Force Balanced Axial Piston Motor Power Loss and Leakage Analysis

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Abstract. Aiming at the axial imbalance force when the traditional axial piston motor works, the force balanced axial piston motor utilized the concept of double stator to improve the structure of the traditional axial piston motor. Because of the special structure, the motor can not only achieve the axial force balance, but also output a variety of torque and speed. The working efficiency of the motor was an important indicator for evaluating the performance of the motor. Therefore, the power loss and leakage of the motor were analyzed in detail, and the prototype was tested. The experimental results showed that: (1) When the corresponding two single motors in the motor worked at the same time, theoretically, the leakage between the distribution cylinder and the cylinder accounts for the largest percentage of total leakage, reaching 56%. When the whole motor worked normally, the total leakage reaches 9.55L/min theoretically, and the leakage between the plunger and the plunger hole accounts for the largest percentage of the total leakage, which is 44% of the total leakage, and the distribution cylinder and cylinder the percentage of leakage between bodies has decreased, but its specific leakage has not changed. (2) The volumetric efficiency of the motor decreased as the pressure at the inlet of the motor increased, and the mechanical efficiency and overall efficiency increased accordingly. (3) When the motor inlet pressure was the same, the volumetric efficiency of two single motors at the same time was lower than the volumetric efficiency of the entire motor.

1. Introduction
The conventional axial piston motors can be divided into swash plate type and inclined axis type. Both of these motors have large axial unbalance forces, which greatly affect the service life of the motor [1]. In order to improve the performance of the motor and extend its life, a plurality of thrust bearings is generally assembled on the shaft to balance the axial imbalance force. The force-balanced axial piston motor proposed in this paper uses the "double stator" idea to achieve the balance of axial force by changing the structure of the motor [2]. In this paper, the main leakage forms and ways of the motor are analyzed, and the basic mathematical models of various leaks are calculated. The power loss is briefly explained. Through the design of the experimental system diagram, the prototype of the motor is tested and the data is carried out. The acquisition laid the foundation for the next step to improve the motor structure and performance.

2. Structure and working principle of force balanced axial piston motor

2.1 the structure of force balanced axial piston motor
The force balanced axial piston motor has made several corresponding designs based on the structure
of the traditional axial piston motor [3]: (1) The motor has two corresponding swash plates, which are called double stators, and the swash plates were respectively located on both sides of the motor through-going shaft, and were connected by the sliding bearing and the through-going shaft of the motor. (2) Since the radial dimension is relatively small, the through-going shaft and the cylinder are designed as one body, collectively referred to as a through-going shaft. (3) The inner surface of the distribution shell is provided with a waist-shaped distribution groove, and its main function is to distribute the flow into the oil passage of the cylinder. (4) The inlet and outlet ports of the motor are all disposed on the surface of the casing of the motor, and have two oil inlets and two oil outlets respectively, and the diameters of the two are substantially the same, and the motor will be distributed from the casing. Figure 1 is a structural schematic diagram of a force balanced axial piston motor.

2.2 Working principle of force balanced axial piston motor

When the high pressure oil enters the waist distribution groove of the distribution case through the two inlet ports, there are 10 pairs of plungers working at the same time, that is, two single motors work at the same time. At this time, the high-pressure oil in the waist-shaped oil distribution tank enters the plunger cavity through the oil-passing hole, so that the plunger moves axially with respect to the through-going shaft. The plunger acts on the surface of the swash plate through the sliding shoe, and the surface of the swash plate generates a corresponding reaction force to the sliding shoe, which generates a radial component force to the plunger to generate a torque with respect to the through-going shaft. When the high-pressure oil entered the waist-shaped distribution groove of the distribution shell through only one inlet port, only five pairs of plungers that were spaced apart generate torque to the through-going shaft, that is, only one single motor operated separately [4].

3. Power loss analysis of force balanced axial piston motor

3.1 Power loss caused by viscous friction between the swash plate and the shoe of the plunger head

The shoe mounted on the head of the plunger performs an elliptical motion on the surface of the swash plate. Figure 2 shows the loss of friction power of the swash plate of the shoe.
According to Figure 2, the viscous friction power $\Delta W_{i\mu}$ when a single shoe slides along the surface of the swash plate is:

$$\Delta W_{i\mu} = \frac{\mu_0 v_s^2 \pi}{h} (d_2^2 - d_1^2)$$

(1)

From the formula $v_s = \omega_s \cdot \rho$ can be obtained:

$$\Delta W_{i\mu} = \frac{\pi \mu_0 (d_2^2 - d_1^2)}{4h} \omega_s \rho^2$$

(2)

Where: $\mu_0$ is the dynamic viscosity coefficient of the hydraulic oil; $v_s$ is the sliding line speed of the shoe along the surface of the swash plate; $h$ is the gap between the bottom surface of the shoe and the surface of the swash plate; $\omega_s$ is the sliding angular velocity of the shoe around the point O in the figure; $\rho$ is the polar diameter of the running track of the shoe; $d_1, d_2$ is the inner and outer diameter of the sealing strip at the bottom of the shoe.

The above formula (2) is the viscous friction power loss between the single shoe and the swash plate. Since there are a plurality of plunger shoes in the motor, it is assumed that the number of plungers of the motor is $z$. When the half of the number of plungers on one side of the motor is an odd number, when there are $z_0 = \frac{z + 1}{2}$ plungers in the high pressure zone of the motor 1, the power loss $\Delta W_{i\mu1}$ caused by the viscous friction between the shoe and the swash plate is:

$$\Delta W_{i\mu1} = \frac{(z + 1) \pi \mu_0 (d_2^2 - d_1^2)}{8h} \omega_s \rho^2$$

(3)

The high pressure zone of the motor 2 on the same side has $z_0 = \frac{z - 1}{2}$ plungers, and the power loss $\Delta W_{i\mu2}$ caused by the viscous friction between the shoe and the swash plate is:

$$\Delta W_{i\mu2} = \frac{(z - 1) \pi \mu_0 (d_2^2 - d_1^2)}{8h} \omega_s \rho^2$$

(4)

Therefore, when the half of the number of plungers on the motor side is an odd number, the power loss $\Delta W_{i\mu}$ caused by the viscous friction between the shoe and the swash plate is:

$$\Delta W_{i\mu} = \Delta W_{i\mu1} + \Delta W_{i\mu2} = \frac{z \pi \mu_0 (d_2^2 - d_1^2)}{8h} \omega_s \rho^2$$

(5)

According to the formula:

$$\rho = R \sqrt{\sin^2 \varphi + \cos^2 \varphi \cos^2 \gamma}$$

(6)

$$\omega_s = \frac{\cos \gamma}{\cos \varphi + \cos \gamma \sin^2 \varphi} \cdot \omega$$

(7)

Where: $\gamma$ is the swashplate inclination angle; $\varphi$ is the rotation angle of the plunger; $\omega$ is the angular velocity when the cylinder rotates; $R$ is the plunger distribution circle radius.

Substituting the above formula (6) and formula (7) into the equation (5), the power loss caused by the viscous friction between the shoe and the swash plate on one side of the motor is:
3.2 Power loss between the plunger and the cylinder due to viscous friction

Assuming that the axial direction of the plunger is parallel to the axial direction of the cylinder, the following briefly analyzes the gap leakage between the plunger and the cylinder. Figure 3 shows the analysis of the friction power loss between the plunger and the cylinder.

\[
\Delta W_{fu} = \frac{z\pi\mu_0(d_2^2 - d_1^2) - \omega R^2\rho^2}{8h}
\]

\[
= \frac{z\pi\mu_0\omega R^2(d_2^2 - d_1^2)}{8h(\cos^2\varphi + \sin^2\varphi\cos^2\gamma)}
\]

4. Leakage analysis and prototype leakage calculation of force balanced axial piston motor

4.1 Motor leakage analysis

(1) Leakage loss between the shoe and the swash plate

The leakage between the surface of the shoe and the swash plate is analyzed. It is assumed that the pressure of the high-pressure hydraulic oil that passes into the cavity at the bottom of the plunger is \( p_1 \), and the pressure of the hydraulic oil after the leakage between the shoe and the swash plate is \( p_2 \), such as Figure 4 shows.
Figure 4. Leakage analysis diagram between the shoe and the swash plate surface

As can be seen from Figure 4, the gap between the surface of the shoe and the swash plate belongs to the parallel plate gap, and the leakage flow between the two parallel plates is:

$$\Delta q_1 = \frac{\pi h^3 \Delta p}{6 \mu_0 \ln(d_2/d_1)}$$  \hspace{1cm} (14)

Where: Pressure difference $\Delta p = p_1 - p_2$.

Due to the particularity of the motor structure, the two sides of the motor cylinder can be divided into two single motors, namely a motor 1 and a motor 2. Since the torque ripple of the odd number of motors is smaller than the number of plungers, it is determined that the number of the single motor plungers is an odd number, and the number of the plunger shoes of the motor 1 in the pressure oil zone is $\frac{z + 1}{2}$. The total amount of leakage of the swash plate surface of the motor 1 in contact with the bottom of the shoe is:

$$\Delta q_1 = \frac{z + 1}{2} \cdot \Delta q'_1 = \frac{(z + 1)\pi h^3 \Delta p}{12 \mu_0 \ln(d_2/d_1)}$$ \hspace{1cm} (15)

The number of plunger shoes in the pressure zone of the motor 2 is $\frac{z - 1}{2}$, and the total leakage of the surface of the swash plate in contact with the bottom of the shoe is:

$$\Delta q_2 = \frac{z - 1}{2} \cdot \Delta q'_2 = \frac{(z - 1)\pi h^3 \Delta p}{12 \mu_0 \ln(d_2/d_1)}$$ \hspace{1cm} (16)

Therefore, the total leakage at the contact between the sliding shoe on the motor side and the swash plate is:

$$\Delta Q = \Delta q_1 + \Delta q_2 = \frac{z\pi h^3 \Delta p}{12 \mu_0 \ln(d_2/d_1)} = \frac{z\pi h^3 (p_1 - p_2)}{12 \mu_0 \ln(d_2/d_1)}$$ \hspace{1cm} (17)

(2) Leakage loss between the plunger and the plunger hole in the motor

As shown in Figure 3, assuming that the plungers are unbiased in the cylinder plunger bores, their gap leakage amount is equal to the concentric cylindrical annular gap flow. Assuming that the hydraulic oil pressure after leakage through the gap between the plunger and the cylinder is $p_3$, the leakage amount $\Delta q_3'$ is:
Where: $\Delta p = p_i - p_4$; $l_i$ is the length of the $i$-th plunger in the high pressure zone and the plunger hole in the cylinder.

The adjacent plunger angle of the motor is $\beta$, so the amount of leakage between the plunger and the plunger bore should be a function of period $\beta$. The number of plungers in which the motor 1 is in the high pressure oil zone is $m_1 = \frac{z+1}{2}$, and the number of plungers in which the motor 2 is in the high pressure oil zone is $m_2 = \frac{z-1}{2}$. It can be seen that the number of plungers in the high pressure zone on one side of the motor is: $m = m_1 + m_2$, that is $m = \frac{z}{2}$. The total gap leakage between the plunger and the plunger hole on the motor side is:

$$\Delta Q_1 = \sum \frac{\pi d^2 \delta^3 (p_1 - p_4)}{12 \mu l}$$

Since the length $l_i$ between the plunger and the plunger bore is a function of the cylinder angle $\varphi$, the total leakage is:

$$\Delta Q_2 = \frac{\pi d^2 \delta^3 (p_1 - p_4)}{12 \mu_0} \sum_{i=1}^{m} \frac{1}{L - R(1 + \cos \varphi) \tan \gamma}$$

Where: $\varphi_i$ is the corner of the $i$-th plunger.

As can be seen from equation (19), the amount of leakage between the plunger and the plunger bore is a function of the cylinder angle $\varphi$, which varies with the cylinder angle $\varphi$.

(3) Leakage loss between the distribution tube and the outer surface of the cylinder

When the high-pressure oil is introduced into the distribution casing, as shown in Figure 5, the high-pressure oil pressures of the two oil-passing zones of the distribution casing are all $p_1$, and no pressure difference is generated between them. The end pressure $p_4$ at the end of the distribution shell and the cylinder must be much smaller than the $p_1$ in the high pressure zone, thus creating a pressure differential $\Delta p$. Due to the influence of the pressure difference, certain leakage will inevitably occur between them. In order to facilitate the calculation, the leakage between the two is regarded as the gap leakage between the parallel plates [5, 6].

![Figure 5. Leakage analysis diagram between the distribution shell and the outer surface of the cylinder](image)

According to the gap leakage theory between parallel plates, the leakage amount $\Delta Q_3$ of the inner surface of the distribution shell and the outer surface of the cylinder can be obtained as:

$$\Delta Q_3 = \frac{\pi d^3 \delta^3 (p_1 - p_4)}{6 \mu_0 \ln(d_4 / d_3)}$$
Where: $\delta_2$ is the small gap between the outer diameter of the cylinder and the inner diameter of the distribution shell; $d_3$ and $d_4$ are the inner and outer diameters of the distribution port of the distribution housing.

Since there are plungers on both sides of the cylinder of the motor, when calculating the sum of the leakage between the shoe and the swash plate surface and the leakage of the plunger and the cylinder hole, the other side of the motor needs to be considered. Inside, the total leakage $\Delta Q$ of the motor is:

$$\Delta Q = 2\Delta Q_1 + 2\Delta Q_2 + \Delta Q_3$$

$$= \frac{z\pi h^3 (p_1 - p_2)}{6\mu_0 \ln(d_2 / d_1)} + \frac{\pi d_3^2 \delta_3^3 (p_1 - p_3)}{12\mu_0}.$$  \hspace{1cm} (22)

$$+ \sum_{i=1}^{n} \frac{1}{L - R(1 + \cos \varphi) \tan \gamma} + \frac{\pi \delta_2^3 (p_1 - p_2)}{6\mu_0 \ln(d_4 / d_3)}$$

4.2 Leakage calculation of motor test prototype

The main design parameters of the motor are: Dynamic viscosity coefficient of hydraulic oil $\mu_0$ is $3 \times 10^{-2} \text{Pa}\cdot\text{s}$; Inner diameter of the sealing strip at the bottom of the shoe $d_1$ is 20mm; Outer diameter of the sealing strip at the bottom of the shoe $d_2$ is 28mm; Number of motor plungers $z$ is 10; Pressure difference $\Delta P$ is 8MPa; Geometric length of the plunger $L$ is 85mm; Plunger diameter $d$ is 20mm; Inner diameter of the distribution port of the distribution housing $d_3$ is 40mm; Inner and outer diameters of the distribution port of the distribution housing $d_4$ is 40mm; Inner and outer diameters of the distribution port of the distribution housing $d_5$ is 70mm; Swashplate inclination $\gamma$ is 16°; Small gap between the outer diameter of the cylinder and the inner diameter of the distribution tube $\delta_2$ is 0.052mm; Radial clearance between the plunger and the plunger bore in the cylinder $\delta_1$ is 0.02mm; Plunger distribution circle radius $R$ is 48mm.

Since the motor is constantly rotating, the leakage created by the gap between the plunger and the plunger bore is a function of the cylinder angle $\varphi$. This leak will vary depending on the corner of the cylinder. According to the formula of the leak, it can be inferred that when the cylinder angle $\varphi=0°$, the corresponding plunger leakage amount is the largest. Here, we only count the case of the largest amount of leakage. Due to the structural specificity of the motor, there is only leakage when the left and right motor sides are operated and leakage when the left and right motors are simultaneously operated. Table 1 shows the theoretical leakage amount when the left and right motors are operated by only one side of the motor, and Table 2 shows the theoretical leakage amount when the left and right motors are simultaneously operated.

| Table 1. Theoretical leakage of the left and right motors when only one side of the motor is working |
|---------------------------------|-----------------|-----------|
| Leak form | Leakage (L/min) | Percentage |
| Leak between the shoe and the swash plate | 0.80 | 12% |
| Leak between the plunger and the plunger hole | 2.12 | 32% |
| Leakage between the distribution shell and the cylinder | 3.72 | 56% |
| Total leakage | 6.64 | 100% |
Table 2. Theoretical leakage when the left and right motors are working at the same time

| Leak form                                      | Leakage (L/min) | percentage |
|-----------------------------------------------|-----------------|------------|
| Leak between the shoe and the swash plate     | 1.63            | 17%        |
| Leak between the plunger and the plunger hole | 4.20            | 44%        |
| Leakage between the distribution shell and the cylinder | 3.72            | 39%        |
| Total leakage                                 | 9.55            | 100%       |

The theoretical flow rate of the motor can be calculated from the theoretical formula \( Q_{tm} = Vn \) of the theoretical flow of the motor. \( n \) is the rated speed of the motor. The rated speed of the motor is \( n = 800 \text{ r/min} \). The total displacement \( V = 172 \text{ mL/r} \) of the motor is the theoretical flow \( Q_{tm} \) of the motor: \( Q_{tm} = Vn = 1.376 \times 10^4 \text{ mL/min} \). According to the definition of motor volumetric efficiency to calculating the volumetric efficiency \( \eta_{vw} \) of the motor in combination with the amount of motor leakage sought:

\[
\eta_{vw} = \frac{Q_{tm} + \Delta Q}{Q_{tm}} \times 100\% = \frac{1.376 \times 10^4 + 9.55 \times 10^3}{1.376 \times 10^4} \times 100\% = 93.5\%
\]

5. Experiment

In order to verify the feasibility of the working principle of the force balance axial piston motor and the rationality of the structure, the experimental test platform of the plunger motor was built, and the performance parameter data was collected and analyzed. Figure 6 is a physical view of a part of the motor, and Figure 7 is a system diagram of the test bench of the motor.

![Figure 6: Physical diagram of some parts of the motor](image)

![Figure 7: Piston motor test system diagram](image)

In the experimental schematic diagram 7, the motor under test is directly connected to the load, which refers to the load pump, which can load the motor under test. The variable pump 3 of the motor supplies high pressure oil. The relief valve 16 regulates the inlet flow of the motor. By adjusting the
working pressure of the load pump in the system, the oil inlet of the motor can get different pressure inputs. The inlet and outlet pressures of the motor are measured by pressure gauges 8, 9, 11, and 12, respectively. The rotational speed and torque of the motor at different pressures can be collected by the torque speed testers 13, 14. Table 3 shows the main design parameters of the test prototype.

| Experimental maximum speed | 800r/min |
|----------------------------|----------|
| Experimental maximum pressure | 8MPa |
| Displacement when two single motors are working | 86mL/r |
| Displacement of the entire motor during operation | 172mL/r |

Due to the small size of the test bed fuel tank, the motor running speed is lower during the actual experiment to verify the theoretical accuracy. The performance parameters of the motor in the above two connection modes are recorded through experiments, and the volumetric efficiency, mechanical efficiency and total efficiency of the motor during actual operation are calculated according to the performance parameters. The specific data are shown in Tables 4 and 5 below.

| Import pressure(MPa) | Input traffic(L/min) | Rotating speed(r/min) | Torque (N·m) | Volumetric efficiency ηv(%) | Mechanical efficiency ηm(%) | Total efficiency η(%) |
|----------------------|----------------------|-----------------------|--------------|----------------------------|----------------------------|----------------------|
| 1                    | 10.0                 | 117                   | 7.1          | 93.6                       | 52.9                       | 49.5                 |
| 2                    | 9.7                  | 112                   | 9.8          | 90.2                       | 64.7                       | 58.4                 |
| 3                    | 9.4                  | 109                   | 16.3         | 88.7                       | 77.0                       | 68.3                 |
| 4                    | 9.1                  | 106                   | 24.7         | 87.4                       | 80.6                       | 70.4                 |
| 5                    | 8.8                  | 102                   | 39.5         | 85.2                       | 83.2                       | 70.9                 |
| 6                    | 8.5                  | 98                    | 59.3         | 83.8                       | 84.1                       | 70.7                 |

| Import pressure(MPa) | Input traffic(L/min) | Rotating speed(r/min) | Torque (N·m) | Volumetric efficiency ηv(%) | Mechanical efficiency ηm(%) | Total efficiency η(%) |
|----------------------|----------------------|-----------------------|--------------|----------------------------|----------------------------|----------------------|
| 1                    | 10.0                 | 61                    | 13.6         | 92.6                       | 53.6                       | 49.6                 |
| 2                    | 9.7                  | 58                    | 21.1         | 89.1                       | 64.6                       | 57.8                 |
| 3                    | 9.4                  | 56                    | 33.5         | 86.9                       | 73.0                       | 63.4                 |
| 4                    | 9.1                  | 54                    | 50.7         | 85.4                       | 74.9                       | 64.0                 |
| 5                    | 8.8                  | 52                    | 76.5         | 83.7                       | 80.6                       | 67.5                 |
| 6                    | 8.5                  | 98                    | 59.3         | 83.8                       | 84.1                       | 70.7                 |

It can be obtained from Tables 4 and 5. Under the operation of two different connection modes, the volumetric efficiency of the motor decreases as the inlet pressure of the motor increases, and the mechanical efficiency and total efficiency gradually increase. The reason is mainly because when the oil pressure is low, the relative sliding surface between the friction pairs fails to form a pressure oil film. At this time, the friction pair is in a dry friction or a semi-dry friction state, and the friction loss is more serious. When the oil pressure rises, a pressure oil film is formed between the friction pairs, and the pressure oil film greatly reduces the friction between the respective moving parts, thereby improving the mechanical efficiency of the motor. Figure 8 below shows the efficiency curve for the two connection modes of the motor.
The working mode 1 indicates the efficiency curve when the left and right motors operate only on one side of the motor. Working mode 2 indicates the efficiency curve when the left and right motors are working at the same time. As can be seen from the above figure, when the motor inlet pressure is the same, the volumetric efficiency of only one side of the motor is lower than that of the left and right side motors. The main reason is that the displacement of the left and right motors when working together is twice that of the former, that is, the degree of displacement is increased to a greater extent than the amount of leakage. Therefore, in comparison, the volume efficiency of the motor on one side is low. As the inlet pressure of the motor increases, the volumetric efficiency of the two working modes will decrease correspondingly, the mechanical efficiency will increase accordingly, and the total efficiency will show a highest point in the process of gradual change.

6. Conclusion
(1) When only one side of the motor is operating, the leakage between the distribution tube and the cylinder is the main leakage in the motor. When the left and right motors are working at the same time, the leakage between the plunger and the plunger hole accounts for the largest percentage of the total leakage.
(2) The volumetric efficiency of the motor decreases as the inlet pressure of the motor increases, and the mechanical efficiency and overall efficiency gradually increase.
(3) When the motor inlet pressure is the same, the volumetric efficiency when only one side of the motor is operated is lower than that when the left and right side motors are operated together. The research results in this paper have a strong guiding significance and reference value for the design and use of the force balance axial piston motor.

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