A Numerical Prediction of Particle Contaminations in the Seal Design for a Bearing by One Way Coupling

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Abstract: In recent years the tightening of fuel economy regulations and the trend towards environmentally friendly products increase demand for compact, lightweight and more efficient automotive transmissions. At the same time, the bearings in the transmission are always more often subject to harsh lubrication conditions. Contaminants in the lubricant have a strong life-reducing effect on rolling bearings. Past practices for protecting bearings from contamination include the use of lip seals make direct contact with raceway, but this solution lead to high frictional torque in the bearing system. Another practice includes the use of labyrinth isolators, but this solution also is not sufficient to prevent premature bearing failures due to contaminants. Therefore, a simple analysis method has been developed for predicting the penetration of contaminants from outside of bearing with non-contact lip seal. The method is fit for labyrinth isolator based on the geometry of the lip seal and the raceway that show the passing through the contaminants during bearing operation, which is driven largely by the flow of the lubricants. The system is analyzed with computational fluid dynamics where the lubricant is modeled with solid grids, while the lip seal and the raceway are represented with stationary and rotational wall. Through a better understanding of the mechanisms that lead to reduced bearing life owing to the effect of such factor, we can improve bearing design and operation.

Keywords: Contamination, Labyrinth, One Way Coupling, Co-simulation, Design Optimization.

1. Introduction

The ulterior motive of automotive original equipment manufacturers is to increase power efficiency and reduce the environmental impact. They also strive to meet the CO2-targets. In order to achieve these goals, there are three of the major directions of developments, which are a reduction in friction (carbon taxes), a light-weight design (carbon taxes) and an increase in lifetime (maintenance costs).

Meanwhile, the transmission bearings are always more often subject to harsh lubrication conditions, for example when low-viscosity oil is used, and are expected to withstand surface-originated flaking (flaking that starts at the surface). The presence of abrasion particles of hardened steel in the transmission oil is quite usual considering the numerous metal-to-metal contacts between the gearbox pinions. These particles can be source of indentations on the raceway surface when it squeezed between the ball and the raceway. In the conventional theory, surface flaking is due to the stress concentration on the indentation edges that firstly initiate to cracks and then to flaking. Surface roughness, particle denting and contamination marks found on bearing raceways can induce stress concentrations and facilitate surface-initiated fatigue. The lubricant film developed at the dent and related local surface stresses are also significant to the crack initiation mechanisms.

Lip seals and contact seals help minimize the amount of dirt and abrasion particles that can contaminate bearing lubricant. These solutions, however, have some drawbacks. Lip seals invariably wear out well before the bearing fails, and sealed bearings inherently foreshorten the life of a bearing to the service life of the contained grease. In addition, contact seals lead to high frictional torque in the bearing system. One successful approach to keeping dirt and abrasion particles out of an operating bearing is to install a labyrinth-type non-contact seal. But this solution also is not sufficient to prevent premature bearing failures due to contaminants. Therefore, a simple analysis method has been developed for predicting the penetration of contaminants from outside of the bearing with non-contact lip seal. And the sealing performance of the labyrinth type seal has been evaluated by endurance test.

2. Effect of contamination in bearings

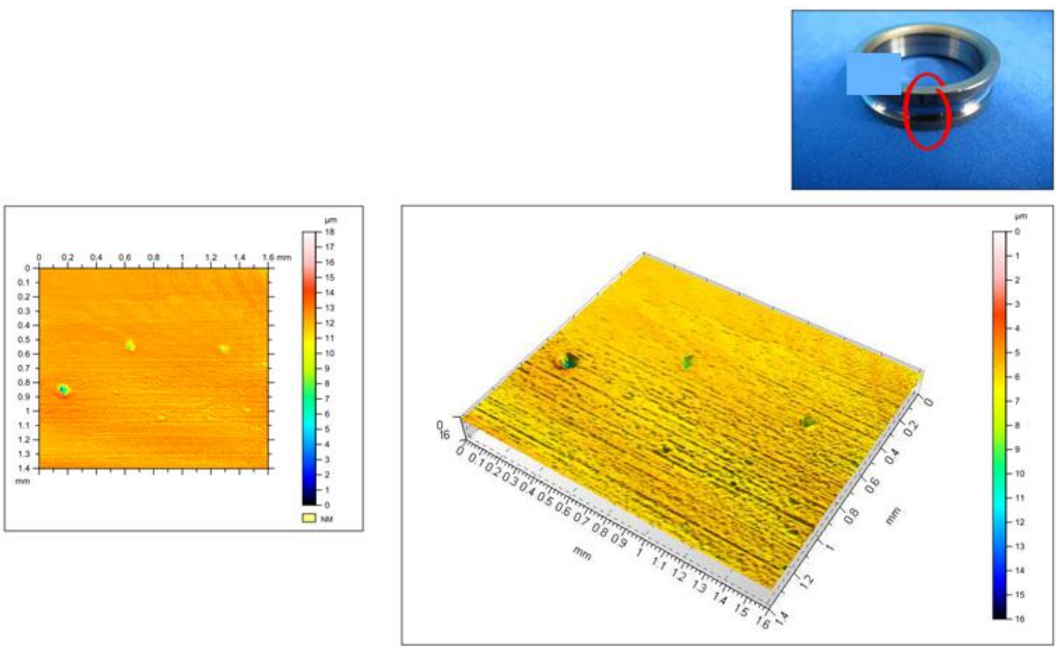

Figure 1. Example of Surface Indentation by Hard Particles.

Hard particles in the lubricating oil can cause indentations in raceways and rolling elements when they get into the rolling contact. When over-rolled such indentations provoke increased stresses which result in surface-originated flaking (flaking that starts at the surface). Figure 1 show an example of the surface indentation by hard particles.

Hard particles in the lubricating oil can be source of indentations on the raceway surface when it squeezed between the ball and the raceway. In the conventional theory, surface flaking is then due to the stress concentration on the indentation edges firstly leading to cracks and then to flaking. Surface roughness, particle denting and contamination marks found on bearing raceways can induce stress concentrations and facilitate surface-initiated fatigue(Figure 2).

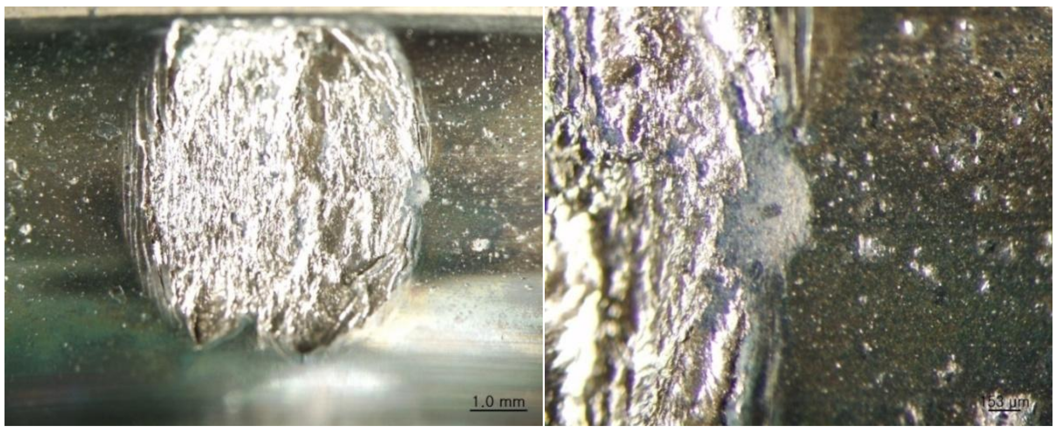

Figure 2. Example of Surface Failure due to Contamination.

3. Friction and frictional torque

The resistance to rotation of a rolling bearing is composed of rolling, sliding and lubricant friction. Rolling contact friction occurs when the rolling elements roll over the raceway; sliding friction occurs at the guiding surfaces of the rolling elements in the cage, at the lip guiding surfaces of the cage and, in roller bearings, at the roller faces and the raceway lips. Lubricant friction is the results of the internal friction of the lubricant between the working surfaces as well as its churning and working action.

3.1 Frictional torque by catalogue method

The total friction of a bearing – that is the sum of rolling, sliding and lubricant friction – is measured as the resistance the bearing exerts against its movement. This resistance represents a moment and it is called the frictional torque M.

$$M = M_0 + M_1$$

where

M = Total frictional torque

M_0=Frictional torque as a function of speed

M_1= Frictional torque as a function of load

Fictional torque as a function of speed for $\nu \cdot n \geq 2000$:

$$M_0 = f_0 \cdot (\nu \cdot n)^{\frac{2}{3}} \cdot d_M^3 \cdot 10^{-7}$$

Fictional torque as a function of speed for $\nu \cdot n < 2000$:

$$M_0 = f_0 \cdot 160 \cdot d_M^3 \cdot 10^{-7}$$

Frictional torque as a function of load for ball bearings:

$$M_1 = f_1 \cdot P_1 \cdot d_M$$

where

n=Operating speed

f_0=Bearing factor for frictional torque as a function of speed

f_1=Bearing factor for frictional torque as a function of load

ν=Kinematic viscosity of lubricant at operating temperature

P_1=Decisive load for frictional torque

d_M=Pitch circle diameter of the bearing

3.2 Frictional torque calculation by numerical method

Friction calculation using Bearinx® is based on an analytical model developed by the Schaeffler Group. This analytical model is characterized by significantly higher model quality than the empirical catalogue method for friction calculation. In addition to the influencing parameters of the catalog method like load, speed, lubricant and temperature further parameters are taken into consideration. Particularly there are the tilting of the bearing rings due to the elasticity of the shaft, taking account of the detailed internal bearing geometry (contact angle, osculation, profiling, rib geometry, ...), the bearing clearance, the lubricant conditions (Oil or grease lubrication and churning losses) and the sealing.

According to the calculation results, maximum torque of 6207 bearing with contact type seal was higher than that of 6207 bearing without sealing device at the given operating condition(Figure. 4). So the friction torque by the contact type seal is an important factor for torque calculation and must be considered.

Table 1. bearing data and operating condition for frictional torque calculation.

Bearing & Lubricant Data	Values	Remarks
Inner diameter	35 mm	Deep Groove Ball Bearing (6207)
Outer diameter	72 mm	
Width	17 mm	
Reference diameter	53.4 mm	
Number of rolling elements	9	
Rolling element diameter	11.112 mm	
Lubrication type	oil / grease lubrication	
Viscosity at 40°C	22 mm²/s	ISO-VG 22
Operating condition	**Values**	**Remarks**
Inner ring rotating speed	3000 rpm	
Cage rotating speed	1188 rpm	
Applied axial load	765 N	
Applied radial load	7650 N	

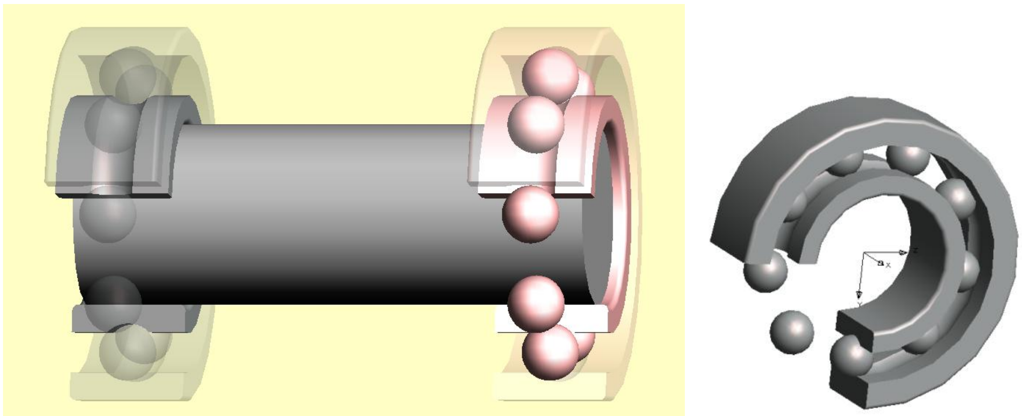

Figure 3. Single shaft and bearing model in Bearinx for frictional torque calculation.

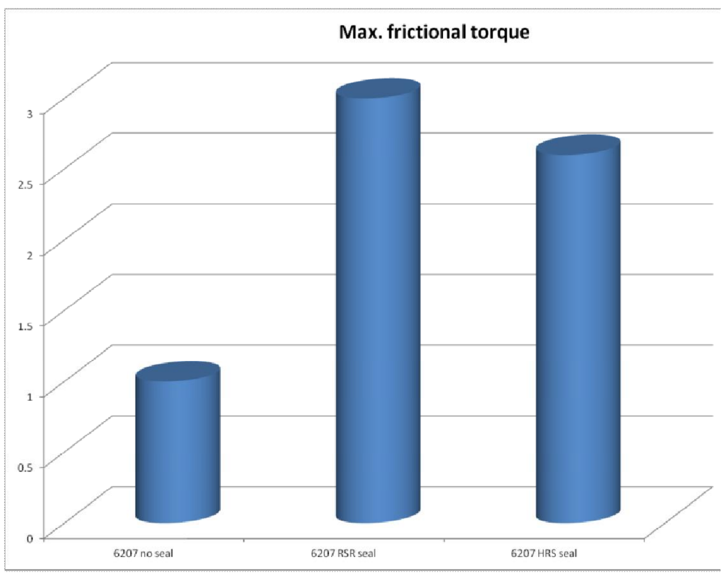

Figure 4. Comparison of max. friction torque with respect to seal types.

4. New seal design for low frictional torque in bearing

4.1 Seal design concept

A seal has two basic tasks: keeping the lubricant in and keeping contaminates out of the bearing system. This separation must be accomplished between surfaces in relative motion, usually a shaft or bearing inner ring and a housing. The selection of a seal design depends on the type of lubrication system employed and the type and amount of contaminates that must be kept out. Other considerations in the decision must include speed, friction, wear, ease of replacement, and economics. Bearings run under a great variety of conditions, so it is necessary to judge which seal type will be sufficiently effective in each particular circumstance.

The concept of new seal applied to the bearings in transmission must correspond to recent requirement for increasing power efficiency and reducing the environmental impact. So it has a characteristic of lower frictional torque than conventional seal. This should be possible to realize by use of non-contact type seal and the sealing performance should be improved by labyrinth effect to prevent ingress of contamination and keep lubrication in bearing.

This new seal for low frictional torque is applicable for high efficiency automobile transmissions as like hybrid car, dual clutch transmission(DCT), continuous variable transmission(CVT) and so on.

4.2 Labyrinth seal

Labyrinth seals consist of an intricate series of narrow passages that protect well against dirt intrusion. A type of mechanical seal that provides a tortuous path to help prevent leakage. For labyrinth seals on a rotating shaft, a very small clearance must exist between the tips of the labyrinth threads and the running surface. Many gas turbine engines, having high rotational speeds, use labyrinth seals due to their lack of friction and long life. Figure 5 shows the mechanism of labyrinth effect and various example of labyrinth type seals in bearings.

4.3 Labyrinth type seals for low frictional torque

Figure 6 shows the new type seals. These design has been based on the concept of new seal applied to the bearings in transmission. It has been considered the more complicated flow path due to the labyrinth and reservior to prevent foreign particles.

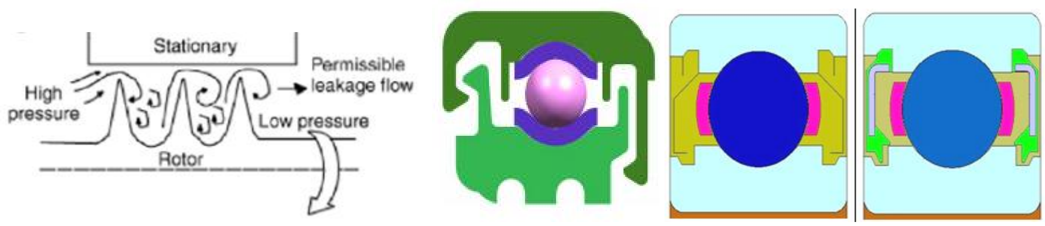

Figure 5. Mechanism of labyrinth effect and example of labyrinth type seals.

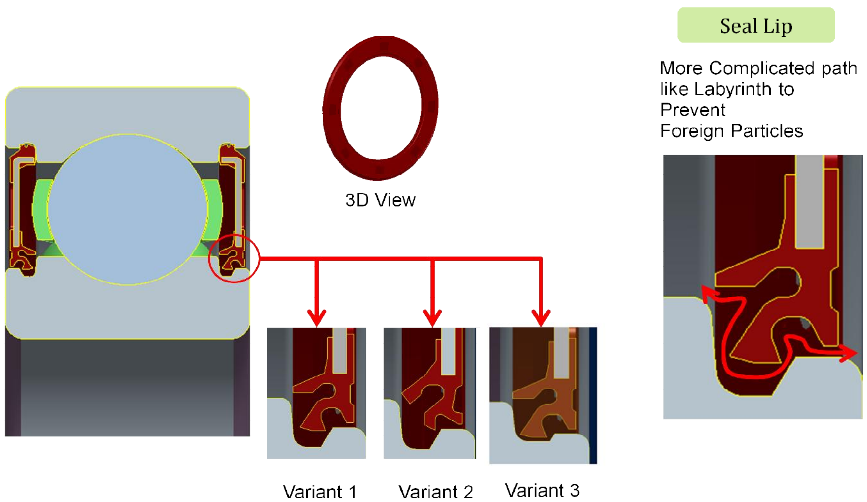

Figure 6. New concepts of seal design.

5. CFD analysis of internal oil flow in bearing

In order to better understand the oil flow inside a ball bearing, internal oil flow has been numerically investigated for inner geometry of ball bearing by CFD technique. The CFD simulation has been conducted by STAR CCM+ v7.

Figure 7 shows the CAD model of the bearing and it has been used to create the domain of fluid volume. And it is also shown the used half-symmetric model for CFD in Figure 7.

Table 2 gives the modeling data and operating conditions for CFD simulation and that is the same as the condition for frictional torque calculation by Bearinx.

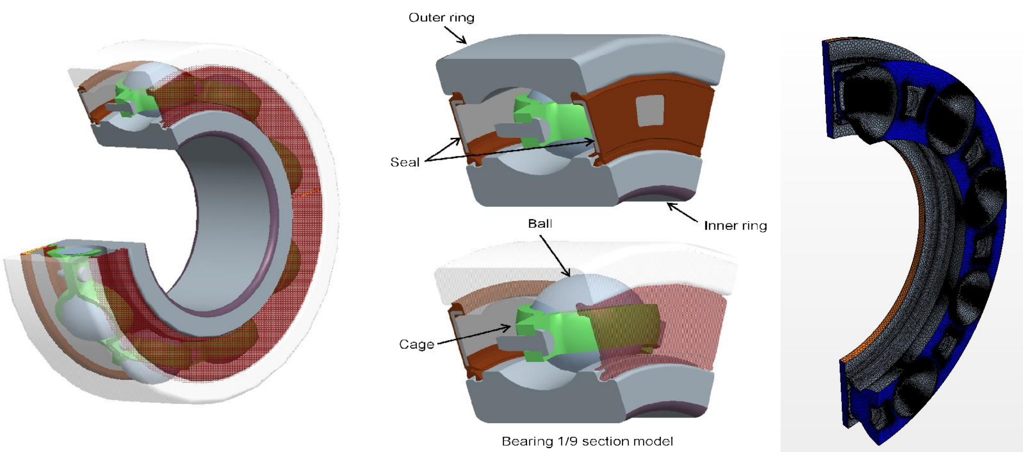

Figure 7. CAD model and half symmetry model for CFD.

Table 2. Modeling data and operating condition for CFD simulation.

Modeling Data	Values
Pre/Post processor / Solver	STAR-CCM+ 7.04
Mesh type	Polyhedral mesh
Number of Meshes	1.2 million
Physics	Constant Density Three Dimensional Liquid Segregated Flow Turbulent
Operating condition	**Values**
Inner ring rotating speed	3000 rpm
Cage, ball rotating speed	1188 rpm
Outer ring, Seal	Fixed all DOF

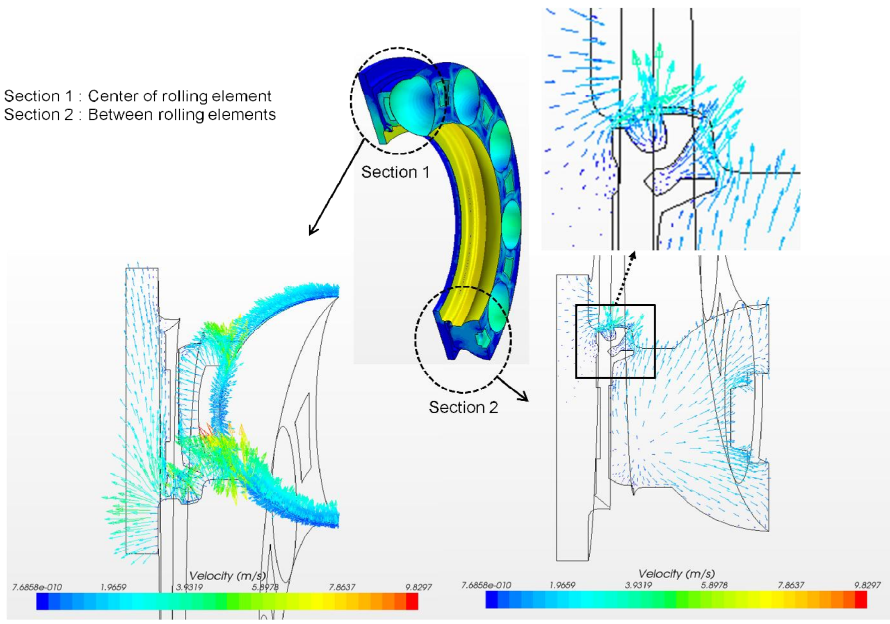

Figure 8. CFD result – half symmetry model, velocity.

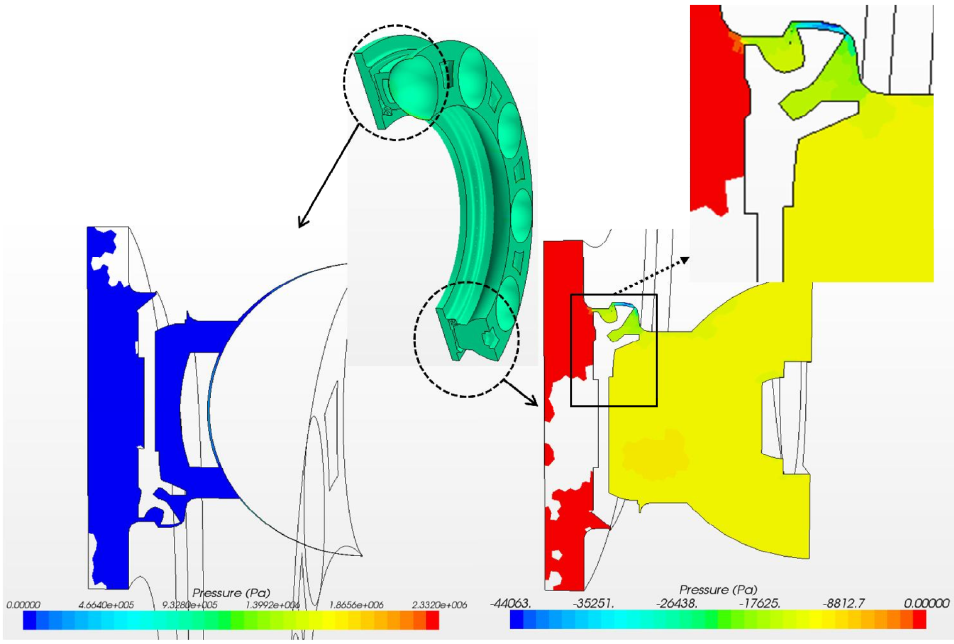

Figure 9. CFD result – half symmetry model, pressure.

The simulation results in Figure 8 and Figure 9 show well the labyrinth effect and the relation of the pressure. On section 1 where is the same plane the rolling element center is, oil flow tend to go outward from bearing inside because of the higher pressure of the inner volume of the fluid which is occurred by the orbital movement of rolling element and cage. On the section 2 where is the middle plane between rolling elements, the oil flow of outer volume tend to go inside of the bearing and the oil flow of inner volume tend to go outward. This conflict on the section 2 should be caused the prevention of the particle inclusion into the bearing.

6. FEA for seal deformation and labyrinth change in bearing

From the flow simulation result, the quantitative pressure value applied to the seal lip has been estimated. These values have been used for the input value in the FE analysis, the seal deformation and the labyrinth change due to oil pressure have been investigated by Abaqus standard 6.12-1.

According to the result, the deformation of the second lip of the seal reduces the gap between inner ring groove and seal lip. So it make difficult to intrude the particle from outside.

For the better understanding and accurate prediction of the interaction between seal and fluid, a co-simulation by Abaqus standard/STAR CCM+ has been planned for the future.

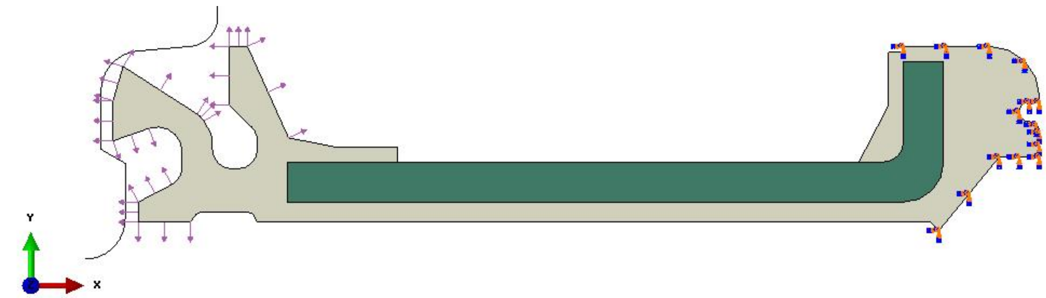

Figure 10. FE model – boundary condition and load condition.

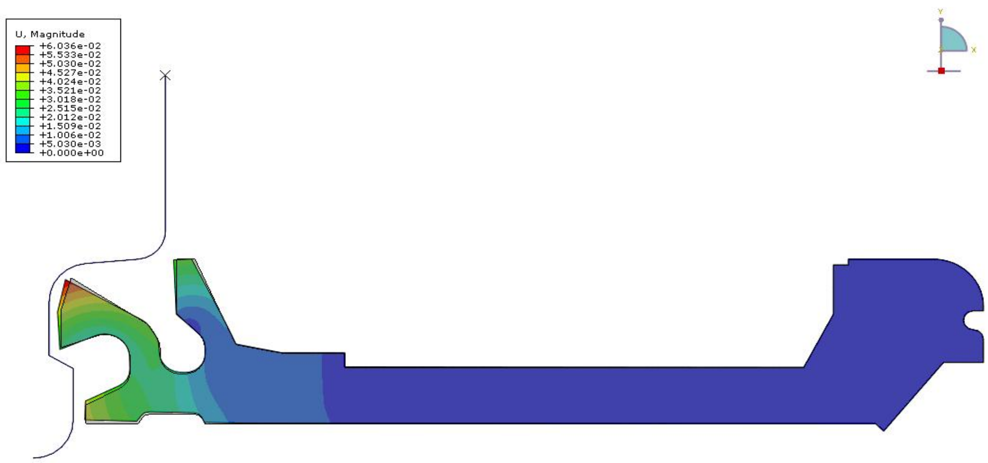

Figure 11. FE result – deformation due to oil pressure (by one way coupling).

7. Endurance test to verify sealing performance

It has been investigated to verify the sealing performance by endurance test for the proposed seal designs.

Table 3. Test conditions

Bearing designation 6207	Values
Radial load applied, Fr	7,650 ± 153N
Axial load applied, Fa	765 ± 50N
Rotational speed	3,000 ± 60 rpm constant
Method of lubrication	Oil bath type
Oil temperature	120 ± 5°C
Quantity of foreign particles	200mg/*l*
Size of foreign particles	0 ~ 100 ㎛

Table 3 shows the test conditions for variant 1 and variant 2 type seal. At a time two bearings are mounted in the test rig. When a bearing in one side has been occurred failure, the bearing in the other side also should be disassembled even though the bearing has not been failed. Table 4 shows the test result for variant 1 and variant 2 type seal. Each test has been performed for 4 different target running times relative to theoretical bearing life (L_{10}) according to ISO 281 as shown below.

Table 4. Test results for variant 1 and variant 2

Bearing No.	Status	Target running time relative to L10	Failure mode	Remarks
1	S	0.2		Variant 1
2	S	0.2		
3	S	0.5		
4	S	0.5		
5	F	1	IR Spalling at 0.46	
6	S	1		
7	S	Sudden Death		
8	S	Sudden Death		
1	S	0.2		Variant 2
2	S	0.2		
3	S	0.5		
4	S	0.5		
5	S	1		
6	S	1		
7	F	Sudden Death	IR Spalling at 1.09	
8	S	Sudden Death		

For the variant 1 there was a failure for bearing no. 5 which was inner ring spalling at 0.46. But other cases were ok. For the variant 2 there was a failure of bearing no. 7 which was inner ring spalling at 1.09. But other cases were ok. Figure 12 shows the whole test results for the variant 1 and variant 2. Test for the variant 3 has been planned but not performed yet. The results shows the effect for the new type seal has not quite satisfied but it shows the possibility to reach the target of required life without failure by optimization of the labyrinth.

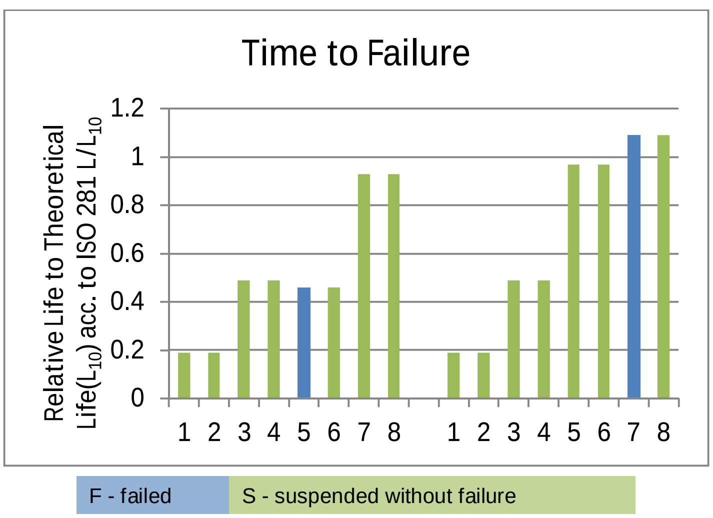

Figure 12. Test results.

8. Conclusion

For investigation of the labyrinth effect between seal lip and inner ring groove, a simple analysis method has been developed. It showed the interaction of the fluid and seal lip due to the fluid flow by one way coupling. It gave a hint to understand the sealing mechanism of labyrinth and qualitative estimation of the sealing performance.

Endurance test for the proposed seal design(variant 1 and variant 2) has been conducted to evaluate the sealing performance. The results were not satisfied but showed a possibility to improve the sealing performance by optimization of the labyrinth.

For an accurate prediction of the interaction between seal and fluid, a co-simulation by Abaqus standard/STAR CCM+ has been planned for the future. This will become the best method to evaluate new seal design, so it will reduce the effort for the preparation of the test and the cost of sample production of new design seals.

9. References

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