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Modelling and Position Control of an Electric Power Steering System

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Abstract: This paper presents the modelling, control and analysis of an axle parallel electric power steering system used for autonomous driving. The purpose of the controller is to ensure accurate and robust following of desired trajectories of front steering angle as well as to deliver a smooth steering wheel movement. Firstly the steering model shows how all nonlinearities for the elastic elements, frictions and gear ratio are considered. This model is linearised for the controller design. Thereafter open loop behaviour of the model is examined to determine system resonances. Next a controller synthesis using a Linear Quadratic Integrator controller as well as pole placement is developed. The performance of these controllers is presented in simulation. These results show good tracking performance from the two approaches presented and supports the use of a state space controller design approach in dealing with the various requirement of steering angle control for autonomous driving.

Keywords: **EPS, Steering Analysis, Linear Control, Disturbance compensation, State Space**.

1 INTRODUCTION

Autonomous driving is currently an ever present trend within the automotive industry. A central point of autonomous driving is using the Electric Power Steering system (EPS) as a primary actuator to fulfil all front steer angle requests. As of yet the control of EPS systems have focused heavily on generating a desired driver hand torque as is shown in works by Mehrabi et al. (2011), Fankem et al. (2014) and Dannöhl et al. (2011). The position control situation used for autonomous driving presents a different problem as the EPS motor is no longer used generally as an amplifier of the driver's hand torque rather has to drive the entire steering system including free moving steering wheel. Moreover during manual driving the driver can constantly adjust his steering wheel inputs to compensate for disturbance from the road or even within the steering system itself, while during autonomous driving the EPS angle control should reliably follow the wishes of the vehicle guidance system irrespective of disturbances. Work by von Groll et al. (2006) shows that the most of the relevant frequencies for driver's inputs are below 4Hz. An additional requirement for autonomous driving is smooth steering wheel behaviour. As a result of a free moving steering wheel attached to a spring, i.e. the torsion bar, the resonance frequency of the steering wheel can result in oscillatory behaviour.

Steer-by-wire systems could also perform front steer angle control. Indeed, due to lack of a fixed connection between the steering gear and steering wheel, the mechanical system to be controlled is greatly simplified. While this is in some ways an advantage, there are currently few steer-by-wire in series production today. Moreover, considering the standard EPS is actually a more difficult control problem to solve due to the addition coupling of the steering wheel, it was chosen as the focus of this study. It should be noted that any solution for the EPS system will be easily implementable for a steer-by-wire system.

For industrial uses the tendency is to adopt some form of PID control as a cost effective and easily tunable solution. This type of control may be ill-suited to dealing with the multi order nature of a steering system and can allow resonance frequencies within the steering system to affect the stability or tracking performance of an EPS angle control. Often to keeps the system stable the use of filters, antiwind-up and no zero structures may be adopted but these limit the controller performance. Furthermore classic control has robustness drawbacks as it is tuned for a specific system state as mentioned by Carrière et al. (2008), Anderson and Moore (1990). Complementary work done by Govender et al. (2016) on PID control for front steer angle control gives an insight into some of the drawbacks mentioned above.

An alternative is the use of more system information to control each state and thereby achieve both improved specific performances as well as better range of performance over a large input spectrum. State space control provides an attractive solution in this regard, see Franklin et al. (2010) Through modelling and analysis of the system a controller designer can understand how best to control each state to achieve a desired performance. The steering is however a complex non-linear system which must firstly be linearised to apply a linear state space solution.

This paper presents a state space approach to the problem of front steering angle control which has as of yet been given little attention in published works. Through detailed modelling and analysis of the steering system a suitable linear system is presented which is then used in the control synthesis. Two state space methods are used namely pole placement and linear quadratic (LQ) control. Pole placement was chosen to demonstrate the effectiveness of achieving

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desired performance on the front steer angle when the system is well described. The optimal control method is well known to maintain a good level of performance as well as robustness when the system shows un-modelled behaviour. Indeed within the linear system design a 60 percent minimum phase margin is maintained as is described in Anderson & Moore (1990). Through the weighting of the states and input it offers a simple way of controlling multiple states without examining the transfer function of these states simultaneously. Once the controllers are designed their performance is presented against relevant performance criteria for autonomous driving. This is done firstly with the linear and thereafter with the non-linear model.

2 MODELLING AND SYSTEM ANALYSIS

2.1 Steering System Components

Conventional steering systems today can be broken up into three general parts. Firstly the ***primary side*** is defined from the steering wheel until the pinion driving the rack. Secondly, the ***steering gear*** is made up of the steering rack connected to the EPS motor. The last part is the ***front axle*** including the front wheel. Figure 1 depicts the setup of these parts.

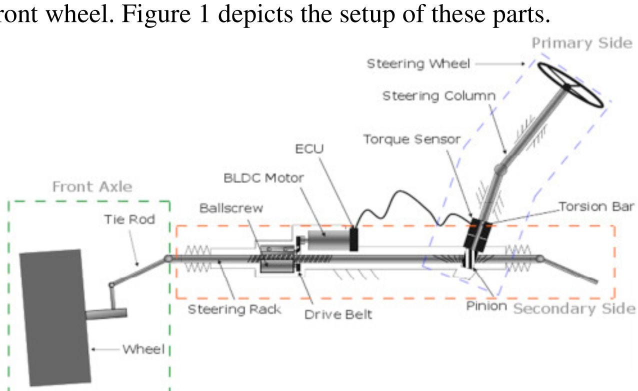

Figure 1: Overall description of the main steering system components. Modified from P.Pfeffer & M. Harrer (2013)

2.2 Non-linearities

Figure 1 shows that the steering system is a 3-mass dynamic system with elastic elements and gears at various points transferring energy. There are however, non-linearities that add to the complexity of the system. The ***variable gear ratios*** between the pinion and rack as well as between the rack and front wheel are non-linear. Furthermore ***friction*** plays an important role in the dynamic behaviour of the system. Lastly the ***self-aligning torque*** at the front wheel is dependent of vehicle velocity and non-linear tyre stiffness. A detailed description of the steering system can be found in Mastinu & Ploechl (2014).

In order to provide a realistic plant to be controlled in simulation, a detailed non-linear model of the steering system as well as a non-linear bicycle model was identified using specific vehicle and steering system measurement data. This process has been presented by Diebold et al. (2006). Throughout this paper the non-linear model developed through the process presented in the aforementioned work may be referred to as Model Based Testing (MbT).

2.3 Derivation of Linearised Model for Controller design

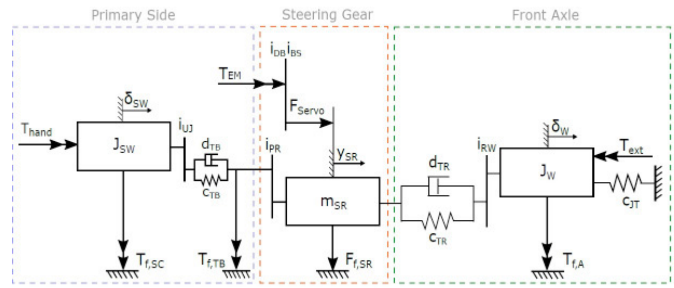

Figure 2: Model description of the steering system

For a linear control approach this system must be linearised. To this effect, friction is modelled as viscous damping, constant gear ratios and a linear vehicle are used. The Lagrange method provides an elegant way to formally derive this system as described by Parmar & Hung (2004).

The state vector is $\boldsymbol{q} = [\boldsymbol{\delta_{SW}} \quad \boldsymbol{y_{SR}} \quad \boldsymbol{\delta_W}]'$, where the elements represent the steering wheel angle, steering rack position and front wheel angle respectively. A full description of all the model parameters is provided in Table 1.

Parameter Name	Symbol	Unit
Driver hand torque	T_{hand}	Nm
Steering wheel & steering column inertia	J_{SW}	Kg.m²
Steering column friction torque	$T_{f,SC}$	Nm
Steering wheel angle	δ_{SW}	rad
Universal joint ratio	i_{UJ}	-
Torsion bar angle	δ_{TB}	rad
Torsion bar stiffness	c_{TB}	Nm/rad
Torsion bar friction torque	$T_{f,TB}$	Nm
Pinion-to-rack ratio	i_{PR}	rad/m
Electric motor torque	T_{EM}	Nm
Drive belt-to-ballscrew ratio	i_{DB}	-
Ballscrew-to-steering rack ratio	i_{BS}	rad/m
Force on steering rack	F_{Servo}	N
Steering rack displacement	y_{SR}	M
Steering rack mass	m_{SR}	Kg
Steering rack friction force	$F_{f,SR}$	N
Tie rod stiffness	c_{TR}	N/m
Tie rod damping	d_{TR}	Ns/m
Steering rack-to-front wheel ratio	i_{RW}	rad/m
Front wheel inertia	J_W	Kg.m²
Front wheel angle	δ_W	Rad
Front axle friction torque	$T_{f,A}$	Nm
External torque acting on wheel	T_{ext}	Nm
Spring stiffness for jacking torque	c_{JT}	Nm/rad
Motor Torque Request	$T_{EM,in}$	Nm
Actual Motor Torque	$T_{EM,out}$	Nm
Period of Motor dynamics	τ	sec

Table 1: Linear model parameters

Lagrange equation incl. dissipation function is given as,

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_k}\right) - \frac{\partial L}{\partial q_k} + \frac{\partial D}{\partial \dot{q}_k} = \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_k}\right) - \frac{\partial T}{\partial q_k} + \frac{\partial V}{\partial q_k} + \frac{\partial D}{\partial \dot{q}_k} = \boldsymbol{Q_k} \quad (1)$$

where $\boldsymbol{L} = \boldsymbol{T} - \boldsymbol{V}$ is the Lagrangian, $\boldsymbol{T}$ is the kinetic term, $\boldsymbol{V}$ is the potential term, $\boldsymbol{Q_k}$ is the kth force or torque acting on the coordinate $\boldsymbol{q_k}$ and $\boldsymbol{D} = \frac{1}{2}\boldsymbol{d\dot{x}^2}$ are the dissipative terms.

Modelling of the Kinematic Term

$$T = \frac{1}{2}J_{SW}\dot{\delta}_{SW}^{\;2} + \frac{1}{2}m_{SR}\dot{y}_{SR}^{\;2} + \frac{1}{2}J_W\dot{\delta}_W^{\;2} \quad (2)$$