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İLKAN HASAN AKPINAR

A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE
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PROF. DR. OSMAN SAİM DİNÇ

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GEBZE TEKNİK ÜNİVERSİTESİ	**YÜKSEK LİSANS JÜRİ ONAY FORMU**

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ONAY

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İMZA/MÜHÜR

ÖZET

Sızdırmazlık konusu turbo-makinalarda yüksek güç, yüksek verimlilik ve genel gas türbini geliştirmeleri için önemli ve gereklidir. Rekabetçi turbo-makina tasarımı için iyi tasarlanmış düşük sızdırmazlık miktarları ile uzun ve süre çalışabilen sızdırmazlık elemanları gerekir. Uygunsuz sızdırmazlık elemanı tasarımı sonucu oluşan yüksek kaçak miktarları verim düşüşü, tasarım kararsızlıkları, yetersiz hava tahliyesi sebebiyle disk boşluklarında sıcaklık artışı ve yüksek yağ tüketimi gibi istenmeyen durumlara neden olur. Tasarımcılar, motor performansını ve dayanıklılığını artırmak için en uygun maliyetli yöntem olduğundan genellikle sızdırmazlık teknolojisine özel ilgi duyarlar. Gas türbinli motorlarda farklı sızdırmazlık teknolojileri kullanılmaktadır. Tasarım mühendisleri, uygun sızdırmazlık teknolojisini seçmek için genellikle ortam sıcaklığı, basıncı, bağıl hareketi, aşınma kapasitesini, ısı üretimini, ağırlığı, kullanılabilir alanı, üretim kolaylığını ve kurulum ve sökme kolaylığını dikkate alır.

Bu tez özellikle odağının karbon keçelerde olduğu turbo-makinelerde kullanılan sızdırmazlık teknolojisine gener bir bakış sunmaktadır. Ayrıca turbo-makine için bir tasarım örneği sunulmaktadır.

SUMMARY

Sealing is extremely important for turbomachinery and essential for high power, high efficiency, and overall gas turbine development. Competitive turbomachinery design requires well-designed and robust seals those can work for very long times with low leakages. High leakage as a result of inadequate sealing can cause efficiency penalties, design instabilities, temperature increase on disk cavities due to insufficient purge, and high oil consumption which are undesired conditions for turbomachinery design. Designers generally have special interest in sealing technology, since it is the most cost effective method to improve engine performance and durability. Different sealing technologies are used in gas turbine engines. Design engineers generally consider the surrounding temperature, pressure, relative movement, wear capability, heat generation, weight, available space, ease of manufacture and ease of installation and removal to select the appropriate sealing technology.

This thesis represents an overview of turbomachinery sealing technology with a specific focus on carbon seals. Moreover, a design example for a turbomachinery is presented.

Key Words: Carbon seal, Turboshaft, Advanced Sealing Technology

ACKNOWLEDMENTS

I would like to express my deep and sincere gratitude to my supervisor, Prof. Dr. Osman Saim DİNÇ, who not only shared his profound scientific knowledge with me but also taught me great lessons of life. His support, suggestions and encouragement gave me the drive and will to complete this work.

I wish to express my warm and sincere thanks to Sinan ŞAL and Cansinem Tüzemen GÜNDOĞAN for his collaboration to this research.

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LIST OF ABBREVIATIONS AND ACRONYMS

Abbreviations and Acronyms		**Explanations**
FOD	:	Foreign Object Damage
HPT	:	High Pressure Turbine
SFC	:	Specific Fuel Consumption
EGT	:	Exhaust Gas Temperature
B	:	Balance Ratio
P_f	:	Pressure, imposed on the face by the action of the sealed pressure
r_o	:	Outer diameter of contact face
r_b	:	Inner diameter of runner
r_f	:	Inner diameter of contact face
P	:	Sealed pressure
P_o	:	Outer side pressure
P_i	:	Inner side pressure
F_s	:	Spring force
F_s	:	Torque
P_{tot}	:	Total Pressure on Contact Face
A_f	:	Contact Face Area
f	:	Frictional Coefficient
D_m	:	Mean Diameter of Contact Face
E_p	:	Produced Energy
N	:	Rotational Velocity
q	:	Removed Energy
s	:	Splash Ratio
m	:	Mass Flow Rate
c_p	:	Specific Heat Capacity
ΔT	:	Temperature Difference
FFKM	:	Perfluoroelastomer
$\Delta Q, \dot{w}$	:	Wear Rate

K	:	Wear Coefficient
W_m	:	Load Supported by Contact
ΔS	:	Sliding Distance
H	:	Hardness
Δw	:	Wear Depth
Δt	:	Time
P_m	:	Pressure on Face
U	:	Speed

LIST OF FIGURES

1. INTRODUCTION

Gas turbine engines require seals that are operating for an extended period of time with minimal leakages. The sealing performance of the seals directly affects the performance of the turbomachinery. Seal leakage amount should be under control for engine mission since material temperature, engine performance are being affected. Thus, gas turbine companies and universities have conducted many studies and researches on this subject. Sealing technology is critical with meeting next generation aero propulsion system goals which are performance, efficiency, safety and cost.

Gas turbine engine shaft is located with the help of bearings and synthetic oils, which generally conform to either MIL-PRF-7808 [1] or MIL-PRF-23699 [2] specifications are utilized for lubrication and cooling purposes. Pressurized air is generally used to keep the oil inside the bearing chamber but sealing system is vital for the performance of the pressurization system. Sealing systems must be provided to prevent overheating of the bearings and oil leakage from the sump.

Oil seals are those adjacent to the bearings. They may be labyrinth type, carbon type or the brush type (EJ200, TP400, etc.). Air seals are next away from the oil seals and bearings. They are separated by a pressurization chamber. The air seals assist in maintaining adequate pressure in the pressurization chamber. Seal pressurization circuits serve two functions: to keep the lubrication oil in the sumps at all attitudes and altitudes expected during the typical mission and the circuit insulates the hot sumps by buffering.

1.1 Scope of Thesis

The main goal of this thesis is to design and perform analytical calculation for mechanical carbon seal as oil seals those are adjacent to bearings. Preliminary design phase of carbon seal are investigated and reported during thesis study.

In scope of this thesis, analytical calculations are performed for carbon seal parts and all adjacent parts in bearing zone with 3D thermal and mechanical models, separately. Some design criteria are checked to size the carbon seal. At the end of the this thesis preliminary geometry which meets with design requirement is generated Addition to this scope, literature information on carbon seal are exhibited.

For instance, in Figure 1.1, carbon seal used in oil (bearing) region is illustrated. This carbon seal is used to restrict oil leakage from bearing area (area $P_1; T_1$) to next region (area $P_2; T_2$).

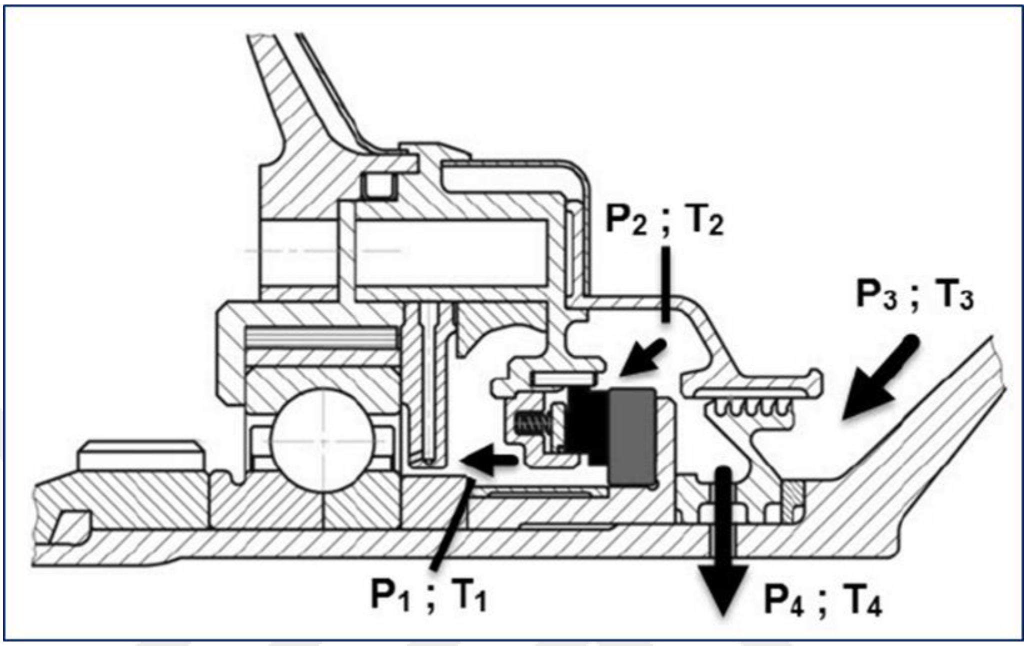


Figure 1.1 The Picture of Bearing Location [3]

2. LITERATURE REVIEW

Seals play an essential role in turbomachinery systems and required and they are listed below.

- to perform safe and efficient operation of gas turbines,
- to control temperature of components by adjusting the sealing air,
- to control internal flows as metering devices,
- to improve performance and efficiency by reducing leakage,
- to prevent the ingestion of hot mainstream gas into turbine rotor cavities,
- to prevent oil leakage,
- to provide thrust balance.

As seen in Figure 2.1, seals shall provide task which indicated above in complex air flow system.

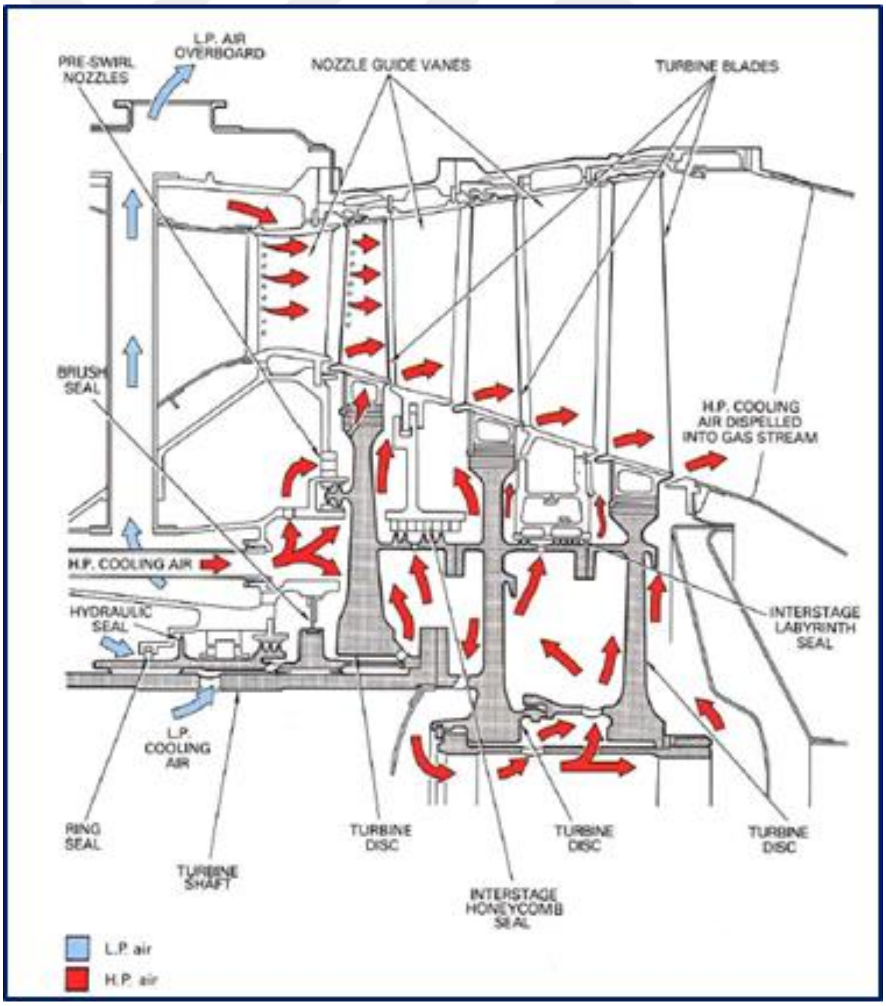


Figure 2.1 Secondary Air System Distribution and Seal Location [4]

Controlling the seal clearances is essential to minimize the leakage and performance penalties. In many cases, sealing interfaces have special coatings, like lubricants to decrease the friction coefficient for the integrity of the components since they are subjected to abrasion, erosion, oxidation, incursive rubs, FOD, and deposits, as well as extremes in thermal, mechanical, aerodynamic and impact loadings [5]. The tribological pairing of materials controls how well and how long these interfaces will be effective.

Performance issues are closely tied to engine clearances. Regarding clearance changes, 0.0254 mm change in HPT tip clearance decreases SFC by 0.1 percent and increases the EGT by 1 °C, producing an annual savings of 0.02 billion gallons for U.S. airlines. For these large, land-based gas turbines, the percentages represent substantial fuel savings and monetary returns with the most significant returns cited for aging power systems [6].

2.1 Labyrinth Seals

Labyrinth seals (knife-edge seals) are the most common seal type used in the modern engines and are very frequently used as bearing chamber seals, including applications as co-rotating or counter-rotating inter-shaft seals. It is easy to design and manufacture compared to other seal types and illustrated in Figure 2.2.

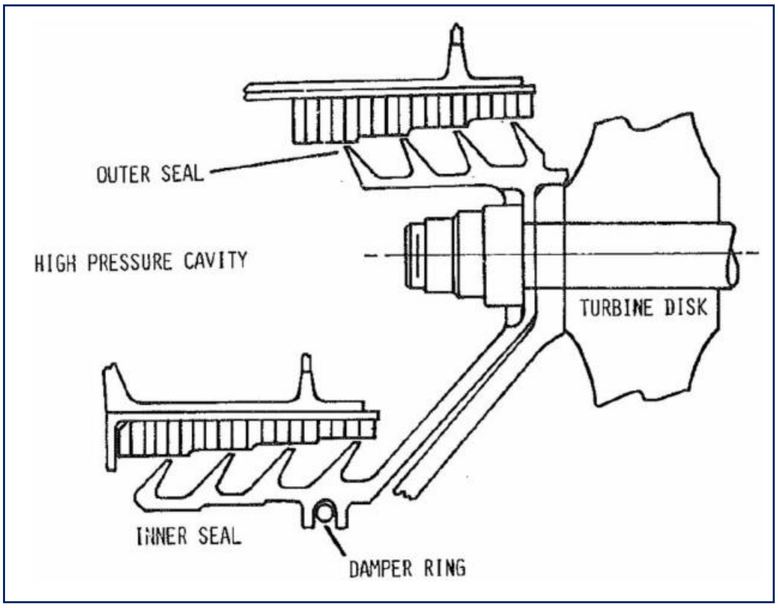


Figure 2.2 Labyrinth Seal [7]

Leakage highly depends on clearance of labyrinth seals and wear / deterioration worsens the leakage as the gas turbine run. Additionally, long seals have instability problems. Labyrinth seals are inexpensive as compared to other seals and they can operate at a wide range of temperature, pressure and speed.

2.2 Brush Seal

Brush seals are most often used as air-to-air seals. They also act as bearing chamber seals in several production engines, an example with air-air and air-oil brush seals. The seals consist of very thin bristles. The example of brush seal is illustrated in Figure 2.3.

Figure 2.3 Brush Seal [8]

2.3 Carbon Seals

Common seal type is face seal that is used in gas turbine engines to meet the challenging requirements of reliability, leak-tightness and less heat generation. Face seals are divided into four groups, which are contacting, self-acting, segmented radial, and self-acting segmented radial. Contacting face seal geometry is shown in Figure 2.4 and self acting face seal is shown in Figure 2.5. Segmented radial face seal is shown in Figure 2.6 and self acting segmented radial face seal is shown Figure 2.7.

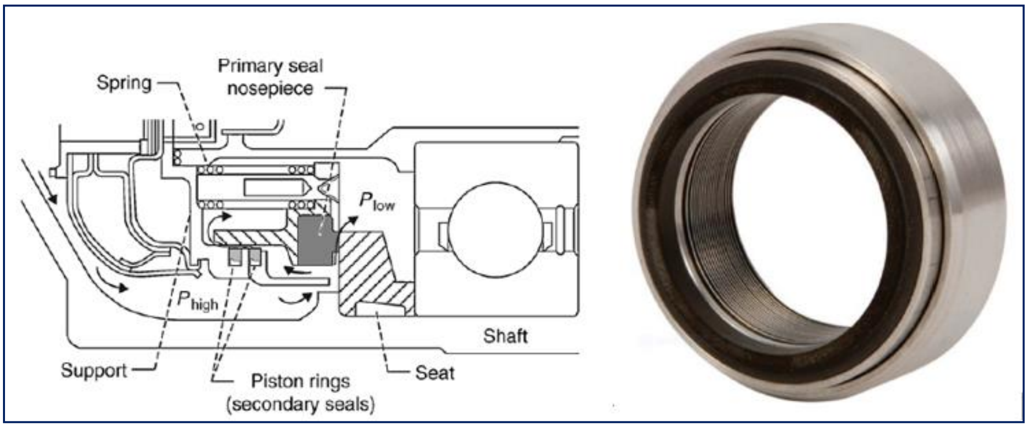


Figure 2.4 Contacting Face Seal [9],[10]

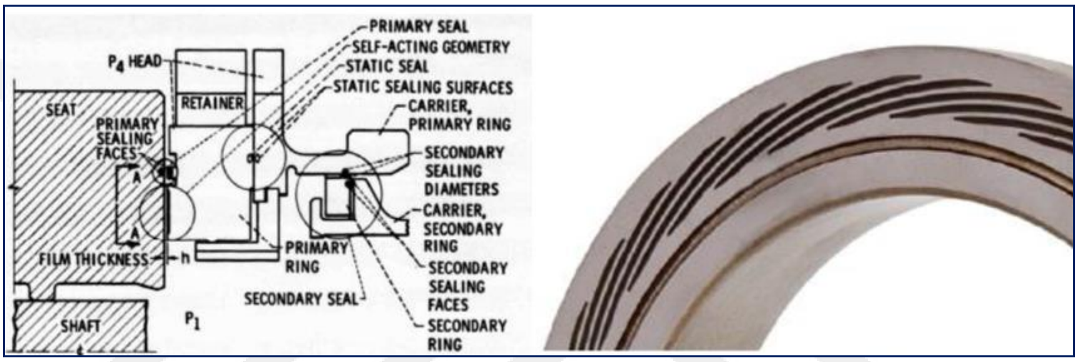


Figure 2.5 Self-acting Face Seal [10],[11]

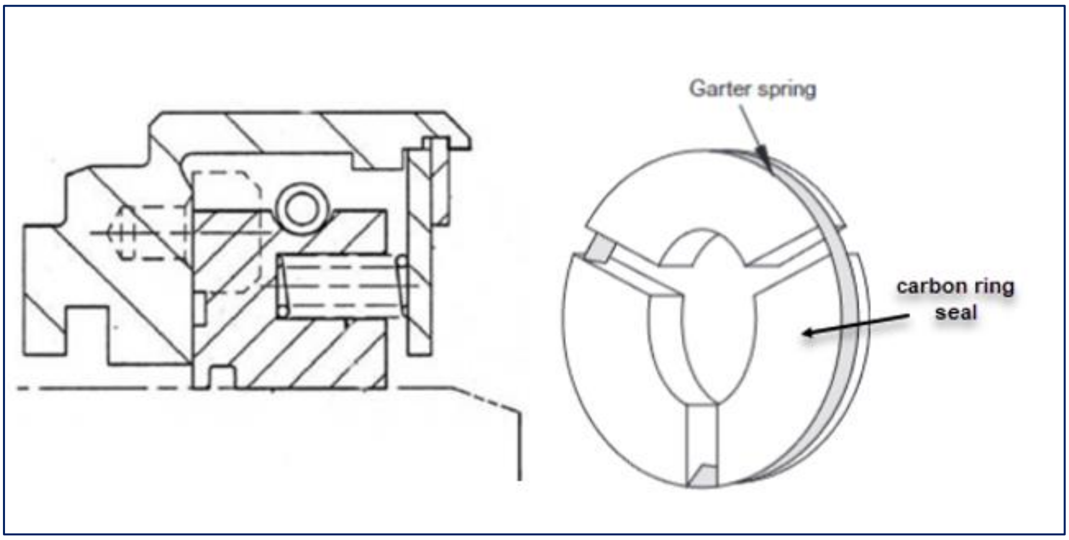


Figure 2.6 Segmented Radial Face Seal [10],[12]

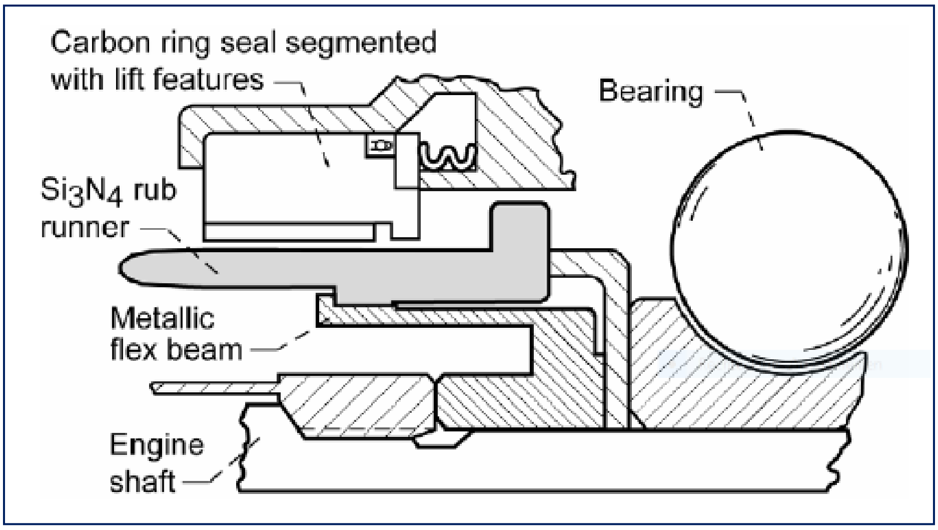


Figure 2.7 Self-acting Segmented Radial Face Seal [6]

Comparison of different seal types was made by several authors. The information should be read in context to the engine and seal technology at the time of publishing. As seen in Figure 2.8, face seal technology exceed in sealing performance (leakage amount) per outdated technology such as labyrinth, rotating ring and wornout circumferential seals. The self-acting face seal with the best sealing performance have some design difficulties compared to conventional face seal [6]. Pros and cons comparison is illustrated in Figure 2.9 for different seal type.

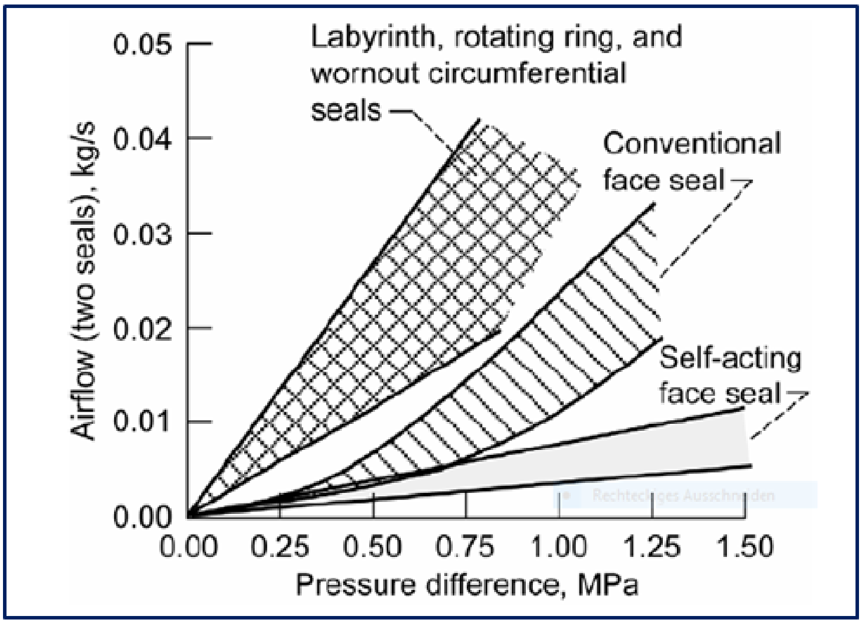


Figure 2.8 Comparison of Seal Types per leakage vs pressure difference [6]

	Type	Location in Aero-engine	Pros	Cons
Labyrinth Seal	Non-contacting (relatively large clearances)	Air to air seal Air to oil seal	Low wear, long service life	High leakage, undesirable contact
Brush Seal	Contacting	Air to air seal	Lower leakage, high pressure application, cheaper	Higher wear, friction, lower service life
Mechanical Seal	Contacting & Non-contacting (minimal clearances)	Air to air seal Air to oil seal	High speed, high pressure, low leakage, wide use	High wear, friction
Radial Lip Seal	Contacting & Non-contacting (minimal clearance)	Air to oil seal	Effective lubricating method with seal, low leakages	Unpredictable, wear due to dry running

Figure 2.9 Pros and Cons of Seal Types [13]

2.4 Terminology

Bearing areas and lubrication systems in aircraft engines must be sealed from hot regions. In these engines, the pressure difference can easily exceed 0.5 MPa, and in addition, the temperature ranges are 500 °C. Speed can be at very different levels. One of the most significant difficulty is coking when the oil in the bearing area comes into contact with hot air. To minimize this hot air, sealing performance is most significant parameter. For this reason, it is desired to reduce the seal clearance as much as possible. Even in some cases, it is required to work in contact but this has a problem such as heat generation due to contact. To be more detailed, cooling oil and groove can be added to the contact surface according to the working conditions and requirements.

For turboshaft engines, many labyrinth seals are being used for bearing chamber locations [6]. It has some disadvantages, such as oil leakage probability during the maneuver and high hot air leakage into the bearing chamber, which increases the heat generation within the chamber and requires more oil for cooling. Thus, advanced seal technology can be used to eliminate this kind of disadvantage. Carbon seal design trial has been performed for bearing chamber of a turboshaft engine.

The primary ring, mating ring, secondary seal, and spring are essential to design the mechanical face seals. The primary ring is one of the sealing surfaces which have a small relative axial or angular motion to achieve the required contact or clearance. Thanks to the secondary seal, the primary can move in the required direction. A mating ring is known as a guiding ring, as well. It is directly mounted to the shaft or housing, which has no motion capability and a second sealing surface. Some type of spring shall be used to hold the annular surfaces together in the absence of fluid pressure. 2D section view is shown in Figure 2.10.

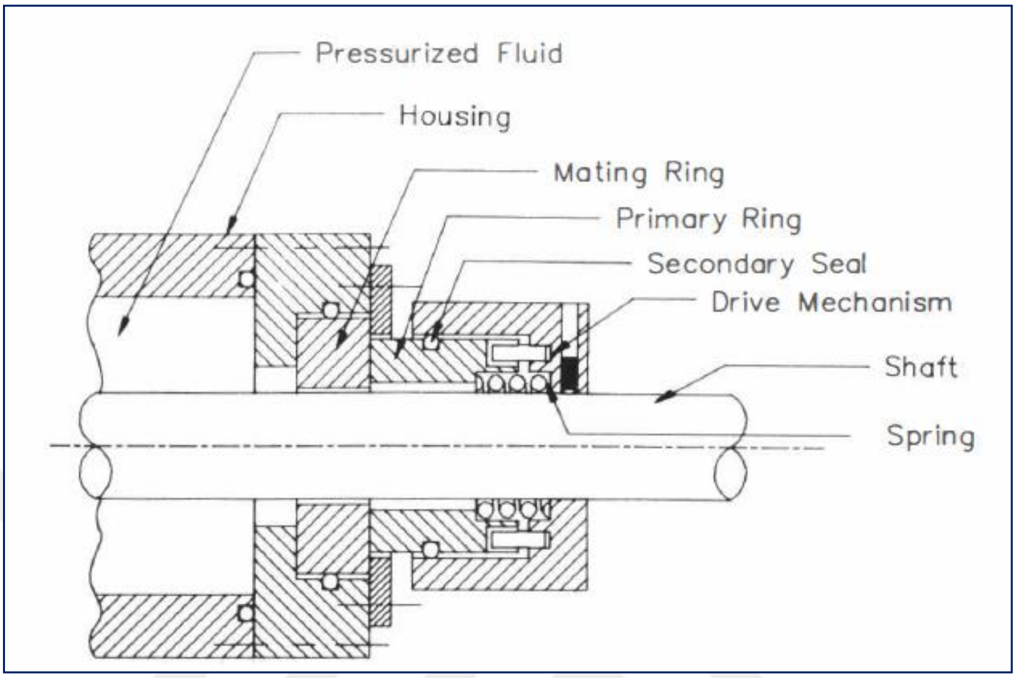


Figure 2.10 Mechanical Face Seal Configuration [14]

3. MATERIAL AND METHODS

Most types of carbon seals are currently used in turboshaft engines. A mechanical face seal (axial) was chosen for this study. The most important reason for making this choice is the available axial space. Circumferential face seals require more space than axial face seals and are more complex in terms of design. This complexity in design brings additional production and supply costs to engine manufacturers. Therefore, mechanical surface seal is the most feasible option for the engine considered within this study. The challenge is the linear velocity for this study. Linear velocity at the related region is recommended to be lower than 100 m/s, and for our case, it is greater than desired value (+20%). However, it is predicted that this speed difference will not cause a problem since the heat generated can be removed from the relevant region. In addition, this speed limit is given as 125 m/s in different sources [10],[14].

3.1 Criteria

Face seal design has been performed according to related area dimensions. Design criteria are listed below.

- Leakage rate < 10% of the current design, which is a labyrinth seal, this parameter is the most important one since the primary task of the seal which is used in turbomachinery is to restrict air or oil leakage from pressurized area to undesired area,
- 0.8 < balance ratio < 1,
- Oil temperature < temperature limit based on oil coking ,
- Component temperature per maximum allowable service temperature of materials,
- Strength,
- Manufacturability/availability and cost,
- Wear/friction capability, this parameter affects the seal leakage performance, heat generation, and life of the carbon ring and runner.

3.2 Geometric Description

General overview and assembly condition can be seen in Figure 3.1. In this section, all required part will be defined in detail one by one. Additionaly, alternative carbon seal has been designed and it is illustrated in Figure 3.2.

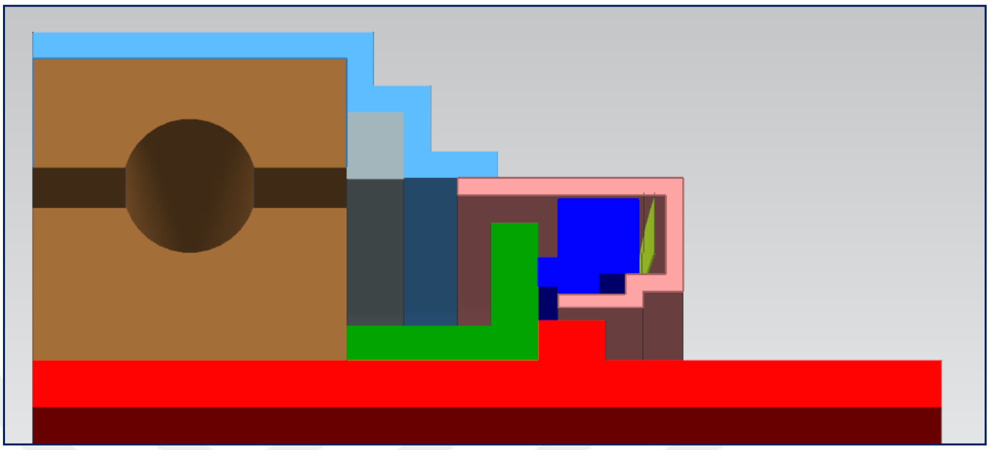


Figure 3.1 Design Configuration of Carbon Seal for No2 Bearing

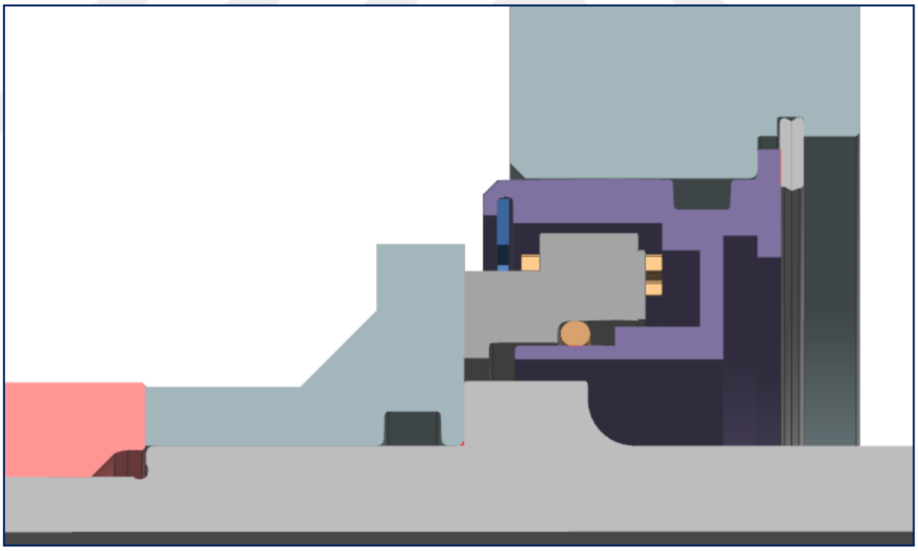


Figure 3.2 Alternative Carbon Seal Design for Same Location

The primary ring illustrated in Figure 3.3 is sized with respect to turboshaft engine bearing location. Two anti-rotation features are used at equal intervals circumferentially to prevent the force that will cause rotation due to friction during operation. An o ring is placed in the inner diameter region to prevent sealing problems that may occur due to the gap between the primary ring and its case. The diameter of the mating face is selected as small as possible in order to reduce the linear velocity of this region. A spring is placed to ensure about continuous contact on the opposite surface of the mating surface.

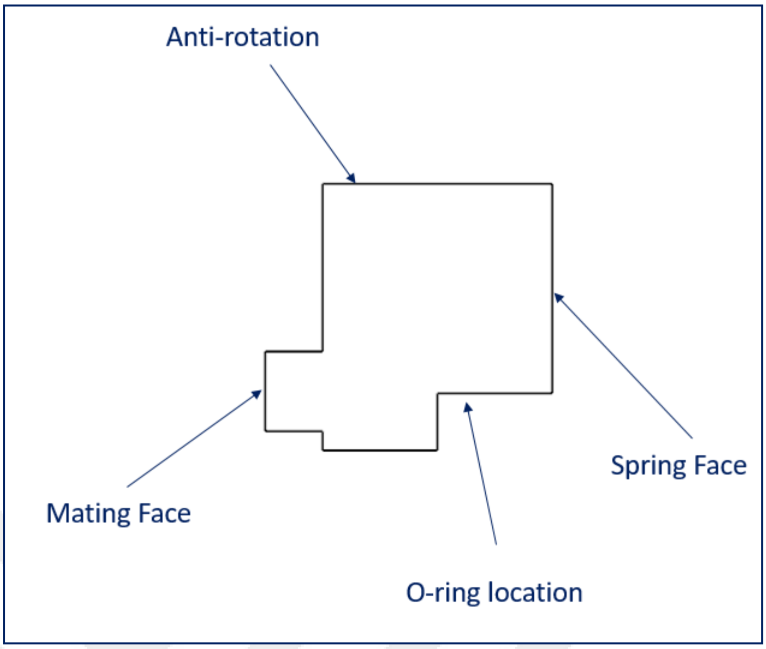


Figure 3.3 Primary Ring Design

A case design illustrated in Figure 3.4 is made to hold the primary ring. This case is mounted directly on the bearing support part. Since it is the conjugate of the primary ring, it has o ring, and spring surfaces similarly. In addition, it also has a retainer ring. The task of this ring is to prevent the movement intent of the primary ring due to the force by the spring placed at the rear.

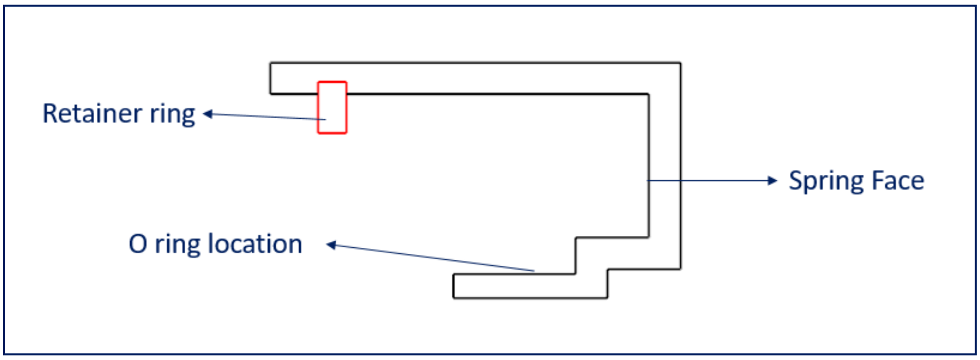


Figure 3.4 Primary Ring Case Design

The bearing support part illustrated in Figure 3.5 holds the bearing, primary ring case, and oil jet all together. These mentioned components are mounted on the bearing support part with an interference fit. In addition to the holes or features added for the engine design, there are oil supply lines for the oil jet.

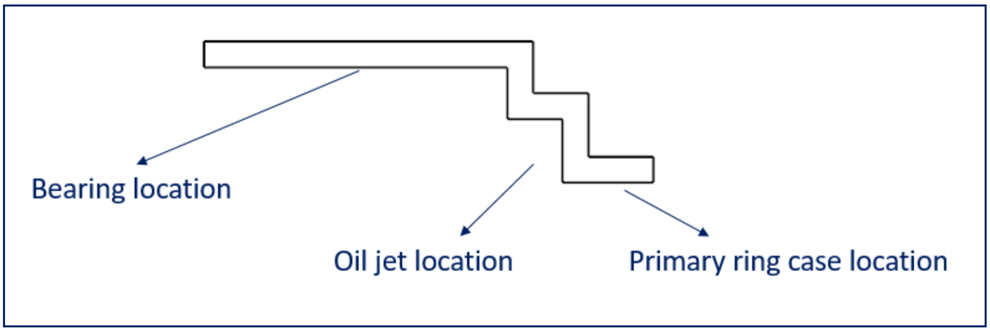


Figure 3.5 Bearing Support Part Design

The runner illustrated in Figure 3.6 is designed to be rotated at desired circumferential speeds with an interference fit with the tie-shaft and preload between tie shaft and bearing. Additionally, the runner lateral surface is in contact with the primary ring. Therefore, the material of the runner shall be chosen in such a way that it will not be damaged in this case or it is necessary to make special coatings on this region.

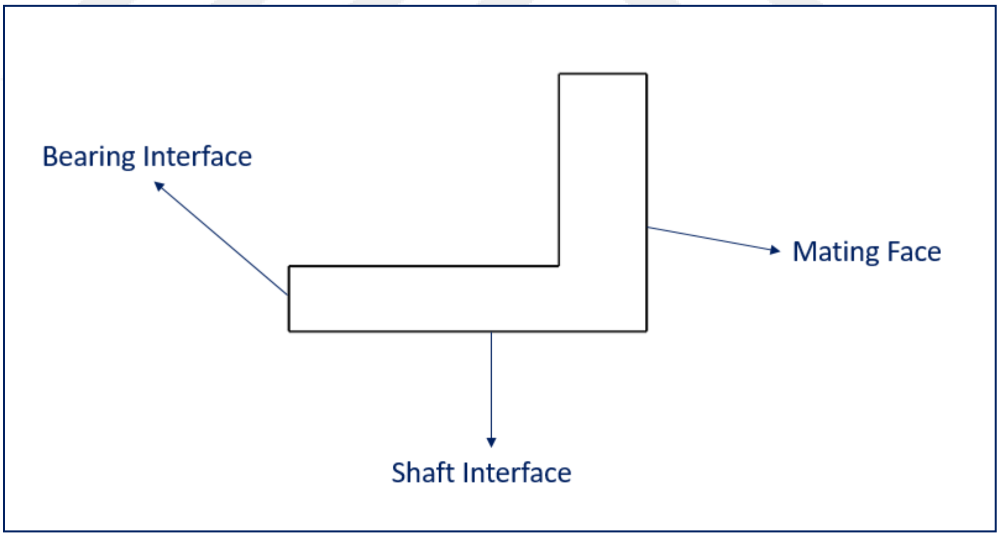


Figure 3.6 Runner Design

Seal leakage can be calculated based on the average gap between the primary ring and runner. During operation, no leakage is expected since there will be no gap between the carbon and runner. However, due to wear, surface roughness and manufacturing tolerances, there will be a small amount of leakage. This leakage level will be calculated per manufacturing tolerances and wear assumptions at manufacturing phase.

Wear is one of the most common problem for mechanical face seals. This is caused by many forces acting on the contact surface. The wavy surface formed as a result of wear increases the stresses in the relevant region and also dramatically worsens the sealing performance. In order to keep the sealing performance under control throughout its entire life and not to see a failure due to stress increases, the amount of wear must be well calculated and validated by testing. During wear life calculation, all forces acting on seal shall be taken into account. These forces illustrated in Figure 3.7.

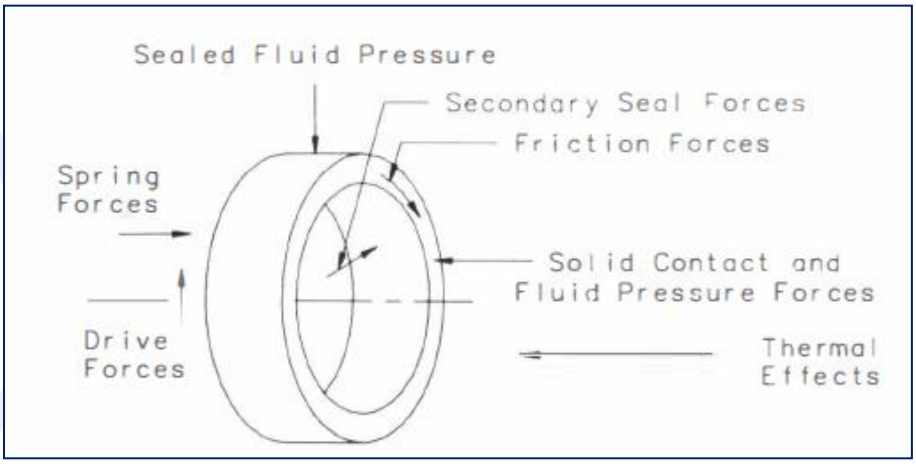


Figure 3.7 Loads Acting on Face Seals [14]

3.3 Design Phase

In design phase, some mechanical design requirement and sealing performance parameter have been checked. All design headings will be detailed in this section.

3.3.1 Balance Ratio

Balance ratio is used to control the axial force equilibrium that is defined in Figure 3.8 and Figure 3.9.

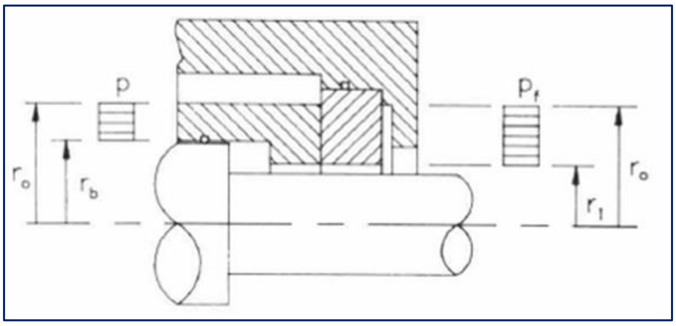


Figure 3.8 Balance Ratio Definition (outside pressurized seal) [14]

$$P\pi({r_o}^2 - {r_b}^2) = P_f\pi({r_o}^2 - {r_i}^2) \tag{3.1}$$

$$B = \frac{P_f}{P} = \frac{{r_o}^2 - {r_b}^2}{{r_o}^2 - {r_i}^2} \tag{3.2}$$

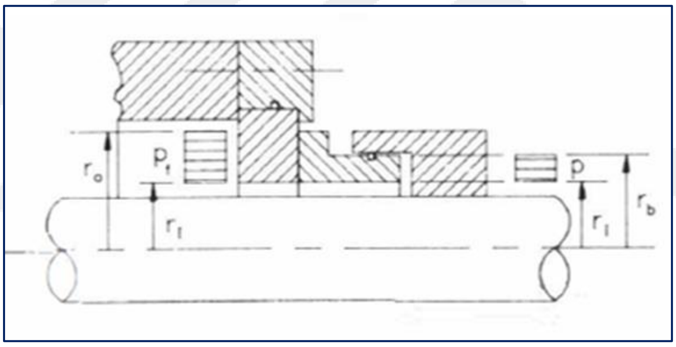


Figure 3.9 Balance Ratio Definition (inside pressurized seal) [14]

$$P\pi({r_b}^2 - {r_i}^2) = P_f\pi({r_o}^2 - {r_i}^2) \tag{3.3}$$

$$B = \frac{P_f}{P} = \frac{{r_b}^2 - {r_b}^2}{{r_o}^2 - {r_i}^2} \tag{3.4}$$

Balance ratio is such an essential and widely used term that it defines the ratio between the average load for inside and outside pressure.

Balance ratio is greater than 1 means the seal is unbalanced (average pressure on the face is greater than sealed pressure). If average pressure is less than sealed pressure, the seal is balanced. Many balanced types of seals are used in the high-pressure area and most low-pressure seals operate at $B > 1.0$ because of the convenience of design.

For max take off condition of turboshaft engine forces are shown in Figure 3.10.

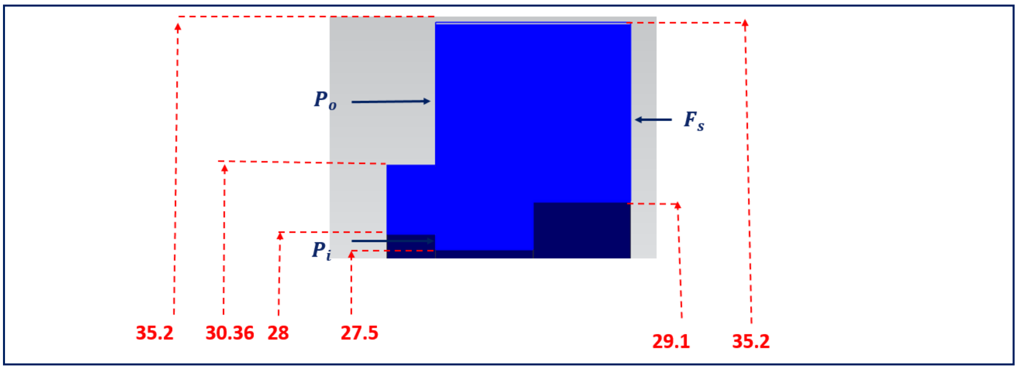


Figure 3.10 Force Equilibrium

$$P_i = 0.196\ MPa$$

$$P_o = 0.13\ MPa$$

$$F_s > 0.13 \times 960 + 0.196 \times 86 = 124\ N$$

To be conservative resultant force multiplied by 2 and spring force is found below.

$$F_s = 250\ N$$

Spring shall be used to hold the surfaces together in the absence of pressurized air. However, it shall be selected appropriately since it affects leakage amount and friction load on the interface. Instead of balance ratio, force balance has been controlled as the runner is fixed between the shaft shoulder and bearing. According to force balance calculation, the required spring force is 250 N and the RW-0125 spring washer has been selected to be used between carbon and carbon case. Geometric description of this spring is illustrated in Figure 3.11. In the assembly condition, stress at the seal interface due to spring force is recommended as between 0.03 MPa to 0.34 MPa, but for some cases, it can be greater. For the current design, this value is about 0.48 MPa and will be evaluated in the following pages.

Part Number	RW-0125
Spring Type	Wavo
D_b: Operates in Bore (in)	1.250
D_s: Clears Shaft Diameter (in)	1.081
Load (lb)	110.0
Work Height (in)	0.110
H_f: Free Height REF. (in)	0.138
CAD Height (in)	CAD Height can only be the Work or Free Height
Number of Waves/Turn	3.0
Number of Turns	1
Number of Shims	0
t: Wire Thickness (in)	0.065
Spring Rate REF. (lbs/in)	3929

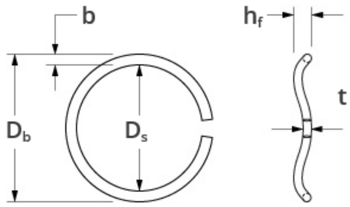


Figure 3.11 Selected Spring Type [15]

3.3.1.1 Material Selection

Seal face materials are selected based on their strength, friction and wear properties, sliding compatibility with each other, heat transfer characteristics, thermal environment resistance, and chemical resistance. Because there are so many criteria to be met, there are only a few materials that can be used for seal face materials.

Mechanical and physical properties to be considered are listed below.

- Tensile & compressive strength: seal rings are loaded by hydrostatic pressures, friction, and drive forces. Thus, tensile & compressive strength of material shall be considered in terms of integrity.
- Modules of elasticity: ease of deforming is an essential parameter because deformed face means more leakage. On the other hand, brittle material can easily be over-stressed due to tight fits and thermal loads. These two cases are directly related with modules of elasticity.
- Coefficient of Thermal Expansion: Seals are running in different temperature conditions and it causes thermal gradient on the seal ring. That’s why, the thermal expansion coefficient shall be considered during material selection to eliminate thermal distortion and thermal shock.
- Thermal Conductivity: removed heat at the surface depends on the thermal conductivity of the seal ring material for the cooling process. Therefore, low

thermal conductivity is an undesirable property because it means high temperature.

- Porosity: Some seal materials are porous. Porosity can cause seal leakage directly through the seal material.

According to these design recommendations, the material of carbon ring is selected as FE679Q [16] because considering these recommendations, it is one of the best options and has widespread use in turbomachinery. In addition to that, bearing and runner materials are selected as M50 [17] and SiC [18], respectively. Other components (shaft, oil jet, squirrel case, carbon ring case) material is Inco718.

One of the most crucial parameters is environmental and metal temperature due to thermal distortion. The thermal behavior of mechanical seal can affect the seal performance, intensionally. Seal geometry is changing by temperature gradient during the operating envelope. Therefore, the environment temperature of the seal shall be predicted and then the effects on the fluid, the mechanics, and the materials shall be assessed. Thermal effects on seal behavior can be listed below.

- Mechanical effects: temperature gradient along the radius or circumference causes radial taper or waviness which is shown in Figure 3.12.

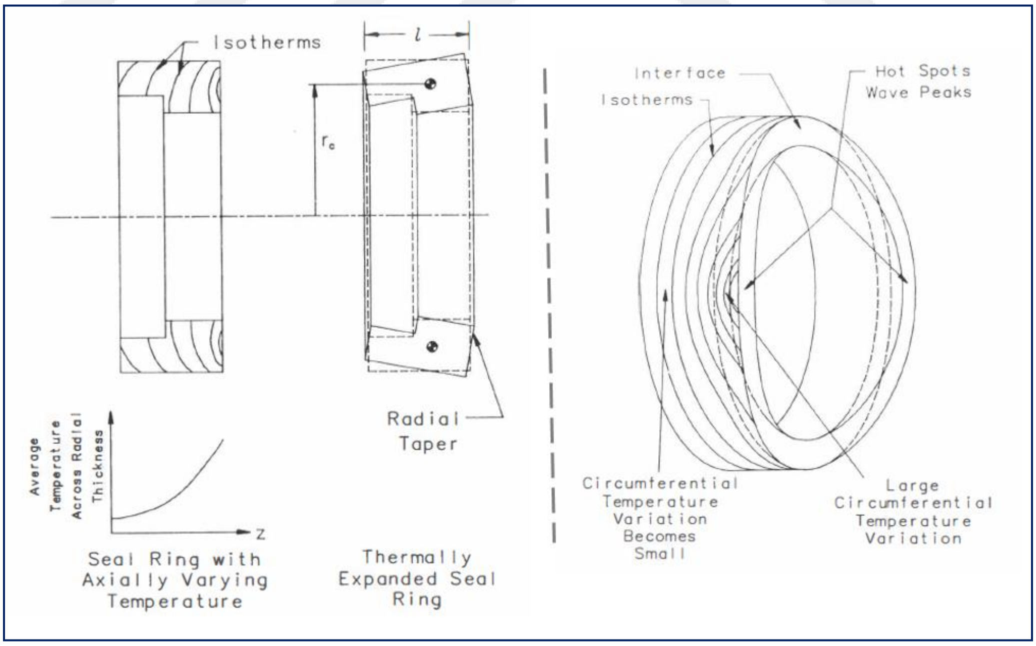
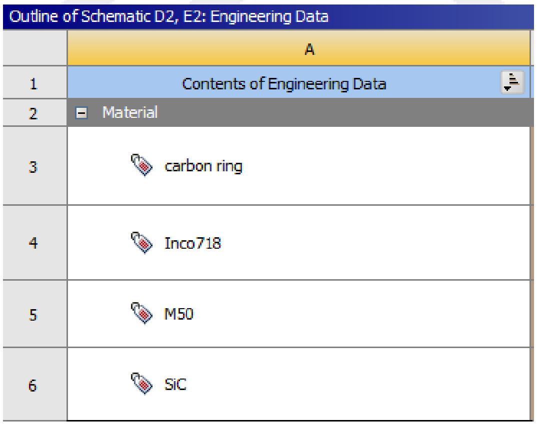


Figure 3.12 Radial Taper and Waviness [14]

- Effects on cooling oil: at elevated temperatures, oils can be cooked and the structure of oil (crystallization, scaling, and solidification) can change. Thus, oil can not do cooling duty.
- Effects on seal materials: All materials' properties in nature vary according to temperature. Therefore, environmental temperature shall be taken into consideration per material limit.
- Seal material temperature may increase due to interface friction which is the primarily dominant parameter, the viscous drag of seal assembly which is seen at very high speeds and process fluid.

3.3.2 Thermal Analysis

Ansys ® workbench steady-state thermal analysis tool has been used to see the maximum temperature and temperature distrubition on related parts. In order to make this analysis, first of all, the necessary material information must be defined, it is shown in Figure 3.13. Density, thermal expansion, young modulus, Poisson's ratio, and thermal conductivity have been defined. In fact, not all of these mentioned are necessary for this analysis, but since the same material data will be used in stress analyses in the future, all of the stated ones are defined.



Outline of Schematic D2, E2: Engineering Data

	A
1	Contents of Engineering Data
2	Material
3	carbon ring
4	Inco718
5	M50
6	SiC

Figure 3.13 Defined Materials for Thermal Analysis

All contacts illustrated in Figure 3.14 are defined as bonded. For current case, the effect of the mesh structure used in temperature analysis on the results is negligible. Therefore, the mesh structure to be used for the temperature analysis was made as course. Mesh size was chosen as one-millimeter body sizing. Generated mesh structure is illustrated in Figure 3.15.

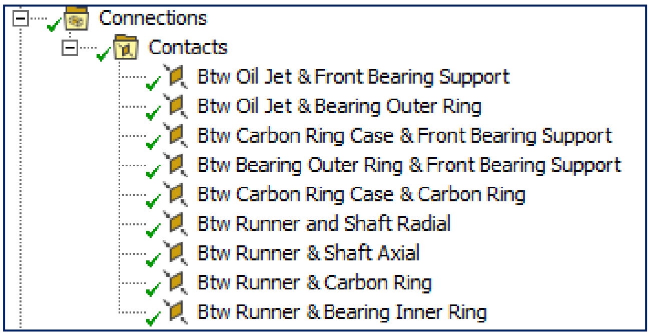


Figure 3.14 Defined Contacts for Thermal Analysis

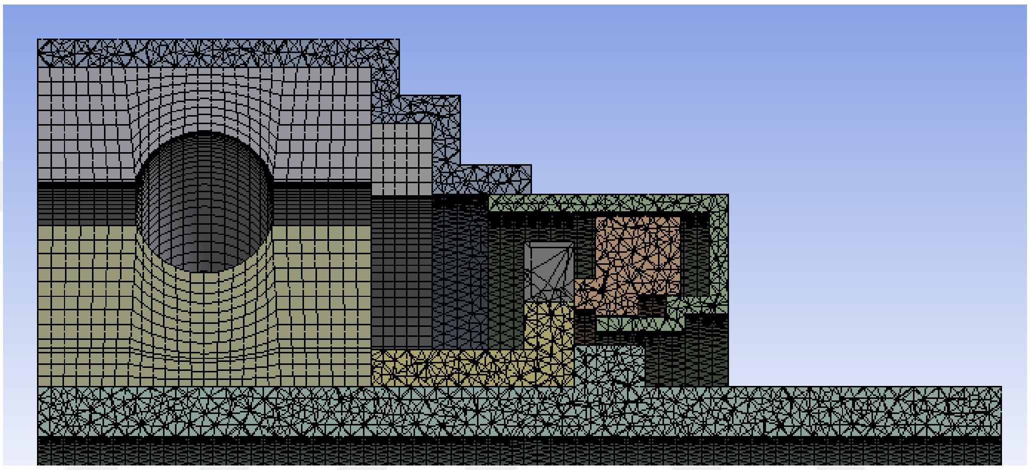


Figure 3.15 Used Mesh Structure for Thermal Analysis

Mesh quality check is applied to check aspect ratio, Jacobian and skewness of the mesh elements, details illustrated in Figure 3.16. To see effect of mesh quality full hex dominant mesh has been applied for different mesh size which is illustrated in Figure 3.17. As seen in figure temperature result distribution is almost same and max temperature result difference is 1.1% and minimum temperatures are same.

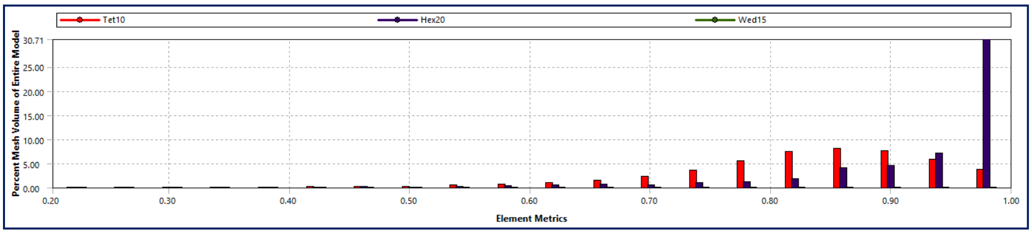


Figure 3.16 Mesh Type Distrubition for Thermal Analysis

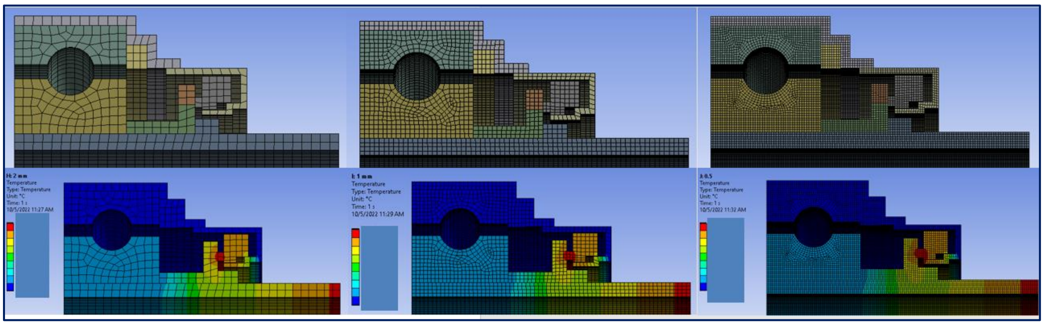


Figure 3.17 Mesh Comparison

As the boundary condition, the possible temperature conditions in the No2 bearing region of the turboshaft engines are given. No boundary condition is defined for the regions close to the contact surface between the carbon and the runner. This is because if the boundary condition is defined for the nearby regions, its effect on the region of interest will be dominant. The generated heat between the runner and the carbon ring was defined as positive heat flux. Conversely, the heat removed from the runner's surface by the oil jet is defined as negative heat flux as runner is cooled with synthetic oil. Boundary locations are shown in Figure 3.18.

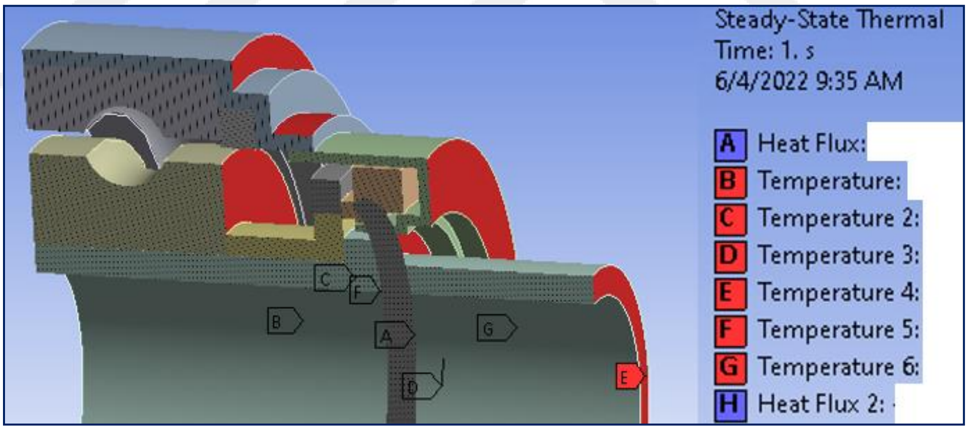


Figure 3.18 Boundary Conditions of Steady State Thermal Analysis

Material temperature is assessed based on heat generation due to friction between carbon and runner. $\mathbf{3.38}\ \boldsymbol{W/mm^2}$ heat flux has been applied on Ansys ® Workbench and the maximum temperature is obtained as $\mathbf{390}\ \boldsymbol{°C}$ in the region between carbon and runner. Temperature result for w/o cooling is shown in Figure 3.19. Equations [19] and result are shown below.

$$T_r = P_{tot} \times A_f \times f \times \frac{D_m}{2000}\ [Nm] \tag{3.5}$$

$$E_P = \frac{T_r \times N}{9548} \; [kW] \tag{3.6}$$

$$f = 0.15$$

$$E_p = 1 \; kW$$

$$applied\ heat\ flux = 3.38 \; W/mm^2$$

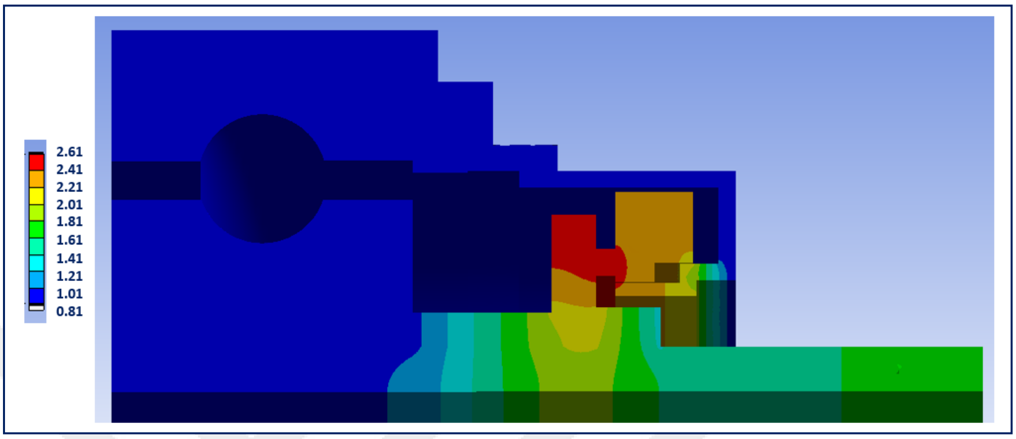


Figure 3.19 Non-dimensional Temperature Distribution w/o Cooling

As the current situation, the calculated metal temperature may be greater than the material serviceable limit. Thus cooling process shall be designed to reduce metal temperature. In some cases, process fluid may be used to cool the seal if the temperature of the process fluid is colder that seal, sufficiently. However, the probability of this case is low. Rarely, if there is enough space, process fluid may be cooled with the help of a mechanism such as a heat exchanger. In addition to this cooling method, separate cooling fluid can be used to reduce seal temperature, but in this case, another fluid flow path shall be considered. This method is called as direct cooling, flushing, or quenching.

For the current case, the temperature of the process fluid in the bearing location is not as cold as enough to quench the seal. Thus separate cooling oil should be added to decrease the maximum temperature in this specific region. Oil flow direction and location is shown in Figure 3.20. Effectiveness of cooling process is added as splash ratio [20]. The oil flow rate is taken as $0.0064 \; kg/s$ and $0.99 \; W/mm^2$ is extracted from the back surface of the runner. Equation [20] is stated below.

$$q = s \times \dot{m} \times c_p \times \Delta T \tag{3.7}$$

$$q = 0.5 \times 0.0064\,\frac{kg}{s} \times 4.193\,\frac{kJ}{kgK} \times 50\ K = 0.67\ kW$$

$$applied\ heat\ flux = -0.99\ W/mm^2\ [19]$$

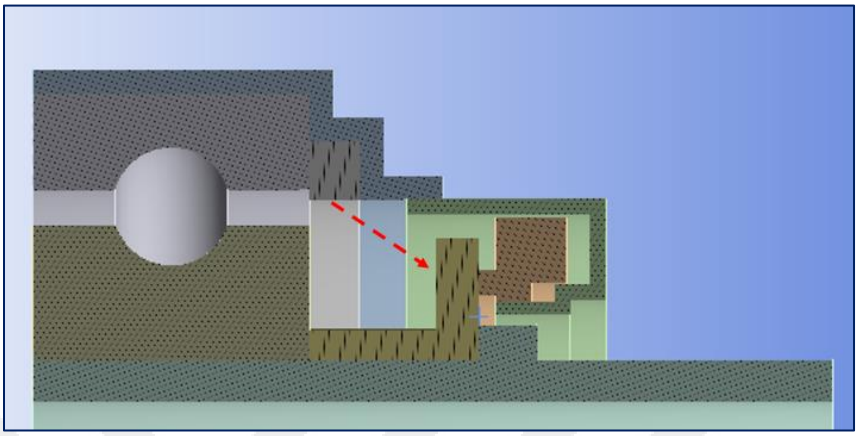


Figure 3.20 Cooling Calculation

The maximum temperature is decreased to 250 $°C$. As a result, with a quenching mechanism, seal temperature illustrated in Figure 3.21 is lower than the material serviceable limit.

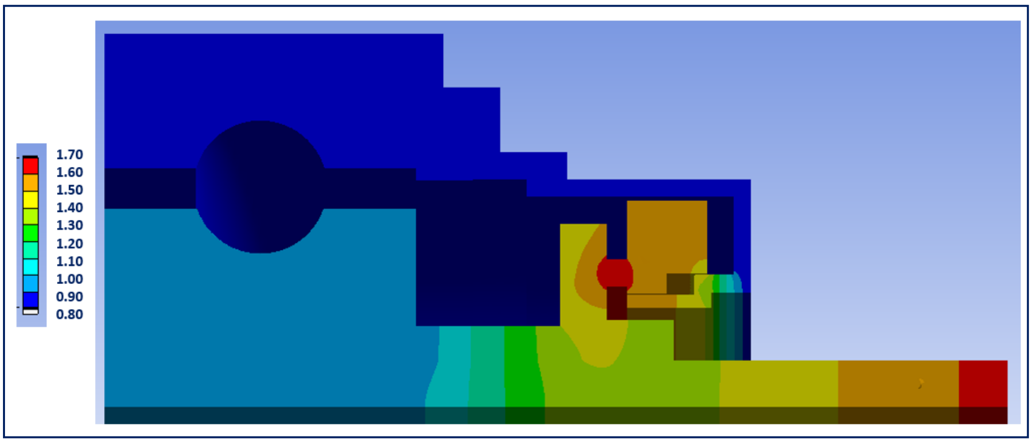


Figure 3.21 Non-dimensional Temperature Distribution w/ Cooling

3.3.3 Secondary Seal Selection

The role of secondary seals is just as important as the role of the sliding faces in a mechanical face seal. Usually, materials of the secondary seal are used as elastomeric materials. The durometer range shall be considered while selecting the secondary seal material since it is hard to extrude the seal which has a high durometer. Softer o rings give lower break-out sliding friction. Due to installation issue, o ring shall has enough elongation. As all materials work under stress, the tensile strength of o ring is also an important parameter. Temperature range is the most critical parameter for o ring selection in case of hot and cold conditions. If o ring is working under hot condition, after some point material is softening. In case of cold condition some o ring materials will be hard and brittle. A compression set can be defined as a dimensional recovery of the seal cross-section from the squeezed condition. Therefore, elastomeric materials are used as a secondary seal.

O-ring selection has been performed according to the service temperature of o rings and they are illustrated in Figure 3.22.

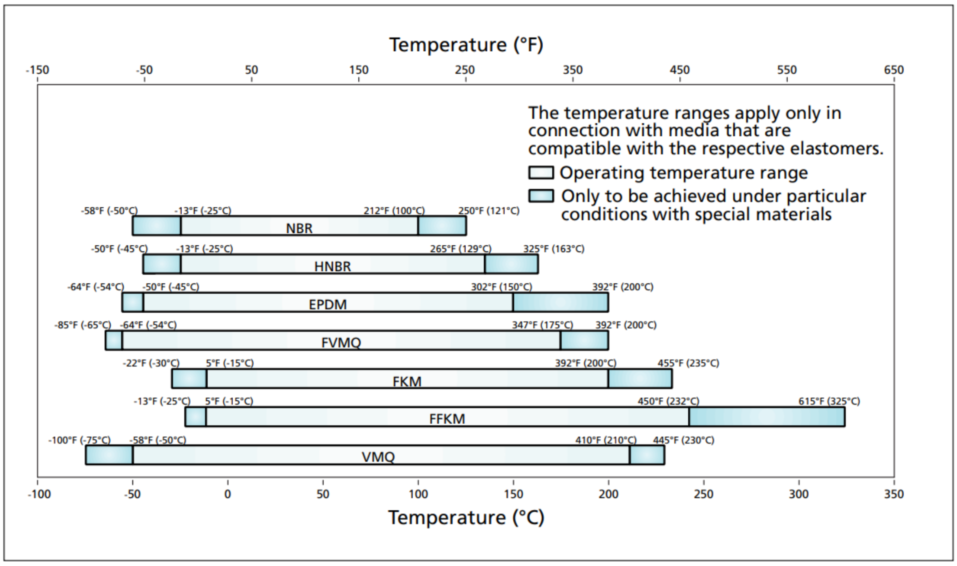


Figure 3.22 O Ring Service Temperature [21]

Related region maximum temperature is 240 ℃ shown in Figure 3.23 and only FFKM was capable to tolerate this elevated temperature. These o ring materials are known material types which are used in turbomachinery. Thus, another properties which shall be evaluated is not stated in this study.

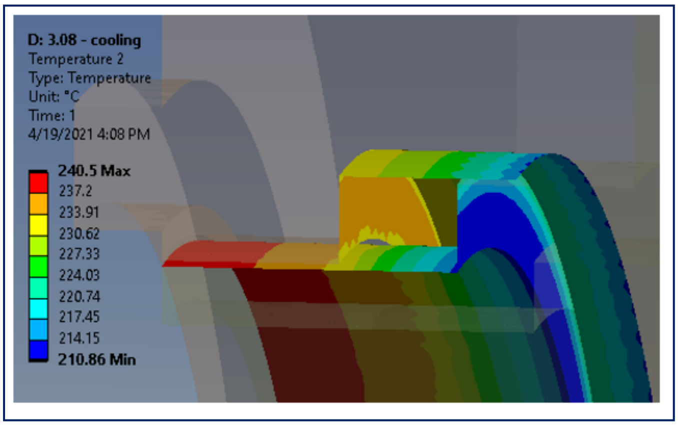


Figure 3.23 O Ring Region Metal Temperature

3.3.4 Strength Analysis

For stress assessment, Ansys ® workbench static structural analysis tool has been used. Similar to the thermal analysis, the same materials were identified, they are shown in Figure 3.24. Unlike the thermal analysis, not all relevant parts were run in a single analysis. Separate analyzes were run for rotating and non-rotating parts and the results were evaluated.

Outline of Schematic D2, E2: Engineering Data

	A
1	Contents of Engineering Data
2	Material
3	carbon ring
4	Inco718
5	M50
6	SiC

Figure 3.24 Defined Materials for Structural Analyses

As in nature, all contacts are defined as frictional, shown in Figure 3.25.

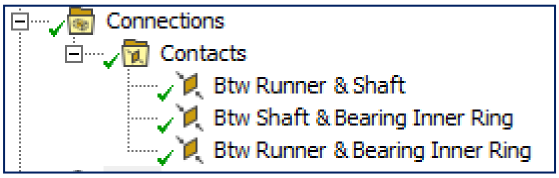


Figure 3.25 Defined Contacts for Structural Analysis for Rotating Parts

The importance of mesh structure in structural analysis is a well-known fact. Obtained mesh structure is shown in Figure 3.26. Therefore, the mesh structure used was chosen as hex dominant, shown in Figure 3.27. In addition, the size of the network structure is also defined smaller as 0.7 mm. This has been used to obtain accurate results even if it prolongs the analysis time.

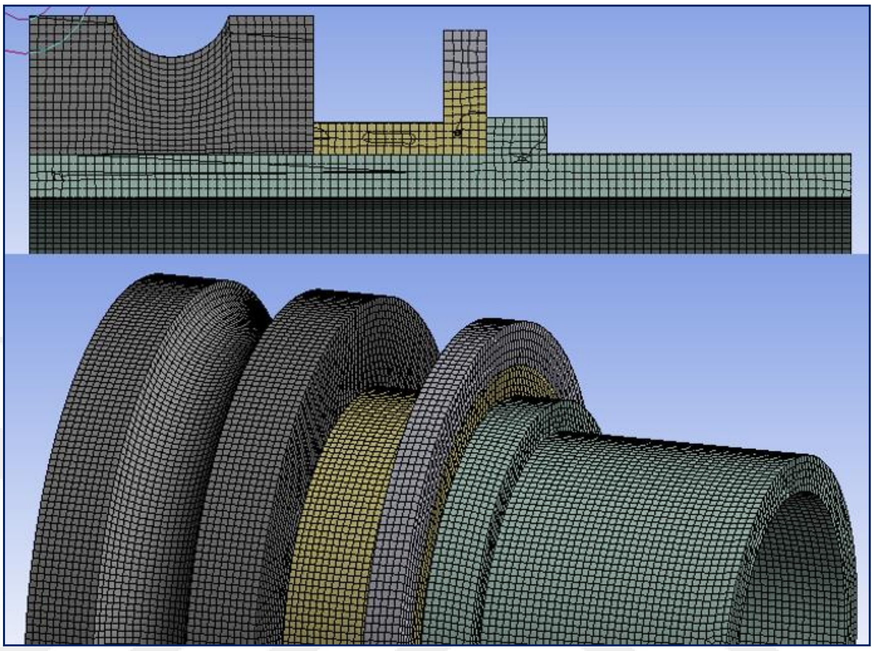


Figure 3.26 Used Mesh Structure for Structural Analysis for Rotating Parts

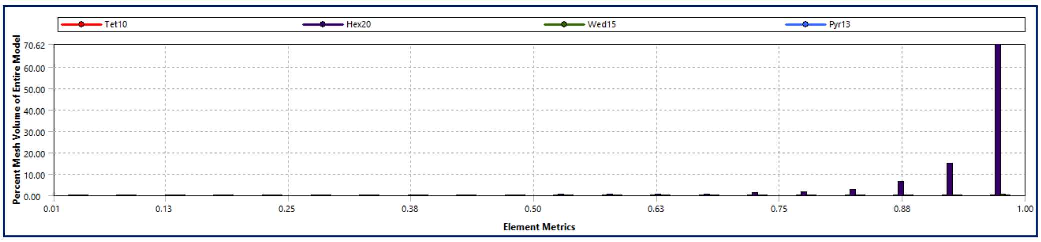


Figure 3.27 Mesh Type Distrubition for Structural Analysis for Rotating Parts

A boundary condition is defined as it is held axially and circumferentially from the contact surface of the balls in the bearing used. The preload force which is applied to hold the rotating parts together during assembly has been applied to the tie shaft. In addition, the axial force exerted by the carbon ring on the runner is also included in the analysis, although it is minimal compared to the aforementioned tie shaft force. These all boundary condition locations are shown in Figure 3.28.

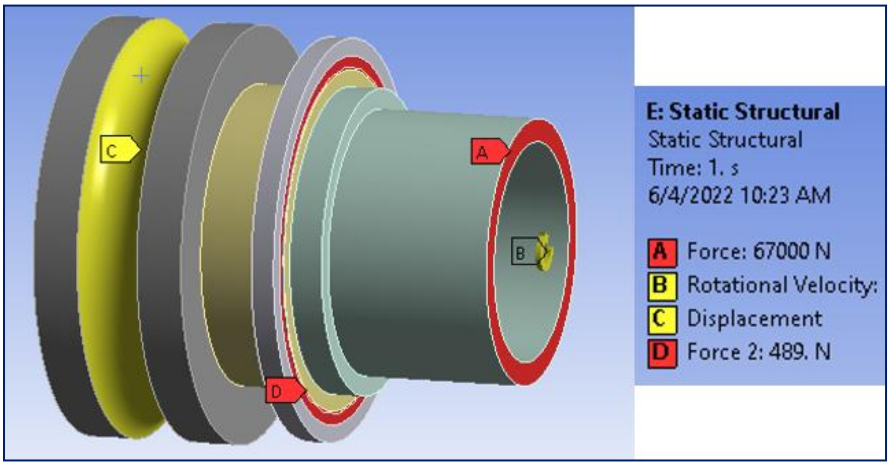


Figure 3.28 Boundary Conditions of Steady State Structural Analysis for Rotating Parts

The output of the temperature analysis was used as the input of the structural analysis. The applied temperature information was validated and no difference was observed in Figure 3.29.

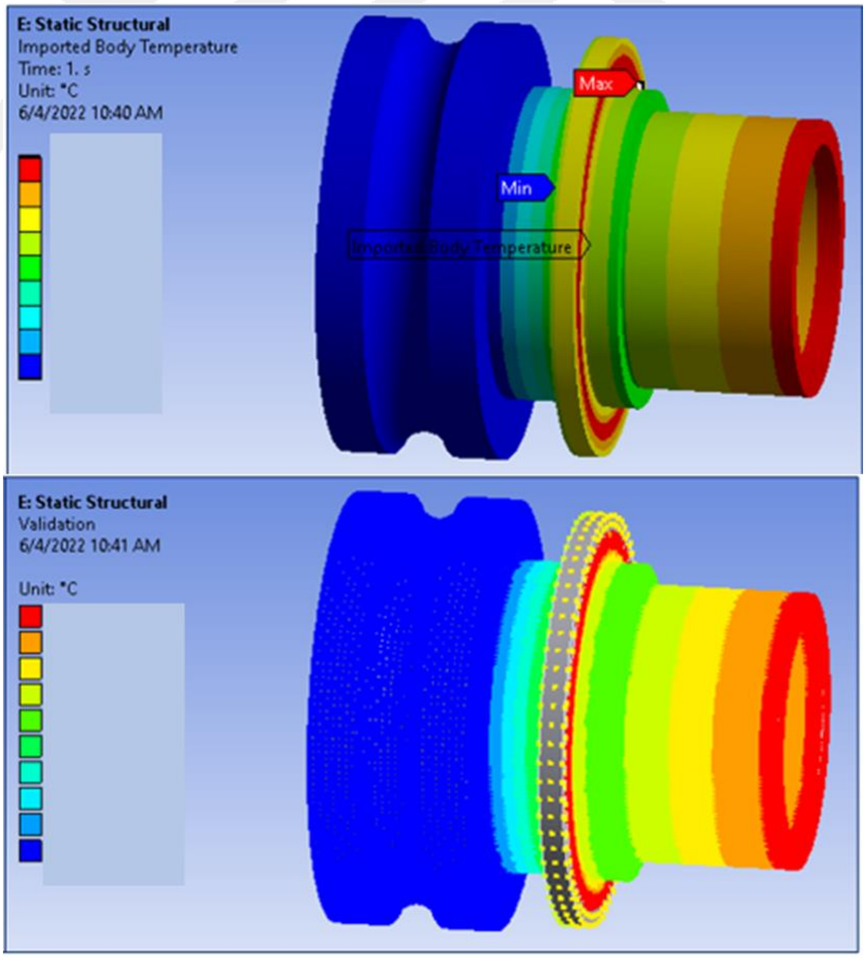


Figure 3.29 Applied Temperature and It’s Validation for Rotating Parts

In this study, only the yield stresses of the materials were used to evaluate the stress results, see Figure 3.30 for stress result. Even if the mechanical face seal is not implemented in the related region, the tie shaft, bearing, and front bearing structure will be included and evaluated by a design engineer. Thus, a stress assessment has been performed for relevant parts, not for all those parts.

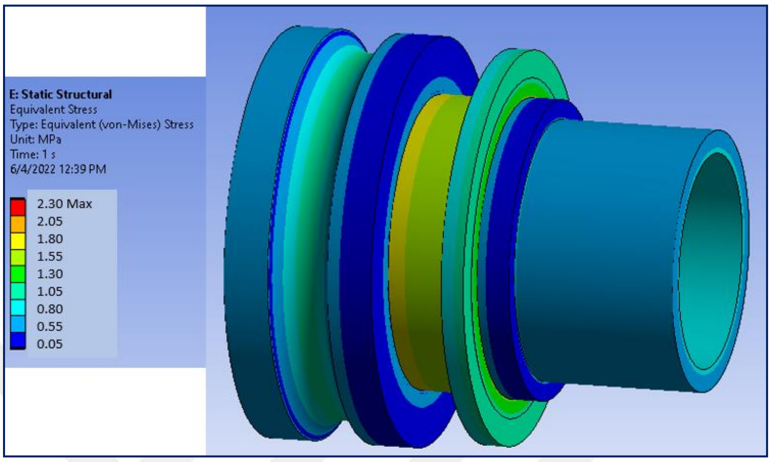


Figure 3.30 Non-Dimensional Stress Result for Rotating Parts

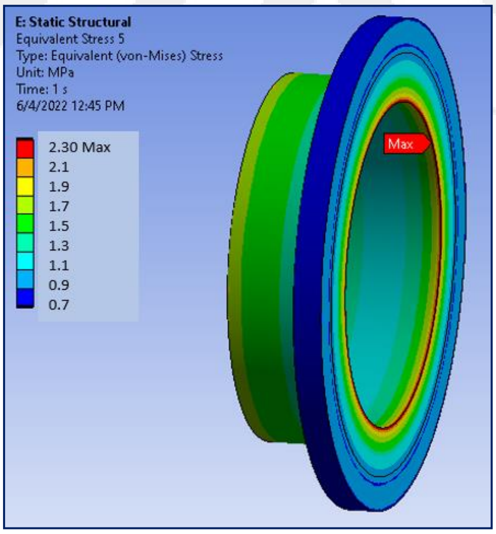


Figure 3.31 Non-Dimensional Stress Result for Runner

The maximum stress observed on the runner is 690 MPa, shown in Figure 3.31. This value is lower than the yield stress of the runner material, silicon carbide (SiC). Therefore, a safe design was made according to this comparison.

The same study has been performed for the stationary parts. Since all steps are similar, they have not been explained in detail again. Only the relevant inputs and outputs are given in this study, Figure 3.32 and Figure 3.33.

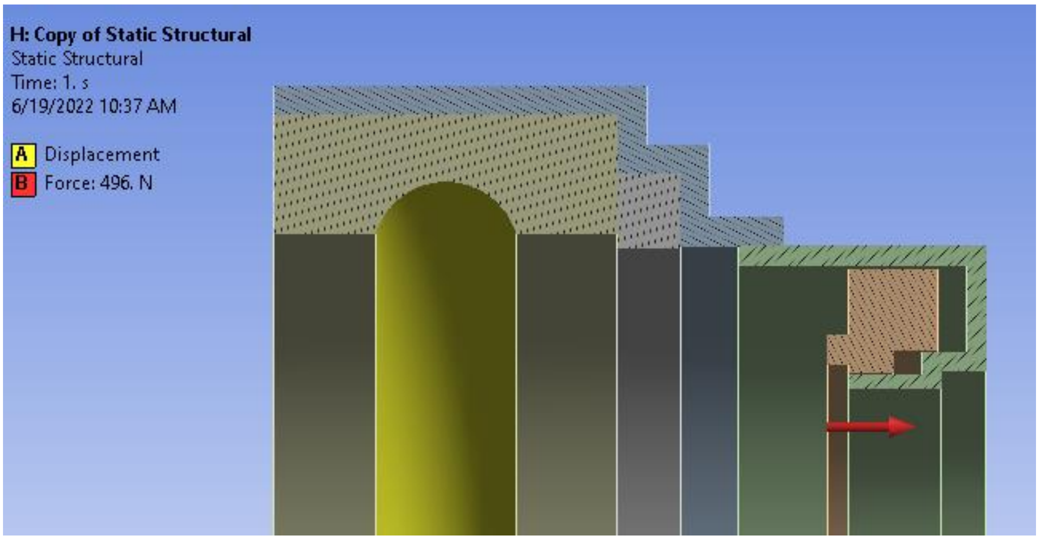


Figure 3.32 Boundary Conditions of Steady State Structural Analysis for Stator Parts

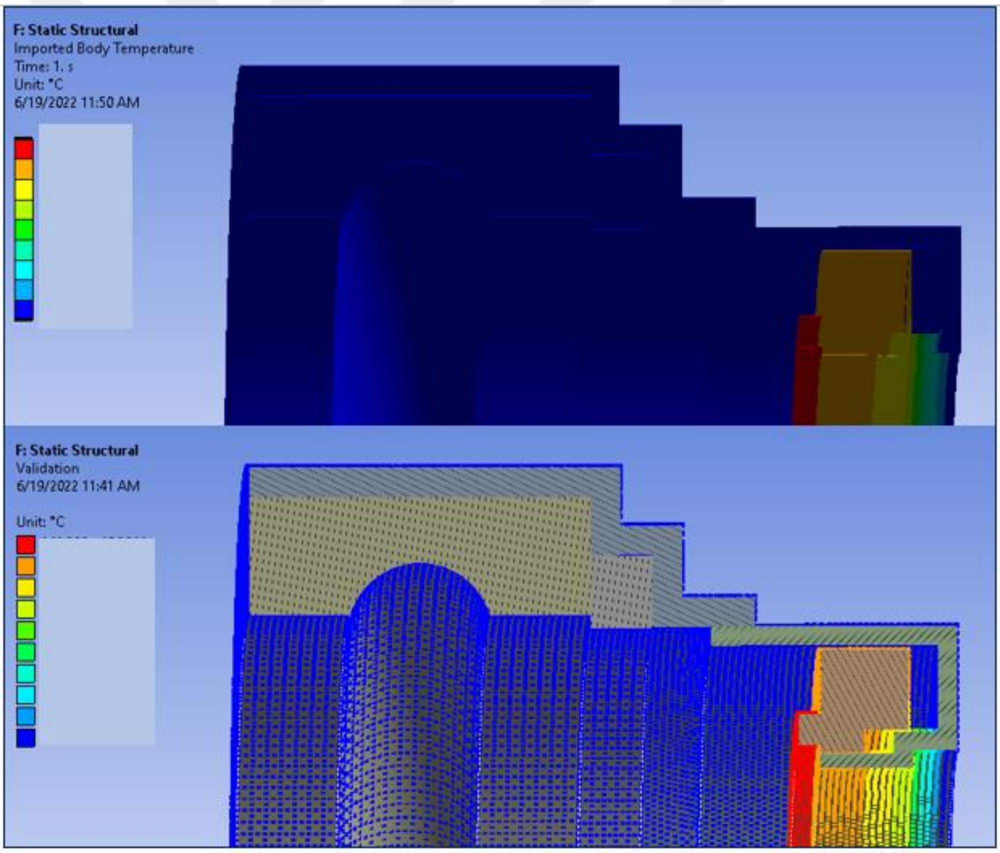


Figure 3.33 Applied Temperature and It's Validation for Stator Parts

As seen in Figure 3.34, the maximum stresses observed on the carbon case.

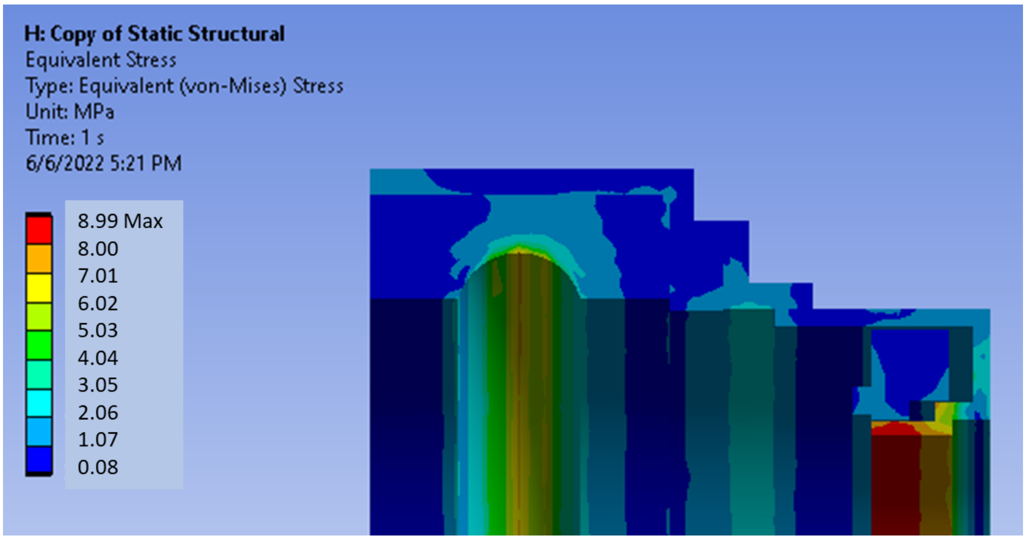


Figure 3.34 Non Dimensional Stress Result for Stator Parts

The carbon ring are 188 MPa shown in Figure 3.35 and 52 MPa shown in Figure 3.36, respectively. These values are lower than the yield stress of the materials for each part (Carbon and Inco718). Therefore, a safe design was made according to this comparison.

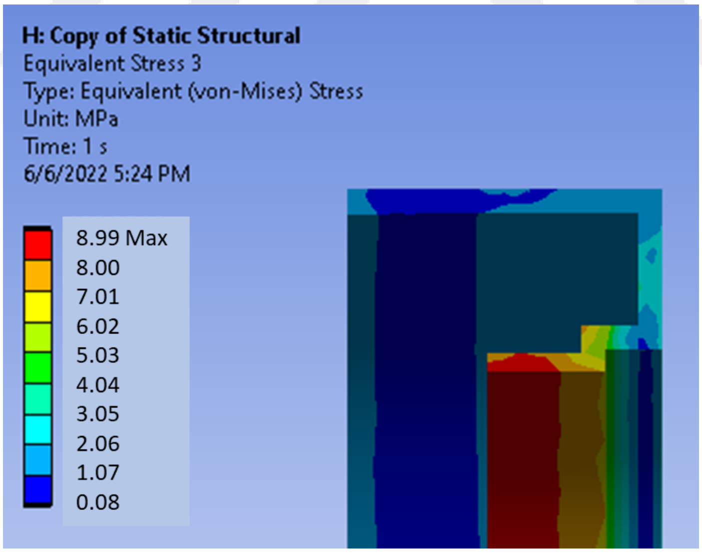


Figure 3.35 Non-Dimensional Stress Result for Carbon Ring Case

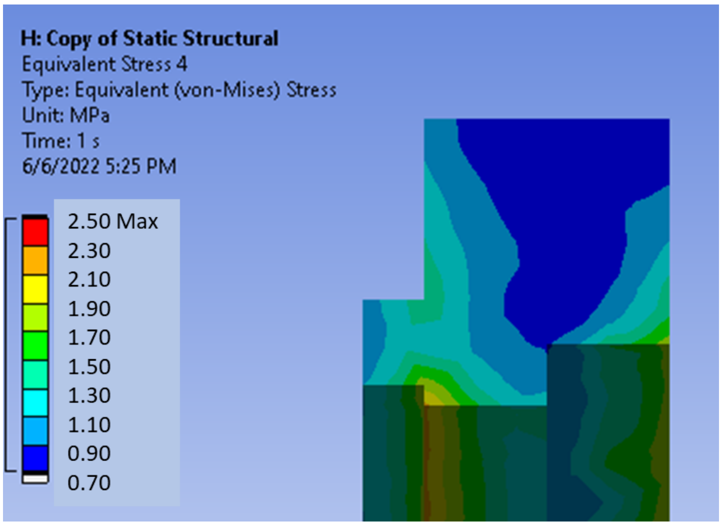


Figure 3.36 Non-Dimensional Stress Result for Carbon Ring

3.3.5 Wear Control

Finally, one of the biggest problems, which is the wear condition, was evaluated. The wear types can be examined under three headings: adhesive, abrasive, and chemical. Adhesive wear occurs as a result of the friction between two surfaces and the removal of small particles from the relevant surface. There is only one solution to this problem is to have fluid between the surfaces and never let them touch each other. This solution is impossible even with non-contact surface seals because they have to work in contact without at some time points in the working envelopes. The wear rate equation [22] can be defined as follows;

$$\Delta Q = \frac{KW_m \Delta S}{H} \tag{3.8}$$

$$\frac{\Delta w}{\Delta t} = \dot{w} = K\frac{P_m U}{H} \tag{3.9}$$

According to the equation shown above, the expected wear amount of the designed mechanical face seal after 1 hour of operation is approximately $\mathbf{7} \times \mathbf{10^{-7}}$ mm. It means carbon seal has to be replaced within $\mathbf{5000}$ cycle. Considering the whole life of a typical turboshaft engine, these calculated values show that the carbon face will not experience critical wear throughout its entire life.

In order to design a mechanical face seal that works well, it is necessary to know well in which situations and factors the mechanical face seal will fail and what consequences it will cause. However, knowing them is not very easy and detectable, contrary to what is believed. Aviation engine companies perform root cause analysis for every failure. As a result of these studies, the reasons for the failures in the mechanical face seals can be listed as follows;

- Environmental factors can be defined as all the effects that may occur naturally. It is one of the most common failure root cause. Therefore, the working conditions and capability of the seal should be calculated and decided very well.
- Design and manufacture are all the details that the designer decides for the final product, mechanical face seal. For example, a nonconformity that may occur as a result of manufacturing tolerances or poorly defined drawing.
- Operation and installation can be defined as an assembly-related error or poorly defined limit condition throughout the engine mission. The working envelope is created according to the environmental condition and power requirement of the engine. The mechanical face seal shall also be suitable and sustainable for the rare upper limit conditions (ultimate condition) of this envelope. The seal may fail even in the first operation due to nonconformity that may occur during assembly.

4. RESULTS

The main concept of the carbon seal has been configured according to design requirements. Force balance on carbon seal has been adjusted by selecting the spring type. Material selection has been done according to carbon seal operating conditions and it has been shown that the material temperature is within material serviceable temperature by performed thermal analysis. By using thermal analysis output and operating load, structural analysis has been made and all parts maintain structural integrity under predicted condition. Lastly, the wear characteristic of the carbon seal has been checked and wear amount would not exceed the critical wear levels throughout the life of the part. The carbon seal design will be fitted to the same design space of the existing labyrinth seal. Carbon seals will be manufactured and tested to observe the leakage performance and wear capability with endurance tests as a future work.

5. CONCLUSION

Preliminary design steps of carbon seal design and details of each step is investigated within this thesis. The design space is selected by considering a turboshaft engine and labyrinth seal of the engine is intended to be replaced with a custom design carbon seal to be used for bearing chamber sealing. The biggest advantage of carbon seal implementation for bearing chamber sealing is reduced pressurization air and prevention of oil leakage.

As a conclusion, it is expected that if designed carbon seal is being used in bearing chamber which currently labyrinth seal is being used in, leakage will be less than 10% of current leakage. Oil cooling should be used to maintain the temperature of related parts within serviceable temperature. Addition to current design runner, carbon seal assembly and oil jet should be implemented on bearing region. Finally, all design steps and results should be validated with test.

It is necessary to start the mechanical seal design by predicting the environmental temperature and pressure values. Having this information helps to choose a better mechanical seal but many times. However, often this information is difficult to find and measure accurately due to inexperience. The best solution is to validate analysis and hand calculations by applying instrumentation during testing. Briefly, all items listed below must be identified and designed.

- Seal Description,
    - Primary rotating or non-rotating,
    - Secondary seal type,
    - Spring / drive / Insert types,
    - Balance ratio,
    - Spring load,
    - Face materials,
    - Face width and mean face diameter,
    - Primary ring stiffness,
    - Primary ring mass,
- Process Fluid,
    - Generic name,
    - Mole fractions of components,
    - Bubble point,

    - Estimated temperature & pressure at seal,
    - Non-steady conditions,
- Seal Environmental Control,
    - Flush & Quench rate and point of entry,
    - Heat exchanger and heat removal rate and / or temperature of inlet,
    - Self-pumping circulation,
- Seal Failure or Life,
    - Number of hours of operation,
    - Reason for removal,
    - Wear on faces,
    - Operation causes for seal failure.

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BIOGRAPHY

İlkan Hasan Akpınar’s professional career started in aviation industry. Currently, he is working as a lead engineer.