The Experimental Analyses of the Effects of the Geometric and Working Parameters on the Circular Hydrostatic Thrust Bearings*

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In this paper, the characteristics of disk-type hydrostatic thrust bearings supporting concentric loads; simulating the major bearing/seal parts of axial piston pumps and motors were investigated. An experimental setup was designed to determine the performance of slippers, which are capable of increasing the efficiency of axial piston pumps and motors, for different conditions. The working parameters and the slipper geometry causing the minimum frictional power loss and leakage oil loss were determined. Since slippers affect the performance of the system considerably, the effects of surface roughnesses on lubrication were studied in slippers with varying hydrostatic bearing areas and surface roughness. The results of the study suggest that the frictional power loss and leakage oil loss were caused by the surface roughness, the relative velocity, the size of the hydrostatic bearing area, supply pressure and capillary tube diameter.

Key Words: Hydrostatic Thrust Bearings, Slipper Bearings, Frictional Power Loss, Leakage Oil Loss

1. Introduction

Disk-type hydrostatic thrust bearings used to prevent contacts between metals is a common the example of supporting elements working under a critical velocity which resist axial load and have a relative motion with respect to each other. These bearings are called axial hydrostatic bearings. The critical elements of axial piston pumps and motors are the slippers and the slipper plates used in hydraulic power transmission systems.

Piston pumps with a wide range of capacities to produce fluids with varying pressures are manufactured in two ways as radial and axial pumps. In these types of pumps, small pistons of rotate in a piston block embedded in the trunk. The axial flow volume in piston pumps and motors is controlled by modifying the stroke of the pistons. The piston stroke is varied by either tilting the piston block or the slipper plate.

The experimental works carried on the behaviour of slippers have involved in the analyses of stationary slippers^{(1),(2)}.

Wang and Yamaguchi⁽³⁾ presented the characteristics of disk-type hydrostatic thrust bearings supporting concentric loads which simulated the major bearing/seal parts of water hydraulic pumps and motors. They evaluated the characteristics by investigating the relationships among the load carrying capacity, recess pressure, film thickness, and leakage flow rate.

Wang and Yamaguchi⁽⁴⁾ also theoretically investigated the load carrying capacity, power losses and stiffness of disk-type hydrostatic thrust bearings under eccentric load for elastic and rigid materials respectively. In their paper, a numerical analysis method was employed to analyses a two-dimensional elastohydrostatic problem using an elastic deformation model.

Koç and Hooke⁽⁵⁾ have examined the design of hydrostatically balanced bearings used in the slippers of high pressure axial pumps, and outlined a design procedure whereby the slipper behavior, minimum film thickness and loss of high pressure fluid could be estimated. It was shown that for a successful operation the slippers needed to have a certain degree of curvature on the running surfaces. In addition, a favorable agreement between the measured and calculated film thickness was demonstrated.

The same researchers presented an experimental investigation of the performance of hydrostatic slipper bearings in axial piston pumps and motors⁽⁶⁾. The effects of

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clamping ratio, offset loading and capillary tube size on the behavior of overclamped and underclamped slippers were outlined. It was shown that the slippers ran satisfactorily with no capillary tube and had their greatest resistance to tilting couples and to minimum film thickness.

Ineffective lubrication might be the cause of metal-metal contact; therefore oil jet pressure must be maintained to prevent oil starvation. In high pressure hydraulic equipments it is a normal practice for many of the bearings and seals to be designed to operate partly hydrostatically. In some pumps, such as the end plates or bushes of gear pumps and the valve plates of axial piston pumps, this is achieved by adjusting the pressurized areas so that the hydrostatic loads are nearly balanced, leaving a small residual clamping load to be carried by the hydrodynamically generated pressures⁽⁷⁾.

Canbulut et al.⁽⁸⁾ developed a neural network predictor for analyzing the rigidity variations of hydrostatic bearing system. They investigated the size of bearing pocket and the orifice diameter was employed to analyze these two parameters. A suitable optimization method based on neural network was designed. The proposed neural predictor exactly followed their experimental results.

A neural network predictor for the analysis of oil leakage in partially hydrostatic slipper bearings was presented in Refs. (9) and (10). Since slippers affect the performance of the system considerably, the effects of surface roughness on lubrication were studied in slippers with varying hydrostatic bearing areas and surface roughnesses. The authors determined that the neural network structure was very suitable for this kind of system and capable of predicting the quantity of oil leakage in the experimental system.

The most common method of lubricating the slippers is to use the oil in the pump and to generate pressure under the slipper in order for the slippers to float on an oil film. This pressure or lift may be generated entirely hydrodynamically. Alternatively, the lift may be obtained largely hydrostatically by tapping the piston pressure. As hydrostatic lubrication mechanism is the most common method, in this investigation this type of slipper is used.

The power loss occurs due to viscous friction of the fluid between slipper and slipper plate in the axial piston pumps and motors. It is important to determine the optimum slipper geometry and working parameters to minimize the frictional power loss. In this paper, the hydrostatic bearing diameter, the capillary tube diameter conducting oil to the bearing and hydrodynamic bearing surface roughnesses were analyzed. The oil viscosity was kept fixed while the supply pressure and relative velocity varied as the other working parameters. The frictional power loss and oil leakage loss were investigated under these conditions

Nomenclature

R_1	: inner radius of bearing (mm)
R_2	: outer radius of bearing (mm)
h_o	: average film thickness (μm)
ρ	: density of oil (kg/m^3)
d	: diameter of the capillary tube (mm)
μ	: dynamic viscosity (cP)
Q_c	: flowrate through the recess (mL/min)
H_f	: frictional power loss (watt)
l	: length of the capillary tube (mm)
W	: load capacity (N)
U_M	: relative velocity (m/sec)
n	: number of revolutions (rpm)
D_s	: piston diameter (mm)
A	: combined area of pad and recess (mm^2)
r	: radius (mm)
p_r	: recess pressure (bar)
R_a	: average surface roughness (μm)
p_s	: supply pressure (bar)
Q_r	: radial flow rate (mL/min)
h	: film thickness (μm)
R	: rotating radius (mm)
u_r	: radial flow velocity (m/sec)
τ	: shear stress (N/mm^2)
ω	: angular velocity of pad (rad/sec)

2. Experimental Technique

2.1 Experimental setup

In this study the effects of slippers which enhance the efficiency of axial piston pumps and motors on lubrication were examined under variable surface roughness and working conditions. An experimental setup was designed to examine the performance of slippers. The schematic experimental setup was presented in Fig. 1⁽¹¹⁾.

Experimental setup consists of three basic units. These are the main test unit where the testing of slippers is performed, the hydraulic power unit where the power needed for the experiment is produced and the driving unit with control and measurement devices.

Slippers in axial piston pumps and motors function on the hydrostatic bearing principles. The main test unit mounted is shown in Fig. 2. This main test consists of a hydrostatic bearing and a slipper plate, a hydraulic loading cylinder and three slippers and leakproof elements.

Since hydrostatic bearings resist sudden increases of load instantly, they enable the moving components to work more steadily. A hydrostatic bearing with a circular pocket is provided for the slipper plate, which is in relative motion with respect to slippers to work without vibration. The slipper plate moved by a shaft from the gearbox is situated on the hydrostatic bearing. The slipper plate is designed in such a way as to prevent the mixing of oil coming from the hydrostatic bearing with the oil from

Part No	Part Name	Part No	Part Name
1	Universal Lathe	12	Hydraulic loading cylinder circuit distributor
2	Main test unit	13	Direction control valve
3	Tank	14	Hydraulic loading cylinder circuit constraint valve
4	Circulating pump	15	Hydraulic loading cylinder circuit pressure adjustment valve
5	Hydraulic loading cylinder circuit power unit	16	Wattmeter
6	Hydrostatic bearing hydraulic circuit power unit	17	Wattmeter
7	Temperature measurement unit	18	Manometer
8	Hydrostatic bearing hydraulic circuit pressure adjustment valve	19	Leakage accumulation hoses
9	Hydrostatic bearing hydraulic circuit constraint valve	20	Oil return hoses
10	Multiple pressure demonstrator	21	Measurement container
11	Hydraulic loading cylinder circuit distributor	22	Compressor

Fig. 1 The view of experimental setup

Fig. 2 Main test unit exploded view

the slipper plate. The surface of the slipper plate which is responsible for slipping is ground to a surface roughness quality of $0.6\text{ }\mu\text{m}$. The measured angular displacement of the slipper plate during the running was $2.5\text{ }\mu\text{m}$.

There are three cylinder barrels on the hydraulic loading cylinder to position the slipper at an angle of 120° (Fig. 3). The connections in the cylinder are designed in such a way as to allow each slipper to be loaded equally.

The supply pressure is controlled by a manometer on cylinder cover. Figure 3 shows the solid model of slippers manufactured in view of the working condition of real slippers, and pressure measurement locations on the bottom sliding surface of the slippers with outer diameters of $\phi 41\text{ mm}$.

Slippers are made of brass. The oil leakage between

the cylinders and the slippers were prevented by using leakproof oil rings. Moreover, the surface roughness of sliding side on the slipper plate is processed with different values in keeping with the aim of the experiments.

Oil barricade made of cast aluminum is used to prevent the oil leakage from the bottom of the slippers from mixing with oil coming out of hydrostatic bearing. The experimental setup is supplied with two hydraulic power units. One of them feeds the hydrostatic bearing with oil to lift the slipper plate and the other unit sends oil to the top of the slipper pocket, this oil moves through the capillary tube and loads slippers situated in the cylinder block.

2.2 Experimental method

In the circular pocket hydrostatic bearing, oil was transferred to the hydrostatic bearing pocket by a capillary tube. While this oil provides lubrication between the slipper plate and slipper it also creates a pressure between slipper plate and slippers on circular pocket bearing and thus produce buoyancy.

Some amount of energy was used to drive the slipper plate and the power drawn from the system was measured by operating the setup without load. When working slippers were loaded by sliding on the slipper plate, friction force occurred between the slipper face and the slipper plate. This force depended on supply pressure, the quality of slipper faces, the amount of oil and velocity of the oil transferred to bearing pocket. The power driving slip-

Fig. 3 The schematic and solid view of slipper and slipper plate as well as their geometric dimensions

Table 1 Factors and their values used experiments

Factor	Average roughness	Hydrostatic pocket ratio	Capillary tube diameter	Supply pressure	Relative velocity
Level	$R_a (\mu m)$	R_1 / R_2	$d (mm)$	$p_s (bar)$	$U_M (m/sec)$
1	0.7	0.7	0.7	10	0.52
2	1.5	0.4	0.5	20	1.08
3	9.5	0.1	0.3	30	2.26
4					3.34

per plate changed depending on the lubrication between two faces which are in relative motion with respect to each other.

In the measurement of frictional power loss and oil leakage loss, the driving unit (turning lathe) was operated without load and the power was measured then the system was operated under load and the power was again measured. The difference between these measured values was the frictional power loss. The oil leakage was measured using a calibrated beaker. All this measurements were performed only for one slipper. The constant geometric and working parameters used in the experimental work are shown in Table 1.

Mobil DTE 26 oil is used as oily fluid in the experiment. The temperature of the oil at one bearing was kept around 25°C ($\mu = 175$ cP). Therefore, an air-cooled system was used to keep the temperature constant. The oil temperature was at 59°C ($\mu = 15$ cP) under the bottom of the slipper.

3. Results and Discussion

The effects of experimental parameters on the power and oil leakage losses were presented as follows.

3.1 Surface roughness

Figure 4 (a) and (b) show the effects of surface roughness on frictional power loss and oil leakage loss as a function of capillary tube diameter. In Fig. 4 (a) the effects of three surface roughnesses (0.7, 1.5, 9.5 μm) on frictional power loss was presented. It is shown that for a surface roughness of 1.5 μm the power loss was minimum for each capillary tube diameter. The power loss was min-

(a) The variation of frictional power loss with roughness average in various capillary tube diameter

(b) The variation of oil leakage rate with roughness average in various capillary tube diameter

Fig. 4 Frictional power loss and oil leakage rate versus roughness average

imum for a 0.7 mm capillary tube diameter for all surface roughnesses. This was because of increased amount of oil due to greater tube diameter. The flow rates of oil entering and exiting slipper pocket are given in Eqs. (1) and (2), respectively.

$$Q_c = \frac{\pi d^4}{128 \mu l} (P_s - P_r) \quad (1)$$

and

$$Q_r = \int_0^h 2\pi r u_r dy = \frac{\pi}{6 \ln(R_2/R_1)} \frac{h^3 p_r}{\mu} \quad (2)$$

Where, d and l are the diameter and length of the capillary tube, and μ is the dynamic viscosity of the fluid.

Where $u_r = \frac{1}{2\mu} \frac{dp}{dr} y(y-h)$ is the (radial) flow velocity^{(3),(4)}. Q_c and Q_r are flow rates entering exiting the pocket. Since the flow is continuous $Q_c = Q_r$. As Q_c is a function of 4th power of capillary tube diameter, this diameter has a very significant effect on Q_c .

Similarly, the frictional power loss was also given in Eq. (3)^{(3),(4)}.

$$W = \pi R_1^2 p_r + 2\pi \int_{R_1}^{R_2} r p dr = \frac{1 - (R_1/R_2)^4}{2 \ln(R_2/R_1)} A p_r \quad (3)$$

The geometry is characterized by the inner (recess) radius R_1 and the outer (pad) radius R_2 . Here A is the combined area of pad and recess, p_s is the supply pressure, p_r is the recess pressure.

Frictional power loss is given by

$$H_f = \int_{R_1}^{R_2} r \omega \tau dA = \frac{1 - (R_1/R_2)^4}{2} \frac{\mu U_M^2 A}{h} \quad (4)$$

where $\tau = \mu(du_r/dr)$ is the shear stress on the runner. $U_M = (\pi n/30)R$ is the relative velocity between the pad and the runner.

As an addition to the information given in Fig. 4 (a) ($R_1/R_2 = 0.7$, $p_s = 20$ bar, $U_M = 3.34$ m/sec) if $d = 0.3$ mm, $l = 5$ mm, $R_2 = 20.5$ mm, $T = 59^\circ\text{C}$, $\mu = 15$ cP, $P_r = 13$ bar and $R = 1.5 \mu\text{m}$ are used the power loss is calculated. If equations of flow, the film thickness calculated as $h_0 = 24.43 \mu\text{m}$. If this film thickness is used in Eq. (4), frictional power loss is calculated as $H_f = 3.43$ W. This calculated theoretical frictional power loss, as agrees with the experimentally measured value ($H_f = 3.5$ W, Fig. 4 (a)). More power is drawn from system as the capillary tube diameter decreases. The reason for this is that the amount of the oil pumped into hydrostatic bearing area decreases as the capillary tube diameter reduces. Therefore, the reaction force on the bearing area fails to reach the sufficient magnitude for maintaining, the balance, than the slippers can not approach the slipper plate, and since it can not provide enough lubrication, the frictional power loss is increased. This balance may be established by increasing capillary tube diameter more easily and as a result the frictional power loss is decreased. For the same working conditions used for the investigation of frictional power loss, oil leakage loss increased with the capillary tube diameter. As Fig. 4 (a) and (b) show the selection of a surface roughness of about $1.5 \mu\text{m}$ minimizes the frictional power loss while the oil leakage loss was kept considerably low.

3.2 Capillary tube diameter

Figure 5 (a) and (b) show the effects of capillary tube diameter on the frictional power loss and oil leakage.

(a) The variation of frictional power loss with capillary tube diameter in various supply pressure

(b) The variation of oil leakage rate with capillary tube diameter in various supply pressure

Fig. 5 Frictional power loss and oil leakage rate versus capillary tube diameter

In order to investigate slipper behaviour under different operating conditions, different capillary tube sizes and loading pressures were used.

Figure 5 (b) shows the variations of oil leakage as a function of capillary tube diameter, under the conditions of 0.7 hydrostatic bearing ratio, $1.5 \mu\text{m}$ surface roughness and speed of 3.34 m/sec. The oil leakage increased with loading pressure. The oil leakage increase in oil leakage is faster between the capillary tube diameters 0.3–0.5 mm than between 0.5–0.7 mm.

In Eq. (1), if ΔP is decreased this increase in oil leakage is confirmed. As the difference between the loading pressure and pocket pressure (ΔP) decreases the oil flow through capillary tube to the bearing pocket also decreased. Moreover, the increases of the orifice diameter from 0.3 mm to 0.5 mm create a 70% change in the oil leakage compared to a 40% change when the diameter is increased from 0.5 to 0.7 mm (compare the slope of the curves).

Figure 5 (a) shows that the frictional power loss is decreased as the capillary tube diameter is increased. This because of improved lubrication as a result of increased oil quantity with increased capillary tube diameter. The choice of capillary tube diameter between 0.5–0.7 mm

decrease the frictional power loss and keeps the oil leakage at a reasonable level.

3.3 Bearing pocket ratio

The effects of bearing pocket ratio on the frictional power loss and oil leakage are shown in Fig. 6 (a) and (b). The figures indicate that between the bearing pocket ratios from 0.1 to 0.4 the frictional power loss is very high while the oil leakage is also significantly high. At a bearing pocket ratio of 0.6 for all the three pressure values the frictional power loss is about the same. As the ratio is further increased, the frictional power loss continues to drop but at the same time the oil leakage increases significantly. Therefore in order to balance these two trends, a selection of bearing pocket ratio between 0.5 and 0.6 give a good solution. Solmaz et al.⁽¹²⁾ have concluded in a work involved the bearing optimization of hydrostatic circular bearing that the optimum bearing pocket ratio for optimum frictional power loss was 0.53.

3.4 Supply pressure

Figure 7 (a) shows the variation of frictional power loss as a function of supply pressure at various pocket ratios. As the figure shows that frictional power loss increased with supply pressure for pocket ratio of 0.1 and

0.4 while it decreased with the increase of supply pressure for a pocket ratio of 0.7. The increase rate was steeper for 0.1 pocket ratio.

Figure 7 (b) shows the variation of oil leakage as a function of supply pressure for the same. In all cases oil leakage increased with supply pressure. The increase was the steepest for pocket ratios of 0.7.

3.5 Relative bearing velocity

Figure 8 (a) shows the variation frictional power loss as a function of relative bearing velocity. The results were obtained for a pocket ratio of 0.7; capillary tube diameter of 0.5 mm and surface roughness of $1.5 \mu\text{m}$ was selected because of the minimum frictional power loss (see Fig. 4 (a)). In Fig. 8 (a) the maximum frictional power loss, 3.7 watt is obtained at a pressure of 10 bars and relative velocity of 1.08 m/sec. On the other hand minimum frictional power loss, 2 watt is obtained at a pressure of 30 bars and relative velocity of 3.34 m/sec. It was shown that the frictional power loss increased between the relative velocities of 0.52–1.08 m/sec and decreased between the relative velocities of 1.08–3.34 m/sec for all supply pressures. For similar working conditions, power loss was with increased supply pressure. When the supply pres-

(a) The variation of frictional power loss with radius ratio in various supply pressure

(b) The variation of oil leakage rate with radius ratio in various supply pressure

Fig. 6 Frictional power loss and oil leakage rate versus radius ratio

(a) The variation of frictional power loss with supply pressure in various radius ratio

(b) The variation of oil leakage rate with supply pressure in various radius ratio

Fig. 7 Frictional power loss and oil leakage rate versus supply pressure

(a) The variation of frictional power loss with relative velocity in various supply pressure

(b) The variation of oil leakage rate with relative velocity in various supply pressure

Fig. 8 Frictional power loss and oil leakage rate versus relative velocity

sure is low the amount of the oil flowing into bearing pocket decreases and the reaction force action on the hydrostatic area becomes insufficient to establish a balance and, therefore frictional power loss.

Similarly, the reasons for the increase of frictional power loss with relative velocity from 0.52 m/sec to 1.08 m/sec initially are the inadequate amount of oil lubrication necessary for the bearing area and the decrease of oil film thickness. In this situation, stable lubrication does not occur. As the relative velocity increased the recess pocket pressure decreases and since the pressure difference ($\Delta p = (p_s - p_r)$ in Eq. (1)) increases as a result oil flow to the pocket also increases and the frictional power loss decreases. This result is shown in Fig. 8 (a) (after relative velocity of 1.08 m/sec).

It is shown that the recess pressure p_r is 23 bars at 1.08 m/sec, and 21 bars at 3.34 m/sec ($R_1/R_2 = 0.7$, $d = 0.5$ mm, $R_a = 1.5$ μ m) at a $p_s = 30$ bars. As it is obtained from the Eq. (1), the amount of the lubrication oil going into bearing pocket is positively proportional to $(p_s - p_r)$. The amount of the lubrication oil in the pocket is increased property to the pressure difference. While, $(p_s - p_r)$ is 7 bars at 1.08 m/sec, $(p_s - p_r)$ is 9 bars at 3.34 m/sec.

As it is seen in the Eq. (4) it is obvious that the

frictional power loss H_f will increase parabolically with relative velocity. However, the experimental results (Fig. 7 (a)) do not show this variation. This can be ascribed to the variation in the viscosity and film thickness. When the velocity increases, the temperature also increases and as a result the viscosity decreases. As a result the quantity of the oil transferred to the slipper increases and consequently the frictional power loss decreases.

Figure 8 (b) shows the variation of oil leakage quantity versus the relative velocity at different loading pressures for slippers with pocket ratio of 0.4, capillary tube diameter of 0.5 mm, surface roughness of 1.5 μ m. The oil leakage quantity increased with relative velocity and loading pressure since the oil quantity sent to the slipper on the sliding surface is increased. While the slope of oil leakage rate does not increase with supply pressure for the same relative velocity ranges, the slope of becoming steeper after the relative velocity reaches 2.3 m/sec.

4. Conclusion

The slipper behavior under different operating conditions, with different capillary tube size and supply pressure was investigated to find exact design parameters for novel applications.

In the case of the 0.7 and 9.5 μ m surface roughness of slippers much power is needed to overcome the friction force between slippers and slipper plates. Frictional power loss is reduced occurs with the slippers with surface roughness of 1.5 μ m. The frictional power loss decreased with increased capillary tube diameter and supply pressure.

The oil leakage quantity increased with loading pressure and capillary tube diameter. Moreover, the oil leakage quantity diminished with capillary tube diameter. For this reason, the oil quantity transmitted to the hydrostatic bearing area also decreased. The leakage dissipation slope for the same loading pressures was less between 0.3 mm and 0.5 mm capillary tube diameters than that of between 0.5 mm and 0.7 mm capillary tube diameters. That is, when the difference between loading pressure and pocket pressure (ΔP) decreased, oil quantity from orifice to bearing pocket also decreased.

The frictional power loss increased with the increase in relative velocity (with 0.4 and 0.1 pocket ratios). The increase in power loss in slippers with 0.1 pocket ratio was higher than that in slippers with 0.4 pocket ratio. However, in the slippers with 0.7 pocket ratio, the increase of frictional power loss was insignificant between the relative velocities of 0.52 – 1.08 m/sec, but it decreased for relative velocities between 1.08 – 3.34 m/sec.

The power loss increased between relative velocities of 0.52 – 1.08 m/sec and decreased between 1.08 – 3.34 m/sec for all supply pressures. For similar working conditions, frictional power loss decreased with increased

supply pressures. When the supply pressure was low, the reaction force in hydrostatic area was not sufficient to establish the balance and, therefore as a result amount of frictional power loss increased.

The work showed that a slipper of surveys roughness of $1.5\text{ }\mu\text{m}$, pocket ratio 0.55 (Fig. 6(a)), relative velocity of 2 m/sec, capillary tube diameter 0.5 mm and supply pressure of 30 bars give good result.

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