Simulation of Rotating Seal Ring

by

Mohd Badaruddin Bin Mahmod 9079

Supervisor: Dr. Hussain H. Al-Kayiem

DISSERTATION submitted in partial fulfillment of

the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

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Universiti Teknologi PETRONAS

Bandar Seri Iskandar

31750 Tronoh

Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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(Dr. Hussain H.Al-Kayiem)
Project Supervisor

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

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 Badaruddin Bin Mahmod)

Universiti Teknologi PETRONAS

Bandar Seri Iskandar

31750 Tronoh

Perak Darul Ridzuan

ABSTRACT

Mechanical seals are simply a means of controlling leakage of a flow process compare to other types of seals that is less capable of performing the task adequately. In order to improve the efficiency of mechanical seals, the first step is to understand what is happening inside the system. It is impossible to achieve that understanding from laboratory testing, since such tests provide only the temperatures of the fluid at the inlet and outlet ports of the system. The purpose of this project is to understand what is the behavior of the seal rings inside the system where it will involve various types of internal flow patterns and heat distribution inside the seal chamber. In order to achieve the objective, computational simulation technique is useful to simulate the mechanical seal. The software used for the analyses is ANSYS. Throughout the simulation process, various types of operating conditions and material of the mechanical seal was considered. The scope of this research is focus on the parameters, materials and spring mechanism that will affect the performance of rotating ring in the seal chamber. The operating condition can be varied easily to understand the behavior of the seal ring. In conclusion, design changes of mechanical seal are vital to improve the performance, and the first step in moving forward with the improvements is the understanding of the behavior of the seal rings. The resulting shear stresses have been correlated to the rotational speed, separating gap between the seal rings as well as viscosity of lubricant in the separating gap.

ACKNOWLEDGEMENT

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CHAPTER 1

INTRODUCTION

1.1: Background

Previously, soft-packed glands are used for the sealing purpose and after some interval of time it was discovered that the seal system is not sufficient enough in controlling the leakage. Instead of that, the maintenance requirement and cost involve make this sealing system not economical justified to be installed. In order to solve the problems, the old system is replaced with mechanical seal which is the most versatile type of seal for rotating shaft. Nowadays, mechanical seals are used extensively in various types of industries to prevent leakage from fluid handling equipment such as centrifugal pump, where it is a mechanical device that is used to transfer liquid through the piping system by increase the pressure on the liquid. A rotating face mechanical seal consist of 2 sealing rings, where one ring rotates with the pump shaft and the other ring remains stationary. The interface between these rings establishes the seal, to control the leakage to acceptable level. The two faces seal which are the rotary and stationary ring run with small gap between the rings, typically less than 0.001mm [2].

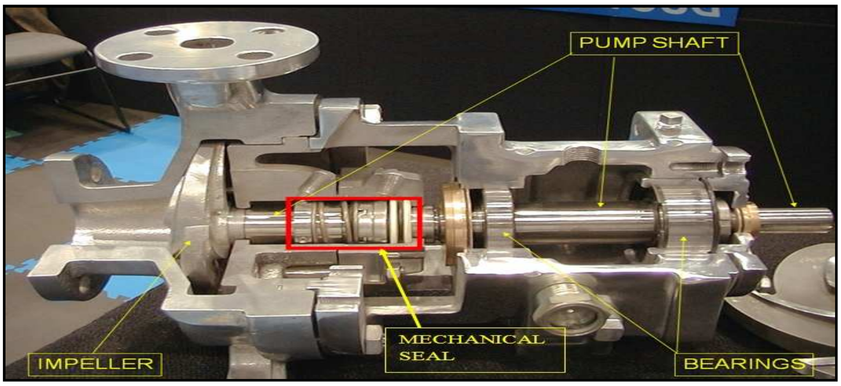

Figure 1.1: Centrifugal pump [5].

As shown in Figure 1, the liquid is supplied to the pump suction and with the centrifugal force created by the shaft and impellers; the liquid is expelled to the other location with high pressure. Based on the physical principles of hydraulics, the liquid inside the suction chamber will seek area of lower pressure. Some form of seal need to be applied to keep the liquid from leaking around the shaft at the point where the liquid enter the case.

Inside the mechanical seal system, there is a barrier fluid that operates at high pressure, which circulates within the seal system. There are 2 functions of the barrier fluid which are to prevent the process fluid leakage between the seals and also act as a coolant. The circulation of the barrier fluid occurs in a closed loop system, where it is pumped from a separate tank through an inlet port into the seal. The fluid then circulates through the seal, picking up heat generated on the mechanical seal (the rings). The heat is removed from the seal via an outlet port and flows back into the tank. The fluid in the tank is cooled by either natural or forced convection [3].

1.2: Problem Statement

The performance of mechanical seal depends greatly on three main factors that results from temperature increase inside the seal chamber. The factors are as following:

- Material selection of the rings
- Operating parameters
- Closing force (spring & hydraulic load)

All of these factors are related to the heat transfer inside the seal chamber and influencing the behavior and performance of a mechanical face seal [1]. Over certain period of seal operation, the characteristics and behavior of the seal rings will change. Same as the spring constant and the parameters involve in mechanical seal. In addition, the heat generated through the friction occurs between the ring and lubrication fluid is detrimental to the lifespan of the seal rings. Friction arises from two sources which are viscous shear and contact friction.

In order to find a design changes that would improve the performance of the mechanical seal; the first step is to understand the behavior of the seal rings. This sort of understanding is impossible to obtain from laboratory testing. Instead of involve high cost of operation and time consuming process, the laboratory testing only provide the temperature of the liquid at the inlet and outlet ports where it is not sufficient to evaluate the performance of mechanical seal.

The best technique that can be applied to obtain all the necessary information for the evaluation is the computer simulation technique. The operating condition can be varied easily to evaluate and compare the performance of the mechanical seal.

1.3: Objectives

The main objectives of this project are as following:

- To built a mathematical model of mechanical seal
- To simulate rotating ring using ANSYS
- To analyze performance of rotating ring in various operating condition

1.4: Scope of Study

Mechanical seal consist of a few components which are installed inside the seal chamber. The scope of this research is focused on the behavior of the seal ring in various operating conditions by using Microsoft Excel and ANSYS. Throughout the simulation process, the force, shear stress, and torque on the rotating ring will be analyzed and compared in order to achieve the objectives. This simulation technique is a very powerful method in analyzing the performance of rotating ring and through this method; it can lead to design changes that will improve the seal operation.

CHAPTER 2

LITERATURE REVIEW

2.1: Mechanical Seal Operation

The mechanical seal was invented by George Cook and was originally called a "Cook Seal". It was first used in refrigeration compressors and in 1990; the world market for mechanical seal was estimated at $1 billion [6]. Mechanical seals are designed to provide a better sealing capability than could be achieved by using common packing system. Basically, mechanical seal consist of 4 essential components that are installed inside the seal chamber which are primary ring, mating ring, secondary seal and spring mechanism [7].

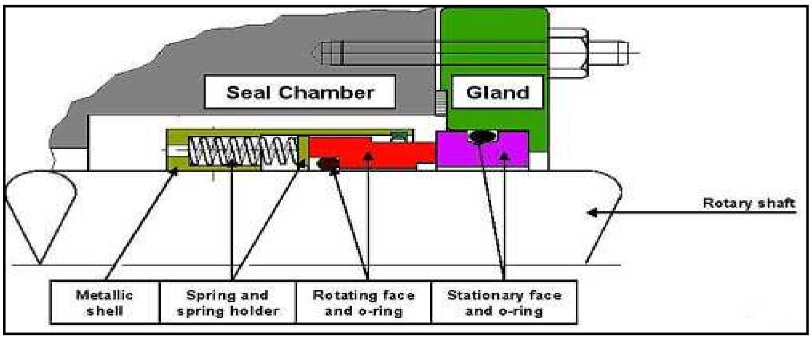

Figure 2.1: Mechanical seal configuration [7]

As shown in Figure 2.1, the primary ring is mounted to provide compensation and flexibility to allow for small relative axial and angular motion for misalignment between the parts that is surfaced-guided ring. The primary ring is the rotating ring while mating ring is rigidly mounted to the seal housing. The secondary seal will allows the primary seal to have axial and angular freedom of motion while retaining sealing integrity and lastly the spring mechanism will create a closing force where it will hold the rings together in the absence of fluid pressure [1]. The spring force will push the rotating ring against the mating ring. For proper operation of mechanical seal system, the seal faces (rotating and stationary ring) must be polished to produce a very flat surface, the seal is installed perpendicular to the shaft (alignment) and lastly the spring force must be sufficient enough to maintain the gap between the rotating and stationary ring [7]. The driving mechanism of this system is the rotation of the rotating seal at the same speed of the pump shaft. The rotating ring is made of harder material compare to the mating ring, but sometimes the design is the other way around where the rotating ring is made from softer material. The rotating ring is the one that defines the gap between the primary seal faces [1].

Lebeck (1991) says that, "A hard face sliding against a soft face has been found to be tribologically desirable. This combination provides the best overall performance" (p.9).

The seal itself is established by the gap (less than 0.001mm) between the rotating and mating rings.

Lebeck (1991) points out that, "The essential character of the surfaces is that they form a type of sliding bearing through which the sealed fluid attempts to flow perpendicular to the direction of sliding" (p.9).

The sliding between those surfaces will generate frictional heat that is vital to be removed. Hence, there is a circulation system of the coolant to the external system for cooling purposes.

Nowadays, mechanical seals are being used extensively in a wide range of industries to prevent leakage from fluid handling equipment such as centrifugal pump. Even though mechanical seal is currently the most efficient sealing system for shaft rotating equipment, there are still lots of design problems that is need to be discovered. All of the problems arise primarily due to the phenomena of heat transfer or thermal behavior inside the seal chamber especially at the face of the seal rings. This behavior will influence the performance of mechanical seal [1].

2.2: Thermal effects on seal performance and behavior

The performance and behavior of mechanical seal is mostly influenced by the thermal behavior of the seal rings. Most importantly, seal geometry may be greatly altered by temperature changes, sealed fluid may become a vapor at higher temperature, and seal material may behave differently or totally unsatisfactorily at high or low temperature [1]. As a result, the thermal environment inside the seal chamber must be carefully predicted in order to evaluate the effect on the fluid, the mechanics, and material.

Karassik (3rd Edition 2001) points out that," In contacting seal, as the shaft begins to rotate a small fluid develops along with frictional heat from the surface of sliding contact. This process occurs at seal faces" (p.2.200).

As shown in Figure 2.2, the basic nature of the seal system where the frictional heat is generated at the sliding interface between the rotating and mating ring. Most of the heat generated flows into the faces of the seal rings by conduction and then immediately transferred out to the surrounding fluid by convection.

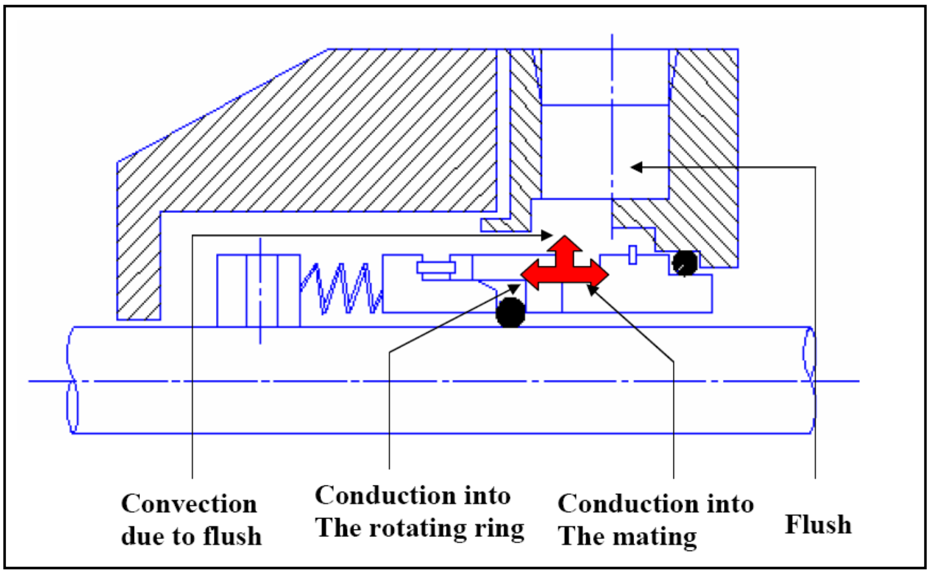

Figure 2.2: Heat generation in mechanical seal [11]

Lebeck (1991) points out that, “Interface friction generally predominates as the major heat source” (p.192).

Noel Brunetiere and Benoit Modolo (2007) points out that, “The temperature of the faces is a key parameter in a mechanical seal”.

Materials used for seals have limits on the temperature at which they can be used. Such limits often will impose a compromise on design or require that a totally different approach be taken to satisfy the design requirements [1]. It suggests that thermal effect and heat transfer on the seal ring and within the seal chamber in very important in determine the performance of the mechanical seal. As mention before, there are two heat transfer processes occur which are conduction and convection. Figure 3 shows the combination of both processes.

Conduction path plays an important role in certain seal parts and significantly influence the heat transfer in the seal assembly. Most of the heat generated at the sliding interface is conducted away from the faces to the part of the seal rings where it is then dissipated away by convection. The conduction path between the interface and the edges of the seal ring are very important to the proper cooling of the interface of the seal. The conduction path is influenced by the thermal conductivity of the seal rings; in fact it is the single most important variable in controlling the temperature difference across the conduction path [1] which means the material selection for the seal rings is vital in determines the seal performance.

2.3: Computer Simulation

Previously the engineers used laboratory testing to evaluate the performance of mechanical seal, but nowadays the engineers move forward by applying computer simulation technique for the research and testing purposes. A simulation of mechanical seal using computer software which is FLUENT has been done previously. In the simulation process, Ray Clark, Henri Azibert and Lanre Oshinowo (Journal: Computer Simulation of Mechanical Seal Leads to Design Change that Improves Coolant

Circulation, 2001) have successfully found a way to improve axial circulation of the barrier fluid. Throughout the simulation process, lots of changes and modification has been considered and FLUENT simulations were used to evaluate the effectiveness. Based on their research, by tapering the bounding surfaces of the stationary seal rings and the shaft sleeve it have promote largest effect on axial circulation. The modified design showed an increase in heat removal of about 50% compared to the traditional configuration [3].

This result is obtained due to the computer software that is useful and very convenient for such case that involves various operating conditions. In conclusion, according to Ray Clark and Henri Azibert in their journal; without computer simulation, engineers would not have known of the problem with axial circulation in the original design, nor would they have had proof that improving it would promote removal of friction-generated heat [3]. Computer simulation led them to an effective design change that might not have otherwise been discovered.

CHAPTER 3

METHODOLOGY AND PROJECT WORK

3.1: Solution Procedure

The primary method of solving the problem arises in evaluating the performance of mechanical seal is the utilization of computer simulation techniques. It is proved that by utilizing this method it will lead to design changes of the system for better performance in sealing the mechanical devices. Prior to utilize all of the necessary software, research on the mechanical seal need to be done at the first stage of this project. In order to ease the research process, the scopes of the research have been brake down into 3 main section associated to the mechanical seal system which is:

- Operating parameter/ Boundary condition
- Material
- Closing force (spring & hydraulic load)

These scopes of research are chosen due to their effects that contribute to the main factors that reflect the performance of mechanical seal. Firstly, the operating/boundary condition will involve the geometries of the mechanical seal, speed of the shaft and the parameter. Parameter of such system that is needed to be considered is the pressure, temperature and flow of the fluid inside the seal chamber. The second section of the research is the material; where it is focused on the primary seal ring which is the rotating ring. The last section of research is the closing force that consists of spring and hydraulic load. Spring mechanism (spring load) and hydraulic load is designed to provide an acceptable compromise between leakage and wear.

In applying the simulation technique, there are two software that will be used which are AUTOCAD and ANSYS. The mechanical seal model which includes the rotating ring will be generated by using AUTOCAD. Then, the area of interest in this project which is the rotating ring will be further processed which is meshing and run the simulation procedure by using ANSYS.

In utilizing this simulation technique, it involves two stages. The first stage is generating a model of an existing rotating ring based on its geometries (size) and boundary condition. At the second stage, by utilizing the same model all of the operating conditions are manipulated for various combination of condition. Finally, by evaluating and comparing all of the result it may leads to design change of mechanical seal for better performance.

3.1: Project Work Flow

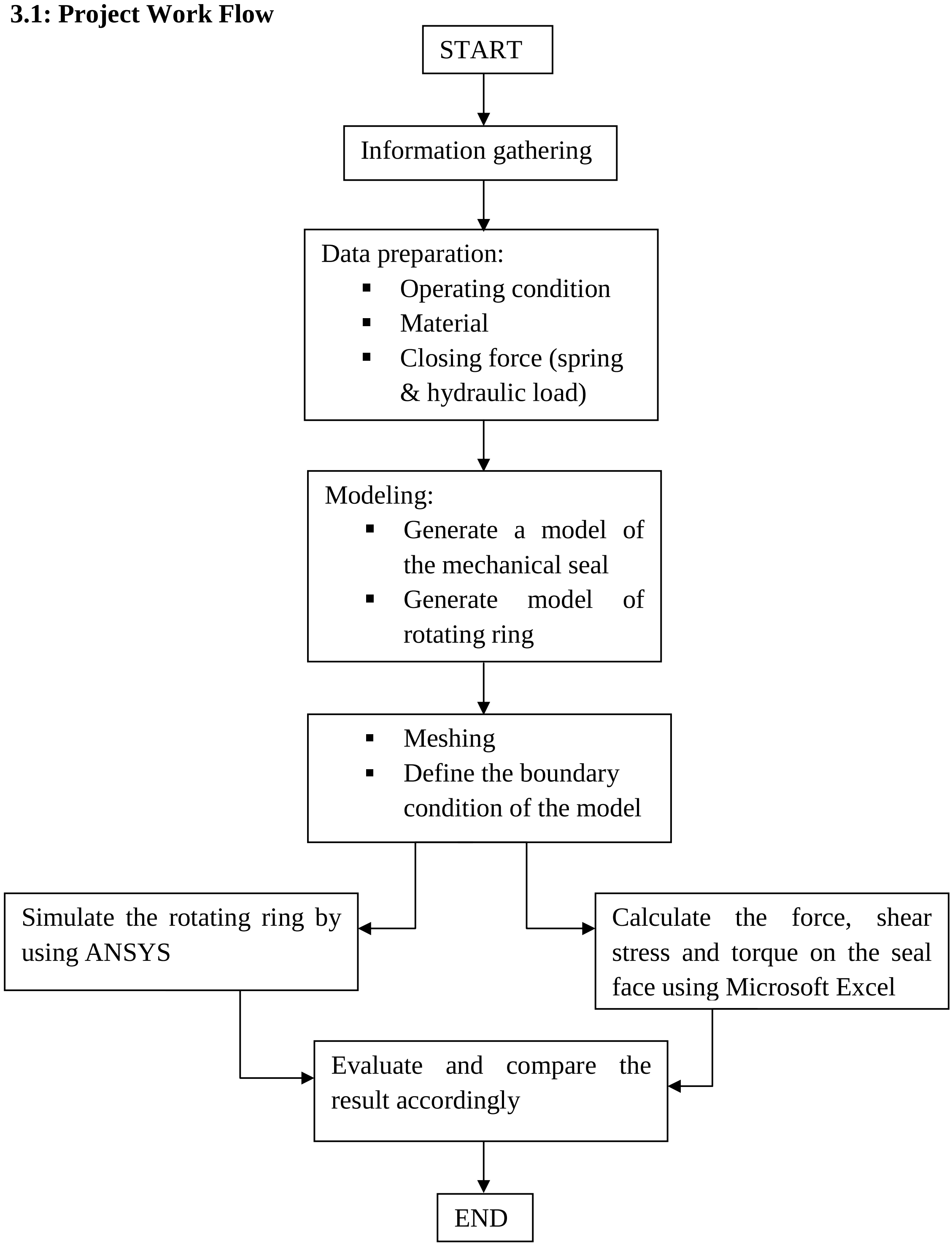

Figure 3.1: Project work flow

CHARTER 4

MATHEMATHICAL MODELLING

4.1: Mechanical Design

There are two types of seal mounting, which are internal and external mounting. External mounted seal is mounted outside the pump where the higher pressure liquid is in contact with the inner diameter of the seal faces. The second type which is internal mounting (Figure 4.1) is the most common arrangement with the seal is mounted inside the pump. In this arrangement, the higher pressure liquid is in contact with the outer diameter of the seal faces.

The fundamental characteristic of a mechanical seal is its balance ratio [2] where it will affect the dimension of the mechanical seal. It can be classified into two, which are:

- Balance seal
- Unbalance seal

Seals with the balance ratio equal or greater than one are referred to as unbalance seal while those in which it is less than one are referred to balance seal. Normally, the balanced seals are used for higher pressure, typically up to 70 barg. While, unbalanced seals are used for lower pressure, typically less than 10 barg [2]. In all mechanical seal system, there is an area where it is responsible for generating the hydraulic closing forces in the seal. The annular area (A_h) enclosed by the outside diameter of the sealing face (Do) and the balance diameter (D_b) as shown in Figure 4. It is the area of the seal face (A_f) compared to the hydraulic area (A_h) which is one of the parameter that will determine whether the seal is balanced or unbalanced.

In mechanical design, spring mechanism also plays an important role in determining the seal performance. All mechanical seals have an arrangement of spring mechanism to apply an initial closing force to the faces and to hold them together in the absence of fluid pressure [2]. The magnitude of spring load must be sufficient to overcome any axial friction from the dynamic secondary seal and the dynamic effects of any face misalignments. Instead of that, the spring mechanism is also for general purpose seals where there is a possibility of a vacuum in the seal chamber which will subjects the seal to a reverse pressure [2].

4.2: Dimension

In generating the model, firstly there are five parameters that need to be considered and they are actually depending on the shaft size [10]. Based on the existing mechanical seal, with the shaft size of 2.75in, the other four parameters are as following (Appendix B):

- Shaft Diameter = 2.75in
- Seal Outer Diameter = 3.5in
- Seal head Outer Diameter = 3.75in
- Seal Head Operating Height = 1.937in
- Seat Thickness = 0.625in

This information is not sufficient enough in order to create a mechanical seal model. There are several other parameters that need to be considered (Figure 4.1) which are:

- Outside Diameter of Sealing Interface (D_o)
- Inside diameter of Sealing Interface (D_i)
- Balance Diameter (D_b)
- Outer Diameter of Stationary Secondary Seal (D_h)

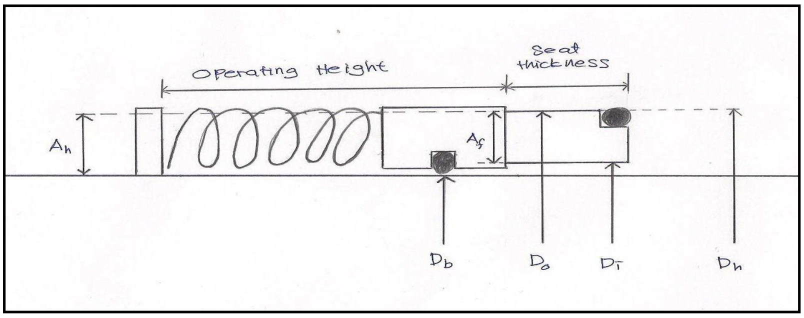

Figure 4.1: Internal mounting and unbalance seal arrangement.

Since it is an internally mounted seal with an unbalanced seal arrangement, the typical value of the balance ratio is around 1.2 [2]. Based on that typical value, the parameter needed to create a model of a mechanical seal can be determined. Based on Figure 4.1 the only unknown is the inside diameter of sealing interface (D_i) and it is determined by applying the equation below.

$$B = \frac{hydraulic\ loading\ area}{sealing\ interface\ area}$$

$$= \frac{\frac{1}{4}.\pi.(D_o^2 - D_b^2)}{\frac{1}{4}.\pi.(D_o^2 - D_i^2)}$$

$$= \frac{(D_o^2 - D_b^2)}{(D_o^2 - D_i^2)}$$

Based on the equation above, the value of D_o and D_b is already known which are 3.5in and 2.75in respectively. By using the typical value of the balance ratio which is 1.2, the value for inside diameter of sealing interface is 2.91in (74mm).

$$1.2 = \frac{89^2 - 70^2}{89^2 - D_i^2}$$

$$D_i^2 = 5403.5$$

$$D_i = 74\ mm$$

Finally, the overall parameters needed to create a mechanical seal model are as following (Figure 4.1):

- Outside Diameter of Sealing Interface (D_o) = 3.5in (89mm)
- Inside diameter of Sealing Interface (D_i) = 2.91in (74mm)
- Balance Diameter (D_b) = 2.75in (70mm)
- Outer Diameter of Stationary Secondary Seal (D_h) = 3.75in (95mm)

4.3: Closing Force

The total closing force (F_t) on the seal ring is sum of the spring load (F_s) and hydraulic load (F_h).

$$F_t = F_S + F_h$$

Spring Load (F_s)

The spring load applied toward the rotating face must be sufficient enough to resist the vacuum loads. So that, the following equation is used to determine the minimum force or spring loads required to overcome the vacuum loads.

$$Fs > \frac{1}{4}\pi(Dh^2 - Db^2)(Pa - P)$$

Where;

P = absolute pressure on outer periphery of seal

Pa = absolute pressure on inner periphery of seal

The value for P and Pa are 0.355 MPa (3.55 bar) and 0.1 MPa respectively. As a result, the value of the spring loads required to overcome the vacuum loads must be higher than 826.1 N.

In mechanical seal design, the typical values of spring load per unit area (P_s) of sealing interface (A_f) are 0.1 MPa to 0.5 MPa. The area of sealing interface is where it is bounded by outside diameter of sealing interface (Do) and inside diameter of sealing interface (D_i). The area of sealing interface (A_f) can be determined by the following equation.

$$Af = \frac{1}{4}\pi(Do^2 - Di^2)$$

As a result, the area of sealing interface is 1920.3 mm^2.

In order to find the sufficient spring load per unit area (P_s), another equation is used.

$$Ps = \frac{Fs}{Af}$$

As calculated before, A_f = 1920.3 mm^2 and the typical value of spring loads per unit area varies from 0.02 MPa to 0.5 MPa [2]. In applying this equation, the indication of the suitability is the spring loads (F_s) where it must be higher than 826.1 N to overcome the vacuum loads.

As a result, the suitable amount of the spring load per unit area is 0.44 MPa to 0.5 MPa (Appendix C).

As per Appendix C with 0.44 MPa to 0.5 MPa of pressure, the amount of force that is being applied toward the rotating ring is as following:

Table 4.1: Force being applied against the rotating ring

P(MPa)	A(mm^2)	F(N)
0.44	1920.3	844.932
0.45	1920.3	864.135
0.46	1920.3	883.338
0.47	1920.3	902.541
0.48	1920.3	921.744
0.49	1920.3	940.947
0.5	1920.3	960.15

NOTE: The amount of the force must higher than 826.1 N to overcome the vacuum loads.

Hydraulic Load (F_h)

Seals are normally subjected to different pressure on the inside and outside. The hydraulic closing force may be obtained by applying the following equation [2]:

$$F_h = A_f \, (B \times \Delta p + p_a)$$

Where Δp is the pressure differential across the seal.

$$F_h = 1920.3 \, mm^2 \, (1.2 \times 0.255 \, MPa + 0.1 \, MPa)$$

As a result the amount of hydraulic load (F_h) is 779.6 N

So that, the amount of total closing forces (F_t) is as following:

Table 4.2: Total closing force (F_t)

F_s(N)	F_h(N)	F_t(N)
844.932	779.6	1624.5
864.135	779.6	1643.7
883.338	779.6	1662.9
902.541	779.6	1682.1
921.744	779.6	1701.3
940.947	779.6	1720.5
960.15	779.6	1739.8

Based on Table 4.2 above, the value at the first row will be used in this project for mathematical calculation of the torque, shear stress and to be used in simulation process as input load.

4.4: Material Selection

The material selection for the seal rings is done properly based on a few criteria. There are relatively few materials that are commonly used. Even though it involves small number of material, the range of properties for each material is very wide. Seal material properties fall into three broad categories which are physical & mechanical, chemical and tribological [1].

There are many different mechanical and physical properties for materials, but only certain ones are of important for rotating ring. The properties are as following:

TENSILE STRENGTH is a very important property that need to be consider since rotating ring are loaded by hydrostatic pressures, friction forces, and drive forces. It is very important for both thermal shock resistance and mechanical shock resistance. Because seal ring materials are selected primarily for tribological and chemical reasons, many of the seal materials have a low tensile strength and are brittle [1].

COMPRESSIVE STRENGTH is usually much higher than tensile strength in brittle materials and may be used to advantage to offset the low tensile strength [1].

MODULUS OF ELASTICITY is found to very more than an order of magnitude for the various seal materials of interest and it is important in determining how well seal faces mate together under load. Low modulus materials deform easily in unwanted ways. On the other hand, for high modulus materials are brittle, easily be over stressed by tight fits and thermal loads [1].

HARDNESS is related to tribological performance than to any other factor. It has been found that seals operate best using some type of carbon graphite mating with various other materials and the range of carbon materials available is wide [1].

COEFFICIENT OF THERMAL EXPANSION is important in relation to the amount of thermal distortion that arises from temperature gradients in the seal ring. It is a key parameter in the thermal shock resistance of a material [1].

THERMAL CONDUCTIVITY is most important in determining how the heat generated at the interfaces flows to the cooling fluid. High thermal conductivity materials will result in lower interface temperatures [1].

DENSITY is important in determining the stress induced by centrifugal effects [1].

POISSON'S RATIO is the ratio when a sample object is stretched, of the contraction or transverse strain (perpendicular to the applied load), to the extension or axial strain (in the direction of the applied load) [14].

The seal by their nature must operate in all types of chemical environments, chemical compatibility between face materials and the sealed fluid is one of the most important criteria for selecting the suitable material for the rotating ring.

Based on the criteria for the material selection, the best material for the rotating ring is Carbon-Graphite. The advantages of carbon-graphite are as following [2]:

- Good lubricating qualities under dry or boundary lubricating conditions.
- An ability to bed in quickly and take up any slight imperfections in face geometry.
- Good chemical resistance
- Wide temperature resistance ranging from cryogenic temperatures to 250^{o}C. This upper limit temperature can be extended to 350^{o}C by using certain metalized grades.
- Reasonably strong in compression.
- Relatively low in cost.

There are a few materials that can be used for the rotating ring. The material that is used in this project is Carbon Graphite Resin Impregnated. The properties of the material are as following [2]:

Table 4.3: Properties of Carbon Graphite Resin Impregnated

PROPERTIES	Carbon Graphite
Density (kg/m^3)	1800
Hardness	90 - 100 Shore A
Poisson's Ratio	0.22
Youngs Modulus (GPa)	23
Tensile Strength (MPa)	41
Tensile Ultimate Strength (MPa)	65
Compressive Yield Strength (MPa)	234
Thermal Conductivity (W/m.^{o}C)	2349.1
Thermal Shock Parameter (W/m)	10 360
Thermal Expansion (1/^{o}C)	8.1945 x 10^{-4}
Specific Heat (J/kg.^{o}C)	2.0486 x 10^5

On the other hand the material that is used for the stationary ring is Silicon Carbide (SiC). The properties are as following:

Table 4.4: Properties of Silicon Carbide (SiC)

PROPERTIES	Silicon Carbide (SiC)
Density (kg/m^3)	3070
Poisson's Ratio	0.22
Youngs Modulus (GPa)	380
Tensile Strength (MPa)	276
Tensile Ultimate Strength (MPa)	460
Compressive Yield Strength (MPa)	1030
Thermal Conductivity (W/m.^{o}C)	35509.5
Thermal Expansion (1/^{o}C)	1.3111 x 10^{-3}
Specific Heat (J/kg.^{o}C)	1.8301 x 10^5

CHAPTER 5

COMPUTER SIMULATION: ANSYS WORKBENCH

5.1: Introduction

In order to analyze the behavior of the rotating seal ring, structural analysis has been performed on the seal ring by using ANSYS WORKBENCH. It is a process to analyze the seal ring in order to predict the responses or behavior of the seal ring under the excitation of expected loading and external environment of the seal ring.

There are three essential components (Figure 5.1) to analyze the behavior of the seal ring which are the model of seal ring, the excitation/loads and response which is the result.

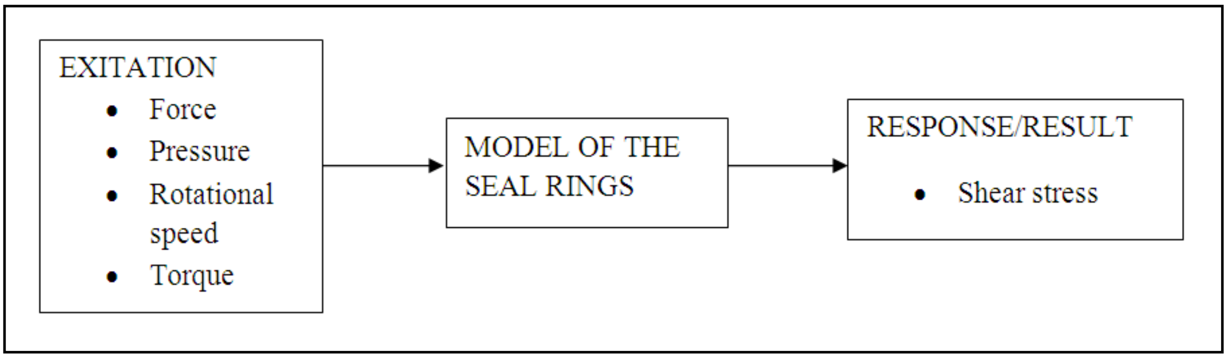

Figure 5.1: Essential components to analyze the behavior

The assumption in this analysis is the steady loading and response condition of the model, where the load and structure's response are assumed to vary slowly with respect to time. All of the data from the basic mathematical calculation that have been obtained (total closing force & torque) is used as load or excitation to the model.

5.2: Steps Involve In ANSYS WORKBENCH Simulation

STEP 1: Generate the mathematical model for the seal rings in ANSYS WORKBENCH

↓

STEP 2: Specify the gap between the seal rings

↓

STEP 3: Specify the material and properties for both rings

↓

STEP 4: Apply rotational velocity to the rotating ring with respect to the required operational condition

↓

STEP 5: Apply spring force to one end of the rotating ring by dividing the total force into eight points as shown in Figure 5.2 & 5.3

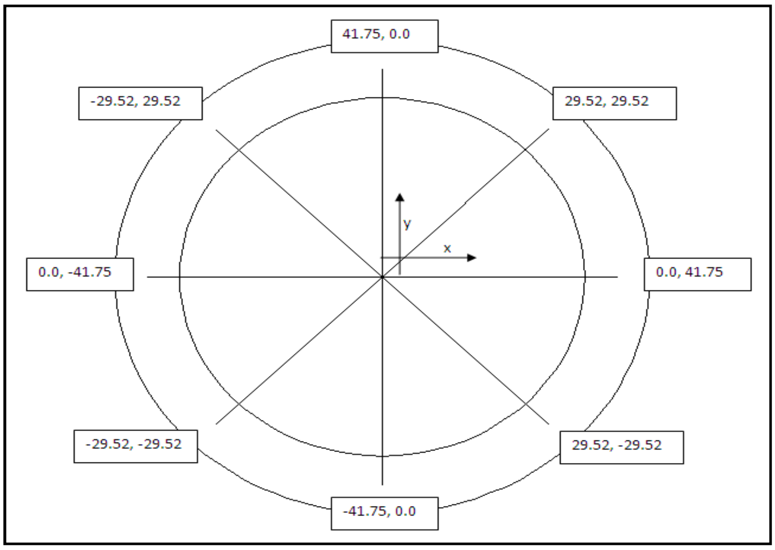

Figure 5.2: Coordinate for the spring force

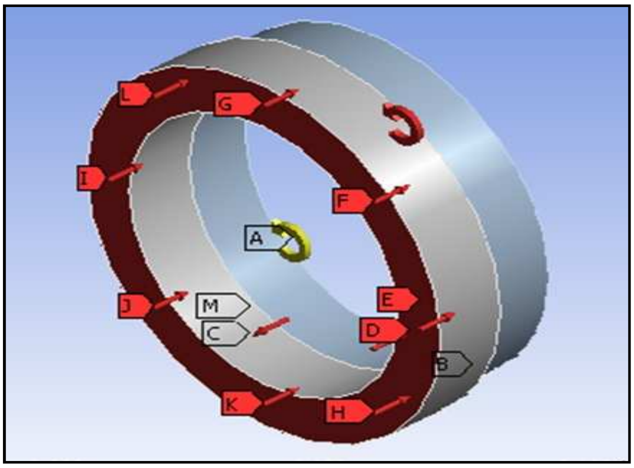

Figure 5.3: Position of spring force

STEP 6: Apply hydraulic pressure load to the same end of the rotating ring

STEP 7: Apply the supporting pressure to the other end of rotating ring as shown in Figure 5.4

NOTE: Supporting pressure is equal to the resulting pressure of total closing force

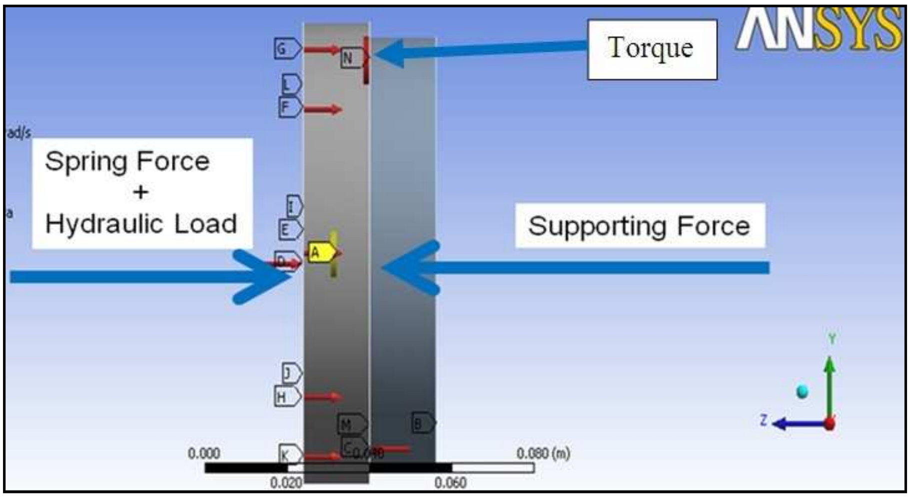

Figure 5.4: Total closing force, supporting force and torque

STEP 8: Apply the torque with respect to the operational condition to the seal face of rotating ring as shown in FIGURE 4

↓

STEP 9: Specify the desired response or result and run the simulation

↓

STEP 10: Analyze the result

CHAPTER 6

RESULT & DISCUSSION

6.1: Model of Mechanical Seal

Based on a few typical values and requirements for a mechanical seal, the dimension of the mechanical seal has been identified in chapter 4. A mathematical model of the mechanical seal has been created using Auto CAD.

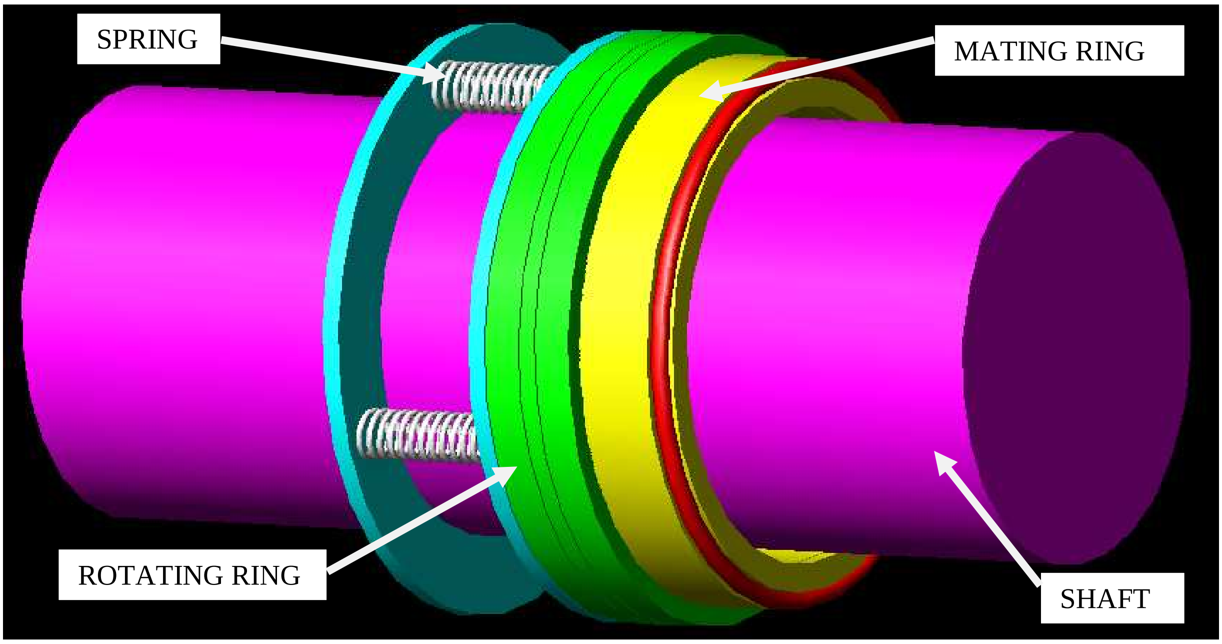

Figure 6.1: Model of mechanical seal

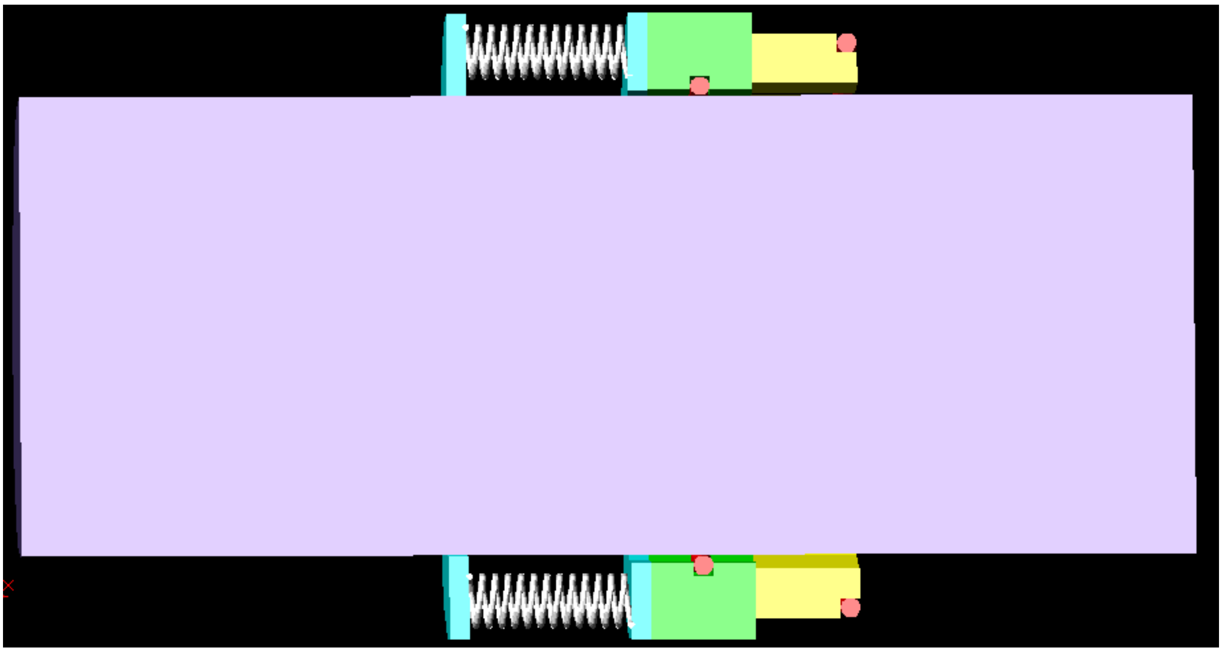

Figure 6.2: Cross section of mechanical seal

Figure 6.3 shows the 3D view of the seal rings and the next figure shows the dimension of the rotating and stationary ring.

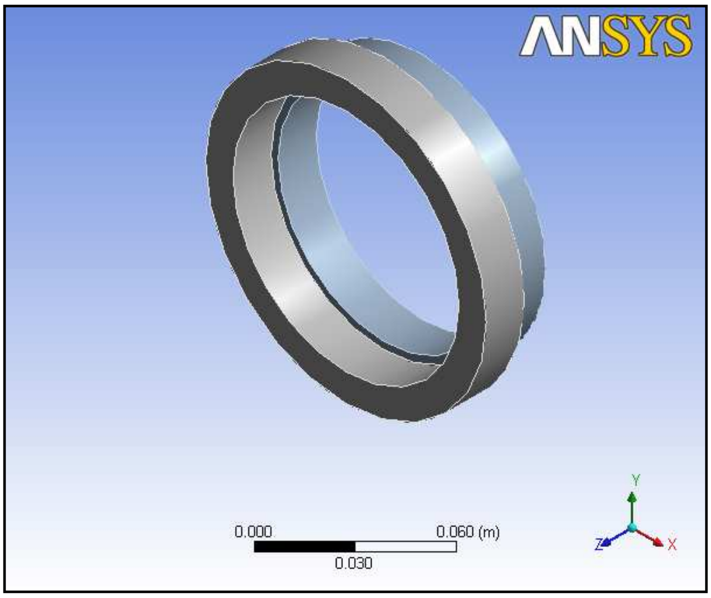

Figure 6.3: Solid model of seal rings

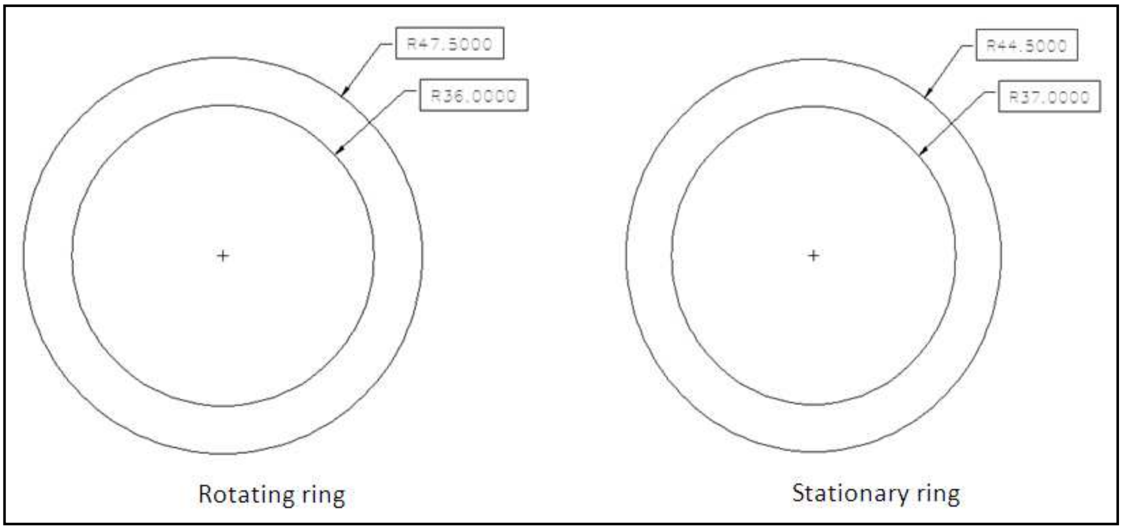

Figure 6.4: Dimension of seal rings (mm).

The thickness for both seal rings is 16mm.

6.2: Force, Shear Stress, & Torque

In order to analyze the forces and shear stress distribution on the seal face of rotating ring, the surface area is divided into ten segments. The outer and inner radius of the rotating ring is 0.0475 m (47.5 mm) and 0.036 m (36 mm) respectively.

$$\Delta r = \frac{r_{out} - r_{in}}{10} = \frac{0.0475 - 0.036}{10}$$

$$= 1.15 \times 10^{-3} m$$

That means the difference between r_{i+1} and r_i (Figure 6.5) is equal to 1.15 x 10^{-3}m

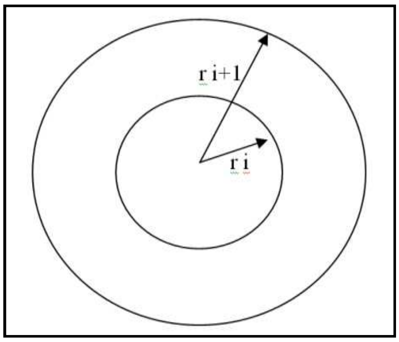

Figure 6.5: Inner and outer radius

$$\therefore r_i = r_{in} + i\Delta r$$

The mean radius for each segment is:

$$r_{i,mean} = r_i + \frac{\Delta r}{2}$$

The inner (r_i), outer (r_{i+1}), and mean (r_{mean}) radius for each segment is shown in Table 6.1 and Figure 6.6 below.

Table 6.1: Inner and outer radius for each segment.

i	r_i(m)	r_{i+1}(m)	r_{mean} (m)
1	0.03600	0.03715	0.03658
2	0.03715	0.03830	0.03773
3	0.03830	0.03945	0.03888
4	0.03945	0.04060	0.04003
5	0.04060	0.04175	0.04118
6	0.04175	0.04290	0.04233
7	0.04290	0.04405	0.04348
8	0.04405	0.04520	0.04463
9	0.04520	0.04635	0.04578
10	0.04635	0.04750	0.04693

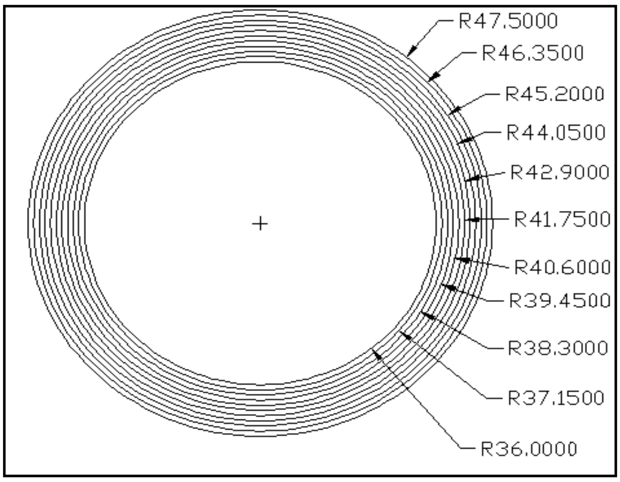

Figure 6.6: 10 segments of the rotating ring.

FORCE & SHEAR STRESS

As shown in Figure 6.7 below, the rotating ring rotates against the stationary ring with a speed of 2950rpm.

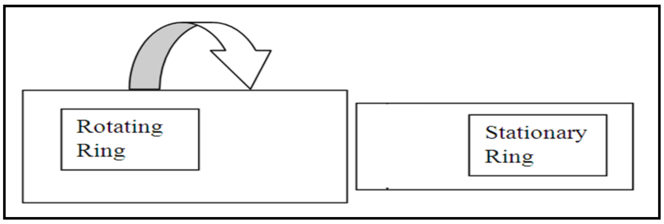

Figure 6.7: Rotating ring rotate against stationary ring.

In order to find the force and shear stress for each segment there are a few equations that need to be considered.

$$dF = \tau . dA$$

$$F = \int \tau . dA = \int \mu \frac{dU}{dy} . dA$$

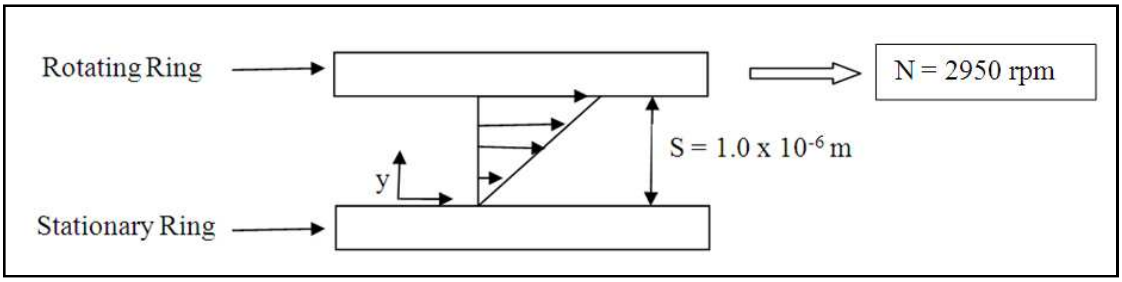

Figure 6.8: Velocity of rotating ring and the gap between the rotating and mating ring.

Since the gap between the rotating and stationary ring (S) is very small, it is possible to consider the linear variation of velocity (U) in that region.

$$U(y) = \frac{y}{S} V$$

$$\frac{dU}{dy} = \frac{V}{S}$$

$$V = \omega r$$

$$\omega = \frac{2\pi N}{60}$$

$$\omega = \frac{2\pi \times 2950rpm}{60s} = 309\, rad/s$$

So that, the force for each segment can be calculated by:

$$F_i = \int_{ri}^{ri+1} \mu . \frac{V}{S}(2\pi r). dr \quad = \int_{rI}^{ri+1} \mu . \frac{\omega r}{S}(2\pi r). dr$$

$$F_i = \mu . \frac{2\pi\omega}{S} \int_{ri}^{ri+1} r^2 \, . dr = \mu . \frac{2\pi\omega}{S} \left(\frac{r_{i+1}^3 - r_i^3}{3} \right)$$

Shear stress is the product of force at each segment divided by the area of the respective segment. The area for each segment can be calculated by the following equation.

$$A_i = \pi(r_{i+1}^2 - r_i^2)$$

After all of the information about the force and area has been obtained, the shear stress for each segment can be calculated together with the total shear stress on the seal face.

$$\tau_i = \frac{F_i}{A_i}$$

$$\tau_{Total} = \sum_{1}^{10} \tau_i$$

TORQUE

Torque, also called moment of force is the tendency of a force to rotate an object about an axis. The torque for each segment of the rotating seal face will be evaluated, so that the magnitude of torque depends on the force exerted to the seal face and the distance of each segment from the center of the ring. The torque for each segment can be calculated as following:

$$dT_i = dF_i \times r_{i,mean}$$

$$T_i = \int_{ri}^{ri+1} dF_i \times r_{i,mean} = \int_{ri}^{ri+1} \mu \frac{2\pi\omega r^2}{S} . dr \times r_{i,mean}$$

$$T_i = \mu \frac{2\pi\omega}{S} \int_{ri}^{ri+1} r^2 . dr \times r_{i,mean} = \mu . \frac{2\pi\omega}{S} \left(\frac{r_{i+1}^3 - r_i^3}{3} \right) \times r_{i,mean}$$

By applying the equations above, the torque and shear stress on the rotating seal face applied by the lubricant is calculated. The shear and torque is evaluated for various number of rotating speeds, gap between the seal rings and viscosity of the lubricant.

Various rotational speeds

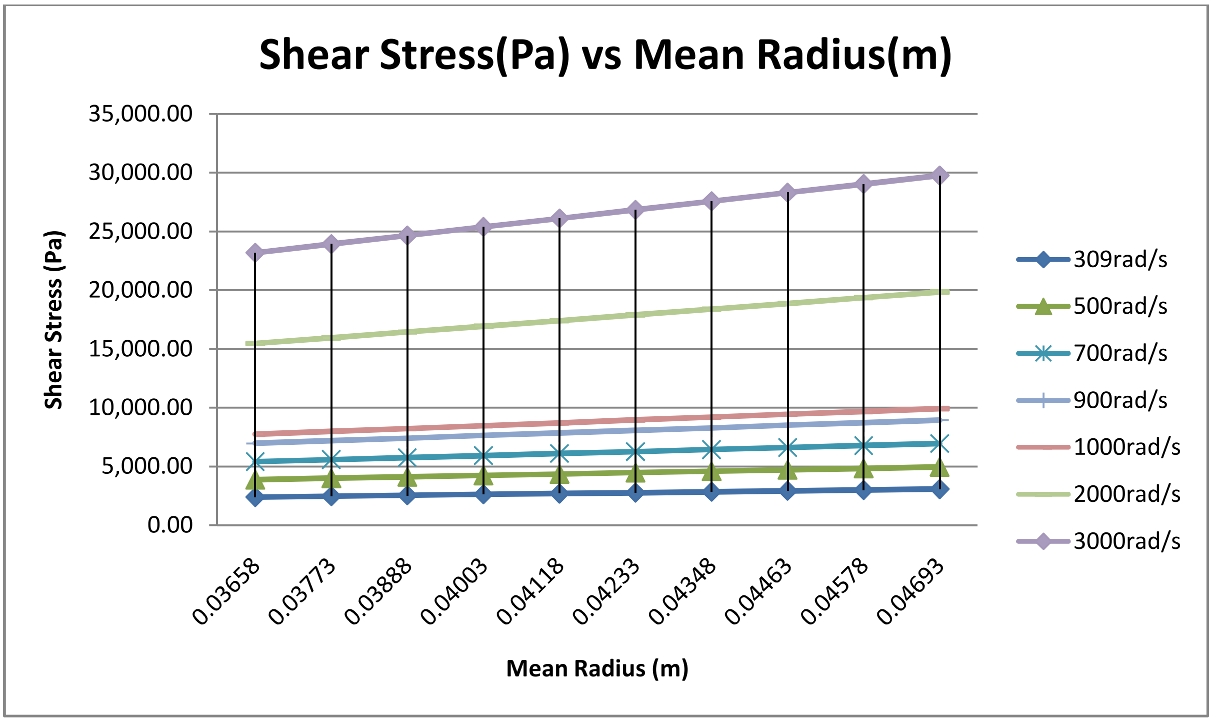

Figure 6.9: Shear stress distribution at various rotational speeds

Figure 6.9 shows the shear stress distribution on the rotating ring seal face for various rotating speeds. From the graph, it shows that for each speed the shear stress increase linearly from inner to outer radius of the ring.

From the graph it shows that the amount of shear stress increase as the rotating speed increase. Small change in rotating speed (309rad/s – 1000rad/s) will not give high impact on the shear stress. With rotating speed of 2000rad/s and 3000rad/s, the different of resulting shear stress is higher.

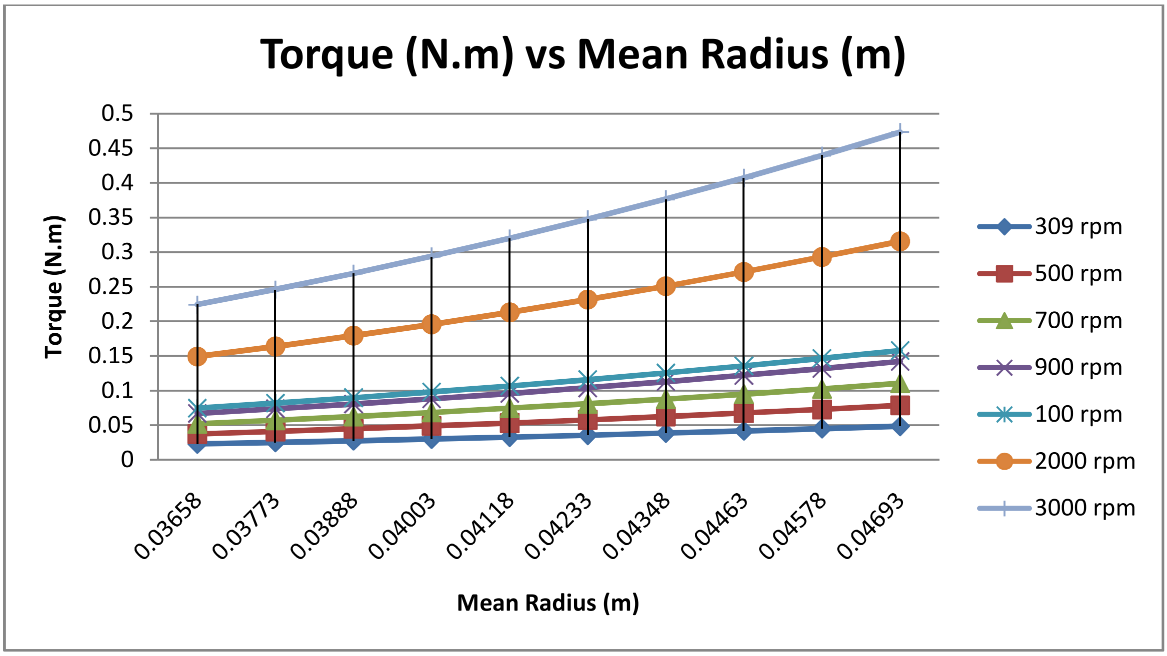

Figure 6.10: Torque distribution at various rotational speeds

Figure 6.10 shows the torque at various rotational speeds. From the graph, it shows that for each rotational speed the torque increase from inner to outer radius of the ring. It also shows that the torque increase as the rotating speed increase.

<u>Various gaps</u>

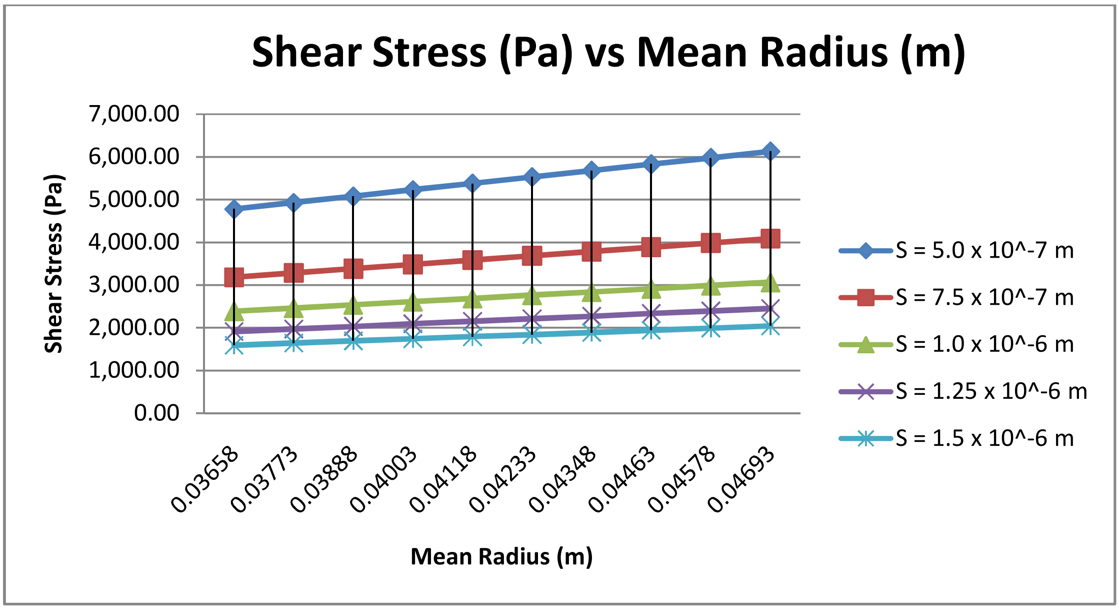

Figure 6.11: Shear stress distribution at various gaps

Figure 6.11 shows the shear stress distribution on the rotating ring seal face for various gaps between the seal rings. For each gap distance, the shear stress increase linearly from inner to outer radius of the ring. It also shows that the shear stress decrease as the gap distance between the seal rings increases.

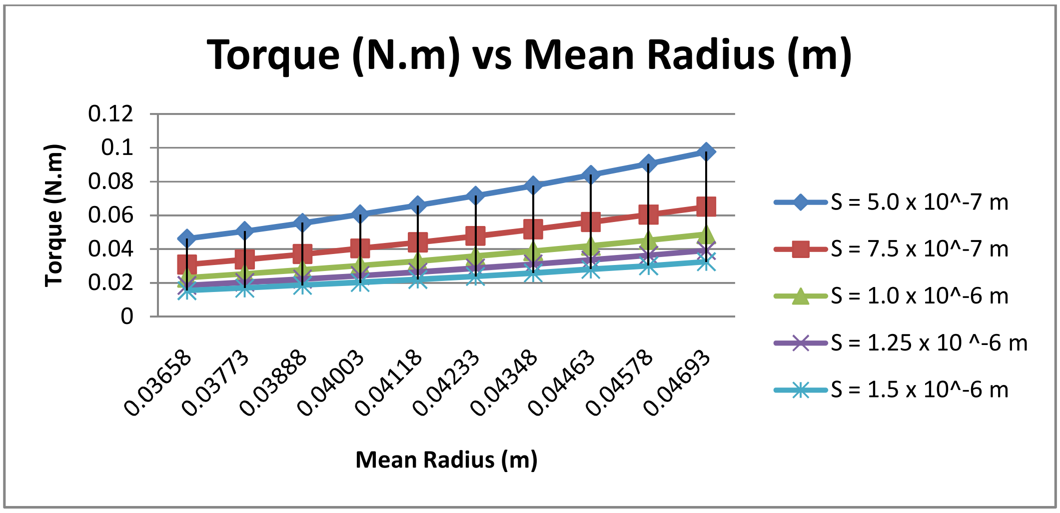

Figure 6.12: Torque distribution at various gaps

Figure 6.12 shows the torque at various gap distances. For each gap, the torque increase from inner to outer radius of the ring. It also shows that the torque decrease as the gap between the seal rings increase.

<u>Various temperatures</u>

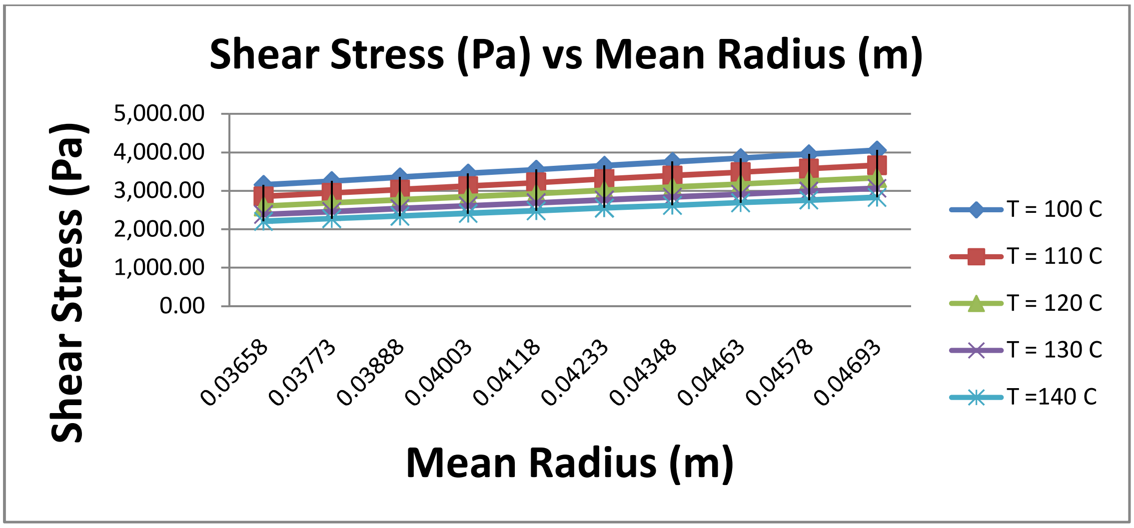

Figure 6.13: Shear stress distribution at various temperature

Figure 6.13 shows the shear stress on the rotating ring seal face for various temperature. For each temperature, the shear stress increase from inner to outer radius of the ring. As the temperature increase, the viscosity of the lubricant decrease. As a result, the shear stress distribution decreases as the temperature increase.

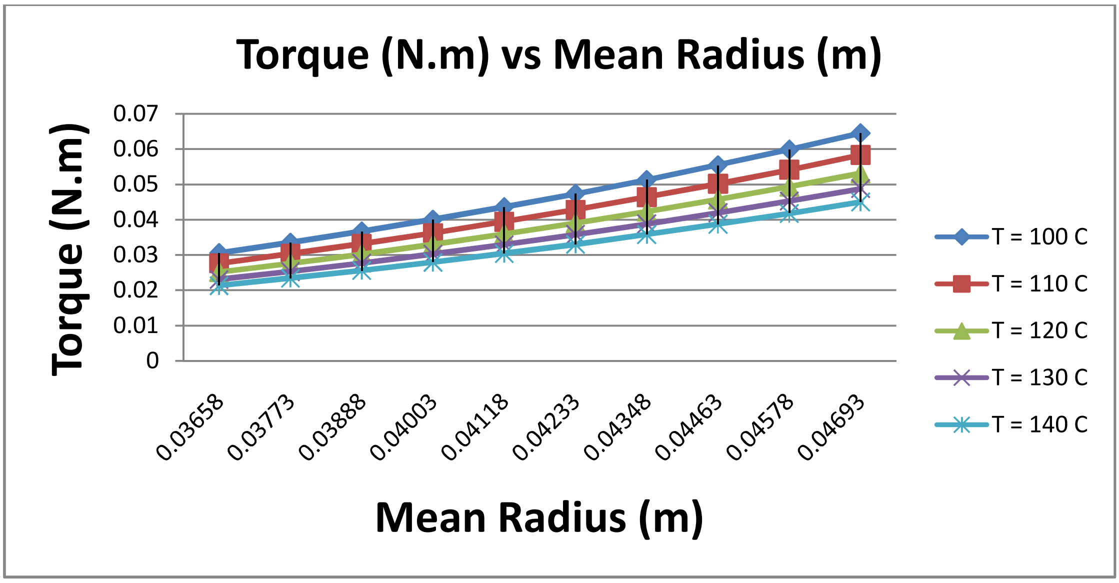

Figure 6.14: Torque distribution at various temperatures

Figure 6.14 shows the torque at various temperature. For each temperature, the torque increase from inner to outer radius of the ring. As the temperature increase, the viscosity of the lubricant decrease and as a result the torque decrease.

6.3: Result and Discussion

Operational Condition: Variation in rotational speed for each gap

- Viscosity, μ (mPa.s): 0.21132
- Rotational Speed, ω (rad/s): 309, 500, 700, 900, 1000, 2000, 3000
- Gap, S (mm): 0.0005, 0.00075, 0.001, 0.00125, 0.0015, 0.00175, 0.002

The result for the maximum shear stress from 309 rad/s t 3000 rad/s for each gap is shown in Appendix D. The data is tabulated in a graph as shown in Figure 6.15 below.

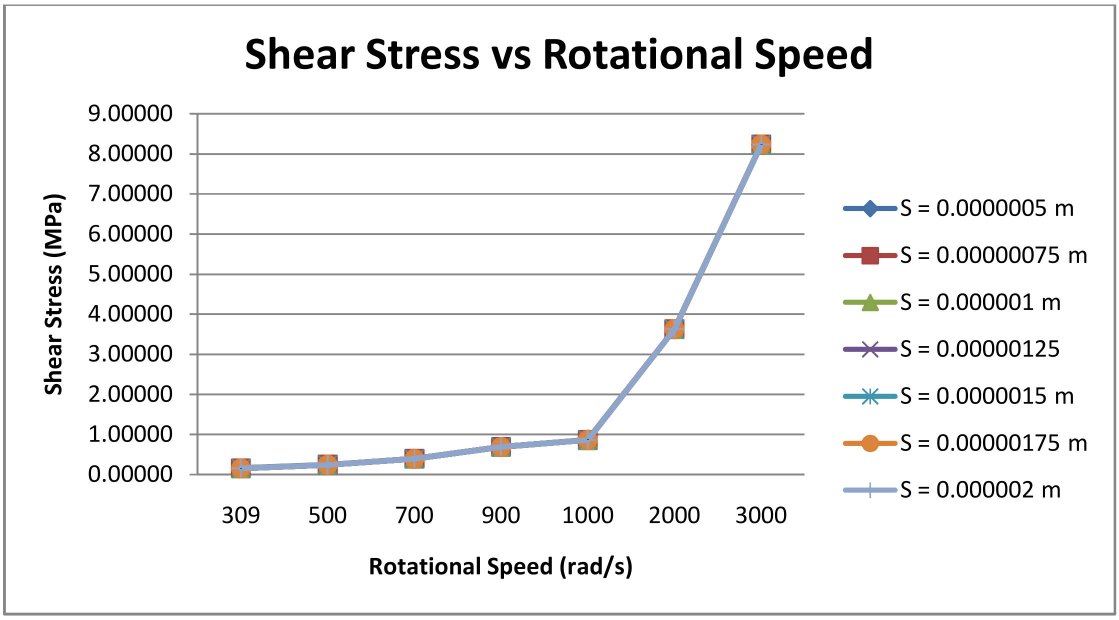

Figure 6.15: Resulting shear stress at various rotational speeds

Figure 6.15 shows the resulting shear stress at various rotational speeds. Based on the graph, the shear stress will increase as the rotational speed increase. It also shows that small increase in the rotational speed will result in small increase in shear stress. Based on Appendix D the shear stress decrease as the gap increase. The difference in shear stress in very small and that explain why the graph above shows a single plot for seven difference gap. It can be concluded that, the variation of gap will slightly changes the resulting shear stress while, the variation of rotational speed will dominantly control the resulting shear stress on the seal face.

Even at the highest rotating speed in this simulation (3000 rad/s), the resulting shear stress is still below the maximum shear stress. It does not mean that the mechanical seal can operate at high rotating speed. There is one parameter that provides limiting value for the rotating speed which is PV (Pressure times velocity) parameter. Exceeding PV limit will lead to rapid failure of mechanical seal. For Carbon Graphite-Silicon-Carbine combination, the PV limit is 90bar.m/s [2].

$$PV = 90\ bar.\frac{m}{s}$$

$$3.55\ bar\ \times V_{rotating\ speed} = 90\ bar.m/s$$

$$V_{rotating\ speed} = \frac{90\ bar.m/s}{3.55\ bar} = 25.35\ m/s$$

$$\omega = \frac{V_{rotating\ speed}}{r} = \frac{25.35\ m/s}{0.0475\ m} = 534\ rad/s$$

Based on the calculation above with seal pressure of 3.55 bar (0.355 MPa) and radius of 0.0475m the limiting speed of the rotating ring is 534 rad/s. If the ring operates higher than 534 rad/s, it will lead to rapid failure of the mechanical seal.

The relationship between shear stress and rotational speed is shown in graph below.

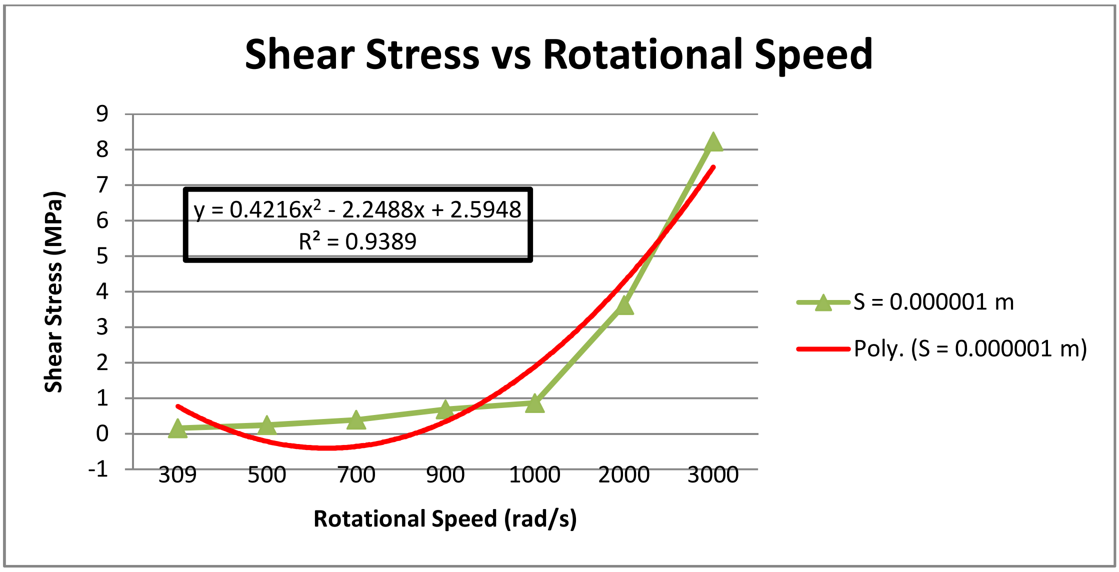

Figure 6.16: Relationship between shear stress and rotational speed.

From the graph there is an equation that relates the resulting shear stress with the variation of rotational speed which is:

$$y = 0.421x^2 - 2.248x + 2.594$$

There is also R-Squared value for the smooth curve which is 0.938. Since the value is close to 1, meaning that the relation can be used to predict the value of shear stress (y) when the rotational speed is varies (x).

Operational Condition: Variation in gap for each viscosity

- Rotational Speed, ω (rad/s): 309
- Gap, S (mm): 0.0005, 0.00075, 0.001, 0.00125, 0.0015, 0.00175, 0.002
- Viscosity, μ (mPa.s): 0.19518, 0.21132, 0.23024, 0.25263, 0.27942

The result for the maximum shear stress from 0.0005 mm to 0.002 mm for each viscosity is shown in Appendix E. The data is tabulated in a graph as shown in Figure 6.17 below.

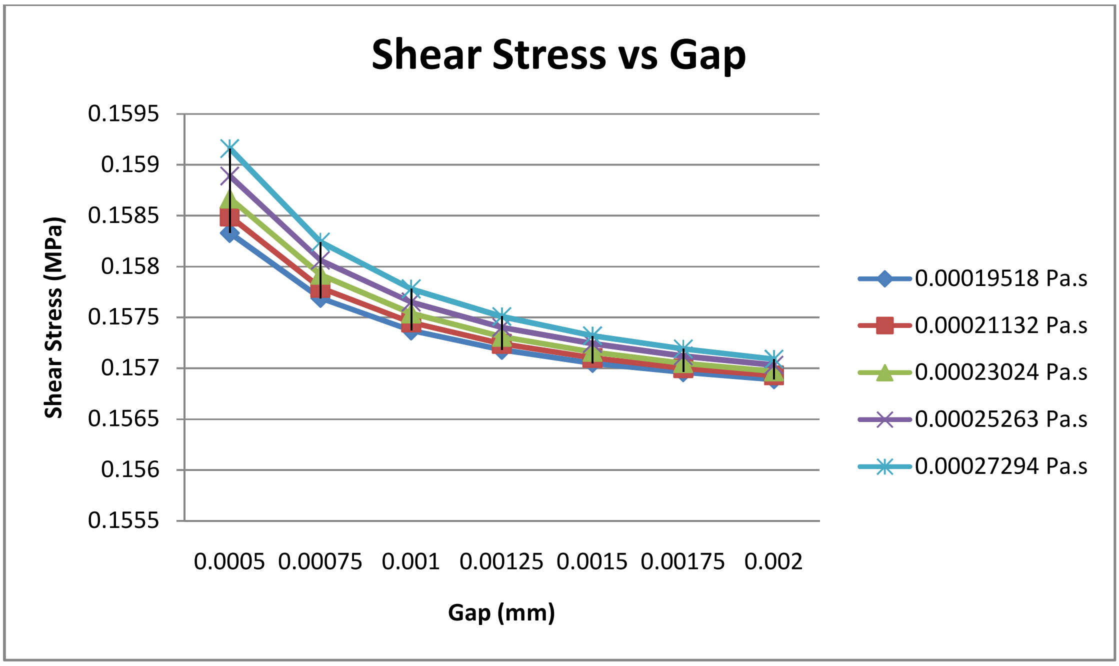

Figure 6.17: Resulting shear stress at various gaps

As shown in the figure, the resulting sheer stress is decreasing as the gap between the seal rings increasing. It also shows that, there is small difference in shear stress for each viscosity, where the shear stresses slightly increase as the viscosity increase. It can be concluded that the variation in gap will significantly affect the resulting shear stress compare to the variation in viscosity.

The relationship between shear stress and gap between the seal ring is shown in graph below:

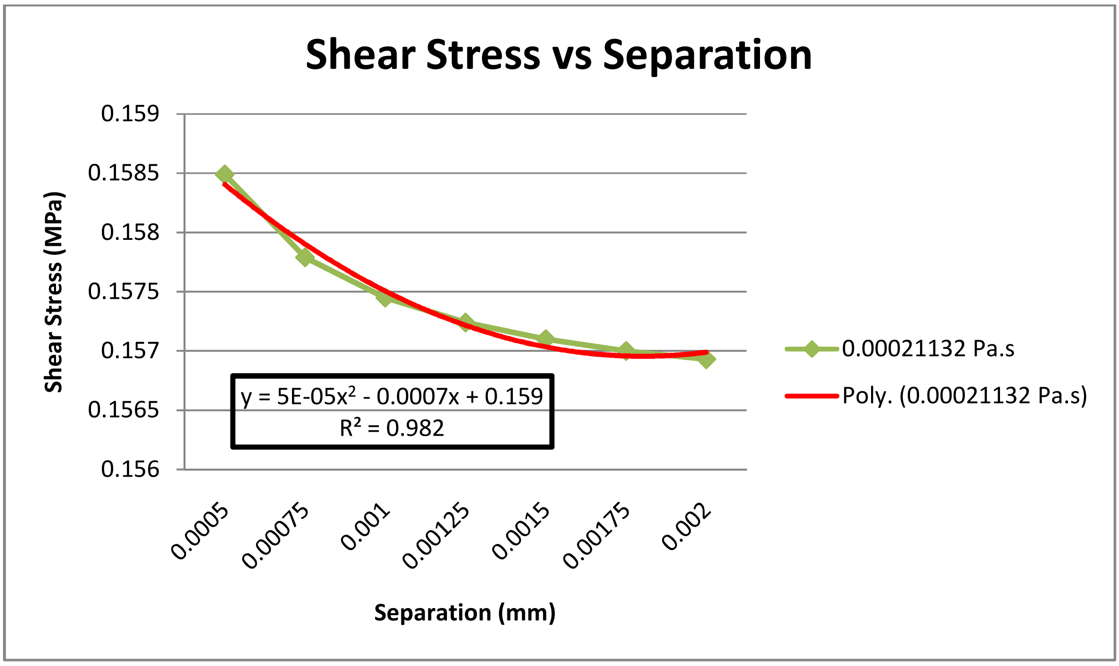

Figure 6.18: Relationship between shear stress and gap.

From the graph there is an equation that relates the resulting shear stress with the variation of rotational speed which is:

$$y = 0.00005x^2 + 2.59$$

There is also R-Squared value for the smooth curve which is 0.982. Since the value is close to 1, meaning that the relation can be used to predict the value of shear stress (y) when the gap between the seal rings is varies (x).

Thicker gap will result in safer operating condition but at the cost of greater leakage. All mechanical seal leak to some extent and in order to predict the leakage rate, the characteristics of the seal product and the environment in which the seal is operating need to be considered [2].

$$Q = 3.6\pi \times d_o \times P \times h_s^2 \times \frac{S}{P_g^2}$$

Where:

Q = Leakage rate (ml/h)

d_o = Outer diameter of seal interface (mm)

P = Seal pressure (bar)

S = Gap factor (bar/s) (Appendix F)

P_s = Face pressure due to spring (bar)

= F_s / A_f

B = Balance ratio

P_g = Seal face pressure (bar)

= $P_s + (B \times P)$

h_s = Face gap (mm)

$$Q = 3.6\pi \times 89 \times 3.55 \times 0.0005^2 \times \frac{1 \times 10^5}{\left(4.4 + (1.2 \times 3.55)\right)^2}$$

By varying the value of face gap (h_s), there will be a variation of leakage rate as shown on Figure 6.19 below.

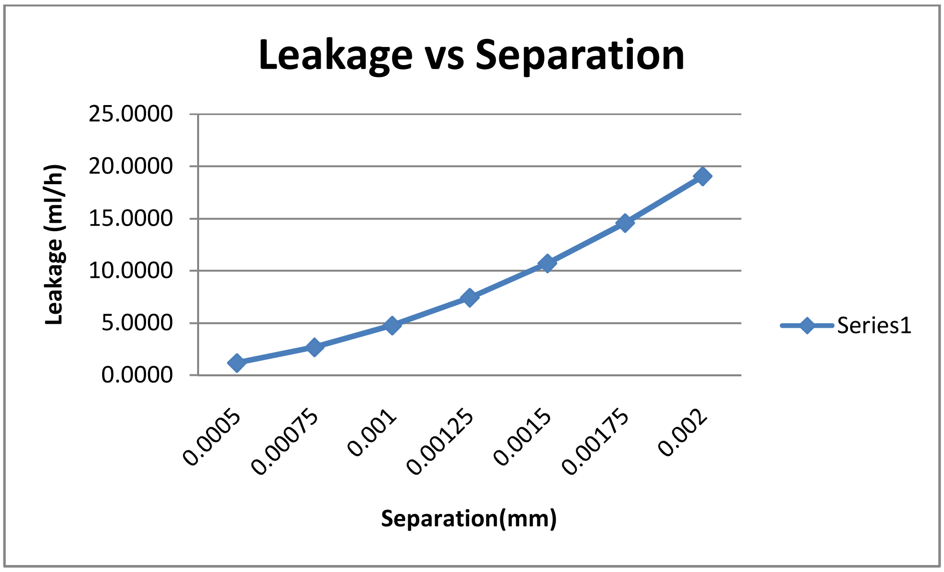

Figure 6.19: Resulting leakage rate at various gaps

The figure clearly shows that the leakage rate is increasing as the gap between the seal rings increases. The level of leakage that can be considered as failure is depending very much on the characteristics of the sealed product and the operating environment. There is a general rule for the leakage, where a seal whose leakage rate is 250 times the theoretical figure is definitely functioning incorrectly [2].

Operational Condition: Variation in viscosity for each rotational speed

- Gap, S (mm): 0.001
- Viscosity, μ (mPa.s): 0.19518, 0.21132, 0.23024, 0.25263, 0.27942
- Rotational Speed, ω (rad/s): 309, 500, 700, 900, 1000, 2000, 3000

The result for the maximum shear stress from 0.19518 mPa.s to 0.27942 mPa.s for each viscosity is shown in Appendix G. The data is tabulated in a graph as shown in Figure 6.20 below.

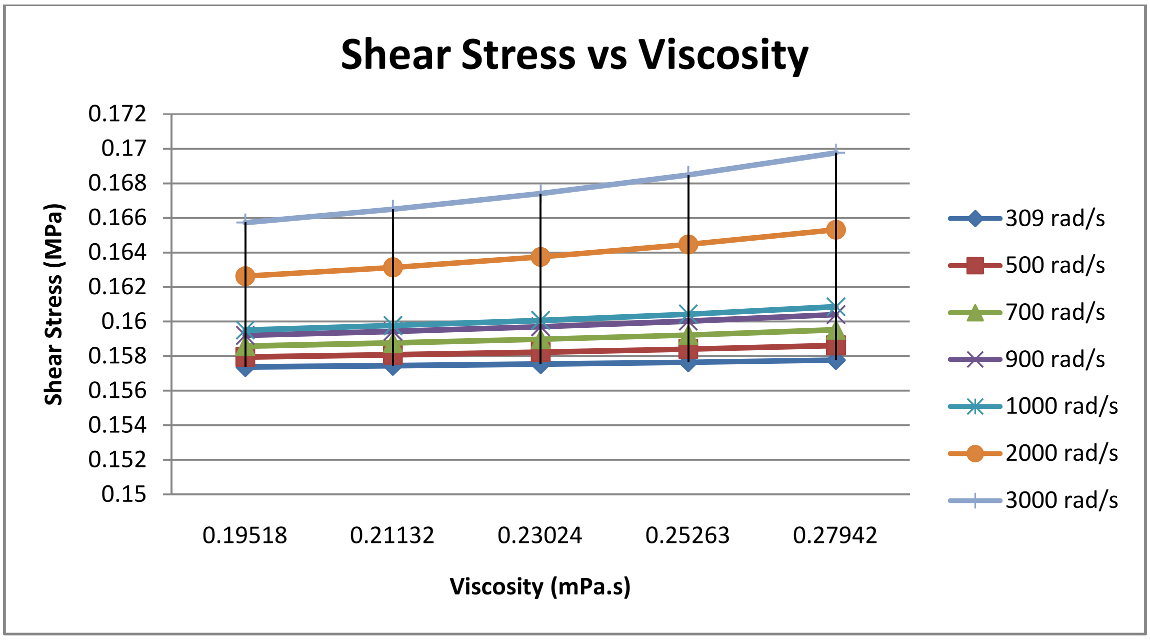

Figure 6.20: Resulting shear stress at various viscosities

As shown in the figure, the resulting shear stress is increasing as the viscosity increasing. It also shows that, the shear stress is increasing as the rotational speed increasing.

The relationship between shear stress and gap between the seal ring is shown in graph below:

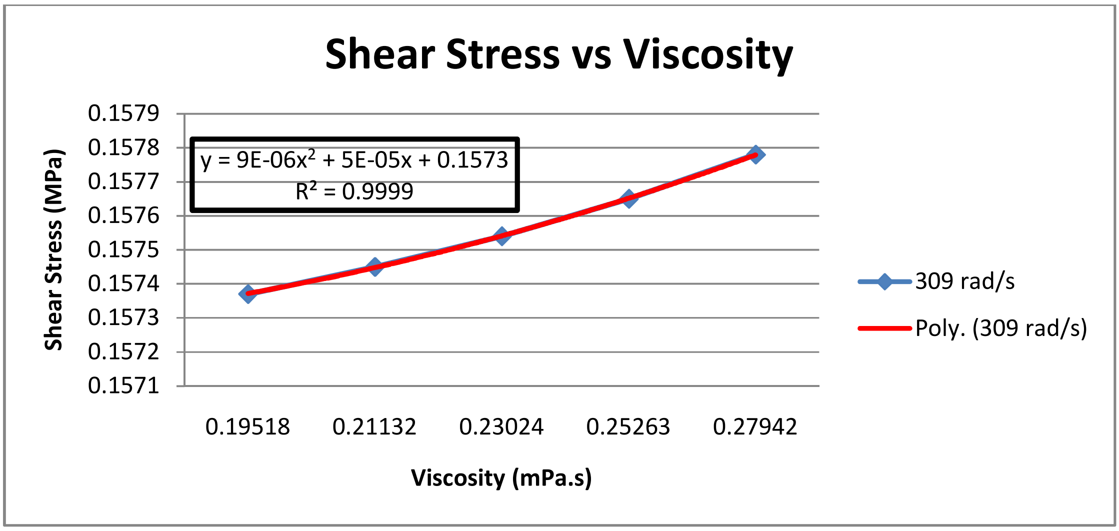

Figure 6.21: Relationship between shear stress and viscosities.

From the graph there is an equation that relates the resulting shear stress with the variation of viscosity which is:

$$y = 0.000009x^2 + 0.00005x + 0.157$$

There is also R-Squared value for the smooth curve which is 0.999. Since the value is very close to 1, meaning that the relation can be used to perfectly predict the value of shear stress (y) when the viscosity of fluid between the rings is varies (x).

Low viscosity is not a problem for liquids, but high viscosity will cause excessive torque during start-up. Instead of that, it will also affect the time taken for the interfacial film to thin down as the seal stop [2]. The time taken for the fluid film to thin down is based on the equation below:

$$\tau = \frac{2 \times S \times h_o^2 \times t}{\mu \times b^2}$$

$$\frac{h}{h_o} = \frac{1}{\sqrt{1 + \tau}}$$

Where:

S = Specific closing force (MPa)

H_o = Initial fluid film (mm)

H = Final fluid film (mm)

μ = Viscosity (Pa.s)

b = Face width (mm)

t = Thinning time

By varying the viscosity of the lubricant fluid, there will be a significant difference in time taken for each different viscosity to thin down as shown in figure below.

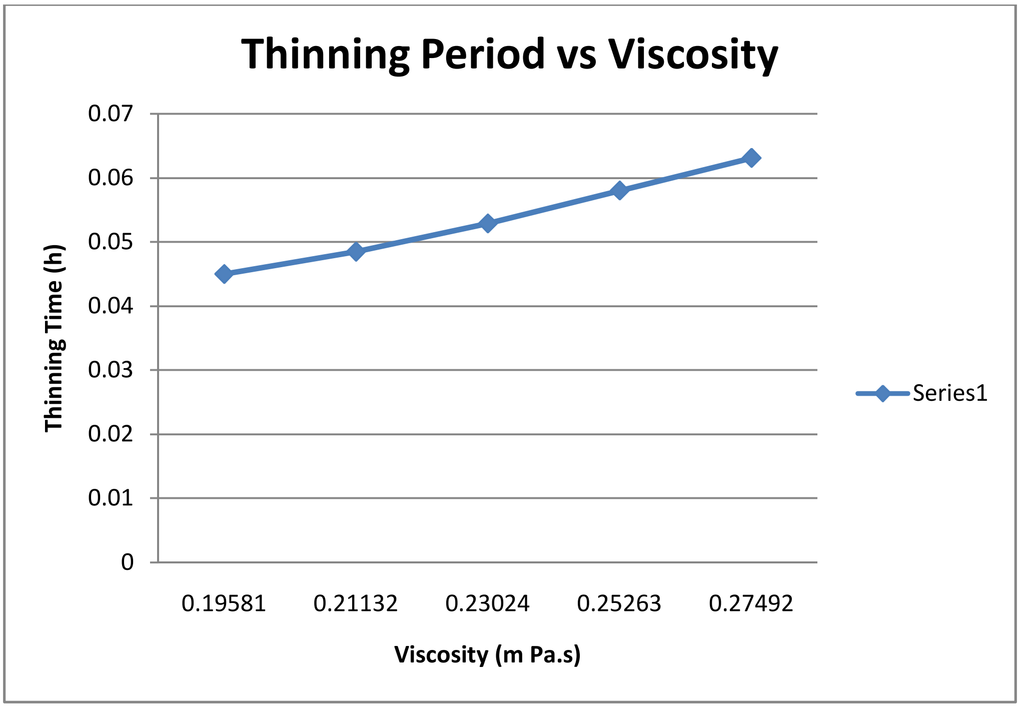

Figure 6.22: Thinning time for each viscosity

Figure 6.19 show the time taken for the fluid film to thin down to 1 % of the initial fluid film. It shows that the thinning time is increasing from 0.045h (161.8s) to 0.0631h (227.2s) as the viscosity increasing. For each viscosity, the thinning time is relatively low.

As shown in Figure 6.23, for a very viscous fluid with viscosity from 0.06 Pa.s to 1 Pa.s, the time-scale becomes days.

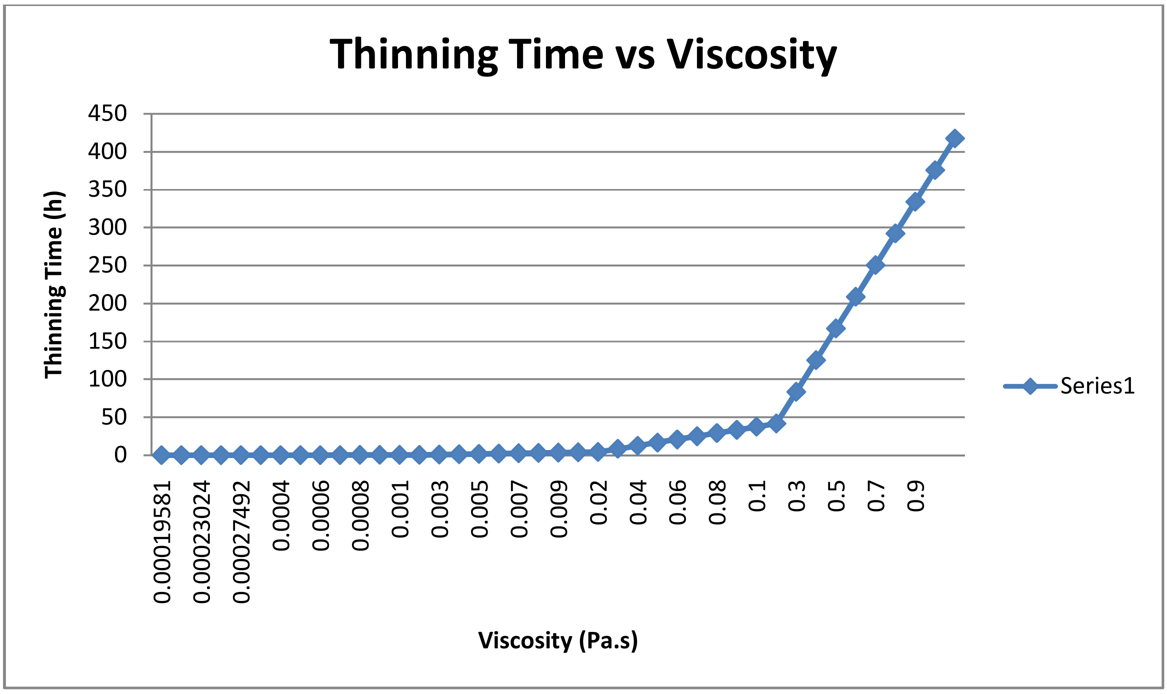

Figure 6.23: Thinning time for each viscosity

It is clearly shows that the viscosity of the fluid will affect the thinning time and consequently will affect maintenance work when it is required.

CHAPTER 7

CONCLUSION AND RECOMMENDATION

7.1: Conclusion

In order to improve the existing design system of mechanical seal, lots of research and testing are required to get a better understanding on how does the system works. At the early stage of research and development, the engineers only rely on the routine laboratory testing in order to get the result of any modification toward the system for improvement. In a way to improve the system, the first step is to understand what exactly happens inside the seal chamber and the best method for that purpose in turn into computer simulation technique. In this project a mathematical model of rotating face seal is build based on a few typical values. Then, the rotating face mechanical seal is simulated by using ANSYS in order to understand the behavior of the seal rings for various operation conditions. From the analysis, there is an equation for each condition that can be used to relate and predict the resulting shear stress on the rotating ring seal face in various conditions. Finally, it will lead to design change for better performance.

7.2: Recommendation

The result of this project is a mathematical model of rotating face seal and the behavior under various operating conditions. In order to improve the result from simulation and obtain more precise data, it should include more input parameter that consider various size and shape of rotating face mechanical seal.

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APPENDIX A: Project Planning for FYP I & II

FYP I

NO	Detail Work Scope/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Selection of Topic														
2	Preliminary research on the importance of the subject														
3	Research/study on the principles of mechanical seal														
4	Research/Data preparation: Operating condition and material selection														
5	Research/Modeling: Generate the existing model of mechanical seal (rotating ring) using AUTOCAD														
6	Calculate the total closing force (spring & hydraulic load)														

FYP II

NO	Detail Work Scope/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Run of basic simulation (ANSYS)														
2	Development of the simulation for various operating condition														
3	Analysis of the result														
4	Documented the result														

APPENDIX B: Mechanical seal dimension based of shaft diameter

2-3/4" (2.750") SHAFT

SEAL HEAD			SEAT			Material	Notes	
Type	0.D.	Oper.Ht.	Type	O.D.	Thick	Code		
C	3.750	1.937	3	3.50	.625	BCFKF		
C	3.750	1.937	3	3.50	.625	BCFJF		
C	3.750	1.937	3	3.50	.625	NCFKF		
C	3.750	1.937	3	3.50	.625	BCFKF	E/F	
C	3.750	1.125	3	3.500	.625	BCFJF		
D	4.000	1.937	3	3.500	.625	BCDKF		
D	4.000	1.937	3	3.500	.625	BCDJF		
E	3.500	2.750	3	3.500	.625	BCDKF		
E	3.500	2.750	3	3.500	.625	BCDJF		
K	3.750	3.750	3	3.500	.625	BCFKF		
K	3.750	1.875	3	3.500	.625	BCFJF		

APPENDIX C: Spring load and force

P(MPa)	A(mm^{2})	F(N)
0.02	1920.3	38.406
0.03	1920.3	57.609
0.04	1920.3	76.812
0.05	1920.3	96.015
0.06	1920.3	115.218
0.07	1920.3	134.421
0.08	1920.3	153.624
0.09	1920.3	172.827
0.1	1920.3	192.03
0.11	1920.3	211.233
0.12	1920.3	230.436
0.13	1920.3	249.639
0.14	1920.3	268.842
0.15	1920.3	288.045
0.16	1920.3	307.248
0.17	1920.3	326.451
0.18	1920.3	345.654
0.19	1920.3	364.857
0.2	1920.3	384.06
0.21	1920.3	403.263
0.22	1920.3	422.466
0.23	1920.3	441.669
0.24	1920.3	460.872
0.25	1920.3	480.075
0.26	1920.3	499.278
0.27	1920.3	518.481
0.28	1920.3	537.684
0.29	1920.3	556.887
0.3	1920.3	576.09
0.31	1920.3	595.293
0.32	1920.3	614.496
0.33	1920.3	633.699
0.34	1920.3	652.902
0.35	1920.3	672.105
0.36	1920.3	691.308
0.37	1920.3	710.511
0.38	1920.3	729.714
0.39	1920.3	748.917

P(MPa)	A(mm^{2})	F(N)
0.4	1920.3	768.12
0.41	1920.3	787.323
0.42	1920.3	806.526
0.43	1920.3	825.729
0.44	1920.3	844.932
0.45	1920.3	864.135
0.46	1920.3	883.338
0.47	1920.3	902.541
0.48	1920.3	921.744
0.49	1920.3	940.947
0.5	1920.3	960.15

APPENDIX D – Shear stress for various revolutions

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.0000005 m Shear Stress (Pa)
309	0.69965	1.5849e+005 Pa
500	1.13213	2.4723e+005 Pa
700	1.58498	3.9478e+005 Pa
900	2.03783	6.889e+005 Pa
1000	2.26425	8.6361e+005 Pa
2000	4.52850	3.6245e+006 Pa
3000	6.79275	8.2286e+006 Pa

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.00000075 m Shear Stress (Pa)
309	0.46644	1.5779e+005 Pa
500	0.75475	2.4611e+005 Pa
700	1.05665	3.957e+005 Pa
900	1.35855	6.9009e+005 Pa
1000	1.50950	8.6493e+005 Pa
2000	3.01900	3.6271e+006 Pa
3000	4.52850	8.2325e+006 Pa

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.000001 m Shear Stress (Pa)
309	0.34983	1.5745e+005 Pa
500	0.56606	2.4555e+005 Pa
700	0.79249	3.9616e+005 Pa
900	1.01891	6.9068e+005 Pa
1000	1.13213	8.6559e+005 Pa
2000	2.26425	3.6284e+006 Pa
3000	3.39638	8.2345e+006 Pa

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.00000125 m Shear Stress (Pa)
309	0.27986	1.5724e+005 Pa
500	0.45285	2.4521e+005 Pa
700	0.63399	3.9644e+005 Pa
900	0.81513	6.9104e+005 Pa
1000	0.9057	8.6599e+005 Pa
2000	1.8114	3.6292e+006 Pa
3000	2.7171	8.2357e+006 Pa

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.0000015 m Shear Stress (Pa)
309	0.23322	1.571e+005 Pa
500	0.37738	2.4499e+005 Pa
700	0.52833	3.9663e+005 Pa
900	0.67928	6.9128e+005 Pa
1000	0.75475	8.6625e+005 Pa
2000	1.50950	3.6298e+006 Pa
3000	2.26425	8.2365e+006 Pa

Various Revolution Viscosity = 0.00021132 Pa.s Revolution (rad/s)	 Torque (N.m)	 Gap = 0.00000175 m Shear Stress (Pa)
309	0.19990	1.57e+005 Pa
500	0.32346	2.4483e+005 Pa
700	0.45285	3.9676e+005 Pa
900	0.58224	6.9145e+005 Pa
1000	0.64693	8.6644e+005 Pa
2000	1.29386	3.6301e+006 Pa
3000	1.94079	8.237e+006 Pa

Various Revolution		
Viscosity = 0.00021132 Pa.s		Gap = 0.000002 m
Revolution (rad/s)	Torque (N.m)	Shear Stress (Pa)
309	0.17491	1.5693e+005 Pa
500	0.28303	2.4471e+005 Pa
700	0.39624	3.9686e+005 Pa
900	0.50946	6.9157e+005 Pa
1000	0.56606	8.6658e+005 Pa
2000	1.13213	3.6304e+006 Pa
3000	1.69819	8.2375e+006 Pa

APPENDIX E – Shear stress for various gaps

Various Gap Revolution (rad/s) = 309 Gap (m)	 Torque (N.m)	 Viscosity= 0.00019518 Shear Stress (Pa)
0.0000005	0.64622	1.5833e+005 Pa
0.00000075	0.43081	1.5769e+005 Pa
0.000001	0.32311	1.5737e+005 Pa
0.00000125	0.25849	1.5718e+005 Pa
0.0000015	0.21541	1.5705e+005 Pa
0.00000175	0.18463	1.5696e+005 Pa
0.000002	0.16155	1.5689e+005 Pa

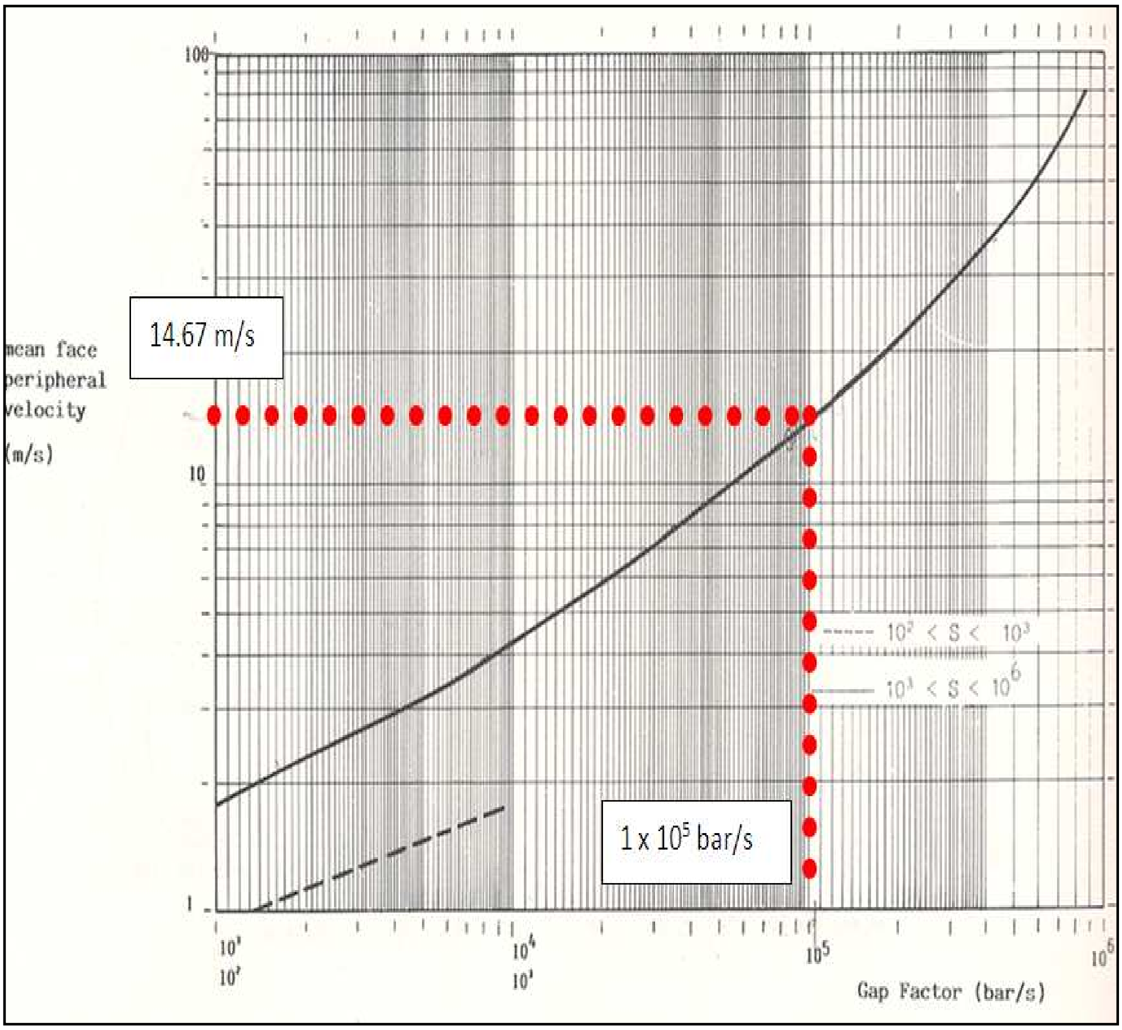
Various Gap Revolution (rad/s) = 309 Gap (m)	 Torque (N.m)	 Viscosity= 0.00021132 Shear Stress (Pa)
0.0000005	0.69965	1.5849e+005 Pa
0.00000075	0.46644	1.5779e+005 Pa
0.000001	0.34983	1.5745e+005 Pa
0.00000125	0.27986	1.5724e+005 Pa
0.0000015	0.23322	1.571e+005 Pa
0.00000175	0.19990	1.57e+005 Pa
0.000002	0.17491	1.5693e+005 Pa

Various Gap Revolution (rad/s) = 309 Gap (m)	 Torque (N.m)	 Viscosity= 0.00023024 Shear Stress (Pa)
0.0000005	0.76229	1.5867e+005 Pa
0.00000075	0.50820	1.5792e+005 Pa
0.000001	0.38115	1.5754e+005 Pa
0.00000125	0.30492	1.5731e+005 Pa
0.0000015	0.25410	1.5716e+005 Pa
0.00000175	0.21780	1.5705e+005 Pa
0.000002	0.19057	1.5697e+005 Pa

Various Gap Revolution (rad/s) = 309 Gap (m)	 Torque (N.m)	 Viscosity= 0.00025263 Shear Stress (Pa)
0.0000005	0.83643	1.5889e+005 Pa
0.00000075	0.55762	1.5806e+005 Pa
0.000001	0.41821	1.5765e+005 Pa
0.00000125	0.33457	1.574e+005 Pa
0.0000015	0.27881	1.5724e+005 Pa
0.00000175	0.23898	1.5712e+005 Pa
0.000002	0.20911	1.5703e+005 Pa

Various Gap Revolution (rad/s) = 309 Gap (m)	 Torque (N.m)	 Viscosity= 0.00027942 Shear Stress (Pa)
0.0000005	0.92512	1.5916e+005 Pa
0.00000075	0.61675	1.5824e+005 Pa
0.000001	0.46256	1.5778e+005 Pa
0.00000125	0.37005	1.5751e+005 Pa
0.0000015	0.30837	1.5732e+005 Pa
0.00000175	0.26432	1.5719e+005 Pa
0.000002	0.23128	1.5709e+005 Pa

APPENDIX F – Gap Factor



APPENDIX G – Shear stress for various viscosities

Various Viscosity		
Rotation (rad/s)= 309		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	0.32311	1.5737e+005 Pa
0.00021132	0.34983	1.5745e+005 Pa
0.00023024	0.38115	1.5754e+005 Pa
0.00025263	0.41821	1.5765e+005 Pa
0.00027942	0.46256	1.5778e+005 Pa

Various Viscosity		
Rotation (rad/s)= 500		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	0.52283	1.5796e+005 Pa
0.00021132	0.56606	1.5809e+005 Pa
0.00023024	0.61674	1.5824e+005 Pa
0.00025263	0.67672	1.5842e+005 Pa
0.00027942	0.74848	1.5863e+005 Pa

Various Viscosity		
Rotation (rad/s)= 700		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	0.73196	1.5858e+005 Pa
0.00021132	0.79249	1.5876e+005 Pa
0.00023024	0.86344	1.5897e+005 Pa
0.00025263	0.94741	1.5922e+005 Pa
0.00027942	1.04787	1.5952e+005 Pa

Various Viscosity		
Rotation (rad/s)= 900		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	0.94109	1.592e+005 Pa
0.00021132	1.01891	1.5943e+005 Pa
0.00023024	1.11014	1.597e+005 Pa
0.00025263	1.21810	1.6003e+005 Pa
0.00027942	1.34727	1.6041e+005 Pa

Various Viscosity Rotation (rad/s)= 1000		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	1.04566	1.5951e+005 Pa
0.00021132	1.13213	1.5977e+005 Pa
0.00023024	1.23349	1.6007e+005 Pa
0.00025263	1.35344	1.6043e+005 Pa
0.00027942	1.49696	1.6085e+005 Pa

Various Viscosity Rotation (rad/s)= 2000		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	2.09131	1.6262e+005 Pa
0.00021132	2.26425	1.6313e+005 Pa
0.00023024	2.46697	1.6373e+005 Pa
0.00025263	2.70688	1.6445e+005 Pa
0.00027942	2.99393	1.653e+005 Pa

Various Viscosity Rotation (rad/s)= 3000		Gap = 0.000001 m
Viscosity (Pa.s)	Torque (N.m)	Shear Stress (Pa)
0.00019518	3.13697	1.6572e+005 Pa
0.00021132	3.39638	1.665e+005 Pa
0.00023024	3.70046	1.674e+005 Pa
0.00025263	4.06032	1.6847e+005 Pa
0.00027942	4.49089	1.6975e+005 Pa