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To cite this article: J Jiang et al 2018 IOP Conf. Ser.: Earth Environ. Sci. 163 012095

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Parameters optimization of the eccentric 8-shaped floating sleeve in external spur-gear pump

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Abstract. In this paper, the mechanism of the wear of floating sleeve was analyzed firstly and according to three different pressure assumptions. The force and moment of the inner part of the floating sleeve were deduced respectively and are compared with the analog value. As a result, the force and moment of the inner part of the floating sleeve based on assumption which has minimal error can be a reference value. Then the expression of outboard force and moment were deduced. Based on the Gamultiobi() Function, the resultant and resultant moment of the eccentric 8-shaped floating sleeve were taken as objective functions. And the outer diameter, inner diameter, angle and elastic strength of the seal ring on the eccentric 8-shaped floating sleeve were taken as independent variables. With the restrained conditions of the width of seal ring, the values of the independent variables were obtained when the values of objective functions were minimum. At last, numerical simulation were carried out on external spur-gear pump with two types of floating sleeves. The numerical simulation of gear pump with optimized floating sleeve shows that the resultant of the eccentric 8-shaped floating sleeve is almost zero and the pressure distribution is the same as the assumption. The research can provide a technical reference for gear pump sealing technology.

1. Introduction

External spur-gear pump has a lot of advantages, and it is widely used in various industries. However, during the operation of the gear pump, the abrasion of its main gear and driven gear with the ends of floating sleeve often occurs. On the one hand, it leads to the scratch phenomenon of the main gear and driven gear, and shortens the life of the external gear pump; on the other hand, it will increase the axial gap between drive and driven gear and the floating sleeve so that it will increase the axial leakage of external spur-gear pump and affect its volumetric efficiency \textsuperscript{[1]}.

As for optimization of the gear pump gap and side plate, domestic and foreign scholars both had relevant researches. such as foreign scholars Mucchi et al. \textsuperscript{[2][3]} who studied the influence of gear shaft eccentricity on the internal flow field and established the gear shaft eccentricity model. Liu Wei et al.\textsuperscript{[4][5]} studied the asymmetry of the pressure distribution in the cavity of the main gear and driven gear. He established the relationship between the deflection of the pump shaft and clearance, and obtained the resultant moment of the floating sleeve. Taking the three-stage high-pressure gear pump as an example, Yang Ji \textsuperscript{[6]} analyzed the factors that influenced the axial floatability of the sleeve. And he also proposed the design method of the sleeve with good axial float by means of concrete examples. In
conclusion, the optimizations of the gap and side plate had been carried out in recent years, but the contents are not full and detailed.

2. Mechanism of floating sleeve wear
As shown in figure 1, the green arrow represents the Suction Oil Chamber of external spur-gear pump and the pressure is low. The red arrows indicate the Pressure Oil Chamber and the pressure is high. The oil pressure in the Pressure Oil Chamber and Suction Oil Chamber will both form a y-axis perpendicular to the paper surface moment for the outer floating sleeve. The action areas of two types of oil are equal. But the oil pressure of Pressure Oil Chamber is much obvious than Suction Oil Chamber. So the moment of high pressure side for the y-axis by counter-clockwise rotation is much bigger than low pressure side of the oil Suction Oil Chamber, resulting in the non-zero moment. Under the action of the moment, one side of the floating sleeve will tend to rotate toward the low pressure side. So the gap between the floating sleeve and the gear is increasingly near. The side of the floating sleeve near the Suction Oil Chamber will finally contact drive gear and driven gear’s end, resulting in wear of the end of floating sleeve.

Figure 1. Sketch map of the inboard force for the floating sleeve. Figure 2. The force diagram of the eccentric 8-shaped floating sleeve.

The force diagram of the eccentric 8-shaped floating sleeve is shown as figure 2. By setting the seal ring on the outside of eccentric 8-shaped floating sleeve, the outer side is divided into high pressure area, sealing area and low pressure area in order to produce the force and eccentric moment to balance the force and moment in inboard floating sleeve. This construction will weaken and even prevent from the rollover trend and wear phenomenon of the floating sleeve.

3. Floating sleeve force and torque

3.1. Comparison of the inboard force and moment of the floating sleeve under different assumptions
On the assumption that pressure distributions of the main and driven gear are symmetrical, the force and moment of single gear for the floating sleeve can be calculated. To improve the reliability of the calculation, the inboard floating sleeve is divided into six areas.

As shown in figure 3, the law of the inboard pressure distribution of the floating sleeve is unknown. It is necessary to assume the pressure distributions of the respective regions in order to obtain the theoretical solutions of the force and the moment. The table 1 lists three hypotheses, in which the region I’s pressure distribution is calculated by using two parallel rotating disk models and the linear distribution assumptions are used for the pressure of II, III, IV, V and VI.
Table 1. The Pressure distribution hypotheses of the divided areas in the inboard floating sleeve.

| Hypotheses | Hypothesis 1 | Hypothesis 2 | Hypothesis 3 |
|------------|--------------|--------------|--------------|
| A area     | 0            | 0            | 0            |
| B area     | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{1}{2} \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{1}{2} \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{2}{6} \frac{\Delta p}{R \ln \frac{r}{R}} \) |
| Region I   |              |              |              |
| C area     | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{2}{3} \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{1}{2} \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{4}{6} \frac{\Delta p}{R \ln \frac{r}{R}} \) |
| D area     | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{3}{4} \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{5}{6} \frac{\Delta p}{R \ln \frac{r}{R}} \) |
| E area     | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{\Delta p}{R \ln \frac{r}{R}} \) | \( \frac{3p}{20} \omega^3 (r^2 - R^2) + \frac{\Delta p}{R \ln \frac{r}{R}} \) |
| Region II  | \( \Delta p \) | \( \Delta p \) | \( \Delta p \) |
| Region III | \( \Delta p \) | \( \frac{3}{4} \Delta p \) | \( \frac{5}{6} \Delta p \) |
| Region IV  | \( \frac{2}{3} \Delta p \) | \( \frac{2}{4} \Delta p \) | \( \frac{4}{6} \Delta p \) |
| Region V   | \( \frac{1}{3} \Delta p \) | \( \frac{1}{4} \Delta p \) | \( \frac{2}{6} \Delta p \) |
| Region VI  | 0            | 0            | 0            |

While the hypothesis 1 is considered, the force and moment of each region for floating sleeve are as in equation (1).

\[
F_n = F_1 + F_2 + F_3 + F_4 + F_5 = \frac{2p}{360} \left[ k_1 T_2 (\theta_1 + \theta_2 + \theta_4 + \theta_6) + k_2 T_2 (\theta_2 + 2\theta_4 + 3\theta_6 + 3\theta_8) \right] + \Delta p \left[ \frac{5p}{N} (R_x - R_y) + \tau_s \right]
+ \frac{1}{2} \left[ \frac{k}{R} + 2iN + V \left( R_x^2 - R_y^2 \right) - \frac{2}{3} \left( R_x - R_y \right)^2 + \frac{4}{3} \left( R_x - R_y \right) \tan \theta_4 \right] + \frac{2p}{3N} \Delta p \left( R_x - R_y \right) + \frac{1}{3} \Delta p \left( R_x - R_y \right)
\]

(1)

Where \( k_1 \) and \( k_2 \) are the parameters defined by formula as in equation (2):

\[
k_1 = \frac{3p}{20} \omega^3 \quad k_2 = \frac{1}{3} \Delta p \left( \ln \frac{R}{R_1} \right)
\]

(2)
\[ T_i \text{ and } T_2 \text{ are the parameters defined by formula as in equation (3):} \]
\[
T_i = \frac{R_i^4}{4} - \frac{R_i^3 R_j^2}{2} + \frac{R_i^2}{4} \quad T_2 = \frac{R_i^4}{2} \ln \frac{R_i}{R_j} - \frac{R_i^3}{4} + \frac{R_i^2}{4}
\]
\[ \alpha \text{ is the angle of Graduated Circle and defined by formula as in equation (4):} \]
\[
\alpha = \tan T - T = \text{inv}T
\]
\[ \theta_1 \text{ is the angle corresponding to low pressure area, } ^\circ; \theta_2 \text{ is the angle corresponding to the first tooth in the two tooth sealing, } ^\circ; \theta_3 \text{ is the angle corresponding to the second tooth in the two teeth sealing, } ^\circ; \theta_4 \text{ is the angle corresponding to High Pressure Oil Tank, } ^\circ; \theta_5 \text{ is the angle corresponding to high pressure area, } ^\circ; N \text{ is the number of teeth of the main or driven gear; } R_f \text{ is the radius of dedendum circle of the main or driven gear, } m; R_n \text{ is the radius of gear shaft of the main or driven gear, } m; R_c \text{ is the radius of addendum circle of the main or driven gear, } m; R_l \text{ is the radius of base circle, } m; S_i \text{ is the area of the sub-region } L \text{ of the region II, } m^2; S \text{ is the tooth thickness of reference circle, } m; R \text{ is the radius of reference circle, } m; T \text{ is the pressure angle of reference circle, } ^\circ; T_a \text{ is the pressure angle of tip circle, } ^\circ; U \text{ is the central angle, } ^\circ. \]

The resultant moment of \( F_{in} \) for the floating sleeve about the y-axis is defined by formula as in equation (5):
\[
M_{sv} = |M_{1s}| + |M_{2s}| - |M_{3s}| = |M_{10s}| - |M_{20s}| - |M_{30s}| = \left[ \cos \theta_1 - \cos (\theta_1 + \theta_2) \right] \left[ k_s S_i + k_s S_2 \right] + \left[ \cos \theta_1 + \theta_2 - \cos (\theta_1 + \theta_2 + \theta_3) \right] \left[ k_s S_i + 2 k_s S_2 \right] + \left[ \cos (\theta_1 + \theta_2 + \theta_3) - \cos (\theta_1 + \theta_2 + \theta_3 + \theta_4) \right] \left[ k_s S_i + 3 k_s S_2 \right] - \frac{\Delta \rho}{3} \left[ R_f^2 (1 - \sin \alpha_3) - \frac{1}{2} \tan^2 \alpha_3 \right] + \Delta \rho R_i x_i \left| - \frac{4 \pi}{9 N} \Delta \rho \frac{R_c - R_f}{R_c - R_f} \sin \phi \right| - \frac{2 \pi}{9 N} \Delta \rho \frac{R_c - R_f}{R_c - R_f} \sin \psi
\]

Where \( S_i \) and \( S_2 \) are the parameters defined by formula as in equation (6):
\[
S_i = \frac{R_i^3}{5} - \frac{R_i^2 R_f^3}{15} \quad S_2 = \frac{1}{3} R_i^3 \ln \frac{R_i}{R_f} - \frac{1}{9} R_i^2 + \frac{1}{9} R_i^2
\]

\( x_i \) is the x coordinate of the geometric center of the region II, m; \( \varphi \) is the angle between Centerline OM of area III and y-axis, \( ^\circ; \psi \) is the angle between Centerline OP of area IV and y-axis, \( ^\circ; \)

When the pressure distribution is on assumption 2 and 3, the formula derivation of the force and moment of the inboard floating sleeve for all regions is similar to the assumption 1.

For the external spur-gear pump studied in this paper, the geometrical parameters and working conditions are shown in the following table 2.

**Table 2.** The main parameters of the main and driven gear for the external spur-gear pump.

| Parameters | Numerical value | Parameters | Numerical value |
|-----------|----------------|-----------|----------------|
| N         | 10             | \( \theta_1 \) (\(^\circ\)) | 81.82          |
| R_c (m)   | 0.009          | \( \theta_2 \) (\(^\circ\)) | 36             |
| R (m)     | 0.015          | \( \theta_3 \) (\(^\circ\)) | 36             |
| R_a (m)   | 0.018          | \( \theta_4 \) (\(^\circ\)) | 108            |
| R_f (m)   | 0.01125        | \( \theta_5 \) (\(^\circ\)) | 98.18          |
| R_b (m)   | 0.01410        | S_i (mm\(^2\)) | 14.2965        |
| s (mm)    | 1.5\pi         | S_f (mm\(^2\)) | 5.0627          |
| T (\(^\circ\)) | 20          | x_i (m) | 0.01238         |
| T_a (\(^\circ\)) | 38.43      | L (m) | 0.01505         |
| U (\(^\circ\)) | 21.58        | \( \varphi \) (\(^\circ\)) | 44.18          |
| m         | 3              | \( \psi \) (\(^\circ\)) | 80.18          |

**Table 3.** The parameters of the oil and the operating parameters of the external spur-gear pump.
Density (kg/m$^3$) 800
Dynamic viscosity (Pa·s) 0.007
Rotating speed (r/min) 3000
Import pressure (Pa) 0
Outlet pressure (MPa) 2

The parameters in the above tables are plugged into the corresponding formula to obtain the forces and the moments of the respective sub-regions based on the three assumptions. The results are shown in the table 4. Simultaneously, the simulation is conducted for the gear pump (in the chapter 5) and the result is also included in the table 4.

Table 4. Comparison of the inboard forces of the floating sleeve

| Force(N) | Hypothesis 1 | Hypothesis 2 | Hypothesis 3 | Simulation valve |
|----------|--------------|--------------|--------------|------------------|
| I        | 45.21        | 38.50        | 41.85        |                  |
| II       | 677.32       | 305.16       | 305.16       |                  |
| III      | 677.32       | 279.12       | 310.13       |                  |
| IV       | 82.70        | 62.03        | 82.70        |                  |
| V        | 41.35        | 31.01        | 41.35        |                  |
| VI       | 0            | 0            | 0            |                  |
| Resultant | 846.58     | 715.81       | 781.20       | 895.61           |
| Errors   | 5.47%        | 20.08%       | 12.77%       | 0                |

The relative error of the calculated force according to hypothesis 1 is minimum compared with others. And the resultant moment of the inboard floating sleeve is 4.25N·m according to hypothesis 1. Finally, this paper use the force and moment calculated by hypothesis 1 as the reference values for structural parameter optimization.

3.2. The force of the outboard floating sleeve

The external force of the eccentric 8-shaped floating sleeve mainly comes from three areas: the low pressure region, the seal region and the high pressure region.

According to the operating conditions of the gear pump, the pressure of the oil in the Suction Oil Chamber is zero so that the pressure and moment in the low pressure region is zero.

Figure 4. The eccentric 8-shaped floating sleeve.

Figure 5. The force diagram of calculation of high pressure region

The force and moment (y-axis) of the seal area for the eccentric 8-shaped floating sleeve are as shown in equation (7).

\[
F_{ul} = 2K_u S_s = 2K_u \pi (R_1^2 - R_2^2) \quad M_{y_s} = \pi r K_u (R_1^2 - R_2^2)
\]

Where $K_u$ is the elastic strength of the seal ring, MPa; $R_1$ is the outside diameter of the seal ring, m; $R_2$ is the inside diameter of the seal ring, m; $e$ is the eccentric distance of the seal ring, m;
The force and moment of the high pressure region for the eccentric 8-shaped floating sleeve are as shown in equation (8) and equation (9).

\[
F_0 = 2p_GS_0 = 2p_G \left( \pi R_1^2 - \pi R_2^2 - 2S_1 + 2S_2 \right) = 2p_G \left[ \pi R_1^2 - \pi R_2^2 - 2 \left( \frac{1}{2} \arccos \left( \frac{L}{R_1} \right) - \frac{1}{2} L \sqrt{R_1^2 - L^2} \right) + 2S_2 \right]
\]

\[
M_{e0} = p_G \pi R_1^2 e
\]

Where \( p_G \) is the pressure of high pressure area.

Therefore, the resultant and moment of the outboard floating sleeve are as shown in equation (10) and equation (11).

\[
F_{\text{out}} = F_L + F_M + F_G = 2K_M \pi \left( R_2^2 - R_1^2 \right) + 2p_G \left[ \pi \left( R_1^2 - R_2^2 \right) - 2 \left( \frac{1}{2} \arccos \left( \frac{L}{R_1} \right) - \frac{1}{2} L \sqrt{R_1^2 - L^2} \right) + 2S_2 \right]
\]

\[
M_{\text{out} - y} = 2 \left( M_{e0} - M_{M0} \right) = 2 \left[ p_G \pi R_1^2 e - \pi eK_M \left( R_1^2 - R_2^2 \right) \right]
\]

4. Parameter optimization of the eccentric 8-shaped floating sleeve

4.1. The objective function of the optimal design for the eccentric 8-shaped floating sleeve

Using the inboard and outboard resultant and moment of the eccentric 8-shaped floating sleeve as the objective function for parameter optimization, the equation (12) can be obtained [7][8].

\[
\begin{align*}
\min f &= |F_{\text{out}} - F_m| \\
\min M &= |M_{\text{out} - y} - M_{\text{e}0}|
\end{align*}
\]

\[
\begin{align*}
\min f &= 2K_M \pi \left( R_2^2 - R_1^2 \right) + 2p_G \left[ \pi \left( R_1^2 - R_2^2 \right) - 2 \left( \frac{1}{2} \arccos \left( \frac{L}{R_1} \right) - \frac{1}{2} L \sqrt{R_1^2 - L^2} \right) + 2S_2 \right] - 1693.16 \\
\min M &= 2 \left[ p_G \pi R_1^2 e - \pi eK_M \left( R_1^2 - R_2^2 \right) \right] - 8.50
\end{align*}
\]

4.2. The constraint condition of the optimal design for the eccentric 8-shaped floating sleeve

The constraints of geometric construction of the eccentric 8-shaped floating sleeve are shown in equation (13). In order to use Gamultiobi() Function, we need to change some linear inequality constraints into the form of matrices(Ax ≤ b). For the bound of the independent variables, they are written directly into the program.

\[
\begin{align*}
0 \leq R_1 &\leq R_1 \\
0 \leq R_2 &\leq L \\
0 \leq e &\leq R_2 \\
0 \leq e + R_2 &\leq L \\
-R_1 + R_2 + 0.001 &\leq 0 \\
-R_2 + e &\leq -R_1 \\
0 \leq e &\leq e \\
0 \leq R_2 &\leq R_2
\end{align*}
\]

\[
\begin{bmatrix}
0 & 1 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix}
\]

\[
\begin{align*}
R_1 &\leq L \\
R_2 &\leq e \\
K_M &\leq -0.001 \\
L &\leq \frac{1}{2} \arccos \left( \frac{L}{R_1} \right) - \frac{1}{2} L \sqrt{R_1^2 - L^2} + 2S_2
\end{align*}
\]

The final optimal solutions are the combinations of independent variables. And by conducting dimensionless summation for the forces and moments obtained by the combinations of independent variables, the value of the independent variable corresponding to the minimum value for the dimensionless summation are selected. The results are as shown in equation (14).

\[
R_1 = 0.01379m, R_2 = 0.01302m, e = 0.00376m, K_M = 0.9920538MPa
\]

5. Numerical simulation

5.1 Pretreatment of the external spur-gear pump

According to the geometric parameters of the optimized eccentric 8-shaped floating sleeve in chapter 4, the geometric model of the ear-type floating sleeve is established by the software, and the final result is shown in figure 6.
Figure 6. The calculative model of external spur-gear pump with the eccentric 8-shaped floating sleeve.

Leading the model into the PUMPLINX for mesh generation, the calculative grid uses structured grid. According to the characteristics of the working process of gear pump, the local grid reconstruction method of dynamic grid is used to deal with the problem that is the computational domain varied with time \[9\]. Figure 7 is the sectional grid view of the model.

5.2. The force analysis of the eccentric 8-shaped floating sleeve

By simulating for the external spur-gear pump with non-optimized and eccentric 8-shaped floating sleeves, the force diagrams of the two types of floating sleeves are shown in figure 8 and figure 9.

Figure 7. Sectional grid view of the external spur-gear pump with eccentric 8-shaped floating sleeve.

Figure 8. The force diagram of the non-optimized floating sleeve

(a) The force acting on the inboard eccentric 8-shaped floating sleeve.

(b) The force acting on the outboard eccentric 8-shaped floating sleeve.
(c) Balance of the force on the inboard and outboard eccentric 8-shaped floating sleeve.

Figure 9. The force diagrams of the eccentric 8-shaped floating sleeve.

In the figure 9, the average force acting on the inboard eccentric 8-shaped floating sleeve is 1837.25N and the outboard is 1849.75N. It basically reached a balance for the force acting on inboard and outboard floating sleeve. However, there is still a large pulsation phenomenon for the inboard and outboard force amplitude. Comparing with the non-optimized floating sleeve, the resultant significantly reduces. Therefore, it can prove that the design of eccentric 8-shaped floating sleeve can reduce the rollover and wear of the sleeve. Table 5 is the comparisons between the resultant of the inboard and outboard forces for the two types of floating sleeves.

### Table 5. The contrast of the force for the inboard and outboard floating sleeve

| Force (N)                        | Inboard force (N) | Outboard force (N) | Resultant (N) |
|----------------------------------|-------------------|--------------------|---------------|
|                                  | Theoretical value | Simulation value   | Error         | Theoretical value | Simulation value | Error         | Theoretical value | Simulation value |
| No optimized floating sleeve     | 1693.16           | 1791.21            | 5.47%         | 1693.16         | 1791.21           |               |
| Eccentric 8-shaped floating sleeve | 1693.16           | 1837.2             | 7.84%         | 1692.92         | 1849.75           | 8.48%         | 0.24             | 12.5             |

5.3. Visualized analysis

As shown in the figure 10, for the external spur-gear pump with no optimization, the two teeth sealing makes a sealed effect \cite{10}\cite{11}. In addition, for other teeth in the high pressure region, the pressure is substantially equal, which is consistent with the assumption of calculating the acting force of floating sleeve and the moment. Finally, in the engaged area of the drive and driven gear, the pressure is significantly higher than the pressure in the Pressure Oil Chamber. For the external gear pump with an eccentric 8-shaped floating sleeve, the pressure distribution is similar.

Figure 10. Static pressure distribution on the z=0 plane
6. Conclusion
The mechanism of the wear of the floating sleeve is analyzed, and it is concluded that the unbalance of the resultant and resultant moment is the cause of the wear.

By designing the eccentric 8-shaped floating sleeve, the objective function is optimized based on genetic algorithm under the restrained conditions. Geometric dimensions of the special structure for floating sleeve are obtained in this research methods when the resultant and resultant moment of floating sleeve are minimal.

The numerical simulation is carried out for the external gear with non-optimized and eccentric 8-shaped floating sleeve. Compared with the non-optimized floating sleeve, the resultant reduces significantly and the pressure distribution is similar to derivation calculation, which verifies the effectiveness of weakening wear for the 8-shaped floating sleeve.

Acknowledgments
The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was financially supported by the High Speed Sprint Engine Key Laboratory Open Fund Project of China (No.20120101018) and the Key Lab of Hydraulic Machinery Transient, MOE, Wuhan University.

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