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# STUDIES BY FINITE ELEMENT METHOD APPLIED ON FACE SEAL

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**Abstract**. *The paper presents a FEM (finite element method) analysis of a face seal. In the first part some theoretical aspects of this kind of seals are presented. The next section emphasizes the FEM model, its constrains and external loads. The analyses were run for different working conditions (temperature, fluid pressure). In the third part the stress and strain state is shown while some conclusions are derived from the results. In the last section some peculiar aspects of contact region are revealed.*

**Key Words**: *Finite Element Method, Face Seal*

## 1. INTRODUCTION

Mechanical face seals are used for sealing fluid at places were a rotating shaft enters an enclosure. Figure 1 shows schematically the configuration of a mechanical seal. The rotating seal is fixed to the shaft and rotates with it, whereas the stationary seal is mounted on the housing. The secondary seal (O rings) prevents leakage between the rotating shaft and the rotating seal, and the housing and the stationary seal, respectively. The rotating seal is flexibly mounted in order to accommodate angular misalignment and is pressed against the stationary seal by means of the fluid pressure and the spring. The primary sealing occurs at the sealing interface of both the seal faces, while the rotating face slides relative to the stationary face. For proper functioning of the mechanical seal a fluid film is maintained between the faces. In the configuration of figure the seal fluid may also act as a lubricant.

The applications of mechanical seals are numerous. The most common example of application is in pumps for chemical industry. Also propeller shafts in ships and submarines, compressors, for conditioners, of cars and turbo jet engines and liquid propellant rocket motors in aerospace industry require mechanical face seals.

The mechanical face seal has become the first choice for sealing rotating shafts operating under conditions of high fluid pressure and high speeds. The reason for this is lower

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leakage, less maintenance and a longer life. A disadvantage of face seals is that when they fail, they do so completely.

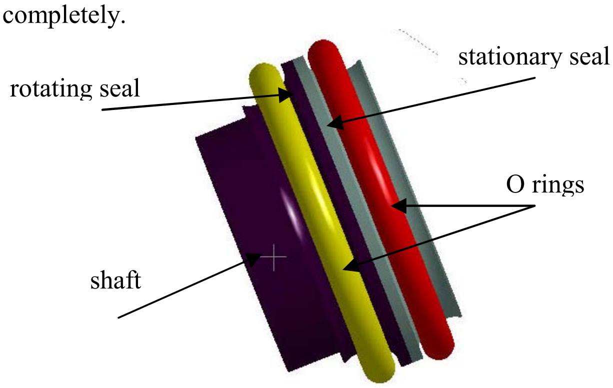



Fig. 1. Mechanical Face Seal

Due to increasing technical environmental requirements, the operational conditions are becoming more severe. Face seals have to operate at higher pressure and higher speeds, so sufficient fluid pressure in the sealing interface is vital if excessive wear, friction and temperature rise (frictional heating) are to be avoided and a long seal life is to be ensured. However, a too thick fluid film is unfavorable with regard to leakage, as this is proportional to the cube of the film thickness. Due to environmental demands, leakage must be minimized by reduction of the clearance between the faces. From the above it is clear that the demands regard to optimum sealing are contradictory. Ideally, a mechanical face seal should operate with a fluid film as thin as possible, to reduce the leakage and to restrict wear.

Advances in a system of computer software based on FEM help in the understanding of the operation and design of the mechanical face seal. Many of the tools or components needed to advance this understanding have now been developed and can be combined into models that give good if not complete, understanding of the whole seal system. These tools are used to explain a mechanical seal operation and to help delineate the regimes of the various known lubrication mechanisms that act in the mechanical face seals. Further, these tools allow analyses of a mechanical seal design problem as a system design problem. This is important because in the mechanical face seal the phenomena of heat transfer mechanical distortion, surface roughness effects, and dynamics have as much influence as the fundamental lubrication theory.

Wear and failure of mechanical seals may be critical in certain applications and should be avoided. A relatively large misalignment between the seal faces is the most likely cause for intermittent contact and the increased friction that eventually brings failure. Adjustment of the seal clearance is probably the most readily implemented method of reducing the relative misalignment and of eliminating seal face contact during operation.

A comprehensive design that takes into account all the information such as seal face geometry, materials, heat transfer, mechanics, system dynamics and empirical data, promotes a long seal life. This is, however, by no means, an easy task, and in many cases a great amount of the information is lacking.

The temperature may vary as a result of some physical phenomena or changes in operating conditions. Particularly, when seal faces are in contact because of large relative misalignments the temperature could still remain low because of the cooling effect of excessive leakage.

## 2. THE FEM MODEL

Using the finite element method (FEM) allows reducing the number of simplified hypothesis used in traditional methods. Computer software based on FEM help in the understanding of the operation and design of the mechanical face seal. Many of the tools or components needed to advance this understanding have now been developed and can be combined into models that give good, if not complete, understanding of the whole seal system. These tools are used to explain a mechanical seal operation. Further, these tools allow analyses of a mechanical seal design problem as a system design problem. This is important because in the mechanical face seal the phenomena of heat transfer mechanical distortion, surface roughness effects, and dynamics have as much influence as the fundamental lubrication theory.

The analysis was run in ANSYS. The model has 5782 four nodded tetrahedron elements and 11823 nodes. The mesh is presented in Figure 2. The analysis was run for different working temperatures, [-50 … +150] ºC, and external loads.

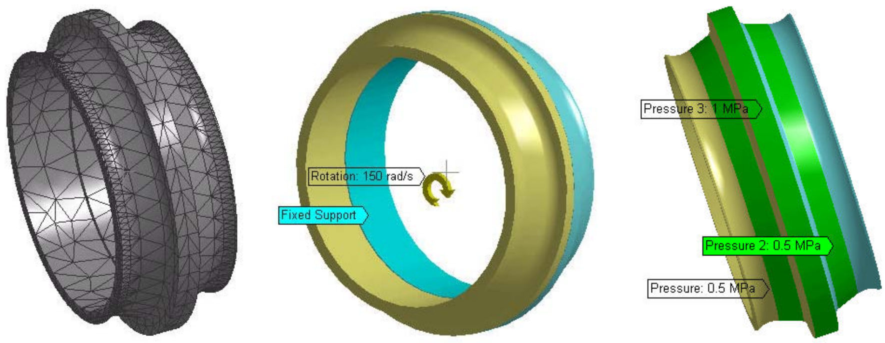



Fig. 2. Mesh, Constrains and External Loads

The imposed constraints are: the stationary seal is fixed on the inner diameter and the rotating seal has an angular velocity of 150 rad/s (Fig. 2). The external applied loads are: a pressure of 0,5 MPa (this value is required for good functioning of the seal and must be generated during the mounting process, [4]) on the contact surface between the O rings and the seals and the fluid pressure of 1 MPa, [2], (see Fig. 2).

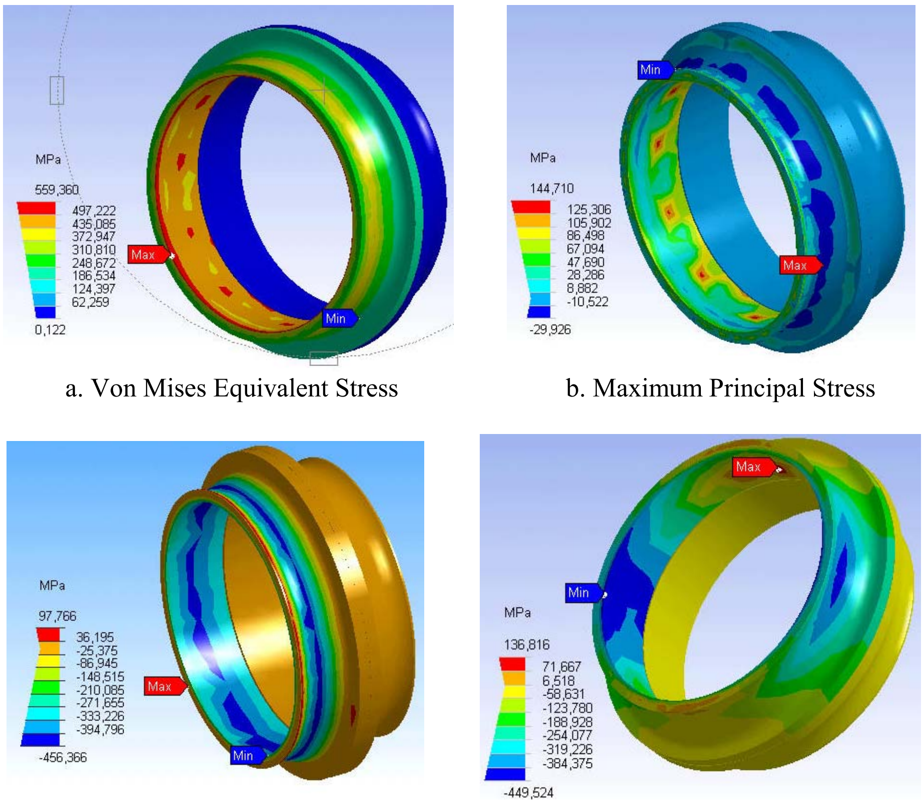
The operating limits for this type of face seal [2] are:

- the diameter shaft: 30 – 600 mm;
- the maximum fluid pressure: 1MPa;
- temperature range: -50 … 150ºC.

A frictional interaction is defined between the two seals. The lubrication regime is modeled in the contact region with different values of frictional coefficient. The chosen values were depicted in order to simulate the dry contact and hydrodynamic regime. The stress and stress strain state is insignificantly influenced by the value of the coefficient of friction.

## 3. STRESS AND STRAIN STATE

Fig. 3 presents the stress map for the assembly at 150 ºC. Similar maps are obtained for the temperature range, but with different values. A good similarity can be seen between von Mises stress and strain. A symmetric map for maximum principal stress is shown in Fig. 3 b. A proper normal stress distribution is revealed considering the orientation of the coordinate system (0x axis is normal to the contact surface).




a. Von Mises Equivalent Stress b. Maximum Principal Stress

c. Normal Stress along x-axis d. Normal Stress along y-axis

Fig. 3. Stresses for Temperature of 150ºC

For example, in Fig. 4 the maximum von Mises and principal stress versus working temperature are given. It can be observed that the minimum values occur around ambient temperature.

Analyzing the results of all analyses, it can be seen that external loads have non-significant influence on the stresses and strains state.

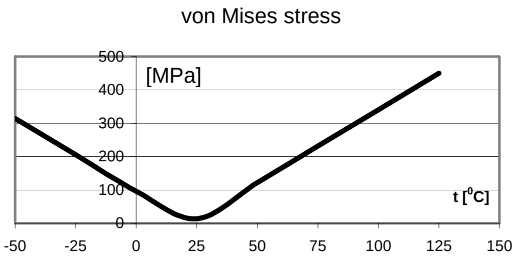
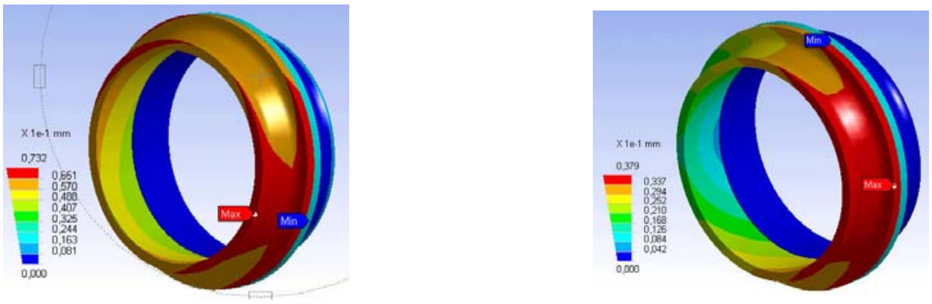
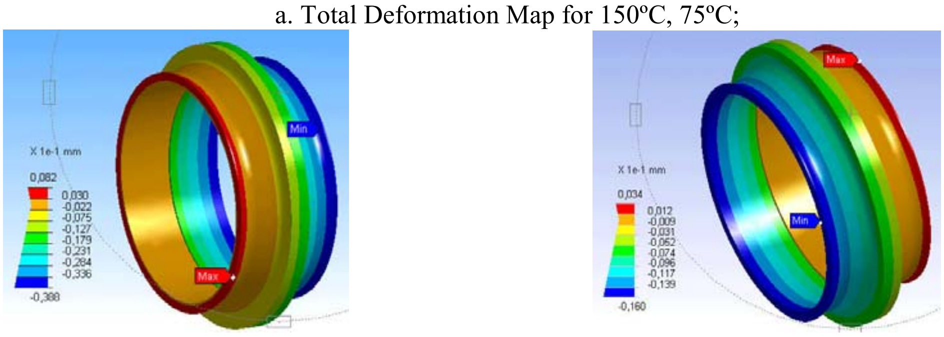


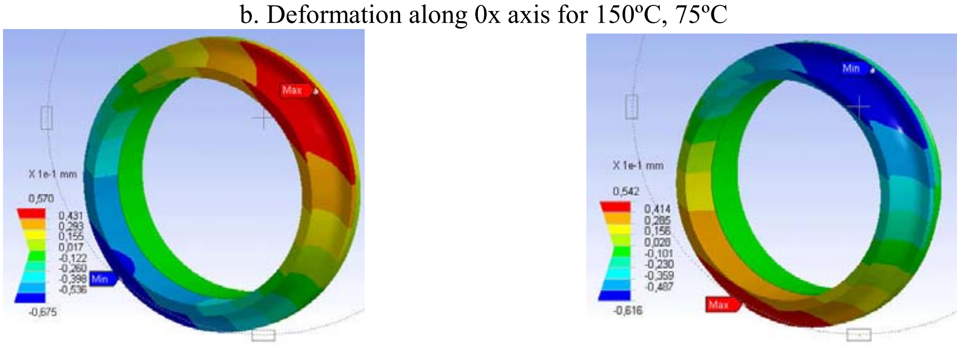

Fig. 4. Temperature Impact on Stress State



a. Total Deformation Map for 150ºC, 75ºC;



b. Deformation along 0x axis for 150ºC, 75ºC



c. Deformation along 0y axis, t = 150ºC; d. Deformation along 0z axis, t = 150ºC;

Fig. 5. Deformation Map for Different Working Temperature

Fig. 5 presents a deformation map for the assembly. It can be seen that the deformation map depends on the seal temperature. There are some similarities in the deformation map, but for lower temperature, the maximum deformation occurs in extended regions, even the values for compression and elongation are smaller.

Another investigated aspect by FEM was a thermal analysis for the assembly. The seal temperature, the thermal characteristics for the seal material (coefficient of convection) and the coefficient of friction were considered. In Fig. 6 the total heat flux is presented.

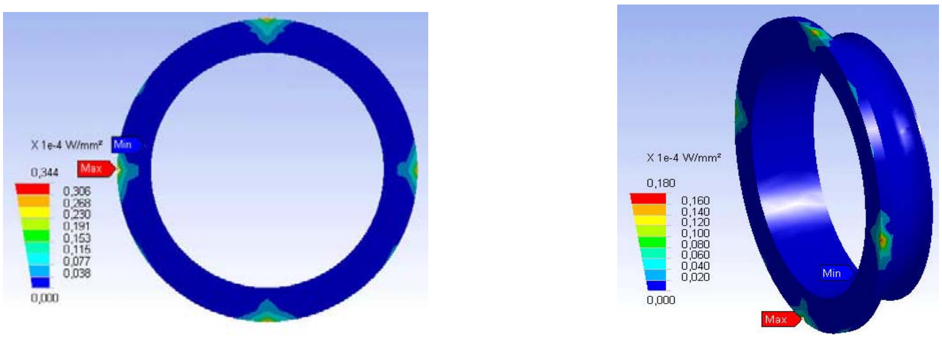


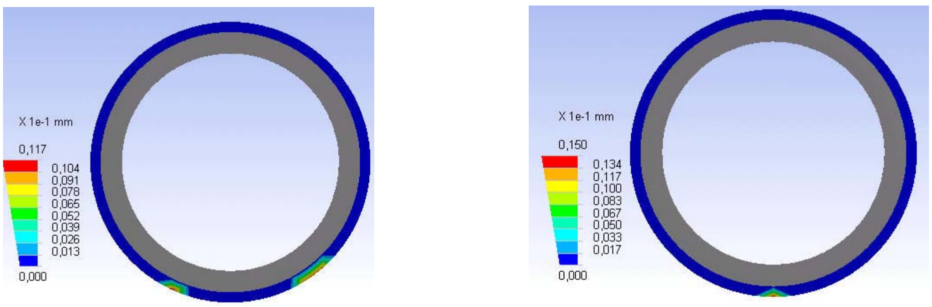

Fig. 6. Total Heat Flux, t = 125 ºC, t = 75 ºC

A symmetrical map can be noticed, containing four regions of maximum heat transfer.

## 4. Investigations on Contact Region

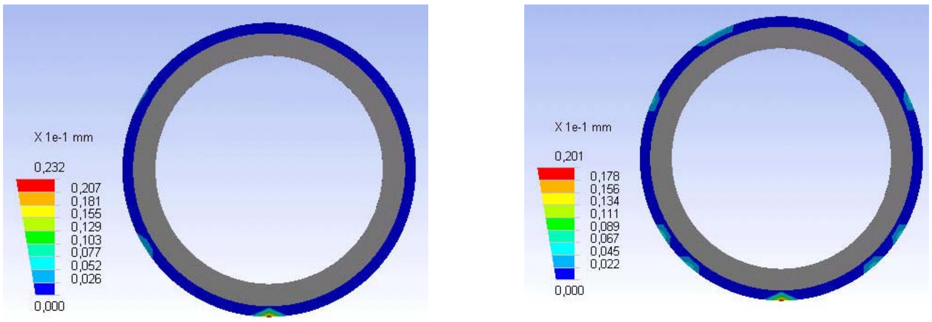
In Figs. 7, 8 and 9 the sliding distance for different temperature and friction coefficient is given. It can be revealed that:

- at lower temperature (negative ones):
  – the number of sliding regions increases regardless of the friction coefficient value;
  – the maximum sliding occurs on the inner side of the contact region;
- at higher temperature (> 100ºC):
  – the maximum sliding occurs on the outer side of the contact region;
  – for dry contact the number of sliding regions increases regardless of temperature;
- at temperature smaller than 100ºC:
  – for elastohydrodynamic and hydrodynamic regime more sliding regions appear.



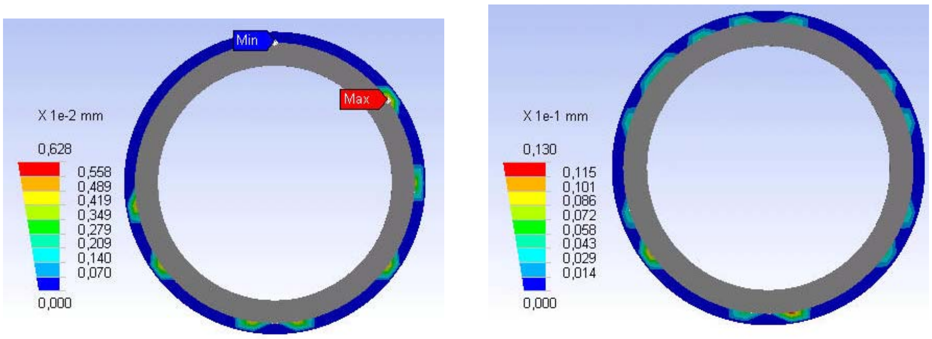

a. Coefficient of Friction f = 0,7; b. Coefficient of Friction f = 0,2;

Fig. 7. Sliding Distance for t = 125º C




a. Coefficient of Friction f = 0,7; b. Coefficient of Friction f = 0,2;

Fig. 8. Sliding Distance for t = 75º C




a. Coefficient of Friction f = 0,7; b. Coefficient of Friction f = 0,2;

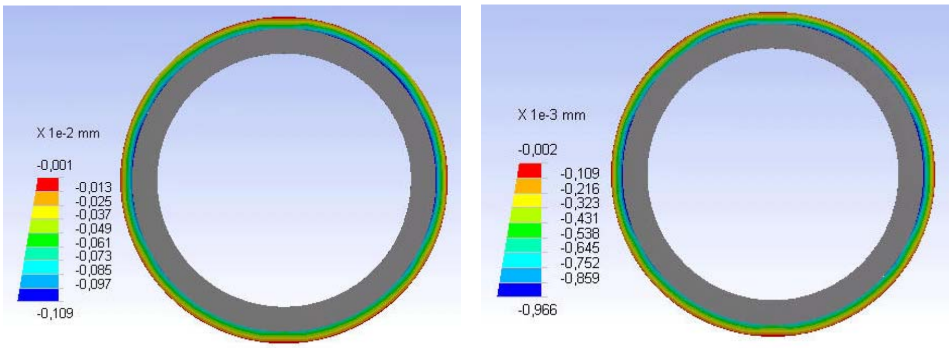
Fig. 9. Sliding Distance for t = –25º C

From Figs. 10, 11 showing a contact gap between the stationary seal and rotating one, some remarks can be made:

– all gap contact values are negative, demonstrating the presence of penetration, but the results values are very small,
– the gap distribution on contact region is like concentric circles regardless of working regime,
– the maximum penetration occurs at the inner side of the contact region at higher temperature, while at lower one it appears on the outer side of the contact region, and,
– in condition of a dry regime, the penetration increases, which develops wear in the contact surface.

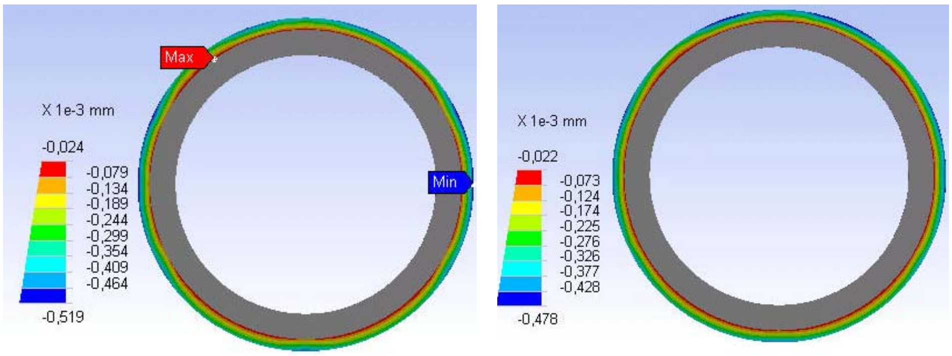
Some aspects stem from Figs. 12, 13 (the frictional stress), namely:

– the results are in good agreement with the ones from Figs. 1, 2, 3, having the same distribution map for similar working conditions, and
– greater values for the frictional stress are obtained at the outer side of the contact region for higher temperature, while at lower temperature the maximum value appears at the inner side.



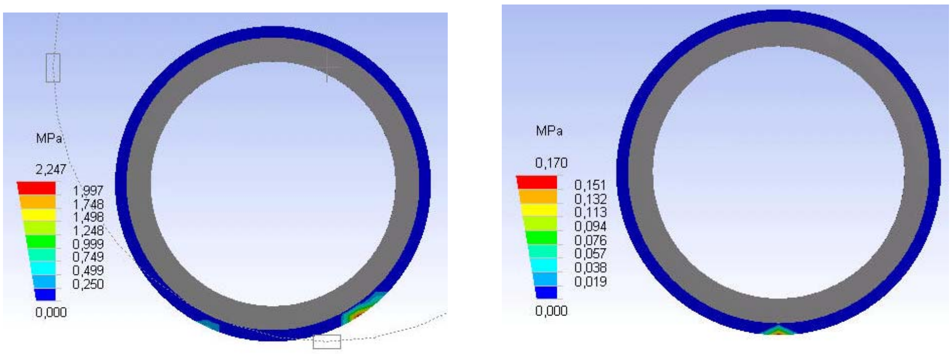

a. Coefficient of Friction f = 0,7; b. Coefficient of Friction f = 0,2;

Fig. 10. Gap at t = 125º C



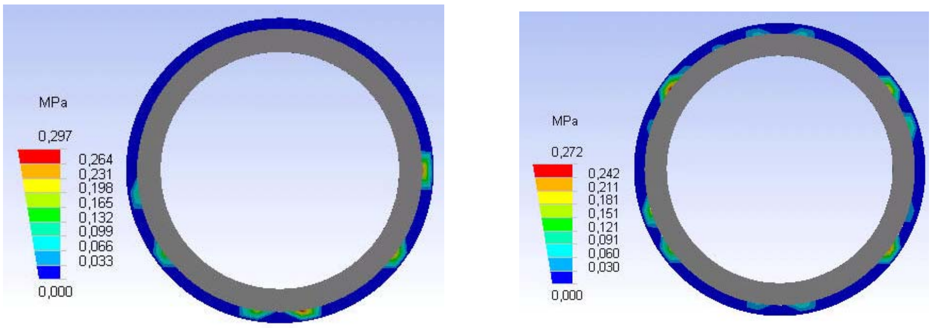

a. Coefficient of Friction f = 0,7 b. Coefficient of Friction f = 0,2

Fig. 11. Gap at t = –25º C




a. Coefficient of Friction f = 0,7 b. Coefficient of Friction f = 0,2

Fig. 12. Frictional Stress at t = 125º C




a. Coefficient of Friction f = 0,7 b. Coefficient of Friction f = 0,2

Fig. 13. Frictional Stress at t = –25º C

## 5. CONCLUSIONS

Using FEM analyses allows us to investigate a proper model, with fewer simplifying hypotheses. The minimum stress and strain state occurs around ambient temperature. The major influence upon stress and strain state is assigned to seal temperature. Analyzing the whole range of the recommended working temperature, [2], it can be stated that functioning at extreme temperatures (< –30ºC; >100ºC) generates rapid failure of the seals, due to the difficult working conditions (high stress and strain).

The results obtained by this analysis are useful for better understanding of the phenomenon that otherwise leads to contact surface damaging and fluid leakage.

Considering the results it is obvious that every face seal has to be modeled and analyzed by FEM, under its own working conditions, in order to assure its good operation.

The working temperature has a major effect upon the contact surface – the number of sliding regions, their distribution, the penetration and frictional stress. All these aspects generate problems in functioning which have to be known in order to explain the leakage of fluid. The hydrodynamic regime (modeled by smaller values for friction coefficient) offers better operating conditions.

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# PROUČAVANJE FRONTALNIH SISTEMA ZAPTIVANJA (FACE SEAL) METODOM KONAČNOG ELEMENTA

**Carmen Sticlaru, Arjana Davidescu**

*Rad prikazuje analizu frontalnog sistema zaptivanja (face seal) uz pomoć metode konačnog elementa. Predstavlja se model realizovan putem ANSYS programa, stanje tenzija (stress) i deformacije. Posebno se insistira na interakciji koja nastaje u zoni kontakta i na uticaj temperature.*

Ključne reči: *Metod konačnih elemenata, frontalni sistem zaptivanja*