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HOW TO MAKE STEER-BY-WIRE FEEL LIKE POWER STEERING

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Abstract: In this paper for the design of a Steer-by-Wire (SbW) system a generic controller structure is proposed with bidirectional position feedback. The design goal for SbW here is to match the dynamics of an (electric/hydraulic power) steering system which may notionally be subdivided into a manual and an assistance steering part. For matching the manual steering part a generic linear controller structure and for matching the assistance steering part a nonlinear unilateral controller structure are suggested. The controller design problem is formulated as a system dynamics equivalence problem, either based on a physical or an identified model, and is solved exactly. This result is then adapted according to practical considerations. For robustness and stability analysis of the linear part of the steer-by-wire system passivity theory is applied and performance is evaluated by Bode magnitude plots and a H_∞-performance criterion. Nonlinear simulations at various operating conditions (vehicle speed, road/tire contact) with a high fidelity vehicle dynamics model demonstrate the robustness of the whole system.

Keywords: Steer-by-Wire, Master-Slave Systems, Man/Machine Systems, Robust Control.

1. INTRODUCTION

The realization of a Steer-by-Wire (SbW) system is a challenge in many technical respects. The steering system in general is an essential part of the interface between the driver and his car, providing to him the possibility of lateral vehicle guidance. Thus it has particular relevance for the reliablity and safety of a vehicle. This paper focusses on the control design part rather than fail-safe considerations. Here, also many aspects have to be considered. First of all, the SbW-system must be stable despite the uncertain biomechanical dynamics of the driver (*driver impedance*) and the dynamics related to the wheel steering mechanics, the vehicle and its interaction with the street (summarized as *vehicle impedance*). The driver may hold the steering wheel very tight or lose. Therefore, he forms an uncertain impedance. The vehicle impedance significantly depends on the vehicle speed, the load and the road/tire contact. Beyond this demand for robust stability, there is the need for supplying an adequate feeling for the actual tire forces to the driver as he is used it from nowadays cars. The goal of this paper is to make a first fundamental control design step on the way to SbW. The aim of this step is to copy the properties of a conventional steering system by SbW as good as possible. Subsequently, additional features can be added to SbW to fully exploit the operational and safety benefits of this technology. These possibilities include amongst others

- variable steering gear ratio (scheduled with vehicle speed and steering wheel angle),
- the implementation of vehicle dynamics control (e.g. for skidding and rollover avoidance) via feedback of vehicle states to the front steering angle, and
- exploiting the haptic interface for forwarding additional information of any kind to the driver.

The conventional steering system to be matched by SbW is either a mechanical steering system without power assistance (manual steering system) or a power assisted steering system like hydraulic power steering (HPS) or electric power steering (EPS). In the sequel we do not want to distinguish between EPS and HPS. We assume a generic power steering system (PS) which represents the desired dynamics.

The SbW-system may be considered as a one degree-of-freedom master/slave-system with an intervening dynamics (also called object dynamics) similiar as it is known from teleoperation systems (Yokokohji and Yoshikawa, 1992). For instance in telerobotics master/slave systems are used to make the human who operates a robot (master) interconnected by wire with another robot (slave) performing a task. The human operator receives a haptic feedback at the master. Thus, he gets a feeling about what happens at the slave. By bidirectional position and/or force feedback three categories of transmission quality (so-called *ideal responses*) are distinguished (Yokokohji and Yoshikawa, 1992): a) Equality of positions, b) equality of forces and c) equality of both positions and forces. These different kinds of transmission quality can be realized by a) bidirectional position feedback, b) bidirectional force feedback and c) bidirectional position and force feedback at master and slave. In robotics the equality of both positions and forces is denoted *transparency*.

Considering the SbW-system as a master/slave system transparency would mean that the driver perfectly feels every little unevenness of the road which is clearly not desired. In fact some low pass filtering is needed as provided by a conventional steering system. The goal of the SbW control design in this paper is to match the feeling of a conventional steering system, i.e. the aim is to attain *equivalence* between SbW and the conventional steering system.

This paper is organized in 7 sections. The scope of the SbW-control system design and the problem formulation is stated in section 2. In section 3 the power steering system is generically modeled which represents what is aimed at by the SbW control design. The SbW-configuration is described in section 4 and a SbW-controller is derived in section 5. The linear part of the SbW control system is analyzed in section 6. Passivity theory is applied for proof of robust stability and a H_∞-performance criterion is used to rate the equivalence, i.e. how good the SbW-system matches the PS-system. In section 7 nonlinear simulations using a high fidelity vehicle dynamics model illustrate the robust performance of the nonlinear SbW-system comprising the linear manual steering part and the nonlinear (power steering) assistance part.

2. SCOPE OF STEER-BY-WIRE CONTROL SYSTEM DESIGN

As outlined in the introduction, the primary scope of SbW control design is to copy the properties of a conventional steering system. This means that the intervening dynamics w.r.t. position and forces between the steering wheel and the front wheel in both directions is equivalent to the conventional steering system dynamics. Hence, the two basic requirements to be satisfied designing the SbW control system are

- equivalence
- robust stability of the entire system including SbW, the driver impedance, and the vehicle impedance. Due to varying operating conditions and varying biomechanics of the driver the vehicle and the driver impedances have to be considered uncertain within certain bounds.

Equivalence

The power steering system may notionally be subdivided into a (approximately linear) manual steering part and a generically nonlinear power assistance steering part. The power steering system may be generally described by

$$\begin{bmatrix}\delta_h\\ x_r\end{bmatrix} = \begin{bmatrix}P_{11} & P_{12}\\ P_{21} & P_{22}\end{bmatrix}\begin{bmatrix}T_h\\ F_r\end{bmatrix} + \begin{bmatrix}0\\ P_{23}\end{bmatrix} F_{a,PS}\ , \qquad (1)$$

where δ_h denotes the handwheel angle and x_r the steering rod position. T_h is the handwheel torque and F_r the environmental force acting on the steering rods. The P_{ij} denote linear transfer functions. The assistance force set point

$$F_{a,PS} = F_{a,PS}(\dot{\delta}_h, T_{TS}, v) \qquad (2)$$

is the output of the power steering electronic control unit (ECU) or the control valve characteristics of the hydraulic system respectively. $F_{a,PS}$ is a nonlinear function depending on the rate of the steering wheel angle $\dot{\delta}_h$, the steering column torque T_{TS} (measured e.g. by means of a torsion bar) and vehicle speed v. Fig. 1 illustrates the static assistance force of an EPS system. For manual steering systems $F_{a,PS} = 0$.

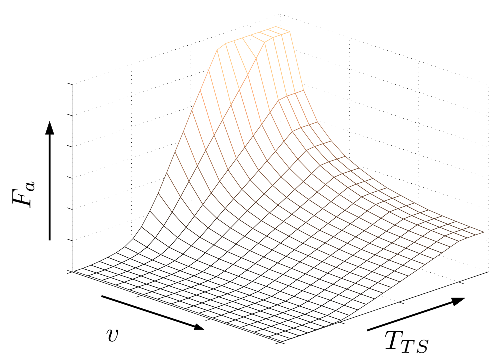

Fig. 1. Static assist characteristics of an exemplary EPS system.

The remaining dynamics, denoted *manual steering part*, is considered linear. The idea which forms the base for the approach introduced in this paper is to notionally subdivide the SbW system into two parts and relate these parts to the manual and assistance steering part respectively, i.e.

$$\begin{bmatrix} \delta_h \\ x_r \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} \\ S_{21} & S_{22} \end{bmatrix} \begin{bmatrix} T_h \\ F_r \end{bmatrix} + \begin{bmatrix} 0 \\ S_{23} \end{bmatrix} F_{a,SbW} \ , \qquad (3)$$

where S_{ij} are linear transfer functions and $F_{a,SbW}$ is the assistance force set point for the SbW-system. Obviously, equivalence of both systems can be achieved if $S_{ij} = P_{ij}$ is established and the same assistance force set point $F_{a,SbW} = F_{a,PS}$ is implemented. Hence, the equivalence of the manual and the assistance part will be considered separately.

However, it will be shown that due to some practical reasons $S_{ij} = P_{ij}$ cannot be perfectly achieved. Therefore, the quality of the SbW linear part design will be rated by means of Bode plots of P_{ij} and S_{ij} respectively and in addition by a H_∞-performance (equivalence) criterion based on scaled admittance matrices. The admittance matrices of the corresponding linear steering parts are

$$\begin{aligned} \boldsymbol{Y}_{PS}(s) &= s \cdot \begin{bmatrix} P_{11}(s) & P_{12}(s) \\ P_{21}(s) & P_{22}(s) \end{bmatrix} \\ \boldsymbol{Y}_{SbW}(s) &= s \cdot \begin{bmatrix} S_{11}(s) & S_{12}(s) \\ S_{21}(s) & S_{22}(s) \end{bmatrix} \ . \end{aligned} \qquad (4)$$

Now, scaling is applied since the tire forces and the driver's handwheel torque or the rack position and the handwheel angle respectively have different units. For this purpose two constant scaling factors n_v and n_f are introduced. Hence, the scaled admittance matrices are

$$\boldsymbol{Y}_{PS,s}(s) = s \begin{bmatrix} P_{11}(s) & \frac{1}{n_f} P_{12}(s) \\ n_v P_{21}(s) & \frac{n_v}{n_f} P_{22}(s) \end{bmatrix} \qquad (5)$$

$$\boldsymbol{Y}_{SbW,s}(s) = s \begin{bmatrix} S_{11}(s) & \frac{1}{n_f} S_{12}(s) \\ n_v S_{21}(s) & \frac{n_v}{n_f} S_{22}(s) \end{bmatrix} , \qquad (6)$$

where n_f and n_v are selected such that the steady state gains of all elements of $\boldsymbol{Y}_{PS,s}(s)$ are identical. This leads to $n_f = 1/n_v = i_P$, where i_P is the pinion/rack gear ratio.

To achieve good performance, i.e. a good level of equivalence, we need to find a SbW-controller such that the H_∞-norm of the scaled admittance difference

$$J = \|\boldsymbol{Y}_{SbW,s}(s) - \boldsymbol{Y}_{PS,s}(s)\|_\infty \qquad (7)$$

becomes as small as possible. This type of specification is commonly used for the design of master/slave systems in frequency domain (Hu *et al.*, 1996; Canudas-De-Wit and Billot, 2001). In this paper we do not apply H_∞-controller design techniques. However, J will be used later in section 6 to rate the quality of the linear part of the SbW-system with an algebraically deduced controller.

The second equivalence postulation is quite easy to accomplish. Therefore, $F_{a,SbW} = F_{a,PS}$ and $S_{23} = P_{23}$ have to be established. If $F_{a,PS}$ is implemented in an ECU (e.g. in case of EPS), then the same algorithm can be used for SbW. For T_{TS} a virtual substitue signal has to be generated by an adequate model. If $F_{a,PS}$ is generated by a mechanical system (e.g. in case of HPS), a model of this system can be used.

Robust stability

For the design of the steer-by-wire control system it is not sufficient to only look at the stability of the SbW-system itself. Also the dynamic interaction between the SbW-system and its environment has to be considered. This is namely the driver impedance connected via the steering wheel and the vehicle impedance connected via the steering rods. For proving stability passivity theory may be applied. Therefore, we transfer results from teleoperation systems to SbW. A bilateral teleoperation system consists of five interacting subsystems: a) human operator, b) master manipulator, c) controller, d) slave manipulator and e) environment (Hu *et al.*, 1996). Analogously, a SbW-system consists of: a) driver, b) actuated steering wheel (SWA), c) controller, d) front wheel actuator (FWA) and e) vehicle as shown in Fig. 2. A sufficient condition for robust stability

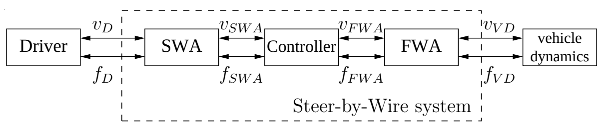

Fig. 2. General steer-by-wire system.

of the system is the passivity of all five subsystems. We merge the three subsystems FWA, SWA and controller into one subsystem since SWA and FWA are active systems. The condition for passivity of the merged subsystem is less restrictive than asking for passivity of any of the single subsystems. In the sequel, only the linear parts of the steering systems are considered for passivity considerations.

Assuming the driver and vehicle impedances to be strictly passive (but otherwise arbitrary) a necessary and sufficient condition for robust stability of the whole system (comprising driver, environment and manual steering part of SbW) is established by applying a criterion based on the structured singular value μ (Colgate and Hogan, 1988) which is given by

$$\bar{\sigma}\big(\boldsymbol{S}_T(s)\big) = \sup_\omega \mu(\boldsymbol{S}_T(j\omega)) \leq 1 \ , \qquad (8)$$

where $\boldsymbol{S}_T$ is the scattering matrix and $\bar{\sigma}$ is the maximum singular value. For the SbW-system the scattering matrix is defined by

$$\boldsymbol{S}_T = (\boldsymbol{Y}_{SbW} - \boldsymbol{I})(\boldsymbol{Y}_{SbW} + \boldsymbol{I})^{-1} \ . \qquad (9)$$

3. POWER STEERING SYSTEM

The power steering system to be matched by the SbW-system might be given either by a

- detailed mathematical model in terms of differential equations with physical parameters or
- by a transfer matrix of frequency responses being obtained from identification experiments.

In any case, the inputs and outputs considered are according to (1): The handwheel torque and reaction force at the steering rods are considered as inputs. The outputs are the position of both the handwheel and the rack. Thus, for linear considerations, the dynamics can be represented by a 2×2 transfer matrix. The following convention for signs of inputs and outputs is used: Both, the hand wheel torque T_h and the external reaction Force F_r act in the direction of δ_h and x_r.

Model based description

The mathematical model of the power steering system introduced in this section is quite generic. Therefore, the power steering system is represented by a set of Laplace transformed differential equations. The first (linear) part of the system only comprises the dynamics of a manual steering system. Nonlinearities due to friction and kinematics are neglected. The second (optional) part is the assistance power steering part, which may also include nonlinear characteristics.

(a) *Manual part:* The first equation describes the dynamics (basically due to inertia) of the steering wheel:

$$\delta_h = P_h(s)(T_h - T_{TS}) \; , \qquad (10)$$

where P_h is the transfer function from the difference of the handwheel torque T_h (applied by the driver) and the torque T_{TS} at the pinion torque sensor to the handwheel angle δ_h. The second body of the manual steering system is the rack. The rack position is given by

$$x_r = P_R(s)\left(\frac{T_{TS}}{i_P} + F_{PS} + F_r\right) \; , \qquad (11)$$

where P_R denotes the transfer function from the rack forces to the rack position. The forces comprise

- the pinion force T_{TS}/i_P (i.e. pinion torque divided by pinion/rack gear ratio i_P),
- the reaction force F_r from the tires acting on the rack via the steering rods,
- and optionally the force F_{PS} applied by the power steering system.

The inertia of the torsion bar is neglected. The pinion torque depends on the torsion angle between hand wheel angle δ_h and the pinion angle x_r/i_P via the transfer function $P_P(s)$:

$$T_{TS} = P_P(s)\left(\delta_h - \frac{x_r}{i_P}\right) \; . \qquad (12)$$

(b) *Assistance power steering part:* If present, the assistance force F_{PS} is applied by a hydraulic piston or an electric motor. It is related to the position via the transfer function $P_{PS_{pos}}$ (e.g. accounting for additional inertia, viscous damping, or mutual induction) and to the assistance force set point via a transfer function $P_{PS_{ref}}$, i.e.

$$F_{PS} = P_{PS_{pos}}(s)x_r + P_{PS_{ref}}(s)F_{a,PS} \; . \qquad (13)$$

Identification based description

If no physically based modeling of the power steering system is feasible, then another approach can be made for gaining knowledge of its input/output behavior. Therefore, it is assumed, that the steering system is separated from the car and attached to a test arrangement. Thereby, independent external torques/forces can be applied to the handwheel (replacing the driver) and the steering rack (replacing the steering rods). Suitable signals should be given to these inputs and the outputs δ_h and x_r should be measured in order to gain data for system identification in frequency domain. The result of the identification are the five transfer functions P_{11}, P_{12}, P_{21}, P_{22}, P_{23} and (if present) the function $F_{a,PS}$.

4. STEER-BY-WIRE CONFIGURATION

Analogously to (10), the dynamics of the steering wheel are

$$\delta_h = S_h(s)(T_h - T_{SWA}) \; , \qquad (14)$$

where S_h is the transfer function from the difference of the handwheel torque T_h (applied by the driver) and the torque T_{SWA} (applied by the steering wheel actuator) to the handwheel angle δ_h. The steering wheel actuator torque

$$T_{SWA} = S_{SWA_{pos}}(s)\delta_h + S_{SWA_{ref}}(s)T_{SWA_{ref}} \qquad (15)$$

is generated by an electric motor via the transfer function $S_{SWA_{ref}}(s)$ which receives a torque set point $T_{SWA_{ref}}$ from the SbW-controller. Inertia and mutual induction can be accounted for with the transfer function $S_{SWA_{pos}}(s)$. The model for the front steering actuator has the same structure as the one for the steering wheel actuator (15):

$$T_{FWA} = i_S\, S_{FWA_{pos}}(s)x_r + S_{FWA_{ref}}(s)T_{FWA_{ref}} \; , \qquad (16)$$

where $T_{FWA_{ref}}$ denotes the torque set point as received from the SbW-controller. The rack dynamics

$$x_r = S_R(s)\left(\frac{T_{FWA}}{i_S} + F_r\right) \qquad (17)$$

are analogous to (11), however the inputs are only the reaction force F_r applied by the steering rods and the force T_{FWA}/i_S from the front wheel actuator which is translated by a gear with gear ratio i_S.

5. DESIGN OF STEER-BY-WIRE SYSTEM

The task of the SbW-controller is to provide the right torque set points for both the steering wheel actuator ($T_{SWA_{ref}}$) and the front wheel actuator ($T_{FWA_{ref}}$). The following SbW controller structure is proposed:

$$\begin{bmatrix} T_{SWA_{ref}} \\ T_{FWA_{ref}} \end{bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} \\ C_{21} & C_{22} & C_{23} & C_{24} \end{bmatrix} \begin{bmatrix} \delta_h \\ x_r \\ T_h \\ F_r \end{bmatrix} + \begin{bmatrix} C_{15} \\ C_{25} \end{bmatrix} F_a \,. \tag{18}$$

Generically, this controller uses dynamic feedback of all system inputs (δ_h, x_r, T_h, F_r). Furhermore, an assistance force F_a is added corresponding to the power steering characteristics used for electric or hydraulic power steering in (2). Therefore, T_{TS} may be determined from a model according to (12). In the first step, an ideal SbW controller is derived such that eqns. (1) and (3) are equivalent. It turns out that the controller structure introduced in (18) has more degrees of freedom than actually necessary. Moreover, doing with fewer sensors is advantagous from a cost point of view. Therefore, in the sequel, only position sensors, but no force and torque sensors are assumed for the SbW system, i.e. $C_{13} = C_{14} = C_{23} = C_{24} = 0$. The power assistant force only acts on the front wheel actuator, meaning that $C_{15} = 0$.

Model based design of Steer-by-Wire

If there are mathematical models of both the power steering system to be copied and the SbW-hardware, then these models can be used to derive a perfect SbW-controller. Algebraic solving of all equations (i.e. $S_{ij} = P_{ij}$) yields a unique solution for the controller dynamics:

$$\begin{aligned} C_{11} &= \frac{P_P - S_{SWA_{pos}}}{S_{SWA_{ref}}} + \frac{S_h - P_h}{S_h P_h S_{SWA_{ref}}}, \\ C_{12} &= -\frac{1}{i_P} \cdot \frac{P_P}{S_{SWA_{ref}}}, \quad C_{21} = \frac{i_S}{i_P} \cdot \frac{P_P}{S_{FWA_{ref}}}, \\ C_{22} &= -\frac{i_S}{i_P^2} \cdot \frac{P_P}{S_{FWA_{ref}}} + \\ &\quad i_S \left(\frac{P_R - S_R}{S_{FWA_{ref}} P_R S_R} + \frac{P_{PS_{pos}} - S_{FWA_{pos}}}{S_{FWA_{ref}}} \right), \\ C_{25} &= i_S \frac{P_{PS_{ref}}}{S_{FWA_{ref}}} \,. \end{aligned} \tag{19}$$

With this theoretical controller, the admittance matrices of both the manual steering system and the SbW-system are identical (see (4)) and the performance index J in (7) becomes 0. If also the same algorithm is used in (18) for computing the assistance force set point F_a as with the power steering system then both the SbW and the PS match exactly. In practice, this controller has to be modified. Before doing so, with this representation it can be seen that some potential controller simplifications may result in special cases:

- C_{11} simplifies significantly if the same steering wheel is used for both PS and SbW (i.e. $S_h(s) = P_h(s)$).
- C_{22} simplifies significantly if the same steering gear is used for both PS and SbW (i.e. $S_R(s) = P_R(s)$).
- C_{22} simplifies even more if the very actuator is used as front wheel actuator which was used for power assistance in case of PS (i.e. $S_{FWA_{pos}}(s) = P_{PS_{pos}}(s)$).
- Moreover, it can be seen that in all controller transfer functions the dynamics of the respective actuator ($S_{FWA_{ref}}(s)$ and $S_{SWA_{ref}}(s)$) needs to be compensated. If the actuator bandwidths are sufficiently high, then compensation of their dynamics can be neglected.
- if necessary, low pass filters can be added to make C_{ij} causal. The ideal frequency responses for a specific SbW-system (which perfectly matches a given real EPS system) are plotted gray in Fig. 3. Dashed linestyle is used for realizable controllers with additional low pass filters.

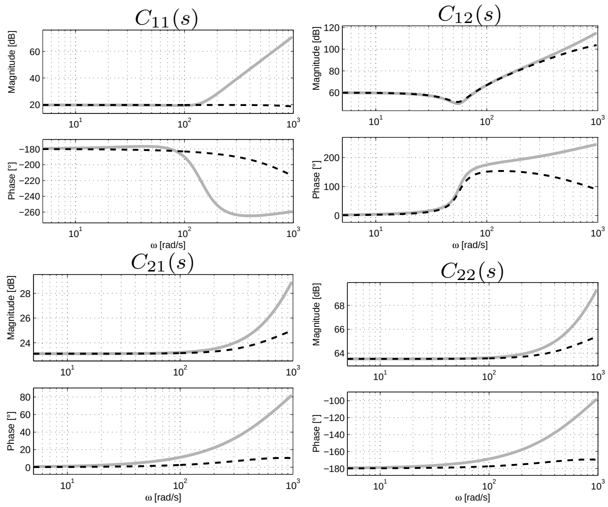

Fig. 3. Frequency responses for ideal controller (19) and implementable controller.

Identification based design of Steer-by-Wire

The controller (18) can also be computed from frequency response models obtained by identification of the power steering system. The equations representing the equivalence postulation $S_{ij} = P_{ij}$ are solved for the controller transfer functions C_{ij}. Thereby, the controller gains and phases are computed for each frequency. An implementable controller can be derived from there by approximation in frequency domain.

6. LINEAR ANALYSIS OF STEER-BY-WIRE

Only the linear part of the steer-by-wire system, i.e. the manual steering part, is considered in this section. Fig. 4 shows that the frequency responses of the elements of the admittance matrices $\boldsymbol{Y}_{PS}(s)$ match

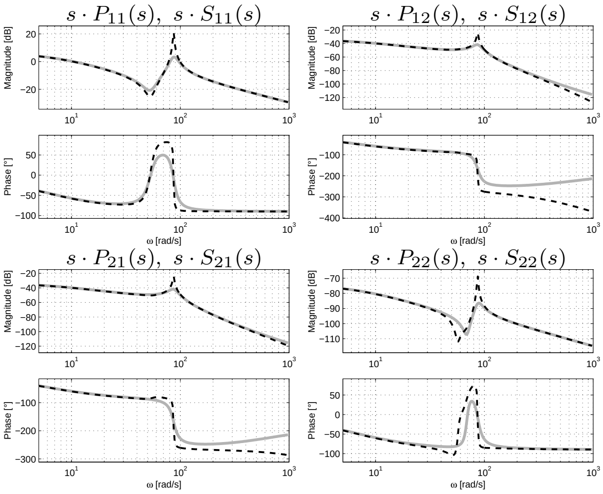

Fig. 4. Admittance matrix frequency responses.

very well their $\boldsymbol{Y}_{SbW}(s)$ counterparts according to (4) (the gray solid lines correspond to PS, the black dashed ones to SbW). Note, that the resonance peaks in the frequency responses of the SbW-system can be reduced by enhancement of the controller. The left plot in Fig. 5 depicts the maximum singular value plot w.r.t. (7). For frequencies in the range 0-10Hz the magnitude of $\bar{\sigma}$ is smaller than -20 dB meaning that a high level of equivalence has been achieved.

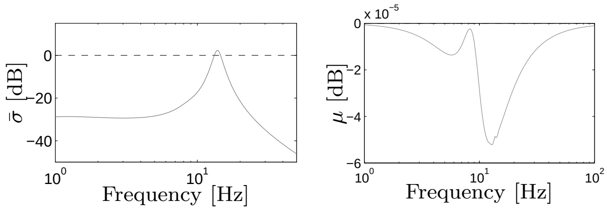

Fig. 5. Magnitude plots of equivalence (left) and passivity condition (right).

The right plot in Fig. 5 shows the structured singular value plot of the scattering matrix $\mu(\boldsymbol{S}_T(j\omega))$. Since $\mu < 1$ $\forall\omega$ the robust stability condition (8) is satisfied meaning that the linear SbW-system is robustly stable w.r.t. arbitrary passive driver and vehicle impedances.

7. SIMULATION RESULTS

The analysis of the nonlinear SbW-system comprising the manual and the assistance steering part, was performed using a high fidelity vehicle dynamics model as well as nonlinear FWA- and SWA-models. The simulations were performed applying a quasi sinusoidal steering torque with a magnitude of 3.2 Nm. Dry road and a speed of 80 km/h were assumed. The results shown in Fig. 6 demonstrate a high level of equivalence. Comparative good results have also been achieved at other speeds and road conditions. However, with practical implementation of the proposed SbW concept, the dynamics of the participating components will not be known perfectly and therefore the expextations established by the shown results need to be qualified. Finally, it should be noted that (instead of a PS system) a system with any desired dynamics may alternatively serve as a reference system. The proposed design process generically supports bidirectional mixed position and force feedback control instead of mere bidirectional position control.

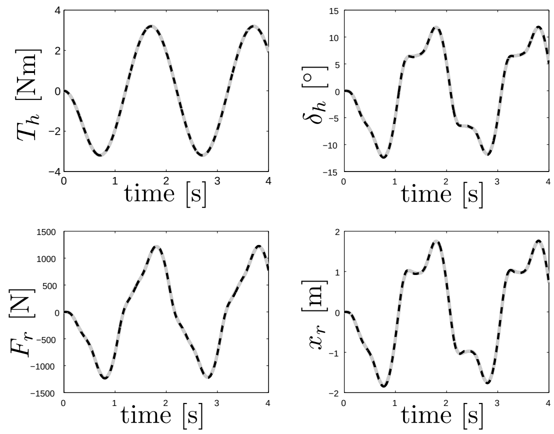

Fig. 6. Simulation results for SbW (black) and EPS (gray).

8. CONCLUSIONS

In this paper a generic control strategy for the design of Steer-by-Wire control systems has been introduced. The goal of making the SbW-system to feel and behave like a power steering system was achieved by applying a model matching (equivalence) approach, either based on a reference model derived from physics or based on identified data. The result of the design procedure is both the controller structure and the corresponding parameterization. A detailed frequency and time domain analysis demonstrated the high level of equivalence between the designed Steer-by-Wire system and a real Power Steering system which served as a reference system. The proposed concept may also be applied to other master/slave force feedback systems.

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