2C4-3

Proceedings of the 6th JFPS International
Symposium on Fluid Power, TSUKUBA 2005
November 7-10, 2005

A STUDY ON THE APPLICATION OF BIRFIELD JOINT TO A WATER HYDRAULIC PISTON PUMP FOR LOW LEAKAGE AND LOW FRICTION PUMPING

Young-Bog Ham*, Yong-Bum Lee**, Kyung-Min Park***, Byung-Oh Choi*

* Intelligence and Precision Machinery Research Division, Korea Institute of Machinery and Materials, 171 Jang-dong, Yusung-gu, Daejeon 305-343, South Korea
(E-mail: hyb665@kimm.re.kr)
** System Engineering Research Division, Korea Institute of Machinery and Materials
*** Powertrain Development Center, P/T Project Planning Team, Ssangyong Motor Company

ABSTRACT

As water hydraulics is more and more highlighted in wide applications, many varieties of research on design, materials and manufacturing methods of water hydraulic components have been carried out. Especially, water hydraulic piston pump used to be the subject of study in relation to its relative sliding movement. In these relative sliding parts, the leakage and friction probably took place because of the characteristics of low viscosity and poor lubrication of water. In order to reduce the leakage and friction, the relative sliding parts are filled up nearly by using engineering plastic as an anti-wear material on sliding parts. Compared with an oil hydraulic system, a water hydraulic system has fundamental disadvantages of large friction, high leakage, high wear, corrosion, etc. A mechanism of rod type piston was considered to reduce friction of the piston's movement in a water hydraulic piston pump. Friction and leakage of the rod type piston was analyzed and compared with those properties of the conventional plunger type piston. An experimental device was set up to evaluate friction and leakage of those pistons. A PEEK piston was also tested as another alternative piston of a water hydraulic pump.

KEY WORDS

Water hydraulics, Piston, Pump, Birfield joint

NOMENCLATURE

A : contact area of viscous friction
c : radial clearance
d : diameter of piston
e : eccentricity of piston
F_c : coulomb friction force
F_f : total friction force
F_L : lateral force
F_R : reaction force under piston shoe
F_v : viscous friction force
h : thickness of oil film
L : passage length
L_1 : overlap length of piston
L_2 : overhang length of piston
P_d : discharge pressure
r : tube radius
v : velocity of relatively sliding
α : inclination angle of tilting cam
λ : coulomb friction coefficient
μ : viscosity of oil
ξ : inclination angle of piston rod

INTRODUCTION

In order to save oil energy and protect the environment, it is needed to reconsider the use of the existing oil hydraulic system, which uses a petroleum fuel. As a result, a water system is aroused our interest again in power transmission, and recently various studies have been conducted on water hydraulic systems that use a water as a hydraulic fluid, since water has cleaner and more economic advantages, as well as excellent safety against fire, ease of access, convenient management, and easy recycling, than oil. [1]
The most basic component in an oil or water hydraulic system is a pump, which make and supply the power on system. And, almost pump in water hydraulic system is adopted the piston pump for more high pressure on the system, when compares to other types. The water hydraulic piston pump is almost the similar structure of oil hydraulic piston one. However, the water hydraulic piston pump requires special design concepts for each sub components due to the difference in physicochemical properties between oil and water. [2]
Also, these researches on water hydraulic components and systems are made actively in Kitakawa Lab. of Tokyo Institute of Technology, Ikeo Lab. of Sophia University, and Sanada Lab. of Yokohama University, etc. In addition, the Nessie Water Hydraulics of Danfoss in Denmark and the Ebara Research Institute in Japan are producing and selling water hydraulic systems as well as pumps/motors, actuators, and valves.

PISTON AND BIRFIELD JOINT

In conventional piston type pumps, the plunger type piston is mostly used. But high friction is occurred between plunger type piston and cylinder bore through relative sliding movements, as shown in Figure 1.

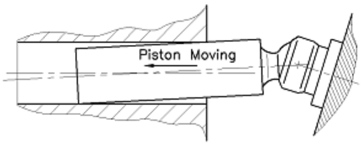


Figure 1 Tilted movement of the plunger type piston

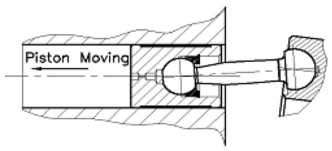


Figure 2 Parallel movement of the rod type piston

Because friction and wear results from the lateral force on the edge of both ends of a plunger type piston is highly intensive, there is a need for another piston design that can reduce this lateral force.
In this paper, as a method to reduce the lateral force, rod type piston with a ball-socket joint of both spherical-surface ends, as shown in Figure 2, is applied. This will be make dispersion the concentrated load on the edges by offsetting the lateral force from the spherical joint, even though the piston is affected by the lateral force as it moves in the bore of the cylinder block.
However, the rod type piston requires an assisting mechanism for special operation. Generally, the plunger type piston is acted on the lateral force when the pump shaft rotates and the piston shoes slides on the swash plate, but the rod type piston is rotated and moved not by the lateral force but by the structural combination of the tilting cam and Birfield joint. Birfield joint is located between the pump shaft and the tilting cam, as shown in Figure 3, and supported to rotate with constant velocity.

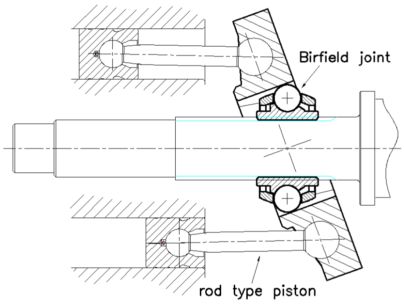


Figure 3 Birfield joint applied on a rod type piston and tilting cam

Besides, for smoother operation, the hydrostatic bearings are applied to the sliding components.
Figure 4 shows the rotary group of water hydraulic piston pump proposed in this study.

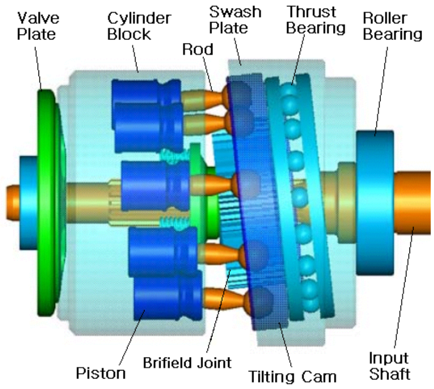


Figure 4 Water hydraulic piston pump consist of rod type piston and Birfield joint

FRICTION

Coulomb Friction of Plunger Pistons

In a plunger type piston pump, the power is transmitted through the side of the piston so that a strong lateral force is generated.

As shown in Figure 5, the lateral force takes or the strong concentrated force acts on each end of the piston, which is tilted to the side. The large concentrated force at the contact point makes coulomb friction and the viscous friction generated equally around the piston. The coulomb friction force is expressed the following Equation (1).

$$F_c = F_{c1} + F_{c2} = \lambda \cdot F_{L1} + \lambda \cdot F_{L2} \quad (1)$$

Here, λ represents the coulomb friction coefficient.

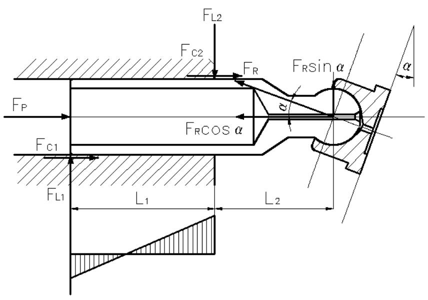


Figure 5 Inclined shape of the plunger piston in transferring the lateral force

In order to predict the effect of the coulomb friction, the equilibrium equation of the force can be used for the analysis.

From the equilibrium equation of the y-axis force and the moment for the zero point, it can be expressed as follows:

$$F_{L2} = F_{L1} + F_R \sin\alpha \quad (2)$$

$$F_R \sin\alpha \cdot L_2 - F_{c2}\frac{d}{2} - F_{L1} \cdot L_1 + F_{c1}\frac{d}{2} = 0 \quad (3)$$

The moment equilibrium equation in Equation (3) omitted the viscous friction because the viscous friction that works around the piston is proportionally distributed in the center point of the piston, zero.

If, F_R, which is the force applied to the piston shoe from the swash plate, is to be measured with the load cell, it is useful to express the coulomb friction F_{c1}, F_{c2} for F_R. If the coulomb friction is induced using Equation (1), (2), and (3), it is as follows:

$$F_c = \lambda(1 + \frac{2L_2}{L_1} - \frac{d}{L_1}\lambda)F_R \sin\alpha \quad (4)$$

Equation (4) shows that the coulomb friction on the piston can be expressed with the proportion function of F_R, which is the force on the piston shoe from the tilting plate. In addition, the proportion coefficient is the value determined with the size and friction coefficients of the piston.

The coulomb friction force on the plunger piston is determined by the tilting angle of the swash plate, and the tilting angle usually ranges from 14 degrees to 18 degrees so it has a very large friction force.

In the motion equation for the piston movement direction (x-axis direction), if the inertia force and friction force of the piston are to be ignored, since they would be smaller than the force of the pressure, F_R, the force on the piston from the swash plate can be approximated as follow:

$$F_R \approx P_d \cdot \frac{\pi d^2}{4} \cdot \frac{1}{\cos\alpha} \quad (5)$$

When Equation (5) is substituted in Equation (4), it is as follows.

$$F_c \approx \lambda(1 + \frac{2L_2}{L_1} - \frac{d}{L_1}\lambda)P_d \frac{\pi d^2}{4} \tan\alpha \quad (6)$$

This shows that the coulomb friction force on the plunger type piston increases in proportion to the discharge pressure of the pump.

Coulomb Friction of a Birfield Joint and Rod type Piston

The configuration of a pump that applied a Birfield joint and rod type piston shown as Figure 4. Most of the torque is transmitted from the main shaft of the pump to the tilting cam through the Birfield Joint. As shown in Figure 6, out of the reacting force generated from the tilting cam; $F_R \sin(\alpha - \xi)$ which is generates torque, is transmitted through the Birfield Joint, so there is no lateral force at the sides of the piston.

The lateral force of the rod type piston, F_L, is affected by the tilting angle, ξ, of the piston rod. This can be expressed as Equation (7).

$$F_L = F_R \cos(\alpha - \xi)\sin\xi \quad (7)$$

Usually, the tilting angle, ξ, ranges from ± 1° ~ ±

2° so the lateral force, F_L, is very small.

$$F_c = \lambda F_L = \lambda F_R \cos(\alpha - \xi)\sin\xi \qquad (8)$$

When Equation (5) is substituted to Equation (8), it becomes Equation (9).

$$F_c \approx \lambda \cos(\alpha - \xi)\sin\xi \cdot P_d \cdot \frac{\pi d^2}{4\cos\alpha} \qquad (9)$$

The coulomb friction force of the rod type piston can be expressed with the proportion function of F_R, the force on the piston shoe from the tilting cam. The proportion function is very small due to the small effect of, and as a result, the coulomb friction is very small. Therefore, it can be determined that the rod type piston used in pump having the Birfield Joint has a smaller friction force than the force of the plunger piston, which is shown in Equation (6) and (9).[3],[4]

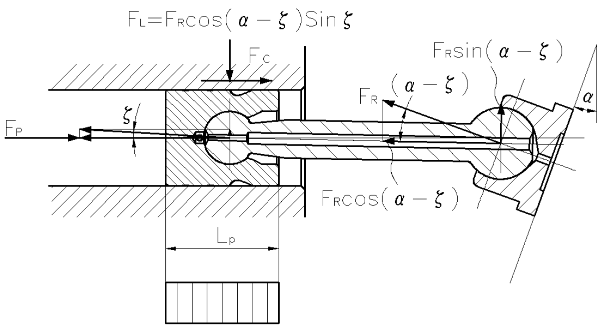


Figure 6 Force diagram of rod piston
Vicious Friction of Piston

Viscous Friction of piston

The friction force, F_f, in a normal hydraulic unit is combined from the coulomb friction force, F_c and viscous friction, F_v.

$$F_f = F_c + F_v \qquad (10)$$

The viscous friction force generated from the united area between the cylinder bore and piston is as follows:

$$\tau = \mu \frac{\partial v}{\partial y} \qquad (11)$$

Here, μ refers to the viscosity and $\frac{\partial v}{\partial y}$ represents the speed inclination. The viscous friction force, F_v, is expressed as follows.

$$F_v = \mu A \frac{v}{h} = \frac{\pi d L \mu}{h} \frac{dx}{dt} \qquad (12)$$

Here, A refers to the contact area. The viscous friction force is irrelevant to the configuration of the piston. However, the rod type piston is very good for shortening the length of the contact point of the piston, $L = L_1$. When the length of the contact point of the piston is shortened, the contact area becomes smaller so that it receives a smaller viscous friction force than the plunger type piston.

LEAKAGE

As shown in Figure 7, the dynamic viscosity of water is 1/100 of that of petroleum oil (in a temperature of 20℃), so the leakage for the same clearance can be very different from that of oil.

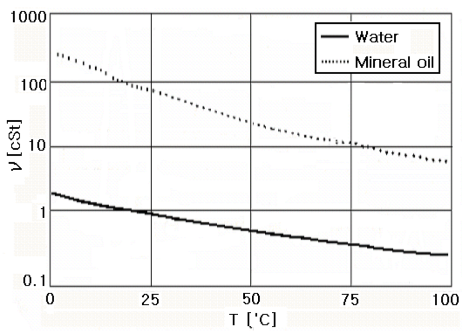


Figure 7 Kinematics viscosity of water and oil

The clearance between the piston and bore can be expressed as Figure 8. The leakage equation, which determines the leakage of this clearance, is expressed as follows: [5]

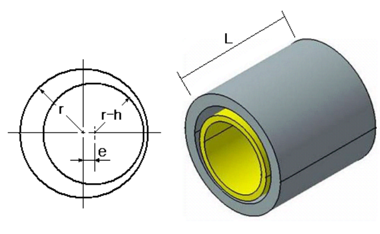


Figure 8 Clearance of piston and cylinder bore

$$Q = \frac{\pi r h^3}{6\mu L}\left[1 + \frac{3}{2}\left(\frac{e}{h}\right)^2\right](P_1 - P_2) \qquad (13)$$

The biggest factors that determine the leakage flow rate, Q, are the contact length and the clearance. Therefore, when considering the viscosity of water, the clearance between the piston and the cylinder should have a very small value and the contact length should be increased in order to reduce the leakage.

However, an increase in contact length increases the viscous friction force, so it is difficult to have the right contact length. Therefore, a precise processing technique that minimizes the clearance of the contact point is required.

Equation (13) shows that the configuration of the piston is irrelevant to the leakage of the piston. However, the eccentricity (e) between the piston and the bore is affected by the configuration. As shown in Figure 5, the plunger piston has a maximum value for the eccentricity (e) when the inside of the bore is tilted. Therefore, the leakage flow rate will have a maximum value.

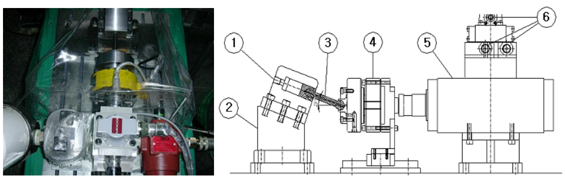
When the rod type piston and Birfied Joint are applied, the eccentric value (e) would be relatively small and so the leakage flow rate would also be small.

TEST

Test Equipment

As shown in Figure 9, 3 types of pistons were prepared for the test; plunger piston, rod piston, and PEEK plunger piston. In order to measure and compare the friction and leakage of these pistons, the test apparatus as Figure 10 was made.

Figure 9 Three types of pistons for test



① Connecting port of water hydraulic pump
② Cylinder block ③ Test piston ④ Load cell
⑤ Servo valve ⑥ Servo valve connector

Figure 10 Schematic diagram of test apparatus

Friction Test

The test was done with water pressure of 10 bar, 20bar, 30bar and 40 bar. In order to measure the friction, the sine wave signal was input to the servo value to move the piston. As mentioned in previous, the shape of the piston (whether it is plunger type, rod type, and PEEK material) affects the Coulomb friction and is irrelevant to viscous friction. Therefore, the research mainly performed tests to measure the Coulomb friction and compared the results.

As shown in Equation (6) and (9), the Coulomb friction can be determined with the force from the tilting cam to the piston shoe, F_R. Therefore, F_R was measured using the load cell sensor.

The test was done on each piston in the same condition. The sizes of 3 types of pistons were all the same overhang length of piston. The cylinder bore and clearance sizes were all same.

Results of Friction Test

Figure 11 shows the signal graphs of load cell measured at the pressure of 30 bar. The signal of the plunger type piston changes greatly in proportion to the sine wave signal. The load cell signal is very high when the plunger piston enters the cylinder bore and it becomes low when the piston reverses from the cylinder bore. On the other hand, the rod type piston had lower load cell signal. There was hardly any difference in the signals of forward motion and reverse motion of the piston.

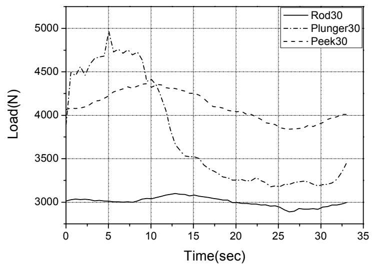


Figure 11 Load cell signals measured at the pressure of 30 bar

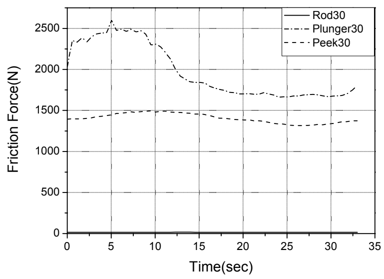


Figure 12 Coulomb friction for three types of piston at the pressure of 30 bar

Figure 12 shows the coulomb friction force for the three types of pistons. The friction of Figure 12 was calculated from the load cell signal of Figure 11.

Such test result matches the theoretical analysis. In other words, when the rod type piston is applied to the water hydraulic pump, the friction is very small and has excellent lubrication.

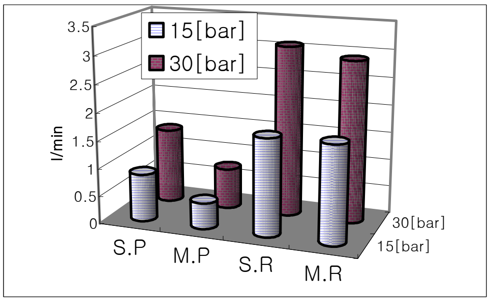
The friction for PEEK plunger type piston is smaller than friction for metal plunger piston, but it is much larger than the friction for rod type piston even though a new material, PEEK is applied as a material of plunger.

Leakage Test

The leakage test was done along with the friction test. The leakage test was done after 30-second initial operation and the system was settled to steady state. The leakage amount was measured using a stop-watch and electric weight scale. The test was done on each piston in the same condition. The sizes of 3 types of pistons were all the same overhang length of piston. The cylinder bore and clearance sizes were all same.

Results of Leakage Test

Leakage is not related to the material of pistons. The leakage test was done only on plunger type and rod type, and Figure 12 shows the leakage flow measured at the test.



S.P : Stopped Plunger, M.P : Moving Plunger
S.R : Stopped Rod, M.R : Moving Rod

Figure 12 Comparison of leakage flow

In 15 bar, the leakage flowrate of the rod piston doubled the leakage flowrate of the plunger piston when the piston is stopped. Even when the piston is operating, the leakage flowrate of the rod piston was four times greater than that of the plunger type. In 30 bar, the results were similar to the results in 15 bar condition.

The leakage of rod type piston was greater than the plunger type piston because the rod type piston was 1/2 the length of the plunger type piston and the resistance length for the leakage flow was shorter.

In other words, the length of the piston should be extended in order to reduce the friction and leakage flow with the rod type piston.

CONCLUSION

In the plunger type piston pump, the lateral forces on the edges of the plunger piston are the source of friction and wear, so a ball socket joint piston rod with spherical surface ends was proposed. When the rod type piston is applied, it is rotated and moved by the tilting cam, so a Birfield joint was applied to the space between the pump shaft and the tilting cam as a constant velocity joint.

The study examined friction in the piston and the cylinder for pumps with a plunger type piston and a rod type piston. As a result, the friction of the rod type piston was a lot smaller than the friction of the plunger type piston. The PEEK plunger piston has low friction compared to the metal plunger type piston but the friction is much larger than the friction of the rod type piston. The engineering plastic, PEEK, seems to be not so effective as the rod piston in reducing the friction of piston.

The leakage flowrate is irrelevant to the structure of the piston but it is determined by the contact or overlap length and clearance between cylinder bore and piston. In the test of the research, the rod type piston had twice larger leakage flowrate than the plunger and this is why the length of the rod type piston is 1/2 length of the plunger type. In order to reduce the leakage flowrate, the overlap length of the piston should be extended.

The study results show that the pump with the rod type piston and Birfield Joint is better than the conventional plunger type piston pump.

ACKNOWLEDGMENTS

The authors gratefully acknowledge financial support from the Korea Energy Management Corporation. (http://racer.kemco.or.kr)

REFERENCES

1. Karl-Erik Rydberg, “Energy Efficient Water Hydraulic Systems”, ICFP, 2001.
2. Songlin Nie and Zhuangyun Li, “Design Techniques for Water Hydraulic Axial Piston Pumps(Motors)”, NFPA, 2000, I00-16.5, pp 581-590.
3. R.A. Wehage, E.J. Haug, “Dynamic Analysis of Mechanical Systems with Coulomb Friction, Stiction, Impact and Constraint Addition-Deletion-I,II,III”, Mechanisms and Machine Theory, 1986, Vol.21, No.5, pp.401-425.
4. Y. B. Ham, S. D. Kim, “A study on the application of tripod joints to transmit the driving torque of axial piston hydraulic motor", Proceedings of the 5th JFPS International Symposium on Fluid Power, NARA, November 14.2002, vol.2, pp 531-536.
5. Herbert E. Merritt, "Hydraulic control systems", John Wiley & Sons Inc, 1967, pp 9-13.