Jurnal Teknologi

Full paper

Energy Optimization of Brushed DC Motor in Electric Power-Assisted Steering

Kumeresan A. Danapalasingam,[a,b*]

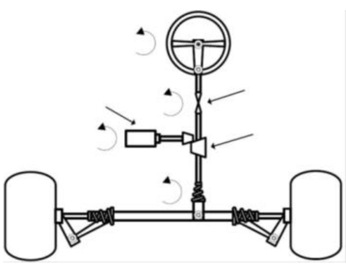
[a]Senior Lecturer, Department of Control & Mechatronics Engineering, Faculty of Electrical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia
[b]Associate Fellow, UTM Centre for Industrial and Applied Mathematics, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia

*Corresponding author: kumeresan@fke.utm.my

Article history

Received :5 February 2014
Received in revised form :
7 April 2014
Accepted :20 May 2014

Graphical abstract



Abstract

Electric power-assisted steering (EPS) is a control system where an electric motor is used to provide assistance in vehicle steering. In this work controllers are designed for a column-type EPS equipped with a brushed DC motor to enable energy optimization. Using a mathematical model of EPS a controller is developed based on nonlinear adaptive regulation method to generate driver torque. PID control is then applied to produce assistance torque in accordance to desired energy saving. Simulation results using Matlab show the trade-off between driver's comfort and energy consumption. The control paradigm introduced here fits appropriately in electric vehicles (EVs) where electrical energy is scarce.

Keywords: Electric power-assisted steering; brushed DC motor; nonlinear adaptive regulation; PID control; energy optimization

1.0 INTRODUCTION

In a vehicle EPS assists a human driver by measuring torque exerted by the driver and providing additional torque by means of an electric motor. Conventionally an EPS system relies on vehicle velocity and driver torque measurements to determine target motor current levels from assist characteristic or boost curve.[1,2] The assist characteristic curve is a set of graphs of motor current versus driver torque and vehicle velocity that is generated experimentally to produce desired assistance torque and thus predetermined steering comfort.[3] While in internal combustion engine vehicles provision of maximum steering comfort to drivers is possible careful considerations are need to be made in EVs. Given a limited power supply in an EV a higher torque assistance would result in a faster battery drain. Therefore the use of the assist characteristic curve to obtain reference assistance torque levels is not applicable in EVs and a better approach enabling energy optimization is justified.

Some previous works on EPS will now be reviewed. In Ref. 3 a modified Linear Quadratic Gaussian controller is shown to be able to track the characteristic curve and attenuate external disturbances for a column-type EPS system. From the simulation results the controller performs well despite the inclusion of nonlinear rotational friction terms even though explicit expressions of the friction are not provided. Chitu *et al*. applied a Linear Quadratic Regulator to derive an optimal controller for an EPS system in Ref. 2. Both simulation and dSPACE ControlDesk real-time application utilizing the assist characteristic curve show stability in frequency, robustness and closed-loop stable step responses during parameter variation. In Ref. 4 a Fuzzy PID control strategy is simulated for assist motor current tracking. The simulation model that includes a simple road surface disturbance demonstrates the effectiveness of the proposed controller.

Out of the four control objectives of an EPS system (cf. Ref. 5) the basic function of an EPS i.e. assistance torque control is considered here. To achieve energy optimization the authors develop a mathematical model of a column-type EPS with a brushed DC assist motor. Nonlinear rotational friction and LuGre dynamic tire friction are included in the mathematical model for accurate EPS representation. The model is then used to design a controller based on nonlinear adaptive regulation for driver torque generation to track a reference wheel angle trajectory. To enable an option to select torque assist level and consequently amount of energy saving desired eco factor E is introduced in later sections of this paper. Target assistance torque is then achieved using PID control of motor current with Pulse Width Modulation (PWM) implementation.

The paper is organized as follows; in Section 2.0 the EPS mathematical model is discussed. A brief explanation on controller designs are provided in Section 3.0 after which Simulation Results and Conclusion are given in Sections 4.0 and 5.0 respectively.

2.0 MATHEMATICAL MODEL

The following is the mathematical model of a column-type EPS (see Figure 1).

Torque sensor model:

$$T_c = K_s(\theta_s - \frac{x_r}{R_s}) \tag{1}$$

Steering column model:

$$J_s\ddot{\theta}_s = T_d - T_c - B_s\dot{\theta}_s - T_f^s \tag{2}$$

Assist motor model:

$$T_m = K_a i_a$$
$$(J_m + \frac{J_G}{G^2})\ddot{\theta}_m = T_m + \frac{T_c}{G} - (B_m + \frac{B_G}{G^2})\dot{\theta}_m - T_f^m$$
$$L\dot{i}_a = V - Ri_a - K_b\dot{\theta}_m$$

Rack and pinion model:

$$\begin{aligned} T_a &= GT_m \\ M_r\ddot{x}_r &= \frac{T_a + T_c}{R_s} - F_{\mathrm{TR}} - B_r\dot{x}_r - K_r x_r, \end{aligned} \tag{3}$$

where

θ_s	-	steering wheel angle
θ_m	-	assist motor armature shaft angle
T_d	-	driver torque
T_c	-	steering column torque sensor measurement
T_m	-	assist motor torque
T_a	-	assistance torque
T_f^s	-	steering column friction
T_f^m	-	assist motor and reduction gear friction
B_s	-	steering column viscous damping coefficient
B_m	-	assist motor viscous damping coefficient
B_G	-	reduction gear viscous damping coefficient
B_r	-	pinion and rack viscous damping coefficient
J_s	-	steering column moment of inertia
J_m	-	assist motor moment of inertia
J_G	-	reduction gear moment of inertia
K_s	-	steering column stiffness
K_a	-	assist motor torque coefficient
K_b	-	assist motor back emf coefficient
K_r	-	rack equivalent spring constant
V	-	assist motor voltage
i_a	-	assist motor current
R	-	assist motor resistance
L	-	assist motor inductance
G	-	reduction gear ratio
x_r	-	horizontal rack displacement
R_s	-	pinion radius
M_r	-	pinion, rack and wheel equivalent mass
F_{TR}	-	dynamic tire friction

The nonlinear rotational friction is given by

$$\begin{aligned} T_f^i &= (\alpha_0^i + \alpha_1^i e^{-\alpha_2^i|\dot{\theta}_i|})\mathrm{sgn}_1(\dot{\theta}_i) + \\ &\quad (\alpha_3^i + \alpha_4^i e^{-\alpha_5^i|\dot{\theta}_i|})\mathrm{sgn}_2(\dot{\theta}_i), \quad i = s, m, \end{aligned}$$

where

$$\mathrm{sgn}_1(\dot{\theta}_i) = \begin{cases} 1 & \dot{\theta}_i \geq 0 \\ 0 & \dot{\theta}_i < 0 \end{cases},$$
$$\mathrm{sgn}_2(\dot{\theta}_i) = \begin{cases} 0 & \dot{\theta}_i \geq 0 \\ -1 & \dot{\theta}_i < 0 \end{cases}, \quad i = s, m$$

and $\alpha_i^j \in \mathbb{R}, \alpha_i^j > 0, j = 0, \ldots, 5$ and $\alpha_0^i \neq \alpha_3^i, \alpha_1^i \neq \alpha_4^i, \alpha_2^i \neq \alpha_5^i, i = s$ in general. [6]

The LuGre dynamic tire friction can be expressed as

$$\tau_a = M_s + M_{sa},$$
$$F_{\mathrm{TR}} = \frac{\tau_a}{r},$$

where r is the steering arm length and M_s and M_{sa} are sticking and self-alignment torques respectively. For complete expressions of M_s and M_{sa} please refer to Refs. 7 and 8.

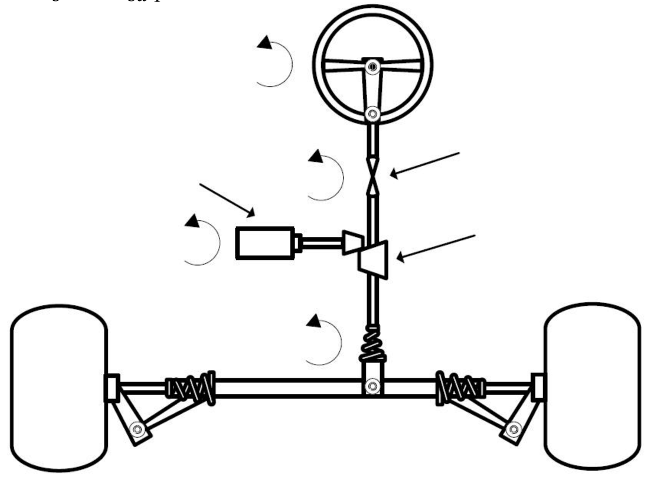


Figure 1 A column-type EPS

3.0 CONTROLLER DESIGN

The main objective of this work is to demonstrate the option made available to choose a desired steering comfort. Since a better steering feel is provided by a higher torque assist a higher energy consumption would result and the reverse is also true. The amount of assistance torque T_a to be produced is determined by the eco factor E and driver torque T_d. Therefore in this section the controller design is done in two stages. In the first part a controller based on nonlinear adaptive regulation is developed to generate driver torque T_d to track a reference wheel angle trajectory. Next a reference assistance torque T_a^* is obtained using the generated T_d and E. In the second stage DC motor control is carried out using PID and PWM application to track T_a^*.

3.1 Driver Torque Generation

Consider (3) with $T_a = 0$,

$$M_r \ddot{x}_r = \frac{T_c}{R_s} - F_{\text{TR}} - B_r \dot{x}_r - K_r x_r \tag{4}$$

and a reference wheel angle trajectory θ_w^* given by

$$\begin{aligned} x_r^* &= \sum_{i=1}^{N} A_i \cos(\Omega_i t + \varphi_i), \\ \theta_w^* &= \frac{x_r^*}{r}, \end{aligned} \tag{5}$$

with a fixed N, unknown amplitudes A_i, phases φ_i and frequencies Ω_i. Then from (4) and (5) a steady state steering column torque T_c^* needed to track θ_w^* is given by

$$T_c^* = (F_{\text{TR}} + B_r \dot{x}_r + K_r x_r + M_r \ddot{x}_r^*) R_s. \tag{6}$$

From (5) and (6) a reference steering wheel angle is obtained as follows.

$$\theta_s^* = \frac{T_c^*}{K_s} + \frac{x_r}{R_s}. \tag{7}$$

By solving for a steady state driver torque T_d^* from (2) using (7) we have

$$T_d^* = T_c + B_s \dot{\theta}_s + T_f^s + J_s \ddot{\theta}_s^*. \tag{8}$$

Note that in steady state T_d^* will produce θ_s^*, T_c^* and consequently x_r^*.

Using the nonlinear adaptive regulation method expressions (7) and (8) yield controllers of the forms

$$\begin{aligned} e_1 &= x_r^* - x_r, \\ u_{\text{st}}^1 &= k_2^1(\dot{e}_1 + k_1^1 e_1), \\ g_{\text{st}}^1 &= G u_{\text{st}}^1, \\ \dot{\xi}_1 &= (F + G\hat{\Psi}_1)\xi_1 + g_{\text{st}}^1, \\ T_c^u &= \hat{\Psi}_1 \xi_1 + u_{\text{st}}^1, \quad k_1^1, k_2^1 > 0 \end{aligned} \tag{9}$$

and

$$\begin{aligned} \theta_s^u &= \frac{T_c^u}{K_s} + \frac{x_r}{R_s}, \\ e_2 &= \theta_s^u - \theta_s, \\ u_{\text{st}}^2 &= k_2^2(\dot{e}_2 + k_1^2 e_2), \\ g_{\text{st}}^2 &= G u_{\text{st}}^2, \\ \dot{\xi}_2 &= (F + G\hat{\Psi}_2)\xi_2 + g_{\text{st}}^2, \\ T_d &= \hat{\Psi}_2 \xi_2 + u_{\text{st}}^2, \quad k_1^2, k_2^2 > 0 \end{aligned} \tag{10}$$

whose performances are subject to the tuning of k_1^1, k_2^1, k_1^2 and k_2^2. For definitions of the control parameters and detailed explanation of control designs (9) and (10) readers are advised to refer to Refs. 9 and 10. The above controllers can be shown to be able to track x_r^* in a globally asymptotically and locally exponentially stable manner.

3.2 Assist Motor Control

The controller designed in the previous section will generate driver torque T_d for reference wheel angle θ_w^* tracking. From the steering column torque measurement T_c desired assist motor torque T_m^* is obtained as follows.

$$\begin{aligned} T_a^* &= E T_c, \\ T_m^* &= \frac{T_a^*}{G}. \end{aligned} \tag{11}$$

Given the value in (11) reference assist motor current $i_a^* = T_a^*/K_a$ is then computed for DC motor control as depicted in Figure 2. A PID controller is tuned to achieve desired accuracy in i_a^* tracking of the assist motor that is powered by PWM.

Note that $E = 1$ would mean that equivalent steering effort has to be put by both a human driver and the assist motor. Even though that results in a comfortable steering it leads to a higher battery usage as compared to setting $E = 0.1$.

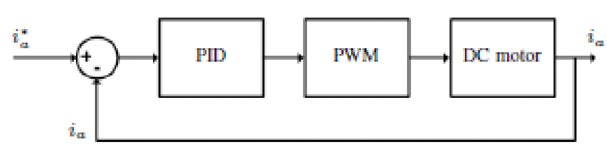


Figure 2 Assist motor control

4.0 SIMULATION RESULTS

The mathematical model of a column-type EPS system given in Section 2.0 together with the control designs in Section 3.0 is simulated using Matlab. Parameters of the EPS model is adopted from Refs. 4, 7 and 8.

Recall that in Section 3.1 a controller is designed to generate driver torque T_d for reference wheel angle θ_w^* tracking. From Figure 3 it could be seen that controllers (9) and (10) performs well since a very close tracking is achieved.

In Figure 4 steering column torque T_c and assistance torque T_a are plotted for $E = 0.6$. As the values of T_a are always approximately 60% of T_c as desired the effectiveness of the PID controller with PWM from Section 3.2 is demonstrated.

Figures 5 and 6 show steering column torque T_c and assist motor torque T_m respectively for different values of E. Note the inverse relationship between T_c and T_m. As a higher T_m is desired (by setting a higher value of E) a lower T_c is required for steering.

As expected assist motor current i_a is proportional to T_m in Figure 7. From Figures 5 and 7 it is clearly shown that a lighter steering feel (a higher E) requires a higher current draw from a car battery. However if users are given the option to choose E accordingly a better management between steering comfort and power usage could be achieved for energy optimization.

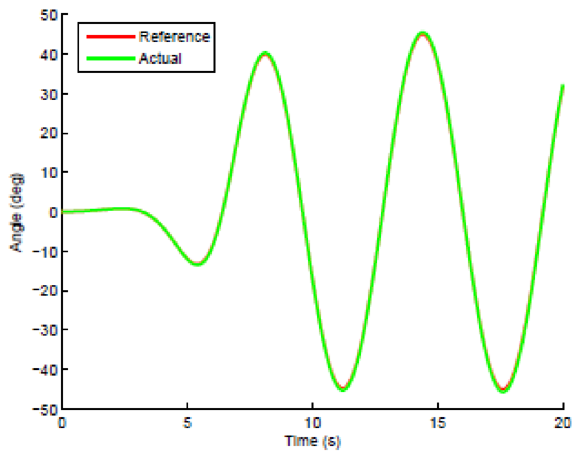


Figure 3 Reference wheel angle θ_w^* and actual wheel angle θ_w

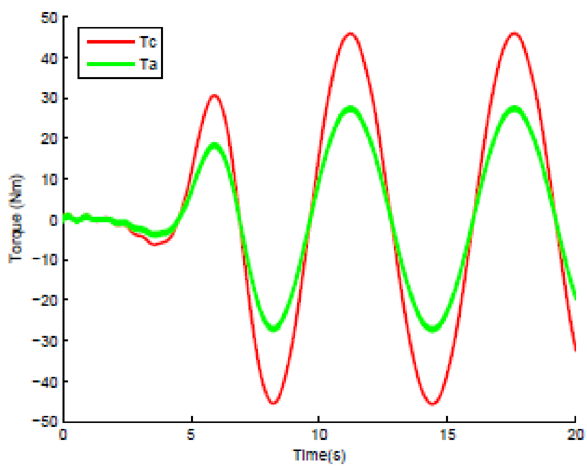


Figure 4 Steering column torque T_c and assistance torque T_a for $E = 0.6$

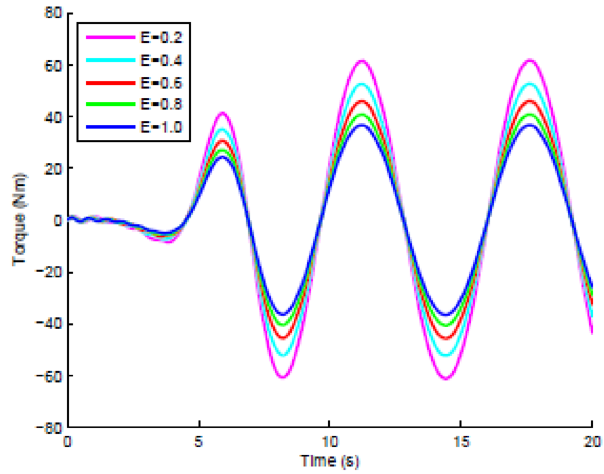


Figure 5 Steering column torque T_c

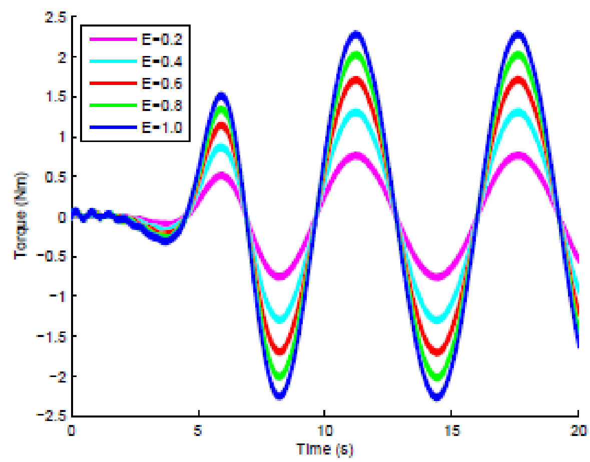


Figure 6 Assist motor torque T_m

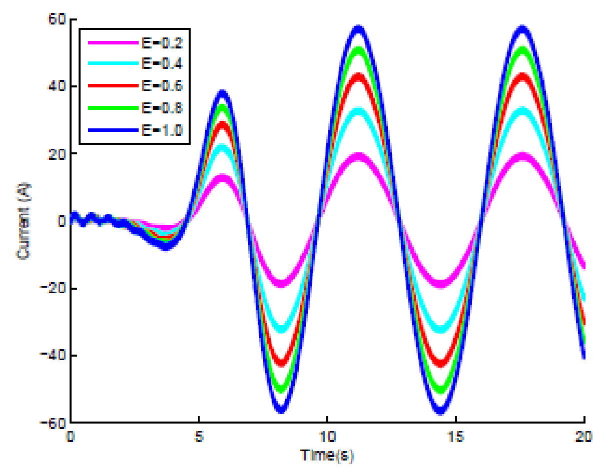


Figure 7 Assist motor current i_a

■4.0 CONCLUSION

Energy optimization of EPS in EVs has been the primary focus of this paper. In this work the conventional method of determining target assist motor current using a lookup table is completely eliminated to give way to an approach enabling battery energy saving. Rather than using fixed values of reference assist motor current to generate required assistance torque here the authors enabled an option to set the level of steering comfort as desired. This flexibility is made available by means of the *eco factor E* introduced in this paper. As a consequence drivers could choose to save battery energy given the inverse relationship between steering comfort and power consumed by the assist motor. Simulation results verify the feasibility of the proposed control methods in achieving energy optimization in EPS.

Acknowledgement

This work was supported by the Fundamental Research Grant Scheme (R.J130000.7823.4F273).

References

[1] H. Zhang, Y. Zhang, J. Liu, J. Ren, and Y. Gao. 2009. Modeling and Characteristic Curves of Electric Power Steering System. In *Proceedings of the International Conference on Power Electronics and Drive Systems, 2009 (PEDS 2009)*. 1390–1393.

[2] C. Chitu, J. Lackner, M. Horn, H. Waser, and M. Kohlbock. 2011. A Robust and Optimal Lqr Controller Design for Electric Power Steering System. In *Proceedings of the 2011 Joint 3rd Int'l Workshop on Nonlinear Dynamics and Synchronization (INDS) & 16th Int'l Symposium on Theoretical Electrical Engineering (ISTET)*. 1–5.

[3] N. Mehrabi, N. L. Azad, and J. McPhee. 2011. Optimal Disturbance Rejection Control Design for Electric Power Steering Systems. In *Proceedings of the 2011 50th IEEE Conference on Decision and Control and European Control Conference (CDC-ECC)*. 6584–6589.

[4] H. Zang and S. Chen. 2011. Electric Power Steering Simulation Analyse Based on Fuzzy PID Current Tracking Control. *Journal of Computational Information Systems*. 7:. 119–126.

[5] Q. Liu, H. Chen, and H. Zheng. 2007. Robust Control of Electric Power Steering System. In *Proceedings of the 33rd Annual Conference of the IEEE Industrial Electronics Society (IECON)*. 874–879.

[6] T. Kara and I. Eker. 2004. Nonlinear Modeling and Identification of a DC Motor for Bidirectional Operation with Real Time Experiments. *Energy Conversion and Management*. 45: 1087–1106.

[7] J. Tordesillas, V. Ciarla, and C. Canudas De Wit. 2011. Oscillation Annealing and Driver/Tire Load Torque Estimation in Electric Power Steering Systems. In Proceedings of the 2011 IEEE Multi-Conference on Systems and Control (MSC 2011), Denver, Colorado, United States, Sep. 2011, p. s/n. [Online]. Available: http://hal.archives-ouvertes.fr/hal-00642035.

[8] E. Velenis, P. Tsiotras, and C. C. de wit. 2002. Extension of the LuGre Dynamic Tire Friction Model to 2d Motion. In Proceedings of the 10th IEEE Mediterranean Conference on Control and Automation-MED2002, July 9-12 2002. Lisbon, Portugal.

[9] A. Isidori, L. Marconi, and A. Serrani. 2003. *Robust Autonomous Guidance*. Secaucus, NJ, USA: Springer-Verlag New York, Inc.,

[10] A. Isidori, L. Marconi, and A. Serrani. 2003. Robust Nonlinear Motion Control of a Helicopter. *IEEE Trans. Automat. Control*. 48(3): 413–426 [Online].