New concepts of active yaw control for electric and hybrid vehicles

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Abstract: To reduce the CO2-consumption of vehicles alternative drive concepts are developed. Two promising concepts are hybrid vehicles and full electric cars. Both concepts need high power electricity which can be also used for a new type of active yaw control. With it safety, efficiency, agility, traction and comfort can be more increased.
The following paper shows principle concepts of active yaw control, mechanical gear structures found by a systematic synthesis and the effect on the vehicle dynamics.

Keywords: vehicle dynamics, torque vectoring, active yaw control, electric drive train, hybrid, active differential, safety, efficiency, agility, sportiness, traction, comfort

1 Introduction

To protect our environment and in order to meet emission regulations the CO_2 emissions must be reduced. For an exemplary compact car[1], driving at 120 km/h, 18 % of the total power is loosed because of the air drag resistance and 6 % because of the rolling resistance. The other power losses of 76 % results of exhaust gas (ca 30 %), engine cooling (ca 27 %), air charge (ca 5 %), engine friction (ca 8 %), auxiliary drive (ca 2 %) and drive train (ca 2 %) [3]. In case of conventional drive concepts with combustion engines the CO_2 emissions can be reduced by downsizing the engine (e. g. by reducing the cubic capacity in combination with a multistage turbocharger) or other actual developments like HCCI[2] and hydrogen engines. Furthermore there are new drive concepts like hybrid transmission and full electric vehicles.

Today, most cars are equipped with mechanical wheel brakes. These are not only used for decelerating but also to get different torques on different wheels e. g. as it is necessary at different traction coefficients (μ-split). By an activation of only one wheel brake a yaw torque along the vertical vehicle axis is generated. This yaw torque can be used to stabilize the vehicle or to increase the sportiness at a cornering driving maneuver. Using the wheel brakes is a cheap but not an efficient way for active yaw control (AYC). Especially for hybrid and full electric vehicles there are solutions with less power loss.

2 Concepts of active yaw control

2.1 Principle concepts

The basis of AYC is an individual drive torque on each wheel based on the actual driving situation. The standard drive concept with an open differential provides only a fixed relation between the wheel torques and does not support AYC. But this can be achieved by two separate engines at one axle (e. g. wheel integrated engines), two coupled engines or by one common engine and an active differential (Fig. 1).

Advantages of the concept with two separate engines are the modularity and the fail-save function because of redundancy. Disadvantages are the higher costs for engine and power inverter, the higher mass and the synchronization problem in case of gear switching. Additional disadvantages of wheel integrated engines are higher unsprung weights, restricted kinematic points of the wheel suspension and higher dynamic loads in case of road bump.

[1] car properties: modern gasoline engine, m=1400 kg, c_w=0.31, A=2.1 m², f_R=0.01, ρ_L=1.26kg/m³; fuel consumption: 6.6 l/100km
[2] Homogeneous Charge Compression Ignition

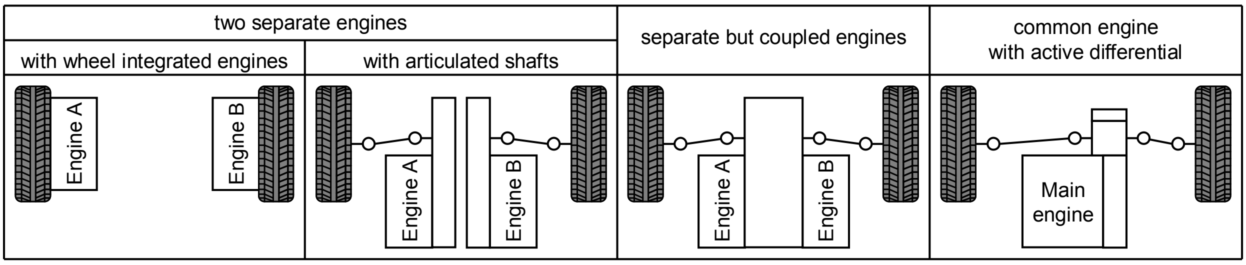

Fig. 1: Principal methods for individual drive torques on each wheel on one axle

The concept with one common engine has an advantage in lower total mass and lower costs for engine and power inverter. However an active differential gear is needed. Fig. 2 shows some gear structures for mechanic active differentials displayed as half-section. The evaluation of the best structure can be done by analyzing the package size, the mechanical effort, the controllability, the efficiency and the modularity. These mechanic gear structures include two brakes or clutches to control the torque distribution. In case of inactivity of AYC (typ. more than 90 % of time) these brakes or clutches are open and produce only a negligible power loss. With active brakes or clutches the power loss in these elements is more than 10 times[3] smaller compared with a system which uses only the wheel brakes.

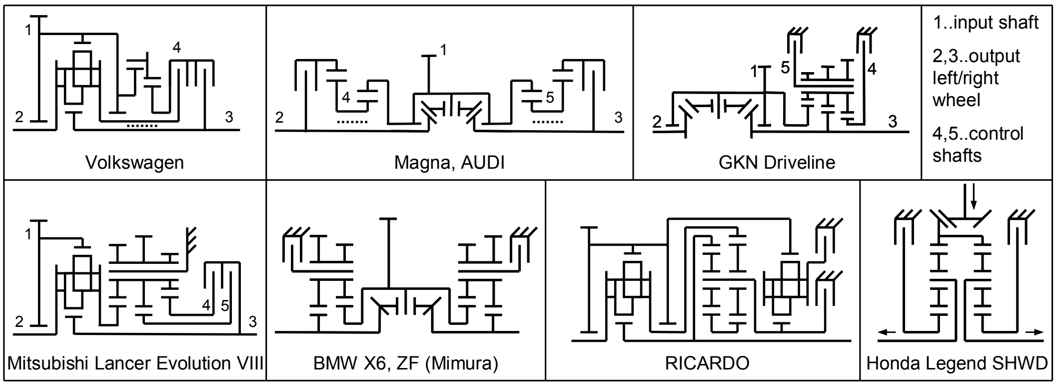

Fig. 2: Several gear structures for mechanical active differentials

2.2 Concepts for electric and hybrid vehicles

Hybrid cars and full electric vehicles have a high power electricity system. This can be used for active differentials too. The mechanic effort may become lower because instead of two brakes or clutches one small electric motor is used. Fig. 3 shows the TOYOTA-Prius hybrid gearbox including an internal combustion engine (ICE), two electric motors (E1, E2) and one planetary gear (I). The spur gear differential has been replaced by two small planetary gears (II+III), some periphery modules and an electric motor (E3). This electric motor could be very small because a power output of ca 2 kW is enough to get a wheel difference torque of 1000 Nm for many applications.

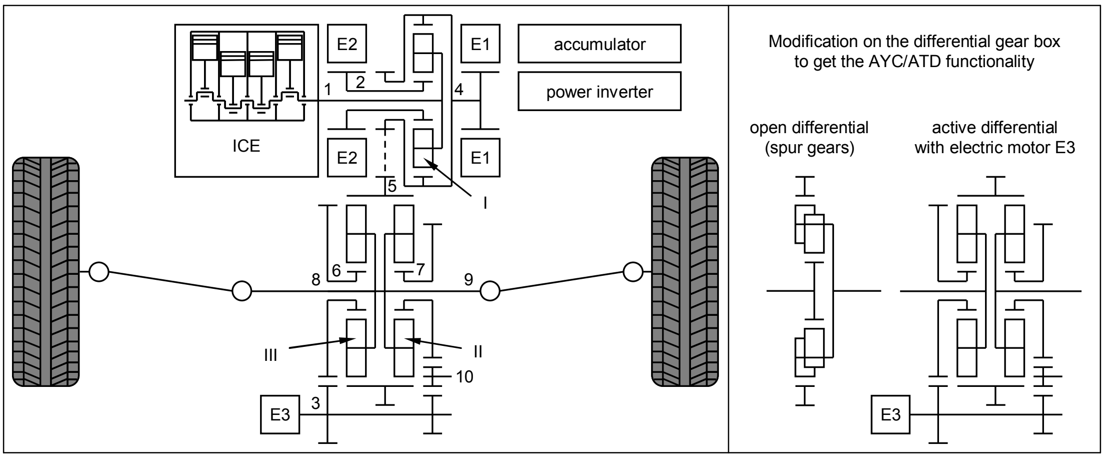

Fig. 3: Extension of the TOYOTA-Prius hybrid gearbox to get active torque distribution

3 depending on the mechanic design and the lowest corner radius, where AYC is possible

3 Mechanic gear structures

3.1 Analysis of planetary gears

Planetary gears are characterized by a high torque capacity. Therefore they are optimal for many vehicle applications. One characteristic value is the stationary ratio i_0, the ratio between the tooth numbers of ring gear and sun wheel. The relation between the speeds ω of sun wheel (S), ring gear (R) and carrier (C) is given by the WILLIS-equation (1). The accelerations α are calculated analogue (2). The torques of each shaft can be calculated based on the equations of equilibrium (4, 5) regarding the moments of inertia J. The stationary efficiency factor η_0 and the direction of the power flow *sig* describe approximately the effect of the power losses in tooth contact and bearings (3).

$$i_0 \cdot \omega_R + \omega_C \cdot (1 - i_0) - \omega_S = 0 \quad (1) \qquad i_0 \cdot \alpha_R + \alpha_C \cdot (1 - i_0) - \alpha_S = 0 \quad (2) \qquad sig = \mathrm{sgn}\left(T_S \cdot (\omega_S - \omega_C)\right) \quad (3)$$

$$i_0 \cdot \eta_0^{sig} \cdot (T_S - \alpha_S \cdot J_S) + (T_R - \alpha_R \cdot J_R) = 0 \quad (4) \qquad (T_S - \alpha_S \cdot J_S) + (T_C - \alpha_C \cdot J_C) + (T_R - \alpha_R \cdot J_R) = 0 \quad (5)$$

With these five equations every coaxial planetary gear can by analyzed. If several planetary gears are connected (like it is in active differentials) these equations have to be applied to every planetary gear with additional conditions for the connections. Because of the linearity the equations can be clearly written as a matrix equation.

3.2 Evaluation of gear structures

There are more than one million possibilities to design active differentials. Fig. 2 shows only a few one. All possible structures can be found by a systematic connection of the shafts of two or more planetary gears. To choose the right gear structure for a certain application the structures have to been rated. At first the package size is important. The radial dimension can be estimated based on the stationary ratio and the axial dimension based on the torque of the sun wheel. Therefore it is useful to define a maximum outer diameter and a minimum sun wheel diameter.

Secondly the mechanical effort can be estimated by counting the shafts, bearings and planetary wheels. In many cases the number of planetary wheels at a positive stationary ratio is twice of a negative stationary ratio. If one shaft is connected with two equal wheels of two different planetary gears the mechanical effort can be reduced. For example if the carrier of one planetary gear is connected with the carrier of another planetary gear there has to be only one carrier.

Thirdly the controllability of the system is decisive. Because all friction forces acts contrary to its relative speed the transmission behavior changes with the sign of the difference speed of left and right wheel, respectively cornering left or right. This effect can be evaluated by a calculation of the required torques at the actuators to get a certain output. If the required torque for the same output shows a big difference between cornering left and right the actuator has to be very fast to reduce the unintentional response of this discontinuous system behavior. Otherwise the controllability may be bad.

All these evaluation factors can be normalized and weighted to one mark so that a ranking is possible.

3.3 Example of design

Fig. 3 (right) shows a gear structure for active yaw control for a hybrid or full electric vehicle. The distinction between a primate engine for driving and a secondary engine for the unequal torque distribution results in a low system mass, a good controllability and a low system cost. This gear structure is characterized by a low mechanical effort because compared with a spur gear differential the ring gear must be enlarged and only one sun wheel and five spur gears have to be added. Even if the radial package size would be large it includes the final drive ration too. One disadvantage is the low speed of the additional electric motor E3 which could be increased by an additional gear.

4 Effect of active yaw control on the vehicle dynamics

4.1 The nonlinear two-track vehicle simulation model

There are many programs on the market to simulate the vehicle dynamics, e. g. MSC Adams/Car, CarSim, dSpace ASM for Matlab/Simulink etc. for complex vehicle models. Only few equations make it possible to simulate the principal dynamic of vehicles for a degree of freedom (DOF) of three. The actual dynamic state X_1 of the vehicle is characterized by its absolute speed v, the slip angle β and the yaw rate $\dot{\psi}$. The first derivation of this state can be calculated in relation to the forces FX and FY as well as the yaw torque MZ.

$$\dot{X}_1 = \begin{pmatrix} \dot{v} \\ \dot{\beta} \\ \ddot{\psi} \end{pmatrix} = \begin{pmatrix} \left(FY \cdot \sin(\beta) + FX \cdot \cos(\beta)\right)/m_v \\ \left(FY \cdot \cos(\beta) - FX \cdot \sin(\beta)\right)/m_v \\ MZ/J_z \end{pmatrix} \quad (6)$$

m_v … vehicle mass
J_z … moment of inertia along vertical axis

According to Fig. 4 the forces and the torque can be calculated with the tire forces and the air drag forces F_{Lx}, F_{Ly}.

$$\begin{aligned} FX &= \sum\left(F_{xi} \cdot \cos(\delta_i) - F_{yi} \cdot \sin(\delta_i)\right) - F_{Lx} \qquad \forall i \in (FR, FL, RR, RL) \\ FY &= \sum\left(F_{yi} \cdot \cos(\delta_i) + F_{yi} \cdot \sin(\delta_i)\right) - F_{Ly} \\ MZ &= \left(F_{yFR} \cdot \cos(\delta_{FR}) + F_{xFR} \cdot \sin(\delta_{FR}) + F_{yFL} \cdot \cos(\delta_{FL}) + F_{xFL} \cdot \sin(\delta_{FL})\right) \cdot l_V \ldots \\ &-\left(F_{yRR} \cdot \cos(\delta_{RR}) + F_{xRR} \cdot \sin(\delta_{RR}) + F_{yRL} \cdot \cos(\delta_{RL}) + F_{xRL} \cdot \sin(\delta_{RL})\right) \cdot l_H \ldots \\ &+\left(F_{xFR} \cdot \cos(\delta_{FR}) - F_{yFR} \cdot \sin(\delta_{FR}) - F_{xFL} \cdot \cos(\delta_{FL}) + F_{yFL} \cdot \sin(\delta_{FL})\right) \cdot b_F/2 \ldots \\ &+\left(F_{xRR} \cdot \cos(\delta_{RR}) - F_{yRR} \cdot \sin(\delta_{RR}) - F_{xRL} \cdot \cos(\delta_{RL}) + F_{yRL} \cdot \sin(\delta_{RL})\right) \cdot b_R/2 \end{aligned} \quad (7)$$

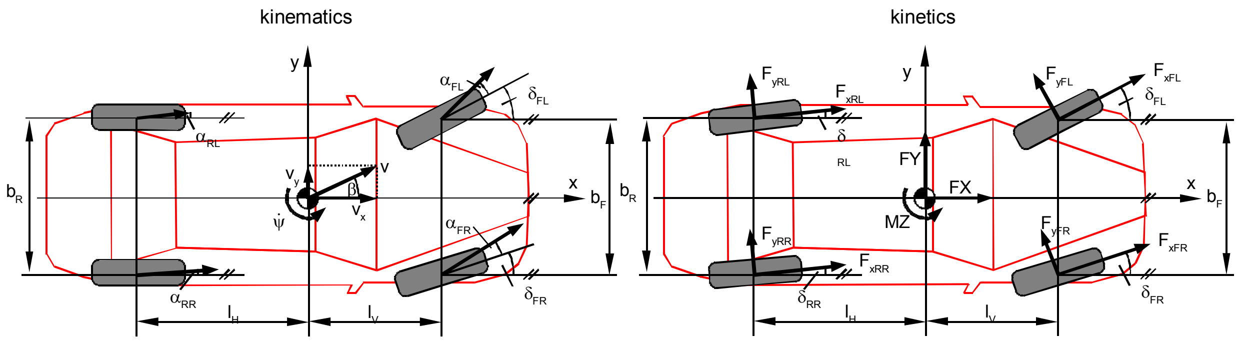

Fig. 4: Kinematics and kinetics of a two track vehicle model

These tire forces for each tire depend on the actual vertical tire load F_z, the longitudinal slip κ, the slip angle α and the local friction coefficient μ. A tire model like "Paceijkas Magic Formula" or the HSRI tire model provide that forces.

HSRI model:

$$F_{xi} = \frac{c_\kappa \cdot \kappa_i}{1-\kappa_i} \cdot \frac{s_{Ri} - 0.25}{s_{Ri}^2} \qquad F_{yi} = \frac{c_\alpha \cdot \tan(\alpha)}{1-\kappa_i} \cdot \frac{s_{Ri} - 0.25}{s_{Ri}^2} \quad (8)$$

$$s_{Ri} = \max\left(0.5\,;\, \frac{\sqrt{\left(c_\kappa \cdot \kappa_i\right)^2 + \left(c_\alpha \cdot \tan(\alpha_i)\right)^2}}{\mu \cdot F_{zi} \cdot (1-\kappa_i)}\right) \quad (9)$$

The vertical tire load can be estimated by neglecting of the damping effects depending to the longitudinal and lateral acceleration a_x and a_y. This estimation is based on the equations of equilibrium, a torsion stiff cassis, the stiffness of the wheel springs of front and rear axle c_F, c_R and the stiffness of the anti-roll-bar c_{SF}, c_{SR}.

$$\begin{aligned} F_{zFR/FL} &= m_v \cdot \left(l_H \cdot g - z_G \cdot a_x\right)/\left(2 \cdot l_R\right) \pm c_{ayF} \cdot m_v \cdot z_G \cdot a_y \\ F_{zRR/RL} &= m_v \cdot \left(l_V \cdot g + z_G \cdot a_x\right)/\left(2 \cdot l_R\right) \pm c_{ayR} \cdot m_v \cdot z_G \cdot a_y \end{aligned} \quad (10)$$

$$c_{ayF/R} = \frac{c_{F/R} + c_{SF/R}}{b_F \cdot \left(c_F + c_{SF}\right) + b_R \cdot \left(c_R + c_{SR}\right)} \quad (11)$$

The slip angles can be calculated according to the kinematics in Fig. 4 based on the actual state X_1.

$$\begin{aligned} \alpha_{FL/R} &= \delta_{FL/R} - \arctan\left(\frac{v \cdot \sin(\beta) + \dot{\psi} \cdot l_F}{v \cdot \cos(\beta) \mp \dot{\psi} \cdot b_F/2}\right) \\ \alpha_{RL/R} &= \delta_{RL/R} - \arctan\left(\frac{v \cdot \sin(\beta) - \dot{\psi} \cdot l_R}{v \cdot \cos(\beta) \mp \dot{\psi} \cdot b_R/2}\right) \end{aligned} \quad (12)$$

There are two possibilities to calculate the longitudinal slip κ. At first the speeds of the four wheels are described by four new degrees of freedom. The additional state is X_2. Its derivation $\dot{X}_2$ can be calculated based on Newton's law.

$$X_2 = \left(\omega_{FR} \quad \omega_{FL} \quad \omega_{RR} \quad \omega_{RL}\right)^T \qquad (13) \qquad \dot{\omega}_i = \left(T_i - F_{xi} \cdot r_{dyn}\right) / J_{wheel} \qquad (14)$$

Another possibility is to resolve the equations of the tire forces to the longitudinal slip. This leads to a quartic function $\kappa_i = f(T_i, \alpha_i, F_{zi}, \mu)$ that can be solved with high mathematic effort. The result is the stationary asymptotic value of the first method.

The accelerations a_x and a_y used in eq. 10 can be calculated from the actual state and its derivation from the last calculation step. The air drag forces used in eq. 7 results from the components of the vehicle speed, the air density ρ, the c_w-value and the characteristic surface A_w.

$$a_x = v \cdot \left(\dot{\beta} + \dot{\psi}\right) \cdot \sin(\beta) + \dot{v} \cdot \cos(\beta) \qquad (15) \qquad a_y = v \cdot \left(\dot{\beta} + \dot{\psi}\right) \cdot \cos(\beta) + \dot{v} \cdot \sin(\beta) \qquad (16)$$

$$F_{Lx} = \left(v \cdot \cos(\beta)\right)^2 \cdot \rho \cdot c_{wx} \cdot A_{wx} / 2 \qquad (17) \qquad F_{Ly} = \left(v \cdot \sin(\beta)\right)^2 \cdot \rho \cdot c_{wy} \cdot A_{wy} / 2 \qquad (18)$$

Now all necessary equations are given to calculate the derivation of the state based on the actual state X_1 (eq. 6). An integration algorithm like Runge-Kutta 4th order provides the results for every time step of the whole simulation. This simulation model can be extended by a more detailed drive train, a driver model, driver assistant systems, a model for engine, clutch, gearbox and differential gear, roll, pitch and vertical moving, single wheel displacement etc. Fig. 5 shows an exemplary vehicle movement for a double lane change with a position and state depending steering angle as input.

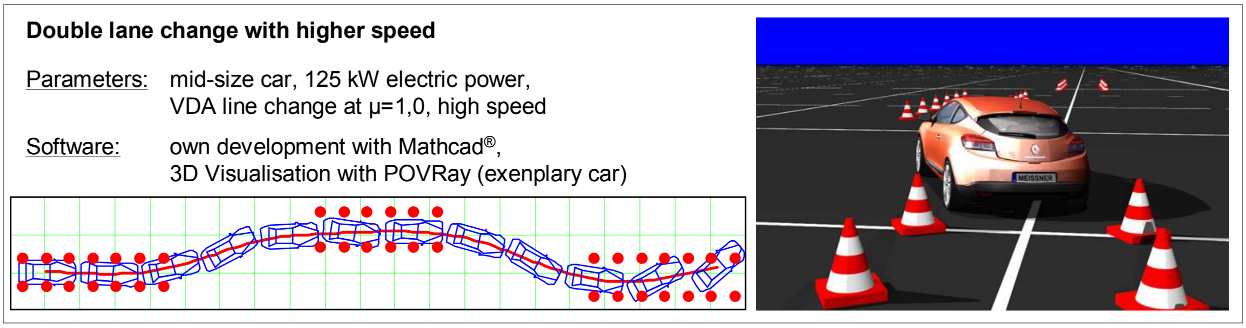

Fig. 5: Exemplary simulation of a double lane change with a high powered electric vehicle

4.2 Effect on safety

In some driving situations like high-speed-cornering, a suddenly evasive maneuver or a suddenly changing of the road surface the vehicle may become instable. Mostly an electronic stability program (ESP) brakes specific wheels to produce an yaw torque to stabilize the car. To get an intended yaw torque only one wheel per axle is braked because braking the other wheel would provide a contrary yaw torque. If the traction potential of this braked wheel is not high enough an accident is not avoidable.

But first of all on surfaces with low friction coefficients like a road with water, snow or ice, an active torque distribution can avoid this accident. There no longer only one traction contact is used but both traction contacts: one with a negative wheel torque (like braking) and one with a positive wheel torque. In common the stabilizing yaw torque with active torque distribution is almost up to twice of the yaw torque with ESP.

4.3 Effect on traction

Hill-starting may be problematical if one wheel is on snow, ice or grit (μ-split). The wheel with the lowest traction potential in combination with an open differential limits the whole drive torque. The vehicle can not start driving. In actual vehicle applications the wheel with the lowest traction potential is braked (electronic differential lock). Therefore the engine has to provide almost twice of the driving torque because one part is braked. If the engine is not powerful enough or the vehicle is equipped with a trailer it still can not start driving. Whereas with an active torque distribution the engine torque is not partly braked but only redirected to the wheel with the higher traction potential. The car starts driving. Fig. 6 shows the maximum hill slope at μ-split with μ_{high}=0.8 (wet road) and μ_{low}=0.2 (snow). It is very interesting that a car with front wheel drive and active torque distribution (ATD) reaches almost the same maximum slope than a more expensive car with all wheel drive and electronic differential lock (EDS) with the same engine in this situation.

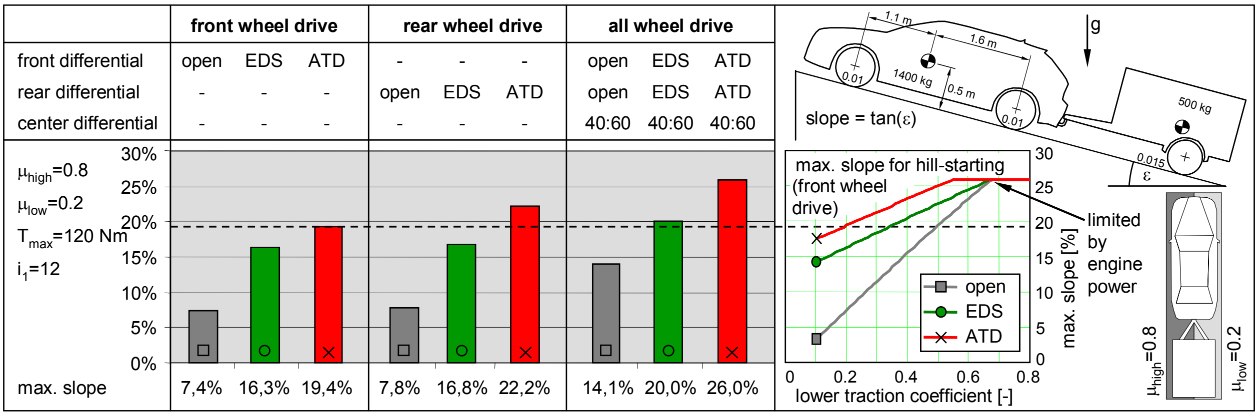

Fig. 6: Maximum slope of a hill-start at μ-split

4.4 Effect on agility

Every vehicle is an inertial mass. It follows the driver intension with a latency period. For a very sporty (agile) driving the vehicle mass and the inertia should be low. However the car-size increases in the last years. This various demands lead to a conflict of interests. AYC provides an additional yaw torque. This can be used to reduce the steering response time, and so the subjective impression of such a vehicle with AYC would be sportier and light-weight.

4.5 Effect on comfort

Because AYC influences the vehicle dynamics the self-steering effects can be manipulated. The AYC control algorithm of an over or under steering vehicle can be adapted to a neutral steering behavior. This may lead to a more comfortable driving.

4.6 Effect on efficiency

A yaw torque can be generated by braking one wheel. With it an enormous power loss occurs. Another possibility is given by mechanical active differentials (Fig. 2). Such systems are characterized by an amplification factor. That means either the torque or the relative speed in the brakes or clutches is about ten times smaller. Therefore the power loss is obviously smaller too. With active differentials which include an electric motor as actuator the power losses can be more reduced depending on the efficiency of the electric system. Additionally this electric motor acts in one direction as generator and in the other direction as motor. Storing the energy like a recuperation will increase the efficiency too.

Furthermore the longitudinal stiffness of a tire increases with the vertical load. That means the slip is being reduced. The power losses of a tire are the result of the drive torque and the slip. Therefore energy can be saved if the wheel with the higher vertical load (corner outside) gets the higher wheel torque.

4 Conclusions

Active yaw control is conventional realized by an activation of the wheel brakes. A method with higher efficiency and a better functionality is to use separate (electric) engines, separated but coupled engines or a single engine with an (electric) active differential. First of all the latter leads to a high efficiency of the whole drive train and e. g. for a front driven car with a medium engine to a driving performance in many situations which is comparable with an all wheel drive.

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