TORQUE VECTORING SYSTEMS TO IMPROVE ENERGY-EFFICIENCY AND VEHICLE DYNAMICS

A primary obstacle for current electric vehicles is the restricted range due to the low energy density of the storage media. Therefore, the Research Centre for Information Technology (FZI) in Karlsruhe is developing driving and operational strategies, which help to utilise the available energy in the most efficient way. The use of wheel-individual propulsion leads to the possibility of torque vectoring and thereby introduces new degrees of freedom to improve both vehicle dynamics and energy-efficiency. This work was awarded the Hermann Appel Price from IAV in 2013.
1 BACKGROUND

The introduction of electric machines for the sole or supporting propulsion of vehicles leads to new challenges for developers in the automotive industry. The dominating application of batteries as energy storage devices leads to strongly restricted range for battery electric vehicles. Current middle-class electric vehicles offer a range of around 150 km. Intelligent and adaptive operational strategies offer the prospect of increasing this range significantly. In contrast to driving strategies, they are transparent to the driver must not be learned and approved. Using a wheel-individual drive concept allows new options to create propulsive or decelerating force. While much research is done to improve vehicle dynamics through torque vectoring, only little focus has been placed on increased energy-efficiency.

Techniques considering Torque Vectoring will be analysed in the following and are based on two principles: Firstly, slipping of the tyres with little load during high longitudinal or lateral acceleration phases can be reduced by adaptive control. Secondly, the force for a small acceleration can be provided by only a part of the electric machines. Hence, declutching non-required machines leads to reduced losses in the powertrain and thus increases energy-efficiency.

2 VEHICLE MODEL

A complex and nonlinear four-wheel model is used to describe the influence of the wheel-individual distribution on vehicle dynamics. The vehicle is controlled by five values, which are set by a controller based on the driver inputs. The steering angle \( \delta \) of the front-axle wheels is directly controlled by the steering wheel and cannot be manipulated by the proposed control systems. The total propulsion torque \( T_{\text{tot}} \) is set by the pedal stroke and distributed to the wheels \( T_i \) by the controller according to Torque Vectoring strategies. The index \( i \) denotes front (f) / rear (r), \( j \) denotes left (l) / right (r). Given these control inputs, the differential equations eq. 1 of the vehicle velocity \( v \), eq. 2 of the sideslip angle \( \beta \) and eq. 3 of the yaw rate \( \psi' \) can be formulated. Shows the mentioned states and the relevant vehicle parameters.

\[
\dot{v} = \frac{1}{m} \left( \cos \beta \sum_{i} F_{x,i} + \sin \beta \sum_{i} F_{y,i} - k_0 - k_2 v^2 \right)
\]

\[
\dot{\beta} = \frac{1}{m v} \left( \cos \beta \sum_{i} F_{y,i} - \sin \beta \sum_{i} F_{x,i} \right) - \dot{\psi}
\]

\[
\dot{\psi} = \frac{1}{J_z} \left[ l_f \left( F_{y,fl} + F_{y,fr} \right) - l_r \left( F_{y,rl} + F_{y,rr} \right) + \frac{b_f}{2} \left( F_{x,fr} - F_{x,fl} \right) + \frac{b_r}{2} \left( F_{x,rr} - F_{x,rl} \right) \right]
\]
The vehicle dependent and constant parameters are: mass of the vehicle \( m \), roll and air drag coefficient \( k_\alpha \) respectively \( k_\omega \), inertia around the vertical axis \( J_z \), distance \( l \) between tyres and centre of mass \( CG \), as well as the track \( b \). The longitudinal and lateral forces in the tyre coordinate system \( t_{\text{long},y} \) respectively \( t_{\text{lat},y} \) result from the tyre characteristics, the current vehicle state and the torque applied to the wheel hub. A coordinate transformation allows to express the forces in the vehicle-fixed coordinate system \( F_{\text{long},y}, F_{\text{lat},y} \). The relation between force in the tyre’s longitudinal direction and the longitudinal slip \( s_{\text{long}} \) can be described by the magic tyre formula (eq. 4) [1]. The behaviour of the tyre is defined by the experimentally acquired parameters \( B, C, E \) and depends on the normal load \( F_y \) as well as the friction coefficient \( \mu \) of the street. The slip angle \( \alpha \) is determined by the Dugoff model [2] in relation to eq. 4.

As eq. 4 shows, the maximum force, which can be transmitted by the tyre to the street, and the resulting slip are strongly dependent on the wheel normal load. This load, on the other hand, is influenced by longitudinal and lateral acceleration. In a curve, the load is shifted to the outer wheels, while for deceleration manoeuvres the load is shifted to the front wheels. A thorough description which also explains the relation of the slip angles \( \alpha \) and the tyre velocity \( v_i \) is given in [3].

The nonlinear relations of the four-wheel model provide an accurate description for high longitudinal and lateral accelerations. However, despite the reduced accuracy, it is often meaningful to use the linearised single-track model to allow the application of linear control theory. The single-track model is derived by merging the wheels of each axle. The tyre characteristics for sideslip are linearised by introducing a constant gradient \( c_{\omega,\alpha} \). Extending the linear single-track model given in literature [4] by the yaw moment \( M_y \), created by torque vectoring as given in eq. 5, results in the linear model of eq. 6.

\[
F_{\text{long},y} = F_{\text{max}}(F_y, \mu) \sin \left[ C \arctan \left( B(1-E)s_{\text{long}} + E \arctan(Bs_{\text{long}}) \right) \right]
\]

\[
\begin{align*}
M_y &= \frac{h}{2} \left( F_{\text{long},r} - F_{\text{long},l} \right) + \frac{h}{2} \left( F_{\text{lat},r} - F_{\text{lat},l} \right) \\
\begin{bmatrix} \dot{\beta} \\ \dot{\psi} \end{bmatrix} &= \begin{bmatrix} -c_{\alpha,\omega} l_x - c_{\alpha,\omega} l_z - c_{\alpha,\omega} l_t - 1 & \frac{mv}{c_{\alpha,\omega} l_t - c_{\alpha,\omega} l_z} \\ -c_{\alpha,\omega} l_z - c_{\alpha,\omega} l_t & \frac{mv}{c_{\alpha,\omega} l_t - c_{\alpha,\omega} l_z} \end{bmatrix} \begin{bmatrix} \beta \\ \psi \end{bmatrix} \\
&\quad + \begin{bmatrix} c_{\alpha,\omega} l_x & 0 \\ \frac{mv}{c_{\alpha,\omega} l_t - c_{\alpha,\omega} l_z} & \frac{1}{J_z} \end{bmatrix} \begin{bmatrix} \delta \\ M_y \end{bmatrix}
\end{align*}
\]

3 TORQUE VECTORING SYSTEM

Within the scope of this study, three systems as shown in 2 have been implemented: one controller is based on the longitudinal slip of the tyres, one system distributes the tyre torque depending on the known normal tyre loads and one system is using a linear-quadratic (LQ) controller with subsequent torque distribution. The slip controller is based on the previously mentioned fact that the longitudinal slip will rise strongly on the inner wheels during a curve manoeuvre, if the propulsive force on the tyres is equally distributed. The torque provided at the wheel hubs is controlled individually by three PID controllers to minimise the longitudinal slip. There is a controller for the distribution \( \omega \) between the front- and rear-axle and one controller for the left-right distribution \( \rho \) on each axle.

A second approach is to set the torque of each electric machine with respect to the known normal load on each wheel. This system is encouraged by [5], in which the authors distribute the recuperation forces during situations with lateral acceleration. The tyre normal load can either be measured by sensors or estimated from the vehicle’s accelerations if the spring-damper dynamics of the car are known. These forces can be the input into a lookup-table, which is considering the normal load distribution as shown in eq. 7 and the ideal braking force distribution given in [4] to provide the optimal torque at the wheel hubs.

\[
\rho_i = \frac{F_{\text{long},i}}{F_{\text{long},t} + F_{\text{long},r}}
\]

The third control system consists of a reference generator, a LQ setpoint tracking controller and a subsequent torque distribution. The system is based on [6], but does not introduce a feedforward control. Additionally, the torque distribution block is enhanced to calculate the introduced distribution factors \( \omega \) and \( \rho \) depending on the optimal machine efficiency, the desired yaw torque and safety constraints. As an input the system receives the control signals in terms of the steering wheel angle, the pedal strokes and the observed vehicle state as given in the four-wheel model.

The reference generation, as well as the LQ setpoint tracking controller utilise the extended linear single-track model (eq. 6). To allow a gain-scheduling approach, the system is linearised for the permissible velocity range of the vehicle. By taking a linear-quadratic cost function into account, the controller minimises both the deviation of the states sideslip angle and yaw rate and furthermore the required yaw torque. By means of this control variable, it is possible to keep the response of the vehicle near that of the linear model, which is assumed to be ideal. While this primarily improves the dynamics of the vehicle, the energy consumption can be reduced compared to the uncontrolled vehicle in certain situations.

The bulk of the energy-efficiency achieved by this system, however, is a result of the torque distribution. To generate the desired acceleration as efficiently as possible, the torque will be distributed by considering the efficiency characteristics of the electric machines. Given a drive torque \( F_{\text{long},i} \) and a yaw torque \( M_y \), calculated by the precedent controller, the factors \( \rho \) respectively \( \omega \) can be determined. Combining the torque distribution, which is achieved by those factors and eq. 5, it can be seen that each distribution factor can be expressed in terms of the other two. Eq. 8 calculates
the left-right distribution at the rear tyres, given the track \( d \) on front- and rear-axle is equal and the tyre radius \( r \) is known.

\[
\rho_f = \begin{cases} 
\frac{(M_r r)/(T_{rot} b) + \sigma \rho_f - 0.5}{\sigma - 1} & \text{if } \sigma \neq 1 \\
0.5 & \text{if } \sigma = 1 
\end{cases}
\]

With the remaining degrees of freedom \( \omega_f \) and \( \omega_r \), the cost in terms of power can be calculated for a requested \( T_{tot} \) and \( M_z \) given the current state considering the four-wheel model. Illustrates the required power, which will be consumed to create an exemplary propulsion and yaw torque. It is obvious that significant gain can only be achieved at the borders of the plot, hence, when at least one electric machine is shut off. By examination of a multitude of possible scenarios, one can see that this is the general case. This motivates abstaining from the solution of a complex and non-convex optimisation problem. Therefore, a torque distribution heuristic as depicted in is proposed. To determine the energy-optimised torque distribution, the required power when using two, three or all electric machines is calculated under consideration of the four-wheel model. The machine combination that is employed depends on whether the requested torque is accelerating or decelerating and on the direction of the yaw torque. The combination leading to the lowest power requirement is chosen. To ensure safety, the currently acting lateral and longitudinal acceleration are taken into account additionally. If necessary, all four machines may be used despite the higher power demand.

4 RESULTS

The results are generated by the vehicle dynamic simulation software IPG CarMaker [7]. To generate an experimental platform, the standard vehicle DemoCar, delivered by the simulation tool, is equipped with wheel-individual electric machines. The parameters for the single-track model of the vehicle are shown in [8], however, for a complete description of the four-wheel model's parameters we refer to IPG CarMaker. According to the three Torque Vectoring proposals, four systems are available: reference vehicle with equal distribution (a), slip controller (b), normal load proportional distribution (c) and the dynamics controller with optimised distribution (d). To evaluate the systems in various situations, four scenarios were selected. To analyse the steering behaviour, steady-state circular tests (ISO 4138) and a manual curve manoeuver are simulated. Furthermore, the performance in extreme situations is examined during a simulated drive on the Hockenheim racing track. To

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**Block diagrams of the presented torque vectoring systems**

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1. **Slip controller**
2. **Normal load proportional distribution**
3. **Dynamic controller with optimised torque distribution**
also acquire an impression of which advantages we can expect in standard driving situations, the Artemis rural driving cycle [8] is assessed. It can be shown that the relevant states longitudinal slip, sideslip angle, acceleration and yaw rate can be observed or measured. For the simulations illustrated within this paper, the variables are assumed to be ideally known.

The required power for a given propulsion force of 1.2 kN and a yaw torque of 400 Nm at a velocity of 80 km/h is shown in the diagram. The reference vehicle (a) shows understeering behaviour from 7 m/s² onwards. The slip controller (b) follows the reference vehicle’s behaviour for low lateral acceleration, keeps a constant gradient and shows slight oversteering when the lateral acceleration exceeds 8.5 m/s². The normal load distribution is illustrated in the flowchart. The heuristic for optimised torque distribution involves calculating the power for a given propulsion force and yaw torque. The process involves checking safety criteria and choosing the cheapest power from the allowed machine combinations.
proportional distribution (c) supports the curve driving for low lateral acceleration from 6 m/s² onwards. Strong oversteering characteristics appear when the acceleration of 8 m/s² is exceeded. At 8.5 m/s² the system begins to become instable and the curve driving cannot be maintained. The dynamics controller (d) leads to the desired linear behaviour and only starts to show understeering behaviour at high lateral acceleration. When considering the energy efficiency, the graph shows that significant gain can only be achieved when exceeding 6 m/s². At low levels b and c act even counterproductive.

The steady-state curve test observes the steering behaviour when all dynamic processes have settled. To evaluate the dynamic response during curve driving, a manual manoeuver is defined. As shown in the upper two graphs in ⑦, the vehicle decelerates with a constant acceleration while approaching the apex of the curve. After the apex is passed, a constant torque is applied to reacquire
the initial velocity. Both yaw rate and sideslip angle indicate that the dynamic controller (d) provides the most harmonic behaviour of all tested vehicle configurations. The worst results are delivered by the normal load proportional distribution, which leads to very asymmetric states when comparing the deceleration with the acceleration phase. Combined with the insight acquired by the steady-state curve test, it becomes clear that the dynamic controller leads to the most advantageous response.

Based on the previous results, only the dynamic controller and the reference vehicle will be compared in the next scenarios. In the case of the Hockenheim racing track, the IPG CarMaker driver can be configured with a maximum acceptable lateral acceleration. If the driver accepts 9.5 m/s² of lateral acceleration for the drive on the track, the vehicle with Torque Vectoring has a lap time of 79.94 s whereas the reference vehicle requires 81.29 s. With a total of 99.81 % compared to the reference vehicle, Torque Vectoring does not significantly reduce the energy consumption. Though, if the lateral acceleration for the Torque Vectoring is reduced to 9.1 m/s², the driver achieves a lap time of 81.07 s. The lap is completed with only 83.85 % of energy compared to the reference vehicle while the lap time is still slightly better. Comparison of equal distribution with the proposed torque distribution on the Artemis rural driving cycle
ence vehicle must still brake to avoid instability (see green force vectors in longitudinal direction of the tyres).

Finally, the reference vehicle and the vehicle with dynamic controller and optimised torque distribution are evaluated for the Artemis rural driving cycle. The simulation aims to give an impression of the potential of the proposed system for a typical rural drive with very little lateral acceleration. Figure 9 shows both the Artemis rural speed profile (top) and a comparison of the combined efficiency of all four machines. As the accelerations during the rural drive are moderate, it suffices to only use two electric machines most of the time. Since in such cases the torque distribution decouples two of the four machines and only uses the remaining to create the propulsion force, the overall friction in the powertrain is reduced. The results for the simulated Artemis rural cycle show that the energy consumption for the drive is reduced by 8.7%.

5 SUMMARY AND OUTLOOK

The simulations of the presented scenarios showed that the Torque Vectoring on basis of the dynamic controller and the optimised torque distribution introduce significant advantages both in situations of high lateral acceleration, as well as in more common driving scenarios. Future work includes the integration of the algorithms in a real prototype, the analysis of the behaviour for low friction surface and the influences of uncertainties when estimating sideslip angle and measuring yaw rate. Also an interesting aspect is the development of an interposed filter between reference generator and LQ setpoint tracker to further improve the dynamic behaviour.

REFERENCES


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