STRESS ANALYSIS ON MECHANICAL SEAL (SINGLE SEAL)

By

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CERTIFICATION OF APPROVAL

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May 2011

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHD AIMRAN BIN ABDULLAH

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CHAPTER 1

INTRODUCTION

1.1 Background of Study

The mechanical seal was first invented by George Cook and called by "Cook Seal". The invention was done as an alternative way to replace the soft packing seal that always produces a leakage when the rotating machinery operates. The seal have the flexibility to accommodate misalignment, shaft deflection, and break away shock loading. It resists clogging in extremely viscous fluids. All mechanical seals are constructed of three basic sets of parts:

1. A set of primary seal faces: one that rotates and one that remains stationary.
2. A set of secondary seals known as shaft packing and insert mountings, such as o-rings, rubber boots, PTFE or Grafoil wedges, or V-Rings.
3. Mechanical Seals have hardware including gland rings, collars, compression rings, pins, springs, retaining rings and bellows.

In order for the mechanical seal to perform over an extended time period with low friction the faces are generally hydrodynamically lubricated. The fluid film will need to carry substantial load. If the load becomes too high for the film surface contact will take place with consequent bearing failure. This lubricating film is generally of the order of 3 micrometers thick, or less. This thickness is critical to the required sealing function. Mechanical seals often have one face of a suitable solid lubricant such that the seal can still operate for a period without the fluid film. Others force such as axial and radial force should be taken into account into the design of mechanical seal.

1.2 Problem Statement

Mechanical seal is designed for most rotating equipment application such as sealing for pumps, mixer and agitator. The function is to helps joint systems or mechanisms together by preventing leakage under extreme pressure, shaft speed and temperature condition. Normally the lifespan is short due to seal material failure. The material failure is caused by the stress exerted during its operation under the extreme condition. Aluminum oxide has been use widely nowadays on mechanical seal but the price is unreasonable.

1. 3 Objective and Scope of Work

The main objectives for this research are:

1. Develop finite element analysis model for single type mechanical seal
2. Perform finite element analysis of single type mechanical seal based on hydrodynamic pressure
3. Investigate the effect of 3 different combination of material which are:
    i. Ductile-ductile material
    ii. Ductile-brittle material
    iii. Brittle-brittle material

The scope of work for this research is to do the analysis on stress profiling using ANSYS software on single type of mechanical seal base design. Once the analysis has been done, the scope continued on theoretical calculation for the stress distribution. Then, the research continued on analysing the type of material to determine the best material for mechanical seal. Some characteristics and raw material price will be determining the material selection.

CHAPTER 2

LITERATURE REVIEW

2.1 Basic Concept of Mechanical Seal

Nowadays, mechanical seal is applied in almost every sector of technology where rotating shaft require control of the leakage of pressurized fluid. Mechanical seal is accepted as the sealing liquid because of their very low leak rates. Compared to soft packing, mechanical seals not only form an extremely fine leakage path but also generate less friction which is important at high speed operation. The clearance for the radial face is small, hence reducing the shear force. The stationary and rotating face seal will act as primary seal members meanwhile other parts such as O-ring, wedges and packing will act as secondary seal members. The secondary seal member usually chosen based on the characteristic such as temperature, compatibility and elastomeric qualities [1].

2.2 Face Load

2.2.1 Closing force

The research will focus on the maximum contact force that the seal can maintain before the leakage occurs. The axial force given by $P_1.A_1$ and $P_2.A_2$, and spring force, F_s should be analyzed critically to get the exact result. From Figure 1, when the force acts axially on the floating ring, it tends to close the sealing interface. Secondary seal members such as o-ring will slides on cylindrical surface of radius r_b. By defining r_b, we can define the area, A_1 and pressure at outer periphery. Others pressure could be ignored as it will not give any big effect to the seal design. At the secondary seal's sliding contact there is sometimes a shear force, due to relative thermal expansion between shaft and housing or seal face wear. Depending on the direction of slip a friction force is transmitted from the secondary seal to the floating ring and contributes to external force [2].

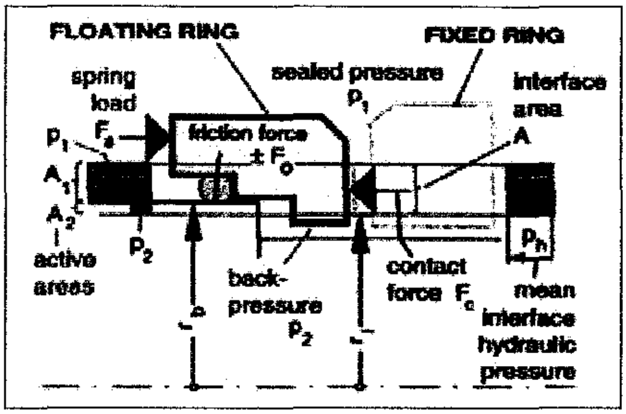

Figure 1: Force acting on seal ring [2]

2.2.2 Opening Forces

The hydrodynamic pressure profile in the radial leakage flow between the seal faces begins at absolute seal pressure, P_1 and end at ambient pressure, P_2. Figure 2 shows the pressure distribution for different interface geometry. The tangential shear flow will interact with film thickness when the shaft rotates. Hence, it will produce hydrostatic and hydrodynamic pressure. The total pressure will denotes as total mean film pressure acting over area, A. Mechanical contact will occur if the mean film pressure is insufficient to counterbalance the specific closed force [2].

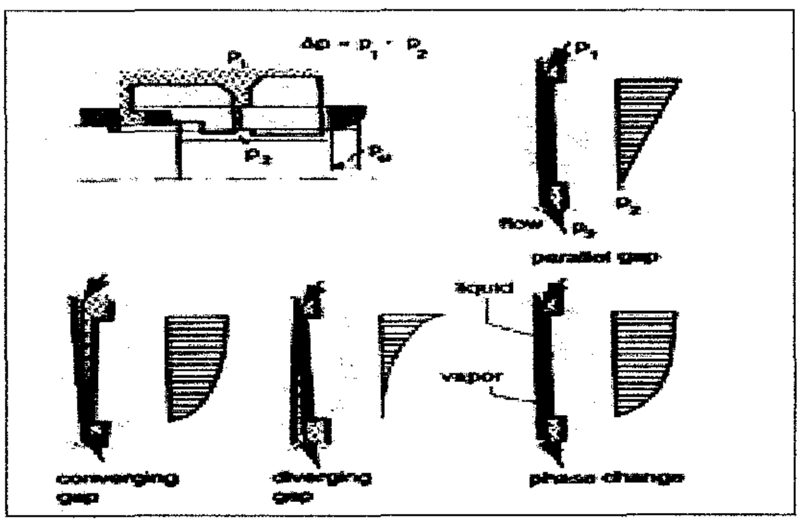

Figure 2: Hydrostatic pressure profile for various geometries [2]

2.3 Hydrodynamic Pressure

The most pressure produce in mechanical seal is Hydrodynamic pressure. When a fluid is present in the mechanical seal, it will create a pressure within the film to separate the face supporting load exerted and preventing the physical contact [7]. When the mechanical seal is under operation, the combination of the pressure will lift the seal ring tilt and developed a thin fluid film. One of the pressures is hydrodynamic pressure. The hydrodynamic pressure mostly is created when the seal ring tilt is moving from its original position. When the tilt is moving, clearance will be produced and the leakage will happened. When the pressure is so high and the compression occur, the deflection in turn affects the hydrodynamic pressure and elastohydrodynamic state occurred.

The hydrodynamic analysis can be determined using Reynolds Equation:

$$\frac{\delta}{\delta x}\left(\frac{\rho h3}{\eta}.\frac{\delta \rho}{\delta x}\right)+\frac{\delta}{\delta y}\left(\frac{\rho h3}{\eta}.\frac{\delta \rho}{\delta x}\right)=6\left[\frac{\delta}{\delta x}(\mathrm{U}\ \rho \mathrm{h})+\frac{\delta}{\delta y}(\mathrm{V}\ \rho \mathrm{h})+2\ \rho\frac{\delta h}{\delta t}\right] \quad (1)$$

Where h is the thickness of fluid film [3].

The left side denote the change of film pressure along coordinate x and y meanwhile the right side denotes the following physical meaning:

$\mathrm{U}\ \rho\frac{\delta h}{\delta x}$, $\mathrm{V}\ \rho\frac{\delta h}{\delta x}$ are physical wedge action which is important for pressure generation [3].

$\rho \mathrm{h}\frac{\delta U}{\delta x}$, $\rho \mathrm{h}\frac{\delta V}{\delta x}$ are strength actions, considering the rate at which the surface velocity change in sliding direction [3].

$\mathrm{Uh}\frac{\delta \rho}{\delta x}$, $\mathrm{Vh}\frac{\delta \rho}{\delta x}$ are density wedge action, concerned with the rate which lubrication density change with temperature rinse or other heat source [3].

$\rho\frac{\delta h}{\delta t}$ is normal squeeze term which provides a valuable cushioning effect when bearing surface tend to be pressed together [3].

2.4 Balance Ratio

Balance ratio *B* is an important and a widely used term. It is defined as the ratio between the average loads, P_f, imposed on the face by the action of the sealed pressure to the sealed pressure, *p* itself. Figures 3 and 4 show how this definition is applied to outside and inside pressurized seals. The pressure P_f is determined simply by the sealed pressure times the net area over which it acts divided by the area of the face area. The balance ratio equations are:

$$P\pi\,(r_o^2 - r_b^2) = P_f\pi\,(r_o^2 - r_i^2) \qquad (2)$$

$$B = B_o = \frac{P_f}{P} = \frac{r_o^2 - r_b^2}{r_o^2 - r_i^2} \quad \text{(Outside pressurized seal)} \qquad (3)$$

$$P\pi\,(r_b^2 - r_i^2) = P_f\pi\,(r_o^2 - r_i^2) \qquad (4)$$

$$B = B_i = \frac{P_f}{P} = \frac{r_b^2 - r_i^2}{r_o^2 - r_i^2} \quad \text{(Inside pressurized seal)} \qquad (5)$$

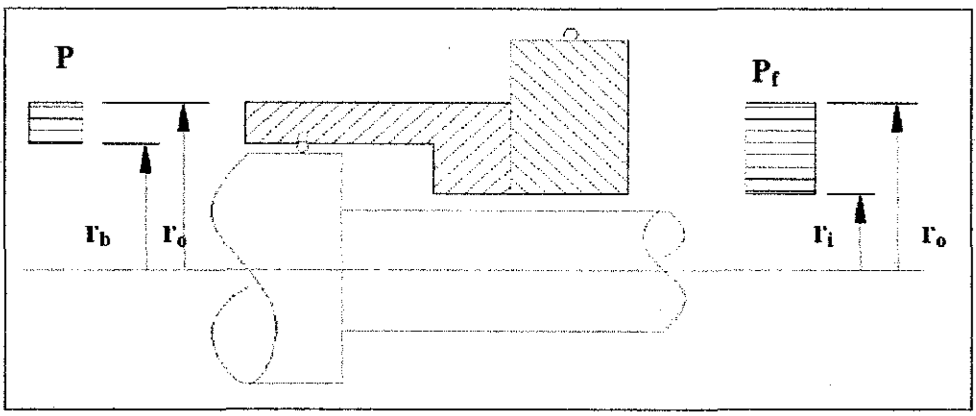

Figure 3: Outside Pressurized Seal, Balance Ratio [5]

Considering the equations from (3) and (5), the balance ratio is the ratio of the net hydraulically loaded face area to the actual face area. If the balance ratio *B* is greater than 1.0, the seal is termed unbalanced. That is the average pressure on the face is greater than the sealed pressure. If *B* is less than 1.0, it is termed to be balanced. In a balanced seal, the average pressure on the face is less than the sealed pressure. While most seals that operate at high pressure are of the balanced type, many low-pressure seals operate at *B* greater than 1.0 because of convenience of design [5].

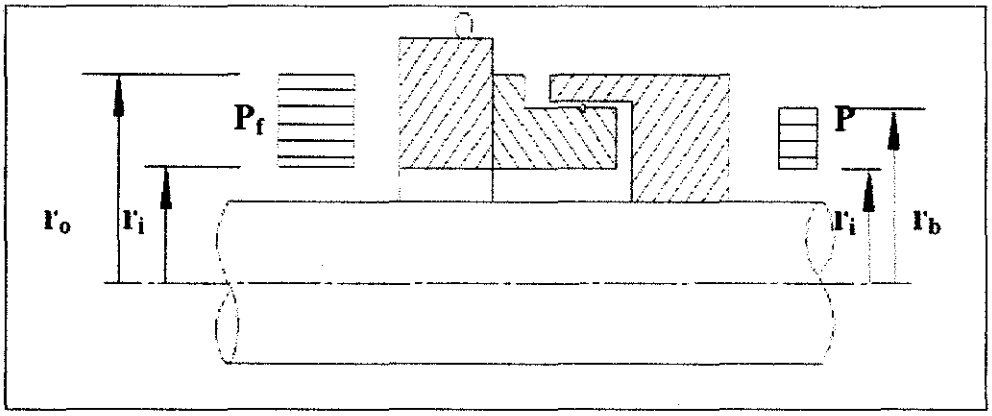

Figure 4: Inside Pressurized Seal, Balance Ratio [5]

2.5 Elementary Theory of Operation

In developing the basic theory, some assumptions and simplifications are considered. The sealed fluid enters between the faces and distributes itself in a manner such that the average value of the fluid pressure between the faces is proportional to the sealed pressure, K_p. This fluid pressure has to at least support some of the applied load. The spring force assures static equilibrium in the axial direction due to the hydrodynamic pressure or contact pressure in between the faces[2].

Summing up all the forces in the axial direction,

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

Thus the mean pressure can be calculated using the equation (5)

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

The value of K greatly affects the contact pressure and it is called the K factor or the pressure gradient factor. If the fluid flow caused by the hydrostatic pressure across the face is laminar and incompressible, the value for K is assumed to be ½ and if it is a compressible flow then K is 2/3 [2].

2.6 Physical and Mechanical Properties

Mechanical seal calculations are considerably simplified by using a coefficient of friction. It is understood that the friction changes from 0.03 to 0.3 and generally it is found to be around 0.1 for most of the applications. Furthermore, the coefficient of friction is reduced when the seal leaks. Others factors such as young modulus and tensile yield strength also determine the lifespan of mechanical seal. The selection of material is based on the process and not all process will suitable to one material of face seal. When material cost increase, the selection will be determine by service life to initial cost. Table 2.1 shows the properties of various mechanical seal face and the cost to produce the mechanical seal face [4].

Table 2.1: Properties of various mechanical seal face and price [4]

Structural	**Carbon Graphite**	**Tungsten Carbide**	**Stainless Steel**	**Silicon Carbide**
Young's Modulus (GPa)	21	620	193	414
Poisson's Ratio	0.31	0.24	0.31	0.19
Density (kg/m^3)	1720	15800	7750	3210
Tensile Yield Strength (MPa)	-	344.8	207	3440
Compressive Yield Strength (MPa)	208	4483	207	462
Tensile Ultimate Strength (GPa)	3.50	1.52	0.58	21.00
Compressive Ultimate Strength (GPa)	-	544.60	-	1.37
Price (USD)	7	20	12	10

2.7 Boundary Condition

In the general case of mechanical seal face, the Reynolds equation requires a solution over entire region of contacting faces. The only boundary condition arises in the general two-dimensional problem as shown in Figure 5 where:

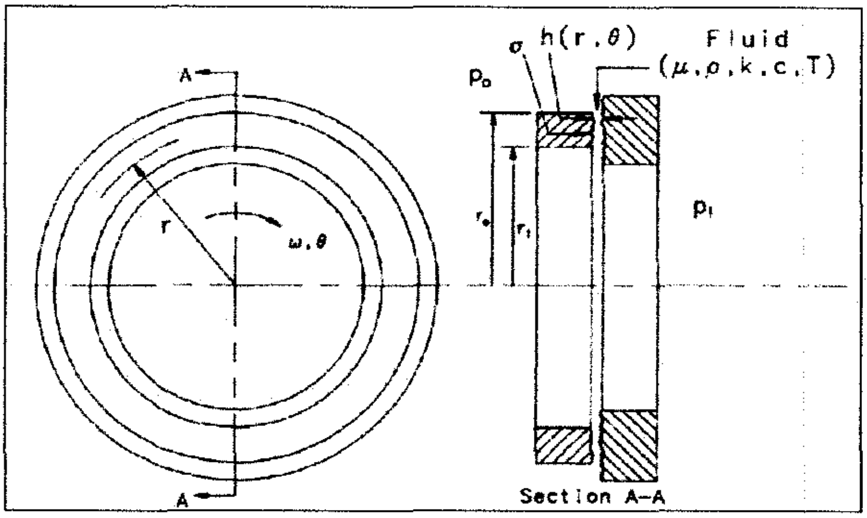

Figure 5: Mechanical seal tribology system [12]

$$P\ (\theta, r = r_i) = P_i \qquad (8)$$

$$P\ (\theta, r = r_o) = 0 \qquad (9)$$

at the inside and outside of the mechanical seal. The condition for equations (8) and (9) only apply for the mechanical seal in steady state condition. For most practical result it will be shown that the solution may be taken as periodic in θ or as axisymmetric. Thus, various special case boundary conditions are developed as needed. Reynolds equation is valid only for region where a liquid extend completely between the two surfaces and is not broken up into region of gas or vapor [12].

CHAPTER 3

METHODOLGY

3.1 Project Flow

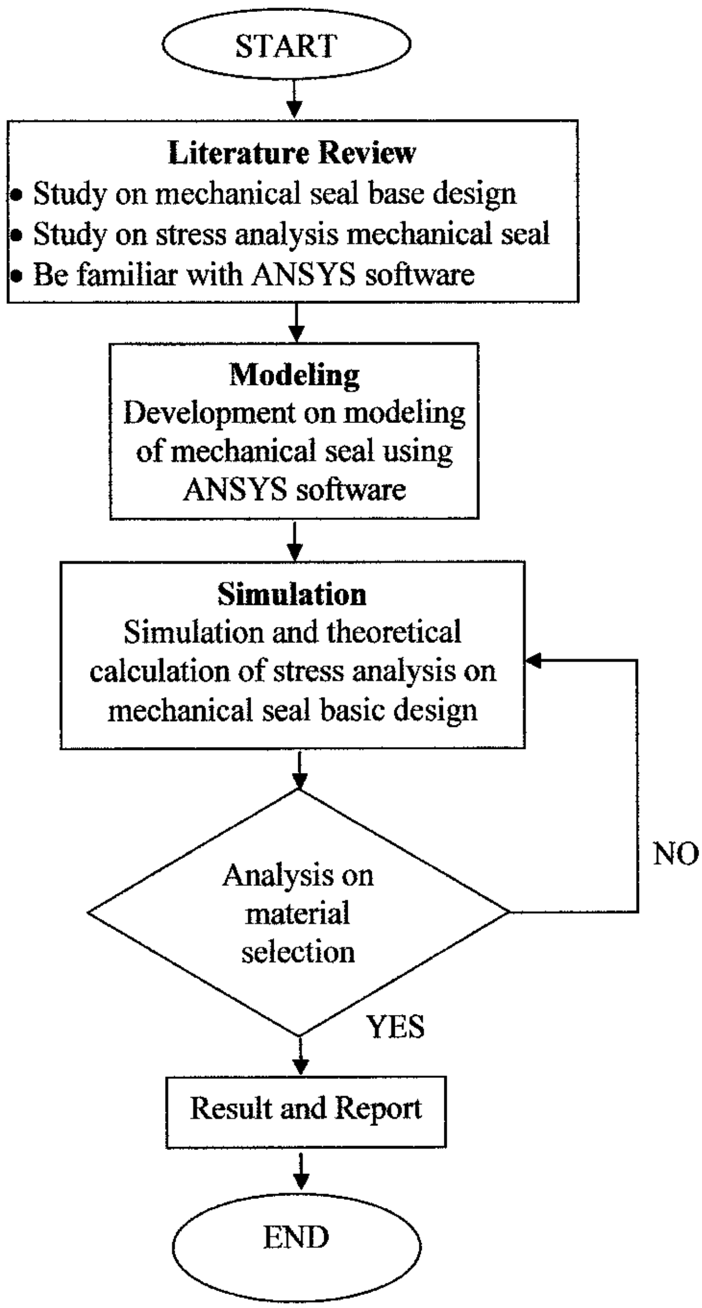

Figure 6: Methodology of the project

3.2 Project Phases

3.2.1 Literature Review

For the first stage of the research, the initial requirement will be based on the information from required gathering method. Among the initial techniques will be information from the journals, books and also case studies that have any relevance to the topics. If there are any changes in requirement, or if there are any refinements on the studies, the requirement gathering phase is revisited to suit any change.

3.2.2 Modeling

There are a few assumptions had to be made to analyze the stress profile on mechanical seal. The assumptions are:

1. Fluid is laminar and not turbulent
2. Fluid is Newtonian
3. Density is constant throughout the fluid
4. Viscosity is constant across the fluid
5. Fluid inertia effect is negligible
6. The effect of roughness on fluid flow is negligible
7. The film is thin such that velocity gradient across the film predominate
8. The effect of micro asperities as they develop pressure on themselves is negligible
9. Temperature will be constant throughout this analysis
10. Zero leakage sealing
11. Characteristic of seal ring and seal medium does not change with temperature

The project is divided into two parts which are numerical analysis and modeling of mechanical seal using ANSYS. Numerical analysis of mechanical seal can be determined using the concept of thick-walled cylinder. This is because the shape of mechanical seal can be assumed as cylinder and the radius of mechanical seal is more than 1/20 of its thickness. Figure 7 shows a typical infinitesimal element of unit thickness which defines two radii parameter, r and $r + dr$ and an angle $d\Phi$. The normal radial acting on the infinitesimal element at distance r will be σ_r meanwhile for variable stress will be $\sigma_r + {}^{d\sigma r}/_{dr}$. The final results from this derivation are:

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

$$\sigma_r = \frac{p_i r_i^2}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right) \tag{10}$$

$$\sigma_t = \frac{p_i r_i^2}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right) \tag{11}$$

The detail derivation as per attach in appendices 5. [12]

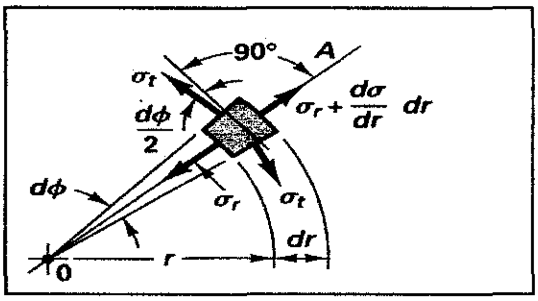

Figure 7: Element in mechanical seal [7]

For modeling using ANSYS, the model will followed the exact dimension of the single seal using ANSYS software. All the information for the operation will be included in this stage to complete the design of the seal.

3.2.3 Simulation

In the simulation, the objective is to analyze the stress distribution given to the seal. The seal will cut into half and the load will be given to the surface. Once the information is complete, the ANSYS software will automatically calculate the area of stress on the seal. Other than using software, the research will also try to calculate manually using theoretical formulations.

3.2.4 Material Selection

In this phase, the design of the mechanical seal will be the same meanwhile the properties of certain part will be changed. As stated in the objective of the project, the only properties that will be change is the mechanical seal face. Various type of material will be used to determine the best material for this type of mechanical seal based on certain environment condition.

3.3 Tooling

3.3.1 ANSYS Workbench

ANSYS Workbench is a process-centric computer-aided design/computer-assisted manufacturing/computer-aided engineering (CAD/CAM/CAE) system that fully uses next generation object technologies and leading edge industry standards. The solid model used is created using Autodesk Inventor 2010 and converted into 'iam' file .This model is used in stress analysis using ANSYS. For a plate-like structure a way to create a solid model is to extend/extrude a cross-section of the plate to form a three dimensional solid model.

The solid model is then imported into ANSYS for stress analysis. The solid model is then imported into ANSYS for stress analysis. Stress analysis entails:

- To specify the type of element(s) to use
- To set the material property values
- To have the software mesh the model
- To specify boundary conditions
- To define the loads that is applied
- To let the program solve the problem

3.3.2 Autodesk Inventor

Autodesk Inventor offers a comprehensive, flexible set of software for 3D mechanical design, product simulation, tooling creation, and design communication. Inventor takes you beyond 3D to Digital Prototyping by enabling you to design, visualize, and simulate your products.
Design - integrate all design data into a single digital model.
Simulate - digitally simulate product's real-world performance.
Visualize - create a virtual representation of final product.

3.3.3 Microsoft Excel

Microsoft Excel is a spreadsheet prepared by Microsoft. The featured includes calculation, graphing tools, pivot tables and a macro programming language called Visual Basic for Applications. It has been a very widely applied spreadsheet for these platforms, especially since version 5 in 1993. For this research, Microsoft Excel will be used to calculate the numerical calculation as well to plot the graph.

3.4 Analysis step using ANSYS Workbench

1. Import file

   The file is imported from the AUTODESK INVENTOR 2010 to ANSYS Workbench in the "iam" format.

2. Setting the properties

   The properties of the model are set before the analysis can be done. Each part of the seal needs to be specifying in order to get an accurate result. The boundary condition such as the symmetry of the seal also needs to determine to simplify the analysis.

3. Meshing the model

   The function of the meshing is to get an accurate result in solving the problems in CAE solution. In this project, the model is meshed using tetrahedral element the specification as in Table 3.1. The shape of the tetrahedral will be different in all area depend on the minimum and maximum edge length of the model.

Table 3.1: meshing specification

Default Face Spacing	
Option	Angular Resolution
Angular Resolution (Degrees)	30
Minimum Edge Length (mm)	0.3
Maximum Edge Length (mm)	6.1

4. Import file to simulation

   The model will be import to ANSYS Extrude where the simulation will be run in this section. All the properties need to be recheck and the connection between the parts in mechanical seal should be joints. 4MPa of pressure will be put on the surface of the seal in tangential direction.

5. Result

   The results such as total deformation, equivalent elastic strain and equivalent stress will be determined in this section in the final.

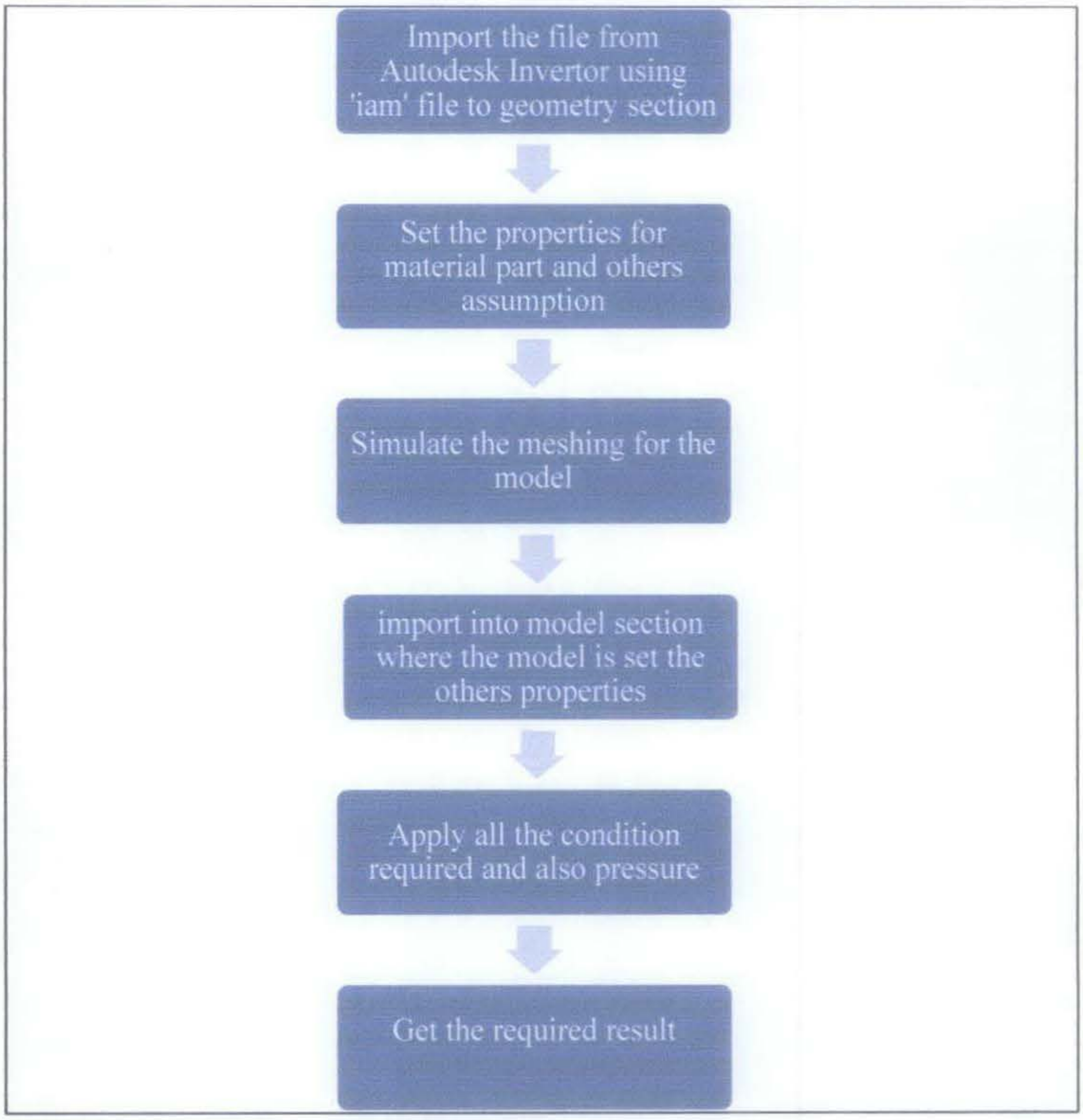

Figure 8: Process in ANSYS

CHAPTER 4

RESULT AND DISCUSSION

4.1 Modeling of Mechanical Seal using Autodesk Inventor

The model was created using Autodesk Inventor 2010. The model was first draw on 2-Dimensional part by part before it had been revolving into 3-Dimensinal view. The dimension of all the parts have been followed the real mechanical seal dimension and the dimension used is inch. The spring has been compressed and the properties have been set in Inventor. Finally, the file has been saving in 'iam' format before it can be import to ANSYS Workbench. Figure 9 shows the full view of mechanical seal in INVERTOR 2010 meanwhile Figure 10 shows the half view of mechanical seal. From Figure 10, the parts are different in color to differentiate the properties of the material. The symmetrical of the model had already applied in this software as well as in ANSYS to make sure the procedure in analyzing the mechanical seal will be smooth.

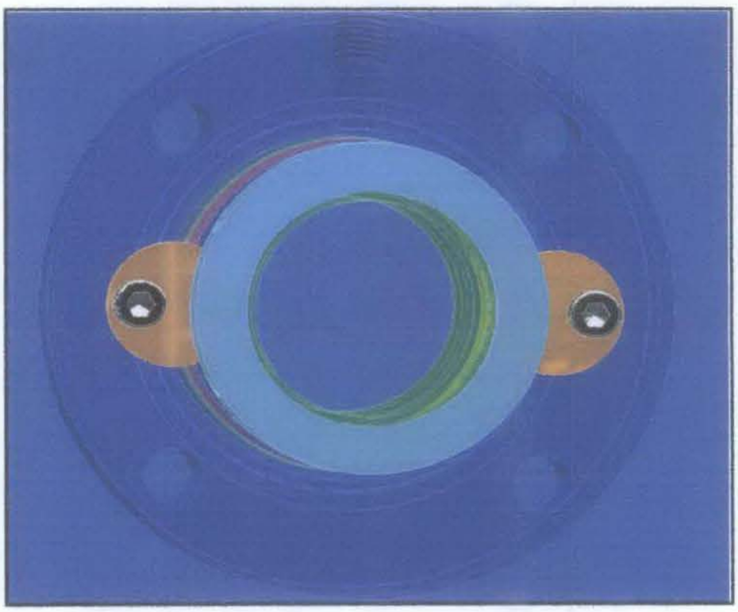

Figure 9: Front View of Mechanical Seal

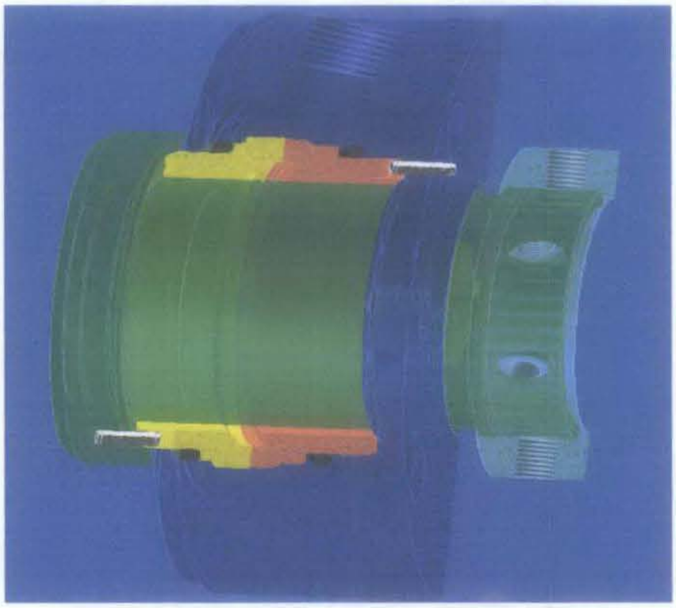

Figure 10: Half View of Mechanical Seal

4.2 Meshing process using ANSYS Workbench

Figure 11 shows the model of mechanical seal before. With 30° in angular resolution, 0.3 mm in minimum edge length and 6.1 mm in maximum edge length, the meshing result is showed in Figure 12. The function of meshing is to simplify in solving step with the correct choice of maximum and minimum edge length.

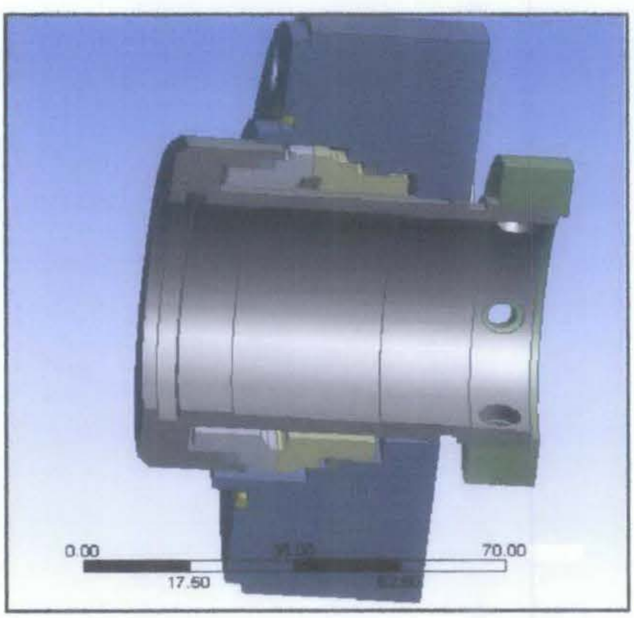

Figure 11: Geometry of Mechanical Seal

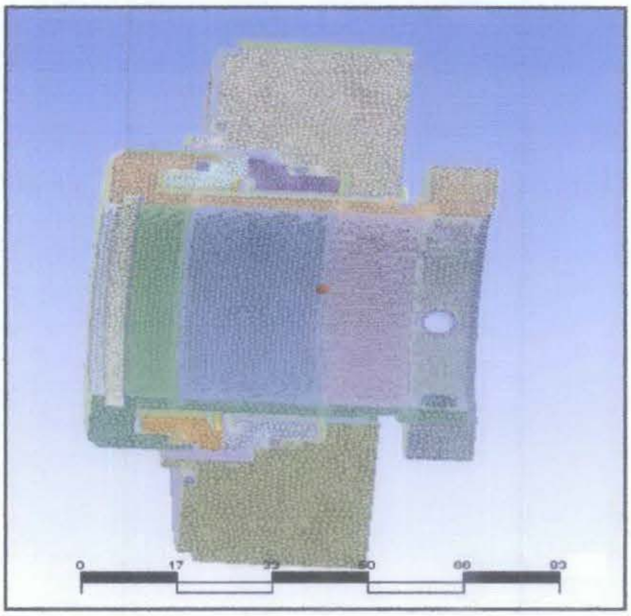

Figure 12: Meshing of Mechanical Seal

4.3 Analysis on Numerical Calculation

Numerical calculation will be use equation (10) and (11) as the reference.

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

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

Using Microsoft Excel, the stress distribution on radial and tangential direction from the above equations can be found.

4.3.1 Radial Distribution Stress

Table 4.1 shows the data used to calculate the stress distribution in the mechanical seal in the radial direction. The initial pressure use is 2.8 MPA with the constant radius in inner and outer. The result shown is in the range 2.8 MPA to 83 Pa. From Figure 12, the radial stress distribution in the mechanical seal is inversely proportional to their distance. As the distance increase, the stress distribution decrease to zero value. The graph shows that the model is having tensile stress on the radial direction. The result is inversely proportional because the seal is assumed as the hollow thick-wall cylinder. For the analysis using ANSYS, the result will be slightly different as the shape of the real mechanical seal is slightly different with the assumption made in this section.

Table 4.1: Radial distribution stress throughout mechanical seal

Number	Initial Pressure, P_i(KPa)	Inner radius, r_i (mm)	Outer radius, r_o(mm)	Distance from inner radius, r (mm)	Stress (σ_r) (KPa)
1	-2800	0.022352	0.060909	2.24E-02	2.800E+03
2	-2800	0.022352	0.060909	2.44E-02	2.284E+03
3	-2800	0.022352	0.060909	2.64E-02	1.882E+03
4	-2800	0.022352	0.060909	2.84E-02	1.563E+03
5	-2800	0.022352	0.060909	3.05E-02	1.306E+03
6	-2800	0.022352	0.060909	3.25E-02	1.095E+03
7	-2800	0.022352	0.060909	3.45E-02	9.204E+02
8	-2800	0.022352	0.060909	3.66E-02	7.740E+02
9	-2800	0.022352	0.060909	3.86E-02	6.502E+02
10	-2800	0.022352	0.060909	4.06E-02	5.444E+02
11	-2800	0.022352	0.060909	4.26E-02	4.533E+02
12	-2800	0.022352	0.060909	4.47E-02	3.744E+02
13	-2800	0.022352	0.060909	4.67E-02	3.055E+02
14	-2800	0.022352	0.060909	4.87E-02	2.451E+02
15	-2800	0.022352	0.060909	5.08E-02	1.917E+02
16	-2800	0.022352	0.060909	5.28E-02	1.444E+02

17	-2800	0.022352	0.060909	5.48E-02	1.023E+02
18	-2800	0.022352	0.060909	5.68E-02	6.454E+01
19	-2800	0.022352	0.060909	5.89E-02	3.065E+01
20	-2800	0.022352	0.060909	6.09E-02	8.873E-02

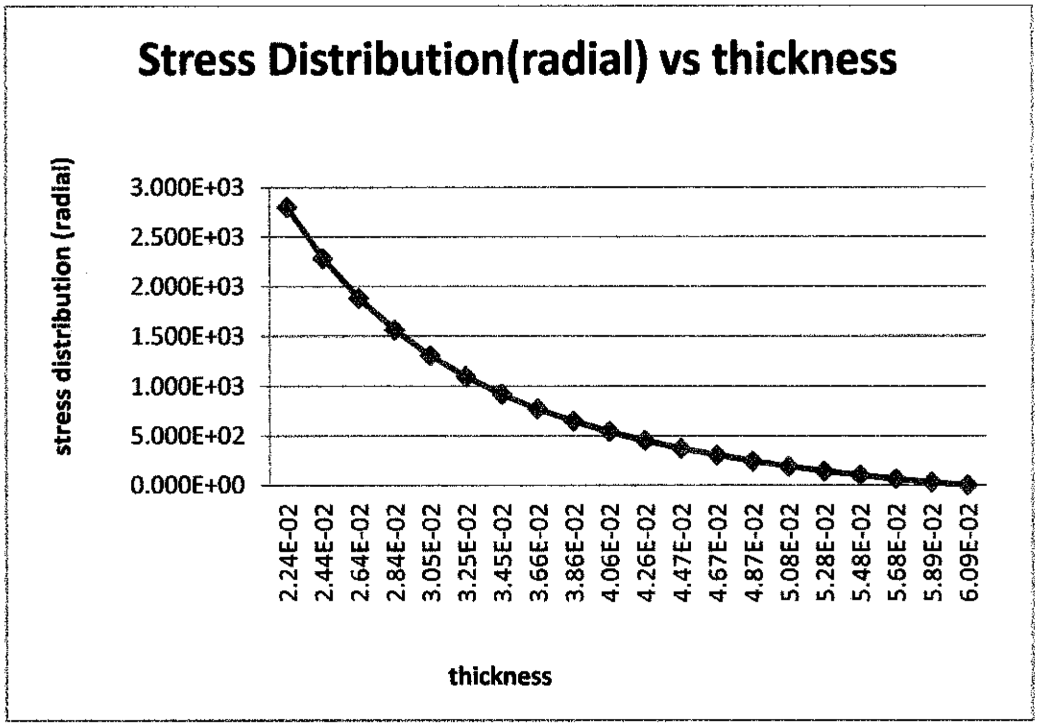

Figure 13: Graph Stress Distribution (radial) VS thickness

4.3.2 Tangential Distribution Stress

Table 4.2 shows the data used to calculate the stress distribution in the mechanical seal in the radial direction. The data is divided into 20 parts which will determine more accurate result. The initial pressure use is 2.8 MPA with the constant radius in inner and outer. The result shown is in the range -3.8 MPA to -0.9 MPa. Figure 14 shows the stress distribution on tangential side is inversely proportional to the thickness of the mechanical seal. As the thickness increase, the stress decrease constantly on radial direction of mechanical seal. This shows that the mechanical seal is having compressive stress on the tangential side. Both figure 12 and figure 13 results will be compared with the analysis using ANSYS in the next section of this study.

Table 4.2: tangential distribution stress throughout mechanical seal

Number	**Initial Pressure, P_i(KPa)**	**Inner radius, r_i (mm)**	**Outer radius, r_o(mm)**	**Distance from inner radius, r (mm)**	**Stress (σ_r) (KPa)**
1	-2800	0.022352	0.060909	2.24E-02	-3671.51
2	-2800	0.022352	0.060909	2.44E-02	-3155.36
3	-2800	0.022352	0.060909	2.64E-02	-2753.53
4	-2800	0.022352	0.060909	2.84E-02	-2434.60
5	-2800	0.022352	0.060909	3.05E-02	-2177.24
6	-2800	0.022352	0.060909	3.25E-02	-1966.57
7	-2800	0.022352	0.060909	3.45E-02	-1791.93
8	-2800	0.022352	0.060909	3.66E-02	-1645.56
9	-2800	0.022352	0.060909	3.86E-02	-1521.67
10	-2800	0.022352	0.060909	4.06E-02	-1415.87
11	-2800	0.022352	0.060909	4.26E-02	-1324.82
12	-2800	0.022352	0.060909	4.47E-02	-1245.89
13	-2800	0.022352	0.060909	4.67E-02	-1177.02
14	-2800	0.022352	0.060909	4.87E-02	-1116.58
15	-2800	0.022352	0.060909	5.08E-02	-1063.23

16	-2800	0.022352	0.060909	5.28E-02	-1015.92
17	-2800	0.022352	0.060909	5.48E-02	-973.77
18	-2800	0.022352	0.060909	5.68E-02	-936.05
19	-2800	0.022352	0.060909	5.89E-02	-902.16
20	-2800	0.022352	0.060909	6.09E-02	-871.60

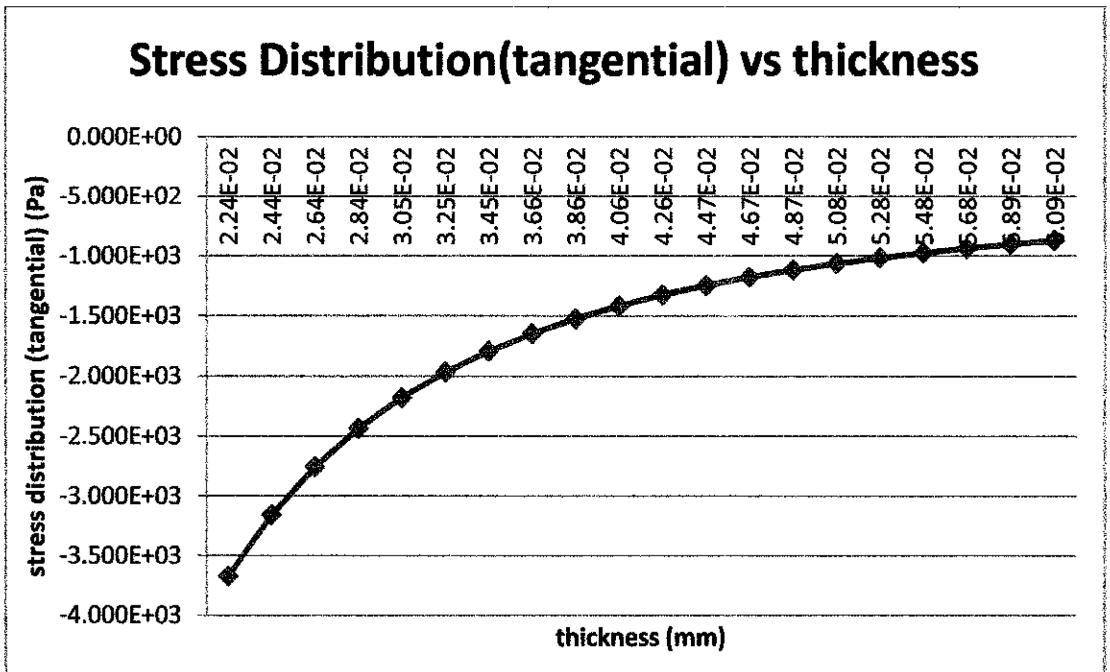

Figure 14: Graph Stress Distribution (tangential) VS thickness

4.4 Graphical analysis using ANSYS Workbench

4.4.1 Graphic result for first analysis using Silicon Carbide and Tungsten Carbide as face seal

1. Equivalent Stress

The equivalent von Mises stress profile shown in Figure 15 is the baseline result for this research. The face combinations are using the silicon carbide and tungsten carbide. The stress effect mainly takes place on the right hand side which places nearly on the O-ring slot. On the other area, the distribution of the stress become equally distribute around the surface of the sleeve. The minimum and maximum stress value is around 8741.8 Pa to 322.39 MPa throughout the mechanical seal. The stress is more focusing on the O-ring slot because the end of the sleeve near to the O-ring slot is fixed. Hence, the stress distribution is high at that area compares to others area. That the reason why the seal is provided with the O-ring to prevent the leakage at that area. As compare with the numerical calculation, stress distribution is same where both results show the stress in having tensile stress.

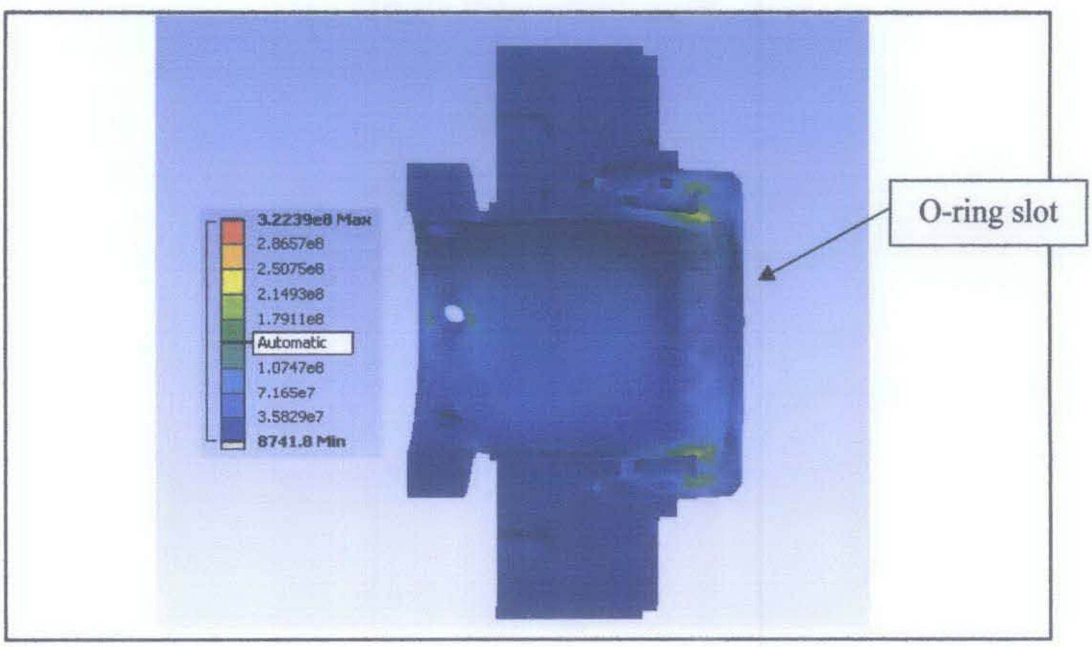

Figure 15: Equivalent stress on mechanical seal face between Silicon Carbide and Tungsten Carbide

2. Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in Figure 16 is the baseline result for this research. The face combination is a combination of silicon carbide and tungsten carbide. The strain profile is quite similar to the stress profile in Figure 15 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the sleeve is around 4.5294e-008 to 1.63e-004 respectively. The positive values show the strain is tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

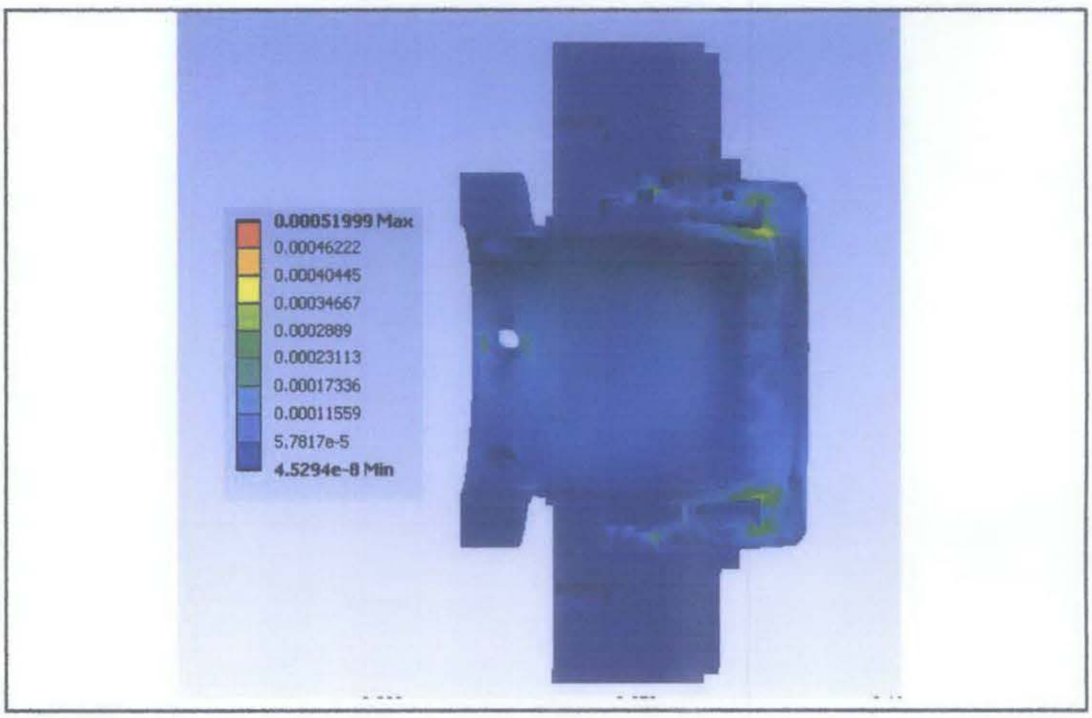

Figure 16: Equivalent elastic strain on mechanical seal face between Silicon Carbide and Tungsten Carbide

3. Total Deformation

From Figure 17, the result is obtained when 4MPa pressure is applied to the mechanical seal sleeve. The range of the deformation on mechanical seal sleeve is from zero m to 6E-5 m. The deformation value is higher at mating ring because of less support in that area. On the other side of the sleeve, the deformation is small as the end side is in fixed condition.

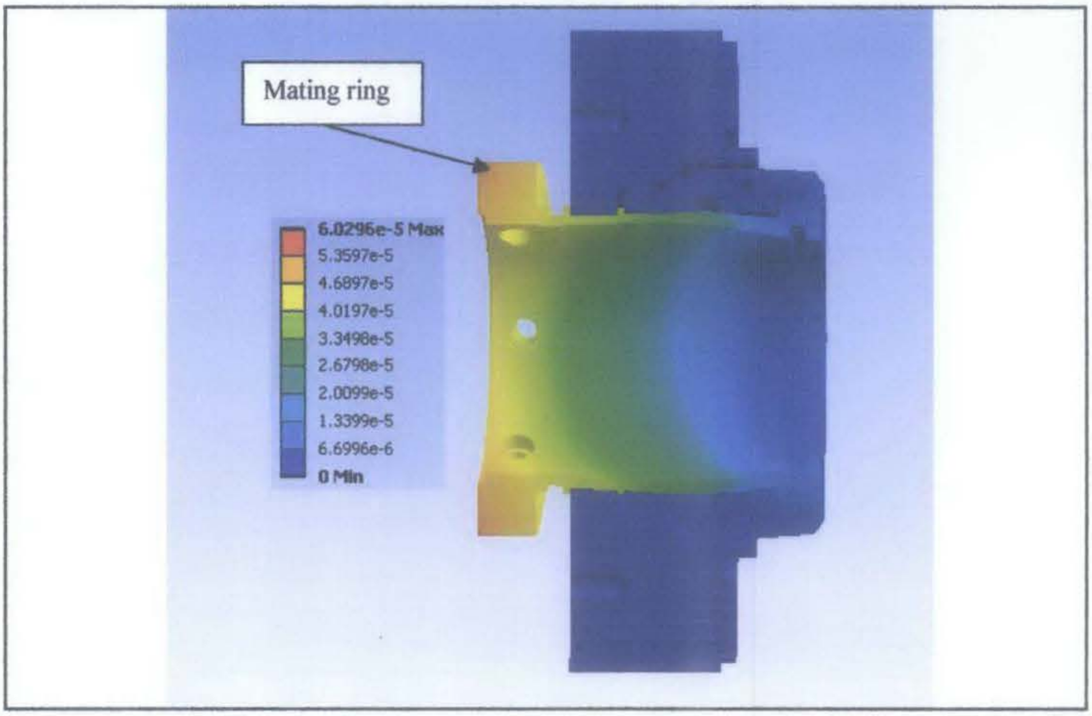

Figure 17: Total deformation on mechanical seal face between Silicon Carbide and Tungsten Carbide

4.4.2 Graphic result for second analysis using Stainless Steel and Tungsten Carbide as face seal (Ductile-Ductile Material)

1. Equivalent Stress

The equivalent von Mises stress profile shown in Figure 18 is the second analysis result for this research. The face combinations are using the silicon carbide and tungsten carbide. The stress effect mainly takes place on the right hand side which places nearly on the O-ring slot. On the other area, the distribution of the stress become equally distribute around the surface of the sleeve. The minimum and maximum stress value is around 7.58 kPa to 305 MPa throughout the mechanical seal. The stress is more focusing on the O-ring slot because the end of the sleeve near to the O-ring slot is fixed. Hence, the stress distribution is high at that area compares to others area. That the reason why the seal is provided with the O-ring to prevent the leakage at that area. As compare with the numerical calculation, stress distribution is same where both results show the stress in having tensile stress.

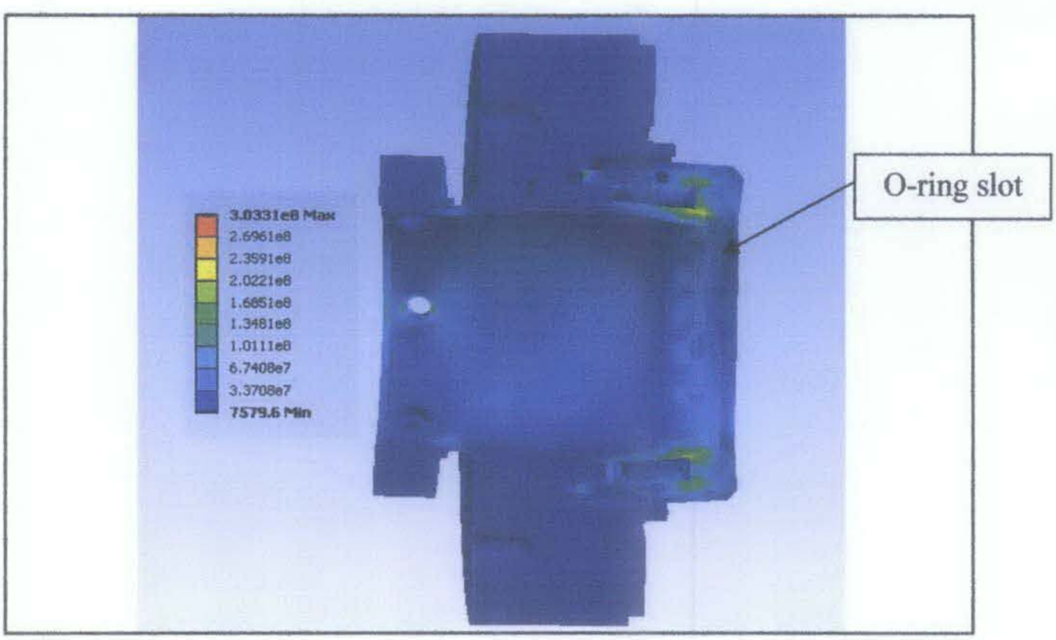

Figure 18: Equivalent stress on mechanical seal face between Stainless Steel and Tungsten Carbide

2. Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in Figure 19 is the second result for this research. The face combination is a combination of silicon carbide and tungsten carbide. The strain profile is quite similar to the stress profile in Figure 18 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the sleeve is around 3.92e-008 to 1.57e-003 respectively. The positive values show the strain is tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

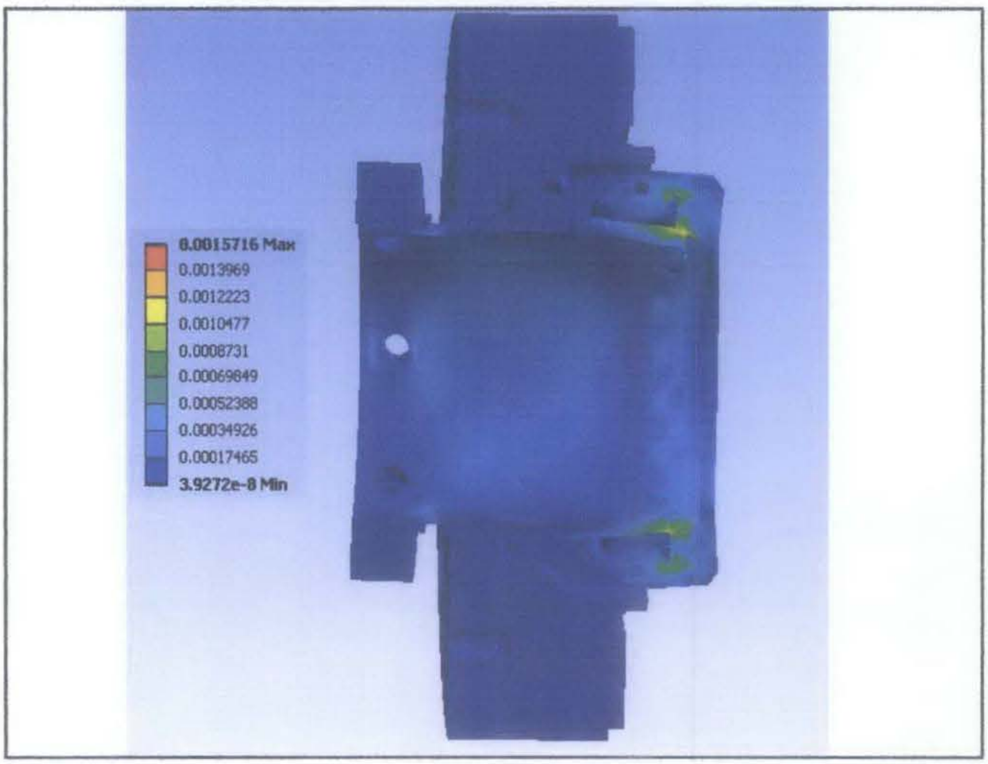

Figure 19: Equivalent elastic strain on mechanical seal face between Stainless Steel and Tungsten Carbide

3. Total Deformation

From Figure 20, the result is obtained when 4MPa pressure is applied to the mechanical seal sleeve. The range of the deformation on mechanical seal sleeve is from zero m to 1.76e-4 m. The deformation value is higher at mating ring because of less support in that area. On the other side of the sleeve, the deformation is small as the end side is in fixed condition.

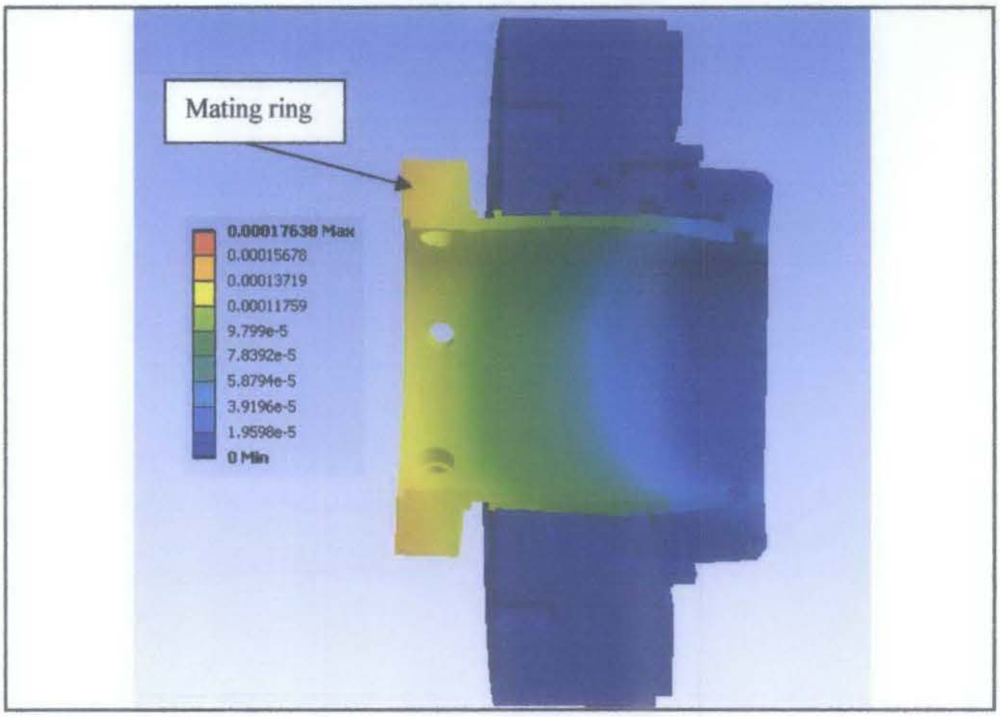

Figure 20: Total deformation on mechanical seal face between Stainless Steel and Tungsten Carbide

4.4.3 Graphic result for third analysis using Carbon Graphite and Stainless Steel as face seal (Brittle-Ductile Material)

1. Equivalent Stress

The equivalent von Mises stress profile shown in Figure 21 is the third analysis result for this research. The face combinations are using the silicon carbide and tungsten carbide. The stress effect mainly takes place on the right hand side which places nearly on the O-ring slot. On the other area, the distribution of the stress become equally distribute around the surface of the sleeve. The minimum and maximum stress value is around 13.76 kPa to 303 MPa throughout the mechanical seal. The stress is more focusing on the O-ring slot because the end of the sleeve near to the O-ring slot is fixed. Hence, the stress distribution is high at that area compares to others area. That the reason why the seal is provided with the O-ring to prevent the leakage at that area. As compare with the numerical calculation, stress distribution is same where both results show the stress in having tensile stress.

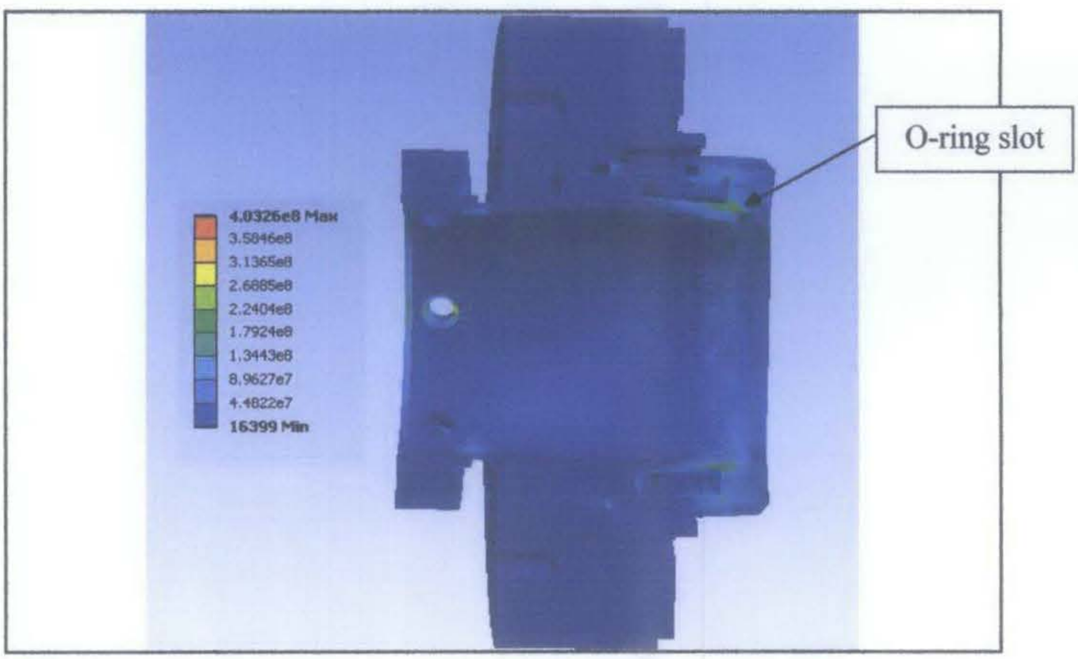

Figure 21: Equivalent stress on mechanical seal face between Carbon Graphite and Stainless Steel

2. Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in Figure 22 is the third result for this research. The face combination is a combination of silicon carbide and tungsten carbide. The strain profile is quite similar to the stress profile in Figure 21 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the sleeve is around 7.13e-008 to 5.81e-003 respectively. The positive values show the strain is tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

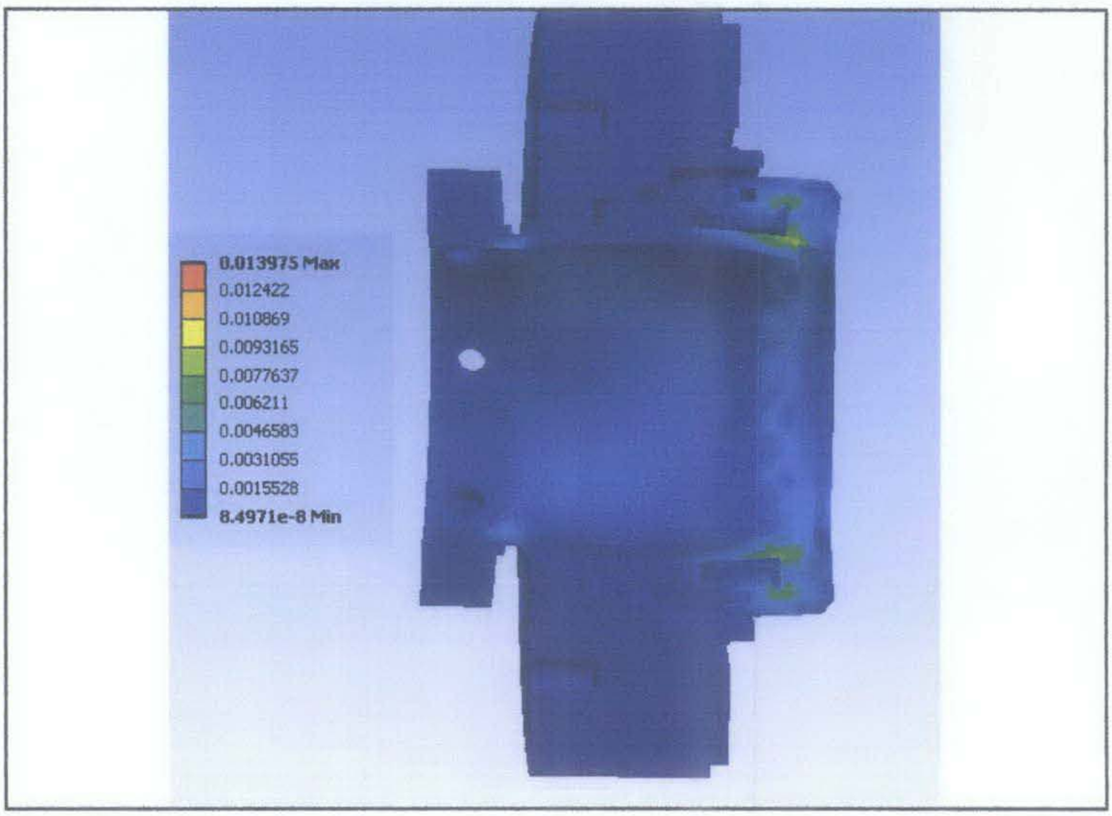

Figure 22: Equivalent elastic strain on mechanical seal face between Carbon Graphite and Stainless Steel

3. Total Deformation

From Figure 23, the result is obtained when 4MPa pressure is applied to the mechanical seal sleeve. The range of the deformation on mechanical seal sleeve is from zero m to 1e-4 m. The deformation value is higher at mating ring because of less support in that area. On the other side of the sleeve, the deformation is small as the end side is in fixed condition.

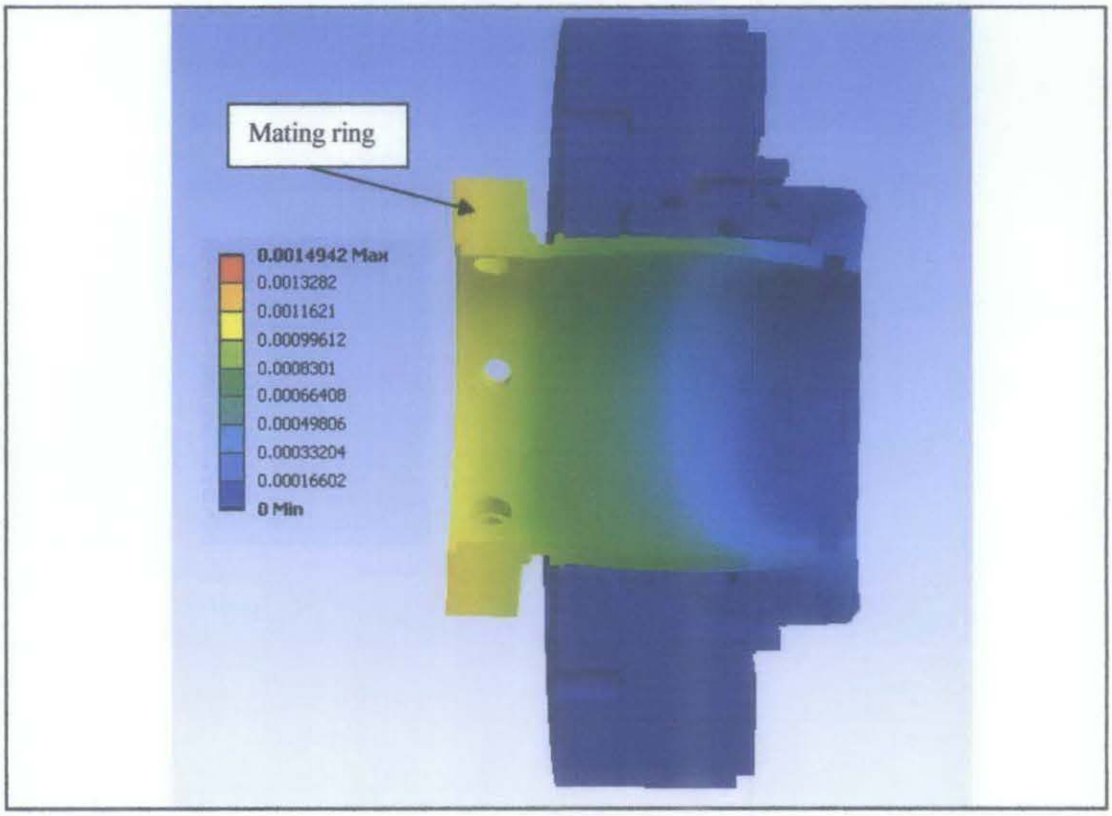

Figure 23: Total deformation on mechanical seal face between Carbon Graphite and Stainless Steel

4.4.4 Graphic result for fourth analysis using Carbon Graphite and Silicon Carbide as face seal (Brittle-Brittle Material)

1. Equivalent Stress

The equivalent von Mises stress profile shown in Figure 24 is the fourth analysis result for this research. The face combinations are using the silicon carbide and tungsten carbide. The stress effect mainly takes place on the right hand side which places nearly on the O-ring slot. On the other area, the distribution of the stress become equally distribute around the surface of the sleeve. The minimum and maximum stress value is around 12.68 kPa to 306 MPa throughout the mechanical seal. The stress is more focusing on the O-ring slot because the end of the sleeve near to the O-ring slot is fixed. Hence, the stress distribution is high at that area compares to others area. That the reason why the seal is provided with the O-ring to prevent the leakage at that area. As compare with the numerical calculation, stress distribution is same where both results show the stress in having tensile stress.

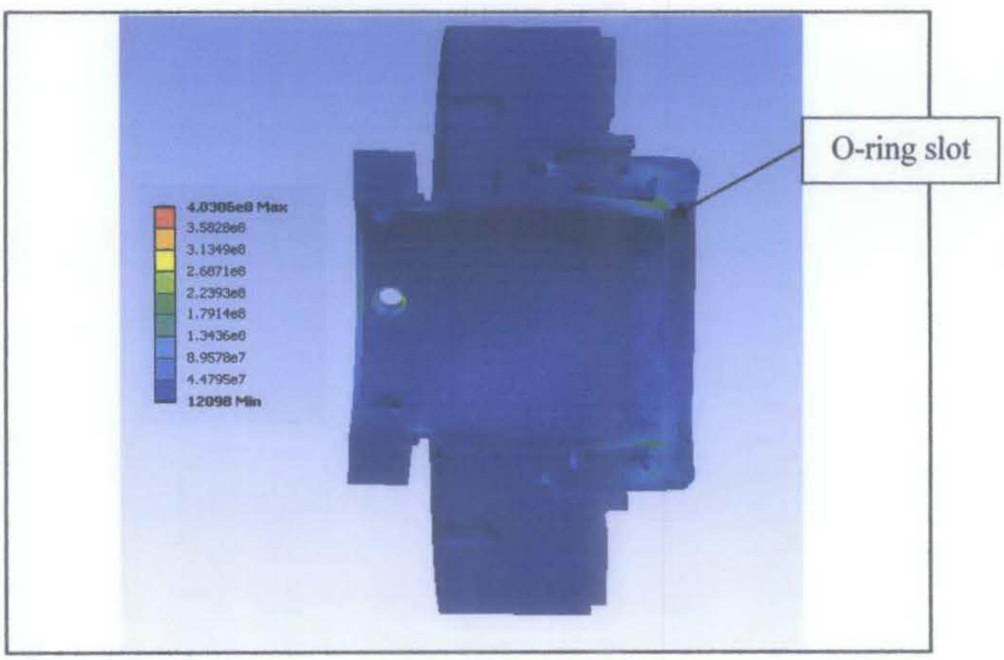

Figure 24: Equivalent stress on mechanical seal face between Carbon Graphite and Silicon Carbide

2. Equivalent Elastic Strain

The equivalent von Mises elastic strain shown in figure 25 is the fourth result for this research. The face combination is a combination of silicon carbide and tungsten carbide. The strain profile is quite similar to the stress profile in figure 24 as the strain is proportionally to the stress. The minimum and maximum value for the strain throughout the sleeve is around 6.57e-008 to 6.08e-003 respectively. The positive values show the strain is tensile condition. This is because the material elongates in the direction of normal stress, contraction in perpendicular direction occur.

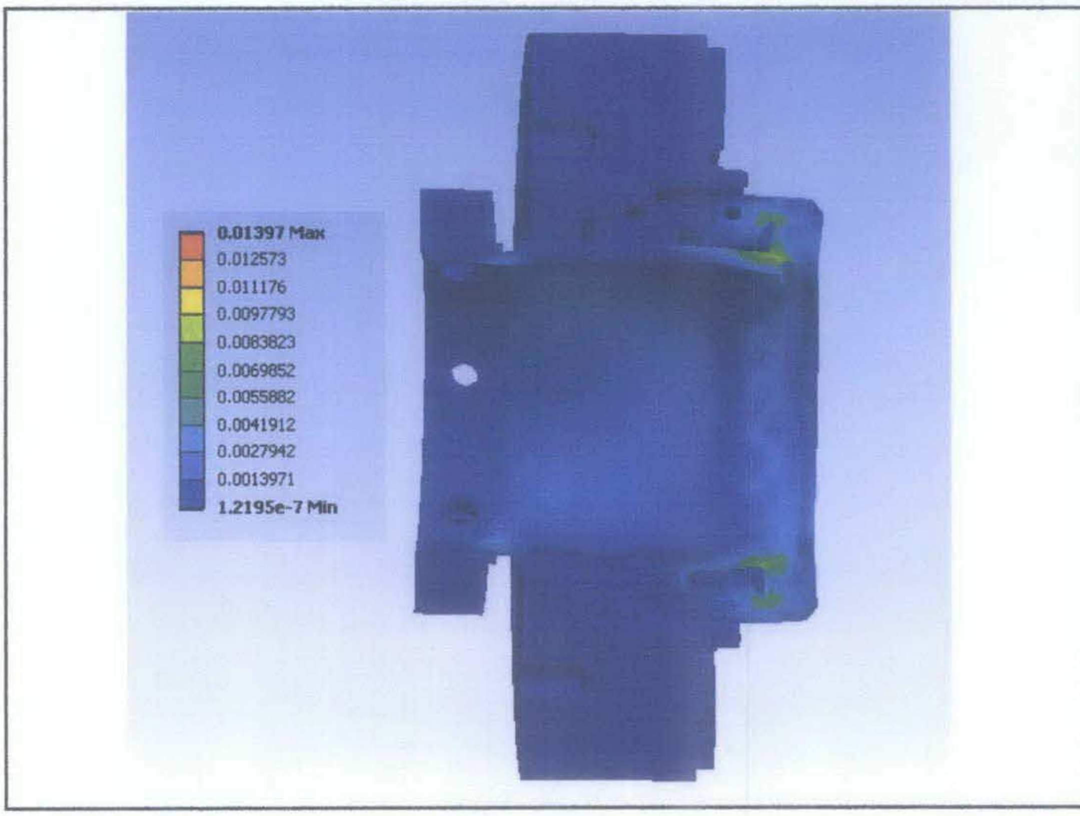

Figure 25: Equivalent elastic strain on mechanical seal face between Carbon Graphite and Silicon Carbide

3. Total Deformation

From Figure 26, the result is obtained when 4MPa pressure is applied to the mechanical seal sleeve. The range of the deformation on mechanical seal sleeve is from zero m to 9.87e-5 m. The deformation value is higher at mating ring because of less support in that area. On the other side of the sleeve, the deformation is small as the end side is in fixed condition.

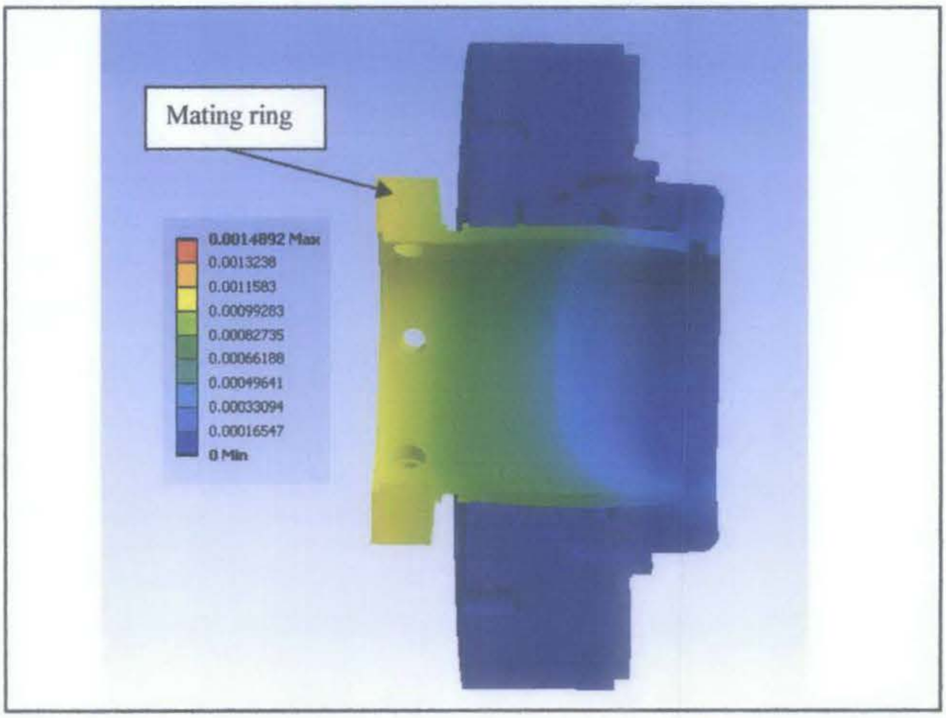

Figure 26: Total deformation on mechanical seal face between Carbon Graphite and Silicon Carbide

4.5 Overall Result

Table 4.3 showed the comparison value for all the analysis that had been done. After a few considerations, a combination of carbon graphite and stainless steel produce a reliable result for this mechanical seal analysis. Hence, the combination of carbon graphite and stainless steel can replace the combination of silicon carbide and tungsten carbide. Based on section 2.6, the price for carbon graphite and stainless steel is quite a reasonable price to use for the mechanical seal face.

Table 4.3: Comparison value for all the analysis

Analysis	Maximum value of total deformation ($x10^{-5}$)m	Rank	Maximum value of equivalent von-Mises elastic strain ($x10^{-4}$)	Rank	Maximum value of equivalent von-Mises stress (MPa)	Rank	Total rank
Combination of silicon carbide and tungsten carbide	16.3	3	5.2	1	322	4	8
Combination of stainless steel and tungsten carbide	17.6	4	15.3	2	305	2	8
Combination of carbon graphite and stainless steel	10.0	2	58.1	3	303	1	6
Combination of carbon graphite and silicon carbide	9.57	1	60.3	4	306	3	8

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

As a conclusion, the objectives of the study to develop finite element analysis model for single type mechanical seal, to perform finite element analysis of single type mechanical seal based on hydrodynamic pressure and to investigate the effect of different combination of material had been achieved by using the finite element analysis method as well as numerical calculation. It is found that the critical stress effect is mainly took place near the O-ring slot 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. It is also found that, the sleeve of mechanical seal deformed critically on mating ring side due to less support on that area. For all the analysis, the pattern of the result is similar but different in the values.

For the rest analysis, the result had been compared with the first analysis which acts as the benchmark to these analyses. The different combination of 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 analyses and considering other factors, the carbon graphite and stainless steel produce the best result and has been selected to replace the face seal available in the market.

5.2 Recommendation

There are several recommendations that can be considered in order to enhance this research. The recommendations are:

1. Experimental study should be done to compare with the analysis of this study.

2. A calibration with the manufacturer will make the study much easier as they would provide more information not only about their product, but also the methodology use to analyze their product

3. More numerical calculation should be done to support the graphical analysis of the study.

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APPENDICES

1. First analysis using Silicon Carbide and Tungsten Carbide as face seal

Apply load condition

Table 1: Apply load condition on mechanical seal for first analysis

Object Name	*Pressure*	*Cylindrical Support*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	8 Faces	2 Faces
Definition		
Define By	Normal To	
Type	Pressure	Cylindrical Support
Magnitude	4e+006 Pa	
Suppressed	No	
Radial		Fixed
Axial		Fixed
Tangential		Fixed

Result from the analysis

Table 2: Result from first analysis

Definition			
Type	Total Deformation	Equivalent (von-Mises) Elastic Strain	Equivalent (von-Mises) Stress
Display Time	End Time		
Results			
Minimum	0. m	4.5294e-008 m/m	8741.8 Pa
Maximum	16.3296e-005 m	5.1999e-004 m/m	3.2239e+008 Pa
Minimum Occurs On	Solid4:1		
Maximum Occurs On	Solid5:1	Solid1:1	
Information			
Time	1. s		
Load Step	1		
Substep	1		
Iteration Number	1		

2. Second analysis using Stainless Steel and Tungsten Carbide as face seal

Apply load condition

Table 3: Apply load condition on mechanical seal for second analysis

Object Name	*Pressure*	*Cylindrical Support*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	8 Faces	2 Faces
Definition		
Define By	Normal To	
Type	Pressure	Cylindrical Support
Magnitude	4e+006 Pa (ramped)	
Suppressed	No	
Radial		Fixed
Axial		Fixed
Tangential		Fixed

Result from the analysis

Table 4: Result from second analysis

Scope			
Type	Total Deformation	Equivalent (von-Mises) Elastic Strain	Equivalent (von-Mises) Stress
Display Time	End Time		
Results			
Minimum	0. m	3.9272e-008 m/m	7579.6 Pa
Maximum	1.7638e-004 m	1.5716e-003 m/m	3.0531e+008 Pa
Minimum Occurs On	Solid4:1		
Maximum Occurs On	Solid5:1	Solid1:1	
Information			
Time	1. s		
Load Step	1		
Substep	1		
Iteration Number	1		

3. Third analysis using Carbon Graphite and Stainless Steel as face seal

Apply load condition

Table 5: Apply load condition on mechanical seal for third analysis

Object Name	*Pressure*	*Cylindrical Support*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	8 Faces	2 Faces
Definition		
Define By	Normal To	
Type	Pressure	Cylindrical Support
Magnitude	4e+006 Pa	
Suppressed	No	
Radial		Fixed
Axial		Fixed
Tangential		Fixed

Result from the analysis

Table 6: Result from third analysis

Definition			
Type	Total Deformation	Equivalent (von-Mises) Elastic Strain	Equivalent (von-Mises) Stress
Display Time	End Time		
Results			
Minimum	0. m	7.1313e-008 m/m	13763 Pa
Maximum	1.0186e-004 m	5.8123e-003 m/m	3.0352e+008 Pa
Minimum Occurs On	Solid4:1		
Maximum Occurs On	Solid5:1	Solid2:1	Solid1:1
Information			
Time	1. s		
Load Step	1		
Substep	1		
Iteration Number	1		

4. Fourth analysis using Carbon Graphite and Silicon Carbide as face seal

Apply load condition

Table 7: Apply load condition on mechanical seal for fourth analysis

Object Name	*Pressure*	*Cylindrical Support*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	8 Faces	2 Faces
Definition		
Define By	Normal To	
Type	Pressure	Cylindrical Support
Magnitude	4+006 Pa	
Suppressed	No	
Radial		Fixed
Axial		Fixed
Tangential		Fixed

Result from the analysis

Table 8: Result from fourth analysis

Definition			
Type	Total Deformation	Equivalent (von-Mises) Elastic Strain	Equivalent (von-Mises) Stress
Display Time	End Time		
Results			
Minimum	0. m	6.5692e-008 m/m	12679 Pa
Maximum	9.8702e-005 m	6.0815e-003 m/m	3.0611e+008 Pa
Minimum Occurs On	Solid4:1		
Maximum Occurs On	Solid5:1	Solid2:1	Solid1:1
Information			
Time	1. s		
Load Step	1		
Substep	1		
Iteration Number	1		

5. Derivation using thick-walled cylinders

i. Static Equilibrium

The element must be in static equilibrium and express mathematically the force acting the element. All the force obtained by multiplying stresses with their respective areas which is 1. From Figure 7, summing the forces along radial line will produce:

$$\Sigma F_r = 0$$

$$\sigma_r \, rd\varnothing + 2\,\sigma_t dr\,(\frac{d\varnothing}{2}) - (\sigma_r + \frac{d\sigma r}{dr}dr)(r + dr)d\varnothing = 0 \qquad (12)$$

Simplify and neglect infinitesimals of higher order:

$$\sigma_r - \sigma_t - r\frac{d\sigma r}{dr} = 0 \qquad (13)$$

ii. Geometric Compatibility

The deformation of an element is described by its strain in the radial and tangential directions. If u represents radial displacement of r and u + $\frac{du}{dr}$ represent radial displacement of r + dr, the strain of radial and tangential will become:

$$\text{Radial direction, } \varepsilon_r = \frac{u+((du/dr)dr)-u}{dr} = \frac{du}{dr} \qquad (14)$$

$$\text{Tangential direction, } \varepsilon_t = \frac{2\,\pi\,(r+u) - 2\pi r}{2\pi r} \qquad (15)$$

iii. Properties of Material

The generalized Hooke's Law relating strains and stresses is given by:

$$\varepsilon_r = \frac{1}{E}(\sigma_r - \nu\sigma_t - \nu\sigma_x) \qquad (16)$$

$$\varepsilon_t = \frac{1}{E}(-v\sigma_r + \sigma_t - v\sigma_x) \tag{17}$$

$$\varepsilon_x = \frac{1}{E}(-v\sigma_r - v\sigma_t + \sigma_x) = 0 \tag{18}$$

(18) into (16) and (17), hence:

$$\sigma_r = \frac{E}{(1+v)(1-2v)}[(1-v)\varepsilon r + v\varepsilon t] \tag{19}$$

$$\sigma_t = \frac{E}{(1+v)(1-2v)}[v\varepsilon r + (1-v)\varepsilon t] \tag{20}$$

iv. Formation of the Differential Equation

Equation (13) can be expressed in term of one variable, *u*. thus, one eliminates the strain ε_r and ε_t from equation (16) and (17) and expressing them in term of *u*.

$$\sigma_r = \frac{E}{(1+v)(1-2v)}[(1-v)\frac{du}{dr} + v(u/r)] \tag{21}$$

$$\sigma_t = \frac{E}{(1+v)(1-2v)}[v\frac{du}{dr} + (1-v)u/r] \tag{22}$$

Substitute these values into equation 13

$$\frac{d^2r}{dr^2} + \frac{1}{r}\frac{du}{dr} - \frac{u}{r^2} = 0 \tag{22}$$

v. Solution of the Differential Equation

As can be verified by substitution, the general solution of equation (13) which gives the radial displacement *u* of any point on cylinder is:

$u = A_1r + A_2/r$; where A_1 and A_2 are constant at boundaries of body (23)

$$\frac{du}{dr} = A_1 + A_2/r^2 \tag{24}$$

And the know pressure are equal to the radial stress acting on elements: $\sigma_i(r_i) = -p_i$ &
$\sigma_r(r_0) = -p_0$ (25)

Equation (24) and (25) can be replace into equation (23) and (24) to become:

$$\sigma_r(r_i) = -p_i = \frac{E}{(1+v)(1-2v)}[A_1 - (1-2v)\frac{A_2}{r_i^2}] \tag{26}$$

$$\sigma_r(r_0) = -p_0 = \frac{E}{(1+v)(1-2v)}[A_1 - (1-2v)\frac{A_2}{r_o^2}] \tag{27}$$

Solving simultaneously for A_1 and A_2 yields:

$$\sigma_r = C_1 - C_2 / r^2 \ \& \ \sigma_t = C_1 + C_2 / r^2 \tag{28}$$

$$C_1 = \frac{p_i r_i^2 - p_o r_0^2}{r_0^2 - r_i^2} \qquad C_2 = \frac{(p_i - p_o) r_i^2 r_0^2}{r_o^2 - r_i^2} \tag{29}$$

Above derivation will resulting in the last equation as stated in (10) and (11).

PROJECT ACTIVITIES		W 1	W 2	W 3	W 4	W 5	W 6		W 7	W 8	W 9	W 10	W 11	W 12	W 13	W 14
Receive the FYP title																
Discuss and update progress with supervisor about the project.																
Literature Review	Gather info from journal, books etc							MID-SEMESTER BREAK								
	Familiarization with ANSYS Software															
	research on mechanical seal design															
Preliminary Report	preparation on report															
	Submission to supervisor															
Develop Research	Develop of mechanical seal model															
	Stress analysis on normal mechanical seal															
	Stress analysis on modified mechanical seal															
Progress Report.	preparation on report															
	Submission to supervisor															
Seminar																
Develop Research	Stress analysis on modified mechanical seal															
	research on material selection															
Submission of Interim Report Final Draft	preparation on report															
	Submission to supervisor															
Oral presentation																

PROJECT ACTIVITIES	DETAIL ACTIVITIES	W 1	W 2	W 3	W 4	W 5	W 6	W 7		W 8	W 9	W 10	W 11	W 12	W 13	W 14
Discuss and post-mortem on FYP 1 result	Repair the result from previous examiner comment								M							
	Planning activities for FYP 2								I							
Continue with the first objective	Finite element analysis on the current model of mechanical seal								D							
	Compare the result with the numerical result															
Continue with the second objective	Finite element analysis on the current model of mechanical seal using hydrodynamic pressure								S							
	Compare the result with the numerical result								E							
Continue with the third objective	Finite element analysis by changing the face material properties								M							
	Compare the result with the first objective															
	Decide which material can be replace current market															
Preparation for the report	Submission of dissertation															