

A Coupled System Simulator for Electric Power Steering System

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ABSTRACT

Electric power steering system has been widely used in passenger cars to replace hydraulic power steering system. In this paper, a coupled system simulator for dual-pinion electric power steering system is developed and its dynamics is verified using experimental data. The mechanical subsystem model is developed using mass, spring-damper and experimental data with hysteresis characteristic. For the control subsystem, the controller model is developed by transplanting control program from the controller in real vehicle to the simulator. For the electric subsystem, the motor model is developed using the map of current to torque which is created by magnetic field analysis in advance. The three subsystems are combined to form the complete system simulator and numerical simulation is performed. The simulation results are compared to the experimental data and the error is less than 10%. The coupled EPS system simulator produces accurate predictions of system dynamics and can support engineers in the analysis and design of real system effectively.

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INTRODUCTION

In recent years, the electric power steering (EPS) system has been widely used in passenger cars due to its distinct advantages over the conventional hydraulic power steering (HPS) system: better fuel efficiency, tidier layout plan and more friendly to the environment. The EPS system uses an electric motor to generate assist torque only when the torque is required, from which the fuel efficiency can improve 3%-5% compared to the HPS system. In addition, the EPS system eliminates the need for hydraulic pump, hoses, belt and pulley, thus it can provide a tidier layout design in a limited engine room.

There are many research activities on the EPS system about modeling, controlling and verification. A reduced EPS system dynamic model introduced in [1] has been used to analyze various closed-loop effects such as torque performance, disturbance rejection, noise rejection, road feeling and stability. In [2], a simplified EPS system model is developed using Lagrangian dynamics and an optimal controller without torque sensor is proposed. In [3], a

nonlinear EPS model is introduced and an optimal control design is performed using this model, effect of the controller is verified by integrated simulation with CarSim®.

Previous works have been mainly focus on the complicated assist control algorithm design, where a simplified EPS system model is often used to verify the effect of controller. For the mass-production EPS system, the control algorithm is simple, usually using the map designed in advance. On the contrary, a precise coupled simulator still needs to be developed, as the engineers in the R&D department require it to support their production development. In this paper, a coupled EPS system simulator, including mechanical, electric and control subsystem, is proposed. Furthermore, the mechanical subsystem model is developed using the experimental data with hysteresis characteristic, the motor model is developed in advance by magnetic field analysis and the results are integrated to the mechanical subsystem. These methods improve the precision of the simulator and haven't been reported by the foregoing papers in this field. Using this simulator, the testing and

tuning that should be performed in real vehicles can be transferred to the virtual environment, the efficiency can be greatly improved and the experiment cost can be greatly reduced.

The main part of this paper is divided into three sections. First, the overall structure of EPS system simulator is introduced. Second, the mechanical subsystem model is proposed. Third, the control and electric subsystem model are proposed. Finally, the results and conclusion are presented.

OVERALL STRUCTURE OF EPS SYSTEM

The column-assist, pinion-assist and rack-assist EPS system have been introduced in [4]. This paper focuses on the dualpinion EPS system made by Hitachi, in which the rack is driven by two pinions. Driver input acts on the rack through the first pinion and motor drives the rack through the second pinion. The diagram of the dual-pinion EPS system is shown in Figure 1. The components of this system mainly include steering wheel, steering column, torque sensor, rack and two pinions, coupling, worm reduction gear, controller, motor and inverter. The inverter and controller are fixed behind the motor and the structure of mechatronics integration is applied.

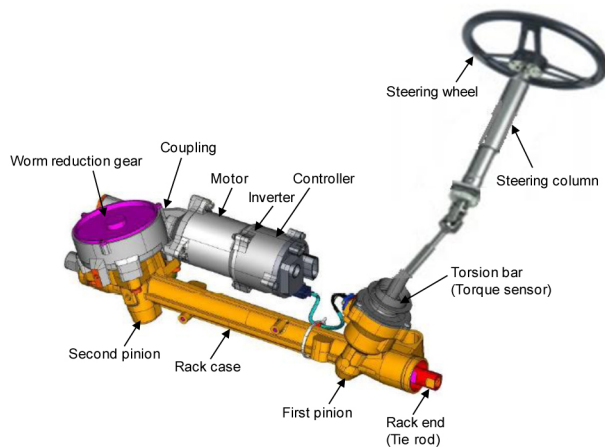


Figure 1. Configuration of dual-pinion EPS system

The rotation of steering wheel is transferred to the first pinion gear through the steering column and the torsion bar, the rack-pinion actuator converts rotational movement of the pinion to translational movement of the rack. The rack turns the wheel by the tie rod and knuckle arm at each end. At the same time, the torque sensor estimates the steering torque enforced on the steering wheel, and transmits this signal to the controller. The controller calculates the required assist torque and converts it to voltage instruction. The voltage is converted to the pulse width modulation (PWM) signal. Using the PWM signal, the inverter converts the 12V DC power to AC power to drive the motor. The motor shaft is attached to the second pinion gear by the coupling, and the

motor torque is enlarged by the worm reduction gear to drive the rack.

The overall of the EPS system simulator developed in MATLAB®/Simulink® is shown in Figure 2, including the mechanical, control and electric subsystem. The mechanical subsystem includes the EPS mechanism and the load. The electric subsystem includes the motor and inverter. In this simulator, the mechanism model, controller model, motor and inverter model, are described in detail, the driving pattern and load are used to test the EPS system dynamics.

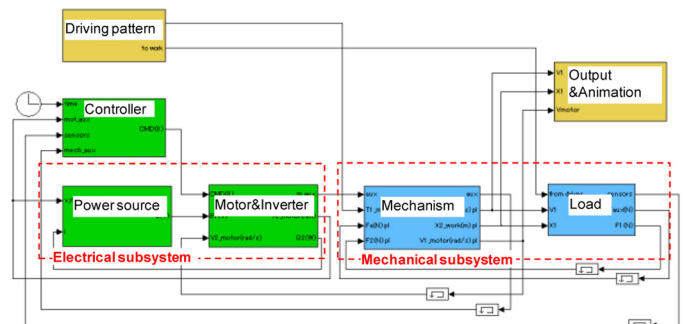


Figure 2. Overall of EPS system simulator

MECHANICAL SUBSYSTEM MODEL

According to Figure 1, the element model of the mechanical system is schematically displayed in Figure 3. In the mechanical subsystem, mechanical parts are modeled as mass interconnected with spring-damper, coupling and gears.

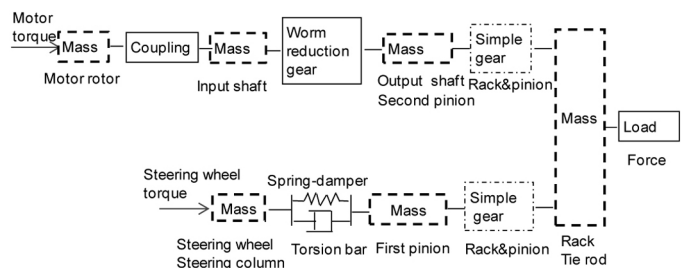


Figure 3. Element model of mechanical subsystem

According to the element model, as shown in Figure 3, the mechanical subsystem model is developed in Figure 4. The models of mass, spring-damper, simple gear are developed using Newton's First Law. The coupling model and worm reduction gear model, explained in detail in the following, are developed using experimental data with hysteresis characteristic. Since the input to the steering wheel is angular velocity, a dummy spring-damper model is used to convert the angular velocity to torque.

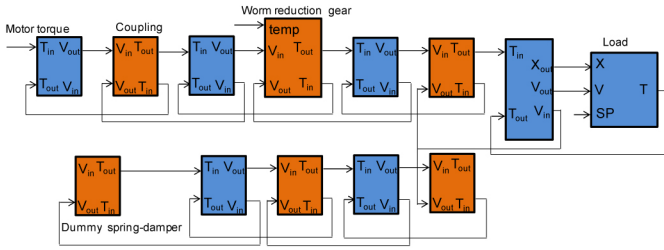


Figure 4. Mechanical subsystem model

The steering wheel and steering column, first pinion, motor rotor, worm reduction gear input shaft, output shaft and second pinion are considered as mass with rotational degrees of freedom (DOF). The rack and tie rod are considered as mass with translational DOF. The motion of rotational system can be expressed as:

$$\Delta T = I \cdot \dot{\omega} \quad (1)$$

Where ΔT is the difference between driving torque and resistant torque, I is the moment of inertia and $\dot{\omega}$ is angular acceleration. The motion of the translational system can be expressed in a similar way.

The torsion bar with torque sensor is modeled as torsion spring-damper, expressed as:

$$T = c(\Delta\omega) + k \int (\Delta\omega) dt \quad (2)$$

where c and k are rotational damping and stiffness, $\Delta\omega$ is the angular velocity difference of the rotational mass attached to the each end of the spring-damper. Similarly, the model of translational system can be developed.

The rack and pinion actuator is modeled as a simple gear, which is expressed by spring-damper with gear ratio. When the power is transferred from the input gear to the output gear, the torques enforced on the input gear and output gear are:

$$\begin{cases} T_{in} = T_{i1} \\ T_{out} = T_{i2} \cdot \eta_f \end{cases} \quad (3-1)$$

When the power is transferred from the output gear to the input gear, the torque is:

$$\begin{cases} T_{in} = T_{i1} \cdot \eta_r \\ T_{out} = T_{i2} \end{cases} \quad (3-2)$$

$$T_{i1} = -[c(v_{in} - r \cdot v_{out}) + k \int (v_{in} - r \cdot v_{out}) dt]$$

$$T_{i2} = [c(v_{in} - r \cdot v_{out}) + k \int (v_{in} - r \cdot v_{out}) dt] \cdot r$$

where v_{in} , v_{out} are angular velocity of the input gear and the output gear, r is gear ratio. η_f and η_r are forward gear

efficiency and reverse gear efficiency respectively. The gear efficiency is estimated from experimental data and simplified to a constant.

Since the hysteresis model is the key component of the coupling, worm reduction gear and load model, the hysteresis model is introduced in advance. The hysteresis characteristic is shown in Figure 5. When the input x increases, the output y increases along the top line, and when the input x returns, the output y returns along the bottom line. Therefore the output is different even to the same input. Furthermore, when the direction of the input changes, the switching between the top line and bottom line is conducted by the middle line.

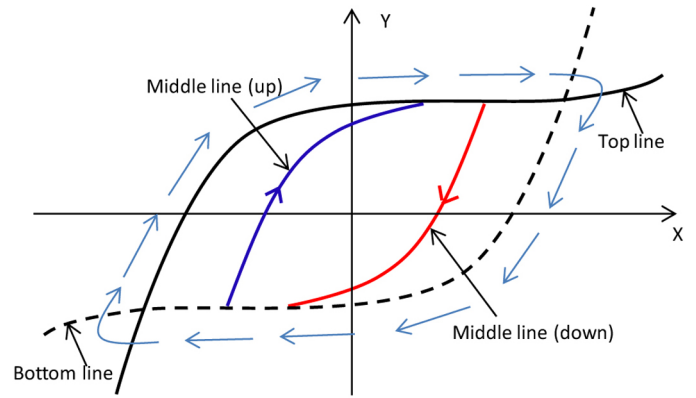


Figure 5. Hysteresis characteristic

The top line is defined by a polynomial (4) of degree n , where $a_n \dots a_0$ are coefficients determined by experimental data. The bottom line is generated symmetrically about the origin to the top line.

$$y = a_n x^n + a_{n-1} x^{n-1} + \dots + a_1 x + a_0 \quad (4)$$

The switching from the top line to the bottom line is conducted by the middle line (down), and the switching from the bottom line to the top line is conducted by the middle line (up). The middle line (up) is defined by a non-dimensional 2th order polynomial (5), whose slope decreases gradually as closing to the top line. Similarly, the middle line (down) is defined by polynomial (6). Since the middle line is defined by non-dimensional polynomial, the slope can be adjusted by multiplying the appropriate factor to match the experimental data.

$$y' = -x'^2 + 2x' \quad (5)$$

$$y' = x'^2 + 2x' \quad (6)$$

Figure 6 describes the algorithm of the hysteresis model. If the direction of input doesn't change, the trajectory progresses along the top line or the bottom line. In this example, the trajectory progresses along the top line and a