International Automotive Engineering Spring School

Vehicle Dynamics

Vehicle Dynamics
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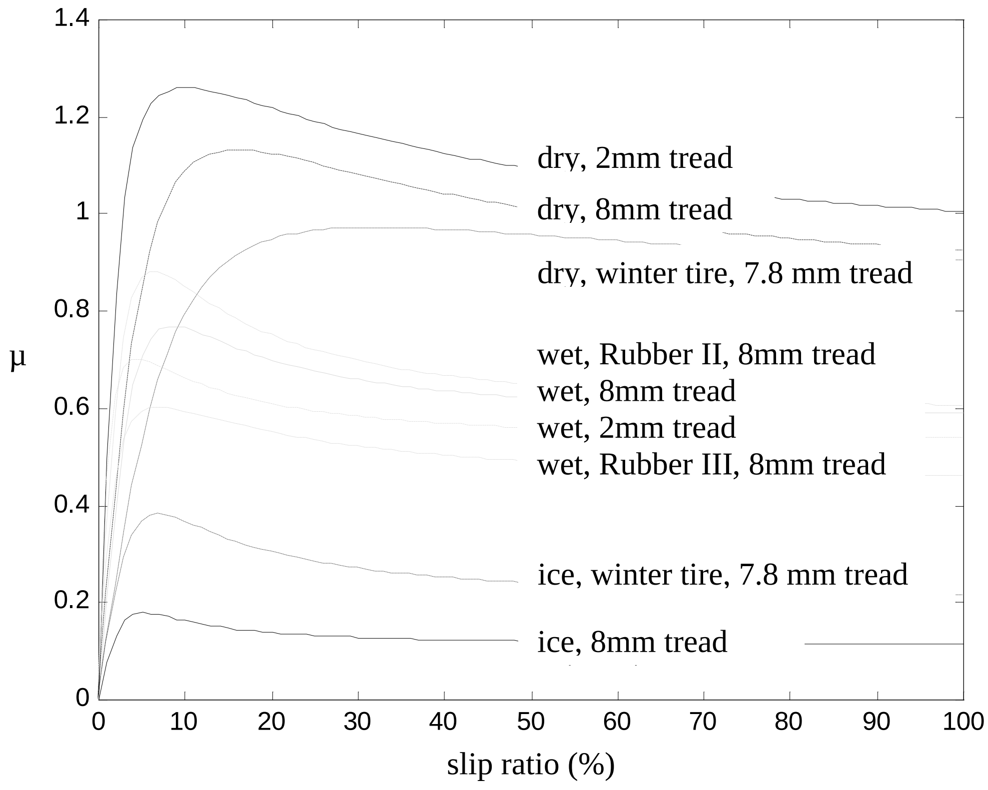
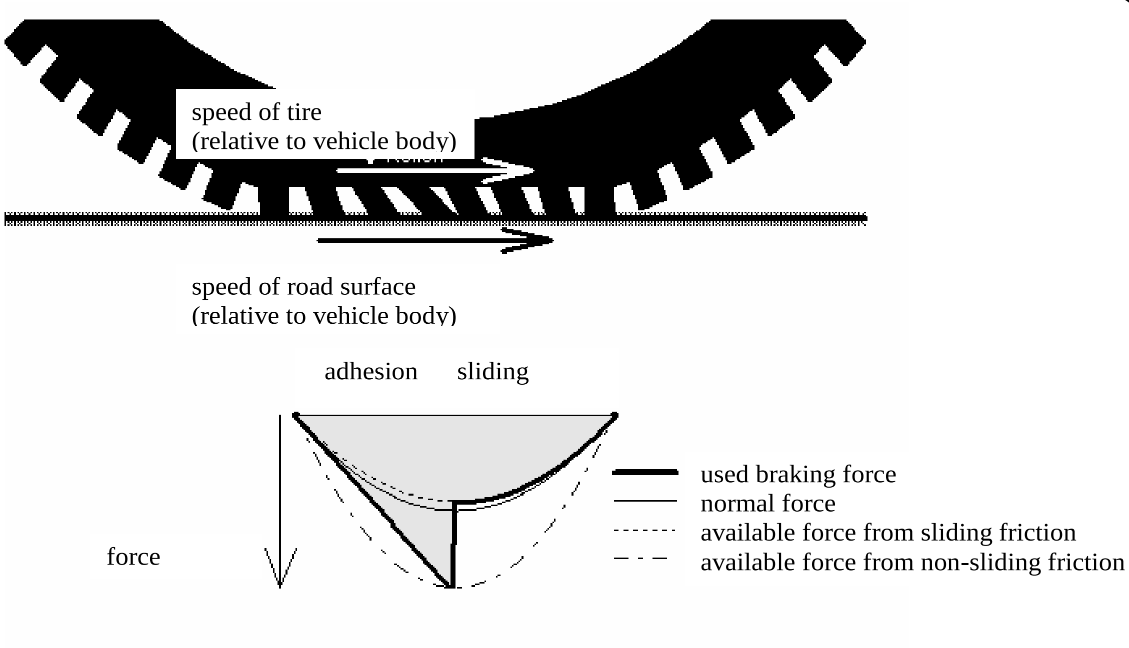
Contents

- Tire – the most important thing to understand for vehicle dynamics
- Vehicle Model - simplified
- Plus Examples
- Plus Exercises



μ-slip tire characteristic, measured

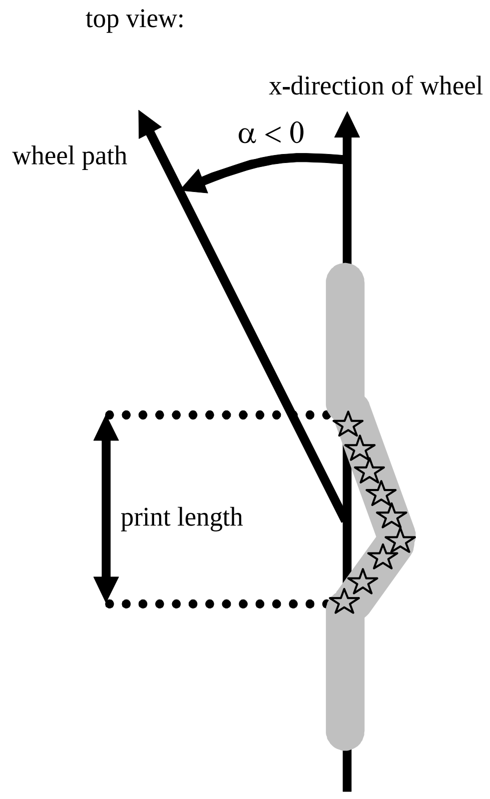
The Brush-Model of the Tire: Combination of sliding and adhesion



$$F_{long}(x) = \Delta(x) \cdot C_{Profile} = \frac{v_{surface} - v_{tire}}{v_{tire}} \cdot x \cdot C_{Profile}$$

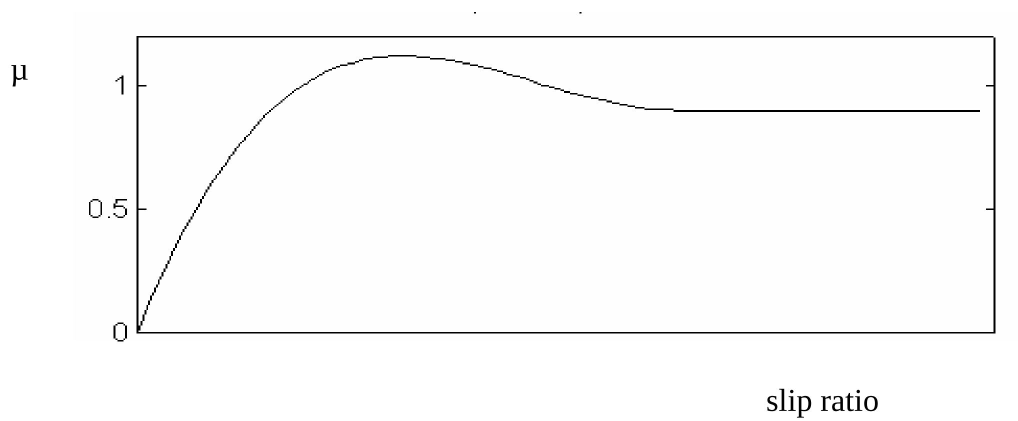
$C_{Profile}$:	mechanical stiffness of the profile elements against distortion
x:	position in the footprint, starting with zero at the left
$\Delta(x)$:	distortion of a profile element at position x
$v_{surface}$:	speed of the surface seen from vehicle body (=the (negative) value of the vehicle's speed)
v_{tire}:	speed of the tire's surface seen from vehicle body

The Brush-Model of the Tire: Same idea works lateraly,too



wheel under lateral forces

The Brush-Model of the Tire: Generated tire characteristics



μ-slip-diagram calculated
on the basis of the brush model

The Brush-Model of the Tire: What’s missing…..

The differences come from some simplifications of the brush model, e.g.

- the normal force distribution in the footprint is more complicated
- the shape of the footprint changes when braking
- there is not just two µ-values for sliding and non sliding friction but states in between
- there are always longitudinal forces within the print even without applied braking torques (the print is 'stretched', means the profile elements are distorted forwards when entering the footprint and distorted backwards when leaving the footprint, these effects eliminate each other except for some remaining rolling resistance)
- rubber is no linear spring
- **µ is load dependent, it typically drops with increasing contact pressure (that means the possible forces in x- and z-direction are not completely proportional to the normal force, plus there is a negative influence if the tire‘s contact area is not perfectly flat on ground!)**
- ......

Wheel Slip Ratio Definitions

SAE J670 defines:

longitudinal slip ratio S_x:

$$S_x = \frac{\omega - \omega_0}{\omega_0} = \frac{\omega}{\omega_0} - 1$$

⇨ values: -1.....0.....inf

with:

ω: angular velocity of the wheel

ω_0: angular velocity of the free-rolling wheel

lateral slip is:

$$S_y = \tan\alpha$$

$$v = R_{effective} \cdot \omega_0$$

with:

$R_{effective}$: Effective wheel radius (between loaded and unloaded wheel radius)

Other Wheel Slip Ratio Definitions exist…..e.g.:

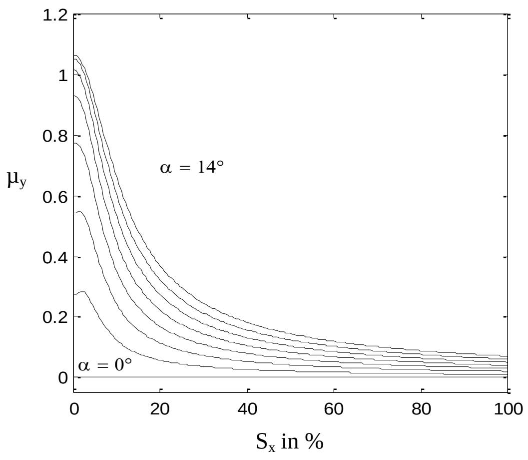
-> Very similar meaning for numbers in the relevant range 0….20%
-> But take care comparing different data!

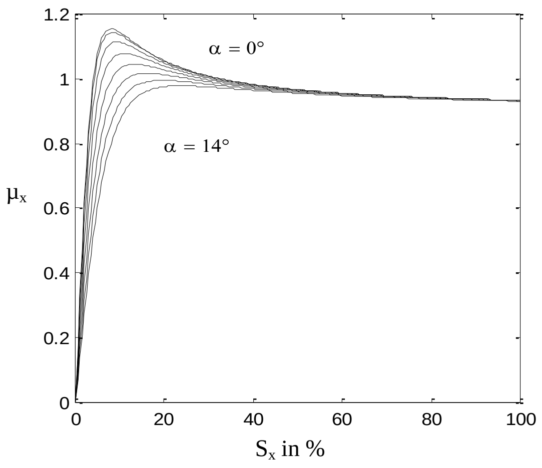
Goodyear: $$S_x = 1 - \frac{v \cdot \cos\alpha}{\Omega \cdot R_{effective}}$$ ⇨ values: -inf….0…..1

Sakai: $$S_x = \frac{v \cdot \cos\alpha}{\Omega \cdot R_{effective}} - 1$$ for traction ⇨ values: 0…..-1

$$S_x = 1 - \frac{\Omega \cdot R_{effective}}{v \cdot \cos\alpha}$$ for braking ⇨ values: 1…..0

Tire Data: Combining longitudinal and lateral slip /forces of course possible



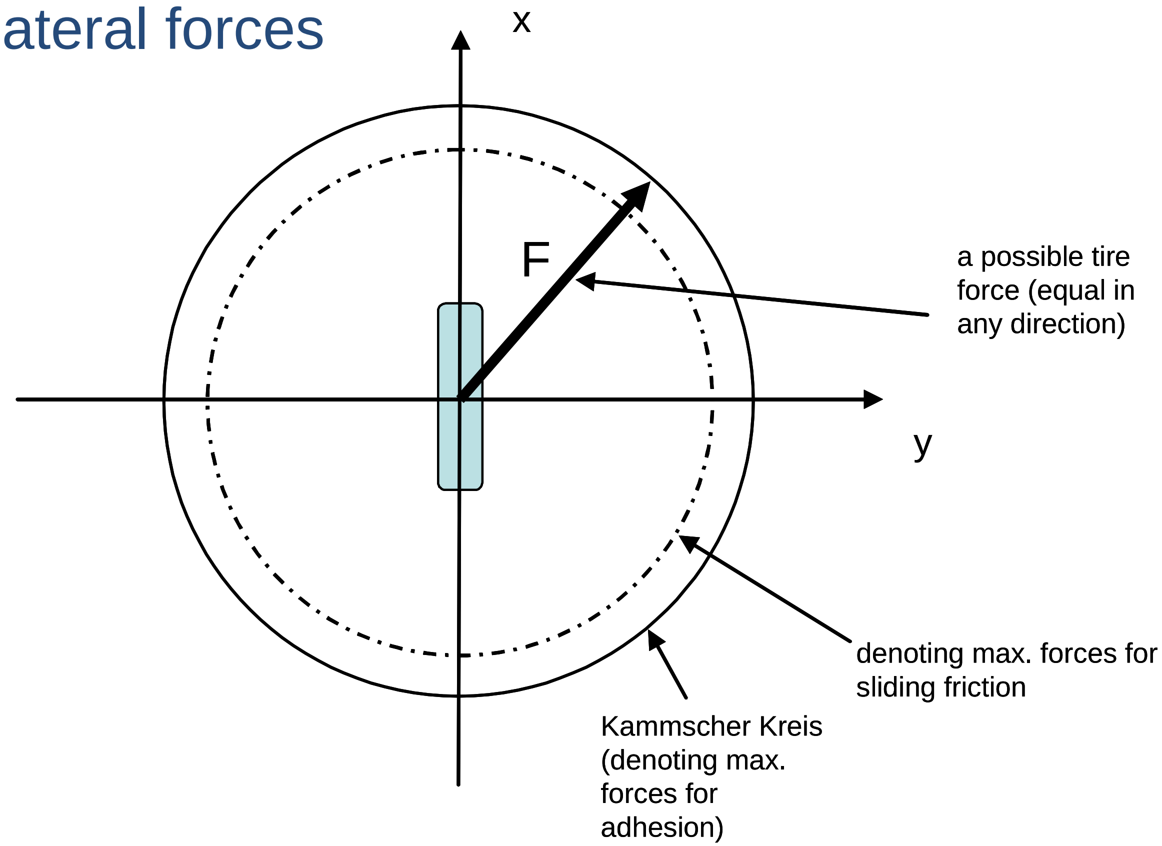


lateral tire characteristics, μ_y (at a certain load) depending on longitudinal slip for different sideslip angles α = 0°, 2°, 4°…14°

longitudinal tire characteristics, μ_x (at a certain load) depending on longitudinal slip for different sideslip angles α = 0°, 2°, 4°…14°

-> Forces in one direction will decrease the forces (or increase the slip) in the other direction

Kamm's Circle: Just looking at the maximum possible forces when combining longitudinal and lateral forces



Kamm's idea of describing a tire:
The tire's potential to transmit forces is equal in any direction in the x-y-plane.

Example of how to use Kamm's Circle:

Useful for checking if a certain combination of longitudinal and lateral acceleration is possible:

1.) Calculate necessary overall Fx, Fy_front and Fy_rear (during steady state cornering Fy is distributed to front and rear in the same percentage like static Fz is distributed on front and rear – as long as Fx is distributed the same left/right, so no torque vectoring)
2.) Estimate Fz for each wheel, represent the available forces by a Kamm‘s circle with radius $\mu_{adhesion}$*Fz
3.) From center of each Kamm‘s circle draw a vector representing Fx (we assume, that Fx distribution is known from drivetrain or brake system)
4) Check, if the remaining available lateral forces of left- and right-hand-tire are more than the required Fy of that axle (so it works) or less (car will slide)

…but no information about sideslip/steering behavior!

Forces not directly due to slip

μ changes due to wheel load, linear approximation:

$$\mu(F_z) = \mu_0 - k_{(\mu(F_z))} \cdot F_z \qquad , with \qquad k_{(\mu(F_z))} > 0$$

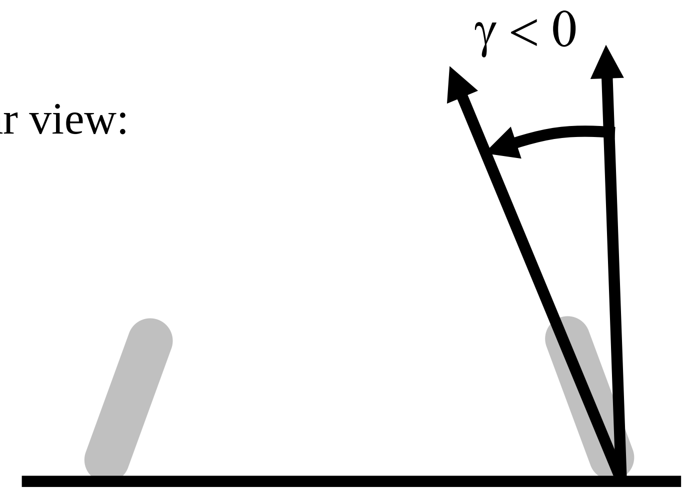
(μ typically drops in the range of 10% when the wheel load exceeds the ‘working point’ of the tire by 25%)

Real behaviour: μ changes little in a certain range, then, with increasing contact pressure, it drops (as well adhesion and sliding friction)
How much depends on rubber and surface! Needn‘t be the same for sliding and adhesion! In some situations with increasing contact pressure even INCREASING μ can occure!

Forces not directly due to slip

Camber and Inclination

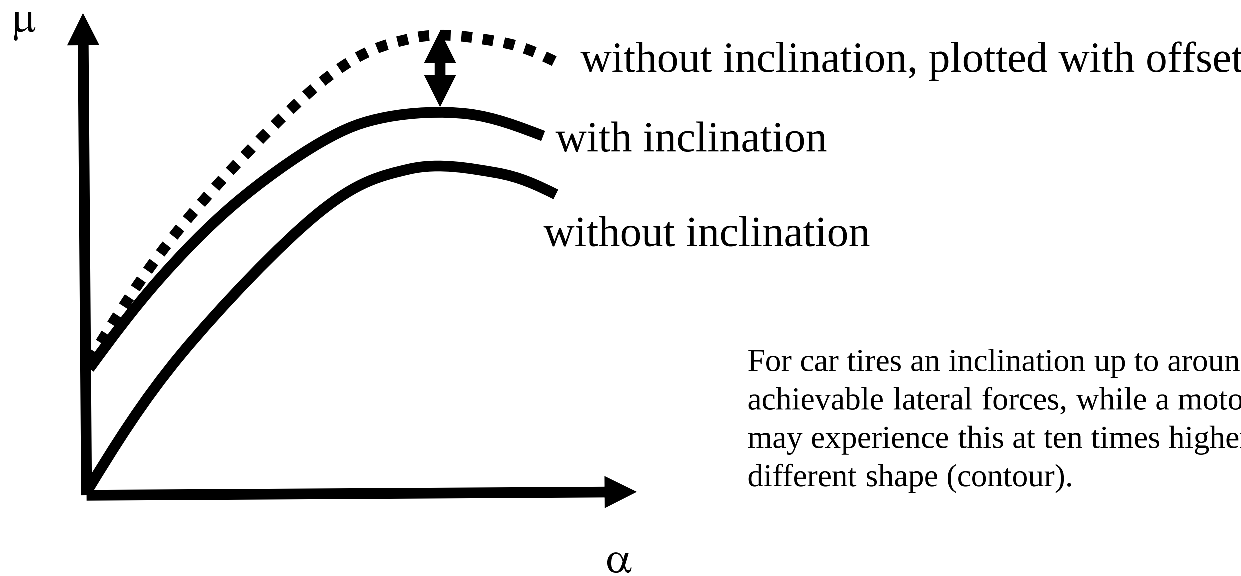
rear view:



camber:	-	-
inclination:	+	-

Forces not directly due to slip

Camber and Inclination Effect:



For car tires an inclination up to around 5^0 leads to the maximum achievable lateral forces, while a motorcycle tire may experience this at ten times higher values of up to 50^0 due to different shape (contour).

-Force at **zero sideslip**: Inclination leads to a lateral component of deformation (in tire coordinate system deformation is still only upwards, but tire system is tilted to side)

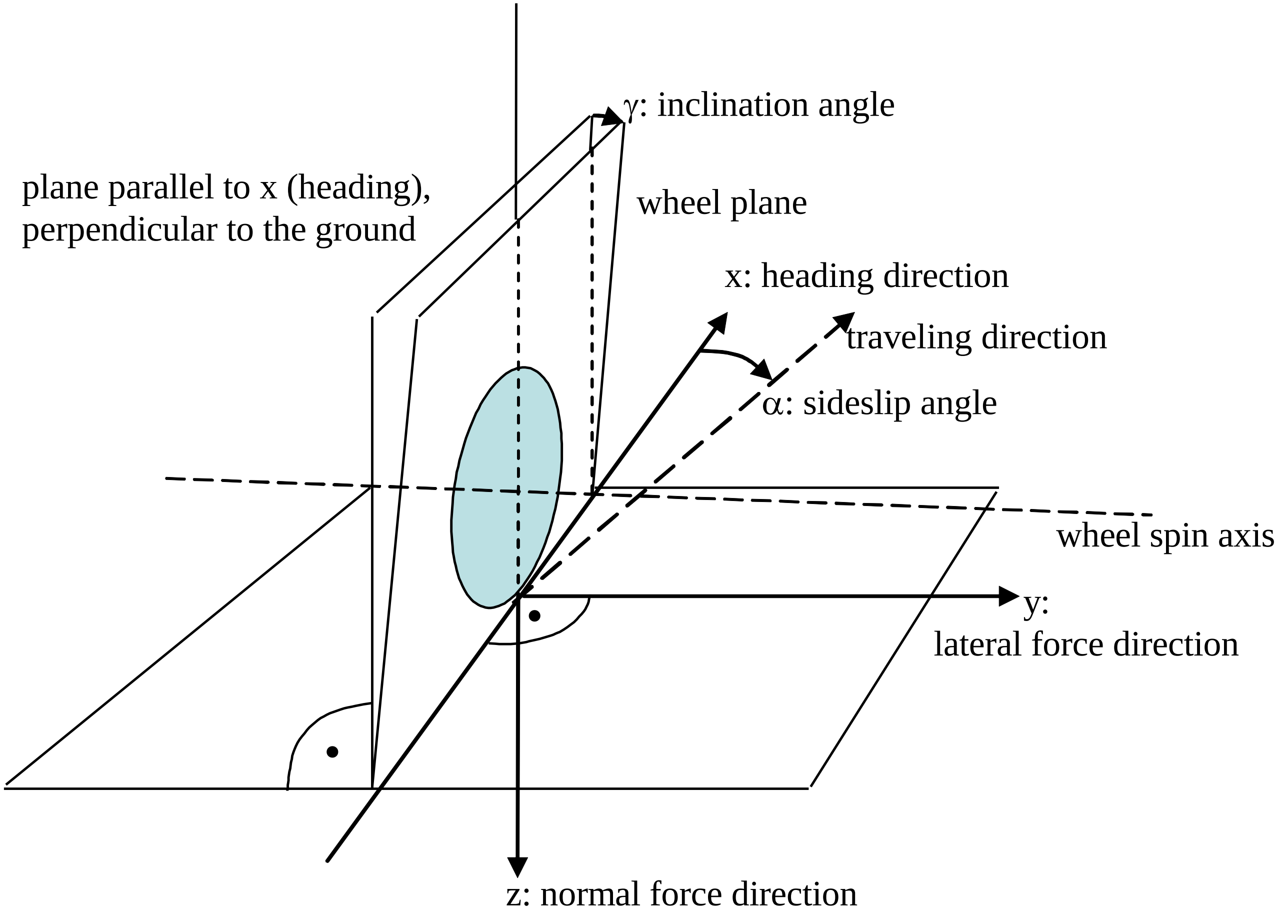
-At **peak adhesion**: Contact area still nice on ground (otherwise: Tire is mainly running on outer shoulder, by this increasing contact pressure)

-The available **longitudinal forces are typically not affected** by camber/inclination, but tire wear (and heat build up) are uneven

.....so that's why suspensions look like this......

Axis Systems and Definitions

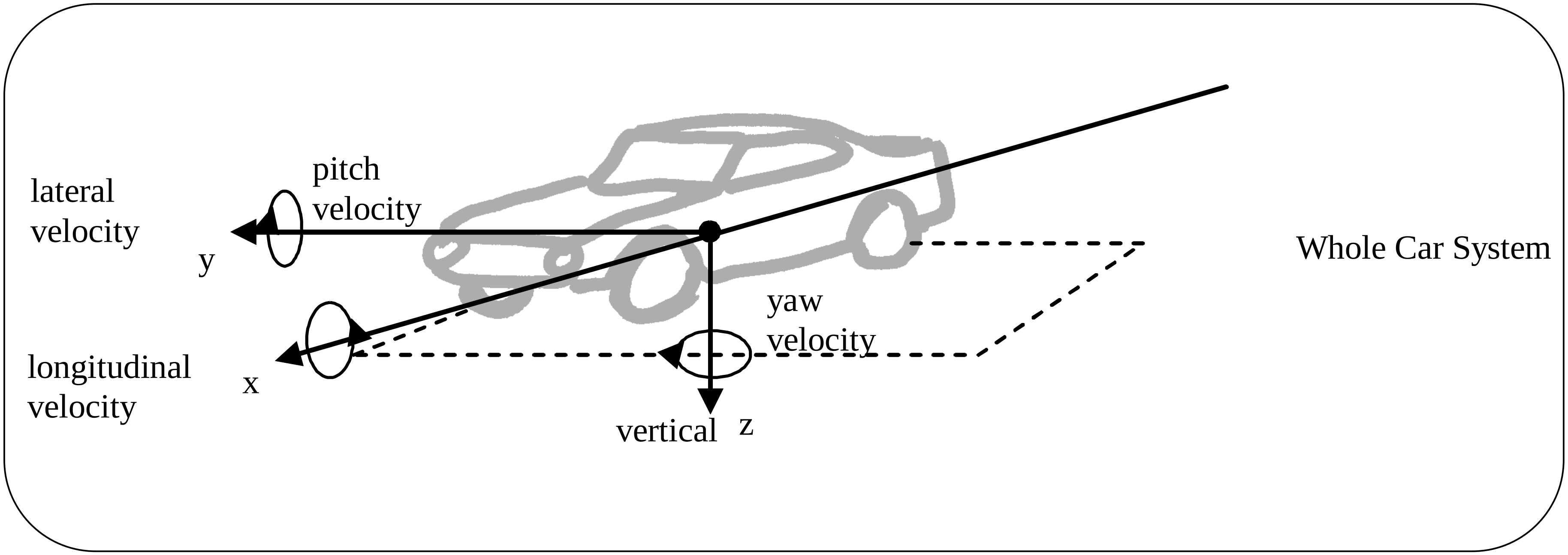
SAE Tire definitions:



Remarks:

The positive directions are shown, take care that a positive sideslip angle creates a negative lateral force!

Axis Systems and Definitions



Moments or rotational velocities are positive, if they are clockwise when looking into the positive direction of the corresponding axes.

Axis Systems and Definitions

We will always use ‘small-angle-approximation!

Basically there are, of course, different coordinate systems:

Body of car (with roll, pitch…)
Unsprung parts of the car.
Plane the car is moving in (x-direction: speed vector, y-direction: parallel to gound)
Each tire
…..

We will ignore the differences, like thee is no (or little) roll, pitch, steering input, sideslip angle…

The Bicycle Model

This following slides are about the ‘bicycle' or single-track-model:

Center of gravity is on ground
Left- and right hand tire are summed up to one tire

This model is already suitable for explaining the basic steering behaviour of a car! As it is moving in a two-dimensional plane, it has basically three degrees of freedom (x- and y-speed, rotation around z-axis: ‘yawing’).

As throttle and brake input is omitted, the absolute value of vehicle speed is assumed to be fixed, so it is only a parameter but not a degree of freedom. So the only two

degrees of freedom:

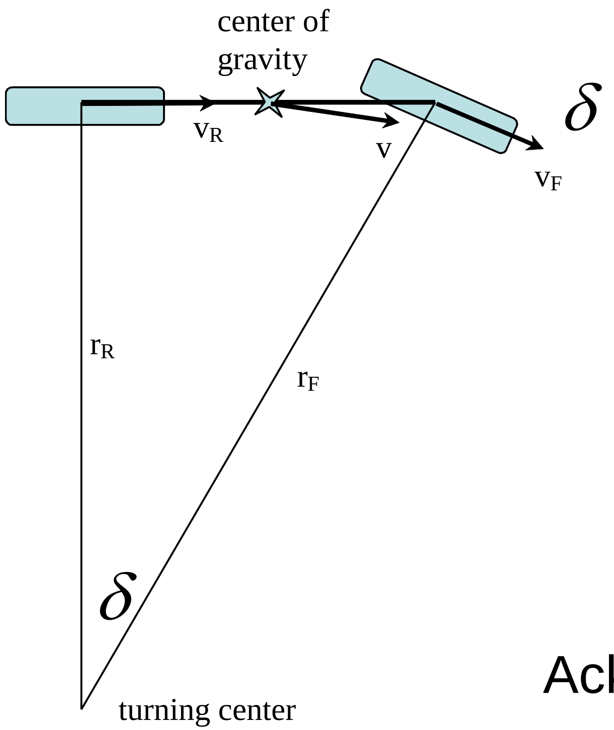
-lateral velocity v_Y (or angle of attack β, $\tan(\beta) = v_y/v_x$ with v_x and v_y measured at center of gravity)
-yawing velocity $\dot{\psi}$

Input variable:

-front wheel steering angle δ

Neutral, Over- and Understeer

Neutral steer without sideslip (or same sideslip on front and rear):



Yaw velocity:

$$\dot{\psi} = \frac{v_x \cdot \delta}{l}$$

Ackermann steering angle:

$$\delta = \frac{l}{r}$$

Definition Neutral, Over- and Understeer

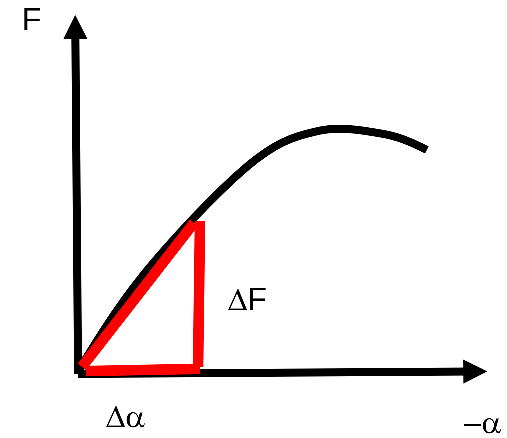
oversteer: $|\alpha_F| < |\alpha_R|$

neutralsteer: $\alpha_F = \alpha_R$

understeer: $|\alpha_F| > |\alpha_R|$

Neutral, Over- and Understeer

In the **linear** range of the tire's characteristic curve the created lateral forces depend nearly linear on the sideslip angle (-> linear bicycle or single track model):



$$F_y = C \cdot \alpha$$

$$C : Sideslipstiffness$$

In order to create the necessary overall Centripetal Force (‚CF‘) during cornering there are forces from front (F_{yF}) and rear (F_{yR}) necessary. Force equilibrium for **steady state** :

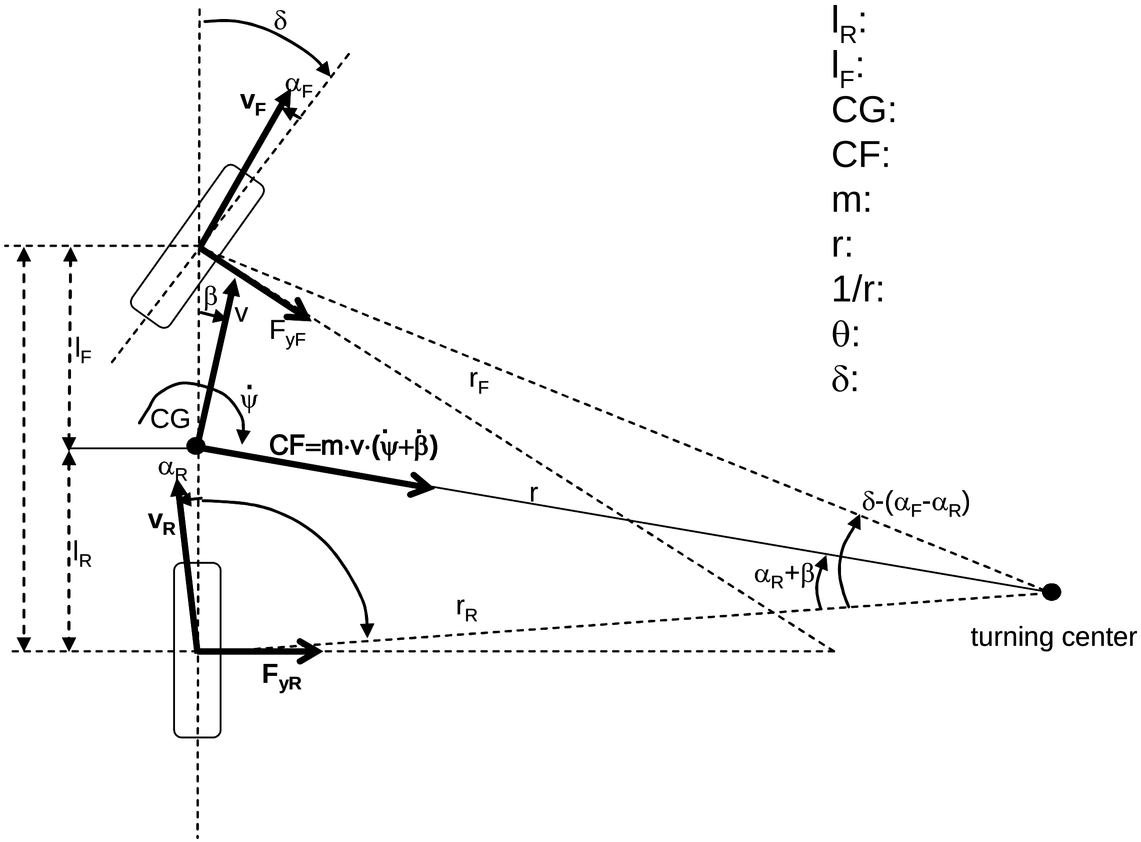
$$CF = F_{yF} + F_{yR} = C_F \cdot \alpha_F + C_R \cdot \alpha_R = m \cdot v^2 / r = m \cdot r \cdot \dot{\psi}^2$$

For zero yaw acceleration (**steady state**!) there must be moment equilibrium as well:

$$C_F \cdot \alpha_F \cdot l_F = C_R \cdot \alpha_R \cdot l_R$$

$$l_F, l_R : \text{distance center of gravity to front or rear}$$

The Bicycle Model with sideslip



ψ :	heading angle
$\dot{\psi}$:	yawing velocity
β:	attitude angle
l:	wheel base
l_R:	distance center of gravity-rear axle
l_F:	distance center of gravity-front axle
CG:	center of gravity
CF:	centripetal force
m:	vehicle mass
r:	path radius
1/r:	path curvature
θ:	moment of inertia around z-axis
δ:	steering angle

The Bicycle Model

How does the car react on steering input? If we add even transitions (i.e. non-steady-state-cornering also included)

The calculations start with the equilibrium of forces at CG:

$$m \cdot v \cdot (\dot{\psi} + \dot{\beta}) = F_{yF} + F_{yR}$$

The second equation to start with is the equilibrium of moments around the z-axis. For small steering angles this means:

$$\Theta \cdot \ddot{\psi} = F_{yF} \cdot l_f - F_{yR} \cdot l_{\mathrm{R}}$$

Then the linear tire approach is used:

$$F_{yF} = C_F \cdot \alpha_F \qquad F_{yR} = C_R \cdot \alpha_R$$

Next the following approximation for the sideslip angle is used, which is valid for small steering input and small yaw velocity:

$$\alpha_F = -\delta + \beta + \frac{\dot{\psi} \cdot l_F}{v} \qquad \alpha_R = \beta - \frac{\dot{\psi} \cdot l_R}{v}$$

This leads to the following equations of motion:

The Bicycle Model

$$m \cdot v \cdot \dot{\beta} + \frac{1}{v} \cdot (m \cdot v^2 - C_F \cdot l_F + C_R \cdot l_R) \cdot \dot{\psi} - (C_F + C_R) \cdot \beta + C_F \cdot \delta = 0$$

$$\Theta \cdot \ddot{\psi} - \frac{1}{v} \cdot (C_F \cdot l_F^2 + C_R \cdot l_R^2) \cdot \dot{\psi} - (C_F \cdot l_F - C_R \cdot l_R) \cdot \beta + C_F \cdot l_F \cdot \delta = 0$$

For the stationary case the yaw acceleration and the change of the angle of attack is zero:

$$\dot{\beta} = \ddot{\psi} = 0$$

Inserting this, one yields the yaw response due to steering input. All minor calculation steps are shown in the following (for those, who are interested). Insertion leads to:

$$\frac{1}{v} \cdot (m \cdot v^2 - C_F \cdot l_F + C_R \cdot l_R) \cdot \dot{\psi} - (C_F + C_R) \cdot \beta + C_F \cdot \delta = 0$$

$$-\frac{1}{v} \cdot (C_F \cdot l_F^2 + C_R \cdot l_R^2) \cdot \dot{\psi} - (C_F \cdot l_F - C_R \cdot l_R) \cdot \beta + C_F \cdot l_F \cdot \delta = 0$$

The goal is to describe the yawing depending on the steering input, so we need to get rid of β. These two equations can be converted to:

The Bicycle Model

$$\left[\frac{1}{v}\cdot(m\cdot v^2 - C_F\cdot l_F + C_R\cdot l_R)\cdot\dot{\psi} - (C_F + C_R)\cdot\beta + C_F\cdot\delta\right]\cdot\left[-(C_F\cdot l_F - C_R\cdot l_R)\right] = 0$$

$$\left[-\frac{1}{v}\cdot(C_F\cdot l_F^2 + C_R\cdot l_R^2)\cdot\dot{\psi} - (C_F\cdot l_F - C_R\cdot l_R)\cdot\beta + C_F\cdot l_F\cdot\delta\right]\cdot\left[-(C_F + C_R)\right] = 0$$

Bringing the part that depends on β on one side for each equation this leads to:

$$\left[\frac{1}{v}\cdot(m\cdot v^2 - C_F\cdot l_F + C_R\cdot l_R)\cdot\dot{\psi} + C_F\cdot\delta\right]\cdot\left[-(C_F\cdot l_F - C_R\cdot l_R)\right] = \left[-(C_F\cdot l_F - C_R\cdot l_R)\right]\cdot(C_F + C_R)\cdot\beta$$

$$\left[-\frac{1}{v}\cdot(C_F\cdot l_F^2 + C_R\cdot l_R^2)\cdot\dot{\psi} + C_F\cdot l_F\cdot\delta\right]\cdot\left[-(C_F + C_R)\right] = (C_F\cdot l_F - C_R\cdot l_R)\cdot\beta\cdot\left[-(C_F + C_R)\right]$$

On the right side of each equation is the same term, so the left side of the above equations equal each other:

$$\left[\frac{1}{v}\cdot(m\cdot v^2 - C_F\cdot l_F + C_R\cdot l_R)\cdot\dot{\psi} + C_F\cdot\delta\right]\cdot\left[-(C_F\cdot l_F - C_R\cdot l_R)\right] =$$

$$= \left[-\frac{1}{v}\cdot(C_F\cdot l_F^2 + C_R\cdot l_R^2)\cdot\dot{\psi} + C_F\cdot l_F\cdot\delta\right]\cdot\left[-(C_F + C_R)\right]$$

Some calculations, so that the terms are sorted depending on yaw velocity and steering input:

$$\frac{1}{v}\cdot(m\cdot v^2 - C_F\cdot l_F + C_R\cdot l_R)\cdot\left[-(C_F\cdot l_F - C_R\cdot l_R)\right]\cdot\dot{\psi} + C_F\cdot\delta\cdot\left[-(C_F\cdot l_F - C_R\cdot l_R)\right] =$$

$$= -\frac{1}{v}\cdot(C_F\cdot l_F^2 + C_R\cdot l_R^2)\cdot\left[-(C_F + C_R)\right]\cdot\dot{\psi} + C_F\cdot l_F\cdot\delta\cdot\left[-(C_F + C_R)\right]$$

The Bicycle Model

All yaw-velocity-dependent parts are brought to the left side, all steering-wheel-dependent terms to the right:

$$\frac{1}{v}\cdot(m\cdot v^2 - C_F\cdot l_F + C_R\cdot l_R)\cdot[-(C_F\cdot l_F - C_R\cdot l_R)]\cdot\dot{\psi} + \frac{1}{v}\cdot(C_F\cdot l_F^2 + C_R\cdot l_R^2)\cdot[-(C_F + C_R)]\cdot\dot{\psi} =$$
$$= +C_F\cdot l_F\cdot\delta\cdot[-(C_F + C_R)] - C_F\cdot\delta\cdot[-(C_F\cdot l_F - C_R\cdot l_R)]$$

After doing all the multiplications:

$$\frac{1}{v}\cdot\{m\cdot v^2\cdot(-C_F\cdot l_F + C_R\cdot l_R) + (C^2{}_F\cdot l^2{}_F + C^2{}_R\cdot l^2{}_R - 2C_F l_F\cdot C_R l_R) - C^2{}_F\cdot l_F^2 - C^2{}_R\cdot l_R^2 - C_F C_R\cdot l_F^2 - C_F C_R\cdot l_R^2\}\cdot\dot{\psi} =$$
$$= \{-C_F\cdot C_R\cdot l\}\cdot\delta$$

Rearranging, so that -finally- the desired expression for the yaw amplification (yaw velocity due to steering input), is on the left hand side:

$$\frac{\dot{\psi}}{\delta} = \frac{v\cdot\{-C_F\cdot C_R\cdot l\}}{\{m\cdot v^2\cdot(-C_F\cdot l_F + C_R\cdot l_R) - 2C_F l_F\cdot C_R l_R - C_F C_R\cdot l_F^2 - C_F C_R\cdot l_R^2\}}$$

Again some rearranging:

$$\frac{\dot{\psi}}{\delta} = \frac{v}{\left\{\frac{m\cdot v^2\cdot(C_F\cdot l_F - C_R\cdot l_R)}{\{C_F\cdot C_R\cdot l\}} + \frac{2C_F l_F\cdot C_R l_R + C_F C_R\cdot l_F^2 + C_F C_R\cdot l_R^2}{\{C_F\cdot C_R\cdot l\}}\right\}}$$

The Bicycle Model

Getting rid of some C_Fs and C_Rs:

$$\frac{\dot{\psi}}{\delta} = \frac{v}{\left\{ \frac{m \cdot v^2 \cdot (C_F \cdot l_F - C_R \cdot l_R)}{\{C_F \cdot C_R \cdot l\}} + \frac{2l_F \cdot l_R + l_F^2 + \cdot l_R^2}{l} \right\}} = \frac{v}{\left\{ \frac{m \cdot v^2 \cdot (C_F \cdot l_F - C_R \cdot l_R)}{\{C_F \cdot C_R \cdot l\}} + \frac{(l_F + l_R)^2}{l} \right\}}$$

So:

$$\frac{\dot{\psi}}{\delta} = \frac{v}{l + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l \cdot C_F \cdot C_R} \right] \cdot v^2} = \frac{v/l}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R} \right] \cdot v^2}$$

$$\frac{\dot{\psi}}{\delta} = \frac{v/l}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R} \right] \cdot v^2}$$

Yaw response

with:

$$\left[\frac{m \cdot (C_R \cdot l_R - C_F \cdot l_F)}{l \cdot C_F \cdot C_R} \right] =: K \text{ stability factor}$$

and:

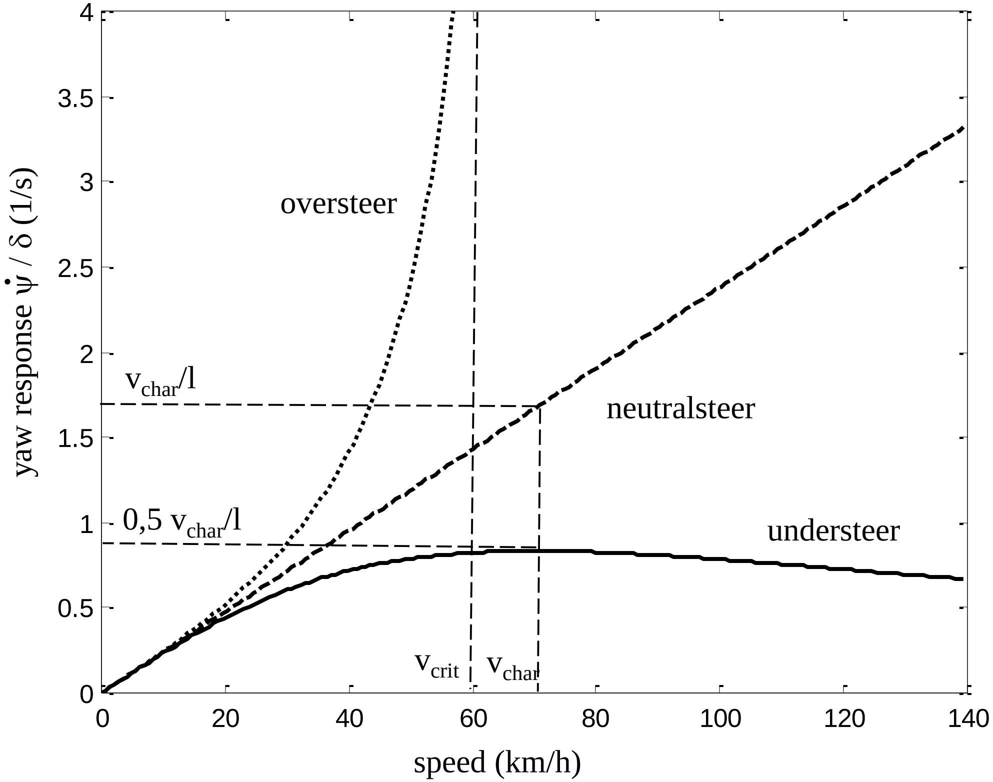
$$\left[\frac{m \cdot (C_R \cdot l_R - C_F \cdot l_F)}{l^2 \cdot C_F \cdot C_R} \right]^{-1} = \frac{l}{K} =: v_{ch}^2 \text{ characteristic / critical speed (dep. on under - or oversteer)}$$

Steady State Yaw Response

$$\frac{\dot{\psi}}{\delta} = \frac{v/l}{1 - \left[\frac{v^2}{vcrit^2}\right]}.$$

$$\frac{\dot{\psi}}{\delta} = \frac{v/l}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R}\right] \cdot v^2}$$

Yaw response



$$\frac{\dot{\psi}}{\delta} = v/l$$

$$\frac{\dot{\psi}}{\delta} = \frac{v/l}{1 + \left[\frac{v^2}{vchar^2}\right]}.$$

The Bicycle Model

Comparison: Two cars drive with same speed on same circle track:

Bicycle Model (describes real car):

$$\frac{\dot{\psi}}{\delta} = \frac{v/l}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R}\right] \cdot v^2}$$

$$\dot{\psi} = \frac{v/l}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R}\right] \cdot v^2} * \delta$$

(Theoretical) neutral-steer-version of car:

$$\frac{\dot{\psi}}{\delta(\text{Ackermann})} = \frac{v/l}{1}$$

$$\dot{\psi} = v/l * \delta(\text{Ackermann})$$

Same yawing velocity (same circle, same speed!):

$$\delta(\text{Ackermann}) = \frac{1}{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R}\right] \cdot v^2} * \delta$$

$$\delta(\text{Ackermann}) * \{1 + \left[\frac{m \cdot (C_F \cdot l_F - C_R \cdot l_R)}{l^2 \cdot C_F \cdot C_R}\right] \cdot v^2\} = \delta$$

At very low speed (no sideslip) we steer according to Ackermann steering angle. With increasing speed we need more input. How much more is described by the factor behind the δ(Ackermann).

Dynamic Yaw Response

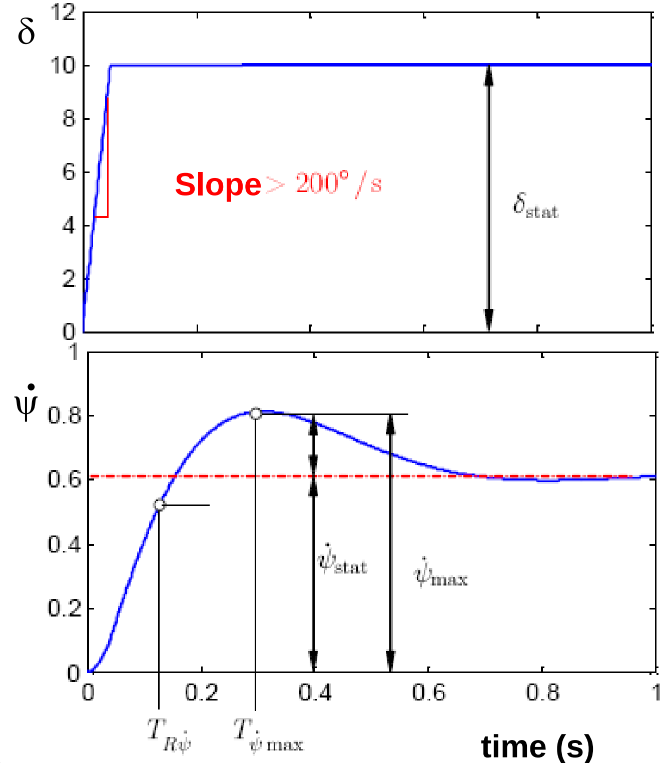
Keep in mind that these calculations were steady state!

The dynamic yaw response to a steering input is similar to a spring-damper system reacting on a force input.

Dynamically interesting:

•How much time is necessary to reach the steady state conditions after a steering angle input? (Typically 90 of steady state after 200...400 ms)

•Is there a big overshoot?

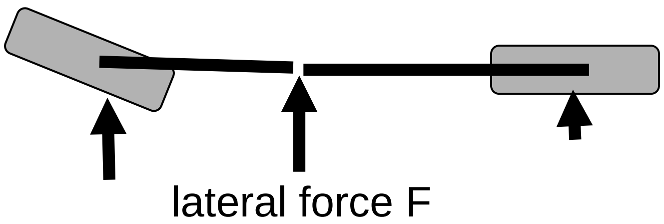


The eigenfrequency will go up a little bit with speed!
But especially: **Damping drops with increasing speed!** (at very low speed it will stop to oscillate and show first-order-behaviour)

Linear Bicycle Model can also be used e.g. if a lateral force occurs e.g. due to inclined road or applies not at center of gravity (e.g. wind gust from side)

......on the other hand some 'mechanical' thoughts:

$$\delta(\text{Ackermann}) = \delta + \left[\frac{(C_R \cdot l_R - C_F \cdot l_F)}{l \cdot C_F \cdot C_R}\right] * \frac{m \cdot v^2}{r}$$



$$F_F = \frac{l_R}{l} F \qquad \text{for cornering}: F = \frac{m \cdot v^2}{r} \qquad F_R = \frac{l_F}{l} F$$

$$C_F a_F = \frac{l_R}{l} F \qquad C_R a_R = \frac{l_F}{l} F$$

$$a_F = \frac{C_R l_R}{C_R C_F l} F \qquad a_R = \frac{C_F l_F}{C_R C_F l} F$$

$$a_F - a_F = \frac{C_R l_R - C_F l_F}{C_R C_F l} F$$

(remember : F may be any lateral force....)

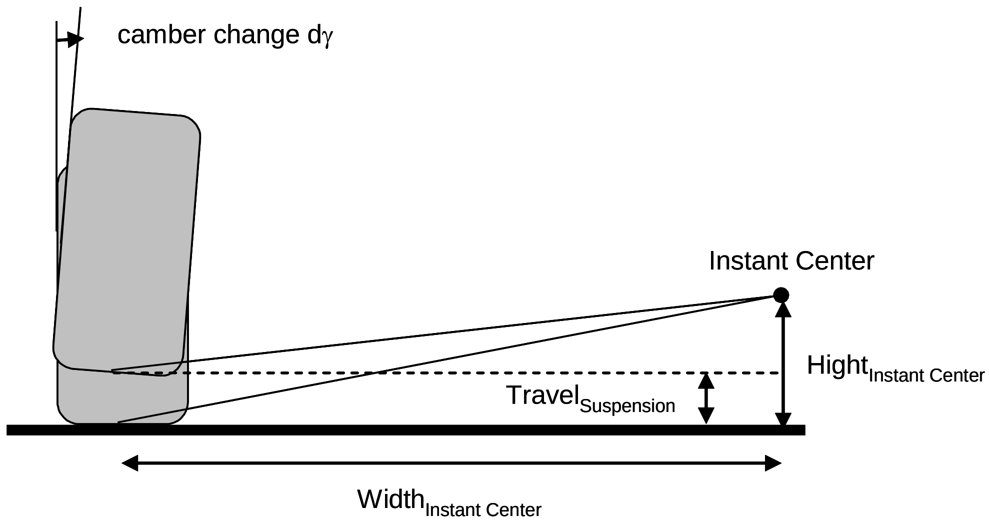
for cornering we get again : $\delta(\text{Ackermann}) = \delta + (a_F - a_R)$

$$\delta(\text{Ackermann}) = \delta + \left[\frac{(C_R \cdot l_R - C_F \cdot l_F)}{l \cdot C_F \cdot C_R}\right] * \frac{m \cdot v^2}{r}$$

...... the same result like the linear bicycle model predicts for centrifugal force

Influencing Driving Behaviour: Camber (static and/or kinematic)

Camber:



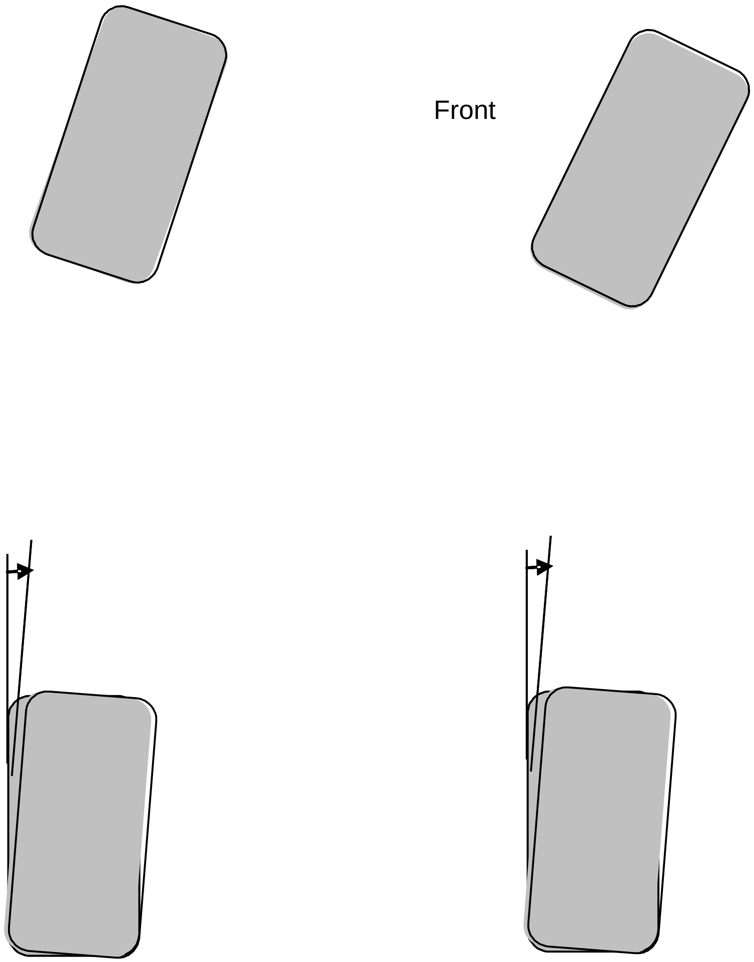
Negative camber change can (over)-compensate the leaning outward of the vehicle body during cornering.

-> Effect is the same like a tire with more peak µ (effective also in non-linear tire range)

But: Inhomogeneous tire wear (and heat build up) may result due to camber

Influencing Driving Behaviour: (Elasto-)kinematic steering

(Elasto-) kinematic steering of rear axle



Rear tires point to the center of the curve (little, maybe 1 or 2 degree). This happens depending on suspension travel ('kinematic') and/or due to elastic elements (rubber bushings) in the suspension ('elasto-kinematic')

<u>-> Effect is the same like a tire with more sideslip stiffness (in linear range) rear</u>

Influencing Driving Behaviour: (Elasto-)kinematics overview

Standard Suspension Measures..... always a compromise

Contribution of linkage without any elastic deflections, depends on suspension travel

The elastic deflections of the rubber parts can add an additional 'elasto-kinematic' effect depending on the longitudinal/lateral forces, that are transmitted.

	Kinematic	Elasto-Kinematic
Steering angle ('Toe-In' and 'Toe-Out')	X	X
Camber	X	o

To provide / increase understeer the rear axle shows typically:

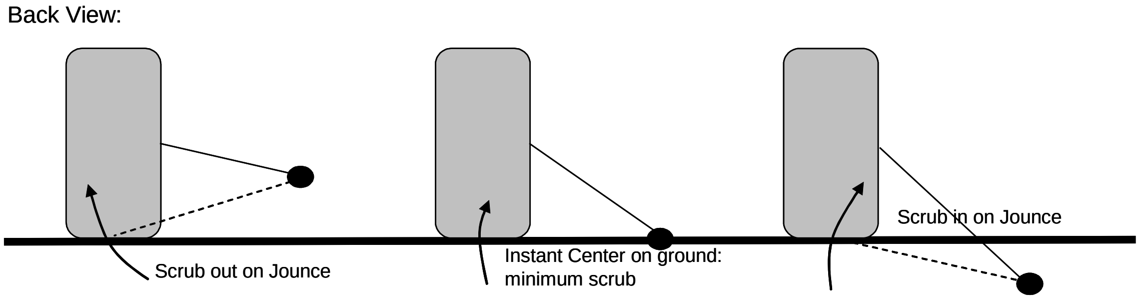
- toe.in on suspension compression, i.e. the rear tires steer to the inside of the turn, which helps especially in the linear tire range
- some negative camber (statically plus/or kinematicaly due to suspension travel) in order to provide some extra lateral grip at the adhesion limit

Influencing Driving Behaviour: Load-transfer left/right

Load transfer left/right typically happens more on front (to increases understeer)

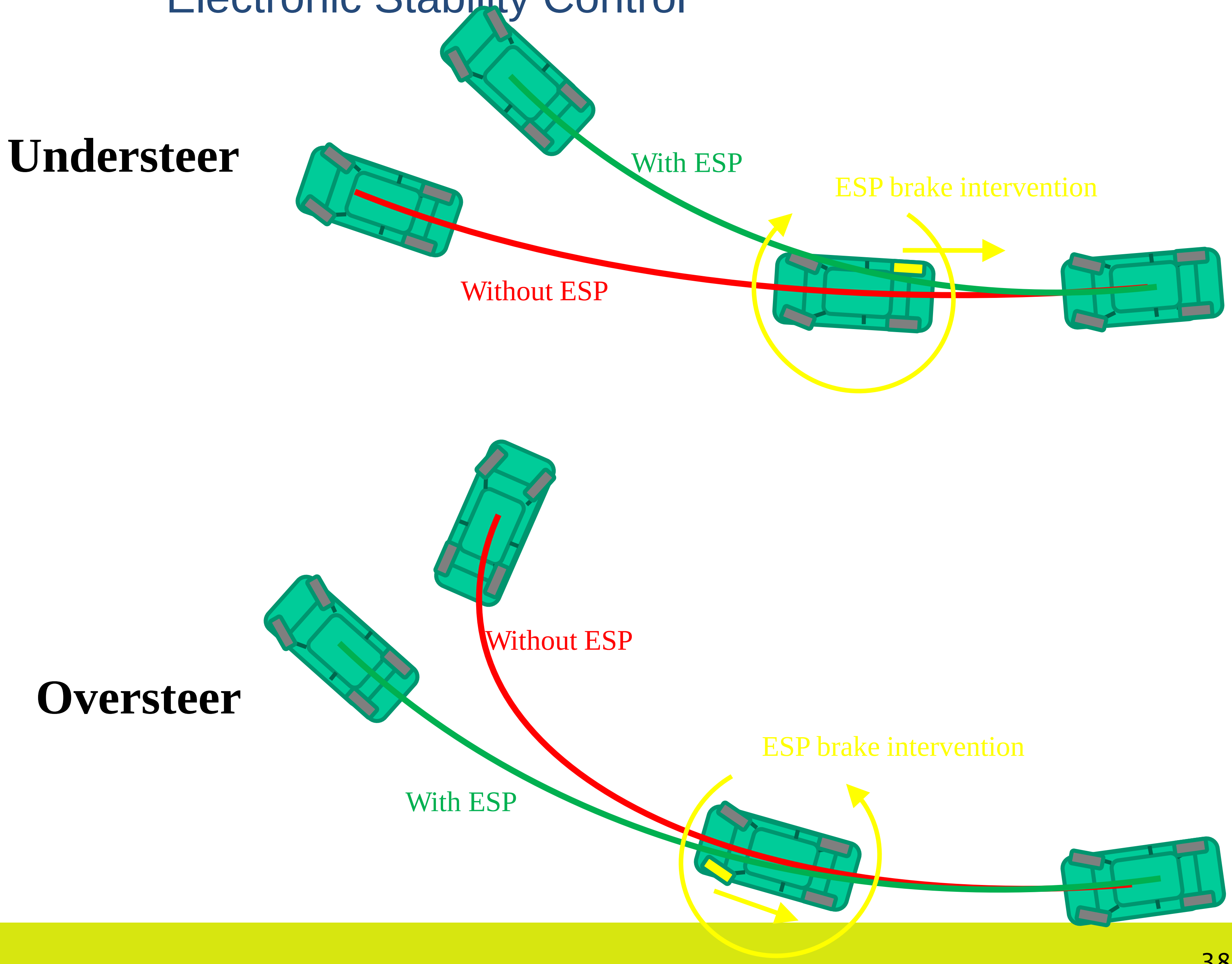
- **typically done by stiffer anti-sway-bar (=anti-roll-bar = stabiliser bar) on front (also possible with stiffer suspension springs, but this reduces ride comfort)**

Take care: Lateral tire movement also contributes to normal-force-changes during cornering! (Typical ,anti-roll-effect at about 20%)

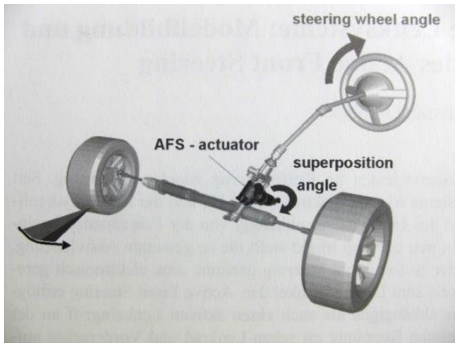


If the wheel is moving outwards during suspension compression, it has to work 'against' the lateral acceleration, by this reducing roll.
But: lateral slip and hence forces occure due to road impurities as well!

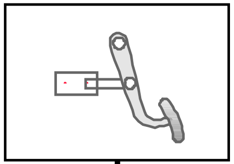
Electronic Stability Control



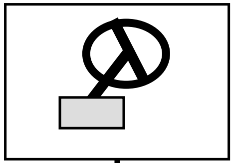
Active (Front) Steering



Brake pedal simulator



Steering head module



Sensors

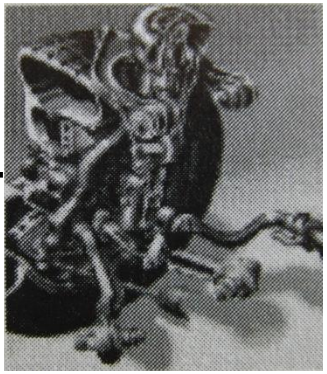
- Yaw rate sensor
- Steering angle sensor
- Transverse acceleration sensor
- Torque sensor steering
-Force sensor braking
-.....

Intelligent Chassis Controller

- Safe data and sensor bus
- Safe operating system (t-triggered)
- System and design reliability (FMEA, TEFT)
- Full X-by-wire functionality
- Networking topology/architecture

Brake-by-Wire (EMB)

Active Suspension (active damping/ springs/swaybars...)



Electric/HybridDrive

Bilder aus ‚Fahrdynamikregelung', R. Isermann, Vieweg 2007

What?	How?	Effects/Possibilities?
Steering:		
Active Steering	Superimposes an additional steering angle to the drivers input, can simply be switched off in case of fault	Stability control without braking Variable steering gear ratio (parking-high speed) Lane keeping aid (together with e.g. a camera)
Rear Wheel Steering	Adds a steering angle to the rear wheels (electically or hydraulically)	Tighter Cornering possible (low speed) At higher speed rear wheels steer in the same direction like the front wheels to improe driving stability
Steer-By-Wire	Completely actuator dependant steering, no mechanical coupling to driver	Like Active Steering, but less restrictions regarding possible steering angle values Packaging advantages

What?	How?	Effects/Possibilities?
Suspension:		
Switchable Bushings	Bushings can be switched hard/soft like modern motor silent blocks	Less restrictions regarding the compromise comfort and vehicle dynamics
Active Bushings/Suspension Linkage	The length of parts of the suspension linkage is changeable (electrically or hydrauliclly), mostly on rear axle. Depending on location steering angle and/or inclination can be controlled independently	Steering angle changes can be used as parking aid/tighter cornering, and for vehicle stability issues. Changes in inclination are suitable for increasing stability and can substantially increase the cornering performance.
Active Dampers	Damping coefficient can be changed	Switching between softer (comfort) and harder damping, which improves comfort and cornering/bad road contact by reducing wheel and body resonances. Able to influence wheel loads during instationary maneuvers to improve driving behaviour.
Active Swaybars	Swaybars are electrically pretensioned	Able to level out roll during cornering. Able to influence wheel loads to improve driving behaviour. Softer sway bars possible which improves the road contact and comfort on bad roads
Air Springs	Suspension springs consist of air cushions instead of steel springs; air cushions can be filled/emptied	Levelling of vehicle possible by rising the pressure, so lower CG or softer springs are possible. Stiffness rises proportional with additional weight if level is maintained, so body frequencies stay the same. Mostly too slow for active intervention.

What?	How?	Effects/Possibilities?
Suspension:		
Hydropneumatic	Suspension springs consist of gas cushions instead of steel springs; air cushions act via hydraulic fluid on the suspension, fluid volume can be changed, gas volume fixed.	Levelling of vehicle possible by changing fluid volume, so lower CG or softer springs are possible. Stiffness rises overproportional with additional weight, so body frequencies increase. Mostly too slow for active intervention. Good combination with steel springs (here the body frequencies go down under additional load)
Hydractive Suspension	Same like Hydropneumatic, but additional gas volumes can be connected/disconnected via valves within milliseconds	Switching suspension firmness (damping and stiffness), which improves cornering and bad road comfort/contact. Able to level out roll during cornering and pitch during braking/acceleration. Able to influence wheel loads to improve driving behaviour. No sway bars needed which improves the road contact and comfort on bad roads
Semiactive Suspension	Additionally to (adaptive) dampers and conventional springs there is a device to pretension the springs electrically	Able to level out roll during cornering and pitch during braking/acceleration. Able to influence wheel loads to improve driving behaviour. No sway bars needed which improves the road contact and comfort on bad roads

What?	How?	Effects/Possibilities?
Drivetrain:		
Hybrid	Acceleration via the drivetrain, which might be one motor per wheel	Depending on the drivetrain the longitudinal forces are free controllable (one electric engine per wheel), so that stability control can become possible without break intervention but by (re)distribution of acceleration
Clutches	second axle of 4-wheel-drive is connected via controllable clutch	Moments can be (partly) redistributed to the connected axle, so that the longitudinal wheel slips can be reduced or the relation between front and rear wheel slip can be influenced, by this improving stability.
Switchable Locking Differential	Moment from the wheel with the higher wheel speed can be transfered to the lower turning wheel.	The additional acceleration potential of cars with locked differentials can be made available without getting the typical disadvantages regarding maneuverability
Active Yaw Control with Active Differentials (often switchable/controlable differentials are called 'active differential', too!)	Differentials have gears and clutches inside, which allow a free moment distribution between both wheels independent of wheel speed	The longitudinal forces are free controllable, so that stability control can become possible without break intervention but by (re)distribution of acceleration