

Adaptive Network Trained Controller for Automotive Steering Systems

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ABSTRACT

Electrical Power Assist Steering (EPAS) systems are currently eliminating the traditional hydraulic steering systems in vehicles. EPAS systems are nonlinear Multi Input Multi Output (MIMO) systems with multiple objectives, including fast response to the driver torque command, good driver feel, and attenuation of load disturbance and sensor noises. Optimal control method is employed to design EPAS system controllers for improved performance and robustness. But these controllers have showed acceptable performance for certain operating conditions and undesired steering feel for high steering gain. In this work, the neural networks are used which replace the optimal controllers of EPAS systems. A Euclidean adaptive resonance theory (EART) networks is trained according to the data collected from an H^∞ optimal controller. The collected data represent the controller input and output signals. The said data are normalized and clustered into categories in the EART modules. The modules are interconnected by a map field. Once the training is accomplished, the EART controller replaces the optimal controller. The proposed controller provides improved robustness and comparatively high steering feel of EPAS system by reducing the amount required for intensive calculation. The rms value of the error signal with 75 number of clusters is lower than that of 15 clusters. The proposed technique is applicable to any arrangements of EPAS namely, rack, pinion and column EPAS.

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INTRODUCTION

The automotive industries are directing significant attention to technical progressions in the vehicle safety, dynamics and stability. In automotive business, the vehicle dynamics plays an important role in the development of a vehicle program. The security and the relief of the passengers are playing very important role in which the steering system is becoming one of the most important fragments of a vehicle development. The steering system of passenger cars are mainly classified into three groups i.e. mechanical steering system, hydraulic steering system and electric power assist steering (EPAS) system. An EPAS system is a feedback control system that electrically amplifies the driver steering torque inputs to the vehicle for improved steering comfort and performance. An EPAS system consists of a steering wheel, a column, a rack, an electric motor, a gearbox assembly, as well as pinion torque, position, and speed sensors. The essential operation of an EPAS system can be depicted in the system diagram shown in [Figure 1](#).

Currently, all EPAS systems employ a pinion torque sensor, between the steering column and pinion, to determine the amount of the torque assist to the driver [1] - [5]. This torque assist is calculated via a tunable nonlinear boost curve [6] - [8]. Then, this signal is used as a control command to the electric motor to achieve a desirable level of assist. A control design must ensure several criteria so that the overall

steering feel and performance will be similar to or better than a conventional hydraulic steering system. Control of EPAS systems is a very challenging task, due to several factors. First, road conditions and behavior of a driver are not predictable. In order to maintain consistency in the steering feel over a large range of the driver's torque, such as when the driver rotates the wheel and produces very high assist torque, the slopes of the boost curve change dramatically [6]. This will affect significantly overall system stability and performance in a fundamental manner. Second, there are several nonlinear components in the system such as friction, saturation and backlash [7]. These nonlinearities change as mechanical systems degrade and system conditions vary (lubrications, temperature, etc.). Finally, road disturbances and sensor noises are significant [10]. The system must be capable of maintaining robustness and eliminating undesirable vibrations or nibble (vertical vibrations in the steering wheel) that the driver might feel.

Analysis of stability of EPAS systems has been reported on several EPAS products. In [8], analytical expressions of transfer functions are derived to model assist dynamic, steering system compliance, and driver feel of the road for a dual-pinion system. Along a similar direction, analysis of stiffness and feel was reported in [10]. Mathematical equations, which relate rack forces to the driver torque and steering angles, were developed to help EPAS gear engineers design an appropriate boost curve. These mathematical equations can

assist in determining the regions of boost curves for which the steering feel varies as a function of vehicle speed. In [11], the authors studied the problem of power steering ‘road feel’. The relationship between the pump pressures vs. steering torque was derived. It was argued that a gear can be designed on the basis of a set of boost curves which can be adapted as a function of vehicle speed. These boost curves can accommodate the desirable reduction of effort at parking and increased effort at straight driving as the vehicle speed increases.

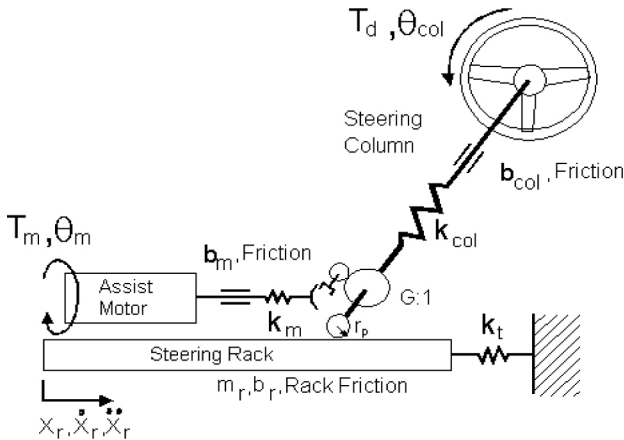


Figure 1. EPAS system block diagram.

Numerous feedback control techniques have been applied to EPAS systems, ranging from traditional lead lag compensation to more advanced control techniques. For example, the work presented by authors in [12] investigated control design by employing an H_∞ -type criterion to provide an appropriate steering feel based on road information. In [6]- [7], several key points in implementing EPAS system in vehicle were described. A three-degree-of- freedom (3-DOF) model of the EPAS was developed. An analysis on the closed-loop system was performed to understand compromises in system design. In [8], it was explained that increasing assist gain reduces steering torque for drivers. However, it may cause undesirable steering vibration at specified frequencies around 30 Hz. Motor angular velocity was needed for this control algorithm. An estimator design was investigated to obtain this signal without installing such a sensor. Two methods were investigated: Observer and back e.m.f.

In recent researches [1]- [15], different control methods are used to design the optimal controllers which provides improved performance and robustness of EPAS system with respect to traditional hydraulic steering system. But these controllers have showed undesired steering feel for high steering gain and acceptable performance for certain operating conditions. So, the aim of the work is to develop a controller which can provide high steering feel and increase in robustness of EPAS system. In this work, the neural networks are used in EPAS system instead of optimal controllers. A Euclidean adaptive resonance theory (EART) networks are trained according to the data collected from an H_∞ optimal controller which represent the controller input and output signals. The said data are normalized and

clustered into categories in the EART modules which are interconnected by a map field. Once the training is accomplished, the EART controller becomes ready to completely replace the optimal controller. The proposed technique is applicable to any arrangements of EPAS namely, rack, pinion and column EPAS with slice modifications in plant and feedback signal.

This work is organized in the following sections. Section II presents the model of 3-DOF nonlinear EPAS system. In section III, the formulation of the control problem is given. Section IV describes the Euclidean adaptive resonance theory (EART) based neural networks. The experimental analysis of EARTMAP neural controller is presented in section V and its performance results are shown in section VI. Finally, Section VII draws the conclusion of the work.

3-DOF NONLINEAR EPAS MODEL

A single pinion EPAS system contains several nonlinear components (Figure 1), such as friction terms in the column, rack, and motor, as well as sticking and backlash in the gearbox. These nonlinear elements can be inserted as required in the simulation environment to accurately represent their nonlinear properties. However, it is possible to approximate these terms by linear ones and treat approximation errors as model uncertainties. Consequently, it is imperative that feedback controllers that are designed on the basis of the linearized systems be sufficiently robust so that they provide stability and satisfactory performance on the original nonlinear system. Following the basic Newton's laws of motions, it can be established that a nonlinear dynamic model of the EPAS system by examining the torque acting on the steering column and the pinion drove electric motor. Finally, by considering the forces acting on the rack, the differential equations of the system can be obtained through equation 1, 2, 3 [5] - [8]. Equations (1), (2), (3) show the single pinion EPAS model having 3-DOF system that are column, rack and electric motor. The input/output relationships of each of these equations are presented by the MATLAB/Simulink model of these equations

A. Column Model

The nonlinear model of the column is described in eq. (1) is shown in Figure 2.

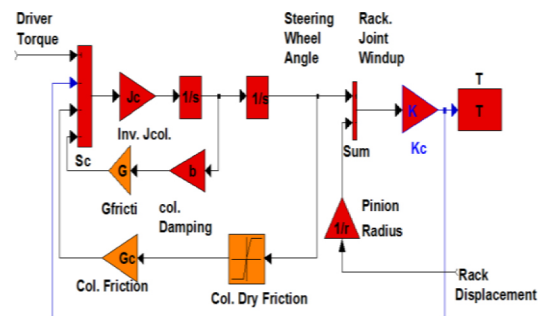


Figure 2. EPAS Column Model.

Driver torque and rack position are the two inputs to the model, and there are five outputs. The outputs are column force (force input to the rack model), column position and velocity, net torque to accelerate the column inertia, and sensed torque. The column is essentially a rotational spring-mass-damper system; consist of the steering column inertia, J_c , the total column damping, b_c , and the torsion spring rate, K_c . The windup of the spring is coupled to the rack displacement through the inverse of the pinion radius. One end of the spring is positioned by the dynamics of the column inertia, and the motion of the rack positions the other end. The difference of these two angles is the net windup of the torsion spring rate of the torque sensor and all other compliance in the column, represented by K_c . The force provided to the rack via the driver's input to the column through the pinion is K_c times the windup, divided by the pinion radius, r_p . The differential equations of the column model can be obtained through [equation 1](#)

$$J_c \ddot{\theta}_c = -b_c \dot{\theta}_c - K_c \theta_c + \frac{K_c}{r_p} X_r + T_d + f_c(\theta_c, \dot{\theta}_c) \quad (1)$$

B. Rack Model

The MATLAB/ Simulink model of the rack based on [eq. \(2\)](#) is shown in [Figure 3](#).

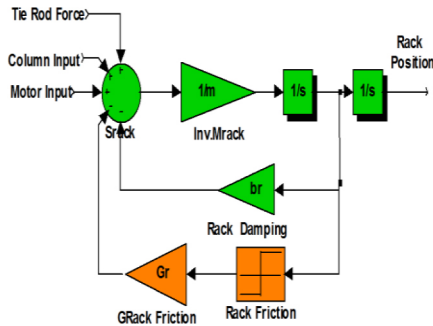


Figure 3. EPAS Rack Model.

The inputs to the model are tie rod force, column input force and motor input force. The three outputs are the rack position, velocity and total input force. Note that the priority of the tie rod force is positive when it opposes the motion of the rack. The parameters associated with the rack model are M (inverse of the rack mass), br (viscous friction or the damping of the rack) and G_r (the dry or static friction of the rack). The rack position output is coupled to the column and the assist actuator or electric motor model, and is the primary output of this block. The differential equations of the rack model can be obtained through [equation 2](#)

$$m \ddot{X}_r = \frac{K_c}{r_p} \theta_c - \frac{K_c}{r_p^2} X_r - b_r \dot{X}_r + \frac{K_m G}{r_p} \theta_m - \frac{K_m G^2}{r_p^2} X_r - K_r X_r + f_r(X_r, \dot{X}_r) \quad (2)$$

C. Electric Motor Model

The model of the electric motor is shown in [Figure 4](#).

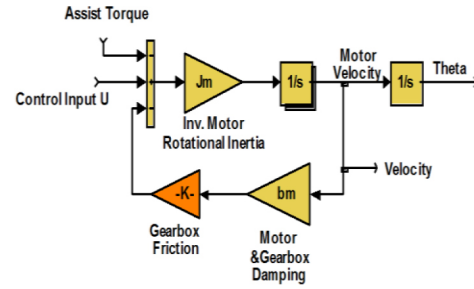


Figure 4. EPAS Electric Motor Model.

[Figure 4](#) shows that there are two main inputs to the model the assist torque T_a and the control input U . The outputs are the motor position and velocity. Note that the inner loop control of the motor is assumed to be of unity gain, implying perfect control of the electric states of the motor. The parameters associated with motor model are Motor rotational inertia J_m , motor and gearbox damping b_m , and gearbox friction gain G_{Br} . The differential equations of the column model can be obtained through [equation 1](#).

$$J_m \ddot{\theta}_m = -b_m \dot{\theta}_m - K_m \theta_m + \frac{K_m G}{r_p} X_r + T_m + f_m(\theta_m, \dot{\theta}_m) \quad (3)$$

The nonlinear system block diagram that describes the 3-DOF EPAS system is shown in [Figure 5](#). This nonlinear model works well for investigating system performance and stability. To investigate controls and to improve system stability this model was linearized and then analyzed with classical design techniques.

The assist torque, T_a can be calculated using the rack displacement and electric motor position as given below:

$$T_a = K_m \left(\theta_m - \frac{G}{r_p} X_r \right) \quad (4)$$

The motor electromagnetic torque input T_m (Nm) is related to motor current input by an inner loop $T_m = G_{in}(s) u$, where u is the motor input and G_{in} is the inner loop transfer function. In the frequency band of interest, $G_{in} \equiv 1$ (per normalization of mechanical units). As a result, one may simply view T_m as the control input. Also, the resistance term $F_t = K_t X_r + F_r$. It has been noted that the resistance force on the rack is mainly $K_t X_r$. F_r will denote the remaining disturbances from road condition changes.

Generically, an EPAS system has two main external inputs: The driver torque T_d that is not measured and the assist motor input current U , which is a control signal. Due to variations in road and tire conditions, sensor inaccuracy and measurement noise, the EPAS system is subject to external disturbances F_r (resistance force on the rack) and d (sensor noises). The dimension of this is identical to the number of sensors utilized in the system.