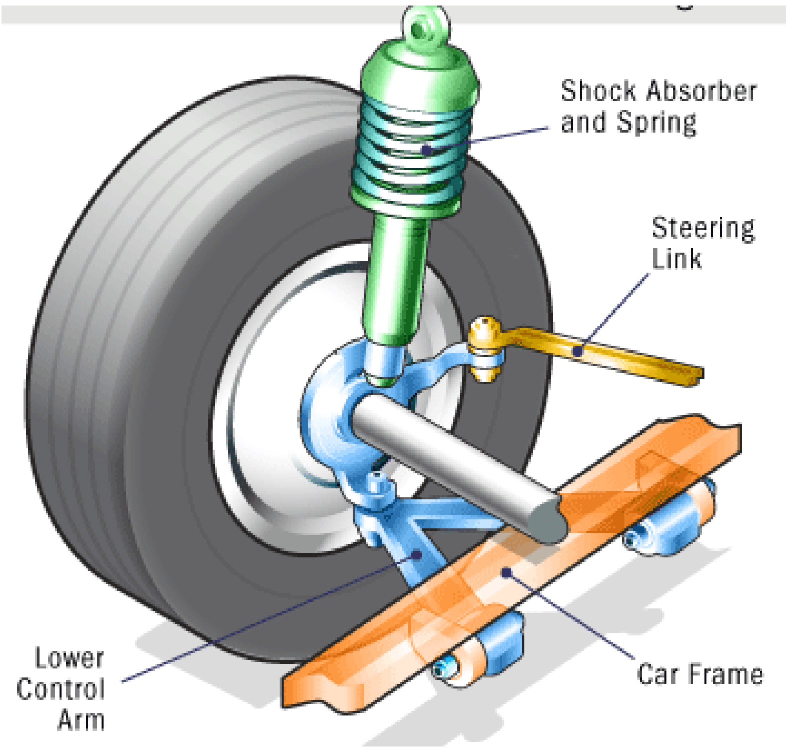
Session-3

Thursday, 17-12-2009;4.00 PM-5.30 PM

Vehicle Suspension-Kinematics and Compliance

- Wheel Alignment
- Wheel Kinematics and Compliance
- Steering Performance Criteria for Handling

Wheel Geometry

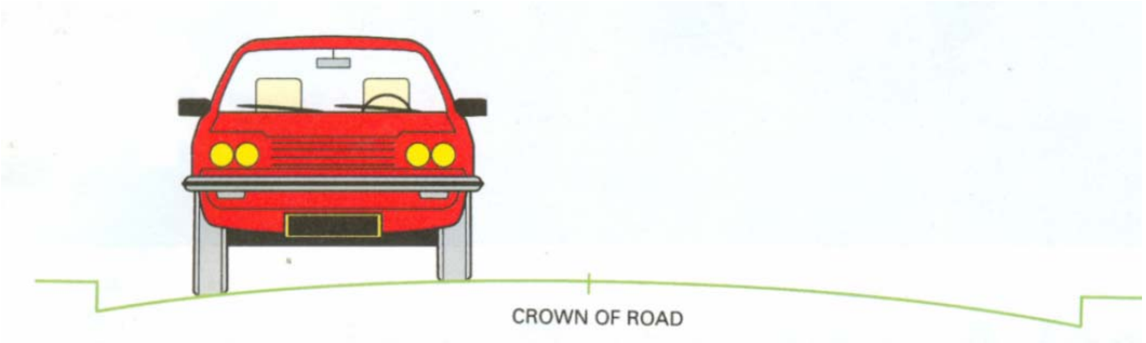


Wheel Alignment

- A wheel Alignment is the adjustment of the suspension and steering to ensure proper vehicle handling with minimum tire wear
- A change in alignment angles may result from one or more of the following factors
  - Wear of the steering and suspension components
  - Bent or damaged steering and suspension parts
  - Sagging springs, which can change the ride height of the vehicle and therefore the alignment angles

Alignment- Related Problems

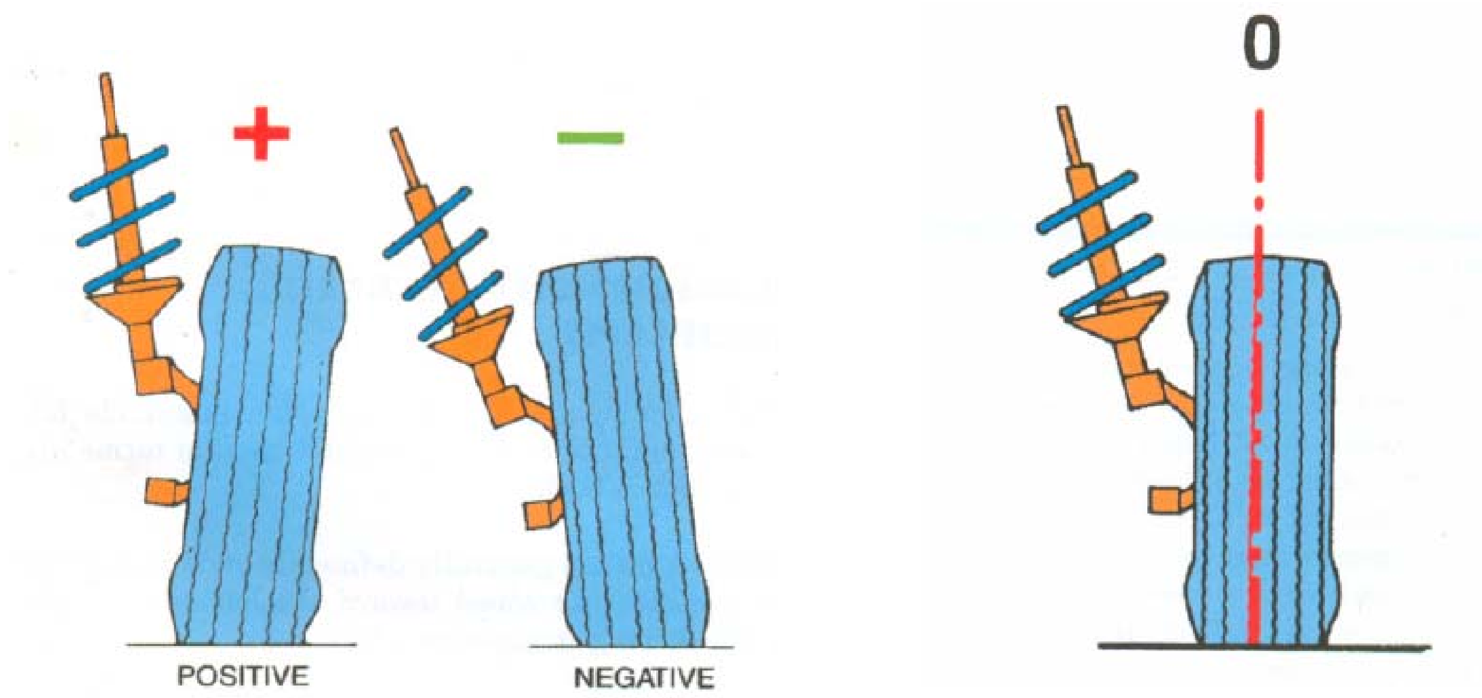
- Pull
  - A pull is generally defined as a definite “tug” on the steering wheel toward the left or the right while driving straight on a level road
- Lead or drift
  - It is a mild pull that does not cause a force on the steering wheel that the driver must counteract. When the vehicle moves toward one side or the other, this is called a lead or drift could be caused by the crown of the road



- Wander
  - A wander is a condition in which almost constant steering wheel corrections by the drive are necessary to maintain a straight-ahead direction on a straight level road

Camber

- Camber is the inward or outward tilt of the wheels from true vertical as viewed from the front or rear of the vehicle

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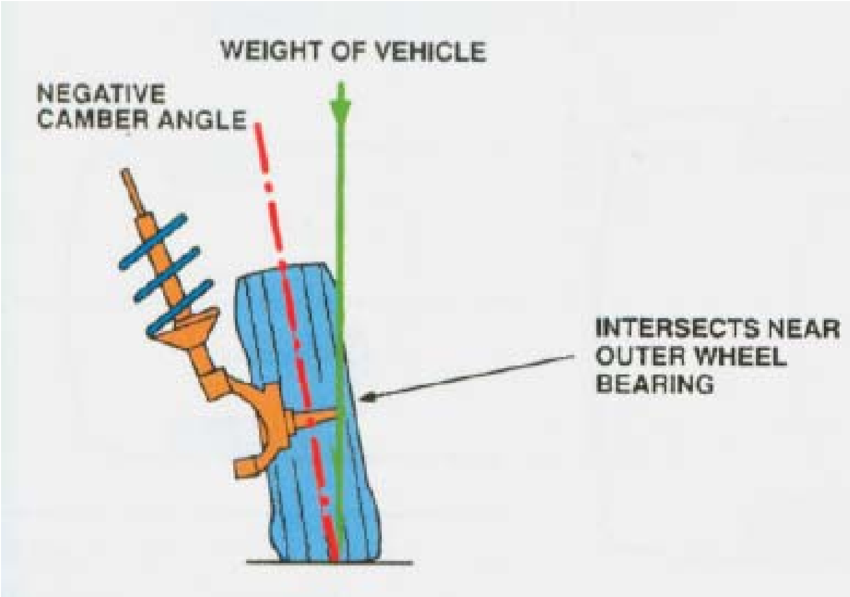
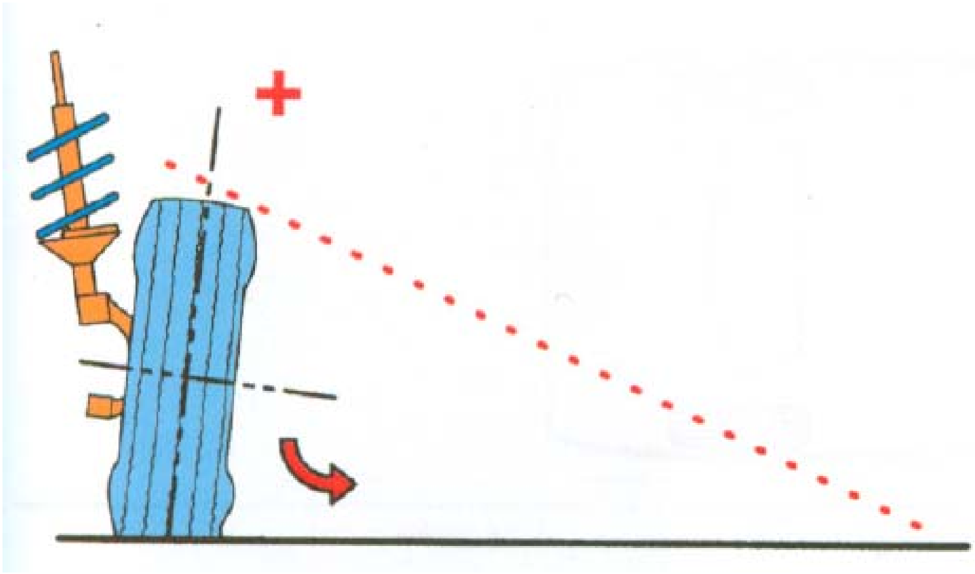
Camber

- Incorrect camber can cause excessive wear on wheel bearings. Many vehicle manufacturers specify positive camber so that the vehicle weight is applied to the larger inner wheel bearing and spindle. As the vehicle is loaded or when the spring sag, camber usually decreases. If camber is kept positive, then the running camber is kept near zero degrees for best tire life



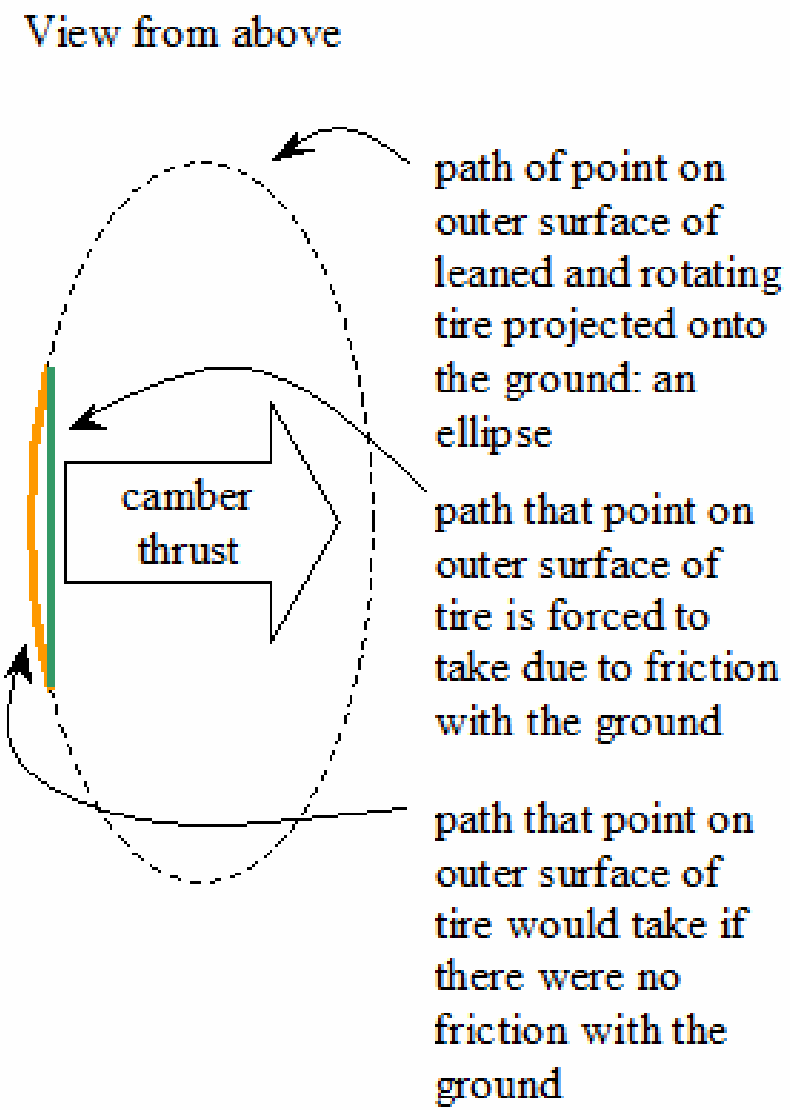
Camber

- Camber can cause pull (Camber Thrust) if it is unequal side to side. The vehicle will pull toward the side with the most positive (or least negative) camber. A difference of more than half a degree from one side to the other will cause the vehicle to pull

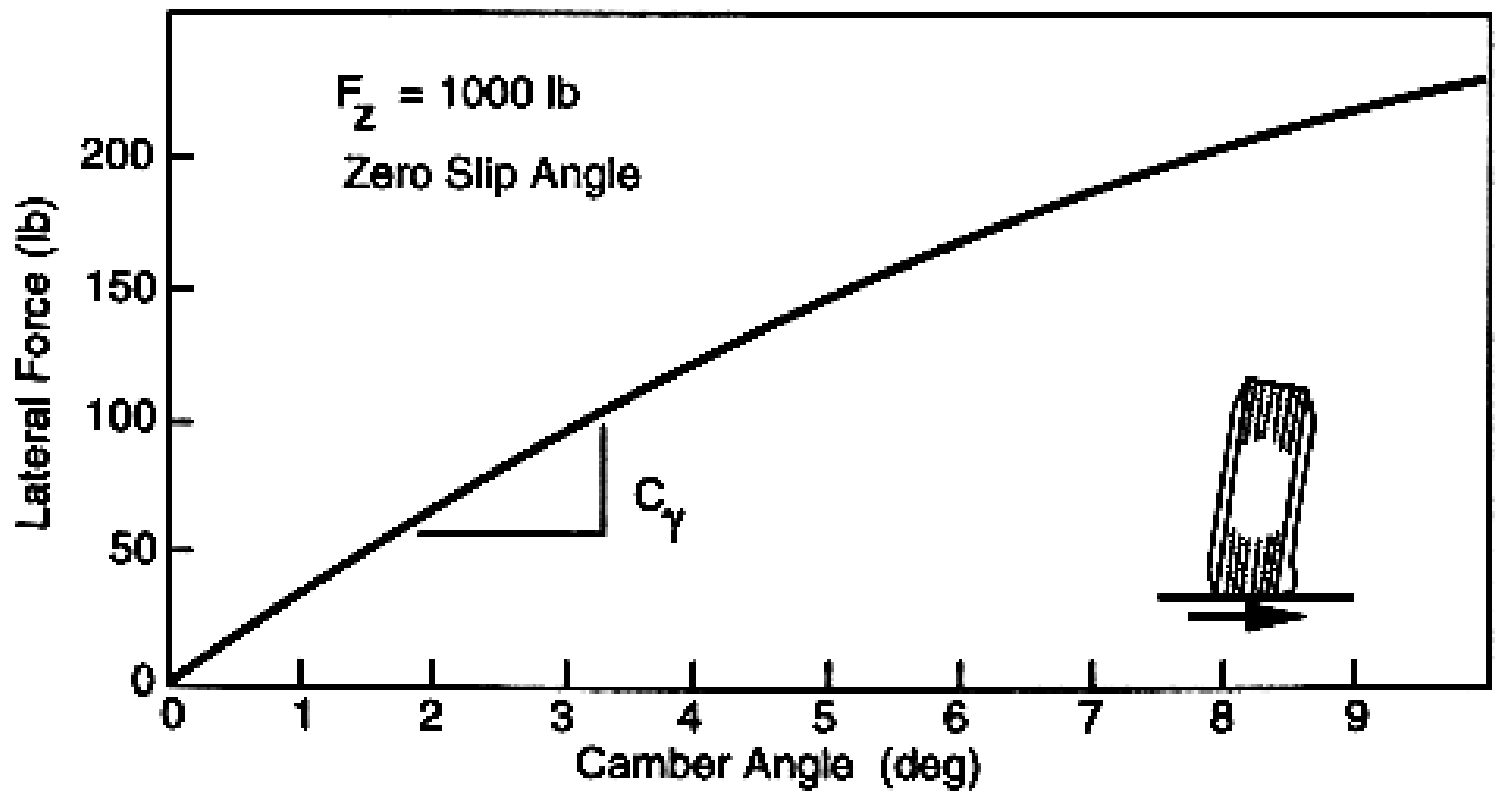


How Camber Produces Camber Thrust

- **Camber thrust** and **camber force** are terms used to describe the force generated perpendicular to the direction of travel of a rolling tire due to its camber angle and finite contact patch. Camber thrust is generated when a point on the outer surface of a leaned and rotating tire, that would normally follow a path that is elliptical when projected onto the ground, is forced to follow a straight path due to friction with the ground. This deviation towards the direction of the lean causes a deformation in the tire tread and carcass that is transmitted to the vehicle as a force in the direction of the lean.

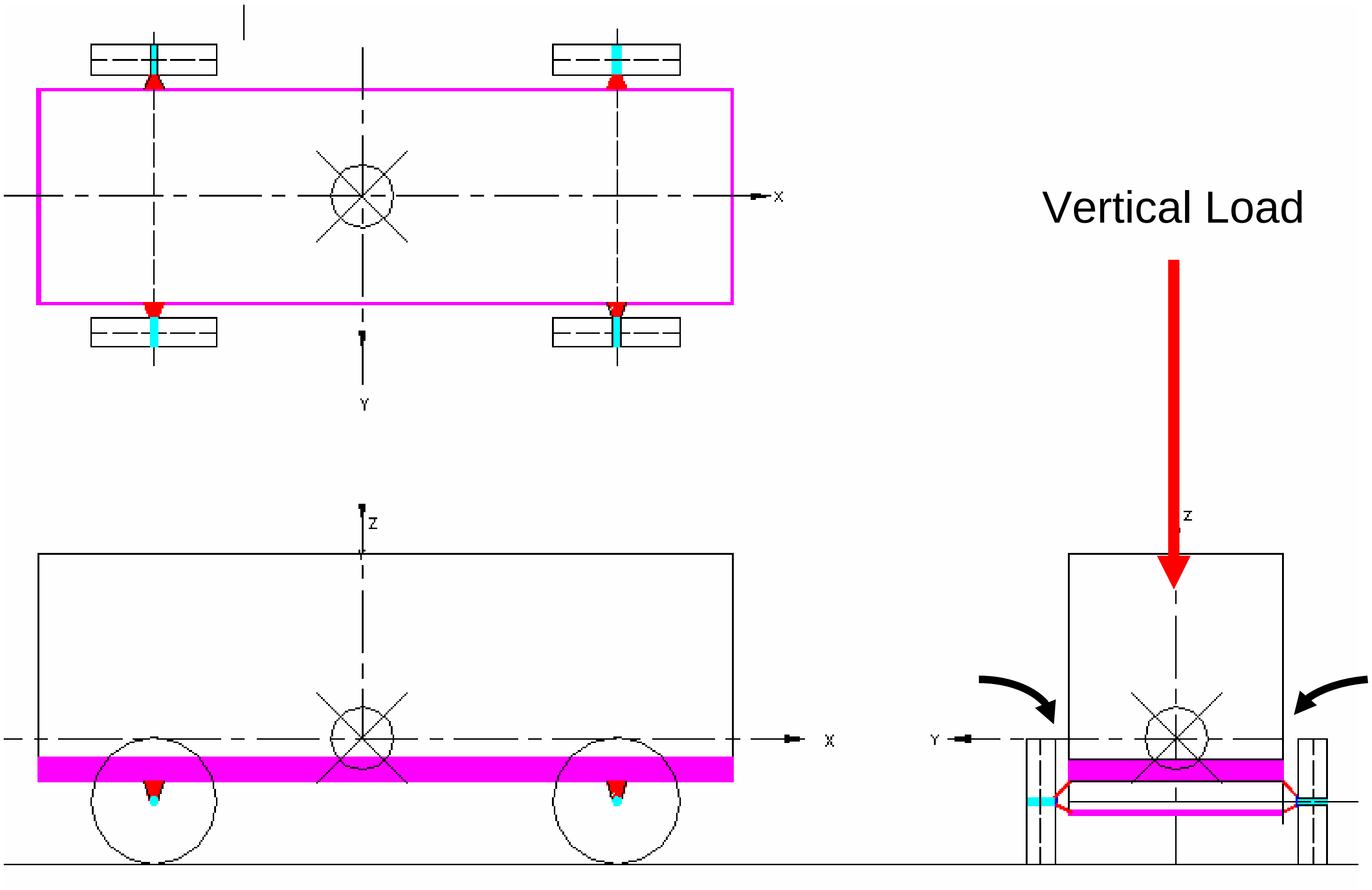


Camber Thrust



- Camber thrust is approximately linearly proportional to camber angle for small angles. Bias-ply tires have been found to generate more camber thrust than radial tires. **Camber stiffness** is a parameter used to describe the camber thrust generated by a tire and it is influenced by inflation pressure and normal load. The net camber thrust is usually in front of the center of the wheel and so generates a **camber torque**.

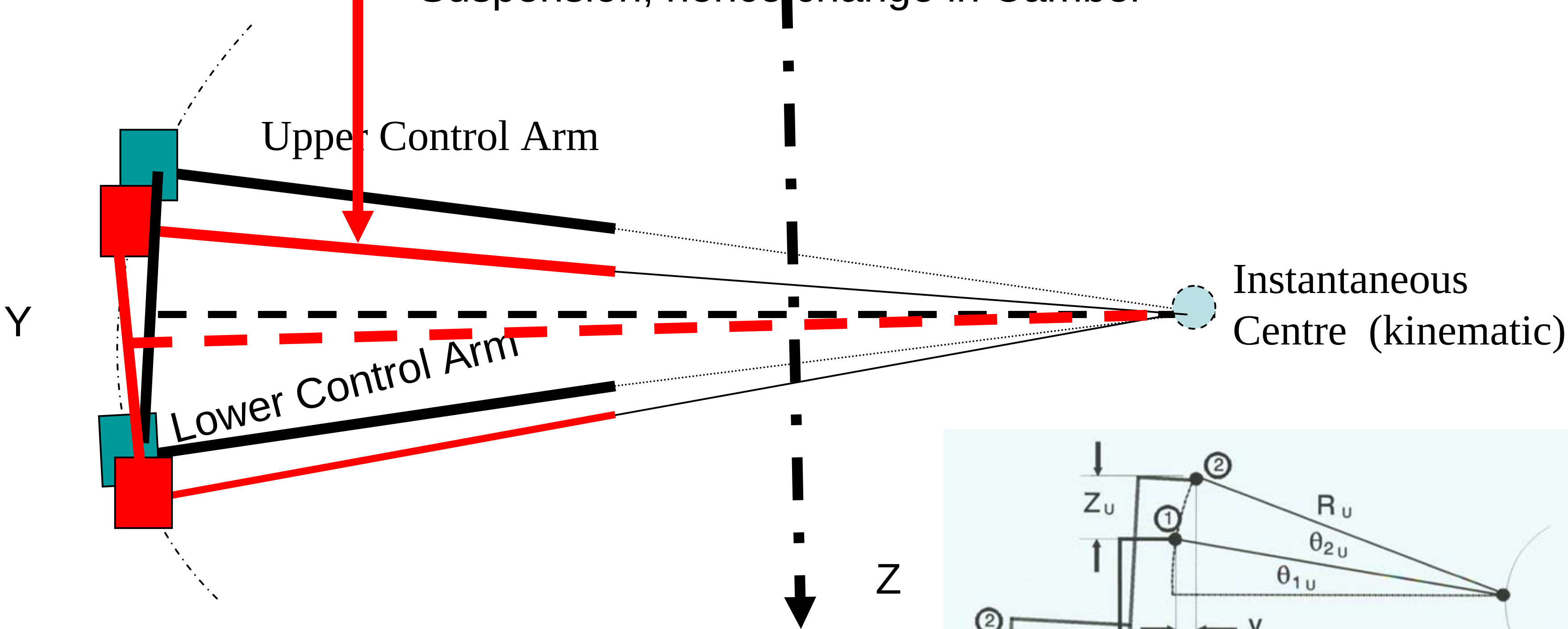
Camber Changes

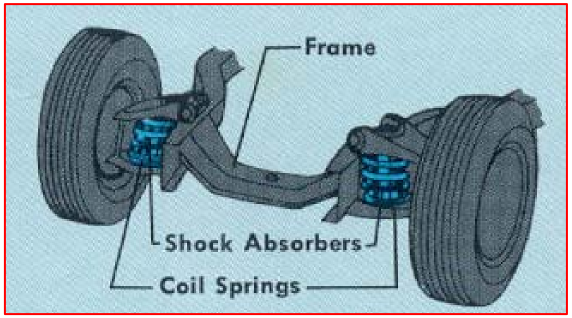


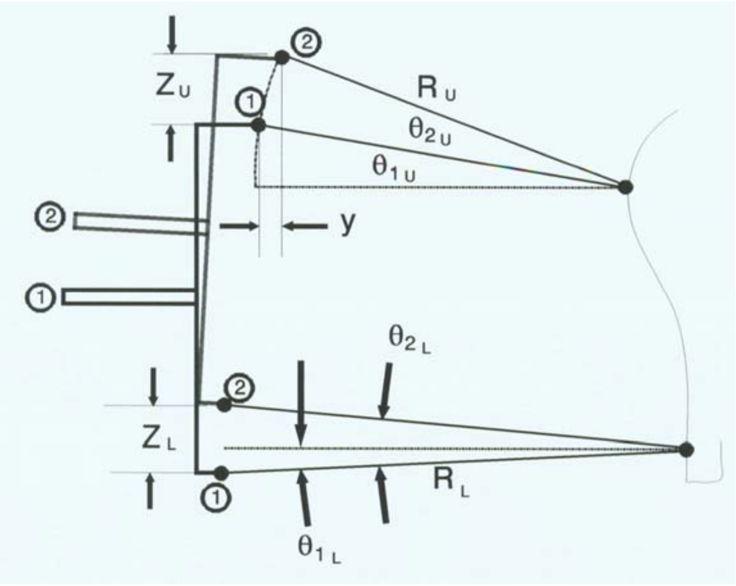
Positive Cambered Wheels move inward when loaded

Camber Change

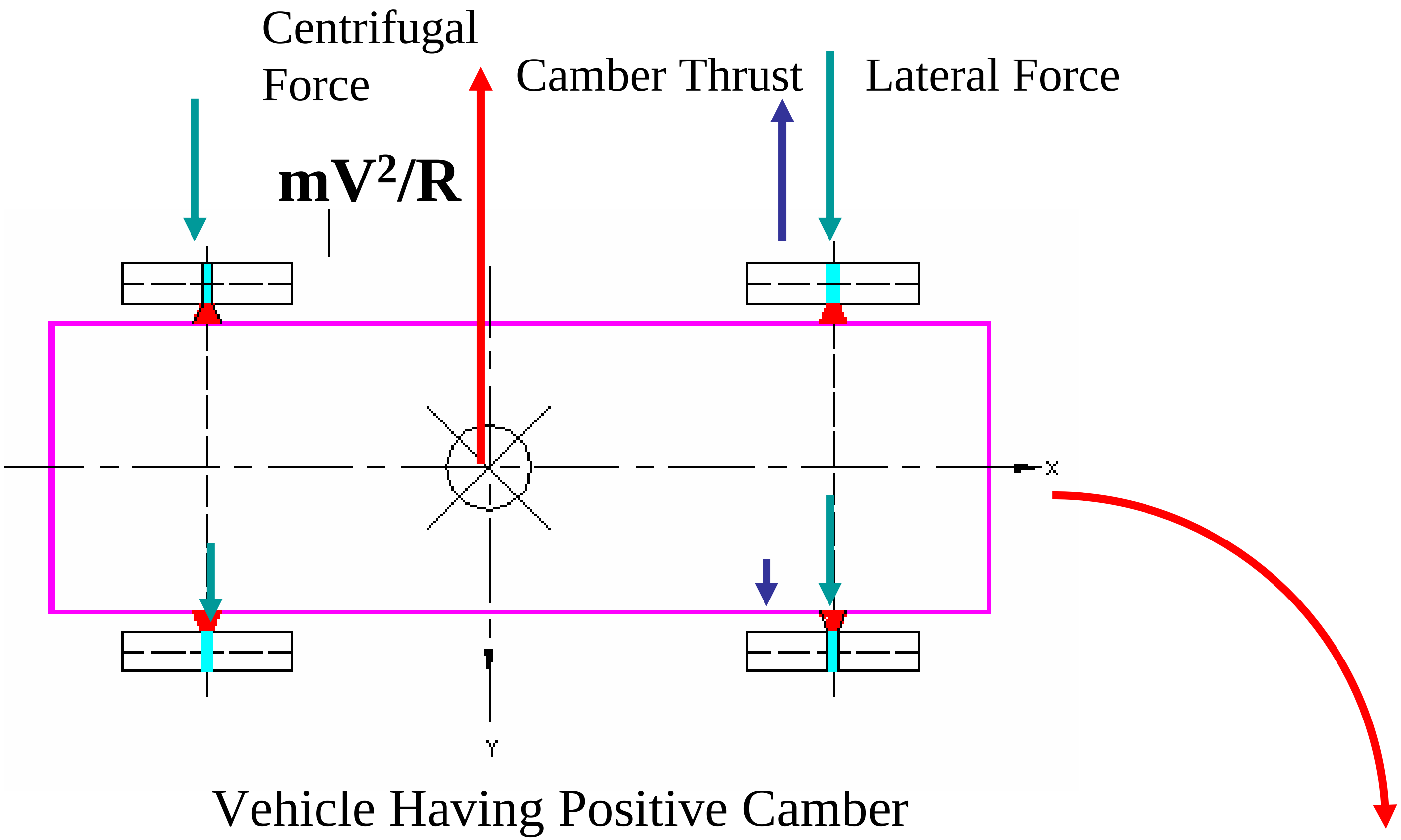
Lateral Load Transfer and compression of Suspension, hence change in Camber







Camber Change Effect in Cornering

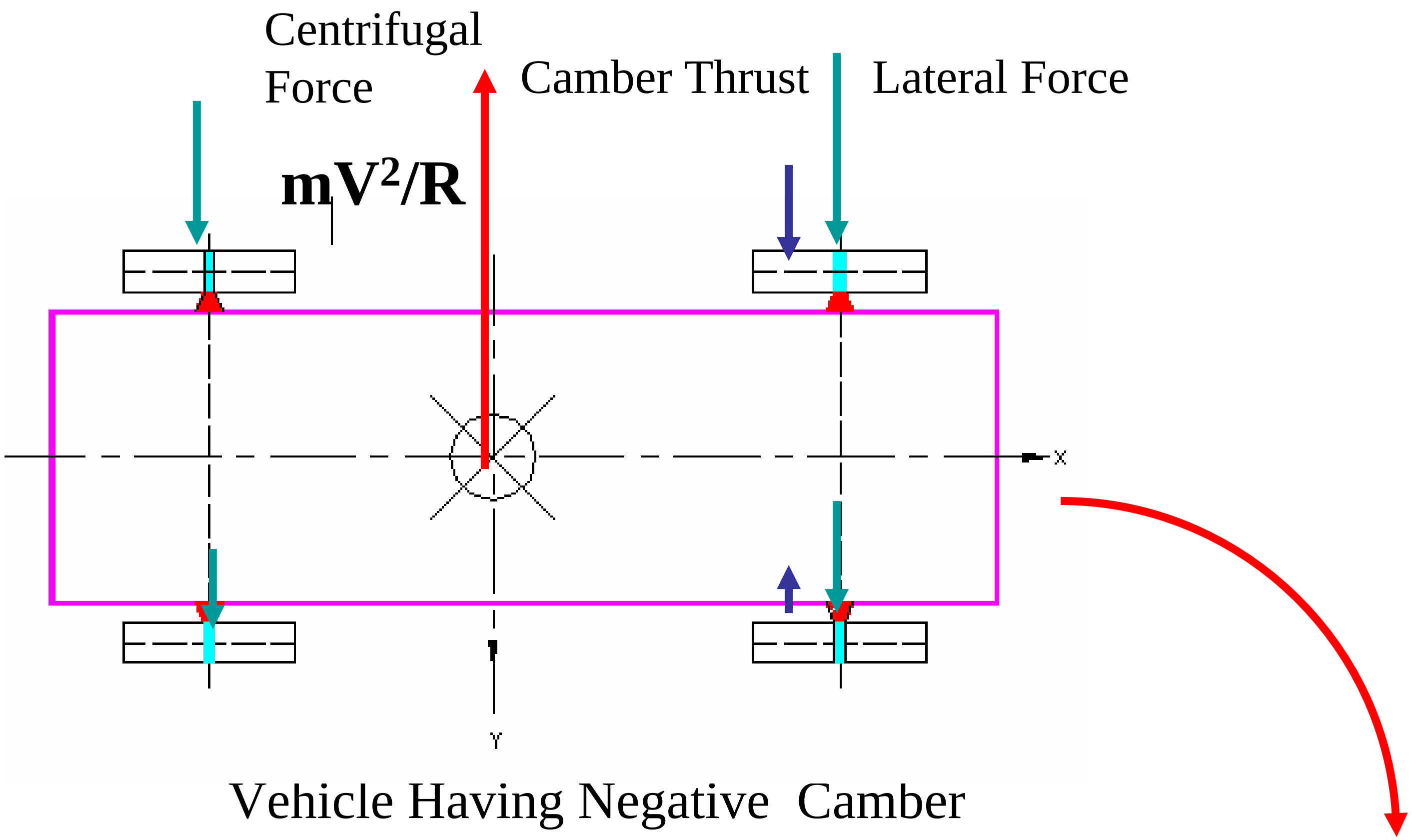


Vehicle Having Positive Camber

Camber Changes Happen

Vehicle Understeers

Camber Change Effect in Cornering

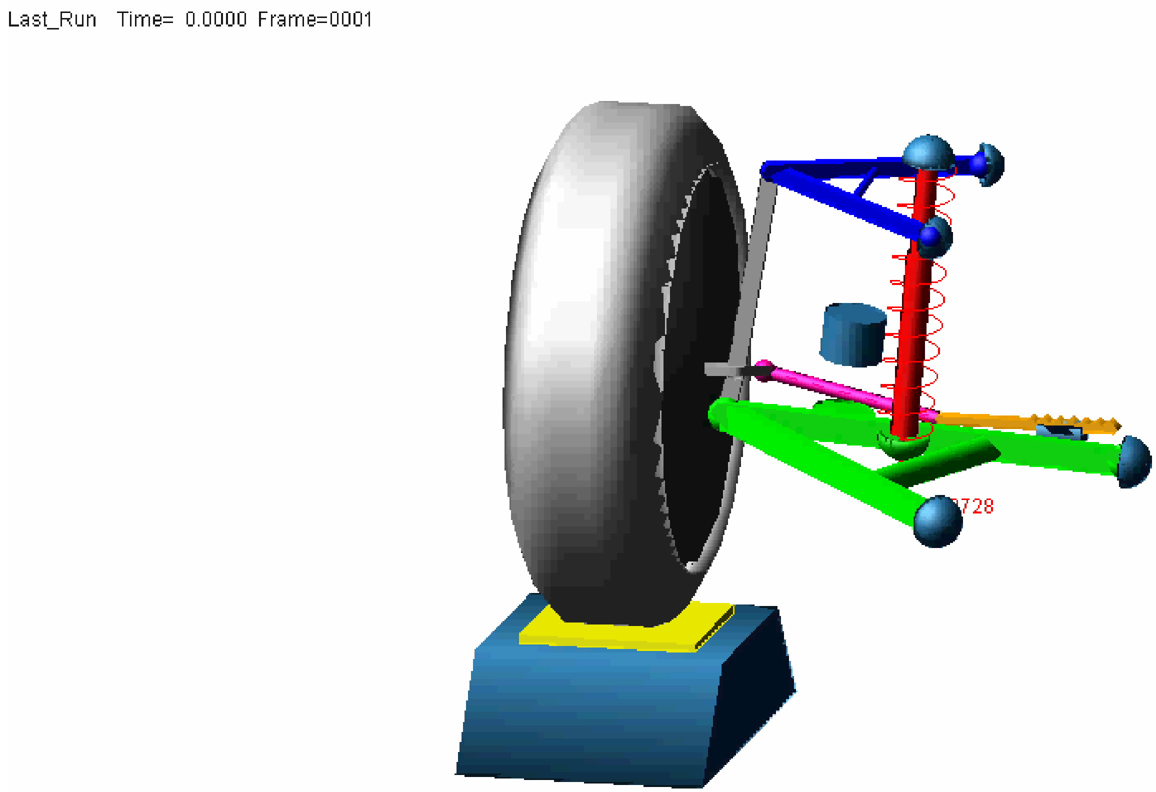


Vehicle Having Negative Camber

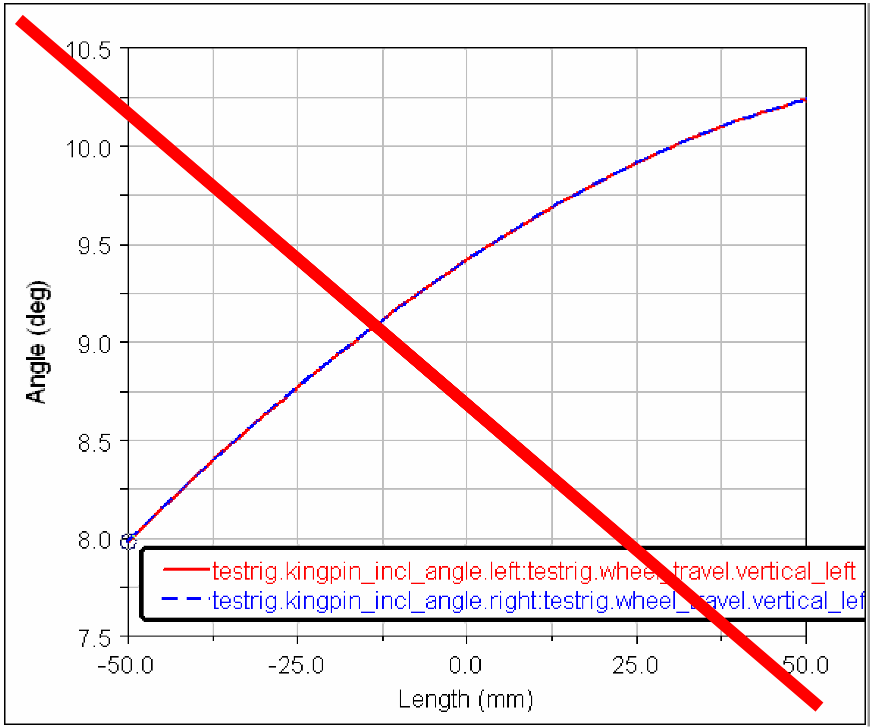
Camber Changes Happen

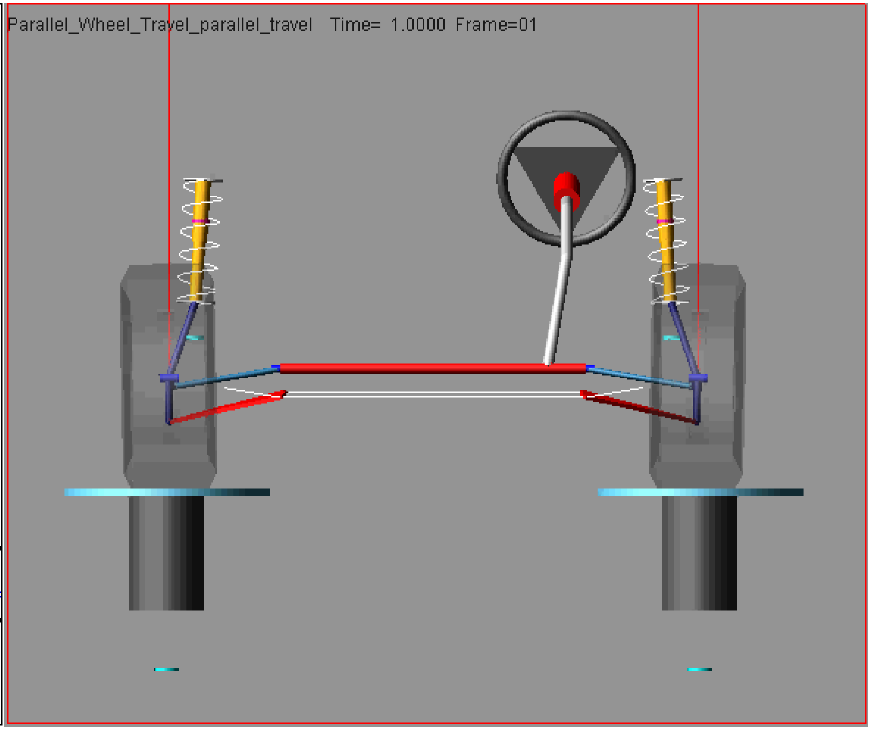
Vehicle Oversteers

Suspension Travel



Parallel Wheel Travel



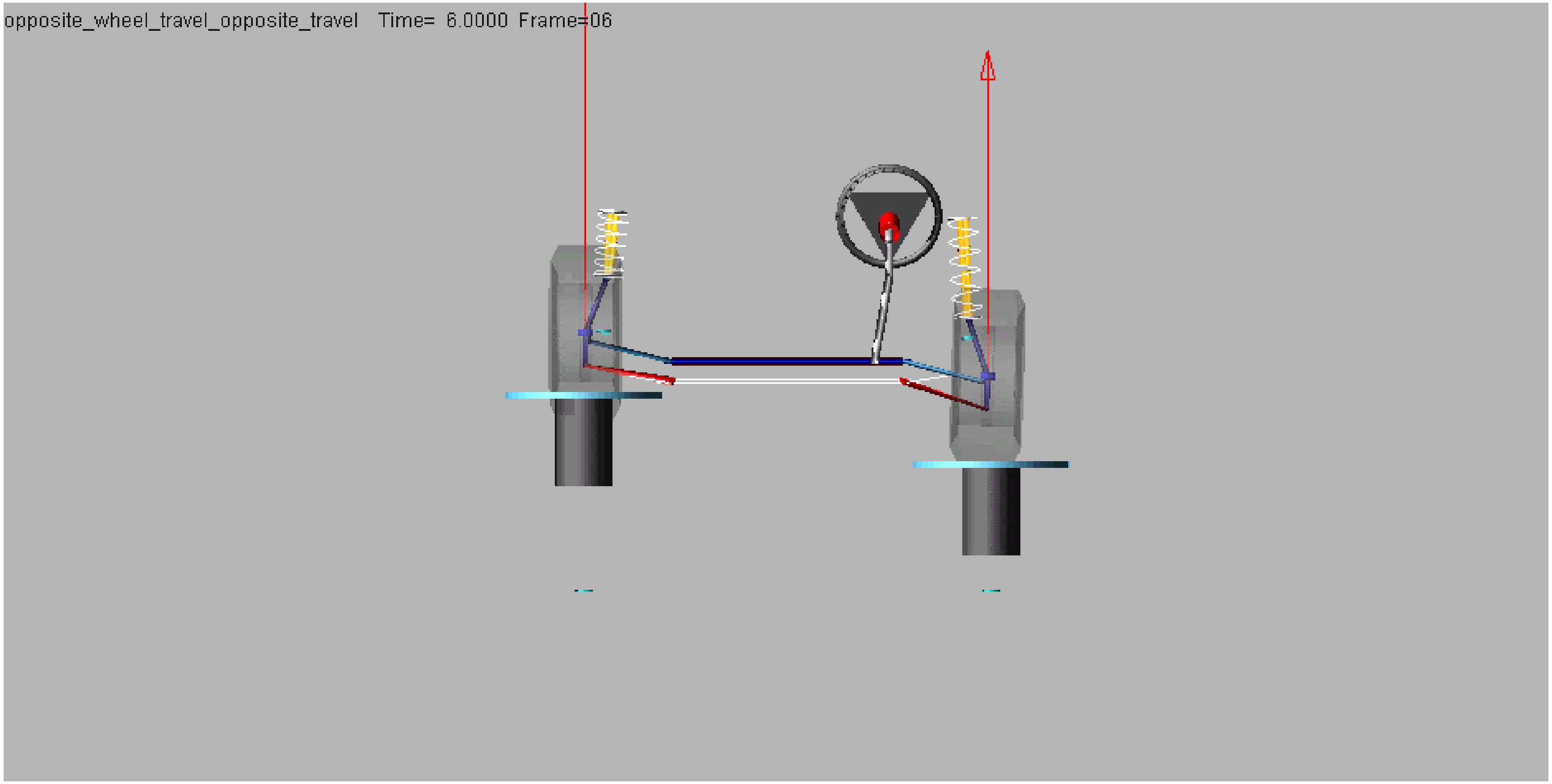


Camber Change

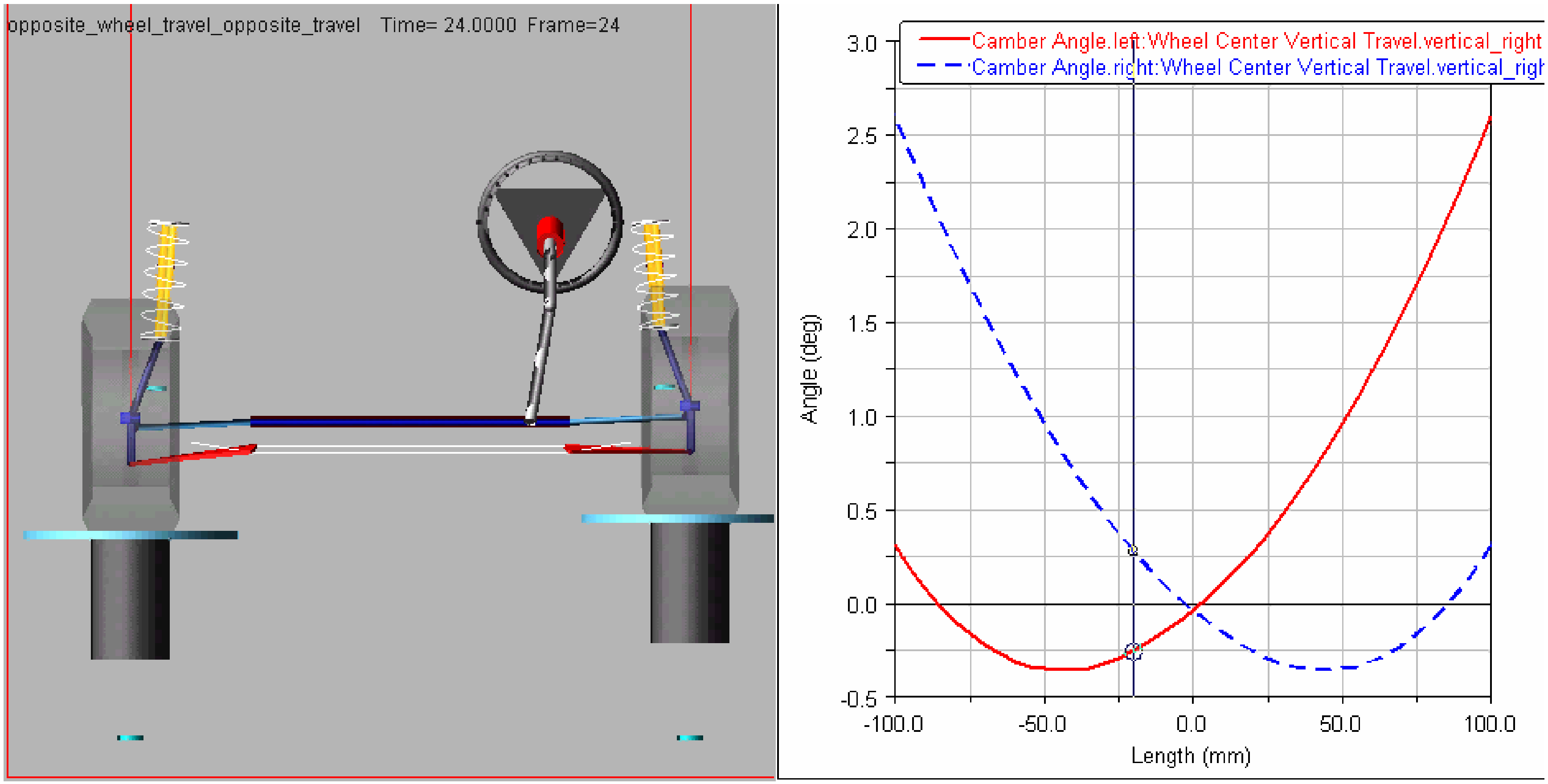
Bump Steer

- **Bump Steer** is the term for the tendency of a wheel to steer as it moves upwards into jounce. It is typically measured in degrees per metre or degrees per foot.
- On modern cars the front of the tire moves outwards, as the suspension is raised, a process known as the front wheels "toeing out". This gives roll understeer. The rear suspension is usually set up to minimise bump steer, where possible.
- A typical value is two degrees per metre, or perhaps more, for the front wheels.
- Excessive bump steer increases tire wear and makes the vehicle more difficult to handle on rough roads.
- Solid axles generally have zero bump steer, but still have roll steer, in most cases. That is, if the wheels move upwards by the same amount, they tend not to steer.
- Roll steer is an important part of the budget used to define a vehicle's understeer, known as a Bundorf analysis.

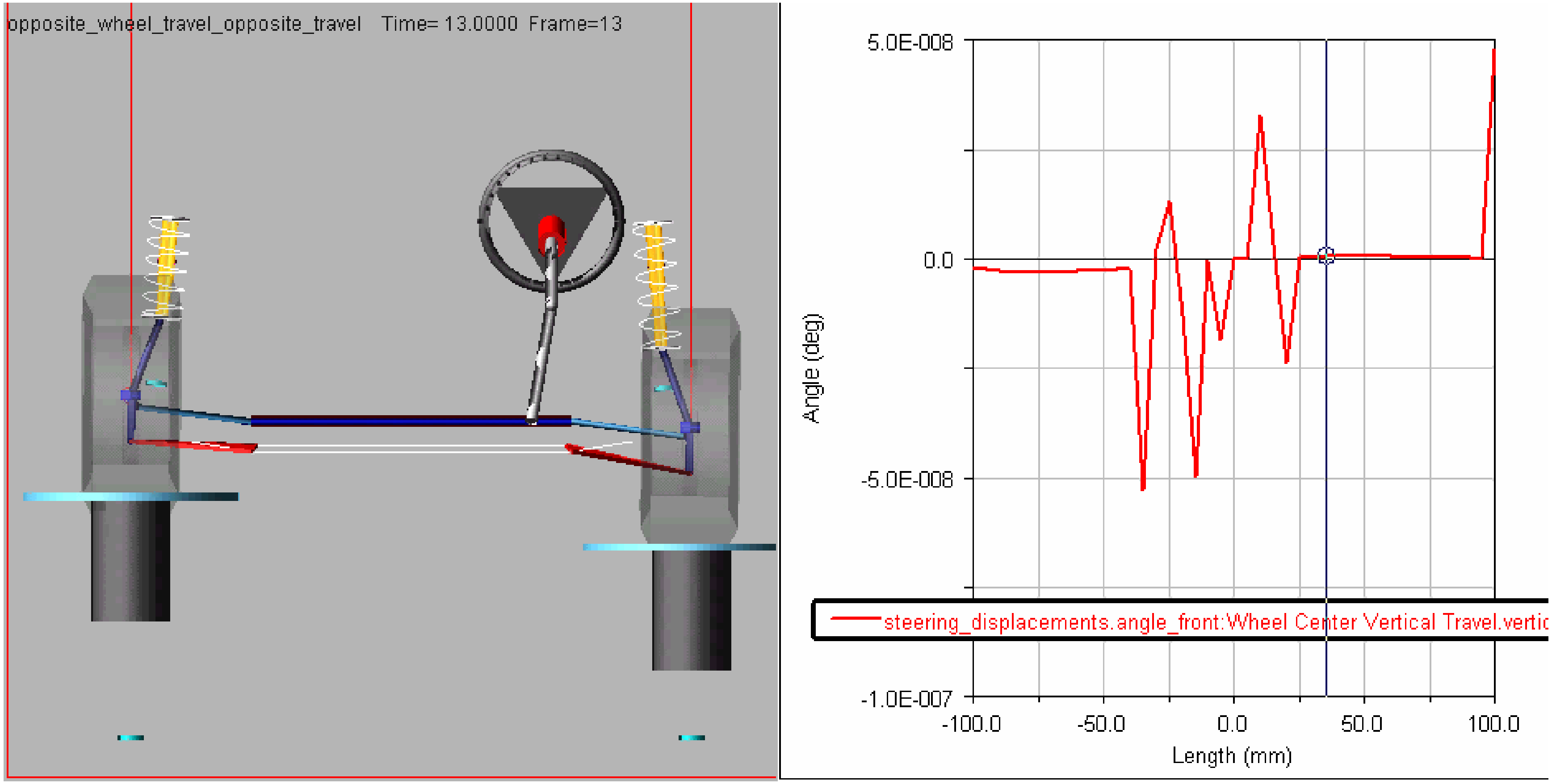
Parallel Opposite Wheel Travel



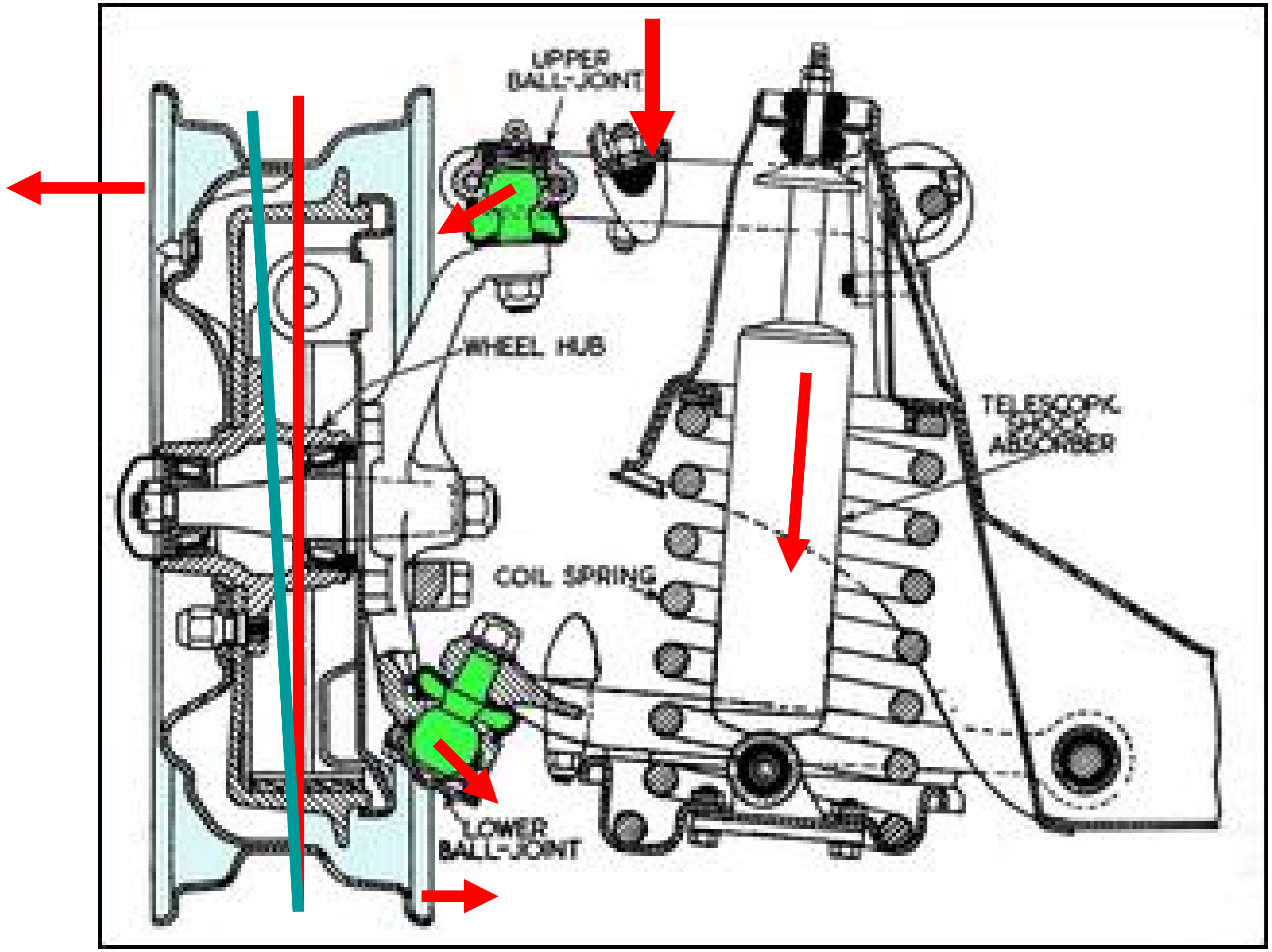
Camber Change



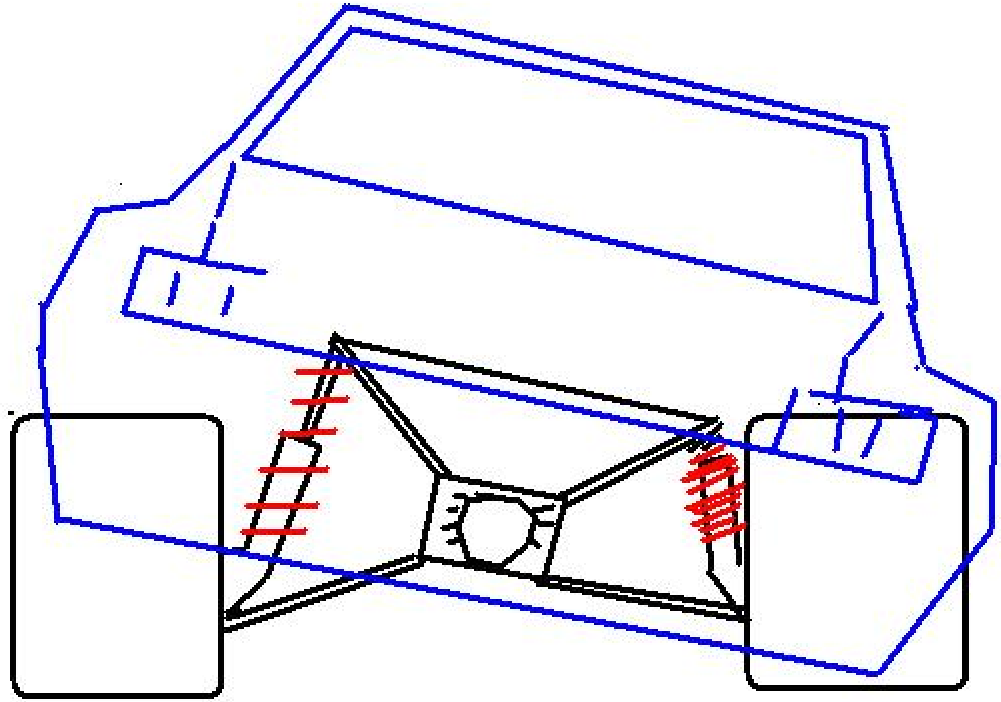
Steering Angle



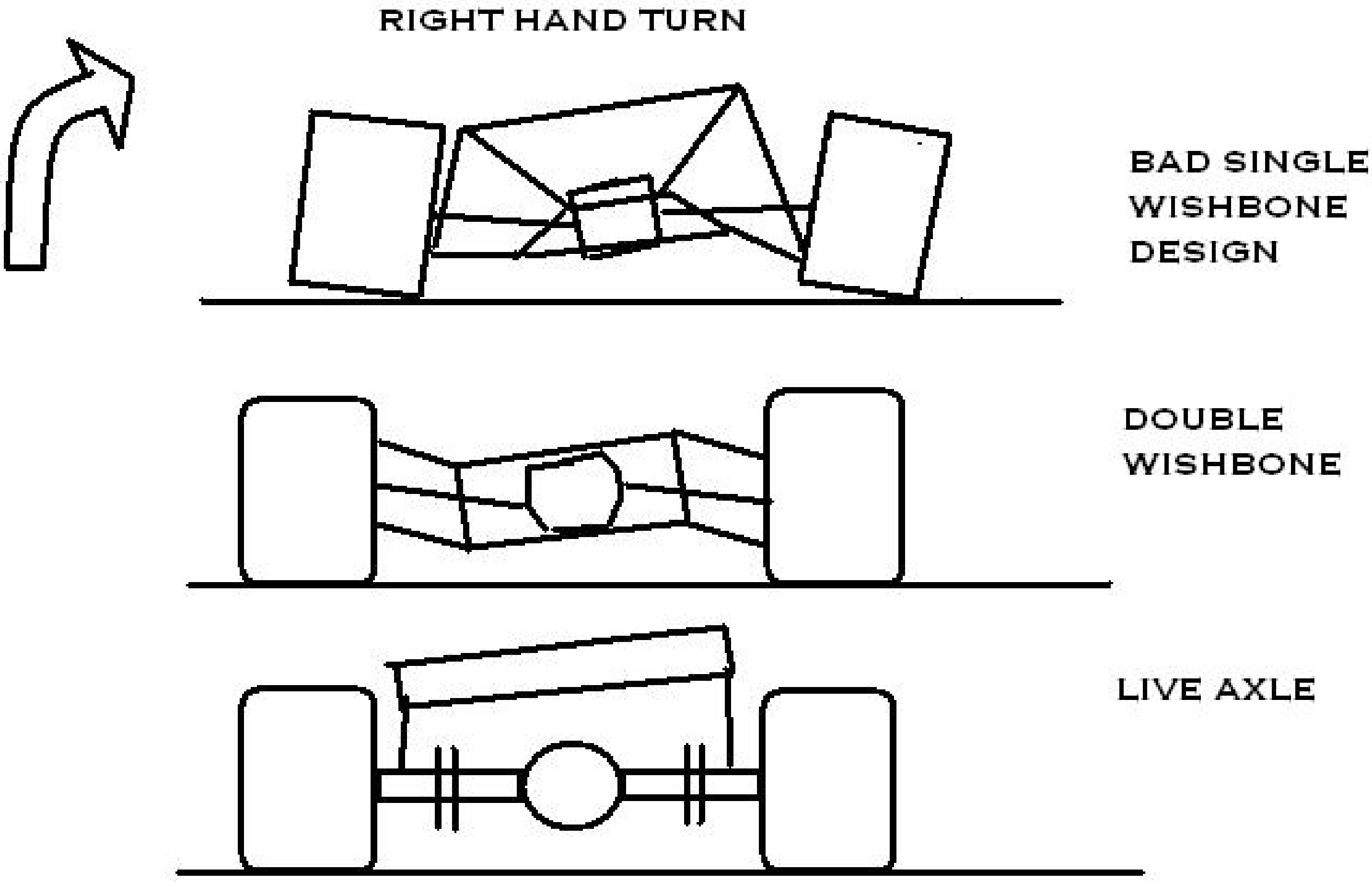
How Camber Changes Happen

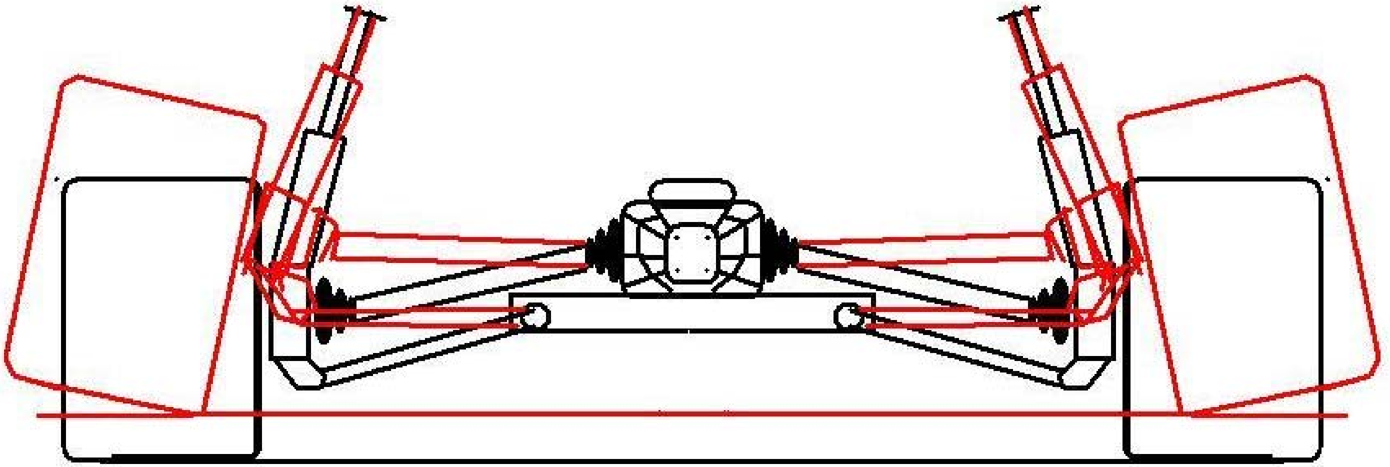


Camber Changes due to the movement of Control arms in Y Direction

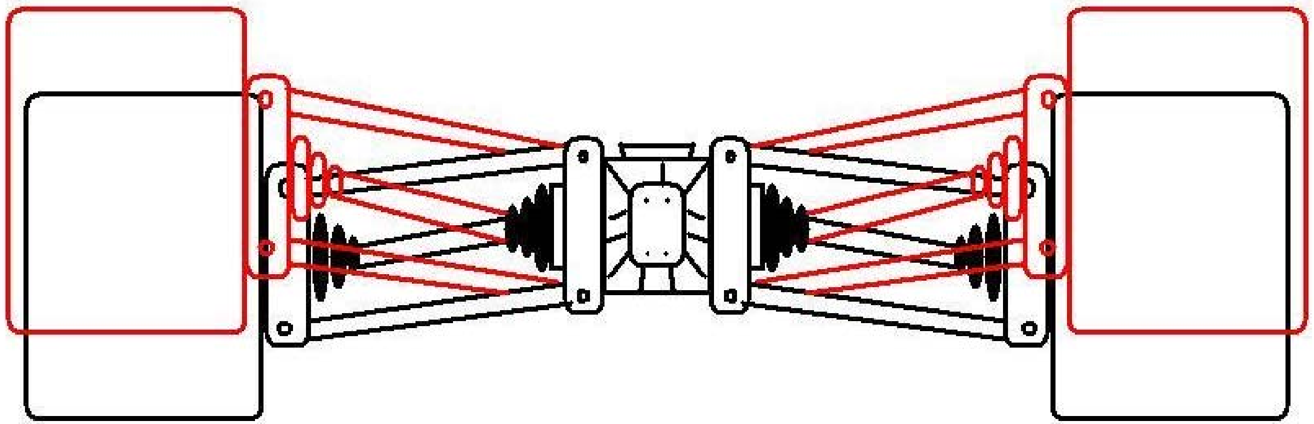


A double wishbone suspension can be made to allow camber change by using un-equal length or diverging/converging arms


RIGHT HAND TURN
BAD SINGLE WISHBONE DESIGN
DOUBLE WISHBONE
LIVE AXLE

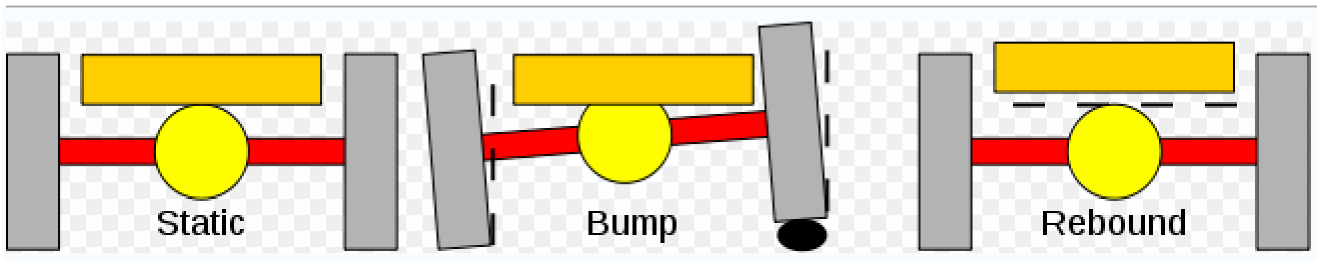


SINGLE WISHBONE SUSPENSION HAS BAD CAMBER CHANGES UNDER SQUAT AND DROOP, RESULTING IN A LOSS OF TYRE FOOTPRINT



UPPER AND LOWER WISHBONES GIVE THE BEST CONTROL ON WHEEL GEOMETRY AND IF DESIGNED RIGHT CAN GIVE POSITIVE AND NEGITIVE CAMBER IN ROLL AND ANTI-SQAUT GEOMETRY ETC ETC (IN PURE SQUAT AND DROOP THE ABOVE SET UP BEHAVES JUST LIKE A LIVE AXLE)

Camber Change in Solid Axle Suspension



Solid axle suspension characteristics: Camber change on bumps, none on rebound, large unsprung weight

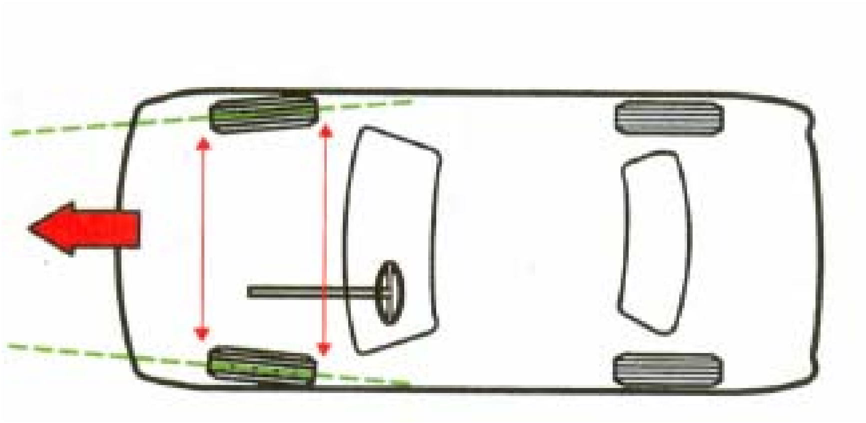
Camber

- Camber is not adjustable on many vehicles
- If camber is adjustable, the change is made by moving the upper or the lower control arm or strut assembly by means of one of the following methods
  - Shims
  - Eccentric Cams
  - Slots
- Camber should be equal on both sides; however, if camber cannot be adjusted exactly equal, make certain that there is more camber on the front of the left side to help compensate for the road crown (half a degree maximum difference) in LHD, opposite for RHD

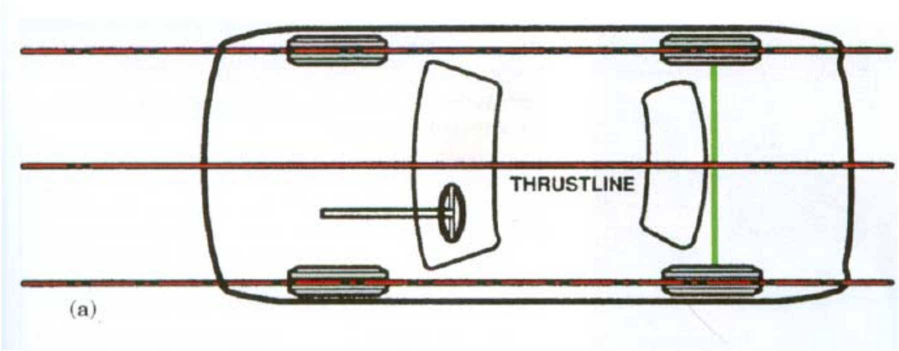
Toe

- Toe is the difference in distance between the front and rear of the tires

Toe- Rear Wheel Drive

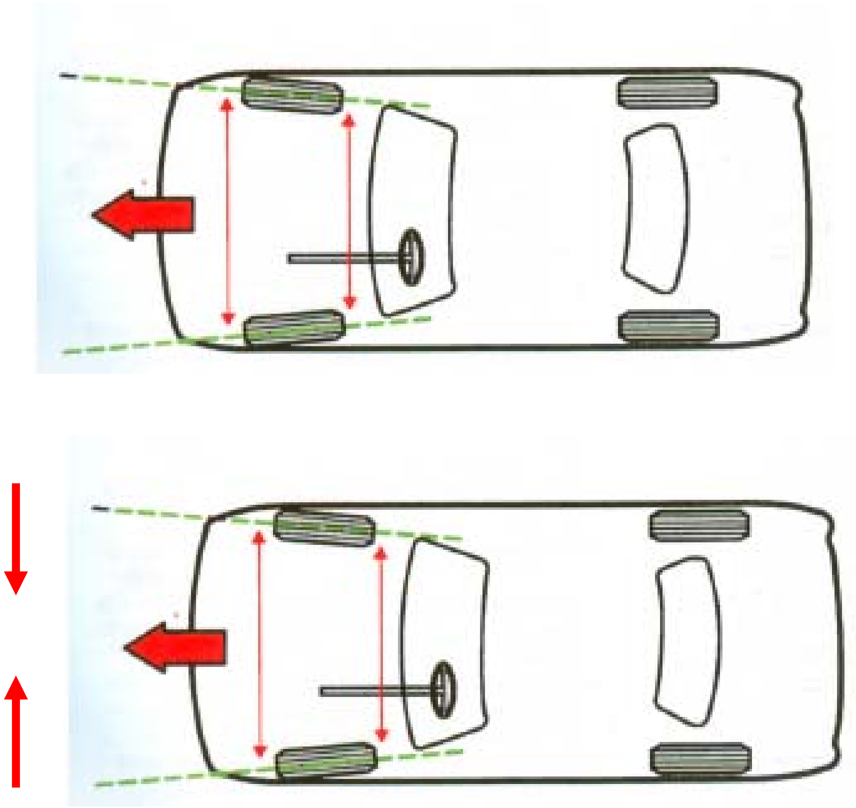


Wheels toe out during running



Toe

- Front wheel Drive



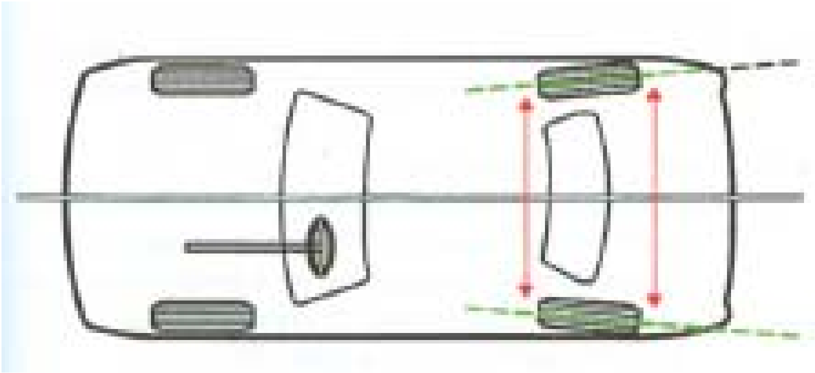
Drive Axles Toe in during running

Toe

- Toe is measured in fractions of degrees or in fractions of an inch (usually sixteenths), millimeters(mm), or decimals of an inch (such as 0.06”)
- Incorrect toe is the major cause of excessive tire wear
- Toe causes camber –type wear on one side of the tire if not correct

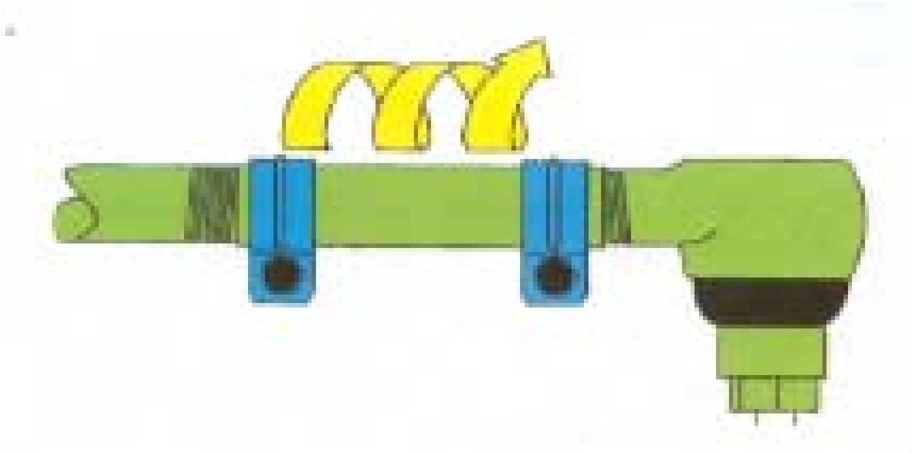
Toe

- Incorrect front toe does not cause a pull condition. Incorrect toe on the front wheels is split equally as the vehicle is driven because the forces acting on the tires are exerted through the tie rod and steering linkages to both wheels
- Incorrect (unequal) rear toe can cause tire wear. If the toe of the rear wheels is not equal, the steering wheel will not be straight and will pull toward the side with the most toe-in



Toe

- Front toe adjustment must be made by adjusting the tie rod sleeves correctly

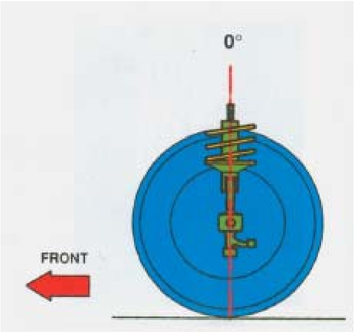


Toe

- Most vehicle manufacturers specify a slight amount of toe – in to compensate for the natural tendency of the front wheels to spread apart (become toed-out) due to the centrifugal force of the rolling wheels acting on the steering linkage
- Some manufacturers of front wheel drive vehicles specify a toe-out setting to compensate for the toe-in forces created by the engine drive forces on the front wheels
- Normal wear to the tie rod ends and other steering linkage parts usually causes toe-out

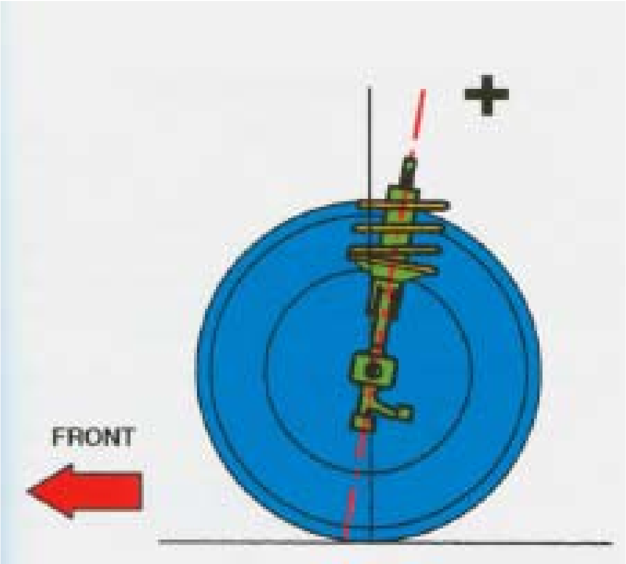
Caster

- Caster is the forward or rearward tilt of the steering axis in reference to a vertical line as viewed from the side of the vehicle. Steering axis is defined as the line drawn through the upper and lower steering pivot points.
- On an SLA suspension system, the upper pivot is the upper ball joint and the lower pivot is the lower ball joint. On a MacPherson strut system, the upper pivot is the centre of the upper bearing mount and the lower pivot point is the lower ball joint.
- Zero Center means that the steering axis is straight up and down, also called zero degrees or perfectly vertical



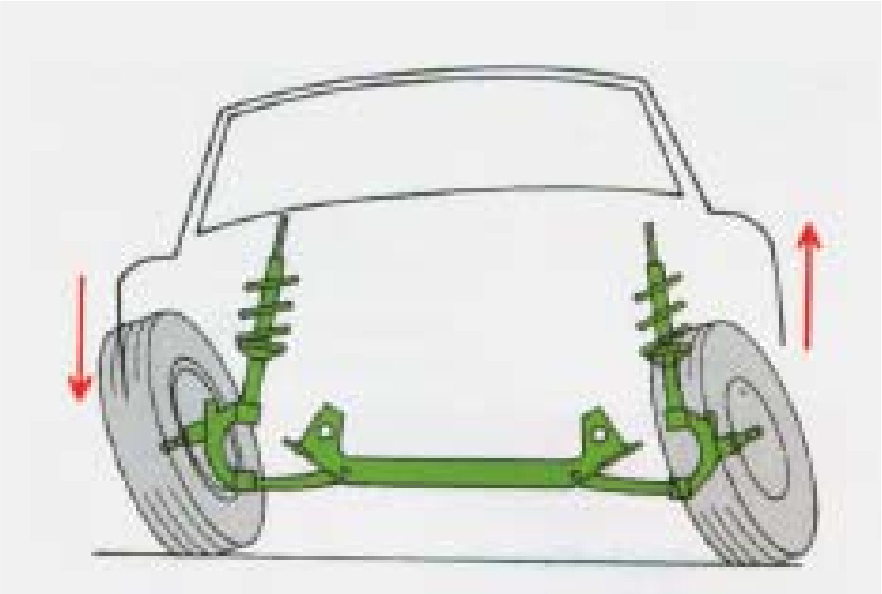
Caster

- Positive caster is present when the upper suspension pivot point is behind the lower pivot point (ball joint) as viewed from the side
- Negative caster is present when the upper suspension pivot point is ahead of the lower pivot point (ball joint) as viewed from the side
- Caster is measured in degrees of fractions of degrees



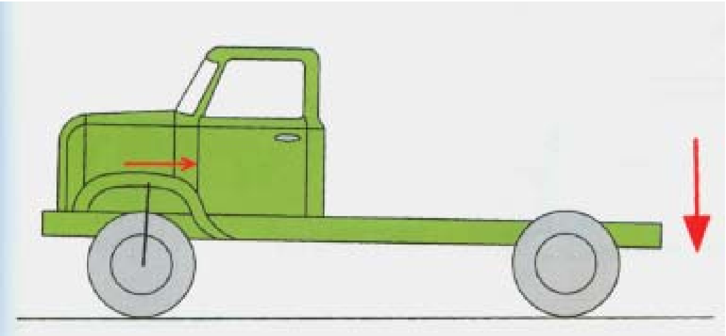
Caster

- Caster is not a tire wearing angle, but positive caster does cause changes in camber during a turn. This condition is called camber roll



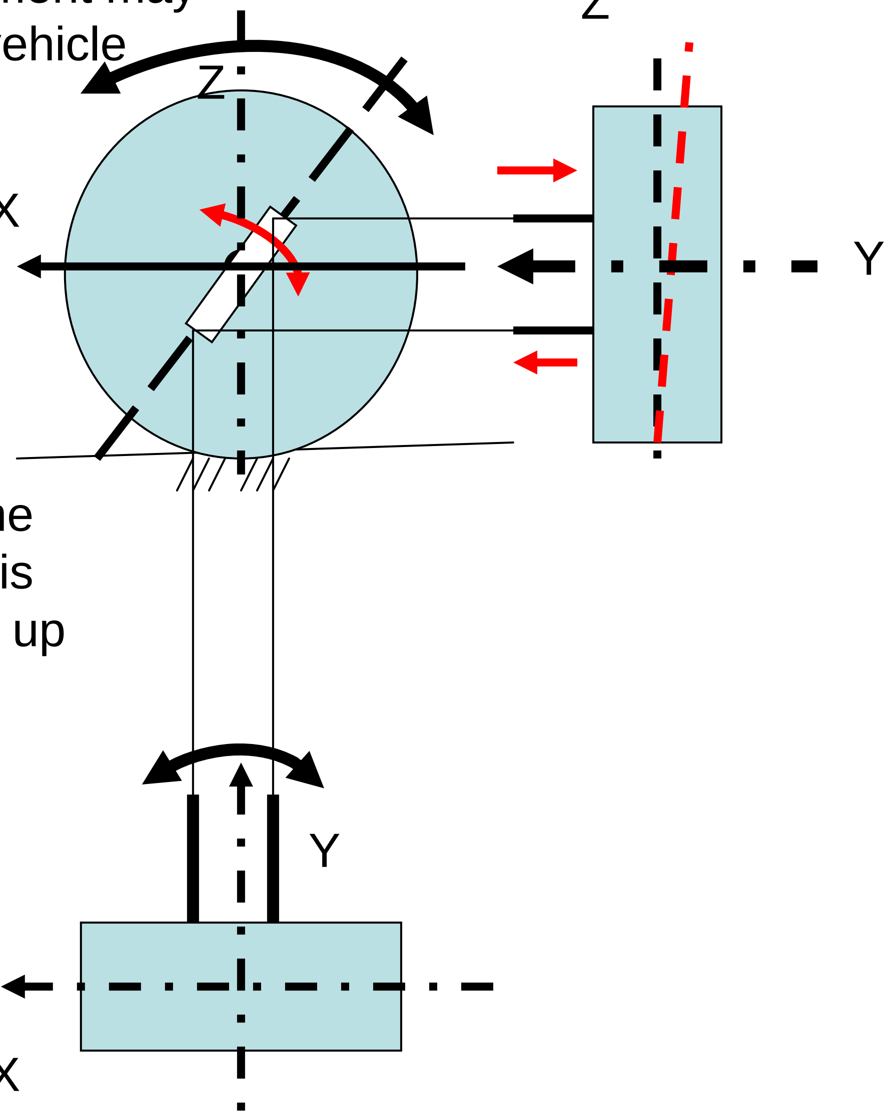
Caster

- Caster is a stability angle
  - If caster is excessively positive, vehicle steering will be very stable (will tend to be straight with little steering wheel correction needed) and help with steering wheel
  - If the caster is positive, the steering effort will increase with increasing positive caster. Greater road shocks will be felt by the driver when driving over rough road surfaces. Vehicles with as high as eleven degrees of positive caster usually use a steering dampener to control possible shimmy at high speeds and to dampen the snap-back of the spindle after a turn
  - If caster is negative, or excessively unequal, the vehicle will not be as stable and will tend to wander. If a vehicle is heavily loaded in the rear, caster increase as shown

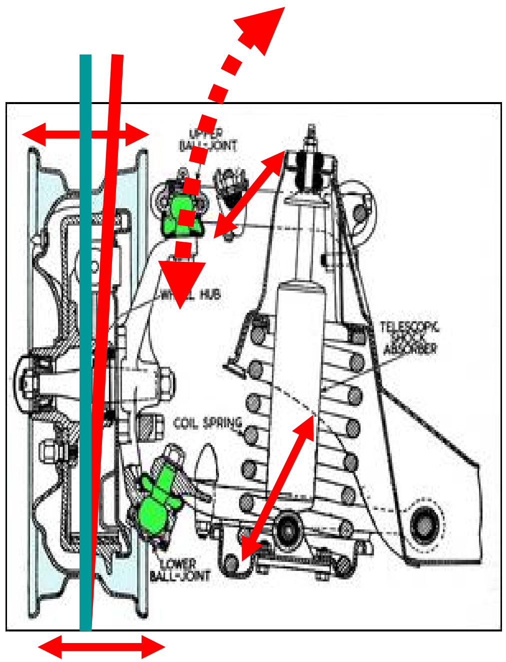


Caster Changes

This movement may be due to vehicle inertia

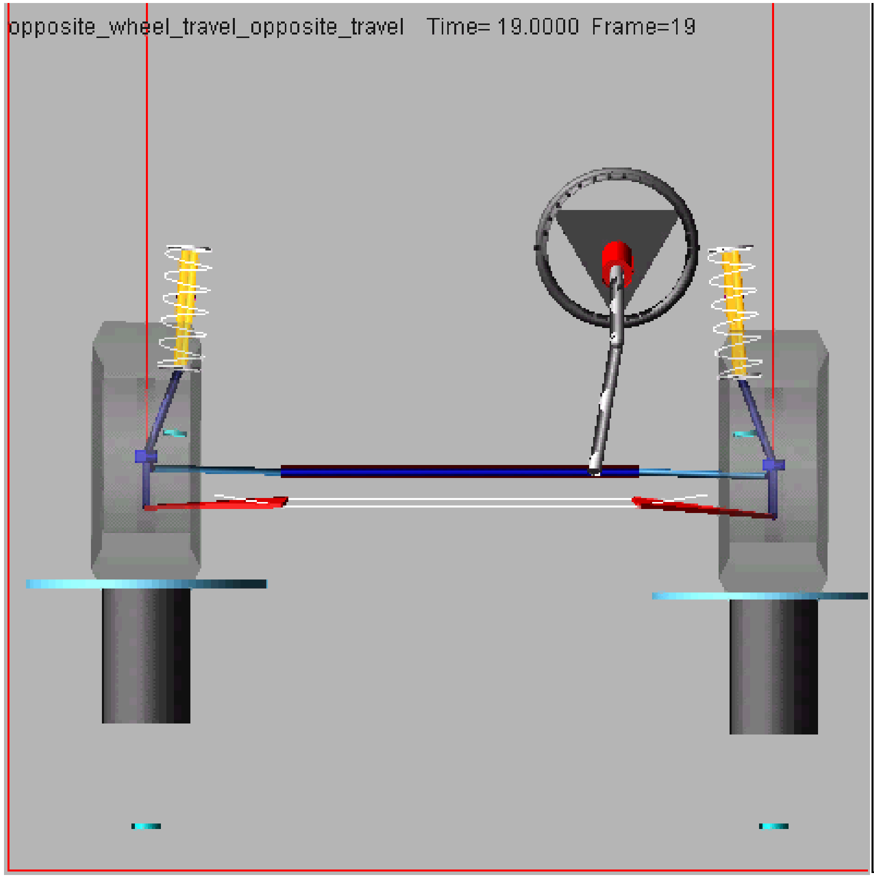


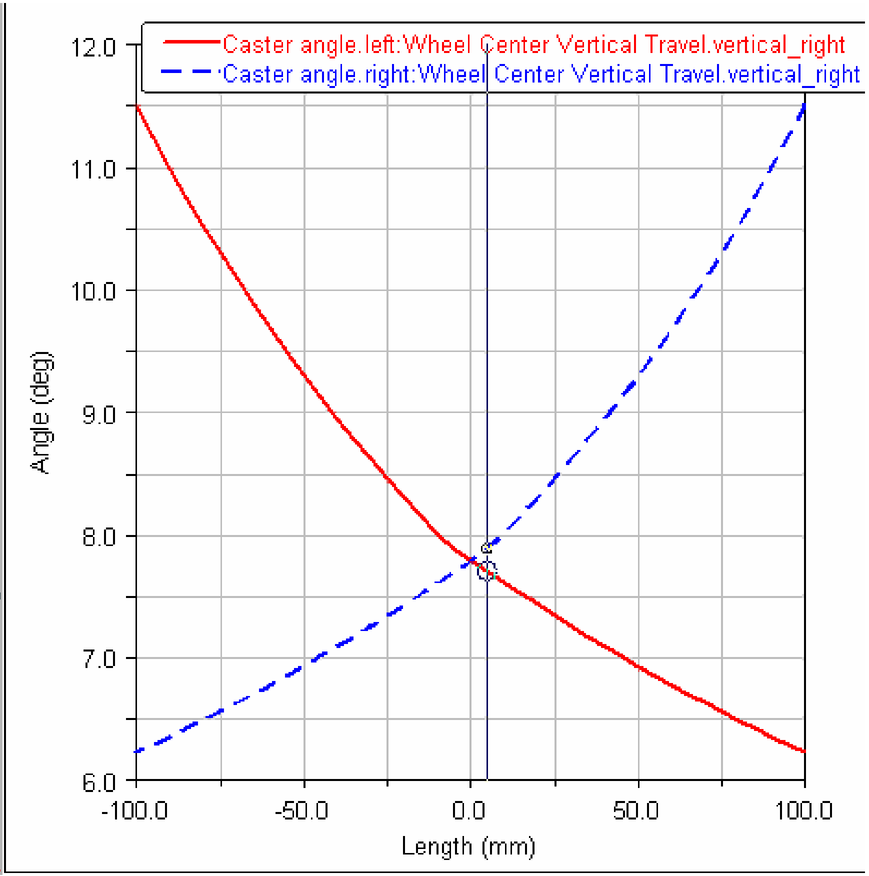
Assume wheel is locked up



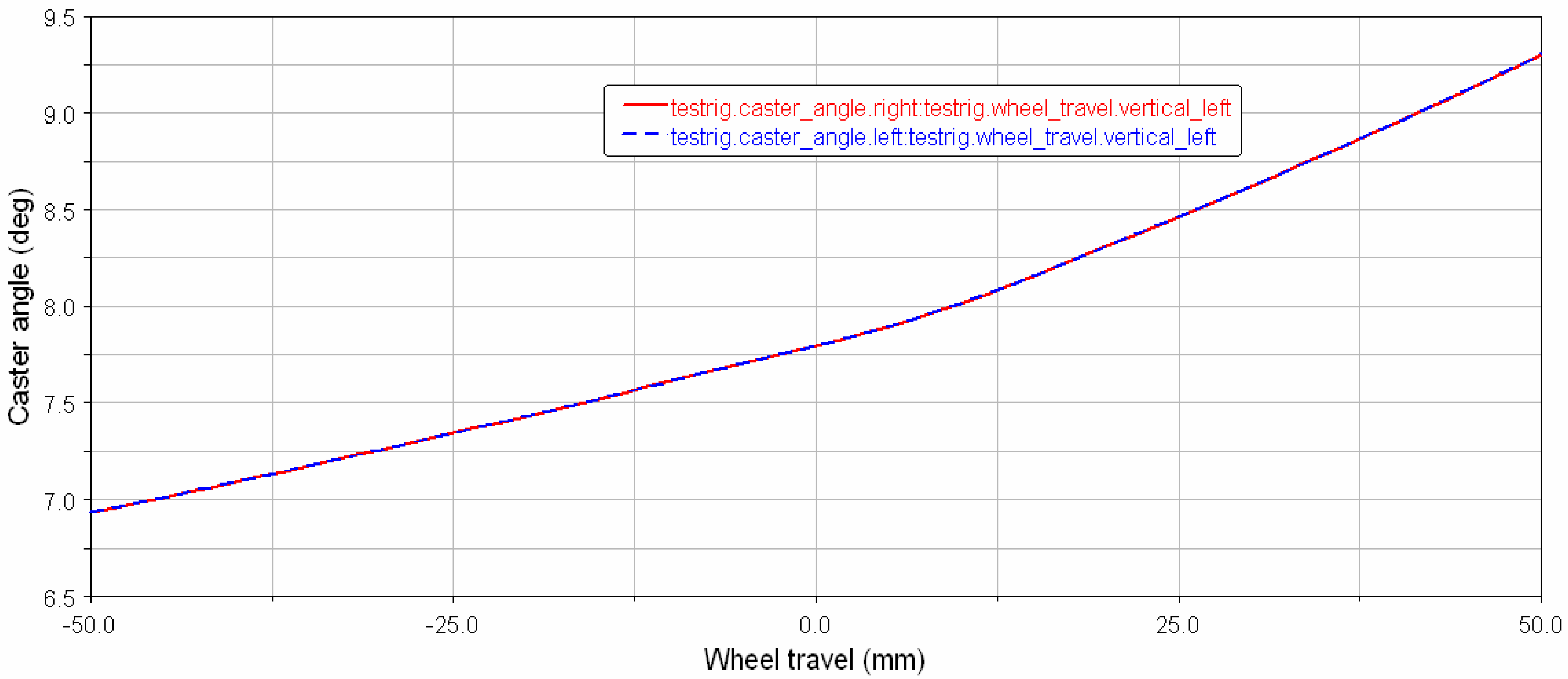
Caster Changes leads to camber change hence camber roll

Caster Change (Parallel Opposite)





Caster angle v/s Wheel Travel (Parallel)

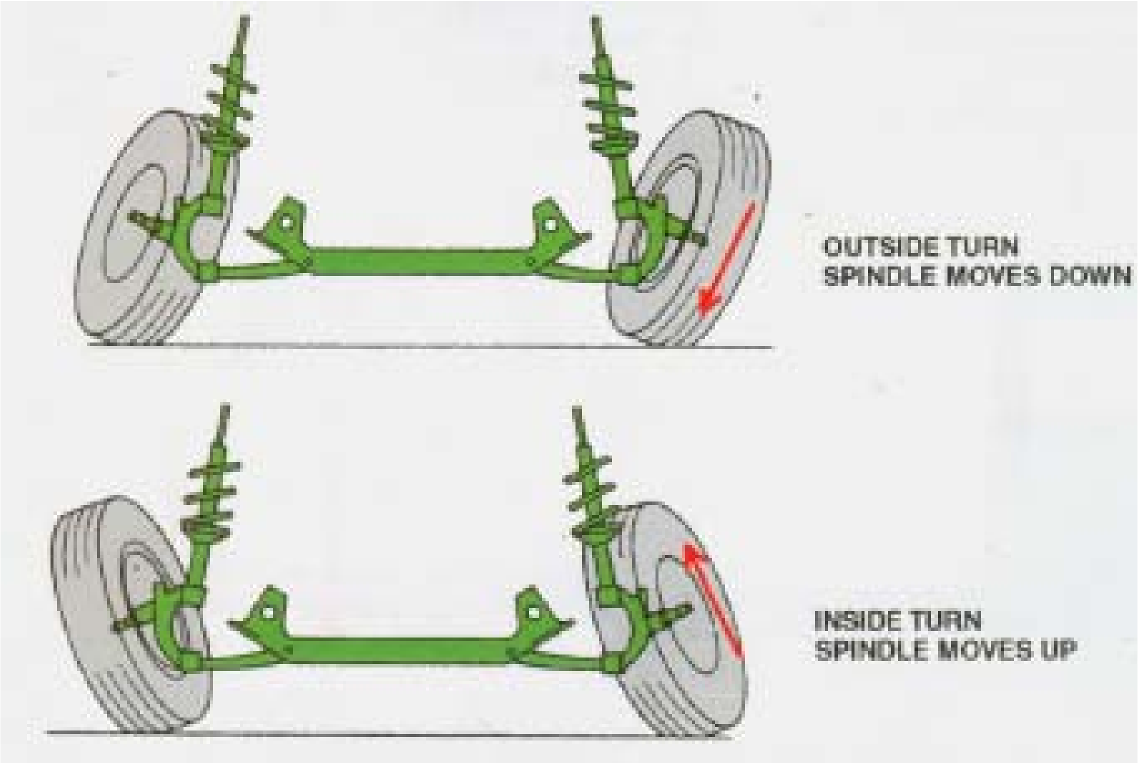


Caster

- Caster could cause pull if unequal. The vehicle will pull toward the side with least positive caster
- Caster is not adjustable on many vehicles
- If caster is adjustable, the change is made by moving either the lower or the upper pivot point forward or backward by means of one of the following methods
  - Shims
  - Eccentric Cams
  - Slots
  - Strut rods

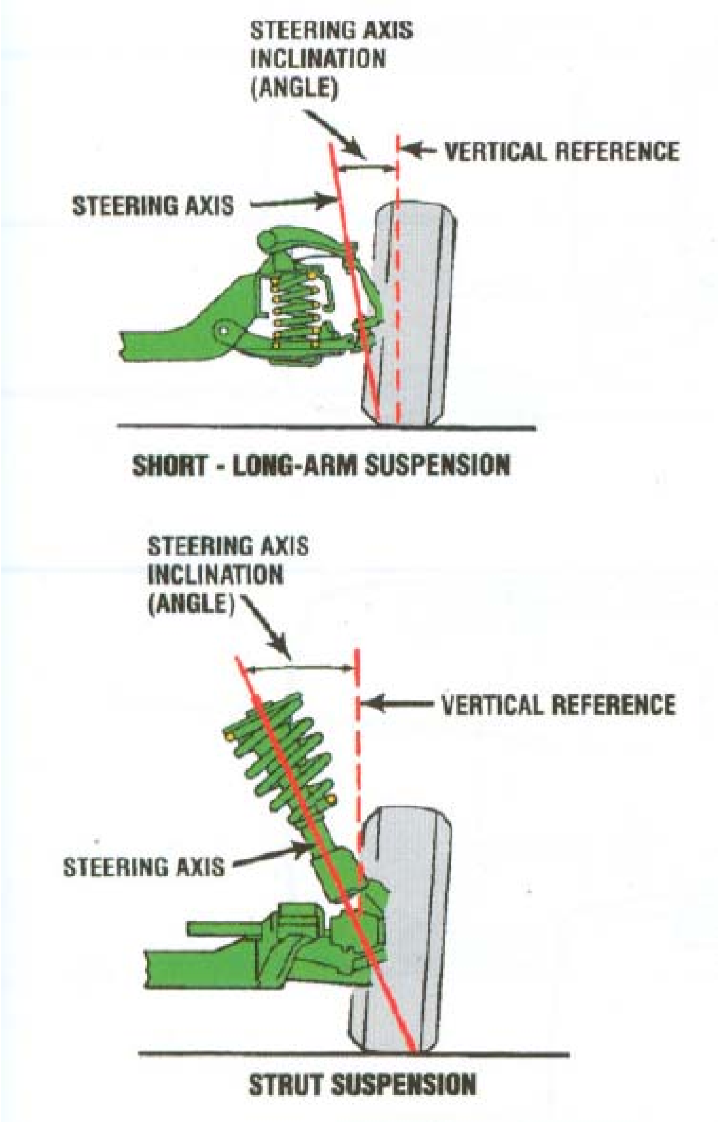
Caster

- Caster affects tire wear indirectly –Changes camber during turns



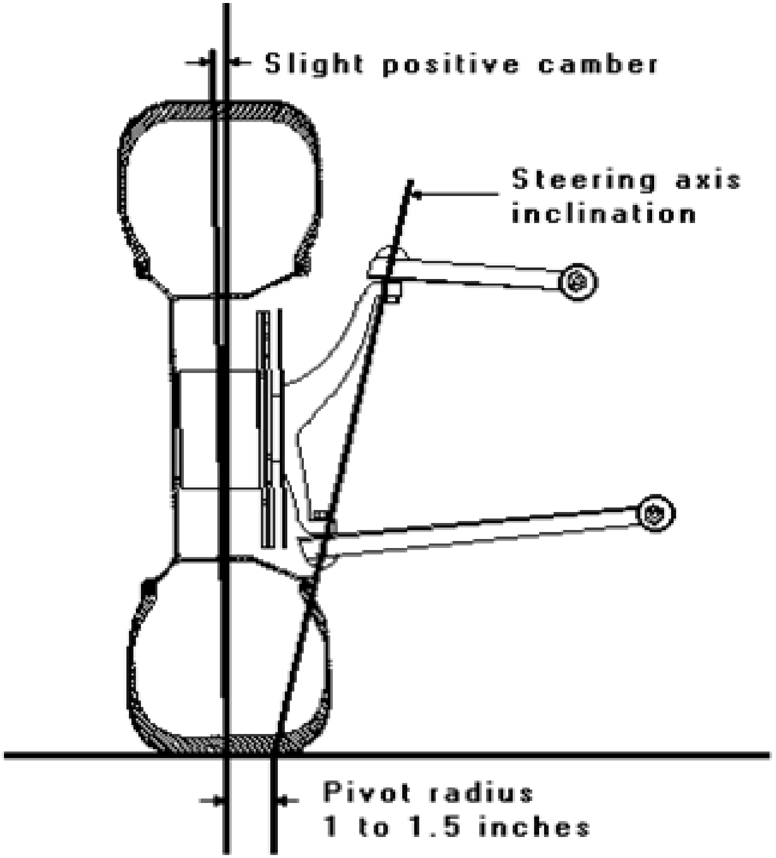
Steering Axis Inclination (SAI)

- The steering axis is the angle formed between true vertical and an imaginary line drawn between the upper and lower pivot points of the spindle.
- Steering axis inclination (SAI) is the inward tilt of the steering axis. SAI is also known as KPI and is the imaginary line drawn through the kingpin as viewed from the front



Steering Axis Inclination (SAI)

- The front view axis inclination angle add steering returnability by lifting the front axle in a turn.

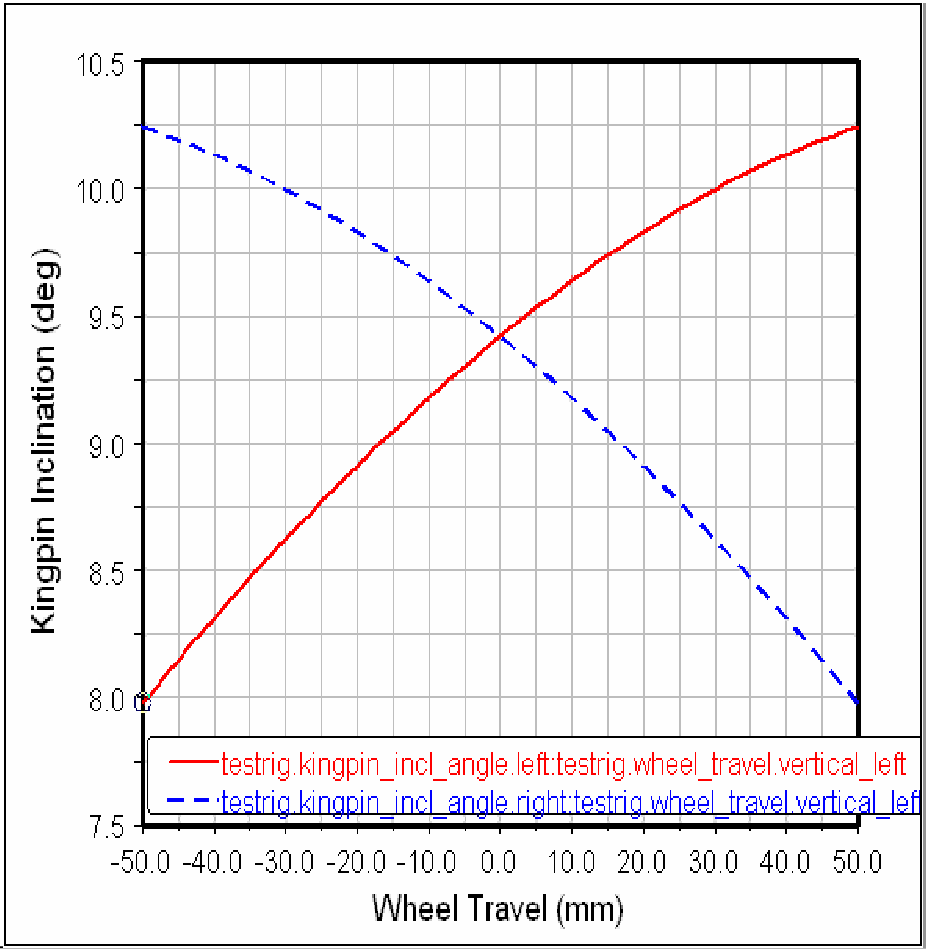


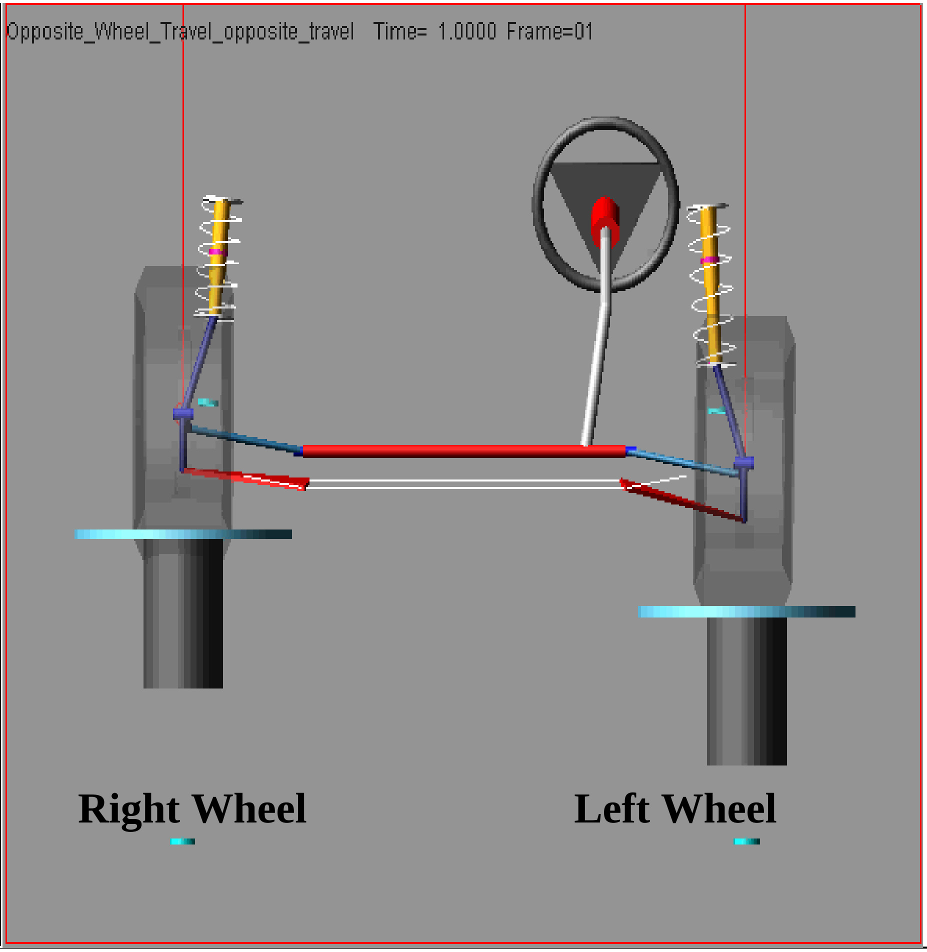
When the wheel is turned, you recognise the lifting of the vehicle (on the ball). If you press the ball, the turned wheel immediately goes into the straight ahead position.

Opposite wheel travel Analysis

Bump travel=50mm, Rebound travel=-50mm(2 inch)

Kingpin Inclination v/s Wheel Travel

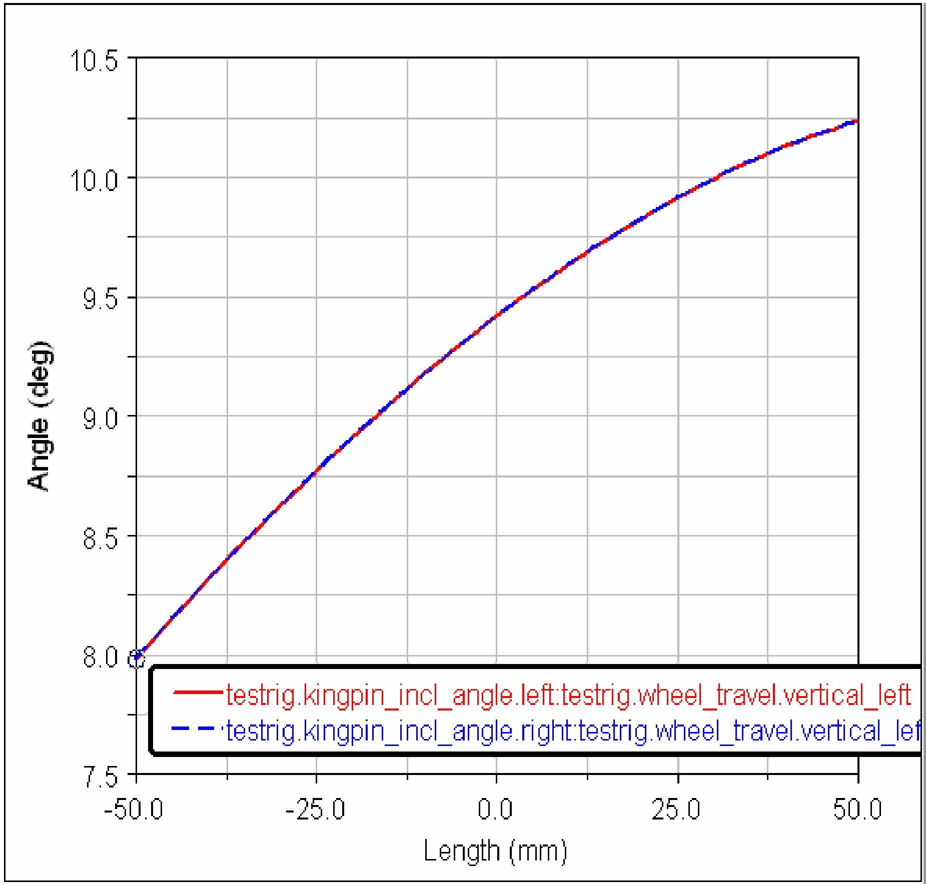


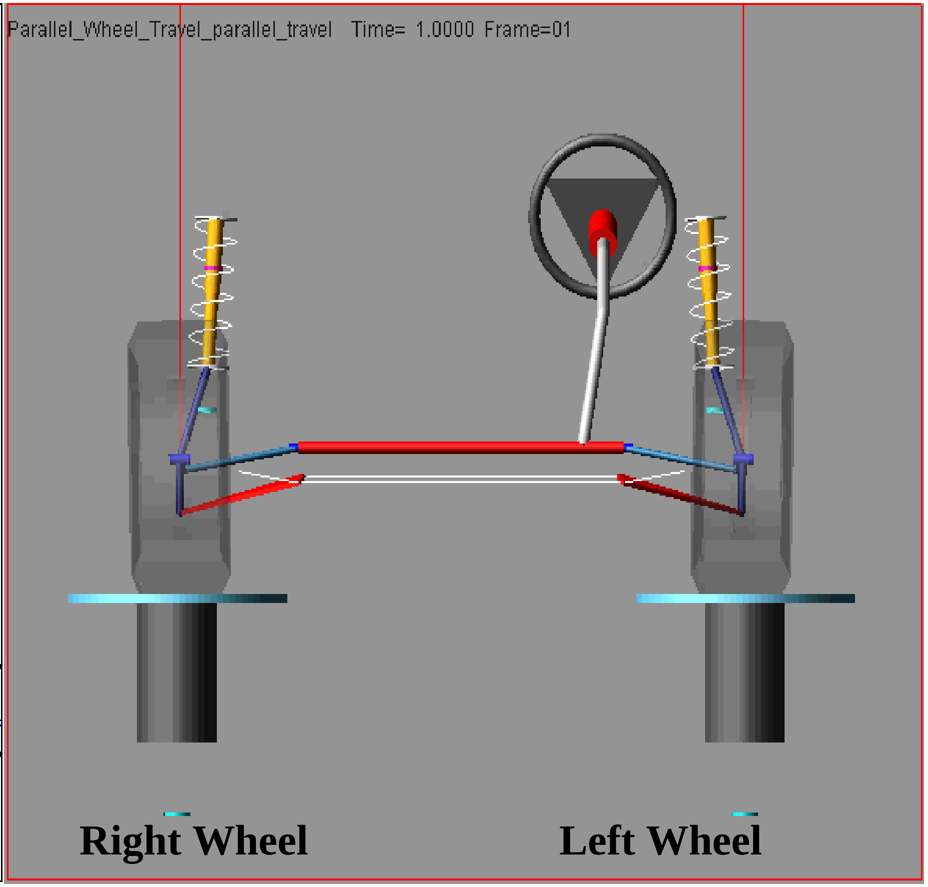


Parallel wheel travel Analysis

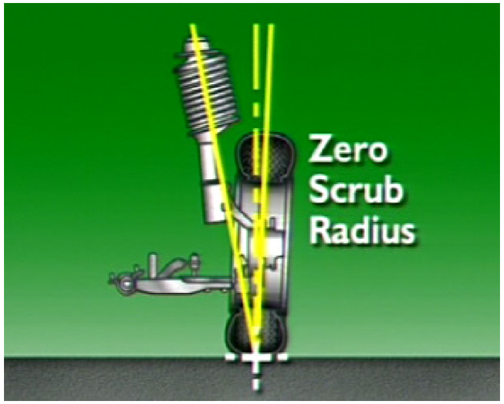
Bump travel=**50mm**, Rebound travel=**-50mm** (2 inch)

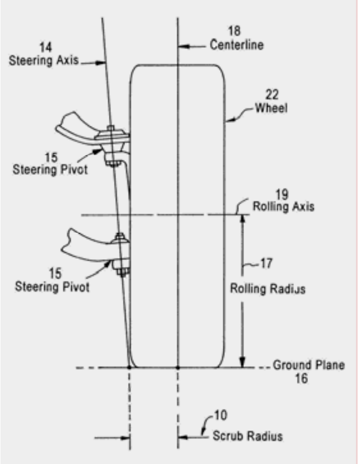
Kingpin Inclination v/s Wheel Travel





Scrub Radius

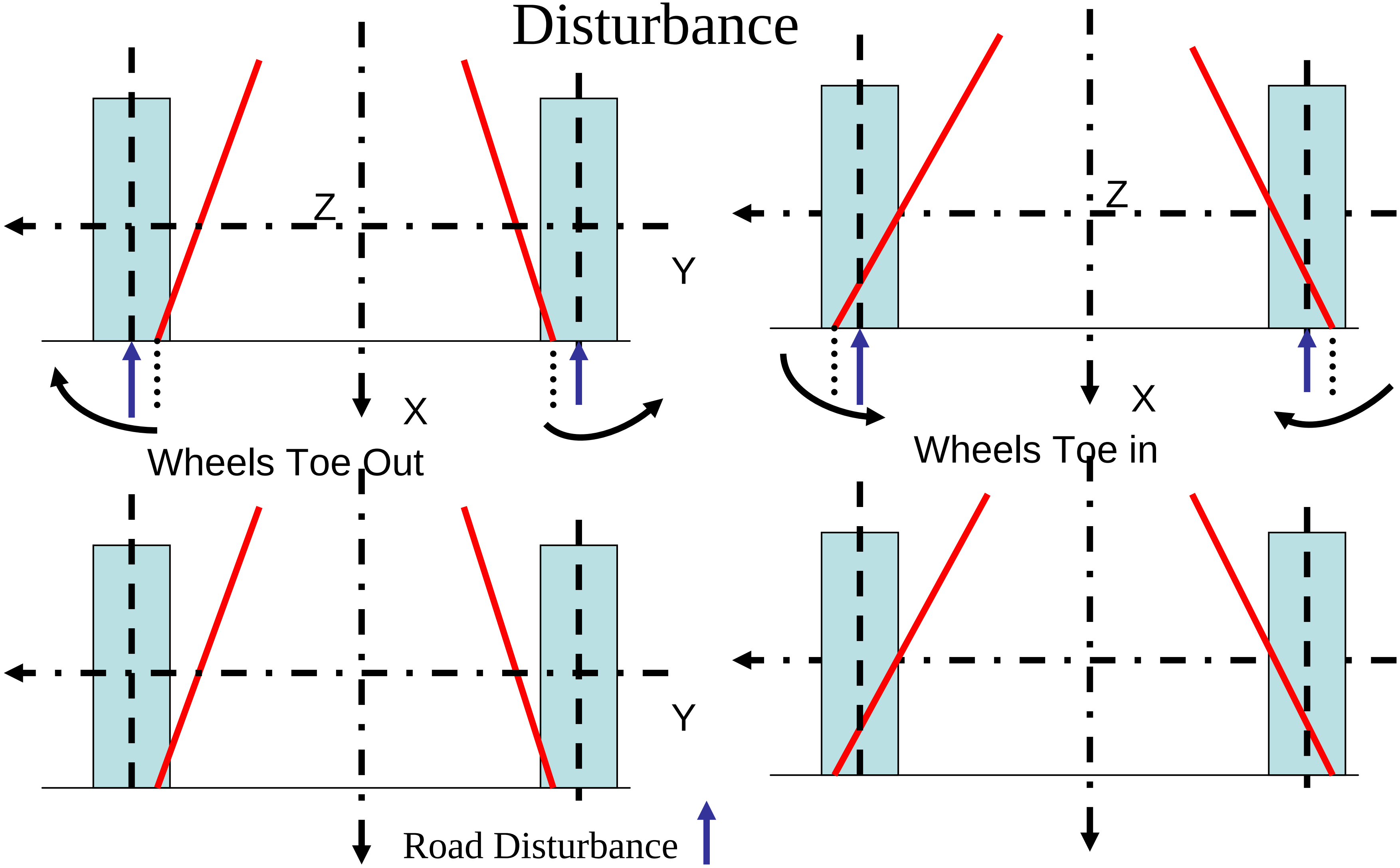




Positive scrub Radius

- Scrub radius is also known as steering offset, and scrub geometry. It is the distance between 2 imaginary points on the road surface - the point of center contact between the road surface and the tire, and the point where the steering-axis center-line contacts the road surface.
- If these two points intersect at the center of the tire, at the road surface, then the scrub radius is zero.
- If they intersect below the road surface, scrub radius is positive.
- If they intersect above the road surface, scrub radius is negative.

Effect of Scrub Radius on Steering Due to Road Disturbance



Positive Scrub Radius Z

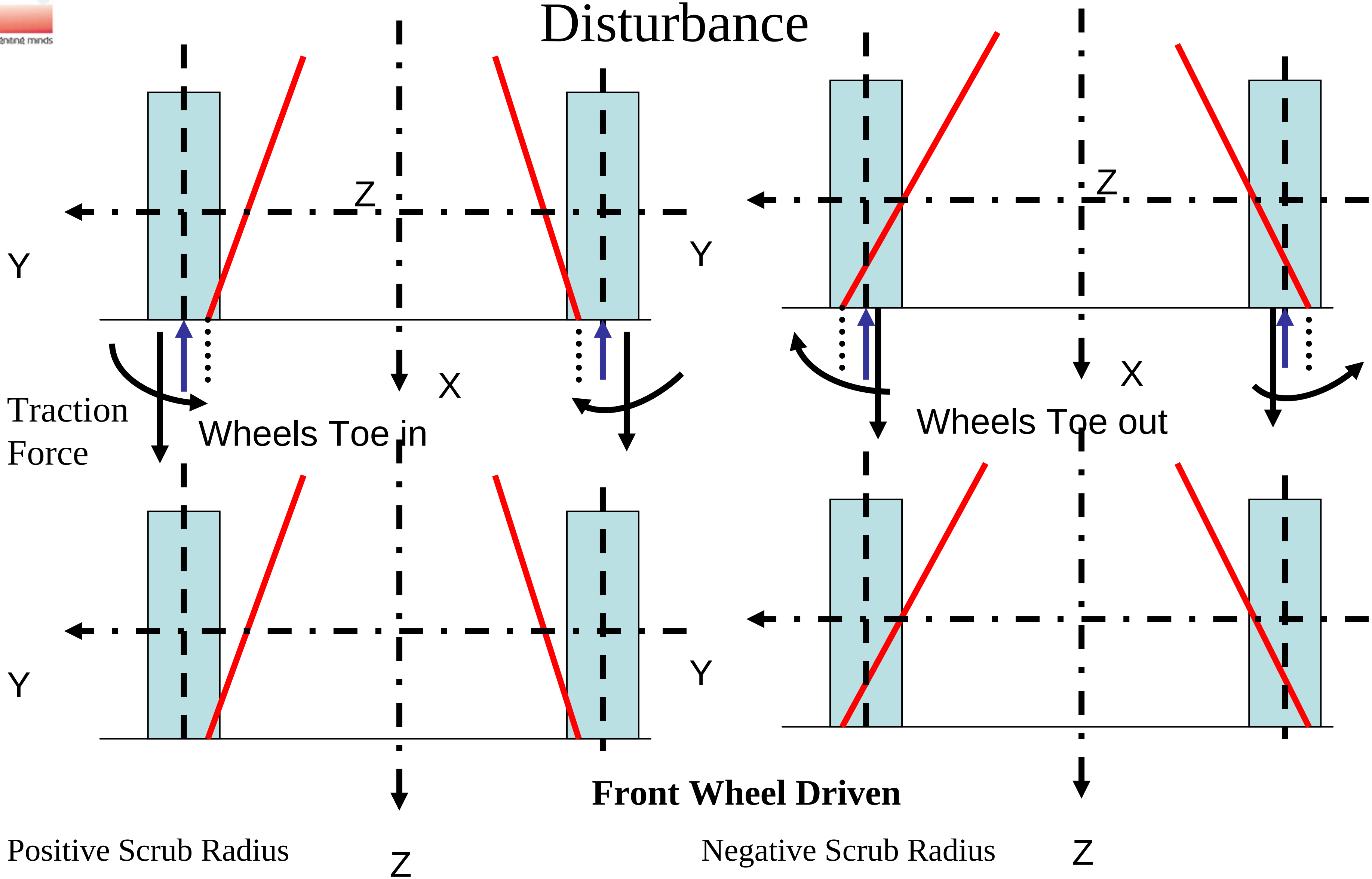
Disturbance Creates outboard moment

Negative Scrub Radius Z

Disturbance Creates inboard moment

Rear Wheel Driven

Effect of Scrub Radius on Steering Due to Road Disturbance



Positive Scrub Radius

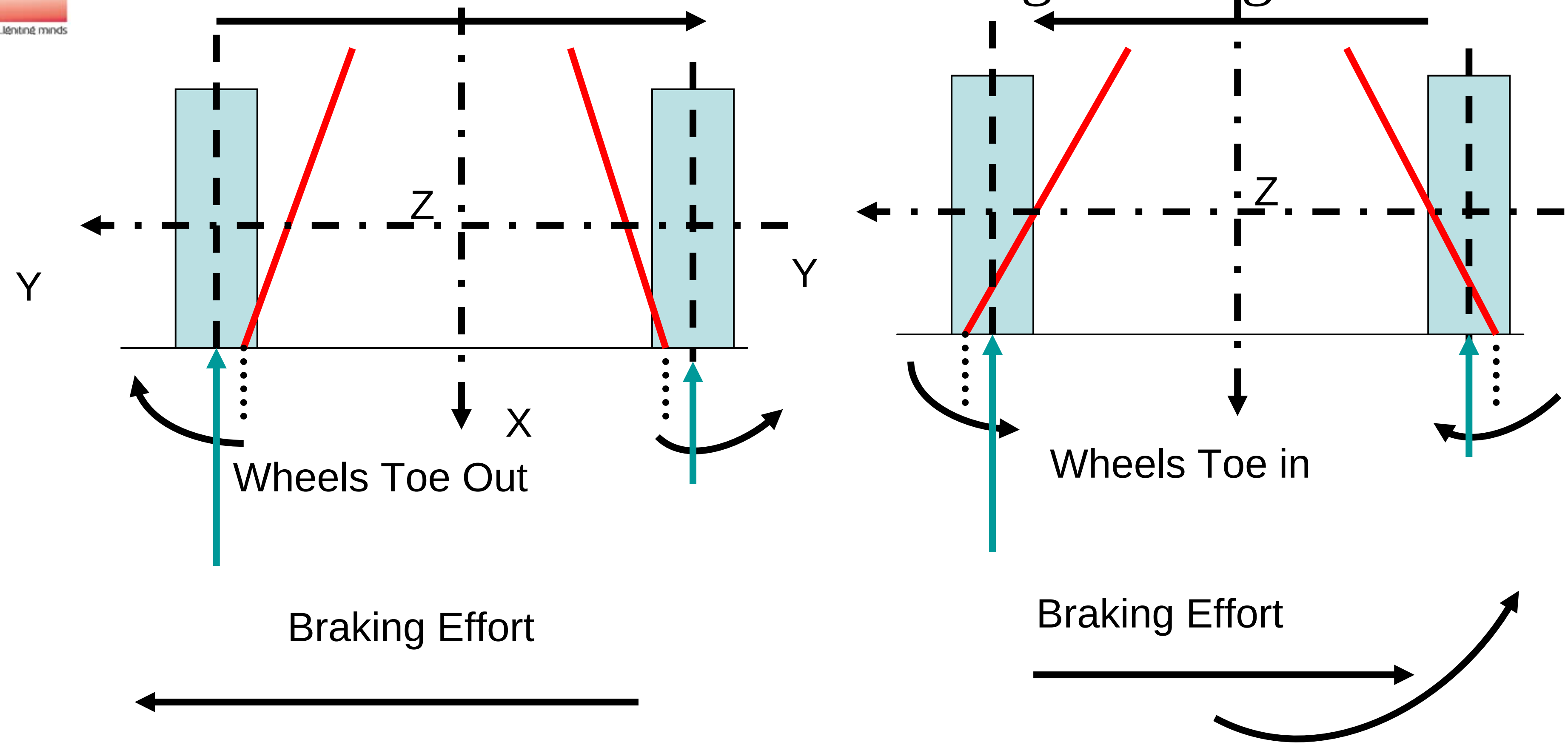
Traction Force Creates inboard moment

Traction is greater than Road Disturbance

Negative Scrub Radius

Traction Force Creates out board moment

Effect of Scrub Radius During Braking

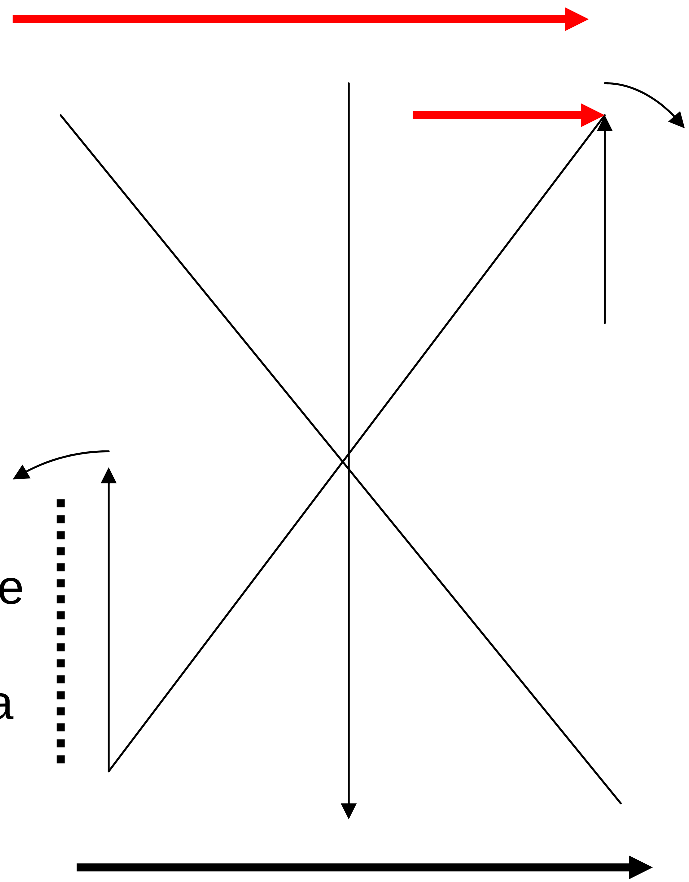


Positive scrub radius will cause the vehicle to veer towards the side with the greater effort

Negative scrub radius will cause the vehicle to veer away from the side with the greater effort

Scrub Radius and Diagonal Split Brake

- Vehicles with a diagonal-split brake system have negative scrub radius built into the steering geometry. If one half of the brake system fails, then the vehicle will tend to pull up in a straight line.

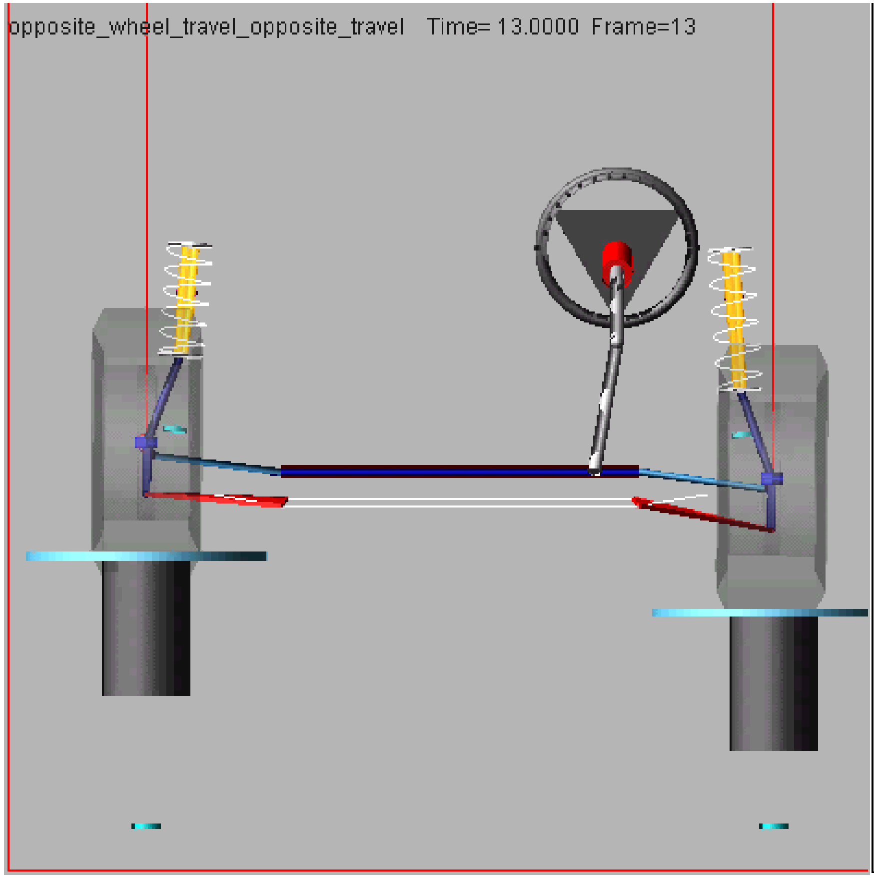


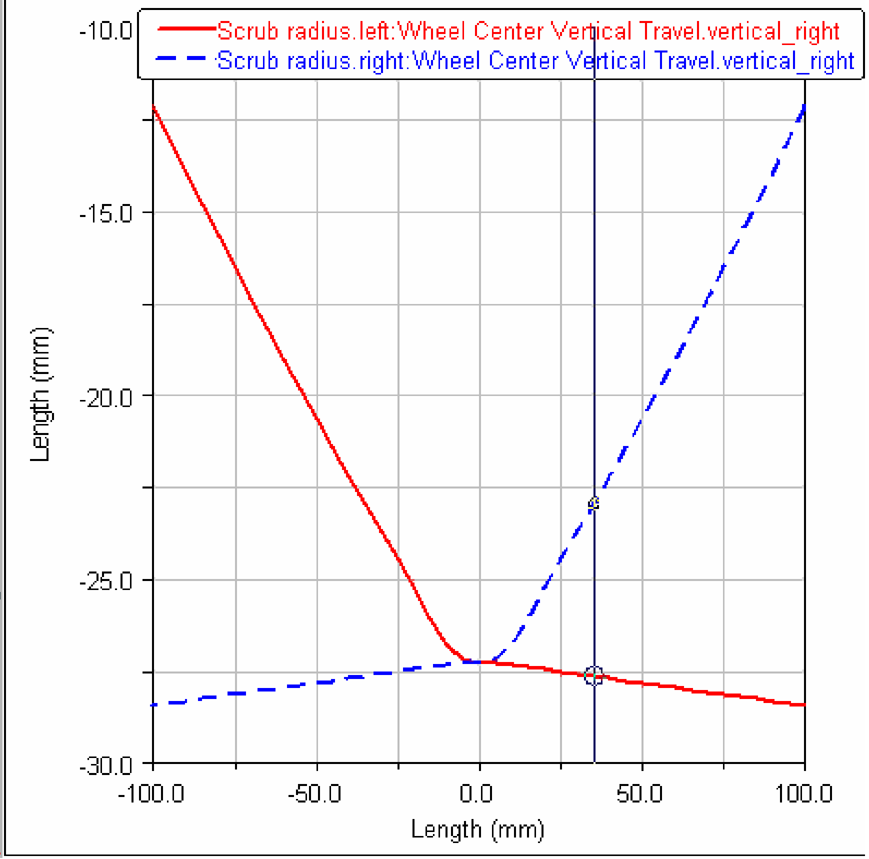
Reaction due to braking produces a counter clockwise couple

More Brake effort produces a couple as shown

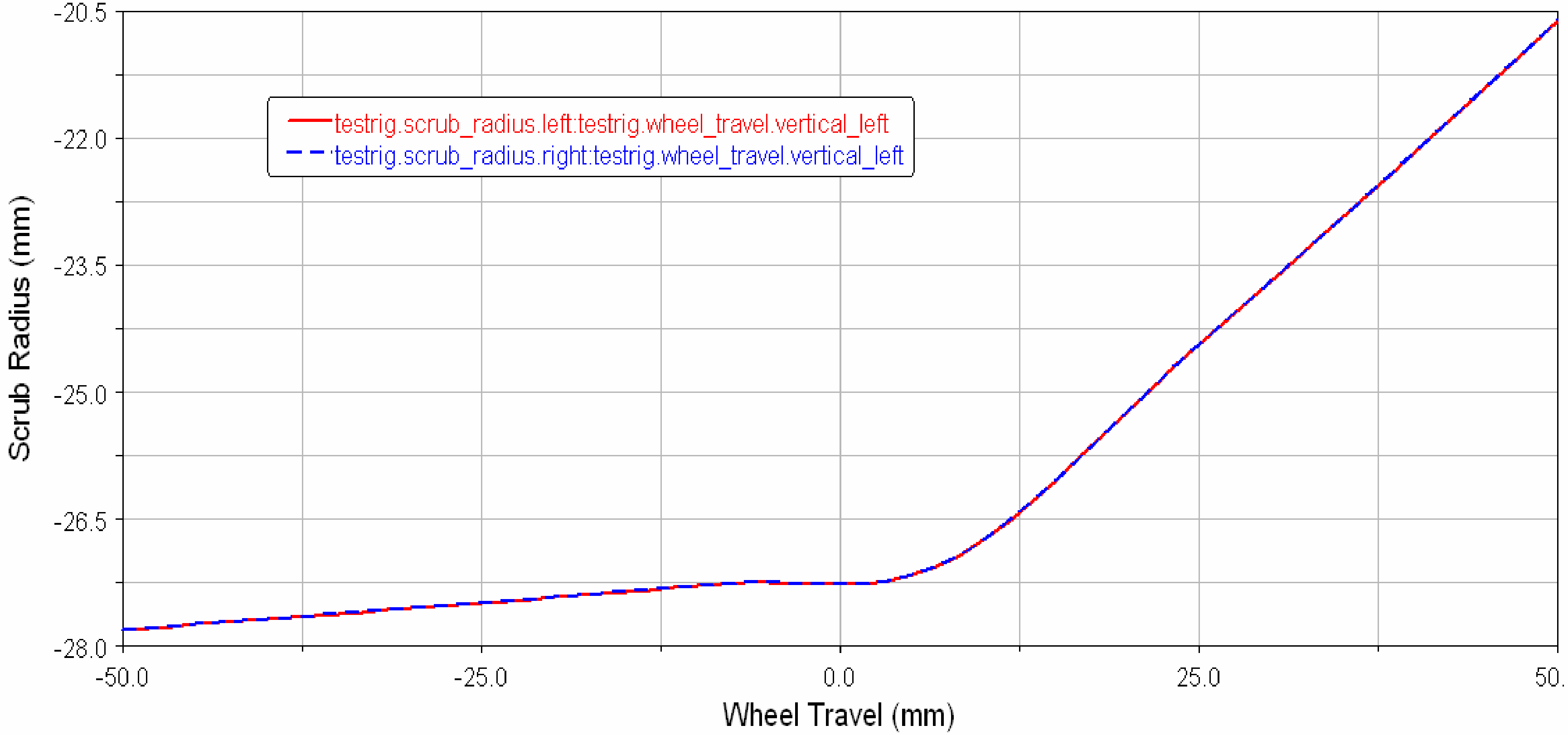
The braking force tries to turn counterclockwise

Scrub Radius Change



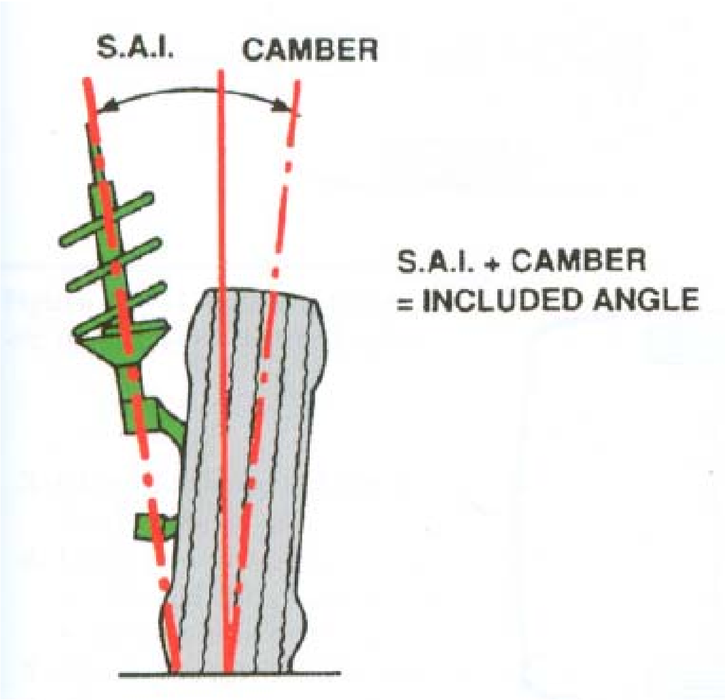


Scrub Radius v/s Wheel Travel



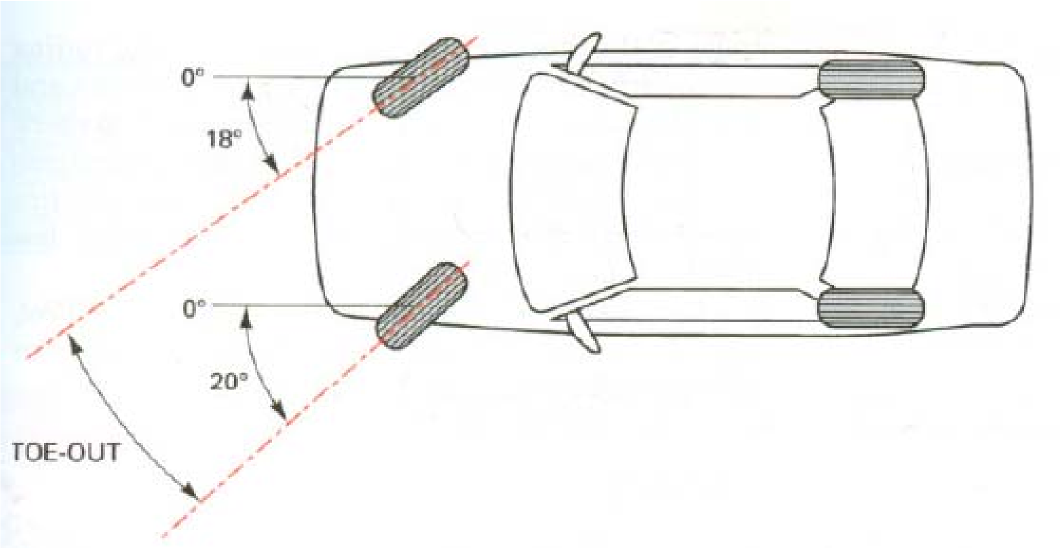
Included Angle

- The included angle is the SAI added to the camber reading of the front wheel only. Included angle is an important angle to measure when diagnosing vehicle handling or tire wear problems



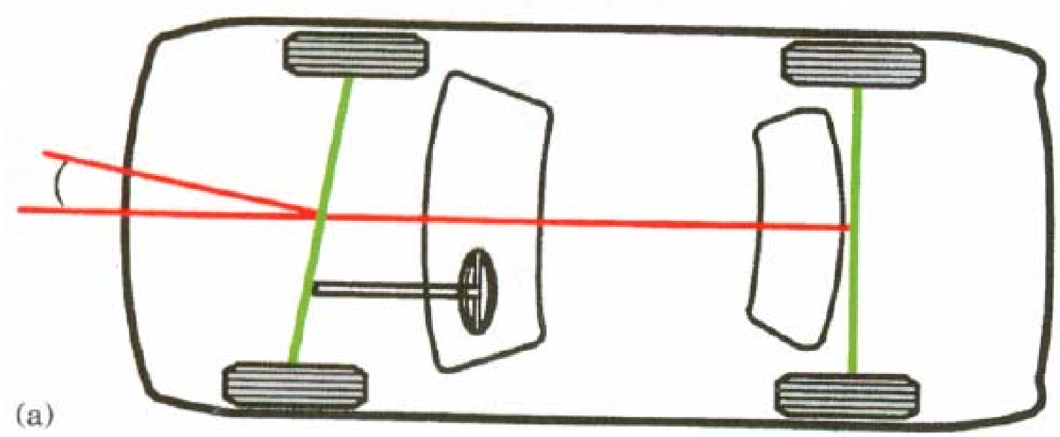
Turning Radius (Toe-out on Turns)

- Whenever a vehicle turns a corner, the inside wheel has to turn at a sharper angle than the outside wheel because the inside wheel has a shorter distance to travel.
- Turning radius is also called toe-out on turns (TOT or TOOT)
- Turning radius is a nonadjustable angle

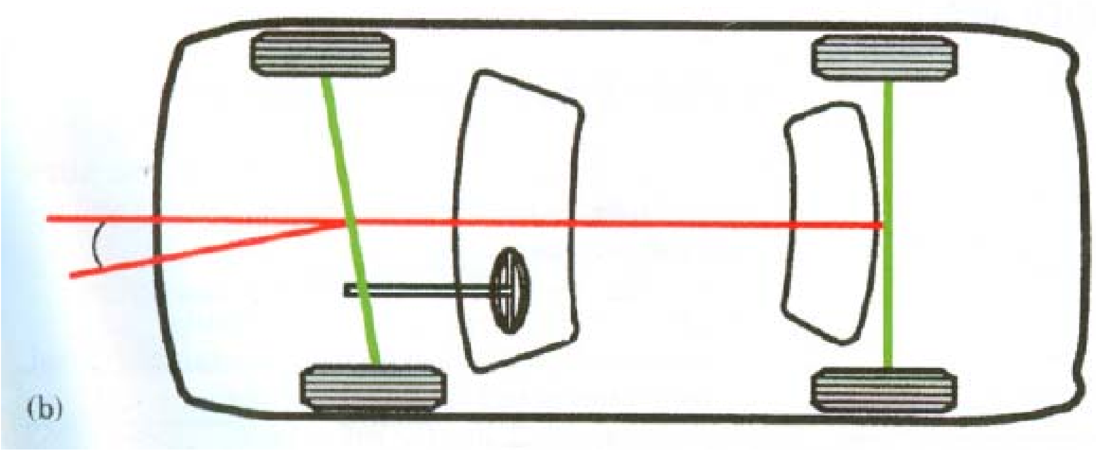


Setback

- Setback is the angle formed by a line drawn perpendicular to the front axle



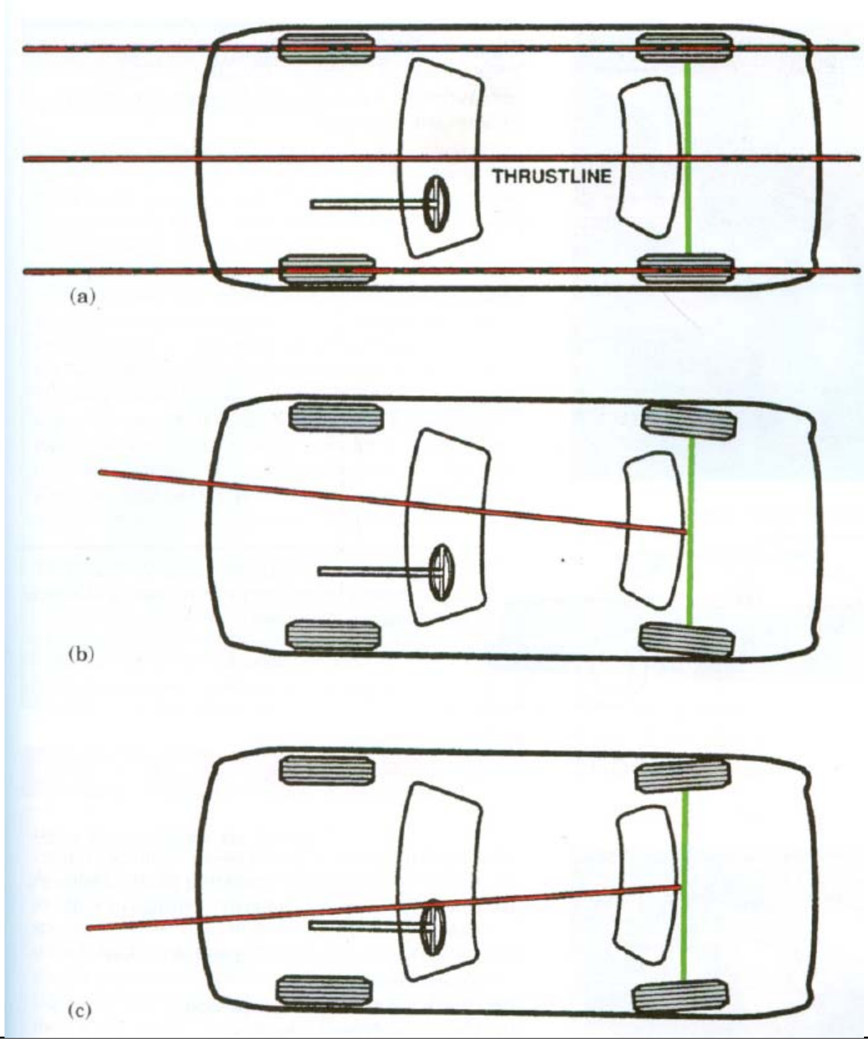
Positive Setback



Negative Setback

Thrust Angle

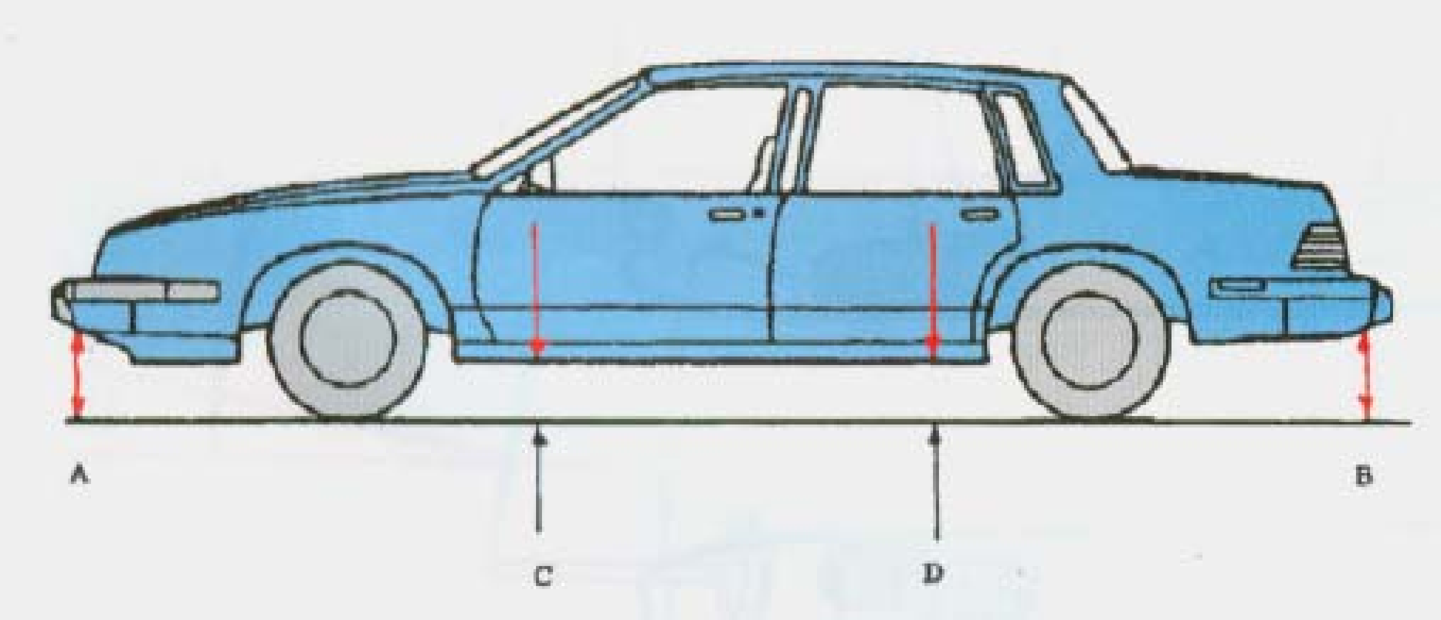
- Thrust Angle is the angle of the rear wheels as determined by the total rear toe



Tracking

- Tracking is the term used to describe the fact that the rear wheels should track directly behind the front wheels.

Ride Height



Load transfer changes wheel alignment angles and ride height

Alignment Specifications at Curb Height

ALIGNMENT SPECIFICATIONS AT CURB HEIGHT

FRONT WHEEL ALIGNMENT	ACCEPTABLE ALIGNMENT RANGE AT CURB HEIGHT	PREFERRED SETTING
CAMBER . . All* *Side To Side Differential	-0.6° to +0.6° 0.7° or less	+0.0° 0.0°
TOTAL TOE All Vehicles (See Note) Specified In Degrees	0.4° In to 0.0°	0.2° In
CASTER*	REFERENCE ANGLE	
All Models *Side To Side Caster Differential Not to Exceed	+2.0° to +4.0° 1.0° or less	+3.0° 0.0°
REAR WHEEL ALIGNMENT	**ACCEPTABLE ALIGNMENT RANGE AT CURB HEIGHT**	**PREFERRED SETTING**
CAMBER . . All Models	-0.6° to +0.4°	-0.1°
TOTAL TOE* All Vehicles (See Note) Specified In Degrees	0.2° Out to 0.4° In	0.1° In
THRUST ANGLE *TOE OUT When Backed On Alignment Rack Is TOE IN When Driving.	-0.15° to +0.15°	

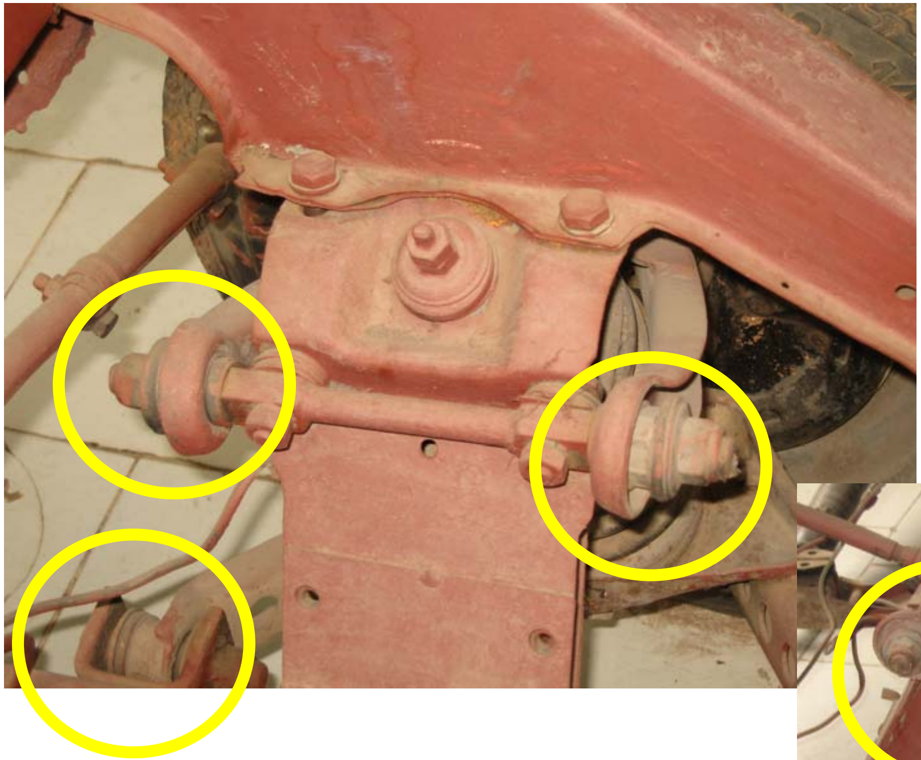
NOTE: Total toe is the arithmetic sum of the left and right wheel toe settings. Positive is Toe-in, negative is Toe-out. Total Toe must be equally split between each front wheel to ensure a centered steering wheel. Left and Right toe must be equal to within 0.02 degrees.

Suspension Kinematics

- A basic characteristic of suspension system is the change in orientation and position of the wheel under the wheel stroke, which is called kinematic characteristic and it strongly influences the handling and the stability of the vehicle
- Kinematic design of a suspension system involves determining the positions of hardpoints or kinematic design points
- Suspension static design factors such as toe, camber and caster are decided by the location of hardpoints.

Compliance

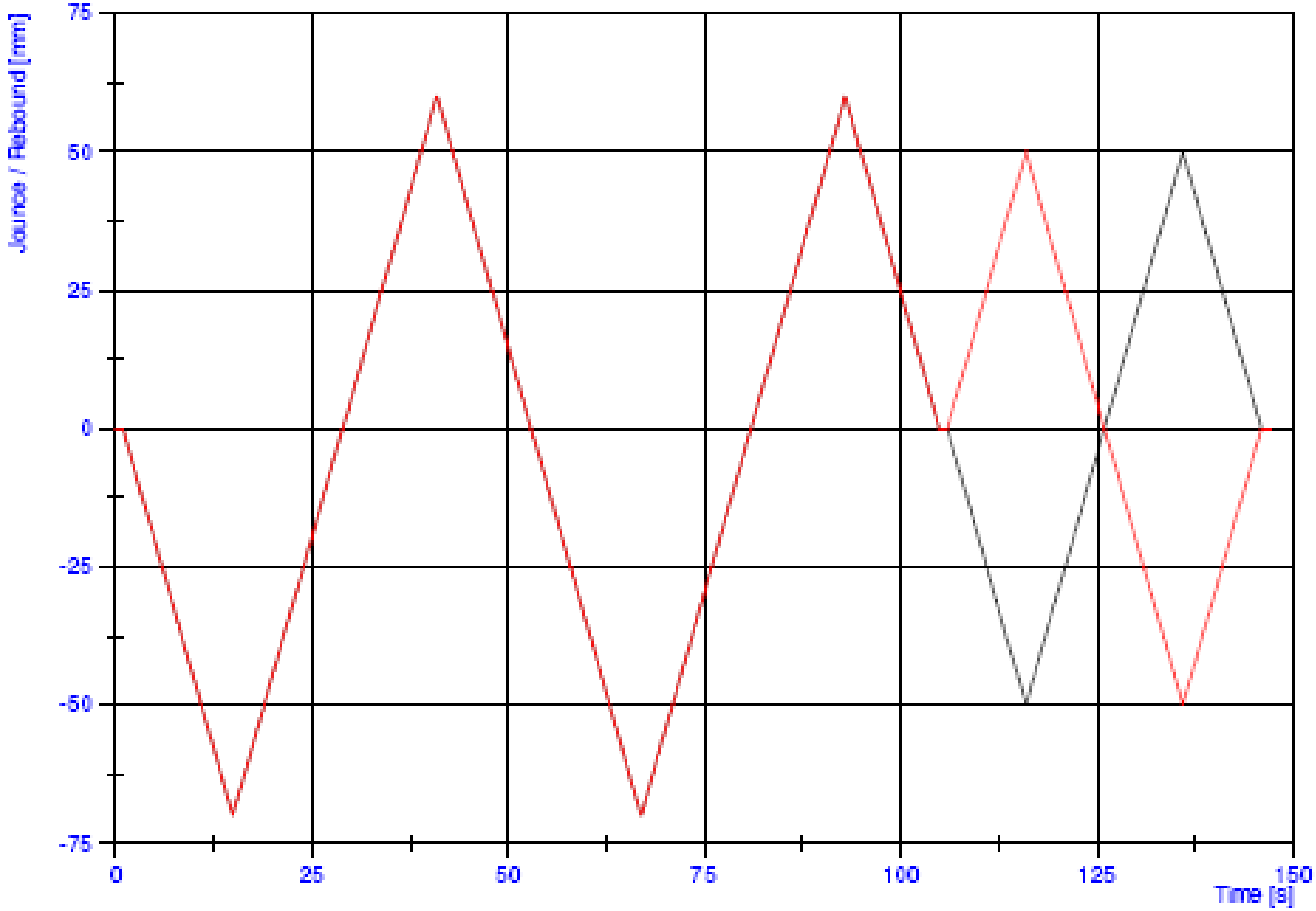
- Compliance is deliberately introduced into the suspension systems through bushings to achieve good ride.
- Bushings are rubber members provided in suspension and steering sub-systems to avoid metal-to-metal friction during kinematic motion.
- Two types of compliance are of interest – lateral and longitudinal force compliance.
- Specific Bushings are required to have desirable stiffness in specific orientations to meet compliance



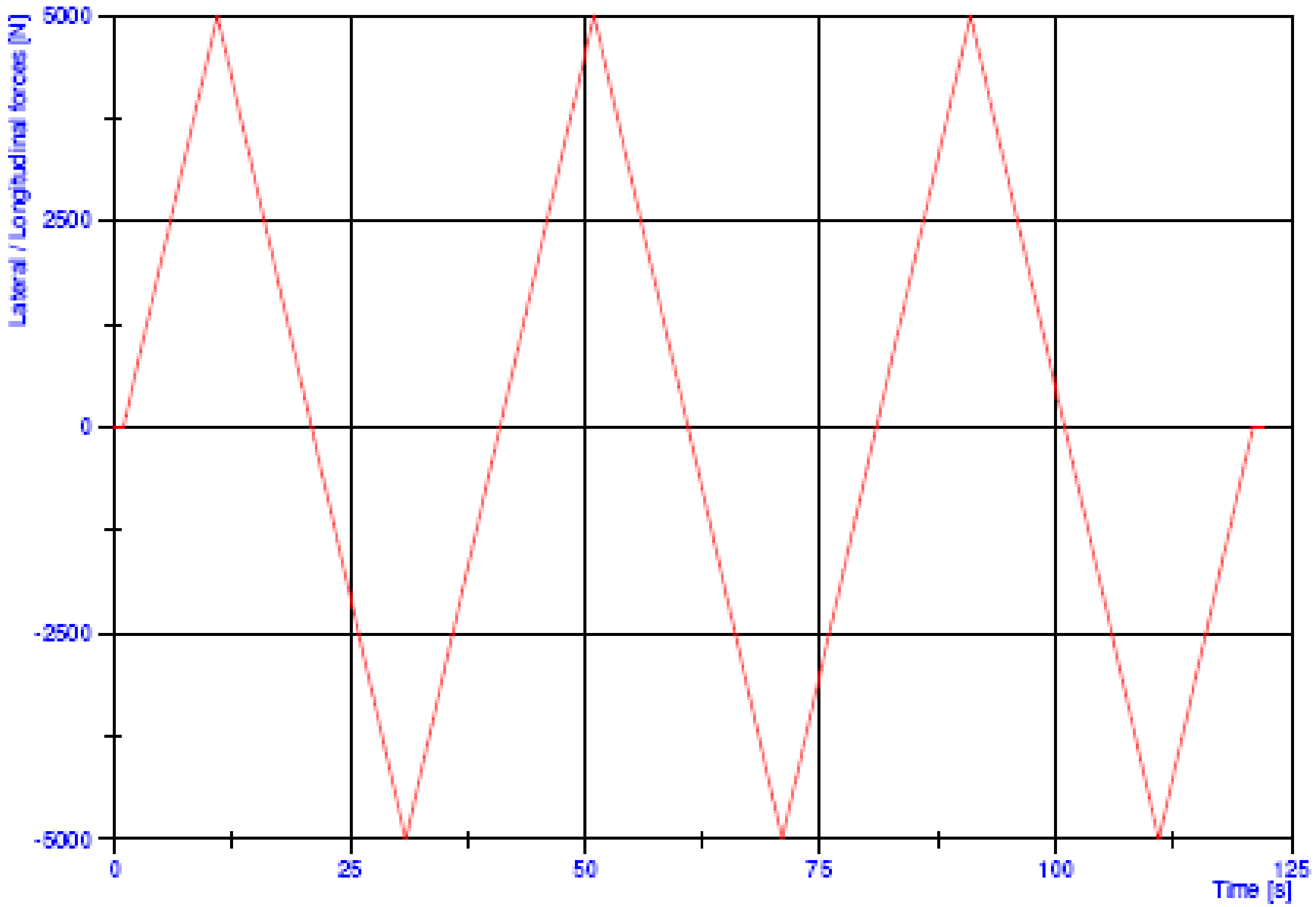
Bush Stiffness

- *For MacPherson Suspension*
- *Y* stiffness of lca_front and lca_rear bushings affects toe and camber in braking and driving
- *X* stiffness of rack_house bushing (steering sub-system) affects toe and camber in braking and driving
- *Z* stiffness of rack_house bushing (steering sub-system) affects toe and camber under lateral forces

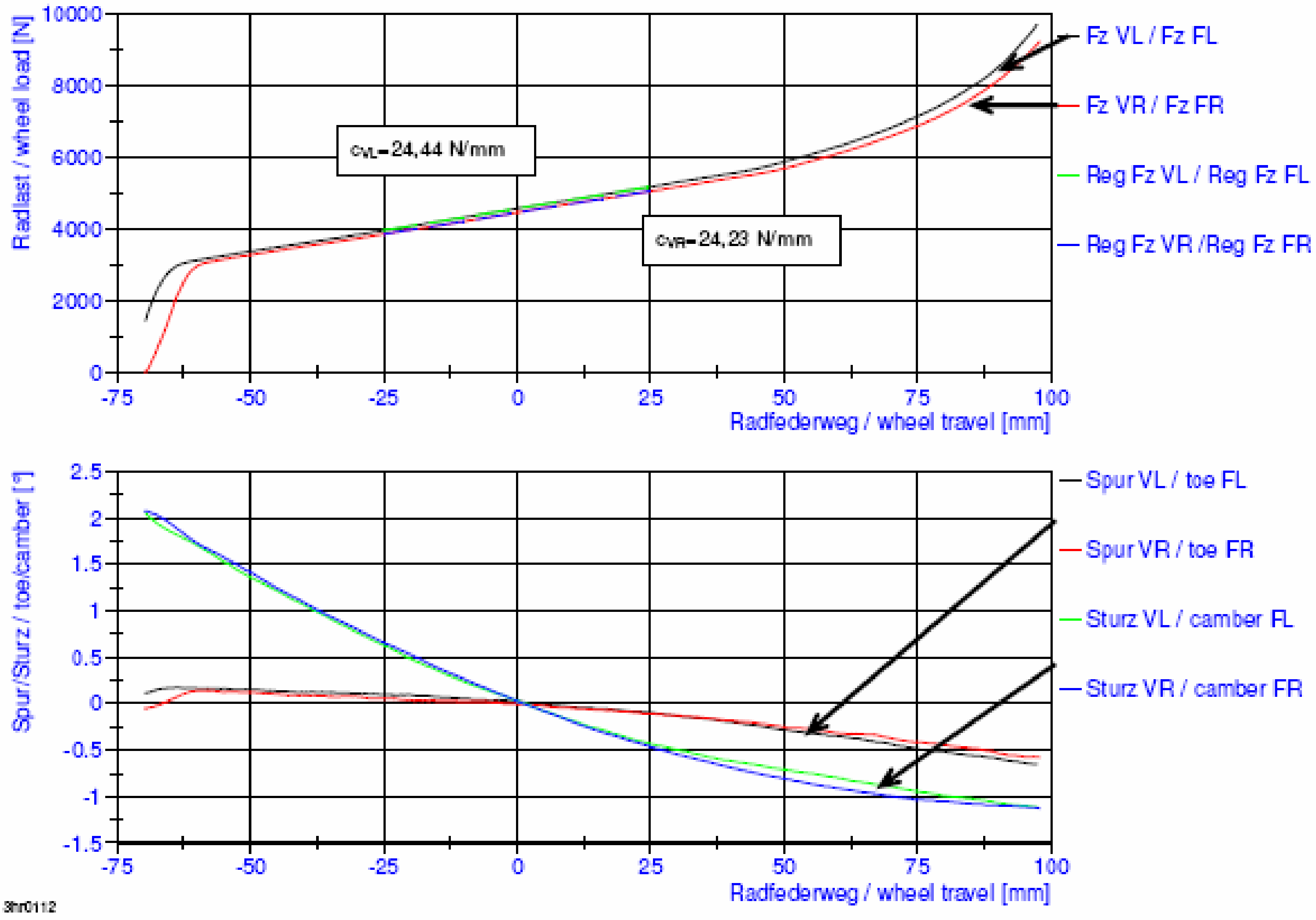
Kinematics and Compliance Test Rig with Test Vehicle



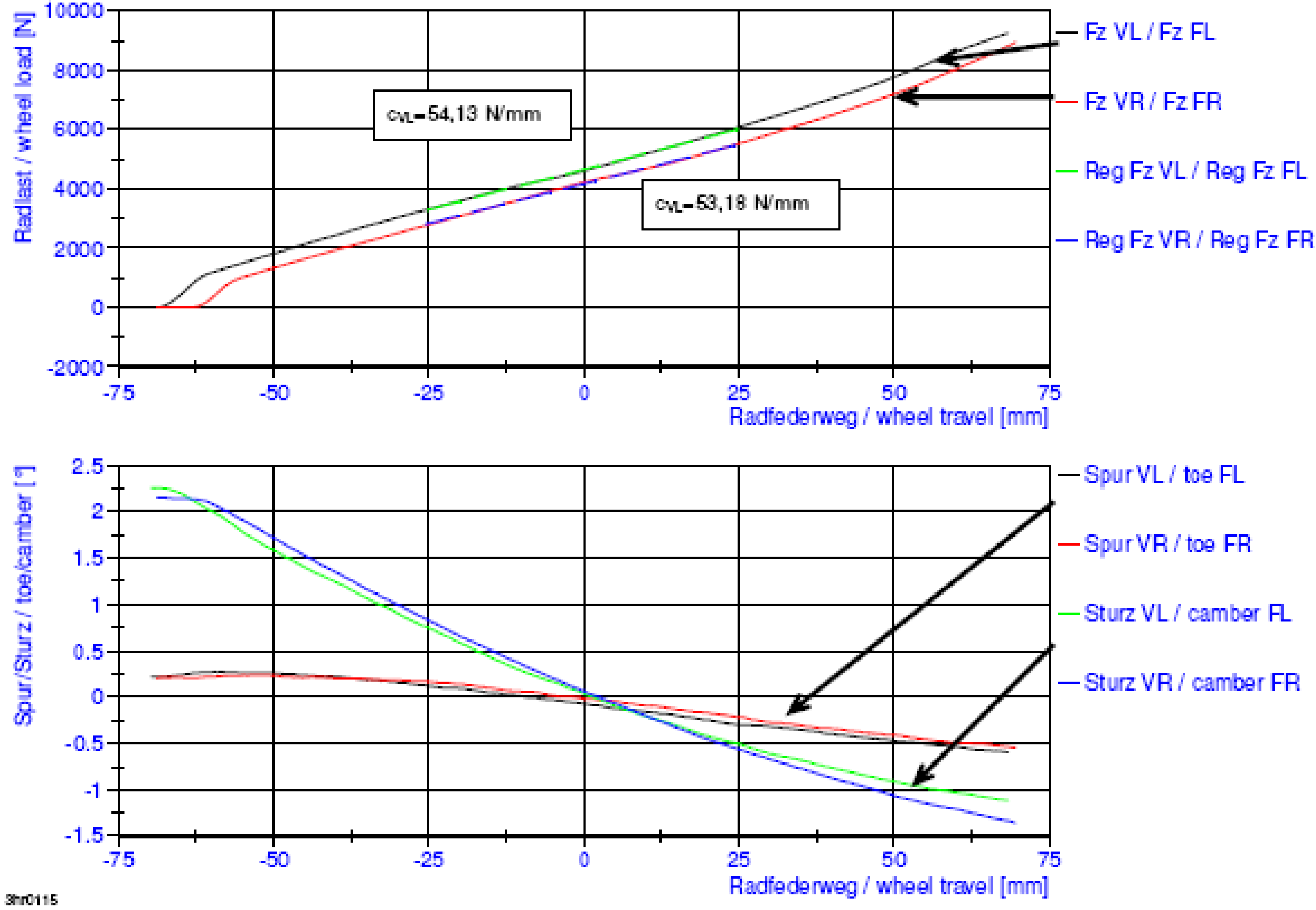
Desired values for the jounce and rebound measurement



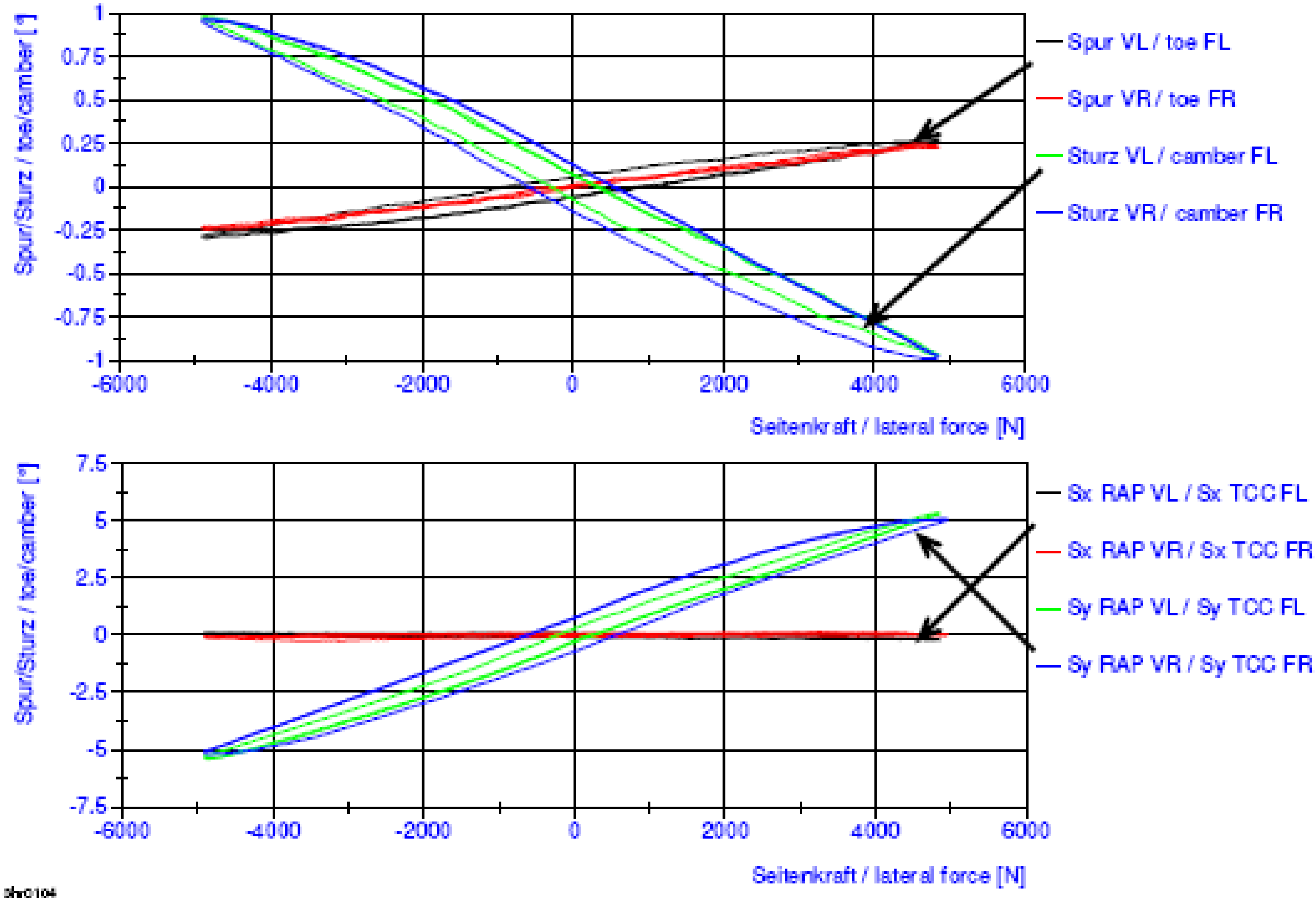
Desired values of the lateral / longitudinal force measurement



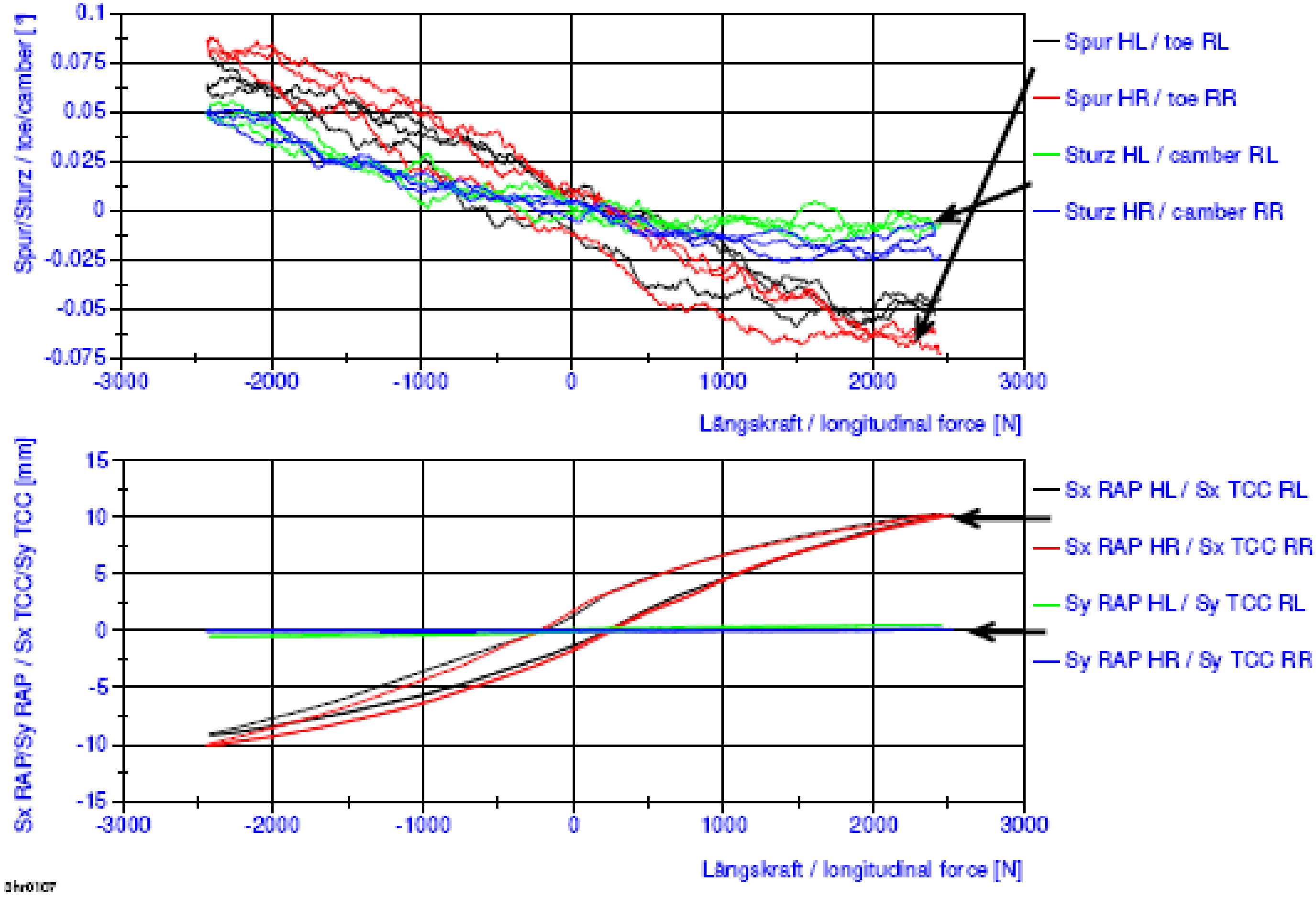
Pic. 5-1: Curve of wheel loads and toe/camber changes during bounce springing on front axis



Pic. 5-2: Curve of wheel loads and toe/camber changes during roll springing on front axis



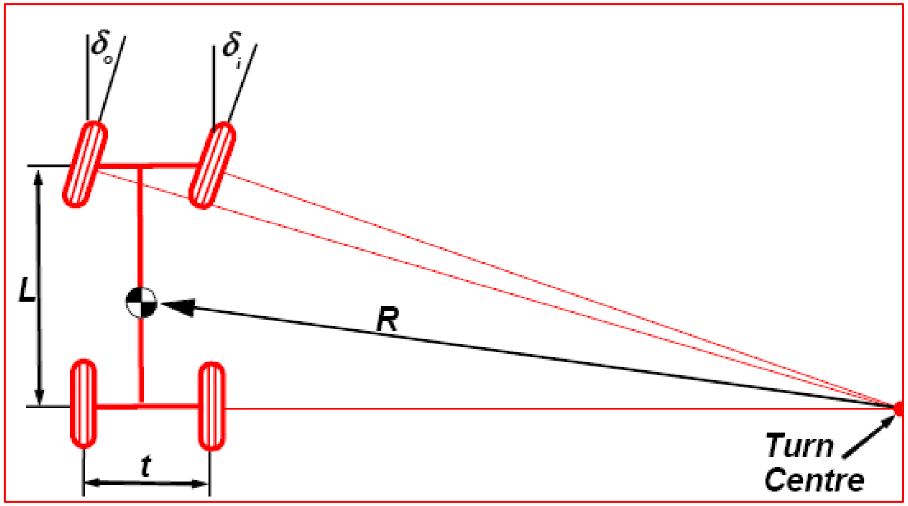
:. 5-3: Toe/camber changes and displacement of the center of tire contact during lateral force initiation



'ic. 5-4: Toe/camber changes and displacement of the center of tire contact during longitudinal force initiation

Steering System

- The steering system must performs important functions like
  - Provide precise control of the front-wheel direction
  - Jolts from irregularities in the road surface must be damped as much as possible during transmission to the steering wheel. However, such damping must not cause the driver to lose contact with the road
  - The basic design of the steering kinematics must satisfy the Ackermann conditions



At Low speeds

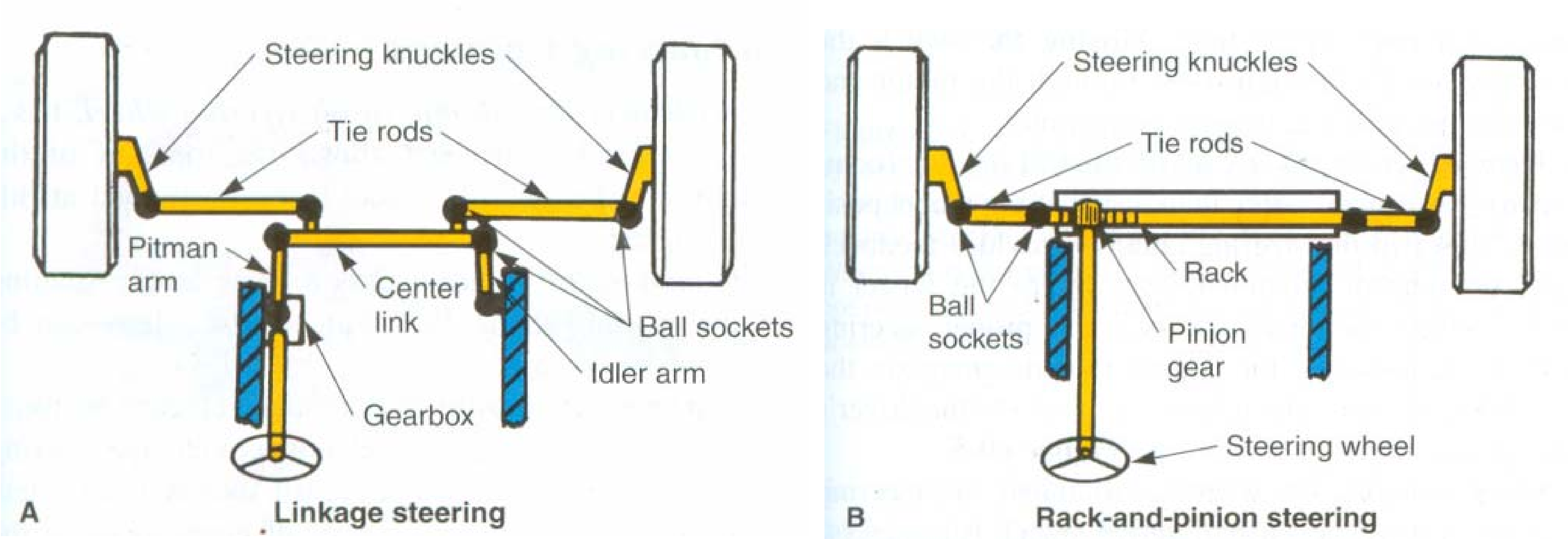
$$Cot\delta_0 - Cot\delta_i = \frac{t}{L}$$

Steering System

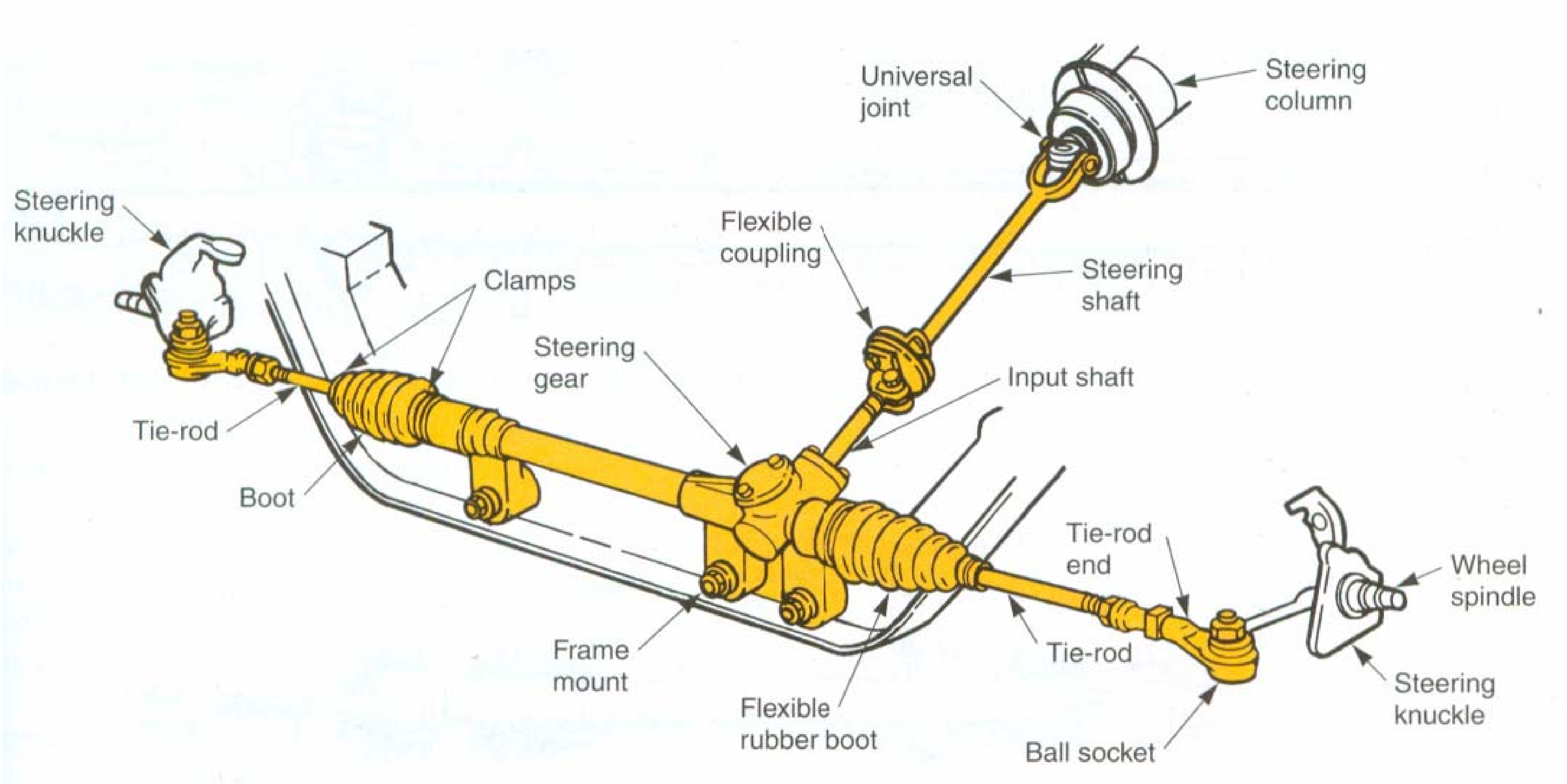
- The vehicle must react to minute steering corrections by means of suitable directness of the steering system and must transmit road feel (slight steering wheel pull caused by road surface) to the driver's hands
- When the steering wheel is released, the wheels must return automatically to the straight-running position and must remain stable in this position
- The steering should have as low a ratio as possible (number of steering-wheel turns from lock to lock) in order to achieve ease of handling. The steering forces involved are determined not only by the steering ratio but also by the front suspension load, the turning radius, the suspension geometry (caster angle, kingpin angle, kingpin offset) and the properties of the tire tread and road surface

Basic Steering Systems

- There are two basic kinds of steering systems in wide use today:
  - Linkage (worm gear) steering systems
  - Rack-and-pinion steering systems
- They may be operated manually or with power assist

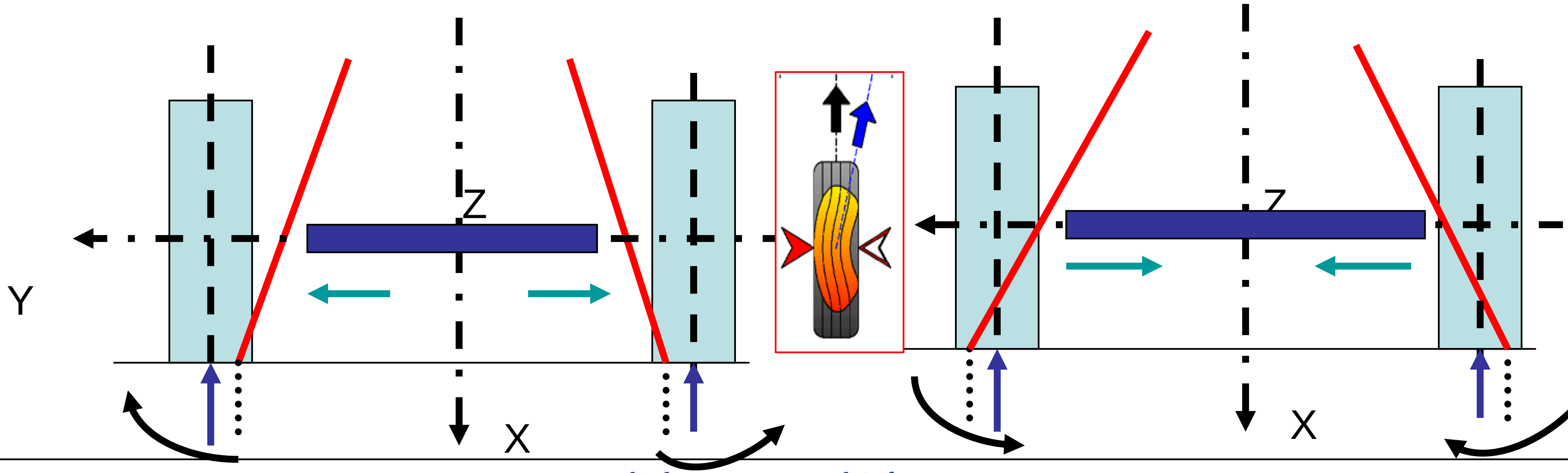


Steering System Nomenclature



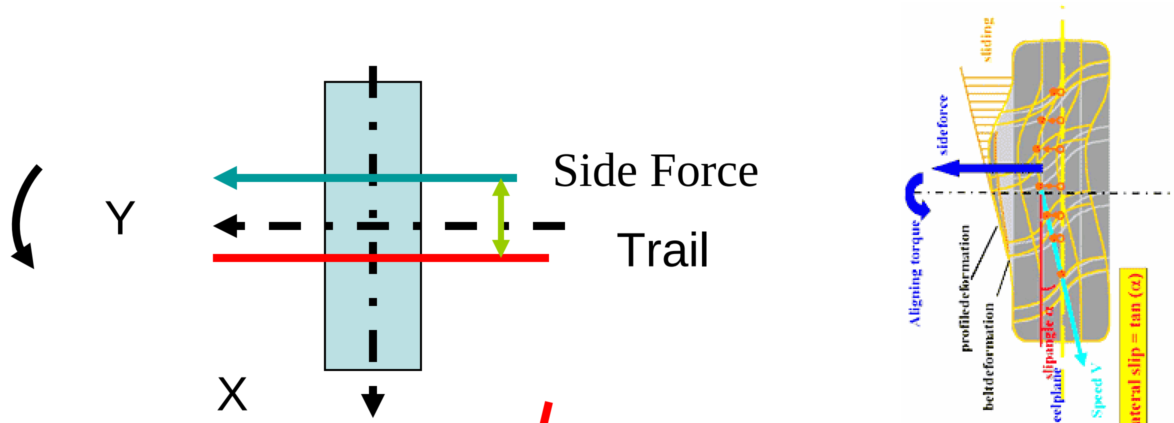
Steering Kinematics

- Steering kinematics and axle design must be such that, although the driver receives feedback on the adhesion between wheels and road surface, the steering wheel is not subjected to any forces from the spring motion of the wheels or from motive forces (front wheel drive)
  - Steering –Axis inclination causes the front section of the vehicle to lift when the wheels are at an angle. This leads to a caster dependent on the steering angle
  - Toe-in (toe-out) is a slip angle present even during straight running travel. This tensions the linkages and causes a rapid build-up of transverse forces when the wheels are at an angle

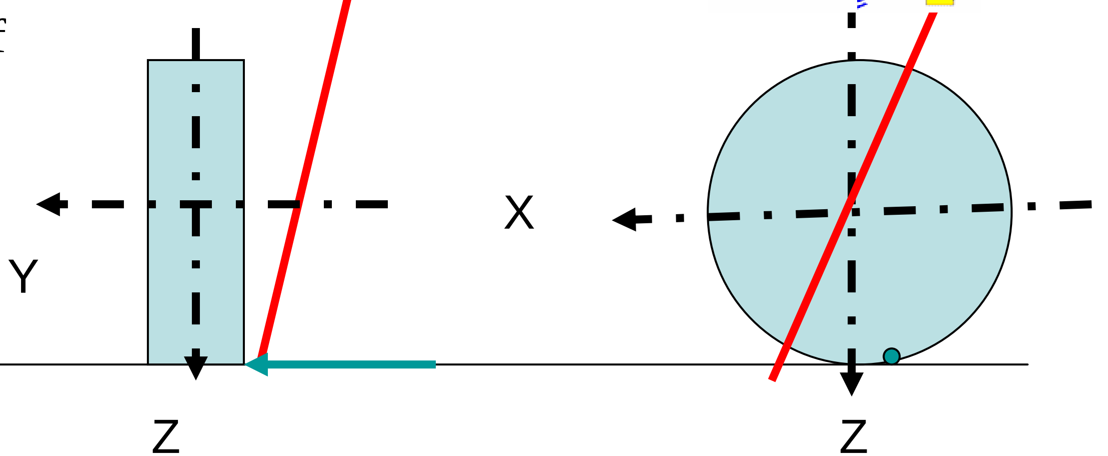


Steering Kinematics

- Caster produces a lever arm for side forces, i.e. speed dependent return torque (Alignment Torque)
- Kingpin offset determines the extent to which the steering system is affected by interference factors :brakes pulling unevenly, motive forces under traction/overrun in front wheel drive vehicles. In modern designs, the aim is to achieve a steering offset which is zero to slightly negative

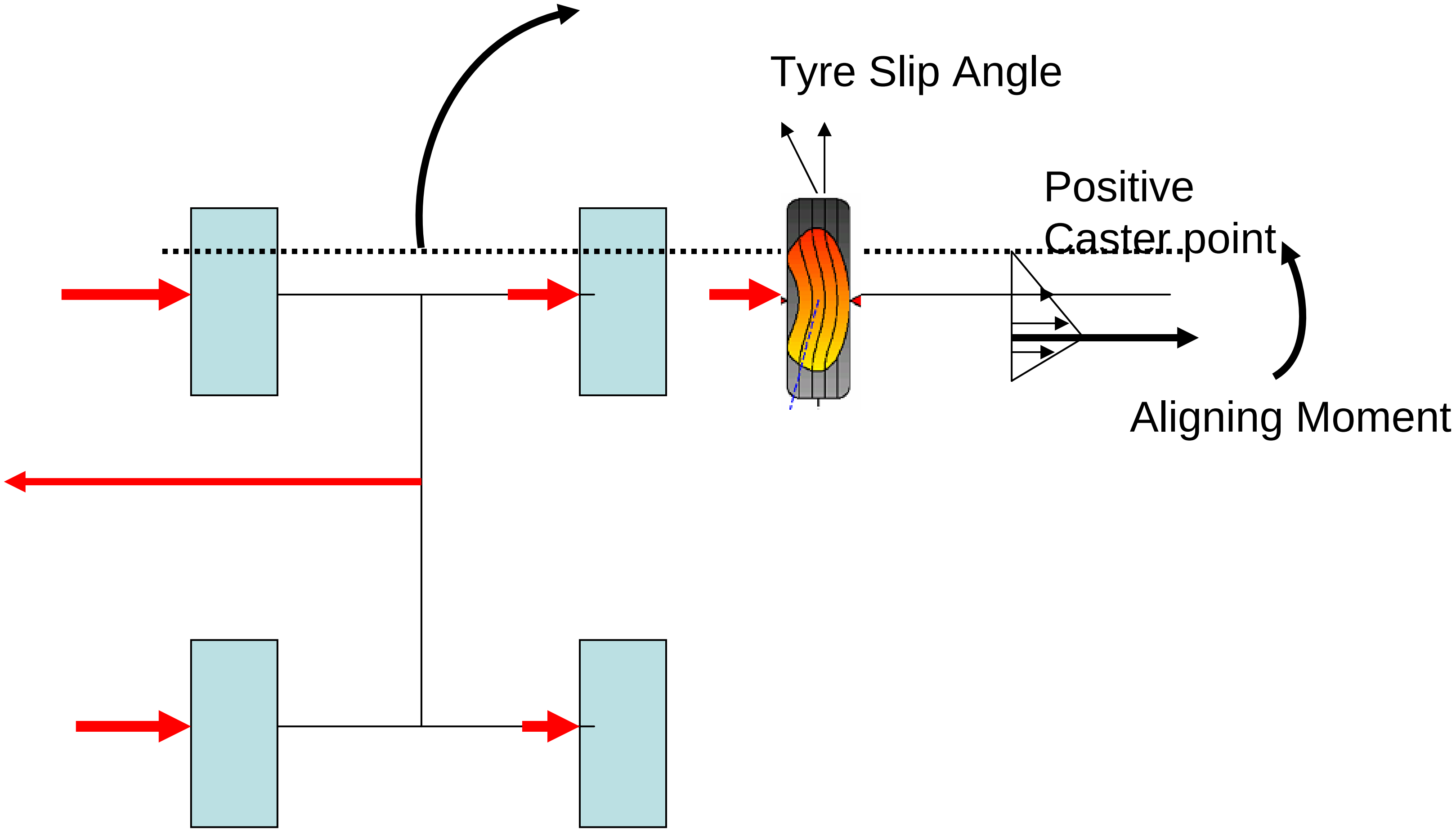


Side force produces self aligning torque about Z axis



Trail Varies as caster varies, hence self aligning torque

Caster Change and Aligning Moment

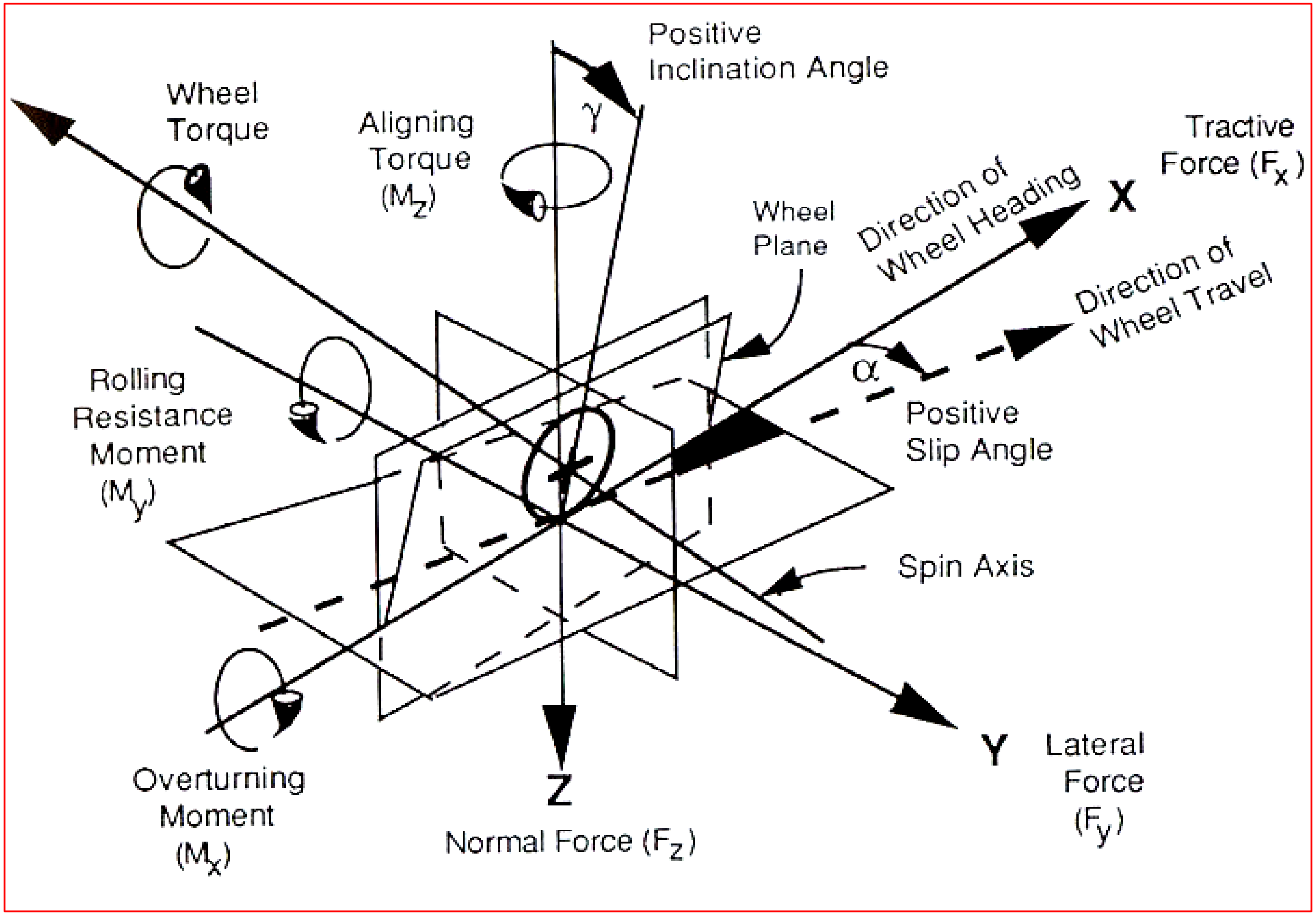


Steering System Forces and Moments

- The ground reactions on the tire are described by three forces and moments, as follows:
- Normal force — Aligning torque
- Tractive force — Rolling resistance moment
- Lateral force — Overturning moment
- On front-wheel-drive cars, an additional moment is imposed by the drive torque.

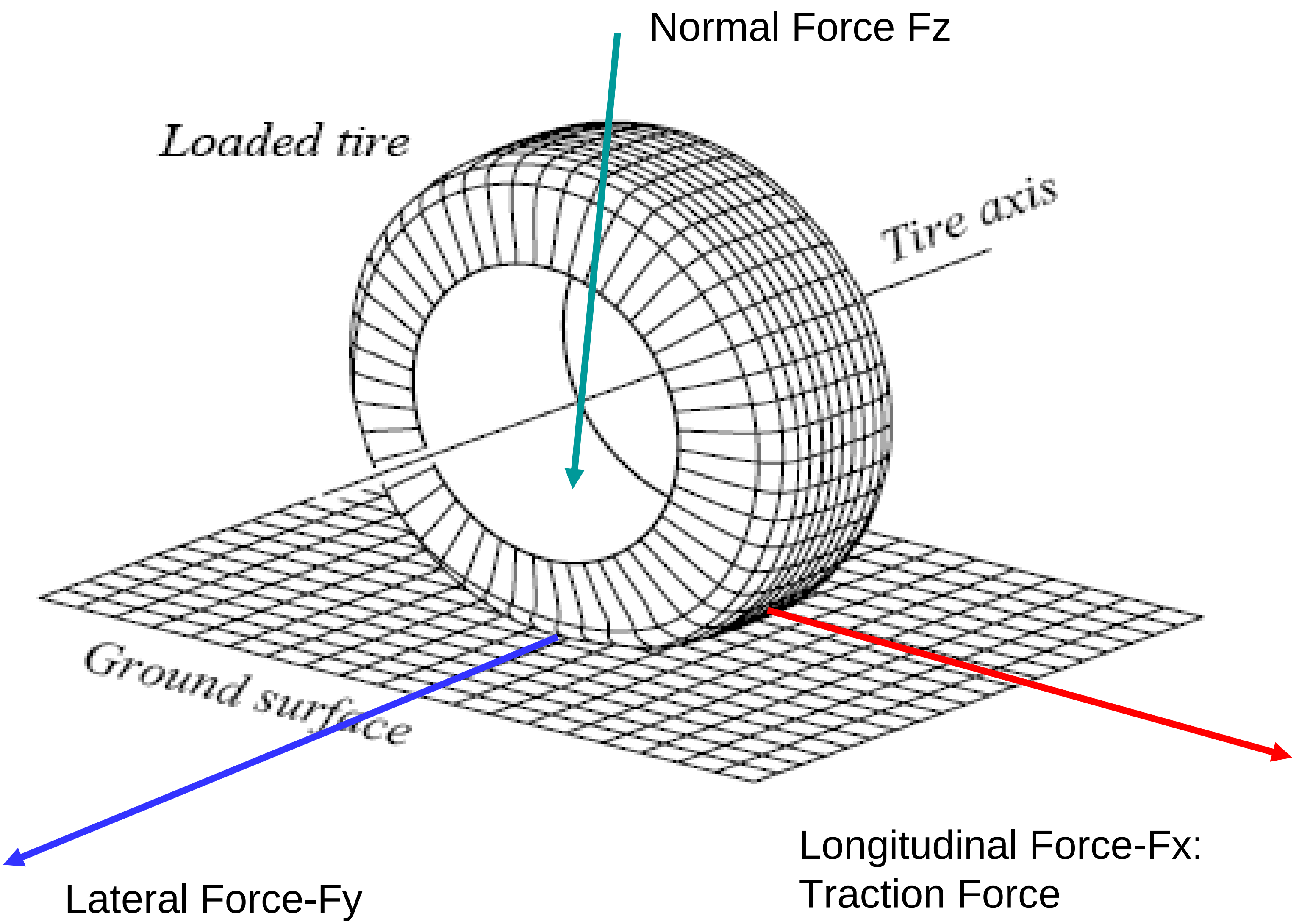
Steering System Forces and Moments

- The forces and moments imposed on the steering system emanate from those generated at the tire-road interface.

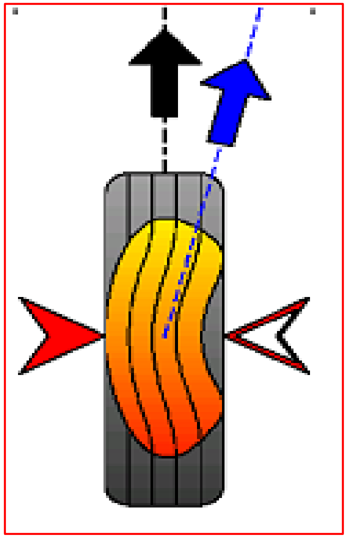


Steering Wheel Torque

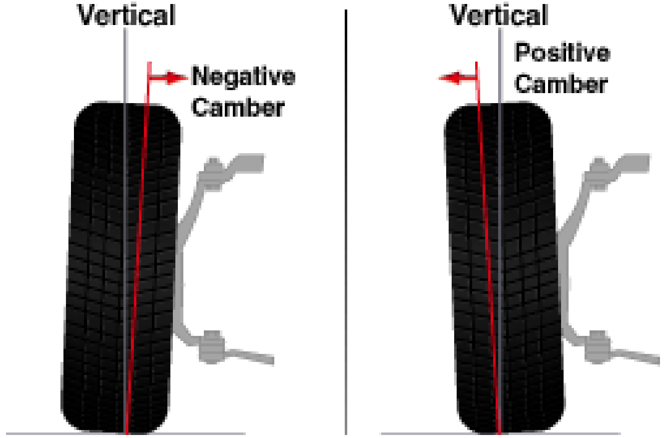
- The reaction in the steering system is described by the moment produced on the steer axis, which must be resisted to control the wheel steer angle.
- The sum of moments from the left and right wheels acting through the steering linkages with their associated ratios and efficiencies account for the steering-wheel torque feedback to the driver.



Normal Force Fz
Loaded tire
Tire axis
Ground surface
Lateral Force-Fy
Longitudinal Force-Fx:
Traction Force

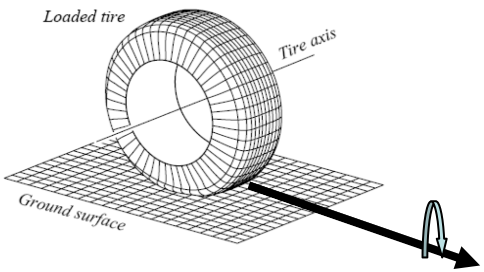


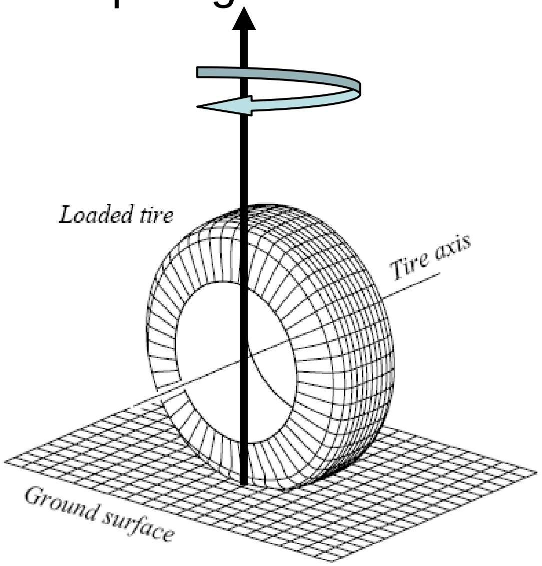
Slip Angle

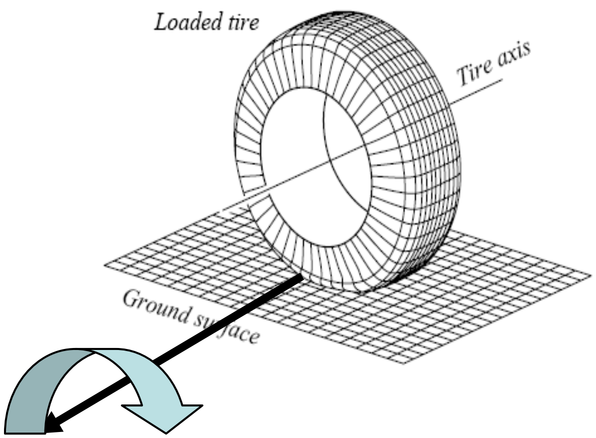


Camber

Overturning Moment Mx



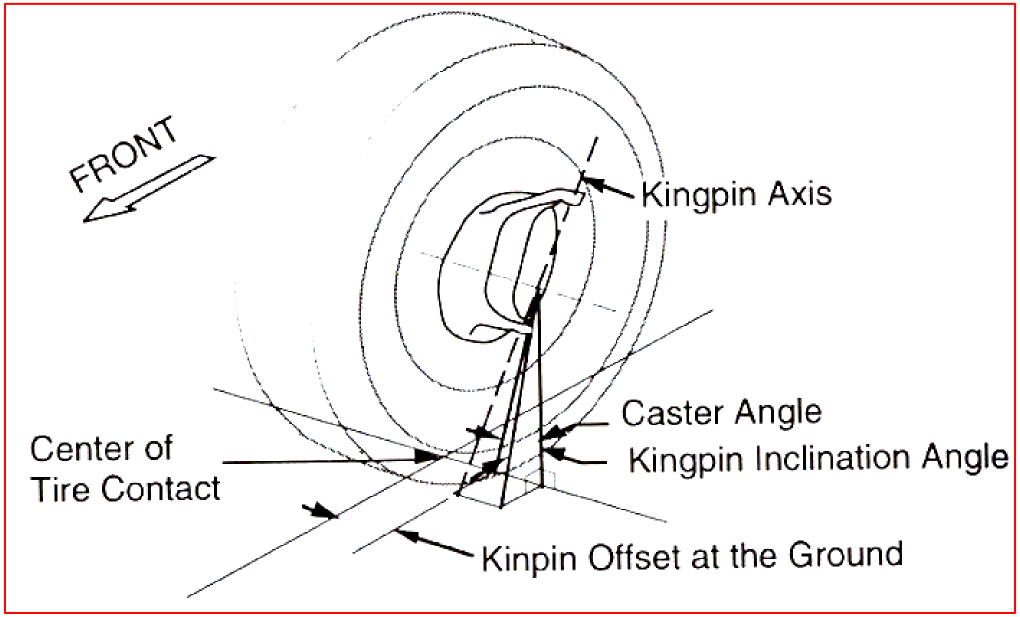




Aligning Moment (Mz)

Rolling Resistance Moment (My)

Estimation of Steering Forces and Moments



M_v= Total moment from left and right wheels

F_{zl}, F_{zr} =Vertical load on left and right wheels

d= Lateral offset at the ground

λ= Lateral Inclination Angle (KPI)

δ= Steer Angle

υ=Caster Angle

Steering Forces and Moments

- The moment arising from vertical force acting

$$M_v = -(F_{zl} + F_{zr})d \sin\lambda \sin\delta + (F_{zl} - F_{zr})d \sin\upsilon \cos\delta$$

- The moment arising from lateral force

$$M_L = (F_{yl} + F_{yr})r \tan\upsilon$$

- The moment arising from traction force

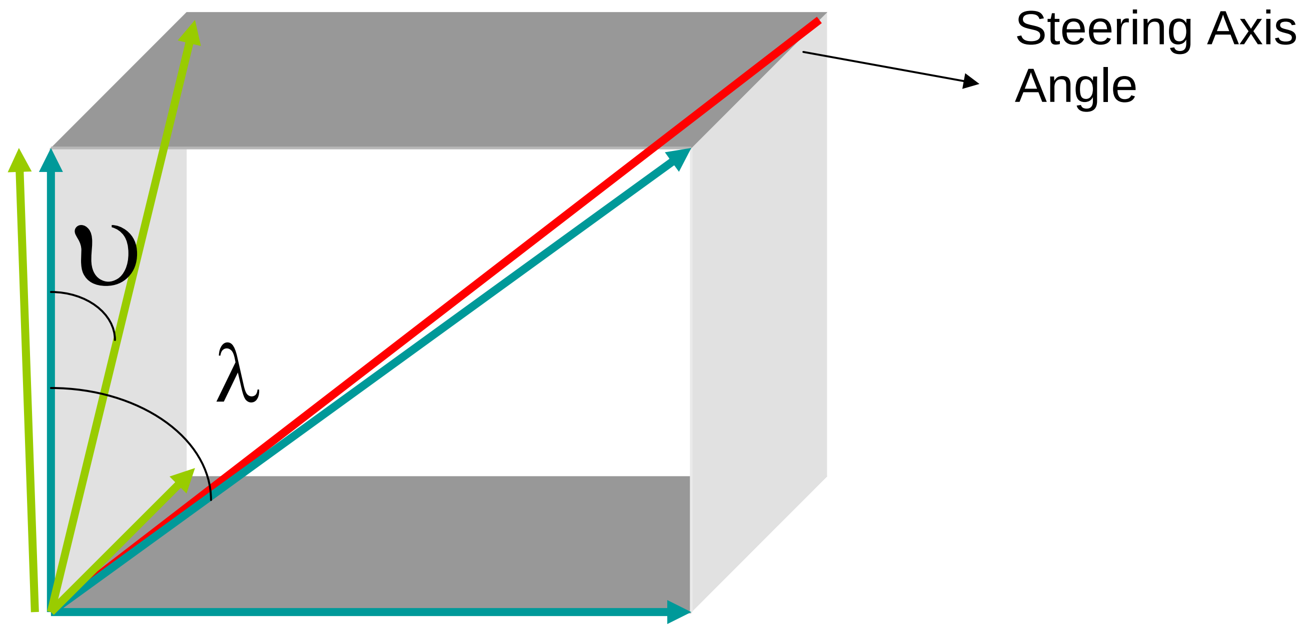
$$M_T = (F_{Xl} - F_{Xr})d$$

- Aligning Torque

$$M_{AT} = (M_{Zl} + M_{Zr})\cos\sqrt{\lambda^2 + v^2}$$

- Rolling resistance and Overturning moments have second order effect and are neglected

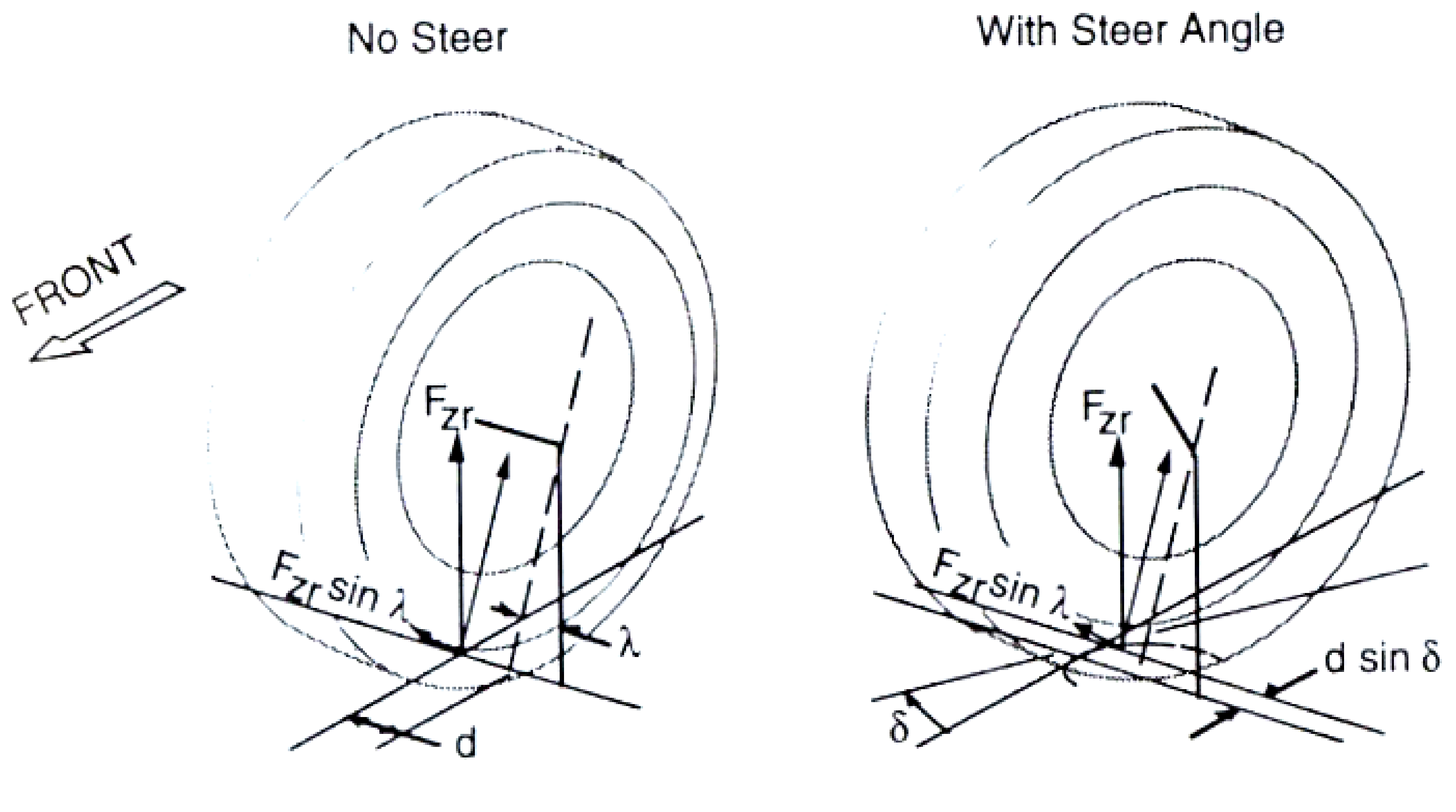
Kingpin Inclination



λ= Lateral Inclination Angle (KPI)

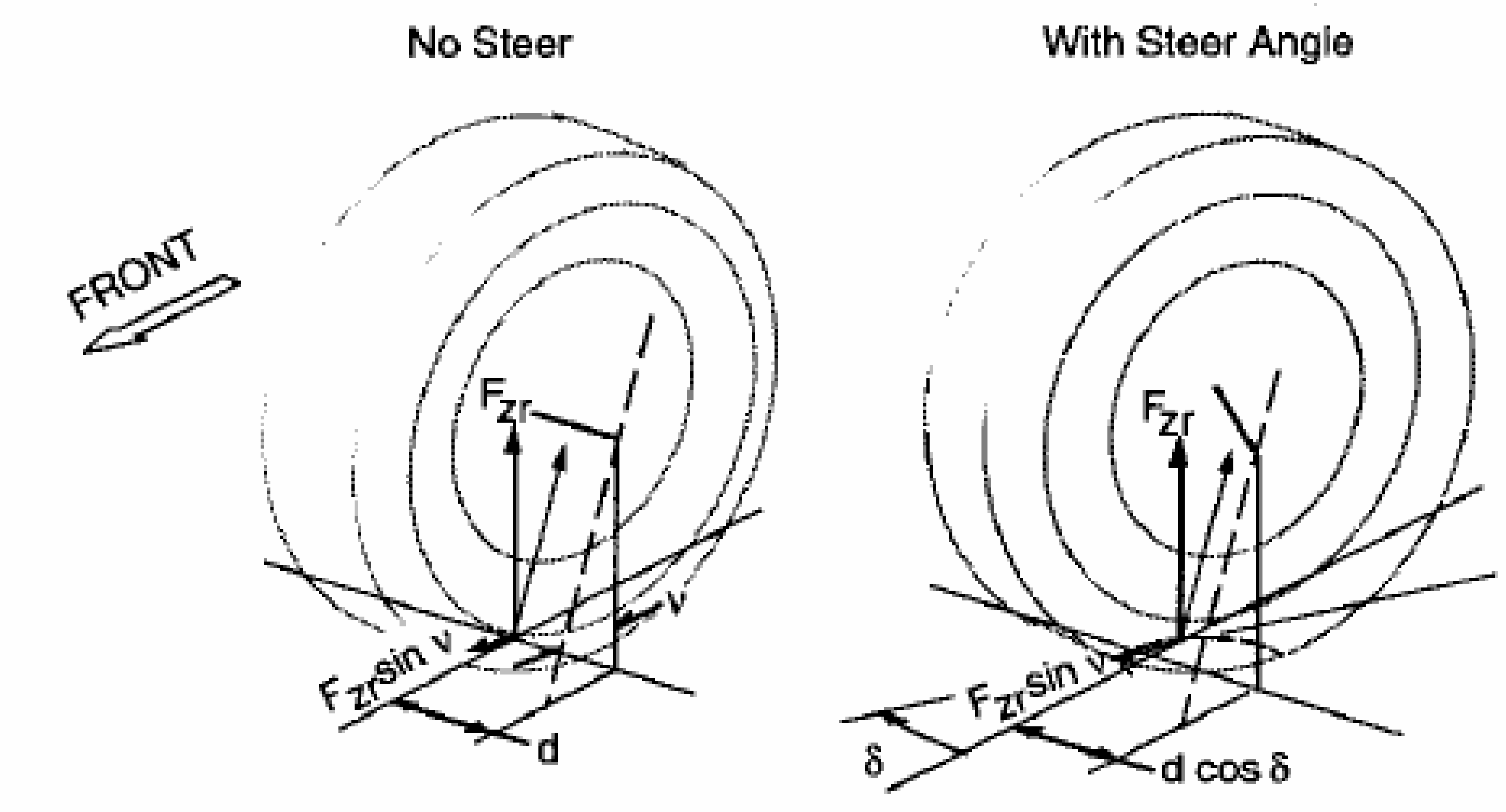
υ=Caster Angle

Moment produced by Vertical Forces acting on Lateral Inclination Angle



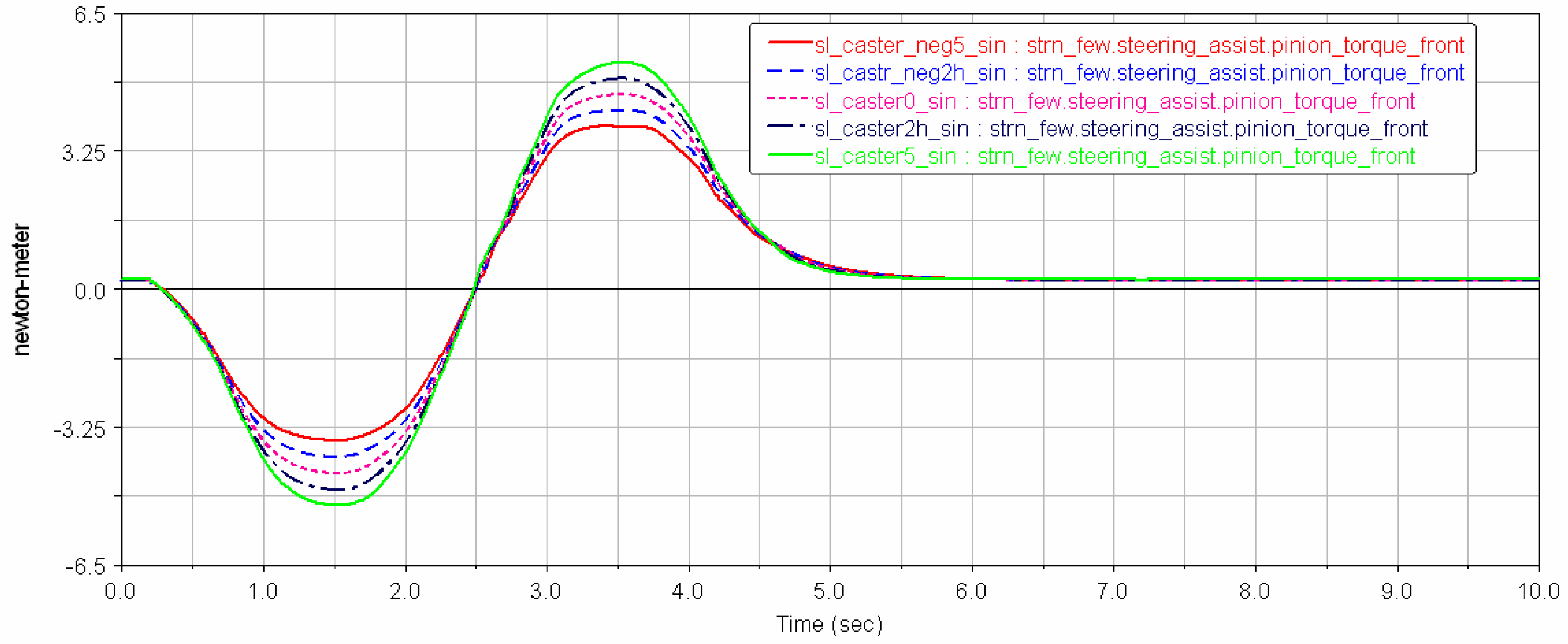
$$M_{V\lambda} = -(F_{zl} + F_{zr})d \sin\lambda \sin\delta$$

Moment Produced by Vertical Force Acting on Caster Angle



$$M_{V\nu} = (F_{zl} - F_{zr})d \sin \upsilon \cos \delta$$

Steering Torque Variation with Caster

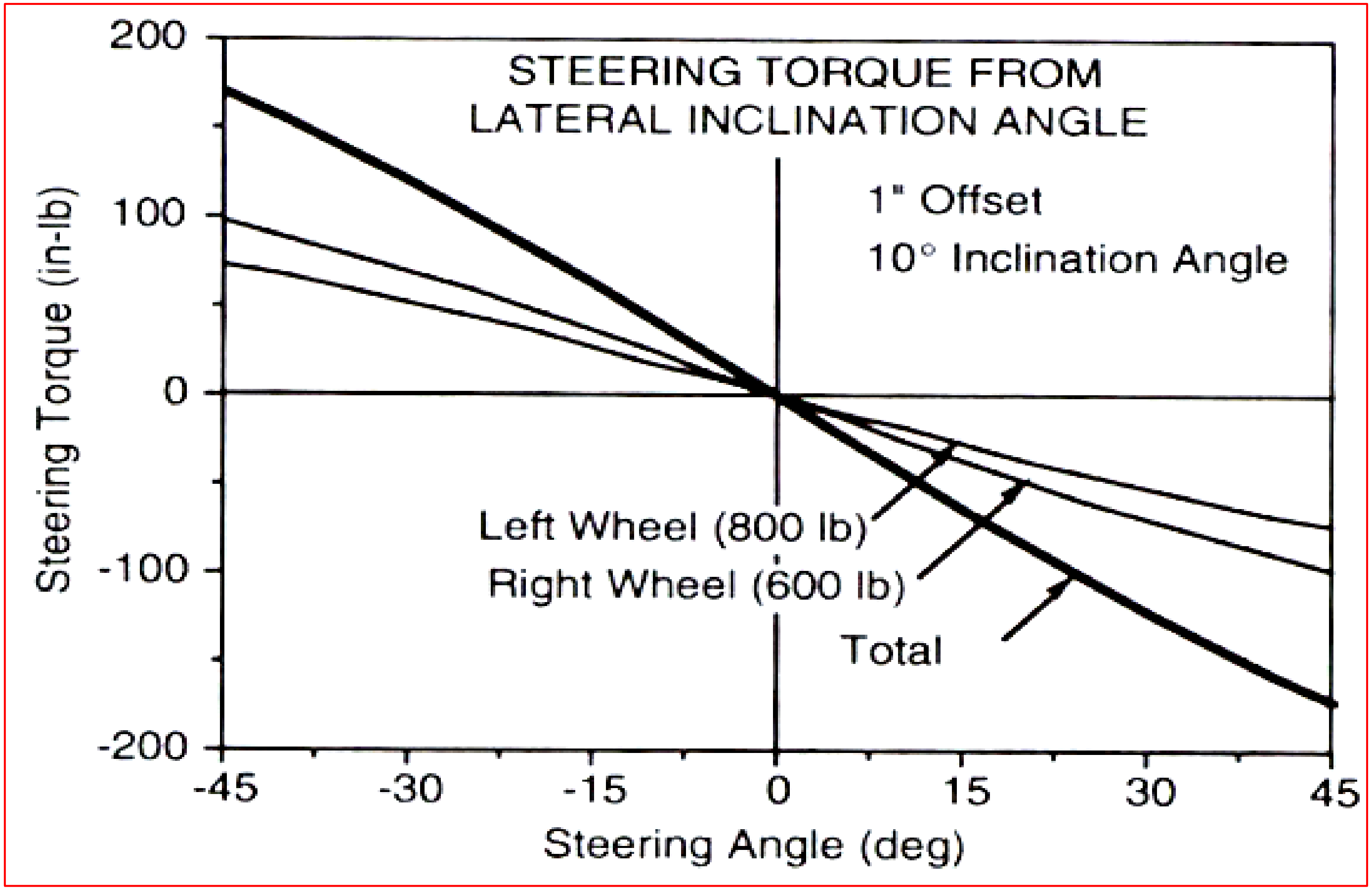


Steering Torque high for positive caster and low for negative caster

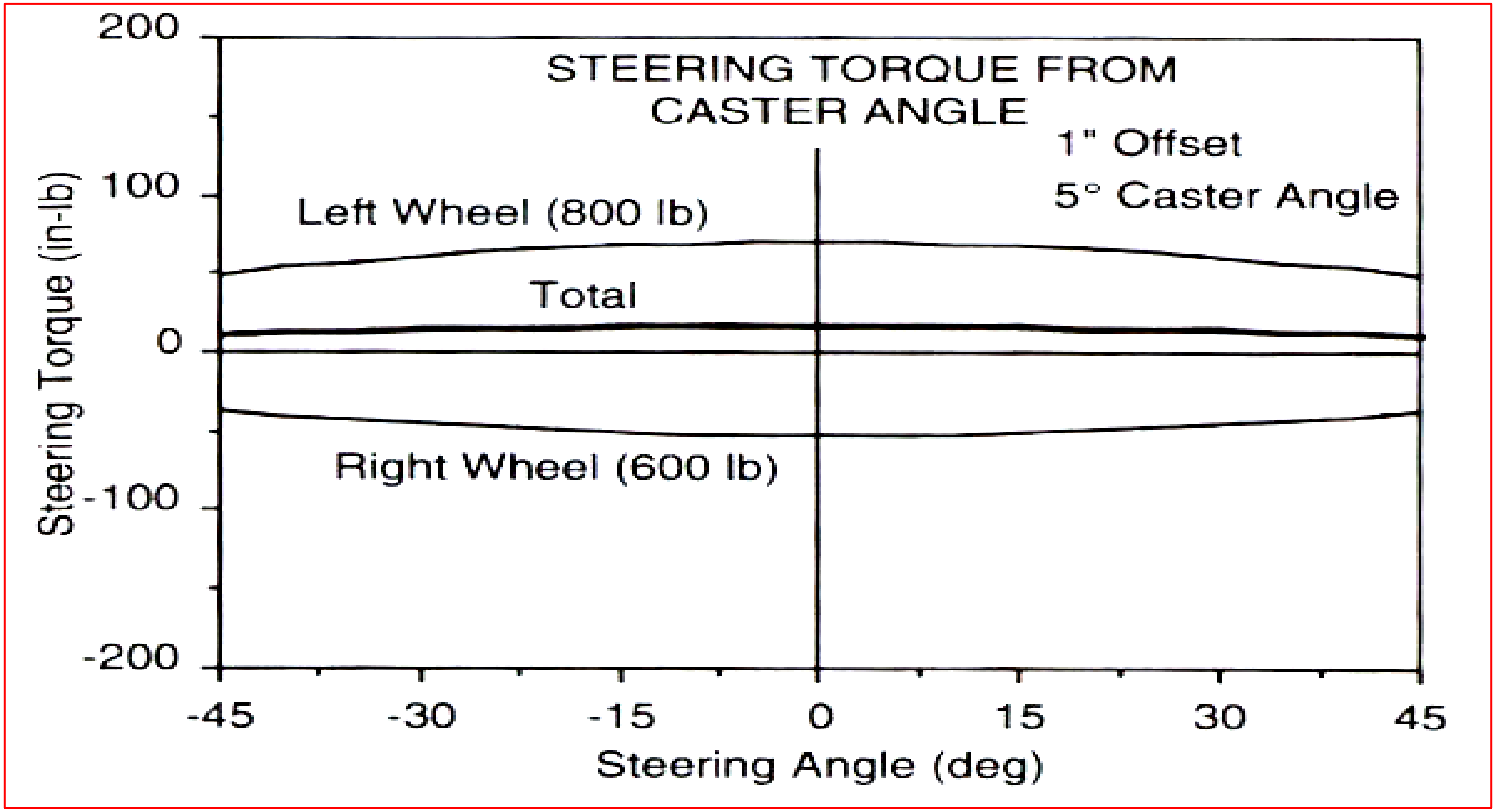
- The moment arising from vertical force acting

$$M_v = -(F_{zl} + F_{zr})d \sin\lambda \sin\delta + (F_{zl} - F_{zr})d \sin\upsilon \cos\delta$$

Steering Torque

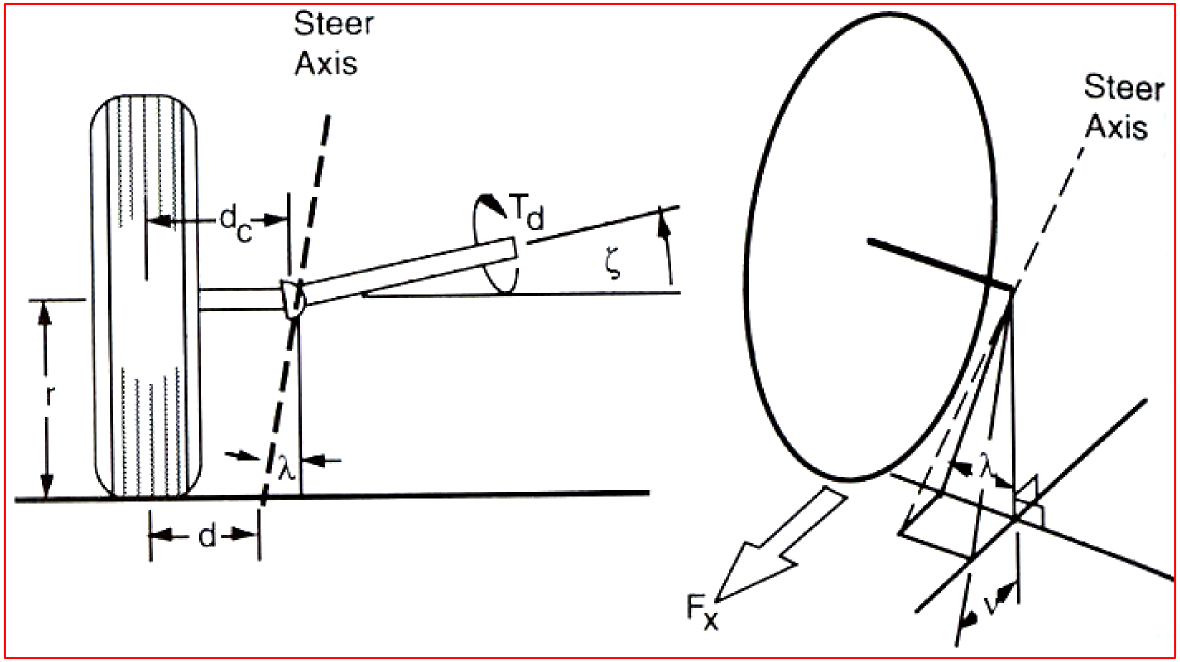


Steering Torque



Moment about SA due to drive line Torque

- The torque in the driveline produces a moment about the steer axis.



$$T_d = F_X r$$

$$M_{SA} = F_X [d \cos \nu \cos \lambda + r \sin(\lambda + \zeta)]$$

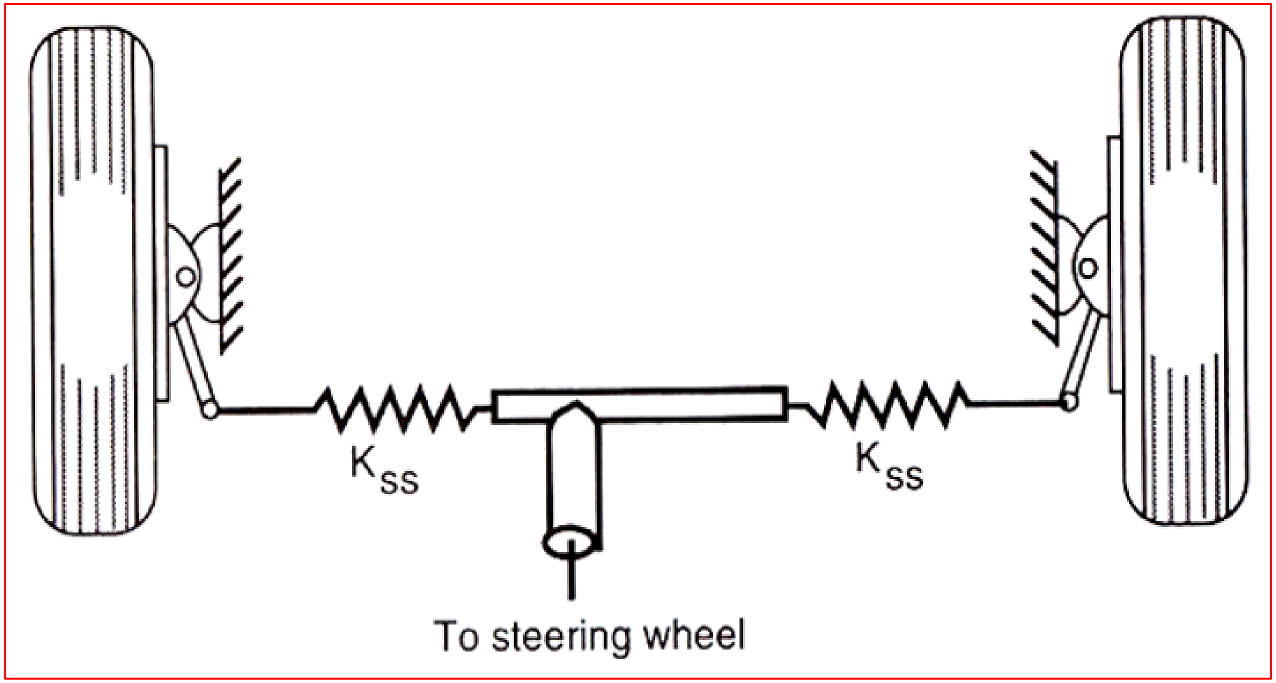
The lateral inclination and caster are small enough that the cosine function can be assumed unity. Hence

$$M_{SA} = F_X [d + r \sin(\lambda + \zeta)]$$

The forward force introduces a moment in the steering system which opposes the steer angle trying to steer the vehicle out of turn-make the vehicle understeer. That is hwy in Front wheel drive vehicles toe out is provided

Steering System Models

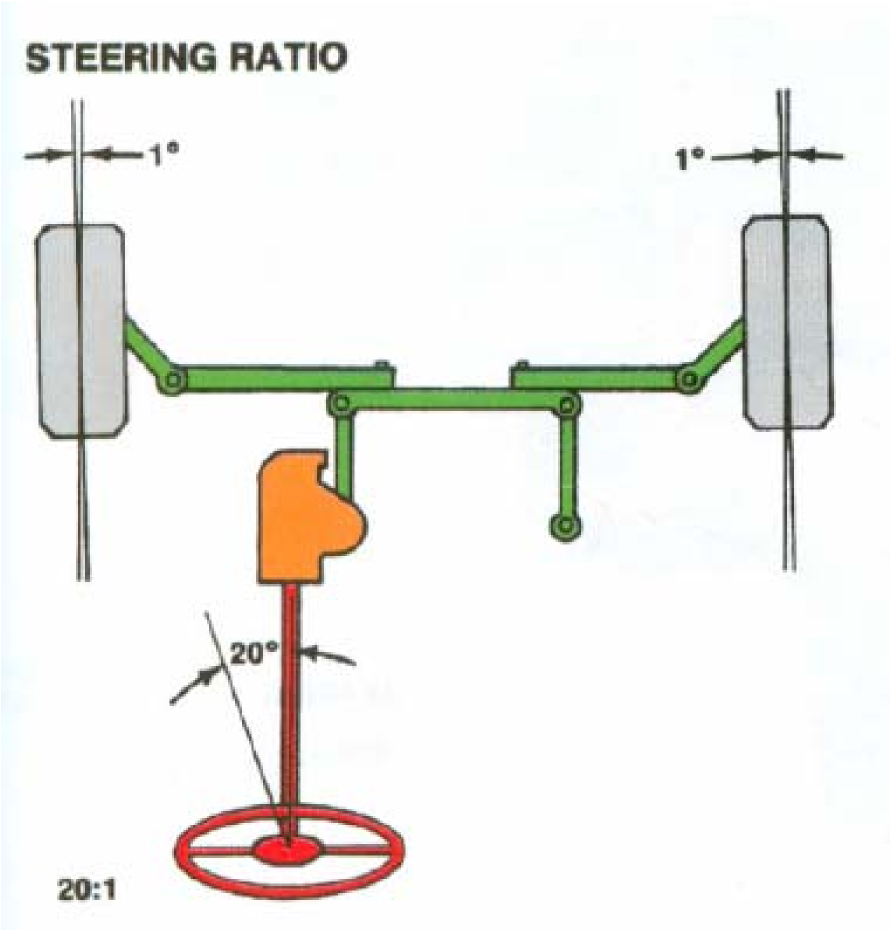
- The effects can be modeled and torque feedback to the steering wheel can be determined



Steering System Performance Measures

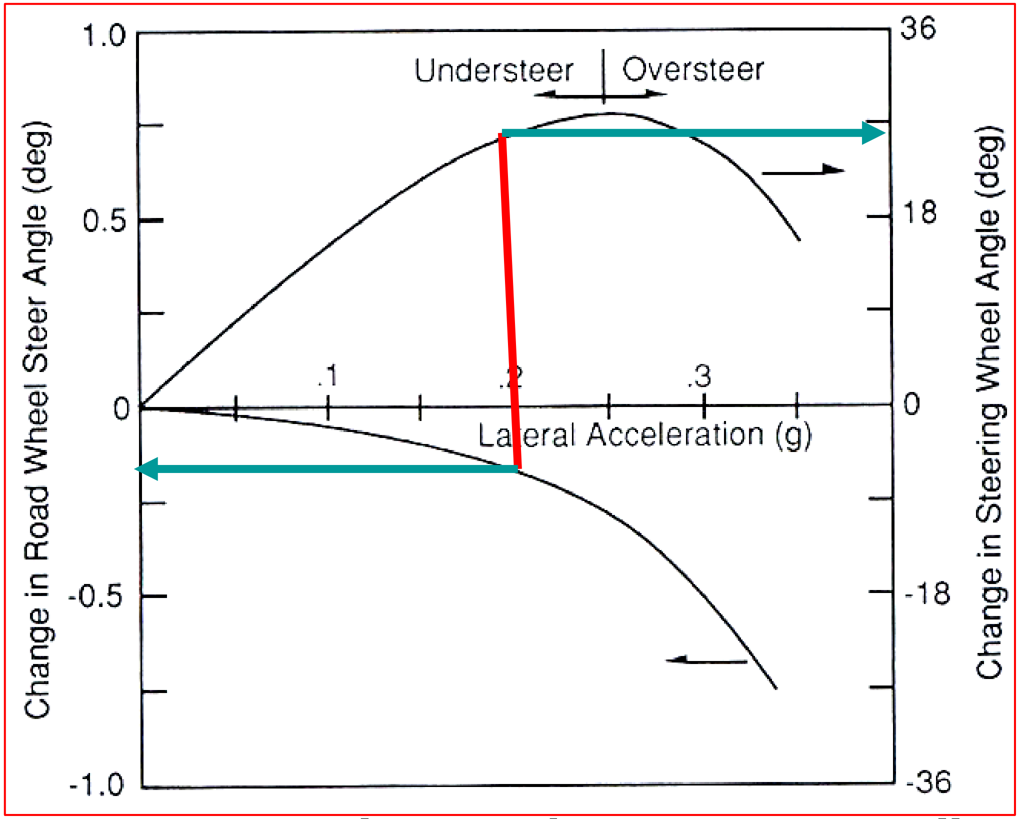
- The specific design of a steering system geometry has a well-recognized influence on steering performance measures such as
  - Center feel,
  - Returnability,
  - Steering efforts
  - Steering ratio to cornering
  - Steering ratio to braking

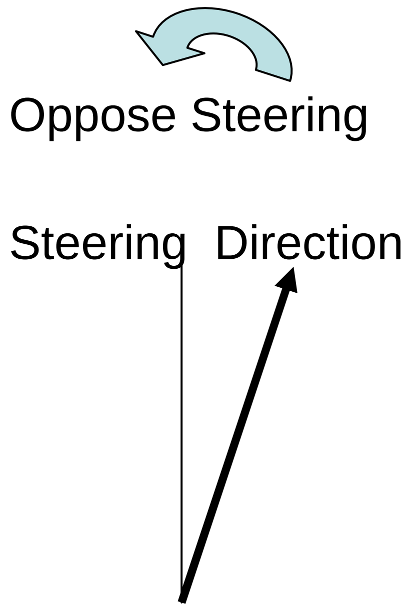
Steering Ratio



- The steering ratio is defined as the ratio of steering wheel rotation angle to steer angle at the road wheels.
- Normally these range from 15 or 20 to 1 on passenger cars, and 20-36 to 1 on trucks.
- Steering ratio allow for easy steering of the front wheels. Low ratios such as 12:1 give quick but stiff steering where as high ratios such as 20:1 provide slow but easier steering
- Because of the compliance and steer torque gradients with increasing steer angles, the actual steering ratio may be as much as twice the designed ratio
- The compliance property is constant on a vehicle, the torque gradient will vary with load on the front tires, tire type, pressure, coefficient of friction, etc. Hence, the actual steering ratio may vary (always exceeding the design value) and influencing the low-speed maneuverability of the vehicle.

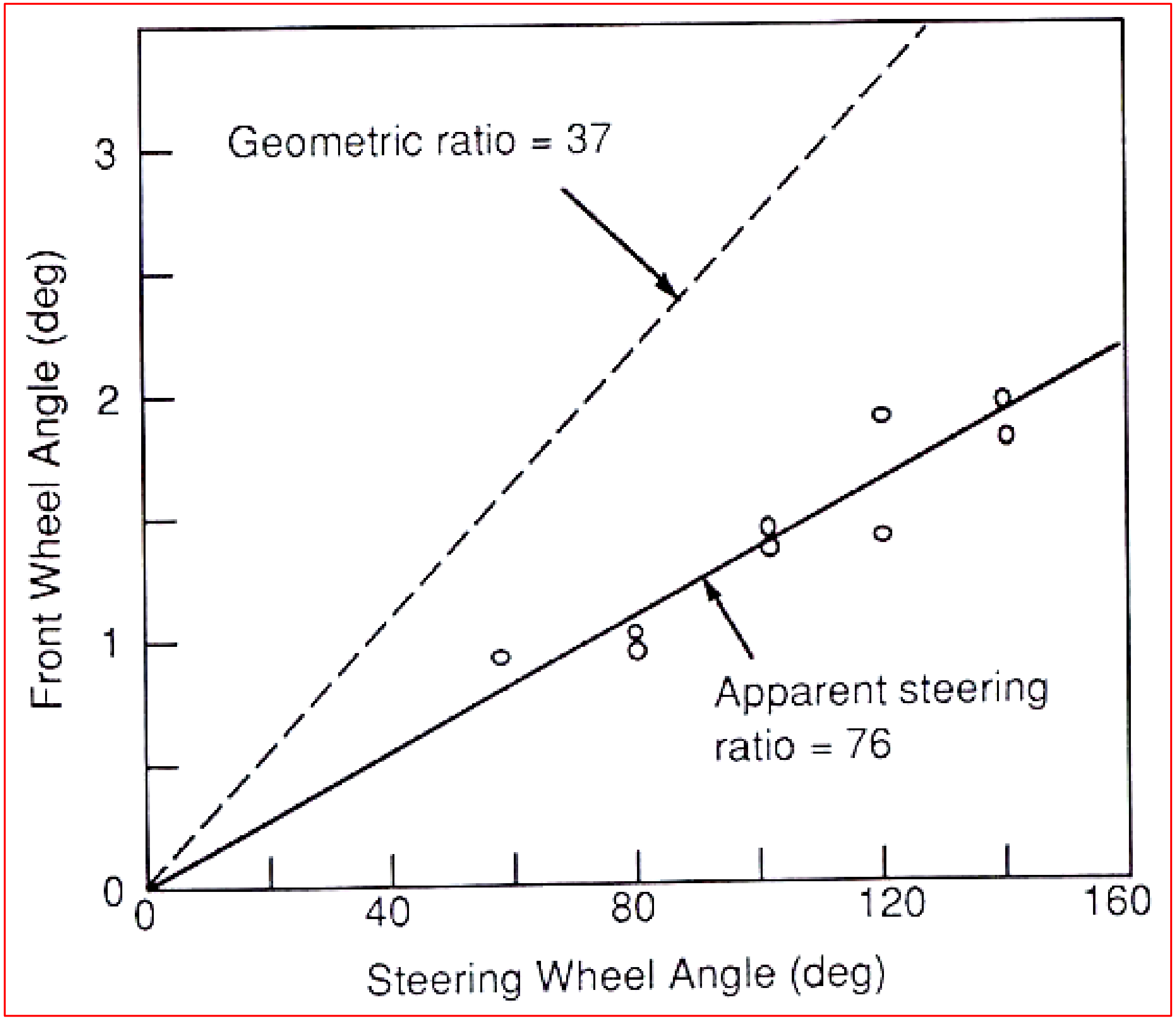
Understeer





- Because compliance in the steering system allows the road wheels to deviate from the steering wheel input, the results obtained are influenced by the steering system properties
- The deviation is equivalent to an apparent gradient of 4 degrees/g at the road wheel
- The difference arises from deflections in the steering linkage as the reactions on the road wheels act against the steering compliances

Steering Ratio

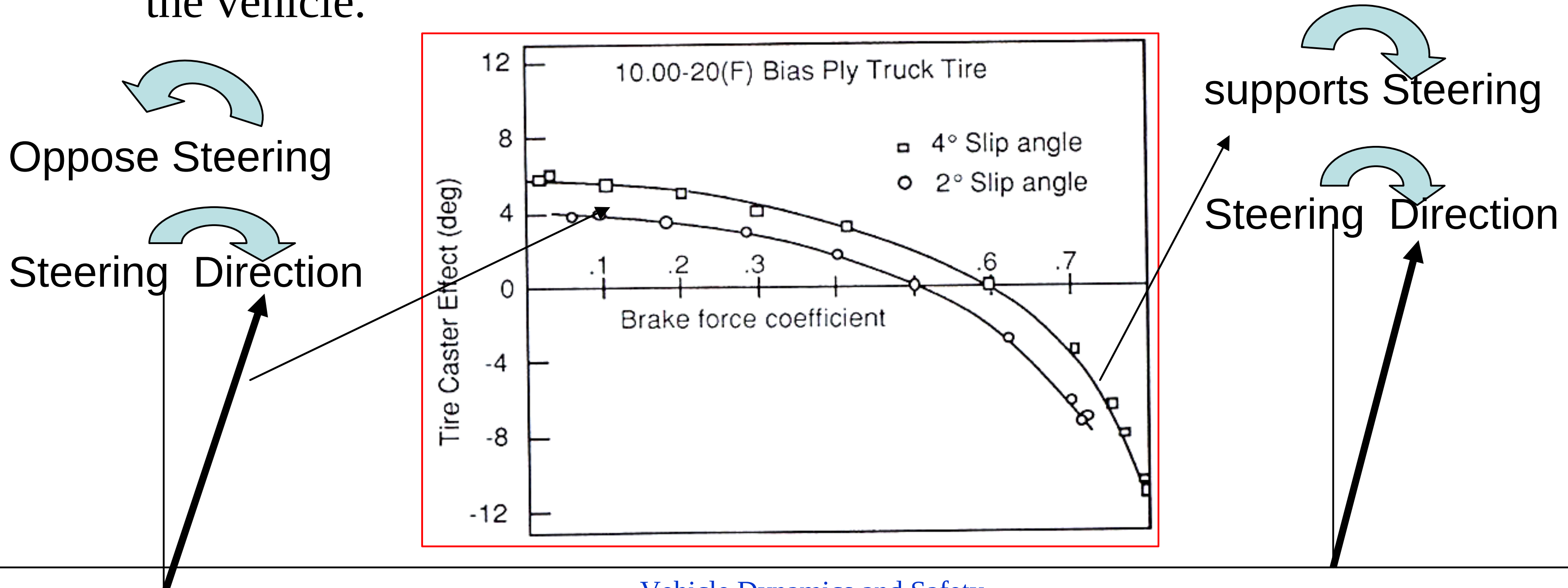


Because of the compliance and steer torque gradients with increasing steer angles, the actual steering ratio may be as much as twice the designed ratio.

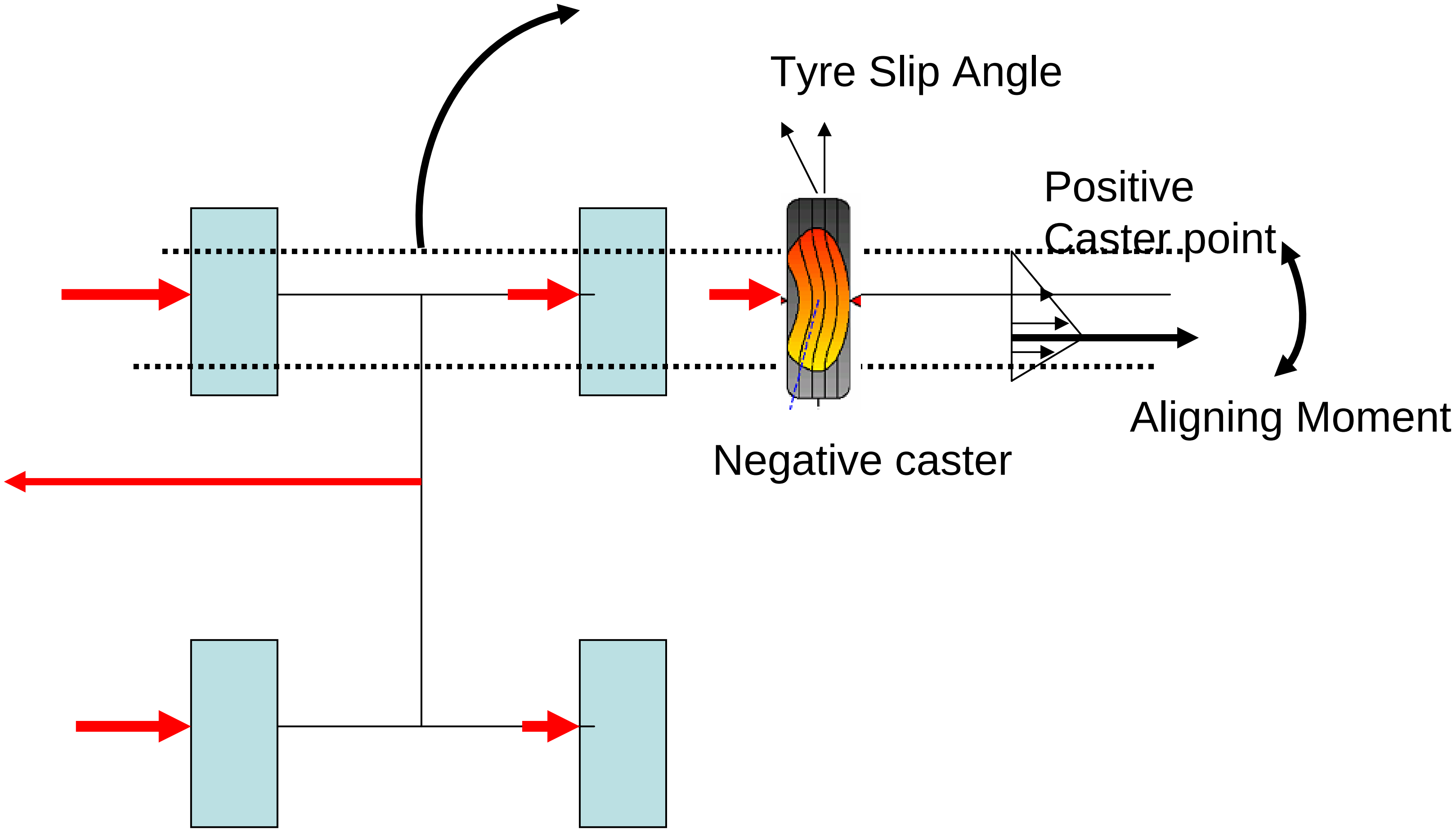
Fig shows experimental measurements on a truck which illustrate the phenomenon

Braking Stability

- Braking is a special case in which steering system design plays an important role in directional response.
- The tire aligning torques which effectively act like 4-8 degrees of caster angle (increase pneumatic trail) under free-rolling conditions can reverse in direction during braking
- Brake force imbalance (due to brake malfunction or a split-coefficient surface) will also act on the compliant steering system, attempting to steer the vehicle.



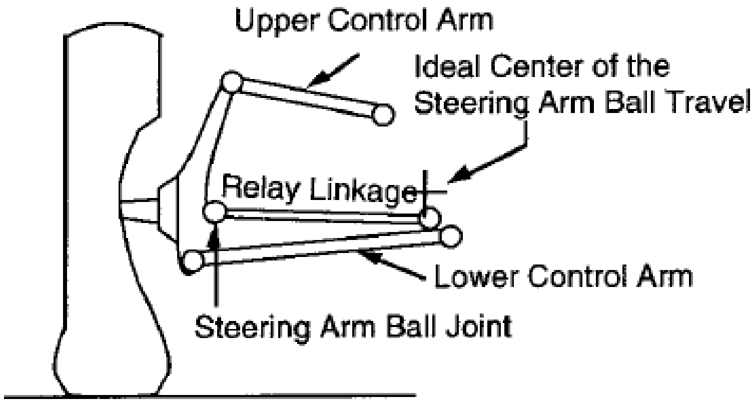
Caster Change and Aligning Moment



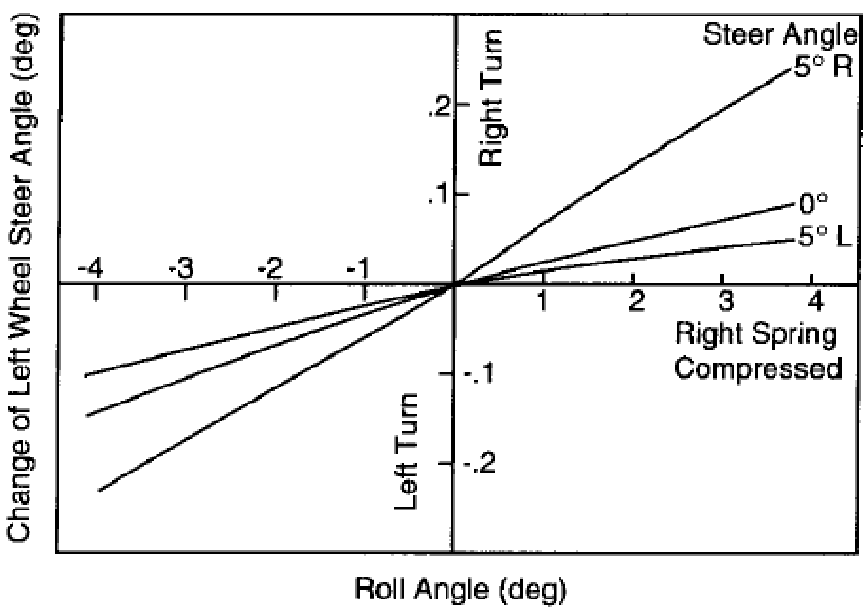
Braking Stability

- At low braking coefficients the aligning torque acts in the direction to steer the tire in its direction of travel, which on the steered wheels attempts to steer the vehicle out of the turn (an understeer influence).
- But at high braking coefficient, the aligning torque reverses direction and may reach elevated negative levels, which will attempt to steer the tire into the direction of turn (an oversteer influence).
- As a result, the normal stabilizing effects of positive caster and tire aligning torque may be substantially reduced or eliminated during high-level braking.

Roll Steer



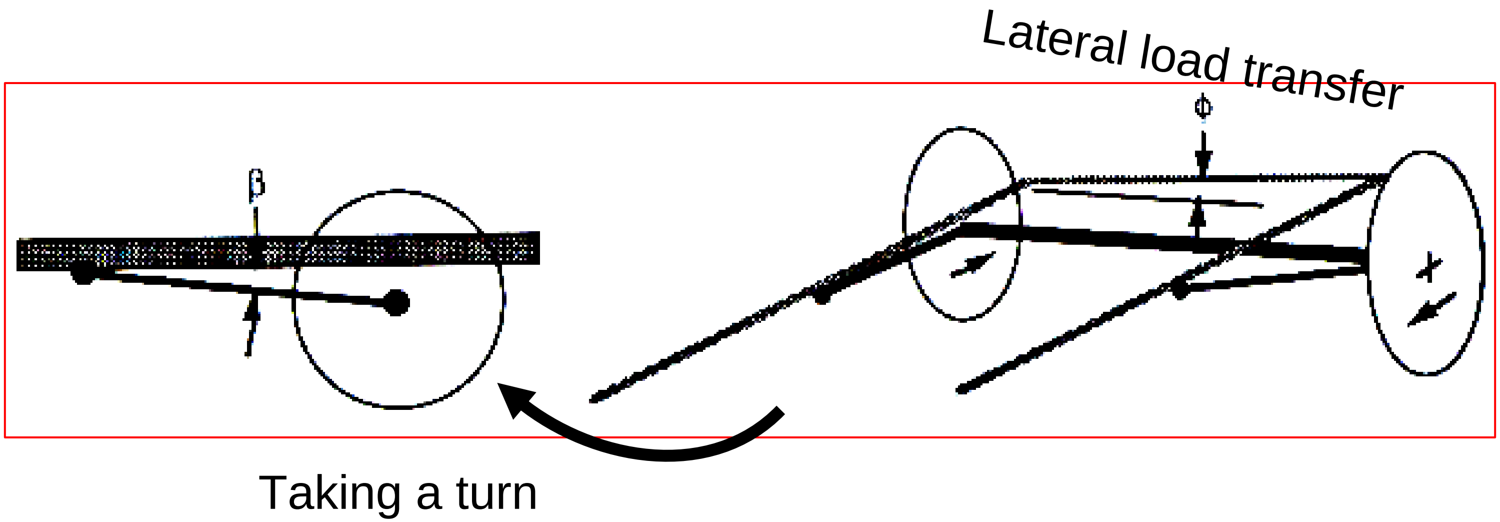
Relay Linkage (Tie Rod) centre is placed below the ideal centre, due to wheel bounce and jounce, the centre point moves up or down steering the wheel.



The slope of the curve is the Roll Steer Coefficient ε

$$K_{roll\ steer} = \varepsilon \frac{d\phi}{d a_y}$$

Roll Steer due to Suspension



Roll Steer and Bump Steer

Difference between Bump Steer and Roll Steer

In a bump steer, both wheels rise together. In roll steer, one wheel rises as the other falls. Typically this produces more "toe in" on one wheel, and more "toe out" on the other, thus producing a steering effect. In a simple analysis you can just assume that the roll steer is the same as bump steer, but in practice things like the sway bar geometry have an effect that modifies it.

Roll steer is usually measured in degrees of toe per degree of roll, but can also be measured in degrees of toe per metre of wheel travel.

Method of Adjustment

The linearity of the bump steer curve is important. If it is not straight then the length of the tie rod needs to be adjusted.

Bump steer can be made more toe out in jounce by lifting the rack or dropping the outer tie rod, if the rack is in front of the axle. The reverse applies if the rack is behind the axle. Usually only small adjustments (say 3mm) are required.

Active Steering

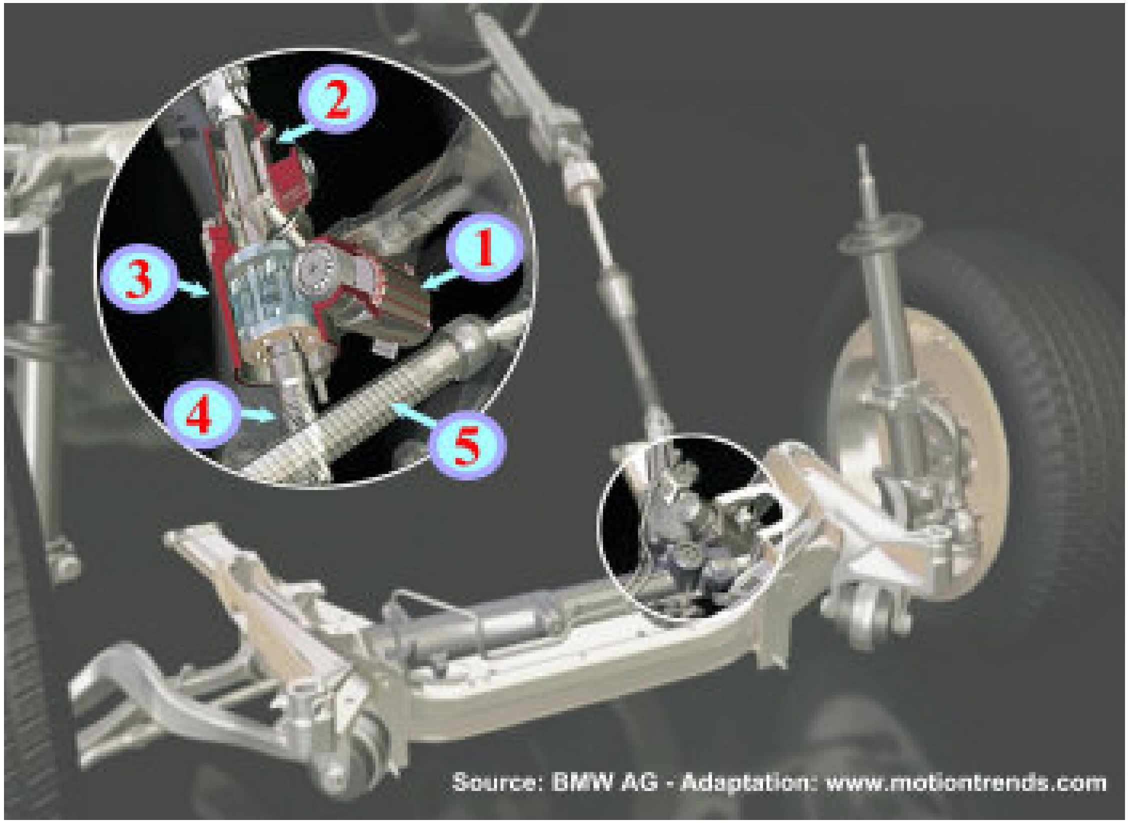
- Need for Active Steering
  - When vehicle Yaws during Braking due to imbalanced braking forces
  - Vehicles yaws due to split Mu
  - Vehicle Aligning moment changes its direction due to caster change
- Steering Ratio should be small to reduce the yaw rate by steering in the opposite direction quickly
- Normal speeds and normal conditions, steering torque rate should not have steep gradient-reasonable steering ratio
- Hence, need for variable steering ratio

Active Steering Concept

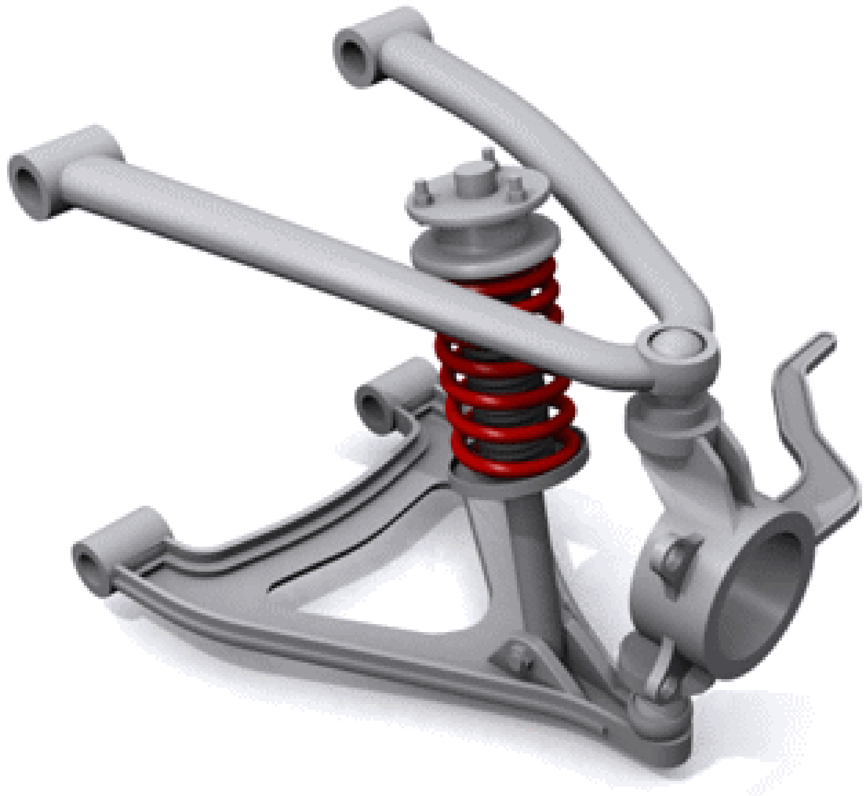
- When driving at lower speeds - such as in city traffic, when parking or on winding mountain roads, Active Steering increases the size of the steering angle- Low steering ratio
- At medium speeds, steering is easier
- To ensure smoothness at higher speeds, as of around 120 to 140 km/h Active Steering becomes more indirect. Active Steering therefore reduces the amount of change in the steering angle for every movement of the steering wheel. This gives the driver the advantage of more precise steering at higher speeds, and ensures great stability and more comfort
- If the vehicle is threatened with instability, such as by oversteering or braking on a changeable surface, Active Steering helps to overcome it. For example, in order to reduce unsafe yaw, Active Steering can increase the angle of steering wheels faster than even the most expert driver.

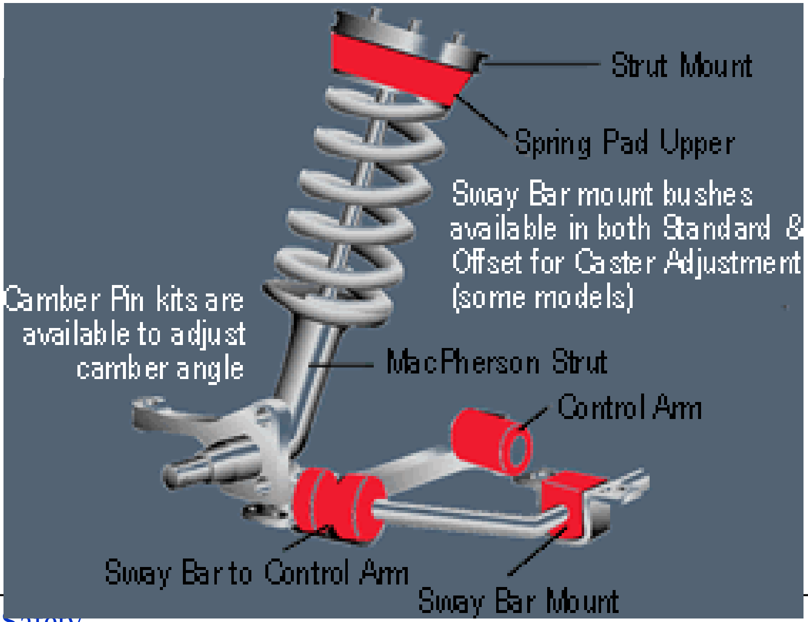
Active Steering Concept

At the heart of Active Steering system is the planetary gear set integrated into the steering column. An electric motor in the joint adjusts the front wheels' steering angle in proportion to the Sedan's current speed.



Elastomer/Rubber Bushes

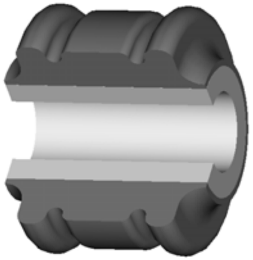



Strut Mount
Spring Pad Upper
Sway Bar mount bushes available in both Standard & Offset for Caster Adjustment (some models)
Camber Pin kits are available to adjust camber angle
MacPherson Strut
Control Arm
Sway Bar to Control Arm
Sway Bar Mount

Suspension Bushes

Single Bonded Bushes

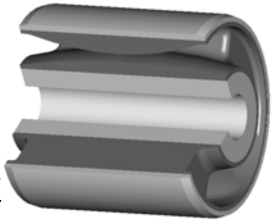
Provides a low cost pivot



Applications:
- Damper bushes
- Engine torque rods
- Low cost suspension arms

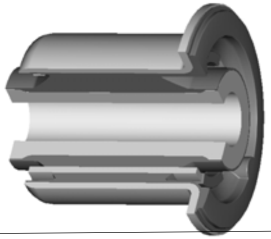
Double Bonded Bushes

Provides a controlled stiffness pivot



Applications:
- Damper bushes
- Suspension arms, where there is insufficient support for a single bonded bush

Interleaved Bushes
Provides a controlled stiffness pivot with low torsional stiffness



Applications:
- High articulation positions
- Multi-link sports suspensions

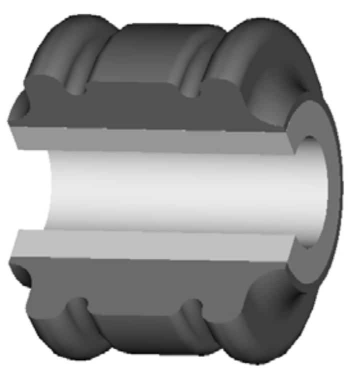
Hydraulic Bushes

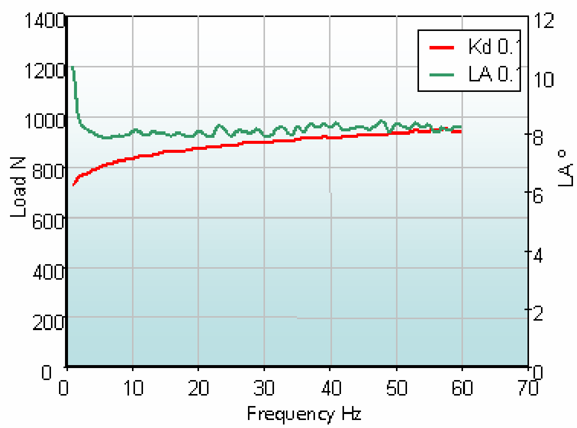
Provides damping control

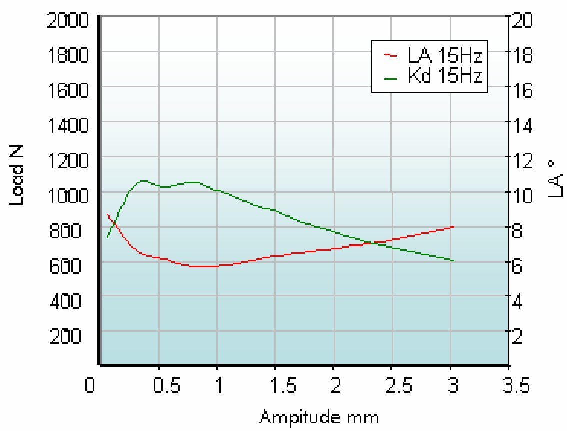
Applications:
- Front suspension compliance bushes
- Rear suspension trailing arm bushes
- Subframe mountings

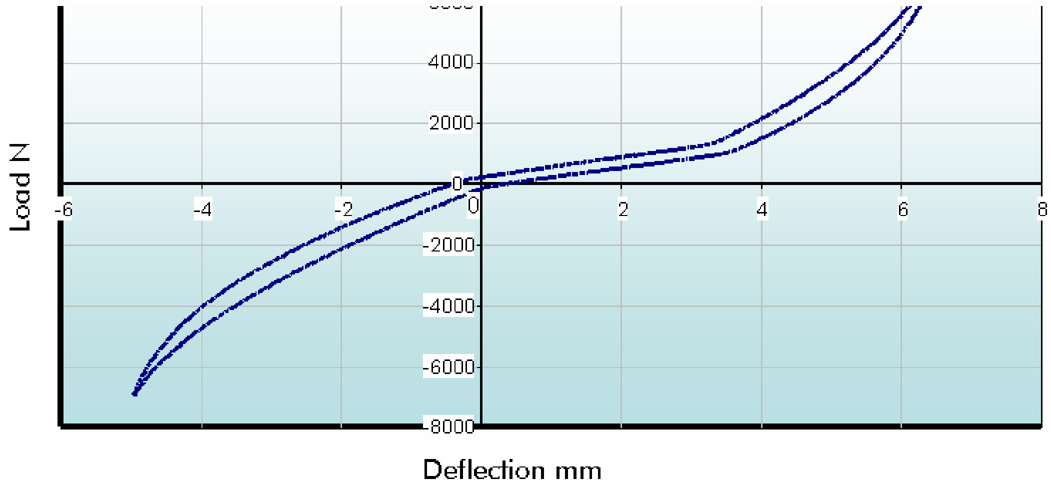
Conventional Suspension Bush

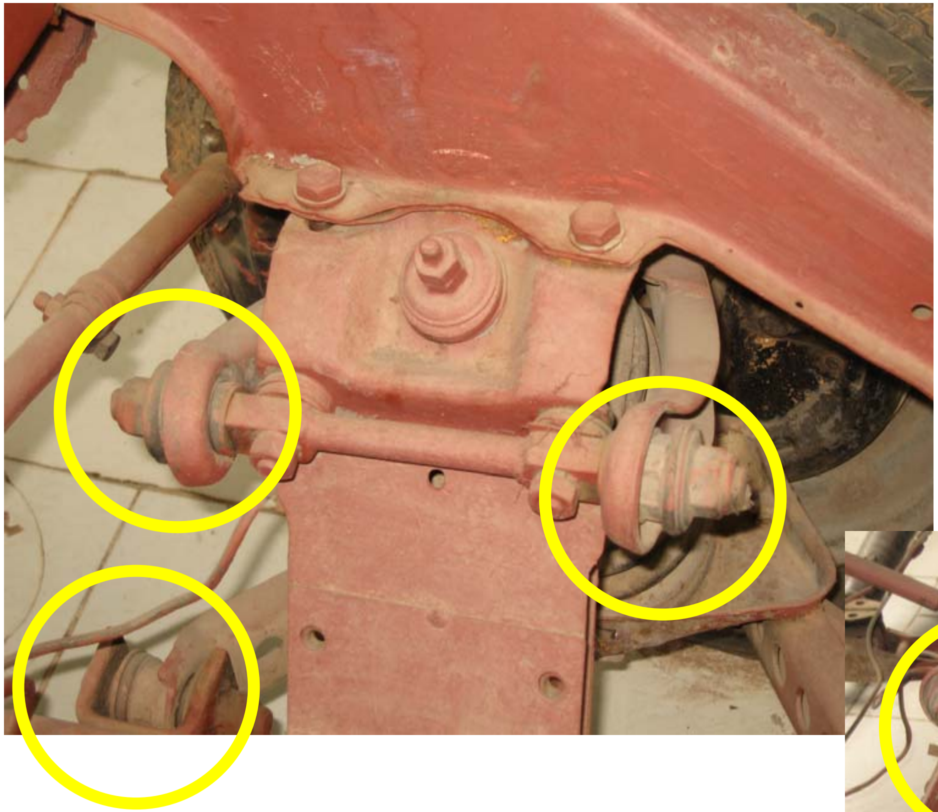
Conventional bush used in a vertical orientation in a front control arm.











Session-3

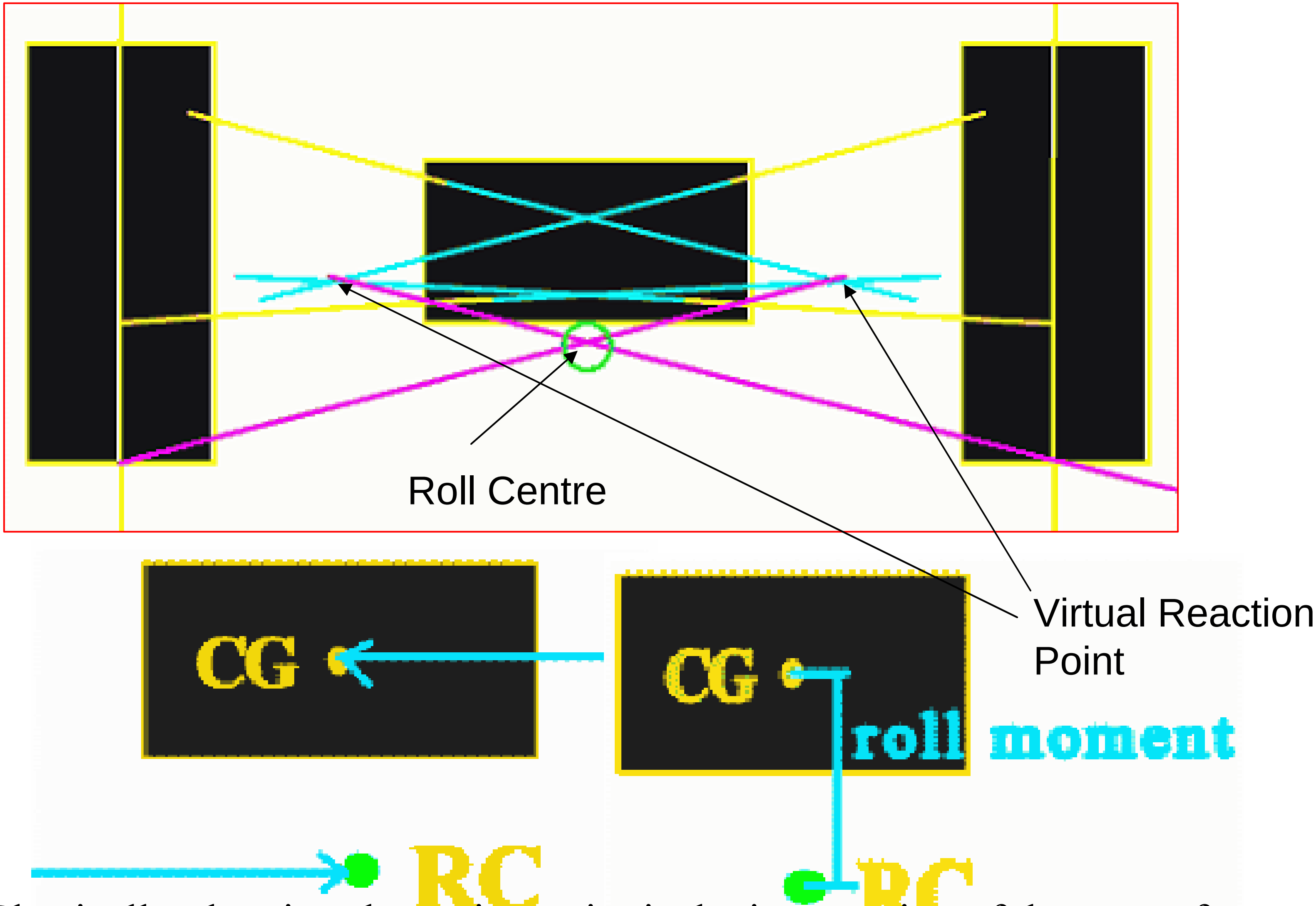
4.00 PM-5.30 PM

Vehicle Suspension-Roll Centre, Squat and Dive

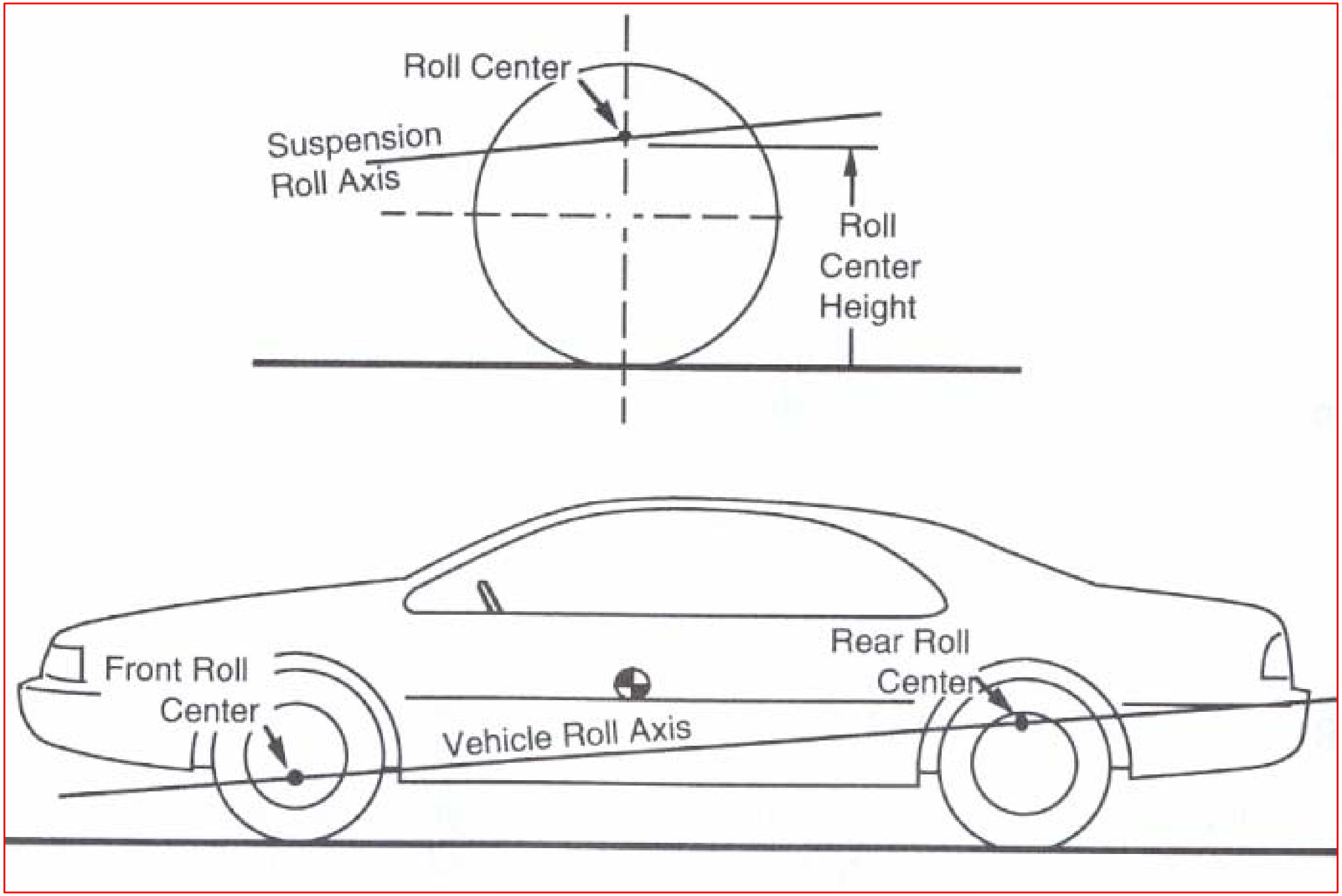
- Roll Centers, Roll Axis
- Squat and Dive

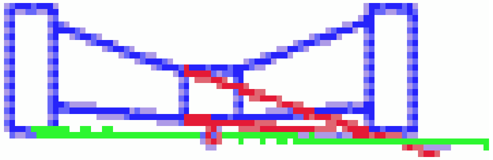
Roll Centre and Roll Axis

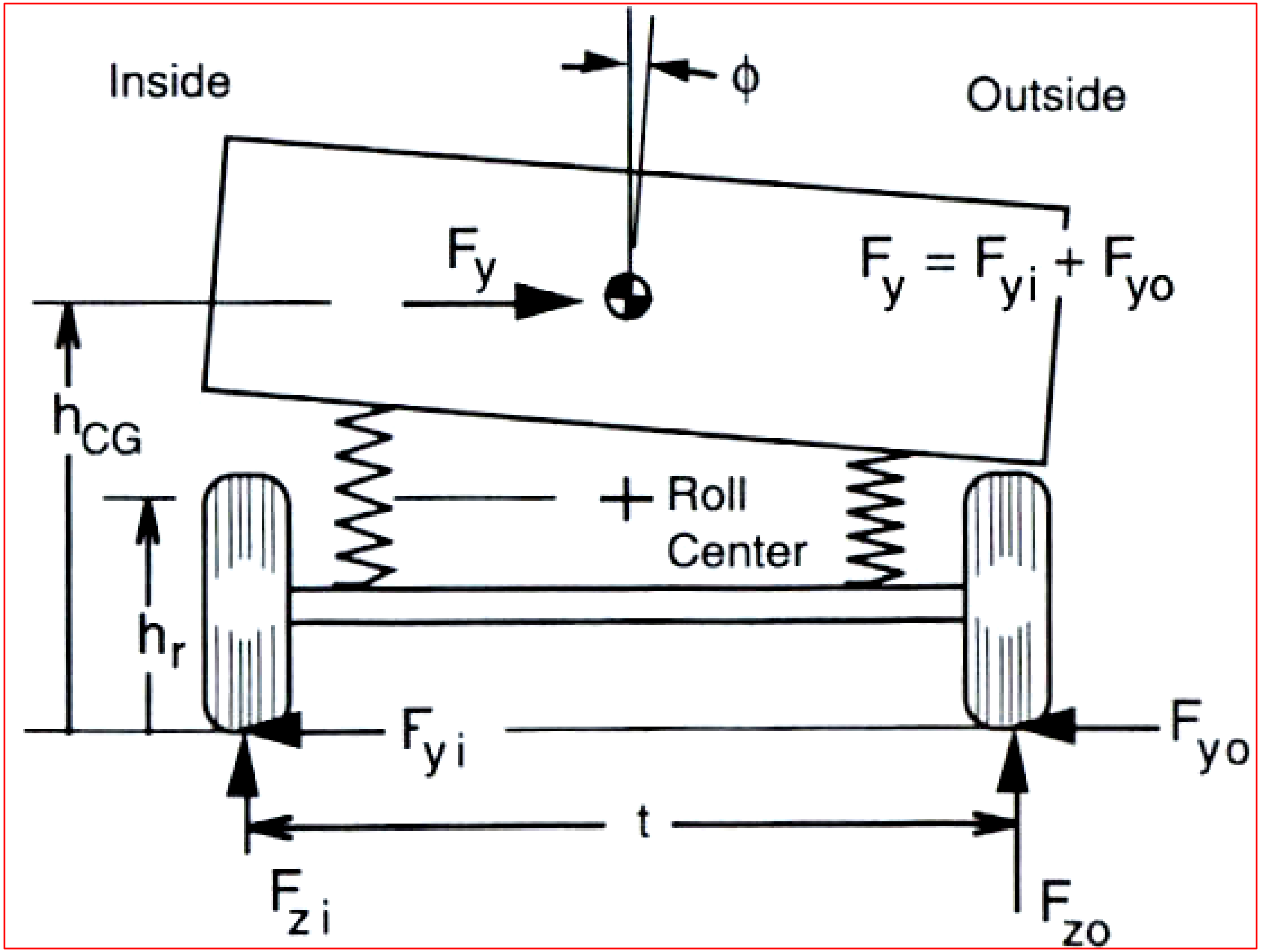
- Roll Centre and Roll Axis
  - The suspension is characterized by a Roll Centre
  - Roll Centre is a point at which the lateral forces are transferred from the axle to the sprung mass
  - Roll centre is a point on the body at which a lateral force application will produce no roll angle
  - It is the point around which the axle rolls when subjected to a pure roll moment
  - Roll Axis; It is the line connecting the roll centres of front and rear suspensions
  - In the median plane of the car, a transverse plane in which horizontal lateral forces applied to the rolling mass of the car will move the car sideways without causing it to roll
  - It follows that forces applied above the roll axis as at the CG will cause roll. Also that forces, if any applied below the roll axis will cause "Banking"



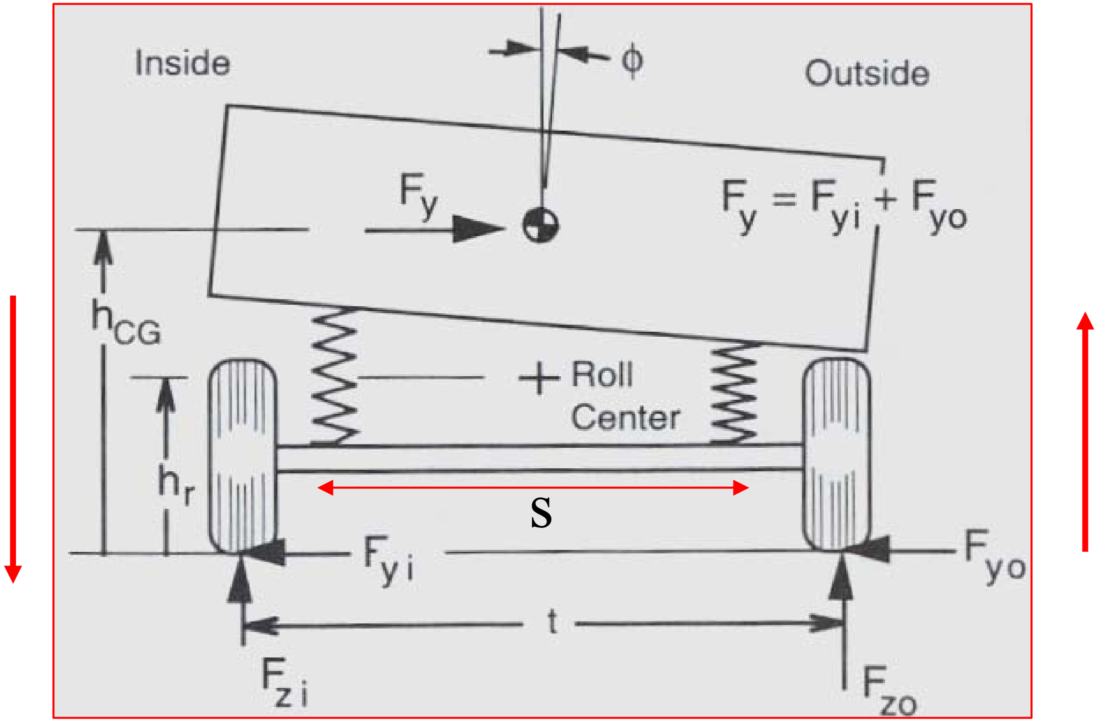
- Physically, the virtual reaction point is the intersection of the axes of any pair of suspension control arms.
- Mechanistically, it is the point where the compression/tension forces in the control arms can be resolved into a single lateral force


Roll Center
Suspension Roll Axis
Roll Center Height
Front Roll Center
Rear Roll Center
Vehicle Roll Axis




Inside
ϕ
Outside
F_y
$F_y = F_{yi} + F_{yo}$
h_{CG}
Roll
Center
h_r
F_{yi}
F_{yo}
t
F_{zi}
F_{zo}

Suspension Roll Stiffness

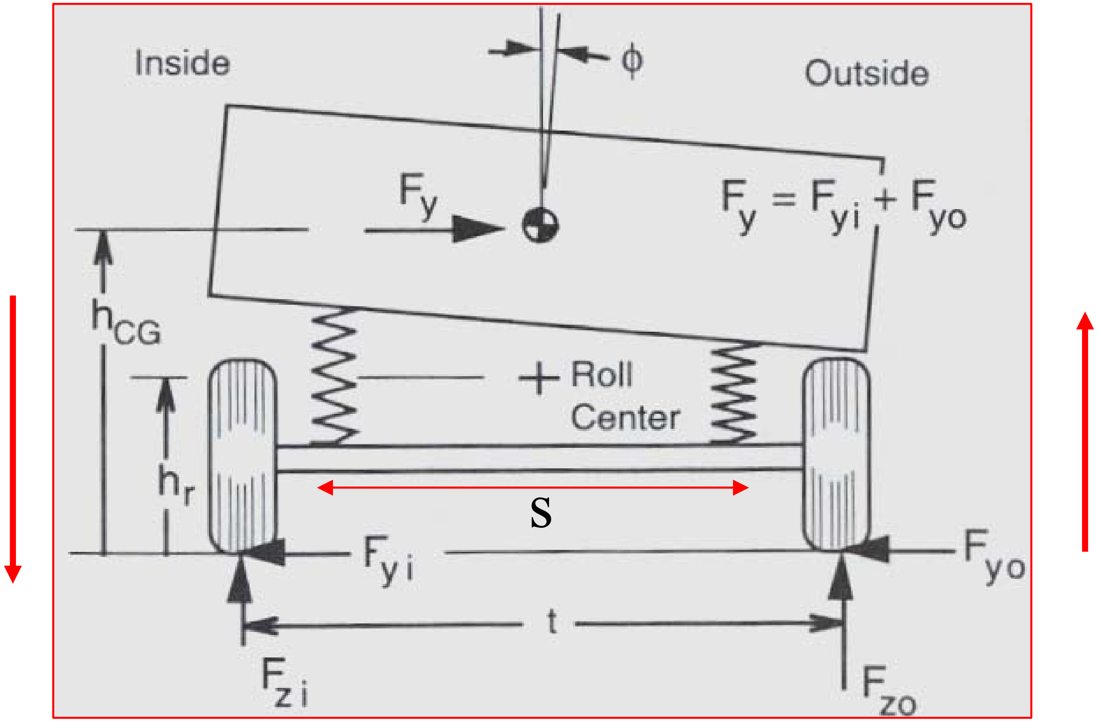


K_ϕ : Roll stiffness of the suspension

K_s : Vertical Spring rate of the left and right springs

s: Lateral separation between the springs

ϕ: Roll angle of the body



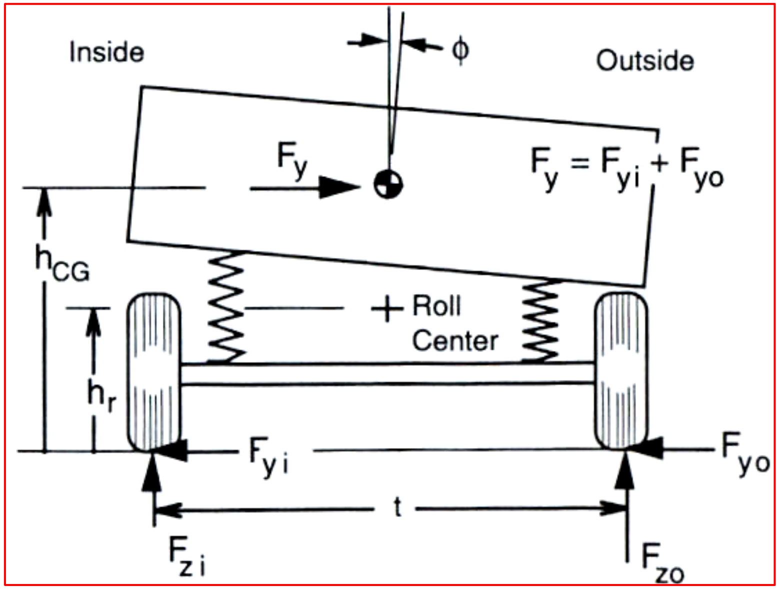
$$\sum M_{cG} = \left(K_s\left(\frac{s}{2}\right)\phi\right)\frac{s}{2} + \left(K_s\left(\frac{s}{2}\right)\phi\right)\frac{s}{2} = \frac{1}{2}K_s s^2\phi = K_\phi \phi$$

If a roll bar is included then

$$\sum M_{cG} = \frac{1}{2}K_s s^2\phi + K_r\phi \;= (K_\phi + K_r)\phi$$

$$\mathrm{K}_\varphi = \text{Roll stiffness of the suspension} = 0.5\mathrm{K}_s \mathrm{s}^2$$

Lateral Load Transfer

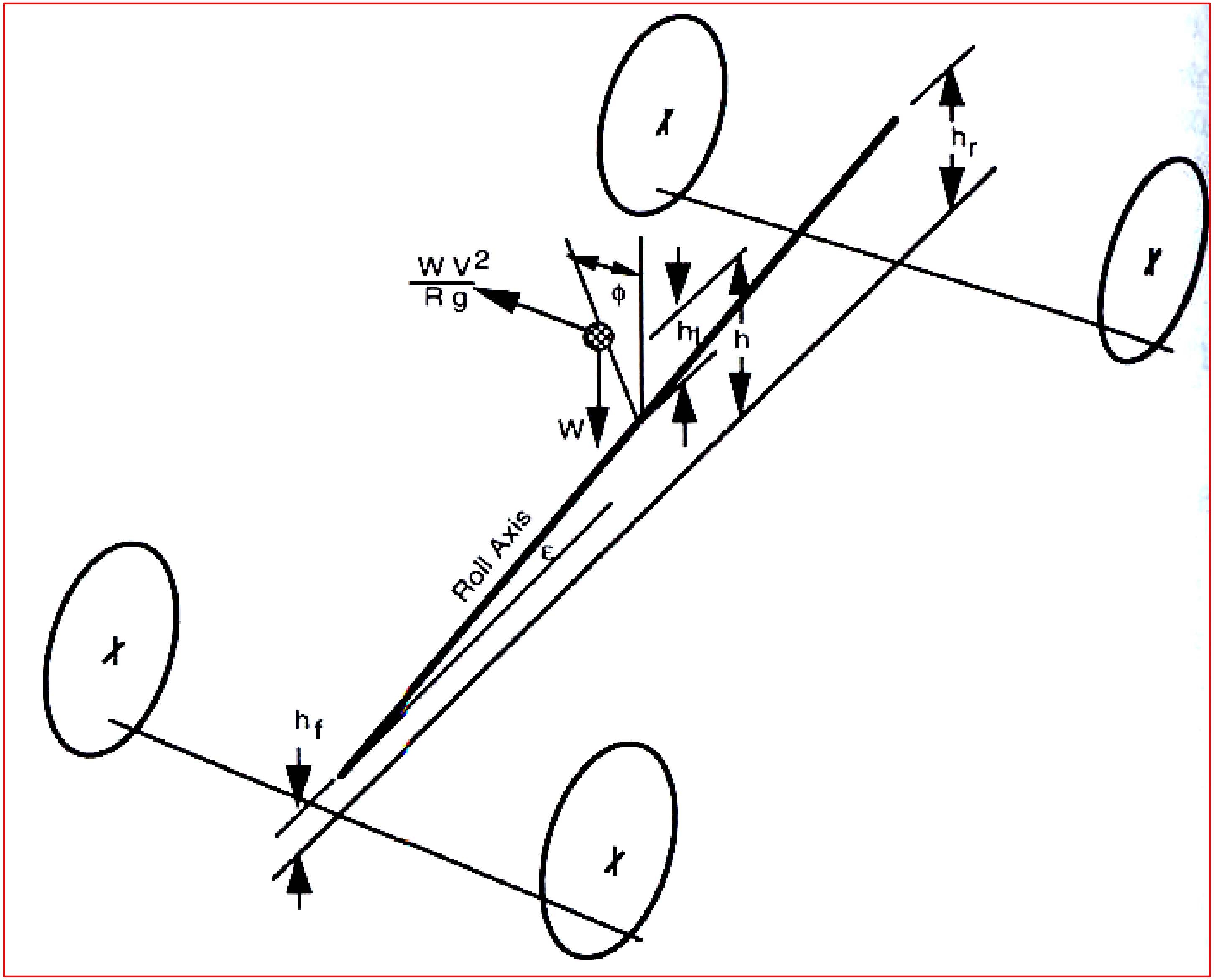


$$F_{z0} - F_{zi} = \frac{2F_y h_r}{t} + \frac{2K_\phi \phi}{t}$$

$$\text{Lateral load transfer due to cornering force} = \frac{2F_y h_r}{t}$$

$$\text{Lateral load transfer due to vehicle roll} = \frac{2K_\varphi \varphi}{t}$$

Roll Angle and Roll Rate



Roll Angle and Roll Rate

$$Roll\ angle = \phi = \frac{Wh_1 V^2/(Rg)}{K_{\phi f} + K_{\phi r} - Wh_1}$$

$$\phi = \frac{Wh_1 a_y}{K_{\phi f} + K_{\phi r} - Wh_1}$$

$$Roll\ Rate = \frac{d\phi}{da_y} = \frac{Wh_1}{K_{\phi f} + K_{\phi r} - Wh_1}$$

The roll rate is usually K in the range of 3 to 7 degrees/g on typical passenger cars

Roll Moment

Roll Moments

$$M'_{\phi f} = K_{\phi f}\frac{Wh_1V^2/(Rg)}{K_{\phi f}+K_{\phi r}-Wh_1} + W_f h_f \frac{V^2}{Rg} = \Delta F_{zf} t_f$$

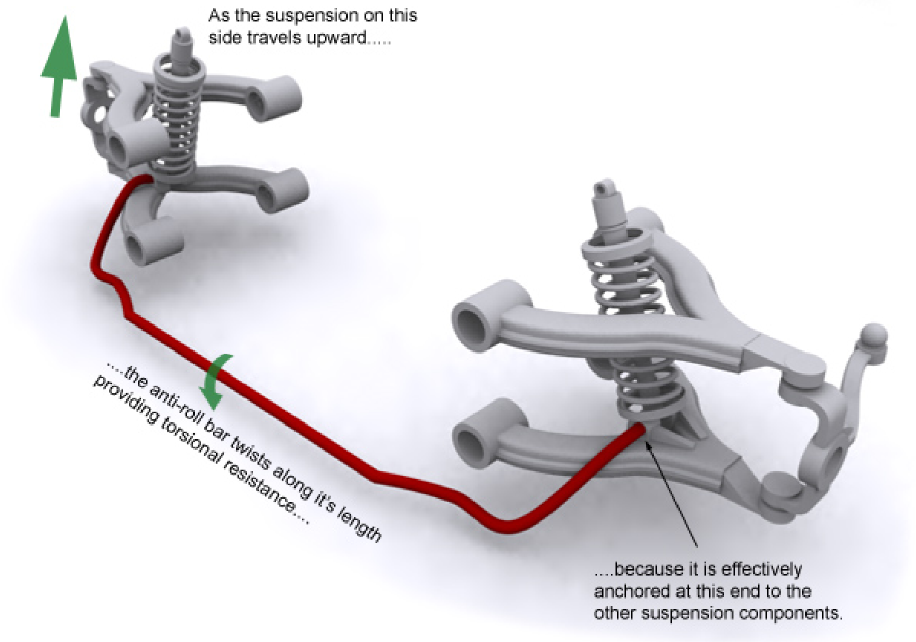
$$M'_{\phi r} = K_{\phi r}\frac{Wh_1V^2/(Rg)}{K_{\phi f}+K_{\phi r}-Wh_1} + W_r h_r \frac{V^2}{Rg} = \Delta F_{zr} t_r$$

The Roll moments magnitude depend on $K_{\Phi f}$ and $K_{\Phi t}$ which in turn depend on suspension stiffness

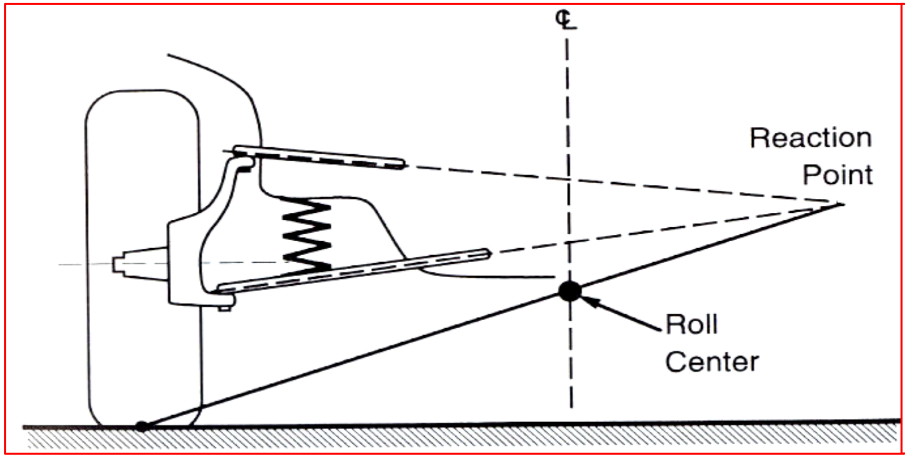
1. In general, the roll moment distribution on vehicles tends to be biased toward the front wheels due to a number of factors:
2. Relative to load, the front spring rate is usually slightly lower than that at the rear (for flat ride), which produces a bias toward higher roll stiffness at the rear. However, independent front suspensions used on virtually all cars enhance front roll stiffness because of the effectively greater spread on the front suspension springs.
3. Designers usually strive for higher front roll stiffness to ensure under-steer in the limit of cornering.

DIMENSIONS	
Overall Length (mm)	3565
Overall Width (mm)	1595
Overall Height (mm)	1550
Wheelbase (mm)	2380
Ground Clearance (mm)	165
Front Track (mm)	1400
Rear Track (mm)	1385
Fuel Tank capacity (l)	35

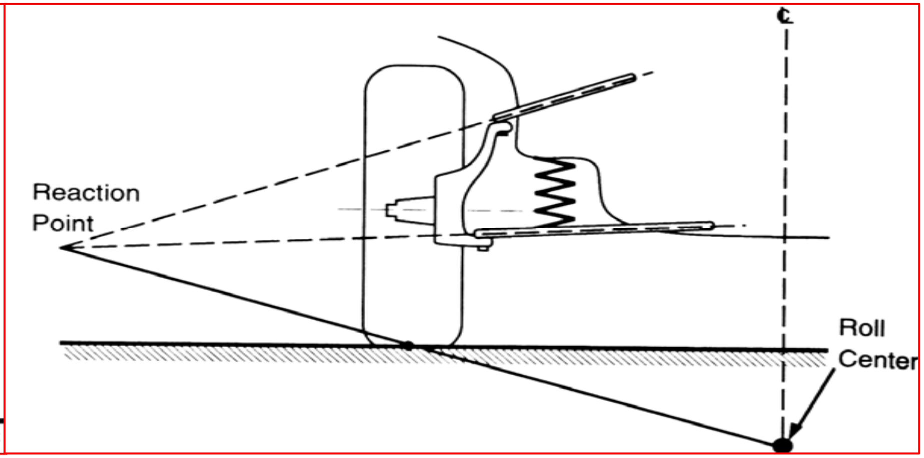
4. Stabilizer bars are often used on the front axle to obtain higher front roll stiffness.
5. If stabilizer bars are needed to reduce body lean, they may be installed on the front or the front and rear. Caution should be used when adding a stabilizer bar only to the rear because of the potential to induce unwanted oversteer.



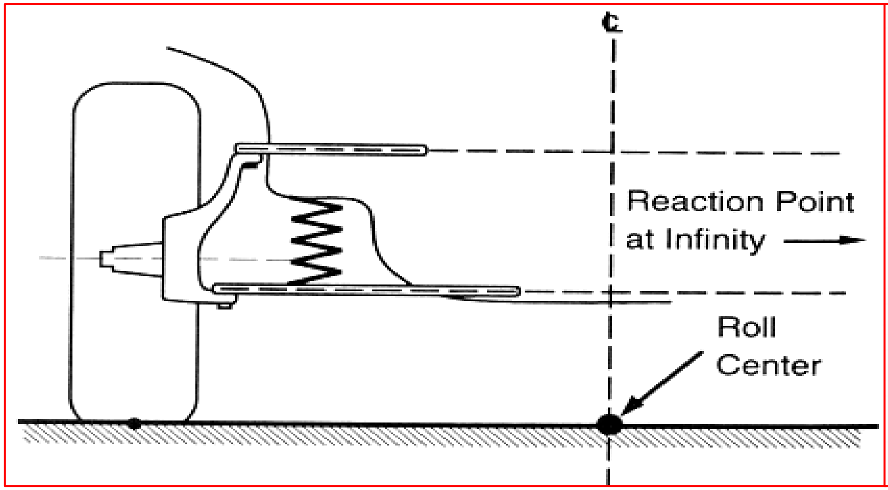
Independent Suspension Roll Centers



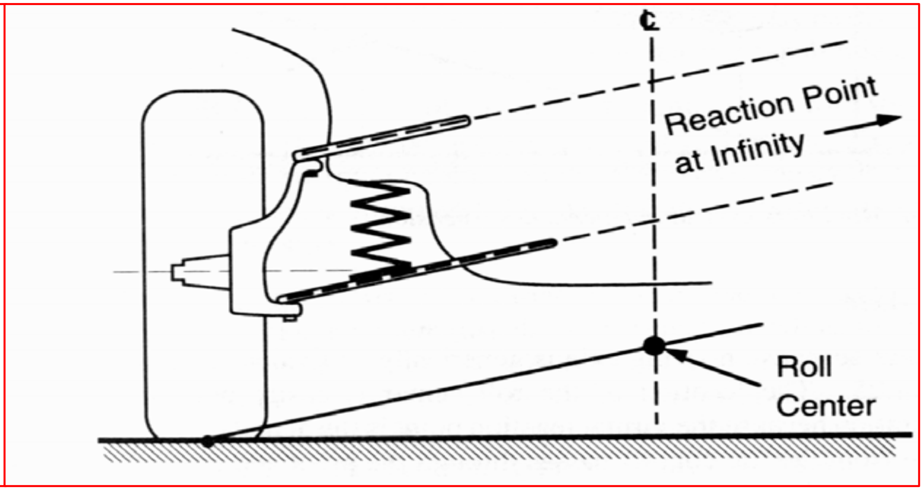
Positive swing arm independent suspension



Negative swing arm independent suspension

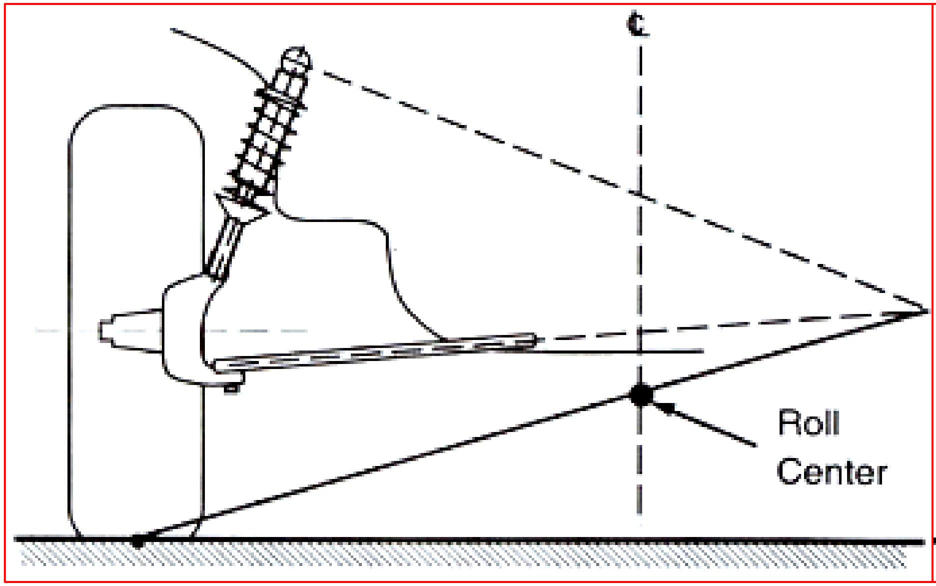


Parallel horizontal link independent suspension

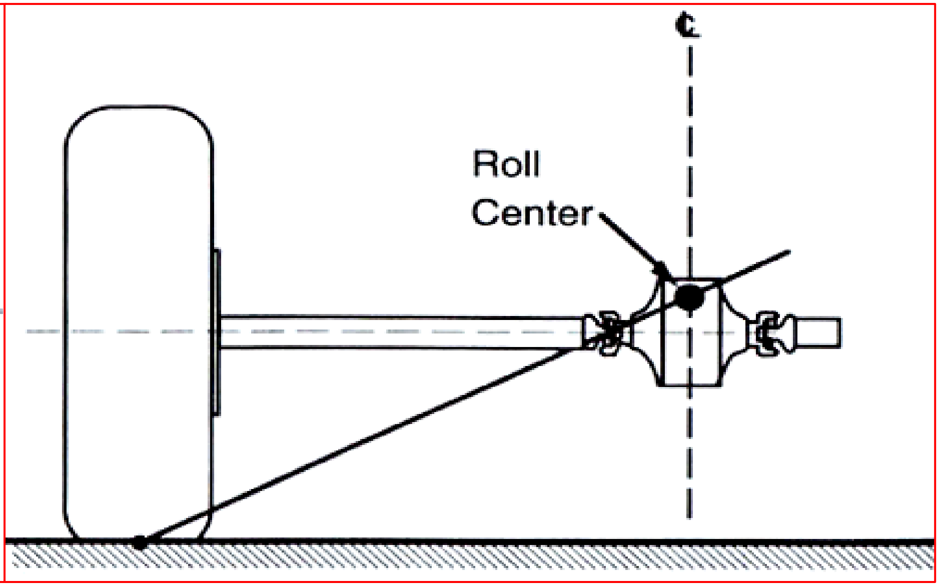


Inclined parallel link independent suspension

Independent Suspension Roll Centers

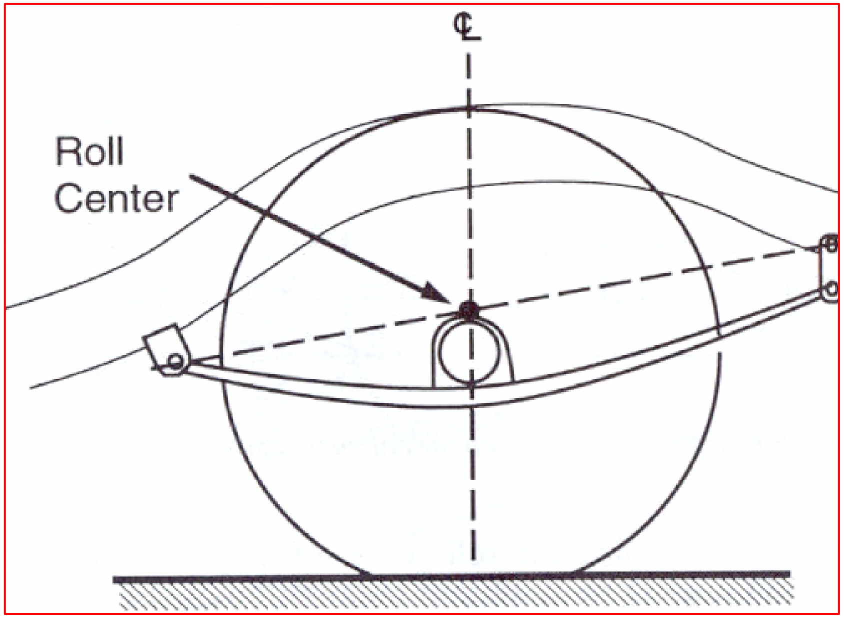


Macpherson strut independent suspension

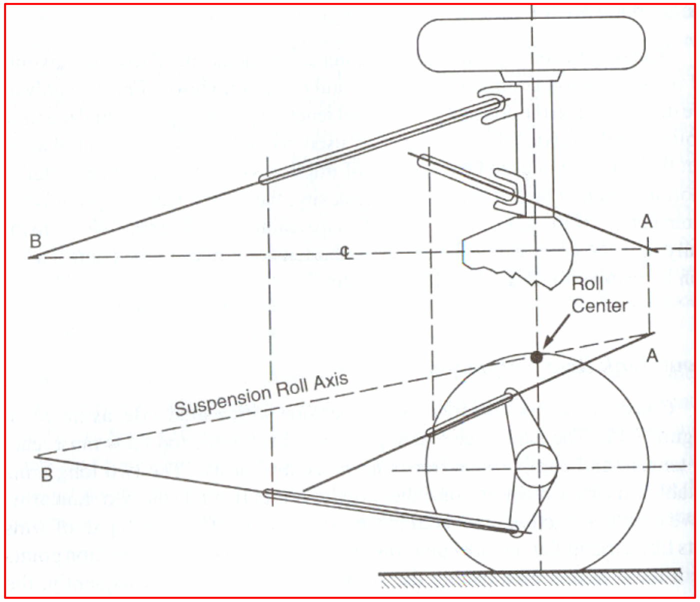


Swing axle independent suspension

Roll Centers of Dependent Suspension

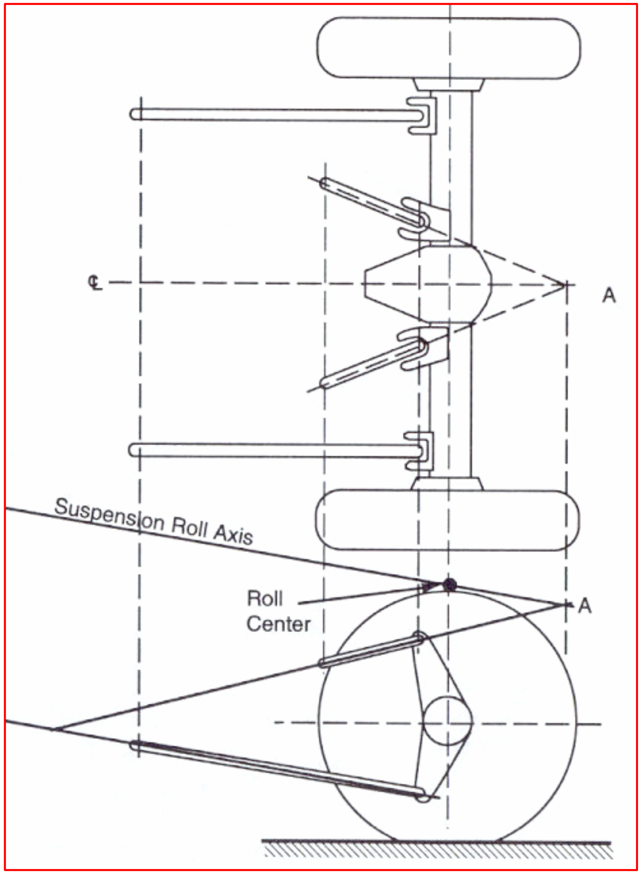


Hotchkiss Suspension

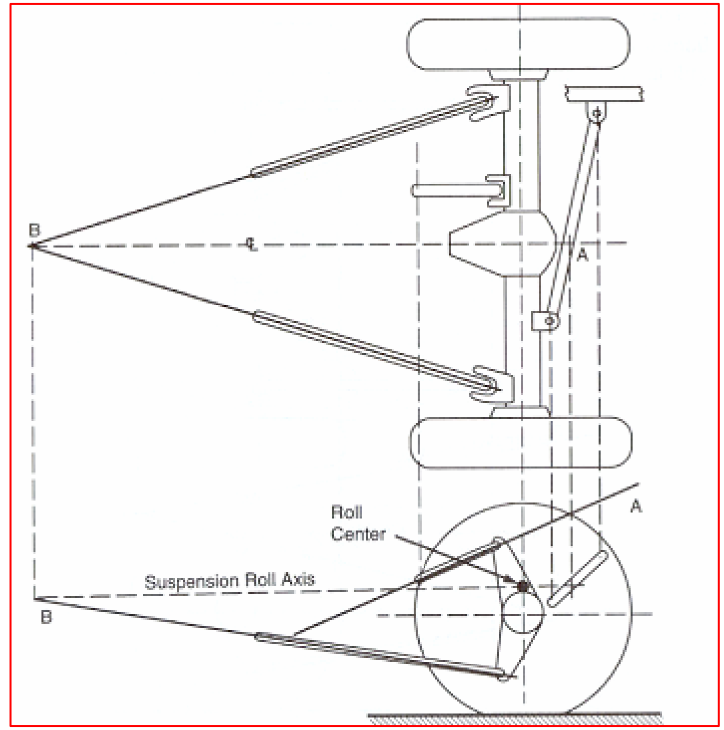


Four Link Rear Suspension

Roll Centers of Dependent Suspension



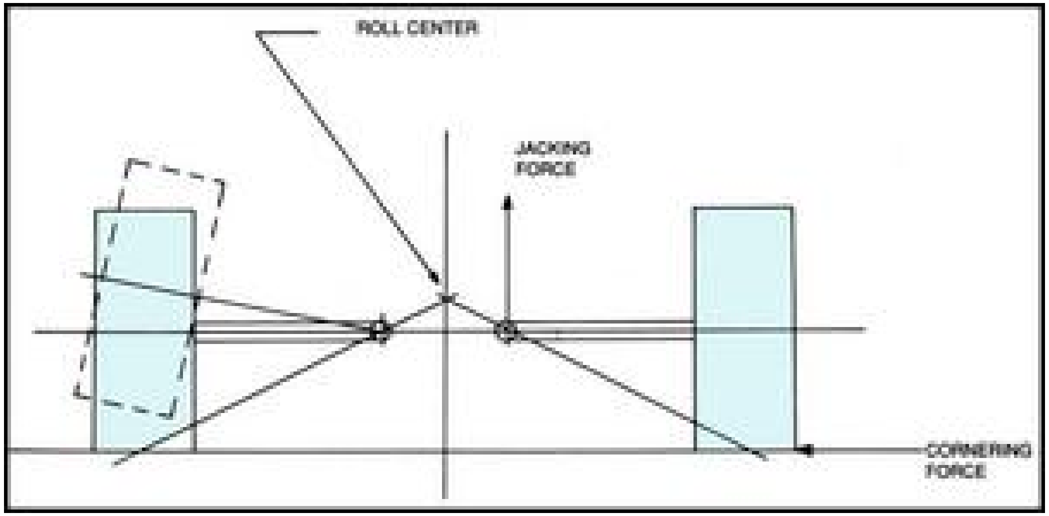
Four Link with Parallel Arms



Three Link Rear Suspension

What is the Effect if Roll Centre Located on Ground or Located Below the Ground?

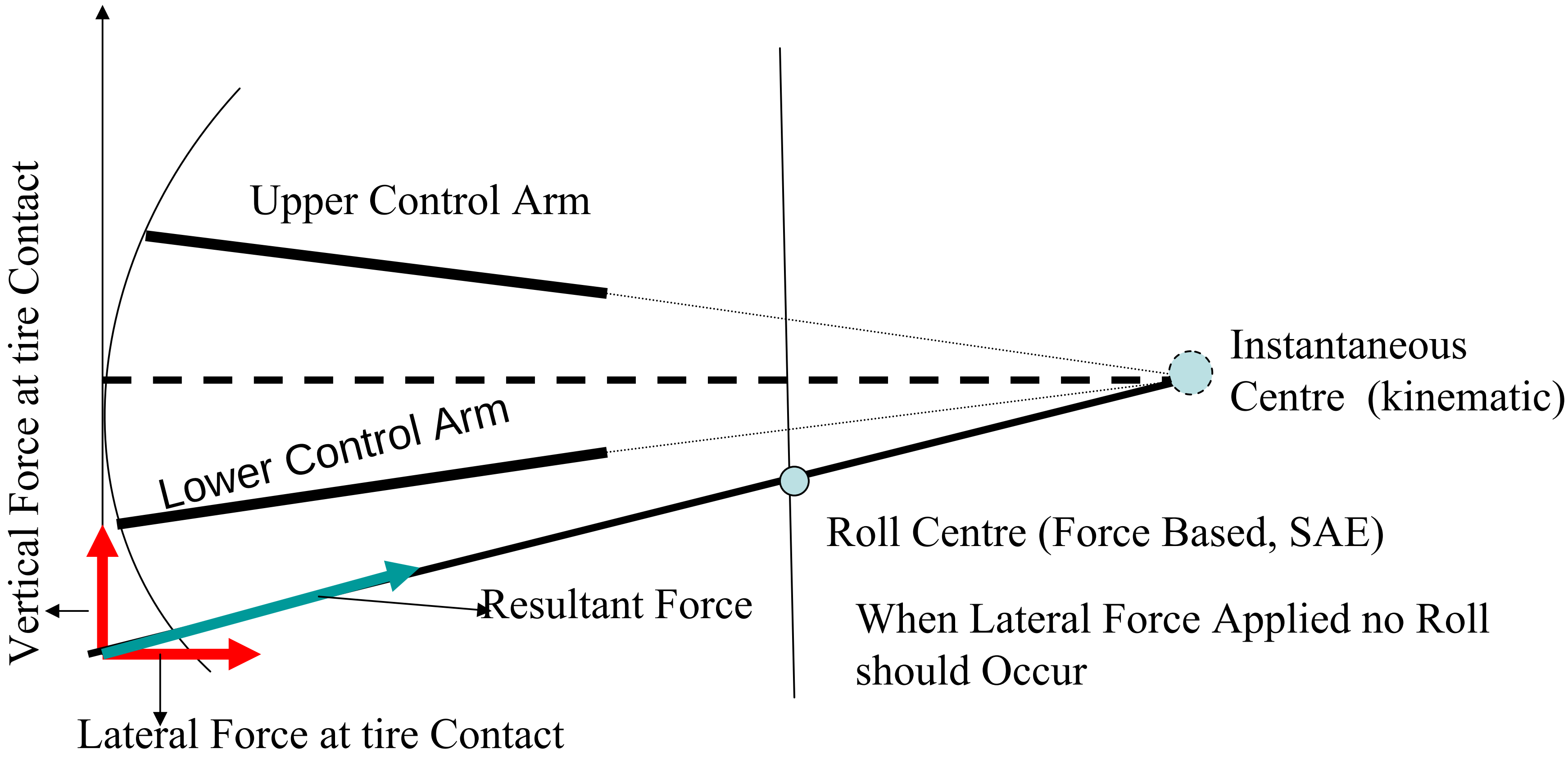
- Jacking forces are the sum of the vertical force components experienced by the suspension links. The resultant force acts to lift the sprung mass if the roll centre is above ground, or compress it if underground. Generally, the higher the Roll Centre, the more Jacking force is experienced



Instant Centre

- Due to the fact that the wheel and tire's motion is constrained by the suspension links on the vehicle, the motion of the wheel package in the front view will scribe an imaginary arc in space with an “instantaneous center" of rotation at any given point along its path. The instant center for any wheel package can be found by following imaginary lines drawn through the suspension links to their intersection point.
- A component of the tire's force vector points from the contact patch of the tire through instant center. The larger this component is, the less suspension motion will occur. Theoretically if the resultant of the vertical load on the tire and the lateral force generated by it points directly into the instant center, the suspension links will not move. In this case all weight transfer at that end of the vehicle will be geometric in nature. This is key information used in finding the force-based roll center as well.
- In this respect the instant centers are more important to the handling of the vehicle than the kinematic roll center alone, in that the ratio of geometric to elastic weight transfer is determined by the forces at the tires and their directions in relation to the position of their respective instant centers.

Kinematic and Force Roll Centre



If the above condition is met the suspension will not move, otherwise suspension moves

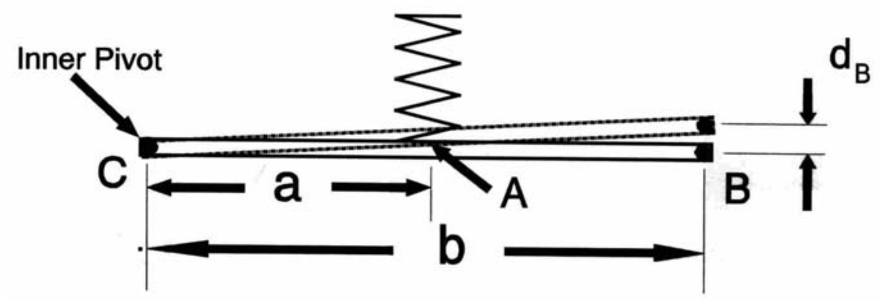
Spring and Wheel Rates

- Spring rates
  - It is the ratio of vertical load on spring to its displacement (N/mm)
- Wheel rates
  - It is the ratio of load acting on the wheel/rise of the wheel
- Motion ratios
  - The displacement relationship between the spring and the wheel determines the actual rate the wheel works against for any spring rate. This displacement relationship may be defined as a motion ratio. The rate at the wheel is defined as the wheel rate (K_w). The rate of the spring itself is called the spring rate (K_s). The displacement relationship is a function of both spring position on the load carrying member and the angular orientation of the spring to that member.

Motion Ratio Analysis

- From the simple lever system a number of relationships can be drawn.

$$F_B = F_A\left[\frac{a}{b}\right]$$

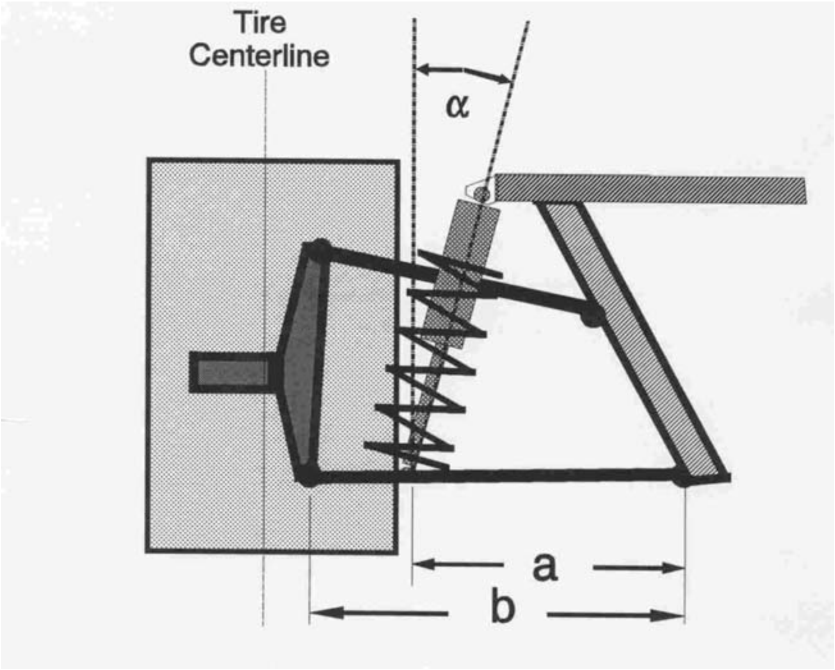


$$d_B = d_A\left[\frac{b}{a}\right]$$

$$\frac{F_B}{d_B} = k_B = \frac{F_A\left(\frac{a}{b}\right)}{d_A\left(\frac{b}{a}\right)} = k_A\left(\frac{a}{b}\right)^2$$

Motion Ratio Analysis

- Motion Ratio in the Road Vehicle.
  - The motion ratio describes the displacement ratio between the spring and the centerline of the wheel. The motion ratio squared times the spring rate gives the wheel rate.



Motion Ratio Analysis

- Using the previous analysis and Figure , the following apply.

$$K_w = K_s \left(\frac{a}{b} \right)^2 \mathrm{Cos}^2 \alpha$$

  – The above analysis assumes minimal camber change at the wheel.

  – The motion ratio can be determined experimentally and the measured distance ratio squared for an accurate value.

$$K_w = K_s \left(\frac{\textit{travel along spring axis}}{\textit{vertical travel of wheel centerline}} \right)^2$$

Squat and Dive

- **Squat** - Squat is the dipping of a car's rear end that occurs during hard acceleration.
- Squat is caused by a load transfer from the front to the rear suspension during acceleration.
- **Dive**- Dive is the dipping of a car's front end that occurs during braking
- Dive is caused by a load transfer from the rear to the front suspension during braking

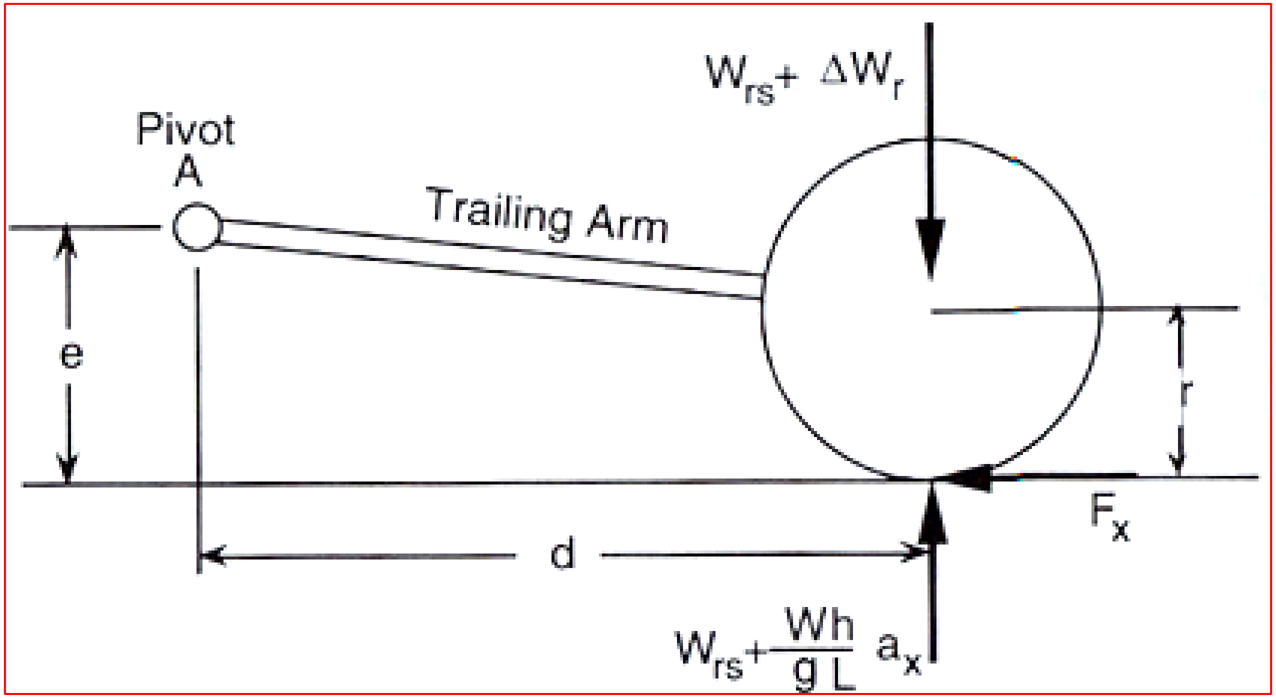
Anti-Squat and Anti-Pitch Suspension Geometry

- During acceleration the load on the rear wheels increases due to longitudinal weight transfer. The load on the rear axle is:

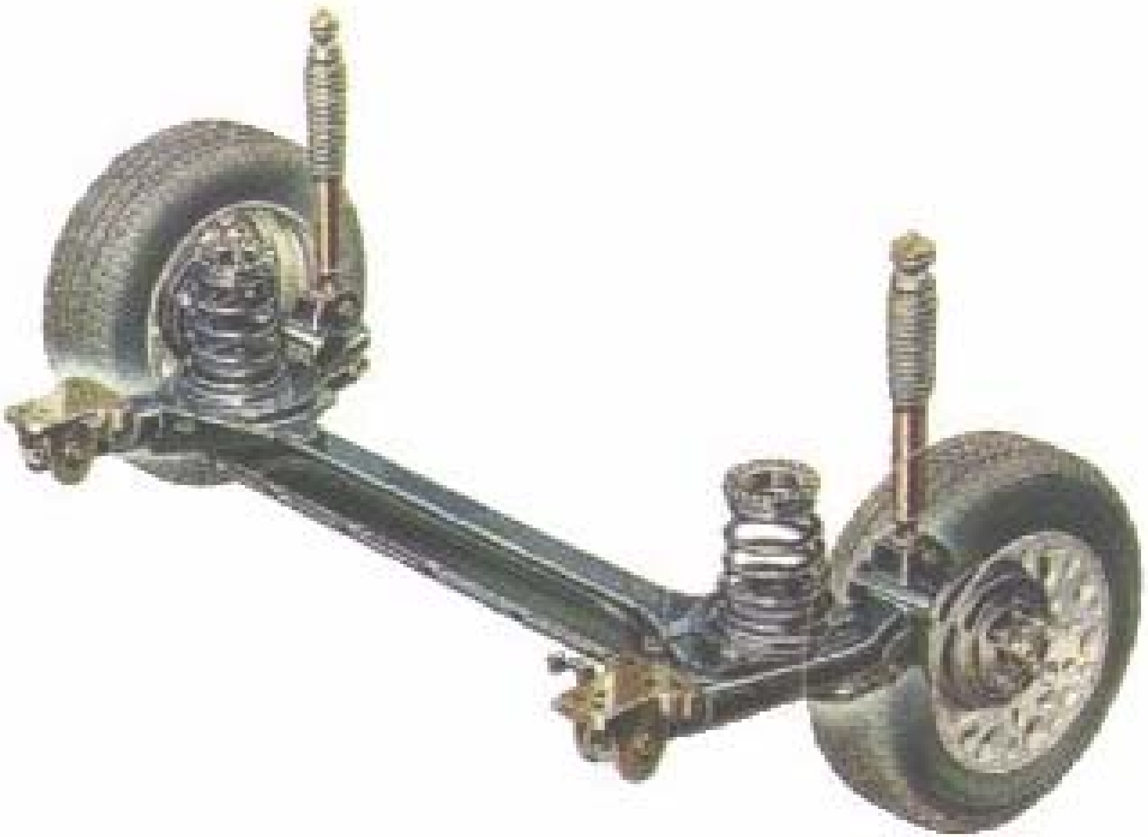
$$W_r = W\left(\frac{b}{L} + \frac{a_x}{g}\frac{h}{L}\right)$$

- The second term on the right side of this equation is the weight transfer effect.
- The weight is transferred to the axle and wheels principally through the suspension. Therefore, there is an implied compression in the rear suspension which, in the case of rear-drive vehicles, has been called “Power Squat.”
- Concurrently, there is an associated rebound in the front suspension. The combination of rear jounce and front rebound deflections produces vehicle pitch. Suspension systems may be designed to counteract the weight transfer and minimize squat and pitch.

Trailing Arm Analysis-Rear Solid Drive Axle



- **Trailing arm** - A suspension element consisting of a longitudinal member that pivots from the body at its forward end and has a wheel hub rigidly attached to its trailing end.



Twist Beam Suspension

Rear Solid Drive Axle

- The system is analyzed by applying NSL for the torques around the pivot point "A." The sum of these torques must be zero when the system is in equilibrium.

$$\Sigma M_A = W_{rs} d + \frac{W}{g}\frac{h}{L} a_x d - W_{rs} d - \Delta W_r d - F_x e = 0$$

Where,

W_{rs} = Static load on the axle = Static load in the suspension

ΔW_r = Change in the suspension load under acceleration

This equation can be solved for the change in rear suspension load.

$$\Delta W_r = \frac{W}{g}\frac{h}{L}a_x - F_x\frac{e}{d} = K_r\delta_r$$

Where:

Kr = Rear suspension spring rate

δr = Rear suspension deflection (positive in jounce)

The front suspension is undergoing a rebound deflection because of the longitudinal load transfer, and has a magnitude of :

$$\Delta W_f = -\frac{W}{g}\frac{h}{L}a_x = K_f\delta_f$$

The pitch angle of the vehicle, θ_p, during acceleration is simply the sum of the suspension deflections divided by the wheelbase. Thus we can write:

$$\theta_p = \frac{\delta_r - \delta_f}{L} = \frac{1}{L}\frac{W}{g}\frac{h}{L}\frac{a_x}{K_r} - \frac{1}{L}\frac{F_x}{K_r}\frac{e}{d} + \frac{1}{L}\frac{W}{g}\frac{h}{L}\frac{a_x}{K_f}$$

Since F_x is simplify the mass times the acceleration, $(W/g)a_x$, the equation can be written :

$$\theta_p = \frac{1}{L}\frac{W}{g}\frac{h}{L}\frac{a_x}{K_r} - \frac{1}{L}\frac{W}{g}\frac{a_x}{K_r}\frac{e}{d} + \frac{1}{L}\frac{W}{g}\frac{h}{L}\frac{a_x}{K_f}$$

$$\theta_p = \frac{1}{L}\frac{W}{g}a_x\left(\frac{1}{K_r}\frac{h}{L} - \frac{1}{K_r}\frac{e}{d} + \frac{1}{K_f}\frac{h}{L}\right)$$

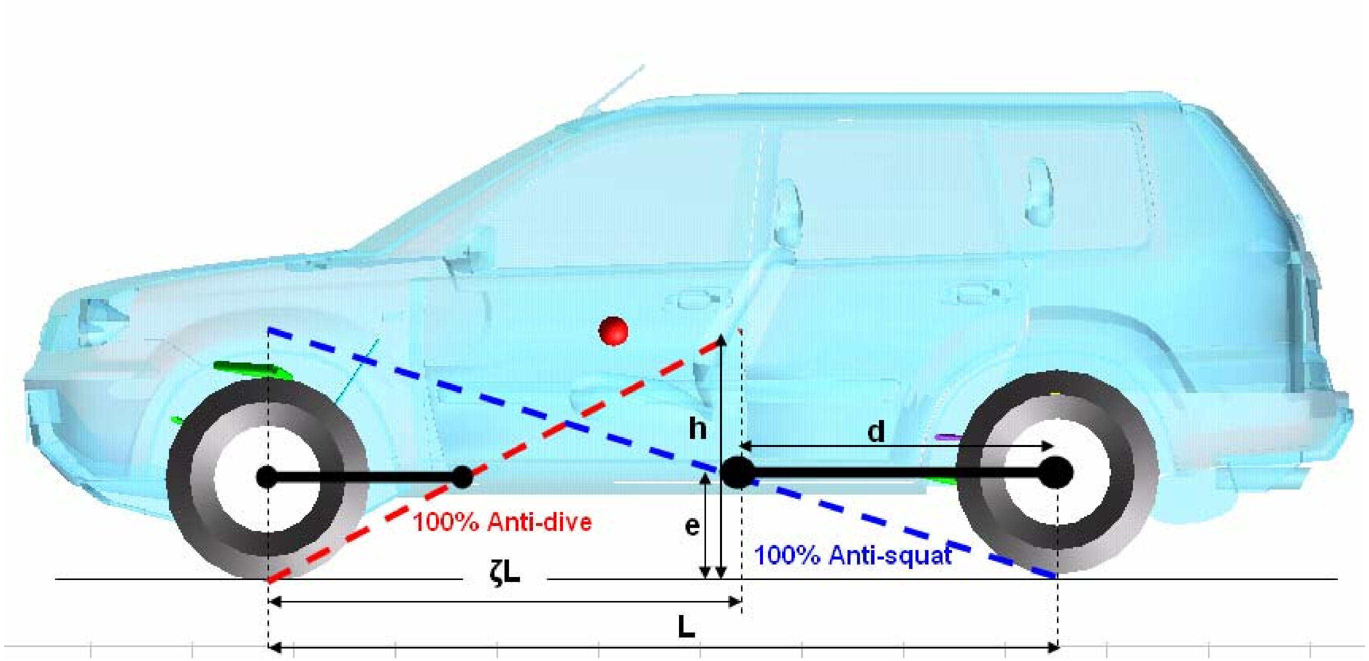
From this equation it is easy to show that zero pitch angle is achieved when the following condition is satisfied :

$$\frac{e}{d} = \frac{h}{L} + \frac{h}{L}\frac{K_r}{K_f}$$

Condition for anti-squat

Rear suspension lift to compensate rebound of front suspension

- The first term on the right-hand side corresponds to the condition by which anti-squat is achieved on the rear suspension.
- That is, if e/d = h/L, the rear suspension will not deflect (jounce) during acceleration. The degree to which this is achieved is described as the percent anti-squat.
- For example, if e/d = 0.5 h/L, the suspension is said to be 50% anti-squat. Since h/L is in the vicinity of 0.2 for most passenger cars, full anti-squat generally requires an effective trailing arm length of about five times the elevation of "e."
- The anti-squat equation (e/d = h/L) defines a locus of points extending from the tire contact point on the ground to the height of the CG over the front axle.


h
d
100% Anti-dive
e
100% Anti-squat
ζL
L

- Locating the trailing arm pivot at any point on this line will provide 100% anti-squat.
- Satisfying the equation with inclusion of the second term implies that the rear suspension will lift to compensate for rebound of the front suspension, thereby keeping the vehicle level.
- The complete equation may be interpreted as the full anti-pitch relationship. Because the ratio of suspension stiffnesses is nominally 1, the anti-pitch condition is approximately:

$$\frac{e}{d} \approx \frac{h}{L} + \frac{h}{L} = 2\,\frac{h}{L} \qquad \text{(Full anti-pitch)}$$

- The locus of points for anti-pitch extends from the tire contact point on the ground to the height of the CG at the mid-wheelbase position.
- Anti-pitch is achieved when the trailing arm pivot is located on the line from the center of tire contact on the ground to the CG of the vehicle.
- Normally some degree of squat and pitch is expected during vehicle acceleration, so full compensation is unusual.
- Anti-squat performance cannot be designed without considering other performance modes of the vehicle as well. When the trailing arm is short, the rear axle may experience "power hop" during acceleration near the traction limit.
- The goals for anti-squat may conflict with those for braking or handling. In this latter case, placing the pivot center above the wheel center can produce roll oversteer.

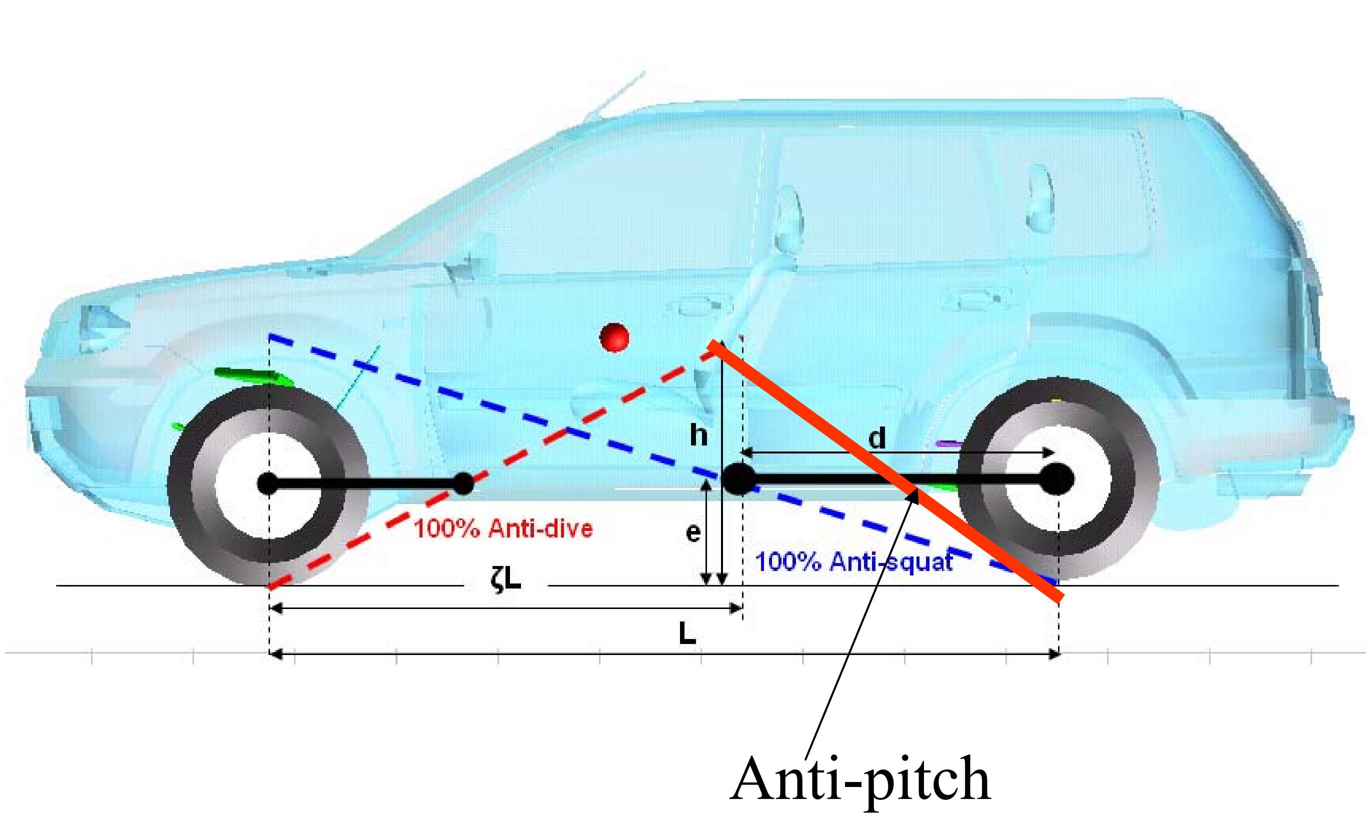
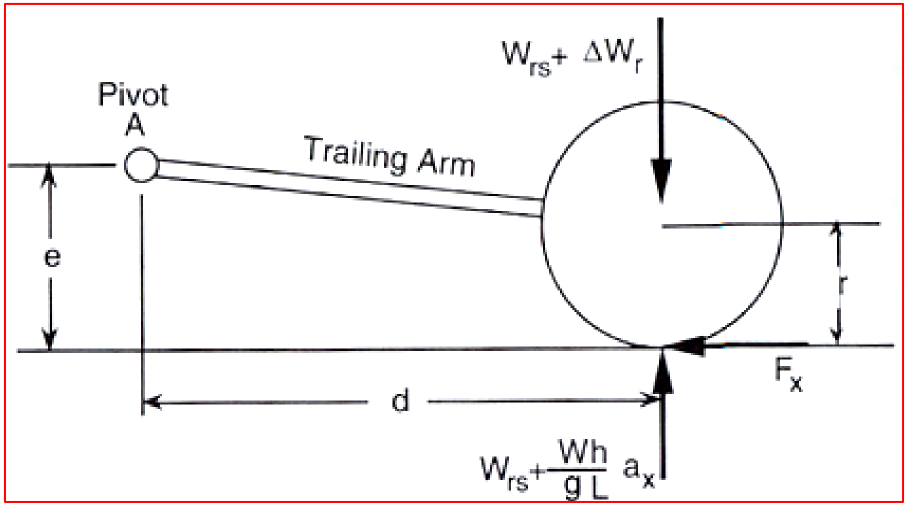


Fig: 16.2b

Independent Rear Drive



$$\frac{e-r}{d}=\frac{h}{L}+\frac{h}{L}\frac{K_r}{K_f}$$

Front Solid Drive Axle

- For a front-drive axle, conditions for anti squat and anti pitch are:

$$\theta_p = \frac{1}{L}\frac{W}{g}a_x\left(\frac{1}{K_r}\frac{h}{L} + \frac{1}{K_f}\frac{h}{L} + \frac{1}{K_f}\frac{e}{d}\right)$$

$$\frac{e}{d} = -\frac{h}{L} - \frac{h}{L}\frac{K_f}{K_r}$$

Independent Front-Drive Axle

- The comparable equations for an independent front-drive axle, as is common on most front-drive cars today, are:

$$\theta_p = \frac{1}{L}\frac{W}{g}a_x\left(\frac{1}{K_r}\frac{h}{L} + \frac{1}{K_f}\frac{h}{L} + \frac{1}{K_f}\frac{e-r}{d}\right)$$

and $$\frac{e-r}{d} = -\frac{h}{L} - \frac{h}{L}\frac{K_f}{K_r}$$

Anti-dive Suspension Geometry

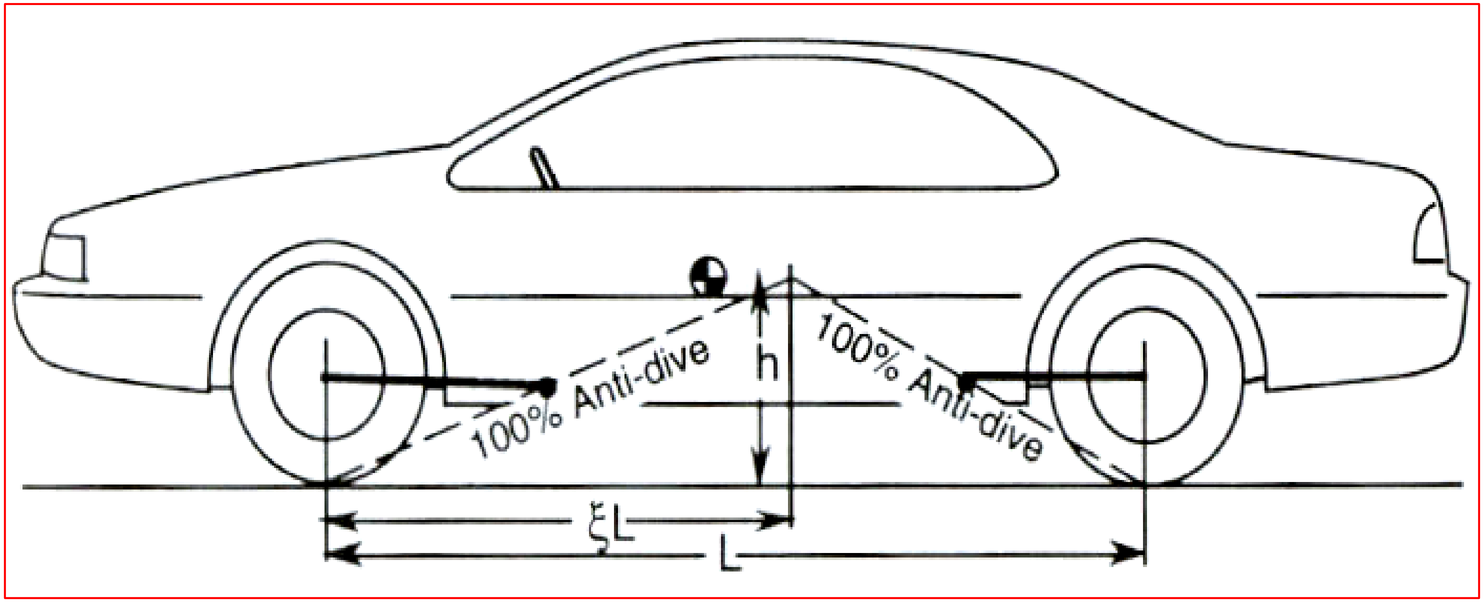
- The longitudinal load transfer incidental to braking acts to pitch the vehicle forward producing "brake dive."
- Just as a suspension can be designed to resist acceleration squat, the same principles apply to generation of anti-dive forces during braking.
- Because virtually all brakes are mounted on the suspended wheel (the only exception is in-board brakes on independent suspensions), the brake torque acts on the suspension and by proper design can create forces which resist dive.
- Using an analysis similar to that developed for the four-wheel-drive anti-squat example given previously, it can be shown that the anti-dive is accomplished when the following relationships hold:

Front suspension: $$\frac{e_f}{d_f} = \tan \beta_f = -\frac{h}{\xi L}$$

Rear suspension: $$\frac{e_r}{d_r} = \tan \beta_r = \frac{h}{(1-\xi)L}$$

Where:

ξ = Fraction of the brake force developed on the front axle eg: 60: 40.



- To obtain 100% anti-dive on the front and 100% anti-lift on the rear, the pivot for the effective trailing arm must fall on the locus of points defined by the defined ratios.
- If the pivots are located below the locus, less than 100% anti-dive will be obtained; if above the locus the front will lift and the rear will squat during braking.
- In practice, 100% anti-dive is rarely used. The maximum anti-dive seldom exceeds 50%.
- Full anti-dive requires that the pivot be located above the point required for full anti-squat. Thus acceleration lift would be produced on solid drive axles.

- Flat stops are subjectively undesirable.
- With full anti-dive, front suspension caster angle changes may increase the steering effort substantially during braking.
- The required steering system geometry may be quite complex.
- Excessive variation in rotational speed can occur in the drivetrain as the wheels move in jounce and rebound causing rattling and noise in the drive gears.
- In the rear suspension, oversteer problems may be created by the high location of the pivot.
- Brake hop may be induced if the effective trailing arm is too short. The propensity for brake hop is reduced by a suspension design with a long effective arm.
- NVH performance may be compromised.

Leaf Spring Windup

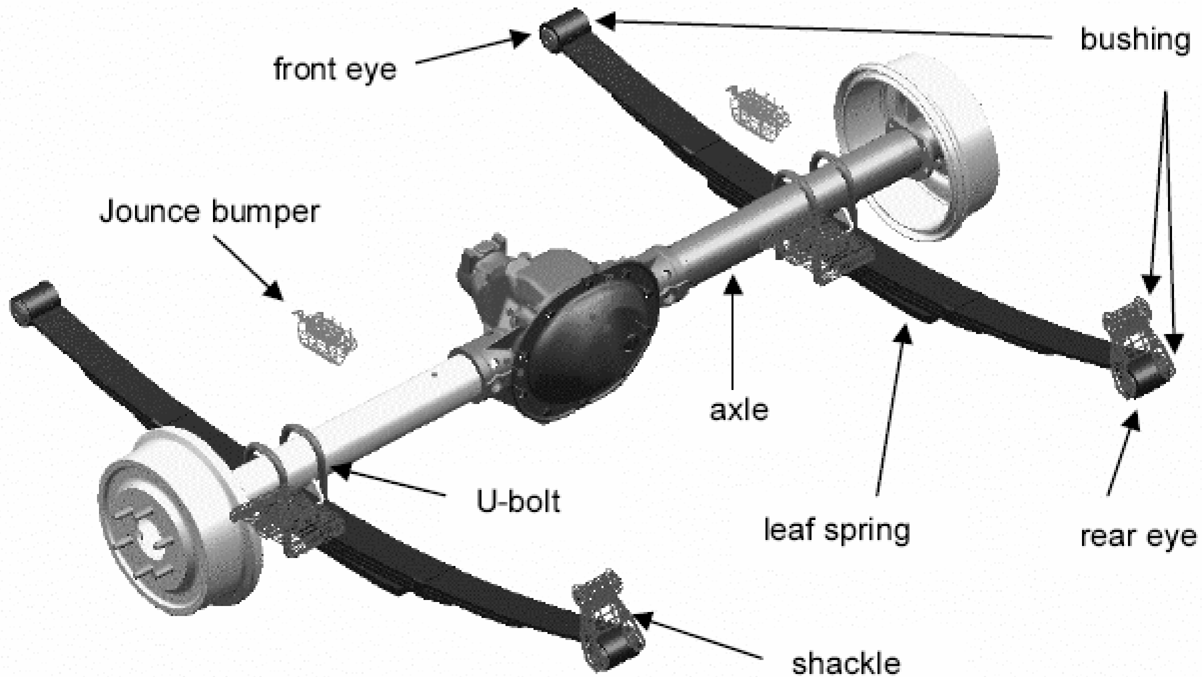
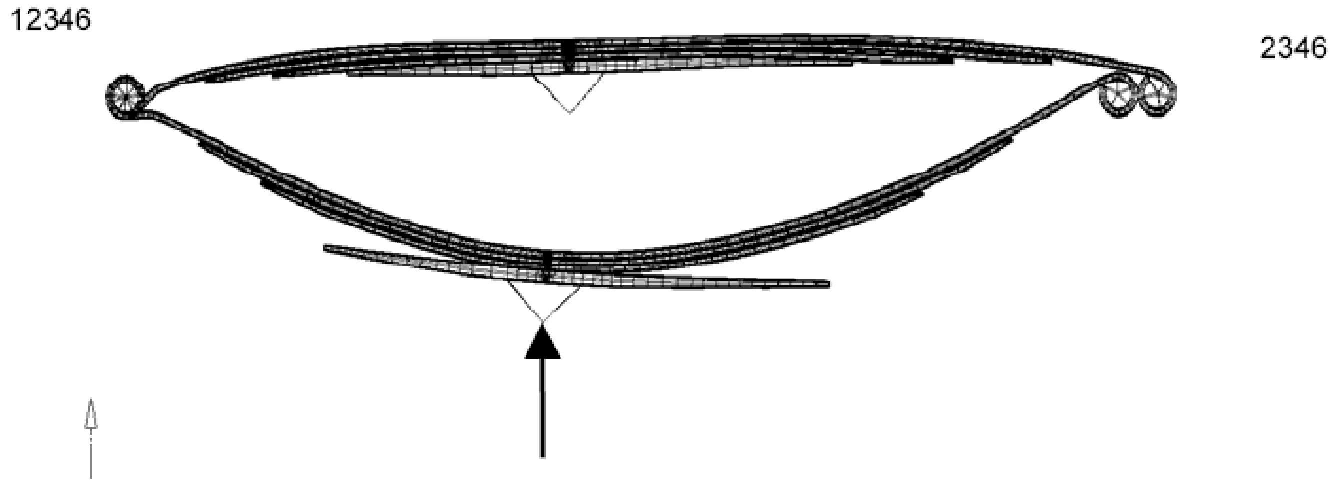


Figure 1. A Hotchkiss suspension.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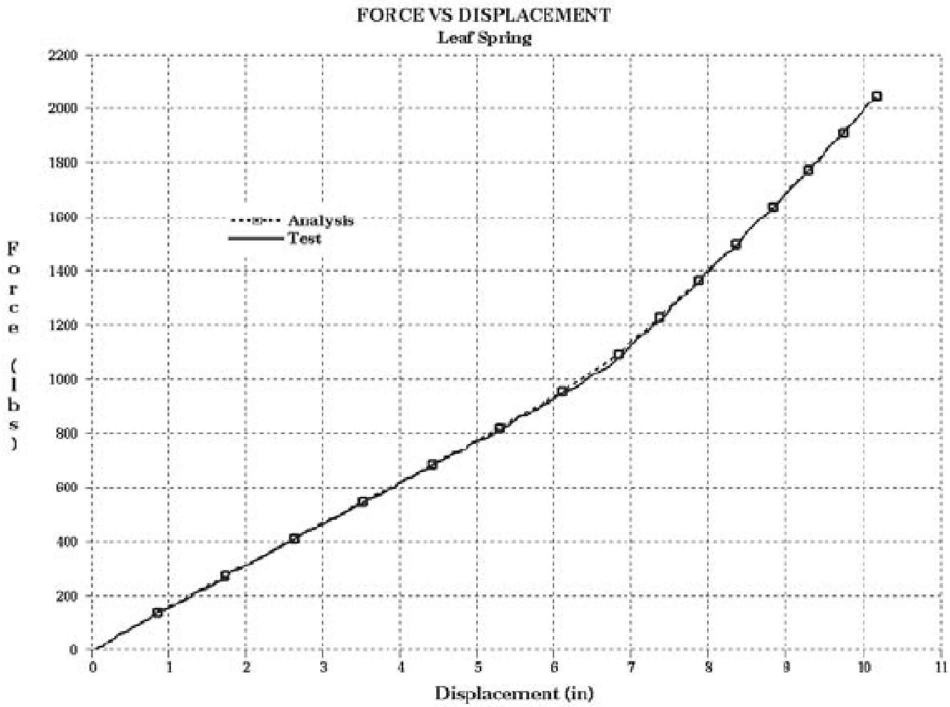
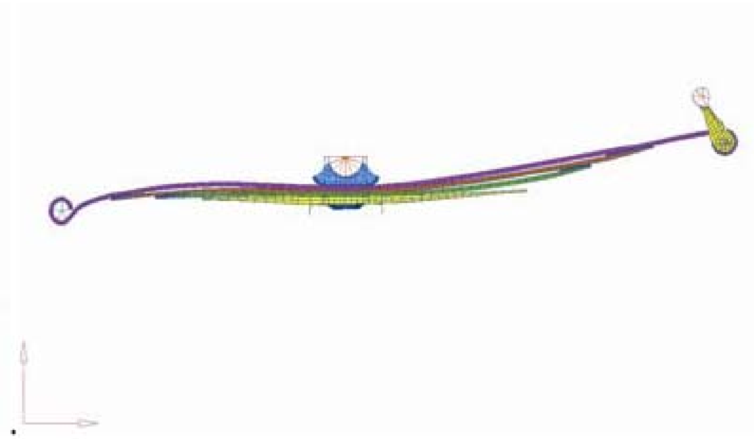
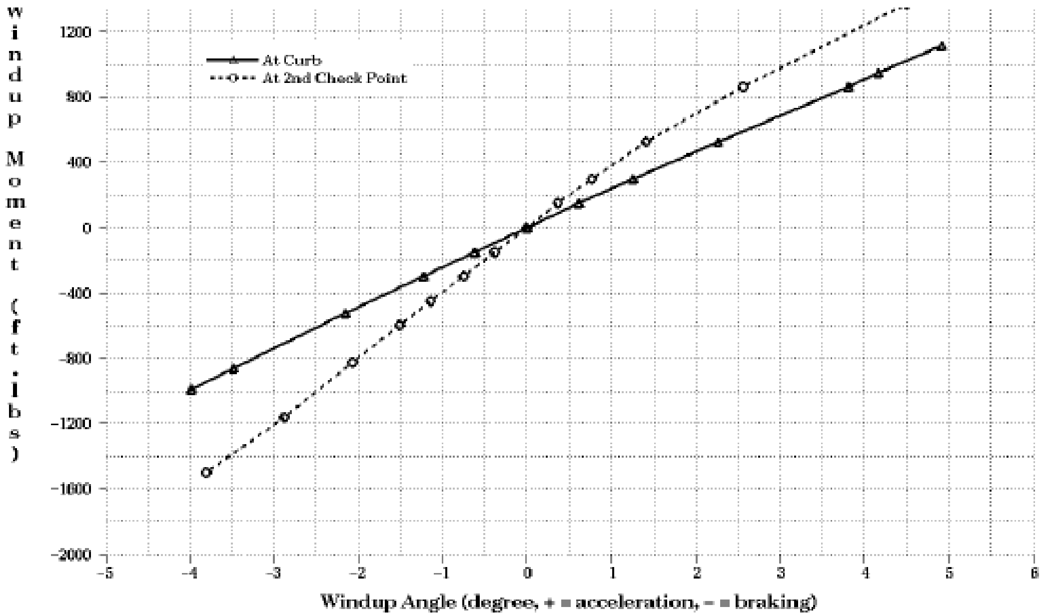
2346


FORCE VS DISPLACEMENT
Leaf Spring
Analysis
Test
Force (lbs)
Displacement (in)
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
1
2
3
4
5
6
7
8
9
10
11

(e) Braking at 2nd checkpoint



(f) Acceleration at 2nd checkpoint



2nd Check Point- Laden Condition