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Analysis of the Influencing Factors on the Oil Film Uniformity of Hydro-Viscous Drive Clutch

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Hydro-viscous drive (HVD) clutch is a type of power transmission device by using shear stress of oil film. Whether the oil film between friction pair are uniformly distributed is the key factor that affecting the performance of HVD clutch. However, it is hard to make sure that the uniformity of oil film between friction pair are the same, which can lead to uneven wear problem for the frictional plates of HVD clutch. In order to study the uniformity of oil film of HVD clutch, the distribution regularity of oil films between friction pair of HVD clutch is researched by establishing the mathematics model and a new HVD clutch with double-piston structure is proposed, which can greatly improve the uniformity of oil film of HVD clutch.

Keywords: Hydro-viscous drive clutch, Distribution of oil film, Uniformity of oil film thickness, Double-piston structure

1 Introduction

The Hydro-viscous drive is a speed regulation device suitable for high-power equipment in fields such as water plants, power plants, construction machinery, special vehicles, and petrochemical industries.It is based on Newton's law of internal friction, relying on liquid viscosity and oil film shearing to transmit power. By adjusting the displacement of the oil cylinder piston and varying the gap between the friction plates, the torque is modified, thereby altering the output speed[1-3].The uniformity of the oil film thickness between the friction pairs in a Hydro-viscous drive is a critical factor affecting its performance. Under ideal conditions, a uniform oil film thickness between the friction pairs not only transmits torque but also dissipates heat. However, in practical experiments, overheating and deformation of the friction plates in Hydro-viscous drives are frequently observed. Researchers have conducted in-depth studies on this issue, with a primary focus on the influence of temperature on the performance of liquid viscous transmission [4-6].In literature, a simulation study based on the CFX model was conducted to investigate the influence of radial grooves on the deformation, stress distribution, and torque transmission capability of the friction pairs in a Hydro-viscous drive. Through the disassembly of a failed clutch, it was found that the friction plates near the piston side experienced severe wear, while those farther from the piston showed almost no wear. This severe uneven wear of the friction plates ultimately led to clutch failure. Hence, it is evident that research on the Hydro-viscous drive from the perspective of its internal structure and the stress model of the oil film between friction plates is of great significance.Research has shown that improving the lubrication channel design of the friction pairs in a Hydro-viscous drive can enhance the separation of the friction pairs, leading to more uniform separation [7-9]. Literature [10] conducted a relevant study on the dynamic characteristics of a Hydro-viscous drive with a double-piston structure, concluding that the response speed of the double-piston Hydro-viscous drive increased by more than twofold[11].

This paper investigates the uniformity of oil film thickness between friction plates based on the force model of the friction plates[12]. It reveals the distribution pattern of oil film thickness between the friction pairs in a Hydro-viscous drive and further compares the oil film uniformity between two different Hydro-viscous drive structures[13; 14]. The results indicate that the double-piston structure significantly improves the uniformity of oil film thickness between the friction pairs, enhancing the overall performance of the Hydro-viscous drive.

2 Working principle of hydro-viscous drive clutch

The hydro viscous drive clutch was the first product developed by Philadelphia Gear Company in the 1970s using hydro viscous transmission technology.

It can adjust speed steplessly, and the larger the speed regulation range, the more significant the energy-saving effect. It can flexible drive and protect the transmission system. It can be driven synchronously, and the ideal transmission efficiency can reach 100 %[15]. The main structure of the traditional hydro-viscous drive clutch is shown in Fig 1.

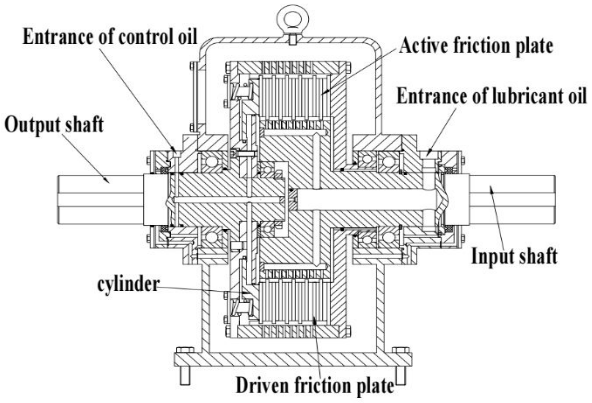

Fig. 1 *The structure of HVD clutch*

Its working principle is: the lubricating oil enters the gap between the active and driven friction plates through the lubricating oil inlet on the input shaft through cover to form an oil film. On the one hand, this oil film provides shear force for the hydro-viscous drive clutch and thus transmits torque, and on the other hand, it can carry away the heat generated by oil film shearing. The control oil enters the piston cylinder through the control oil inlet on the output shaft cover. By adjusting and controlling the oil pressure, the piston displacement can be controlled, so as to adjust the clearance of the friction pair and then adjust the torque. Due to friction among the spline teeth of the active and driven friction plates, along with the external teeth of the input shaft and the internal teeth of the passive rotation device, under the traditional single-piston structure, the pressing force decreases from the side close to the piston to the side away from the piston, so the uniformity of the oil film between the friction pairs is uneven[16].

3 Oil film bearing capacity analysis

The bearing capacity of the oil film is mainly composed of the following parts.

$$F_d = F_1 \text{-} F_2 \text{-} F_3 + F_4 \,, \tag{1}$$

In the equation F_d is the bearing capacity of the oil film. The first item F_1 on the right of the equals sign is the static pressure bearing capacity generated when the lubricating oil flows through the gap between the two still and parallel parallel friction plates. The second item F_2 is the centrifugal bearing force generated by the pressure drop caused by centrifugal force when the lubricating oil passes through the gap between the rotating friction plates, the symbol is a negative number. The third item F_3 is the squeezing force generated when the friction plates move against each other, the symbol is a negative number. The fourth item $F4$ is the dynamic pressure bearing force generated by grooves the surface of the friction plates[17].

3.1 Hydro-static bearing capacity

When lubricating oil flows through the gap between two stationary parallel friction plates, a gap flow is formed [5]. Its basic equation is:

$$q = \frac{b\delta^3 \Delta p}{12\mu l} \,, \tag{2}$$

Where:

q…The flow rate through the parallel plate,

b…The width of the gap between the pair of parallel plate [m];

σ…The thickness of the oil film between the plates [m];

Δp…The pressure difference between the gap flow within the length of l [Pa];

μ…The dynamic viscosity of the oil [Pa.s];

l…The length of the plate [m].

The final derivation is obtained by equation:

$$F_1 = \frac{\pi P}{2\ln(r_2 / r_1)} \left[r_2^{\,2} - r_1^{\,2} - 2r_1^{\,2} \ln(r_2 / r_1) \right], \tag{3}$$

Where:

F_1…The hydro-static bearing capacity [N],

P…The lubricating oil pressure [Pa],

r_2…The radius outside the friction plates [m],

r_1…The radius inside the friction plates [m].

3.2 Centrifugal bearing capacity

The centrifugal bearing capacity generated by the pressure drop caused by centrifugal force when the lubricating oil flows through the gap between rotating friction plates can be derived by equation[12]:

$$F_2 = \frac{\rho\pi}{6}\left(r_2^{\,4} - r_1^{\,4}\right)\left(\omega_1^{\,2} - \omega_1\Delta\omega + \frac{3}{10}\Delta\omega^2\right), \tag{4}$$

Where:

F_2…The centrifugal bearing capacity [N],

ϱ…The density of lubricating oil [kg/m^3],

r_2…The radius outside the friction plates [m],

r_1…The radius inside the friction plates [m],

ω_1…The angle of the active friction plates [rad/s],

$\Delta\omega$…The main driven friction plates angular velocity difference [rad/s].

3.3 Extrusion force

When the friction plates move axially with each other, the uniformity of the thickness of oil film changes, resulting in an extrusion force[18] , which is derived from the equation:

$$F_3 = \frac{3\pi\mu\left[r_2^{\,4} - r_1^{\,4} - \frac{\left(r_2^{\,2} - r_1^{\,2}\right)^2}{\ln(\mathrm{r}_2/\mathrm{r}_1)}\right]}{\delta^3}\frac{\mathrm{d}\delta}{\mathrm{d}t}, \tag{5}$$

Where:
F_3…The extrusion force [N],
μ …The dynamic viscosity of the lubricating fluid [Pa.s],
r_2…The radius outside the friction plates [m],
r_1…The radius inside the friction plates [m],
σ …The thickness of the oil film [m].

3.4 Dynamic pressure bearing capacity

The actual friction plates surface is often grooved, resulting in dynamic pressure bearing capacity, and the expression of dynamic pressure bearing capacity is more complicated. For the sake of analysis, the friction plate can be regarded as a flat plate with length $\pi(r_1 + r_2)$ and width $r_2 - r_1$, the dynamic pressure bearing capacity can be simplified to:

$$F_4 = \frac{\left[m\mu l_1 l_2\left(l_1 + l_2\right)(r_2 - r_1)(r_1 + r_2)h_a\right]\Delta\omega}{2l_1\delta^3 + 2l_2(\delta + h_a)^3}, \tag{6}$$

Where:
F_4…The dynamic pressure bearing capacity [N],
$\Delta\omega$…The main driven friction plates angular velocity difference [rad/s],
σ…The thickness of the oil film [m],
μ…The dynamic viscosity of the lubricating oil fluid [Pa.s],
z…The number of grooves,
l_1…The groove width [m],
l_2…The step width [m],
h_a…The groove depth [m].
For ease of analysis, $l_1 = l_2 = l$ can be further assumed:

$$\begin{aligned} l_1 + l_2 &= \pi(r_1 + r_2)/z \\ l &= \pi(r_1 + r_2)/2z \\ K &= [z\mu l^2(r_2 - r_1)(r_1 + r_2)h_a]\Delta\omega \\ &= \frac{[\mu\pi^2(r_2 - r_1)(r_1 + r_2)^3 h_a]\Delta\omega}{4z} \end{aligned}, \tag{7}$$

It can be obtained:

$$F_4 = \frac{K}{\delta^3 + (\delta + h_a)^3}, \tag{8}$$

4 Force analysis of friction plates with single-piston structure

As shown in Fig. 2 each friction plates is balanced by the combined action of oil film bearing force and static friction, and the oil film bearing capacity has been analyzed earlier, and the friction plates spline teeth are also subjected to static friction[7].

$$F_m = \frac{2Tf}{nd}, \tag{9}$$

Where:
F_m …The friction generated at the spline teeth of the friction plates [N],
T…The torque transmitted by the hydro-viscous drive clutch [N.m],
f…The coefficient of friction of the spline,
n…The number of oil films of the hydro-viscous drive clutch,
d…The diameter of the index circle at the spline tooth [m].

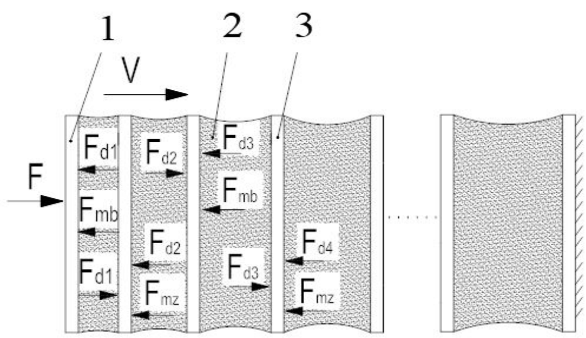

Fig. 2 *The mechanical model of HVD clutch*

Where:
1…Driven friction plate,
2…Oil film,
3…Active friction plate.
Easy to draw:

$$\begin{cases} F_{d1} = F_{d2} + F_{mz}, F_{d2} = F_{d3} + F_{mb} \\ F_{d3} = F_{d4} + F_{mz}, F_{d4} = F_{d5} + F_{mb} \\ \vdots \\ F_{dn-2} = F_{dn-1} + F_{mz}, F_{dn-1} = F_{dn} + F_{mb} \end{cases}, \tag{10}$$

Where:
F_{dn}…The bearing capacity of the oil film [N],
F_{mz}…The friction generated at the spline teeth of the active friction plates [N],
F_{mb}…The friction generated at the spline teeth of the driven friction plates [N].

When analyzing the force of the friction plates, it is assumed that the hydro-viscous drive clutch is in a stable speed regulation state[19], that is, the relative angular velocity difference between the friction plates is equal, and the oil film centrifugal bearing capacity F_2 of each friction plates is also equal. At the same time, the uniformity of the thickness of oil film between the active and driven friction plates reaches balance, and there is no relative extrusion movement, the oil film extrusion force F_3 is zero; And the lubricating oil pressure between the friction plates is equal, so it can be seen from equation (3) that the hydro-static bearing capacity F_l of the oil film subjected to each friction plates is also equal.

Substituting equations (3), (4), (5) and (8) into equation (1) and then bringing in equation (10), the uniformity distribution rule of each friction plates thickness of the hydro-viscous drive clutch can finally be derived:

$$\frac{K}{\delta_1^3+(\delta_1+h_a)^3}=\frac{K}{\delta_n^3+(\delta_n+h_a)^3}+\frac{n-1}{2}\left(F_{mz}+F_{mb}\right), \tag{11}$$

It can be seen from equation (11) that under the single-piston unidirectional displacement structure, the first oil film thickness value between each friction pair is the smallest, and the subsequent oil film thickness value increases sequentially. The sequence formed after the reciprocal of the oil film thickness cube between the friction pairs is approximately a arithmetic progression.

Since the oil film transmission torque of the hydro-viscous drive clutch is inversely proportional to the thickness of the oil film[20], further analysis shows that the thickness of the oil film between the friction pairs close to the piston side is very small, the friction plates are even in direct contact, and the torque transmitted is very large, so the wear of the friction pair is serious; the oil film between the friction pairs far from the piston side has a large thickness, and even does not participate in the transmission of torque, so there is almost no wear. The deflection wear of the friction plates will aggravate the failure of the hydro-viscous drive clutch.

5 Uniformity analysis of oil film thickness in single-piston structures

In order to reasonably illustrate the uniformity of the thickness of oil film distribution between the friction pairs of the hydro-viscous drive clutch, the relevant parameters of the hydro-viscous drive clutch (including the size of friction plates[20], the number of friction plates, number of grooves, the depth of grooves, dynamic viscosity of oli and speed control conditions, etc.) in actual work are analyzed, and the corresponding parameters are shown in the following table.

Tab. 1 *The parameters of HVD clutch*

Parameter	Parameter values	Unit
Rated rotational speed of the HVD	750	r/min
Rated torque of the HVD	6000	N.m
The number of friction plates	19	
The outer diameter of the friction plate	300	mm
The inner diameter of the friction plate	400	mm
Modulus of spline teeth	4	mm
Number of spline teeth of the passive friction plate	98	
Number of spline teeth of the active friction plate	78	
Coefficients of friction of the spline teeth	0.07	
The depth of the groove	0.4	mm
The number of grooves	16	
Dynamic viscosity of the # 6 hydraulic oil	0.02	Pa.s
Speed ratio	0.7	
Power of the load water pump	234	KW

The data in the table can be obtained by bringing them into equations (7) and (9) respectively:

- $K = 6.2 \times 10^{-8}$
- $F_{mz} = 72.9$
- $F_{mb} = 58.3$

Whether a set of data is uniform or analyzes its degree of dispersion can be described by mathematical variance, so we can introduce mathematical expectations and the definition of variance to analyze the uniformity of the thickness of oil film between the friction pairs of the hydro-viscous drive clutch[21].

$$\begin{cases} E(\delta)=\dfrac{\sum_{i=1}^{n}\delta_i}{n} \\ D(\delta)=\dfrac{\sum_{i=1}^{n}\left(\delta_i-E(\delta)\right)^2}{n} \end{cases}, \tag{12}$$

For ease of analysis, convert the left oil film thickness unit in equation (11) to mm, then K=62. For the array shown in equation (11), in the case of h_a=0.4, F_{mz}=72.9, F_{mb}=58.3, then take K=65.5, 69.5, 73.5, respectively, and calculate the thickness of oil film σ and thickness variance D with the change curves of n are shown in Fig. 3 and Fig. 4, respectively.

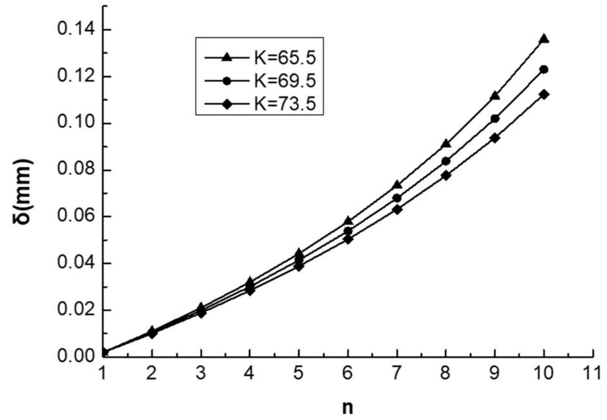

Fig. 3 *σ-n diagram with different K*

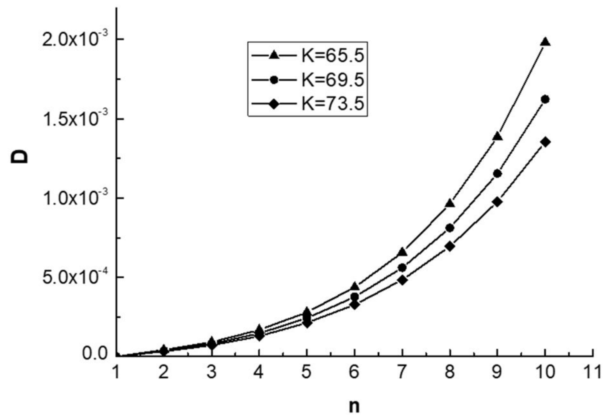

Fig. 4 *D-n diagram with different K*

In the case of K=65.5, F_{mz}=72.9 , F_{mb}=58.3, then take h_a=0.38, 0.39, 0.4, respectively, and calculate the thickness of oil film σ and thickness variance D with the change curves of n are shown in Fig. 5 and Fig. 6, respectively.

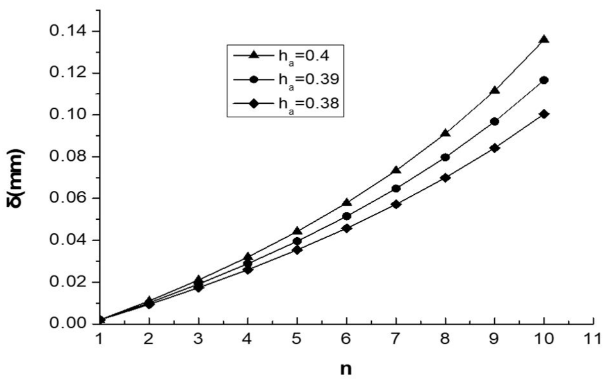

Fig. 5 *σ-n diagram with different h_a*

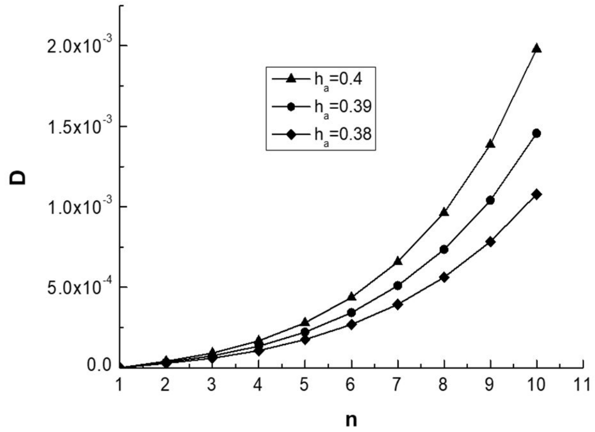

Fig. 6 *D-n diagram with different h_a*

In the case of h_a=0.4, K=62, take them separately:

- F_{mz}=41.4，F_{mb}=33，
- F_{mz}=55，F_{mb}=44.2，
- F_{mz}=72.9，F_{mb}=58.3.

The thickness of oil film σ and thickness variance D with the change curves of n are calculated as shown in Fig. 7 and Fig. 8, respectively.

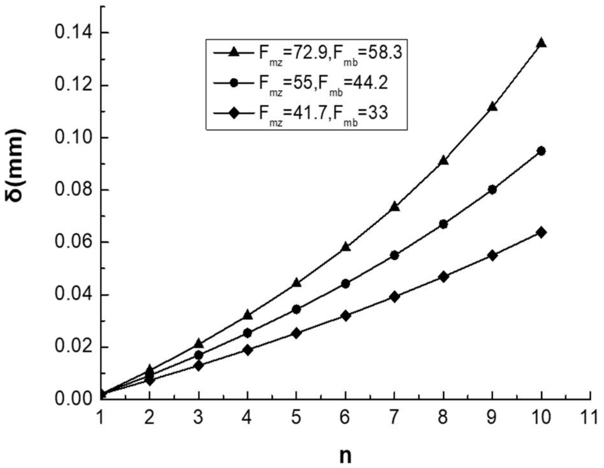

Fig. 7 *σ-n diagram with different F_m*

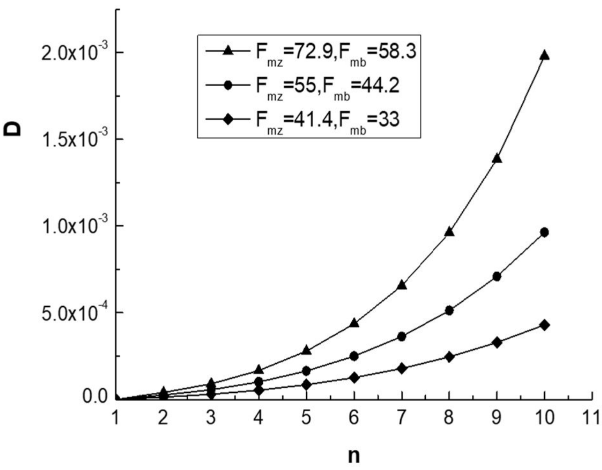

Fig. 8 *D-n diagram with different F_m*

Obviously, from equation (11), it can be concluded that the uniformity of the thickness of oil film between the friction pairs of the hydro-viscous drive clutch is mainly related to the coefficient K and the number of oil films n, the friction force F_m at the spline teeth and other factors. It can be seen from Fig. 3, Fig. 5 and Fig. 7 that with the increase of the number of oil films n, the thickness of oil film value σ increase significantly.

From Fig. 3 and Fig. 4, it can be concluded that the larger the coefficient K, the smaller the variance value of the thickness of oil film, so the better the uniformity of the thickness of oil film. On the contrary, the variance of oil film thickness is bigger.

From Fig. 5 and Fig. 6, it can be concluded that the smaller the trench depth h_a, the smaller the variance value of the thickness of oil film, so the better the uniformity of the thickness of oil film. On the contrary, the variance of oil film thickness is bigger.

From Fig. 7 and Fig. 8, it can be concluded that the smaller the friction Fm at the spline teeth, the smaller the variance value of the thickness of oil film, so the better the uniformity of the thickness of oil film. On the contrary, the variance of oil film thickness is bigger.

The comparison Fig. 4, Fig. 6 and Fig. 8 also shows that relative to the coefficient K and groove depth h_a, the friction force F_m at the spline teeth has a greater influence on the thickness of oil film variance value D, and the number of oil film n has the greatest influence on the thickness of oil film variance value D, and the thickness of oil film variance value increases significantly with the increase of the number of oil films.

When the hydro-viscous drive clutch is in a stable speed regulation condition, if the transmission is relatively small, due to the small torque transmitted, the friction at the spline teeth is small, and due to the large speed difference and the coefficient K, the degree of non-uniformity of the thickness of oil film is better, and the non-uniformity of the thickness of oil film has little influence on the deflection wear of the friction plates; When the transmission ratio increases, the transmitted torque increases, so the friction at the spline teeth increases, and due to the small speed difference, the coefficient K is small, and the degree of non-uniformity of the thickness of oil film increases, and the non-uniformity of the thickness of oil film has a greater impact on the deflection wear of the friction plates.

6 Uniformity analysis of oil film thickness of double-piston structure

From the above analysis, it can be concluded that in order to improve the uniformity of the thickness of the oil film, you can start from several aspects, such as increasing the coefficient K, reducing the number of oil film n, and reducing the friction F_{mz}, F_{mb} at the spline teeth, etc.

From equation (7), it can be seen that the coefficient K is related to the μ of lubricating oil dynamic viscosity, the h_a of depth of grooves on the surface of the friction plates, and the number of grooves m. Since the increase in the dynamic viscosity of the lubricating oil will increase the resistance of the lubrication and control system, increase the power consumption of the oil pump and the heating of the oil, it is not suitable to use hydraulic oil with a large viscosity. The groove on the surface of the friction plates is mainly for cooling effect, but the depth and width of the groove are not suitable to be too large, and the number is not suitable to be too much, otherwise it will greatly reduce the effective working area of the friction plates and reduce the torque transmission capacity of the friction plates. Therefore, it is not suitable to improve the uniformity of the thickness of oil film of the hydro-viscous drive clutch by increasing the coefficient K.

Due to the limitations of the design of the hydro-viscous drive clutch[22], the number n of friction pairs and the friction force F_{mz} and F_{mb} at the spline teeth are not easy to change. Because when designing a hydro-viscous drive clutch, the input raw data is the rated torque transmitted under synchronous conditions. The first thing that can be determined is the number and size of the friction pair, because the number, material, and size of the friction plates are determined, and the friction at the spline is basically determined by equation (9).

From the above analysis, it can be seen that to improve the uniformity of the thickness of oil film of the hydro-viscous drive clutch, it is necessary to find another way. Under the premise of not reducing the actual number of friction pairs, a double-piston symmetrical compression structure is proposed[23, 24], which indirectly halves the number of friction pairs through the double-way symmetrical displacement of the friction plates, thereby greatly improving the uniformity of the thickness of oil film between the friction pairs of the hydro-viscous drive clutch. The structure is shown in Fig. 9.

Compared with the single-piston structure, the double-piston structure not only adds a set of piston cylinder mechanism on the right, but also changes the control oil circuit, and the pressure oil of the output shaft control oil passage is cleverly introduced into the right piston cylinder by adding a control oil pipe in the axial lubricating oil passage of the input shaft. Thus, under the action of the same controlled oil pressure, the friction plates changes from single-way displacement to symmetrical double-way displacement.

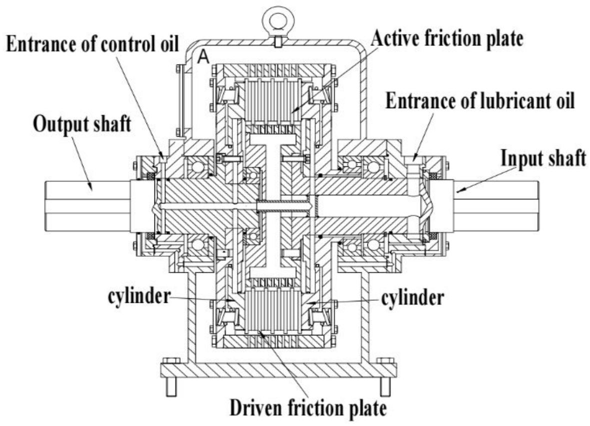

Fig. 9 *HVD clutch with two piston*

As shown in Fig. 10, the bearing capacity of the oil film and the force condition of the friction plates under the double-piston pressing structure are consistent with the single-piston pressing structure, but the displacement of the friction plates changes from unidirectional to bidirectional. Assuming that the double-piston structure is completely symmetrical, the middle friction plates does not move, and the displacement of the left and right friction plates is completely symmetrical, and only the thickness of oil film distribution rule between half of the friction plates can be analyzed. Because in the symmetry case, the expectation and variance of half of the thickness of oil film is equal to the expectation and variance of the overall oil film thickness.

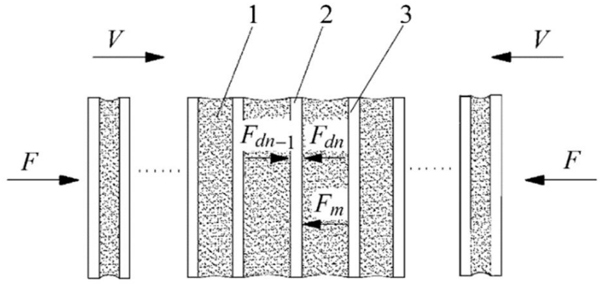

Fig. 10 *Bearing capacity*

Where:
1…Driven friction plate,
2…Oil film,
3…Active friction plate.
The mathematical proof process is as follows.

$$n_1 = \frac{n}{2}; \quad E_1(\delta) = \frac{2\sum_{i=1}^{\frac{n}{2}} \delta_i}{\frac{n}{2}} = \frac{2\sum_{i=1}^{\frac{n}{2}} \delta_i}{n} \tag{13}$$

$$D_1(\delta) = E_1(\delta^2) - E_1^2(\delta) = \frac{2\sum_{i=1}^{\frac{n}{2}} \delta^2{}_i}{n} - E_1^2(\delta) \tag{14}$$

$$n_2 = 2n_1 = n \tag{15}$$

$$E_2(\delta) = \frac{(\delta_1 + \delta_2 + \ldots\ldots + \delta_{n/2}) + (\delta_1 + \delta_2 + \ldots\ldots + \delta_{n/2})}{n} = \frac{2\sum_{i=1}^{\frac{n}{2}} \delta_i}{n} = E_1(\delta) \tag{16}$$

$$D_2(\delta) = E_1(\delta^2) - E_1^2(\delta) = \frac{\sum_{i=1}^{\frac{n}{2}} \delta^2{}_i + \sum_{i=1}^{\frac{n}{2}} \delta^2{}_i}{n} - E_1{}^2(\delta) = D_1(\delta) \tag{17}$$

Since the force model has not changed essentially, the oil film of the double-piston structural hydro-viscous drive clutch is taken out in half, and the distribution rule of the thickness of each half of the oil film is also satisfied equation (11). Under the condition that the coefficient K, the number of oil films n, and the friction force F_{mz} and F_{mb} at the spline teeth are the same, the variance value of the thickness of oil film between the friction pairs under the double-piston compression structure can be directly derived from Fig. 4, 6 and 8.

Since the double-piston structure indirectly halves the number of friction pairs, the variance value of the thickness of oil film is significantly smaller than the variance value of the thickness of oil film under the single-piston structure. And the larger the number of friction pairs n, the more obvious the improvement effect is after the single-piston structure is improved to a double-piston structure.

7 Conclusion

Through the force analysis of friction plates and the equation derivation, the distribution rule for the oil film between the friction pairs in the hydro-viscous drive clutch is determined: under the single-piston unidirectional displacement structure, the first oil film thickness value between each friction pair is the smallest, and the subsequent oil film thickness value increases in turn. The progression formed after reciprocating the cube of oil film thickness between friction pairs is approximately an arithmetic progression.

By introducing the definition of expectation and variance, analyze the uniformity of the oil film thickness between the friction pairs in the hydro-viscous drive clutch, the analysis shows that increasing the coefficient K, reducing the number of oil films, n, and decreasing the friction forces, F_{mz} and F_{mb}, at the spline teeth can decrease the variance value of the oil film thickness between the friction pairs in the hydro-viscous drive clutch and enhance the uniformity of the oil film thickness.

The traditional single-piston compression structure of the hydro-viscous drive clutch friction pair has a high degree of unevenness, so it is easy to lead to the uneven wear problem of friction plates, thereby affecting the service life of the hydro-viscous drive clutch. By proposing a double-piston symmetrical compression structure, the number of friction pairs is indirectly halved by double-way symmetrical displacement of the friction disks, thus significantly enhancing the uniformity of the oil film thickness between the friction pairs in the hydro-viscous drive clutch.

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