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Design of Industrial Electro-Hydraulic Valves, New Approach

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ABSTRACT
A fully flexible valve actuation systems are being under consideration as an enabling technology for advanced internal combustion engine concepts. Electro-hydraulic valve actuator systems are being considered as a dominating variable valve technology. Compared to the servo control system, the system using a proportional valve has the advantages of low price, high anti-pollution ability and high reliability. Unlike existing electromechanical or servo actuated electro-hydraulic valve actuation systems, precise valve motion control is achieved with an internal feedback mechanism. This feedback mechanism can be turned on or off in real-time using simple two state valves which helps reduce the system cost and enables mass production.
Keywords: Electro-hydraulic, Valve. Dynamic characteristics, Control, Engine

I. INTRODUCTION

In Practice the design, manufacture and application of electro-hydraulic control valves embraces many disciplines such as :

- Materials selection, new materials
- Fluid mechanics, wear and lubrication, alternative fluids
- Thermodynamics
- Energy efficiency
- Vibration & noise analysis
- Components & system steady state & dynamic design
- Sensors and electromagnetic technologies
- Signal processing and associated algorithms
- Computer control techniques

Figure (1) shows a basic directional control valve which is really an on-off valve for selecting flow rate direction. Applying a voltage, either ac or dc, to one of the solenoids crates an electromagnetic force that throws the spool to the end of its stroke thus opening the appropriate flow ports. Usually with no solenoid actuation the spool returns to its central position. The spool ports can have variety of configurations depending upon the requirements, and this type of valve is the backbone of many industrial applications requiring sequence-type operations.

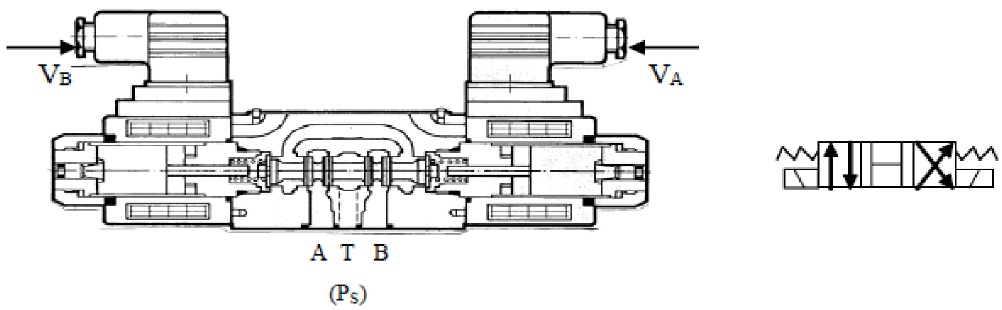


Figure (1) A directional control valve with solenoid actuation

The application of increasing voltage to the solenoid produces an electromagnetic force that attempts to move its poppet to the left thus increasing the resistance to flow through the restrictor and hence the pressure at the valve [Watton 2016],

It has been shown that a flexible air handling system with the capability of varying the valve lift , timing, duration or a combination of these parameters can offer significant improvements in the performance and efficiency over a wide range of operating conditions [Turner 2004].

For any system intended to be mass produced, it is required to have the following characteristics:

- Flexibility in lift, timing and duration
- Low valve seating velocities
- Low power consumption
- A feedback system with simple and inexpensive components that require minimum calibration while being capable of precise valve motion control.

- Subsystems which can be packaged compactly and efficiently [Pradeep 2009]

II. LITERATURE SURVEY

The electro-hydraulic valve actuator was studied in Stanford University already in the beginning of the 1980s [Richman 1984], when one of the first publications of this kind of device was produced, and a test rig was manufactured. In 2008, Liao et al.(Stanford university) published “Repetitive Control of an Electro-Hydraulic Engine Valve Actuation” based on simulation in which they presented a framework covering system identification, feedback controller design and a feed forward repetitive controller [Liao 2008].

Increased efficiency, reduced emissions, and improved power over existing internal combustion engines are the three primary objectives of the research currently underway at the University of South Carolina. The research focuses on the development of a camless engine and addresses several of the design imitations of earlier camless attempts. One of their first publications dealing with camless engines is from 2001: John Brader’s thesis “Development of a Piezoelectric Controlled Hydraulic Actuator for a Camless Engine”. The University of California has been studying electrohydraulic camless valve trains since the 1990s and has established cooperation with the Ford Motor Company. In 2000, they introduced different control methods for the electrohydraulic valve train by means of simulations [Ashhab2000a], [Ashhab 2000b], and test rig tests were performed by Tai et al. [Tai 2000].

The Department of Mechanical Engineering at the University of Minnesota includes the Center for Diesel Research. Zongxuan Sun is one of the assistant professors and his recent research work includes novel time-varying control methodologies for rotational-angle based systems, developing key enabling technologies for clean, efficient and multi-fuel capable automotive power train, such as camless engine, precise pressure regulation for common-rail systems, modeling and control of efficient transmission systems, and various advanced hybrid concepts. In 2009, he published a journal paper on electrohydraulic fully flexible valve actuation. The paper presents an electrohydraulic valve actuation concept with an internal feedback mechanism. Key technical issues, such as dynamic range capability, seating velocity and closing timing repeatability, area schedule, internal feedback spool dynamics, and energy consumption, are modeled and analyzed. A test rig was also manufactured. [Sun 2009],[Heinzen 2011].

III. SYSTEM DESIGN

The block diagram of the proposed valve actuation system is shown in figure (2)

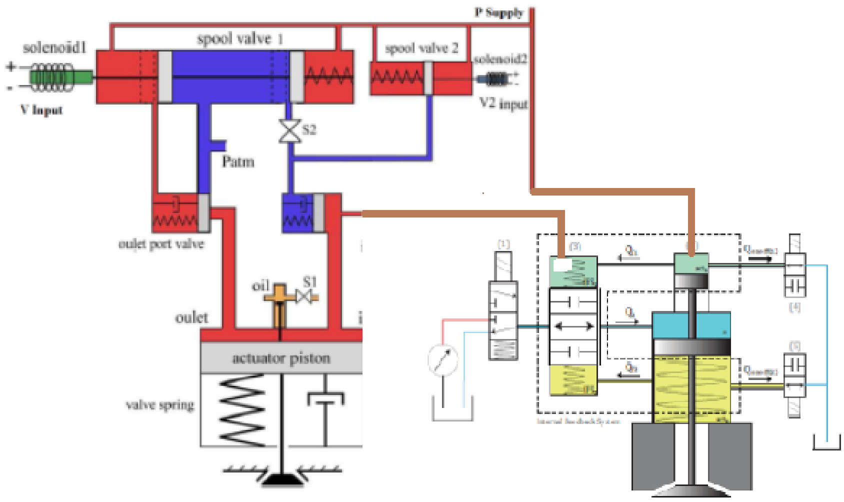


Figure (2) The Proposed System

It is a modification of the system used by [pradeep 2009] , so we used some of the equations mentioned in that reference.

IV. MATHEMATICAL MODEL OF THE SYSTEM

The dynamic of the feedback system depend on number of parameters like the physical dimension of each component, the stiffness of the various springs and the timing of on-off of the valves etc. It is therefore necessary that we have an accurate mathematical model of the system, which will help to evaluate the effects of various parameters and verify any new model that proposed. The valve operates at 50Hz when the engine is running at 7000 RPM. For modeling the pressure variations, the effect of fluid compressibility needs to be considered since it becomes prominent at such high frequencies. The dynamics of the actuator, spool and pressures in the various chambers are described as follows.

$$\dot{X}_{act} = V_{act} \quad (1)$$

$$\dot{V}_{act} = \frac{1}{M_{act}}[P_a \cdot A_a + P_{act_1} \cdot A_{act_1} - P_{act_2} \cdot A_{act_2} - K_{act} \cdot X_{act} - b_{act} \cdot \dot{X}_{act} - F_{preload}] \quad (2)$$

where, X_{act} and V_{act} are the position and velocity of the actuator, M_{act} is the moving mass of the actuator and the engine valve assembly, P_a is the pressure in the actuation chamber, A_a is the area of the actuator's piston, P_{act1} , P_{act2} are the pressures in the actuator's top and bottom feedback chambers, A_{act1} and A_{act2} are the areas of the actuator's top and bottom feedback chambers, K_{act} is the stiffness of the actuator spring, $F_{preload}$ is the spring preload and b_{act} is the damping coefficient of the actuator.

$$\dot{X}_{spool} = V_{spool} \quad (3)$$

$$\dot{V}_{spool} = \frac{1}{M_{spool}}[P_{IFS_1} \cdot A_{IFS_1} - P_{IFS_2} \cdot A_{IFS_2} - (K_{IFS_1} + K_{IFS_2}) \cdot X_{spool}] \quad (4)$$

where, X_{spool} and V_{spool} are the position and velocity of the spool, M_{spool} is the mass of the spool, P_{IFS1} and P_{IFS2} are the pressures in the spool valve's top and bottom feedback chambers, A_{IFS1} and A_{IFS2} are the areas of the feedback regulator's top and bottom feedback chambers, K_{IFS1} and K_{IFS2} are the stiffness's of the IFS springs and b_{IFS} is the damping coefficient of the IFS.

$$\dot{P}_a = \frac{\beta\,(Q_a - V_{act} \cdot A_a)}{(x^*_{act} + X_{act}) \cdot A_a} \quad (5)$$

$$\dot{P}_{act_1} = \frac{\beta\left(-Q_{f1} - Q_{onoff_1} - V_{act} \cdot A_{act_1}\right)}{\left(x^*_{act_1} + X_{act}\right) \cdot A_{act_1}} \quad (6)$$

$$\dot{P}_{act_2} = \frac{\beta\left(-Q_{f2} - Q_{onoff_2} + V_{act} \cdot A_{act_2}\right)}{\left(x^*_{act_2} - X_{act}\right) \cdot A_{act_2}} \quad (7)$$

$$\dot{P}_{IFS_1} = \frac{\beta\left(Q_{f1} - V_{spool} \cdot A_{IFS_1}\right)}{\left(x^*_{spool_1} + X_{spool}\right) \cdot A_{IFS_1}} \quad (8)$$

$$\dot{P}_{IFS_2} = \frac{\beta\left(Q_{f2} + V_{spool} \cdot A_{IFS_2}\right)}{\left(x^*_{spool_2} - X_{spool}\right)} \quad (9)$$

where, β is the bulk modulus of the hydraulic fluid and x_{act} , x_{act1}, x_{act2}, x_{spool1} and $x\_{s\ pool2}$ are the clearances in each of the corresponding chambers when the engine valve is in the closed position and the spool in the center position.. The flow rates are calculated using the orifice equation.

$$Q = A \cdot C_d \cdot \sqrt{\frac{2 \cdot |P_1 - P_2|}{\rho}} \cdot sign(P_1 - P_2) \qquad (10)$$

where, A is the associated orifice area between the chambers, C_d is the discharge coefficient, r is the density of the fluid, P_1 and P_2 are the upstream and downstream pressures. In the case of Q_a, the orifice area A_{spool} is a function of the displacement of the spool X_{spool} . This relation $A_{spool} = f_{(Xspool)}$ is called the area-schedule. In the case of $Q_{ono\ f\ f1}$ and $Q_{ono\ f\ f2}$, the corresponding orifice areas $A_{ono\ f\ f1}$ and $A_{ono\ f\ f2}$ are set to either 0 or maximum depending on the state of the particular on-off valve. The lift of the engine valve is controlled by varying the timing of the bottom on-off valve. Simulations are performed by activating the IFS when the engine valve reaches 6 mm, 7 mm and 8 mm which controls the lift of the engine valve to 8 mm, 9 mm and 10 mm accordingly. The results of these simulations are shown in Fig. 3(a). For valve closing, activating the top on-off valve when the engine valve reaches 2 mm, decelerates the engine valve and ensures that the valve seating velocity is minimized. Fig. 3(b) shows the velocity of the valve corresponding to the 10mm lift case. The position and velocity traces agree closely with the results obtained from AMEsim *(a physics based multi-disciplinary modeling & simulation package)* model simulation in [**Z Sun 2009].**

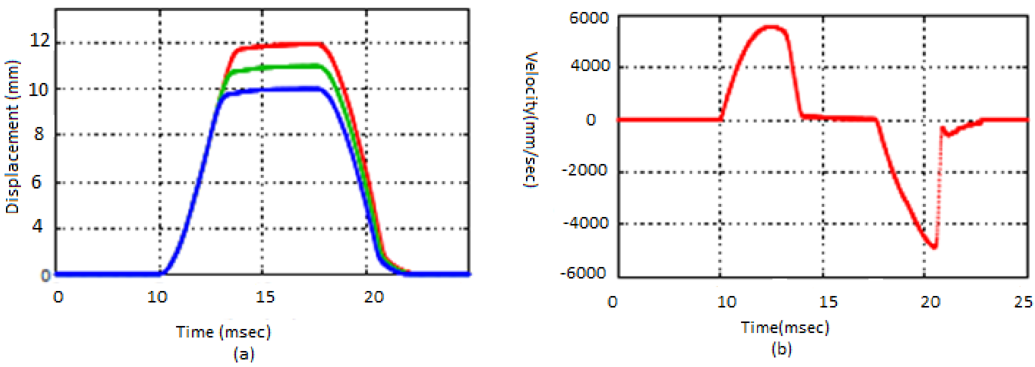
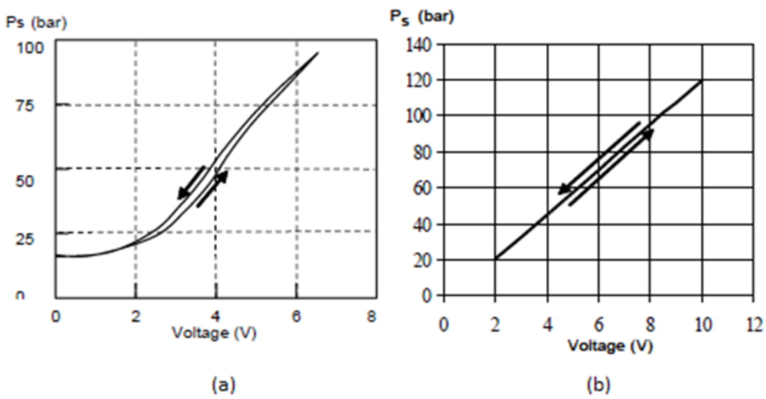


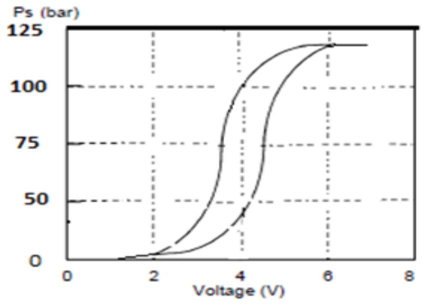
Figure (3) Comparison between reduced order model and full order model (a) Actuator position; (b) Actuator velocity

Test: of three different valves

Figure (4) shows measurements taken on three commercial PRVs. For the first valve the pressure/voltage relationship is highly non-linear at low pressures, and with poor hysteresis characteristics. Hysteresis is dominated by the electromagnetic components of the valve such that if the applied voltage direction is changed then the pressure does not immediately respond until a small voltage threshold change has occurred. The second valve shows a much better characteristics with a much reduced hysteresis characteristics. The third valve shows also a very poor characteristics due to the hysteresis.

These characteristics do show the importance of understanding the application performance required and then selecting the correct proportional pressure relief valve. However, a poor hysteresis characteristic may well be acceptable for particular application.





(c)

Figure (4) Test of three different valves

V. CONCLUSIONS

This paper presents a new approach towards a model based control design of a new engine valve actuation system. The nonlinear feedback control is built into the design of the system which greatly simplifies the external control and reduces the calibration effort required for the system. To improve the performance and simplicity of the system, a critical control parameter is determined and its design is formulated as a problem to be optimized. The dimensions of the valve became an obstacle that we had to solve, and so we proposed and calculated a numerical solution based on a reduced order model.. To check the effectiveness of the obtained designs is verified using simulations. The most important issues associated with the design based on the reduced order model results into the development of a computationally feasible method for solving the higher order optimization problem.

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