# A Study on the Improvement of Reliability of High-Performance Hydraulic Servo Actuator

Yong-Bum Lee[1] and Tae-Seok Kim[2]

[1,2]*Korea Institute of Machinery & Materials, Reliability Assessment Center, Daejeon, Korea*
*lyb662@kimm.re.kr, tskim@kimm.re.kr*

## ABSTRACT

A tension-compression fatigue testing equipment is broadly used to secure the reliability of the components related to mechanics, automobiles, electronics and materials. This test equipment has the proper characteristics, varying the loads, displacements, and frequencies, for the acceleration life test of mechanical components. Also, this has used the hydraulic servo actuator for operation, which is required higher performance and durability than any other types of equipment. The functions and durability of the hydraulic servo actuator of the test equipment were backed up and have occurred the failures by overloaded, because of extending the operation life of the components due to the reliability improvement, increasing the accelerated load condition due to the extended operation environments, the life test time, and test frequency. This study analyzed the failure reasons for the hydraulic servo actuator of the tension-compression fatigue testing equipment requiring extended durability and introduced the examples improved on the reliability through the fundamental solutions.

## 1. INTRODUCTION

High-performance hydraulic servo actuators are mounted on tension-compression fatigue test equipment to improve the reliability of machines, automotives, electronic parts and materials and conduct warranty tests. These devices have the ability to differentiate load, displacement and frequency, which make them ideal for mechanically accelerated life testing. A hydraulic servo actuator used in test equipment requires much longer life than that used in the product to be tested. Despite a longer service life of products thanks to their greater reliability, greater accelerated load condition due to expanded usage environment, significantly increased duration and counts of life testing, the functionality and service life of hydraulic servo actuators for test equipment have become stagnant, which causes frequent failure in them.

The purpose of this study is to identify the causes of failure in the hydraulic servo actuator for tension-compression fatigue testing equipment that requires high performance and long service life and present cases where its reliability is enhanced by finding the right solutions.

## 2. METHODOLOGY AND RESULTS

### 2.1. Configuration of equipment and analysis

The equipment for tension-compression fatigue testing consists of test mechanism, a control panel, a hydraulic unit and others as shown in Figure 1. A test product is placed in the test mechanism before testing is performed by the control panel using hydraulic power. The test displacement and maximum load as well as frequency can be controlled. For this control to be possible, friction in the hydraulic servo actuator must be minimized to prevent stick slip due to internal friction from occurring and ensure signal-based control. Traditionally, two types of actuators using low friction seals for the cylinder and the hydrostatic bearing were used.

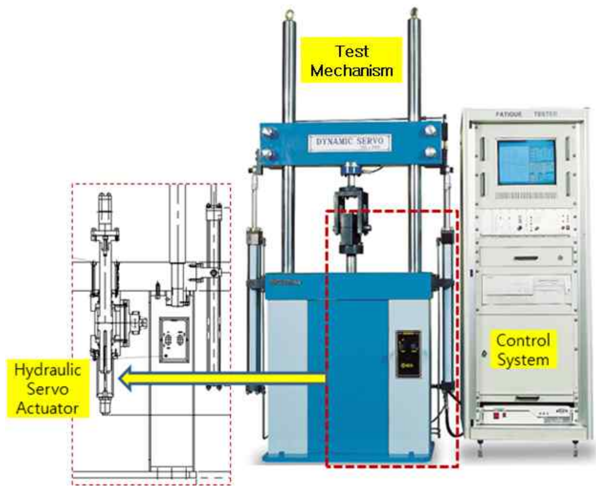


Figure 1. Components of tension-compression fatigue test equipment

Each of them has its advantages and disadvantages. First, if a low friction seal is used and the displacement is minimal, overload may take place at the seal. Burning may also result due to overheating caused by gas adiabatic compression. If a hydrostatic bearing is used, friction decreases, but significant hydro energy loss can occur. Other disadvantages include potential leakage of oil and higher cost due to the need for high-precision machining.

### 2.2. Modes of faults in hydraulic servo actuators

One of the primary sources of fault in tension-compression fatigue test equipment is the hydraulic servo actuator. The problem usually lies in burning at the contact with the seal arising out of friction between the piston's tube and rod as shown in Figure 2.

A closer look into the burning reveals that its causes include overload on the seal, misalignment between the piston and the rod, and adiabatic compression of gas contained in the hydraulic oil.

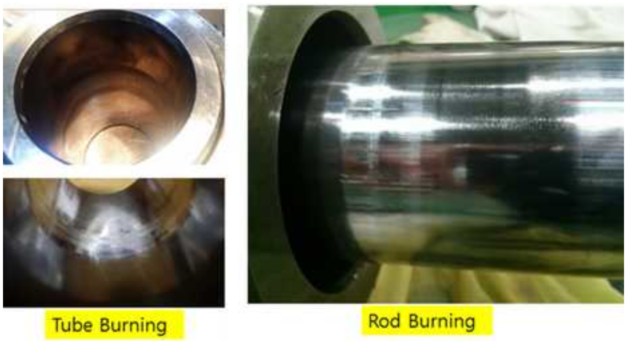


Figure 2. Photo of a fault in the hydraulic servo actuator

### 2.3. Designing a high-performance, durable hydraulic servo actuator

Designing a high-performance, durable hydraulic servo actuator for tension-compression fatigue testing equipment can be made possible by addressing the main sources of fault and applying new technology.

The problem of seal overload can be addressed by adopting a new patented technology including a low friction hydrostatic bearing. The problem of misalignment can be addressed by adopting another patented technology to ensure eccentricity. The problem of adiabatic compression can be solved by using a patented technology that enables discharge of air.

New technologies used include energy-saving capability achieved by a hydrostatic bearing, independent displacement, and control of load and frequency.

#### 2.3.1. Designing a hydrostatic bearing for piston rod

A hydrostatic bearing having four pockets capable of minimizing friction in the actuator and controlling lateral force during the testing was designed as shown in Figures 3 and 4. In addition, a pressure relief valve circuit was applied as shown in Figure 5 in order to minimize loss of hydro energy.

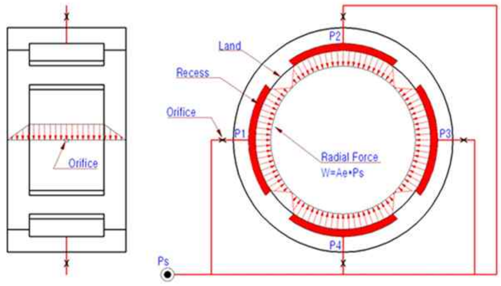


Figure 3. Operation of a hydrostatic bearing

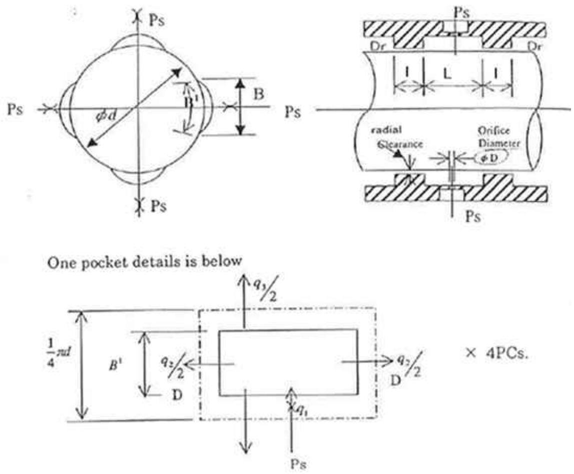


Figure 4. Flow characteristics of a hydrostatic bearing

For a hydrostatic bearing used in a regular hydraulic servo actuator, the flow from orifice $q_1$ having a pressure of 210 kgf/cm$^2$, is calculated as shown in formula (1), and the flow through clearance $q_2$ is calculated as shown in formula (2). The flow $q_3$ through clearance from one pocket of the bearing to another is so minimal that the difference in pressure between the adjacent pockets is negligible. This results in no flow in formula (3). The pressure inside the pocket can be calculated as shown in formula (4). Oil leakage consumed during the flow through the hydraulic bearing can be calculated as shown in formula (5). Therefore, a power loss of approximately 3.1542 kW occurs when supply pressure is 210 kgf/cm$^2$ and the leakage flow of oil amounts to 9.012 L/min/600.

$$q_1 = CA\sqrt{\frac{2g}{r}(P_s - p)} \times 60 \tag{1}$$

$$= C \times \frac{\pi}{4} D^2 \sqrt{\frac{2g}{r}(P_s - p)} \times 60 (cc/\min)$$

$$q_1 = 0.6 \times \frac{\pi}{4} 0.045^2 \sqrt{\frac{2 \times 980}{0.87 \times 10^{-3}}(210 - p)} \times 60$$

$$= 79.2\sqrt{210 - P}$$

$$q_2 = \frac{\delta_o^3 P}{12 \times \frac{\alpha}{100} \times \frac{\gamma}{g} \times l} \times \frac{\pi}{4} d \times 2 \times 60 \tag{2}$$

$$q_2 = \frac{0.005^3 \times P}{12 \times \frac{32}{100} \times \frac{0.87 \times 10^{-3}}{980} \times 2.5} \times \frac{\pi}{4} \times 7.5 \times 2 \times 60$$

$$= 10.4P$$

$$q_3 = 0 \tag{3}$$

$$q_1 = q_2 + q_3 (q_3 = 0)$$

$$q_1 = q_2$$

$$85.938\sqrt{210 - P} = 29.49P \tag{4}$$

$$P = 38.2 (kgf/cm^2)$$

$$q = 29.49 \times 38.2 \times 4pcs \times 2set \tag{5}$$

$$= 9012 (cc/\min)$$

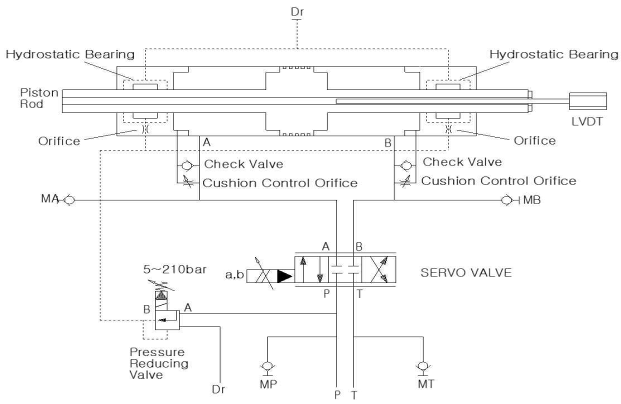


Figure 5. A hydraulic bearing using a manual pressure relief valve

To reduce hydro energy loss in the hydraulic bearing, the supply pressure for the bearing (Ps) is lowered to 30 percent of its original level (approximately 70 kgf/cm$^2$), as shown in Figure 5. This results in a power loss of 0.5642 kW when the supply pressure is 70 kgf/cm$^2$ and the leaked oil is 4.836 L/min/600, which corresponds to 17.9 %(4836 cc/min) of the level when a maximum pressure of 210 kgf/cm$^2$ is supplied.

### 2.4. Designing to protect from damage due to adiabatic compression in the piston

The hydraulic servo actuator for tension-compression fatigue testing equipment is used in the 1 to 80 percent range of their maximum stroke. The actuator's minimal displacement causes hydraulic oil inside it to fail to be fully discharged and maintain the initial pressurization. This results in just enough supply of hydraulic oil for control of micro-displacement. Gas flowing in with the oil remains in the actuator, impairing the precision of control, and, if trapped between the cylinder and the piston, causes adiabatic compression as shown in formula (6). This results in a rise in temperature which in turn accelerates deterioration of lubrication and, as a result, leads to an increase in friction which can cause a drop in efficiency or fault due to excess wearing.

$$T_2 = T_1 \times \frac{P_2 \bullet V_2}{P_1 \bullet V_1} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \tag{6}$$

The oil temperature for the hydraulic system is maintained at 50℃ and the maximum service pressure at 21 MPa. Therefore, gas remaining inside can be heated up to over approx. 500℃ to burn the seal, resulting in an increase in friction.

$$T_2 = (273 + 50) \times \left(\frac{21}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$\fallingdotseq 771 (°K) \fallingdotseq 498°C$$

To address this problem, this study uses the labyrinth seal on the right instead of the seal on the left as shown in Figure 6. This way, gas can be discharged when a small amount of fluid flows out.

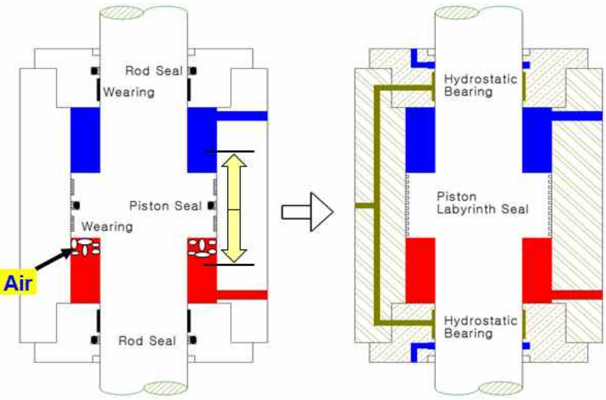


Figure 6. Comparison of the existing piston seal type and the new labyrinth seal type

### 2.5. Designing of prototype and test results

Testing of a hydraulic servo actuator created for tension-compression fatigue testing equipment as shown in Figure 7 reveals that displacement control was accurate both in low and high-speed ranges as shown in Figure 8, and load control was also accurate as shown in Figure 9. In addition, when the engine mount springs with large unbalanced loads were tested for 1 million cycles, no damage was found to have been caused to the actuators..

Figure 7. The prototype of the hydraulic servo actuator mounted

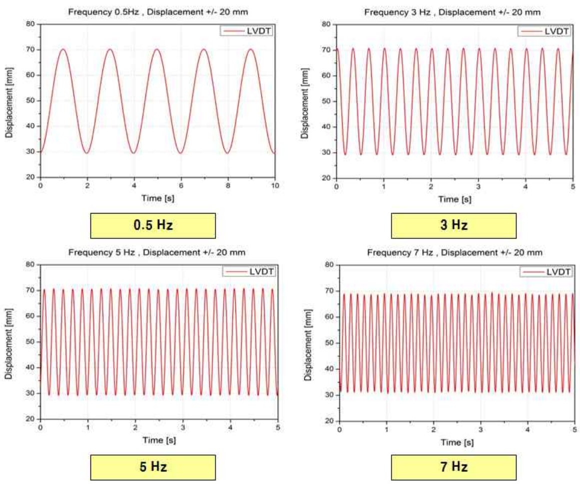


Figure 8. Results from displacement control testing

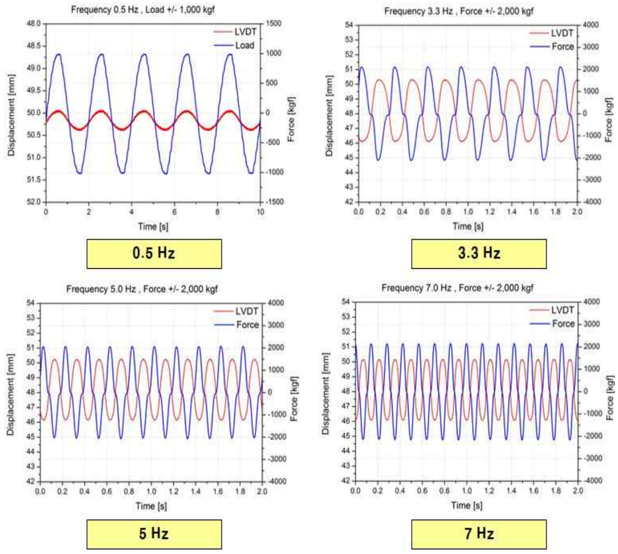


Figure 9. Results from load control testing

## 3. CONCLUSIONS

Assessing the reliability of machine parts requires test equipment having much higher proven duration and performance than the products to be tested.

Locally developed products are being launched to take the place of their imported, expensive counterparts, many of which, however, turn out unsuccessful due to low technological competitiveness.

This paper proposes a technology that can identify the cause of fault in a hydraulic servo actuator, which is a critical component of tension-compression fatigue testing equipment, and help enhance the competitiveness of the testing equipment.

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## NOMENCLATURE

| | |
|---|---|
| C | flow coefficient |
| D | orifice diameter |
| $r$ | oil specific gravity |
| $g$ | gravity acceleration |
| $P_s$ | supply pressure |
| P | internal pressure of pocket |
| $\delta_o$ | radial clearance |
| a | oil viscosity |
| l | clearance length |
| $d$ | rod diameter |

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