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Abstract: This paper is with the open invited track for the GdR MACS Young PhD researchers. It concerns applied research in the field of automotive control, in particular on the control of Electric Power Steering systems (EPS) to reduce the production costs. This study is meant to design a new control strategy with less available outputs (especially without torque sensor) and preserved steering performance. Thus, two approaches could be considered: first, a driver torque estimation coupled to a conventional controller; second, a control strategy not using the driver torque information. Moreover, from the obtained controller design, an analysis of the new system shall be developed in order to draw conclusions on an appropriate mechanical structure. For this purpose, the study is performed on a column-type EPS system for which a model is expressed by Newton's law of motion. This model is validated using experimental data obtained on a prototype vehicle. Then, the control objectives are defined and potential designs of the controller are discussed, with a reminder on driver torque estimation method. The simulation environment to analyse the strategies is also described.

Keywords: automotive control, electric power steering, linear observer, robust control

1. INTRODUCTION

Interactions between driver and vehicle mainly go through a steering system. Manual steering involves heavy or low-geared steering. Hence, power-assisted steering systems have been introduced to overcome those issues. They help the driver turn the vehicle wheels in the desired direction. Nowadays, modern vehicles are more and more equipped with Electric Power Steering (EPS) systems, substituting Hydraulic Power Steering (HPS) ones. Some advantages of EPS could be listed such as: fuel economy, tunability of the steering feel, ease of integration with other subsystems and ecological impact. In EPS systems, an assistance torque is provided by an electrical assistance motor, to reduce the amount of torque required from the driver to turn the wheels. The amount of supplied power is defined by the Electronic Control Unit (ECU) according to a motor torque control policy, containing an assistance rule depending on the vehicle speed and measurements from a torque sensor.

Indeed, the torque sensor measures the applied steering torque and defines the required torque assistance to provide (typically, through base assist curve and other functions to be tuned). Thus, a failure of the torque sensor commonly leads to a sudden loss of assistance (SLOA). However, such an event should be avoided or its effect attenuated, regarding safety and comfort. Several studies have been carried out on the steering torque estimation without using torque sensor. Such a control strategy could prevent SLOA. Otherwise, a control law not using the steering torque information might also be a way, based only on steering wheel angle. Nevertheless, to the best of the authors' knowledge, there have been very few works on this topic. In practice, a controller that satisfies those above proposals while keeping a good steering feeling would result to lower EPS production costs. This is one of the main challenge for EPS system suppliers to deal with competition. Furthermore, a wider range of vehicles might profit of EPS systems leading to a market expansion.

Some studies on EPS systems are presented below. Badawy et al. (1999) presented a controller divided in three parts: assist, return and damping algorithms. From this, a good steering feeling is achieved by tuning. In Tian et al. (2004), a H_∞ controller has been developed to minimize the effect of disturbances (measurement noises, driver torque and reaction torque) on steering feeling and motor assist errors. System performance is verified in simulation. Also in Liu et al. (2007) a H_∞ controller has been designed to improve stability and performance of the system. Moreover, controller order reduction methods are presented for implementation issues. El-Shaer et al. (2009) presented a two degrees of freedom controller: a state feedback and a phase compensator. According to the design, the closed-loop system is strictly passive and performance outputs, defined by torsion bar signal, control input, steering wheel and rack speed, satisfied the objectives. Chabaan and Wang (2001) developed a H_∞ controller where robust stability and performance have been improved, since a driver torque estimation using the

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torque sensor is taken into account. In Parmar and Hung (2004), an optimal linear quadratic regulator (LQR) using a Kalman filter was designed, where the weighting matrices are function of the effort applied by the driver. However, the only output of the EPS system is the motor angle. Hence, a Kalman filter has been introduced to estimate full-state. Marouf et al. (2011) proposed a reference model, where the desired motor angle is tracked by a sliding mode controller. Regarding implementation issue, a driver torque estimation is required, so a sliding mode observer has been developed. The reference motor angle depends on the three inputs and ideal performances are achieved through model design parameters. In this work, the steering wheel angle and motor angle are also the outputs of the EPS system.

The main contribution of this paper is the design of a robust controller for EPS systems allowing a reduction of production costs while keeping a high steering performance. Consequently, the control objective should be reached reducing the number of sensors (more specifically capable of operating without torque sensor) and combining optimal mechanical parameters that differ from nominal EPS systems (especially stiffness). Two strategies are examined: a controller associated with driver torque estimation (extending authors' previous results) or a control strategy which does not require steering torque information. The later, which only requires measurements on the steering wheel angle, requires reconsidering classical performance evaluation criteria since they usually rely on torque sensing. Furthermore, an adapted design of EPS system is expected, as there is no more need to detect the torsion torque induced by the driver. Thus, additional cost reduction is examined through mechanical parameters influence, unlike previous work, the whole EPS system design is covered.

The paper is organized as follows. In section 2, the EPS system model is presented and the specific concern of its structure is shown. In section 3, the control objective and design are described. The simulation environment is introduced in section 4, and in section 5 future works are defined.

2. EPS SYSTEM MODEL

Column type EPS (C-EPS) systems have been the early EPS systems to be marketed, which means a deeper knowledge and experience feedback. Moreover, C-EPS systems are ideal for compact vehicles (small engine compartment), and even for light vehicles with manual steering. Hence, this study on high stiffness seems appropriate to be applied, at first, on C-EPS type. Fig. 1 shows the whole steering mechanics for a C-EPS system. Three inputs act on the system: the driver torque τ_d, the assist motor torque τ_m and the rack force F_r (sum of left F_{rL} and right F_{rR}).

A simplified mechanical model is deduced, applying Newton's laws of motion and neglecting dry friction. The C-EPS model is governed by the equations (1) to (2):

$$J_c\ddot{\theta}_c = \tau_d - D_c\left(\dot{\theta}_c - \frac{\dot{\theta}_m}{R_m}\right) - K_c\left(\theta_c - \frac{\theta_m}{R_m}\right) - B_c\dot{\theta}_c \quad (1)$$

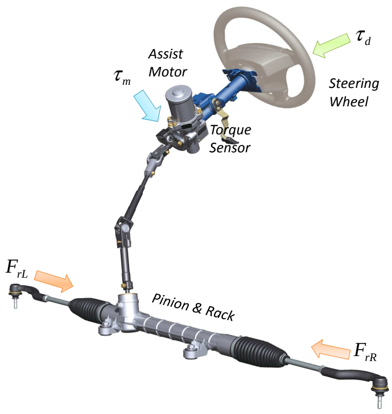

Fig. 1. Column type EPS system

$$\begin{aligned} J_{eq}\ddot{\theta}_m &= \tau_m + \tfrac{D_c}{R_m}\left(\dot{\theta}_c - \tfrac{\dot{\theta}_m}{R_m}\right) + \tfrac{K_c}{R_m}\left(\theta_c - \tfrac{\theta_m}{R_m}\right) \\ &\quad - B_m\dot{\theta}_m - K_r\tfrac{R_p^2}{R_m^2}\theta_m - D_r\tfrac{R_p^2}{R_m^2}\dot{\theta}_m - \tau_r \end{aligned} \quad (2)$$

where $J_{eq} = J_m + \frac{R_p^2}{R_m^2}M_r$ and $\tau_r = F_r\frac{R_p}{R_m}$. Table 1 describes the mechanical parameters.

Table 1. EPS system mechanical parameters

Notation	Description	Unit
J_c	Steering column inertia	$\mathrm{kg\,m^2}$
B_c	Steering column viscous friction	$\mathrm{N\,m/(rad/s)}$
K_c	Column stiffness	$\mathrm{N\,m/rad}$
D_c	Column damping	$\mathrm{N\,m/(rad/s)}$
R_p	Pinion/rack reducer	$\mathrm{m/rad}$
M_r	Rack and tie rods mass	kg
B_r	Rack viscous friction	$\mathrm{N/(m/s)}$
K_r	Rack stiffness	$\mathrm{N/m}$
D_r	Rack damping	$\mathrm{N/(m/s)}$
R_m	Motor reduction ratio	-
J_m	Motor inertia	$\mathrm{kg\,m^2}$
B_m	Motor viscous friction	$\mathrm{N\,m/(rad/s)}$

The state-space representation is given by (3) and the corresponding notations is presented in Table 2.

$$\begin{cases} \dot{x} = Ax + Bu + Ed + Ww \\ y = Cx \end{cases} \quad (3)$$

Table 2. State-space representation variables

Notation	Variables	Description
x	$\begin{pmatrix}\dot{\theta}_c \\ \theta_c \\ \dot{\theta}_m \\ \theta_m\end{pmatrix}$	*Steering wheel speed* *Steering wheel angle* *Assist motor speed* *Assist motor angle*
u	τ_m	*Assist Motor Torque*
d	τ_d	*Driver Torque*
w	τ_r	*Road Reaction Torque*
y	$\begin{pmatrix}\theta_c \\ \theta_m\end{pmatrix}$	*Steering wheel angle* *Assist motor angle*

Fig. 2 shows a comparison between transfer functions, from nominal EPS and prototype EPS (new structure with a higher column stiffness). The top figure represents the frequency response from the motor torque to the driver torque, and the bottom figure represents the frequency response from the motor angle to the motor torque. In both cases, a sinus of amplitude $0.5Nm$ with frequency of $0.5Hz$ to $40Hz$ has been applied as the motor torque input. In low frequencies the behaviour does not change,

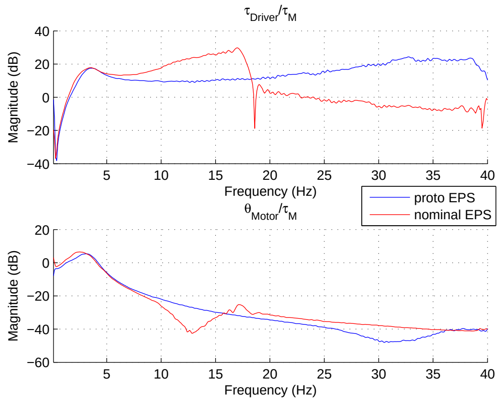

Fig. 2. Frequency responses comparison between Nominal and Prototype EPS system

nevertheless the second resonance frequency of the system has evolved becoming inconsequential to the fundamental behaviour of the system.

3. EPS CONTROL DESIGN

The objectives of EPS control are as follow:

(1) provide an assistance torque to the driver while ensuring a suitable road-feedback, so that the driver feels the resistance torque due road conditions (ice, pavement, asphalt) and vehicle speeds, to be able to adapt his driving
(2) obtain a closed-loop stable system
(3) get a bandwidth of the closed-loop system at least of $\omega = 15Hz$
(4) have robustness to model uncertainty (mainly on the part concerning the road external forces)
(5) have a low complexity regarding implementation issue

Although, steering feeling and comfort are subjective criteria related to driver, those above points illustrate the general guidelines to satisfy it.

3.1 Synthesis using driver torque estimation

In this part, a previously developed driver torque estimations is recalled. The system is given by:

$$\begin{cases} \dot{x} = Ax + Bu + Ed + Ww \\ y = Cx + Nn \end{cases} \tag{4}$$

where additional disturbances have been introduced on the output by n the measurements noise.

The Proportional Integral Observer (PIO) under the assumption that $\dot{d} = 0$, is in the form:

$$\begin{cases} \dot{\hat{x}} = A\hat{x} + E\hat{d} + L_p(y - C\hat{x}) + Bu \\ \dot{\hat{d}} = L_i(y - C\hat{x}) \end{cases} \tag{5}$$

Hence, the state-space representation of the estimation error:

$$\begin{cases} \dot{e_x} = (A - L_pC)e_x + Ee_d + Ww + L_pNn \\ \dot{e_d} = -L_iCe_x - L_iNn \end{cases}$$

$$\Leftrightarrow \begin{cases} \begin{pmatrix} \dot{e_x} \\ \dot{e_d} \end{pmatrix} = \left(\begin{pmatrix} A & E \\ 0 & 0 \end{pmatrix} - \begin{pmatrix} L_p \\ L_i \end{pmatrix} (C \;\, 0) \right) \begin{pmatrix} e_x \\ e_d \end{pmatrix} \\ \qquad + \begin{pmatrix} W \\ 0 \end{pmatrix} w + \begin{pmatrix} L_pN \\ -L_iN \end{pmatrix} n \\ z = (0 \;\, I) \begin{pmatrix} e_x \\ e_d \end{pmatrix} \end{cases} \tag{6}$$

where $e_x = (x - \hat{x})$, $e_d = \left(d - \hat{d}\right)$.

Then, the objective is to find $\left(L_p^T \; L_i^T\right)^T$ such that

(1) $||T_{zw}||_\infty < \gamma_\infty^2$ minimizes the effect of road disturbances on the driver torque estimation error
(2) $||T_{zn}||_2 < \gamma_2^2$ minimizes the effect of measurements noise on the driver torque estimation error.

The inputs of the observer are the steering wheel angle and the assist motor angle. PIO is designed by solving a multi-objective optimization problem subject to H_∞ and H_2 description (i.e Linear Matrix Inequalities), see Yamamoto et al. (2015) for more details. In Fig. 3, similarly as in the previous work, the PIO has been tested using experimental data on a vehicle equipped with a C-EPS system. This shows good performance of the observer.

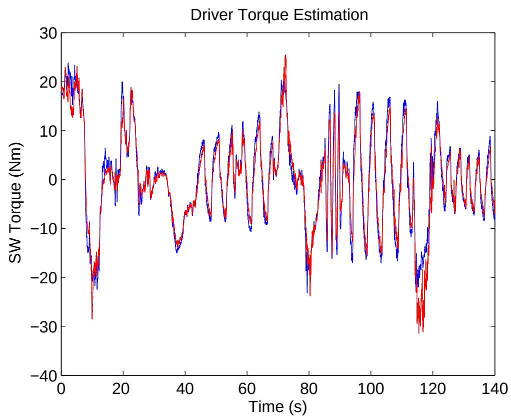

Fig. 3. Driver torque estimation

Then a controller based on the driver torque estimation shall be developed, for instance: a conventional controller (base assist with additional sub-functions) as illustrated in Fig. 4.

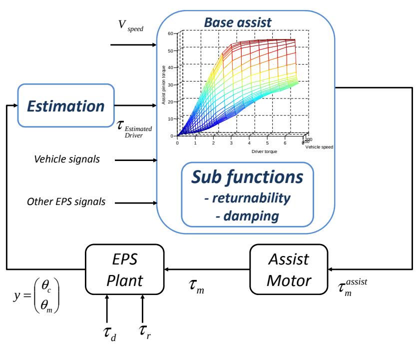

Fig. 4. Conventional controller with driver torque estimation

In Fig. 5 a simple conventional controller associated to the previous PIO has been applied on vehicle. It could be noticed that with this assistance law, the driver torque level does not increase as the steering wheel angle increases. Hence, it helps the driver in handling the steering wheel compared to no assist. Indeed, the driver applied less than $5Nm$ rather than $12Nm$ at $7km/h$ for a lock-to-lock steering wheel manoeuvre, the assist level reached is good despite appearing ripples.

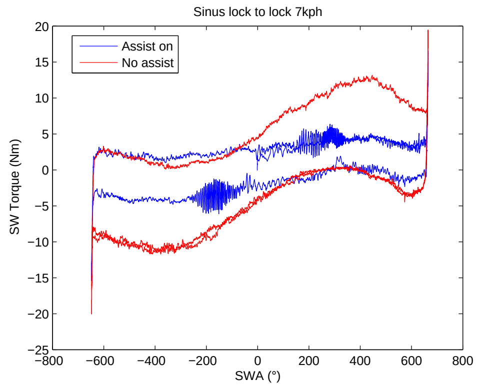

Fig. 5. Comparison between no assist and conventional controller with PI observer on vehicle at 7km/h

3.2 Control synthesis without driver torque information

Since the driver torque information is not considered, the conventional base assist could not be used. Then, the first step is to verify that it is still possible to provide convenient steering assistance. To first design the controller and simplify the problem, the road disturbance τ_r is included as an uncertainty on the tire spring part $K_r \frac{R_p^2}{R_m^2}\theta_m + D_r \frac{R_p^2}{R_m^2}\dot{\theta}_m$. Thus, the input w is removed from (3) and the steering wheel angle is the only measured output i.e $y = \theta_c$, resulting in the transfer function

$$y = G_{yd}(s)\tau_d + G_{yu}(s)\tau_m \qquad (7)$$

The synthesis of the H_∞ controller is carried out regarding Fig. 6. The following transfer functions $T_d(s)$, $W_z(s)$ and $W_u(s)$ have been introduced only for synthesis purpose (not implemented) to get the controller $K(s)$, with EPS plant represented by $G_{yd}(s)$ and $G_{yu}(s)$.

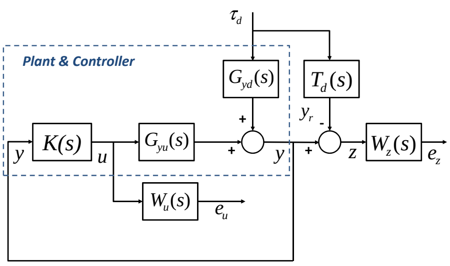

Fig. 6. H_∞ control synthesis

The design parameters are:

(1) Matching function

$$\tau_d = T_d(s)y_r \qquad (8)$$

- This defines the performance level, according to the transfer function of the steering wheel angle and the driver torque, for example $T_d = \frac{K}{1+\tau s}$

(2) Weighting functions for tracking error and actuator constraints:

$$\frac{1}{W_z} = \frac{s + \omega_b \epsilon}{\frac{s}{M_s} + \omega_b} \qquad (9)$$

- The driver characteristics are the following: acting frequency up to $5Hz$ and maximum driver torque expected is $10Nm$ (corresponds usually to the torque sensor saturation value).

$$\frac{1}{W_u} = \frac{\epsilon_1 s + \omega_{bc}}{s + \frac{\omega_{bc}}{M_u}} \qquad (10)$$

- The actuator characteristics is such that the bandwidth is from $30Hz$ to $50Hz$ and motor torque maximum is $5Nm$.

Fig. 6 is usually transformed into the standard H_∞ control configuration of Fig. 7 in order to design the controller.

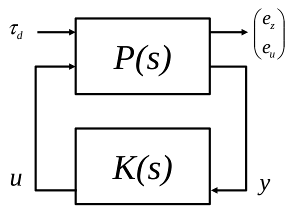

Fig. 7. General H_∞ control problem

Then, the objective is to find $K(s)$ subject to $||T_{zd}(s)||_\infty = \left\| \begin{matrix} W_e S \\ W_u K S \end{matrix} \right\|_\infty \leq 1$ with $S = \frac{y - y_r}{d}$, $KS = \frac{u}{d}$ and $d = \tau_d$.

4. SIMULATION ENVIRONMENT

To analyse the control part, the whole system has to be considered involving not only the mechanical model but also the electrical motor dynamic, expressed in a simplified way by:

$$\tau_m = \frac{K}{1 + \frac{2z}{\omega_0}s + \frac{s^2}{\omega_0^2}}\tau_m^{assist} \qquad (11)$$

Moreover, a non-linear bicycle model (see Fig. 8 and Table 3) is added to simulate the road force acting on the rack F_r (convert with ratio $\frac{R_p}{R_m}$ to get EPS input τ_r), and also to obtain the behaviour of the vehicle through lateral acceleration γ_t and yaw rate $\dot{\psi}$.

Then, the vehicle lateral dynamics are given by:

$$\begin{cases} \dot{V}_y = \dfrac{1}{M}(F_{yf}cos(\alpha_f) + F_{yr}) - V_x\dot{\psi} \\ \ddot{\psi} = \dfrac{1}{C_y}(L_1 F_{yf}cos(\alpha_f) - L_2 F_{yr}) \end{cases} \qquad (12)$$

with $\gamma_t = \dot{V}_y + V_x\dot{\psi}$.

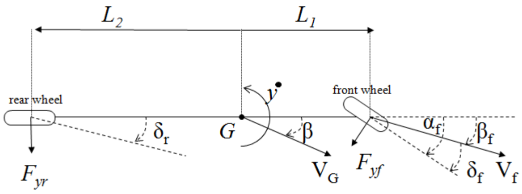

Fig. 8. Bicycle model

The front wheel angle α_f is obtained from the steering wheel angle θ_c through the relation: $\alpha_f = \theta_c R_{ct}$, and slip angles are expressed by:

$$\begin{cases} \delta_v = \alpha_v - \beta_v \\ \delta_r = -\beta_r \end{cases}, \quad \begin{array}{l} tan(\beta_f) = \dfrac{V_y + L_1\dot{\psi}}{V_x} \\ tan(\beta_r) = \dfrac{V_y - L_2\dot{\psi}}{V_x} \end{array} \tag{13}$$

Table 3. Bicycle model parameters

Notation	Description	Unit
M	Vehicle mass	kg
L_1	Distance center of gravity and front axle	m
L_2	Distance center of gravity and rear axle	m
C_y	Yaw inertia	kg m^2
V_x	Longitudinal speed	m/s
V_y	Lateral speed	m/s
R_{ct}	Convert steering wheel to front wheel angle	-
β_f	Slip angle of the front axle	rad
β_r	Slip angle of the rear axle	rad

The lateral forces F_{yf} (front), F_{yr} (rear) and road reaction force F_r acting on the rack are calculated using the Pacejka tire model, where variables are defined in Table 4.

$$F = D_p sin\left\{C_p arctan{-}1[(1-E)B_p\delta + E_p arctan(B_p\delta)]\right\} \tag{14}$$

Table 4. Pacejka tire model descriptions

Notation	Description
B_p	Stiffness factor
C_p	Shape factor
D_p	Peak factor
E_p	Curvatur factor

Fig. 9 presents a comparison between real measurements and vehicle model for a typical test. The test consists of increasing the steering wheel angle at a constant steering and vehicle speed until loss of grip. Simulation results show that a realistic vehicle environment is obtained.

Fig. 10 shows the simulation structure including the suggested controller (either with an estimation of driver torque or without, the output y is then adapted to each case).

5. FUTURE WORKS

In future works, variations of specific EPS mechanical parameters influencing the behaviour of the closed-loop system shall be analysed. Thus, an adapted mechanical design of the EPS system is deduced from the proposed control strategy. Then, an implementation of the whole structure on vehicle is expected.

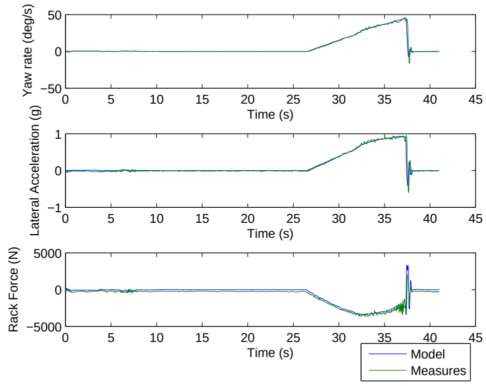

Fig. 9. Validation of the proposed vehicle model

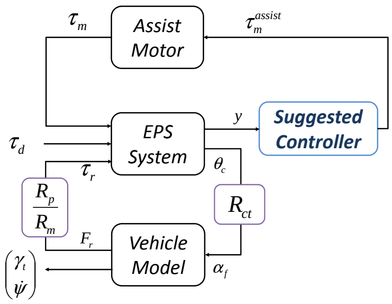

Fig. 10. Simulation structure for the suggested control strategy

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