Chapter 2

Automobile Suspension Overview and Characterization

2.1 The Need for Basic Understanding

Dynamic analysis of an automobile is very complex and not easily simplified. Sufficiently describing the engineering properties necessary for vehicle design continues to be difficult to comprehend and explain. The interaction between tire and suspension systems contain many subtleties which greatly influence vehicle control. Doing research in the vehicle dynamics field without prior exposure to vehicle design or analysis can be a frustrating experience. This area of engineering is terminology heavy. Basic terms like; understeer, oversteer, lateral load transfer distribution, roll center, slip, etc. have very involved meanings. It is not enough to understand the meaning of these terms but the designer must recognize the subtle interactions to apply them effectively. Even further, the designer must know the modes of interaction that are most desirable for optimum performance of a specific vehicle. This chapter expands on information presented by Cole [8] and presents a "physical feel" to some of the basic parameters needed to do a vehicle-suspension-design analysis. The S.A.E. publication on "Vehicle Dynamics Terminology" [18] may also prove useful in this area.

First, the force and moment generating capabilities of the pneumatic tire will be characterized. Then the method through which the automotive suspension augments these capabilities will be outlined.

2.2 Pneumatic Tire Mechanisms for Exerting Tractive/Braking and Cornering Effort

The pneumatic tire is a complex force and moment generating element. As the vehicle's interface with the environment it must provide many functions. The pneumatic tire must:

- support the vehicle's weight over a finite contact area;
- generate the shear forces necessary for traction, braking, and cornering;
- provide isolation over road irregularities;
- provide adequate steering control; and
- guide the vehicle and insure directional stability.

More than any other single component the tire can influence the handling characteristics of a vehicle. Consequently, the pneumatic tire is the subject of much research. To facilitate this research tire analysts have devised a standard tire axis system. This axis system allows the six forces and moments generated at the tire/road interface to be conveniently defined, see Figure 2-1. The performance characteristics of a tire are defined in terms of this standard axis.

The primary function of the pneumatic tire in handling analysis is to generate a lateral cornering force. Many parameters influence the tire's ability to do this. The most significant of these are listed below and shown graphically in Figure 2-1:

·normal force;
·camber angle; and
·slip angle.

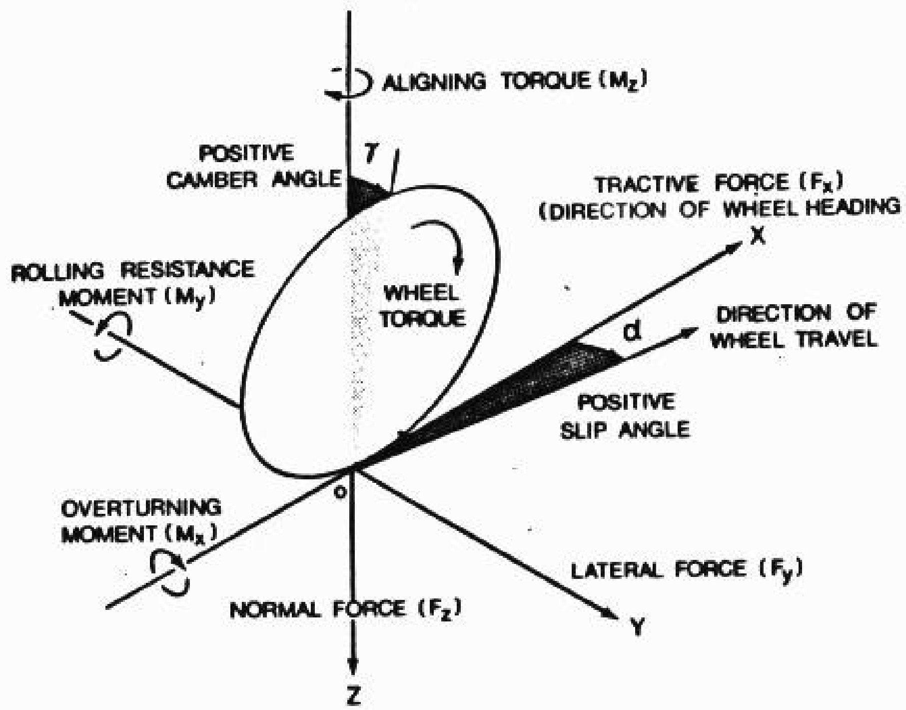


Figure 2-1: The tire axis system

Normal force has a weak influence on tire lateral force generation and is often neglected. As normal force increases the tire can generate more lateral force. After a transition point, as the normal force increases the tires ability to generate more lateral force diminishes. This non-linear phenomenon is called "normal load roll off" and is shown in Figure 2-2.

The influence of camber angle on the cornering capabilities of a tire is more appropriately explained in conjunction with suspension camber effects. Camber effects on tire force generation will be discussed in section 2.5.1.

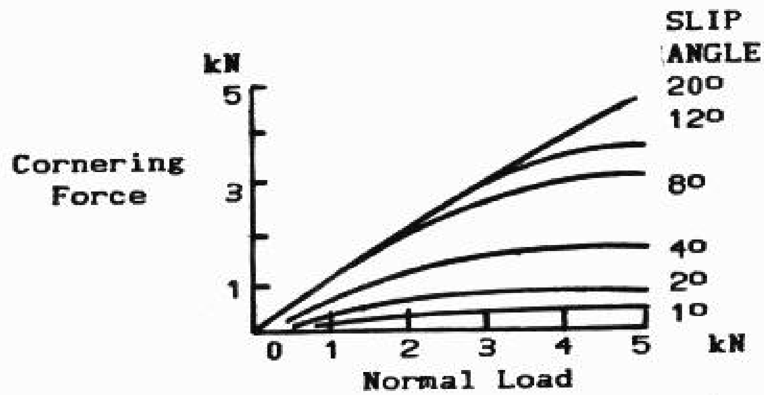


Figure 2-2: Effect of normal load on cornering characteristics

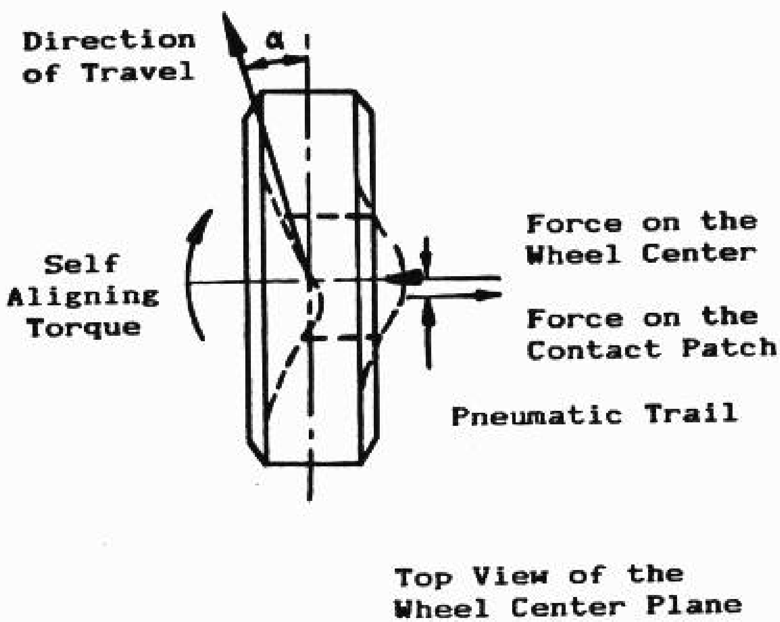


Figure 2-3: Top view of a cornering tire

The slip angle represents the primary mechanism through which the tire generates lateral force. During cornering the vehicle applies a lateral force at the wheel center. This lateral force is reacted by shear forces at the tire/ground interface. Figure 2-3 shows that the resultant of these shear forces is located at a distance behind the wheel center. This distance, called the pneumatic trail, causes a self-aligning torque and contributes greatly to the creation of the slip angle. The slip angle phenomenon is the deformation of the tire in such a way that an angle is created between the wheel plane and the directional heading of the tire. This is the same as saying that a free-rolling tire, with no lateral forces applied to it, will move in a direction parallel to the plane of the wheel. When lateral forces are applied at the wheel center the directional heading of the tire will be rotated an angle ALPHA from the wheel plane. As lateral forces are increased the slip angle will also increase until a non-linear limit is reached. This point is called the limit of adhesion or tire saturation.

This explanation of the slip phenomenon is used to lend physical feel to the mechanisms of tire cornering and is not intended to be comprehensive. In actuality the causes for cornering slip are many and very complex and this explanation is presented to lay ground work for later topics.

The mechanism that a tire uses to apply tractive/braking effort to the ground is also called slip. However, tire "longitudinal slip" should not be confused with the tire "slip angle". Even though these phenomena have similar names the mechanisms themselves are very different. Two explanations of the tractive/braking effort vs longitudinal slip mechanism are given.

Referring to Figure 2-4 we see the side view of a pneumatic tire rolling at a nominal speed, w. In an un-deformed state the tire is broken down into equally shaped sectors. The inner diameter of the tire is rigidly fixed to the wheel. As a tractive moment is applied to the wheel center, the tire distorts near the ground contact area. As a given sector moves toward the tire/road interface, the tangential dimension of the sector is

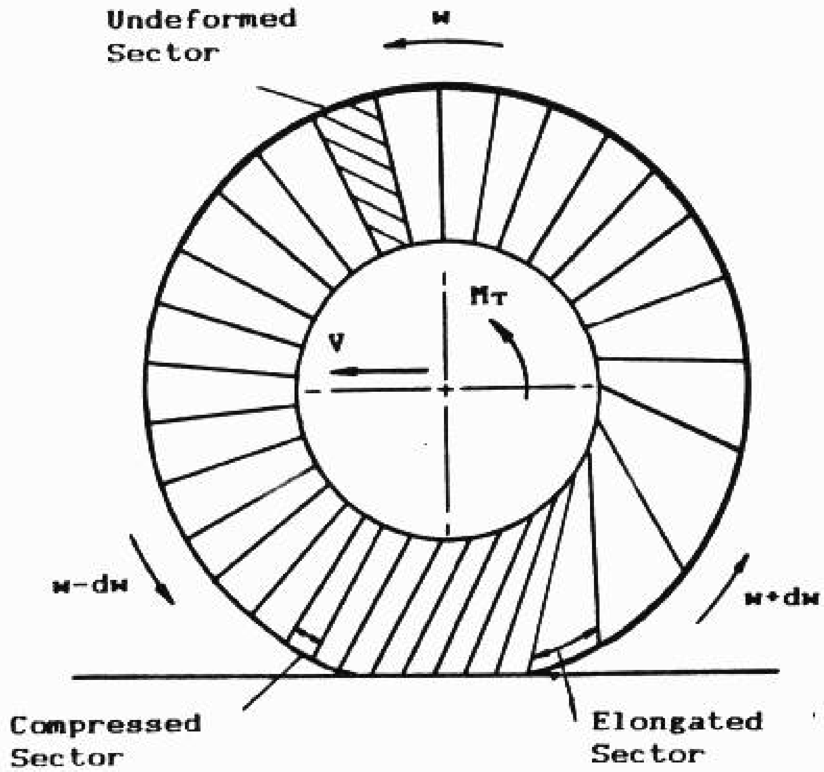


Figure 2-4: Sector model of longitudinal slip

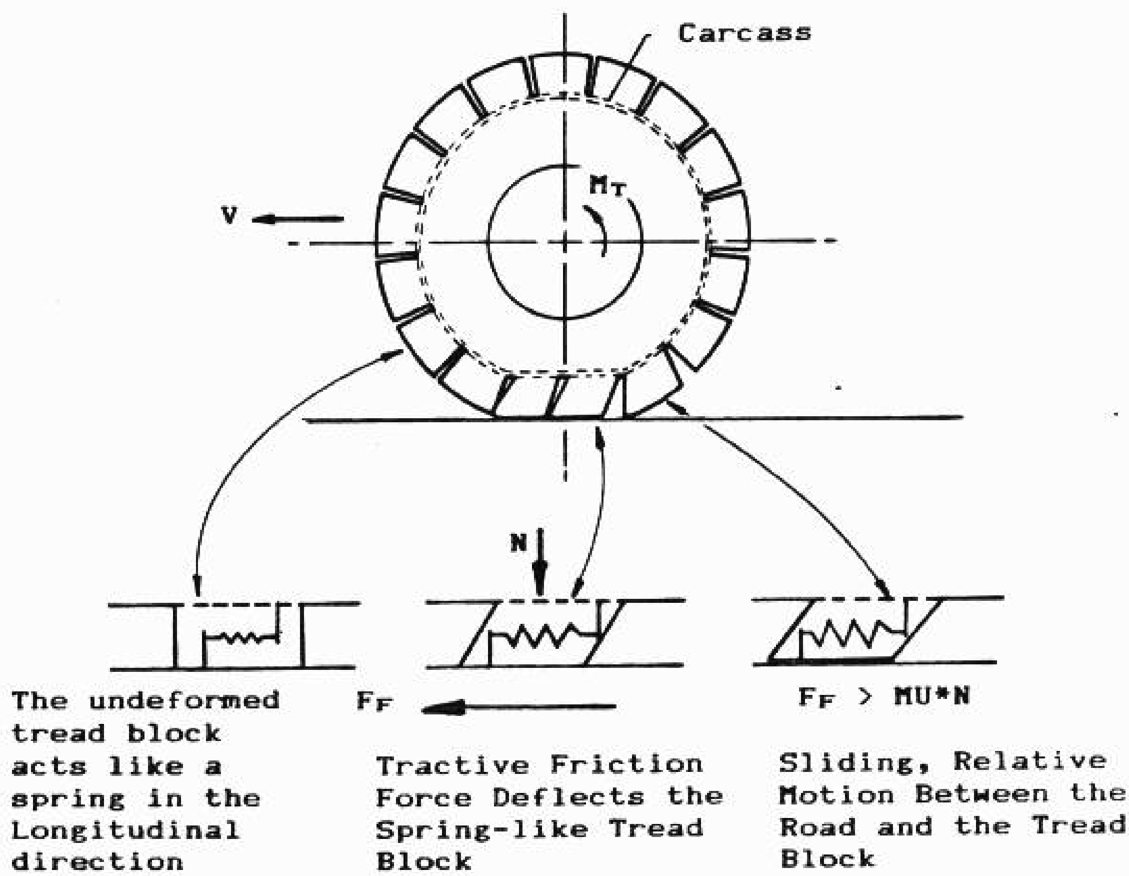


Figure 2-5: Rubber block model of longitudinal slip

compressed. This causes the perimeter of the tire to be shortened in this area and consequently the perimeter slows to a speed of w-dw. The shortening continues until the sector moves into the tire/road contact patch where the perimeter length remains constant (for the most part). When the sector is contacting the ground, and there is no sliding, the perimeter of the tire has the same velocity magnitude as the wheel center, V. Once the same sector reaches the back edge of the contact patch it is subject to tension and elongates beyond the equilibrium size. At this stage the perimeter speed increases to a speed of w+dw. Finally, as the sector moves to a point that is above the wheel center it will have the nominal angular speed, w. Therefore, with tractive effort applied at the wheel center there exists a rotational speed, w, that is different from the translational speed, V, divided by the tire radius.

The second explanation is facilitated by making the analogy that the tire tread is a small block of rubber. If fact, the tire tread is made up of many small rubber blocks attached to the tire carcass. Referring to Figure 2-5 as one of these blocks comes into the ground contact area, part of the tire normal load pushes this block onto the road. The normal load allows the block to exert a friction force. During this condition the block acts like a longitudinal spring between the road surface and the tire carcass. As the tractive moment on the wheel center is increased the block deflects and the frictional force builds. When the frictional force reaches the static limit, MU*N, the block will begin to slide. During sliding the road-contacting surface of the block will have a speed that is greater than the road surface or the carcass. If we expand this so that all the tread blocks that come into the contact area undergo this phenomenon, then it is possible for the tire to spin a minute amount without any translational progression and no sliding. Therefore from a macroscopic perspective the translational velocity of the wheel center can be different than the angular velocity times the loaded radius.

The longitudinal slip phenomena is represented by a non-dimensional relation:

$$s = \frac{V - r\,w}{V} \qquad (2\text{-}1)$$

In the general case longitudinal slip is non-linearly related to tractive/braking effort. As with cornering slip, when longitudinal slip increases the tire will saturate and no more effort will be generated for increasing values of slip. This is the longitudinal adhesion limit and usually implies impending sliding of the tire on the road surface.

The mechanisms that allow pneumatic tires to generate tractive /braking and cornering effort have been explained separately. However any given tire can experience both of these phenomena simultaneously. Further, at moderate to high values of tire effort these phenomena do not add together linearly. Few comprehensive studies of tire performance under combined loadings have been done. Combined loadings is an area of current interest and data, both experimental and theoretical, are becoming available. Therefore, until recently it has been rather challenging to do combined maneuver analysis. One early representation of this type of data is called the tire ellipse and is shown in Figure 1-2

Hopefully these explanations of the slip mechanisms will increase the physical appreciation for how a pneumatic tire performs its function. Next the characterization of automotive suspensions and how they interact with tire systems to create the handling traits of a given vehicle will be explained.

2.3 Understeer and Oversteer Defined

The understeer/oversteer characteristic is generally used to provide information about the stability of a vehicle. Basically if a car understeers it is stable, if it oversteers it is unstable, and the borderline condition is called neutralsteer. Some vehicle dynamicists will argue this point but it is generally true for domestic vehicles. Rather than discussing the reasons why, the terms will just be defined and a more full explanation is given in Jacobson [9]. Consider a vehicle going around a constant radius curve at a constant speed, steady state. At this point understeer/oversteer is defined as what happens to the steering characteristics of the vehicle as the speed is slowly increased. Also when defining this parameter in terms of slip angles, one must consider the "effective slip angles" that include: all weight transfer steer effects, roll steer, camber steer, compliance steer, etc. (refer to later sections or [9]).

A vehicle is understeering when the driver must give additional steering lock (turn into the curve), as the speed is increased, for the vehicle to maintain the same curve radius. Another way of defining understeer is; as the speed increases the vehicle must counteract higher lateral acceleration forces. This requires that the slip angles at the tires must increase. Understeer occurs if the front slip angles increase at a greater rate than the rear slip angles.

A vehicle is oversteering when the driver must give opposite steering lock as the speed is increased, for the vehicle to maintain the same curve radius. Or, oversteer occurs if the rear slip angles increase at a greater rate than the front slip angles as speed (ergo lateral acceleration) is increased.

A vehicle is neutralsteering when the driver must maintain the same steering angle, as the speed is increased, for the vehicle to maintain the same curve radius. Or, neutralsteer occurs if the front and rear slip angles increase at the same rate as speed is increased.

2.4 Ride and Handling Implications of Unsprung Mass Magnitude

In vehicle modelling reference is made to the sprung and unsprung mass. The meaning of these terms becomes much more critical when they are related to different suspension types. The unsprung mass includes the masses that are suspended between the tire and the suspension springs. These include the:

- wheels;
- tires;
- brakes(if located at the wheels);
- part of the suspension linkage;and
- part of the suspension springs.

The sprung mass includes everything that the unsprung mass does not; the major portion of which includes the body or passenger compartment. The sprung mass is the entity which the suspension is to support, isolate, and control. The ideal situation occurs when the sprung mass can be maintained motionless under any external disturbance. Approaching this ideal situation depends on the magnitude of both the sprung and unsprung mass while considering ride and handling.

The magnitude of the unsprung mass affects both the ride and handling properties. An optimal value is a trade off between ride and handling. For example, consider a vehicle with a very low unsprung mass. With low unsprung mass, more wheel control is possible. This is because the forces necessary to accelerate and decelerate the wheel system and maintain wheel contact with the road, are smaller if there is less mass to control. Also, a more accurate wheel position can be maintained and the suspension responds more rapidly over road irregularities. This implies the handling characteristics of the vehicle are improved because more of the large primary ride motions (pitching and hopping over road swells and pot holes) are taken up in the suspension and less forces are transmitted into the sprung mass. The disadvantage of a very low unsprung mass is that low-amplitude high-frequency road inputs are fed into the sprung mass. If the unsprung mass were large it would act as an anvil and would absorb high-frequency road buzzes. The components of a low unsprung mass system can start to vibrate with little road input. This is perceived as ride harshness by the passengers. Ride harshness is commonly observed when the tire impacts a tar strip or crack in the road surface. Rubber isolation helps this harshness but there is a practical limit to its use.

Summarizing, the magnitude of the unsprung mass has implications to the ride and handling performance of a vehicle.

2.5 Suspension Characterization for Vehicle Modeling

The suspension system provides essential handling characteristics to the automotive vehicle. The major suspension functions are to support, isolate, and control the body of the vehicle over road irregularities. During a "pure cornering" analysis, the representation of the suspension is not necessary. This is because during a cornering analysis the motion of the suspension is symmetric from one side to the other. This implies that tire performance modifications due to suspension travel can be easily incorporated into the cornering stiffness of the tire, or incorporated into the tire model in some other way. However, when a combined-maneuver analysis is being performed the motions of the suspension are not symmetric; which has major implications in tire force/moment generation and over-all vehicle handling characteristics. As a result, some representation of the suspension must be modelled in a combined-maneuver analysis.

2.5.1 Suspension Functions and General Parameters

Above, the major functions of a suspension were listed briefly as; to support, isolate, and control the sprung mass over road irregularities. However, to understand fully the workings of a suspension, a more specific list is provided below. The suspension should:

- support the weight of the total vehicle;

- ·cushion the sprung mass over road irregularities;
- ·constrain the motion of the unsprung mass to one degree-of freedom, essentially vertical (where small variations in the other five wheel dof's are prescribed by the suspension geometry);
- ·distribute driving and braking forces to the chassis;
- ·provide steering control and enhance directional stability of the vehicle;and
- ·provide control of the sprung mass orientation during cornering, braking, and acceleration.

The way the suspension performs these functions is to place the tire in contact with the road in such an orientation so it will maintain sufficient adhesion to perform a given maneuver. This is not to say that the only reason for a given suspension trajectory is to give the tire the proper orientation. On the contrary, the primary reasons for a specific suspension geometry are sometimes packaging and cost with just an eye toward optimal wheel/road orientation. Within these constraints, and for a given suspension type, many parameters can be adjusted to optimize the road adhesion properties of the pneumatic tire.

Suspension types vary in cost, complexity, and geometry but there exists a set of parameters that are common to all. These parameters define the wheel orientation to the sprung mass and the suspensions ability to control the position of the sprung mass and wheel relative to the road. These parameters are listed below and shown on Figure 2-6:

- ·camber;
- ·caster;
- ·kingpin offset or scrub radius;
- ·anti-dive;
- ·anti-squat; and
- ·anti-roll.

A brief explanation of how these parameters affect vehicle handling is given.

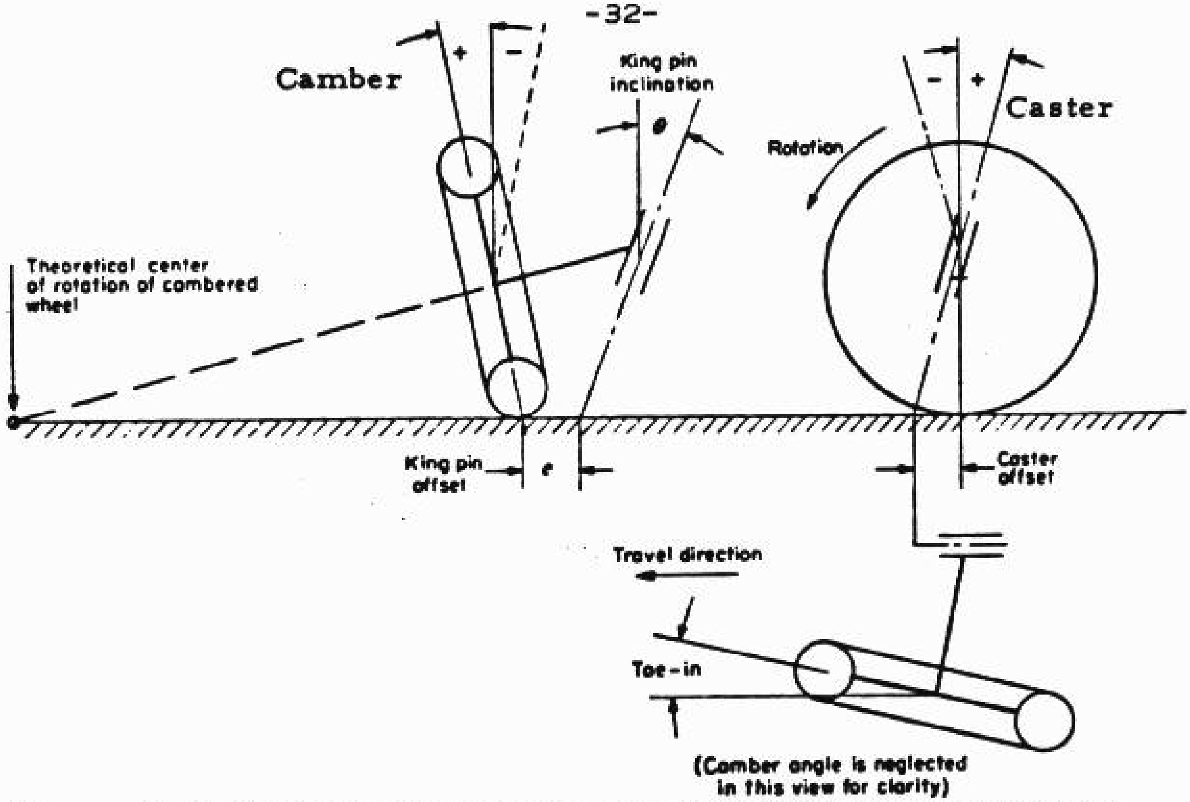


Figure 2-6: Vehicle suspension characterization parameters

Camber

Camber has a significant effect on the force generating capabilities of the pneumatic tire. A wheel is cambered when the rolling or wheel plane is inclined to the vertical with respect to ground, refer to Figure 2-6. Positive camber is defined as the inclination of the top of the rolling plane away from the sprung mass. Camber can enhance or detract the cornering capabilities of the tire depending on the sign of the camber angle and the direction the vehicle is turning. A cambered wheel develops what is known as camber thrust. Camber thrust was first used to hold wagon wheels on their axles. If a wheel is cambered in the right direction some of the normal load (as described by the tire axis system) will be redirected into a lateral force that pushes the wheel onto the axle. In automotive applications, if the outside tire of a cornering vehicle has positive camber then the normal load is redirected into lateral force in a direction which detracts from the total cornering force the tire can generate. This condition can be pictured as the top of the tire leaning away from the center of the turn. However, if the same wheel has negative camber the normal load has a lateral component with respect to the tire that adds to the total cornering force the tire can generate.

Camber thrust is the primary mechanism that allows a bicycle to corner. The rider leans the bicycle so that the normal load is redirected to create the needed cornering force. Very little of the slip mechanism or steering angle is used to allow a bicycle to corner at high speeds. Although this lends understanding to the effects of camber on cornering, the primary mechanism for cornering in the automotive case is the slip angle.

The effect of camber on the steady-state straight-line motion of a vehicle is small. Although the effects on the adhesion properties of tires during cornering or load-transfer-induced steer effects are significant. If the outside wheel of a cornering vehicle has positive camber, the tire must produce a larger slip angle so it can counteract a given cornering-induced lateral force. This is because a significant portion of the normal load has been directed into a degradation of lateral force. Therefore, if the tire is to produce the same cornering capability, it must have a larger slip angle for a given normal load. This means the understeer characteristic of a vehicle can be changed by altering the camber of the wheel during suspension travel.

The camber angle of an independent suspension changes during jounce and rebound travel. The camber of a static vehicle with an independent suspension is set to minimize the camber angle of the most heavily loaded tire during cornering. Camber is negligible for most solid axle suspensions. Excessive camber will cause tire wear and the reduction of cornering forces in any case.

Caster

Referring to Figure 2-6, caster is the inclination of the steering axis to the vertical in the side view. Positive caster is when the top of the steering axis is leaned rearward. Moments applied about the wheel center causes caster to change as a result of suspension compliance. These moments can be caused by rolling resistance or tractive effort at the wheel/road interface. The primary reason for caster is to increase the self-aligning torque on the tire by creating a caster offset as seen in the figure. The self-aligning torque enhances the directional stability of the vehicle by utilizing the "shopping cart" caster effect. This effect maintains the wheel plane in a heading parallel to the present heading of the total vehicle. This same self-aligning torque increases the amount of steering-wheel fight that the driver must overcome. So there is a trade-off; directional stability is improved by increasing caster but the torque applied by the driver to turn the wheel is increased. Power steering can help in this case but caster angles also have implication on the anti-squat anti-dive parameters of a suspension. This subject will not be treated here. In addition, the self-aligning torque discussed above adds to the mechanisms that create the tire slip angle.

Toe-in Toe-out

Toe is observed in the top view and modifies the steering angle of a wheel. When the leading edge of the wheel is inclined toward the center of the vehicle the wheel is toed-in. Toe-in is used to overcome the tendency of the wheel to toe-out during longitudinal suspension compliance caused by rolling resistance and tractive/braking effort. This "compliance steer" must be considered during suspension design or understeer can be adversely affected by a change in the effective slip angle.

Consider the front suspension; the outside wheel of a cornering vehicle moves up because it is the most heavily loaded. If the wheel toes-out as it moves up, the steer angle must be increased to maintain the same radius turn. The inside wheel will deflect downward during this roll motion. This wheel will toe-in and also require more steer angle to maintain the same turn radius. This steering effect is usually referred to as roll steer. If the back suspension toes-in as the suspension moves up the total roll understeer of the vehicle will increase by changing the "effective slip angle". If the front suspension toes-out as the suspension is deflected upward roll understeer will be increased also.

At design position (for the static vehicle) the front wheels are generally toed-in to provide the driver with "road feel" by increasing directional stability during straight-line motion.

Toe is measured as the distance between the front edges of the left and right wheel subtracted from the distance between the back edges of the the left and right wheel. This difference is generally three to six millimeters. Excessive toe will cause tire wear.

King Pin Angle/Offset

King pin angle and offset are shown in Figure 2-6. King pin angle is the inclination of the steering axis to the vertical in the front view. The steering axis, or king pin axis, is defined by a line drawn between the ball joint centers. The wheel spindle rotates about this line. The distance from the center of the tire contact patch to the point where king pin axis intersects the road plane is called the king pin offset or scrub radius. A small king pin offset reduces steering effort, wheel fight, and brake pull. These reductions are possible by making the moment arm around the steering axis smaller. If the king pin axis were vertical the moment arm around the king pin axis would be the intersection of the spindle with the wheel plane and the king pin axis. Therefore the king pin angle makes the moment arm equal the scrub radius and not the spindle length.

The king pin inclination as shown in Figure 2-6 forces the vehicle to be lifted as the wheel is turned about the steering axis. This can be pictured if the path of the spindle tip is traced as it rotates around the steering axis. As the end of the spindle traverses this path, it moves downward and lifts the vehicle. In doing this the vehicles own weight is used as a centering spring on the steering system. This works in a similar fashion to a pendulum seeking the lowest position. As the wheel is turned a torque is produced around the steering axis forcing it to return to the equilibrium position. So the king pin inclination improves the stability of straight line handling but also increases steering effort.

Anti-Squat

Anti-squat reduces the amount that the sprung mass sinks in the back and raises in the front during acceleration. When this condition occurs it is referred to as "power squat". This pitch motion of the sprung mass is a result of the weight transfer from the front to the back. By proper selection of the back suspension geometry anti-squat forces can be generated, refer to Figure 2-7. For any given suspension the trajectory of the back wheel can be defined. Once this trajectory is known in the side view the imaginary link shown in the figure is prescribed. This link is called the "side view swing arm", SVSA. At any given wheel position we can further define point A as the instant center of the back wheel. The tractive force acting on the tire at the tire road interface causes a moment about the wheel center. This moment is reacted through the SVSA and causes an upward force on the sprung mass at point A. The force at point A counteracts some of the weight transfer caused by the tractive effort. By judicious selection of point A the amount of anti squat can be controlled.

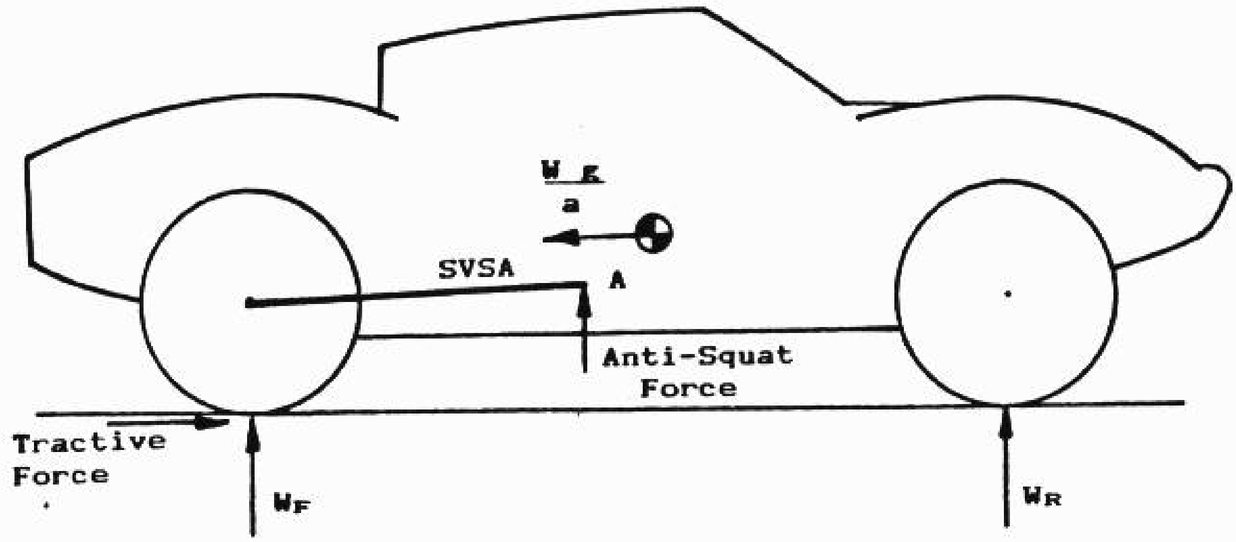


Figure 2-7: Schematic of anti-squat geometry

Ride and handling must be considered when applying the geometry for the desired amount of anti-squat. If the location of point A is selected without this consideration roll-oversteer may be designed into the suspension. This can happen because the location of point A can affect the toe characteristics of some suspensions. Refer to the subsection on toe-in and toe-out, above. Anti-squat values of over 100 percent are possible. High values of this parameter can be dangerous because the vertical action of the suspension is reduced; reducing the suspension deflection under tractive acceleration can cause a condition known as "power hop" [18].

Anti-Dive

Anti-dive reduces the amount that the sprung mass sinks in the front and lifts in the back during deceleration or braking. When this condition occurs it is referred to as "brake dive".

This pitch motion of the sprung mass is a result of the weight transfer from the back to the front. By proper selection of the front (or back) suspension geometry anti-dive forces can be generated, refer to Figure 2-8. For any given suspension the trajectory of the front wheel can be defined. Once this trajectory is known in the side view the imaginary link shown in the figure is prescribed. This link is called the "side view swing arm", SVSA. At any given wheel position we can further define point C as the instant center of the front wheel. The braking force acting on the tire at the tire road interface causes a moment about the wheel center. This moment is reacted through the SVSA and causes an upward force on the sprung mass at point C. The force at point C counteracts some of the weight transfer caused by the braking effort. By judicious selection of point C the amount of anti-dive can be controlled.

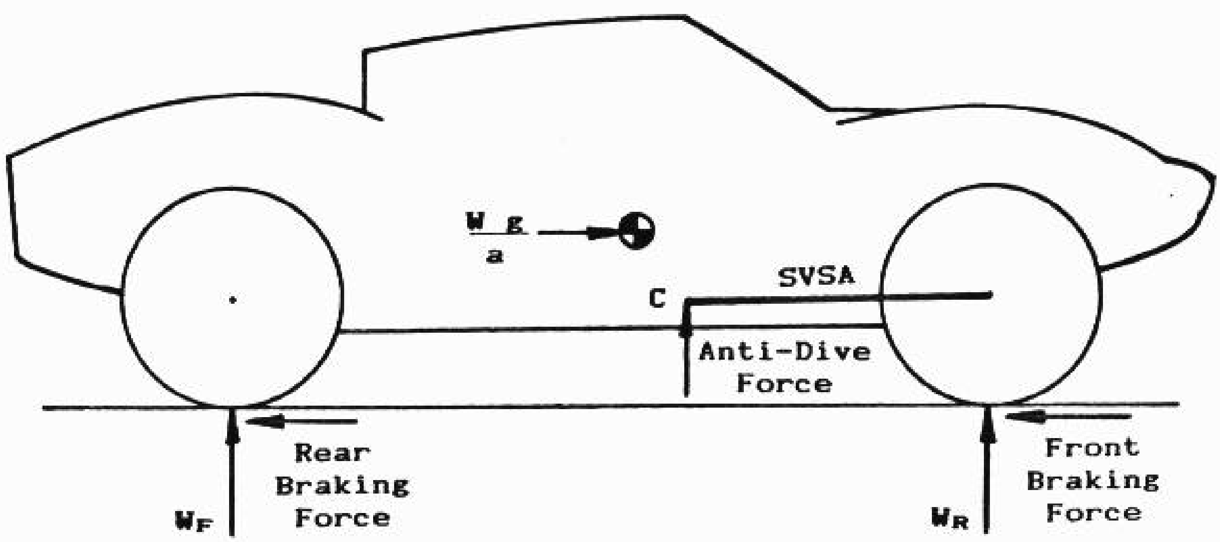


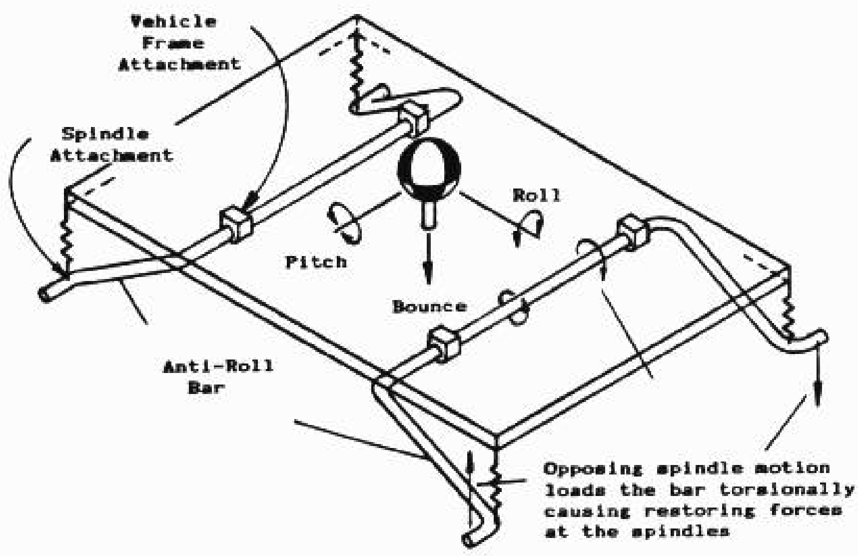
Figure 2-8: Schematic of anti-dive geometry

The maximum amount of anti-dive desirable is approximately fifty percent for the following reasons:

- "flat stops" are undesirable because they reduce the feed back to the driver during deceleration;
- large anti-dive forces could remove some of the suspension action and cause severe shock loads;
- the caster angles necessary for 100 percent anti-dive can increase steering wheel fight;
- if high anti-dive forces are generated in the back, oversteer problems may result; and
- large anti-dive forces can limit the action of the suspension during braking and result in brake hop [18].

Anti-Roll

Anti-roll is an after-the-fact, add-on feature to suspension design. The anti-roll feature is used to control the rolling motion of the sprung mass during cornering. This rolling of the sprung mass affects "roll couple" and "roll steer", explained below. Anti-roll is added to a vehicle through the use of an "anti-roll bar" (also called a stabilizer bar or anti-sway bar , see Figure 2-9). The anti-roll bar is a torsional spring that is attached to the vehicle frame and wheel spindles. This spring generates vertical forces at the spindles as a function of the difference between the suspension travel from one side to the other. When the vehicle pitches or bounces the difference is zero and no forces are generated by the roll bar. If the vehicle rolls or tramps [18] there is relative wheel motion from side to side; the torsional spring is loaded and adds to the total roll rate of the suspension. The roll rate of the suspension tends to restore the sprung mass to an equilibrium position. The amount



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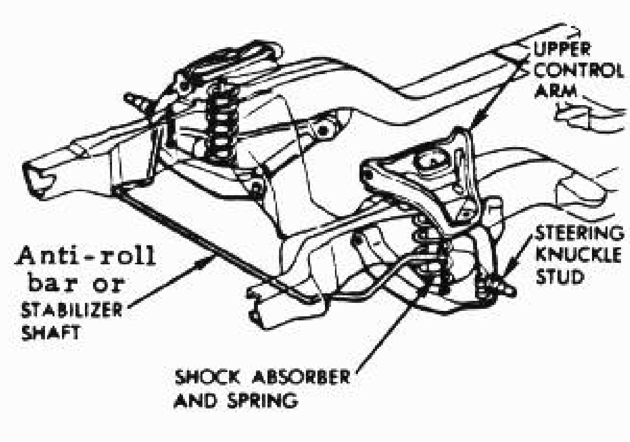

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Figure 2-9: Anti-roll bar a) model b) actual application of roll stiffness that can be added to a suspension is controlled by the diameter of the roll bar.

Roll couple is an inertial moment that is applied to the sprung mass which causes the suspension to be deflected and the vehicle to roll. This deflection causes roll steer effects. Roll steer effects occur if the suspended wheel tends to toe-in or toe-out during suspension travel. The influence of these steer effects are presented in the subsection on toe-in and toe-out.

Because the sprung mass is a rigid body the roll couple is distributed from front to rear based on the relative roll stiffnesses from front to rear. This distribution of roll couple is simply called "roll couple distribution". The roll couple distribution from front to rear has a major impact on the understeer/oversteer characteristics of a vehicle.

Lateral load transfer (or the weight transfer from one side of the vehicle to the other) is a function of only lateral acceleration, center-of-gravity height, and track width. The amount of lateral load transfer reacted at the front or rear can be different based on the roll stiffnesses at the front and rear. The amount of normal load increase on the outside front or outside rear tire during cornering is also proportional to the difference between the front and rear roll stiffnesses. Therefore, if the front has a higher roll stiffness more of the roll couple acting on the sprung mass is applied to the front suspension. With the increase of normal load on the outside front tire being greater than at the outside rear, the front tire must generate a larger slip angle for a given lateral acceleration. With the front slip angle growing faster than the rear, the vehicle will understeer for moderate acceleration levels (at larger accelerate levels "normal load roll off" must be considered). For the same reason the placement of an anti-roll bar in the rear could increase the rear roll stiffness sufficiently to cause oversteer.

Handling kits for modern cars would include a stiffer front anti-roll bar and add an anti-roll bar to the rear to control sprung mass roll and excessive understeer. This makes the vehicle's lateral response much quicker and so the vehicle handles better.

2.6 Summary of Suspension Characteristics

The general parameters that can be used to to characterize any suspension are discussed in the previous section. Generally the motivations for using a given suspension are from many considerations like; how the vehicle is intended to ride and handle, cost of manufacturing the components, cost of assembly and installation, the intended image of the vehicle (marketing), the intended final selling price, etc. Therefore when comparing suspension types all motivations must be considered. In this chapter some basic terms needed to understand vehicle handling have been covered. The most important of these terms is the understeer characteristic because more than any other parameter it indicates the handling and stability of the automotive vehicle.