Calculation and FE analysis for sustainability of seal for ball bearing

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Abstract

Dust and water ingress in bearings can lead to early failure of bearings Contact seals are frequently used to keep contaminants out and lubricant inside the bearing. The sustainability of the seals is dependent on lip contact forces. In this research work, an attempt has been made to arrive at the mathematical model to calculate the sustainability of the seal lip with respect to its resultant contact forces. The results has have been further validated with the non linear FEA approach by using ABAQUS. The Finite element analysis for an NBR seal was conducted using Mooney-Rivlin hyperelastic material model, the Seal was modeled as an incompressible hyperelastic material under an isotropic flow assumption.

The forces were derived and found to be dependent on the following parameters of the lip; original length of lip (L), inclined angle of seal lip (β) and groove angle of inner ring (α). The FEA results have been compared with the analytical results and formulated both are in close correlation.

Keywords: Contact force; FEA; Non linear; Mooney-Rivlin; Hyperelastic NBR material; Original length of lip; Inclined angle of seal lip; Groove angle of inner ring

1. Introduction

Seals play a very vital role in deciding the life of the bearing. Their main function is to prevent ingress of foreign contamination and avoid the leakage of lubricant. For this, contact pressure should be maintained between the seal lip and the inner ring of the bearing. Under the running conditions of the bearing, the seal lip experiences a number of forces acting on it. To achieve the designed bearing life, a constant or optimum pressure needs to be maintained at the contact surface.

The objective is to find the optimum axial force acting on the seal lip. All the forces are derived and then the axial component is taken into consideration. Bearings with bore size 12mm, 15mm, 17mm have been considered for this research and further a mathematical relation has been established between the contact pressure and the various dimensions of bearing[6].

Nomenclature

δ	Deflection in axial direction
β	Initial inclined angle of lip
f	Deformed inclined angle of lip
L	Lip length
l	Elongated lip length
$\in_{y}$	Error in y direction
ψ	f / β
α	Inclined angle of the groove
ω	Coefficient of thermal expansion
ΔT	Temperature gradient
ΔP	Pressure gradient

P_{atm}	Atmospheric pressure
ρ	Density of lubricant
v	Velocity of lubricant
θ	Inclined angle between the lip end and bearing groove
E	Young's modulus of NBR material
I	Area moment of inertia
t	Thickness of seal lip

2. Seal material

The material considered for seals is Nitrile rubber, being widely used in industries. The primary reasons for using Nitrile rubber are low material cost, injection moldable and one-step processing. Nitrile rubber, also known as Buna N or NBR, consist of copolymers of butadiene and acrylonitrile, with acrylonitrile content varying from 20 – 50%. It may also contain various antioxidants.

The low temperature of lubricant and thermal resistance are the major deciding factors for the ratio of butadiene and acrylonitrile in NBR. Other physical properties of NBR include excellent resistance against most lubricants and the metal adhesion property is excellent, providing compact stiffener structure and better sealing [1][5].

3. Analysis of sustainable forces

Various forces act on the bearing seal in its application. The net force on the lip, acting in axial direction, results in contact pressure. This is an important criterion for designing the seal. The resultant force depends on various factors and is the vector addition of the impact force due to lubricant, deflection force of the lip, stretch force, pressure force of lubricant and thermal force. The mathematical form can be represented as,

$$F_N = F_I + F_D + F_S + F_T + F_P$$

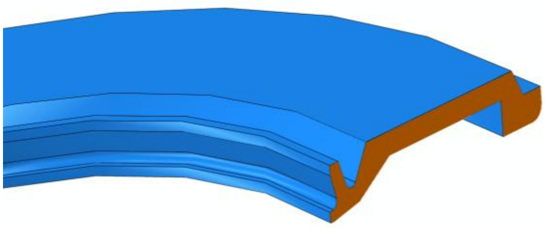

Figure 1 : Sectional view of seal lip with stiffener

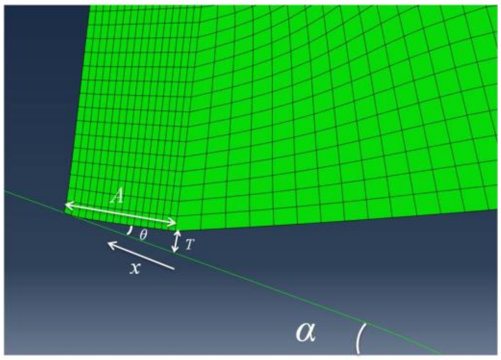

Figure 2 : Cut section showing impact force dimensions

3.1. Force due to impact (F_I)

The contact between the seal lip and the inner of the bearing is not uniform. The edge of the lip makes an angle with the inner and hence creating a venturi effect. The lubricant entering this venturi will have some velocity gradient due to which, an impact force will be acting on the seal lip. The impact force on the seal lip can be calculated as,

$$F_I = \frac{\pi \rho A v^2 T\left[2d + A[cos(\alpha - \theta) + cos\,\alpha . cos\,\theta]\right]^2}{(A - x)\left[2d + (A - x)[cos(\alpha - \theta) + cos\,\alpha . cos\,\theta]\right]}$$

Here, A & d represents length of varying cross section & diameter of lip edge respectively, T is the height of the section at any distance x from the fixed end of cantilever beam.

3.2. Force due to deflection (F_D)

The lip of the seal is considered as a cantilever beam which is fixed at the point where secondary and primary lips meet. The lip is considered as a straight beam which experiences stress after being installed. The axial deflection δ is considered for the extreme point as the deflection of each point is different. Also net displacement δ' is considered at the end point of the lip seal. To derive the axial force, the simple beam theory was applied. The force at distance x from a fixed support can be calculated as,

$$F_D = 6EI\frac{(L-l)}{[3L-x]x^2}(\sin\beta - \sin\psi\beta)$$

To calculate the force it is necessary to find out δ,

$$\delta = (L-l)(\sin\beta - \sin\psi\beta) + \epsilon_y$$

Also, after deflection a small portion of seal lip will remain in contact with the inner ring of the bearing at the free end. Taking this deflected length as l, we will find the force at length $(L-l)$. Hence we also need to calculate l by using simple trigonometry [2].

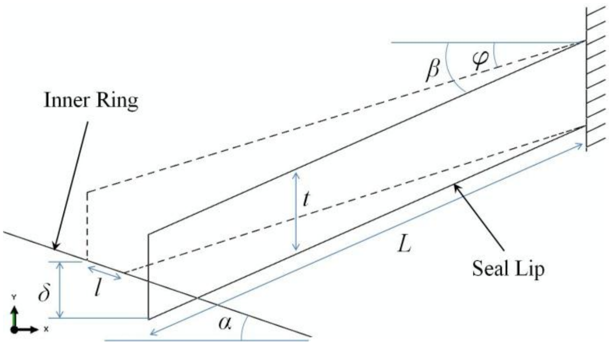

Figure 3 : Schematic diagram of seal lip

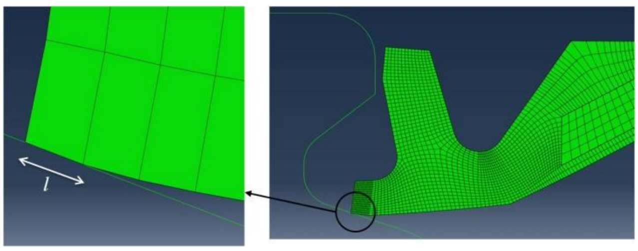

Figure 4 : Deformed seal lip with exaggerated view

3.3. Force due to Stretch (F_S)

When the seal lip is fitted inside the bearing, the outer fibers undergo stretching. The fibers under stretch try to retain their original position, hence exert a force, which can be calculated by,

$$F_S = \int_0^{2\pi}\left[\frac{\pi EL(R+r)(L-L'')}{3t} \times sin\,\psi\beta \times \frac{1}{360}\right]d\theta$$

R & r are the radius of extreme points of the primary lip, L' is the length of the lip after deformation.

3.4. Force due to Thermal stress (F_T)

Temperature in the working environment of bearings can be near to 150˚C and non uniform [3]. The fitment of seals is in such a way that it do not allow further expansion of the material. This induces certain stresses in the seal which can be referred as thermal stresses.

Due to existing temperature difference in the atmosphere and the inside of bearing, thermal stresses will be induced in the seal lip. This will result in a force in the axial direction.

$$F_T = \pi El(r_1 + r_2)(\omega\Delta T)$$

Where r_1 & r_2 are the radii of the extreme point of the deflected length l.

3.5. Force due to pressure gradient (F_P)

Due to the lubricant, there will be pressure gradient at the seal. This pressure, typically higher on the inside, will result in an outward force on the seal. This force can be calculated using the equation,

$$Fp = \pi L(R + r)(\Delta P)\cos\psi\beta$$

4. Results and discussion

The conditions considered for the analysis of the NBR (Nitrile Butadiene Rubber) seal and the dimensions taken are tabulated in Table 1. Figure 5 shows the theoretical results for the net contact force Fnet as a function of length of the lip for various ratios of the deformed angle and initial inclined angle of the seal lip. As the Length of the seal lip increases, the axial contact force of the seal lip decreases but with a gradual slope. The results represented in figure 6 show the increases in the contact force with increase in the initial angle of inclination. The slope is linear and positive. Figure 7 represents the variation in contact force with respect to the thickness of the lip. As the thickness increases, the force increases exponentially. This also makes the lip thickness a very important parameter for designing the seals. Hence to increase the contact force, it is convenient to increase the thickness of the seal lip and hence obtaining an exponential growth.

Table 1 : Dimensions & Properties of standard seal

Diameter of seal lip edge	9.15mm
Thickness of seal lip	0.2mm
Temperature difference	50-70^{o}C
Coefficient of thermal expansion	23*10^{-5}/ ^{o}C
Pressure difference	(0.01-0.015)*P_{atm}
Young's Modulus	2-5MPa
Velocity of impact	150-300 mm/sec

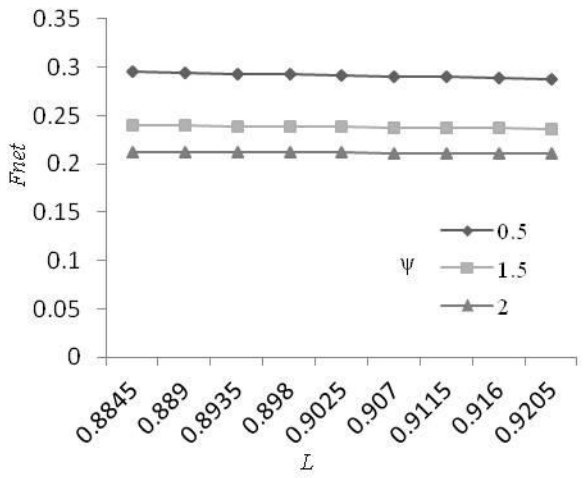

Figure 6 Total contact force as a function of the lip length

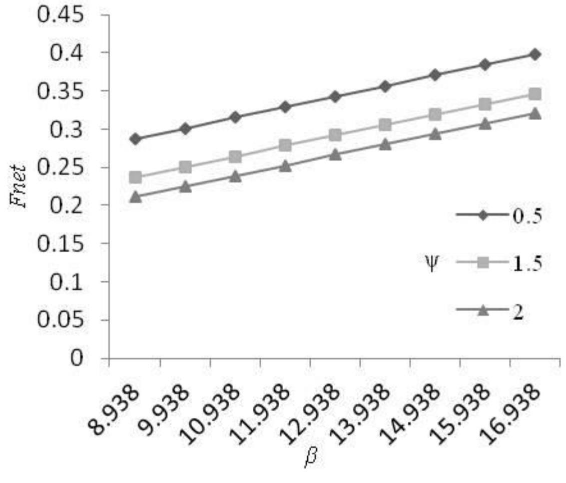

Figure 5 Total contact force as a function of Inclined angle of seal lip

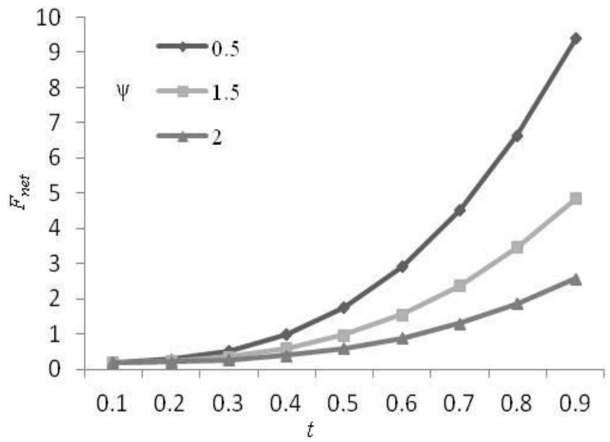

Figure 7 : Total contact force as a function of the lip thickness

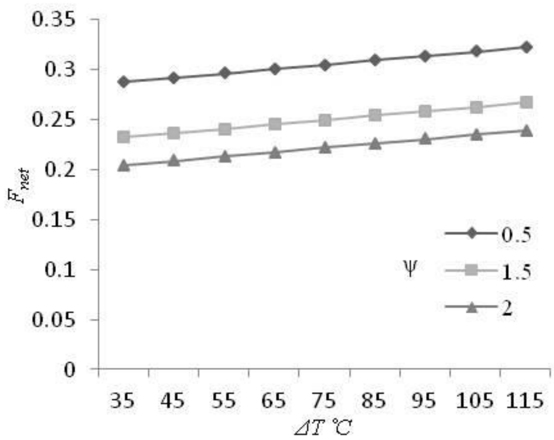

Figure 8 : Total contact force as a function of temperature difference

The results for variation in contact force with change in temperature have been represented in Figure 8. As the temperature difference inside the bearing and atmospheric temperature increases, the net contact force increase. Hence if a bearing is working at a very low temperature, the seal must be designed accordingly to perform in optimum way.

5. FEA approach

To conduct the FEA analysis, an advanced methodology has been conducted using the FEA tool ABAQUS with a non-linear standard model. An axis symmetric model, as shown in figure 9, is constructed to reduce the intricacy of the modeling steps and bring down the iteration time and resources used to a minimum. The inner and outer rings of the bearing are considered rigid with six degrees of freedom restricted.

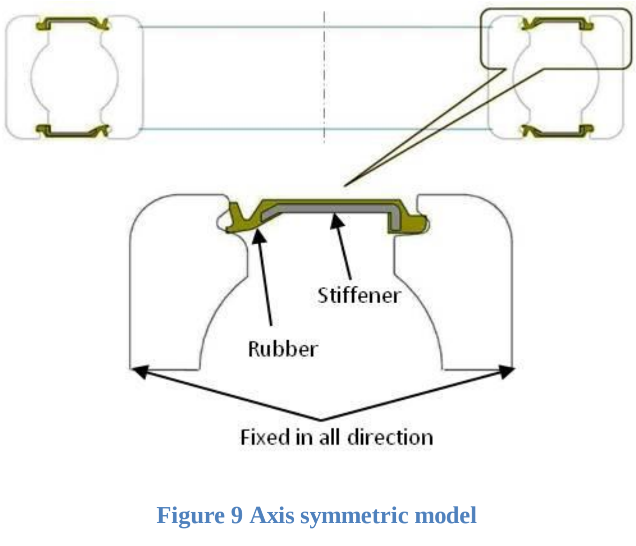

Figure 9 Axis symmetric model

The basic properties of an NBR seal were considered and nonlinearity in material, boundary and geometry was taken. As the Young's Modulus for NBR cannot be defined because of the high non-linearity existing in NBR material, the Mooney-Rivlin model approach has been used and Mooney-Rivlin constants were used instead of material properties. The coefficients were obtained from uniaxial test data [3] [5].

Surface to surface interaction property has been given between the seal and the groove at inner and outer of the ring whereas for seal itself, self contact has been considered. All the contacts are assumed frictionless. A gradual interference fit approach has been used to provide realistic fitment conditions. To restrict the relative motion between seal and stiffener, Tie constraint has been used. The meshing type used is four node bilinear axis symmetric quadrilateral elements with hybrid formulation. Total number of element account for 4235. A partition is also made, separating the contact region from the rest, so as to refine the mesh and hence obtain improved results. A meshed cross sectional view has been shown in figure 10.

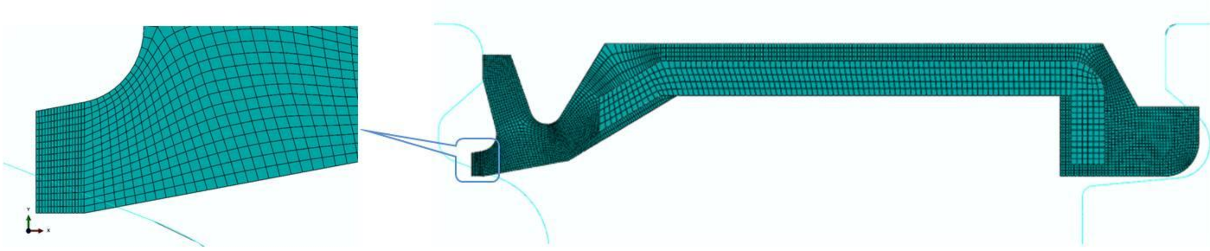

Figure 10: Meshed Seal with Quadrilateral element

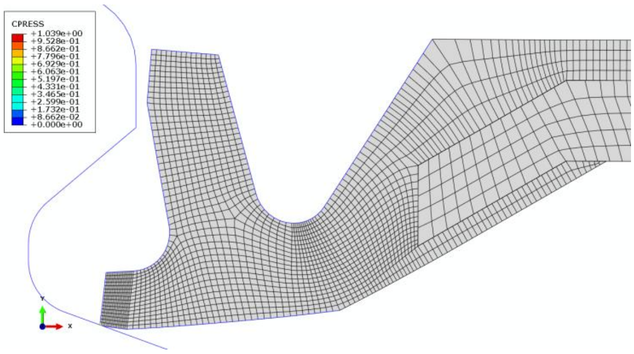

Figure 11: Contact Pressure at Seal lip

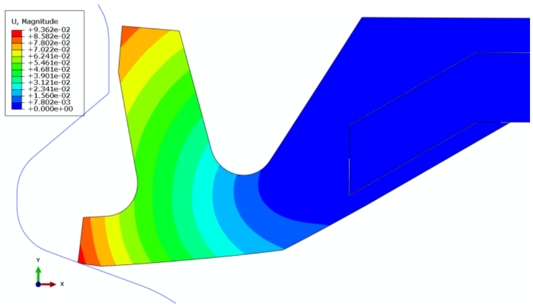

Figure 12: displacement of seal

Using the FEM technique, the forces were calculated in the axial direction as a function of the initial seal dimension [4]. The maximum contact pressure on the seal is experienced on its lip. Figure 11 shows the location where the pressure is found to be acting. Since the deflection force is the main governing force of contact pressure, the deflection will have great influence on the net pressure and so should be maximum at the point where pressure is maximum i.e. the edge of seal lip. The colored counters in figure 12 show the respective displacement of points and hence validate the above expected result.

6. Conclusions

It is concluded that the contact pressure of seal lip depends on various parameters of the lip which include length of the lip, inclined angle, lip thickness and the contact lengths. With the help of this model, a wide scope for contact pressure optimization has been established.

7. Future scope

1. Knowing the contact pressure, research can be done in order to create low torque bearings.
2. Design criteria can be established to optimize the contact pressure without compromising with the life of bearing.
3. Hinge and groove can be considered for further optimization of seal design.

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