Shape Optimization and Leakage Analysis of the Oil Cavity of Hydrostatic Sliding Table in Down-pressing Board of the Ultra-large Tonnage Compression Shear Tester

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Abstract. This paper proposes a method to reduce the oil leakage of the hydrostatic slide table. The method reduces the friction loss of the oil cavity through structural optimization, and derives the optimal ratio of the length and width of the oil cavity sealing oil edge in the oil cavity size. The clearance seal design is added around the down-pressing board, and the advantage of the extremely small thickness of the oil film is combined with an appropriate edge width and an oil-saving groove to ultimately reduce the leakage of oil. The theoretical calculations show that this method can better solve the oil in the down-pressing board problems with leaks.

Keywords: hydrostatic sliding table, compression shear tester, down-pressing board, clearance seal.

1. Introduction
In order to meet the testing needs of modern large-scale bridges and large-tonnage bearings and columns in construction projects, it is now necessary to develop a 60mn ultra-large tonnage static and dynamic multi-directional loading system and equipment (Figure 1) to achieve the mechanical performance inspection of ultra-large-tonnage bearings.

Figure 1. Model of compression shear tester.
The friction coefficient between the lower plate and the table of the testing machine has a greater impact on the detection of the bearing performance. The traditional compression shear testing machines use rolling contact to reduce friction. However, with the increase of pressure, the deformation of the internal support of the rolling guide block and the rolling contact surface with the table will increase with the increase of pressure, then affect the friction coefficient with the change of pressure, which will definitely affect the performance detection accuracy of the support. In addition, the assembly accuracy of the multiple sliding guide blocks installed on the down-pressing board, which will affect the uniformity of the plane under the rear roller assembled on the down-pressing board.

The hydrostatic support uses high-pressure oil film to achieve contact between the relative moving parts, eliminating direct contact dry friction, and the coefficient of friction of the oil film is low and basically constant. Therefore, the hydrostatic support is widely used in mechanical transmission mechanisms (Figure 2).

**Figure 2.** Hydrostatic support model of down-pressing board.

The large hydrostatic slides analyzed in this article are supported and guided by multiple hydrostatic oil cavities, and their characteristics meet the requirements of large size, strong bearing capacity, small sliding friction resistance, and high accuracy. It is planned to optimize the hydrostatic oil cavity leakage through calculation. And friction power loss, the size relationship between the length and width of the oil cavity is obtained, and the optimal pressure bearing model of the oil cavity arrangement and the sealing scheme to reduce leakage are obtained through theoretical calculations.

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2. **Working principle and size model of hydrostatic oil cavity**

2.1. **Technical requirements of the lower plate**
Maximum vertical load of the down-pressing board: 60mn;
   Down-pressing board size: 2.8 × 2.8m;
   Maximum horizontal shear displacement: ± 0.3m
2.2. Oil cavity structure

The oil cavity must have sufficient initial lifting force. If the slide table is to be floated, the initial lifting force must be designed to be greater than the dead weight of the slide table itself and the maximum weight of the workpiece. There must be a certain initial oil cavity bearing area.

The internal structure of the hydrostatic support oil chamber is usually divided into oil chamber type and oil groove type according to the internal shape. Under the same contour size, as long as the corresponding oil seal surface size is equal, the effective bearing area of the two oil chambers is equal. Oil groove type cavity is composed of several oil grooves, with incomplete connectivity in the middle, good vibration resistance, large oil film extrusion force, and can be used for large machine tools with low speed and large dead weight. According to the design of this model, the oil cavity width will not be less than 200mm. Therefore, reference is made to the structure characteristics of the shape of Chinese word "油嘴" oil pocket[5]. It is planned to initially determine the following oil cavity models(Figure 3).

![Figure 3. Oil cavity model.](image)

2.3. The aspect ratio of the oil cavity when the total power loss is minimum

From the perspective of the optimal solution for the power loss of the oil chamber, it is mainly divided into friction loss and flow loss. From these two perspectives, the calculation and calculation of the length and width ratio of the oil chamber seal oil side are performed.

2.3.1. Friction power loss. Let the relative sliding speed of the friction pair on the down-pressing board of the hydrostatic slide table be \( v \), The shear stress on the support surface of the sliding table is:

\[
\tau = \mu \frac{v}{h}
\]

(3)

The friction of the bearing surface is:

\[
F_i = A_\varepsilon \tau = \mu A_\varepsilon \frac{v}{h}
\]

(4)

where, \( \mu \) For oil dynamic viscosity, \( \mu = \nu \rho \) (\( \nu \) is the kinematic viscosity of the oil, \( \rho \) is the oil density), \( A_\varepsilon \) is the effective hydrostatic support area of the oil chamber,

\[
A_\varepsilon = (L + l)(B + b) / 4 = 0.81 LB
\]

Friction power loss is:

\[
\Delta P_i = F_i v = \nu \rho A_\varepsilon \frac{v^2}{h}
\]

(5)
2.3.2. *Leakage power loss*. The bearing surface leakage flow is:

\[ q = k_q h^3 p_s \] (6)

where, \( k_q \) is the leakage coefficient, which depends on the structure and dynamic viscosity of the hydrostatic bearing \( \mu \):

\[ k_q = \frac{1}{6\mu} \left( \frac{B+b}{L-l} + \frac{L+l}{B-b} \right) = \frac{3}{8\mu} \left( \frac{B}{L} + \frac{L}{B} \right) \] (7)

Among them, input pressure \( p_s = \alpha p_t \) (\( \alpha \) is for dimensionless pressure ratio), \( \varepsilon \leq 0.3 \). Therefore, no value range for this pressure ratio:

\[ \alpha \geq (1-\varepsilon_{\text{max}})^3 = 0.343 \] (8)

According to the support characteristics of the hydrostatic guide rail, when \( \alpha = 2/3 \), when the oil film has the maximum stiffness.

To eliminate viscosity \( \mu \) effect, and only reflects the structural shape of the bearing surface, the value of \( k \) is:

\[ k = \mu k_q = \frac{3}{8} \left( \frac{B}{L} + \frac{L}{B} \right) \] (9)

Then the leakage power loss is:

\[ \Delta P_q = p_s q = p_s k_q h^3 p_s = \frac{\alpha k h^3}{\mu} p_s^2 \] (10)

2.3.3. *Determine the relationship between length and width*. If it is required that the power loss of the oil cavity is minimum under a preset working condition oil film thickness, the proportional relationship between the size and length of the oil cavity can be derived, and the design will be based on this relationship when designing the layout later. \( L = f(B) = xB \), then:

\[ \Delta P = \frac{4\nu^2}{h^2} B f(B) + \frac{3\alpha h^3 p_s^2}{\mu} \left( \frac{f(B)}{B} + \frac{B}{f(B)} \right) \]

\[ = \frac{4\nu B^2 x}{h^2} + \frac{3\alpha h^3 p_s^2 (1 + x)}{8\mu} \] (11)

The optimal oil cavity size relationship refers to the relationship between the length and width of the oil cavity when the total power loss is the smallest.

\[ \frac{d\Delta P}{dx} = \frac{\mu B^2 \nu^2}{h^2} + \frac{3\alpha h^3 p_s^2}{8\mu} (1 - x^{-2}) = 0 \] (12)

Find when \( x = \sqrt[4]{\frac{h^2 p_s^2}{4\mu^2 B^2 \nu^2 + h^5 p_s^2}} \), \( \Delta P \) is minimal, and \( x \) the size varies with the given side length.
3. Controlling leaks

In the selection of the sealing scheme, due to the complex environment of the on-site test site and more impurities in the air, it is necessary to keep as little as possible the leakage of hydraulic oil to the exposed hydrostatic sliding table contact surface table (picture), and try to pass the internal leakage from the oil return hole at the bottom of the lower plate through the oil return pocket to the oil pocket. For these relative moving sealing devices, traditional stuffing boxes or mechanical seals cannot meet this requirement, special sealing devices is necessary. Due to the small thickness of the clearance of the hydrostatic slide table, it is objectively easy to meet the "clearance seal" condition, that is, the ratio of the length $B$ of the seal area to the seal clearance $b$, $B / b$, is reachable $10^4$~$10^5$ So that it fails to reach the critical Reynolds number and meets the performance requirements of its contact seal (relatively low leakage) [8].

The following is an optimized design from the perspective of the clearance and width of the edge seal area (Figure 4):

![Figure 4. Clearance seal profile.](image)

The principle of the clearance seal is the thrust caused by the pressure difference between the two ends of the clearance $F_T$. If it is less than the friction of the supporting surface $\epsilon_i$. The sealing conditions are satisfied. The oil outflow speed is $v$, the kinematic viscosity coefficient is $\mu$, and the coordinates of the clearance thickness direction are $x$, then:

$$\epsilon_i = -\mu B b \frac{dV}{dx}$$

(13)

Combined with the shape of this model (picture), then:

$$\epsilon_i = 2B(2.8 + 2B)\mu V / b$$

(14)

Assume that the oil pressure drop is linear. In the edge leakage model, the inlet pressure is the internal oil chamber working pressure $P_s$, then the thrust of oil film formation is:

$$F_T = B(2.8 + 2B)P_s$$

(15)

According to the requirements of the clearance seal principle, $F_T > \epsilon_i$, got the Edge clearance to prevent oil from leaking out is $b > \frac{2\mu V}{P_s}$, sealing area width is $B > 2 \times 10^4 \frac{\mu V}{P_s}$.
At this time, the leakage amount at both ends of the movement direction considering only the main leakage pathway is significantly reduced, and its shape coefficient $k_o = \frac{1}{6\mu} \frac{2.8+2B}{B}$, the leakage does not exceed:

$$Q_o = \frac{b^3 P}{6\mu} \frac{2.8+2B}{B}$$ (17)

In order to further reduce the leakage, two rows of throttling grooves and throttling labyrinths are uniformly distributed on the surface of the sealing area (Figure 5). The principle is to rely on the turbulent flow and expansion of the seal to reduce the pressure, and play a role of auxiliary sealing.

4. Expand application
Due to the large oil cavity pressure-bearing area of the entire down-pressing board, sealing measures should be set between the internal oil cavity pressure-bearing areas to prevent local leakage of oil chambers from affecting the adjacent oil chambers. Therefore, the design scheme is as shown below (Figure 6):

5. In Conclusion
(1) The hydrostatic slide table of the down-pressing board of this 60mn compression shear tester uses the overall oil cavity distribution design, which greatly improves the utilization of the bottom support and effectively reduces the oil pressure and power required for the pressure reducing oil cavity. The ideal size of the oil chamber is an effective way to improve support capacity.

(2) The non-contact sealing design of the multi-level different internal structure is adopted to reduce the leakage of oil, while avoiding the mutual interference of oil between the oil chambers.
(3) The calculation method of the clearance sealing clearance and size for the edge of the hydrostatic slide table is determined, which is convenient for adjusting the thickness of the oil chamber and the slide table to modify the clearance thickness and the width of the sealing area.

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References
[1] Yingzhou Zhang. Experimental study on the mechanical properties of a new type of bridge support [D]. Lanzhou: Lanzhou Jiaotong University, 2018. (in Chinese)
[2] Wenqin Xu, Yingda Sun. Study on determination of oil film thickness of machine tool hydrostatic guideway [J]. Manufacturing Technology and Machine Tool, 2009 (6): 56-58. (in Chinese)
[3] Feng Shen, Miao Liu. Influence of oil seal edge shape on flow field structure and bearing capacity [J]. Journal of Beijing University of Technology, 2012 (8). (in Chinese)
[4] Shengshuai He, Xianfeng Sun. Research on mechanical characteristics and structural optimization of large horizontal slides [J]. Intensity and Environment, 2010 (2): 1-7. (in Chinese)
[5] Zhenqian Ding. Hydrostatic bearing design [M]. 1989. (in Chinese)
[6] Yong Wu, Hongyuan Li, Ye Tian. Research on High Pressure and Heavy Duty Plane Hydrostatic Support System% [J]. Coal Mining Machinery, 2014, 035 (012): 95-97. (in Chinese)
[7] Zhao J H, Xiao-Chen W U, Wang J, et al. Influence of oil sealing belt on dynamic and static characteristic of closed type liquid hydrostatic slide [J]. Machine Tool & Hydraulics, 2018.
[8] Naibi Tong. Study on the mechanism of clearance seal [J]. Lubrication and Sealing (02): 33-38. (in Chinese)