

Analysis and optimization of nitrile butadiene rubber sealing mechanism of ball valve

Xue-Guan SONG, Lin WANG, Young-Chul PARK

Department of Mechanical Engineering, Dong-A University, Busan 604-714, Korea

Received 2 March 2009; accepted 30 May 2009

Abstract: An approach for analyzing and optimizing sealing mechanism of ball valve made of nitrile butadiene rubber(NBR) with finite element method was presented. The Mooney-Rivlin hyperelastic material model was chosen to characterize NBR sealing, as it has been recommended in the similar applications. That is, NBR sealing was modeled as incompressible hyperelasticity, as well as the assumption of isotropic flow. The results illustrate the structural pressure and contact pressure on the contact surface, which shows that the NBR sealing mechanism is very suitable for sealing after dimension optimization.

Key words: nitrile butadiene rubber(NBR); elastomeric seal; finite element method; contact pressure

1 Introduction

The use of nitrile butadiene rubber(NBR) in valve industry has increased in recent years. A wide group of ball valves work as main/important sealing components. The chemical name of nitrile rubber is butadiene acrylonitrile. However, most are referred to as NBR or Buna-N. As a kind of elastomers, NBR is capable of recovering substantially in size and shape after removal of a load or from large deformations. Hence, NBR is considered as the standard material for sealing, since abrasion and tear resistance of NBR is very excellent.

NBR owes its many applications to its special mechanical properties; however, on the other hand, the specially nonlinear mechanical properties make the analysis of NBR very difficult. It is necessary to take into account of the nonlinear stress—strain relations and incompressible behavior of such material, in addition to its vast amount of deformation. In the past, a great deal of work has been devoted to such incompressible hyperelasticity and its implementation into various finite-element codes. Especially, the finite element method is popular to investigate the performance of elastomeric seals when designing a new geometry or analyzing an existing one[1–5]. It is very useful to evaluate the stress and strain state, contact pressure on surfaces, friction force and so on.

In this study, it is focused on designing a new seal from an existing seal installed in ball valve. The procedure of seal design can be divided into two stages. In the first stage, an existing seal was investigated to understand the stress and contact pressure distribution on the seal. In the second stage, based on the result of first stage, the cross section of seal was optimized to make the contact pressure distribute uniformly. To this end, plane finite element model, which took material nonlinearities and frictional contact between the seal and ball into account, was developed. The contact pressure on the contact surfaces between seal and ball is illustrated, to compare the performance of different cross sections and verify the best one.

2 Geometry and material properties

The seal studied in this research is a commercial seal designed for individual installation for the leakage protection of ball valve with a diameter of 50 mm. Fig.1 shows both the cross section and 3/4 3D model of the seal. In practice, the seal is clamped among the valve body, fix ring and ball with a slight interference fit, designated as 0.2 mm in tangential direction and 0.05 mm in axial direction.

The material properties of NBR were obtained from experimental tests of samples. The stress—strain characteristic was determined by means of a uniaxial

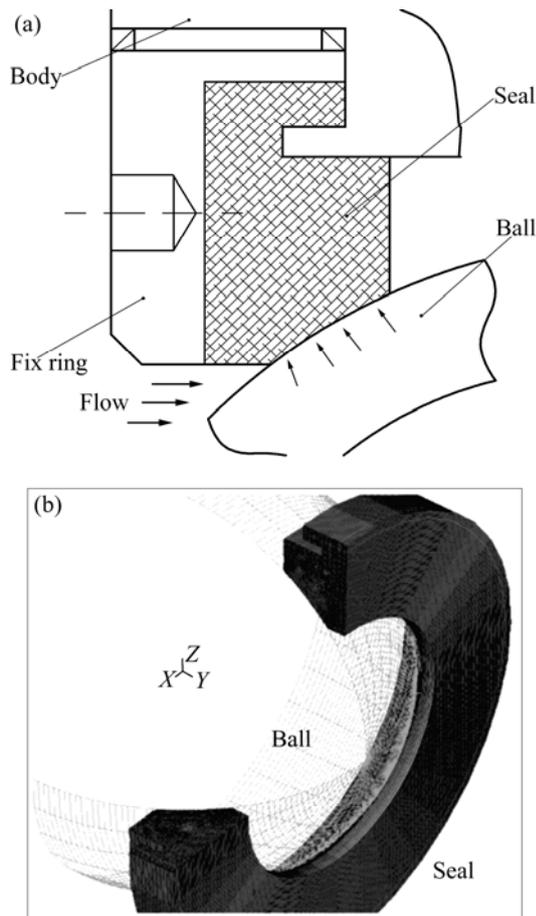


Fig.1 Cross section (a) and 3-D modeling (b) of ball valve seal

tensile-compression test using a dynamometer rig and standard test specimen in accordance with KS M 6518. And then, the Mooney-Rivlin hyperelastic material model was chosen to characterize the seal, as it has been historically popular in the modeling of elastomers. The Mooney-Rivlin model assumes the material to be isotropic and incompressible, and determines stress based on the derivation of a strain energy function W that can include two, three, five, or nine parameters.

3 Numerical model

A finite element model of the seal was built using ANSYSTM Release 11.0 in order to evaluate seal performance under working conditions. The FEM model is shown in Fig.2. A plane model of the seal was developed using four-node plane elements hyper 56 and contact elements contact 169 to simulate contact between the seal and the valve ball. The valve ball was represented by means of rigid elements, meaning that there are no deformations compared with the initial shape. In addition, the axial symmetry constraint was taken into account for simulating the real model.

The full working condition during normal operation is divided into three kinds: installation in seat with an

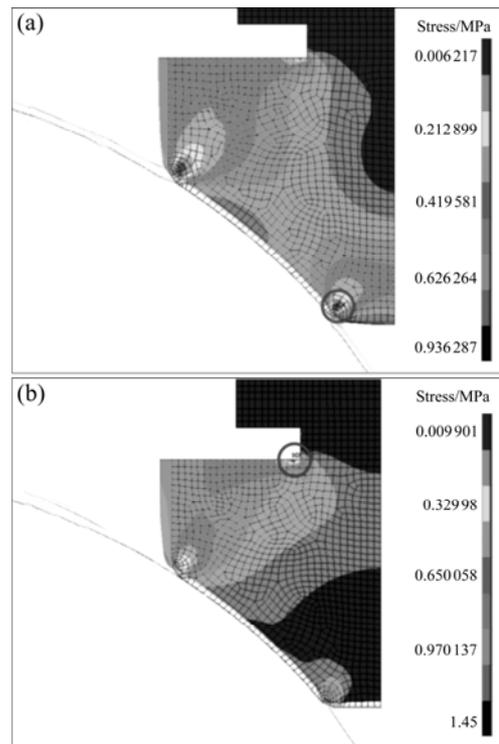


Fig.2 Deformed cross section and von Mises stress under two load conditions: (a) Load condition A; (b) Load condition A+B

initial interference fit (load condition A), the effect of fluid pressure (load condition B), and valve ball rotation or opening (load condition C). Load condition A was achieved by displacing the ball, and moving it against the seal in the tangential direction and axial direction by a distance equal to 0.2 mm and 0.05 mm in order to establish the nominal condition. Load condition B was represented by a constant load per unit length on the inner edge of the seal. Load condition C was achieved by rotating the ball in Z direction, but was not discussed in this work. Load conditions were applied incrementally for each small load step, which is helpful to encouraging solutions to converge. Besides, boundary conditions also took into account of the friction on the contact surfaces between seal and ball. Based on some simple test between NBR and cast steel in water lubricated, the contact between seal and ball surface has a friction coefficient (f) of 0.1. The assumption of a fixed friction coefficient is a simplification with respect to a more rigorous study that takes into account of the mixed phenomena between the seal and the rotating contact surface.

As discussed above, NBR can be well represented by the strain energy function according to the Mooney-Rivlin equation. Herein, the two-parameter option was chosen since it provided enough fitness with the test curve. The two constants, $C10$ of 1.87 and $C01$ of 0.47, were computed by means of uniaxial tensile and uniaxial compression tests, respectively[6]. The Poisson's ratio

was assumed to be 0.49.

4 Analyses of existing seal

Before doing the optimization of seal geometry, the FE analysis of existing seal was performed to study its behavior under the working conditions. Figs.2 and 3 show the stress and contact pressure distribution under the load condition A and load condition A+B, respectively. The mesh areas represent the undeformed shape of seal before being applied load conditions, and the solid planes indicate the new shape of seat after deformation. As can be seen in Fig.2, under the load condition A, the maximum stress marked as “MX” in circles occurs at the inner corner portion of the seal, but under load condition A+B, it occurs near the outer corner in contact with valve body, which probably may be due to the stress concentration at right-angled corner. Contact pressure at the seal/ball interface is of particular interest, as the sealing efficiency and the abrasion of seal directly depend on it directly. Fig.3 shows the contact pressure at the interface under load condition A and load condition A+B. It can be seen from Fig.3 that the maximum contact pressure marked as “MX” in circle always occurs near the outer corner portion of the seal, and the distributions are very similar no matter under which condition. Fig.4 demonstrates the contact pressure along the seal/ball interface. It can be seen that the contact pressure at the central portion is very uniform and

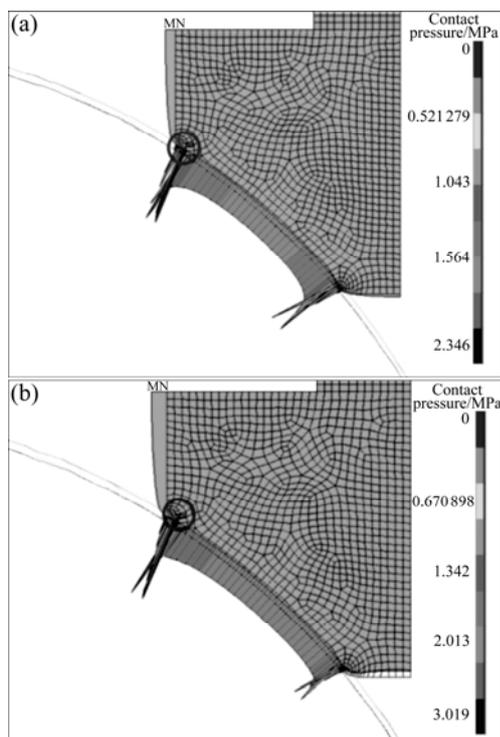


Fig.3 Deformed cross-section and contact pressure under two load conditions: (a) Under load condition A; (b) Under load condition A+B

appropriate, but increases very sharply at the two side portions, that is, the pressure distribution is highly non-homogeneous there. Also, both of the maximum contact pressures under two conditions are excessively high compared with the fluid pressure (0.88 MPa), which means this seal provides an excellent restriction to prevent the leakage and wear out easily by means of friction.

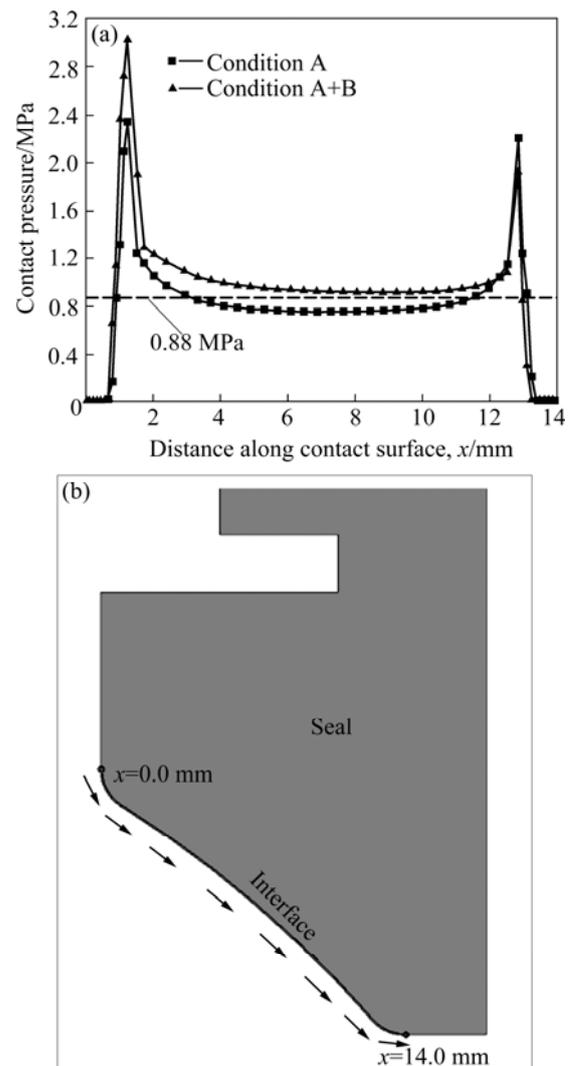


Fig.4 Contact pressure at initial seal/ball interface (fluid pressure: 0.88 MPa)

5 Optimization

As the starting data, the results of FE analysis of initial model demonstrate that the existing cross section has good ability of preventing leakage, but disadvantage of preventing material abrasion, because of the overhigh contact pressure. The purpose of the seal optimization is to find a good cross section that can provide the best trade-off between leakage prevention and material abrasion. From the point of preventing leakage, the contact pressure at interface should be as high as

possible; however, on the other hand, a little of excessive contact pressure, or very non-uniform pressure at very small region, may wear/break the seal quickly and heavily. Hence, there come to be two requirements for the design of new seal. First, the maximum contact pressure (p_{max}) should be greater than the fluid working pressure (p_f), $p_{max} > p_f$. Second, the contact pressure should be as uniform or homogeneous as possible.

To complete the optimization at these two requests, a two-stage procedure was used. First, a new cross section was designed, which can satisfy the second requirement. In this stage, the real value of contact pressure was not considered, that is to say, just make the contact pressure distribute uniformly. In the second stage, based on results obtained in the first stage, a slight interference fit was adjusted to make the whole contact pressure increase or decrease to a necessary and proper level.

Stage I : Four new models were studied in the first stage. The difference between these four models and the initial model is that the curvatures of contact surfaces are different. A schematic view of them is depicted in Fig.5, where *A* represents the initial model with the same curvature as the ball, *E* is a straight line, also thought to be infinite large curvature, *B*, *C* and *D* are three special curvatures between *A* and *E*.

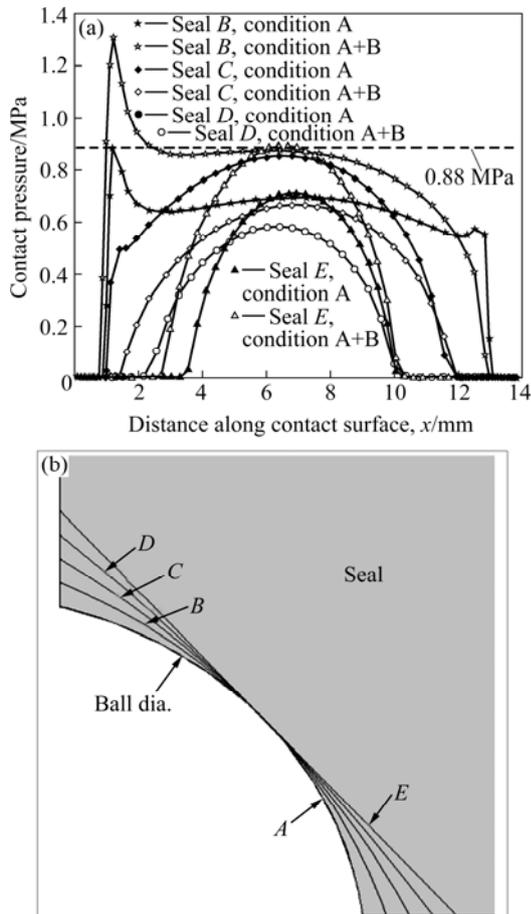


Fig.5 Contact pressure at seal-ball interfaces of different seals

Fig.5 shows the contact pressure at each corresponding interface, from the point of uniform distribution, seals *B* and *C* have better effect than seals *D* and *E*, even there is some abrupt increase at the side portion on seal *B*, because the contact pressure on seals *D* and *E* just occurs at a small central portion. Hence, the curvature between *B* and *C* is better. Based on this assumption, by using experiment design method, the optimized surface with a curvature between *B* and *C* was obtained at last.

Stage II : Having obtained the special curvature, the analysis of this seal was conducted first. As shown in Fig.6(a), even though the distribution of contact pressure is uniform, the maximum contact pressure p_{max} is smaller than the fluid pressure of 0.88 MPa, hence, the interference fit has to be adjusted to ensure the first requirement. Fig.6(b) illustrates the contact pressure along the seal/ball interface after increasing 0.05 mm interference in the axial direction, that is, the total interference fit is 0.2 mm in tangential direction and 0.1 mm in axial direction. This shows that about 65% interface has contact pressure a little higher than fluid working pressure, meaning that this seal can be designed as the optimum seal. Compared with the initial geometry

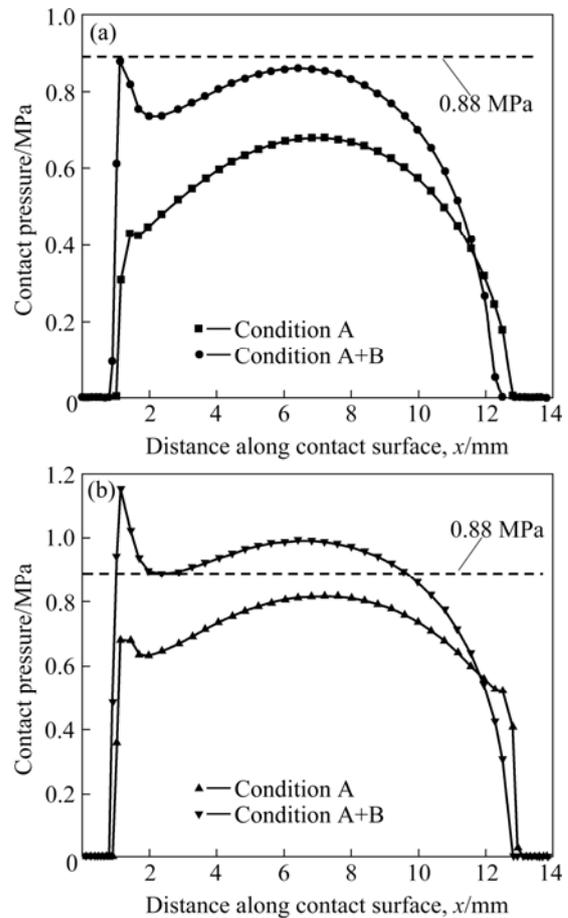


Fig.6 Contact pressure at seal/ball interface of optimized seal: (a) Before adjustment of interference; (b) After adjustment of interference

depicted in Fig.2, the final geometry of the seal only changes the curvature of the contact surface from the initial value of 39.5 mm to 63.0 mm.

6 Conclusions

A new geometry of seal for ball valve was proposed. Compared with the initial seal, the optimized seal has two advantages. First, the optimized seal produces proper contact pressure while working under the loading conditions, not as the initial seal producing excessive pressure. Second, the contact pressure at the interface is more uniform than that at the interface of initial seal, which reduces the abrasion effectively. In a word, the optimized seal minimizes friction abrasion while maintaining sealing efficiency.

The method employed proposed a new concept of uniform distribution of contact pressure. This concept is helpful to minimizing the abrasion in advanced design. In particular, the finite element analysis shows to be a reliable and convenient design tool.

The analysis and design presented does not consider temperature effect and true durability condition. Thus, in the future work, it is necessary to include the temperature analysis and fatigue analysis.

Acknowledgement

This work was supported by Technical Center for High-Performance Valves from the Regional Innovation Center (RIC) Program of the Ministry of Knowledge Economy (MKE), Korea.

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(Edited by LI Xiang-qun)