EXPERIMENTAL STUDY ON HYDROSTATIC SUPPORTS 
OF WATER HYDRAULIC VALVES

Kenji SUZUKI* and Eizo URATA*

* Department of Mechanical Engineering, Faculty of Engineering 
Kanagawa University 
3-27-1 Rokkakubashi, Kanagawa-ku, Yokohama, 221-8686 Japan 
(E-mail: suzuki@mech.kanagawa-u.ac.jp)

ABSTRACT

This paper deals with characteristics of hydrostatic bearings, which is used in water hydraulic valves for supporting a valve spool to avoid wear and sticking of the spool to the sleeve. The hydrostatic bearings should have a sufficient spring constant for the supporting, while minimizing the leakage flow through the clearance between the spool and the sleeve. To satisfy these requirements, a strict combination of dimensions of the hydrostatic bearings and the clearance is required. Using a newly designed experimental rig, pressure distribution in the clearance and leakage through the clearance for various eccentricities are measured and compared with a computer simulation. The experimental value of supporting force is calculated integrating the measured pressure. The result of this study will be used to improve accuracy of the designing.

KEY WORDS

Hydrostatic Bearing, Water Hydraulics, Servovalve, Pressure distribution

NOMENCLATURE

$C_t$: difference of spool and sleeve radii 
$e$: eccentricity (distance between the spool center and the sleeve center) 
$h$: clearance between spool and sleeve 
$L$: clearance length (length of spool land) 
$L_b$: distance from spool edge to bearing center 
$p_i$: pressure at bearing port of number $i$ ($i = 1, 2, 3, 4$) 
$p_n$: nozzle pressure (= pressure at spool edge) 
$p_s$: supply pressure 
$q_i$: flow rate from bearing port of number $i$ 
$x$: spool axial coordinate on a developed plane of clearance 
$\alpha$: azimuth of the eccentricity of the spool 
$e$: dimensionless eccentricity ($= e/C_t$)

INTRODUCTION

In our water hydraulic servovalve, hydrostatic bearings support the spool to avoid abrasive wear and adhesion of the spool to the sleeve [1],[2]. In this water hydraulic servovalve, a part of the flow from the bearings is led to the nozzle in the servovalve and used in the flapper-nozzle system. In this way, the bearing fluid is utilized...
as the working fluid to control the position of the spool. Consequently, the power loss due to the leakage is reduced.

Dimensions of the hydrostatic bearings and the clearance between the spool and the sleeve should be designed to satisfy the following three confronting requirements: (1) to minimize the internal leakage, (2) to supply the flow rate required by the hydrostatic bearings, (3) to supply sufficient fluid for controlling of the spool position. For a design satisfying these requirements, the authors [31 reported a simulation result. However, measurement of actual pressure distribution has not been made yet.

Aims of this study are the following items: (1) the pressure distribution in the clearance of the spool and the sleeve, (2) the supporting force of the hydrostatic bearing, (3) the flow rates to the spool ends and the tank port.

The results of this study will improve accuracy of the designing, and become the base to determine the dimensions of the hydrostatic bearings and other parts of the valve.

**GEOMETRY OF CLEARANCE**

Figure 1 illustrates the hydrostatic bearings in this paper. Water is supplied to the four ports, and flows through the throttles, the pockets, and the clearance between the spool and the sleeve. The water flowing the right side of the pockets returns to a tank. The water flowing the left side reaches the end of the spool and is led to a nozzle in the servovalve.

The hydrostatic bearing ports must be placed at regular intervals on the circumference. Its number must be greater than three; in our case it is four for convenience of manufacturing. Each pocket is numbered by 1, 2, 3 and 4. The pressure at the pocket No. $i$ is denoted by $p_i$.

The clearance between the sleeve and the spool is sufficiently small compared with the spool radius. Then, the distribution of the clearance by an eccentric position of the spool is represented by

$$h = c_i - e\cos(\theta - \alpha), \quad (1)$$

where the base line of the angle is taken at the middle of the ports No.1 and 4. Figure 2 shows the geometry when the eccentricity occurred in the direction of azimuth $\alpha$.

We can develop the clearance to a plane as shown in Fig. 3. Since the thickness of clearance in the eccentric direction is thinner, the flow delivered from the port in that direction becomes smaller. Then, by the effect of the throttle between the port and the pocket, the pressure at the pocket of the eccentric direction increases [4].

**EXPERIMENT**

**Experimental apparatus**

An experimental apparatus in that the eccentricity between spool and sleeve are arbitrarily adjustable was constructed. A photograph and a schematic drawing of the experimental apparatus are shown in Fig. 4 and 5, respectively. The experimental rig is composed of a valve body, a sleeve and a spool unit that can rotate in the sleeve and can slide along the sleeve axis. Representative dimensions of the rig are: spool radius $R = 9$ mm, pocket radius $r = 3.75$ mm, $L = 52.5$ mm, $L_b = 15$ mm, $c_i = 30 \mu$m, and the diameter of the bearing throttle is 1.2 mm.
The axial displacement of spool \( x \) is measured with a scale set on the valve body. Spool angle \( \theta \) is measured using a protractor mounted on the spool. The eccentricity of the spool to the sleeve is controlled with two differential screws. The first threads of the differential screws are fixed on the valve block while the second threads of the screws are connected to the supporting blocks fixed to the spool. The positioning of the spool and the parallelism with the sleeve are adjusted using these two screws. Displacement and deflection of the spool are detected with two dial gages of 1-\( \mu \text{m} \) resolution.

A pressure-guiding hole of 0.8mm diameter was bored on the circumference of the spool. The boring is connected through a central hole in the spool to a pressure transducer. An arbitrary direction of the spool to the sleeve is realized by rotating the spool. By a rotation and an axial translation of the spool, we can put the pressure-guiding hole to any point in the clearance. This enables us pressure distribution measurement.

The supply pressure and the pressure at the spool ends are also measured with the other pressure transducers. The transducers are connected to a personal computer via strain amplifiers and an A/D board.

The pressure at the spool end is determined using an adjustable throttle. The flow passing through the adjustable throttle and the flow to the return port correspond to the nozzle flow and leakage flow in an actual servovalve, respectively. Each flow rate was measured with a flow meter or a measuring cylinder.

The water pressure source of this equipment is a three-plunger-type water pump of maximum pressure 21 MPa with rated flow 21 l/min.

The water temperature is kept at 40\( \pm 2 \)°C operating a water cooler and a thermostat installed in the reservoir.

**Experimental procedure**

Experiments are made with following steps:

1. Setting of the supply pressure \( p_s \).
2. Alignment of the spool axis to the sleeve axis. The spool is displaced to \( x = L_b \). The differential screw is adjusted so that the pressures at four ports \( (\theta = \pi/4, 3\pi/4, 5\pi/4 \text{ and } 7\pi/4) \) are equal. Similar operation is made at a place near \( x = L \). If their circumferential pressure distributions are uniform, then dial gages are set to zero position.
3. Setting of the nozzle pressure \( p_n \).
4. Setting of the eccentricity \( e \).
5. Measuring pressure distribution.

The pressure at the spool end is set by the adjustable throttle.

The spool is put in a desired eccentricity using two differential screws. The dial gages point the eccentricity.

**Experimental results**

Figure 6 and 7 shows the experimental results of pressure distribution for \( \alpha = \pi/4 \). In this case, the spool is moved to the port No.1. Figure 6(a) and (b) show the pressure distributions for \( e = 0 \) and 0.4, respectively. This overview is close to the distribution obtained by the simulation [3], as shown in Fig. 8.

Figure 7 shows circumferential distributions of pressure at various \( x \)-coordinates. This result reveals some irregularity of the distribution along the circumference. Although the pressure reduction in \( \theta = 7\pi/4 \) is detected, its quantity is smaller than the simulation. Pressure unbalance induces a small bending of the spool; the resultant warp of the sleeve reduces the clearance change given by the lateral displacement of the spool. Consequently, resultant pressure distribution approaches...
to that for zero eccentricity. It is also supposed that the effect of the bearing throttle may not be adequate. A more extensive experimental work including these factors should be made in a future study.

Fig. 6  Overviews of measured pressure distribution

Fig. 7  Pressure distribution along circumferences

CONCLUSION

Pressure distribution in the clearance of the spool and the sleeve are measured controlling the eccentricity of the spool. Although the distribution is similar to the simulation, changes of pocket pressures are smaller than the simulation. Deformation due to the induced lateral force should be considered in future study.

REFERENCES