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DEVELOPMENT OF A SENSOR-LESS ELECTRO-HYDRAULIC VALVE ACTUATOR FOR A CAM-LESS ENGINE

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ABSTRACT

A fully electronically controllable valve-actuator has potentials of improving volumetric efficiency at gas exchange, and of presenting variable compression-ratio. This paper describes a new type of hydro-mechanical position-servo electro-hydraulic valve actuator for a cam-less engine. The piston is positioned hydro mechanical feedback mechanism, which balances the flow rate of the feedback slot to the pilot valve. The feedback-slot opens its area in proportion to the piston displacement. This mechanism succeeds to decelerate the landing-velocity of 0.2m/s at a working speed of 1.0m/s for valve lift of 12mm, and to actuate the valve at a volumetric efficiency of 95% within lag times of opening 18ms and closing 8ms, that is possible to work at an equivalent engine rotational speed of 2100rpm. The actuator is energized by the hydraulic power source of flow-rate of 6.7L/min and pressure of 14MPa.

KEYWORDS

Engine-valve actuator, Hydro-mechanical servo piston, Landing-velocity control, Switching valve

INTRODUCTION

A fully electronically controllable valve-actuator has a potential not to improve volumetric efficiency of intake or exhaust gas exchange, but to present variable compression ratio. An electro-hydraulic actuator(1) has a feature of high force density but motion speed is usually not so quick at a limited power compared to an electro-magnetic actuator(2). This paper describes a new trial of a hydro-mechanical position servo actuator that improves speed and efficiency for a 2800rpm cam less engine. The actuator positions the valve without any electronic position sensor by using a new type of hydraulic piston with feedback slot, and this makes it possible to reduce the landing velocity. This type of actuator falls into unstable at a design of too small under lap of the feedback slot and high-pressure operation. The stability is examined by linear theory first, and more detail design parameters are fixed by using bond graphs analysis. It clarifies power waste and landing speed simultaneously by simple expression. The prototype is designed to actuate a valve of 430g mass, 12mm in displacement within delay of 18ms, and to close the valve within the landing velocity of 0.2m/s at a control pressure of 14MPa.

HYDRO-MECHANICAL POSITION CONTROL ACTUATOR

A schematic view of the pilot operated hydraulic piston with feedback slot is shown in Fig.1. The sectional area of the upper chamber is smaller than the pilot one, and it closes the valve stationary by high-pressure fluid through the slot. The piston opens when the pressure in the pilot chamber is reduced by opening the pilot valve. The flow rate from the pilot valve and the feedback slot should be balanced, and then the piston is stopped at a settled pilot valve opening because the slot is designed to have a opening area in proportion to the piston displacement.

LANDIMG VELOCITY CONTROL

Linear analysis is important for over viewing the dynamics of the actuator. The piston motion is expressed by

$$m\frac{d^2x}{dt^2}+b\frac{dx}{dt}+k_s x = A_s P_s - A_c P_c - F_d \qquad (1)$$

where *m, b* and k_s are equivalent mass, damping coefficient and spring constant, A_s and A_c are cross sectional areas of piston, and P_s and P_c are constant supply pressure and control pressure in the pilot chamber.

The continuity of fluid flow in the pilot chamber including fluid compressibility is expressed by

$$Q_c - Q_p + A_c\frac{dx}{dt} = \frac{V_c}{K}\frac{dP_c}{dt} \qquad (2)$$

where the pilot flow rate Q_p is expressed by

$$Q_p = C_{dp}A_p\sqrt{\frac{2}{\rho}P_c} \qquad (3)$$

Flow rate from the feedback slot Q_c by

$$Q_c = C_{dc}S_x\sqrt{\frac{2}{\rho}(P_s - P_c)} \qquad (4)$$

where the opening area of the feedback slot S_x is given by

$$S_x = a_x W_c (x + x_l)\cos\theta\,. \qquad (4)$$

W_c is the width of the feedback-slot and a_x is the area gain expressed by depth against stroke, □ is slope angle of slot, and x_l is under lap of opening slot at valve closure.

Linearlized flow characteristics of pilot valve in Eq.(3) and slot (4) are expressed by

$$Q_p = k_{qp}\,x_p + k_{cp}\,P_c \qquad (5)$$

$$Q_c = k_{qc}x - k_{cc}P_c \qquad (6)$$

where k_{qp},k_{qc} are flow gains, and k_{cp},k_{cc} are pressure–flow coefficients. The linearlized flow continuity is

$$k_{qc}x - k_{cc}P_c - k_{qp}x_p - k_{cp}P_c + A_c\frac{dx}{dt} = \frac{V_c}{K}\frac{dP_c}{dt} \qquad (7)$$

Laplace transformed formula of valve motion and flow

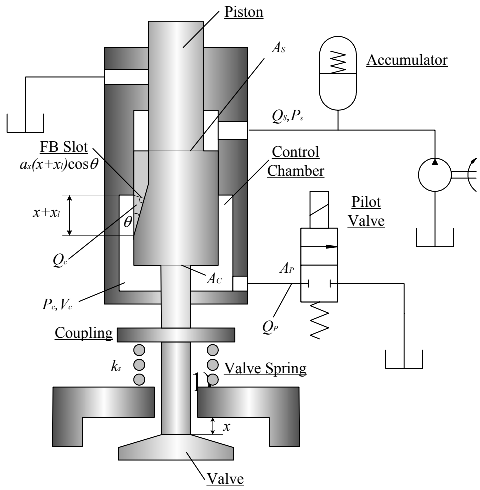


Fig.1 Schematic view of the sensor-less valve actuator with a hydro-mechanical feedback slot

continuity are expressed by

$$\left(k_{cc}+k_{cp}+\frac{V_c}{K}s\right)P_c = \left(k_{qc}+A_c s\right)x - k_{qp}x_p \qquad (8)$$

$$\left(m_s{}^2 + b_s + k_s\right)x = -A_c P_c - F_d\,. \qquad (9)$$

The block diagram of the valve actuator is described in **Fig.2**, where parameters are expressed as follows. (3)

$$k_{ce} = k_{cc} + k_{cp} \quad , \quad \omega_s = \sqrt{\frac{k_s}{m}} \quad , \quad \zeta_s = \frac{b}{2\sqrt{mk_s}} \qquad (10)$$

The stability limit of this mechanism is expressed by using Hurwitz criteria.

$$4\zeta_s{}^2 + \frac{2\zeta_s\{\alpha + \omega_s{}^2 T_v (T_c + \alpha T_v)\}}{\alpha\omega_s T_v} + \frac{1}{\alpha}\left(\frac{T_c}{T_v} - 1\right) > 0$$

(12)

where $\alpha = k_s k_{ce}/(k_{qc}A_c)$, $T_v = V_c/(Kk_{ce})$ and $T_c = A_c/k_{qc}$

Evaluating the stability of Eq.12, we can get T_v = 1.31x10^{-9} (s), T_c = 5.56x10^{-3} (s) and T_c / T_v >1, so that the Eq.12 is always realized.

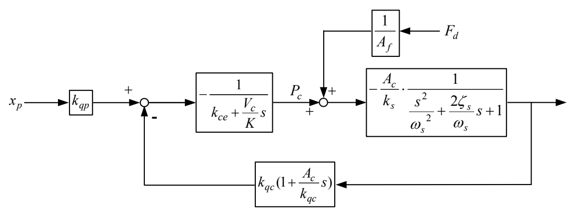


Fig.2 Block diagram of the electro-hydraulic valve actuator

BOND GRAPHS ANALYSIS AND EXPERIMENT

Valve landing-velocity and power waste are examined by using bond graph analysis in this section.

Bond graphs of the actuator A single bond expresses two physical meanings of flow and effort that denote a combination of velocity and force in mechanical elements, or flow-rate and pressure in hydraulic elements. **Figure 3** expresses the bond graph of the actuator. The discharge flow *SF* (Q_s) from a pump is regulated by a relief valve *R* (resistance) through the hydraulic accumulator *C* (capacitance), and pressurized fluid is flown into the pilot valve *R* through the feedback slot *R*. The opening area of the feedback-slot *R* is expressed by a linear function of the actuator displacement in the mechanical spring *C* (k_s).

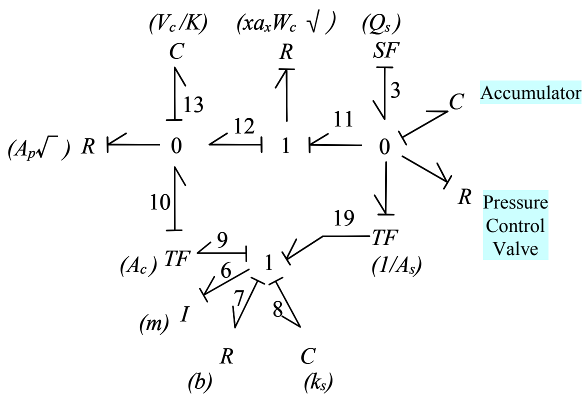


Fig.3 Bond graphs of the electro-hydraulic valve actuator operated by a switching pilot valve

Effect of mass on the response and landing speed **Figure 4** simulates the effect of mass on the response and landing velocities at a limited design of flow rate 6.7L/min and pressure 14MPa. We can see that the opening delay is a shortly improved by using lighter mass of 200g from 430g. **Figure 5** shows also a simulated result of the velocity and flow-rate for positioning the 430g-actuator. The actuator is decelerated within 0.2m/s at landing.

Comparison of bond graphs analysis and measurement **Figure 6** shows a comparison of bond graphs analysis and measurement of the step response *x* of the 430g-actuator. The high speed switching pilot valve is operated within lag time of 0.8ms by using an electric quick driver [3]. The actual landing of the actuator is decelerated strongly at x=–0.5mm

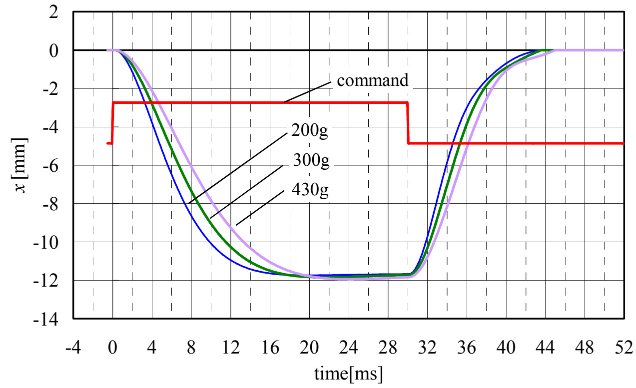
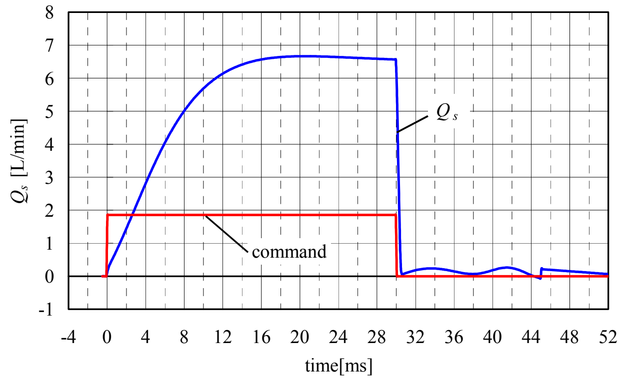
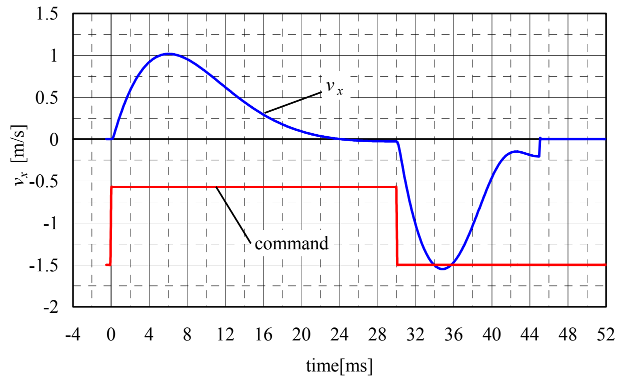


Fig.4 Effect of mass on the step response of the valve actuator by Bond Graphs Analysis at P_s = 14MPa



(a) Flow rate from the inlet port



(b) Valve landing velocity

Fig.5 Flow rate (a) and valve landing velocity (b) by Bond Graphs Analysis at P_s = 14MPa

that will be caused by the reduction of the discharge coefficient of the feedback slot at a high pressure control. **Figure 7** is a calculated example of using the reduced discharge coefficient from 0.7 to 0.2 at x=–0.5mm.

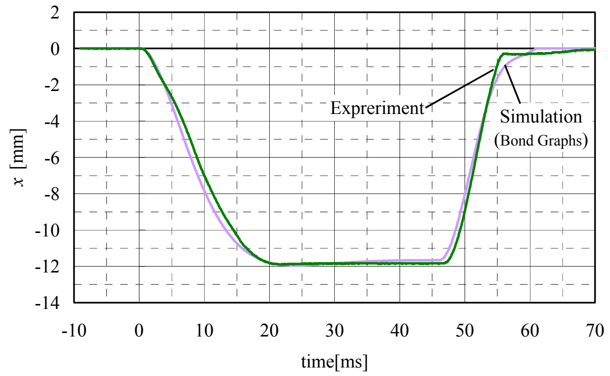


Fig.6 Comparison of the measurement and Bond Graphs Analysis of the valve motion at Ps=14MPa.

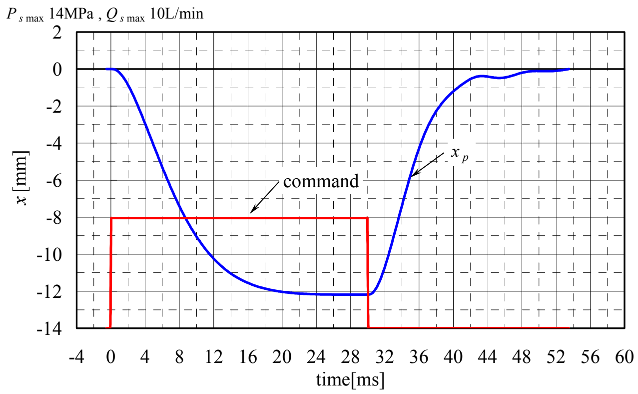


Fig.7 The strong deceleration of the landing velocity at x=-0.5mm is explained by the reduction of the flow coefficient of the feedback slot from 0.7 to 0.2 in the bond graphs analysis.

Displacement control by pressure The valve lift is influenced by the supply pressure due to the valve spring. The spring is used to close the valve at normal condition. **Figure 8** shows experimental results of the effect of the supply pressure on the lift. The lift changes from 6.8mm at 10MPa to 12mm at 14MPa linearly. This means that the lift is also controllable by the pressure besides the pilot valve opening..

Response speed measurement **Figure 9** shows measurement of frequency response of the actuator at the fixed valve opening crank angle of 120 degrees. The frequencies of 6, 12 and 18 Hz are equivalent to engine rotational speeds of 700, 1400 and 2100 rpm. The valve actuator opens 12mm in stroke and closes at a regulated landing speed. The actuator is difficult to work over the engine rotational-speed of 2800 rpm at a limited hydraulic

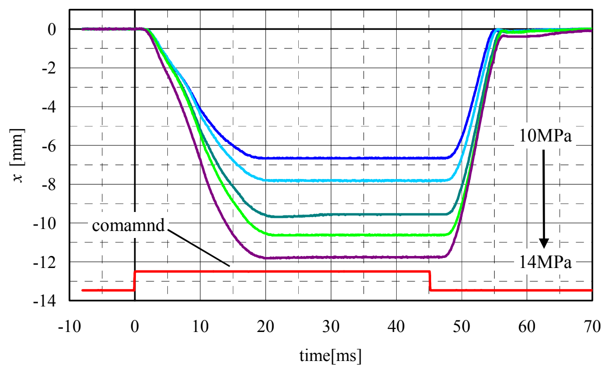


Fig.8 Measurements of the effect of the supply pressure on the valve lift

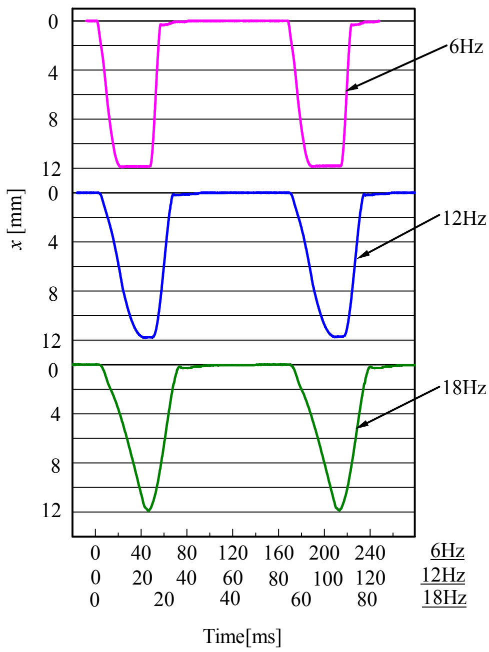


Fig 9 Measurement of step responses at three cyclic frequencies (We can see that all of the landing velocities are decelerated. Frequencies correspond to the engine rotational speeds of 700, 1400 and 2100rpm.)

power; however using high power-source in addition to larger pilot valve will solve this limit.

Landing velocity control by two pulse command The landing velocity is also controlled by using two-pulse command for strong deceleration. The pilot valve is opened again in a short time before landing as shown in **Fig.10.** We can see that the landing velocity becomes under 0.1mm/s.

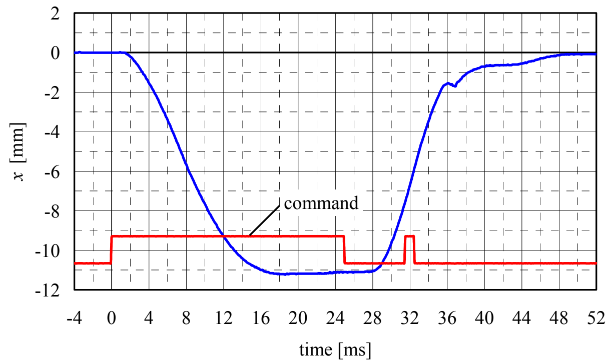


Fig.10 Measurement of the landing velocity deceleration by the two-pulse control

Figure 11 shows a quick response test result by using 2 switching valves per piston, which makes it possible to work the engine at a maximum rotational speed of 2400rpm

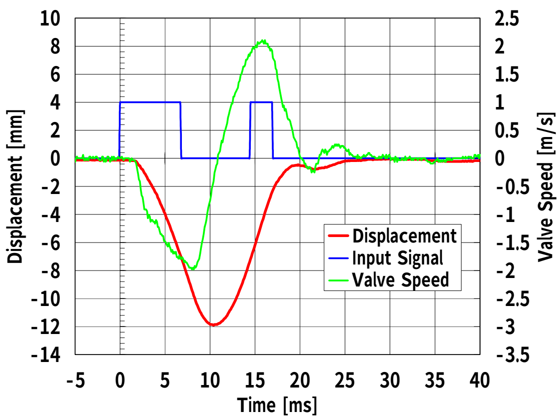


Fig.11 Quick response test by using 2 switching valves per piston for working the engine at a maximum rotational speed of 2400rpm

MESUREMENT OF VOLUMETRIC EFFICIENCY OF THE ACTUATOR

This actuator is controlled by a half-bridge hydraulic-circuit for saving the hydraulic power-waste. The loss is caused only by the piston leakage, which is measured as shown in **Fig.12**. The volumetric efficiency is given by the volume of leakage against that of one stroke of the piston

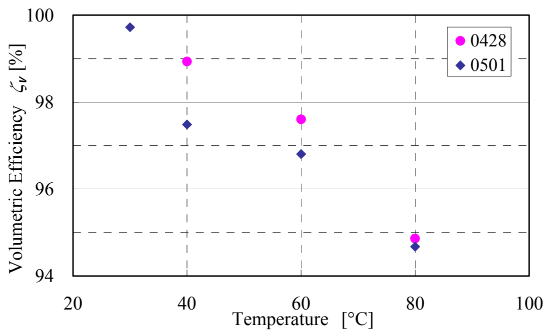


Fig.12 Measurement of volumetric efficiency of hydraulic valve actuator

CONCLUSION

A newly designed hydro-mechanical position-servo electro-hydraulic valve actuator for a cam-less engine is proposed. The piston is positioned by balancing the flow rate of the feedback slot and the pilot valve. The feedback slot is designed to be open the area in proportion to the piston displacement. This mechanism succeeds to decelerate the landing velocity of 0.2m/s at a working speed of 1.0m/s for valve lift of 12mm by the single pulse control, and to actuate the valve at a volumetric efficiency of 95% at lag times of opening 18ms and closing 8ms, that is possible to work at an equivalent engine rotational speed of 2100rpm. The actuator is energized by the hydraulic power source of flow-rate of 6.7L/min and pressure of 14MPa. A pair of intake- and exhaust-valve actuators is shown in Photo.1, and their outlook of mounting on a 6-cylinder 12L displacement engine. head in Photo 2..

ACKNOWLEDGEMENT

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Photo.1 Electro-hydraulic valve actuator for gas exchange valve operated by high speed switching valves

Photo 2 Valve actuators mounted on a 12L engine head