Oil film characteristics and failure mechanism analysis of one kind of mechanical seal under the effect of fluid-structure-thermal coupling

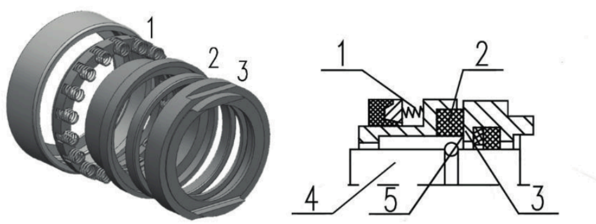
Yuheheng Song*, Shengrong Guo**, Sheng Liu** and Jiming Ma***

Beihang University, School of Reliability and System Engineering, 37#, Xueyuan Rd, Beijing, P.R. China *
Nanjing Engineering Institute of Aircraft System Jincheng, AVIC, 33#,Shuige Rd, Nanjing, P.R.China **
Beihang University, Sino-French Engineering School37#, Xueyuan Rd, Beijing, P.R. China***
E-Mail: songyh125@buaa.edu.cn

A method aims at analysing the fluid-solid-thermal coupling effect of one type of mechanical seal on failure is proposed. The method considers the coupling effects among fluid, solid and heat into one numerical simulation procedure. The deformations of one specific seal are calculated by ANSYS. Samples collected form workshop and field are used for validation of the analysis. Results show that excessive stress is the main reason for the internal ring fractures of the stationary graphite ring. The method proposed in this paper can well predict the location of the fractures. This analysis method is helpful for exploring the failure mechanism, and designing optimization of mechanical seal with longer lifetime and higher reliability.

Keywords: Fluid-structure-thermal coupling, mechanical seal, oil film, failure mechanism, impregnated graphite
Target audience: Hydraulic source, Design Process

1 Introduction



1-compression spring; 2-stationary ring; 3-rotating ring; 4-shaft; 5-liquid space

Figure 1 Structure of mechanical seal

Malfunction of mechanical seal is one of the main causes resulting in failure of electromechanical equipment. Analysis and research on the mechanical failure mechanism is beneficial to improve the performance and life of mechanical seal. The selected mechanical seal by this research, one type of external mechanical seal with specific structure (see Figure 1), is constituted by stationary ring (impregnated graphite with furan resin), rotating ring (mold steel 25Cr3Mo3VNb), compression spring and some accessories. Table 1[1] gives the material and properties of the components and oil.

Component and material	Description	Value	Unit
Rotating Ring (25Cr3Mo3VNb)	Density	7810	kg/m^3
	Specific heat capacity	4.60E+02	J/(kg.K)
	Thermal conductivity	44	W/(m.k)
	Coefficient of thermal expansion	1.01E-05	K^-1
	Reference temperature	20	°C
	Young's modulus	2.20E+11	MPa
	Poisson's ratio	0.28	/
Stationary Ring (M298K)	Density	1650	kg/m^3
	Specific heat capacity	710	J/(kg.K)
	Thermal conductivity	129	W/(m.k)
	Coefficient of thermal expansion	6.50E-06	K^-1
	Reference temperature	200	°C
	Young's modulus	4.04E+04	MPa
	Poisson's ratio	0.27	/
Oil	Density	889	kg/m^3
	Specific Heat capacity	1845	J/(kg.K)
	Heat conductivity	0.145	W/(m.k)
	Coefficient of thermal expansion	1.06	kg/(m.s)
	Reference temperature	25	°C
	Susurlands constant	124	K

Table 1 Material and properties of the components and oil

Statistics of the mechanical seals failure mode from field data, collected by the manufacturer, show that the most common failure is fracture of the graphite stationary ring, see Figure 2. The fracture becomes more serious in inner ring than the outer ring. The purpose of this paper is to analyze the causes of the failure of graphite ring in mechanical seals by simulation method.

Figure 2 Fracture of stationary ring

Many researches on performance and life prediction of mechanical seal have been carried out. Gu [2] summarized the failure modes of mechanical seals as wear, blister and thermocrack. Wear and blisters cause pits and scratches on the surface, and thermocrack means radial crack under excessive thermal stresses. None of the failure modes matches the one shown in Figure 2. The purpose of this paper is to propose a method to reveal the mechanism of this special failure mode.

The numerical analysis method on behavior of mechanical seal has the most outstanding achievements. Mayer [3] systematically expounds the fluid mechanics and lubrication, friction, wear theory and design methods of mechanical seals. Buck [4, 5] proposed a method to calculate the temperature distribution of the seal ring: the fin method. The heat transfer and the efficiency of the seal were calculated by establishing a fin model of the seal ring. Pascovici [6] studied the thermodynamic properties mechanical seal. The effects of Nusselt number, flow direction and the thermal conductivity on the temperature distribution of the ring were analyzed by solving the heat energy equation.

For validation of the numerical results, there are some researchers focusing on the experiment of mechanical seals. Lebeck [7] used experimental methods to measure and predict the oil film coefficient of mechanical seal.

Several tests with varying speed, temperature and running time were performed to obtain temperature distribution data. Through the temperature distribution of the seal ring, the oil film coefficient of the mechanical seal was predicted, and the result was compared with the measured value. Parviz [8] built the similar experimental equipment to validate the results of a computational model for flow and thermal analysis of mechanical seal. The computational model adequately predicts the temperature distribution within the stator of a mechanical seal. these above studies focused only on the seal rings and their thermal and mechanical behaviors, and did not reveal the characteristics of the flow in the seal chamber.

Thermohydrodynamic(THD) has gradually become the focus of researches in recent decades. Brunetière[9] analyzed the thermal performance of lubrication by establishing the three-dimensional THD model of the sealing face. The factors, which affected lubricating property, such as the sealing ring material, fluid flow type, heat transfer conditions of the seal face, were considered in the model. In the following study [10, 11], numerical models for thermal effects was investigated and the influence of the design and running parameters on the TEHD behavior of Mechanical Face Seals (MFS), in steady dynamic tracking mode, was analyzed. Then, Erwan [12] applied the same model to a specific seal used in reactor coolant pumps, and obtained similar results. Luan [13-15] discussed flow behavior and cooling effect of oil in the seal chamber by means of numerical simulation method. In recent years, researchers began to use analytic method gradually to establish the heat transfer equations of seal rings. [16-18]

Above researches' solutions were obtained based on some assumptions, such as uniform temperature and constant viscosity of the fluid film, neglect of the cooling effect of the oil and so on. These assumptions caused errors in the results.

Under the condition of high temperature, mechanical and thermal deformations are the main factor leading to seal failure. Deformations will aggravate wear, leading to high temperature on seal rings' surfaces. Thermal stress on the seal rings may exceed the stress limit of the material, resulting in fracture. Therefore, the study of mechanical and thermal deformation of mechanical seals is the most important part of failure analysis.

Zhu [19] established the mechanical equation for seal of ship stern shaft, and verified the deformation equation by a certain type of seal. Brunetière [20] proposed a semi-analytical method of non-contact seal considering the thermal deformation. Wang [21] established finite element model of rotating and stationary ring with ANSYS, results indicated that the thermal deformation of the rotating and stationary ring leads to the formation of gap in the seal chamber, resulting in the increase of leakage.

In conclusion, fluid-structure-thermal coupling analysis on mechanical seal has made some progress. However, there are few studies on the pressure distribution of the oil film and the elastic deformation of seal rings caused by inhomogeneous oil pressure distribution. Theoretical researches focused on structure-thermal coupling deformation analysis are also few. The purpose of this study is to propose a method to analyze the temperature and pressure distribution of oil film in seal chamber and deformations of seal rings under fluid-structure-thermal coupling effect. Based on this method, failure mechanism analysis is carried out on one specific type of mechanical seal.

2 RESEARCH CONTENTS AND SCHEME

This research presents a fluid-structure-thermal coupling analysis method on behaviors of oil film in the seal chamber. The oil film characteristics (thickness, temperature distribution, pressure distribution) are obtained by simulation. Then, temperature and pressure distributions are used to calculate the structure-thermal coupling deformation and the temperature contour of seal rings. Thanks to the simulation result, the effect of oil film characteristics on the fracture creating process could be revealed efficiently. By comparing the surface morphology of broken seal, the possible failure showed by the analysis results could be validated. The research scheme is shown in Figure 3.

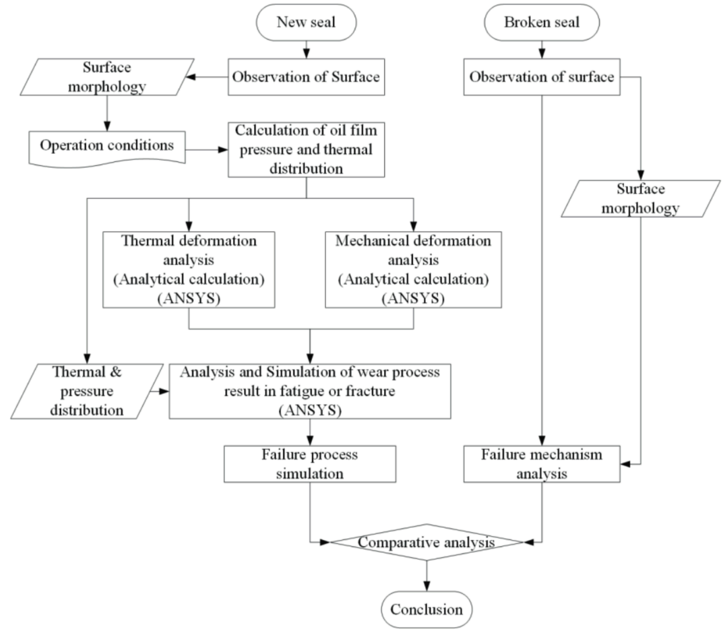


Figure 3 Research Scheme Diagram

3 THEORETICAL MODEL FOR CACULATION OF OIL FILM PRESSURE AND THERMAL DISTRIBUTION UNDER FLUID-STRUCTURE-THERMAL COUPLING EFFECT

3.1 Fluid flow model

The structure of studied mechanical seal and cross section of the seal is shown in Figure 1. The following assumptions are made:

1. Ignore the all body force (gravity or magnetic force).
2. Fluid at the interface is not sliding, that is to say the fluid flow affixed to the rotating ring and stationary ring surface has the same velocity with the rings.
3. Because the oil film thickness is only a few microns or even smaller, the pressure does not change along oil film thickness. So the pressure distribution has only two dimensions, circumferential and radial.
4. Comparing with the oil film thickness, the oil film curvature radius is very large, so the influence of oil film curvature is neglected, and the translational speed is used instead of the rotational speed.
5. Oil in the chamber is Newton fluid.
6. The flow is laminar, there is no turbulence.
7. Comparing with the viscous force, the influence of inertia force can be neglected.

On the basis of the above assumptions, the Reynolds equation can be used to solve the pressure distribution. The general form of the Reynolds equation is shown in the Equation (1). [22]

$$\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial r}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial r}\right)=6\left[\frac{\partial}{\partial\theta}\left(\left(U_h-U_0\right)h\right)+\frac{\partial}{\partial r}\left(\left(V_h-V_0\right)h\right)\right]+12\frac{\partial h}{\partial t} \qquad (1)$$

In this study, oil film thickness h in one specific time step is constant. U_h is the circumferential velocity of the rotating ring, the value is ω_r, and U_h stay constant in circumference. U_0 is the circumferential velocity of the stationary ring, the value is 0. V_h is the radial velocity of rotating ring and V_0 is the radial velocity of the stationary ring, the values of both are 0. Equation (1) can be simplified and rewritten as Equation (2).

$$\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial r}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial r}\right)=12\frac{\partial h}{\partial t} \tag{2}$$

3.2 Temperature distribution equation of the fluid film

For the condition of hydrodynamic lubrication, the variation of kinetic energy and potential energy of fluid flow can be neglected, so that the energy is only a function of temperature. Suppose that the flow is in a stable state, so all variables do not change with time. Under the circumstance of hydrodynamic lubrication, heat convection is much more than heat conduction. So heat conduction along the direction of the film thickness z is negligible. The temperature changes only along direction θ and direction r.

Under these conditions, the energy equation applicable to hydrodynamic lubrication is Equation (3).

$$\frac{\mu_k U^2}{J\rho_0 c_0 h_k}+\frac{h_k^3}{12\mu_k J\rho_0 c_0}\left[\left(\frac{\partial p_{k+1}}{\partial\theta}\right)^2+\left(\frac{\partial p_{k+1}}{\partial r}\right)^2\right]-\lambda\left(T-T_{\mathrm{ref}}\right)=q_\theta\frac{\partial T_{k+1}}{\partial\theta}+q_r\frac{\partial T_{k+1}}{\partial r} \tag{3}$$

In which, q_θ and q_r are volume flow of micro-unit in direction θ and r respectively, U is the circumferential velocity of the flow, $U=\omega\frac{R_1+R_2}{2}$.

The dynamic viscosity of oil varies with pressure and temperature, the calculation of dynamic viscosity requires reference viscosity, pressure and temperature. The dynamic viscosity is given by Equation (4).

$$\mu=\mu_0\exp\left(\alpha p+\beta\left(T-T_{\mathrm{ref\_o}}\right)\right) \tag{4}$$

3.3 Calculation of oil film thickness

The rotating ring has no movement in the axial direction, while the stationary ring can move in the axial direction under the action of the spring. Therefore, the oil film thickness is only related to the motion of the stationary ring and the oil pressure.

Movement analysis on the stationary ring is mainly determined by the force analysis on the stationary ring. According to the structure of seal in the piston pump, the forces acting on the stationary ring include spring force and bearing force of oil film. In addition, the stationary ring is also subjected to a damping force due to the viscous action of the oil. According to the parameters listed in Table 2, the spring force F_N depends on the oil film thickness h_k. The spring force is given by Equation (5).

$$F_N=F_{N_0}+K\left(h_k-H_0\right) \tag{5}$$

The supporting force of oil film is related to the pressure distribution. The supporting force of the oil film on the contact surface between the stationary ring and the oil can be obtained by Equation (6).

$$F_o=\sum_i\sum_j p_{k+1}(i,j)\cdot\left[\left(R_1+j\cdot d_r\right)\cdot d\theta\cdot dr\right] \tag{6}$$

The damping force is one of the main forces that cannot be neglected. It is only related to the motion. The coefficient K_f is an experimental value. In this paper, $K_f = 526$. The damping force can be calculated by Equation (7).

$$F_f=-K_f\cdot v_k \tag{7}$$

After force analysis of the stationary ring, according to Newton's second law, the acceleration a, velocity v_{k+1} of stationary ring, and the next step time oil film thickness h_{k+1} are obtained.

$$\begin{cases} a=\dfrac{F_N-F_o-F_f}{M_s+M_t} \\ v_{k+1}=v_k+a\cdot dt \\ h_{k+1}=h_k-v_{k+1}\cdot dt \end{cases} \tag{8}$$

4 FLUID-STRUCTURE-THERMAL COUPLING ANALYSIS PROCEDURE

A fluid-structure-thermal coupling analysis procedure for the oil film in the seal chamber is introduced in this section. The oil film thickness, temperature and pressure distribution, which all effect the deformation of seal structure, are studied. All the equations in the previous section are solved based on the finite element method and boundary element method, the temperature and pressure characteristics of oil film are simulated in order to investigate their influences on the wear process. The analysis procedure is shown in Figure 4.

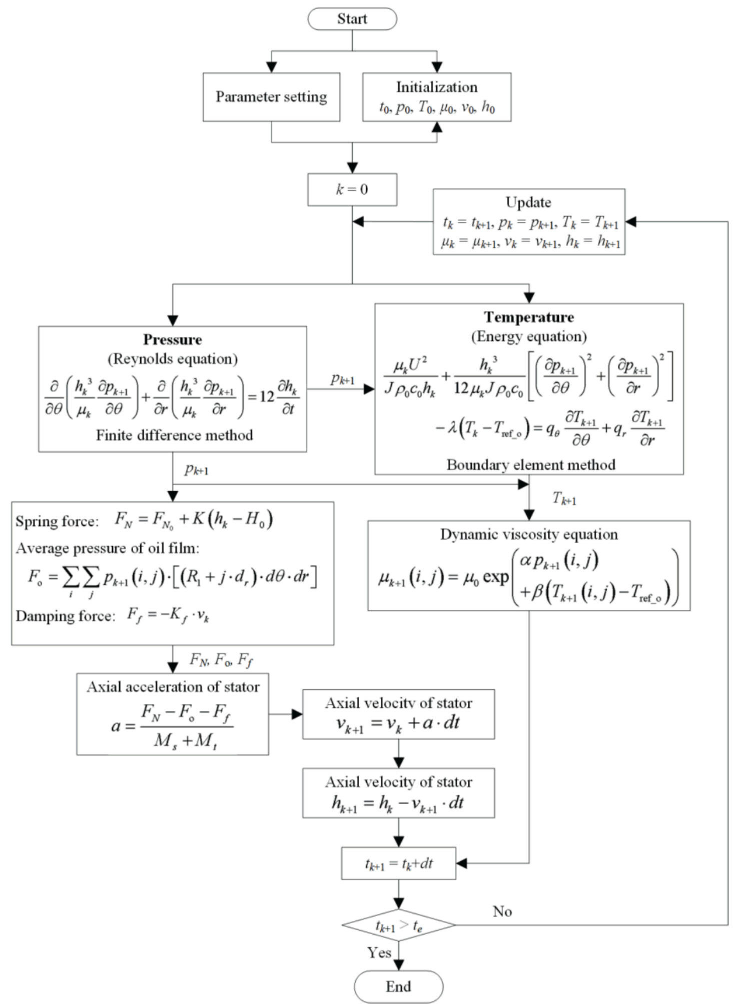


Figure 4 Flow chart of analysis procedure

The instructions for the calculation process are as follows:

1. Parameter setting

The parameters used in the whole calculation process are determined in this step. Most of the values of parameters are theoretical or measured, there is a small amount of them are obtained by experiential hypothesis, such as K_f.

2. Initialization

The time, pressure distribution, temperature distribution, dynamic viscosity of oil film, axial velocity of stationary ring and oil film thickness are initialized for the calculation of the starting state.

3. Calculation of pressure distribution

The Reynolds equation is solved by finite difference method. The calculation requires the pressure distribution, the dynamic viscosity distribution, the axial velocity of stationary ring and the oil film thickness at the previous time step. The first calculation of the initial time takes the initializations as inputs, and then the results obtained at the previous time step can be used at next time step.

4. Calculation of temperature distribution

The energy equation considering the heat conduction is solved by boundary element method. This equation is applicable to the mixed lubrication state. It must be solved after the calculation of pressure distribution because of the need of pressure distribution as input.

5. Calculation of dynamic viscosity of oil film

The dynamic viscosity equation is solved.

6. Calculation of oil film thickness between the stationary and rotating ring

The oil film thickness between the two rings is determined by the axial movement of stationary ring. The motion analysis requires force analysis of stationary ring. After the calculation of the spring force, the oil film pressure and damping force which are the three forces acting at the stationary ring, the axial velocity of stationary ring and the oil film thickness can be calculated.

7. Update the values of variables

Variables (pressure distribution, temperature distribution, dynamic viscosity, stationary ring speed, oil film thickness) are updated to the calculation results of previous time step.

The procedure is repeated until the time t_e has reached the scheduled simulation time. Then the overall calculation is completed.

5 RESULTS AND DISCUSSION

The numerical calculations were performed in order to determine the effect of selected parameters characterizing the seal operation, i.e. the inlet oil temperature (40°C, 100°C and 200 °C), on the distributions of pressure and temperature in the oil chamber.

Table 2 and Figure 5 show the analyzed geometrical and operational parameters of the seal.

Type	Parameter name	Symbol	Value	Unit
Time	Time step	d_t	1e-4	s
	Simulation time	t_e	300	s
Structure & Operation Parameters	Inner radius	R_1	18.3e-3	m
	Outer radius	R_2	20.85e-3	m
	Oil pressure in chamber	P_{oil}	4.5e5	Pa
	Pressure atmosphere	P_{air}	1e5	Pa
	Ambient temperature	T_{ref}	25	°C
	Angular velocity	ω	4200*2*pi/60	rad/s
	Initial velocity of stationary ring	v_0	0	m/s
	Mechanical equivalent of heat	J	4.1840	J/cal
	Spring pretightening force	F_{N0}	88	N
Material	Mass of stationary ring	M_s	0.002022	kg
	Mass of sleeve of stationary ring	M_t	0.025	kg
	Spring stiffness coefficient	K	3460	N/m
	Density of oil	ρ_0	889	Kg/m^3
	Specific heat capacity of oil	c_0	1845	J/(kg·°C)
	Coefficient of heat transfer	λ	475	W/(m^2·°C)
	Thermal conductivity of oil	k_0	0.145	W/(m·°C)
	Pressure viscosity coefficient	α	2.2e-8	/
	Thermal viscosity coefficient	β	-0.03	/
	Reference oil viscosity	μ_0	15e-6	N.s/m^2(Pa·s)
	Reference temperature	T_{ref_o}	50	°C

Table 2 Parameters setting for numerical calculations

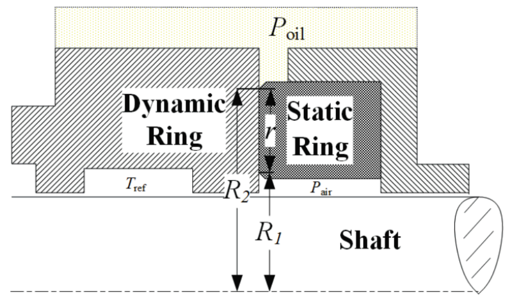


Figure 5 Diagram of operational parameters

After the simulation under the condition of different inlet oil temperature, the calculation results were presented in the graphical form. Figure 6 shows distributions of temperature in the oil chamber for three values of rotational speed of the inlet oil temperature: 40°C, 100°C and 200 °C along the radius. Figure 7 shows the distributions of pressure.

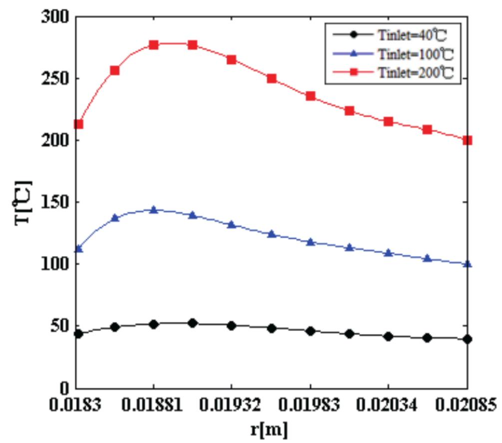


Figure 6 Temperature distribution in oil chamber for different values of inlet oil temperature

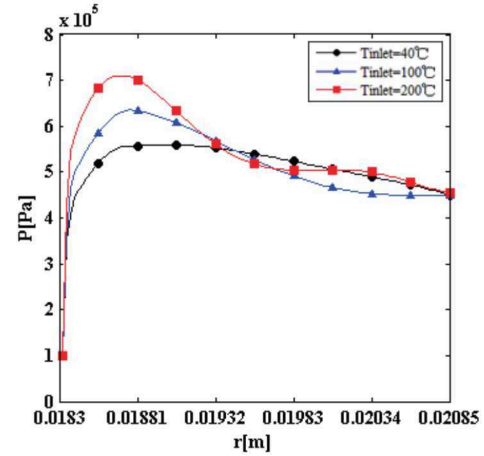
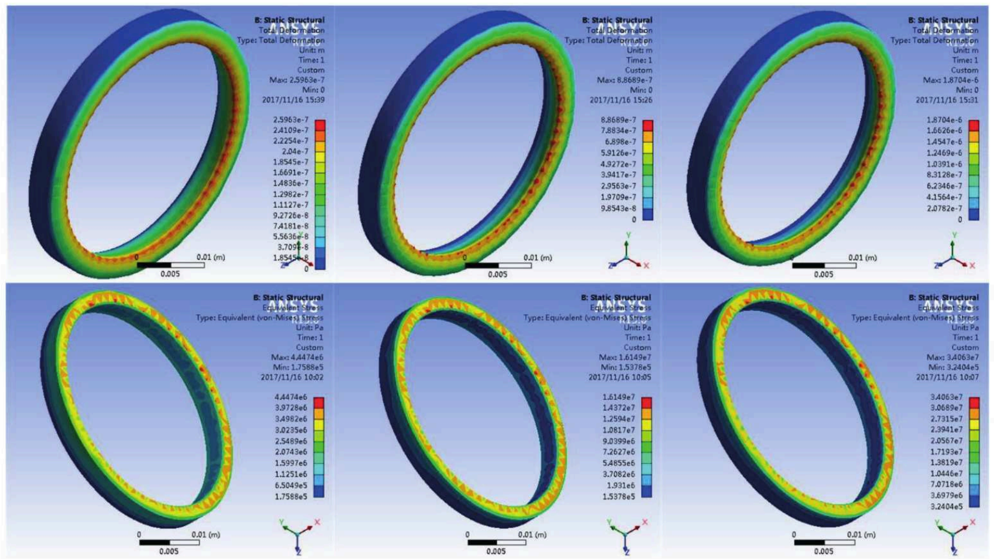


Figure 7 Pressure distribution in oil chamber for different values of inlet oil temperature

It can be seen from Figure 6 and Figure 7, the higher temperature of inlet oil will cause the more significant difference in temperature in the chamber. The position of the higher temperature in the oil chamber is also the position of higher pressure. The temperature and pressure results obtained by simulation are used as input conditions and analyzed in the commercial software ANSYS to determine whether the graphite seal ring will be damaged under the action of the temperature and pressure loads.

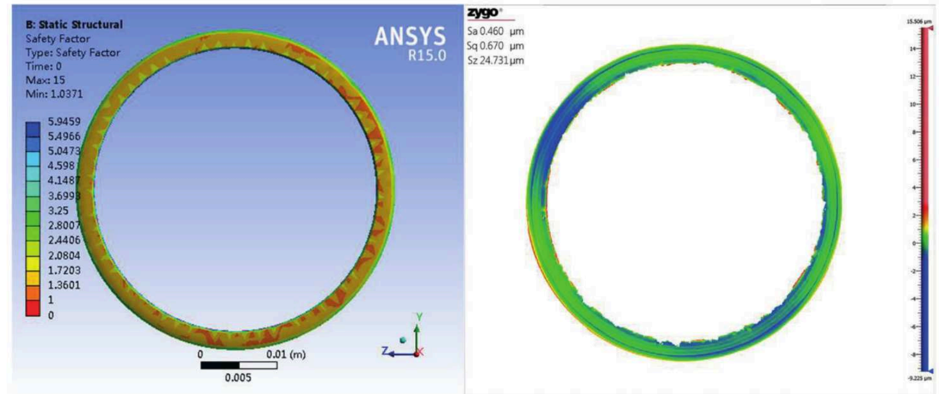


(a) 40°C *(b)100°C* *(c)200°C*

Figure 8 Total deformation and Equivalent (von-Mises) stress results in ANSYS

According to Figure 7, the inner side protrudes and bears tensile stress. The inner side of the graphite ring is subjected to larger stress, and the weak parts are distributed on the inner side. With the increase of the inlet oil temperature, the inner side is subjected to more tensile stress. Under the condition of temperature of 40°C, 100°C and 200°C, the maximum stresses are 4.45MPa, 16.15MPa and 34.06MPa respectively. According to the description of [1], the compressive strength of the graphite material M298K is 120MPa, the shear strength is 57MPa, and the tensile strength is only 0.57 of the shear strength, which is 32.5MPa. When the inlet oil temperature is up to 200 degrees, the stress of the weak part of the inner side has slightly exceeded the tensile strength of the material, and the fracture may appear in the continuous operation process.

The comparison between the measured results and the simulation results is shown in Figure 9. According to Figure 9, the positions of simulated damage (where safety factor are less than 1) are consistent with the position of actual damage in the seal using in the outfield.



(a) *(b)*

Figure 9 Comparison of (a) simulated damage and (b) actual damage

6 CONCLUSION

In this paper, the failure mode of one type of graphite-steel mechanical seal is analyzed. A numerical model under the effect of fluid-structure-thermal coupling, which is suitable for describing the thermal and mechanical behavior of oil in the chamber, has been developed. This model incorporates the Reynolds equation for fluid-flow, energy equation considering the heat conduction for fluid and force analysis of the stationary ring. The effect of the inlet oil temperature on the distributions of pressure and temperature in the oil chamber is analyzed using this model. Results show that the higher the inlet oil temperature is, the more intense the temperature difference is in the oil chamber and the intense temperature difference in the oil chamber will lead to strong squeeze effect of oil, then the pressure in the chamber will increase.

Analysis of Equivalent (von-Mises) stress in ANSYS under the pressure and temperature loads calculated by proposed model of graphite ring indicates that under the condition of inlet oil temperature at 200°C, the graphite bears the stress which exceeds the strength limit of the material. The damage locations predicted by the model are consistent with the observed damage locations. Based on the failure analysis results, excessively high inlet oil temperature is the fundamental cause of failure, and the mechanical seal should avoid long-term operation at high temperature.

The method proposed in this paper is appropriate to describe structural behavior of graphite ring used in mechanical seal in case of inner ring fractures. Indeed, the model proposed is appropriate to calculate the loading details and predict the effects caused by the loading details, which markedly influence the failure mechanism of this type of structures.

References

/1/ X. Shen, *Handbook of sealing materials*. Beijing, China: China Petrochemical Press, 1993.

/2/ Y. Gu, "Sealing Failure Modes and Critical Balance Ratio for Mechanical Seals," *Journal of the*

University of Petroleum, vol. 20, no. 6, pp. 52-56, 1996.

3/ E. Mayer, *Mechanical Seals*. Dusseldorf, Germeny: VDI-Verlag, 1966.

4/ B. GS, "Heat Transfer In Mechanical Seals," presented at the Proceeding of 6th internation pump user symposium, Baton rouge, LA, USA, 1989.

5/ B. GS, "Estimating Heat Generation, Face Temperature and Flush Rate for Mechanical Seals. ," presented at the PumpUsers Expo'99, Baton rouge, LA, USA, 1999.

6/ M. D. Pascovici and I. Etsion, "A Thermo-Hydrodynamic Analysis of a Mechanical Face Seal," Journal *of Tribology,* vol. 114, pp. 639-645, 1992.

7/ A. O. Lebeck, M. E. Nygren, S. A. Shirazi, and R. Soulisa, "Fluid Temperature and Film Coefficient Prediction and Measurement in Mechanical Face Seals--Experimental Results," *Tribology Transactions,* vol. 41, no. 4, pp. 411-422, 1998.

8/ P. Merati, O. Nori Aki, P. Robert L, and J. Larry E, "Experimental and Computational Investigation of Flow and Thermal Behavior of a Mechanical Seal," *Tribology Transactions,* vol. 42, no. 4, pp. 731-738, Oct 1999.

9/ B. Tournerie, J. C. Danos, and J. Frêne, "Three-Dimensional Modeling of THD Lubrication in Face Seals," *Journal of Tribology,* vol. 123, no. 1, pp. 196-304, 2001.

10/ N. l. Brunetièree, B. Tournerie, and J. Frêne, "TEHD Lubrication of Mechanical Face Seals in Stable Tracking Mode: Part 1-Numerical Model and Experiments," *Journal of Tribology,* vol. 125, no. 3, pp. 608-616, 2003.

11/ N. l. Brunetièree, B. Tournerie, and J. Frêne, "TEHD Lubrication of Mechanical Face Seals in Stable Tracking Mode: Part 2- Parametric Study," *Journal of Tribology,* vol. 125, no. 3, pp. 617-627, 2003.

12/ E. Galenne and I. Pierre-Danos, "Thermo-Elasto-Hydro-Dynamic Modeling of Hydrostatic Seals in Reactor Coolant Pumps," *Tribology Transactions,* vol. 50, no. 4, pp. 466-476, 2007.

13/ Z. Luan and M. M. Khonsari, "Numerical Simulations of the Flow Field Around the Rings of Mechanical Seals," *Journal of Tribology,* vol. 128, no. 3, pp. 559-569, 2006.

14/ Z. Luan and M. M. Khonsari, "Computational Fluid Dynamics Analysis of Turbulent Flow Within a Mechanical Seal Chamber," *Journal of Tribology,* vol. 129, no. 1, pp. 120-128, 2007.

15/ Z. Luan and M. M. Khonsari, "Heat transfer correlations for laminar flows within a mechanical seal chamber," *Tribology International,* vol. 42, no. 5, pp. 770-778, 2009.

16/ G. Zhu, "Computer Prediction of Mechanical Seal Performance and Experimental Validation," *Proceedings of the Institution of Mechanical Engineers. Part J: Journal of Engineering Tribology,* vol. 213, no. 6, pp. 433-449, 1999.

17/ S. Błasiak, C. Kundera, and J. Bochnia, "A Numerical Analysis of the Temperature Distributions in Face Sealing Rings," *Procedia Engineering,* vol. 39, pp. 366-378, 2012.

18/ S. Blasiak, "An analytical approach to heat transfer and thermal distortions in non-contacting face seals," *International Journal of Heat and Mass Transfer,* vol. 81, pp. 90-102, 2015.

19/ Z. Hanhua, L. Zhenglin, and W. Shizhu, "Mechanical Deformation of Ship Stern-shaft Mechanical Face Seals," *Tsinghua Science and Technology,* vol. 9, no. 3, pp. 299-302, 2004.

20/ N. Brunetière and A. Apostolescu, "A Simple Approach to the ThermoElastoHydroDynamic Behavior of Mechanical Face Seals," *Tribology Transactions,* vol. 52, no. 2, pp. 243-255, 2009.

21/ W. Heshun and D. Lin, "Research on Thermal Deformation of Hydrostatic Pressure Mechanical Seal," *Energy Education Science and Technology Part A: Energy Science and Research,* vol. 31, no. 1, pp. 473-476, 2013.

/22/ W. Shizhu and H. Ping, *Principle of Tribology*. Tsinghua university press, 2012.