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Synthesis and Validation of a Rack Position Controller for an Electric Power Steering

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Abstract: This paper provides insight into the position control of an electric power steering system to meet the challenges of autonomous driving. It builds on previous works in this field to present advancement in terms of tracking performance and robustness. Furthermore the approach presented here, including controller synthesis, has been validated in a test vehicle under representative test conditions.

Keywords: Electric Power Steering, Linear Control, Tracking Performance, Actuator Control, State-Space, Lead-Lag Control

1. INTRODUCTION

With the advent of autonomous driving, new challenges are placed on the Electric Power Steering (EPS). No longer is the EPS merely used as an amplifier to provide assistance to the drivers input, rather, through the electric motor, the EPS is used as an actuator. This is in-turn used to deliver a front steering angle which can be commandeered from a higher level controller which takes over the guidance of the vehicle. Although this steering control is of vital importance for autonomous driving, this area of research has received little attention, with work over the past two decades focusing on the problem of steering feel, for example Dannöhl et al. (2012), Yamazaki et al. (2009), Fankem et al. (2014), Zaremba et al. (1997). Another area where position control has been investigated is in steer-by-wire systems. Here, however, the control problem dealt with by steer-by-wire systems is considerably simplified by the removal of the steering column, which introduces additional resonance as well as increases the order of the system dynamics. Recently this author has described theoretical control methodologies using a PID approach and a state space approach Govender and Müller (2016), and Govender et al. (2016), provide insight into the problem of position control for the EPS system.

During the investigation into this area further practical consideration of dealing with time delays and filter behaviours have arisen. These affect the stability of the controller and its resulting performance. Furthermore an analysis into the desired bandwidth requirement for the position controller has been conducted. Beck (2017) concludes that a bandwidth of 4Hz is required for the position controller to replicate 99% of all current driver steering inputs. In order to present a reliable control strategy for steering control during autonomous driving, the robustness of the controller under real world conditions must be investigated.

The basis of this paper uses a state space controller synthesis described in Govender and Müller (2016) and extends the structure to deal with the practical consideration in order to reach the goal performance. An analysis into the controller stability is conducted. The additional effects of various system and controller characteristics on the stability margins are presented. Finally a controller, that is robust against model mismatch and disturbances, is implemented in a test vehicle, which provides a validation of the controller designs with experimental data.

2. SYSTEM MODELLING

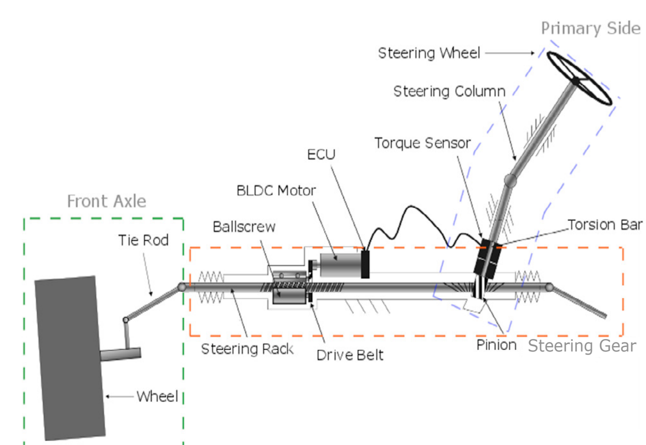

Fig. 1. Description of the Electric Power Steering. Modified from Harrer and Pfeffer (2017)

The conventional EPS system can generally be broken up in three sections, namely the primary side, steering gear and lastly the front axle. In this work the ***Primary Side*** is defined from the steering wheel until the pinion driving the rack. The ***Steering Gear*** begins at the interface

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between pinion and rack and ends at the tip of the rack. This part includes the Brushless DC Motor and its assembly into the steering rack. The last section is the ***Front Axle***, which begins with the tie rod connection to the rack and ends at the tyre-road interface.

For controller synthesis in the linear domain, an appropriate linear model must be derived. In Govender and Müller (2016) the steering system described previously, with all its non-linearities including friction and torsion bar stiffness, is reduced to a two mass linear model without losing crucial system behaviour. This reduced system will now be presented and forms the plant around which the controller will be designed. A description of the system variables can be found in appendix A.

The system is governed by the following two differential equations describing the steering wheel and steering rack behaviours respectively.

$$J_{sw}\ddot{\delta}_{sw} + c_{tb}(\delta_{sw} - i_L y_{sr}) + d_{sc}\dot{\delta}_{sw} + d_{tb}(\dot{\delta}_{sw} - i_L\dot{y}_{sr}) = 0. \quad (1)$$

The right hand side of the previous equation shows that no driver hand torque is applied. Equation 2 contains the term $\frac{c_j}{{i_T}^2}y_{sr}$ which described the external forces from the road as a linear spring.

$$m_{sr}\ddot{y}_{sr} + c_{tb}i_L(i_L y_{sr} - \delta_{sw}) + \frac{c_j}{{i_T}^2}y_{sr} + d_{sr}\dot{y}_{sr} + \frac{d_{rax}}{d{i_T}^2}\dot{y}_{sr} + d_{tb}i_L(i_L\dot{y}_{sr} - \dot{\delta}_{sw}) = T_{em}i_{bd}i_{bs}. \quad (2)$$

The system can now be described in a state space framework, $\dot{x} = Ax + Bu$, as shown in equation 3a and will be used for the control synthesis.

$$\begin{bmatrix}\dot{q}\\ \ddot{q}\end{bmatrix} = \begin{bmatrix}0_{2\times2} & I_{2\times2}\\ -M^{-1}C & -M^{-1}D\end{bmatrix}\begin{bmatrix}q\\ \dot{q}\end{bmatrix} + \begin{bmatrix}0_{3\times1}\\ \frac{i_{bd}i_{bs}}{m_{sr}}\end{bmatrix}T_{em} \quad (3a)$$

$$y = [0\ 1\ 0\ 0]\begin{bmatrix}q\\ \dot{q}\end{bmatrix} \quad (3b)$$

where $q = \begin{bmatrix}\delta_{sw}\\ y_{sr}\end{bmatrix}$, is a position vector and M, C and D are matrices representing the mass, stiffness and damping parameters.

This linear 4th order system is a representation of the non-linear behaviour of steering system shown in Fig. 1 and is used for the control synthesis. It consists of 4 states, i.e. the steering wheel angle, steering rack position and their respective velocities. The electric motor torque is the input of the system with steering rack position as the output.

3. CONTROLLER SYNTHESIS

The main aim of the control synthesis is to design a controller based on the linear model. This should track a desired steering rack position by manipulating the electric motor torque. We have performance requirement of a bandwidth of over 4Hz as described in Groll et al. (2006).

The proposed solution is a cascade control structure with a state space controller $\boldsymbol{K}$ in the inner loop and a lead/integrator controller $\boldsymbol{C}(s)$ in the outer loop, see Figure 2. We choose a state space controller for the inner loop as it is a convenient and functional way of including the steering wheel angle measurement into the controller. The lead/integrator controller $\boldsymbol{C}(s)$ was chosen to improve the performance and guarantee zero steady state error.

We now define further parts of the steering system as well as auxiliary variables. We first define the one input two output system $\boldsymbol{P}(s) \in \mathbb{C}^{2\times1}$ which is the plant as defined in section 2 with an additional output being the steering wheel angle. The electric motor, $\boldsymbol{G}_m(s)$, is modelled as a second order system with a bandwidth of 60Hz. The communication delay (t_d) of the system is 2ms. The measurements of steering wheel angle and steering rack position are filtered with a second order system with a bandwidth of 50Hz. Their derivatives are also calculated to get the whole state vector $\boldsymbol{x}$. We combine this into the filter $\boldsymbol{G}_{f,in}(s) \in \mathbb{C}^{4\times2}$ which will be used in the inner loop. For the outer loop we define the filter $\boldsymbol{G}_{f,out}(s) \in \mathbb{C}^{1\times2}$ which takes the steering wheel angle and steering rack position and returns the filtered steering rack position. The same second order filter is used in the inner loop.

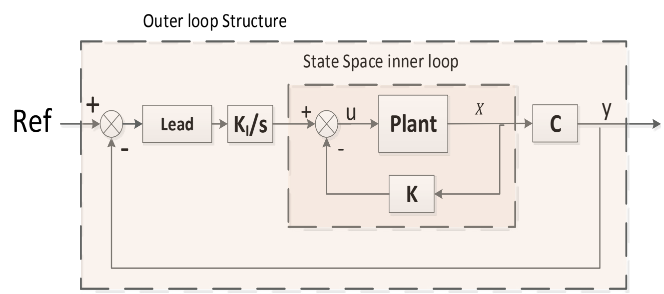

Fig. 2. Controller Scheme

3.1 Inner Loop

The controlled variable is the steering rack position. The steering wheel angle is also an available measurement in modern vehicles and will be used within the controller structure as it provides information about the effects of the steering wheel on the steering rack. Using this measurement therefore enables us to improve the steering rack position control.

The state space controller $\boldsymbol{K} \in \mathbb{R}^{1\times4}$ calculates the command signal from the four states of the linear system. The four control parameters are tuned for disturbance rejection.

The closed loop function of the inner control loop $\boldsymbol{T}_{in}(s) \in \mathbb{C}^{2\times1}$ is

$$\boldsymbol{T}_{in}(s) = \frac{\boldsymbol{P}(s)\boldsymbol{G}_m(s)e^{-st_d}}{1 + \boldsymbol{K}\boldsymbol{G}_{f,in}(s)\boldsymbol{P}(s)\boldsymbol{G}_m(s)e^{-st_d}}. \quad (4)$$

$\boldsymbol{T}_{in}(s)$ has two outputs, the steering wheel angle and the steering rack position. The sensitivity transfer function $\boldsymbol{S}_{in}(s)$ of the inner loop is:

$$\boldsymbol{S}_{in}(s) = \frac{1}{1 + \boldsymbol{K}\boldsymbol{G}_{f,in}(s)\boldsymbol{P}(s)\boldsymbol{G}_m(s)e^{-st_d}}. \quad (5)$$

Even though we use a state space controller, both transfer functions are derived in the frequency domain to be able to easily include time delays. From the sensitivity transfer function $\boldsymbol{S}_{in}(s)$ one can see the advantage of high $\boldsymbol{K}$ values