STRESS ANALYSIS OF MECHANICAL SEAL IN API 610 PUMP

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2021

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DISSERTATION SUBMITTED IN FULFILMENTOF THE REQUIREMENTS FOR THE MASTER OF MECHANICAL ENGINEERING

**FACULTY OF MECHANICAL ENGINEERING
UNIVERSITY OF MALAYA
KUALA LUMPUR**

2021

UNIVERSITY OF MALAYA

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STRESS ANALYSIS OF MECHANICAL SEAL IN PUMP (API610)

ABSTRACT

This research project uses Ansys Finite Element Analysis software to conduct Stress Analysis of Mechanical Seal used in API 610 pump. Maximum Sealing Pressure will be applied as per API682 requirement and maximum and minimum stress will be monitored. The finite element method has become a powerful tool for the numerical solutions of a wide range of engineering problems. Besides equivalent stress, strain and deformation of the mechanical seals will also be analyzed. The modeling of the mechanical seal was done by Ansys Design Modeler and the analysis was performed by using Ansys workbench 2021 R1 version.

In the model, the mechanical seal face parts are going to be in contact and the remaining parts like ring and steel support not combined.

At the end of research, we will be able to select the best candidate for the critical part of mechanical seal in API 610, which is the sleeve part.

Keywords: Mechanical Seal, FEA, ANYSY, API610

ANALISA TEKANAN TERHADAP PENGEDAP MEKANIKAL DI DALAM PAM (API610)

ABSTRAK

Projek Penyelidikan ini menggunakan perisian Ansys Finite Element Analysis untuk mengendalikan Analisa Tekanan kepada Pengedap Mekanikal yang digunakan di dalam Pam API610. Tekanan maksima pengedapan akan diaplikasikan menurut keperluan standard API682 dan maksimum dan minimum stress akan diperhatikan. Kaedah Element Terhad telah menjadi kaedah yang mapan sebagai solusi kepada banyak cabang masalah kejuruteraan.

Selain tekanan kumulatif, tegangan kumulatif and pengubahan kepada bentuk Pengedap Mekanikal dibuat menggunakan Ansys Design Modeler dan Analisa telah dijalankan menggunakan Ansys workbench versi 2021 R1.

Di dalam model, permukaan-permukaan pengedap mekanikal akan dicantumkan manakala komponen-komponen lain seperti cincin dan besi sokongan adalah berasingan.

I akhir kajian, kita dapat menemukan calon material terbaik untuk pengedap mekanial bagi pam API610, iaitu bahagian pelindung.

Keywords: Pengedap Mekanikal, FEA, ANYSY, API610

ACKNOWLEDGEMENTS

I would like to express my gratitude to my research supervisor (Assoc. Prof. Ir. Dr. Wong Yew Hoong) from the Department of Mechanical Engineering of University Malaya. He has allowed this paper to be my own work and always so responsive.

I would also like to acknowledge all of other professors who have contributed in teaching of all valuable subjects of Mechanical Engineering during my journey in Master of Mechanical Engineering in University of Malaya. I am gratefully indebted of the valuable lessons and commitment of all professors in making sure that all the students learn something, even though have to conduct classes and exam during the COVID-19 pandemic.

Finally, I must express my very profound gratitude to my husband, parents and family members for providing me with unfailing supports and encouragement throughout my years of study. This accomplishment would not have been possible without them.

Thank you

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LIST OF SYMBOLS AND ABBREVIATIONS

API	:	American Petroleum Institute
ANSYS	:	Analysis System
FEA	:	Finite Element Analysis
MPa	:	Mega Pascal
MAWP	:	Maximum Allowable Working Pressure

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INTRODUCTION

1.1 Background of Study

A mechanical seal is one of the important parts of a pump which directly affects the performance of the pump unit. Consistent operation of a mechanical seal is vital as this clearly affects the flowrate of the pump.

Unfortunately, failure of mechanical seals is the most common type of pump downtime. As claimed by a research, by Grundfos Industry (Pump Handbook, 2004), mechanical seals account for 39% of pump failures [(1)]. Other failures of pumps are contributed by Roller Bearing (15%), Operation (5%) and miscellaneous.

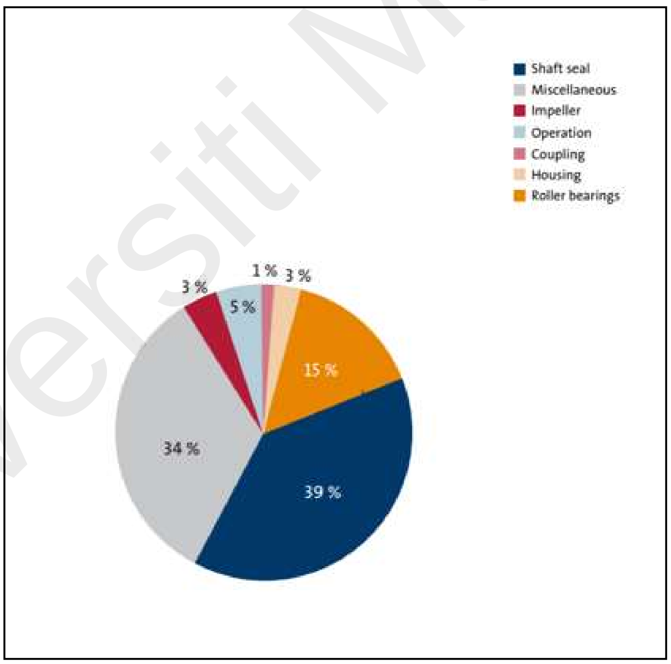


Fig 1 *Common causes of pumps failure*

Because of the requirement of compliance of the stringent requirement for hydrocarbon emission, Oil & gas industry are evaluating possible alternative to cut the releases of fugitive emissions.

Almost 70% of centrifugal pump maintenance is due to mechanical seal failures, which is a primary contributor of fugitive emissions. The new standard-API 682 was developed to set guidelines that determine mechanical seal performance and specifications. API-682 describes centrifugal seal-sealing system performance and design criteria that will improve reliability and increase pump-seal life. Axial and radial force must be investigated in the design process of mechanical seal.

1.2 Problem Statement

Mechanical seal is designed for all pumps, especially heavy-duty pumps such as API 610 pumps. The function is as a mechanism of sealing to prevent leakage under intense shaft speed, extreme pressure, and temperature condition. The life cycle of the pump is short contributed from the seal material failure. The material failure is mainly impacted form excessive stress during operation under the extreme condition, especially for API610 pumps.

1.3 Objectives and Scope of Work

The main objectives of this research are:

1. To observe on which mechanical seal component affected by stress (critical part) when applied API610 pumps MAWP.
2. To conduct Finite Element Analysis for mechanical seal for different materials.
3. To observe minimum and maximum stress and select best material candidate for the critical component of API610 pumps Mechanical Seal.

The scope of work for this research is to monitor the simulation of the stress profile by using ANSYS software on the mechanical seal. At the same time, theoretical calculation will be done to observe on the trend of the stress distribution, and then the trend will be compared with ANSYS simulation result. After the analysis, results between different materials will be observed and which material is the most suitable will be recommended based on API682 standards for mechanical seals.

LITERATURE REVIEW

2.1 Basic Design and Mechanism

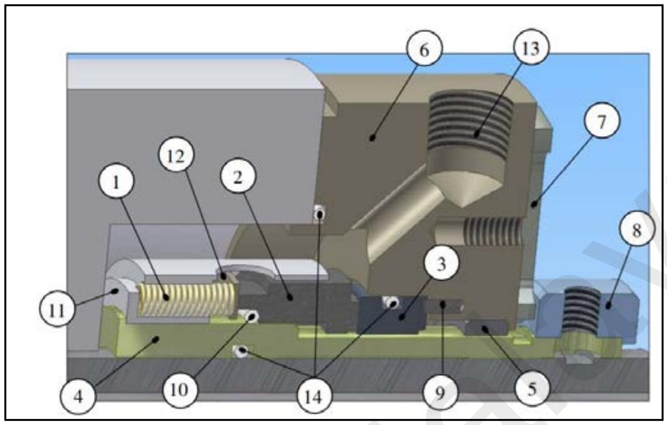


Fig. 2 *Basic Components of Mechanical Seal*

The typical components and their primary function of mechanical seal are: -

1. Spring Component: This item presses the seal faces all together, giving the appropriate amount of close force to suppress fluid under static conditions, while permitting the attached face differing degrees of freedom. In this way, the seal interface can suppress the fluid, despite certain level of wear, misalignment, run-out, distortion, vibration and thermal expansion.
2. Rotating Face: The shape of this ring has been designed to adapt to the different forces present during the operation. This component allows the seal to operate parallel to the stationary face for appropriate sealing performance, despite pressure and thermal affects. The ring usually made from carbon graphite.
3. Stationary Seal Face: This ring rests within the stationary housing which is connected to the shaft housing.

4. Shaft Sleeve: This item is intended to slip over the shaft, to allow the entire seal assembly to be positioned instantly as a container. The sleeve offers a controlled surface over the dynamic portion of the mechanical seal can slide and protects the shaft from wear.
5. Secondary Containment Bushing: This item is to improve the containment of fluid if failure occurs. It does not provide full containment not unlike the sealing interface, but it can stop the flow.
6. Seal Gland: This item bolts up to the shaft housing to help remove particulate from the seal interface. It acts as a secondary housing to take the stationary sealing ring, the secondary containment bushing, and link components together as a cartridge.
7. Setting Clips: These clips are used to align the sealing components as a cartridge before installation and protect the seal during delivery.
8. Drive Collar: It is the coupling between the shaft and the sleeve that causes the rotating parts to move together with the shaft.
9. Drive Pins: It is a pin used to prevent the face from rotating together in relative motion with other components that hold the face.
10. Dynamic gaskets: Gaskets are used where relative motion between components is big.
11. Spring Holder: It is to hold the rotating components together. It has drive pins to engage the rotating face and allow the face to move axially to maintain the appropriate contact during operation.
12. Backing Ring: It distributes the spring load evenly across the back of the rotating face to eliminate any localized spring distortion on the face.
13. Flush Port: It allows control of the fluid environment of the seal by providing a method for positive displacement to remove particulate and vapor blocked above the sealing interface and for temperature and pressure control

14. Static gaskets: They are used in locations where relative motion between components is not significant.

These components are designed to help beat the sealing environment, letting the seal interface to run as close together and parallel as possible, while retaining fluid within the interface. This is for mechanical seal to maintain appropriate sealing contact even though being exposed to environment where vibration, miss-alignment, expansion, and wear are present.

2.2 Seal Operation and Sealing Components

A mechanical seal is a method of containing fluid within pumps where a rotating shaft allows through a stationary housing where the housing revolves around the shaft.

When sealing a pump, the challenge is to permit a rotating shaft to go in the 'wet' area of the pump, without allowing huge quantities of pressurized fluid to leak. A typical mechanical seal comprises of a rotary seal face with a driving system which turns at the same speed as the pump shaft. The primary seal is achieved by two very flat faces, which make it particularly difficult for the fluid to escape between them. All mechanical seals are made with three elementary set of parts (J. Edward Pope, 1997) [(4)]. The first and most important set is the mechanical seal faces, as shown in Figure 1. The rotating seal face is mounted on the rotating shaft, while the stationary seal face, is attached to the housing via the gland ring. The faces are pressed against each other by a combination of hydraulic from the fluid and spring force of the seal plan.

In this way, a sealing is maintained to prevent the fluid from leaking between the rotating and stationary areas of the pump. The second set contains of the secondary sealing members. These members consist of a wedge ring located under the rotor, an O-ring located on the stator, and the gland ring gasket. The third set is the seal hardware, consisting of the spring retainer, springs, set screw and gland ring. The spring retainer is used to mechanically drive the rotating seal face, as well as house the springs. The springs are a vital component for assuring that the seal faces remain in contact during any axial movement from normal seal face wear, or face misalignment. The set screw is used for transmitting the torque from the shaft.

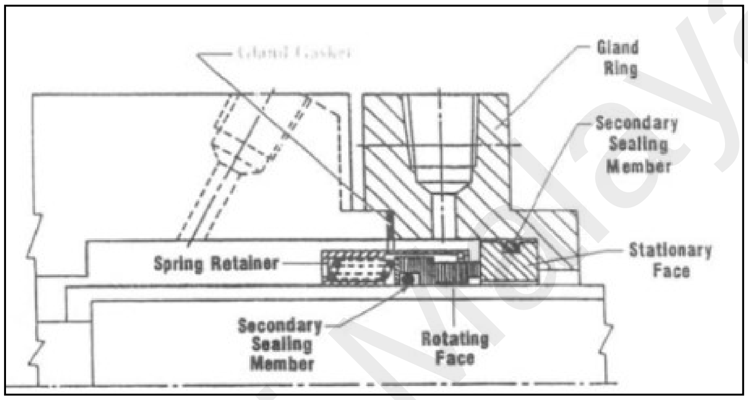


Fig. 3 *Sealing Component of Mechanical Seal (Courtesy of J. Edward Pope)*

The primary seal is essentially a spring-loaded vertical bearing - consisting of two extremely flat faces, one fixed, one rotating, running against each other. The seal faces are pushed together using a combination of hydraulic force from the sealed fluid and spring force from the seal design. In this way a seal is formed to prevent process leaking between the rotating (shaft) and stationary areas of the pump.

If the seal faces rotated against each other without some form of lubrication, they would wear and quickly fail due to face friction and heat generation. For this reason, some form of lubrication is required between the rotary and stationary seal face; this is known as the fluid film. There are four main sealing points in a mechanical seal, as illustrated in Figure 4.0

The seal faces are the primary sealing point (Point A). This point is achieved by pushing against each other two very flat, lapped surfaces, perpendicular to the shaft, that creates a very treacherous leakage path. Leakage is also minimized by the rubbing or sliding contact between the rotating and stationary faces.

The second leakage point, Point B, is under the rotating seal face along the shaft. This point is blocked by a secondary O-ring. At Point C an additional secondary is used to prevent leakage between the gland ring and the stationary seal face. Point D is the gland ring gasket which prevents leakage between the equipment case and the gland (J. Edward Pope, 1997).

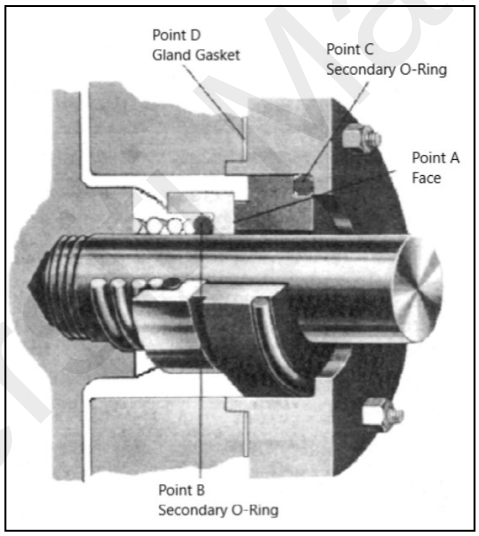


Fig. 4 *Sealing Faces of Mechanical Seal (Courtesy of J. Edward Pope)*

2.3 Operation Conditions and Theory.

In this analysis, pressure that is applied is 4.0 MPa, according to API610 maximum sealing pressure. Some assumptions and simplifications are considered. The sealed fluid enters between the faces and distributes itself in a way such as the average value of the fluid pressure between the faces is proportionate to the sealed pressure, K_p. This fluid pressure must support some of the applied load. The spring force ensures static equilibrium in the axial direction because of the hydrodynamic pressure or contact pressure in between the faces.

Summarizing all the forces in the axial direction,

$$P\pi(r_o^2 - r_b^2) + F_2 = KP\pi(r_o^2 - r_i^2) + P_m\pi(r_o^2 - r_i^2) \quad (6)$$

The mean pressure can be calculated by this equation

$$P_m = P(B - K) + \frac{F_2}{\pi(r_o^2 - r_i^2)} = P(B - K) + P_2 \quad (7)$$

The value of *K* significantly affects the contact pressure, and it is called the *K* factor or the pressure gradient factor. If the fluid movement affected by the hydrostatic pressure throughout the face is laminar and incompressible, the value for *K* is presumed to be ½ and if it is a compressible flow, then K is 2/3.

2.4 Boundary Condition

In typical case of mechanical seal face, the Reynolds equation involves a solution for the whole area of contacting faces. The boundary condition is shown by the arrow in Figure 5.0.

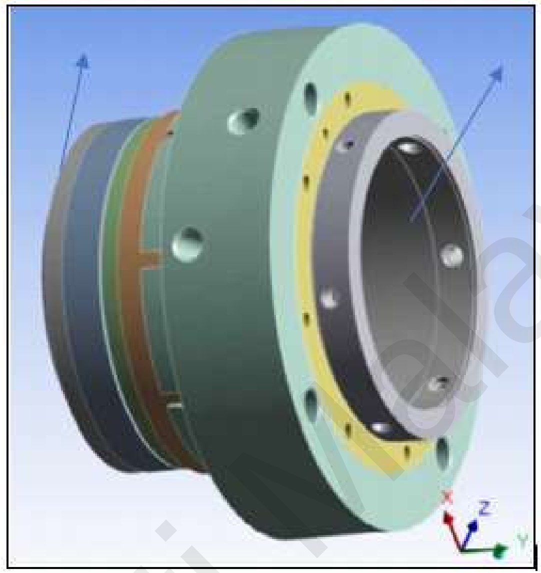
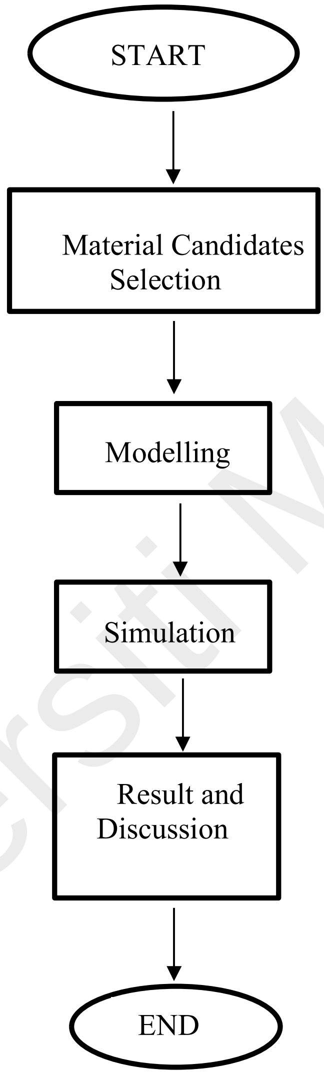


Fig. 5 *Boundary Condition*

METHODOLOGY

3.1 Project Flow



3.2 Material Candidates Selection

In order to identify material candidates, we will refer to the Ashby charts of materials. For mechanical seals, two main properties to determine good material candidates are:

1. High Yield Strength
2. High Elasticity (Young modulus)

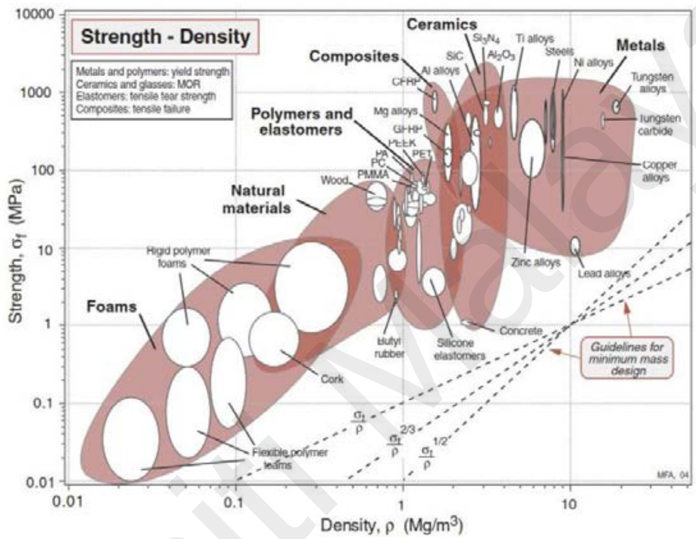


Fig. 6 *Ashby chart: Yield Strength plotted against density.*

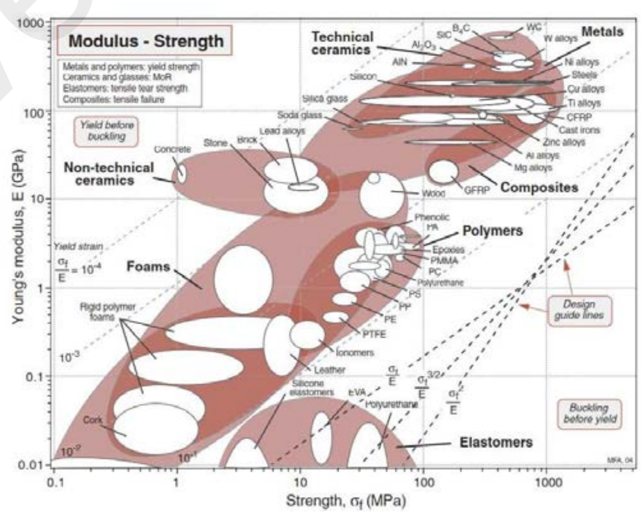


Fig. 7 *Ashby chart: Young's modulus plotted against strength*

Based on this, and after counterchecking the API 682 standard, we have selected four (4) material candidates.

Table 1: Material candidates Yield Strength

MATERIAL CANDIDATES	YIELD STRENGTH
Carbon Graphite	208 MPa
Tungsten Carbide	344.8 MPa
Stainless Steel	207 MPa
Silicon Carbide	3440 MPa

Table 2: Material candidates Young Modulus

MATERIAL CANDIDATES	YOUNG MODULUS
Carbon Graphite	21 GPa
Tungsten Carbide	620 GPa
Stainless Steel	193 GPa
Silicon Carbide	414 GPa

3.3 Modeling

3.3.1 Geometry

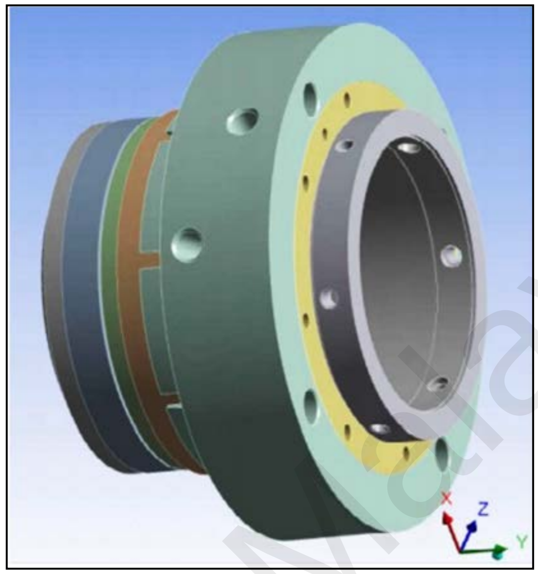


Fig 8 *Assembly view of mechanical seal*

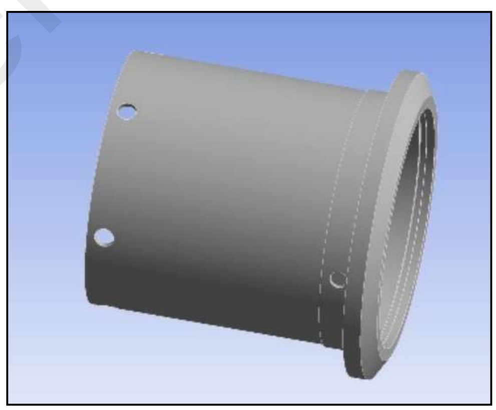


Fig 9 *Mechanical seal model- sleeve part*

3.3.2 Assumption used in analysis

1. Fluid is Newtonian
2. Fluid is laminar and not turbulent
3. Density is constant throughout the fluid
4. Viscosity is constant throughout the fluid
5. Fluid inertia effect is negligible
6. The impact of roughness on fluid flow is negligible
7. The film is thin as velocity gradient across the film predominate
8. The impact of micro asperities as they create pressure on themselves is negligible
9. Temperature is constant during the course of this analysis
10. Sealing is zero leakage
11. Characteristic of seal ring and seal medium do not change with temperature.

For the numerical analysis, the stress calculation is concluded based on the theory of thick-walled cylinder. This is because of the shape of mechanical seal is cylinder and the radius of mechanical seal is more than 1/20 of its thickness. Figure 10 shows a microscopic element of unit thickness which classifies two radii parameter, r and r + r and angle Θ. The typical radial acting on the microscopic element at distance r will be a σ_r while for variable stress will be $\sigma_{r\,+}\,{}^{dor}/_{dr.}$ The results from this derivation are:

For internal pressure case ($P_i \neq 0$) & ($P_o = 0$)

For radial: $$\sigma_r = \frac{p_i\, r_i^2}{r_0^2 - r_i^2}\left(1 - \frac{r_0^2}{r^2}\right) \qquad (1)$$

For tangential: $$\sigma_r = \frac{p_i\, r_i^2}{r_0^2 - r_i^2}\left(1 + \frac{r_0^2}{r^2}\right) \qquad (2)$$

The detail of derivation is as per attached in appendix I

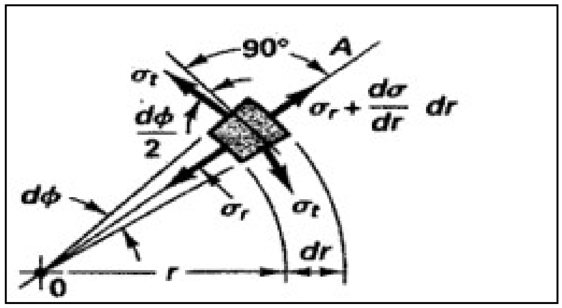


Fig 10 *Element in mechanical seal*

3.4 Simulation

The objective of the simulation is to analyze the stress distribution given to the seal. The load given will be as per maximum sealing pressure for API 610 pumps according to the standard, which is 4.0 MPa in tangential direction. Manual calculation is also conducted using theoretical formulations.

3.4.1 Material Selection & Engineering Properties

The simulation will be using four (4) material candidates with different mechanical and physical properties as per stated in Table 1.0. The simulation result using different type of material will be compared and the most suitable material will be recommended.

Properties of Outline Row 3: Carbon Graphite

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	1720	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
5	Coefficient of Thermal Expansion	1.2E-05	C^-1
6	Isotropic Elasticity		
7	Derive from	Young...	
8	Young's Modulus	21	GPa
9	Poisson's Ratio	0.31	
10	Bulk Modulus	1.8421E+10	Pa
11	Shear Modulus	8.0153E+09	Pa
12	Strain-Life Parameters		
13	Display Curve Type	Strain-...	
14	Strength Coefficient	9.2E+08	Pa
15	Strength Exponent	-0.106	
16	Ductility Coefficient	0.213	
17	Ductility Exponent	-0.47	
18	Cyclic Strength Coefficient	1E+09	Pa
19	Cyclic Strain Hardening Exponent	0.2	
20	S-N Curve	Tabular	
24	Compressive Yield Strength	208	MPa
25	Tensile Ultimate Strength	3.5	GPa

Table 3 Material and Physical Properties input for Carbon Graphite

Properties of Outline Row 7: Tungsten Carbide

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	15800	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
5	Coefficient of Thermal Expansion	1.2E-05	C^-1
6	Isotropic Elasticity		
7	Derive from	Young...	
8	Young's Modulus	620	GPa
9	Poisson's Ratio	0.24	
10	Bulk Modulus	3.9744E+11	Pa
11	Shear Modulus	2.5E+11	Pa
12	Strain-Life Parameters		
13	Display Curve Type	Strain-...	
14	Strength Coefficient	9.2E+08	Pa
15	Strength Exponent	-0.106	
16	Ductility Coefficient	0.213	
17	Ductility Exponent	-0.47	
18	Cyclic Strength Coefficient	1E+09	Pa
19	Cyclic Strain Hardening Exponent	0.2	
20	S-N Curve	Tabular	
24	Tensile Yield Strength	344.8	MPa
25	Compressive Yield Strength	4483	MPa
26	Tensile Ultimate Strength	1.52	GPa
27	Compressive Ultimate Strength	544.6	GPa

Table 4 Material and Physical Properties input for Tungsten Carbide

Properties of Outline Row 5: Stainless Steel

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	7750	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
5	Coefficient of Thermal Expansion	1.2E-05	C^-1
6	Isotropic Elasticity		
7	Derive from	Young...	
8	Young's Modulus	193	GPa
9	Poisson's Ratio	0.31	
10	Bulk Modulus	1.693E+11	Pa
11	Shear Modulus	7.3664E+10	Pa
12	Strain-Life Parameters		
13	Display Curve Type	Strain-...	
14	Strength Coefficient	9.2E+08	Pa
15	Strength Exponent	-0.106	
16	Ductility Coefficient	0.213	
17	Ductility Exponent	-0.47	
18	Cyclic Strength Coefficient	1E+09	Pa
19	Cyclic Strain Hardening Exponent	0.2	
20	S-N Curve	Tabular	
24	Tensile Yield Strength	207	MPa
25	Compressive Yield Strength	207	MPa
26	Tensile Ultimate Strength	0.58	GPa

Table 5 Material and Physical Properties input for Stainless Steel

Properties of Outline Row 4: Silicon Carbide

	A	B	C
1	Property	Value	Unit
3	Density	3210	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
5	Coefficient of Thermal Expansion	1.2E-05	C^-1
6	Isotropic Elasticity		
7	Derive from	Young...	
8	Young's Modulus	414	GPa
9	Poisson's Ratio	0.19	
10	Bulk Modulus	2.2258E+11	Pa
11	Shear Modulus	1.7395E+11	Pa
12	Strain-Life Parameters		
13	Display Curve Type	Strain...	
14	Strength Coefficient	9.2E+08	Pa
15	Strength Exponent	-0.106	
16	Ductility Coefficient	0.213	
17	Ductility Exponent	-0.47	
18	Cyclic Strength Coefficient	1E+09	Pa
19	Cyclic Strain Hardening Exponent	0.2	
20	S-N Curve	Tabular	
24	Tensile Yield Strength	3440	MPa
25	Compressive Yield Strength	462	MPa
26	Tensile Ultimate Strength	21	GPa
27	Compressive Ultimate Strength	1.37	GPa

Table 6 Material and Physical Properties input for Silicon Carbide

3.4.2 Meshing

In this project, the model is meshed as per specifications in Table 7. The meshing shape will be different in all areas, depending on the minimum and maximum edge of the model.

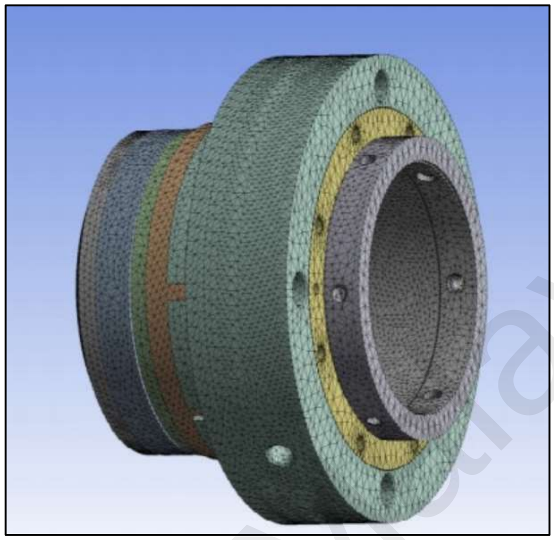


Fig. 11 *Model mesh view*

Table 7 Meshing Details

Details of "Mesh"

Display	
Display Style	Use Geometry Setting
Defaults	
Physics Preference	Mechanical
Element Order	Program Controlled
Element Size	5.e-003 m
Sizing	
Quality	
Check Mesh Qua...	Yes, Errors
Error Limits	Aggressive Mechanical
Target Quality	Default (0.050000)
Smoothing	Medium
Mesh Metric	Skewness
Min	1.1691e-003
Max	0.99986
Average	0.43246
Standard Devi...	0.21743
Inflation	
Advanced	
Statistics	
Nodes	388959
Elements	211742

3.4.3 Simulation

All the properties will be examined, and pressure will be applied to the model. The deformation will be observed and critical parts prone to deformation will be noted. Then the result of Total Deformation, Equivalent Stress and Equivalent Strain will be monitored.

RESULT AND DISCUSSION

4.1 Numerical Calculation

For numerical calculation analysis, equation (1) and (2) are used.

This is because manual calculation can only be show for one point, so we have to distribute the stress based on the distance from the center point, and we take note on the maximum and minimum value. Then we will compare with the maximum and minimum value from ANSYS simulation result.

For radial:
$$\sigma_r = \frac{p_i r_i^2}{r_0^2 - r_i^2}\left(1 - \frac{r_0^2}{r^2}\right) \quad (1)$$

For tangential:
$$\sigma_r = \frac{p_i r_i^2}{r_0^2 - r_i^2}\left(1 + \frac{r_0^2}{r^2}\right) \quad (2)$$

The stress is distributed on radial and tangential direction using these formulas.

4.1.1 Radial Distribution Stress

Table 8 shows the date used to calculate the stress distribution in the mechanical seal for the radial direction. The initial pressure used is 2.21 MPa (according to datasheet as reference), with the constant inner and outer radius. The numerical result shows range from 2.20 MPa to 58Pa. From Figure 16.0, the radial stress distribution in the mechanical seal is inversely proportional to their distance. When the distance increases, the stress distribution decrease to zero. The graph shows that the mechanical seal is having tensile stress on the radial direction. The result is inversely proportional due to the seal is calculated using formula for hollow thick-wall cylinder while ANSYS test result is using the real geometry and dimension of the model.

Table 8 Radial stress distribution throughout mechanical seal

NO	INITIAL PRESSURE Pi (MPa)	INNER RADIUS, ri (mm)	INNER RADIUS, ro (mm)	DISTANCE FROM INNER RADIUS, r (mm)	STRESS (σr) (MPa)
1	-2.21	0.022352	0.060909	0.0224	2.1991
2				0.0244	1.7993
3				0.0264	1.4868
4				0.0284	1.2381
5				0.0304	1.0368
6				0.0324	0.8716
7				0.0344	0.7343
8				0.0364	0.6191
9				0.0384	0.5214
10				0.0404	0.4378
11				0.0424	0.3658
12				0.0444	0.3033
13				0.0464	0.2487
14				0.0484	0.2008
15				0.0504	0.1584
16				0.0524	0.1208
17				0.0544	0.0872
18				0.0564	0.0572
19				0.0584	0.0302
20				0.0604	0.0058

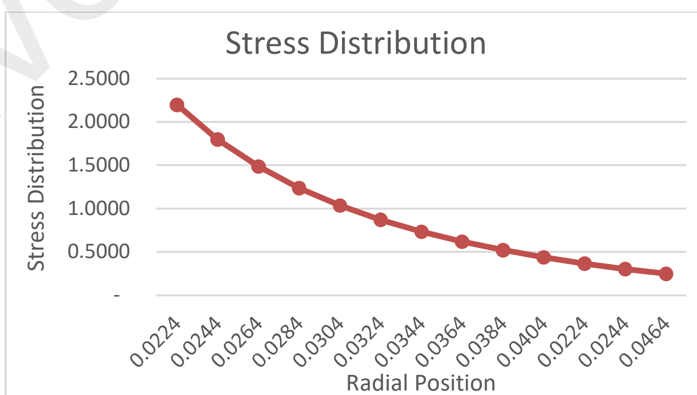


Fig. 12 *Radial stress distribution*

4.1.2 Tangential Distribution Stress

Table 9 shows the data used to calculate stress distribution in the mechanical seal in tangential direction. The initial pressure used is 2.21 MPa (according to datasheet as reference), with constant inner and outer radius. The numerical result shows range from -2.89 MPa to -0.69MPa. From Figure 17.0, the tangential stress distribution in the mechanical seal is inversely proportional to the radial distance of the mechanical seal. As the thickness increase, the stress decreases constantly on radial direction of mechanical seal. This shows that the mechanical seal is having compressive stress on the tangential side.

Table 9 Tangential stress distribution throughout mechanical seal

NO	INITIAL PRESSURE Pi (MPa)	INNER RADIUS, ri (mm)	INNER RADIUS, ro (mm)	DISTANCE FROM INNER RADIUS, r (mm)	STRESS (σr) (MPa)
1	-2.21	0.022352	0.060909	0.0224	(2.8869)
2				0.0244	(2.4871)
3				0.0264	(2.1747)
4				0.0284	(1.9259)
5				0.0304	(1.7246)
6				0.0324	(1.5594)
7				0.0344	(1.4222)
8				0.0364	(1.3070)
9				0.0384	(1.2093)
10				0.0404	(1.1257)
11				0.0424	(1.0537)
12				0.0444	(0.9912)
13				0.0464	(0.9366)
14				0.0484	(0.8886)
15				0.0504	(0.8463)
16				0.0524	(0.8086)
17				0.0544	(0.7751)
18				0.0564	(0.7451)
19				0.0584	(0.7181)
20				0.0604	(0.6937)

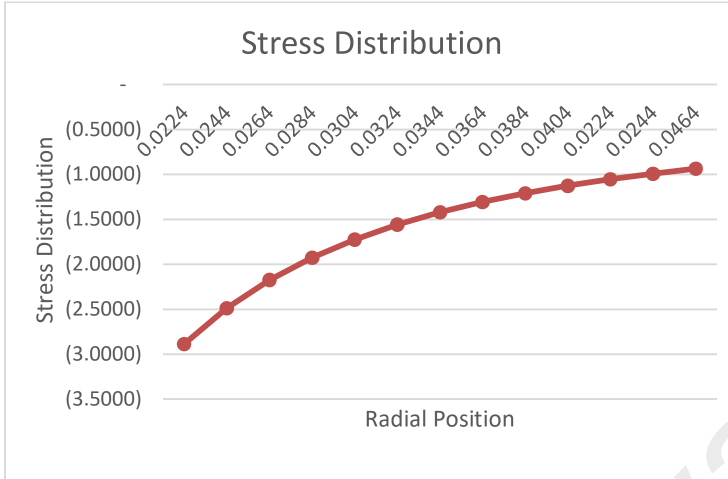


Fig. 13 *Radial stress distribution graph*

4.2 ANSYS Simulation Result

4.2.1 Simulation using Carbon Graphite as material

4.2.1.1 Equivalent Stress

The equivalent von Mises stress profile shown in Figure 14 is the baseline result for this research., where the maximum stress shown is 98MPa, and minimum stress shown is 797.69Pa. The stress mainly takes place on the Sleeve, so the Sleeve stress distribution is being focused on. Based on this distribution, it is aligned with numerical calculation, where both results show the model having tensile stress.

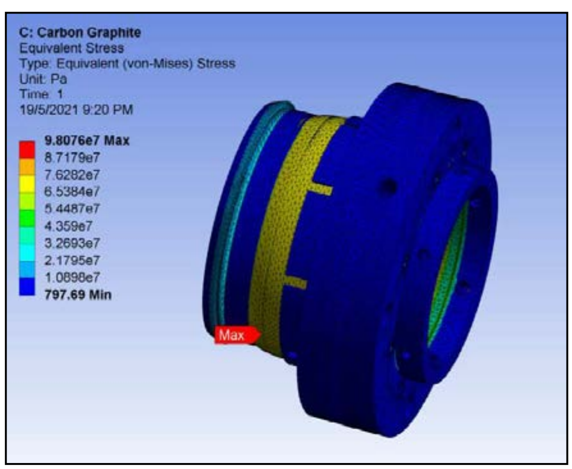


Fig. 14 *Simulation result of Equivalent Stress on Carbon Graphite*

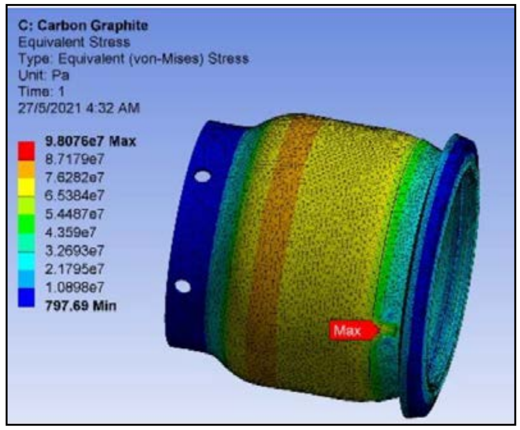


Fig. 15 *Simulation result of Equivalent Stress on critical component of mechanical seal- Sleeve (using Carbon Graphite)*

4.2.1.2 Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in Figure 16 is the baseline result for this research, using Carbon Graphite material. The strain profile is almost like the stress profile in Figure 17 as the strain is proportionally to the stress. The minimum and maximum value if the strain throughout the mechanical seal is 0.000000047m/m and 0.004739m/m respectively. The positive values indicate that the strain is in tensile condition. This is because material elongates in the direction of normal stress.

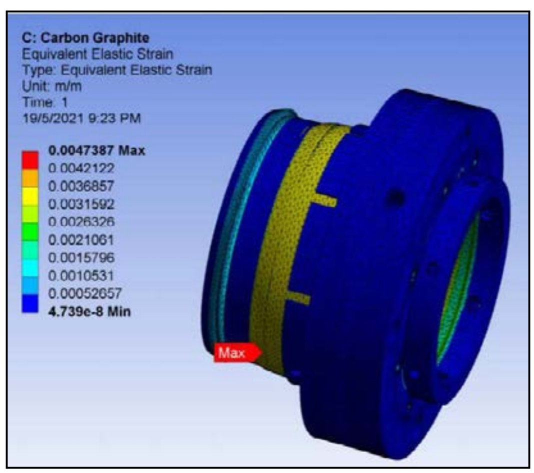


Fig. 16 *Simulation result of Equivalent Elastic Strain on Carbon Graphite*

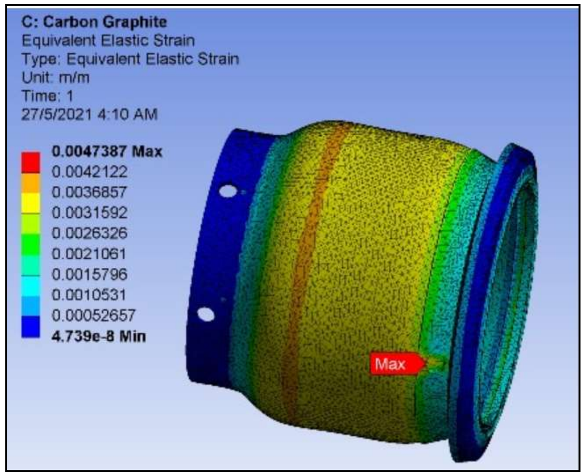


Fig. 17 *Simulation result of Equivalent Elastic Strain on critical component of mechanical seal- Sleeve (using Carbon Graphite)*

4.2.1.3 Total Deformation

The deformation is observed when 4.0 MPa pressure applies to the mechanical seal. The range of deformation on mechanical seal is from 0m to 0.000356m. The deformation is highest at the sleeve as can be seen in Figure 19.

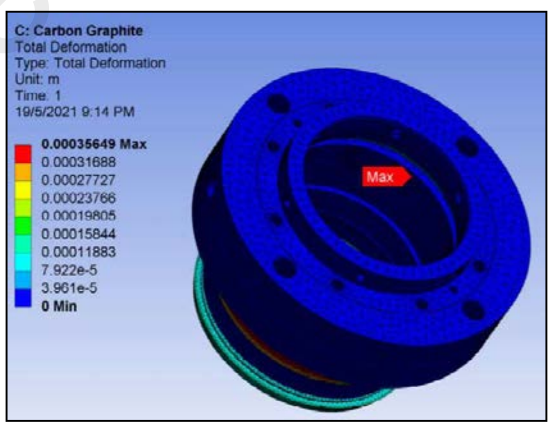


Fig. 18 *Simulation result of Total Deformation on Carbon Graphite*

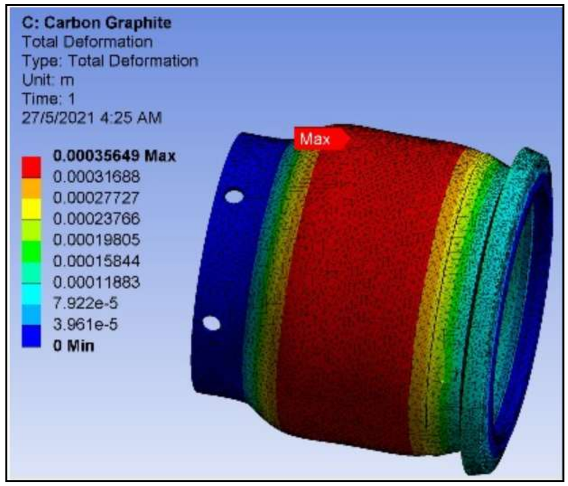


Fig. 19 *Simulation result of Total Deformation on critical component of mechanical seal- Sleeve (using Carbon Graphite)*

4.2.2 Simulation using Tungsten Carbide as material

4.2.2.1 Equivalent Stress

The equivalent von Mises stress profile shown is Figure 20 is the second analysis result for this study, using Tungsten Carbide as material. The minimum and maximum stress value is 1156Pa and 9.865867MPa respectively, all over the mechanical seal.

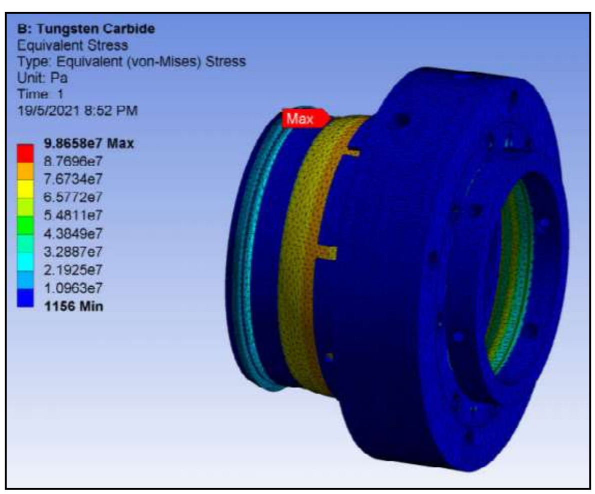


Figure 20 *Simulation result of Equivalent Stress on Tungsten Carbide*

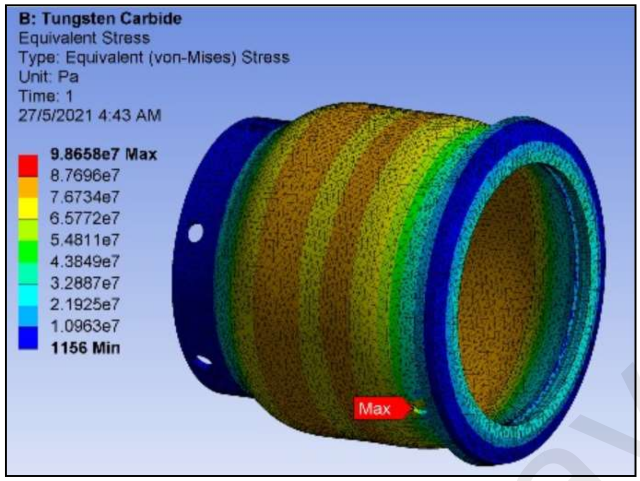


Fig. 21 *Simulation result of Equivalent Stress on critical component of mechanical seal- Sleeve (using Tungsten Carbide)*

4.2.2.2 Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in Figure 22 is the second result for this research, using Tungsten Carbide material. The strain profile is quite similar to the strain through mechanical seal as the strain is proportional to the stress. The minimum and maximum value for the strain is around 0.0000000021m/m and 0.00016137 m/m respectively. The positive value shows the strain is under tensile condition. This is because when the material stretches in the direction of normal stress, contraction in perpendicular in direction occur.

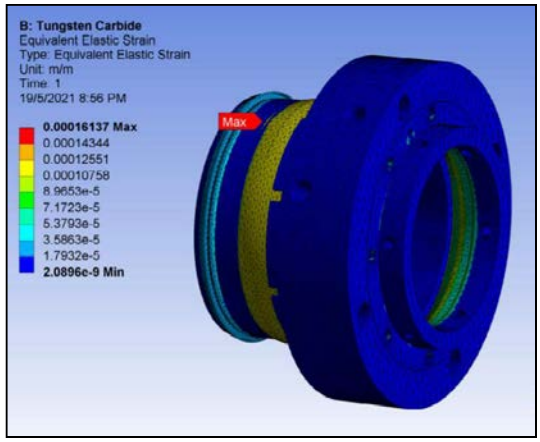


Figure 22 *Simulation result of Equivalent Elastic Strain on Tungsten Carbide*

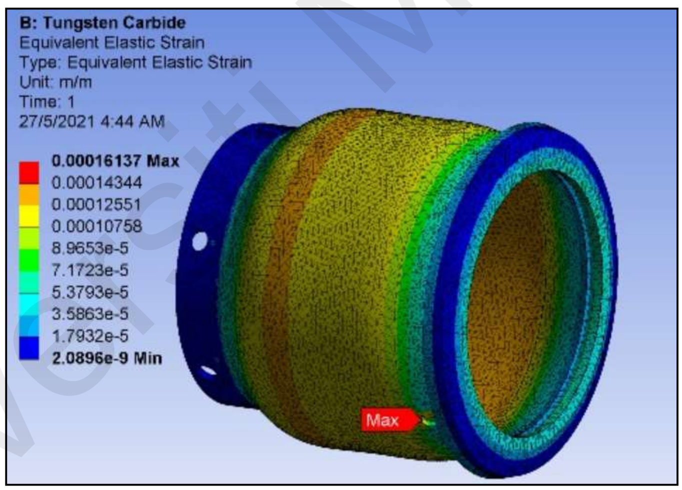


Fig. 23 *Simulation result of Equivalent Stress on critical component of mechanical seal-Sleeve (using Tungsten carbide)*

4.2.2.3 Total Deformation

The result as per Figure 24 is obtained when 4.0MPa pressure is applied to the mechanical seal. The range of the deformation on mechanical seal sleeve is from 0m to 0.000012366 m.

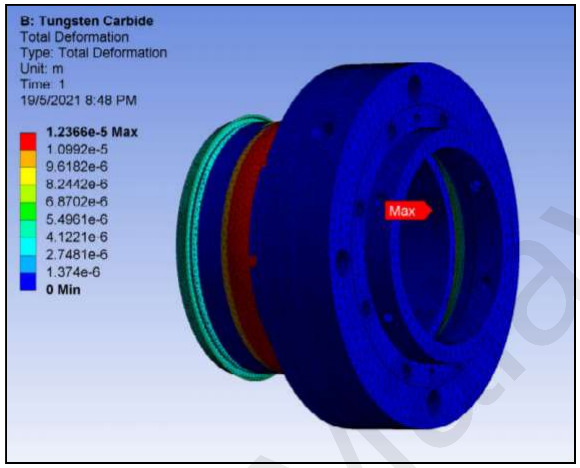


Figure 24 *Simulation result of Total Deformation on Tungsten Carbide*

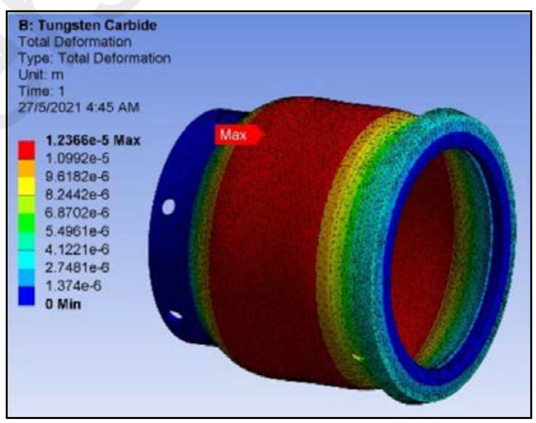


Fig. 25 *Simulation result of Total Deformation on critical component of mechanical seal- Sleeve*

4.2.3 Simulation using Stainless Steel as material

4.2.3.1 Equivalent Stress

The equivalent von Mises stress profile shown in Figure 26 is the third analysis result for this research, using Stainless Steel as the material. The stress mostly takes place on the sleeve. The minimum and maximum values is 802.01Pa and 98.073MPa. The stress is concentrating more on the 0-ring slot due to the end of the sleeve near to the 0-ring slot is fixed. As compared to the numerical calculation, stress distribution is same where both results demonstrate the stress in having tensile stress.

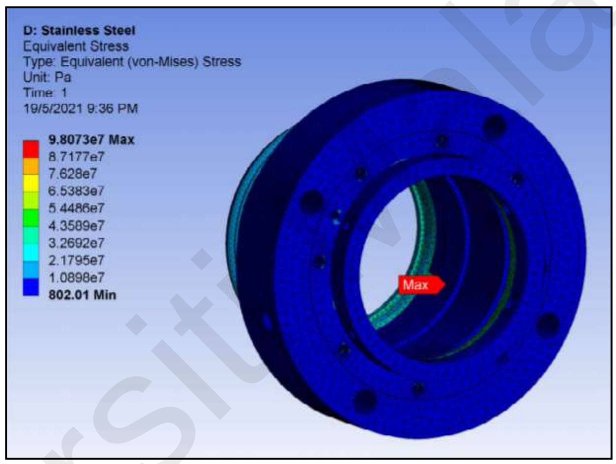


Figure 26 *Simulation result of Equivalent Stress on Stainless Steel*

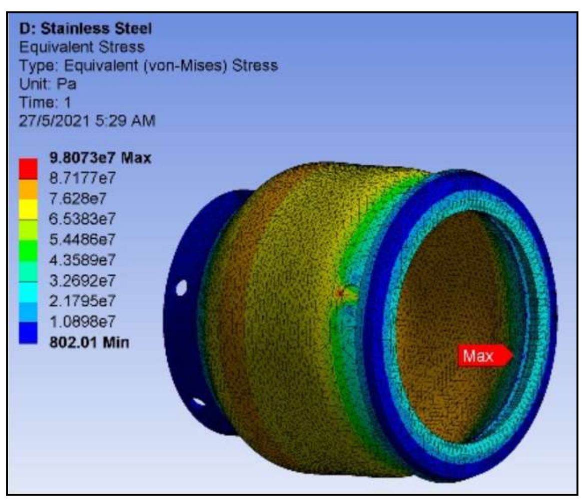


Figure 27 *Simulation result of Equivalent Stress on critical component of mechanical seal- Sleeve (using Stainless Steel)*

4.2.3.2 Equivalent Elastic Strain

The equivalent von Mises elastic strains shown in Figure 28 is the third result, using Stainless Steel as the material. The strain profile is quite like the stress profile in Figure 30.0 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the mechanical seal is around 0.0000000071818m/m to 0.00051561m/m. The positive values show that the strain is under tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

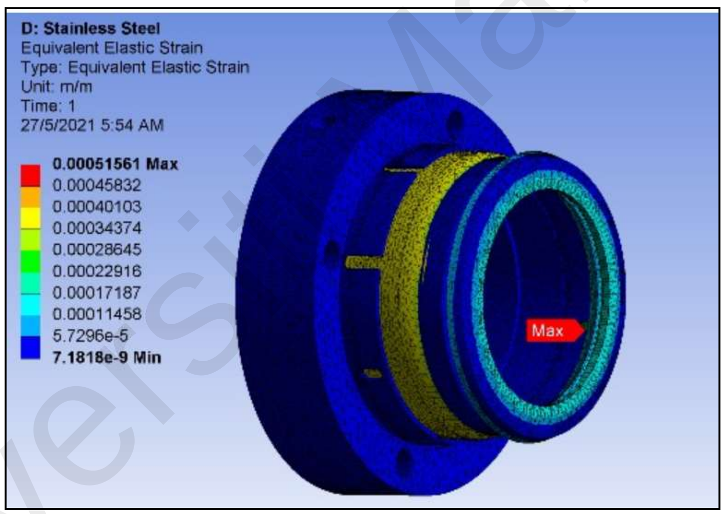


Figure 28 *Simulation result of Elastic Strain on Stainless Steel*

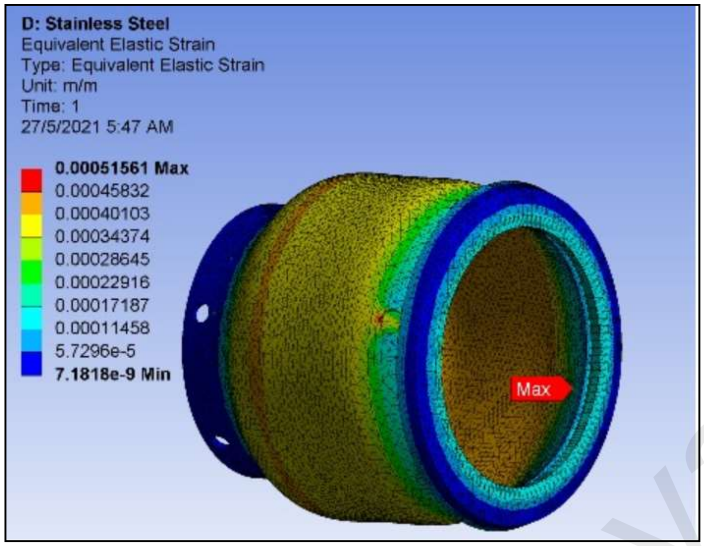


Figure 29 *Simulation result of Elastic Strain on critical component of mechanical seal- Sleeve (using Stainless Steel)*

4.2.3.3 Total Deformation

The result as per Figure 30 is obtained when 4.0MPa pressure is applied to the mechanical seal. The range of the deformation on mechanical seal sleeve is from 0m to 0.000038772 m.

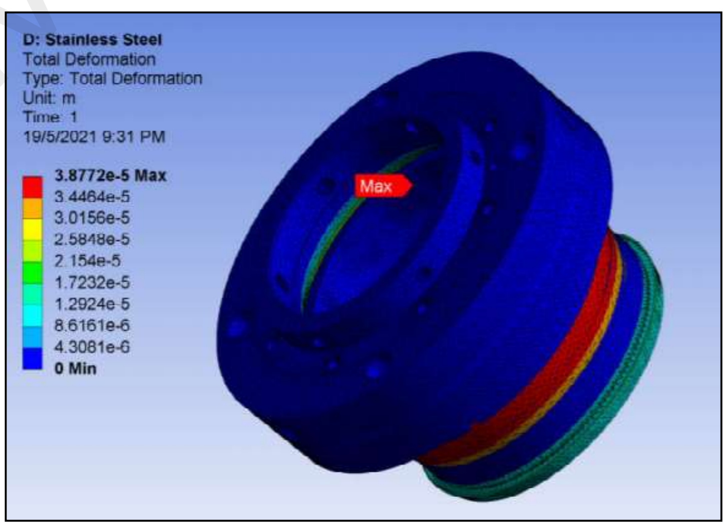


Figure 30 *Simulation result of Total Deformation on Stainless Steel*

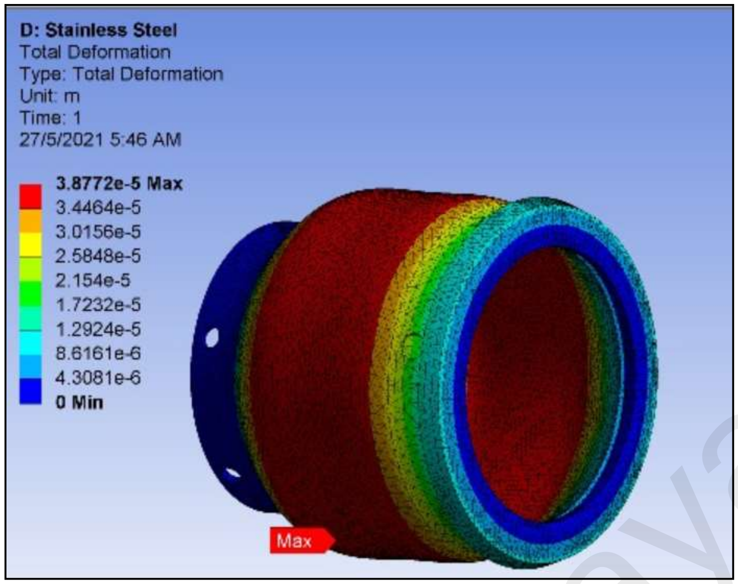


Figure 31 *Simulation result of Total Deformation on critical component of mechanical seal- Sleeve (using Stainless Steel)*

4.2.4 Simulation using Silicon Carbide as material

4.2.4.1 Equivalent Stress

The equivalent von Mises stress profile shown in Figure 32 is the third analysis result for this research, using Stainless Steel as the material. The stress mainly takes place on the sleeve. The minimum and maximum values is 880.03 and 98.073MPa. The stress is more focusing on the 0-ring slot because the end of the sleeve near to the 0-ring slot is fixed. As compared to the numerical calculation, stress distribution is same where both results show the stress in having tensile stress

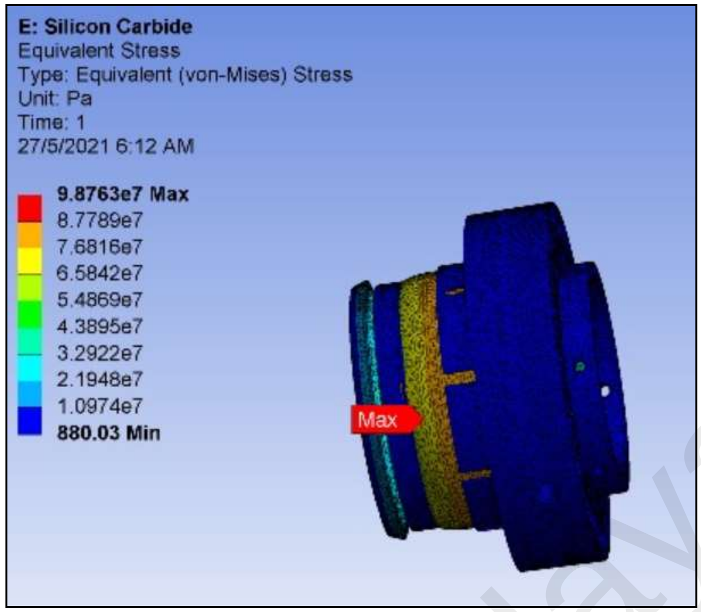


Figure 32 *Simulation result of Equivalent Stress on Silicon Carbide*

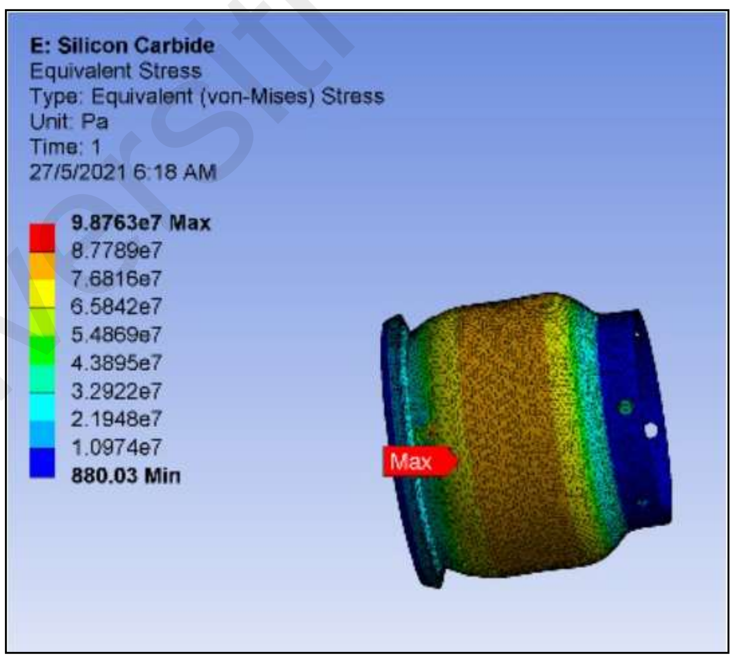


Figure 33 *Simulation result of Equivalent Stress on critical component of mechanical seal- Sleeve (using Silicon Carbide)*

4.2.4.2 Equivalent Elastic Stress

The equivalent von Mises elastic strains shown in Figure 34 is the third result, using Stainless Steel as the material. The strain profile is quite like the stress profile in Figure 35 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the mechanical seal is around 0.0000000071818m/m to 0.00051561m/m. The positive values show that the strain is in tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

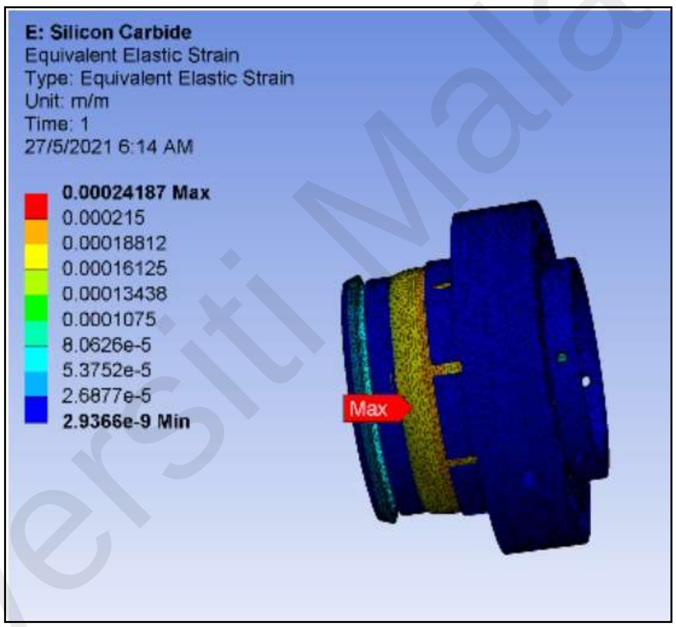


Figure 34 *Simulation result of Elastic Strain on Silicon Carbide*

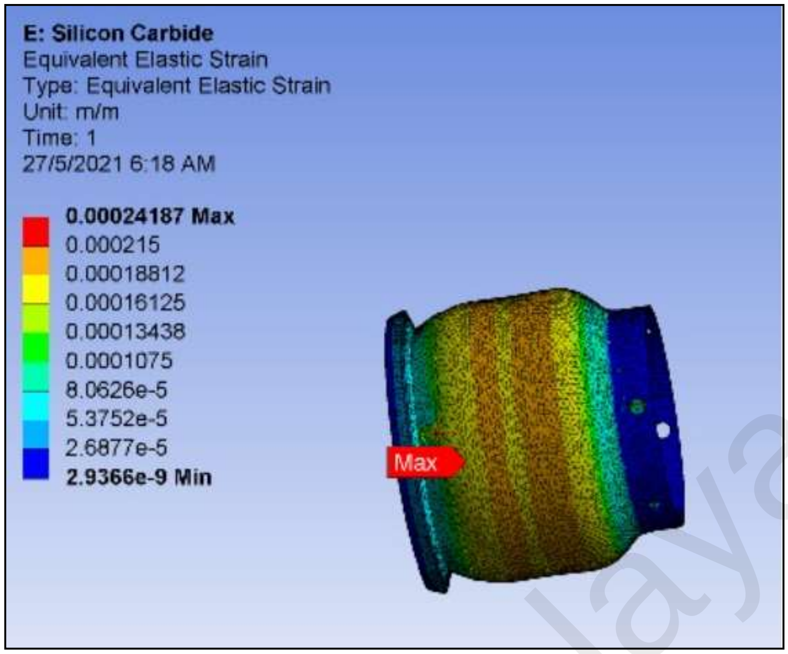


Figure 35 *Simulation result of Equivalent Elastic Strain on critical component of mechanical seal- Sleeve (using Silicon Carbide)*

4.2.4.3 Total Deformation

The result as per Figure 36 is obtained when 4.0MPa pressure is applied to the mechanical seal. The range of the deformation on mechanical seal sleeve is from 0m to 0.000018754 m

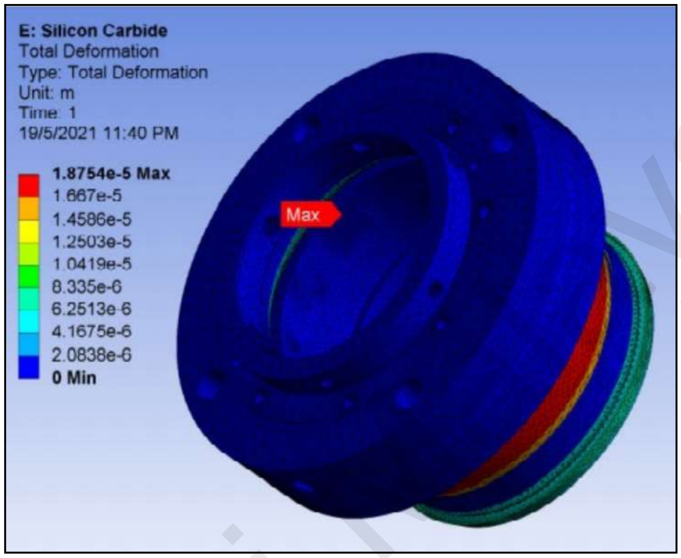


Figure 36 *Simulation result of Total Deformation Silicon Carbide*

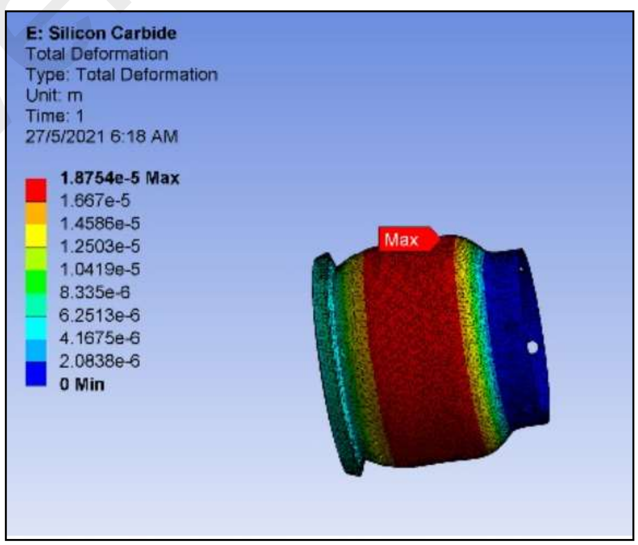


Figure 37 Simulation result of Total Deformation on critical component of mechanical seal- Sleeve (using Silicon Carbide)

4.3 Overall Result

Table 10 shows the comparison value for all the analysis and simulation that has been done. After a few considerations, Silicon Carbide material has produced the most reliable result.

Table 10 Value comparison and ranking based on analysis

ANALYSIS / MATERIAL	MAXIMUM VALUE OF EQUIVALENT VON MISES STRESS, Mpa	RANK	MAXIMUM VALUE OF EQUIVALENT VON MISES ELASTIC STRAIN, m/m	RANK	MAXIMUM VALUE OF TOTAL DEFORMATION, m	RANK	TOTAL RANK
Carbon Graphite	98.000	2	0.000000047	1	0.000356000	2	5
Tungsten Carbide	98.659	4	0.000161370	2	0.000012366	1	7
Stainless Steel	98.073	3	0.000515610	3	0.000038772	3	9
Silicon Carbide	97.63	1	0.00016137	2	0.000012366	1	4

CHAPTER 5: CONCLUSION AND RECCOMENDATION

5.1 Conclusion

As a conclusion, the objectives of the study to observe on which mechanical seal component affected by stress (critical part) when applied API610 pumps MAWP, to conduct Finite Element Analysis for mechanical seal for different materials, and to observe minimum and maximum stress and select best material candidate for the critical component of API610 pumps Mechanical Seal, using FEM (Finite Element Method) as well as numerical calculation.

It is found that the critical stress effect is on the Sleeve part. Meanwhile on the other areas, the distribution of stress equally distributed around the surface. Because the stress is proportional to the strain, the result for the strain was found similar to the stress pattern.

For the rest analysis, the results are compared with the first analysis which acts as the benchmark to these analyses. Different material was used to differentiate the value of stress, strain and total deformation for all the analysis. The results are in shown in section 4.5.

It was found from the analysis and considering other factors, Silicon Carbide produced the best result and recommended to as the suitable material for mechanical seal of API610 pumps.

5.2 Recommendation

Few recommendations that can be studied to improve this research. Which are: -

1. Actual experiment should be conducted and result to be compared with the analysis of this study for more accurate result.
2. To include thermal analysis as for working temperature is a major influence that can cause mechanical seal failure.
3. To include hydrodynamics analysis finite-volume method, as fluid condition is also a major influence that can cause mechanical seal failure.

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