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Modelling and Control Design for Electric Power Steering system with Cascaded Lead Compensation Method

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Abstract

This paper presents a new control strategy for an Electric Power Steering system. Due to higher steering assist gain and non linear torque map, which reduces stability and causes vibrations in steering system, an stabilizing compensator should be included in the controller system. The role of compensator is to provide stability to the system and also to accelerate the system response time. A faster response is required as a slight delay or overshoot in steering control can lead to severe consequences. The stability of the system with torque map is detailed. Cascaded lead compensators are designed and applied to the system and the theoretical analysis is verified from the simulation and experimental results.

Keywords: Cascaded lead controller, Electric Power Steering (EPS), Stability analysis.

INTRODUCTION

Electric power steering is a motor based power steering system that provides the driver with a easy and less compact steering control.Nowadays,mostly cars are provided with an EPS system rather then hydraulic power steering system as it holds many advantages[1].These advantages includes higher fuel efficiency,a compactvolume,a easily tunable steering,and also the ability to combine other electrically controlled systems in the vechile.A torque map is the important aspect of an EPS controller;it calculates how much torque is to be provided by the motor.The torque map provides a nonlinear functions between the measured torque from the motor.The torque map curves determines how the steering feels to driver[2].The slope of torque map is the steepest when the vechile is idle,and then decereases when it starts to accelerate,because the torque required to steer is high when the vehicle is idle,and the steering should be heavier when a vechile is going fast to achive stability.A higher controller gain at low velocity and the nonlinear torque can be a reason for instability and vibration in the system[3],[8];hence,the stabilizing compensator is needed along with e torque map to get the EPS controller.But designs ing a proper EPS conroller has become a challenge for several reasons[4].The controller should be robust if there are any unexpected dynamics and parameter can be hard,because even for for the same type of vechile,each car have different system parameters.In addition as the interaction of the steering system with driver may be sensitive,the controller is to be designed to eliminate the various studes suggesting different forms of EPS conrollers to provide the stability to the system.The stable conditions are suggested based on an EPS model and utilized a fixed structure compensator to minimize the torque vibrations and to provied stability to the system[5].A frequency-weighted damping compensator is used to increase the phase margin of the system,but the range of the margin eas restricted[6].An H-infinity control is employed to assit torque and to improve the steering feel as well as to enhance the closed-loop robustness[7].The drawback is that the proposed control has higher order and unable to eliminate vibrations.A robust integral sliding-mode controller is suggested to generate the assist torque and to stabilize the system,hence improves the damping characteristics of EPS[8].An optimal linear quadratic regulator controller is proposed for a dual-pinion EPS[9].Previous studes of EPS controller design has crtain drawbacks.First is most studies,the influence of nonlinearity on stability of the system is not properly analyzed.Linearized system represents dynamic charactristics near by to an operating point only,and nonlinear elements like torque map may reduce stability of the system,which results indivergence or severe vibration of the system.Second,the proposed controllers were only analyzed and verifed with simulations or bench tests which do not refelect the nonlinear characteristics of the tires,especically when the vechicle is parked,which is the worst case for system stability due to the large assit gain.The other suggestions consists of too complex contollers to be implemented practically and large computational effort is not ideal in the development of a commercial vechile.In this,we presents a new method to analyze the srability of an EPS system and to design a new stabilizing compensator for given torque map.The EPS system consists of two combined masses coupled with a torsional spreong and a suggestion for model parameter identification method is made.A separate analysis is made for effects of two potential source of instability,large gain and nonlinearity of torque map in the laplace domain,and each provides guidelines for designing a suitable stabilizing compensator.A cascaded lead compensator is designed as a stabilizing compensator for loop in the frequency domain and a methodology to select parameters for the design of compensator is also proposed.The simulations and experimental results are obtained in order to verify the theoretical analysis.

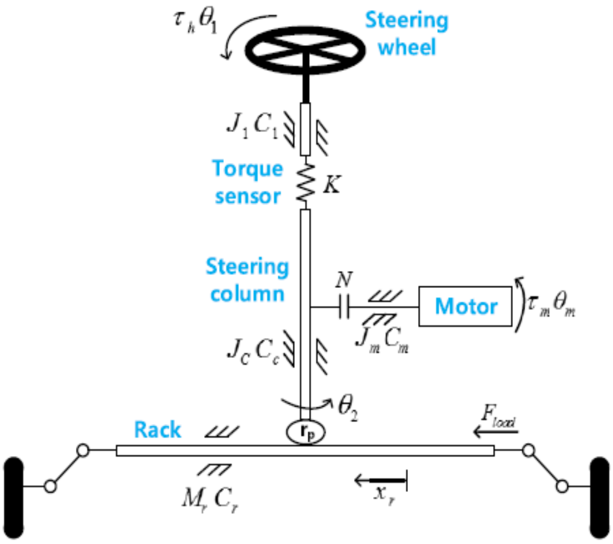
EPS SYSTEM MODELING AND IDENTIFICATION:

A.EPS System model

Based on the location of motor the EPS systems are categorized as:

- Column-type,
- Pinion-type, and
- Rack-Type.

In this, the column-type EPS system is studied; the dynamic model of the above mentioned type is shown below



The each parameter used in the above figure is explained in the Table shown below:

MODEL PARAMETERS AND VARIABLES

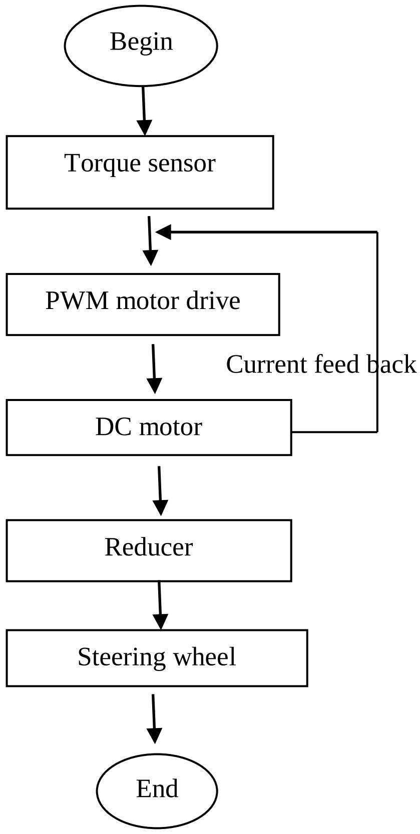
Notation	Description
J_1	Moment of inertia of steering wheel
C_1	Damping coefficient of steering wheel
J_c	Moment of inertia of column
C_c	Damping coefficient of column
K	Torsional stiffness of torque sensor
M_r	Mass of rack
C_r	Damping coefficient of rack
J_m	Moment of inertia of motor
C_m	Damping coefficient of motor
θ_1	Steering wheel angle
θ_2	Column angle
θ_m	Motor angle
x_r	Rack displacement
τ_h	Driver torque
τ_m	Motor torque
F_{load}	Load force of rack from tire
i_q	q-axis current
K_m	Motor constant between q-axis current and assist torque(τ_a)
r_p	Pinion radius
N	Gear ratio

The EPS system comprised of four main parts:

- The steering wheel,
- The column,
- The motor, and
- The rack.

This system is termed as a position servo system, and its working is explained below: when steering turns, torque generated by the steering wheel is sensed by the torque sensor installed in the axle sleeve of steering wheel. And the voltage signal generated is converted into a torque signal by the torque converter. Then it will be converted into digital quantity by the A/D module. The current through the motor will be sensed, and provided as the current signal feedback to the controller. According to the torque signal and feedback current signal, The controller will calculate the booster current value according to the torque and current feedback signal and it will drive the motor using the PWM control. The detection of motor speed is one of the most essential parts. With various speed conditions, the power curve also changes accordingly: if the speed is higher, the power will be lesser. When it attains a certain speed, the power will be zero, which reduces the vibrations and make the system a stable one. This paper mainly focused in designing the controller.

The principle structure is as shown below.



The transfer function for the given blocks in the flowchart above is:

PWM power amplifier:

$$\frac{K_1}{T_1 s + 1},$$

Motor

$$\frac{K_2}{(T_d s + 1)(T_m s + 1)},$$

Reducer

$$= \frac{K_3}{s}$$

Angle-feedback = K4 ,

The simplified transfer function of the system is

$$G(s) = \frac{K_\theta}{s(T_\mu s+1)(T_m s+1)}$$

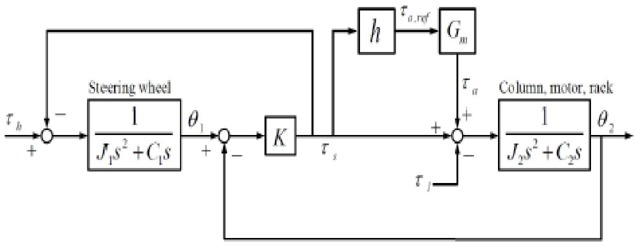
Where

$K\theta = K1K2K3K4$ are magnification factors ;

$T\mu = T1 + Td$ is time constant,

T1 is delay time,

Td and Tm are time constant ;



The EPS system is provided in above block diagram.

B. Model parameter Identification

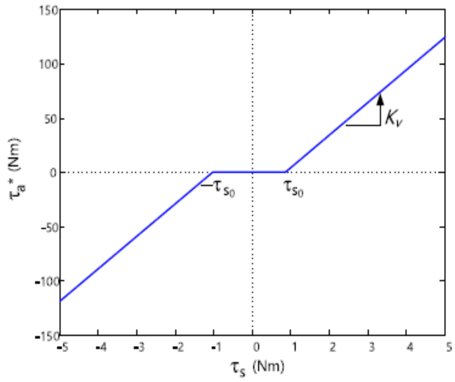
The system identification is essential for model based controller to identify the parameters. To determine the parameters, a comparison of the system response of excitation of the motor at different frequencies with the transfer function of the system from the model is made. Zero external torque is provided ($\tau l = 0$ and $\tau h = 0$). The motor excites the system with different frequencies, and the current, the column angle and the torque were measured as the output variables. The following table provides experimental constraints for the system:

Signal type	Sine function
Amplitude	5,7,9 Nm
Frequency	F= a_0 r^(i-1)
a_0	1 hz
r	1.0615
km	0.9764

STABILITY ANALYSIS AND CONTROLLER DESIGN

A. Controller Structure

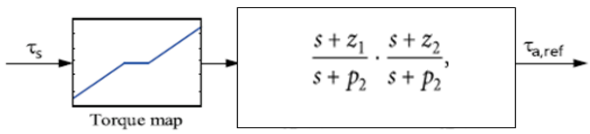
When the torque is applied to steering wheel, the torque sensor senses the torque and send it to the controller of the system. The sensed torque will be provided to the torque map and then to the cascaded lead compensator and then the assist torque was determined by the EPS controller. The steering characteristic map is designed similar to torque map using the torque equation.



$\tau a*$ is the input to stabilizing compensator(assist torque)

$$\tau a* = \begin{cases} 0, & 0 \le \tau_s \le \tau_{s0} \\ K_v(v_{car})(\tau_s - \tau_{s0}), & \tau_{s0} < \tau_s . \end{cases}$$

The torque map has a deadband under $\tau s0$while driving at high speeds as the system is too sensitive to driver torque. This is a reason for nonlinear functionality of the controller. In the parking state, the driver needs a larger assist torque, whereas lesser assist torque is needed at high speeds. Hence *Kv* reduces with higher the vehicle speed. The controller also needs a stabilizing compensator, as the higher assist gain and nonlinear nature of the torque map results in unstable system. Here, we are using a cascaded lead compensator as stabilizing filter and the structure of the EPS controller is shown as below.



The cascaded lead compensator is placed after the torque map and provides a dynamic characteristic to the controller while securing stability and reducing the vibrations in the EPS system. The transfer function for the cascaded lead compensator is given below:

$$C(s) = \frac{s+z_1}{s+p_2} \cdot \frac{s+z_2}{s+p_2},$$

B. Cascaded lead compensator Design

The designing of compensator is provided with the idle vehicle condition (*Vcar* = 0), which is the worst case scenario. At idle condition the torque map provides the steepest slope, because more assist torque is required to cancel the frictional force between the tire and road. This results in large controller gain, which in turn reduces the stability of the system. Hence, if we design the compensator for idle condition, the system will be stable for all the conditions. To achieve a much higher response time, a cascaded phase-lead controller is designed.

The design principles are similar to what has been discussed in the phase-lead section. To achieve maximum desired output, consider the following:

1) Both poles of the two lead controllers are placed farther from the imaginary axis into the left half plane.
2) Both zeros of the two lead controllers are placed near the imaginary axis.
3) Maximum distance between the poles (p_1, p_2) and zeros (z_1, z_2) of the two lead controllers is kept.

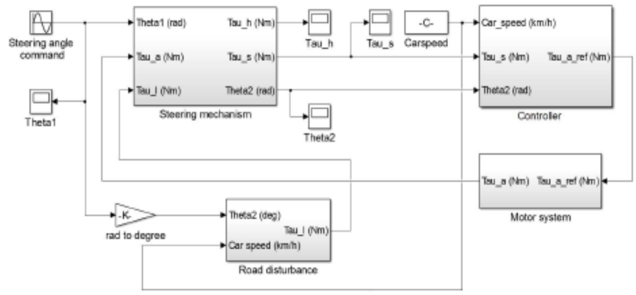
Mathematically,

$$p_1 - z_1 \gg \text{average distance},$$
$$p_2 - z_2 \gg \text{average distance},$$

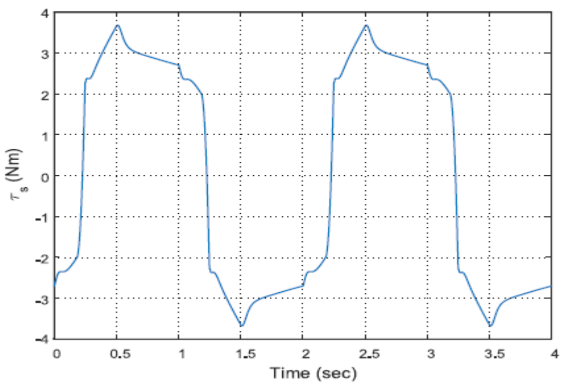
where p_1 and p_2 represent the poles and z_1 and z_2 represent the zeros of the two cascaded lead controllers.

SIMULATION AND VEHICLE TEST RESULT:

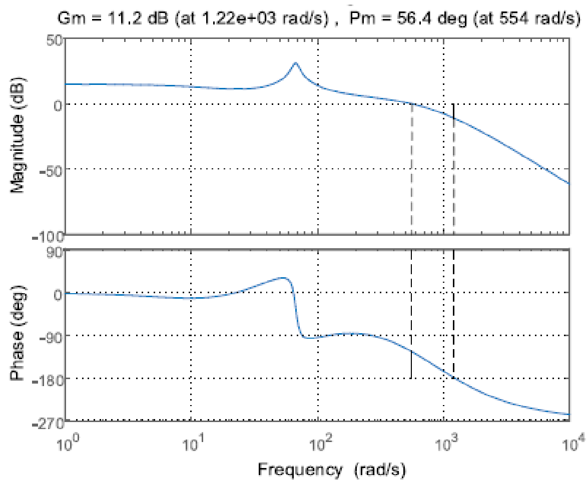
The simulation is built with simulink and consists of the EPS,controller,motor and road disturbance as shown below:

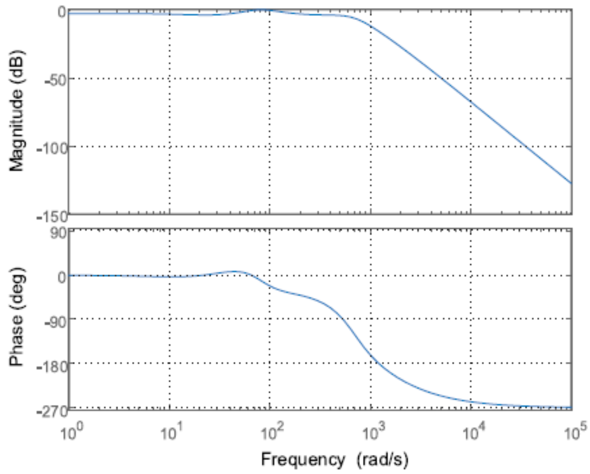


The input command for simulation is obtained from the steering angle.The road disturbance is split into two cases:the idle state and the driving condition.In idle state and thedriving condition.In idle condition,road disturbance is a satutation function with hysteresis,and in the driving mode,road disturbance is proportional to the steering angle.EPS system with proposed controllers are simulated with steeringangle.EPS system with proposed controllers are simulated with steering angle commands of 120◦ and a frequency of 0.7Hz in idle condition 1 and 2,the system is stable as shown in the graph below.



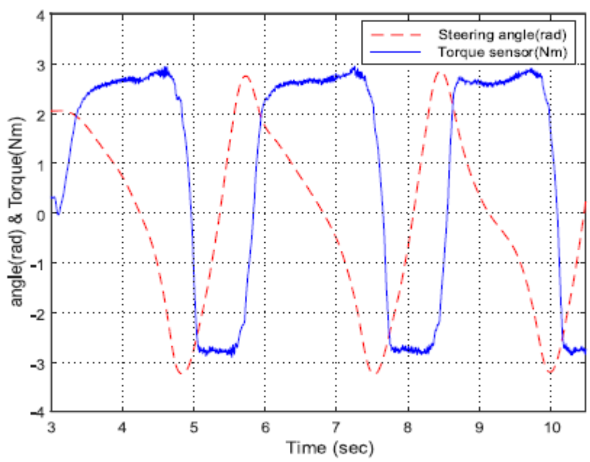
The Bode plot diagrams for the Peq*H* and *Tzw* of proposed controller are as follows





B. Vechile test result

The same constraints used in the model paprameter identification is used here and the experiment is conducted while the vechile is in idle condition.



The results satisfied both the stability conditions and very low vibration (<0.3 Nm) is induced. As per the results from the experiments, the system is stable on both conditions with minimum vibrations. Hence, we can conclude that the proposed controller can be utilized for real time applications.

CONCLUSION

This paper proposed an complete design procedure for a controller for an electric powered steering system based on the torque map.This starts with modeling of the EPS system and proposing identificatio methods for system parameters.Then,the design procedure for a stabilizing compensator for the controller are provided to ensure the stabiltiy of the system.Simulations show that the proposed stabiltiy analysis and controller design methods are accepted well with the results.Finally,the vechile test results shows that the propoced controller performs well in a real vechile.

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