Finite Elements in Analysis and Design 83 (2014) 10–21

Contents lists available at ScienceDirect

Finite Elements in Analysis and Design

journal homepage: www.elsevier.com/locate/finel

Prediction of seal wear with thermal–structural coupled finite element method

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ARTICLE INFO

Article history:
Received 2 May 2013
Received in revised form
12 October 2013
Accepted 14 January 2014
Available online 13 February 2014

Keywords:
O-seal
R-seal
Wear
Thermal–structural coupled simulation
FE

ABSTRACT

In this paper, a thermal–structural coupled finite element analysis method for wear prediction of seal is proposed. The methodology is built with iterative wear prediction procedure in which the geometry of the contact interface is progressively changed according to the wear model. To perform finite element simulation of wear process, a novel mesh reconstruction strategy to reflect the evolution of geometry caused by the wear is presented. Considering that friction heat in the sealing area normally has important effects on wear process, thermal behavior of wear process analysis is carried out and the friction heat flux density is calculated as one of the boundary conditions in each wear simulation circle. Two kinds of seals, namely O-shape seal (O-ring) and rectangular-section seal (R-ring), are investigated in this paper. As the sealing force is important for seals to evaluate the sealing capability, the effect of wear is reflected through the change of sealing force. The numerical results supply a suggestion for the applications of both seals.

1. Introduction

In hydraulic systems, seals may contact directly with a moving components due to high extrusion force of the fluid. Wear will be one of the most important factors which essentially affects the lifetime of the seals. In addition, material removal of the seals will decrease the contact pressure in the sealing area and the leakage will be increased in the same time. Extent leakage is able to lead to the failure of the entire hydraulic system, even safety problems sometimes. Thus, a comprehension of the wear evolution of the seals and its effect on the change of the sealing pressure can be useful for the designers.

Standard pin-on-disc experiment is widely applied to investigate the wear of the materials. As the boundary and initial conditions can be highly different in different cases even for the same material, pin-on-disc experiment is not satisfactory for wear prediction of the seals in designing phase. A full-scale experiment is sometimes applied to simulate the true operating conditions to predict the wear of the seals precisely. However, much time and much money are needed for this kind of experiments.

Numerical simulation methods have gradually replaced the position of the experimental methods because the former is able to overcome the shortcomings of the latter. For example, numerical simulation methods assist to investigate the effect of the wear on the contact pressure and the deformation of the seals which are hardly obtained through experimental methods. In addition, numerical simulations can be applied for parametric investigation if validated by the reliable experiment. More similar complicated problems can be predicted through the verified numerical methods.

Many numerical methods have been developed to simulate the wear process. Among them finite element technology is a popular method. An FE simulation was applied to model the wear behavior of metals and ceramics by Podra and Andersson [1]. The process of the particle removed from the steel when the steel sliding on a rough surface was investigated by Balogh [2] and Eleod [3]. A wear of pin-pivot oscillating contact was investigated by Mukras et al. [4]. The wear process of the disc brake system was studied by Sorderberg and Andersson [5]. Theoretical analysis combined with FE model was proposed by Salib et al. [6]. The model was simplified as spherical contact in this research. Ashraf et al. [7] modeled the wear process of the rear view mirror. A wear procession of the aircraft clutch was simulated by Zhao et al. [8]. The wear processes of some composite materials were studied. The friction and wear behaviors of polymer composite materials were studied through FE models by Goda et al. [9]. Teoh et al. [10] studied the wear process of a polyethylene cup of hip prosthesis.

The researches mentioned above focused on the wear simulation of hard materials. However, there is a limitation: only the element layer of the surface can be worn. The accuracy of FE

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0168-874X/$ - see front matter
http://dx.doi.org/10.1016/j.finel.2014.01.001

simulation method is not severely influenced for hard materials because a little loss of the hard materials is able to lead to the huge variation of the contact area and the pressure distribution. But for soft materials such as rubber, the loss of the top material in micrometer magnitude cannot influence the contact condition obviously for its hyperelastic property. The wear volume which leads to the failure of the rubber material is much more than that of the steel. Therefore, only the top layer of the elements worn in FE model cannot satisfy the need for the simulation of the rubber. One way to solve this problem is to increase the size of the elements on the top layer. But the accuracy will be decreased in the same time. Some other methods combined with FE models are given in [11]. However, the model is limited to a theoretical model. No specified operating conditions are considered. In addition, as the seals used in the hydraulic systems may contact with the shaft which rotates in a high speed, the effect of friction heat on the wear cannot be neglected.

As the sealing force is a criterion that is always applied to evaluate the capability of the seals, the prediction of the effect of wear on the sealing force for the seals which contact with a rotating shaft in the hydraulic system is the goal of this paper. Since the material of the seals is traditionally defined as rubber, a strategy that is able to simulate the wear process of soft materials without the limitation of the size of the elements is presented in this paper. In addition, the pressure and the temperature of the hydraulic oil are also incorporated in boundary conditions to model the real operating environment. Considering that friction heat normally has important effects on wear process under high speed rotation situation, another contribution of this paper is to employ thermal–structural coupled analysis into seal's wear prediction.

2. Finite element wear analysis

2.1. Structural finite element model

The subject of this paper focuses on the rotating sealing system. In the rotating sealing system, O-seal and R-seal are commonly used. Thus, these two kinds of seals are the research objects in this paper. Fig. 1 shows a typical geometry of a common rotating sealing structure using O-shape and R-shape seals. The seal is installed in the slot in the chamber and fixed with the chamber while the shaft rotates independently. There is a relative motion in the area where the seal contacts with the shaft. The function of the seal is to prevent the flowing of the fluid on both sides of the seal.

According to the geometric model of sealing system, a finite element model of this structure is built. In this paper, the material of the chamber and the shaft is set as steel while the material of the seals is selected as rubber. Since rubber is a kind of hyperelastic material, hyperelastic element is adopted for the seals in the FE model. As the elastic modulus of the steel is much bigger than that of the rubber, the deformation of the chamber and the shaft is ignored. It means that both the chamber and the shaft can be treated as rigid bodies in our finite element model. Therefore,

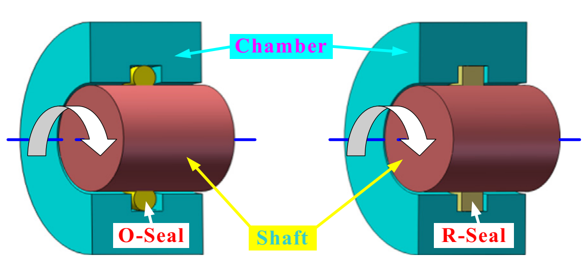

Fig. 1. Typical rotating sealing system with O-seal and R-seal.

the profiles of the chamber and the shaft will be taken as constraints on the boundary of the seals.

Besides, in order to save the computing cost, considering the symmetrical feature of the sealing systems shown in Fig. 1, only half of the 2D finite element model is needed. Asymmetrical meshing strategy is adopted when meshing the domain of the seal. As mentioned above, there is a relative motion between the seal and the shaft. So the wear of the seal mainly occurs on this area. In order to simulate the wear behavior of the seal precisely and reduce the computing time, the area where the seal contacts with the shaft is densely meshed while the other area of the seal is coarsely meshed because there is no relative motion and the wear cannot be considered.

The FE models of O-seal and R-seal are given in Fig. 2. The aim of the structural analysis is to obtain the contact pressure distribution. Thus, analysis type for this model is static mechanics. In order to compare the wear behavior of the two kinds of seals, the geometries of the two models are set with the same values.

In reality, deformation often occurs when the seal is installed in the slot. But the deformation cannot be pre-calculated in the FE model. So the pre-compression effect is not included in the FE model but will be accomplished in the following step.

Considering the true working conditions, there is fluid scattering in the outer and inner of sides the chamber. Therefore, both sides of the seal contact directly with the fluid. That means the pressure of the fluid will be acted on the seal. The left side of the seal is assumed to bear the pressure from the inner side fluid while the right side of the seal bears the pressure from the outer side and the pressure from inner side is supposed to be higher than the outer side in this study.

To sum up, the nodal contact pressure distribution can be obtained through two steps. The compression of the seal is accomplished in the first step. The height of the slot is set as D_s and the height of the clearance between the chamber and the shaft is set as Δ. The sectional diameter of O-seal is marked with Φ_O and the sectional width and the height of R-seal are marked with W_R and H_R, respectively. Both the horizontal and vertical freedom degrees of the boundary of the shaft are fixed and only the horizontal freedom degree of the boundary of the chamber is fixed. A y-displacement d_h ($d_h = \Phi_O - D_s - \Delta$ for O-seal and $d_h = H_R - D_s - \Delta$ for R-seal) is applied on the boundary of the chamber to achieve the required compression.

The effect of the fluid pressure is taken into account in the second step. As mentioned above, the left side of the seal is assumed to bear a higher pressure. Supposing that the seal will be compressed to right wall of the slot due to the higher pressure from left side and the fluid on the right side can rarely contact with the seal, only the left-side pressure is applied. When the seal is deformed, part of the left side of the seal contact with the fluid directly which is marked with red curves in Fig. 2. Therefore, the fluid pressure P_{fluid} is set at the nodes on the red line in the horizontal direction. In the meantime, both the horizontal and the vertical freedom degrees of the shaft and the chamber are constrained.

2.2. Friction heat analysis

The influence to the wear of the seal caused by the friction heat in the sealing area cannot be neglected when the shafts rotates with a high speed. The circumferential shear stress at node i on the seal's surface can be given as

$$\tau_i = f p_i \tag{1}$$

where f is the friction coefficient, p_i is the normal nodal pressure.

In high speed rotating sealing system, the friction heat will affect the wear of the seal while the loss of the material will affect