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Suction Performance of Axial Piston Pump

(1st Report, Analysis and Fundamental Experiments)*

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In this paper, the pressure losses that occur in the cylinder port of an axial piston pump are analyzed and the critical condition for a suction operation is obtained. And using a simulated model of a rotating cylinder block, the values of loss coefficients that appear in the analysis are examined experimentally.

As the result of this investigation, it becomes clear that, for the purpose of improvement of suction performance, the radius of pitch circle of cylinder ports should be small and the crosssectional area of cylinder ports should be large.

1. Introduction

Axial piston pumps are widely used in oil hydraulics. In many axial piston pumps, switching between suction and delivery is performed by a rotating cylinder block and a valve plate. In order to maintain the force balance at the sliding surface between the cylinder block and the valve plate, the crosssectional area of the cylinder port is made smaller than the piston area. As the result, hydraulic oil is restricted at the cylinder port and a pressure loss is produced. This pressure loss is harmful to the suction performance of an axial piston pump.

In the case of closed circuit, the suction performance is not so important. But in the case of open circuit, the allowable maximum speed of a pump is limited by the critical suction speed.

In order to increase the maximum suction speed, enlarging the suction pipe or smoothing the duct in a pump has been tried. However theoretical foundations of these efforts are not established.

In this paper, the pressure drop at the cylinder port is calculated and its influences on the suction performance are discussed.

The pressure drop at the cylinder port arises from the pressure loss due to the axial flow velocity and the tangential

velocity and the pressure drop due to the centrifugal and inertia force. The flow rate at the cylinder port and the opening area of cylinder port change with the rotational angle of cylinder block. Therefore the pressure in the cylinder changes with time.

In the analysis of this paper, the pressure in the cylinder is obtained as a function of time introducing the pressure losses mentioned above. And then, the critical condition of the suction operation of an axial piston pump is obtained.

In the latter part of this paper, the values of loss coefficients that appear in the analysis are obtained using a simulated model of a rotating cylinder block.

Nomenclature

A_0 : crosssectional area of cylinder port

A_p : piston area

$F = A_p \delta / A_0 r$

P_g : air separation pressure

P_t : total pressure at valve plate

$\Delta P_c = P_t - P_g$

R_s : minimum radius from center line of cylinder block to cylinder wall

r : radius of pitch circle of cylinder ports

v_2 : mean axial flow velocity in cylinder port

v_{2m} : mean axial flow velocity in cylinder port at 90° rotation $= A_p \delta \omega / A_0$

w : tangential velocity of cylinder port $= r\omega$

w_{max} : critical value of w

δ : one half of piston stroke

ζ : loss coefficient related to w

ζ_0 : coefficient related to ζ

$\eta = 1 - (R_s/r)^2$

ξ : loss coefficient related to v_2

ρ : density of oil

ω : angular velocity of pump

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2. Analysis

Figure 1 shows a suction duct, a valve plate and a cylinder block. In the passage from the reservoir to the valve plate (from point 1 to point 2 in Fig.1), the flow is steady. And the total pressure at the port of the valve plate is given by subtracting the suction head and the friction losses in the suction duct from the atmospheric pressure as follows.

$$P_t = P_1 + \rho v_1^2 / 2 = P_a - \gamma H - (\kappa Q + \lambda Q^2) \quad \dots (1)$$

where P_t is total pressure at valve plate, P_1 is static pressure at valve plate, ρ is density of oil, v_1 is flow velocity at valve plate, P_a is atmospheric pressure, γ is specific weight of oil, H is suction head, Q is flow rate of pump and κ , λ are loss coefficients. The sign of H is positive when the pump sucks up oil.

In Fig.2, the cylinder ports and the valve plate are expanded along the pitch circle of the cylinder ports. As shown in Fig.2, the opening area of cylinder port changes with the rotational angle of cylinder block.

In the case of Fig.3, the opening area of cylinder port is given by the following equations.

For $0 \leq \theta < \theta_2$

$$A = A_0 [1 - 2\beta_1 / \pi - (1/\pi) \sin 2\beta_1] \quad \dots (2)$$

for $\theta_2 \leq \theta \leq \pi - \theta_2$

$$A = A_0 \quad \dots (3)$$

for $\pi - \theta_2 < \theta \leq \pi$

$$A = A_0 [1 - 2\beta_2 / \pi - (1/\pi) \sin 2\beta_2] \quad \dots (4)$$

where

$$\beta_1 = \sin^{-1} \left[\frac{2r}{d} \sin \left\{ \frac{\theta_2 - \theta}{2} \right\} \right] \quad \dots (5)$$

$$\beta_2 = \sin^{-1} \left[\frac{2r}{d} \sin \left\{ \frac{\theta_2 - (\pi - \theta)}{2} \right\} \right] \quad \dots (6)$$

and θ is rotational angle of cylinder port from start of suction stroke, A_0 is cross-sectional area of cylinder port, r is radius of pitch circle of cylinder ports and d is diameter of cylinder port.

In order to prevent a sudden change of the pressure inside of the cylinder, the port timing is slightly shifted in actuality. And then, in actual pumps, the opening area is a little different from Eqs. (2) - (4).

The relationship between the displacement of piston x and the rotational angle of cylinder port θ is given by the following equations:

$$x = \delta (1 - \cos \theta) \quad \dots (7)$$

or

$$x = \delta (1 - \cos \omega t) \quad \dots (8)$$

where δ is one half of piston stroke, ω is angular velocity and t is time.

The flow rate passing through the cylinder port is obtained from the piston area A_p and the piston velocity given by differentiating Eq. (8) with time as follows:

$$q = A_p \delta \omega \sin \omega t = A_p \delta \omega \sin \theta \quad \dots (9)$$

The mean flow velocity v_2 at the inlet of cylinder port is given by the following

Fig.1 Suction passage of axial piston pump

Fig.2 Arrangement of ports along pitch circle

Fig.3 Ports of cylinder block and valve plate

Fig.4 Branched duct

equation:

$$v_z = q/A \quad (10)$$

The tangential velocity w of the cylinder port is as follows:

$$w = r\omega \quad (11)$$

When oil flows into the cylinder from the valve plate, the following four kinds of pressure drops occur.

- (1) Pressure loss due to v_z
- (2) Pressure loss due to w
- (3) Pressure drop due to centrifugal force
- (4) Pressure drop due to inertia of oil

First, the pressure loss due to the axial flow velocity v_z is treated. At the start and the end of the suction stroke, the flow characteristic becomes like that of an orifice. And even when the cylinder port is fully opened, the flow characteristic of the cylinder port is also like that of an orifice because the length of the cylinder port is same order as its diameter. Then the pressure loss due to v_z is expressed as follows:

$$\Delta P_1 = \xi \rho v_z^2 / 2 \quad (12)$$

where ξ is loss coefficient.

There are several formulas which describe the flow characteristic of a sharp edged cylindrical choke. Among them, the following experimental formula⁽¹⁾ is available for the condition of the flow passing through the cylinder port at the critical condition of suction operation:

$$\xi = (1.16 + 6.25\sigma^{-0.61})^2 \quad (13)$$

where $\sigma = q/(\nu A)$ and ν is kinematic viscosity.

Second, the pressure loss due to the tangential velocity w is treated. This pressure loss appears when the oil has the velocity component in tangential direction. And it is similar to the pressure loss which appears at a branched duct as shown in Fig.4, because the relative motion between the oil and the cylinder port is the same. The pressure loss due to the tangential velocity is given by the following equation:

$$\Delta P_2 = \zeta \rho w^2 / 2 \quad (14)$$

where ζ is loss coefficient. And ζ is a function of the ratio v_z/w of the flow velocities similarly to the case of a branched duct and it increases linearly with v_z/w as follows:⁽²⁾

$$\zeta = \zeta_0 (v_z/w) \quad (15)$$

where ζ_0 is constant.

From Eqs. (14) and (15), one can obtain the following equation:

$$\Delta P_2 = \zeta_0 \rho v_z w / 2 \quad (16)$$

Third, the pressure drop due to the centrifugal force which acts on the oil in the cylinder is treated. The pressure drop ΔP_3 due to the centrifugal force which is produced by the radius difference between r and R_s is given by the following equation:

$$\Delta P_3 = \eta \rho w^2 / 2 \quad (17)$$

where

$$\eta = 1 - (R_s/r)^2 \quad (18)$$

and R_s is the minimum radial length from center line of cylinder block to cylinder wall as shown in Fig.3.

Finally, the pressure drop ΔP_4 due to the inertia force which acts on the oil in and near the cylinder port is treated. Modifying the pressure drop due to a uniform acceleration of oil in the cylinder port, ΔP_4 is given by the following equation:

$$\Delta P_4 = \varphi \frac{\rho l}{A_0} \frac{dq}{dt} \quad (19)$$

where φ is the correction factor and φ is larger than unity because the oil near the inlet and the outlet of the cylinder port is also accelerated and decelerated.

From Eqs. (12), (16), (17) and (19), the total pressure drop ΔP at the cylinder port is obtained as follows:

$$\begin{aligned} \Delta P &= \Delta P_1 + \Delta P_2 + \Delta P_3 + \Delta P_4 \\ &= \xi \frac{\rho v_z^2}{2} + \zeta_0 \frac{\rho v_z w}{2} + \eta \frac{\rho w^2}{2} + \varphi \frac{\rho l}{A_0} \frac{dq}{dt} \end{aligned} \quad (20)$$

And the pressure P_2 in the cylinder is as follows:

$$P_2 = P_t - \Delta P \quad (21)$$

One can obtain the following equation from Eqs. (9), (10), (11), (20) and (21).

$$\begin{aligned} P_2 &= P_t - \frac{\rho w^2}{2} \left\{ \xi \left(\frac{A_p \delta}{A_r} \right)^2 \sin^2 \theta + \zeta_0 \frac{A_p \delta}{A_r} \sin \theta \right. \\ &\quad \left. + \eta + 2\varphi l \frac{A_p \delta}{A_0 r^2} \cos \theta \right\} \end{aligned} \quad (22)$$

The calculated examples of Eqs. (2) - (4), (7) and (22) are shown in Fig.5. The values of ξ and ζ_0 in Fig.5 are the experimental values obtained in the latter part of this paper.

It is clear from Fig.5 that the pressure in the cylinder reduces severely in the short range at the beginning and at the end of the suction stroke. The reason for this is the small opening area of the cylinder port. And in this short range, cavitation will occur.

The cavitation bubbles which will appear at the beginning of the suction stroke will be able to disappear during the suction stroke, but the cavitation bubbles which will appear at the end of the suction stroke will be carried to the delivery side. However, the piston movement in this region is less than 0.2 % of the piston stroke. Therefore, the cavitation which occurs at the end of the suction stroke makes little trouble.

On the other hand, when the cavitation occurs in the region near $\theta = 90^\circ$, the delivery flow rate reduces severely and the noise of the pump becomes large.

Because of the inertia effect of oil, the pressure in the cylinder becomes minimum momentarily a little before $\theta = 90^\circ$. But one can regard the value of P_2 at $\theta = 90^\circ$ as the minimum value in this region approximately. This minimum value P_{2min} can be obtained from Eqs. (3) and (22) as follows:

$$P_{2min} = P_t - (\xi F^2 + \zeta_0 F + \eta) \rho w^2 / 2 \quad (23)$$

where

$$F = A_p \delta / (A_0 r) \quad (24)$$

From Eqs. (3), (9) and (10), the axial flow velocity v_{2m} at $\theta = 90^\circ$ can be obtained as follows:

$$v_{2m} = A_p \delta \omega / A_0 \quad (25)$$

Therefore F which is defined by Eq. (24) means the ratio of v_{2m} to w as follows:

$$F = v_{2m} / w \quad (26)$$

When the value of P_{2min} which is given by Eq. (23) reaches the air separation pressure P_s , the situation of the pump can be assumed as the critical condition of suction operation. Then the critical condition of suction operation is given by the following equation:

$$P_s = P_i - (\xi F^2 + \zeta_0 F + \eta) \rho w_{max}^2 / 2 \quad (27)$$

where w_{max} is critical value of w .

Equation (27) can be rewritten as follows:

$$w_{max} = k \sqrt{2 \Delta P_c / \rho} \quad (28)$$

or

$$\frac{\Delta P_c}{\rho w_{max}^2 / 2} = \frac{1}{k^2} \quad (29)$$

where

$$\Delta P_c = P_i - P_s \quad (30)$$

$$k = 1 / \sqrt{\xi F^2 + \zeta_0 F + \eta} \quad (31)$$

In the special case of $P_i = P_s$, Eq. (28) becomes as follows:

$$W_{max} = k \sqrt{2(P_s - P_s) / \rho} \quad (32)$$

In the case of self suction operation, the critical tangential velocity can not exceed the value given by Eq. (32) because the value of P_i is always less than P_s .

Figure 6 shows calculated examples of Eqs. (29) and (31). The values of ξ and ζ_0 are almost constant as shown in the latter part of this paper. So the value of $\Delta P_c / (\rho w_{max}^2 / 2)$ is only a function of F .

In order to increase the critical suction speed, it is necessary to minimize ρ and P_s and also necessary to maximize k and P_i . To maximize the value of k , it is most effective to enlarge the cross-sectional area of the cylinder port as seen from Eqs. (24) and (31).

In the case of geometrically similar axial piston pumps, the values of η and F are constant and the values of ξ and ζ_0 are almost independent of the displacement volume, so the value of k is almost constant. Therefore the value of W_{max} which is given by Eq. (32) is also constant. Then the critical suction speed has a tendency to reduce with an increase of the displacement volume.

3. Measurements of ξ and ζ

The values of ξ and ζ are investigated experimentally using a simulated model of a rotating cylinder block.

About the value of ξ , Eq. (13) can be used. But about the value of ζ , there is

Fig. 5 Calculated examples of P_2 , A/A_0 , $x/(2\delta)$ and θ

Fig. 6 Results calculated by Eqs. (29) and (31)

no reference except the experiments about a branched duct.

Additionally, it is also necessary to confirm the validity of Eq.(20).

The experiments in this work are performed under a steady flow condition.

3.1 Experimental apparatus and method

Figure 7 shows the experimental apparatus. A cylindrical choke is located at the radius of r and it corresponds to a cylinder port of a pump. The upstream pressure of the choke is measured by the pressure tap on the casing. The downstream pressure of the choke is transmitted to the center of the rotor and is taken out by the shaft.

Principal dimensions of the apparatus are shown in Table 1.

The pressure difference which is measured by the mercury manometer in Fig.7 is the sum of the pressure drop due to the axial flow velocity v_z , that due to the tangential velocity w and that due to the centrifugal force. From Eqs.(12), (14) and (17) the pressure difference Δp which is measured by the manometer can be written as follows:

$$\begin{aligned}\Delta p &= \Delta P_1 + \Delta P_2 + \Delta P_3' \\ &= \xi \frac{\rho v_z^2}{2} + \zeta \frac{\rho w^2}{2} + \eta' \frac{\rho w^2}{2} \dots\dots\dots(33)\end{aligned}$$

The value of η' in Eq.(33) is different from the value of η in Eq.(17) because the radial length subjected to a centrifugal force is different.

3.2 Experimental results

Figure 8 shows the experimental results of ξ and σ . The measurements in Fig.8 are performed when the rotor is stopped. The experimental results under variable oil temperatures coincide with the calculated result by Eq.(13) shown by a solid line in Fig.8. In actual axial piston pumps, the value of σ is more than 200 at the critical condition of suction operation. In this range, it can be supposed that the value of ξ is constant as $\xi = 1.75$.

Figure 9 shows the experimental results of ζ and v_z/w . As shown in Fig.9, the experimental results under variable Reynolds number can be represented by a single line. The characters which can be seen between ζ and v_z/w are similar to those of a branched duct. The solid line in Fig.9 is calculated by Eq.(15) with $\zeta_0 = 1.08$. In the range of small values of v_z/w , the experimental values of ζ are not represented by Eq.(15). However, at the critical condition of suction operation, the value of v_z/w becomes about 1.5 or 2.0 and the range of small values of v_z/w is not important. Therefore, Eq.(15) is available for the prediction of the critical condition of suction operation.

Figure 10 shows the relationship between $(\Delta P_1 + \Delta P_2) / (\rho w^2 / 2)$ and v_z/w . The experimental values of $\Delta P_1 + \Delta P_2$ are obtained by subtracting $\Delta P_3'$ in Eq.(33) from the measured pressure difference. As shown in Fig.10, the experimental results coincide

Table 1 Principal dimensions of experimental apparatus

Tested cylindrical choke		
Diameter d cm		0.85
Length l cm		1.20
Diameter of downstream D cm		1.50
Pitch radius r cm		1.60
Properties of used oil		
Oil temp. $^{\circ}\text{C}$	Density ρ kg s ² /cm ⁴	Kinematic viscosity ν cm ² /s
30	0.891×10^{-6}	0.887
40	0.882	0.527
50	0.874	0.338
60	0.865	0.231
Experimental conditions		
Reynolds number $Re (=v_z d / \nu)$		0 - 3600
Mean flow velocity v_z m/s		0 - 10
Tangential velocity w m/s		0.7 - 6.4
Rotational speed rpm		440 - 3800

1 Cylindrical choke	6 Drive shaft
2 Rotor	7 Thermometer
3 Casing	8 Back pressure valve
4 Pressure shaft	9 Tank and scale
5 Hg manometer	

Fig.7 Experimental apparatus

Fig.8 Comparison of experimental results and calculated results of ξ

Fig.9 Experimental results of ζ and v_2/w

with the solid line calculated by the following equation:

$$\frac{\Delta P_1 + \Delta P_2}{\rho w^2/2} = 1.75 \left(\frac{v_2}{w} \right)^2 + 1.08 \frac{v_2}{w} \quad \text{.....(34)}$$

Thus, the values of $\xi = 1.75$ and $\zeta_0 = 1.08$ can be used when one calculates the critical condition of suction operation from Eq.(31).

The value of loss coefficient of a branched duct is not so influenced by the shape of crosssection of the duct.⁽⁸⁾ And the value of ξ does not so change either. Therefore the values of ξ and ζ_0 that are obtained in this work will be applicable to other crosssectional shapes of cylinder ports.

4. Conclusions

In this paper, the pressure in the cylinder of an axial piston pump is analyzed and the critical condition of suction operation is obtained. And the values of loss coefficients are obtained experimentally using a simulated model of a rotating cylinder block. As the result of this investigation, the following informations are obtained.

Fig.10 Comparison of experimental results and calculated result by Eq.(34)

(1) The critical condition of suction operation is given by the following equation:

$$\frac{\Delta P_c}{\rho w_{\max}^2/2} = \xi F^2 + \zeta_0 F + \eta$$

(2) The values of loss coefficients are $\xi = 1.75$ and $\zeta_0 = 1.08$.

(3) In order to increase the critical suction speed, it is necessary to minimize the pressure loss between the reservoir and the valve plate, to minimize the radius of pitch circle of cylinder ports and to maximize the crosssectional area of cylinder port.

(4) The tangential velocity for geometrically similar axial piston pumps is constant at the critical condition of suction operation.

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