

Transient Responses of a Hydraulic Power Assisted Vehicle Steering System

2011-01-0984

Published
04/12/2011Nong Zhang
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doi:10.4271/2011-01-0984

ABSTRACT

This paper presents a comprehensive model of a hydraulic power steering system for predicting the transient responses under various steering inputs. The first principles of multi-body system dynamics and fluid mechanics are applied to model key nonlinear components and in particular, the rotary spool valve, piped fluid lines, the frictional coupling between multiple contacting surfaces with use of the empirical data. The system model, which integrates together all of lump masses, fluid line elements and hydraulic components, is formulated using the state space representation approach. It contains time-variant coefficient matrices resulting from the nonlinearities in the fluids systems. A numerical simulation scheme is developed to obtain the system transient responses and the results are compared with those measured from the tests. Large transient vibration (or judder) of multi-body subsystem, or pressure ripple in the hydraulic subsystem, are observed from both of simulation and tests under certain operating conditions. In general, the simulation results agree with those obtained from the tests under the same steering inputs and operating conditions. The presented model can predict the pressure boosting characteristics of the rotary spool valve with a good accuracy, which could otherwise only be done through testing on prototypes.

INTRODUCTION

Hydraulic power steering systems are widely used in modern vehicles to assist drivers in steering vehicles through reducing the required input torque on the steering wheel and regulating the ratio between steering wheel angular displacement and the front wheel angular displacement. A hydraulic power steering system consists of many parts and components including a steering wheel, input shaft, rotary spool valve, rack and pinion, pump, hydraulic cylinder mounted on the

chassis, hydraulic hoses, tie rod linkage and tires. Although it is well developed technology, there are remained noise and vibration issues associated with the kinetic coupling between hydraulic and mechanical components of the steering system. In the design analysis, existing modelling methods and simulation tools are limited to free and steady state vibration analysis of the system due the high nonlinearity existing in the rotary spool valve and various surface contacts existing within the system. Special rotary spool valves and rack and pinion set with variable ratio were developed to obtain desirable pressure boost characteristics to ensure safe steering.

Some vibration and noise problems, like steering shudder and shimmy, have been observed. Wong [1] stated that the steering shudder was an excitation of the fundamental frequency at the frame support and it was due to the periodic fluid force caused by the engine-driven vane steering pump. Sahinkaya [2] suggested that the steering vibration was caused by an interaction of the mechanical components and the periodic high pressure fluid of the steering system, and the fluid oscillation in the power cylinder of a hydraulic power steering system was the main reason of the steering shudder and shimmy. The steering shudder occurred when parking a vehicle and its frequency was from 25 to 200 Hz, and the frequency of steering shimmy was from 6 to 15 Hz.

A simplified mathematical steering system model including the supply line, the feeding pipes, and power cylinder was presented by Matsunaga et al [3]. This research mainly focused on the hydraulic components in a power steering system, and it did not include steering wheel, pinion and front wheels. The transfer matrix method was applied to investigate the fluid circuit dynamics of the hydraulic power steering system [4, 5]. The system transfer matrix was formed by combining the impedances of the pump and the steering

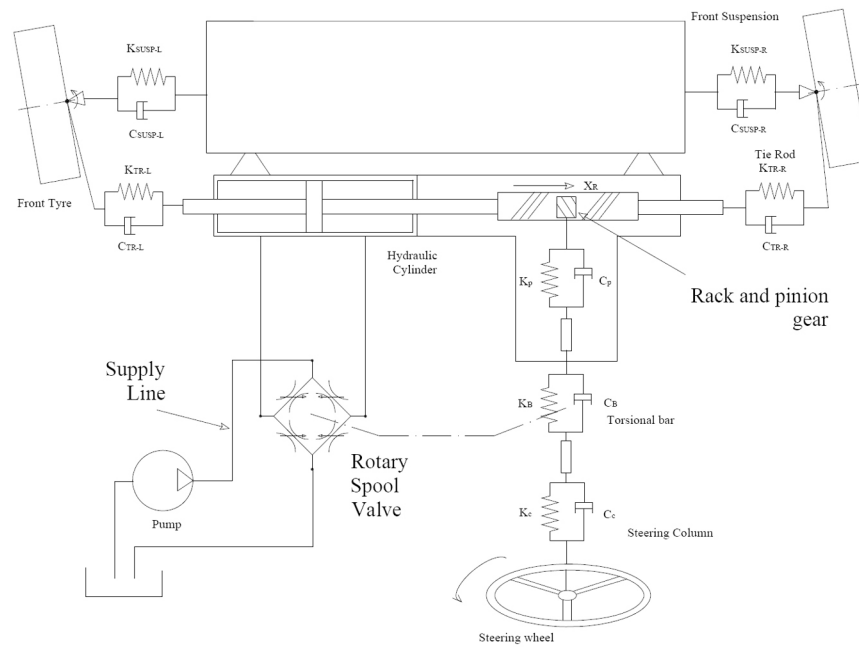


Figure 1. Schematic of a typical power steering system

gear with the impedance matrixes of the tubes and hoses. It showed that the computer simulation partly agreed with the experimentally measured data. The limitation of the presented work was that the key component of the system, the rotary spool valve, was simplified as a point impedance, and the detailed characteristics of the valve were not integrated in the system model.

Chen [6] also adopted the transfer matrix method to study the noise and vibrations generated by the comprising the mechanical and hydraulic components of the hydraulic power steering system was developed and tests were conducted to verify the model. The properties of the hose and the fluid-structure interaction were discussed in detail, but the coupling between the mechanical and hydraulic components was not included. The noise characteristics of some individual components in the hydraulic power steering system, such as the pump and the steering gear, were measured and reported by Qatu et al [7]. Although, the detailed measurement results were supplied, the systematic modelling and analysis was not detailed. Zhang et al [8] presented a model of hydraulic power steering system fitted with variable ratio steering rack-pinion unit and determined the system's dynamic characteristics. However, the method is again based on transfer matrix method and has a limitation on transient analysis that reveals the dynamic coupling between fluid, structure and rigid body system under a large step steering input.

The aforementioned studies indicate there is lack of a comprehensive model for unveiling the dynamic coupling and interaction of the mechanical and hydraulic components of a steering system. In particular, little work has been

reported on the transient analysis of the power steering system consisting of pressurised fluid circuits and kinetically linked multi-rigid bodies.

In this paper, a mathematic model of a hydraulic power steering system based on the first principles of multi-body system dynamics and fluid mechanics is presented. The appropriate theories and methods are pressure ripple and flow pulsation through the fluid lines of power steering systems. A mathematical model applied to model main nonlinear components and in particular, the vehicle speed sensitive hydraulic power steering mechanism, the supply line and control valve with the use of empirical data supplied by the manufacturer. The system model, which integrates together all of lump masses, fluid line elements and hydraulic components, is then formulated using the state space representation approach. It consists of time-variant coefficient matrixes, which accommodate the nonlinearities in the fluids systems and can predict the system transient responses under large and harsh steering inputs. A numerical simulation scheme is developed to obtain the system transient responses. The obtained results are compared with those measured from the tests. Large transient vibration, so called judder and pressure ripple, are observed from both of simulation and tests when certain steering inputs are applied. Discussion and concluding remarks are given on the both simulation results and those obtained from the laboratory rig tests.

MATHEMATICAL MODEL FOR TRANSIENT ANALYSIS

HYDRAULIC POWER STEERING SYSTEM

Figure 1 shows a schematic of a typical power steer system. The hydraulic power steering system consists of subsystems of steer input wheel together with a torsional bar that produces a twist angular displacement controlling the spool valve's status; spool valve system with a pump fluid supply and output fluid lines connected to the left and right fluid chambers of the cylinder; fixed or variable ratio rack-pinion unit that converts rotation of steering wheel into translation of the rack, which is also the position rod inside the hydraulic cylinder and linked with the front wheel steering linkages; and two front wheels. When the steering is turned from its neutral position, the spool valve will generate a pressure difference in the left and right fluid chambers of the hydraulic cylinder and results in the assisted translation of the rack and thereby the front wheel steering rotation.

MULTI-LUMPED MASS MECHANICAL SYSTEM

To perform the transient analysis, for simplicity, the rigid body components are modelled as lumped masses and they are connected by linear springs and dampers. A total of 11 independent coordinates, i.e., 11 degree of freedoms (DOFs), are defined to represent the translational or angular displacement of rigid bodies' motions as shown in schematic in Fig. 1. The equations of motion are derived using free body diagrams approach and expressed in the following form.

$$[M]_{Lz \times Lz} [\ddot{Z}]_{Lz} + [C]_{Lz \times Lz} [\dot{Z}]_{Lz} + [K]_{Lz \times Lz} [Z]_{Lz} = [F_M]_{Lz} \quad (1)$$

where

$$Z = [A_C \ A_P \ X_H \ X_R \ A_{FWL} \ A_{FWR} \ A_{CPL} \ A_{CPR} \ X_{FWL} \ X_{FWR} \ X_V]^T,$$

$$\dot{Z} = [\dot{A}_C \ \dot{A}_P \ \dot{X}_H \ \dot{X}_R \ \dot{A}_{FWL} \ \dot{A}_{FWR} \ \dot{A}_{CPL} \ \dot{A}_{CPR} \ \dot{X}_{FWL} \ \dot{X}_{FWR} \ \dot{X}_V]^T$$

the subscripts mean: C - steering column; P - pinion; H - hydraulic cylinder; R - rack; FW- front wheel; L & R - left & right; CP - contact patch of tyre; and V- vehicle.

Equation (1) can be rewritten into a state space form:

$$T_M \dot{X}_M = S_M X_M + F_M \quad (2)$$

where $F_M = [0 \ F_M]^T$, Length of X_M , $LM=2LZ=22$.

HYDRAULIC SUBSYSTEM

For the hydraulic fluid system ($Lh=length \ of \ X_H=M$); the key hydraulic elements and components are modelled with a total of 12 first order differential equations.

Cylinder Chamber Element

Figure 2 shows the cylinder piston component, the fluid goes out from the upper-chamber; and into the bottom-chamber. Their governing equations of the unit are

$$\dot{P}_{OUT} = \frac{\beta_{oil} (A_{OUT} \dot{Z}_{p-ch} - Q_{OUT})}{V_{OUT_0} - Z_{p-ch} A_{OUT}}, \quad (3)$$

$$\dot{P}_{IN} = \frac{\beta_{oil} (Q_{IN} - A_{IN} \dot{Z}_{p-ch})}{V_{IN_0} + Z_{p-ch} A_{IN}}. \quad (4)$$

Where, the 2nd term (with algebraic value) is contribution of $\dot{P}$ to Q .

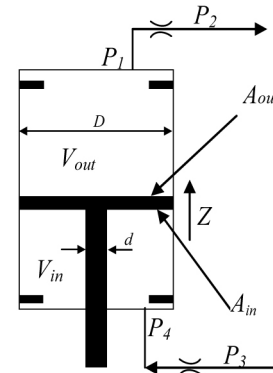


Figure 2. Schematic of Cylinder

$$\dot{Z}_{p-ch} = \dot{Z}_{piston} - \dot{Z}_{cham_body}; \quad Z_{p-ch} = Z_{piston} - Z_{cham_body} \quad (5)$$

$$Q_{OUT} = K_C^{OUT-Orif} (P_1 - P_2); \quad Q_{IN} = K_C^{IN-Orif} (P_3 - P_4); \quad (6)$$

Where C notates cylinder; orif for orifice; V_{IN} & V_{OUT} for volumes of the chambers.