Research Article

Guoping Yan, Lan Xiao, Fei Zhong*, and Weiguo Lin

Optimum research on the temperature of the ship stern-shaft mechanical seal end faces based on finite element coupled analysis

<https://doi.org/10.1515/nleng-2021-0041>

Received Aug 04, 2021; accepted Oct 27, 2021

Abstract: Due to the complicated working conditions of the ship stern-shaft mechanical seals, it is very difficult to evaluate and optimize the temperature on the sealed end faces. In the paper, one FEA model of the small taper convergent stern-shaft mechanical seals was put forward, and the nonlinear end-face liquid film pressure distribution was derived from different end-face gaps based on the simplified Reynolds equation as the end-face input parameters in FEA model. To prove the correction of the FEA results, one experiment was designed and completed. At the same time, combined with a parametrically coupled FEA model and an orthogonal experimental design, one PSO-BP-GA method based on the combination of particle swarm optimization (PSO), BP neural network and genetic algorithm (GA) was proposed to optimize the temperature of the sealed end faces. The results show that aiming at the prediction accuracy of the temperature on the sealed end faces, the PSO-BP algorithm is higher than BP algorithm and GA-BP algorithm, which error interval is shortened about 40% and 29%, respectively. Subsequently, PSO-BP-GA algorithm is obviously better than BP-GA algorithm and GA-BP-GA algorithm, which validates its effectiveness and feasibility. It will help to lay a theoretical foundation for temperature evaluation on the sealed end faces.

Keywords: stern-shaft mechanical seals; coupled field analysis; temperature optimization; PSO-BP-GA combined method; accuracy

1 Introduction

As one part of the ship's power unit, the stern-shaft mechanical seals device mainly plays a sealing role when sailing in a certain depth of sea conditions, and its stress is more complicated. On one hand, with the effect of the combined force, the sealed end faces will be deformed, and uneven contact high-pressure areas will be formed locally. On the other hand, the frictional heat generated by the relative rotation of the rotating ring and stationary ring will cause the temperature to rise on the sealed end faces. Subsequently, the thermal gradient can generate thermal deformation and thermal stress on the sealed end faces, which in turn causes the contact area of the sealed end faces to change. As a result, the tendency of local high temperature and high stress will appear on the sealed end faces. This is a strongly coupled cyclic process. If the thermal stress of the seal rings is too large, it will cause thermal cracking of the sealed end faces, which will affect the using life and safety of the mechanical seals. Therefore, it is necessary to analyze the coupled field of the mechanical seals. With the development of mechanical seals in the direction of large size and high load, the requirements for the excellent performance of mechanical seals are increasing. How to obtain the optimal mechanical seals performance under severe working conditions has always been a big problem that plagues engineering practice.

A lot of currently useful research have been done on the coupled field analysis of mechanical seals. Xin Zhou *et al.* [1] considered a modified W-M fractal function and adopted a three-dimensional rotating sliding contact model to study transient dry friction characteristics of a mechanical seal. During the simulation process, software ABAQUS was applied to the analysis of the mechanical action and thermal-mechanical coupling. Warda *et al.* [2] conducted a numerical study to simulate the performance of barrier fluid flow through a bi-directional dual mechanical seals with an integral radial pumping ring. In the simulation, standard K- ϵ model was implemented and a multi-block hybrid mesh was used. Meng Xiangkai *et al.* [3], Tournerie

*Corresponding Author: **Fei Zhong:** School of Mechanical Engineering, Hubei University of Technology, Wuhan, 430068, China, Email: iamypg@163.com

Guoping Yan: School of Mechanical Engineering, Hubei University of Technology, Wuhan, 430068, China

Lan Xiao: School of Economics, Wuhan Institute of Shipbuilding Technology, Wuhan, 430050, China

Weiguo Lin: School of Mechanical Engineering, Huazhong Agricultural University, Wuhan, 430070, China

et al. [4] and Gao Binchao *et al.* [5] established a coupled mechanical model for evaluating the steady state of mechanical seals. Yan Guoping *et al.* [6, 7] and Zhou *et al.* [8] analyzed the steady-state coupled field of the ship stern-shaft mechanical seals. Gu Boqin [9] and Peng Xudong [10] analyzed the coupled conditions of the frictional heat of the liquid film and the thermal deformation on the sealed end faces. Srivastava *et al.* [11] studied the thermoelastic dynamic pressure characteristics of the reactor cooling pump seals. Wang Jihui *et al.* [12] set up a 3D transient sliding contact model for the contact mechanical seals according to the W-M fractal function. However, the present research about the optimization of conventional mechanical seals mostly focuses on processing technology or improvement of material performance of the seals. For example, Xiangkai Meng *et al.* [13] performed numerical analysis on the slotted mechanical seals performance. Zhao Shuai *et al.* [14] and Liu Xukang *et al.* [15] made a comparative study on the material properties of seal rings. Mosavat *et al.* [16] used configuration and numerical methods to carry out temperature conduction on the sealed end faces of mechanical sealing equipment with cooling fin-shape under different working conditions. Blasiak *et al.* [17–19] used numerical methods to study the characteristics of the thermal conductivity parameters of the non-contact mechanical sealed end faces. Thomas *et al.* [20] presented a numerical modeling of thermo-elasto-hydro-dynamic mechanical face seals behavior which was axisymmetric to high pressure compressible flow. Zhu *et al.* [21], Zhou *et al.* [22] and Gu *et al.* [23] had carried out some research about the coupled effect of mechanical seals with one FEA method. Anoop *et al.* [24] made a novel mechanical seal's design with superior thermal characteristics in 2004.

On the other hand, some research of performance of mechanical seals was optimized. Luo *et al.* [25] established the rotating ring-fluid film-stationary ring 3D model to analyze the dynamic characteristics of mechanical seals in the specific extrusion fault, its feasibility and effectiveness of the new analysis method was proved. Guo *et al.* [26] prepared polycrystalline diamond (PCD)-coated mechanical seal rings by hot filament chemical vapor deposition (HFCVD) on graphite-loaded silicon carbide (GSiC) substrates, and an extremely low and stable friction coefficient and super mechanical properties under harsh conditions could be approved as the source of the high PV limit of PCD coating for more demanding applications. Based on the TRIZ theory, Heshun *et al.* [27] chose two improving parameters and deteriorating elements to analyze the problems of poor stability and large leakage of dynamic pressure non-contact mechanical seals in a specific working environment. Guiyue Kou *et al.* [28] established a two-dimensional

axisymmetric thermal-fluid-solid coupled mathematical model of a contact mechanical seal. During the process, the finite difference method was used to solve the control equations for the fluid pressure and temperature of the seal end faces, and the finite element method was used to determine the thermal deformation state of the seals. All these could be applied for mechanical seal design and optimization. Chen *et al.* [29] proposed a model of the reliability estimation for mechanical seals based on bivariate dependence analysis. According to the model, friction torque and leakage rate were two degradation performance indicators.

In view of the unique large size and high pressure of the ship stern-shaft mechanical seals, the paper adopted an orthogonal experimental design and ANSYS parametric simulation technology in order to establish one strong non-linear overall mapping relation between the given design variables and objective function. The relation could be obtained by means of BP neural network method (BP) according to the steady-state coupled field analysis of the ship stern-shaft mechanical seals. Meantime, for a more efficient and accurate non-linear global mapping model, particle swarm optimization (PSO) was used to optimize some parameters in the BP neural network prediction. Finally, the genetic algorithm (GA) was used to perform an optimal search in the global scope based on the prediction results of the optimized BP neural network. As a result, it can find the global optimum within the scope of design variables and objective function in order to achieve optimization purposes. All this proposes a stern-shaft mechanical seals optimization design method based on PSO-BP-GA method, which can quickly obtain the critical dimensions of the sealed end faces to verify the feasibility of the optimization method.

2 Optimized design method of mechanical seals based on PSO-BP-GA

For the mechanical seals optimization problem with multiple design variables, particularly in the prediction of the nonlinear mapping relationship between design variables and target values, traditional BP neural network algorithm is usually used to adjust network connection weights based on gradient descent and backward propagation of errors [30], so the prediction is often easier to fall into the local optimal solution. To solve this problem, the particle swarm optimization method can be used to optimize the weights and thresholds of the neural network and search

Figure 2: Map of the ship stern-shaft mechanical model

The coupled field calculation model and boundary conditions are described as follows: The pressure of the seawater medium is respectively applied on the surfaces presented by ABCDERFLMNPS, and the seawater temperature is respectively applied on the surfaces presented by ABC and NPST, respectively. Simultaneously, the spring pressure is directly applied on the surface represented by PS, and the convective heat transfer boundary conditions are respectively applied on the surfaces represented by CDERFL, LMN, AI, and IHWGKU according to the specific working environments [6]. The corresponding boundary heat transfer coefficients are respectively shown in Eqs. (1)–(3), respectively [7, 8].

$$\begin{cases} \alpha_{ci} = \frac{\lambda'}{D} \cdot 0.135 \cdot [(0.5 \cdot Re_c^2 + Re_{ef}^2) \cdot P_r]^{0.33} \\ Re_c = \frac{\omega \cdot D^2}{\nu_l^2}, Re_{ef} = \frac{V \cdot D}{\nu_l} \end{cases} \quad (1)$$

$$\alpha_{ha} = 0.023 \cdot \left(\frac{2V \cdot S_{dz}}{\nu_l} \right)^{0.8} \cdot P_r^{0.4} \quad (2)$$

$$\alpha_{id} = \epsilon \cdot \frac{\lambda'}{D} \cdot 0.135 \cdot \left(\frac{V \cdot D}{\nu_l} \right)^{0.66} \cdot P_r^{0.33} \quad (3)$$

where, α_{ci} , α_{ha} , α_{id} is the convective heat transfer coefficient of CDERF, IA, and FLMN sections respectively; λ' is the thermal conductivity of the fluid medium; D is the diameter of the seals ring; Re_c is the Reynolds number which can reflect the influence caused by the rotary stirring of fluid medium; P_r is Planck constant; Re_{ef} is the Reynolds number which can reflect the influence caused by the lateral flow of the fluid medium, ν_l is the kinematic viscosity of the fluid; V is the axial flow velocity of the fluid medium; S_{dz} is the gap between the rotating ring inside and the marine stern-shaft; ϵ is the correction factor considering the stirring heat.

Corresponding constraints are imposed on the joints of the axial retaining ring and the seat body, and liquid film

pressure and the heat flux density generated by friction between the two sealed rings are applied to the nodes on the sealed end faces. Here, the heat flow density on the sealed friction surfaces is related to the contact pressure and rotating speed, which is obtained by Eq. (4) [7, 8].

$$q(i) = f \cdot p(r) \cdot \omega \cdot r(i) \quad (4)$$

where, f is the friction coefficient of the sealed end faces; $p(r)$ is the contact pressure of a certain node on the sealed end faces with radius r ; ω is the angular velocity of the mechanical seals rotating ring; $r(i)$ is radial coordinates of a certain node on the sealed end faces.

Because of the strong coupled process between the successive calculations, the heat flow density is iteratively self-corrected for a higher accuracy according to Eq. (5).

$$\max_{i=1,n} \left| \frac{q(i, k) - q(i, k+1)}{q(i, k+1)} \right| \leq \epsilon'' \quad q(i, k+1) \neq 0 \quad (5)$$

where, $q(i, k)$ is the calculated heat flux density of node i on the sealed end faces after calculation of k times, $q(i, k+1)$ the calculated heat flux density of node i on the sealed end faces after calculation of $k+1$ times, ϵ'' is the given iterative convergence tolerance.

In addition, due to the seawater environment, one thin layer of seawater liquid film is attached to the small opening gap between the sealed end faces. Considering its influence on the characteristics of the sealed end faces, the liquid film pressure in a linear manner is applied on the sealed end faces in reference [6, 15]. In fact, this linear loading method does not reflect the influence of the deformation of the sealed end faces on the liquid film distribution during the coupling process. In the paper, the actual contact deformation of the sealed end faces is profoundly considered. As a respond, liquid film pressure in a non-linear manner derived from the simplified Reynolds equation according to the opening gap of the sealed end faces is applied to the corresponding nodes on the sealed end face, as shown in Eq. (6) [6, 7].

$$\begin{cases} p_{ymp} = \frac{(y-r_i)(p_o-p_i)}{r_o-r_i} + p_i \\ p_{ymc} = p_i + \frac{[h_{min}+(r_o-r_i)\tan\beta]^2(y-r_i)}{(r_o-r_i)[h_{min}+(y-r_i)\tan\beta]^2} \\ \quad \times \frac{[2h_{min}+(y-r_i)\tan\beta](p_o-p_i)}{[2h_{min}+(r_o-r_i)\tan\beta]} \\ p_{ymd} = p_i - \frac{(y-r_i)h_{min}^2}{(r_o-r_i)[2h_{min}+(r_o-r_i)\tan\beta]} \\ \quad \times \frac{[r_i+y-2r_o]\tan\beta-2h_{min}}{[h_{min}+(r_o-y)\tan\beta]^2}(p_o-p_i) \end{cases} \quad (6)$$

where, on the sealed end faces, h_{min} is the minimum film thickness; y is the radial coordination, $r_i \leq y \leq r_o$; β is the angle between the sealed end faces; p_{ymp} , p_{ymc} and p_{ymd} are the distribution function of the film pressure respectively when the gap is parallel, convergent and divergent; p_i and

p_o are the pressure value respectively on the outboard part and inboard part; r_i and r_o are the radial coordination respectively on the outboard part and inboard part.

Based on the calculation result, for the convergence gap, the liquid film pressure is distributed nonlinearly along the radial direction of the sealed end faces. When the minimum liquid film thickness is the same, the liquid film pressure rises with the increase of the cone angle of the sealed end faces. When the end cone angle is the same, the liquid film pressure decreases with the increase of the minimum liquid film thickness. Meantime, a self-correction is also performed continuously according to the results of each iteration in the calculation, which is illustrated in Eq. (7) until the coupled field calculation is completed.

$$\sum_{k=1}^m \sum_{i=1}^n \left| \frac{p(i, k+1) - p(i, k)}{\sum_{k=1}^m \sum_{i=1}^n |p(i, k+1)|} \right| \leq \varepsilon' \quad (7)$$

where, n is the node number on the sealed end faces, k is the calculation number of iteration convergence, ε' is the given iterative convergence tolerance, $p(i, k)$ is the liquid film pressure of node i on the sealed end faces after calculation of k times, $p(i, k+1)$ is the liquid film pressure of node i on the sealed end faces after calculation of $k+1$ times.

All boundary conditions for calculation are shown in Figure 3. Considering the medium distribution and heat exchange environment around the ship stern-shaft mechanical seals, the calculation boundary conditions around the mechanical seal are as follows: S1 and S4 represent medium pressure and temperature from the seawater, S2 means medium pressure from the seawater and convective heat transfer boundary conditions around the rotation ring, S3 means medium pressure from the seawater and convective heat transfer boundary conditions around the stationary ring. S5 means spring pressure. S6 means inside temperature. S7 and S8 mean convective heat transfer boundary conditions around the rotating ring. The finite element analysis meshing of the sealed end faces is shown in Figure 4.

According to some basic calculation steps, some boundary conditions are set to the FEA model of the mechanical seals by means of some script codes. During the calculation, iteration judgements are also realized by some programming programs and final results are shown in Figures 5–6, where seawater pressure is 0.60MPa and the rotating speed is 200r/min , respectively. Due to the difference between the medium pressure and the thermal boundary conditions, under the thermal coupling effect, the contact area and high temperature area of the sealed end faces are distributed on the inner side, and the outer contact pressure is 0. All of these indicate that convergent taper deformation exits on

Figure 3: Boundary conditions for calculation

Figure 4: Meshing map of sealed rings

Figure 5: Contact pressure result

Figure 6: Temperature result

the sealed end faces, which will contribute to the redistribution of the liquid film pressure on the sealed end faces.

3.2 FEA results test

Compared with the above FEA results, one experiment was designed to test the results, which can be used to prove the correction of the FEA results. In this paper, one self-developed multifaceted mechanical seals equipment was

Figure 7: Mechanical seals test device

Figure 8: Temperature measurement map

Figure 9: Temperature sensors distribution

designed, as shown in Figures 7–9. In the previous simulation part, a specific example was given to show the feasibility of the finite element method in this paper. In the following test part, a mechanical seal device provided by a manufacturer that met the Chinese standard was simulated again by using the above simulation method and tested, so as to ensure the consistency of model parameters between the simulation and test. The model structure in Figure 2 is slightly different from that in Figure 7 only because of the spring installation position, and this will have no effect on the analysis. In test analysis model, the spring force will be applied to the back of the rotating ring and other calculation boundary conditions do not need to be changed.

The test device included the following parts: (1) a self-designed controller, (2) a servo motor (IP67, 0–3000r/min, 380V), (3) two couplings (aluminum alloy 7075), (4) a reducer (transmission ratio is 3), (5) a mechanical seals device around the rotating shaft (medium tank volume: 8L, seals type: 105 type with 60mm rotating shaft diameter (Chinese general specification of mechanical seals, GB/T33509-2017 and JB/T1472-2011, $r_A = 30$ mm, $r_G = 31.5$ mm, $r_F = 35$ mm, $r_M = 40$ mm, r_A , r_G , r_F , r_M are the distances from points A, G, F, M to the center of the rotation axis as shown

in Figure 2, respectively), 51 is the spring, 52 is the rotating ring, 53 is the stationary ring, 54 is the rotating shaft), (6) a pair connection pipe system, (7) a pressurizing device (0–3MPa, precision: $\pm 0.5\%$), (8) three temperature sensors (MBT5250, -10 – 100°C , precision: $\pm 0.5\%$), uniformly distributed on a stationary ring) are arranged on the circumference with a radius of 32.5mm and 2mm away from the sealed contact end faces, as shown in Figure 9. Here, 1 represents sensor one, 2 represents sensor two, 3 represents sensor three, 4 represents the stationary ring, 5 represents the rotating ring base, 6 represents the rotating ring base support, 7 represents the rotating shaft, 8 represents the supporting bear, 9 represents the supporting bear base, (9) a transmission shaft.

During this experiment, the medium temperature in the sealed tank was 20°C and the rotating speed of the shaft is 200r/min, the medium pressure is from 0 MPa to 0.8MPa. Every test time was 30 minutes, and each working condition parameter was tested three times. In order to reduce the measurement error, the average test results values were taken. When the test started, the test parameters such as medium pressure and shaft speed were set. Some variable frequency control operations were performed to achieve

speed requirement on the console of the self-designed controller and one valve in the pressurizing device was used to adjust the medium pressure. When the specified test requirements were met, the operation was kept moving for a period of time, and then started recording the test results.

A comparison between the FEA simulation results and the test results for the temperature on the stationary ring end face is shown in Figure 10. It indicates that the trend of the test values is consistent with the FEA simulation results and the maximum error is within 10%. This can be used to prove that the FEA model of mechanical seals coupling analysis in the paper is reasonable, and this lays a good theoretical foundation for the subsequent parameter optimization of mechanical seals as well.

Figure 10: Temperature measurement curve

3.3 Mathematical model and orthogonal test design of the sealed end faces temperature optimization

There are many factors which have an effect on the temperature of mechanical sealed end faces, such as material, structure, size, etc. In the article, the structural size optimization will be considered, and the influence of materials on the temperature of sealed end faces will be further discussed in future research. As can be seen from Figures 5–6, the sealed end faces characteristics are an important reflection of the mechanical seals performance. Some important geometries such as protruding length l_d (ER as shown in Figure 2) of the rotating ring, protruding length l_j (FL as shown in Figure 2) of the stationary ring and width b (FG as shown in Figure 2) determine the geometry size of the sealed end faces and they can have a direct impact on the performance

such as heat dispelling and load. Therefore, l_d , l_j and b can be seen as the design variables during the optimization process in order to achieve the best optimized parameters when other parameters are not changed. In addition, it can provide a theoretical guidance for the engineering design. The optimization mathematical model is described as in Eq. (8).

$$\left\{ \begin{array}{l} \text{objective function :} \\ \min f(x) = T_{\max} \\ \text{design variables :} \\ j_1 \leq l_j \leq j_2; \\ d_1 \leq l_d \leq d_2; \\ b_1 \leq b \leq b_2 \\ \text{state variables :} \\ g_{\max} \leq 1\mu\text{m}; \\ p_w = 0; \\ p_{\max} \leq [p_{\max}]; \\ \sigma_{\max} \leq [\sigma_{\max}] \end{array} \right. \quad (8)$$

where, $T_{\max}$ is local maximum temperature on the sealed end faces; j_1 and j_2 are the minimum and maximum of the protruding length of the stationary ring, respectively; d_1 and d_2 are the minimum and maximum of the protruding length of the rotating ring, respectively; b_1 and b_2 are the minimum and maximum of the width of sealed end faces, respectively; $g_{\max}$ is the biggest opening gap and p_w is the most lateral contact pressure on the sealed end faces; $p_{\max}$ is the maximum contact pressure and $\sigma_{\max}$ is the largest von mises stress on the sealed end faces.

In general, the quality and quantity of training samples have a great impact on the prediction accuracy of BP neural networks. At this time, the orthogonal experiment design can determine a certain number of the typical training samples and improve the accuracy of BP neural network prediction. Based on one generic orthogonal table, five levels of each factor are taken and orthogonal design on each design variable is performed to obtain a total of 25 test groups listed in Tables 1–2. During the calculation, the rotating speed is 150 r/min, the medium pressure is 1.2 Mpa.

Table 1: Orthogonal experimentation

level	factor		
	l_j/mm	l_d/mm	b/mm
1	4	1.2	9
2	4.2	1.4	9.5
3	4.4	1.6	10
4	4.6	1.8	10.5
5	4.8	2.0	11

Table 2: Training samples and results

Index	l_j /mm	l_d /mm	b /mm	FEA $T_{max}/^{\circ}\text{C}$	Index	l_j /mm	l_d /mm	b /mm	FEA $T_{max}/^{\circ}\text{C}$
1	4	1.2	9	29.6593	14	4.4	1.8	9	29.453
2	4	1.4	11	36.0999	15	4.4	2	11	35.9218
3	4	1.6	10.5	34.6836	16	4.6	1.2	10	31.7931
4	4	1.8	10	32.3233	17	4.6	1.4	9.5	30.5086
5	4	2	9.5	30.8984	18	4.6	1.6	9	29.3484
6	4.2	1.2	11	35.9557	19	4.6	1.8	11	34.6711
7	4.2	1.4	10.5	33.6522	20	4.6	2	10.5	33.1653
8	4.2	1.6	10	32.1196	21	4.8	1.2	9.5	30.3973
9	4.2	1.8	9.5	30.7751	22	4.8	1.4	9	29.2594
10	4.2	2	9	29.5522	23	4.8	1.6	11	34.4633
11	4.4	1.2	10.5	33.3288	24	4.8	1.8	10.5	32.9894
12	4.4	1.4	10	31.9460	25	4.8	2	10	31.5981
13	4.4	1.6	9.5	30.6398					

3.4 Specific implementation of PSO-BP-GA

According to the results obtained by the orthogonal test, the design variable samples are sequentially used to analyze the steady-state coupled field of the mechanical seals for the corresponding input data. The results are shown in Table 2. Firstly, the samples matrix of BP neural network is listed, and a three-layer grid structure is initialized. In the BP neural network, there are 3 neurons, 3 neurons and 1 neuron in the input layer, the hidden layer, and the output layer, respectively. At the same time, the ‘tansig’ transfer function is used in the hidden layer and the ‘purelin’ function is used in the output layer. The total training step size is 1000 steps, and the mean square error is 10^{-4} . Subsequently, PSO is used to optimize the weights and thresholds of the BP neural network, and then the optimized weights and thresholds are given back to the network for back-propagation correction, thereby a highly accurate BP neural network is obtained. In the PSO algorithm, some parameters need to be listed such as maximum particle velocity, goal error, particle swarm size and velocity coefficients, which are 100, 10^{-4} , 30 and 1.49, respectively. Finally, the optimization results are shown in Figures 11 and 12. As can be seen from Figure 11, under given computational conditions, the PSO algorithm reaches convergence in about 75 generations. In Figure 12, the prediction accuracy of the BP neural network is obviously lower, and its prediction absolute error interval is from -0.75 to 0.2 , while the prediction accuracy of the PSO-BP neural network is higher, and its prediction absolute error interval is relatively stable which is from around -0.35 to 0.22 . All these show that the BP neural network optimized by PSO has higher accuracy and

can be moderately used in the optimization design of this paper. Meantime, a GA-BP algorithm is also completed to compare the accuracy of the PSO-BP algorithm. The process of GA-BP algorithm is similar to that of PSO-BP algorithm. The only difference is that the former uses GA method to optimize BP weight instead of PSO method, and the specific process is what the optimization of GA shows. Obviously, the prediction absolute error interval in GA-BP algorithm is from -0.4 to 0.38 . From the above, it can be drawn that the prediction accuracy of GA-BP is higher than that of BP and lower than that of PSO-BP. Eventually, the error interval predicted by PSO-BP is shortened by about 40% and 29% compared to that predicted by traditional BP algorithm and GA-BP algorithm, respectively.

Figure 11: PSO-BP fitness curve

Figure 12: Prediction error curve

Figure 13: Genetic iterative process

The predicted values mapped by the optimized BP neural network can be used as the objective function required by the genetic algorithm which can research the optimum parameters within the range of global design variables. The parameters in the genetic algorithm are set as follows: the number of individuals in the population is 20, the maximum number of genetic generations is 100, the generation gap is 0.9, and the cross probability is 0.2. To avoid design variables exceeding the sample values given by BP neural network, the range of design optimization variables l_d , l_j and b are taken as $[4.2, 4.6]$, $[1.5, 1.8]$ and $[9.5, 10.5]$, respectively, and one optimization is carried out by means of three separately different algorithms. The results are shown in Figure 13. In addition, one first-order optimized finite element analysis is used to compare the above results obtained by the three different methods. The results are still shown in Table 3.

It can be seen from Figure 13 that the convergence generations in the three different methods can be reached at the 58th, 27th, and 63rd generation, respectively. At this time, the optimal solution of the local maximum temperature of the sealed end faces is basically found. In subsequent genetics, the optimized objective function solution is hardly changed. As can be seen from Table 3, three opti-

mal design variables obtained by the PSO-BP-GA algorithm are 4.5217, 1.7677, and 9.5044, respectively, and the error is 0.8% compared with FEA optimum result. The three optimal design variables obtained by the BP-GA algorithm are 4.4867, 1.5182 and 9.5027, respectively, and the error is 1.04% compared with FEA optimum result. The three optimal design variables obtained by the GA-BP-GA algorithm are 4.5391, 1.6559 and 9.5055, respectively, and the error is 1.93% compared with FEA optimum result. From the calculation results, the errors of the three results are all within the acceptable range of the project (within 2%), which indicates that it is feasible to adopt BP neural network prediction and use optimization algorithms to search the global optimal design variables for engineering application. In addition, as far as the result accuracy is concerned, since PSO-BP has one higher accuracy for the prediction result of the BP neural network, its optimization result is improved by about 20% and 50% in details, respectively, compared with BP algorithm and BP-GA algorithm. It can be also seen that the prediction accuracy of the previous BP neural network has a certain influence on the accuracy of the optimization result especially for this case. On the other hand, because related algorithms have a certain limitation and volatility, they sometimes converge to local solving

Table 3: Results comparison

	l_j /mm	l_d /mm	b /mm	T /°C	Error %
PSO-BP-GA	4.5217	1.7677	9.5044	30.2382	0.8
BP-GA	4.4867	1.5182	9.5027	30.2882	1.04
GA-BP-GA	4.5391	1.6559	9.5055	30.5524	1.93
FEA Optimum	4.5958	1.6895	9.5290	29.9743	

range or lead to the inability to obtain ideal optimized values, which require multiple adjustments and comparisons during the global optimization process. Therefore, it will be the focus to vigorously improve the prediction accuracy of the algorithm and try to avoid the volatility of the algorithm for the similar optimization research.

4 Conclusion

With the finite element analysis and PSO-BP-GA algorithm, one specific model for the coupled field of the ship stern-shaft mechanical seals was optimized. The main conclusions in the paper are as follows:

1. According to the finite element analysis model of mechanical seals coupling field proposed in this paper, the calculation was carried out, and the experiment was designed for verification. The results show that the error between the experimental results and the simulation results is less than 10%, and the analysis model is reasonable.
2. Based on the finite element analysis and PSO-BP-GA optimization algorithm, a fast and efficient mathematical model for the coupled field of the ship stern-shaft mechanical seals could be obviously established. Finally, it was verified better for its good performance applicability through an example in details.
3. Through the comparative analysis of the mechanical seals coupled field data, the BP neural network optimized by PSO had higher prediction accuracy, which would be more conducive to the prediction for the coupled field of the ship stern-shaft mechanical seals. Besides, the prediction accuracy of the PSO algorithm was higher than the GA method which was used to optimize the BP neural network. On the other hand, the prediction accuracy of the previous BP neural network had a certain influence on the accuracy of the optimization result especially for this case. Previous prediction accuracy of the previous BP neural network had a certain influence on the accuracy of the subsequent optimization result especially for this case. Therefore, it would be the focus to vigorously improve the prediction accuracy of the algorithm and try to avoid the volatility of the algorithm for the similar optimization research.
4. PSO-BP-GA optimization method which was proposed had a certain generality and could be used in the optimization design of nonlinear systems with

multi-dimensional design variables. In addition, it still has a good engineering application prospect.

Acknowledgement: This study is funded by supported by major project for technological innovation of Hubei Province of China (No. 2018AAA026). Hubei University of Technology Doctoral Startup Fund (BSQD2016003)

Author contributions: All authors have accepted responsibility for the entire content of this manuscript and approved its submission.

Conflict of interest: The authors state no conflict of interest.

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