

Steering Torque Disturbance Rejection

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ABSTRACT

The layout of a vehicle steering system has to resolve a compromise. While it is important for lateral vehicle control to feel steering torque feedback of lateral tire to ground interaction, disturbing forces shall not be present in the feedback steering torque. These disturbing forces result from road irregularities, wheel rotor imbalance, suspension asymmetry caused by production tolerances, wear or impacts, and additional vehicle internal forces, e.g. the steered wheels also driven by the engine or braked. In general these disturbances are reduced by an optimization of the suspension geometry to decrease the impact of the unintended forces on the steering system. The remaining disturbance is controlled to an acceptable level via force feedback sensitivity calibration of the steering system, what in return influences the intended driver sensitivity to feel lateral tire forces.

Modern electrical assisted steering systems allow overlay of additional assist torques to compensate for disturbance torques. This paper describes a function intended to counteract disturbances resulting from longitudinal tire forces, usually called Torque-Steer during acceleration and Brake-Pull under deceleration. The developed system identifies the disturbances and decides on the appropriate amount of compensation torque, which is applied by the steering system. To limit the cost of implementation Steering Torque Disturbance Rejection (STDR) function has been realized in software with emphasis on not requiring sensors on top of what is already commonly available in today's vehicles.

STDR consists of two models calculating rackforce. While a vehicle based model is used to determine the force suitable for the driving situation, a steering system based model ascertains the force currently present. The presence of Torque-Steer and Brake-Pull is detected by comparing both forces during substantial acceleration or deceleration. The disturbance rejection torque request is based on the difference of both calculation models.

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INTRODUCTION

Importance of Steering Torque Feedback

The steering torque felt is one of the most important information to allow a vehicle driver lateral control. Although most of the information to build the driving path is based on visual information, the haptic feedback of the steering wheel to inputs is much faster than the delayed vehicle reaction can be seen. If the steering torque builds consistently on the lateral forces of the front suspension this helps the driver to understand upfront, what the reaction and path of the vehicle will be. To allow precise display of lateral tire forces at the steering wheel a steering system with minimized internal friction is required. Especially for electrical assisted systems emphasis needs to be placed on mechanical optimization. Unfortunately the intended superior sensing of lateral tire forces goes hand in hand with unintended sensitivity for erroneous disturbance forces. These result from road irregularities, wheel rotor imbalance (Nibble), suspension asymmetry caused by production tolerances, wear or impacts (Pull), and additional vehicle internal forces, e.g. the steered wheels also driven by the engine (Torque-Steer) or braked (Brake-Pull).

The Impact of Driving and Braking Torque on Steering Feedback

While Nibble and Pull disturbances seem to be decreasing due to improved manufacturing quality and robust designs, Torque-Steer and Brake-Pull disturbances take precedence with the trend to stronger engines and large, low aspect tires in the automotive industry.

Dornhege [1] and Frantzen [2] describe background and mechanical root cause of Torque-Steer. Frantzen follows his structured analysis by the Revo-Knuckle approach (Figure 1) to minimize the influence by change of front suspension geometry. As an update to a McPherson design the Revo-Knuckle Concept splits the upright into an inner- and an outer part. The rotation of the outer knuckle around the McPherson steering axis formed by Top Mount and Lower Ball Joint is prevented by connection to the stabilizer bar. Two additional pivot points connecting inner and outer knuckle form a new steering axis much closer to the wheel center. The reduced King Pin Offset, which is the lever arm from acceleration wheel forces to the steering axis, strongly improves robustness against longitudinal force

asymmetries and therefore improves Torque-Steer performance. Drawback of this front suspension concept is a negative impact on weight and cost.

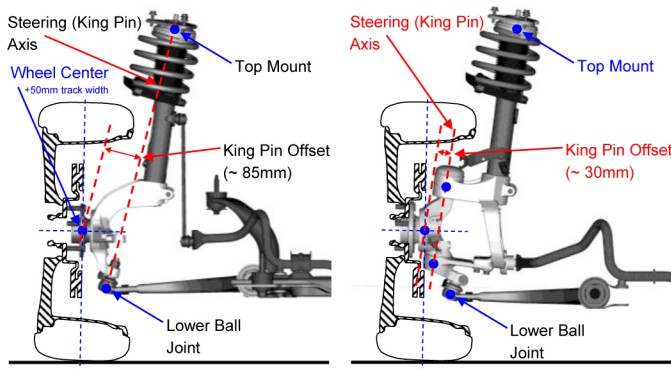


Figure 1. McPherson (left) and Revo Knuckle Concept (right) according to Frantzen [2]

Although internal design parameters of Torque-Steer robustness are understood a residual disturbance cannot be excluded, as the geometric optimization is bounded by cost, weight and package constraints. Furthermore uneven ground can move the midpoint of the tire to road contact patch laterally. This will lead to asymmetric driving torque application and finally to residual torque disturbing the driver feedback torque.

Opportunity of Corrective Functions with EPAS

The nature of electric power assisted steering systems (EPAS) allows overlaying disturbance rejection forces on the usual steering assist. Figure 2 identifies two potential interfaces to be used. On both interfaces software functions embedded in the EPAS control algorithm can request torque offsets. The Hand-Wheel-Torque-Interface (HTI) reacts to a torque request in a similar way, as if the driver would input that amount of torque at the handwheel. Therefore the requested torque offsets the measured value from the Torsion-Bar-Sensor parallel to applying the torque directly at the assist motor to simulate the mechanical torque from the driver. The Pinion-Torque-Interface (PTI) on the other hand applies the requested torque directly at the motor, thereby simulating the application of rackforce.

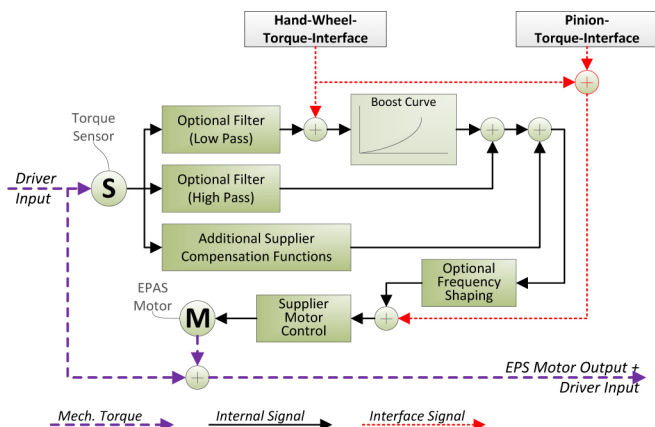


Figure 2. Torque Interfaces in EPAS Boost Curve Concept

FUNCTION DEVELOPMENT

Approach

The basic idea of the disturbance rejection function presented in this document is to calculate forces in the steering rack using several methods. One rack force is based on overall vehicle data, depicting the overall driving situation, serving as reference for the current driving situation. This will be called the reference rack force (RRF) in the following text. A second rack force calculation bases on steering system data, representing the current state of the system. This will be called the observed rack force (ORF). Both models must match the physical rack force sufficiently close in vehicle driving situations without disturbance. While in situations with steering torque disturbance this influence will be reflected in the ORF, it will not be seen in the RRF. Therefore in situations, where a disturbance is likely, e.g. high propulsion torque requested, the disturbance rejection algorithm will become active and request the difference of RRF and ORF as disturbance rejection torque. As the function is intended to work as embedded software in an automotive steering control unit, the execution time for calculations has to be reduced to the absolutely required level. Even if better approaches are known to solve partial problems, simpler solutions have to be chosen for efficiency reasons.

A similar function has been developed before [3], known as Torque-Steer-Compensation (TSC). In difference to the Steering Torque Disturbance Rejection (STDR) function described here the RRF was converted to a reference steering wheel torque by an inverted steering assist function and the difference to the measured column torque used for disturbance rejection.

Reference Rack Force

Background for the described calculation of the reference rack force is the finding that for the majority of driving situations rack force is basically linearly dependent on vehicle lateral acceleration. Unfortunately the vehicle lateral acceleration is significantly phase delayed to the rack force. To calculate a representative vehicle lateral acceleration signal without delay a simplified vehicle model is used. As Torque-Steer is a phenomenon mostly recognized during mainly linear driving situations with minor steering angle change a decision was made to use a linear representation. Isermann [4] derives from the single track model equations the reference yaw rate:

$$\dot{\psi}_{Ref} = \frac{v_x \cdot \delta}{l \cdot \left(1 + \frac{v_x^2}{v_{ch}^2}\right)} \quad (1)$$

Next to input signals vehicle speed (v_x), front wheel steer angle (δ) and vehicle parameter wheelbase (l) the only calibration value is the characteristic vehicle speed (v_{ch}). With the simple relations of constant circle driving follows the reference lateral acceleration:

$$a_{y,Ref} = \frac{v_x^2 \cdot \delta}{l \cdot \left(1 + \frac{v_x^2}{v_{ch}^2}\right)}$$

The value derived from equation (2) now forms the linear basis for RRF calculation with the tunable gain k_1 .

$$RRF_1 = k_1 \cdot a_{y,Ref} \quad (3)$$

In an additional calculation step vehicle geometry and mass distribution is used to derive a vehicle side slip angle from rear cornering stiffness. Combined with the steering angle a front tire side slip angle α_f is calculated. It is further assumed, that the tire will provide a self-aligning torque, which is proportional to the tire slip angle. In a further simplification the transfer of this tire aligning torque into a rack force is assumed to be proportional in the steering angle range, where Torque-Steer occurs. This leads to the secondary portion of the rack force calculation:

$$RRF_2 = k_2 \cdot \alpha_f \quad (4)$$

The front axle load of the vehicle will remain in a well-defined range between empty vehicle and maximum loading specification. This allows finding a suitable value for the tuning factor k_1 of equation (3). But the fact that customers will replace tires with aftermarket tires of unknown specification, e.g. different size, manufacturer, summer vs. winter or all season, we can't find a suitable compromise for k_2 . Therefore this tire relevant parameter is adjusted during operation to minimize the distance of RRF and ORF. While this parameter adaptation helps to adjust the model as long as no disturbance is present, it would convert to wrong values during disturbance situations. Therefore this parameter update is suppressed during active phases of the disturbance rejection algorithm.

The finalized reference rack force is calculated as:

$$RRF = k_1 \cdot a_{y,Ref} + k_2 \cdot \alpha_f \quad (5)$$

During driving manoeuvres RRF_1 and RRF_2 show similar behavior with slightly different phasing and different amplitude dependent on the driving situation. With proper selection of k_1 also the weighting of both portions is influenced.

Current Rack Force

To calculate the rack force that is physically present at the steering system using a disturbance observer an appropriate steering gear model is required. Based on a complex model including linkages, rims and tire to road contact patch a simplified model is derived.

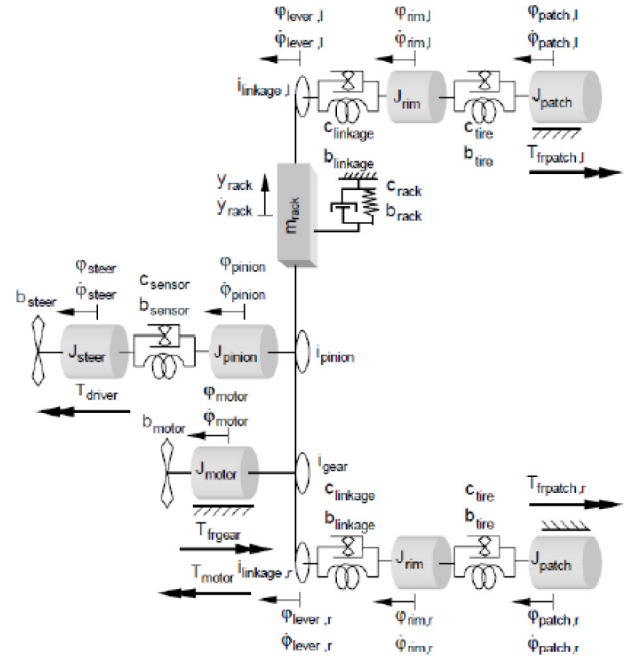


Figure 3. Complex Steering system model, Henrichfreise [5]

As described the approach is rack force reference based. Hence the complex steering system model is reduced to a rack force based model. All external forces acting at the lower end of the steering system directly to the tie rods are combined to F_{Rack} .

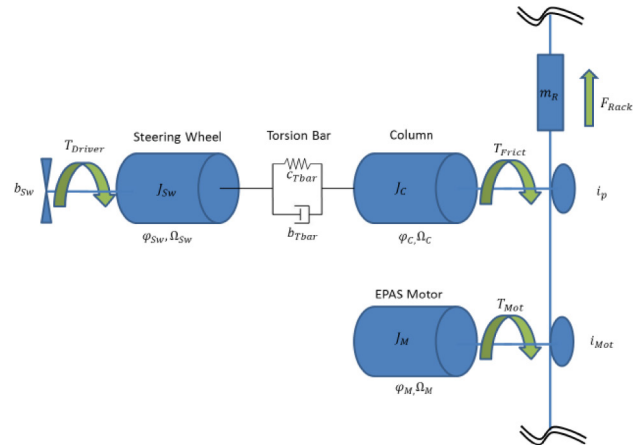


Figure 4. Reduced three-mass system model

Since the stiffness between EPAS motor and steering column is modeled as infinite the masses can be combined by mechanical rules [6,7] to one single mass using equations (6)-(9).

$$J_{driven} = J_{drive} i_{gear}^2 \quad (6)$$

$$J_{C_{Red}} = J_{Col} + i_{EPAS}^2 J_{Mot} + \frac{m_{Rack}}{i_p^2} \quad (7)$$

$$i_{EPAS} = \frac{i_{Mot}}{i_p} = \frac{i_b i_{belt}}{i_p} \quad (8)$$