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RESEARCH

Design of the Springs Tightening for a Double Cartridge Mechanical Seal

Andrei Stanciu[a], Costin Ilinca[a], Razvan George Ripeanu[a,*]

[a]*Petroleum-Gas University of Ploiesti, Blvd. Bucuresti, no.39, Ploiesti, Romania.*

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* Corresponding author:

Razvan George Ripeanu
E-mail: rrapeanu@upg-ploiesti.ro

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A B S T R A C T

Mechanical seals are used in a wide range of industrial applications, including pumps, compressors etc., in order to avoid leakage of the working fluid. The design construction, and the operation of the mechanical seals continue to evolve to meet the demands of new technologies and industries. These developments are the increase of operating parameters, in the condition of strictly respecting the environment protection. A mechanical seal consists of two rings, one being static and the other one dynamic, which are actioned by one or two fluid pressure, and an elastic force (or magnetic) which keeps the two faces of the rings in permanent contact. A direct contact between the frontal face of the rings will increase the surface temperature and the rings wear, so a small fluid film, as a lubricant and coolant, it is kept between the two face sealings. At modern mechanical seal are used also a secondary fluid. The tightening force in the seal, which it is very important, depends on seal construction, materials used, working fluid type, pressure and temperature, rotational speed, secondary fluid type, pressure, and temperature. Paper aims to present the studies made by the authors on a mechanical seal of the AESSEAL, cartridge seal type CDSA of a 1" in order to obtain the correct sealing pressure between seal rings. Calculation was made analytical, with the respect of regulations of the standards, and by FEA method. Was obtained a CFD model, to simulate the pressure inside the chamber of secondary fluid at different inlet/outlet pressures to determine the real value of it. The obtained results by CFD analyses were integrated in FEM analyses by ANSYS Static Structural to calculate the contact pressure on the rings faces and to calculate the minimum spring force for internal and external sealing cartridges for different working and secondary fluid pressure, also considering the seal construction design, materials, and rotational speed of 3000rpm. The results are very important in operation of the mechanical seal because the graphs presented in the paper give the operators the correct value of the spring tightening in the seal depending on the different ranges of fluids pressures.

1. INTRODUCTION

The history of mechanical seals dates back to the early 20th century. The first patent for a mechanical seal was issued in 1909 to John Landefeld, an engineer in Ohio, who developed a seal for use in pumps. The design was relatively simple, consisting of a stationary ring and a rotating ring, which were held in contact by spring pressure.

Throughout the 20th century, mechanical seals continued to evolve, with the introduction of new materials, such as carbon and silicon carbide, and improvements in seal design. In the 1940s, the development of the "balanced" mechanical seal, which used two springs to press the seal faces together, improved the reliability of seals in high-pressure and high-temperature applications.

In the 1960s and 1970s, manufacturers began to develop mechanical seals specifically for use in the chemical and petrochemical industries, which required seals that could withstand harsh operating conditions.

Today, mechanical seals are used in a wide range of industrial applications, including pumps, compressors, mixers, and agitators, and continue to evolve to meet the demands of new technologies and industries.

New technological developments make the problems of sealing machines and equipment more and more difficult. These developments are the increase of quantity of industrial equipment, the increase of operating parameters and making rigid the requirements to ecological safety of system of machines and their units, [1].

There are various standards that have been developed for mechanical seals, depending on the industry and application. Here are a few examples:

API 682 (American Petroleum Institute): This standard is widely used in the oil and gas industry and covers the selection, installation, and maintenance of mechanical seals for pumps and other rotating equipment. The latest version of API 682, which is the 4th edition, was published in 2014, [2].

ISO 21049: This international standard covers the general principles of mechanical seals for pumps and other rotating equipment. It was first published in 2000 and the latest version is from 2019, [3].

DIN 24960: This is a German standard that covers mechanical seals for pumps and other rotating equipment. It was first published in 1980 and the latest version is from 2019, [4].

ASME B73.1: This American standard covers mechanical seals for centrifugal pumps. The latest version of ASME B73.1 is from 2020, [5].

ISO 3069: This international standard covers the testing of mechanical seals for pumps and other rotating equipment. It was first published in 1974, withdrawn a reissued in 2000, with the latest version is from 2018, [6].

It's worth noting that standards are updated periodically to reflect the latest advances in technology and to ensure that they remain relevant to the industry. Therefore, it's important to check for the latest version of a standard before using it.

2. BASE OF CALCULATION

The mechanical seals can be found in different forms and design, due to vas area of utilization. One of the standard types of mechanical seals are presents in Figure 1.

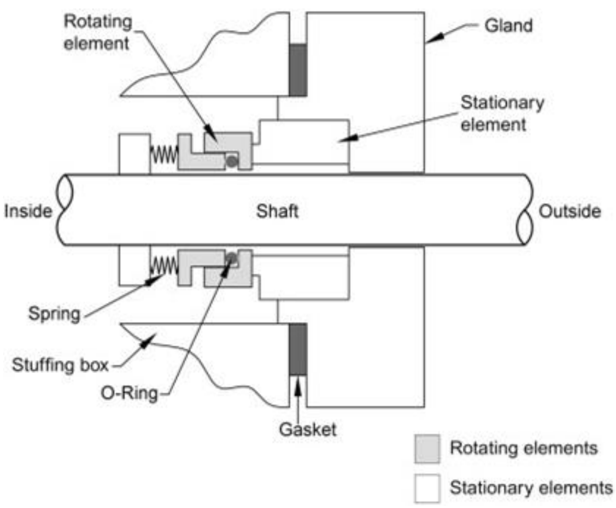

Fig. 1. Mechanical seal structure, [7].

The mechanical seals have typically 2 seals, one in work condition and one in stand-by, when the pump is not working. For the situation when the pump is not working, the equipment is only held together by the springs, who are pushing the dynamic (rotating) face to the stationary element.

On the other side, when the equipment is running, the mechanical seal is working on a complex way. Different loads are acting on the mechanical seal components, creating a sealing between the two sealing faces. Additional sealing components are needed to prevent leakage between the components of mechanical seal, typical, these components are represented by O-Rings or gaskets.

As mentioned, different types of loads are acting on the mechanical seal:

- The hydraulic force that it is applied to the mechanical seal frontal section by the fluid;
- Spring force;
- The hydraulic force that it is acting between the static and dynamic ring of the seal.

Also, the mechanical seal is influenced by different factors, as:

Pressure: The seal must be able to withstand the operating and designed pressure of the fluid or gas being sealed, as well as any pressure spikes that may occur.

Temperature: The seal must be able to withstand the operating and designed temperature of the fluid or gas being sealed, as well as any temperature changes that may occur.

Speed: The seal must be able to withstand the rotational speed of the shaft on which it is mounted, and any vibration that may occur.

Chemical Compatibility: The seal must be able to withstand the chemical properties of the fluid or gas being sealed, and the materials must not react with the fluid or gas.

Friction: The seal must be able to withstand the friction between the seal and the shaft, and between the seal and the gland.

Lubrication: The seal must be able to operate effectively with or without lubrication, depending on the application.

Erosion: The seal must be able to withstand erosion caused by the fluid or gas being sealed, and any particulates that may be present.

Installation: The seal must be easy to install and align properly with the shaft and the gland.

These forces must be taken into account when designing a mechanical seal to ensure that it will be effective and long-lasting in the specific application, [8, 9].

For the analytic analysis of the mechanical seal, we are going to use the information offered in API 682, [2].

A seal to be considered closed, the closing force, F_c, must have a value equal or higher than that the opening force, Fop. As per Ion Pana and Ion Preda analysis [8], we can corelate the closing force with the contact pressure, pc, which will be considered as closing pressure divided by the sealing contact surface:

$$F_{\mathrm{op}} = A \cdot \Delta p \cdot K, \text{ [N]}. \tag{1}$$

$$p_{\mathrm{c}} = \Delta p\ (B - K) + p_{\mathrm{sp}}, \text{ [MPa]}. \tag{2}$$

were,

A, it is the contact face area, [mm^2]:

$$A = \frac{\pi(D_{\mathrm{o}}^2 - D_i^2)}{4}. \tag{3}$$

D_{o}, - outside diameter, [mm];

D_{i}, - inside diameter, [mm];

Δp, - pressure across the seal face, [MPa];

B, - seal balance ratio. The balance ratio value representation it is showed in the Figure 2, and the values could be calculated with relations (4.a) and (4.b);

K, - pressure drop coefficient. K, it is a number between 0 and 1 which represents the pressure drop as the sealed fluid migrates across the seal faces. For flat seal faces (parallel fluid film) and a non-flashing fluid, K it is approximately equal to 0.5, [8]. For convex seal faces (converging fluid film) or flashing fluids, K it is greater than 0.5,[8]. For concave seal faces (diverging fluid film), K it is less than 0.5, [8];

P_{sp}, - spring pressure, [MPa].

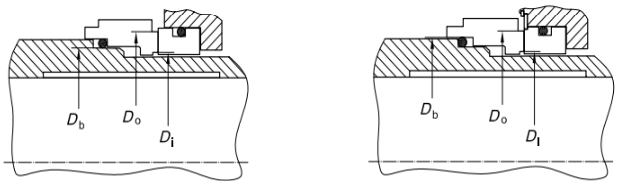

Fig. 2. Balance ratio measurement points, [2].

For external pressured seals (where the external pressure is higher than the barrier fluid), B it is:

$$B = \frac{D_o^2 - D_b^2}{D_o^2 - D_i^2}. \tag{4}$$

For internal pressured seals (where the barrier fluid pressure is higher than the pump fluid pressure), B it is:

$$B = \frac{D_b^2 - D_i^2}{D_o^2 - D_i^2}. \tag{5}$$

The contact pressure will have to be equal or higher than the operating force (or pressure), as per this affirmation, we have the following equations:

$$p_o = \Delta p\,(B - K) + p_{sp,}\ [\text{MPa}]. \tag{6}$$

were,

p_o, it is the opening pressure, [MPa].

$$p_o = \frac{F_{op}}{A}, [\text{MPa}]. \tag{7}$$

For the aim of this paper, we need to determine the spring tightening force, which depends on the spring constant that can be calculated with formula (7), [10]:

$$k = \frac{G \cdot d^4}{64 \cdot n \cdot R^3}. \tag{8}$$

were,

k, it is the spring constant, [N/mm];

G - shear modulus, [MPa];

d - spring wire diameter, [mm];

n - number of spires;

R - average radius of the spring, [mm]. The average radius could be obtained with the formula (8):

$$R = \frac{D_{sp} - d}{2}. \tag{9}$$

were,

D_{sp} it is the spring diameter, [mm].
As Hooke law, the spring force it is calculated by formula (9):

$$F_{sp} = -k \cdot x. \tag{10}$$

were,

F_{sp} it is the spring force, [N/mm];

x - spring displacement [mm].

3. ANALITIC IMPLEMENTATION

All the information presented in the section 2 of the paper was applied to a double cartridge mechanical seal type CDSA 1", from Aesseal, on both cartridge seals. The structure of the equipment can be found in the Figure 3 and 4, [11].

As previous stated, the equipment is having 2 different seals, one internal (Figure 5), which has a role of stopping the leakage of the pump fluid out in atmosphere, and a secondary (external) one that is having the role to stopping the barrier fluid exiting the seal (Figure 6).

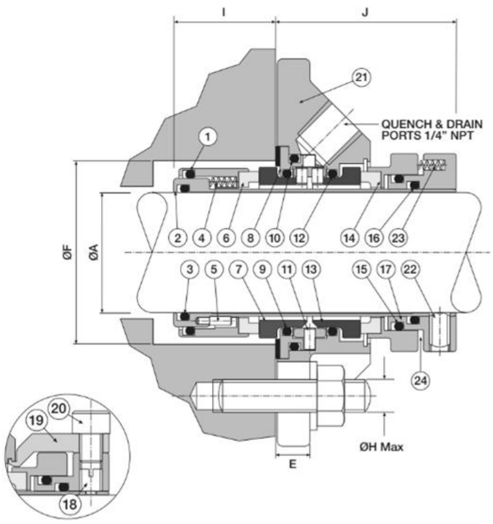

Fig. 3. CDSA mechanical seal configuration, [11].

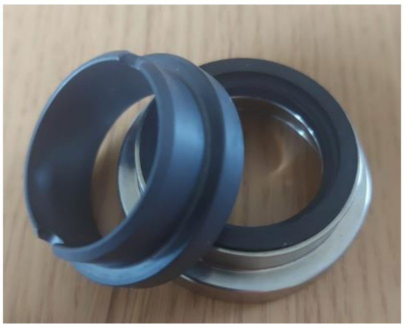

Fig. 4. Aesseal CDSA pair of seals.

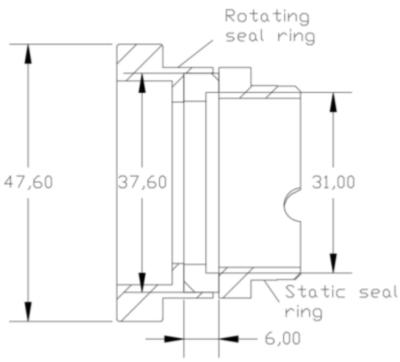

Fig. 5. Aesseal CDSA external seal.

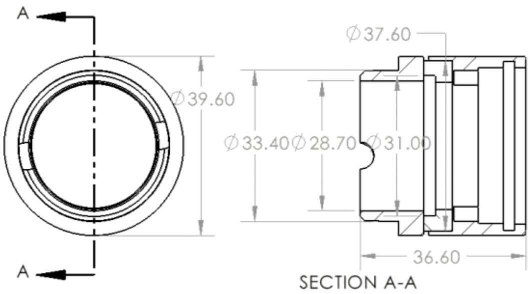

Fig. 6. Aesseal CDSA internal seal.

For this analysis, we are going to focus more on the external seal, as due to how the fluid is acting on the seal surface, and it is the more affected one.

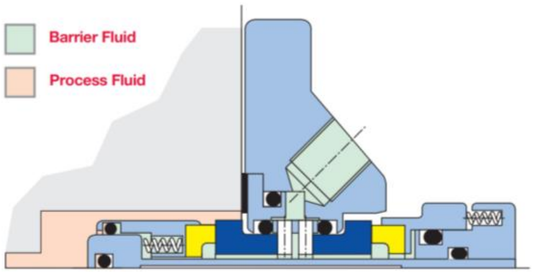

Fig. 7. Fluid acting on the cartridge mechanical seal, [11].

Table 1. External seal characteristics.

Description	Value
Pressure, *P*	0.44 MPa
Outside diameter of the contact surface, D_0	37.6 mm
Inside diameter of the contact surface, D_i	31 mm
Balance diameter of the seal, D_b	35 mm
Pump shaft speed	3000 RPM
Pressure drop coefficient, *K*	0.5
Number of springs, *n*	8

We are going to proceed, step by step as per calculation presented in section 2 of the paper:

$$F_{op} = 78.12 \text{ N}$$

$$A = 355.41 \text{ mm}^2$$

$$p_c = \Delta p\,(B - K) + p_{sp} \tag{11}$$

$$p_{sp} = p_c - \Delta p\,(B - K) \tag{12}$$

Considering that the contact pressure must be at least equal with the opening pressure, the value of the spring pressure it is:

$$p_{sp} = p_o - \Delta p\,(B - K) \tag{13}$$

$$p_o = 0.22 \text{ MPa}$$

$$B = 58.3\%$$

$$p_{sp} = 0.18348 \text{ MPa}$$

The resulting value of the spring pressure will have to be transformed in force, by multiplying the value to due area.

$$F_{sp} = p_{sp} \cdot A, [\text{N}]. \tag{14}$$

$$F_{sp} = 65.21 \text{ N}$$

To have the value for which every spring, Fsp,1 needs to act on the seal, we must divide the spring force F_{sp} by the number of springs:

$$F_{sp,1} = \frac{F_{sp}}{n}, [\text{N}]. \tag{15}$$

$$F_{sp,1} = 8.15 \text{ N}$$

By applying the same steps, for different pressures, we can determine the value of the minimum required springs tightening force as presented in Figure 8.

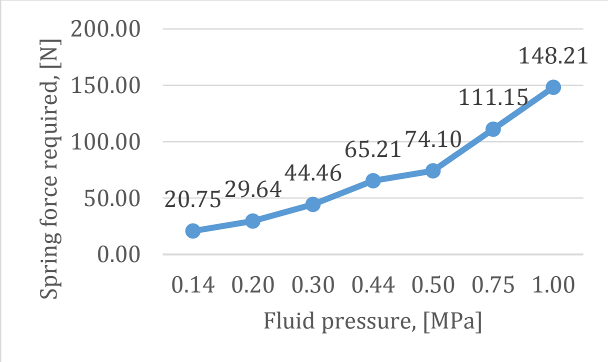

Fig. 8. Minimum required spring tightening force for different fluid pressure.

In construction design of the mechanical seals, manufacturers could use an different number of springs. In table 2, we can find the value of the tightening force needed, per each spring component, based on different values of the number of springs.

Table 2. Different tightening forces per spring at different fluid pressure.

Number of springs	Fluid pressure, [MPa]			
	0.14	0.20	0.30	0.44
	Minimum spring force per spring, [N].			
1	20.75	29.64	44.46	65.21
2	10.37	14.82	22.23	32.61
3	6.92	9.88	14.82	21.74
4	5.19	7.41	11.12	16.30
5	4.15	5.93	8.89	13.04
6	3.46	4.94	7.41	10.87
7	2.96	4.23	6.35	9.32
8	2.59	3.71	5.56	8.15
9	2.31	3.29	4.94	7.25
10	2.07	2.96	4.45	6.52

For the calculated tightening forces, we will determine the spring displacement, using the spring construction presented in Figure 9:

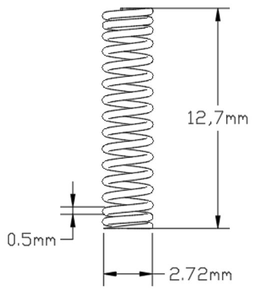

Fig. 9. Spring dimensions.

The spring characteristics considered are as follows:

G = 78600 MPa (C-276 Alloy);

d = 0.5 mm;

N = 13 spires;

D_{sp} = 2.72 mm;

R = 1.11 mm;

k =4.3133 N/mm.

In table 3 are presented the springs displacements for different fluid pressures and in Figure 10 is showed the spring required displacement vs. fluid pressures.

Table 3. Different tightening forces per spring at different fluid pressure.

P, [MPa]	F_{sp}, [N]	$F_{sp,1}$, [N]	x, [mm]
0.14	20.75	2.59	0.60
0.20	29.64	3.71	0.86
0.30	44.46	5.56	1.29
0.44	65.21	8.15	1.89
0.50	74.10	9.26	2.15
0.75	111.15	13.89	3.22
1.00	148.21	18.53	4.30

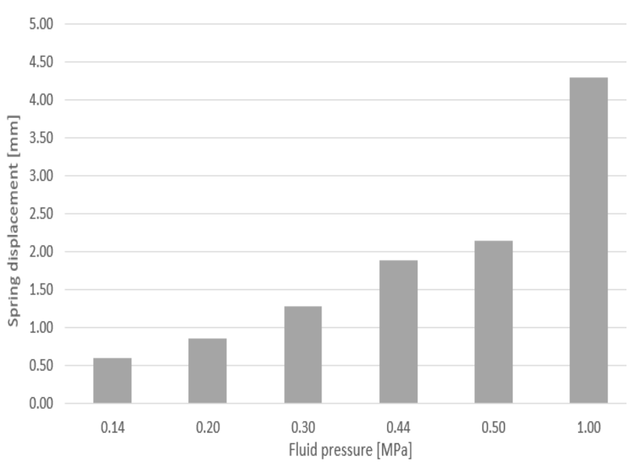

Fig. 10. Spring required displacement vs. fluid pressure.

Was determined experimentally, for the spring used, the maximum displacement with a value of 7.6 mm. This maximum displacement of the springs corresponds to a maximum fluid pressure sealing. At the value of around 2 MPa, the spring displacement it is higher than 7.6 mm, creating an instability into the seal. To exclude any risks, we must increase the number of springs, or to select a more appropriate seal for the job.

For the internal seal, the calculation process it is identical, and the parameters are presented in table 4.

As one pressure is acting inside the seal and the other one is acting outside, as per Figure 7, the pressure used in calculation will be as per equation (16).

Table 4. Internal seal characteristics.

Description	Value
Barrier fluid pressure, P_i	0.44 MPa
Pump fluid pressure, P_e	0.30 MPa
Rotational speed	3000 RPM
Outside diameter of the contact surface, D_0	37.6 mm
Inside diameter of the contact surface, D_i	31 mm
Balance diameter of the seal, D_b	35 mm
Pressure drop coefficient, K	0.5
Number of springs, n	6

$$\Delta p = P_i - P_e, \text{[MPa]}. \quad (16)$$

$$\Delta p = 0.14 \text{ MPa}$$

From now one, the calculation is going to be similar, and the results are presented already in tables 2 and 3, including Figures 8 and 10.

4. FEM ANALYSIS

The finite element analysis was done using the software Ansys, static structural analytic system. We will start with the external seal.

Components were measured, resulting the dimensions from figures 5 and 6, and the running parameters are in the tables 1 and 4.

Materials used for the 2 rings, were considered as follows (tables 5 and 6):

1. Static ring → Silicon Carbide.

Table 5. Silicon carbide properties, [12].

Property	Value	Unit
Density	3.1	g/cm^3
Coefficient of Thermal Expansion	$4\cdot10^{-6}$	C^{-1}
Young Modulus	410	GPa
Poisson's Ratio	0.14	-
Bulk Modulus	189.81	GPa
Shear Modulus	179.82	GPa
Tensile Yield Strength	390	MPa
Compressive Yield Strength	3900	MPa
Tensile Ultimate Strength	390	MPa

2. Dynamic ring contact surface → Tungsten Carbide.

For these two items, the characteristics were introduced manually.

Table 6. Tungsten carbide properties, [13].

Property	Value	Unit
Density	15.7	g/cm^3
Coefficient of Thermal Expansion	$4.5\cdot10^{-6}$	C^{-1}
Young Modulus	669	GPa
Poisson's Ratio	0.22	
Bulk Modulus	398.21	GPa
Shear Modulus	274.18	GPa
Tensile Yield Strength	344	MPa
Compressive Yield Strength	2683	MPa
Tensile Ultimate Strength	344	MPa

3. Dynamic ring body → Stainless Steel.

The last item, had the characteristics imported from Ansys library.

The analysis of the two seals, have started with the modelling. After realising the model and material selections, the contacts between the 2 rings seal are entered into the Ansys, as per Figures 11 and 12.

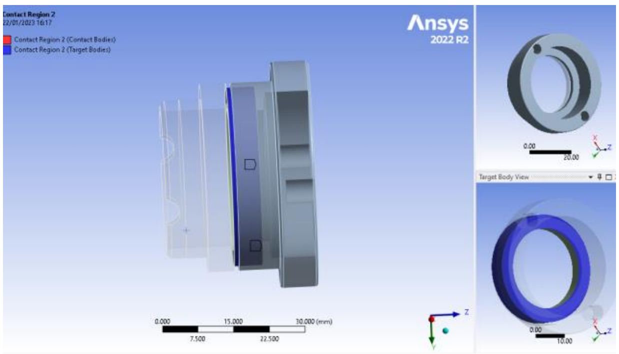

Fig. 11. Bonded connection between the contact surface of the dynamic ring and body.

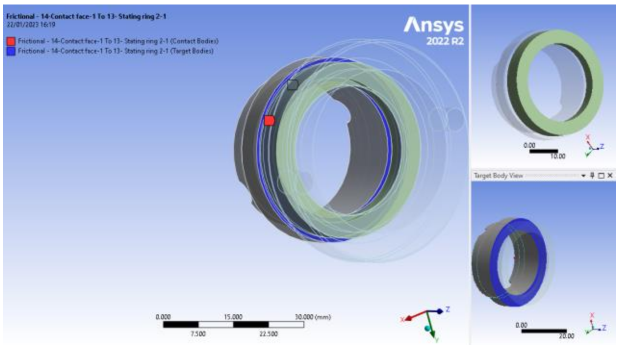

Fig. 12. Frictional connection between the static and dynamic rings.

The friction coefficient between the tungsten carbide ring and silicon carbide ring was considered 0.08, [14].

For meshing value, different topics were taken into account:

1. Number of contact elements;
2. Results accuracy;
3. Running time.

Based on the above, the mesh has results presented in Figure 13.

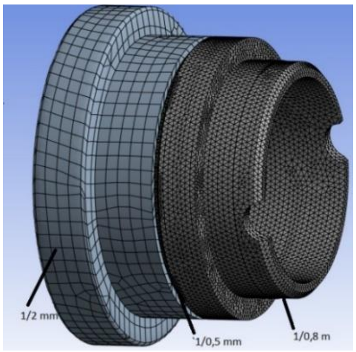

Fig. 13. Mesh quantity (unit per mm).

Body of the dynamic ring, had a mesh size of 1 unit per 2 mm, dynamic ring contact surface, 1 unit per 0.5 mm and static ring, 1 unit per 0.8 mm. In every important region, the sizing of the mesh was increased using refinement or face sizing.

For the contact region, the meshing value was considered after executing different tests on the program.

Using the values from the table 7, the graphic from Figure 14 was created, and it shows, that at the value of the analysis is not changing significantly, but the number of elements and nodes tripled from 0.8 units of mesh to 0.4, which is adding a long time of running and a lot of data to be storage.

Table 7. Number of elements, nodes and pressure results per different mesh values.

Mesh [unit/mm]	Elements	Nodes	Average pressure results [MPa]
0.8	342502	515761	0.25237
0.6	508654	753912	0.25734
0.5	700658	1026611	0.25186
0.4	1053091	1522996	0.25364

Last step, before running the program, is to determine the right loads to be used, loads such pressure, spring force, supports etc.

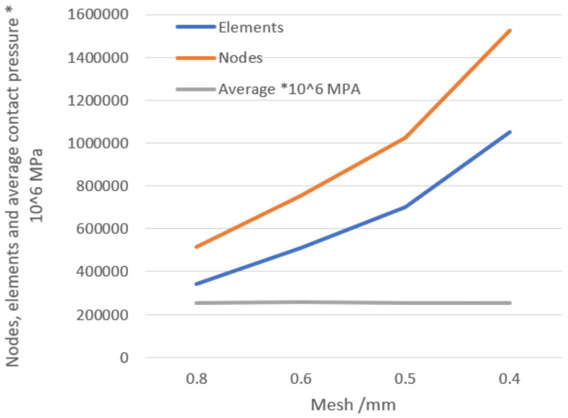

Fig. 14. Number of elements, nodes and pressure results per different mesh value.

The pressure was determined using the CFX analysis, considering a turbulent flow inside the mechanical seal. The inputs were entered at the entry into the seal, 0.44 MPa, and on the exit, having 0.44 MPa, as per Figure 15.

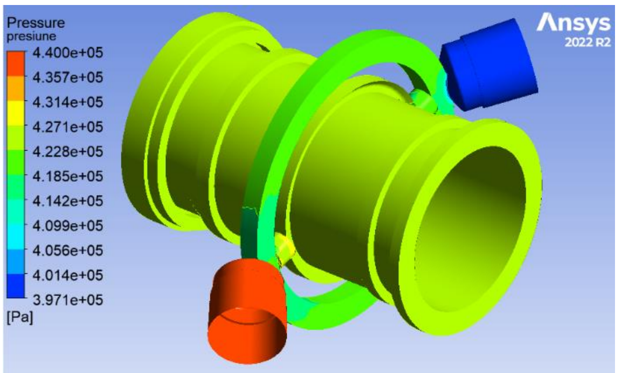

Fig. 15. CFX usage for determination of the inside pressure of mechanical seal

The pressure drop inside the seal it is not so significant, having almost 0.01 MPa lost. For the accuracy of the calculation, we are considering the maximum value of the pressure that we are applying of 0.44 MPa.

The values of spring forces were taken from figure 8, the rotational speed from refinery and remote displacements as per drawing and working conditions of the seal.

Interstitial load (load D from Figure 16), was considered linear, from the inside of the seal, where it is a pressure of 0.44 MPa, to the exterior, where the pressure it is atmospheric.

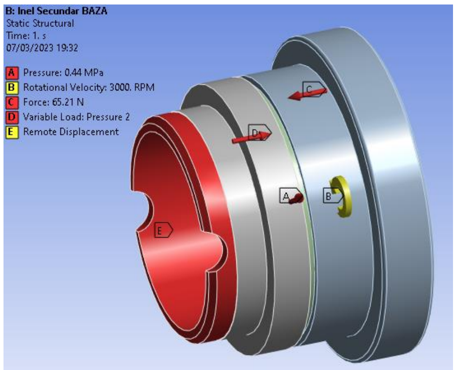

Fig. 16. Static structural inputs for the external seal.

From the analysis, the results are presented in Figure 18a and 18b.

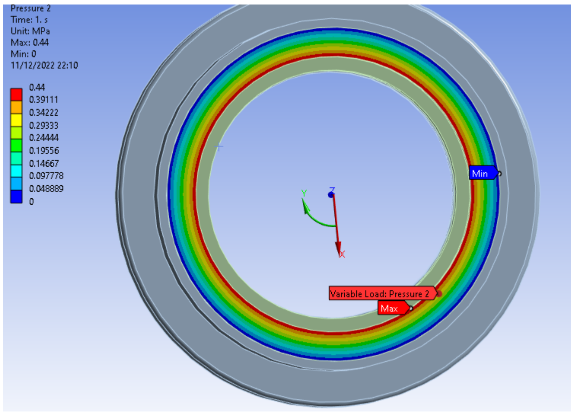

Fig. 17. Interstitial load of the external seal

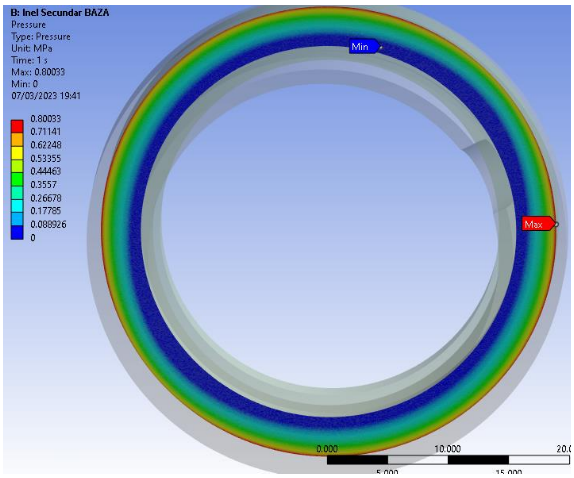

Fig. 18a. Static structural results on the external seal

Results	
Minimum	0. MPa
Maximum	0.80033 MPa
Average	0.25186 MPa
Minimum Occurs On	14-Contact face-1
Maximum Occurs On	14-Contact face-1
Information	

Fig. 18b. Static structural results on the external seal – minimum / maximum / average values.

As could be observed from figure 18b, the average pressure of the seal has a value of 0.251 MPa, which it is with 0.031 MPa, higher than the opening pressure calculated at stage 3 of this paper.

As can be seen, there are some areas of the seal that are having the contact pressure smaller than the opening pressure, due to this, the barrier fluid will enter between the two seals, creating a fluid film, cooling down the seal.

To better understand the behaviour of the seal, different analysis, of the seal, with different values of pressure and springs were considered, resulting the values from table 8. A difference of 12-15% occurs between the analytic and FEA methods. This difference appears due to different factors, that the standard does not consider, such as materials, friction coefficient, centrifugal force etc.

Taking into the account that the standard role is to keep everything safe, we can expect that the values offered from the standard are stricter than the real scenario.

Table 8. Different results using FEA.

Pressure [MPa]	FEA Contact pressure [MPa]	Analytic contact pressure [MPa]	Delta (%)
0.14	0.082747	0.07	15%
0.20	0.11638	0.10	14%
0.30	0.17277	0.15	13%
0.44	0.25186	0.22	12%
0.50	0.28578	0.25	13%
1.00	0.56964	0.50	12%

For the internal seal, the discussion it is identical until the inputs, and they can be found in Figure 19.

As presented in table 4, the characteristics are similar with the external seal, but the pressure and the number of springs was changed.

Using the Ansys module, we can figure out that the seal is more stable due to pressure equilibrium, and the spring force it is necessary only to keep the seal closed, as Figure 20 where interstitial loads are presented.

In Figure 21a can be seen the contact pressure values, having the same principle as per external seal, and there is a gap for the barrier fluid to enter between the seals.

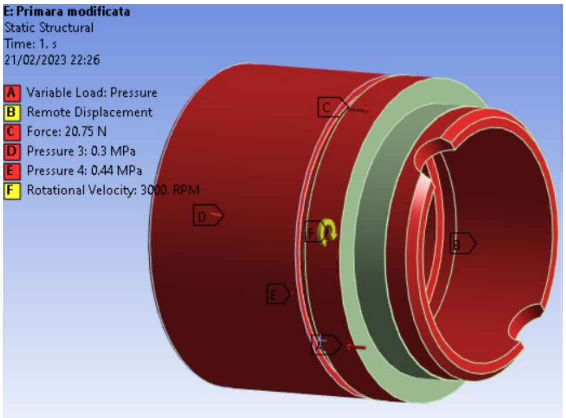

Fig. 19. Static structural inputs for the internal seal

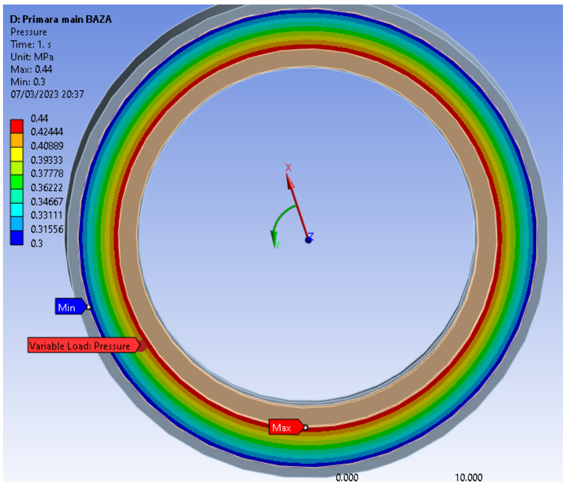

Fig. 20. Interstitial load of the internal seal

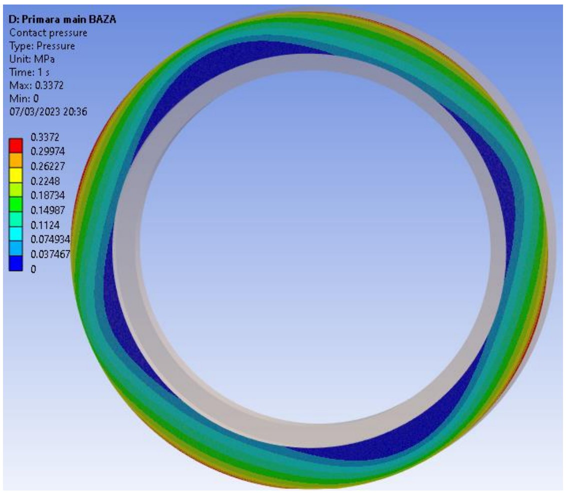

Fig. 21a. Static structural results on the internal seal

Results	
Minimum	0. MPa
Maximum	0.3372 MPa
Average	0.11205 MPa
Minimum Occurs On	6- Contact surface-1
Maximum Occurs On	6- Contact surface-1
Information	

Fig. 21b. Static structural results on the internal seal – minimum / maximum / average values.

This time, due to configuration and the importance of the seal, the average contact pressure has a value of 0.11 MPa, but analytic, the result was 0.07 MPa. This difference, as per previous one, is due to friction, design, material [15, 16] etc., but this seal importance is higher than the external one, so a higher level of security it is needed.

However, spring role it is not only to keep the seals together in operation condition, but it needs to keep it also on the stand-by moments, when the seal it is transported, or not mounted into the pump shaft.

Considering only the spring force into the internal seal, we have the following displacements into the seal showed in Figure 22.

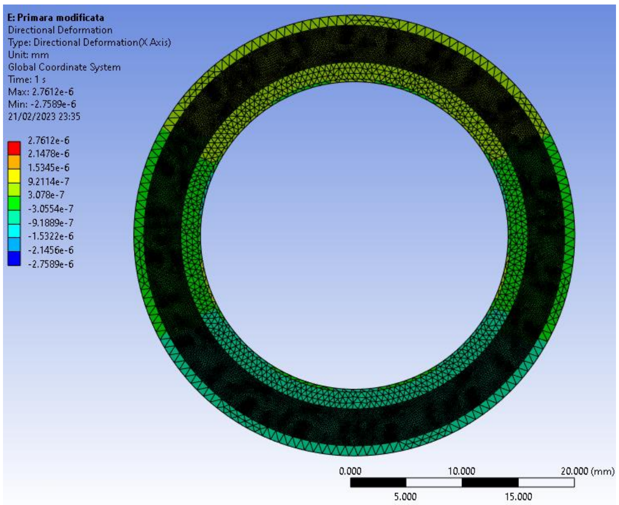

Fig. 22. Internal seal displacement under only spring effect.

As the figure 21 presents, the maximum displacement it is around 2.7 micrometres, which is impossible to be recognized with bare eye, so the seal will be kept together in the case of transportation or any other movements that will be implemented.

5. CONCLUSION

The spring role in the safe functioning of a mechanical seal it is very important, as it is one of the main players into closing the seal, especially when on the seal only internal or external fluid is acting. Even when the seal it is stabilized only from the pressure, the springs will still need to energize the two seals, pushing them, and also keeping them together in time of movements.

Was obtained a nomogram of necessary springs tightening force and spring displacements vs. fluid pressure, so mechanical seals operators could avoid leakage without increasing the wear of the sealing rings. The importance of a good design sealing it is very important, as even a small mistake can lower the durability of the seal, as a lower pressure contact can cause leakage, while a higher contact pressure will create friction, generate heat, and will result into a failure.

A difference of 12-15% occurs between the analytic and FEA methods. This difference appears due to different factors, that the standard does not consider, such as materials, friction coefficient, centrifugal force etc. Taking into the account that the standard role it is to keep everything safe, we can expect that the values offered from the standard are stricter than the real scenario.

For the internal seal, the springs tightening force are necessary only to keep the seals together in operation condition, but it needs to keep it also on the stand-by moments, when the seal is transported, or not mounted into the pump shaft.

The model, of a real double cartage seal and FEM analysis realized, could be applied for different rings materials, and for different working and barrier fluids pressures, in order to improve the design of the mechanical seals, and to increase the durability, respecting the leakage requirements.

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