A Study on the Modeling and Analysis of an Electro-Hydraulic Power Steering System

Ji-Hye Kim and Sung-Gaun Kim

***Abstract*—**Electro-hydraulic power steering (EHPS) system for the fuel rate reduction and steering feel improvement is comprised of ECU including the logic which controls the steering system and BL DC motor and produces the best suited cornering force, BLDC motor, high pressure pump integrated module and basic oil-hydraulic circuit of the commercial HPS system.

Electro-hydraulic system can be studied in two ways such as experimental and computer simulation. To get accurate results in experimental study of EHPS system, the real boundary management is necessary which is difficult task. And the accuracy of the experimental results depends on the preparation of the experimental setup and accuracy of the data collection. The computer simulation gives accurate and reliable results if the simulation is carried out considering proper boundary conditions. So, in this paper, each component of EHPS was modeled, and the model-based analysis and control logic was designed by using AMESim

***Keywords*—**Power steering system, Electro-Hydraulic power steering (EHPS) system, Modeling of EHPS system, Analysis modeling.

I. Introduction

POWER steering is a system which helps to steer the wheels with some source of power other than the driver's manual force when he tums the steering. This feature adds to the comfort while driving as less effort is needed to turn the steering wheel by the driver [1].

In recent years the efforts to produce more efficient vehicles have led to a trend to replace the traditional engine belt-driven hydraulic power assisted steering (HPS) system. In the current market two systems are increasingly being used. Firstly, in electro-Hydraulic power steering (EHPS) system, the hydraulic pump is driven by an electric motor and runs independently from the engine. Because the speed of the pump is not subjected to the wide range of engine speeds, this reduces the power demand of the system dramatically and maintains all the basic properties of a hydraulic system, e.g. the good road feel. And also, they have many advantages over traditional hydraulic power steering systems.

Secondly, in electrical power steering (EPS) systems the steering assistance comes directly from an electric motor. This gives a further improvement in efficiency.

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EHPS and EPS systems are currently suitable mainly for small and medium sized cars because of electrical power limitations.

Most development work is currently aimed at EPS systems, focusing particularly on improvement of the steering feel and cost reduction. EPS systems are likely to become more and more common, particularly when higher voltage supply systems become commonplace and when technological advances enable compact, low inertia, higher torque motors to be used. However HPS and EHPS systems are likely to remain in widespread use for the foreseeable future, and EHPS systems still have great potential to reduce the energy consumption [2].

The EHPS system is comprised of ECU including the logic which controls the steering system and BLDC motor and produces the best suited cornering force, BLDC motor, high pressure pump integrated module and basic oil-hydraulic circuit of the commercial HPS system.

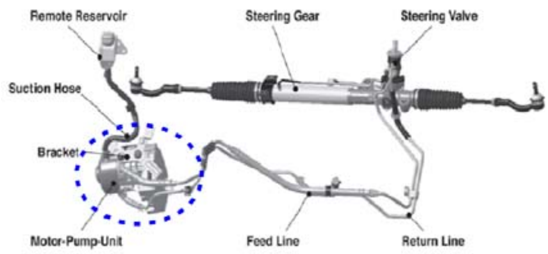

Fig. 1 Structure of EHPS System

II. Modeling of Electro-Hydraulic Power Steering System

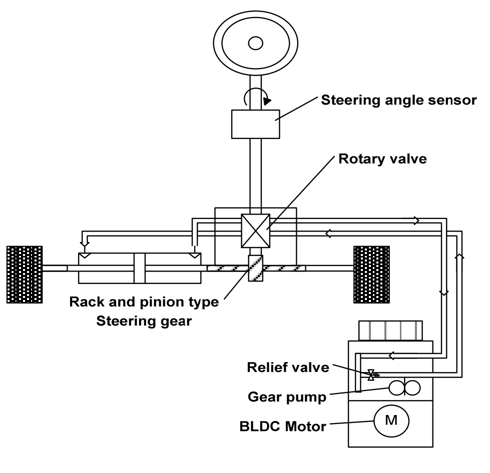

Fig. 2 Schematic diagram of EHPS system

An electronic control unit (ECU) supplies the electric motor with an appropriate pulse-width modulated (PWM) voltage according to the actual power demand. The electric motor drives a positive displacement pump which supplies the oil flow to the steering valve. The steering valve supplies a differential pressure to the cylinder to provide the required assistance force to the tie rods. The differential pressure is governed by the torsion bar twist angle and hence by the steering wheel torque.

A. Rotary valve

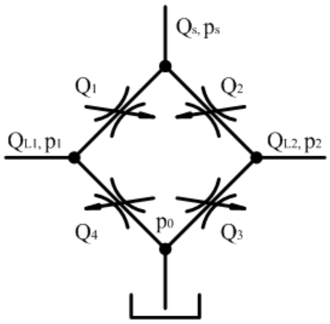

Fig. 3 Schematic diagram of Rotary valve

The sleeve connected with the torsion bar, and the spool also connected with the torsion bar on the other side, so when the torsion bar twists, there is an angle error between the sleeve and the spool. There are axial slots in the sleeve and the spool. The slots in the sleeve are larger than their mating lands in the spool. In one groove of the valve, there are two orifices caused by the spool and sleeve. When a steering torque is applied, the torsion bar twists and the spool is allowed to rotate relative to the sleeve. When cross section area of one orifice increases, and the other decreases, which causes the output pressure difference on the two side of the valve.

The flow rate orifice equations given as below describe the flow rate and pressure relations

$$Q_i = C_d A_i \sqrt{\frac{2}{\rho}\left|\Delta p_i\right|} \qquad i = 1.2.3.4 \tag{1}$$

where Q_i = flow rate through the valve port i, m^2/s; C_d = flow coefficient; A_i = open area of the orifice, m^2; Δp_i = pressure error of orifice, Pa; ρ = density of oil, kg/m^2

$$\begin{cases} Q_s = Q_1 + Q_2 \\ Q_{L1} = Q_1 - Q_4 \\ Q_{L2} = Q_2 - Q_3 \end{cases} \tag{2}$$

$$\begin{cases} Q_1 = C_d A_1 \sqrt{2(p_s - p_1)/\rho} \\ Q_2 = C_d A_2 \sqrt{2(p_s - p_2)/\rho} \\ Q_3 = C_d A_3 \sqrt{2(p_2 - p_0)/\rho} \\ Q_4 = C_d A_4 \sqrt{2(p_1 - p_0)/\rho} \end{cases} \tag{3}$$

where p_s = output pressure, Pa; Q_{L1} = flow rate to the left chamber of cylinder, m^2/s; Q_{L2} = flow rate to the right chamber of cylinder, m^2/s; p_1 = pressure to the left chamber of cylinder, Pa; p_2 = pressure to the right chamber of cylinder, Pa; [3].

B. Gear pump

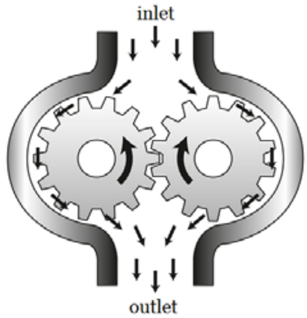

Fig. 4 Structure of Gear pump

The net area flow rate $\dot{A}_N$ can be written as

$$\dot{A}_N = \dot{A}_O - \dot{A}_E \tag{4}$$

where $\dot{A}_O$ is the entering area into the control region and $\dot{A}_E$ is the exiting area out the control region per unit time.

Firstly, the entering area flow rate $\dot{A}_O$ can simply be written as

$$\dot{A}_O = \frac{1}{2} r_{a,1}^2 \varpi_1 + \frac{1}{2} r_{a,2}^2 / \varpi_2 = \frac{1}{2} \varpi_1 (r_{a,1}^2 + \frac{z_1}{z_2} r_{a,2}^2) \tag{5}$$

where ϖ_i is the rotation speed, $r_{a,i}$ is the addendum radius, and z_i is the number of teeth for gear *i*.

Then, the exiting area flow rate is written as

$$\dot{A}_E = \frac{1}{2} \varpi_1 (r_1^2 + \frac{z_1}{z_2} r_2^2) \tag{6}$$

where r_i are the radial distances of the tooth pair , which can be written as

$$r_i = (r_{op,i}^2 + l^2 - 2 r_{op,i} \cdot l \cdot \cos\beta_i)^{\frac{1}{2}} \tag{7}$$

In the above l is obtained as

$$l = l_0 - r_{bi} \varpi_i t \tag{8}$$

where l_0 is the length of line of action in the approaching are and is given as

$$l_0 = r_{a2} \cos[\alpha_{op} + \sin^{-1}(\frac{r_{op.2}}{r_{a,2}} \cos\alpha_{op}) / \cos\alpha_{op} \tag{9}$$

where $r_{op.2}$ denotes the operation radius of gear 2 and α_{op} is the operation pressure angle [4].

C. BLDC motor

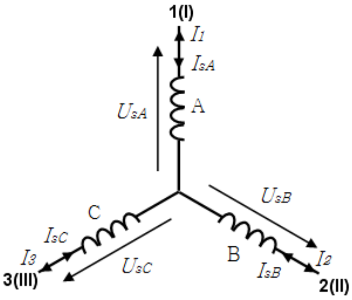

Fig. 5 Schematic diagram of brushless dc motor

The electromagnetic torque is computed according to energy conservation:

$$r = -p(T_{coef}(\theta_e, T_{emp})I_{sA} + T_{coef}(\theta_e - \frac{2\pi}{3}, T_{emp})I_{sB} + T_{coef}(\theta_e + \frac{2\pi}{3}, T_{emp})I_{sC}) \tag{10}$$

where I_{sA} , I_{sB} , I_{sC} is stator winding current, T_{emp} is temperature, T_{coef} is torque coefficient datafile or expression as a function of electrical angle and temperature, θ is electrical angle [8].

D. Steering wheel and pinion

The inputs to this system are the torque generated by the driver and the assisting force of the piston.

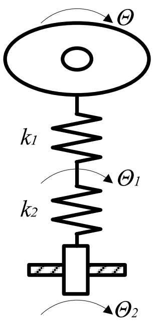

Fig. 6 Schematic diagram of Steering wheel and pinion

The equation of motion for the steering column, pinion and rack can be described as follows:

$$k_1(\theta - \theta_1) = k_1(\theta_1 - \theta_2) \tag{11}$$

where θ =steering wheel angle, rad; k_1 =steering column stiffness, Nm/rad; k_2 =torsion bar stiffness, Nm/rad; θ_1 =upper column angle, rad; θ_2 =pinion angle, rad [3].

E. Rack and wheel

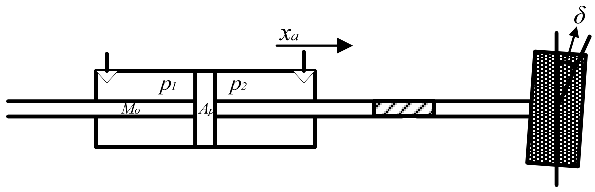

Fig. 7 Schematic diagram of rack and wheel

In steering system, the parts of mechanical power mainly consists of rack, hydraulic cylinder, piston, tie-rod, knuckle arm and wheels. A schematic of rack and wheel is showed in Fig 7

The equation of motion for steer angle (δ), kingpin and knuckle arm can be described as follows:

$$I_w\ddot{\delta} = F_r d - k_w\dot{\delta} - C_w\delta - K_1 e\delta \tag{12}$$

where I_w = moment of inertia front tread, kgm^2; δ = steer angle, rad; d = moment arm length of kingpin, m; k_w = vicous damping coefficient of steering system, Nms/rad; C_w = equivalent stiffness of steering system, N/rad; K_1 = front tread cornering stiffness, N/rad; e = moment arm length of front tread, m;

The relationship between displacement of piston (x_a) and pinion angle (θ_2) is as follows [3]:

$$\theta_2 = x_a \cos\alpha / r \tag{13}$$

III. Simulation

Simulation of the EHPS system was performed. The simulation model using AMESim consists in a steering wheel, a rotary valve, a rack, a hydraulic jack, the lines, a gear pump, a BLDC motor and control logic. When the driver inputs a steering angle command, the steering valve opens. This allows oil to flow into the hydraulic cylinder, with a pressure proportional to the amount of valve opening. The oil pressure acts on the cylinder piston to create an actuating force proportional to the pressure. This actuating force then assists the driver in moving the cylinder piston and the mechanism connected to it (rack, wheels...). To provide the optimum actuating force, flow of gear pump must be controlled. In order to control the flow, the current of each phase of the BLDC motor is controlled by the vector control. Through the vector control, the stator current vector and field flux of motor is always maintained the perpendicularity.

IV. Conclusion

This paper presents the modeling and simulation of the electro-hydraulic power steering system. And the model-based analysis and control logic was designed by using AMESim. This model is the multiplicity of domain system consisting of BLDC motor part, the power steering and control logic. The computer simulation was performed.

Further work will complete the model-based system for the Matlab/Simulink integrated control and improve the whole vehicle-based control algorithm by driving scenario.

ACKNOWLEDGMENT

This paper was supported by the Ministry of Knowledge Economy of Korea

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