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Design and control of model based steering feel reference in an electric power assisted steering system

T. Chugh, W. Chen, M. Klomp & S. Ran
Volvo Car Corporation, Gothenburg, Sweden

M. Lidberg
Chalmers University of Technology, Gothenburg, Sweden

ABSTRACT: Electric Power Assisted Steering (EPAS) system is a current state of the art technology for providing the steering torque support. The interaction of the steering system with the driver is principally governed by the EPAS control method. This paper proposes a control concept for designing the steering feel with a model based approach. The reference steering feel is defined in virtual dynamics for tracking. The layout of the reference model and the control architecture is discussed at first and then the decoupling of EPAS motor dynamics using a feedback control is shown. An example of how a change in steering feel reference (as desired by the driver) creates a change in steering feedback is further exhibited. The ultimate goal is to provide the driver with a tunable steering feel. For this, the verification is performed in simulation environment.

1 INTRODUCTION

The automotive industry is in a transformation where software and electronics are revolutionizing the engineering behind the vehicles. This is particularly true for steering systems, which have developed from passive mechanical to EPAS, now enabling the advanced driver support systems and the evolution towards fully automated driving. With all these benefits as well as reduced fuel consumption, EPAS systems have an inherent downside in a loss of steering torque feedback largely due to the effects of motor dynamics (Harrer and Pfeffer, 2017). For these shortcomings, there are additional steering functions as requirements, namely active damping, etc., for improving the steering feel. As a result, several control methods have been implemented over the years. For instance, (Bröcker, 2007) has implemented the desired functions within a motor torque controller as feedforward. This has been a common approach in recent years. With a feedforward approach it is difficult to give the driver a genuine steering feel since very less feedback information regarding the vehicle state is used. The aim of the proposed control strategy in this paper is twofold; to decouple the system from EPAS motor dynamics and to provide a desired tunable steering feel based on reference. The concept of steering feel reference is similar to the steering feel design in either steer-by-wire vehicles or driving simulators where the complete steering feedback is artificial (Balachandran and Gerdes, 2015), (Fankem and Müller, 2014). In this context, the reference is model based and the target is to emulate it. (Lee et al. 2016) has presented an algorithm for reference steering feel tracking with a different control strategy (torque control) and the reference is derived from the experimental data rather than model based. A simple example of how a change in steering feel reference causes a subsequent change in steering feedback of a decoupled system is discussed. This will exhibit the notion of tunable steering feel with position/velocity control. However, the paper will not include how and which feedback component from the vehicle state is used in the reference model because it requires separate discussion.

In this paper, section 2 will introduce three sub-systems which combines to form a complete model. The controller development is segregated in two parts—Section 3 covers steering feel reference model design followed by the controller layout in section 4. Lastly, the initial result of this concept highlights the potential of design to feedforward-feedback approach for steering feel.

2 SYSTEM MODELLING

The steering system of a vehicle comprises of three main sub-systems; mechanical steering unit, EPAS servo motor (rack assisted) and tires (representing vehicle), refer Figure 1. The input to the system is steering wheel torque (SWT) M_S from the driver, steering rack force $F_{Rack,veh}$ obtained from the vehicle model (and tires) and EPAS assist force $F_{Rack,EPAS}$ from the servo motor. The output is all other states including steering wheel angle (SWA) δ_S, torsion bar torque M_{TB}, rack displacement x_{Rack}, etc. The entire system is assumed to be linear for the analysis.

2.1 *Steering model*

A 2-DOF steering model has been considered. The steering wheel inertia J_S is coupled to rack mass m_{Rack} via torsion bar (TB) compliance $\left(\text{and gear ratio } i_{R/P}\right)$. No friction element is included. The Equation 1 represents the force balance at the steering rack.

2.2 *Vehicle model*

A single track model is used to include the lateral dynamics through steering rack force. It has been derived using front axle lateral force (or wheel steer torque) as shown in Equation 2:

$$\left(m_{Rack} + i_l^2\, J_{Whl}\right)\ddot{x}_{Rack} + k_{Rack}\,\dot{x}_{Rack} = M_{TB}\, i_{R/P} + F_{Rack,EPAS} - F_{Rack,veh} \tag{1}$$

$$F_{Rack,veh} = C_{\alpha F}\left(x_{Rack}\, i_l - \left(v_Y + l_F\,\dot{\psi}\right)/\,v_X\right) n_T\, i_l \tag{2}$$

where i_l–wheel rotation to rack displacement ratio; J_{Whl}–wheels inertia; k_{Rack}–rack damping; $v_X\,/\,v_Y$ –longitudinal/lateral velocity; $\dot{\psi}$ –yaw rate; n_T–combined pneumatic and caster trail; $C_{\alpha F}$–front axle cornering stiffness; and l_F–distance between the center of gravity and front axle.

2.3 *Electric motor model*

The electric motor comprises of various compliance elements as shown in Figure 1. The inertia J_{Mot} and viscous damping b_{Mot} of the servo motor are considered without any stiffness. The actual

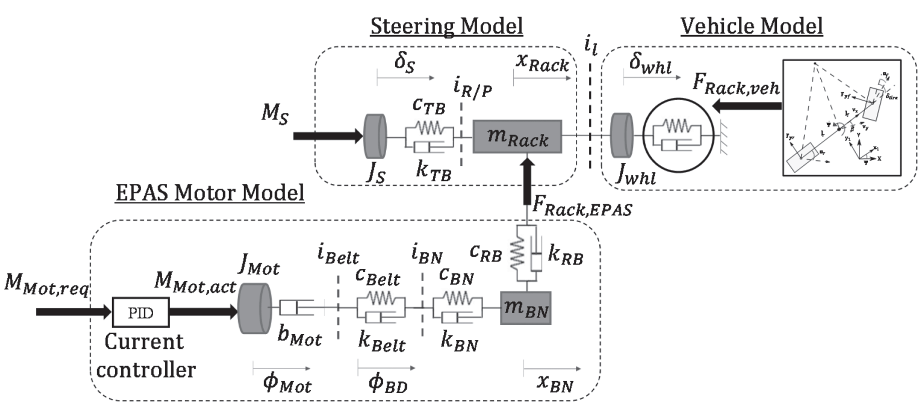

Figure 1. Steering system model with its description.

torque $M_{Mot,act}$ is converted to rack assist force via belt drive (BD), ball nut (BN) and recirculating balls (RB). Motor inertia determines the mechanical bandwidth. Higher inertia results in lower bandwidth. Therefore, a current controller is integrated to ensure the torque request $M_{Mot,req}$ is obtained with fast dynamics (Dannöhl et al. 2007). If this condition is true, then the simplified motor model results in Equation 3, implying a twofold effect of inertia—determining the bandwidth and creating a dynamic rack mass and damping. Also, the effect of elasticities determine the motor gear ratio i_{Mot} bandwidth which is relatively higher (~120 Hz). Hence the ratio is considered static (a rigid connection between the steering rack and motor), but the transfer function from request to actual motor torque is given by a time delay.

$$F_{Rack,EPAS} = i_{Mot}\, M_{Mot,act} - i_{Mot}^2\, J_{Mot}\, \ddot{x}_{Rack} - i_{Mot}^2\, b_{Mot}\, \dot{x}_{Rack} \tag{3}$$

3 STEERING FEEL REFERENCE MODEL

The need of steering feel reference is to create an artificial steering feedback. The model consists of virtual dynamics and basic functionalities which already exist in conventional EPAS for the purpose of stability and desired feel. The rack dynamics balances out the motor assist and driver input (approximated by torsion bar torque). The critical difference between an EPAS and a mechanical system is the inertial effect of servo motor which can be viewed as an equivalent rack mass (~1500 kgs.) in Equation 4 (from Equation 1 and 3). The passive response of the rack is equivalent to a low pass filter and the bandwidth of useful road information is decreased (deteriorated feedback) because of higher dynamic rack mass. The bandwidth theoretically can be shifted higher by compensating the motor inertia. This part is covered in section 4.1 but the main discussion point is steering feel design and not rather altering the steering system bandwidth.

$$\left(m_{Rack} + i_l^2\, J_{Whl} + i_{Mot}^2\, J_{Mot} \right) \ddot{x}_{Rack} + \left(k_{Rack} + i_{Mot}^2\, b_{Mot} \right) \dot{x}_{Rack} + F_{Rack,veh} = M_{TB}\, i_{R/P} + M_{Mot,act}\, i_{Mot} \tag{4}$$

The reference model is also based on Equation 4 with virtual parameters; an equivalent mass-spring-damper. The driver input in terms of torsion bar torque from the sensor generates the reference behavior. In principle, the reference model parameters have been designed in simulation environment with different steering functions (mentioned below) that fulfill typical requirements.

3.1 *Basic steering assistance*

This function supports the driver by reducing the steering effort with a vehicle speed dependent gain (non-linearly) proportional to torsion bar torque under different scenarios. The motor torque assist is defined in Equation 5. With increasing gain, the steering dynamics is also affected and the overall damping is reduced, refer data 1, 2 and 3 in Figure 2. The steady state value increases with the gain as expected, but the overshoot is also increased around 1 Hz (vehicle yaw eigenfrequency) due to lower damping. This is due to higher relative steering

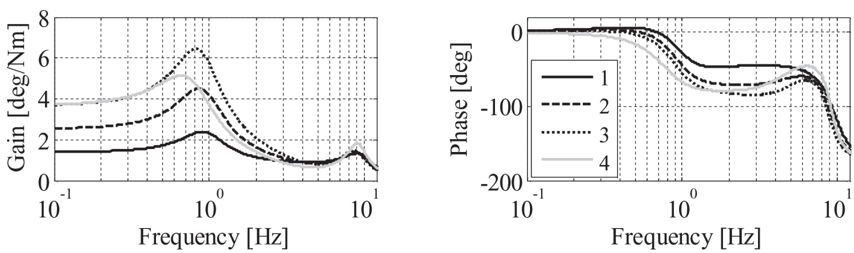

Figure 2. SWA/SWT frequency response at v_X = 100 km/h. 1–No assist, 2–0.05 EPAS gain, 3–0.10 EPAS gain and 4–0.10 EPAS gain with active damping.

rack stiffness with an increasing assist. The effect of lower damping can be counteracted by introducing active damping.

$$M_{Mot,BSA} = k_{EPAS}\left(v_X\right) M_{TB,act} \tag{5}$$

3.2 *Active functions*

Active damping imparts more system damping for both improved stability and steering feel. As described earlier the steering rack acts as a filter between the driver and road, therefore the control of the rack state is essential for a desired steering feedback. This function acts against the rack movement (refer Equation 4) to increase the overall rack damping. The active damping torque demand (in Equation 6) is a function of rack speed and proportional to velocity dependent damping coefficient k_{AD} and hyperbolic function of vehicle speed. The effect can be seen in Figure 2. With active damping, SWA has lower overshoot (data 3 and 4 with same assist gain). As a result, an improved phase lag steering response is achieved at lower frequencies for a desired steering feel. Similarly, an active return function has been implemented for self-centering of the steering wheel during its release. Both these functions are desired steering characteristics.

$$M_{Mot,AD} = -k_{AD}\left(v_X\right)\ \tanh\left(\propto_{v_X}\ v_X\right)\dot{x}_{Rack,ref} \tag{6}$$

4 STEERING CONTROL ARCHITECTURE

The control layout is shown in Figure 3, a conventional 2-DOF controller (Åström and Murray, 2008) where the feedforward part controls the driver's command directly and the feedback controller reacts to disturbances. The virtual model provides the reference rack state (position and velocity). The overall concept is based on rack position/velocity control. The basic steering assistance (also as a feedforward) is directly sent to the motor torque request because of two reasons; to achieve a lower steady state error in rack state and lower phase delay in steady state steering response. However, the target is to completely eliminate this direct assistance and depend purely on feedforward and feedback control. This will be investigated in the later work.

4.1 *Feedforward control*

The feedforward control output is given in Equation 7 (in Laplace domain). In general, the transfer function $G_{FF}(S)$ is derived from the inverse of model dynamics. The dependency on

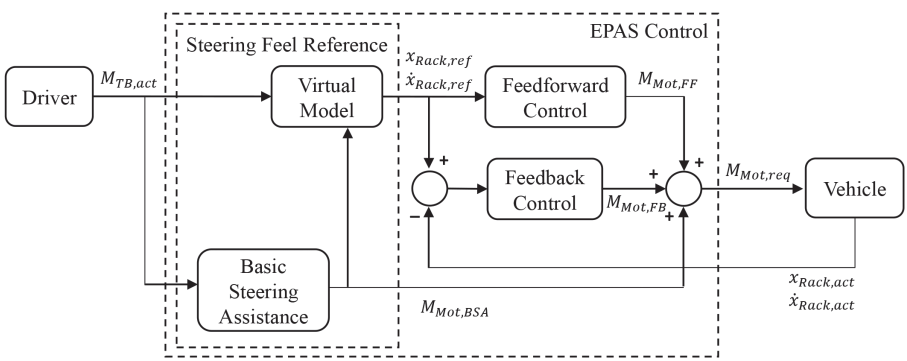

Figure 3. EPAS control layout.

this part is very limited in the present controller and as a result, a constant gain is set as a function of vehicle speed to fine tune the required steering response.

$$M_{Mot,FF}(s) = G_{FF}(s)\, X_{Rack}(s) \tag{7}$$

Apart from this, the motor inertia plays a key role in determining the rack bandwidth. Relatively higher inertia has a negative influence on the steering feel. An additional motor torque is requested generally based on the estimated acceleration as feedforward compensation. Although there are numerous methods for estimating the acceleration but the accuracy depends on the calculation time step, signal noise and time delay which affects the stability. Figure 4(a) illustrates the open loop stability of SWA/SWT transfer function in pole-zero map for increasing inertia compensation (appearance of right hand plane pole). The effect of time delay has been taken into account using Padé approximation (Åström and Murray, 2008). Also in standalone simulation with SWT as step input (in Figure 4(b)), a feedforward compensation has stability issues due to above stated reasons (compare data 1 and 2). Only a partial compensation can be achieved with feedforward. The refinement of this part requires advanced control methods which will be considered in the future. An effective compensation of motor dynamics is realized by a feedback control.

4.2 *Feedback control*

As mentioned in section 3, torsion bar torque from the sensor determines the reference rack position and velocity. The virtual model is composed of lower system mass/inertia and higher damping governing the functionality of the feedback control. A simple PI-controller is implemented. The actual rack state is estimated from the onboard pinion angle sensor. The feedback motor torque (in Laplace domain) is given in Equation 8 below:

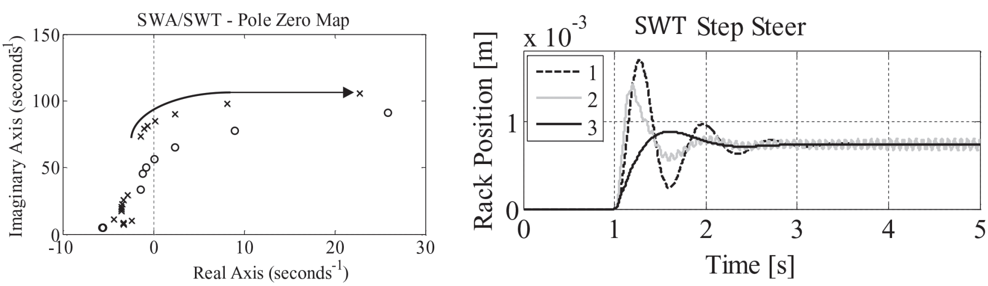

Figure 4. a) SWA/SWT pole-zero map with increasing inertia compensation. b) SWT as step input. 1–Passive EPAS response, 2–Feedforward inertia compensation (at stability limit), 3–Feedback control.

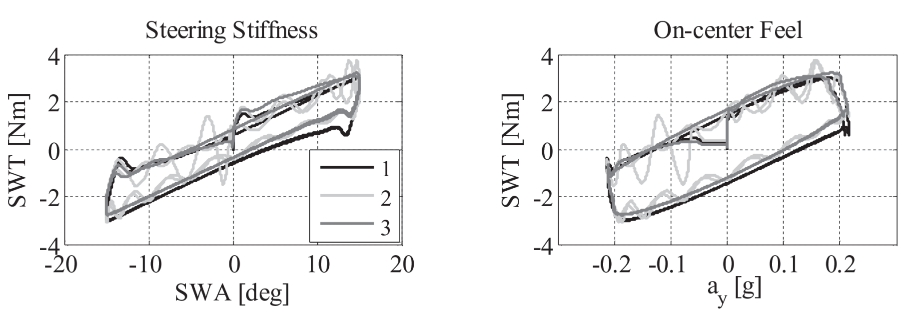

Figure 5. On-center weave maneuver for a steering frequency of 0.4 Hz at $v_X = 100$ km/h. 1–Reference (mechanical system), 2–EPAS without feedback control and 3–EPAS with feedback control.

$$M_{Mot,FB}(s) = G_{X_{Rack}}(s)\,E_{Rack,pos}(s) + G_{\dot{X}_{Rack}}(s)\,E_{Rack,vel}(s) \tag{8}$$

shere $E_{Rack,pos}$ –rack position error; $E_{Rack,vel}$ –rack velocity error. The feedback controller transfer functions $G_{X_{Rack}}(s)$ and $G_{\dot{X}_{Rack}}(s)$ have vehicle speed dependent gains.

The first task of the feedback controller is compensation of servo motor dynamics which has been achieved as seen in Figure 4(b) – data 3. Extending this result on full vehicle simulation, on-center weave maneuver has been performed to evaluate the steering feedback and stability. The steering response can be seen in Figure 5, SWT vs SWA and SWT vs lateral acceleration a_y. The steering sensitivity, SWA vs lateral acceleration, is almost independent of the control strategy. A static gain is used for basic steering assistance rather than a non-linear boost curve. The reference model sets the target steering feel which is equivalent to a mechanical system but with auxiliary steering functions. The result (with feedback control) shows a promising outcome of achieving an almost similar on-center steering feel with EPAS control by decoupling the effects of motor dynamics. If not compensated, the motor inertial effects can cause stability issues and hampers the steering response. Moreover, the tracking of the reference steering feel does produce some deviations while returning to the center largely due to the presence of friction in steering rack and column (in full vehicle model) which has not been considered in this controller.

The performance and stability of the controller for reference tracking is dependent not only on the controller gains but also on the virtual model parameters, especially mass and damping properties because of dynamic compensation. This limitation will not be discussed in detail.

5 RESULTS

The second aspect is to exhibit the potential of steering feel reference model and how actually the steering feedback can be altered. An example of stiffness variation within reference model has been considered. The on-center weave maneuver was performed for 0.2 Hz steering frequency and the results are shown in Figure 6. Higher reference model stiffness results in a heavier steering feel making the driver feel more directly connected to the road and vice versa. This correlates with SWT gradient over lateral acceleration. The transient response can be seen in Figure 7. The steady state gain variation is similar to the on-center response. Comparing the phase delay at 1 Hz for dynamic response, higher stiffness results in lower damping and vice versa. Hence, the driver can feel more about the yaw rate amplification around 1 Hz with higher stiffness. In all the cases, vehicle handling was found almost similar. To sum up, higher stiffness results in a more aggressive steering feedback whereas reduced stiffness creates a comfortable steering response objectively. This analysis shows that it is possible to emulate a desired steering characteristic as defined in the reference given that the relevant information for the driver has been used.

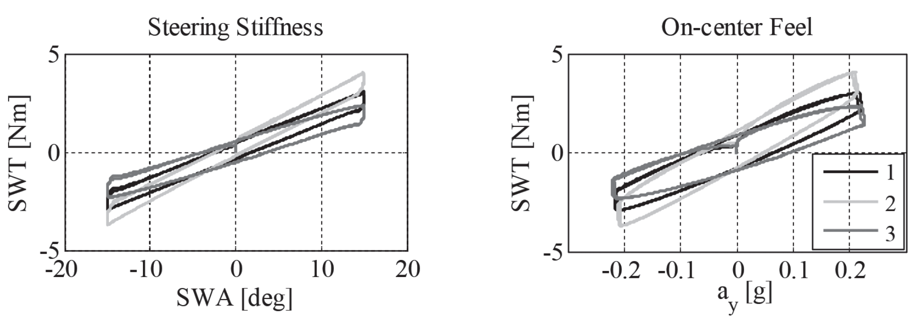

Figure 6. On-center weave at v_X = 100 km/h. 1–Standard, 2–High stiffness and 3–Low stiffness.

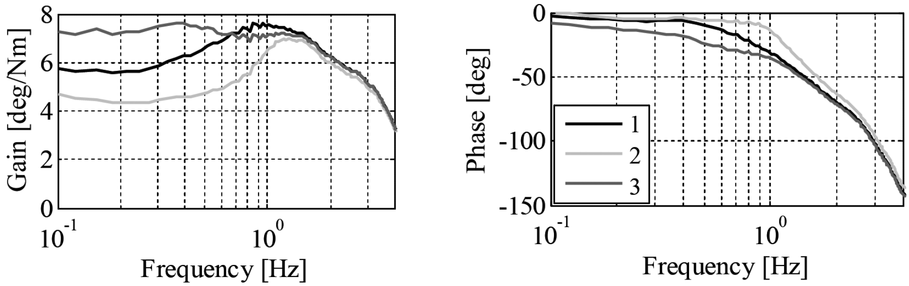

Figure 7. SWA/SWT frequency response at $v_X = 100$ km/h. 1 – Standard, 2 – High stiffness and 3 – Low stiffness.

6 CONCLUSION

This paper has proposed a control concept for designing the steering feel with a feedforward-feedback approach in an EPAS system. Design of the steering feel reference model and basic functions have been discussed. The main challenge in EPAS is to compensate the motor dynamics. The presented steering controller performs decoupling of these effects and almost replicates the target steering feel (same as mechanical steering) which can be observed from the simulation results. The potential of this concept has been shown in an example by changing the steering feel reference (for a desired behavior) and consequently observing the change in steering feedback.

For future tasks, the controller will be further evaluated for its robustness, performance and limitations and then performing the experiments. The development of the steering feel reference model needs improvement and further investigation, for instance inclusion of the friction element and its compensation module.

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REFERENCES

Åström, K.J. & Murray, R.M. 2008. *Feedback Systems*, Oxfordshire,UK, Princeton University Press.

Balachandran, A. & Gerdes, J.C. 2015. Designing Steering Feel for Steer-by-Wire Vehicles Using Objective Measures. *Mechatronics, IEEE/ASME Transactions on,* 20, 373–383.

Bröcker, M. 2007. New control algorithms for steering feel improvements of an electric powered steering system with belt drive. *Vehicle System Dynamics,* 44, 759–769.

Dannöhl, C., Müller, S. & Ulbrich, H. 2012. H∞-control of a rack-assisted electric power steering system, *Vehicle System Dynamics,* 50, 527–544.

Fankem, S. & Müller, S. 2014. A new model to compute the desired steering torque for steer-by-wire vehicles and driving simulators. *Vehicle System Dynamics*, 52, 251–271.

Harrer, M. & Pfeffer, P.E. 2017. *Steering Handbook,* Springer International Publishing Switzerland.

Lee, D., Jang, B., Yi, K., Chang, S. et al. 2016. A Novel Electric-Power-Steering (EPS) Control Algorithm Development for the Reference Steering Feel Tracking. *SAE Technical Paper,* 2016-01-1546.