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Analysis and Key Parameter Optimization Design of Leningrader Seal Performance

□ YANG Dongya^{1,2}, WANG Xuelin¹, WANG Feng¹, ZHANG Hailong¹, GAO Gui³

1. College of Mechanical and Electrical Engineering, Lanzhou University of Technology, Lanzhou 730050, Gansu China;

2. Wenzhou Pump and Valve Engineering Research Institute, Lanzhou University of Technology, Wenzhou 325000, Zhejiang, China;

3. State Key Laboratory of Solid Lubrication, Lanzhou Institute of Chemical Physics, Chinese Academy of Sciences, Lanzhou 730000, Gansu, China

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Abstract: In order to improve the performance and service life of the Leningrader seal of the Stirling engine piston rod, interference, preload and friction coefficient were taken as influencing factors, and the curved surface response method was adopted to reduce the contact stress of sealing surface and von Mises stress of the sealing sleeve as the response index, with the optimization goal of reducing wear and extending life. The above three key parameters are analyzed and optimized, the influence of each parameter on the sealing performance and service life is obtained, and the best combination scheme of the three is determined. The results show that the interaction between pre-tightening force and interference fit has the greatest impact on contact stress. The interaction between interference fit and friction coefficient has the most significant effect on von Mises stress. The optimized parameters can reduce the maximum contact stress and maximum von Mises stress of the sealing sleeve by 26.3% and 20.6%, respectively, under a media pressure of 5-9 MPa. Test bench verification shows that the leakage of the optimized sealing device in 12 h is reduced by $0.44 \text{ cc} \cdot \text{min}^{-1}$ ($1 \text{ cc} = 1 \text{ cm}^3$). The wear rate of the sealing sleeve is 1.08% before optimization and 0.45% after optimization, indicating that the optimized parameters in this paper are effective.

Key words: Leningrader seal; Stirling engine; performance analysis; optimized design; parameter configuration

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0 Introduction

The piston rod Leningrader seal is the last seal of the Stirling engine, and its sealing performance directly affects the working efficiency and service life of the engine^[1]. The failure mechanism of the Leningrader seal is comprehensively affected by working conditions, material characteristics and structural parameters, especially

high temperature and pressure. Working conditions will aggravate the deformation and wear of the seal, thus affecting its sealing, reliability and service life. At present, Leningrader seals still have durability problems, which will lead to incomplete seals or leaks, thus affecting the performance and reliability of the engine, and seals require high-precision processing and assembly, as well as high production costs. Furthermore, maintenance and re-

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Biography: YANG Dongya, male, Associate professor, research direction: Stirling machine sealing technology. E-mail: yangdy@lut.edu.cn

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placement of seals also require high costs, increasing the cost of use. Therefore, on the basis of ensuring the piston rod seal has "zero" leakage, the optimization design is of great significance to improve the working efficiency of the Stirling engine.

Numerous scholars have studied the performance of piston rod seals. Zhou *et al*^[2] studied the sealing characteristics of a rubber X-shaped combination seal under high-pressure hydrogen gas, and developed a subroutine using Abaqus software to simulate the sealing performance coupled with hydrogen expansion, and evaluated the applicability of the X-shaped seal ring. Zhang *et al*^[3] established a finite element model of O-ring, and studied the effects of friction coefficient, precompression amount, and medium pressure on the static and dynamic sealing performance of the O-ring. Zhang *et al*^[4] established a two-dimensional finite element model of O-ring using ANSYS software, and compared it with experimental results to determine the effects of clearance size changes and fluid pressure transients on O-ring. Kaushal *et al*^[5] conducted optimization on the shape of the sealing ring structure to enhance the sealing performance of the labyrinth V-shaped dynamic seal. The results demonstrated that the modified shape improved the sealing performance by 50% compared to the original structure. Hu *et al*^[6] established a numerical model of V-ring seals based on experimental data and studied the comprehensive effects of geometric dimensions on the sealing performance and application of the sealing ring. Azzi *et al*^[7] studied the influence of different sealing elements and operating parameters on the frictional force of sealing elements. Luo *et al*^[8] conducted a simulation of the frictional behavior of the cylinder seal, combining the analysis results with the experimental results to reveal the dynamic changes of the seal caused by friction. Yakovlev^[9] provided a detailed experimental setup to study the wear of lip seals in frictional contact with rotating shafts. An empirical dependency was proposed to evaluate the life of rubber and polyurethane lip seals depending on their degree of wear. Hu *et al*^[10] designed a combination seal of tilted pad and slip ring made of polytetrafluoroethylene device, which can compensate for seal wear automatically. Zhang *et al*^[11] employed response surface methodology to investigate the changes in maximum contact stress response of rubber cylinders under various factors and levels, with the aim of enhancing the sealing performance of the sealant. Androsovich *et al*^[12] utilized response surface methodology to generate a geometric

parameter function for labyrinth seals, with the objective of improving the operational efficiency and sealing performance of gas turbines. Subsequently, an optimized labyrinth seal was developed and its features were compared with those of the initial seal. Zhang *et al*^[13] applied response surface methodology and multi-objective optimization design approach to optimize and analyze the contact pressure and leakage rate of dynamic seals under hydraulic pressure, and obtained the variation patterns of contact pressure and leakage rate. Liu *et al*^[14] utilized a combination of response surface methodology and Box-Behnken experimental design to obtain functional expressions for optimization objectives and constraint conditions in relation to optimization parameters. By means of a particle swarm optimization algorithm, the inherent frequency value was significantly improved, thereby achieving optimization of the dynamic characteristics of dry gas seals. Jiang *et al*^[15] aimed to improve the sealing performance of the metal spring C-ring and established an optimization model based on the multi-island genetic algorithm. The optimization design of key structural parameters was investigated. Cao *et al*^[16] employed an orthogonal experimental design method to optimize the structural parameters of sealing rings, with the minimum reduction of maximum contact pressure between cap seal and piston rod and the most extended seal life as optimization objectives.

The above studies investigated the sealing performance of different structures seals in three main aspects: operating parameters, sealing structure, and sealing materials. However, there is little research on Leningrader seals, and no use of regression design methods for parameter optimization design. It is crucial to minimize the contact stress and von Mises stress while ensuring "zero" leakage of in the seal for optimizing sealing parameters. Firstly, the distribution of contact stress on the sealing surface and von Mises stress on the sealing sleeve were investigated using the finite element method. The influence of different sealing parameters, namely assembly interference fit, spring preload force, and sealing surface friction coefficient, on the sealing performance was analyzed. Based on this analysis, the key sealing structural parameters were optimized and configured using regression design methods. Finally, the optimization results were verified for leakage and wear through an experimental platform, providing theoretical guidance for the optimization of the Leningrader sealing structure.

1 Modeling

1.1 Geometric Model

Figure 1 illustrates the structure of the Stirling seal system. The sealing sleeve is commonly made of a PTFE-filled modified composite material, while the locating seat, casing, and brace ring are typically made of

brass. As the geometric structure, constraints, boundary conditions, loads, stresses, and strains of the Leningrad seal are all axisymmetric, which can be regarded as two-dimensional axisymmetric. The contact stress at the sealing interface is denoted by σ_c , the maximum contact stress is $\sigma_{c\max}$, the von Mises stress is σ_v , and the maximum von Mises stress is $\sigma_{v\max}$.

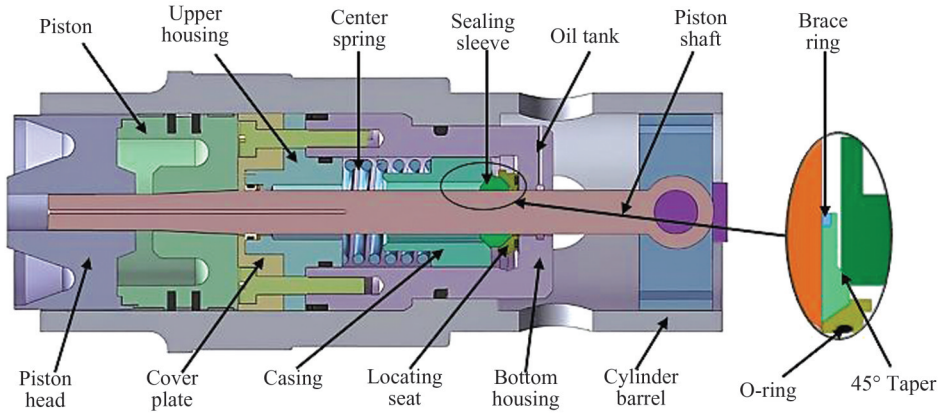


Fig. 1 Structural diagram of the Stirling engine sealing system

1.2 Mathematical Model

1.2.1 Interference fit model analysis

In engineering applications, the piston rod and the sealing sleeve are assembled using a base shaft system interference fit, which corresponds to a radial compressive stress denoted by P_r , as illustrated in Fig. 2.

The formula for calculating the radial force P_r is shown in Eq. (1)^[17]:

$$P_r = \frac{\Delta E s}{r^2} \quad (1)$$

where Δ is half of the interference fit amount between the sealing sleeve and the piston rod, $\Delta = (d - d_0)/2$; d_0 (mm) is the inner diameter of the casing; E (MPa) is the compressive modulus of elasticity of the material; r (mm) is the radius of the piston rod axis, $r = 0.5d$; s (mm²) is the minimum cross-sectional area of the sealing sleeve, $s = \pi(d_1^2 - d^2)$, d_1 (mm) is the minimum outside diameter of the sealing sleeve, d is the outer diameter of the piston rod.

1.2.2 Preload force model analysis

The forces of the seal sleeve under spring preload P_c (MPa) and radial compression P_r with applied work pressure P_0 are shown in Fig. 2, where $P_c = 4kx/\pi(D_0^2 - d_0^2)$, $k = 8$ N/mm is the spring stiffness coefficient, and the equilibrium equation for the force acting on the sealed sleeve in the y -axis direction^[17] is

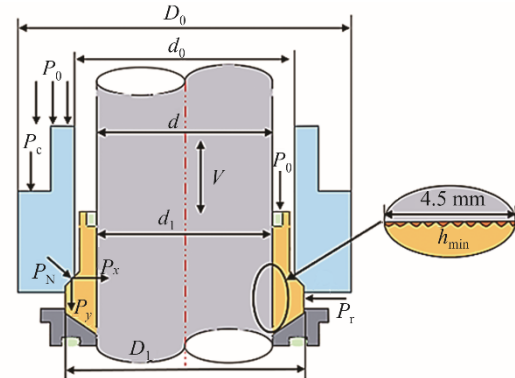


Fig. 2 Schematic diagram of force analysis of Leningrader parts

$$\pi(d \cdot R_{a1} + D_1 \cdot R_{a2}) \cdot P_x dy - \frac{\pi}{4}(D_1^2 - d_1^2) \cdot dp_y = 0 \quad (2)$$

where D_1 is the maximum outer diameter of the sealing sleeve, R_{a1} and R_{a2} are the friction coefficients of the contact surface of the piston rod and the sealing sleeve, respectively.

According to the principle of combination seal, the relationship between radial specific pressure P_x and axial specific pressure P_y is

$$P_x = R P_y \quad (3)$$

Substitute (3) into (2) and solve

$$\ln P_y = 4R \left[\frac{R_{a1} + R_{a2}(D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} + C \quad (4)$$

Substitute boundary condition $P_{y|y=0}=P_0$ into Eq. (4) to obtain Eq. (5):

$$C = \ln P_c \quad (5)$$

$$P_y = P_c \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (6)$$

$$P_x = R \cdot P_c \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (7)$$

where y takes the maximum displacement of 2 mm, pressure transfer coefficient $R = 1$, pressure $P_x = P_y$, then the contact stress on the 45° cone is P_N :

$$P_N = \sqrt{2} P_c \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (8)$$

When considering the media pressure P_0 , after filling the media gas at a certain pressure, the seal sleeve is tightened against the piston rod. At this time the axial force on the seal sleeve is P_y and the radial force is P_x :

$$P_y = (P_c + P_0) \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (9)$$

$$P_x = R \cdot (P_c + P_0) \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (10)$$

$$P_N = \sqrt{2} (P_c + P_0) \cdot \exp \left\{ 4R \left[\frac{R_{a1} + R_{a2} (D_1/d_1)}{(D_1/d_1)^2 - 1} \right] \frac{y}{d_1} \right\} \quad (11)$$

1.3 Analysis Steps and Basic Assumptions

Engineering experience shows that the Leningrad seal casing is affected by the media pressure, the initial pre-tightening force and interference fit during the installation stage. Therefore, the entire sealing process is simulated in three load steps during the numerical simulation stage. Step 1: Apply an interference fit to the sealing sleeve corresponding to a radial pressure and pre-

loading force, causing the sealing sleeve to undergo pre-compression and simulate its initial state, as shown in Fig. 3(a). Step 2: As shown in Fig. 3(b), H_2 gas is introduced to apply media pressure to the combination seal. Step 3: After loading the payload, the piston rod is subjected to a Y -axis reciprocating motion to simulate the dynamic sealing process, as depicted in Fig. 3(c).

Initial boundary conditions are set as follows: the media pressure is 9 MPa, the preload force is 32.5 MPa, the interference fit amount is 0.12 mm, and the average velocity of the piston rod is 2.5 m/s. The friction coefficient is set to 0.2 for all contact surfaces except for that between the piston rod and the sealing casing, which is affected by the oil film.

Based on actual working conditions, assumptions are made for the Leningrad seal: 1) The pressure variation in the direction of lubricating oil film thickness is negligible; 2) The influence of oil film curvature is negligible; 3) Lubricating oil is assumed to be a Newtonian fluid; 4) Viscosity is assumed to be constant in the lubricating film thickness direction.

2 Leningrader Seal Performance Analysis

The maximum contact stress on the sealing surface and the maximum von Mises stress distribution on the sealing sleeve were obtained through finite element analysis, as illustrated in Fig. 4 and Fig. 5, respectively. The contact stress is an indirect indicator of the sealing performance. In theory, leakage can be prevented when the maximum contact stress is greater than the media pressure. Moreover, the greater the contact stress, the better the sealing effect^[18]. However, excessive contact stress can increase power consumption and aggravate wear, and thereby affect service life. The analysis shows that the maximum contact stress of Leningrader sealing surface is greater than the medium pressure (Fig. 4), and

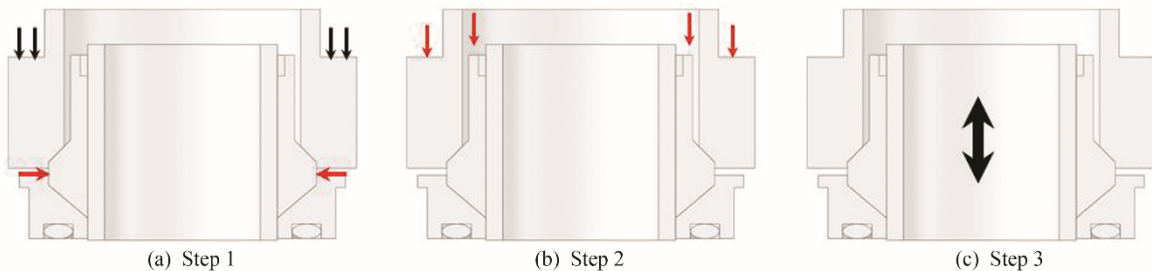


Fig. 3 Analysis steps of the Leningrader seal

the sealing effect is up to standard. On this basis, the contact stress should be as low as possible.

Von Mises stress reflects the size of the cross-section. It is a key parameter for evaluating the fatigue failure of the sealing ring. The greater the von Mises stress value is, the more prone it is to relaxation and cracks of the material seal failure, and the shorter its life

will be^[19]. As shown in Fig. 5, under the impact of boundary conditions, the von Mises stress concentration of the Leningrad seal sleeve is mainly located at the 45° conical surface. Particularly during the return stroke, the stress concentration area spreads to the sealing surface, which can result in cracks and easy failure of the sealing surface.

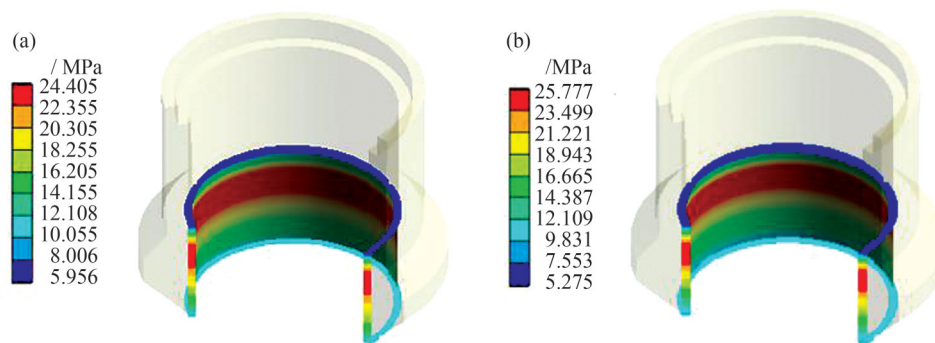


Fig. 4 Cloud chart of maximum contact stress of sealing sleeve under positive stroke (a) and return stroke (b)

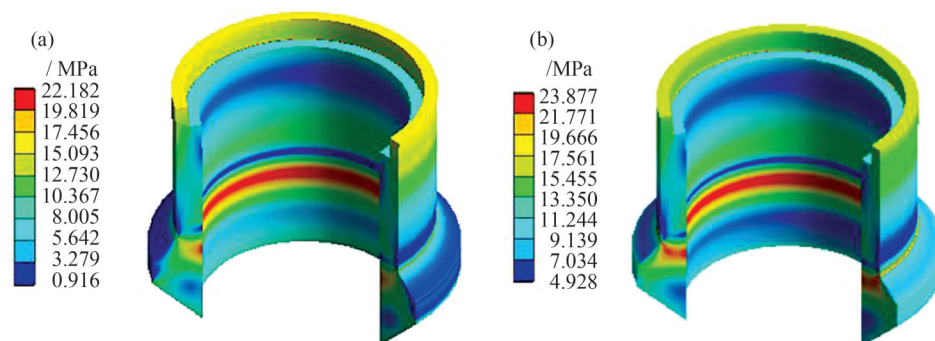


Fig. 5 Cloud chart of maximum von Mises stress in the sealing sleeve under positive stroke (a) and return stroke (b)

3 Analysis of the Influence of Key Parameters on Sealing Performance

3.1 Analysis of the Influence of Interference Quantity

In the reciprocating movement of the piston rod, the interference fit can improve the stability and sealing property between the piston rod and the sealing sleeve. However, the large interference fit I will increase the contact stress, equivalent stress and wear of the sealing surface, and reduce the service life of the sealing casing. If the interference fit I is too small, when the preload force cannot provide enough radial extrusion, the sealing structure will not meet the required working conditions, and the piston rod and the sealing sleeve cannot fit perfectly, resulting in the distortion of the sealing sleeve.

Therefore, it is indispensable to study the effect of the interference fit I on the sealing performance.

Through mechanical analysis, the friction coefficient of the sealing surface was limited to $\mu=0.2$, and under normal conditions, the interference fit varied between 0.1-0.15 mm^[20]. The finite element analysis is combined with Eq. (1). The corresponding change curve of the contact stress at each node of the sealing surface is shown in Fig.6. During the operation of the sealing device, the piston rod reciprocates, and the sealing element undergoes wear, resulting in a decrease in radial thickness. The greater the interference fit of the sealing element, the smaller the radial clearance between the sealing surfaces. Therefore, under the action of media pressure, a larger radial compensation force is generated when the sealing surfaces are in contact, leading to an in-

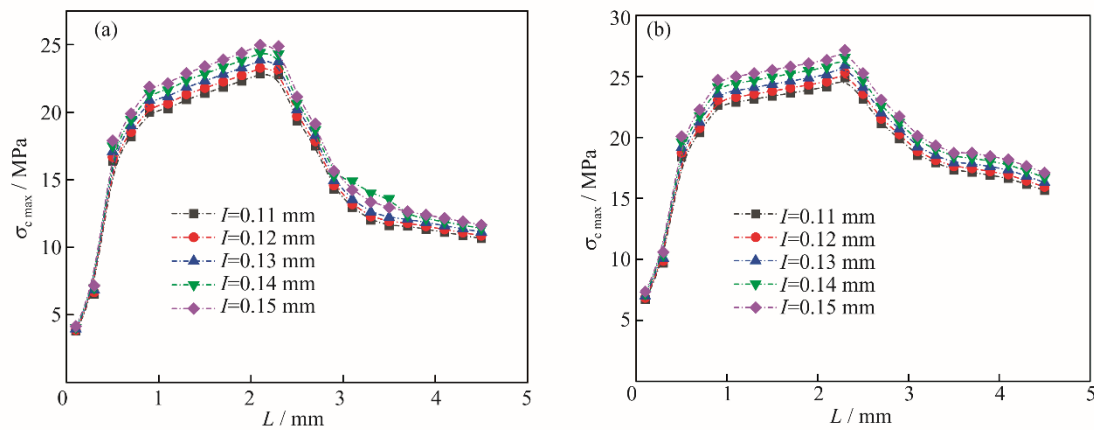


Fig. 6 Curve of contact stress on the sealing surface with interference of different interference fit I under positive stroke (a) and return stroke (b)

crease in contact pressure on the sealing surfaces^[21].

Figure 7 shows the von Mises stress values of the sealing sleeve under different interference fit. The contact stress between the sealing surfaces is derived from the combined effect of external forces and the elastic deformation of the sealing sleeve. The elastic deformation generates internal stress in the sealing sleeve, which affects the magnitude of the von Mises stress. Moreover, as the interference fit increases, the axial length of the sealing sleeve changes more significantly, leading to more elastic deformation and internal stress, further increasing the von Mises stress. Therefore, the larger the interference fit, the larger the von Mises stress^[22], which can affect the sealing performance and service life of the sealing sleeve. Considering the contact pressure and equivalent stress, it is necessary to select an appropriate interference fit based on specific applications to ensure the reliability and safety of the sealing sleeve.

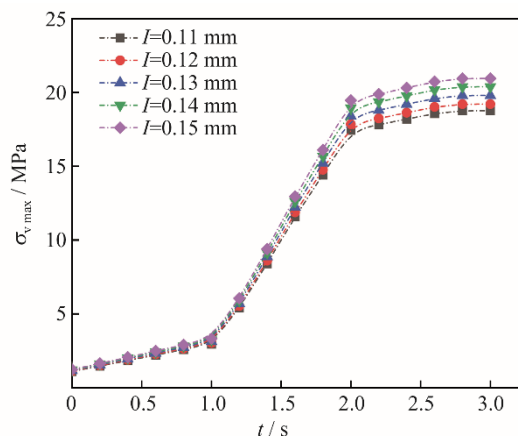


Fig. 7 Effect of interference of different interference fit I on von Mises stress of sealing sleeve

3.2 Analysis of the Influence of Spring Preload Force

The preload force of the spring has an essential effect on the contact pressure. As the preload force of the spring increases, the elastic deformation of the spring increases, causing slight changes in the relative position between the sealing surfaces, which can affect the shape and size of the sealing surfaces and thus the sealing performance. Suppose the preload force of the spring is too large. In this case, the pressure between the sealing surfaces may exceed the bearing limit of the sealing material, causing wear and deformation of the sealing surfaces and thus reducing the sealing performance. If the preload force is too small, the pressure between the sealing surfaces may be insufficient to ensure good sealing performance. Therefore, in the Leningrader seal, the preload force of the spring should be selected reasonably based on the material characteristics of the sealing element and the requirements of the working environment to ensure optimal sealing performance.

According to mechanics analysis, the compression amount of the sealing sleeve spring is between 12-14.2 mm. Combined with equations (2)-(11), five groups of preload force P_c were selected as 29.7, 31.1, 32.5, 34.0, and 35.4 MPa. The corresponding change curve of contact stress on the sealing surface is shown in Fig. 8. When the seal is subjected to work pressure, the spring will produce a greater reaction force due to the increased elastic deformation of the spring, and the seal surface contact pressure will increase^[23]. In addition, the increase of the spring preload force will reduce the length of the spring, change the initial shape of the spring, lead to

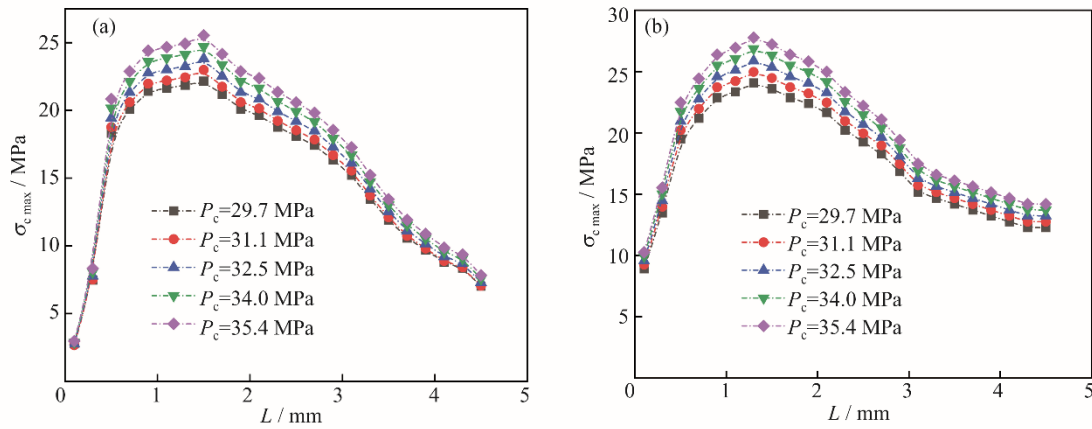


Fig. 8 Curve of contact stress on the sealing surface with interference of different preload force P_c under positive stroke (a) and return stroke (b)

changes in the shape and size of the sealing sleeve, and affect the contact stress.

Figure 9 shows the curve of the von Mises stress of the sealing sleeve under different preload forces. As the contact stress increases, the deformation of the sealing sleeve also increases, leading to an increase in the von Mises stress. Increasing the preload force of the spring makes the contact stress distribution on the sealing sleeve more uneven, thereby making the von Mises stress more concentrated. If the von Mises stress exceeds the bearing limit of the material, it can lead to damage and failure of the sealing sleeve. Therefore, combined with the above analysis, the actual application needs to be reasonably designed and selected according to the specific situation.

3.3 Analysis of the Impact of Friction Coefficient

The power consumption and service life of sealing

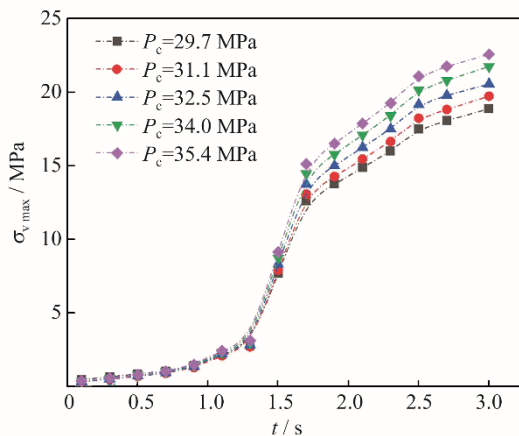


Fig. 9 Effect of preload force of different preload force P_c on von Mises stress of sealing sleeve

elements are closely related to the quality of the sealing surface, including surface roughness, hardness, and other factors, which are important causes of failure in reciprocating motion seals. The friction coefficient u of the modified polytetrafluoroethylene sealing sleeve is between 0.1 and 0.4, and the friction coefficient u of the five groups is 0.1, 0.15, 0.2, 0.25 and 0.3, respectively, and the finite element analysis is carried out. The variation curve of the contact stress between the sealing surfaces is shown in Fig.10, with a limiting interference of $I = 0.12$ mm and a preload force of $P_c = 32.5$ MPa. The contact stress exhibits a parabolic curve during both the positive and return strokes of the reciprocating motion. When the piston rod experiences reciprocating motion, the greater the friction coefficient, the more serious the wear, and the larger the clearance of the sealing surface. Under the same interference and preload, the smaller the friction coefficient, the greater the contact stress. Figure 11 shows the variation curve of von Mises stress in the sealing sleeve. According to the variation of the graph and the stress pressure difference, it can be seen that the friction coefficient has a greater influence on the von Mises stress value. In combination with contact stress, von Mises stress, the friction coefficient of the sealing surfaces, needs to be considered when designing and selecting seals to ensure reliability and service life of the seals.

4 Optimization Design of Sealing Parameters Based on Response Surface Methodology

Optimization of the Stirling engine Leningrader

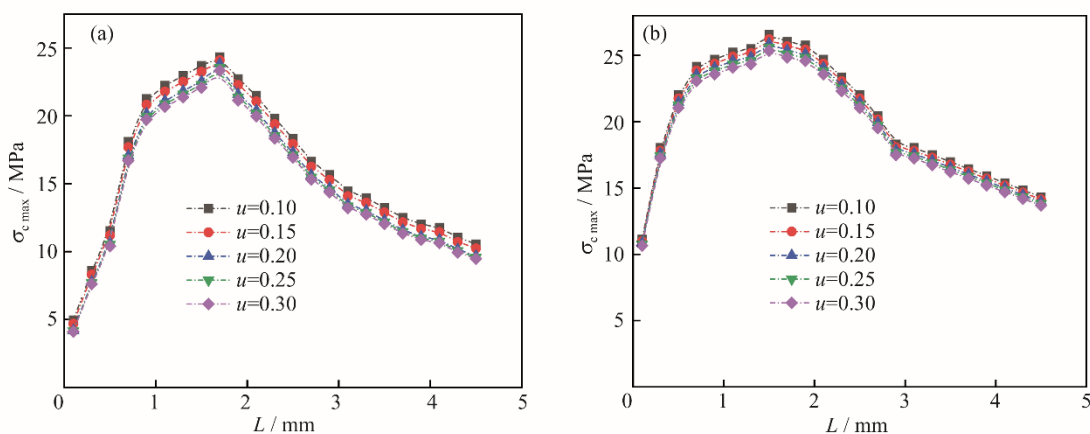


Fig. 10 Curve of contact stress on the sealing surface with interference of different friction coefficient u under positive stroke (a) and return stroke (b)

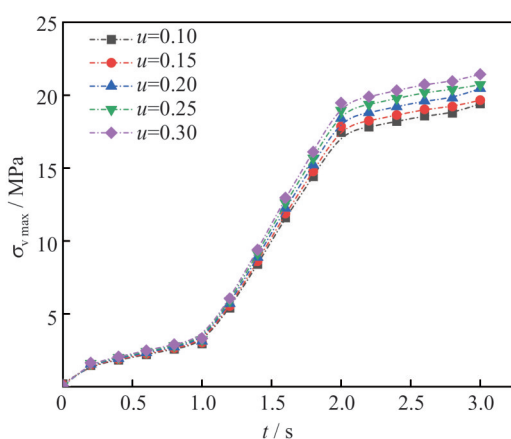


Fig. 11 Effect of different friction coefficient u on von Mises stress of sealing sleeve

seal parameters requires a combination of two aspects: maximum contact stress and maximum von Mises stress. Based on the regression design method and the premise of ensuring reliable sealing performance while reducing the maximum contact stress of the sealing sleeve and piston rod contact surface, as well as minimizing the maximum von Mises stress of the sealing sleeve as the re-

sponse optimization target, three factors, assembly interference, spring preload, and sealing surface friction coefficient, are selected for parameter optimization design. Multiple quadratic regression equations were used to fit the functional relationship between the factors and the response values, and the best sealing parameter combination scheme was obtained by analyzing the response surface and ANOVA.

4.1 Response Surface Analysis with the Maximum Contact Stress as the Index

The design results of the response surface method with the maximum contact stress as the index are shown in Fig. 12, which illustrates the surface plot of the maximum contact stress of the sealing casing with different parameters crossing each other. The response surface has a high inclination, significant color change, and obvious ellipticity of contour diagram, and their interaction is significant. Therefore, the combination factors that have the greatest influence on the contact stress are preload force and interference fit. The corresponding estimated regression coefficients are shown in Table 1. It can be observed that the model value (P -value) of the

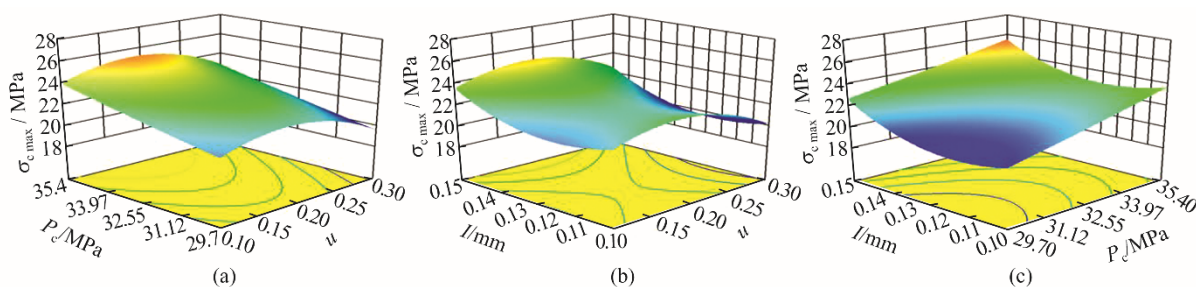


Fig. 12 Surface diagram of contact stress response of sealing surface with the maximum contact stress as the index under the interaction of preload force P_c , friction coefficient u and interference fit I

Table 1 Regression coefficients and analysis of variance for estimating the maximum contact stress at the sealing interface

Source	Free degree	Adj SS	Adj MS	<i>F</i> value	<i>P</i> -value
mold	10	28.142 4	2.814 2	44.43	0.001
block	1	0.000 0	0.000 0	1.50	0.233
linearity	3	22.013 4	7.337 8	53.89	0.000
<i>u</i>	1	0.001 9	0.001 9	77.42	0.000
<i>P_c</i>	1	17.335 9	17.335 0	41.61	0.002
<i>I</i>	1	4.602 4	4.602 4	89.31	0.003
square	3	0.000 3	0.000 1	4.26	0.682
<i>u</i> × <i>u</i>	1	0.000 1	0.000 1	2.10	0.197
<i>P_c</i> × <i>P_c</i>	1	0.000 1	0.000 1	2.21	0.187
<i>I</i> × <i>I</i>	1	0.000 2	0.000 2	9.77	0.020
two factor interaction	3	0.002 4	0.000 8	32.80	0.012
<i>u</i> × <i>P_c</i>	1	0.000 0	0.000 0	1.29	0.057
<i>u</i> × <i>I</i>	1	0.000 8	0.000 8	33.85	0.015
<i>P_c</i> × <i>I</i>	1	0.001 6	0.001 6	63.26	0.001
error	6	0.000 1	0.000 0	—	—
lack of fit	52	12.1470	2.412 5	0.73	0.612
pure error	2	0.000 0	0.000 0	0.73	0.612
summation	16	28.142 6	—	—	—

Note: Adj SS means adjusted sum of squares of deviation from mean; Adj MS means adjusted mean square

maximum contact stress of the sealing surface is 0.001, indicating that the model is significantly responsive to the response value. Since the lack of fit value is greater than 0.05, it is determined that there is no lack of fit phenomenon in this model.

The *P*-value represents the significant impact of the three factor parameters on the maximum contact stress in the model. A smaller *P*-value indicates a greater influence of the relevant factor parameters on contact stress. In this model, the individual effects of interference fit, preload force, and friction coefficient on the sealing sleeve are all significant. Regarding dual factor interaction, the most significant impact is on preload force and interference fit, while the *P*-values of friction coefficient and preload force, and friction coefficient and interference fit, are both greater than 0.05, indicating a lower significance for contact stress.

Figure 13 shows the maximum contact stress of the sealing surface of the residual map, the model residual spatial distribution is in the uniform distribution or normal distribution. From Fig. 13(a), it can be seen that all the residuals are almost distributed in the same straight

line, i. e., a linear relationship. The residuals approximately obey the normal distribution. The histogram of Fig. 13(c) is nearly bell-shaped, which conforms to the shape of normal distribution. From Fig. 13(b) and (d), the fitted value does not appear "trumpet-shaped", or "funnel-shaped", proving that the residual graph has no obvious defects, and the observed values arranged in chronological order fluctuate up and down, indicating that there is no dissimilarity with the fitted value and the order of the residuals. It is judged that the residuals are normal.

4.2 Response Surface Analysis Based on Maximum von Mises Stress

Analysis of the results of the response surface method using the maximum von Mises stress as an indicator shows that the combination of factors that have the greatest influence on the von Mises stress is the interference fit and friction coefficients. The inclination of the response surface is high, and the elliptic curve of the contour diagram is obvious, as shown in Fig. 14. The corresponding estimated regression coefficients are

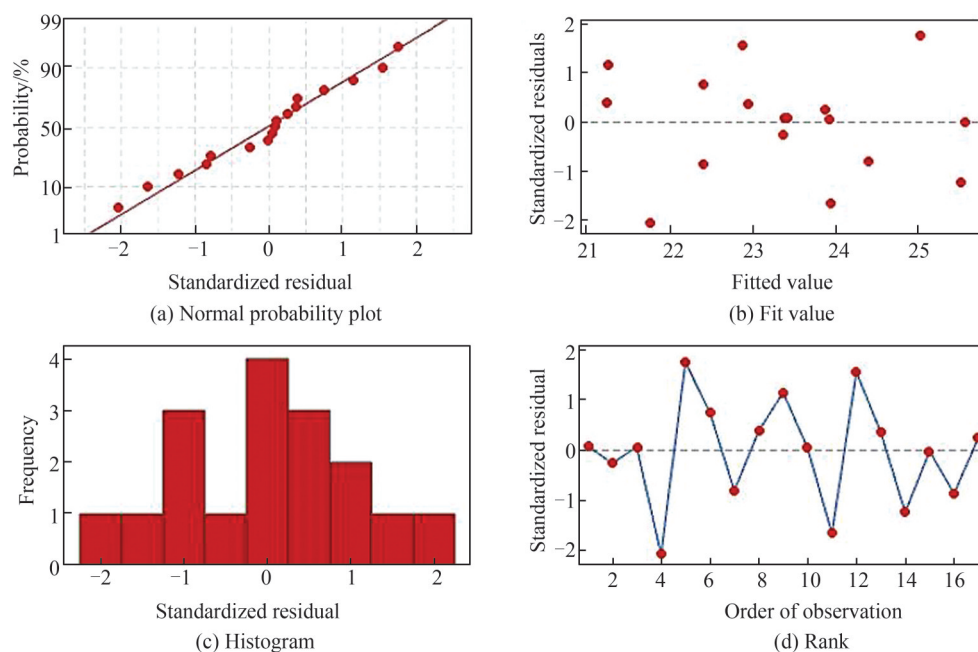


Fig. 13 Residual diagram of the maximum contact stress on the sealing surface

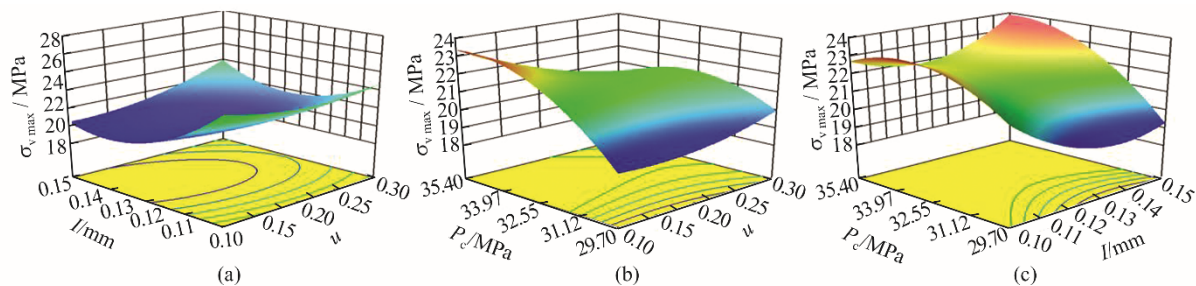


Fig. 14 Surface diagram of contact stress response of sealing surface with the maximum von Mises stress as the index under the interaction of preload force P_c , friction coefficient u and interference fit I

shown in Table 2. It can be seen that the model value (P -value) of the maximum von Mises stress of the sealing sleeve is 0, which is less than 0.05, indicating that the model is significantly responsive to the response value. At the same time, the maximum von Mises stress to lack of fit value is greater than 0.05, which is judged not to be a misfit.

In this model, the significance of all the individual factor effects on the sealing casing under the action of a single factor is high. In the two-factor interaction, the most significant influence was $u \times I$, and the P values of $u \times P_c$ and $P_c \times I$ were both greater than 0.05, which was consistent with the response surface results. Figure 15 shows the residual diagram of the maximum von Mises stress in the sealing sleeve.

Figure 15 shows the residual plots of the maximum von Mises stress of the sealing casing. In Fig. 15(a), we found that all the residuals are uniformly distributed in

the same straight line, that is, a linear relationship, and Fig. 15(c) is similar to a bell shape, in line with the morphology of the normal distribution. Fig. 15(b) and (d) show that the fitted value does not appear as "trumpet", and the observed values arranged in chronological order fluctuate up and down, that is, the fitted value and the order of the residuals are all normal, and the normal distribution performance is good.

4.3 Dual Objective Collaborative Optimization and Validation

With the different target parameters, there is a significant difference in the improvement results. In order to synthesize the results of design of experiment analysis under the maximum contact stress and maximum von Mises stress metrics, 15 sets of optimal solutions are predicted by response surface simulation analysis for the maximum contact stress (Table 3) and maximum von

Table 2 Estimated regression coefficients and analysis of variance for the maximum von Mises stress of seal sleeve

Source	Free degree	Adj SS	Adj MS	<i>F</i> value	<i>P</i> value
mold	10	30.525 6	3.052 6	52.98	0.000
block	1	0.000 0	0.000 0	0.04	0.852
linearity	3	26.921 3	8.973 8	63.13	0.071
<i>u</i>	1	0.000 3	0.000 3	24.94	0.000
<i>P_c</i>	1	26.90 8	26.908 0	16.96	0.000
<i>I</i>	1	0.012 4	0.012 4	2.58	0.000
square	3	0.000 0	0.000 0	26.36	0.001
<i>u</i> × <i>u</i>	1	0.000 0	0.000 0	77.28	0.115
<i>P_c</i> × <i>P_c</i>	1	0.000 0	0.000 0	0.07	0.804
<i>I</i> × <i>I</i>	1	0.000 0	0.000 0	0.00	0.987
two factor interaction	3	0.000 7	0.000 2	25.65	0.001
<i>u</i> × <i>P_c</i>	1	0.000 0	0.000 0	4.74	0.072
<i>u</i> × <i>I</i>	1	0.000 6	0.000 6	1.92	0.001
<i>P_c</i> × <i>I</i>	1	0.000 1	0.000 1	20.30	0.062
error	6	0.000 0	0.000 0	—	—
lack of fit	5	11.54 7	1.125 8	0.57	0.414
pure error	2	0.000 0	0.000 0	—	—
summation	16	30.525 6	—	—	—

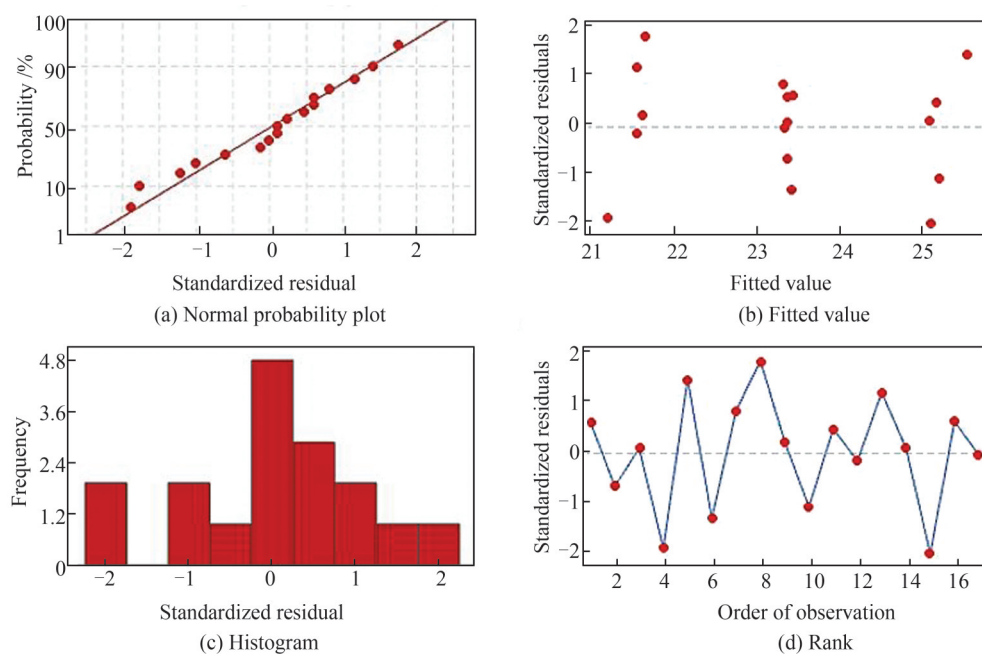


Fig.15 Residual plot of the maximum von Mises stress in the sealed casing

Mises stress (Table 4), respectively, under the interaction of each parameter.

The analysis shows that the optimal parameter values of the 7th group in Table 3 and the 8th group in

Table 4 are consistent, with a maximum contact stress of 19.711 2 MPa and a maximum von Mises stress of 18.927 9 MPa, a friction coefficient *u* of 0.22, an interference fit amount *I* of 0.113 mm, and a preload force *P_c*

Table 3 Parameter combination of an optimal solution for maximum contact stress

Number	u	l/mm	P_c/MPa	$\sigma_{c\max}/\text{MPa}$	Solvability
1	0.30	0.113	29.7	19.614 2	0.892
2	0.30	0.12	29.7	19.614 4	0.892
3	0.30	0.126	29.7	19.616 7	0.892
4	0.29	0.111	29.7	19.619 9	0.892
5	0.29	0.111	30.0	19.630 5	0.891
6	0.25	0.11	29.7	19.630 8	0.891
7	0.22	0.113	29.71	19.711 2	0.891
8	0.22	0.121	29.71	19.732 7	0.884
9	0.30	0.119	29.8	19.754 7	0.883
10	0.30	0.12	29.99	19.774 2	0.882
11	0.29	0.11	29.7	19.994 6	0.867
12	0.20	0.10	29.7	20.121 1	0.859
13	0.10	0.121	29.71	20.888 4	0.807
14	0.10	0.121	29.9	20.889 7	0.807
15	0.10	0.121	30.0	20.893 5	0.807

Table 4 Parameter combination of an optimal solution for maximum von Mises stress

Number	u	l/mm	P_c/MPa	$\sigma_{v\max}/\text{MPa}$	Solvability
1	0.13	0.13	29.7	18.810 2	0.909
2	0.14	0.13	29.7	18.813 3	0.908
3	0.15	0.135	29.7	18.813 6	0.908
4	0.16	0.135	29.7	18.822 2	0.906
5	0.17	0.135	29.7	18.850 2	0.901
6	0.17	0.14	29.71	18.879 1	0.895
7	0.19	0.14	29.71	18.891 4	0.873
8	0.22	0.113	29.71	18.927 9	0.871
9	0.22	0.113	30.0	19.105 8	0.858
10	0.27	0.11	34.9	19.127 6	0.848
11	0.28	0.11	34.92	19.130 2	0.847
12	0.29	0.12	35.1	19.138 2	0.846
13	0.30	0.121	35.4	19.171 8	0.839
14	0.30	0.124	35.4	19.180 7	0.837
15	0.30	0.126	35.4	19.254 8	0.823

of 29.71 MPa. To confirm the optimality of the parameters, a simulation analysis was conducted on the optimized sealing parameters using finite element analysis. The results showed that the maximum contact stress was 20.12 MPa and the maximum von Mises stress was 18.95 MPa, which were close to the predicted values. After optimization, under the same operating conditions, compared with those before optimization, the maximum contact stress and maximum von Mises stress of the sealing ring decreased by 26.3% and 20.6%, respectively.

In order to effectively verify the applicability of the optimized parameters, five groups of different working media pressures of 5, 6, 7, 8 and 9 MPa were selected to compare and analyze the corresponding maximum contact stress and maximum von Mises stress before and after optimization. As shown in Fig.16, the optimized contact stress and von Mises stress values are significantly reduced greater than 20% under the above five media pressures, which proves that the optimized parameters are effective.

5 Test Bench Verification

In order to test the effectiveness of the optimized design and the correctness of the performance analysis, this paper takes the piston rod Leningrader sealing experimental device for experimental verification, and the sealing device principle is shown in Fig. 17. The test device can complete the piston rod seal workpiece leakage detection test. The piston rod dynamic sealing device must ensure reliable gas sealing ability, in order to avoid material leakage brought by the engine power drop or shutdown. Therefore, the test device can simulate the Stirling engine working conditions, and, on this basis, realize the collection of leaking gas mass and its leakage measurement.

In the actual operation of the Stirling engine, the medium leaked by the piston rod sealing device enters the crankcase directly. Then it enters the oil and gas separator through a one-way valve for oil and gas separation, and is finally measured by a micro flowmeter. To ensure that the power of the Stirling engine is not reduced due to the medium leakage, the crankcase is generally pressurized. However, the sealing problem between the extended shaft and the housing when the motor is connected to the crankshaft has not been reliably solved, which will cause an incomplete collection of leaked media (hydrogen), and the low accuracy of the

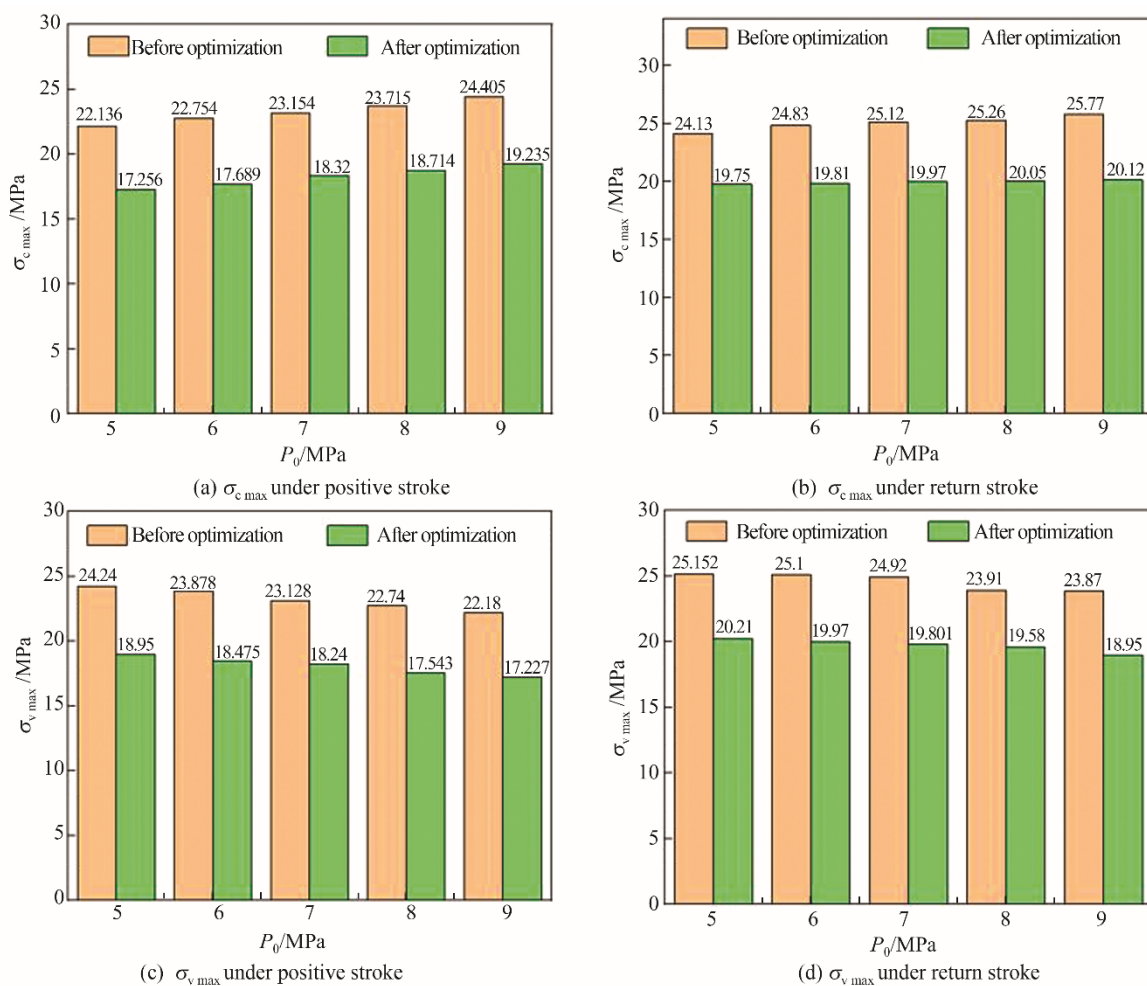


Fig. 16 Comparison of maximum contact stress and maximum von Mises stress before and after optimization under the five media pressures

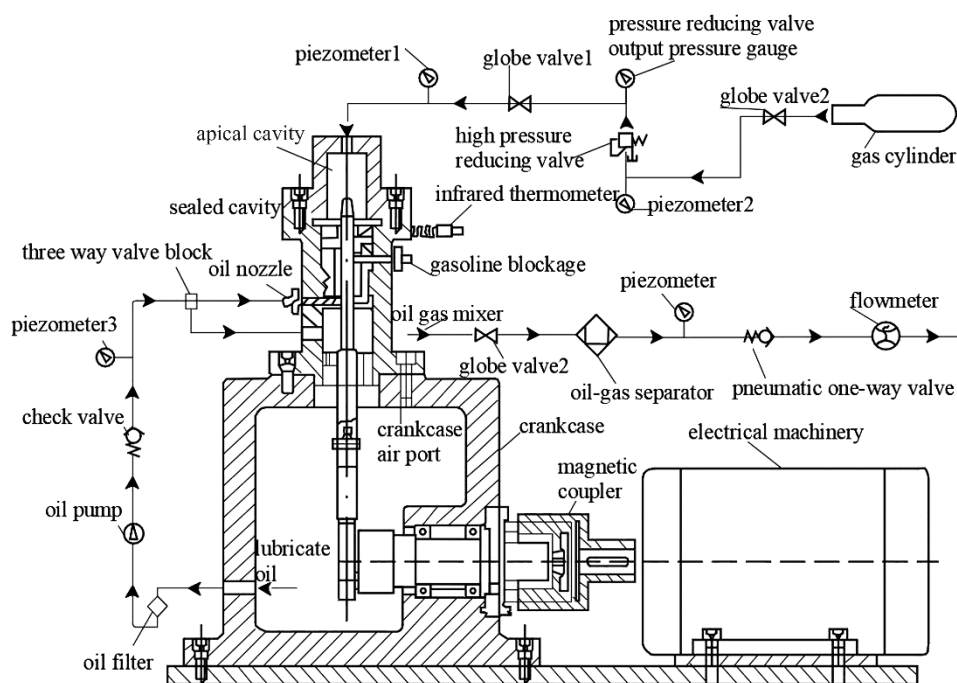


Fig. 17 Schematic diagram of the Leningrader piston rod sealing performance experiment

test. So, a magnetic drive has been utilized to ensure that the leaked medium through the rod sealing device is all concentrated in the crankcase without leaking outside. On this basis, the crankcase of the test bench is also filled with medium gas during the test, so as to eliminate the gas miscibility that may exist between the medium gas and the gas (non-medium gas) after the medium gas leaks into the crankcase. The test bench uses the same medium gas to ensure the sensitivity and accuracy of gas leakage measurement. In order to address the thermal equilibrium issue of the system, the measurement starting point of the test bench is set when the system reaches a (pressure, thermal) balance during operation. The experiment shows that the system takes ~3.25 h to reach the steady state from startup. In order to ensure the accuracy and reliability of the system's leakage measurement, the starting point of each measurement is set to 4 h after the start of operation, and the operation time is 12 h. The measurement records before and after optimization are shown in Table 5.

Table 5 Test record of leakage of sealing components before and after optimization

f/Hz	Top chamber air pressure P/MPa	Tank temperature $/^{\circ}\text{C}$	Leakage $/\text{cc}\cdot\text{min}^{-1}$	
			Before optimization	After optimization
10	3.5	36.9	0.12	0.11
	7.0	37.2	0.31	0.28
30	3.5	43.8	0.64	0.35
	7.0	44.2	2.1	1.67
50	3.5	48.3	1.33	0.87
	7.0	49.1	2.71	2.15

Note: f : rotational frequency; 1 cc=1 cm³

The experimental steps are as follows: 1) Open the switch of the hydrogen cylinder (10 MPa)→Open the pressure-reducing valve→Adjust the inlet pressure of the pressure-reducing valve to 10 MPa →Adjust the outlet pressure of the pressure-reducing valve to 7 MPa→Open the globe valve1→The piezometer1 displays that the input pressure P_0 in the top cavity is constant at 7 MPa→Close the globe valve1 to maintain pressure. 2) Hydrogen gas (P_0) from the top chamber leaks into the piston rod sealing chamber (P_1) through the Leningrader sealing element→The pressure drops from P_0 to P_1 →The lubricating oil flows into the oil line through the filter→The oil pump begins to work→The outlet oil pres-

sure of the oil pump rises to 0.1 MPa→The hydraulic control check valve opens (the minimum opening pressure is 0.1 MPa)→Piezometer3 shows that the oil pressure of the entire oil circuit is 0.1 MPa→The lubricating oil is divided by the three-way valve block and flows into the nozzle and the crankcase oil inlet, respectively →The split oil circuit lubricates the sealing sleeve and the brace ring respectively. 3) Start the electrical machinery and increase the operating frequency to the maximum→The rotor of the linear electrical machinery drives the magnetic coupling to start running at the maximum speed→The speed of the reciprocating motion of the piston rod increases from the minimum to the maximum→Attach an infrared thermometer to the corresponding position on the outer wall of the cylinder wall →Monitor and record the real-time temperature of the Leningrader seal. 4) When the temperature measured by the infrared thermometer is constant (the entire device's sealed hydrogen gas reaches thermal equilibrium)→Record the reading of piezometer1→Display it as the top chamber pressure P_0 (around 7 MPa). 5) The hydrogen gas in the piston rod sealing chamber (P_1) leaks into the crankcase through the sealing sleeve (P_2)→The pressure drops from P_1 (0.1-8 MPa) to P_2 →Open globe valve2→The oil and gas mixture flows out after being filtered and dried by the oil and gas separator→When the pressure of the dry hydrogen gas flowing through piezometer2 reaches 0.14 MPa→The pneumatic one-way valve (minimum opening pressure difference of 0.035 MPa) is opened→The pressure difference between the two ends of the flowmeter is 0.035 MPa (the inlet pressure of the flowmeter is 0.135 MPa, and the outlet pressure is atmospheric)→The flowmeter valve (minimum opening pressure difference of 0.035 MPa) is opened→Piezometer2 displays the pressure of the entire gas circuit as 0.035 MPa. 6) When the temperature measured by the infrared thermometer is constant→Piezometer2 begins to record the pressure reading (MPa) →The flow meter begins to count (SCCM, under standard conditions mL·min⁻¹)→Record the instantaneous flow rate and cumulative flow rate of hydrogen gas flowing through the flow meter→When the hydrogen gas is completely leaked and the reading on piezometer2 is 0, stop counting→Close globe valve2.

As shown in Table 5, when the rotational frequency is 10 Hz, the difference in leakage before and after optimization is small, and the optimization effect is insignificant, indicating low wear and leakage at the low rota-

tional frequency stage. As the speed increases, the frequency of seal surface wear increases and the wear loss increases. The difference in leakage between before and after optimization increases. At a speed of 50 Hz, with a top chamber pressure of 7 MPa and an oil tank temperature of 49.1 °C, the maximum difference in leakage is 0.56 cc·min⁻¹. When the limited rotational frequency is 30 Hz, the air pressure in the head chamber is 7 MPa, and the temperature of the tank is 44.2 °C. The variation curves of the leakage volume and the wear volume of the system during 12 h of operation are shown in Fig. 18, and the weighing method was chosen to obtain its wear loss.

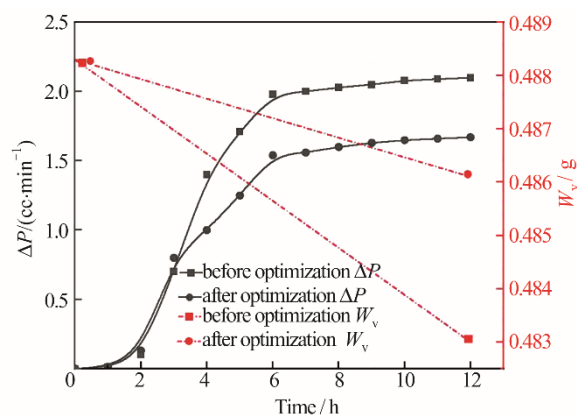


Fig. 18 Curve chart of leakage and wear before and after optimization

To ensure the weighing accuracy, the test specimens were cleaned and dried before weighing. The experiment was then started, and after completion, the above steps were repeated to obtain the wear loss. A comparison of the wear loss before and after optimization is shown in Table 6. When the piston rod starts reciprocating (0-1 h), due to the large interference fit between the sealing sleeve and the piston rod during assembly, the radial compression stress caused by the interference fit is much larger than the hydrogen pressure, which ensures that there is basically no leakage on the sealing surface. After 1-2 h of operation of the test rig, due to the wear of the sealing sleeve, the radial interference fit decreases, causing the contact stress on the sealing surface to be less than the hydrogen pressure, resulting in slow leakage. Currently, the leakage of the optimized sealing structure is greater than that before optimization. After running for 3 h, the wear of the sealing sleeve increases, and the leakage of the optimized sealing structure starts to become less than that before opti-

Table 6 Wear amount of the sealing sleeve before and after optimization

Sealing sleeve	Mass before wear /g	Mass after wear /g	Wear rate /%
Before optimization	0.488 3	0.483 0	1.08
After optimization	0.488 3	0.486 1	0.45

mization. After running for 6 h, due to the continuous increase in wear, the film thickness ratio between the sealing sleeve and the piston rod surface increases. At this point, the lubrication state of the sealing sleeve changes from mixed lubrication to elastohydrodynamic lubrication, and the oil film thickness increases, slowing down the leakage rate of hydrogen. When the running time reaches 10 h, at this time, the sealing sleeve reaches the stable wear stage in the state of oil lubrication, the check valve discharges the same amount of leaking hydrogen, and the entire crankcase is in equilibrium.

In the experiment, the measured leakage before the optimization is 2.1 cc·min⁻¹, the optimized leakage is 1.67 cc·min⁻¹, and the leakage difference is 0.44 cc·min⁻¹. According to Table 6, the wear rate of the sealing sleeve before optimization was 1.08%, which was reduced to 0.45% after optimization. It is proved that the finite element analysis results are correct, and the sealing performance and service life of seals are enhanced under this parameter, and the optimization parameters are effective.

6 Conclusion

Taking the maximum contact stress and maximum von Mises stress in the Leningrader seal of the Stirling engine as the performance and service life evaluation indicators of the seal sleeve, a set of optimal parameters was obtained through response surface optimization, with the interference fit I of 0.113 mm, preload force P_c of 29.71 MPa, and friction coefficient u of 0.22 as parameters.

After optimization, the maximum contact stress was 20.12 MPa with a decrease of 26.3%, and the maximum von Mises stress was 18.95 MPa, a decrease of 20.5%. The applicability of the optimization parameters was verified by using a medium pressure of 5-9 MPa, proving that the optimization is effective at 5-9 MPa and the decrease is greater than 20%.

The experimental results show that with leakage

and wear as the core, the simulation results are verified by running for 12 h. After the optimization, the leakage of the seal structure is $1.67 \text{ cc} \cdot \text{min}^{-1}$, $0.44 \text{ cc} \cdot \text{min}^{-1}$ less than that before the optimization, and the wear rate after the optimization is 0.45%, 0.63% less than that before the optimization, which proves that the optimization parameters are effective.

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