A HIGH PRESSURE 2 PORT VALVE FOR CYCLIC PRESSURE TEST OF A DIESEL FUEL PIPE

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Abstract

A high pressure electro-hydraulic 2 port valve of 6mm in diameter and 0.4mm stroke is developed for a cyclic pressure test of diesel fuel pipes. The valve compensates steady-state flow force and reduces leaks from both valve seat and lands as minimum as possible for controlling 200MPa diesel fuel. This paper describes the valve design, test results of flow force compensation and of relation between leak and Hertzian pressure at the valve seat, leak from the clearance of land. It is applied to the cyclic pressure test of a pipe of 560mm in length and 1.59mm in bore.

Key words

2 port valve, diesel fuel, high pressure, flow force, leaks

1.Introduction

Recent diesel fuel injection systems are worked in high pressure for improving tail pipe exhaust emissions, which closes up the reliability of pipes or connectors in cyclic pulsated pressures. For medium pressure test, a confined volume system using an intensifier is usually applied, however it has two disadvantages. One is batch process and the other is requirement of large power for pressurizing the large volume of chamber over to 200MPa due to the compressibility of fuel. On the other hand, valve control testing system

has advantages of continuous test and low power for small volume pressurization. Putting the latter system into actual use, it is necessary to develop an electronically controllable high pressure 2 port valve.

This paper shows design principle of 200MPa 2 port valve for diesel fuel and test results of flow force compensation and of leak characteristics at the seat and radial clearance. Following to these basic tests, pressure transients in a pipe of 560mm in length and 1.59mm in bore at a supply pressure of 200MPa are measured by using this valve.

μ :viscosity at working pressure and temperature
 μ_0 :viscosity at open air pressure and room temperature
 ρ :density of diesel fuel

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2.1 Valve design

2.2 Steady-state flow force compensation

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momentum in the valve chamber through a compensating orifice in Fig.1.

2.2.1 Suitable diameter of orifice

The valve motion is expressed by

$$m(d^2x_v/dt^2)+b(dx_v/dt)+kx_v = P_c A_b - \rho l_v(dQ_v/dt) + \rho Q_v(v_s \cos \alpha_s - v_v \cos \alpha_v) \quad (1)$$

where, 2nd term in the right hand expresses the unsteady-state flow force, and 3rd term momentum supplied by a jet from the compensating orifice. The flow rate is put to be $Q_s=Q_v$ by neglecting the leak from the valve lands.

The velocity v_v is expressed by v_s with using the flow continuity of $Q_s=Q_v$.

$$Q_s = 2(C_{dc} A_c \sqrt{2(P_s - P_v)/\rho}) = (\pi/2) C_{dc} d_c^2 v_s \quad (2)$$

$$Q_v = C_{dv} A_v \sqrt{2(P_v - P_d)/\rho} = (C_d \pi d_0 x_v \sin \alpha_v) v_v \quad (3)$$

In Eq.(2) the multiplier 2 means the valve has 2 inlet ports with orifices.

The steady-state flow force is balanced by giving the 3rd term in Eq.(1) to be zero. Then the diameter of the compensating orifice should be

$$d_c = \sqrt{2C_{dv} d_0 x_v \tan \alpha_v \cos \alpha_s / C_{dc}} \quad (4)$$

Figure 2 shows a calculated example of d_c at $C_{dc}=0.70$, $C_{dv}=0.72$, $\alpha_s=45\text{deg}$, $\alpha_v=30\text{deg}$ expressed by

$$d_c = 0.92 \sqrt{d_0 x_v} \quad (5)$$

For the design of valve stroke of $x_v=0.2\text{mm}$, suitable diameter of the compensating orifice is $d_c=1.0\text{mm}$.

Figure 3 shows an experimental result of valve motion x_v against pressure difference of $P_L=P_s-P_d$. We can see that the valve is kept to be the initial position of $x_{v0}=0.2\text{mm}$ by the flow force compensation.

2.3 Leaks from the seat and lands

2.3.1 Leak from the valve seat

Leak from the valve seat is usually in relation to the set force of the spring, which will have some relationship with the Hertzian pressure of the seat.

Table 1 Specification of the 2 port valve

Diameter of Spool	d_0	6 [mm]
Angle of Suction port	α_s	45 [deg.]
Angle of Main valve	α_v	30 [deg.]
Diameter of Boost-piston	d_b	13 [mm]
Spring-constant	k	1.6×10^5 [N/m]
Supply Pressure (max.)	$P_{s\text{max}}$	200 [MPa]
Displacement of Valve	x_v	0.4 [mm]
Radial clearance	C_l	2 [μm]

Fig.2 Theoretical relationship between compensating orifice diameter and valve displacement ($C_{dc}=0.7$, $C_{dv}=0.72$)

Fig.3 Steady-state flow force compensating test results (effect of supply pressure and initial valve-opening)

Fig.4 Calculated Hertzian pressure of valve seat

Figure 4 shows a calculated chart of maximum Hertzian pressure P_{max} against spring set force F_s . Figure 5 shows test results of leak at four spring set forces of $F_s=160\text{N}$, 320N , 480N and 540N . From Fig.5 we can see if we settle the $F_s=320\text{N}$ which corresponds to the maximum Hertzian pressure of $P_{max}=210\text{MPa}$. For controlling diesel fuel pressure of $P_s=250\text{MPa}$, we can see that the spring set force of $F_s=480\text{N}$ is adequate for zero leak as shown in Figs.4 and 5.

2.3.2 Leak from the valve lands

This valve has two sliding lands and their radial clearances are designed to be $C_1=C_2=2\mu$, and the lengths are $L_1=12\text{mm}$ and $L_2=19\text{mm}$ in both sides. The leaks are measured at $P_v=P_s$ by closing the outlet port. The leaks increase with pressure exponentially shown in Figs.6(a) and (b) of which characteristics are different from the usual Couette flow of linear expression of pressure through an axis symmetrical clearance.

$$Q_{11} = [\pi r_1 / (6 \mu L_1)] C_1^3 P_s. \quad (9)$$

For theoretical analysis, both changes of viscosity and clearance are considered. The viscosity is expressed by introducing parameters of λ and κ for temperature and pressure increases.

$$\mu = \mu_0 \exp(-\lambda \Delta T + \kappa \Delta P). \quad (10)$$

The temperature increase ΔT is put by assuming that pressure energy of leak flow at a pressure difference ΔP is absorbed by an averaged temperature increase.

$$\Delta T = \Delta P / (\rho C_{v11}) \quad (11)$$

where ρ is density and C_{v11} specific heat of diesel fuel.

Then Eq.(10) is expressed by

$$\mu = \mu_0 \exp(\kappa - \alpha) \Delta P \quad (12)$$

where

$$\alpha = \lambda / (\rho C_{v11}) \quad (13)$$

Substituting the following values of

$\lambda = 1.5 \times 10^{-2} [^\circ\text{C}^{-1}]$, $C_{v11} = 1.97 \times 10^3 [\text{J}/(\text{kgK})]$, $\rho = 840 [\text{kg}/\text{m}^3]$ and $\mu_0 = 0.003 \text{Pa}\cdot\text{s}$, we get $\kappa = 8.1 \times 10^{-9}$ and $\alpha = 9.06 \times 10^{-9}$ in which

Fig.5 Experimental leak characteristics of valve seat

(a) Pilot chamber side (L_1)

(b) Spring chamber side (L_2)

Fig.6 Leak characteristics of valve lands

Fig.7 Calculated deformation of h_1/h_2

the viscosity increase by pressure is almost cancelled by the temperature increase.

Next, the effect of elastic deformation of the valve body by pressure on the leak characteristics are considered. Figure 7 is a calculated clearance of h_2 normalized by h_1 (h_1 : high pressure side, and h_2 : low pressure side of $h_2 = C_1 = 2\mu$) of the valve body with 140mm in outer diameter and 6mm in bore at a linear pressure fall on the land. The leak is expressed by

$$Q_{l,1} = \pi d_0 (h_1 \cdot h_2)^2 / (6 \mu L_1) (h_1 + h_2) P_s \quad (14)$$

In Fig.6 (a) and (b), we can see calculated results of leaks including elastic deformation agree with experiments.

3. Measurement of Pressure in a Pipe Controlled by a Pair of Two Port On/Off Valves

Figure 8 shows a high pressure testing circuit of a pipe controlled by upstream and downstream 2 port on/off valves. The length and bore of the pipe are $L_p = 560\text{mm}$ and $d_p = 1.59\text{mm}$, respectively. Figure 9 shows measurements of valve displacements x_{vu} (upstream) and x_{vd} (downstream), and pressure transients of upstream P_u (solid line) and of power source P_s (broken line). We can see the both valves work alternatively. In this test, the supply pressure is regulated to be $P_s = 200\text{MPa}$, however the pressure at charging is lower than that due to lack of flow capacity for filling the pipe, and a small amount of fuel leaks from the downstream valve. The valve view is shown in Photo.1

Fig.8 High pressure pipe testing circuit

(a) Spool displacement

(b) Pressure of upstream side

Fig.9 Transient pressure characteristics of the pipeline ($P_s = 200\text{MPa}$, $x_{vu0} = 0.2\text{mm}$, $x_{vd0} = 0.2\text{mm}$)

Photo.1 View of 2 port valve

4. Conclusion

A high pressure 2 port on/off valve with 6mm in diameter and 0.4mm in stroke with a radial clearance of 2μ for diesel fuel of 200MPa pressure is developed.

(1) The steady-state flow force is compensated by a compensating orifice in the inlet port, which stabilizes the valve motion at a high pressure difference.

(2) The leak from the valve seat is in relation to the Hertzian pressure at the contact.

(3) The leak from the valve land is estimated by considering both viscosity change of fuel and elastic deformation of the valve body.

(4) The combination of 2 port valve operated by a 3 port pilot valve controls the pressure of pipe by a simple electric circuit.

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