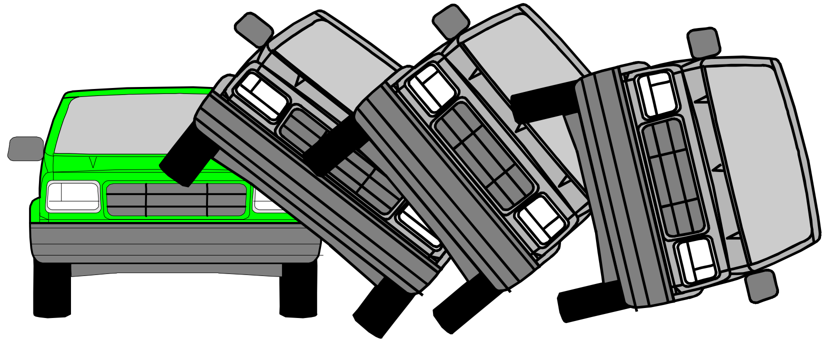
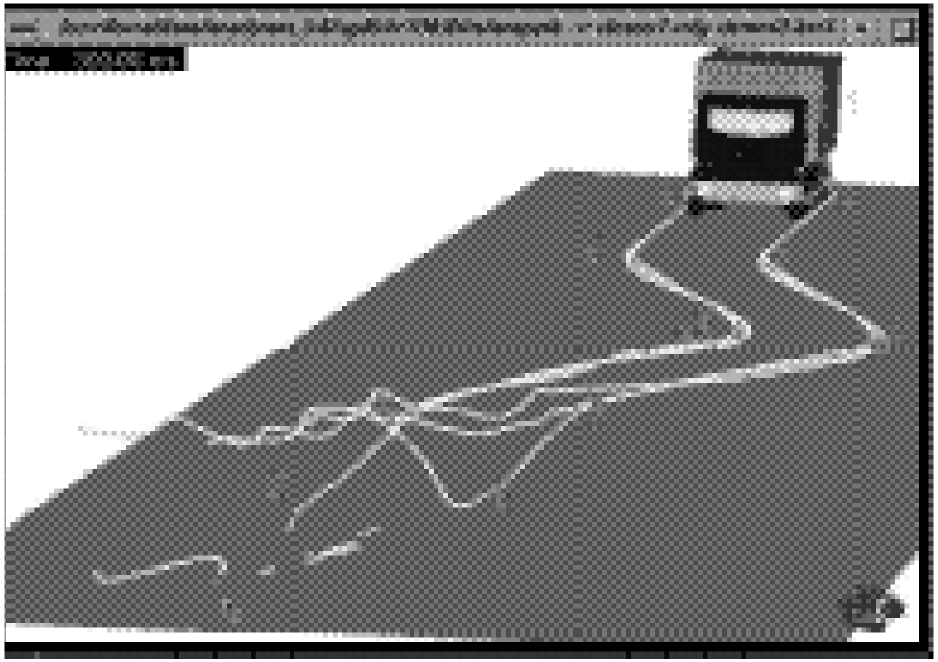
VEHICLE ROLLOVER ANALYSIS



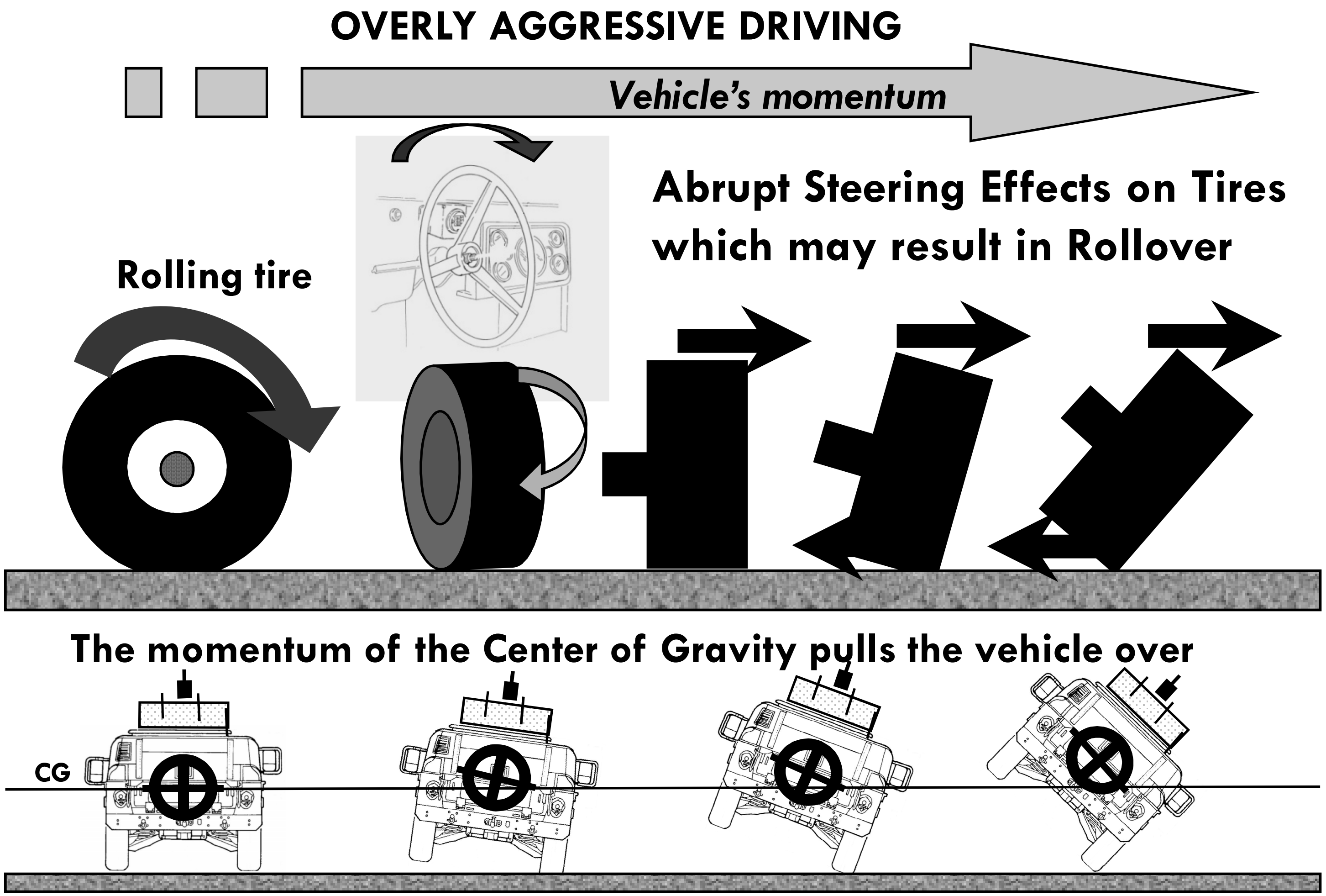
MECHANICAL ENGINEERING DEPARTMENT
PALESTINE POLYTECHNIC UNIVERSITY

Rollover Events

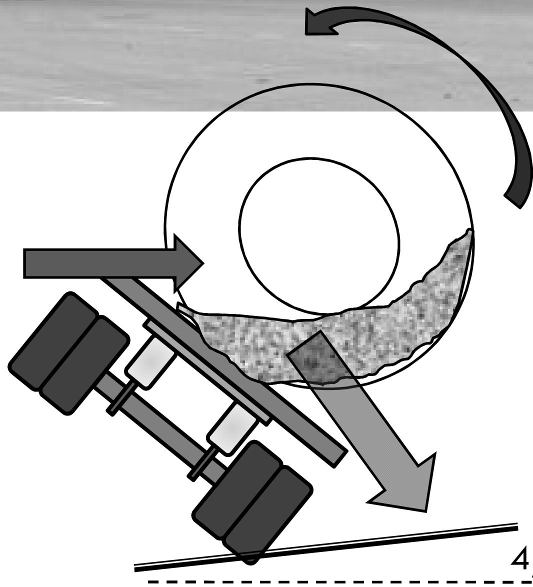
- A rollover is defined as any vehicle rotation of 90° or more about any true longitudinal or lateral axis.



Rollover Dynamics

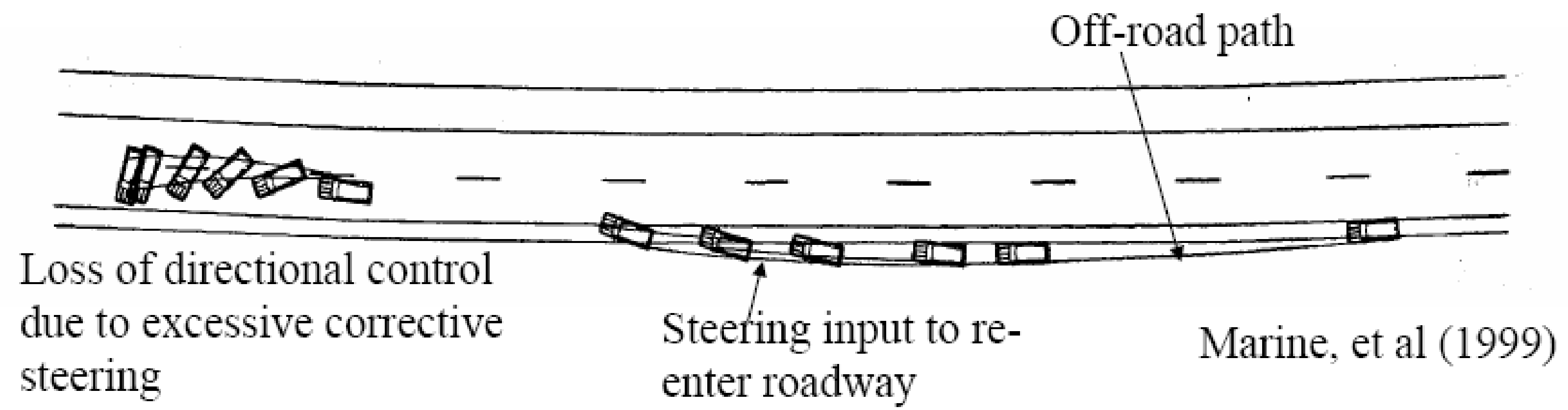


Rollover Dynamic Testing



Rollover Events

- Rollover can occur on flat and level surfaces (on-road). On-road rollovers typically arise from loss of directional control, which may result from driver steering actions.
- Off-road rollover may result from the cross-slope effect adding to lateral forcing from curb impacts, soft ground/soil, or other obstructions that "trip" the vehicle.

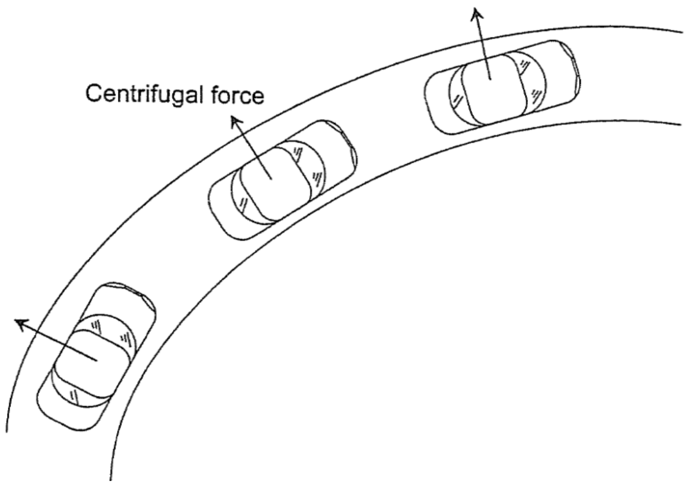


Rollover Occurrence Conditions

- Traveling at high speed on curved road.
- Sever cornering maneuver.
- Traveling on collapsing road and suddenly providing steering input for a vehicle with a low level of roll stability.
- Losing control due to a rapid decrease of friction, such as driving on icy road.
- Laterally sliding of the road.
- Sliding from a cliff.

Factors affect the vehicle to rollover.

- These factors are tire and vehicle characteristics, environmental conditions, and drivers.
- Rollover can happen on a flat road, on a cross-slope road, or off road.
- Rollover can be divided into two categories: tripped rollover, and untripped rollover.
- Tripped rollover is caused by a vehicle hitting an obstacle.
- Traveling at high speed on a curved road: When a vehicle travels on a curved road, lateral centrifugal force will pull it in an outboard motion, as shown in Figure.



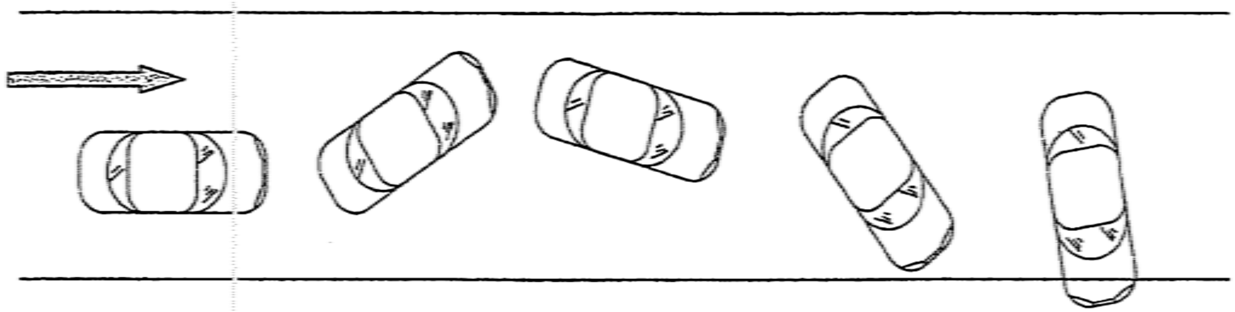
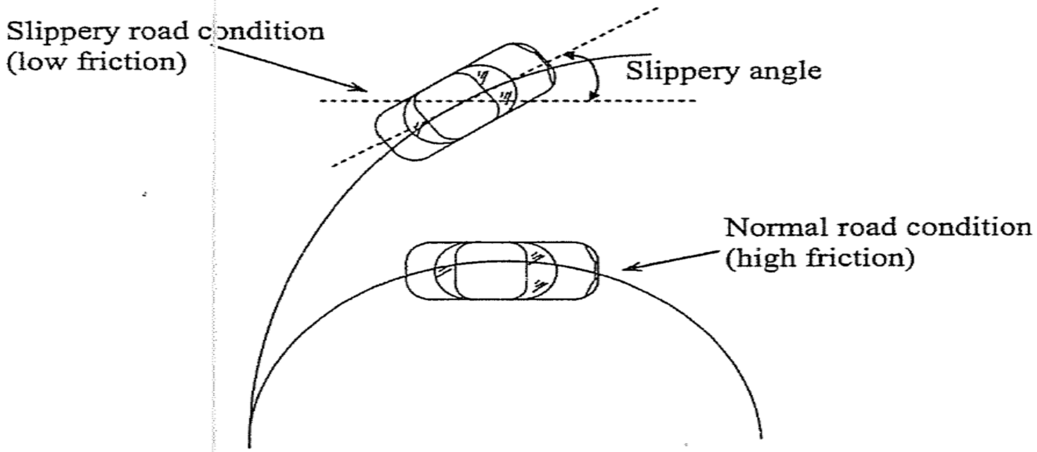


Figure 5.3 A vehicle rollover scenario caused by severe steering input.

- Traveling on a collapsing road and suddenly providing steering input for a vehicle with a low level of roll cause a yaw disturbance. Figure 5.3
- Losing control due to a rapid decrease of friction, such as driving on an icy road:

  Steering can cause yaw motion because forces on the tires in the lateral direction are strong enough to roll the vehicle.



Vehicle and non vehicle factors of roll over

- The study of vehicle rollover is a ***complicated topic***.
- The phenomenon of rollover is ***difficult to predict***.
- Rollover involves many ***vehicle factors*** and some ***non vehicle factors***, such as the road friction coefficient, obstacles on the road, the type of road shoulder, road and shoulder inclination angles, driver steering patterns, and others.
- To simplify the complicated overall analysis***, a steady-state rigid model*** and ***a steady-state suspended model*** are most commonly used.
- Some simple metrics have been developed to evaluate the static and dynamic stability of vehicles, such as the
  - static stability factor (SSF),
  - the tilt table ratio (TTR),
  - the critical sliding velocity,
  - the dynamic stability index,
  - the static stability factor and the tilt table ratio

OVERTURNING AND SKIDDING OF A VEHICLE ON A CURVED HORIZONTAL AND BANKED TRACK

Zammit …chapter 9

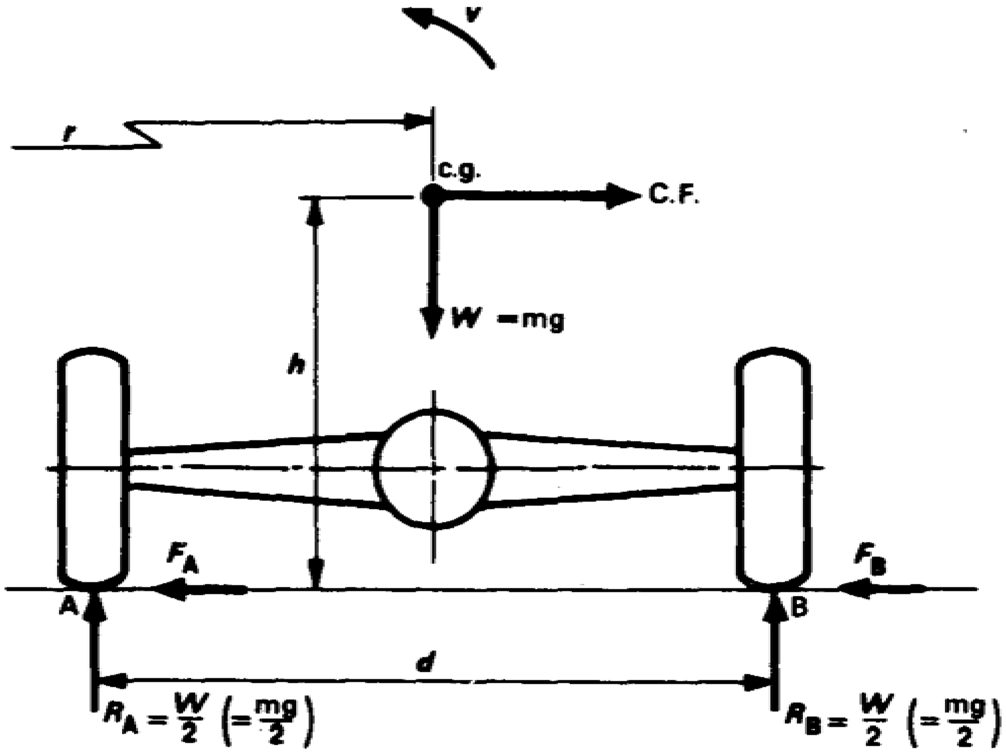
Overturning on a curved horizontal track

limiting speed at which overturning just occurs on a curve.

W = mg, weight.

R_A, R_B, normal reactions.

(C.F.) inertia or centrifugal force



When the vehicle is at rest,

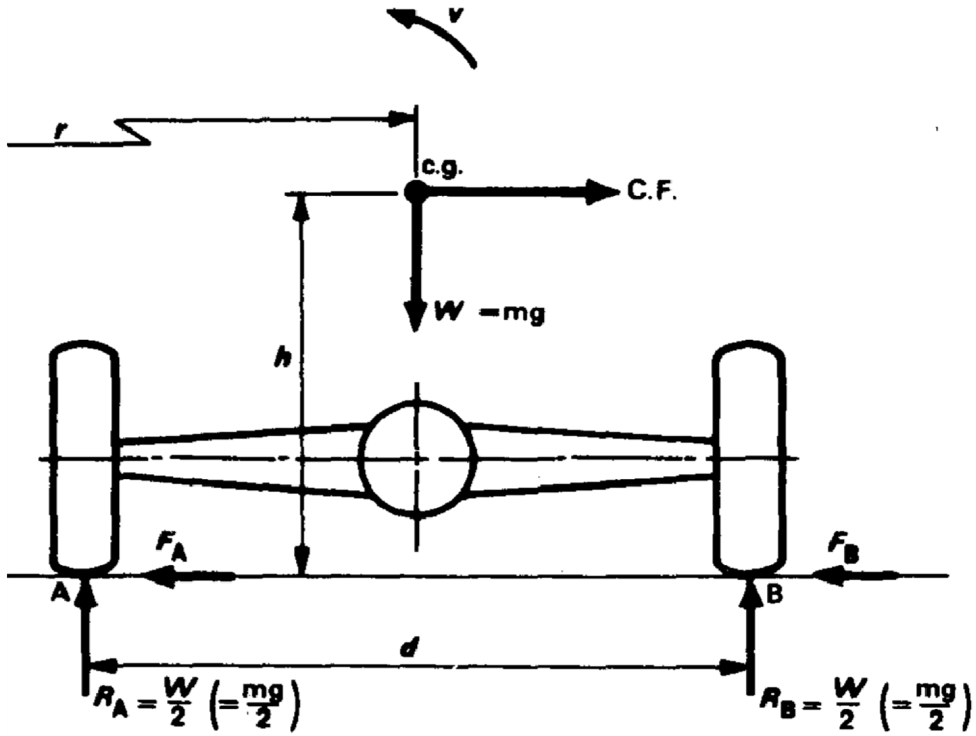
$$\begin{pmatrix}\text{Normal reaction at}\\ \text{inner wheels at A}\end{pmatrix} = \begin{pmatrix}\text{Normal reaction at}\\ \text{outer wheels at B}\end{pmatrix}$$

i.e.,

$$R_A = R_B = \frac{W}{2} = \frac{mg}{2}$$

Overturning on a curved horizontal track

On the point of overturning
$R_A = 0$, and $R_B = W = mg$
Then, taking moments about an axis through the c.g



Overturning moment = Stabilizing moment

$$\text{C.F.} \times h = R_B \times \frac{d}{2}$$

$$\frac{mv^2}{r} \times h = mg \times \frac{d}{2}$$

$$v^2 = \frac{mgd}{2} \times \frac{r}{mh}$$

$$\boxed{\text{Overturning speed, } v = \sqrt{\left(\frac{dgr}{2h}\right)}}$$

Overturning speed may be increased by:-

- **Lowering the height h.**
- **increasing the wheel track d.**
- **increasing the radius r.**

Overturning speed:- speed over which vehicle will overturn

Skidding on a curved horizontal track

F_A = frictional resistance to inner wheels at A

F_B = frictional resistance to outer wheels at B

$$\begin{pmatrix}\text{Inertia or centrifugal} \\ \text{force}\end{pmatrix} = \begin{pmatrix}\text{Frictional resistance} \\ \text{to skidding}\end{pmatrix}$$

,

$$\frac{mv^2}{r} = F_A + F_B$$

$$= \mu R_A + \mu R_B = \mu(R_A + R_B)$$
$$= \mu W = \mu mg$$

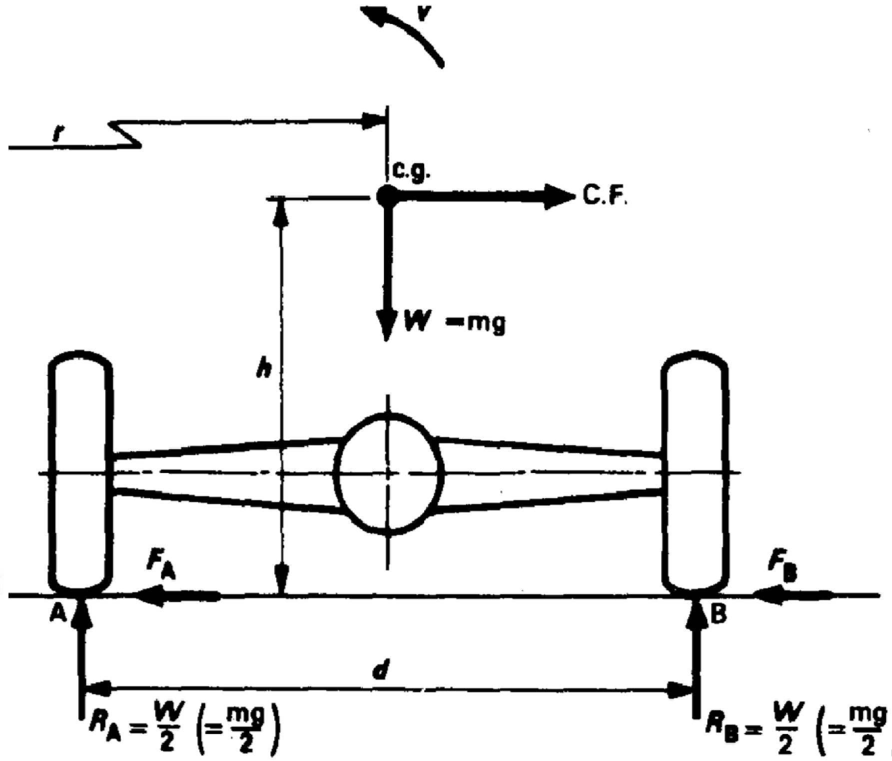
that

$$v^2 = \mu mg \times \frac{r}{m}$$

$$\boxed{\text{Skidding speed, } v = \sqrt{(\mu gr)}}$$

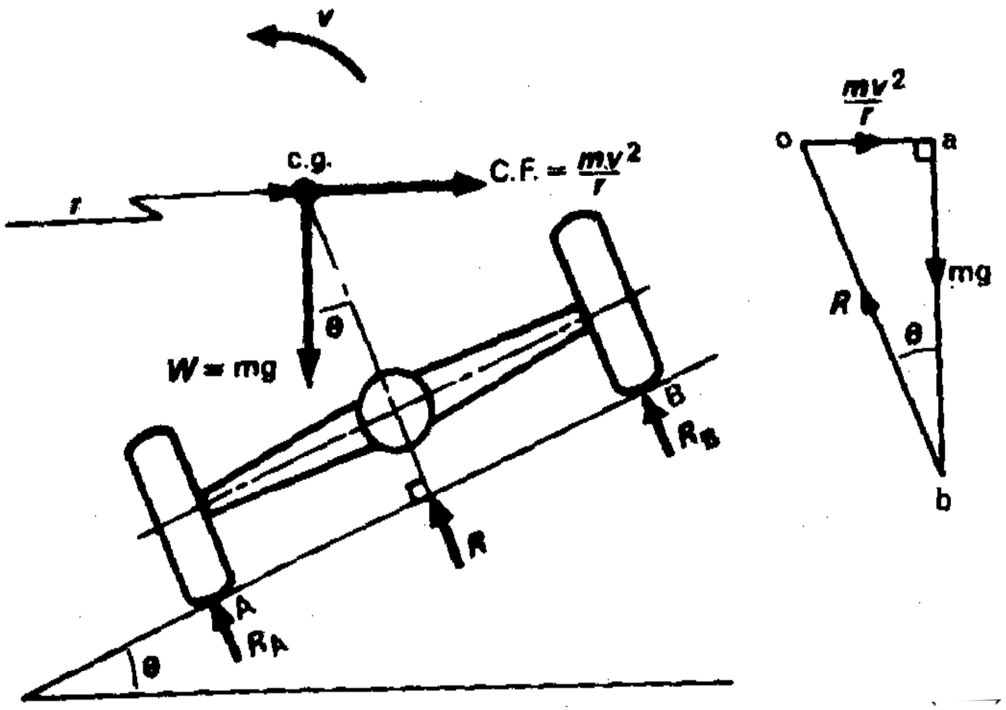
(

If this maximum speed is exceeded, the vehicle will skid.



Effect of banked tracks

Track can be suitably banked to allow a vehicle to traverse safely at a higher speed round a curve



$$\sec\theta = \frac{\text{ob}}{\text{ab}} = \frac{R}{mg}$$

$$mg\sec\theta = R = R_A + R_B$$

$$\tan\theta = \frac{\text{oa}}{\text{ab}}$$

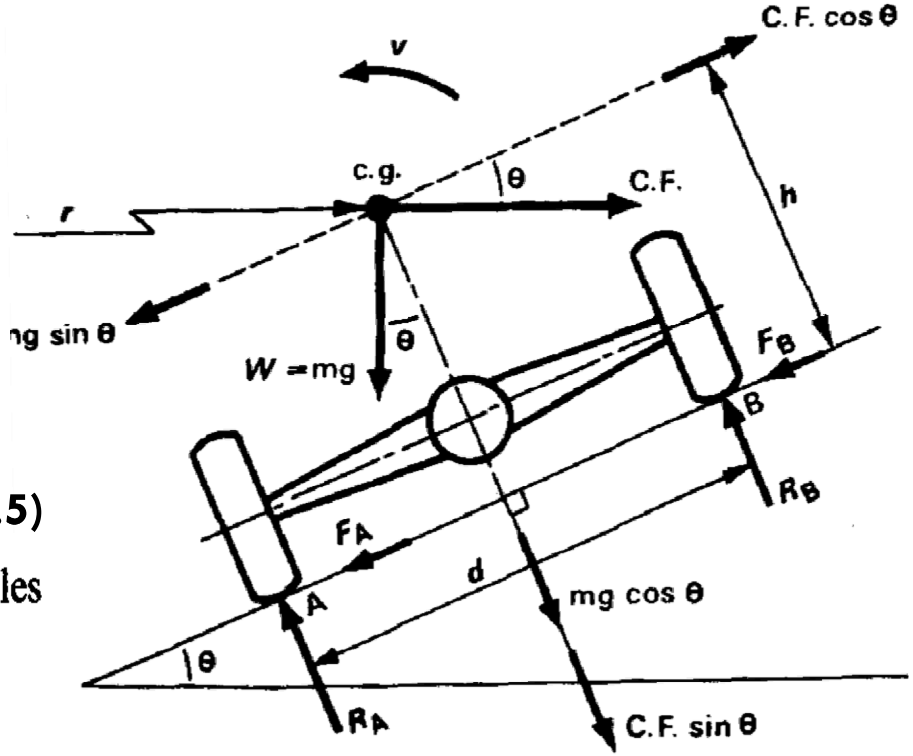
$$= \frac{mv^2}{r} \times \frac{1}{mg}$$

where, v = speed at which vehicle may take the curve (m/s)
r = radius of curve (m)
g = 9·81 m/s^2
θ = angle of banking required to eliminate lateral force on the tyres (deg)

$$\boxed{\tan\theta = \frac{v^2}{gr}}$$

Overturning on a curved banked track

The forces acting on the vehicle when it is turning left on a track banked at an angle θ



Now,

Resolving at right angles to the track, we get:

$$mg\cos\theta + \frac{mv^2}{r}\sin\theta = R_A + R_B \quad (9.5)$$

Taking moments about an axis through the c.g. at right angles to the wheel axles, we get:

$$\left(\frac{mv^2}{r}\cos\theta - mg\sin\theta\right)h + \left(R_A \times \frac{d}{2}\right) = R_B \times \frac{d}{2}$$

or

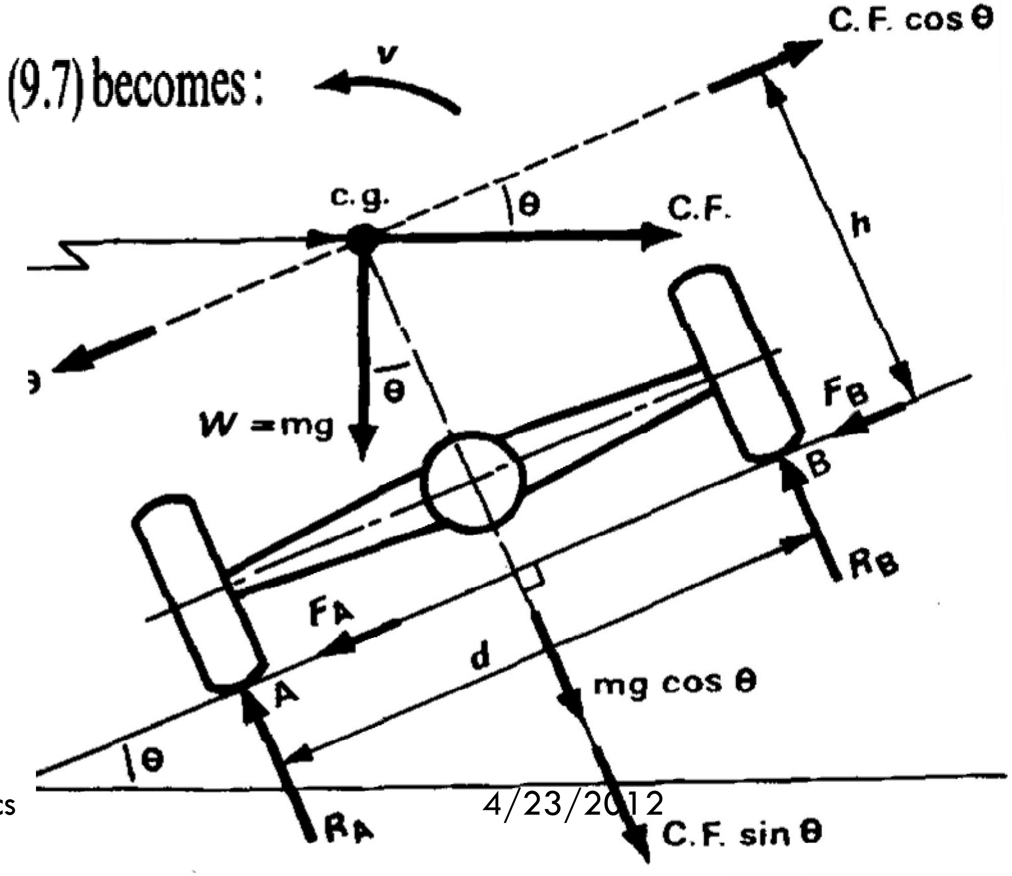
$$\frac{2mh}{d}\left(\frac{v^2}{r}\cos\theta - g\sin\theta\right) = R_B - R_A \quad (9.6)$$

Overturning speed on a curved banked track

Subtracting eq. (9.5) from both sides of eq. (9.6), we get :

$$\frac{2mh}{d}\left(\frac{v^2}{r}\cos\theta - g\sin\theta\right) - m\left(g\cos\theta + \frac{v^2}{r}\sin\theta\right) = -2R_A$$

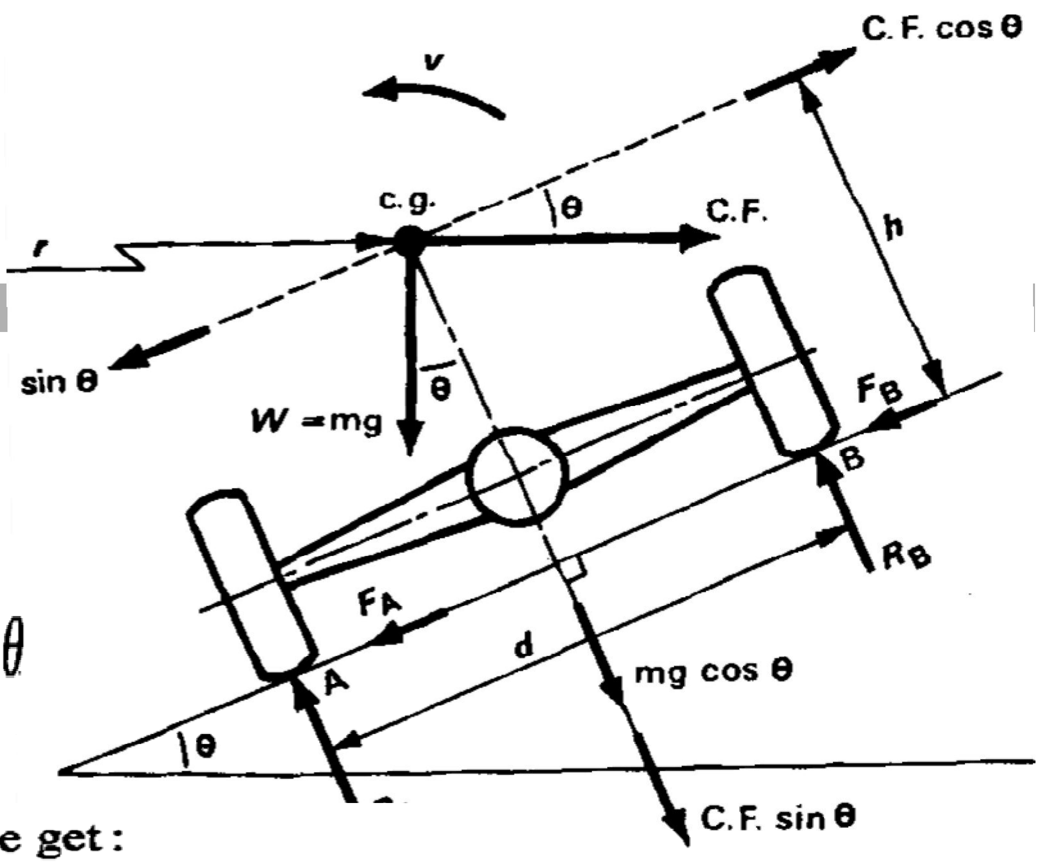
But on the point of overturning, $R_A = 0$, so that eq. (9.7) becomes :



$$\frac{2h}{d}\left(\frac{v^2}{r}\cos\theta - g\sin\theta\right) = g\cos\theta + \frac{v^2}{r}\sin\theta$$

or

$$\frac{2v^2h}{dr}\cos\theta - \frac{v^2}{r}\sin\theta = \frac{2gh}{d}\sin\theta + g\cos\theta$$



$$\frac{2v^2h}{dr}\cos\theta - \frac{v^2}{r}\sin\theta = \frac{2gh}{d}\sin\theta + g\cos\theta$$

Multiplying throughout by $\frac{d}{2\cos\theta}$, **we get:**

$$\frac{v^2}{r}\left(h - \frac{d}{2}\tan\theta\right) = g\left(h\tan\theta + \frac{d}{2}\right)$$

so that $$v^2 = gr\left(\frac{h\tan\theta + \frac{d}{2}}{h - \frac{d}{2}\tan\theta}\right)$$

<u>Overturning speed on a curved banked track</u>

$$\therefore \quad \boxed{v = \sqrt{\left\{gr\left(\frac{h\tan\theta + \frac{d}{2}}{h - \frac{d}{2}\tan\theta}\right)\right\}}}$$

Skidding speed on a curved banked track

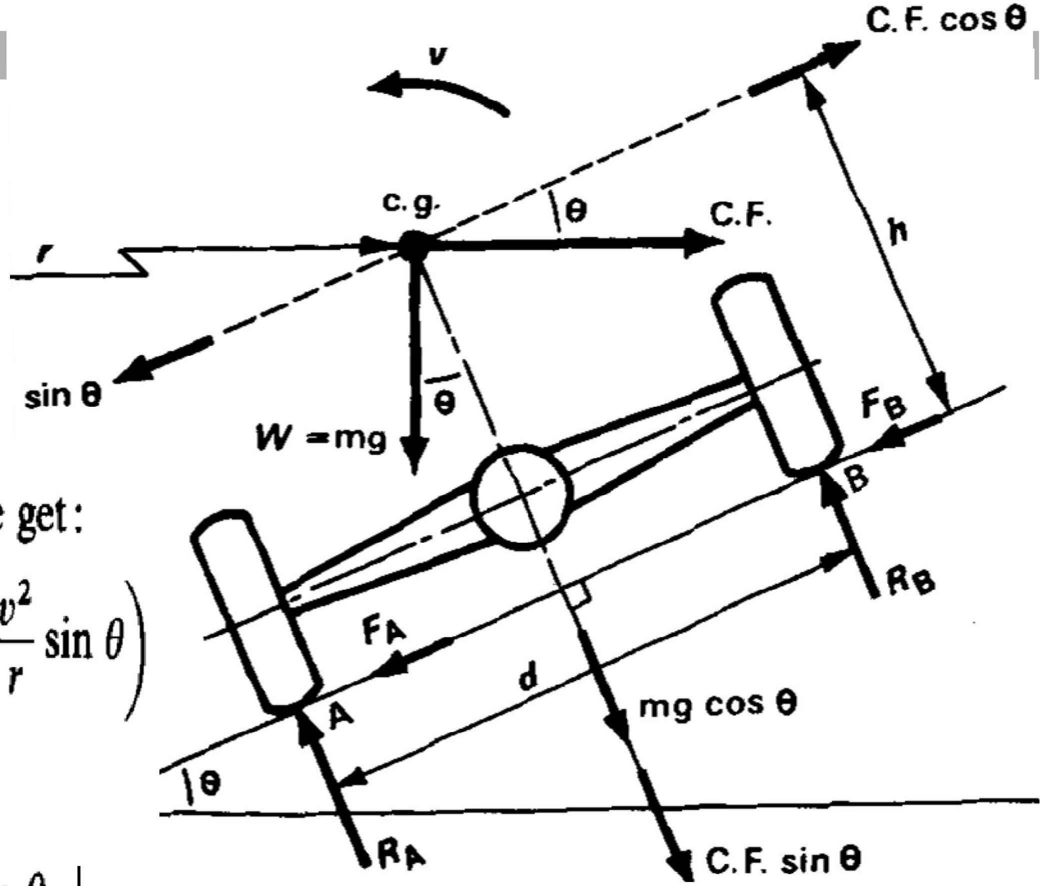
let be the limiting coefficient of friction between tires and road.

Resolving along the track, we get :

$$\frac{mv^2}{r}\cos\theta - mg\sin\theta = F_A + F_B$$

$$= \mu(R_A + R_B) \quad (9.9)$$

$$mg\cos\theta + \frac{mv^2}{r}\sin\theta = R_A + R_B \quad (9.5)$$



substituting $R_A + R_B$ from eq. (9.5) in eq. (9.9), we get :

$$\cancel{m}\left(\frac{v^2}{r}\cos\theta - g\sin\theta\right) = \mu\cancel{m}\left(g\cos\theta + \frac{v^2}{r}\sin\theta\right)$$

or

$$\frac{v^2}{r}\cos\theta - \frac{v^2}{r}\mu\sin\theta = \mu g\cos\theta + g\sin\theta$$

Dividing throughout by $\cos\theta$, we get :

$$\frac{v^2}{r}(1 - \mu\tan\theta) = g(\mu + \tan\theta)$$

$$v^2 = gr\left(\frac{\mu + \tan\theta}{1 - \mu\tan\theta}\right)$$

$$\boxed{\text{Skidding speed, } v = \sqrt{\left\{gr\left(\frac{\mu + \tan\theta}{1 - \mu\tan\theta}\right)\right\}}}$$

Overturning and skidding of a vehicle on a curved banked track

$$\text{Skidding speed, } v = \sqrt{\left\{gr\left(\frac{\mu + \tan\theta}{1 - \mu\tan\theta}\right)\right\}}$$

$$\text{Overturning speed, } v = \sqrt{\left\{gr\left(\frac{h\tan\theta + \frac{a}{2}}{h - \frac{d}{2}\tan\theta}\right)\right\}}$$

Overturning and skidding of a vehicle on a curved horizontal track

$$\text{Skidding speed, } v = \sqrt{(\mu gr)}$$

$$\text{Overturning speed, } v = \sqrt{\left(\frac{dgr}{2h}\right)}$$

EXAMPLE 9.2

A car is coasting round a bend of 80m radius at 50.4 km/h. If the road is horizontal find the minimum value of the coefficient of friction to prevent skidding and prove that the car will not overturn.

The height of the centre of gravity of the car is 0.61 m and the track width of the wheels is 1.22 m.

$$\text{Skidding speed, } v = \frac{50{\cdot}4 \times 5}{18} = 14 \text{ m/s}$$

Radius of bend, $r = 80$ m

Height of c.g., $h = 0{\cdot}61$ m

Wheel track, $d = 1{\cdot}22$ m

When the car is on the point of skidding, the inertia or centrifugal force is balanced by the total resistance to skidding

$$\frac{mv^2}{r} = \mu mg$$

so that

$$\mu = \frac{v^2}{gr}$$

$$\therefore \quad \mu = \frac{14 \times 14}{9{\cdot}81 \times 80}\left[\frac{\text{m}^2}{\text{s}^2} \times \frac{\text{s}^2}{\text{m}} \times \frac{1}{\text{m}}\right]$$

$$= 0{\cdot}25$$

We will now proceed to show that the car will not overturn at 50·4 km/h.

From eq. (9.1)

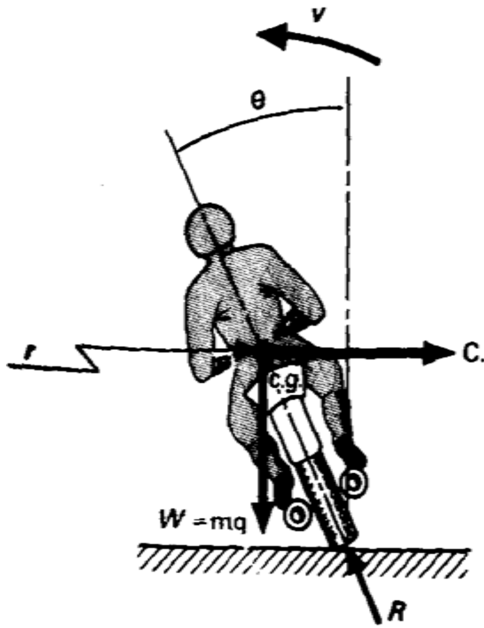
$$\text{Overturning speed} = \sqrt{\left(\frac{dgr}{2h}\right)}$$

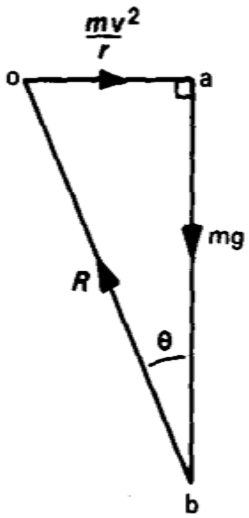
$$= \sqrt{\left(\frac{1{\cdot}22 \times 9{\cdot}81 \times 80}{2 \times 0{\cdot}61}\,\frac{m \times m \times m}{\text{m} \times \text{s}^2}\right)}$$

$$= 28 \text{ m/s}$$

EXAMPLE 9.4

A motor cycle travels round a bend of 50 m radius at 72 km/h on a flat road. The total mass of the motor cycle and rider is 180 kg. Find the angle which the motor cycle makes with the vertical and the resultant force exerted on the road.





Force diagram

From the force diagram,

$$\tan\theta = \frac{\text{oa}}{\text{ab}}$$

$$= \frac{mv^2}{r} \times \frac{1}{mg}$$

$$= \frac{v^2}{gr} \qquad \text{[same eq. (9.4)]}$$

$$\therefore \qquad \tan\theta = \frac{20^2}{9{\cdot}81 \times 50}\left[\frac{\text{m}^2}{\text{s}^2} \times \frac{\text{s}^2}{\text{m}} \times \frac{1}{\text{m}}\right]$$

$$= 0{\cdot}8155$$

From tables, $\theta = 39°12'$

Also, from the same force diagram,

$$\sec\theta = \frac{\text{ob}}{\text{ab}} = \frac{R}{mg}$$

i.e., $R = mg\sec\theta$ [same eq. (9.3)]

$$\therefore \qquad R = 180 \times 9{\cdot}81 \times \sec 39°12'$$

$$= 180 \times 9{\cdot}81 \times 1{\cdot}2904\,[\text{kg} \times \text{m/s}^2]$$

$$= 2278{\cdot}6\ \text{N} = 2{\cdot}2786\ \text{kN}$$

Angle of motor cycle with the vertical = 39° 12′
Resultant force exerted on the road = 2·279 kN

EXAMPLE 9.5

The wheel track width of a vehicle is 1.5 m and the centre of gravity is situated at a height of 0.9 m above road level.

(a) If the coefficient of friction between tyres and road is 0.6, calculate the maximum speed, in km/h, at which the vehicle can traverse round a curve of 70 m radius which is banked at an angle of 20° to the horizontal without skidding.

(b) Assuming that friction is sufficient to prevent skidding, calculate the maximum speed, in km/h, at which the vehicle can traverse round the curved banked road without overturning.

(*a*) From eq. (9·10),

$$\text{Skidding speed} = \sqrt{\left\{gr\left(\frac{\mu + \tan\theta}{1 - \mu\tan\theta}\right)\right\}}$$

E

$$= \sqrt{\left\{9{\cdot}81 \times 70\left[\frac{0{\cdot}6 + 0{\cdot}364}{1 - (0{\cdot}6 \times 0{\cdot}364)}\right]\right\}}$$

$$= \sqrt{\left(\frac{9{\cdot}81 \times 70 \times 0{\cdot}964}{0{\cdot}7816}\right)}$$

$$= \sqrt{847} = 29{\cdot}1 \text{ m/s}$$

$$29{\cdot}1 \text{ m/s} = \frac{29{\cdot}1 \times 18}{5} = 29{\cdot}1 \times 3{\cdot}6 \text{ km/h}$$

$$= 104{\cdot}8 \text{ km/h}$$

(*b*) From eq. (9·8),

$$\text{Overturning speed} = \sqrt{\left\{gr\left(\frac{h\tan\theta + \frac{d}{2}}{h - \frac{d}{2}\tan\theta}\right)\right\}}$$

$$= \sqrt{\left\{9{\cdot}81 \times 70\left[\frac{(0{\cdot}9 \times 0{\cdot}364) + 0{\cdot}75}{0{\cdot}9 - (0{\cdot}75 \times 0{\cdot}364)}\right]\right\}}$$

$$= \sqrt{\left(\frac{9{\cdot}81 \times 70 \times 1{\cdot}078}{0{\cdot}627}\right)}$$

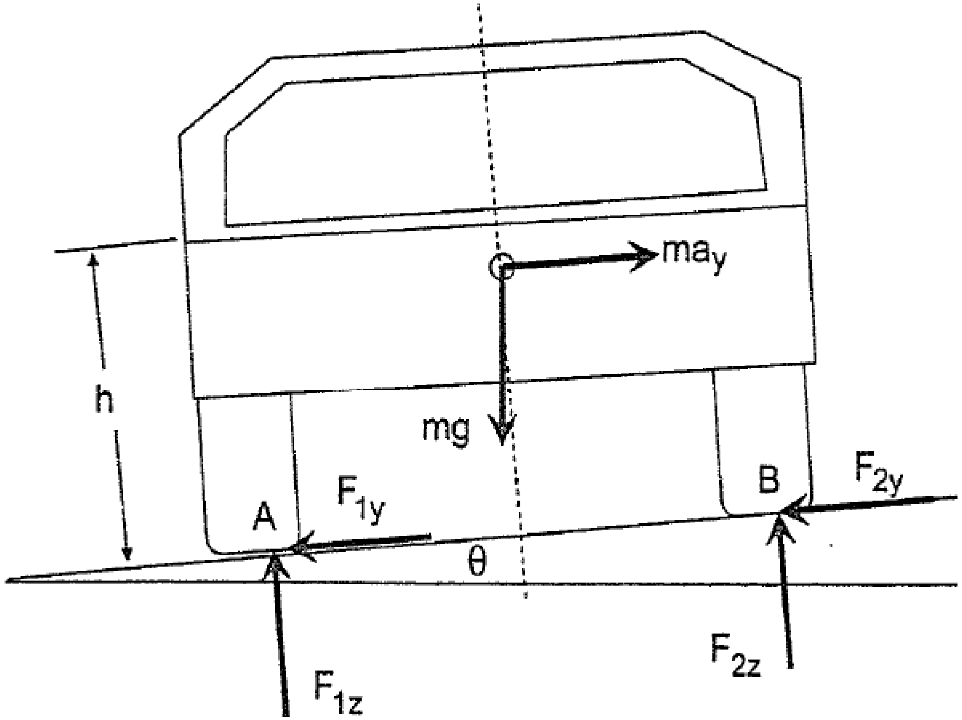
$$= \sqrt{1181} = 34{\cdot}36 \text{ m/s}$$

$$34{\cdot}36 \text{ m/s} = 34{\cdot}36 \times 3{\cdot}6 = 123{\cdot}7 \text{ km/h}$$

(*a*) Maximum speed without skidding = 104·8 km/h
(*b*) Maximum speed without overturning = 123·7 km/h

Rigid Vehicle Model

- The deflection of the tire and suspension is neglected.



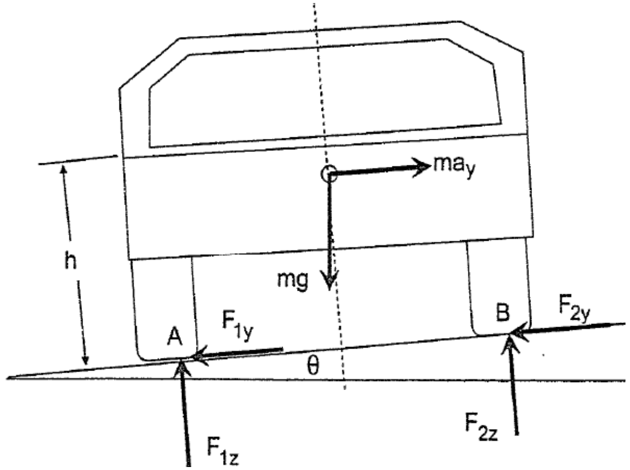
When the vehicle undergoes a turn, centrifugal force caused by its body pulls it outward from the turning center. <u>The centrifugal force equals the lateral acceleration of the vehicle multiplied by the weight of the vehicle, or ma_y</u>

Rigid Vehicle Model – Quasi Static-State

Taking the moment at point B, the vehicle dynamic equation of Figure 5.6 can be written as

I_b is the vehicle moment of inertia around point B,

$$I_b\ddot{\varphi} = F_{1z}d - (mg\sin\theta)h - (mg\cos\theta)\frac{d}{2} + ma_y h$$



Usually, the slope angle θ is very small; therefore, $\sin\theta \approx \theta$ and $\cos\theta \approx 1$. can be simplified as

$$I_b\ddot{\varphi} = F_{1z}d - mgh\theta - mg\frac{d}{2} + ma_y h$$

At the beginning moment when rollover occurs, the rotation velocity and acceleration are neglected (i.e., $\ddot{\varphi} = 0$). This situation is called the quasi-static state (steady state). The discussions in this section and the next section are based on an assumption of the quasi-static state. Equation 5.2 can be simplified as

$$F_{1z}d - mgh\theta - mg\frac{d}{2} + ma_y h = 0 \qquad (5.3)$$

Static Stability Factor (SSF) $\frac{a_y}{g} = \frac{d}{2h}$

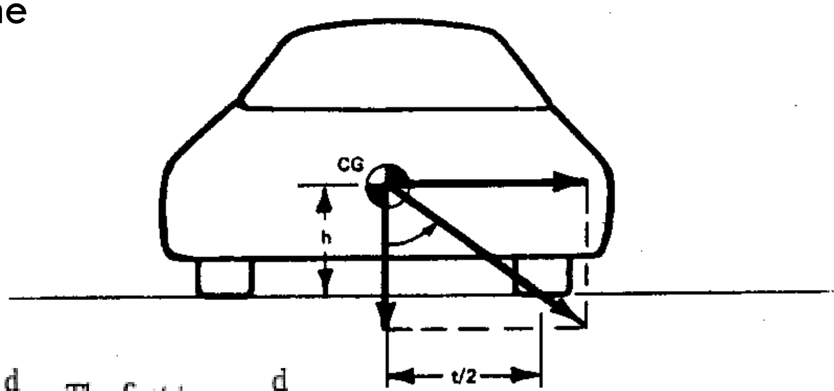
Rollover begins if the load on the inner tires is zero (i.e., $F_{1z} = 0$). Equation 5.3 can be simplified as

$$F_{1z}d - mgh\theta - mg\frac{a}{2} + ma_y h = 0 \qquad \frac{a_y}{g} = \frac{d}{2h} + \theta \tag{5.4}$$

The larger the slope angle the higher the rollover lateral acceleration.

On a flat road surface, $\theta = 0$.

$$a_y > \frac{d}{2h} g\text{, rollover will occur}$$



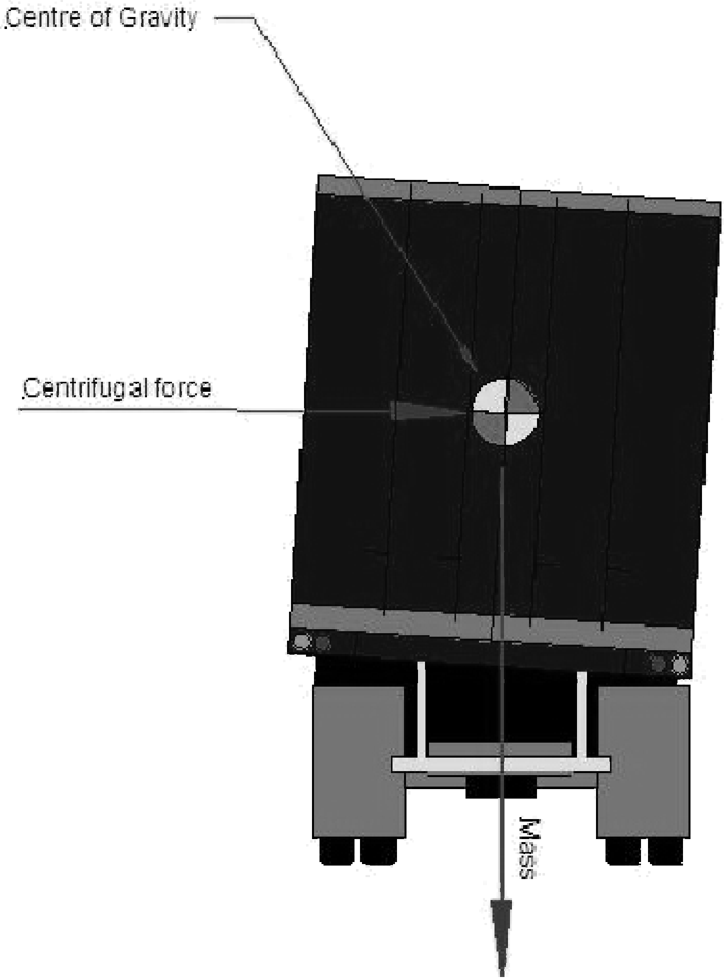
acceleration. Usually, the slope angle θ is much smaller than $\frac{d}{(2h)}$. The first term, $\frac{d}{(2h)}$, often is called the first-order estimate of the static rollover threshold.

- First order estimate of steady state lateral acceleration when rollover begins (d/2h).

Centre of Gravity

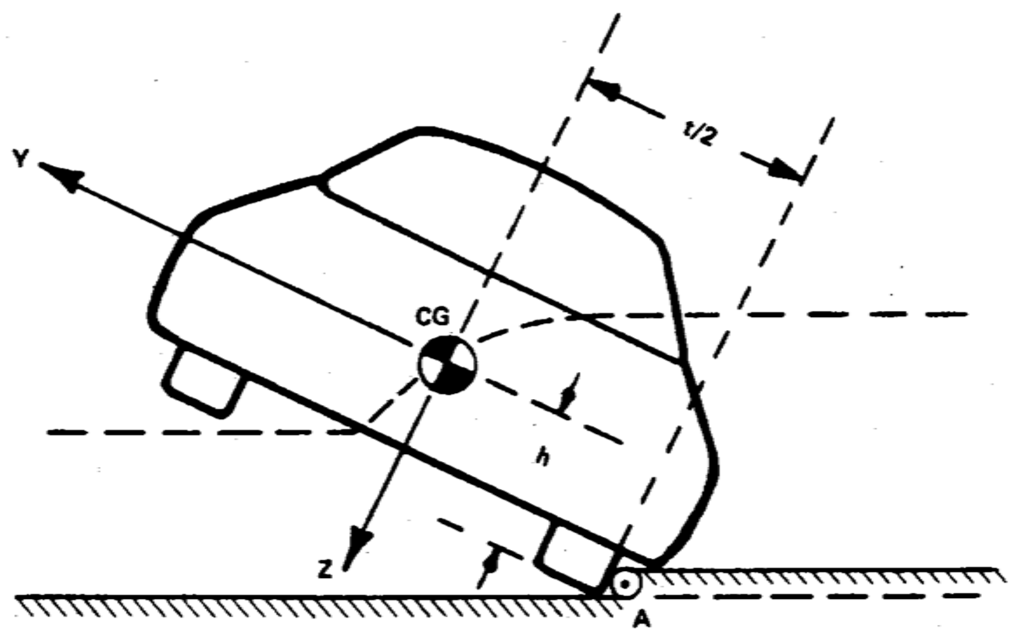
- Increasing load highest.

$$a_y > \frac{d}{2h} g$$



Critical Sliding Velocity (CSV)

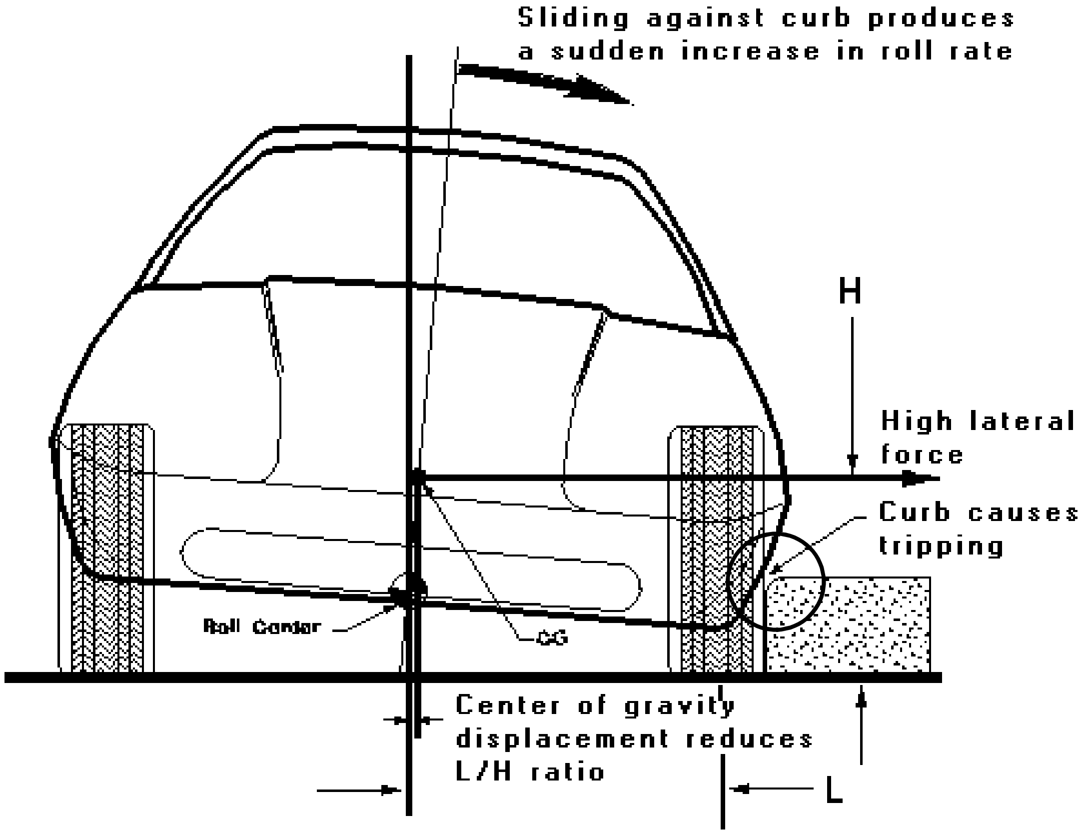
- Theoretical lowest speed at which sliding sideways into a curb causes rollover.



Critical Sliding Velocity (CSV)

$$a_y > \frac{d}{2h} g$$

FIGURE 4: ROLLOVER CAUSED BY TRIPPING

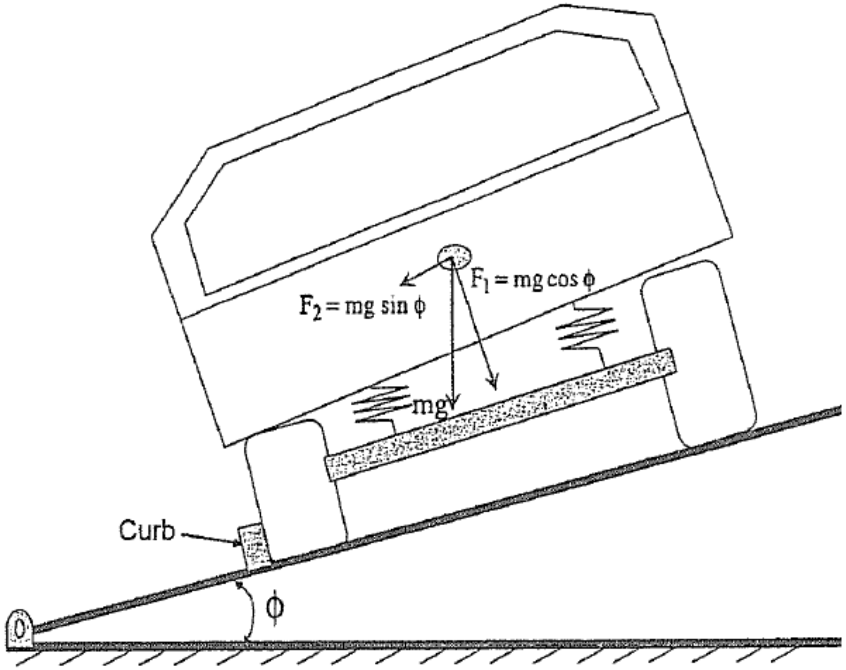


Tilt Table Ratio (TTR)

- Minimum table angle at which a vehicle on the table will tip over.

$$F_1 = m_s g = mg\cos\phi$$

$$F_2 = m_s a_s = mg\sin\phi$$



$\mathbf{m_s}$:-simulated weight
$\mathbf{a_s}$:-simulated lateral acceleration

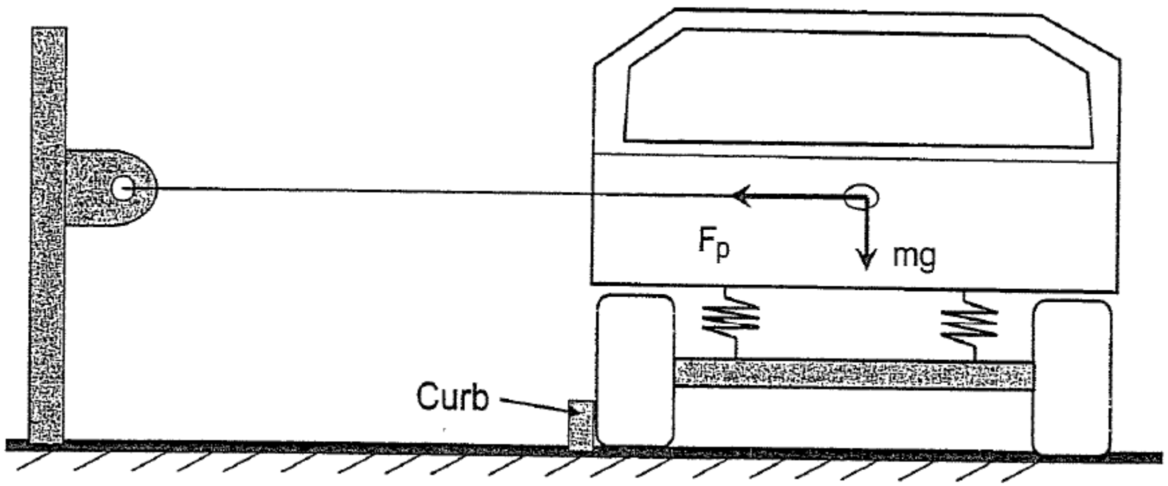
$$TTR = \frac{a_s}{g} = \tan\phi$$

During the test, the tilt of the table is increased slowly until the tire at the high side loses contact with the surface of the table. The angle at which this occurs is called the tip-over angle. Dividing above Eq., the tilt table ratio (TTR), Its used to estimate the rollover of unstable vehicles, where there is a small tilt angle, than stable vehicles.

Side Pull Ratio (SPR)

- The minimum side pull force required to left all tires on one side of the vehicle from the ground to the vehicle weight.
- A vehicle is placed on a high-friction surface. Then a pull force through a belt is applied to the sprung mass at the vehicle center of gravity and slowly pulls laterally.

$$SPR = \frac{F_p}{mg}$$



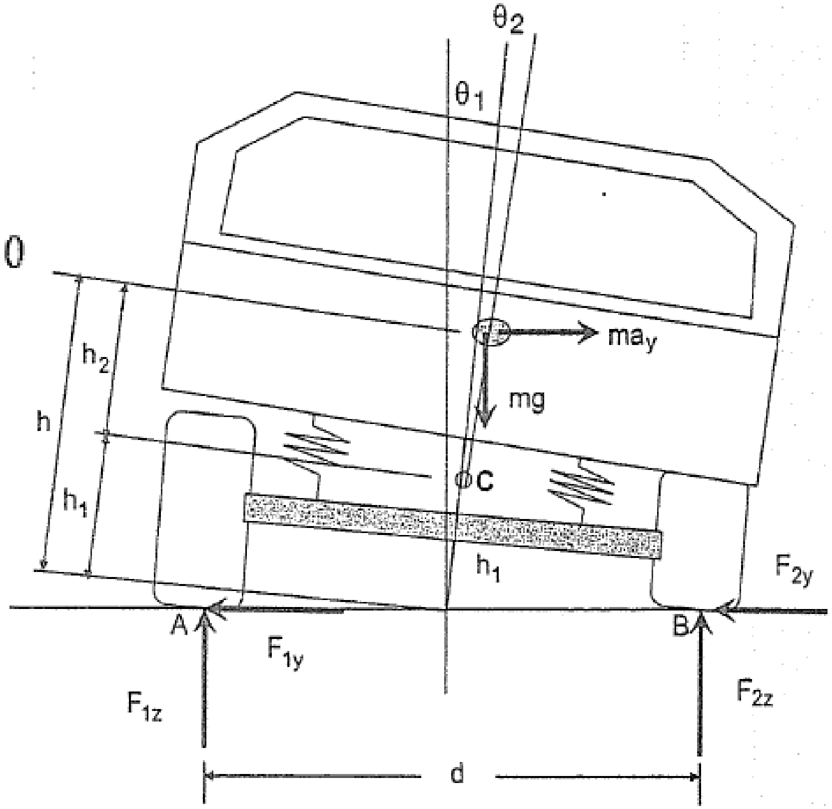
Suspended Rollover Model

angle θ_1 due to the deflection of the tires and another angle θ_2 due to the suspension deflection. Thus, the total angle at which the rollover line rotates is $\theta_1 + \theta_2$, as shown

$$F_{1z}d + ma_y\left[h_1\cos\theta_1 + (h - h_1)\cos(\theta_1 + \theta_2)\right]$$

$$-\, mg\left[\frac{d}{2} - h_1\sin\theta_1 - (h - h_1)\sin(\theta_1 + \theta_2)\right] = 0$$

$$\frac{a_y}{g} = \frac{d}{2h} - \theta_1 - \frac{h_2}{h}\theta_2$$



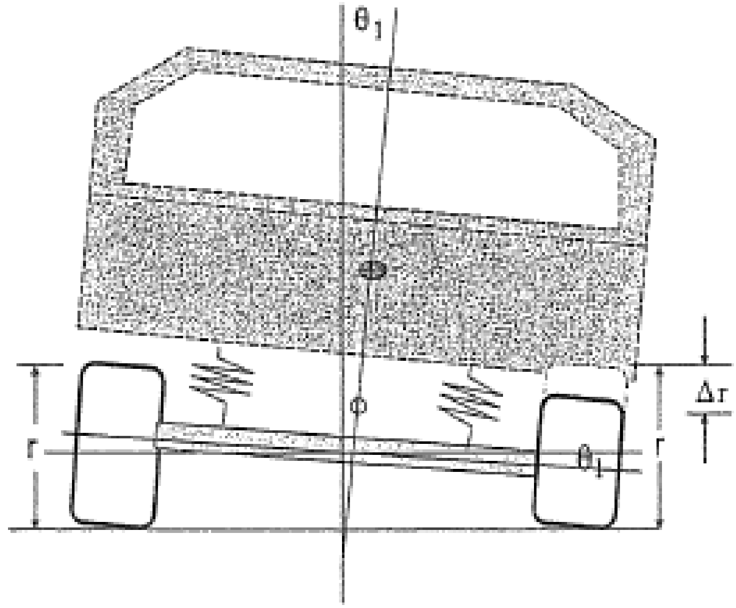
Tire deflection

The tire deflection, Δr, compared with the stationary condition, is

$$\Delta F_{2z} = \frac{mg}{2} \qquad \Delta r = \frac{\frac{mg}{2}}{K_{tire}}$$

where K_{tire} is the tire stiffness.

$$\theta_1^R = \frac{\Delta r}{\frac{d}{2}} = \frac{mg}{dK_{tire}}$$



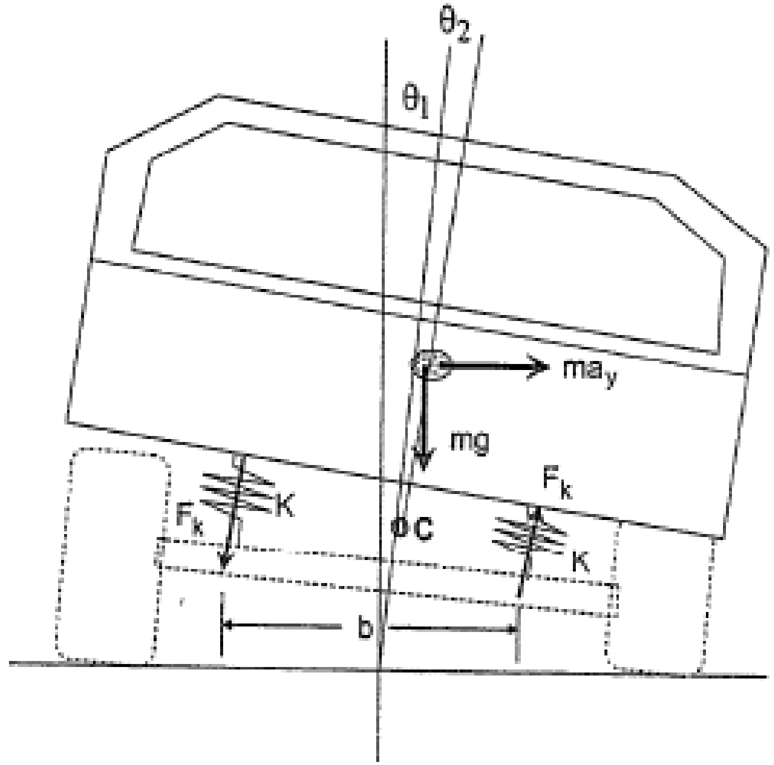
Suspension deflection

$$ma_y h_2 \cos(\theta_1 + \theta_2) + mgh_2 \sin(\theta_1 + \theta_2) - F_K b = 0$$

$$\theta_2 = \frac{a_y}{g} \frac{mgh_2}{\frac{b^2}{2}K - mgh_2} + \frac{mgh_2}{\frac{b^2}{2}K - mgh_2}\theta_1$$

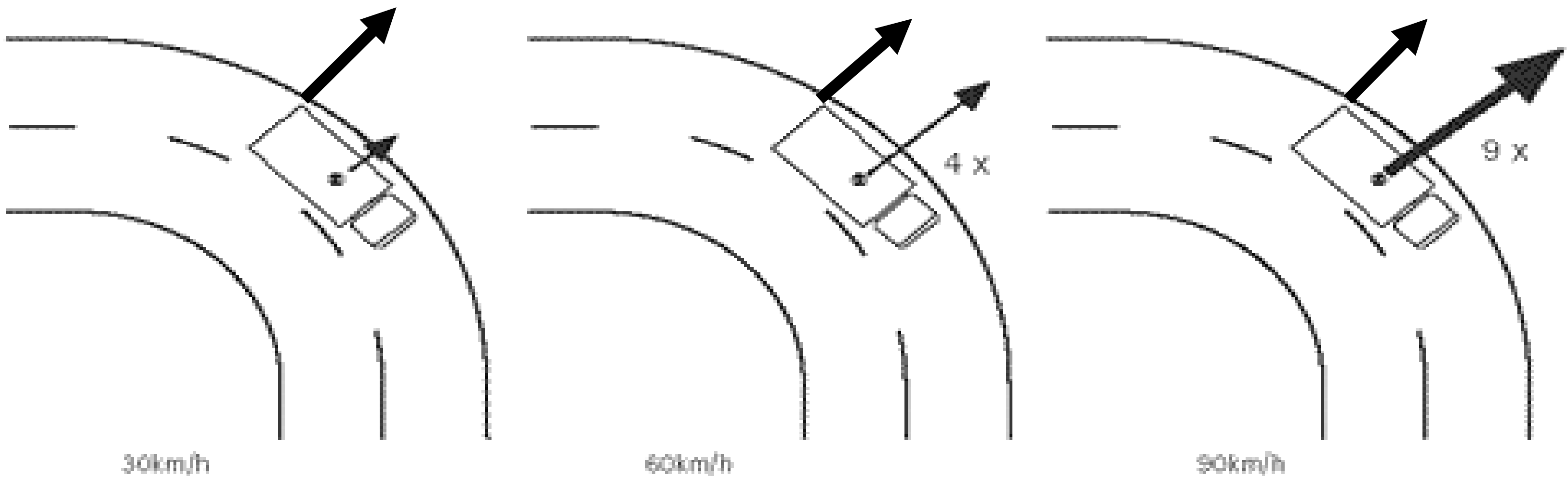
Hence, $\theta_2 \geq 0$, so we obtain the relation

$$b \geq \sqrt{\frac{2mgh_2}{K}}$$



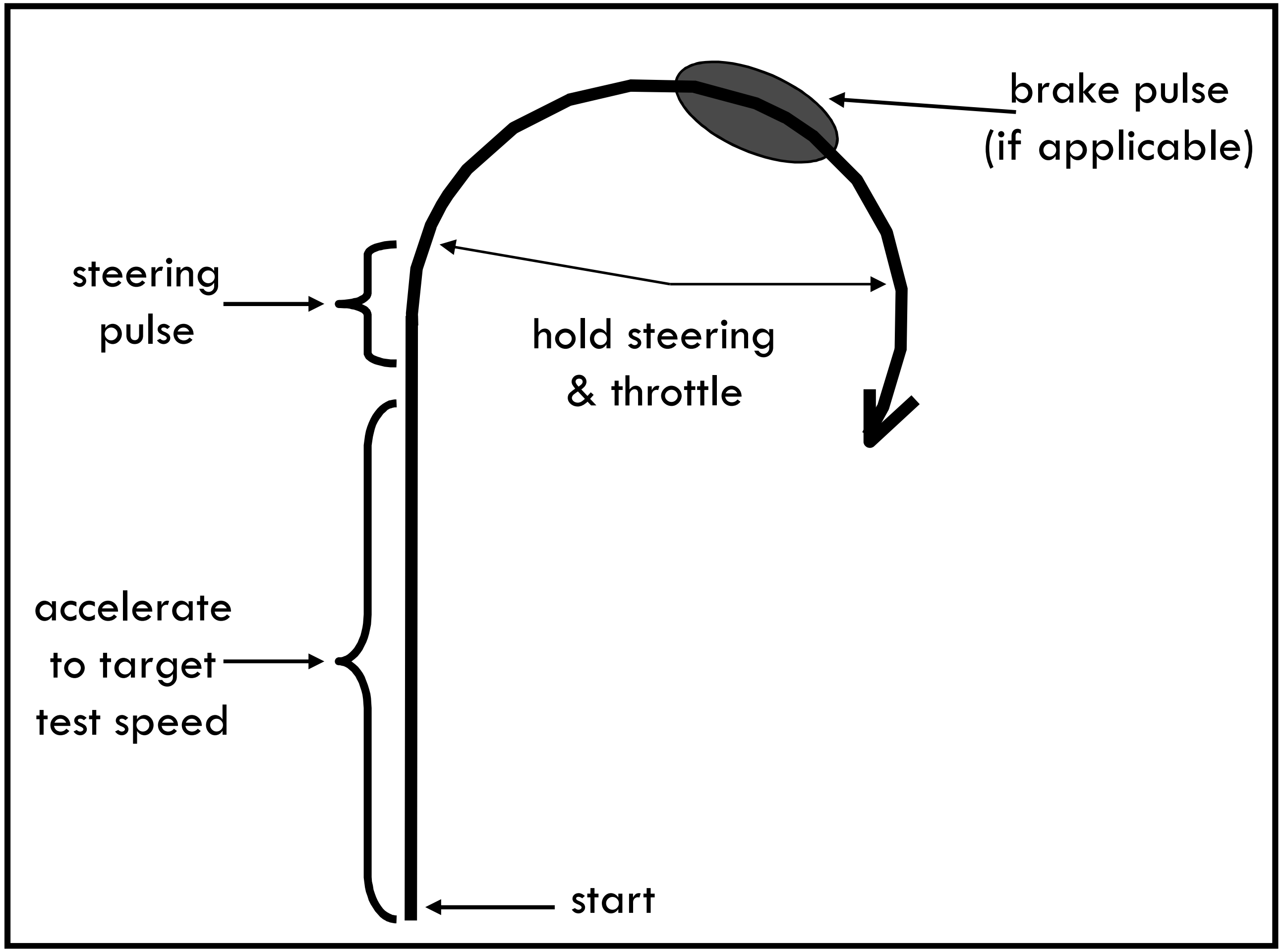
Speed and Centrifugal force

- If a vehicle is forced to take evasive action, these forces are further multiplied.

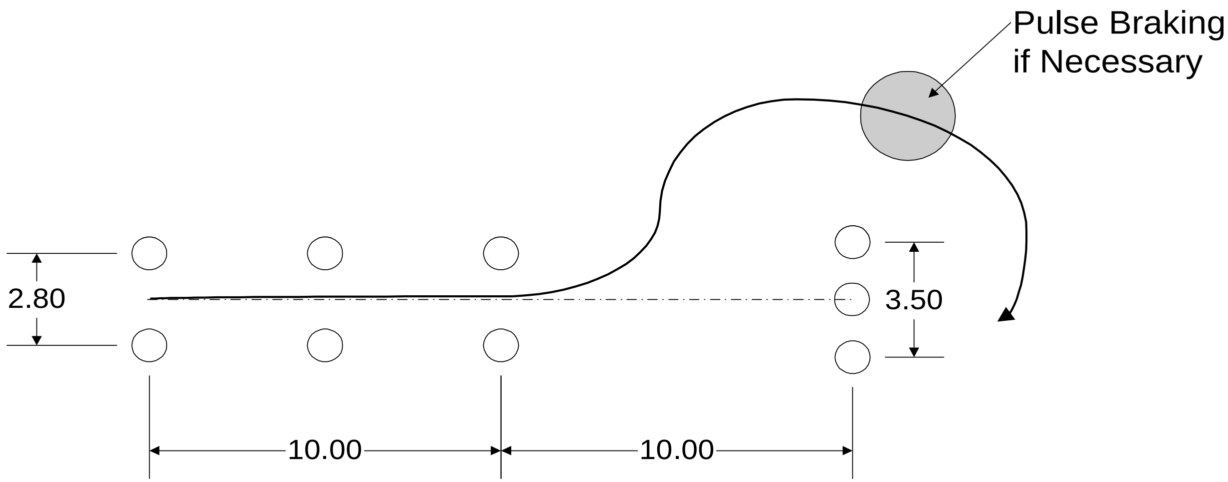


- If you double your speed, the overturning force will be four times higher. As the speed increases the trailer tracks wider and forces increase on rear axle. This means that a slight increase in speed can be critical.
- It increases by squaring so that 10% increase in speed causes 100% increase in force! Double the speed is four times the force!

J-Turn Maneuver



Toyota Fishhook Maneuver



1 cm = 2 m

Approximately 270 degree initial steer and then turning to steering lock in opposite direction

Vehicle driven at incrementally higher speed until vehicle tip up occurs - throttle off while in course

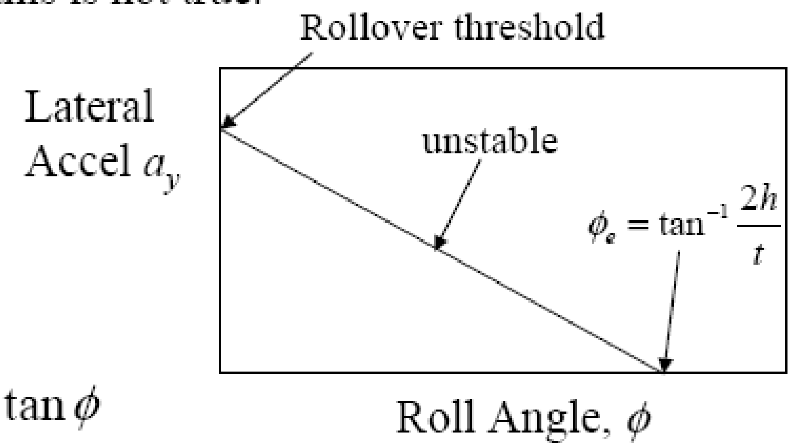
Rollover Threshold

Vehicle Type	CG Height	Tread	Rollover Threshold
Sports car	18-20 inches	50-60 inches	1.2-1.7 g
Compact car	20-23	50-60	1.1-1.5
Luxury car	20-24	60-65	1.2-1.6
Pickup truck	30-35	65-70	0.9-1.1
Passenger van	30-40	65-70	0.8-1.1
Medium truck	45-55	65-75	0.6-0.8
Heavy truck	60-85	70-72	0.4-0.6

Note that these values can exceed the cornering capabilities that arise from friction limits (about 0.8).

$$\mu = \frac{F_y}{F_z}$$

So vehicle could spin out in such a case, implying rollover would not occur. We know this is not true.



We have examined 'rigid body' rollover. The effect of roll angle shows that, at least for a simple steady-state case, there is more to rollover prediction than this simple analysis.

$$\left.\frac{a_y}{g}\right|_{F_{zi}=0,\varphi=0} = \frac{t}{2h} - \tan\phi$$

Dynamic Testing Versus Metrics

- These dynamic tests give reasonable results that correspond to real-world performance.
- Dynamic tests are not better than metrics at predicting rollover involvement.
- Extra expense of dynamic testing is substantial.
- Several practical problems remain with vehicle testing:
  - Use of human driver leads to safety concerns and mandates use of outriggers.
  - Outriggers affect handling.
  - Tire debeading may mask true limit behavior

Analysis of load restraint test

Factors Influencing Rollover

- **Gravity**
- **Speed**
- Changing direction
- Acceleration.
- Driver experience.
- Sloshing.
- Load movement.
- Type of load.
- Restraints
- **Friction**
- Stiction
- **Centre of Gravity**
- Driver alertness
- Sun
- Wind
- Brake condition
- Couplings
- Number of trailers
- Roundabout size
- R-A-Bout Camber
- Lack of time intersection
- Intersection size
- Tight Corners
- Wrong Camber
- Road condition
- On ramps
- Off ramps
- Load location
- Load viscosity
- Load packing
- Low tare weights
- Gross weight
- Load heights
- Bed heights
- Trailer format
- Camber change in turn
- Road litter
- Engine failure
- Missed gear
- Inappropriate selection
- Lane change downhill
- Suspension condition
- Tyre condition
- Tyre pressure
- New tyres
- Axle alignment
- **Suspension type**
- Time pressure
- Other road users
- Mobile phones
- Road knowledge
- **Centrifugal Force**

Rollover Safety Control

control applications:

- Active suspensions
- Active roll-bars
- Active steering
- Anti-rollover braking systems

Active suspensions and active roll-bars directly control the vehicle roll motion. The active steering reduces vehicle oversteering and the vehicle yaw moment.

Rollover Safety Control