Chapter 11

- Suspension Design

Suspension Design Process

- Selecting appropriate vehicle level targets
- Selecting a system architecture
- Choosing the location of the 'hard points', or theoretical centres of each ball joint or bushing
- Selecting the rates of the bushings-compliance
- Analysing the loads in the suspension
- Designing the spring rates
- Designing shock absorber characteristics
- Designing the structure of each component so that it is strong, stiff, light, and cheap
- Analysing the vehicle dynamics of the resulting design

Vehicle Level Targets

- Maximum steady state lateral acceleration (in understeer mode)
- Roll stiffness (degrees per g of lateral acceleration)
- Ride frequencies
- Lateral load transfer percentage distribution front to rear
- Roll moment distribution front to rear
- Ride heights at various states of load
- Understeer gradient
- Turning circle
- Ackermann
- Jounce travel
- Rebound travel

System Architecture

- For the front suspension the following need to be considered
- The type of suspension (Macpherson strut or double wishbone suspension)
- Type of steering actuator (rack and pinion or recirculating ball)
- Location of the steering actuator in front of, or behind, the wheel centre
- For the rear suspension there are many more possible suspension types, in practice.

Location of Hardpoints

The hardpoints control the static settings and the kinematics of the suspension.

The static settings are

- Toe
- Camber
- Caster
- Roll center height at design load
- Mechanical (or caster) trail
- Anti-dive and anti-squat
- Kingpin Inclination
- Scrub radius
- Spring and shock absorber motion ratios

The kinematics describe how important characteristics change as the suspension moves, typically in roll or steer. They include

- Bump Steer
- Roll Steer
- Tractive Force Steer
- Brake Force Steer
- Camber gain in roll
- Caster gain in roll
- Roll centre height gain
- Ackerman change with steering angle
- Track gain in roll

Compliance

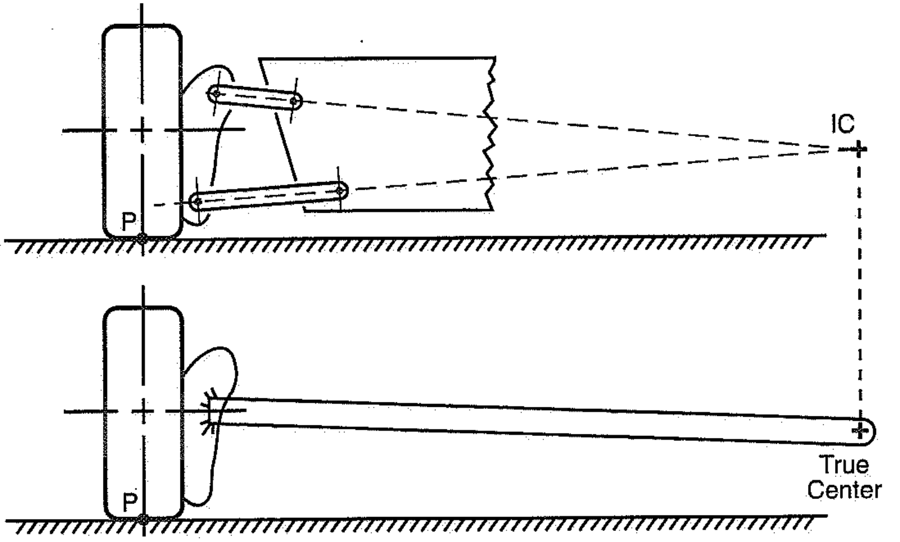
- The compliance of the bushings, the body, and other parts modify the behaviour of the suspension.
- In general it is difficult to improve the kinematics of a suspension using the bushings, but one example where it does work is the toe control bush used in Twist-beam rear suspensions.
- More generally, modern cars suspensions include an NVH bush. This is designed as the main path for the vibrations and forces that cause road noise and impact noise, and is supposed to be tunable without affecting the kinematics too much;

Loads

- Once the basic geometry is established the loads in each suspension part can be estimated. This can be as simple as deciding what a likely maximum load case is at the contact patch, and then drawing a Free body diagram of each part to work out the forces, or as complex as simulating the behaviour of the suspension over a rough road, and calculating the loads caused. Often loads that have been measured on a similar suspension are used instead - this is the most reliable method.

Instant Centre

- It is a projected imaginary point that is effectively the pivot point of the linkage at that instant
- Two short links can be replaced with one longer one
- As the linkage is moved the centre moves, so proper geometric design not only establishes all the instant centers in their desired positions at ride height, but also controls how fast and in what direction they move with suspension travel

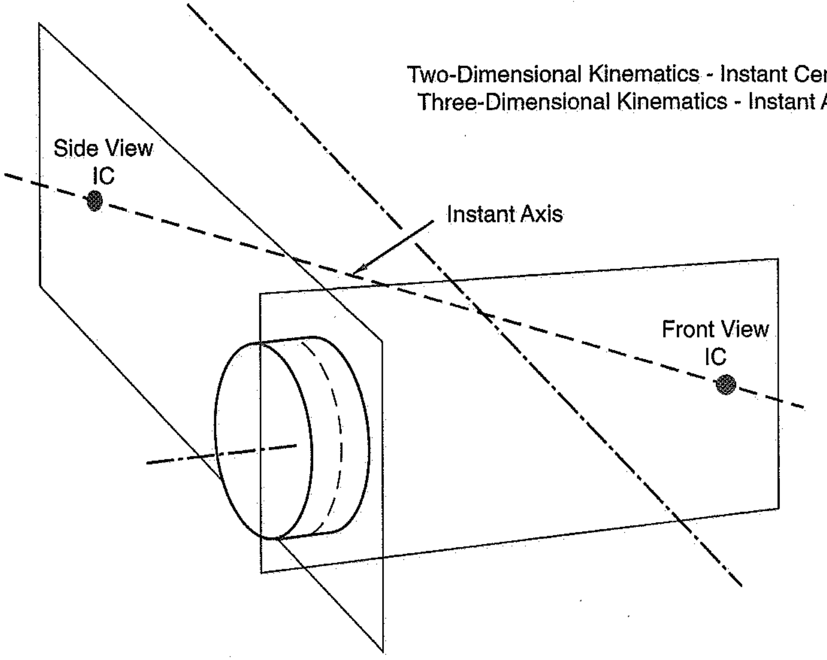


Different Views of Instant Centre

- Instant centre – Front view
  - The instant centre defines the camber change rate, part of the roll centre information, scrub motion, and data needed to determine the steer characteristics
- Instant Centre-Side View
  - The instant centre define the wheel forth and aft path, anti-lift and anti dive/squat information, and caster change rate

Instant Axis

- The instant axis is the line connecting the instant centre in front and side view. This line can be thought of as the instant axis of motion of the knuckle relative to the body
- Rear axles have two instant axes, one for parallel bump and one for roll, these also may move with changes in ride height

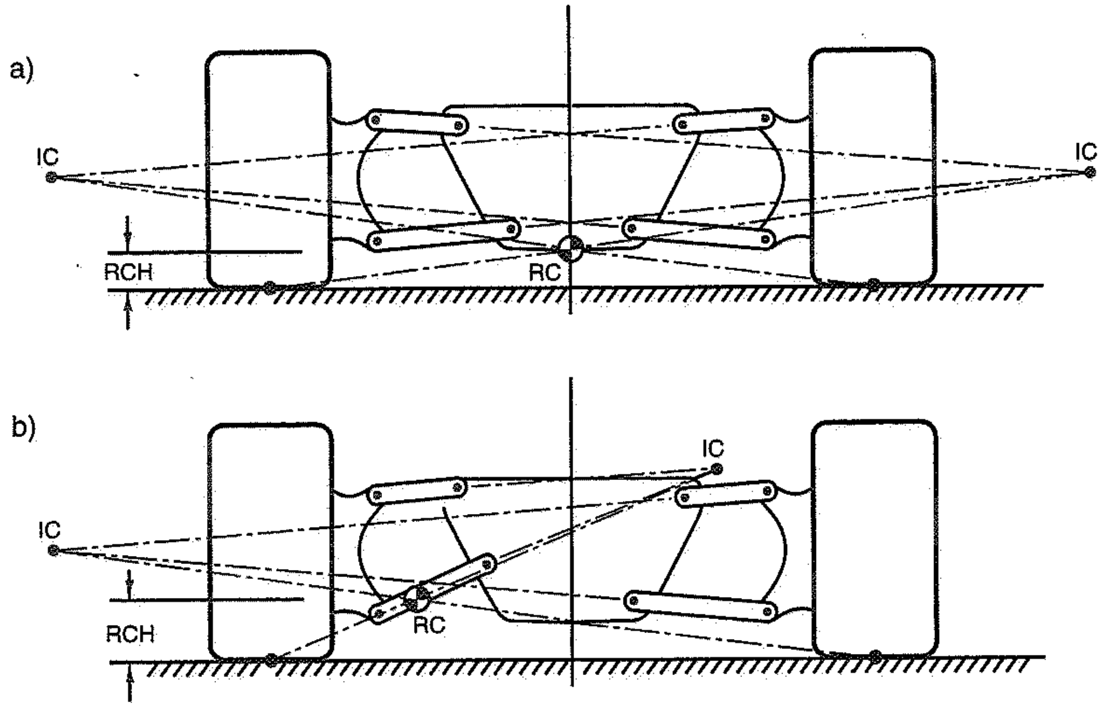


Independent Suspensions

- For all independent suspension there are two instant centres that establishes the properties of that particular design
- The side view instant centre (bump and droop) controls force and motion factors predominantly related to fore and aft accelerations
- The front view instant (or swing) centre controls force and motion factors due to lateral accelerations

Front View Swing Arm Geometry (fvsa)

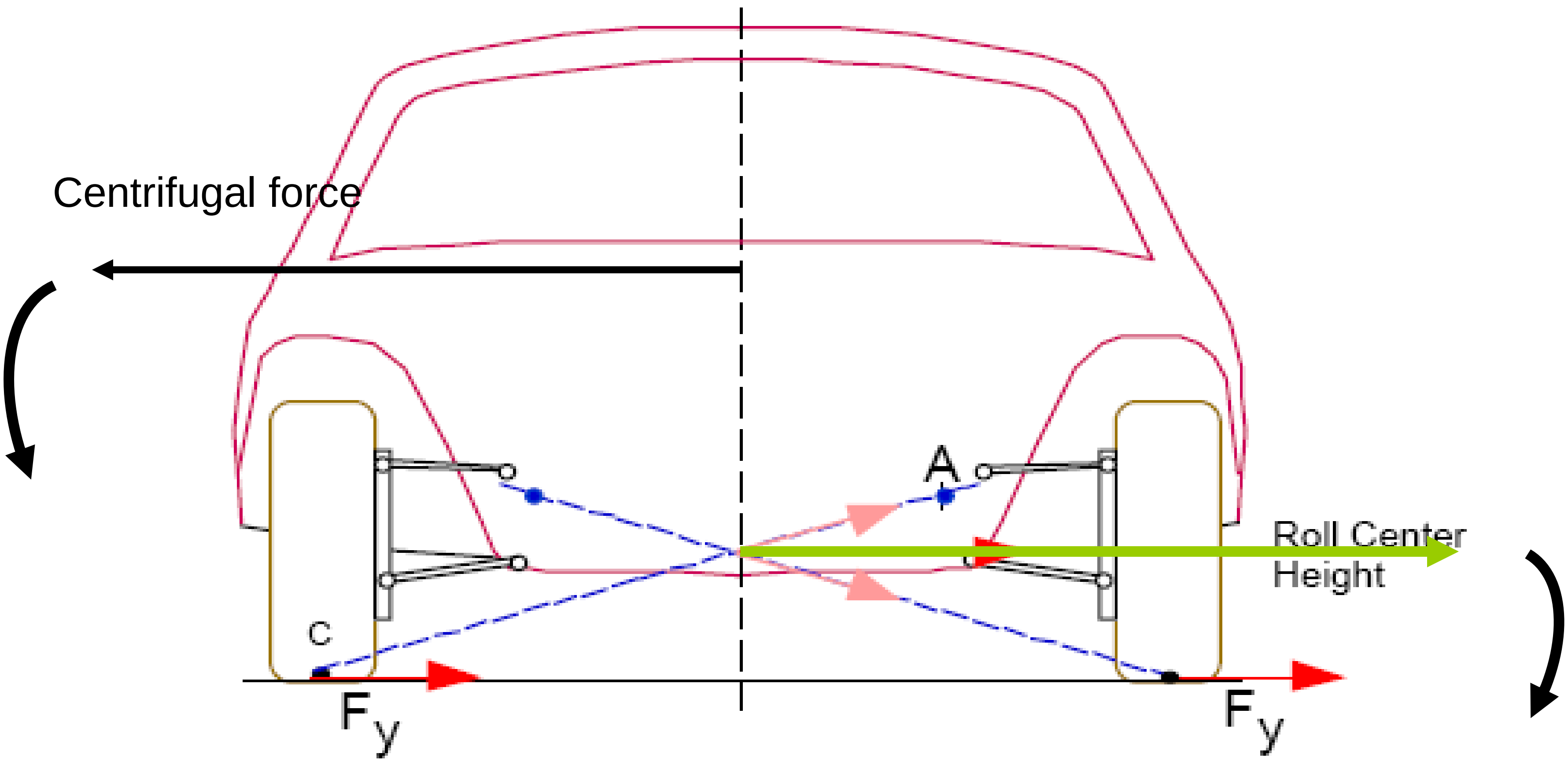
- The front view swing arm instant centre controls the roll centre height (RCH), the camber change rate, and tire lateral scrub
- The IC can be located inboard of the wheel or outboard of the wheel, it can be above ground level or below ground. The location is up to the designers performance requirements



Roll Centre Height

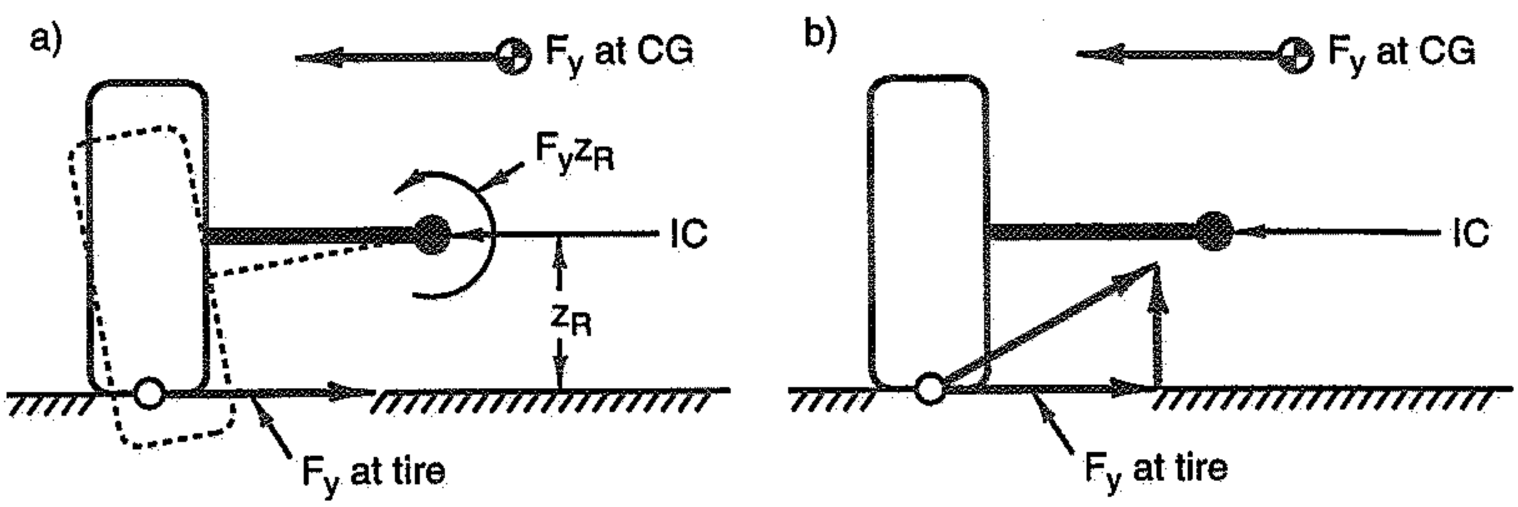
- The roll centre establishes the force coupling point between the unsprung and sprung masses
- Whenever a vehicle corners, the centrifugal force acts at the Centre of Gravity of the vehicle
- The Centrifugal force is reacted at the tyre road contact as lateral force
- The lateral force at the tyres can be translated to the roll centre if the appropriate force and moments are shown
- The higher the roll centre the smaller the rolling moment about the roll centre-the rolling moment is resisted by suspension springs, the lower the roll centres the larger the rolling moment
- The higher the roll centre the lateral force that acts at the roll centre is higher off the ground
- Lateral force * the distance to the ground is called non rolling overturning moment

Roll Centre heights are trading off the relative effects of the rolling and non rolling moments

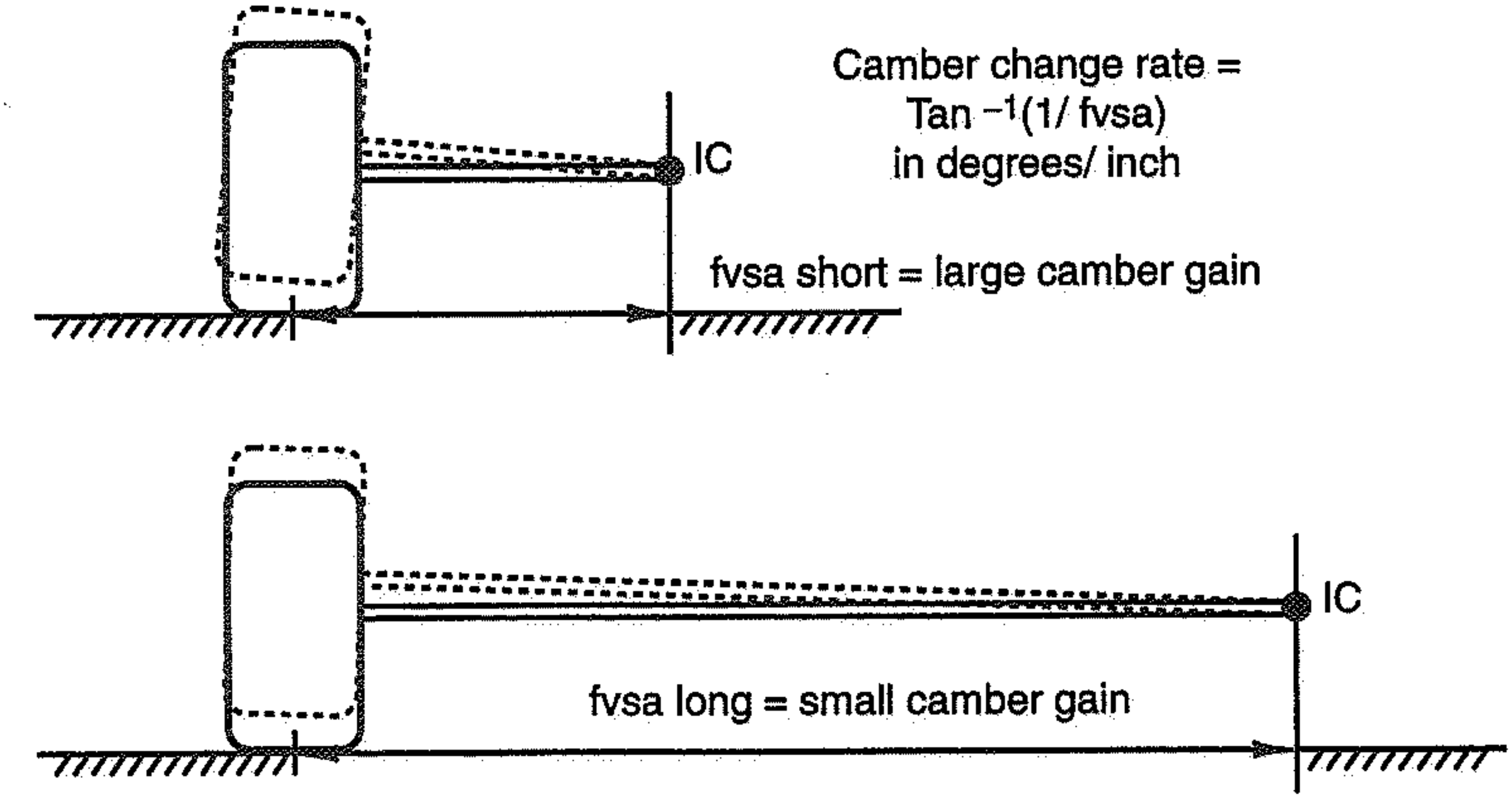


Jacking Effect

- If the roll centre height is high, the lateral force acting at the tyre generates a moment about the instant centre. This moment pushes the sprung mass up and wheel down-jacking effect. The reverse happens if the roll centre is below the ground



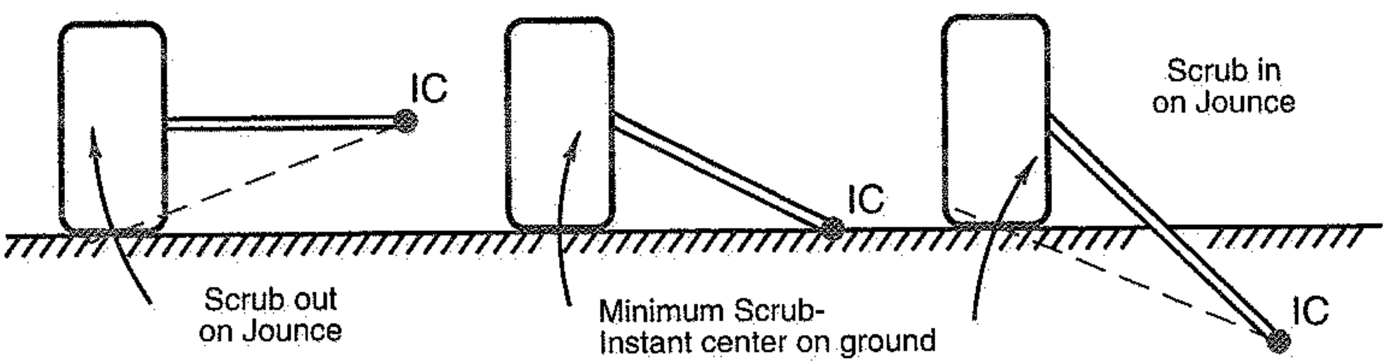
Camber Change Rate



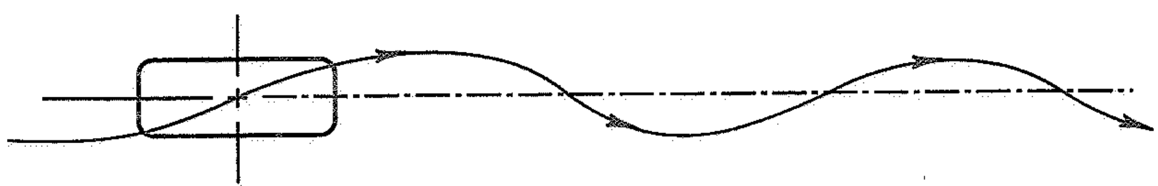
Short and long arm-why?

Scrub

- This is the lateral motion relative to the ground that results from vertical motion of the wheel. Scrub occurs in every suspension system
- The amount of scrub is a function of the absolute and relative lengths of the control arms and the position of the front view instant centre relative to ground
- If the front view instant centre is at any position other than the ground level scrub radius is increased



On a rough ground the wheel path is not a straight line if there is a scrub



Side View Swing Arm Geometry (svsa)

- Typically, the instant centre is behind and above the wheel centre on front suspensions and it is ahead and above on most rear suspensions

Anti Features

- The Anti effect in suspension is a term that actually describes the longitudinal to vertical force coupling between the sprung and un sprung masses. It results purely from the angle or slope of the side view swing arm
- Suspension “anti’s” do not change the steady state load transfer at the tire patch

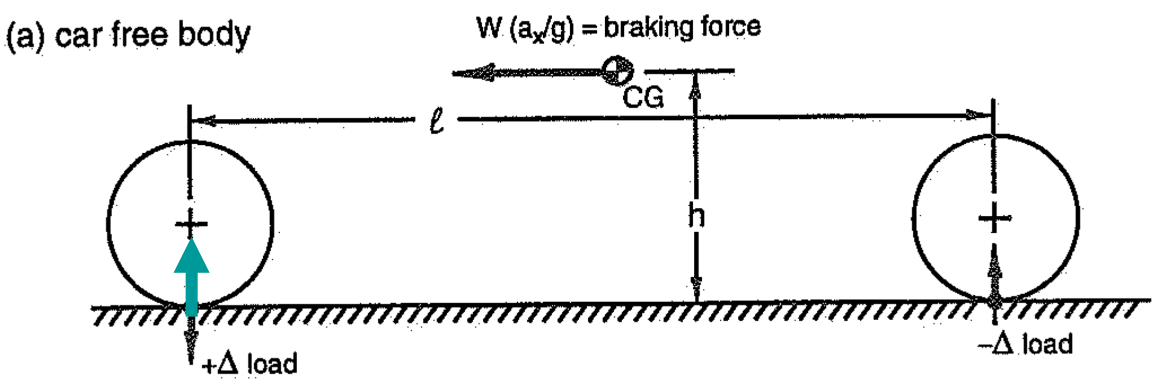
Anti- Dive (Braking)

- Dynamic Load Transfer during braking to the front axle

$$\Delta \text{Load} \times \ell = W \times \frac{a_x}{g} \times h$$

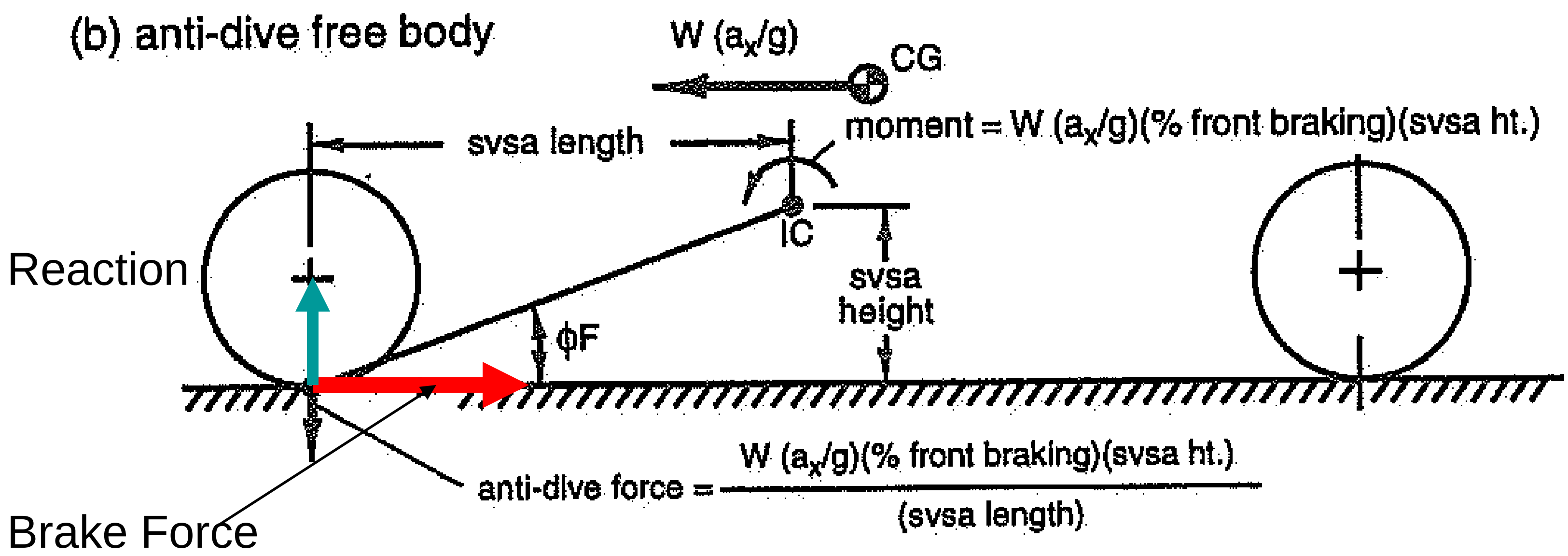
$$\Delta \text{Load} = W \times \frac{a_x}{g} \times \frac{h}{\ell}$$

Reaction

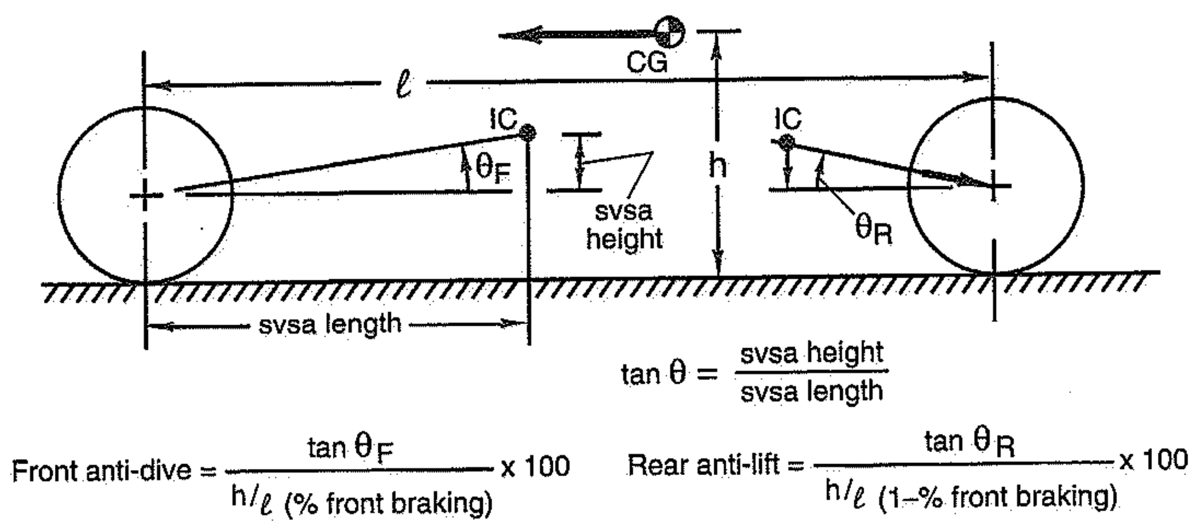


- This load transfers through the suspension springs resulting its deflection, hence dive moment.

- What essentially required is to reduce the load that is passing through the spring and make it pass through the control arm



Derivation of braking anti features with outboard brakes.



Braking anti features with inboard brakes.

$$\% \text{ anti-dive front} = \frac{m(a_x/g)(\% \text{ front braking})(\text{svsa - height}/\text{svsa - length})}{m \times (a_x/g) \times (h/\ell)}$$

$$= (\% \text{ front braking})(\tan \phi_F)(\ell/h)$$

For rear anti-lift calculation, substitute $\tan\phi_R$ and % rear braking.

If a suspension has 0% anti, then all the load transfer is reacted by the springs and the suspension will deflect proportional to the wheel rate, none of the transferred load is carried by the suspension arms; 0% anti occurs when θ or ϕ in the figures equals zero

Anti Lift

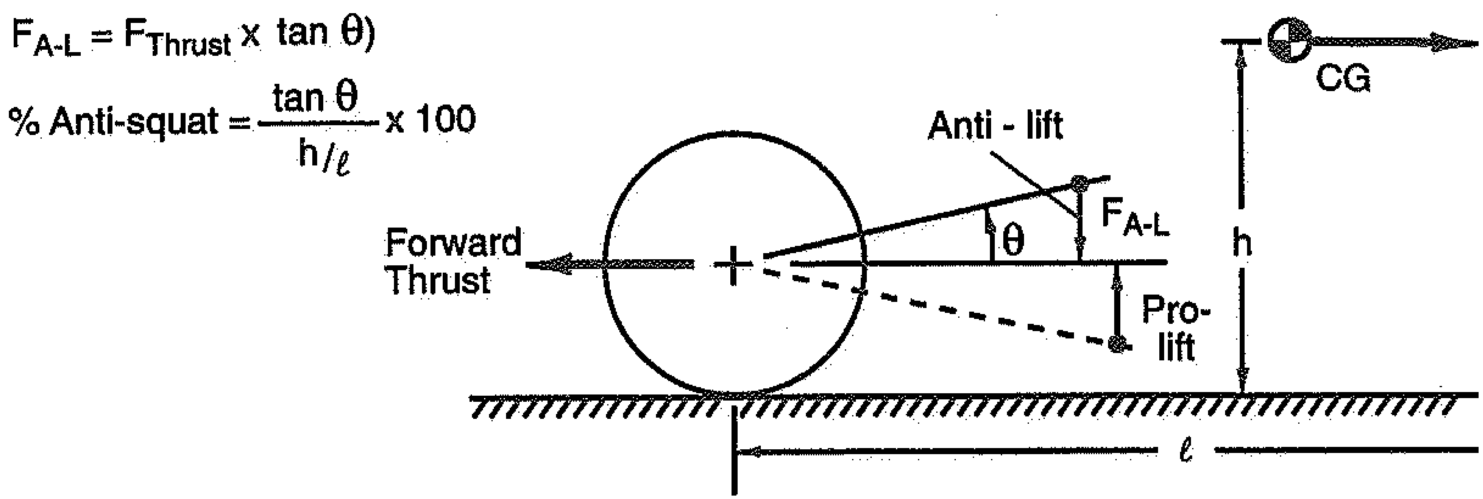


Figure 17.14 Front-wheel-drive anti-lift (or pro-lift).

Anti Squat

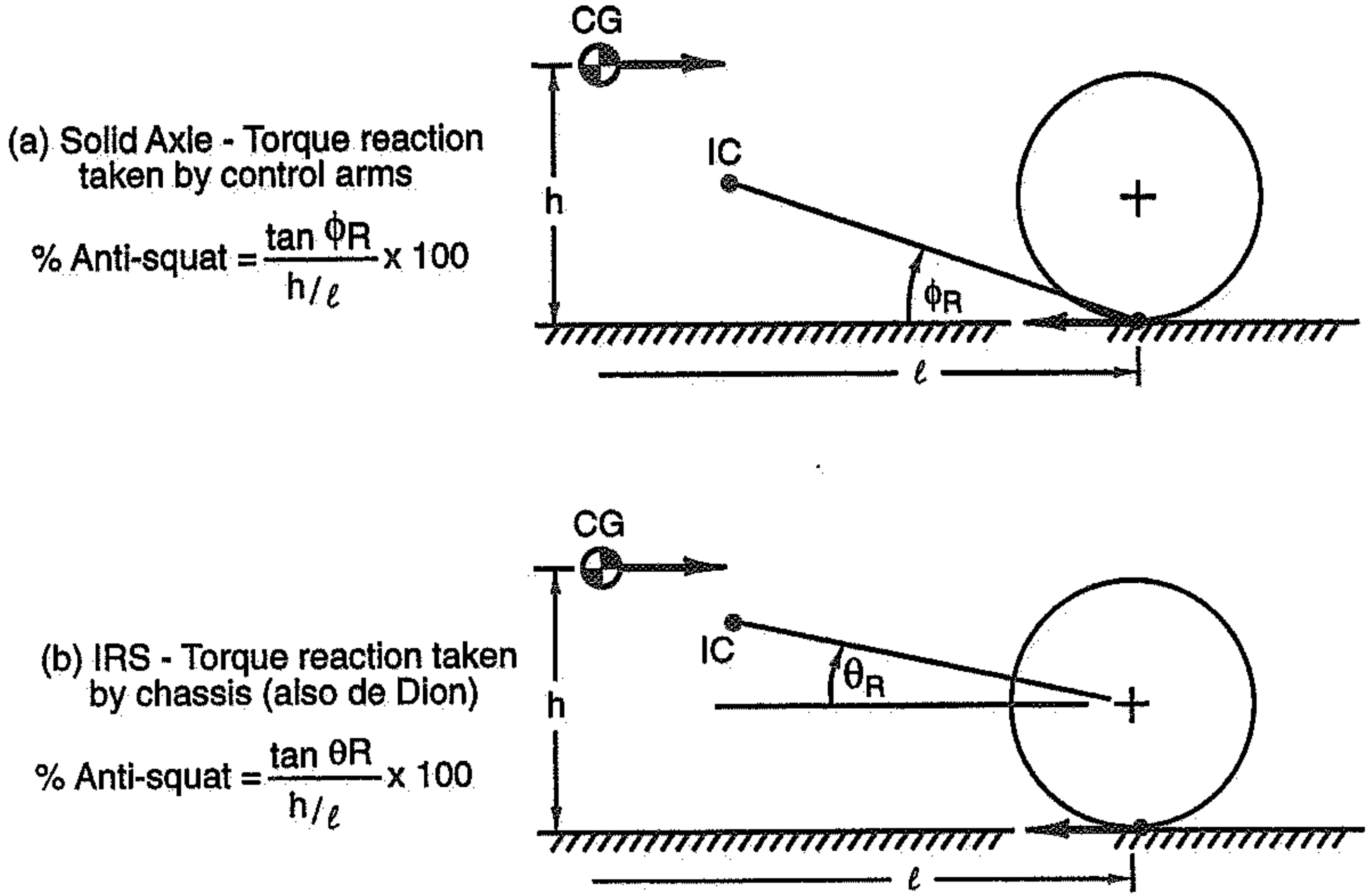
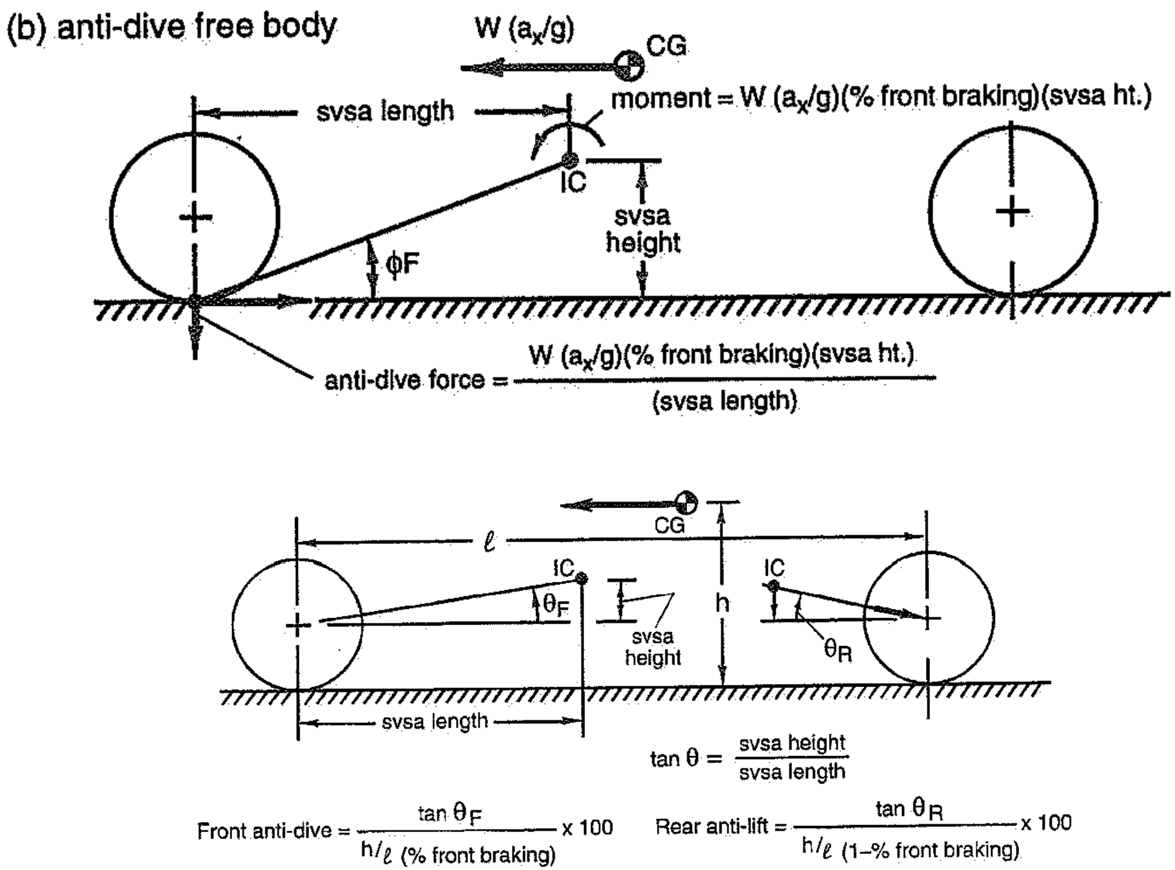


***Figure 17.15** Rear anti-squat, (a) solid axle and (b) independent rear suspension.*

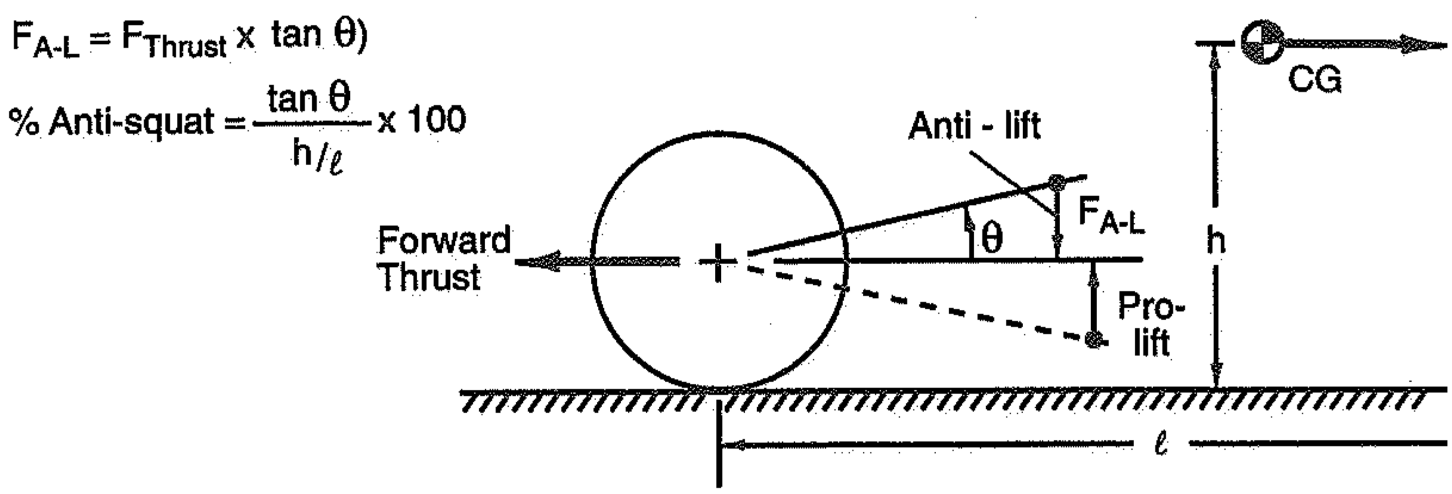
Anti's

- Anti-dive geometry in front suspension reduces the bump deflection under forward braking



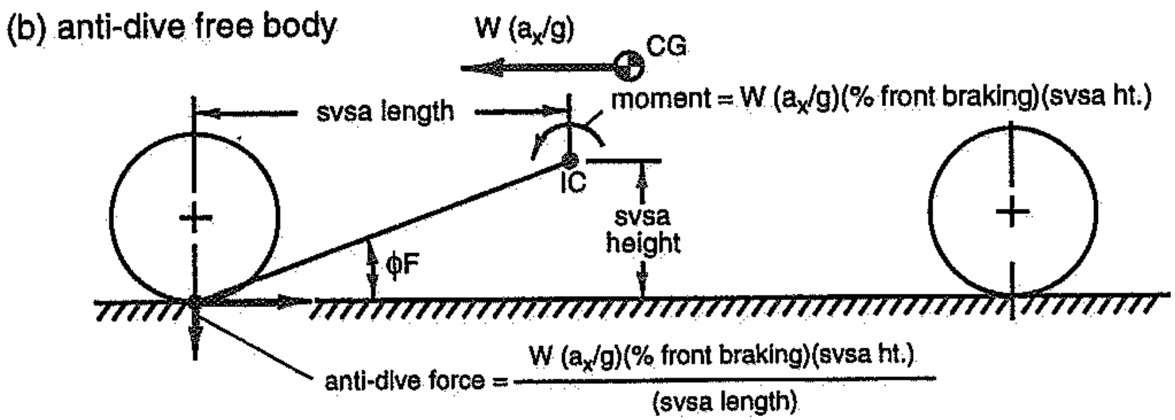
Anti's

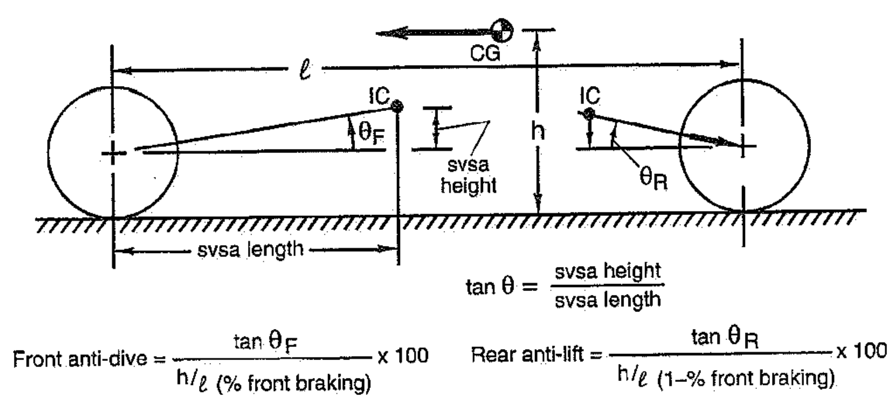
- Anti-lift geometry in front suspension only occurs with front wheel drive and it reduces droop deflection under forward acceleration



Anti's

- Anti-lift in rear suspension reduces droop travel in forward braking





Anti's

- Anti-squat in rear suspension reduces the bump travel during forward acceleration on rear wheel drive cars only

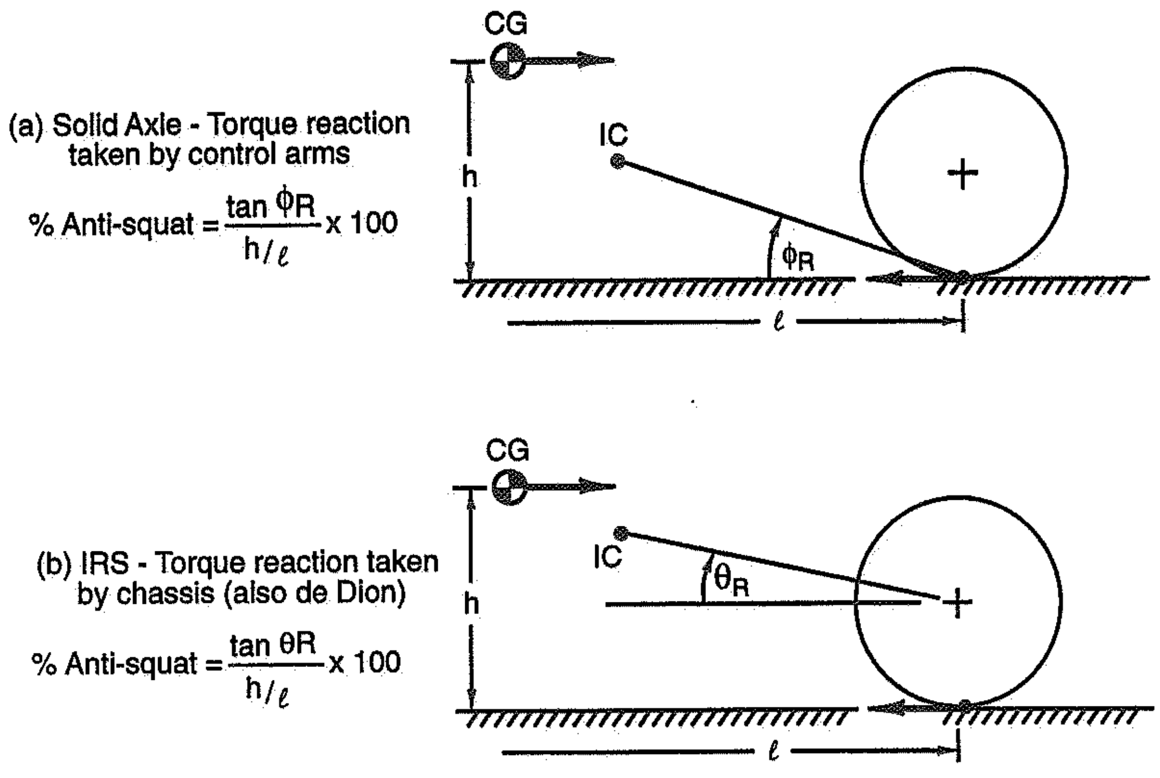


Figure 17.15 Rear anti-squat, (a) solid axle and (b) independent rear suspension.

Wheel Path

The path of the wheel center relative to the sprung mass in the side view is totally controlled (for "perfect" kinematics, no compliances) by the position of the instant center.

- If the IC is rearward and above wheel center height or forward and below, the wheel will move forward as it rises.
- If the instant center is behind and below wheel center or ahead and above, the wheel will move rearward as it rises.

The amount of curvature that the wheel center path has as the wheel rises or falls is totally a function of the swing arm length. The wheel path is not generally a concern in a race car, but on production cars the wheel center path affects the isolation capability when hitting impact bumps. The situation is similar to scrub but rotated 90°.

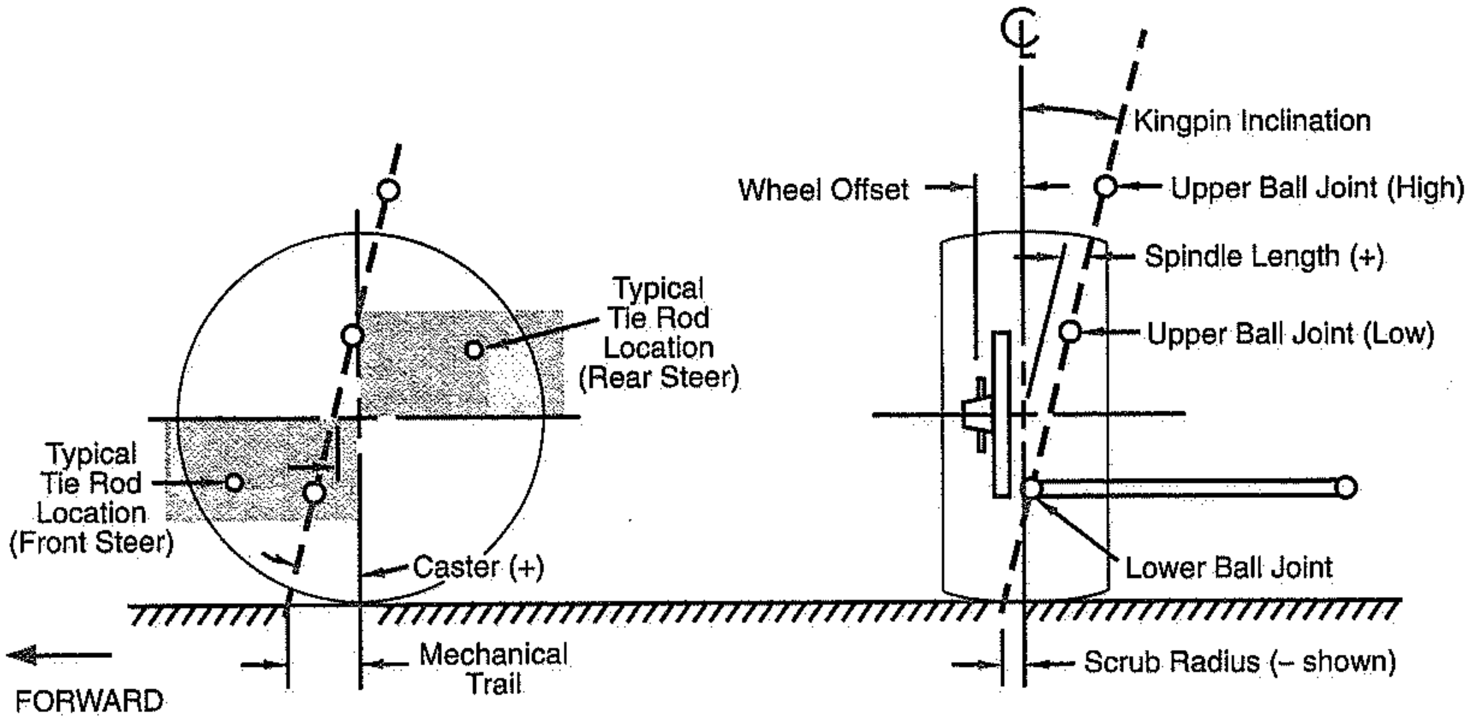
Caster Changes

Just like camber in the front view, caster changes in side view purely as a function of the length of the side view swing arm. There is very little reason to intentionally have caster change with suspension travel. What results is generally accepted as a function of some other parameter establishing the length of the swing arm. One result of caster change with suspension travel is that the bump-steer curve is more difficult to make linear throughout the total range of travel.

Suspension Design

Front Suspension-independent

- Design issues- Establish Packaging parameters



Packaging parameters

- Tire size, rim diameter and width
- Wheel offset
- Brakes, bearings
- Kingpin length, angle, scrub radius, spindle length
- The caster, The camber
- The knuckle design
- Tie rod position
- Rack location
- Trackwidth
- Decide the upper and lower ball joint positions
- Tie rod outer position

SLA-Suspension Design-front view Geometry

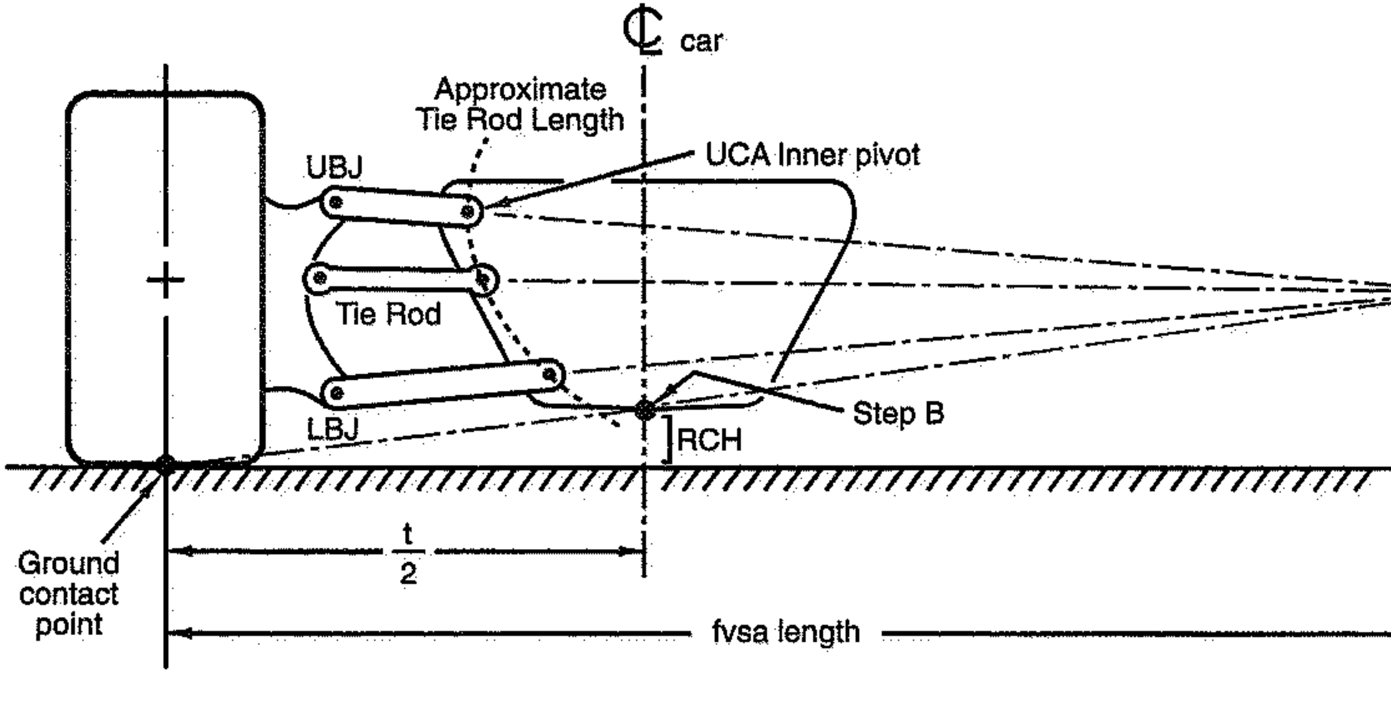
- Locating front view swing arm instant centre

$$\text{fvsa} = (t/2)/(1-\text{roll camber})$$

where t = track width

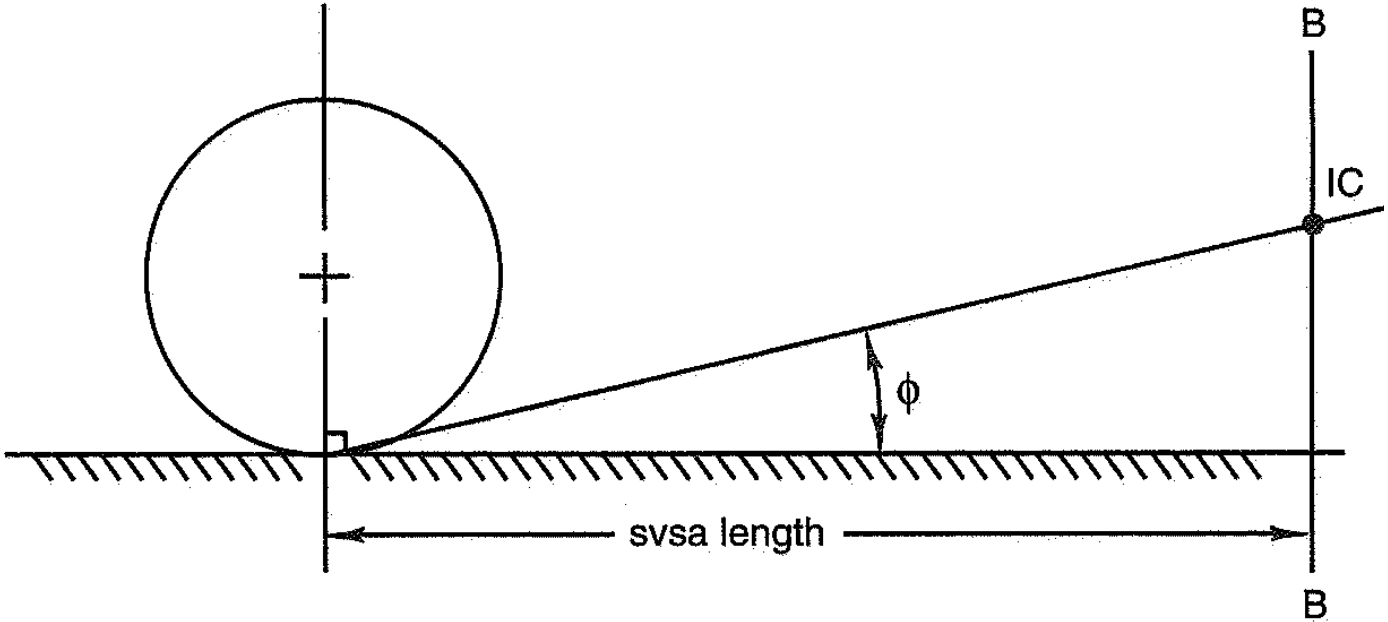
$$\text{Roll camber} = \frac{\text{wheel camber angle}}{\text{chassis roll angle}} \quad \text{(with both measured relative to the road)}$$

SLA-Suspension Design-front view Geometry



Step A—Establish front view swing arm length (line A-A)
Step B—Establish roll center location and project from ground contact point through RC to line A-A, establishing IC
Step C—Project lines from outer ball joints to IC
Step D—Choose control arm lengths to get inner pivot locations
Step E—Connect tie rod outer pivot to IC
Step F—Establish tie rod length.

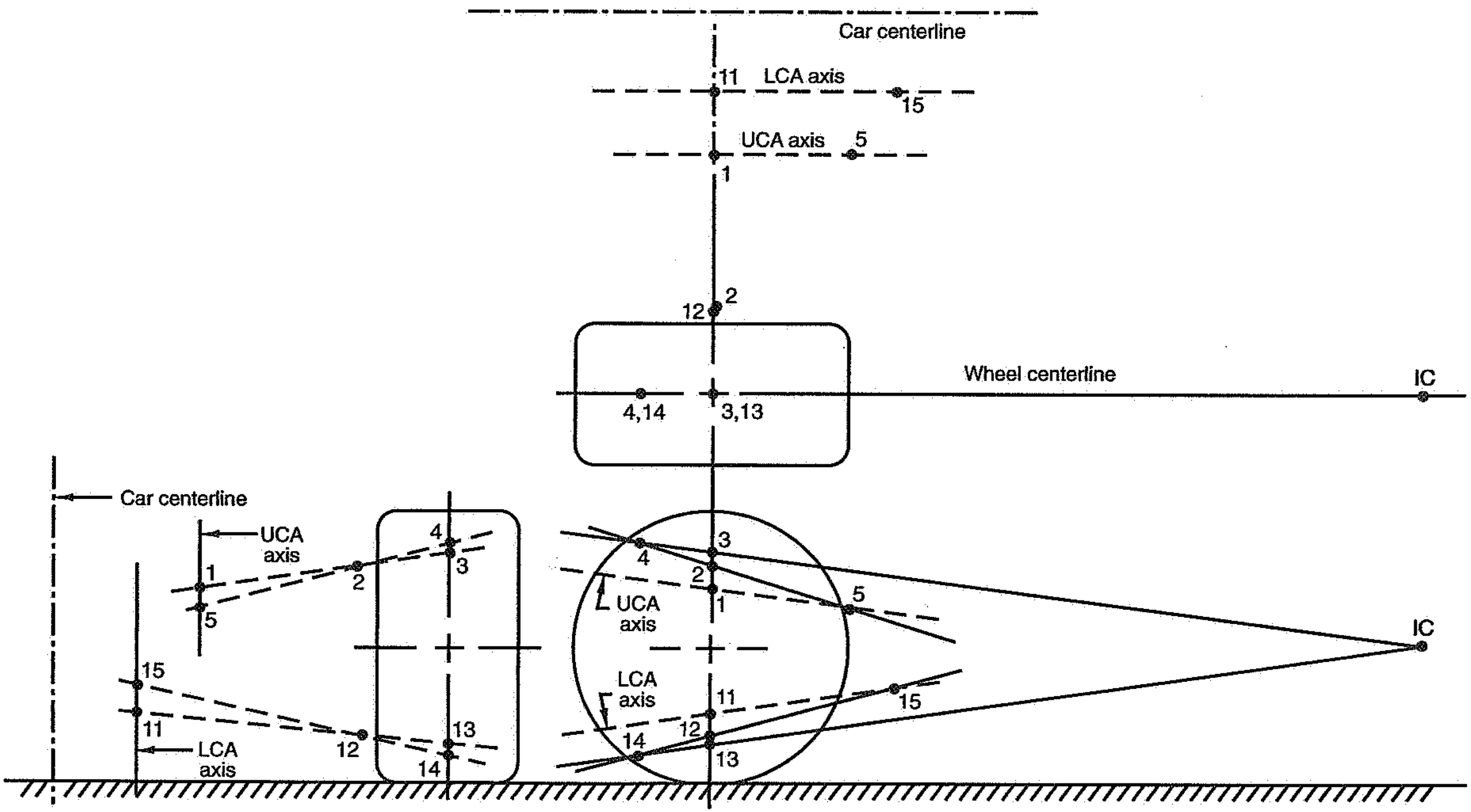
Side view Geometry



Step A—Angle ϕ establishes anti-dive
Step B—svsa length gives line B-B
Side view IC is the intersection of steps A and B.

Decide required anti-dive, carefully choose svsa length

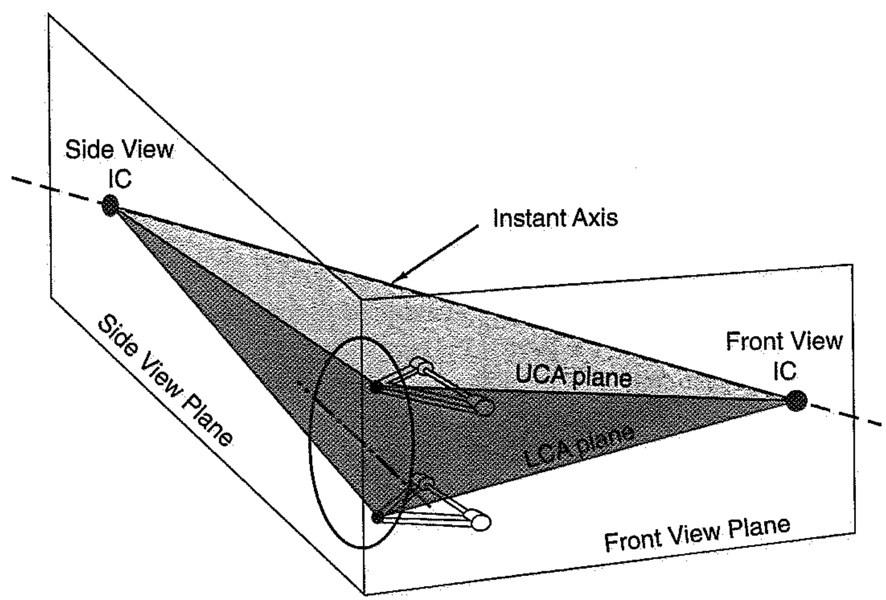
Inner pivot Axis Construction



In front view, the upper control arm inner pivot is point #1, the upper ball joint is point #2, and the extension into the longitudinal plane is point #3. For the lower control arm the corresponding points are #11, #12, and #13. These six points are transferred to the side view. Two lines in the side view from the side view instant center should be extended through and beyond points #3 and #13. Next we choose an arbitrary point in the side view on the line between the IC and point #3 that is a few inches ahead of point #3 and number it point #4. Repeat this procedure for the lower arm creating point #14. Next we project these points into the front view so that both the side view and the front view contain points #1 through #4 and #11 through #14.

To maintain the desired geometry all upper arm points (#1 through #4) must be in a single plane, and all lower arm points (#11 through #14) must be in a single plane. As long as we always project straight lines through two points in a plane and establish new points on these lines, the new points will remain in the plane. Next we project a line from point #4 through point #2. This is done in both views and the line is extended in the front view at least as far inboard as point #1. Repeat for the lower arm using point #14 through #12 inboard at least to point #11.

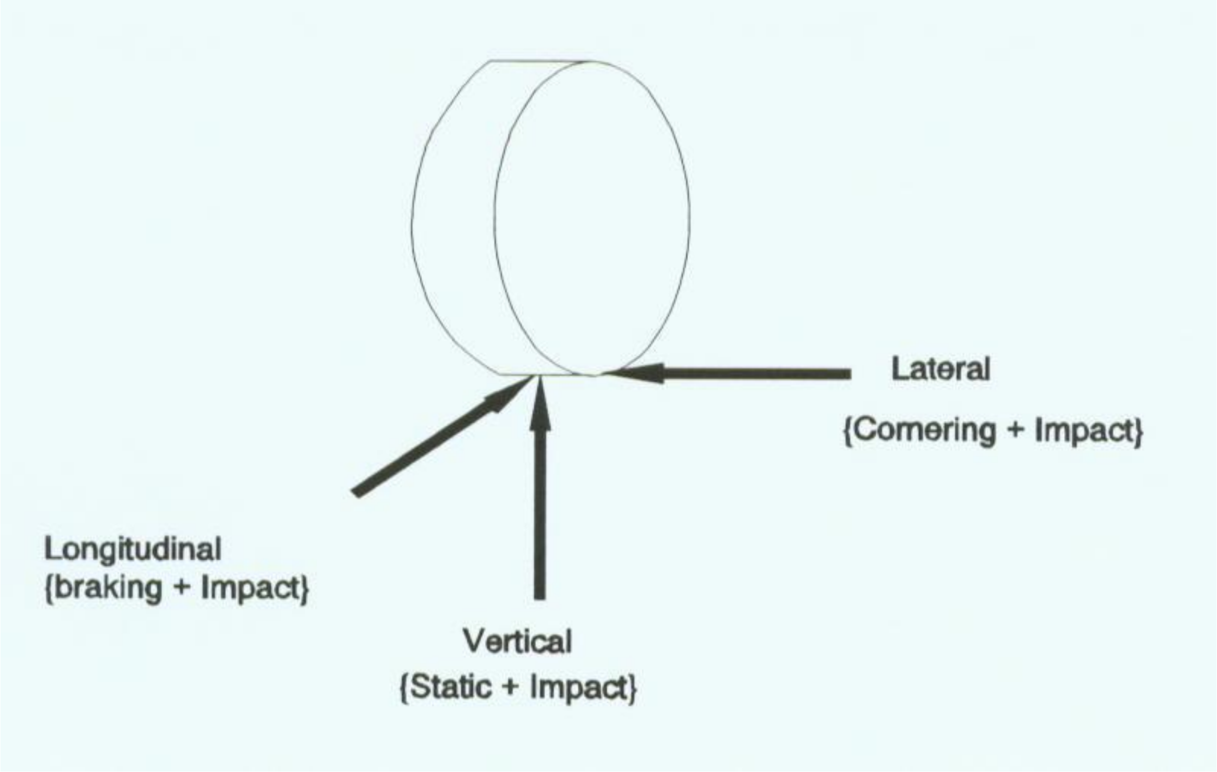
For almost all suspension designs it is acceptable and even desirable to have the inner pivots of the control arms parallel to the centerline of the car. Therefore the next step will be to draw a vertical line through point #1 in the front view; this line is the front projection of the upper control arm (uca) axis. Identify point #5 on this vertical line as the extension of a line from points #4 through point #2. Repeat for the lower control arm (lca) via a vertical line through point #11, finding point #15 from #14 through #12. Next project points #5 and #15 into the side view. Draw a line between #5 and #1 and between #15 and #11 (shown dashed in the side view). The control arm inner pivots must lie on these lines. They can be spread wider or narrower than the points but they must fall on the lines.



Design of other Suspension

- Ref. Race Car Vehicle Dynamics _Milliken

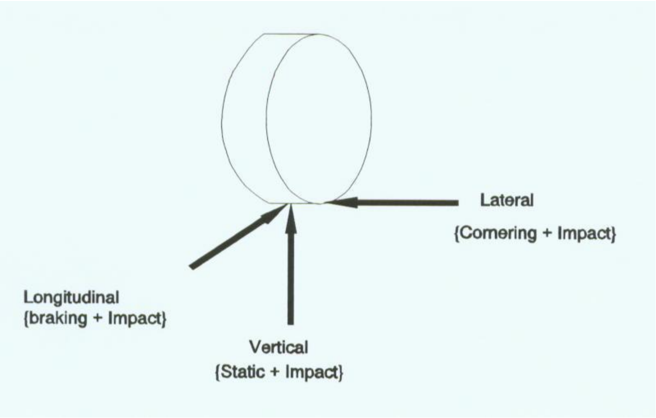
Suspension Load Distribution



WHEEL LOADS AND DIRECTIONS

Suspension Load Distribution

- Front Axle Braking per wheel:



$$F_B = \frac{\mu}{2}[static + dynamic \quad load]$$

$$= \frac{\mu}{2}[W\frac{b_{cg}}{l} + m\bar{a}\frac{h_{cg}}{l}] \quad = \frac{\mu}{2}W[\frac{b_{cg}}{l} + \frac{\bar{a}}{g}\frac{h_{cg}}{l}]$$

μ = tire-road coefficient of friction

b_{cg} = Cg-to-rear axle distance (L_R)

a = ave. longitudinal acceleration (deceleration)

h_{cg} = Cg height

W = total vehicle weight

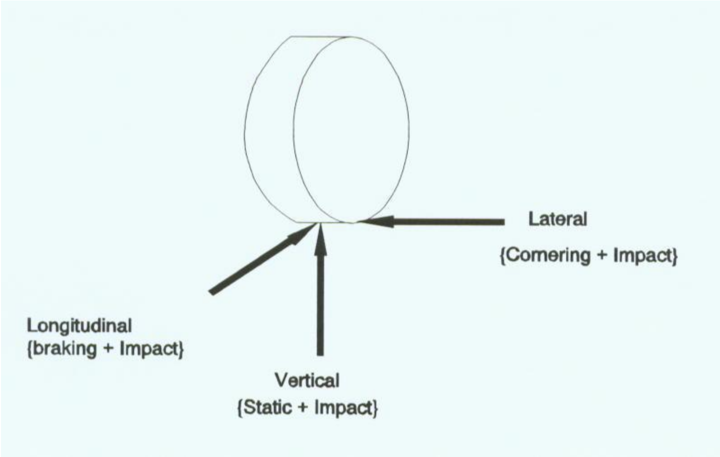
l = wheelbase length

m = vehicle mass

g = acceleration of gravity

Suspension Load Distribution

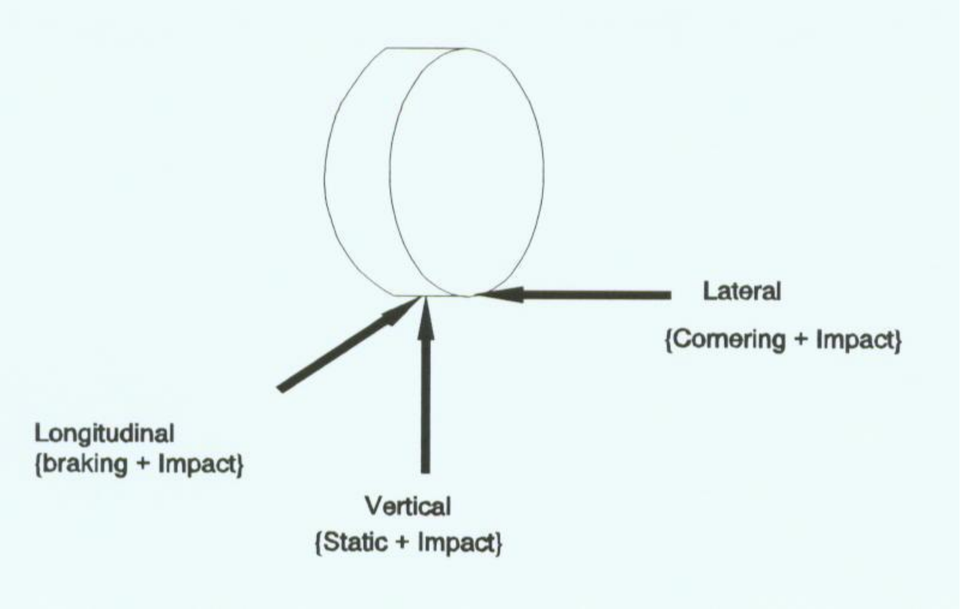
- VERTICAL: (Total is commonly considered as a 3 g load)



$$V = \frac{3}{2}[W\frac{b_{cg}}{l} + m\bar{a}\frac{h_{cg}}{l}] \quad = \frac{3}{2}W[\frac{b_{cg}\,g + \bar{a}\,h_{cg}}{gl}]$$

Suspension Load Distribution

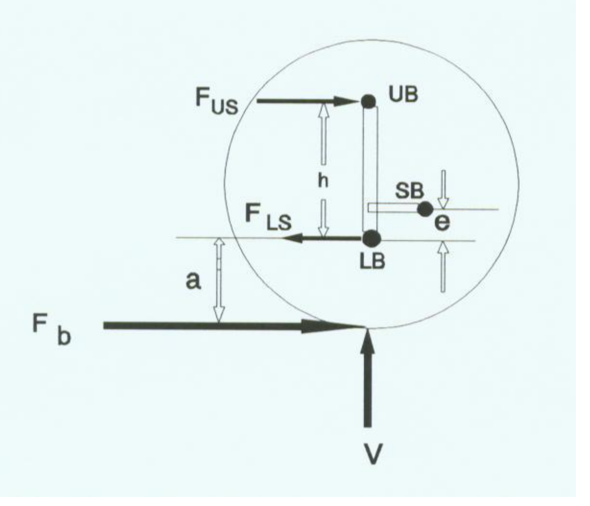
- LATERAL: (commonly considered as a 2 g load)



$$F_L = W[\frac{b_{cg}\, g + \bar{a}\, h_{cg}}{gl}]$$

Suspension Load Distribution

Side view front wheel SLA front suspension



LB is lower ball joint

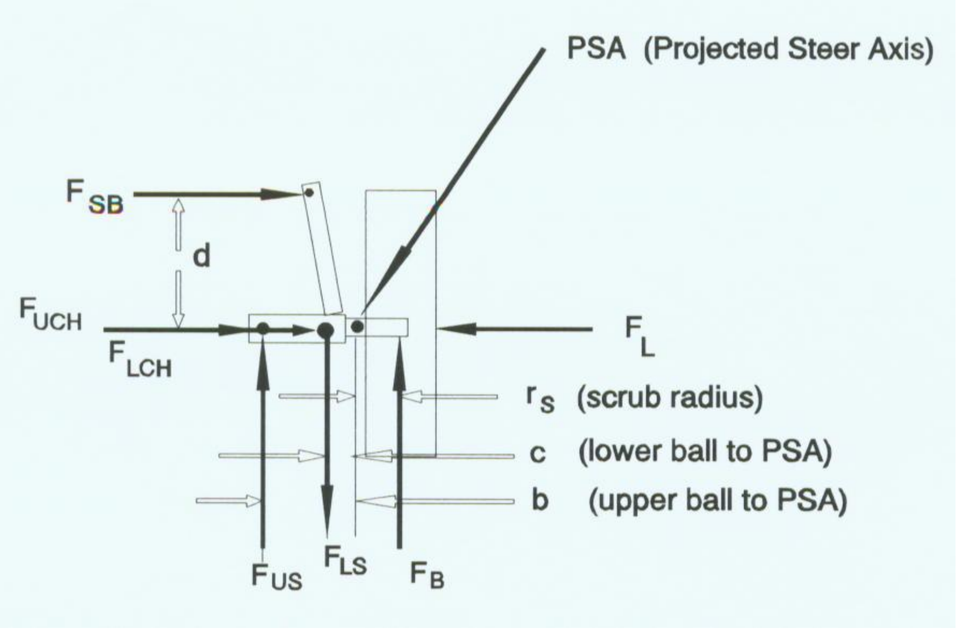
$$\sum M_{LB} = 0 \qquad F_{us} h = F_B a \qquad F_{us} = F_B \frac{a}{h}$$

$$\sum F_x = 0 \quad F_{US} - F_{LS} + F_B = 0$$

$$F_{LS} = F_{US} + F_B \qquad F_{LS} = F_B [\frac{a}{h} + 1]$$

Suspension Load Distribution

SLA front SUSPENSION TOP VIEW

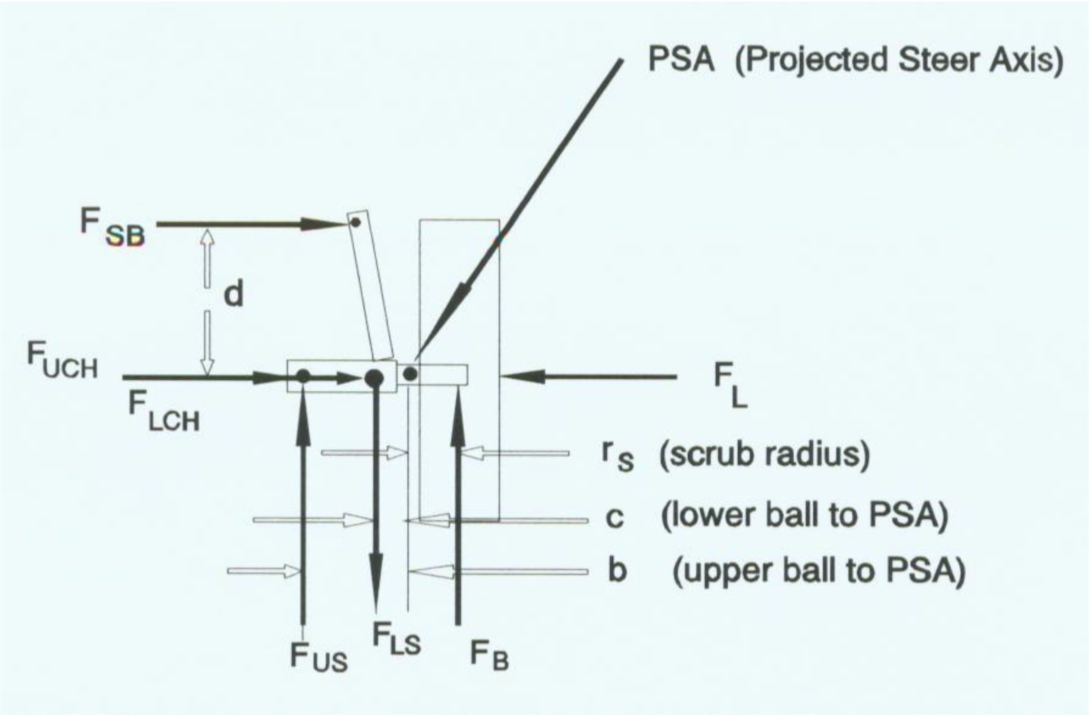


$$\sum M_{PSA} = 0$$

$$F_{SB} d + F_{US} b - F_{LS} c - F_b r_s = 0$$

$$F_{SB} = \frac{1}{d}[F_{LS} c + F_b r_s - F_{US} b] = \frac{F_b}{d}[r_s + c - \frac{a}{h}(b - c)]$$

Suspension Load Distribution



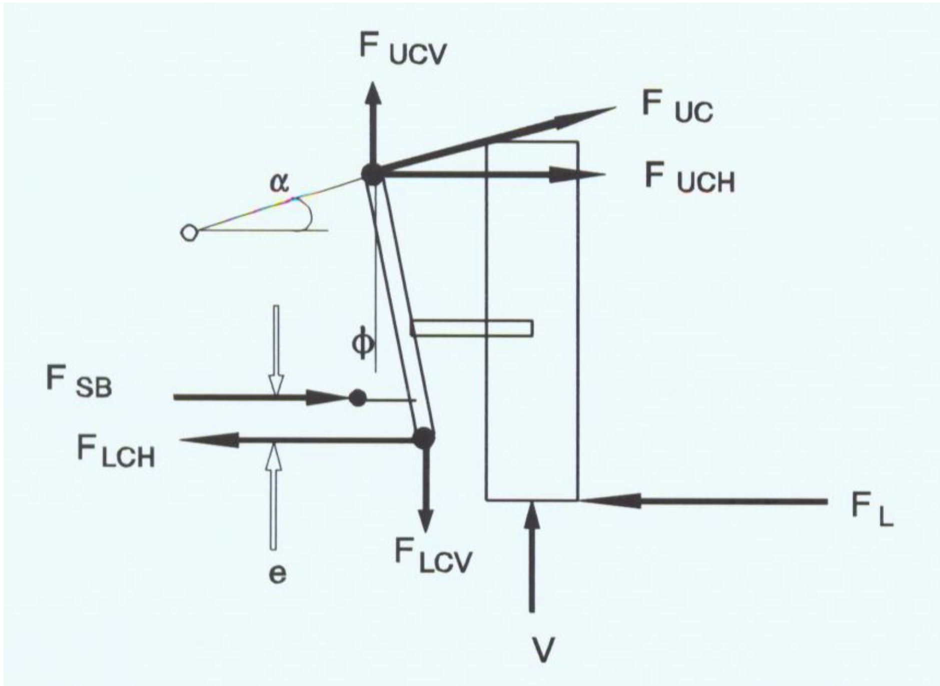
$$\sum F_y = 0$$

$$F_{UCH} + F_{SB} - F_{LCH} - F_L = 0$$

$$F_{LCH} = F_{UCH} + F_{SB} - F_L$$

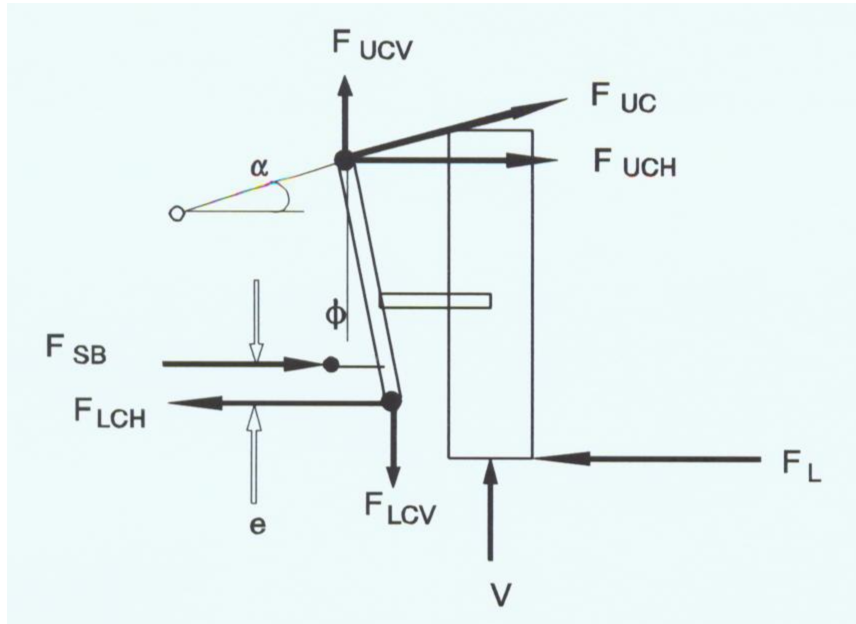
Suspension Load Distribution

SUSPENSION FRONT VIEW



$$\sum M_{LB} = 0 \quad F_{UCV}\, h \,\mathrm{Tan}\,\varphi + F_{UCH}\, h + F_{SB}\, e + F_L\, a - V\,(r_s + c) = 0$$

Suspension Load Distribution

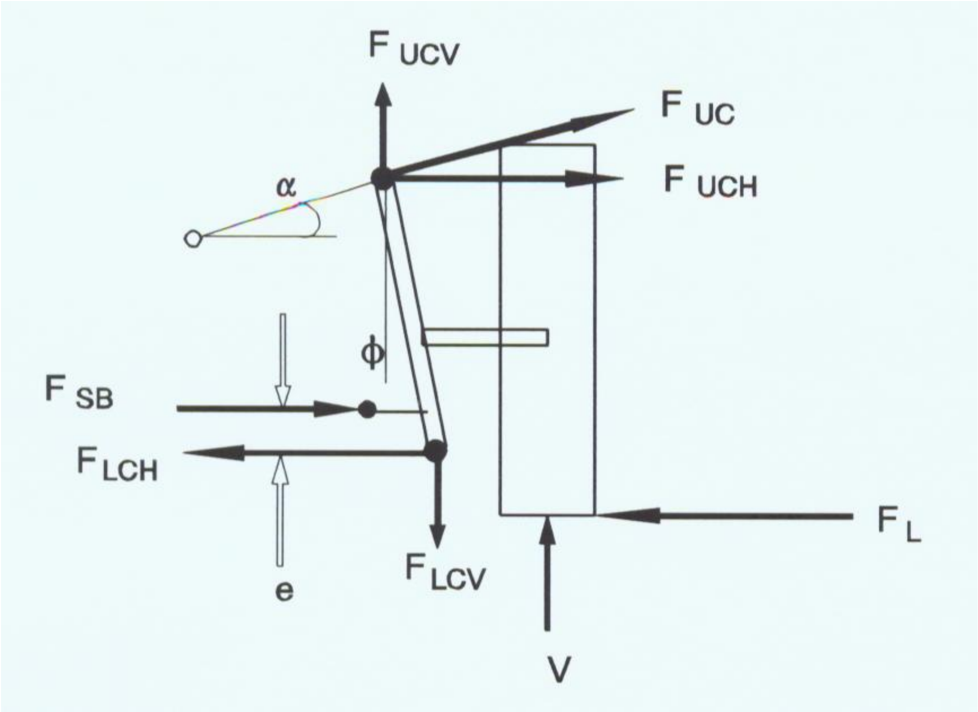


$$F_{UCV} = F_{UC} \sin \alpha \qquad F_{UCH} = F_{UC} \sin \alpha$$

$$F_{UC} = \frac{V(r_s + c) - F_{SB}\, e - F_L\, a}{\sin \alpha \tan \phi + \cos \alpha\, h}$$

$$F_{LCH} = \frac{F_B}{d}\left[(r_s + c) - \frac{a}{h}(b - c)\right] + F_{UC} \cos \alpha - F_L$$

Suspension Load Distribution



$$\sum F_z = 0$$

$$V - F_{LCV} + F_{UCV} = 0$$

$$F_{LCV} = V + F_{UC} \sin\alpha$$

Rear Suspension Design

Beam Type Axle suspension (Solid axle/Dependent)

- The parallel jounce axis and roll axis control the characteristics of this type of suspension
- Anti-features are similar as explained earlier
- The roll axis is found by determining the two lateral restraints and connecting them with a line
- The slope of the roll axis is the roll steer value. If the roll axis tilts down to the front of the vehicle when viewed from the side then the suspension has roll understeer for a rear suspension, if it tilts up to the front, then the suspension has roll oversteer geometry
- Axle roll does occur in solid axle suspension unless the point of force application is at ground level

Lateral restraint Forces