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Sensorless Control of Electric Power Assisted Steering System

A. Marouf, M. Djemaï, C. Sentouh and P. Pudlo

***Abstract*— This paper deals with the sensorless control of Electric Power-Assisted Steering (EPAS) system with the permanent magnet synchronous motor (PMSM) using sliding mode techniques. Using steering wheel angle and motor currents measurements, two cascaded sliding mode observers with unknown inputs are designed. The whole observer generates the driver torque needed to determine the amount of the assistance and the other pieces of information needed to implement the controller. H-infinity sliding mode control is designed to generate the basic assist torque, improve damping characteristics of EPAS and attenuate the mismatched disturbances. Simulation results show the effectiveness of the proposed control.**

***Index Terms*— Electric power assisted steering (EPAS), sliding mode control, observer with unknown inputs, permanent magnet synchronous motor.**

I. INTRODUCTION

ELECTRIC Power Assisted Steering (EPAS) Systems are replacing hydraulic power steering in many new vehicles today. They have many advantages over traditional hydraulic power steering systems [1]. The main objective of EPAS control is to reduce the driver's steering effort by generating assist torque using electric motor. The amount of assist torque is typically calculated from tunable torque boost based on the vehicle's speed and driver torque [1][2][3].

The requirements for EPAS motors has increased as EPAS system become to be used in medium size and large size vehicles. As a result, the use of brushed dc motor become unsuitable for EAPS system [4][5], because of its large size; high moment of inertia; high friction; and high noise characteristics. Brushless Permanent Magnet Synchronous Motor (PMSM) is a good candidate for EPAS system, because of its high power density; high efficiency; reliability; low moment of inertia; reduced friction; and reduced noise [4][5][6]. However, the control of PMSM is more complicated and more expensive than the brushed DC motor. To achieve high performance in PMSM motor drive, Vector control, also known as Fieled Oriented Control (FCO), is used. This control requires accurate measurement of motor position. The motor angle can be measured using encoder, resolver or Hall sensors. However, using position sensors increases system cost in competitive automotive industry, increases the EPAS size and reduces the reliability. Therefore, it is desirable to estimate the motor angle measurement. To eliminate the need of motor angle measurements, several approaches have been proposed to estimate motor position and/or speed [7][8][9][10][11][12].

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Conventionally, in EPAS system, the driver torque is not measured directly. Thus, the measured torque is used as approximation of the driver torque to determine the desired assist torque. However, the driver torque and the measured torque are different in transition dynamics and it is difficult to determine exactly the values of steering column stiffness and steering column inertia [3]. Moreover, the torque sensor introduces an element of compliance and reduces the steering column stiffness. Furthermore, it is shown in [13] that the control design with steering torque measurement increases the control requirements and makes the system response slower than the control with driver torque. Road reaction torque also is an important information in the design of the EPAS control system. Road reaction torque can be used to improve the steering wheel returnability and to achieve good steering feel [14]. So, it is highly desirable to estimate driver torque and road reaction torque. Chabaan and Wang [3] proposed a driver torque estimator to determine the desired assist torque. However, the pinion torque measurement and motor speed are used. Also, the estimator used improper transfer function, which are approximated by a stable and a proper transfer functions. Marouf et al. [15] proposed a sliding mode observer with unknown inputs to estimate a driver torque and road reaction force. The observer uses steering wheel angle and motor angle measurements. Parmar and Hung [16] proposed a sensorless optimal control for dual pinion EPAS with brushed DC motor. The control was designed to reduce torsion vibrations. However, the measurement of motor position is required and the generation of the basic assist torque curves, which need the driver torque information, does not discussed. We aim here to estimate the driver torque and road reaction torque without the need of the motor position measurement in order to achieve sensorless control of EPAS.

In this paper, we propose a sensorless sliding mode control for EPAS system with PMSM. The control is designed to generate the assist torque, attenuate the vibrations and improve the damping characteristics. The control is implemented via sliding mode observer with unknown inputs. The controller requires the measurements of steering wheel angle and motor currents.

The remainder of this paper is organized as follows. Section II presents the EPAS system model. In section III, the control objectives and the control design are presented. Section IV describes the observer design. Section V reports the simulation results in order to validate the performance of the control strategy, while section VI presents our conclu-

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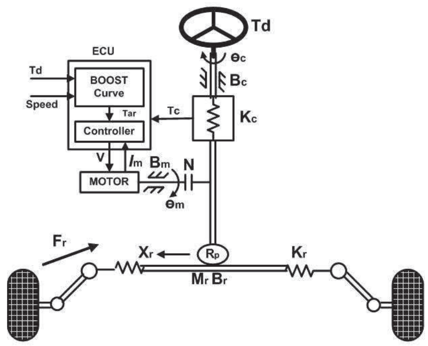

Fig. 1. EPAS Dynamic model

sions.

II. SYSTEM MODELING

A. Dynamic Model of EPAS System

The dynamic model of the EPAS system establishes relation between steering mechanism, electric dynamics of the motor and tire/road contact forces. Figure. 1 shows the model of steering mechanism equipped with brushless PMSM motor. According to Newton equation, the equations of motion can be written as follows:

$$J_c\ddot{\theta}_c = -K_c\theta_c - B_c\dot{\theta}_c + K_c\frac{\theta_m}{N} - F_c sign\left(\dot{\theta}_c\right) + T_d \quad (1)$$

$$\begin{array}{c} J_{eq}\ddot{\theta}_m = K_c\frac{\theta_c}{N} - \left(\frac{K_c}{N^2} + \frac{K_r R_p^2}{N^2}\right)\theta_m - B_{eq}\dot{\theta}_m + T_e \\ -F_m sign\left(\dot{\theta}_m\right) - \frac{R_p}{N}F_r \end{array} \quad (2)$$

The motor's electrical dynamics in the synchronous reference frame (d-q) is given by [6]:

$$\left\{ \begin{array}{l} L_s\dot{i}_d = -R_s i_d + w_e i_q + v_d \\ L_s\dot{i}_q = -R_s i_q - w_e i_d - \psi_m w_e + v_q \\ T_e = 3P\psi_m i_q/2 \ = K_t i_q \end{array} \right. \quad (3)$$

where $J_{eq} = J_m + \frac{R_p^2}{N^2}M_r$, $B_{eq} = B_m + \frac{R_p^2}{N^2}B_r$, $\theta_m = \frac{Nx_r}{R_p}$, $w_e = P\dot{\theta}_m$ and $\theta_e = P\theta_m$. Table I defines and quantifies the other EPAS model parameters. The road reaction torque T_r and the assist torque T_a can be written as:

$$T_r = R_p F_r, \quad T_a = NT_e \quad (4)$$

Considering the equations (1) and (2), the mechanical dynamic equations of EPAS system can be expressed in the following state-space form:

$$\left\{ \begin{array}{l} \dot{x} = Ax + B_1u_1 + B_2u_2 \\ y = Cx = \theta_c \end{array} \right. \quad (5)$$

where $x = \left[\begin{array}{cccc} \theta_c & \dot{\theta}_c & \theta_m & \dot{\theta}_m \end{array} \right]^T$ is the state vector; $u_1 = T_e$ is the electromagnetic torque generated by the motor; $u_2 = \left[\begin{array}{cc} T_d & T_r \end{array} \right]^T$ represents the vector of unknown inputs, which is composed with driver torque and road reaction torque; and y is the measurement signal.

TABLE I

NOMENCLATURE DEFINITIONS

Symbol	Description	Value [units]
T_d	driver torque	
θ_c	steering wheel angle	
θ_m, θ_e	mechanical and electrical motor angle	
$\dot{\theta}_m, w_e$	mechanical and electrical motor speed	
J_c	steering column moment of inertia	$0.04Kg.m^2$
B_c	steering column viscous damping	$0.072N.m./(rad/s)$
K_c	steering column stiffness	$115N.m/rad$
F_c	steering column friction	$0.027N.m$
M_r	mass of the rack	$3kg$
B_r	viscous damping of the rack	$3820N/(m/s)$
R_p	steering column pinion radius	$0.007m$
K_r	tire spring rate	$43000N.m/m$
J_m	motor moment of inertia	$0.0004Kg.m^2$
B_m	motor shaft viscous damping	$0.0032N.m./(rad/s)$
F_m	motor friction	$0.056N.m$
N	motor gear ratio	13.65
Ls	stator inductance	$43e-5H$
Rs	stator resistance	0.019Ω
P	pole pairs	4
ψ_m	magnetic flux	$0.0056Wb$
I_d, I_q	Stator currents in (d-q) frame	A
v_d, v_q	Stator voltages in (d-q) frame	V

B. Vehicle Model

The four-wheel vehicle model is used [17] to generate the road reaction force F_r, which acts on the rack. This vehicle model is used for the simulation (Figure. 2). The vehicle model input is the front wheel angle $(\delta = \theta_c/N_v)$, and the vehicle model output is the road reaction force. To generate this force, Pacejka tire model is used. The block diagram shown in Figure. 2 illustrates the system inputs and outputs.

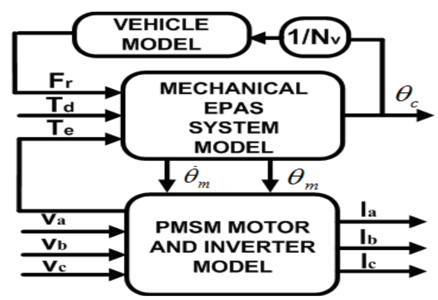

Fig. 2. System block diagram

III. CONTROL DESIGN

A. Control Objectives

The EPAS control must insure the generation of the desired assist torque (Figure. 3), a stable system with large amount of assistance, and the motor's rapid response to the driver torque command [2][3]. The control must also provide damping compensation to attenuate the vibrations and make the steering wheel come back to center without excessive overshoots [1][18][13].

B. Sliding Mode Control Design

Define the reference assist torque [2]:

$$T_{ar} = K_a T_d \ , \quad 0 \leq K_a \leq K_a^{\max} \quad (6)$$

Define the reference currents:

$$I_{qref} = \frac{T_{ar}}{NK_t} = \frac{K_a}{NK_t}T_d, \qquad I_{dref} = 0 \quad (7)$$

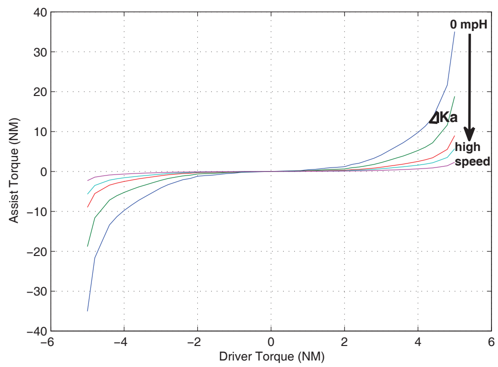

Fig. 3. Assist boost curve characteristics

Let $X = \left[\begin{array}{cccccc} \theta_c & \dot{\theta}_c & \theta_m & \dot{\theta}_m & I_q - I_{qref} \end{array}\right]^T$ the tracking vector and $z = C_q X$ is the desired output. Assuming that the driver torques is slowly time varying $\dot{T}_d = 0$, then the equation dynamics of X is given by

$$\dot{X} = A_a X + B_u v_q + B_w u_2 + B_u(-w_e i_d + d) \qquad (8)$$

with $B_w = \left[\begin{array}{ccccc} 0 & \frac{1}{J_c} & 0 & \frac{K_a}{NJ_{eq}} & 0 \\ 0 & 0 & 0 & \frac{-1}{NJ_{eq}} & 0 \end{array}\right]^T$

$B_u = \left[\begin{array}{ccccc} 0 & 0 & 0 & 0 & 1/L_s \end{array}\right]^T, \quad d = \frac{-R_s K_a}{NK_t} T_d$

$$A_a = \left[\begin{array}{ccccc} 0 & 1 & 0 & 0 & 0 \\ \frac{-K_c}{J_c} & \frac{-B_c}{J_c} & \frac{K_c}{J_c N} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ \frac{K_c}{J_{eq}N} & 0 & -\left(\frac{R_p^2 K_r + K_c}{J_{eq}N^2}\right) & \frac{-B_{eq}}{J_{eq}} & \frac{K_t}{J_{eq}} \\ 0 & 0 & 0 & \frac{-P\psi_m}{L_s} & \frac{-R_s}{L_s} \end{array}\right]$$

We observe that the system (8) exhibits matched disturbances d and mismatched disturbances u_2. To achieve the control objectives we define the following sliding surface:

$$\sigma = \left[\begin{array}{c} \sigma_1 \\ \sigma_2 \end{array}\right] = \left[\begin{array}{c} SX \\ i_d - i_{dref} \end{array}\right] = \left[\begin{array}{c} SX \\ i_d \end{array}\right] \qquad (9)$$

where $S \in \Re^{1\times 5}$ to be designed, SB_u is non-singular.

$$\dot{\sigma} = D_{12}u + F_{12} \qquad (10)$$

with $D_{12} = \left[\begin{array}{cc} SB_u & 0 \\ 0 & \frac{1}{L_s} \end{array}\right], u = \left[\begin{array}{cc} v_q & v_d \end{array}\right]^T$

$F_{12} = \left[\begin{array}{c} SA_a X + SB_w u_2 + SB_u d - SB_u w_e i_d \\ \frac{-R_s}{L_s} i_d + \frac{w_e}{L_s} i_q \end{array}\right]$

- The discontinuous control u_{dis} is given by:

$$u_{dis} = -\rho D_{12}^{-1} \frac{\sigma}{\|\sigma\|} \;, \quad \rho > \|F_{12}\| \qquad (11)$$

Consider the Lyapunov function as: $V = \frac{1}{2}\sigma^t \sigma$.

$$\begin{array}{l} \dot{V} = \sigma^T (D_{12}u + F_{12}) \\ \dot{V} = \sigma^T \left(-\rho \frac{\sigma}{\|\sigma\|} + F_{12}\right) \\ \dot{V} \le -\|\sigma\| (\rho - \|F_{12}\|) < 0 \end{array}$$

So, we guarantee that the system will reach the sliding surface in finite time.

- Sliding mode motion

The equation representing the motion when confined to the sliding surface $\sigma = 0$ $(\sigma_1 = 0, \sigma_2 = i_d = 0)$. When the system reaches the sliding surface $\sigma = 0$, we have

$$\sigma_1 = SX = 0, \;\; \dot{X} = A_a X + B_u v_q + B_w u_2 \qquad (12)$$

The system (12) can be written as:

$$\begin{array}{l} \sigma_1 = SX = S_1 X_1 + X_2 \,, \;\; X_1 = \left[\begin{array}{cccc} \theta_c & \dot{\theta}_c & \theta_m & \dot{\theta}_m \end{array}\right]^T \\ \dot{X}_1 = A_{a11}X_1 + A_{a12}X_2 + B_w u_2 \\ \dot{X}_2 = A_{a21}X_1 + A_{a22}X_2 + \frac{1}{L_s}v_q \end{array} \qquad (13)$$

Now, the objective is to find the matrix S. When $\sigma_1 = 0$, $X_2 = -S_1 X_1$, then

$$\dot{X}_1 = (A_{a11} - A_{a12}S_1)X_1 + B_w u_2 \qquad (14)$$

The system (14) is a standard H_∞ control problem, the state feedback H_∞ gain that stabilizes the system (14), provides desirable performance and guarantees disturbance attenuation $\|z\|_2 \le \gamma \|u_2\|_2$ is given by [19]:

$$S_1 = -A_{a12}^T P \qquad (15)$$

where the symmetric positive definite matrix P is the solution to the algebraic Riccati equation (ARE):

$$PA + A^T P + P\left(\frac{1}{\gamma^2} B_w B_w^T - A_{a12}A_{a12}^T\right)P + Q_c = 0, \;\; Q_c \ge 0 \qquad (16)$$

where the weighting matrix Q is selected as:

$$Q_c = \left[\begin{array}{cccc} q_{11} & 0 & 0 & 0 \\ 0 & q_{22} & 0 & -q_{24} \\ 0 & 0 & q_{33} & 0 \\ 0 & -q_{24} & 0 & q_{44} \end{array}\right] \qquad (17)$$

where q_{11} is a weight on the steering column angle, q_{22} is a weight on the steering column speed, q_{33} is a weight on the motor angle and q_{44} is a weight on the motor speed. An appropriate selection of these parameters enures that good tracking to the desired assist torque and provide damping compensation. Figure. 4 shows the proposed control block diagram.

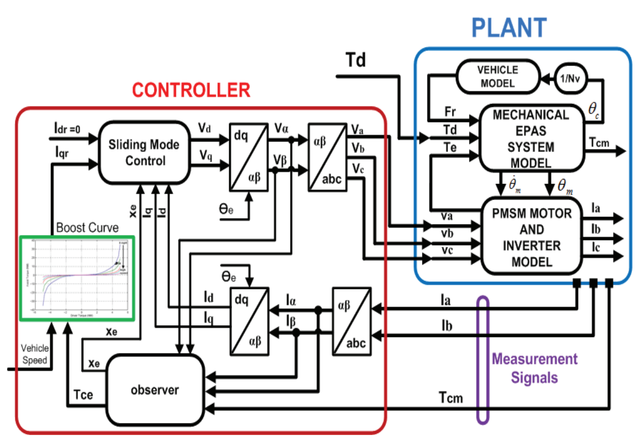

Fig. 4. Controller block diagram

IV. OBSERVER DESIGN

In this section, we propose a sliding mode observer to estimate the driver torque and the road reaction torque (i.e., unknown inputs) using the measurements of steering wheel angle and motor currents. The proposed observer is based on the sliding mode observer, Walcott and Zak [20], used in conjunction with higher order sliding mode differentiators to estimate the additionally outputs needed to construct the observer; since the system doesn't satisfy the observer matching condition.

A. The Sliding-Mode Observer with Unknown Inputs

The Sliding-Mode Observers (SMO) have been used for their robustness, minimization of error dynamics, finite time convergence and reconstruction of unknown inputs [21][20][22][23]. In the literature, it is well known that the observer matching condition is required to construct the observer. To extend the existence of these observers when the observer matching condition does not satisfied, the idea of generating additionally outputs using Higher Order Sliding Mode Differentiators (HOSMD) was proposed [23].

The system (5) is observable, but don't satisfy the observer matching condition $(rank(CB_2) = rank(B_2))$. Therefore, we can't construct the SMO directly. The relative degree of the driver torque with respect to the measurement signal (i.e., steering wheel angle) is 2. So we need to estimate the steering wheel speed using HOSMD. If the motor position and speed are available in finite time, then we can design the sliding mode observer [20][23].

In order to estimate the states and the unknown inputs of the system (5), the following sliding mode observer is proposed:

$$\dot{\hat{x}} = A\hat{x} + B_1 u_1 + L\left(y_a - \hat{y}_a\right) - B_2 E\left(\hat{y}_a, y_a\right) \tag{18}$$

with $y_a = \left[\begin{array}{cccc} y_1 & y_{12} & y_2 & y_{22} \end{array}\right]^T$, y_1 is the measured steering wheel angle, (y_{12}, y_2, y_{22}) are the additionally outputs (steering wheel speed, motor angle and motor speed); $\hat{y}_a = C_a\hat{x},\ C_a = I_4$; and $E\left(\hat{y}_a, y_a\right)$ is the discontinuous output injection:

$$E\left(\hat{y}_a, y_a\right) = \begin{cases} \eta\frac{F(\hat{y}_a - y_a)}{\|F(\hat{y}_a - y_a)\|_2} & for\, F\left(\hat{y}_a - y_a\right) \neq 0 \\ 0 & for\, F\left(\hat{y}_a - y_a\right) = 0 \end{cases}$$

η is a positive constant larger than the upper bound of u_2 and the pair of matrices (P, F) satisfying (19) and (20) for some L and Q , where P and Q are matrices symmetric positive definite (see [20] for more details).

$$\left(A - LC_a\right)^T P + P\left(A - LC_a\right) = -Q \tag{19}$$

$$FC_a = B_2^T P \tag{20}$$

In order to obtain the steering wheel speed y_{12} from the measured steering wheel angle, HOSMD is used. The main advantages of this differentiator are that it is exact and robust with respect to measurement errors, input noises, and finite convergence process time [24]. The estimation of the motor speed and position is discussed in the next section. Fig 5. shows the architecture of the proposed sliding-mode observer.

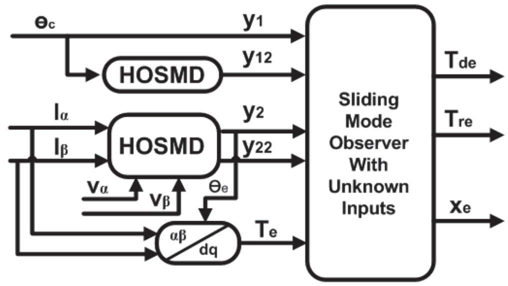

Fig. 5. Block diagram of the proposed sliding-mode observer

B. Motor Speed and Position Estimation

The observer required here must provide finite-time observation of the motor speed and position. For this purpose, we use the HOSMD based on the motor currents measurement and the electrical motor model. Considering that the electrical dynamics is faster than the mechanical one and the finite-time convergence characteristics of the proposed HOSMD observer, this objective can be achieved. The current dynamic equation of PMSM in the stationary reference frame $(\alpha - \beta)$ can be written as follows:

$$\frac{di_{\alpha\beta}}{dt} = \frac{-R_s}{L_s} i_{\alpha\beta} + \frac{1}{L_s} v_{\alpha\beta} - \frac{1}{L_s} e_{\alpha\beta} \tag{21}$$

where $i_{\alpha\beta} = \left[\begin{array}{cc} i_\alpha & i_\beta \end{array}\right]^T$, $v_{\alpha\beta} = \left[\begin{array}{cc} v_\alpha & v_\beta \end{array}\right]^T$, and $e_{\alpha\beta} = \left[\begin{array}{cc} e_\alpha & e_\beta \end{array}\right]^T$ are, respectively, the stator currents vector, the stator voltages vector, and the Back Electromotive Forces (BEMFs) vector in the stationary reference frame $(\alpha - \beta)$.

$$e_\alpha = -\psi_m w_e sin\left(\theta_e\right),\ \ e_\beta = \psi_m w_e \cos\left(\theta_e\right) \tag{22}$$

$e_{\alpha\beta}$ are the unknown inputs and currents $i_{\alpha\beta}$ are measured. The high-order sliding mode observer proposed by Fridman et al. [25] for system (21) is the following:

$$\begin{cases} \dot{z}_{\alpha\beta} = \frac{-R_s}{L_s} z_{\alpha\beta} + l_o(i_{\alpha\beta} - z_{\alpha\beta}), \\ \dot{v}_{1\alpha\beta} = w_{1\alpha\beta} = -2M^{1/3}\left|v_{1\alpha\beta} - i_{\alpha\beta} + z_{\alpha\beta}\right|^{2/3} \cdot \\ \qquad sign\left(v_1 - i_{\alpha\beta} + z_{\alpha\beta}\right) + v_{2\alpha\beta} \\ \dot{v}_{2\alpha\beta} = w_{2\alpha\beta} = -1.5M^{1/2}\left|v_{2\alpha\beta} - w_{1\alpha\beta}\right|^{1/2} \cdot \\ \qquad sign\left(v_{2\alpha\beta} - w_{1\alpha\beta}\right) + v_{3\alpha\beta} \\ \dot{v}_{3\alpha\beta} = -1.1M sign\left(v_{3\alpha\beta} - w_{2\alpha\beta}\right) \\ \hat{e}_{\alpha\beta} = -L_s(v_{2\alpha\beta} - (\frac{-R_s}{L_s} - l_o)v_{1\alpha\beta}) \\ \hat{\dot{e}}_{\alpha\beta} = -L_s(v_{3\alpha\beta} - (\frac{-R_s}{L_s} - l_o)v_{2\alpha\beta}) \end{cases} \tag{23}$$

where $l_o > \frac{R_s}{L_s}$ and $M > \frac{1}{L_s}\left|e_{\alpha\beta}\right|$. According to Fridman et al. [25], the observer (23) provides exact finite-time identification of smooth $e_{\alpha\beta}$. Using the estimated value of $e_{\alpha\beta}$ and $\dot{e}_{\alpha\beta}$, the electrical motor speed and position can be calculated as follows.

$$\hat{w}_e = \left(\psi_m\right)^{\frac{-2}{3}}\left(\hat{e}_\alpha\hat{\dot{e}}_\beta - \hat{e}_\beta\hat{\dot{e}}_\alpha\right)^{\frac{1}{3}} \tag{24}$$

$$\hat{\theta}_e = \int_0^t \hat{w}_e dt + \theta_{e0}\,,\ \ \theta_{e0} = \arctan_2\left(\frac{-\hat{e}_{\alpha 0}}{\hat{e}_{\beta 0}}\right) \tag{25}$$

Then the mechanical motor speed and position can be calculated as follows.

$$\hat{\theta}_m = P\hat{\theta}_e \,, \;\; \hat{\dot{\theta}}_m = P\hat{w}_e \tag{26}$$

C. Overall HOSMDs/Sliding-Mode Observer

According to the equations (5) and (18), the error dynamics of the observer is given by:

$$\dot{e} = Ae + B_1u_1 + L\left(y_a - \hat{y}_a\right) - B_2E\left(\hat{y}_a, y_a\right) \tag{27}$$

where $e = x - \hat{x}$. After a finite time T, the HOSMDs converge; from that moment the following condition hold:

$$y_a = C_a x \tag{28}$$

Thus for all $t > T$ the error dynamics (27) are given by:

$$\dot{e} = \left(A - LC_a\right)e - B_2u_2 - B_2E\left(\hat{y}_a, y_a\right) \tag{29}$$

Since $rank\left(C_aB_2\right) = rank\left(B_2\right)$ and the invariant zeros of the triple $\left(A, B_2, C_a\right)$ remains in the left half plan, the design methods given in [23][20][22] can be applied, so the state estimation error e, enters sliding mode along $\{e : S = FC_ae = 0\}$ after a finite time (see [23][20][22] for more details) and the unknown inputs vector is given by:

$$u_2 = -E\left(\hat{y}_a, y_a\right) \tag{30}$$

V. SIMULATION RESULTS

The purpose of this section is to assess the feasibility and the effectiveness of the proposed approach. Figure. 4 illustrates the simulation block diagram implemented using Matlab/Simulink. White noise was added to the measurement signals and the system inputs, using Simulink uniform noise generator. The magnitude of noise added to the measurement signals is 5% of the signal magnitude with a sampling time of $0.001s$. The magnitude of noise added to the road reaction force is 10% of the signal magnitude with the sampling time of $0.001s$ and the magnitude of noise added to the driver torque is 10% of the signal magnitude with the sampling time of $0.1s$. In the following simulations, the driver steers the steering wheel left from 0 to $4Nm$, right $4Nm$ back to $3Nm$ and then releases the steering wheel (Figure. 8). The vehicle speed is set to $10m/s$ when the amount of assistance is high.

Figures 6 and 7 compare the estimated EMBFs and electrical motor angle to their simulated values. We can see that the estimation values are closer to that simulated ones. Therefore, the estimation results of the HOSMD observer are conclusive. Figure. 8(a)(b) shows the comparison between the simulated driver torque and road reaction force and their estimated values. It can be seen that the unknown inputs can be reconstructed with good accuracy. Also, it can be seen in figure. 8(c)(d) that the steering column velocity and the mechanical motor speed can be estimated with good accuracy. These curves show the robustness of the observer and the differentiators versus measurement noise. Figure. 9 shows the response of the EPAS plant to the driver's torque command with the proposed control. In figure 9(a), we can see that the road disturbances (noise added to road reaction torque) are not transmitted to the driver via the steering torque and the driver can get a good steering feeling. It can be seen in 9(c) that the steering wheel angle returns to its center position rapidly without overshoots after that the driver releases the steering wheel. Also, no undesired vibrations shown on the steering wheel velocity and motor speed after

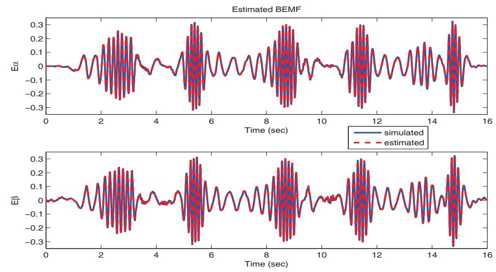

Fig. 6. EMBFs estimation

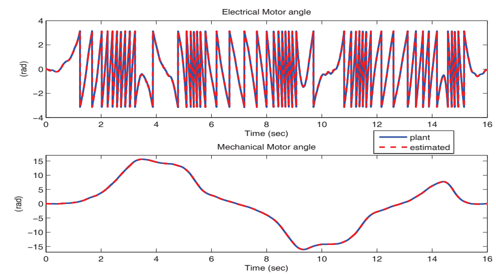

Fig. 7. Motor angle estimation

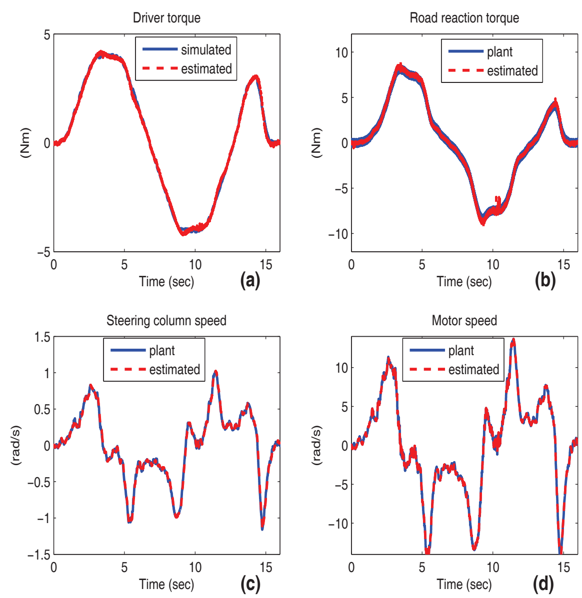

Fig. 8. Estimation performance

the driver released the steering wheel. The figure. 9(b) shows the comparison between the reference assist torque and the generated assist torque. It can be seen that the control method provides a good response to the reference assist torque. The figure. 9(d) shows the relationship between the driver torque and the generated assist torque produced by the proposed structure. It can be seen that the shape is close to the desired boost curve (figure. 3).

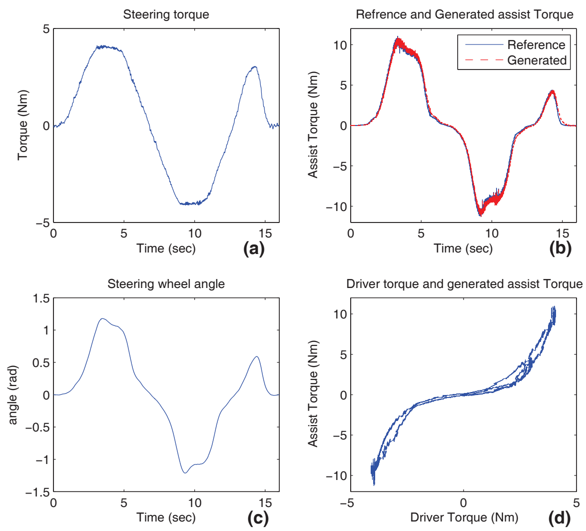

Fig. 9. Control performance

VI. Conclusion

In this paper, we have proposed a sensorless control of EPAS system with PMSM. The control is designed using sliding mode control via H-infinty to attenuate the mismatched disturbances and improve damping characteristics of EPAS. The sensorless control is implemented using sliding mode observer with unknown inputs. Simulation results have shown the feasibility of the proposed controller. In the future work, the controller will be validated experimentally with the necessary hardware modifications using experimental simulator actually under development.

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