

Numerical Investigation of the Effect of Radial Lip Seal Geometry on Sealing Performance

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Abstract: Sealing elements are often needed in industry and especially in machine design. With the change and development of machine technology from day to day, sealing elements show continuous development and change in parallel with these developments. Many factors influence the performance of the sealing elements such as shaft surface roughness, radial force, lip geometry etc. In addition, the radial lip seals must have a certain pre-load and interference in order to provide a good sealing. This also affects the friction torque. Researchers are developing new seal designs to reduce friction losses in mechanical systems. In the presented study, the effect of the lip seal geometry on sealing performance will be examined numerically. The numerical model created for this purpose will be verified with experimental data firstly. In the numerical model, shaft and seal will be modeled as hyper-elastic in 2D and 3D. NBR (Nitrile Butadiene Rubber) as seal material will be analyzed for the rotating shaft state at constant speed by applying a uniform radial force.

Key Words: Lip Seal, Lip geometry, sealing performance, NBR (Nitrile Butadiene Rubber), Mooney Rivlin

1. Introduction

The use of nitrile butadiene rubber (NBR) in seals has increased significantly in recent years and is now the most used sealing material in the industry. In the majority of industry, lip seals are used as important sealing equipment also elastomer class material NBR with chemical name butadiene acrylonitrile has the ability to avoid of deformations after the applied force is removed or after large deformations. The NBR, which has high wear resistance for this reason, is considered to be a standard sealing element material.

Mechanical properties of NBR is very useful in many applications, but particularly non-linear mechanical properties of NBR make difficult to analyze the elastomer. Nitrile rubber is more resistant to oil and some additives are resistant to airborne degradation. But, NBR is not favorable to work in high temperature and cold environments. The working temperature is between -40 and 110°C [1]. In this study, analysis has been carried out at room temperature.

Except for the high amount of deformation, calculations must be done according to non linear mechanical properties and incompressibility of material. When the recent studies are examined, finite element applications are performed to material which has hyper-elastic behavior. The sealing performance of the



new geometry designs was investigated by analysis with finite element method before prototyping. The results such as stress, deformation, contact pressure, etc. in the seal and shaft contact area were investigated during the studies.

It was found that the stress concentration in the contact area of the NBR radial lip seals were investigated in the study of Chung Kyun Kim and Woo Jeon Shim and it was concluded that the stress concentration were changed by the spring force and design parameters such as seal geometry, oil side angle, air side angle and spring position [2].

When the study of Xue-Guan SONG, Lin WANG and Young-Chul PARK is examined, 4 new seal models are formed for different loading conditions and different interferences in NBR radial lip seal. In these models, the maximum contact pressure change in the contact area and if it is greater than internal pressure it prevents leakage. The study of Xue-Guan SONG, Lin WANG, Young-Chul PARK 's the study of Chung Kyun Kim and Woo Jeon Shim showed that loading and geometry affect sealing performance of the seal [3].

How sealing performance is affected by α and β angles (see Fig. 1) is already studied, but in this study, how sealing performance is affected by lip geometry is investigated by using frictional torque values by using the stress values generated from the friction between seal and shaft.

2. Numerical Method and Verification

Hyperplastic modeling methods were used because of the hyperplastic behaviors of material when numerical model was formed. In this study, radial lip seal and shaft having 40 mm in diameter were modelled with two-dimensional asymmetric modeling instead of 3D modelling which is required more resource. As shown in Figure 1, the finite element model which was built in the ANSYS software and boundary conditions of seal and shaft were given. Then, the Mooney-Rivlin hyperelastic material model was chosen to characterize the seal. Mooney Rivlin is very popular in the elastomer modelling because Mooney Rivlin assumes that the material is homogeneous, isotropic and incompressible.

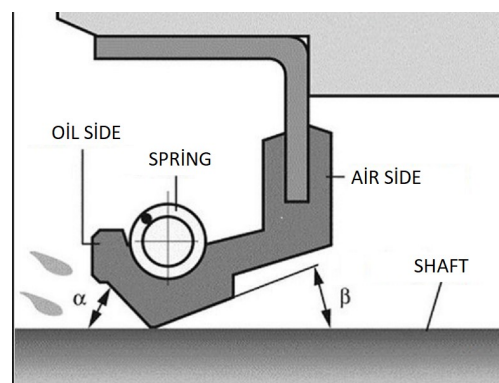


Fig. 1: 2D view of radial lip seal and shaft

After the FEM model (Fig. 2) is established, a uniformly distributed radial force which is caused by garter spring and shaft seal tightness, was applied. This force data was obtained by previous experiments which include NBR material [4]. In addition, the results are confirmed by the results of this previous experiment's data.

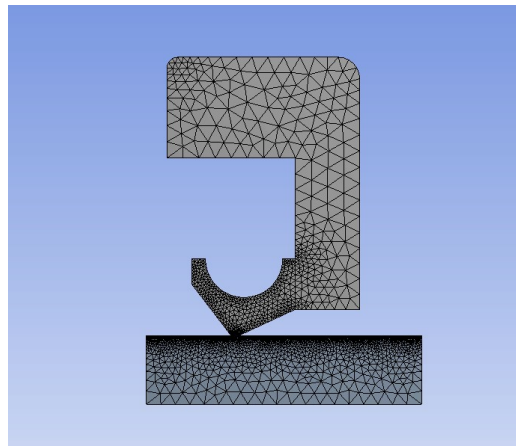


Fig. 2: FEM model of Lip seal and shaft

In order to verify model, some assumptions were made regarding previous experimental work [4]. The internal pressure which is applied by oil side, was modeled as 0.05 MPa and the external pressure was accepted as 0.1 MPa, provided that the angles α and β of the lip remain within the known valid ranges. Frictional contact is defined due to shaft rotation at the contact area and according to result of the researches, friction coefficient is defined as $\mu=0.1\dots0.01$. As a result of the conducted analysis, stress values at the contact area which are caused by radial force, were examined and average stress values were obtained and then frictional torque was calculated using average stress at the contact area. Calculated frictional torque values were verified with experimental data of the “Experimental Study of Friction torque of Rotary Lip Seals” [4]. Also calculations are shown below;

$$T_f = \sigma_{avg} \times L \times \pi \times D \times r$$

$$\sigma_{avg} \text{ (average contact stress)} = 7 \text{ MPa}$$

$$L \text{ (Contact width)} = 0.3 \text{ mm}$$

$$D \text{ (diameter of the shaft)} = 40 \text{ mm}$$

$$r \text{ (radius of the shaft)} = 0.02 \text{ m}$$

$$\mu \text{ (friction coefficient)} = 0.05$$

T_f is found as 0.26 Nm and it was seen that the frictional torque value in the experimental study [4] was 0.24 Nm. When the analysis performed is compared with experimental data, the error rate is 8%.

In order to model The NBR radial lip seal whose boundary conditions was determined above, 2 parameters option was chosen for Mooney Rivlin equation [3, 5-7]. These two parameters, $C10=1.87$ and $C01=0.47$, were calculated by uniaxial tensile and uniaxial compression test data. Also the Poisson ratio was assumed as 0.49.

3. Geometry

It is known that sealing performance of the lip seal is influenced by many parameters but researches show that the geometry of the contact area is deformed by applied force and it affects the strain values at contact zone. When the study of A. Tasora, E.Prati, T.Marin [8] is examined, the magnitude of the applied force appears to influence the stress values in the contact zone. It means that it affects friction torque at contact

area. In this study, friction torque and stress values at contact area, were investigated based on the change in geometry of the seal lip. In the contact zone, the parameters that affect friction torque primarily are thickness of the lip (d) and eccentricity between garter spring and seal lip (e) (see. Fig. 3). Then models were created with different e and d values, as seen in the Table 1. In commercially available standard seal models values of $e = 0.3$ and $d = 1.3$, are taken as reference values.

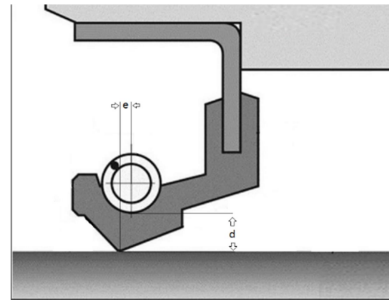


Fig. 3: Dimensional parameters of the lip which were studied; e : eccentricity between garter spring and seal lip d : thickness of lip

Table 1. Models and parameters are designed for analysis

Eccentricity (mm)	Model No
$e=0.15$	Model 2
$e=0.3$	Model 1
$e=0.6$	Model 3
$d=1.3$	Model 1
$d=1.6$	Model 4
$d=1.9$	Model 5
$e=0.15$ $d=1.3$	Model 6

Model 1 is the reference model and in other models, e and d values were changed. Also two parameters (e and d) were combined to form mixed model (model 7) and then in these models friction torque was obtained.

4. Results

The eccentricity and lip thickness values of reference model shown in the Fig. 4, were changed and new model was analyzed. Since the contact pressure is higher than the internal pressure, it can be said that sealing is provided. As seen in Table 2, when eccentricity was reduced from 0.3 to 0.15, friction torque caused by friction between seal and shaft, reduced. But when the eccentricity was increased, friction torque also increases. It means that increase in eccentricity also increases friction torque.

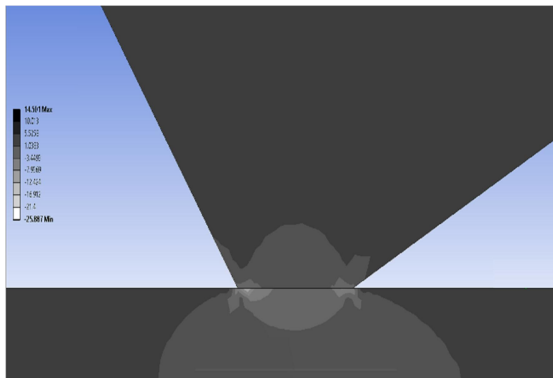


Fig. 4. Reference Model (model 1)

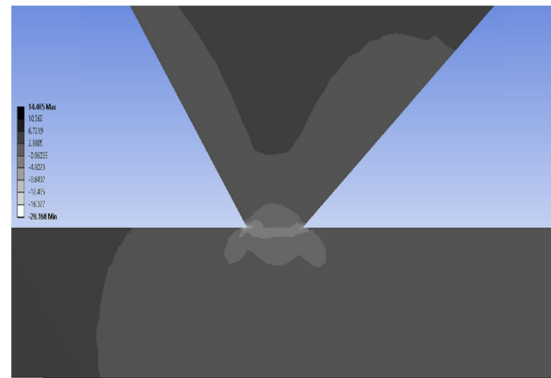


Fig. 5. Model 5 (Lip thickness $d = 1.9$ mm)

Lip thickness which is other geometrical parameter, is at first reduced from 1.3 to a smaller value, but it has been seen that when the lip thickness is reduced from the reference size, α and β angles become out of optimum range so it affects the sealing performance negatively. Therefore models were created by increasing d value (lip thickness) (Fig. 5). Then as a result, it was decided to increase lip thickness from 1.3 to 1.9 mm. In the new model, strain and stress values at the lip decreased, but friction torque increased.

Table 2. Friction torque results of the change in lip geometry

Eccentricity (mm)	T_f (Nm)
$e=0.15$	0.21
$e=0.3$	0.26
$e=0.6$	0.34
$d=1.3$	0.26
$d=1.6$	0.32
$d=1.9$	0.38
$e=0.15$ $d=1.3$	0.49

In the last model, analysis was performed in a mixed model in which the eccentricity was reduced ($e=0.15$ mm), and the lip thickness was increased ($d = 1.6$ mm). In this model, it is seen that the effect of decreasing friction torque is lost as a result of decreasing the eccentricity and the effect caused by the increase of lip thickness is dominant and the friction torque is increased in the model.

5. Conclusion

In this study, the effects of the lip geometry on the sealing performance were investigated and the effects of two parameters were emphasized; Garter spring and seal lip (e) and lip thickness (d). It was seen in the analyzes that when the eccentricity was reduced, the friction torque decreased, but an increase in the friction torque was observed with the increase of the eccentricity and lip thickness.

As a result, researchers have found that reducing the eccentricity improves the sealing performance, but it has been obtained that increasing the lip thickness affects the sealing performance negatively. In the mixed models, we have seen that the advantage of the eccentricity is lost due to the effect of lip thickness and then it becomes the worst model. For the future study, the value or range of values in which the eccentricity have optimum can be found, and new geometric models may be developed that may improve the sealing performance.

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