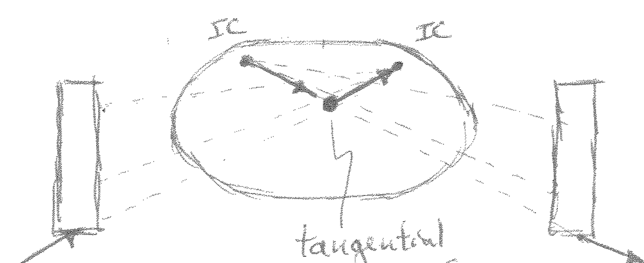
$$(F_{rul} + F_{rll})_x - (F_{ti})_x = 0$$
$$(F_{rul} + F_{rll})_y - (F_{ti})_y = 0$$

$$\sum M_{IC} = 0$$

The fact that the moment of each force about the I.C. is zero enables us to develop an equivalent force to the reaction forces on the sprung mass...



So with our idealized spring model, we can translate the tangential forces at the contact patch to the sprung mass at the instantaneous center.



tangential tire forces have no moment about this point - the roll center

So there is a point around which the suspension link forces produce no sprung mass roll.

This is also the point where lateral forces are applied to the sprung mass, just like our model.

This is really a clever abstraction of the suspension design process to think in terms of roll centers and instantaneous centers. Instead of treating all of the complexity of individual suspensions in the handling calculations, handling is usually discussed in terms of centers and then mapped back to specific suspension parameters.

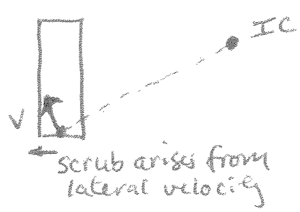
Where can this simplification cause problems?

(1) In turning, the inside and outside forces are not the same. This means the net force at the roll center has a vertical or "jacking" component. Physically, the suspension carries some of the load, thus unloading the springs. We'll talk about this more in terms of the swing axle suspension.

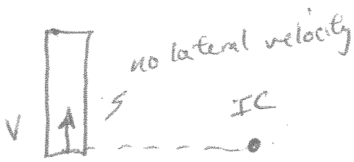
(2) The roll center is not fixed, but changes as the vehicle rolls and the suspension deflects (this problem can also combine with problem 1 above). You can make suspensions where the roll center moves a lot (though people try not to do this). The same construction holds (mostly - see Dixon for exceptions) when the vehicle rolls only the roll center drifts from the centerline and changing roll center height + distance must be included in the analysis.

In practice, there are several reasons why designers try to keep instantaneous centers and roll centers from moving too much. This follows from a quick examination of what intuition we can get from roll or instantaneous centers.

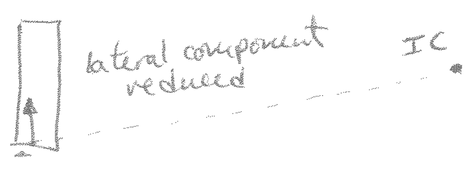
First, remember that the velocity of the tire must be perpendicular to the line joining the I.C.



This means that as the tire moves up and down there is a lateral component to its motion. In other words, the tire must "scrub" the contact patch as it moves. This is bad for wear and driver feel (since scrub produces side forces).



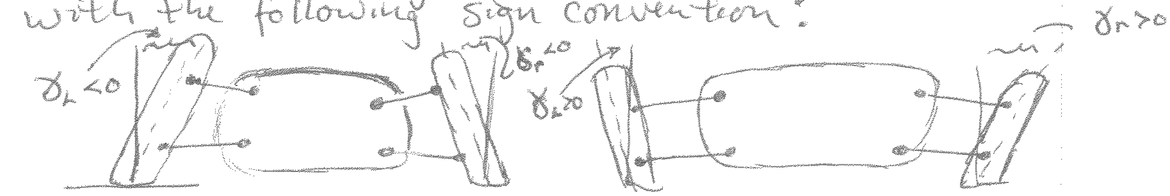
These problems can be eliminated if the instantaneous center is on the ground. Placing it farther from the wheel also helps minimize this effect.



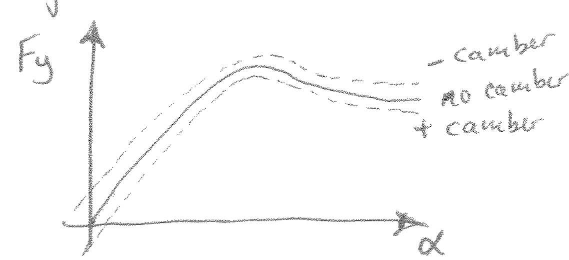
Thus from a scrub standpoint, it makes sense if the roll center falls near the ground and close to the vehicle centerline. Some designers claim that roll centers that move great distances from this location produce bad subjective handling.

Keeping the instantaneous center (and hence the roll center) relatively stationary is generally associated with long link lengths. This, however, conflicts with one requirement for performance handling...

As the suspension deflects, the camber angle of the tire changes. The camber angle is defined as the inclination angle of the tire from the vertical with the following sign convention:

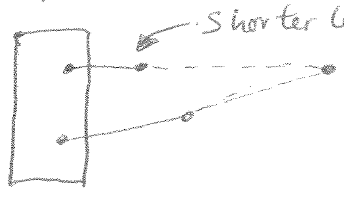


Negative camber occurs when the top of the tire leans inwards towards the vehicle body. In hard cornering, negative camber on an outside wheel is a good thing since it can increase side force.



This effect holds only up to about 5° for radials on passenger cars and up to an even lower value for wide racing tires.

For performance, it is helpful for the negative camber to increase on the outside wheel as the vehicle corners. This can be accomplished by shortening the upper control arm length

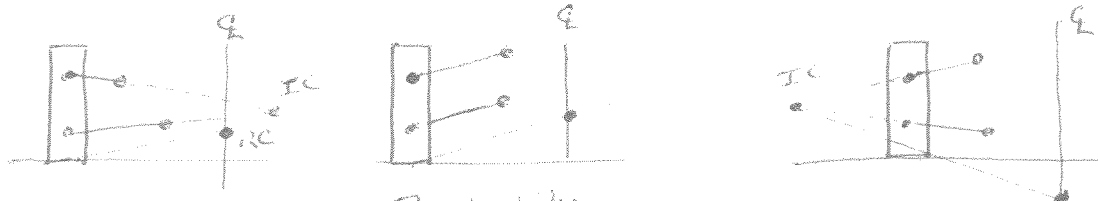


Shorter length means the top of the wheel is "pulled" towards the vehicle as the suspension deflects.

Note that camber gain can be set after the nominal value of the instantaneous center is established since it requires changing only the length of the control arm.

In the real world, of course, ideal suspension geometry must sometimes take a back seat to the need to package brakes, steering, springs and shocks and the fact that these links need to be physically attached to the body.

Roll centers for double wishbone suspensions are found in the same manner regardless of link arrangement!



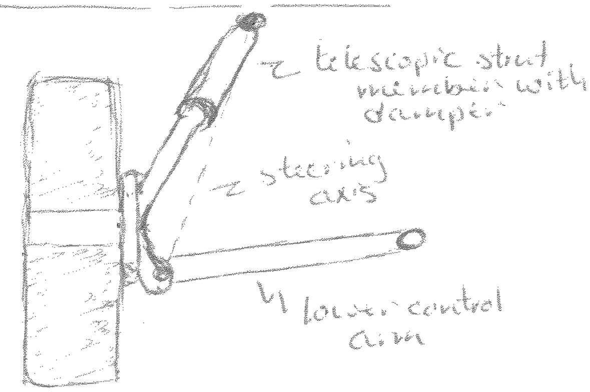
Positive swing arm – Roll center above ground

Parallel links – IC at infinity

Negative swing arm – Roll center below ground

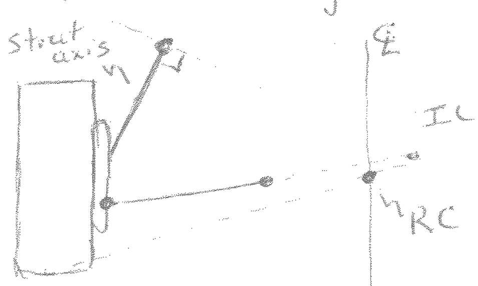
Will this work for other suspensions? Yes! For example, the...

MacPherson Strut



The MacPherson strut suspension has a lower control arm with a telescopic strut replacing the upper control arm. The damper is integrated into the strut instead of being a separate element.

The strut can be considered like an infinitely long upper control arm since it allows only linear motion along the strut axis.



The instantaneous center is found by drawing a line perpendicular to the strut axis and locating the point where this crosses a line drawn extending the lower control arm.

From the discussion of camber gain, one of the main disadvantages of this suspension type becomes clear – the MacPherson strut cannot be set up for the same negative camber gain as the double wishbone. It also has the disadvantage of requiring a higher hood height to accomodate the strut.

The advantages of the strut include simplicity, cost, suitability for unibody construction (since the mounting points are spread widely) and compatibility with transverse mounted engines (since it leaves lots of open space).

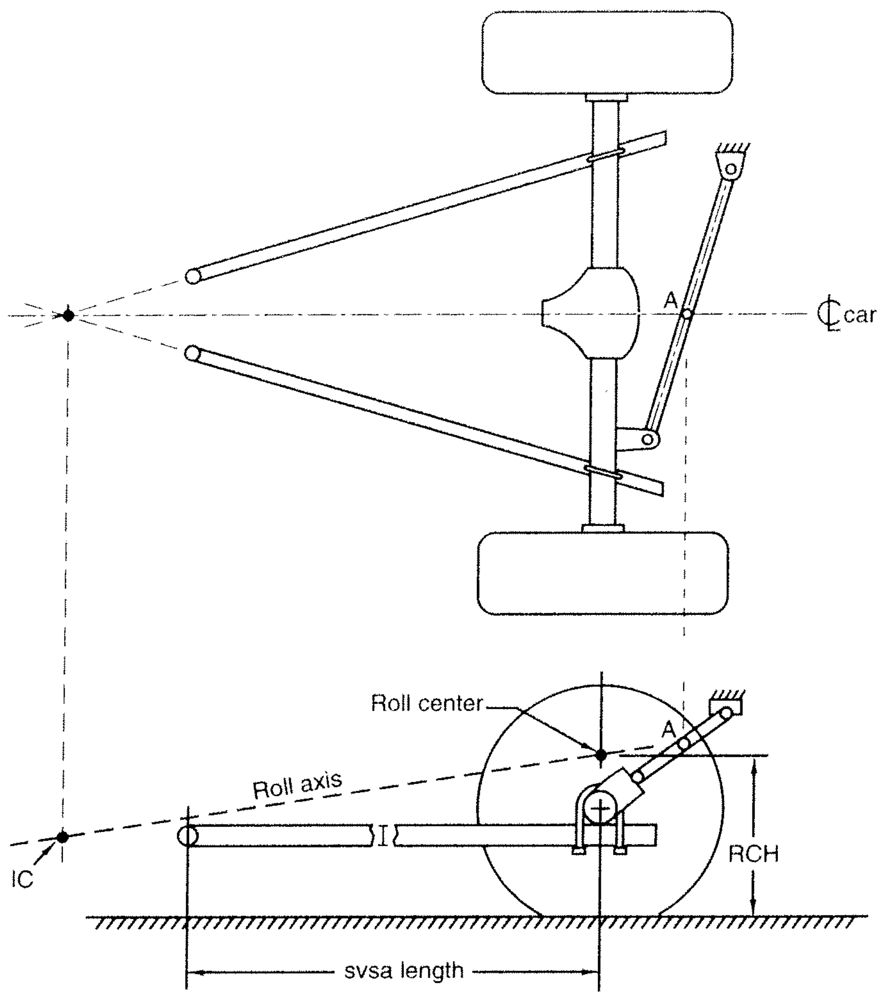


Figure 17.40 *"NASCAR" type rear axle.*

Torque Arm Suspensions

There are other styles of suspensions using an arm in bending to react torque. The General Motors F-car line (Firebird and Camaro) has one such example, shown in Figure 17.41, which is also similar to the Type 35 Bugatti of 1924. Here the system is very much like the three-link and track bar except instead of the third link upper arm there is a torque reaction beam rigidly attached to the axle housing and extending forward to the transmission end housing. The torque beam is mounted in rubber and is free to slide fore and aft but is not allowed to move up and down or side to side. There are two lower control arms and a Panhard bar to complete the kinematic control.

The kinematics are a bit unique for this case in that the side view instant center is not obvious. First a line is extended through the lower control arm pivots forward past the